Development of a Compact and Efficient Truck APU

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Abstract

Trucks parked at idle create excessive fuel consumption and exhaust emissions. One of the solutions to this issue is to institute a separate power system, the APU. It will be in operation when vehicle is parked and engine is off to provide the electricity and support the necessary needs (refrigerator, air conditioning, TV, etc.). This paper offers the insight of development of an Electrical Power Cell (EPC) as the candidate of the APU. The device consists of a unique Opposed Piston and Opposed Cylinder (opoc) engine and a high speed rotary generator. An extensive investigation has been conducted during the engine development program to create a special unniflow scavenging process of this two stroke engine. As the result, it realizes zero fresh charge loss with high scavenging efficiency. This paper provides the technical challenges and solutions in achieving the design targets.

1. Introduction

Hundreds of Thousands of 400-500 hp engines produce less than 10 hp power for 8 – 10 hours every day when the truck is parked in the United States. The power is necessary to provide electricity to operate the utility devices in the driver compartment, such as air conditioner, refrigerator, TV, and so on. However, under this condition the engine has to produce many times greater power than its output to overcome the unnecessary friction and parasitic losses. Therefore, the fuel consumption rate is many times higher than its normal operation conditions. In the mean time, the engine works at low speed and cold combustion conditions, which results in the highest HC, CO and PM emissions. US EPA calls for immediate attention and actions to stop this "Truck Idle" energy loss and air pollution [1].

Accessory Power Units (APU) is rated as the number one solution to this issue. With an APU on board, the required electrical power can be provided without running the big engine when the truck is parked. Since the small engine that is integrated in the APU is optimized for the specific application, it has the opportunity to reduce dramatically the fuel consumption and emissions, compared to the "truck idling" situation. Unfortunately, the current available APUs are judged to be too big, too heavy and too expensive [1].

In the case of the military vehicles ever increasing use of electronics for navigation, detection and advanced countermeasures dictates the need for a dedicated compact, quiet and efficient APU while still the needing to reduce fuel consumption and improve performance.

There are on-going efforts to electrify the engine cooling fan, water pump; oil pump, power steering pump and air conditioning compressor drive wherever air cooling is required in tactical vehicles. These and other combat electronics' power requirement is estimated to be from 5 to 30 KW.

A high power density generator set weighing no more than 37 lbs can serve as an APU on the vehicle and be easily removed and carried by one person. In the off the vehicle mode it can run off a standard military fuel container or any other fuel source depending on the situation, providing the flexibility and utility power necessary for combat operations.

The following pictures and tables provide the information of military APUs that are currently available. These APUs are too big and too heavy for the future combat mobility requirements.
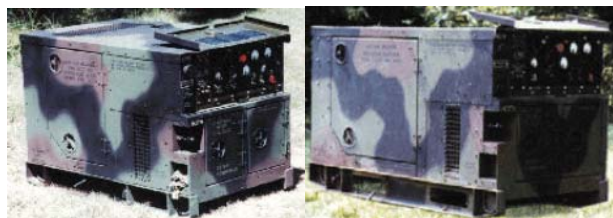

Fig 1: 5 and 10 KW Tactical Quiet Generator (TQG)

Table 1. 5 KW TQG Specifications

Nomenclature	60 Hz TQG	400 Hz TQG
MEP Model Number	802A	812A
NSN	6115-01-274-7387	6115-01-274-7391
LIN	G11966	G12102
ZLIN	Z31532	Z47570
SSN	M535	M518
Wet Weight	888 lb.	911 lb.
Length	50.4 in.	50.4 in.
Width	31.8 in.	31.8 in.
Height	36.2 in.	36.2 in.
Cubic Feet	34	34
Noise at 7 meters	70 dBA	70 dBA
Voltage Connection	120 V, single phase, 2 wire; 120/240V, single phase, 3 wire; 120/208V, 3 phase, 4 wire	120 V, single phase, 2 wire; 120/240V, single phase, 3 wire; 120/208V, 3 phase, 4 wire

Table 2. 10 KW TQG Specifications

Nomenclature	60 Hz TQG	400 Hz TQG
MEP Model Number	803A	813A
NSN	6115-01-275-5061	6115-01-274-7392
LIN	G74711	G74779
ZLIN	Z47289	Z47366
SSN	M52900	M56500
Wet Weight	1182 lb.	1220 lb.
Length	61.7 in.	61.7 in.
Width	31.8 in.	31.8 in.
Height	36.2 in.	36.2 in.
Cubic Feet	41	41
Noise at 7 meters	70 dBA	70 dBA
Voltage Connection	120 V, single phase, 2 wire; 120/240V, single phase, 3 wire; 120/208V, 3 phase, 4 wire	120 V, single phase, 2 wire; 120/240V, single phase, 3 wire; 120/208V, 3 phase, 4 wire

Nomenclature	APU, 5kW, DED, 28VDC	No AC APU
MEP Model Number	952B	N/A
NSN	6115-01-452-6513	N/A
LIN		N/A
ZLIN		N/A
SSN		N/A
Wet Weight	550 lb.(includes housing)	N/A
Length	34.5 in.	N/A
Width	30.5 in.	N/A
Height	17 in.	N/A
Cubic Feet	11.4	N/A
Noise at 7 meters	70dBA	N/A
Voltage Connection	28VDC, NATO slave receptacle	N/A

Fig 2: 5 KW APU on a Personnel carrier

opoc engine driving a high speed generator presented in this paper offers an optimal solution to these demands. The unit is named Electric Power Cell (EPC). It meets all the fuel consumption and emissions requirements with a fraction of package size and weight compared to the currently available APU. The working principle of the opoc engine and some insight of the development process of it are provided in this paper.

The program is conducted at APT with its existing patented technology. FEV provides extensive engineering support to the engine development subjects. The U.S. Army Tank Automotive Research and Development Engineering Command (TARDEC) – National Automotive Center (NAC) funds the program and is deeply involved the development process.

2. Description of the opoc Engine

opoc engine is short for Opposed Piston Opposed Cylinder engine. It is a two-stroke Compression Ignition engine, consisting of opposed cylinders, each having two opposed pistons. All pistons are connected to a single crankshaft located between the two opposed cylinders by unconventional connecting rods. Intake and exhaust ports are located at each end of the piston travel (BDC) positions. These ports provide the means
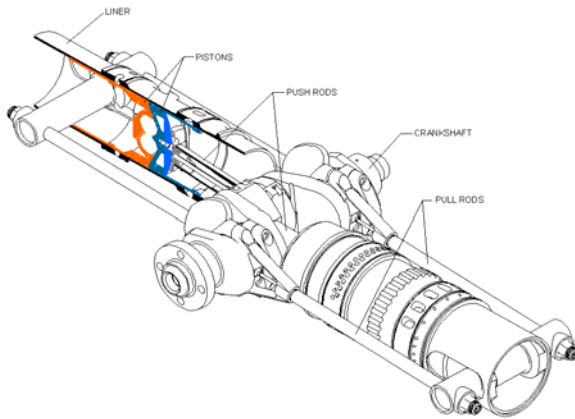
Fig 3: opoc Engine Configuration

for cylinder scavenging without valve mechanism. Fig 3 shows the principle of the design.

One of the most important feature of this two-stroke engine is its asymmetric uniflow scavenging process. The crankshaft journals, which control the position of the pistons, are arranged asymmetrically such that, for each cylinder, the exhaust ports open and close before the intake ports. With an innovative intake air flow arrangement, the scavenging efficiency achieves 85% range with 100% trap efficiency. The high scavenging efficiency offers the high power output per piston stroke. Therefore, it allows the opoc engine approach the close to the ideal two stroke engine power density: two times greater than the four stroke engine. The perfect air trap efficiency results in zero fresh air leaving to the exhaust and thus eliminating the HC emissions as conventional two stroke engines do. An integrated piston driving scavenging pump is designed to provide sufficient intake boosting to provide positive scavenging at engine starting and high intake pressure for further improved high power density.

The opposed piston structure of the opoc engine allows for doubling the speed of the crankshaft with the same mean piston speed, and thus doubling the engine power for the same cylinder displacement. The combination of these two factors enables the opoc engine to be superior in power density and specific power. The conrod of the opoc engine is unique. The pair of the inner and outer rods are constructed in such a way that torque is created directly on the crank from the piston forces. Little resultant main bearing forces exist as the conventional engine does. This feature allows for the engine designer to use lightweight design and materials for the crank case to further improve the specific power. In addition, the engine noise induced

by the crank mechanism is significantly reduced. Considering the elimination of the cylinder head and valve train driving mechanism and the noisy cover, the engine becomes a quiet machine, which is extremely important for the truck APU application.

The currently designed opoc engine for truck APU application is on Diesel fuel. A low cost and low pressure fuel system is integrated to realize "Assisted HCCI" combustion process. Fuel is meted by the fuel system and evaporated by the hot and high speed air. This pre-mixed fuel/air mixture is then introduced into the cylinder through intake ports. The compression ratio is arranged to such a level that pre-chemical reactions take place without significant combustion (ignition) until the timed spark provides the trigger. With the speed of 10,000 rpm, the close-to-HCCI process will be realized with a reliable combustion phasing control due to the longer ignition delay in terms of crank angle.

opoc engine design offers the unique machine to reduce mechanical friction. With the much reduced friction surfaces, such as valve train bearings and number of main bearings, the friction and wear are significantly reduced. The piston side force is also greatly reduced due to the long effective conrod length. The outer rod is geometrically longer than normal rods by 2-3 times. The inner piston connecting rod on the opoc engine is subject only to compression loading, which eliminates a need for a wrist pin. This is replaced by a concave radius of large diameter on which a sliding crosshead slipper impinges, and on which the pull-rod slides. With this configuration the effective rod length (L_{eff}) is increased by over 67%.

Fig 4 shows the package layout of the EPC that is designed for 5 kW capacity with full emission control and up to 10 kW with less emissions regulation. It consisted with an opoc engine with its muffler, a high speed generator, power electronics and a battery.
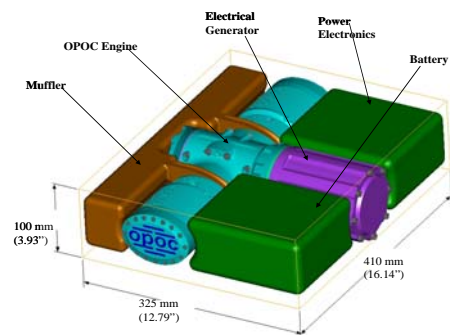
The engine and generator have been designed and built for the 5/10 kW EPC application. Fig 5 shows a picture of the hardware of the engine and the comparison of it with a conventional 4 stroke engine having similar displacement. Thanks for the unique architecture of the opoc engine design, the engine is simple in structure and has much less part counts compared to the conventional engines.

opoc engine, one model and two cylinders

Conventional four stroke engine with two cylinders

Fig 5: opoc Engine Parts Counts and Compared with That of a Conventional Engine

3. Development of the opoc Engine

Engine performance prediction, design optimization, structure evaluation and the initiation of the testing investigations are described in this section to present the critical process of the opoc engine development.

3.1 Engine Design Analysis and Optimization

3.1.1 Engine Performance Prediction and Optimization

For two stroke engines, gas exchange (scavenging) process is critical to both the engine performance and emissions. The combination of 1-dimensional gas exchange and 3-dimensional CFD analysis are conducted to optimize the scavenging process to achieve the challenging targets.

GT-Power software is used to simulate the engine process, focusing on the gas exchanging. The simulation model incorporates all the actual geometry of the gas exchange system, and also the engine operation conditions. Engine performance information, such as torque, power, volumetric efficiency, peak cylinder pressure etc., as well as gas dynamics parameters, e.g. dynamic pressure waves, gas temperature and mass flow rate at any location of the gas exchange system, are available output data of the simulation.
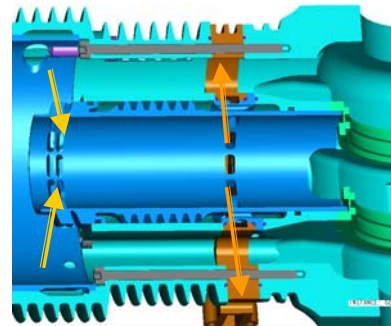

Fig 6: Location of intake/exhaust ports.

One of the unique features of the OPOC engine is its ability to utilize uniflow scavenging with asymmetric port timing. To fully realize this advantage, the simulations were conducted to optimize the port heights and timing. The simulation matrix of port heights is given in Table 3Table 3.

Table 3: Simulation matrix of port heights

Ex. Port	Intake Port Heights		
	3 mm	4 mm	5 mm
3 mm	X	X	X

4 mm	X	X	X
5 mm	X	X	X

The simulation results for different port heights/timing combinations are shown in Fig 7, which indicates that the optimum exhaust port height is 4 mm, while the optimum intake port height is between 4 – 5 mm. This optimized port height is the compromising of the delivery ratio, scavenging efficiency, trapping efficiency and power expansion ratio. The efficiency ratio was modeled in this 1-D simulation per experience, which is subject to be confirmed with the 3-D CFD simulations.
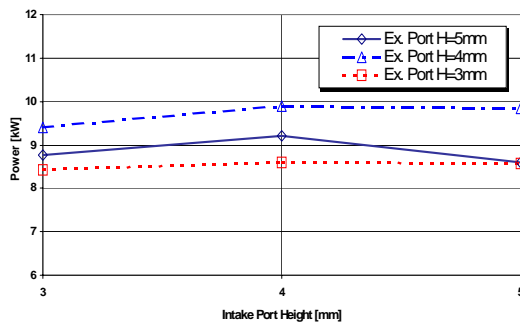
Fig 7: Optimization of intake and exhaust port height at 10,000 rpm

3.1.2.   Exhaust Port Back Pressure Study

Challenging targets of the engine noise control require high pressure drop mufflers/silencers design. This high exhaust back pressure may influence the engine performance. Therefore, a separate study was conducted to evaluate this influence.

It is concluded from this study that the predicted engine performance is little affected by the exhaust backpressure level up to 2.0 bar (absolute). It is discovered that, during most of the scavenging period, the pressure ratio of exhaust port over the cylinder is less than 0.528, i.e. less than the critical pressure ratio for sonic flow, therefore the exhaust gas flow through the exhaust port is supersonic flow. In the sup-sonic flow region the exhaust gas mass flow rate only depends on the upstream conditions (in this case the cylinder conditions) and the downstream condition (exhaust port) has no influence. This engine characteristic offers the opportunity of achieving high exhaust noise reduction without effecting engine performance.

3.1.3.   3-Dimensional CFD Study

Since the scavenging process is of 3-dimensional nature and is beyond the modeling capability of the 1-D GT-Power code, 3-D StarCD simulations of the scavenging process were conducted. The obtained scavenging efficiency was fed back to the 1-D simulation to update the engine performance simulation. Several interactive loops between the 1-D and 3-D simulation were conducted and the converged results are discussed below.

Fig 8 displays the simulation results of a scavenging process. The gas composition of the supercharging chamber, in-cylinder and exhaust manifold at different timing are displayed to show the progress of the scavenging process. In the picture, "Red" represents the burnt gas from the previous cycle, "Blue" depicts the fresh charge, and any color in between represents mixing of the burnt gas and fresh charge, as indicated by the color band scales.

Exhaust ports open before intake ports, taking the advantage of the blow down process for an effective exhaust scavenging process. By the time of intake ports open, the positive pressure ratio between intake and cylinder allows for rapid intake flow process. The shape of the intake ports and piston top design force the fresh air goes into the cylinder center and leave the hot exhaust gases stay along the cylinder wall. This gas distribution profile facilitates the residual gases exit the cylinder through exhaust ports located along the cylinder wall before fresh air "leak" through. A high scavenging efficiency is achieved with minimum fresh air short circuits. Fig 8 demonstrates this scavenging process.
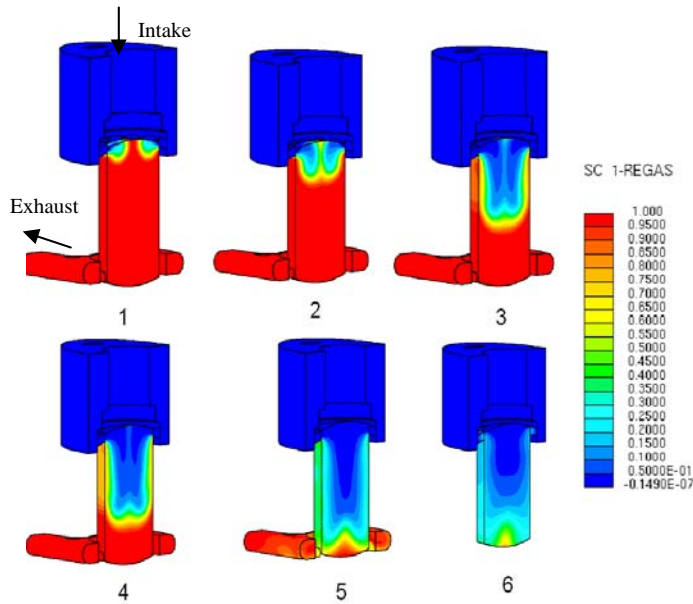
Fig 8: CFD Simulation Results: Gas Composition Distribution Progress of the Scavenging Process

Extensive 1-D GT-Power and 3-D Star-CD interactive simulations were conducted to realize the converging solutions and approach the optimum port heights design.

Fig 9 summarizes the optimization results of the gas exchange process. It is very clear that with the 4 mm exhaust port height, due to larger overlap between the intake and exhaust port timing, the scavenging time duration is longer and more fresh air mass will be delivered, which results in higher scavenging efficiency. However, more fresh charge will be lost into the exhaust port.

On the other hand, reducing the exhaust port height from 4 mm to 3 mm reduces the exhaust/intake port overlap and scavenging duration. Fresh charge delivery is reduced and so is the scavenging efficiency, but the amount of fresh charge lost to the exhaust port is almost eliminated. With the combination of these two effects, fresh charge trapped in cylinder is actually increased, although the delivered fresh charge is slightly reduced.

Reducing the intake port from 4 mm to 3 mm also shows a significant difference on the scavenging process. A shorter intake port would have an increased cylinder volume to trap, and a larger effective compression ratio which in return results in higher boost pressure. As the overall results show, reducing the intake port height from 4 mm to 3 mm increases both, the compression ratio and the trapped cylinder volume, therefore increased the trapped fresh charge.

Together with the 3 mm exhaust port height, the fresh charge loss to the exhaust port is also eliminated.

Based on these analyses, both exhaust and intake port heights are determined to be 3 mm for the current engine design.

3.1.4. Predicted Engine Performance

Based on the simulation results a set of optimized engine design parameters is specified. The corresponding engine performance is then predicted. Three (3) typical engine speeds: i.e. 8,000 rpm, 10,000 rpm, and 12,000 rpm, were selected for the performance prediction as shown in Fig 10.
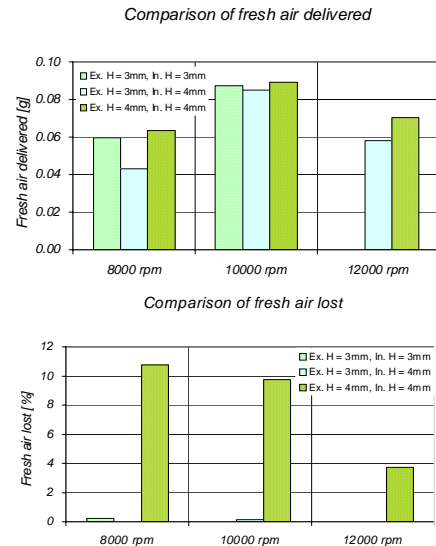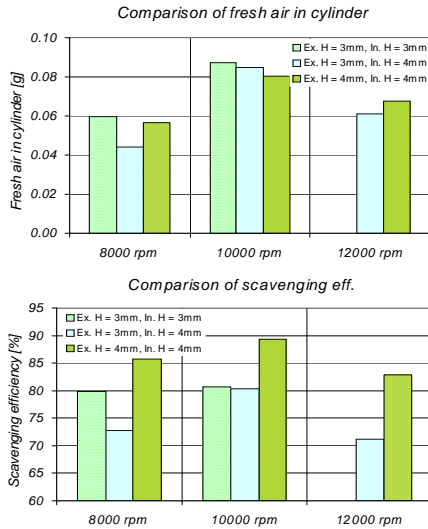
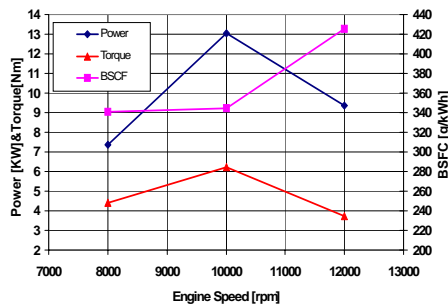Fig 9: Summary of 3-D CFD optimization of exhaust/intake port height

Fig 10: Predicted Engine Performances

3.2. MBS Analysis

As the first step of structure analysis, mechanical kinematical and dynamic analyses were conducted to specify the boundary conditions, using MBS (Multi-Body System) analysis tools. Fig 11 shows the moving components inside the ADAMS environment, the direction of rotation of the crankshaft, and the global coordinate system that is used to reference the forces acting on the main bearings, the balancing system, and the sum forces on the housing. Fig 12 shows the gas forces acting on each component of this mechanism, including the cylinder compression and combustion force and the forces induced from the scavenging pumping.
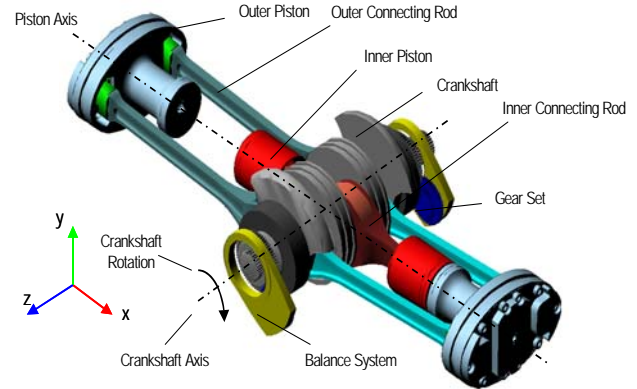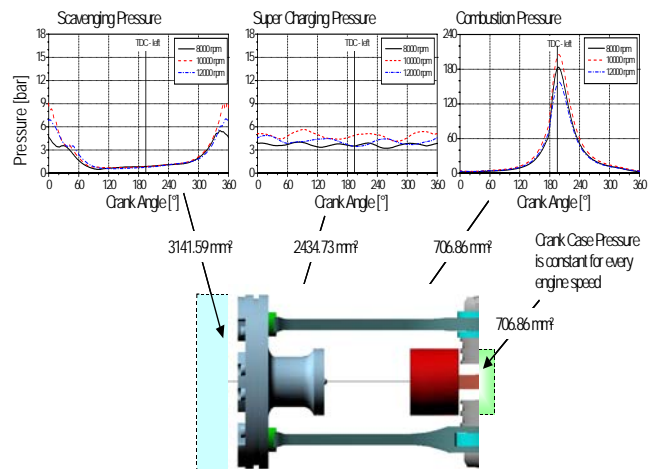

Fig 11: ADAMS MBS Simulation Model

Fig 12: Gas Pressure Forces Acting on Each Component

3.2.2. Crankshaft Main Bearings

Fig 13 shows the calculating results of the main bearing force with and without combustion and scavenging pressures on the mechanical mechanism. It is displayed in two graphs where one graph shows the main bearing forces in global x-direction, the other in global y-direction. The black (solid) line shows the force at bearing #1, the red (dash) line the force at bearing #2, and the blue (dash-dot) line gives the sum value. It is indicated that the gas forces have significant influence on the bearing force.
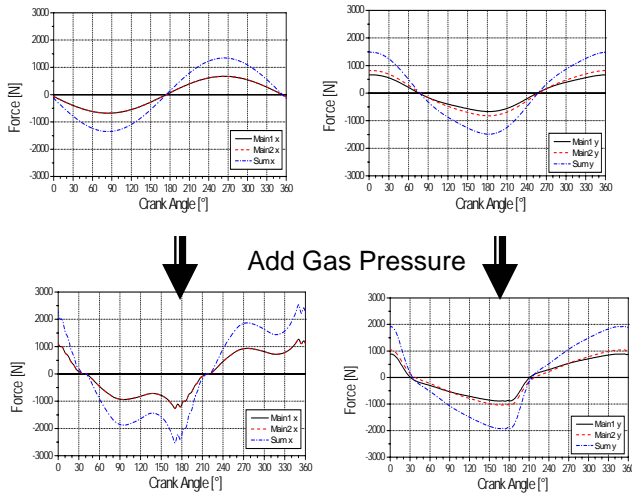
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**Fig 13: Main Bearing Forces @ 10,000 rpm
With and Without Gas Pressure**

3.3. FE Analysis

FE analyses of the moving opoc engine components (crankshaft, bridge, outer connecting rod assembly, inner connecting rod) are conducted with the load cases provided by the MBS analysis.

3.3.1. Crankshaft Analysis

The Model analysis of the crankshaft shows that the first bending and torsional modes of the crankshaft assembly are at a frequency of 1723 Hz and 2957 Hz respectively, which are well above the main excitation engine orders at 12,000 rpm.

The stress and deflection analysis is conducted for strength evaluation of the crankshaft. The result of this analysis is shown in Figs 14.
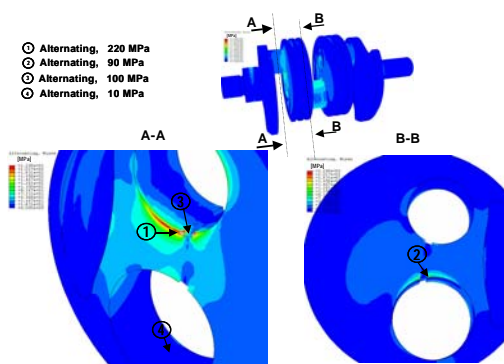

Fig 14: Alternating Von Mises Stress at 10,000 rpm

The deflection analysis of the crankshaft was performed for clearance evaluation. The maximum deflection of 0.58 mm occurs on the inner webs at 12,000 rpm. Fig 15 shows the radial deflection relative to the center axis of the crankshaft.
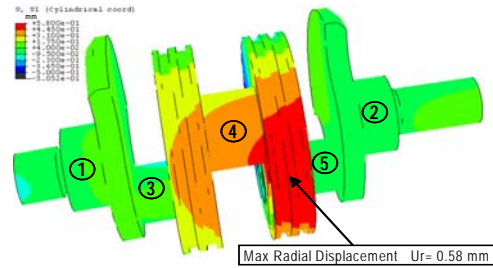

Fig 15: Radial Deflection of the Crankshaft

3.3.2. Outer Connecting Rod Assembly Analysis

FEA analysis is conducted on outer connecting rod assembly as shown in Fig 16, with the bridge, the two connecting rods spacers, and the two different connecting rods (fork & central). The fork design of the outer connecting rod includes the raceway of the needle bearing between the outer connecting rods. The model does not consider assembly loads due to bolt assembly or press fit between parts.
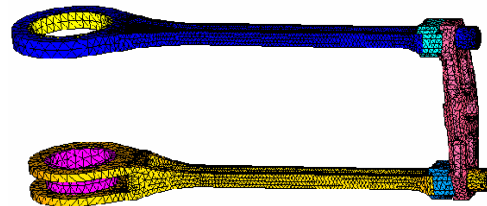

**Fig 16: FE model of the Outer Connecting
Rod – Bridge Assembly**

The load from the combustion forces and the inertia forces of the outer piston are applied to the needle bearing running face on the bridge.

The results of the stress distribution at maximum and minimum tensile load are shown in Fig 17.

At the indicated locations (1 and 2) the Von Mises stress has a maximum of 500 MPa and 480 MPa. The maximum (tensile) stress occurs on the inside of the connecting rods.
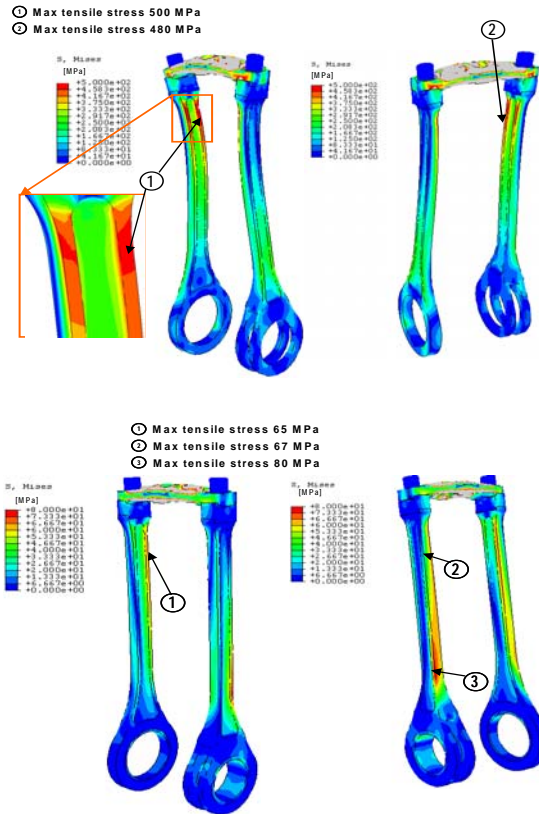
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Fig 17: Von Mises stress distribution for max./min. tensile load at 10,000 rpm

To evaluate the maximum deflection values of the outer connecting rod assembly, the maximum tensile load (10,000 rpm) is used. The deflections in the three Cartesian directions are analyzed: along the crankshaft axis (direction 1), perpendicular to the assembly plane (direction 2), and along the outer connecting rods (direction 3)

The results are shown in Fig 18, where the direction 1 is displayed to the left, direction 2 in the middle, and direction 3 to the right.

The maximum deflection along the journal / crankshaft axis is 1.99 mm, bending inwards. The design clearance needs to be ensured at this location.

The maximum deflections out of the assembly plane (0.13 mm) and along the connecting rod axis (1.07 mm) do not indicate interference with other engine components at that crank angle.
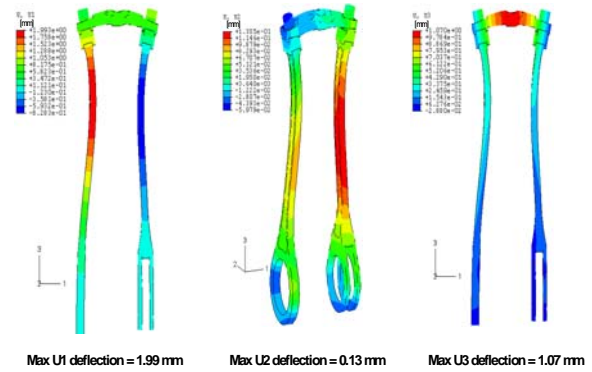

Max U1 deflection = 1.99 mm   Max U2 deflection = 0.13 mm   Max U3 deflection = 1.07 mm

Fig 18: Deflection results of the outer connecting rod assembly

The analysis reveals that the safety factor for the first buckling load is 25.8, which is sufficient against buckling.

3.4.   Muffler Design

The target for the exhaust muffler was to reduce overall noise to less than 60 dBA at 7 meters while maintaining a minimum package size (approximately muffler volume is 1.3 liters). The exhaust muffler should not also not to negatively impact engine performance. Detailed models of the intake and exhaust system were added to the GT-Power model developed for engine performance simulation and used to develop a muffler concept. The initial overall sound pressure level with no muffler in place was predicted to be 104 dBA at 10,000 rpm at 7 meters.

Using a combination of exhaust pressure loss, pressure wave reflection, and pressure energy absorption, a concept muffler was designed to minimize acoustic pressure pulsations. Attention has been paid that relatively low gas velocities are ensured to minimize flow noise.

Fig 19 shows the proposed muffler concept, which consists of three chambers separated by solid baffles. Exhaust gas enters the muffler at the middle chamber via the two exhaust ports marked 1 and 2 and indicated with red arrows. The exhaust flow within the muffler is indicated with orange arrows. Exhaust flow enters the resonance pipe opening in the middle chamber and exits in the left chamber. The length and diameter of the resonance pipe was tuned to create the necessary transmission loss. A middle pipe connects the left and right-hand chambers. The end of the middle pipe in the left chamber is closed. Flow enters the middle pipe via

perforations in the middle pipe. The number and size of these perforations can be adjusted to control the back-pressure. In the right chamber the middle pipe branches to two perforated tailpipes. The right chamber is packed with wool, which absorbs the acoustic energy. Exhaust gas then exits the two tailpipes.
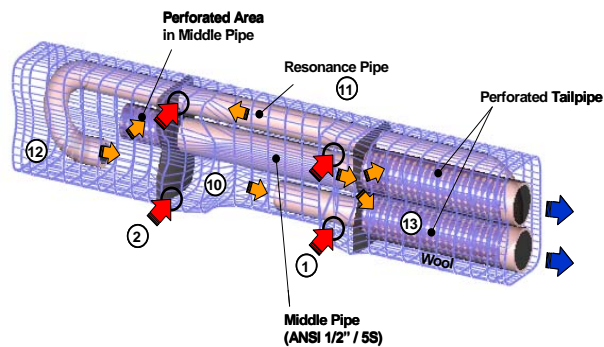

Fig 19: Exhaust muffler concept

Fig 20 shows the performance of the muffler with two different back-pressure values relative to the base engine without a muffler. Back-pressure ranges from 2-2.5 bar. The left figure shows engine torque, while the right figure shows overall sound pressure level at 7 meters. It can be concluded that with the designed muffler, a superb acoustic performance with minimal impact on engine torque is achieved.
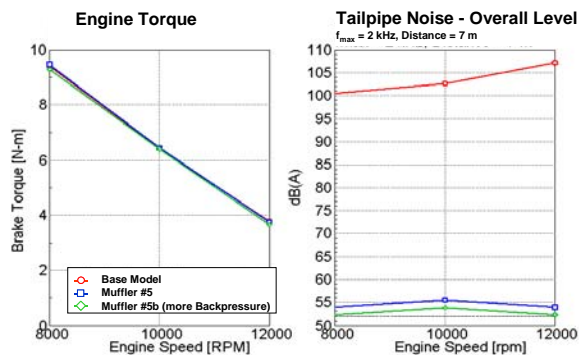

Fig 20: Engine torque & tailpipe noise with muffler concepts

4.   Testing Investigations

The opoc engine that has been built in this program is under extensive testing investigations when this paper is written. Fig 21 shows the setups of the engine at APT's test cell that was specifically configured for the EPC engine development. Test results will be verbally

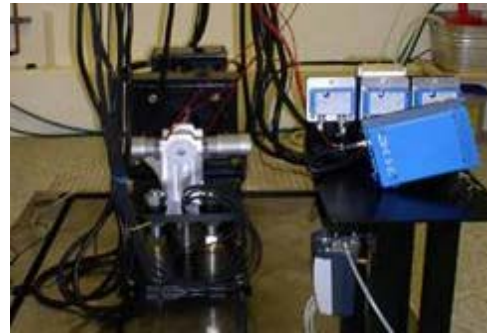presented during the SAE conference, by which time results will be available.

Fig 21: opoc Engine Setup at APT's Test Cell

5.   High Speed Generator

A rotary generator is developed in parallel to the engine development process. The generator operates at the speed of 40,000 rpm through a 4:1 gearbox. Cooling fan is integrated in the generator rotor to ensure reliable operation. Fig 22 shows opoc EPC with the generator.
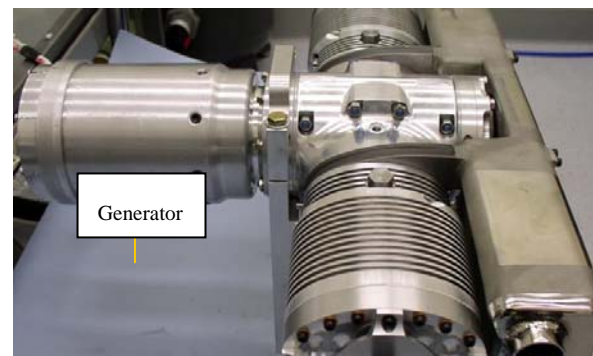

Fig 22 opoc EPC with Generator

6.   Conclusions

Excessive fuel consumption, emissions and noise associated with truck idling must be significantly reduced. The unique Electric Power Cell (EPC) presented in this paper offers the solution as the truck Accessory Power Units (APU).

opoc engine is the key component of the EPC that offers high power density, high fuel efficiency, low operational noise and low exhaust emissions. Engine performance simulation indicated that the engine performance satisfy all requirements.

The designed asymmetric uniflow scavenging system realizes over 85% scavenging efficiency and close-to 100% trap efficiency. This achievement will make this two stroke engine, the first time in the industry, to be competitive in emissions and almost double the power compared with the four stroke engines.

Taking the advantage of allowable high engine back pressure, the designed exhaust muffler is predicted to be able to reduce the engine exhaust noise to the level of less than 60 dBA. This feature enables the EPC offer significant advantage when quiet truck parking is critical requirement.

The conducted finite element analysis of the key engine components assisted the mechanical design optimization and ensures the robust engine structure.

Engine testing investigation is under way in realizing all the engine features and discovering techniques of further optimizing the engine design.

The advancement of the art in terms of power density and quietness of the EPC offers critical features to meet the special military demands.

Reference:

[1] Proposal to Reduce Idling from New 2007+ Heavy-Duty Diesel Trucks, Public Workshop, California Environmental Protection Agency Air Resources Board, June 4, 2003