Heavy duty heat recovery with a Rankine cycle using a piston expander

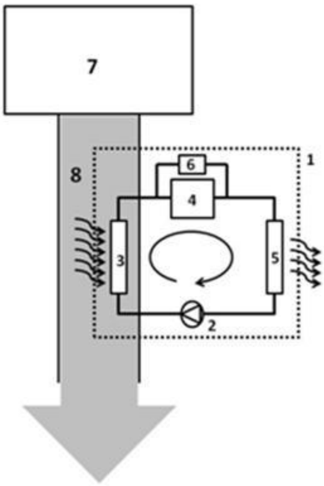
Darmedru Antoine, Mélis Julien, Daccord Rémi

EXOES, France, info@exoes.com

1 Introduction

Nowadays, it seems clear that the truck industry will implement bottoming Rankine cycles (RC) on long haul trucks before 2020 as an answer to the future stringent regulations and the still increasing customer requests for operating cost reduction [1]. The next step of this challenging project is the choice of the best technical solution allowing 4 to 5% fuel economy, which seems to be approved by the industry as a viability threshold for the waste heat recovery system (WHRS) regarding the total cost of ownership [2][3].

The key components of a RC are shown hereafter (**fig.1**).



1: Rankine cycle (RC)
2: Pump
3: Evaporator
4: Expander
5: Condenser
6: Bypass
7: ICE
8: Exhaust gas

Figure 1 : Bottoming Rankine cycle

Key choices when designing such a complex system are the selection of the hot source, the working fluid, the expander and the cold sink:

- Hot source:

As a consensus, the heat rejected into the cooling water and dissipated through the radiator (22% including EGR [4]) is not exploitable because of its too low temperature. Exhaust waste heat (22% [4]) is thus preferred as the favorite heat source for the bottoming cycle. The evaporator will preferably take place downstream the after-treatment system not to penalize pollutants emissions (**fig.2**). EGR (**fig.3**) must also be considered: its smaller flow but higher temperature makes it easier to exploit. Regarding Euro6 compliance on pollutants emission, a majority of truck makers consider implementing EGR on their long haul trucks. Since EGR must already be cooled down, no additional heat is rejected under hood which simplifies the integration of the cooling circuit. Both exhaust and EGR sources can be combined to maximize heat recovery. In this case, the system may become much more complex, with two evaporators in parallel (**fig.4**) or in series (**fig.5**). It can then boost the fuel economy of a WHRS, but increases weight, integration issues and control complexity.

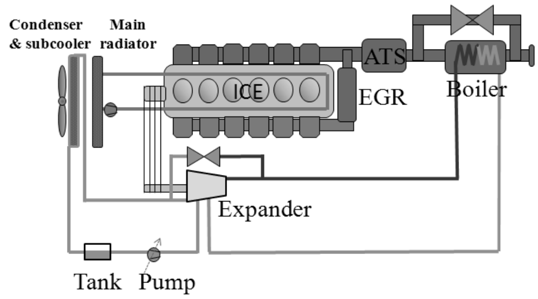


Figure 2: bottoming cycle on exhaust line

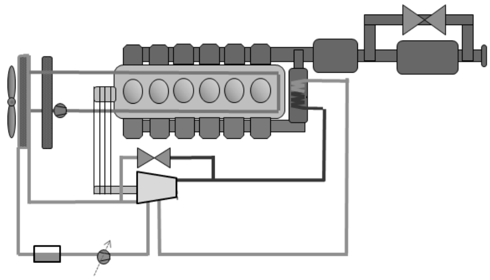


Figure 3: bottoming cycle on EGR

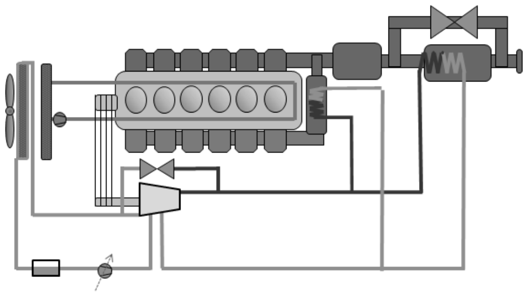


Figure 4: bottoming cycle on exhaust and EGR in parallel

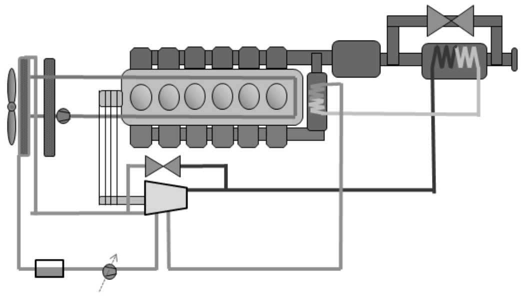


Figure 5: bottoming cycle on exhaust and EGR in series

- Working fluid:

Usual selection criteria are a low environmental impact, being harmless, a low flammability, a high thermodynamic performance, low operating pressures, a good stability at high temperature, a condensation temperature at atmospheric pressure and a low freezing point [5]. Up to now, no fluid meets all the requirements. Water and ethanol, or a mixture of both, are often selected for their better performance than organic fluids like R245fa [6][7]. Water will be selected for high temperature applications: exhaust of gasoline or CNG engines and EGR cooling on Diesel engines. Ethanol is more appropriated for lower hot source temperatures like exhaust on Diesel engines. At last, one must find a solution to manage the freezing problem of water and the ageing of ethanol. Water, ethanol and R245fa will be studied for a RC used in a Diesel engine.

- Expander and transmission:

Piston and Scroll expanders are often selected as primary candidates thanks to their efficiency and their ability to handle droplets. Scroll expanders are nearly off-the-shelf products and have low expansion ratios only compatible with R245fa, contrarily to piston expanders. Thus scroll expanders may be considered for first step demonstrators, with a quick access to the market with poor efficiencies, when piston expanders could be part of the second generation bottoming cycles, with high efficiency but requiring a longer development effort. Impulse turbines are also of main interest but their droplets intolerance leads to a higher control complexity and a reduced recovered energy on a dynamic cycle (the bypass of the turbine may be often actuated because of the risk of droplets) [8]. The following study focuses only on piston expanders.

A mechanical transmission must be considered for classic HCV because of the too low electricity needs of such vehicles. For hybrid trucks, an electrical production through the use of a generator will be the most appropriate.

- Cooling and packaging:

While using RC, the majority of the tailpipe exhaust heat will be rejected in the cooling fluid under hood. A trade-off due to weight, packaging, fan on time and aero complexities must be found. Given the fact that aerodynamic drag represents 18% of the losses of a truck and that a 175kg additional weight reduces fuel economy of one percent [4], those challenges are also of main interest to keep the promised 5% fuel consumption reduction.

The working fluid's choice will also determine the cold source configuration. While water will allow using the existing cooling loop of the HCV's engine (**fig.7**), the use of ethanol or R245fa will impose an additional radiator to condense the working fluid (60°C for R245fa and 80°C for ethanol for example) (**fig.6 and 8**).

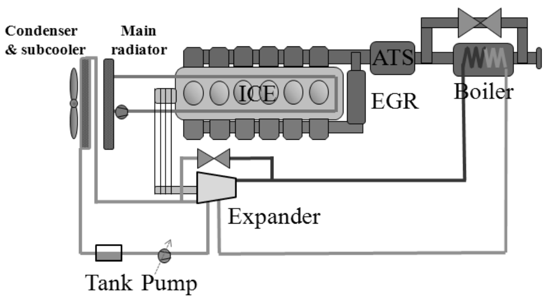


Figure 6: Dedicated cooling loop

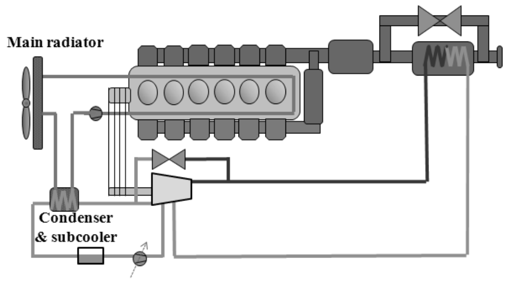


Figure 7: Existing cooling loop

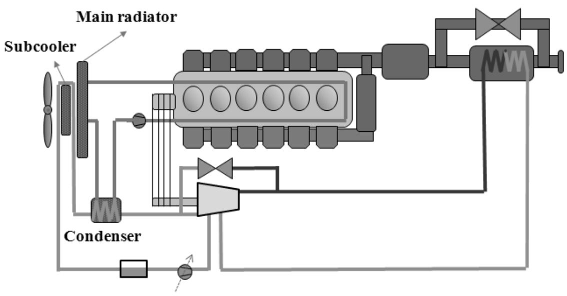


Figure 8: Mix of both solutions with additional subcooler

2 Simulation

After a period of tests and model calibration on single cylinder, Exoès has developed a static expander model to strengthen and speed up the design process and development phases of its prototypes. The model used for these calculations is a simple 0D model that takes into account the pinch phenomenon in the evaporator, the internal leaks and the mechanical efficiency of the expander.

2.1 Model assumptions

2.1.1 Evaporator(s) modeling

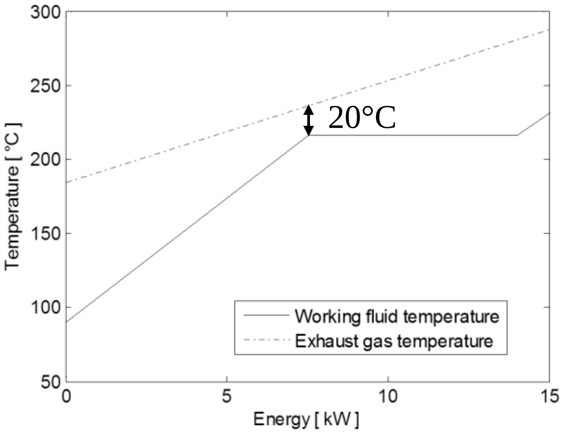


Figure 9: heat exchange for exhaust boiler

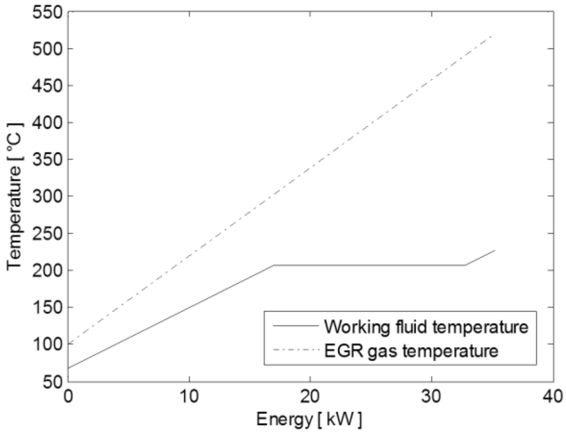


Figure 10: heat exchange for EGR boiler

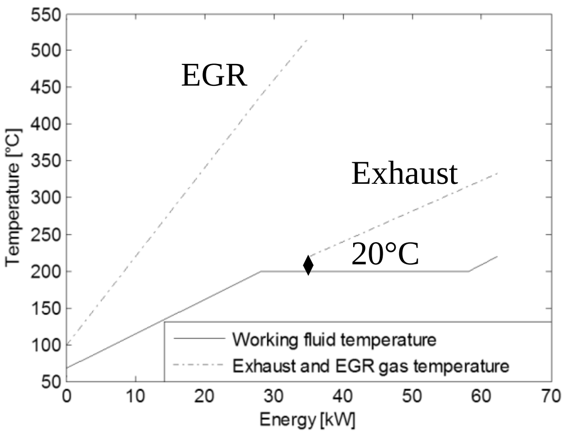


Figure 11: heat exchange for EGR and exhaust boilers in series

Exhaust gases are considered at 1 bar. At the outlet of the evaporators, we limit the exhaust gases minimal temperature to 100°C to prevent the formation of acidic chemicals. The EGR evaporator is controlled in order to cool the exhaust gases down to 100°C. We also assume a minimal pinch of 20°C in the evaporators. Since piston expanders may accept droplets at their exhaust ports, the overheat of the vapor is fixed at only 50°C for water, 20°C for ethanol and 5°C for R245fa. The subcooling of the working fluid is 10°C. No pressure drops are considered in the condenser and evaporator. Figures 9, 10 and 11 illustrate the calculations based on these assumptions with ethanol for different hot source configurations (exhaust, EGR and both in series).

We observe on figures 9 or 11 that the energy of the exhaust gases can only be partly recovered because of the evaporation plateau occurring at too a high temperature. A lower working fluid pressure would enable a higher energy recovery but would deteriorate the RC efficiency and lead to a bigger expander. A trade-off must then be found.

2.1.2 0D expander cycle

2.1.2.1 Theoretical cycle

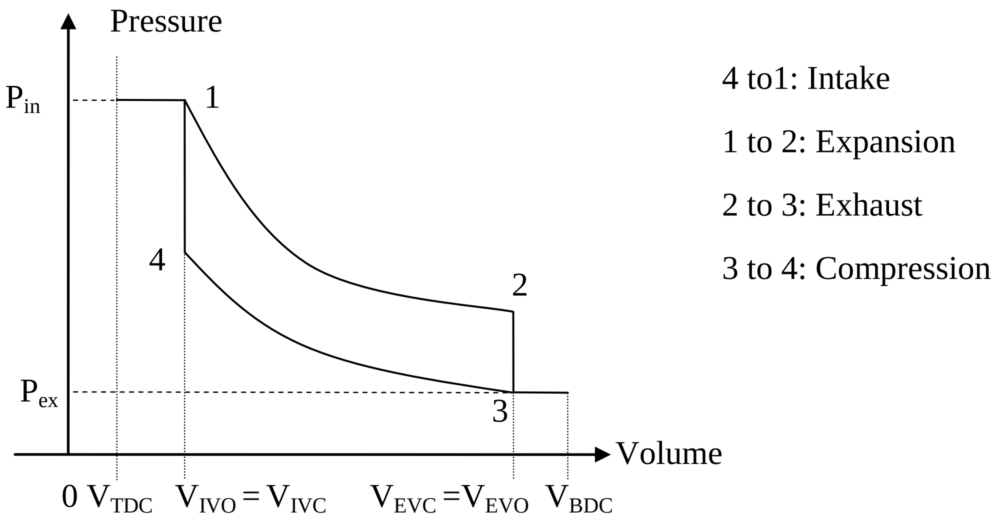


Figure 12: Theoretical cycle for 0D calculation

Exoès expander technology is based on a very simple and robust inlet and exhaust system which imposes “symmetrical” valve timing both at top and bottom dead centers. The expander is then characterized by only two parameters: its swept volume and its expansion ratio, R_E (***Equ.(1)***).

$$R_E = \frac{V_{EVO}}{V_{IVC}} = R_C = \frac{V_{EVC}}{V_{IVO}} \tag{1}$$

The modeling strategy consists in developing a 0D model (***fig.12***) set by geometric data and corrected by coefficients calculated from both test bench measurements and more complex 1D models: the isentropic effectiveness η_{is} and the filling factor ∅ [9] (***Equ.(2) and (3)***). The isentropic effectiveness characterizes the expander’s ability to use the potential energy of the steam while the filling factor stands for the ability to fill the cylinder with the right mass of vapor. According to these definitions, a perfect expander would have a filling factor equal to 1 and its isentropic effectiveness can be calculated with its swept volume, its expansion ratio and the inlet and exhaust conditions. Note that we include the mechanical efficiency in the isentropic effectiveness.

$$\eta_{is} = \frac{\dot{W}_{mecha}.\eta_{mecha}}{(h_{wf,in} - h_{wf,ex,is}).\dot{m}_{wf,in,meas}} \tag{2}$$

$$\phi = \frac{\dot{m}_{wf,in,meas}}{(V_{IVC}.\rho_{in} - V_{EVC}.\rho_{ex,is}).N_{\exp}.n} \tag{3}$$

Several phenomena will change these indicators: the internal leaks, the pressure drops, the heat exchanges and the friction losses. To alleviate the following simulations, only the mechanical efficiency and the internal leakage have been modeled.

2.1.2.2 Mechanical Efficiency

The mechanical efficiency is modeled by a polynomial law extracted from literature [10] on an expander presenting a similar architecture.

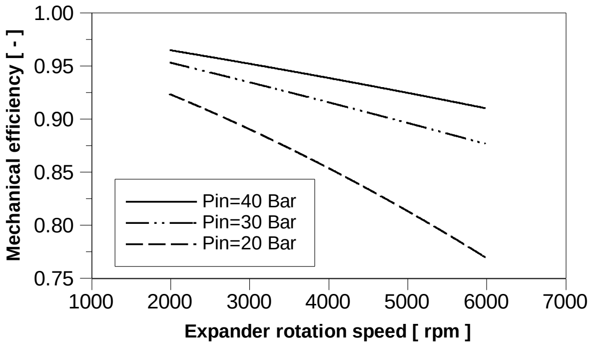


Figure 13: Calculated mechanical efficiency

2.1.2.3 Internal leakage

The expander leaks model is a subsonic or sonic flow through a simple nozzle directly between inlet and exhaust, with the hypothesis that superheated steam behaves comparable to an ideal gas. The equivalent diameter of the nozzle is set to 0.55mm [10].

$$\dot{m}_{leak} = A_{leak}.\rho_{thr,leak}.\sqrt{2.(h_{in} - h_{thr,leak})} \tag{4}$$

$$P_{thr} = \max(P_{crit}, P_{ex}) \tag{5}$$

$$P_{crit} = P_{in}.(\frac{2}{\gamma+1})^{\frac{\gamma}{\gamma-1}} \tag{6}$$

2.1.3 Pump modeling

The pump is modeled with a 80% mechanical efficiency and a 70% isentropic effectiveness, which is conservative according to our daily experience.

$$\dot{W}_{pp} = \left(\frac{P_{wf,out} - P_{wf,in}}{\eta_{mecha}.\eta_{is}.\rho_{wf,in}}\right).\dot{m}_{wf} \tag{7}$$

2.1.4 Definitions of the indicators

The simulation results use different indicators allowing us to compare the efficiency of the whole cycle or component by component:

- The energy transformed from the vapor into mechanical power by the expander.

$$R_{exp/vap} = \frac{\dot{W}_i.\eta_{mecha} - \dot{W}_{pp}}{\sum_i \dot{m}_{wf_i}.\Delta h_{wf_i,evap_i}} \tag{8}$$

- The energy extracted from the exhaust gases and transferred to the vapor. Because of the possible use of two evaporators, we compare each calculation result with this indicator.

$$R_{vap/waste} = \frac{\sum_{i=1}^{2} \dot{m}_{wf_i}.\Delta h_{wf_i,evap_i}}{\sum_{j=1}^{2} \dot{m}_{hf_j}.\Delta h_{hf_j,evap_j}} \quad \text{with: } \Delta h_{hf_j,evap_j} = c_{p,hf}\left(T_{hf,in,evap} - 100°C\right) \tag{9}$$

- The energy in the exhaust gases to ICE power ratio.

$$R_{\text{waste/ICE}} = \frac{\sum_i \dot{m}_{hf_i}.\Delta h_{hf_i,evap_i}}{\dot{W}_{ICE,mecha}} \tag{10}$$

- The expander power to ICE power ratio.

$$R_{\text{exp/ICE}} = \frac{\dot{W}_i.\eta_{mecha} - \dot{W}_{pp}}{\dot{W}_{ICE,mecha}} \tag{11}$$

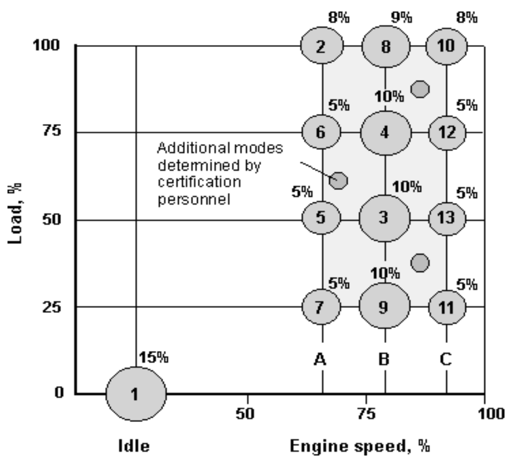
- And an average of this ratio over the four loads conditions.

$$R_{\exp/ICE}\ \text{avr} = \sum_{1}^{4}\left(\frac{\dot{W}_i.\eta_{mecha} - \dot{W}_{pp}}{\dot{W}_{ICE,\ mecha}}.O_{cc}\right) \tag{12}$$

This last average expander power to ICE power ratio is the closest to the fuel cuts (neglecting positive and negative side effects like quicker ICE warm up, possible ICE downsizing and transmission efficiency, extra-weight on the vehicle,…).

2.2 Data

The ICE input data used to assess the performance comes from 4 operating points extracted from the European Stationary Cycle (ESC) (fig.14) at B speed and at four different loads.



$A = n_{lo} + 0.25(n_{hi} - n_{lo})$

$B = n_{lo} + 0.50(n_{hi} - n_{lo})$

$C = n_{lo} + 0.75(n_{hi} - n_{lo})$

n_{hi}: highest engine speed where the power equals 70% of the maximum net power.

n_{lo}: lowest engine speed where the power equals 50% of the maximum net power.

Figure 14: ESC for HCVs

The engine data used for all calculations are extrapolated to future EURO6 engines. Note that the occurrences of the working points are different from the cycle, because more representative of real life conditions.

Table 1: input ICE data

Pts E.U. norm	**T_{ice} [%]**	**N_{ice} [rpm]**	**Occurrence [%]**
9: B25	25	1200	20
3: B50	50	1200	30
4: B75	75	1200	40
8: B100	100	1200	10

2.3 Results

An example of a performance calculation is explained in this section. The input data are: pure ethanol, EGR-only, swept volume 800cc, expansion ratio 20, exhaust pressure 1bar and expander speed 3600rpm (drive ratio of 3). This case has been chosen for its simplicity and popularity: ethanol is well adapted for the EGR temperature range and there is neither air infiltration nor additional cooling load.

Table 2: example of performance calculation

Points	**P_{wf} [Bar]**	**$\dot{W}_{i,exp}$ [kW]**	**$\dot{m}_{wf}$ [kg/h]**	**$R_{exp/vap}$ [%]**	**$R_{vap/waste}$ [%]**	**$R_{waste/ICE}$ [%]**	**$R_{exp/ICE}$ [%]**
B25	28,4	2,8	63,3	13,4	40,4	52,0	2,8
B50	31,1	4,7	99,5	14,3	37,7	51.2	2,8
B75	32,7	5,8	120,3	14,5	39	48	2,7
B100	35,8	8,0	163,4	14,9	39,2	43,3	2,5
$R_{exp/ICE}$ average = 2,7%							

If one decides to design the evaporator for full load (B100), which implies a huge area, the performance calculated for the 3 partial load points will be achieved. To optimize the cost, weight and packaging of the system, a smaller evaporator can be chosen, optimal for B75 point for example. In this case, performance on B100 will be lower than those calculated.

It is also considered for calculation that all the heat can be rejected under hood. In real life there will surely be an upper limit to heat rejection ability, forcing to bypass the evaporator at high load in severe conditions (hot ambient air temperature, altitude, etc.).

There is a low dispersion of the results depending on the engine load because the expander chosen for this simulation is generously sized.

2.4 Sensitivity analysis

Many different configurations of the RC can be imagined when designing such a system. Each choice will have an impact on fuel consumption reduction (efficiency and weight), price and packaging complexity. Hereafter is illustrated the sensitivity of the system regarding performance on B75 point only. When not specified, the simulation configuration retained is the same as explained in §2.3.

The following figures show what performance Exoès technology may achieve on different evaporators configurations. We limit the expansion ratio to 20. We observe that the performances plateau around V_{cc}=800 and R_E=20 with EGR-only for 3,600 rpm (***fig.16 and 17***). A different set of results would be found with another hot source configuration. For instance the exhaust-only configuration would require an even bigger expander to lower the vapor pressure and recover more energy. We limit the swept volume to 800cc leading to the biggest expander acceptable in terms of size, weight and cost according to our projection for a mass market product. This configuration was run on the different evaporator configurations to finally demonstrate that a WHRS may reach up to 6.7% fuel cuts.

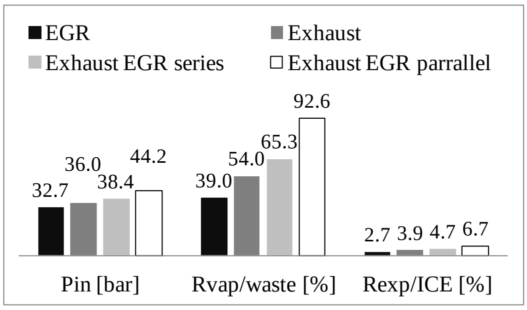


Figure 15: hot source sensitivity

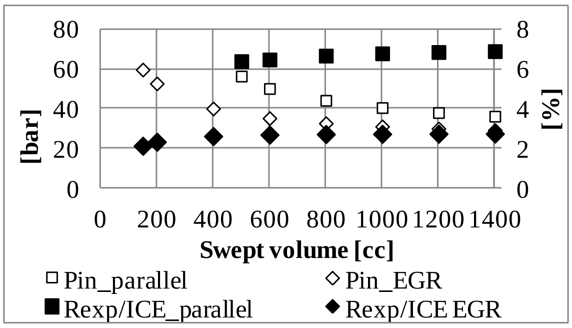


Figure 16: swept volume sensitivity

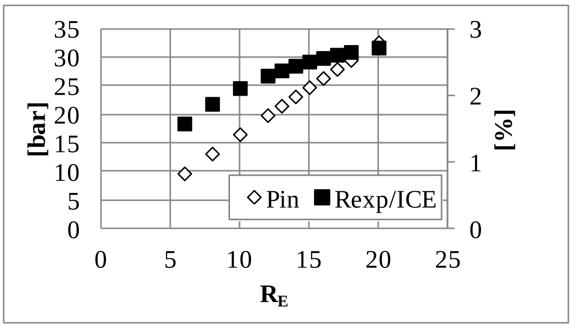


Figure 17: Expansion ratio sensitivity

In figure 18 and 19, we can remark the influence of the condensing temperature and the exhaust pressure on the efficiency for different fluids. Note that the expansion ratio for R245fa is set to 6 instead of 20 for the others and that we consider two evaporators in series. The trend is simple: the lower the condensing temperature the higher the efficiency. Ethanol and water present the same efficiency but we can observe that water suffers a bit less from high condensing temperatures than ethanol. On the other hand, ethanol reaches very high vapor pressures: the expander would have to be bigger or run at higher speed to maintain an acceptable level of pressures. At last, air infiltration has to be managed when the exhaust pressure is below the atmospheric pressure which was satisfactorily done on our test bench. We can conclude that ethanol is more adapted for 80°C and water for 100°C.

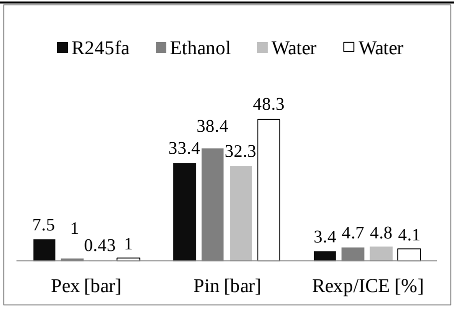


Figure 18: comparison at 80°C condensing temperature

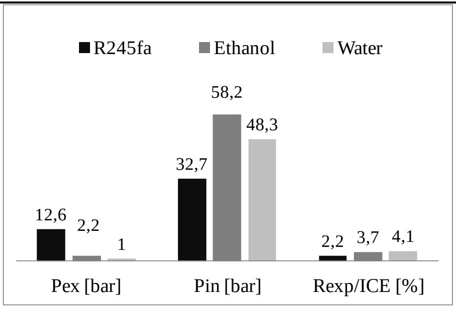


Figure 19: comparison at 100°C condensing temperature

2.5 Best in class configurations

Regarding pure efficiency without any other considerations, we would suggest a parallel assembly of evaporators, the use of ethanol as working fluid and a 80°C condensing temperature. The expander parameters are an expansion ratio of 20 and a swept volume of 800cc and a drive ratio of 3. The results are presented in table 3.

Table 3: Highest performance configuration

Points	**P_{wf} [Bar]**	**$\dot{W}_{i,exp}$ [kW]**	**$\dot{m}_{wf}$ [kg/h]**	**$R_{exp/vap}$ [%]**	**$R_{vap/waste}$ [%]**	**$R_{waste/ICE}$ [%]**	**$R_{exp/ICE}$ [%]**
B25	32,9	6,0	123,8	14,6	79,4	52,0	6,0
B50	41,4	12,1	243,3	15,0	92,9	51,2	7,1
B75	44,2	14,2	283,8	15,0	92,6	48,0	6,7
B100	49,7	18,4	367,7	14,8	88,8	43,3	5,7
$R_{exp/ICE}$ average = 6,6%							

To achieve the best performance-cost trade-off, we would suggest a serial assembly of evaporators. The exhaust evaporator is then smaller. It has less constraint in terms of pressure drops and integration. And above all, there is less heat to dissipate under hood. Control of the system will also be easier with suppression of a distributing valve used in the parallel configuration. The expander has a nominal speed of 3,600 rpm, a swept volume of 800cc and an expansion ratio of 15 that is less demanding than 20 in terms of design. The fluid remains ethanol, with a condensing temperature of 80°C (table 4).

Table 4: Best compromise configuration

Points	**P_{wf} [Bar]**	**$\dot{W}_{i,exp}$ [kW]**	**$\dot{m}_{wf}$ [kg/h]**	**$R_{exp/vap}$ [%]**	**$R_{vap/waste}$ [%]**	**$R_{waste/ICE}$ [%]**	**$R_{exp/ICE}$ [%]**
B25	23,8	4,6	104,1	13,2	66,1	52,0	4,5
B50	28,2	8,4	182,9	13,8	69,2	51,2	4,9
B75	29,9	9,8	213,0	13,9	68,8	48,0	4,6
B100	33,3	12,8	275,3	14,1	66,0	43,3	4,0
$R_{exp/ICE}$ average = 4,6%							

3 Tests

An expander and a test bench have been manufactured and tests are in progress both with water and ethanol. The datasheet of the expander prototype (fig.20) is presented in table 5.

Table 5: expander datasheet

Data	Value / Description
Engine architecture	Swashplate – 5 cylinders
Capacity	185cc
Nominal speed	3,600rpm
Expansion ratio	Adjustable from 6 to 12
Weight	18kg
Size	L280mmxD160mm
Working fluid	Water or Ethanol
Maximal inlet pressure	60 bar
Maximal exhaust pressure	2.5 bar
Maximal inlet temperature	300°C
Minimal oil mass fraction at inlet	0% - it can run oil-free
Peak shaft power	8kW

The test bench is represented in fig.21. The maximal steam power reachable on this bench is around 25kW. The maximal pressure is 60bar and the maximum temperature is 300°C.

Figure 20: piston expander

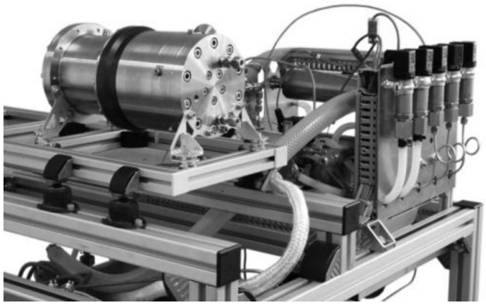


Figure 21: test bench

4 Conclusion

Exoes has developed a static performance model allowing us to weight the influence of several parameters on performance and design of WHRS for HCVs. As output of this model we can propose today what we think to be the best compromise, presented in section 2.5. The projected fuel cuts with this configuration is 4,6% without considering any side effects, and we have shown that 6,6% fuel cuts may be reachable with a more complex and bigger system provided it can be cooled. Mixture of water and alcohol are now under integration in our model as we look at it as a good trade-off to solve freezing, ageing and cooling needs issues.

We have also developed a piston expander prototype with oil free hot parts and a test bench to run it with water and ethanol. Tests are under progress but we have already experienced the major influence of the internal leakage on the isentropic effectiveness of the expander as no oil film helps to “vaportight” the valves and the piston rings. The results will be available very soon and will be compared to the model results.

5 Literature

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6 Nomenclature

General

ATS: (exhaust) After Treatment System
EGR: exhaust gas recirculation
HCV: heavy commercial vehicle
ICE: internal combustion engine
WHRS: waste heat recovery system
RC: Rankine cycle

Calculation

A: area [m²]
BDC: bottom dead center
cc: cubic centimeter
cyl: cylinder
EVC: exhaust valve closing
EVO: exhaust valve opening
h: specific enthalpy [J.kg-1]
IVC: intake valve closing
IVO: intake valve opening
$\dot{m}$: flow [kg/s]
n: cylinder number
N: rotation speed [s-1]
Occ: occurrence
P: pressure [bar]
$\dot{Q}$: heat flow [kW]
R: ratio [-]
rpm: round per minute
T: temperature [K]
T_{ICE}: ICE torque
TDC: top dead center
V: volume [m3]
$\dot{W}$: power [kW]

Greek

η: efficiency [-]
ρ: density [kg.m-3]
Ø: filling factor [-]

Subscripts and Superscripts

avr: average
C: compression
cond: condensation
crit: critical (sound of speed)
E: expansion
evap: evaporator
ex: exhaust
exp: expander
hf: hot fluid (exhaust gas)
i: indicated
in: intake
is: isentropic
meas: measured
mecha: mechanical
pp: pump
thr: throat
vap: vapor
w: wall
wf: working fluid