Variable Geometry Turbine Design for Off-Highway Vehicle Organic Rankine Cycle Waste Heat Recovery

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Abstract. Although modern ultra-efficient heavy duty diesel engines exhibit thermal efficiencies of well over 40%, a substantial part of fuel energy will continue being wasted as heat in the exhaust system due to the Diesel cycle limitations. Recovering this potential source of energy could increase the overall thermal efficiency of the engine as well as reduce the exhaust gas emissions and the operational cost of the heavy duty diesel engine. Organic Rankine Cycle (ORC) is regarded as a promising candidate technology for transforming exhaust gas waste heat into electricity (or direct power) due to the nature of the untapped source of energy that it allows to be capitalised on, its relative simplicity and small back pressure impact on engine performance and fuel consumption. This study was carried out for an off-highway engine project but its application is generic to heavy duty applications. An ORC model with a radial expander submodel is implemented in a heavy duty diesel engine powertrain, to evaluate the impact of the ORC on fuel consumption and exhaust gas emissions under various engine operating conditions. Finally, the potential benefit of utilizing a variable geometry expander (VGE) in the ORC model is investigated in this study. Compared to its fixed geometry expander equivalent, the VGE shows a wider range of high efficiency operation within the engine operating window and is therefore a promising addition for consideration in ORC application of this type.

Notation

bsfc	*Break specific fuel consumption*
CO_2	*Carbon dioxide*
FGE	*Fixed Geometry Expander*
ORC	*Organic Rankine Cycle*
SCR	Selective Catalytic Reduction
TC	*Turbocompounding*
TEG	*Thermo-electric generation*
VGE	*Variable Geometry Expander*
WHR	*Waste Heat Recovery*

1. Introduction

The increasingly stringent European emissions legislation on CO_2 and other pollutants is what drives to a large extent manufacturers for ultra efficient heavy duty diesel engines. Waste heat recovery (WHR) technologies are gaining ground in heavy duty diesel engine (including off-highway applications) as in these cases the maximum thermal efficiency exceeds 40% and the majority of the fuel energy is wasted (Teng H., Klaver J. et al. 2011; Liang, Wang et al. 2015). It is known that a 10% reduction on fuel consumption could be achieved if 6% of the enthalpy contained in the exhaust gases convert into electric power (Vázquez J., Sanz-Bobi M. A. et al. 2002). At the same time, an up to 20% increase in the engine maximum power could be achieved by the WHR system without additional fuel consumption (Teng H., Regner G. et al. 2007).

WHR technologies can be classified into three mainstreams, namely; thermo-electric generation (TEG), turbocompounding (TC), and organic Rankine cycle (ORC). Experimental studies proved that fuel savings of 3.9 up to 4.7% could be achieved by using thermo-electric generation (Stobart R. and Milner D. 2009; Stobart R., Wijewardane A. et al. 2010); however this technology is currently highly expensive and faced with a longer development time. On the other hand, mechanical turbocompounding can potentially improve brake specific fuel consumption up to 6% (Wilson 1986), while electrical turbocompounding contributes to fuel economy by up over 5% (Hopmann U. 2004; Katsanos C. O., Hountalas D. T. et al. 2013). The main disadvantage of TC is the increase of backpressure and finally the higher pumping losses, which compromise fuel savings from the recovery of exhaust gas heat (Mamat A.M.I. 2012). Last but not least, the Organic Rankine Cycle is probably the most promising candidate for conversion of exhaust heat into power due to its performance and practical elements of cost and ease of maintenance. The heat exchanger of the ORC system produces less backpressure compared to the TC technology, while the thermal efficiency can reach up to 13% at maximum engine power of a heavy duty diesel vehicle (Sekar R. and Cole R.L 1987).

The Rankine cycle is a closed loop cycle where heat is transferred to a working fluid at constant pressure. It consists of four main components, namely; evaporator, expander, condenser and pump (Fig.1). The working fluid is vaporized in the evaporator and then expands in the expander that drives a generator to produce electricity. Finally, the working fluid is condensed at constant pressure and pumped again to the evaporator. In recent years, a great number of studies deal with the implementation of ORC systems in vehicle powertrains. Yang et al. found that the implementation of an ORC system operating with R245fa improves bsfc from 2.5% to 7.4% (Yang K. and Zhang H. 2015). In another study, the engine water and the exhaust gas were employed to predict ORC efficiency of around 9.6%, while the total engine thermal efficiency was increased by 9.0% (Shu G.Q., Yu G. et al. 2013). The efficiency of the ORC system is a function of its specification, including the available heat sources employed, the heat exchanger design, the working fluid selected and the expander type chosen and its design, to name the most important.

Among the ORC system components, the expander is the most crucial and expensive component in Organic Rankine Cycle (ORC) systems (Wong C. S., Meyer D. et al. 2013). Expanders can be classified into two main groups, namely; positive displacement expanders (Screw, Scroll, Piston and Rotary Vane) and turbomachine expanders (Axial or Radial). The selection of the appropriate expander depends on the application; however for waste heat recovery applications scroll expanders and radial turbines are the most common solutions in literature (Rosset K., Mounier V. et al. 2015; Zhen L., Guohong T. et al. 2015). ORC efficiency is increased at higher pressure ratios; therefore radial turbines appear as more suitable for vehicular applications where mass flow rates are in the low to middle range and pressure ratios are middle to high. In terms of manufacturing cost, it is less expensive compared to axial turbines as they can be converted from standard production designs, being less sensitive to their blade profile (Paltrinieri N. 2014), while radial turbine geometry allows higher peripheral speeds than the axial turbines and therefore a higher enthalpy drop per stage (Quoilin, Broek et al. 2013). On the disadvantages of radial turbines, they are inefficient at part load, don't operate efficiently at variable speeds (Petchers N. 2003) and their efficiencies drop when operating under off-design conditions (Teng H., Regner G. et al. 2007).

The implementation of a variable geometry turbine can potentially mitigate many of the performance disadvantages of a radial turbine expander in an ORC system. A recent study on the aerodynamic evaluation of a VGT for organic Rankine cycle showed that turbine power and efficiency is improved in a higher range of mass flow rate and expansion ratios compared to the fixed geometry turbine (Wong S.C. and Krumdieck S. 2015). However in another study the implementation of a variable geometry turbine in a low temperature ORC system that uses geothermal heat showed little benefit in terms of average power output compared to a fixed geometry turbine. (Read M., Kovacevic A. et al. 2015). However, the literature review performed for this work has failed to uncover a detailed study to evaluate the impact of variable geometry radial expander (VGE) performance for organic Rankine cycle waste heat recovery in vehicular applications. Additionally, the evaluation of this technology in terms of fuel consumption and emissions at partial engine load conditions is crucial, as internal combustion engines will only infrequently operate at the ORC design point.

This study explores the impact of a variable geometry expander (VGE) in an ORC system for waste heat recovery from an off-highway vehicle. An integrated in-house model has been developed for this reason, which includes the engine map row data from a heavy duty diesel engine, the ORC model and the variable geometry radial expander model. In order to evaluate the potential benefit on fuel consumption and NOx emissions, the model was employed at various engine load and speed operating points. The aim of this study is to evaluate the impact of the VGE at design and off-design operating points of an ORC system and to compare these with the results of a conventional FGE. This is achieved by employing the integrated engine-ORC powertrain model across a range of important engine load and speed operating points in steady-state condition.

2. Powertrain Modeling Approach

The proposed integrated powertrain model is schematically presented in Fig. 1. The inputs of the model are the geometric characteristics of the heat exchanger (evaporator), the working fluid properties, the diesel engine maps and the expander maps. The model solution includes the calculation of the turbine power output, the ORC efficiency as well as the combined fuel consumption, NOx specific emissions and powertrain power output.

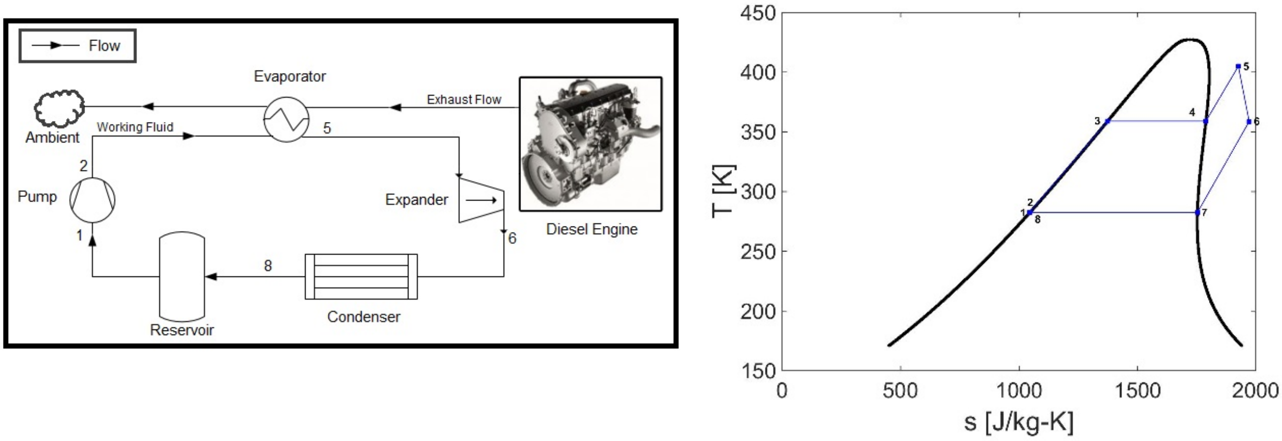


Fig. 1. (Left) Schematic presentation of the ORC powertrain; (Right) Schematic presentation of the thermodynamic ORC cycle

2.1 Organic Rankine Cycle optimization

An in-house MATLAB code has been developed for the thermodynamic modeling and optimazation of the ORC system. The code utilizes CoolProp v5 to calculate the thermodynamic properties of the organic fluid at liquid and gaseous conditions. In this version of the ORC model, the system is optimized to operate at steady state conditions, while the heat exchanger is assumed ideal. In addition, for simplicity, the heat and pressure losses in the connecting pipes are neglected. The heat input from the exhaust gas is given by equations (1) and (2). The number indexes are schematically described in the right section of Fig.1.

$$\dot{Q}_{in} = \dot{m}_{WF} \cdot (h_5 - h_2) \tag{1}$$

$$\dot{Q}_{exh} = \dot{m}_{exh} Cp_{air} \cdot (T_{exh,in} - T_{exh,out}) \tag{2}$$

$$T_{exh,out} \geq 200^{o} C \tag{3}$$

The working fluid mass flow ($\dot{m}_{WF}$), the ORC peak pressure (which controls the superheating percentage) and the exhaust temperature can be optimized from the in-house code, using as objective function the cycle thermodynamic efficiency and by fulfilling the constraints (equation 3). However in this study, the working fluid mass flow was constant for all cases and equal to 0.34 kg/s, as the target of this study is to show the potential benefit of a variable geometry turbine when only the inlet enthalpy varies. Regarding the rejected heat, it is assumed ideally that the exit temperature of the organic fluid is equal to 320K and it is described by equation (4).

$$\dot{Q}_{out} = \dot{m}_{WF} \cdot (h_6 - h_8) \tag{4}$$

The consumed power by the pump is determined by equation (5). The pump efficiency was assumed constant in this study and equal to 0.65, and was considered as a realistic value to reduce impact on the total ORC thermal efficiency calculation.

$$\dot{W}_{pump} = \dot{m}_{WF} \cdot \frac{P_2 - P_1}{\rho_1 \eta_{pump}} \tag{5}$$

The efficiency of the expander is given by the expander model through an interpolation and extrapolation module, as expander efficiency varies at different nozzle (stator) positions, expander rotational speeds, pressure ratios and mass flow rates. Then the ORC model calculates the power produced by the expander through equation (6). The net electric power produced by the ORC is given by equation (7). The efficiency of the generator was assumed constant and equal to 0.92, while the mechanical losses are negligible, as the transmission ratio is 1:1 there are no gears between the expander and the generator.

$$\dot{W}_{expander} = \dot{m}_{WF} \cdot \left(h_5 - h_{6,is}\right) \cdot \eta_{expander} \tag{6}$$

$$\dot{W}_{net} = \dot{W}_{expander} - \dot{W}_{pump} \tag{7}$$

Finally, the overall ORC efficiency is given by:

$$\eta_{ORC} = \frac{W_{net}}{Q_{in}} \tag{8}$$

2.2 Engine modeling

The engine model was based on a 10.3ℓ heavy duty diesel engine which basic characteristics are given in Table 1 (Biaggini G. and Knecht W. 2000). This version of this engine fulfills the Euro 3 emission standards and it is equipped with a turbocharger and a common rail injection system, while its maximum engine power is 316 kW at 2100 rpm. This six cylinder engine appears to be a reasonable choice to apply a waste heat recovery system on, considering its high exhaust flow rate and the level of exhaust gas power available for conversion.

The modeling of this engine was performed using a commercial engine simulation tool (GT-Power), in order to develop the required engine maps. The final calibrated engine model calculates not only the fuel consumption, but also the exhaust gas temperature, the exhaust mass flow rate (exhaust waste heat) as well as the engine NOx emissions, which formation is based on a calibrated extended Zeldovich mechanism submodel. Regarding the engine modeling, it is expected an up to 10% error on the estimation of the exhaust waste heat, as a lot of information is missing, especially on combustion modeling. However, by assuming a maximum 10% efficiency for the ORC cycle, the estimated error for the ORC system is up to 1%, which seems reasonable for this study.

Table 1. Characteristics of the heavy duty diesel engine

Main specification		
Bore	125	[mm]
Stroke	140	[mm]
Compression Ratio	17	[-]
Valve Number/Cylinder	4	[-]
Cylinder Number	6	[-]
Cam timing		
IVO	16	BTDC
IVC	32	ABDC
EVO	51	BBDC
EVC	11	ATDC
Turbocharging		
Type	VGT mixed flow turbine	
Charge air cooling	Air/Air	
Performance		
Max Torque	1900 Nm / 1000-1600 rpm	
Max Power	316 kW / 2100 rpm	

2.3 Expander modelling

The expander modelling includes the modelling of a radial inflow turbine which consists of three main components namely volute, stator and rotor. A 0D/1D commercial software (Rital^{TM}) has been employed for the modelling of the radial expander at the optimum operating point. Then a variable geometry turbine map has been scaled, in order to match the optimum efficiency of the designed expander with the one provided from the map. This scaling is assumed that it also includes the impact of different working fluids. Finally, the efficiency of the turbine is imported in the ORC model as a function of mass flow rate and pressure ratio. The reason that this simplified approach had been followed is to present the potential benefit of a variable geometry turbine on the performance of an ORC system, rather than focusing on the turbine design itself. The next step of this study is the development of a variable geometry radial expander map using CFD tools.

3 Results

3.1 Engine modeling

A heavy duty diesel engine model has been developed in this study, using the commercial engine simulation tool. The model was calibrated at five (5) different load/speed operating points. The engine model calibration was based on experimental fuel consumption values from (Biaggini G. and Knecht W. 2000). Fig. 2 shows the experimental fuel consumption map and the five simulation points. The reason that these five points were selected is to evaluate the improvement of the engine characteristics such as the maximum power, maximum torque and the bsfc by implementing an ORC system in this engine. Additionally, P1 and P2 are representative points on partial load conditions, where engine works at 30% of the operating time at normal off-highway conditions.

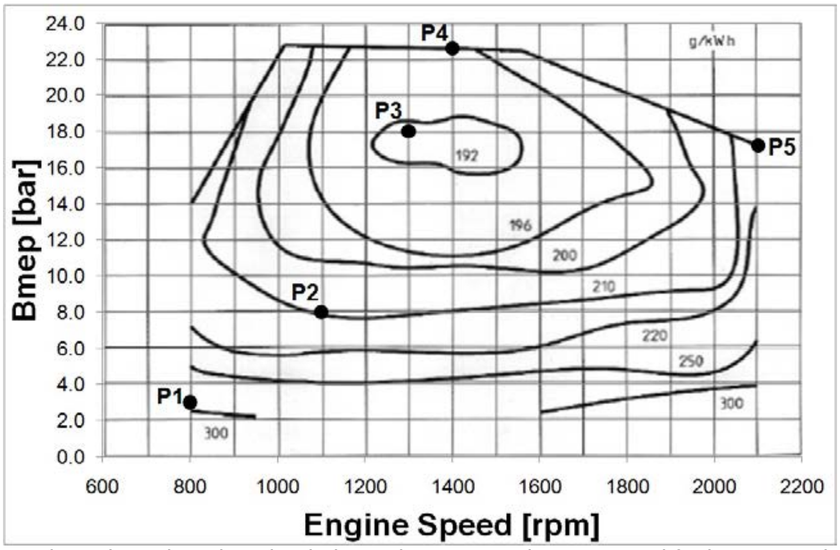


Fig. 2. The selected engine simulation points versus the measured fuel consumption map

The results of the engine modeling include the calculation of the exhaust gas conditions and the engine fuel consumption and emissions, which are briefly presented in Table 2. Comparative results between Table 2 and Fig. 2 show that brake specific fuel consumption (bsfc) is predicted within a 2% error band compared to the experimental values. It can be observed that exhaust flow rate and temperature are increased at higher load and speed conditions; this means that the available exhaust gas waste enthalpy for the WHR system is higher.

Table 2. Summary results from engine simulation

Testing Points	P1	P2	P3	P4	P5
Engine Speed [rpm]	800	1100	1300	1400	2100
Bmep [bar]	3	8	18	23	18
Power [kW]	21	76	201	277	325
Torque [Nm]	246	656	1476	1887	1477
Bsfc [g/kWh]	283	210	192	194	212
Exhaust Mass Flow Rate [kg/s]	0.07	0.16	0.30	0.39	0.46
Exhaust Temperature [K]	695	670	681	719	804

3.2 Optimization of Organic Rankine Cycle using Variable Geometry Expander

The exhaust gas conditions at the selected five operating points were fed into the ORC model, which was parametrically executed for a realistic range of expander nozzle positions (0.3 – 1, which correspond to 30% open (expander throat area) to 100% open). Fig. 3 shows the impact of the nozzle positions on ORC efficiency (left) and the expander efficiency (right) under various engine conditions. As the nozzles are closed (moving towards 0.3 from an initial value of 1.0), expander total to static isentropic efficiency is initially increased and after reaching a maximum the efficiency is decreased. This trend stays true for most cases except for a slight trough in the case of the peculiar engine point P1 (at idle).However, the trend corresponds to typical variable radial turbine geometry behavior. Similarly ORC thermal efficiency follows qualitatively the trend of the expander efficiency but not quantitatively. In fact, VGE position affects the organic fluid mass flow which seems to affect the efficiency of the heat exchanger (evaporator). Finally the ORC maximum thermal efficiency is a trade-off between the efficiency of the evaporator and the expander, while the same trend can be observed for the net power of the ORC system (Fig. 4).

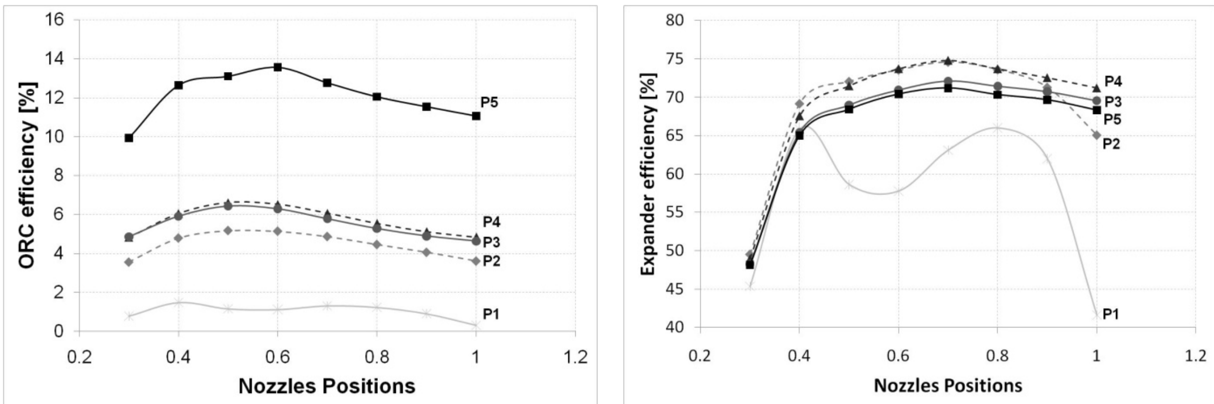


Fig. 3. Impact of the VGE nozzle position on (a) ORC system efficiency and (b) expander total-to-static isentropic efficiency for five different operating points at 50000rpm using working fluid R245fa.

Another important observation is coming from the strange behavior of operating point P1, regarding the expander efficiency (Fig. 3). In fact at low heat and mass flow conditions, the organic fluid may not evaporate fully resulting in significant drop in expander efficiency. As nozzle position is increased, the mass flow rate of the organic fluid is increased, although the available exhaust heat energy remains constant. This is the main reason that ORC systems at partial to low load conditions present very low efficiency, while the VGE technology seems to be significantly beneficial at these points.

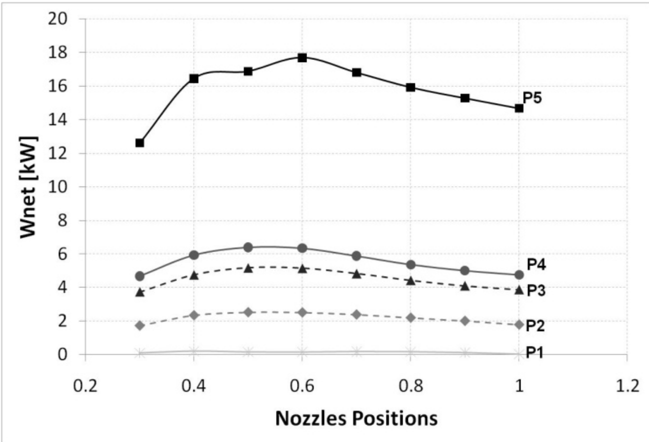


Fig. 4. Impact of nozzles rack positions on ORC net power for five (5) different operating points at 50000rpm using working fluid R245fa.

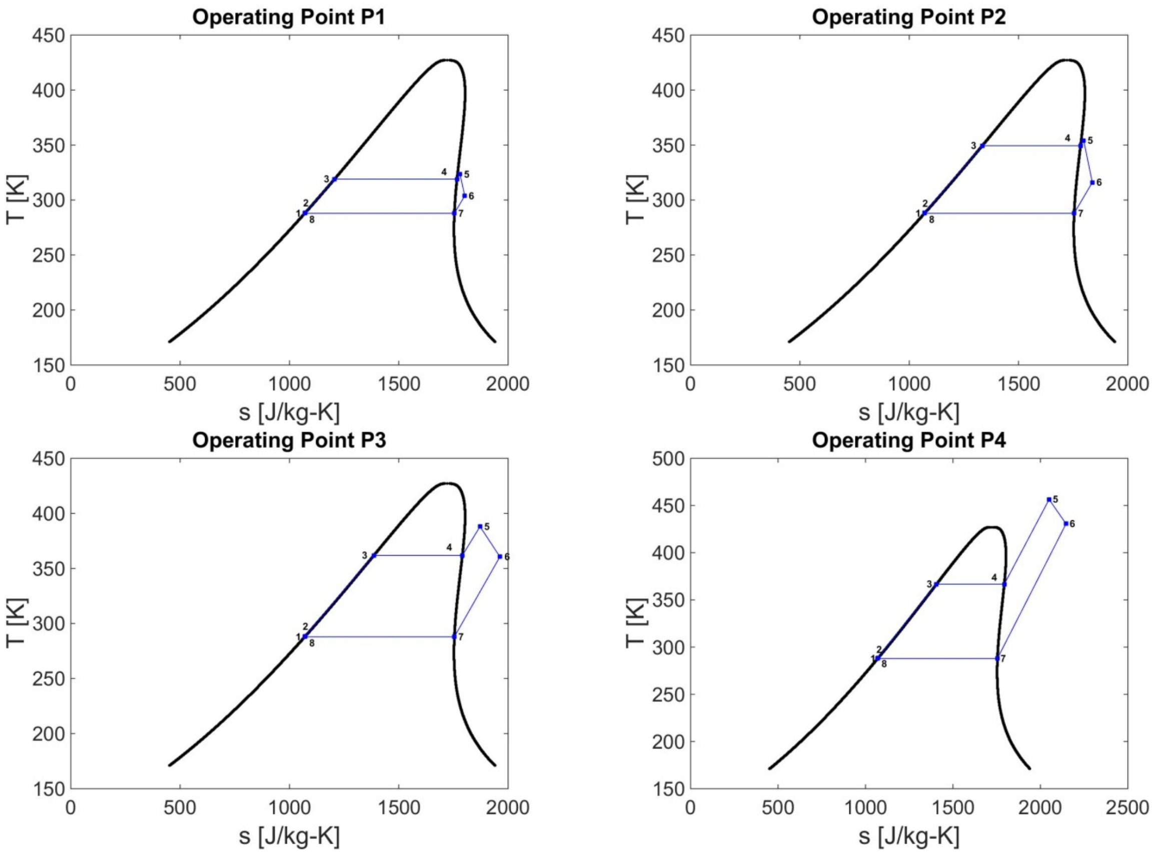
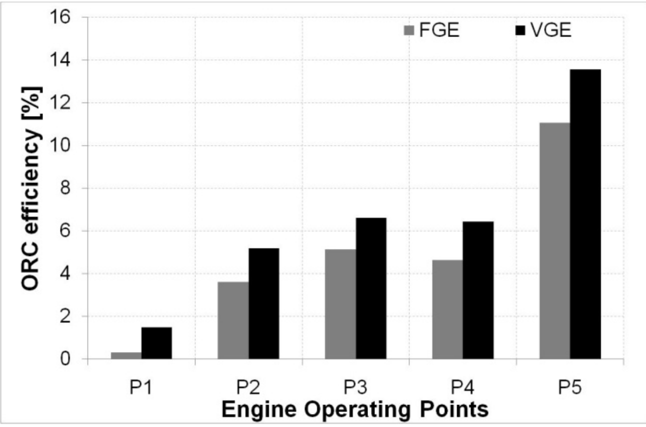


Fig. 5. Schematic presentation of the ORC temperature – entropy (T-S) diagram for four different engine operating points at the optimum, corresponding, nozzle positions using working fluid R245fa.

The mass flow rate of the organic fluid has an important role in the ORC system. In most ORC studies, mass flow rate remains constant and a similar methodology has been adopted in this study, although nozzle position does throttle the mass flow resulting in slightly lower mass flow with decreasing nozzle area. Fig. 5 illustrates the temperature – entropy diagrams for four different operating points at the optimum nozzle position, regarding ORC efficiency. At engine operating condition P1, cycle point 5 is very close to point 4, with almost no superheating, which validates the explanation on the expander efficiency above, at not complete evaporation. As engine load and speed are increased, more exhaust heat energy is available due to the almost constant organic fluid mass flow and therefore superheating is increased. Although superheating is important to ensure the full evaporation of the organic fluid, too much superheating as that obtained at operating point P4 (or P5 as it present similar thermodynamic points) is detrimental to ORC efficiency. Superheating can be decreased by increasing the organic fluid mass flow rate. Additionally, expander power is proportionally related with the increase in mass flow rate, which means that increasing organic fluid flow rate can only be beneficial in the final ORC efficiency. The latter declares that in the hypothetic case of a variable flow rate ORC system, a variable geometry expander can potentially provide even better results under all operating points compared to the benefit it can provide from a fixed flow rate ORC system.

The Variable geometry expander achieves ORC efficiency benefits compared to the fixed geometry expander even under fixed organic fluid mass flow. Fig. 6 presents the impact of the VGE on ORC efficiency and ORC net power compared to a fixed geometry expander (no moving nozzles). It is observed that the variable geometry expander achieves higher ORC efficiency and net power through all engine points. Especially under low to partial load conditions, where ORC efficiency suffers, VGE appears to enhance the ORC system performance. In addition, at high loads, the extra power of the VGE technology compared to the FGE is almost 3 kW, while the ORC efficiency is extended beyond the 12%, which appears in most studies to be a the higher limit in the performance of an ORC system.



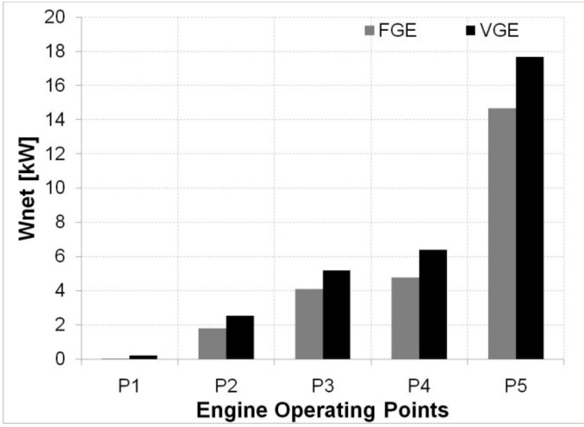


Fig. 6. Impact of a Variable Geometry Expander (VGE) compared to a Fixed Geometry Expander (FGE) on (a) ORC system efficiency and (b) ORC net power for the five (5) different operating points.

Fig. 7 illustrates the VGE contribution on the net power and the thermal efficiency of an ORC system. It is presented that VGE increases both ORC net power and efficiency by up to 350% at low load conditions, while in partial to high engine loads VGE contributions is between 20 and 50%. However it has to be mentioned that the accuracy at very load operating points is very sensitive on the engine simulation, while it is expected that the ORC efficiency error is within 1%, which is very close to the predicted ORC efficiency value at idle conditions. Therefore the contribution of VGE is expected to range between 20% and 50% in terms of ORC efficiency.

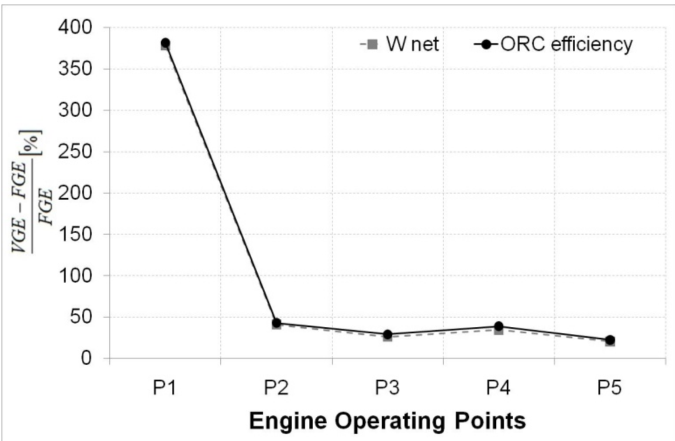


Fig. 7. Percentage improvement on ORC net power and ORC thermal efficiency by utilizing a variable geometry expander (VGE) compared to a fixed geometry expander (FGE).

3.3 ORC-equipped powertrain

The implementation of an ORC system on the off-highway vehicle powertrain can improve powertrain power, fuel consumption and emissions. The target of this paper is to explore the potential for improvement of the ORC powertrain when a VGE expander is implemented. Fig. 8 presents the improvement on the powertrain power due to the ORC system with and without the VGE technology. At maximum engine power, powertrain power is increased by 15 kW (4.5% increase) while the implementation of a VGE technology gives an additional, approximate 1% increase on powertrain power (transmission losses neglected as this is a feasibility study). Regarding the other operating points, VGE shows an improvement on the ORC powertrain exit power from 0.9% to 1.2% compared to the FGE.

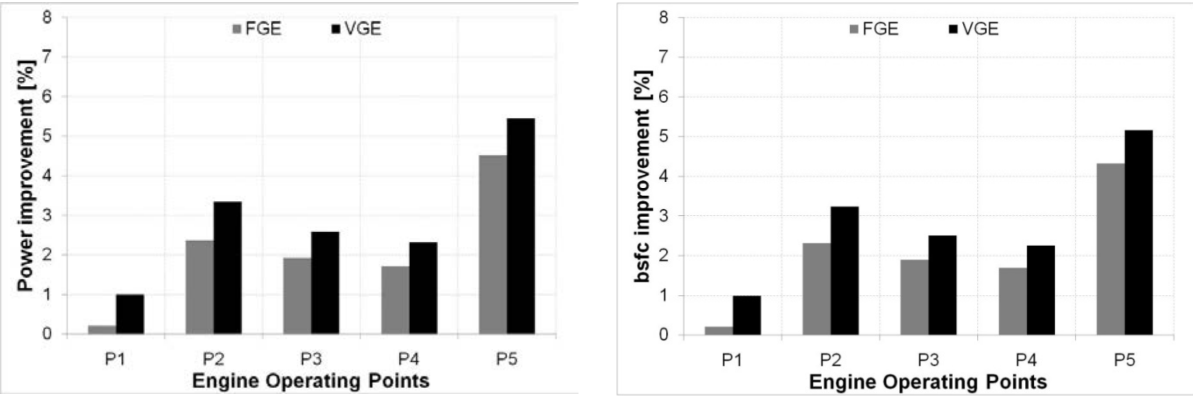


Fig. 8. Comparison between a fixed geometry expander (FGE) and a variable geometry expander (VGE) on the improvement of engine power (left) and engine bsfc (right).

The main purpose of waste heat recovery is the improvement of engine (or powertrain) fuel consumption. The right side of Fig. 8 shows the percentage on the powertrain bsfc improvement due to the ORC system. In general, bsfc can be improved up to 4.2% with a fixed geometry expander and over 5% when a VGE is implemented. In fact, VGE benefits fuel consumption additionally by up to 1%, which is proportional to the fuel cost. Last but not least, it is observed that the ORC impact on fuel consumption is more intensive at engine operating points where engine thermal efficiency is not very high; these points are partial to low loads (P1, P2) and full load conditions (P5).

Finally, ORC system was found to improve NOx emissions. In off-highway vehicles and marine applications emissions are measured in g/kWh; therefore the increase of the combined output power due to the ORC leads to lower brake specific NOx. Fig. 9 shows that an ORC system itself can't keep emissions under legislation emission standards; however it can assist an aftertreatment system to handle lower NOx emissions. On the other hand, an ORC system can assist the SCR catalyst to operate within the temperature range of 350-450^{o}C by controlling the available thermal power of the exhaust gas.

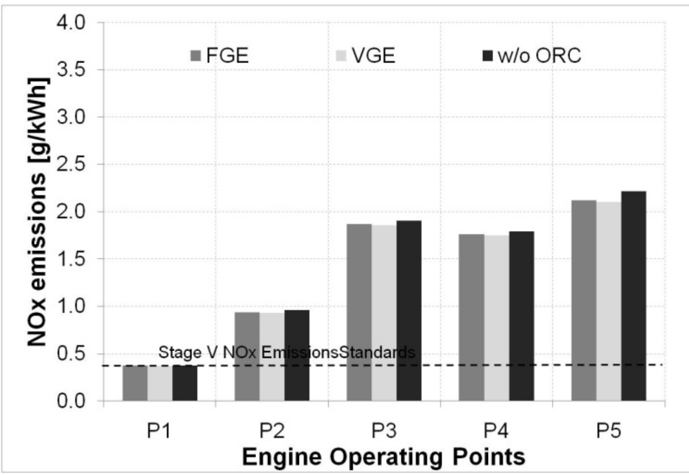


Fig. 9. Impact of the ORC system using a fixed geometry expander (FGE) and a variable geometry expander (VGE) on engine NOx emissions.

Conclusions

In this study, a variable geometry radial turbine expander (VGE) for waste heat recovery using an organic Rankine cycle (ORC) for an off-highway vehicle application has been investigated. This simulation study also includes the comparison between a fixed geometry expander and a variable geometry expander, in order to evaluate the potential benefit of the VGE. The comparison has been applied at five different engine operating points such as maximum power, maximum torque, optimum fuel consumption (bsfc) operating point, partial load and idle.

The fixed geometry expander showed that the ORC thermal efficiency achievable is between 4% at partial load and 11% at maximum power conditions. In terms of power, this represents between 2 and 15 kW additional power. The optimum fuel consumption point is further improved by 1.9%, while the maximum power that is achieved by improving fuel consumption by 4.2%. Last but not least, brake specific NOx was also improved by up to 4.2% at maximum power conditions.

When the VGE was applied, a further improvement is observed on ORC efficiency, fuel consumption and brake specific NOx emissions. It was found that VGE can improve ORC efficiency and net power by an unweighted point average of 34% at partial to high load conditions while benefits are even higher at the lower loads. This technology can also have an impact on the powertrain of off-highway vehicles and many other medium and heavy duty diesel engines. Compared to the fixed geometry WHR system, the VGE WHR system presents an additional 20% to 50% percentage improvement on the ORC thermal efficiency. Fuel consumption is also decreased by an additional 1% while output power in increased by the same percentage. Brake specific NOx emissions are decreased by the same order of magnitude, although this is not enough in order to reach the stage V emissions standards for off-highway vehicles. Finally, it may be stated the performance of a VGE offers a substantial improvement in terms of relative fuel consumption gain compared to a conventional FGE. This makes it an attractive consideration given also that its on-cost is likely to be only a fraction of the relative on-cost of implementing variable geometry to a turbocharger which is a simpler, lower cost system (compared to an ORC system) making the latter less sensitive to the addition of variable geometry.

Acknowledgement

The authors would like to thank Ian May for the GT Power support in this study.

References

Biaggini G. and Knecht W. (2000). The Advanced Iveco Cursor 10 Heavy Duty Truck Diesel Engine. Seoul 2000 FISITA World Automotive Congress. Seoul, Korea. 2000-05-0071.

Hopmann U. (2004). Diesel Engine Waste Heat Recovery Utilizing Electric Turbocompound Technology. 2004 DEER Conference, San Diego, California.

Katsanos C. O., Hountalas D. T., et al. (2013). "Simulation of a heavy-duty diesel engine with electrical turbocompounding system using operating charts for turbocharger components and power turbine." Energy Conversion and Management 76: 712-724.

Liang, X., X. Wang, et al. (2015). "A review and selection of engine waste heat recovery technologies using analytic hierarchy process and grey relational analysis." International Journal of Energy Research 39(4): 453--471.

Mamat A.M.I. (2012). Design and Development of a High Performance LPT for Electric Turbocompounding Energy Recovery Unit in a Heavily Downsized Engine. Department of Mechanical Engineering Imperial College London. PhD.

Paltrinieri N. (2014). A Mean-Line Model to Predict the Deaign Performance of Radial Inflow Turbines in Organic Rankine Cycles. Department Of Industrial Engineering, University of Padua. Master of Science in Energy Engineering.

Petchers N. (2003). Combined Heating, Cooling & Power Handbook: Technologies & Applications : An Integrated Approach to Energy Conservation. , The Farmont Press.

Quoilin, S., M. V. D. Broek, et al. (2013). "Techno-economic survey of Organic Rankine Cycle (ORC) systems." Renewable and Sustainable Energy Reviews 22: 168-186.

Read M., Kovacevic A., et al. (2015). Comparison of organic rankine cycle systems under varying conditions using turbine and twin-screw expanders. 3rd International Seminar on ORC Power Systems. Brussels, Belgium: 1-10.

Rosset K., Mounier V., et al. (2015). Potential of small-scale turbomachinery for waste heat recovery on automotive internal combustion engines. 3rd International Seminar on ORC Power Systems. Brussels, Belgium.

Sekar R. and Cole R.L (1987). Integrated Rankine bottoming cycle for diesel truck engines. Argonne National Laboratory. Master thesis.

Shu G.Q., Yu G., et al. (2013). "Simulations of a Bottoming Organic Rankine Cycle (ORC) Driven by Waste Heat in a Diesel Engine (DE)." SAE Technical Paper 2013-01-0851.

Stobart R. and Milner D. (2009). "The Potential for Thermo-Electric Regeneration of Energy in Vehicles " SAE Technical Paper 2009-01-1333.

Stobart R., Wijewardane A., et al. (2010). "The Potential for Thermo-Electric Devices in Passenger Vehicle Applications." SAE Technical Paper 2010-01-0833. .

Teng H., Klaver J., et al. (2011). "A Rankine Cycle System for Recovering Waste Heat from HD Diesel Engines - WHR System Development." SAE Technical Paper 2011-01-0311.

Teng H., Regner G., et al. (2007). "Waste Heat Recovery of Heavy-Duty Diesel Engines by Organic Rankine Cycle Part I: Hybrid Energy System of Diesel and Rankine Engines." SAE Technical Paper 2007-01-0537.

Vázquez J., Sanz-Bobi M. A., et al. (2002). State of the Art of Thermoelectric Generators Based on Heat Recovered from the Exhaust Gases of Automobiles. Proceedings of the 7th European Workshop on Thermoelectrics, Pamplona, Spain

Wilson, D. (1986). "The Design of a Low Specific Fuel Consumption Turbocompound Engine " SAE Technical Paper 860072.

Wong C. S., Meyer D., et al. (2013). Selection and Conversion of Turbocharger as Turbo-Expander for Organic Rankine Cycle (ORC). 35th New Zealand Geothermal Workshop, Rotorua, New Zealand.

Wong S.C. and Krumdieck S. (2015). Aerodynamic evaluation of automotive turbocharger as gas turbine for Organic Rankine Cycle. Proceedings of the 2015 ASME Power & Energy. San Diego, US. PowerEnergy2015-49591.

Yang K. and Zhang H. (2015). "Performance Analysis of the Organic Rankine Cycle (ORC) System under Engine Various Operating Conditions." Journal of Clean Energy Technologies 35(5): 340-344.

Zhen L., Guohong T., et al. (2015). "Modelling and Optimisation on Scroll Expander for Waste Heat Recovery Organic Rankine Cycle." Energy Procedia 75: 1603-1608.