DEVELOPMENT OF A 48V ORC TURBO-PUMP FOR WASTE HEAT RECOVERY IN THE COOLANT OF LIGHT DUTY AND COMMERCIAL VEHICLES

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ABSTRACT

In the next 10 years, a more than 30% CO2 emission reduction is awaited on road vehicles in Europe. Complementary to electrification and hybridization, waste heat recovery (WHR) appears to be a solution to improve vehicle efficiency and provide significant reduction in fuel consumption of passenger cars and commercial vehicles.
In this context IFP Energies nouvelles accompanied by ENOGIA have actively worked on Organic Rankine Cycle (ORC) dedicated to transport to adapt this mature technology in stationary application to the small scale road transport industry.
A cost-driven approach using low temperature waste heat recovery was selected has key driver in this development. It is based on an Organic Rankine Cycle with the engine coolant as heat source and an integrated electrical compact kinetic turbine and pump as compression and expansion devices. Compared to other WHR based on exhaust gas heat recovery with high efficiency, this solution offers a safe, lightweight and low-cost module readily pluggable onto engines and compatible with 48V hybridization. Simulations and experimental results highlighted a potential of 2% and up to 3% fuel economy on regulatory cycle WLTP for light duty vehicle with thermal management.
In this paper results of system simulations, CFD simulations, design process of the electric ORC turbo-pump as well as experimental results will be presented.

1. INTRODUCTION

In a vehicle cruising on expressway, more than half of fuel's energy content is wasted as thermal losses in exhaust gases or in the cooling system; this value is higher for low load conditions where engine has a lower brake thermal efficiency generally limited below 40% for modern car engines.
Recovering part of these losses to improve engine efficiency would allow significant reduction in fuel consumption. As Waste Heat Recovery (WHR) system, Rankine cycle shows promising potential and is investigated by original equipment manufacturers and vehicle manufacturers.
Most of R&D projects related to ORC systems (Serrano *et al.* , 2015), (Horst *et al.,* 2014), (Ringler *et al.*, 2009), (Smague *et al.* , 2013) propose WHR solutions with direct recovery on exhaust gas or EGR to maximize recovery gains. Recent changes in legislation introduced in 2017 the Worldwide Harmonized Light Vehicles Test Procedure (WLTP) and the corresponding Test Cycle (WLTC) to replace the New European Driving Cycle (NEDC) procedure for the determination of emissions and fuel consumption for light-duty vehicles. The new cycle is of longer duration but much more transient than the previous one, leading to more discontinuities in available exhaust waste heat from the ICE to the WHR solutions. Some recent work (Kraljevic *et al.* , 2018) proposed to store exhaust energy as

sensible heat in steam accumulator to damp this transient recovery and facilitate the control strategy of ORC system.
Some other recent works have investigated waste heat recovery with ORC on engine coolant for hybrid heavy duty trucks combined with thermo-management (Furukawa *et al.* , 2014). By adding WHR, fuel economy can be expected since hybrid system can obtain not only regenerative energy in coast down cruising, but also waste heat energy from engine in a flat way or an uphill. Main advantages of HT engine cooling loss recovery versus exhaust gas recovery is the high latent heat of water leading to a stable temperature for ORC system even for vehicle cruising on transient cycle. Moreover about 1/3 of exhaust gas energy is uncollectable due to condensation concern in the exhaust.
Other recent works have investigated similar waste heat recovery with ORC on engine coolant with a mechanical output for expander and with an integrated ORC turbo-pump to maximize efficiency of ORC recovery and propose solution for non HEV truck (Smague *et al.* , 2018b).
In the future with the always more stringent emission regulations and after treatment solution, temperature of exhaust gas could decrease even more leading to always lower exhaust gas recovery potential for ORC.
Considering all these observations, a project was launched at IFP Energies nouvelles in partnership with ENOGIA to evaluate the relevance of engine coolant heat recovery ORC system for light duty vehicle and design the key components of the cycle: the ORC turbo-pump (Leveque *et al.* 2018).

2. SIMULATIONS AND ECONOMIC ANALYSIS

2.1 Hybrid simulation

To be relevant and attractive for customer, an ORC system must be cheap enough to reach a payback time within 24 and 36 months. Economic analysis can define the maximum cost of such an ORC system for the customer depending on its efficiency.
Numerical simulations have been done to evaluate a coolant heat recovery ORC system relevance on a light duty hybrid vehicle equipped with gasoline engine. Generic architecture for MT ORC system onboard HEV is considered (Figure 1) for the simulations. ORC system is plugged on ICE cooling circuit where evaporator recovers heat from coolant before it is released through HT front radiator. ORC condenser releases heat in a LT cooling circuit equipped with LT front radiator similar in size and performance as radiators used for charge air cooling on supercharged ICE. ORC Turbine and pump are connected to 48V HEV electrical network.

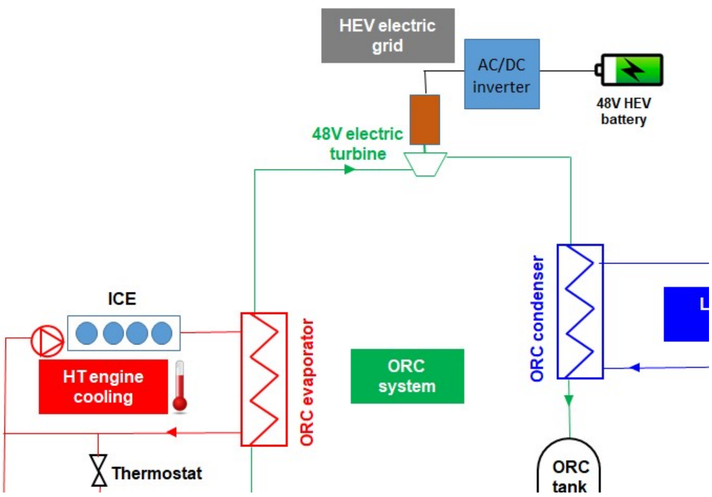

Figure 1: MT ORC system schematic view onboard light duty HEV

For plug-in hybrid vehicle, ICE is mainly used for highway driving conditions when electric traction is irrelevant due to electric battery autonomy. The simulations consider two highway driving conditions for this van C-segment vehicle. As expansion device, a turbo expander is selected with different matching suited to the heating power recoverable defined for the highway driving conditions and the working fluid selected.

Table 1: Driving conditions for light duty hybrid van vehicle

Vehicle speed ($km.h^{-1}$)	Engine power (kW)	Heat power dissipated at radiator (kW)
110	24	14
130	36	22

For electric production as well as fuel saving estimation, realistic assumptions for vehicle and ORC system are considered in the simulator as well as for cost analysis.

Table 2: Hypothesis for ORC cost analysis

C-segment vehicle	
Typology	Medium European van car
ICE Brake-specific fuel consumption on highway condition	215g/kWh (2025 roadmap trends)
Vehicle CO2 score on WLTC (thermal ICE only version)	120g CO2/km
Engine regulation temperature	90°C and 100°C (high temperature regulation)
Highway annual mileage	From 5 000 to 50 000km Highway time ratio = 40% - 110km/h / 40% - 130km/h / 20% dead time for ORC system
Electric production to vehicle fuel gain factor	1.4 (electric to fuel energy equivalence considering a 70% efficiency state of the art automotive alternator)
Gasoline fuel cost	1.6€/L (trend)
ORC system	
ORC turbine efficiency	52% global efficiency 65% (isentropic/mechanical conversion) 80% (electric conversion)
ORC turbine permeability	3 matchings for each engine regulation temperature 60mm² / 70mm² / 80mm² for 90°C regulation 40mm² / 50mm² / 60mm² for 100°C regulation
ORC pump / LT pump efficiency	Max 40%
Vehicle fuel penalty due to ORC mass	-0.5%

ORC electrical production is evaluated using system simulation with Simcenter Amesim. A 0D physical model of vehicle cooling circuits of the C segment vehicle is described. An ORC system model connected to the both cooling circuits of vehicle is set-up. This simulator allows simulations on transient cycles like WLTC to reproduce thermal behavior of vehicle and to evaluate ORC performance on transient cycle. For economic evaluation on C-segment van hybrid vehicle it is used only on hot engine stationary condition related to use of vehicle on highway driving condition.

Each component of ORC system is detailed in the sketch with previously described assumptions on performance. This model describes the characteristics of HT and LT cooling components (exchangers, radiators) as well as engine thermal and fluidic behavior of the C segment vehicle considered. For ORC cycle a physical description is also used considering simplified constant efficiency for condenser and evaporator, commercial pump performance and efficiency map, and estimated turbine performance coming from CFD. Simulations are run during enough time to reach engine cooling circuits thermal stabilization and ORC cycle stabilized electric production . Electric net output production including turbine production reduced by ORC pump and LT cooling pump consumption is then used for economic analysis.

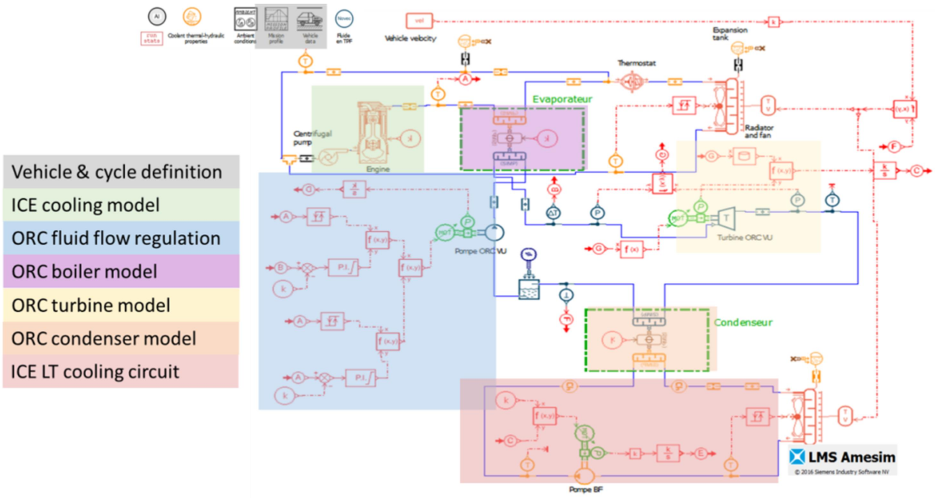

Figure 2: Overview of MT ORC model

Fuel saving and ROI are then estimated with ORC electric net output, a targeted cost for customer of 300€ for the whole ORC system and taking into account lump penalty due to ORC system mass as well as dead time to stabilize conditions for ORC production. The following equations (1), (2), (3), (4), (5) are used to evaluate the fuel gain and ROI :

$$\text{ORC net fuel saving (\%)} = \frac{\text{Electric to fuel gain} * \text{P ORC net (W)}}{\text{P vehicle (W)}} - \text{Mass penalty (\%)} \quad (1)$$

With

$$\text{P ORC net(W)} = \text{P ORC turbine (W)} - \text{P ORC pump (W)} - \text{P LT cooling pump (W)} \qquad (2)$$

$$Payback\ \text{time(year)} = \frac{\text{ORC annual gain} \left(\frac{€}{\text{year}}\right)}{\text{ORC system cost (€)}} \qquad (3)$$

With

$$\text{ORC annual gain} \left(\frac{€}{\text{year}}\right) = \frac{\text{Engine BSFC}\left(\frac{\text{g}}{\text{kWh}}\right) * \text{Highway power mix (kW)}}{1000 * \rho\text{fuel} \left(\frac{\text{kg}}{\text{m3}}\right)} * \text{Highway time} \left(\frac{\text{hours}}{\text{year}}\right) *$$

$$\text{Fuel cost} \left(\frac{€}{\text{L}}\right) * \text{ORC net fuel saving (\%)} * (1 - \text{dead time for ORC system (\%)}) \qquad (4)$$

And with

$$\text{Highway power mix (kW)} = \text{Engine Power 110kmh (kW)} * \text{Time ratio 110kmh (\%)} + \text{Engine Power 130kmh (kW)} * \text{Time ratio 130kmh (\%)} \qquad (5)$$

Fuel saving estimation is estimated between 1.9 and 4.5% depending on the turbine matching, the engine duty, and the ICE regulation temperature.

Table 3: MT ORC electric production and fuel saving estimation for HV with 100°C engine regulation

Electric power(W)	110km/h			
	Turbine e-power (W)	ORC pump e-power (W)	LT pump e-power (W)	ORC net e-power (W)
Turbine 40mm²	670	44	20	606
Turbine 50mm²	750	57	20	673
Turbine 60mm²	667	52	20	595

Electric power(W)	130km/h			
	Turbine e-power (W)	ORC pump e-power (W)	LT pump e-power (W)	ORC net e-power (W)
Turbine 40mm²	693	44	20	629
Turbine 50mm²	827	60	20	747
Turbine 60mm²	934	82	20	832

100°C ICE temperature regulation			
Net fuel saving (%)	Highway 110km/h (%)	Highway 130km/h (%)	Average
Turbine 40mm²	3.1	2.0	2.6
Turbine 50mm²	3.5	2.5	3.0
Turbine 60mm²	3.0	4.5	3.7

Considering a highway mix of 40% - 110km/h / 40% -130km/h / 20% - dead time for ORC system and a variable year mileage for vehicle on highway, an estimation of payback time for customer can be evaluated. As it could be seen on the graph below, payback time for customer has an asymptotic decrease, and a 3years payback is achievable for a 30kkm/year highway profile and best matching turbine selection for each engine regulation condition. Such calculation can easily be updated considering other conditions for selected vehicle. Moreover the gain for the customer, additional benefit can be considered for the car manufacturer with a reduction of the CO2 emissions and benefit on CO2 fees.

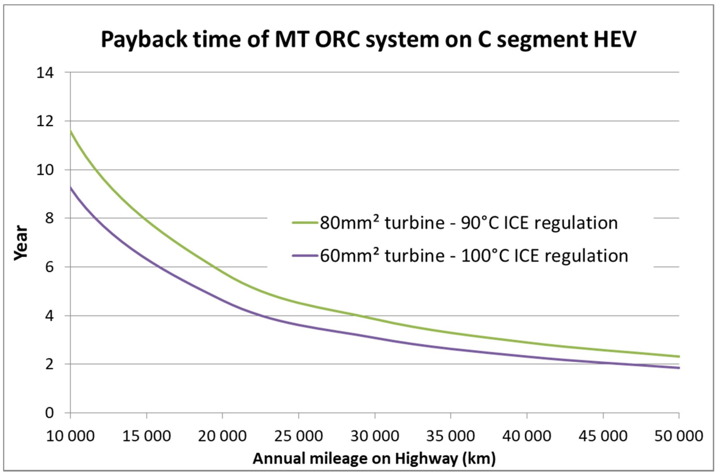

Figure 3: MT ORC system payback time estimation

2.2 Conventional gasoline vehicle simulation

Additionally to this fuel saving potential for customer on HEV, MT ORC system can also help car manufacturers to reduce CO2 emissions for vehicles equipped with conventional thermal ICE.
Simulations on regulatory WLTC have been done with Amesim physical model considering now a lighter compact C-segment vehicle equipped with gasoline engine. For this vehicle the whole cycle is performed with ICE. CO2 reduction potential of ORC can be evaluated. Simulation shows that mainly half of the cycle is dedicated to engine warm-up where ORC system can't be operated due to low temperature of engine HT cooling.
Heat release through HT radiator is up to 25kW in transient for highway driving conditions but is much lower in average and very variable depending on parts of the cycle . Synthesis graph below presents an outlook of WLTC for compact C-segment example with mean mechanical and thermal power available for ORC system as well as simulation of the evolution of ICE HT and LT cooling temperatures during WLTC with ORC running. With simulation ORC net electric production is estimated and cumulated along the cycle. This cumulated net energy coming from ORC is compared to vehicle moving energy to estimate a CO2 gain (g/km) on the cycle taking into account the mass penalty (Table 2). This simulation is done considering 2 turbines matching and a 95°C ICE temperature regulation. We can see on the graph below that turbine matching has a significant influence on energy recovery shape during the transient cycle. A small turbine recovers more energy on low ICE load condition but recovery is limited on extra-high part of the cycle whereas a bigger turbine can recover maximum power on highway driving condition but is less efficient at low load. For the whole cycle with the retained hypothesis mainly 2g CO2/km reduction can be expected for the vehicle.

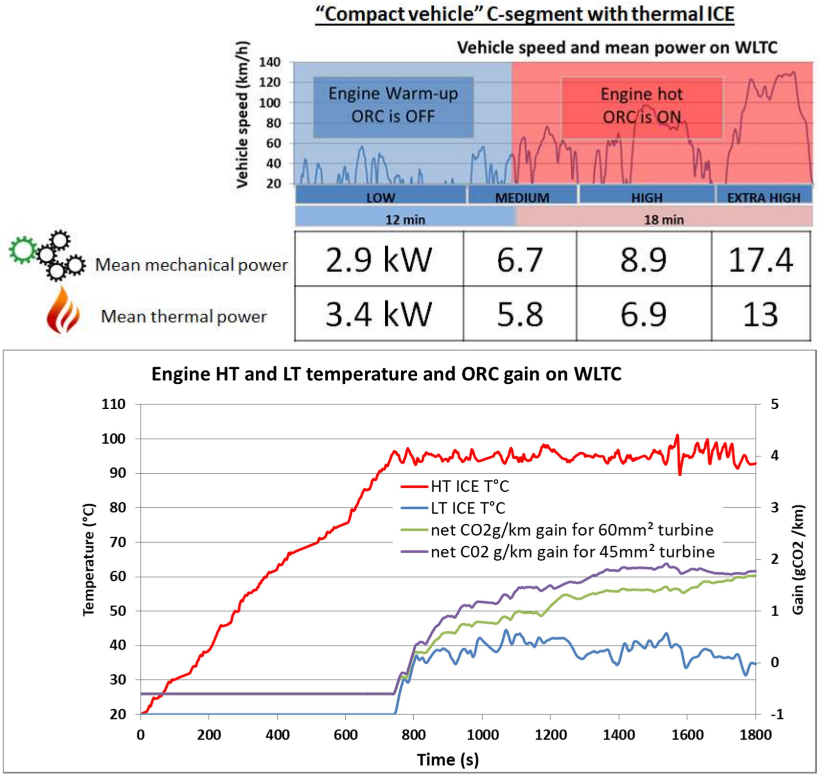

Figure 4: ICE temperature and ORC gain estimation on WLTC for C-segment vehicle

3. ORC TURBO-PUMP DESIGN

3.1 Definition of the application range

The whole ORC system was designed considering available heat flux for heat recovery on compact C-segment conventional vehicle ranging from 7 to 15kW. This mean value is chosen considering mean heat power available on "high" and "extra-high" part of the WLTC at a temperature of 95°C (Figure 4).

Thermodynamic simulations have been done to select the working fluid. As input parameters, it was considered a heat recovery on engine coolant at 368 K (95°C). ORC system is combined with a cold temperature source at mainly 308 K (35°C) present in vehicle for charged air or EGR cooling. Considering realistic pinch in both ORC heat exchangers, the thermodynamic cycle was designed between 363 K and 313 K (90°C / 40°C).
Working fluid selected for ORC system is NOVEC 649 not only due to its better thermodynamic properties compared to the others but also for these following reasons:

- Low pressure cycle compared to working fluid commonly used for MT ORC cycle (R1233zd, R245fa) allowing to design a lighter ORC system dedicated to mobile applications
- High mass flow rate, making it easier the design of small scale turbine and maximize efficiency
- Lower environmental impact: the global warming potential of NOVEC 649 is 1.

Table 4: ORC working fluid properties

Fluid property	NOVEC 649
Chemical formula	$C_6F_{12}O$
Boiling point (°K)	322

Molecular weight (g/mol)	316
Critical temperature (°K)	442
Critical pressure (MPa)	1.88
Vapor pressure @ 25°C (kPa)	40
Heat of vaporization (kJ/kg)	88
Liquid density (kg/m3)	1600
Specific heat (J/kgK)	1103
GWP	1

3.2 Turbine fluidic design

Once defined the working fluid, the temperature and pressure in the cycle, an axial turbine was iteratively optimized using Ansys CFX CFD software.
For turbine blade optimization a DOE methodology was used with the help of a dedicated CAD shape optimization platform CAESES™. After defining a reference design for the turbine stage, we proceeded to blade profile optimization under constraints of turbine permeability to find the most efficient geometry.

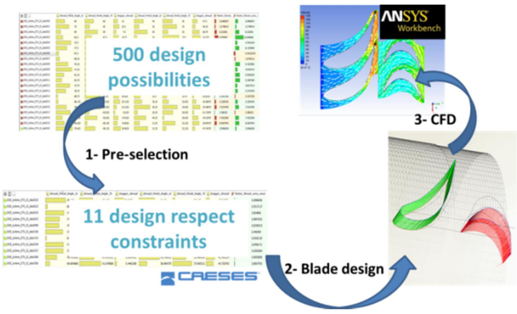

Figure 5: Methodology for turbine blade profile optimization

The final turbine stage performance estimated with CFD at turbine nominal condition are presented in the table below and illustrated with a flow velocity picture. For CFD simulations, steady state conditions on only one turbine sector is evaluated for fast calculation. Meshing is automated and a constant tip clearance is taken into account between turbine wheel and casing. Fluid model description is coming from Refprop. Heat transfer model selected is “Total Energy” in Ansys CFX and “k-epsilon” is used as turbulence model. Simulation is done considering fluid massflow inlet and static pressure outlet to ensure calculation convergence.
Simulation shows a potential of 784W for turbine mechanical output power in design condition corresponding to 15kW of heat power related to WLTC “extra-high” turbine matching.

Property	Calculated value
High pressure	2.75 bar
Low pressure	1 bar
High temperature	85°C
Working fluid mass flow ($\dot{m}$)	0.113 kg.s^{-1}
Isentropic power (P_{is})	922 W
Isentropic efficiency (η_{isen})	0.85
Effective power (P_{eff})	784 W

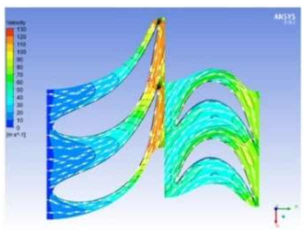

Figure 6: Turbine performance estimation and visualization of velocity vector through stage

3.3 Turbo-pump mechanical design

Considering system integration as well as cost optimization it was proposed to integrate the ORC fluid pump with the turbine in an unique component corresponding to an ORC turbo-pump. As energy output a small high speed electric machine was selected and integrated inside the turbo-pump. Electric efficiency was maximized on nominal turbine speed for a 48V output.

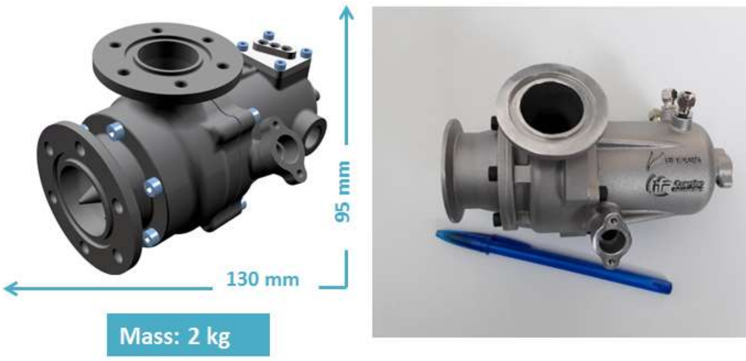

Figure 7: Generation 1 electric ORC turbo-pump

4. ORC TURBO-PUMP EXPERIMENTAL TEST

4.1 ORC skid presentation

To evaluate ORC prototype performance, a test bench was set-up in IFPEN premises. It allows to test the whole electric turbo-pump as well as the turbine and the pump stage separately. The skid is connected on the one hand to a hot water heat source electrically managed to allow variable thermal power up to 15kW (upgrade to 25kW is on-going) and heat temperature up to 105°C to emulate any kind of light duty and commercial vehicle with thermal-management; and in the other hand to a water cold sink with also variable temperature and flow management. A view of the ORC skid is presented in the picture below.

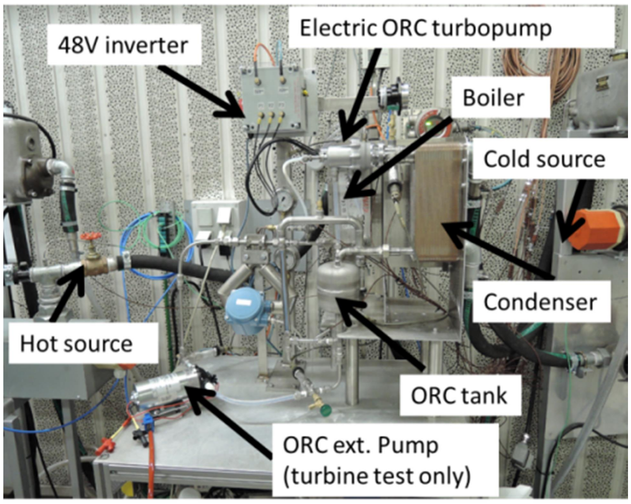

Figure 8: Test bed for ORC turbo-pump evaluation

4.1 Turbine experimental result

Turbine power and efficiency were measured for different heat powers, hot source temperatures, turbine rotational speeds and turbine matching. Two turbines matching performance curves are presented in the figure below. A first matching called WLTC "extra-high" corresponding to the turbine design optimized for the extra-high phase of WLTC and a second matching with smaller turbine permeability called WLTC "high" optimized for high phase of WLTC. Whereas turbine production is low for low vehicle speeds, it can reaches up to 500W for 16kW heat power (3.1% raw thermal efficiency) corresponding to 130km/h driving condition for compact C-segment vehicle. With the second matching, turbine electric production reach mainly 300W (3.2% raw thermal efficiency) for condition related to 80km/h driving condition.

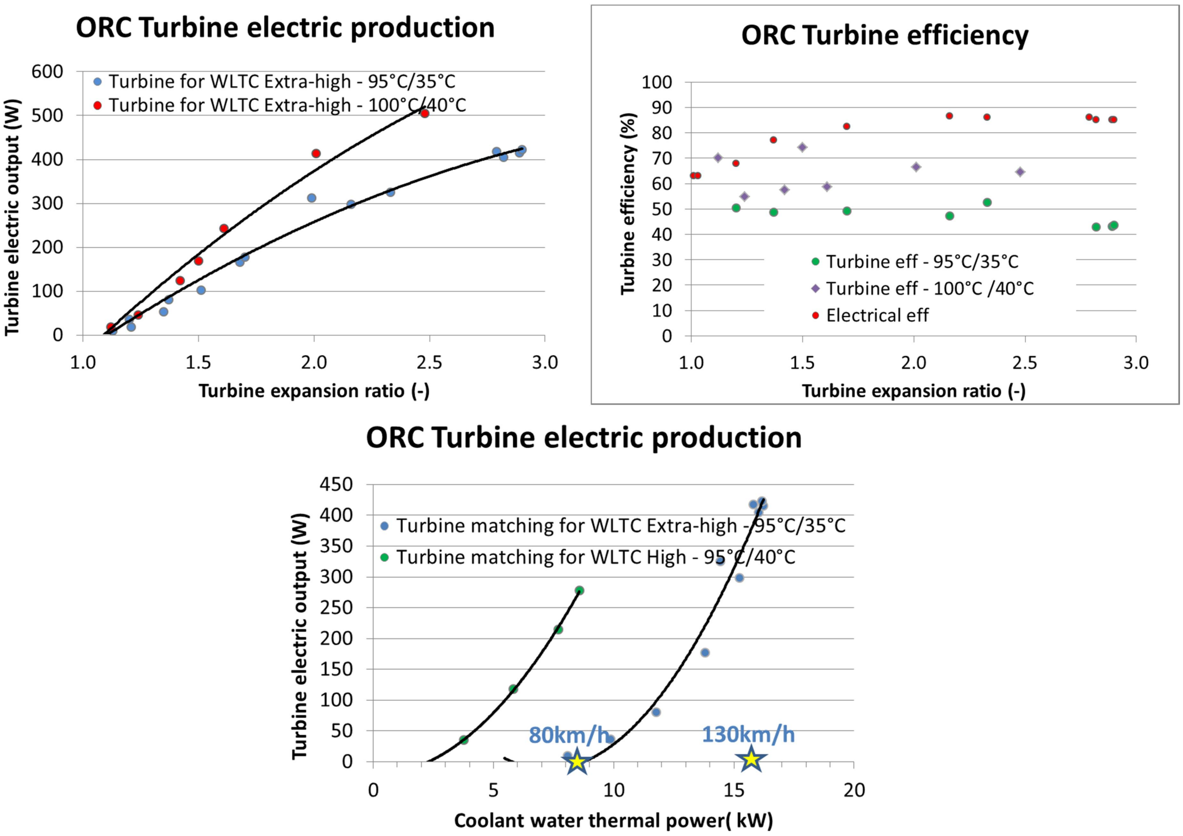

Figure 9: Experimental turbine performance

Additional work is under process to push turbine isentropic efficiency beyond 70% reducing tip clearance effect and optimizing blade profiles. Reflection is also ongoing to maximize turbine electric production whatever vehicle load conditions and limit turbine matching impact on global performance.
Pump result are not presented in this paper because test are still on-going. Target is to reach pump consumption below 10% of the turbine production.
ORC pump and whole electric turbo-pump behavior will be then evaluated in the next months on vehicle demonstration with an industrial partner.

5. CONCLUSION AND OUTLOOK

This paper describes the evaluation with simulations, design, and test of a 48V electric ORC turbo-pump dedicated to light duty as well as commercial vehicles using HT engine coolant as thermal heat source. The current design allows heat recovery up to 20kW / 105°C but is fully scalable. A full R&D process was led to define a functional system ready for implementation onboard vehicle. Improvement on engine thermal management with high temperature regulation as well as heat sink performance are key parameters for that medium temperature ORC solution. Optimization of the small scale turbine is also a key point to open up better fuel gain horizons. Following this first performance evaluation, new developments are under process with a car manufacturer to evaluate the technology onboard a commercial vehicle.

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NOMENCLATURE

CAD	Computer-Aided Design	
CFD	Computational Fluid Dynamics	
DOE	Design Of Experiment	
EGR	Exhaust gas recirculation	
Engine BSFC	Engine brake specific fuel consumption	(g/kWh)
HEV	Hybrid electric vehicle	
HT	High Temperature	
ICE	Internal Combustion Engine	
LT	Low Temperature	
Mass penalty	Fuel penalty impact of ORC mass	(%)
MT	Medium Temperature	
ORC net fuel saving	Fuel saving estimation with ORC system	(%)
ORC transient penalty	Fuel penalty impact of ORC transient behavior	(%)
Payback time	Payback time estimation for ORC system	(€/year)
P LT pump	Electric consumption of low temperature cooling circuit pump	(kW)
P ORC pump	Electric consumption of ORC pump	(kW)
P vehicle	Vehicle power resistance	(kW)
ROI	Return On Investment	