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Vincent Grelet, Thomas Reiche, Vincent Lemort, Madiha Nadri, Pascal Dufour. Transient performance evaluation of waste heat recovery rankine cycle based system for heavyduty trucks. Applied Energy, 2016, 165 (1), pp.878-892. 10.1016/j.apenergy.2015.11.004 . hal-01265044

HAL Id: hal-01265044

<https://hal.science/hal-01265044>

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**V. Grelet , T. Reiche, V. Lemort, M. Nadri, P. Dufour,
"Transient performance evaluation of waste heat recovery rankine cycle based
system for heavyduty trucks",
Applied Energy, 2016, 165(1), pp. 878–892.
[DOI : 10.1016/j.apenergy.2015.11.004](https://doi.org/10.1016/j.apenergy.2015.11.004)**

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TRANSIENT PERFORMANCE EVALUATION OF WASTE HEAT RECOVERY RANKINE CYCLE BASED SYSTEM FOR HEAVY DUTY TRUCKS

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Abstract

The study presented in this paper aims to evaluate the transient performance of a waste heat recovery Rankine cycle based system for a heavy duty truck and compare it to steady state evaluation. Assuming some conditions to hold, simple thermodynamic simulations are carried out for the comparison of several fluids. Then a detailed first principle based model is also presented. Last part is focused on the Rankine cycle arrangement choice by means of model based evaluation of fuel economy for each concept where the fuels savings are computed using two methodologies. Fluid choice and concept optimization are conducted taking into account integration constraints (heat rejection, packaging ...). This paper shows the importance of the modeling phase when designing Rankine cycle based heat recovery systems and yields a better understanding when it comes to a vehicle integration of a Rankine cycle in a truck.

Keywords: Waste heat recovery system, Modeling, Thermodynamic,

*Corresponding author

1 **1. INTRODUCTION**

2 Even in nowadays heavy duty (HD) engines, which can reach 45% of effi-
3 ciency, a high amount of the chemical energy contained in the fuel is released
4 as heat to the ambient. Driven by future emissions legislation and increase
5 in fuel prices, engine gas heat recovering has recently attracted a lot of inter-
6 est. Over the last decades, most of the research has focused on waste heat
7 recovery systems (WHRS) based on the Rankine cycle [1, 2, 3]. These sys-
8 tems can lead to a decrease in fuel consumption and lower engine emissions
9 [4, 5]. Recent studies have brought a significant potential for such systems
10 in a HD vehicle [6, 7]. However, before the Rankine cycle based system can
11 be applied to commercial vehicles, the challenges of its integration have to
12 be faced. The work done in [8] and [9] show that one of the main limita-
13 tion is the cooling capacity of the vehicle. But other drawbacks, such as the
14 back pressure, weight penalty or transient operation should not be minimized
15 [10, 11]. Before tackling the problem of the control strategy of this system
16 [12, 4], the architecture and components need to be selected to achieve a
17 certain objective that could be to maximize the fuel savings or minimize the
18 impact on the vehicle. This study focuses more on maximizing the system
19 performance by taking into account the different penalties induced by the
20 integration of the system on a heavy duty truck. Technical challenges and
21 optimization of stationary organic Rankine cycles (ORC) are well addressed
22 [13, 14] but for mobile applications only few studies deal with fuel saving po-
23 tential of WHRS on dynamic driving cycles [15, 16] and the latter is generally
24 reduced to a certain number of steady state engine operating points [17, 18].
25 This last approach leads to an overestimation of the WHRS performance [3]
26 and therefore of the fuel economy. In [19] different concepts are analyzed
27 taking into account the system integration into the vehicle cooling module.
28 The concepts differ in the number of heat sources used and the temperature
29 level of the cooling fluid. Each is simulated on different steady state engine
30 operating points and the fuel economy is calculated taking into account the
31 increase in cooling fan consumption, exhaust back pressure or intake manifold
32 temperature. Depending on the Rankine configuration and the location of
33 the condenser, improvements from 2.2% (recovering heat only from exhaust

34 gases and condenser placed in front of the cooling package) to 6.9% (exhaust
35 gas recirculation and exhaust heat are recovered and condenser is fed with
36 engine coolant) are achieved. In [15], dynamic fuel economy is evaluated
37 on a light duty vehicle taking into account the main penalties induced by
38 the integration of the WHRS. Fuel savings from 3.4% to 1.3% are presented
39 depending on the level of integration of the system into the vehicle architec-
40 ture. However, no optimization is proposed either on the system architecture
41 or on the condenser integration into the cooling package. This paper is or-
42 ganized as follows. The second section explains the different considerations
43 to take when designing a Rankine cycle for a HD application. In the third
44 section, the different models used in the rest of the study are explained. In
45 the fourth section, the scope of the study and the different methodologies are
46 explained. In the fifth section, simulation results are analyzed and possible
47 improvements are proposed. Finally, conclusions are drawn and directions
48 for future research work are discussed.

49 **2. DESIGN ASPECTS TO CONSIDER**

50 Figure 1 shows a simple waste heat recovery system mounted on a 6 cylinder
51 heavy duty engine. Working fluid flows through four basic components which
52 are: the pump, the evaporator linked to the heat source, the expansion ma-
53 chine and the condenser linked to the heat sink. For sake of clarity, the link
54 between the expander and the engine driveline is represented by a dashed
55 line since it can be either mechanical or electrical (by coupling a generator to
56 the expansion machine and reinject the electricity on the on board network).
57

58 *2.1. Working fluid choice*

59 There are several aspects to take into account when choosing a working
60 fluid for this application. Unlike stationary power plants where the main
61 consideration is the output power or the efficiency, here other aspects have to
62 be considered such as fluid deterioration, environmental aspects or freezing.
63 Up to now, several studies have tried to identify the ideal fluid for WHRS
64 [20, 21, 22] but no single fluid has been found. Recently, new performance
65 indicators have been introduced [14, 23], where cost and design issues enter
66 into consideration.

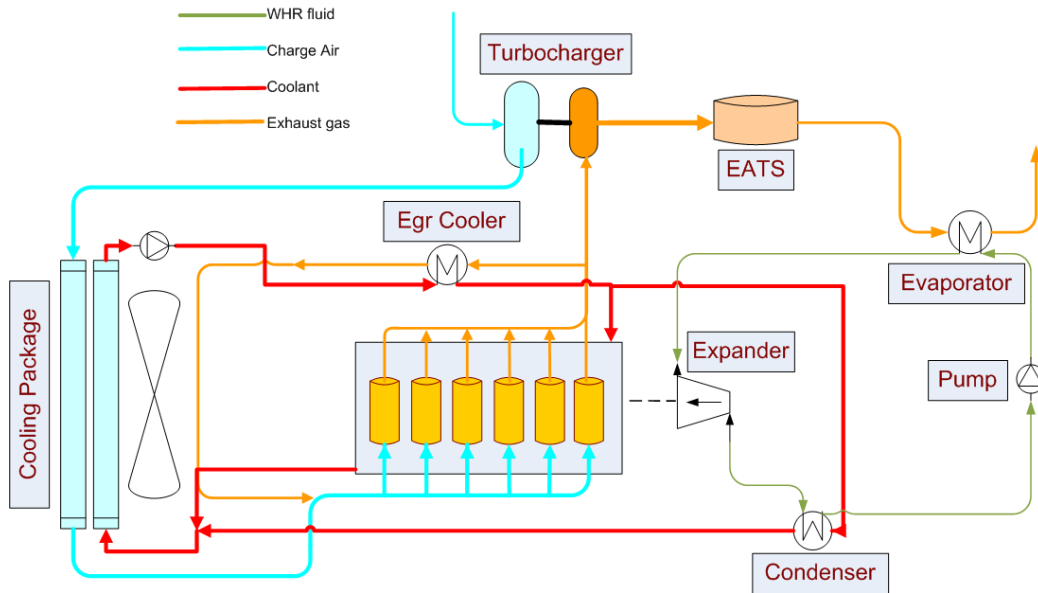


Figure 1: Simple waste heat recovery Rankine based system

67 *2.2. Heat sources*

68 On a commercial vehicle, a certain number of heat sources can be found
 69 such as exhaust gases, cooling water or engine oil. These ones have several
 70 grade of quality (temperature level) and quantity (amount of energy). If the
 71 number of heat sources often yields higher fuel savings, it also brings more
 72 complexity and more challenges for the design of the system (fluid, expansion
 73 machine, control).

74 *2.3. Heat sink*

75 On a HD Truck, the only heat sink available is the vehicle cooling package
 76 which is a module including radiators for the compressed air and the engine
 77 coolant and cooled down by means of the ram air effect and the cooling
 78 fan. Integration of a WHRS into the cooling module results on a higher load
 79 on the latter and limits the amount of waste heat that can be converted
 80 into useful work. As such, complete system analysis is necessary to find the
 81 optimal way of recovering heat from a vehicle.

82 2.4. Subsystem interaction

83 The engine operation is influenced by the introduction of a WHRS. For ex-
84 ample, as the WHRS shares the cooling system of the vehicle, the charge air
85 cooling capacity can be lower, which has a negative behavior on the engine
86 performance. Another example is the use of exhaust gas recirculation (EGR)
87 as heat source. This leads to a trade-off between EGR cooling and Rank-
88 ine cycle performance, which could impact negatively the engine emissions.
89 Several other interactions such as the exhaust back pressure or the weight
90 penalty could be cited.

91 The WHRS performance, and so the fuel economy induced by this later, is
92 then dependent on all these aspects. It is therefore critical to model the
93 complete system and its environment in order to optimize its architecture. It
94 helps to select the best design and reduce the number of experimental tests
95 to carry out.

96 3. RANKINE MODELING

97 3.1. Rankine process

98 The Temperature Entropy (T-s) diagram represented in figure 2 shows the
99 associated state changes of the working fluid through the Rankine cycle:

- 100 • The pressure of the liquid is increased by the pump work up to the
101 evaporating pressure ($1 \rightarrow 2$).
- 102 • The pressurized working fluid is pre-heated ($2 \rightarrow 3a$), vaporized ($3a \rightarrow$
103 $3b$) and superheated ($3b \rightarrow 3c$) in a heat exchanger, by recovering heat
104 (\dot{Q}_{in}) from the heat source.
- 105 • The superheated vapor expands from evaporating pressure to condens-
106 ing pressure ($3c \rightarrow 4$) in an expansion device creating mechanical power
107 (\dot{W}_{out}).
- 108 • The expanded vapor condenses ($4 \rightarrow 1$) through a condenser (linked to
109 the heat sink) releasing heat (\dot{Q}_{out}).

110 In this process the changes of states in both the pump and the expander are
111 irreversible and increase the fluid entropy to a certain extent. To correctly
112 assess the performance of a system based on the Rankine cycle, two different

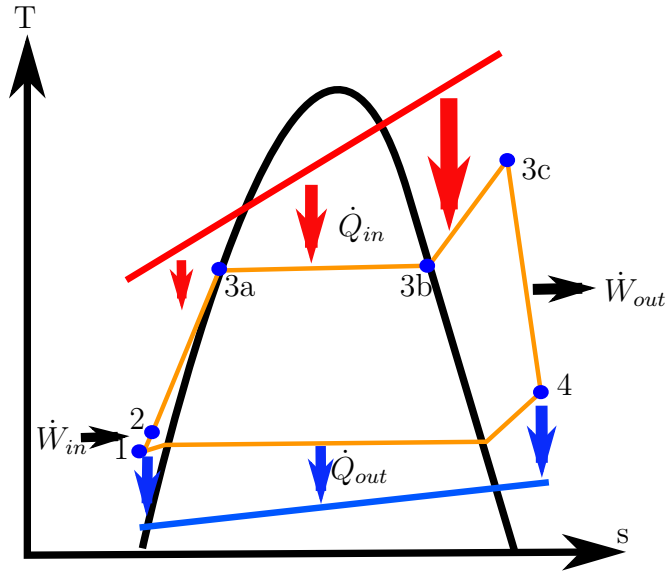


Figure 2: Temperature-Entropy diagram of the Rankine cycle

113 models have been developed: a simple 0D steady state model based on the
 114 enthalpy change that undergoes the working fluid which does not intend to
 115 represent components performance and where the dynamic is not taken into
 116 account and a second, based on conservation principles applied on one spatial
 117 dimension. This is required to represent real performance of the components
 118 constituting the system either in steady state or in transient.

119 3.2. 0D steady state modeling of a Rankine cycle

120 In order to simulate a high number of working fluids, a 0D model of a Rankine
 121 cycle using one heat source is developed. It does not represent a real system
 122 but it allows a fast assessment of a various number of working fluids. It helps
 123 to select the suitable working fluids for the studied application. This model
 124 is based on the enthalpy changes in the process described in section 3.1.
 125 This model is able to perform either subcritical or supercritical cycle, which
 126 avoids the vaporization process and leads to a smaller system and a better
 127 heat recovery process [24]. Those relations are verified as long as the heat
 128 losses in the system and in the components are neglected. The 0D model

129 used is given by the system of equations (1):

$$\left\{ \begin{array}{l}
 P_{cond} = P_{sat}(T_{cond}), \\
 P_{f_{in},pump} = P_{cond}, \\
 T_{f_{in},pump} = T_{sat}(P_{f_{in},pump}) - \Delta T_{subcooling}, \\
 h_{f_{in},pump} = h(T_{f_{in},pump}, P_{f_{in},pump}), \\
 s_{f_{in},pump} = s(h_{f_{in},pump}, P_{f_{in},pump}), \\
 P_{f_{out},pump} = P_{evap}, \\
 h_{f_{out},pump} = h_{f_{in},pump} + \frac{(h_{f_{out},pump_{is}} - h_{f_{in},pump})}{\eta_{pump_{is}}}, \\
 T_{f_{out},pump} = T(h_{f_{out},pump}, P_{f_{out},pump}), \\
 s_{f_{out},pump} = s(h_{f_{out},pump}, P_{f_{out},pump}), \\
 P_{f_{out},boiler} = P_{f_{out},pump}, \\
 h_{f_{out},boiler} = h_{f_{out},pump} + \frac{\dot{Q}_{gas}}{\dot{m}_f}, \\
 T_{f_{out},boiler} = T(h_{f_{out},boiler}, P_{f_{out},boiler}), \\
 s_{f_{out},boiler} = s(h_{f_{out},boiler}, P_{f_{out},boiler}), \\
 P_{f_{out},exp} = P_{cond}, \\
 h_{f_{out},exp} = h_{f_{out},boiler} + (h_{f_{out},boiler} - h_{f_{out},exp_{is}})\eta_{exp_{is}}, \\
 s_{f_{out},exp} = s(h_{f_{out},exp}, P_{f_{out},exp}), \\
 P_{f_{out},cond} = P_{f_{out},exp}, \\
 T_{f_{out},cond} = T_{sat}(P_{f_{out},cond}) - \Delta T_{subcooling}, \\
 h_{f_{out},cond} = h(T_{f_{out},cond}, P_{f_{out},cond}), \\
 s_{f_{out},cond} = s(h_{f_{out},cond}, P_{f_{out},cond}).
 \end{array} \right. \quad (1)$$

130 where

$$\left\{ \begin{array}{l}
 h_{f_{out},pump_{is}} = h'(P_{f_{out},pump}, s_{f_{in},pump}), \\
 \dot{Q}_{gas} = \dot{m}_{gas} c_{p_{gas}} * (T_{gas_{in},boiler} - T_{gas_{out},boiler}), \\
 h_{f_{out},exp_{is}} = h'(P_{f_{out},exp}, s_{f_{out},boiler}), \\
 h_{f_{out},exp} \geq h''(x_{f_{out},exp_{min}}, P_{f_{out},exp}).
 \end{array} \right. \quad (2)$$

131 In table 1 one can find the simulation model parameters, and the abbrevia-
 132 tions are given in the appendix. In addition to that, a routine verifying that
 133 the pinch point (PP) is respected during the evaporation process. Refprop
 134 database [25] is used to compute the following quantity: h , s , T , P_{sat} and
 135 T_{sat} . Input variables of the model (1) are the gas mass flow and temperature
 136 (denoted by \dot{m}_{gas} and $T_{gas_{in},boiler}$) entering in the system and the condensing
 137 temperature (T_{cond}). Outputs of the model are the power produced by the

Model parameters	Variable in (1)	unit	value
Pump isentropic efficiency	η_{pumpis}	%	65
Expander isentropic efficiency	η_{expis}	%	70
Maximum evaporating pressure	P_{evap}	bar	40
Minimum condensing pressure	P_{cond}	bar	1
Maximum pressure ratio	$\frac{P_{evap}}{P_{cond}}$	-	40:1
Pinch points HEX	PP	K	10
Pressure ratio among HEX	$\frac{P_{f_{out,boiler}}}{P_{f_{out,pump}}}$	-	1
Minimum quality after expansion	$x_{f_{out,exp,min}}$	-	0.9

Table 1: 0D model parameters

138 expansion \dot{W}_{exp} , the power consumed by the compression \dot{W}_{pump} and the
139 net output power NOP which are defined as:

$$\begin{cases} NOP &= \dot{W}_{exp} - \dot{W}_{pump}, \\ \dot{W}_{exp} &= \dot{m}_f * (h_{f_{in,exp}} - h_{f_{out,exp}}), \\ \dot{W}_{pump} &= \dot{m}_f * (h_{f_{in,pump}} - h_{f_{out,pump}}). \end{cases} \quad (3)$$

140 The model 1 is not dynamic and does not represent any real components
141 performance. A dynamic 1D model is therefore developed to evaluate the
142 system performance on more realistic dynamic driving conditions.

143 3.3. 1D dynamic modeling of a Rankine cycle

144 3.3.1. Tank

145 The reservoir is modeled by a fixed volume, which can be either vented to
146 the atmosphere or be hermetic (depending on the condensing pressure) in
147 order to avoid sub atmospheric conditions. Mass and energy conservation
148 equations are:

$$\begin{cases} \dot{m}_{f_{out,tank}} - \dot{m}_{f_{in,tank}} &= \frac{\partial m_{f_{tank}}}{\partial t}, \\ \dot{m}_{f_{in,tank}} h_{f_{in,tank}} - \dot{m}_{f_{out,tank}} h_{f_{out,tank}} &= m_{f_{tank}} \frac{\partial h_{f_{tank}}}{\partial t}. \end{cases} \quad (4)$$

149 *3.3.2. Working fluid pump*

150 The working fluid pump is simply represented by a fixed displacement and
151 isentropic efficiency. The volumetric efficiency is a function of the outlet
152 pressure. This law is identified thanks to experimental data:

$$\dot{m}_{f_{out,pump}} = \rho_{f_{in,pump}} \frac{N_{pump}}{60} C_{cpump} \eta_{pumpvol}. \quad (5)$$

153 The outlet enthalpy is calculated as shown in the equation for $h_{f_{out,pump}}$ in
154 the 0D model (1).

155 *3.3.3. Heat exchangers: Evaporator(s) and condenser(s)*

156 The models are developed to dynamically predict temperature and enthalpy
157 of transfer and working fluid at the outlet of each heat exchanger (HEX).
158 When coming to dynamic models of those components, two methodologies
159 can be found in the literature: moving boundary (MB) and finite volume
160 (FV) models. Usually more complex in terms of computational capacity
161 needed due to the high number of system states, the FV approach has the
162 advantage to be more powerful and robust concerning the prediction. Both
163 approaches have been widely used in large power recovery system and control
164 system design [26, 27, 28, 29] and results in a simplification of the heat
165 recovery boiler/condenser geometry in a great extent (i.e. by representing
166 the boiler by a straight pipe in pipe counterflow heat exchanger). In this
167 study, the FV approach is preferred since it easily handles starting and shut
168 down phases [30] when only few papers adressed those cases with a MB
169 approach [31].

170 ***Model assumptions***

171 Several assumptions are done to simplify the problem in a great extent. These
172 ones are usually admitted when coming to heat exchanger modeling [32, 26]:

- 173 • The transfer fluid is always considered in single phase, i.e. no conden-
174 sation in the EGR/exhaust gases is taken into account.
- 175 • The conductive heat fluxes are neglected since the predominant phe-
176 nomenon is the convection.
- 177 • All HEX are represented by a straight pipe in pipe counterflow heat
178 exchanger of length L .

- 179 • Fluid properties are considered homogeneous in a volume.
- 180 • Pressure dynamics is neglected since it is very fast compared to those
- 181 of heat exchanger.

182 ***Governing equations***

183 Boiler(s) and condenser(s) models are based on mass and energy conservation
184 principles.

- 185 • Working fluid (internal pipe):

$$\begin{cases} A_{cross_f} \frac{\partial \rho_f}{\partial t} + \frac{\partial \dot{m}_f}{\partial z} = 0, \\ A_{cross_f} \frac{\partial \rho_f h_f}{\partial t} + \frac{\partial \dot{m}_f h_f}{\partial z} + \dot{q}_{conv_{f_{int}}} = 0, \\ \dot{q}_{conv_{f_{int}}} = \alpha_f P e_{exch_f} (T_f - T_{wall_{int}}). \end{cases} \quad (6)$$

- 186 • Internal pipe wall: An energy balance is expressed at the wall between
- 187 the working fluid and the gas and is expressed as follows:

$$\dot{Q}_{conv_{f_{int}}} + \dot{Q}_{conv_{g_{int}}} = \rho_{wall} c_{p_{wall}} V_{wall_{int}} \frac{\partial T_{wall_{int}}}{\partial t}. \quad (7)$$

- 188 • Gas side (external pipe): The energy conservation is then formulated
- 189 under the following form:

$$\rho_g A_{cross_g} c_{p_g} \frac{\partial T_g}{\partial t} + c_{p_g} \dot{m}_g \frac{\partial T_g}{\partial z} + \dot{q}_{conv_{g_{int}}} + \dot{q}_{conv_{g_{ext}}} = 0, \quad (8)$$

190 where the convection on the external side is used to represent the heat
191 losses to the ambient.

- 192 • External pipe wall: As for the internal pipe an energy balance is ex-
- 193 pressed between the gas and the ambient:

$$\dot{Q}_{conv_{g_{ext}}} + \dot{Q}_{conv_{amb_{ext}}} = \rho_{wall} c_{p_{wall}} V_{wall_{ext}} \frac{\partial T_{wall_{ext}}}{\partial t}. \quad (9)$$

194 In equation (7) and (9) the convection heat flow rate (\dot{Q}_{conv}) is expressed as:

$$\begin{aligned} \dot{Q}_{conv_{j_k}} &= \alpha_j A_{exch_{j_k}} (T_{wall_k} - T_j), \\ \text{where } j &= g, f, amb \\ \text{and } k &= int, ext. \end{aligned} \quad (10)$$

195 Furthermore, to complete the system, one need boundary and initial condi-
 196 tions. Time-dependent boundary conditions are used at $z = 0$ and $z = L$
 197 ($t > 0$):

$$\dot{m}_f(t, 0) = \dot{m}_{f_0}(t), \quad (11)$$

$$h_f(t, 0) = h_{f_0}(t), \quad (12)$$

$$\dot{m}_g(t, L) = \dot{m}_{g_L}(t), \quad (13)$$

$$T_g(t, L) = T_{g_L}(t). \quad (14)$$

198 The initial conditions for the gas and wall temperatures and working fluid
 199 enthalpy are given by ($z \in [0, L]$):

$$h_f(0, z) = h_{f_{init}}(z), \quad (15)$$

$$T_{wall_{int}}(0, z) = T_{wall_{int_{init}}}(z), \quad (16)$$

$$T_g(0, z) = T_{g_{init}}(z), \quad (17)$$

$$T_{wall_{ext}}(0, z) = T_{wall_{ext_{init}}}(z), \quad (18)$$

200 ***Heat transfer and pressure drop***

201 To model the convection from the transfer fluid to the pipe walls and from
 202 the internal pipe to the working fluid, a heat transfer coefficient (α) is needed.
 203 The convection from a boundary to a moving fluid is usually represented by
 204 the dimensionless number Nusselt (Nu) which is the ratio of convective to
 205 conductive heat transfer.

$$Nu(\alpha) = \frac{\alpha l}{\lambda}, \quad (19)$$

206 where l represents a characteristic length and is, in this case, the hydraulic
 207 diameter. Numerous correlations to approach this number can be found in
 208 the literature and are usually derived from experiments, see for example [33].
 209 In single phase, the Gnielinski correlation is chosen for both fluids. In two
 210 phase, Chen (for evaporation) and Shah (for condensation) correlations are
 211 used. Pressure drop in both fluids are taken into account in order to simulate
 212 the real performance of the system. The pressure drop can be split into three
 213 main contributors:

$$\Delta P = \Delta P_{static} + \Delta P_{momentum} + \Delta P_{friction}, \quad (20)$$

214 where the static pressure drop (ΔP_{static}) is function of the change in static
 215 head (i.e. the height), the momentum pressure drop ($\Delta P_{momentum}$) depends

216 on the change on density during phase change and the friction contribution
 217 ($\Delta P_{friction}$) is function of the speed of the fluid and the considered geometry.
 218 Table 2 shows the different correlations used depending on flow conditions.
 219 In laminar single phase, the assumption of a constant heat flux at the wall
 is made.

		Laminar	Turbulent
Heat transfer	Single phase	Nu = 4.36	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 HEX

220

221 3.3.4. Valve(s)

222 The fluid flow \dot{m} through the valve is modeled using a compressible valve
 223 equation of the form:

$$\dot{m}_{f_{in,v}} = C_{d_v} S_{eff_v} \sqrt{\rho_{f_{in,v}} P_{f_{in,v}}} \phi, \quad (21)$$

224 where the compressibility coefficient ϕ is defined as:

$$\phi = \frac{2\gamma_f}{\gamma_f - 1} \left(\varphi^{\frac{2}{\gamma_f}} - \varphi^{\frac{\gamma_f+1}{\gamma_f}} \right), \quad (22)$$

with

$$\varphi = \begin{cases} \frac{P_{f_{out,v}}}{P_{f_{in,v}}} & \text{if } \frac{P_{f_{out,v}}}{P_{f_{in,v}}} > \frac{2}{\gamma_f+1} \frac{\gamma_f}{\gamma_f-1} \\ \frac{2}{\gamma_f+1} \frac{\gamma_f}{\gamma_f-1} & \text{if } \frac{P_{f_{out,v}}}{P_{f_{in,v}}} \leq \frac{2}{\gamma_f+1} \frac{\gamma_f}{\gamma_f-1}, \end{cases} \quad (23)$$

225 where γ_f is the ratio of the specific heats of the working fluid and depends on
 226 the temperature and the pressure. Equation (23) means that the parameter
 227 φ is either the pressure ratio if the flow is subsonic or the critical pressure
 228 ratio when the flow is supersonic.

229 *3.3.5. Expansion machine*

230 Several studies have been carried out in order to choose the correct expansion
 231 machine for Rankine based recovery system [34, 35]. In most of them where
 232 vehicle installation is considered, turbine expanders are preferred for their
 233 compactness and their good performance [8, 36] since the major advantage
 234 of volumetric expander such as piston machines is the expansion ratio [37].
 235 Though, recent study [38] has shown turbine with expansion ratio over 40:1
 236 on a single stage with really good performance at tolerable speed for a vehicle
 237 installation. In this study, only a kinetic expander is modeled. The turbine
 238 nozzle is represented by the following equation:

$$\dot{m}_{f_{in,exp}} = K_{eq} \sqrt{\rho_{f_{in,exp}} P_{f_{in,exp}} \left(1 - \frac{P_{f_{in,exp}}}{P_{f_{out,exp}}}\right)^{-2}}. \quad (24)$$

239 And the isentropic efficiency is calculated according to the following relation:

$$\eta_{exp_{is}} = \eta_{exp_{is_{max}}} \left(\frac{2c_{us}}{c_{us_{max}}} - \frac{c_{us}^2}{c_{us_{max}}^2} \right), \quad (25)$$

where

$$c_{us} = \frac{u}{c_s} = \frac{\omega_{exp} R_{exp}}{2\sqrt{h_{f_{in,exp}} - h_{f_{in,exp_{is}}}}}. \quad (26)$$

240 Model parameters are fitted using data from supplier and similarity relation
 241 [39].

242 *3.3.6. Other heat exchanger(s)*

243 In order to describe the vehicle cooling system, the number of transfer unit
 244 (NTU) approach is used. It is commonly adopted when it comes to single
 245 phase heat exchanger modeling. For an air cooled radiator the following
 246 relations are used:

$$\dot{Q}_{air} = \dot{m}_{air} c_{p_{air}} \varepsilon (T_{coolant_{in}} - T_{air_{in}}). \quad (27)$$

247 For a given geometry, ε can be calculated using correlations based on the heat
 248 capacity ratio. By considering parallel flow configuration for the radiators,

249 the effectiveness can be written:

$$\varepsilon = \frac{1 - e^{-NTU \left(1 + \frac{(\dot{m}c_p)_{min}}{(\dot{m}c_p)_{max}}\right)}}{1 + \frac{(\dot{m}c_p)_{min}}{(\dot{m}c_p)_{max}}}, \quad (28)$$

$$\text{with } NTU = \frac{UA}{(\dot{m}c_p)_{min}}. \quad (29)$$

250 3.3.7. Coolant pump and fan

251 The coolant pump model used is a map-based model function of engine speed
 252 and pressure rise. This one is sized to deliver enough subcooling even at high
 253 engine load. The engine fan is also a map-based model delivering a given
 254 mass flow at a given speed. The fan consumption is calculated according to:
 255

$$\dot{W}_{fan} = C_{fan} \rho_{air} N_{eng} G_{ratio} N_{fan}^2, \quad (30)$$

256 where the coefficients C_{fan} and G_{ratio} are dependent on the fan model and
 257 vehicle. The mass flow rate blown by the fan is mapped according to data
 258 from supplier and depends on the fan speed and atmospheric conditions. The
 259 air mass flow rate going through the cooling package (\dot{m}_{air}) is a combination
 260 of the natural air mass flow rate (corresponding to a fraction of the vehicle
 261 speed) and the forced mass flow rate (corresponding to the mass flow blown
 262 by the fan).

$$\dot{m}_{air} = \rho_{air} A_{cool\ pack} S r_{air} V_{vehicle} + \dot{m}_{fan}(N_{fan}, \rho_{air}), \quad (31)$$

263 where $S r_{air}$ is the ratio between the vehicle speed and the air speed in front of
 264 the cooling package and is either calculated via computational fluid dynamics
 265 (CFD) or measured in a wind tunnel.

266 4. SYSTEM OPTIMIZATION

267 4.1. Key aspects

268 In this section, the degrees of freedom used to optimize the WHRS are de-
 269 tailed:

- 270 1. Fluid choice: the fluid selection is a critical part of the system opti-
271 mization. The correct fluid choice has to match both heat source and
272 cold sink in order to generate as much power as possible [40, 41]. From
273 environmental and legal points of view, the working fluid has to respect:
- 274 • Its chemical class: chlorofluorocarbons (CFCs) have been ban-
275 ished by the Montreal Protocol and hydrochlorofluorocarbons (HCFCs)
276 production is planned to be phased out by 2030.
 - 277 • Its presence on the global automotive declarable substance list
278 (GADSL).
 - 279 • Its chemical properties such as the global warming potential (GWP),
280 the ozone depletion potential (ODP) or the risk phrases (R-phrases).
 - 281 • Its classification according the national fire protection agency (NFPA)
282 704 classification (ranking above 1 in Health or Instability class)

283 In top of that, the freezing point which has to be below 0 °C.

- 284 2. Components choice and design: the correct choice of components and
285 particularly the expansion machine have an important impact on the
286 system performance and the control design. Indeed, a volumetric ex-
287 pander is less stringent in terms of degree of superheat and tolerate
288 a given amount of liquid during the expansion process whereas a kinetic
289 expander requires a higher degree of superheat in order to have a full
290 vapor expansion (liquid droplets can cause blade erosion and broke the
291 machine). The design of all other components of the Rankine system
292 is also critical to maximize its potential. For example, too big heat
293 exchangers show higher performance but also inertia which could be a
294 disadvantage when coming to highly dynamic driving cycle since the
295 more interesting points (i.e. high load engine operating points) are not
296 lasting for long. A heavy evaporator is therefore not catching up the
297 maximum potential of this high heat flow rate.
- 298 3. Heat sources and sinks arrangement: the architecture of sources and
299 sinks has to be adapted to increase overall performance. Heat sources
300 choice and arrangement impact a lot the system performance by chang-
301 ing the heat input to the system. The cold sinks choice is influencing
302 the condensing pressure so the overall pressure ratio (and therefore the
303 power generated by the expander).

304 4. Other system interactions: as the final goal is to implement the system
 305 in a heavy duty vehicle, the WHRS must be considered not as a stand-
 306 alone system but as a connected sub system of the complete vehicle.
 307 The interactions of the Rankine system on the other sub-systems have
 308 to be taken into account (e.g. increase in fan consumption due to the
 309 heat rejection coming from the condenser).

310 *4.1.1. Investigated architectures and components*

311 Several studies have been conducted in the field of waste heat recovery Rank-
 312 ine based systems for mobile applications. A screening of the different heat
 313 sources available is reported in [5] and shows that the most promising ones
 314 are the EGR and the Exhaust streams. In the present study, only these
 315 two heat sources are considered since they present the higher grade of tem-
 316 peratures among other sources. Therefore four different Rankine layout are
 317 studied:

- 318 1. Exhaust recovery only where the only heat source are the exhaust gases.

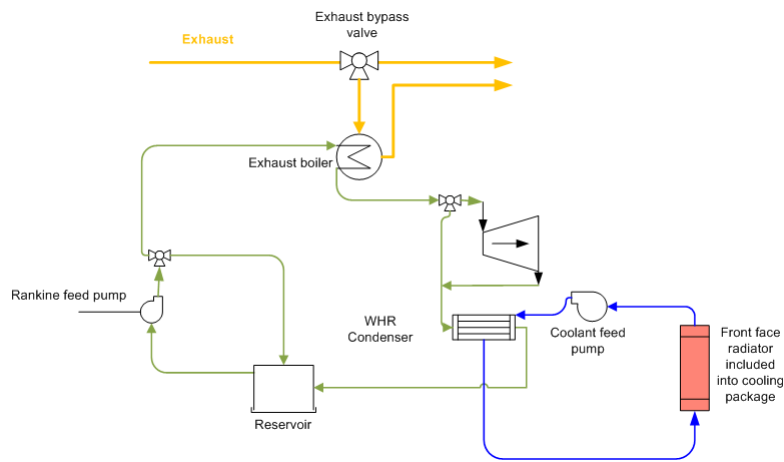


Figure 3: Exhaust only system schematic

- 319 2. EGR recovery only where only the EGR gases are used as the only heat
 320 source.

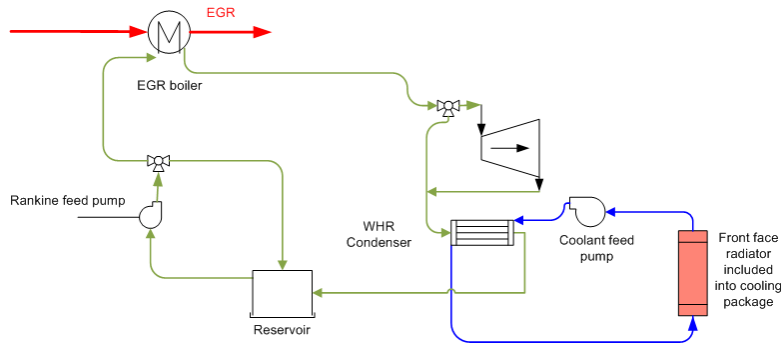


Figure 4: EGR only system schematic

- 321 3. Both sources in parallel where the working fluid is split into two streams
 322 heated up separately by each source and then mixed before the ex-
 323 pander.

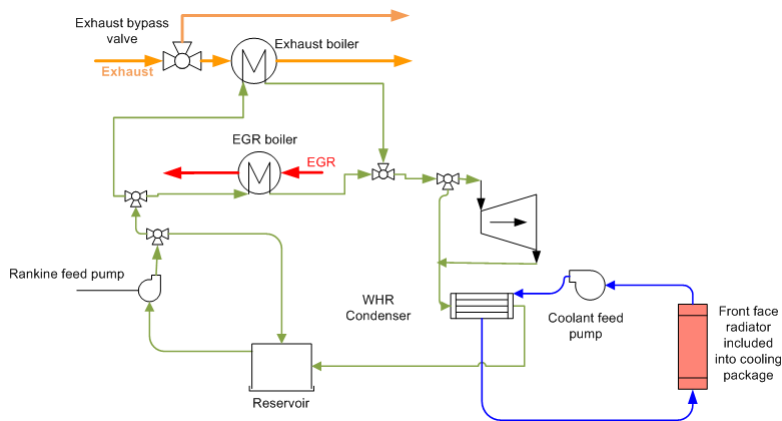


Figure 5: Exhaust and EGR in parallel system schematic

- 324 4. EGR and exhaust in series where the EGR gases are used to preheat
 325 the fluid and the exhaust gases to vaporize and superheat. Using the
 326 EGR as a preheater, instead of a superheater, is chosen to lower the
 327 EGR gases temperature after the evaporation process.

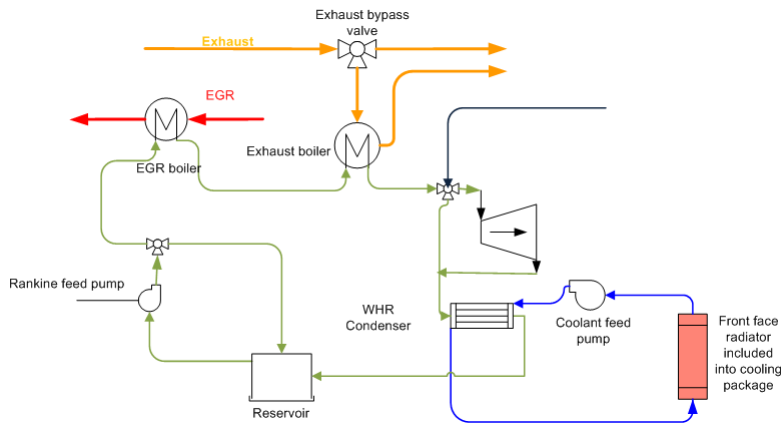


Figure 6: Exhaust and EGR in series system schematic

328 Coupled to that, two different cooling architecture are approached:

- 329 • A first one (called in the following Cooling Config 1) which uses a low
 330 temperature radiator dedicated to the Rankine condenser and is placed
 331 between the charge air cooler (CAC) and the engine radiator.

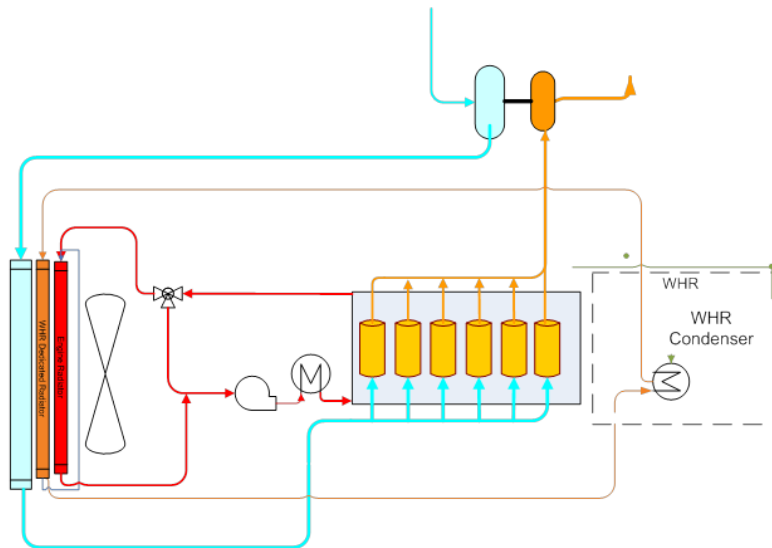


Figure 7: Cooling config 1

- 332 • A second one (called in the following Cooling Config 2) using the engine

333 coolant as heat sink for the Rankine cycle. In that case, a derivation
 334 of the coolant is done in front of the engine to benefit from the lowest
 335 temperature grade.

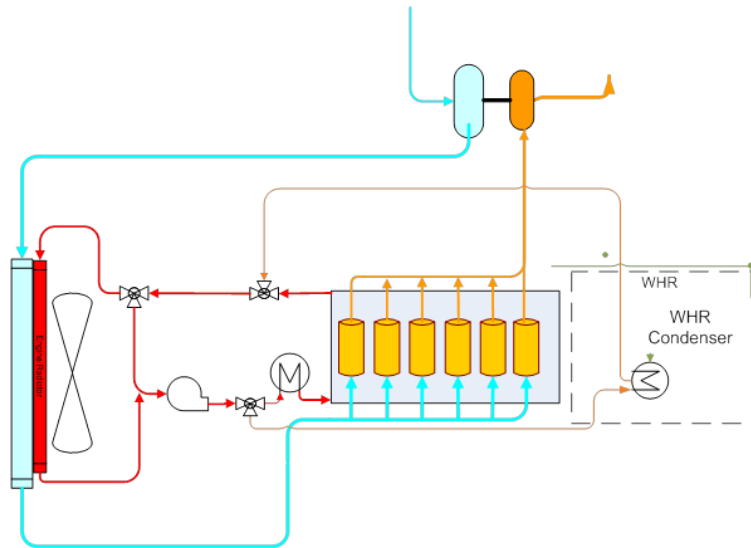


Figure 8: Cooling config 2

336 Concerning the components, as previously said, the study is limited to one
 337 expansion machine technology: turbine expander. For the heat exchangers
 338 (evaporator(s) and condenser), only counter-current configurations which are
 339 usually used in this kind of applications [42] are considered.

340 4.2. Duty cycles

341 Driving conditions are acting as input disturbances and therefore, their im-
 342 pact on the target performance must be studied with care.

343 4.2.1. Steady state evaluation

344 Under steady state driving conditions, the performance is evaluated by ex-
 345 pressing the weighted average net output power of the 1D model (3.3) (the
 346 NOP, which is the additional power that the engine receives, therefore cor-
 347 responds to the fuel economy) on 13 engine operating points (summarized

Name	Vehicle speed	EGR mass flow	EGR temperature	Exhaust mass flow	Exhaust temperature	Weight factor
Parameter	$V_{vehicle}$	\dot{m}_{egr}	T_{egr0}	\dot{m}_{exh}	T_{exh0}	w_i
Unit	km/h	g/s	$^{\circ}C$	g/s	$^{\circ}C$	$\%$
1	20	31.5	263.1	78.7	237.9	6.9
2	85	38	409.5	119.8	338.2	9.0
3	60	59.5	635.0	309.3	443.9	4.9
4	85	54.6	544.0	252.4	413.0	2.6
5	75	46.1	454.0	154.6	366.4	18.9
6	85	56.3	247.5	85.7	212.5	10.5
7	30	85.9	631.0	352.7	425.1	2.8
8	85	69.4	562.5	290.5	405.5	3.6
9	50	58	473.0	183.2	336.2	12.7
10	85	59.8	251.0	95.2	216.0	11.2
11	45	87.1	581.0	326.8	400.8	2.3
12	75	68.9	472.0	198.4	359.6	10.7
13	85	62.9	252.5	102.8	217.5	3.9

Table 3: Steady state evaluation: Driving conditions and weight for 13 engine operating points

348 in table 3) These operating points are chosen to represent a classical long
349 haul driving cycle and weighted according to the percentage of energy used
350 on each operating point. Operating point number 5 is identified as the de-
351 signing point whereas the operating points 3 and 11 are considered critical
352 due to the high engine load and the low vehicle speed.

353 4.2.2. Dynamic evaluation

354 In order to accurately assess the potential of the WHRS, dynamic driving cy-
355 cles are also used to complete the study and check whether the performance
356 found with the previous method is correct. This is really important when
357 coming to thermal systems performance estimation since they generally have
358 long response time [43]. The driving cycle used is split into 7 phases (sum-
359 marized in table 4) supposed to represent all conditions of a long haul truck
360 usage. In the previous 1D model (3.3), each phase is considered, in the rest
361 of the study, as a driving cycle of approximately the same length (denoted by

Driving cycle	1	2	3	4	5	6	7
Road Type	Extra urban	Highway	Highway	Extra urban	Extra urban	Extra urban	Rolling
Vehicle speed	Medium	High	Medium	Low	Medium	High	High
Weight factor w_i (%)	10	10	50	7.5	10	7.5	5

Table 4: Dynamic evaluation: Driving conditions and weight for the 7 phases

362 a number from 1 to 7) and weighted according to their real life importance
363 (for a long haul truck highway is predominant).

364 4.3. System performance evaluation

365 The criterion used for the performance evaluation under steady state and
366 dynamic driving conditions, is the total net reinjected power to the conven-
367 tional driveline. This is done by taking into account the producer (WHRS
368 expander) and different consumers (cooling fan, WHRS pump and WHRS
369 coolant pump) and assuming them to be mechanically driven (this is not
370 always true for the pumps but efficiencies are detuned to take into account
371 the mechanical to electrical conversion). A complete vehicle model integrat-
372 ing engine, exhaust after treatment system (EATS), transmission, cooling
373 package, WHRS and road environment is used to simulate the total vehicle
374 approach and calculate the power needed to drive the vehicle. The perfor-
375 mance criterion (PC) is then calculated as the ratio of this reinjected power
376 to the engine power:

$$PC_i = \int_{t_{init}}^{t_{final}} \frac{\dot{W}_{exp} - \dot{W}_{pump} - \dot{W}_{cool,pump} - \dot{W}_{fan}}{\dot{W}_{eng}}, \quad (32)$$

377 where the engine power (\dot{W}_{eng}) taking into account the mechanical auxiliaries
378 consumption mounted on it and the increase in exhaust backpressure (due
379 to the exhaust evaporator). The vehicle gross weight is assumed constant
380 and equal to 36 tons which intends to represent the average load on a long
381 haul truck. The performance criterion (PC) over the different steady state
382 operating points or driving cycles is then calculated by summing the weighted
383 PC on each points/cycles:

$$PC = \sum_{i=1}^k w_i PC_i, \quad (33)$$

384 where $k \in [1 \ 13]$ for steady state evaluation (presented in section 4.2.1) and
385 $k \in [1 \ 7]$ for evaluation on dynamic driving cycle (presented in section 4.2.2).

386 5. RESULTS AND DISCUSSION

387 5.1. 1D model validation

388 In this section some component models, judged as critical for the overall
389 system performance evaluation, are first validated thanks to supplier or ex-
390 perimental data. The different studied configurations being made of the same
391 components model it has been decided to validate the models components
392 by components. The validation is done by comparing experimental to the
393 modeling results. A model is further considered as valid if the average mod-
394 eling error is below 5% of the predicted quantity. Since the main dynamic of
395 the system is contained in the evaporators [43] and the final aim is to predict
396 the power generated by the system only validation figures are presented for
397 the evaporators and the expansion machine. It should be said that a more
398 detailed validation on the whole system mounted on the vehicle should be
399 carried out but this requires the system to be built. Unfortunately this tech-
400 nology is still under investigation at the truck makers level and no figures
401 are available yet. This study intends then to compare the architecture and
402 analyze their impact on the truck fuel consumption.

403 5.1.1. Heat exchangers

404 A high attention is paid to the evaporators in order to accurately predict the
405 steady state and dynamic performance of those components (corresponding
406 to the model presented in section 3.3.3). In this paper, a finite volume ap-
407 proach has been chosen to implement the continuous set of equations (equa-
408 tions 6, 7, 8, 9). Figure 9 shows the schematic of the discretized model.
409 Table 5 and 6 show respectively steady state and dynamic prediction errors.
410 Note that in both cases the relative error is computed according to the max-
411 imum temperature difference between the heat exchanger bounds (usually
412 $T_{gL} - T_{f0}$). The steady state validation is conducted on a lot of operating
413 conditions supposed to represent the complete range of operation for those
414 components. The dynamic behavior is evaluated on different load point varia-
415 tions but obviously need further validation especially on fast change that can

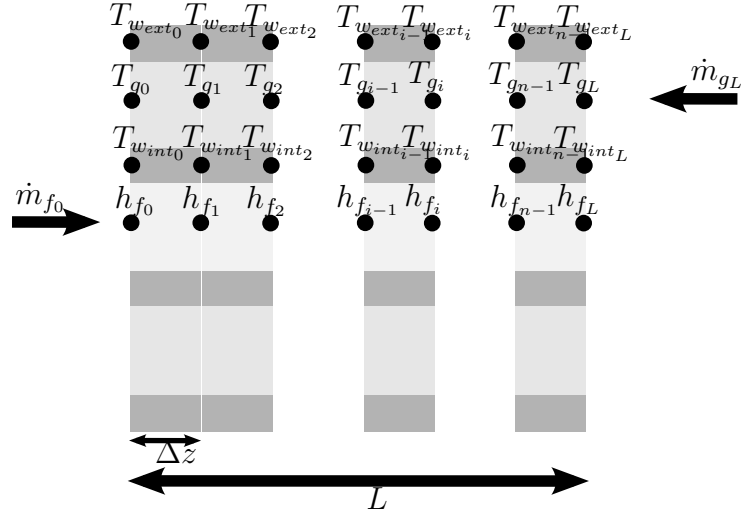


Figure 9: HEX model schematic

416 take place on real driving conditions. However, the mean relative modeling
 417 error remains lower than 3.5%, which is considered acceptable.

	$T_{f_{LEGRB}}$		$T_{f_{LExhB}}$		$T_{egr0_{EGRB}}$		$T_{exh0_{ExhB}}$	
Error	max	mean	max	mean	max	mean	max	mean
Absolute (K)	2.95	1.30	9.15	4.16	7.54	2.54	15.47	4.71
Relative (%)	0.57	0.29	8.84	3.28	2.34	0.61	8.61	3.40

Table 5: Evaporators steady state validation

	$T_{f_{LEGRB}}$		$T_{f_{LExhB}}$		$T_{egr0_{EGRB}}$		$T_{exh0_{ExhB}}$	
Error	max	mean	max	mean	max	mean	max	mean
Absolute (K)	4.5	1.5	25.9	2.3	7.9	2.8	20	4.2
Relative (%)	1.38	0.46	14.37	1.28	2.43	0.86	11.1	2.33

Table 6: Evaporators dynamic validation

418 5.1.2. Expansion machine

419 The turbine expander model presented in 3.3.5 is fitted thanks to supplier
 420 data. Figure 10 and 11 respectively show the working fluid inlet pressure and
 421 the generated power predicted by the model versus the normalized working

422 fluid mass flow entering in the turbine. Those two quantities are well fitted
423 and this model is further considered validated.

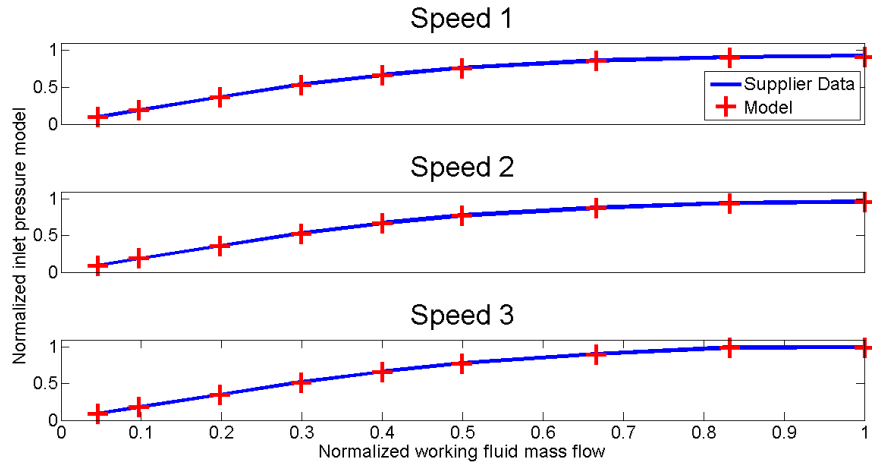


Figure 10: Turbine pressure model validation

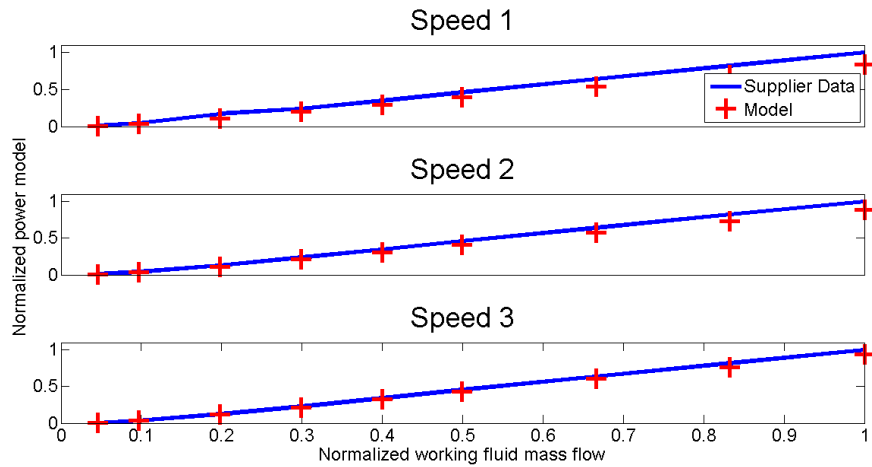


Figure 11: Turbine power model validation

424 *5.1.3. Model analysis*

425 The whole 1D models are built from the same component models. The iner-
426 tial effect of the pipes are neglected since their effect are negligible compared

427 to the other components dynamics (namely the evaporators) [43]. The full
428 model is then a combination of validated detailed models (e.g. heat exchang-
429 ers) and quasi-static models (pumps expansion machine and fan) used for
430 comparison purpose. This study then intends to compare the different heat
431 sources and sinks configurations possible on a heavy duty vehicle to select
432 the best system in terms of performance.

433 5.2. Optimization of the WHRS

434 5.2.1. Working fluid selection

435 From an exhaustive fluid list [25], all those that do not respect the different
436 criteria mentioned in 4.1 have been removed. However, as water is a good
437 reference fluid since it is generally used in power plant [44], it has been kept.
438 The results presented hereafter are coming from the ideal thermodynamic 0D
439 model presented in section 3.2 where all 13 operating points are simulated for
440 two condensing temperatures 60°C and 90°C, which intends to represent the
441 two cooling configurations presented in the previous section. The parameters
442 of the cycle, $P_{f_{out,pump}}$ and \dot{m}_f are optimized to reach the highest performance
443 (i.e. maximize the *NOP*). Here, each hot stream is simulated separately in
444 order to see the impact of the heat source on the Rankine fluid selection. The
445 simulation matrix contains 13 operating points (listed in section 5.2.2) and
446 13 selected working fluids. For the sake of simplicity, the results presented
447 in figure 12 show the number of occurrences where the fluid is in the top
448 five ¹ regarding the *NOP*. When analyzing each operating point and config-
449 uration separately among the 13x13 simulations, water is the best fluid for
450 heavily loaded operating points. For low and medium engine load, as gases
451 temperatures are lower and due to the large enthalpy of vaporization of water
452 and the high level of superheating required, it is not recommended to use it.
453 Acetone and ethanol show good performance at mid and high engine load no
454 matter of the cold sink temperature. Refrigerants such as R1233zd or Novec
455 649 show good results for heat source temperature under 280 °C for the low-
456 est condensing temperature. More exotic fluids such as cis-butene or MM
457 (silicon oil) could be attractive for low and medium engine load respectively
458 at 60°C condensing temperature for the first one and 90°C for the second one.

459

¹top five means the *NOP* related to the fluid is ranked in one of the five first position

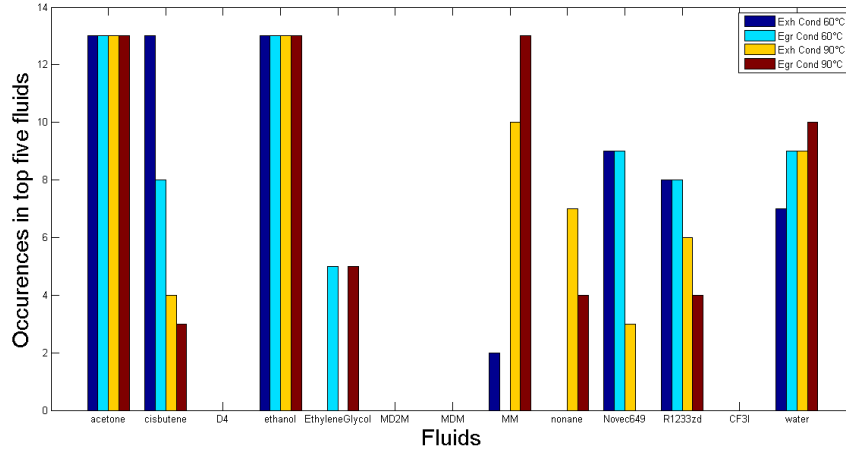


Figure 12: Number of occurrences of each fluid in top five ¹ for different boundary conditions

460 These first simulations results limit the number of investigated working fluids
 461 for the remaining part of this paper to the following ones: Acetone, Ethanol.
 462 Those two fluids represent the highest number of occurrences for the differ-
 463 ent configurations considered. As these fluids have similar volumetric flows
 464 it would be possible to use the same components' characteristics with only
 465 some minor changes (e.g. throat diameter for the turbine model and pump
 466 displacement). However due to the low flash point of Acetone (-20°C) only
 467 ethanol is then considered suitable for a mobile application.

468 5.2.2. Steady state performance analysis

469 Now, the performance criterion is analyzed on the 13 operating points and
 470 the 2 cooling architectures (Cooling config 1 and 2) for the previously cho-
 471 sen working fluids. The savings are computed thanks to the weight factors
 472 presented in table 3. Figure 13 presents the NOP to engine power ratio eval-
 473 uated for the 2 cooling configurations. It can be observed that the decrease
 474 due to higher condensing temperatures induced by cooling configuration 2 is
 475 somehow constant (between 11 and 15 %) no matter of the Rankine cycle
 476 arrangement. This drop in performance is due to the increase in condensing
 477 pressure which affects the overall pressure ratio through the expansion ma-
 478 chine and therefore its performance. This could be partially balanced by a

479 specific design of the expansion machine in order to have a variable nozzle ge-
 480 ometry that keeps the pressure ratio constant when the condensing pressure
 481 increases. A similar approach is done in [38] to adapt the nozzle geometry to
 482 the mass flow entering in the turbine. With those components, the parallel
 483 arrangement of the heat sources gives the best PC , followed by the serial
 484 one, the exhaust only and the EGR recovery. However the difference be-
 485 tween series and parallel layout is not so important and the lower number of
 486 valves needed by the first one could compensate this drop in performance.
 487 Moreover, in this configuration, as the working fluid mass flow is controlled
 488 to get a superheated vapor state at the outlet of the tailpipe boiler the mass
 489 flow rate is then higher than in any other configurations. It results into lower
 490 EGR temperature which could be a benefit in terms of engine performance
 491 and pollutant emission control [4]. Last but not least, with the EGR only
 492 solution even if the weight and installation impact is low (the heaviest com-
 493 ponent is the EGR evaporator that replaces the traditional EGR cooler),
 494 the PC seems too low for a vehicle installation. This obviously needs further
 495 analysis taking into account also the cost impact of each solution on the total
 cost of ownership.

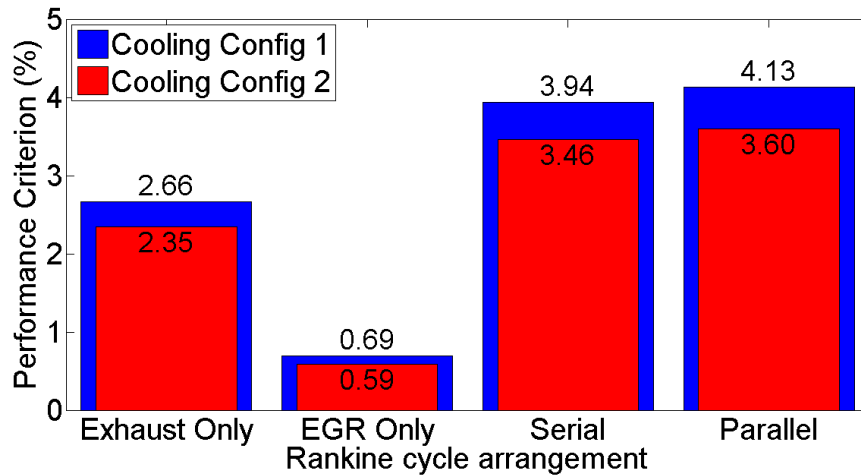


Figure 13: Steady state PC assessment

496

497 5.2.3. *Dynamic performance analysis*

498 Then, in order to validate the previously used method, dynamic simulations
 499 are run to further assess the performance criterion of the WHRS. Indeed,
 500 as previously said, the Rankine based recovery systems could have long time
 501 constant due to the boiler(s) inertia (wall capacity). This could help in terms
 502 of control by filtering some high transient of the heat sources but reduce the
 503 heat transferred to the fluid, since only a fraction of the heat contained in
 504 the hot gases is then used. In the following, the performance is assessed on
 505 7 different driving cycles (see table 4) representative of a long haul truck
 usage. An example of two of those road profiles is presented in fig 14. Each

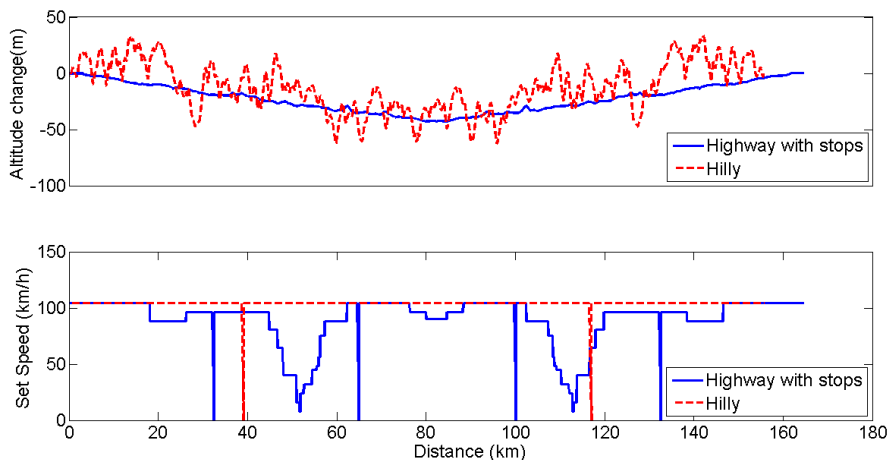


Figure 14: Road profiles examples

506 driving cycle is simulated separately starting from ambient conditions that
 507 can result in a lower PC due to the long warm up time of the exhaust after
 508 treatment system. Weights (see 4) are applied to the different driving cycles
 509 to calculate the total performance criterion of the WHRS.

510 Figures 15 and 16 show the PC reached by each Rankine configuration re-
 511 spectively for cooling configuration 1 and 2. As shown is 5.2.2 the decrease in
 512 performance using cooling configuration 2 rather than cooling configuration
 513 1 is more or less constant and around 11%. The main information brought
 514 by this study remains the lower fuel savings when simulating the system in
 515 dynamic instead of steady state, which can be as big as 50% for the systems
 516 using exhaust as heat source. This is due to two main reasons:
 517

- 518 • the exhaust after treatment system, which has a very important con-
519 stant time, causes big temperature drop during fast highly loaded en-
520 gine conditions where a lot of heat is supposed to be available.
- 521 • the non optimal design of the tailpipe boiler used in the simulation
522 model. Indeed the validation of the model shown in section 5.1 is
523 based on prototypes components that do not represent the optimum in
524 terms of size and transient performance.
- 525 • the constraint implemented on the EGR temperature at the evaporator
526 outlet not to derate the emission control. The maximum EGR temper-
527 ature is set to 150 °C which on some phases is not going hand in hand
528 with the superheat level control. The EGR temperature becomes the
529 control objective and when the superheat is not sufficient the expander
530 is not fed with working fluid and therefore the power production is null.

531 Table 7 resumes the time (relative to the duty cycle time) where the superheat
532 is sufficient to feed the expansion vapor. Systems recovering heat from the
533 exhaust stream mainly suffer from a long start-up phase but then the system
534 never lose the superheat level needed to expand the working fluid in the
535 kinetic turbine. This long start-up is due to the boundary conditions used
536 where all the sub-systems initial temperatures are set to ambient. For the
537 system recovering heat only from the EGR the start-up is not significant since
538 on some cycles the system is generating power more than 99% of the time
539 (the EGR gets its normal operation temperature after few seconds whereas
540 the thermal inertia of the EATS makes the temperature rise very slow).
541 Nevertheless on highly loaded cycles (namely 3 and 7) the high engine load
542 results into high EGR temperature and to not interact too much with the
543 engine emissions system, the superheat is dropped to the detriment of the
544 EGR temperature. Superheated vapor is no longer generated by the boiler
545 and the fluid goes back to a diphasic state and the expansion machine is
546 bypassed.

547 Anyway, similarly to the previous results in steady state, the best system in
548 terms of fuel savings remains the EGR and exhaust in parallel with cooling
549 configuration 1 that brings 2.2% savings on the overall weighted driving cycle.
550 In addition to that, it can be seen that the relative performance are kept from
551 arrangement to arrangement (compared to section 5.2.2).

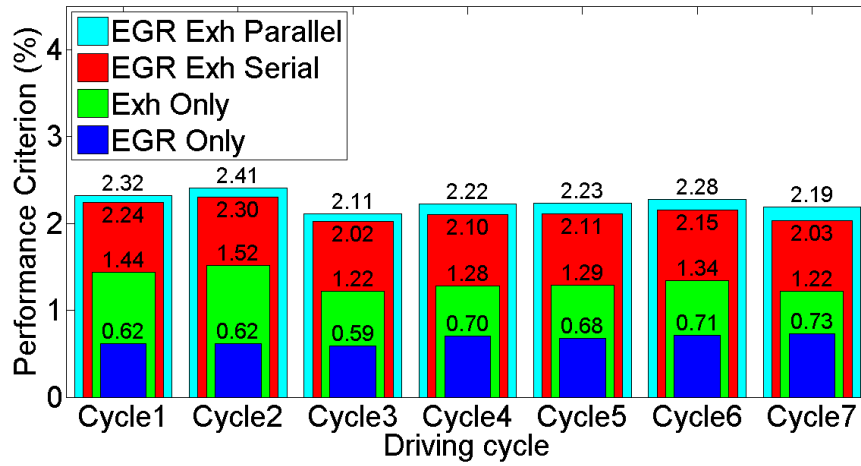


Figure 15: *PC* for cooling configuration 1 over dynamic driving cycles

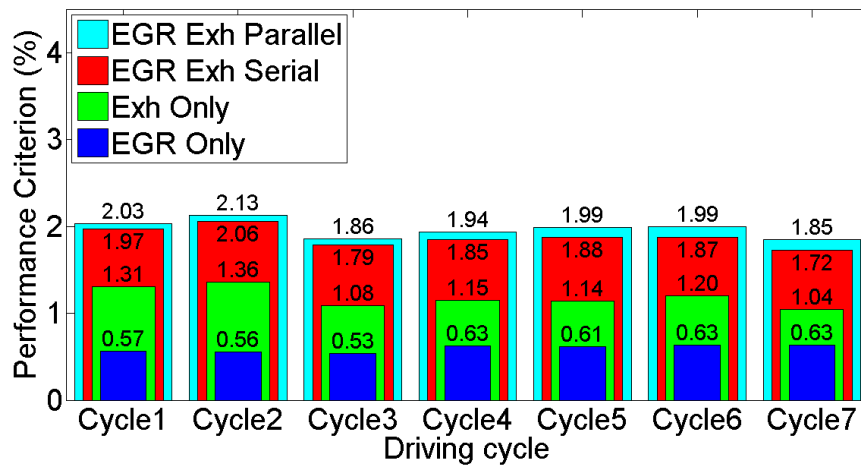


Figure 16: *PC* for cooling configuration 2 over dynamic driving cycles

Configuration	Exhaust	EGR	Serial	Parallel
Driving cycle				
1	93.37 %	99.11%	93.71%	93.93%
2	93.06%	99.30%	93.41%	93.25%
3	95.27%	83.38%	95.64%	95.44%
4	92.50%	93.37%	91.07%	92.06%
5	91.18%	97.26%	90.67%	91.55%
6	92.00%	98.30%	90.42%	91.46%
7	93.53%	88.94%	90.87%	92.92%

Table 7: Vapor creation time ratios summary

552 *5.2.4. Optimal WHRS*

553 The low performance figures presented in the previous sections are mainly
554 due to non optimized components for the considered application. In order to
555 evaluate what could be the economy brought by an optimized system, the dif-
556 ferent components constituting the WHRS are redesigned to perfectly match
557 the targeted application. In addition to that, a perfect insulation of these
558 different components is then considered. In this section, only cooling config-
559 uration 1 is evaluated since it has been shown that it leads to larger savings.
560 Both approaches previously used (steady state and dynamic analysis) are
561 presented in figure 17. Optimization has been done on boilers and condenser
562 size with respect with the additional weight. Pump and expansion machine
563 performance are increased to reach standards in power plant Rankine cycles
564 ($\eta_{pump, is} = 70\%$ and $\eta_{exp, is, max} = 78\%$). More acceptable results are reached
565 for a vehicle implementation of such a system, especially when considering a
566 system recovering from both EGR and exhaust in parallel. Again, a big step
567 is observed between the two evaluation methodologies which tends to prove
568 that the cycle division into a certain number of steady state engine operating
569 points is not adapted for performance evaluation of thermal systems which
570 generally have a long response time.

571

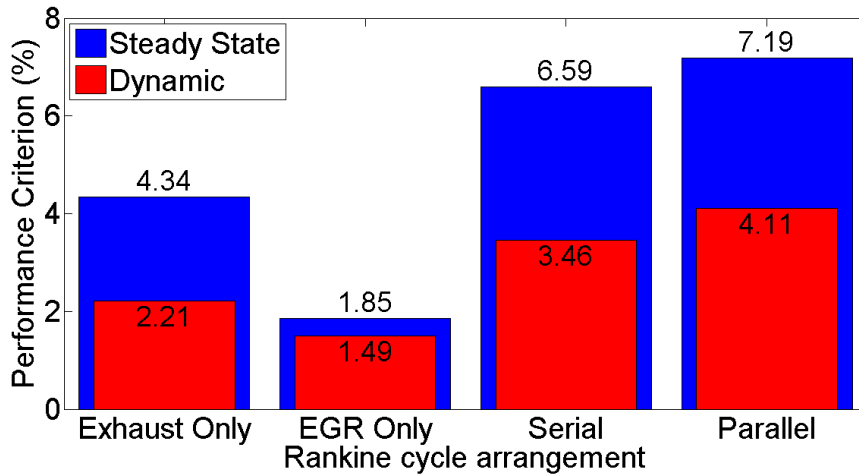


Figure 17: *PC* for optimal sizing of the components

572 **6. CONCLUSION**

573 Performance simulations of different WHRS for heavy duty trucks applica-
 574 tion was conducted to understand the potential of such a system in terms
 575 of fuel consumption decrease. Two different methodologies are used and dis-
 576 cussed. Usually, only the first approach, which consists to split a driving
 577 cycle into several steady state engine operating points, is used to assess the
 578 performance without any regards of the transient behavior of the different
 579 components composing the WHRS [3]. The second one, where a total vehicle
 580 approach is simulated over a wide variety of dynamic driving cycles repre-
 581 senting the usage of a long haul HD vehicle. In both methods architecture
 582 to architecture ranking is the same which tends to prove that the first ap-
 583 proach could be used for qualitative but not quantitative study. Using the
 584 second approach results into lower fuel savings and needs to be balanced in
 585 regards of the model validation done based onto prototypes, which are not
 586 representing what could be a mass-produced system. However, the absolute
 587 numbers should not be interpreted as the maximal potential for WHRS in
 588 HD trucks, since transient control of the system and components are not
 589 optimal. Different systems layout have been analyzed to maximize the sys-
 590 tem performance over a broad variety of driving cycles. However the results
 591 presented in this paper need to be treated carefully and further completed
 592 with the cost and the packaging effort for each configurations. An optimized

593 scenario is also presented where a specific attention has been paid to the
594 components size and performance in order to perfectly match the applica-
595 tion. However fuel savings are rather low compared to what can be found
596 in the literature [45, 3] and need to be further validated by experimental
597 results on a system mounted onto a vehicle. In addition to that, control
598 issues are not approached in this paper but remain a big part of the system
599 performance maximization. In this study, perfect sensors and actuators are
600 used, which reduce the control effort. Moreover, actual mass-produced con-
601 trol units are not as powerful as current laptop central processing unit and
602 reduce considerably the possibility in terms of advanced control algorithm
603 development. Recent studies have brought significant advances in this field
604 [46, 12, 47] but this still needs to be addressed when vehicle implementation
605 is touched.

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766 APPENDIX

767 Nomenclature

768 Acronyms

- 769 *CAC* Charge air cooler
770 *CFC* Chlorofluorocarbon
771 *CFD* Computational fluid dynamics
772 *EATS* Exhaust after treatment system
773 *EGR* Exhaust gas recirculation
774 *GADSL* Global automotive declarable substance list
775 *GWP* Global warming potential
776 *HCFC* Hydrochlorofluorocarbon
777 *HD* Heavy duty
778 *HEX* Heat exchanger
779 *NFPA* National fire protection agency

780	<i>NOP</i>	Net output power
781	<i>NTU</i>	Number of transfer unit
782	<i>ODP</i>	Ozone depletion potential
783	<i>ORC</i>	Organic Rankine cycle
784	<i>PC</i>	Performance criterion
785	<i>WHRs</i>	Waste heat recovery system
786	Greek letters	
787	α	Heat transfer coefficient ($W/m^2/K$)
788	ϵ	Heat exchanger efficiency ($-$)
789	η	Efficiency ($-$)
790	γ	Specific heat ratio ($-$)
791	λ	Heat conductivity ($W/m/K$)
792	ω	Angular velocity (rad/s)
793	ϕ	Compressibility factor ($-$)
794	ρ	Density (kg/m^3)
795	φ	Critical pressure ratio ($-$)
796	Latin letters	
797	\dot{m}	Mass flow (kg/s)
798	\dot{Q}	Heat flow rate (W)
799	\dot{q}	Linear heat flow rate (W/m)
800	\dot{W}	Power (W)
801	A	Area (m^2)
802	C_c	Cubic capacity (m^3)
803	C_d	Discharge coefficient ($-$)
804	c_p	Specific heat ($J/kg/K$)
805	G	Gear ratio ($-$)
806	h	Enthalpy (J/kg)
807	K_{eq}	Equivalent throat diameter (m^2)
808	N	Rotational speed (rpm)
809	Nu	Nusselt number ($-$)
810	P	Pressure (Pa)
811	Pe	Perimeter (m)

812	<i>PP</i>	Pinch point (K)
813	<i>r</i>	Ideal gas constant ($J/kg/K$)
814	<i>S</i>	Section (m^2)
815	<i>s</i>	Entropy ($J/kg/K$)
816	<i>Sr</i>	Vehicle to ram air speed ratio ($-$)
817	<i>T</i>	Temperature (K)
818	<i>t</i>	Time (s)
819	<i>V</i>	Volume (m^3)
820	<i>w</i>	Driving cycle weight ($-$)
821	<i>x</i>	Quality ($-$)
822	<i>z</i>	Spatial direction (m)

823 **Subscripts**

824	<i>air</i>	Air
825	<i>amb</i>	Ambient
826	<i>conv</i>	Convection
827	<i>cross</i>	Cross section
828	<i>eff</i>	Effective
829	<i>egr</i>	EGR gas
830	<i>EgrB</i>	EGR boiler
831	<i>eng</i>	Engine
832	<i>exh</i>	Exhaust gas
833	<i>ExhB</i>	Exhaust boiler
834	<i>exp</i>	Expander
835	<i>ext</i>	External wall
836	<i>f</i>	Working fluid
837	<i>fan</i>	Cooling fan
838	<i>g</i>	Gas
839	<i>in</i>	Inlet port
840	<i>int</i>	Internal wall
841	<i>max</i>	Maximum
842	<i>min</i>	Minimum
843	<i>out</i>	Outlet port

844	<i>pump</i>	Pump
845	<i>tank</i>	Tank
846	<i>v</i>	Valve
847	<i>vol</i>	Volumetric
848	<i>wall</i>	Heat exchanger wall