CONCEPTUAL DESIGN AND PRELIMINARY TESTING OF AN ORGANIC RANKINE CYCLE THERMAL ARCHITECTURE

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ABSTRACT

Waste heat recovery is a key path in improving the overall thermal efficiency, and hence, reducing CO_2 emissions in the mid to large scale internal combustion engines. However, realisation of the cost-effective deployment of Organic Rankine Cycles (ORC) are hindered by several key factors. Amongst these are the, utilisation of low-grade ORC practice for high-grade applications, disconnect between parameters considered in simulation studies to those demonstrated experimentally, and integrating multiple heat recovery sources.

To investigate and address these challenges, a programme of 'concept-to-demonstration' is in progress at the University of Brighton. This paper describes some of the key features of a new ORC test facility that can contribute towards reduced system costs and increased overall conversion efficiency. These features include, firstly, a variable heat source setup, allowing the potential to replicate a wide range of realistic gaseous heat source quality and quantity levels. Secondly, the direct utilisation of the High-Temperature (HT) exhaust gases, which is expected to reduce the overall system cost when compared to a system utilising an intermediate thermal-oil loop. Thirdly, deployment of HT blends, this is estimated to increase the overall conversion efficiency when compared to a system employing a conventional organic working fluid. Fourthly, a flexible thermal architecture, offering a dual source heat recovery for effective heat utilisation and internal heat recuperation for increased thermal efficiency. Finally, the HT and high-pressure cycle operating capability, offering a near-optimal process condition. The potential benefits of the above features are quantified using a combination of literature survey, simulation results and experimental measurements. The paper concludes with a brief overview of the research direction intended to be undertaken in the next phase of the work.

INTRODUCTION

Heavy Duty (HD) Diesel engines are the most common prime mover in road, marine and rail freight transportation. Due to the high absolute fuel consumption, they represent a significant challenge in terms of CO_2 emissions reduction. A key global imperative is therefore the substantial improvement of HD engine efficiency. Numerous technology road maps identify a specific need for waste heat recovery to target the portion of the lost fuel energy (Stanton 2013). The heat to power conversion technologies are additionally of particular interest to the process industry and the renewable sector.

Waste heat recovery on HD Diesel engines has been demonstrated using various methods (Saidur et al, 2012). Amongst these, Organic Rankine Cycles (ORC) are being preferred when considering overall conversion efficiency and technology readiness level. Seher et al, 2012 showed two-stage turbine and piston expander mechanical efficiencies in the region of 65-85%, while Yang et al, 2015 and Zhang et al, 2014 have demonstrated prototypes of fin-tube and spiral-tube evaporators, respectively. Adaptability of ORCs has also led to the proposition of ORCs combined with other technologies. Shu et al, 2012 analysed a thermoelectric generator + ORC system to recover the coolant and exhaust heat. Additionally, Pandiyarajan et al, 2011 demonstrated the ability to store a noticeable level of fuel energy in the combined storage system.

Despite the above advancements, there still exist some key challenges hindering the cost-effective deployment of ORCs. Extending upon the earlier simulation studies performed using HYSYS V8 and AMESim V12 simulation environments (Panesar 2015), this paper summarises the 'concept-to-demonstration' of an ORC experimental facility which can contribute towards addressing these challenges. The paper consists of two main sections, the methodology phase and the demonstration phase, and finally concludes with an overview of the next stage of the work. Utilising literature, simulation (HYSYS V8) and experimental findings, the factors relating to the heat source, the heat exchange, the working fluid, the thermal architecture and the process operating conditions are discussed.

METHODOLOGY PHASE

Heat source setup

Figure 1a presents the range of heat quality and proportion of the fuel chemical energy wasted by the typical mid to large scale (0.1-1 MW) Diesel engines utilising a range of different regulated emissions strategies (Panesar 2015). Such engines are typically operated at near-steady state conditions, between the mid-speed mid-load point and the engine rated conditions. The continued trend of cooler engine intake temperatures and higher engine intake pressures now means that, from an exergy perspective, the charge air is a potential source of waste heat alongside the exhaust and coolant heat. From figure 1a, it is evident that a suitable heat source setup for testing ORCs must offer a wide range of gaseous heat quality and quantity levels.

To offer this flexibility, while avoiding the challenge of integrating and operating an engine test-bed in parallel, the waste heat was experimentally simulated using a compressor (by Elmo Rietschle; side channel blower) and burner (by Maxon.; TUBE-O-THERM) combination, as shown schematically in figure 1b. The compressed air flow was distributed into two streams, the first stream acted as the main air supply to the gas burner, and the second stream was used for dilution purposes. This was since, firstly, the gas stream exiting a gas burner is typically well above the maximum target temperature of 500°C. By using the secondary compressed air stream, the exhaust gas temperature can be reduced. Secondly, the gas burner must be accompanied with a higher pressure air stream since ambient gas burners cannot operate when faced with the backpressures that will be introduced by the exhaust heat exchangers (HEX). As a result, depending on the fuel supply to the burner, the proportion of diluted air and the backpressure, the heat source setup has the potential to deliver gaseous heat qualities between 200-500°C and heat quantities between 12-120 kW, at a pressure of 1.05-1.25 bar.

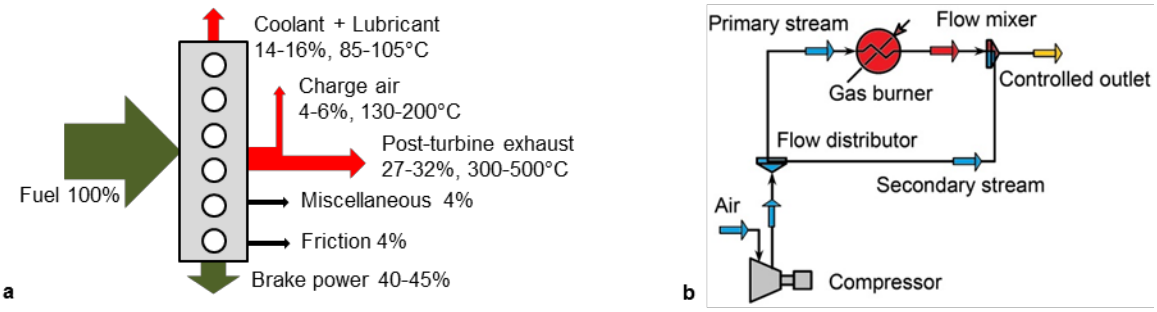


Figure 1: (a) Fuel energy distribution in the mid to large scale engines at relatively higher speeds and loads, (b) Schematic of the experimental heat source setup

Utilisation of exhaust gases

To recover the High-Temperature (HT) exhaust heat, an intermediate thermal-oil loop is often proposed (Shu et al, 2013). As shown in figure 2, the diathermic oil recovers the HT exhaust heat, and transfers it to the ORC working fluid. Similar process integration was proposed by Shu et al, 2013 for vehicle engine heat recovery using conventional refrigerants like R245fa and isopentane. Due to the reduced heat source temperature, this approach has two key advantages. Firstly, a reduced risk of thermal degradation in the working fluid, and secondly, utilisation of conventional refrigerants and off-the-shelf components. In addition, the oil loop may allow recovering heat from multiple sources, and transferring it to a single HEX. If the pressure and temperature limits of the ORC do not change with the addition of the oil loop, and the maximum available waste heat is transferred to the oil, then the system power of the ORC will not reduce noticeably. This is because, the total heat transfer losses in the heat recovery process are divided between the two HEXs.

Unfortunately, the oil loop also introduces a number of disadvantages. The additional sub-system components include the exhaust-oil HEX, the oil tank and the oil pump. Therefore, for roughly equal power, the system complexity, size, weight and potential failure points will increase. Furthermore, information gathered during the pre-procurement research stage and literature survey indicated that the cost/kW ratio of a Low-Temperature (LT) ORC coupled with an oil loop can be 15-20% higher compared to the HT ORC (Guillen et al, 2011). Thermodynamically, the oil loop fails to provide any opportunity to reduce the overall system irreversibility. Fundamentally, such systems cannot take full advantage of the high-grade waste heat, and in fact, mimic low-grade heat recovery systems. For the above drawbacks, the oil loop approach was excluded as a potential solution. Hence, efforts were focused in engaging the process industry supply chain in designing a HEX for direct exhaust and working fluid use. As a result, after several iterations, a shell-and-plate HEX manufactured by Vahterus was procured with the capability to operate at high temperatures and pressures.

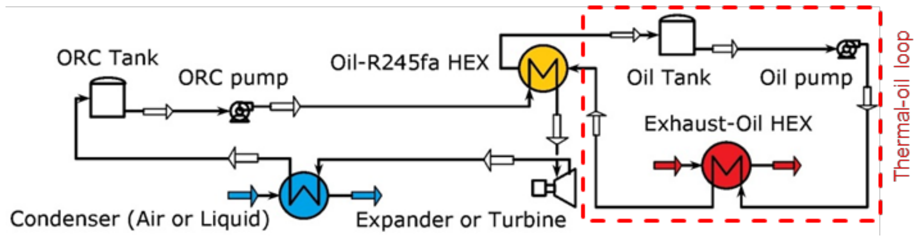


Figure 2: Schematic of a thermal-oil loop coupled with a conventional ORC

Working fluid

Waste heat recovery below 200°C is considered mature using Hydrofluorocarbons such as R245fa (Sprouse and Depcik, 2013). Hence, these fluids have also been suggested for HT exhaust heat recovery (either directly or coupled with an oil loop). However, from an exergy perspective, they result in higher irreversibilities, and hence, lower overall conversion efficiencies.

To investigate and address this issue, a two-part simulation study was undertaken, firstly to identify suitable working fluids, and secondly to propose a flexible thermal architecture. Recent simulation studies have indicated that, water blends can provide an improved case for 300-500°C heat recovery (Panesar 2015). Amongst the numerous water blends, ethanol-water and propanol-water blends (mass fraction 50% each), were considered suitable in view of thermodynamic, thermophysical, chemical, environmental, safety, cost, availability, compatibility, miscibility and decomposition properties.

Utilising the parameters, boundary conditions and assumptions presented in table 1, figure 3 presents the theoretical net ORC power for R245fa and the chosen alcohol-water blends for the 5°C superheated expansion with respect to (a) maximum cycle pressure and (b) maximum cycle temperature. For R245fa operation, a near-critical pressure (≈ 35 bar) was considered optimal, offering 2.7 kW of net power for 57 kW of heat recovery (figure 3a). At the same maximum pressure, the alcohol-water blends offered a 2.3 times improvement in the net power for 48 kW of heat recovery. Furthermore, the 30-40 bar pressure limit was considered near-optimal for the two blends as improvement in recovered work was negligible above 40 bar. However, compared to R245fa, which required a maximum temperature below 150°C, the alcohol-water blends required a much higher value of 230°C (figure 3b) due to the relatively higher normal boiling points (82-89°C vs. 15°C).

Table 1: Parameters, boundary conditions and assumptions representative of truck exhaust heat recovery

Parameters	Values	Parameters	Values
$T_{exhaust}$	400°C	$T_{pinch-point}$	30°C (exhaust HEX)
$T_{charge\ air}$	140°C	$T_{pinch-point}$	10°C (charge air, IHE)
$\dot{m}_{exhaust}$	0.2 kg/s	$T_{condensing}$	90°C
$c_{p\ exhaust}$	1.15 kJ/kg°C	$T_{sub-cooling}$	5°C
$T_{cooling\ air}$	35°C (inlet), 55°C (exit)	$T_{superheat}$	5°C
η_{pump}	50%	$\Delta P_{all\ HEXs}$	0.1 bar
$\eta_{expander}$	65%		

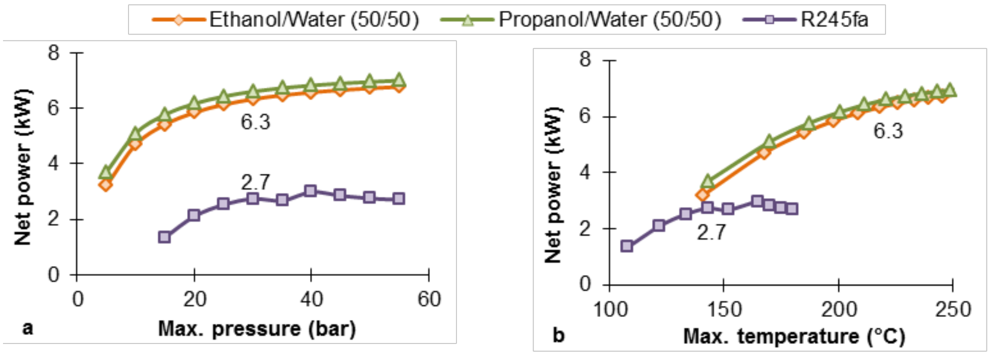
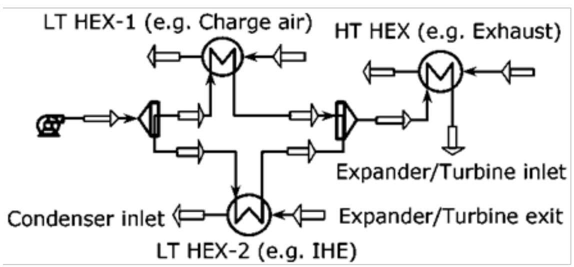


Figure 3: Comparison of theoretical net power between R245fa and alcohol-water blends at (a) Maximum cycle pressure, and the corresponding (b) Maximum cycle temperature

Thermal architecture

The thermal architecture of ORCs are expected to differ amongst the variety of HD Diesel engine applications (Sprouse and Depcik, 2013). Hence, a flexible experimental platform with the potential to be adapted for either multiple heat recovery and/or efficient heat utilisation is needed. Figure 4 presents the schematic of the proposed heat recovery architecture which comprises of two LT HEXs in a parallel branched flow followed by one HT HEX. The tabulated information summarises the key results for comparing the three possible heat recovery options when using propanol as the working fluid. All the parameters were normalised to the net heat recovered in option 1, which corresponded to exhaust heat recovery only. Additionally, the three options were targeted for the same net power (at 12%) and with an equal maximum cycle pressure (20 bar).

It can be noted that in option 2, which corresponds to series charge air and exhaust heat recovery, the exhaust HEX duty was lowered by 14% for the same level of power output. This was since, the lower temperature exhaust heat recovery was replaced with the charge air heat recovery, which was already a load on the engine cooling module. Finally, option 3 may be considered for exhaust heat recovery only, but in demanding condenser packaging applications. In this option, the working fluid was slightly superheated compared to options 1 and 2 (from 215 to 250°C). The level of superheating paired with the drying nature of the working fluid allowed the use of an Internal Heat Exchanger (IHE) to partially recover the exergy that may be lost in the condenser. Due to the internal heat recuperation, the exhaust heat recovery was reduced by 13%, but more importantly, the condenser heat rejection was lowered by 15%.



IN USE / NOT IN USE

Option	HT HEX	LT HEX-1	LT HEX-2	Fluid	Net power (kW)	Exhaust (kW)	Charge air (kW)	IHE (kW)	Condenser (kW)
1	IN USE	NOT IN USE	NOT IN USE	Propanol	12	100	0	0	88
2	IN USE	IN USE	NOT IN USE	Propanol	12	86	14	0	88
3	IN USE	NOT IN USE	IN USE	Propanol	12	87	0	14	75

Figure 4: Schematic of the experimental heat recovery architecture and normalised comparison of the architecture options

In addition to the working fluid and the thermal architecture studies, a range of parametric studies using alcohol-water blends, alcohols and water were also conducted. The overall range of the simulation results were translated into: system specification and layout, process and instrumentation diagram, and procurement plan. These acted as the design reference for the ORC facility. Furthermore, due to the prototype experimental nature of this project, control and instrumentation, hazard and operability, and risk assessment studies were undertaken.

INITIAL DEMONSTRATION PHASE

ORC experimental facility

Figure 5 presents the ORC facility at its current status. The overall facility is not aimed at demonstrating the power/density ratio of a mature commercial system, but is rather aimed at demonstrating HT ORC operation, conducting working fluid research and evaluating components. The relatively large foot-print of the facility is due to the fact that the balance of the plant was sourced from the process industry to achieve the target thermal and pressure rating. Furthermore, the facility is built with the aim of testing different thermal architectures and characterising components of varying capacities from different sectors.

The facility comprises of three primary loops: the heat source, the heat sink and the working fluid loop. In addition, two secondary loops exist, one relating to the chiller, and the other relating to the expander auxiliaries. The major components for the ORC loop include: pump-motor, 3-way flow control valve, needle valves, flow meters, LT HEXs, HT HEX, 3-way flow by-pass valve, condenser, liquid receiver and chiller. Furthermore, the outlet line from the tank includes a filter dryer, a sight glass and a sampling line with bottle. The sampling line is included to collect samples to quantify thermal degradation in alcohols and alcohol-water blends after appropriate time, pressure and temperature exposures. Additionally, the outlet line from the pump includes a pressure relief valve and a pulse dampener.

The exhaust loop can be arranged to give either exhaust only or exhaust + charge air heat recovery options. For the heat recovery architecture presented in figure 4, a further option is possible in which both the LT HEXs are utilised. To simulate this option, the 3-way flow control valve at the outlet of the pump can be utilised for flow distribution into the two LT HEXs and the flow rate can be monitored using the two branched flowmeters.

A liquid cooled plate HEX was utilised as the condenser. This was since, plate HEXs can withstand a wider range of condensation pressures and temperatures. As a result, condensing characteristics relevant to both stationary and transport applications can be simulated. The tank was modified to receive the chiller cooling loop as a safety measure and as a means to draw down lower boiling point fluids from the system.

The pressure limit of the re-fabricated tank, paired with the temperature limit of the sight glass provided the limit of pressure and temperature capability of $\leq$ 9 bar and $\leq$ 70°C at the low-pressure LT side. Similarly, the pressure limit of the HT HEX, paired with the temperature limit of the HT gasket provided the limit of pressure and temperature capability of $\leq$ 40 bar and $\leq$ 250°C at high-pressure HT side. The ORC loop pipework was continuous orbit welded to offer high quality consistent welds to guarantee the integrity of the pipe system.

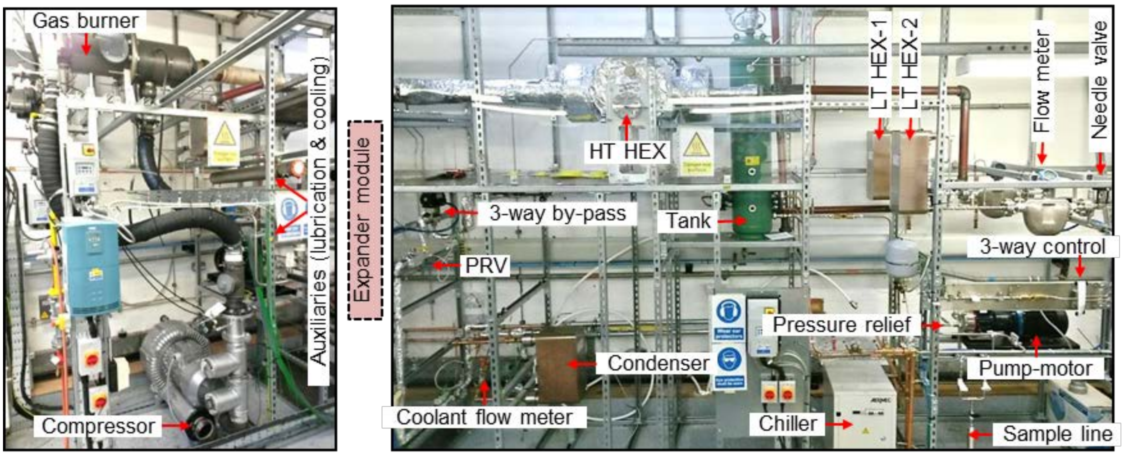


Figure 5: High-temperature high-pressure ORC facility employing three heat exchangers in mixed series and parallel combination for alcohols and alcohol-water blends

Results and discussion

Figure 6a-e presents some of the key parameters from the initial steady-state testing. The ORC thermal architecture was configured as option 1 (figure 4) and controlled using LabVIEW. The exhaust temperatures at the inlet and exit of the HT HEX were 388°C and 242°C respectively (figure 6a), while the coolant (water-glycol mixture) temperatures at the inlet and exit of the condenser were 20°C and 28°C respectively (figure 6b).

In this case, water was used as the working medium to demonstrate the relatively high-pressure and high-temperature capability. Note that, the facility is fully compatible with ethanol, propanol and their water blends. The reason to utilise water was to facilitate commissioning and system development prior to introducing alcohols in the system.

The HT HEX exit temperature and pressure (248°C, 32 bar, figure 6c-d) was targeted to be comparable to that required by the alcohol-water blends (figure 3). To simulate the inlet and exit pressures of a two-stage expander or turbine, which will be required under HT differential ORCs, a two-stage Pressure Reducing Valve (PRV) was utilised (figure 6d). Furthermore, the successful demonstration of throttling via the PRV is critical. This is since, under start-up and transient conditions, which may correspond to two-phase at the HT HEX exit, the flow needs to be by-passed when using conventional expanders and turbines. A total of 42 kW of heat was recovered in the HT HEX, with the mean working fluid flow rate of 55.9 kg/hr (figure 6e).

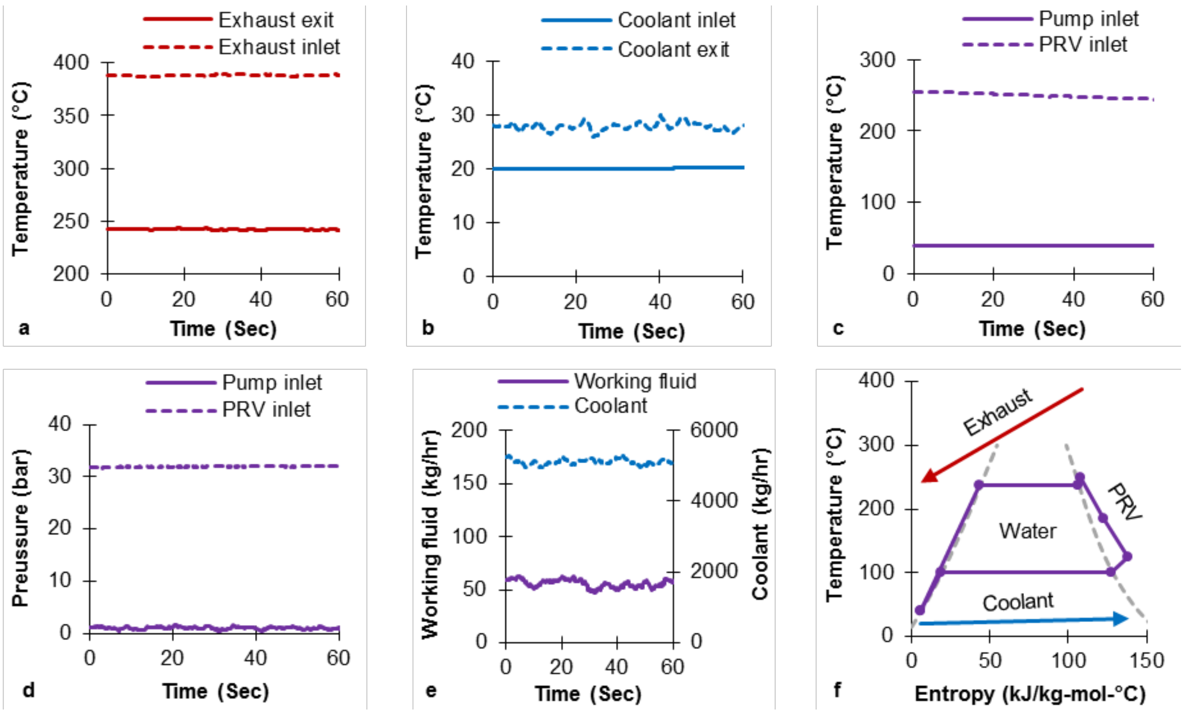


Figure 6: Off-design steady-state test results (a) Source quality, (b) Sink quality, (c) Working fluid temperature limits, (d) Working fluid pressure limits, (e) Working fluid and coolant flow rates, (f) T-S diagram of the process using a PRV

Finally, figure 6f presents the T-S diagram for this process. A total of 9 thermocouples, 5 pressure transducers and 2 flow meters were utilised to derive the thermodynamic conditions at the key points in the cycle. Due to the relatively benign heat load condition simulated, approximately 20 minutes were utilised in achieving the steady-state operation from a cold start. The excessive sub-cooling can also be explained owing to the off-design conditions.

Potential of power generation

To estimate the recoverable power for the test condition, the experimental parameters were utilised in the ORC model and the two-stage PRV was replaced with an efficient (65%) two-stage expander (5.7:1 each).

Figure 7a shows the resulting T-S diagram of the cycle, which offered a potential expander power of 6 kW for 42 kW of heat recovery. Furthermore, when the working fluid was replaced to the ethanol-water blend in the model, the expander power increased by 15% to 7 kW (figure 7b). This was since, assuming a similar Log Mean Temperature Difference (LMTD) value in the HT HEX, the blend recovered a higher quantity of exhaust heat (55 vs. 42 kW). This was principally due to the tailored, and rather reduced, latent heat of vaporisation of the blend compared to pure water. This also supports in explaining why water has been suggested as a working fluid for heat source qualities above 500°C (Saidur et al, 2012), and will in fact offer lower overall conversion efficiencies for typical exhaust heat recovery (300-500°C). Finally, assuming similar heat to expansion power conversion rate as that of figure 7a, the ORC facility can be tailored for efficient expansion machines up to 15 kW.

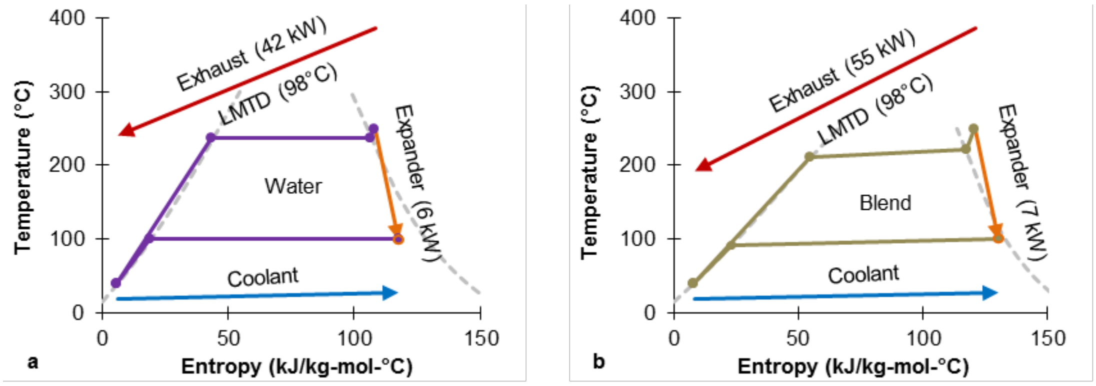


Figure 7: (a) Potential T-S diagram when integrating a high-pressure ratio expansion process (b) Potential T-S diagram when utilising alcohol-water blend paired with a high-pressure ratio expansion process

CONCLUSION

A programme of 'concept-to-demonstration' of a new ORC experimental facility is currently underway at the University of Brighton in order to investigate some of the challenges hindering the cost-effective deployment of HT ORCs. Resulting from the simulation to the initial demonstration phase of the work, the key features of the facility which may contribute towards reduced system costs and increased overall conversion efficiency can be summarised as:

- Variable heat source setup: Offering the potential of research and development for a wide capacity range of components. This is since, the setup can be tailored for heat qualities and quantities between 200-500°C and 12-120 kW, respectively.
- Direct exhaust heat utilisation: Expected to reduce the overall system cost by 15-20% when compared to LT ORC coupled with an oil loop. Furthermore, the reduced number of components, complexity and weight, for equal net power, is vital for transport applications.
- Ethanol-water and propanol-water blends: Estimated to increase the theoretical overall conversion efficiency by 2.3 times when compared to a system using R245fa. This result also supports the need for new ORC practise for high-grade applications.
- Flexible thermal architecture: Allowing combined charge air and exhaust heat recovery for effective low temperature heat utilisation (e.g. equal net power with 14% lower exhaust heat recovery), and internal heat recuperation for reduced condenser load (e.g. equal net power with 15% lower condenser load). As a result, the two LT HEXs in the parallel branch flow followed by the one HT HEX can be configured for varied engine applications.
- Advanced operating capability: Providing continuous maximum working fluid pressure and temperature of 40 bar and 250°C, respectively, which corresponds to the near-optimal region identified in the simulation studies.

The design, manufacturing, commissioning and demonstrating the operational capability of the thermal architecture (using PRV), as summarised in this paper, concludes the phase 1 of the works. In phase 2, parametric studies will be conducted using the alcohol-water blends by varying the evaporation and condensation temperatures, and the degree of superheat. Following this, in phase 3, high overall expansion ratios, using two-stage expansion machines that are tolerant to wetness at exit will be investigated. Note that, the experimental facility in its current status includes auxiliary loops which can be adapted for lubrication and cooling of future expansion machines.

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