**Experimental comparison of the performance of a Waste Heat Recovery Organic Rankine Cycle system for truck application using R245fa and R1233zd**

**Ludovic Guillaume a, Arnaud Legros a, Vincent Lemort a,**

a Thermodynamics laboratory, University of Liège, Campus du Sart Tilman, B49, B-4000 Liège, Belgium

\* Corresponding author. Email: Ludovic.guillaume@ulg.ac.be; Tel. +32(0)3664823

**Abstract**

The reduction of CO2 emissions is a strategic goal of the EU where Heavy Duty Vehicles (HDV) can contribute in a relevant way. A very promising solution is to recover the waste heat, which is about 50 to 60% of the combustion energy. Transforming this heat in mechanical or electrical energy will enable to increase the overall energy efficiency of the vehicle. Consequently, the fuel consumption and the CO2 emissions will be reduced.

As being adopted for large stationary applications, the heat re-use can be performed by means of an external combustion engine, such as the Organic Rankine Cycle (ORC), using the waste heat as energy source. However, the adoption of such technology in the automotive domain requires specific R\& D activities to select and develop the components, identify the most appropriate system architecture and integration level so to achieve sustainable cost and the reliability requirements.

A significant part of these activities is devoted to the selection of the expansion machine and working fluid. Within this context, the objective of this study is to compare the performance of an ORC system using a radial turbine and exploiting the waste heat out of a truck for two working fluids: R245fa and R1233zd.

A test rig integrating the turbine was built. This turbine was developed mainly using components of truck turbochargers and was directly coupled to an electrical generator. The waste heat of the exhaust gases and the recirculated gases of the truck were simulated using an electric oil boiler associated with the ORC loop. The electrical power supplied by the turbine, was then dissipated in a load bench while the condenser was cooled by a water loop.

Measurements in steady-state were performed in order to evaluate the performance of the turbine when varying its pressure ratio, its rotational speed and the mass flow rate for various oil temperatures and mass flow rates.

**Keywords** waste heat recovery, organic Rankine cycle, experimental comparison, R245fa, R1233zd

**2 Material and methods**

The schematic layout of the ORC test rig is shown in figure 1. The system is a regenerative cycle equipped with radial turbine characterized by a power limited to 3.5 kW. This turbine was developed mainly using components of truck turbochargers and was directly coupled to an electrical generator. The bearings are lubricated and the generator is cooled through two by-pass pipes that go from the pump outlet to the turbine as shown in figure 1.The variable speed membrane pump, through the asynchronous generator, was connected to an inverter that allows controlling the rotational speed of the machine. The waste heat of the exhaust gases and the recirculated gases of the truck were simulated using an electric oil boiler associated with the ORC loop. The electrical power supplied by the turbine generator was then dissipated in a load bench while the condenser was cooled by a water loop. A liquid receiver is placed between the condenser outlet and the subcooler inlet.



Figure 1: Schematic layout of the ORC test rig

**2.1 Experimental Investigation**

Several points were measured for both R245fa and R1233zd. These points were obtained by keeping the system at a stable condition for a minimum of 15 minutes and by averaging the measurements over a period of 2 minutes. Only some of these points are presented here.

The condenser pressure was varied between a value of 2.5 and 4.5 bar for R245fa and between 3 and 4 bar for R1233zd.

The working fluid flow rate was imposed by varying the pump rotational speed, the evaporating pressure being imposed by the turbine. The condensing pressure is imposed by adjusting the cooling flow rate in the condenser.

The rotational speed of the turbine was varied as well as the bearings lubrication flow rate, responsible of windage losses.

Comparing the performance of the ORC systems could be done for same saturation temperature levels of the two fluids. This would enable an objective comparison of both fluids in case of exact same heat sources and heat sinks temperature and mass flow profiles. The result of this comparison would mainly be that the pressure level corresponding to the same saturation temperature is lower for R1233zd than for R245fa. The condensing pressure is therefore reduced but just as the evaporating pressure is reduced. But the latter is much more reduced and so is the pressure ratio and the turbine power production (figure 2).

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| Figure 2(a): Evaporating and condensing temperature levels for both R245fa and R1233zd | Figure 2(b): Evaporating and condensing pressure levels for both R245fa and R1233zd |

In this study the performance of the system is compared for same condensing temperature levels and evaporating pressure levels. This enables a comparison more adapted to the WHR on truck application for which the main constraint is generally the condensing temperature because of the mid-to-high temperature of the heat sink. But the evaporating pressure can be optimized and so there is no reason to keep the same temperature levels on the evaporator side.

**3 Results & discussion**

The heat balances over the heat exchangers are calculated in order to ensure the consistency of the measured data. As it can be seen on figure 3(a) and (b) , the balances are good. Both exchangers being insulated, the difference between the heat flow rate on the secondary fluid side and the one on the working fluid side is practically null. The maximal difference is around 3% for the highest temperature levels for which the uncertainty of the type-t thermocouples is slightly increased.

The energy conservation was then also verified for the rotating machines.

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| Figure 3 (a): Heat balance over the evaporator for both R245fa and R1233zd | Figure 3(b): Heat balance over the condenser for both R245fa and R1233zd |

The energy conservation being respected for all the components of the rig,, the measurements appearing coherent, the investigated comparison could be started.

For WHR on trucks applications, the main constraint is generally the condensing temperature because of the high temperature of the possible inboard heat sink. A fluid whose condensing pressure level is lower for the same saturation temperature will be preferred.

On the other hand, the evaporating pressure is generally a optimization variable of the WHR system. It is optimized to maximise the power output of the ORC.

Therefore it was decided to compare both fluids for the same condensing temperature and the same evaporating pressure (Figure 4).

Results are then compared in terms of

* working fluid mass flow rate
* pump power consumption: assuming a constant isentropic efficiency
* turbine power production

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| Figure 4(a): Evaporating and condensing pressure levels for both R245fa and R1233zd | Figure 4(b): Evaporating and condensing temperature levels for both R245fa and R1233zd |

Regarding mass flow, it can be observed that achieving the conditions of pressure of this comparison leads to the same mass flow rates for both fluids. Indeed, both fluids are practically identical regarding densities. The turbine imposing the flow rate for an inlet pressure, the evaporating pressures being the same, mass flow rates are also the same (Figure 5(a)).

Nonetheless, assuming a constant for the pump in each case, the pump power consumption would be slightly lower in case of R1233zd (Figure 5(b)).

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| Figure 5(a): Mass flow rate of working fluid both R245fa and R1233zd | Figure 5(b): Pump power consumption both R245fa and R1233zd |

Finally, the power produced by the turbine is compared for both fluids. For confidentiality reasons, but to give order of magnitudes:

* It is first compared from a hypothetical aspect (Figure 6(a)) assuming, as for the pump, a constant isentropic efficiency (35%).
* The real active powers produced during the experiment are then made dimensionless and compared (Figure 6(b)).

As it could be expected, the power produced is always higher using R1233zd. Because of the comparison method, the pressure ratio over the turbine is in each case higher than for R245fa. Since the mass flow rates are the same, the fluids begin very similar, the power produced is increased when using R1233zd.

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| Figure 6(a): Theoretical power produced by the expansion machine assuming a constant isentropic efficiency of 35 % both R245fa and R1233zd | Figure 6(b): Experimental power produced by the turbineboth R245fa and R1233zd |

**4 Conclusions**

The objective of this study was to compare the performance of an ORC system using a radial turbine and exploiting the waste heat out of a truck for two working fluids: R245fa and R1233zd.

A test rig integrating the turbine was built and several measurements points were realized for both fluids. Some of these points were then used to compare the performance of the ORC components for same condensing temperature and same evaporating pressures. This method was identified as appropriate considering the particular case of waste heat recovery for truck applications. Indeed, the constraint for the ORC, in this case, is generally the condensing temperature because of the mid-to-high temperature heat sinks only available onboard. A fluid with a lower condensing pressure for a same saturation temperature level is therefore suited. On the other hand, the evaporating pressure is generally optimized for these ORC systems. There was therefore no point to compare the same evaporating temperature levels

Results were then compared in terms of working fluid mass flow rate, pump consumption and turbine production. It was shown that the temperature and pressure conditions used for the comparison led to the same mass flow rates for both fluids. Nonetheless, the pump consumption was slightly lower in case of R1233zd. The power produced by the turbine was always higher when using R1233zd because of the higher pressure ratio resulting of the comparison method.

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