Volvo Powertrain North America

DOE/NETL
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Project Final Report
September 2005 – March 2008

Project Title:
"Very High Fuel Economy, Heavy Duty, Constant Speed, Truck Engine Optimized Via Unique Energy Recovery Turbines and Facilitated by High Efficiency Continuously Variable Drivetrain"

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DISCLAIMER

Summary

September 2005 – August 2006

The project began under a corporative agreement between Mack Trucks, Inc and the Department of Energy starting from September 1, 2005. The major objective of the four year project is to demonstrate a 10% efficiency gain by operating a Volvo 13 Litre heavy-duty diesel engine at a constant or narrow speed and coupled to a continuously variable transmission.

The simulation work on the Constant Speed Engine started on October 1st. The initial simulations are aimed to give a basic engine model for the VTEC vehicle simulations. Compressor and turbine maps are based upon existing maps and/or qualified, realistic estimations.

The reference engine is a MD13 US07 475 Hp. Phase I was completed in May 2006 which determined that an increase in fuel efficiency for the engine of 10.5% over the OICA cycle, and 8.2% over a road cycle was possible. The net increase in fuel efficiency would be 5% when coupled to a CVT and operated over simulated highway conditions.

In Phase II an economic analysis was performed on the engine with turbocompound (TC) and a Continuously Variable Transmission (CVT). The system was analyzed to determine the payback time needed for the added cost of the TC and CVT system. The analysis was performed by considering two different production scenarios of 10,000 and 60,000 units annually. The cost estimate includes the turbocharger, the turbocompound unit, the interstage duct diffuser and installation details, the modifications necessary on the engine and the CVT. Even with the cheapest fuel and the lowest improvement, the pay back time is only slightly more than 12 months.

A gear train is necessary between the engine crankshaft and turbocompound unit. This is considered to be relatively straight forward with no design problems.

September 2006 – November 2006

Both steady state and high altitude (1675 m/25°C) simulations have been performed for the MD13 TC engine with updated boundary conditions and turbo maps. The results are obtained by cycle optimization at "engine out" conditions, i.e. NOx and soot emissions.

Two "engine out" NOx levels are considered. The level of "engine out NOx" is dependent on many factors as SCR efficiency, DPF etc. By analyzing two levels certain sensitivity information can be obtained.

A simplified load cycle has been defined which is used as the basis for the stress analyses of the turbines and compressor.

The mechanical design of the turbocharger and the turbo compound unit has started. The aero design and the stress analyses of the compressor and turbocharger turbine are more or less finalized.

December 2006 – February 2007

The mechanical design of the compressor is finished. The parts including the compressor, housing and diffusers are being ordered.

The mechanical design of the turbocharger is finished. Ordering for the turbocharger assembly, housing and bearing housing will be done as soon as appointing manufacturer.

The mechanical design for the most parts of the turbo compound unit is finished.

No simulations have been performed during this period

March 2007 – May 2007

Baseline engine testing for ESC, FTP and Road cycle on a MD 13 production using nominal performance curve is done. Engine testing for a suggested narrow range speed performance curve is done.

All the mechanical design of the compressor, turbocharger and turbocompound is finished. The parts are ordered some are manufactured including the compressor.

June 2007 – Sept. 2007

The compressor has been tested successfully. Engine simulations have been conducted with the tested compressor data. The cycle specific fuel consumption (SFC) is only 0.2 % below previous data run with a "theoretical map".

Turbo and TC design work is completed and drawings and related documents ready.

Regarding hardware deliveries most of parts have been received and they are in good shape. The rest of the parts including the radial turbine and the axial turbine should be available early November and functional testing will be commencing in December.

The immediate remained activity in this phase are to verify the turbocharger and turbocompound turbines and their. This will include characteristic parameters such as flow and efficiency versus pressure ratio at different rotational speeds. In the functional test rotor dynamics and other mechanical features will be verified. Also the other remained activity is to test the base engine with and without the turbocompound to verify the efficiency improvement.

Oct. 2007 – Dec. 2007

Parts for the turbocharger and turbocompound units have been delivered, balanced and assembled in good order.

Turbo tests have been run with the following results:

- The rotor dynamic displacements are well within the limits.
- The compressor performance complies with the previous test.
- The turbo turbine efficiency is in line with the predictions.
- The turbo turbine flow is about 3 % low (which is a good agreement).

Remaining activities in this phase are to verify the turbo turbine. This will include characteristic parameters such as flow and efficiency versus pressure ratio at different rotational speeds.

Jan. 2008 – Mar. 2008

Parts for the turbocharger and turbocompound units have been delivered, balanced and assembled in good order.

Rotor dynamic and performance testing of the turbo systems have been done and was shown that are well within the limits.

The project was terminated in March 2008 as a result of mutual agreement by DOE and Mack.

CONTENTS

1 BACKGROUND

Today the engines operate over a relatively wide speed range. This is a compromise of the typical drivetrains available today. If an engine can operate at a constant, or nearly constant, speed, efficiency improvements can be made. For the turbo charger, operating the engine at a constant speed will allow the turbine and compressor to operate closer to their peak efficiency through most of the driving cycle. By using, for example, a vaned compressor diffuser and a vaned turbine stator, the efficiency can be further increased.

Also the heat recovery turbine suffers from the demand of a wide speed range. Today it operates far below optimum tip speed at the design point because of high overspeed demand (the turbine must stand the increased speed at faulty gear shifts without bursting). By reducing the engine speed range, the design point speed can be chosen closer to the maximum turbine tip speed.

Besides the turbo charger and the turbo compound turbine other systems can be tuned to better efficiency if the speed range is decreased. Such systems can be injection system, cam shaft, engine after treatment system, inlet manifold etc.

The reduced speed range makes it necessary to have a CVT transmission alternatively to use a close ratio mechanical transmission with automated gear shifts.

Mack Trucks has set up a project with a Department of Energy (DOE) co-funding. The purpose of the project is to demonstrate an engine with the technology mentioned above.

The project is divided into four different phases. The first phase, which was engine and vehicle simulations, is finalized and reported. The second phase which was cost and feasibility study is also finalized and reported. The third phase is to design and build the needed components including compressor, turbocharger, turbo-compound and rest of modification to achieve the higher fuel efficiency. The focus of this quarterly report is simulation and design of the working conditions and components.

2 PHASE 1 SIMULATION RESULTS

During the first phase Volvo Aero Corporation simulated the engine and Volvo Technology simulated the vehicle with a Continuously Variable Transmission. The engine used in the simulations is the MD13. The maximum power was set to 354 kW @ 1300-1400 rpm and maximum torque 2700 Nm @ 1000-1200 rpm.

The engine simulations showed that it is possible to decrease the fuel consumption with 10.5% acc. to the OICA cycle and with 8.2% on three defined specific roads with certain weight factors. With a vehicle having this engine coupled to a CVT, the efficiency gain decreased to 5% when this driveline was simulated over the various roads.

The engine out NO_X level in the simulations was 1.18 g/kWh while the soot level was 0.058 g/kWh.

3 DESCRIPTION

3.1 Turbocharging system

To fully utilize the advantages with the turbocompound concept, a high efficiency turbocharger has been assumed. The lay-out is fairly conventional with radial compressor and turbine impellers. The compressor has a Low Solidity Airfoil diffuser, tuned for maximum efficiency at a higher pressure ratio than normally found on standard turbochargers. Due to the relatively high pressure ratio, and hence outlet temperature, a titanium impeller is needed.

For highest possible turbo turbine efficiency, a vaned turbine stator has been chosen together with a double-entry turbine housing.

The turbocompound unit has an axial turbine with stator. The concept is very similar to what is found on the D12D 500 Horsepower Euro 3 engine. The turbine is connected to the engine crankshaft via two reduction gear sets and a fluid coupling. The fluid coupling is necessary to dampen the crankshaft oscillations. The gearings are located in the rear end of the engine.

Also the gas conduit between the turbocharger and turbocompound unit (also called the Interstage Duct Diffuser) is similar to the D12D 500.

3.2 Engine

Contrary to the D12D engine, the MD13 has the engine transmission in the flywheel end. This makes it much easier to install the turbocompound unit to the engine as the gear on the crankshaft already exists as well as the transmission casing. Two new gears (one of them is a bull-gear) are assumed to be necessary. As the transmission casing must be somewhat enlarged, a new flywheel housing and a new timing gear plate are necessary. It is possible that a minor modification to the engine block must be done but this will not affect the product price.

4 COST ESTIMATE

A cost estimate has been performed. This has been done at two different annual numbers; 10,000 and 60,000 units per year. The procedure has been to compare every component in the systems with known costs for similar components. An adjustment for the complexity and size compared to the reference part has been made.

When comparing different annual numbers, the learning factor 0.87 has been used (except for standard details as screws, nuts, studs, etc.).

Included in the unit costs are the assembly cost and an OH of 20%.

The engine cost estimate has been divided into four different sub-groups:
- Turbocharger
- Turbocompound unit
- Interstage duct diffuser and installation details
- Modification necessary on the engine to accommodate the TC unit

The EGR system will be similar to the US10 engine and no additional costs have been charged. Due to the turbocompound system the packaging of the components possibly must be different.

The US10 reference engine has a Variable Geometry Turbocharger. The cost of this has been deducted from the added engine cost.

5 PAY BACK TIME

For the calculation of the pay back time the following premises have been used:

Truck mileage per day	600 miles (966 km)
Working days per year	240 days
Average fuel consumption	6.5 MPG (reference)(36.2 lit/100 km)
Increased fuel economy	5 – 7 - 10 % improvement
Fuel price	2.50 - 3.00 - 3.50 - 4.00 - 4.50 - 5.00 $/gallon

These figures result in an annual mileage of 144,000 miles (231,746 km) and that 22,154 gallons (83,861 liters) are spent in the reference case. With a fuel price of $3 per gallon this corresponds to $66,462 annually.

With the increased fuel efficiency and the fuel prices stated above, the annual savings are listed in table 1.

Table 1, Annual cost savings at various fuel prices and increased fuel efficiencies

	Fuel saving	5%	7%	10%
Fuel price $	2.50	$2 769	$3 877	$5 538
	3.00	$3 323	$4 652	$6 646
	3.50	$3 877	$5 428	$7 754
	4.00	$4 431	$6 203	$8 862
	4.50	$4 985	$6 978	$9 969
	5.00	$5 538	$7 754	$11 077

Using the driveline cost premium together with the increased fuel efficiencies and fuel prices above, the additional driveline cost will be paid back as a latest in slightly less than 15 months. Please see table 2.

Table 2, Pay back time in months at two different annual numbers and at various fuel prices and increased fuel efficiencies

		10,000/year			60,000/year		
Fuel saving		5%	7%	10%	5%	7%	10%
Fuel price $	2.50	14.7	10.5	7.3	12.5	9.0	6.3
	3.00	12.2	8.7	6.1	10.5	7.5	5.2
	3.50	10.5	7.5	5.2	9.0	6.4	4.5
	4.00	9.2	6.6	4.6	7.8	5.6	3.9
	4.50	8.2	5.8	4.1	7.0	5.0	3.5
	5.00	7.3	5.2	3.7	6.3	4.5	3.1

6 POTENTIAL INSTALLATION RISKS

The gear train between the crankshaft and fluid coupling is not designed; it exists only as principal lay-out. However, at the design stage of the MD13 engine, the turbocompound concept was considered and a CAD-model was created. This gear train is believed not to cause any problem.

Greater problem can be the location of the Exhaust Gas Recirculation Cooler. If the US07 location is kept to the US10 engine, the cooler and the turbocompound transmission housing will interfere.

Contrary to the US07 engine, the US10 will most likely have the EGR valve on the cold side of the gas path. Different EGR concepts have been simulated and the difference in engine performance was found to be relatively small. The idea is then to have dual pipes from the exhaust manifold (cylinders 1-3 and 4-6) which will join into a common pipe just before the EGR cooler. The EGR valve can be located on the inlet manifold side, close to the mixer. A relocation of the cooler seems necessary if this is possible with respect to packaging. If the EGR cooler can't be relocated, the turbocompound unit must be raised to avoid interference. In this case three additional gears are necessary due to the increased center distance.

September 2006 – November 2006

7 STEADY STATE AND HIGH ALTITUDE SIMULATIONS

Both steady state and high altitude (1675 m/25°C) simulations have been performed for the MD13 TC engine with updated boundary conditions and turbo maps. The results are obtained by cycle optimization at "engine out" conditions, i.e. NOx and soot emissions. No Engine After Treatments (EATS) has been considered during this study which would naturally affect the results.

The boundary conditions used origin from the US10 project.

Both the compressor and turbo turbine have been updated from the achieved progress in both structural and aero dynamical analysis.

Two different "engine out" NOx levels are considered. The level of "engine out NOx" is dependent on many factors such as SCR efficiency, DPF etc. By analyzing two levels certain sensitivity information can be obtained.

During optimization some parameter limits have been applied. In the case considered with high NOx level just over 1% improvement is obtained over the low NOx level case in road BSFC. The improvement would probably be higher if not the PCP was limiting. Also, NTE NOx has been set to 2.5 g/kWh in the high level NOx case due to limitations in the emission model used. This also deteriorates the improvement.

High altitude simulations have also been performed with NTE limits on NOx/Soot ~2.5/0.06 g/kWh and exhaust manifold and compressor temperature limits of ~685/230°C. The obtained points are seen in the compressor map below. No derating problems were encountered. The points are used as reference values when simulating transient load changes at high altitude described in the next section.

8 LIFE REQUIREMENTS

For the stress analysis (LCF) thermodynamic input in form of transient turbo speed, temperature, pressures etc. are required. In order to extract these data, a simplified truck operating cycle is defined seen in the table below for cold starts, warm starts and load- speed changes.

Following requirements applies for:
Turbo turbine

9 3.8E3 cold starts defined in Table 3-3
9 15E3 warm starts defined in Table 3-2
1E6 load cycles defined in
9 Table 3-1
Compound turbine 120E3 engine braking cycles defined in
9 Table 3-1

Table 3-1, Transient load cycle

Load cycle at high altitude, 1675m/25°C		
Time (s)	**Speed**	**Load**
0-4	1000	25%
10	1000	100%
20	1300	100%
30	1000	0% (Motoring)
60	1000	25%
63	1690	Engine brake
73	1900 (max.)	Engine brake
75	1900 (max.)	Engine brake (steady state)

Table 3-2, Transient warm start

Warm start, 0m/25°C		
Time (s)	**Speed**	**Load**
0-15	1000	Scaled data from test
18	1000	25%
23	1000	100%
34	1300	100%
124	1300	100% (steady state, "warm engine")

Table 3-3, Transient cold start

Cold start, 0m/25°C		
Time (s)	**Speed**	**Load**
0-15	1000	Scaled data from test
18	1000	25%
23	1000	100%
34	1300	100%
224	1300	100% (steady state, "warm engine")

December 2006 – February 2007

9 MECHANICAL DESIGN

9.1 Turbocharger

9.1.1 General

The turbocharger has a high performance compressor and a high performance radial turbine. Besides the compressor and turbine, it is fairly conventional with floating ring radial bearings and a hydrodynamic thrust bearing.

The turbocharger was briefly described in the last quarterly report. The design speed is around 110,000 rpm. The turbine has a housing which is of the double entry type. This means that each group of cylinders feeds 180° of the turbine's circumference. The figure below shows the turbocharger.

The lay-out is close to completion and all parts with long lead-time are under process of ordering.

9.1.2 Compressor

The design of the compressor was described in last quarterly report. Parts will be manufactured by milling in titanium. The compressor is of an advanced design, it has 9 full blades and 9 splitters. It has a LSA diffuser in order to have good efficiency together with an acceptable operating width. The inlet hub diameter is 17 mm, the inlet tip diameter is 54 mm and the outlet diameter is 94 mm. The compressor and three different diffusers are in the process of ordering. The diffusers which are being manufactured and tested are: nominal, 1.5° opened and 1.5° closed.

9.1.3 Turbine

The turbine, which was described in last quarterly report, is also in the ordering process. The turbine will be made in cast Inco 713C. To be able to make the investment castings, milled waxes will be produced.
For highest efficiency, the turbine is equipped with fixed guide vanes. The turbine has 11 blades and the stator has 18 blades. No scalloping is done on the turbine.

9.2 Turbo-compound unit

9.2.1 General

The compound is rather similar to the unit used in the Europe version of D12D500. The design has not progressed as far as the turbocharger, but most of the lay-out is finished. No detailing is done.

The turbocharger and the compound unit are located on a common axis which means the gas conduit between turbocharger and the compound unit is annular.

Design of the interface to the rear transmission in the flywheel housing and the design of the geartrain in the rear transmission are completed. Most of these parts are under process of ordering.

9.2.2 Turbine

The power turbine is of the axial type and it has a stator. Like the turbocharger turbine, this will be manufactured in Inco 713C. The manufacturing process is the same.
The turbine has an OD of Ø118 mm and an ID of Ø80 mm. The number of rotor blades is 23 and the stator blades are 17.

9.2.3 Fluid Coupling

Similar to the D12D turbo compound unit, a fluid coupling is needed. The reason for this is to dampen the oscillations from the piston engine. The coupling will be equal to the D12D except the gears. Due to new speeds and slightly changed center distance, the new gears are necessary.

10 ENGINE TEST

10.1 Reference Engine Tests

For the reference engine test one standard engine from the production line has been used. No change has been done to the engine. Also the data file used in the engine was latest production file for US07 engine with no changes.

10.1.1 General

A standard torque curve map for MD 13 engine D13F 360 kW - 2240 N-m (485 hp – 1650 ft-lb) is used for the baseline engine test. The nominal engine performance curve (torque and power vs. speed) was used.

All the tests are done with no after-treatment device and the emission results are engine out data. A set of restrictions are applied as the test conditions:

Exhaust pressure	: 25 kPa
Inlet pressure	: 3.5 kPa
CAC Temp	: 44 °C

10.1.2 European Standard Test (ESC)

The composite data from the 13 modes OICA points for the baseline test engine and the result from the simulation reported in the previous reports are compared. There was a negligible difference (about 0.5%) in bsfc between the experiment and simulation. As it was mentioned in the introduction the baseline engine test is conducted with no after-treatment while the result of the simulation are considering DPF unit. For the mentioned situation the soot result from the simulation is much less than the engine-out results of the baseline engine test.

10.1.3 Federal Transient Protocol Test (FTP)

Although no simulation has been done based on FTP while conducting test on MD13 engine, federal transient test also was done to have a reference for emission analysis.

10.1.4 Road Cycle Test

The road simulation was based on a combination of roads for a typical long haul truck. Three roads were used: flat, rolling hills and hilly. The simulated road cycle is a combination of 70% flat, 25% hilly and 5% rolling hill. After comparing result of the road cycle test with the simulation result reported previously the change was around 1.8% increase in bsfc.

10.2 Narrow Range Speed Engine (NSE) Tests

In this section the result of the engine testing under the narrow range torque curve will be presented. The narrow range test has been conducted on a standard MD13 engine with no modification in hardware or software. These are preliminary test and are not intended to be solid result to focus at. The reason of these sets of test is to get a baseline to set goals and directions for further modification on the hardware and calibration of the engine.

10.2.1 General

The torque curve used in these tests is the modified version of the torque curve from the simulation result with an extra modification for turbocompound. The previously simulated engine performance torque curve is included the turbo compound. To compensate for the effect of the turbo compound the maximum power level has been altered to match maximum torque value in the original base engine performance curve at its own corresponding speed. For the rest of the points in the torque curve graph the corresponding torque for each speed has been shifted down as low as maximum torque.

10.2.2 European Standard Test (ESC)

The composite result of the ESC for the specific fuel consumption (bsfc) for the narrow range speed engine test shows an improvement of 4.5%. This proves that using an engine with no hardware or software modification in a narrow range speed performance curve will improve the fuel efficiency. This result is just preliminary result and can be used as a reference only.

10.2.3 Federal Transient Protocol Test (FTP)

The suggested narrow range speed performance curve is also used to perform FTP test. The result showed even a better fuel efficiency improvement for the FTP around 11.2%. The reported values are with no changes in hardware or software of engine and just for the construction of a baseline for the next stage engine development.

10.2.4 Road Cycle Test

To perform a narrow range road cycle test the suggested power and speed distribution in previous report [1] is used. According to this distribution two ranges of speed and power (as it can be seen in the figure 2.1) are considered:

"Narrow range"	70% of time at 185kW@1000 rpm -1100 rpm
	30% of time at 354kW@1300 rpm

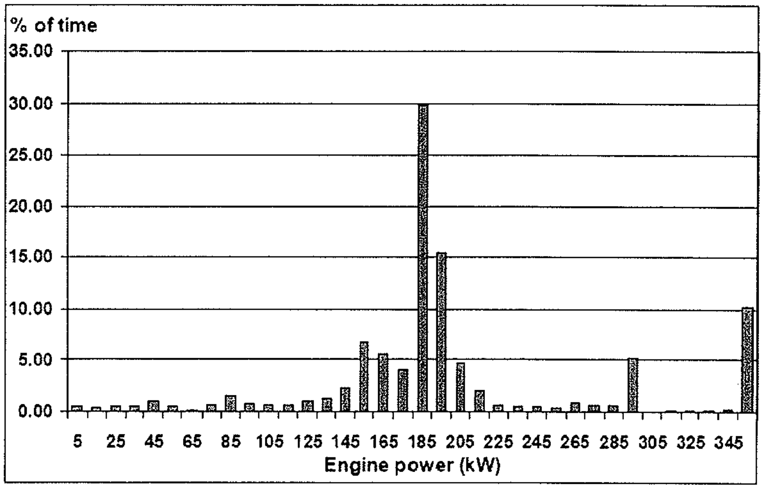

Figure 1. Power distribution in a combined road cycle (Flat 70%, Hilly 25%, Rolling Hill 5%)

In this part of the engine testing a two-hour combined road cycle performed such that the 185 kW considered as the switching point between the low speed and high speed. The other consideration in the road cycle test is to compare the effect of changing the low speed. It was previously reported that narrow range speed is considered to work in two speeds of 1000 rpm and 1300 rpm. In addition to the 1000 rpm, 1100 rpm is also tested and the results show a strong dependency of the bsfc to the working speed. The fuel efficiency gain with 1000 rpm and 1100 rpm as low speed was respectively 3.6% and 2.7% while the high speed was fixed at 1300 rpm. The NOx level at 1000 rpm exceeded the standard level which with increasing the speed to 1100 rpm NOx level was corrected. The reason of such high NOx level at lower speed was using the variable geometry turbine (VGT) in the base engine as the baseline engine map was forcing to close the EGR valve at low speed to avoid surging the VGT. The turbocharger and compressor under this project are designed to work with no difficulty in lower speed.

11 TURBOMACHINERIES

All the simulations are finished and the most of the design for the compressor and turbines are done. The finalized components are ready to be ordered or are already ordered.

11.1 Compressor

The compressor is of an advanced design, it has 9 full blades and 9 splitters. It has a LSA diffuser. The compressor and three different diffusers (nominal, 1.5° opened and 1.5° closed) are manufactured and will be tested. The compressor and compressor housing has been delivered. The compressor diffuser will be delivered in June 2007. The compressor is planned to be tested on a standard turbocharger.

11.2 Turbocharger

The turbine is now in the manufacturing process. As it was explained in previous reports, the turbine is equipped with fixed guide vanes and has 11 blades and its stator has 18 blades.

June 2007 – Sept. 2007

12 COMPRESSOR TESTING

The compressor has been tested with good result in a turbo test rig. In this rig a standard turbo turbine is connected to the compressor. Drive power is utilized via a high flow compressed air system and boosted in a combustor just up-stream the turbine inlet.

The compressor was tested with three different diffusers. The total to total efficiency is at the design target. At pressure ratio 4 the max efficiency is less than 1% below target. The pressure ratio is somewhat low at high speeds and requires a speed increase of 1-2 % to compensate. This should be manageable from a stress point of view. The map width is good but the choke flow at higher revs is about 3 % too low.

Oct. 2007 –Mar. 2008

13 TURBO SYSTEM DESIGN AND MANUFACTURING

All the parts for turbocharger and turbocompound units are manufactured and delivered. All turbo testing are done.

13.1 Simulation and mechanical design

No engine simulations were performed during this period. The mechanical design of the turbocompound system was completed and reported earlier. Design of the turbo turbine test rig has been done.

13.2 Manufacturing

Turbocharger and turbocompound prototype units have been delivered in good order. No critical deviations have been observed and the rotors have been balanced in line with the balancing criteria.

14 TURBO TESTING

In this period the turbocharger was tested with focus on rotor dynamics and also to get an indication of turbo turbine performance. In the previous test (compressor performance) the compressor was assembled in a commercial turbo.

14.1 Rotordynamic test

The purpose of the turbo test was to check that the unit was mechanically sound. The displacement was measured to verify its rotor dynamics. In the operating regime the displacements were well within limits.
The mechanical efficiency was estimated to be about 97%. A separate test with the bearing oil as cold as possible, was also carried out. At maximum rotational speed the bearing loss was 4000 W which corresponds to 95% efficiency.

The compressor component testing carried out earlier was performed with a bearing housing and turbine from a commercial VGT turbo charger. It is therefore possible to compare the bearing losses of the CSE bearing housing with the "commercial" bearing housing. The analyses indicate that the bearing losses are lower for the CSE turbocharger. At maximum rotational speed the bearing efficiency is approximately 2%-points higher, i.e. 97% instead of 95%, for the CSE turbo.

14.2 Turbo performance test

The turbine isentropic efficiency is calculated as

$$\eta_{tt} = \frac{\dot{m}_{comp} \cdot C_{p,comp} \cdot (T_{tot,comp,out} - T_{tot,comp,in})}{\dot{m}_{turb} \cdot C_{p,turb} \cdot T_{tot,turb,in} \cdot \left[1 - \left(\frac{p_{tot,turb,out}}{p_{tot,turb,in}} \right)^{\frac{\kappa-1}{\kappa}} \right]} / \eta_{mech}$$

The analysis of the performance test has been focused on the turbine performance. The compressor performance seems to be in agreement with the component test results presented in the third quarterly report of the phase III.

Turbine isentropic efficiency has been calculated. Note that the efficiency is expressed as total-to-total and includes the rig exhaust diffuser. Since the divergence angle of the rig diffuser (2θ) is as high as 30°, the efficiency loss is judged to be approximately 3%.

At pressure ratios above 2.2 the calculated efficiency is 1-2% below the prediction but this is before correction for rig diffuser loss. Taking this into account the tested turbine seems to be 1-2% better than predicted. At pressure ratios below 2.0 the tested turbine seems to be 1-2% worse than predicted.

The turbine results at this point should be seen as indicative only. In the planned cold rig test more reliable data will result and these will hopefully confirm the encouraging performance measured in the turbo rig.

Turbine corrected mass flow was also measured. Compared to the prediction the measured level is about 3% lower, which is seen as a good agreement.

14.3 Turbo turbine test

To test the turbo turbine and turbocompound units engine test is done to verify that the turbo and TC unit operates mechanically sound. Under these testing the rotor dynamic behavior for both of the components is monitored. The turbo turbine has been run at "cold" conditions to verify its aerodynamic performance.

15 ENGINE MODIFICATION PARTS

All of the modification parts are manufactured. The major parts are:

- Camshaft
- Timing Gear Plate
- Flywheel Housing
- Exhaust manifold

Engine assembly will be started upon receiving all the modification parts and turbo testing.

16 FUTURE ACTIVITIES

The remaining activity from the phase III of the project is to develop narrow range speed control system of the engine to verify the engine efficiency and emission in the test cell. Unfortunately the project was terminated at this point as a result of mutual agreement by DOE and Mack.