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A NOVEL ORGANIC RANKINE CYCLE (ORC) FOR HIGH TEMPERATURE APPLICATIONS

Angad Panesar*, Morgan Heikal, Marco Marengo and Nicolas Miché

School of Computing, Engineering and Mathematics, University of Brighton, UK

* Corresponding author: a.panesar@brighton.ac.uk

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ABSTRACT

The current experimental ORC setups for long-haul Heavy Duty Diesel Engines (HDDE) are not reaching the desired fuel savings within the expected costs. Pathways to improve ORC performance and cost-effectiveness remain a major challenge facing the automotive sector. This paper presents the conceptual overview and simulation results (using Aspen HYSYS) of a novel ORC especially tailored for high temperature heat sources (300-400°C) in truck applications. With a fundamental revision of the expansion and heat transfer characteristics, the advantage of the proposed novel ORC included an equivalent performance to the conventional ORC despite a 20% reduction in the total heat transfer area and a 40% reduction in the size of the expansion machine. This resulted in a 15% improvement in the Cost/kW value of the system, whilst offering 5.1% improvement in engine thermal efficiency at highway driving conditions.

INTRODUCTION

A typical internal combustion engine rejects up to 50% of the total fuel energy in the form of heat. Due to increasing CO_2 emissions and fuel costs, there is a growing interest in techniques that can even partially utilise the exhaust heat to improve the overall system efficiency (Xu et al., 2013, Fu et al., 2012, Shu et al., 2012, Saidur et al., 2012). The use of fluid bottoming cycles offer one means towards converting waste heat into usable mechanical or electrical power for long-haul HDDEs. Performance and system considerations under mobile applications with heat source temperatures of 350-450°C and power output capacities of 5-20 kW favour the use of ORC over Rankine and Kalina cycles (Wang et al., 2012, He et al., 2011, Wang et al., 2011, Tchanche et al., 2011).

The heat-to-power conversion potential of an ORC largely depends on the selected working fluid (e.g. refrigerants, hydrocarbons), its associated cycle operating mode (e.g. subcritical, supercritical) and the system architecture (e.g. thermal, sub-system) (Sprouse III and Depcik, 2013, Saadatfar et al., 2013, Domingues et al., 2013, Tian et al., 2012). Although, the molecular make-up of organic fluids fundamentally precludes the possibility of an ideal fluid, the two fluids commonly proposed for high temperature applications in the literature include toluene and hexamethyldisiloxane (MM) (Yamaguchi et al., 2013, Seher et al., 2012, Fernández et al., 2011, Calm and Hourahan, 2011). These conventional organic fluids, proposed as subcritical ORCs, offer the advantages of a relatively high thermal stability (300-400°C), high molecular weight and the use of an Internal Heat Exchanger (IHE). However, these fluids are not without their distinctive challenges since they have sub-atmospheric condensing pressures at typical engine radiator temperature level (85-95°C) and involve near isothermal evaporation/condensation processes. As a result, they require large expansion pressure ratios and result in significant heat transfer irreversibilities.

To offer improved thermal matching, supercritical compression and heat addition using ethanol as the working fluid has also been recommended (Teng and Regner, 2009). Nonetheless, under high source temperatures (≈400°C) and high source-to-sink temperature differentials (≈300°C), this is typically challenging in practice due to the excessive system pressures (> 65 bar) and the need for large pressure ratios (> 30:1) (Latz et al., 2012).

METHODOLOGY

The key consideration in the research and development effort for ORCs is to investigate and identify technical paths that may improve the practicality of such a heat to power conversion concept. For this, simple yet innovative solutions are vital for a timely deployment of the technology to meet the anticipated CO_2 regulations cost-effectively in the long-haul trucking sector. To provide a potential solution, this paper presents a fundamental revision of the conventional high temperature ORC process without increasing the system integration complexity. Fig. 1b presents the hypothetical T-S sketch of this desired ORC. This is compared to Fig. 1a, which depicts the T-S sketch of a conventional ORC. The improved expansion and the unique heat transfer characteristics, which collectively can offer an improved energy conversion concept and a higher energy density solution are summarised below:

- Area 1: A noticeably higher density at the minimum cycle pressure for a nearly equivalent mass flow rate. This increases the system pressure differential for a fixed expansion machine size.
- Area 2: A high ΔT variation during evaporation to mirror the high heat source temperature drop. This reduces the heat transfer irreversibilities and offers higher average evaporation temperatures.
- Area 3: A low ΔT variation during condensation to mirror a low heat sink temperature rise. This offers lower average condensation temperatures for the same cooling air flow rate.

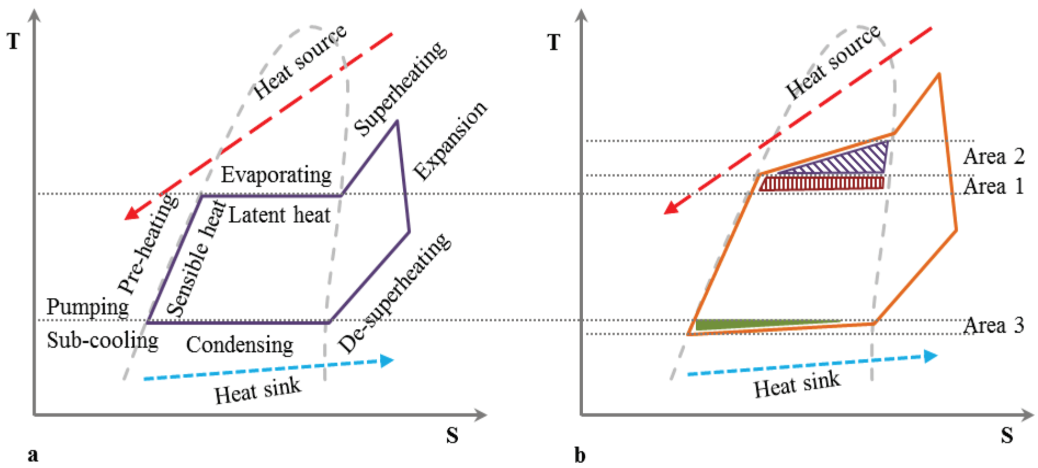

Figure 1: Comparing conceptual T-S diagrams and areas of loss reduction (a) Conventional ORC (b) Proposed ORC

The pathway identified to meet the above requirements was to formulate fluid blends, in particular positive pressure swing homogenous azeotropes, while retaining Toluene and MM as a noticeable blend component. This approach then represents a single method, which for the smallest change to the conventional ORC with an IHE, would translate to a noticeable benefit. Such an approach is contrary to the current research trends in the automotive sector, which are more tailored towards component development and improvements in component efficiencies to make the ORC systems cost-effective (Seher et al., 2012, Halliwell, 2012, Edwards et al., 2012, Hossain and Bari, 2011). Although, efficient heat exchangers (HEX) and high Volume Flow Ratio (VFR) expansion machines will be vital in the deployment of ORC systems, these approaches only indirectly address the cycle drawbacks.

RESULTS AND DISCUSSIONS

This section firstly presents the simulation results of toluene and MM to act as a reference for comparison. Secondly, the simulation results of the two particularly useful blends meeting the conceptual requirements of Fig. 1b are presented. The exhaust heat recovery, downstream of the aftertreatment devices, was considered at highway cruise conditions from a 12.8 litre truck engine model (Ricardo Software, 2008). To limit the total engine cooling-module size, the direct air cooled ORC condenser was positioned to provide engine radiator like condensing temperatures. The ORC simulations were conducted in an advanced chemical process modelling tool, Aspen HYSYS (Aspen

Technology, 2011). Fig. 2 shows the layout of the simulated ORC. The purpose of using an IHE was to internally utilise the considerable exergy exiting the expansion machine with the selected drying fluids. An ideal IHE increases the cycle thermal efficiency, decreases the heat recovery efficiency and has no impact on the net power (Li et al., 2011, Desai and Bandyopadhyay, 2009). The improvement in the thermal efficiency results in reduced load on the condenser. The exclusion of low temperature exhaust heat recovery avoids corrosion and fouling in the HEX. Also, since the pressure difference across the expansion remains equal, the specific work is unchanged.
Prior to a detailed heat transfer equipment calculation and design, the following was considered for the absolute size comparison. It was assumed that the overall heat transfer coefficient (U, W/m^2°C) was similar for the pure fluids and their blends. Note that this assumption is subject to an inaccuracy (±15%) when comparing vastly different types of fluids (e.g. refrigerants vs. hydrocarbons) (Perry and Green, 2007). Nonetheless, since the blends contain ≥80% by mass the counterpart pure fluids, the inaccuracy is expected to be low. Therefore, UA (W/°C), i.e. overall heat transfer coefficient multiplied by the heat transfer area (A, m^2), was considered as an indicator for the absolute heat transfer size comparison for HEX, IHE and condenser. Similarly, VFR defined as the ratio between the volumetric flow rates at the expansion outlet to inlet was considered as an indicator of the absolute size of the expansion machine.

$$Q_{HEX} = Q_{preheater} + Q_{evaporator} + Q_{superheater}$$

$$Q_{HEX} = m_{source}\, c_{p\,source}\left(T_{source\,HEX\,in} - T_{source\,HEX\,out}\right)$$

$$= m_{fluid}\left(h_{fluid\,HEX\,out} - h_{fluid\,HEX\,in}\right)$$

(Similar approach to model IHE and cond.)

$$W_{exp} = \eta_{exp}\, m_{fluid}\left(h_{exp\,in} - h_{exp\,out\,ideal}\right)$$

$$W_{pump} = \frac{m_{fluid}\left(h_{pump\,out\,ideal} - h_{pump\,in}\right)}{\eta_{pump}}$$

$$\eta_{thermal} = \frac{W_{exp} - W_{pump}}{Q_{in}}$$

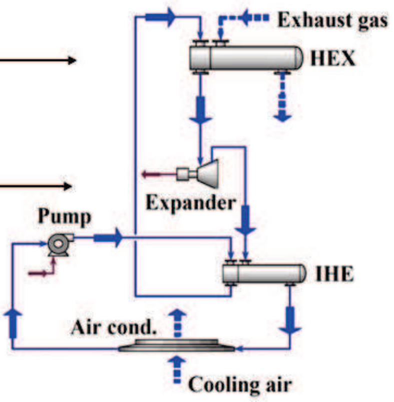

Figure 2: Simulated ORC setup and brief modelling overview

Table 1 summarises the system parameters and performance index for toluene and MM with fixed UA (UA $_{HEX}$ = 1500 W/°C, UA $_{IHE}$ = 750 W/°C, UA $_{cond}$ = 1250 W/°C) and VFR (16:1). Toluene and MM generated a net power of 8.3 and 6.8 kW, respectively. Fig. 3a depicts the T-S diagram for the resultant toluene ORC. Fig. 2 and 3a also collectively summarise the ORC modelling overview and assumptions used corresponding to realistic temperatures and component efficiencies. The maximum fluid temperature and cooling air flow rate was restricted to 280°C and 2 kg/s, respectively in all cases.

Table 1: Key system, sizing, performance and fluid properties for the conventional and the proposed ORC

		Toluene	MM	T80	MM90			Toluene	MM	T80	MM90
		System parameters						Performance index			
P_{max}	bar	13.2	12.5	22	21.3	W_{net}	kW	8.3	6.8	8.3	6.9
P_{min}	bar	0.5	0.5	1.7	1.7	$\eta_{thermal}$	%	16.6	13.4	15.9	13.3
T_{evap}	°C	235	217	223	206						
T_{cond}	°C	86	77	105	101						
m_{fluid}	kg/s	0.104	0.158	0.094	0.137						
Q_{HEX}	kW	49.7	50.8	52.2	52						
Q_{IHE}	kW	20.8	38.4	20.1	32.6			Fluid Properties			
Q_{cond}	kW	41.3	44	43.9	45.1	Molecules		C,H	C,H,O,Si	C,H,OH	C,H,O,Si,OH
						$T_{boiling}$	°C	112	100	79	73
		Sizing parameters				M_{weight}	g/mol	92	162	77	130
UA_{HEX}	W/°C	1500	1500	1200	1200	ρ_{liquid}	kg/m^3	780	677	807	724
UA_{IHE}	W/°C	750	750	600	600	c_p	kJ/kg°C	1.96	2.07	2.33	2.17
UA_{cond}	W/°C	1250	1250	1000	1000	λ	W/m°C	0.11	0.11	0.12	0.12
VFR_{exp}	-	16:1	16:1	9.6:1	9.6:1	μ	cp	0.24	0.23	0.36	0.34

The two blends that meet the requirements of Fig. 1b were termed T80 and MM90, based on toluene and MM percentage by mass, e.g. T80 corresponding to 80% toluene. Note that the second blend constituents are undisclosed, however the key fluid properties for T80 and MM90 are given in table 1. T80 and MM90 properties were calculated using the Wilson property package (Aspen HYSYS V 7.3, 2011). These blends are expected to offer similar freezing temperatures, thermal decomposition temperatures, auto-ignition temperatures, NFPA rating and environmental impact compared to the pure fluid counterparts. Additionally, liquid density, specific heat and thermal conductivity favour T80 and M90.

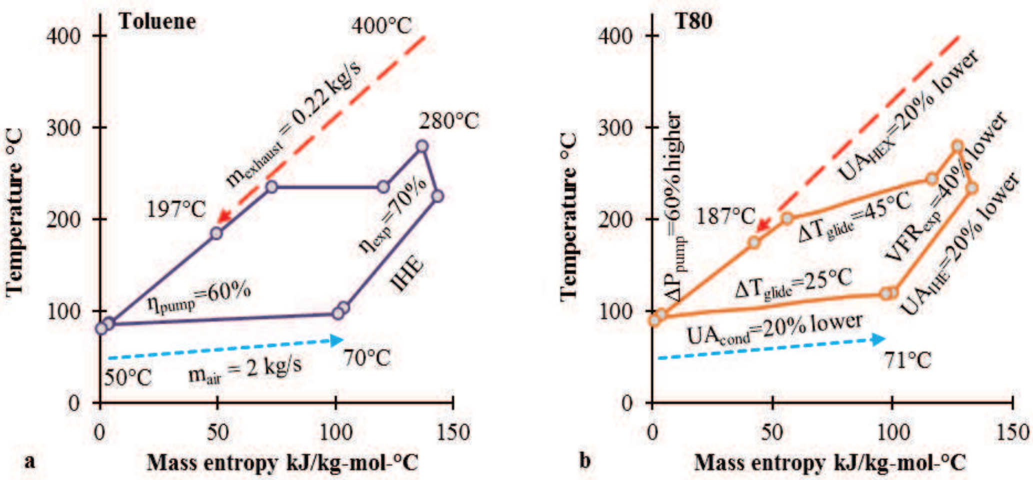

Figure 3: Comparing real T-S diagrams (a) Conventional toluene ORC (b) Reduced design intensity T80 ORC

T80 and MM90 were simulated for the same ORC setup (Fig. 2), but with 20% reduced heat transfer equipment size (i.e. 20% lower UA for HEX, IHE, condenser) and 40% reduced expansion machine size (i.e. 40% lower VFR). Table 1 also presents the system parameters, sizing parameters and performance index for T80 and MM90. Despite the system size reduction, T80 and M90 gave equal net power to the pure fluid counterparts. This corresponded to 5% of additional engine crankshaft power, increasing the engine thermal efficiency by 5.1% (from 42.9% to 45.1%). Fig. 3b depicts the T-S diagram for the resultant T80 ORC. These fluids allow the continued use of an IHE, which is vital for reducing the size of the air cooled condenser. The relatively lower VFR (9.6:1 vs. 16:1) and the retention of higher molecular weights (77-130 vs. 92-162 g/mol) bodes particularly well for single stage radial turbines as expansion machines (Saadatfar et al., 2013, Tchanche et al., 2011).

Additionally, T80 is expected to offer virtually zero electric conductivity and good lubrication properties. Hence, T80 could also be used for cooling the electrical components of the ORC system and lubricating the expansion machine.

TECHNO-ECONOMICAL ANALYSIS

The current market niche for ORCs is dependent on simplicity and affordability, with initial technology deployment on commercial vehicles expected in the 2020-2025 timeframe. A detailed economic review and an original cost analysis study conducted for small scale (5-20 kW), high-production (>1000 units), ORC systems (with a focus on automotive application) and its associated components resulted in the cost distribution shown in Fig. 4a (Lopes et al., 2012, Roos, 2009, Quoilin and Lemort, 2009, Thekdi, 2007, Dickey, 2007, Peters et al., 2003). The three relevant points in the present case were that:

- Approximately 25% of the total system costs were fixed and non-scalable.
- Scalable fluid expansion, fluid compression and heat transfer costs were closely related to expansion VFR, pump pressure differential and heat transfer UA, respectively.
- A similar distribution trend also existed for the size and weight of ORC components (Note: fixed and non-scalable percentage was around 30%).

To estimate a techno-economical trade-off between T80, MM90 and their pure fluid counterparts, the following was considered. It was assumed that the ratio of Cost/kW of the toluene and MM cycles were 1. Therefore, when considering 20% lower UA, 40% lower VFR, and 60% higher ΔP (for T80 and MM90), the Cost/kW reduced to 0.85, i.e. a 15% savings (Fig. 4b). Furthermore, T80 and MM90 showed noticeably decreased sensitivity to the absolute system pressure losses. The results presented in table 1 included a 0.2 bar pressure loss on the working fluid side in each heat transfer element. When this pressure loss was increased to 0.4 bar (for the same UA and VFR values), the maximum loss in power was only 3%, compared to 9% with the pure fluids (Fig. 4c). This was largely attributed to a higher minimum cycle pressure (1.7 vs. 0.5 bar). The system size reduction and reduced pressure loss sensitivity advantages collectively bode well for the use of compact and inexpensive equipment.

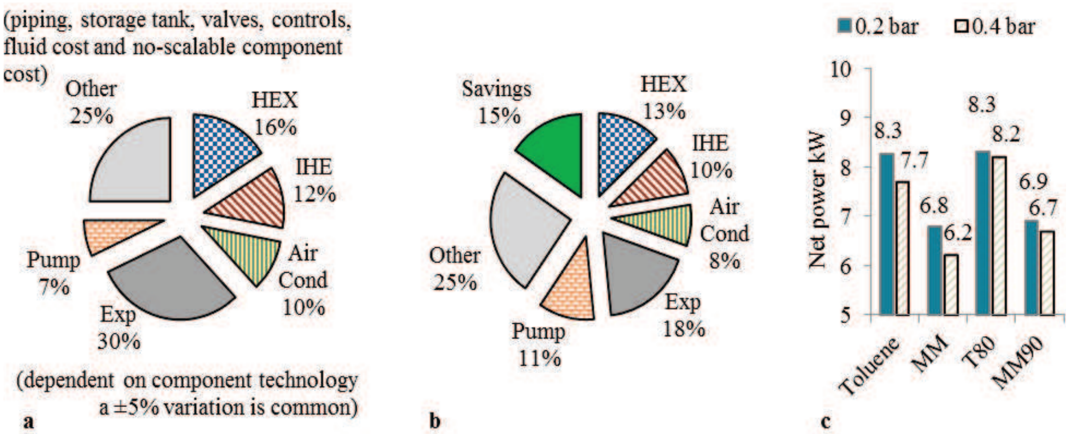

Figure 4: (a) Small scale conventional ORC component cost distribution (b) Relative expected savings using the proposed ORC (c) Fluid pressure drop sensitivity to net power

CONCLUSIONS AND FURTHER RESEARCH

A novel thermodynamic approach has been presented in this paper to facilitate the introduction of ORC systems for truck applications. The novelty of the concept developed here involved, adapting the working fluid make-up to offer a higher system pressure differential for the same size of the expansion machine, and efficiently match to the source/sink streams for the same heat transfer footprint. This was as a result of a noticeably higher density at the minimum cycle pressure for a nearly equivalent mass flow rate (despite high toluene and MM concentrations), and reduced heat transfer irreversibilities encountered during the phase-change processes (despite subcritical pressures). Therefore, for an equal system power to toluene and MM cycles, the T80 and MM90 cycles presented the advantage of a 15% lower system cost for the same net power, thus improving commercialisation potential. Additionally, the lower VFR (9.6:1) and the higher molecular weights

calculated (77, 130 g/mol) allows the possibility of using both single stage radial turbines and piston expanders.
The approach for an efficient ORC presented in this paper has been investigated using a simulation tool (Aspen HYSYS) widely used in the chemical and process industries. Limited but crucial parameters were selected for analysis and the results form the basis for comparison with the pure fluid counterparts. The possibility to source T80 has been confirmed from a well-known chemical supplier. Future works will include verification of T80 fluid properties given in Table 1 (calculated using Wilson property package) against experimental results.

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