DOE Final Report 2004 DE-FC05-00OR22810

Final Technical Report
Clean Diesel Engine Component Improvement Program
Diesel Engine Waste Heat Recovery
Utilizing Electric Turbocompound Technology
Department of Energy Contract
DE-FC05-00OR-22810

Date: 17 December 04 1 Caterpillar, Engine Research DOE Final Report 2004 DE-FC05-00OR22810

Project Title: Clean Diesel Engine
Component Improvement Program
Diesel Engine Waste Heat Recovery
Utilizing Electric Turbocompound Technology

Recipient: Caterpillar Inc. DOE Award No.: DE-FC05-00OR-22810 DOE Project Officer: John Fairbanks Principal Investigator: Ulrich Hopmann Program Manager: Ulrich Hopmann

Date: 17 December 04 2 Caterpillar, Engine Research DOE Final Report 2004 DE-FC05-00OR22810

Summary

Caterpillar's "Technology and Solutions Division" carried out a research program on waste heat recovery with support from DOE and the DOE Office of Heavy Vehicle Technology. Electric turbocompound technology has been chosen to improve the fuel economy of on-highway truck engines. The basic principle is to extract surplus exhaust heat with an electric generator and convert the energy into electrical energy which is then transferred to an electric motor, which contributes to the power output of the diesel engine. An electric turbocompound (ETC) system has been conceived and designed. As part of this effort, a novel turbocharger, sized for a 15-ltr on-highway truck engine, has been analyzed, designed, built, and tested. The design is a radical change from conventional turbochargers and incorporates an electric motor/generator (M/G). All major components of the turbocharger have been designed for highest efficiency. The prototype turbocharger has been tested on a gas stand where a significantly higher efficiency than production turbochargers has been measured. Another essential milestone was the development of the electrical machinery. Two electrical machines, one high-speed generator and one electric motor have been analyzed, designed, built, and tested. High efficiency and compactness were the design targets. Both machines were tested on the test rig in generating and motoring mode. In the case of the electric motor the measured efficiency was satisfactorily high and within the expected range. The high-speed generator has been tested and was able to cover the required speed and torque range with additional air-cooling. Because of the excessive heat generation the efficiency was below the target. A detailed technical path to improved performance has been identified and should be followed prior to production. Another milestone was the development of the control system and control strategy. Considering the complexity and the additional degrees of freedom of the electric turbocompound technology, a system model has been built which allowed for the development of a special control system. A rig test of the ETC hardware proved the proper functionality of turbocharger, high-speed generator and control system. Waste heat was recovered by producing electrical power

over a wide range of typical turbocharger operating conditions. The control system was able to control power flux and turbocharger operation for the entire range. A production engine was tested to determine baseline engine performance. In a final step the same engine was equipped with the complete ETC system. The engine was tested in the lab while running in turbocompound mode; thereby demonstrating the principle of operation. Unfortunately, a bearing system hardware failure prevented a system level BSFC demonstration. Future development activity would include the development of a robust bearing system.

Introduction

Caterpillar's "Technology and Solutions Division" has developed a new waste heat recovery technology that improves the fuel economy of on-highway truck engines. A cooperative program between the DOE Office of Heavy Vehicle Technology and Caterpillar was aimed at demonstrating electric turbocompound technology on a Class 8 truck engine. The goal was to demonstrate the level of fuel efficiency improvement attainable with an electric turbocompound system. The system consisted of a turbocharger with an electric motor/generator integrated into the turbo shaft. The generator extracts surplus power at the turbine, and the electricity it produces is used to run a motor mounted on the engine crankshaft, recovering otherwise wasted energy in the exhaust gases. The electric turbocompound system also provides more control flexibility in that the amount of power extracted can be varied. This allows for control of engine boost and thus air/fuel ratio.

The team developed an electric turbocompound system for a Caterpillar heavy-duty on-highway truck engine. The scope of this project was to design, fabricate, and test an Electric Turbo-

Compounding (ETC) technical demonstration module including testing of lab engine with the ETC module. The primary objectives were:

Demonstrate ETC Technology on an Engine

• • Demonstrate improved Fuel Economy (with goal of 5% - Cycle Improvement from Exhaust Energy Recovery)

A multi-discipline approach was used in order to address the following key development areas: aero design, electrical machine design, engine performance, control system, structural analysis and test.

1. Layout and Design

1.1 System Architecture

The architecture of the ETC system is represented by the schematic shown in fig. 1.

Fig. 1 Electric Turbocompound System Schematic

The principle is as follows:

When the power produced by the turbocharger turbine exceeds the power requirement of the compressor, this surplus power is converted into electrical power by the electrical machine located on the turbocharger shaft. This surplus power can be recovered by the ETC system through an electric motor, mounted on the crankshaft, which assists the engine. The result of this process is an increase in system efficiency and improved fuel economy. Alternatively to compounding, the surplus power can be used to drive other electrical on-board devices, or truck electrical loads, or it can be stored. Thus, the principal mode of operation is when the electric machine on the turbocharger shaft acts as a generator, and the electric machine on the engine crankshaft works as a motor. The power extraction from the turbo can be varied to control boost, or alternately power can be drawn from the crankshaft and use to help accelerate the turbo.

Apart from the lower fuel consumption the ETC system offers other attractive benefits:

- No wastegate or VGT (Variable Geometry Turbine) needed
- Flexible engine operation versus fixed geometry of mechanical compound systems
- Turbo assist capability
- Capability for improved braking power

1.2 Computer Simulation

Due to the complexity of this research program, extensive system simulation was utilized. A set of test data for a production C15 engine was used for calibration of the model. Turbocharger maps for the production engine were incorporated into the simulation tool. In-depth modeling of engine, air system, and combustion was conducted by using Caterpillar’s own engine simulation tool. The high-level simulation model was completed in Simulink where the electric machines and open loop controls were included in the simulation model. After calibration the model essentially duplicated the production engine C15 test data. This model served as a baseline for all system development efforts.

1.3 Concept Development

Air-Handling System

Based on engine simulation results a single stage, centrifugal compressor utilizing vaned diffusers was identified as the best compressor for the electric turbocompound system. The compressor design fulfilled the specific truck engine requirements for pressure ratio, efficiency, and range (see Fig. 2).

Fig. 2 Compressor Impeller

The chosen turbine concept was a high work, single stage radial turbine. That turbine concept proved to be a good match for the high work requirements of the electric turbocompound operation.

A concept design layout of the electric turbocompound turbocharger is shown in fig. 3. The incorporation of a motor/generator on the turbo shaft was a major change in structure and functionality and required a complete new design of a turbocharger. As a result of a PUGH ranking analysis a conventional design with a mid-mount electrical machine was chosen. Other concepts, e.g. back-to-back configuration of turbine and compressor with an overhung generator,

were not pursued because of detrimental heat transfer effects from turbine to compressor and obstructed compressor inlets or turbine outlets.

In order to keep frictional losses low, it was decided to use rolling element bearings. The concept also included a water-cooled bearing housing and a water-cooled generator stator.

The concept design layout fulfilled the requirements of air system, bearing system and turbo shaft motor/generator. Analysis work, including shaft dynamics analysis, determined shaft diameter, bearing type, bearing span and bearing loads, and predicted whether shaft modals will occur in the turbo operating range. The shaft system in the concept phase is shown in fig. 4.

Fig. 4 Concept Phase Turbocharger Shaft Assembly

Rotor

Bearings

Fig. 3 Concept Design Layout of Electric Turbocompound Turbocharger

<u>Crank Starter/Generator/Motor</u>

A crankshaft starter/generator was already developed as part of the Caterpillar/DOE cooperative “MEI Truck” program. This concept was selected for the electric turbocompound technology demonstration. The initial plan was to utilize two crank motors, each rated at 30 kW, in order to meet the rating requirement. Because of space limitation in the flywheel housing this idea was abandoned; it was decided to redesign the existing machine to be able to absorb the high power generated by the electric turbocompound system. Fig. 5 shows the concept design of the crankshaft motor/generator.

Fig. 5 Concept Design of Crank Shaft Motor/Generator

<u>Turbo Motor/Generator</u>

The selection of an appropriate concept for the turbo shaft motor/generator was mainly driven by high shaft speeds in combination with high power output and packaging constraints. Since the shaft speed was given at this point by the chosen compressor design, a small rotor OD was important in regards to low centrifugal stresses in the rotor lamination. Also, rotor inertia had to be kept low for good engine response. High machine and thus system efficiency is essential for all turbocompound systems.

A PUGH analysis of candidate technologies was conducted, and a switched reluctance design emerged as the best alternative.

<u>Electronic Systems Integration and Controls</u>

Three approaches were investigated as candidates for the primary control signal:

A. Control of exhaust temperature

B. Control of turbocharger speed

C. Control of turbocharger boost

<u>A. Exhaust Temperature Control</u>

Investigation of control of engine exhaust temperature as a parameter for system control revealed difficulties with this approach. The simulation model predicted that an increase in demand on the engine would result in an initial dip in engine torque, see fig. 6. As the fuel-air ratio increases, the control system increases the turbocompound speed to reduce the quickly increasing exhaust

Stator and cooling

Rotor & hub

SRController

temperature. As more power is taken from the crankshaft to increase turbocompound speed, net crank torque is temporarily reduced at a time when the operator is demanding increased torque. This type of control would require modifiers in the controller that may increase the calibration effort. Also, the durability of the sensor was a concern in the hot, corrosive exhaust environment. The required shielding would have resulted in a slow sensor response with a time constant of many seconds. Therefore, the exhaust temperature control was not chosen.

Fig. 6 Simulation Results with Control of Exhaust Temperature

B. Turbo speed control

Control of turbo speed would consist of controlling shaft speed to a setpoint speed for an engine operating point. This approach also resulted in a dip in engine torque (fig. 7), but not as severe as the exhaust temperature approach. Therefore, the turbo speed control was not chosen.

Fig. 7 Simulation Results with Turbo Speed Control

C. Boost control

Simulation of boost control resulted in a fast and stable loop (see fig. 8). Some overshoot/undershoot is evident in a constant pressure demand step response as the control brings pressure to setpoint following a disturbance. The setpoint will change with engine operating point to obtain best efficiency. Simple filtering of the setpoint prevented non-minimum phase behavior and minimized overshoot/undershoot.

Initial drop in torque
Can be mitigated by filtering the setpoint

Increase power demand results in an initial dip in engine torque

0 5 10 15 20 25 30 35 0 0.51 Boost Control (Engine Speed = 1800 rpm, Variable Pressure Setpoint) Demand (0-1) 0 5 10 15 20 25 30 35 150 200 250 300 Intake Pressure Setpoint Intake Pres. (kPa) 0 5 10 15 20 25 30 35 500 1000 1500 2000 2500 Crank TQ (N.m) Time (sec)

Fig. 8 Simulation Results with Boost Control

A boost pressure sensor is already on production engines. System dynamics for boost and speed signals change much less than for exhaust temperature, resulting in simpler algorithms and less calibration effort. The boost pressure approach appeared to offer slightly less variation in response over the engine operating range. In conclusion, boost pressure proved to be the best control signal.

1.4 Design

Air Handling System - Compressor

The compressor chosen under "1.3 Concept Design" was further refined in ProE and a compressor volute was included in the design. Fig. 9 shows the compressor stage with vaned diffusor and volute.

In the preliminary design of the compressor, an FEA analysis to predict stress and vibration of the impeller was performed. The objectives of the FEA analysis were to determine if the design would meet the low cycle fatigue requirement and the blade vibration frequency requirement.

Fig. 9 Compressor Impeller with Vaned Diffuser and Scroll

(filtered)

Overshoot/
undershoot

The FEA results indicated that the stress levels of the initial design were too high to meet the low cycle fatigue limit, fig. 11. Therefore, it was decided to have the impeller made of a titanium alloy. The stress goal was reached with the change in material and by modifying the backface geometry and increasing the blade fillet radius. The blade vibration frequency was found to meet the design target, see fig. 10.

Fig. 10 Compressor FEA Modal Analysis Image Fig. 11 Compressor Blade Stress Analysis Image

The performance prediction for the final compressor design showed a peak efficiency of approximately 85% (total-to-static), a very good value.

Air Handling System - Turbine

In the preliminary Design, a radial and a mixed flow configuration were evaluated. Both met the application requirements. However, the radial configuration was selected for further development as it had higher efficiency, lower mass, and lower inertia.

A 1D aerodynamic analysis for preliminary design optimisation and performance prediction, including maps, was performed. This was followed by a 3D analysis, which included blade generation. A 3D flow field analysis using CFD was performed (fig. 13), followed by stress and modal analysis using finite element methods (fig 12).

Fig. 12 Turbine FEA Stress Contour Plot Fig. 13 Turbine CFD Flow Field

Modifications to fillet radii, blade thickness distribution and back face led to a final rotor design with stresses below the limits.

After the preliminary assessment of the structure the aerodynamics were further refined. The turbine scroll and nozzle designs were completed. A scroll area schedule was developed and translated into a solid model of the flow volute, see fig. 15. Two nozzle vane geometries were developed: flat (uncambered) vanes (fig. 14) and curved vanes. Aerodynamics analysis of the two vane types indicated little difference in performance. Therefore, the flat vane design was chosen because of its simpler geometry and compatibility with possible future use of a variable geometry turbine (VGT).

Fig. 14 Turbine Rotor and Nozzle Fig. 15 Turbine Inlet Flow Volute Geometry

The dominant aero design goal was high efficiency. Turbine efficiencies exceeding 85% (total-to-static) were predicted. Considering the size of the stage this is an excellent figure.

Performance Prediction

Once the aerodynamic design of the wheels was complete and the predicted performance maps were developed, the complete turbo compound system, consisting of engine, ETC turbo with compressor and turbine maps, control unit and crank motor/generator, was simulated with a Caterpillar cycle simulation tool. Target application for this system is the Caterpillar C-15 heavy-duty on-highway truck engine.

Engine test data of a model year 2000 engine were used for model calibration and reference. Cycle simulation focused on five typical road load conditions plus the rated point. Each operating point was calibrated against measured engine data. On the electrical side of the system, an 85% efficient power conversion from the turbo shaft generator to the crankshaft motor including power electronics was assumed.

Fig. 16 shows a typical characteristic of BSFC vs. compound power. At this operating point approximately 15% of the total engine power comes from the turbo shaft. Increasing the turbo compound power, i.e. extracting more power from the turbo shaft generator, improves engine fuel economy. By doing that the exhaust waste heat is recovered. Beyond the optimum power extraction the engine boost pressure is still decreasing. This ultimately leads to poor combustion efficiency, which offsets the turbo compound benefit. Fig. 16 also shows that the boost pressure is directly dependent on the amount of extracted power; the implication of this behavior is:

• By varying the extracted power A/F ratio can be controlled
• Boost signal is ideal for controlling the ETC system

Fig. 16 Boost pressure, turbine inlet pressure and bsfc at rated power

The engine simulation of the complete ETC system showed an encouraging potential for fuel consumption. At rated power fuel consumption can be reduced by 10%. Considering a typical road load for an on-highway truck with different weighting factors for the prevailing operating speeds and loads, fuel consumption is reduced by approximately 5%. An important result of the engine simulation was that the turbo shaft generator specifications were relaxed. Whereas in the concept phase it was estimated that the power requirements would be 60kw continuous and 70kw intermittent, the simulation showed that optimum BSFC could be achieved with generator power levels of 40kw continuous and 60kw intermittent.

Air Handling System - Shaft System

Analysis from the Concept Design phase indicated that the third critical speed of the conceptual shaft design was within the turbo speed range. Revised turbo shaft generator specifications discussed in the previous section allowed for a shorter generator length, which in turn alleviated rotor dynamics. The first two critical speeds were moved below the typical operating speed range and the third critical was moved above the maximum shaft speed.

Angular contact ball bearings were chosen because of their axial and radial load capability; ceramic balls were chosen for increased stiffness and higher lifetime. After having determined the geometry of compressor, turbine, generator, and shaft system an overall Pro-Engineer model was created, see fig. 17.

Fig. 17 Pro-Engineer Model of the Electric Turbocompound Assembly

Turbo Motor/Generator

The switched reluctance machine, chosen in the Concept Design phase, was further refined. The revised turbo shaft generator specifications allowed for a shorter rotor length. The electrical capabilities of the machine were changed to rated power of 40 kW for continuous operation, and 60 kW for peak-power/intermittent operation. This relaxation in generator specifications did not have a significant effect on electrical machinery efficiency. Since the reduction in length, and thus surface area for heat dissipation, was approximately equal to the reduction in power, the thermal management of the generator was not affected. As a precaution, the generator was equipped with embedded thermal sensors. A thermal derate strategy was implemented as part of the overall control system. A three-phase configuration with 6 stator poles and 4 rotor poles was selected for the final design, see fig. 18.

Fig. 18 Preliminary Design of Rotor and Stator

Cooling
Ports

Case

Stato

Windings

Rotor

The rotor lamination geometry is a compromise between power requirement, efficiency and centrifugal stress. It was found that a larger rotor OD might offer higher efficiency, but at the same time centrifugal forces increase. Once the overall rotor dimensions were finalized, a centrifugal stress analysis was completed. Calculated stress levels for design speed and 20% overspeed were acceptable, see fig. 19.

Fig. 19 Rotor Stress Contour Plot

Due to the changes in power requirement and geometry a heat transfer analysis was conducted. Heat management and interaction between rotor and stator were examined with a FEA model of the turbo shaft generator. The maximum allowable temperature in the stator coil is 180 deg C for continuous operation and 205 deg C for intermittent operation. The corresponding generator power was 40 kW continuous and 60 kW intermittent operation. The FE model showed temperatures below the limit for both operating conditions. The housing was provided with a separate air path and air in and out connections for additional cooling.

Air Handling System - Overall Assembly Modeling

Thorough FE analyses of the turbocharger structure was carried out at both steady state and transient conditions. The main task was to keep thermal gradients, and thus stress gradients, low and at the same time provide enough cooling for the bearings and motor/generator. Also, for turbo performance reasons, the heat at turbine inlet should be retained. In addition to that, care had to be taken of thermal growth during operation. All requirements were met by decoupling the turbine housing and turbine back plate from the center housing.

The final structural analysis consisted of a combined heat transfer and stress analysis of the complete turbocharger structure for steady state and transient conditions; the steady state results are shown in fig. 20. In a final check the model was run under transient conditions, i.e. simulated engine start from idle to peak torque and hot shut down. All stresses and deflections were acceptable.

Fig. 20 Stresses at Rated Speed and Steady State

Crank Starter/Generator/Motor

The design of the crank starter/motor/generator was completed. A continuous rating of 40 kW at 1,500 rpm, and 60 kW at 2,100 rpm for peak power were chosen.

The motor was designed so that it was contained within the engine flywheel housing. In order to integrate the motor to the driveline (in between the engine and the transmission) a new flywheel housing was designed and fabricated.

Electronic Systems Integration and Controls

The simulation model was used to map the required boost at different engine operating conditions. Fig. 21 shows the result for various operating points in turbo compound mode. With such a map the ETC test engine will operate under optimal settings.

Fig. 21 Boost Map vs. Engine Speed and Load

A control strategy was developed which uses boost as the main parameter; turbo speed as an input to the system to assure rotational speed is within set limits. This is a particularly important feature since a failure of the electronic system could result in overspeeding the turbocharger.

The block diagram in fig. 22 shows the basic control principle. Boost pressure is the primary control signal; depending on engine speed and load a certain amount of power is being extracted from the turbo shaft generator that has a direct impact on the boost level.

Fig. 22 Electric Turbocompound Controller Block Diagram

The ETC has its own control system, which takes care of the two electric machines and the power flux. Output from the Electric Turbocompound Controller is used to set the power levels of the turbo generator and crank motor. The engine keeps its engine control module. Fig. 23 shows how the two controllers are connected.

Date: 17 December 04 16 Caterpillar, Engine Research DOE Final Report 2004 DE-FC05-00OR22810

Fig. 23 Motor and Generator Control Block Diagram (Caterpillar Confidential)

2. Procure, Build and Test Electrical Machinery

2.1 Crank Starter/Generator/Motor

The crankshaft motor electrical testing and characterization were carried out at 340 Vdc in the motoring and generating mode. Fig. 24 shows the crankshaft M/G in the flywheel housing attached to the test engine. Fig. 25 shows the assembled machine before testing on a separate rig.

Fig. 24 Flywheel housing with Crank Shaft M/G Fig. 25 340 V Crank Shaft Machine

The target figures for power, 60 kW for motoring and 40 kW for generating, were met. Tests were carried between 0 and 2400 rpm and at loads between 0 and 450 Nm; measured maximum temperatures were 65 deg C. Peak efficiencies in motoring mode exceeding 90% were measured.

Date: 17 December 04 17 Caterpillar, Engine Research DOE Final Report 2004 DE-FC05-00OR22810

2.2 Turbo Motor/Generator

Fig. 26 shows stator frame, lamination, rotor and encapsulated motor/generator with sealed wire exits. The stator was fitted with 6 thermocouples to monitor temperature during operation. One test unit was used for the characterization on a test rig; the second unit was part of the ETC turbo.

Fig. 26 Parts and Assembly of Turbo Shaft Motor/Generator

The motor/generator was tested on the dynamometer rig. These tests characterized the machine, which enabled an optimal setting of the machine for each operating point. Because of rig limitations a speed range between 0 and 30,000 rpm was covered. This corresponds to a power of up to 30 kW. The measurements were used to calibrate a model, which was used for a virtual characterization for shaft speeds above 30,000 rpm. Tests were carried out in motoring and generating mode. The maximum temperature of the stator-mounted thermocouples was 120 deg C, well below the limit of 180 deg C. The measured peak efficiency on the rig was 80.5 %. The simulation tool for the electrical machine was calibrated with these measurements. Predicted efficiencies for higher loads and speeds were between 85-90%, thus the measured values were slightly below the goal for this electrical machine.

2.3 Impact on Overall Turbocompound System

The original cycle simulations for the overall turbocompound system were done with predicted efficiencies for the electrical machines and power electronics. The predicted efficiency per electrical machine was 92%, giving a system efficiency from turbo shaft to crank shaft of 85%.

The measured turbo shaft M/G efficiency was below target, as stated above. Rig test results of the turbo shaft machine with air cooling were used to re-calibrate the analysis tools. With that the best point of the electrical system was predicted to be at 80.5%. M/G efficiency is critical to realizing the engine system efficiency targets. Translated into engine fuel economy, the theoretical improvement in fuel consumption is between 2 and 3% instead of the originally predicted 5%.

The development team identified a number of options to improve the efficiency, which are discussed in the following section.

Turbo Motor/Generator Improvements

A study was carried out to carefully review the design of the high-speed turbo-compound generator and identify potential improvements that could contribute towards a higher overall efficiency. This review was to be constrained to use the same physical size as the original prototype but any other design parameter could be changed.

Key elements that were studied in detail:

• Basic topology
• Core material selection
• Winding losses
• Windage losses
• Control Strategy
• Lamination geometry

An improved design was conceived and analyzed. An improvement in generator efficiency of 3 to 5 percentage points was predicted. This new design path would be pursued prior to moving this system to production.

3. Procurement Turbo Machinery and System Hardware

Fig. 27 shows the main components of turbomachinery hardware.

Fig. 27 Turbocharger Hardware

4. Turbocharger Assembly

Turbocharger assembly consisted of parts inspection, minor remachining and documentation of tolerances, see fig. 28.

Fig. 28 Inspection and Assembly of Turbocharger

Due to the longer shaft and the increased number of parts the assembly requires much more attention compared to a conventional turbocharger. Several trial assemblies were necessary to insure proper fit of parts. The rotor assembly with the bracket for balancing can be seen in fig. 29. Because of the high speed of the machine the complete rotor was assembly balanced.

Fig. 29 Rotor Assembly of Turbocharger with Balancing Bracket

5. Build Rigs

5.1 Gas Stand - Turbocharger

The ETC turbocharger was prepared for gas-stand testing, with connections to an oil circuit for lube oil supply and drain, to a water circuit for the stator frame, and to shop air for air-cooling inside the motor/generator. Fig. 30 shows the prepared turbocharger in detail.

Fig. 30 Assembled Turbocharger

Views of the ETC turbo installed on the gas stand are shown in figures 31 and 32.

Fig. 31 Turbo with Oil, Water, Air Connections Fig. 32 Turbo on Gas Stand

6. Test Turbocharger – ETC Rig Test

The test program for the turbocharger on the gas stand was as follows:

A. Rotor dynamics test
B. Aerodynamic test – compressor map
Date: 17 December 04 21 Caterpillar, Engine Research DOE Final Report 2004 DE-FC05-00OR22810
C. Aerodynamic test – turbine map
D. Characterization of motor/generator

A. Rotor dynamics test

The initial rotor dynamics test was carried out with two proximity probes in the compressor inlet to measure radial excursions. The results are shown in Fig. 33: The measured critical speed was 16,000 rpm. At higher speeds the radial excursions are exceptionally small. Another indication of high shaft stability is the fact that the measured orbits at the compressor end are almost circular for all shaft speeds up to the max. allowable shaft speed of 66,000 rpm (see fig. 34).

Fig. 33 Radial Excursions of Turbo Shaft at Compressor End

Fig. 34 Turbo Shaft Orbit Measured at Compressor End

Date: 17 December 04 22 Caterpillar, Engine Research DOE Final Report 2004 DE-FC05-00OR22810

B. Aerodynamic test – compressor map

The compressor was fully mapped. The measured peak efficiency was 82%, which is 3% points below the prediction. The difference becomes larger at high pressure ratios. The discrepancy can be attributed to a larger than targeted clearance at the compressor exit. The final clearances are a result of a complicated force balance between the aerodynamic thrust loads and the preload built into the bearing/shaft system. Both areas contributed to the shift from the targeted clearance. The discrepancy would be easily correctable with a design change in the next generation unit.

C. Aerodynamic test – turbine map

Turbine performance was measured during gas stand testing. The measured flow vs. expansion ratio was in good agreement with the predictions. Due to the cooler running temperatures of the turbine during the mapping exercise, the efficiencies obtained were in excess of predicted values. For this mapping technique, lack of instrumentation and running the turbine at cooler temperatures most likely contributed to the large differences in these efficiencies

D. Characterization of motor/generator

The characterization of the turbo shaft M/G was carried out in two steps: up to 30,000 rpm on a separate test rig for the M/G only and between 30,000 and 60,000rpm on the gas stand for the complete ETC turbocharger. The tests confirmed the virtual (predicted) characterization.

The results of the characterization were as follows:

o Turbo shaft M/G generates power
o Highest measured power was 44 kW at 58,600 rpm
o Highest recorded speed while generating power was 61,750rpm
o Generator demonstrated the required power output at the five tank mileage points
o Measurements were very close to virtual characterization
o ETC controller and CAN bus worked flawlessly

The measured power of the turbo shaft generator is shown in fig. 35. 0 5000 10000 15000 20000 25000 3000025000 30000 35000 40000 45000 50000 55000 60000 65000 Turbo Shaft Speed rpm Generator Power W 010 20 30 40 50 60 Elecronics O/ P power (W) % Torque

Fig.35 Measured Power of Turbo Shaft Generator

Tests results show that all operating conditions up to the M/G power of approximately 15 kW can be run under steady state conditions. This means about three of the five engine operating points can be measured and run steady state. For higher generator power it was determined that the machine will overheat. Despite the water jacket around the stator frame and the air-cooling between rotor and stator heat buildup was higher than anticipated. This was due to the higher than predicted electrical losses in the generator and greater than anticipated heat flux from the hot exhaust gases and turbine. As a result, the generator can only run for limited time when greater than 15kW is extracted. As an example, the demonstrated high power point of 44 kW was run for approximately 30 to 60 seconds and then load was removed to let the machine cool down.

7. Overall System Integration and Engine Performance

Prior to engine testing of turbocompounding the subsystems were integrated. The turbocharger with the ETC controller and turbo shaft generator had proved its functionality. The crankshaft M/G had been successfully tested on a test rig and the control system had been lab tested. These subsystems were integrated and mounted onto the engine in the test cell (see fig 36).

Fig.36 ETC Turbocharger Mounted on Engine

ETC Engine Testing

The baseline C15 production engine was tested at 13 engine operating points. After that, the engine was converted into ETC configuration with the ETC turbocharger.

Although the turbo ran already about 50 hours on the gas stand, preliminary engine testing was conducted with great care. Turbo operation was monitored during several engine idling phases with system checks in between.

The engine was tested at an engine speed of 1500 rpm at 25% and 50% load. For these operating points the engine ran in turbocompounding mode, power was generated at the turbo shaft and recovered with motor on the crankshaft. Since the focus was on the electronics and controls system no engine performance data were taken. During these tests a turbocharger bearing failure occurred and prevented further testing. Cause of the failure was believed to be due to rotor imbalance, possibly caused by a shifting of one of the rotor endplates.

Conclusion

An electric turbocompound (ETC) system, including a novel turbocharger with an electric motor/generator (M/G), has been analyzed, designed and tested. During the course of the research program many obstacles like the high speed M/G on the turbo shaft were overcome. The ETC turbocharger and ETC control system were successfully tested on the gas stand; the turbo shaft M/G generated up to 44kW of electrical power at 58,600 rpm; the highest recorded speed while generating power was 61,750rpm. This covers the required speed and power range to impact the truck engine's fuel economy and the five tank mileage points.

The complete ETC system was successfully tested on a lab engine where exhaust energy was recovered with the generator on the turbo shaft and converted to shaft power with an electrical motor on the crankshaft.

The program successfully demonstrated that the technology can be developed and can operate on a class 8 on-high truck engine. The primary goal of a technology demonstrator had been accomplished. A path to higher turbo shaft generator efficiency had been identified and should be implemented prior to production. Follow on work is needed in the area of shaft/bearing system reliability.

Ulrich Hopmann Rich Kruiswyk
Program Manager Air Systems Technical Lead
Caterpillar Inc. Caterpillar Inc.