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OPTIMAL WASTE HEAT RECOVERY RANKINE BASED FOR HEAVY DUTY APPLICATIONS

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ABSTRACT
Even in nowadays engines which can reach 45% of efficiency a high amount of energy is released as heat to the ambient. The increase in oil prices compels manufacturers to focus on new solutions to improve fuel efficiency of truck powertrain such as Waste Heat Recovery Systems (WHRS). Over last few years a lot of studies have proven that there are a lot of hurdles (cooling margin, expansion machines, …) for a perfect match of such a system on a vehicle. The objective of this study is to define an optimal WHRS for a heavy duty vehicle.
The results presented hereafter are obtained thanks to steady state modeling. Two kinds of simulations will be used for this study. Simple thermodynamic simulations for the comparison of several fluids then a more complex model will be presented in the second part. This part will be more focused on the expansion machine and Rankine cycle arrangement choice thanks to the evaluation of fuel economy for each concept. The fuel savings will be determined by reducing real driving cycle to a relevant number of steady state operating points weighting by their fraction of operating type over the cycle
In the first part a comparison of the net output power (Shaft Power minus Pump consumption) and the cycle efficiency is done for several fluids in order to choose the best one from a performance and environmental point of view. On the second part the fuel economy in percentage is evaluated for different concept in order to determine the best system architecture answering to our integration constraints (heat rejection, cold sink, packaging …). New or often forgiven constraints will be presented in this section.
The optimization of a waste heat recovery system under vehicle constraint is rarely done. This analysis presents what are the limitations to take in account for having a perfect match between a Rankine cycle and a heavy duty vehicle.
This paper showed the importance of the application when designing WHRS. It yields to a better understanding when it comes to a vehicle integration of a Rankine cycle in a truck. The consideration of these limitations allow to maximize the WHRS output therefore of the fuel economy.

TECHNICAL PAPER

INTRODUCTION

Driven by future emissions legislations and increase in fuel prices engine gas heat recovering has recently attracted a lot of interest. Over the last decade most of the research focuses on energy recovery systems based on the Rankine Cycle ([1] [2] [6] [7]). These systems are called Waste Heat Recovery Systems (WHRS) can lead to a decrease in fuel consumption and lower engine emissions.
Recent studies have brought a significant potential for such a system in a HD vehicle ([3] [4] [5]). However before the cycle can be applied to commercial vehicle the challenge of its integration have to be faced. The work done in [3] shows that the main limitation to that installation will be the cooling system of the vehicle.
This paper is organized as follows. The first part will explain the different considerations to take of when designing a Rankine cycle for a HD application. In the second part studied system and mathematical models are given. In section three methodology is explained and results are analysed. Finally conclusions are drawn directions for future research work is discussed.

ASPECT TO CONSIDER WHEN DESIGNING WHRS FOR HD TRUCK

Hot Sources

On a commercial vehicle a certain number of heat sources can be found like Exhaust, Coolant or Oil flow. These ones have several grade of quality (temperature level) and quantity (amount of energy). The number and the arrangement of these heat sources in the cycle can vary depending on the fuel economy targeted. However more heat sources will bring more complexity and more challenges for the design of the system (fluid, expansion machine, control).

Cold Sinks

On a HD Truck the only heat sink available for this application is the vehicle cooling module. Integration of a WHRS into this one will result on a higher load of the cooling system and limit the amount of waste heat that can be converted into useful work or lead to an increase in fan consumption which can cancel any benefits due to the WHRS. This is dependant of many factors such vehicle speed and ambient temperature. As such, complete system analysis is necessary to find the optimal way of recovering heat into a vehicle.

Impact on the engine performance and emissions

The engine operation will be influenced by the introduction of a WHRS. For example as the WHRS will share the cooling system of the vehicle the charge air cooling can be can be lower and have a negative behaviour on the engine performance. Another example is the use of Exhaust Gas Recirculation (EGR) as heat source. This will lead to a trade-off between EGR cooling and Rankine cycle performance which could impact negatively the engine emissions.

Working Fluid Choice

There are several aspects to take care when choosing a working fluid for this application. Unlike to stationary power plants where the main consideration is the output power here we have to deal with many things such as fluid deterioration, environmental aspects (GWP and ODP) or freezing. Similarly to the heat sources the working fluid will lead to different components design, operating strategies and fuel savings. Up to now several studies have tried to identify the ideal fluid for a Heavy Duty truck but no single fluid has been found.

WHRS Components

The integration of a Rankine cycle into a commercial vehicle presents new constraints. Packaging issues are often raised. Another drawback is the payload diminution therefore the volume power density must be maximized. The components must also fulfil the durability and the maintenance requirements of a commercial vehicle. The cost impact is also important since the cost pressure inherent to the automotive industry is more and more important.

RANKINE CYCLE MODELING

Rankine process

The Temperature Entropy (T-s) diagram in figure 1 shows the associated state changes of the working fluid through the Rankine cycle.

- The pressure of the liquid is increased by the pump work up to the evaporating pressure
- The pressurized working fluid is pre-heated, vaporized and superheated in a heat exchanger. This heat exchanger is linked to the heat source.
- The superheated vapour expands for Evaporating pressure to Condensing pressure in an expansion device doing mechanical work.
- The expanded vapour condenses through a condenser (linked to the heat sink) releasing heat.

In this process the changes of states in both the pump and the expander are irreversible and increase the fluid entropy to a certain extent.

Simple modelling of a Rankine cycle

In order to simulate a high number of working fluids an ideal model of a Rankine cycle using one heat source have been developed. It won't represent a real system but allow a fast assessment of a various number of working fluid. It will give us the suitable working fluids for our applications.
This model is based on the enthalpy increase due to the heat input into an ideal heat exchanger. A Matlab routine has been specially developed in order to find the optimality into the Waste Heat Recovering Process in order to maximize the net output power (mechanical work produced by the expansion minus the pump work). This model is able to perform either subcritical or supercritical cycle which avoid the need for an evaporator and lead to a smaller system. In table 1 you can find the simulation model parameters.

Simulation Limitations		
Pump Isentropic Efficiency	%	65
Expander Isentropic Efficiency	%	70
Maximum Evaporating Pressure	bar	40
Minimum Condensing Pressure	bar	1
Maximum Pressure Ratio	-	40
Pinch Point	K	1
Minimum Quality after Expansion	-	0,9

Table 1 Ideal model parameters

Complete system model

Following to what have been said previously the need for a complete system model is obvious. This model including its control system is described below components by components.
Tank

The reservoir is modelled by a fixed volume which can be either vented to the atmosphere or hermetic (depending on the condensing pressure) in order to avoid sub atmospheric conditions.
In this volume mass and energy conservation equations are solved

$$\dot{m}_{out} - \dot{m}_{in} = \frac{\partial}{\partial t} m_{res} \quad (1)$$

$$\dot{m}(h_{in} - h_{out}) = m_{res} \frac{\partial}{\partial t} h \quad (2)$$

Pump

The working fluid pump is simply represented by a fixed displacement and an isentropic efficiency (volumetric efficiency is set to 1).

$$\dot{m} = \rho_{in} \frac{N}{60}\, Displacement \quad (3)$$

$$h_{out} = \frac{(h_{out\ is} - h_{in})}{\eta_{is}} + h_{in} \quad (4)$$

Evaporator and Condenser

Two main methodologies have been found in the literature to model heat exchangers with phase change: three zones or finite volume model ([8] xx). Here the finite volume approach is preferred.
Boilers and Condenser will be represented by a straight pipe in pipe counterflow heat exchanger which is a commonly adopted assumption when it comes to dynamic modelling of heat exchangers. This heat exchanger model is split into n longitudinal lumped volume of equal length Δx, where the conservation equations are applied. For each finite volume, three nodes can be defined: one referring to the fluid state in the internal pipe (Rankine fluid), one to the state of the metal constituting the pipe wall and another to the fluid state in the external pipe.

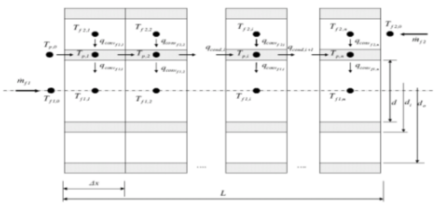


Figure 1 Heat Exchanger Schematic

Partial differential equations are solved in these three nodes:

- Outer Pipe (transfer fluid):

$$\dot{m}c_p(T_{in} - T_{out}) - \dot{Q}_{conv} = \rho V c_p \frac{\partial T}{\partial t} \quad (5)$$

- Pipe Wall:

$$\dot{Q}_{conv\ external\ side} - \dot{Q}_{conv\ internal\ side} = \rho V c_{p_{wall}} \frac{\partial T_{wall}}{\partial t} \quad (6)$$

- Inner Pipe (Working Fluid):

$$\dot{m}^t_{out} = \dot{m}^t_{in} - V\frac{\partial \rho}{\partial t} \quad (7)$$

$$\dot{m}(h_{in} - h_{out}) + \dot{Q}_{conv} = V\frac{\partial}{\partial t}(\rho h - P) \quad (8)$$

Since the working fluid will vaporize both energy and mass conservation have to be solved.

$$Q_{conv} = \frac{Nu * \lambda}{Dh} * S_{exchange} * \Delta T_{pipe-fluid} \quad (9)$$

The convections fluxes are computed using correlation for the Nusselt Number.In single phase Dittus Boelter is chosen for both fluids. In two phase Chen (for evaporation) and Shah (for condensation) correlations are used. Pressure drop in both fluids have been taken into account in order to simulate the real performances of the system.
Table 2 shows the different correlations used depending on flow conditions.

		Laminar	*Turbulent*
Heat transfer	*Single phase*	Nu = 4	Gnielinski
	Two Phase evaporation	Chen	Chen
	Two Phase condensation	Shah	Shah
Pressure drop	*Single phase*	Poiseuille	Blasius
	Two Phase	Friedel	Friedel

Table 2 Correlations used in heat exchanger

Valve

Compressible valve model have been done to represent the expander by pass valve which is used to protect the expander in order to not fill it with liquid. The fluid flow through the valve is modeled using a compressible valve equation of the form:

$$\dot{m} = c_d S_{eff}\sqrt{\rho_{in} P_{in} \phi} \quad (10)$$

Where the compressibility coefficient (Φ)is defined as

$$\phi = \frac{2\gamma}{\gamma - 1}\left(\varphi^{2/\gamma} - \varphi^{\gamma+1/\gamma}\right) \quad (11)$$

And the parameter φ is either the pressure or the critical pressure ratio depending on the flow conditions (sub or supersonic).

Impulse turbine model

Several studies have been done in order to choose the correct expansion machine for a WHR Rankine based system ([15]). Most of them where vehicle installation is considered turbine expanders are preferred for their compactness and their good performances ([6] [13]). Bigger advantage of volumetric expander like piston machines was the expansion ratio ([5]) but Kunte & Al ([16]) have shown a single stage partial admission turbine with an expansion ratio of 40 and really good performance at tolerable speed for a vehicle installation. Here a simple approach is chosen for kinetic expander types similar at [8].
The turbine nozzle is represented by the following equation

$$\dot{m} = K_{eq}\sqrt{\rho_{in}P_{in}(1 - PR^{-2})} \quad 12$$

And the isentropic efficiency is computed thanks to equation

$$\eta_{is} = \eta_{is\,max}\left[2\frac{c_{us}}{c_{us\,max}} - \left(\frac{c_{us}}{c_{us\,max}}\right)^2\right] \quad 13$$

Where

$$c_{us} = \frac{u}{c_s} = \frac{\omega R}{2\sqrt{h_{in\,turb} - h_{out\,turb\,is}}} \quad 14$$

Model is developed using literature data such as [16], [17] and the similarity relations [14].

Other heat exchangers

As described earlier the Rankine model influences the cooling system of the vehicle. This one is modeled using a classical NTU approach.

$$\dot{Q}_{air} = \dot{m}_{air}cp_{air}\epsilon(T_{coolant\,in} - T_{air\,in}) \quad 15$$

$$NTU = \frac{UA}{(\dot{m}cp)_{min}} \quad 16$$

For a given geometry ϵ can be calculated using correlations based on the heat capacity ratio.
In our case radiators are parallel configuration and the effectiveness can be written:

$$\epsilon = \frac{1 - e^{-NTU\left(1+\frac{(\dot{m}cp)_{min}}{(\dot{m}cp)_{max}}\right)}}{1 + \frac{(\dot{m}cp)_{min}}{(\dot{m}cp)_{max}}} \quad 17$$

Coolant pump

The coolant pump model used is a map based model function of engine speed and pressure rise. This one has been sized to deliver enough subcooling even at high engine load.

Control System

Simple PID has been used to control the superheating after each boiler (by acting on the working fluid pump rotational speed). When using Egr as heat source another function is added to respect the Egr temperature after the boiler in order to keep it between certain limits and not impact too much engine performance.
A logical controller is also needed for the expander bypass valve. This one is based on superheating and evaporating pressure in front of the turbine in order to avoid droplets formation which can rapidly destroy the machine. The minimum superheating to open the bypass is defined as the superheating giving a saturated vapor at the end of the expansion process for the maximum isentropic efficiency of the expander (this is only needed for wet fluids).

Thermodynamic properties

The thermodynamic properties are computed from the "Refprop" software ([11]). This tool is based on the National Institute of Standards and Technology (NIST) database. For sake of simplicity and computational time 2-D table have been created function of Pressure and Enthalpy.

MODELING RESULTS

<u>Boundary conditions used</u>

In table 3 you can find the different inputs data used for this study.
These operating points have be chosen to represent a classical long haul driving cycle and weighted according to the percentage of energy used on each operating point. Operating point number 8 have been identified as designing point whereas operating points 7 and 9 are considered here as critical due to the high engine load and the low vehicle speed.

Operating Point	-	1	2	3	4	5	6	7	8	9
Engine Speed	rpm	859	978	1000	1038	1157	1157	1157	1194	1500
Engine Load	%	25	50	100	75	25	50	75	100	100
Vehicle Speed	km/h	20	85	85	85	75	85	30	85	50
EGR mass flow	g/s	43	33	53	57	63	65	71	59	83
EGR temperature	°C	280	374	668	543	329	450	525	674	624
Exhaust mass flow	g/s	69	156	304	238	150	227	271	369	393
Exhaust temperature	°C	259	291	434	365	266	315	352	435	396
Weight Factor	%	0,47	3,03	8,13	4,67	21,32	15,62	11,28	29,11	6,38

Table 3 Boundary conditions

For the working fluid side the Evaporating pressure has been limited to 40 bars and no sub-atmospheric condensing pressure has been retained.

Methodology

This section shows the process used during this study. This one is divided into four steps:

- First the simple model is used to define the fluids which can be suitable for the considered boundary conditions. This is done taking into account environmental legislation which eliminates different fluids according to their chemical properties such as GWP or their attendance on the GADSL. Each operating points will be simulated using two condensing temperatures 60°C and 90°C.
- Then the cooling package model is simulated in order to know the temperature levels corresponding to different heat rejections. This will help when it comes to components parameters design.
- Last the complete rankine model including cooling loop is simulated on the 9 operating points allowing us to evaluate the fuel economy.

Rankine architecture investigated and components:

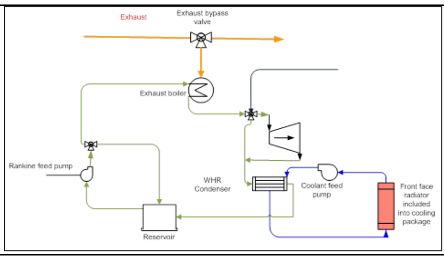
Several studies have been done in the field of waste heat recovery Rankine based systems for mobile applications. A screening of the different heat sources available have been done in [4] and shows that the most promising ones are the Egr and the Exhaust streams. For this study we will only focused on these two hot sources since they present the higher grade of temperatures among other sources.
In figure 2 you can find the different Rankine cycle arrangement studied.
In these sketches the different components modeled are shown: a working fluid pump, Evaporators (for Egr or Exhaust streams), an expander by pass valve to protect it if flow conditions are not sufficient to drive the expander, an expansion machine, an indirect condenser and a tank. Moreover similar to [3] the coolant loop for the water cooled condenser is also represented and includes a front face radiator and a coolant pump to drive the cooling fluid through the Rankine condenser. The serial arrangement showed below is recovering heat from Egr then Exhaust. It has been chosen in this order to limit the Egr temperature after the heat exchanger since the boiler configuration is counter-current (the lower the fluid temperature at the inlet of the heat exchanger is the lower the Egr gas temperature at the outlet is)

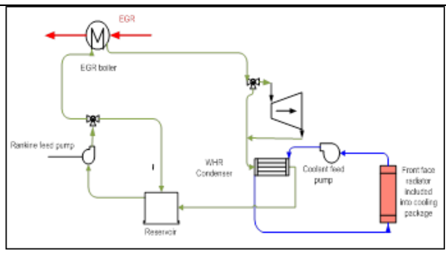
Fluid selection

The working fluid in a WHRS has to be chosen not only from a performance point of view but also from a legal and environmental one. Several aspects have to be taken into account such as:

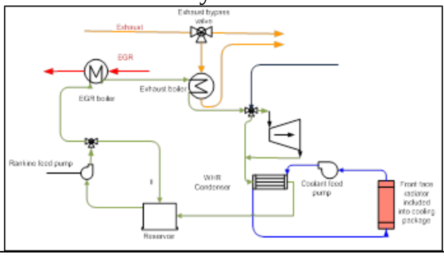
- Its chemical class: CFCs have been banished by the Montreal Protocol and HCFCs production is planned to be phased out by 2030.
- Its presence on the GADSL.
- Its chemical properties like the GWP, the ODP or the Risk Phrase.
- Its classification according the NFPA 704 classification (ranking above 1 in Health or Instability class)
- The freezing point which has to be above 0°C.

Exhaust Recovery:	Egr Recovery:





Egr and Exhaust recovery in serial:



Egr and Exhaust Recovery in parallel:

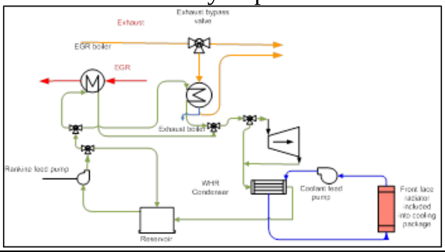


Figure 2 Rankine cycle architectures investigated

From a very important fluid list ([11]) all those which not respect the different criteria mentioned above have been removed. However as water is a good reference fluid we will keep it for the rest of the study.
The results presented hereafter coming from an ideal thermodynamic model presented in section 2 where all 9 operating points have been simulated for two condensing temperature 60°C and 90°C in each streams Egr and Exhaust separately. Results are presented in table 4.

			OPERATING POINT								
		Fluid Ranking	1	2	3	4	5	6	7	8	9
Heat taken from Egr	Condensing Temperature 60°C	1	Cyclopentane	Acetone	Water	Water	Acetone	Acetone	Water	Water	Water
		2	Acetone	Ethanol	Acetone	Acetone	Ethanol	Ethanol	Acetone	Acetone	Acetone
		3	R1233zd	Cyclopentane	Ethanol	Ethanol	Cyclopentane	Cyclopentane	Ethanol	Ethanol	Ethanol
		4	Ethanol	R1233zd	Cyclopentane	Cyclopentane	R1233zd	Water	Cyclopentane	Cyclopentane	Cyclopentane
		5	Novec649	MM	R1233zd	R1233zd	MM	R1233zd	R1233zd	R1233zd	R1233zd
	Condensing Temperature 90°C	1	MM	Acetone	Water	Water	Acetone	Water	Water	Water	Water
		2	Cyclopentane	Ethanol	Acetone	Acetone	Ethanol	Acetone	Acetone	Acetone	Acetone
		3	Acetone	Cyclopentane	Ethanol	Ethanol	Cyclopentane	Ethanol	Ethanol	Ethanol	Ethanol
		4	Ethanol	MM	Cyclopentane	Cyclopentane	MM	Cyclopentane	Cyclopentane	Cyclopentane	Cyclopentane
		5	R1233zd	Water	MM	MM	Water	MM	MM	MM	MM
Heat taken from Exhaust	Condensing Temperature 60°C	1	R1233zd	Cyclopentane	Acetone	Acetone	R1233zd	Acetone	Acetone	Acetone	Acetone
		2	Acetone	Acetone	Ethanol	Ethanol	Cyclopentane	Cyclopentane	Ethanol	Ethanol	Ethanol
		3	Cyclopentane	R1233zd	Cyclopentane	Cyclopentane	Acetone	Ethanol	Cyclopentane	Cyclopentane	Cyclopentane
		4	Ethanol	Ethanol	Water	R1233zd	Ethanol	R1233zd	R1233zd	Water	R1233zd
		5	Novec649	MM	R1233zd	MM	Novec649	Novec649	MM	R1233zd	Water
	Condensing Temperature 90°C	1	R1233zd	Acetone	Water	Acetone	R1233zd	Acetone	Acetone	Water	Acetone
		2	MM	Cyclopentane	Acetone	Ethanol	MM	Cyclopentane	Ethanol	Acetone	Ethanol
		3	Acetone	Ethanol	Ethanol	Cyclopentane	Acetone	Ethanol	Cyclopentane	Ethanol	Water
		4	Cyclopentane	MM	Cyclopentane	MM	Cyclopentane	MM	MM	Cyclopentane	Cyclopentane
		5	Ethanol	R1233zd	MM	Water	Ethanol	Water	Water	MM	MM

Table 4 best working fluids for each operating points

Fluids are ranked from 1 to 5 according to their performance (1 gives the best shaft power). This table shows that Water is the best fluid when considering Egr as heat source and Acetone for Exhaust heat recovering. Refrigerants such as R1233zd or Novec 649 show good results for hot source temperature under 280 for a low condensing temperature.
These first simulations results will limit the number of studied working fluid for the remaining part of this paper to the following fluids: Acetone, Cyclopentane, Ethanol and Water. These four fluids represent the highest number of occurrences at the three first places. As these fluids have similar volumetric flows it will be possible to use the same components characteristics with only some minor changes (e.g. throat diameter for the turbine model and pump displacement).

Cooling Package simulations

In order to know what will be the temperature level in front of the Rankine condenser we need to simulate the complete cooling package with the additional radiator dedicated to the WHRS. To not derate too much the engine performance and avoid excessive fan engagement only two architectures have been investigated: one where the additional radiator is housed between the CAC and the engine radiator (Cooling Architecture 1) and another where the coolant filling the rankine condenser coming from the engine radiator (Cooling Architecture 2).

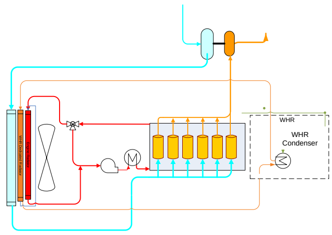
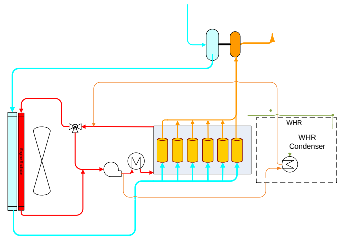
Cooling Architecture 1	Cooling Architecture 2
	

Figure 3 Cooling system architectures

For each configuration above we vary the heat rejection coming from a Rankine system from an Egr recovery only to heat from Egr added to the one from Exhaust on the 9 operating points cited in xx. These heats have been calculated with the formula $\dot{Q}_x = \dot{m}_x * cp_x * \Delta T_x$ where x is the source considered specific heat has been assumed at 1.09kJ/kg/K. For the ΔT the low egr temperature coming from engine control and for the exhaust rejection it has been assumed to 100°C. For these simulations an average ambient temperature of 20°C has been assumed.

Below you can find in table 5 the different heat rejections used.

Operating Point		1	2	3	4	5	6	7	8	9
Qegr	kW	9,2	10,1	31,9	27,5	16,2	24,9	32,3	35,8	45,9
Qexh	kW	12,1	32,2	104,0	67,6	26,8	52,1	71,5	123,4	117,6
Qegr + Qexh	kW	21,3	42,3	135,9	95,1	43,0	77,0	103,8	159,2	163,5

Table 5 WHRS Heat Rejection applied to cooling system

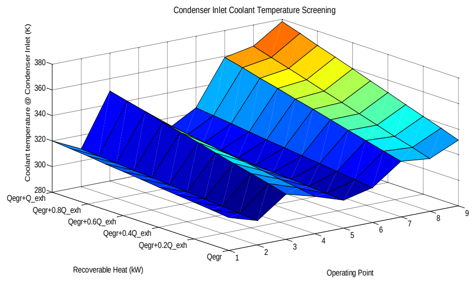
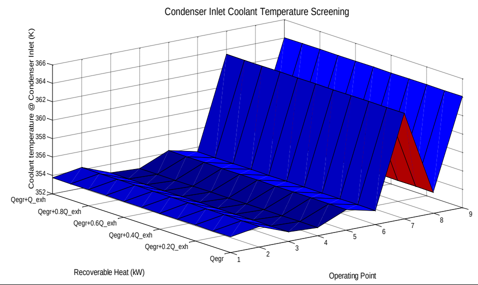
Cooling Architecture 1	Cooling Architecture 2
	

Figure 4 Coolant Temperature mapping

For the cooling architecture 1 the coolant temperature entering the Rankine condenser is varying between 50°C and 108°C. This is in the worst ideal case (i.e. all the heat is recovered and the expander is by-passed) which validates the assumption of a maximum condensing temperature of 90°C took previously.

On Cooling architecture 2 temperature is quite constant since it is regulated by the thermostat (the most important factor is the vehicle speed which will affect the air speed facing the cooling module). This will result into a higher opening of this one. It shows that a good installation of a Rankine cycle into a heavy duty vehicle will ask a smarter heat management especially in hot ambient conditions (to avoid fan engagement) and reduce coolant temperatures.

Fuel Economy analysis

Now the fuel economy has been analyzed on the 9 operating points and the two cooling architectures (Config 1 and 2) for the four working fluids chosen previously. The savings are computed thanks to the weight factors presented in table 3.

All components have been calibrated thanks to performance data from literature ([16] [17] [18]) since no test data were present at the time of this study.

Exhaust Recovery:	Egr Recovery:

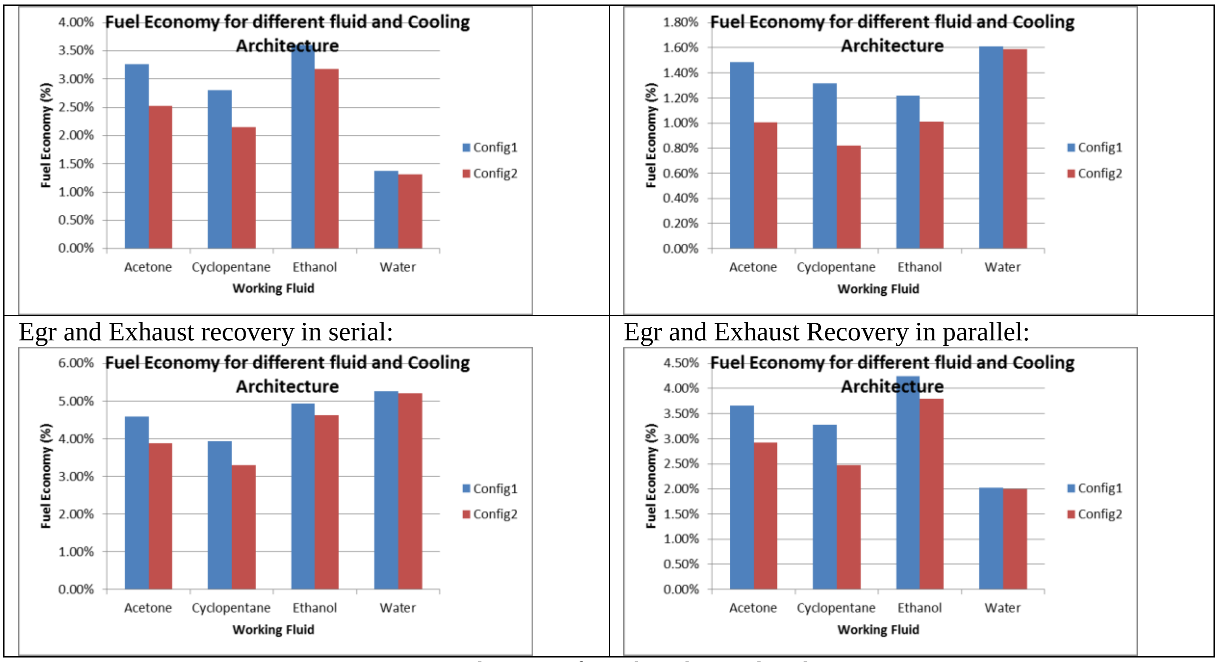


Figure 5 Fuel Economy for each Rankine cycle architecture

For Exhaust recovery contrary to what have been predicted in section "Fluid choice" Ethanol seems to be the best candidate for such a system. These differences are due to the high pumping loss when using Acetone or Cyclopentane resulting from higher mass flows. Water is not adapted and gives the lower fuel savings due to the high superheating needed to open the expander bypass valve.

For Egr recovery results are similar to the one presented in table 4 Water gives the best fuel economy followed by Acetone Ethanol and Cyclopentane.

In case of using both Egr and Exhaust in parallel Water is the best fluid since Egr stream allows high superheating at relatively high pressure.

For a serial recovering of both sources (Egr and Exhaust) results are slightly higher than in case of only Exhaust recovering. This can be explained by the non-total usage of exhaust heat. An improvement of this configuration will be to divide the Egr flow into two streams: one used to pre-heat and the second to superheat.

All fluids present a decrease in performance in cooling configuration 2. This lower fuel economy coming from the increase in condensing pressure due to higher coolant temperatures. However Water shows similar results in both configuration since the normal boiling point is 100°C. From this consideration we can see that Acetone and Cyclopentane are more impacted.

To sum up the main differences between results presented here are due to:

- The pumping loss. The higher the molecular mass is the higher the pumping work required is. Even if the pressure has been optimized for the power recuperation the pump power will be always higher with Acetone and Cyclopentane (this effect will be even more present when choosing refrigerant as working fluid).
- The high variation of condensing pressure for these two fluids in cooling configuration 1. As they have a boiling point lower than ethanol and water they will be more affected by the change in coolant temperature on the 9 operating points (see Cooling system simulations). This will affect a lot the turbine performance and especially the efficiency as showed in equation 13.
- Recover both gas streams in parallel give the best fuel economy.

DISCUSSION

The fuel savings presented in this paper are somewhat lower to the majority of values previously published ([3] [1] [6]). It needs to be considered why this trend is here observed.

First in most of the study only the benefits from the Rankine cycle is observed and the impact on the Cooling system is rarely taken into account.

The second aspect is the heat sources Egr has the advantage to present highest temperatures among the sources present on the truck. Therefore it would seem a good opportunity to move the engine calibration to increase the Egr rates and the total efficiency of the vehicle. This is true as long the tradeoff fuel consumption versus tailpipe emissions is not worsened.

The third important aspect is the heat sink. The fuel economy can be easily improved by a smarter thermal management. For example in winter conditions the thermostat is rarely fully open an electric actuation of this

one combined with a good thermal management and WHRS control will lead to higher fuel economy since the condensing pressure will decrease. Special attention has to be paid to the expansion device. A good matching between the expander design and cycle conditions is really important to maximize the fuel economy brings by a Rankine cycle installation.
The working fluid plays a very important role in the system performance. Ethanol and Water seems to be the best candidates but they have both drawbacks: poor thermal stability for Ethanol and freezing point of 0°C for the second. This could be solved by mixing these two fluids but the mixture behavior is not well known.
Last but not least is the transient behavior of the different components since here only steady state operating points have been simulated. The fuel economy seems to be therefore over predicted and that even if the models seem quite conservative in terms of components performance. On a real driving cycle it is not sure that the heat would be totally used especially on high loaded operating points which usually are energy consuming but not last a long time. This will result in a smooth behavior due to the thermal inertia of the boilers and will not allow to recover the high amount of energy present on this operating point.

CONCLUSION

WHRS introduction will be driven by future emissions legislations and especially CO_2 (which is directly proportional to the fuel consumption).
This study shows that upon the components improvements a system based approach is required when it comes to commercial vehicle integration. A new control based on the overall efficiency seems to be the key to a successful installation of a Rankine system into the truck.
Future work will focused more on transient aspect, model validation through components and overall system testing and interaction with the engine (change in egr temperature, exhaust backpressure, …).

REFERENCES

[1] R Freymann, W Strobl, A Obieglo. "The Turbosteamer: A System Introducing the Principle of Cogeneration in Automotive Applications". MTZ 05I2008 Volume 69 p20-27
[2] Wang, T., Zhang, Y., Peng, Z. and Shu, G., "A review of researches on thermal exhaust heat recovery with Rankine cycle", Renewable and Sustainable Energy Reviews 15 (2011) 2862-2871
[3] S Edwards, J Eitel, E Pantow, P Geskes, R Lutz. "Waste Heat Recovery: The Next Challenge for Commercial Vehicle Thermomanagement". SAE Technical Paper 2012-01-1205 doi:10.4271/2012-01-1205
[4]N Espinosa. "Contribution to the study of waste heat recovery systems on commercial truck diesel engines". PhD Thesys University of Liege, 2011
[5] Howell, T., Gibble, J.and Tun, C., "Development of an ORC system to improve HD truck fuel efficiency", presentedat Deer Conference 2011, USA, October 5th, 2011.
[6] E Doyle, L Dinanno, S Krammer. "Installation of a Rankine Compound Engine in a Class8 Truck for a single Vehicle Test". SAE Technical Paper 790646
[7] Teng, H., Klaver, J., Park, T., Hunter, G. et al., "A Rankine Cycle System for Recovering Waste Heat from HD Diesel Engines - WHR System Development," SAE Technical Paper 2011-01-0311, 2011, doi:10.4271/2011-01-0311.
[8] I Vaja. "Definition of an object oriented library for the dynamic simulation of advanced energy systems: Methodologies, Tools and Applications to Combined ICE6ORC Power Plants". PhD Thesys University of Parma 2009
[9] Gräber M, Strupp NC, Tegethoff W. "Moving boundary heat exchanger model and validation procedure". Proceedings of the 7th EUROSIM congress on modelling and simulation; September 5–10, 2010. Prague, Czech Republic.
[10] Badr, O., Probert, S. D. and O'Callaghan, P.W., "Selecting A Working Fluid for a Rankine-Cycle Engine", Applied Energy 21 (1985) 1-42
[11] Lemmon, E.W., Huber, M.L. and McLinden, M.O., REFPROP Nist Standard Reference Database 23 (Version 9.0), Thermophysical Properties Division, National Institute of Standards and Technology, Boulder, CO, 2011
[12] G Latz, S Andersson, K Munch. "Selecting an Expansion Machine for Vehicle Waste-Heat Recovery Systems Based on the Rankine Cycle". SAE Technical Paper 2013-01-0552 doi:10.4271/2013-01-0552
[13] Freymann, R., Ringler, J., Seifert, M.and Horst, T., "The second generation turbosteamer", MTZ 02/2012, Volume 73, p. 18-23.
[14] Baljé, O. E., "A Study on Design Criteria and Matching of Turbomachines: Part A - Similarity Relations and Design Criteria of Turbines", Transactions of the ASME, Journal of Engineering Power 84 (1962) 83-102.

[15] Kenneth, E., Nichols, P.E., "How to Select Turbomachinery For Your Application", http://www.barbernichols.com/resources, Oct. 2012.
[16] H Kunte, J Seume. "Partial Admission Impulse Turbine for Automotive ORC Application". SAE Technical Paper 2013-24-0092 doi:10.4271/2013-24-0092
[17]D Seher, T lengenfelder, J Gerhardt. "Waste Heat Recovery for Commercial Vehicles with a Rankine Process". 21st Aachen Colloquium Automobile and Engine Technology 2012
[18] S. Mavridou, G.C. Mavropoulos, D. Bouris, D.T. Hountalas, G. Bergeles. "Comparative Design Study of a Diesel Exhaust Gas Heat Exchanger for Truck Applications with Conventional and State of the Art Heat Transfer Enhancements". Applied Thermal Engineering (2010), doi: 10.1016/j.applthermaleng.2010.01.003

LEXICAL

Abbreviations

Cd: Discharge Coefficient
CFC: Chloro Fluoro Carbon
Cp: Specific Heat
Dh: Hydraulic Diameter
EGR: Exhaust gas recirculation
GADSL: Global Automotive Declarable Substance List
GWP: Global Warming Potential
H: Entahlpy
HCFC: Hydro Chloro Fluoro Carbon
HD: Heavy Duty
K: Impulse Turbine Throat
$\dot{m}$: Mass Flow
m: Mass
N: Rotational Speed
Nu: Nusselt number
NTU: Number of Transfer Unit
ODP: Ozone Depletion Potential
P: Pressure
PR: Pressure Ratio
$\dot{Q}$: Heat Flux
R: Wheel Radius
S: Area
T: Temperature
V: Volume
WHR: Waste Heat Recovery
WHRS: Waste Heat Recovery System

Greek Letters

γ: Specific heat ratio
η: Efficiency
λ: Heat Conductivity
ρ: Density
ω: Angular speed

Subscript

conv: Convective
eff: Effective
eq: Equivalent
in: Inlet
is: Isentropic
out: Outlet
res: Reservoir