Final Technical Report

High Efficiency Engine Technologies Program

**An Engine System Approach to
Exhaust Waste Heat Recovery**

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High Efficiency Engine Technologies Program

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Executive Summary

Caterpillar's Product Development and Global Technology Division carried out a research program on waste heat recovery with support from DOE (Department of Energy) and the DOE National Energy Technology Laboratory. The objective of the program was to develop a new air management and exhaust energy recovery system that would demonstrate a minimum 10% improvement in thermal efficiency over a base heavy-duty on-highway diesel truck engine. The base engine for this program was a 2007 C15 15.2L series-turbocharged on-highway truck engine with a LPL (low-pressure loop) exhaust recirculation system.

The focus of the program was on the development of high efficiency turbomachinery and a high efficiency turbocompound waste heat recovery system. The focus of each area of development was as follows:

- 1) For turbine stages, the focus was on investigation and development of technologies that would improve on-engine exhaust energy utilization compared to the conventional radial turbines in widespread use today.
- 2) For compressor stages, the focus was on investigating compressor wheel design parameters beyond the range typically utilized in production, to determine the potential efficiency benefits thereof.
- 3) For turbocompound, the focus was on the development of a robust bearing system that would provide higher bearing efficiencies compared to systems used in turbocompound power turbines in production.

None of the turbocharger technologies investigated involved addition of moving parts, actuators, or exotic materials, thereby increasing the likelihood of a favorable cost-value tradeoff for each technology. And the turbocompound system requires less hardware addition than competing bottoming cycle technologies, making it a more attractive solution from a cost and packaging standpoint.

Main outcomes of the program are as follows:

1. Two turbine technologies that demonstrated up to 6% improvement in turbine efficiency on gas stand and 1-3% improvement in thermal efficiency in on-engine testing.
2. A compressor technology that demonstrated 1.5% improvement in compressor efficiency on gas stand compared to production available compressors.
3. A power turbine with high efficiency bearing system that demonstrated excellent rotordynamic stability throughout the required speed range, up to 60,000 rpm
4. A predicted improvement (using engine simulation) in engine thermal efficiency of 7% at the peak torque design point, when combining the technologies developed in this program.

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1. Introduction

1.1 Program Background

On 24May2004 the United States Department of Energy released an announcement of funding opportunity for “Energy Efficiency Renewable Energy (EERE) High Efficiency Clean Combustion and Waste Heat Recovery for Internal Combustion Engines, DE-PS26-04NT42099-00”. On 9August2004 Caterpillar Inc. submitted a proposal for a program of work in response to ‘area of interest 2’ of that DOE announcement, which was Exhaust Energy Recovery (DE-PS26_04NT42099-02). The Caterpillar proposal was accepted, and DOE contract DE-FC26-05NT42423 was awarded, to begin on 1August2005. The title of the program was “An Engine System Approach to Exhaust Waste Heat Recovery”.

1.2 Program Objectives

The objective of this program was to develop a new air management and exhaust energy recovery system that would demonstrate a minimum 10% improvement in thermal efficiency over a base heavy-duty on-highway diesel truck engine. The developed system was to deliver this efficiency improvement with the following requirements:

- No increase in emissions rate
- No reduction in engine power density
- Compatible with aftertreatment requirements for EPA 2007 and 2010 on-highway emissions regulations.

The emphasis of the program was to develop cutting-edge technologies that meet demanding cost, packaging, and driveability (response) requirements. The base engine would be a 2007 C15 15.2L on-highway truck engine with series turbocharging and low-pressure-loop (LPL) exhaust recirculation system.

This program supports the DOE / NETL goals of decreasing our nation’s dependency on imported petroleum. The amount of diesel fuel consumed by on-highway trucks represents 20% of the total US surface transportation petroleum energy consumption. In addition, fuel economy is a major factor in the total owning and operating costs of these trucks. Therefore, small improvements in diesel engine fuel efficiency will dramatically affect both our nation’s total consumption of fuel and the profit margins of fleet owners. Approximately 20% of the fuel’s energy availability is lost in the exhaust of today’s diesel engines. Consequently, recovering energy from the diesel engine exhaust is a technological approach that should be able to provide a 10 percent reduction in engine fuel consumption. If one assumes a \$3.00/gallon fuel cost, the 10% efficiency improvement represents a staggering \$10.2 billion annual savings in fuel cost! The higher efficiency engines will also be more desirable to the customer and cause a faster than normal fleet turnover. The fast fleet turnover will replace the older, higher emitting engines with the new, lower emitting, engines.

1.3 Program Timeline / Funding

This program was initially scoped as a 5-phase, 4-year 5-month program (1Aug2005 to 31Dec2009). The high-level program plan was as follows:

- Analyze candidate technologies to define the recipe to achieve 10% engine thermal efficiency improvement (Phase I)
- Design components for focused testing to validate the technology recipe (Phase II)
- Procure components and conduct focused bench and gas stand testing to validate the technical approach (Phase II and Phase III)
- Refine the designs and the technical recipe as needed based on component test results and refined analyses (Phase III and Phase IV)
- Demo the final system on-engine (Phase V)

Delays in availability of funds early in the program resulted in extensions of the early phases, and the program was ultimately terminated at the end of Phase3 in conjunction with the commencement of the DOE Supertruck programs. The original and final program timelines are shown in Figure 1:

Figure 1. Original and Final Program Timeline and Phases

The final timeline and funding for this program were therefore as follows:

Budget Period / Phase	Time Period	Cat Share	DOE Share	Total Funding
BP1, Phase 1,2	8/1/05 – 11/30/07	\$879,800	\$875,346	\$1,755,146
BP2, Phase 3	12/1/07 – 7/31/10	\$1,293,717	\$1,298,152	\$2,591,869
Total		\$2,173,517	\$2,173,498	\$4,347,015

2. Technical Approach

The goal of a 10% improvement in engine thermal efficiency via improvements in exhaust energy utilization was aggressive, and in the opinion of the Caterpillar team would require improvement in all aspects of exhaust energy handling and utilization. So a systems solution was anticipated.

The base engine for this project was a Caterpillar C15 15.2L on-highway truck engine with series turbocharging and low-pressure-loop (LPL) exhaust recirculation system, as shown schematically in Figure 2.

Figure 2. Baseline C15 On-Highway Truck Engine, LPL Configuration

The peak torque operating point (1200rpm, full load) was selected as the design point on which the system would be evaluated. An availability analysis was conducted on the energy exiting the cylinders at this peak torque condition, and the results are shown in figure 3:

Figure 3. Availability Analysis of Energy Leaving Cylinder at Peak Torque

This figure shows that roughly 1/3 of the available energy leaving the ports is used to produce boost, roughly 1/3 is lost via heat transfer or irreversibilities, and roughly 1/3 exits via the exhaust stack. Therefore, a solution to maximize thermal efficiency through improved utilization of exhaust energy will have to minimize losses in all components as well as recover stack energy.

Extensive sensitivity analysis was conducted using engine simulation to determine the path to 10% thermal efficiency that would balance the technical risk equally amongst the components of the solution, and thereby maximize the chances of program success. The technical recipe that was developed is shown in Figure 4:

Figure 4. Technical Recipe for Achieving 10% Improvement in Thermal Efficiency

In Figure 4, the numbers in parenthesis are the expected thermal efficiency improvement contributions from each element of the solution – together they comprise a 10% improvement solution.

Four of these seven solution elements were worked in the first 3 phases of this program: the HP (high pressure) turbine stage, the LP (low pressure) turbine stage, the compressor stages, and the stack recovery. Together these four elements comprise nearly 80% of the solution, i.e. 7.7% out of the 10% improvement goal. The intercooling and port insulation elements were developed in a previous program and would be simply applied to this program. The piping element was scheduled for development in Phases 4 and 5 of this program, when all of the solution elements would be brought together for an on-engine demonstration. As mentioned in section 1, this program was terminated prior to entering Phases 4 and 5, so the piping element of the solution was never developed.

The developments undertaken to meet the goals outlined in Figure 4 for the HP turbine stage, LP turbine stage, compressor stages, and stack recovery elements are described in Section 3 below.

3. Technology Developments

3.1 HP Turbine Developments

As shown in Figure 4, improvements to the HP turbine stage were targeted to produce a 2% improvement in engine thermal efficiency. This was a very aggressive goal. According to engine simulation sensitivity studies, a 2% improvement in engine thermal efficiency would require roughly a 10% improvement in turbine stage efficiency. Alternately, a 2% improvement in engine thermal efficiency would require roughly an 8% improvement in turbine stage efficiency along with an improvement in exhaust pulse energy utilization. This is explained in the following paragraphs.

On 6-cylinder engines with divided exhaust manifolds, much of the exhaust energy is concentrated in the ‘pulses’ that result from the initial opening of the exhaust valve and the release of high-temperature, high-pressure exhaust gasses into the manifold. Today’s conventional radial turbines are not ideally suited to capture the energy in these exhaust pulses, the reasons for which can be explained by looking at typical radial turbine efficiency contours.

A ‘typical’ radial turbine efficiency map is shown in Figure 5, where turbine efficiency is plotted on the vertical axis and blade speed ratio is plotted on the horizontal axis. Blade speed ratio is simply a non-dimensionalized turbine tip speed. As shown in the figure, the radial turbines commonly used in production turbochargers typically have their peak turbine efficiency at a U/C of 0.7. However, the peak on-engine exhaust energy that is contained in the exhaust pulses occurs in a lower U/C range, where the turbine efficiency is starting to drop off. If the turbine peak efficiency could be shifted to lower U/C, even without an increase in the peak efficiency value, an improvement in exhaust energy utilization and engine thermal efficiency would result.

Therefore, the goal for the HP turbine stage was set at an 8% improvement in peak turbine efficiency, with the efficiency peak shifted to lower U/C for improved on-engine exhaust pulse utilization. To achieve this level of improvement, two technologies were explored and developed: the RND (radial, nozzled, divided) turbine and the mixed flow turbine.

Figure 5. Typical Radial Turbine Map

3.1.1 RND Turbine

The concept of the RND (radial, nozzled, divided) turbine was initially explored as part of the Caterpillar – DOE HTCD (Heavy-Truck Clean Diesel) program. The idea was to combine three key elements into one turbine stage:

- A divided turbine housing. This is important for retaining a fully-divided exhaust system on 6-cylinder engines, which helps to preserve exhaust pulse energy
- A nozzled turbine inlet. This is important for providing uniform inflow conditions to the turbine wheel and thereby maximizing efficiency of the wheel
- A high efficiency turbine wheel design

These three elements are not combined in production turbochargers of this size, due primarily to the challenges of designing a robust divided and nozzled turbine housing. On the HTCD program, the RND concept was evaluated using individual nozzle vanes inserted into a modified, divided turbine housing as shown in Figure 6.

Figure 6. Concept Version of RND Turbine from HTCD Program

For this program, the goal was to move the concept toward production viability and further optimize the performance.

To that end, the individual nozzle vanes were replaced with a single-piece cast nozzle ring which was inserted into a specially designed turbine housing. Gas stand results for this Gen2 version of the RND are shown in Figure 7, in comparison to the baseline production radial turbine. The turbine efficiency is plotted on the vertical axis.

Figure 7. Gas Stand Results – Gen2 RND vs Baseline Radial Turbine

The results in Figure 7 show that roughly a 6% improvement in peak efficiency relative to the base turbine was achieved, or approximately 3/4 of the targeted 8% improvement. In addition, there has been some shift in the peak efficiency to lower U/C, though not quite as much as desired for peak exhaust pulse utilization.

This Gen2 turbine stage was also tested on the dynamometer gas stand at Imperial College in London in order to get full-width (wider U/C range) turbine maps, to verify that the efficiency improvements were not lost at higher or lower U/C values. These results are shown in Figure 8. Again a 5-6% improvement in turbine efficiency was demonstrated, and this improvement held throughout the full U/C range.

The RND turbine was also tested on-engine. An engine in the base configuration (series turbocharged, LPL recirculation) was not available in a test cell at the time of testing, so the RND was tested on a single-stage turbocharged C15 with HPL (high-pressure-loop) EGR system. The resulting fuel economy improvements are shown in Figure 9. Because the efficiency benefit of the RND is applied to all the boost generation in a single-stage turbocharged engine (as opposed to being applied to only a portion of the boost generation when used as the HP stage in a series turbocharged engine), the thermal efficiency gains were quite high on this engine ; 3% at the peak torque design point.

Response testing was also performed on this engine and is also shown in Figure 9. Roughly a 30% improvement in response was achieved at all engine speeds, with lower smoke. The inertias of the rotating assemblies were virtually identical between

Figure 8. Imperial College Gas Stand Results – Gen2 RND vs. Baseline Radial Turbine

Figure 9. BSFC and Response Test Results – RND vs. Baseline Turbo on C15 Single-Stage Turbocharged Engine

the RND turbine and the base turbine, as were the turbine swallowing capacity curves. Therefore, the response benefit can be attributed almost entirely to the improved efficiency of the turbine stage.

As impressive as these results are for the RND turbine, it still falls short of the aggressive goal of an 8% improvement in peak efficiency with the peak shifted to lower U/C. Therefore the team investigated a second turbine technology, the mixed flow turbine.

3.1.2 Mixed Flow Turbine

A mixed flow turbine is a turbine in which the flow into the turbine wheel is neither radial nor axial, but in-between, as shown schematically in Figure 10.

Figure 10: Mixed Flow vs. Radial Flow Turbine Wheels

The mixed flow turbine wheel offers several efficiency advantages relative to the radial turbine wheel. First, because the flow does not have to turn a full 90 degrees in the meridional plane there are fewer flow losses which means the mixed flow can achieve a higher peak efficiency. Secondly, the mixed flow turbine can have positive inlet blade angle while retaining radial blade fibers. Positive inlet blade angle means the leading edge of the wheel can be angled in the direction of flow rather than being straight out from the axis of rotation, and radial blade fibers are necessary to keep blade root stresses down for long wheel life. The reason for wanting a positive inlet blade angle is that it has the desired effect of moving the turbine peak efficiency to lower U/C for effective exhaust pulse utilization. Further explanation of these effects can be found in turbo texts and technical papers on mixed flow turbines.

While the effect of using a positive inlet blade angle to shift peak efficiency to lower U/C has been extensively documented on the gas stand, relatively little has been published showing the resulting positive fuel economy benefit of this change on 6-cylinder engines with divided exhaust systems. Demonstrating these on-engine benefits was the focus of this program.

Two generations of mixed flow turbine stages were designed, procured, and tested, and the gas stand results are shown in Figure 11. The first design, Gen1, is shown in red. While the peak turbine efficiency was successfully shifted to lower U/C in this design, the peak efficiency value was 2-3% lower than that of the baseline turbine. This is attributed to inexperience in using the design codes to develop mixed flow wheels and to being overly aggressive in the selection of the positive inlet blade angle. These results led to the development of the second design, Gen2, which is shown in green. For this design, a less aggressive positive inlet blade angle was used, and the result is an improvement in peak efficiency of about 2-3% relative to the baseline turbo, as well as successfully shifting the peak efficiency to lower U/C.

The Gen2 mixed flow turbine was also tested on the dynamometer gas stand at Imperial College to get full-width maps, and the result compared to the baseline radial turbine is shown in Figure 12. While the mixed flow turbine has lower efficiency at U/C values greater than approximately 0.7, this region is relatively unimportant for engine efficiency because most of the exhaust energy is contained in the pulses that occur at U/C below 0.7. In this region the mixed flow turbine has efficiencies up to 5% higher than the baseline turbine.

Figure 11. Gas Stand Test Results – Mixed Flow Gen1 and Gen2 Turbines

Figure 12. Imperial College Gas Stand Test Results – Mixed Flow Gen2 Turbine

The Gen2 mixed flow turbine was tested on the base C15 engine configuration (series turbocharged, LPL configuration) as the HP turbine stage, and the results are shown in Figure 13. A 1.3% fuel economy benefit was achieved at the peak torque design point. This is more than would have been expected from simply a 2-3% increase in peak turbine efficiency, and this is evidence that improved exhaust pulse energy utilization has been achieved due to the shifting of the peak turbine efficiency to lower U/C.

**Figure 13. Engine Test Results – C15 On-Engine Test of Mixed Flow Turbine
BSFC Benefit Relative to Baseline Radial Turbine**

From the above discussions, it is clear that neither the RND turbine nor the mixed flow turbine met the goals of 8% improvement in peak efficiency and efficiency peak shifted to lower U/C. This is shown clearly in Figure 14 below.

Figure 14. Gas Stand Results of RND and Mixed Flow Turbines Compared to Target

However, one can see that both technologies brought significant benefits, and that a combination of the technologies may yield a turbine stage that meets the target. Thus the idea of a mixed-flow, nozzled, divided (MND) turbine can be envisioned. Replacing the radial wheel in the RND with a mixed flow wheel should bring an increase in peak efficiency and a shift in the peak efficiency to lower U/C, i.e. should move the red curve closer to the purple target. The concept development and preliminary design work for the MND turbine stage was started late in the program; a preliminary picture of the concept is shown in Figure 15.

Figure 15. Mixed-Flow, Nozzled, Divided Turbine Stage Concept (divider wall not shown)

3.2 LP Turbine Technologies

As shown in Figure 4, improvements to the LP turbine stage were targeted to produce a 1% improvement in engine thermal efficiency. According to engine simulation sensitivity studies, a 1% improvement in engine thermal efficiency would require roughly a 6% improvement in turbine stage efficiency relative to the baseline radial turbine. Two technologies were considered to try to achieve this turbine efficiency improvement: 1) an axial turbine stage, and 2) a nozzled radial turbine stage.

3.2.1 Axial LP Turbine Stage

An axial turbine stage can provide higher peak turbine efficiency than a radial stage because the turning of the flow in the meridional plane is eliminated. So confidence was high that an axial stage could be developed to meet the efficiency targets of this program.

An axial turbine stage was designed and analyzed to meet the flow and efficiency requirements of this program, see Figure 16. Analysis results indicated that the 6% improvement in turbine efficiency relative to the base turbo could be achieved. However, packaging an axial LP turbine between a radial HP turbine and a radial or axial turbocompound power turbine was considered extremely challenging due to the sizeable collector required for the axial turbine. As a result, the decision was made to evaluate the ability to meet the LP turbine performance requirements with a more easily packaged radial turbine stage.

Figure 16. Axial LP Turbine Stage – Concept Design

3.2.2 Radial LP Turbine Stage

In a series turbocharged engine, the response of the air system is dominated by the HP stage, with relatively little impact on response from the LP stage. Therefore, when designing an LP turbine stage for a series turbocharged engine, the focus of the wheel design should be on efficiency, with less regard or concern for the resulting wheel inertia. This was the approach taken with the radial turbine design for this program.

A high efficiency turbine wheel was designed with a vaned nozzle inlet to reach maximum possible efficiency for a radial turbine stage in this application, see Figure 17. The length/diameter ratio of the turbine wheel was greater than typically found in production radial turbines in order to allow for more gradual turning of the exhaust flow and higher efficiency.

Figure 17. Radial LP Turbine Stage

Gas stand results for the radial LP turbine stage are shown in Figure 18. A 3.5 to 4% improvement in peak turbine efficiency was achieved relative to the base radial turbine. This falls short of the 6% improvement goal and the improvement predicted from the axial turbine stage. But the 2% shortfall in efficiency must be weighed against the improved packaging that results with a radial LP turbine. Caterpillar has already successfully demonstrated the ability to package series turbochargers with radial turbines on both HP and LP stages with the 2007 C15 on-highway truck engine, see Figure 19. Therefore, the radial LP turbine stage was selected as prime-path to carry into the final system solution for this program.

Figure 18. Radial LP Turbine Gas Stand Results, Turbine Efficiency

Figure 19. Caterpillar 2007 C15 On-Highway Truck Engine

3.3 Compressor Technologies

In a series turbocharged application, each compressor stage is only carrying a portion of the total engine boost requirement. In production, the compressors used for series turbocharged applications are typically designs that were originally created for single-stage applications. Designing a compressor specifically for a series turbocharged application will open up the possibility to be more aggressive with certain design parameters to achieve high efficiency at the expense of overall boost pressure capability. This was the approach pursued for this project.

As shown in figure 4, improvements to the compressor stages were targeted to produce a 0.7% improvement in engine thermal efficiency. Achieving this would require a 2.5% improvement in compressor efficiency for both the HP and LP compressor stages, based on engine simulation sensitivity analysis. The focus of the first three phases of this program was on improving the HP compressor stage.

To achieve high efficiency on the HP compressor stage, a highly backswept compressor wheel was designed with a matched low-solidity vaned diffuser. High blade backsweep can be an effective means to improve efficiency in a compressor stage at the expense of boost pressure – for a series turbocharged application this reduction in boost pressure in each stage can be tolerated. The blade backsweep angle used for this design exceeded that conventionally used in production compressors. The HP compressor stage that was designed and tested is shown in Figure 20, and gas stand test results are shown in Figure 21. A peak efficiency of 82% was measured, with a broad flow range of high efficiency islands. This peak efficiency is 1 to 1.5% higher than the baseline compressor stage, short of the 2.5% improvement goal. Further improvements are anticipated in Gen2 refinements of the design.

No work was completed on the LP compressor stage in the first three phases of the project. The plan was to apply lessons learned on the HP stage to improve the LP stage during later phases of the project.

Figure 20. High Efficiency HP Compressor Stage

Figure 21. Gas Stand Map for High Efficiency HP Compressor Stage

3.4 Stack Energy Recovery Technologies

As shown in Figure 4, the recovery of exhaust stack energy is the single biggest element of this system solution, expected to provide for 4% engine thermal efficiency improvement, or 40% of the total system improvement.

There are a number of ways to recover the thermal energy available in the engine exhaust stack, and two technology ‘groups’ were considered under this program. First, bottoming cycle technologies, where the exhaust stack energy is captured via a heat exchanger and transferred to a thermodynamic cycle for conversion of the energy into a useful form. Second, turbocompound technologies, where the exhaust stack energy is captured via a turbine, which typically then feeds the energy back into the engine crankshaft. Schematics of these two systems for a single-stage turbocharged engine are shown in Figure 22.

Figure 22. Bottoming Cycle and Turbocompound Stack Energy Recovery

Early in the program, the advantages and disadvantages of turbocompound and two bottoming cycle technologies, Brayton cycles and Rankine cycles, were evaluated. A high level summary of this evaluation is shown as Figure 23. While turbocompound is a known technology and requires the least additional hardware, its benefits can be eroded by backpressure on the engine. Backpressure is created by aftertreatment components implemented to meet emissions regulations. With this in mind, turbocompound was considered high risk and the team initially focused on the development of a Brayton

bottoming cycle solution.

	<u>Advantages</u>	<u>Disadvantages</u>
Turbocompound	<ul style="list-style-type: none"> Known Technology 'Easy' to package 	<ul style="list-style-type: none"> Sensitive to BP
Rankine Cycle	<ul style="list-style-type: none"> Cycle Efficiency Insensitive to BP 	<ul style="list-style-type: none"> Packaging / cost ORC - additional fluid
Brayton Cycle	<ul style="list-style-type: none"> No additional fluid Insensitive to BP Lower cost (compared to Rankine) 	<ul style="list-style-type: none"> Cycle Efficiency Packaging

Figure 23. High Level Comparison of Stack Energy Recovery Technologies

3.4.1 Brayton Bottoming Cycle

An air Brayton bottoming cycle was initially selected as prime path technology for exhaust stack energy recovery. A schematic of the system on the base engine is shown in Figure 24.

Figure 24. Air Standard Brayton Bottoming Cycle

In this system, a heat exchanger is placed in the system downstream of the LP turbine to capture exhaust energy remaining in the stack. The energy is used to heat an air stream that drives a turbine. The turbine drives a compressor, which provides the boosted air stream to the heat exchanger. The excess turbine energy (beyond that needed to drive the compressor) is fed into the engine crankshaft via a mechanical or electronic transmission, thereby improving engine thermal efficiency. The system therefore requires a heat exchanger and a turbocharger modified to feed energy back into the crankshaft via a transmission system.

The efficiency of the Brayton cycle is defined as the ratio of the power output delivered over the energy input from the heat exchanger. This efficiency is highly dependent on the temperature of heat addition (the exhaust gas temperature entering the heat exchanger) and the component efficiencies and transmission efficiency. Engine simulation analysis indicated that the targeted 4% improvement in engine thermal efficiency could be achieved with the following component performance specifications: 85% efficient Brayton compressor, 85% efficient Brayton turbine, 93% transmission efficiency, and 90% heat exchanger effectiveness. The ability to achieve 93% transmission efficiency has been demonstrated with mechanical transmissions used on earlier turbocompound programs. The team was also confident that it could deliver aero components with efficiencies close to the 85% targets based on development results from other projects. That left the heat exchanger as the key remaining challenge to making the Brayton cycle solution work.

The primary difficulty with the heat exchanger is not achieving the 90% effectiveness target (effectiveness being the ratio of the actual heat transfer rate to the maximum possible heat transfer rate of the heat exchanger), but in doing so in a reasonable package size for a mobile application. The team evaluated several compact heat exchanger technologies in an effort to achieve a satisfactory solution.

First, the team evaluated a primary surface heat exchanger technology. This is a compact heat exchanger technology that has been used in recuperators for gas turbines. Sizing calculations using the requirements for the Brayton bottoming cycle were performed, and the results are shown in Figure 25.

Figure 25. Primary Surface Heat Exchanger and Sizing for Brayton Cycle

To achieve an effectiveness of 90% would require a heat exchanger volume of approximately 2.5 cubic feet. This would pose significant packaging challenges for installation in a class 8 on-highway truck chassis. Figure 26 shows concept layouts for the Brayton cycle components with several assumed heat exchanger sizes. The concepts show that even a heat exchanger volume of 1.0 cubic feet is sizeable and would be challenging to install in the available under-hood space.

A micro-channel heat exchanger technology was investigated next, in the hopes that the package size could be reduced. The required core volume to achieve 90% heat exchanger effectiveness with this technology was approximately 1.5 cubic feet. This is still 50% larger than the larger of the two concepts shown in Figure 26.

Figure 26. Packaging of Brayton Cycle Components on C15 On-Highway Truck Engine

At this point the Brayton cycle solution was not considered a highly practical solution for the class 8 truck market, given the packaging challenges. The decision was therefore made to revisit the turbocompound solution.

3.4.2 Mechanical Turbocompound

The Caterpillar team has conducted numerous investigations into mechanical turbocompound on past programs, most recently on the DOE-funded HTCD (Heavy Truck Clean Diesel) program. Design work from the HTCD program proved that a series-turbocharged engine with mechanical turbocompound could be packaged in a Class 8 truck (Figure 27).

Two questions that had to be addressed in this program were:

- 1) Can turbocompound deliver the needed 4% thermal efficiency improvement when used with aftertreatment (and the resulting backpressure)?
- 2) Can the high efficiency power turbine bearing system operate robustly through the required speed range?

The second question arose because of numerous bearing-related failures that occurred during the testing of the turbocompound engine on the HTCD program.

Engine simulation analysis was used to investigate the issue of thermal efficiency benefit versus engine backpressure, and results are shown in Figure 28.

Figure 27. Series-Turbocharged Turbocompound Engine Concept, HTCD Program

Figure 28. Turbocompound Fuel Economy Benefit vs. Engine Backpressure

Figure 28 shows that with high efficiency turbomachinery (as described in sections 3.1 to 3.3 of this report) and turbocompound transmission efficiencies of 90-95%, the 4% improvement target can be reached with backpressures as high as 25kPa at peak torque conditions. This level of backpressure is generally higher than what is anticipated at peak torque conditions for on-highway 2010 emissions-compliant engines, with DPF's (diesel particulate filters) and possibly SCR (selective catalytic reduction). Note the importance of the high efficiency turbomachinery; with production turbochargers the value of turbocompound is cut by roughly 1/3.

With these simulation results, turbocompound became the clear method of choice for stack energy recovery due to it's packaging and cost advantages relative to bottoming cycle technologies. The remaining challenge was to develop a robust and efficient bearing system for the power turbine.

Mechanical turbocompound is a technology that has made it into production in a limited number of on-highway applications over the last 10-20years. Those production systems relied on traditional turbo-style journal bearings for the power turbine. A higher efficiency ball-bearing system would reduce power turbine bearing losses and improve the thermal efficiency benefit of the turbocompound system. Caterpillar has utilized ball bearings in turbocompound demos for a number of years, most recently on the DOE HTCD (Heavy Truck Clean Diesel) program. However, there have been a number of bearing-related failures on those programs, evidence that a robust bearing system had not yet been solidly established. So the focus of this program was development and test of a robust, high-efficiency, ball-bearing power turbine unit.

Traditionally, power turbines are tested on-engine in an engine test cell rather than on the gas stand. This is because there is no compressor on the other end of the turbine shaft to load the power turbine wheel. But testing on a firing engine in an engine test cell is not the proper environment for performing the detailed shaft motion measurements necessary for bearing evaluation and optimization. So a new power turbine test methodology had to be developed.

A method was developed where the complete power turbine and gear train system was installed on the gas stand, with the engine acting as load compressor for the turbine gas stand testing (Figure 29)

Figure 29. Power Turbine on Gas Stand

Bearing system testing was conducted on a total of 5 different bearing system variants, at speeds ranging from 25 to 60krpm. A sampling of the results is shown in Figure 30. With the best bearing system, shaft motion of less than ± 0.002 inches was measured at all speeds. These results confirm a robust bearing system had been developed and that the issues observed on earlier programs had been resolved.

The power turbine performance was also measured and was consistent with previous performance tests and with performance predictions, with peak efficiencies of 82-84% total-static turbine efficiency.

Figure 30. Shaft Motion Measurement on Power Turbine at 45krpm

3.4.3 Electrical Turbocompound

Another way to do turbocompound is to eliminate the separate power turbine and mechanical geartrain and incorporate an electrical generator on the turbocharger shaft. The generator could harness the excess exhaust energy (energy beyond what's needed to provide boost) and feed it to an electrical motor on the engine crankshaft. This concept was developed under the Caterpillar-DOE program titled "Diesel Engine Waste Heat Recovery Utilizing Electric Turbocompound Technology" (contract DE-SC05-00OR-22810). The program ran from 2001 to 2005, and resulted in the design, development, and testing of an electric turbocompound system as shown in Figure 31.

Figure 31. Electric Turbocompound System

The program was a mixed success. On the plus side, the electric turbocompound system was successfully demonstrated on-engine: power was generated in the turbo-generator and fed to the electric motor on the crankshaft. However, the efficiency of the electrical machine on the turbo shaft fell short of the target values, and partially as a result of this the turbo generator had a tendency to overheat at higher power levels. The plan for this program was to update and improve the design during the first three phases of the program, with intent to procure and test the Gen2 turbo generator in Phases 4 and 5.

The Caterpillar team worked with Barber-Nichols Inc. to evaluate two concepts of electric turbocompound turbo-generator. The first was an improved version of the concept developed and tested under the previous DOE program, with the motor generator mounted on the shaft of the turbo between the turbine and compressor. The second had the motor generator mounted in front of the compressor, away from the heat of the turbine. Both concepts are shown in Figure 32.

Figure 32. Concepts for Gen2 Turbocompound Turbo-Generator

Preliminary analysis indicated that concept 2, while an improvement from a generator temperature standpoint, had significant Rotordynamic challenges. In particular, the rotor bending critical speed was within the turbocharger operating speed range, a situation that would likely lead to very high and potentially damaging shaft motion. For this reason the team elected to pursue development of Concept 1.

Concept 1 was developed and performance, structural, and thermal analyses were performed to validate the feasibility of the design. Generator efficiencies > 96% over the majority of the speed-load range were predicted. Predicted peak stator and rotor temperatures were acceptable (see Figure 33).

Figure 33. Steady State Thermal Plot – Electric Turbocompound

Results of the Rotordynamic analysis are shown in figure 34. Use of a compliant bearing gives a wide operating speed range between the first two rigid body critical speeds and the third critical speed, which is a bending mode. The bending critical is at 87krpm, which is 21% above the peak operating speed of 72krpm, a good margin.

The results of these analyses suggest that the Concept 1 design has overcome the issues of low efficiency and overheating that were inherent in the Gen1 design from the previous program. The design was deemed ready for detailing and procurement.

Figure 34. Critical Speed Map for Turbo-Generator Shaft System

4. Results and Conclusions

As discussed in section 1.3 above, this program was originally scoped as a 5-phase program and the main deliverable of Phase 3 was to be a mid-program on-engine demo of all the technologies developed to date. Early in Phase 3 it was decided that the program would be concluded at the end of Phase 3 to accommodate the transition of DOE funds to the Supertruck programs. Because this program would not be continuing into Phases 4 and 5, and because of the substantial expense involved in a full on-engine technology demonstration, agreement was reached with DOE to do a “virtual” demo. A “virtual” demo would entail taking all the measured component maps from the bench testing of the developed technologies and loading them into engine simulation for prediction of the resulting engine thermal efficiency benefit.

For this “virtual” demo, the following component maps were used:

- HP turbine – used RND turbine map as measured on gas stand (see section 3.1.1)
- LP turbine – used radial turbine map as measured on gas stand (see section 3.2.2)
- HP compressor – used high efficiency HP compressor map as measured on gas stand (see section 3.2)
- Power turbine – used axial power turbine map as measured on gas stand (see section 3.4.2)
- Aftercooler – used performance characteristics of the aftercooler as developed under the HTCD (Heavy Truck Clean Diesel) program

So the virtual demo was conducted using measured maps from 5 of the 7 solution elements, as shown in Figure 35. These 5 elements were anticipated to provide 9% improvement in thermal efficiency at the peak torque design point, or 90% of the total goal.

Figure 35. Solution Elements Included in "Virtual" Demo

With the measured component maps for these 5 solution elements the predicted improvement in thermal efficiency at the peak torque design point was 7%. This is an impressive accomplishment and just 2% short of the 9% improvement goal for these elements. The 2% shortfall from the goal and the remedies to get to the full 9% thermal efficiency improvement were accounted for as follows:

Solution element	Thermal Efficiency Improvement			Explanation
	Target	Simulated	Shortfall	
HP Turbine	2.0	1.3	0.7	Used RND. Need MND developed
LP Turbine	1.0	0.6	0.4	Used radial turbine
Compressors	0.7	0.2	0.5	No modification to LP stage yet
Stack Recovery	4.0	3.6	0.4	Due to shortfalls in above 3
Total Shortfall			2.0	

So for the HP turbine stage, the RND turbine map was used. As discussed in section 3.1.1, the development of the MND turbine is expected to improve the HP turbine stage performance and make up most, if not all, of the shortfall on that solution element. For the LP turbine, the target can be met with the axial turbine technology as discussed in section 3.2.1, but the performance shortfall was accepted to go with the more practical (from a packaging standpoint) radial turbine. For the compressors, only the HP compressor stage was modified and measured in Phases 1-3 of this program; rolling the lessons learned into an improved LP compressor stage would decrease the shortfall on this solution element. Finally, the turbocompound shortfall is directly attributable to the efficiency shortfalls of the other elements, and would be remedied with improvements to those turbocharger technologies.

A key point to note is that if the improved turbine and compressor stages and the aftercooling are left out, the benefit of turbocompound drops to 2%, or half the 4% target. This dramatically illustrates the need for a system solution and affirms the value of improved turbocharger technologies for effective waste heat recovery.

5. Collaborations / Consultations / Partners

The Caterpillar team worked with the following organizations to meet the requirements and objectives of this DOE-sponsored program:

- Barber-Nichols Inc. – Collaboration on the design and analysis of the Gen2 electric turbocompound turbo-generator
- ConceptsNREC – Consulted on turbomachinery design and optimization
- Honeywell Turbo Technologies – Provided prototype turbomachinery components.
- Imperial College, London – Performed gas stand testing of advanced turbomachinery components
- Oak Ridge National Lab – Performed on-engine testing of advanced turbomachinery components
- Turbo Solutions Engineering LLC - Consulted on turbomachinery design and optimization

Each of these partners provided a critical service in the successful execution of this program.

6. Publications / Conference Presentations

Results of this project were presented at the following conferences and reviews:

- 12th DOE DEER Conference, August 23 2006, Detroit Michigan
- 13th DOE DEER Conference, August 15 2007, Detroit Michigan
- 14th DOE DEER Conference, August 6 2008, Dearborn Michigan
- 15th DOE DEER Conference, August 5 2009, Dearborn Michigan
- 16th DOE DEER Conference, September 29 2010, Dearborn Michigan
- FY2007 DOE Vehicle Technologies Program Annual Merit Review, June 19 2007, Washington DC
- FY2008 DOE Vehicle Technologies Program Annual Merit Review, February 27 2008, Bethesda Md
- 2009 DOE Hydrogen Program and Vehicle Technologies Program Annual Merit Review and Peer Evaluation Meeting, May 21 2009, Arlington Va
- 2010 DOE Hydrogen Program and Vehicle Technologies Program Annual Merit Review and Peer Evaluation Meeting, June 10 2010, Washington DC

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