February 2000 • NREL/SR-540-27500

Subcontractor Report

Development of a Direct-Injected Natural Gas Engine System for Heavy-Duty Vehicles

Final Report Phase I

Caterpillar, Inc.
Peoria, Illinois

National Renewable Energy Laboratory

1617 Cole Boulevard
Golden, Colorado 80401-3393

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Contract No. DE-AC36-99-GO10337

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NREL Technical Monitor: Keith Vertin

Prepared under Subcontract No. ZCI-6-15107-01

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FINAL REPORT
PHASE 1

SUBCONTRACT NO. ZCI-6-15107-1

DEVELOPMENT OF A DIRECT INJECTED NATURAL GAS ENGINE SYSTEM FOR HEAVY-DUTY VEHICLES

BACKGROUND

The transportation sector accounts for approximately 65% of US petroleum consumption. Petroleum consumption for light- duty transportation has tended to stabilize in the last 10-15 years, due largely to more efficient automotive systems. Petroleum consumption in the heavy-duty sector (approximately 150-550 hp) on the other hand, has continued to increase. For economic and national security reasons, the US must reduce dependance on petroleum. One significant way to reduce our dependence on petroleum is to substitute "alternative fuels", such as natural gas, propane, alcohols and others in place of the petroleum fuels in heavy-duty applications. Most of the alternative fuels also have the additional benefit of reduced exhaust emissions relative to petroleum fuels, thus providing a cleaner environment.

Homogeneous-charge, spark ignited engines can burn most of these alternative fuels with relatively minor modifications; however, generally, they do not match diesel power density and and they have lower thermal efficiency over the typical operating range. They also require additional radiator capacity because of increased heat rejection. Modifications to incorporate pilot diesel igntion in place of the spark plug, have potential to further improve efficiency, but such systems only approach diesel efficiency and they reduce the substitution of alternative fuel for diesel fuel (pilot diesel systems typically result in 50-80% substitution over a typical operating range). The power density of homogeneous-charge alternative fuel engines is sensitive to fuel quality (i.e., they achieve their highest power density with pure methane and have lower power density with fuels such as propane). Regardless of the ignition system, homogeneous-charge engines are inherently limited to "less-than diesel" power density and efficiency by detonation. To be commercially viable, alternative fuel engines will have to match the diesel in power density and thermal efficiency, and will have to achieve 100% fuel subtitution.

The best long-term technology for heavy-duty alternative fuel engines is the 4-stroke cycle, direct injected (DI) engine using a single fuel. This DI, single fuel approach maximizes the substitution of alternative fuel for diesel and retains the thermal efficiency and power density of the diesel engine.

This contract focuses on developing 4-stroke cycle, DI single fuel, alternative fuel technology that will duplicate or exceed diesel power density and thermal efficiency, while having exhaust emissions equal or less than the diesel. Although current focus is on DI natural gas (DING) engine technology, the technology can relatively easily be applied to other alternative fuels such as propane, DME, alcohols and hydrogen. DING engine technology was chosen for this initial development because it is the most challenging from a technical standpoint (natural gas has the poorest ignition characteristics and gaseous fuels are the most challenging from a fuel handling/injecting standpoint).

The work in this contract is a continuation of work that Caterpillar had initiated with GRI. In the preceding work, DING engine power & efficiency equal or better than that of a diesel had been demonstrated in a single cylinder version of a Caterpillar 3500 Series engine (4.3 liters per cylinder). A 3516 DING engine (16 cylinders) had been built. No emissions development or durability development had been performed.

This report summarizes the results of the 1st year (Phase 1) of this NREL/Caterpillar contract, in which development is focused on DING engine component durability, exhaust emissions and fuel handling system durability. Although the work is currently on a 3500 Series DING engine, the work is viewed as "basic technology" development which can be applied to any engine. Therefore, it will be important in future work to develop design models that can be used to efficiently scale the technology to other engines and to other alternative fuels.

EXECUTIVE SUMMARY

1. Phase 1 has been completed. The following summarizes the results from Tasks 1-3 in Phase 1.

A. Task 1 (Initial Component and Engine Development): Ignition assist and gas injector systems have been identified as the primary areas for future durability focus. No significant distress was observed on other components. Phase 2 work will focus on component durability development using the 3501 (1 cylinder engine), with guidance from analytical models that are being developed for the DING engine. Because the 3516 DING engine has been shown to be an inefficient development platform from time and cost standpoints, a 3126 DING engine (6 cylinder, 1.2 liters per cyl) will be procured for DING and/or DI propane system development. The best component modifications identified on the 3501 will then be demonstrated on the 3126 DING engine. The 3126 is currently used for pickup/delivery trucks and for buses. In addition to being a more efficient development platform, a 3126 DING or DI propane engine will be better suited for eventual transportation sector field demonstration than the 3516 DING engine.

B. Task 2 (Low NOx Development on 3501): 8 mode cycle-averaged NOx emissions were reduced from 11.8 gm/hp-hr ("baseline" conditions) to 2.5 gm/hp-hr (modified conditions) on 3501 DING engine. Thermal efficiency of the engine was substantially compromised to meet this NOx objective. Paths have been identified for future reduction with minimal sacrifice in efficiency. Phase 2 work will focus on demo < 2.5 gm/hp-hr NOx + HC with minimal thermal efficiency sacrifice and will identify the feasibility of achieving NOx < 1 gm/hp-hr.

C. Task 3 (Durability Development of 3000 psi Fuel Handling System): State-of-the-art system technology has been identified. The current level of 3000 psi fuel handling technology was evaluated. It was determined that for direct injected gas engines to be practical in the transportation sector, a liquid natural gas fuel delivery system will generally be required. Cryogenic pump technology requires additional development for mobile applications. Diesel thermal efficiency and lower incremental costs are also required to make the DING engine ecomomically practical. Phase 2 work will demonstrate the feasibility/durability of a 3000 psi LNG pump.

SUMMARY OF PHASE 1 : INITIAL COMPONENT AND ENGINE DEVELOPMENT

Task 1: DI Natural Gas Engine Development

Subtask 1.1 - Initial Durability Evaluation

Objective: The prototype DING 3516 Caterpillar engine shall undergo an initial durability test of 250 hours. At the end of the test the engine shall be disassembled and components inspected to identify areas that need improvement. Life target goals for key components are:

glow plugs > 2000 hours
injectors, valves, etc. > 10,000 hours

Accomplishment Summary:

The 3516 DING engine was inspected after accumulating approximately 250 hours of operation at a variety of running conditions. Indications are that the gas injector and ignition assist system are areas that need improvement.

Accomplishment Details:

The 3516 DING engine has accumulated approximately 250 hours of operation at a variety of operating conditions, up to 2037 hp at 1500 rpm (98% of rated diesel power). By comparison, the homogeneous-charge, lean burn, spark ignited (SI) 3516 gas engine is rated at 1435 hp at 1500 rpm. The majority of 3516 DING engine operation was between 50-75% of rated diesel power. The majority of components in the 3516 DING were basically the same as are in the 3501 DING engine, upon which power and efficiency = to the diesel were previously demonstrated. There were, however, several new systems in the 3516 that were not on the 3501 (initial demonstration engine). These new systems included turbochargers, multicylinder electronic control, hydraulic system for the 16 HEUI-type gas injectors, and a gas compressor system capable of providing 3000 psi gas for the 16 cylinder engine. Photographs and schematics of the system are shown in the attached figures. Although rated power had previously been demonstrated on the 3501 DING engine, the additional complexity of 16 cylinders and these new systems made obtaining full power and the accumulation of test hours difficult. Generally, the "margin for error" of the components in this initial DING system design was not sufficient to allow for the cylinder-to-cylinder manufacturing variations that exist in a multicylinder engine. No significant engine running was possible until all cylinders were operating properly. The improper operation was frequently due to variations in the ignition assist system or in the gas injection system. The lack of sufficient "margin for error" is due to both manufacturing variations in these "early development" components and the fact that additional "design margin" must be incorporated into the DING engine component/system designs. For example, the glow plug ignition assist system that operates satisfactorily in a 6 cylinder methanol engine, has lead to premature glow plug failures in the DING engine. This is caused by a combination of natural gas being more difficult to ignite than methanol (higher glow plug power required) and manufacturing variations between glow plugs. Similarly, gas injector sealing problems (sealing 3000 psi gas from 3000 psi oil) caused improper gas injector operation, due both to manufacturing variations and due to the fact that the sealing design needs to be more robust. Other frequent problems included malfunctioning of system electronics, the gas compressor system and the test facilities (such as dynamometer). It frequently took considerable time to resolve these problems so that all 16 cylinders were functioning properly and significant stable running time was obtained. The problems were typically resolved by either replacing individual components with similar components or by making minor system modifications.

This task showed that the 3516 engine is not an efficient platform for DING system development at this stage of the program. The components are large and costly and 16 cylinders are complex and costly to work with. A 6 cylinder DING engine, preferrably in a smaller bore size, would be much more time and cost efficient for development purposes.

The following pages show the 3516 DING engine hardware that was inspected after testing.

Cylinder heads

Figure 1 shows an overview of the bottom of the head from cylinder number 9. Figure 7 shows this same view of cylinder head number 2.

Intake valve seats

Figure shows a close up of the intake valve seat that had the intake air shield on it, as well as a view of the glow plug shield.

Intake air shields

The air shield (figures 3 and 4) broke off on two separate occasions. Further work is needed to solve this problem. As part of the next phase of the contract, a computational fluid dynamics (CFD) model will be created to help determine the influence of the intake air shield and other variables on the glow plug. Testing has shown that this shield reduces the power required by the glow plug, but the actual mechanism is not completely understood.

Pistons

Figures 5 and 6 show the piston crown in cylinder number 9. The close up view shows areas where the normal carbon deposits were knocked off by the intake air shield. Figure 8 shows the piston in cylinder number 2 for comparison.

Valves

Figures 9 through 12 show the intake and exhaust valves from the above cylinders. No abnormal wear was found on any of these valves. The valve seats were inspected and showed no signs of wear.

Camshaft followers

Camshaft followers (figure 13 and also seen in figure 7) displayed acceptable wear.

Cylinder liners

The cylinder liner in figure 14 shows that the honing pattern is still visible, indicating no wear.

Natural gas injectors

Figure 15 shows a disassembled gas injector. Problems were encountered with o-rings that came in contact with high pressure natural gas. A phenomenon known as explosive decompression (explained elsewhere in this report) caused these o-rings to fail frequently (see figures 17 and 18). Figure 19 shows that an o-ring inside the case had been polishing the stop. Additional work on the dynamics of the high pressure natural gas and hydraulic oil is needed and will be performed in Phase 2. Figure 20 shows an injector tip that was sectioned to study the internal wear surfaces. Figures 21 and 22 show that the check tip seat and orifices are in good condition. Figure 23 shows a close up of the check in the guided stem portion. There is some evidence of scuffing of this part as well as some, as yet, unexplained wear in the injector tip seen in figure 24

Ignition assist system

Glow plug durability continues to be well below the 2000 hour goal. Caterpillar will continue our ongoing glow plug durability program. The glow plug controller used on the 3516 DING was a prototype model. The durability of this controller is unknown since it was remotely mounted and therefore not subjected to engine or vehicle operating conditions. A complete glow plug controller package needs to be developed for mobile applications.

In summary, the main problems at this point, from a reliability and durability standpoint, are injector o-ring, intake air shield failures, and glow plug durability. Future computational fluid dynamic modelling will help determine if the intake air shield is actually required and identify other options for minimizing glow plug power. Additional design work will be undertaken in Phase 2 to resolve the explosive decompression of the o-rings. Caterpillar is working on glow plug durability in an internally funded program, but combustion chamber modifications (including various glow plug and intake air shields) to reduce dependencies on the glow plug will be examined in the next phase of this contract. At this point the wear seen on other components currently does not appear serious enough to warrant undue concern.

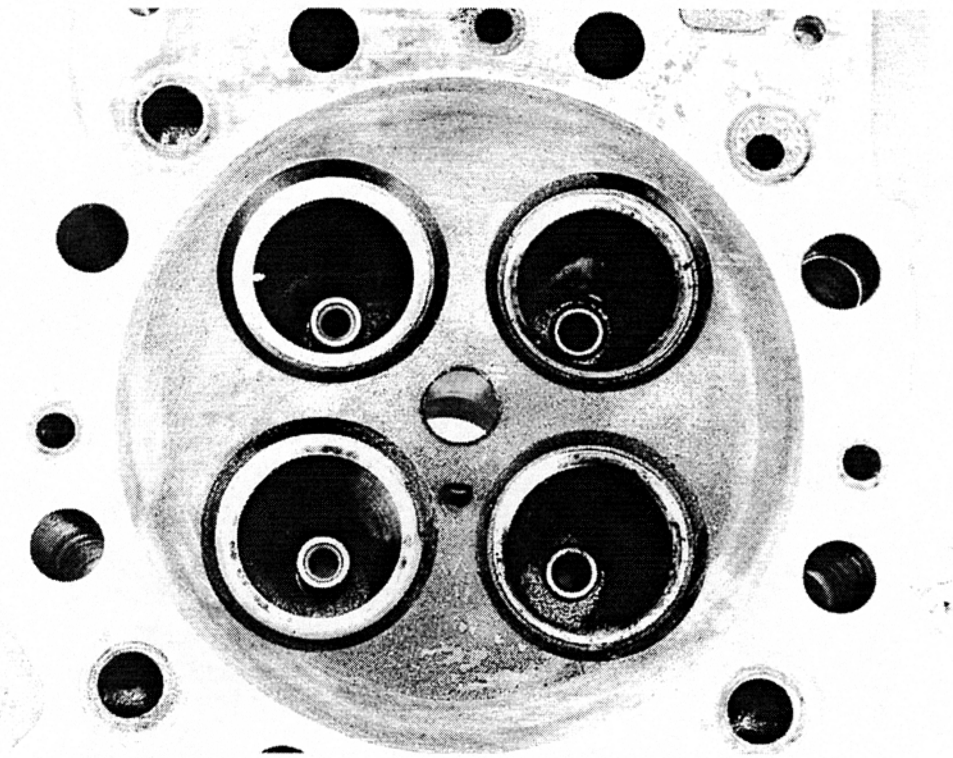


Figure 1: Bottom deck of cylinder head #9

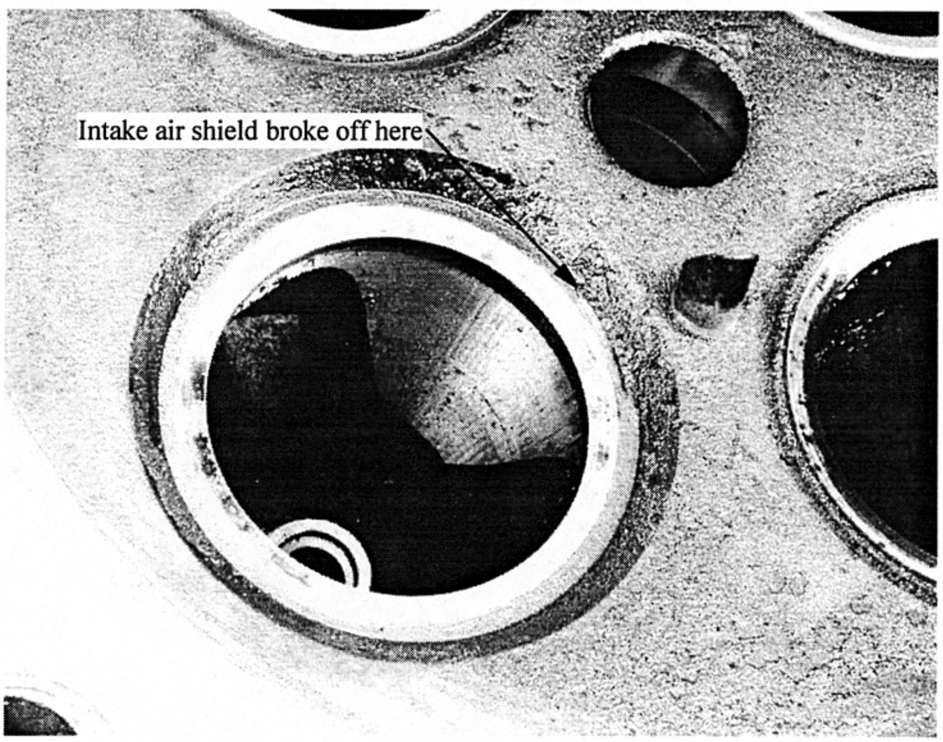


Figure 2: Close up of intake seat and glow plug shield

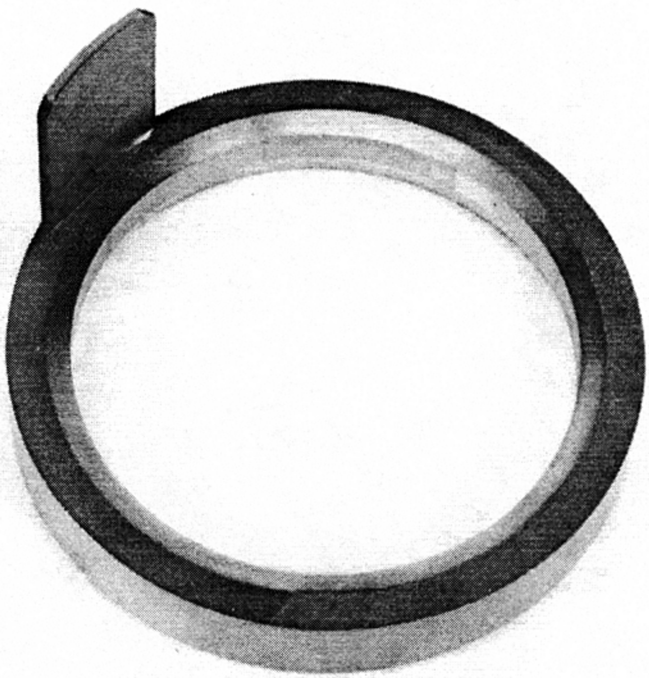


Figure 3: Intake valve seat with integral air shield

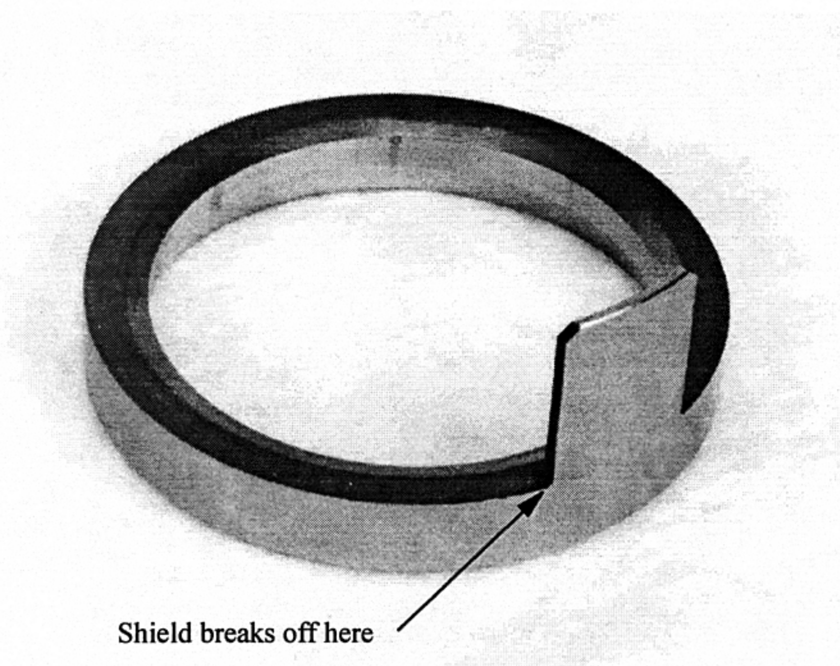


Figure 4: Intake valve seat

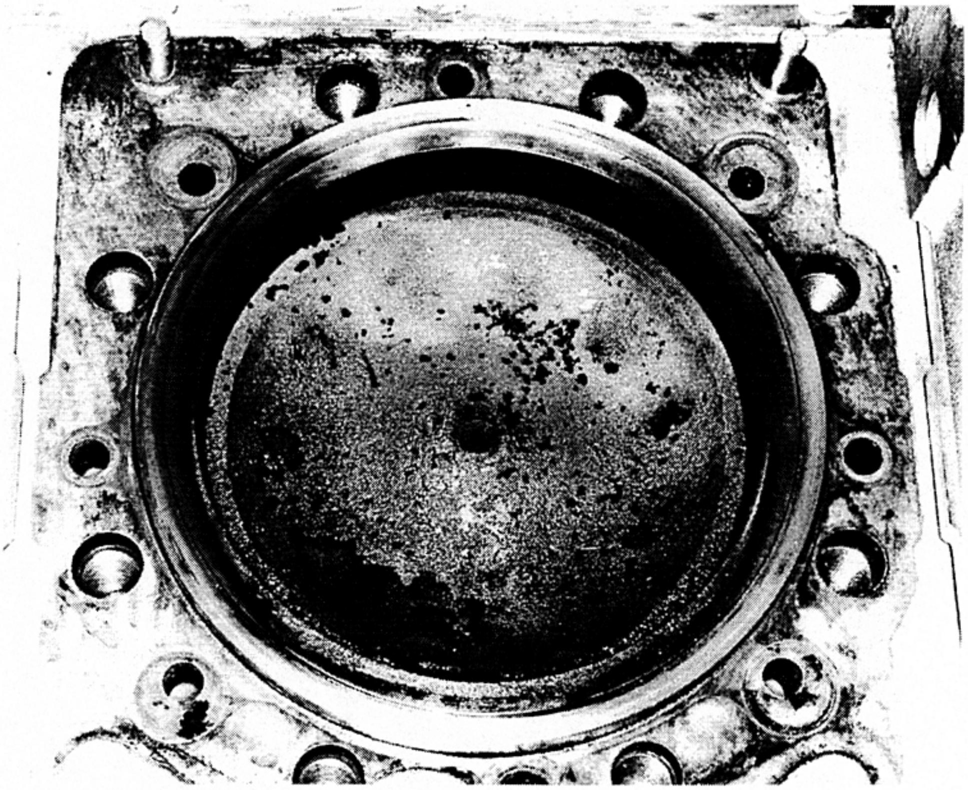


Figure 5: Piston in cylinder #9

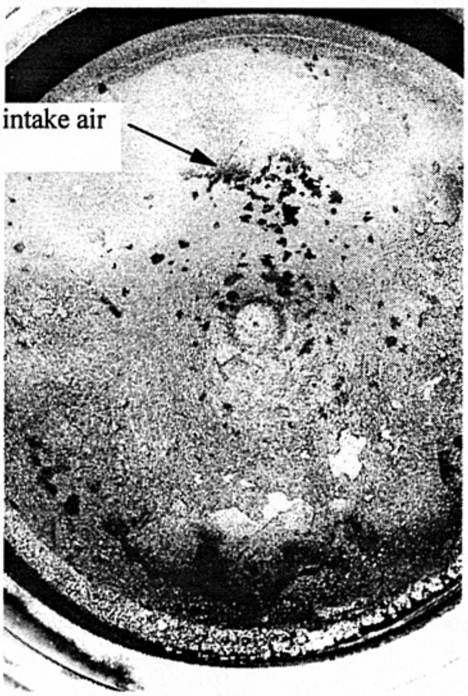


Figure 6: Close up of piston crown

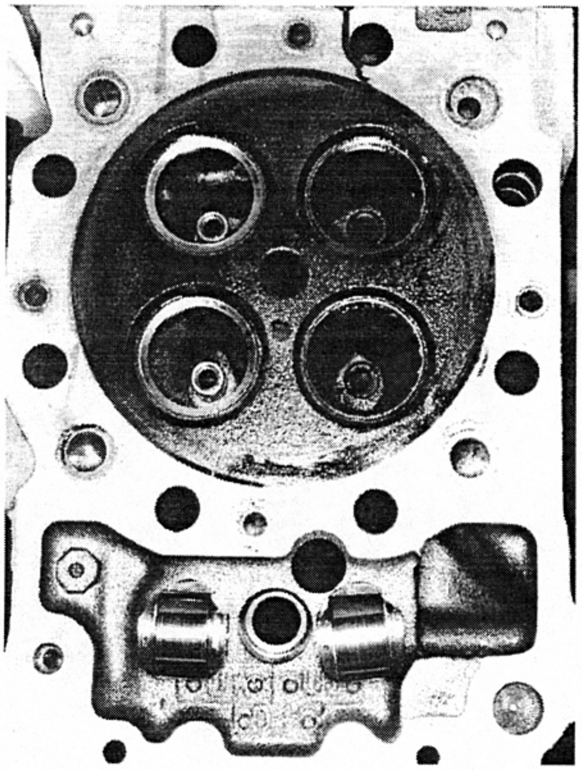


Figure 7: Cylinder head #2

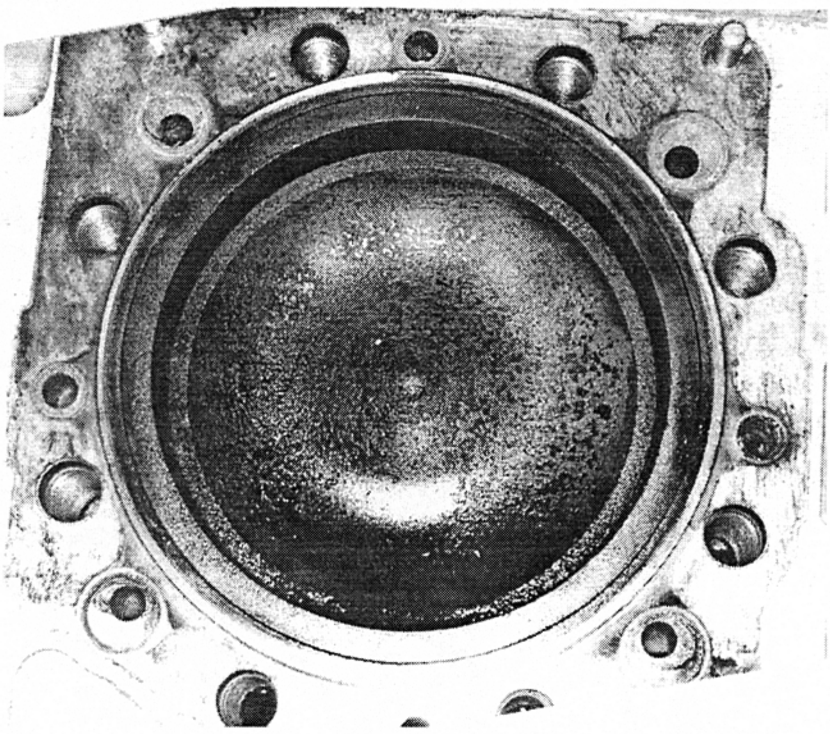


Figure 8: Piston #2

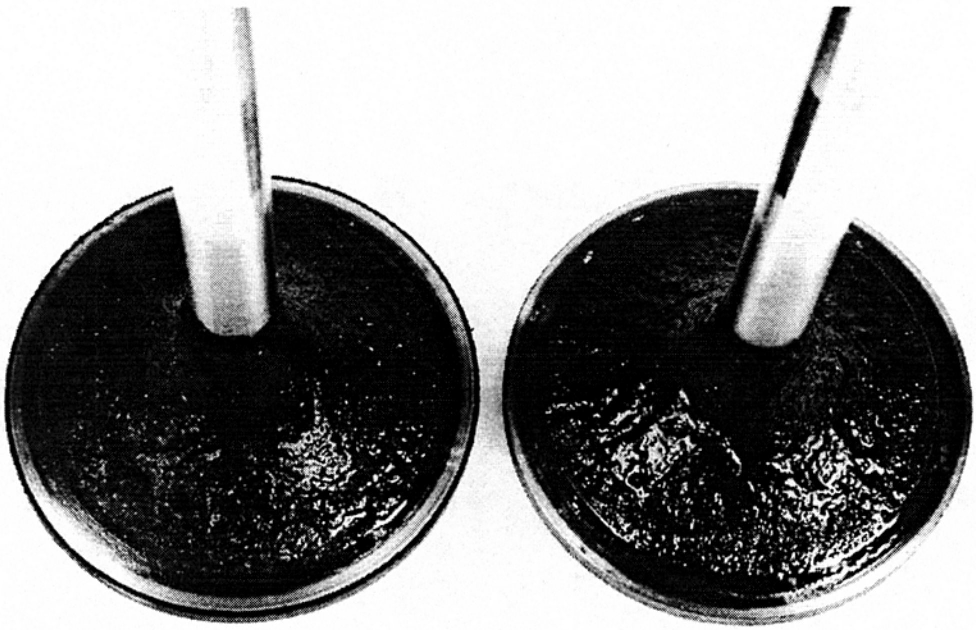


Figure 9: Intake valves from cylinder #9

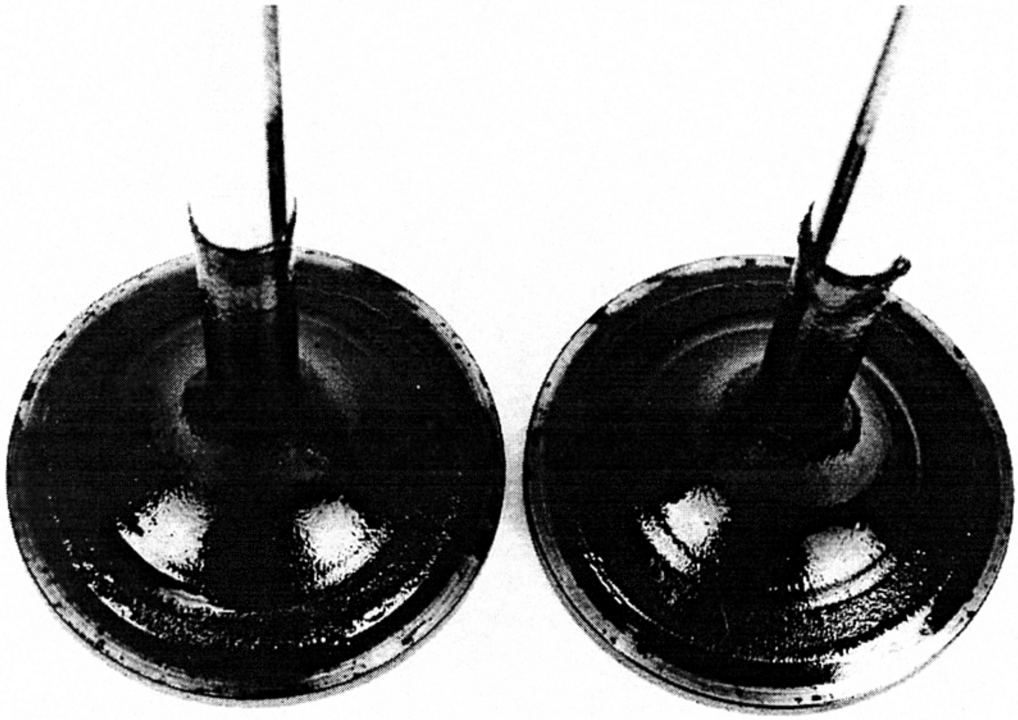


Figure 10: Exhaust valves from cylinder #9

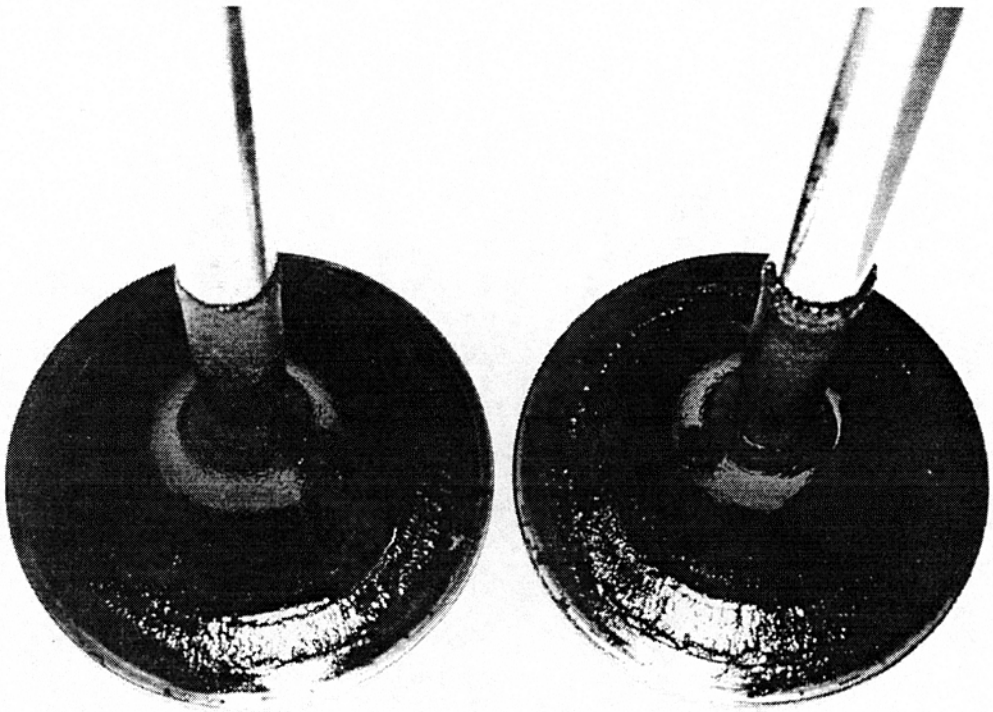


Figure 11: Intake valves from cylinder #2

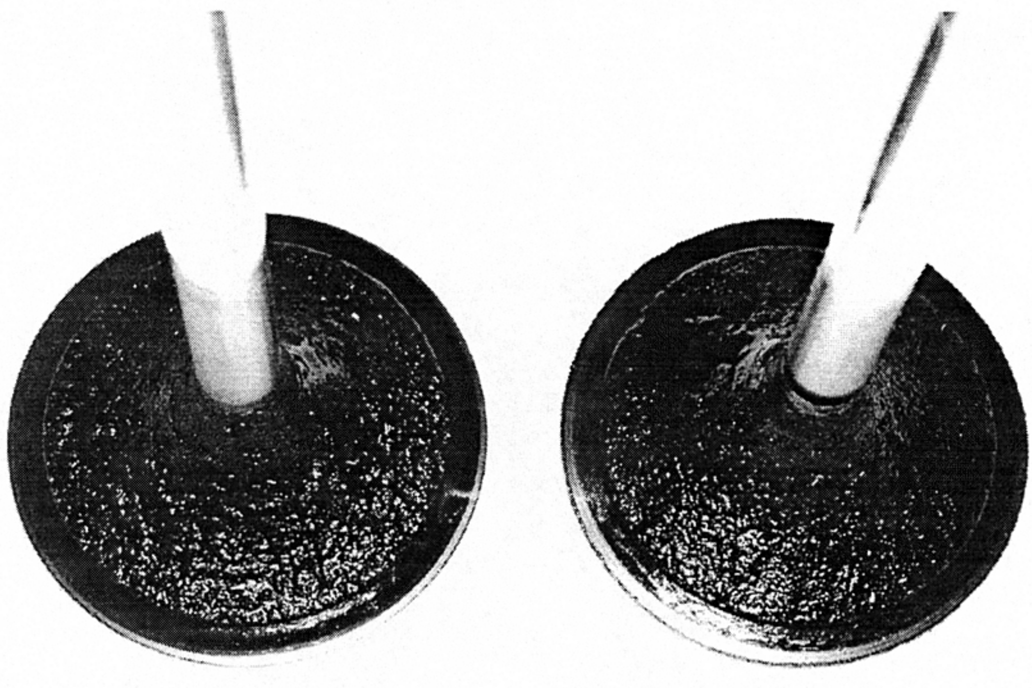


Figure 12: Exhaust valves from cylinder #2

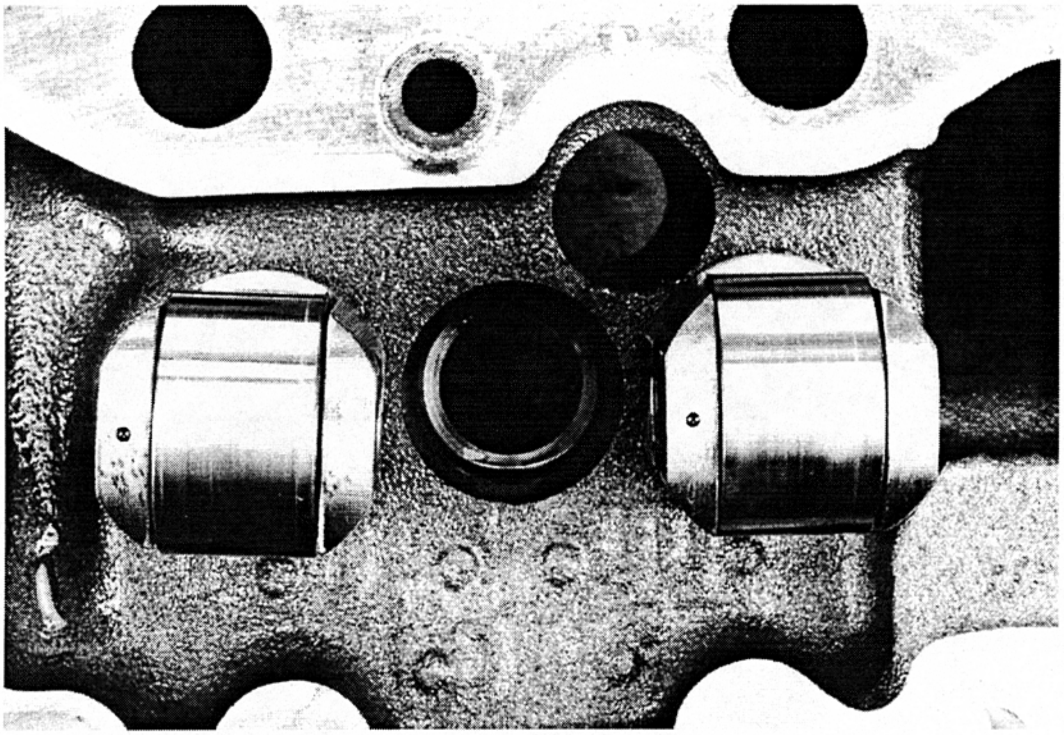


Figure 13: Camshaft followers for cylinder #9

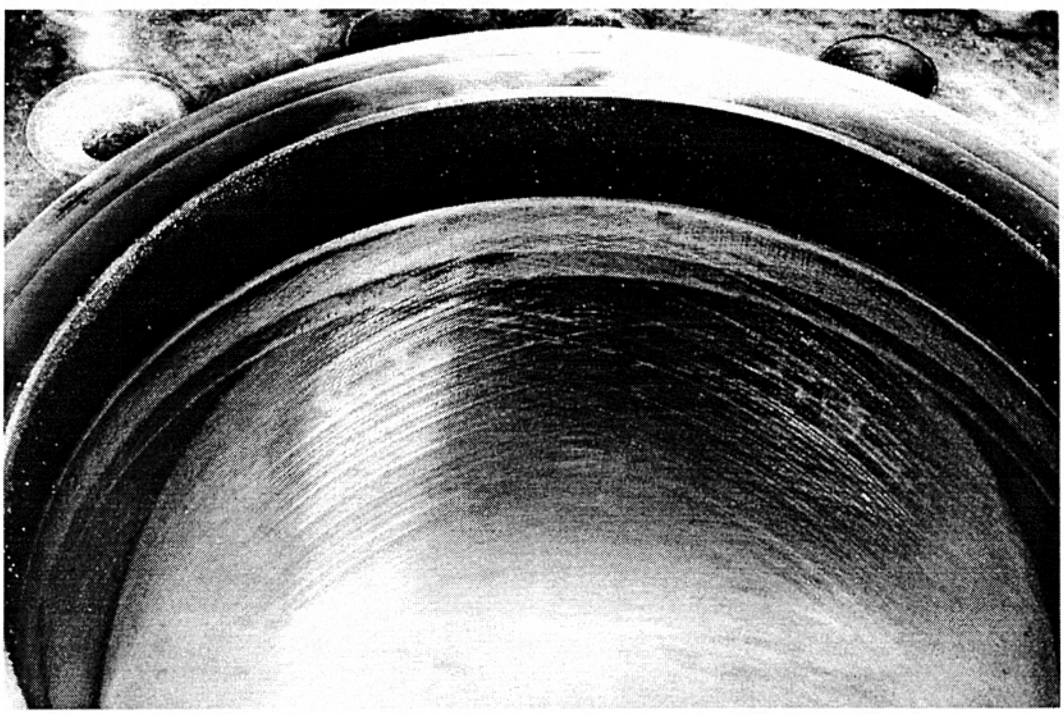


Figure 14: #9 cylinder liner - honing pattern is still visible

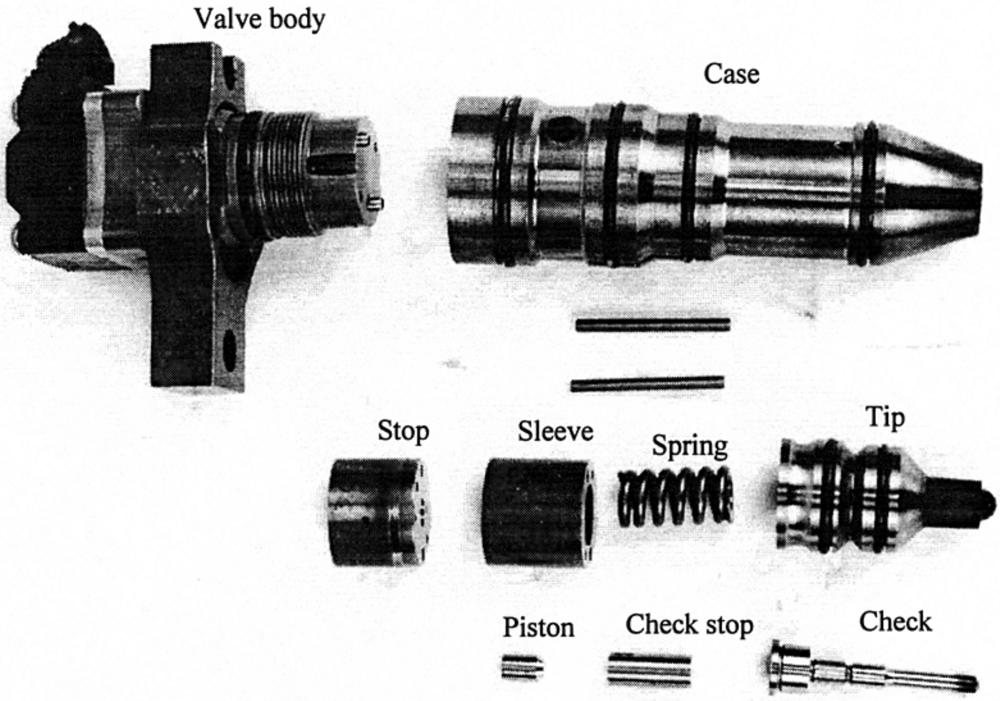


Figure 15: Disassembled injector

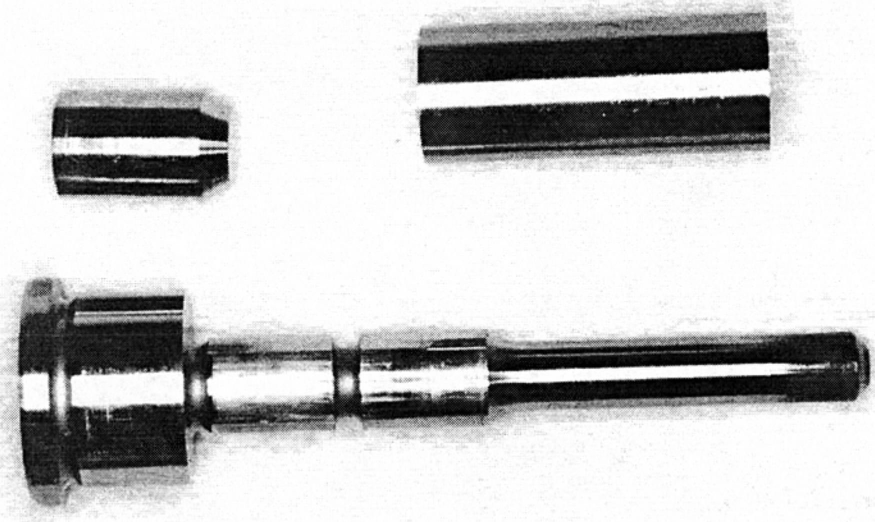


Figure 16: Close up of injector check, piston, and check stop

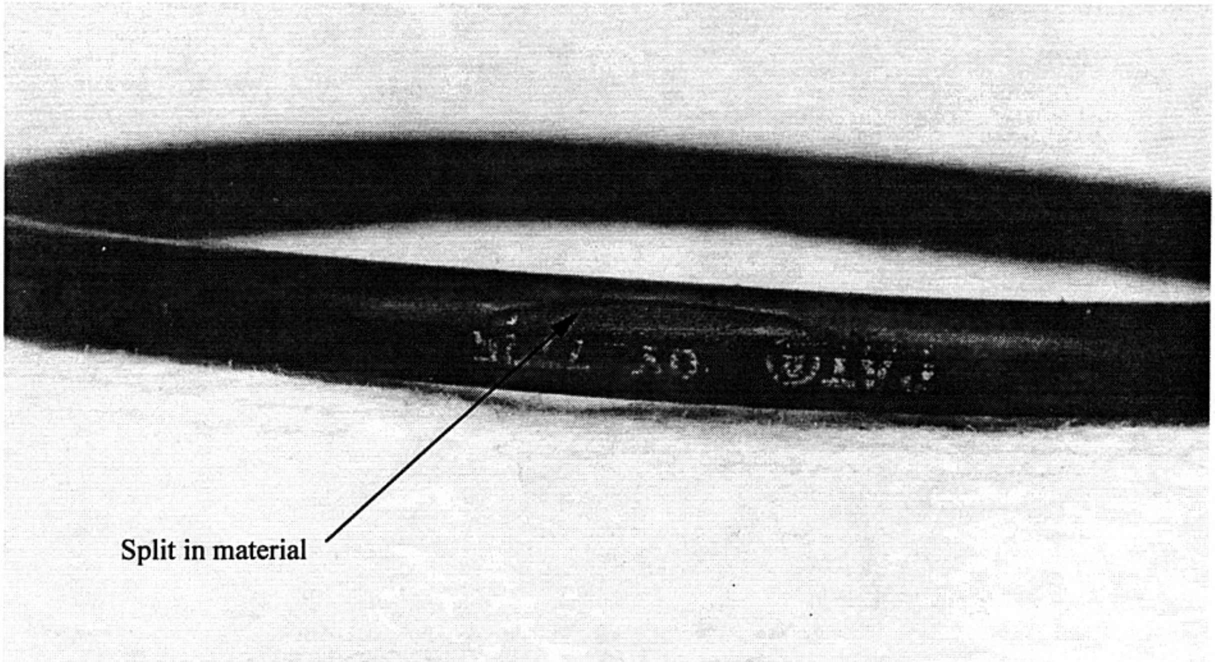


Figure 17: O-ring showing damage due to explosive decompression

Figure 18: O-ring failure due to explosive decompression

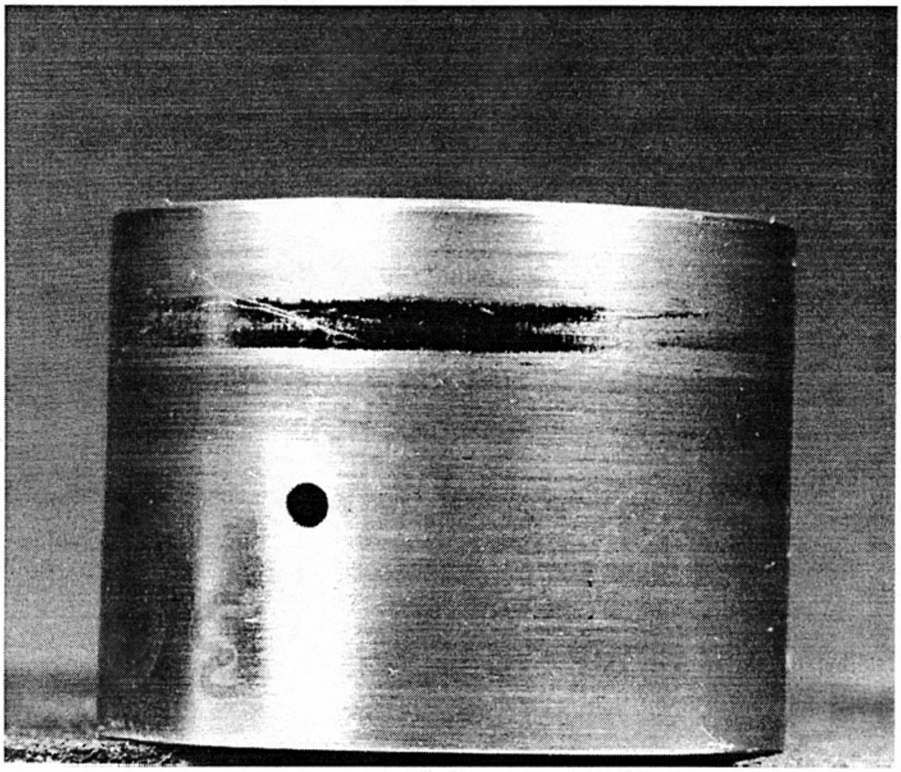


Figure 19: Injector stop showing polishing from case o-ring

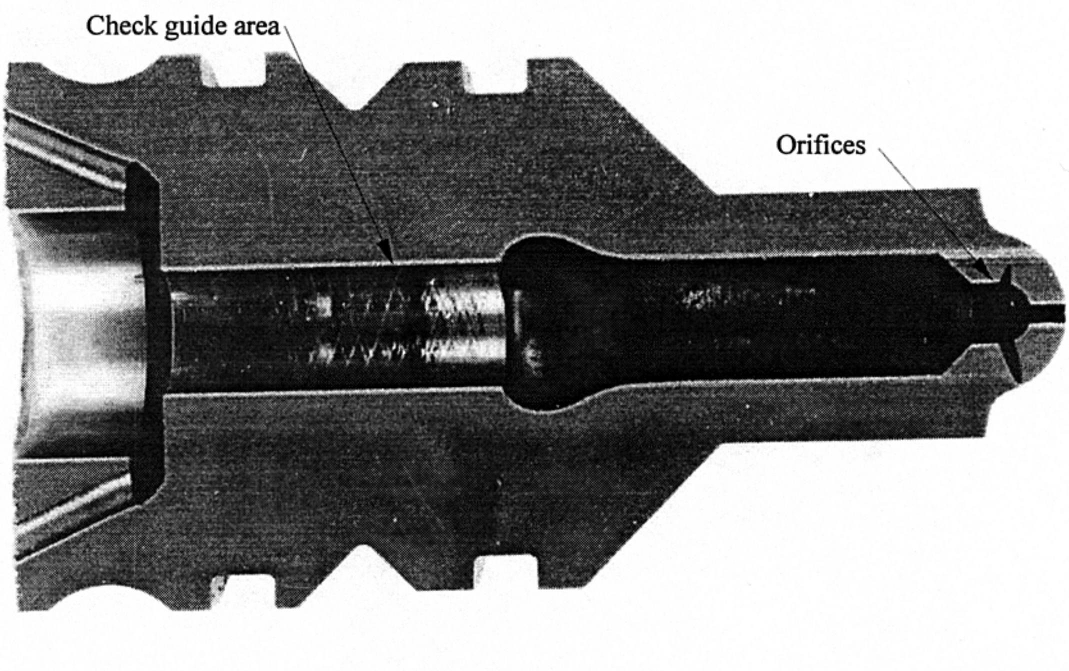


Figure 20: Cross section of injector tip

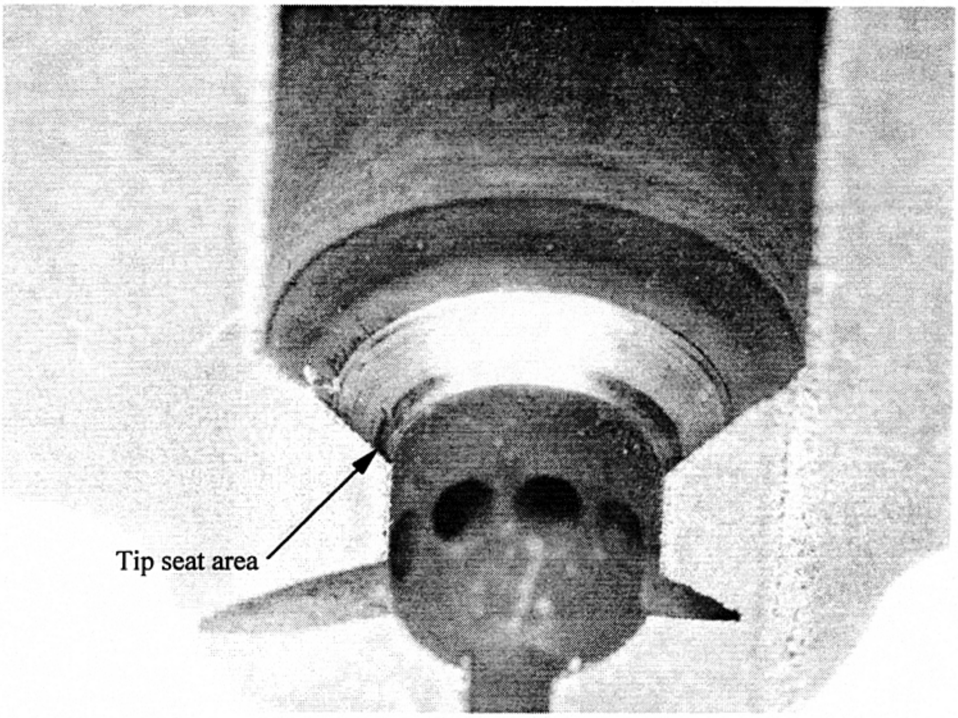


Figure 21: Injector check seat

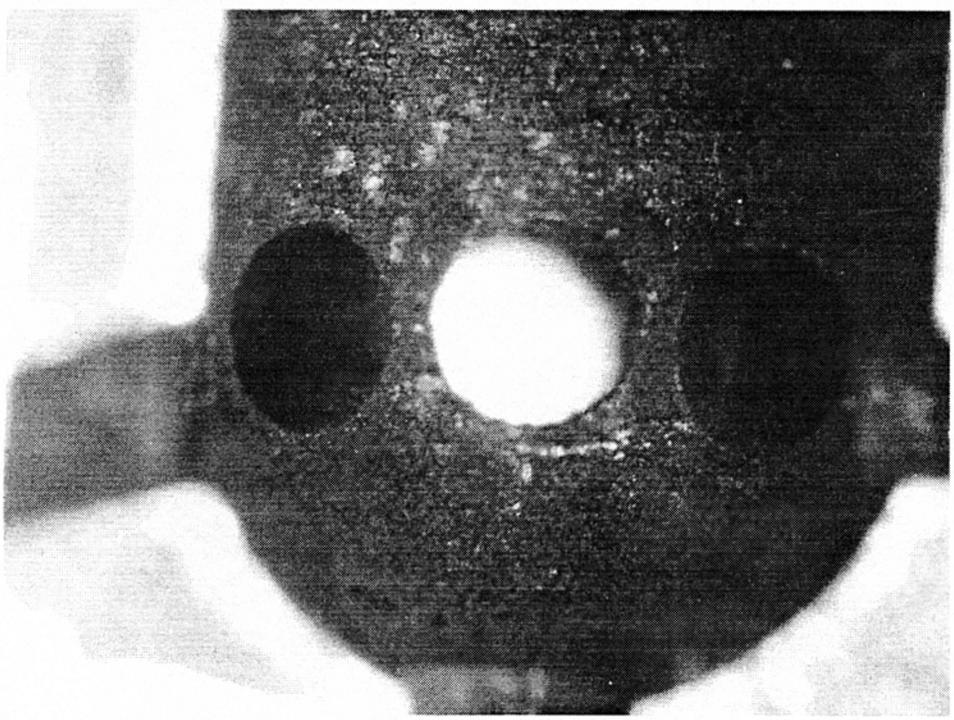


Figure 22: Close up of injector tip orifices

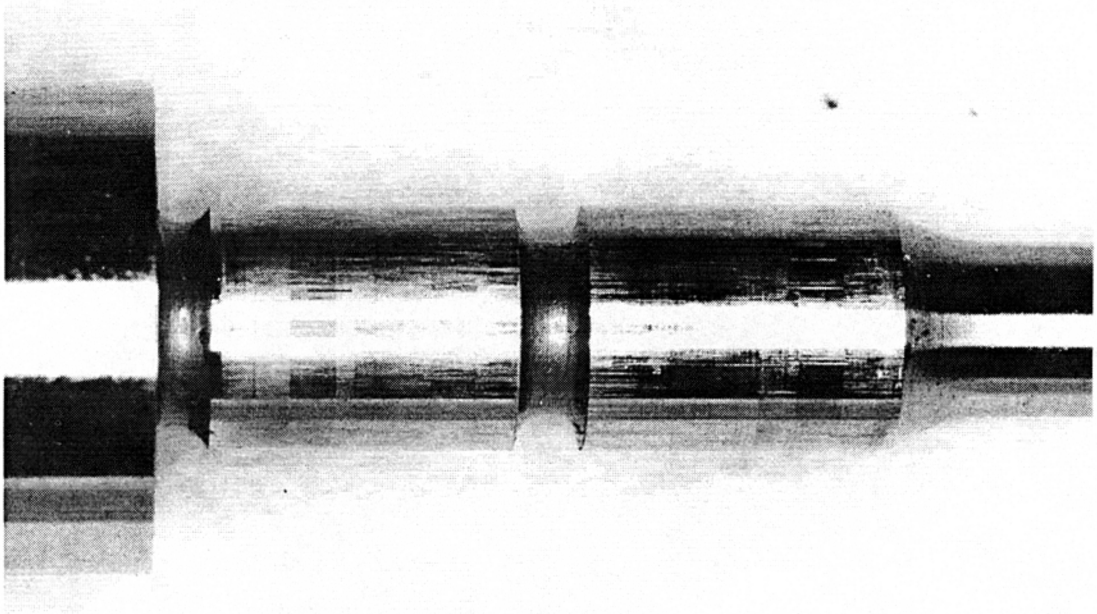


Figure 23: Check valve area showing wear pattern

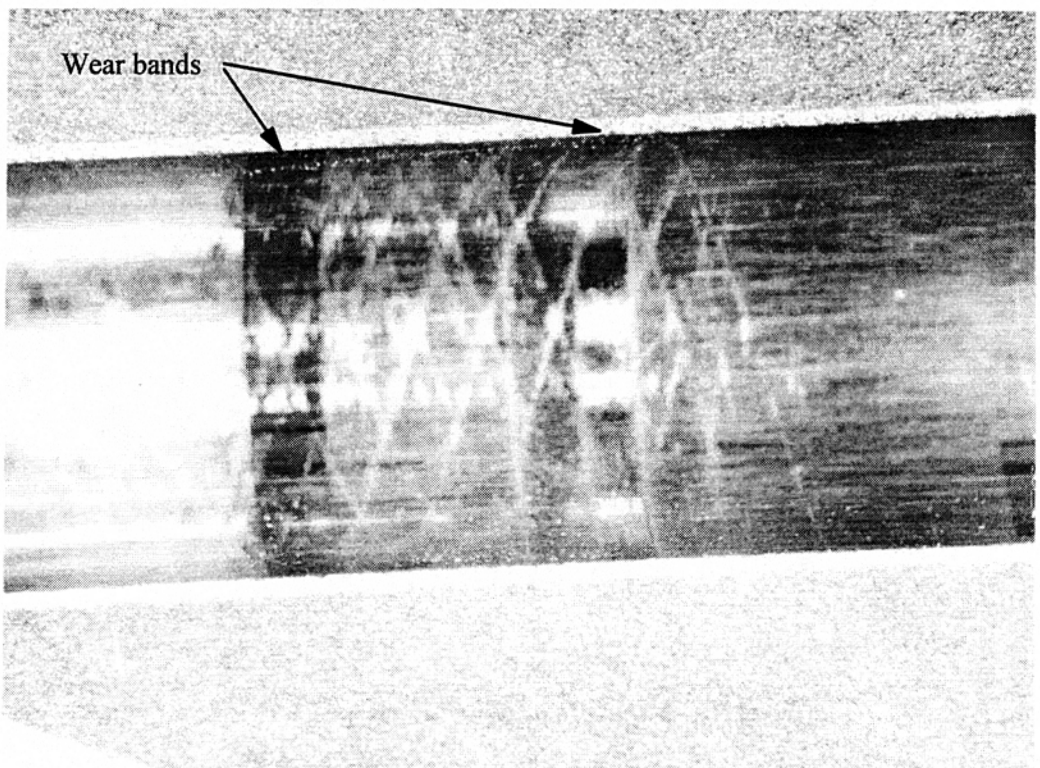


Figure 24: Injector check guide bore corresponding to figure 20 above

Subtask 1.2 - Component Design Modification

Objective: Based on the results from the initial durability testing, design modifications shall be made to the individual components. Procure components and incorporate in the engine.

Accomplishment Summary:

Modifications to help the gas injector sealing and glow plug ignition assist were implemented. These modifications have improved the engine operation.

Accomplishment Details:

Because of the "margin for error" problems that were identified during operation of the 3516 DING engine in Subtask 1.1, several modifications were implemented during the course of attempting to accumulate test hours, instead of waiting for the end of the 250 hour durability test. These modifications were effective in improving engine operation. The engine is much better able to sustain steady operation now than at the beginning of this task. For example, the engine has been run for several hours to obtain detailed performance data that can be used to analyze engine performance. Such data was used to identify a dynamometer malfunction that was giving erroneous readings. Data analysis indicated that in previous tests, the engine had probably been run well above 100% diesel power, even though the dynamometer reading indicated much lower power. The highest officially measured power to which the engine has been run following the dynamometer correction, has been 98% of rated diesel power. Attempts to obtain detailed performance data at 100% diesel power have been unsuccessful so far because of the "margin of error" problems previously described and due to an electronic control problem. Additional running to document performance at 100% power was suspended so the engine inspection in Subtask 1.1 could be made.

Design modifications that were examined or implemented during Subtask 1.1 are summarized below:

1. Gas injector sealing: The current 3500 DING gas injector contains o-rings that seal 3000 psi natural gas from 3000 psi oil (figure 15). If the seals fail, improper injector operation can result. Occasional o-ring failure due primarily to "explosive decompression" has frequently been observed (figures 17 and 18). The seal damage occurs when high pressure natural gas permeates and gets trapped in the pores of the o-ring. When the pressure surrounding the o-ring is suddenly reduced (due to pressure waves or sudden engine shut-down) the pores "explode", thus causing damage. A system was incorporated to prevent the sudden release of gas pressure (i.e., let the pressure bleed down at a slow rate). Early indications are that this system has greatly alleviated the problem. Additional modifications have been considered/procured that may be incorporated later, if additional testing indicates further improvement is required. Such modifications include Teflon-coated o-rings, spring-energized Teflon rings, and modifying the seal design to eliminate o-rings from high pressure gas sealing.

2. Glow plug electronic control: Initial testing of the 3516 DING engine was performed using a constant voltage glow plug controller. For any given operating point, the voltage to the glow plugs was fixed. This required manually adjusting the voltage for different conditions due to the varying demand on the glow plugs, or overpowering the plugs at some points so they would operate all the time. Later testing was performed with a constant resistance controller. A resistance level was set in the controller. The resistance of an operating glow plug is related to it's temperature. The controller measured the current and voltage from the alternators to the glow plug and calculated the glow plug resistance. It compared this calculation to the set point, then varied the output voltage of the alternators to keep the resistance at the set point. This improves glow plug life by reducing glow plug power to the minimum required at each operating point, as well as allowing for autonomous operation.

3. Glow plug (funded entirely by Caterpillar): Various glow plug material configurations were evaluated. None of the configurations significantly reduced the voltage needed for ignition, but they may help extend glow plug life. Modifications such as shielding, combustion chamber geometry and injection modifications are expected to have a favorable affect on reducing glow plug voltage. Such modifications will be evaluated in Phase 2.

Two items can significantly facilitate identification and verification of the needed improvements in Phase 2. They are:

(1.) further development and utilization of DING injection/combustion models. Such models can be an extension of models that Caterpillar has previously developed for diesels. This model development needs to be accelerated and the models used to facilitate identifying modifications to reduce glow plug power.

(2.) procurement of a 6 cylinder, smaller bore size DING engine for system development purposes. As mentioned in Subtask 1.1, the 3516 DING engine is an inefficient DING technology development platform. A Caterpillar 3126 engine (6 cylinders, 1.2 liters/cylinder) has been selected as the best DING technology development platform. In addition to being compact (6 cylinders, smaller size), it (1.) incorporates a HEUI injector system, therefore, facilitating making a HEUI DING injector, as is in the 3500 DING engine, and (2.) is a popular engine in transportation applications (pickup/delivery and buses), thus providing an engine that can be used in transportation system field demonstrations. The current 3500 DING technology will be scaled to such an engine.

Task 2: DI Natural Gas Engine NOx Development

Subtask 2.1 - Design or Modify Prototype DING Engine to Meet 2.5 g/hp-hr NOx Goal

Objective: Make initial engine design/modifications on a 3501 DING engine to meet the initial goal of 2.5 g/hp-hr NOx.

Accomplishment Summary:

A camshaft that had an early exhaust valve closing providing for in-cylinder exhaust gas recirculation (EGR) was designed, procured, and tested. A piston that provided a higher compression ratio, which had been used in previous performance testing, was installed and tested to determine NOx reduction potential. Exhaust valves with a catalytic coating that will provide a more controlled combustion, thus reducing NOx, have been procured.

Accomplishment Details:

Background:

There are many ways of reducing NOx emissions. Usually these methods result in lower average in-cylinder temperatures. Table 1, lists many traditional methods of reducing NOx emissions. Typically there is a tradeoff between reduced NOx emissions and efficiency. However, this tradeoff can be minimized by reducing the spread of the in-cylinder temperature distribution while still maintaining the same average in-cylinder temperature. Better mixing of fuel and air so the turbulent diffusion flame, controlled by injection rate and mixing, occurs at leaner than stoichiometric conditions could result in the same average cylinder temperature with lower localized peak cylinder temperatures. Reducing ignition delay would result in less premixed combustion and the high heat release rates characteristic of this phase. Both the increased compression ratio and exhaust gas recirculation (EGR) cam tested are expected to reduce ignition delay.

Modification	Trends
Retard injection timing	Reduces amount of combustion that occurs before top dead center (TDC) which would be compressed, heated, and speed the NOx formation rate further. Reduces cylinder pressures and temperatures. Retarding timing alone will reduce NOx at the expense of thermal efficiency.
Reduce Injection pressure	Reduces the amount of fuel in the cylinder before combustion occurs (Premixed). Reduces rate at which fuel is injected. Reduces mixing turbulence and burn rate. There should be an optimum injection pressure based on speed & load. Reducing below this will reduce NOx at the expense of thermal efficiency.

Injection Rate Schedule	Injecting fuel slow at first and fast later should reduce premixed combustion and heat release rate before TDC but increase later. Should reduce NOx with lesser effect on thermal efficiency.
Catalyst equipped glow plugs	Unique to our engine, this can reduce ignition delay and the amount of premixed combustion. Does reduce power required to sustain combustion.
EGR	EGR dilutes unburned mixture and acts as a thermal sponge. Also reduces the availability of oxygen which reduces the heat release rate. Should offset NOx - thermal efficiency tradeoff.
Increased Compression Ratio	Will reduce ignition delay and amount of premixed combustion. Also allows engine to operate near peak cylinder pressure (PCP) with less combustion before TDC. Can reduce NOx with small effect on thermal efficiency. Also, works well when combined with lower inlet air temperatures. However, can require retarded timing to meet PCP limits.
Reduced inlet manifold air temperature	Reduces absolute temperature reached after combustion. Reduces NOx & can increase thermal efficiency. Can increase ignition delay.
Exhaust Catalyst	NO is removed by reduction using the CO, H2, and hydrocarbons in the exhaust. Typically requires stoichiometric combustion.
Water Injection	Reduces inlet charge temperature when water evaporates as well as reducing peak cylinder temperatures by absorbing large amounts of thermal energy. Will reduce NOx with small effect on thermal efficiency.
Air - Fuel ratio	Increasing amount of air in cylinder reduces overall average in cylinder temperature reached after combustion.
Injector Nozzle	Effects mixing and local temperature phenomena. Believed to be optimized in our system at higher loads for efficiency.
Split Injection	Injecting fuel early, to allow it to mix homogeneously into a lean mixture that will not burn until the main injection event brings the air-fuel ratio above the flammability range, should reduce NOx.

Table 1, NOx reduction strategies

There are several ways to achieve exhaust gas recirculation (EGR). From a hardware standpoint one very attractive way is to modify the camshaft to close the exhaust valve early and trap the exhaust gas in the cylinder. The drawbacks to this system are; the exhaust gas can not be cooled, the amount of EGR is fixed, the EGR is always active, and there is a penalty in volumetric efficiency. The penalty in volumetric efficiency derives from having to open the intake valve later and the reduced valve lift. The intake valve is opened later so the work that went into compressing the trapped exhaust gas is not all lost.

One way of inducing EGR in a turbo charged engine is to use a control valve located off the exhaust manifold and route the exhaust gas back into the intake manifold. This will only work when the back pressure created by the turbocharger is greater than the boost. This type of system is more complicated but allows the amount of EGR to be regulated. It also allows for the option of cooling the exhaust gas that is to be regulated.

An engine cycle simulation was run to identify cam modifications that would provide approximately 10% and 20% EGR. Two cams for the 3501 DING engine were designed and procured. Both cams closed the exhaust valve early to trap exhaust in the cylinder. Both cams also delayed the intake valve opening until the trapped exhaust gas had re-expanded. The maximum lift was reduced to come up with acceptable jerk, accelerations, and valve closing velocities. Opening the intake valve earlier to prevent the compression of the exhaust gas and increase the volumetric efficiency would have required notching the piston with valve pockets. Figure 25 compares the base cam's valve events to the 10% EGR cam's valve events.

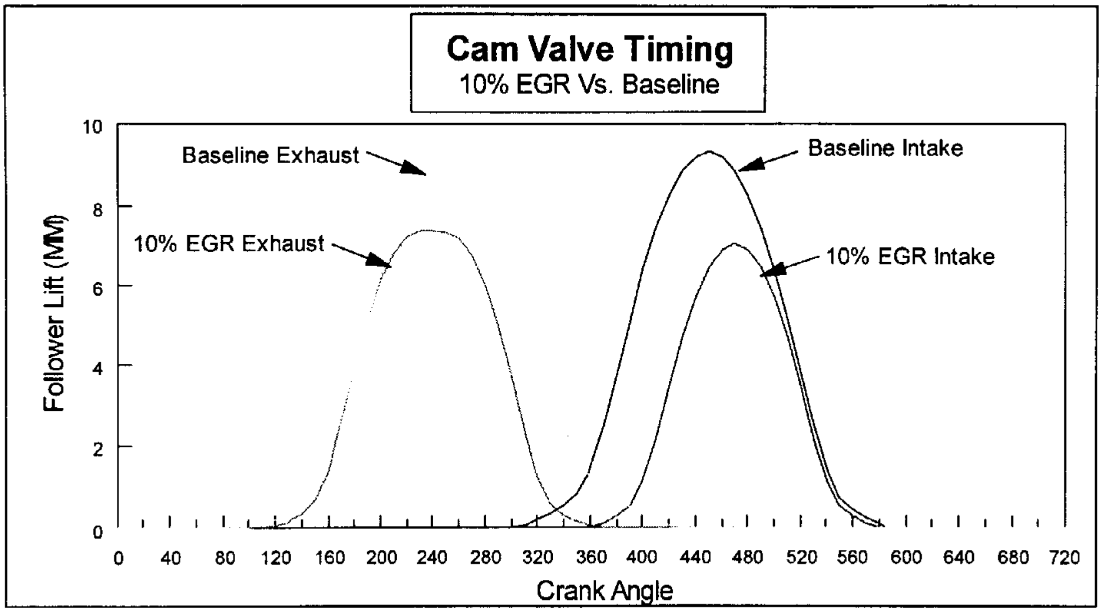


Figure 25 - Comparison of Camshaft Valve Events

Subtask 2.2 - Demonstrate < 2.5 g/hp-hr on the Modified DING Engine

Objective: Demonstrate that the DING engine can meet the 2.5 g/hp-hr NOx goal over an applicable heavy-duty vehicle engine cycle.

Accomplishment Summary:

The DING engine successfully demonstrated the 2.5 g/hp-hr NOx goal over an 8 mode steady state test cycle. The 8 mode NOx emission was reduced from 11.8 g/hp-hr to 2.5 g/hp-hp by increasing the piston compression ratio, reducing inlet air temperature, reducing gas injection pressure, and retarding injection timing. Thermal efficiency of the engine was substantially compromised to meet this NOx objective. Future work, including cooled EGR, water injection, and injection characteristic modifications, will be aimed at reducing the impact on thermal efficiency. The EGR cam was successful at reducing NOx emissions. However, the NOx - efficiency tradeoff did not favor using the camshaft due to a reduced efficiency caused by poor volumetric efficiency, increased heat rejection, and hotter intake charge that resulted from the cam change.

Accomplishment Details:

Background:

Conventional (Otto cycle) natural gas engines are limited in power and thermal efficiency relative to a diesel engine due to detonation and the need to run a nearly stoichiometric air/fuel ratio. Previously Caterpillar has demonstrated power density and thermal efficiencies equal to a diesel fueled engine with a direct inject natural gas (DING) single cylinder test engine. Previous test data had indicated a NOx advantage relative to the diesel in part due to the lower adiabatic flame temperature. Another advantage is not having the particulate / NOx tradeoff problem associated with diesel fuel. The simple hydrocarbon structure, as well as injecting the fuel as a gas, leads to a virtually smoke free operation characteristic of a spark ignited engine.

Increasingly stringent regulations on NOx emissions could make DING a "good alternative" to diesel. Natural gas should exhibit lower NOx emissions while still maintaining acceptable particulate emissions. U.S. on highway heavy duty truck NOx emissions are presently regulated at 5.0 g/hp-hr in a transient test. In 1998 this maximum will drop to 4.0 g/hp-hr and is proposed to drop to 2.0 g/hp-hr in 2004, or 2.5 g/hp-hr NOx + HC. Off-Highway emission standards are presently set based on the rated power of the engine. Engines above 750 Hp, such as the 3516, are presently unrestricted. However, by the year 2000 these engines will be regulated at 6.9 g/hp-hr using the 8 mode steady state (SS) off-highway heavy duty cycle. To make an equivalent comparison to the diesel fueled engines the 3501 DING engine was run at the same conditions as a 3516 configured for use in a 789C mining truck. The 789C mining truck engine was chosen because it was rated at a similar power to the 3516 DING engine and recent exhaust gas emission data had been taken. The 8 mode test is a steady state approximation of the conditions that off highway trucks are expected to operate at with weighting factors that indicate the amount of time the engine is expected to operate around those conditions. Table 2, shows the operating conditions, weighting factors, and power output of this specific configuration.

Mode	Speed (RPM)	Load (%)	Weighting Factor (%)	Power/Cylinder (kW)
1	700	0 (33)	15	7.2
2	1,300	50	10	39.6
3	1,300	75	10	59.4
4	1,300	100	10	78.8
5	1,750	10 (25)	10	9.2
6	1,750	50	15	44.8
7	1,750	75	15	66.5
8	1,750	100	15	89.5

Table 2 - 8 Mode SS emission points, (x) are increased loads to accommodate frictional differences.

Due to frictional differences between a single and multi cylinder engine it is not possible to operate the single cylinder test engine at an equivalent idle or 1750 RPM 10% load conditions. That is, the frictional difference between the single and multi is greater than the set load. To correct the test for this the first and fifth modes were modified by increasing their load percentages to points that could be compared between the single and multi cylinder engines.

Nitric oxide (NO) and nitrogen dioxide (NO2) are usually grouped together as NOx emissions. NO is typically the predominate oxide that is produced. Sources of NO are principally the oxidation of atmospheric nitrogen as well as nitrogen that is mixed with the natural gas that is supplied, typically 2 to 3.5% by mole fraction. Principal reactions governing the formation of NO from N2 have large activation energies which dictate a strong temperature dependence on the rate of formation.

Test Results:

Catalyst-equipped glow plugs were tested as part of the baseline data. No significant reduction in ignition delay was achieved, but a definite reduction in the power required to operate the glow plugs was observed. The overall engine performance was the same regardless if a catalyst was used on the glow plug or not. The DING engine's performance compared closely to that of the 793C diesel engine. Under most operating conditions the DING engine was able to achieve greater thermal efficiencies while staying below the 2200 PSI peak cylinder pressure (PCP) limit. See Table 3 for the 3516 diesel data and table 4 for the DING baseline comparison. Although the DING engine achieved greater thermal efficiencies, the brake specific nitric oxides (BSNOx) were also higher. The diesel's cycle-averaged-8-mode-BSNOx emissions were 9.5 g/hp-hr compared to 11.8 for the DING engine. The DING engine was tested at the same thermal efficiencies as the diesel and BSNOx were found to be very similar, slightly lower in some modes and slightly higher in others (+,- 5%).

Mode	Inlet Manifold temp (C)	Injection Timing (BTDC)	Injection Pressure (PSIG)	Thermal Efficiency (%)	BSNOx (g/hp-hr)
1	38	-	-	34.4	9.9
2	60	-	-	39.1	9.2
3	68	-	-	40.6	8.9
4	74	-	-	41.1	7.8
5	57	-	-	31.1	9.2
6	66	-	-	37.5	10.3
7	70	-	-	39.7	10.8
8	75	-	-	40.3	9.4

Table 3 - 793C Diesel comparative performance

Mode	Inlet Manifold temp (C)	Injection Timing (BTDC)	Injection Pressure (PSIG)	Thermal Efficiency (%)	BSNOx (g/hp-hr)
1	43	13	1,200	31.4	5.9
2	60	24	2,650	43	12
3	67	17	2,800	41.4	12.9
4	71	20	3,000	42.3	13.4
5	59 (-)	20	1,800	25.7	3.4
6	64 (-)	30	2,000	37.1	13.9
7	71	29	2,700	41	11.9
8	70	27	3,000	40.6	11.1

Table 4 - DING baseline BSNOx at greatest tested thermal efficiencies
(-) = Air flow reduced from diesel baseline

To achieve greater efficiencies at the light-load high-speed points (modes 5 & 6) the air flow was reduced. This reduces the volume of air and fuel that are out of the flammability range of methane. This was confirmed by a 50% reduction in the unburned hydrocarbon (HC) emissions. The mode 5 HC emissions were reduced from 14.6 g/min to 8.2 g/min (15.6 g/hp-hr) when the air flow was reduced. However, 8.2 g/min is approximately 8% of the total fuel rate at this load.

Further reduction in unburned hydrocarbons are necessary to increase the light load efficiency. The unburned hydrocarbon problem is typical for alternative fueled engines that have flammability ranges that don't allow for extremely lean burning. To comply with the proposed 2004 regulations of NOx + HC = 2.5 a solution to the hydrocarbon problem will have to be found. The hydrocarbon emissions could possibly be corrected with an oxidizing exhaust catalyst, but this would not help the light load thermal efficiency of the engine. Increasing the compression ratio should also help this problem. In some applications this light load performance would be critical to maintain. This is the DING engine's largest performance problem.

To determine the lowest BSNOx emissions, injection timing and pressure were reduced until the exhaust port temperatures were near the maximum operating condition of 750C. In some cases the exhaust port temperature was not near the 750C maximum but combustion was no longer stable, or the glow plug was requiring more power than it could realistically be expected to sustain. Because these reasons, the injection timing and pressure were not reduced further. Table 5 shows the effects that retarding injector timing and reducing injection pressure have on BSNOx and thermal efficiency. No reduced NOx data is available for this exact diesel arrangement, but the NOx / Thermal efficiency tradeoff of the DING is typical of a diesel engine. With only these changes, the 8 mode BSNOx were reduced from 11.8 g/hp-hr to 4.0 g/hp-hr.

Mode	Inlet Manifold temp (C)	Injection Timing (BTDC)	Injection Pressure (PSIG)	Thermal Efficiency (%)	BSNOx (g/hp-hr)
1	43	13	1,200	31.4	5.9
2	62	14	1,600	35.5	3.7
3	70	14	2,200	35.3	4.5
4	74	14	2,800	37.6	5.2
5	59	20	1,800	22.1	2.9
6	69	19	1,800	27.2	2.2
7	71	18	2,700	34.9	4.2
8	76	20	2,900	34.2	4.1

Table 5 - DING baseline lowest tested BSNOx emissions and corresponding efficiencies

The inlet air temperatures of this application are quite high and detrimental to low NOx emissions. Table 6 shows the results of running at lower intake manifold temperatures that would occur if an air to air aftercooler (ATAAC) were used. NOx emissions were further reduced without adversely affecting BSFC. The best tested 8 mode cycle average BSNOx emissions were 3.6 g/hp-hr @ 34.4% thermal efficiency.

		Lowest BSNOx tested		Highest Thermal efficiency tested	
Mode	Intake Manifold temp (C)	Thermal Efficiency (%)	BSNOx (g/hp-hr)	Thermal Efficiency (%)	BSNOx (g/hp-hr)
1	43	31.4	5.9	31.4	5.9
2	41	34.5	3.2	40.4	10.5
3	44	36.7	4.5	42.1	11.8
4	42	37.8	4.3	43.2	12.4
5	59	22.1	2.9	25.7	3.4
6	42	31.9	3		
7	43	33.3	3	39.5	10.9
8	55	35	3.8	40	8.6

Table 6 - DING NOx - thermal efficiency tradeoff with reduced air temperatures

The 10% EGR cam was installed and some differences were noticeable from the start. The power required by the glow plug was substantially reduced. The engine would idle smoothly with the injector timing approximately 10 degrees less advanced. All of these benefits came as a result of having the higher initial charge temperatures associated with trapping exhaust gas in the cylinder. These higher initial charge temperatures allowed the engine to operate with less advanced injection timings than previously without exceeding the maximum glow plug power. Once again data was taken to find the greatest thermal efficiencies and lowest BSNOx following the same PCP and exhaust temperature limitations as previously tested. Table 7 shows the results of the 8 mode test and injector settings used to obtain the greatest thermal efficiencies. As would be expected, the thermal efficiency was not as high as previously due to the increased initial charge temperatures, reduced volumetric efficiency, and slower heat release that results from the reduced availability of oxygen. Also, due to the reduction in volumetric efficiency, more boost had to be simulated to maintain the baseline air flow. As a result, injection timing could not be advanced as far due to peak cylinder pressure limitations. The 8 mode-cycle-averaged BSNOx emissions were 5.3 g/hp-hr @ 36.9% efficiency. To reduce BSNOx further injection timing and pressure were reduced. Table 8 shows the results of this testing. These injection timings and pressures resulted in an 8 mode BSNOx of 2.8 g/hp-hr @ 31.9% efficiency. Exhaust port temperatures at most modes, including 7 & 8, were not at their maximum temperature. If injection timing was reduced at modes 7 & 8, 2.5 g/hp-hr could have been achieved. However, it would have further reduced thermal efficiency and combustion stability even more.

Mode	Inlet Manifold temp (C)	Injection Timing (BTDC)	Injection Pressure (PSIG)	Thermal Efficiency (%)	BSNOx (g/hp-hr)
1	34.1	7	1,650	17	3.7
2	31 (+)	22	2,200	38.6	3.9
3	34 (+)	20	2,800	41.5	7.9
4	32 (+)	18	2,900	40.6	5.1
5	33	27	1,900	25.9	5
6	27	26	2,000	34.4	5.6
7	32	20	2,800	37.4	5.4
8	32	24	2,800	37.8	4.6

Table 7 - DING BSNOx at greatest tested thermal efficiencies with EGR cam
(+) = Increased air flow

Mode	Inlet Manifold temp (C)	Injection Timing (BTDC)	Injection Pressure (PSIG)	Thermal Efficiency (%)	BSNOx (g/hp-hr)
1	34.1	7	1,650	17	3.7
2	31	13	2,200	34.6	2.5
3	32	14.5	2,200	33.7	1.8
4	36	14	2,800	35.5	3.3
5	31	24	1,900	23.4	3
6	33	20	2,000	27.4	1.9
7	33	20	2,800	32.6	3.1
8	27	21	2,850	33.1	3.3

Table 8 - DING thermal efficiency at lowest BSNOx tested with EGR cam

The intake manifold air temperatures had to be reduced to permit running with the 10% EGR cam due to problems with knock and end gas autoignition. Figure 26 is a photo of a cylinder pressure trace that occurred at 1300 RPM 75% load. The engine was running at this point without the glow plug turned on. That means, the autoignition temperature was being reached due to the hot exhaust gas being mixed in with the warm intake charge. The ignition delay was still long, approximately 10 degrees; however, this ignition temperature was reached throughout the entire cylinder so all the fuel that had been injected and mixed was ignited and burned at a rapid rate. Even though there is a long delay when the glow plug initiates combustion, the heat release rate is limited by the propagation of the flame. Therefore, the cylinder pressure rise rate is not as great as that seen when the glow plug did not initiate combustion. A engine would not last operating under those rapid pressure rise conditions. Similarly, to help prevent preignition, the 1300 RPM points were run at increased air flow rates for the advanced injection timing testing.

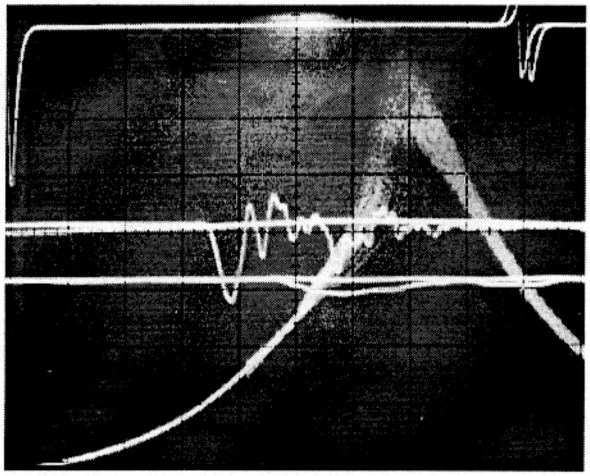


Figure 26 - Photo of knock resulting from high initial charge temperatures

Figure 27 below, shows the NOx - thermal efficiency trade off curve for the baseline cam and the EGR cam at peak torque. The EGR cam proved to be more efficient for specific NOx emissions between 3.0 and 5.0 g/hp-hr, when the airflow was increased from the baseline condition. However, when the baseline air flow was tested BSNOx were actually higher at all points. This is believed to be due to the higher initial and final charge temperatures that result from mixing the exhaust gas with the intake charge. Increasing the air flow cools the exhaust temperature that is mixed with the intake air as well as reducing the average cylinder temperature. The cooler exhaust temperature also allows injection timings to be retarded further without running above the exhaust port temperature restrictions.

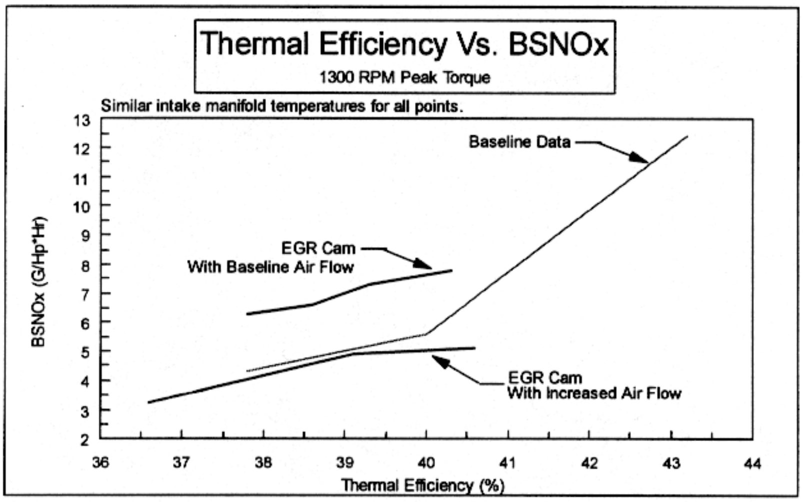


Figure 27 - NOx Vs. Thermal Efficiency Trade Off

Figure 28 is another curve showing the NOx - thermal efficiency tradeoff at rated power. The figure has a point where the two curves crossover and it appears the base cam would have been more efficient for the lowest NOx levels. This crossover was seen at lighter loads more clearly. This crossover is believed to occur as a result of the higher exhaust temperatures that occur when the injection timing is retarded. This hotter exhaust gas mixes with the cool intake charge and warms it to a higher initial temperature. This increasing temperature reduces the effect that retarding injection timing has on reducing the peak cylinder temperatures. This causes the slope of the tradeoff curve to drop off at a higher rate than the baseline test for retarded, less efficient operating conditions. This is a problem with uncooled EGR when attempting to further reduce NOx emissions by retarding injection timing or pressure. Another reason the NOx - thermal efficiency trade off curve seems to indicate that the baseline setup was better is because the EGR cam further reduced the efficiency of the engine due to the poor volumetric efficiency. While harder to quantify, another reason for the reduced efficiency derives from a higher heat rejection that may have occurred as a result of the higher intake charge velocities. The primary reason the baseline rated testing did not achieve BSNOx levels as low as the EGR cam was due to the exhaust port temperature limitation that occurred earlier due to the hotter intake manifold temperatures that were used for most rated conditions. The baseline data taken with intake manifold temperatures similar to the EGR cam data suggests that the efficiency would have been greater for a given BSNOx level at all rated conditions.

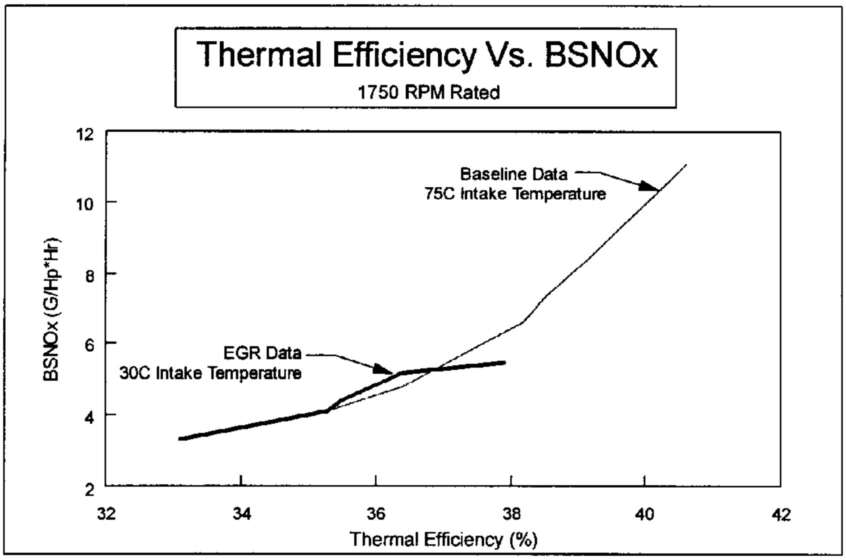


Figure 28 - Rated power NOx - thermal efficiency trade off

Comparing equivalent air flow rates and intake temperatures at lighter loads, the EGR cam did reduce BSNOx emissions for a given efficiency. However, as stated previously if timing was retarded to a point where the exhaust temperatures rose significantly, the baseline data became more efficient for a given NOx output. For these reasons the EGR cam was removed and the baseline cam was reinstalled along with a 15.2:1 compression ratio piston. To help overcome the limitations the glow plug placed on the baseline test, catalyst coated exhaust valves and a catalyst coated glow plug shield are being procured for the next phase of the contract. Previous experiences with catalysts indicate this will reduce the demand on the glow plug.

Testing of the higher compression ratio piston has just begun but a preliminary 8 mode test resulted in BSNOx equal to 2.46 g/hp-hr @ 28.5% efficiency. Table 9 shows the points that lead to this 8 mode average. To achieve this 8 mode average the thermal efficiency was substantially reduced. Increasing the efficiency while still maintaining BSNOx emissions of 2.5 g/hp-hr could be accomplished with injector nozzle changes and injection characteristic changes. This testing is planned for the next phase of the contract. Cooled EGR would not have the knock limitations that the EGR cam imposed. Cooled EGR should allow the injection timing to be advanced while maintaining the same BSNOx emissions. This is expected to result in a greater thermal efficiency while still maintaining the 2.5 g/hp-hr target. The cooled EGR should also help to achieve BSNOx emissions less than 2.5 g/hp-hr. For these reasons cooled EGR will be tested for the next phase of the contract.

Mode	Inlet Manifold temp (C)	Injection Timing (BTDC)	Injection Pressure (PSIG)	Thermal Efficiency (%)	BSNOx (g/hp-hr)
1	35	7	1,650	23	3
2	45	12	1,700	27.4	1.7
3	45	14	2,300	30.8	2.1
4	41	15	2,900	33.5	2.9
5	45	20	1,800	23	2.9
6	45	20	2,000	22.1	1.5
7	40.1	20	2,700	28.6	2
8	43	23	2,900	31.5	3.3

Table 9 - Best NOx tested with the 15.2:1 compression ratio piston

To show that NOx emissions of the DING engine follow a consistent and understandable pattern a model was correlated using the cycle averaged flame temperature. The model integrates NOx formation as a function of the cylinder temperature that is based on several initial conditions and the heat release rate of the fuel. A heat release model for natural gas is being developed as part of the low NOx and durability work. The NOx model is part of an engine simulation program developed by Caterpillar. The simulation allows NOx to be estimated using Arrhenious' equation below:

$$NOx = A*\exp(E/Tf) \quad gNOx/kgf \qquad \text{Arrhenious Equation}$$

Tf is the adiabatic flame temperature, and A and E are empirical "constants". This equation is integrated with respect to fuel mass since the flame temperature varies throughout the engine cycle. It was found that the 3501's measured NOx could be correlated in terms of an average Tf defined by applying the above equation to the total NOx estimated by engine simulation. Figure 29, a graph of log(NOx) vs.. 1/Tf average was expected to yield a straight line, and it very nearly does for the rated and peak torque conditions where the model was tested. Corresponding values of A and E can be determined from this graph and used in the engine simulation program to predict NOx for the 3500 engine with or without EGR. Scaling by time results in the curve on the graph designated Rated RPM Corrected. The realization that time plays a role in NOx formation gives this single approximate curve. The Tf average determine as described above is not exactly independent of E, but the variation has mathematically been shown to be very small if E does not change significantly.

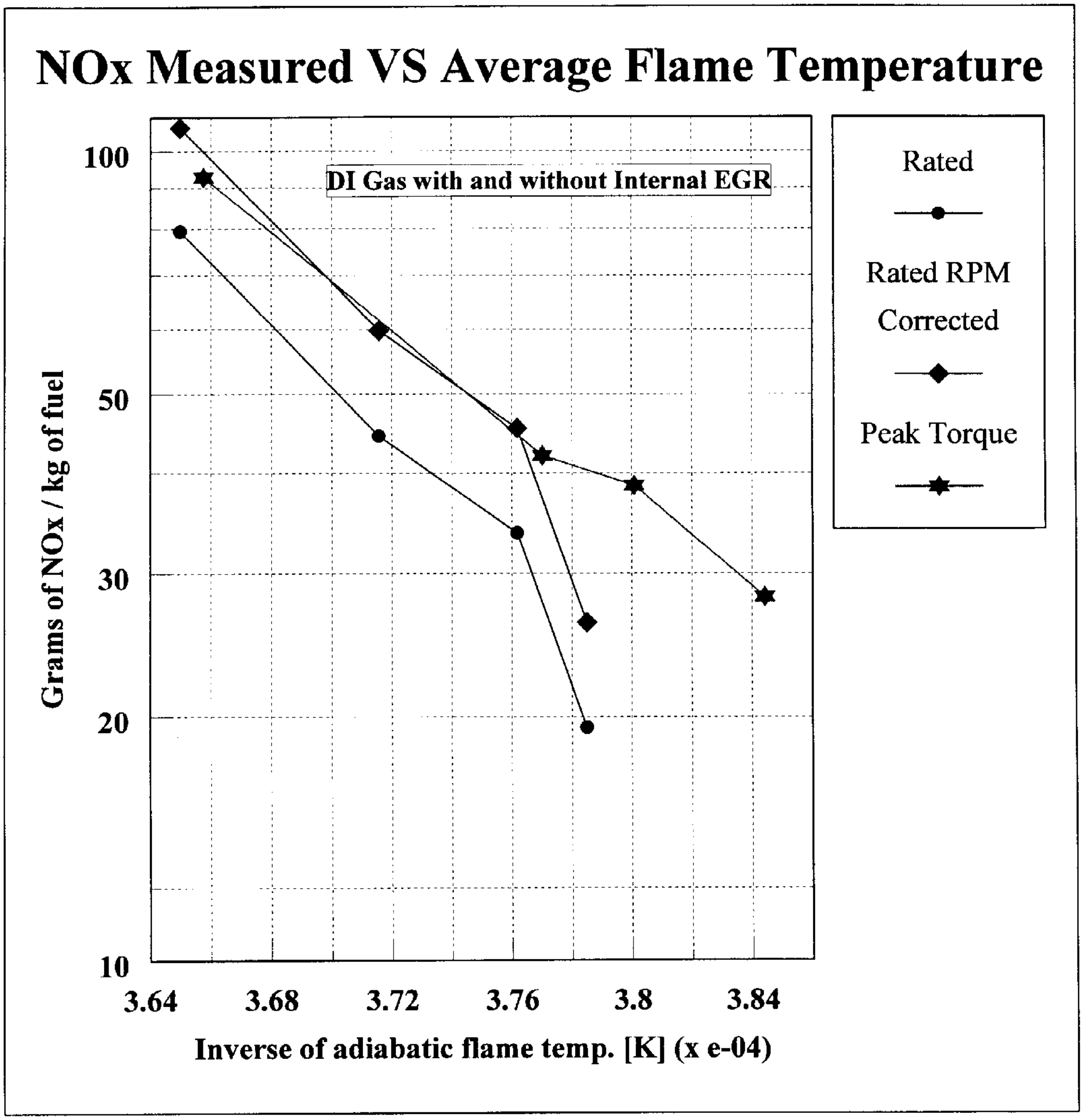


FIGURE 29 - 1/(cycle averaged flame temperature) vs.. BSNOx/lbm air

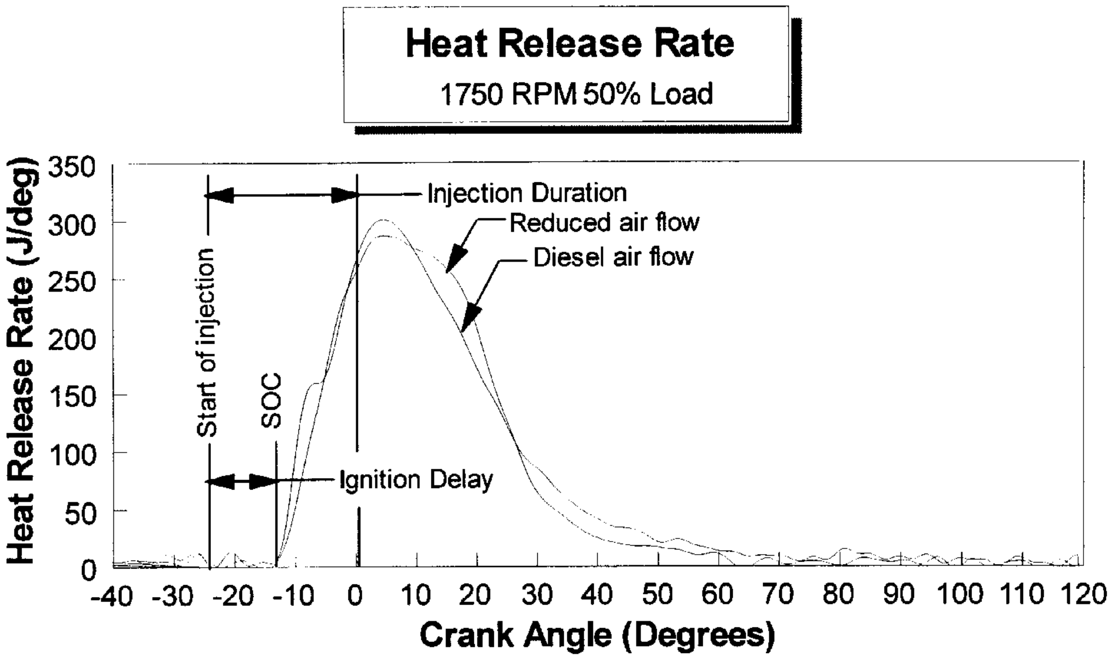


Figure 30 - Heat release rate for decreased air flow

Figure 31 shows how the EGR affected the heat release rates at peak torque . Comparing the two retarded timings it is apparent the heat release rate was always higher with the EGR cam. The explanation for this comes from the hotter initial charge temperature and the slightly earlier SOC that resulted. Advancing the timing reduced the initial charge temperature and the BSNOx were now lower than that of the baseline data. However, the baseline data had an earlier start of combustion as a result of a slightly earlier injection. Both EGR heat release rates have a slight spike at the end. This can be indicative of end gas autoignition. This was not evident in the pressure trace but had been seen previously. One final aspect to realize is although the EGR cam was producing less BSNOx for approximately the same conditions the thermal efficiency of the EGR setup was 3% lower. If the EGR cam BSNOx were compared to a baseline point of equivalent thermal efficiency the BSNOx of the baseline would have been lower.

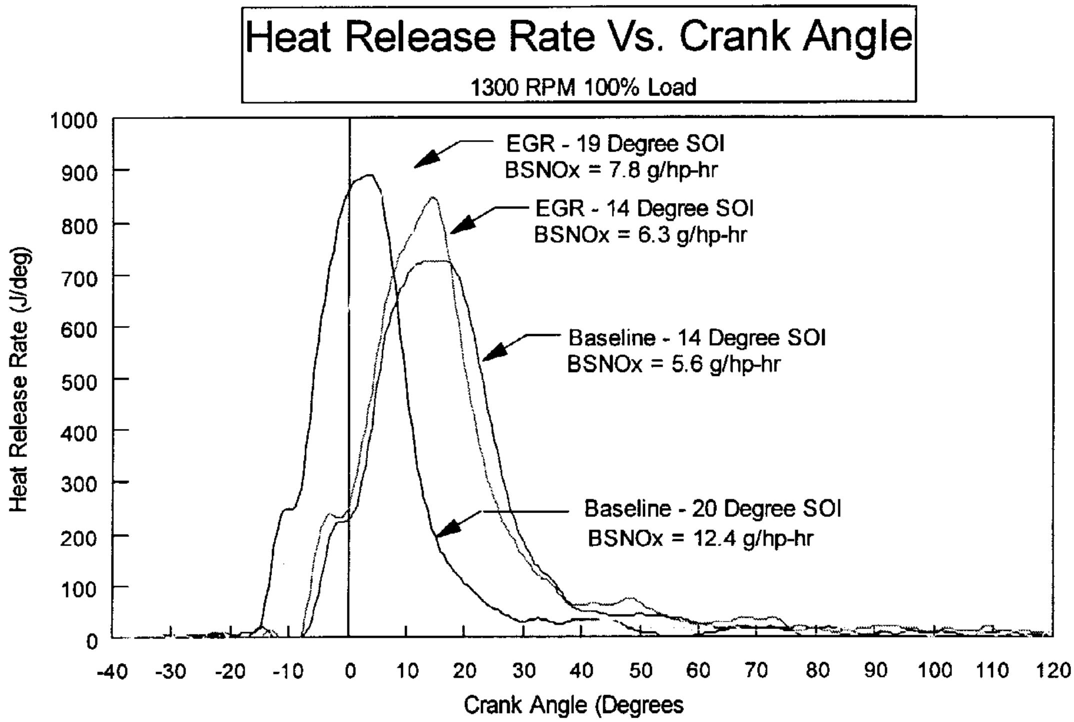


Figure 31: Heat release comparison between baseline and EGR cam

Future Work:

As stated in previous sections catalyst coated exhaust valves and glow plug shields are being procured and scheduled for testing in early February. Several injector tips will be procured and effects on NOx, HC, and efficiency will be tested. It is expected that the thermal efficiency at light loads and reduced NOx points can be improved. Since the hot EGR combined with the reduced efficiency that the cam caused proved not to substantially help the NOx - thermal efficiency tradeoff cooled EGR will be tested. The exact EGR configuration that would be used on a multicylinder engine has not yet been decided, but testing on the single should help further define this. Caterpillar has shown the emission benefits of injecting water mixed with the fuel into the cylinder with the A-21 fueled 3176. Water cannot be mixed with the natural gas and injected into the cylinder, but water can be injected into the intake port and mixed with the intake air. Procurement and testing of port water injection is planned for the next phase of the contract. Work involving selective catalytic reductioning of NO exhaust emissions is also expected to begin as part of the next phase of the contract.

Conclusion:

Baseline DING data was taken for an 8 mode steady state emission test and compared to a 3516 diesel engine. The diesel's 8-mode-cycle-averaged-BSNOx emissions were 9.46 g/hp-hr @ 39.2% thermal efficiency. Operating at the same conditions as the diesel, the DING engines 8 mode BSNOx were 11.82 @ 39.4% thermal efficiency. Looking at this alone the DING engine BSNOx look substantially higher for a similar thermal efficiency. This is only the case because of the DING engines lower efficiency at light loads which are factored into this thermal efficiency. This lower efficiency is caused by incomplete combustion at light loads and is talked about in greater detail in this report. Reducing injection pressure and retarding injection timing lead to a cycle averaged BSNOx of 4.02 g/hp-hr @ 33.1% thermal efficiency. These emissions were made with intake manifold air temperatures as high as 75C at rated power. Reducing intake manifold air temperatures helped further reduce these emissions without adversely affecting thermal efficiency. The best tested 8 mode BSNOx emissions were 3.6 g/hp-hr @ 34.4% Thermal efficiency. The BSNOx could have been reduced further at the expense of efficiency and glow plug durability. To further reduce BSNOx and minimize the effect on thermal efficiency a 10% EGR cam was procured and tested. Only reduced intake manifold air temperatures were tested due to a problem with knock and end gas auto-ignition. With the EGR cam installed in the engine the 8 mode BSNOx at the engines greatest efficiency were 5.32 g/hp-hr @ 36.9% thermal efficiency. The lowest tested 8 mode BSNOx were 2.81 g/hp-hr @ 31.9% thermal efficiency. Due to the problem with knock and reduced efficiency the EGR cam was removed and the baseline cam reinstalled. The 13.0:1 compression ratio piston was removed and replaced with a 15.2:1 piston. Testing with this piston has just begun, but preliminary data indicates an 8 mode BSNOx of 2.46 @ 28.5% thermal efficiency. This piston has shown that the DING engine is capable of NOx emissions as low as 2.5g/hp-hr.

To reduce the impact on thermal efficiency and recognize the next deliverable of BSNOx equal to 1 g/hp-hr; cooled EGR, water injection, various injector nozzle changes, split injection, exhaust gas after treatment with catalysts, and ignition catalyst tests are all planned for the next phase of the contract.

Task 3: Durability Development of the 3000 psi Fuel Handling System

Subtask 3.1 - Review Current Level of On-Board 3000 psi Fuel Handing Systems

Objective: Evaluate the current level of 3000 psi fuel delivery technology for both Compressed Natural Gas (CNG) and Liquefied Natural Gas (LNG). Investigate the technology's vehicle applications and economics. Identify components that will require additional development for the 3000 psi fuel delivery system to be commercially viable.

Accomplishment Summary:

The current level of 3000 psi fuel handling technology was evaluated. It was determined that for direct injected gas engines to be practical in the transportation sector, a liquid natural gas fuel delivery system will generally be required. Cryogenic pump technology requires additional development for mobile applications. Diesel thermal efficiency and lower incremental costs are also required to make the DING engine economically practical.

Accomplishment Details:

Introduction:

Concerns about the environmental effects of fossil fuel use and the dependence of the United States on foreign oil are providing the incentive for the increased use of alternative fueled vehicles. According to the Energy Information Administration (EIA) [1], the number of alternative fueled vehicles in the United States has increased from 250,000 units in 1992 to

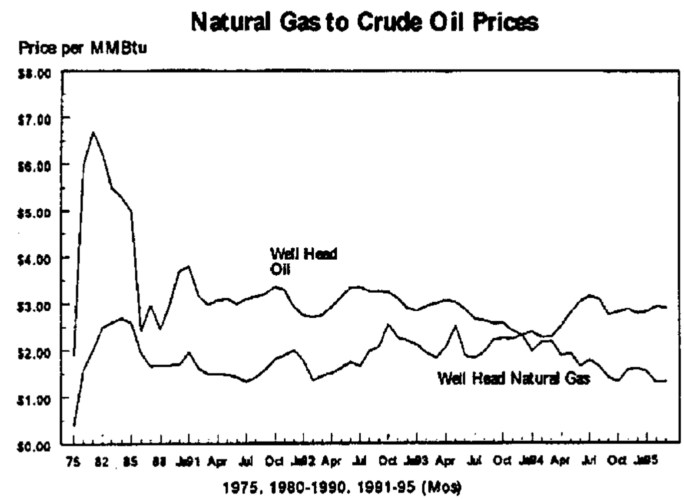


Figure 32 (Source: DOE, Oil & Gas Journal)

320,000 units in 1994 to an estimated current level of 420,000 units. Of these units, liquefied petroleum gas (LPG) is by far the most popular alternative fuel, at approximately 70% of the total population, but natural gas is emerging as a popular fuel. Over this same period, the number of natural gas vehicles has grown by over 300% to attain 20% of the market. A plentiful supply of domestic natural gas has resulted in lower fuel costs, as shown on Figure 32, which is increasing interest in gas fuel vehicles.

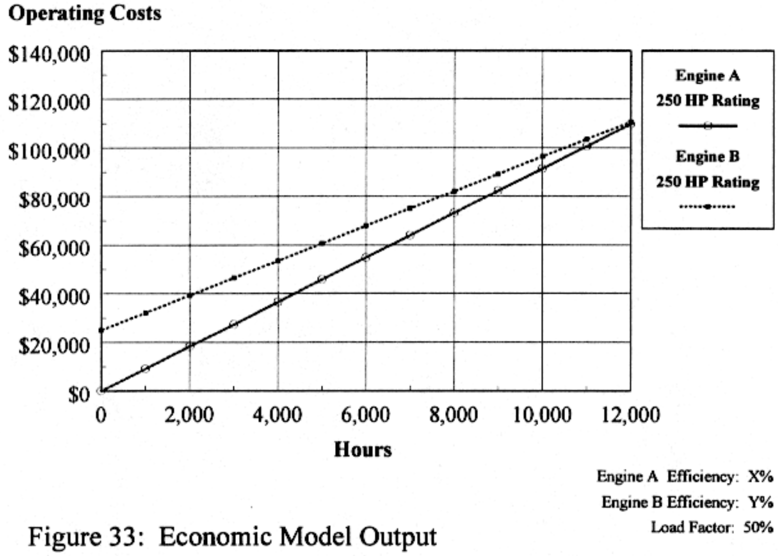


Figure 33: Economic Model Output

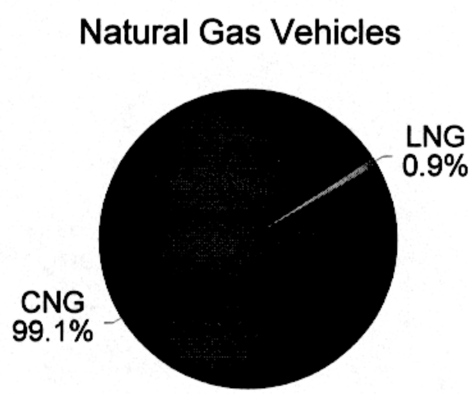


Figure 34: Natural Gas Vehicle Distribution

For natural gas vehicles to be viable to most customers, they must show an economic benefit, especially for large fuel consumers. Showing an emissions benefit alone won't satisfy the customers needs. Low natural gas prices certainly provide an incentive, but many other factors affect the economic viability such as engine efficiency and power rating, engine costs, load factor and fuel system costs. To help demonstrate the economics of a natural gas fueled engine system, a cost spread sheet model has been developed to compare the natural gas operating costs verses a diesel baseline. A generic example of the spread sheet is shown on Figure 33.

Inputs into the model include engine power, efficiency, load factor, fuel costs, and incremental costs for operating on natural gas as compared to a diesel baseline. Taxation of the fuel is included in the fuel costs. The important factor is the relative difference in costs between diesel fuel and natural gas.

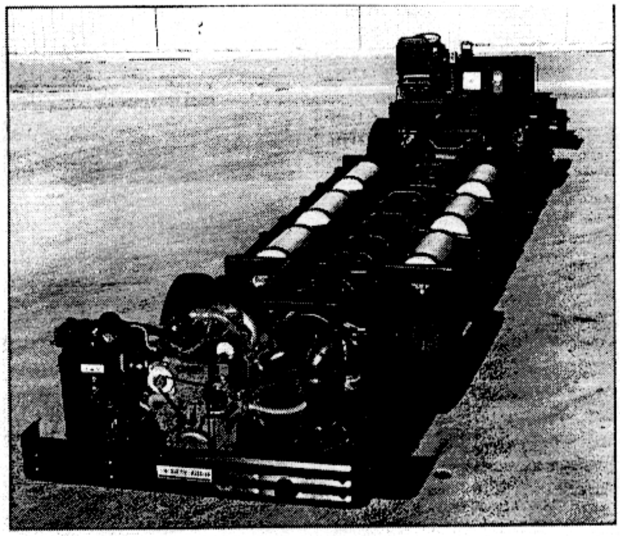


Figure 35: John Deere Bluebird Bus Chassis

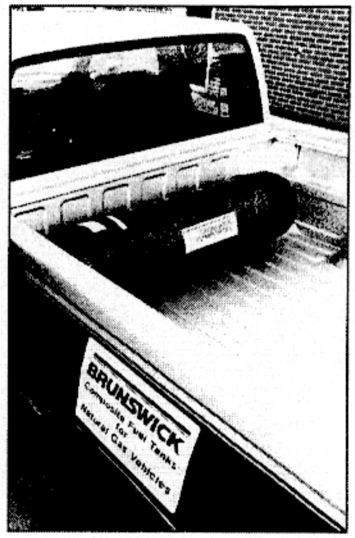


Figure 36: Light-Duty Truck

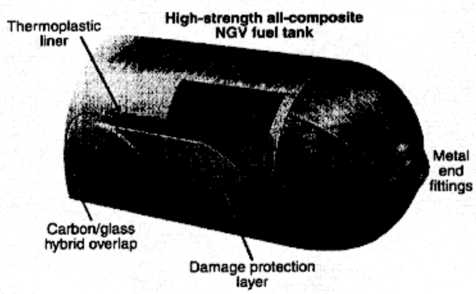


Figure 37: Composite CNG Tank

Status:

A significant percentage of the incremental costs for operating on natural gas is the fuel handling system. Two types of natural gas fuel systems exist today: CNG and LNG. CNG systems are by far the most commonly used system today as shown on Figure 34. There exist only 716 LNG vehicles today according to GRI [2]. An example of a CNG system from the chassis of a BlueBird bus is shown on Figure 35 [3]. The fuel handling system consists of the 6 CNG tanks, initially charged to 3600 psi, which are connected in series with a pressure regulator valve that controls the gas pressure supplied to the engine. Depending on the engine design, the regulated pressure supplied to the engine ranges from 50-300 psi. Most CNG vehicles are used in light-duty applications such as the truck shown on Figure 36 [4].

Size (inches)	External Volume (Gal)	Empty Weight (lbs)	SCF @ 3600 psi	CNG Useable Storage (Gal)	Diesel Eq. Gallons
12.2 x 34.4	17	35	474	12	3.7
13.7 x 74.6	47	90	1,416	36	11.1
15.4 x 74.1	60	109	1,828	46	14.3
16.0 x 97.3	85	152	2,615	66	20.4
18.3 x 121.6	138	243	4,350	110	33.9
Assume 296.34 SCF/ft³ H_2O vol. @ 3600 psi LHV natural gas = 1000 btu/scf LHV diesel = 128,100 btu/gal					
Table 10: CNG Tank Specifications					

The CNG tank used in this application is a all-composite material design produced by Brunswick. This tank utilizes a high density polyethylene with an over wrap of carbon and glass fibers with an epoxy resin as shown on Figure 37 [5]. The primary advantages of the composite cylinders are the weight savings over steel tanks (70%) and from 30-50% over aluminum lined composite tanks. The composite walls are thinner which allows approximately 15% more volume with the same external volume. CNG tanks also have the advantage of indefinite hold time assuming they are not heated significantly above ambient temperatures. Even with the improvements the composite design CNG tanks offer in terms of efficiency in storing high pressure gas, the primary disadvantage of the CNG tank is energy density storage. Table 10 shows the specifications of commercially available CNG tanks from EDO Corporation [6]:

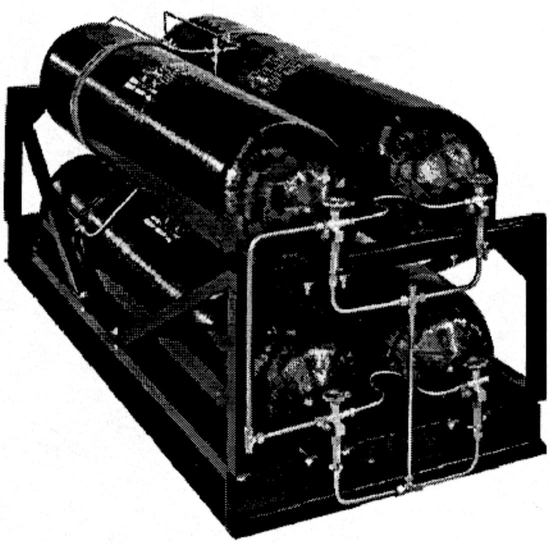


Figure 38: CNG Tank Cascade

As can be seen by the chart, the size of the CNG tanks needs to be approximately four times the size of a diesel tank to store an equivalent amount of fuel. For this reason, standard size CNG tanks are often bundled together in a cascade as shown on Figure 38 [7].

As mentioned earlier, the tanks are charged to a pressure of approximately 3600 psi during refueling at a CNG station. A schematic of a CNG station is shown on Figure 39 [8].

CNG stations draw the natural gas from underground pipelines operated by gas companies. Typically, the pipeline gas contains water which can cause condensation problems in the refueling station and/or the vehicle fuel system. To remove the excess water, the natural gas passes through a dryer before compression. The compressor uses multiple stages to compress the pipeline gas to tank filling pressures. To minimize starts and stops of the compressor, large high pressure gas buffer (cascade) systems are installed with sequencing and priority valving to direct the gas to the vehicle.

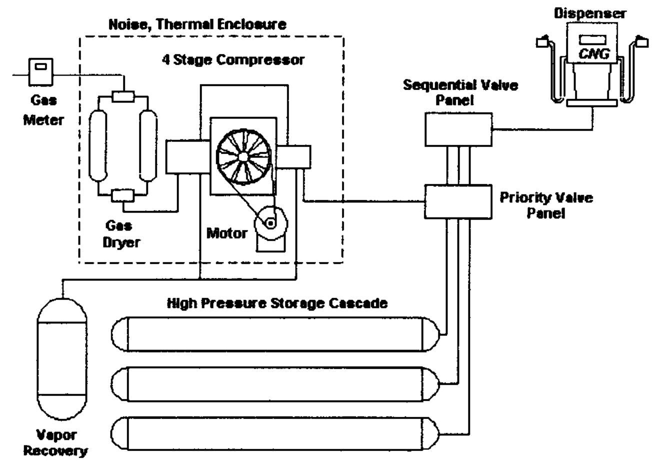


Figure 39: CNG Station (Courtesy CH-IV Corp.)

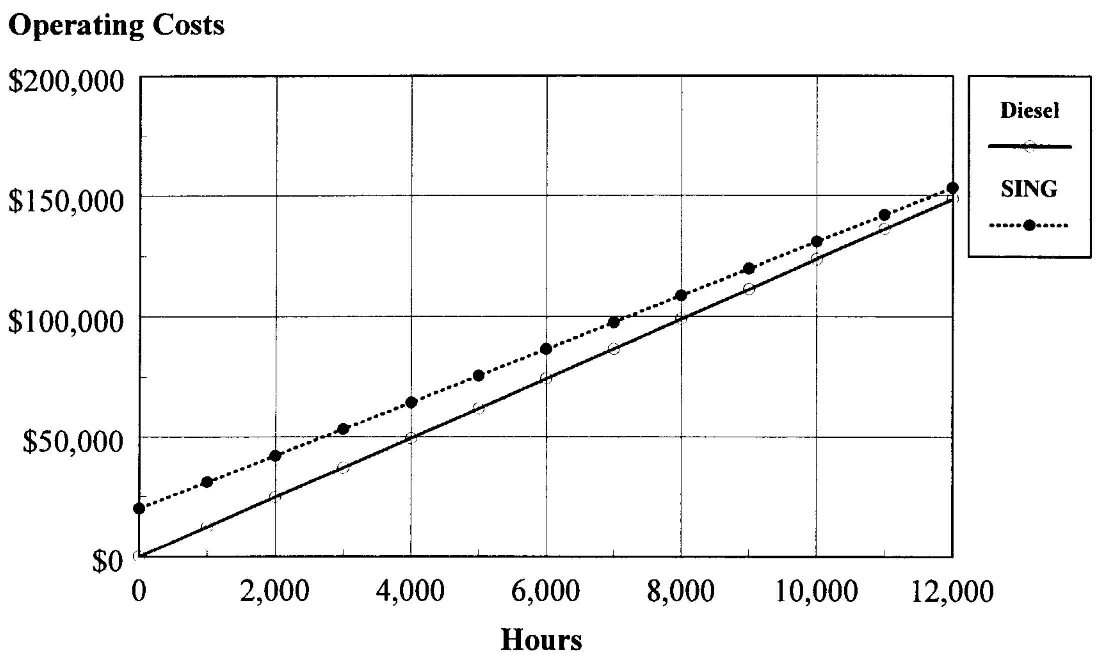
These compressor stations are quite expensive with installed capital costs estimated to be $350,000 for a 300 scfm system which would be adequate only for light duty vehicles. A vehicle with a CNG tank that holds 11 diesel equivalent gallons (deg) would require about 5 minutes to fully fill the tanks. According to CH-IV Corporation, the operating and maintenance costs associated with a typical CNG installation range from $0.15 to $0.35 a gallon of gasoline equivalent (gge) or $0.16 to $.38 per deg. Also, heat introduced by the compression of gas makes it virtually impossible to fast-fill CNG tanks to their rated capacity. This further reduces the useable range of a cng fueled vehicle.

As mentioned before, for a natural gas fueled vehicle to be practical, it must show an economic benefit. The use of a model helps to show the economic benefits of using natural gas. Inputs into the model include fuel costs, vehicle conversion costs (to operate on natural gas), engine efficiency, and load factor. Table 11 shows the fuel costs per deg to be used for the model.

Fuel	Heating Value	Base Cost (Average)	Federal Excise Tax	Retail Price (Average)
Diesel	128,100 btu/gal	$0.70/gal	$0.244/gal	$1.30/gal[9]
LNG	77,500 btu/gal	$0.50/gal[10]	$0.31/deg	$1.14/deg
CNG	1000 btu/scf	$0.30-$0.45/therm[11] (uncompressed)	$0.06/deg	$0.87/deg[12]

Table 11 Typical Fuel Costs

Figure 40 shows the output of the economic model using the fuel costs from Figure 20. The engine used in the model is a Caterpillar 3406 engine (14.6 liter) rated at 350 hp using diesel fuel. The spark ignited natural gas version (SING) matches the diesel power rating.



Diesel Efficiency: 40%
SING Efficiency: 30%
Load Factor: 55%

Figure 40: 3406 CNG @ SI Gas Efficiency

The model assumes a 55% load factor which is typical for trucks using engines in this horsepower range and an incremental cost of $20,000 [13] to convert a diesel fueled vehicle to operate on natural gas. The lower cost of natural gas reduces the engine operating costs even though some of this advantage is lost due to the lower efficiency of operating an Otto cycle (30%) as opposed to a diesel cycle (40%). Operation on natural gas at current prices saves approximately $1.28/hour on fuel costs. However, to pay for the incremental cost to convert a diesel engine to operate on natural gas results in a payback period of over 15,000 hours (7-8 years) of engine operation. To be viable to a customer, the payback period must be significantly reduced. The $20,000 vehicle price premium to operate on natural gas is typical according to Booz-Allen [13] with today's small volumes of natural gas vehicles. Even with additional volume, its difficult to expect this price to drop by more than 50%, which will only reduce the payback period to 3-4 years.

Reducing customer costs further requires developing engine technology that allows operation of natural gas at higher efficiencies such as the direct injection natural gas (DING) . A DING engine can potentially meet the diesel efficiency (40+ %) and improved driveability (better torque, response, and reduced heat rejection) as compared to the SING engine. This results in significantly reduced operating costs as shown on Figure 41 .

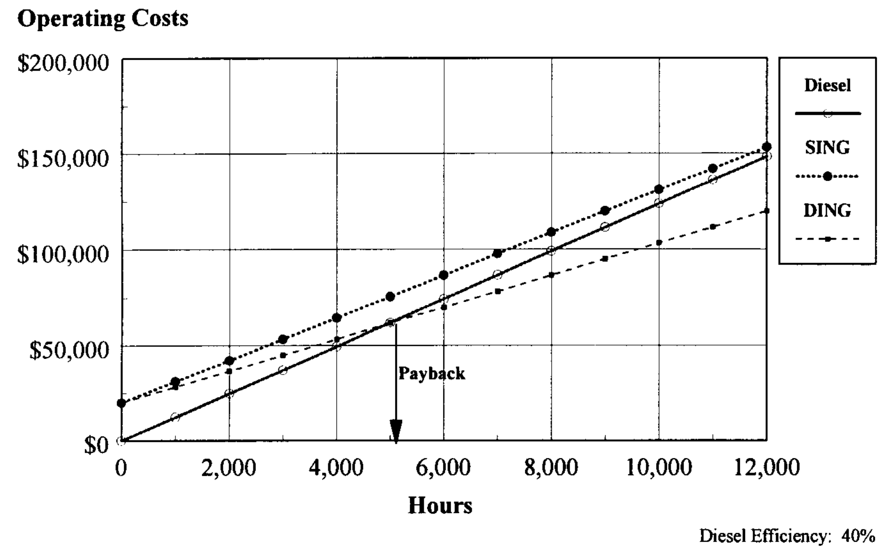


Figure 41: 3406 CNG @ DI Gas Efficiency

Operating at the diesel efficiency reduces the payback period from 15000 hours to 5000 hours at current CNG & diesel prices. But to operate a DING engine requires a high pressure fuel handling system (3000 psi). CNG tanks are initially charged to 3600 psi but would not be suitable for a DING application. As shown on Figure 42, after using only 10% of the fuel in a CNG tank, its pressure would drop below 3000 psi. To maintain pressure, an on-board compressor would be required. These types of systems are used strictly in stationary applications and are not practical for mobile applications.

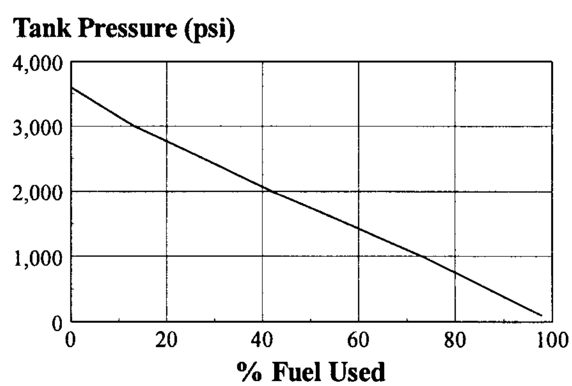


Figure 42: CNG Fuel Usage

A potential solution would be to use an LNG fuel handling system. As mentioned earlier, LNG vehicles are far less common than CNG vehicles, but LNG offers two major advantages. First, LNG has a much higher lower heating value (LHV) by volume (73,000 btu/gal) than CNG (34,400 btu/gal @ 3600 psi) and LNG can be pumped up to pressure as a liquid more efficiently than compressing natural gas. LNG is made by cooling natural gas to approximately -260 degrees F. Storage of LNG requires specially insulated cryogenic tanks as shown on Figure 43.

The LNG fuel tank is a double-walled stainless steel design with vacuum insulation. The inner tank or pressure vessel contains the fuel normally at 50-150 psi and is rated at approximately 230 psi. The outer tank provides for structural integrity and a vacuum seal. Between the tanks is the insulation material which increases hold time. The Minnesota Valley Engineering (MVE) designed tank includes a fuel pressure regulator, manual supply and vent valves, excess flow valve, primary and secondary relief valves, and an electronic fuel gage.

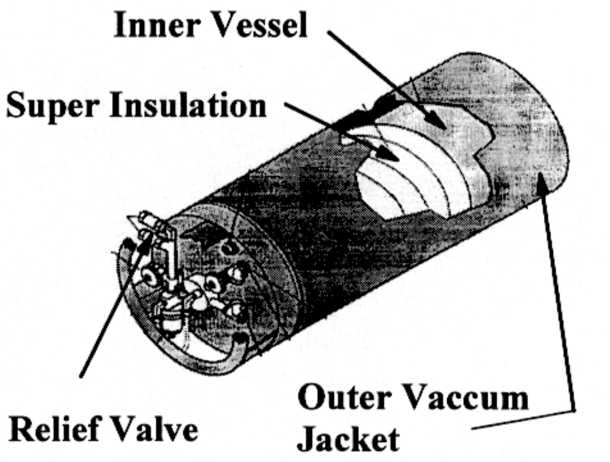


Figure 43: Cryogenic Tank [14]

An example of an installed LNG cryogenic tank is shown on Figure 44. The refuse hauling vehicle shown is a Mack MR6000. These tanks are designed to be installed in the same location as a diesel tank. Operation of this fuel system is transparent to the customer.

Figure 44 LNG Fueled Truck [15]

The typical LNG fuel handling system also contains a vaporizer to convert the LNG to CNG for delivery to the engine. The vaporizer uses engine coolant to vaporize the LNG which is then directed to the engine intake manifold for SING applications. Also, the MVE design for a SING engine has the vaporizer integral with the tank. For DING applications, the LNG would need to be pumped up to operating pressures using a cryogenic pump. This type of fuel handling system would be similar to a LCNG (using LNG to produce CNG) station produced by CH-IV Corporation as shown on Figure 45.

This station produces CNG by pumping LNG using a cryogenic pump up to the desired pressure and then vaporizing the LNG. The advantage of this type of system as compared to a CNG station is that it takes substantially less power and equipment to pump a pound of liquid than to compress a pound of gas to the same pressure. As an example, Table 12 compares a gas compressor to a cryogenic LNG pump.

As can be seen, the LNG pump is much smaller and takes much less power to operate as compared to the gas compressor. According to CH-IV, LCNG stations are much quieter, safer, and cheaper to operate. The capital costs for LCNG stations are significantly less than that of comparably sized CNG systems, ranging from 85% (for 350 SCFM) to as little as 30% for larger systems. The savings increase as the system's refueling capacity increases. Also, the operating and maintenance costs associated with a CH-IV system is about 2 to 4 cents/gge as compared to a CNG station cost of 15 to 35 cents/gge.

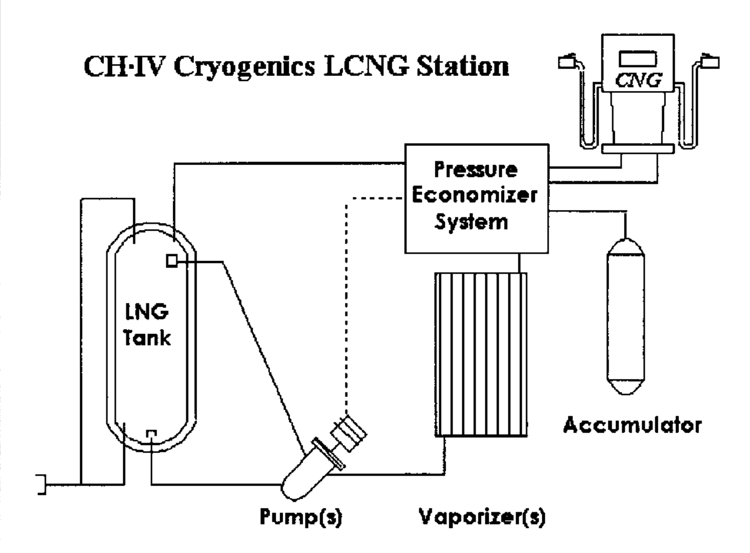


FIGURE 45

	Gas Compressor [16]	LNG Cryogenic Pump [17]
	Knox Western Eagle 2245	ACD 1-GAPD
Size	30" x 90"	12" x 40"
Fuel	CNG	LNG
Flow Rate	5 GPM	5 GPM
Inlet Pressure	50 psi	5 psi
Outlet Pressure	3000 psi	3000 psi
Power	139 HP	20 HP

Table 12 Gas Compressor/Cryogenic Pump Comparison

The advantages of an LCNG station would also apply to a mobile on-board fuel handling system. The on-board system would consist of the same components, just scaled down for the application. The heart of the system is the cryogenic pump. Cryogenic pumps, as shown on Figure 46, are typically used in high pressure cylinder filling and bulk-storage pumping applications. Their use in mobile applications is limited. Even though approximately 50% of the LNG vehicles in use today use cryogenic pumps, this amounts to only approximately 300 units [19]. With cryogenic pumps, system pressures can operate in the 150-300 psi range. Due to cost and reliability concerns, natural gas engine developers are reducing fuel pressure requirements to levels that would not require a pump. For DING applications, a high pressure (3000 psi) pump would be mandatory. The high pressure pump technology does exist but requires further development to reduce costs for mobile applications. Current pumps cost approximately $4000 for pumps sized for passenger bus applications which approaches the cost of the base engine. Improvements in cryogenic pump technology to improve reliability will be necessary for DING technology to be viable.

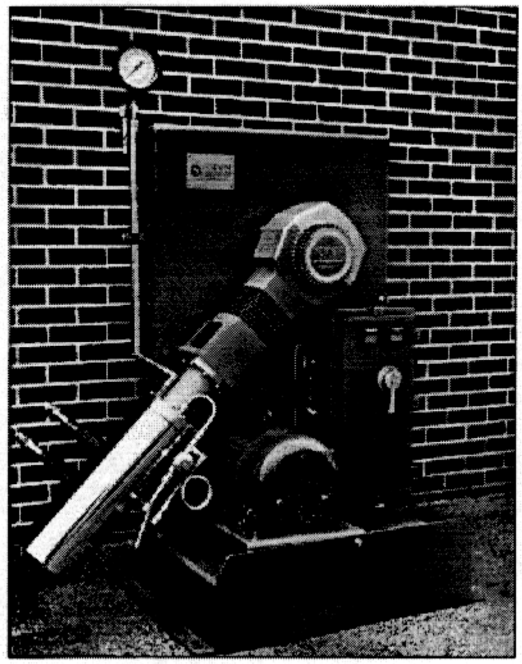


Figure 46 Cryogenic Pump [18]

Due to the low volumes and demand, the cost to convert a vehicle to operate on natural gas can be very expensive. Typical costs including the engine, fuel system, and installation to convert a truck or a passenger bus to operate on LNG are $32,000 to $34,000 per vehicle [20]. This cost will vary depending on many factors such as LNG tank size, engine horsepower, cryogenic pump necessity, and number of vehicles ordered. To analyze the benefits of using LNG to fuel a vehicle, the data on Table 13 will be used as input into the economic model. For the purposes of this analysis, an incremental cost of $33,000 will be used to establish the baseline economic analysis of using LNG in a DING application. Engine sizes were selected that would be used in a passenger bus or a Class 8 truck application with appropriate engine efficiencies and load factors. The engine efficiencies shown are typical for Caterpillar engines in these particular engine ratings. It will be assumed the DING versions of these engines can match the diesel efficiencies. The estimated $12,000/unit anticipated cost premium assumes a cost reduction due to higher volumes of LNG vehicles produced and sold. Also, to be consistent, it will be assumed that fuel taxes will be paid in all analyses including the passenger bus applications. No emission reduction credits will be assumed for this analysis. Focus will be maintained to look at the benefits from a purely economic point of view.

	Passenger Bus	Class 8 Truck
Engine Rating	250 HP	350 HP
Efficiency	38%	42%
Load Factor	42%	54%
LNG Vehicle Cost Premium	$33,000	$33,000
Future Vehicle Cost Premium	$12,000	$12,000

Table 13: Economic Model Assumptions

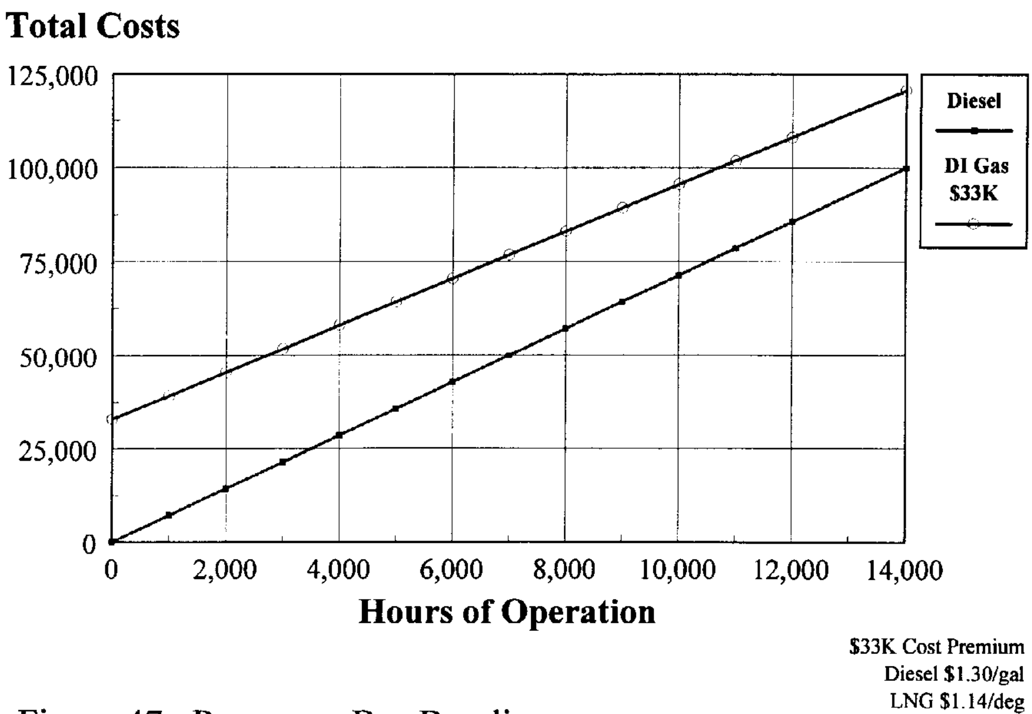


Figure 47: Passenger Bus Baseline

Figure 47 compares the baseline operating costs of a diesel engine verses an LNG fueled DING engine operating in a passenger bus or similar lighter duty application required to pay current fuel taxes. The taxed fuel costs and the premium for operating on LNG are in 1996 dollars.

Operation on LNG results in a lower operating costs per year due to the lower fuel costs but not at a rate to pay off the investment in an acceptable period of time. The lower fuel costs will save approximately $1800/year but with an initial investment of $33,000 this results in a rate of return of approximately -3% figured over a 15 year period. The initial investment would not be paid off. To be viable, lower fuel costs and/or lower incremental cost premiums are required.

As mentioned before, CNG (6 cents/deg) currently has favorable Federal excise tax status relative to LNG (31 cents/deg). Currently, the LNG industry is lobbying Congress to tax LNG at the CNG rate. If successful, this will dramatically lower the cost of LNG by 25 cents/deg. Figure 48 shows the effect of taxing LNG at the same rate of CNG. The lower taxes reduce operating costs by $4500/year which provides a 10% rate of return calculated over a 15 year period. The initial investment is repaid in approximately 2 ½ years. Ideally, the pay back period should be approximately 1 year. To accomplish this, the initial cost premium must be reduced, which will happen with increasing production volumes. As mentioned earlier, GRI estimates the cost premium to fall to approximately $12,000/vehicle. Since the DING application will need a cryogenic pump, for this analysis the cost premium will be reduced to only $14,000 (assuming cryogenic pump costs are reduced as well). This results in a payback period of approximately 1 year as shown on Figure 49. This calculates to a rate of return of 32% which is very attractive considering the relatively low fuel usage of a passenger bus as compared to a class-8 truck. Its important to note that lower LNG fuel taxes are required for the passenger bus to be viable.

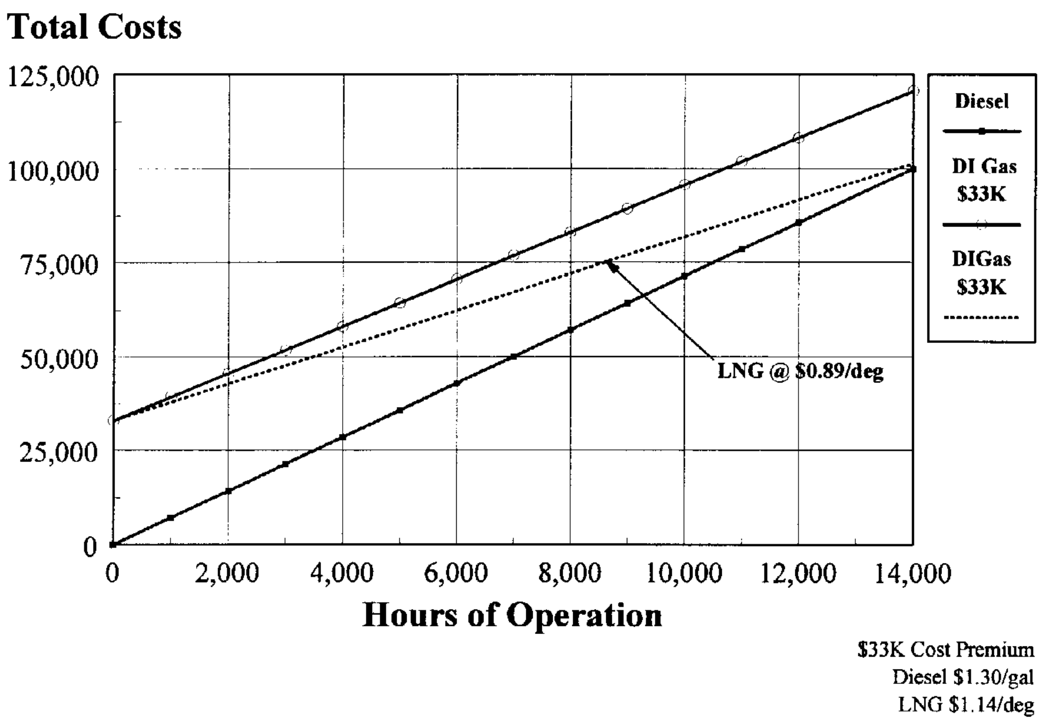


Figure 48: Passenger Bus with Lower LNG Taxes

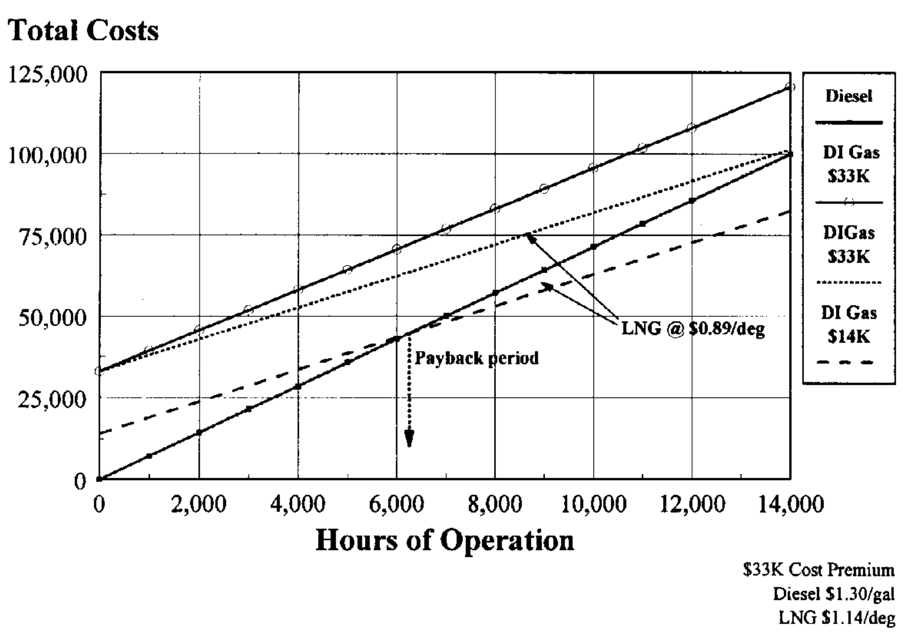


Figure 49: Passenger Bus with Lower LNG Taxes and Reduced Cost Premium

On-highway trucks, with their larger engines and load factors, have significantly higher fuel useage rates and fuel costs as compared to passenger buses. Plus, they have longer range requirements. For these reasons, they would be even better candidates for conversion to LNG. The economic model is assuming a 350 hp engine (similar to a Caterpillar 3176) operating at a load factor consistent with this application.. Experience at Caterpillar shows these engines operate at a higher efficiency as compared to the passenger bus engines, and it will be assumed the DING version can match the efficiency. As mentioned earlier, today's cost to convert the vehicle for LNG operation will be assumed to be $33,000.

Figure 50 shows the results of converting a class 8 truck to operate on LNG in a DING application. Operating on LNG results in an annual cost savings of $2860/year which provides a rate of return of $3.5% based on a 15 year period at current costs. Even though this return is much better than the passenger bus application, its still not acceptable due to the long payback period at today's economics.

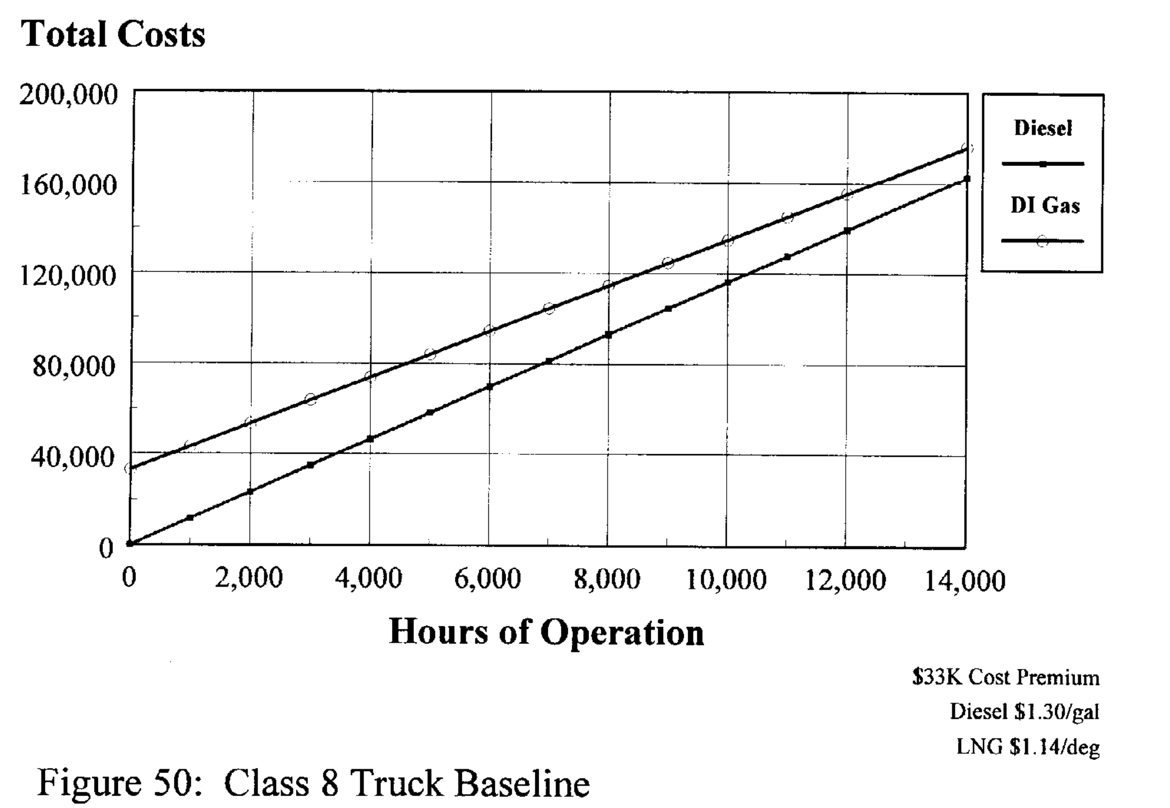


Figure 50: Class 8 Truck Baseline

Fuel costs are a significant portion of the owning and operating costs of an on-highway truck. Reducing these costs will provide a significant benefit. The current costs of $0.50/gal of LNG represents an average cost which can vary depending on location. With additional infrastructure, the cost of LNG is expected to drop further. Lowering the tax rate of LNG to the same rate as CNG will provide an immediate cost relief. As mentioned earlier, this would reduce the cost of LNG by approximately $0.25/deg. Figure 51 shows the results of reduced fuel costs.

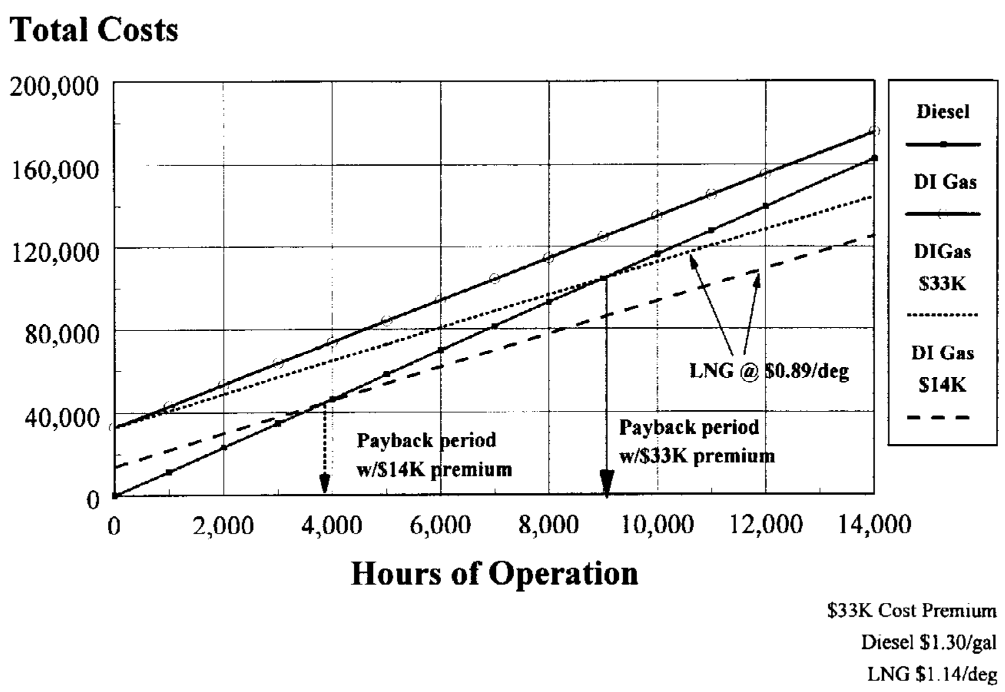


Figure 51: Class 8 Truck w/Lower Taxes and Cost Premium

Lowering the LNG tax rate provides an immediate and significant reduction in operating costs due to lower fuel costs of $7340 annually as compared to the diesel baseline. This increases the rate of return to 21% and reduces the payback time to approximately 9000 hours of operation at today's fuel handling system costs. Accounting for the expected drop in fuel handling costs to $14,000/vehicle reduces the payback period to under 4000 hours which results in a very substantial improvement in rate of return to approximately 50%.

Summary:

Due to the lower cost of natural gas as compared to diesel fuel, technology is being developed to utilize natural gas as a vehicular fuel. Engine manufacturers are modifying their diesel engine designs to operate on natural gas using an Otto cycle. These engine designs require natural gas supplied in the 50-150 psi range. On-board storage of the natural gas is a difficult and expensive challenge. Most natural gas fueled vehicles use tanks of CNG for fuel storage. These systems can easily provide the natural gas at the desired pressures for SING operation but can store only approximately ¼ the fuel in the same volume as compared to a diesel tank which severely impacts vehicular range. Even with the lower price of natural gas, it is difficult to justify the cost premium of an SING engine system with CNG tanks due to the lower efficiency of an SING engine relative to the diesel. To justify the cost premium, the natural gas engine must operate at efficiencies nearly equal to the diesel. With DING technology, engine operating efficiencies can equal those of the diesel, but requires a high pressure (3000 psi) fuel handling system. A CNG system initially charged at 3600 psi would require an on-board compressor to maintain pressure. Gas compressors are strictly used in stationary applications and are not practical for mobile applications. For DING applications, an LNG system would be much more appropriate as it is much more efficient to pump a liquid up to operating pressure as opposed to compressing an equal amount of gas. LNG fuel handling systems store the natural gas as a liquid in a special cryogenic tank which can store the liquid for approximately 2 weeks. These tanks are considered a mature technology and are being utilized for vehicular use. Costs of the LNG tanks are expected to be reduced as manufacturing volumes are increased. The LNG fuel handling system will vaporize the LNG through a heat exchanger using engine coolant. For DING applications, a cryogenic pump will be required to maintain desired pressure and will be the most critical technology to be developed. Operation of a DING engine can provide a significant return on investment if the natural gas engine system cost premium can be reduced (which is expected with increasing volumes) and the Federal Excise taxes on LNG can be reduced to the same levels as that of CNG.

PHASE 2 PLANS

The plan for Phase 2 will be similar to that in the original proposal (dated August 1995) of this 3-phase program . Task 1 will be modified, while Tasks 2-4 will be essentially the same as in the original proposal.

Task 1 "DING Engine Development" will continue, with focus on selecting and demonstrating the best options for meeting the component durability targets in key areas such as the ignition assist and gas injection systems. These systems in the preceding phase (Phase 1) were adapted to the DING engine based primarily on scaling from previous Caterpillar DI alternative fuel experience (i.e., methanol engine). They worked relatively well in this early Phase 1 demonstration, but the results indicate that several of the current systems (primarily the ignition assist and gas injection systems) need additional "margin for error" to be acceptable commercially. For example, the ignition assist technology in the current DING engine has less "margin for error" than with methanol because natural gas is more difficult to ignite than methanol. This lack of "margin for error" has slowed the ability to accumulate test hours on the 3516 engine in Phase 1. Therefore, ongoing Task 1 work will focus on identifying/selecting ignition assist and gas injection systems modifications that provide the additional "margin for error". Identification and selection of the best modifications will be performed utilizing the 3501 DING engine. Input from 3500 DING CFD models will provide guidance. As planned, later demonstration of the best configurations will be performed on a multicylinder DING engine. The relatively large, 16 cylinder engine (3516) has also added significant complexity to this early technology development process and has slowed the development. Because the 3516 DING engine is an inefficient developmental test bed (from both development time and cost standpoint) at this early development stage and because of the desire for an eventual DI gas field demonstration in a "transportation" vehicle, Task 1 will also focus on designing, procuring, and assembling a Caterpillar 3126 DING engine (typically 250 hp application in pickup/delivery and buses) to perform this multicylinder component durability demonstration. The current DING technology can be scaled to the 3126 engine in a relatively straightforward process. The primary modification will be designing the gas injector. Since the 3126 diesel currently incorporates a HEUI injection system (also used in DING engine), incorporating a DING injector in the 3126 engine will not pose significant difficulty. The 3126 DING engine is planned to be available late in 1997 for total DING system demonstration, following the component design option screening on the 3501. This change from focusing DING multicylinder development on the 3516 DING to focusing system development on the 3126 DING engine will provide a more efficient path for DI gas technology development. It will also provide a DING engine that can be used for field demonstration in the "transportation" sector, as compared to a 3516 DING engine that would be applied in earthmoving, locomotive or stationary power generation. The 3126 DING engine could also be adapted to a DI propane (DIP) engine, which may be a candidate for early field demonstrations.

Task 2 "DING Engine NOx Development" will continue as in the original plan, utilizing the 3501 DING engine as the demonstration platform. Development to achieve NOx = 2.5 gm/hp-hr with minimal thermal efficiency sacrifice will continue. The deliverable will be to identify the feasibility of DING engine NOx = 1 gm/hp-hr. This engine will also be used for component option screening in Task 1. This low NOx technology will be scaled to the 3126 DING engine.

Task 3 " Durability Development of 3000 psi Fuel Handling System" will continue as originally planned. The deliverable will be to demonstrate the feasibility of a 3000 psi LNG pump. This work will be performed by MVE, under subcontract to Caterpillar.

Task 4 "Commercial Application Study" will be initiated as originally planned. The deliverable will be to identify the specifications needed to make DING (and other DI gas engines, such as DI propane) commercially viable. Variables will include engine power output, load factor, application, initial cost of engine and system, emissions and fuel costs.

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13. ABSTRACT *(Maximum 200 words)*

The transportation sector accounts for approximately 65% of U.S. petroleum consumption. Consumption for light-duty vehicles has stabilized in the last 10–15 years; however, consumption in the heavy-duty sector has continued to increase. For various reasons, the United States must reduce its dependence on petroleum. One significant way is to substitute “alternative fuels”(natural gas, propane, alcohols, and others) in place of petroleum fuels in heavy-duty applications. Most alternative fuels have the additional benefit of reduced exhaust emissions relative to petroleum fuels, thus providing a cleaner environment. The best long-term technology for heavy-duty alternative fuel engines is the 4-stroke cycle, direct injected (DI) engine using a single fuel. This DI, single fuel approach maximizes the substitution of alternative fuel for diesel and retains the thermal efficiency and power density of the diesel engine.

This report summarizes the results of the first year (Phase 1) of this contract. Phase 1 focused on developing a 4-stroke cycle, DI single fuel, alternative fuel technology that will duplicate or exceed diesel power density and thermal efficiency, while having exhaust emissions equal to or less than the diesel. Although the work is currently on a 3500 Series DING engine, the work is viewed as “basic technology” development that can be applied to any engine. Phase 1 concentrated on DING engine component durability, exhaust emissions, and fuel handling system durability. Task 1 focused on identifying primary areas (e.g., ignition assist and gas injector systems) for future durability testing. In Task 2, eight mode-cycle-averaged NO_x emissions were reduced from 11.8 gm/hp-hr (“baseline” conditions) to 2.5 gm/hp-hr (modified conditions) on a 3501 DING engine. In Task 3, a state-of-the-art fuel handling system was identified.

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