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Single Crankshaft Opposed Piston Heat Rejection Measurement and Simulation on High Power Density Engines for Future Ground Combat Vehicle Power Pack Configurations

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ABSTRACT

This paper discusses inherent advantages and additional design changes that can be made to a single crankshaft opposed piston engine (SCOPE) in order to satisfy military engine heat rejection-to-power requirements of 0.45.

The paper starts off with a discussion of the currently demonstrated heat rejection to power levels being obtained with the commercial version of the SCOPE configuration. Here, it is seen that heat rejection-to-power ratios are approximately 0.69. Tests are ongoing and this value is considered preliminary in nature.

Analytical results are then presented that decompose where the heat is being generated - for the intake air system, the coolant system, and also the oil lubrication system. The model includes consideration of heat generated from the engines turbochargers, cylinders, pistons, and gear train. The model is anchored to measurements made with a commercial version of the SCOPE engine. Engine heat rejection results for this baseline configuration (experimentally measured and/or predicted values) are made for all of the critical heat rejection areas of interest, including a decomposition of heat rejection areas within the gear train.

The heat rejection model is then extended to an engine configurations of military interest – those satisfying the 0.45 heat generation to power ratio. To facilitate various options available, identification of potential reductions in various heat generating areas of the engine are reviewed, including identification of technology limitations in each areas. In every particular heat generating regime, various design options, features and considerations are described, and the down-selected configuration is selected

Finally, a SCOPE engine configuration is presented that satisfies military engine heat rejection-to-power ratio requirement of 0.45, while at the same time utilizing engine design and manufacturing technologies that are of relatively low development risk.

The paper highlights that SCOPE engines offer lower heat rejection than conventional V-shaped reciprocating engines, in addition to their inherent packaging efficiency and power density. Additionally, design modifications that improve the heat rejection-to-power ratio of the engine tend to be more advantageous for the SCOPE engine than a V-shaped engine, thereby resulting in the ability to meet military engine heat rejection requirements without the need to pursue high risk technologies or design concepts.

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INTRODUCTION

The US Army Tank Automotive Research, Development, and Engineering Center (TARDEC) has required an engine heat rejection to power ratio of 0.45 for recent Request for Proposals (RFPs). This number is very demanding as conventional four stroke engines operate at a heat rejection to power ratio near unity.

Two stroke engines inherently offer advantages that are favorable towards reducing the heat generation to power ratio. This is the result of the elimination of a dedicated engine stroke for air intake and exhaust, such as exists on a 4 stroke engine. During this time frame heat continues to be transferred from the cylinder walls and pistons to the cooling medium, while no power is being produced.

A 2 stroke opposed piston engine design offers additional benefits with respect to the desire for a low heat generation to power ratio as the ratio of the surface area/volume is lower than other styles of 2 stroke engines (~ 30 %), owing this to the fact that no cylinder head exists in an opposed piston configuration.

Additionally, the traditional opposed piston engine can be configured as a Single Crankshaft Opposed Piston Engine (SCOPE), which offers even greater benefits in reducing the heat rejected to power ratio of the engine, as the displaced volume of this configuration is reduced significantly. A summary of the various configurations is shown in Table 1 below.

Table 1. Summary of displaced volumes for various engine configurations.

JP8 Tank-Engines

	Inputs								Results			
	i	g	z	P	P	vm	pe	S/d= k	V	n	S	D
				hp	kW	m/s	bar	-	Liter	rpm	mm	mm
4-Stroke 1P/C	2	2	10	1500	1119	12.71	24.3	1.147	22.5	2450	156	136
4-Stroke OP	2	1	6	1500	1119	14.0	18.0	2.2	26.8	2782	302	137
2-Stroke 1P/C	1	2	6	1500	1119	13.0	10.0	1.2	38.8	1730	225	191
2-Stroke OP	1	1	6	1500	1119	12.8	10.0	2.2	26.2	2564	300	136
2-Stroke SCOPE	1	1	6	1500	1119	11.4	12.4	1.385	14.3	3800	180	130

Results presented in Table 1 are based on most recently published data (as identified by the authors) and then normalized for consistent horsepower. Focusing on the displaced volume, it is seen that the SCOPE engine requires approximately one-half the displacement of the other engine configurations. The reduced volume translates into reduced surface area for heat transfer.

Recently, the SCOPE configuration has undergone considerable commercial investment and is planned for entering production in 2017. Figures 1 through 3 represent the commercial version of the SCOPE engine.

Figure 1. Two-cylinder commercial version of the SCOPE configuration.

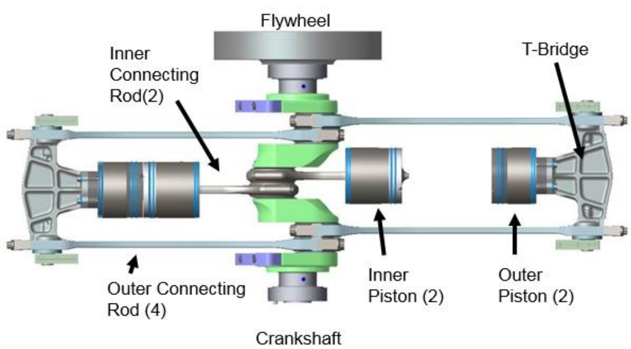

Figure 2. Exploded view of the SCOPE engine (powertrain components).

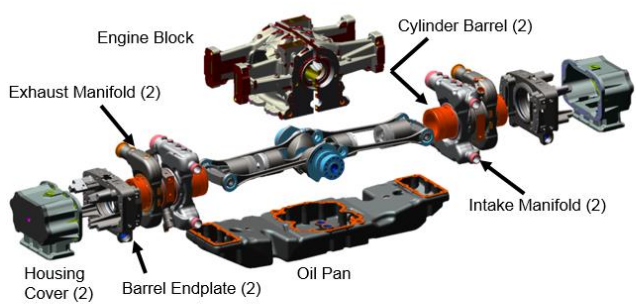

Figure 3. Exploded view of the commercial version of the SCOPE configuration (outer core engine parts).

Additional components that are important considerations to the heat generated by the engine assembly are the combustion air compression components. In the case of the SCOPE engine, a combination of a supercharger and a turbocharger are used for this purpose. As will be discussed later, the supercharger can be eliminated from the militarized version.

The SCOPE configuration offers additional advantages with regards to reduced heat rejection (in addition to reduced surface area/volume as identified in Table 1) as the system requires no cylinder head with valve train, fewer bearings and gears for operation. The reduced parts count of the SCOPE configuration contributes both to a reduction in heat generated and an improvement in power generation.

In this investigation, baseline performance of the commercial variant of the SCOPE engine is presented.

Next, relatively low risk means of reducing the heat generated from the engine are presented, which includes:

- Optimizing the coolant channel design of the barrel and replacing ethylene glycol/water coolant with engine oil.
- Adding a ceramic insert to the piston crown
- Eliminating the operation of the supercharging system during full power operation.

Details of the test data obtained with the commercial SCOPE engine and prediction methodology (which includes both one dimensional and Computational Fluid Dynamic predictions) are presented. Additionally, a near term schedule of planned demonstrations and milestones is also shown.

Experimental Testing of the SCOPE Engine

A fully instrumented ‘production-intent’ SCOPE engine was tested for determination of the heat rejected to power ratio of the system. The instrumented engine is shown in Figure 4 below.

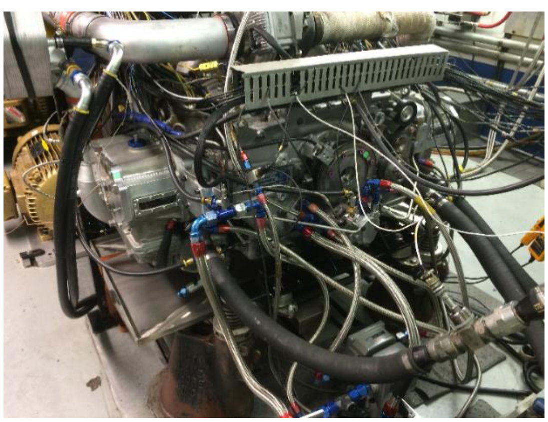

Figure 4. Fully instrumented SCOPE engine used for determination of the heat rejection to power ratio.

It should be noted that the supercharger is not an integral part of the SCOPE engine. Appropriate accounting of this aspect is accounted for and discussed in detail.

Measured heat generating parameters are identified in Table 2 below.

Table 2. Measured heat generating components in the SCOPE engine

Fuel System	
	Inlet Pressure
	Inlet Temperature
	Flow Rate
Combustion Air Intake System	
	Intake Air Pressure
	Intake Air Temperature
	Air Flow
	Supercharger Cooler -Combustion Air Temperature and Pressure (Upstream and Downstream)
	Compressor Aftercooler - Combustion Air Temperature and Pressure (Upstream and Downstream of cooler)
	Exhaust Gas Before Turbocharger - Pressure and Temperature
	Exhaust Gas After Turbocharger - Pressure and Temperature
Oil Lubrication System	
	Oil flow rate
	Oil temperature and pressure entering and exiting oil cooler
Cooling System	
	Coolant flow rate to cylinder #1
	Coolant flow rate to cylinder #2
	Coolant temperature and pressure entering coolers (cylinders #1 and #2 separately cooled)

Additional heat generating parameters were measured (e.g., controller heat generated, blow-by exhaust gas flow energy, etc.) but were found to be negligible for the intent of this investigation.

Measured power production and usage is presented in Table 3 below.

Table 3. Measured power produced or consumed by engine and various components

Engine Output Power	Measured with a water brake dynamometer
Supercharger Power Consumption	Measured with a torque meter.

The fuel injection pump, water pump, and oil pump, were all directly driven by the core engine. Therefore, no consideration of the power requirements of these components is necessary for the purpose of this investigation.

Power requirements for operation of the externally powered engine controller and the fuel supply pump were also examined but found to be negligible for the purposes of this investigation.

Test Results

Testing was performed at the peak torque point of the SCOPE engine, which corresponds to approximately 118 kW of net power at 2200 rpm. It should be noted that the engine was designed for 234 kW of peak power at 3000 rpm. Several test runs were made and the results presented below (in Tables 3 and 4) represent the mean values obtained from these tests. ***As the configurations and test procedures changed during the course of these tests (and continues to do so as the final production configuration is approached), the information published below is preliminary, with finalized results expected in the 3rd quarter of 2016.***

Table 3. Heat generated by the commercial SCOPE engine.

Subsystem		Heat Removed from System (kW)*
Combustion Air	Supercharger Cooling	5.8
	Compressor Cooling	24.2
Oil Lubrication System		24.5
Cooling System		27.2
Total		**81.7**
*Preliminary Results		

Engine power generation and consumption is presented in Table 4 below. Similar to the heat rejection values presented, the power generation numbers are considered preliminary in nature.

Table 4. Power generation and consumers measurement values for the commercial SCOPE engine.

Engine Flywheel Power (kW)*	126.8
Supercharger Power (kW)*	8.7
Net Power (kW)	**118.1**
***Preliminary results**	

Based on experimental measures, the commercial SCOPE engine has a heat generation to power ratio of 0.69. It should be noted that this value is substantially lower than most 4 stroke engines, which, in general, operate at heat rejection to power ratios near unity.

SCOPE Engine Improvements for Military Applications

Measured values of the heat rejection to power ratio of the SCOPE engine of 0.8 show promise as a candidate for use in military environments. As the commercial engine prepares for its final stages of introduction to production, additional favorable (but modest) improvements are expected as the fuel injection and control system are optimized.

For military applications, further enhancements to the SCOPE configuration can be realized by introducing the following changes:

- Increasing combustion air temperatures from the aftercooler. The commercial SCOPE engine is designed for an 80 C exit temperature from the aftercoolers. Increasing the exit temperature to be in the vicinity of 130 C will significantly reduce heat rejection levels.
- Optimizing the operating temperature of the cylinder barrel. Currently, the commercial version of the SCOPE engine is cooled with an ethylene glycol/water mixture in a barrel chamber that was optimized for minimum development effort (low priority) and minimum cost.
- Adding a ceramic insert to the pistons. Ceramic inserts offer substantial durability benefits over ceramic coatings and are the preferred configuration.
- Eliminating operation of the supercharger at full power operation. Test data and analysis have shown that with a properly matched turbocharger system, the militarized SCOPE engine (utilizing oil cooler liners and ceramic inserts in pistons) does not require supercharging at full load points.

An option is to replace the superchargers with electronically controlled turbochargers as boosting is necessary during engine start, acceleration, and part load. This option is outside of the intention of this investigation.

The latter 3 improvements to an engines heat transfer to power ratio are considered in greater depth in this investigation. The aspect of increasing aftercooler temperatures is another potential improvement in heat to power ratio of an engine but was not considered in this analysis.

Optimization of the Operating Temperature of the Cylinder Barrel

The cylinder barrel of the commercial SCOPE engine accounts for almost 40 % of all heat transferred from the engine. The distribution of barrel surface temperature (predicted) is shown in Figure 5 below.

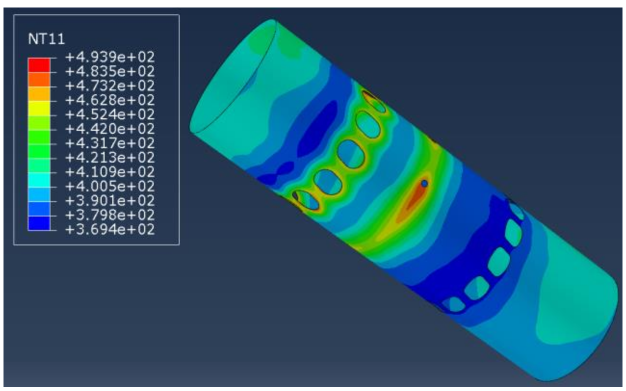

Figure 5. Predicted temperature distribution of barrel surface temperature for the commercial SCOPE engine.

A review of the temperature contours of this engine reveal the following:

- Barrel surface temperatures are substantially below the maximum operating temperature of the metal (steel) in nearly all locations
- 'Hot spots' are localized and limited to regions near the top dead center of the engine and the exhaust port region of the engine.

In consideration of the above situation, substantial reductions in heat transfer, can be realized by simply replacing the ethylene glycol/water cooling system with oil cooling and optimizing the geometry of the coolant channel to minimize overcooling. This is the path taken with the militarized SCOPE engine. Results are shown in Figure 6 and also summarized in Table 5 below.

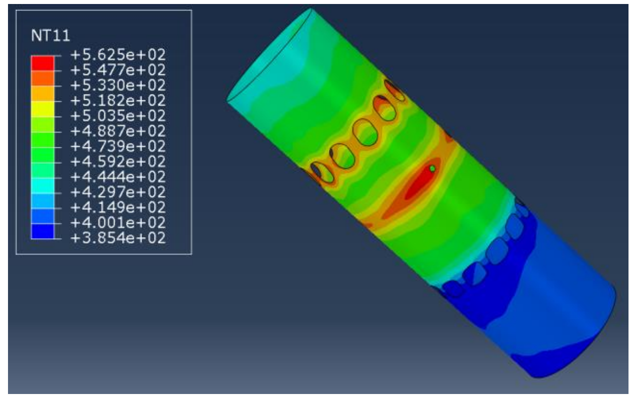

Figure 6. Predicted temperature distribution of barrel surface temperature for the military SCOPE engine.

Table 5. Heat transfer comparisons from the barrel of the engine

	SCOPE with water cooling	SCOPE with engine oil cooling and optimized coolant channel geometry
Exhaust Port Region	8.6 kW	6.1 kW
Top Dead Center Region	9.4 kW	5.9 kW
Outer Piston Travel Region	6.0 kW	4.5 kW
Inner Piston Travel Region	3.2 kW	2.0 kW
Total Heat Transfer to the Barrel	27.2 kW	18.5 kW

A summary of the improvements expected to be realized by optimizing the coolant channel design of the barrel and changing to an oil cooled configuration is provided in Table 6 below. It should also be noted that elevated wall temperatures are expected to result in modest improvements in power generation and bsfc but have not been considered in this preliminary analysis, leaving these expected benefits as additional margin to be validated later.

Table 6. Heat to Power ratio comparison of an ethylene glycol/water cooled engine to an oil cooled engine

	Ethylene Glycol/Water Cooled SCOPE Engine	SCOPE Engine with a coated piston crown
Combustion Air Cooling	30 kW	30 kW
Oil Lubrication System Cooling	24.5 kW	24.5 kW
Barrel Cooling	27.2 kW	18.5 kW
Total Heat Generated	95 kW	73.0 kW
Net Power Generated	118.1 kW	118.1 kW
Heat Generation to Power Ratio	0.69	0.62

Consideration of Change to a High Temperature Metal Piston and Consideration of Addition of a Ceramic Insert to the Piston Crown

The oil lubrication system cools the engine pistons, bearings and gears, in addition to work input by the engine oil pump. By far, the most significant amount of heat transfer is the result of the pistons as outlined in Table 7 below.

Table 7. Oil lubrication system heat transfer predictions for the commercial SCOPE engine.

	Heat Transfer
Piston Heat Transfer	21.2 kW
All other heat transfer (Crankshaft Bearing Heat Transfer, Turbochargers, Oil Pump, etc.)	3.3 kW
Total Heat Transfer to the Oil Lubrication System	24.5 kW

2 Modifications to the commercial SCOPE engine were considered: 1.) Changing the baseline high temperature steel piston to high temperature nickel based alloy (Haynes 282) and 2.) Adding a ceramic insert to the crown of the piston. The 'path forward' configuration chosen was the ceramic insert, as this system was found to reduce piston heat transfer approximately 3 X more than the Haynes 282 alloy piston, as will be shown later.

Thermal barriers on the piston crown offer significant improvements to a reciprocating diesel engine. These benefits include reduced heat load to the coolant system, reduced smoke and emissions, and improved fuel consumption. Due to the higher heat loads experienced in a 2 stroke engine, the potential benefits of usage increase with this configuration. And, as will be shown later, the application of a ceramic crown increases the exhaust gas temperatures and allows elimination of the need for the supercharger at rated power conditions.

In order to properly estimate the impact of operation with a ceramic insert, 3-dimensional oil flow models (including modeling of oil shaking) and 3 dimensional heat transfer models on the combustion side (CONVERGE) of the piston were performed, along with a 3 dimensional piston conduction analysis. Sample results are shown in Figures 7-10

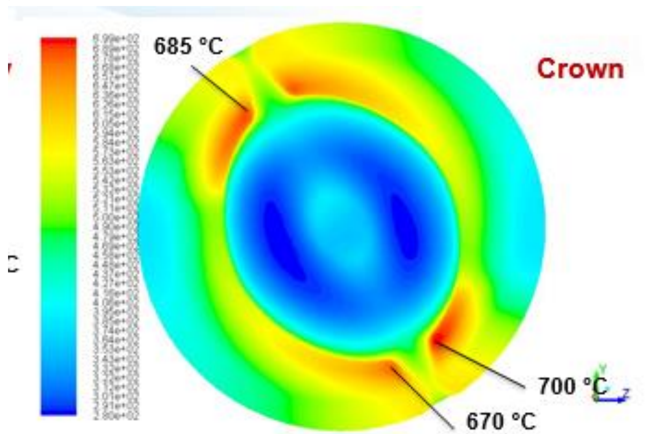

Figure 7. Predicted piston crown temperature on the hot side of the inner piston (commercial SCOPE piston).

Figure 8. Predicted back side piston temperatures of the inner piston (commercial SCOPE piston)

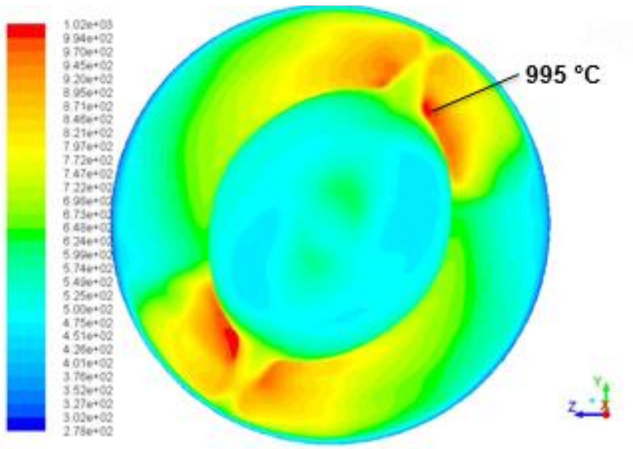

Figure 9. Predicted piston crown temperature on the hot side of the inner piston with a ceramic insert added (militarized SCOPE piston).

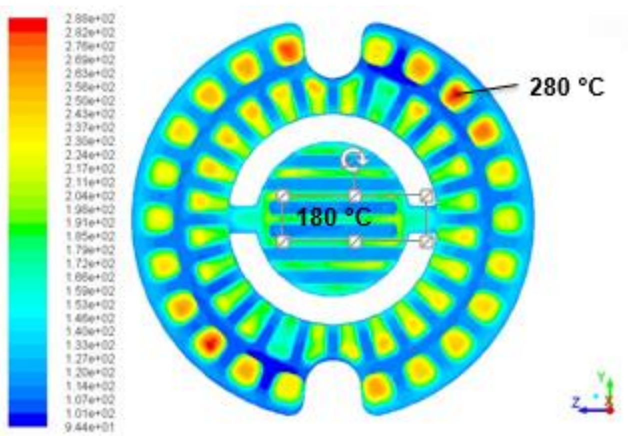

Figure 10. Predicted back side piston temperatures of the inner piston with a ceramic insert added (militarized SCOPE piston).

Comparison of the above predictions shows the significant advantages of adding a ceramic insert to the piston – hot side wall temperatures increase (with a corresponding decrease in combustion gas heat transfer), while coolant temperature rise and back side wall temperatures decrease. Results of the reduction in heat transfer and its overall impact to the heat-to-power ratio of the militarized SCOPE engine are shown in Table 8 below.

Table 8. Reduction in the heat to power ratio of the SCOPE engine with the application of a ceramic insert to the piston crown.

	SCOPE Engine with oil cooled liner and conventional pistons	SCOPE Engine with oil cooled liner and pistons with ceramic inserts
Air Induction System Cooling	30 kW	30 kW
Oil Lubrication System Cooling	24.5 kW	8.8 kW
Ethylene Glycol/Water Cooling System	18.5 kW	18.5 kW
Total Heat Generated	73.0 kW	57.3 kW
Net Power Generated	118.1 kW	118.1 kW
Heat Generation to Power Ratio	0.62	0.49

In the above predictions, it is assumed that the addition of a ceramic insert to the piston crown will have no impact on after-cooling, power generation, and liner heat transfer. In reality, there will be a modest, and overall beneficiary, impact to all of these systems. These secondary impacts were not considered in this analysis.

Elimination of the Supercharger at Full Power Operation

As a result of the reduction in heat transfer of the combustion gasses to the cylinder walls and the pistons, the energy of the exhaust gasses increases significantly. As such, operation of the superchargers at full power is unnecessary, which will result in a significant additional net crankshaft power. Other details, such as the impact of the elimination of the intercooler (which cooled the supercharged air) will have a modest impact on the performance of the engine, but were not considered in this investigation. Various options exist to address this aspect. For this investigation, it was assumed that the temperatures exiting the aftercooler have sufficient margin and that there was also sufficient margin in the design air-to-fuel ratio of the engine to accommodate this product simplification. Initial analysis support these assumptions.

The predicted results of eliminating the supercharger are shown in Table 9 below.

Table 9. Predicted impact in the heat-to-power ratio of the SCOPE configuration when the superchargers are eliminated

	SCOPE Engine with oil cooled liner and pistons with ceramic inserts	Fully Militarized SCOPE Engine (oil cooled liner, pistons with ceramic inserts, supercharger eliminated)
Air Induction System Cooling	30 kW	30 kW
Oil Lubrication System Cooling	8.8 kW	8.8 kW
Ethylene Glycol/Water Cooling System	18.5 kW	18.5 kW
Total Heat Generated	**57.3 kW**	**57.3 kW**
Supercharger Power	8.7 kW	0 kW
Gross Engine Power	126.8 kW	126.8 kW
Net Power Generated	**118.1 kW**	**126.8 kW**
Heat Generation to Power Ratio	**0.49**	**0.45**

Summary and Recommendations

A commercial 'production intent' SCOPE diesel engine configuration is currently in the final phases of production readiness and is projected to have a heat to power ratio of 0.69. The data and projections presented in this paper are preliminary, with final results expected to be obtained in the 3rd quarter of 2016. Modest 'militarization' improvements to the engine can be made that are predicted to lower the heat rejection to power ratio from 0.69 to 0.45. The improvements are realized by 3 reasonably simple actions:

1.) Replacing water cooling of the barrel with a newly designed engine oil cooling system.
2.) Addition of a ceramic insert to the crown of the piston.
3.) Eliminating the operation of the supercharger when at maximum power conditions.

Further work should focus on the following:

- Continued development of the SCOPE configuration for commercial applications (gensets, automotive, off-highway)
- Reconfiguring a SCOPE configuration to satisfy US military land vehicle requirements (packaging size, etc.)
- Experimental validation of the improvements in heat rejection to power ratio predicted by this investigation.