

**ORC SYSTEM OPTIMISATION: INNOVATIVE EXHAUST HEAT EXCHANGER,  
IN-ENGINE BLOCK HEAT EXCHANGER AND ZERO IMPACT CONDENSOR.**

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**ABSTRACT**

Waste Heat Recovery (WHR) systems, especially Organic Rankine Cycle (ORC) based, have the potential to significantly reduce fuel consumption. Current state of the art systems typically reduce the fuel consumption by 3-6%. However, the ORC systems still add excessive mass and complexity in order to become economically attractive. Focus on the optimization of a single aspect will not bring the fuel reduction percentages in the double digit numbers. In this paper we propose to combine various proposed system improvements in combination with a new condenser method. Implementation of the stable evaporation method developed by TNO (Dutch organisation for Applied Technological Research) in evaporative heat exchangers may improve their capacity by 30%. Next, additional heat sources may be used. The Ingenieurgesellschaft Auto und Verkehr (IAV) showed the opportunity to retrieve the heat from the engine itself as well. This innovation may largely benefit from integrating the stable evaporation method of TNO. Finally, TNO has found an innovative implementation of the condenser which is able to release tens of kW's with no penalty in the form of increased drag or fan power. This paper reasons, by combining all these aspects, a potential fuel reduction of 12-15% can be achieved.

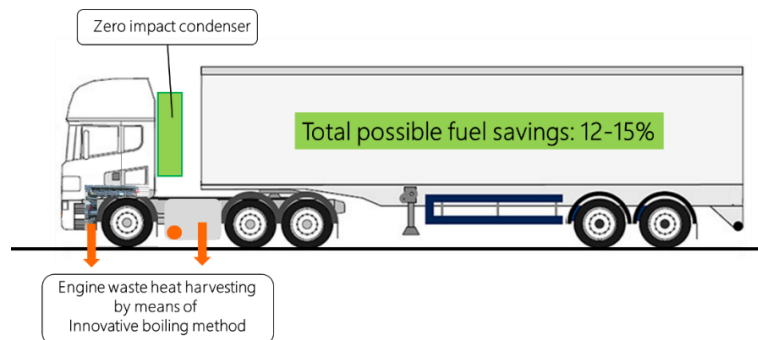


Figure 1 - Visual representation of optimizing WHR combination.

**1 INTRODUCTION**

In the last decade reduction of fuel consumption got a lot of attraction, for both the obvious economical reasons as well as environmental reasons. Less fuel consumption leads to less grams of CO<sub>2</sub> per 100km and the emission limits get more strict over time (White, 2017). The concept of re-using waste heat, so-called Waste Heat Recovery (WHR), is a viable path to reduce the fuel consumption. Organic Rankine Cycle (ORC) systems have the potential to regain significant amounts of thermal energy generated by the heavy duty combustion engine. Typically, state-of-the-art ORC systems achieve 3-6% reduction in fuel consumption (Marlok, 2019). However, current ORC systems still add excessive mass and complexity, and thus adding system investment costs and reliability challenges. In order to become economically attractive, the fuel reduction percentage should be in the double digits. The proposed

system improvements often focus on one aspect of the ORC. For example, the characteristics of the liquid pump are improved (Abergucci, 2019), or the heat transfer relations for the heat exchangers are being revised (Baldasso, 2019). Similarly, the quest for retrieving the optimal working fluid (Imre, 2018) can be regarded as a single focus system optimisation as well. In order to become economically attractive, all sources of heat need to be addressed and system induced losses need to be minimised.

The way forward to assemble an economically viable system is to combine various proposed system improvements. In the next section the foreseen improvements are discussed separately. First, the stable evaporation method developed by TNO, which allows a unique counterflow layout in evaporative Heat Exchangers, is discussed. Next, the usage of all available heat sources, as shown by IAV, to retrieve the heat from the engine itself as well is reviewed. Finally, we present the innovative implementation of the condenser by TNO, which is able to release tens of kW's with no penalty in the form of increased drag or fan power. The third section discusses how these three innovations can be brought together, and what total gain potentially can be achieved. Finally, the main conclusions will be given in section 4.

## 2 THREE INNOVATIONS TO COME TO A VIABLE WHR SYSTEM

In order to achieve double digit percentage fuel savings various separate innovations must be combined. In this chapter we discuss each of the innovations separately. We show the individual improvement potential of implementing the innovative flow boiling method, in-engine block energy harvesting and a zero impact condenser.

### 2.1 Innovative Flow Boiling Method

Boiling explosions, due to the explosive vapour bubble growth, in small diameter/ thin volume channels are identified as the origin of flow maldistributions leading to sub-optimal thermal system performance. Insights to properly control the explosive bubble growth are used to employ the phenomenon to our advantage (Rops, 2009). The innovative flow boiling method (Rops 2019) has led to the elimination of the large pressure fluctuations which is the root cause of the unwanted fluid backflow. Additionally, the fast liquid propulsion through the evaporator is annihilated, which provides enough time to fully evaporate the fluid. Therefore, the full thermal capacity of the evaporator can be used, even in a counter flow layout.

The benefit of a counter flow layout evaporator compared to a parallel flow layout evaporator is common knowledge in industry and is visualized in the schematic overview in Figure 2.

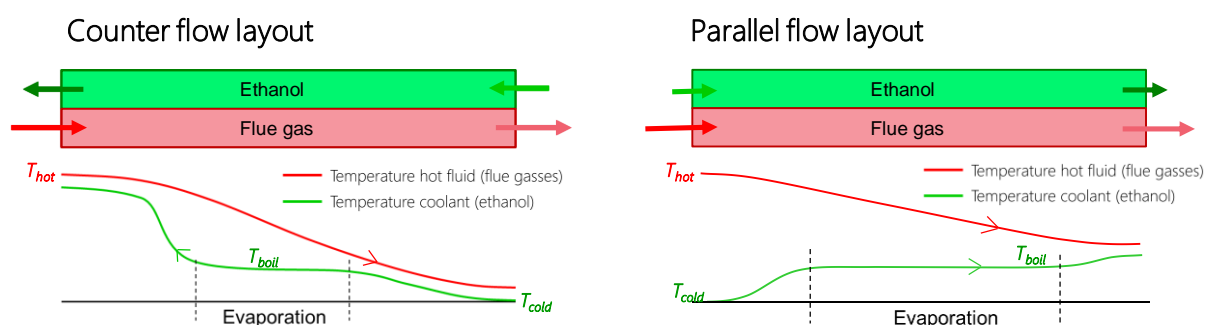


Figure 2 - Schematic overview evaporator layout and typical temperature profiles over the length of the evaporator. Left: counter flow layout. Right: parallel flow layout.

In a counter-flow evaporator the flue gasses can be cooled down to approximately the cold entrance temperature of the ethanol. While in a parallel-flow layout the exit temperature of the flue gasses must remain above the boiling temperature to ensure complete evaporation of the liquid. Of course, for both layouts holds that the flue gas energy below the boiling temperature can not be used to realize the phase change. However, in case of a counter-flow layout the energy contained by the flue gas temperatures below the ethanol boiling point can be used to warm up the liquid ethanol. This preheating tapping from the low quality heat is not possible in a parallel-flow configuration.

Currently many evaporators have implemented some kind of parallel-flow or hybrid cross-flow layout. This implementation is most likely due to the fact that at high wall temperatures evaporative flow instabilities occur more severely. Therefore, single phase heat transfer means are used to first reduce the wall temperature before the boiling starts. These methods therefore correspond in some way to a parallel-flow evaporator layout and will not be able to fully harvest all potentially available energy from the flue gasses.

Although the fact that a counterflow heat exchanger outperforms the parallel layout is commonly known, textbooks often only show this for single phase layouts. The theoretical benefit of a counter-flow *evaporator* over a parallel-flow *evaporator* is less often shown, but can be estimated using some assumptions. The schematic figure for the counter-flow evaporator shows that all heat above the boiling temperature can be used to achieve the phase change of the ethanol. However, if not enough heat is present in the flue gas mass flow to have the liquid ethanol to be warmed up to the boiling point, some of the heat from above the boiling temperature needs to be used for this as well. Therefore, the maximum ethanol mass flow to be evaporated using a counter-flow layout is:

$$\left\{ \begin{array}{l} \phi_{m,eth\_maxCounter} = \phi_{m,flue} \cdot \frac{c_{p,flue} (T_{hot} - T_{boil})}{h_{ev}} \quad \text{if } \phi_{m,eth} \leq \phi_{m,flue} \cdot \frac{c_{p,flue}}{c_{p,eth}} \quad (a) \\ \phi_{m,eth\_maxCounter} = \phi_{m,flue} \cdot \frac{c_{p,flue} (T_{hot} - T_{cold})}{h_{ev} + c_{p,eth} (T_{boil} - T_{cold})} \quad \text{if } \phi_{m,eth} > \phi_{m,flue} \cdot \frac{c_{p,flue}}{c_{p,eth}} \quad (b) \end{array} \right. \quad (1)$$

Similarly, for a parallel-flow layout the schematic figure shows that the heat of the flue gasses above the boiling temperature is used for both warming up the liquid as well as the phase change. This results in the following relation for the maximum ethanol mass flow to be evaporated using a parallel-flow layout:

$$\phi_{m,eth\_maxParallel} = \phi_{m,flue} \cdot \frac{c_{p,flue} (T_{hot} - T_{boil})}{h_{ev} + c_{p,eth} (T_{boil} - T_{cold})} \quad (2)$$

For demonstration purposes typical constant values are applied for the thermal heat capacity for both the ethanol and the flue gasses ( $c_{p,eth} = 3350 \text{ J/kgK}$ ,  $c_{p,flue} = 1160 \text{ J/kgK}$ ) and setting the heat of evaporation,  $h_{ev, ethanol}$ , to a constant value of  $588 \text{ kJ/kg}$  the maximum evaporation mass flows can be estimated. Figure 3 shows the theoretical maximum ethanol mass flow to be evaporated using a counter flow layout and a parallel flow layout assuming a  $0.1 \text{ kg/s}$  flue gas mass flow.

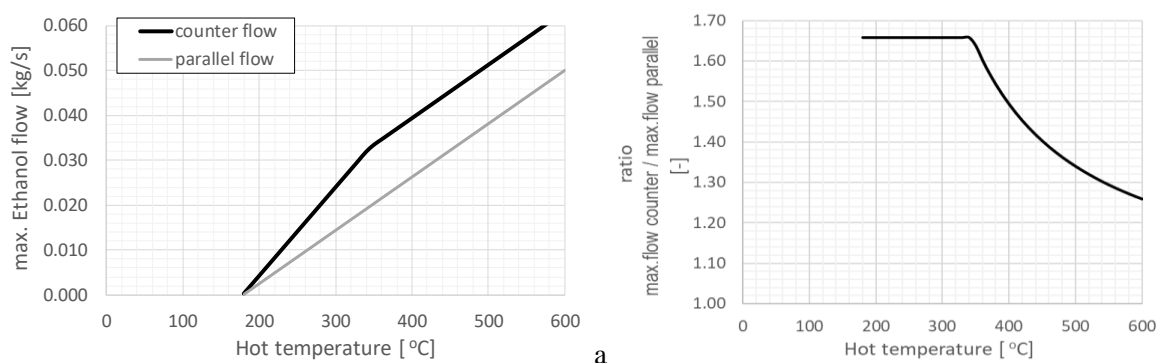


Figure 3 - Theoretical maximum Ethanol evaporation mass flow as a function of hot flue gas temperature,  $T_{hot}$ . Boiling temperature,  $T_{boil} = 179 \text{ °C}$  (20bar), cold Ethanol inlet temperature,  $T_{cold} = 70 \text{ °C}$ , and a flue gas mass flow of  $0.1 \text{ kg/s}$ .

The left hand graph (Figure 3a) shows that the counter-flow evaporator has the potential to evaporate more ethanol per 100gram flue gasses at the same hot entry temperature than the parallel-flow

evaporator. The right hand graph (Figure 3b) shows the ratio of the maximum evaporative mass flows. For entry temperatures not too far above the boiling temperature the counter-flow evaporator may outperform the parallel-flow evaporator by more than 65% (equation 1a over 2). However, as the maximum ethanol mass flow increases, the flue gasses with a temperature below the boiling temperature do not contain enough energy anymore to warm up the liquid ethanol to its boiling temperature. Therefore, part of the higher quality energy needs to be used for warming up the liquid. Figure 3 shows that the absolute mass flow increase remains, however the relative improvement reduces to about 35% at 500 °C (equation 1b over 2). One should note that the thermal properties are temperature dependent adding additional effects. For example, the higher heat capacity of the flue gasses at higher temperatures implies that relatively less energy is available at lower temperatures. Thus the ratio, as shown in the right hand graph (Figure 3b), will start to decrease at lower temperatures.

## 2.2 In-Engine Block Energy Harvesting

In order to perform better on fuel economy IAV proposes to include the engine heat as a source of waste heat (Dingel, 2019) as well. So far this potential big energy source has not been in the focus of the ORC system. Currently, this waste energy is dumped into the outside air via the radiator using conventional engine cooling. The coolant in the conventional cooling system is at a low temperature, which negatively impacts the recuperation efficiency. However, incorporating the engine in a high temperature phase change ORC system promises several advantages: more heat recuperation, less wall heat losses and higher exhaust gas temperatures, which is beneficial for the exhaust gas treatment (Leahu, 2018).

In their investigations (eg. Ambrosius, 2017) IAV designed new high pressure (10-60 bar) cooling ducts to overcome the main disadvantages of the flow boiling phenomena and to tune the boiling temperature to the desired value. However, practical difficulties are still encountered preventing stable and robust operation. In order to estimate the fuel reduction a simulation model has been made combining the engine combustion model, the ORC model and the phase change cooling physics.

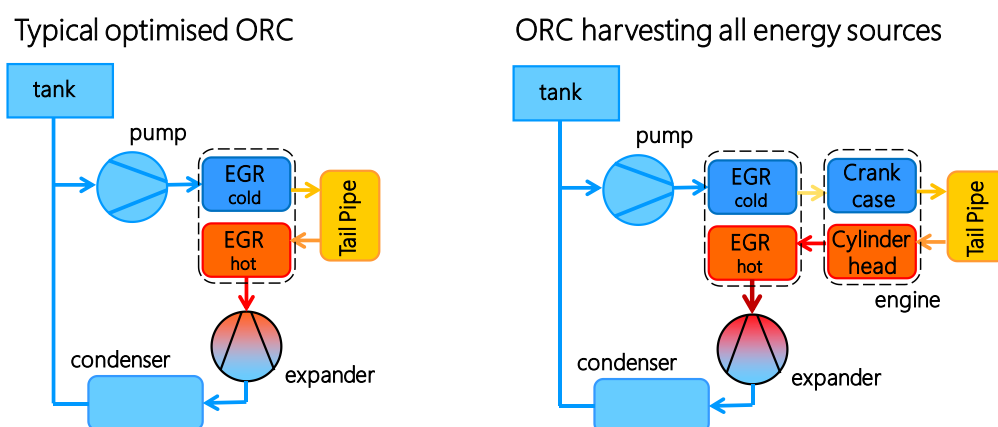


Figure 4 - ORC schematics -after IAV publication (Dingel, 2019)- Left: typical optimized ORC system. Right: ORC system including the engine heat as energy source.

The ORC-system may optimise the output of the combustion system in two ways: lowering the flue gas output temperature of the Exhaust Gas Recirculation (EGR), and increasing the superheat of the ethanol. The schematics on the left of Figure 4 show that these two conditions can be met by separating the EGR heat exchanger in two parts: a cold one at the flue gas exit section and a hot one at the flue gas entry section to boost the superheating of the ethanol. The right schematics of Figure 4 show the implementation with the engine included as well.

The simulations of IAV indicate significant fuel reductions on introducing the ORC to the drive system. For three operating points (OP) with different engine speed ( $n$ ) and torque ( $T_{or}$ ) (OP1:  $n=1100\text{rpm}$  &  $\text{tor}=1000\text{Nm}$ ; OP2:  $n=1100\text{rpm}$  &  $\text{tor}=1500\text{Nm}$ ; OP3:  $n=1300\text{rpm}$  &  $\text{tor}=1800\text{Nm}$ ) the potential fuel gain is given in Figure 5.

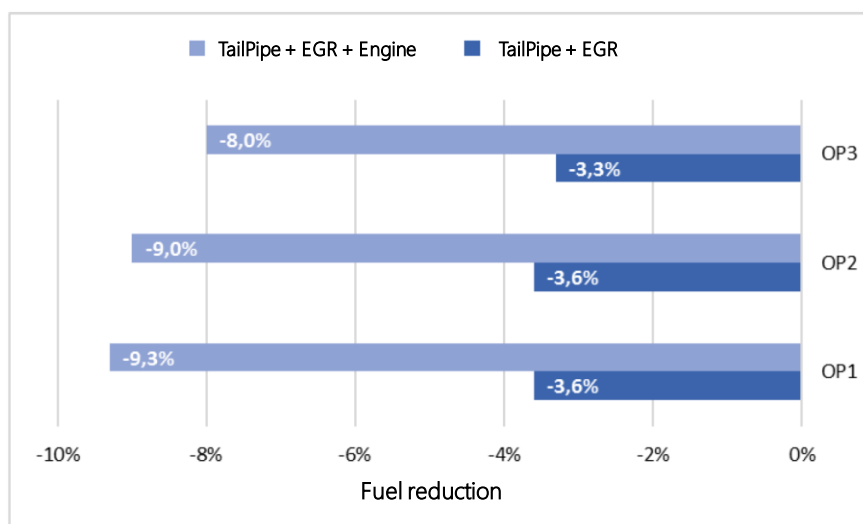


Figure 5 - IAV figure: fuel reduction by ORC for 3 operating points with only TailPipe and EGR included, and ORC with TailPipe, EGR and Engine included (Dingel, 2019).

The above figure shows that the current typical ORC system could realise fuel reduction values between 3% and 4%. However, including the engine heat in the ORC system those values will be lifted to about 9%. The main reasons indicated for this increase are: First, the engine heat is added to the ORC at a high temperature level. Next, the exhaust heat is extracted at a higher temperature with a larger fluid mass flow. Finally, the engine power is increased due to lower wall losses and reduced friction. Therefore, replacing the components of the engine cooling system for the components needed for the ORC system will generate the 9% fuel reductions with similar hardware components as a drive system without ORC.

### 2.3 Zero Impact Condenser

Further fuel reductions can be obtained by improving the overall economy of the vehicle. In the end significant amounts of waste heat need to be dumped into the air. If no ORC is applied the energy of the flue gasses is simply released in the air by the hot flue gasses leaving the tailpipe. The waste heat of the engine is released into the air via the radiator. However, the more energy needs to be disposed via the radiator, the more air needs to pass through. This is either realised by an internal (visco-slip) fan, or ram air intake. Both realisations have a negative effect on the fuel consumption: the fan is extra weight and needs significant (engine) power, and the ram air intake deteriorates the aerodynamics of the vehicle. Additionally, if more heat needs to be disposed, typically the weight of the heat disposing element goes up, since the heat exchanging area must become larger. The quest would be to dispose the unused heat present in the ORC, while having no negative impact on the above mentioned aspects.

In order to take away the cost of the reduced aerodynamics or the engine power needed to drive the fan, TNO has investigated the possibility to realise such a zero impact condenser. Of course, it is obvious that this zero impact condenser can function as any heat disposing element. The key element of the zero impact condenser is the creation of enough buoyancy such that natural convection allows the air to pick up enough energy. In order to achieve this goal, the zero impact condenser needs to be high enough, and the spacing between the heat exchanging parts should be tuned that not too much flow resistance is encountered. Next, the total heat exchanging area should be several square metres (Rops, 2021). Finally, it should be positioned such that the air flows entering and exiting the condenser do not deteriorate the aerodynamics of the vehicle. Figure 6a shows a possible implementation of such a zero impact condenser. Computational Fluid Dynamics (CFD) simulations were conducted to validate the hand calculations. A 6 million cells mesh was created to model a simplified geometry comprising a stylistic truck-trailer combination and its surroundings. The buoyancy space between the plates was modelled using 30 cells between the plates. A grid refinement did not show significant (<5%) deviation. Ideal gas law was applied for modelling the air. In case of the truck standing still, CFD calculations show that the design functions as expected: the hot air rises and a plume is ejected at the top removing 8.5 kW.

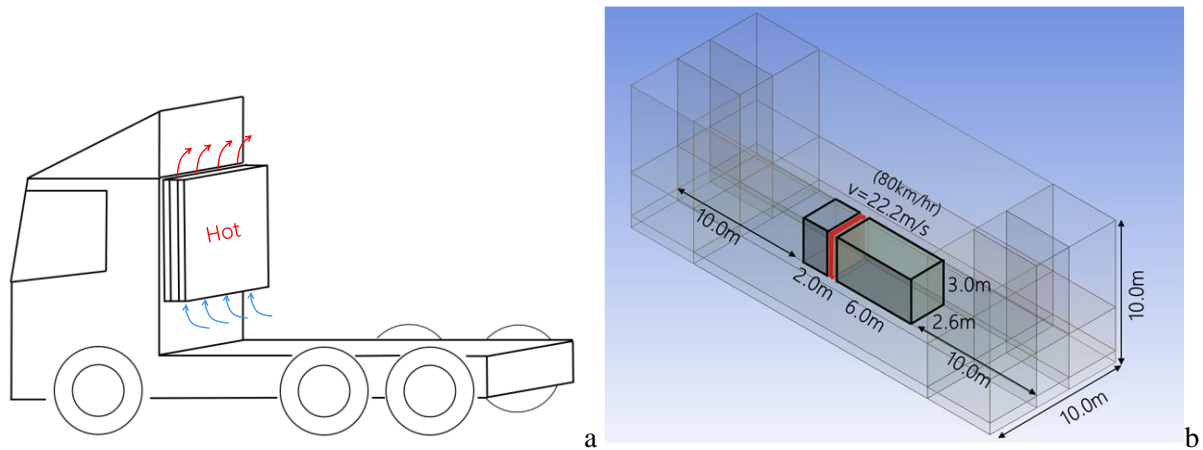


Figure 6 - Schematic of a zero impact condenser. Left (a): Truck schematic adapted from Rops, 2021. Right (b): Box simplified geometry of Truck-trailer combination used for CFD investigation (condenser location in red).

Figure 6b shows the simplified geometry used to investigate whether sufficient buoyancy drag is created between the plates. Especially due to the driving, counteracting pressures may be introduced which can cancel the envisioned natural convection. Next to this, a comparison regarding the pressure drop over the vehicle is made to ensure the drag of the vehicle is not affected due to the hot air mixing with the dragged air around the vehicle. We are aware that truck-trailer combinations have much better aerodynamics and have many features such as windshields in place. However, for reasons of comparison we calculated for the same geometry the drag (pressure drop) over the combination with a hot condenser and an ambient temperature condenser. The condensation temperature of the ethanol is about  $78\text{ }^{\circ}\text{C}$ . Therefore, it may be expected that the condenser walls are at a similar temperature, although slightly below. So, the condenser walls are set at  $77\text{ }^{\circ}\text{C}$ . The main results are shown in Figure 7.

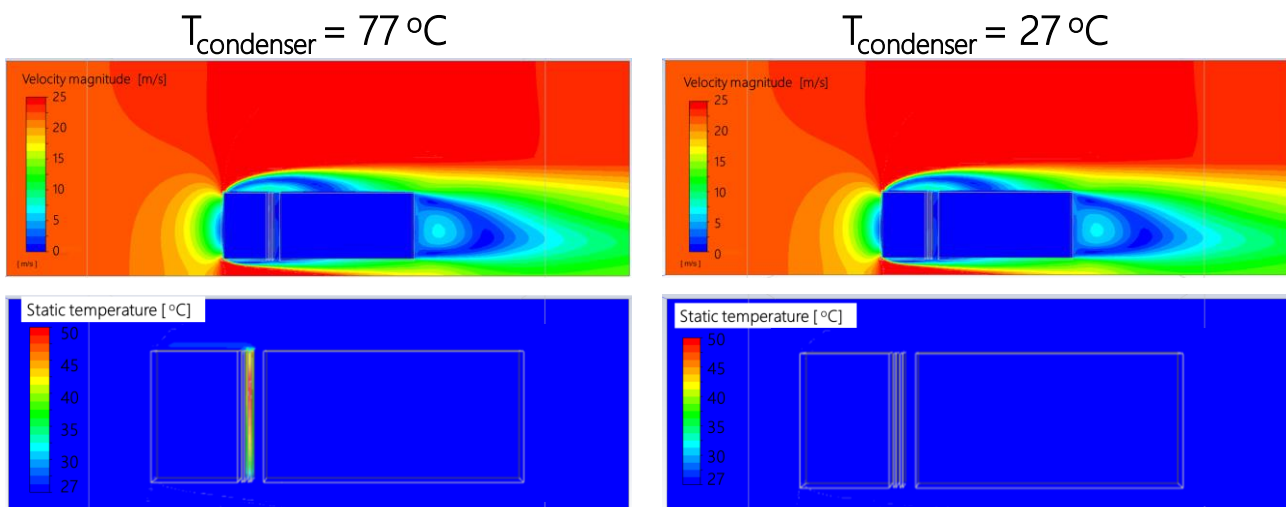


Figure 7 - CFD results on the mid plane. Left column: hot ( $77\text{ }^{\circ}\text{C}$ ) condenser case. Right column: ambient ( $27\text{ }^{\circ}\text{C}$ ) condenser case. Top: velocity magnitudes (relative to truck-trailer combination) Bottom: temperature around the truck-trailer combination.

The ambient temperature is set  $50\text{ }^{\circ}\text{C}$  below the condenser temperature to  $27\text{ }^{\circ}\text{C}$ . The above figures show that from a flow perspective little differences may be observed. Only at the location of the condenser the air temperature is significantly higher. This increased temperature induces enough buoyancy to get air rising and eject a hot plume. The hot air is immediately mixed with the recirculation on top of the truck cabin. This hot plume entrainment is not changing the flow patterns around the truck-trailer combination. Since we set in our simulations all velocities relative to the truck-trailer combination we can calculate the total pressure drop of the combination by subtracting the outlet pressure from the inlet pressure. Any difference in friction will affect this pressure drop and can be assigned to an altered flow pattern. In case of ambient conditions the pressure drop is calculated to be



29.0 Pa and in case of the hot condenser the pressure drop is equal: 29.0 Pa. The extracted heat disposed into the air is 21.6 kW. This corresponds to 0.43 kW/°C.

The simplified simulations show that the wall temperature difference of 50 °C is enough to drive the air despite the pressure field present around the truck and creates a buoyant flow. This flow can dispose significant amounts of heat into the air without increasing the drag of the combination.

### 3 OPTIMISING ORC SYSTEM BY COMBINING THREE INNOVATIONS

The three innovations: the innovative flow boiling method, the in-engine block energy harvesting and the zero impact condenser can be combined, enforcing each other. Adding the engine heat to the ORC allows to regain part of this energy instead of blowing it into the air via the tail pipe. However, making the engine cooling part of the ORC, and not being a separate cooling system, sets additional robustness requirements onto the entire ORC-system. Implementation of the inherently stable boiling method offers this robustness and opens the path towards this innovative ORC-system realisation.

In the previous chapter the IAV study has shown that the inclusion of the engine waste heat into the ORC has a potential fuel reduction of 8 - 9.3%. The underlying simulations assume the current typical heat exchanger layout. The IAV innovation is much suited to implement the innovative flow boiling method of TNO. Next to adding robustness, this may also increase the heat exchange by 20-40%, due to the counter-flow layout improvement as shown in section 2.1. Since the heat exchange, ethanol mass flow and the expander output power are linearly dependent (eg. Guo, 2018), the fuel reduction may be increased by about a factor 1.3: 10.4 – 12.1%.

Adding the zero impact condenser will add robustness to the ORC system as well. The system contains a self-regulating aspect since more buoyancy is generated in the air when the more energy needs to be disposed. The heat drives the energy disposing system instead of an engine fan or ram air. Several studies discuss the effect of the air intake on the fuel consumption (eg. Larson 2013). Typical values vary between 2 - 3% fuel reduction. Therefore, it is reasonable to assume that combining all three innovations may increase the fuel reductions up to 12 - 15%. Additionally other heat dissipation modules could be integrated in the zero impact condenser, like an AC condenser or even an engine coolant radiator which could increase the aerodynamic efficiency even more. This is visually represented in Figure 8.

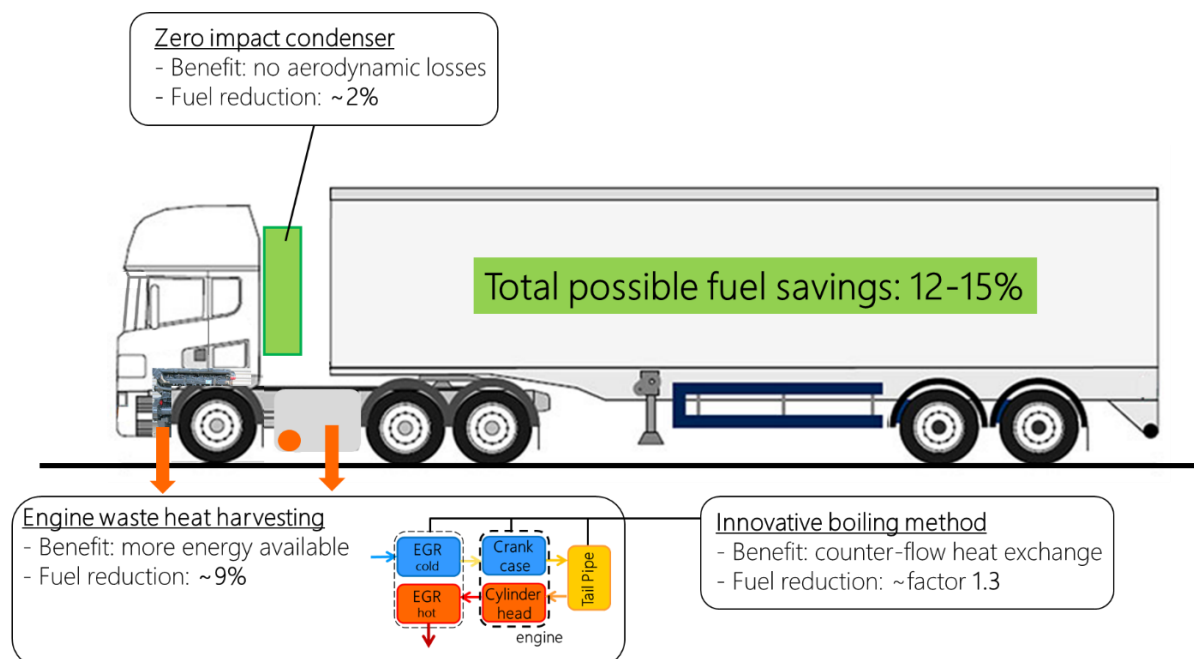


Figure 8 - Visual representation of fuel savings potential.

## 4 CONCLUSIONS

ORC optimization should not focus on a single system item but combine various system improvements which enhance each other. Section 2.1 shows that implementation of the stable evaporation method in evaporative heat exchangers may improve their capacity by 20-40% due to their counter flow layout. Next, additional heat sources may be used to retrieve the heat from the engine itself as well. This could lead to fuel reductions between 8 - 9.3%, as shown in section 2.2. Finally, it is discussed that implementation of the zero impact condenser, which is able to release tens of kW's with no penalty in the form of increased drag or fan power, could add 2 - 3%. It is argued that by combining all these aspects may bring fuel reductions of 12-15% within reach.

## NOMENCLATURE

$c_p$	heat capacity	(J/kgK)	$\phi_m$	mass flow	(kg/s)
$h_{ev}$	heat of evaporation	(J/kg)			
$n$	engine speed	(rpm)		<b>Subscript</b>	
$T$	temperature	(°C)	flue	flue gas	
$tor$	torque	(Nm)	eth	ethanol	

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