The split cycle engine and its impact on the vehicle cooling system

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Abstract

The split cycle engine represents a radical departure from conventional internal combustion engines. In the split cycle engine, the compression and combustion processes occur in different cylinders, enabling extreme charge air cooling, intra cylinder waste heat recovery and independent optimisation of the compression and combustion chambers. Combining these benefits, thermal efficiencies of over 50% and approaching 60% in some applications are theoretically possible. The split cycle engine is of particular interest in commercial vehicle applications where load factors are high and savings in fuel consumption of major economic benefit to end users.

The energy balance of the split cycle engine is markedly different to conventional engines with significantly more energy being rejected through charge air cooling but correspondingly less through the engine cooling system. How the charge air is cooled will affect how thermal energy is rejected to the environment and the consequential impact on the vehicle cooling system.

In this paper, the split cycle engine will be described and the energy balance compared with a conventional heavy duty diesel engines. Methods of charge air cooling; through multi stage compression with intercooling, direct cooling with a water spray and by the injection of a cryogenic liquid spray will be discussed. The overall energy balance, impact on powertrain efficiency and heat exchanger sizing for the water spray cooling case is compared with a comparable diesel engine. Finally, the consequences of the split cycle on component thermal loading will be discussed and strategies for controlling critical component temperatures described.

1 INTRODUCTION

The long haul commercial vehicle presents a number of challenges in improving the fuel economy of the vehicle. The mean load factor on a typical journey is higher than other on road applications, such as passenger cars and light commercial vehicles, limiting the benefit of downsizing and hybridisation of the power train [1]. As such, the Internal Combustion Engine (ICE) is likely to remain the primary power source in this application for the foreseeable future

[2]. Improvements to the IC engine have been proposed, including waste heat recovery and advanced thermodynamic cycles. [3, 4]. Inevitably, recovery of waste heat for conversion to power and modification to the ICE cycle will impact the energy balance of the system and the vehicle cooling system.

In this paper, we describe a new thermodynamic cycle with integrated waste heat recovery and the impact of the new cycle on the system energy balance and vehicle cooling system. The new cycle, referred to as the 'split cycle', separates the compression and combustion strokes into separate chambers. This has a number of benefits including:

1) Reduction of the compression work through quasi-isothermal compression.
2) Decoupling of the compression and expansion strokes enabling a Miller cycle and independent optimisation of the chambers, in particular the surface temperatures to manage thermal losses.
3) High pressure waste heat recovery between the compression and combustion cylinders by recuperation of exhaust energy.

The split cycle engine is described, focusing on the impact of the cycle on the vehicle cooling system. Three methods of achieving isothermal compression are described and assessed in the context of the commercial vehicle transport application. The energy balance of the split cycle engine is then compared with a conventional heavy duty diesel engine and the relative sizes of key components are presented. Finally, a comparison is made with other methods of waste heat recovery.

2 THE SPLIT CYCLE ENGINE

2.1 Description of the Cycle

The split cycle engine was first described by Coney [5], and later by Jackson [6] and Dong [7]. The variant of the split cycle engine considered in the present work is shown in figure 1. Air is compressed in a supercharger (1-2) and then cooled (2-3) in an air-air intercooler. The charge air is further compressed in a compression cylinder (3-4), into which water is injected to cool the charge air during the compression stroke. The water is separated from the charge air (4-5) and the charge air heated in a recuperator (5-6) using heat from the engine exhaust.

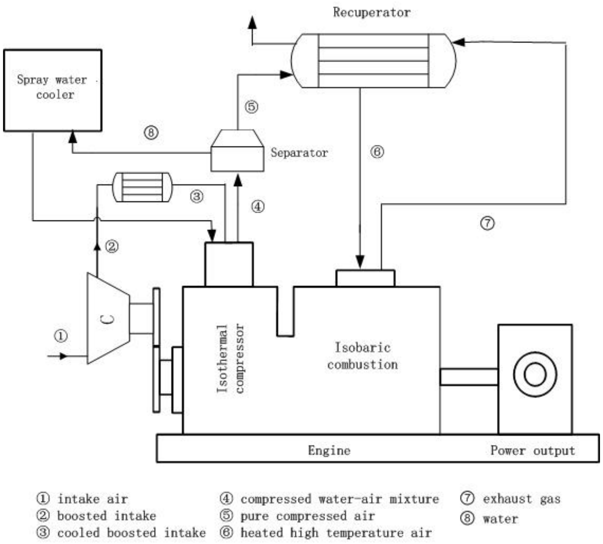

Figure 1: Schematic of a split cycle engine

The hot high pressure air is then transferred to a combustion cylinder, mixed with fuel which is burnt and work is extracted through the piston (6-7). After expansion, the combustion products are removed from the combustion cylinder and used to pre-heat the charge air in the recuperator before being exhaust to atmosphere. The recovered water from the compressor cylinder is cooled and re-injected into the compressor cylinder completing the split cycle. Heat is therefore rejected at low temperature through the spray water cooling, but recovered at high temperature from the exhaust between the compressor and combustor cylinders. There are other methods of achieving isothermal compression, which will be described in section 4.

2.2 Cycle Optimisation

A comprehensive analysis of the split cycle engine was recently reported by Dong [7], using the AMESim software [8]. The key assumptions of the modelling results used in the current work are listed in table 1 and optimised engine parameters for a 350kW shaft output engine in table 2.

Table 1: Modelling assumptions

Supercharger Efficiency	85%
Recuperator Effectiveness	0.73-0.86 (upon to engine load conditions)
Combustor chamber wall temperature	610 K
Phase of heat release CA 50	14 ° ATDC
Air Fuel Ratio	21:1

Table 2: Optimised engine parameters (derived from [7])

Stroke Type	2 stroke	Intake valve opening (IVO)	17° BTDC (compression chamber) 5° BTDC (expansion chamber)
Number of cylinders	6 (3 combustor +3 compressor)	Intake valve closing (IVC)	174° BTDC (compression chamber) 10° ATDC (expansion chamber)
Bore [mm]	136	Exhaust valve opening (EVO)	5° BTDC (compression chamber) 152° ATDC (expansion chamber)
Stroke [mm]	143	Exhaust valve closing (EVC)	10° ATDC (compression chamber) 2° ATDC (expansion chamber)
Compression ratio	23	Design speed [rpm]	1200
Expansion Ratio	27.5	Target power	350 kW
Compressor discharge pressure	110 bar		

The temperature of the spray water is a key factor which affects the isothermal compression process. The impact of spray water temperature on the cycle efficiency is presented in figure 2, using the theoretical analysis method described by Dong [7]. Maintaining the spray water temperature below 72°C has a minimal impact on the overall efficiency, whereas the efficiency significantly degrades above this value. The reason for this is the effectiveness of heat recovery is diminished as the compressor outlet temperature approaches the exhaust temperature of the combustor cylinder. For the stationary application described by Coney [5] a spray water temperature of 35°C was assumed, which is practical for such an application. However, for the vehicle application where packaging and aerodynamic constraints limit the area available for heat rejection to the environment, it is beneficial to raise the spray water temperature and a value of 72°C was selected for further analysis.

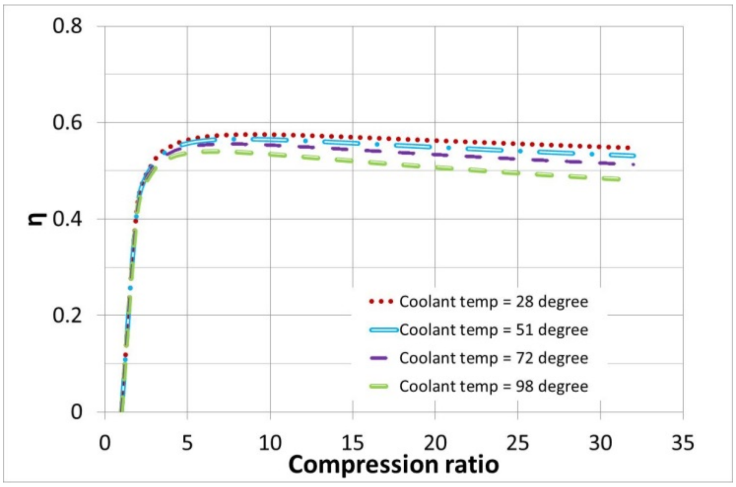

Figure 2. Effect of spray water temperature on cycle efficiency as a function of compression ratio

3 ANALYSIS OF COMPRESSION COOLING OPTIONS

The achievement of effective isothermal compression is essential to realising the potential of the split cycle engine. Isothermal compression could be achieved in a number of ways including:

1. Multi stage compression with intercooling
2. Direct cooling during the compression stroke through water injection
3. Direct cooling during the compression stroke through liquid nitrogen injection

3.1 Multi Stage Compression With Intercooling

To practically achieve the required combustor inlet pressure of 110bar using conventional turbo-machinery, at least three stages with a compression ratio of 3.3 would be required. More stages are desirable to reduce the overall compression work but the number of stages must be traded off against complexity of the intake system and therefore cost. A simple analysis was undertaken to investigate the impact of the number of compressor stages with (a) compression work and (b) heat rejection to the intercoolers. An ambient air temperature of 15°C was assumed and intercooler outlet temperature of 44°C. The compressors were assumed to be adiabatic, with an efficiency of 85%. The results of this analysis, compared to the ideal isothermal case are presented in figure 2. A 20% work saving was achievable using four compressor stages. The final gas temperature will also be of the order of 44°C and so significant heat will be recoverable from the vehicle exhaust. Although theoretically viable, this solution would require a complex chain of compressors and large intercoolers. The high pressure stages and intercoolers would also be heavy components to contain the high pressures, increasing cost and weight. This option was discounted from further analysis for these reasons.

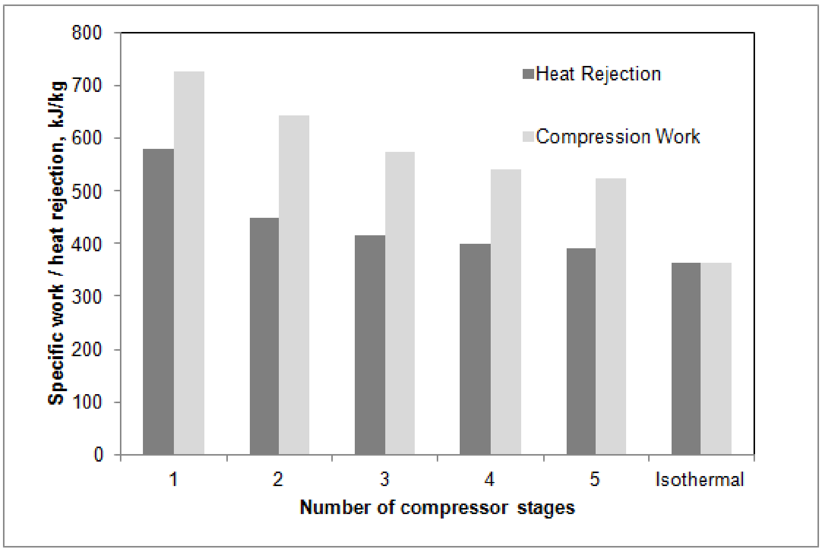

Figure 3: Comparison of work and heat rejection from multi stage compression to achieve 110 bar pressure

3.2 Direct Cooling with a Water Spray

Cooling of the charge air by the injection of a water spray directly into the cylinder during the compression stroke was proposed by Coney [5] and more recently by Iglesias [9] and Qin [10] . The former work was on an early split cycle engine concept, known as the isoengine whereas the later work was for a high efficiency compressed energy storage concept. To be effective the heat of compression must be rapidly absorbed by the spray water. The process can be modelled using the following relationship, relating the initial charge air temperature T_1 and the temperature, T_2 after a compression process of ratio CR:

$$\frac{T_2}{T_1} = CR^{(\gamma-1)exp\left[-\frac{ua}{C_p}\left(1-\frac{T_W}{T_1}\right)\Delta t\right]} \qquad (1)$$

Where u is the heat transfer coefficient between the water spray of temperature T_w and the charge air, a is the total surface area of the water droplets and Δt is the incremental time over which the compression process occurs. C_p is the specific heat capacity and γ is the ratio of the specific heats of the charge air. Inspection of equation (1) shows the process tends to the adiabatic case when the value Ua is small, and the isothermal case when Ua is high. In the isothermal case, the end of compression temperature, T_2 will converge to the injected water temperature T_w. To achieve effective isothermal compression it is therefore necessary to:

1. Ensure good mixing of the air and water through charge air motion and/or high water droplet velocity to maximise the heat transfer (u).
2. Maximise atomisation of the water spray to increase the effective area for heat transfer (a).
3. Maximise time for heat transfer (Δt) by limiting the maximum speed of the compressor.

The spray water must not change phase during the compression process as this would lead to a significant volume change as the water changes phase and rise in pressure and compression work. Assuming a 10°C rise in temperature during the compression process this would mean the maximum water temperature is 82°C under normal operating conditions. Referring to figure 4, as the compressor inlet pressure is 3bar, the maximum spray water temperature is still 50°C below the boiling point at the minimum cycle pressure. This provides sufficient margin to avoid boiling in a local hot spot due to uneven distribution of water in the cylinder or during transience where the water injection rate may drop below the target value.

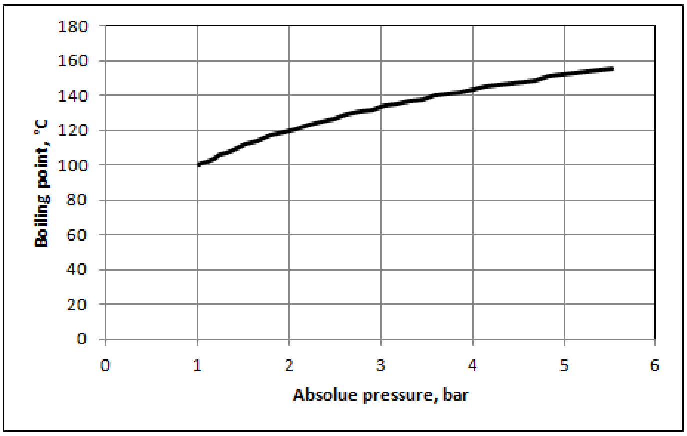

Figure 4: Relationship between boiling point and pressure for water

3.3 Direct Cooling through Liquid Nitrogen Injection

A novel concept was proposed by Jackson [6] using a cryogenic fluid, such as liquid nitrogen, as the cooling media. The cryogen is injected directly into the compressor cylinder, not only absorbing the heat of compression but also contributing to the compression through the change of phase of the cryogen from liquid to gas. The thermal energy required to achieve isothermal compression is therefore stored in 'a tank of cold'. The cryogen can be treated as a fuel, and manufactured in a stationary refrigeration unit or possibly on the vehicle, using energy regenerated during breaking. This promising concept is at an early stage and the subject of ongoing research. Based on the effectiveness and technical maturity the spray water cooling option was selected for further analysis.

4. HEAT BALANCE AND IMPACT ON VEHICLE COOLING

4.1 Comparison of Heat Balance

Based on the research in reference [7], the heat balance of a 350kW heavy split cycle engine with spray water compression cooling was calculated and summarised in table 3. For comparison, the heat balance for a heavy duty Euro 6 diesel engine using a mixed strategy of Exhaust Gas Recirculation (EGR) and Selective Catalytic Reduction (SCR) after-treatment for NOx control is also presented.

From table 3, the heat rejection requirements for the vehicle cooling pack can be calculated and are summarised in table 4. It was assumed the EGR heat was transferred to the vehicle cooling system and then rejected to the environment through the radiator in the diesel engine case. Also from table 3, it can be seen that 20.7% less thermal energy is rejected from the split cycle vehicle compared to the diesel baseline. A new cooler is required to reject the heat transferred to the spray water from the isothermal compressor.

Table 3: Comparison of heat balance in terms of kW and percentage of fuel energy for a Euro 6 heavy duty diesel engine and equivalent split cycle engine.

	Diesel Engine		Split Cycle Engine	
	kW	%	kW	%
Fuel Energy	875	100	701.4	100.0
Break Power	350	40	350	49.9
After-cooler	61	7	32.4	4.6
Spray Water Cooler	0	0	118	16.7
Water jacket (radiator)	149	17	65	9.3
EGR (addition to radiator)	61.3	7	0	0
Exhaust	315	36	137	19.5

The reduction in vehicle thermal load is unsurprising given the 19.8% reduction in fuel consumption and hence primary heat input into the combustion chamber. Waste heat is recovered internally to the cycle, unlike in external waste heat recovery concepts such as the Organic Rankine Cycle (ORC), where significantly more thermal energy needs to be rejected to the environment to recover work from the vehicle exhaust. This will be discussed in more detail later in the paper. The after-cooler heat rejection is significantly lower for the split cycle engine as less air is required at lower boost than for the equivalent diesel engine. As the after-cooler is often as big as the radiator in trucks, significant frontal area would be freed up for the spray coolers.

Table 4: Comparison of cooling requirements

	Diesel Engine kW	Split Cycle Engine kW	Change in Heat Rejection
Radiator	210	65	-68.9%
Spray water cooler	0	118	-
Aftercooler	61	32.4	-46.9%
TOTAL	271	215	-20.6%

To understand the impact of the changes to the vehicle heat balance the temperature at which heat is rejected must also be considered and will be discussed in the following sections.

4.2 Spray Water Cooler

The spray water cooler must cool the water recovered in the phase separator, this could be achieved by passing the water through a fin and tube type radiator. However, the spray water is recovered at high pressure which would lead to a significant increase in the weight of the radiator and loss of performance due to increase in tube thickness to contain the elevated pressure of the spray water. It is advantageous to maintain the water at the recovered pressure to facilitate the injection of water back into the isothermal compressor and reduce the risk of boiling. Reducing the pressure was therefore considered impractical. A supplementary cooling loop (figure 5) is proposed between the spray water cooler and radiator. The spray water cooler can therefore be a compact plate type heat exchanger. The secondary cooling loop then circulates coolant from between the spray water cooler and radiator at low pressure, comparable with existing engines.

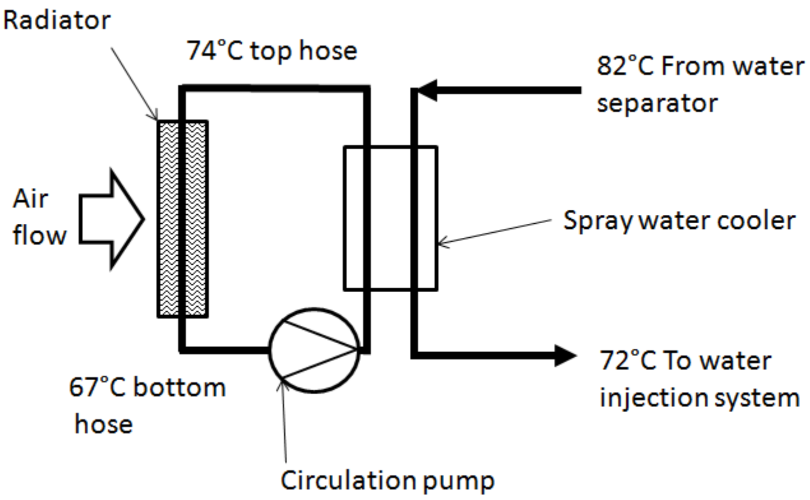

Figure 5: Spray water cooling circuit

The coolant in the spray water loop is water. A temperature rise of 10°C and an approach temperature of 5°C was assumed in the spray water cooler. The spray water radiator bottom hose temperature of 67°C is lower than the 90°C more typical of heavy duty engine radiator cooling systems. To estimate the impact on the vehicle front end cooling pack the Log Mean Temperature Difference (LMTD) method was used:

$$LMTD = \frac{\Delta T_A - \Delta T_B}{ln\left(\frac{\Delta T_A}{\Delta T_B}\right)}$$

Where ΔT_A is the temperature difference across the hot side and ΔT_B across the cold side. Assuming the thickness and design of the heat exchanger is unchanged, the change in frontal area can be calculated using:

$$A_2 = A_1 \frac{Q_2\, LMTD_1}{Q_1 LMTD_2}$$

Where *A* is the frontal area of the radiator, *Q* is the heat rejection and subscripts *1* and *2* refer to the two radiators in question. The resulting LMTD values for the main heat exchangers, compared with values calculated for the baseline diesel engine, are presented in table 5.

4.3 Cooling System Design

The LMTD and heat rejection data presented above was used to calculate the change in cooler sizes compared to a diesel engine for the split cycle engine. For this analysis, the air conditioning condenser and transmission oil coolers were not considered as these would be unchanged. The change in heat exchanger area are summarised in table 5.

Table 5. Comparison of relative cooler sizes

	LMTD	Change in required area relative to diesel case
Aftercooler	45.5	-47%
Radiator	47	-69%
Spray water cooler	22.9	+115% (relative to the diesel radiator)

From table 5, it can be seen that the spray water cooler is a substantial cooler, requiring 15% more frontal area than the Diesel vehicle radiator. However, the substantial reductions in the engine coolant radiator (-69%) and after-coolers (-47%) would provide sufficient space to package the spray water cooler. In most heavy duty vehicles, the after-cooler is positioned in front of the radiator. The reduction in after-cooler frontal area would provide a more direct air path to the spray water cooler of lower temperature air, further reducing the size of this component.

Conventional engine coolants would be used for the engine cooling circuit and the secondary spray water cooling loop. However, water would be used for the injected spray which could freeze in cold climates when the vehicle was not in use. Other fluids with a lower freezing point could be used but as the fluid sprayed into the compressor is mixed with the combustion air any unrecovered fluid would be exposed to the hot recuperator heat transfer surfaces. For these reasons, water is probably the best choice but some form of freezing protection will be required to protect the water injection circuit in cold climates.

4.4 Engine Thermal Design

Although not the main focused of the current work, some comments can be made on the thermal design of the engine. The thermal loading on the combustor cylinder is significantly higher than conventional engines;

1. High intake air temperature, increasing loading on the intake valves and loss of charge air cooling of the combustion chamber
2. Two stroke cycle, increasing power density and specific thermal loading
3. Increased combustion chamber wall temperatures to reduce chamber heat losses, increasing mean metal temperatures

Aspects of the thermal design of a large (megawatt scale) split cycle engine was reviewed by Coney [11]. In particular, the cooling of the top areas of the liner is of concern to preserve the integrity of the oil film. Coney concluded these issues could be addressed on a large engine format, which is encouraging considering the downsized engine for a heavy duty transportation application is likely to be easier to cool.

In the previous work by Coney the piston and cylinder head were manufactured from high temperatures alloys. This was necessary to contain the high mean combustion temperatures rather than mechanical loading, which will be lower for the split cycle engine due to the quasi-isobaric combustion process. As with the liner cooling, the smaller format engine will reduce the heat path between the combustion chamber wall and cooling channels and so more conventional materials may be feasible.

To investigate component thermal loading further a more accurate measure of the combustion heat release characteristics is required. This and further research on the impact of the slit cycle on engine cooling is the subject of ongoing research between Ricardo and the University of Brighton.

5. COMPARISON WITH OTHER HIGH EFFICIENCY CONCEPTS

The split cycle engine is a very novel concept with excellent potential, but other routes to achieving high efficiency in the heavy duty vehicle application have been proposed. Incremental improvements in friction, combustion, vehicle aerodynamics have been proposed, with waste heat recovery being the most significant single improvement. The most promising waste heat recovery solutions, and an assessment of the impact of the system on vehicle cooling is summarised in Table 6:

Table 6: Comparison of waste heat recovery technologies

	Efficiency improvement	Impact on vehicle cooling system
Turbo compounding (mechanical)	5%	Reduced overall heat rejection due to efficiency improvement
Turbo compounding (electrical)	10%	Reduced overall heat rejection due to efficiency improvement
Organic Rankine Cycle (ORC)	5%	Significant increase in heat rejection from ORC condenser
Thermoelectric Generation (TEG)	5%	Significant increase in heat rejection
Split cycle engines	25-36%	Change in heat balance but not overall package size

From table 6, it is apparent that the two thermal energy recovery methods (TEC and ORC) will impact the vehicle cooling system. The analysis presented in this paper suggests the impact of the split cycle engine on the overall cooling requirements is minimal, which is a significant benefit of the split cycle over other waste heat recovery methods.

7 CONCLUSIONS

A novel split cycle engine concept is described with separate compression and combustion chambers and intra cylinder heat recovery. Energy efficiencies of over 50% are possible for the concept, in excess of the combined improvements of conventional diesel cycles with exhaust heat recovery. Three concepts for compression cooling were presented and the spray water concept selected for further analysis. Multi stage turbocharging was discounted due to the high pressures and impact on cost and weight. Direct cooling using a cryogenic fluid is interesting and merits further research.

The heat balance of the engine was compared with a conventional diesel engine and a 20.6% reduction in overall thermal loading to the vehicle cooling system was predicted. A detailed review of the impact of the new cycle on heat exchanger components showed the vehicle cooling pack would be largely equivalent in overall dimensions to a conventional powered diesel vehicle of equivalent rating. The reduction in heat rejection to the radiator and after-cooler heat exchangers is sufficient to accommodate the new spray water cooler.

Further analysis of the component thermal loading is required and is the subject of ongoing research.

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