EXHAUST WASTE HEAT RECOVERY FOR INTERCITY BUS CLIMATISATION USING RANKINE TECHNOLOGY WITH FOCUS ON TOPOLOGY DESIGN

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ABSTRACT

In this work, a preliminary simulative comparison of two possible topologies for combining waste heat recovery and compartment refrigeration is carried out. The focus is on the energetic assessment using a detailed simulation model of a long-haul intercity bus. The first topology includes a conventional Organic Rankine Cycle (ORC) that is integrated into the vehicle by directly coupling the expansion machine to the crankshaft. The applied working fluid is cyclopentane. For the means of compartment air conditioning in this configuration a R-717-refrigeration system is used, which has been derived from an R134a refrigeration system. The second topology uses the approach of an Organic Rankine Vapor Compression (ORVC) cycle, in which the refrigeration system and the waste heat recovery system share the same refrigerant and condenser. The used refrigerant here as well is ammonia (R-717). Expansion machine and compressor are both connected to the drive belt of the vehicle. In order to evaluate the fuel consumption reduction potential of these two topologies the intercity bus simulation model, equipped with the aforementioned R-717-refrigeration system, is used as a reference.
The results show, that the ORC topology reduces fuel consumption by 3.7 % and the ORVC topology by 6.5 %. Hence, in this specific scenario, the ORVC topology outperforms the ORC topology.
Keywords: ORC, ORVC, Ammonia, R-717, Cyclopentane.

1. INTRODUCTION

The compressor of the Air-Conditioning (AC) System of an intercity bus uses up to 15 kW of additional mechanical power from the engine, thereby reducing the effective work available for vehicle traction. For a long-haul journey of several hundred kilometers, this energy input accounts for around 8 % of the overall diesel fuel consumption. On the other hand, approximately one third of the supplied chemical energy is rejected as hot exhaust gas into the environment and another third is rejected by the cooling system leaving only one third for vehicle traction. In order to use the exergetic potential of the exhaust gas, a Waste Heat Recovery (WHR) system can be applied. The recovered energy can be used for reducing the engine load mechanically or by driving auxiliary loads like the alternator or the refrigerant compressor. Therefore, combining the waste heat recovery and the air-conditioning system can be a promising method for reducing primary energy usage. Combining these two systems efficiently is a challenge, as many aspects and interactions have to be considered.
WHR systems can roughly be distinguished in systems that use an additional medium, such as an Organic Rankine Cycle (ORC), and systems without additional fluids (e.g. direct exhaust gas expansion or thermoelectric generators). The work described in this paper is part of the overall research effort of the author that aims to develop and compare different topologies and methods for intercity bus climatisation using exhaust waste energy, with the help of an additional medium (e.g. ORC, absorption

or ejector processes). The aim is to completely cover the energy need of the refrigeration system and all its components. The focus within that research lays on system optimization with respect to dynamics and environmental regulations. In this study, a preliminary comparison of two possible topologies for combined waste heat refrigeration is carried out using a detailed simulation model of a long-haul intercity bus equipped with an R-717-refrigeration system. The basis in this case is the classical Organic Rankine Cycle and its derivative, the Organic Rankine Vapour Compression Cycle (ORVC). In terms of the latter, the carried-out research is mainly distinguished to other researches (e.g. Yilmaz (2015), Wang *et al.* (2011b), Saleh (2016 & 2018)) by using ammonia as working fluid/refrigerant.

2. ORGANIC RANKINE (VAPOR COMPRESSION) CYCLE

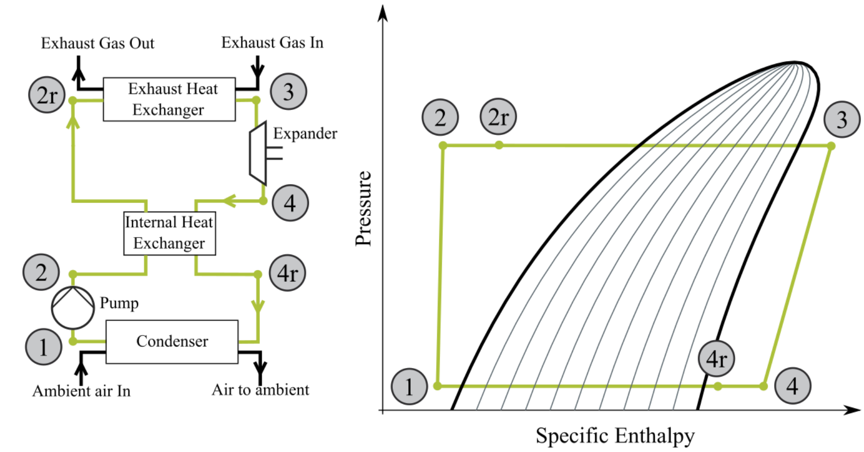

Figure 1: Schematic overview of a conventional Rankine Cycle with internal heat exchanger. On the left hand side, the diagram shows all essential components, the right hand side shows the corresponding thermodynamic points of state for the working fluid cyclopentane.

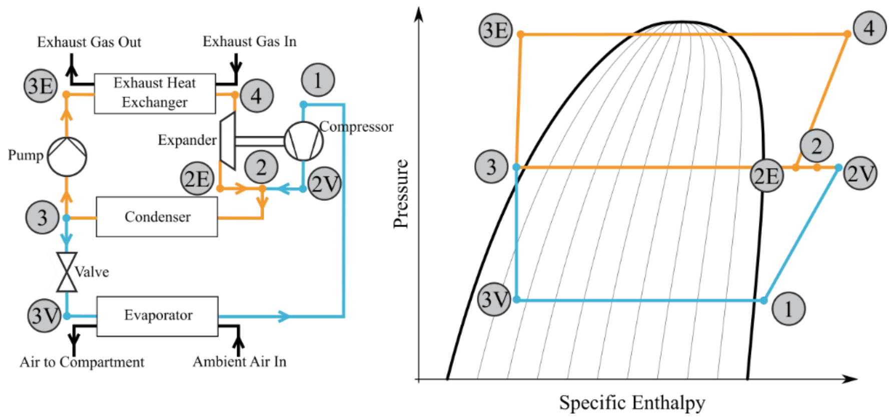

Figure 2: Schematic overview of an Organic Rankine Vapor Compression Cycle. On the left hand side, the diagram shows all essential components, the right hand side shows the corresponding thermodynamic points of state for the working fluid ammonia.

Figure 1 shows the scheme of a conventional ORC with its components and corresponding thermodynamic points of state. The system comprises of two main heat exchangers as sink and source, an expansion machine and a pump. In order to increase the efficiency an internal heat exchanger (IHX) is used as well. Depending on the system design, the sink can be another liquid flow, e.g. a cooling circuit of the vehicle or ambient airflow. In addition, it is possible to install a separator after the condenser on the working fluid side in order to assure that the pump is fed by saturated liquid only. **Figure 2** shows an example configuration of an Organic Rankine Vapor Compression Cycle (ORVC), as commonly shown the literature (e.g. Yilmaz (2015), Wang et al. (2011b), Saleh (2016)). All relevant components and thermodynamic points of state are depicted. The conventional refrigeration process, consisting of evaporator, condenser, compressor and expansion valve, is extended by a Rankine Cycle. Thus, the presented process is a combination of a heat engine and refrigeration machine, sharing the same working fluid. The remaining heat of both cycles is rejected via the same condenser on middle pressure level. The expansion machine and the compressor are connected mechanically. Depending on the system design, the expansion machine or the compressor can be connected additionally to an auxiliary load or drive.

3. SIMULATION MODEL AND INVESTIGATED TOPOLOGIES

The used omnibus simulation model comprises all necessary subsystems and consumers of a long-haul omnibus, such as the electrical system, drive-train, cooling system, passenger cabin, and the HVAC-(Heating, Ventilation and Air Conditioning) unit. The modeling resolution is high and takes all relevant dynamic effects into account. Furthermore, longitudinal dynamics have been considered. Vertical dynamics have been neglected. The model was created by Kaiser (2019) using the object orientated programming environment Modelica and has been validated with experimental data. All relevant energetic interactions are considered and therefore the model allows to evaluate energetic measures with respect to sub-system interactions such as implementing novel refrigeration systems or waste heat recovery systems. The actual vehicle represents an omnibus with a passenger capacity of 48 people and an engine peak power of 295 kW.

Table 1: Time average values of the evaluated reference system on a summer day in august on a journey from Hanover to Munich.

Description	Average value
Cooling load front- and roofbox evaporator	28 kW
Engine power	140 kW
Exhaust exergy flow at SCR-catalyst outlet	30 kW

The vehicle is originally equipped with an R-134a refrigeration cycle. In order to evaluate the impact of topology design on system performance, the refrigerant is replaced by Ammonia (R-717) in form of a drop-in. The gear ration between compressor and drive belt is changed in order to achieve the same cooling power as in the original R-134a system. In that manner, in all three refrigerant cycles the same refrigerant is used to make them comparable. Ammonia is here chosen as a place holder refrigerant, because of its high volumetric cooling capacity and a good compromise in usability as a working fluid in terms of waste heat activated cooling. In future work of the author, it is planned to use a novel refrigerant, which has similar thermo-physical properties without the drawback of health and safety issues. **Figure 3** on the left hand side shows the reference R-717-refrigeration system and its connections into the powertrain of the omnibus. The refrigeration system has two evaporators in parallel, one for the climatisation of the driver and one for the climatisation of the passenger compartment. For the sake of simplification, only the main evaporator for the passenger compartment is depicted, since it provides more than 90 % of the whole cooling load. The compressor is driven by the drive belt of the combustion engine. All fans are fed by the vehicle’s electrical system, respectively the alternator. Further explanation of the refrigeration system can be found in detail in Kaiser (2019). **Table 1** gives information about the evaluated reference system on a typical summer day in august on a journey from Hanover to Munich.

In the following, the investigated topologies and their implementations into the omnibus are described. All topologies have been created with the standard TIL- and TIL-Media library (Richter (2008), Gräber *et al.* (2010), Schulze *et al.*(2011)) in Modelica. A detailed description with all dynamic modelling approaches of the standard TIL and TIL-Media library can be found in Schulze (2013).

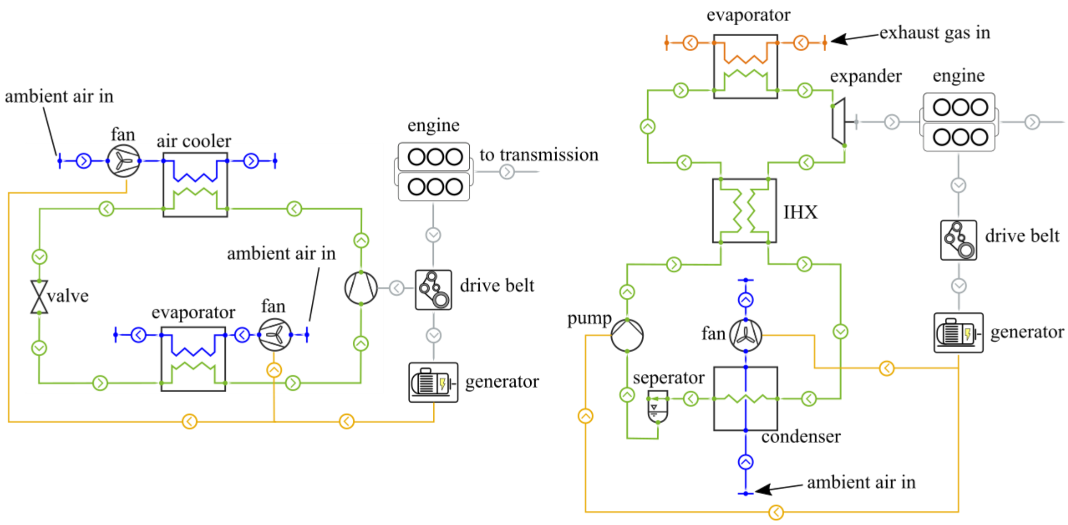

Figure 3: Schematic of the R-717 reference refrigeration system on the left hand side, and the ORC-topology on the right hand side. Both systems are depicted including connections to engine, drive belt and generator. Orange represents exhaust gas, blue represents air, green represents working fluid and yellow represents electrical current. Mechanical work is depicted in gray.

Figure 3 on the right hand side shows the ORC-topology and its connections into the powertrain of the omnibus. The expansion machine is coupled to the combustion engine with a transmission ratio of two, i.e. the speed of the expansion machine is twice the speed of the engine. This is a result of an iterative design process in order to achieve high pressures close to the pressure of maximum dew enthalpy. Investigations by Ebeling (2019) have shown, that under full load the optimal high pressure is close to that region. Fan and pump are fed by the vehicle's electrical system, respectively by the alternator. The working fluid is cyclopentane. In the literature it is quite often identified as one of the best performing workingfluids in terms of mobile waste heat recovery, as for example in Ebeling (2019) and Reiche (2018). Fan and expansion machine are assumed to have a constant overall efficiency of 70 %. The interdependence of pressure increase and volume flow rate of the pump at nominal speed is modeled by a quadratic function with the parameters pressure increase at zero flow rate and volume flow rate at zero pressure increase. Furthermore, a parameter for nominal effective efficiency is used, which is set to 55 %. The dependence of the aforementioned characteristic on rotational speed is modeled by affinity laws. All heat exchanger models are based on physically motivated models with implemented commonly used heat and pressure drop correlations (e.g. Haaf (1988), Shah (1979), Gnielinski (1975)). The heat exchanger design has been carried out roughly with an expected amount of available space in the vehicle. The evaporator is a Fin-And-Tube heat exchanger, the condenser is an Multi-Port-Extrusion heat exchanger in a crossflow configuration. The internal heat exchanger is a plate heat exchanger.
For dynamic controlling the process two PI-Controllers are used. The first one sets the speed of the pump and controls the inlet state of the expansion machine to 10 K superheated vapor. The second controller sets the speed of the condenser fan and hereby the condensing pressure of the cycle. The control variable in this case is the partial derivative of the exergetic efficiency with respect to the low pressure level of the system. The setpoint is correspondingly zero, hence the exergetic efficiency is maximized if the derivative gets zero. The partial derivative is calculated using a steady state model of the system. The applied controlling technique has been introduced by Noeding (2019) as "zero-gradient-control".
Since the compressor of the refrigeration system is connected to the drive belt as well, the shown configuration is similar to the one described in Wang *et al.* (2011a), where compressor and expansion machine are directly coupled and the applied working fluid and refrigerant are R-245fa and R-134a. In this work cyclopentane and R-717 are used. The author is not aware of any literature, that describes the investigation of this combination of working fluid and refrigerant.

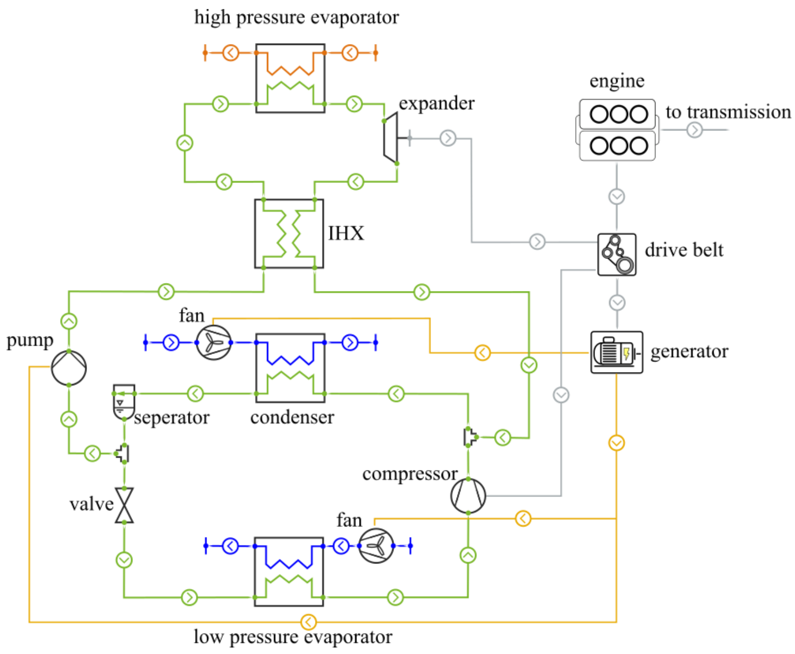

Figure 4: Schematic of the ORVC-topology including connections to engine, drive belt and alternator. Orange represents exhaust gas, blue represents air, green represents working fluid and yellow represents electrical current. Mechanical work is depicted in gray.

Figure 4 shows the ORVC-topology and its connections into the powertrain of the omnibus. This topology is a combination of the vehicle's reference refrigeration system and the described ORC-topology. The condenser of the reference refrigeration system replaces the ORC-condenser. The expected condensation heat is approximately twice the condensation heat of the reference refrigeration system, hence the size of the condenser is doubled. The number of condenser fans is increased as well. The chosen working fluid is again ammonia (R-717). The volume flow rate parameter of the pump is adapted from cyclopentane to R-717 in order to match the increased evaporation enthalpy. The expansion machine and the compressor are both coupled to the drive belt, the gear ratio between belt drive and expansion machine is adapted so that similar pressure ratios as in the ORC-topology are achieved. All other components and operating strategies of the system remain unchanged.

For controlling the outlet state of the expansion machine, a PI-Controller is used, which sets the speed of the pump for controlling the outlet state of the expansion machine to 60 K superheated vapor. In that manner, the following internal heat exchanger is provided with enough temperature difference to transfer heat from mid pressure to high pressure level. The controls of the expansion valves, condenser and evaporator fans are identical to the reference refrigeration system and remain the same as in the model of Kaiser (2019).

The described topology is similar to Yilmaz (2015), where R-134a and R-245fa are investigated as working fluid. In the system described by Yilmaz (2015), the compressor and expander are directly coupled with no other external connection. However, in this work the expansion machine and compressor are coupled via the drive belt of the vehicle. Hence, the shaft work of expander and compressor may be unequal, so that power is drawn from or supplied to the engine. Furthermore, as mentioned, in this work ammonia is used as working fluid.

4. BOUNDARY CONDITIONS AND SIMULATION RESULTS

4.1 Boundary conditions

In order to evaluate the integration of the presented topologies into the vehicle, a real driving scenario from Hanover to Munich is applied as simulation input. The scenario considers vehicle speed, slope of route and weather conditions. For the applied scenario a typical august summer day has been chosen, further details are explained in Kaiser (2019). **Figure 5** shows the vehicle speed and the corresponding driving slope with respect to time as well as the system input boundary conditions and system response of pressure at high pressure evaporator outlet for a representative time interval. In case of the ORC- and ORVC-topology both systems are only activated during highway conditions. The fuel consumption measurement interval has been chosen from 2000 until 23000 seconds of the journey. The mentioned interval is depicted in **Figure 5**, as well. All topologies are compared within this period. In both configurations, the ORC and the ORVC, the exhaust gas evaporator is integrated after the SCR-catalyst of the exhaust after treatment system of the vehicle.

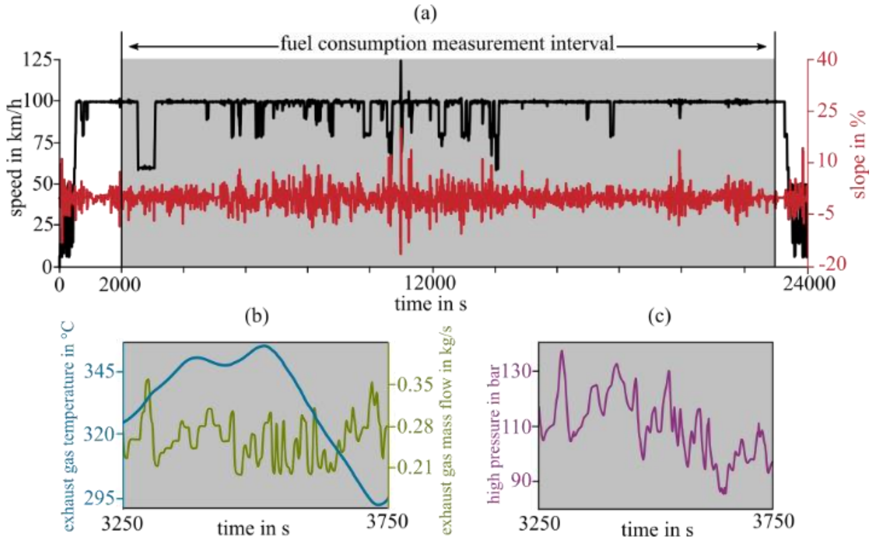

Figure 5: Real life driving scenario from Hanover to Munich (Kaiser, 2018). Vehicle speed and slope of route depicted with respect to time (a). System input boundary conditions (b) and system response of pressure at high pressure evaporator outlet (c) for a representative time interval of 3250 s to 3750 s for the ORVC-topology.

4.2 Results

Figure 6 shows the result for the exergy analysis in the mentioned driving scenario for the three topologies. Exergy source and exergy demand are compared for each topology. The exergy balance is formed around the engine, the drive belt and the gearbox, taking all necessary consumers into account. The exergy demand is hereby divided into driving resistance and gearbox losses, engine auxiliaries (all kinds of consumers like pumps or fans etc.), vehicle auxiliaries (lighting and electronic control units etc.), HVAC auxiliaries (condenser and evaporator fans and control units), compressor shaft power and the resulting drive belt losses. Electrical conversion losses of the electrical components are included as exergy demand of the corresponding component. In case of ORC and ORVC applications, the exergy demand of the pump and fan (only ORC) are depicted as well. As exergy source the corresponding component is depicted in opposition and is divided into engine crankshaft and, if existing, expansion machine.

It can be seen that the exergy demand covered by the engine decreases from the reference topology to the ORC and ORVC topology. In case of the ORC topology, the exergy output of the engine decreases by 4 % compared to the reference, in case of ORVC by 7.1 % compared to the reference. This decrease corresponds to a fuel consumption decrease by 3.7 % and 6.5 % respectively. The reason for this decrease is mainly due to the WHR system, since the expansion machine reduces the engine load, as mentioned above. In case of the ORC, there is no direct impact on the refrigeration system itself. Both topologies, reference and ORC, are using the same refrigeration cycle, hence there is no significant change in power demand for refrigeration. On direct comparison of the ORC and the ORVC topology, it can be seen, that the necessary work input of the compressor is decreased by around 4 kWh and the work output of the expansion machine is increased by around 9 kWh. The ORVC condenser size is increased, as well as the number of fans for condensation. This results in an increase of energy usage of the HVAC utilities by around 1.5 kWh.

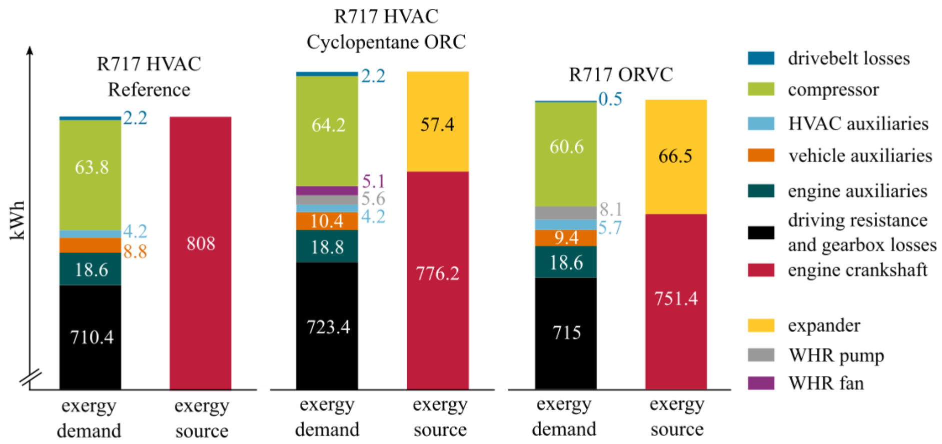

Figure 6: Exergy demand and source for the evaluated topologies with the for the period of a real life driving cycle from Hanover to Munich between 2000 and 23000 seconds of the journey. The results show a primary energy usage improvement between all topologies only in application of the described set of parameters.

The necessary work input for the pump is increased by 2.5 kWh as well. Still, the overall work input for the WHR is lower in direct comparison to the ORC topology, since the expansion machine is able to supply more energy and also there is no dedicated condenser fan needed. In case of the ORVC toplogy, the engine has to provide 8 kWh of mechanical energy and in case of the ORC topology 22 kWh of mechanical energy in order to supply air conditioning for the passenger compartment.
Another aspect that can be seen from the simulation results is the impact of the expansion machine on the drive train of the engine. As mentioned, in all three topologies identical boundary conditions were applied. Due to that, the exergy demand for the driving resistance should be equal in all three simulations. As shown in **Figure 6**, this is not the case. The exergy demand differs by about 13 kWh in case of the ORC and by about 5 kWh in case of the ORVC topology compared to the reference. Because of the thermal capacity of the WHR and the exhaust manifold, the time constant of the WHR is several magnitudes bigger than that of the drivetrain's mechanics, hence the expansion machine is still providing energy while the driver is applying the brake pedal. Due to the additional power input of the expansion machine into the drive train, the driver, represented by a PI-controller, tends to accelerate and decelerate more aggressively, which leads to the shown increase in driving resistance, respectively fuel consumption. Consequently, the fuel reduction potential for the shown ORC and ORVC configuration is slightly higher than depicted in **Figure 6**. The adaption of the driver model as function of drivetrain design is part of future work of the author.

5. CONCLUSIONS

The results show that with the applied parameters and model approach it is possible to effectively decrease the fuel consumption of an intercity bus on a long-haul journey using both systems. In direct comparison of the shown configurations with the applied set of parameters, the fuel reduction potential of the ORVC topology is about twice of that of the ORC topology. Still, it has to be stressed, that in this comparison two different approaches of condenser design has been applied. In case of the ORC topology, it is assumed that the condenser is placed near the engine with little packaging space available. In case of the ORVC, the original HVAC unit has been modified, which uses an condenser installed on the roof of the vehicle, with obviously more space available. Also, it has not been taken into account that the use of R-717 in direct evaporation systems can lead to grave safety issues, since it is a highly toxic substance. Among others, the scope of future work is therefore to implement an ORVC with alternative refrigerants which have no impact on the environment or vehicle passengers, similarly as shown in Saleh (2018). It has to be evaluated, if yet a fuel reduction potential is given. In that case the use of an ORVC would be highly promising, since the needed cooling load could be maintained with simultaneously meeting environmental regulations, such as GWP and ODP. It then has to be evaluated, if other topologies (e.g. absorption or ejector processes) can compete.
In summary, it has to be pointed out, that further extended research has to be carried out in order to make the two systems more comparable. Several effects have not been analyzed, which could improve

the presented ORC configuration's performance. Both, ORC and ORVC topology, have not been fully optimized and will be presented in future publications.

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