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# Advanced Hydrogen Utilization Technology Demonstration

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# Executive Summary

Hydrogen fuel has long been considered as a possible replacement for hydrocarbon-based fuels. This is due to the abundance of hydrogen in nature, the fact that carbon-based emissions are not produced in hydrogen combustion, and our ability to produce the fuel from sustainable energy sources (i.e., hydro, solar, biomass, and wind). In the future, hydrogen fuel may be used predominantly in fuel cells; however, this technology is still in its infancy. The use of hydrogen in internal combustion (IC) engines could fill the gap between existing hydrocarbon-fueled IC engines and future hydrogen fuel-cell technology.

In this study, a Detroit Diesel Corporation (DDC) 6V-92TA engine was used for experiments using hydrogen fuel. The methanol coach configuration of this engine was chosen because it was most easily adapted for efficient combustion of hydrogen. The engine was baseline tested using methanol fuel and methanol unit injectors. One cylinder of the engine was then converted to operate on hydrogen fuel, which was injected at high pressure near top dead center and burned in a compression-ignition diesel cycle. Methanol fueled the remaining five cylinders.

This early testing with only one hydrogen-fueled cylinder was conducted to determine the operating parameters that would later be implemented for multicylinder hydrogen operation. Some of these parameters included:

- Injector tip hole number and size
- Optimum injector pulse width
- Optimum beginning of injection
- Idle operation requirements.

Researchers then operated three cylinders of the engine on hydrogen fuel to verify single-cylinder idle tests. Once it was determined that the engine would operate well at idle, the engine was modified to operate with all six cylinders fueled with hydrogen.

Six-cylinder operation on hydrogen provided an opportunity to verify previous test results and to more accurately determine the performance, thermal efficiency, and emissions of the engine. This included testing at minimum for best torque (MBT) timing and additional testing at retarded timings and reduced injection pressures. Results from these tests established that the engine performed well and provided rated power and torque. Researchers achieved an estimated Federal Test Procedure transient $NO_x$ emissions level of less than 5 grams per horsepower-hour (g/hp-hr), based on steady-state measurements. A maximum brake thermal efficiency of more than 40% (based on the lower heating value of hydrogen) was measured at peak torque speed.

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# Objective

The objective of this project was to demonstrate the ability of a Detroit Diesel Corporation (DDC) 6V-92TA engine to operate on hydrogen and to document its power, thermal efficiency, and exhaust emissions. A compression-ignition diesel cycle was achieved by establishing in-cylinder conditions for auto-ignition and injecting hydrogen at high pressures late in the compression stroke.

# Background

Hydrogen fuel has long been considered a possible replacement for hydrocarbon-based fuels. This is due to the abundance of hydrogen in nature and our ability to produce it from sustainable energy sources (i.e., hydro, solar, biomass, and wind). The fact that no carbon-based emissions are produced in the combustion of hydrogen fuel could be an important advantage. The preferred use of hydrogen fuel may be in fuel cells; however, this technology is still in its infancy. Hydrogen-fueled internal combustion (IC) engines could fill the gap between existing hydrocarbon-fueled IC engines and future hydrogen fuel-cell technology (1,2).

Much research and development is needed to determine the best methods for hydrogen introduction and combustion in IC engines in order to create efficient hydrogen-fueled engines. Introducing hydrogen into IC engines is complicated because of its wide flammability range, low energy density by volume, and high flame speed (3,4,5,6).

Hydrogen's wide combustion range and high flame speed make introduction into the intake manifold of an engine problematic. Flashback through the intake manifold of the engine can occur if ignition occurs in the combustion chamber during the intake stroke. Also, the susceptibility of hydrogen to surface ignition can cause preignition in the combustion chamber during the compression stroke. The high flame speed of hydrogen leads to rapid combustion, which is sometimes perceived as knock, although rumble is the preferred term. As the equivalence ratio decreases, the tendency for the engine to knock is reduced along with the power output of the engine. The low volumetric energy density of the fuel also results in lower engine power output because of the displacement of air by the fuel during the intake stroke (3,5,6).

Introduction of the fuel directly into the cylinder early in the compression stroke eliminates the problems of flash back through the manifold and lower power output caused by displacement of the air by fuel. The introduction of the fuel after the intake valve closes actually supercharges the cylinder. However, knock is still a concern with this method of fuel introduction (3,5,6). One way to eliminate the knock problem is to inject the fuel under high pressure late in the compression stroke. The rate of heat release can be controlled by the rate at which the hydrogen fuel is injected into the combustion chamber (4,6).

There are a number of ways to ignite fuel with late-cycle injection. These include, but are not limited to, ignition by compression, hot surface (glow plug), spark, laser, and catalytic surface. Compression ignition has the disadvantage of requiring a great deal of residual heat, or high compression ratio, because of the high auto-ignition temperature of hydrogen. Glow plugs have shown a problem with durability because of the severe environment of the combustion chamber. The spark ignition system requires a narrow equivalence ratio window in the spark plug gap in order to ignite the fuel. Therefore, spark ignition is dependent on injection timing, injection pressure, engine speed and load, the geometry of the injector and spark plug, and the mixing of hydrogen and air in the combustion chamber (4,6).

A disadvantage to late-cycle injection involves the compression needed to produce the required injection pressures. This is typically in the range of 2000–3000 psi. The cost and complexity of high-pressure compression and fuel delivery systems are substantial compared to low-pressure systems. Nevertheless, burning hydrogen fuel in a diesel cycle shows engine performance and emissions benefits compared to injecting hydrogen early in the compression cycle or adding it to the inlet air (6). Because the engine is not knock-limited with late-cycle injection, higher boost pressures, compression ratios, and loads can be used. As a result, higher brake thermal efficiency (BTE) is obtainable.

$NO_x$ emissions of a late-cycle-injected hydrogen engine are similar to those of a standard diesel-fueled engine of the same power output (4). The combustion rate depends on the rate of fuel injection in the late-cycle injection system. This contrasts with systems where flame speed is a function of the equivalence ratio. This can reduce the in-cylinder temperatures that create $NO_x$ emissions to levels equivalent to that of a diesel-fueled engine (4).

Most of the research described above involved either a spark plug or a glow plug as an ignition source. These were used to keep cylinder pressures low in the automotive-based engines.

Compression ignition of late-cycle-injected fuel is commonly referred to as the diesel cycle. The methanol-based DDC 6V-92TA engine provided an excellent base engine for modification of compression ignition for late-cycle-injected hydrogen fuel. This engine possesses the following features, which make it attractive for high-pressure hydrogen injection and combustion research:

- High compression ratio (23:1)
- Rugged construction capable of withstanding high cylinder pressures
- A glow plug system to assist in cold start and ignition at idle
- An electronic control system capable of operating a gaseous fuel injector.

# Results and Discussion

## Test Cell Setup

### *Fuel System*

The hydrogen-fuel system design was based on Southwest Research Institute's (SwRI) previous experience in hydrogen fuel systems, with input from Air Products and DDC. As much hardware as possible was located outside the test cell and was positioned to ensure the safety of personnel working on or near this project.

Air-actuated valves located near the fuel supply and the engine controlled the flow of hydrogen. The air supplied to these valves was controlled by palm button valves located at the test cell console, inside the test cell, and near the fuel supply. This air supply was also controlled by a computer, which monitored a safety shutdown system consisting of lean limit sensors, a flame detector, jacket water temperature, oil pressure, dynamometer water temperature and pressure, and hydrogen mass flow. If a safety concern was perceived, the computer would shut off and vent the fuel system by reducing the air supply pressure.

Researchers leak tested the hydrogen-fuel system with helium and a liquid soap leak detector product called Snoop. The system was then pressurized to operating pressures with helium and left alone for 2 days. No measurable pressure loss was observed during that time.

Figures 1 through 3 are fuel system schematic drawings. The drawings include some of the safety systems and a numbering system for most of the major components. This numbering system also was used in the standard operating procedures (SOPs), which were compiled to provide proper startup/shutdown/emergency shutdown and other safety procedures.

A fuel manifold supplied hydrogen to the injectors inside the engine. The manifold was run through a small plate in the outside edge of the heads, underneath the valve cover, and in front of the electronic unit injectors. This provided for easy modification when injectors were changed from methanol to hydrogen.

A hydrogen compressor was installed and connected to a high-pressure bottle manifold. The Micro-Motion fuel meter was connected in the system with a pair of high-pressure flexible hoses to isolate this instrument from engine vibrations that could cause erroneous readings. The fuel system was completed and used to test the injector in the engine on both helium and hydrogen.

The fuel flow measurement was not accurate when operating only one cylinder on hydrogen because the Micro-Motion mass flow meter was sized for six-cylinder operation and was too large for single-cylinder operation. Fuel flow measurements were verified using emissions data (7).

Air Products provided a 2700-psi hydrogen tube trailer that was placed 100 ft from the engine test cell. Some precautions were taken in the selection of the tube trailer parking site to ensure safety. These included:

- The trailer was located at least 50 ft from any building that did not contain an automatic fire sprinkler system.

- The trailer was located at least 15 ft from the vertical plane of any overhead power lines.

# HYDROGEN SUPPLY

## Option B, 16 Bottles, Compressor, and Tube Trailer

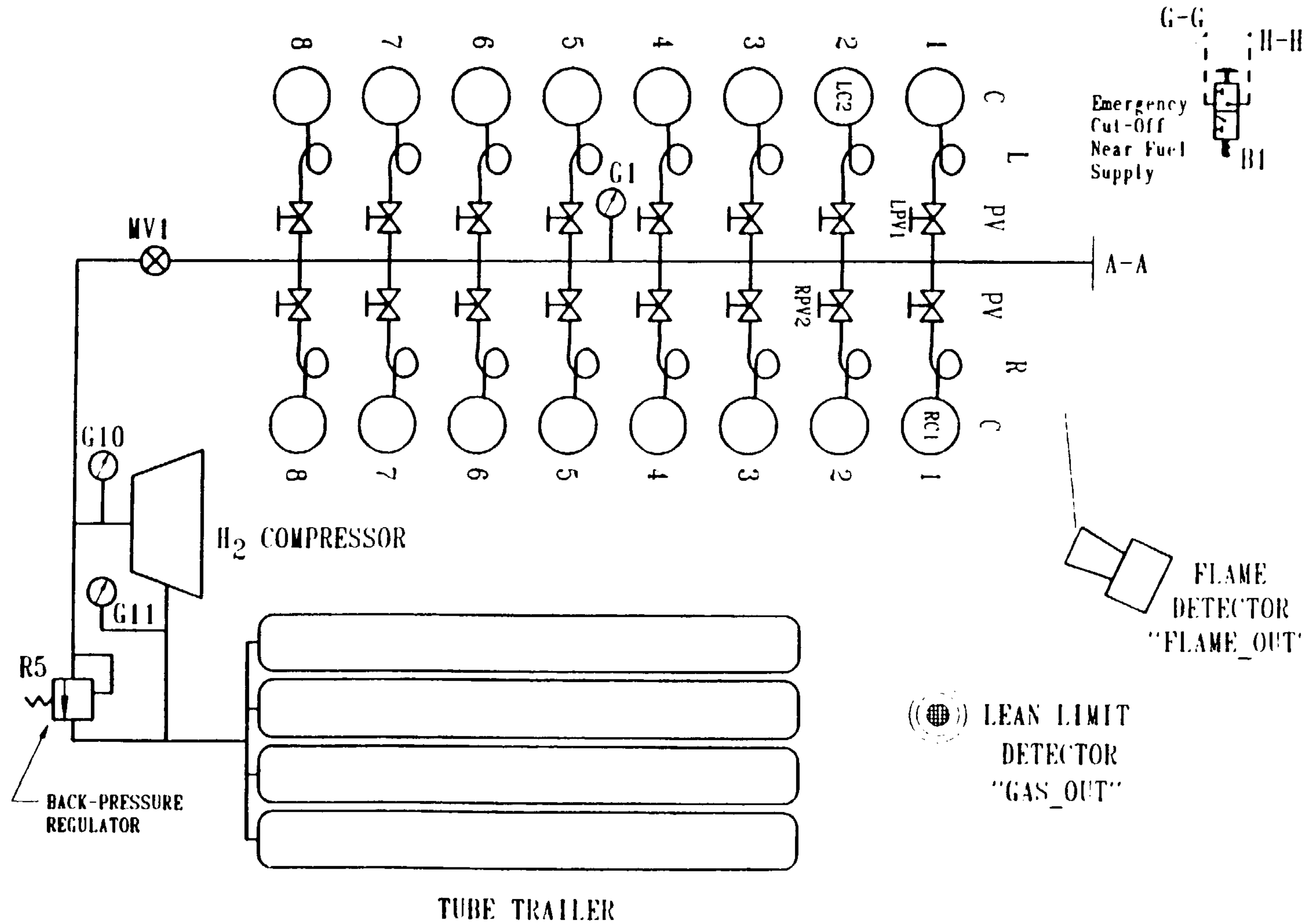


**Figure 1. Fuel supply system**

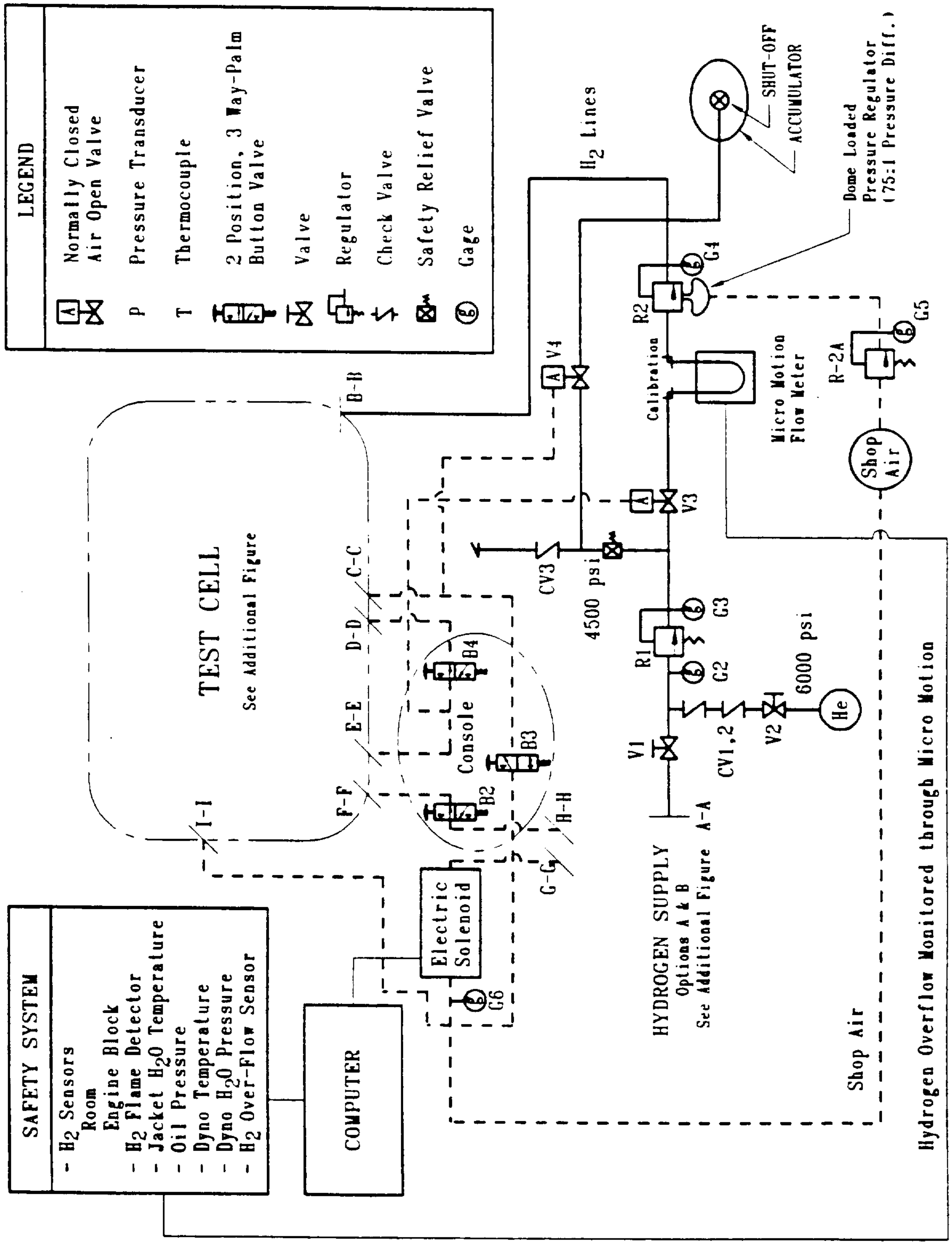


**Figure 2. Fuel measurement and control system**

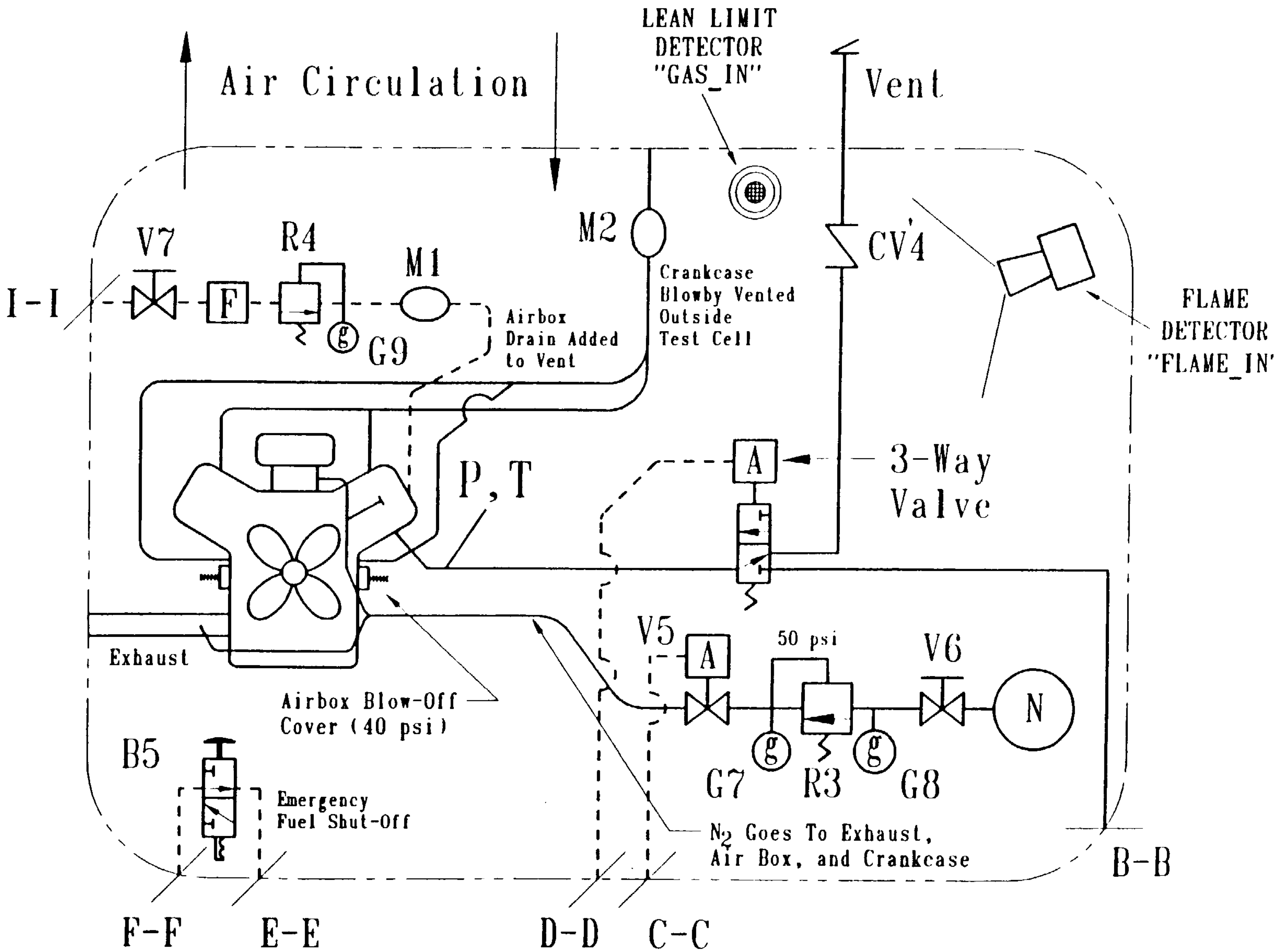


Figure 3. Test cell fuel system

- The trailer was located at least 10 ft from any sprinkled building (a cooling tower with a continuous flow of water was considered a building with a sprinkler system).

- The slope of the trailer location was less than 3%.

- The ground under the trailer was defoliated.

- Curb stops were installed to prevent the trailer from rolling.

- Signs were placed around the trailer to warn that smoking was not permitted in the area.

- The trailer jacks were well supported to prevent them from sinking into the ground.

A 14-ft flexible line connected the trailer to 0.5-inch outside diameter, thick-walled, stainless steel tubing, which was connected to the suction side of the compressor. A hydrogen compressor supplied the high-pressure hydrogen needed to operate the engine. The tube trailer provided a maximum of 2700 psi to the compressor, which in turn raised the pressure to 3300 psi to supply the engine.

Researchers found a number of gas leaks in the plumbing of the compressor upon initial startup. Many of these leaks had resulted from using single feral fitting on the stainless steel lines of the compressor during assembly by the manufacturer (Burton Corblin). The fittings were tightened and adjusted to stop the leaking. Two of the pipe fittings on the compressor assembly also leaked and were repaired by tightening and applying Teflon tape on the threads.

The final problem encountered with the compressor and fuel system was the lack of a "Cadillac mode" on the compressor. At maximum operating pressure, this "Cadillac mode" unloads the compressor by holding the intake valve open during the compression stroke. Once the pressure drops below a set point, the compressor is loaded by rendering the intake valve operational, and gas compression resumes. As delivered, the compressor was simply set to shut off with a "Trouble" light activated on the control panel when it obtained the maximum pressure (3300 psi). The compressor would not reset automatically. The solution to this problem was a back pressure regulator (labeled R-5 in Figure 1) that allowed excess flow through the compressor to recirculate to the suction side of the compressor. This system maintained a constant pressure at the compressor outlet but required an additional heat exchanger to keep at a constant near-ambient temperature.

### *High-Speed Data Instrumentation*

A shaft encoder providing 720 pulses per revolution was mounted to the front of the crankshaft. A Kistler Model 6061 water-cooled pressure transducer was fitted into the cylinder head in place of the glow plug in Cylinder 1 on the right side of the engine (see Figure 1). This provided a nonflush mounting of the pressure transducer and created several problems during the project:

- A "ringing" in the tube to the transducer, documented in numerous motoring traces. This problem was noticed in both motoring and firing traces in the instantaneous heat release rate.

- A slight reduction in compression ratio as a result of the increased combustion chamber volume created by the nonflush mount of the transducer in the cylinder head.

- A problem with combustion initiation during cold starting and at light loads caused by removal of the glow plug. The exhaust temperature of the cylinder without the glow plug was often lower than that of other cylinders during light-load operation. The presence of the glow plug assisted combustion initiation even when it was not energized. Apparently, the temperature of the glow plug remained high enough to cause surface ignition of hydrogen at light loads with the glow plug system inactive.

### *Safety System*

Flame detectors and combustible gas detectors were installed and tested. A pressure relief valve was located downstream of the main pressure regulator. If the primary regulator failed, this pressure relief valve would protect the rest of the fuel system (including the flow meter) by venting pressure greater than 4500 psi outside the test cell.

In a fire or other emergency situation, a nitrogen supply could be manually released to flood the engine's crankcase, airbox, and exhaust system. The nitrogen release was controlled at the test cell console. A three-way, air-actuated valve located near the engine provided another safety precaution. When this valve was closed, hydrogen in the lines between the valve and the engine was vented outside the test cell through a check valve.

## Engine Operation

### *Methanol Baseline Operation*

A baseline test of the engine operating on methanol was completed to help identify operating parameters for future hydrogen operation. The data taken during this test are shown in Figures 4 through 8.

Figure 4 shows indicated horsepower (IHP), calculated from the cylinder pressure data from one cylinder, and brake horsepower (BHP) versus engine speed. Note that the BHP at 2100 rpm is much lower than the 277 BHP anticipated. From the difference between these two numbers, Figure 5 shows calculated friction and pumping horsepower versus engine speed. There was approximately a 53-horsepower difference between IHP and BHP at 2100 rpm.

Figure 6 shows preturbine exhaust temperature versus engine speed, which was obtained to help diagnose the reason for power loss at rated speed. Figure 7 shows peak cylinder pressure and location of peak cylinder pressure versus engine speed at these test conditions. As can be seen in Figure 7, at 1900 rpm the engine comes very close to exceeding the 2000 psi cylinder pressure limit set by DDC. The pulse width (PW) and beginning of injection (BOI) versus engine speed are shown in Figure 8.

### *One-Cylinder Hydrogen Operation*

During the early stages of testing, the engine was operated extensively with one cylinder fueled with hydrogen and the remaining five fueled with methanol. The engine was operated using five different injectors with various nozzle configurations and different electronic control units (ECUs) to change multipliers for PW and BOI. The ECU numbers and multipliers for PW and BOI are provided in Table 1, along with the injector hole sizes and the number of holes. This portion of the project was designed to determine operating conditions/requirements and combustion stability of the hydrogen-fueled cylinder before multi-cylinder operation was attempted. This also provided time to gain confidence in the test cell safety system, the fuel control system, and the SOPs with a minimum of risk and a minimum consumption of hydrogen fuel.

All low-speed and high-speed data and graphs are catalogued in two three-ring binders, with data for one-cylinder hydrogen operation catalogued in Volume I. Data for these tests include the BHP of the engine with one cylinder operating on hydrogen and the remaining five on methanol. Injector PW for the methanol-fueled cylinders was held constant at each test mode while fuel rate and injection timings were explored on the cylinder fueled by hydrogen.

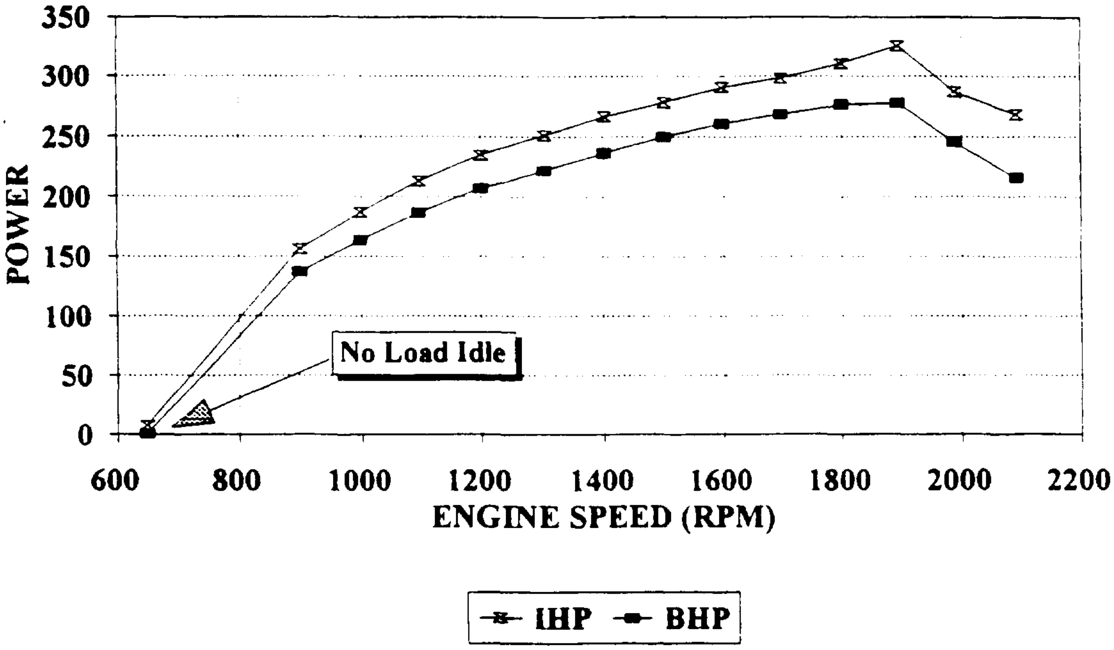


Figure 4. Baseline IHP and BHP versus engine speed

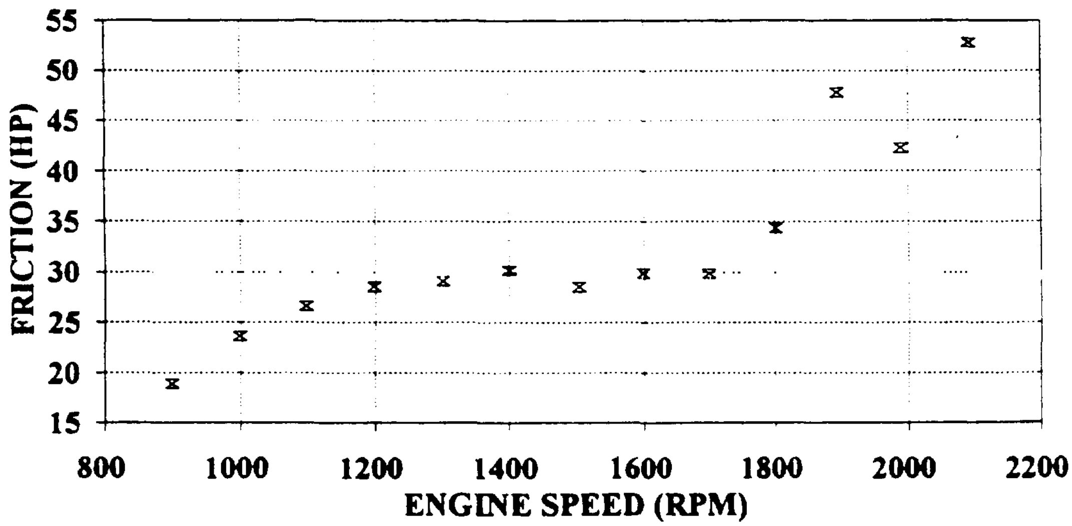


Figure 5. Baseline calculated friction HP versus engine speed

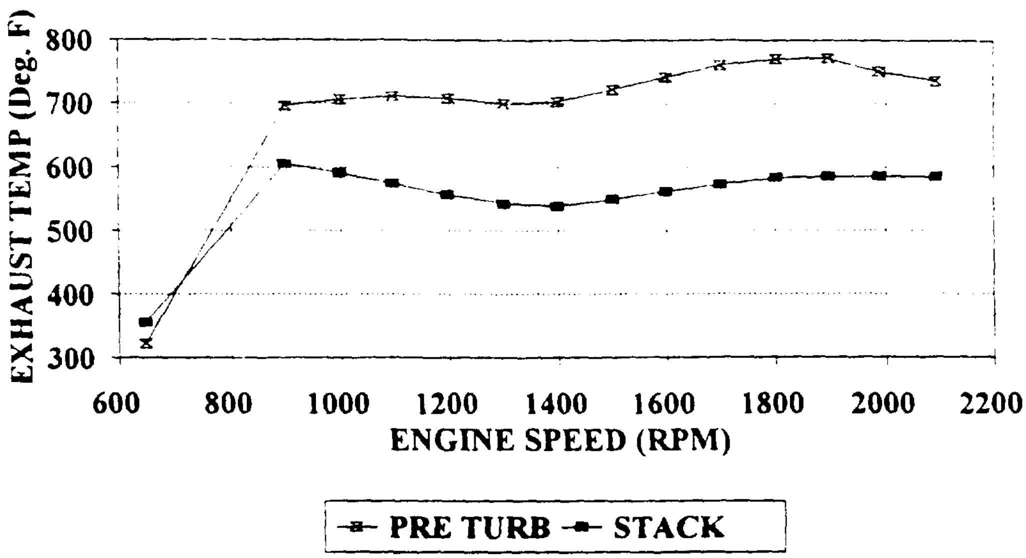


**Figure 6. Baseline preturbine and exhaust stack temperature versus engine speed**

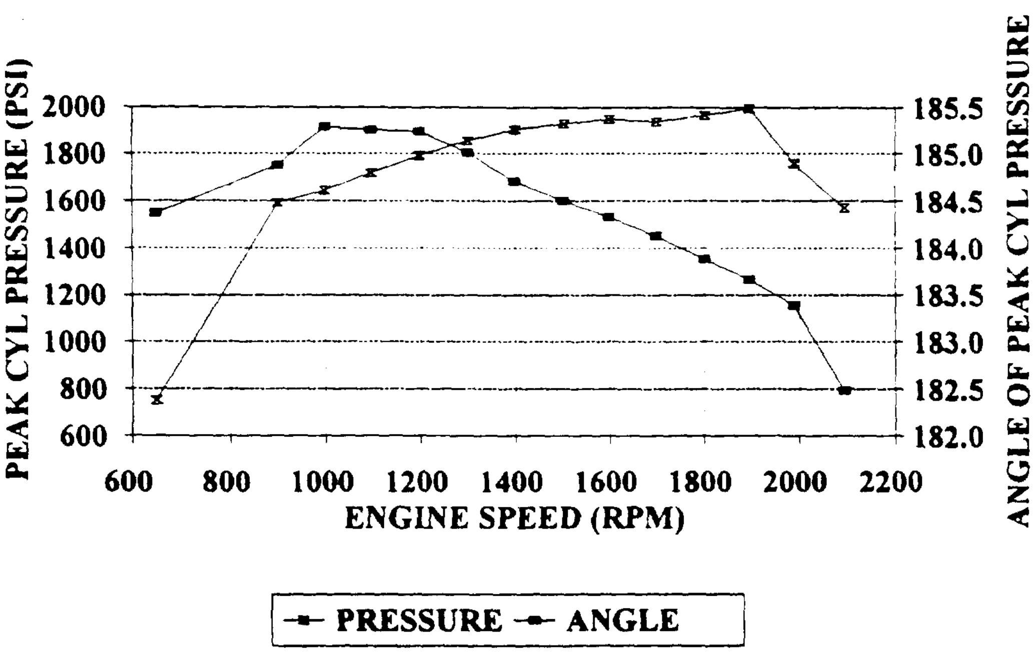


**Figure 7. Baseline peak cylinder pressure and angle of peak cylinder pressure versus engine speed**

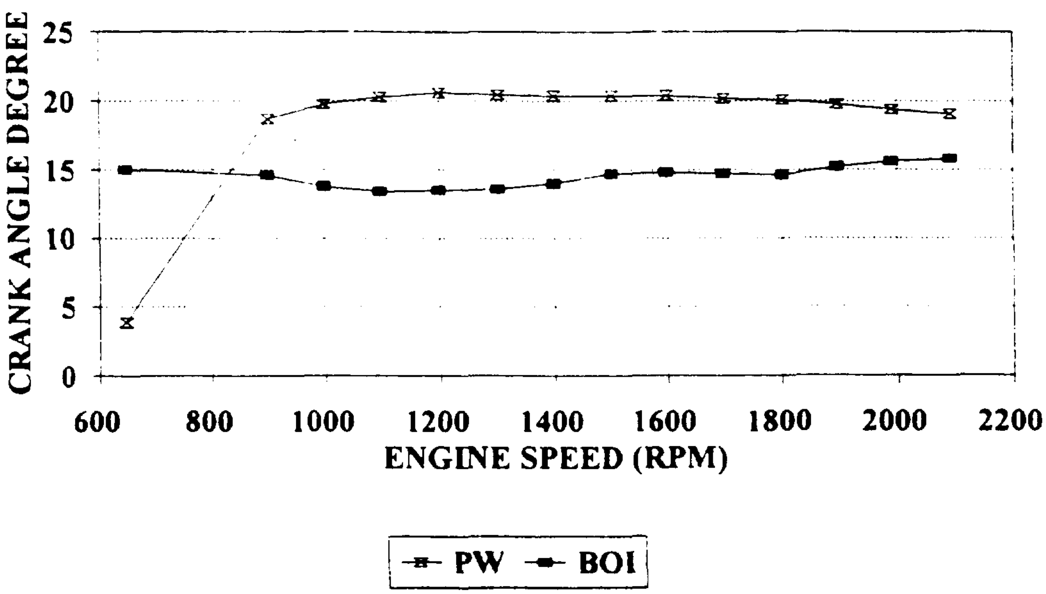


**Figure 8. Baseline PW and BOI versus engine speed**

**Table 1. ECU and Injector Notation**

| ECU Number | # of $H_2$ Cylinders | PW Multiplier | BOI Multiplier | Injector Number | Number of Holes | Holes Size (inch dia) |
|---|---|---|---|---|---|---|
| 1 | 1 | 1.49 | 1 | 1 | 7 | 0.016 |
| 2 | 1 | 3 | 1 | 2 | 7 | 0.016 |
| 3 | 1 | 3 | 2 | 3 | 1 | 0.125[a] |
| 4 | 3 | 3 | 2 | 4 | 7 | 0.022 |
| 5 | 6 | 3 | 2 | 5 | 7 | 0.025[b] |

[a] Injector hole generated by machining the end of the tip until Sac volume exposed.
[b] This injector was modified to provide additional needle lift.

***Injector 1—ECU 1.*** Injector 1 was one of two original injectors shipped from DDC. This injector was chosen for its lower leak rate from the tip. Leak rate was determined by simply applying liquid leak detector on the tip of the injector and visually checking the rate at which bubbles were formed by the leaking gas. Injector 1 accumulated more than 2 hrs of operation on helium during bench testing and on hydrogen inside the engine. The injector was finally removed because of a gas leak from the coil pack. The leak rate from the tip also appeared to increase, although the rate was not measurable with the Micro-Motion mass flow meter.

The main thrust of the testing was to determine the effect of injection timing on power output for a given PW at peak torque speed. This testing was conducted from runs 59 to 84, and the results are shown in Figure 9. This graph indicates that power output was relatively insensitive to the range of injection timings tested. From cylinder pressure data taken during these tests, it appeared that a more advanced BOI might increase power output and improve engine efficiency.

***Injector 2—ECU 2.*** After the leakage from the first injector increased, the second of the two original injectors sent from DDC was placed in the engine. A second ECU was programmed at DDC and installed with a PW multiplier of 3 to increase the fuel flow through the injector. This new ECU allowed increased fuel rates needed for rated power operation.

After reviewing the high-speed combustion data, researchers concluded that the fuel injection event was taking too long and fuel was burning in the exhaust. Additional testing was done with this configuration to determine the effect of the hydrogen gas supply pressure on power at a fixed PW and BOI. The results are shown in Table 2.

These results show no performance benefit of operating above a fuel supply pressure of 3000 psi. The elevated exhaust temperatures were caused by excessively long (28 CAD) injection duration. The experimenters felt that larger injector tip holes would increase the injection rate, significantly increase engine efficiency, and reduce exhaust temperatures.

***Injector 3—ECU 2.*** Injector 3 was a single-hole injector constructed by cutting off the bottom of the injector tip to maximize flow. The choke point of the hydrogen flow into the cylinder was in the needle seat area. It was hoped that the injected hydrogen would diffuse and burn after hitting the top of the piston. The injector was run only for a brief period because of poor performance at rated power. The data from this test are shown in Table 3.

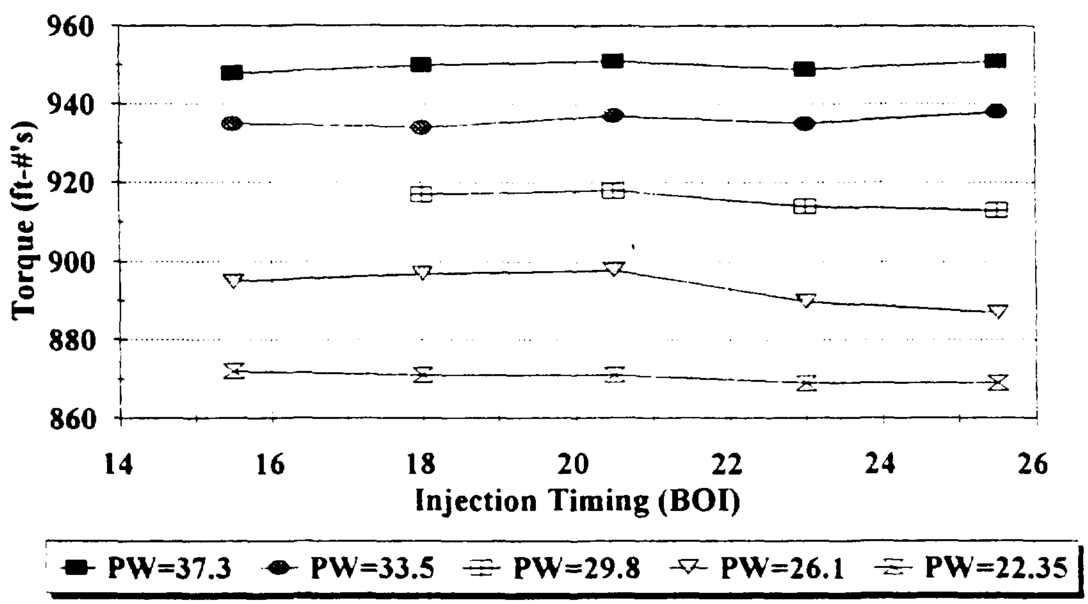


**Figure 9. Effect of injection timing of hydrogen on torque at 1200 rpm**

**Table 2. Effects of Hydrogen Supply Pressure at Fixed PW and BOI**
(PW = 28.31 CAD; BOI = 25 CAD)

| $H_2$ Pressure (psi) | Total BHP | Hydrogen Fueled Cylinder 1 IHP[a] | Exhaust Temperature (°F) | Run Number |
|---|---|---|---|---|
| 3181 | 262.1 | 383.9 | 1173 | 98 |
| 3050 | 262.5 | 366.2 | 1109 | 97 |
| 2823 | 259.5 | 340.7 | 1058 | 96 |
| 2440 | 247.1[b] | 286.3 | 975 | 99 |
| -- | 198.2[b] | -33.0 | 460 | 95 |
| -- | 193.4[b] | -33.0 | 523 | 100 |

Note: Runs 95 and 100 were operating on five cylinders of methanol.
[a]Assumes remaining five cylinders are fueled in same manner as $H_2$-fueled cylinder.
[b]Total engine power is affected by changes in turbocharger boost pressure caused by the cylinder operating on hydrogen.

**Table 3. Performance Results with Injector 3**

| $H_2$ Pressure (psi) | Total BHP | Hydrogen Fueled Cylinder 1 IHP[b] | Exhaust Temperature (°F) | PW | BOI | Run Number |
|---|---|---|---|---|---|---|
| 2928 | 197.3 | -34.6 | 468 | 0 | 0 | 90 |
| 3052 | 239.5 | 245.0 | 1145 | 31.25 | 25 | 91 |
| 3062 | 238.0 | 216.1 | 1033 | 33.0 | 25 | 92 |
| 3370 | 236.4 | 224.9 | 1080 | 31.25 | 25 | 93 |
| 3435[a] | 195.7 | -23.3 | 631 | 0 | 0 | 94 |

[a]Motoring run—Cylinder 1 not fueled with remaining five cylinders fueled on methanol.
[b]Assumes remaining five cylinders are fueled in same manner as $H_2$-fueled cylinder.

Although the injector operated for less than 15 minutes, the tip showed signs of exposure to extreme temperatures. The tip was a golden color at its end and blended to a dark purple toward the injector body.

The top of the piston was inspected to determine if any damage had occurred during operation with this injector. The top of the piston had some slight discoloration, but there appeared to be no significant damage. The piston skirt, however, showed a light scuff mark. This damage was not significant enough to warrant replacement.

***Injector 5—ECU 3.*** A series of tests was performed to determine the capability of the engine to reach rated horsepower at 2100 rpm. ECU 3 provided a PW multiplier of 3 and a BOI multiplier of 2. Injector 5 had seven 0.025-in.-diameter holes in the injector tip and increased needle lift.

**Table 4. IHP for Injector 5 for PW of 45 CAD and BOI of 40 CAD**

| Run Number | Exhaust Temperature (°F) | Percent of Temperature Limit | Cylinder Pressure (psi) | Percent of Pressure Limit | IHP[a] | BHP |
|---|---|---|---|---|---|---|
| 127 | 1001 | 91 | 2016 | 101 | 376.2 | 323 |
| 129 | 1044 | 95 | 2016 | 101 | 381.5 | 329 |
| 150 | 1038 | 94 | 1954 | 97 | 390.0 | 337 |
| 151 | 1029 | 94 | 2053 | 103 | 379.7 | 327 |
| Average | 1028 | 94 | 2010 | 101 | 381.9 | 329 |

[a]IHP was calculated from the cylinder pressure data taken from the hydrogen-fueled cylinder with the assumption that the other five cylinders were fueled in the same manner.

The engine limits and estimated BHP generated by Injector 5 are shown in Table 4. Friction horsepower was determined from baseline operation on methanol fuel (see Figure 5). IHP was calculated from the cylinder pressure data taken from the hydrogen-fueled cylinder with the assumption that the other five cylinders were fueled in the same manner.

A PW of 45 CAD and a BOI of 40 CAD provided an estimated BHP well over the rated 277 BHP of the methanol configuration engine. The test showed that the engine was capable of producing these high powers without exceeding the exhaust temperature limit of 1100°F. The cylinder pressure was at the limit of 2000 psi for these tests. In addition, the engine produced more than 445 IHP (run 141) with a PW of 52.5 CAD and BOI of 44 CAD. At this condition, the exhaust temperature limit was exceeded by 100°F and the cylinder pressure limit was exceeded by 135 psig (107%).

***Single-Cylinder Operation at Idle.*** Replacing the glow plug with the cylinder pressure transducer, the cylinder pressure trace initially gave no indication of combustion at idle. Also, the exhaust temperature from the hydrogen-fueled cylinder was very low (approximately 100°F to 150°F below the other cylinders firing on methanol). Researchers concluded that no combustion was taking place in the cylinder.

Table 5 shows exhaust temperature data with functioning glow plugs for both the hydrogen-fueled cylinder and the average methanol-fueled cylinder. The exhaust temperature from the hydrogen-fueled cylinder showed an expected trend that followed the BOI. This table shows that the exhaust temperature was affected by BOI and was repeatable.

The engine was then shut down and the glow plug was replaced with a cylinder pressure transducer in the hydrogen-fueled cylinder. After a warmup period, the exhaust temperature was taken again at idle with a BOI of 20.0 and PW of 5.7 CAD. For this test condition, the exhaust temperature was 255°F, which is within 5°F of previous tests at this BOI even though the glow plug was not in the cylinder to assist with combustion initiation.

Three additional startup tests with and without the glow plug were completed to verify the idle characteristics of the engine. This verification was done to alleviate concern over the engine's ability to start and idle during future six-cylinder hydrogen operation. Between each of the three idle tests, the engine was shut down and a large fan was placed on the engine to allow it to cool to near-ambient temperature.

**Table 5. Exhaust Temperature at Idle with Glow Plugs Operational** (Fixed PW of 5.7 CAD)

| Run No. | BOI (CAD) | Hydrogen Exhaust Temperature (°F) | Methanol Average Exhaust Temperature (°F) |
|---|---|---|---|
| 179 | 10 | 299 | 219 |
| 180 | 20 | 260 | 228 |
| 181 | 30 | 220 | 235 |
| 182 | 10 | 300 | 219 |

Figure 10 shows the exhaust temperature for the hydrogen cylinder versus time for the first idle test with glow plugs. Any changes in PW or BOI during the test are noted in this figure. Figure 10 also shows when the engine was started.

Figure 11 shows the same sample interval for idle Test 2. A more advanced BOI of 15.0 CAD was chosen with a PW of 6.0, using the glow plug to assist in ignition. Past the 200-second mark, the PW was changed to 6.3 to verify the elevation of exhaust temperature with the increased PW.

Idle Test 3 was performed with a pressure transducer in the cylinder in place of the glow plug. The cylinder pressure was monitored on an oscilloscope during this test to determine if any combustion was taking place without the glow plug. Figure 12 shows the exhaust temperature. As viewed on the oscilloscope, combustion did not start until about 100 seconds. Between 100 and 150 seconds, combustion was sporadic and ragged. After 150 seconds, combustion appeared to be relatively stable. Figure 13

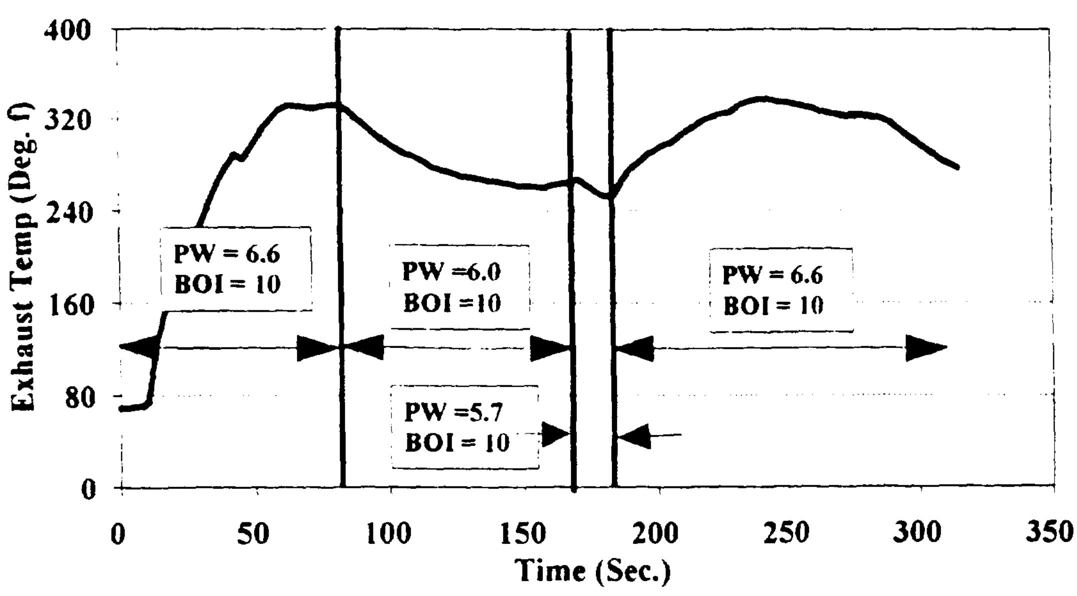


**Figure 10. Hydrogen exhaust temperature versus time for idle Test 1 with glow plug**

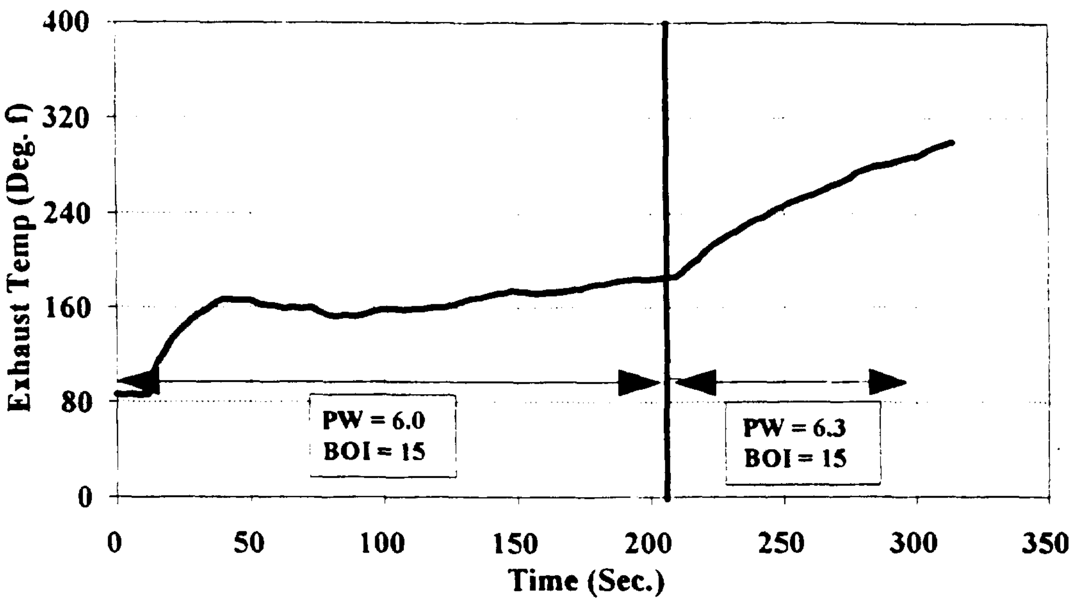


**Figure 11. Hydrogen exhaust temperature versus time for idle Test 2 with glow plug**

shows that two methanol cylinders dropped in exhaust temperature after 100 seconds, about the same time the hydrogen began to ignite.

## Three-Cylinder Hydrogen Idle Tests

For these test, the engine was fitted with three hydrogen injectors, and the three remaining cylinders operated on methanol. These tests were conducted to determine if the entire engine could be operated on hydrogen without starting and idle difficulties. The three hydrogen injectors were fitted with 0.022-inch diameter seven-hole injector tips. Two different BOIs (20 and 15) were tested with a fixed PW of 6 degrees at idle. These tests indicated that the best BOI for startup and idle is 15 degrees before top dead center. This is the same BOI determined during previous single-cylinder testing.

## Six-Cylinder Hydrogen Operation

In these tests, the engine was fitted with six hydrogen injectors. The engine was run with all six cylinders on hydrogen fuel to investigate idle operation, rated power operation, effects of retarded injection timing, effects of reduced injection pressure, and exhaust emissions. All raw data from these tests are contained in Volume II of the two-volume data set.

### *Maximum Power Operation*

In tests conducted to determine maximum power output, the engine produced 324 BHP at 2100 rpm (rated power on methanol fuel was 277 BHP) with a PW of 60 CAD and BOI at 22 CAD. The average exhaust temperature was 1151°F, and peak cylinder pressure was over the 2000 psi limit. The exact cylinder

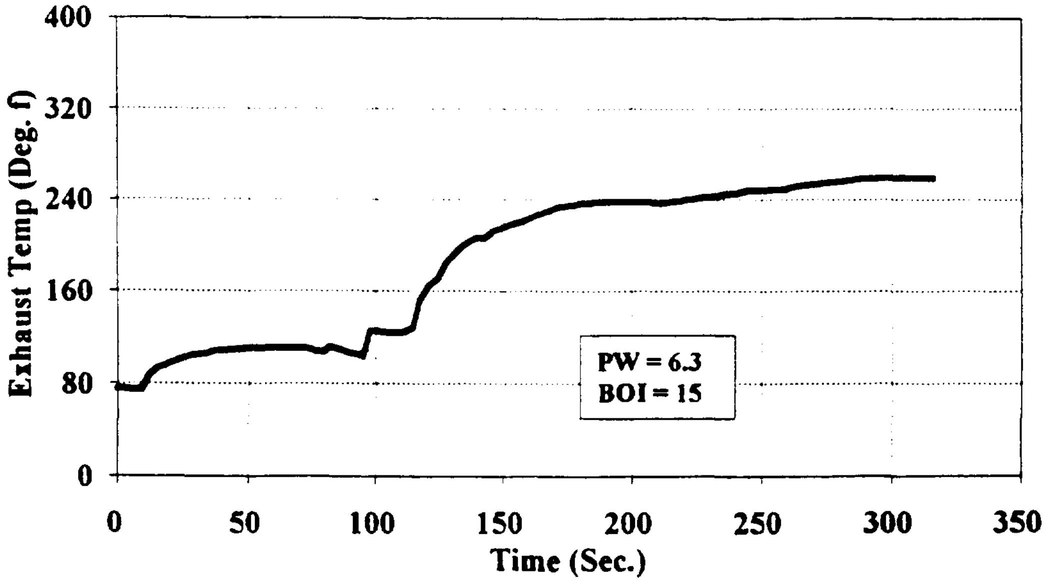


**Figure 12. Hydrogen exhaust temperature versus time for Test 3 without glow plug**

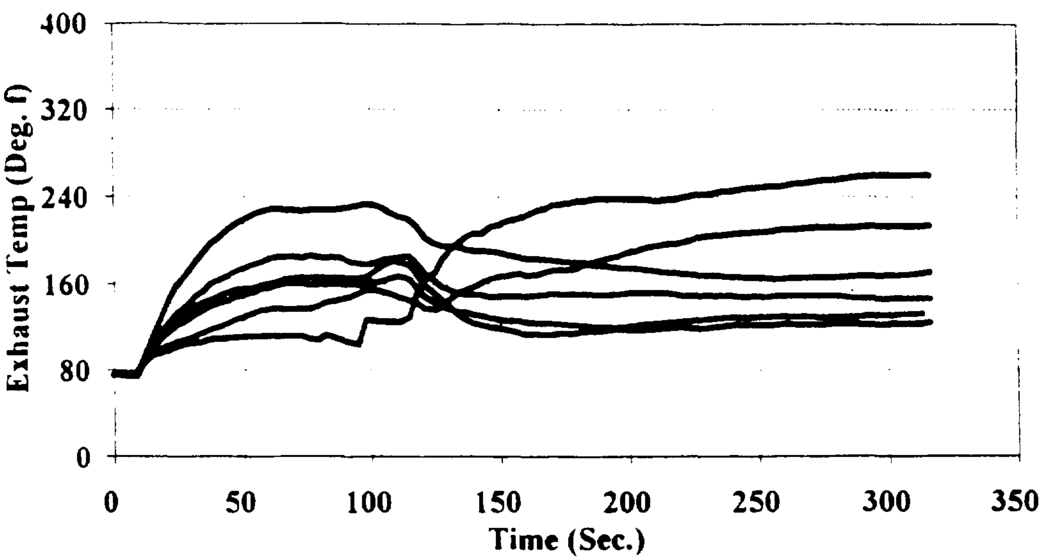


**Figure 13. Exhaust temperature for all six cylinders versus time for idle Test 3**

pressure could not be determined due to thermal failure of the pressure transducer (exhaust temperature on the cylinder with the transducer was 1324°F).

### 11-Mode Emissions Test

Researchers used an SwRI 11-mode steady-state emissions test. Results from this test are shown in Table 6. This table shows performance and corrected emissions (7) with the BOI set at minimum for best torque (MBT). The BTE at rated power (227 hp and 2100 rpm) was 32.1% based on the lower heating value of the fuel. The BTE at peak torque (935 lb-ft at 1200 rpm) was 41.5%. The equivalence ratios for these two points were 0.36 and 0.28, respectively.

Table 7 shows estimated Federal Test Procedure (FTP) transient and other steady-state emissions based on the 11-mode test results. The most interesting result is the predicted FTP $NO_x$ emissions of 4.75 g/hp-hr. This is below the 5 g/hp-hr $NO_x$ standard currently legislated for heavy-duty diesel engines from 1994 through 1997. Hydrocarbon and carbon monoxide emissions result from oil burning in the engine or in the hydrogen compressor.

### Effect of Injection Pressure

The effects of injection pressure on performance and emissions was tested. Performance results from this test can be seen in Table 8. Note that injection timing and engine speed were not varied, while the PW was changed to keep the power constant. Figure 14 shows PW and BTE for these injection pressures. As the injection pressure decreases, the necessary PW becomes significantly longer. The thermal efficiency tends to drop off rather quickly after the injection pressure falls below 2800 psi. Figure 15 shows turbine inlet temperature versus injection pressure. This figure indicates that the exhaust temperature increases as injection pressure decreases. This is due to the longer injection durations needed to maintain power, causing the fuel to burn later in the expansion cycle, thus raising exhaust temperatures.

Figure 16 shows $NO_x$ emissions versus injection pressure. As expected, $NO_x$ decreased significantly with lower injection pressure. Lower injection pressures result in later combustion due to increased injection duration. Lower peak cylinder pressures and temperatures, in turn, result in reduced $NO_x$ production.

### Effect of Injection Timing

Researchers investigated the effect of retarding injection timing at speeds of 2100 and 1200 rpm.

***Retarded Injection Timing at 2100 rpm.*** Figure 17 shows $NO_x$ emissions for MBT, MBT minus 3 degrees CAD, and MBT minus 6 degrees CAD injection timings at 2100 rpm. This graph shows an ever-widening $NO_x$ reduction, down to approximately 150 BHP. Below that point, the 6 CAD retard created a knock condition, so this timing retard was not pursued. Figure 18 shows the BTE for the three test conditions (MBT, MBT minus 3 degrees CAD, and MBT minus 6 degrees CAD). As injection timing was retarded, the expected decrease in thermal efficiency was observed.

Figure 19 shows airbox pressure versus horsepower for MBT and the retarded timings. It should be noted that at 0% load, the airbox pressure increased for the 3-degree retarded condition. Boost air pressure supplied by the turbocharger did not show this increase, indicating that the blower bypass system changed its operating condition as a result of the retarded timing.

## Table 6. 11-Mode Performance Emissions Data at Minimum Best Torque (MBT) Timing

| Mode No. | Run No. | Percent of Rated Speed | Percent Load | Speed (rpm) | Load (lb-ft) | BOI (MBT) | EQ Ratio (Total) | BSFC | BTE (%) | HC (g/hr) | CO (g/hr) | $NO_x$ (corr) (g/hr) | BHP |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 1 | 224 | 100 | 100 | 2097 | 686 | 22 | 0.3557 | 0.1534 | 32.14 | 49.50 | 43.60 | 781.10 | 274 |
| 2 | 225 | 100 | 75 | 2092 | 526.5 | 22 | 0.3033 | 0.1426 | 34.56 | 15.60 | 33.50 | 761.70 | 210 |
| 3 | 226 | 100 | 50 | 2093 | 340.4 | 20 | 0.3080 | 0.1612 | 20.58 | 11.30 | 30.90 | 547.40 | 136 |
| 4 | 227 | 100 | 25 | 2088 | 165 | 18 | 0.2957 | 0.2290 | 21.53 | 12.10 | 22.10 | 220.60 | 66 |
| 5 | 228 | 100 | 0 | 2089 | 5.4 | 18 | 0.1857 | 4.2701 | 1.15 | 5.90 | 15.90 | 145.30 | 2 |
| 6 | 229 | 67* | 100 | 1199 | 934.7 | 16 | 0.2828 | 0.1189 | 41.47 | 13.30 | 19.70 | 1613.30 | 213 |
| 7 | 230 | 67* | 75 | 1199 | 699.4 | 16 | 0.2842 | 0.1209 | 40.79 | 5.20 | 13.70 | 1397.00 | 160 |
| 8 | 231 | 67* | 50 | 1199 | 466.7 | 14 | 0.2855 | 0.1232 | 40.02 | 2.90 | 10.70 | 823.70 | 107 |
| 9 | 232 | 67* | 25 | 1191 | 287 | 14 | 0.2922 | 0.1429 | 34.49 | 2.70 | 9.40 | 344.80 | 65 |
| 10 | 233 | 67* | 0 | 1231 | 2.3 | 12 | 0.1251 | 5.7600 | 0.86 | 1.60 | 5.00 | 125.40 | 1 |
| 11 | 235 | Idle | 0 | 648 | 1.7 | 10 | 0.0772 | 2.8857 | 1.71 | 0.90 | 3.40 | 83.80 | 0 |
| | | | | | | | | **Sum of Columns** | | 121.00 | 207.90 | 6844.10 | 1232.45 |
| | | | | | | | | **G/HP-HR for Cycle** | | 0.10 | 0.17 | 5.55 | |

***Retarded Injection Timing at 1200 rpm.*** When the injection timing was retarded at 1200 rpm, a 6 CAD retard created audible knocking at all loads. Operation at a 3 CAD retard also created an audible knock at no-load. Measurements were therefore not made at these conditions. Figure 20 shows $NO_x$ emissions for MBT and MBT minus 3 degrees CAD. This figure shows that $NO_x$ production for 25% load was actually higher with the 3 CAD retarded timing than at MBT.

Figure 21 shows airbox pressure. Again, below 150 hp, the airbox pressure increased dramatically for the 3 CAD retard when compared to MBT. This could account for the increased $NO_x$ at 25% load shown in Figure 20. At operating points above 150 hp at 1200 rpm, the airbox pressure for the retarded timing was almost identical to that of MBT. It is likely that the ECU is closing the blower bypass at the retarded injection timing.

**Table 7. Emissions Predictions from 11-Mode Steady-State Testing**

| | HC (g/hp-hr) | CO (g/hp-hr) | $NO_x$ (g/hp-hr) |
|---|---|---|---|
| Estimated CARB 8-Mode Emissions | 0.10 | 0.16 | 5.13 |
| Estimated European R-49 Emissions | 0.07 | 0.12 | 6.59 |
| Estimated Japanese 13-Mode Emissions | 0.06 | 0.14 | 7.79 |
| Estimated FTP Emissions | 0.13 | 0.21 | 4.75 |

**Table 8. Effect of Injection Pressure on Efficiency at Rated Power**

| Run No. | rpm | Fuel Flow (lb/hr) | Air Flow (lb/hr) | BOI | PW | Equivalence Ratio | BHP | BTE (%) | Injector Pressure (psi) |
|---|---|---|---|---|---|---|---|---|---|
| 248 | 2096 | 41.2 | 3899 | 22 | 46.5 | 0.36 | 277 | 33.2 | 2940 |
| 249 | 2095 | 41.0 | 3968 | 22 | 51.0 | 0.35 | 275 | 33.1 | 2820 |
| 250 | 2096 | 42.9 | 4055 | 22 | 55.8 | 0.36 | 278 | 31.9 | 2680 |
| 251 | 2096 | 45.9 | 4163 | 22 | 64.8 | 0.38 | 275 | 29.5 | 2480 |

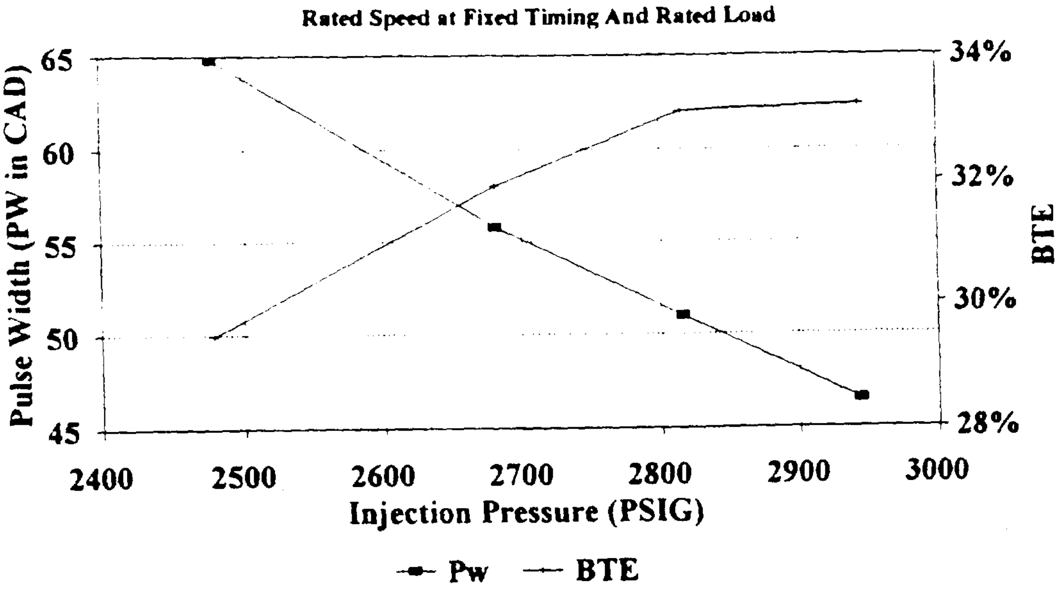


Figure 14. PW and BTE versus injection pressure

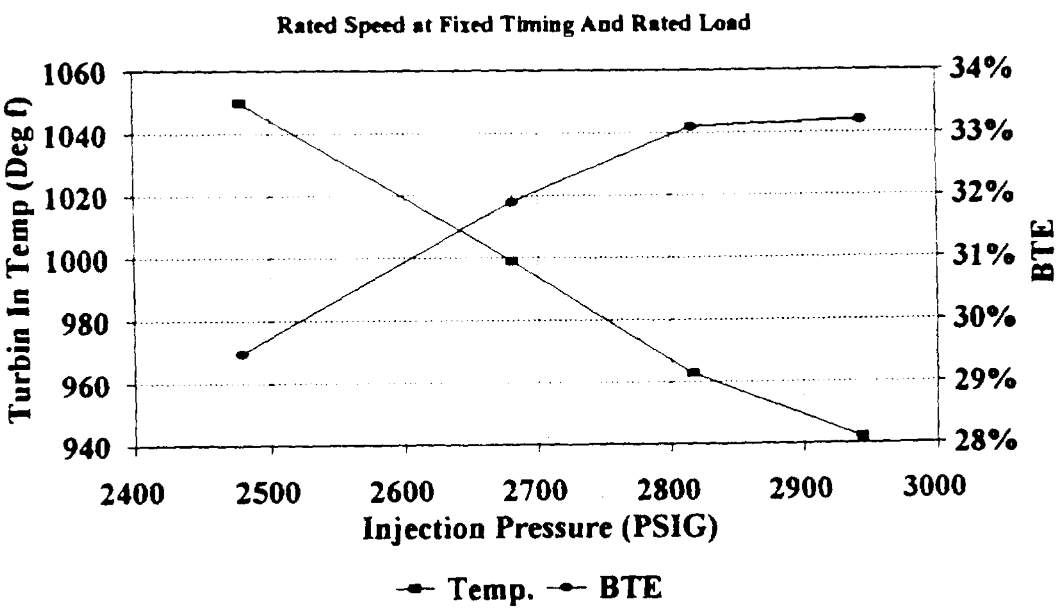


Figure 15. Turbine inlet temperature and BTE versus injection pressure

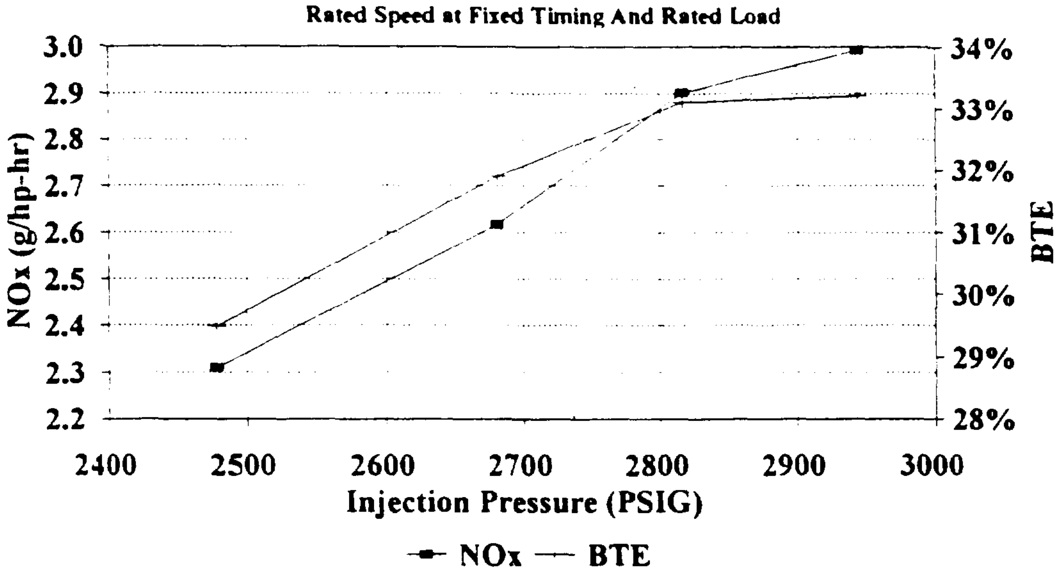


Figure 16. $NO_x$ emissions versus injection pressure

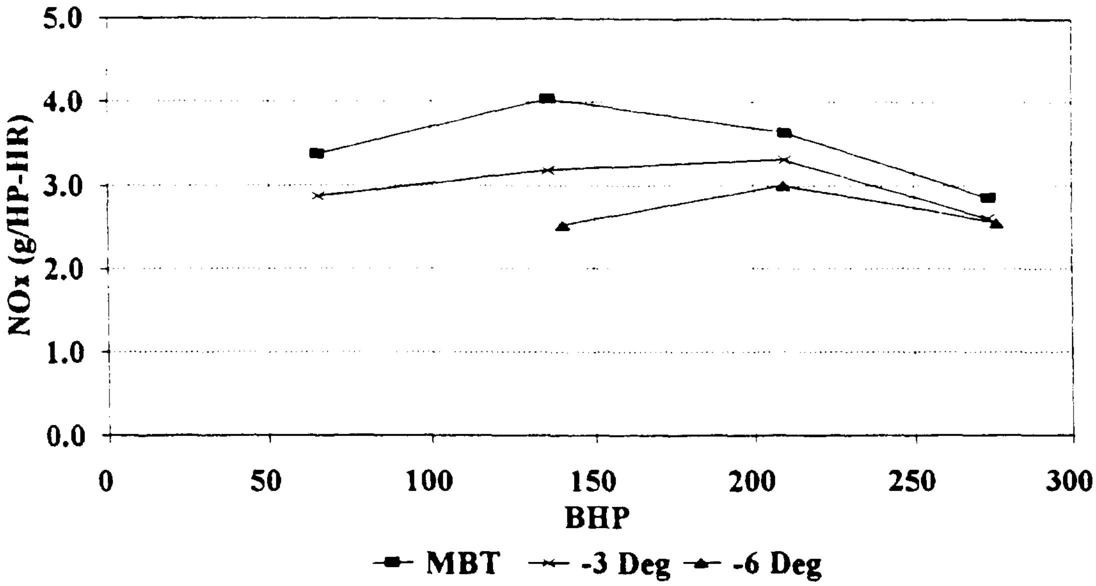


Figure 17. $NO_x$ emissions versus BHP for three different injection timings at 2100 rpm

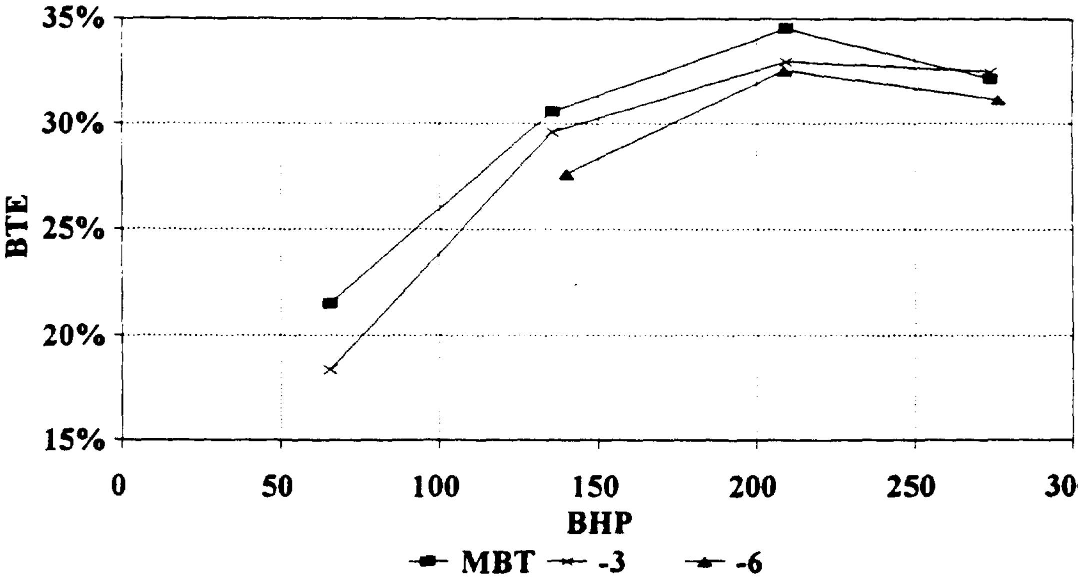


**Figure 18. BTE versus BHP for three different injection timings at 2100 rpm**

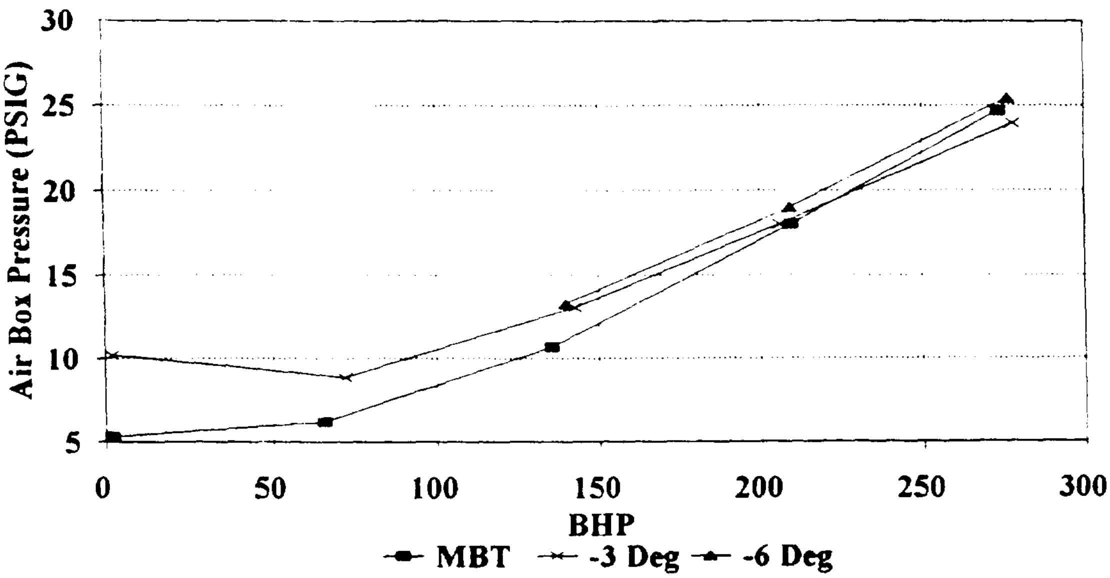


**Figure 19. Airbox pressure versus BHP for three different injection timings at 2100 rpm**

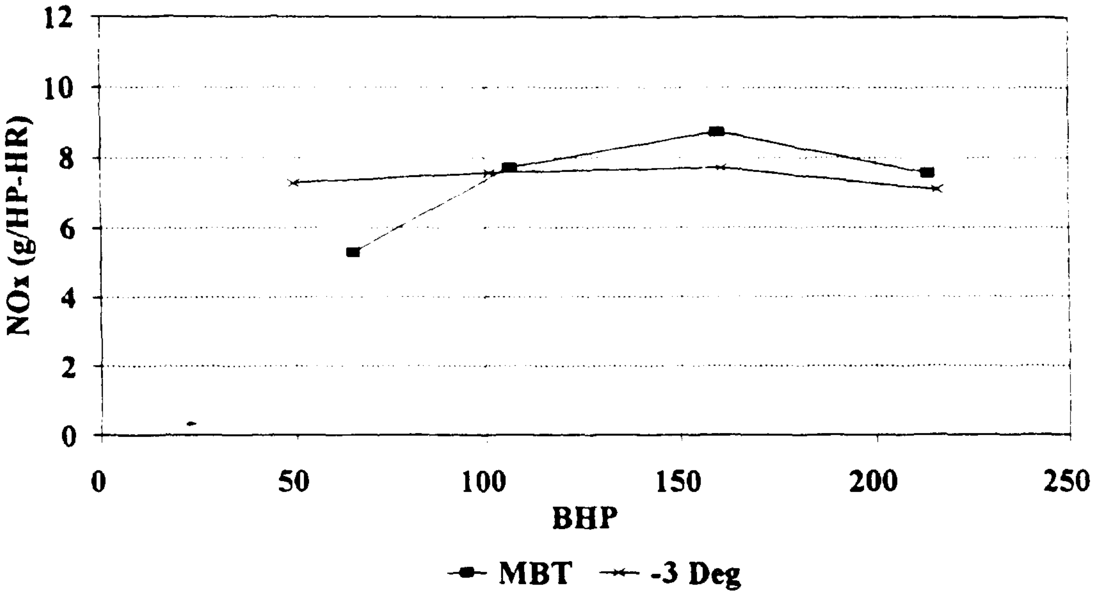


**Figure 20. $NO_x$ versus BHP for two different timings at 1200 rpm**

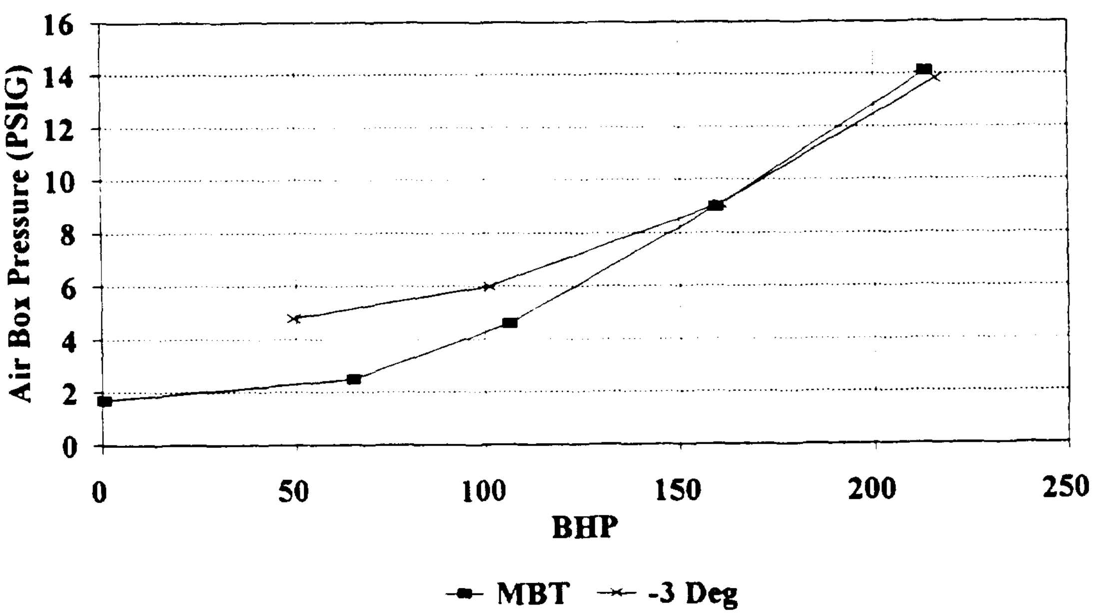


**Figure 21. Airbox pressure versus BHP at 1200 rpm**

# Conclusions

This project demonstrated that the Detroit Diesel Corporation's 6V-2TA methanol engine could be converted to operate on hydrogen with few modifications. In-cylinder injection of hydrogen was shown to be an effective means for fueling an internal combustion engine. This engine performed with good efficiency, performance, and emissions when compared to the baseline operation on methanol. Exhaust emissions were estimated to be below the 1994-1997 heavy-duty diesel standards. Brake thermal efficiencies of 32.1 and 41.5 were obtained at rated power and peak torque conditions, respectively.

The following are some specific conclusions from the testing:

1. Auto-ignition of hydrogen in a suitably modified diesel engine is possible.

2. The 6V-92TA methanol engine rated power and peak torque values of 277 hp at 2100 rpm and 880 lb-ft at 1200 rpm can be achieved using hydrogen.

3. Relatively low exhaust emissions can be obtained by auto-igniting hydrogen, although $NO_x$ emissions are significant (see Table 7).

4. The diesel engine used in this study will start on hydrogen using glow plugs as an ignition source.

5. For the gaseous injector used, an injection pressure of about 3000 psi was necessary for good engine performance (see Figure 14).

6. Considerable development of the hydrogen injector is necessary for reasonable performance and durability.

# Recommendations

This project demonstrated that the 6V-92TA engine is capable of performing satisfactorily on hydrogen fuel. Some issues require additional investigation before this engine will be ready for a demonstration project. After these issues are resolved, it would be appropriate to evaluate the hydrogen operation of this engine in a transit bus. The engine is very popular for this application and hydrogen's low exhaust emissions would be most attractive in an urban environment.

The hydrogen injector used in these tests showed an unacceptable mean time between failures. The injector design was based on the high-pressure natural gas injector developed by DDC, with increased injector tip hole size to accommodate the hydrogen fuel. The most common failure mode for the injector was leakage in the solenoid coil that actuated the injector needle. This leak allowed hydrogen to enter the crankcase. A second failure mode was fuel leakage into the combustion chamber, past the injector needle and seat. The injector design must be modified to solve these problems.

Considerable combustion system development is needed to improve the mixing of hydrogen and air for greater fuel efficiency. Research should include studies of the compression ratio, piston bowl shape, and injector hole number, size, and spray angle. Optimization of the engine air system could entail changing the turbocharger and blower displacement rates as well as resetting the blower bypass.

Once engine performance issues are addressed, the engine should be durability tested to determine if there are any additional long-term problems not encountered during the short time period of this project, such as valve recession or glow plug failure.

For this study, all testing was done under steady-state conditions. Actual transient testing is required to prove the system is applicable for operational vehicles. This would include running FTP transient emission tests and tuning engine operation for best emissions, efficiency, and driveability.

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# Appendix A

## To Convert American Units to SI Units

| American | SI | Multiply By |
|---|---|---|
| psi | Pa | $6.894757 \times 10^3$ |
| ft | m | $3.0480 \times 10^1$ |
| in. | cm | 2.540 |
| °F | °C | (°F–32)/1.8 |
| g/hp-hr | g/kW-hr | 9.80950 |
| lb/hr | g/s | 0.126262 |
| hp | kW | 9.80950 |
| BHP | kW | 9.80950 |

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**13. ABSTRACT** *(Maximum 200 words)*

This report presents the results of a study done by Detroit Diesel Corporation (DDC). DDC used a 6V-92TA engine for experiments with hydrogen fuel. The engine was first baseline tested using methanol fuel and methanol unit injectors. One cylinder of the engine was converted to operate on hydrogen fuel, and methanol fueled the remaining five cylinders. This early testing with only one hydrogen-fueled cylinder was conducted to determine the operating parameters that would later be implemented for multicylinder hydrogen operation. Researchers then operated three cylinders of the engine on hydrogen fuel to verify single-cylinder idle tests. Once it was determined that the engine would operate well at idle, the engine was modified to operate with all six cylinders fueled with hydrogen. Six-cylinder operation on hydrogen provided an opportunity to verify previous test results and to more accurately determine the performance, thermal efficiency, and emissions of the engine.

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