Radial Turbine Expander Design for Organic Rankine Cycle, Waste Heat Recovery in High Efficiency, Off-Highway Vehicles

Fuhaid Alshammari[1], A.Karvountzis-Kontakiotis[1], and A.Pesiridis[1*]

1. Brunel University London, Department of Mechanical, Aerospace & Civil Engineering, Centre of Advanced Powertrain and Fuels (CAPF)

* Corresponding author: apostolos.pesiridis@brunel.ac.uk

Abstract: Although state-of-the-art, heavy duty diesel engines of today can reach peak thermal efficiencies of approximately 45%, still most of the fuel energy is transformed into wasted heat in the internal combustion process. Recovering this wasted energy could increase the overall thermal efficiency of the engine as well as reduce the exhaust gas emissions. Compared to other Waste Heat Recovery (WHR) technologies, Organic Rankine Cycle (ORC) systems are regarded favourably due to their relative simplicity and small back pressure impact on engine performance and fuel consumption. The key elements affecting the efficiency of the ORC system are the type of working fluid selected and the design of the expander. In this simulation study, a zero-dimensional, design code has been developed to explore the impact of two, common, refrigerant working fluids on the design of a radial turbine expander. In addition, an off-design turbine analysis has been applied in order to evaluate the performance of the expander in the ORC cycle at various engine operating points. Moreover, the evaluation of ORC-diesel engine on improving fuel consumption, brake power, brake torque and exhaust gas emissions is investigated. Compared to a conventional diesel powertrain system, WHR showed an up to 5.7 % increase in brake torque and brake power and a 5.4% reduction in the brake specific fuel consumption (bsfc). The results also showed that the working fluid selection and the expander speed are critical parameters on the performance of the proposed hybrid powertrain configuration.

Keywords— diesel engines, organic Rankine cycle, radial turbine design, waste heat recovery.

1. Introduction

The transportation sector is responsible for one third of the world CO_2 emissions and approximately 15% of overall greenhouse gas emissions[1]. With respect to internal combustion engine (ICE) technology, heavy duty diesel engines in off-highway vehicles (OHV) are receiving increasing levels of attention, recently. According to [2], UK total sales on off-road construction equipment increased by 33% since 2013, making UK the largest producer of off-road construction equipment in EU. In the EU, emissions from off-road vehicles in the construction and agricultural sectors equate to 12.7% of the total on-road transport emissions (11.2% in UK). Moreover, diesel consumption from OHV sector costs the UK 10.8 £bn, annually [3, 4]. Hence, waste heat recovery systems offer opportunities to manufacturers to reduce fuel consumption and achieve lower pollutants and CO_2 emissions.

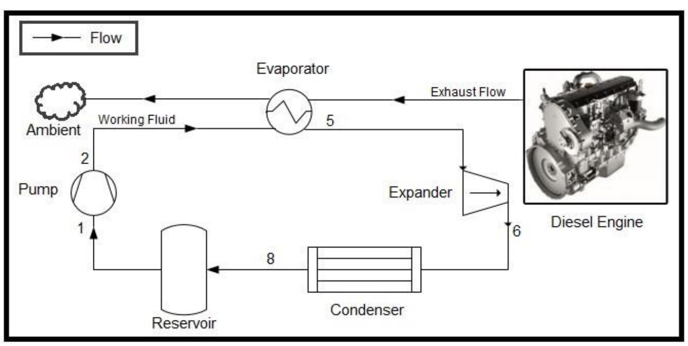
Compared to other waste heat recovery technologies, Organic Rankine Cycle (ORC) technology is probably the most promising candidate for the conversion of exhaust heat into power; this is due to its performance and practical elements of cost and ease of maintenance. Moreover, the heat exchanger of the ORC system produces less backpressure compared to technologies such as turbocharging and turbocompounding, while the thermal efficiency can reach up to 13% at maximum engine power conditions for a heavy duty diesel engine [5]. The Rankine Cycle is a closed cycle where the working fluid exchanges heat with a hot medium in an evaporator at constant pressure. The evaporated fluid then expands in an expander that produces the power output of the system. After the expansion process, the working fluid is condensed in a condenser at constant pressure, and then pumped again to the evaporator. Utilization of ORC systems for waste heat recovery in ICEs has been receiving great attention. The study of Boretti [6] proved that 6.4 % better fuel economy could be achieved when cruising at 120 km/h by implementation an ORC system. Also, Chen et al. [7] concluded that the BSFC of the diesel engine decreases by up to 6.1%.

Among the ORC system components, the expander is the most crucial and expensive component in Organic Rankine Cycle (ORC) systems [8-10]. Moreover, the properties of the working fluid and the expansion machine have significant effects on Rankine cycle thermal efficiency. Expansion machines are classified into two main groups: turbomachine and positive displacement. Selection of the appropriate expander strongly depends on working conditions, type of working fluid, space and weight restrictions and the size of the system [11]; however for waste heat recovery applications scroll expanders and radial turbines are the most common solutions in literature [12]. In applications under high pressure ratio and low to medium mass flow rate conditions such as vehicular applications, radial expanders are generally preferred. Moreover, the radial inflow expander is less expensive, lighter, and simpler in design, and doesn't need a lubrication system [13]. However, radial turbines are less efficient at part load, don't operate efficiently at variable speeds [14], on other words, their efficiencies drop when operating under off-design conditions [15].

This study presents an optimized modelling approach for the radial inflow turbine where the input parameters are optimized to deliver a turbine with superior performance. In order to predict the performance (off-design) of the optimized turbine under steady state conditions, the geometry of the turbine is imported into a commercial turbomachinery simulation software (Rital) [16] where these results can be calculated. This off-design model can be used in the first stage of the design process to decide whether a certain approach is delivering the expected results before moving to the detail design of the CFD. Furthermore, the impact of the optimized radial inflow expander on the ORC system for an off-highway vehicle is presented using two working fluids, R245fa and R123. To achieve this, an integrated in-house model has been developed which includes the engine map operating points from a heavy duty diesel engine, the ORC model and the proposed radial expander model. In order to evaluate the potential benefit of the ORC system on diesel engine fuel consumption, brake power and NOx emissions, the model was employed at various engine load and speed operating points.

2. Powertrain Modelling Approach

The proposed integrated powertrain model is schematically presented in Figure 1.



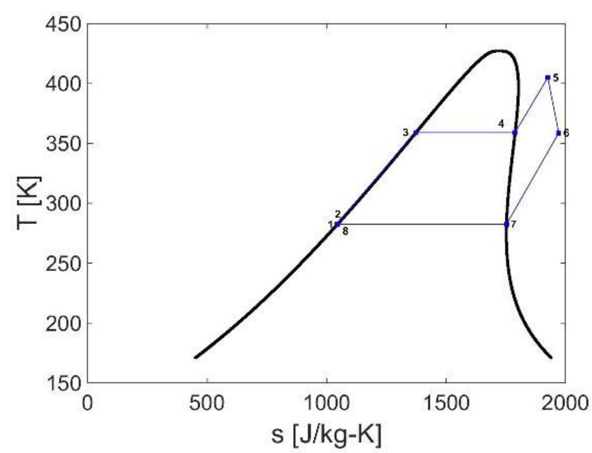


Figure 1: (Left) Schematic Representation of the ORC Powertrain; (Right) Schematic representation of the ORC cycle

The model integrates the exhaust thermodynamic conditions predicted by the engine simulation and the ORC modelling which are both presented below in more detail. The powertrain modelling results include the optimum ORC efficiency and the net ORC power output for various engine operating conditions.

2.1 Engine Modelling

The engine mode was based on a 10.3ℓ heavy duty diesel engine the basic characteristics of which are given in Table 1. This version of the engine fulfils the Euro 3 emission standards and it is equipped with a variable geometry mixed flow turbocharger and a common rail injection system, while its maximum engine power is 316 kW at 2100 rpm. The high engine capacity appears to be a reasonable choice to apply a waste heat recovery system to, due to the high exhaust flow enthalpy.

The modelling of this engine was performed using a commercial engine simulation software, in order to calibrate the engine model on the experimental given fuel consumption. The final calibrated engine model calculates the fuel consumption, the exhaust waste heat as well as the engine NOx emissions, by using the well-known extended Zeldovich mechanism.

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	BTDC
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/1200 rpm	

2.2 ORC System Optimization

An in-house MATLAB code has been developed for the thermodynamic modelling and optimization 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 Figure 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 were optimized by the in-house thermodynamic ORC code, using as objective function the cycle thermodynamic efficiency that fulfils the constraints (equation 3). 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 the 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 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.

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

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

Finally, the overall ORC efficiency is given by:

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

2.3 Turbine Modelling

The expander modelling considers the turbine volute, stator and rotor. For simplicity, a circular cross section was assumed in the volute. Since the volute is responsible for the turning of the fluid, the aim of the stator vanes is to accelerate the flow and deliver it to the rotor inlet uniformly and at the correct angle. Hence, uncambered vanes were assumed in this model.

The convergence flowchart for the expander design methodology is briefly described in Figure 2. At the beginning an isentropic efficiency is assumed, then the actual enthalpy drop on the rotor, stator and volute are calculated. After the calculation of the losses the new isentropic efficiency is calculated. The loop ends when the estimated and the calculated isentropic efficiency converge. The whole modelling approach is based on the design procedure of Baines et al. [17].

In order to start with this procedure, a set of input parameters are required at the beginning of the design method. The thermodynamic parameters such as total inlet temperature and pressure, pressure ratio, and mass flow rate are imported from the ORC thermodynamic model in this paper. The aforementioned parameters are shown in Table 2. However it has to be mentioned that the results of the expander design code have to fulfil some geometric constrains which are briefly described in Table 3.

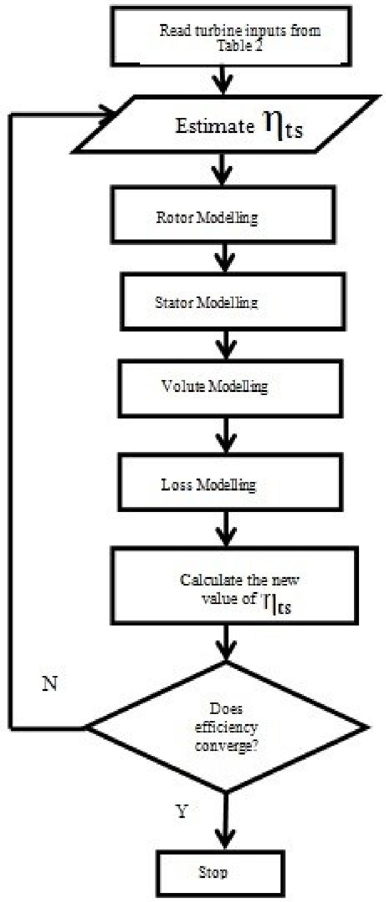


Figure 2: Flow chart of the expander design procedure.

Table 2: Input Parameters of the Turbine model

Parameters		R245fa	R123
Total inlet Temp, To1	K	319 - 374	337 - 405
Total inlet Press, Po1	bar	3.0-10.0	3.0 - 10.0
Pressure ratio, PR	-	3.0 - 10.0	3.0 - 10.0
Mass flow rate, m	Kg/s	0.28 - 0.58	0.21 - 0.65
Loading Coefficient, Ψ	-	0.8 - 1	0.8 - 1
Flow Coefficient, Φ		0.2 - 0.5	0.2 - 0.5
Rotor exit r5h/r4	-	0.2 - 0.3	0.2 - 0.3
Rotational speed, N	RPM	40k - 60k	40k - 60k
Nozzle inlet r2/r3	-	1.2 - 1.3	1.2 - 1.3
Flow blockage factor	-	0.1	0.1

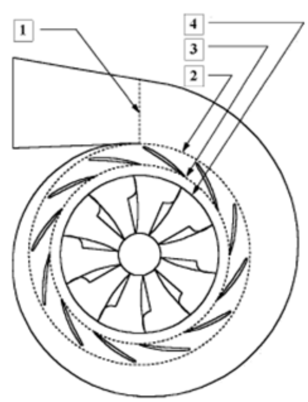


Figure 3: Schematic Representation of the Radial Turbine Stage

Finally, an off-design procedure was adopted in this study. The isentropic efficiency of the expander is not always the same under varied thermodynamic inlet flow conditions. The latter has an impact on the performance of the total ORC system. For the off-design analysis of the expander a commercial tool was utilized (Rital [16]).

Table 3: Constrains of the expander design code

Parameter	Unit	value	Explanation
Rotor inlet diameter, d4	m	> 0.02	The minimum value considered for manufacturability.
Inlet blade height, b4	m	> 0.001	The minimum value considered for manufacturability.
Inlet absolute flow angle, α4	degree	< 80	To achieve reasonable number of blade and minimize associated losses
Radius ratio, r5tip/r4	-	< 0.85	To prevent excessive tip curvature

3. Results

3.1 Engine Modelling:

A heavy duty diesel engine model has been developed in this study, using a commercial engine simulation tool. The model was calibrated at three (3) different load/speed operating points, such as the optimum bsfc engine operating point, the maximum engine torque and maximum engine power. The engine model calibration was based on experimental fuel consumption values obtained in literature [18]. Table 4 briefly presents the engine simulation values, regarding fuel consumption and exhaust gas conditions.

Table 4: Brief presentation of the engine simulation values

Parameters	Bsfc point	Max Torque	Max Power
Engine Speed [rpm]	1300	1400	2100
Bmep [bar]	18.00	23.00	18.00
Power [kW]	201	277	325
Torque [Nm]	1476	1887	1477
Bsfc [g/kWh]	192	194	212
Exhaust Mass Flow Rate [kg/s]	0.30	0.39	0.46
Exhaust Temperature [K]	681	719	804

3.2 Design and Off-Design Analysis of the Radial Inflow Turbine

The design of the ORC system is performed on the maximum engine power operating point, as this point is characterized by the higher content of exhaust waste heat, which can potentially lead to high inlet pressure and temperature for the expander and hence higher cycle efficiency. Initially, the proposed ORC thermodynamic model was utilized for the calculation of the optimum ORC parameters regarding the organic fluid mass flow and the pressure ratio by assuming an isentropic efficiency for the expander equal to 0.8. Then, the operating conditions of the expander were further optimized; these are the flow coefficient, the loading coefficient, the expander rotational speed and some basic geometric parameters described in Table 2. The optimization of the expander returns the real isentropic total to static efficiency which was fed to the ORC thermodynamic model to calculate the new ORC thermal efficiency. After a few iterations, the two models converge and the final values are calculated.

Table 5: Expander optimized geometry calculated from the expander design code for the design point

Parameter	Units	R245fa	R123
Diameter d_2	[mm]	105	94
Diameter d_3	[mm]	81	72
Diameter d_4	[mm]	77	69
Diameter d_5	[mm]	38	37
Diameter $d_{\{5,tip\}}$	[mm]	60.6	59
Diameter $d_{\{5,hub\}}$	[mm]	15.4	14
Blade Height b_4	[mm]	3.8	3
Blade Height b_5	[mm]	22.6	23
Number of rotor blades	[-]	16	14
Reaction	[-]	0.41	0.512
Specific speed, Ns	[-]	0.486	0.625
Total to Static Efficiency	[-]	0.758	0.753
Turbine power output	[kW]	17.68	18.6

The optimized expander geometry and its performance are described in Table 5. The results show that both examined fluids present almost the same total-to-static isentropic efficiency, but R123 presents approximately 1kW higher expander power output and an almost 10% more compact expander size. In fact, the two tested R245fa and R123 organic fluids are considered as dry fluids which are favourable to avoid fluid droplets at the turbine blades as these droplets may result in corroded blades. According to Wang et al [19], R123 shows better thermodynamic performance, which is also validated in this study. In terms of ORC thermal efficiency, R123 presents 0.7% higher thermal efficiency compared to R245fa. On the other hand, in many studies [20, 21] is suggested that R245fa is also promising fluid because of its environmental friendly and economic characteristics.

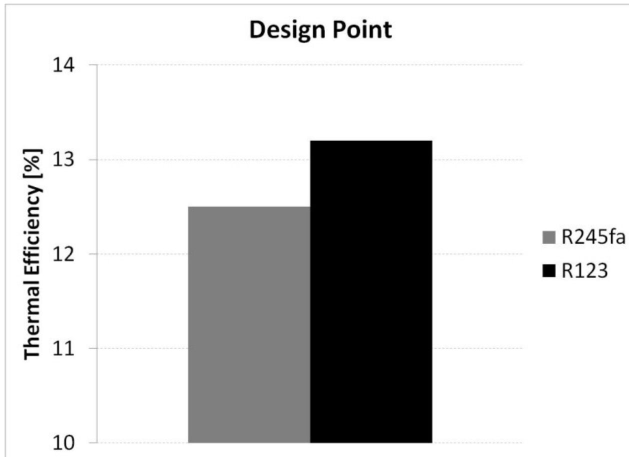


Figure 4: ORC thermal efficiency at design point (Engine MaxPower).

After the design-point result analysis that is presented above, an off-design method was utilized in order to explore the performance characteristics of the expander and the ORC system at different engine operating conditions. This corresponds to the maximum engine torque point and the minimum bsfc engine point conditions. The off-design analysis of the expander was performed for three different rpm cases: 41000rpm, 52000rpm and 63000rpm. The expander geometry was kept constant and was imported from the design procedure.

Figure 5 illustrates the off-design analysis of the expander efficiency and shaft output for the two tested working fluids. In the same figure is also illustrated the expander design point as a reference. It is observed that the expander total to static isentropic efficiency and its power output are highly related to the rotational speed and the velocity (U/C) ratio. Small changes of the inlet thermodynamic conditions can lead to significant variations of the expander performance. It is also presented that maintaining the expander rotational speed in the design speed limits, isentropic efficiency slightly changes but power output drops at a more substantial rate.

The expander map presented in Figure 5 was utilized to predict the ORC system efficiency under various engine operating conditions. Table 6 summarizes the thermodynamic conditions and the optimum operating conditions for both the expander and the ORC system at various engine operating conditions and for the two tested fluids. It is illustrated that ORC efficiency achieves its maximum efficiency value at maximum torque while the bsfc engine operating point efficiency drops by 40% for both fluids. R123 shows consistently superior performance compared to R245fa by almost 5%. The reason that the ORC efficiency is higher at the maximum torque operating point is related to the performance of the evaporator which in this study is assumed ideal without any losses. However the general trend is that ORC characteristics do not change significantly at full torque and power conditions.

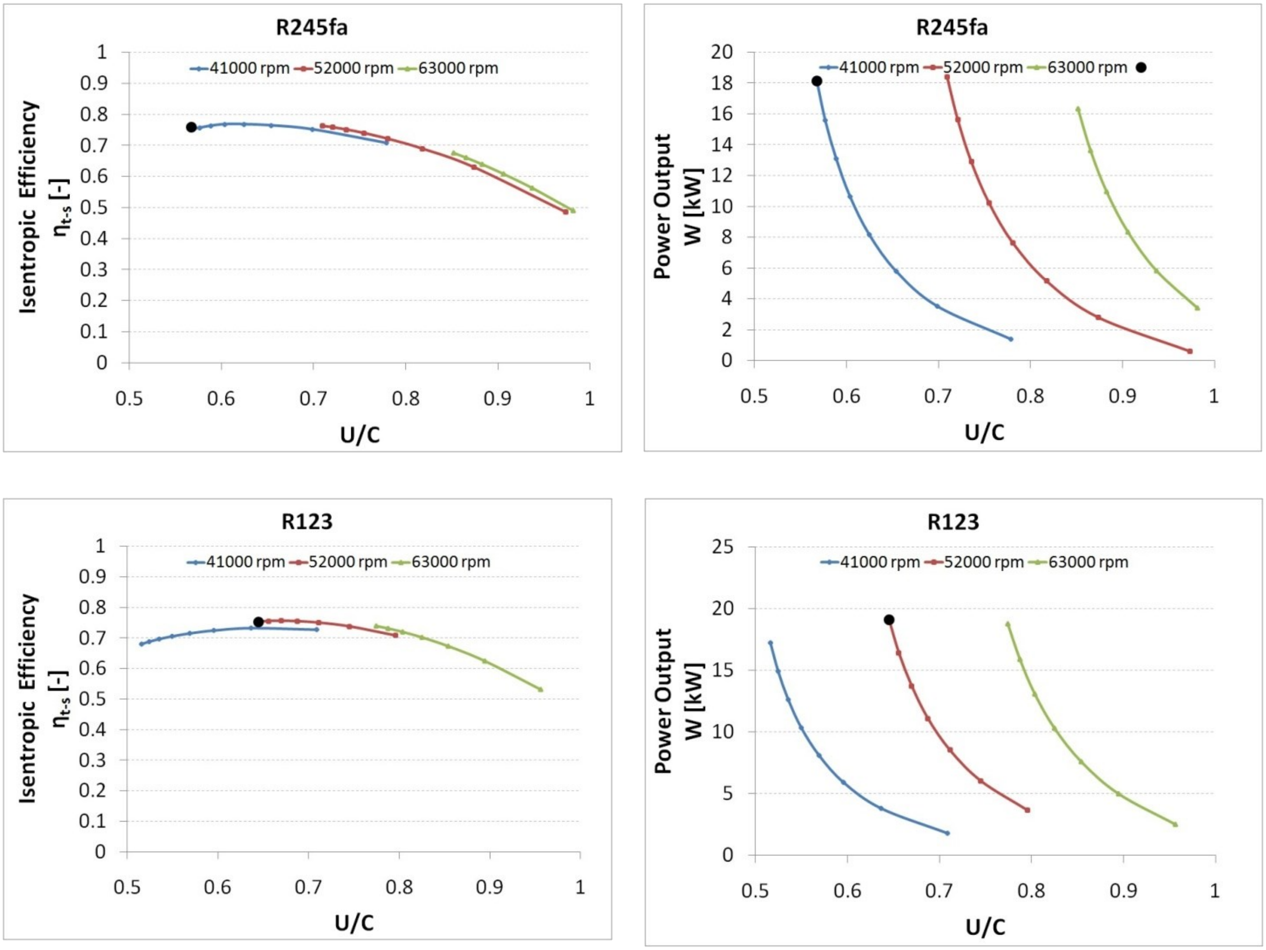


Figure 5: Off-design analysis for R245fa and R123 with the reference of the design point.

Another important observation from Table 6 is the performance of the expander under off-design conditions. It is clear that the expander efficiency doesn't change significantly; however the power output does significantly change at the three tested engine operating points. The main reason for this observation seems to be related with the available extracted heat from the evaporator. Although the expander efficiency doesn't change significantly, the available heat is lower and therefore the available heat to be converted into useful shaft power is lower.

Table 6: Optimum Cycle Parameters for Different Engine Points for R245fa and R123.

Parameters	R245fa			R123		
	BSFC	**MaxTorque**	**MaxPower**	**BSFC**	**MaxTorque**	**MaxPower**
To1 (K)	320	375	375.1	338	401	405
Po1 (bar)	3.14	10	10	3.25	10	10
PR (-)	3.14	10	10	3.25	10	10
$\dot{m}$ (kg/s)	0.28	0.34	0.54	0.32	0.43	0.60
N (RPM)	41000	41000	41000	52000	52000	52000
$\dot{W}_{net}$ [kW]	4.30	11.4	17.68	4.54	13.3	18.60
$\dot{Q}_{evap}$ [kW]	61.88	89.70	141.10	61.87	100.78	141.10
η_{ORC} [%]	6.96	12.70	12.53	7.35	13.21	13.18
$\eta_{\exp ander}$ [%]	76.60	76.82	75.80	74.50	75.50	75.30
η_{pump} [%]	75.00	75.00	75.00	75.00	75.00	75.00

3.3 ORC-equipped powertrain

One of the challenges facing the manufacturers of off-highway vehicles is to meet the CO_2 and exhaust emissions standards. At the same time, ORC system can also improve fuel consumption and increase powertrain power. One of the main targets of this paper is to investigate the potential for improvement of internal combustion engines by implementing an ORC system that has an efficient expander. In this study, the benefits of the ORC system on the performance and fuel consumption of the engine are investigated, while the effects of the choice of working fluid are explored.

Figure 6 presents the improvement of the brake specific fuel consumption (bsfc) for both R245fa and R123 working fluids at different engine operating points.

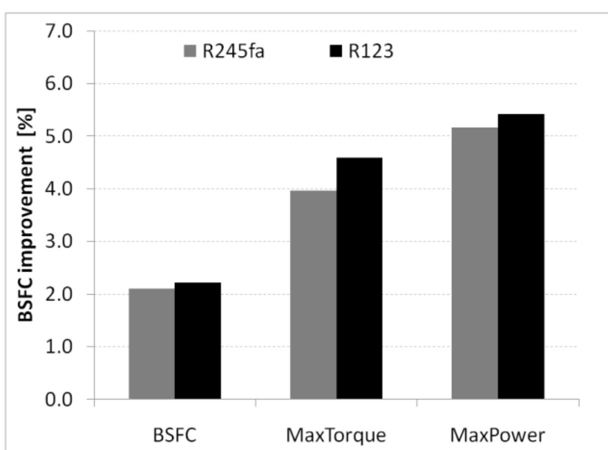


Figure 6: Impact of the ORC system on brake specific fuel consumption (bsfc).

At the BSFC engine operating point, the ORC system improves bsfc by 2.09% when using R245fa as working fluid and by 2.21% when using R123.

The potential benefit of the ORC system on fuel consumption is even higher the when engine works at full load conditions. At engine maximum torque and engine maximum power brake specific fuel consumption can be improved by 3.95% and 5.16% when R245fa is applied while R123 offers even higher efficiency equal to 4.6% and 5.4% respectively. It also has to be mentioned that these numbers are directly related with the CO_2 reduction of the proposed powertrain unit.

The effect of the ORC system on the integrated powertrain power was, also, investigated. As can be seen in Figure 7, the trends for the improvement on engine power are almost the same as with the bsfc results. At the BSFC engine operating point, the powertrain power output was increased by 2.14% or 2.26%, when the employed working fluid was either R245fa or R123, respectively.

At the maximum torque engine operating point, it was found that the available torque is increased by 4.12% in the case of R245fa and by 4.8% in the case of R123. Last but not least, the maximum power of the integrated unit increased by 5.45% when R245fa was utilized and 5.75%% when R123 was employed.

Finally, the 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. Figure 8 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 through exhaust thermal management. This can be done by assisting the SCR catalyst to operate within the temperature range of 350^o-450^oC by controlling the available thermal power of the exhaust gas.

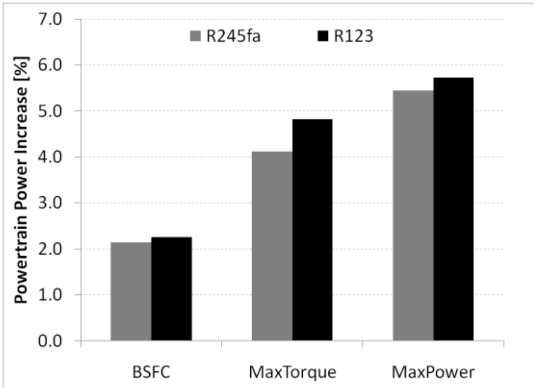


Figure 7: Impact of the ORC system on the powertrain power output.

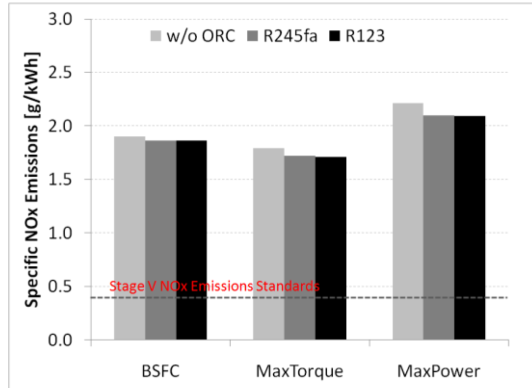


Figure 8: Contribution of the ORC system on specific NOx emissions reduction

4. Conclusions

In a number of ORC system-level studies, the efficiency of the expander is considered to be constant for all of the operating points and with different working fluids. In this study, this assumption is shown to be insufficient for accurate modelling of system efficiency and management of the expectations of what is achievable from such a technology. For this analysis, an in-house design and optimization code was employed and briefly presented as was a commercial turbomachinery analysis software to collectively analyse on- and off-design cases. According to the analysis presented, the expander efficiency changes substantially depending on the input conditions and the fluid used.

It was also shown that the cycle thermal efficiency is highly affected by the engine operating conditions. At high engine speeds, the cycle shows improved performance due to the higher temperature available and hence the higher quantity of the fluid being evaporated. Also, a trade-off was noticed between the cycle and expander efficiencies in terms of fluid selection. Generally, R123 showed better cycle thermal efficiency, while R245fa showed better expander efficiency.

Overall, the goal of this study was to investigate how ORC systems can improve diesel engine performance and reduce exhaust emissions, through a selected parametric study of key parameters. The study showed that the engine power can be increased by up to 5.7%. Moreover, the results proved that ORC systems can have a significant impact on fuel consumption reduction. The results showed that a decrease of up to 5.44% both for fuel consumption and NOx emissions.

References

[1] ODEC, "Reducing transport greenhouse gas emissions, Trends & Data," 2010.

[2] S. Popp. (March 2015). *Constructing the Europe of Tomorrow*. Available: http://www.oemoffhighway.com/article/12050035/the-european-construction-equipment-market

[3] E. Georges, S. Declaye, O. Dumont, S. Quoilin, and V. Lemort, "Design of a small-scale organic Rankine cycle engine used in a solar power plant," *International Journal of Low-Carbon Technologies,* 2013.

[4] A. Krasenbrink and A. Dobranskyte-Niskota, "2007 Technical review of the NRMM Directive 1997/68/EC as amended by Directives 2002/88/EC and 2004/26/EC," European Commission Joint Research CenterSeptmeber, 2008.

[5] R. Sekar and L. R. Cole, "Rankine bottoming cycle for diesel truck engines," Master thesis, Argonne National Laboratory, 1987.

[6] A. A. Boretti, "Transient operation of internal combustion engines with Rankine waste heat recovery systems," *Applied Thermal Engineering,* vol. 48, pp. 18-23, 2012.

[7] C. Bei, H. Zhang, F. Yang, S. Song, E. Wang, H. Liu, Y. Chang, H. Wang, and K. Yang, "Performance Analysis of an Evaporator for a Diesel Engine–Organic Rankine Cycle (ORC) Combined System and Influence of Pressure Drop on the Diesel Engine Operating Characteristics," *Energies,* vol. 8, 2015.

[8] S. Wong, D. Meyer, and S. Krumdieck, "Selection And Conversion Of Turbocharger As Turbo-Expander For Organic Rankine Cycle (Orc)," presented at the 35th New Zealand Geothermal Workshop, Rotorua, New Zealand, November 2013.

[9] K. George, M. Dimitris, and P. George, "Experimental testing of a small-scale supercritical orc at low-temperature and variable conditions," presented at the 3rd International Seminar on ORC Power Systems, Brussels, Belgium, October, 2015.

[10] D. E. Wilson, "The Design of a Low Specific Fuel Consumption Turbocompound Engine," *SAE International,* 1986.

[11] T. Wang, Y. Zhang, Z. Peng, and G. Shu, "A review of researches on thermal exhaust heat recovery with Rankine cycle," *Renewable and Sustainable Energy Reviews,* vol. 15, pp. 2862-2871, 2011.

[12] K. Rosset, V. Mounier, E. Guenat, O. Pajot, and J. Schiffmanm, "Potential Of Small-Scale Turbomachinery For Waste Heat Recovery On Automotive Internal Combustion Enginges," presented at the 3rd International Seminar on ORC Power Systems, Brussels, Belgium, October, 2015.

[13] A. P. Weib, "Volumetric Expander Versus Turbine – Which Is The Better Choice For Small ORC Plants," presented at the 3rd International Seminar on ORC Power Systems, Brussels, Belgium, October, 2015.

[14] N. Petchers, *Combined Heating, Cooling & Power Handbook: Technologies & Applications: An Integrated Approach to Energy Conservation*: The Fairmont Press Inc., 2003.

[15] H. Teng, G. Regner, and C. Cowland, "Waste Heat Recovery of Heavy-Duty Diesel Engines by Organic Rankine Cycle Part I: Hybrid Energy System of Diesel and Rankine Engines," *SAE International,* 2007.

[16] (2012). *Concepts NREC*. Available: http://www.conceptsnrec.com

[17] H. Moustapha, M. Zelesky, N. Baines, and D. Japikse, *Axial and Radial Turbines*: Concepts ETI, Inc. d.b.a. Concepts NREc, 2003.

[18] G. Biaggini and W. Knecht, "The Advanced Iveco Cursor 10 Heavy Duty Truck Diesel Engine," in *Seoul 2000 FISITA World Automotive Congress. Seoul, Korea. 2000-05-0071*, 2000.

[19] E. H. Wang, H. G. Zhang, B. Y. Fan, M. G. Ouyang, Y. Zhao, and Q. H. Mu, "Study of working fluid selection of organic Rankine cycle (ORC) for engine waste heat recovery," *Energy,* vol. 36, pp. 3406-3418, 2011.

[20] C. R. Nelson, "Application Of Refrigerant Working Fluids For Mobile Organic Rankine Cycles," presented at the 3rd International Seminar on ORC Power Systems, Brussels, Belgium, October 12-14, 2015.

[21] S. Masheiti, B. Agnew, and S. Walker, "An evaluation of R134a and R245fa as the working fluid in an Organic Rankine Cycle Energized from a Low Temperature Geothermal Energy Source," *Journal of Energy and Power Engineering 5 (2011) 392-402*.