WORKFLOW DESIGN METHODOLOGY FOR ORC SYSTEMS DEDICATED TO WASTE HEAT RECOVERY FROM THE ENGINE COOLANT OF COMMERCIAL VEHICLES

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ABSTRACT

Road transport is today the biggest contributor to CO2 emissions in transport (75% of transport total CO2 emissions) (IEA, 2019). Commercial vehicles, including Heavy Duty (HD) trucks, account for a significant contribution mainly due to their important mileage. In France for example, the 650 000 trucks registered represent 1.5% of all vehicles but 27% of their overall CO2 emissions. The new European CO2 regulation aims to reduce road vehicles CO2 emissions by 30% in 2030. Starting from 2021 all new light vehicle registration must comply with a maximum fleet average emission limit of 95 gCO2/km. For HD trucks, a target limit in gCO2/ton.km will be enforced starting from 2025. Electrification with batteries appears today as a reasonable way to reduce CO2 emissions for light vehicles; many prospective studies highlight this market trend (ADEME, 2018) and the change is already underway. Yet, for long mileage and heavy commercial vehicles, using conventional internal combustion engines (ICE) with low carbon content fuels (biofuel or hydrogen) or hydrogen fuel cell (FC) combined with electric powertrains seems to be more in alignment with vehicle range constraints. For these commercial vehicles, for which low carbon mobility is necessary, waste heat recovery (WHR) appears to be relevant to improve fuel consumption and reduce total cost of ownership (Gravel, 2019), (Reiche et al., 2020). In this article, a cost-driven approach using low temperature WHR is proposed. It is based on an electric organic Rankine cycle coupled with the vehicle cooling circuit as a heat source. This solution offers a safe, lightweight, and low-cost module compatible with both conventional ICE and low temperature Proton Exchange Membrane Fuel Fell (PEMFC) technology, which is investigated for the future decarbonized hydrogen mobility. Simulation results highlighted that this ORC solution can break even within 2 years for northern European conditions with the current hydrogen cost trend and can be improved thanks to the split cooling radiator architecture proposed.

1 INTRODUCTION

In a conventional road vehicle equipped with an ICE, more than half of the fuel energy content is wasted as thermal losses in exhaust gases or in the cooling system; this value is higher for low load vehicle operating conditions where the ICE has a lower brake thermal efficiency but the global efficiency can be improved by hybridization / electrification. For highway driving conditions with low positive hybridization effect, the ICE can reach peak efficiencies limited to 42% on modern light commercial vehicle and 45% for state-of-the-art long-haul trucks. Future hydrogen fuel cell commercial vehicles will enable electric mobility with zero tank-to-wheel CO2 emissions along with extended driving range as compared to pure battery electric solutions but with residual waste heat despite a better thermal efficiency than conventional ICE. Recovering part of these thermal losses to improve efficiency would significantly reduce the fuel consumption and benefit the profitability of FC vehicles. A recent global study for ICCT on technology potential for fuel efficiency of commercial road transport (Ricardo, 2017) estimates a 4.5% fuel saving potential with waste heat recovery with an ORC, a €3000 to €5000 system cost and a 2.8 years payback period considering a tractor unit.

Most R&D projects related to ORC systems for commercial vehicle are proposed by OEMs (Gravel, 2018) or suppliers (Carballido *et al*., 2018), (Dingel *et al*., 2017) for highway trucks and promote high grade recovery sources, such as exhaust gas and/or EGR, to maximize recovery gains, with 48V electrified or mechanical output based on volumetric or kinetic turbine expanders. Some recent works have investigated low-grade waste heat recovery with ORC on engine coolant and compared it to exhaust heat recovery (Reiche *et al*., 2020). Obviously, heat recovery on engine cooling has lower fuel efficiency than on exhaust and leads to a major concern: access to cold sink to cool down the ORC condenser. This is the main drawback for low temperature ORC in road transportation for which the ORC cooling system should be implemented on an existing vehicle whose front face radiator area is already saturated with radiators for vehicle service (air conditioning (AC), charge air cooler (CAC)...). To improve low-grade recovery efficiency, other works (Furukawa *et al*., 2014) combine ORC with the engine thermo-management strategy to increase the temperature of the hot source. The main advantages of engine cooling recovery over exhaust gas recovery is that temperature and flow is much more stable. Moreover, about a third of exhaust gas energy at low temperature is unrecoverable due to condensation concerns in the exhaust. A recent study investigates HD truck exhaust gas heat recovery and highlights that the recovery is generally done at an intermediate temperature compared to gas temperature leading to high irreversibility losses in the ORC boiler (Cipollone *et al*., 2017). Another recent work (Smague *et al*., 2018) has investigated waste heat recovery on engine cooling with a mechanical output expander and with an integrated ORC turbo-pump to maximize ORC net efficiency and propose a cheaper ORC solution for low-grade recovery for non-HEV trucks or light commercial vehicles (Smague *et al*., 2019). With an integrated solution despite a lower raw expander efficiency as compared to exhaust gas recovery, a reasonable payback time can be achieved thanks to a lower pump ancillary energy consumption and a lower ORC cost. The future evolutions of commercial vehicles (Lejeune *et al*., 2018), driven by more and more stringent pollutant emission regulations, will involve new mobility solutions using lower carbon content or carbon-free fuels and leveraging fuel cell or hydrogen ICE technologies. Considering all these observations, a project was launched at IFP Energies nouvelles so as to promote ORC low-grade heat recovery in engine cooling circuit for commercial vehicles and HD trucks, to propose an original layout for its integration in the cooling circuit of conventional ICE or FC and to design the key component of the system: the ORC turbine.

2   PRE-DESIGN SIMULATIONS AND ECONOMIC ANALYSIS

2.1   ORC layout for optimal low-grade waste heat recovery

2.1.1 ICE heat power distribution
An energy balance of the fuel heating value can depict the average ICE heat losses distribution in real driving conditions and hot sources availability for ORC systems. Considering a long-haul heavy-duty truck propelled by a Diesel engine, three main heat sources can be considered: EGR as a high-grade energy source, exhaust gas as a medium grade energy source, and CAC and HT coolant as a low-grade energy source. Direct low-grade energy can represent mainly 20% of the fuel heating value in these conditions, and cumulative indirect low-grade energy, including EGR cooling with HT coolant, up to 30% as depicted in **Figure 1** – left. A similar distribution can be considered for hydrogen-fueled ICE, which are candidates for future low-carbon, heavy duty mobility. Indeed, the hydrogen flame front comes closer to the cylinder wall and induces higher heat flux than in Diesel ICE. But H2 combustion is done at a lower air-fuel ratio than Diesel and is achieved with a lower engine compression ratio (Demuynck *et al*., 2012). Considering these observations, similar heat losses through engine walls to engine cooling circuit could be considered for hydrogen and Diesel ICE in a first approach.

2.1.2 FC heat power distribution
Fuel cells dedicated to vehicles are generally Polymer Electrolyte Membrane Fuel Cells (PEMFC). They operate at low temperature - typically 80°C - and benefit from a higher peak efficiency than ICE at approximately 60% and consequently show less heat losses.

The main difference with ICE is that PEMFC heat losses are mainly rejected in the cooling circuit and marginally in the cathode exhaust that contains hot humid air (Costa Neto et *al.,* 2013). This situation is favorable to low-grade Medium Temperature ORC (MT-ORC) recovery that converts 80-100°C heat losses from the cooling circuit. A recent study analyses a hybrid system including PEMFC coupled with an ORC system and estimates a 1.5 to 2% fuel efficiency improvement with ORC (Choi et *al*., 2019). Briefly, the performance and the efficiency of the fuel cell are governed by its polarization curve. Cooling circuit heat losses are a combination of kinetic/activation losses (caused by the slowness of the reaction taking place at the electrodes), ohmic losses (caused by the membrane protonic resistance), and mass transfer losses (caused by diffusive transport through the gas diffusion layer of the cell). This low temperature waste heat concentrated in the fuel cell cooling circuit is very similar in amount and temperature to low-grade waste heat of conventional ICE and can represent roughly 30-35% of consumed hydrogen net calorific value (**Figure 1** – right). Thus, ORC systems for ICE or PEMFC of equal power level can be imagined with similar performance levels and operating temperatures.
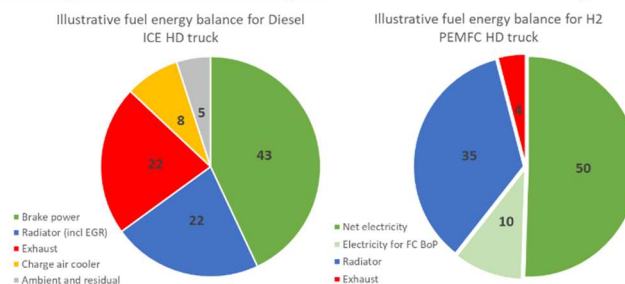
Figure 1: Illustrative breakdown of fuel energy in a truck

2.1.3 ORC layout for low-grade heat recovery
Figure 2 shows a general Rankine cycle layout for low-grade energy recovery with electrical output for road transport applications. In this cycle a low boiling point working fluid is vaporized by the ICE / FC coolant in an evaporator, it then rotates a turbine connected to an electric generator (48V electrical output for a mild hybrid HD truck powered with low carbon content fuel ICE, or 400V for a compatibility with a high voltage FC vehicle electric grid). Then, the working fluid is condensed by a cooling source in the condenser, and finally pumped again to close the cycle.
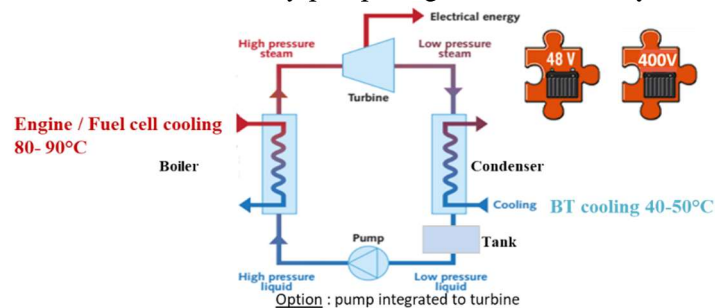
Figure 2: Low-grade electric Rankine cycle general layout

As presented in the introduction, the main drawback of low-grade waste heat recovery for road vehicles is the cold sink availability to cool down the working fluid to a targeted low temperature required for ORC cycle performance. As a solution, an additional radiator can be inserted in the front grille of the vehicle to take advantage of vehicle speed induced air cooling or with a deported radiator equipped with a fan that has a large energetic cooling cost impact (Reiche et *al*., 2020). Whatever the solution, it often leads to severe architecture constraints with generally limited space available in the vehicle. To improve integration of low-grade ORC and make it favorable for commercial vehicles, an original layout is proposed: ORC system and energy converter (ICE or PEMFC) are coupled together with a split cooling circuit including a unique radiator. In this solution, the radiator is alternatively used as a low temperature radiator for ORC condenser cooling and high temperature radiator for the ICE / FC cooling when ORC is non-operative. **Figure 3** explains the general layouts for both converters.
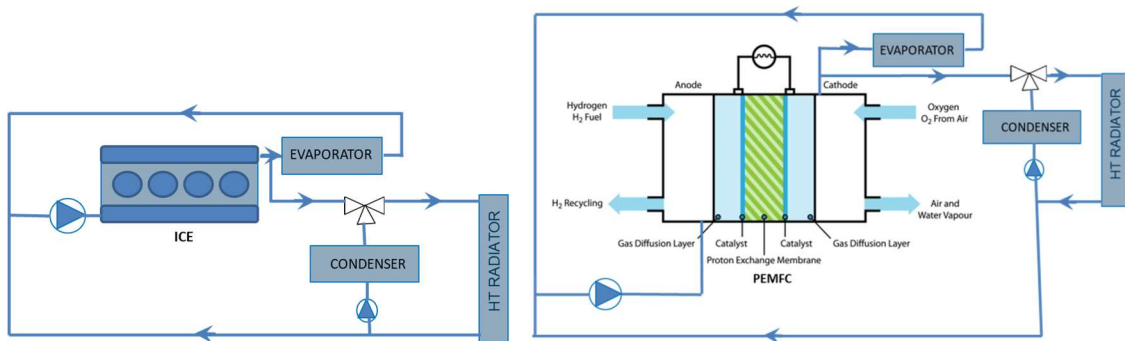
Figure 3: *Cooling circuit layout with split cooling radiator for ICE or PEMFC with ORC*

As described in the previous figures, in this solution, the ORC evaporator is in parallel with a three-way thermostat to allow heat recovery at ICE / FC coolant maximum temperature and to ensure, whatever the load, a thermal safety for the energy converter. The ORC condenser is located on a branch of thermostat, creating a low temperature cooling loop with the front radiator if the thermostat is closed considering the engine side. For an HD truck application, the ORC system can be optimized for a flat average 80-90km/h driving condition and is operative if the thermostat remains closed. Three operating phases can be considered as depicted in **Figure 4**. During powertrain warm-up or idle condition, the ORC system as well as the thermostat and the radiator are non-operative. The cooling loop is bypassed to improve ICE / FC warm up for fuel efficiency. When the converter is hot, the ORC system starts to recover heat losses from the coolant. The thermostat remains closed and the radiator is used to cool down at low temperature the ORC condenser. During this phase heat losses from ICE or FC are converted into electricity by the ORC system and residual heat is released by ORC through the front radiator at low temperature. When the maximum heat power recoverable by the ORC turbine is reached, the thermostat starts opening to regulate cooling circuit temperature. The hot water from the energy converter is released and mixed with the low temperature coolant for ORC cooling. In these conditions, the ORC electric production is decreased and then voluntarily stopped by the control unit when the ICE / FC heat load becomes too high. During this phase the ORC system is bypassed, the radiator is only dedicated to ICE / FC cooling at high temperature as in a standard cooling circuit. With this solution only one radiator is necessary for the ORC system and the ICE / FC cooling, but ORC waste heat recovery is only limited to favorable defined conditions of use.
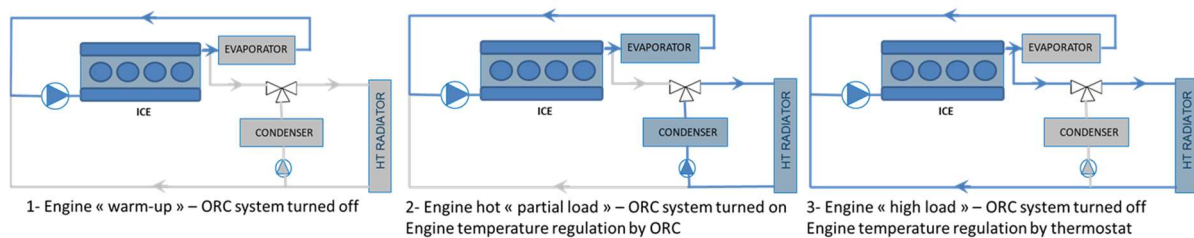

Figure 4: *ICE / ORC cooling operating phases*

To evaluate the performance of low-grade energy ORC with this specific cooling circuit layout, and properly size the ORC system, system simulations are performed considering two vehicle configurations where ORC is relevant : a long haul HD truck equipped with a hydrogen PEMFC or with a hydrogen ICE as energy converter.

2.2 Vehicle simulations

2.2.1 Introduction
For simulations, 1D physical models were set up to evaluate ORC performance onboard the vehicle. For simplicity, simulators built with Simcenter Amesim platform are split down to three technological levels. At the first level a general vehicle simulator is set up with the IFP Drive library. It describes the vehicle characteristics with its energy converter (ICE or PEMFC with battery and electric motor) and

estimates along a driving cycle, the power requested from the energy converter and battery if present to move the vehicle. For the simulation a standard ACEA VECTO long haul driving cycle is selected. This cycle simulates a generic representative 100km highway driving mission for a Class 5 tractor.

Then at the second level, a physical simulator of the ICE / FC estimates heat losses that are rejected depending on the power request (computed with the first level simulator) and which could be recovered by the ORC system. For the PEMFC, an efficiency map-based model of the stack with its ancillaries (namely the Balance of Plant) is available at IFPEN. This FC system efficiency map has been experimentally determined on a state-of-the-art FCEV with vehicle chassis dynamometer tests (Sery et al., 2021). For the conventional ICE, the heat losses rejected in the coolant circuit are estimated also from an experimental map-based model interpolated with the engine speed and torque (computed by the first level simulator).

At the third level, the ORC system with tank, pump, boiler, turbine, and condenser modelled with the specific Two-Phase Flow library in Simcenter Amesim receives as input parameters the heat losses estimated from the second level simulator. It is connected to the vehicle cooling circuit considering the radiator split cooling architecture proposed. With the connection, thermal coupling between truck cooling temperature and ORC system is modelled into this third simulator.

The simulator estimates the ORC electric net production to evaluate power to power efficiency, fuel saving of energy converter and economic profitability of ORC system. The goal of this 3-step system simulation from vehicle to ORC system is to estimate the ORC performances onboard the vehicle on realistic/ representative driving conditions considering a set of parameters for vehicle, future ICE / FC, and ORC system. The general simulation workflow is illustrated in **Figure 5**. In future works stronger coupling between simulators could be considered to refine impact of ORC electric power production on energy management strategy and truck fuel efficiency.
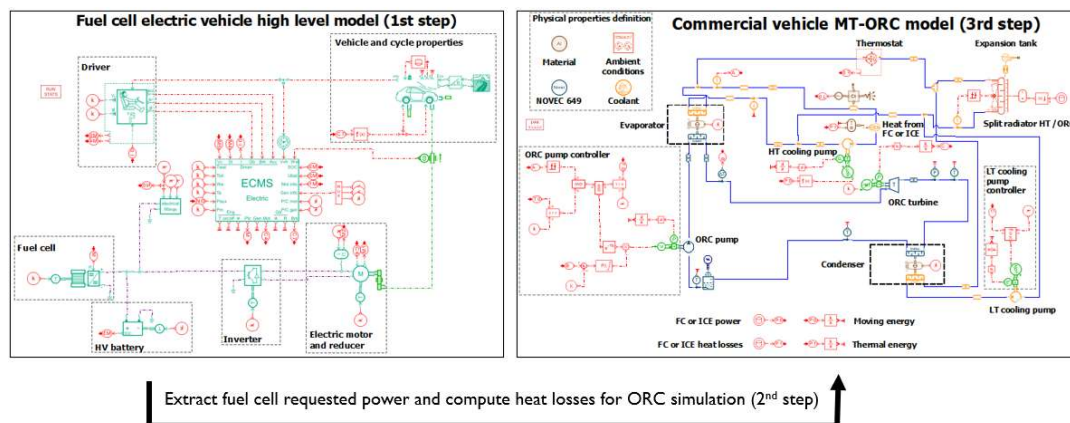
Figure 5: Simulation workflow illustration (fuel cell case)

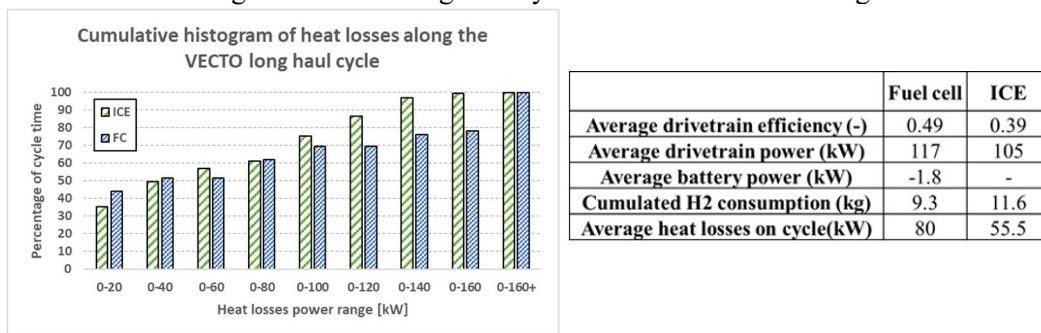2.2.2 Vehicle simulator description and results

The vehicle simulator includes a full description of the truck with its powertrain and the driver. For FCEV, an energy manager control unit called ECMS estimates the power split between battery and fuel cell to minimize hydrogen consumption throughout the cycle, considering current battery state of charge. **Figure 6** illustrates the main parameter values used for the simulated truck and powertrain.

Parameter	HD truck with H2 ICE	HD TRUCK with H2 Fuel cell
Vehicle	state of the art European Euro VI Class 5 truck effective mass : 33 t (15t empty + 18t payload) SCx : 6.3m^2 / Crr : 6.28kg/T	
Energy converter	Diesel ICE : inline 6 cylinders / 410kW / 2700Nm	PEMFC : 373kW / 1000 cells Battery : 13kWh / 120kW / 800V Electric motor : 500kW / 2000Nm / 95% peak eff
Transmission	12 speeds manual gearbox	1 speed reduction 10:1
Driving cycle	VECTO Long Haul	

Figure 6: Simcenter Amesim vehicle simulators main parameters

For the FC electric drivetrain, a hybrid configuration is selected for fuel efficiency with a high-power FC coupled with a medium battery capacity. The simulation is done considering a "charge sustaining strategy" keeping a constant high battery state of charge throughout the cycle. The battery is used to assist the fuel cell during high power requests to improve truck dynamic performance, for fuel efficiency at low truck load and to enable regenerative breaking. For the truck powered with a hydrogen ICE a conventional drivetrain without any electric assistance is considered. Compared to the FC with electric drive system, the ICE drivetrain has a lower mean efficiency, furthermore without any battery electric assistance, ICE efficiency is degraded on part load conditions. The conventional vehicle is less dynamic due to a lower converter torque capacity as compared to the electrified solution. The average power request is higher for FC. Hydrogen consumption along the cycle is lower for FC solution due to a better fuel efficiency of the converter and thanks to the battery assistance and capacity to partially recover vehicle braking energy. Heat losses rejected in the coolant is more scattered and higher for the FC despite better fuel efficiency because heat is mainly rejected in the coolant circuit whereas for the ICE heat losses are split between coolant and exhaust gas. **Figure 7** illustrates heat power losses distribution in the coolant circuit along the VECTO long haul cycle for both ICE/ FC configurations.

	Fuel cell	ICE
Average drivetrain efficiency (-)	0.49	0.39
Average drivetrain power (kW)	117	105
Average battery power (kW)	-1.8	-
Cumulated H2 consumption (kg)	9.3	11.6
Average heat losses on cycle(kW)	80	55.5

Figure 7: Thermal converter heat losses to the cooling circuit and main energy efficiency results for ICE and FC powertrains on the VECTO long haul cycle

2.3 ORC simulations

The previously estimated profiles of the ICE / FC heat losses to coolant are then injected as an input in the ORC simulator combining vehicle cooling circuit and ORC system. ORC simulator estimates recovery performance throughout the driving cycle, considering ORC components characteristics and truck cooling circuit performance.

2.3.1 ORC simulator description

The ORC simulator, like the vehicle simulator, is built with the Simcenter Amesim platform. The ORC system with fluid, tank, pump, boiler, turbine, and condenser is modelled with the specific two-phase flow library dedicated to air conditioning or ORC system design modelling. The working fluid is NOVEC 649, which was chosen based on previous works on thermodynamical performance optimization regarding heat source temperature, and low-speed turbine design constraints (Smague et *al.,* 2019). This working fluid is not the most performant for this low temperature cycle configuration, but it has a low Global Warming Potential (GWP) as well as a low flammability propensity, which are key factors for ORC dedicated to mobility.

In the system, ORC tank is modeled as a fixed volume stratified chamber with an initial pressure and liquid fraction in the volume. Fluid densities inside depend on the liquid volume percentage.

ORC pump is described as a volumetric compressor with a fixed displacement and considering constant volumetric, isentropic, and mechanical efficiencies for simplicity. Global pump efficiency (volumetric*isentropic*mechanical) is fixed to 40%.

The mass flowrate of the pump is computed with the following equation:

$$\dot{m}_{fluid} = \eta_{vol} * \rho_{suc} * N * disp \qquad (1)$$

Where: η_{vol} is the volumetric efficiency (-), ρ_{suc} is the fluid density on the suction side (kg/m^3), N is the pump speed (rpm) and *disp* is the pump displacement (m^3). Considering the pump isentropic efficiency, the enthalpy flow rate increase P_{pump} (W) is calculated with the following equation:

$$P_{pump} = \dot{m}_{fluid} * \frac{h_{dis} - h_s}{\eta_{is}} \qquad (2)$$

Where: h_{dis} is the isentropic discharge specific enthalpy (J/kg), h_s is the suction specific enthalpy (J/kg), and η_{is} is the pump isentropic efficiency (-), $\dot{m}_{fluid}$ is the fluid mass flow rate in the ORC loop (kg/s).

Both ORC evaporator and condenser are modelled with two half heat-exchangers: the ORC side with a two-phase flow half-exchanger and the water side with a single-phase half-exchanger. For both sides, thermal calculations are computed based on the steady-state efficiency NTU method (Incropera et *al.*, 2006). For hydraulic calculations, Mac Adams frictional correlations (McAdams, et *al.*, 1942) is used for two-phase flow side to estimate pressure drop. Both evaporator and condenser are modelled considering a fixed 98% effectiveness for heat exchange between half heat-exchangers model.

ORC turbine is modelled as an equivalent orifice with a fixed effective area and with a tabulated isentropic efficiency depending of the turbine pressure ratio. The selected efficiency values are relevant regarding previous experience (Smague et *al.*, 2018) and are depicted in **Figure 8**.

Turbine mechanical power is computed with the following equation:

$$P_{mech} = \dot{m}_{fluid} * \Delta h_{tur} \qquad (3)$$

Where Δh_{tur} is the difference of the specific enthalpy at the inlet and the outlet of the turbine (J/kg) estimated from fluid conditions.

The turbine mechanical output is then converted into electricity considering a fixed 85% efficiency for electric conversion (machine and inverter).
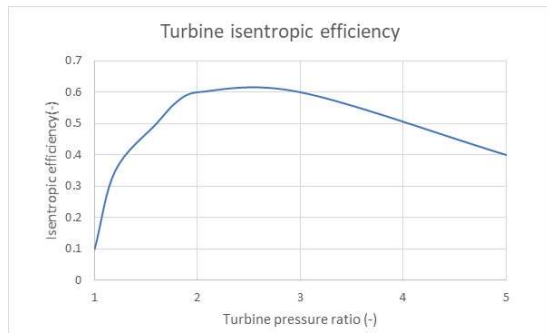
Parameter	Value
External air temperature	0 - 10 - 20°C (Northern EU)
Cooling circuit thermal inertia	ICE : 700kg / FC: 300kg / Coolant : 60L
HT Radiator heat exchange	function of air velocity and coolant flow
Air flow speed at radiator inlet	15% of vehicle speed
Engine thermostat	82°C or 92°C regulation
Turbine efficiency	Isent. : 10-65% / Elec : 85%
Heat exchanger effectiveness	98%
ORC pumps efficiency	40% (vol. mech.) and 85% (elec.)
ORC working fluid	NOVEC 649 (20kg)

Figure 8: Turbine isentropic efficiency and set of parameters for the ORC simulator

In simulation, the vehicle cooling circuit is connected to the ORC system using the unique radiator split cooling architecture proposed in this paper. From experience on light duty vehicle, a simple model reducing the air velocity on main radiator at an unfavorable 15% of the vehicle speed is implemented. The thermal simulator aims at evaluating ORC performance and effect of ORC on vehicle cooling temperature evolution under transient driving conditions and with this specific cooling circuit architecture. The simulator sketch is depicted in **Figure 5** – right and a general set of parameters is reported in **Figure 8** – right.

Detailed strategy for ORC pump control will not be detailed in the paper. Basically, the strategy purposed is to regulate the fluid mass flow regarding the two following constraints : trying to keep the outlet engine coolant temperature under thermostat opening temperature to prevent engine heat waste in the radiator, and keeping a superheated working fluid in the ORC system to prevent turbine liquid flooding. Therefore, the superheating profile varies throughout the driving cycle with thermal load.

The tuning strategy for ORC system is to select a turbine effective area to maximize the heat power transferred to the ORC system with the highest pressure considering the heat source, while keeping the energy converter thermostat closed to avoid coolant heat losses through the split radiator. The best turbine effective area gives the highest cumulative electric energy on the cycle. For both energy converters (ICE and FC), the ORC system and turbine effective area was sized to accept a level of heat power that frame mainly 60% of the VECTO cycle (73kW for fuel cell and 77kW for ICE). An 80kW heat recovery was selected as optimum setup for both configurations. Considering this thermal power, the converter temperature regulation and the external ambient temperature, an optimum turbine effective area can be defined for the selected working fluid, with the following equation:

$$S_{turbine\ throat} = \frac{\frac{P_{heat}}{\Delta h_{evap}}}{\rho_{WF\ turbine\ throat} * V_{WF\ sound(Ps,Ts)}} \qquad (4)$$

Where P_{heat} is the heat power at evaporator (kW), Δh_{evap} is the difference of the specific enthalpy at the inlet and the outlet of the evaporator (J/kg), $\rho_{WF\,turbine\,throat}$ is the fluid density at turbine throat (kg/m^3) and $V_{WF\,sound(Ps,Ts)}$ is the fluid speed of sound at turbine throat (m/s). To determine the conditions at turbine throat, the fluid density and the speed of sound in the working fluid are evaluated considering temperature and pressure at turbine inlet and an isentropic expansion in the turbine:

$$H_{S(PS,Ts)} = H_{0(P0,T0)} - \frac{V_{WF\,sound(P0,T0)}^2}{2} \tag{5}$$

$$S_{0\,(P0,T0)} = S_{s\,(Ps,Ts)} \tag{6}$$

Where H_s and H_0 are the static and total specific enthalpy at the turbine inlet (J/kg), and S_S and S_0 are the static and total specific entropy at the turbine inlet (J/kgK).

The fluid thermodynamic properties are estimated considering static conditions and thermodynamic properties of the fluid available in REFPROP NIST database.

Six optimum turbine capacities are defined considering the combination of 2 converter regulation temperatures (82°C and 92°C) and 3 ambient air temperatures (0-10-20°C). The other components of the ORC system are kept similar for all the simulations. The HT radiator has the same performance for the FC and the ICE powertrain configurations.

2.3.2 ORC simulation results

Status on power-to-power efficiency of ORC for both drivetrains is given in **Figure 9**. Net heat-to-power and power-to-power efficiencies are calculated as follows:

$$Heat\ to\ Power\ efficiency\ (\%) = \frac{ORC\ turbine\ energy\ (kWh) - ORC\ pump\ energy\ (kWh)}{ICE/FC\ thermal\ energy\ (kWh)} \tag{7}$$

$$Power\ to\ Power\ efficiency\ (\%) = \frac{ORC\ turbine\ energy\ (kWh) - ORC\ pump\ energy\ (kWh)}{ICE/FC\ energy\ (kW\ )} \tag{8}$$

		Ambient air temperature					
		FC configuration			ICE configuration		
		0°C	10°C	20°C	0°C	10°C	20°C
Energy converter regulation temperature	82°C	1.3%	1%	0.9%	1.6%	1.6%	1.4%
	92°C	1.4%	1.3%	1.1%	1.7%	1.7%	1.6%

Figure 9: ORC power to power efficiency evaluation synthesis

In the equation (7) and (8), *ICE /FC thermal energy* is the cumulative thermal energy losses from converter in the coolant circuit along the cycle, and *ICE/FC energy* is the cumulative mechanical/electrical energy produced by the converter to move the vehicle along the cycle.
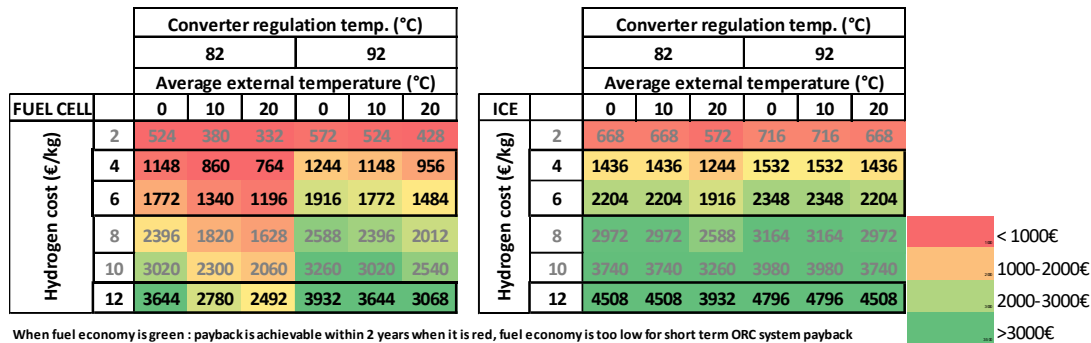
Despite higher supplied thermal energy along the cycle with FC, ORC efficiency is better for ICE. This is mainly due to lower heat flow peak and smaller standard deviation with ICE that lead to a better waste heat recovery by the selected ORC system, and less heat wasted due to thermostat opening. A larger turbine effective area for FC configuration will certainly decrease the ORC performance difference between powertrain configurations. Moreover, in the simulation the same radiator performance is considered whereas the FC radiator will be probably oversized compared to the ICE radiator in the vehicle due to higher losses in the cooling for FC. In addition, another working fluid with better thermodynamic properties regarding heat source temperature could improve both powertrains power-to-power efficiency. The most favorable configuration has the highest regulation temperature and the lowest air temperature. To be attractive for HD trucks, an ORC system must be cheap enough to reach payback time within 24 months. An economic evaluation can help to estimate the maximum cost of the MT-ORC system depending on its net efficiency. Considering general assumptions and power-to-power efficiency from the simulations, one can estimate ORC fuel benefits for 2 years driving profile (**Figure 10**). Fuel saving is estimated following the equations (9) and (10) and is accounted for different H2 cost roadmaps (Hydrogen council, 2020; IEA, 2019), engine regulation temperatures and external air mean temperatures. In the equation, the considered fuel consumption comes from VECTO simulation.

$$Fuel\ saving\left(\frac{€}{year}\right) = Mileage\left(\frac{km}{year}\right) * M_{fuel}\left(\frac{L}{km}\right) * \eta_{power\ to\ power}(\%) * Fuel\ cost\left(\frac{€}{L}\right) \tag{9}$$

$$Cost(€) = [Fuel\ saving\left(\frac{€}{year}\right) - maintenance\left(\frac{€}{year}\right)] * Payback(year) \qquad (10)$$

Parameter	Value
H2 consumption	10kg/100km
H2 cost	From 2€/kg to 12€/kg
Truck annual mileage	120 000km
ORC maintenance cost	50€/year
Payback time target	2 years

FUEL CELL — Converter regulation temp. (°C)

Hydrogen cost (€/kg)	82, 0	82, 10	82, 20	92, 0	92, 10	92, 20
2	524	380	332	572	524	428
4	1148	860	764	1244	1148	956
6	1772	1340	1196	1916	1772	1484
8	2396	1820	1628	2588	2396	2012
10	3020	2300	2060	3260	3020	2540
12	3644	2780	2492	3932	3644	3068

ICE — Converter regulation temp. (°C)

Hydrogen cost (€/kg)	82, 0	82, 10	82, 20	92, 0	92, 10	92, 20
2	668	668	572	716	716	668
4	1436	1436	1244	1532	1532	1436
6	2204	2204	1916	2348	2348	2204
8	2972	2972	2588	3164	3164	2972
10	3740	3740	3260	3980	3980	3740
12	4508	4508	3932	4796	4796	4508

Legend: < 1000€ ; 1000-2000€ ; 2000-3000€ ; >3000€

When fuel economy is green : payback is achievable within 2 years when it is red, fuel economy is too low for short term ORC system payback

Figure 10: General assumptions for cost analysis and fuel saving over a two-year operation

The current high cost of hydrogen (>10€/kg) makes the system clearly relevant for hydrogen mobility but not realistic. Future trend price for hydrogen dedicated to mobility (4-6€/kg) will imply more challenges for ORC system engineering and manufacturing but the low-grade energy ORC system recovery on energy converter cooling circuit with a split cooling radiator solution seems a realistic way to improve efficiency of decarbonized mobile applications despite a moderate efficiency compared to high-grade exhaust heat recovery. In this context, future hydrogen cost will be one of the main drivers on system profitability.

3   CONCLUSION

In this paper simulations were performed to evaluate low-grade waste heat recovery MT-ORC onboard future heavy duty truck equipped with decarbonized energy converters (H2 ICE or PEMFC). Heat recovery with ORC on energy converter cooling circuit appears to be an efficient and low-cost way to reduce fuel consumption for high mileage trucks. Whatever the energy converter solution (FC or ICE), a common or closed ORC system solution can be considered. The economic evaluation reveals that this solution can be profitable for northern European conditions with the current hydrogen cost trend and can be emphasized thanks to the split cooling radiator architecture proposed in order to reduce the global system integration cost and complexity. An optimized turbine design for each powertrain coupled with an improved working fluid selection and an enhanced radiator cooling device may increase this first fuel efficiency evaluation.

NOMENCLATURE

AC	air conditioning
CAC	charge air cooler
CO2	carbon dioxide
ECMS	equivalent consumption minimization strategy
EGR	exhaust gas recirculation
FC	fuel cell
FCEV	fuel cell electric vehicle
GWP	global warming potential
HD	heavy duty
ICE	internal combustion engine
ORC	organic Rankine cycle
PEMFC	polymer electrolyte membrane fuel cell
OEM	original equipment manufacturer

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