INVESTIGATION OF AN ORC SYSTEM WITH INTEGRATED PHASE CHANGE ENGINE COOLING

Oliver Dingel*, Daniel Luederitz, Thomas Arnold

IAV GmbH
Kauffahrtei 45, 09120 Chemnitz, Germany
oliver.dingel@iav.de
daniel.luederitz@iav.de
thomas.arnold@iav.de

* Corresponding Author

ABSTRACT

Conventional engine cooling systems, which are based on the use of a liquid coolant, are normally controlled by a thermostat that keeps the engine out temperature on a constant level between 85 and 100°C. The cooling heat of the engine is rejected to the ambient by a radiator without making further use of it. Only in winter, the dissipated cooling energy is partly used for cabin heating.
When replacing the conventional cooling system of an engine by a dedicated phase-change-cooling (PCC) system and substituting the coolant by an Organic Rankine Cycle (ORC) working fluid, the engine cooling power can be integrated into the exhaust gas and EGR heat recuperating circuit. In a detailed simulation study based on GT-power and MatLab/Simulink models, IAV has investigated the fuel saving potentials of such a combination in a long-haul truck application.
The investigation was carried out at three characteristic engine operating points. Ethanol was used as working and cooling fluid in a single circuit. By the help of pressure sweeps, the overall optimum conditions in each operating point could be determined. By a detailed loss split analysis, the obtained results were examined in detail. As wall temperatures have impact upon emissions formation too, the influence of PCC upon NOx-emissions was evaluated in addition.
As a result it can be said, that the potential to reduce fuel consumption by an ORC system using energy from exhaust gas and EGR heat exchangers only, can be more than doubled when integrating the PCC into the circuit.

1. INTRODUCTION

The CO_2-emissions of on-road vehicles have to be reduced significantly in the upcoming years in order to achieve overall climate goals (European Commission, 2018). ORC applications are an interesting option to contribute to these goals. Due to the high fuel consumption and mileage of long haul trucks, they are of special interest for ORC systems since a long time. The upcoming CO_2 limits for these vehicles in the EU are an additional driver for this combination.
The main reason why ORC systems have not been introduced into the market so far is the very ambitious goal of the shipping companies to achieve a return of invest of these systems within two years. To reach this goal, future systems will have to be either cheaper in production or perform better fuel economy. IAV found an approach to achieve the latter one.
Most ORC system approaches introduced as models or prototypes for long haul truck applications make typically use of wasted exhaust heat only (Bettoya, *et al*., 2016, Glensvig, 2017, Marlok *et al*., 2018) or exhaust and EGR heat (Bettoya, *et al*., 2016, Lutz *et al*., 2012, Xu et al., 2016) . Another big source of wasted heat, the engine cooling was not in the focus so far, as this energy is only on much lower temperature level available, which in turn has negative impact upon the recuperation efficiency. This drawback can be overcome by replacing the conventional liquid convection cooling by a high temperature phase change cooling system. As this system is integrated into the exhaust heat ORC circuit, it promises several advantages at the same time: more heat recuperation, less wall heat losses,

and higher exhaust gas temperatures. Due to the integration in the ORC circuit, it does not need additional components but replaces some of them, which at the end leads to a much better performance to cost ratio.

2. PHASE CHANGE COOLING

2.1 Basics and history of PCC

Conventional convection cooling systems are designed for the maximum amount of heat dissipation under full load conditions. High mass flows provided by a coolant pump being directly connected to the engine ensure safe component temperatures even at a very low coolant temperature difference. In order to avoid adversely low wall temperatures at part load as well as during engine warm-up, a thermostat controls the mass flow through the radiator and therefore the engine out coolant temperature. A specific rise of wall heat temperatures at part load conditions or during engine warm-up is limited to a small range when coolant evaporation should be avoided. In order to overcome this restriction, several evaporative or phase change cooling systems have been investigated in the past. Müller *et al.* (1995) described an approach based on an open low-pressure system layout using a conventional cooling jacket. However, due to various unsolved problems like volume increase or hot spots caused by film boiling, these PCC systems never came to series production.

2.2 New pressurized PCC system with enforced boiling

A new approach, based on a high-pressure layout with completely new designed cooling ducts is the basis for this report. In Ambrosius *et al.* (2017) the main features and the underlying physics of this approach is described in detail. Due to the high pressure of the system (10 – 60 bar) and adequate flow velocities of the cooling fluid, the main disadvantages of former systems could be avoided. Furthermore it could be shown, that by varying the pressure and therefore the evaporation temperature of the coolant, the wall heat temperatures can be adjusted in a wide range independent from engine load and speed. That allows reducing wall heat losses and increasing exhaust gas temperatures by higher coolant temperatures at part load without endangering the thermal health of the engine under full load conditions. Another positive effect is the reduced power demand of the coolant pump due to the significantly reduced mass flow. This effect overcompensates the enhanced power demand to reach the high pressure level of the system and it offers the opportunity to combine it with an ORC based exhaust heat recovery system.

In opposite to the approach of Ziviani *et al.* (2018) the fluid used is ethanol and it evaporates not only in the EGR and exhaust gas heat exchanger (HX) but also in the engine. Arnold *et al.* (2018) described the basic approach and the idea behind that layout, which is called Advanced Thermal Management (ATM). The author shows results from an ATM set-up using a 1.4-liter T-GDI engine equipped with a cylinder head modified for PCC. It could be proved that coolant pressures of up to 60 bar and temperatures of up to 275°C did not lead to critical component temperatures in the cast iron cylinder head with the adapted cooling duct design.

As these results looked very promising in a next step it should be investigated how much fuel consumption benefit can be achieved on a heavy-duty truck engine equipped with ATM. That was done using a suitable simulation environment.

3. SIMULATION MODEL

The entire simulation model for the present study consists of an engine model, a PCC-model and an ORC model, where the latter ones are build up in MatLab/Simulink, whereas GT-power, a wide spread commercial simulation software for internal combustion engines (ICE) from Gamma Technologies; was used for the engine model. Figure 1 shows the schematic of the simulation model and what values had been exchanged between the models. In order to handle this value exchange in an automated way and to carry out iterations to achieve a given set value a dedicated MatLab/Simulink based tool had been created.

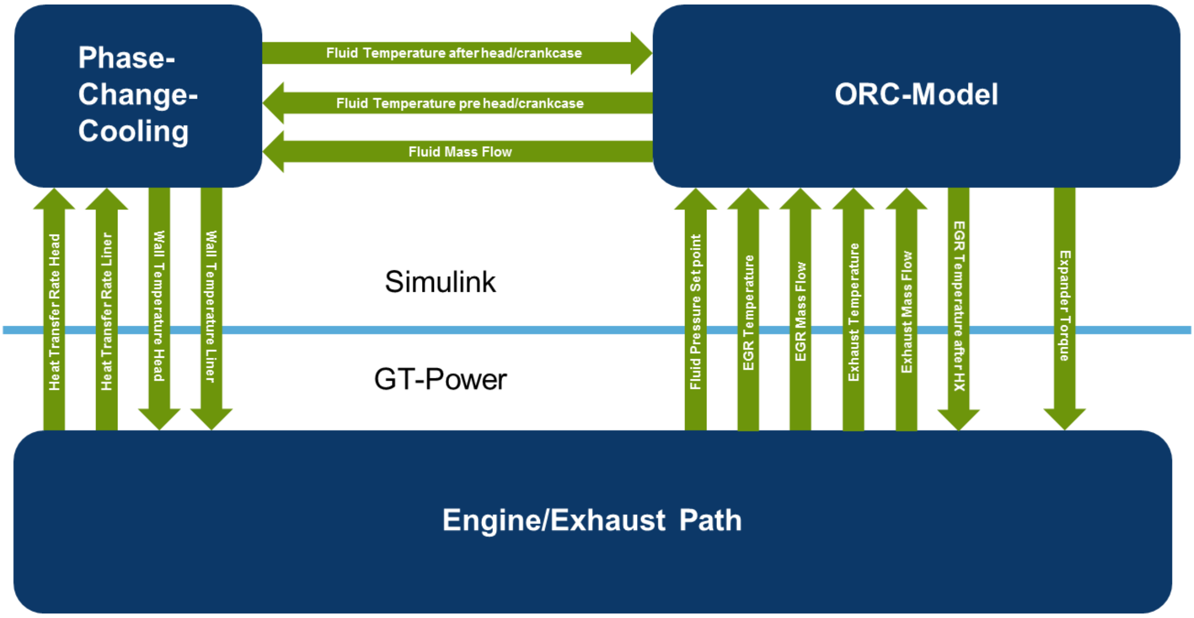

Figure 1: Schematic configuration of the simulation model

3.1 Engine Model

The engine model represents a typical inline 6-cylinder heavy-duty truck engine with a displacement of around 11 liter. It fulfills EURO VI emissions regulations and is equipped with a single stage turbo-charger, a charge air cooler and a cooled EGR system.

In order to evaluate the impact of the coolant temperature upon wall temperature and wall heat transfer the "FECylinderStructure" object of GT power is used for the cylinder wall modeling. The "HeatExchangerConn" object represents the heat transfer of the original EGR cooler. When replacing the EGR cooler by an EGR-HX, the model was represented in the ORC-simulation environment.

By using the basic Chen-Flynn friction model the impact of the wall temperature upon piston and piston ring friction could not be modeled in an appropriate way. Therefor the Chen-Flynn model was extended by the Schwarzmeier-Reulein approach.

Based on measurement results from the dyno the model was calibrated at three representative steady state operating points defined by engine speed and torque:

- OP1: n = 1100 rpm; T = 1000 Nm
- OP2: n = 1100 rpm; T = 1500 Nm
- OP3: n = 1300 rpm; T = 1800 Nm

The matching of the simulated and measured results regarding fuel and air mass flow, temperatures, pressures and NOx emissions was satisfying for the purpose of this project. A detailed loss split of the reference model was basis for the analyses of the subsequent results gained by the use of ORC and PCC.

3.2 ORC Model

ORC systems, such as described in Reiche and Galuppo (2017) currently offer suitable preconditions for mobile use of waste heat recuperation. The results of the present paper are based on calculations using ethanol as working fluid. A detailed study in the framework of the FVV (Research Association for Combustion Engines) project "Ideal Rankine Fluid" showed ethanol to be one of the most suitable working fluids, thanks to its specific combination of properties (Preissinger and Schwoebel 2016).

The ORC system includes a heat exchanger downstream of the exhaust gas aftertreatment, an EGR heat exchanger and a piston expander. Both heat exchanger models and the reference piston expander model represent prototypes designed and manufactured by IAV. They have been validated by dyno measurements of the corresponding prototype parts. The piston expander model later on was adapted to different heat and mass flow demands. For more information, see chapter 3.4.

Figure 2 shows the topology of the ORC circuit where all components are arranged in serial configuration. As a special feature, the EGR heat exchanger is split in two parts, which provides benefits for the EGR and the fluid side. After being pressurized by the pump (2), the fluid enters the outlet side of the two-part EGR-HX. As the fluid mass flow is not split up like in ORC-systems with parallel arrangement of EGR and tailpipe-HX and the liquid fluid has a temperature t = 70°C, the recirculated exhaust gas leaves the HX at a low temperature, which is beneficial for a low NOx formation. After being evaporated in the tailpipe-HX (3c), located downstream of the exhaust gas aftertreatment (EAT) box, the fluid enters the inlet side of the EGR-HX. Here the exhaust gas has the highest temperature so that the fluid can be superheated before entering the expansion machine (3f). This in turn is beneficial for the efficiency of the piston expander.

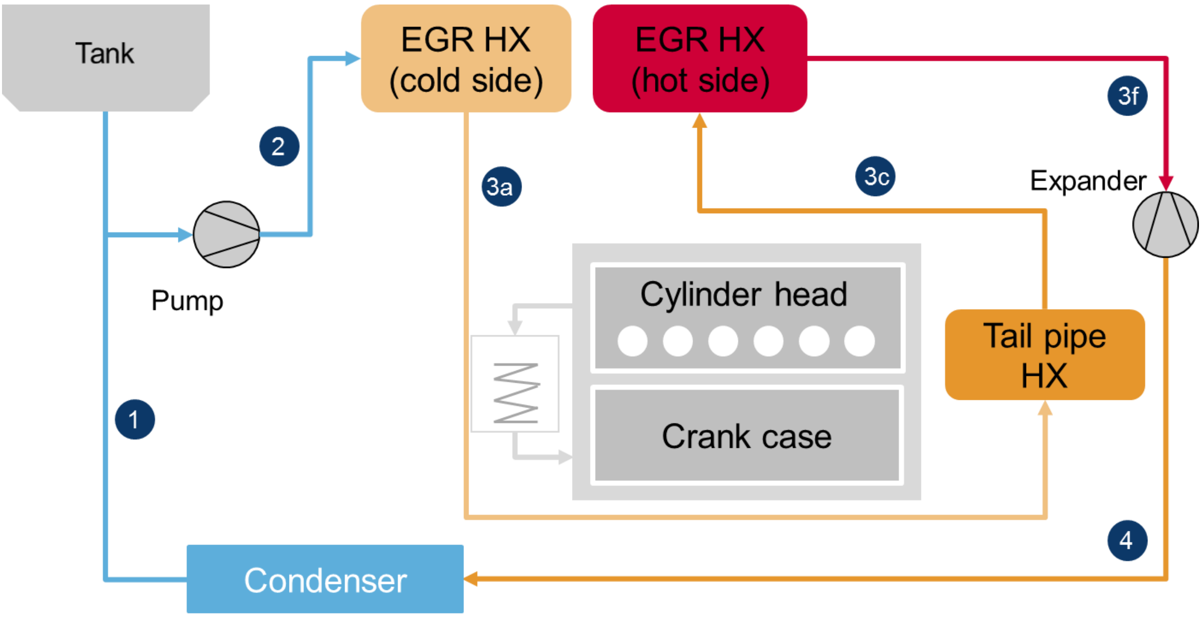

Figure 2: Topology of ORC circuit w/o PCC

3.3 PCC Model

Main task of the PCC model is the evaluation of the heat transfer on the coolant side in the engine considering the flow velocity and the condition of the fluid including the vapor quality in the two-phase area. Together with the heat load on the combustion chamber side calculated in the GT-power engine model the wall temperatures of liner and cylinder head can be calculated.

Figure 3 shows the topology of the ORC circuit with the phase change cooled engine integrated. As there is no additional cooling circuit necessary, there is also no radiator and no water pump in place. The high-pressure pump and the condenser of the ORC circuit replace their purposes. It can be seen that engine cooling is split between crankcase and cylinderhead. After having left the cold outlet side of the EGR-HX (3a), the fluid first enters the crankcase of the engine and then the tailpipe-HX (3b). In the operating points investigated in this study, the fluid is in the two-phase area when leaving the crankcase. After absorbing heat from the exhaust gas in the tailpipe-HX (3c), the fluid enters the cylinder head where it first flows to the combustion roofs of the cylinders and then to the exhaust ports. A connection to the inlet ports is avoided in order not to heat up intake air and thereby increase NOx emissions. After leaving the cylinder head (3d) a throttle valve controls the pressure and thereby the evaporation temperature of the fluid. After the throttle valve (3e), the fluid superheats in the hot inlet side of the EGR-HX and then enters into the piston expander (3f).

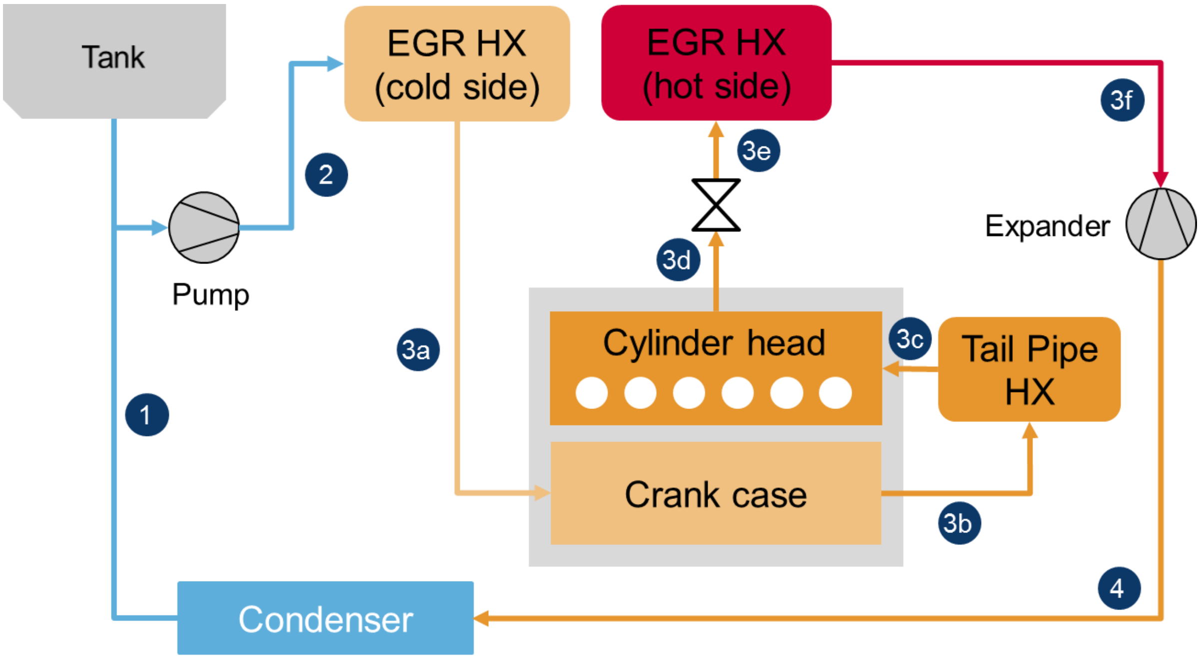

Figure 3: Topology of ORC circuit with PCC

3.4 Boundary conditions and controls strategy

For all simulations with ORC and PCC system it was assumed that the working fluid has a temperature of t = 70°C in between condenser and EGR-HX (1,2).

All heat exchangers work according to the counter-flow principle. After having left the last HX and before entering the piston expander (3f) the fluid should always have a superheated temperature of t = 295°C in order to ensure best expander efficiency. This could be achieved by controlling the fluid mass flow by pump speed accordingly.

In order to find out the optimum conditions for best fuel consumption the pressure of the working fluid and thereby the evaporation temperature of the fluid had been varied in both, the ORC and the PCC system.

When looking at the ORC system only, there is a trade-off between a high expander efficiency at high pressures on one side and high heat exchanger efficiencies at low pressures and temperatures on the other side. In operating point 3, the pressure was varied by scaling the expander in terms of size and speed. Here a maximum expander power output was achieved at a pressure of 50 bar. For OP 1 and OP2 the expander size and the transmission ratio to the engine speed was kept constant which lead to the according pressure levels shown in chapter 4.1.

When adding the PCC to the engine the additional effects upon wall heat temperature and losses as well as upon exhaust gas temperature have impact upon the best pressure set value. For this reason, a pressure sweep using the throttle valve after cylinder head was carried out in a range of 20 to 50 bar at each operating point. Expander size and speed then was scaled in order to achieve a two bar lower pressure level upstream expander.

Recuperated power from the expansion machine can be rejected to the engine on either a mechanical or an electrical way as shown in Treutler *et al.* (2017). In the present study, the piston expander is mechanically coupled to the crankshaft of the engine, so expander power and torque contributes directly to the system performance consisting of engine and expander power.

BTE_{exp} here is defined as the ratio between mechanical power of the expander and the heat power of the fluid upstream expander. The heat power in this case is related to the reference enthalpy of the fluid at 1 bar and 20°C.

The values for air mass flow and air/fuel ratio as well as charge air pressure and EGR rate of the reference engine were kept constant when adding the ORC system and PCC to the engine. Therefore, the absolute fuel consumption also was kept constant. Thus, the reduction of BSFC results from the higher system power output caused by the additional expander power and – in case of PCC - increased engine power output due to a better BTE.

4. RESULTS

4.1 Exhaust heat recovery by ORC only

Using the described ORC system only the BSFC of the engine can be reduced by 3.6% at OP1 and OP2 and 3.3% at OP3 (see table 1). These are quite common values for such a system. As exhaust gas temperatures rise when load increases, the optimal pressure also rises from 30 bar at OP1 to 40 bar at OP2 and 50 bar at OP3. The BTE of the expander is in a small range between 11.3 and 11.5%. This leads to expander power outputs of 4.3 kW (OP1), 6.5 kW (OP2) and 8.7 kW (OP3). As there is no impact of the ORC system upon engine friction or wall heat losses, the expander power and torque is the only contribution to rising system torque and thereby to the reduced BSFC values.
As said it was also investigated in how far NOx raw emissions are effected by the different waste heat recovery approaches. In case of the ORC system, the only impact can result from a different temperature of the EGR when entering the intake manifold. Whereas an EGR cooler lowers the temperature at the reference engine, a split EGR-HX takes over this task when introducing an ORC system.
The results in table 1 show, that only in OP1, where the EGR temperature raised by 20°C, a moderate rise in NOx emissions of 5.4% can be observed. In both other operating points, no noteworthy change can be reported.

Table 1: Results relative to reference engine from use of ORC only

Value	Unit	OP1	OP2	OP3
Temp. upstream expander	°C	295		
Pressure upstream expander	bar	30	40	50
Fluid mass flow	g/s	28	41	55
Heat power upstr. expander	kW	40	57	75
Expander power	kW	4.3	6.5	8.7
BTE expander	%	11.5	11.3	11.5
Δsystem torque	Nm	37	56	60
ΔBSFC	**%**	**3.6**	**3.6**	**3.3**
ΔTemp. exhaust TP-HX	K	152	158	126
ΔTemp. EGR downstr. HX	°C	20	10	0
ΔNOx raw emissions	%	5.4	0	0

4.2 Combined recovery of exhaust and engine cooling heat

Looking to the change of BSFC in table 2, a significant increase can be stated when adding PCC to the ORC system. The highest overall value of 9.3% could be achieved at OP1 followed by 9.0% in OP2 and 8.0% in OP3.
When analyzing these results, it can be seen, that the additional recovery of cooling heat is not the only contribution of PCC for these remarkable results. The higher wall temperatures of cylinder liner and head lead to increased exhaust gas temperatures and thereby improved heat transfer in the exhaust gas and EGR-HX. At the same time, reduced wall heat losses to the coolant result in a rising engine BTE. The heat power from cylinder head and block increases the overall heat power in the fluid and thereby the expander power significantly. As the additional heat load requires a higher fluid mass flow also the heat power that is rejected to the fluid in both ORC-HX exceeds the values from the use of the ORC

system only in all operating points. This is also reflected in the increased exhaust gas temperature difference across the tailpipe-HX.
The rise in system power additionally benefits from the higher engine torque to the amount of 22 Nm to 30 Nm depending on the operating point. This is mainly a result from reduced wall heat losses resulting in a better indicated BTE_{ICE} of the engine. Reduced friction only plays a minor role in this context.
When looking at the NOx raw emissions shown in table 2, it can be seen that they rise in a range of 5.2% to 15%. As this obviously does not result from higher EGR out temperatures, the reason must be the higher cylinder wall temperatures that lead to higher combustion temperatures and therefor support NOx formation. On the other hand, the increased exhaust temperatures have positive impact upon the conversion rate of the SCR catalyst, leading to lower tailpipe NOx emissions. As this could not be simulated in this study, it has to be verified in an experimental set-up.

Table 2: Results relative to reference engine from use of ORC + PCC

Value	Unit	OP1	OP2	OP3
Temp. upstream expander	°C	295		
Pressure upstream expander	bar	46	44	47
Fluid mass flow	g/s	57	85	118
ORC heat power from TP and EGR-HX	kW	46	64	86
PCC heat power from cylinder head and block	kW	25	41	60
Heat power upstr. expander	kW	80	119	164
Expander power	kW	9.2	13.7	17.9
BTE expander	%	11.5	11.6	10.9
Δengine torque	Nm	22	30	25
Δsystem torque	Nm	102	149	156
ΔBSFC	**%**	**9.3**	**9.0**	**8.0**
ΔTemp. exhaust upstream TP-HX to reference	K	45	39	36
ΔTemp. exhaust TP-HX	K	200	207	177
ΔTemp. EGR downstr. HX	°C	2.6	-2.8	-16
ΔNOx raw emissions	%	15.0	5.2	11.4

5. CONCLUSIONS

The introduction of PCC in addition to an ORC system increases the BSFC benefit about 2.5 times up to a range between 8.0% and 9.3%. Main reasons for this remarkable improvement are:

- The entire cooling heat of the engine is added to the ORC system on a high temperature level
- Exhaust heat recovery is increased due to higher exhaust gas temperatures and a bigger fluid mass flow
- BTE_{ICE} and thereby engine power output is improved by lower wall heat losses and reduced friction

It can be assumed, that the small increase in NOx raw emissions is potentially compensated by a higher SCR efficiency rate.

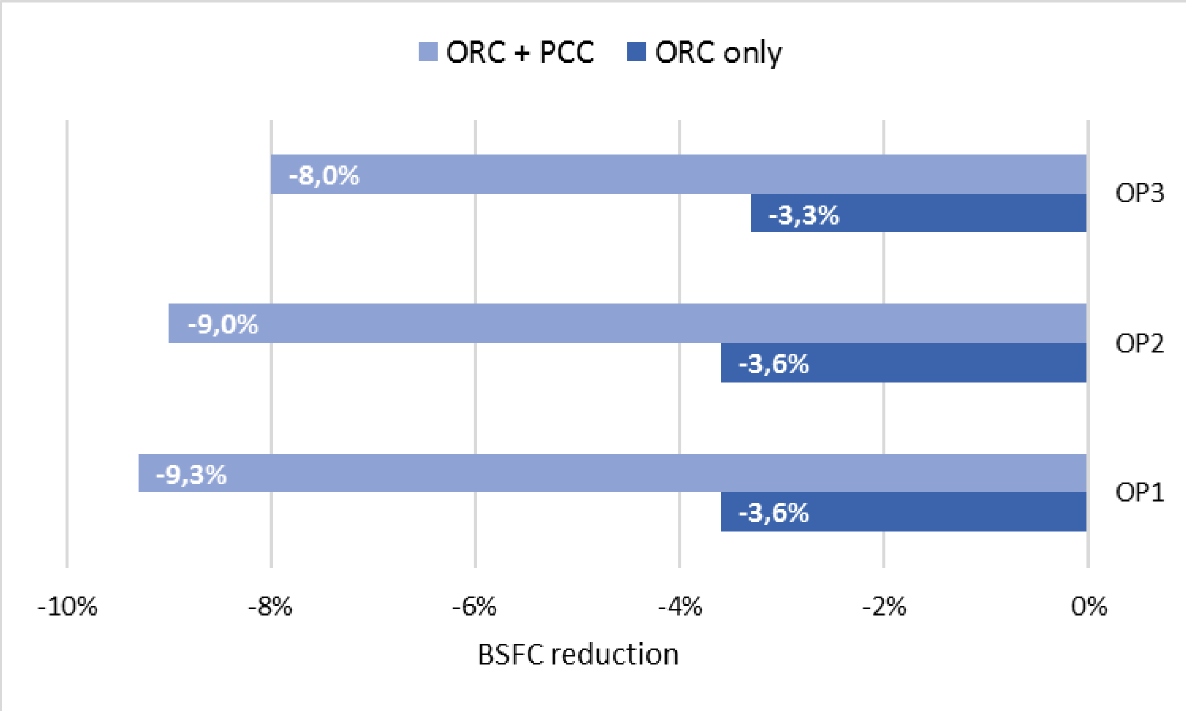

Figure 4: Comparison of BSFC reduction with ORC only and ORC + PCC

The BSFC improvement by PCC is achieved with low additional technical effort and expenses. Beside a new cooling core design, it only requires a throttle valve as additional part, so the cost/benefit ratio is significantly improved compared to the use of an ORC system only. This makes it a very attractive solution in order to meet the 2030 CO_2-goals in the EU.

NOMENCLATURE

ATM	Advanced Thermal Management	(–)
BTE	Break Thermal Efficiency	(%)
BSFC	Break Specific Fuel Consumption	(g/kWh)
EG	Exhaust Gas	(–)
EGR	Exhaust Gas Recirculation	(–)
HPP	High pressure Pump	(–)
HX	Heat Exchanger	(–)
NOx	Nitrogen Oxide	(–)
OP	Operating Point	(–)
ORC	Organic Rankine Cycle	(–)
PCC	Phase Change Cooling	(–)
SCR	Selective Catalytic Reduction	(–)
n	Engine speed	(rpm)
t	Temperature	(°C)
T	Torque	(Nm)
TP	Tail pipe	(–)

Subscript

exp	Expander
ICE	Internal Combustion Engine

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