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# US10 Capable Prototype Volvo MG11 Natural Gas Engine Development: Final Report

## December 16, 2003 — July 31, 2006

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*Hagerstown, Maryland*

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*Woodland Park, Colorado*

***Subcontract Report***
**NREL/SR-540-40756**
**October 2006**

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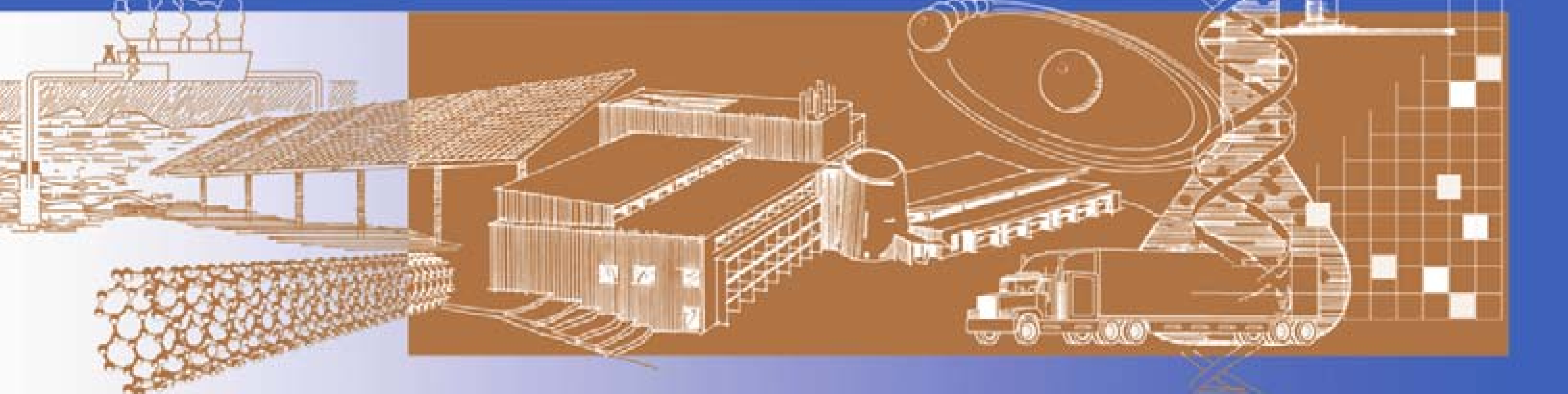

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## EXECUTIVE SUMMARY

The objective of this project is to develop a low emissions natural gas engine. The emission targets for this project are 0.27 g/kW-hr (0.2 g/bhp-hr) of $NO_x$ and 0.013 g/kW-hr (0.01 g/bhp-hr) of particulate matter. To meet the objective, a chemically correct combustion (stoichiometric) natural gas engine with exhaust gas recirculation (EGR) and a three-way catalyst (TWC) was developed. In addition, a Sturman camless Hydraulic Valve Actuation (HVA) system was used to improve efficiency.

A Volvo 11 liter diesel engine was converted to operate as a stoichiometric natural gas engine. Operating a natural gas engine with stoichiometric combustion allows for the effective use of a TWC, which can simultaneously oxidize hydrocarbons and carbon monoxide and reduce $NO_x$. High conversion efficiencies are possible through proper control of air-fuel ratio. Adding EGR lowers combustion chamber temperatures, which can improve efficiency, reduce the tendency to knock and lower engine out $NO_x$ emissions.

A camless HVA system by Sturman Industries was applied to the engine. Using the HVA system can reduce pumping losses at light loads through either an early intake valve closing or a late intake valve closing (Miller cycle). The HVA system can also enable the use of high compression ratio pistons, where a lower effective compression ratio can be used at the high loads through either an early or a late intake valve closing, and higher compression ratios can be used at light to medium loads. Pistons with a 14.3:1 compression ratio were used on this engine.

Emission tests were run at the 13-mode steady state test points and at three additional points (15% load at the low, intermediate, and high engine speeds). The results show a weighted 13-mode $NO_x$ emission level of 0.005 g/kW-hr, which easily meets the emissions target of 0.27 g/kW-hr (0.2 g/bhp-hr). Due to time and budget constraints, particulate matter was not measured, but is expected to be below the target of 0.013 g/kW-hr (0.01 g/bhp-hr).

The high compression ratio pistons and variable valve timing available from the HVA system improved efficiency by 6.1% over a fixed valve timing cam based system using the weighted 13-mode results. Further efficiency improvements may be possible through reduced losses of the HVA system and through improvement in light load combustion by using the HVA system to increase in-cylinder motion (swirl).

## 1.0 INTRODUCTION

Natural gas (NG) is an abundant energy source in this country that can be used as an automotive fuel. With rising fuel prices, many companies are seeking alternatives to imported oil to help contain fuel cost increases. However, only a small portion of current vehicles are powered by natural gas and gas vehicles are mainly operated by local and state municipalities. The National Renewable Energy Laboratory (NREL), led by the Center for Transportation Technologies and Systems, has the goal to help industry introduce alternative fueled vehicles into the marketplace by working with public and private organizations to develop and demonstrate innovative technologies to help reduce the nation's dependence on imported oil. California has perhaps the most aggressive programs to help reduce pollution and curb the use of imported oil. Stringent fleet rules in California require that whenever public fleet operators with 15 or more vehicles replace or purchase new, the vehicle must use alternative fuels. NREL is leading the effort to develop the next generation heavy-duty natural gas engine to help reduce emissions in nonattainment areas.

Emissions standards are also becoming increasingly stringent across the United States. Regulations for 2007 require that emissions be reduced from the current 2.5g/bhp-hr of $NO_x$+NMHC to a phase-in level of 1.18 g/bhp-hr of $NO_x$ with NMHC less than 0.14 g/bhp-hr and particulate matter (PM) less than 0.01 g/bhp-hr. Meeting the PM standard with diesel engines will require the use of a diesel particulate filter (DPF). Heavy exhaust gas recirculation (EGR) will be required for $NO_x$ control. The $NO_x$ standard for 2010 is 0.2 g/bhp-hr for all heavy-duty engines. For the diesel engine to meet that standard, new $NO_x$ aftertreatment technology will be required. Natural gas engines have an advantage over diesel in that they operate on a gaseous fuel and are typically spark-ignited which emits lower PM by nature. This allows the combustion system to operate at or near stoichiometric air to fuel ratios (AFR), and maintain high tailpipe exhaust temperatures over the typical duty cycle when compared to a diesel engine. This allows a three-way catalyst (TWC) to be coupled to the exhaust system to oxidize CO and hydrocarbon (HC) emissions and reduce $NO_x$.

TWC technology was developed in the 1970's for the automobile industry to clean up the emissions from the gasoline engine and also proved to be the most cost effective solution. Since then, TWC technology has evolved and demonstrated extremely low CO, HC, and $NO_x$ emissions for the auto industry, which allows even lower emitting vehicles to be manufactured. Operating a heavy-duty natural gas (NG) engine with a TWC does not require special hardware. Proven control systems and sensors used for the light-duty industry can be applied to the heavy-duty NG engine. EGR however, will need to be added for knock and exhaust temperature control as stoichiometric operation inherently increases the tendency for knock and engine out exhaust temperatures. Applying TWC technology to a heavy-duty NG engine leverages 35 + years of development as applied to light-duty gasoline engines and is a very cost effective solution for meeting future emissions standards. Mack Trucks Inc. is the leader in heavy-duty NG engines for use in the refuse industry. Waste Management (WM) is the leader in natural gas refuse haulers and operates a fleet of 400+ natural gas refuse haulers in southern California.

NREL is funding a program with Volvo Powertrain (formerly Mack Powertrain) to demonstrate a next generation, stoichiometric natural gas engine with a TWC and a camless Hydraulic Actuated Valve (HVA) system. The engine is based on the Volvo MD11 diesel engine with natural gas components from the MG9 engine currently in production.

## 1.1 Project Objective

The objective of this project is to develop a low emissions natural gas engine. The emission targets for this project are 0.27 g/kW-hr (0.2 g/bhp-hr) of $NO_x$ and 0.013 g/kW-hr (0.01 g/bhp-hr) of particulate matter. To meet the objective, a chemically correct (stoichiometric) combustion, natural gas engine with exhaust gas recirculation (EGR) and a three-way catalyst (TWC) was developed.

In addition to low emissions, diesel-like efficiencies are targeted through use of a Sturman camless Hydraulic Actuated Valve (HVA) system. The HVA system can reduce pumping losses at light loads through either an early intake valve closing or a late intake valve closing (Miller cycle). The HVA system can also enable the use of high compression ratio pistons. The compression ratio of a gas engine is limited by knock at high loads. With the HVA system, a lower effective compression ratio can be used at the high loads through either an early intake valve closing or a late intake valve closing, and higher compression ratios can be used at light to medium loads.

## 2.0 TECHNICAL DISCUSSIONS/RESULTS

A Volvo MD11 diesel engine (known as the MP7 engine in North America) was modified for operation on natural gas. Although there is not a gas version of the MD11 engine, there is a production gas version of the MD9 diesel engine known as the MG9 engine offered in Europe for Volvo Bus Corp., which is a stoichiometric, external high pressure EGR engine. Since the MD9 and MD11 engines share the same engine block and head, the conversion of the MD11 engine to gas used many of the parts from the MG9 engine in addition to parts from the Mack E7G engine. The gas version of the MD11 engine will be designated as the MG11 engine. In addition to converting the engine to operate on gas, the Sturman HVA system was developed as well. Since this MG11 engine is the only Volvo 11 liter engine converted to operate on natural gas, baseline tests with a normal cam were not possible; therefore tests were conducted using the HVA system to simulate a fixed valve timing of a diesel cam with a typical compression ratio used for gas engines to obtain a baseline. Tests with throttleless operation and high compression ratio pistons were conducted using the HVA variable valve timing system and the results were compared to the baseline results. All efficiency comparisons (% increase) are in relative improvements, and not in efficiency points.

### 2.1 Engine Description

The engine arrived at SwRI as a natural gas engine with a high pressure EGR loop, a variable geometry turbocharger, and 10.25:1 compression ratio pistons with a Sturman HVA system partially installed. Components were added and or modified to convert the engine to a low pressure EGR loop with exhaust and intake breathing modified to minimize cylinder to cylinder distribution problems. The turbocharger was changed for a better match for this engine, high compression ratio pistons (14.3:1) were installed, and the Sturman HVA system installation was completed.

Engine specifications are:

- Volvo MG11 - 11 liter (659 cu. in.) displacement, 6-cyl inline, 4 valves per cylinder
- Bore x Stroke = 123 cm x 152 cm
- 325 bhp@1950 rpm
- Peak torque: 1180 lb-ft@1250 rpm
- Wastegated turbocharger with after cooler
- Low pressure, cooled EGR
- Compression Ratio = 14.3:1 **(**Baseline Compression ratio = 10.25:1)
- Spark plugs-- 6 – Denso
- Ignition System – Volvo/Bosch Smart Coil on Plug

Fuel system specifications are:

- Woodward Flotec “drive by wire” throttle control. Min Max automotive type
- Servojet (SP021) natural gas injectors, 8 available, 6 used
- Mack developed air / natural gas mixing ring and elbow

- Stoichiometric, Closed – Loop
- Fuel = LNG, CNG
- Fuel pressure set point = 100-120 psi

### *2.1.1 Engine Breathing*

The engine breathing includes the intake system, the exhaust system, the turbocharger and the EGR loop. The engine arrived at SwRI with an end fill intake manifold, a variable geometry turbocharger, and a high pressure V-Pulse EGR loop. The engine was modified as described below.

#### *2.1.1.1 Intake Manifold*

The end fill intake manifold was replaced with a center fill manifold. The center fill provides a more even cylinder-to-cylinder distribution of the fuel and air mixture. A picture of the end fill intake manifold is shown in Figure 1. A picture of the center fill intake manifold is shown is Figure 2. The center fill manifold was mounted upside down so that the throttle and mixing elbow would not interfere with the HVA transfer plate.

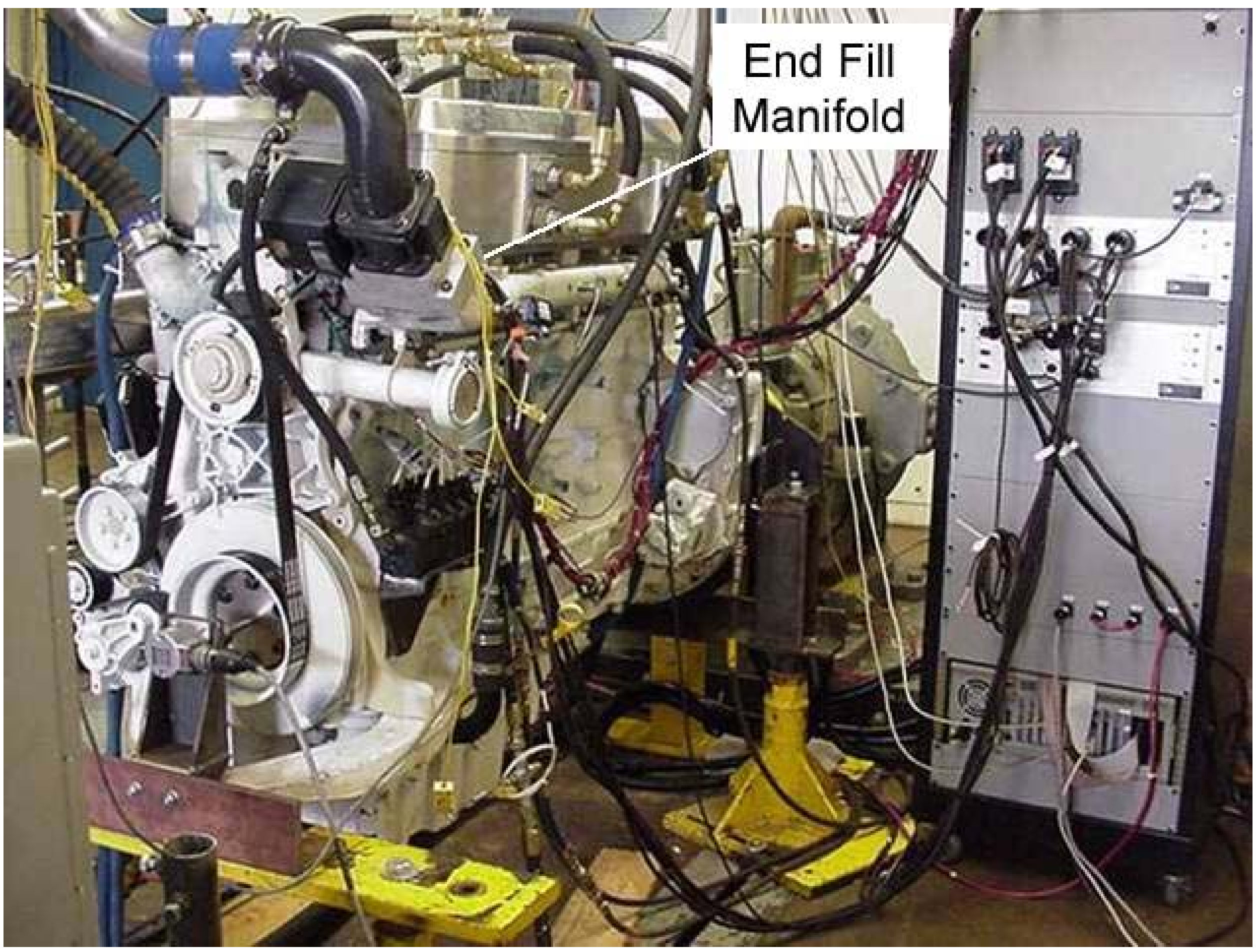


**Figure 1. End Fill Intake Manifold**

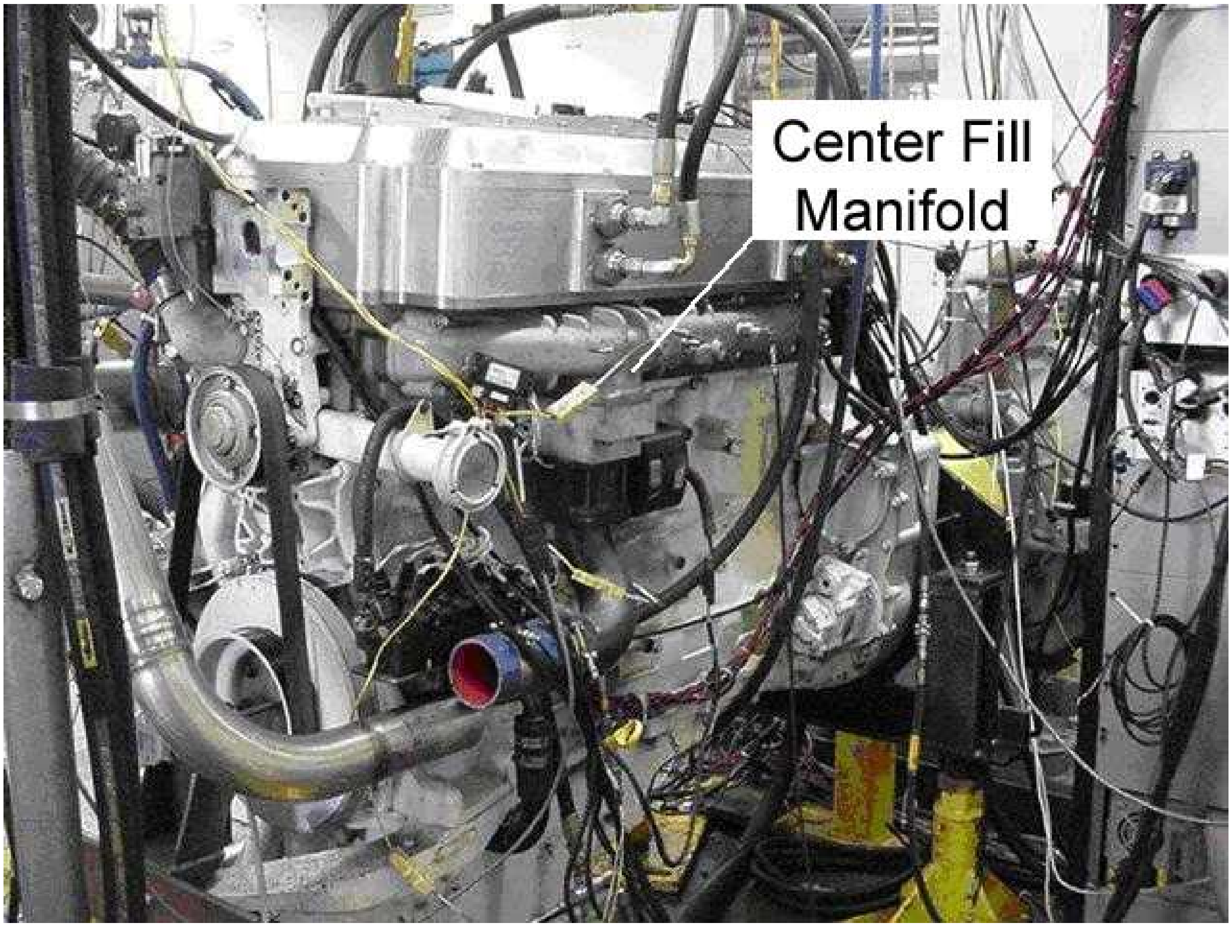


**Figure 2. Center Fill Intake Manifold**

### *2.1.1.2 Exhaust System and Turbocharger*

The turbocharger on the engine was a variable geometry turbocharger. Since a low pressure EGR loop was used, a VGT was not needed to drive EGR, and it was decided to use a fixed geometry turbocharger with a wastegate. The wastegate on the turbocharger only bypassed exhaust gases on one side of the divided housing; therefore an external wastegate was used. A wastegate system was setup to bypass exhaust gases from both sides of the divided exhaust manifold. A picture of the external wastegate is shown on Figure 3.


**Figure 3. External Wastegate**

### *2.1.1.3 EGR System*

The EGR system was changed from a high pressure loop to a low pressure loop, where the EGR is routed from the exhaust system after the turbine, though a laboratory heat exchanger, to an EGR valve, and into the intake system before the compressor on the turbocharger. A picture of the EGR system is shown in Figure 4. The fan in the picture was used to cool the wastegate.

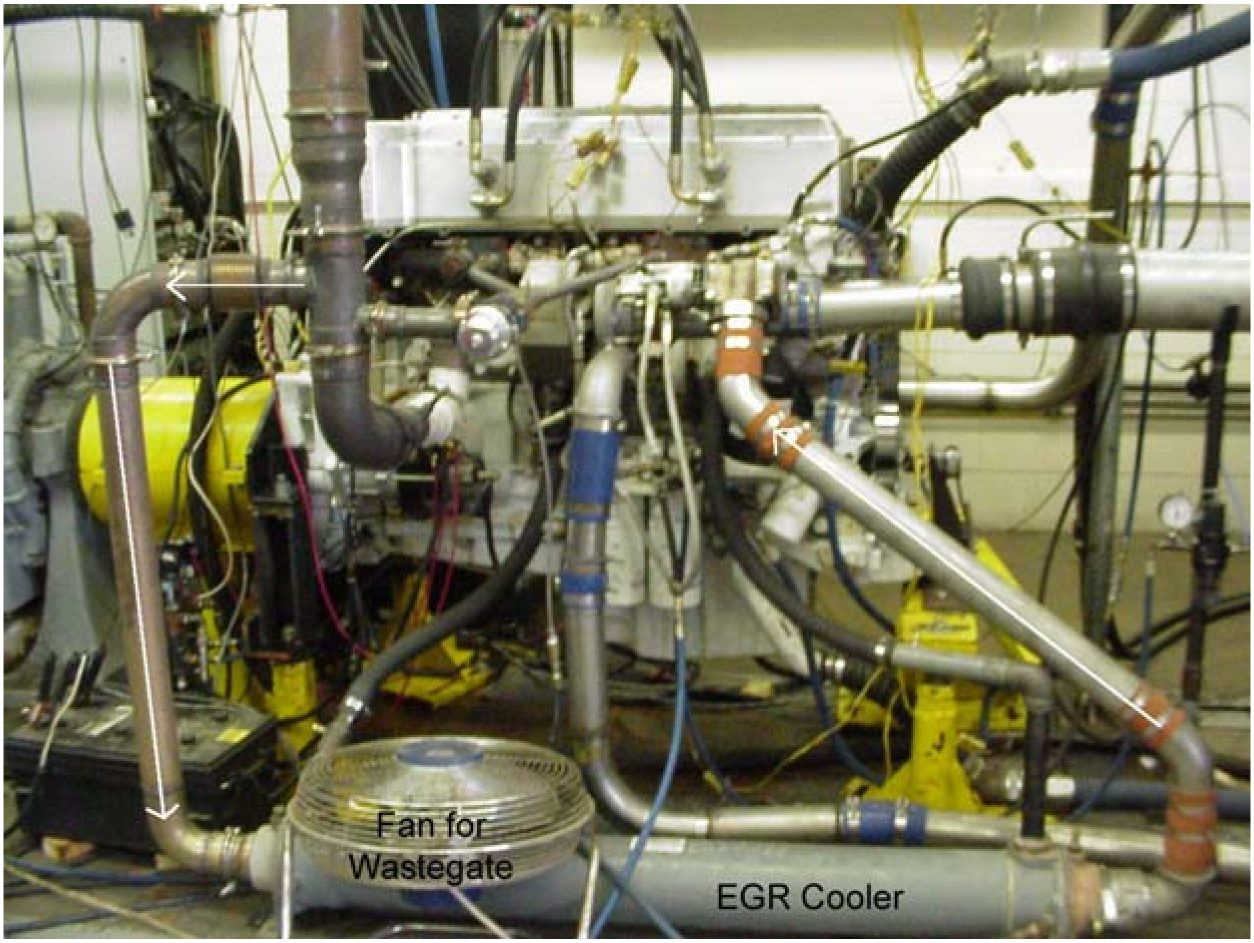


**Figure 4. Low Pressure EGR Loop**

### *2.1.2 Engine Ignition System*

The engine ignition system is adapted from the MG9 ignition system. The coil extensioner and coil tube have been elongated to fit with the special valve space made for the HVA system.

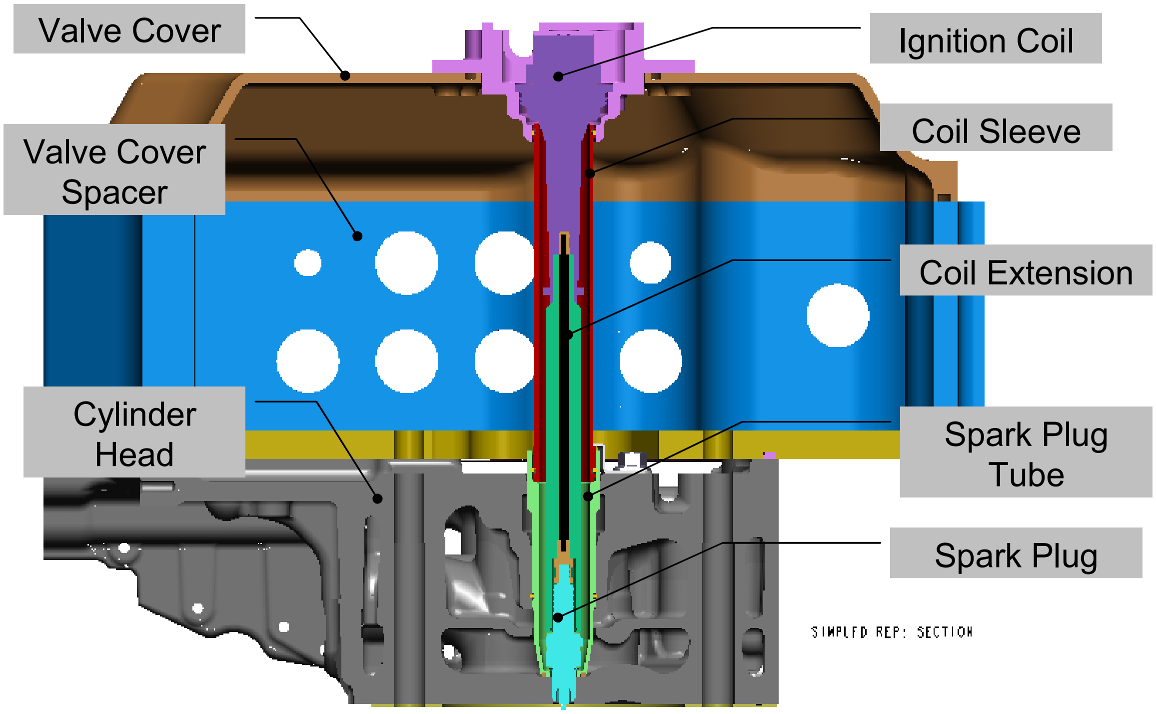


**Figure 5. Engine Ignition System**

### *2.1.3 Engine Pistons*

Aluminum pistons are chosen to avoid high surface temperature and they are also easy to machine. Fourteen (14) low compression ratio (10.25:1) pistons are made for baseline testing. Compared to the MD11 pistons, MG11 pistons have the cooling gallery removed in order to have enough space for reduced compression ratio. The compression height was reduced to accommodate the higher thermal deformation on the crown. The piston ring groves were redesigned as well.

Seven (7) high compression ratio (14.3:1) pistons are designed to better utilize the variable valve timing capability.

Seven (7) blank pistons are reserved for unexpected situations during engine testing.

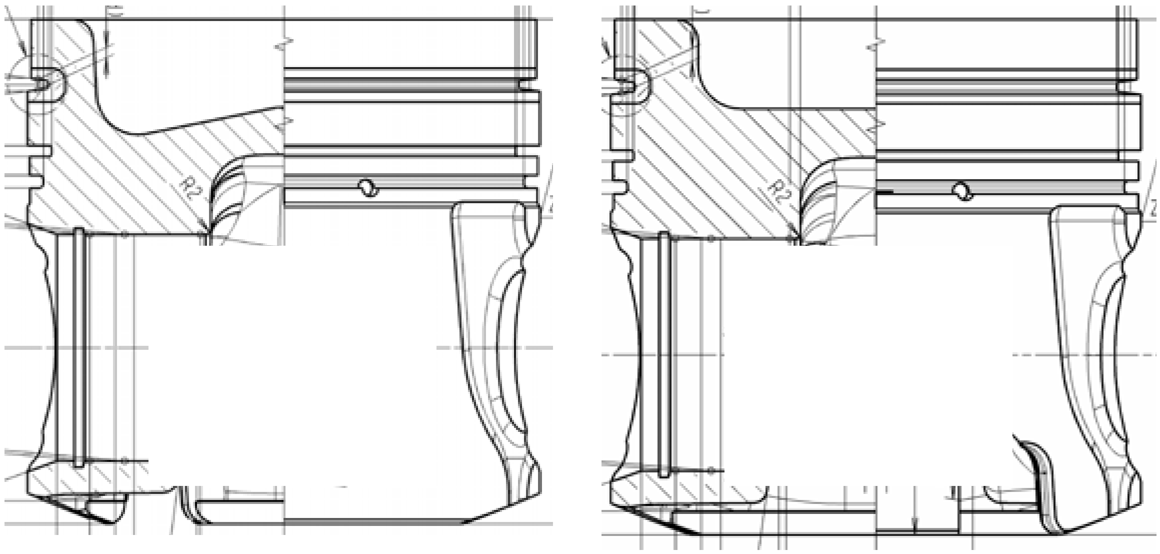

**Figure 6. Piston Designs, Left – 10.25:1 CR, Right – 14.3:1 CR**

### *2.1.4 Engine Control System*

For this natural gas engine, a Rapid Prototyping Electronic Control System (RPECS) was used. The RPECS is a PC based control system that uses C++ as the programming language, which provided full user flexibility for electronic controls. The RPECS allows for "on the fly" changes in the engine calibration. Typical algorithms for a natural gas engine were used.

### *2.1.5 HVA System*

The major parts of the Hydraulic Valve Actuation (HVA) system are the pump cart, the Valve Drive Module (VDM), the desktop computer, the transfer plate to mount the actuators and oil rails to the head, the oil rails, and the actuators. The possible variable valve motion includes:

- Exhaust Valve Opening
- Exhaust Valve Lift
- Exhaust Valve Closing Landing Rate
- Exhaust Valve Closing
- Intake Valve Ramp Hold Duration
- Intake Valve Opening
- Intake Valve Lift
- Intake Valve Closing Landing Rate
- Intake Valve Closing

A picture of the HVA oil rails and actuatorsis shown in Figure 7. Figure 8 shows the VDM and Figure 9 shows the hydraulic pump cart. Further information of the Sturman HVA system can be obtained from Sturman.

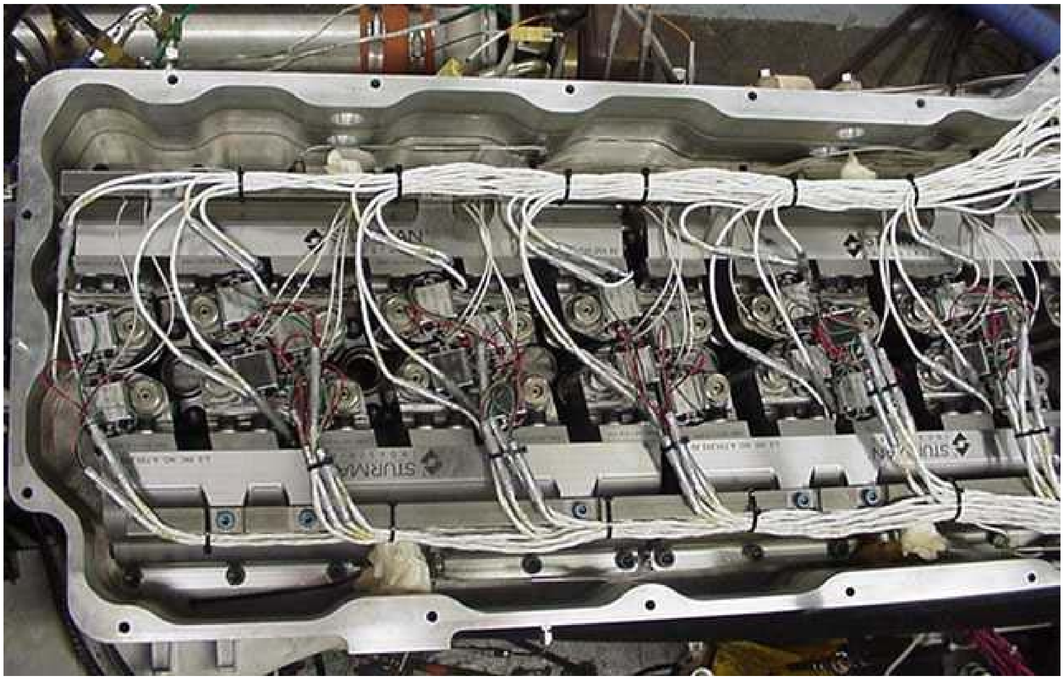

**Figure 7. Sturman HVA System**

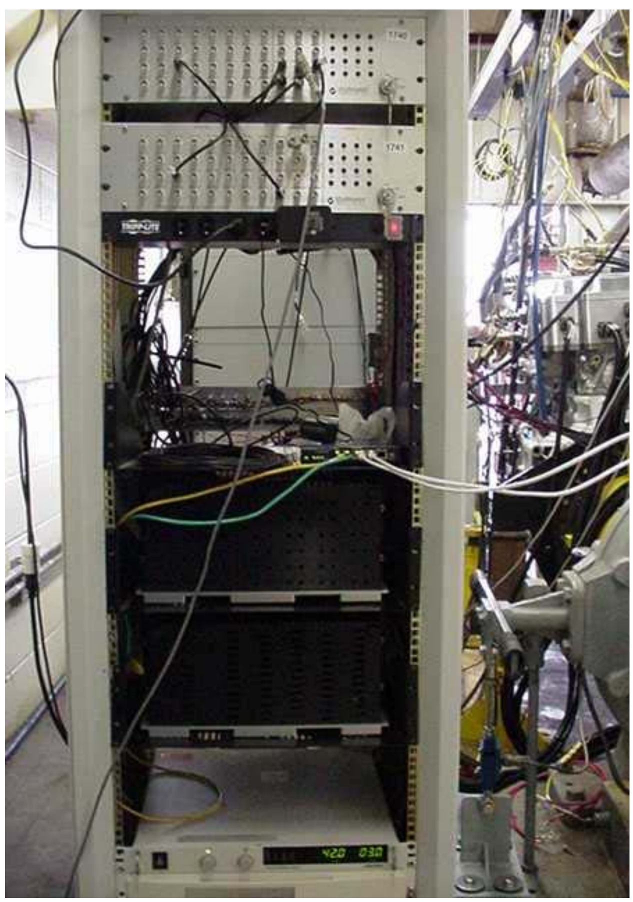

**Figure 8. Sturman Valve Drive Module (VDM)**

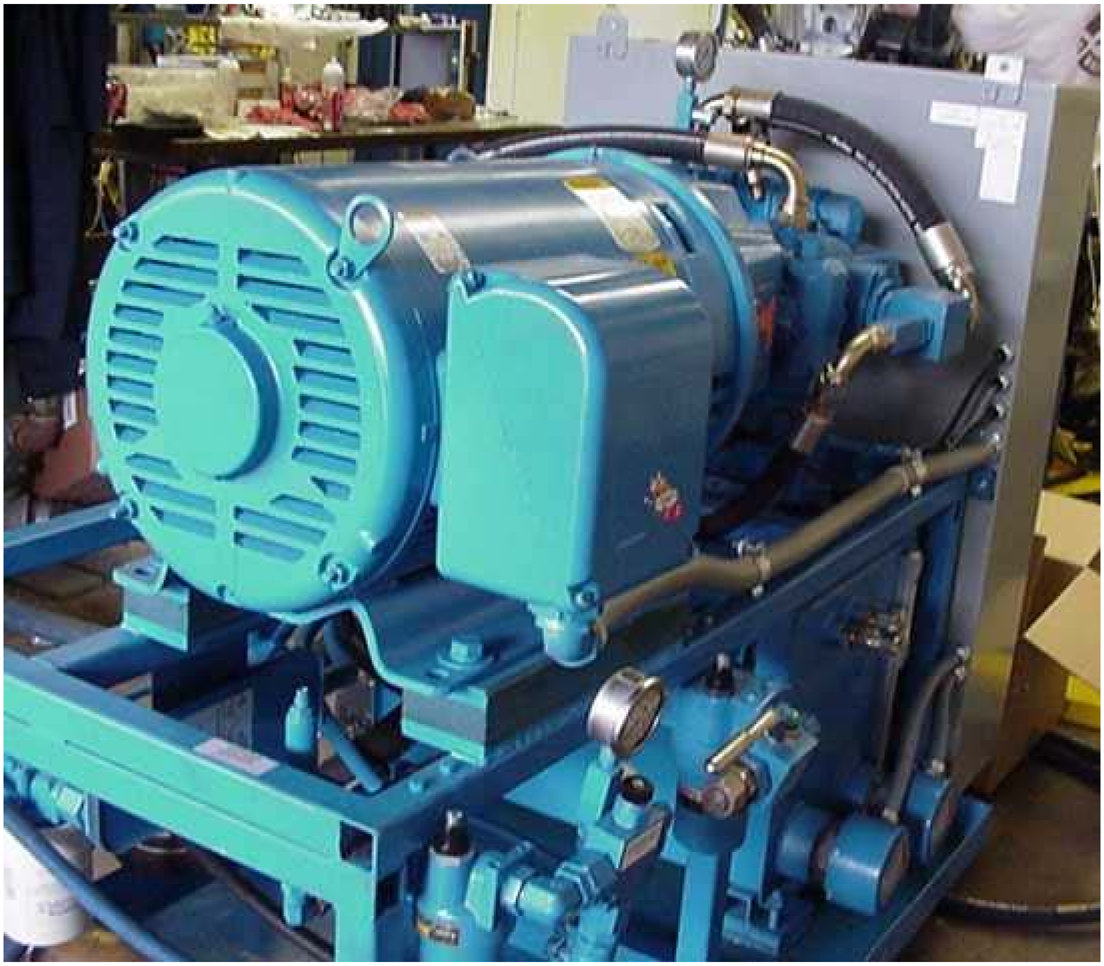

**Figure 9. Hydraulic Pump Cart for the HVA System**

## 2.2 Engine Development

Engine development was conducted in a steady-state test cell.

### *2.2.1 Test Cell Description*

The steady-state test cell uses an eddy current dynamometer to absorb power. A heat exchanger is used for the engine coolant where the engine out coolant temperature is controlled to a specified temperature. A heat exchanger is also used for the intercooler, where the intercooler out temperature is controlled to a specified temperature. Natural gas was used for the testing, where it is compressed into a storage tank, and then delivered to the test cell where regulators reduce the pressure to a specified level (~110-120 psi). A gas chromatograph records the gas composition on an hourly basis. The average gas composition during the engine testing is shown in Table 1 along with the standard deviation.

## TABLE 1. AVERAGE GAS COMPOSITION AND STANDARD DEVIATION FOR DEVELOPMENT WORK

| Component | Concentration (vol %) | Standard Deviation |
|---|---|---|
| Methane | 94.22 | 0.40 |
| Ethylene | 0.00 | 0.00 |
| Ethane | 2.82 | 0.34 |
| Propane | 0.54 | 0.12 |
| Butanes | 0.21 | 0.06 |
| C5+ | 0.12 | 0.03 |
| Carbon Dioxide | 1.06 | 0.13 |
| Nitrogen | 1.02 | 0.16 |
| Hydrogen Sulfide | 0.00 | 0.00 |

Data were recorded with the SwRI data acquisition system. The data recorded are shown in Table 2. All equipment was calibrated according to SwRI Standard Operating Procedure for Calibration and Maintenance. The efficiency recorded by the SwRI data acquisition system does not include the power required to drive the hydraulic pump for the Sturman HVA system.

## TABLE 2: DATA RECORDED

| | | | |
|---|---|---|---|
| Misc | Run Number<br>Date<br>Time<br># of Points Averaged<br>Engine Speed<br>Torque<br>Fuel Flow<br>Air Flow<br>Relative Humidity | Calculated | BTE<br>BSFC<br>BSEC<br>BMEP<br>Power<br>Air Fuel Ratio<br>Equivalence Ratio<br>Fuel H/C Ratio<br>Fuel O/C Ratio<br>Fuel N/C Ratio<br>Stoich Air Fuel Ratio<br>Fuel Molecular Weight<br>Fuel High Heating Value<br>Fuel Low heating Value<br>Reactive H/C Ratio<br>Specific Humidity<br>EGR Rate |
| Pressure | Fuel<br>Barometric<br>Inlet Restriction<br>Boost Before Intercooler<br>Boost After Intercooler<br>Intake Manifold<br>Pre-Turbine<br>Exhaust Restriction<br>Oil Gallery | Temperature | Coolant In<br>Coolant out<br>Fuel<br>Ambient Air<br>Boost Before Intercooler<br>Boost After Intercooler<br>Compressor Inlet<br>Intake Manifold<br>Exhaust Stack<br>Individual Exhaust (6)<br>Pre-Turbine<br>Oil Gallery<br>Oil Sump<br>Catalyst (3) |

| | | | |
|---|---|---|---|
| Fuel Composition | Nitrogen<br>Methane<br>Carbon Dioxide<br>Ethane<br>Hydrogen Sulfide<br>Propane<br>Iso-Butane<br>Butane<br>Iso-Pentane<br>Pentane<br>Hexanes<br>Heptanes<br>Octanes<br>Iso-Nonanes<br>Ethylene | Emissions<br><br><br><br><br><br><br>Emissions Calculations | CO<br>HC<br>NOx<br>CO2<br>O2<br>Intake CO2 (for EGR)<br><br>BSCO<br>BSHC<br>BSNOx<br>Corrected BSNOx<br>Equivalence Ratio<br>EGR Rate |

### *2.2.2 Engine Calibrations*

A typical calibration for a natural gas engine was used. EGR rates were selected based on previous engine data on a Mack E7GT engine. Ignition timing was set to maintain ~15° ATDC location of peak pressure. Equivalence ratios were set to provide low $NO_x$ and CO emissions after the catalyst. Typically, just slightly rich of stoichiometric provides the lowest emissions for natural gas engines.

### *2.2.3 Test Points*

Steady state testing was conducted. The plan was to perform a 22-mode test. The 22-mode test is a combination of the 13-mode steady state test, also known as the European Stationary Cycle (ESC) or Organisation Internationale des Constructeur D'Automobiles (OICA) test, and the SwRI 16-mode test. The 13-mode test is used in certification of the engine. The 16-mode test was developed to cover some of the areas that the 13-mode does not, such as light loads (<25% load) and/or low engine speeds (< peak torque speed). These are typically the areas where the calibration could be significantly different from higher speed and load points. The test points are shown in Table 3 and Figure 10. Speed values have been rounded to a convenient test point. The speeds for the 16-mode and 22-mode test points are based on a more typical low speed at idle (600 rpm) and a more typical high speed at rated power (1950 rpm). The 13-mode speeds are based on the requirements of the Code of Federal Regulations (CFR), where the low speed is defined as the lowest engine speed where 50% of the maximum power occurs and the high speed is defined as the highest engine speed where 70% of the maximum power occurs. The 25% speed, as defined for the 13-mode, is typically the same as the more typical 50% speed or peak torque speed of the engine. The 75% speed, as defined for the 13-mode, is typically the same as the more typical 100% speed or peak power speed of the engine. Due to time and budget constraints, the testing was reduced to mode 1 and modes 8-22. These points are shown in Figure 11.

## TABLE 3: STEADY STATE TEST POINTS

| 22 Mode | Speed ** | Load | Speed (rpm) | Torque (N-m) | 16 Mode | 13 Mode | 13 Mode Weight | Speed for the 13 Mode *** |
|---|---|---|---|---|---|---|---|---|
| 1* | 0% | 0% | 600 | 160 | 1 | 1 | 15% | idle |
| 2 | 15% | 15% | 800 | 146 | 2 | | | |
| 3 | 15% | 50% | 800 | 488 | 3 | | | |
| 4 | 15% | 100% | 800 | 976 | 4 | | | |
| 5 | 30% | 15% | 1000 | 203 | 5 | | | |
| 6 | 30% | 50% | 1000 | 678 | 6 | | | |
| 7 | 30% | 100% | 1000 | 1356 | 7 | | | |
| 8* | 50% | 15% | 1250 | 240 | 8 | | | |
| 9* | 50% | 25% | 1250 | 400 | | 7 | 5% | 25% |
| 10* | 50% | 50% | 1250 | 800 | 9 | 5 | 5% | 25% |
| 11* | 50% | 75% | 1250 | 1200 | | 6 | 5% | 25% |
| 12* | 50% | 100% | 1250 | 1600 | 10 | 2 | 8% | 25% |
| 13* | 75% | 15% | 1600 | 206 | 11 | | | |
| 14* | 75% | 25% | 1600 | 344 | | 9 | 10% | 50% |
| 15* | 75% | 50% | 1600 | 688 | 12 | 3 | 10% | 50% |
| 16* | 75% | 75% | 1600 | 1032 | | 4 | 10% | 50% |
| 17* | 75% | 100% | 1600 | 1376 | 13 | 8 | 9% | 50% |
| 18* | 100% | 15% | 1950 | 178 | 14 | | | |
| 19* | 100% | 25% | 1950 | 297 | | 11 | 5% | 75% |
| 20* | 100% | 50% | 1950 | 593 | 15 | 13 | 5% | 75% |
| 21* | 100% | 75% | 1950 | 890 | | 12 | 5% | 75% |
| 22* | 100% | 100% | 1950 | 1186 | 16 | 10 | 8% | 75% |

* Tested
** With idle as the low speed and rated power as the high speed
*** With the low and high speeds as defined by the 40CFR86

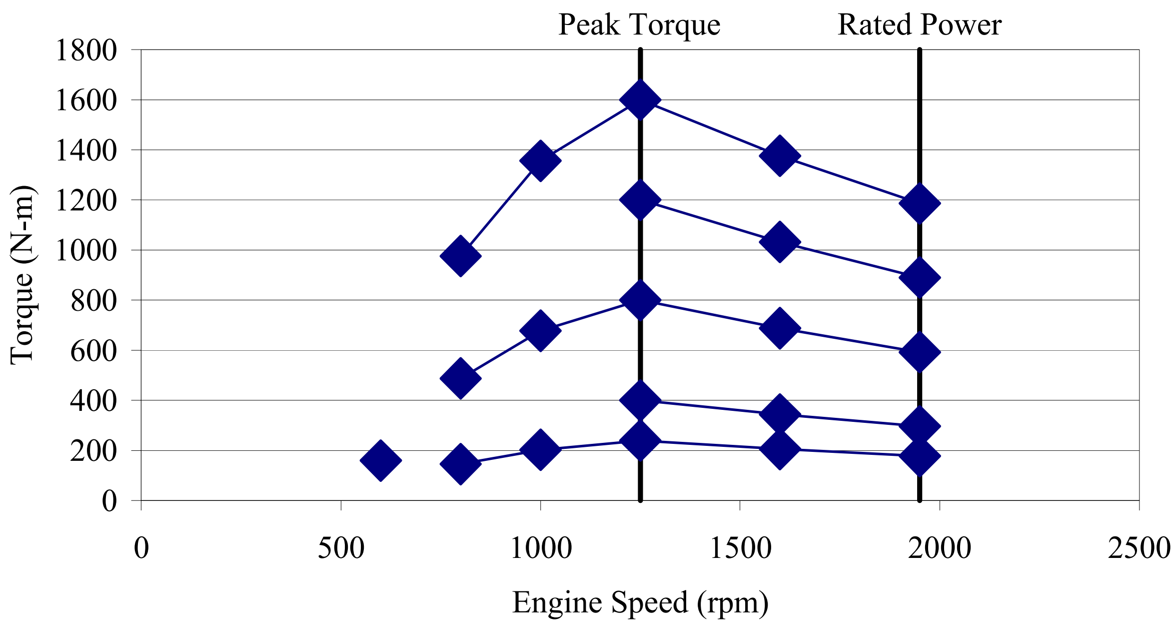


**Figure 10. 22-Mode Steady State Test Points**

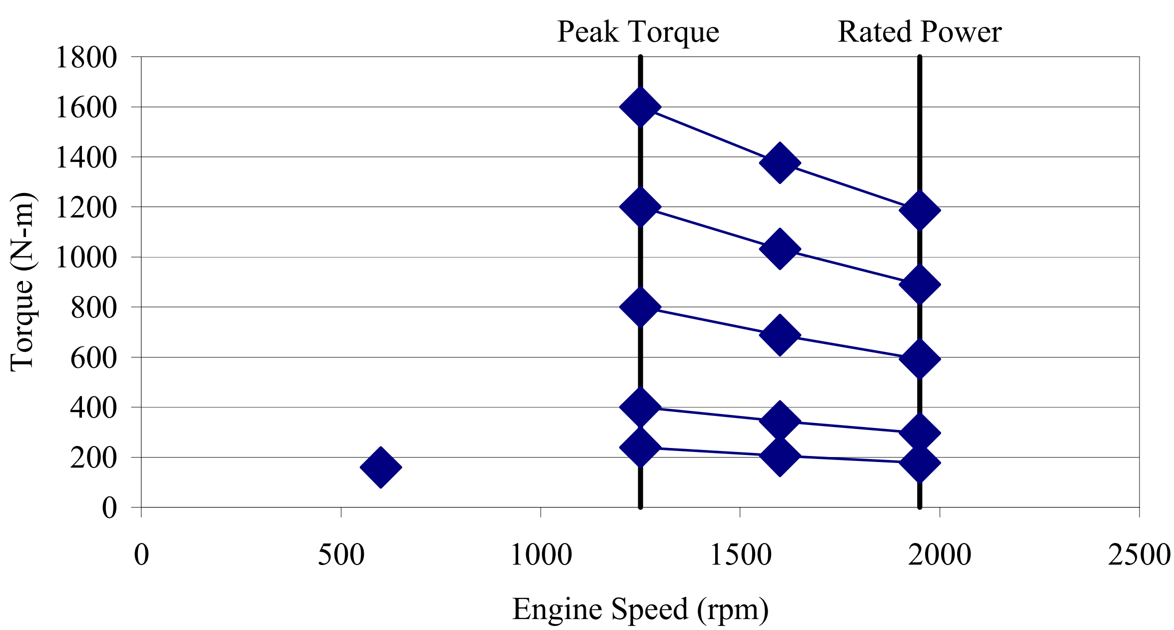


**Figure 11. 16 Steady State Modes Tested**

### *2.2.4 Problems*

During the course of the program, there were some problems that occurred. These include a broken valve, foreign material entering the engine through the intake system, excessive valve wear, and noise on the HVA system.

The first problem that occurred was a broken valve. The head on an intake valve on the number 3 cylinder fractured and embedded itself into the top of the piston. A picture of the cylinder head with the missing valve head is shown in Figure 12. The valve head embedded in piston is shown in Figure 13. It is assumed that the valve had contact with the piston multiple times and failed due to bending fatigue. All of the valves that were hitting the piston showed some bending. A picture of the result from a piston versus valve collision is shown in Figure 14. The software for the HVA system was updated to disable the valve when a valve to piston collision is detected through the valve position feedback signal. This problem occurred with the 10.25:1 compression ratio piston, which were then replaced with the 14.3:1 compression ratio pistons. The original valves and valve seats in the head were from the MG9 engine. The valve seats were ground and the valves were replaced with MD11 valves.


**Figure 12. Broken Intake Valve Head on Cylinder 3**

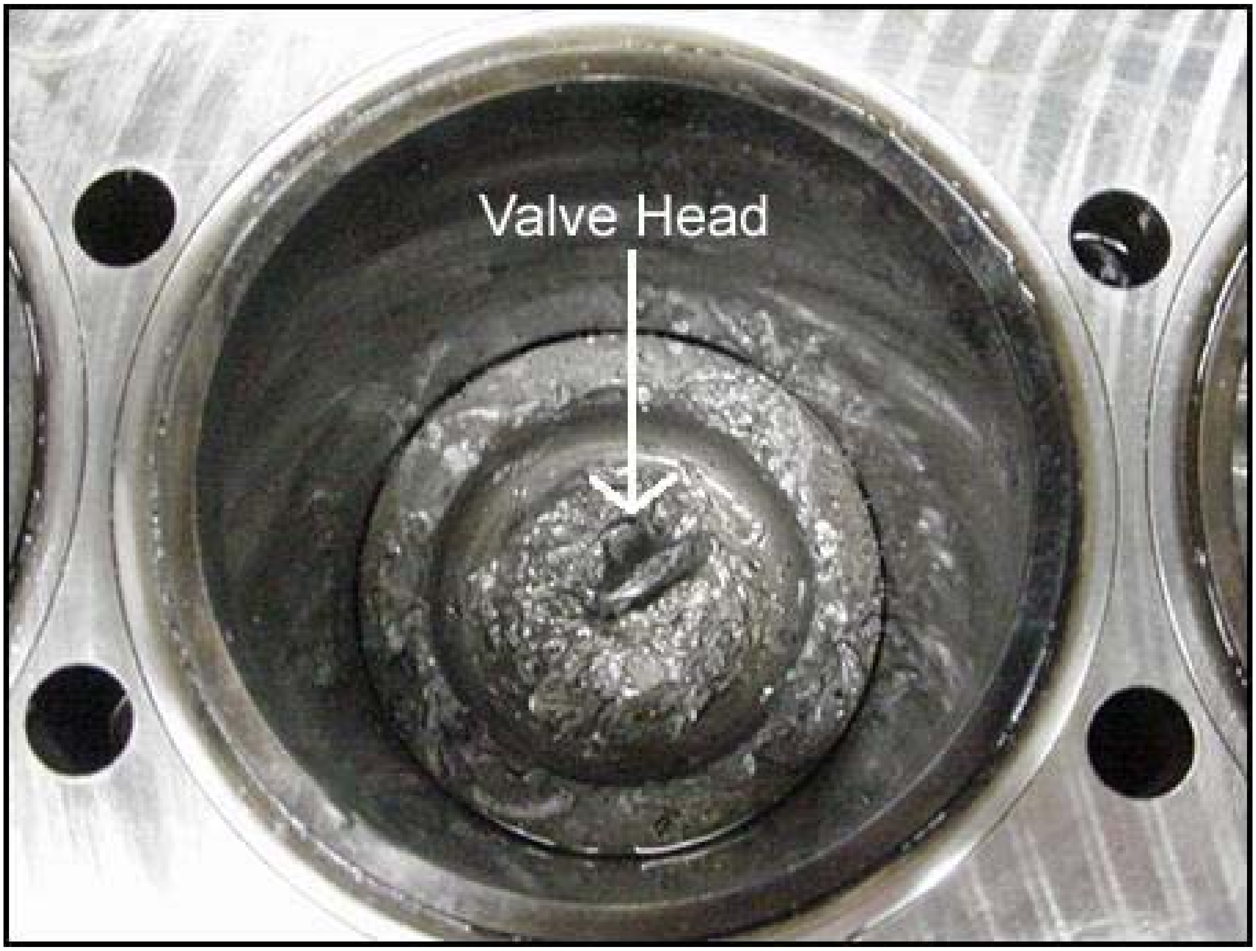


**Figure 13. Valve Head Embedded in the Piston**

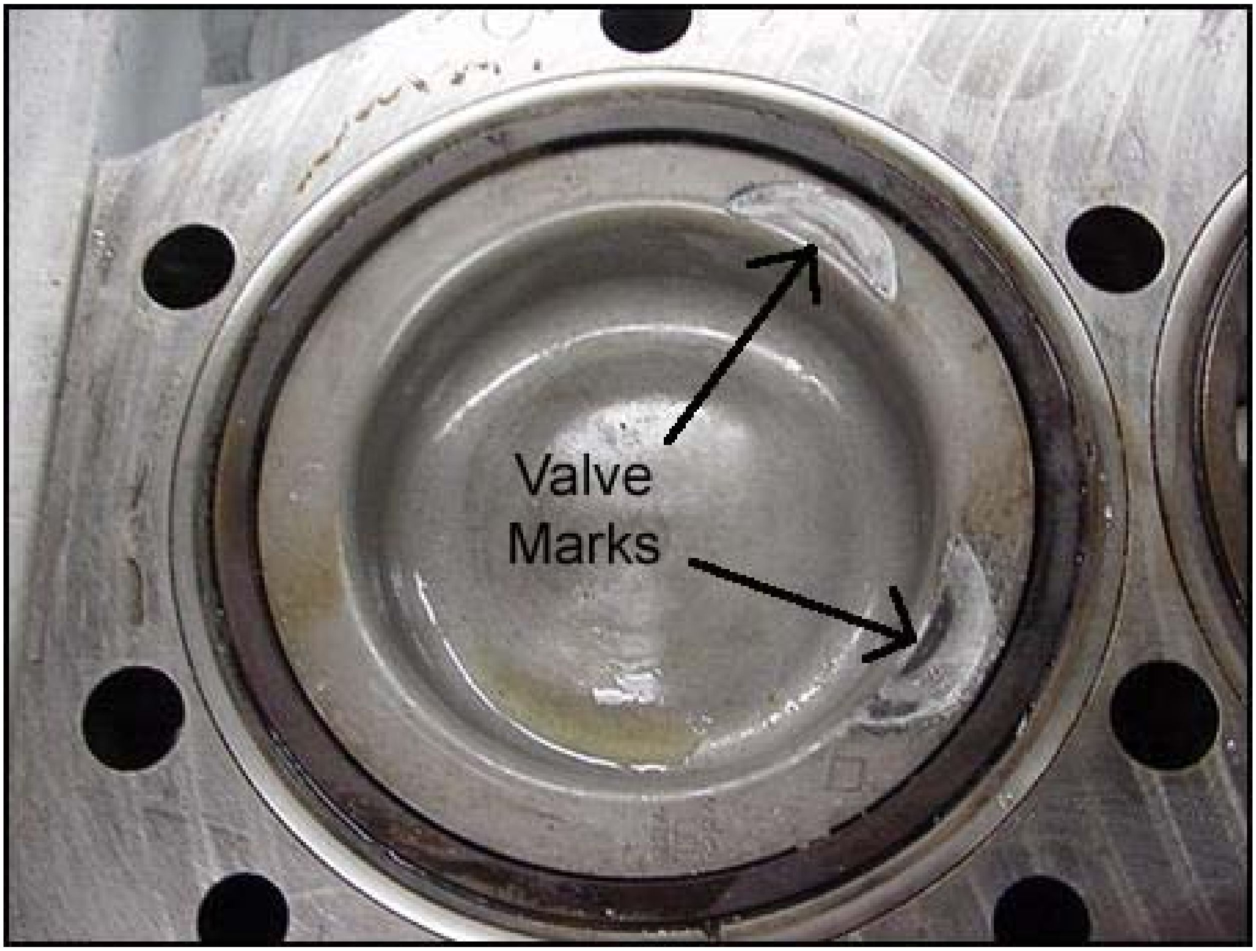


**Figure 14. Marks on the Piston from Valve/Piston Collisions**

The second problem that occurred during testing was foreign material entering the engine and embedding itself on the number one piston. A picture of the foreign material is shown in Figure 15. The foreign material appeared to be valve material from the previous valve failure. It is assumed that the material had lodged itself somewhere in the engine before the rebuild, and dislodged when running the engine after the rebuild. The number one piston was replaced and the intake valves and valve seats on the number one cylinder were reground.

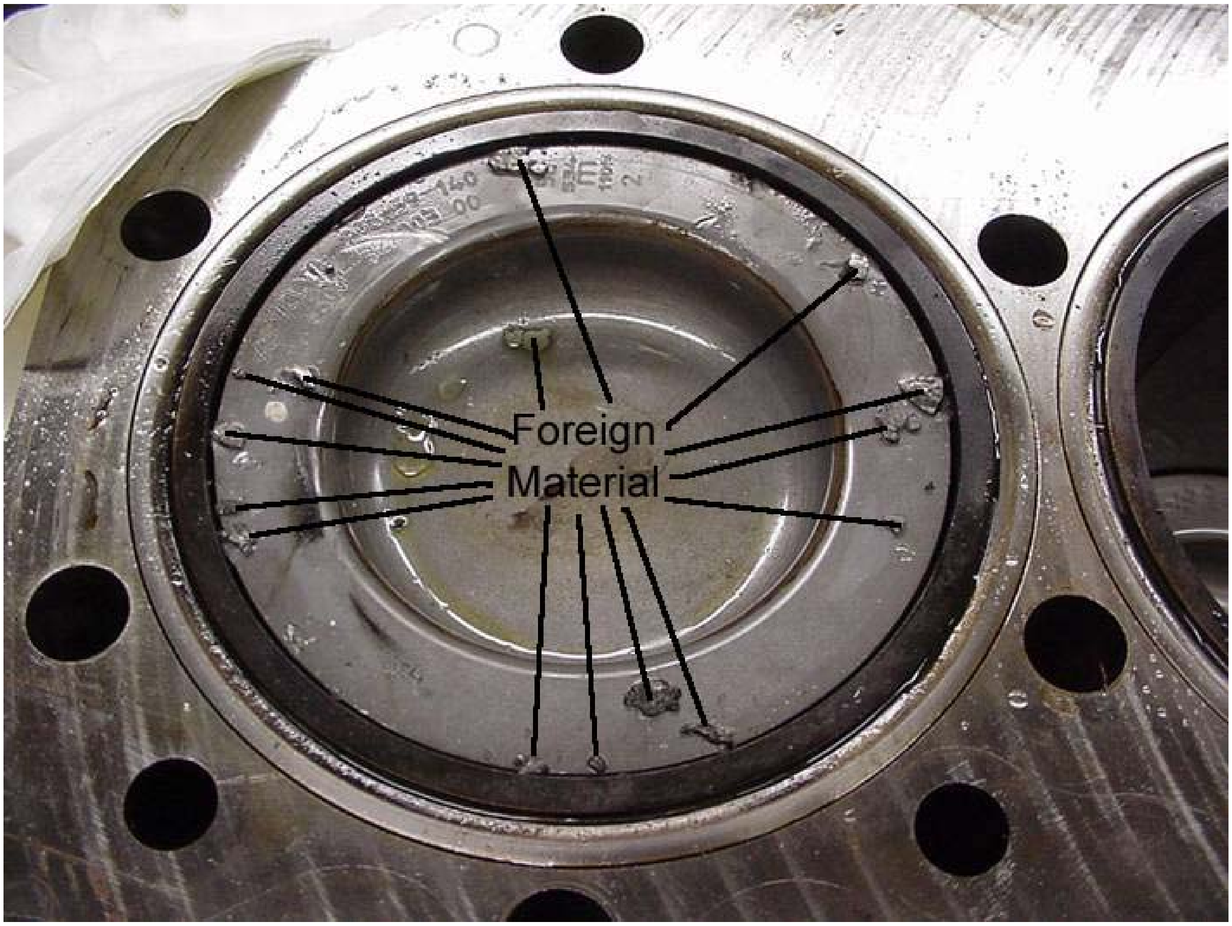


**Figure 15. Foreign Material on the Number One Piston**

The third problem that occurred was excessive valve wear. Figure 16 shows one of the valves, but all of the valves showed excessive wear after approximately 50 hours of testing. The head was rebuilt with new MD11 valve seat inserts and valves. The MG9 valve seats, which were originally in the head, induce more swirl than the MD11 valve seats. The excessive wear was a result of material incompatibility, where the diesel MD11 valves were used with the natural gas MG9 valve seats. The head was rebuilt with new MD11 valve seat inserts and valves. Changing the valve seats from the original MG9 design to the MD11 design cause a reduction in the in cylinder swirl. Only the preliminary data was run with the MG9 seats.

**Figure 16. Excessive Valve Wear**

The fourth problem was noisy signals on the HVA system. Both the command signals for the actuators and the feedback signals were noisy. A picture from an oscilloscope of the feedback signal on one of the valves is shown in Figure 17. This data were recorded in a simulation mode (engine not actually running, but the HVA system detects that it is running). This figure shows that the early intake valve opening was caused by a noisy command signal. The HVA system was also detecting valve to piston collisions when the intake valve ramp and hold function was turned on. The clearance between the valve and piston is small when the piston is at top dead center (TDC) and the intake valve opened. If the noise on the feedback system is larger than the clearance, the Sturman system will detect a collision and disable the valve. This problem was solved by insulating the metal connectors under the valve cover for both the command signals and feedback signals.

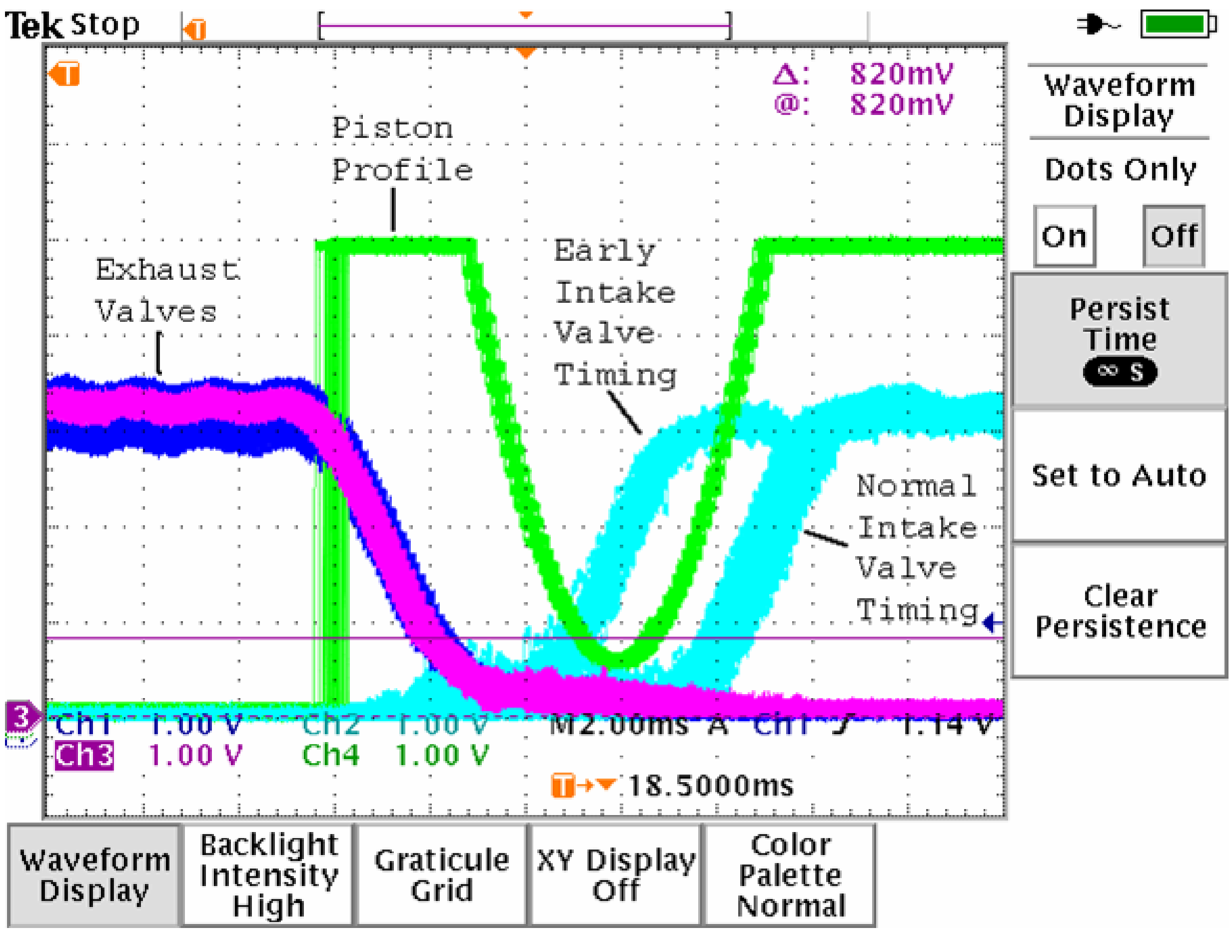


**Figure 17. Oscilloscope Picture Showing an Early Intake Valve Opening Command**

## 2.3 Valve Timing Sweeps

With the Sturman HVA system, the time needed for testing with all the possible valve timing configurations (as described in section 2.1.3) is much greater than the time allowed for this project. Therefore reduced testing as described in the following sections was conducted.

### *2.3.1 Preliminary Valve Timing Sweeps*

Preliminary valve timing sweeps were conducted on the engine after the HVA system installation was completed but before all of the other modifications were performed. The engine had a low pressure EGR loop, the center fill manifold, and the 10.25:1 compression ratio pistons. The purpose of these preliminary timing sweeps was to show the efficiency improvements of late and early intake valve closing at light loads. A late or early intake valve closing at light loads reduces the effective compression ratio but also reduces pumping losses. Figure 18 shows a 4.7% improvement in efficiency for a late intake valve closing and a 6.2% improvement for an early intake valve closing. The exhaust valve opening, exhaust valve closing, and intake valve opening were set to simulate a normal cam. Both intake and exhaust valve lifts were set to 10 mm and the intake ramp and hold was turned off.

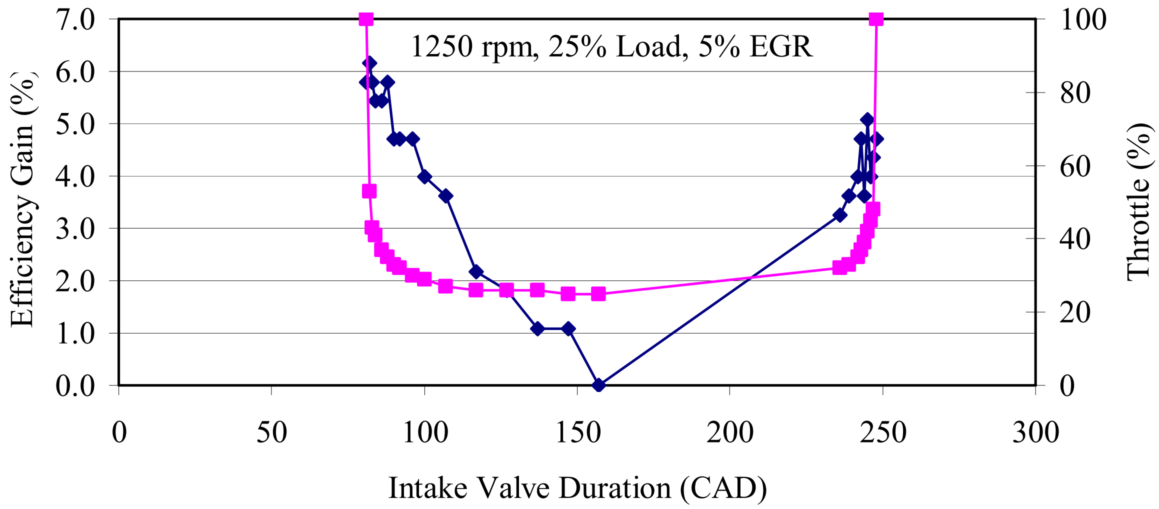


**Figure 18. Preliminary Testing for Late and Early IVC**

### *2.3.2 Valve Timing Sweeps*

Valve timing sweeps were conducted on the HVA variable timing engine as shown below.

- Vary exhaust valve opening
  - Exhaust valve closing set to simulate a normal cam
  - Intake valve opening set to simulate a normal cam
  - Intake valve closing used to set torque though an early closing at light loads or set to simulate a normal cam at medium to high loads with early intake valve closing used to lower the effective compression ratio at the highest loads where knock is encountered
  - Intake and exhaust valve lifts set for 10 mm, unless reduced exhaust valve lift was needed to lower exhaust temperatures
  - Intake ramp and hold set for the minimum duration, unless it was turned off to lower exhaust temperatures

- Vary Exhaust Valve Closing
  - Exhaust valve opening set for maximum efficiency based on above results
  - Intake valve opening set to simulate a normal cam
  - Intake valve closing used to set torque though an early closing at light loads or set to simulate a normal cam at medium to high loads with early intake valve closing used to lower the effective compression ratio at the highest loads where knock is encountered
  - Intake and exhaust valve lifts set for 10 mm, unless reduced exhaust valve lift was needed to lower exhaust temperatures
  - Intake ramp and hold set for the minimum duration, unless it was turned off to lower exhaust temperatures

- Vary Intake Valve Opening
  - Exhaust valve opening set for maximum efficiency based on above results
  - Exhaust valve closing set for maximum efficiency based on above results
  - Intake valve closing used to set torque though an early closing at light loads or set to simulate a normal cam at medium to high loads with early intake valve closing used to lower the effective compression ratio at the highest loads where knock is encountered
  - Intake and exhaust valve lifts set for 10 mm, unless reduced exhaust valve lift was needed to lower exhaust temperatures
  - Intake ramp and hold set for the minimum duration, unless it was turned off to lower exhaust temperatures

### *2.3.2.1 Full Timing Sweeps*

Testing began with Mode 15 of the 22 modes, which is 1600 rpm, 50% load (700 N-m). Results are shown in Figures 19-21. All of these test points are at a 100% throttle with load controlled by an early intake valve closing. Figure 19 shows the effect of varying the exhaust valve opening. Efficiency peaks at an opening of 164° for 5% EGR, 152° for 10% EGR, and 168° for 15% EGR. Figure 20 shows the effect of varying the exhaust valve closing using the peak efficiency point for exhaust valve opening. The trend for exhaust valve closing is increasing efficiency with a later closing. The latest exhaust valve closing is limited by the piston to valve clearance. Figure 21 shows the effect of varying the intake valve opening using the peak efficiency point for exhaust valve opening and closing. Efficiency peaks at an opening of 374° for both 5% and 10% EGR and 390° for 15% EGR. For all of these curves, efficiency differences are small.

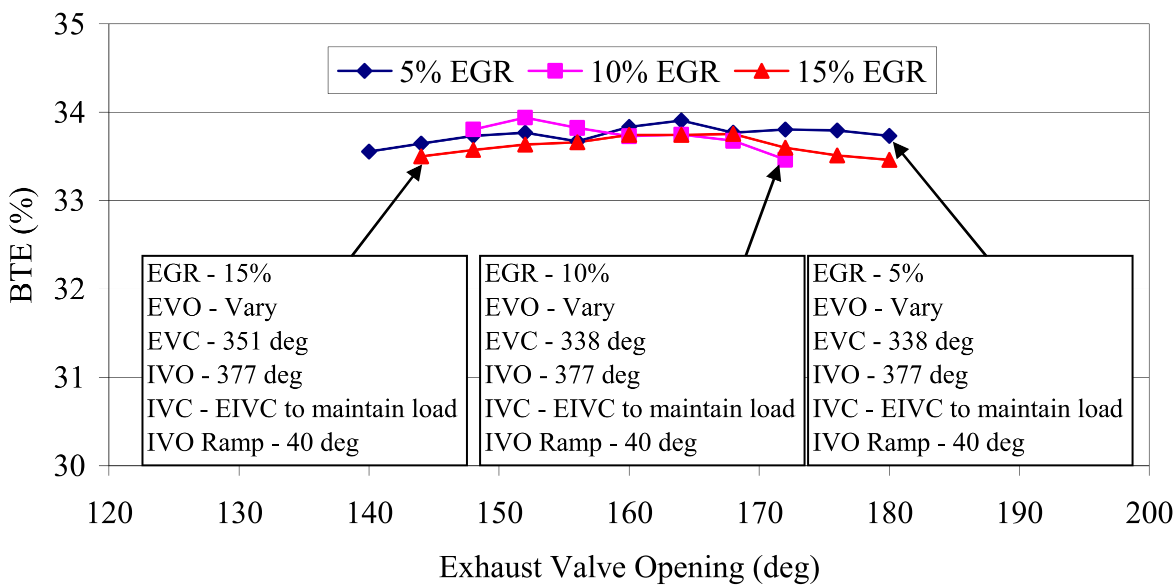


**Figure 19. Mode 15 of the 22 Mode Test, 1600 rpm, 50% Load Varying Exhaust Valve Opening**

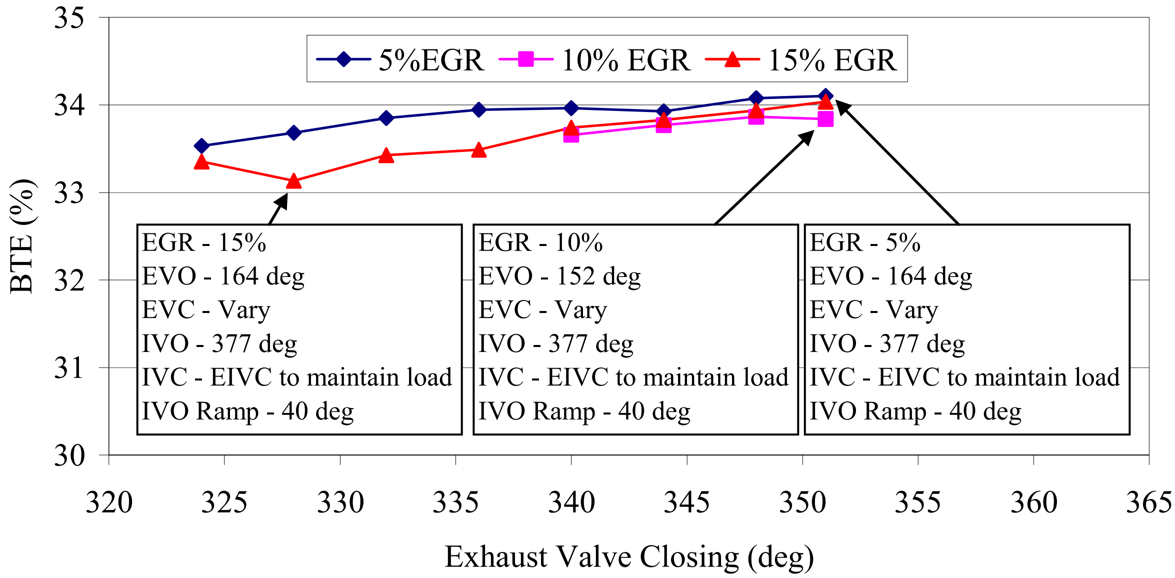


**Figure 20. Mode 15 of the 22 Mode Test, 1600 rpm, 50% Load**
**Varying Exhaust Valve Closing**

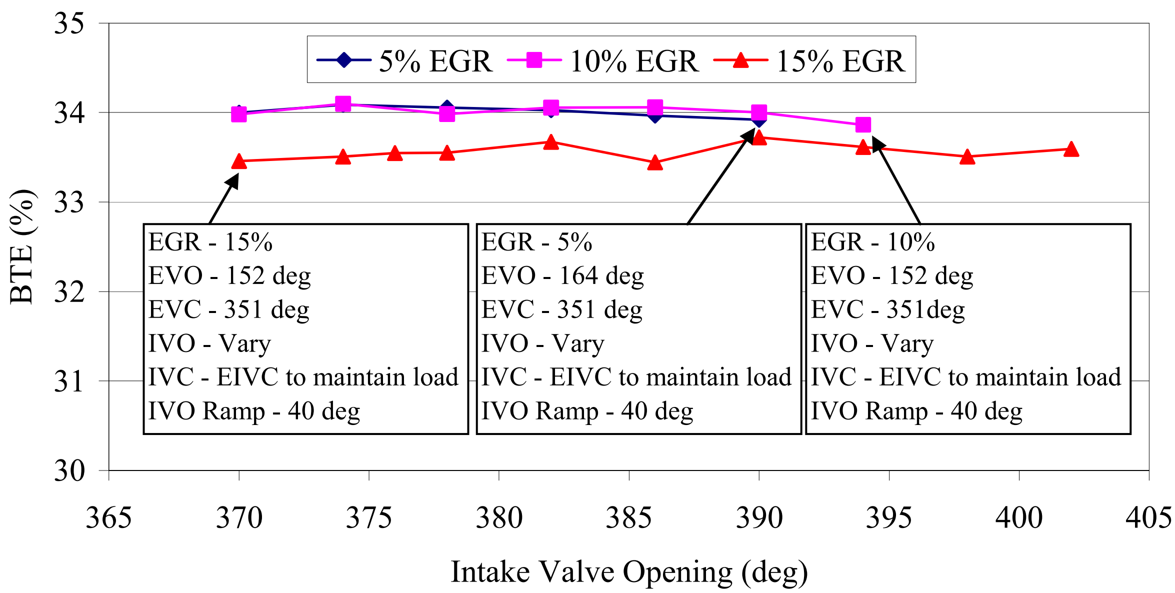


**Figure 21. Mode 15 of the 22 Mode Test, 1600 rpm, 50% Load**
**Varying Intake Valve Opening**

Full timing sweeps continued with modes 9, 19, 11, 14, and 10. Similar efficiency trends were observed for these modes, the results of all the full timing sweeps are presented in Appendix A. The full timing sweep tests indicate that:

- Exhaust valve opening has little effect on efficiency within the range of openings tested. Further changes in exhaust valve opening, beyond the range tested, are expected to reduce efficiency

- Exhaust valve closing should be as late as possible for highest efficiency

- Intake valve opening should be close to as early as possible for highest efficiency

The speeds and loads for these modes are shown in Figure 22.

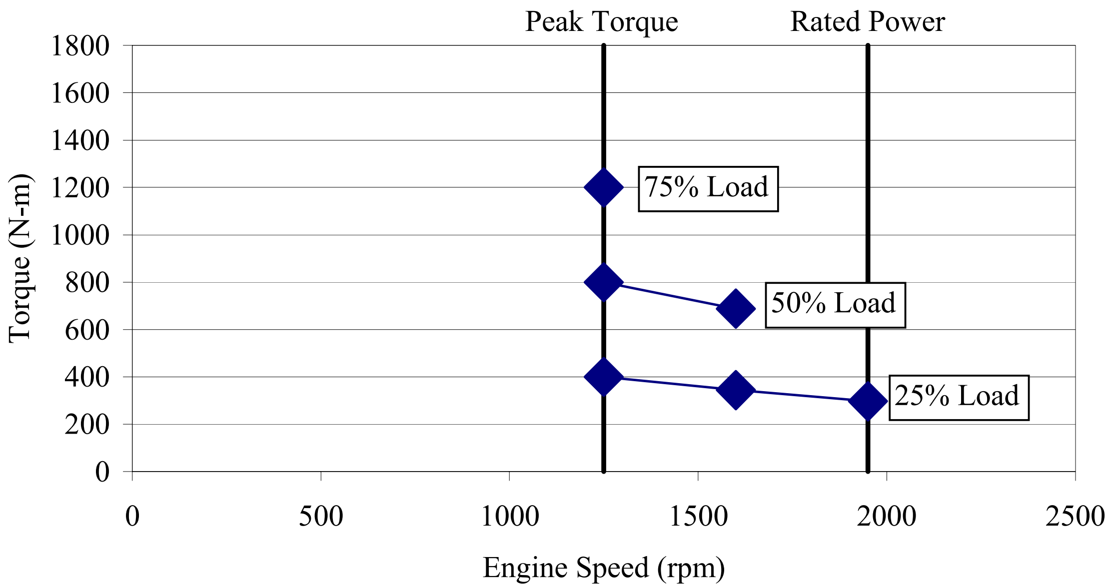


**Figure 22. Full Timing Sweeps Test Points**

### *2.3.2.2 Reduced Timing Sweeps*

Based on the information from the full timing sweeps, it was decided to shorten the testing at each mode to three exhaust valve opening timings, three exhaust valve closing timings, and three intake valve opening timings at a single EGR rate. The valve timings are shown in Table 4, and the EGR rates will be determined by previous testing on the Mack E7GT engine (NREL Report No. SR-540-38222). The latest possible exhaust valve closing and earliest possible intake valve opening are limited by piston to valve clearance.

**TABLE 4. REDUCED VALVE TIMING MATRIX**

| EVO | EVC | IVO |
|---|---|---|
| 160 degrees | The latest possible setting | +4 degrees from the earliest possible setting |
| 164 degrees | -4 degrees from the latest possible setting | +8 degrees from the earliest possible setting |
| 168 degrees | -8 degrees from the latest possible setting | +12 degrees from the earliest possible setting |

Results at Mode 16, which is 1600 rpm, 75% load (1032 N-m), are shown in Figures 23-25 for varying exhaust valve opening (EVO), exhaust valve closing (EVC) and intake valve opening (IVC) respectively. The throttle was set to 100%, the intake valve closing set to 536°, and the wastegate was used to set the load.

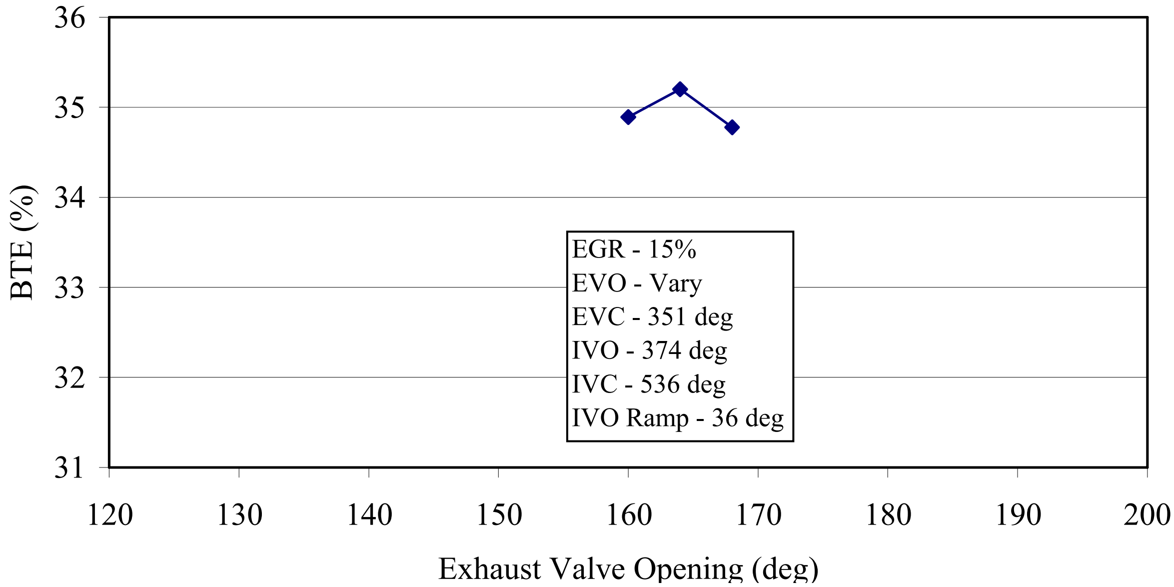


**Figure 23. Mode 16 of the 22 Mode Test, 1600 rpm, 75% Load Varying Exhaust Valve Opening**

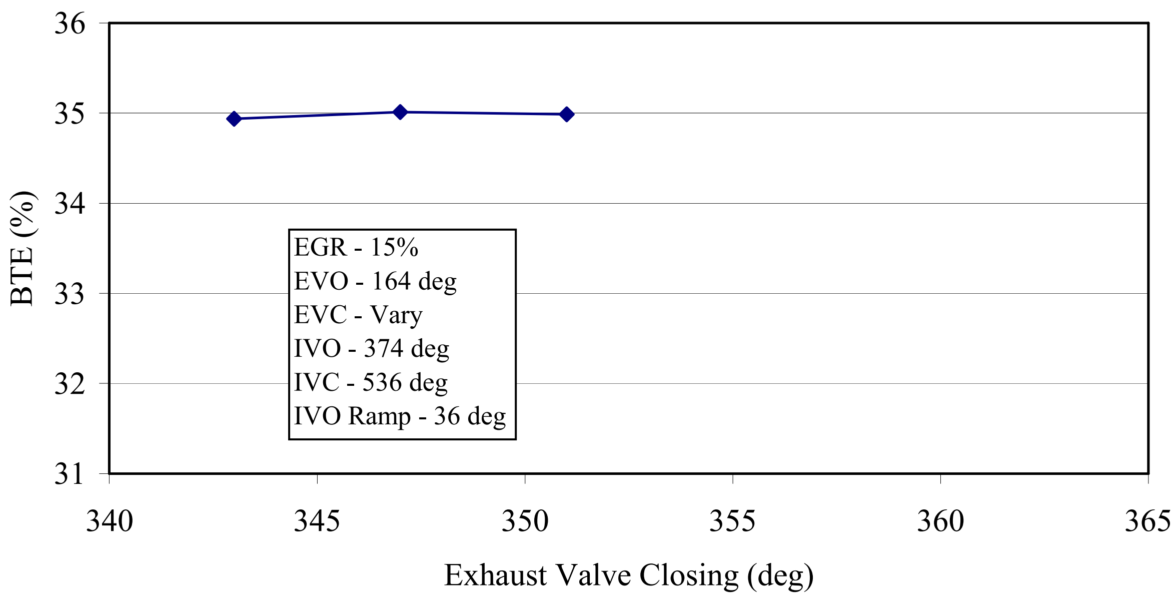


**Figure 24. Mode 16 of the 22 Mode Test, 1600 rpm, 75% Load Varying Exhaust Valve Closing**

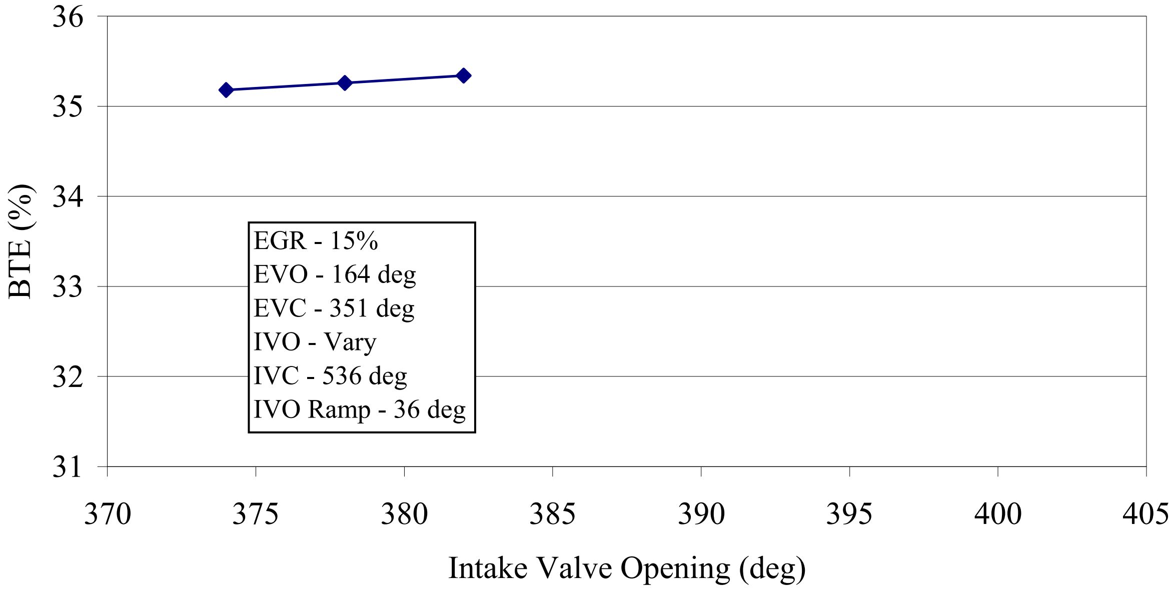


**Figure 25. Mode 16 of the 22 Mode Test, 1600 rpm, 75% Load Varying Intake Valve Opening**

Reduced timing sweeps continued with modes 16, 8, 13, 18, 12, 20, and 21. Similar efficiency trends were observed for these modes, the results of all the reduced timing sweeps are presented in Appendix B. Single test points were recorded for modes 22, 17, and 1. Modes 22 and 17 are 100% load at 1950 and 1600 rpm respectively. At these test points, there was no knock margin, and only one test point was recorded. Mode 1 is idle, and only one test point was recorded due to time and budget constraints. Results for the single point tests are included in Appendix C.

### *2.3.3 Final Calibration*

Based on the above results, emissions tests at each of the modes were conducted at the highest efficiency point for each mode. The emissions and efficiency results for each mode are shown in Table 6. The composite (weighted) emissions for the 13-mode are 0.005 g/kw-hr of $NO_x$, 1.166 g/kW-hr of CO, and 0.536 g/kW-hr of THC.

## TABLE 6. EFFICIENCY AND EMISSIONS RESULTS

| Mode * | Speed | Load | Speed (rpm) | Torque (N-m) | MG11 BTE (%) | $NO_x$ After Cat (g/kW-hr) | CO After Cat (g/kW-hr) | THC After Cat (g/kW-hr) |
|---|---|---|---|---|---|---|---|---|
| 1 (1) | 0% | 0% | 600 | 160 | 17.5 | 0.015 | 1.337 | 10.165 |
| 8 | 50% | 15% | 1250 | 240 | 24.1 | 0.087 | 0.555 | 0.116 |
| 9 (7) | 50% | 25% | 1250 | 400 | 28.6 | 0.001 | 0.664 | 0.095 |
| 10 (5) | 50% | 50% | 1250 | 800 | 33.4 | 0.002 | 0.055 | 0.060 |
| 11 (6) | 50% | 75% | 1250 | 1200 | 36.3 | 0.005 | 1.621 | 0.299 |
| 12 (2) | 50% | 100% | 1250 | 1600 | 37.9 | 0.006 | 1.707 | 0.427 |
| 13 | 75% | 15% | 1600 | 206 | 20.9 | 0.003 | 1.570 | 0.000 |
| 14 (9) | 75% | 25% | 1600 | 344 | 26.3 | 0.003 | 0.785 | 0.019 |
| 15 (3) | 75% | 50% | 1600 | 688 | 32.9 | 0.006 | 1.967 | 0.317 |
| 16 (4) | 75% | 75% | 1600 | 1032 | 35.3 | 0.003 | 0.166 | 0.125 |
| 17 (8) | 75% | 100% | 1600 | 1376 | 37.6 | 0.004 | 1.754 | 0.775 |
| 18 | 100% | 15% | 1950 | 178 | 17.9 | 0.003 | 1.214 | 0.037 |
| 19 (11) | 100% | 25% | 1950 | 297 | 23.6 | 0.000 | 1.549 | 0.108 |
| 20 (13) | 100% | 50% | 1950 | 593 | 30.7 | 0.007 | 1.175 | 0.874 |
| 21 (12) | 100% | 75% | 1950 | 890 | 34.4 | 0.008 | 0.957 | 0.485 |
| 22 (10) | 100% | 100% | 1950 | 1186 | 35.3 | 0.007 | 0.819 | 0.572 |
| Weighted 13-Mode | | | | | 33.9 | 0.005 | 1.166 | 0.536 |
| * Number in parentheses is the corresponding mode for the 13-mode test | | | | | | | | |

The operational settings for the above data are shown in Table 7.

## TABLE 7. OPERATIONAL SETTINGS

| Mode | EVO (°) | EVC (°) | IVO (°) | IVC (°) | Ramp (°) | EGR (%) | Spark Timing (°btdc) | In Lift (mm) | Ex Lift (mm) |
|---|---|---|---|---|---|---|---|---|---|
| 1 | 164 | 344 | 380 | 429 | 0 | 0 | 9 | 10 | 8 |
| 8 | 164 | 345 | 376 | 444 | 28 | 5 | 22 | 10 | 10 |
| 9 | 162 | 343 | 376 | 465 | 28 | 5 | 20 | 10 | 10 |
| 10 | 164 | 349 | 376 | 536 | 28 | 10 | 19 | 10 | 10 |
| 11 | 160 | 349 | 378 | 536 | 28 | 10 | 19 | 10 | 10 |
| 12 | 164 | 345 | 376 | 489 | 28 | 10 | 21 | 10 | 10 |
| 13 | 160 | 351 | 374 | 450 | 36 | 5 | 24 | 7 | 10 |
| 14 | 164 | 343 | 382 | 467 | 36 | 5 | 20 | 10 | 10 |
| 15 | 164 | 351 | 377 | 509 | 36 | 5 | 21 | 10 | 10 |
| 16 | 164 | 351 | 378 | 536 | 36 | 15 | 26 | 10 | 10 |
| 17 | 168 | 351 | 374 | 480 | 0 | 12.5 | 28 | 9 | 10 |
| 18 | 152 | 348 | 373 | 456.5 | 44 | 5 | 25 | 7 | 10 |
| 19 | 152 | 351 | 373 | 481 | 44 | 10 | 26 | 6 | 10 |
| 20 | 160 | 351 | 373 | 499 | 0 | 10 | 25 | 10 | 7 |
| 21 | 160 | 351 | 373 | 485 | 0 | 12.5 | 30 | 7 | 7 |
| 22 | 168 | 351 | 373 | 475 | 0 | 14 | 32 | 7 | 9 |

The emissions results before the catalyst for each mode are shown in Table 8. The composite (weighted) emissions for the 13-mode are 6.896 g/kw-hr of $NO_x$, 13.178 g/kW-hr of CO, and 6.933 g/kW-hr of THC. The conversion efficiencies of the TWC are over 99% for $NO_x$, 91% for CO, and 92% for THC.

**TABLE 8. BEFORE CATALYST EMISSIONS RESULTS**

| Mode * | Speed | Load | Speed (rpm) | Torque (N-m) | $NO_x$ Before Cat (g/kW-hr) | CO Before Cat (g/kW-hr) | THC Before Cat (g/kW-hr) |
|---|---|---|---|---|---|---|---|
| 1 (1) | 0% | 0% | 600 | 160 | 4.308 | 142.094 | 31.517 |
| 8 | 50% | 15% | 1250 | 240 | 6.548 | 11.416 | 4.936 |
| 9 (7) | 50% | 25% | 1250 | 400 | 10.377 | 13.984 | 6.628 |
| 10 (5) | 50% | 50% | 1250 | 800 | 5.811 | 11.290 | 7.372 |
| 11 (6) | 50% | 75% | 1250 | 1200 | 5.689 | 14.709 | 7.897 |
| 12 (2) | 50% | 100% | 1250 | 1600 | 9.053 | 12.530 | 5.995 |
| 13 | 75% | 15% | 1600 | 206 | 5.693 | 16.911 | 9.987 |
| 14 (9) | 75% | 25% | 1600 | 344 | 6.991 | 11.292 | 5.479 |
| 15 (3) | 75% | 50% | 1600 | 688 | 5.046 | 18.099 | 9.849 |
| 16 (4) | 75% | 75% | 1600 | 1032 | 5.586 | 11.648 | 6.154 |
| 17 (8) | 75% | 100% | 1600 | 1376 | 2.827 | 19.112 | 12.202 |
| 18 | 100% | 15% | 1950 | 178 | 6.861 | 12.359 | 7.299 |
| 19 (11) | 100% | 25% | 1950 | 297 | 8.621 | 13.464 | 7.690 |
| 20 (13) | 100% | 50% | 1950 | 593 | 4.308 | 142.094 | 31.517 |
| 21 (12) | 100% | 75% | 1950 | 890 | 6.548 | 11.416 | 4.936 |
| 22 (10) | 100% | 100% | 1950 | 1186 | 10.377 | 13.984 | 6.628 |
| Weighted 13-Mode | | | | | 6.764 | 14.213 | 7.024 |
| * Number in parenthesis is the corresponding mode for the 13-mode test | | | | | | | |

The emissions target for $NO_x$ for this engine has been achieved, but efficiencies do not show an improvement over the Mack E7GT stoichiometric EGR engine (efficiency comparisons are presented in the following sections). Particulate matter was not measured due to time and budget constraints, but is expected to be below the target. The efficiencies for the Volvo 11 liter HVA engine should show a higher efficiency than an engine with a cam due to both a higher compression ratio and reduced pumping losses (throttleless operation, particularly at light loads). In addition, the efficiency comparison shown in Table 6 does not include valve train losses (power needed to run the hydraulic pump) for the Volvo 11 liter HVA engine. These efficiency results are not higher than a typical natural gas engine such as the Mack E7GT. The major reason for the difference in efficiencies is that the E7GT is a mature engine whereas the Volvo 11 liter is a new gas engine which has not been optimized.

## 2.4 Baseline Testing

To evaluate the efficiency improvement an HVA system can provide, baseline tests were conducted on the MG11 engine. The 14.3:1 compression ratio pistons were replaced with the original 10.25:1 pistons, which is the same compression ratio used on the MG9 engine. Tests at the 16 modes were conducted with valve timings set to approximate a cam. The throttle was used to control torque where the wastegate was fully open at light loads and was used to control manifold air pressure at higher loads. The valve timings and efficiencies for the baseline tests are shown in Table 9.

**TABLE 9. BASELINE TEST CONDITIONS**

| Mode | Speed (rpm) | Load (%) | EVO (deg) | EVC (deg) | IVO (deg) | IVC (deg) | Ramp (deg) | Int Valve Lift (mm) | Exh Valve Lift (mm) | BTE (%) |
|---|---|---|---|---|---|---|---|---|---|---|
| 1 | 600 | 0% | 168 | 347 | 379 | 536 | 0 | 10 | 10 | 17.7 |
| 8 | 1250 | 15% | 164 | 349 | 376 | 545 | 28 | 10 | 10 | 20.8 |
| 9 | 1250 | 25% | 164 | 349 | 376 | 545 | 28 | 10 | 10 | 25.5 |
| 10 | 1250 | 50% | 164 | 349 | 376 | 545 | 28 | 10 | 10 | 31.5 |
| 11 | 1250 | 75% | 164 | 349 | 376 | 545 | 0 | 8 | 8 | 34.1 |
| 12 | 1250 | 100% | 164 | 349 | 376 | 545 | 0 | 5/10 | 10 | 35.7 |
| 13 | 1600 | 15% | 162 | 350 | 374.5 | 550 | 36 | 10 | 10 | 19.2 |
| 14 | 1600 | 25% | 162 | 350 | 374.5 | 550 | 36 | 10 | 10 | 24.3 |
| 15 | 1600 | 50% | 162 | 350 | 374.5 | 550 | 0 | 5/10 | 10 | 29.9 |
| 16 | 1600 | 75% | 162 | 350 | 374.5 | 550 | 0 | 5/10 | 10 | 32.2 |
| 17 | 1600 | 100% | 162 | 350 | 374.5 | 550 | 0 | 5/10 | 10 | 33.6 |
| 18 | 1950 | 15% | 160 | 351 | 373 | 555 | 44 | 10 | 10 | 16.7 |
| 19 | 1950 | 25% | 160 | 351 | 373 | 555 | 0 | 5/10 | 10 | 21.3 |
| 20 | 1950 | 50% | 160 | 351 | 373 | 555 | 0 | 5/10 | 10 | 27.7 |
| 21 | 1950 | 75% | 160 | 351 | 373 | 555 | 0 | 5/10 | 10 | 30.3 |
| 22 | 1950 | 100% | 160 | 351 | 373 | 555 | 0 | 5/10 | 10 | 33.4 |

## 2.5 Results Analysis

The efficiency improvements of an HVA system on a natural gas engine are discussed below.

### *2.5.1 Baseline Efficiency*

The baseline testing on the MG11 engine was conducted with the HVA valve timing set to simulate a normal cam and without HVA hydraulics pumping losses included in the efficiency calculation. To obtain a more accurate baseline, simulations were conducted on this engine to obtain the difference between the valve profile of using the HVA system to simulate a cam and the actual valve lift profile with a cam. Figure 26 shows the valve lift difference between the HVA system and a cam.

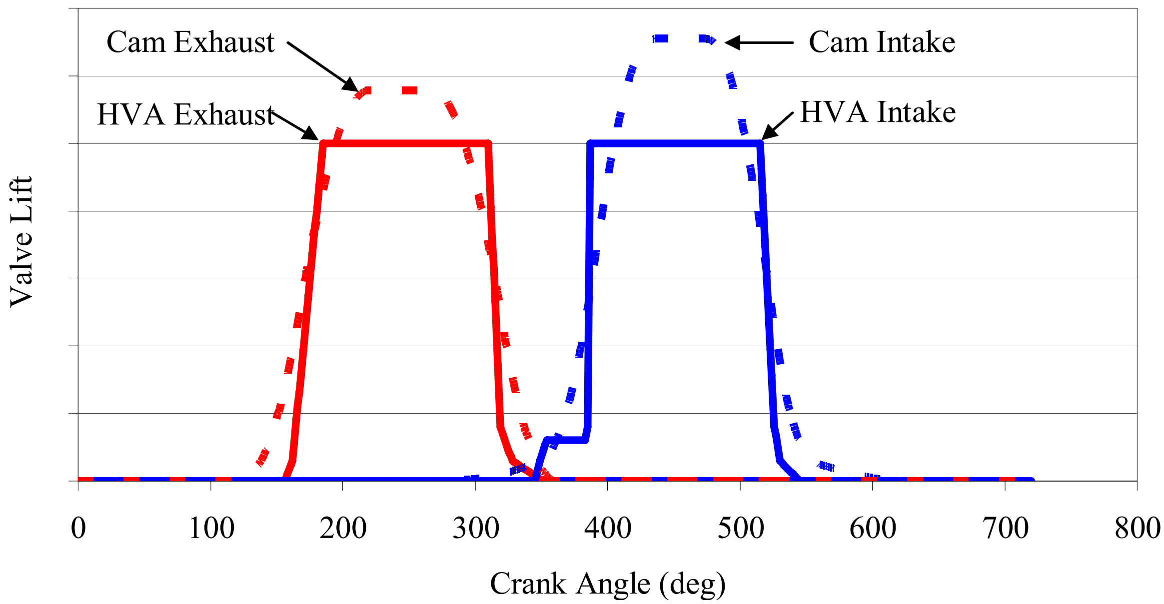


**Figure 26. Valve Lift Profile for the HVA system and a Cam**

The simulation provides a relative performance difference between the HVA system and a cam, therefore the relative change in efficiency from the simulations were applied to the test data. The efficiency data represents a normal cam, but still does not include the cam losses. Cam losses were applied to the data to arrive at a baseline efficiency which represents an engine with a cam. The results are shown in Table 10. The last column represents the baseline case. Idle was not modeled therefore a relative efficiency change was not applied at idle.

**TABLE 10. BASELINE EFFICIENCY RESULTS**

| Mode | Speed (rpm) | Load (%) | BTE from Test Data, HVA Fixed Timing (%) | BTE from HVA Sim (%) | BTE from Cam Sim (%) | BTE w/ cam w/o cam losses (%) | BTE w/cam & cam losses (%) |
|---|---|---|---|---|---|---|---|
| 1 | 600 | 0% | 17.7 | - | - | 17.7 | 17.0 |
| 8 | 1250 | 15% | 20.8 | 23.0 | 23.3 | 21.0 | 20.5 |
| 9 | 1250 | 25% | 25.5 | 26.9 | 27.0 | 25.6 | 25.2 |
| 10 | 1250 | 50% | 31.5 | 33.6 | 34.1 | 31.9 | 31.6 |
| 11 | 1250 | 75% | 34.1 | 35.3 | 35.5 | 34.2 | 34.1 |
| 12 | 1250 | 100% | 35.7 | 38.8 | 39.2 | 36.0 | 35.9 |
| 13 | 1600 | 15% | 19.2 | 20.9 | 21.9 | 20.1 | 19.4 |
| 14 | 1600 | 25% | 24.3 | 24.8 | 25.2 | 24.7 | 24.2 |
| 15 | 1600 | 50% | 29.9 | 31.8 | 31.3 | 29.4 | 29.1 |
| 16 | 1600 | 75% | 32.2 | 33.6 | 34.7 | 33.3 | 33.1 |
| 17 | 1600 | 100% | 33.6 | 34.5 | 35.2 | 34.2 | 34.1 |
| 18 | 1950 | 15% | 16.7 | 17.8 | 18.0 | 16.9 | 16.3 |
| 19 | 1950 | 25% | 21.3 | 21.7 | 21.7 | 21.3 | 20.9 |
| 20 | 1950 | 50% | 27.7 | 28.3 | 28.5 | 27.9 | 27.5 |
| 21 | 1950 | 75% | 30.3 | 31.6 | 32.0 | 30.7 | 30.5 |
| 22 | 1950 | 100% | 33.4 | 31.6 | 32.3 | 34.1 | 33.9 |
| Weighted 13-Mode | | | 31.1 | | | | 31.3 |

### 2.5.2 *HVA Efficiency*

The HVA testing on the MG11 engine was conducted without HVA hydraulics pumping losses included in the efficiency calculation. Pumping requirements were obtained on this system (a research type system) and applied to the data. Table 11 shows the HVA efficiency results. The last column represents an HVA system with hydraulic pumping losses included.

**TABLE 11. HVA EFFICIENCY RESULTS**

| Mode | Speed (rpm) | Load (%) | BTE from Test Data, HVA Variable Timing (%) | BTE w/Pumping losses (%) |
|---|---|---|---|---|
| 1 | 600 | 0% | 17.5 | 13.3 |
| 8 | 1250 | 15% | 24.1 | 20.4 |
| 9 | 1250 | 25% | 28.6 | 26.0 |
| 10 | 1250 | 50% | 33.4 | 31.8 |
| 11 | 1250 | 75% | 36.3 | 35.2 |
| 12 | 1250 | 100% | 37.9 | 37.0 |
| 13 | 1600 | 15% | 20.9 | 17.4 |
| 14 | 1600 | 25% | 26.3 | 23.6 |
| 15 | 1600 | 50% | 32.9 | 31.2 |
| 16 | 1600 | 75% | 35.3 | 34.1 |
| 17 | 1600 | 100% | 37.6 | 36.6 |
| 18 | 1950 | 15% | 17.9 | 14.3 |
| 19 | 1950 | 25% | 23.6 | 20.8 |
| 20 | 1950 | 50% | 30.7 | 28.8 |
| 21 | 1950 | 75% | 34.0 | 33.0 |
| 22 | 1950 | 100% | 35.3 | 34.2 |
| Weighted 13-Mode | | | 33.9 | 32.4 |

### 2.5.3 *Optimized Natural gas Efficiency*

The efficiency data from the Mack E7GT data was used as an optimized natural gas engine. The E7GT engine has an 11.5:1 compression ratio, whereas efficiency improvements with the HVA system will be compared to a baseline engine with a 10.25:1 compression. The E7GT efficiency data was modified to reflect the efficiency at a compression ratio of 10.25:1, so that the efficiency improvement is applied to an engine with the same compression ratio as the baseline.

The theoretical efficiency is given by the following equation:

$$\eta = 1 - \frac{1}{CR^{(\gamma - 1)}}$$

Where:

CR = compression ratio
$\gamma$ = ratio of specific heats = 1.4

For a 10.25:1 compression ratio, the theoretical efficiency is 60.6%, and for an 11.5:1 compression ratio, the theoretical efficiency is 62.4%. The efficiency of the 10.25:1 compression ratio is 97% of the efficiency of the 11.5:1 compression ratio. Although the actual efficiency numbers are unrealistic, the relative change in efficiency is realistic. Therefore the efficiency of the E7GT engine was reduced by 3% to arrive at the 10.25:1 compression ratio E7GT efficiency. The efficiencies are shown in Table 12.

**TABLE 12. E7GT EFFICIENCY RESULTS**

| Mode | Speed (rpm) | Load (%) | E7GT 11.5 CR BTE (%) | E7GT 10.25 CR BTE (%) |
|---|---|---|---|---|
| 1 | 600 | 0% | 21.3 | 20.7 |
| 8 | 1250 | 15% | 25.2 | 24.5 |
| 9 | 1250 | 25% | 30.6 | 29.7 |
| 10 | 1250 | 50% | 35.9 | 34.9 |
| 11 | 1250 | 75% | 38.4 | 37.3 |
| 12 | 1250 | 100% | 39.1 | 38.0 |
| 13 | 1600 | 15% | 21.8 | 21.2 |
| 14 | 1600 | 25% | 28.2 | 27.4 |
| 15 | 1600 | 50% | 33.8 | 32.9 |
| 16 | 1600 | 75% | 36.8 | 35.8 |
| 17 | 1600 | 100% | 37.9 | 36.8 |
| 18 | 1950 | 15% | 18.8 | 18.2 |
| 19 | 1950 | 25% | 24.5 | 23.8 |
| 20 | 1950 | 50% | 31.8 | 30.9 |
| 21 | 1950 | 75% | 35.5 | 34.5 |
| 22 | 1950 | 100% | 36.6 | 35.5 |
| Weighted 13-Mode | | | 35.1 | 34.1 |

### *2.5.4 Diesel Efficiency*

Efficiency data for a 2010 diesel engine (same emissions level target as the MG11 HVA engine) are not currently available, but data for engines certified US EPA 2004 standards are.

The data for the heavy-duty engines are an average of four engines ranging from 12.0 to 15.2 liters (average of 14.0 liters). The data for the medium-duty engines are an average of three engines ranging from 5.9 to 8.8 liters (average of 6.9 liters). Since the 11 liter MG11 engine falls in between the medium and heavy duty engine size, the average of the two was used for the diesel comparison to the MG11 HVA engine. In addition, it is expected that efficiencies will be reduced by 2% for 2007, and will not change for 2010 (efficiency for 2010 includes both fuel costs and cost of operating an aftertreatment device).

Table 13 shows the 2004 and expected 2010 diesel efficiencies. Only data at the ESC 13-mode test points are available.

**TABLE 13. DIESEL EFFICIENCY**

| Mode | Speed (rpm) | Load (%) | 2004 Diesel BTE (%) | 2010 Diesel Estimated BTE (%) |
|---|---|---|---|---|
| 1 | 600 | 0% | 5.6 | 5.4 |
| 8 | 1250 | 15% | | |
| 9 | 1250 | 25% | 35.7 | 35.0 |
| 10 | 1250 | 50% | 38.2 | 37.4 |
| 11 | 1250 | 75% | 39.7 | 38.9 |
| 12 | 1250 | 100% | 40.7 | 39.9 |
| 13 | 1600 | 15% | | |
| 14 | 1600 | 25% | 33.2 | 32.5 |
| 15 | 1600 | 50% | 37.5 | 36.7 |
| 16 | 1600 | 75% | 39.2 | 38.4 |
| 17 | 1600 | 100% | 40.2 | 39.4 |
| 18 | 1950 | 15% | | |
| 19 | 1950 | 25% | 29.6 | 29.0 |
| 20 | 1950 | 50% | 34.7 | 34.0 |
| 21 | 1950 | 75% | 37.1 | 36.4 |
| 22 | 1950 | 100% | 38.0 | 37.3 |
| Weighted 13-Mode | | | 37.9 | 37.1 |

### *2.5.5 Efficiency Comparison*

A comparison of the efficiencies is shown in Table 14 and a bar chart of the weighted 13-mode results is shown in Figure 27. Each of the efficiency columns in Table 14 is described below.

1. This column is the estimated efficiency of the MG11 engine with a cam and 10.25:1 compression ratio pistons. It is estimated from actual data on the MG11 HVA engine using valve timings that simulate a fixed cam (hydraulic pumping losses are not included in this data). The efficiencies were then modified based on simulations of this engine with HVA valve timing and normal cam valve timing to get a efficiencies of an MG11 with a cam (but still without valve train losses). Finally, valvetrain losses of the cam system were applied to arrive at the MG11 Cam Baseline BTE.

2. This column shows the efficiency of the MG11 engine with the HVA system using valve timings that simulate a fixed cam and 10.25:1 compression ratio pistons. It is estimated from actual data on the MG11 HVA engine using valve timings that simulate a fixed cam (hydraulic pumping losses are not included in this data) and then an estimate of hydraulic pumping losses of the research oriented system were applied to arrive at the MG11 HVA Fixed Timing BTE.

3. This column shows the efficiency of the MG11 engine with the HVA system using variable valve timings and 14.3:1 compression ratio pistons. It is estimated from actual data on the MG11 HVA engine using variable valve timings (hydraulic pumping losses

are not included in this data). Hydraulic pumping losses of the research oriented system were applied to arrive at the MG11 HVA Variable Timing BTE.

4. This column shows the efficiency from the E7GT engine with 11.5:1 compression ratio pistons. This efficiency is from actual data.

5. This column shows the estimated efficiency of the E7GT engine at a compression ratio of 10.25:1. It is estimated from the actual data with the efficiency reduction of changing the compression ratio from 11.5:1 to 10.25:1 CR to arrive at the E7GT 10.25:1 CR BTE.

6. This column shows the estimated efficiency of the E7GT engine if it had the HVA system with valve timing set to simulate a normal cam. It is estimated by applying the same change in efficiency of the MG11 engine from the HVA fixed timing (with pumping losses) to the cam baseline (with valvetrain losses) to arrive at the E7GT 10.25 CR HVA Fixed Timing BTE.

7. The column shows the estimated efficiency of the E7GT engine with a 14.3:1 compression ratio and a variable timing HVA system. It is estimated by applying the change in efficiency from the MG11 cam baseline (including valvetrain losses) to the MG11 HVA variable timing engine (with pumping losses of a research oriented system) to the E7GT engine at a compression ratio of 10.25:1 to arrive at the BTE Improvement Applied to the E7GT. The research system has full capabilities for valve control full, which requires a certain amount of power to properly operate the system

8. The column shows the estimated efficiency of the E7GT engine with a 14.3:1 compression ratio and a variable timing HVA system with a production oriented system. It is estimated by applying the change in efficiency from the MG11 cam baseline (including valvetrain losses) to the MG11 HVA variable timing engine (with pumping losses of a production oriented system) to the E7GT engine at a compression ratio of 10.25:1 to arrive at the BTE Improvement Applied to the E7GT. The production system assumes that the full power of a research system will not be necessary for proper operation, and reduced pumping losses can be realized with a system that only includes the required valve control for a production system.

9. This column shows the estimated efficiency of a 2010 diesel engine. It is based on data for engines certified to 3.35 g/kW-hr (2.5 g/bhp-hr) of $NO_x$+NMHC. The efficiencies were reduced by 2% (relative change) to represent the loss in efficiency and/or increase operating cost to arrive at the 2010 Diesel BTE.

## TABLE 14. EFFICIENCY COMPARISON

| Mode | Speed | Load | 1<br>MG11 Cam Baseline BTE (%) | 2<br>MG11 HVA Fixed Timing BTE (%) | 3<br>MG11 Research Oriented HVA Variable Timing BTE (%) | 4<br>E7GT 11.5 CR BTE (%) | 5<br>E7GT 10.25 CR BTE (%) | 6<br>E7GT 10.25 CR HVA Fixed Timing BTE (%) | 7<br>Research Oriented HVA BTE Increase Applied to E7GT (%) | 8<br>Product Oriented HVA BTE Increase Applied to E7GT (%) | 9<br>2010 Diesel BTE Est (%) |
|---|---|---|---|---|---|---|---|---|---|---|---|
| 1 | 0% | 0% | 17.0 | 13.5 | 13.3 | 21.3 | 20.7 | 16.5 | 16.2 | 20.7 | 5.4 |
| 8 | 50% | 15% | 20.4 | 17.7 | 20.4 | 25.2 | 24.5 | 21.2 | 24.5 | 26.9 | - |
| 9 | 50% | 25% | 25.1 | 23.2 | 26.0 | 30.6 | 29.7 | 27.5 | 30.8 | 32.4 | 35.0 |
| 10 | 50% | 50% | 31.5 | 30.0 | 31.8 | 35.9 | 34.9 | 33.2 | 35.2 | 36.1 | 37.4 |
| 11 | 50% | 75% | 34.0 | 33.1 | 35.2 | 38.4 | 37.3 | 36.3 | 38.7 | 39.3 | 38.9 |
| 12 | 50% | 100% | 35.8 | 34.9 | 37.0 | 39.1 | 38.0 | 37.0 | 39.3 | 39.8 | 39.9 |
| 13 | 75% | 15% | 19.4 | 15.9 | 17.4 | 21.8 | 21.2 | 17.3 | 18.9 | 20.8 | - |
| 14 | 75% | 25% | 24.2 | 21.8 | 23.6 | 28.2 | 27.4 | 24.7 | 26.6 | 28.1 | 32.5 |
| 15 | 75% | 50% | 29.0 | 28.3 | 31.2 | 33.8 | 32.9 | 32.1 | 35.4 | 36.3 | 36.7 |
| 16 | 75% | 75% | 33.0 | 31.1 | 34.1 | 36.8 | 35.8 | 33.7 | 37.0 | 37.6 | 38.4 |
| 17 | 75% | 100% | 34.0 | 32.7 | 36.6 | 37.9 | 36.8 | 35.4 | 39.6 | 40.1 | 39.4 |
| 18 | 100% | 15% | 16.3 | 13.5 | 14.3 | 18.8 | 18.2 | 15.1 | 16.0 | 17.8 | - |
| 19 | 100% | 25% | 20.8 | 18.8 | 20.8 | 24.5 | 23.8 | 21.5 | 23.8 | 25.2 | 29.0 |
| 20 | 100% | 50% | 27.5 | 26.0 | 28.8 | 31.8 | 30.9 | 29.2 | 32.4 | 33.3 | 34.0 |
| 21 | 100% | 75% | 30.4 | 29.1 | 33.0 | 35.5 | 34.5 | 33.1 | 37.5 | 38.2 | 36.4 |
| 22 | 100% | 100% | 33.8 | 32.4 | 34.2 | 36.6 | 35.5 | 34.0 | 35.9 | 36.4 | 37.3 |
| Weighted 13-mode | | | 31.2 | 29.8 | 32.4 | 35.1 | 34.1 | 32.3 | 35.4 | 36.2 | 37.1 |

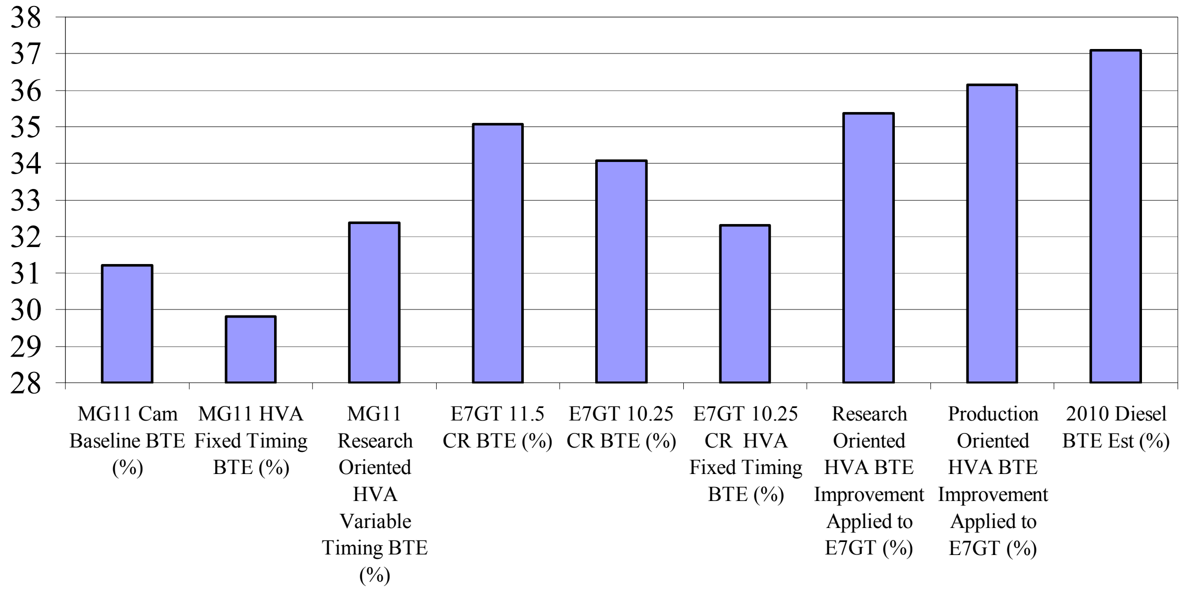


**Figure 27. 13-Mode Weighted Efficiencies**

Adding valvetrain losses to the MG11 baseline cam engine reduces the efficiency by 1.0% (relative efficiency) on the weighted 13-mode results, whereas adding pumping losses of a research system to the MG11 HVA variable timing engine reduces efficiency by 4.3%, but adding pumping losses of a production system to the MG11 HVA variable timing engine reduces efficiency by 2.2%.

Figure 28-30 shows the efficiencies at 1250, 1600, and 1950 rpm respectively. Figure 31 shows the efficiency change of an engine with a HVA system compared to an engine with a cam. The highest efficiency improvements are at higher loads, whereas there is an efficiency loss at the lower loads. Part of this is due to the increase in losses associated with operating an HVA system, where the losses are more significant with the HVA system at the lighter loads. Part of this may be the loss of turbulence with the 100% throttle position used with the HVA system. At light loads, the turbulence created by the throttle assists in the combustion process. Further testing with the HVA system should be conducted with one intake valve open more than the other to create turbulence at light loads. Figure 32 shows the efficiency changes of a HVA system compared to a 2010 diesel engine. At the high loads, the efficiencies approach the diesel efficiency, but are still lower at the lighter loads. The research oriented pumping losses for the HVA system were based on engine speed only. A production system that can reduce pumping losses could improve efficiencies.

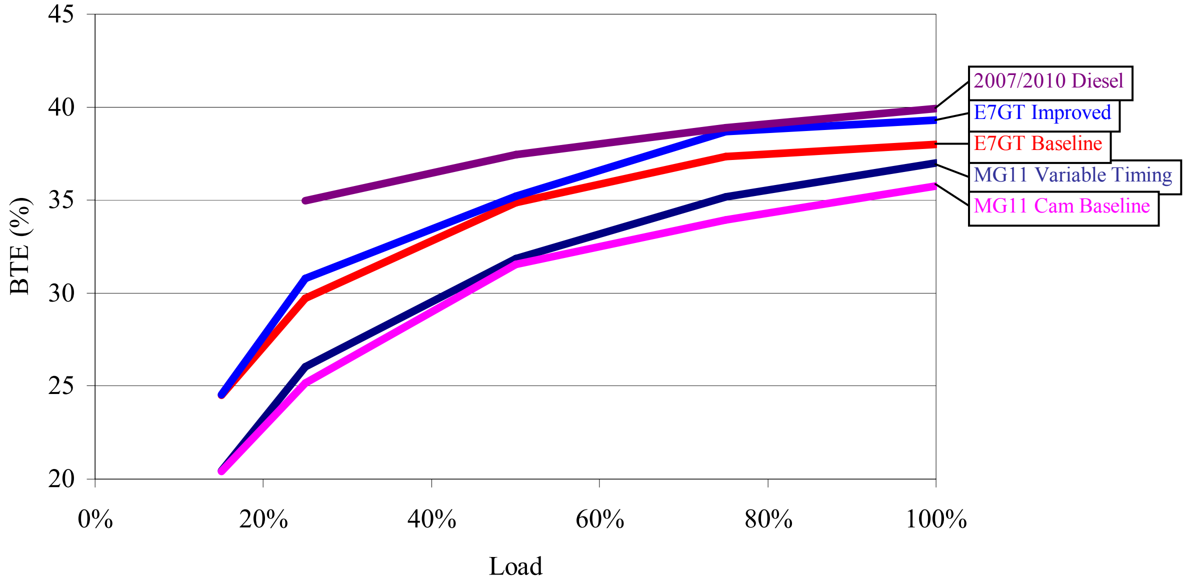


**Figure 28. Efficiencies at 1250 rpm**

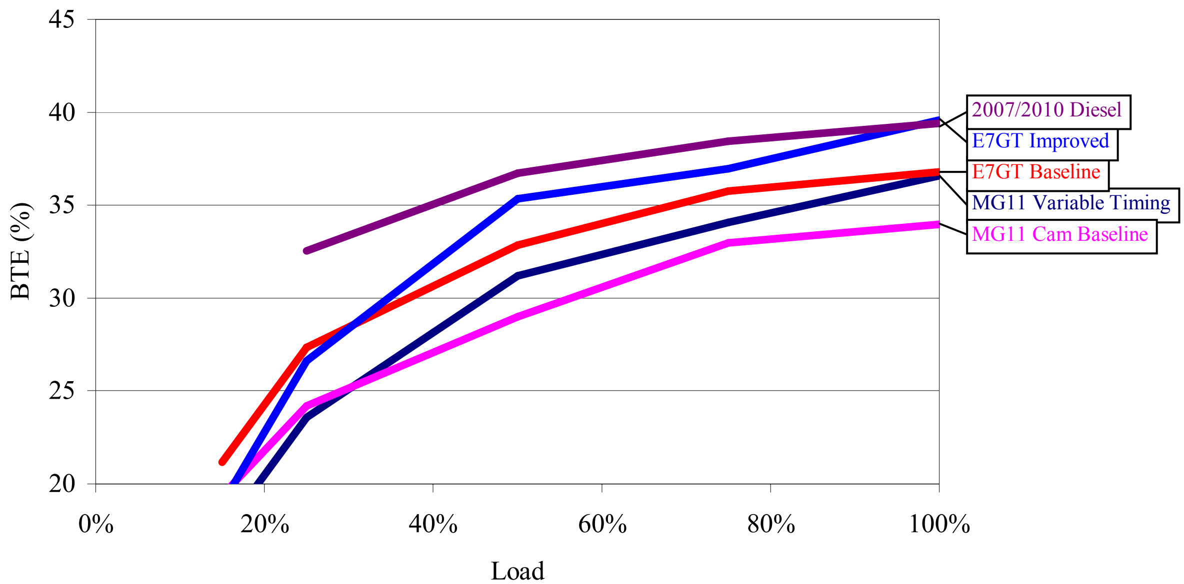


**Figure 29. Efficiencies at 1600 rpm**

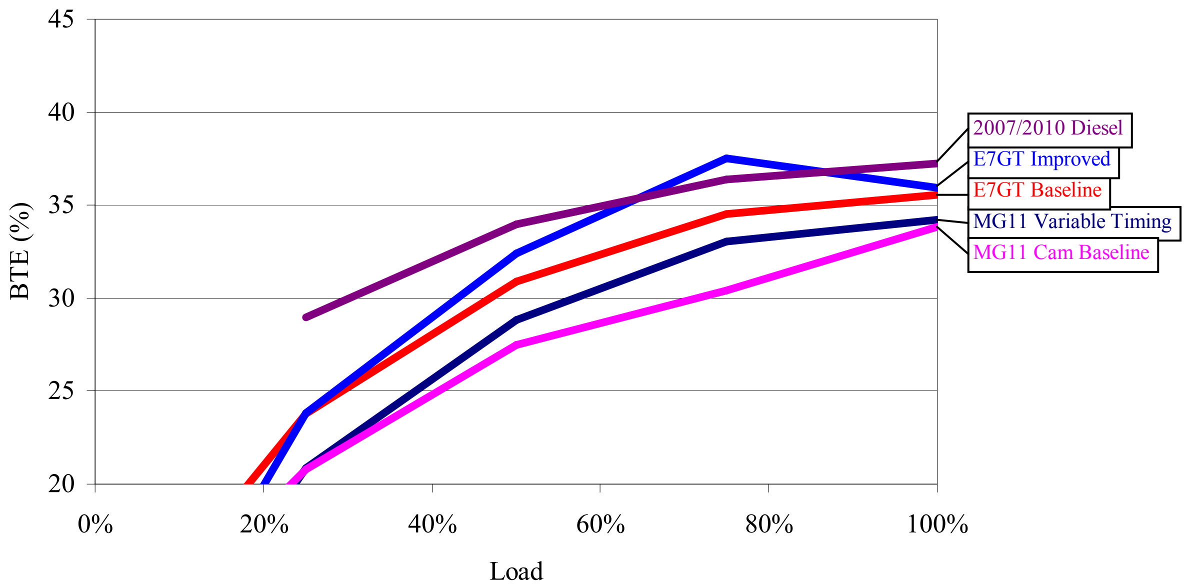


**Figure 30. Efficiencies at 1950 rpm**

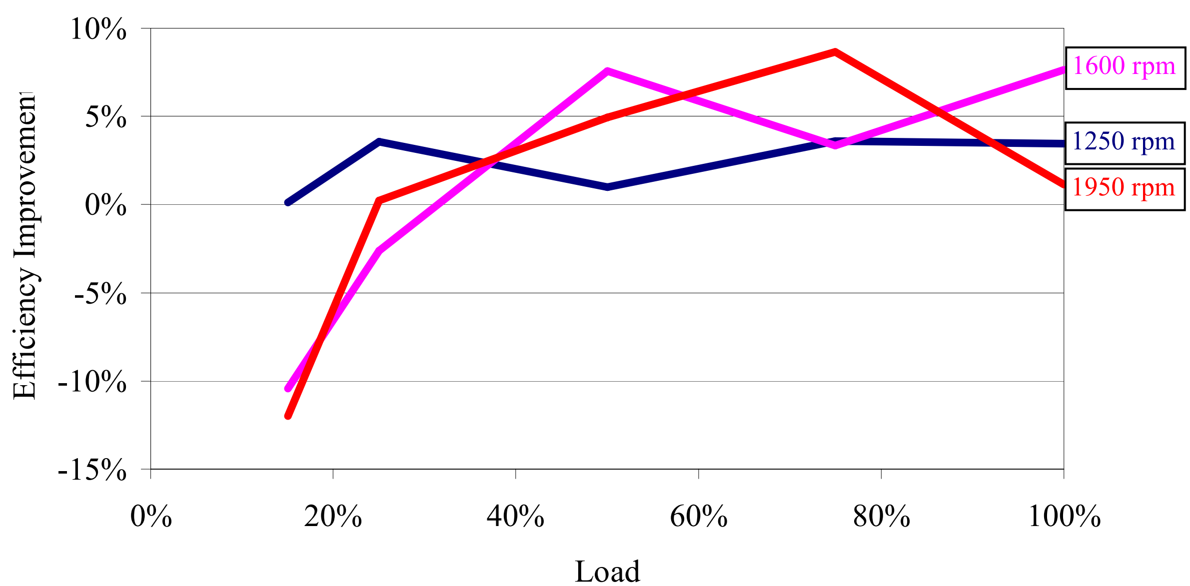


**Figure 31. HVA System Efficiency Improvement over a Cam System**

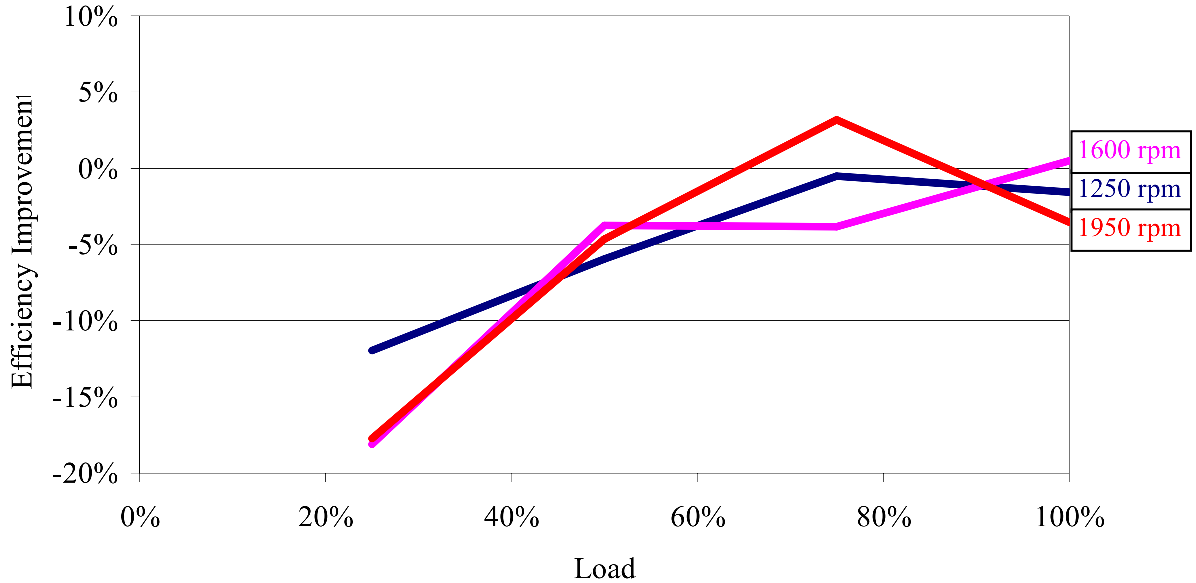


**Figure 32. HVA System Efficiency Improvement over a 2010 Diesel Engine**

# 3. SUMMARY AND CONCLUSIONS

The summary and conclusions of the project are:

- A Volvo 11 liter diesel engine was converted to natural gas with a camless hydraulic valve actuation system.
  - A Sturman HVA system was installed on the engine.
  - The engine had not been optimized for operation on natural gas.

- Tests were conducted with high compression ratio pistons using an early intake valve closing to either operate throttleless at light loads or reduce the effective compression ratio at high loads.
  - The Sturman system enables throttleless operation, using early intake valve closing to control load.
  - The Sturman HVA system enables full use of high compression ratio pistons, where full compression ratio can be used at light loads and a reduced effective compression ratio can be used at high loads to avoid knock.

- The engine efficiency is not sensitive to intake opening, exhaust opening, or exhaust closings within a fairly large range of valve timings.

- The HVA system can improve the efficiency of a heavy duty gas engine.
  - With pumping losses of a research system included, efficiency improvements of an HVA system over a cam system are realized down to 25% load, at 15% load, there is an efficiency penalty due to high hydraulic pumping losses of the HVA system and lack of turbulence from throttleless operation.
  - More intelligent control of hydraulic pumping requirements for the HVA system at light loads could improve efficiency.
  - Turbulence can be created with a throttleless engine through different valve lifts on the two intake valves.

- When comparing a natural gas engine with a research HVA system to a 2010 diesel engine, efficiencies at the higher loads are equivalent, but efficiencies at the lower loads are still higher with the diesel engine.
  - A research HVA system has 4.7% lower relative efficiency on the weighted 13-mode than an equivalent diesel engine, but a production HVA system has 2.5% lower relative efficiency.

- Low emissions are possible with stoichiometric combustion, EGR and a TWC.
  - The $NO_x$ emissions on the 13-mode test are 0.005 g/kW-hr, which are well below the target of 0.27 g/kW-hr.
  - Particulate matter was not measured, but is expected to be below the target of 0.013 g/kW-hr.

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