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The potential of bottoming cycle applied to a high exhaust gas recirculation engine for maximum fuel consumption improvement

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Abstract

Increasing fuel prices and CO_2 emissions have increased interest in the application of waste heat to power conversion for heavy duty Diesel engines. A systems approach is used to research the benefits of a Bottoming Cycle (BC) applied to an engine utilising a high Exhaust Gas Recirculation (EGR) emissions control strategy. The fuel economy improvement from the use of a BC largely depends on the selected working fluid, the cycle operating condition and the associated process integration. The present simulation study uses water, ethanol, R30, acetone, R245fa and E152a as working fluids for the BC while recovering heat from combinations of high temperature after-cooler, EGR cooler and exhaust gas streams. Starting the working fluid expansion from saturated, superheated and supercritical phase, 10 different cycle arrangements are investigated for maximum power recovery with least impact on the engine cooling module. The two best BC arrangements and operating conditions from optimal performance and system related trade-offs show an additional 9 and 9.5% engine power recovered at a high load condition.

Keywords: Heavy duty Diesel engine, Exhaust gas recirculation, Waste heat recovery, Bottoming cycle, Working fluid selection, Process integration

1 Introduction

Set to take effect in 2014, the Euro 6 emissions standards of NOx limits of 0.4 g/kWh stipulate tighter emissions controls. The use of such regulatory technique to limit greenhouse emissions (CO_2) for heavy duty Diesel engines is also expected. To comply with the Euro 6 NOx legislations, three different emissions control strategies are possible, these being, combined Selective Catalytic Reduction (SCR) and Exhaust Gas Recirculation (EGR), SCR only, and EGR only. Studies have shown an EGR only engine to be the least favourable, with a 6% fuel penalty compared to a combined SCR and EGR platform [1].

With over 40% of the fuel energy wasted through untapped exhaust heat, Waste Heat Recovery (WHR) offers a valuable alternative approach in improving overall fuel economy of an engine with an EGR NOx control strategy. WHR technologies exist in several forms (mechanical/electrical turbo-compounding, thermo-electric generators, bottoming cycles), most of which are still under development. Despite the various challenges, Bottoming Cycle (BC) systems are shown to be one of the most promising solutions for long haul application, leading to a fuel efficiency improvement of around 8%[2].

2 Combined EGR and WHR approach

An EGR only engine relies on an efficiently cooled EGR system, high pressure fuel injection and two-stage of charging arrangement for in-cylinder NOx reduction. EGR only engines are expected to have lower overall weight, and lower installation and operational costs[3]. However, due to the use of extremely high recirculation rate of 35-45%, EGR only engines have the drawback of increased load on the engine cooling module. The total heat rejection is expected to be around twice than that of the combined EGR and SCR platform.

With an innovative approach for the post 2014 demands of efficiency and emission requirements, this paper demonstrates the use of a systems approach to WHR and NOx reduction. Mitigation of an EGR only engines inefficiency is explored through WHR. To generate waste heat data typical of a long haul engine at a high load condition (C100), a 10 litre (L) EGR only engine model was developed in Ricardo WAVE 8.1 [4] running with the NOx strategies derived from a 2L research engine [5].The running conditions used were, air to fuel ratio (24.6:1), inlet pressure (4.8 bar), inlet temperature (69°C), EGR gas fraction (35.5%), start of injection (5°BTDC), injection duration (30°) and fuel rail pressure (2300 bar).

Nomenclature

A/C	After-Cooler
BC	Bottoming Cycle
BTDC	Before Top Dead Centre
DPF	Diesel Particulate Filter
E152a	Difluoromethyl-Methyl-Ether
EGR	Exhaust Gas Recirculation
EXH	Exhaust
EXP	Expander/Turbine
HT	High Temperature
HP	High Pressure
L	Litre
LMTD	Log Mean Temperature Difference [°C]
LT	Low Temperature
LP	Low Pressure
R245fa	1,1,1,3,3-Pentafluoropropane
R30	Dichloromethane
SCR	Selective Catalytic Reduction
WF	Working Fluid
WHR	Waste Heat Recovery

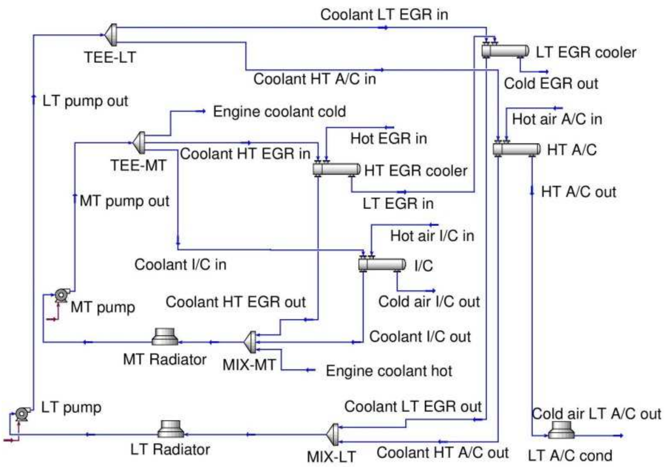


Fig. 1 Cooling module of the 10L engine

Table. 1 Available waste heat in the 10L engine

	Duty (kW)	Inlet °C	Outlet °C
High temperature EGR	37.2	465	325
Inter-cooler	10.76	146	121
Engine cooling	53.76	80	90
Low temperature EGR	58.02	325	95
High temperature after-cooler	45.22	200	95
Low temperature after-cooler condenser	14.13	95	62
Low temperature radiator	101.36	80	65
High temperature radiator	103.6	88.8	80
Air, low temperature radiator		20	35
Air, high temperature radiator		35	50

The arrangement of the cooling system for the 10L engine is shown in Fig. 1. The cooling system comprises of two indirect cooling circuits. The low temperature cooling circuit cools the Low Temperature EGR (LT EGR) cooler and the High Temperature After-Cooler (HT A/C). The high temperature cooling circuit dissipates heat from the High Temperature EGR (HT EGR) cooler, Inter-Cooler (I/C) and engine coolant. As shown in Fig.1, the after-cooler is divided into two stages, the first is integrated into low temperature circuit and the second covers the remaining cooling process by employing direct air cooling (LT A/C cond). Table 1 shows the quantities and qualities of the available heat sources. The coolant (water/glycol, 50:50) in the low and high temperature condensers is cooled down to 65 and 80 °C, respectively.

At the selected C100 condition the 10L engine model has an efficiency of 40.2% and produces 296 kW of shaft power. The total heat being rejected by the two condensers adds up to 205 kW. A further of 98.2 kW of heat (from 300-95°C) is available in the exhaust flow downstream of the Diesel Particulate Filter (DPF). WHR is investigated recovering heat from the combinations of high temperature after-cooler, EGR cooler and exhaust gas streams. BC simulations were conducted using Aspen HYSYS V7.3 [6]. The thermodynamic properties of the various fluids have been calculated using the Peng-Robinson equation of state.

3 Bottoming cycles with single expansion stage

3.1 Arrangement 1: EGR and HT A/C

At the selected C100 condition, the EGR cooler offers high quality (from 465-95°C) and quantity (95.2 kW) of heat. However, in a high EGR rate engine there is also a low quality (from 200-95°C) and high quantity (45.2 kW) of heat available in the HT A/C. In this context, arrangements that convert this low grade heat source into power are of great significance. When the arrangement proposed in Fig. 2a is implemented, it can convert heat from HT A/C and EGR cooler to power with minimal impact on the engine cooling load whilst increasing engine power.

In Fig. 2a the working fluid leaving the Low Pressure (LP) pump is distributed into the HT A/C and LT EGR cooler where it is preheated. The working fluid then enters the High Pressure (HP) pump where the pressure is further raised. The high pressure working fluid is evaporated in the HT EGR cooler. The working fluid leaving the HT EGR cooler is at saturated vapour state in the case of dry and isentropic fluids and slightly superheated in the case of wet fluids. After expansion, the condenser finally sub-cools the working fluid by 1°C.

In arrangement 1, increasing the working fluid temperature in the preheating section of the cycle increases the thermal efficiency and net power. The pressure at the LP pump exit was controlled to provide a working fluid maximum preheating temperature of ≤ 170°C, whilst maintaining the heat exchanger pinch point limitation ≥ 25°C. As the condensing pressure was kept constant, increasing the evaporation pressure in the HT EGR cooler by the HP pump to slightly subcritical levels can remarkably increase the efficiency. For such pressures the isobar of the working fluid matches much better to the isobar of the high temperature EGR and the enthalpy drop across the expansion increases. However, as arrangement 1 uses a single stage expansion, the maximum pressure was limited to 40 bar.

Fig. 2b and 2c show the cycle T-S diagram using an isentropic fluid, acetone and a dry fluid, R245fa respectively. These two fluids are not superheated as dry and isentropic fluids may leave the expansion stage with substantial amount of superheat, increasing the load on the condenser. However, in the presented T-S diagrams, the expansion carried out is such that the expansion goes through the two-phase region. In such a process only extremely fine droplets (fog) are expected to form in the two-phase region without any liquid that will actually damage the turbine [7].

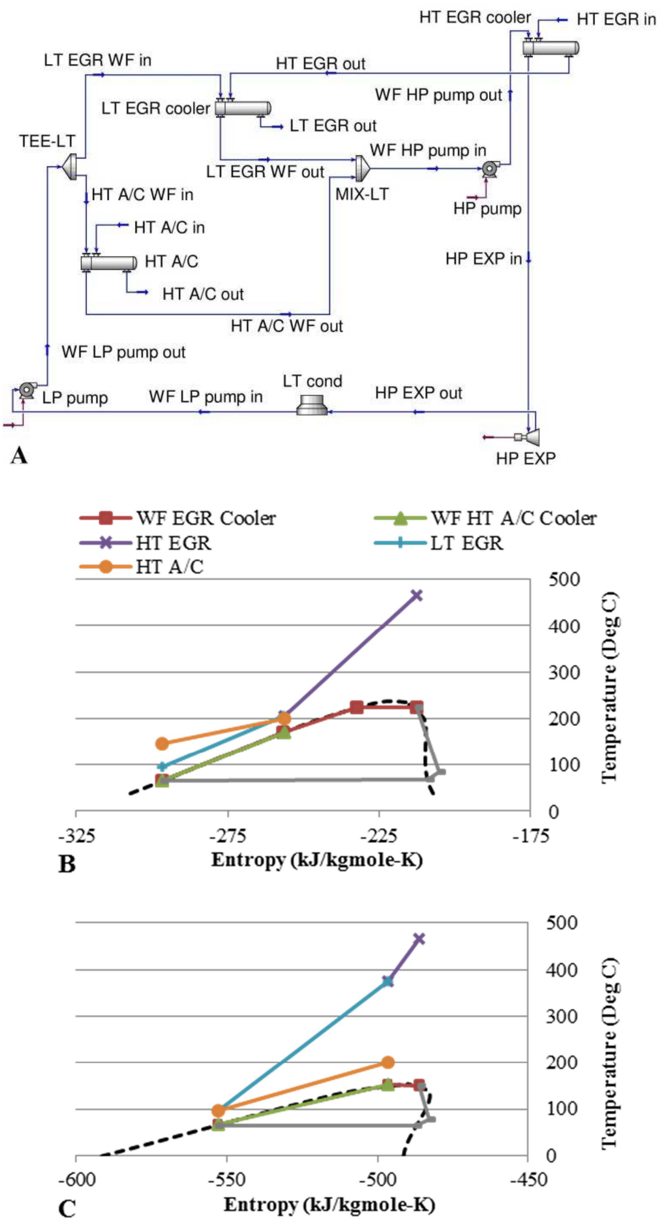


Fig. 2 (a) Arrangement 1: EGR and HT A/C, (b) Cycle operation using acetone, (c) Cycle operation using R245fa

For all the simulations, heat in the BC condenser is rejected directly to the air as it offers higher conversion potential. Boundary conditions and assumptions regarding the simulations are detailed in table 2.

The selection of the working fluid is critical in achieving high overall conversion efficiencies (product of heat recovery and thermal efficiency). The critical point of a working fluid suggests the possible operating temperature and pressure range.

Table. 2 Boundary conditions and assumptions

Cooled HT A/C, EGR and EXH temperature ≥ 95 °C
Max. BC pressure (single stage) = 40 bar or 98% of critical pressure (whichever is lower)
Max. BC pressure (dual stage) = 60 bar
All heat exchanger pinch point ≥ 25 °C
ΔP, HT A/C, EGR and EXH (gas side) = 0.05 bar/heat exchanger
ΔP, working fluid = 0.2 bar/heat exchanger
ΔP, working fluid condenser = 0.1 bar
Turbine/expander efficiency = 75%
Pump efficiency = 65%
Min. condenser temperature = 65°C

From table 3 and Fig. 2b , 2c it can be seen that R245fa can absorb more HT A/C heat (45.22 kW) than acetone (23.18 kW) but only at a lower evaporation temperature (151 vs. 224°C). Although the net heat input to the R245fa BC is higher (140.42 vs. 118.37 kW), it has lower thermal efficiency (8.67 vs. 15.17 %). This is as a result of its much lower critical temperature (154 vs. 235°C) which results in higher irreversibilities in the heat exchangers (magnitude difference between the source and working fluid profile). As the area enclosed by the cycle curve is also small the net power produced is much lower (12.18 vs. 17.95 kW). Therefore, it is necessary to select working fluids that optimise the heat absorption profile for maximising the work output.

Table. 3 Heat duty, net power and relative size for arrangement 1

	Ethanol	R245fa	R30	Acetone
HT EGR cooler duty (kW)	68.18	23.90	67.94	67.94
LT EGR cooler duty (kW)	27.02	71.30	27.26	27.25
HT A/C cooler duty (kW)	13.79	45.22	12.37	23.18
LT cond. duty (kW)	91.95	128.24	90.96	100.42
Net power (kW)	17.04	12.18	16.60	17.95
Thermal efficiency (%)	15.64	8.67	15.43	15.17
Total area/Net power	1.04	1.71	1.10	1.01

In table 3, one criterion used as an indicator in estimating the relative size and cost of BC is the ratio of total heat exchanger area to the net power produced. For calculating the heat exchangers surface area, numerical correlations described in [8] are used. Assumptions and restrictions relative to the design of the heat exchangers are given in table 2. All the heat transfer elements are divided into 50 equal working fluid enthalpy change sections. The working fluids are considered for flow inside the tubes and the air and exhaust gas flows through the shell. Ethanol, R30 and acetone show net power and relative heat exchanger size within ±10%. However, R245fa shows a 70% increase in relative heat exchanger size (1.01 vs. 1.71 m^2/kW). For arrangement 1, the maximum generated net power is 17.95 kW with acetone as the working fluid. In addition to the energy recovered there will be a reduction in the fan power requirement and engine cooling load.

3.2 Arrangement 2: EGR and EXH parallel

A source of medium quality (from 300-95°C) and high quantity (98.2 kW) heat is the exhaust (EXH) flow downstream of the turbocharger and DPF. The combination of EGR and EXH are the most promising heat sources due their high exergy value. Fig. 3a considers a parallel arrangement for EGR and post DPF exhaust. The distributed working fluids streams are preheated, evaporated and superheated (if necessary) in the two heat exchangers. In this analysis the maximum exhaust gas temperature was fixed at 300°C. However, in a high EGR rate engine the more frequent active DPF regeneration will provide higher exhaust temperatures (500-600°C) with longer resident times. If the BC can harness this added energy, the overall conversion efficiency will improve.

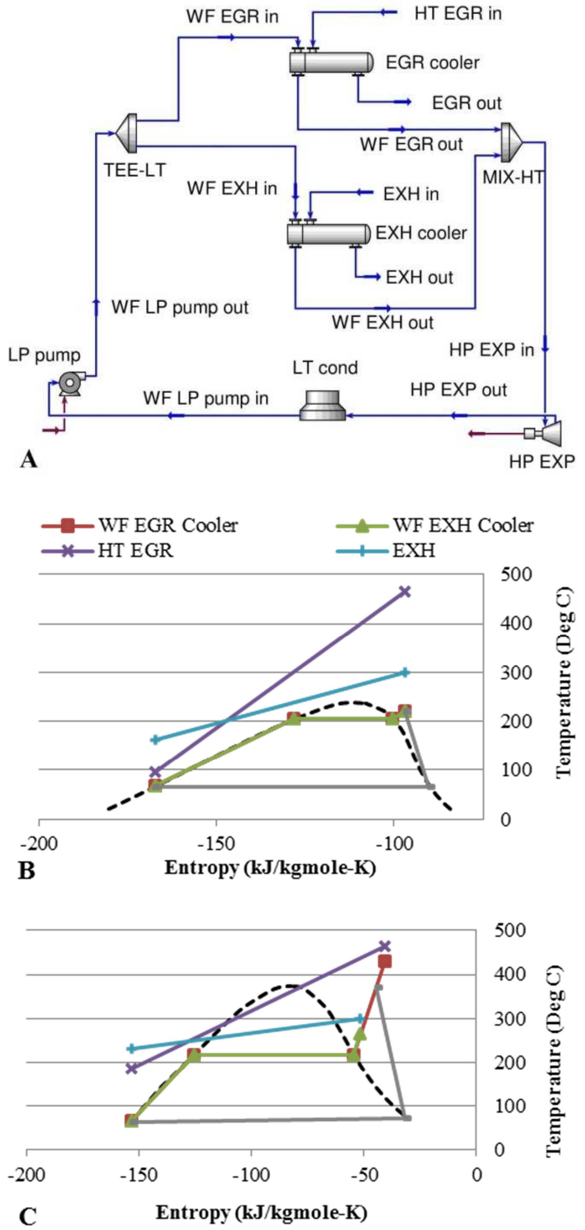


Fig. 3 (a) Arrangement 2: EGR and EXH parallel, (b) Cycle operation using R30, (c) Cycle operation using water

Fig. 3b and 3c show the cycle diagram using wet fluids R30 and water, respectively. With wet fluids, increasing evaporator pressures shifts the expansion exit state left and increases the moisture content. Wet fluids are superheated to have expansion exit vapour fraction ≈0.99. As water is the wettest known fluid the control strategy to reduce moisture at the expansion exit is different from that of R30. When using R30 slight superheating in the two heat exchangers is sufficient to reduce moisture. Hence, the working fluid profiles overlap for the EGR and EXH cooler (Fig. 3b). However for water, as sufficient superheat cannot be generated in the EXH cooler, large superheating takes place in the EGR cooler. As can be seen from Fig. 3c, the steam temperature exiting the EGR and EXH cooler is at 430 and 265°C, respectively. These two streams mix to form a superheated vapour at 370°C before entering the turbine.

For ethanol, R30 and acetone the maximum cycle pressure was limited to 40 bar. For the selected organic fluids higher pressures increase the net power produced, contrary to water. For water the cycle pressure corresponding maximum power recovery was 23 bar. This is due to the largest latent heat of water, leading to the largest latent to sensible heat ratio of any known fluid. Raising the evaporator pressure beyond the 23 bar limit will only increase the thermal efficiency but reduce the heat absorbed by the cycle. So these positive and negative effects result in an optimal evaporator pressure much lower than that of the organic fluids.

It is vital for working fluids in a high EGR rate engine to not only provide high net power but also to cool the EGR temperature close to 95°C, without requiring an additional air cooled LT EGR cooler. R30 is able to recover all the 95.2 kW heat available in the EGR cooler (table 4). Whereas water recovers only 72.42 kW of heat available in the EGR cooler, rejecting 25% of the EGR heat. R30 provides 20% more net power (25.1 vs. 20.5 kW), nonetheless with reduced cycle efficiency (15.56 vs. 19.38 %) due to the energy being produced by steadily decreasing EGR and EXH temperatures. The higher net power advantage with low boiling point of R30 (39.8°C) becomes obvious for these temperature levels.

Another interesting conclusion is not only that high overall conversion efficiencies are attainable with R30 but also that these efficiencies are possible with lower temperature difference across the expansion stage (155 vs. 305°C). Low temperature drop during expansion will reduce thermal stress and can be an interesting solution to reduce expansion machine cost.

Table. 4 Heat duty, net power and relative size for arrangement 2

	Water	Ethanol	R245fa	R30	Acetone
EGR cooler duty (kW)	72.43	95.20	95.20	95.20	95.20
EXH cooler duty (kW)	33.37	64.70	96.95	66.15	77.58
LT cond. duty (kW)	85.30	134.73	175.49	136.25	146.25
Net power (kW)	20.50	25.17	16.65	25.10	26.53
Thermal efficiency (%)	19.38	15.74	8.67	15.56	15.35
Total area/Net power	0.67	0.73	1.38	0.76	0.71
Δ Rad. load (kW)	12.87	39.53	80.29	41.05	51.05

The large latent heat of water reduces the required mass flow by 12 times (0.036 vs. 0.44 kg/s), this results in a feed pump power consumption that is 16 times lesser than R30 (0.13 vs. 2.06 kW). However, in transport applications controlling a small water flow rate precisely with the changing driving condition will be a challenge. For higher overall conversion efficiency using water, other arrangements or source temperatures higher than 600°C will be needed.

Like arrangement 1, ethanol, R30 and acetone show net power and relative heat exchanger size within ±10%. As acetone has the lowest latent to sensible heat ratio among the three fluids it recovers the largest tail pipe heat (77.58 kW). With nearly equal thermal efficiency (15.35-15.74 %), acetone produces the maximum net power of 26.63 kW for the parallel EGR and EXH arrangement.

BCs that capture heat from the exhaust will increase the engine cooling load. The increase in the engine cooling load is dependent on the amount of exhaust heat recovered and the cycle efficiency. For such a trade off it may be beneficial to select working fluids that will provide net power approaching that of acetone but with lower increase in the engine cooling load. Selecting R30 may offer such a solution. R30 will deliver around 10% lower net power (25.1 vs. 26.53 kW) but have 20% lower increase in heat rejection (41.05 vs. 51.05 kW).

As the exhaust heat exchanger is size constrained it should stay efficient and limit backpressure. Simulations using the 10L model have shown around 1% increase in fuel consumption due to the increased back pressure. At conditions where increased heat rejection by the engine cooling module is not possible, an exhaust bypass has to be provided.

3.3 Arrangement 3: EGR and EXH for maximum power

In arrangement 2, R245fa recovers 96.95 kW form the EXH cooler, cooling the exhaust stream to 98°C. Due to the pinch point limitations, higher critical temperatures, and higher latent heat to sensible heat ratio, ethanol, R30 and acetone recovered lower amounts of exhaust heat (64.7-77.7 kW). The arrangement 3, proposed in Fig. 4a is the only solution that will recover maximum EGR and exhaust heat for highest net power for these fluids.

Fig. 4b shows the cycle operating condition with R30 as the working fluid. The distributed working fluid is only preheated in the LT EGR cooler whereas the working fluid is preheated and partially evaporated in the EXH cooler. Working fluid leaving the EXH cooler mixes with the preheated working fluid reducing its dryness fraction from 0.77 to 0.42. The 2 phase fluid finally enters the HT EGR cooler where is evaporated and slightly superheated.

Comparing Fig. 3b and 4b for the heat source and working fluid profile match in the heat exchange process, the superior profile match of Fig. 4b becomes evident. This is also reflected in the lower Log Mean Temperature Difference (LMTD) in the EXH cooler (44.3 vs. 51.7°C).

The additional heat rejection requirements owing to the future utilisation of exhaust gas heat for waste heat recovery is anticipated. This makes new low temperature radiators necessary or, alternatively, the efficiency of existing engine cooling modules must be significantly increased. For arrangement 3, maximum power benefit can be achieved using acetone (29.32 kW, table 5). However, the drawback of arrangement 3 is the severe impact on the engine cooling module. The total cooling module heat rejection increases by 66.54 kW to 271.5 kW. Since this cooling requirement must compete with those for engine cooling and charge air coolers, effects upon the fan power requirement and engine combustion have to be minimised.

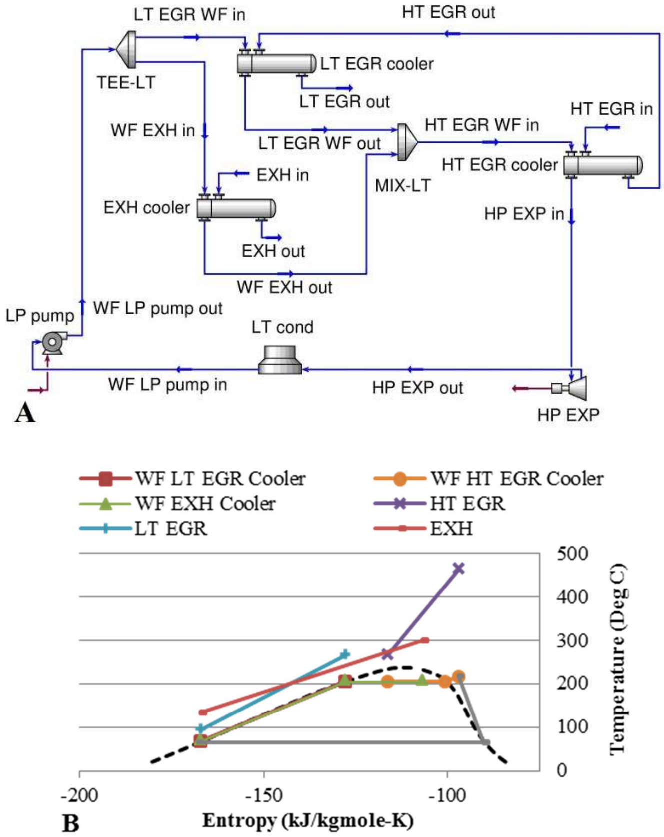


Fig. 4 (a) Arrangement 3: EGR and EXH for maximum power, (b) Cycle operation using R30

Table. 5 Heat duty, net power and relative size for arrangement 3

	Ethnaol	R30	Acetone
HT EGR cooler duty (kW)	43.43	52.37	43.43
LT EGR cooler duty (kW)	51.76	42.83	51.76
EXH cooler duty (kW)	76.01	80.01	95.87
LT cond. duty (kW)	144.31	147.99	161.74
Net power (kW)	26.89	27.21	29.32
Thermal efficiency (%)	15.71	15.53	15.35
Total area/Net power	0.80	0.87	0.86
Δ Rad. load (kW)	49.12	52.80	66.54

3.4 Arrangement 4: EGR and EXH series

The drawback of arrangement 2 and 3 is the increased load on the engine cooling module (e.g. R30: 41.05 and 52.8 kW, respectively). For EGR and EXH heat recovery with the least impact on the engine cooling module series arrangement as shown in Fig. 5a is proposed. As exhaust temperature is lower than EGR temperature (300 vs. 465°C), to obtain a higher thermal efficiency, the exhaust stream should be used for pre-heating and partial evaporation and the EGR stream should be used for evaporating and superheating. The approach used in arrangement 4 as depicted by the cycle operating condition as shown in Fig. 5b involves EGR for preheating and partial evaporation and exhaust for evaporating and superheating. This approach resulted in slightly lower cycle efficiencies but was nonetheless able to provide the required EGR cooler outlet temperatures with least impact on the engine cooling module.

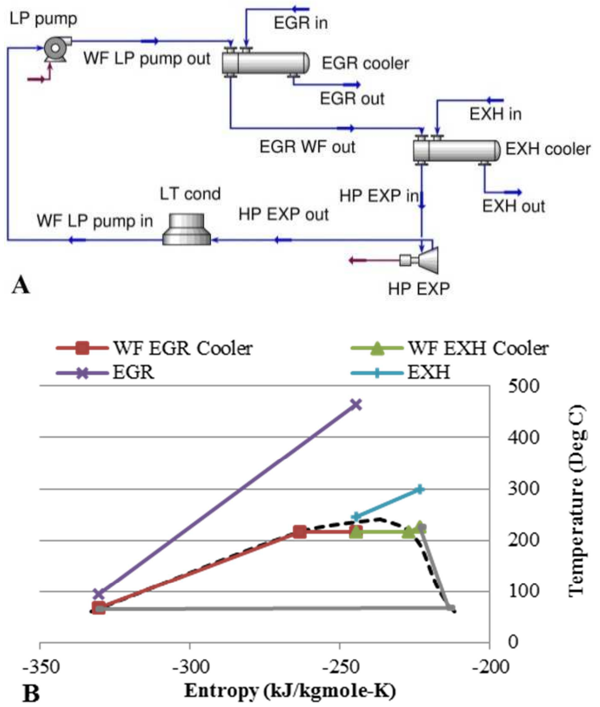


Fig. 5 (a) Arrangement 4: EGR and EXH series, (b) Cycle operation using ethanol

An additional parameter of interest in this arrangement is the cooled exhaust temperatures. Using the three fluids shown in table 6, the exhaust temperatures were kept above 230°C. Contrary to arrangements 2 and 3, arrangement 4 may allow the EXH cooler to be placed post turbine but pre DPF. Temperature levels over 230°C may allow uninterrupted DPF operation and reduce the thermal cycling in the exhaust heat exchanger. From table 6, BC using acetone cooling EGR to 95°C and receiving 95.2 kW of heat, and cooling exhaust to 254°C by topping up with 22.27 kW of heat is considered the most optimal. It will produce 18.4 kW of net power, will require the smallest heat exchanger size and will increase the engine heat rejection by only 4.23 kW.

Table. 6 Heat duty, net power and relative size for arrangement 4

	Ethanol	R30	Acetone
EGR cooler duty (kW)	95.20	95.20	95.20
EXH cooler duty (kW)	26.16	30.89	22.27
LT cond. duty (kW)	102.24	106.46	99.43
Net power (kW)	19.12	19.63	18.04
Thermal efficiency (%)	15.75	15.57	15.36
Total area/Net power	0.83	0.85	0.79
Δ Rad. load (kW)	7.04	11.26	4.23

3.5 Arrangement 5: EGR, HT A/C and EXH

To realise the benefits of arrangement 1, i.e. recovery from HT A/C heat and arrangement 4, i.e. least impact on engine cooling module, arrangement 5 is considered. In Fig. 6a, the working fluid is distributed in the LT EGR cooler and HT A/C where it is preheated to a maximum temperature of 170°C. After the preheated working fluid is mixed, the HP pump raises the cycle pressure to a maximum of 40 bar. The working fluid is again distributed between the HT EGR cooler and EXH cooler where it is evaporated and superheated (if necessary).

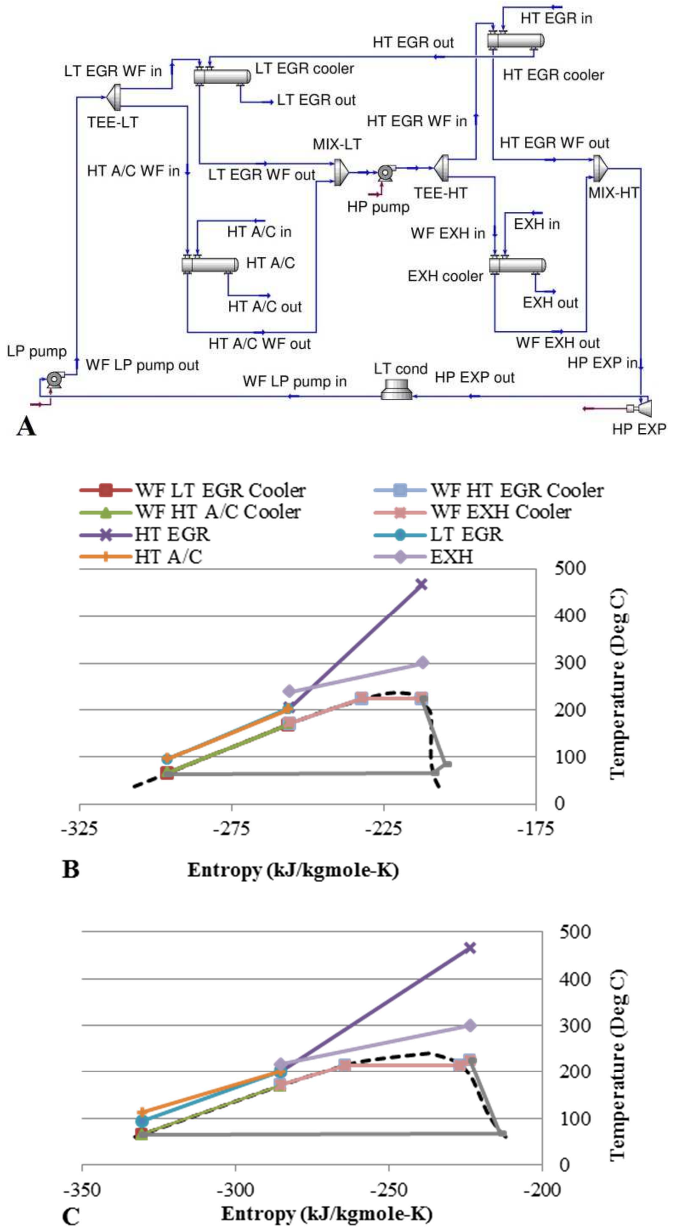


Fig. 6 (a) Arrangement 5: EGR, HT A/C and EXH, (b) Cycle operation using acetone, (c) Cycle operation using ethanol

Fig. 6b and 6c describe the cycle operating condition using acetone and ethanol as working fluids. When considering the heat sources (Fig. 6b, 6c and table 7), two differences are observed in the HT A/C and exhaust heat profiles. Ethanol cools exhaust stream to 225 vs. 238°C recovering larger portion of exhaust heat (40.54 vs. 30.07 kW) and cools HT A/C stream to 112 vs. 95°C recovering lower portion of HT A/C heat (37.72

vs. 45.22 kW). With nearly equal thermal efficiency (15.63 vs. 15.17%), ethanol results in higher increase in engine cooling load (13.43 vs. 4.21 kW).

With arrangements 1 and 5 the average heat addition temperature reduces due to heat recovery from HT A/C. In such arrangements acetone is most suited offering highly cooled EGR and HT A/C temperatures. For the arrangement in Fig. 6, using acetone will result in a net power of 25.86 kW with an increase of only 4.21 kW in the heat rejection by the engine cooling module (table 7). When compared to arrangement 4, for equivalent increase in engine cooling load (4.21 vs. 4.23 kW) arrangement 5 delivers 40% greater net power (25.86 vs. 18.04 kW).

Table. 7 Heat duty, net power and relative size for arrangement 5

	Ethanol	R30	Acetone
HT EGR cooler duty (kW)	68.32	68.15	68.01
LT EGR cooler duty (kW)	26.87	27.04	27.19
HT A/C cooler duty (kW)	37.72	35.36	45.22
EXH cooler duty (kW)	40.54	41.96	30.07
LT cond. duty (kW)	146.33	145.91	144.63
Net power (kW)	27.11	26.61	25.86
Thermal efficiency (%)	15.63	15.42	15.17
Total area/Net power	0.98	1.02	1.04
Δ Rad. load (kW)	13.43	15.35	4.21

3.6 Suitability of acetone and R30

To date there is no consensus as to which working fluid is best suited for HDDE. Working fluids under investigation by other authors include water, ethanol, and R245fa [2, 9, 10]. A fluid selection study conducted by the authors has highlighted the advantages of acetone and R30 over these fluids [11].

At the selected 65°C condensing temperature acetone and R30 offer super atmospheric condensing pressures (1.33, 2.28 bar) compared to sub-atmospheric pressures shown by water and ethanol (0.25, 0.62 bar). With sub-atmospheric pressures the ambient air may leak into the system and the thermodynamic efficiency will reduce.

Acetone and R30 also offer a higher overall conversion efficiency when compared to water and R245fa. This is also evident when comparing the results of net power for arrangement 2. Acetone and R30 produce 26.53 and 25.1 kW, respectively compared to water and R245fa which produce only 20.5 and 16.65 kW, respectively.

The development of a suitable expansion machine is lagging that of the heat exchangers, whether it is turbine or positive displacement expander coupled electrically or mechanically to the driveline. The properties of the selected working fluid greatly influence the design parameters. For the arrangements 1 to 5, ethanol, R30 and acetone show net power, relative heat exchanger size and change in engine cooling load within ±10%. A second valuable criteria used as an indicator of estimating the relative size and cost of BC is expansion volume flow ratio. To compare all the five fluids discussed so far, Fig. 7 shows the expansion volume flow ratio vs. net power produced for arrangement 2. The volume ratio depends on the properties of the fluid and gives an idea of the compactness of the expansion machine. When comparing the 5 fluids, R30, R245fa and acetone provide the best combination of highest net power for the lowest volume flow ratios, resulting in smaller expanders/turbines.

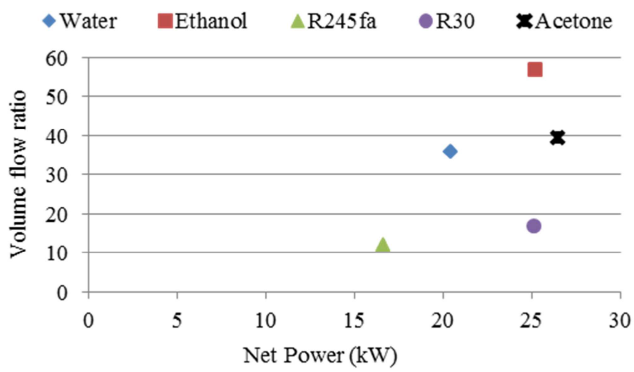


Fig. 7 Expansion volume flow ratio vs. net power produced for arrangement 2.

When acetone and R30 are compared to the only other high net power working fluid i.e. ethanol, they offer low pressure ratios (27.9:1, 16.7:1 vs. 55.2:1) and condensing specific volume (0.35, 0.13 vs. 0.84 m^3/kg). They also offer higher mass flow rate (0.27, 0.44 vs. 0.15 kg/s), molecular weight (58.1, 84.9 vs. 46.1 g/mol) and comparable or higher liquid densities (785, 1318 vs. 792 kg/m^3). All these properties shown by acetone and R30 are preferable for turbines and positive displacement expanders. With the lowest volume flow ratio per net power output, R30 is considered the most suitable. For maximum power with least impact on the cooling module (table 7) R30 produces an additional 9% of engine shaft power at the C100 condition with a 15.35 kW increase in the cooling load.

4 Bottoming cycles with two expansion stages

4.1 Arrangement 6: Dual loop parallel expansion

As the selected engine platform offers heat at different temperature levels, cycle arrangements using 2 different evaporating temperatures are also analysed. To investigate arrangements that deliver higher power than arrangement 3 whilst recovering heat from HT A/C, arrangement shown in Fig. 8a was examined. Fig. 8b shows the corresponding cycle operating condition with R30.

The working fluid leaving the condenser is distributed and pumped by 2 pumps for different temperature levels of evaporation. The high temperature high pressure loop recovers heat from the EGR and HT EXH cooler, whereas the low temperature low pressure loop recovers heat from the LT EXH cooler and HT A/C. The working fluid in all heat exchangers are preheated, evaporated and superheated (if needed). After expansion at 2 different pressure levels, working fluid streams are mixed prior to the condenser inlet.

BCs with 2 pressure levels have to operate under the best thermodynamic conditions to achieve a better integration. As the high pressure expansion inlet condition (temperature and pressure) is same as that of arrangements 1 to 5, optimisation of such cycles are subjected to parametric study of the low pressure cycle.

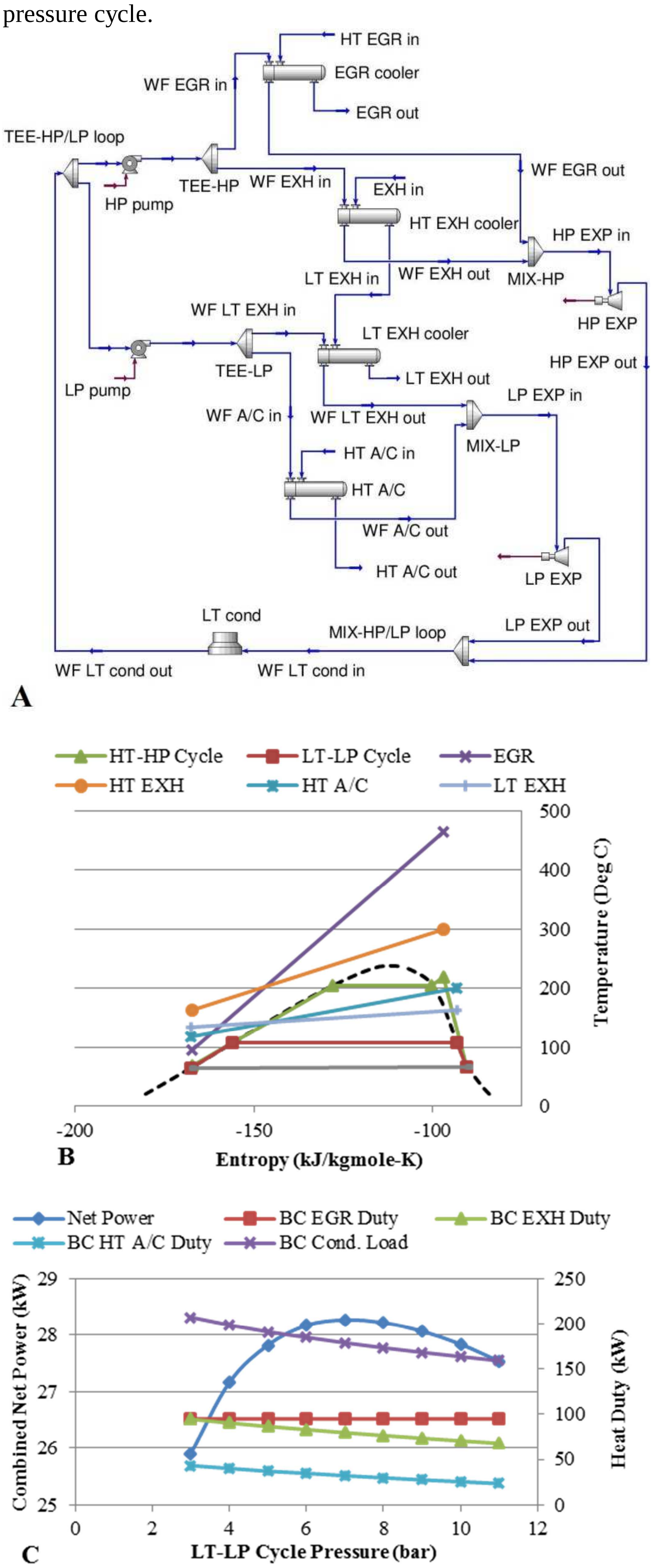


Fig. 8 (a) Arrangement 6: Dual loop parallel expansion, (b) Cycle operation using R30, (c) Parametric optimisation of the LP cycle

Fig. 8c shows the results of the parametric optimisation by varying the pressure of the low pressure cycle. Also included are the heat load of all the heat exchangers and the BC condenser. For maximum combined net power the pressure levels in the 2 cycles is 40 and 7 bar, respectively.

Since the heat input into the low pressure loop is of lower quality (163-134 and 200-126 °C) and quantity (13.7 and 32.15 kW), the evaporation temperature is only 107 °C. As a result the low pressure loop has lower pressure ratios (2:1) and expander power (4.7 kW). Due to the capacity and pressure ratios positive displacement expanders like the scroll expanders are most suitable. For the high pressure loop, evaporation conditions are the same as that considered in arrangement 1 to 5. The high pressure loop will require a high pressure ratio piston expander (16.6:1) for R30.

At this condition the net power produced using R30 is 28.27 kW and the total increase in heat rejection is 51.65 kW (table 8). When this result is compared with arrangement 3 the net power increases only by 4 % (28.27 vs. 27.21 kW) while the increased heat rejection load remains nearly same (51.65 vs. 52.8 kW). Arrangement 6 recovers an additional 32 kW of heat compared to arrangement 3, however at a 12% lower cycle efficiency (13.36 vs. 15.53%). The added complexity and cost of a scroll expander for such small benefits in net power by arrangement 6 is not justifiable.

Table. 8 Heat duty, net power and relative size for arrangement 6

	Ethanol	R30	Acetone
EGR cooler duty (kW)	95.20	95.20	95.20
HT EXH cooler duty (kW)	64.69	66.15	77.58
HT A/C cooler duty (kW)	29.43	32.15	34.95
LT EXH cooler duty (kW)	11.51	13.77	3.23
LT cond. duty (kW)	172.55	179.00	182.20
Net power (kW)	28.29	28.27	28.75
Thermal efficiency (%)	14.08	13.64	13.63
Total area/Net power	0.78	0.84	0.79
Δ Rad. load (kW)	47.92	51.65	52.06

4.2 Arrangement 7: Dual loop series expansion

An important limitation of the 40 bar maximum cycle pressure considered in arrangements 1 to 6 with a pure working fluid is the large heat absorbed during isothermal boiling. To improve high temperature thermal match, working fluids will require higher pressures. As higher pressure cycle will require higher average source temperature the most suitable arrangement to explore high pressure expansion using ethanol, R30 and acetone will be arrangement 2. However, considering R30 with increased maximum pressure in arrangement 2 to 60 bar increases the cycle efficiency (16.7 vs. 15.56%) without increase in the net power (25 vs. 25.1 kW). This is because at such high pressures, following the pinch point limitations imposed, the heat recovered in the EXH cooler reduces from 66.15 to 55.3 kW. As a result, the exhaust stream is only cooled to 186°C instead of the 163°C as shown in Fig. 3b. Furthermore, to realise the high pressure cycle two stage expansions will be needed. The additional cost of the second expansion machine for no gains in net power produced is unfavourable.

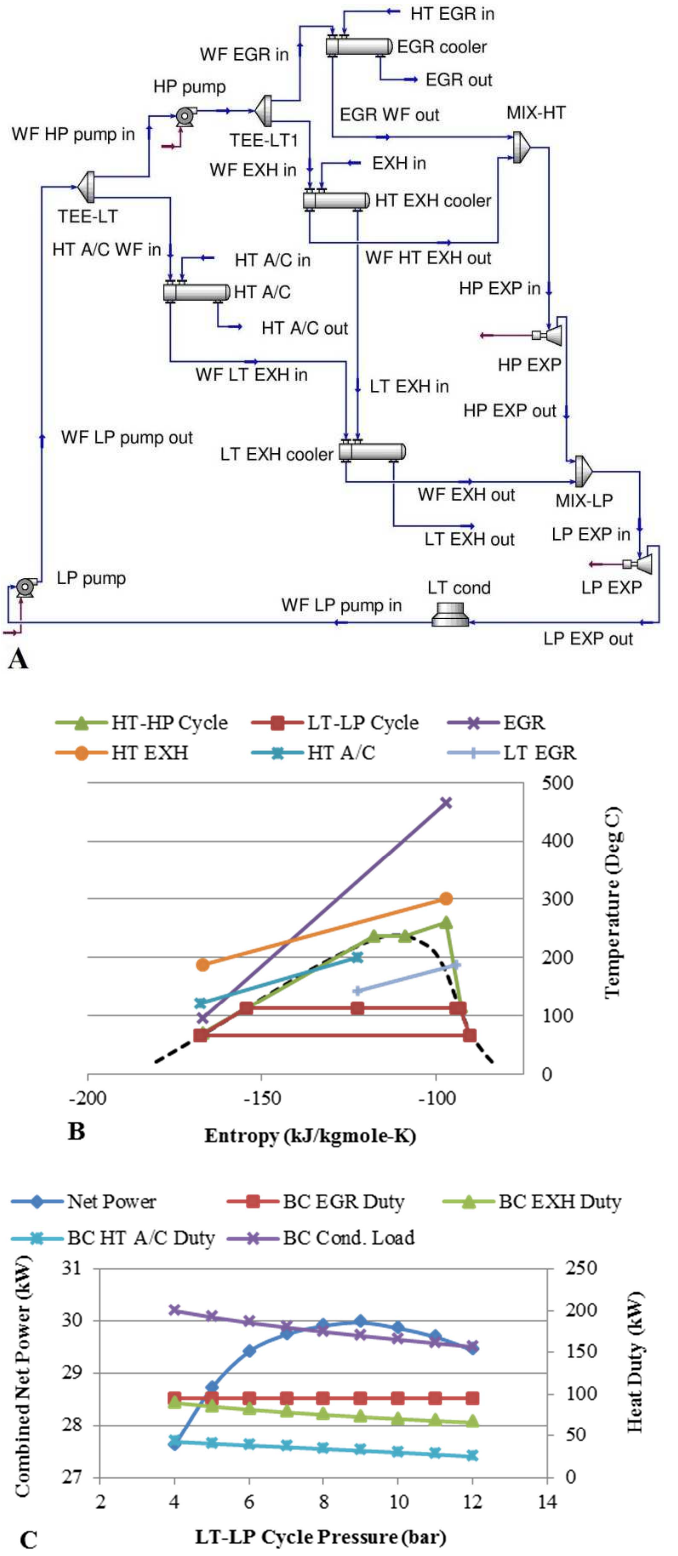


Fig. 9 (a) Arrangement 7: Dual loop series expansion, (b) Cycle operation using R30, (c) Parametric optimisation of the LP cycle

To exploit maximum exhaust heat with higher cycle pressures whilst recovering HT A/C heat, arrangement shown in Fig. 9a is proposed. Like arrangement 6, this layout also consists of two different evaporating pressures and the cycle operating condition is shown in Fig. 9b using R30. The working fluid leaving the low pressure pump is separated into 2 streams. For the high pressure loop the high pressure pump further raises the pressure to 60 bar. The working fluid is then distributed in the high pressure loop. In the EGR and HT EXH cooler the working fluid is preheated, evaporated and superheated. Apart from an increase of maximum cycle pressure from 40 to 60 bar the working fluid is also superheated to 260°C. It is assumed that ethanol, R30 and acetone do not undergo thermal degradation at this temperature. This maximum temperature is ≤ 25°C above the critical temperature of the working fluids.

For the low pressure loop, the maximum pressure is governed by the low pressure pump. In the low pressure loop heat from the HT A/C is used to preheat and partially evaporate the working fluid. The remaining portion of the exhaust heat is then used to fully evaporate the working fluid. The dry working fluid stream exiting the high pressure expansion is subsequently mixed with the low pressure stream and injected into the low pressure expansion stage.

Fig. 9c shows the results of parametric optimisation of the combined arrangement subjected to varying the maximum pressure in the low pressure cycle. With the high pressure expansion inlet condition fixed at 60 bar and 260°C, Fig. 9c shows the maximum net power being generated when the low pressure cycle is at 9 bar. Similar optimisation was also performed for other fluids resulting in high and low pressure ratios of 13:1 and 6.4:1, 6.3:1 and 4.1:1, 13:1 and 3.3:1 for ethanol, R30 and acetone, respectively. Such pressure ratios are favourable for use with positive displacement expanders. Furthermore, this is the only arrangement considered so far that has sufficiently low pressure ratios with ethanol and allows the use of only positive displacement expanders.

Considering R30, arrangement 6 has shown the greatest power output (28.27 kW). However due to greater hardware complexity for only 4% increase in net power compared to arrangement 3, the arrangement was considered unsuitable. Table 9 shows the optimised results for arrangement 7. When compared to arrangement 3, R30 produces almost 10% greater net power (29.86 vs. 27.21 kW). The advantage of this arrangement is not restricted to higher net power but it also reduces the increase in engine cooling load from 52.8 to 40.35 kW.

Table. 9 Heat duty, net power and relative size for arrangement 7

	Ethanol	R30	Acetone
EGR cooler duty (kW)	95.20	95.20	95.20
HT EXH cooler duty (kW)	54.80	54.15	65.55
HT A/C cooler duty (kW)	30.17	29.54	35.19
LT EXH cooler duty (kW)	16.00	16.06	11.14
LT cond. duty (kW)	166.62	165.09	177.52
Net power (kW)	29.54	29.86	29.55
Thermal efficiency (%)	15.06	15.32	14.27
Total area/Net power	0.82	0.83	0.81
Δ Rad. load (kW)	41.26	40.35	47.13

Furthermore, due to improved thermal match in high pressure loop and additional benefits of increase in the evaporation temperature of low pressure loop from 107 to

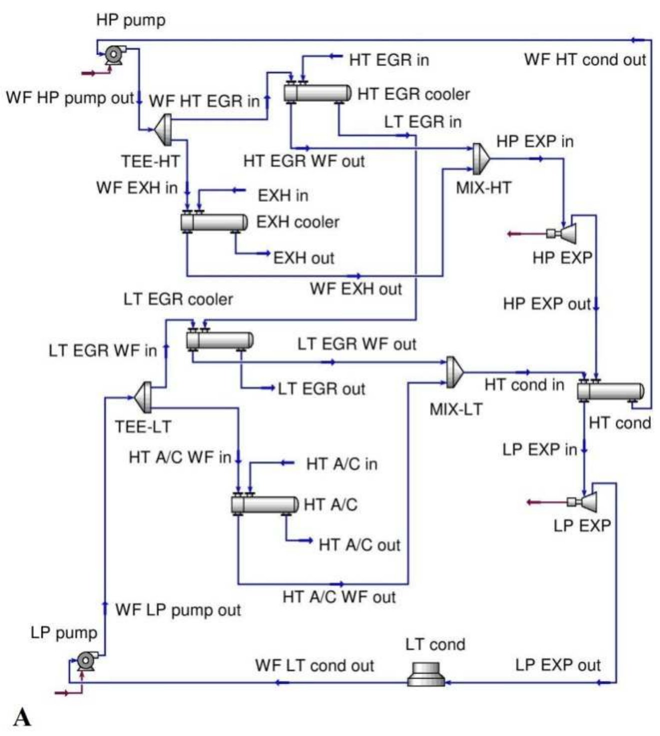
114°C the combined cycle efficiency for arrangement 7 improves from 13.64 to 15.32 % over arrangement 6.

The cycle operating condition of arrangement 7 with acetone differs to that of ethanol and R30. This is because the critical pressure of acetone is reached at much lower pressure (44.7 vs. 61.4, 60.8 bar). Therefore supercritical fluid parameters are realised. Supercritical state bypasses the two-phase region, which allows it to have a better thermal match as evident with the reduced LMTD in the EGR cooler from (76.78 to 71.78 °C) when compared to arrangement 6.

4.3 Arrangement 8: Cascade

As shown, arrangement 7 is considered most favourable with a maximum power output of 29.86 kW. However, in order to evaporate the working fluid in the low pressure loop, the cycle has to use the low quality exhaust heat exiting the HT EXH cooler. The low cycle efficiency of the low pressure loop results in an increased load on the engine cooling module (40.35 kW). Considering acetone, with the exception of arrangement 5 which recovers all the HT A/C heat (45.22 kW), the other arrangements that recover HT A/C heat do not show this benefit (arrangement 1,6 and 7: 23.18, 35.19 and 34.95 kW, respectively). Hence, high pressure solution that recover large amounts of HT A/C heat, with least impact on the engine cooling module were investigated by the use of cascade arrangement as shown in Fig. 10a.

Cascade arrangement consists of two inter-related BCs. The high temperature cycle utilises heat from the HT EGR cooler and EXH cooler and low temperature cycle recovers heat from the LT EGR cooler and HT A/C. The condensing load of the high temperature cycle is used to fully evaporate the working fluid of the low temperature cycle.



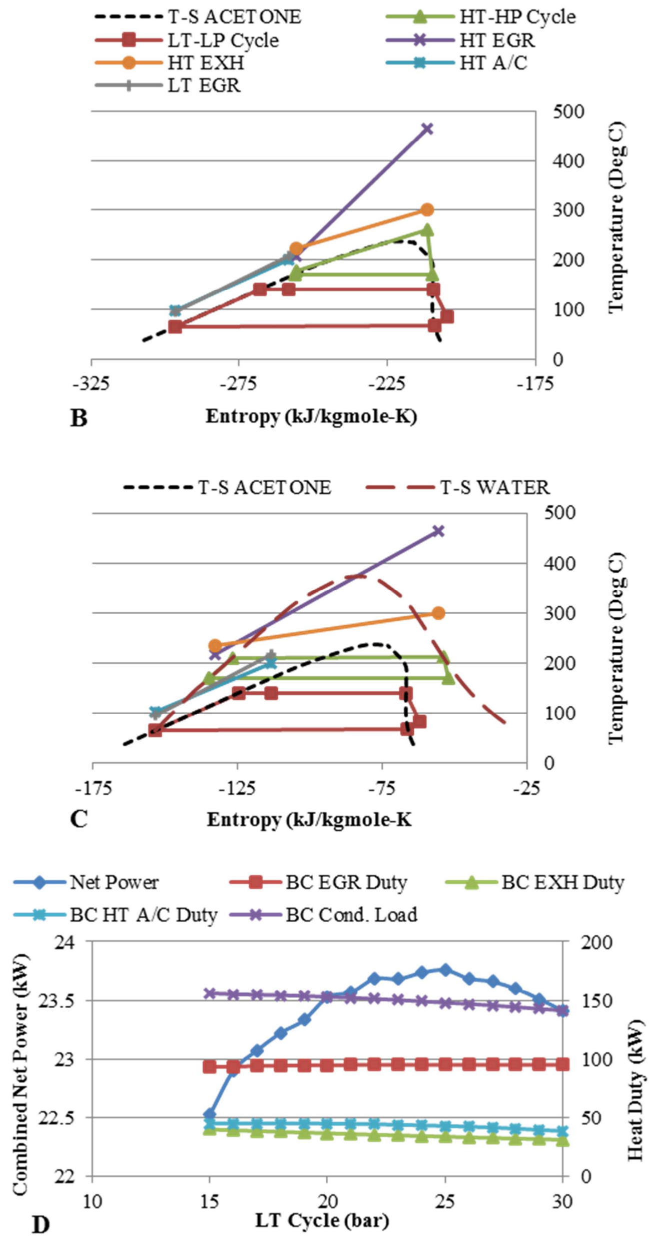


Fig. 10 (a) Arrangement 8: Cascade, (b) Cycle operation using acetone, (c) Cycle operation using water/acetone (d) Parametric optimisation of the LT R30/acetone cycle

Fig. 10b shows the cycle operating condition with acetone as the working fluid. When using organic fluids the high pressure expander inlet conditions are kept constant at 60 bar and 260°C. Fig. 10b shows the high temperature cycle with supercritical mode. Depending on the heat load available in high temperature condenser, the working fluid in the low temperature cycle may either leaves as saturated or as two phase liquid. In Fig. 10b, the low temperature working fluid enters the high temperature condenser with a dryness fraction of 0.17.

Using single fluids cascade cycles were investigated using ethanol and acetone. However, if leakage and mixing of the working fluids in the high temperature condenser can be guaranteed, the possibility also exists of using two different working fluids for more suitable operation and/or performance. Fig. 10c shows such a case. Here water is used as the working fluid for the high temperature cycle and acetone for the low temperature cycle. Contrary to the maximum pressure of 60 bars when using organic fluids, the maximum pressure was kept fixed at 20 bars when using water. This upper pressure of 20 bars provided the highest overall conversion efficiency for the high temperature cycle.

With the high pressure expander inlet pressure and temperature fixed, optimisation of net power was investigated by varying the pressure in the low temperature cycle. Fig. 10d shows the results with R30 (high temperature) and acetone (low temperature). With cycle pressures of 60/25 bar the cascade cycle generated a maximum net power of 23.68 kW (table 10). For this condition 44.23 kW of HT A/C heat is recovered with an increase in the engine cooling load of 11.75 kW.

Table. 10 Heat duty, net power and relative size for arrangement 8

	Ethanol	Acetone	R30/ Acetone	Water/ Acetone
HT EGR cooler duty (kW)	67.12	67.15	67.98	64.48
EXH cooler duty (kW)	34.13	37.62	35.43	31.22
HT cond. duty (kW)	94.77	98.34	96.64	90.35
HT A/C cooler duty (kW)	37.86	44.16	44.23	41.73
LT EGR cooler duty (kW)	27.20	27.92	27.00	30.58
LT cond. duty (kW)	142.81	152.73	150.97	145.85
Net power (kW)	23.50	24.11	23.68	22.17
Thermal efficiency (%)	14.13	13.64	13.56	13.19
Total area/Net power	1.18	1.23	1.23	1.12
Δ Rad. load (kW)	10.63	13.51	11.75	9.05

When results of acetone are compared for arrangement 8 and 5, it is seen that arrangement 8 produce 6% lower net power (24.11 vs. 25.86 kW) and increases the additional heat to be rejected by the engine cooling module from 4.12 to 13.51 kW. With marginally higher net heat input (176.85 vs. 170.5 kW), arrangement 8 delivers a 13% lower cycle efficiency (13.19 vs. 15.17%).This is primarily due to the 30°C minimum approach considered for the high temperature condenser. As a consequence of this the heat exchanger footprint per net power is 20% greater compared to arrangement 5 (1.23 vs. 1.04 m^2/kW). Hence, cascade systems are considered unsuitable due to no added benefit in net power, higher system pressures, and instillation cost and size (due to the HT condenser).

4.4 Arrangement 9: Regenerative power cycle

As shown by Fig. 2c, R245fa shows the worst performance due to its lower critical temperature. The maximum thermal efficiency using R245fa is only 8.67%, this is due to its lowest evaporation temperature of 151°C. However, the benefit of fluids with lower critical temperatures then ethanol, R30 and acetone are the lower volume flow ratios (Fig. 7).

Dry fluids discussed in arrangements so far involved saturated vapour expansion. To increase the average cycle temperature, working fluids can be superheated. The effect of superheating a working fluid can be governed by the rate at which the isobars diverge in the superheated region. With increasing superheat between two isobaric curves, the thermal efficiency decreases for dry fluids (converging superheated isobars) while it increases for wet fluids (diverging superheated isobars). As the increase in rate of heat addition temperature is lower than that of the heat rejection temperature, superheat contributes negatively to the cycle efficiency for dry fluids.

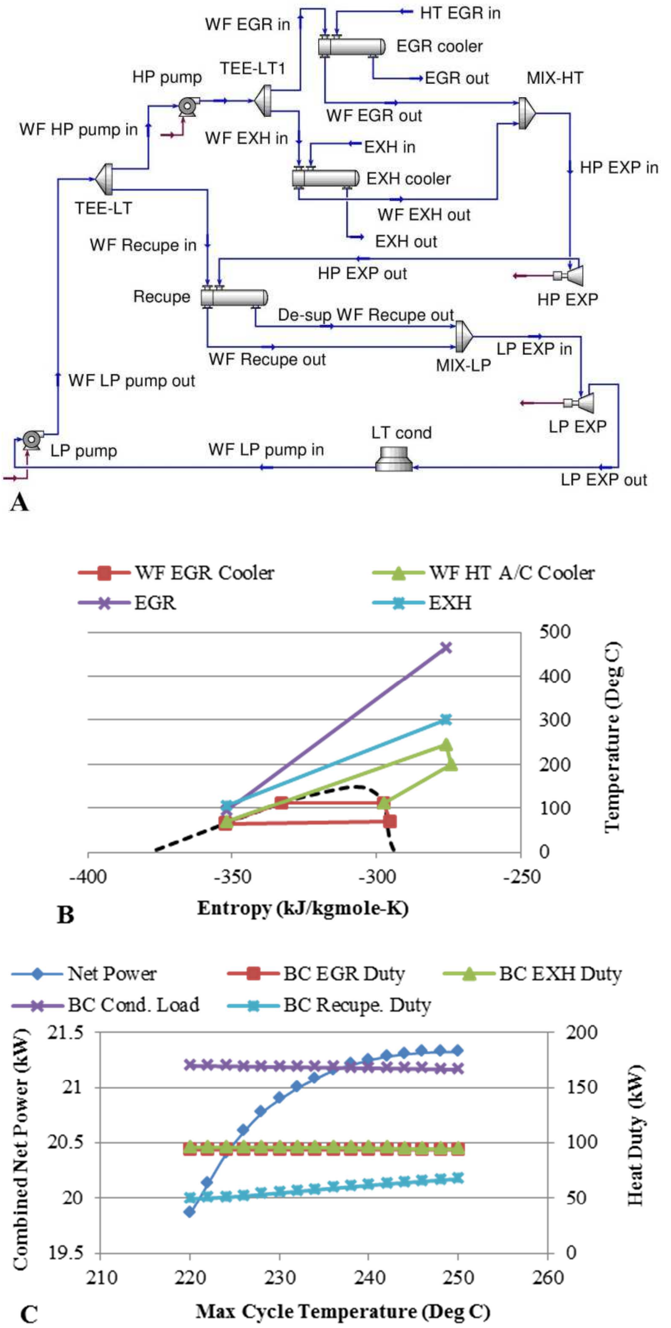


Fig. 11 (a) Arrangement 9: Regenerative power cycle , (b) Cycle operation using E152a, (c) Sensitivity analysis of the high pressure expander temperature

The expansion of superheated vapours of dry fluids shifts the expansion exit further away from the two-phase region towards the superheated region. This substantial increased amount of superheat will adds to the cooling load in the condenser. As the temperature of the expanded vapour will be more than that of the liquid at the inlet of the preheater, it is possible to use a regenerator. A regenerator improves thermal efficiency in the case of dry fluids, due to increase in the average heat addition temperature. However, a regenerator also reduces the net heat recovered, increasing the quantity of low temperature unrecovered heat. Therefore, a standard regenerative cycle will increase the system cost, complexity, total heat exchanger area without increasing the net power output.

To take the advantage of lower volume flow ratios offered by low critical temperature fluids, while increasing the thermal efficiency and net power produced arrangement 9 shown in Fig. 11a is proposed. To comply with directive 2006/40/EC (Global Warming Potential < 150 years), E152a is considered as a substitute to R245fa (110 vs. 1030 years). E152a offers comparable critical temperature (148.9 vs. 154.1 °C) and pressure (43.3 vs. 36.4 bar).

Fig. 11b shows the cycle operating condition using E152a. Arrangement 9 consists of two evaporating temperature levels where the high pressure pump is used to raise the working fluid pressure to 60 bar. The working fluid exiting EGR and EXH cooler is at high temperature and supercritical pressures. After the high pressure expansion, the working fluid with large amounts of superheat is used to preheat and evaporate the low pressure fluid entering the regenerator. In this heat transfer, the post expansion superheated stream is reduced to saturated vapour levels. This stream is mixed with the evaporated low temperature stream and injected into the low pressure expander. A pinch point of 30 °C was considered in the regenerator.

Table. 11 Heat duty, net power and relative size for arrangement 9

	R245fa	E152a
EGR cooler duty (kW)	95.50	93.46
EXH cooler duty (kW)	95.50	93.46
Recupe. duty (kW)	58.06	58.61
LT cond. duty (kW)	169.41	166.63
Net power (kW)	21.28	22.02
Thermal efficiency (%)	11.14	11.78
Total area/Net power	1.49	1.48
Δ Rad. load (kW)	74.22	71.43

The pressure at the exit of the low pressure pump is controlled by the amount of heat available in the regenerator. With the maximum pressure of the cycle fixed at 60 bar to complete the optimisation of the combined unit, a sensitivity analysis of the high pressure expander inlet temperature was performed. Fig. 11c shows no added benefit in the net power produced with high pressure expander temperatures above 245°C. At this point the maximum net power produced is 22.02 kW with cycle efficiency of 11.78% (table 11). At this condition, E152a shows very favourable pressure and volume flow ratios of 2.6:1, 2.7:1 and 2.8:1, 3.1:1 across the high and low pressure expanders, respectively.

When results of R245fa for arrangement 9 are compared with arrangement 2, which also recovers EGR and exhaust heat. For nearly equal heat addition (191 vs. 192.15 kW), arrangement 8 has a 28% higher thermal efficiency (11.14 vs. 8.67%). As a result arrangement 9 produces higher net power (21.28 vs. 16.65 kW). When the impact of increased heat exchanger size due to the regenerator is considered arrangement 8 shows a 9% increase in the heat exchanger area per unit net power output (1.49 vs. 1.38 m^2/kW). No other arrangement considered generates higher net power for low critical temperature (≈150 °C) fluids than arrangement 9.

4.5 Arrangement 10: Superheated power cycle

For ethanol, R30 and acetone, one means of improving the performance was shown in arrangement 7. By using high pressures (60 bar), the improved thermal matching was observed in high temperature loop. However, even arrangement 7 has drawbacks. The increased high pressure from 40 to 60 bar will require relatively more expensive heat exchangers. Furthermore, the heat recovered in the LT EXH cooler to evaporate the working fluid (R30) in the low temperature loop contributes significantly to the 40.35 kW added to the cooling module. This 40.35 kW is much lower than 51.65 kW as seen in arrangement 6. Nonetheless further reduction of load on the engine cooling module is desirable.

For the arrangement of wet fluids like R30 and isentropic fluids like acetone considered so far, the maximum expansion inlet temperature was at minimum superheat and saturated vapour, respectively (with the exception of arrangement 7). Superheat levels were restricted to satisfy the minimum dryness fraction (≈0.99) at the outlet of the expansion. From a thermodynamic point of view, if the thermal level in the source is high enough as is the case of arrangement 2, superheated cycles can deliver improved efficiencies. With diverging isobars in the superheated region, cycles with ethanol, R30 and acetone will see higher average working fluid temperatures and higher net power. However, large superheating to higher temperatures has the undesirable effect of increased load on the condenser due to de-superheating.

To benefit from superheating, reducing pressures from 60 to 40 bar and avoid de-superheating in the condenser, arrangement 10 shown in Fig. 12a is proposed. The corresponding cycle operating condition is shown in Fig. 12b using acetone. In the high pressure loop, the high pressure pump increases the pressure to 40 bar. The working fluid exiting the EGR and EXH cooler is superheated at 260°C. In the low pressure loop, the working fluid is preheated and partially evaporated in the HT A/C. The large superheated vapour exiting the high pressure expansion is mixed with the two phase liquid exiting the HT A/C to form saturated vapour. This saturated vapour is injected into the low pressure expansion stage. To recover larger amounts of heat from the HT A/C, the working fluid exiting the high pressure expansion should have high superheat levels.

With the high pressure and temperature fixed at 40 bar and 260°C, parametric optimisation of the low pressure cycle was performed for maximum power recovery. Fig. 12c shows the influence of the low pressure cycle on the net power produced using R30. With high and low pressures of 40 and 10 bar, the combined system shows maximum net power of 28.17 kW (table 12).

Comparing to the highest net power arrangement i.e. arrangement 7, arrangement 10 shows a 6% reduction in maximum power (29.86 vs. 28.17 kW). However, arrangement

10 is considered more suitable as the increase in heat rejection is much lower (30.69 vs. 40.35 kW) and the cycle will produce an additional 9.5% of engine shaft power. Furthermore, both the arrangements recover nearly equal amount of heat form HT A/C (29.54 vs. 28.68 kW), have similar thermal efficiencies (15.32 vs. 15.42%) and heat exchanger area per unit power output (0.83 vs. 0.81 m^2/ kW).

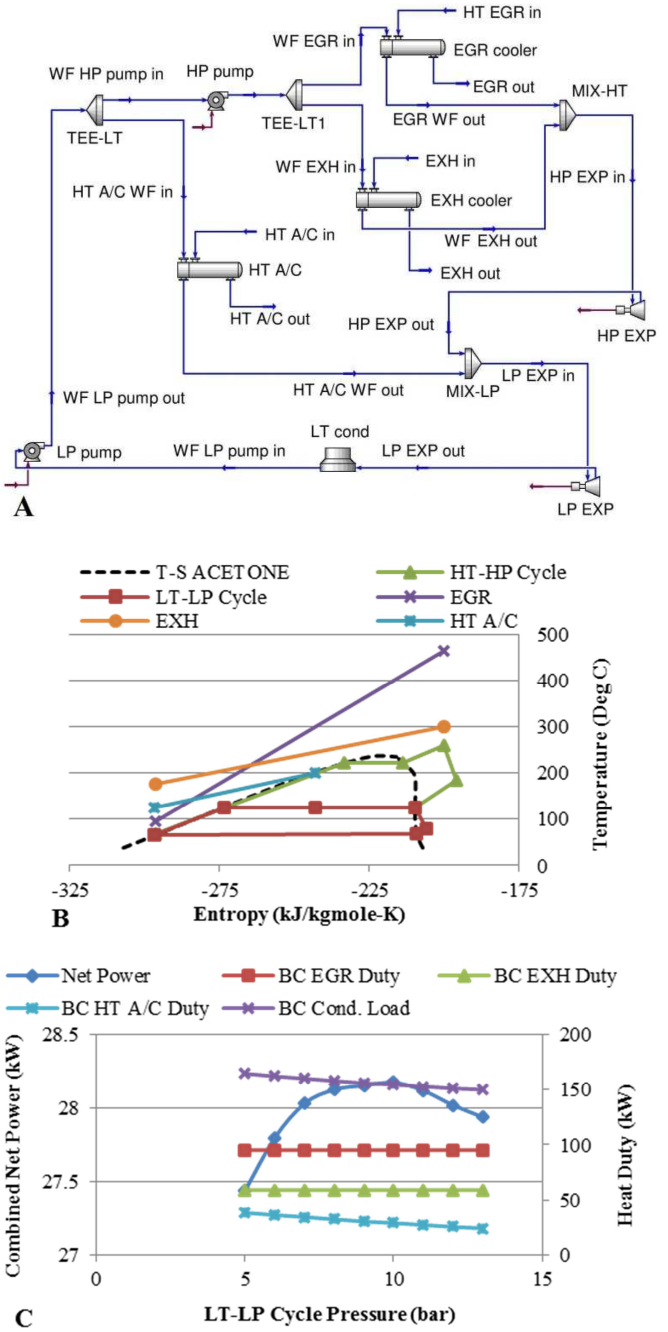


Fig. 12 (a) Arrangement 10: Superheated power cycle, (b) Cycle operation using acetone , (c) Parametric optimisation of the LP cycle using R30

Arrangement 10 also provides a flexible control strategy over a wider engine operating map. For high waste heat conditions the BC will operate as shown in the T-S diagram in Fig. 12b. When the available waste heat decreases, especially in HT A/C for example at A25 (low speed, low load), the low pressure loop is by-passed and the BC will operate as shown in Fig. 3b. For such conditions the BC will effectively operates as an EGR and EXH parallel layout.

Table. 12 Heat duty, net power and relative size for arrangement 10

	Ethanol	R30	Acetone
EGR cooler duty (kW)	95.20	95.20	95.20
HT A/C cooler duty (kW)	28.41	28.68	32.67
EXH cooler duty (kW)	57.42	58.86	59.96
LT cond. duty (kW)	153.06	154.57	159.27
Net power (kW)	27.96	28.17	28.56
Thermal efficiency (%)	15.45	15.42	15.21
Total area/Net power	0.77	0.81	0.76
Δ Rad. load (kW)	29.46	30.69	31.40

5 Optimal performance and system related trade-off

To summarise the impact of the different BCs and operating conditions on the 10L engine, Fig. 13 shows the net BC power (green), increase in the engine cooling load (red) and relative BC heat exchanger size (purple) for the 10 arrangements. The working fluid for arrangement 9 is E152a. All other arrangements in Fig. 13 use R30.

As the negative impact of the BC on the engine cooling load should be minimal, arrangement 3, 6 and 9 are considered unsuitable. They will increase the heat rejection load on the cooling module by 52.8, 51.65 and 71.73 kW, respectively. This is an increase of over 25% heat load on the cooling module.

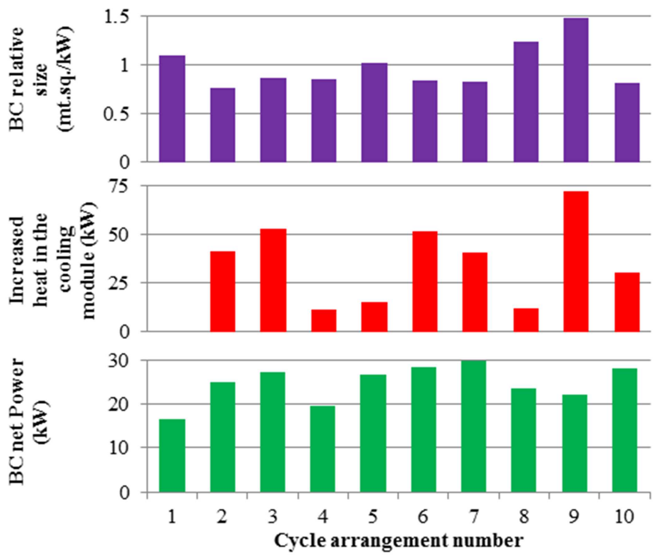


Fig. 13 Comparision of cycle arrangements for optimal performance and system related trade-off

Comparing the remaining single stage expansion arrangements (1, 2, 4 and 5), arrangement 2 (EGR and EXH parallel) and 4 (EGR and EXH series) are unfavourable in the present application as they do not recover any heat from the HT A/C. These two arrangements are more suited to a low EGR rate engine, where the load on the HT A/C is much lower. Hence, for maximum power recovery, with the least impact on the cooling module arrangement 5 is considered the most suitable within the single stage expansion arrangements.

Within the remaining two stage expansion cycles (7, 8 and 10), the arrangement 8 (cascade) has a low impact on the cooling module but due to one of the largest BC footprints, cascade arrangements are also discounted. For the two stage expansions arrangement 10 is considered more suitable over 7 due to nearly equal net power (28.17 vs. 29.86 kW), BC heat exchanger size (0.81 vs. 0.83 m^2/ kW) and lower increased load on the cooling module (30.69 vs. 40.35 kW).

The two optimal arrangements i.e. arrangement 5 and 10 also have lower maximum pressures than arrangement 6, 7, 8 and 9 (40 vs. 60 bar). The benefits of acetone and R30 were highlighted in section 3.6, however, due to the highest power for the lowest volume flow ratio R30 is considered the most suitable for 40 bar single stage expansion. For arrangement 5, a piston expander with volume flow ratio and pressure ratio of 16.7 and 16.6, respectively, suffices. Arrangement 10 incurs the additional cost of a compact expander. However, the low pressure ratios due to the 2 stage series expansion also allow acetone and ethanol as alternatives. The two selected arrangements will generate over 9% of engine shaft power at C100, improving the fuel consumption by over 8.2%.

6 Conclusion

With increasing fuel prices and anticipated regulations for CO_2 emissions from long haul trucks, the focus will be even more towards effective thermal management of engines by converting waste heat into mechanical or electrical power. This paper analysed a systems approach to Euro 6 NOx reduction and WHR using a BC on a 10L heavy duty Diesel engine. With expansion states including saturated vapour, superheated and supercritical phase, 10 different cycle topologies were investigated using thermal energy recovered from the high temperature after-cooler, EGR cooler and exhaust gas streams. The topologies were analysed in terms of cycle efficiency, net power output, relative heat exchanger size and volumetric expansion ratio. Optimisation of the BC was investigated by the selection of working fluid, its cycle operating condition and associated process integration. The paper discussed the influence of working fluid on operating pressures, latent heat, expander design, overall conversion efficiency and effectiveness of superheating.

A single stage expansion (arrangement 5: EGR, HT A/C and EXH) and a two stage expansion (arrangement 10: Superheated power cycle) are considered the most optimal for performance and system related trade-offs. The 2 proposed arrangements use sub-critical pressures of 40 bar, have favourable heat exchanger footprint, offer high net power and have a low negative impact on the engine cooling load. For single stage expansion, R30 is suggested as the best working fluid due to the highest power for the lowest volume flow ratio. A piston expander with volume flow ratio and pressure ratio of 16.7 and 16.6, respectively, suffices. For the two stage expansions, acetone and ethanol can also be considered, as parametric analysis resulted in pressure and volume flow ratios $\leq$ 10:1 for all the three fluids in both expanders. The two arrangements will generate over 9% of engine shaft power at a high load condition, improving the fuel consumption by over 8.2%.

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