February 2000 • NREL/SR-540-27501

Subcontractor Report

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

Final Report Phase II

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J.D. Hiltner, R. Nagaraj, and C. Emmer
Caterpillar, Inc.
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National Renewable Energy Laboratory

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

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Table of Contents

FINAL REPORT - PHASE 2

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 dependence 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 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 ignition 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 normally 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 closely approach 100% fuel substitution.

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 to 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).

Prior to this NREL contract, Caterpillar had initiated DING engine technology development with GRI. In this prior work, DING engine power and 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.

In Phase 1 of this NREL contract, the goal of operating a 3516 DING engine for 250 hours at various operating conditions to evaluate major component durability was achieved. The glow plug ignition assist and the gas injector system were identified as the primary components requiring additional development. The 3516 was determined to be an inefficient test bed for DING technology development, and it was decided to design and procure a 3126 DING engine for future DING development. A 3126 DING engine will also be a good candidate for field demonstration in the transportation sector. The goal of demonstrating DING engine NOx emissions = 2.5 gm/hp-hr was achieved on a single cylinder 3500-series DING engine (3501), although thermal efficiency was significantly reduced. Paths for minimizing the efficiency loss were identified and were planned for Phase 2. A survey of the state-of-the-art of 3000 psi fuel handling systems was completed.

This report summarizes the results of Phase 2 of the NREL contract. Additional DING technology development has been performed on a 3501 DING engine, and a 3126 DING engine has been designed, procured and preliminary performance testing initiated. A DING engine commercial application analysis has been performed, and a concept study identifying the major issues and options in designing a 3000 psi LNG system for a DING engine has been completed.

EXECUTIVE SUMMARY

1. A computational fluid dynamics (CFD) model of a 3500 DING engine gas injection/ combustion system has been developed and has been utilized to identify DING ignition/ combustion system improvements. Significant improvements in the thermal efficiency - NOx tradeoff have been demonstrated on a 3501 DING engine. For example, using EGR and split injection, the 8 mode, cycle-weighted average NOx levels of 2.5 g/hp-hr were achieved with 37.8% thermal efficiency. This is approximately a 20% improvement in efficiency compared to Phase 1 testing. Catalyst bench testing has demonstrated the ability to reduce NOx emissions to less than 1 g/hp-hr.

2. Components for a 3126 DING engine (300 hp) have been designed and procured , and the 3126 DING engine has been assembled. Preliminary performance testing has been initiated. The engine ran successfully at low loads for approximately 2 hours before injector tip and check failures terminated the test. The injector failures are believed to be solvable problems; however, it was decided to terminate this program phase at this point.

3. A Decision & Risk Analysis model has been developed comparing DING engine technology with various engine technologies in a number of commercial applications. The model shows the most likely commercial applications for DING technology and can be used to identify the sensitivity of variables that impact commercial viability. The model will be updated as needed and used as an ongoing tool in identifying DING engine commercial viability in various applications.

4. A preliminary concept design study by MVE Inc. has been completed that examines the major design issues involved in making a reliable and durable 3000 psi LNG pump. Primary concern is the life of pump seals and piston rings.

5. Plans for the next phase of this program (Phase 3) have been put on indefinite hold. Caterpillar has decided not to fund further Direct Injected Natural Gas work at this time due to limited current market potential for the DING engine. However, based on results from this program, it is believed that DI natural gas technology is viable for allowing a natural gas-fueled engine to achieve diesel power density and thermal efficiency for both the near and long terms

SUMMARY OF PHASE 2: FINAL ENGINE DEVELOPMENT

Task 1: DI Natural Gas Engine Development

Sub task 1.3: Component Development, 3126 DING Assembly and Durability Testing

Objective:

The subcontractor shall use the 3501 DING engine to identify and select glow plug ignition assist and injection system modifications that will provide acceptable performance/durability in a DING application. Analytical models will be utilized to aid in selecting the best modifications. The subcontractor shall adapt the technology to a 3126 DING engine which will move the project forward on a platform conducive to demonstration in an "over-the-road" transportation vehicle. The 3126 DING engine shall initiate performance/durability demonstration incorporating DING component modifications identified on the 3501 engine. The performance goal will be to demonstrate power and efficiency equal to a diesel engine.

Accomplishment Summary:

1. A computational fluid dynamics (CFD) model of the 3500 DING engine has been developed and is being utilized to identify thermal efficiency and NOx improvements on the 3501 DING engine. Significant efficiency - NOx tradeoff improvements have been demonstrated and are reported in Task 2.

2. A 3126 DING engine has been designed, procured and preliminary performance testing initiated. The engine ran successfully on all cylinders for approximately two hours before injector tip failures caused the tests to be terminated. No performance and durability data was acquired.

Accomplishment Details:

Computational Fluid Dynamics Modeling of In-Cylinder Events

Advanced computer modeling of in-cylinder events has been utilized to aid the design of the DING engine piston shape and intake air shield geometry. In addition to these direct applications, computational fluid dynamics (CFD) models have been constructed to provide a better understanding of the events leading up to combustion. Three models have been created which cover a range of quantitative accuracy and time required for model execution. The lowest level model consists of a fixed geometry piston/cylinder arrangement and allows the evaluation of various piston bowl shapes and injector configurations. A 72 degree sector model of the 3500 cylinder with moving piston geometry has been used to evaluate the penetration and dispersion of the natural gas fuel spray. The largest model is a full model of the 3501 engine, including intake port, that is being used to evaluate the in-cylinder flow field prior to ignition
.

The long computational times that are required by large-scale, detailed, in-cylinder models preclude their use as a means to evaluate multiple component geometries. For example, the running time for a full cycle on a single cylinder moving mesh problem is over two months using 4 processors on a Silicon Graphics Inc. Onyx 2 24 processor computer. The CFD files are typically run on 4 processors which maximizes the speed of the current code. In order to use CFD as a viable design tool, smaller models have been constructed to offer insight into the qualitative impact of piston and injector nozzle geometry on gas mixing. The mesh shown in Figure 1 is one example of this variety of model. The piston face geometry can be easily modified to simulate potential piston designs. In addition, the location, size, and number of nozzle holes can be manipulated to determine the effect of various injection patterns on the in-cylinder distribution. The separation between cylinder head and piston face can be modified but is typically set to the location of the piston near the beginning of the injection event. The cylinder pressure and turbulence level are also given as inputs to the model. Figure 2 shows some of the basic results from this model. The impact of the geometry of the impingement point on the distribution of natural gas in the cylinder is shown, with the raised center region allowing the natural gas to separate from the piston face. It is believed that the attachment of this center jet to the piston causes elevated CO and HC emissions at low loads. Because the piston location does not vary with time, the impact of the upward motion of the piston on in-cylinder flow is neglected in this model. Despite this drawback, several important design modifications have been guided by models of this type. With turnaround times on the order of a few hours, a large number of designs can be evaluated in a short period of time. The piston configurations that produce the most stagnant, flammable mixture in the region of the glow plug have been procured for testing in the 3501 DING engine.

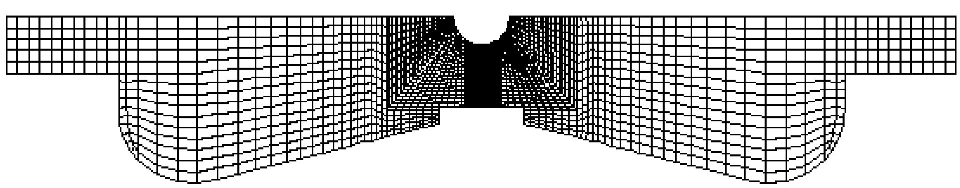


Figure 1: Section View of Fixed Piston 3501 CFD Model

In order to determine more accurately the distribution of fuel in the region of the glow plug, a pie sector model of the 3501 cylinder was created. This model includes 72 degrees of the cylinder with two of the nozzle holes straddling the glow plug and shield. Figure 3 shows the mesh geometry for this model. Piston motion is included so that the impact of increasing cylinder pressure and squish are present. Extensive information has been garnered from this model including the importance of the glow plug shield in retaining flammable mixture near the glow plug, the impact of piston bowl shape and angle of inclination on the attachment of gas jets to solid surfaces and the effect of injection parameters on the velocity of the gas jet at the nozzle outlet. A section view of one of the gas jets is shown in Figure 4. The shading in the picture distinguishes the mixture that is above the rich flammability limit of methane from that which is below the lean flammability limit. It is only the outside shell of the injection plume which can be ignited by the glow plug. This is a major contributing factor to the long ignition delays that have been observed in the DING engine. In the future, this model, or variations of it, will be used to optimize the glow plug shield geometry to: 1) retain flammable mixture in the region of the glow plug, 2) prevent cooling of the plug by the gas jet, and 3) minimize the cooling of the glow plug by in-cylinder air flow.

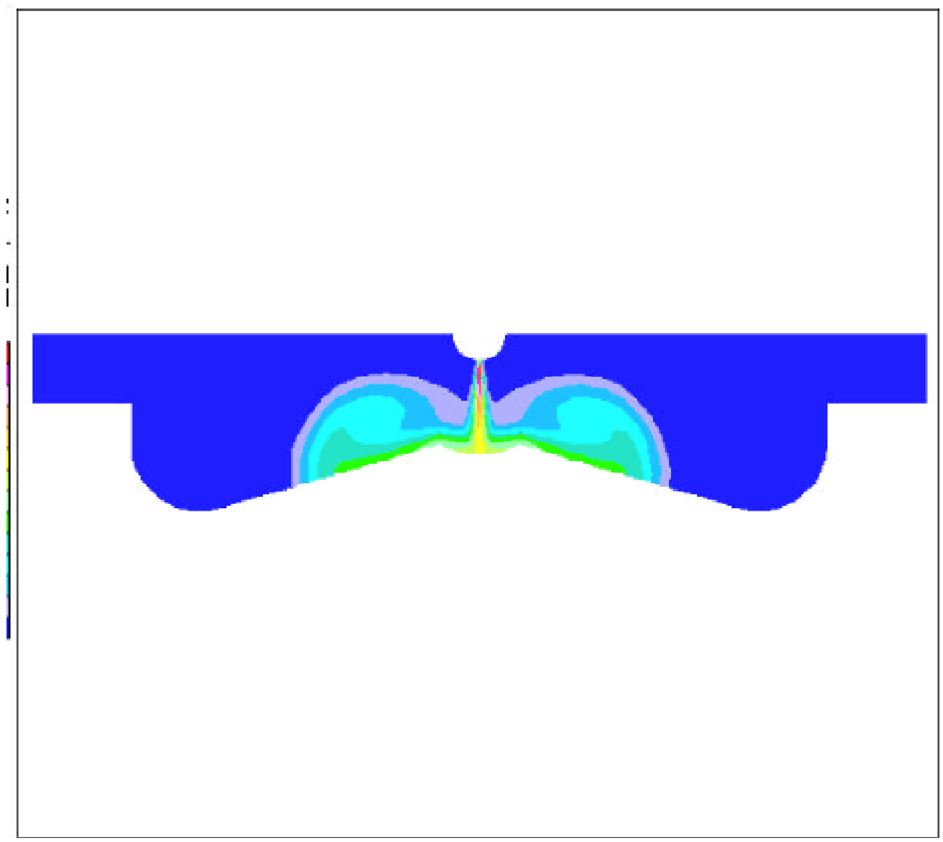


Figure 2: Methane Concentration Calculated From 3501 Model (Side View of Combustion Chamber Near Top Dead Center)

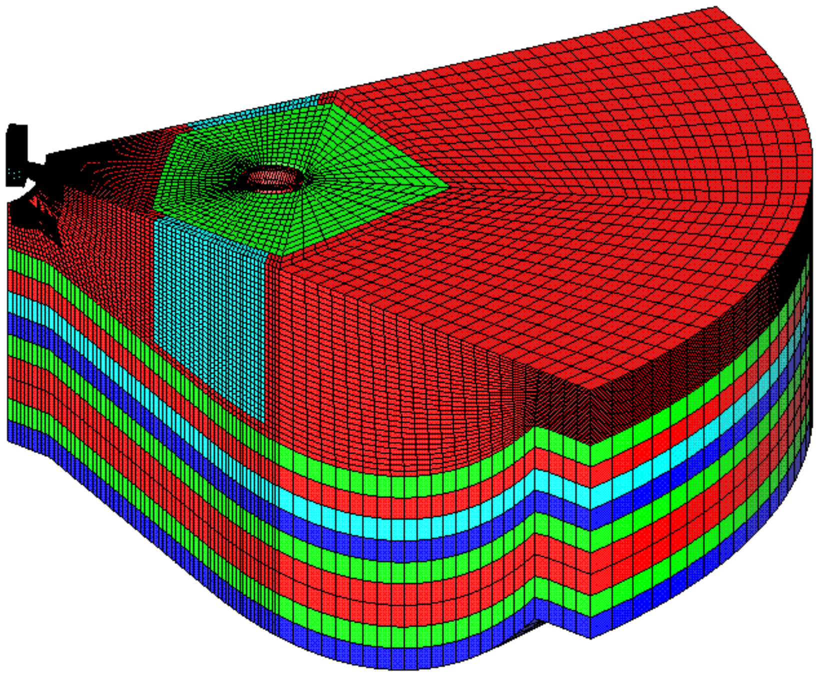


Figure 3: 72 Degree Sector Model of 3501 DING Engine Combustion Chamber

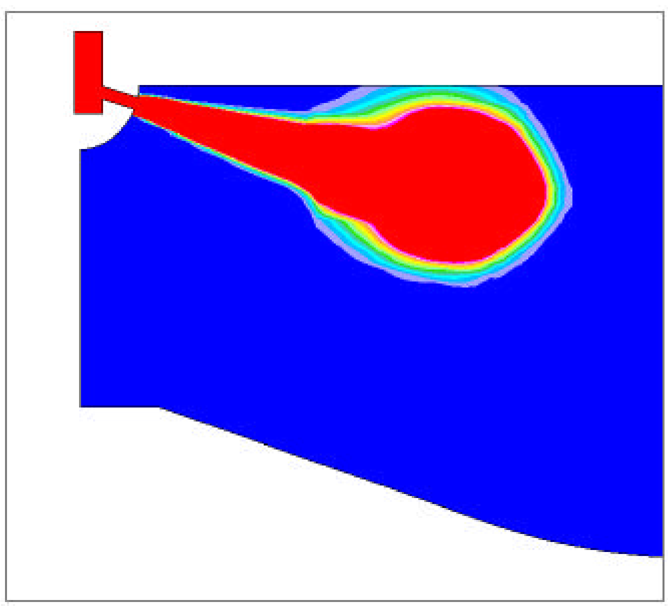
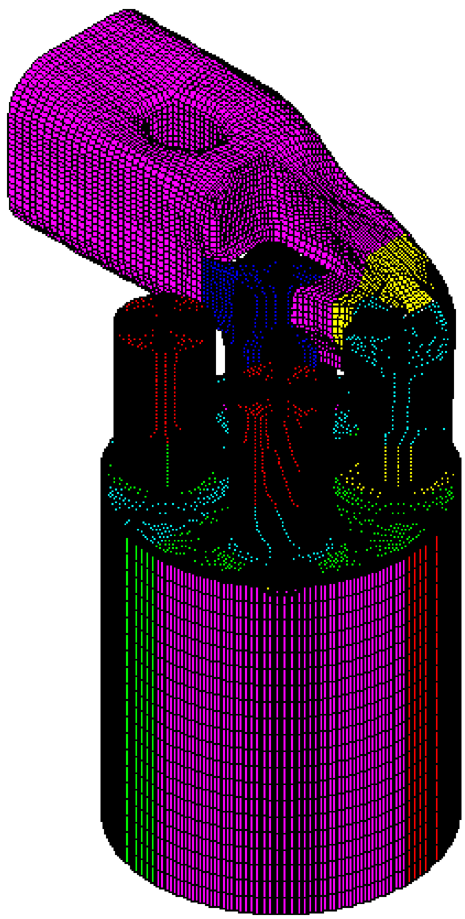


Figure 4: Fuel Concentration Profiles in 3501 DING Engine

In addition to the two limited-scope models that have been described, a comprehensive model has been constructed which includes the entire 3501 cylinder and piston geometry as well as the intake port geometry. The computational mesh for this model is shown in Figure 5. This model is being run from the time of intake valve opening until the beginning of injection, a period of 360 crank angle degrees. The extensive running time of the model prevents its use as a design tool, but the results to be obtained from it will aid in increasing the accuracy of the other in-cylinder models. Specifically, the turbulence intensity and velocity field at the time of injection are important determinants of the gas mixing rate. The sub-models described previously are currently using initial velocity and turbulence fields obtained from a significantly larger engine. The results of this analysis will provide both increased accuracy in the sub-models and a better understanding of intake flow as it affects glow plug temperature. The combination of the fixed piston sub-model, the pie sector moving piston sub-model and the full 3501 moving mesh model have provided significant input towards the improvement of the DING engine (discussed in Task 2).

Figure 5: 3501 DING Engine Cylinder , Piston and Port Model



3126 DING Engine Design, Procurement and Test

The primary design challenges centered around trying to incorporate natural gas injectors and glow plugs into a cylinder head that was designed for the diesel engine. The design of the gas injector is based on the 3500 DING injectors and also on an injector designed for a direct-inject propane engine. Figure 6 shows cross sections of the injector design. A major challenge in designing the injector was incorporating seals to seal the 3000 psi gas from the 3000 psi oil within the constraints of the 3126 diesel cylinder head. The sealing arrangement that has been incorporated is expected to work adequately for performance demonstration, but will be closely examined for durability. Additional sealing improvement options have been identified and may be examined in Phase 3 of the program. One feature that has been added to the 3126 DING injector that was not previously included in the 3500 DING injector, is the Teflon seal around the check valve. High pressure oil in the oil seal line is used to keep the natural gas from getting under the check valve. The oil pressure in this line is always higher than the gas pressure, and, therefore, without the Teflon seal, a small amount of oil can leak down the check valve into the natural gas. This oil adds to the overall particulate emissions. Therefore, incorporation of the seal is expected to reduce particulate emissions.

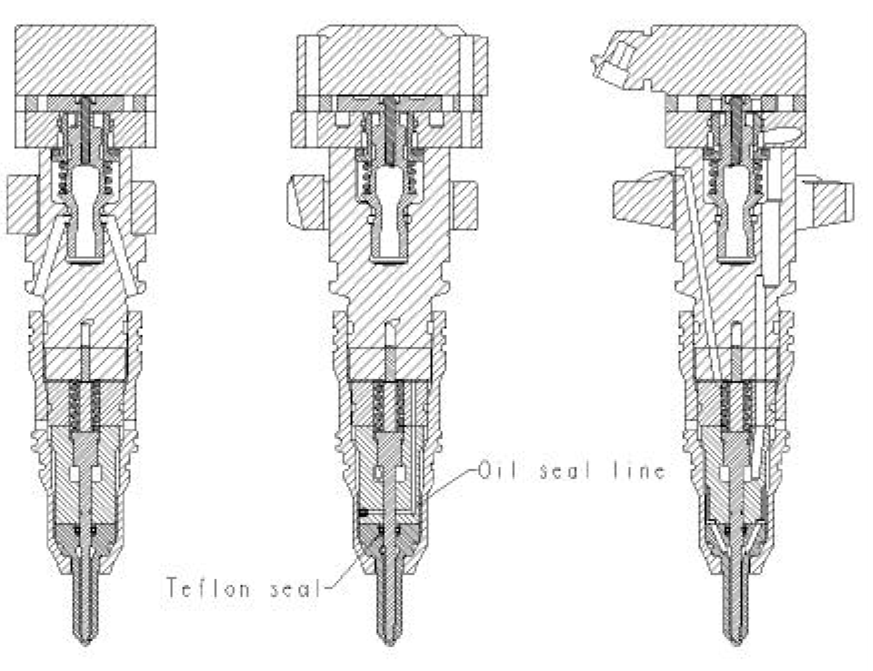


Figure 6: 3126 DING Injector

Another challenge was locating the glow plug in the cylinder head. Figure 7 is a section of the cylinder head through the glow plug hole. The glow plug was located as close to the injector as the head casting would allow. In this location, the hot spot of the glow plug is approximately 30% closer than the "production" location (the diesel "production" cylinder head incorporated a

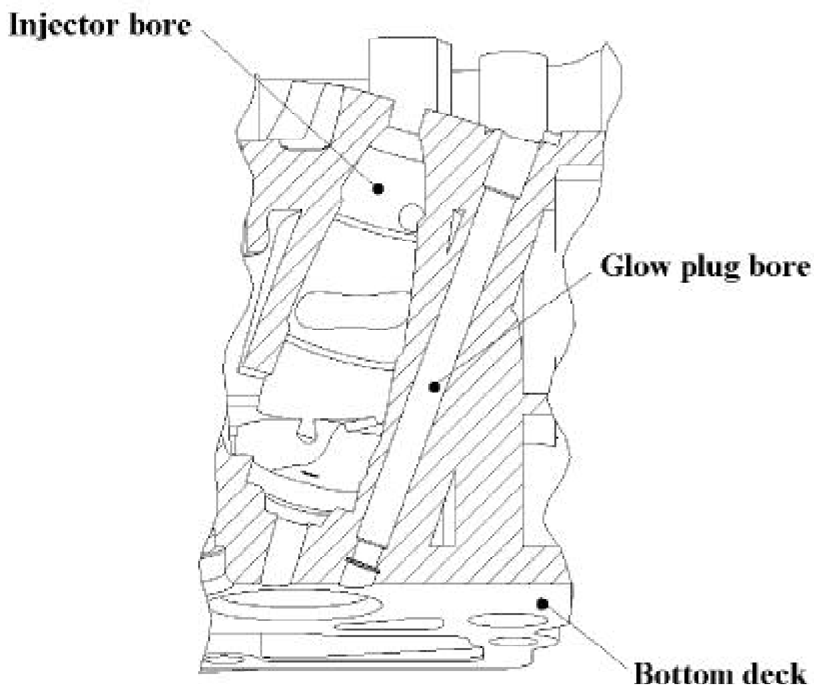


Figure 7: Section Through 3126 DING Cylinder Head

boss for "future" glow plug application). The closer location required a modification to the cylinder head casting. The initial testing will use
the production cylinder head since it was installed prior to the arrival of the modified heads. This will provide additional performance data. Table 1 shows the distance from the injector tip to the glow plug (distance X) for various engines versus the Caterpillar 3406 methanol engine (all the engines listed are from research projects). The 3406 was chosen as the "ideal" because it performed very well. Further work at the University of Illinois and Sandia National Lab will help determine the relative penetration distances of natural gas versus methanol. This work may change the "ideal" distance of the glow plug to the injector for DING applications.

Engine	Ratio of distance X vs. 3406
3116	1.14
3126(p)	2.24
3126(m)	1.51
3171	1.03
3501(o)	1.36
3501(c)	1.08

Table 1: Glow plug to injector distance ratios

In the previous table the (p) denotes the production boss location in the 3126 cylinder head. The (m) denotes the modified design with the glow plug closer to the injector. The (o) stands for the original 3500 cylinder head design, the (c) means the closer version.

Another design issue was the exhaust gas recirculation (EGR) system. Several modifications and additions to the diesel EGR system were required for the DING engine. The production system did not include a cooler. To be comparable to the 3501 data, cooled EGR was needed. A heat exchanger capable of cooling 20% EGR at rated power from 660° C to 140° C was designed and procured.

The 3126 DING engine was assembled and installed in the test cell to initiate performance testing. Figures 8 through 11 show the 3126 DING engine installed in the test cell.
The 3126 DING engine was initially run at low loads, with all six cylinders firing the very first time the engine was run. A problem with the injector check valve spring appeared shortly after testing commenced. The spring did not meet print specifications and was breaking. The problem was corrected by obtaining springs that met print specifications (from a different supplier). Testing was restarted, but again halted a short time thereafter. Two problems arose during the brief testing. Inspection revealed that the tip had failed on several injectors. This is shown in Figure 12. The resultant damage to the check is shown in Figure 13. Figure 14 shows an undamaged check for comparison. The second problem was that the check valve stem was breaking at the location of the undercut. This undercut is the annulus for the oil to seal the high pressure gas from the control oil. A broken check is shown in figure 15.

At this time, it is unclear why the tip would fail. The material is the same as that used on the 3500 DING injector tips. The 3126 DING injector tip is a "VCO-type" design, versus a "sac-type" design for the 3500 DING. It may be that the load of this check on the tip in the area of the orifices was too great. A complete structural analysis of the tip should be undertaken before proceeding with further testing.

The injector problems are likely solvable. The most probable solution would be to change the material of the tip. It may be necessary to provide cooling to the tip as well. An analysis of the parts will determine steps needed to correct the situation. Also, the check valve would be redesigned to remove the undercut and place the annulus in the guide instead. Fixing these problems will allow the performance goal of matching or beating diesel power and thermal efficiency to be achieved.

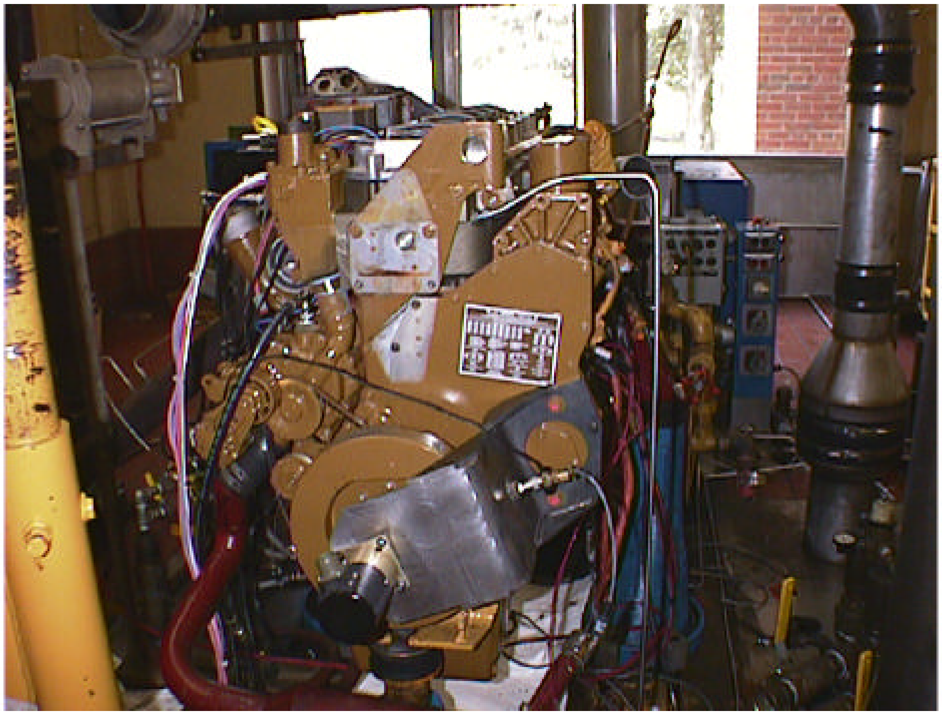


Figure 8: 3126 DING Engine in Test Cell

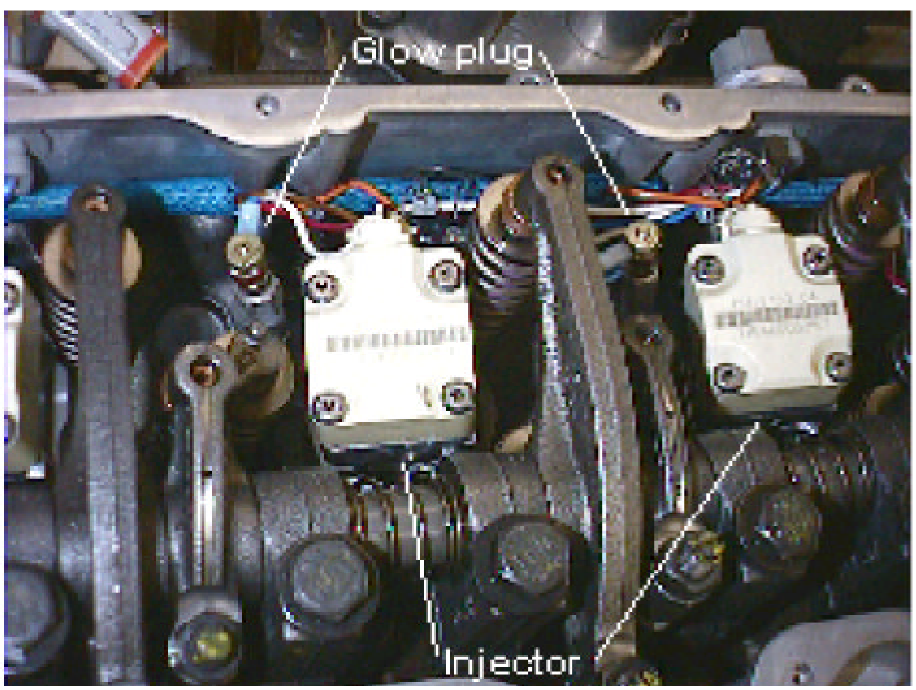


Figure 9: View Showing 3126 DING Glow Plug Location

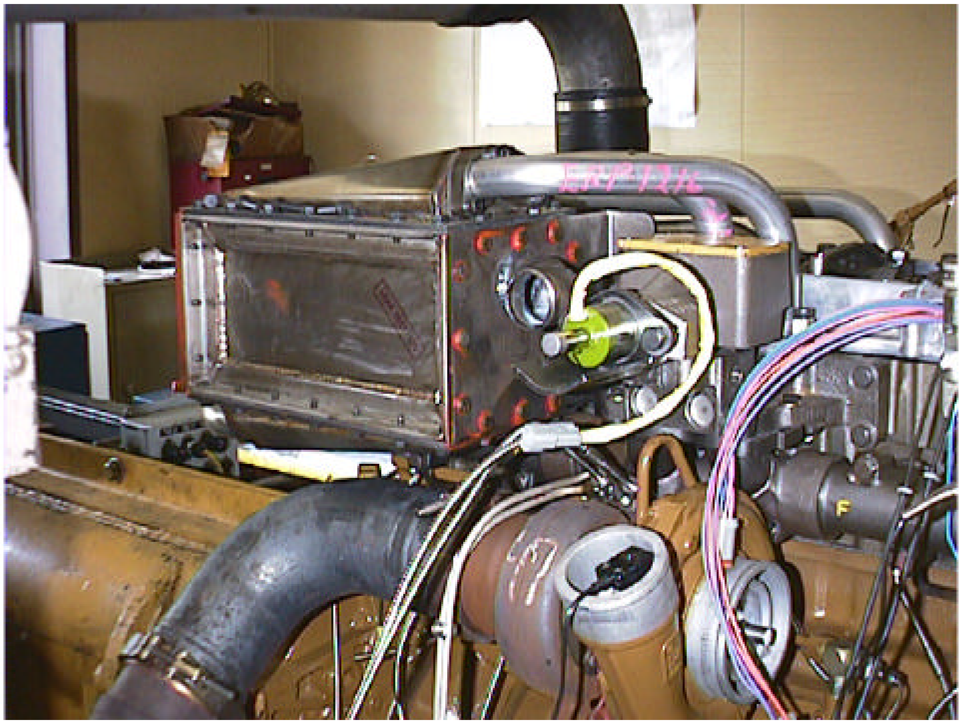


Figure 10: 3126 DING EGR Cooler Installation

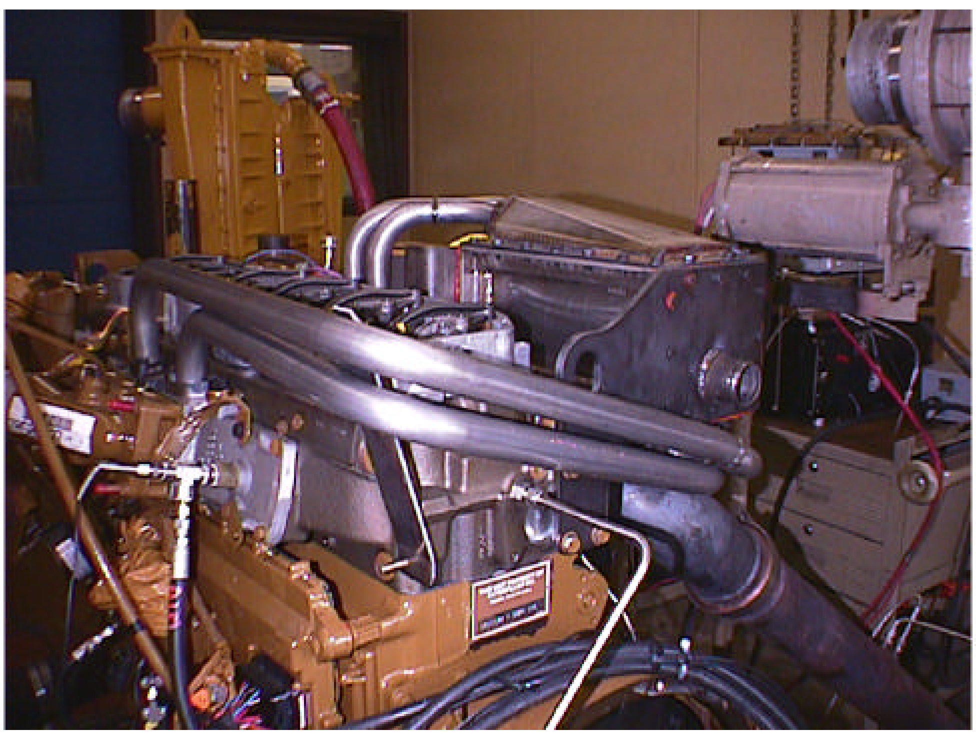


Figure 11: 3126 DING EGR Jumper Tubes

Figure 12: Damaged injector tip
(the small piece fell off after the injector was removed)

Figure 13: Damaged check - from injector shown above

Figure 14: Undamaged check

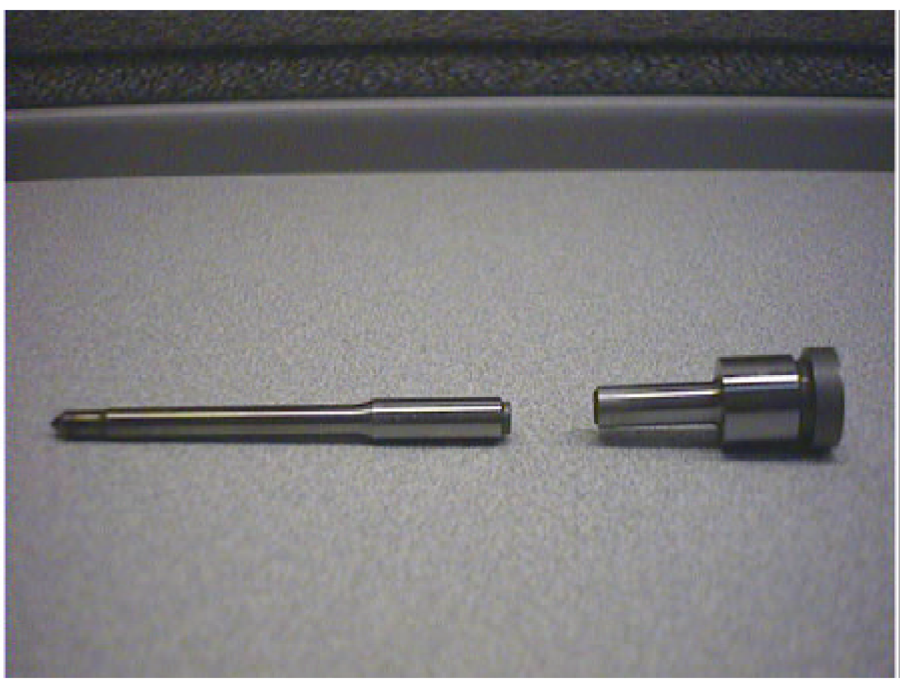


Figure 15: Injector check broken at undercut

Task 2: DI Natural Gas Engine NOx Development

Sub task 2.3: Evaluate Feasibility of Meeting 1.0 gram/hp-hr NOx Goal

Objective:

Based on the results from Sub task 2.2 (Phase 1), the subcontractor shall explore the feasibility of meeting 1.0 g/hp-hr NOx goal. In this task, the subcontractor shall determine whether additional engine modifications and aftertreatment systems are necessary to meet this goal. The subcontractor shall then design, fabricate, procure, and integrate these changes to the engine and perform emissions testing as specified in Sub task 2.2.

Accomplishment Summary:

1. With the aid of computational fluid dynamics (CFD) modeling, significant improvements in the NOx - thermal efficiency tradeoff have been demonstrated on the 3501 DING engine, relative to Phase 1 of the program. Without exhaust aftertreatment, rated power thermal efficiency at 2.5 gm/hp-hr NOx is 39%, while 8-mode cycle-averaged NOx levels of 2.5 gm/hp-hr were demonstrated at above 37% thermal efficiency. Light load performance has increased as much as 20% utilizing split injection.

2. Selective catalyst reduction bench tests have demonstrated that NOx levels below 1.0 gm/hp-hr are achievable on the DING engine. Such techniques are planned for demonstration on the 3126 DING engine in Phase 3.

Accomplishment Details:

<u>3501 DING Engine Tests to Improve Thermal Efficiency - NOx Tradeoff</u>

From a combustion standpoint, natural gas is a desirable fuel because it has a lower adiabatic flame temperature as well as a simpler hydrocarbon structure that allows equivalence ratios as rich as 2:1 to occur before significant soot formation begins. This has allowed direct inject natural gas engines to run with further retarded injection timings than diesel fueled engines while still maintaining acceptable particulate levels at reduced NOx emissions. However, retarding injection timing increases fuel consumption and therefore CO2 emissions which are a known green house gas. To find a better solution to this NOx - efficiency trade off, many methods other than retarded injection timings were evaluated in this phase of the program. Table 2 gives a list of the methods tested as well as a brief matrix showing other variables of concern that would be affected. Improved modeling strategies were needed to meet the NOx goal of 2.5 g/hp-hr at 40% thermal efficiency.

Computational Fluid Dynamics (CFD) is being utilized to fill this need. CFD has proven to be extremely beneficial in reducing NOx in turbine combustors by allowing leaner air/fuel ratios and achieving a more uniform air-fuel mixture. Reciprocating engines are more complicated to model due to their moving valves, pistons, and complicated geometry. This type of modeling has therefore lagged the turbine industry. In this project CFD was used to help determine the optimum piston bowl diameter and depth, injector spray angle, injector orifice size, glow plug shielding, and piston impingement pin. CFD was also used to understand where the flame kernel initiates and how it propagates, leading to a diffusion controlled burn.

	Reduced NOx	Light Load Performance	Reduced COV	Peak Efficiency	Ignition Durability	Production / Marketability
EGR	+ +		-	-	-	
SCR (Bench Test)	+ +			-		-
Lean Premix	+				+	
Split Injection		+ +	+		+	
High Squish		+	+	+		
Injector Nozzle Configuration	+	+	+	+	+	
Fuel Heating	-	+	+		+	
Glow plug shielding		+	+	+	+	
Glow plug location		+	+	+	+	

Table 2 - Methods tested to reduce NOx
Note: + refers to a positive effect
- refers to a negative effect

To achieve EGR with the 3501 single cylinder test engine, a system was designed to allow exhaust gas to flow from the exhaust manifold through a valve and heat exchanger into the intake manifold. The valve combined with boost andbackpressure was used to regulate the amount of EGR. Figure 16 shows the overall system. This system is conceptually similar to the design incorporated into the 3126 DING engine. Systems like this are typically used in applications where the back pressure created by the turbo is greater than the boost for the operating range that EGR is desired. For the smaller 3126 engine, this is the case for most of the operating range. However, on larger engines with more efficient turbochargers the back pressure may never be greater than the boost which would necessitate a venturi or a means other than back pressure to get exhaust gas into the intake manifold. On the single cylinder engine, when EGR was activated, the mass flow of fresh intake air was reduced to simulate what would happen in amulticylinder engine. Fresh intake air flow was reduced in proportion to the amount of EGR and the new corresponding overall turbo efficiency. A test matrix was run to determine the amount of EGR that would be required to achieve the cycle NOx and efficiency deliverable.   Rated power and peak torque are the primary contributors to the overall weighted NOx and efficiency measures. The NOx efficiency tradeoffs are shown for those points in Figure 17. As seen in Figure 17, 15% EGR was needed to produce NOx emissions of 2.5 g/hp-hr at peak torque and rated power. Smoke was not measurable at either point. EGR is successful at reducing NOx with lesser an impact on efficiency because it reduces peak flame temperatures while maintaining a similar heat release rate to achieve high efficiency with low NOx.

Using EGR, split injection, and the 10 hole 70° nozzle with the 15.5:1 CR Mexican hat piston bowl, eight-mode BSNOx emissions were reduced to 2.52 g/hp-hr at 37.2% thermal efficiency Table 3 shows the contributions from each mode point to this average. Figures 18-23 show the relationship between NOx-EGR and efficiency for 1300 and 1750 RPM at 50%,75%, and 100% load cases with the hardware described above

Speed (RPM)	Load (%)	Efficiency (%)	BSNOx (G/hp-hr)	EGR (%)	Weighting Factor (%)	Percent of total cycle NOx (%)	Comments
700	33	26.3	5.6	0	15	2.5	Split injection used
1300	50	36	2.7	14.5	10	8	
1300	75	38.8	2.2	13.7	10	9.7	
1300	100	40.1	2.5	14	10	14.5	
1750	25	32.1	3.9	0	10	7.2	Split injection used
1750	50	35.4	3.3	16.5	15	17.8	
1750	75	36.5	2.5	18.2	15	19.2	
1750	100	38.5	2.1	15.4	15	21.1	
Cycle Average		37.2	2.52		100		

Table 3 - 8 Mode NOx Summary

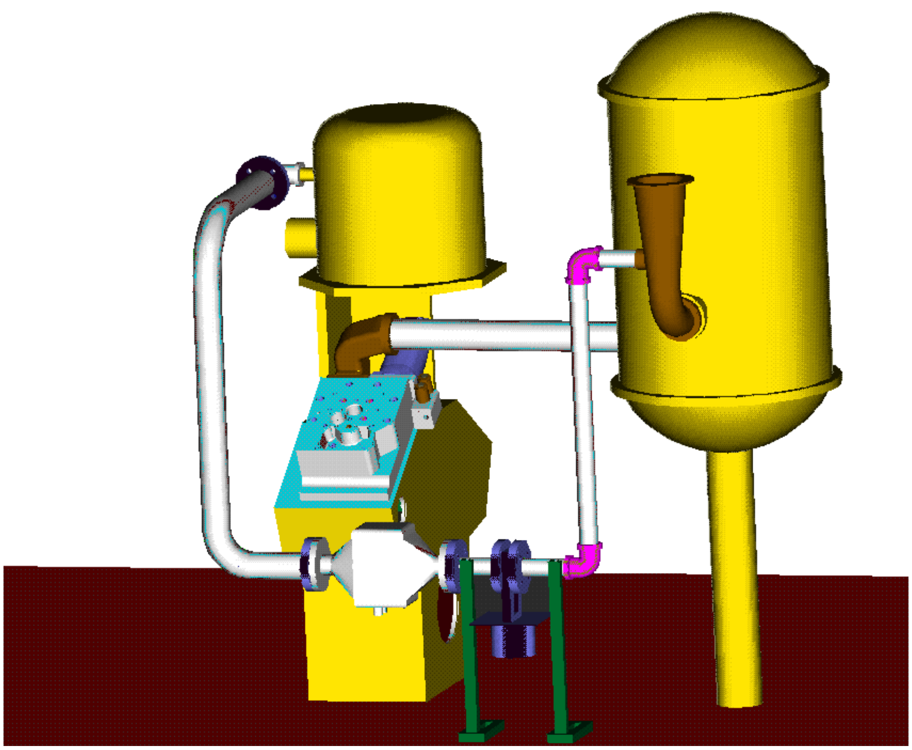


Figure 16: EGR Configuration for 3501 DING Engine

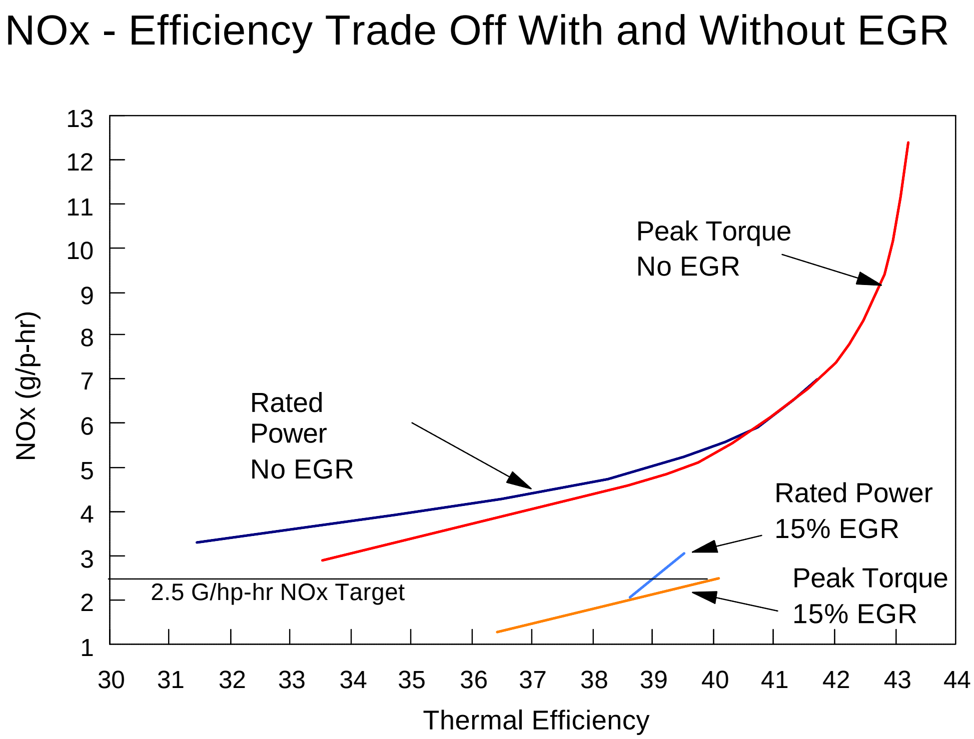


Figure 17 - EGR Required for BSNOx Target of 2.5 g/hp-hr (3501 DING)

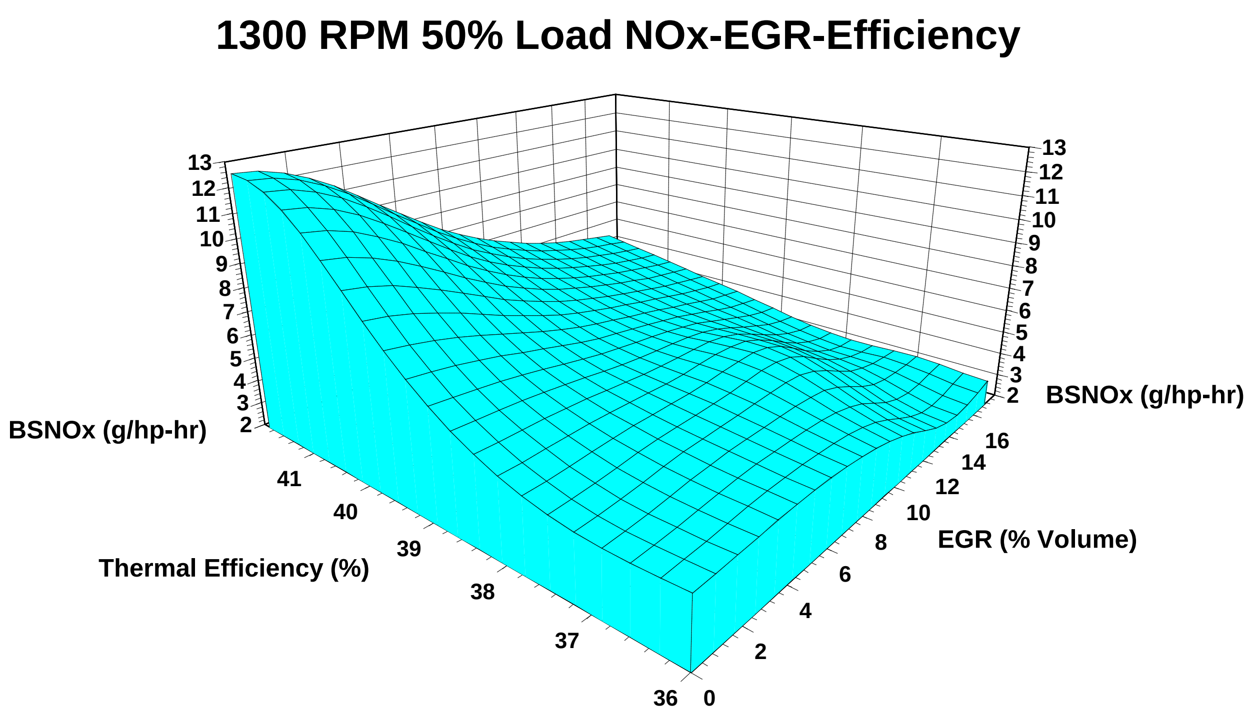


Figure 18 - Effects of EGR on NOx and Efficiency (3501 DING)

1300 RPM 75% Load NOx-EGR-Efficiency

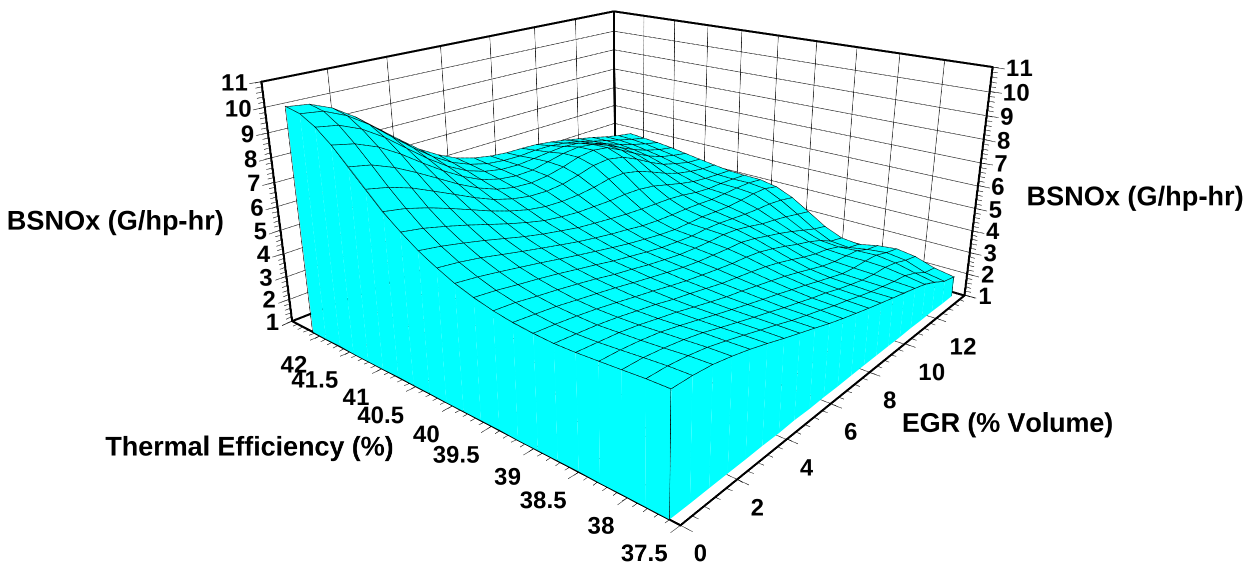


Figure 19 - Effects of EGR on NOx and Efficiency (3501 DING)

1300 RPM 100% Load NOx-EGR-Efficiency

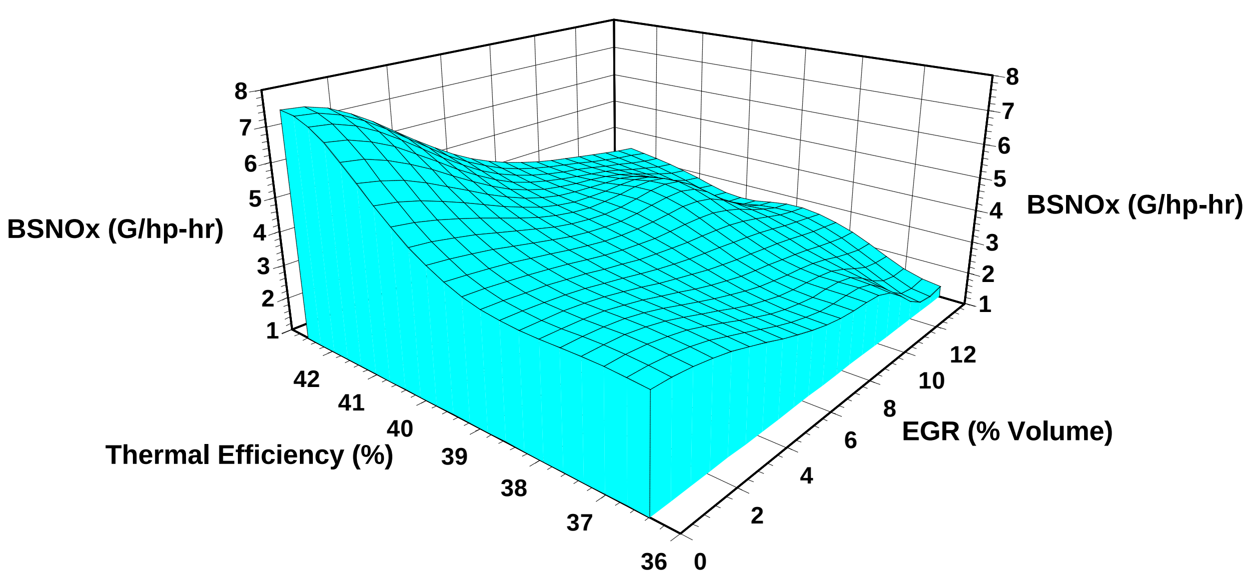


Figure 20 - Effects of EGR on NOx and Efficiency (3501 DING)

1750 RPM 50% Load NOx-EGR-Efficiency

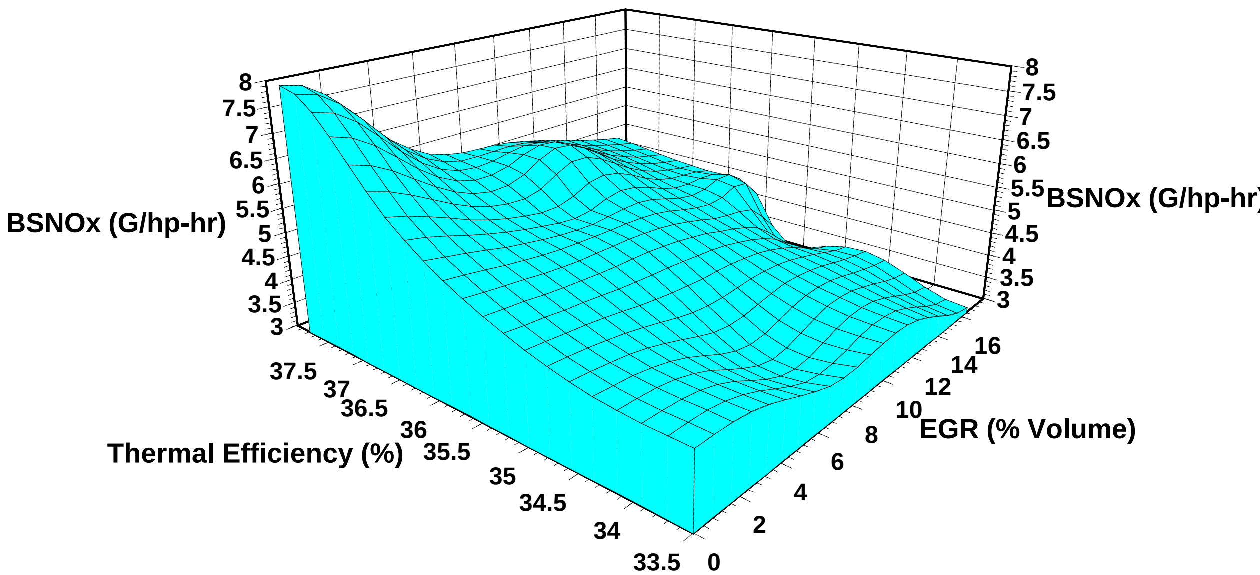


Figure 21 - Effects of EGR on NOx and Efficiency (3501 DING)

1750 RPM 75% Load NOx-EGR-Efficiency

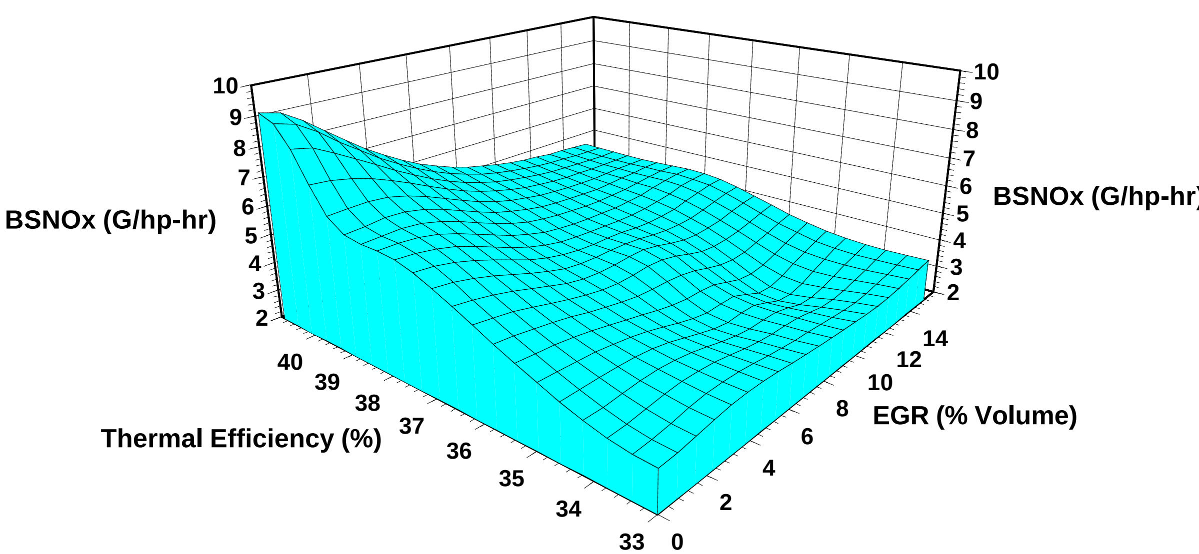


Figure 22 - Effects of EGR on NOx and Efficiency (3501 DING)

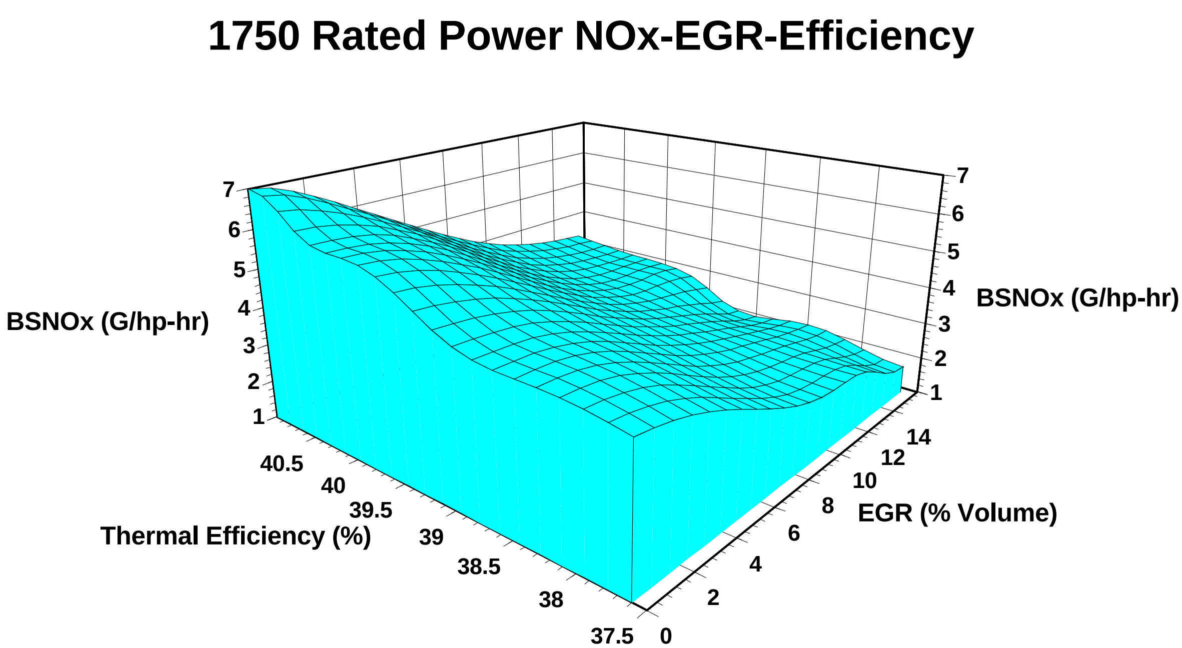


Figure 23 - Effects of EGR on NOx and Efficiency (3501 DING)

The lean premix concept was another means of trying to reduce the peak flame temperatures and their higher corresponding NOx formation rates. Diffusion flames are known to occur near stoichiometric conditions. Stoichiometric flames produce the highest flame temperatures. It was thought that by mixing a fraction of the fuel with the intake air a significant portion of the burning could be forced to occur at a mixture leaner than stoichiometric. Natural gas is a very favorable fuel to mix with the intake air since its lower flammability limit is approximately 5% by volume. This allows fuel to be mixed with the intake air at ratios as rich as 36:1 before ignition should occur. This concept also provides the benefit of reducing the amount of natural gas that has to be compressed to 3000 PSI. This would result in either a smaller LNG pump or open the possibility of using a small gas compressor. Figure 24 shows a solid model of the lean premix port injection manifold that was tested on the 3501 engine.

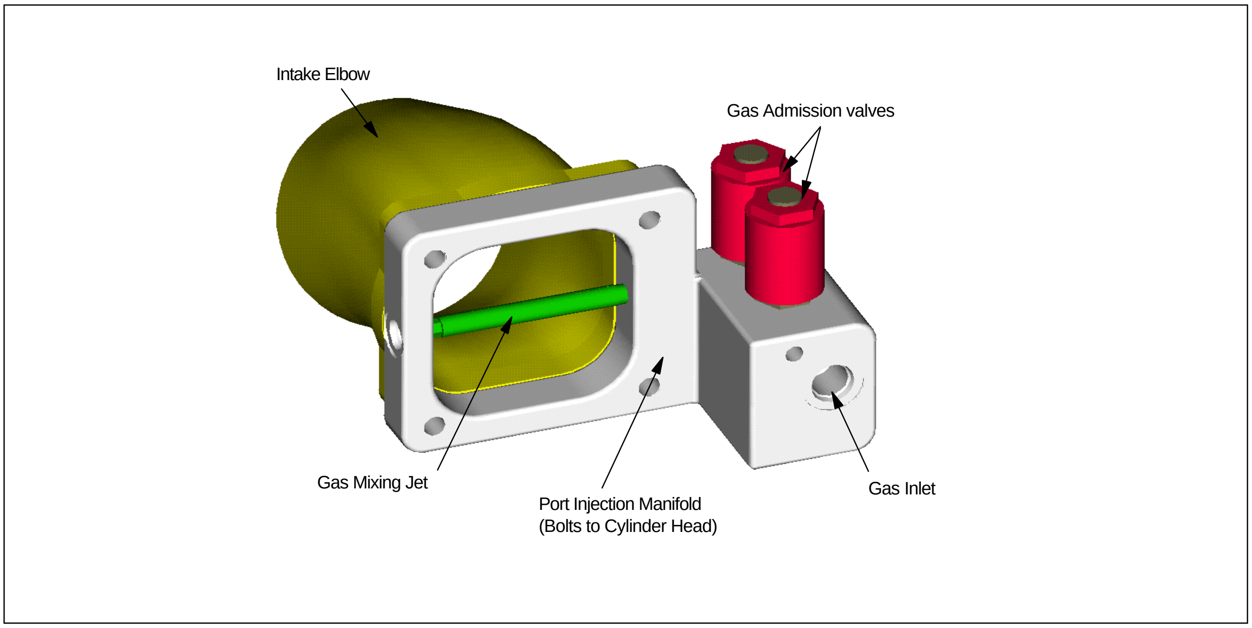


Figure 24 - Lean Premix Port Injection Manifold (3501 DING)

For maximum flow control a pressure ratio of minimally 2:1 is required to ensure choked flow. Boost pressures can be as high as 60 in-hg (30 PSIG) so a supply pressure of minimally 60 PSIG is needed for this type of system. For best mixing, gas valve timings of 10 degrees after intake valve opening and 30 degrees before intake valve closing are needed. From this it was determined that two gas admission valves would be required to achieve an air-fuel ratio of 36:1 at rated power. This system was first tested with the original direct inject nozzle. This led to a very rapid rate of heat release at the end of combustion. To stretch the main injection duration out and reduce the tendency to knock, an injector nozzle with approximately half the flow area was procured. The new nozzle had 8 holes instead of 10 and did not have the center hole.

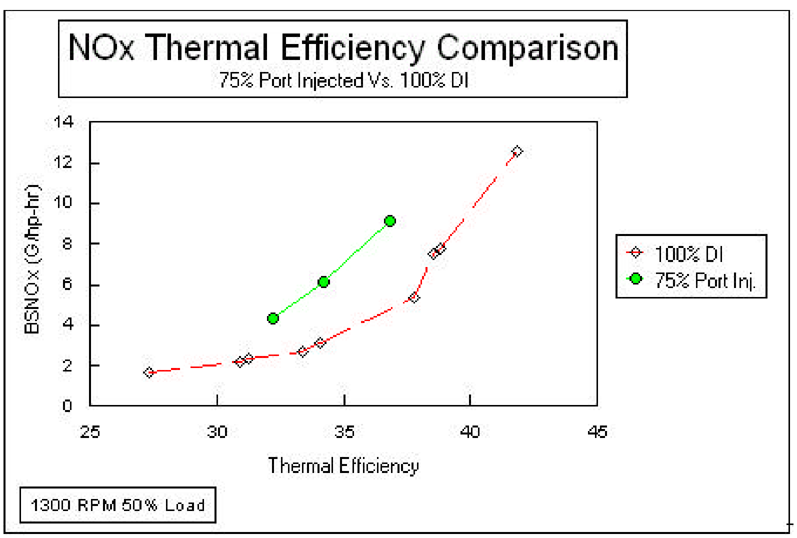


Figure 25 - 3501 Lean Premix Summary

Fewer holes were used in order to maintain maximum penetration. The angle between the sprays was 36 degrees for the sprays that straddled the glow plug and 46.3 degrees for all the other sprays. Data acquired at 50% load with this nozzle is presented above in Figure 25. As shown in the figure, the NOx-efficiency trade off was worse with the port injection. With lower port injection substitution rates the NOx-efficiency trade off approached that of the 100% direct inject case. The negative impact on the NOx-efficiency trade off is a result of a significant increase in the unburned hydrocarbons and a decrease in the rate of heat release that occurred as a result of less turbulence. At rated power and peak torque the NOx-efficiency relationship was found to be much the same. However, the maximum substitution to avoid knock was between 40-60% of the total fuel rate depending on speed and load. With the large flow area nozzle the maximum substitution was between 25-50% of the total fuel rate. Results of this testing indicate there is no NOx advantage to port injecting a fraction of the fuel with the intake air and igniting with a DI natural gas pilot.

Several nozzle configurations were tested in an attempt to reduce NOx and increase performance by increasing the combustion stability or reducing the coefficient of variation (COV). Typical NOx-efficiency curves are nonlinear. This is largely a function of the NOx activation energy's exponential dependence on temperature. If the combustion event does not repeat itself each cycle, the in-cylinder temperature will be different from cycle to cycle. This difference causes the engine to run at a different point on the NOx-efficiency curve each cycle. When these points are averaged together, the NOx is slightly higher at a given efficiency than what would occur if combustion always initiated at the same time (Figure 26). It was expected that any change that would occur would be a small incremental step in terms of NOx emissions. However, if a nozzle was found that reduced COV, it would typically reduce the amount of power that the glow plug would require. Also, efficiency can increase since individual cycle peak cylinder pressures that limit the amount of timing advance that can be used will more closely approach the mean peak cylinder pressure of many cycles.

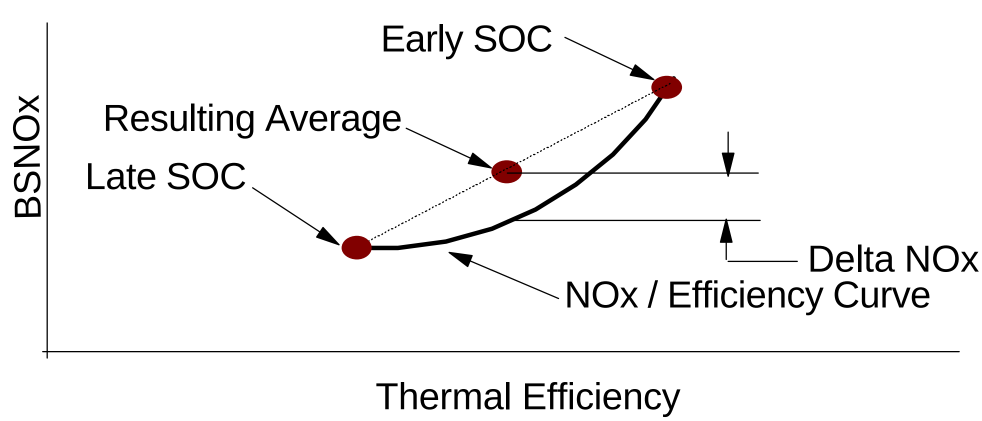


Figure 26 - NOx Reduction Due to Reduction in COV

Eight different nozzles were procured. Table 4 gives a brief description, design intent, and results for each nozzle. Each nozzle was tested at the extreme points of the engine operation; idle, high speed light load, rated power, and peak torque. Peak efficiency and BSNOx were measured at each point and compared. At rated and peak torque several timings were run to get a indicator of the NOx efficiency tradeoff.

Nozzle Description	Design Intent	Results
4 Hole Stratified Charge 4 x .900mm 74º Angle	Stratify a combustible mixture of fuel and air near the glow plug. Stratification was to increase efficiency by reducing amount of fuel that was mixed below flammability limit.	Increased low load efficiency to near baseline diesel. However, nozzle did produce smoke at the higher speeds. Did not work well with the lean premix port inject concept.
10 Hole - No center 10 x .965mm 74º Angle	Remove center hole to reduce smoke at retarded injection timings. Larger holes also increase penetration.	Reduced smoke to immeasurable values for most operating points.
12 Hole - No center 12 x .88mm 74º Angle	Increase air utilization, aid flame propagation.	Increased COV, CFD models indicate a lean condition near glow plug.
16 Hole - No center 16 x .83mm 74º Angle	Increase air utilization, aid flame propagation.	Not tested to date due to problems encountered with 12 hole nozzle.
10 Hole plus center 90% flow area 10 x .85mm 74º Angle 1 x 1.04mm	Reduce amount of fuel in cylinder before SOC as well as reduce rate of heat release.	NOx / efficiency curve not dramatically altered. COV was reduced slightly. Peak efficiency reduced at most .5% due to later end of injection.
10 Hole no center 10 x .965mm 70º Angle	Stop attachment of natural gas to cylinder head, which would increase penetration and air utilization, as well as reduce CO.	CO and HC reduced when combined with split injection.
8 Hole no center 50% flow area 8 x .760mm 70º Angle	Reduce rate of heat release and increase detonation margin with lean premix concept.	Increase in knock resistance when used with split injection. Did not improve NOx-efficiency tradeoff with lean premix.
8 + 2 asymmetric 8 x .920mm 70º Angle 2 x 1.125mm 70º Angle	Form a combustible mixture near the glow plug earlier and increase the initial rate of heat release at the ignition source.	Decreased COV, Increased initial heat release rate. Particulates measured between .02 and .05 g/hp-hr depending on condition.

Table 4 - Injector Nozzles Procured / Tested (3501 DING)

The 4 hole nozzle was successful at reducing COV, HC, and ignition delay, as well as increasing the heat release rate and thermal efficiency. Figure 27 demonstrates this through a comparison of the cylinder pressure vs. crank angle plots of the baseline and four hole nozzle injector tips. The 4 hole nozzle had a significantly higher cylinder pressure rise rate so a less advanced timing was required for maximum efficiency. As a result of the charge stratification and reduced ignition delay, the fuel rate was reduced through a reduction in the unburned hydrocarbons. This represented approximately a 4 g/min (18%) improvement in idle fuel rate. The four hole nozzle was tested with the port injection to see if the combination could efficiently cover the load range. At part loads total hydrocarbon (THC) emissions were high and efficiency suffered. At elevated load, end gas autoignition became a problem.

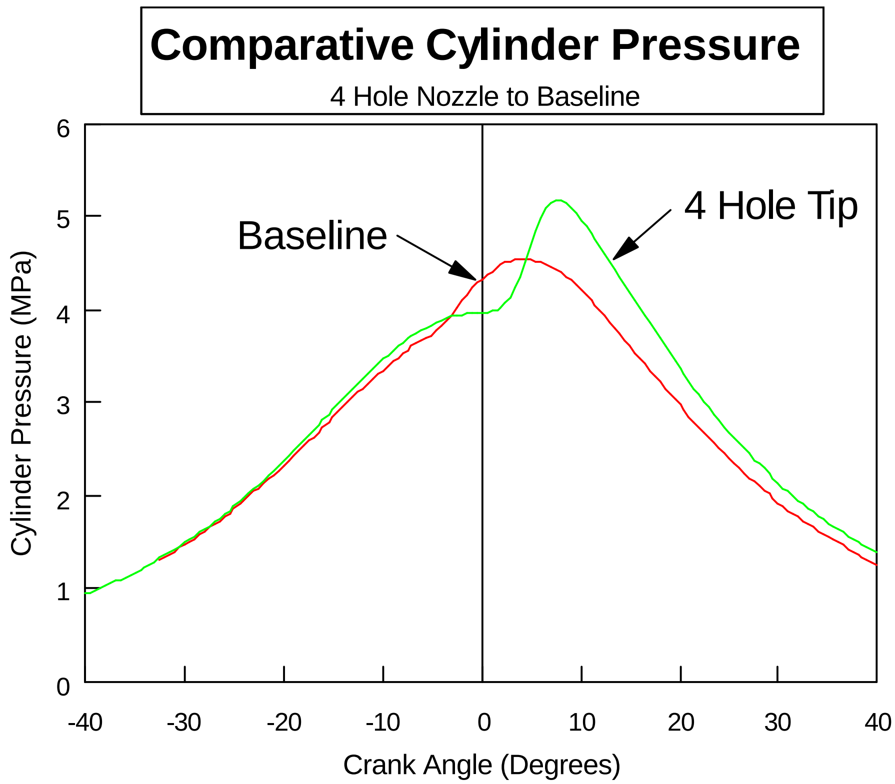


Figure 27 - Cylinder Pressure with Baseline and 4 Hole Nozzles (3501 DING)

The 10 hole nozzle cured the smoke problem that was caused by the center hole. There was approximately a 10% penalty in light load performance by removing the center hole. However, this was later overcome with split injection which allowed smoke free, efficient operation across the entire load and speed range. The NOx efficiency tradeoff was not measurably different than previously reported.

A second 10 hole nozzle was tested with a 70° angle. The angle was decreased from 74° (see figure 28) as a result of the CFD modeling. The CFD simulation with the 74° angle predicted that the gas jets would curve and attach to the cylinder head as a result of a low pressure zone created between the head and gas jet. This attachment can cause poor air utilization and flame quenching, which can lead to increased CO emissions as well as an increased COV. Test results with the 70º angle nozzle have indicated up to a 50% reduction in HC emissions and 20% reduction in CO emissions at low loads. Figure 29 shows the concentrations predicted by CFD modeling with the redesigned 70º nozzle. The model provides information that shows areas that are within the flammability limits of natural gas.

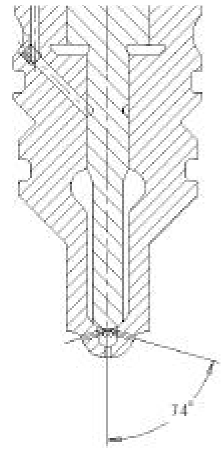


Figure 28 - Orifice spray angle

The asymmetric nozzle reduced COV as well as increased the initial rate of heat release. This nozzle was designed to be used with the offset elliptical bowl piston described later. However, the nozzle was originally tested with the standard piston. COV was reduced approximately 10%. There was no measurable change in the NOx-efficiency tradeoff. Particulate data was taken with this nozzle to verify the low particulate levels of the DING engine. At peak torque with no EGR, the engine produced .033 g/hp-hr with a BSNOx of 6.2 g/hp-hr at 42.6% thermal efficiency. Retarding timing lead to a BSNOx of 4 g/hp-hr and a .036 g/hp-hr particulate level at 39.5% thermal efficiency. For all points tested, excluding idle, particulates were in the range of .02 to .04 g/hp-hr. It is suspected much of this particulate matter was coming from oil being injected into the cylinder. This problem has been addressed in the new 3126 DING injector.

To enhance mixing and increase the burn rate, particularly when EGR was being used, a piston that could increase the air motion was desired. A squish area ratio of approximately .5 and a piston-to-head clearance of 1.5 mm was needed to achieve air velocities of the same magnitude as the spray velocities near the edge of the piston crater. Figure 30 compares the gas jet velocities to the squish velocities near the edge of the piston crater.

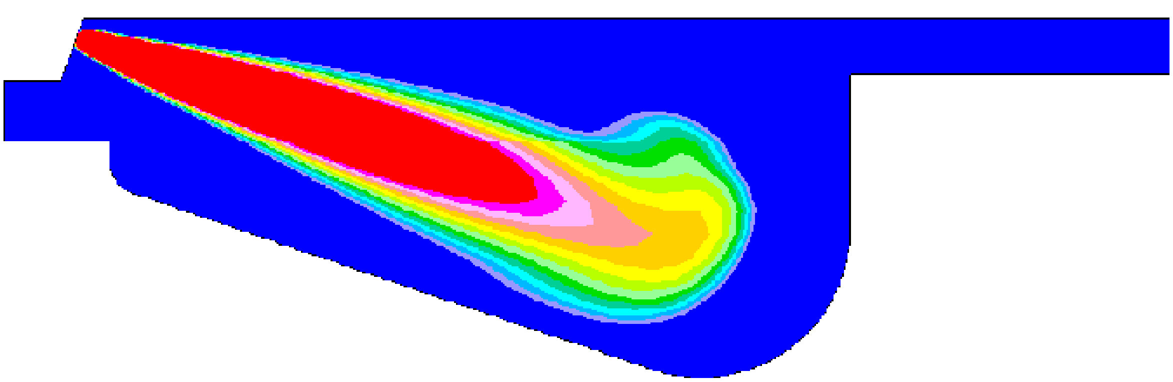


Figure 29 - Concentration Plot of 10 Hole Nozzle with 20 Degree Angle (3501 DING)

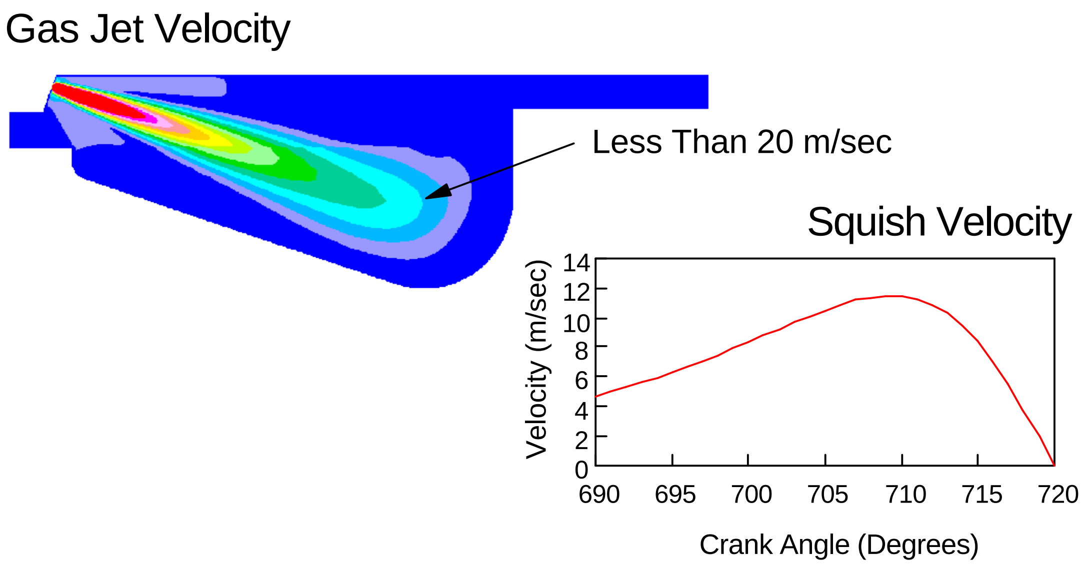


Figure 30 - Comparison of Gas Jet Velocities to Squish Velocities (3501 DING)

With the present 3500 camshaft the 1.5 mm clearance required machining valve pockets into the piston top. Also, CFD results indicated that the center gas jet was attaching to the piston face and not spreading as desired. Engine testing indicated that there was a smoke/particulate tradeoff associated with using the center hole. The center hole was originally designed to be a means of helping the flame propagate from one side to the other. However, CFD along with test results, indicated that the main advantage of the center hole was that it helped provide a stagnated, combustible mixture at the glow plug surface earlier, which reduced ignition delay. For this reason it was desired to keep the center spray but stop the resulting attachment. Several different piston impingement geometries were modeled with CFD. A sharp edge that could break the attachment was a part of each of these geometries. The final geometry chosen was a pin that was part of the piston. The pin diameter and height were key parameters in allowing the center spray to be used without attachment to the piston. Other geometries were predicted to be capable of stopping attachment to the piston. However, these other geometries resulted in a jet that attached to the cylinder head surface instead . To determine the effects of the increased squish and bowl geometry, the compression ratio was maintained at 15.5:1. Also incorporated into the piston was an elliptical bowl that was offset to better center combustion around the glow plug. This piston was designed to be used in conjunction with the asymmetric nozzle that had larger orifices that straddled the glow plug. Figure 31 shows the resulting design of the piston.

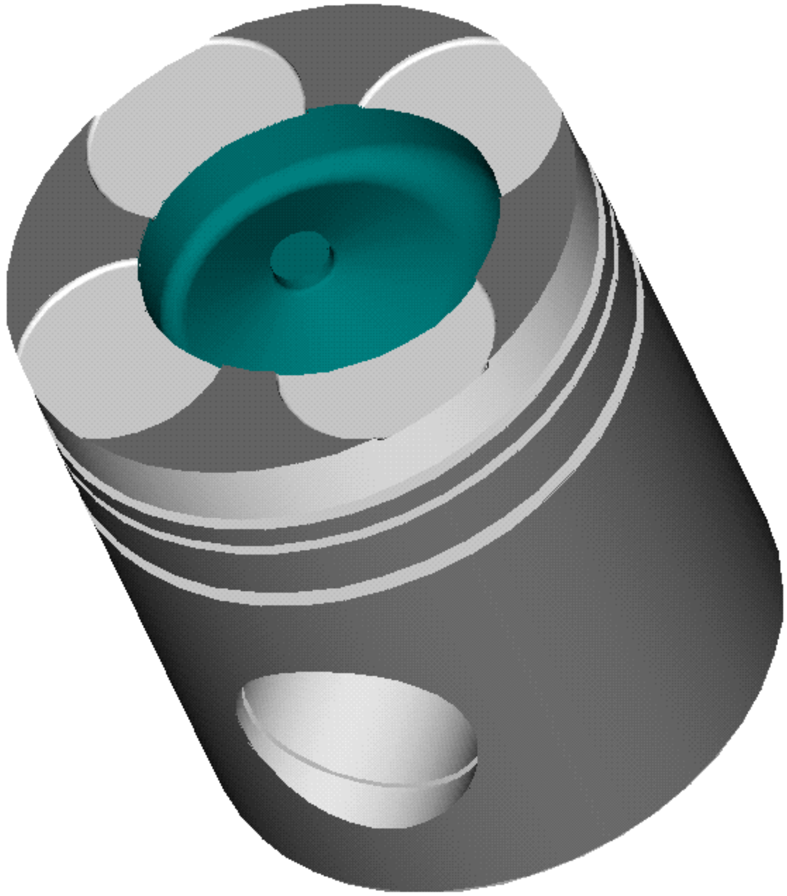


Figure 31 - Piston with Offset Elliptical Bowl (3501 DING)

COV was reduced as much as 20% at air fuel ratios that were richer than 26:1. Performance and NOx at points leaner than this did not show any significant improvement. Light load performance with split injection was reduced slightly with this piston. This was presumably due to the increase in mixing that allowed more fuel to be mixed below the flammability limit. This was partially verified with an increased measure of THC. Comparing results from the 15.5:1 Mexican hat piston bowl to this design indicates there is an advantage to having the increased air motion associated with the squish region. It is suspected that the offset bowl had a minor impact due to the limited offset. The initial 3126 DING build will use 16:1 CR pistons with a similar squish ratio.

A cylinder head with the glow plug located closer to the injector was procured and tested. The closer glow plug decreases the physical time required for the gas jet to reach the glow plug. This can reduce the ignition delay which would help improve COV, light load efficiency, and hydrocarbons. A slight reduction in ignition delay was measured with the new cylinder head. However, glow plug power had increased significantly due to the higher gas velocities by the plug. After comparing the benefit of a slightly shorter ignition delay (.5^0 to 1^0) to the decrease in expected glow plug life, it was decided to return to the original cylinder head to complete the rest of the test plans. The closer glow plug could be better optimized with an injector nozzle with a larger separation angle between the jets that straddle the plug. However, this was not available for testing at this time. Therefore, the cylinder head was removed and testing was continued with the original cylinder head.

Split injection has proven to reduce noise and NOx emissions in diesel fueled engines. This is due to a reduction in the amount of fuel that is premixed and autoignites, which would not be a diffusion controlled burn. The DING engine does not have this rapid initial heat release, even with its longer ignition delay. This is due to the fact that the mixture is ignited at a point source and the flame must propagate. However, it was thought that a split injection could reduce the amount of fuel in the cylinder before the start of combustion, which in turn, would reduce CO, HC, and COV, and would help the NOx - efficiency trade off. Previously, a mechanical split injection was built into the original gas injector. Those early test results indicated there was no benefit in using the split injection. However, this split was very fast, similar to a diesel split. There was not sufficient time for a significant amount of gas to form a combustible mixture near the glow plug or to allow sufficient residence time to ignite the fuel. It was decided that an electronic split injection may provide better results than the previous mechanical split. The electronic split is simply a modification to the controller software. This modification allows a variable pilot quantity of fuel to be injected into the cylinder and pauses a variable amount of time before the controller injects the remaining amount of fuel that is needed to maintain engine speed. This was very successful at reducing the light load fuel rate and hydrocarbon problem at low and high speeds. In fact, the DING engine was able to closely match the diesel efficiency across the entire load range. This was done without reducing air flow or skip firing, as was done in previous attempts at matching the diesel efficiency. Table 5 shows two points tested with and without the split injection.

	Speed (RPM)	Load (%)	Thermal Efficiency (%)	BSNOx (g/hp-hr)
No Split	700	10%	20%	5.3
Split	700	10%	26.3%	5.6
No Split	1,750	25%	26.2%	4
Split	1,750	25%	32.1%	3.9
Diesel	1,750	25%	32%	-

Table 5 - Low Load Efficiency with Split Injection (3501 DING)

Figure 32 graphically shows the dramatic reduction in THC emissions when utilizing the split injection at light loads. As can be seen in the figure the unburned hydrocarbons were reduced by a factor of three or more when split injection was activated. Utilizing the 10 hole 20° injector nozzle with split injection further reduced HC and CO. The DING engine still has higher HC emissions than the diesel but is now significantly closer. At higher loads the DING engine THC emissions are typically in the 75-450 PPM range. A similar diesel fueled engine typically produces 50-150 PPM THC. For clarity the PPM THC measurements were converted to a mass flow rate and labeled above each bar. The impact on thermal efficiency is pointed out below the chart. Figure 33 shows the required logic pulse for split injection operation and the resulting measured check lift. Since the split is done electronically it can be mapped to only function at light loads where needed.

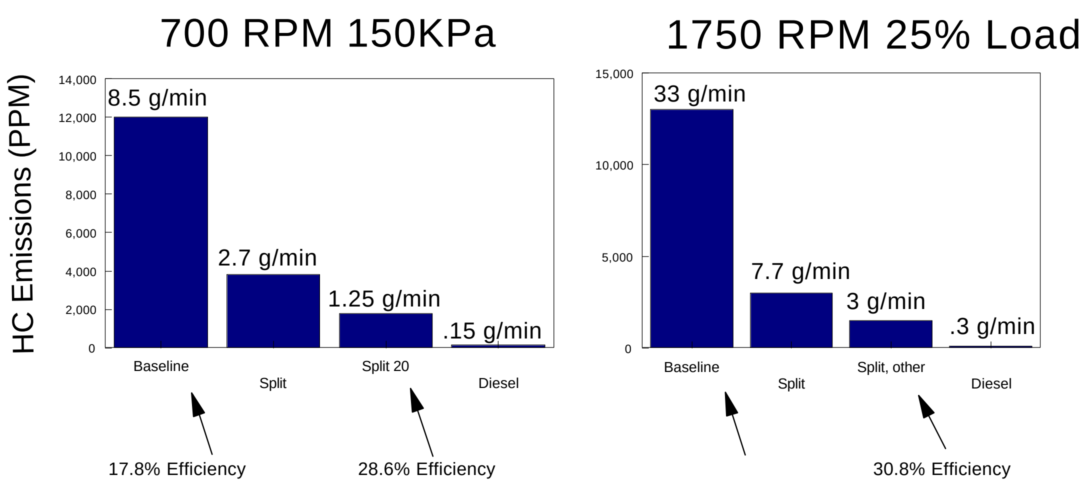


Figure 32 - Chart comparing HC emissions with and without split injection at light loads

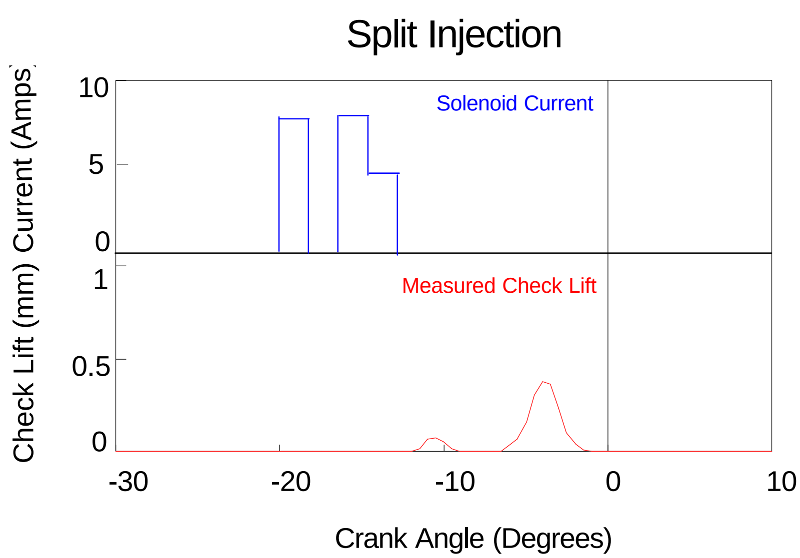


Figure 33 - Plot of injector logic pulse and resulting measured check lift (3501 DING)

Select points were taken from the final data ran using the offset bowl piston, cooled EGR, split injection at light load, and the 7-2 asymmetric nozzle. This data is shown below in Table 6. The data shows the effects of injection timing and amount of EGR on emissions and other performance parameters. Points indicated in bold font lead to an 8 mode cycle BSNOx of 2.48 g/hp-hr at 37.8% thermal efficiency. Once again, no EGR was used at light load points to maintain peak efficiency.

Mode	Speed (RPM)	Power (KW)	Inj. Timing (BTDC)	Inj. Pressure (PSIG)	Fuel Rate (G/min)	A/F	EGR (%)	Int. (C)	Exh. (C)	Th. Eff. (%)	BSCO2 (G/hp-hr)	BSCO (G/hp-hr)	BSNOx (G/hp-hr)	BSHC (G/hp-hr)
1	**700**	**3.6**	**8 Split**	**1500**	**16.2**	**89.5**	**0**	**44**		**27.8**	**468.8**	**7.12**	**5.92**	**14.94**
2	1300	40.1	16	2250	125.7	37.4	0	41	354	40.45	361.8	2.54	6.09	4.53
2	1300	40.2	16	2250	128.9	31.1	8.48	49	394	39.5	359.9	2.97	3.62	4.61
2	1300	40.2	16	2250	133.5	27.75	14.11	50	416	38.3	366.9	4.08	1.82	4.93
2	**1300**	**40.3**	**18**	**2250**	**129.1**	**28.5**	**14.34**	**51**	**399**	**39.5**	**355.4**	**3.17**	**2.34**	**4.46**
3	1300	59.7	18	2500	175.9	28.8	1.99	43	450	43	344.7	3.3	7.69	0.32
3	1300	59.8	18	2500	181.2	26.1	8.37	50	469	41.8	349.5	5	4.61	0.51
3	1300	59.8	18	2500	186	24.6	11.05	53	489	40.7	356.4	7.49	3.13	0.7
3	**1300**	**60**	**18**	**2500**	**198**	**22.5**	**12.78**	**54**	**518**	**38.4**	**366.5**	**14.16**	**1.79**	**0.79**
4	1300	78.3	16	2850	248	23.5	5.07	53	565	40	357.7	11.34	4.02	0.13
4	**1300**	**77.6**	**16**	**2850**	**265**	**20.9**	**9.26**	**60**	**609**	**37.1**	**380.1**	**15.3**	**1.75**	**0.35**
5	**1750**	**23**	**35 Split**	**1800**	**90**	**36.8**	**0**	**45**		**31.9**	**365.9**	**5.52**	**4.56**	**14.03**
6	1750	46.8	25	2000	153.4	38.5	0	42	371	38.8	361.8	3.63	4.63	6.67
6	1750	46.6	25	2000	157	33.3	7.78	51	405	37.8	371.9	4.31	2.55	7.39
6	**1750**	**46.8**	**25**	**2000**	**166.1**	**29.6**	**11.77**	**52**	**427**	**35.8**	**384.4**	**4.97**	**1.67**	**7.72**
6	1750	46.7	28	2000	154.7	31.82	10.97	50	393	38.4	361.4	3.66	3.11	6.65
7	1750	66	26	2500	199.6	32	2.24	43	431	42.1	352.3	2.24	6.26	1.62
7	1750	65.6	26	2500	209.5	29.4	8.98	49	446	39.8	365	3.28	3.59	2.68
7	**1750**	**65.8**	**26**	**2500**	**212.8**	**28**	**10.77**	**53**	**459**	**39.4**	**364.7**	**3.86**	**2.79**	**2.95**
8	1750	90.4	25	2850	271.6	30	1.31	43	477	42.3	341.7	3.21	6.62	0.42
8	1750	90.2	25	2850	278.8	28.3	5.64	53	489	41.2	346.2	3.95	4.3	0.89
8	**1750**	**90**	**26**	**2850**	**282.8**	**26.9**	**9.56**	**53**	**509**	**40.5**	**355.1**	**5.64**	**2.98**	**1.01**
8	1750	90.1	27	2850	323	22.9	13.25	54	559	35.5	391.2	17.8	0.85	1.73

Table 6 - Performance and Emissions (3501 DING)

In summary EGR is required to efficiently achieve the 2.5 g/hp-hr NOx goal. The asymmetric nozzle increased the initial heat release rate and increased combustion stability. Removing the center hole cured the smoke problem at retarded injection timings. Split injection allowed the DING engine to match diesel efficiency at all loads that could be tested. A piston with a deeper bowl and squish region improved combustion stability and increased the rate of heat release. The port injection concept did not demonstrate the ability to reduce NOx below standard DI operation.

Bench Tests of Selective Catalyst Reduction to Achieve NOx < 1 g/hp-hr.

Selective catalytic reduction (SCR) is another means of reducing NOx emissions in an exhaust stream containing oxygen. Selective catalytic reduction requires injecting a fuel into the catalyst bed where a surface reaction occurs converting fuel, oxygen, and NO to N_2, CO_2, and H_2O. Ammonia and ethanol are common fuels to use as a reductant. To eliminate the need to carry two fuels, a catalyst that is reactive with the simpler CH_4 hydrocarbon is being sought. Several catalyst materials have been tested with methane but did not show a significant conversion efficiency. To demonstrate SCR catalyst capabilities, a catalyst sample was tested using ethanol as a reductant with an exhaust stream that matched the concentrations of the DING engine's exhaust. Conversion efficiencies were above 90%, for almost all testing performed at exhaust temperatures between 350C and 450C. Part of this high conversion efficiency is due to the exhaust stream not containing sulfur, a known catalyst poison. Another part of the high conversion efficiency was due to the formation of nitrogen-containing intermediate gases that are not measured as NOx. Since these emissions are undesirable as well, follow-on bench tests were conducted incorporating an oxidation catalyst after the SCR catalyst to ensure NOx was being catalyzed to pure nitrogen. The addition of the oxidizing catalyst reduced conversion efficiency from the original 90% to approximately 80%, which is still very good.

An experiment was set up to evaluate catalyst performance at various engine operating points as well as to gain quantitative measurements of what the effects the different exhaust gas species have on conversion efficiency. Throughout this testing the reductant ratio was maintained at 3:1, while the space velocity was maintained at 50,000/hr. Figure 34 shows the effect increased water concentration has on conversion efficiency. Water in the exhaust increases when the air fuel ratio becomes richer. This can happen as a result of a decrease in air flow due to utilizing EGR or a decrease in efficiency due to retarding injection timing. Figure 35 demonstrates the effect of decreasing the initial NOx (ppm)/exhaust (ppm) ratio. Also shown on Figure 35 is the resulting decrease in conversion efficiency found when using the oxidizing catalyst. Even when using the oxidizing catalyst conversion efficiencies were as high as 90%. Figures 32 and 33 show the effects of reducing the amount of oxygen and carbon monoxide in the exhaust stream. Neither CO or O_2 had a large impact on conversion efficiency.

The resulting conversion efficiency of three different fuel/NOx (F/NO) ratios are shown in Figure 38. In all the following cases, the fuel used was ethanol. Performance at an F/NO ratio of 1 is very poor (<45%) over the entire temperature range. At this ratio, there is insufficient ethanol reductant present in the exhaust stream to react with all the NOx present. As the ratio is increased to 2, the NOx conversion performance improves dramatically with a high of 89% conversion at 350° C. As is typical for this catalyst, performance decreases with increasing temperature. Increasing the F/NO ratio from 2 to 3 yields a slight improvement in the NOx conversion efficiency (3% at 350° C increasing up to 8.5% at 520° C). However, this is only a marginal improvement in NOx conversion for the extra 50% of ethanol that is being consumed. Engine testing is needed to arrive at the optimum amount of ethanol that should be added to achieve the appropriate balance between NOx conversion performance and operating cost.

The effect of F/NO ratio on the performance of the SDS-16 catalyst system at a space velocity of 100,000 hr^{-1} is shown in Figure 39. The same performance trends are present as were observed for the 50,000 hr^{-1} space velocity tests. However, the improvement in NOx conversion performance seen when the ratio is increased from 2 to 3 is greater at 100,000 hr^{-1} than 50,000 hr^{-1}. This trend is likely due to the deNOx reaction being diffusion limited and therefore at the higher space velocity (i.e. shorter residence time) the additional ethanol provides a greater opportunity for the species to arrive at the catalyst sites.

Figure 40 shows the effect of both space velocity and SO_2 presence on the performance of the SDS-16 deNOx catalyst system. All tests were run at an F/NO of 3. In comparing the performance at the two space velocities, it appears that there is slight improvement (4-6%) in the NOx conversion performance at the lower temperatures (350° and 400° C) for the 50,000 hr^{-1} test. This is understandable since the longer residence time results in an improved chance for the NOx to nitrogen reaction to occur. This trend, however, disappears and then reverses itself as the temperature increases to 450° and 520° C respectively. At the higher temperatures there is a greater tendency for the SDS-16 catalyst to form the nitrogen containing intermediate species. At the lower space velocity, there is a greater quantity of intermediates formed due to the increased residence time. These intermediate species are subsequently re-oxidized back to NOx by the downstream oxidation catalyst and hence are reflected in the overall poorer NOx conversion performance. The presence of SO_2 in the gas stream results in a general reduction of the NOx conversion performance of the SDS-16 DeNOx catalyst system. SO_2 tends to adsorb onto the catalyst sites at lower temperatures (<450° C) and thus inhibit the NOx conversion reaction. At higher temperatures (>450° C) the SO_2 is desorbed and thus the catalyst performs the same as if there were no SO_2 present. The presence of the SO_2 resulted in an overall flattening of the NOx conversion performance to a value of approximately 70 % over the temperature range of 350°-520° C. This test does not reflect how the catalyst will perform with exposure to SO_2 for extended times. Often there can be permanent catalyst degradation due to long term exposure to SO_2. A value of 15 ppm SO_2 may be an unrealistically high simulation of what would actually be present in engine exhaust since lubricating oil is the only source of SO_2 for the DING engine. However, 15 ppm was the lowest achievable limit with the current bench test setup and at the very least represents a "worst case" scenario.

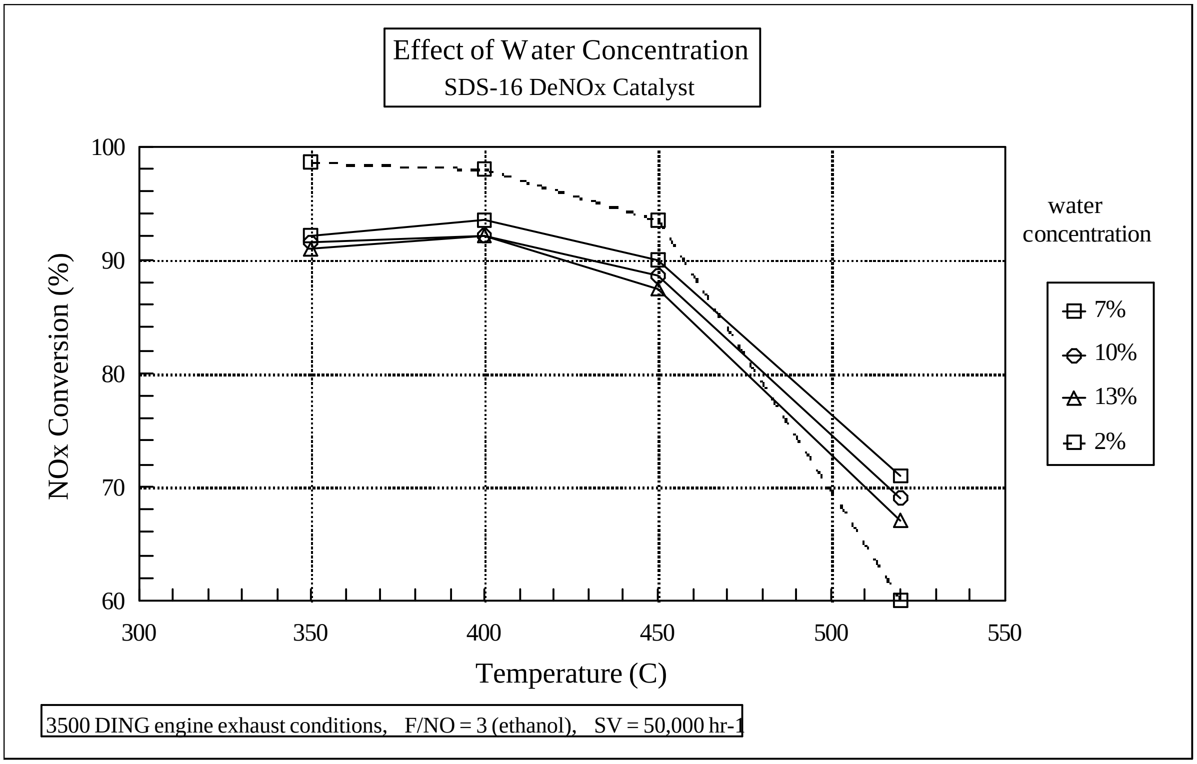


Figure 34 - Effect of H2O on Conversion Efficiency (Bench test data)

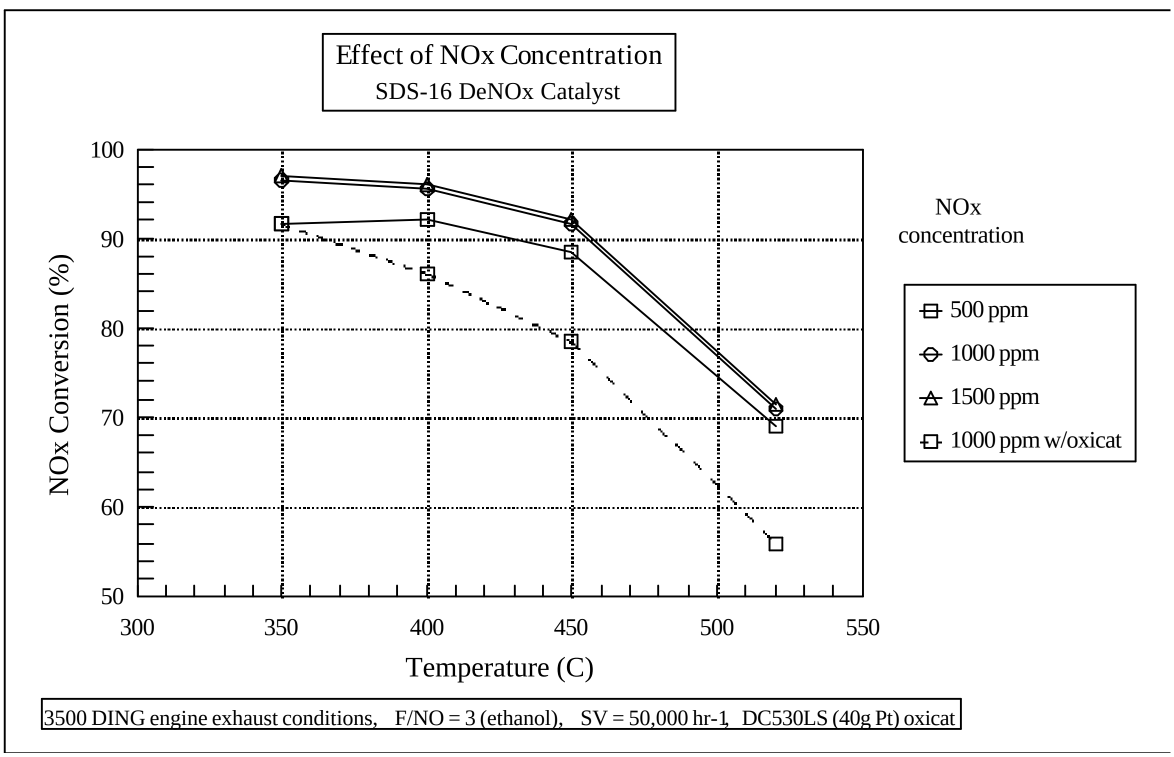


Figure 35 - Effect of NOx Concentration on Conversion Efficiency (Bench test data)

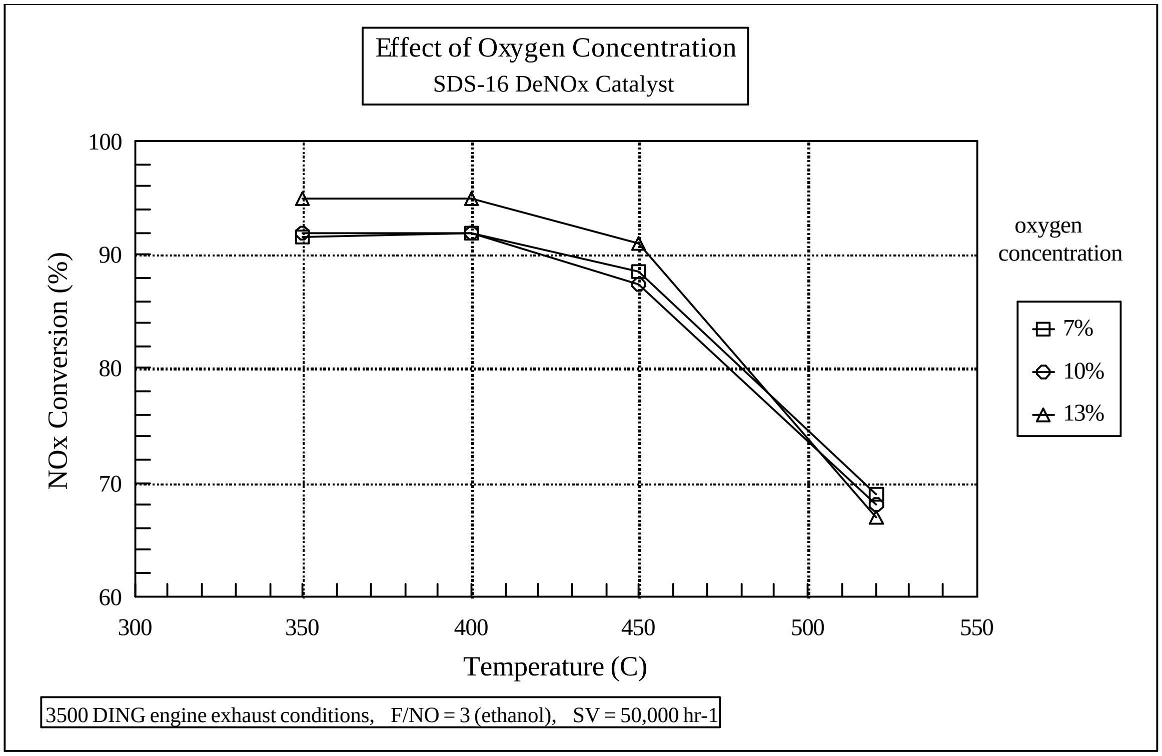


Figure 36 - Effect of O2 on Conversion Efficiency (Bench test data)

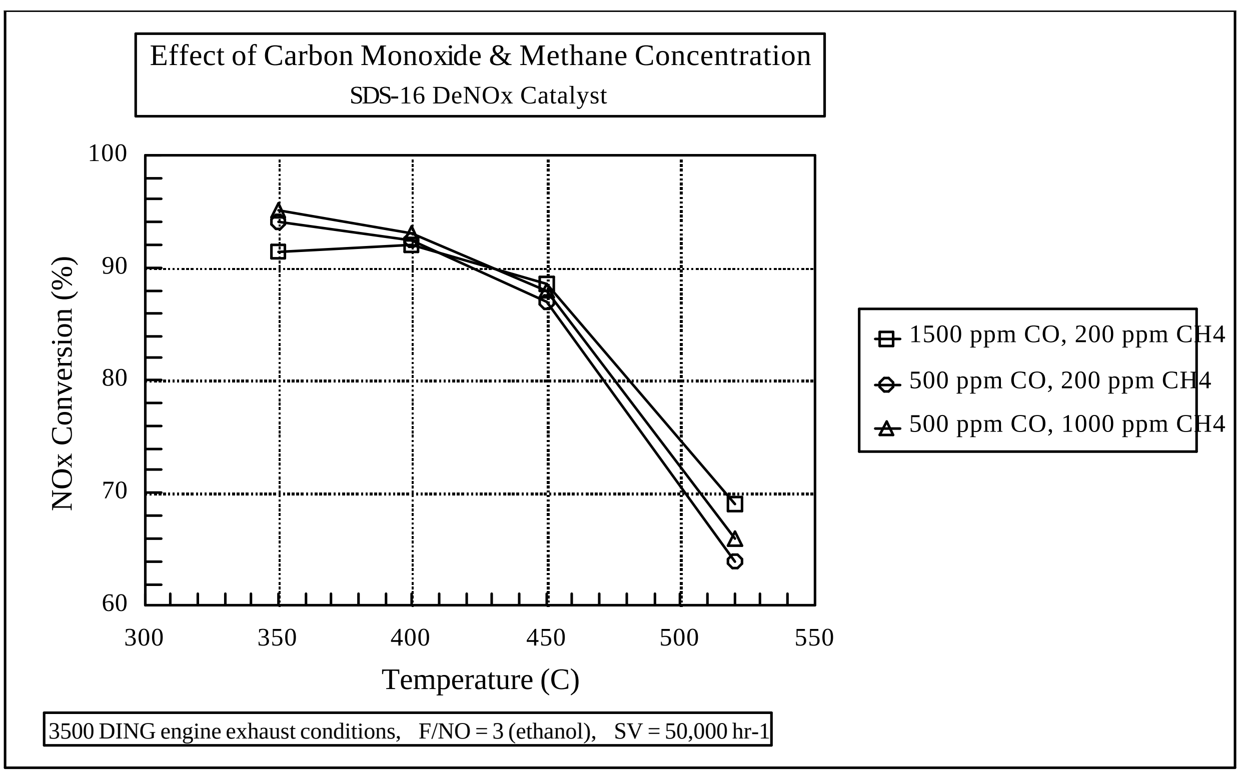


Figure 37 - Effect of CO on Conversion Efficiency (Bench test data)

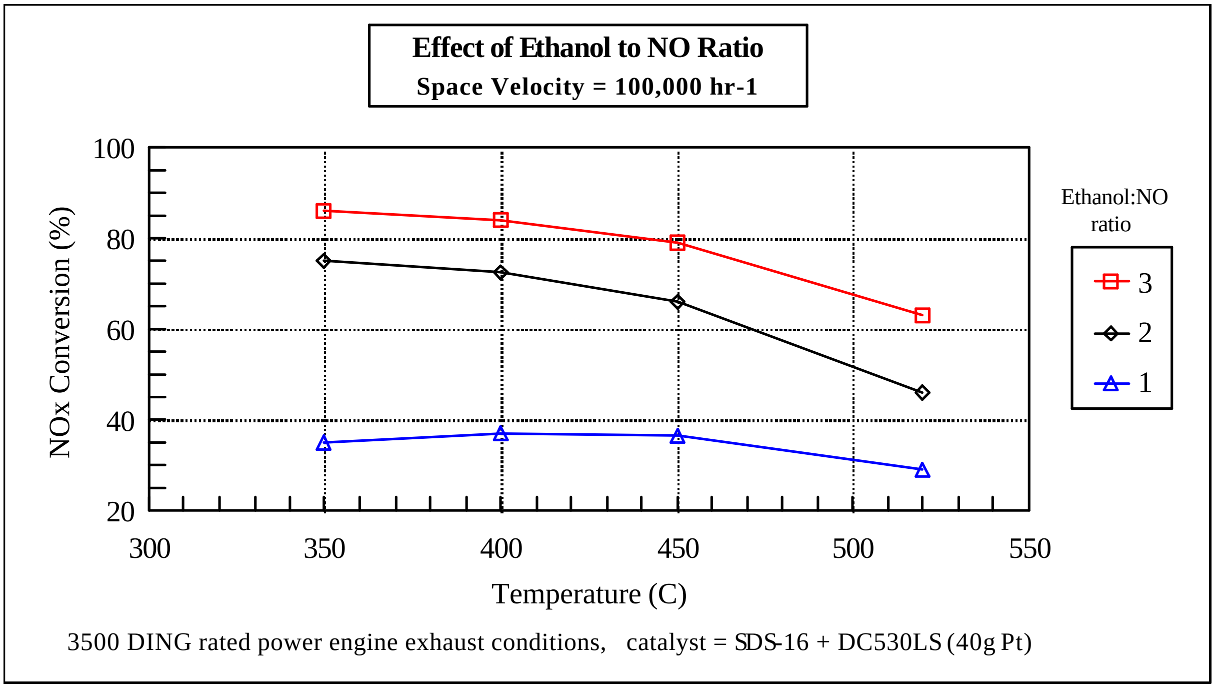


Figure 38 - Effect of ethanol-to-NOx ratio on NOx conversion efficiency
(Space velocity of 100,000 hr-1) (Bench test data)

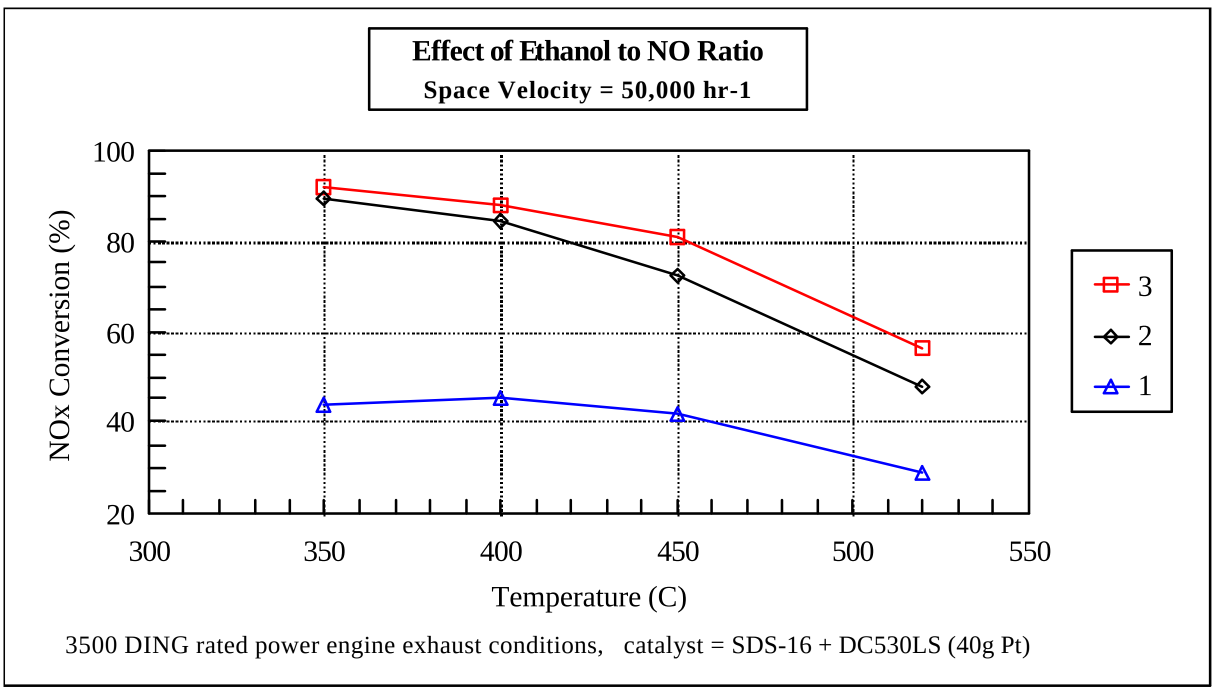


Figure 39 - Effect of ethanol to NOx ratio on NOx conversion efficiency
(Space velocity of 50,000 hr-1) (Bench test data)

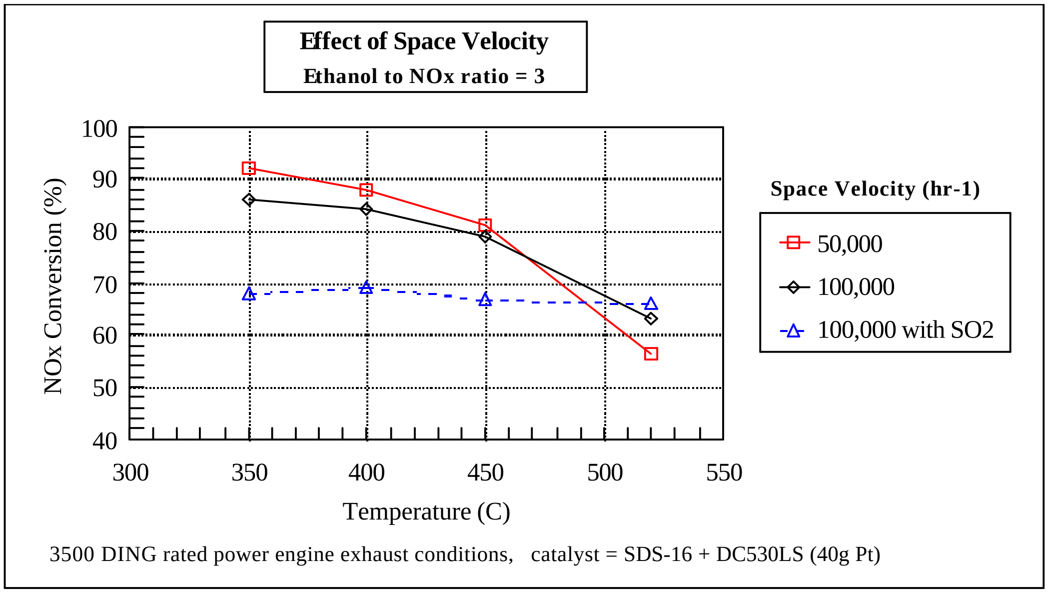


Figure 40 - Effect of space velocity and SO2 presence on NOx conversion efficiency at an F/NO ratio of 3 (Bench test data)

Combining 3501 DING engine NOx - efficiency data with bench test conversion efficiency data Figure 41 was created to show the potential of an SCR system to reduce NOx to below 1 gm/hp-hr, with and without EGR. Conversion efficiencies from tests were run with a F/NO of 3:1 and no sulfur in the exhaust. Figure 42, demonstrates a worst case scenario including 15 PPM SO2 in the DING engines exhaust due to lube oil being burned. BSNOx emissions of less than 1 g/hp-hr were still achievable if EGR was used in conjunction with the catalyst.

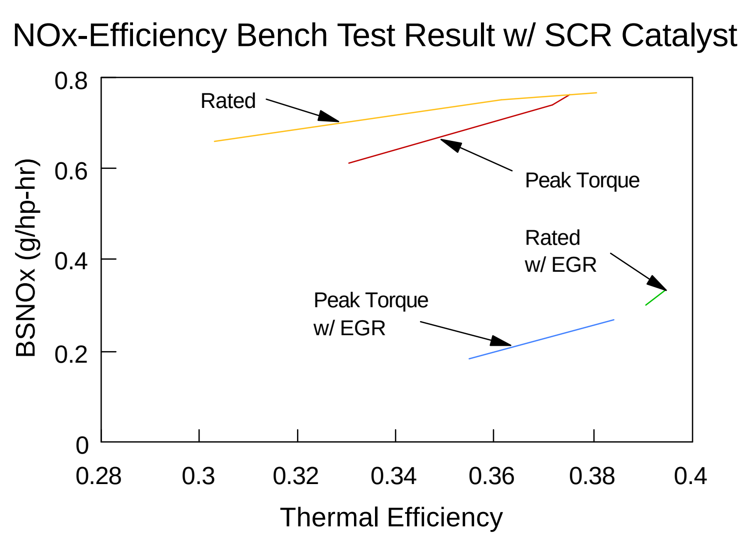


Figure 41 - Calculated NOx Emissions and Resulting Efficiency

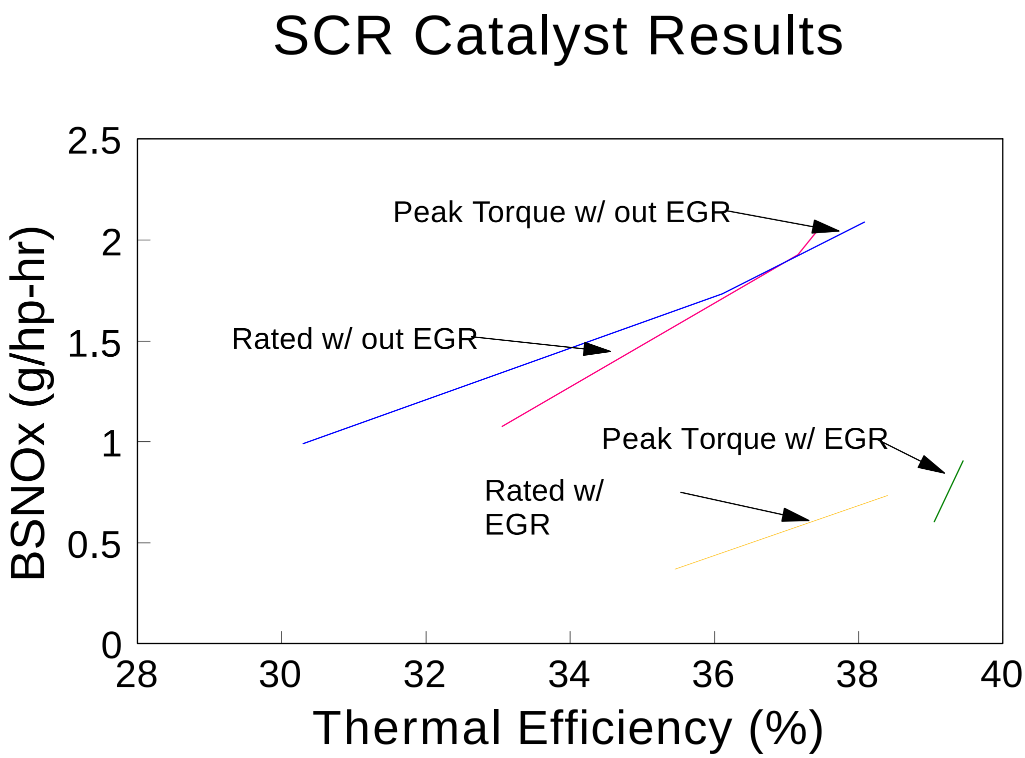


Figure 42 - Calculated NOx Emissions with SO2 in Exhaust

Summary

SCR bench test results show conversion efficiencies of approximately 70% can be expected Using the conversion efficiencies found in the bench testing, BSNOx of less than 1 g/hp-hr is achievable on the DING engine. The catalyst life on a DING engine should prove to be longer than that of a comparable diesel engine, due to the lower sulfur content in the DING exhaust gas. Catalysts have been ordered for the 3126 DING engine.

Task 3: Durability Development of 3000 psi Fuel Handling System

Sub task 3.1: Demonstrate 3000 psi LNG Pump State-of-the Art

Objective:

In this sub task the subcontractor shall be responsible for demonstrating currently-available 3000 psi LNG pumps to determine their suitability for DING engine applications. These currently-available LNG pumps have not been utilized in vehicular applications. The pumps will be tested to see if they can handle the continuous operation through transient conditions that is required of a vehicular pump. The subcontractor shall investigate the different options, such as submersing the entire pump/motor assembly in the fuel tank and determine the best design for bench performance evaluation.

Accomplishment Summary:

The planned 3000 psi LNG pump development that was planned in Phase 2 was not performed due to the inability of the subcontractor (MVE) to perform the work in this time frame. A report on the state-of -the-art of high-pressure positive displacement cryogenic pumps was performed This includes a history of cryogenic pump designs. Also included are some of the parameters which drive the design of the cryogenic pump.

Accomplishment Details

The following pages were taken from a report created by MVE, the subcontractor for this sub task.

Principles

High-pressure cryogenic pumps are positive displacement pumps, used to pump a liquid cryogen to high pressures, force it through a heater, and then use the resulting gas at high pressures. Typical applications are to fill high-pressure storage systems, such as cylinders and storage tubes. In other cases, the gas is used directly in high-pressure autoclaves, to fracture rock formations in oil wells, etc. High-pressure pumps will pump a variety of cryogenic fluids – from Hydrogen at -432 Deg F to Carbon dioxide at 0 Deg F. Obviously, each fluid has its particular problems; oxygen requires special cleaning and materials, hydrogen requires special insulation, and CO2 requires judicious selection of materials.

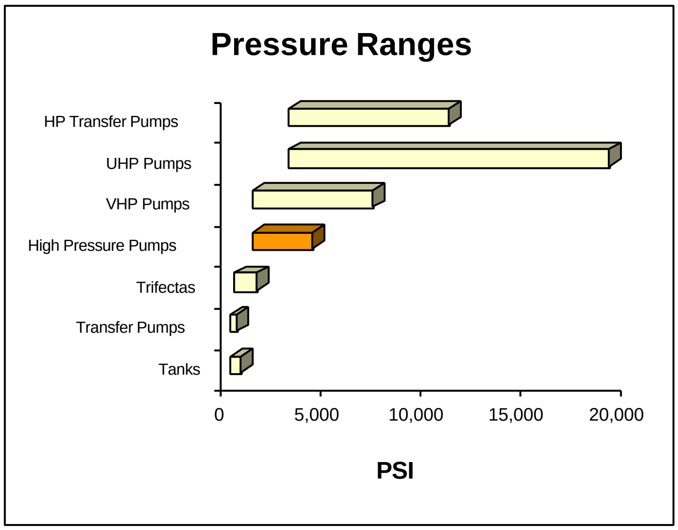


Figure 43

Pressure ranges for high-pressure pumps range from approximately 1,200 PSI to 16,000 PSI. HP Transfer Pumps and UHP pumps are prohibitively expensive for normal applications. With the exception of certain rocket engine pumps, high pressures are achieved through positive displacement pumps.

Positive Displacement

Positive displacement cryogenic pumps are – essentially – a variation of one design. Pumps have pistons with rings within a cylinder, inlet and outlet valves, and some mechanism to push the piston in and out of the cylinder. Most pumps have piston packing that prevents the cryogen from bleeding past the piston to the outside. Some recent pumps have intermediate compression strokes.

Pump output is a function of piston diameter, stroke, and speed (otherwise known as liquid displacement). Typically, pump manufacturers will standardize on a piston diameter and stroke, and vary the speed to displace more or less fluid over a given unit of time. There is a limited range of adjustment – too slow and too much product seeps by the rings – too fast and not enough fluid can enter the cylinder through the inlet valve.

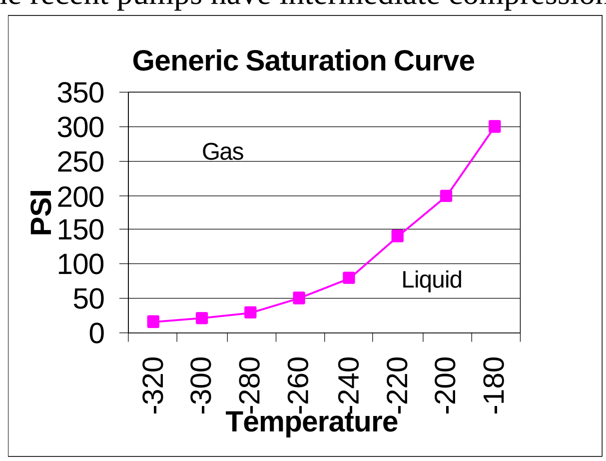


Figure 44

There are three basic challenges in positive displacement pumps for cryogens:

1. The fluid that is being pumped is almost always on the verge of boiling – "flashing" - to vapor.

2. Low lubricity of liquid cryogens – CO_2 even acts as a solvent and degreaser.

3. Dealing with atmospheric and ambient conditions while pumping at cryogenic temperatures.

Cryogenic fluids are in a state of equilibrium with pressure and temperature. If the pressure on a fluid is reduced, it will boil to cool until it is in equilibrium with the lowered pressure. Conversely, a cryogenic liquid will not boil until the temperature of the liquid is in equilibrium with the increased pressure.

The challenge in positive displacement cryogenic pumps is to prevent the fluid from flashing into vapor as it enters into the cylinder on the suction stroke. Subcooling[1], artificial pressure heads, and two stage pumps are used to ensure that the liquid does not flash into vapor. When flashing does occur, and the pump goes into cavitation, catastrophic damage to the drive train may occur.

Cryogenic liquid has a relatively low viscosity. This is typically a problem when dealing with wearing of parts such as packing and piston rings. Effective pump designs must address and deal with this issue with specific material selection that may provide greater resistance to wear and some added lubricity.

Since most cryogenic liquid boils well below the normal ambient temperature, any added heat due to operational conditions (such as pump cavitation or advance component wear) may prohibit the pump from performing in the normal manner. This condition, if left in a prolonged state, may provide aggressive wear and excessive damage to the pump.

US Manufacturers

The Predecessors

Cryogenic pumps have been in use for a considerable length of time. Initially, cryogenic pumps were built into the cold boxes of Air Separation Units (ASU), and the product was pumped out of the rectification column through a vaporizer into high-pressure cylinders. The liquid going through the vaporizer was used to help cool incoming air to make the operation of the ASU more efficient. Withdrawal from the column was controlled by the speed of the pump.

As distribution systems were set up, liquid was shipped from increasingly larger – and thus more efficient - ASU's to storage tanks. Major gas producers – AIRCO, Air Products, Liquid Carbonic, Linde, NCG, etc. developed versions of cryogenic pumps that could operate from storage vessels. Gradually, various designs were weeded out – largely due to cost. Current

[1] To bring a fluid below the saturation curve of the particular cryogen. It is done either by maintaining the same pressure and cooling the liquid, or maintaining the same temperature and raising the pressure - or a combination of both.

pump technology – though not mature – has not been very innovative. Advances typically are limited to small improvements in sealing materials.

Airco Cryogenics

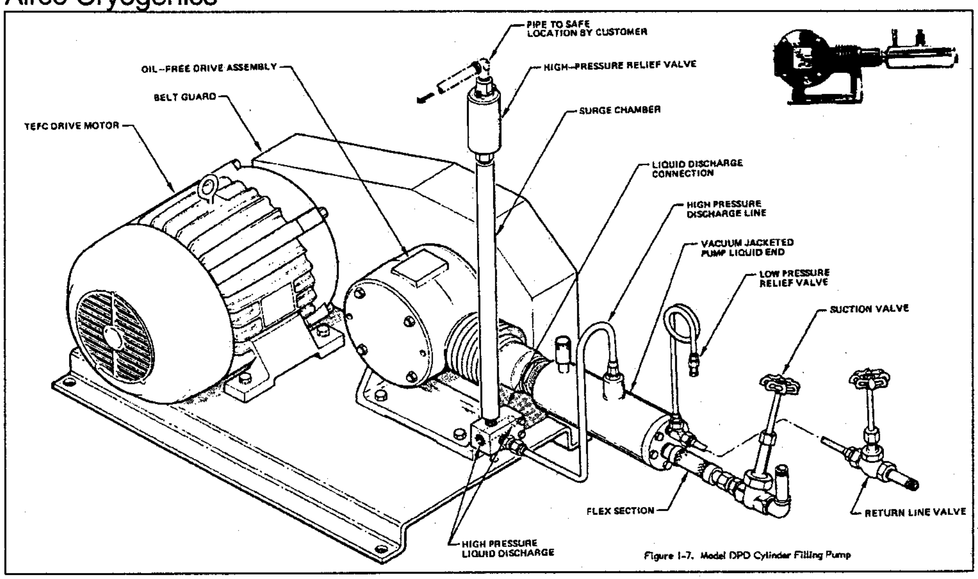


Figure 45

Airco Cryogenics was what could be considered the originator of the low cost positive displacement cylinder filling pump. Designed by Paul Duran, the pump was a small, very high piston speed pump. Piston shaft seals were "hat seals" made of Teflon™ gaskets supported by metal supports. Airco Cryogenics resulted in two spin-off companies –Cryo-Mech and CCI. AIRCO Cryogenics was later divested to become ACD. The European division was not divested, and now operates under the name of CryoStar. Reported MTBF's of the Paul pump is around 560 hours.

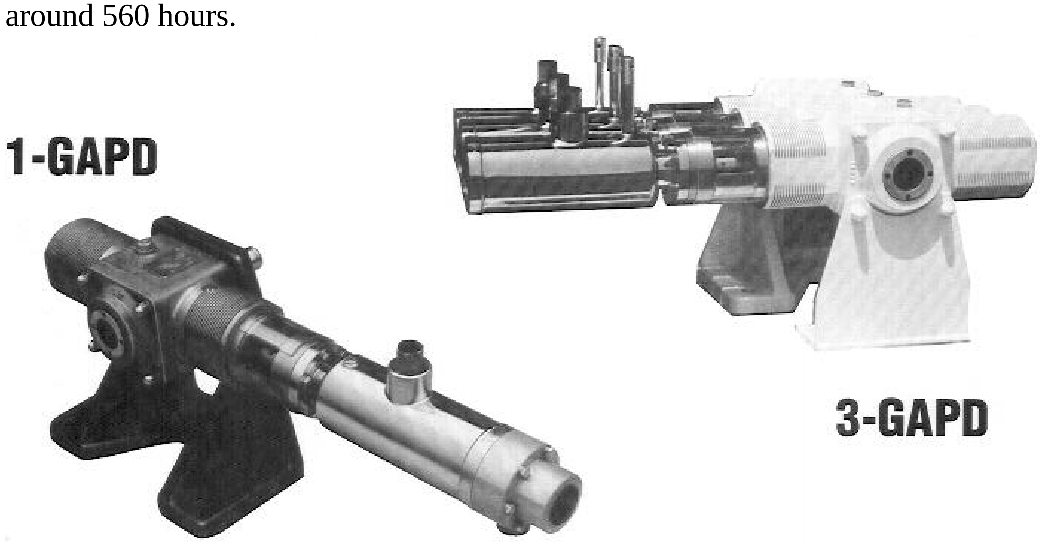


Figure 46

Liquid Carbonic / LouTyree

Liquid Carbonic's Lou Tyree, though extremely expensive, developed a particularly reliable pump. The Tyree pump was successful through careful material selection and appropriate tolerancing to make a pump that required virtually no sub-cool to operate. The large bulk of the pump – as well as the cost – made it difficult to market commercially. Reported MTBF's of 8000 to 9000 hours have been achieved.

Chemetron/National Cylinder Gas

Chemetron developed a very reliable slow pump. Highlights of this pump are the use of commercially available common chevron Teflon™ seals, as well as a very efficient 6 opening ball cage inlet valve. The pump had a relatively large insulated sump. Operation was at 150 and 300 RPM. Typically, very robust commercial drive ends were used to move the piston. Consistent MTBF's of 3,400 hours are achieved[2].

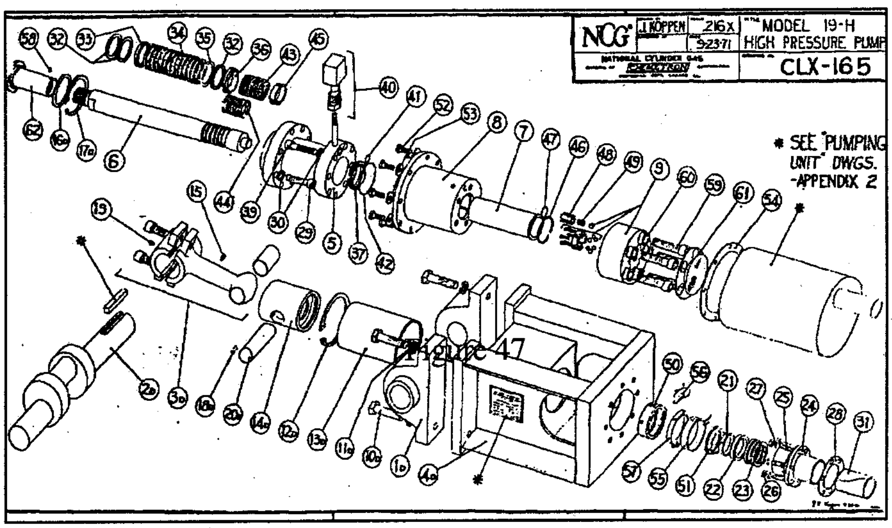


Figure 47

Union Carbide/Linde

Union Carbide developed several pump designs – vertical and inclined.

Current US Manufacturers

ACD

AIRCO Cryogenics, was divested by AIRCO (Now BOC) and continued in the pump business. They acquired all of Airco's designs, tooling, etc. Currently ACD is the largest manufacturer of cryogenic pumps in the US.

ACD has three models of pumps:

[2] Claus Emmer – Production Manager, Chemetron Venezuela – 1972 to 1978

The NDPD Family

Originally developed byCryoStar (Former sister company) the NDPD pump is a horizontal – grease lubricated drive or slightly inclined - pump with relatively small vacuum jacketed insulated sump. Advantages are a relatively low cost, low space requirements, and low liquid wastage. Problems with the pump are difficulties in achieving subcool, resulting in cavitation and excessive wear. Subcool is achieved entirely through appropriate external plumbing. Due to the different designs of cryogenic tanks, and all of the installation standards, sub-optimal piping is very common[3]. Particularly damaging to the NDPD pump is starting the pump against existing pressure. Seal design is based on the early Paul pump, and could be improved. NDPD pumps are typically used in cylinder filling applications, with pressure ranging from 2,200 to 2,800 PSI.

The GAPD Family

The GAPD series of pumps are more robust and larger versions of the NDPD pumps with oil lubricated drives. Operating problems are the same. These pumps are typically used in the process industry, where prolonged delivery of high-pressure gas is required. They are particularly suited for higher pressure pumping, and automatic systems. Even these pumps can be damaged by starting against pressure, and typically have anunloader valve that is open until the pump begins to operate.

The P-1600 Family

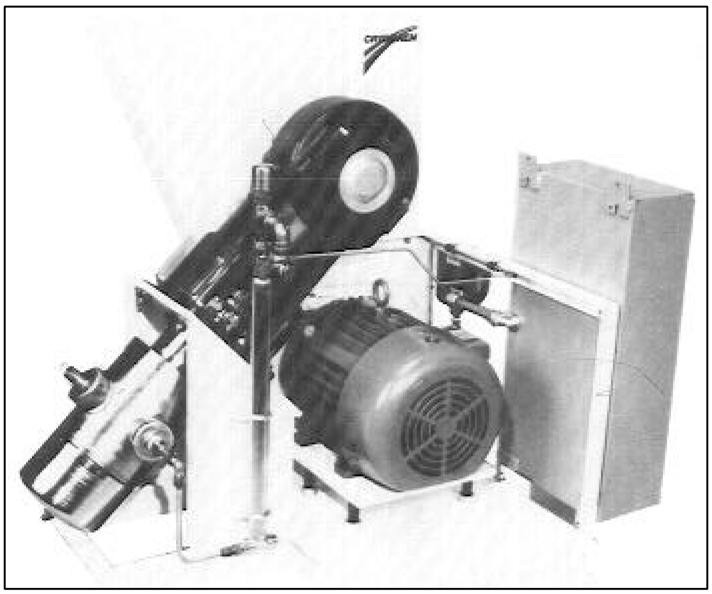


Figure 48

This series of pumps has a relatively deep, inclined sump. The depth of the sump is used to achieve additional subcool inside the pump, thus making piping to the pump less critical than with the NDPD or GAPD series. The sump is inclined for easier maintenance. Liquid flows into the sump at the lower part, and vapor flows out of the sump some distance above. Liquid losses are higher, since the liquid in the sump evaporates when the pump is stopped. Several versions and sizes exist.

The SZNDP Family

The "sub-zero" pump was developed by ACD in response to the difficulties in achieving sufficient subcool from conventional cryogenic vessels. The pump essentially is a double acting pump, with an initial compression stroke pressurizing the incoming liquid to a medium pressure, and then forcing it into the high-pressure chamber. Since the valve requirements in the initial suction side are for very low pressures, they can open very

[3] Often, installation decisions for cryogenic pumps are made by personnel who have not been trained in the effects of piping on subcool and pump performance. Cryogenic pump installations bear no resemblance to water pump installations.

easily, thus requiring very low NPSH. Further improvements are achieved by an additional vertical liquid conditioning sump on the inlet of cold end that allows warm liquid and gas to rise to the top. In effect, the small vertical sump performs the same function as the large sump on the inclined pumps.

CryoStar

CryoStar was the European division of Airco Cryogenics, and still remains associated with Airco (now BOC). CryoStar produces a version of the NDPD pump called the SDPD. CryoStar has continued to improve various areas of the pump such as: extending the packing life, improved suction design, and a more robust drive. CryoStar pioneered the combination of thermosiphon vessels and pumps in Europe, becoming the de-facto standard. In particular, CryoStar optimized cold end angles for maximum vapor removal.

CryoStar also manufacturers a line of larger, heavier duty reciprocating pumps (LOPD, LAPD). These pumps are designed for higher flows and pressures.

There is little outside difference between CryoStar and ACD base designs. The SDPD is the equivalent of the NDPD, the GDPD is the equivalent of the GAPD. Due to earlier adoption in Europe of the thermosiphon principle in cryogenic tanks designed for pumping, CryoStar has not actively marketed an equivalent of the P-1600 or SDPD series.

CVI

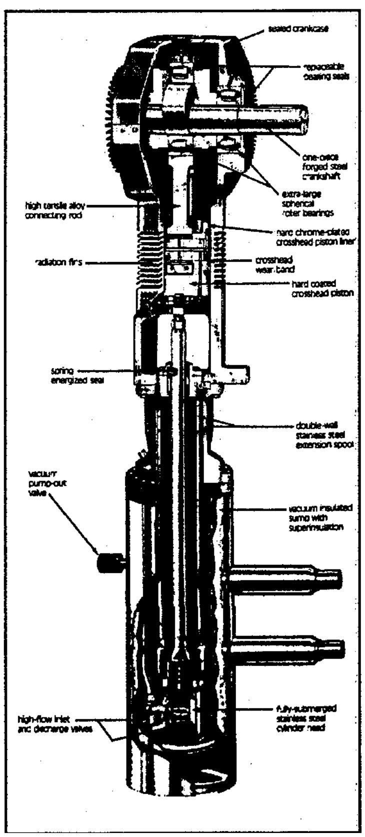


Figure 49

CVI has specialized in an inclined pump – the P-3000. This pump is considered by the industry to be extremely reliable.

One of the interesting design features of this pump is that the main shaft sealing is done by 2 spring energized Teflon™ seals, as compared to the multiple sets of hat seals. There is also a fairly long distance between the warm end and the cold end of the pump, ensuring that the bearing lubrication remains fluid.

CVI also manufactures a small narrow pump that can be immersed into a cylinder or a tank. It has been used extensively by the US Air Force for small, mobile, pumping units.

Woodland

Woodland manufactures a clone of the NDPD pump, called the WDPD. Woodland has developed a vertical pump – the WCP for rigorous applications. Maintenance is simplified by allowing the pump to be easily tipped for disassembly.

CCI

CCI manufactures a clone of the NDPD pump. CCI designed a smaller pump – the LXR-1000 was designed jointly with MVE Inc.

Pumps manufactured by CCI or MVE for the US Navy have proved extremely reliable, with a tested MTBF of 1,800 hours.

Pump Design Philosophy

Positive Displacement

Positive displacement pumps work by forcing liquid in and out of a cylinder. Pumps have pistons with rings within a cylinder, inlet and outlet valves, and some mechanism to push the piston in and out of the cylinder. The cylinder, piston, and valve assembly is usually inside an insulated sump containing the liquid being pumped. This sump serves to cool the cylinder and the piston. Cooling is required to remove heat generated by friction. The challenge of the positive displacement pump for cryogens is to prevent the liquid being pumped from boiling. Most pumps have piston packing that prevents the cryogen from bleeding past the piston to the outside. Other pump designs have variations of a labyrinth[4] or close tolerance seal to prevent excessive flow. Some recent pumps have intermediate compression strokes.

Cold End

As previously discussed, there are many challenges posed when pumping liquid cryogens. These challenges must be overcome by effective engineering designs that address and combat some of these elements. Some of these areas and associated engineering design solutions are the following:[5]

Vacuum Jacketed Cold-end – This feature helps assist the liquid from ambient temperature influence. A slight increase in liquid temperature will result in a boiling liquid preventing effective pumping.

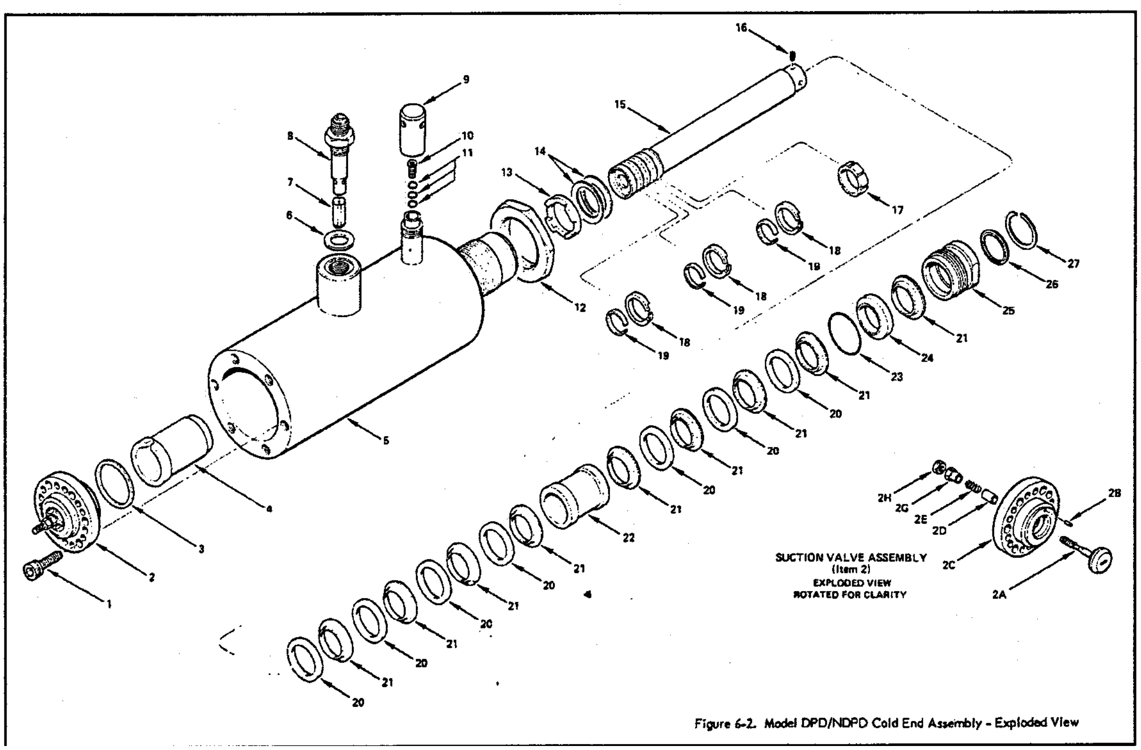


Figure 6-2. Model DPD/NDPD Cold End Assembly - Exploded View

Figure 50

[4] Labyrinth seals are most common in centrifugal process pumps.
[5] Cutaway of NDPD Cold End from ACD User Manual

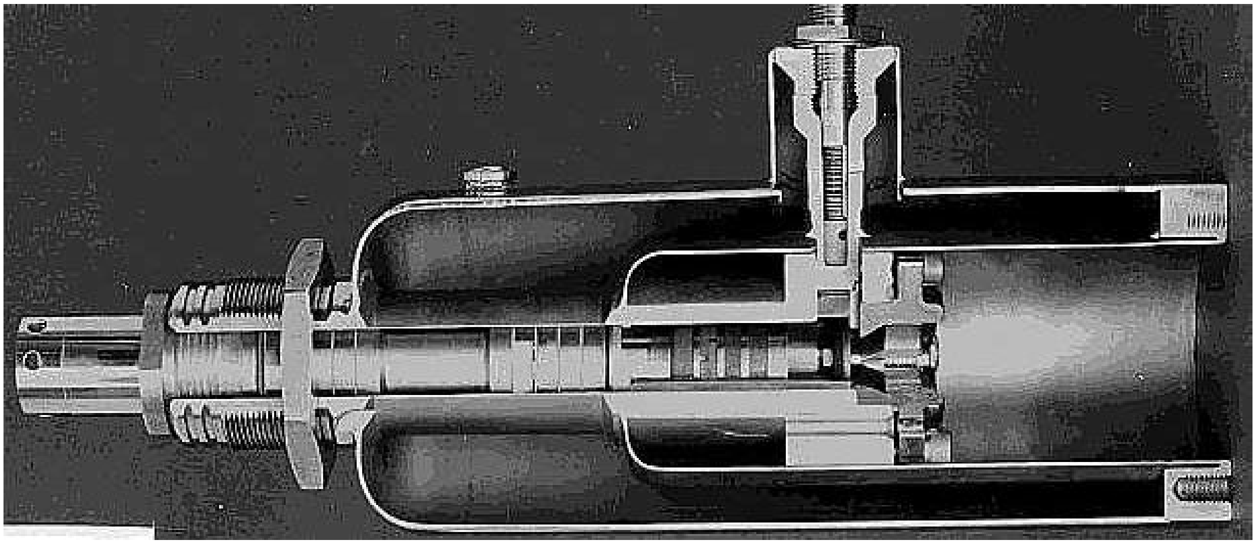


Figure 51 – Typical cold end[6]

Lubricationless packing – Since no known liquid lubricant can exist as an effective liquid lubricant at cryogenic temperatures, appropriate packing material and the piston riding surface must provide the best “wear resistant” properties during operation.

Piston rings and internal moving (Rubbing) components - Must also have great resistance to wear due to the potential two-phase flow (Liquid and gas both present) and poor liquid lubricity.

Cold-end material compatible with low temperatures - High thermal shrinkage rates must be addressed due to such low operating temperatures and behavior of material properties at these conditions must be taken into consideration.

Separation of Cryogenic temperatures from drive-end - Effective elimination of transference of cryogenic cold temperatures must be minimized as this approaches the drive-end and its components.

[6] Courtesy of Cryostar USA

Warm End

Pump warm-end (drive-end)[7] must be designed to withstand the load imposed by the piston (known as “rod-load”). Some type of eccentric device is used to provide the stroke during one revolution of the rotation. This is sometimes referred to as the “Crank” or “Crankshaft” as it applies to the mechanism. Typically ball bearings are used to support this crank device during rotation.

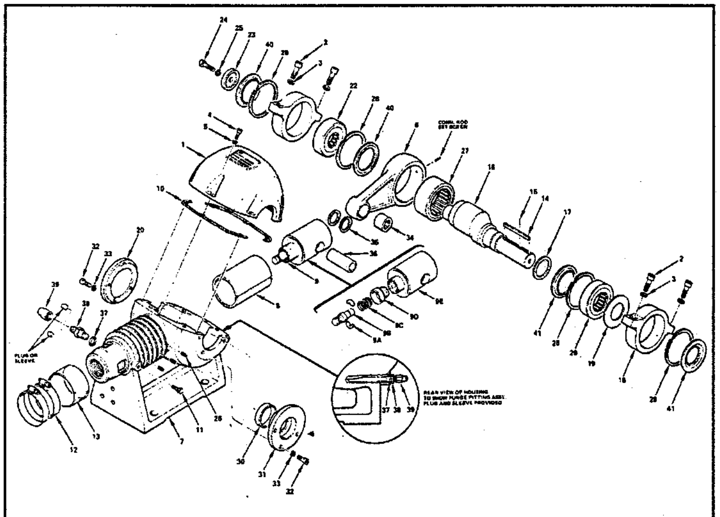


Figure 5-1A. Model NDPD Foot Mount Drive Assembly - Exploded View

Figure 52

An alignment device must be provided to maintain the alignment of the connecting component of the drive-end to the cold-end. This is usually known as the “Crosshead Piston”. This alignment must be maintained or accelerated wear will occur in the packing of the cold-end. Some type of “Wet” lubricant must be provided to the drive-end components that are under load or advance fatigue will occur, resulting in ultimate component failure.

The Role of the Sump

Implementing the use of a sump will typically resolve the issue of “Acceleration Head” losses to the inlet of the pump. This means by having a cold-end immersed in the cryogenic liquid, the pump is less likely to be sensitive to improper pump installation due to poor suction piping. A horizontal pump which has an “In-line” suction (i.e. NDPD style), will be more prone to experience cavitation when the pump has long suction piping due the additional losses while the liquid accelerates during the normal pumping action of a “Positive displacement pump”.

Horizontal

Horizontal sumps – such as those on the NDPD, SDPD, etc. have the advantage of having very small amounts of cryogen in them. As a result, there is little wastage when the pump is shut down.

Disadvantages to horizontal sumps are that there is very little opportunity for liquid to become conditioned in the sump. There is no opportunity for bubbles to separate from the liquid, nor is there enough vertical distance for stratification to occur, so the liquid in the pump is consistently the same temperature, nor is there opportunity to add to the NPSH through a liquid column.

[7] Cutaway of NDPD Warm End from ACD User Manual

Inclined

The inclined design, such as the P-1600 and the CVI PD-3000 or the Woodland WCP tends to provide better gas/liquid separation as the liquid enters the sump and the gas exits the vent. Further, there is ample opportunity for the liquid to become conditioned, as the bubbles and the warm liquid rise to the top of the sump, while the colder, denser liquid, sinks toward the bottom.

Problems with inclined sumps are large amounts of liquid lost after shutdown, and relative problems in maintenance.

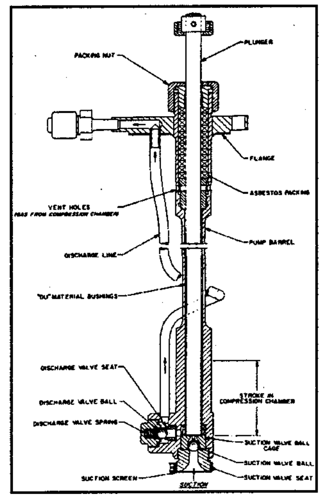


Figure 53

Immersed Pumps

In immersed pumps, the vessel that the pumps are immersed in the vessel which acts as the sump as well as the source of supply. This is technically the most elegant solution in that it:[8]

Allows the pump to be kept cold and ready to operate at all times.

May not require piston shaft seals, since the means to drive the piston could also be maintained under pressure.

Has no piping requirements.

The difficulties in maintenance have however made most immersed pumps a commercial failure. The pumps must be quite long to reach to the bottom of the vessel, and complex procedures are required to remove the pump from the vessel.

Contamination is hard to prevent from moist air.

MVE has recently had good success in immersing centrifugal pumps spinning at 4,500 RPM in liquid nitrogen or liquid natural gas.[9] Reliability data on immersed centrifugal pumps is still being collected.[10]

Effects of the Cryogenic Vessel on Pump Performance

NPSH

Net positive suction head requirements for cryogenic pumps[11]

More Pump problems result from incorrect determination of Net Positive Suction Head (NPSH) than from any other single cause.

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[8] Immersed pump – Linde design, circa 1962
[9] MVE's ORCA vessel for fast delivery of cryogen to sites.
[10] LNG Data – 1,400,000 gallons pumped in 40 to 80 gallon lots at LNG fill station. Teardown showed no damage to bearings. Liquid Nitrogen Data – 6,000 Gallons delivered weekly in 40 gallon lots for 16 months at MG Industries.
[11] Industrial Gases Data Book, Cryopump, India

Liquids at any temperature above their freezing have a corresponding vapor pressure that must be taken into account when planning a pumping system. NPSH can be defined as the difference between the actual pressure and the vapor pressure of the liquid at the suction port of the pump. This is also sometimes referred to as "sub-cooling" or super pressure

While sitting idle, the liquid in a storage vessel will gradually absorb heat and, with all the vents closed, will generate pressures that are directly related to the temperature of the liquid. These pressures are called the "saturated vapor pressure." This saturated condition exists as long as the liquid is at its boiling point for any given pressure in the vessel. The important point to remember is that no matter what the tank pressure is, any reduction in pressure will cause the saturated liquid to boil.

No cryogenic pump can operate on saturated liquid since in order to establish flow into the pump suction there must be lower pressure in the pump. This pressure drop causes the saturated liquid to boil, and the resultant vapors enter the pump causing it to cavitate and loose prime.

To prevent cavitation, some NPSH must be provided to the pump. The amount of minimum NPSH varies with size, type and make of pump, and is generally indicated on the nameplate. The NPSH can be provided by static head, or elevation of liquid above the pump suction and/or by building an artificial pressure in the supply tank with a pressure building coil. This artificial pressure must be maintained throughout the pumping cycle to insure proper and efficient pump operation.

It is easy to lose, or offset, this "artificial" pressure or liquid head, by warming the liquid in the suction line to the pump by heat from the atmosphere. It is possible to have a high "super pressure" in the storage lank so that the liquid is highly subcooled and still have saturated liquid at the pump suction. To prevent this, pump suction lines should be short and well insulated.

One important factor that must be recognized is that the condition of ... liquid products are very frequently near the boiling point.[12]

"A simple statement of what is necessary to make pumps operate is "the pressure pushing the liquid into the pump must be great enough to

a) Overcome the frictional and turbulent pressure losses in the suction path
b) Accelerate the movement of liquid product into the compression chamber
c) Lift the suction valve.
d) Have enough pressure left to keep the liquid from boiling when it enters the compression chamber

Unless these requirements are fulfilled the pump will not operate satisfactorily even though the pump is in perfect mechanical condition."
"... pressure is needed to fill the compression chamber while the pump is on the suction part of the stroke. The faster the pump runs the faster the liquid has to accelerate and move to be sure the compression chamber is full when the compression part of the stroke is started. A pump,

[12] 1962 Linde Distribution Equipment Manual:

incidentally, cannot be run faster than it is possible to push an adequate amount of liquid into its compression chamber."[13]

Saturation as a function of Vessel

There are essentially two types of cryogenic vessels used for pumping. The most common is the standard customer station that can be used to connect to a pump, or to provide medium pressure (200 psi) gas, or liquid. This type of vessel is adequate in all applications, but excels only at providing medium pressure gas.

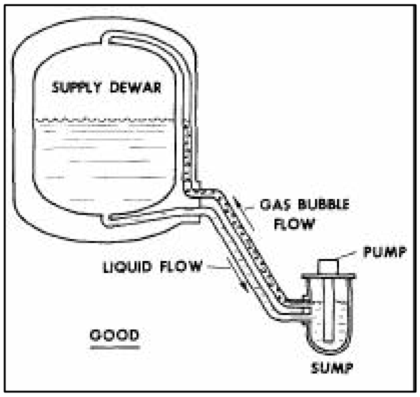

Figure 54

The customer station

The customer station is typically a vertical vessel with a liquid outlet that is connected to the pump inlet, and a vapor line from the top of the tank that is connected to the pump vent. The liquid in the vessel is at a certain level, and as such provides a hydrostatic head to "push" liquid into the pump. The pump is located below the vessel, and the vertical piping distance adds to the hydrostatic head. The more head available, the better the pump will operate. The liquid in the tank will tend to stratify, bringing cooler (denser) liquid to the bottom, and allowing the warmer liquid to float on the top. This cool liquid then is piped to the pump.

On the other hand, the moment the liquid leaves the storage vessel it begins to warm up. The warmer the liquid the worse the pump will operate as NPSH is lost. In piping systems that are too long, or where traps are located, the liquid may become so warm that it cannot pump, but rather turns to vapor on each suction stroke.[14]

In conventional pumping hookups, liquid first pours into the pump sump and is immediately flashed into vapor that is returned to the top of the tanks. As the sump cools down, less and less liquid is converted into vapor, and liquid begins to rise in the pump return until it reaches the same height as the liquid in the vessel. Once equilibrium is reached, flow of liquid into the pump is only sufficient to make up the product pumped out and the liquid evaporated keeping the system cool. This slower speed will cause the liquid to warm up more, as it moves more slowly though the piping heated by ambient temperature. Poorly designed piping will also have vapor traps that further impede flow through the system.

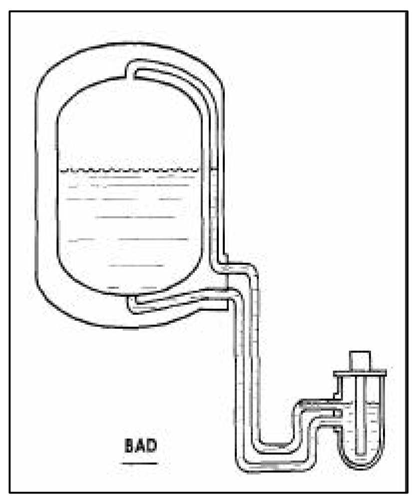

Figure 55

In conventional systems then, the flow of liquid into the pump slows down dramatically once the pump is cool. The

[13] 1962 Linde Distribution Equipment Manual
[14] Pictures from Navair 06-30-501

lower flow rates will allow more heat to enter the liquid going into the pump.

The Thermosiphon vessel

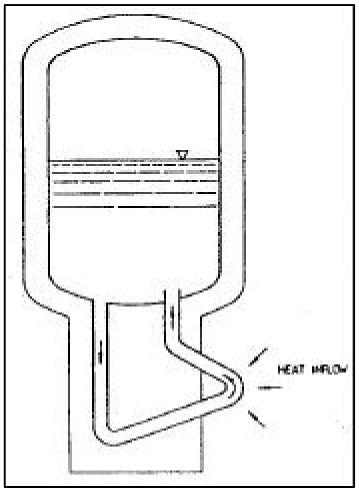

Figure 56

The thermosiphon vessel is a vessel deigned for optimal pumping. The principle of the vessel is to establish a natural convective flow of liquid, to ensure that the pump always has access to fresh, cold liquid. MVE has developed an extremely successful model, the S-100™ thermosiphon.

As can be seen from the sketch at the left, the normal thermosiphon vessel has an insulated pod extending downwards to the base of the vessel. The densest, coldest liquid is found at bottom of this pod. When the piping with liquid goes outside the pod, it warms up slightly, and becomes less dense. The liquid therefore rises back into the tank, and is replaced by fresh cold liquid.

The pump is attached to this stream, and draws whatever fresh liquid it needs to pump from it. The remaining liquid returns to the tank.

There are several caveats when connecting a pump to a thermosiphon. The lines must be as short as possible, and the must be angled such that liquid flow is always upward. That is to say, the liquid flows upward into the pump from the bottom of the pod, and flows upward into the tank from the pump. This allows any bubbles that may have formed to be removed quickly and effectively.

Pumps properly connected to a thermosiphon do not cavitate. Further, they typically will prime – be ready to pump – in three minutes or less.

Areas to investigate

Seals

Typical atmospheric seals (Packing) in cryogenic service yield limited life (1000-2000 hours of service). This number is based on typical high-pressure cylinder filling pump applications. Other applications may provide different service life. The assumed life cycle, while operating an on-board liquid fuel pump, will probably be quite reasonable, as it is assumed that a LNG system will be used on 200+ mile routes. Thus it can be assumed that the pump will operate continuously, at varying pressures, for four or more hours at a time. Further, being able to tailor the storage vessel and the pump as one optimized system also allows for creative seal design

Our starting point is that, at slower piston speeds we have demonstrated long seal life – at least 35% of the 10,000 hour target life. Thus, the piston will probably be quite large in diameter, and move quite slowly.

Further, we plan to investigate if simple planned maintenance – similar to changing an oil filter or a spark plug – is possible. A probable acceptable solution would be a 3000-hour change-out of a seal cartridge, as long as it was as simple and fast as changing an oil filter.

Piston Rings

Potential the same issue as described with seals may exist with piston rings as well. Typically, piston rings have twice the life of seals. Probably, by using modern space age materials a 10,000-hour target life can be achieved.

It is possible that some sort of close tolerance metal seal or metal/Teflon combination may be used.

Custom Vessels

Is probable that the cryogenic vessel for the system will be a custom design to immerse the pump in liquid at all times. The only available NPSH will be that of the liquid column, since the motion of the vehicle will remove any possible stratification. Ideally, the vessel would be vertical, rather than horizontal like the conventional gas tank. Caterpillar's input in truck/tank configuration will be used in the final vessel design.

Further, conventional LNG vessels almost like to have some heat leak going in so as to maintain LNG pressures of 70+ PSI. In the Mamba vessel, cold liquid is desirable, so the insulation system and support system may well be made significantly more efficient. Aerogel/multilayer composite insulations may be used.

Conclusions

Reliability

It has been very difficult to obtain meaningful reliability data. Efforts have been made to contact ACD, Woodland, CTR, CryoStar. With the exception of CryoStar, reliability and life information was limited to “...you know – about a thousand hours, maybe – depends how you use it… ”

The next report will:

Address existing reliability data

Address new data and patents that have been reviewed.

Propose Test Plans

Propose designs to be discussed at the preliminary design review.

Summary

The information provided in this report serves as a background to the existing state-of-the-art in high pressure LNG pumps, as well as insight into the considerations of the design of a reliable and durable LNG pump and fuel handling system. A system capable of handling the conditions present in a vehicle will be required for a field test demonstration of the 3126 DING engine.

Task 4: Commercial Applications Study

Objective:

In this task, the subcontractor shall perform a study on the commercial potential of DING engine technology for a variety of heavy-duty engine applications. The study shall include use of the technology on a variety of applications ranging from pickup/delivery trucks, buses, and on-highway trucks to earthmoving, marine and locomotive applications. The subcontractor shall also examine the potential of DING engine technology for alternative fuels such as propane, the alcohols, dimethyl ether and hydrogen. The study shall incorporate the effect the total system (3000 psi fuel handling system, fuel storage, emissions results and requirements, etc.) will have on the commercial viability of the technology. The study shall identify the most likely applications for the DING technology.

Accomplishment Summary:

An investigation was completed to determine the applications where the DING engine would offer the most economic advantage. Decision and Risk Analysis (DRA) was applied to a spreadsheet which was constructed to dissect the costs associated with engine owning and operating expenses into: 1) the cost of the engine, 2) the cost of the onboard fuel system 3) the cost of the required infrastructure enhancements, 3) the yearly maintenance expense and 4) the yearly fuel expense. In addition to these factors, the emissions reduction produced by each technology in any given application is included so that the environmental impact of new technology can be quantified. The above information is summarized in terms of the overall capital expenditure required, the yearly operating costs, and the period of time required for the new technology to pay back the needed capital expenditure. This information will lead to improved understanding of the sensitivity of the overall cost/benefit balance to each of the contributing factors.

The study concluded that, as there is no current economic incentive to reduce NOx emissions in mobile sources, only vehicles which have a very high fuel consumption compared to engine cost will offer any economic advantage to customers, and these market opportunities exist only if the fuel cost differential between natural gas and diesel fuel remains consistently above 30 cent/DEG for the duration of the payback period.

Accomplishment Details:

Introduction:

The primary reason for the development of DING engine technology is the potential economic benefits such an engine would offer over its competitors in terms of fuel cost, power density, and initial engine cost. The engine under development essentially duplicates diesel engine fuel efficiency over the entire operating range, yielding significant reductions in fuel consumption in comparison with a traditional lean burn gas engine. Engine cost benefits are expected to accrue due to the increased power density of the DING engine, resulting in a smaller required engine and thus lower manufacturing costs.

In order to evaluate the potential benefits of the DING engine and explore the emissions reduction potential of the engine as well, a spreadsheet analysis has been conducted. The spreadsheet compares the DING engine with its most likely competitors for a number of different possible applications. In order to succeed in the engine marketplace the DING engine must not only offer cost savings over a diesel engine, but must also outpace the other gas engine competitors, namely: lean burn natural gas engines, stoichiometric natural gas engines, and dual fuel natural gas engines. A spreadsheet has been created that takes into account initial engine cost, fuel system cost, engine maintenance expense, and fuel efficiency in comparing each of these engine configurations for five different applications. These applications were chosen to span the range of engine cost/fuel consumption ratio, which is the main factor in determining the number of years required for an investment in natural gas technology to yield a savings to the customer. In addition to analyzing the costs involved, the total NOx emissions of each engine platform have been calculated such that the cost of emissions reduction can be portrayed accurately. This information may offer insight into the most cost-efficient way a customer with multiple engine platforms, for example railroads or large industrial plants, can meet impending emissions regulations.

The purpose of this analysis is to offer insight during the development process into the cost and emissions implications of various technologies. In order to be useful in this capacity a number of assumptions must be made in order for the study to yield comprehensible results. These include assumptions as to the costs of engines which are not currently produced, the costs to the customer beyond the incremental cost of the initial engine purchase, the future differential between the available supply of natural gas and diesel fuel etc. Assumptions that have been made in the course of the analysis are described in detail in the following discussion. Changes in these basic assumptions can yield significant alterations in the overall economic analysis, so whenever possible, a range of values has been tested which span the envelope of reasonable quantities. Specifically, the cost differential between natural gas and diesel fuel has been allowed to vary over a wide range (10 cents to 80 cents per diesel equivalent gallon (DEG)) with the resulting variation in payback period expressed in the discussion section. In the future, this spreadsheet analysis will allow the optimum engine to be developed to suit any desired application and will aid in revealing markets in which natural gas engines can effectively compete.

<u>Analysis Method</u>:

The five applications that were selected for evaluation in this study were line-haul locomotive, stationary power generation, switcher locomotive, line haul truck, and transit bus. Each of these applications either already has significant penetration by natural gas engines or is expected to offer potential for high levels of penetration in the future. Natural gas locomotives have been under consideration at least since the early 1970's and several demonstration projects have been carried out by Burlington Northern Santa Fe and Union Pacific. These projects utilize dual-fuel engines due to their relatively high power density and potential for fuel flexibility. Stationary power generation is one of the largest current markets for gas engines, with penetration expected to increase rapidly in the near future due to deregulation of the utilities industry. This market is dominated by lean burn natural gas engines as a result of their efficiency at full load and low emissions relative to their diesel counterpart. Several switcher locomotives are in operation utilizing lean burn natural gas engines. The primary driving factor for this application is emissions reduction in regions with poor air quality. Simplicity in fuel storage and transportation makes natural gas a reasonable choice for switcher locomotives. Line haul trucking has the potential for natural gas engine penetration due to the fixed nature of the truck route and very high yearly fuel consumption relative to engine cost. One of the main limiting factors at present is the absence of a high power density natural gas engine that can directly replace a diesel engine. The transit bus industry has seen a large influx of natural gas powered vehicles in the last decade. This transition has been driven by the desire to reduce emissions in urban areas where governing agencies do not solely rely on economics to drive decision making. Central refueling and limited route lengths also contribute to the viability of natural gas engines in transit buses. In each of the potential applications described, the DING engine offers some advantages over its competitors, typically in either overall fuel consumption or lower initial engine cost due to increased power density. The degree to which these advantages influence the overall economies of the engine purchase decision are discussed in the results section.

The five different engines considered for each application are the traditional diesel engine, a stoichiometric natural gas engine, a lean burn natural gas engine, a dual fuel engine, and the direct injection natural gas engine. Data for the traditional diesel engine was taken directly from production engines for each application. Engine cost was extracted directly from the Caterpillar price list and represents the retail price for the engine unit only, i.e. generator sets were not included for the locomotive or power generation applications. The fuel economy and emissions are for current engines and do not include any of the upcoming advances in EGR and injector technology. Data for stoichiometric natural gas engines are based on the 3516 stoichiometric engine sold by Caterpillar. Emissions are based on a catalyst efficiency of 90% NOx conversion, a typical value for catalyst systems. Lean burn gas engine data was taken from current production cost and performance literature. The engine used for the study is a "high" compression ratio (11:1) 3516 which runs on high methane number gas. Additional derating would be required if a lower quality fuel was to be utilized. The dual fuel engine data is based on a compilation of engines taken from the literature. It is not representative of the current "micro-pilot" engines that are becoming available, but is similar to traditional dual fuel engines. The basic assumptions for this engine are that the replacement rate varies linearly from 0% at idle to 95% at full load, with the peak load 5% less than the diesel and the efficiency equal to the diesel at full load and varying linearly down to 80% of the diesel at idle. The NOx emissions were taken to be 2.5 g/hp-hr throughout the load range. As additional data on modern micro-pilot engines becomes available they can be included in the model. The DING engine performance data has been taken from the most recent DING engine tests. It does not include EGR in order that a direct comparison with non-EGR equipped diesel engines can be made. The NOx emissions are those taken at efficiency levels that closely mimic the diesel engine.

The cost of a natural gas engine with respect to its diesel counterpart is one of the main impediments to penetration of natural gas engines into the marketplace. Because many of the markets being considered in this study are not currently served by Caterpillar products, a method of accurately gauging the price of a production version of these engines had to be established. The technique that has been applied is based on the assumption that the incremental engine cost is a combination of the cost increase due to limited production runs and the cost associated with the need to overcome the BMEP penalty inflicted by most natural gas engines. For each application, the base engine cost is the current price of the diesel engine which would satisfy the customer requirements, for example a 3612 diesel engine rated at 4964 h.p. was used for the line haul locomotive application. This cost was then multiplied by a BMEP size factor for each engine. The size factors were determined by comparing the highest rated 3516 diesel engine with the highest rated engine in each other category. For instance, a 3516 diesel engine is capable of 1900 hp at full load, while the stoichiometric version of the 3516 gas engine is rated at only 1085 hp, yielding a BMEP size factor of 1.75. The base engine cost is thus multiplied by the BMEP size factor to account for the larger engine that would be required if a natural gas engine were chosen. This method assumes that the cost of a given engine scales linearly with engine displacement, which is relatively valid for most of the Caterpillar engine line. For example the cost of the base diesel 3608 is $412,000 compared to $842,000 for the 3616. This scaling of engine price with size is a large impediment to the penetration of spark ignited natural gas engines into the marketplace. The second contribution to increased prices for natural gas engines is the cost of limited production versus mass production. While natural gas engines share many components with their diesel counterparts, a number of key features including fuel injectors, pistons, valves, camshafts, etc. are not shared between applications, significantly increasing the cost of limited production gas engines. For each application in the study, limited production incremental engine cost was determined by comparing the cost of the base diesel with the natural gas engine of the same size, i.e. 3516 diesel versus 3516 natural gas. This incremental cost varies from about 30% for large engines up to over 100% for smaller engines. A portion of this price difference is the actual cost of producing limited numbers of distinct engines, a second portion is the increased profit margin required by the producer to be induced into investing in a limited market outside of the traditional diesel field. These two components have not been analyzed separately and are both taken under the heading of incremental engine cost.

Another weakness of natural gas engines, particularly for mobile applications is the incremental fuel system cost in comparison with the diesel engine. This incremental cost can include the fuel tanks, fuel pump, regulators, safety systems, etc. While light duty natural gas vehicles are often powered with compressed natural gas, the higher fuel consumption of heavy duty vehicles typically dictates liquefied natural gas as the fuel source. For the purpose of this study, all costs beyond the fuel tanks and the fuel pump have been neglected. This provides a best case scenario for comparing the natural gas engine to its diesel competitor. In addition, the availability of fueling stations is assumed. The cost of an LNG fueling station is at present a large impediment to natural gas engine penetration into the mobile market and will need to be overcome in some way if natural gas is ever to become a significant mobile engine fuel. Fuel system costs for this study are based on the best available data for each application. Line haul locomotives are easily equipped for LNG operation as cryogenic rail cars are already available for general transportation applications. The $225,000 cost used in the study is based on the demonstration projects being carried out. The cost of an LNG fuel pump is an unknown as there are not any mass produced, high pressure, mobile LNG pumps available. It is estimated that a high pressure LNG pump will fall between $5000 and $15000 for the various applications used. The fuel tank cost for other applications is based on a price of $50/DEG which is well below the cost of current tanks but should be realizable within the next 5 years. The stationary power application does not include a tank cost as fuel would undoubtedly be pipeline natural gas. The DING engine for this application includes a stationary gas compressor which is required to provide the high pressure gas for direct injection. For most of the applications, the fuel system cost is significant but it is typically lower than the engine incremental cost.

Engine maintenance is included in the study but current projections of maintenance costs for mobile gas engines vary widely. For large stationary engines, natural gas engine maintenance is about 15% lower than diesel maintenance due to longer oil change periods and lower overall engine loading. For small mobile applications such as transit buses, the maintenance cost can be as much as double the cost of the diesel engine. For this study reasonable values have been selected based on the body of literature available but more specific data is required to improve the accuracy of the model. Overall, the contribution of the incremental maintenance cost between diesel and natural gas engines does not play a large part in determining the payback period of natural gas engines.

On an economic basis, the primary advantage that is offered by current gas engines over diesel engines is lower overall fuel cost due to the price of natural gas being lower than the cost of diesel fuel. In order to evaluate the cost benefits that can be accrued, the total fuel consumption of each engine in each application was determined. While the total fuel consumption in any application is a complex function of many variables, for this study the fuel usage has been reduced to fuel efficiency at five load levels multiplied by the amount of time spent at each load level. For example, stationary power generation engines spend the vast majority of their time at full load, therefore full load efficiency is much more important than low load efficiency. On the other hand, switcher locomotives spend nearly 65% of their operating time at idle, and only a small amount under full load, meaning that part load efficiency is at least equal in importance with peak efficiency. For each application, the amount of time spent at any given load was determined by multiplying the percentage of time spent at that load by the total yearly operating hours. These load factors were found in the literature for each of the applications. The engine efficiency was determined from engine test data for idle, 25% load, 50% load, 75% load and full load. All of the data included in the study was based on 3500 series data, as this is the only engine line for which DING efficiency and emissions data is available. The diesel data is based on diesel baseline tests run on the 3501 DING engine. Lean burn and stoichiometric data is based on production Caterpillar 3516 engines of these types. The most distinguishing feature of the fuel consumption data is the poor low load fuel efficiency of the lean burn and stoichiometric gas engines. In each case, fuel consumption at idle is nearly twice the diesel baseline. The total yearly fuel cost for each application is determine by totaling the natural gas and diesel fuel consumption and multiplying by the given fuel costs. The primary variable in this study is the cost differential between natural gas and diesel fuel. This differential was allowed to vary between 10 cents/DEG and 80 cents/DEG. Diesel fuel and natural gas costs vary from location to location as well as in time such that selecting a single cost differential would be of little value.

While in the vast majority of cases, the engine emissions play no part in the economic decision between diesel and natural gas engines, the total NOx emissions have been determined as part of this study. The purpose of this is to reveal how much NOx reduction can be achieved for any given application and allow an estimation to be made of what cost is inflicted by the desire to reduce emissions. The NOx emissions were calculated in the same way as the fuel consumption, with weighted averages taken over the operable load range. Also revealed by the NOx calculations is the relative emissions of the various applications.

The engine cost, fuel system cost, maintenance expense and fuel cost are all combined in determining the payback period for each application. The payback period is simply the number of years that the engine must be in service before the fuel cost savings recognized by the natural gas engine overcomes the higher initial engine cost. The payback period has no component to include emissions reductions achieved with natural gas engines. If emissions credits were to be instituted, or if limitations are placed on total fleet emissions, an emissions cost column could be added to the spreadsheet to account for the emissions reduction on an economic basis.

Results and Discussion

The basic spreadsheet used for this analysis is shown in Table 7 (fold out on next page). The properties of each application which make it unique from the others are given in the rows immediately below the application title. For each application the peak horsepower is given along with load factors for each of the five load levels, the total number of hours of operation each year and fuel prices for diesel and natural gas. The analysis for each application is summarized in the four rightmost columns giving the total NOx emissions, the total capital expenditure, the yearly operating cost, and the payback period for the engine.

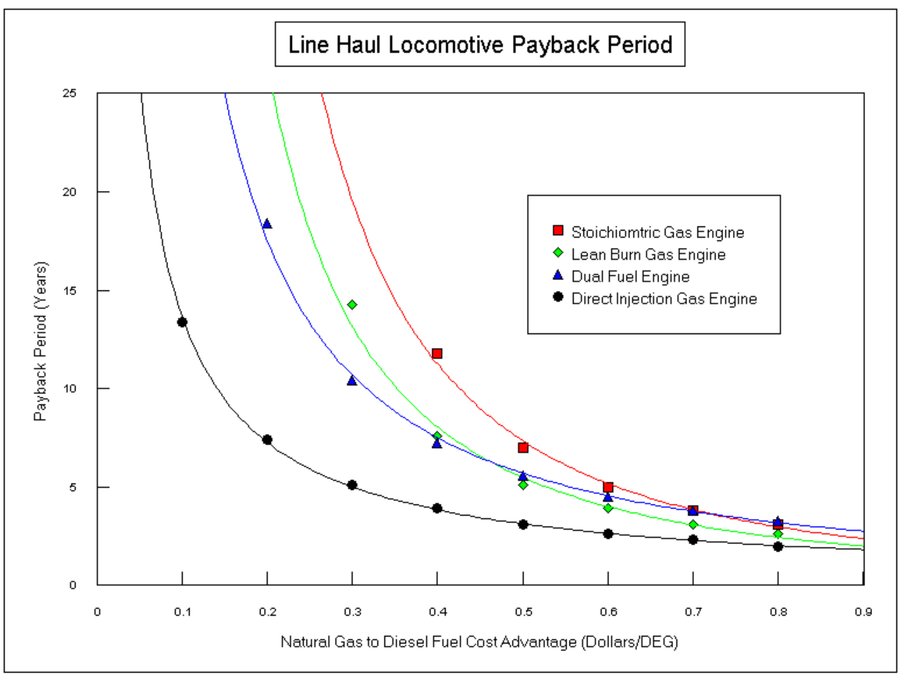

Figure 56 - Line haul locomotive payback period

The first application shown is a line haul locomotive. The specific application is a typical 5000 hp locomotive running almost 8000 hours per year. Despite the high required engine rating, due to long idle times the overall load factor is quite low. For this application the yearly fuel cost is comparable to the engine cost, making the locomotive a good candidate for DING engine application. The payback period as a function of fuel cost differential is shown in Figure 56.

Clearly, from the figure, in the most likely fuel cost differential region, ranging from 10 cents to 30 cents per diesel equivalent gallon, the DING engine outperforms the other natural gas engines by a substantial margin. Despite this, the payback period for the locomotive application is still between 5 and 15 years. These figures represent a best case scenario as no capital return on investment has been included. In order for a DING engine to be selected over a diesel engine on a strictly economic basis, a fuel cost differential of at least 20 cents per gallon would have to exist and would have to persist for at least the duration of the payback period. While in some regions and at some points in time this type of differential does exist, it would be a very risky capital investment to convert a large number of locomotives based on these figures. If a larger cost differential existed, due to dwindling crude oil supplies, nearly any of the gas engine would offer a reasonable diesel substitute. Depending on the cost differential, either a lean burn gas engine or a traditional dual fuel engine would be the next best substitute for the DING engine. As the cost differential increases beyond a certain point the impact of the low substitution rate in the dual fuel application begins to negatively impact its economic advantage. In summary, with today's fuel cost differential, no natural gas engine offers enough economic benefit to displace the diesel engine in any significant number of locomotive applications.

The payback period for a stationary power generation engine as a function of fuel cost differential is shown in Figure 57. While the horsepower rating of the engine is significantly lower than the locomotive engine, the load factor is so high that fuel costs for stationary power generation dwarf the initial engine cost within the first year. It is in this high fuel usage type of application where an engine operating on low cost natural gas has its greatest advantage. This is clear both from the current market for gas engines and from the short payback periods shown in Figure 57. While the DING engine and dual fuel engine both outperform the lean burn gas engine due to superior peak efficiency, they are largely excluded from this market due to emissions constraints. The only production DING engine, offered by Wartsila, has made some penetration into this high fuel use market, but high NOx emission compared to the lean burn gas engine has limited potential applications. The lean burn gas engine outperforms the stoichiometric engine based mainly on lower initial cost and higher peak efficiency. The need for a high pressure compressor for the DING engine has little influence in this application, as the initial capital cost is quickly overcome by high fuel usage. Overall, the potential of DING engine penetration into the stationary power generation market is limited due to NOx emissions regulations that are currently below levels the DING engine is capable of and will likely continue to be reduced in the future. The application of an SCR catalyst or other exhaust gas aftertreatment system to the DING engine may alter this conclusion, but with the peak efficiency of the lean burn engine so close to that of the DING engine any displacement is unlikely.

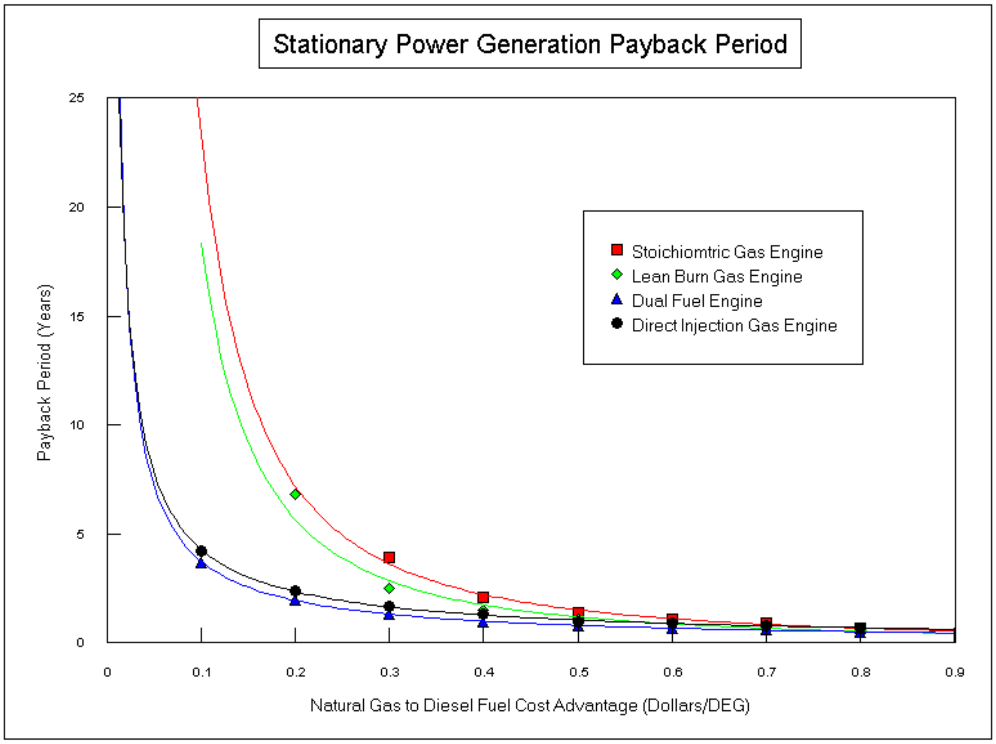

Figure 57 - Stationary power generation payback period

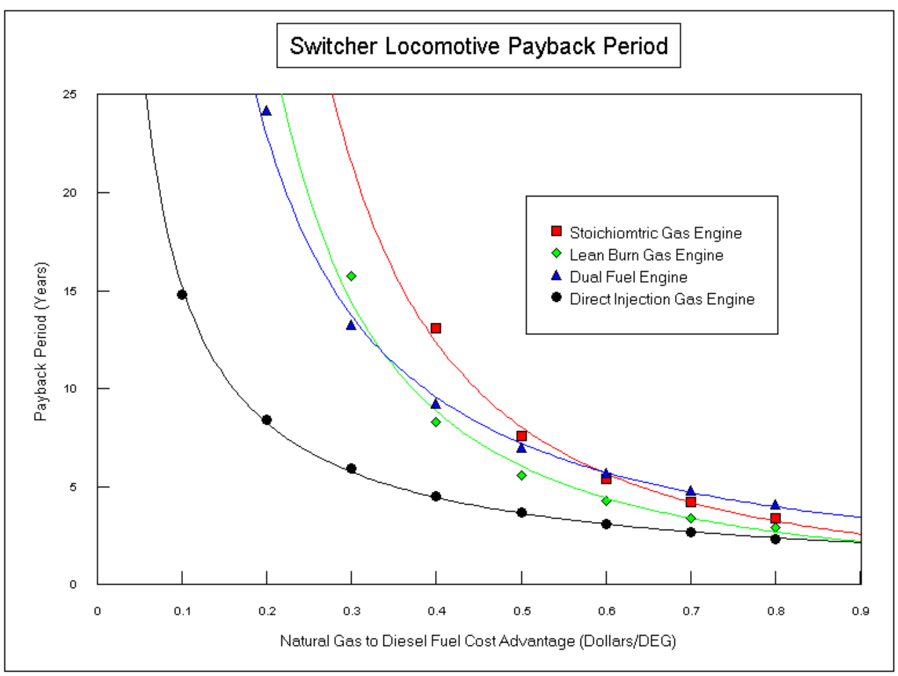

Figure 58 - Switcher locomotive payback period

Lower horsepower switcher locomotives offer some potential for natural gas engine usage due to their limited operating range and the sensitivity of urban areas to NOx emissions. A number of Caterpillar gas engines have already been sold into this market for use in emissions restricted areas. The economic advantage of the DING engine over other natural gas engine types is shown in Figure 58. The typical lifetime of this type of locomotive is over 20 years, which somewhat extends the acceptable payback period. If the cost differential between fuels is on the order of 30 cents/DEG it would be reasonable to expect some market penetration for the DING engine in this application. The main impediments to this are NOx emissions and the inconsistency of gas prices. If the purpose of switching from diesel to natural gas is at least partially attributable to the desire to reduce NOx emissions, a lean burn gas engine offers significantly lower NOx output. In addition, for a DING engine to be economically viable the cost differential between fuels must persist for at least a decade. Considering the large changes in cost differential even from month to month and certainly from location to location, a long term purchase contract between the customer and the gas supplier would be needed to insure a favorable differential over the payback period. Neglecting NOx emission reduction, switcher locomotives offer some potential for DING engine market penetration.

Line haul trucks typically operate at relatively high load factors, over fixed routes, for a large number of hours per year. This operating schedule leads to a high ratio of fuel consumption to

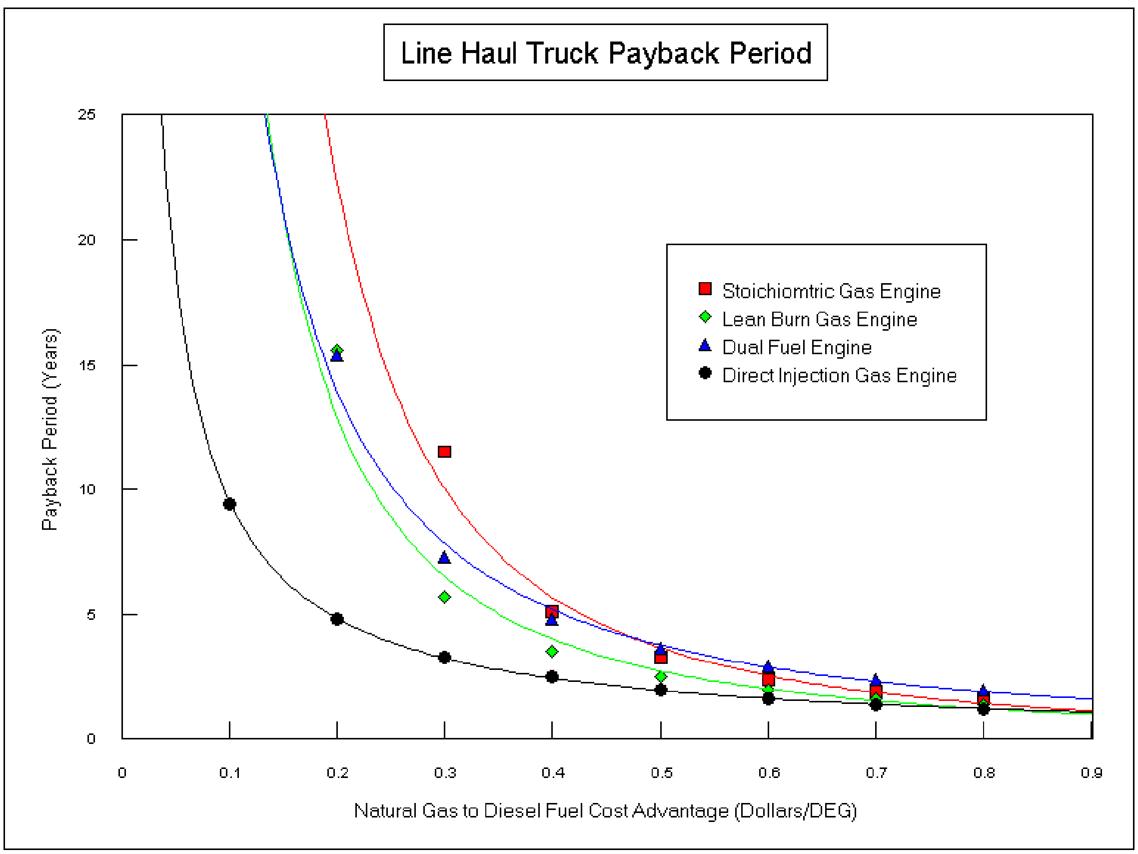

Figure 59 - Line haul truck payback period

engine cost which makes these trucks a promising application for DING engine application. Figure 59 shows the payback period for a 550 horsepower line haul truck operating 12 hours per day. The engine used in this application is a 3406E Caterpillar engine. In order for the payback period to reach its desired value of 2-3 years the fuel cost differential must be at least 30 cents/DEG. In this region the DING engine significantly outperforms all of the other natural gas engine options. Once again, if NOx reduction is the primary purpose of the conversion, other natural gas engines probably offer greater reductions in NOx, though at significantly lower efficiency.

The final application considered is the transit bus market, where natural gas engines have made their largest market impact. The desire to reduce emissions in urban areas has been the primary force pushing this conversion. Clearly, from Figure 60, the economic incentive for the conversion of transit buses to natural gas does not exist. The high initial cost of the conversion, coupled with the relatively low yearly fuel consumption require an unreasonably high fuel cost differential to make transit bus conversion economically viable. The conversion of a large number of buses to date can be attributed almost entirely to government mandates and has little to do with transit bus economics. The main impediment to natural gas transit bus engines is not

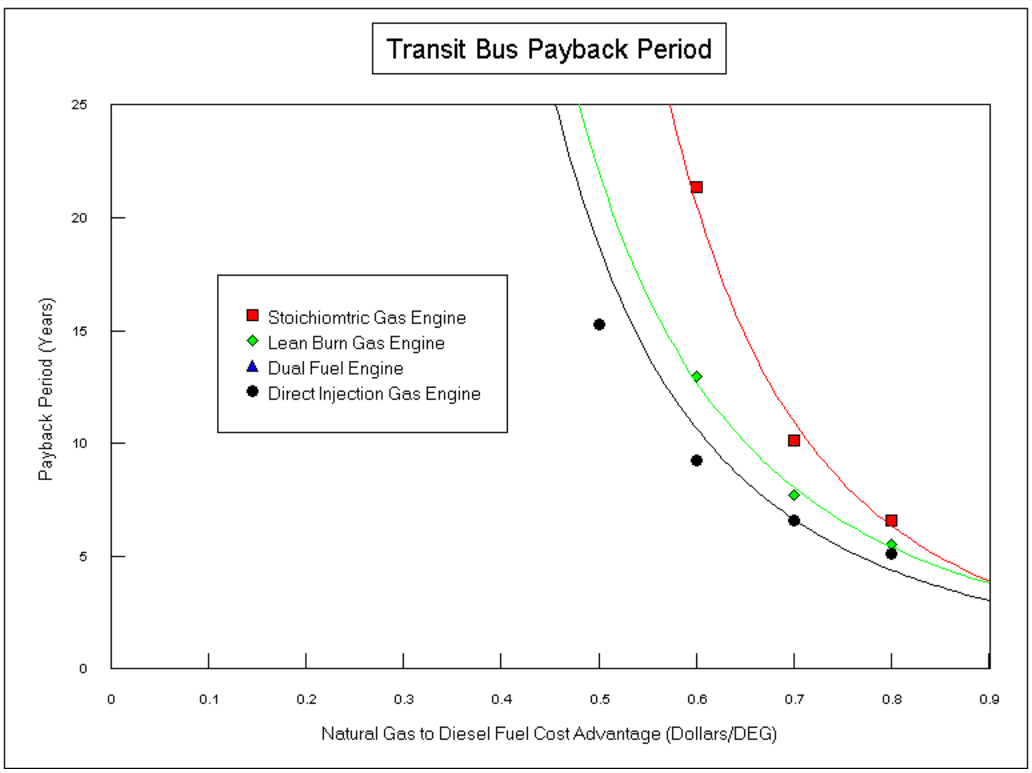

Figure 60 -Transit bus payback period

fuel efficiency but the incremental cost of the initial engine purchase. With yearly fuel costs only reaching about 1/3 of the initial engine cost, the payback period for transit buses is inherently long. The relatively longer lifetime of transit buses, as mandated by transit authorities helps in allowing longer payback periods, but with payback periods being upwards of 25 years for any fuel price differential less than 50 cents/DEG, the potential for economically viable natural gas engines in transit buses is presently unforeseeable. In this market, the DING engine offers little economic incentive over other gas engines, due to the low fuel consumption, and offers little advantage over the diesel engine due to its relatively minor NOx reduction potential.

Figure 61 shows the NOx emissions of each engine type for each application. This figure gives some indication of the relative contribution of the various applications to total NOx generation. A line haul locomotive operating nearly 8000 hours/year can produce over 100,000 kg of NOx per year. With 20,000 such locomotives in operation in the United States these emissions make a significant contribution to overall NOx emissions. Even more substantial is the contribution of the 2 million class eight trucks on the road which produce over half of the total national NOx

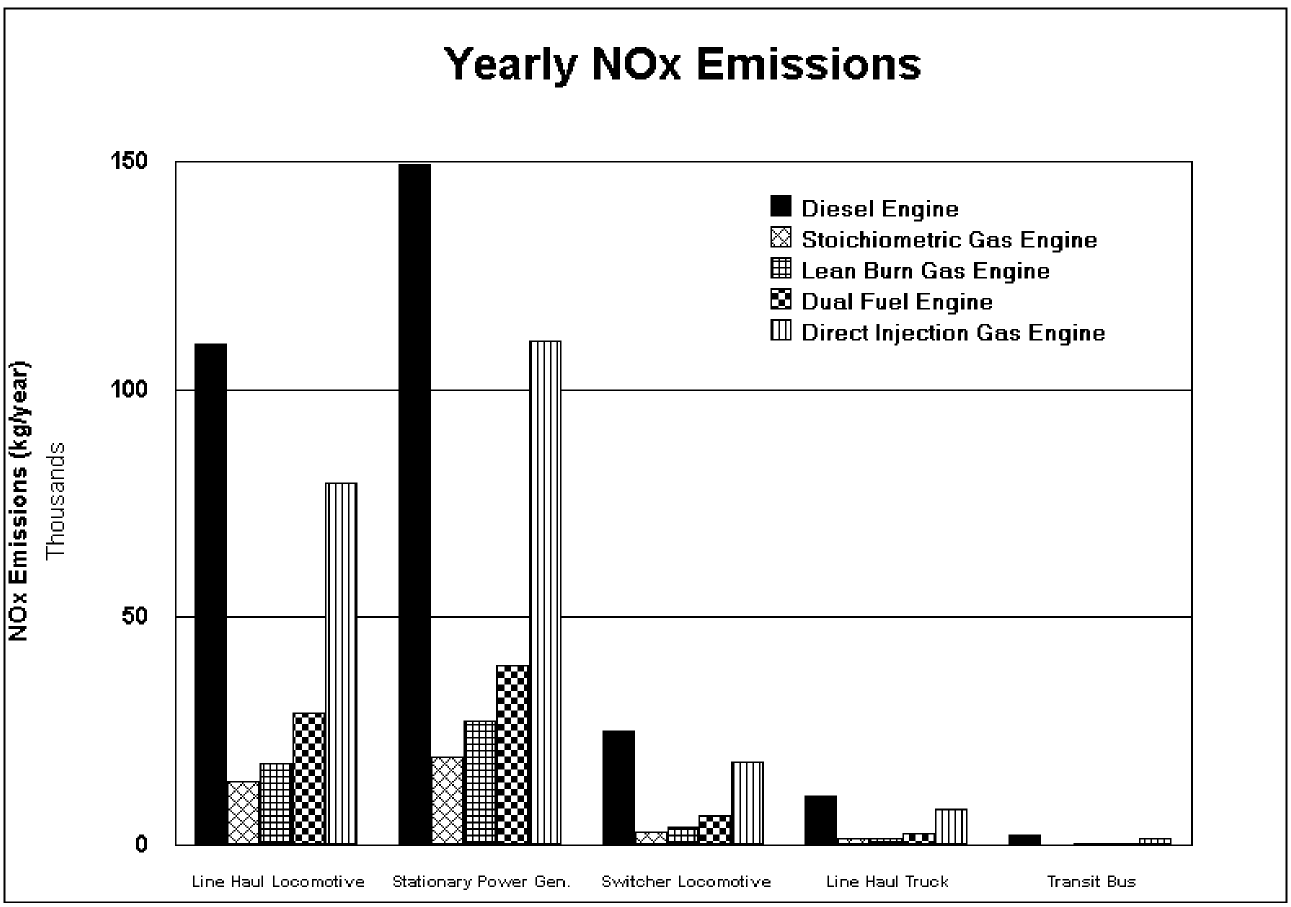

Figure 61 - Comparison of yearly NOx emissions

emissions from mobile sources. The relatively low fuel usage combined with a small number of vehicles (2000 switcher locomotives and 60,000 transit buses) makes emissions from either transit buses or switcher locomotives inconsequential in terms of overall NOx. These applications may still make a significant contribution on a local level, as their emissions are typically concentrated in high population density urban areas. In summary, as there is no current economic incentive to reduce NOx emissions in mobile sources, only vehicles which have a very high fuel consumption compared to engine cost will offer any economic advantage to customers, and these market opportunities exist only if the fuel cost differential between natural gas and diesel fuel remains consistently above 30 cent/DEG for the duration of the payback period.

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13. ABSTRACT (Maximum 200 words)
This report summarizes the results of Phase 2 of this contract. We completed four tasks under this phase of the subcontract. (1) We developed a computational fluid dynamics (CFD) model of a 3500 direct injected natural gas (DING) engine gas injection/combustion system and used it to identify DING ignition/combustion system improvements. The results were a 20% improvement in efficiency compared to Phase 1 testing. (2) We designed and procured the components for a 3126 DING engine (300 hp) and finished assembling it. During preliminary testing, the engine ran successfully at low loads for approximately 2 hours before injector tip and check failures terminated the test. The problems are solvable; however, this phase of the program was terminated. (3) We developed a Decision & Risk Analysis model to compare DING engine technology with various other engine technologies in a number of commercial applications. The model shows the most likely commercial applications for DING technology and can also be used to identify the sensitivity of variables that impact commercial viability. (4) MVE, Inc., completed a preliminary design concept study that examines the major design issues involved in making a reliable and durable 3000 psi LNG pump. A primary concern is the life of pump seals and piston rings.

Plans for the next phase of this program (Phase 3) have been put on indefinite hold. Caterpillar has decided not to fund further DING work at this time due to limited current market potential for the DING engine. However, based on results from this program, we believe that DI natural gas technology is viable for allowing a natural gas-fueled engine to achieve diesel power density and thermal efficiency for both the near and long terms.

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