COMBINED EFFECTS OF JP-8 FUEL AND CERAMIC THERMAL BARRIER COATINGS ON THE PERFORMANCE AND EMISSIONS OF A DI DIESEL ENGINE

FINAL PROGRESS REPORT

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An experimental study was conducted on the combined effects of using JP-8 Fuel in conjunction with thin thermal barrier coatings on the specific fuel consumption and emissions of UHC, NO, and smoke of a DI diesel engine. The experiments were conducted on a Ricardo Hydra single-cylinder DI diesel engine. Thin ceramic thermal barrier coatings were applied to various combustion chamber surfaces including the piston crown, cylinder head, and cylinder liner. Tests were run with the insulated surfaces installed individually, and with all coated parts installed together. The results were compared with those obtained from the baseline all-metal engine, and the results for all cases with JP-8 fuel were compared with runs conducted with hexadecane as a baseline fuel. The emissions and fuel consumption were observed for each case over a test matrix of three engine speeds, three load levels, and three injection timings.

The engine performed satisfactorily on JP-8 fuel. No abnormal or adverse operating characteristics were observed in any of the configurations tested. The individual coating schemes can produce improvements in ore or more categories of emissions, but at the expense of an increase in another category. The choice of one insulation scheme over another depends on which emission product is considered most critical.

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SUMMARY

An experimental study was conducted on the combined effects of using JP-8 fuel in conjunction with thin thermal barrier coatings on the specific fuel consumption and emissions of UHC, NO, and smoke of a DI diesel engine. The experiments were conducted on a Ricardo Hydra single-cylinder DI diesel engine. Thin ceramic thermal barrier coatings were applied to various combustion chamber surfaces including the piston crown, cylinder head, and cylinder liner. Tests were run with the insulated surfaces installed individually in the engine, and with all three coated parts installed together. The results were compared with those obtained from the baseline all-metal engine, and the results from all cases with JP-8 fuel were compared with similar runs conducted with hexadecane as a baseline fuel. The emissions and fuel consumption were observed for each engine configuration and fuel over a test matrix of operating conditions consisting of three engine speeds, three load levels, and three injection timings.

The engine performed satisfactorily on JP-8 fuel. No abnormal or adverse operating characteristics were observed in either the baseline all-metal configuration, or with any of the insulation schemes investigated. It is therefore concluded that JP-8 is a viable alternative fuel for use in DI diesel engines with thin ceramic thermal barrier coatings applied to one or more combustion chamber surfaces.

For the Ricardo Hydra DI diesel engine, insulating the various surfaces offered mixed results with either performance or emissions being improved or compromised depending on the insulation scheme and the operating conditions. It was generally found that different operating conditions affected the engine in diverse ways, with speed, load, and to a lesser extent, injection timing, affecting the relative levels of emissions and BSFC for the various insulation schemes. In analyzing the results, greater emphasis was placed on the high-speed (2500 rpm), high-load (16 N-m torque) condition because of the applicability to typical ground-based vehicle applications and because changes in BSFC and emissions have the greatest total impact under these conditions. Even then, no individual scheme was significantly more beneficial than any of the others.

The individual coating schemes investigated can produce improvements in one or more categories of emissions, but at the expense of an increase in another category. The choice of one insulation scheme over another, depends on which emission product is considered most critical. If hydrocarbons are the major emission concern, then coating the head is the best choice, with the added benefit of providing the greatest improvement in fuel economy. If oxides of nitrogen are of greatest importance, then coating the piston is the best choice. If soot is the major consideration, then coating all of the combustion chamber surfaces is the only configuration that consistently reduced soot emission below the baseline level, while also significantly improving UHC emissions (but at the expense of slightly increased NO and fuel consumption). Given all of the trade-offs, (see Table 4.1, page 24) the best choices might well be either coating all surfaces of the combustion chamber to achieve reduced particulate and hydrocarbon emissions with little effect on NO emission and fuel consumption, or the coating the head surface only to achieve the improved fuel economy and UHC emission.

1. INTRODUCTION

1.1 PROBLEM STATEMENT

The objective of this work was to study the effects of using thin ceramic thermal barrier coatings in conjunction with JP-8 jet fuel on the performance and emissions of a direct injection (DI) diesel engine. The Army has been interested in the use of ceramics in diesel engines for a number of years, and several studies have been conducted on various aspects of this topic [1-23]. The DoD has also investigated the suitability of JP-8 jet fuel for use in DI compression ignition engines. This study focused on the use of JP-8 fuel in a Ricardo Hydra DI diesel engine with thin ceramic coatings applied to the piston crown, the cylinder head inner surface and the cylinder liner for the purpose of determining if this combination of fuel and selective insulation of combustion chamber surfaces yields any particular advantages or disadvantages in terms of fuel economy and emissions. Each of the ceramic insulated components was tested individually in the engine under a variety of operating conditions (speed, load and injection timing). Additionally, a set of tests were conducted with all insulated components installed simultaneously. Of particular interest were the effects of the coatings and fuel on brake specific fuel consumption, NOx emission, unburned hydrocarbon emission, and soot emission. A pure hydrocarbon fuel, Hexadecane ($C_{16}H_{34}$), was used as a baseline fuel for comparison purposes.

1.2 PERTINENT LITERATURE REVIEW

LHR Engines

Diesel engines reject about two-thirds of the energy content of the fuel, leaving one-third as useful power output. Approximately one-third of the chemical energy is rejected through the cooling system, while exhaust heat accounts for an additional one-third. Low heat rejection (LHR) engines reduce heat transfer through the walls of the combustion chamber, to the coolant, and have the potential to harnesses some of the rejected energy. Ceramics have been used for thermal barrier applications in diesel engines to reduce heat loss to the cooling system.

The selection of materials with low conductivity, good high temperature strength, low coefficient of expansion, and low cost has been the focus of many studies. However, it has been difficult to identify materials that satisfy all these requirements. Silicon nitride has often been used due to its high temperature strength and toughness, although it is not the best insulator [2,4,8,9,10,11]. Partially stabilized Zirconia (PSZ) has also been widely used because of its good insulating properties. Wong et.al [10] studied the improvement in engine efficiency as a function of coating thickness and material. They found that thin coatings offered the best thermal efficiency. The optimal thickness for PSZ ranged from 0.25-0.5 mm for the cylinder liner, and was around 0.1 mm for the non-friction parts; the piston crown and cylinder head.

The prevailing thought has been that thermal efficiency should improve with a reduction in heat rejection. However, the results of many investigations have shown that simply insulating the combustion chamber surfaces to reduce heat loss does not necessarily increase efficiency. The way in which the resulting hot surfaces affect both the combustion process and volumetric efficiency plays a significant role.

Various theoretical studies using different combinations of combustion chamber insulation of naturally aspirated, turbocharged, and turbocompound engines [12,13,14,15] have shown that LHR engines significantly reduce heat rejection. The increase in thermal efficiency, however, has been small. The turbocompound and turbocharged engines have recorded the most improvement. Kamo and Bryzik [1] predicted the effects of combustion chamber insulation on the performance of both naturally aspirated and turbocharged engines. They simulated a turbocharged engine that was insulated using PSZ. The results showed that insulating the piston and cylinder head only, reduced the heat rejection by 41 percent and increased thermal efficiency by 1.4 percent. Insulating the liner only reduced the heat rejection by 55 percent and increased thermal efficiency by 1.2 percent. The non-turbocharged insulated engine increased thermal efficiency roughly 0.5 percent for zero heat rejection. Miyairi et. al [2] indicated that theoretical models used in calculating the performance of LHR engines could not sufficiently simulate the stratified combustion process of a diesel engine. They experimentally studied the effect of selective insulation of the cylinder head, piston crown, and cylinder liner, using thick monolithic ceramic inserts, on the performance and emission characteristics of a single-cylinder, normally-aspirated DI diesel engine. They showed that fuel economy and NO emissions of the engine were improved by insulating the cylinder head and liner, but were made worse by insulating the piston crown. Part of the degradation in BSFC with the insulated piston was attributed to the increased reciprocating mass due to the heavy monolithic ceramic piston crown. Wong et. al [10] concluded that these studies, in total, showed inconclusive and contradictory effects of thermal barriers on engine efficiency.

Cummins engine company experimentally determined the effect of insulated combustion chambers on performance using an insulated piston and cylinder head separately [16]. The cylinder head and the piston combustion surface were plasma spray coated to a thickness of 1.25 mm. The insulated piston results showed a loss in thermal and volumetric engine efficiency, but the insulated piston was effective in reducing engine heat rejection to the coolant and increasing exhaust gas energy. Insulation of the cylinder head resulted in no change in the brake specific fuel consumption or heat rejection.

Alkidas [17] experimentally investigated performance and emission characteristics of an uncooled, thermally-insulated, single-cylinder, open-chamber diesel engine. The results of his investigation showed this LHR engine had lower volumetric efficiency, higher BSFC, and higher NO/NOx and smoke emissions. Woshni et. al [18] theorized that the deterioration in BSFC from metal to insulated engine by direct component substitution, without component optimization, was due to an increase in heat transfer in the insulated engine. He proposed that the increase is a result of the thinning of the reactive boundary layer caused by higher wall temperature. Cheng et al [19] compared the performance and heat transfer characteristics of a single-cylinder diesel engine. They ran the engine in both metal and ceramic-coated configurations at the same speed, load and airflow rates. They found the insulated engine had a higher BSFC that was attributable to a slower combustion process since the complete burn duration in the insulated engine was longer. They also found the exhaust and the time averaged surface temperature of the insulated engine were higher. In contrast to Woshni [18], Cheng et al [19] found that the total heat transfer in the insulated engine was lower. They blamed the poorer BSFC on a degradation of the combustion system due perhaps to the poor performance of the injector in the hot environment [17]. To optimize engine performance comparison, they suggested a redesign of the injector system to match the charge environment.

Mavinahally, et. al [23] also discuss matching the fuel injection system to the different combustion characteristics of the LHR engine. They found that a combination of high injection pressure and retarded injection timing resulted in improved BSFC on the order of five percent for a six-cylinder, turbocharged, DI diesel with thin thermal barrier coatings on the piston crown, head, and cylinder liner.

Assanis, et. al [2] conducted a series of tests on a supercharged DI diesel engine with and without piston surface insulation to determine the effect of ceramic coating the piston crown on engine performance and emissions. In their study, they emphasized the significance of the heat release profile, and indicated that insulating the piston with a thin coating of PSZ resulted in better engine efficiency and reduced emissions over the baseline engine.

Dickey [1] studied the effect of applying thin ceramic coatings to all combustion chamber surfaces in a supercharged single-cylinder Caterpillar 1Y-540 DI diesel engine. The results showed decreased thermal efficiency, but also decreased specific NOx and UHC for the ceramic coated engine relative to the baseline engine, especially at higher loads.

JP-8 Fuel

To reduce the logistical problems associated with the different fuel requirements of ground vehicles and aircraft, the Department of Defense (DoD) has sought to establish a single fuel suitable for both applications. Diesel fuel composition varies widely. Typically, the choice of fuel depends on engine design and usage. Lighter, distillate fuels are used in higher speed engines, while heavier (residual) fuels are reserved for slower, larger engines. The chemical composition of diesel fuels depends mainly on the composition of the crude oil, from which it is distilled and which varies widely in physical characteristics, and the subsequent treatment during the refining process. The thinner the crude, the higher the proportion of distillate components. The composition of crude oil is mainly hydrocarbons with traces of other elements such as sulfur, nitrogen and oxygen.

The molecular pattern of hydrogen and carbon atoms has a strong influence on the ignition quality and low-temperature fluidity of diesel fuel. A fuel with a lower cetane number has a lower ignition quality and a longer ignition delay period. Increased ignition delay can lead to increased diesel knock and rougher running at lighter loads. Therefore the quality of the fuel needs to be carefully addressed while trying to maintain the ease and cost of refinement. In this regard, JP-8 jet fuel has shown considerable promise as a single-source fuel.

JP-8 is essentially the same as commercial Jet A-1 fuel with the exception of the addition of three additives, a corrosion inhibitor, an icing inhibitor, and an antistatic additive. Relative to Jet A-1, the corrosion inhibitor in JP-8 (usually dilinoliec acid) reduces oxidative wear of engine parts, particulary rotary fuel pump components which are susceptible to increased wear with the use of low lubricity fuels (24). Table 1.1 (excerpted from ref. 25 and 26) compares typical values of selected fuel properties for JP-8 and Number 2 Diesel fuel. Compared to No. 2 Diesel, JP-8 is a lighter, less viscous, more volatile fuel with a 12 percent lower cetane number, comparable gravimetric heating value, and somewhat lower (5 - 8 percent) volumetric heating value.

A number of laboratory and field tests on a variety of compression ignition engines have established the suitability of JP-8 as a fuel for use in DoD diesel powered ground equipment. Owens, et. al (27) reported on laboratory engine tests and vehicle field tests that compared engine performance characteristics for operation on both JP-8 and No. 2 diesel fuel. Five different diesel engines, representative of engines used in a variety of ground-based equipment, were subjected to either the 210 hour Army Coordinating Research Council (ACRC) cycle for wheeled vehicles or the 240 hour ACRC cycle for tracked vehicles, depending on the typical application of the engine. The test engines included examples of two and four stroke, normally aspirated and turbocharged, and in-line 6, V6, and V8 designs. Field tests were conducted on eight different vehicles ranging from battle tanks and armored personnel carriers to utility cargo vehicles. Some conclusions that were drawn from the tests were: (1) No drivability or idle problems occured with JP-8; (2) JP-8 reduced the acceleration of six of the eight vehicles tested and increased it for one of the vehicles; (3) Maximum power thermal efficiency increased with JP-8 for three of the five engines tested, and the increase was sufficient to offset the reduced volumetric energy content so that improved range would be projected for vehicles powered by these engines; (4) JP-8 produces less contamination of engine lubricant, improves top ring wear, and produces less combustion chamber deposits than DF-2; (5) Use of JP-8 resulted in severe wear of the rotary fuel injection pump of one of the laboratory test engines, while a 10,000 mile vehicle test with the same model engine did not produce unusual wear.

Lestz and Lepera (26) reported on a U.S. Army technology demonstration of the use of JP-8 fuel in ground vehicles. The two and one-half year demonstration program was conducted at Fort Bliss, Texas between 1989-1991 and involved over 2,800 pieces of military ground equipment, accumulating over 71,00 hours of operation and 2,621,000 miles. Results of the demonstration showed that: (1) There was no statistically significant difference in fuel consumption between JP-8 and the reference No. 2 diesel fuel (DF-2); (2) Power loss was evident in a few types of equipment, but it was commensurate with the reduced volumetric heating value of JP-8; (3) No catastrophic failures could be attributed to the use of JP-8; (4) JP-8 is suitable for use in diesel powered military ground equipment.

Table 1.1 - Comparison of JP-8 and No. 2 Diesel Properties

Property	No. 2 Diesel	JP-8
Specific Gravity at 15 °C	0.85	0.80
Flash Point, °C	83	45.6
Kinematic Viscosity at 40°C, cSt	2.95	1.25
Sulfur, mass %	0.40	0.07
Net Heat of Combustion, MJ/kg	42.5	43.0
Cetane Number	51	45

2. DESCRIPTION OF EXPERIMENTS

2.1 EQUIPMENT AND INSTRUMENTATION

The present work used JP-8 fuel in a Ricardo Hydra E6 single-cylinder DI diesel engine treated with various ceramic coating insulation schemes to observe fuel consumption and emissions under a variety of operating conditions. Hexadecane ($C_{16}H_{34}$, CN = 100) was used as a baseline fuel for comparison purposes. Table 2.1 provides the basic specifications of the engine and Figure 1 provides a cross-sectional view of the combustion chamber showing the toroidal bowl piston. Measured data included load, speed, crank angle, needle lift, cylinder and fuel line pressure, fuel mass flow rate, air volume flow rate, intake and exhaust gas temperatures, coolant and lube oil temperatures, and emissions of NO and NOx, unburned hydrocarbons (UHC), and smoke.

Engine loading was provided by a digitally-controlled DC motoring dynamometer interconnected to the utility grid through a KTK power control and signal conditioning system. Speed, load and injection timing could be remotely controlled from an instrument panel which provided readouts of these parameters, plus coolant, lubricant, and intake and exhaust manifold temperatures. Additional digital thermocouples were used to monitor room air temperature, NOx sampling line temperature and air filter intake temperature. A sling psychrometer was used to determine combustion air humidity. The engine coolant temperature and oil temperature could be individually controlled with a pair of thermostats that regulate the flow of laboratory water through heat exchangers that provide cooling for the engine fluids. The coolant temperature was maintained at 80 °C and the oil temperature was maintained at 55 °C.

Exhaust smoke was measured with An AVL 415 Variable Sampling Smoke Meter. Oxides of nitrogen were measured with a Thermo Environmental Instruments Model 10AR Chemiluminescent NO-NOx Analyzer in conjunction with a Model 800 Heated Sample Conditioning Unit and a heated sampling line. The NOx system was calibrated periodically against 2000 ppm NO in nitrogen calibration gas. Unburned hydrocarbons were measured with a two-channel Cambustion HFR400 Fast Response FID system. One sampling probe was located in the exhaust port adjacent to the head, 73 mm downstream from the exhaust valve stem. The close proximity of this probe to the exhaust valve, and the rapid time response of the instrument, allowed time resolved measurements of cyclic variations of UHC. The other probe was mounted in the exhaust pipe about 4 m downstream of the exhaust port to obtain an average UHC reading after thorough mixing of the exhaust stream.

Intake air flow was measured with a Meriam Model 50MC2-2F Laminar Flow Element equipped with a digital manometer to read the pressure differential, and an inclined water manometer for calibration purposes. Fuel mass flow rate was monitored with an AVL Model 730 gravimetric fuel balance. Cylinder pressure was monitored with a Kistler 6121 Piezo transducer and charge amplifier. The fuel injector was equipped with a Wolff Controls Hall effect needle lift sensor which was used in conjunction with a Wolff Injector Signal Processor.

The cylinder pressure, fuel pressure, needle lift and crank angle were recorded with a Data Precision DATA 6000 digital wave form analyzer at a data rate of 25 KHz. The data was stored on diskettes for subsequent analysis. Ignition delay and injection duration were determined from the digital wave form data. The 25 KHz data rate translates to 40 microseconds per data point, which then represents the uncertainty for both the ignition delay and the injection duration. Table 2.2 gives the test matrix of load, speed and injection timings used for the experiments.

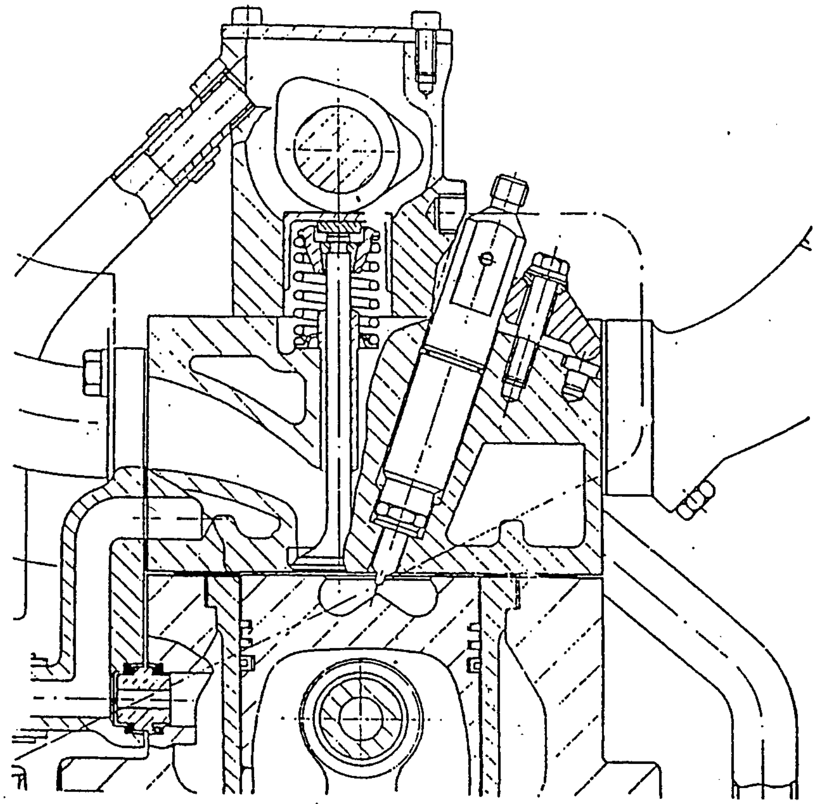

Figure 2.1
Cross Section of Ricardo Hydra DI Diesel Combustion Chamber

Table 2.1 - Specifications of Ricardo Hydra DI Diesel Engine		
Engine		
Number of Cylinder		1
Bore		80.26 mm
Stroke		88.90 mm
Swept Volume		450 ml
Maximum Speed		75 rev/s
Maximum Power		8 KW
Maximum Cylinder Pressure		120 bar
Compression Ratio		20 : 1
Connecting Rod Length		15.80
Squish Height		0.82542 mm
Swirl		3.57
Valve Timing:	Intake Opens	10 BTDC
	Intake Closes	42 ATDC
	Exhaust Opens	58 BBDC
	Exhaust Closes	10 ATDC
Injection System		
Injector Pump		Micro Bosch size *A* type EA 4000 L900
Nozzle		4 holes * 0.21 mm dia. * 155
Nozzle Opening Pressure		250 bar = 25 Mpa
Injector		KBEL 88 PV 1187
Lift Pump		Mico Bosch 9440 030 003
Dynamometer		
Manufacturer		McClure
Type		Shunt wound dc with separate excitation
Rating		30 KW continuous
Max. Speed		100 rps
Control		KTK type 6P4Q3D converter for motoring and regenerative loading

2.2 CERAMIC COATINGS

The piston, cylinder head, valves and cylinder liner were sent to Adiabatics, Inc. in Columbus, IN for coating. The piston crown and bowl received a 0.25 mm slurry- sprayed coating of partially stabilized zirconia (PSZ) consisting of 85 percent partially calcium stabilized cubic zirconia, 10 percent tungsten cobalt chrome powder, and 5 percent chrome oxide. The head and valves were coated with a 0.5 mm thermal barrier coating incorporating 5 percent hollow alumina spheres in a slurry of 65 percent silica, 15 percent PSZ, 7 percent tungsten chrome powder, and 8 percent chrome oxide.

The cylinder liner was bored 0.2 mm over the entire length plus an additional 0.75 mm in the region above the top ring reversal (TRR). The region above the TRR was plazma sprayed with ytria stabilized zirconia, and then the entire length of the liner was given a 0.2 mm wear coat of slurry sprayed PSZ.

2.3 TEST MATRIX

A full set of data runs were conducted with each of the two fuels, hexadecane and JP-8, and for each of five different engine builds utilizing different combinations of coated components including: (1) baseline (no coatings), (2) coated piston alone, (3) coated head alone, (4) coated liner alone, and (5) coated piston, head and liner together. A full set of runs consisted of the test matrix shown in Table 2.2 involving three engine speeds, three loads, and three injection timings, for a total of 27 runs per engine build, per fuel (270 total runs). A constant coolant temperature of 353 K (80 °C) and constant oil temperature of 328 K (55 °C) were used for all runs.

TABLE 2.2 - Test Matrix

ENGINE SPEED (RPM)		1500	2000	2500
INJECTION TIMING (Degrees BTDC)		8	12	16
LOAD	TORQUE (NM)	8	12	16
	BMEP (BARS)	2.235	3.35	4.47

A typical set of test data for one engine build (all surfaces coated) and one engine speed (2500) rpm for JP-8 fuel is shown in Table 2.3.

Table 2.3 - Sample Experimental Data

Test Data For JP-8 Fuel At 2500 RPM With All Coated Parts Installed									
RUN NUMBER	1	2	3	4	5	6	7	8	9
TORQUE (NM)	8			12			16		
INJ. TIMING (DBTDC)	8	12	16	8	12	16	8	12	16
INTAKE AIR TEMP. °C	33	33	34	34	34	34	33	33	34
COOLANT TEMP. °C	81	82	82	82	82	82	81	82	82
OIL TEMP. °C	55	56	56	56	56	56	56	56	56
EXHAUST TEMP. °C	242	238	239	287	282	297	338	336	336
UHC (ppm C_3H_8)	700	640	600	520	570	210	410	410	650
SOOT CONC. (mg/m^3)	1	1	2	7	7	12	40	28	15
Nox (ppm)	450	600	800	.750	950	1375	1050	1325	1875
No (ppm)	375	550	750	650	825	1200	925	1150	1650
INTAKE AIR (CFM)	19.41	19.41	19.41	19.14	19.01	19.27	18.74	18.87	18.61
FUEL CONS. (g/min)	12.3	12.3	12.6	15.8	15.4	16.9	20	19.7	20.2
IGNIT. DELAY (msec)	0.72	0.69	0.73	0.66	0.66	0.68	0.64	0.64	0.71
INJ. DURATION (msec)	0.74	0.71	0.68	0.89	0.85	1	1	1	1
AIR/FUEL RATIO	50	50	49	38	39	36	30	30	29
POWER OUTPUT (W)	2094	2094	2094	3141	3141	3141	4188	4188	4188
BSFC (gr/kwh)	352	352	361	302	294	323	287	282	289
BMEP (bars)	2.23	2.23	2.23	3235	3.35	3.35	4.47	4.47	4.47
INJ. DURATION (Deg)	11	11	10	13	13	15	15	15	15
IGNIT. DELAY (Deg)	11	10	11	9.9	9.9	10	9.6	9.6	10.6
VOL. EFFICIENCY (%)	98	97	98	96	95	97	94	95	94
FILTER TEMP. °F	85	80	79	84	80	80	79	86	79
ROOM TEMP. °F	83	77	78	82	79	78	77	83	80

3. EXPERIMENTAL RESULTS

3.1 GENERAL OBSERVATIONS

Insulating the combustion chamber of an internal combustion engine should improve the thermal efficiency according to the second law of thermodynamics, but, over the years, researchers who have investigated the use of ceramic insulation in diesel engines have obtained conflicting results. Some have obtained improvements, others have noted a deterioration in performance. Some of the reasons for the discrepancies can be attributed to different engine configurations and operating parameters, while others can be attributed to the complex nature of the diesel combustion process. It becomes apparent that pros and cons have to be weighed against each other when dealing with selectively insulating combustion chamber surfaces, leaving one to select the option that works best for any given application. Operating parameters, e.g., speed and load, play a large role in the performance and emissions results.

A detailed analysis of the experimental data obtained from this project will be limited to the high-speed (2500 rpm), high-load (16 N-m) condition because this is the most important operating regime in terms of total fuel consumption and total emissions. Detailed analysis will also be limited to the results for the least advanced injection timing (8°BTDC) because, as illustrated in Table 3.1 below, with the noted exception of soot production, the less advanced timing generally produced better results than the more advanced timings for both JP-8 and hexadecane at 2500 RPM and 16 N-m load.

Table 3.1 - Effect of Injection Timing on BSFC and Emissions at 2500 RPM and 16 N-m

		BSFC			NO			Soot			UHC		
Timing (DBTDC)		**8**	**12**	**16**	**8**	**12**	**16**	**8**	**12**	**16**	**8**	**12**	**16**
Baseline	**JP-8**	✓			✓			✓			✓		
	Hexa	✓			✓			✓			✓		
Coated Piston	**JP-8**	✓			✓					✓	✓		
	Hexa	✓			✓			✓			✓		
Coated Liner	**JP-8**	✓			✓					✓		✓	
	Hexa	✓			✓					✓	✓		
Coated Head	**JP-8**	✓			✓					✓	✓		
	Hexa			✓	✓					✓	✓		
All Coated	**JP-8**		✓		✓					✓		✓	
	Hexa			✓	✓					✓	✓		

(Note: ✓ denotes condition where lowest reading was obtained)

Before analyzing in detail the specific results for the case of high speed and load, a few general observations can be made based on the entire data set. A full set of data plots for BSFC, NO, UHC, and soot for the entire test matrix are provided in Appendices A, B, C, and D, respectively. General observations that can be drawn from these plots include the following.

BSFC (Figures A.1 - A.18)

1) The coated liner produced the highest fuel consumption under all conditions for hexadecane fuel, but had among the lowest fuel consumption figures for JP-8 fuel.
2) The coated piston and coated head reduced fuel consumption relative to the baseline engine under most conditions for hexadecane and at high speed (2500 RPM) for JP-8.

NO (Figures B.1 - B.18)

1) With hexadecane fuel: All insulation schemes reduced NO emission below baseline at low speed, but at higher speeds only the coated piston and coated liner were effective at reducing NO.
2) With JP-8 fuel: The coated liner effectively reduced NO at low speed, while at higher speeds, both the coated liner and the coated piston reduced NO relative to the baseline engine by up to 50 percent.

UHC (Figures C.1 - C.18)

3) With hexadecane fuel: The coated piston and coated head effectively reduced UHC emission by up to 50 percent at the low speed and the high speed, but not at 2000 rpm. The coated liner and coating all surfaces increased UHC above baseline.
4) With JP-8 fuel: The coated head greatly reduced UHC emission below baseline by as much as 60 percent, while the coated liner showed less dramatic, but still significant reductions in UHC. The coated piston generally increased UHC above baseline.

Soot (Figures D.1 - D.18)

1) For both fuels, the coated liner generally increased soot emission above baseline, particularly at low speed and high load when smoke production is highest.
2) At low speed the other insulation schemes reduced smoke slightly relative to baseline, while at high speed the coated piston increased smoke slightly.

3.2 DETAILED ANALYSIS OF HIGH SPEED (2500 RPM) HIGH LOAD (16 N-m) DATA

Figure 3.1 compares the values of BSFC and emissions for each of the five engine builds for JP-8 fuel at 2500 RPM, 16 N-m, and 8 °BTDC injection timing, while Figure 3.2 does the same for hexadecane. From the bar graphs, it is evident that the coated piston reduced fuel consumption slightly (3-4 percent) and NO emission by 20-30 percent relative to the baseline engine, but at the cost of increased UHC and soot emissions. The coated head reduced fuel consumption by 3 percent and UHC emission by 50 percent, but at the cost of increased NO and soot. The coated liner by itself has little effect on BSFC, but effectively reduced NO. Operating the engine with all of the coated components installed resulted in increased NO, UHC, and soot for hexadecane fuel, and increased NO, but reduced UHC, for JP-8.

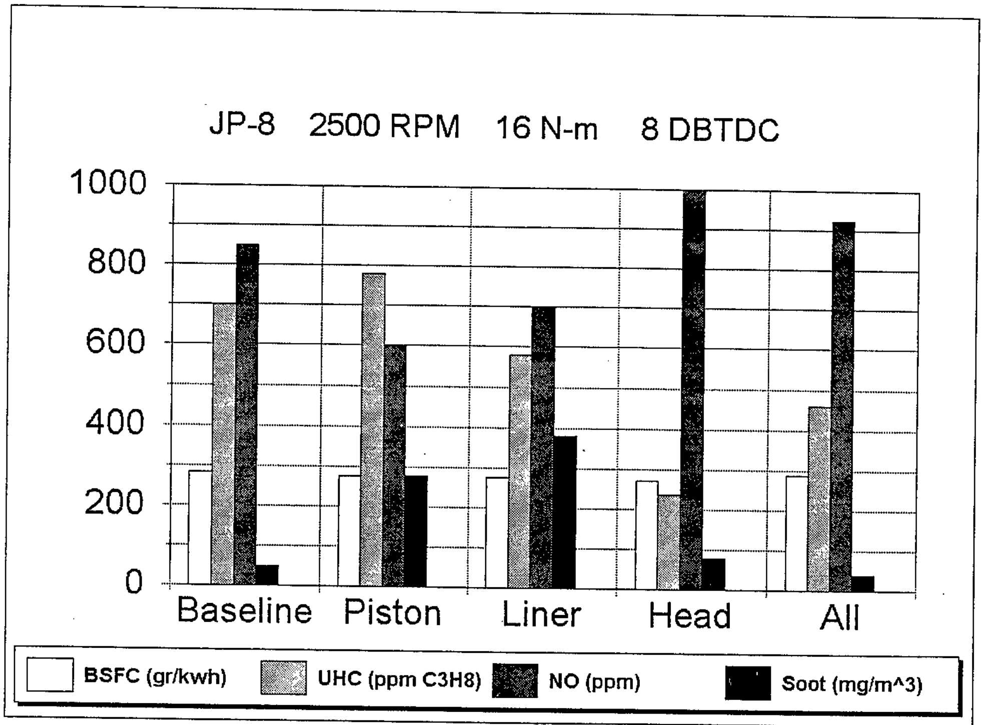

Figure 3.1 - Comparison of BSFC and Emissions for JP-8 Fuel at 2500 rpm, 16 N-m, and 8 °BTDC Injection

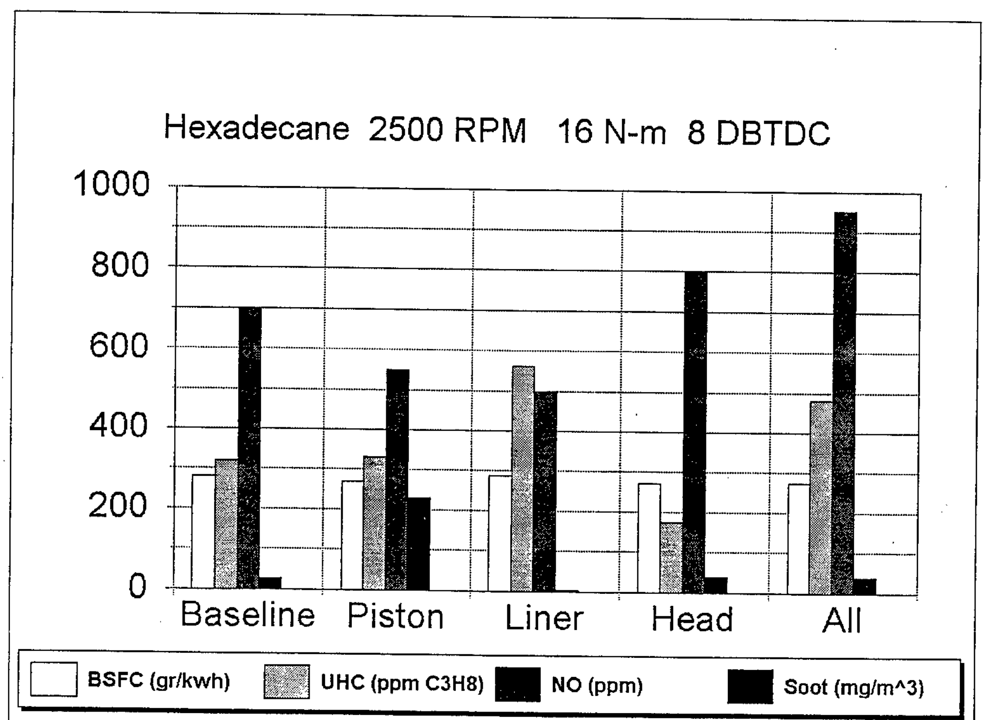

Figure 3.2 - Comparison of BSFC and Emissions for Hexadecane Fuel at 2500 rpm, 16 N-m, and 8 °BTDC Injection

3.2.1 Effect of Coatings on Ignition Delay

Figure 3.3 summarizes the ignition delay data for the high speed, high load case. With hexadecane fuel, all of the insulated cases reduced the ignition delay with respect to the baseline engine as one would expect a priori based on the assumption that insulated combustion chamber surfaces will increase the intake air temperature, which should result in shorter ignition delay. However, in the case of JP-8 fuel, the coated piston caused increased ignition delay, while the other insulation schemes resulted in negligible variation from the baseline engine. The result with the coated piston and JP-8 is in agreement with results reported by Dickey [11] and Assanis, et. al, [18]. They offer various explanations for why the insulated piston produces longer ignition delays than the baseline case, contrary to intuition. They postulate that the higher gas temperatures alter the dynamics of the fuel injector causing lower intitial rates of injection, and thus a decrease in spray velocity and a slowing down of the breakup and vaporization processes, increasing the time required to form a combustible mixture.

Beg, et. al, [22] expanding on the arguments of Dickey and Assanis, state that the spray velocity decreases in the case of a coated piston, resulting in a slowing down of the breakup and vaporization processes of the impinged fuel. The possibility also exists that, even though the piston crown coating in this study went through a densification process to reduce porosity, the process may not have totally eliminated porosity in the coating of the torroidal bowl. This area may indeed act as a capacitor as suggested by Beg, et. al, [24], absorbing and storing some of the fuel that comes into contact with the porous surface, producing both a longer ignition delay and a lower premixed combustion fraction.

At 2500 rpm, 8 degree injection, and high load for JP-8 fuel, the coated liner and coated head had the shortest delay periods as seen in Figure 3.3. In the coated head case the non-insulated piston and non-insulated liner are each cooled effectively by thermostatically controlled circulating oil and coolant respectively. Thus the air in the cylinder receives less heat transfer from the cooler temperature walls and therefore the intake air mass is higher than for the coated piston

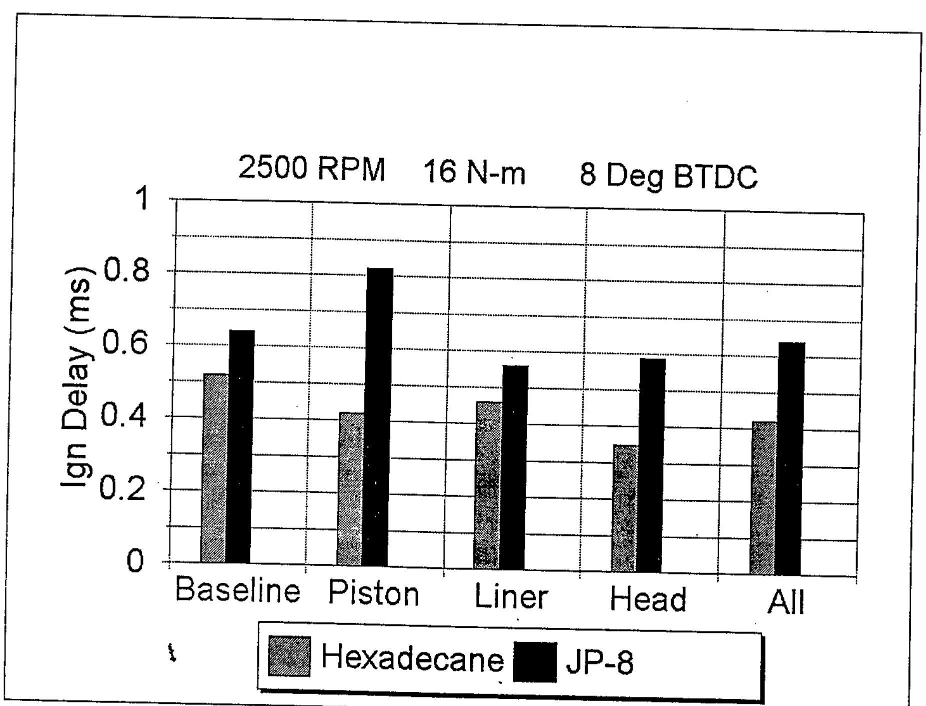

Figure 3.3 - Ignition Delay Comparison for all Insulation Schemes

case. The result is better fuel-air mixing and a more thorough combustion process as indicated by the shorter ignition delay, and the low BSFC and UHC. There may also be heat transfer from the hotter head surface to the injector tip resulting in reduced fuel viscosity and improved spray characteristics [8].

The ignition delay period with JP-8 fuel increased for all engine builds relative to hexadecane, which was to be expected given the lower cetane number (45 vs 100).

3.2.2 Effect of Coatings on BSFC

The BSFC results at 2500 RPM, 16 N-m, and 8 °BTDC are summarized in Figure 3.4 flor all insulation schemes. Minor improvements in BSFC on the order of 2-4 percent were obtained for JP-8 with the coated piston, coated liner, and coated head configurations. But coating all the surfaces increased fuel consumption slightly. With hexadecane fuel, the coated piston produced the best fuel economy, while the coated liner produced the worst. In general, fuel consumption with JP-8 was higher than for hexadecane as a result of the lower cetane number and higher viscosity.

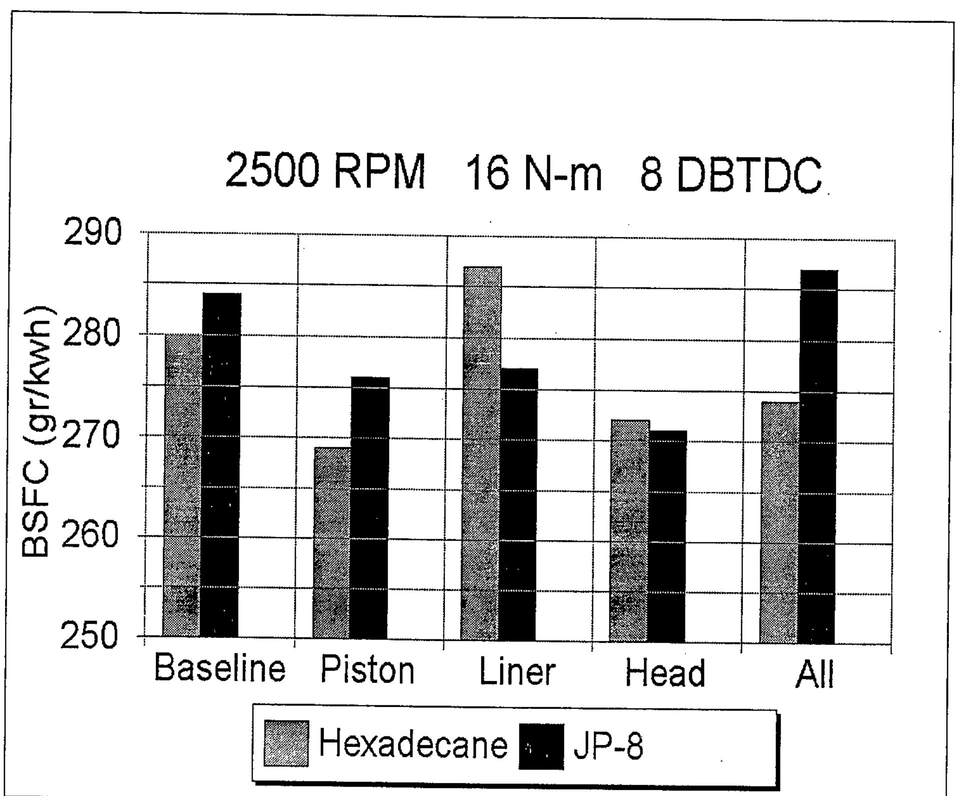

Figure 3.4 - BSFC Comparison for all Builds at 2500 RPM, 16 N-m and 8 °BTDC

3.2.3 Effect of Coatings on Nitric Oxide (NO) Emissions

NO emissions are compared between the five different engine builds at 2500 RPM, 16 N-m, and 8 °DBTDC in Figure 3.5. Nitric oxide is formed in diesel engines when nitrogen and oxygen in the cylinder react at high temperature during the combustion process. In general, higher peak cylinder temperatures result in higher levels of NO. Typically, longer ignition delays tend to increase the premixed combustion fraction thereby causing higher peak temperatures and NO emissions. But in our experiments the coated piston had low NO emissions even though the ignition delay was the longest. This would imply that the peak cylinder temperatures, and consequently peak cylinder pressures, were relatively low, even with the longer ignition delay periods. This is confirmed by the cylinder pressure traces (P-Θ diagram) shown in Figure 3.6. It is seen that the lower NO readings for the coated piston and coated liner correspond with the lower cylinder pressure for these cases, while the high NO readings for the coated head and all-surfaces cases correspond with the higher cylinder pressures for these cases. The higher cylinder pressures, which accordingly imply higher peak gas temperatures, suggests a more complete burning of the fuel in the coated head and all-surfaces cases. The improved fuel economy and low UHC for these builds would also substantiate such a condition. The coated head, which is also common to all-surfaces build would appear to be the controlling factor for the improved combustion. As stated earlier, this could be a result of the hotter cylinder head surface temperature increasing the temperature of the injector tip and thereby affecting the injection process in a positive way in terms of improved atomization and mixing as proposed by Alkidas [8] and Dickey [11].

JP-8 generally produced higher levels of NO emissions throughout the test spectrum. The higher NO levels can be attributed to the longer ignition delay characteristics (or larger premixed fraction) of the JP-8 fuel resulting in higher peak cylinder temperatures.

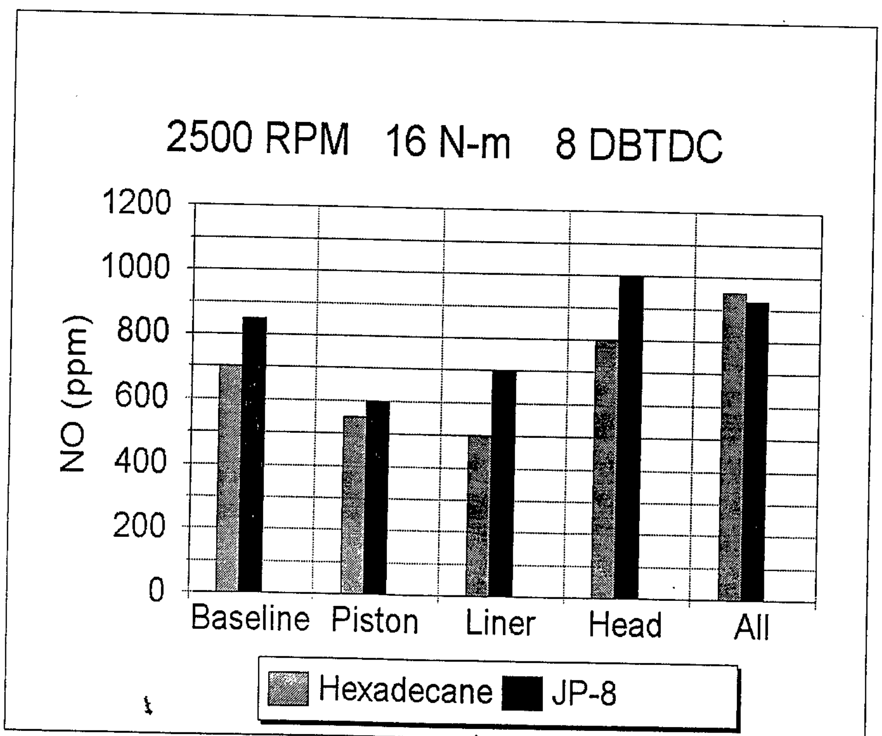

Figure 3.5 - NO Comparison at 2500 RPM, 16 N-m, and 8°BTDC

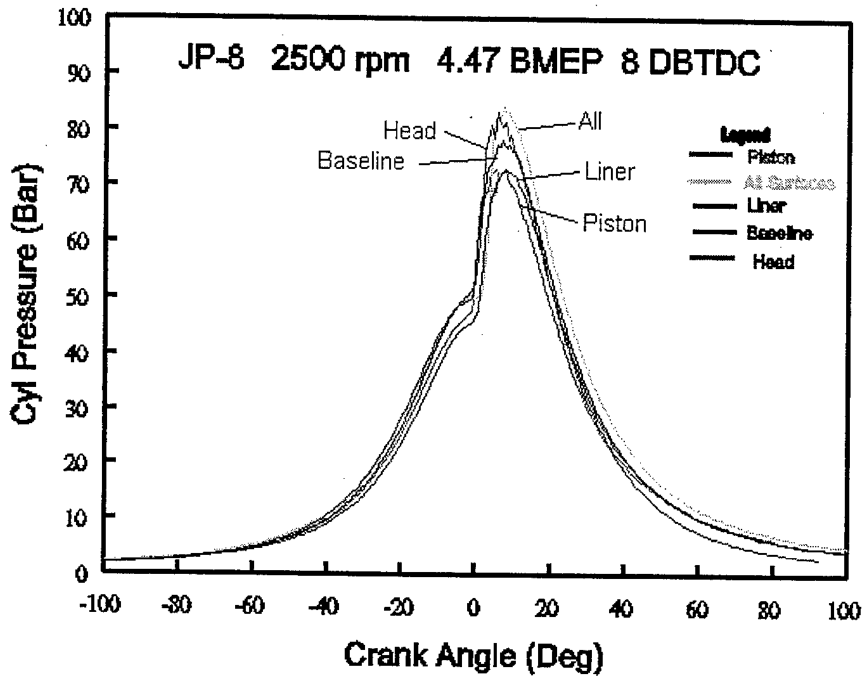

Figure 3.6 - P-Θ Diagrams for 2500 RPM, 16 N-m, and 8 °BTDC Injection.

3.2.4 Effect of Coatings on Hydrocarbon Emissions

Generally, UHC emissions were load sensitive and minor fluctuations in load conditions greatly affected the UHC present in the exhaust line. For JP-8 at 2500 rpm and high load, the coated piston yielded the highest UHC readings while the coated head produced the lowest as seen in Figure 3.7. The long ignition delay associated with the coated piston case (Figure 3.3) may explain the high UHC readings for the coated piston since more fuel is burned later in the cycle during the expansion stroke with a higher probability of wall quenching. On the other hand, better combustion efficiency as evidenced by the lowest BSFC and the short ignition delay period, would explain why the coated head exhibited the lowest UHC. There is in fact reasonable correlation between ignition delay and UHC emission for all the builds, with shorter ignition delay resulting in lower values of UHC.

Hydrocarbon readings were typically higher with the JP-8 fuel than for hexadecane, in particular for the baseline and coated piston builds. An increase in UHC with JP-8 would be expected as a result of the lower cetane number and longer ignition delay resulting in more fuel not being fully combusted. In the case of the coated liner, the UHC emissions were nearly the same for the two fuels, as were the lengths of the ignition delay period.

The large reduction in UHC for the coated head is, again, most likely related to temperature effects on the injector spray characteristics. These same effects are present, as well, for the all-surfaces case, and help to mitigate the tendency of the coated piston to increase UHC.

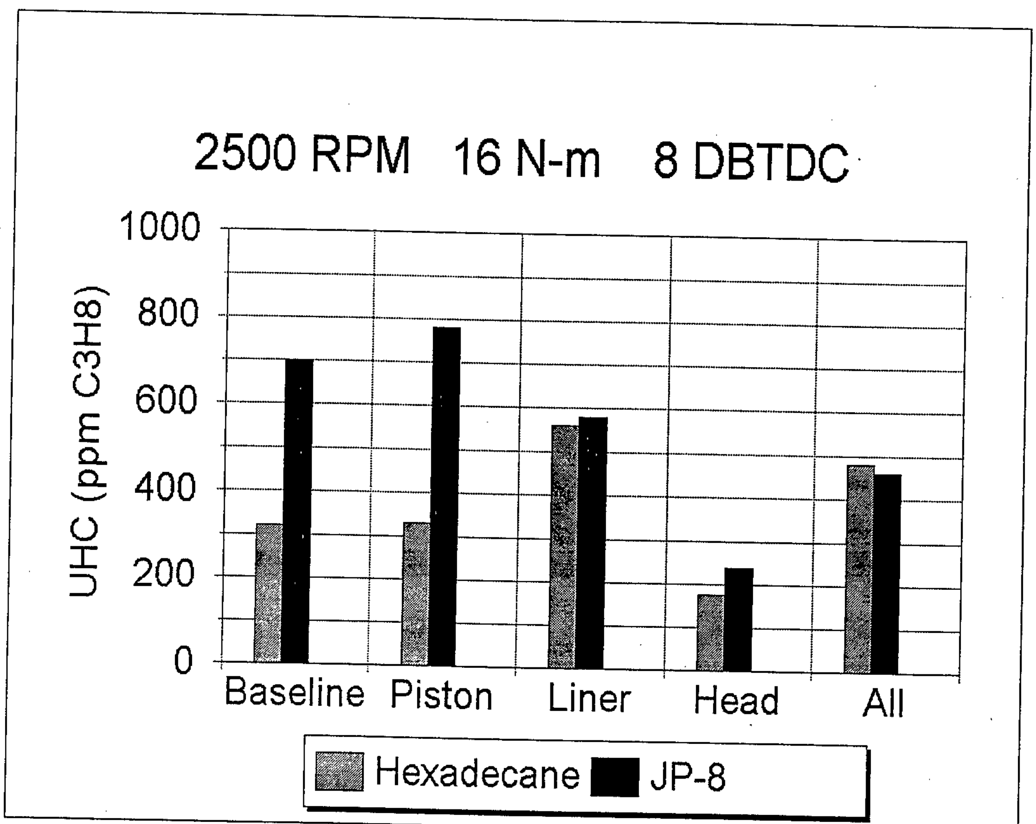

Figure 3.7 - UHC Comparison for 2500 RPM, 16 N-m, and 8 °BTDC

3.2.5 Effect of Coatings on Soot Emissions

In order to lower soot emissions, a diesel engine needs a lean, high-temperature environment. Unfortunately, the conditions that reduce soot create an environment that increases NO emission. Smoke levels in gr/m^3 of soot for all engine builds at 2500 RPM, 16 N-m, and 8 °BTDC are shown in Figure 3.8. At this RPM, the coated piston and the coated liner consistently exhibited the highest soot emission. At lower speeds the coated liner, in particular, produced very heavy smoke under heavy load (see Figures D.1 - D.12). The high levels of soot production for the coated head and coated liner cases correlate with the reduced levels of NO emission observed. Correspondingly, the low smoke characteristics of the baseline, coated head, and all-surfaces builds, correlates with the higher levels of NO observed for these cases. The high levels may also be attributed to the extended diffusion combustion mode, caused by late burning of the absorbed/trapped fuel in the piston crown. The result would be a richer fuel mixture. Coating all the surfaces resulted in the lowest amount of soot due to the presumed higher combustion temperature persisting throughout the combustion cycle.

In all cases soot levels went up with JP-8 relative to hexadecane, indicative of the late combustion of the lower cetane number fuel. Since, at 2500 RPM, soot levels were fairly low to begin with, increases in soot emissions for JP-8 were not significant except for the coated liner which, for unexplainable reasons, produced very low smoke with hexadecane fuel.

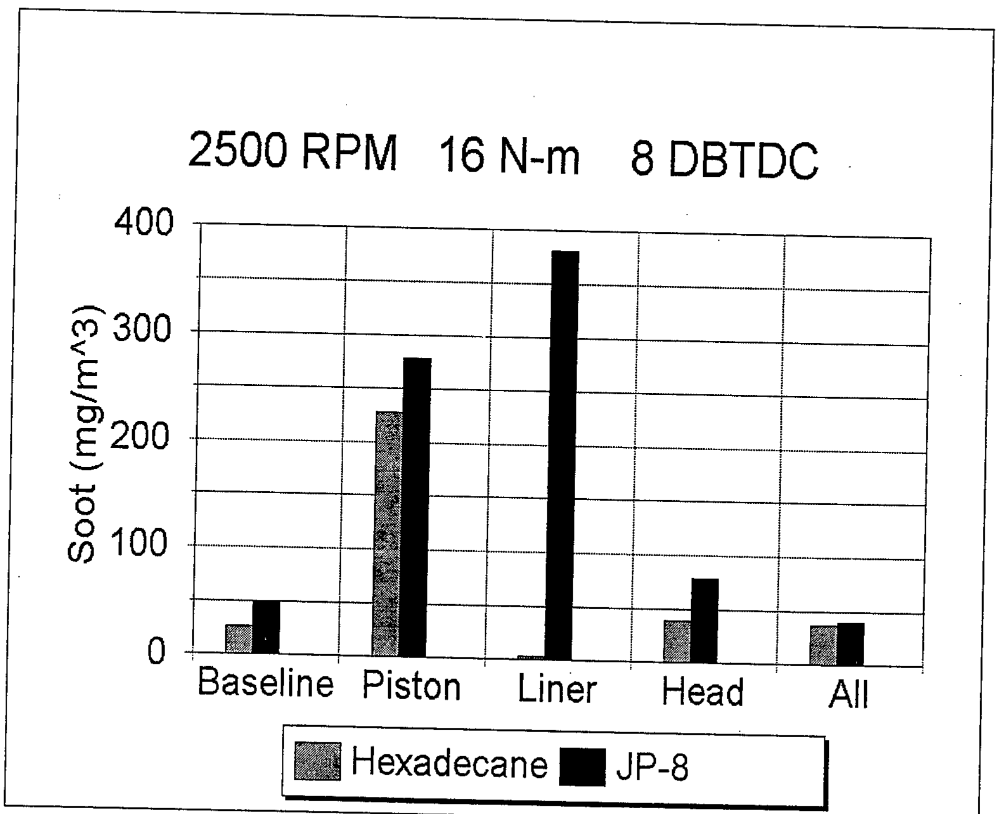

Figure 3.8 - Comparison of Soot Emissions for 2500 RPM, 16 N-m, and 8 °BTDC

3.2.6 Effect of Coatings on Exhaust Gas Temperature

Figure 3.9 summarizes the results for exhaust gas temperature at 2500 RPM, 16 N-m, and 8 °BTDC injection for all engine builds. Figures 3.10 a-c on the next page plot exhaust temperature versus load for JP-8 fuel for each of three engine speeds. From Figure 3.10 it is observed that exhaust temperature increases with engine speed as would be expected due to decreasing time for heat loss from the gas to the surrounding chamber walls. The coated liner and the coated piston consistently produce the highest exhaust temperature and the coated head consistently produces the lowest exhaust temperature. This would appear to be in disagreement with the NO emissions and cylinder pressure data, but exhaust temperature is not necessarily directly related to peak combustion temperature (which controls NO emission and peak cylinder pressure). Exhaust temperature is dependent on the heat release profile and the magnitude of the premixed combustion fraction relative to the diffusion fraction, and also on the conductivity of the solid surfaces exposed to the expanding gas. The insulated head produces low exhaust temperature as a result of its relatively short ignition delay shifting heat release earlier in the cycle which allows more time for expansion cooling and heat transfer from the hot gases. In addition, the uninsulated piston and liner provide effective heat transfer paths to the oil and the coolant. The coated piston and liner produce higher exhaust temperatures through a combination of heat release shifted later in the expansion stroke and effective thermal barriers on the highly-cooled piston and liner walls.

As seen in Figure 3.9, There was an increase in the exhaust temperature for JP-8 fuel relative to hexadecane for all the coating schemes except the coated piston which registered a slight decrease. It is unclear why the decrease occurred for the coated piston case since it would be expected that a longer ignition delay for JP-8 (0.82ms vs 0.42ms for hexadecane) would result in later burning of the fuel. All other cases showed a correlation between the ignition delay period, and the exhaust temperature change, i.e. a longer ignition delay would cause a rise in the exhaust temperature. A summary of these results is shown in Table 7.

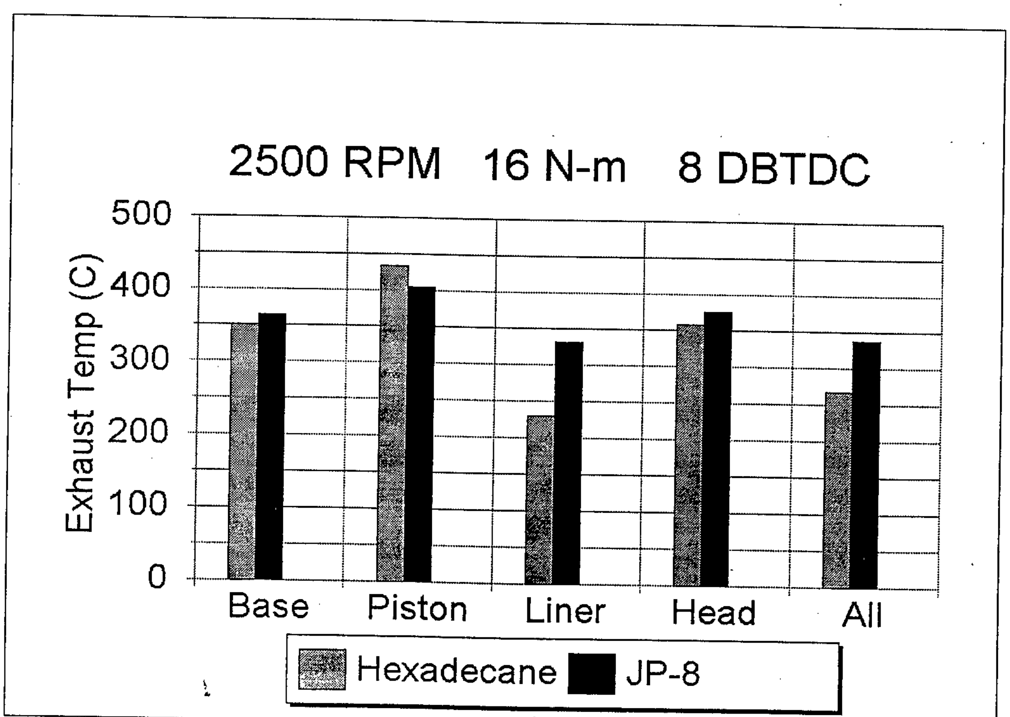

Figure 3.9 - Comparison of Exhaust Temperature for 2500 RPM, 16 N-m, and 8 °BTDC

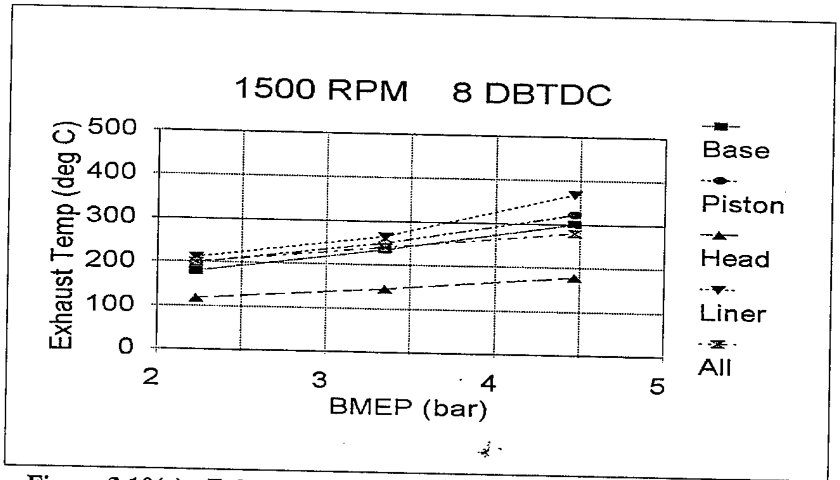

Figure 3.10(a) - Exhaust Temperature Vs Load for JP-8 at 1500 RPM

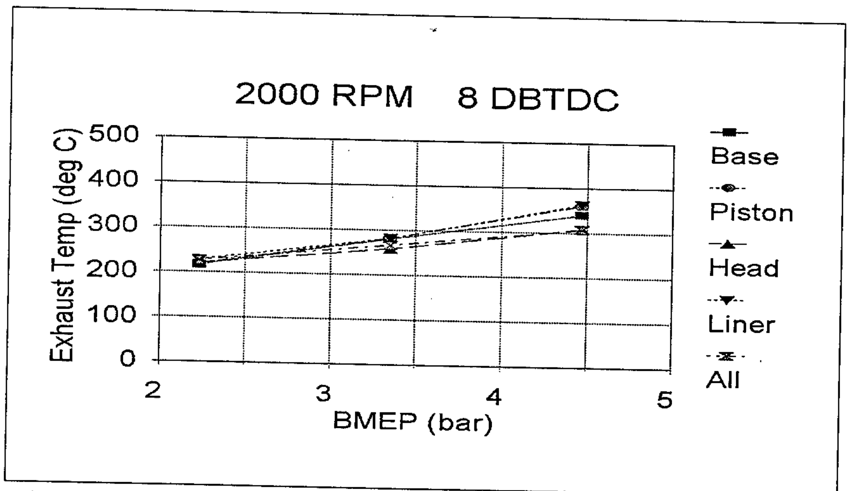

Figure 3.10(b) - Exhaust Temperature Vs Load for JP-8 at 2000 RPM

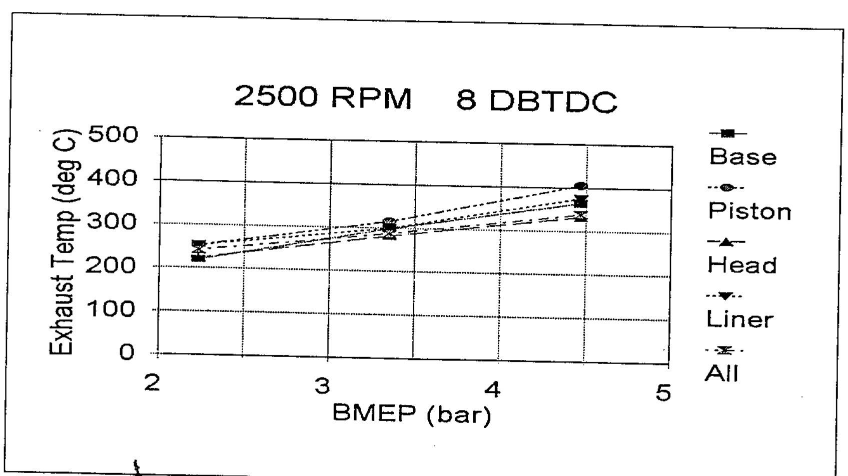

Figure 3.10(c) - Exhaust Temperature Vs Load for JP-8 at 2500 RPM

TABLE 3.2 - Ignition Delay And Exhaust Temperature Comparison at 2500 RPM, 16 NM, 8°BTDC

	IGNITION DELAY (msec)		% CHANGE	EXHAUST TEMP (°C)		% CHANGE
FUEL	HEX	JP-8		HEX	JP-8	
BASELINE	0.52	0.64	+23	350	364	+4
COATED PISTON	**0.42**	**0.82**	**+95**	**434**	**405**	**-7**
COATED LINER	0.46	0.56	+30	358	384	+7
ALL SURFACES	0.42	0.64	+53	268	338	+26
COATED HEAD	0.35	0.59	+59	230	332	+44

(Note: coated piston is highlighted to indicate the anomaly)

3.2.7 Effect of JP-8 on Cylinder Pressure

Illustrations of the differences in cylinder pressures for JP-8 and Hexadecane for the five different engine builds are shown in Figures 3.11 a-e. They clearly show the increased ignition delay for the JP-8 fuel which results in a larger premixed combustion fraction and therefore a higher rate of pressure rise at the start of combustion and higher peak cylinder pressures.

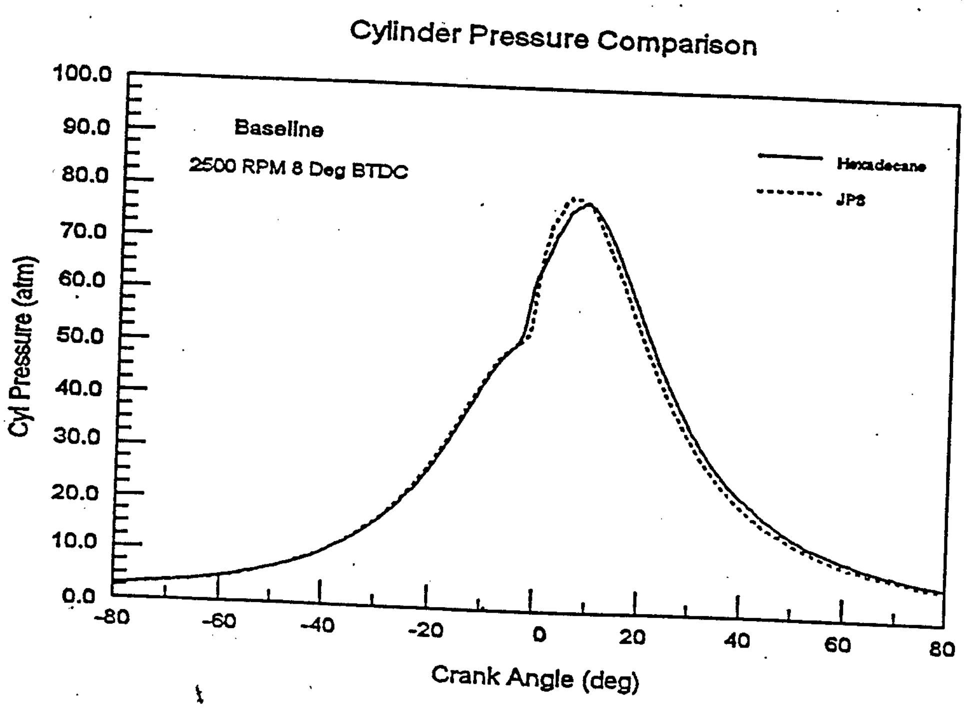

Figure 3.11(a) - P-Θ Comparison for JP-8 and Hexadecane for Baseline Engine

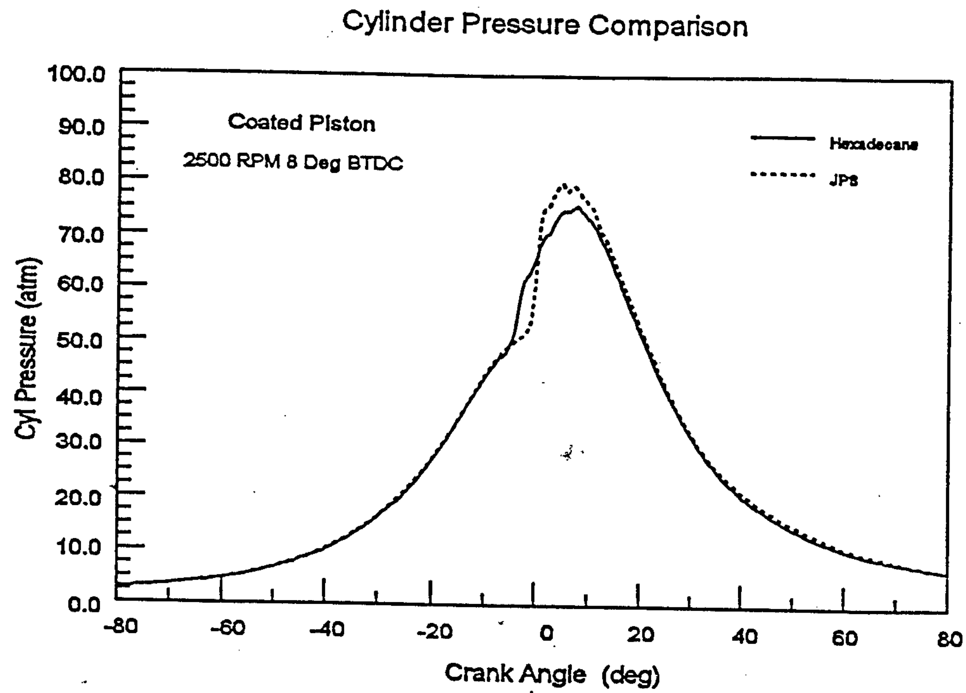

Figure 3.11(b) - P-Θ Comparison for JP-8 and Hexadecane for Coated Piston Build

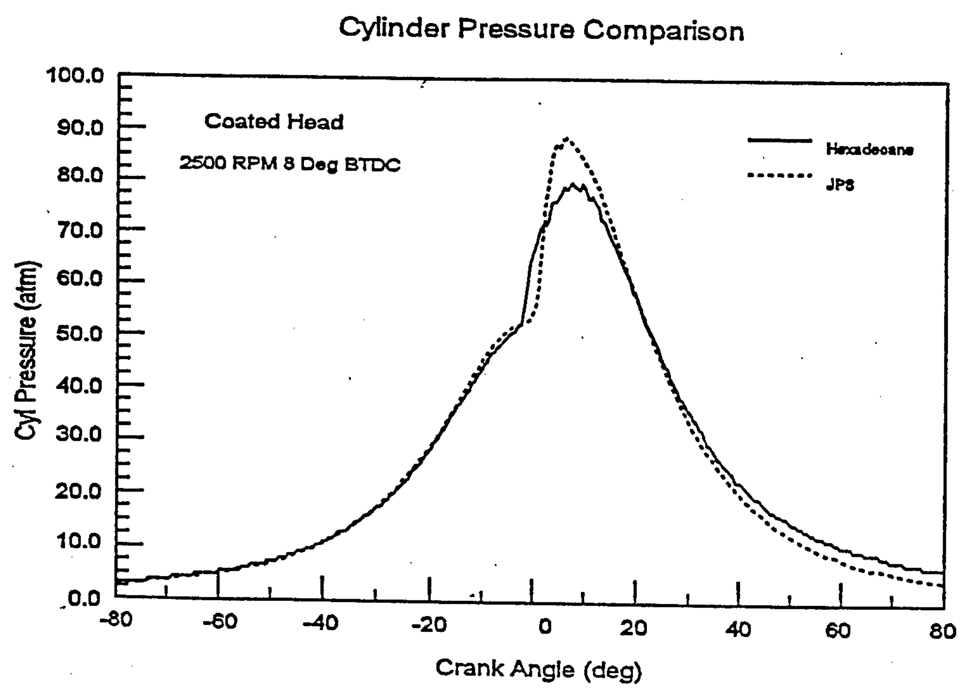

Figure 3.11(c) - P-Θ Comparison for JP-8 and Hexadecane for Coated Head Build

Cylinder Pressure Comparison

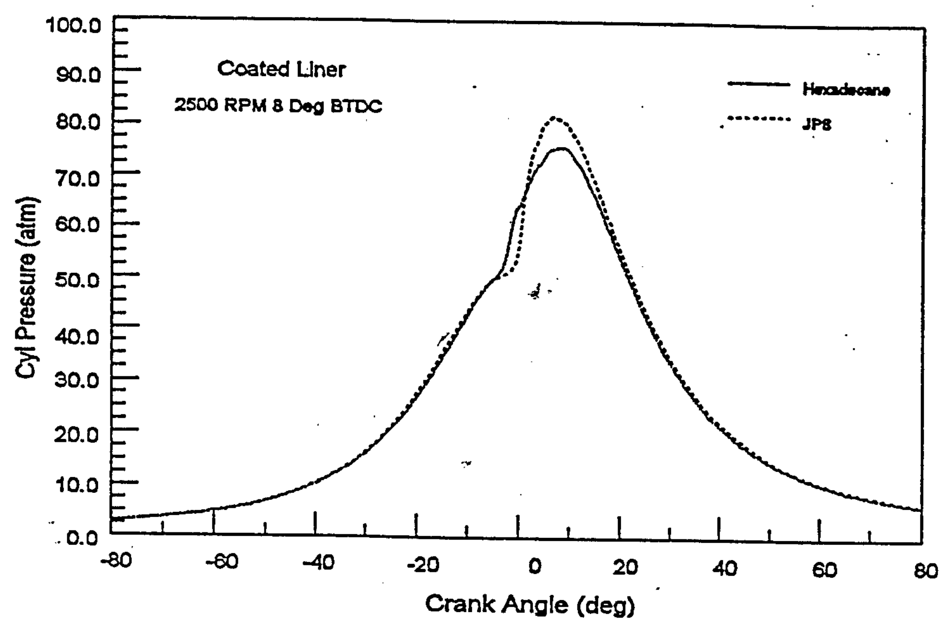

Figure 3.11(d) - P-Θ Comparison for JP-8 and Hexadecane for Coated Liner Build

Cylinder Pressure Comparison

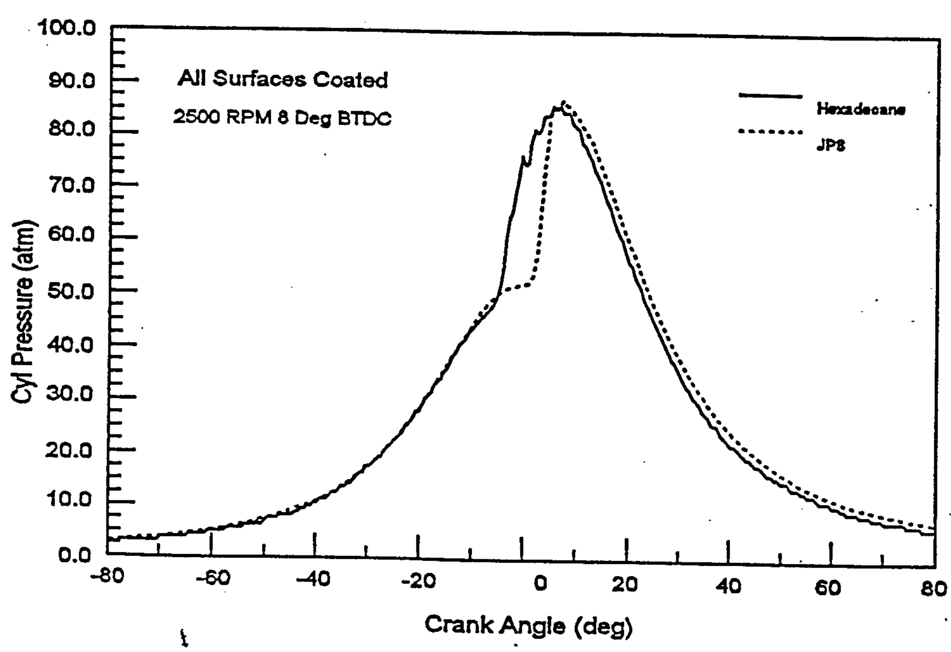

Figure 3.11(e) - P-Θ Comparison for JP-8 and Hexadecane for All-Surfaces Build

4. CONCLUSIONS

The engine performed satisfactorily on JP-8 fuel. No abnormal or adverse operating characteristics were observed, in either the baseline all-metal configuration, or with any of the insulation schemes investigated. It is therefore concluded that JP-8 is a viable alternative fuel for use in DI diesel engines with thin ceramic thermal barrier coatings applied to one or more combustion chamber surfaces.

For the Ricardo Hydra DI diesel engine, insulating the various surfaces offered mixed results with either performance or emissions being improved or compromised depending on the insulation scheme and the operating conditions. It was generally found that different operating conditions affected the engine in diverse ways, with speed, load, and to a lesser extent, injection timing, affecting the relative levels of emissions and BSFC for the various insulation schemes. It was decided to place greater emphasis on the high-speed, high-load condition because of the applicability to typical ground-based vehicle applications and because changes in BSFC and emissions have the greatest total impact under these conditions. Even then, no individual scheme was significantly more beneficial than any of the others.

Based on the results at 2500 RPM, 16 N-m load, and 8 °BTDC injection timing for JP-8 fuel (Figure 3.1, page 12), the percentage increase/decrease in BSFC, UHC, NO, and Soot for each of the insulation schemes compared to the baseline engine are given in Table 4.1.

Table 4.1 - Percent Increase/Decrease for Insulation Schemes Relative to Baseline For 2500 RPM, 16 N-m, 8 °BTDC Injection

Insulation Scheme	BSFC	UHC	NO	Soot
Coated Piston	-2.8%	+11.4%	-29.4%	+467%
Coated Liner	-2.8%	-17.1%	-17.6%	+677%
Coated Head	-4.6%	-66.1%	+17.6%	+61%
All Surfaces	+1.0%	-34.3%	+8.8%	-18%

It is obvious from Table 4.1 (and Figure 3.1) that the individual coating schemes investigated can produce improvements in one or more categories of emissions, but at the expense of an increase in another category. The choice of one insulation scheme over another, depends on which emission product is considered most critical. If hydrocarbons are the major emission concern, then coating the head is the best choice, with the added benefit of providing the greatest improvement in fuel economy. If oxides of nitrogen are of greatest importance, then coating the piston is the best choice. If soot is the major consideration, then coating all of the combustion chamber surfaces is the only configuration that consistently reduced soot emission below the baseline level, while also significantly improving UHC emissions (but at the expense of slightly increased NO and fuel consumption). Given all of the trade-offs represented by the data in Table 4.1, the best choices might well be either coating all surfaces of the combustion chamber to achieve reduced particulate and hydrocarbon emissions with little effect on NO emission and fuel consumption, or coating the head alone to achieve the improved fuel economy and UHC emission.

Other combinations of insulated surfaces are possible, but were not investigated, i.e. coated piston and head together, coated piston and liner together, and coated head and liner together. These combinations should be investigated in the future for any possible advantage that they might offer.

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APPENDIX A

Plots of Brake Specific Fuel Consumption Versus Load
For Three Engine Speeds and Three Injection Timings

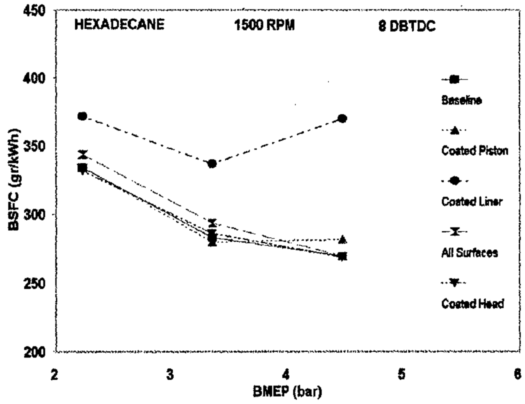

Figure A.1 - BSFC Vs Load for Hexadecane at 1500 RPM and 8 DBTDC

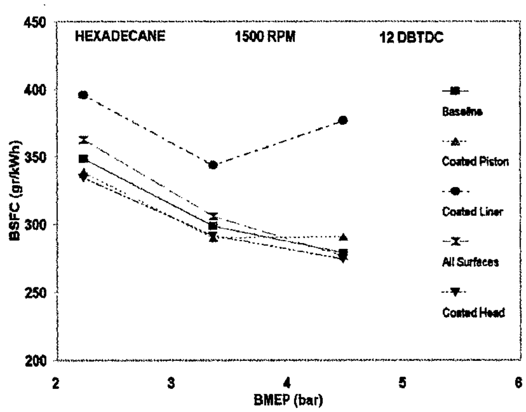

Figure A.2 - BSFC Vs Load for Hexadecane at 1500 RPM and 12 DBTDC

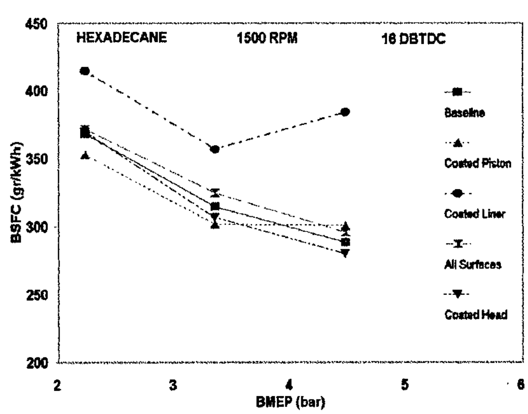

Figure A.3 - BSFC Vs Load for Hexadecane at 1500 RPM and 16 DBTDC

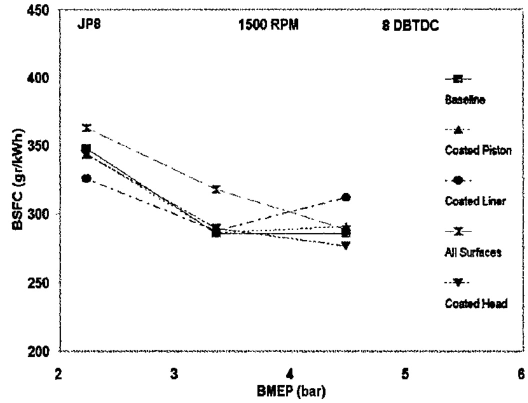

Figure A.4 - BSFC Vs Load for JP-8 at 1500 RPM and 8 DBTDC

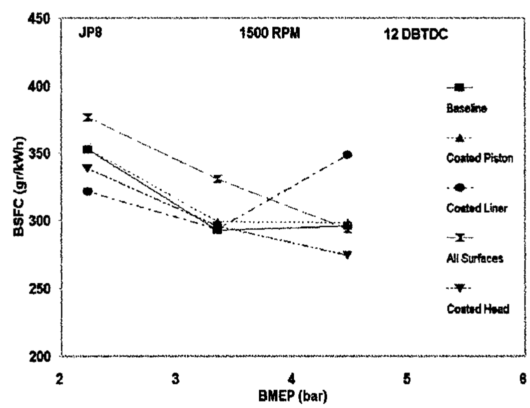

Figure A.5 - BSFC Vs Load for JP-8 at 1500 RPM and 12 DBTDC

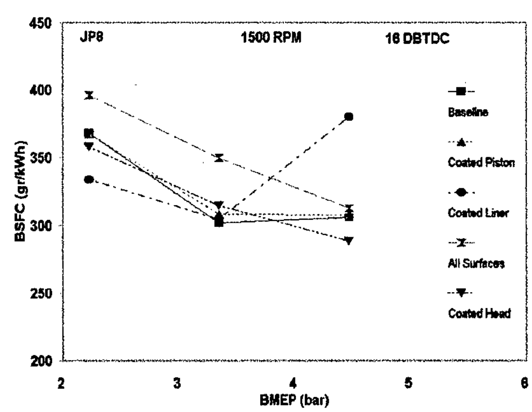

Figure A.6 - BSFC Vs Load for JP-8 at 1500 RPM and 16 DBTDC

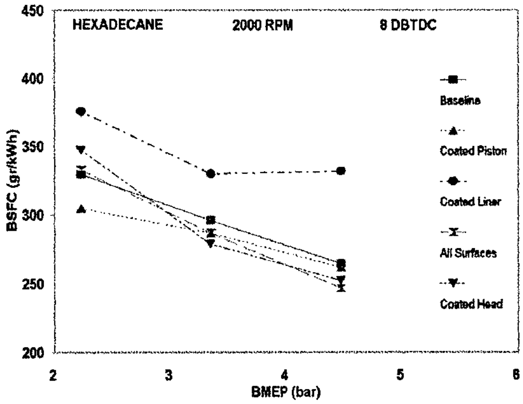

Figure A.7 - BSFC Vs Load for Hexadecane at 2000 RPM and 8 DBTDC

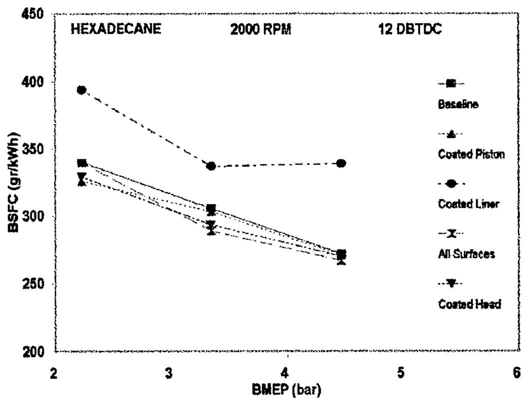

Figure A.8 - BSFC Vs Load for Hexadecane at 2000 RPM and 12 DBTDC

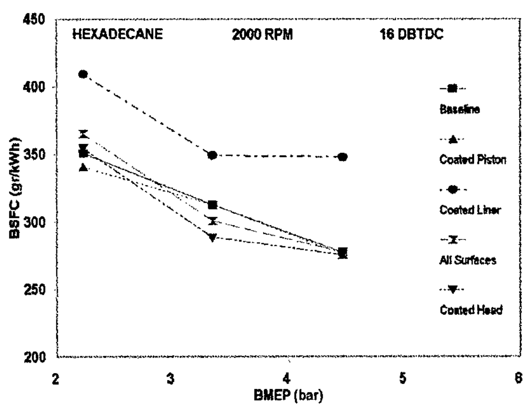

Figure A.9 - BSFC Vs Load for Hexadecane at 2000 RPM and 16 DBTDC

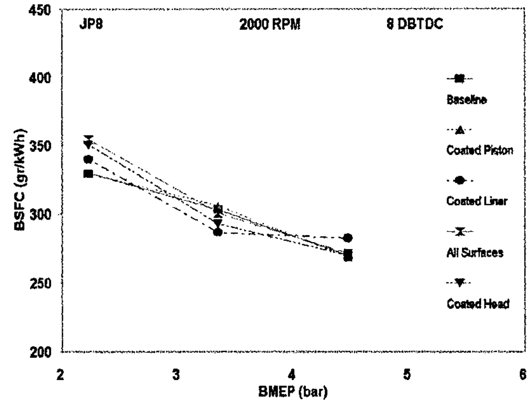

Figure A.10 - BSFC Vs Load for JP-8 at 2000 RPM and 8 DBTDC

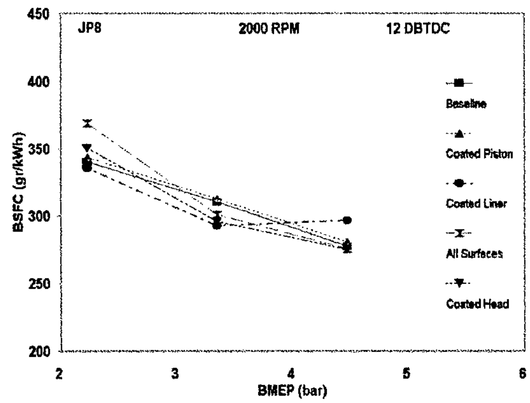

Figure A.11 - BSFC Vs Load for JP-8 at 2000 RPM and 12 DBTDC

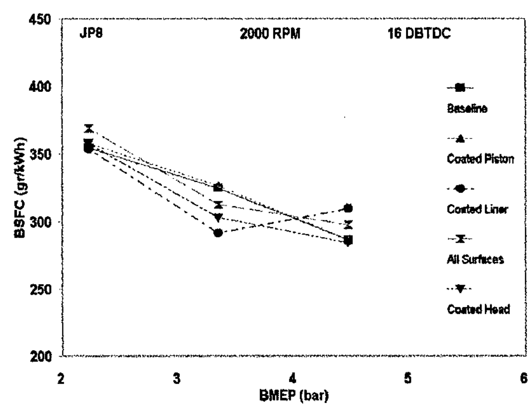

Figure A.12 - BSFC Vs Load for JP-8 at 2000 RPM and 16 DBTDC

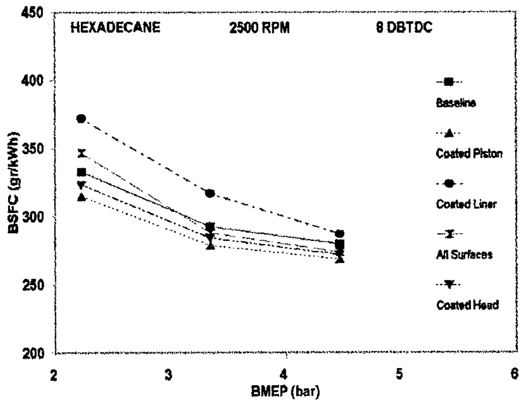

Figure A.13 - BSFC Vs Load for Hexadecane at 2500 RPM and 8 DBTDC

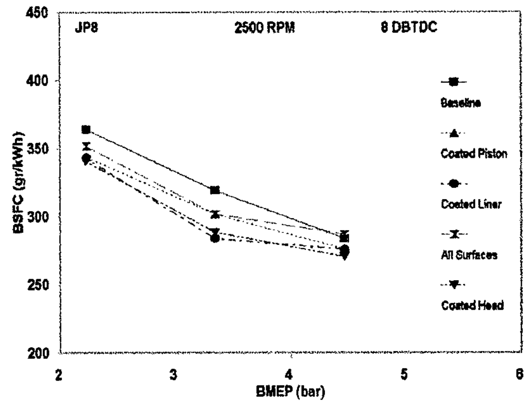

Figure A.16 - BSFC Vs Load for JP-8 at 2500 RPM and 8 DBTDC

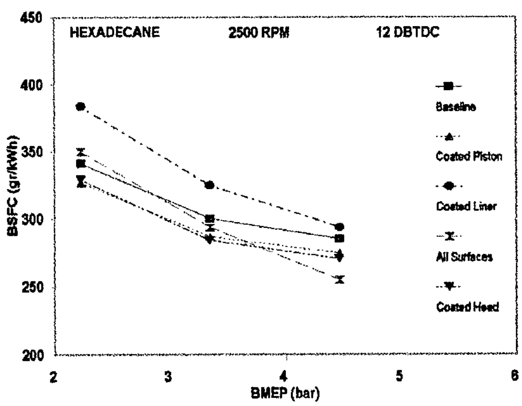

Figure A.14 - BSFC Vs Load for Hexadecane at 2500 RPM and 12 DBTDC

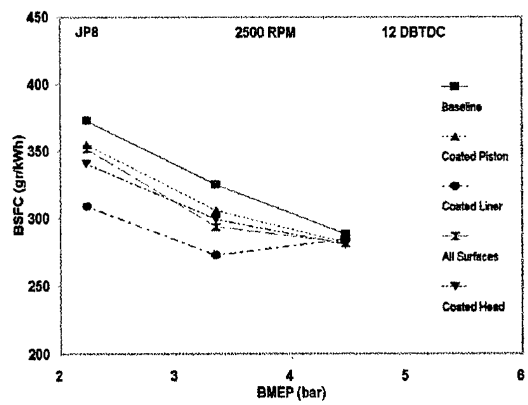

Figure A.17 - BSFC Vs Load for JP-8 at 2500 RPM and 12 DBTDC

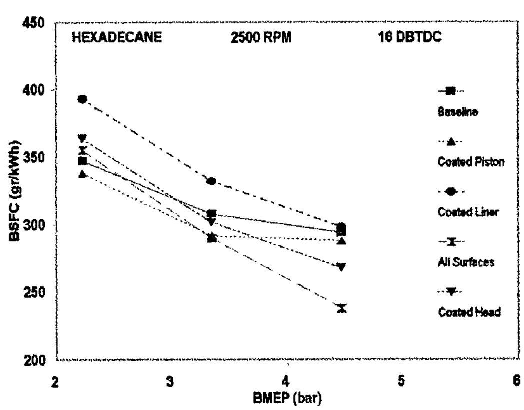

Figure A.15 - BSFC Vs Load for Hexadecane at 2500 RPM and 16 DBTDC

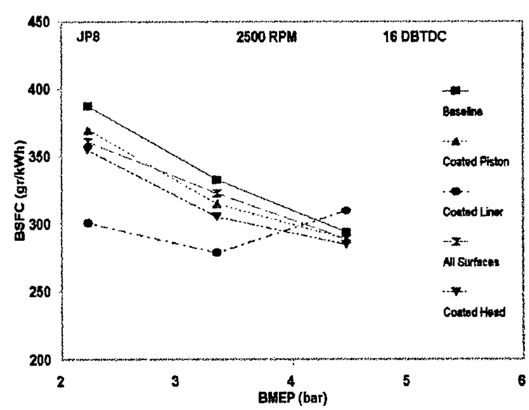

Figure A.18 - BSFC Vs Load for JP-8 at 2500 RPM and 16 DBTDC

APPENDIX B

Plots of Nitric Oxide Emission Versus Load
For Three Engine Speeds and Three Injection Timings

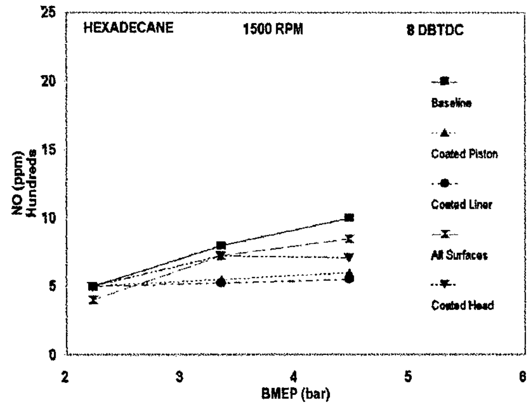

Figure B.1 - NO Vs Load for Hexadecane at 1500 RPM and 8 DBTDC

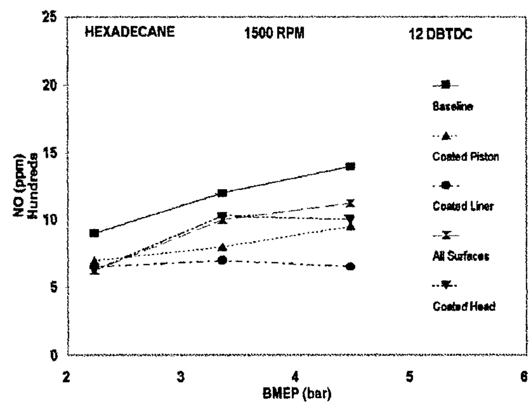

Figure B.2 - NO Vs Load for Hexadecane at 1500 RPM and 12 DBTDC

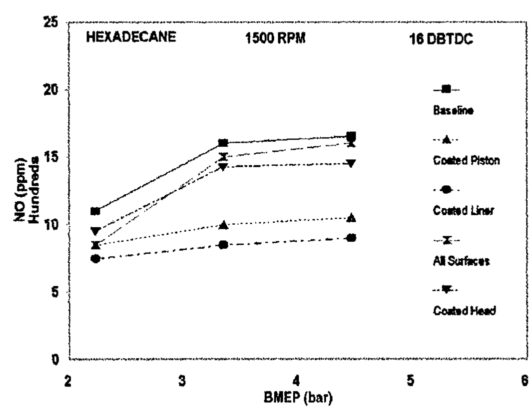

Figure B.3 - NO Vs Load for Hexadecane at 1500 RPM and 16 DBTDC

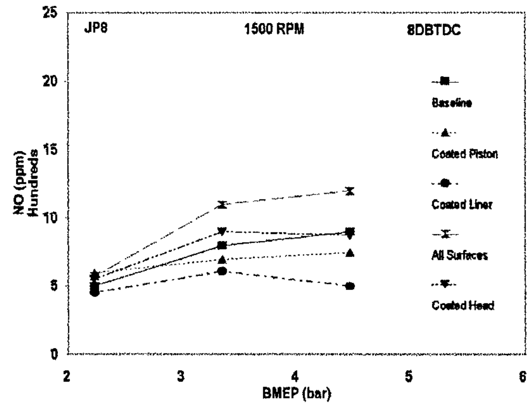

Figure B.4 - NO Vs Load for JP-8 at 1500 RPM and 8 DBTDC

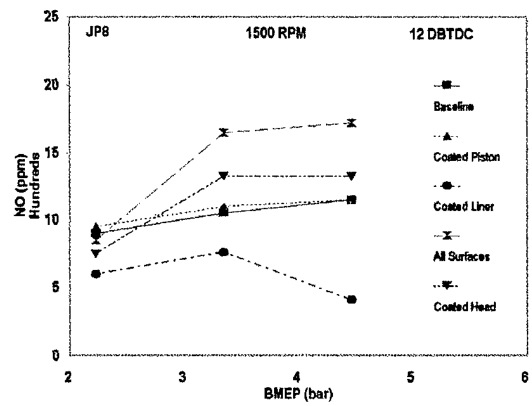

Figure B.5 - NO Vs Load for JP-8 at 1500 RPM and 12 DBTDC

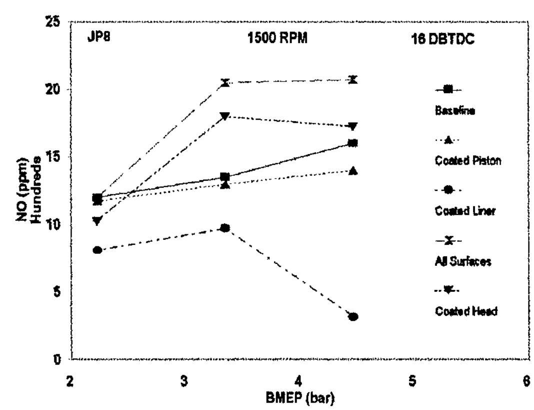

Figure B.6 - NO Vs Load for JP-8 at 1500 RPM and 16 DBTDC

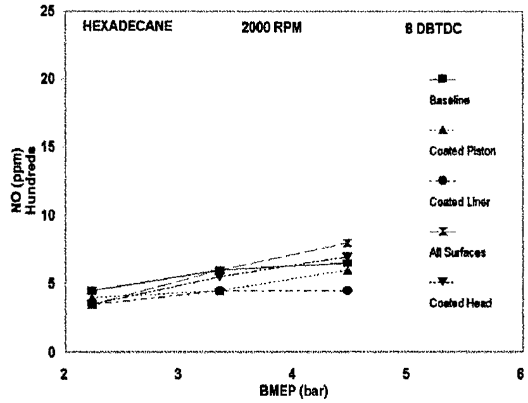

Figure B.7 - NO Vs Load for Hexadecane at 2000 RPM and 8 DBTDC

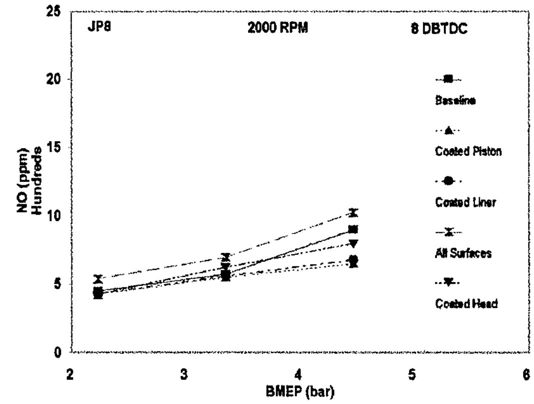

Figure B.10 - NO Vs Load for JP-8 at 2000 RPM and 8 DBTDC

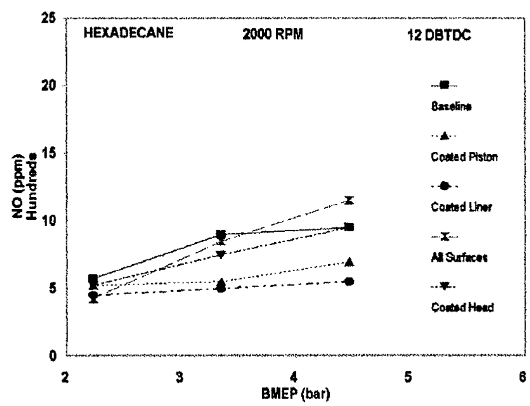

Figure B.8 - NO Vs Load for Hexadecane at 2000 RPM and 12 DBTDC

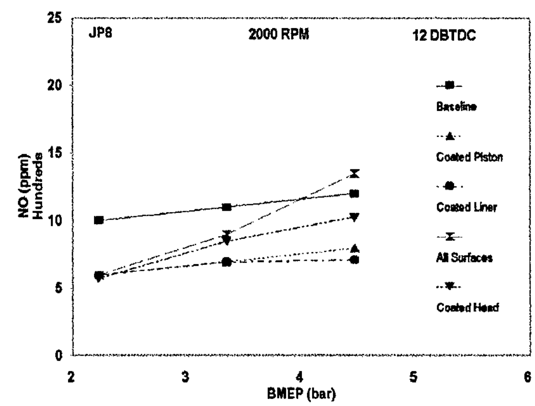

Figure B.11 - NO Vs Load for JP-8 at 2000 RPM and 12 DBTDC

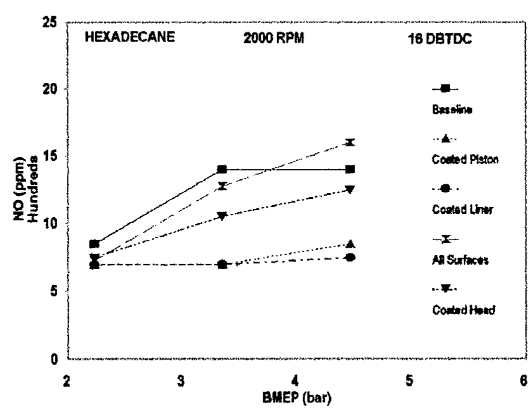

Figure B.9 - NO Vs Load for Hexadecane at 2000 RPM and 16 DBTDC

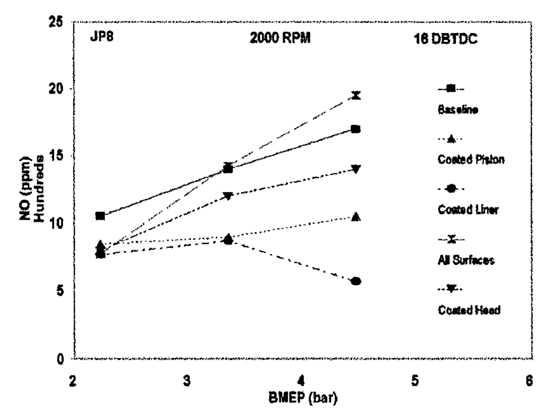

Figure B.12 - NO Vs Load for JP-8 at 2000 RPM and 16 DBTDC

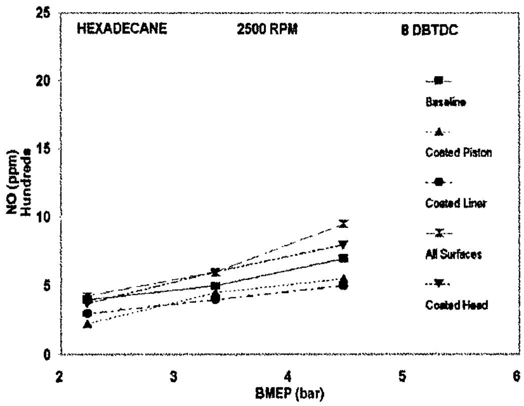

Figure B.13 - NO Vs Load for Hexadecane at 2500 RPM and 8 DBTDC

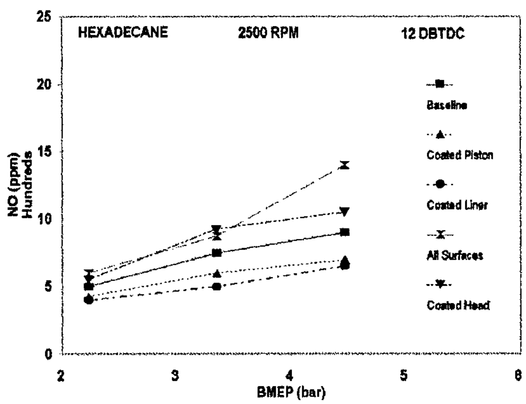

Figure B.14 - NO Vs Load for Hexadecane at 2500 RPM and 12 DBTDC

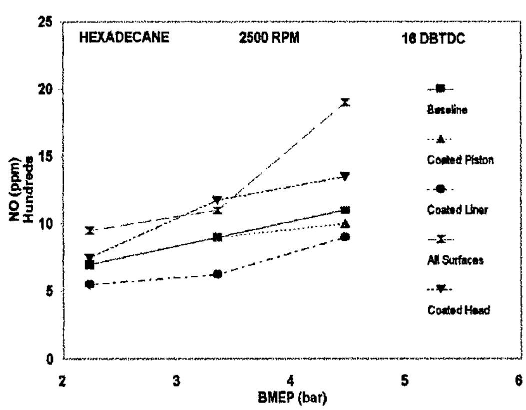

Figure B.15 - NO Vs Load for Hexadecane at 2500 RPM and 16 DBTDC

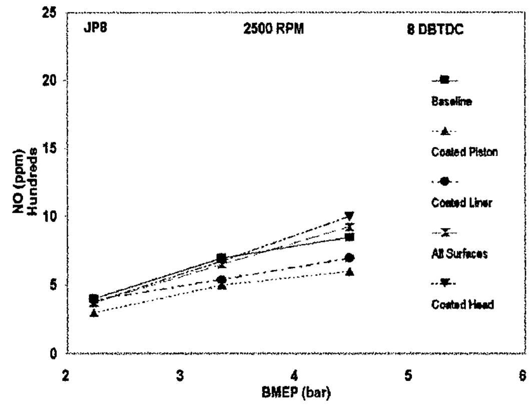

Figure B.16 - NO Vs Load for JP-8 at 2500 RPM and 8 DBTDC

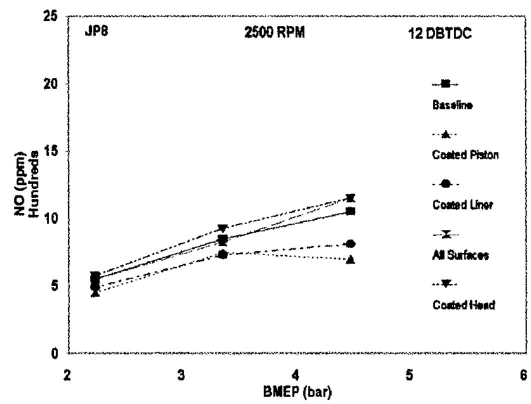

Figure B.17 - NO Vs Load for JP-8 at 2500 RPM and 12 DBTDC

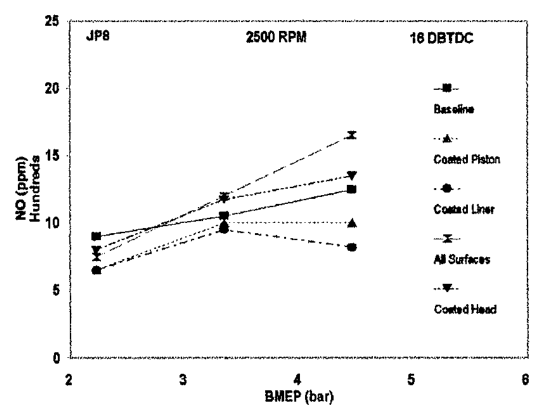

Figure B.18 - NO Vs Load for JP-8 at 2500 RPM and 16 DBTDC

APPENDIX C

Plots of Unburned Hydrocarbon Emission Versus Load
For Three Engine Speeds and Three Injection Timings

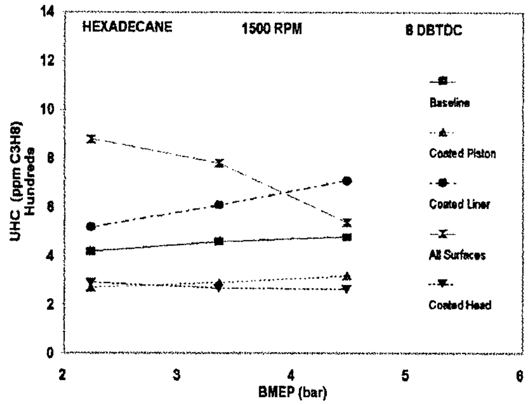

Figure C.1 - UHC Vs Load for Hexadecane at 1500 RPM and 8 DBTDC

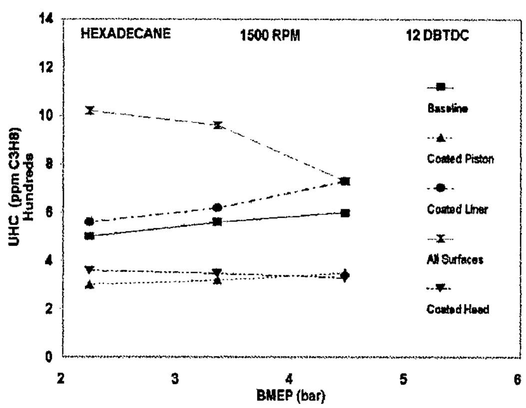

Figure C.2 - UHC Vs Load for Hexadecane at 1500 RPM and 12 DBTDC

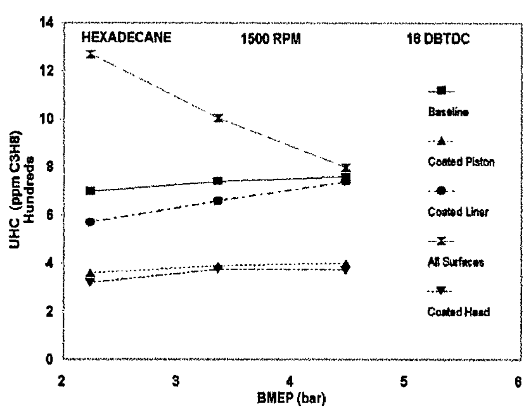

Figure C.3 - UHC Vs Load for Hexadecane at 1500 RPM and 16 DBTDC

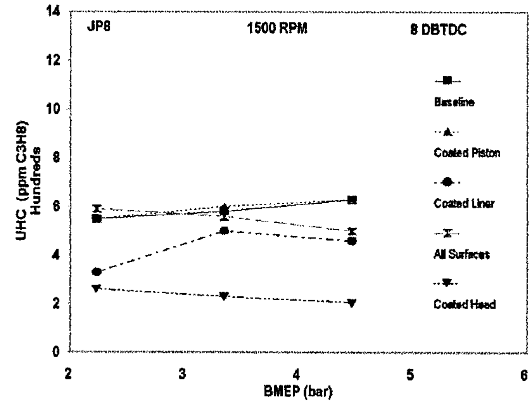

Figure C.4 - UHC Vs Load for JP-8 at 1500 RPM and 8 DBTDC

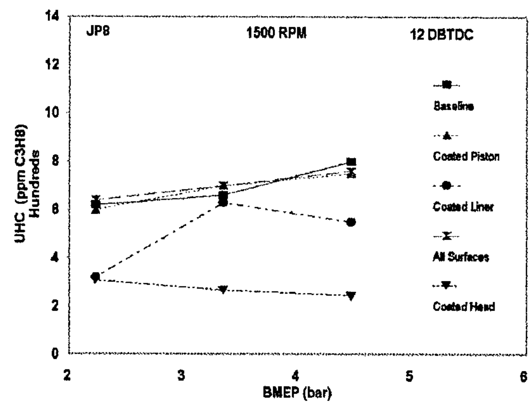

Figure C.5 - UHC Vs Load for JP-8 at 1500 RPM and 12 DBTDC

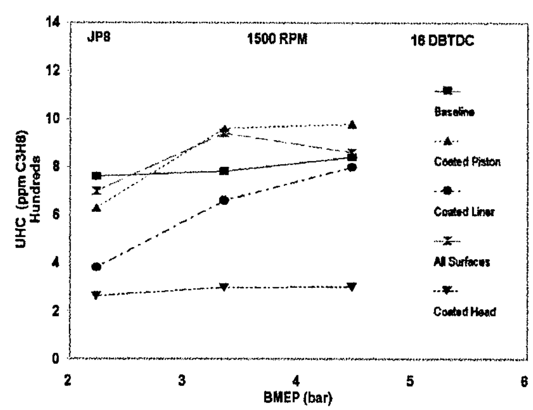

Figure C.6 - UHC Vs Load for JP-8 at 1500 RPM and 16 DBTDC

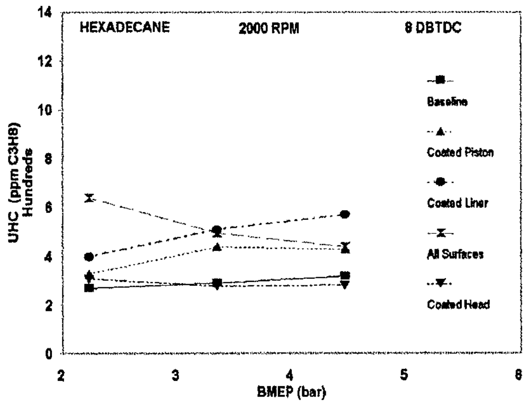

Figure C.7 - UHC Vs Load for Hexadecane at 2000 RPM and 8 DBTDC

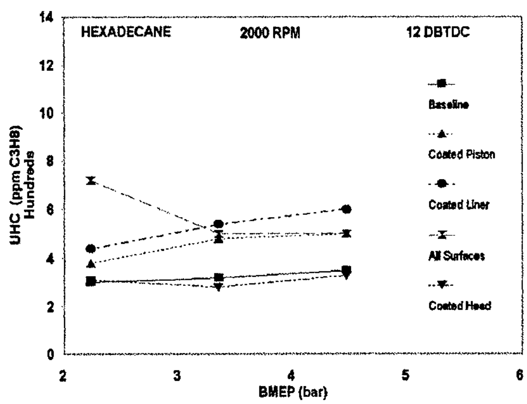

Figure C.8 - UHC Vs Load for Hexadecane at 2000 RPM and 12 DBTDC

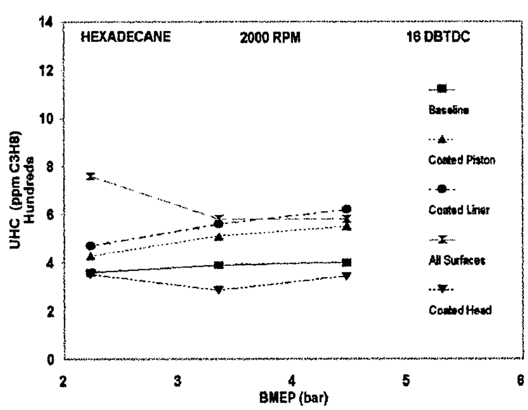

Figure C.9 - UHC Vs Load for Hexadecane at 2000 RPM and 16 DBTDC

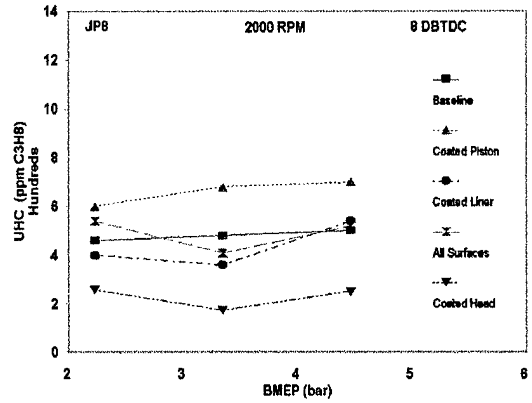

Figure C.10 - UHC Vs Load for JP-8 at 2000 RPM and 8 DBTDC

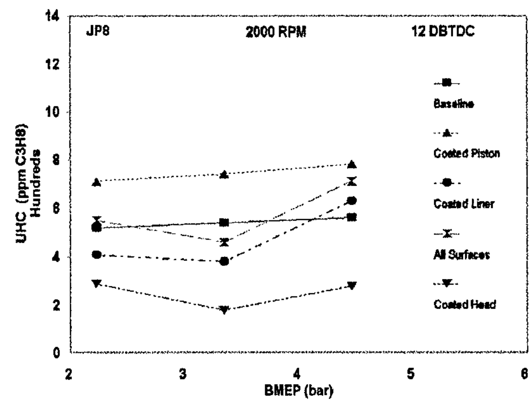

Figure C.11 - UHC Vs Load for JP-8 at 2000 RPM and 12 DBTDC

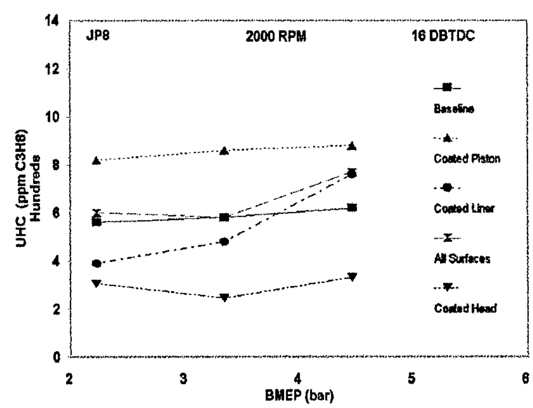

Figure C.12 - UHC Vs Load for JP-8 at 2000 RPM and 16 DBTDC

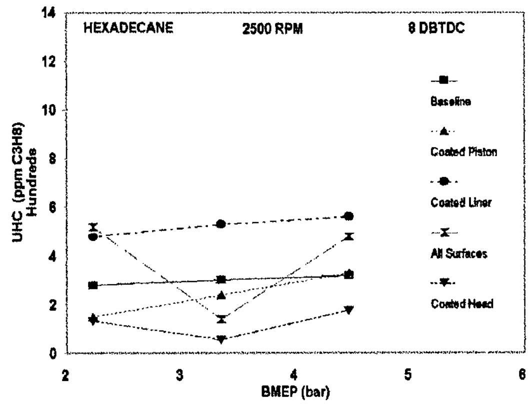

Figure C.13 - UHC Vs Load for Hexadecane at 2500 RPM and 8 DBTDC

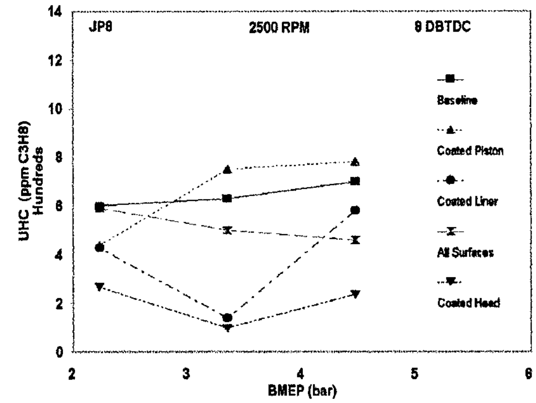

Figure C.16 - UHC Vs Load for JP-8 at 2500 RPM and 8 DBTDC

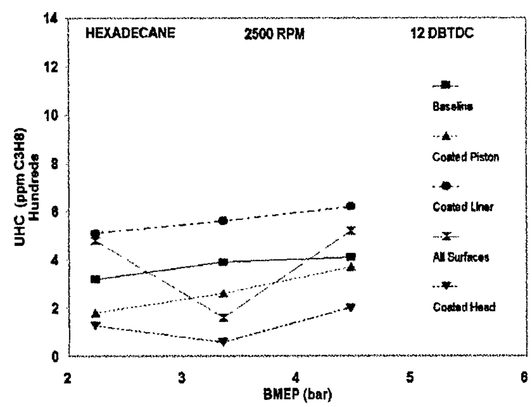

Figure C.14 - UHC Vs Load for Hexadecane at 2500 RPM and 12 DBTDC

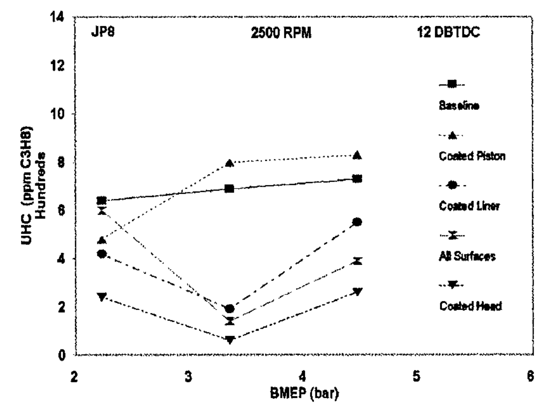

Figure C.17 - UHC Vs Load for JP-8 at 2500 RPM and 12 DBTDC

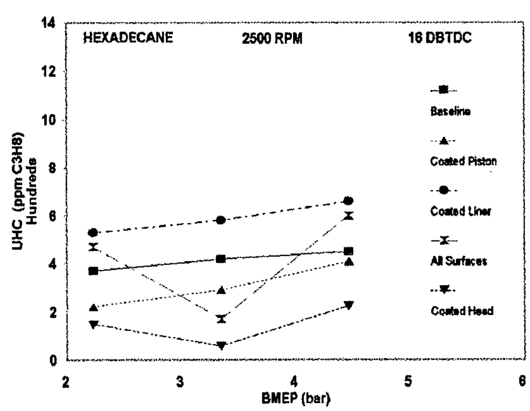

Figure C.15 - UHC Vs Load for Hexadecane at 2500 RPM and 16 DBTDC

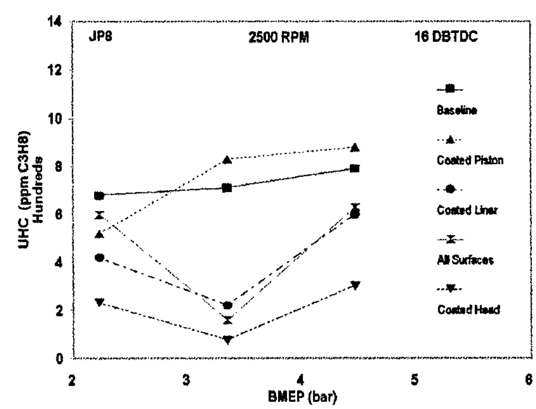

Figure C.18 - UHC Vs Load for JP-8 at 2500 RPM and 16 DBTDC

APPENDIX D

Plots of Soot Emission Versus Load
For Three Engine Speeds and Three Injection Timings

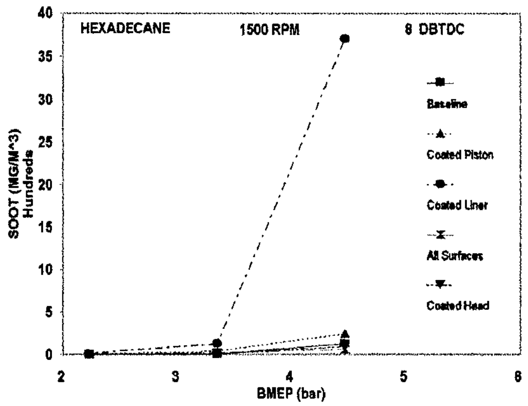

Figure D.1 - Soot Vs Load for Hexadecane at 1500 RPM and 8 DBTDC

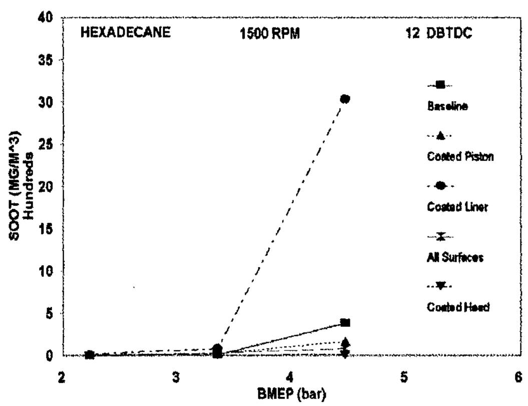

Figure D.2 - Soot Vs Load for Hexadecane at 1500 RPM and 12 DBTDC

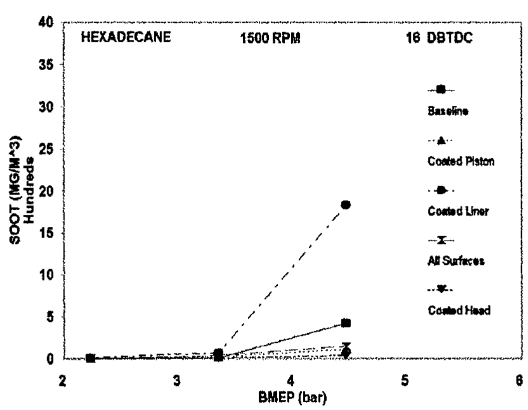

Figure D.3 - Soot Vs Load for Hexadecane at 1500 RPM and 16 DBTDC

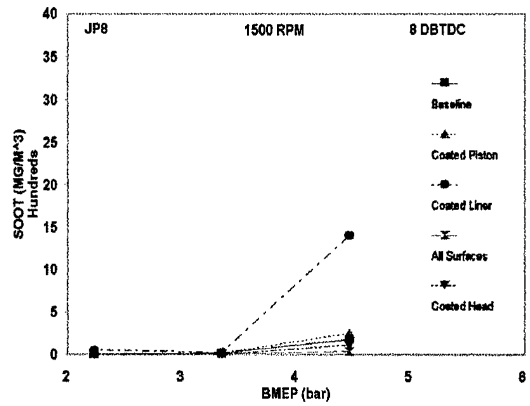

Figure D.4 - Soot Vs Load for JP-8 at 1500 RPM and 8 DBTDC

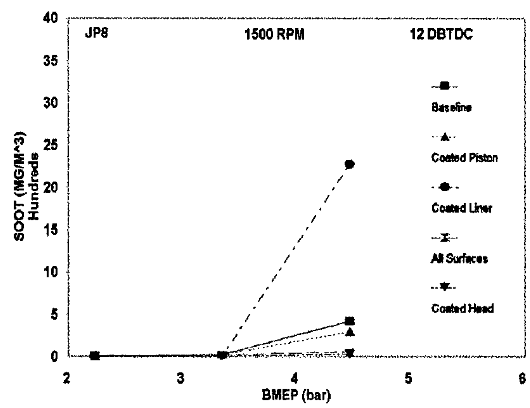

Figure D.5 - Soot Vs Load for JP-8 at 1500 RPM and 12 DBTDC

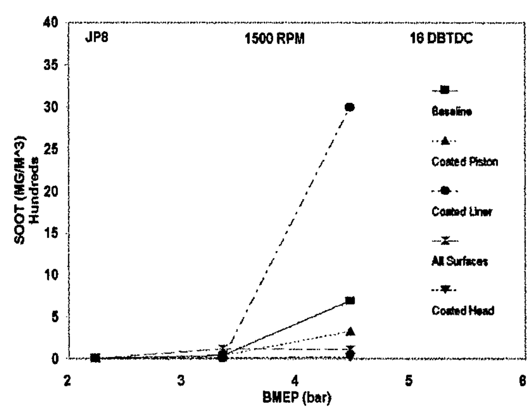

Figure D.6 - Soot Vs Load for JP-8 at 1500 RPM and 16 DBTDC

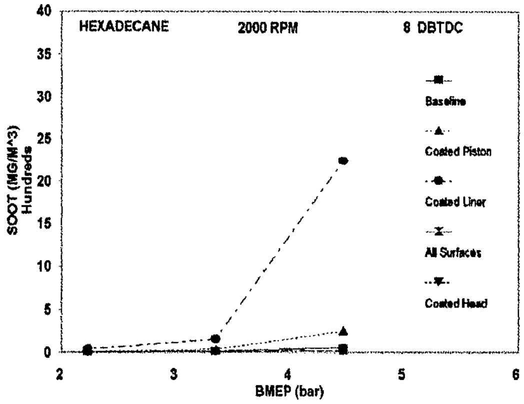

Figure D.7 - Soot Vs Load for Hexadecane at 2000 RPM and 8 DBTDC

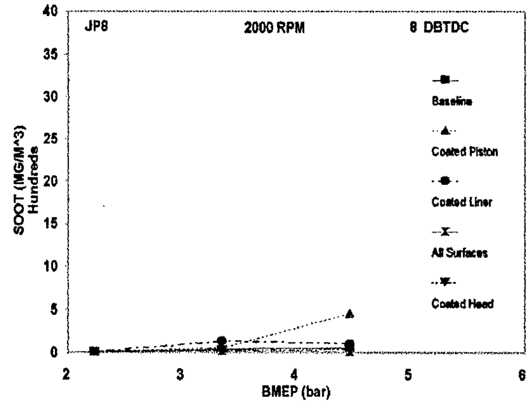

Figure D.10 - Soot Vs Load for JP-8 at 2000 RPM and 8 DBTDC

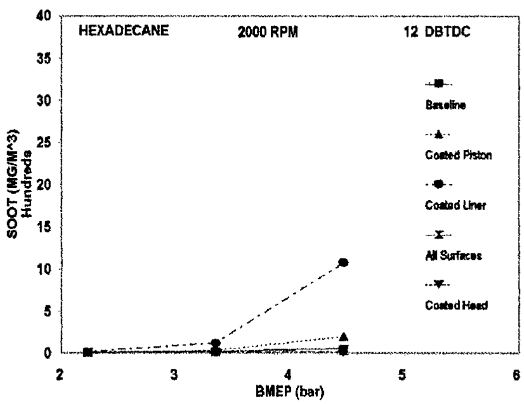

Figure D.8 - Soot Vs Load for Hexadecane at 2000 RPM and 12 DBTDC

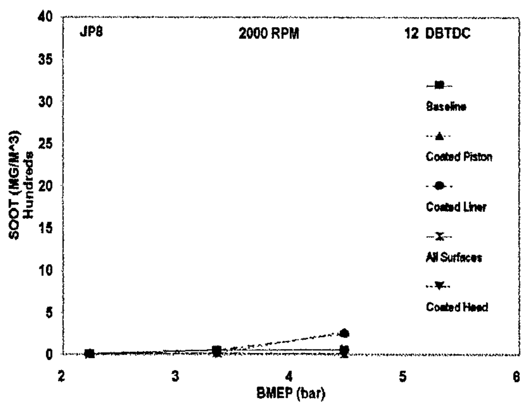

Figure D.11 - Soot Vs Load for JP-8 at 2000 RPM and 12 DBTDC

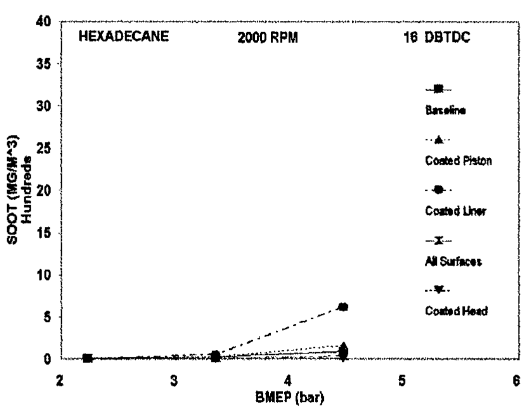

Figure D.9 - Soot Vs Load for Hexadecane at 2000 RPM and 16 DBTDC

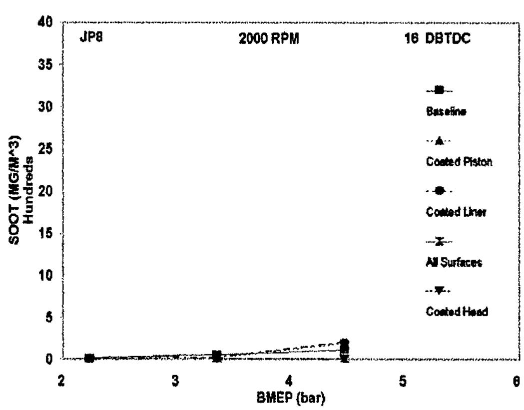

Figure D.12 - Soot Vs Load for JP-8 at 2000 RPM and 16 DBTDC

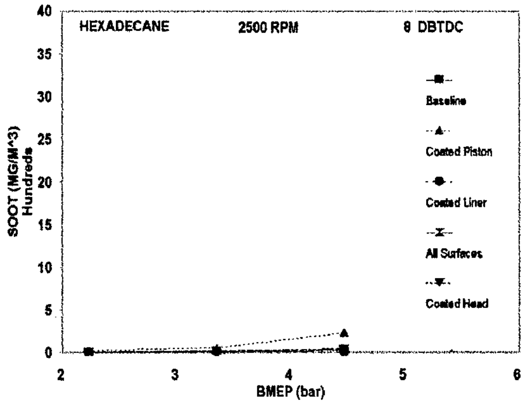

Figure D.13 - Soot Vs Load for Hexadecane at 2500 RPM and 8 DBTDC

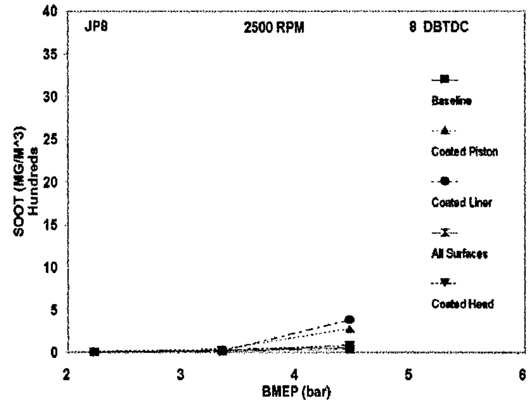

Figure D.16 - Soot Vs Load for JP-8 at 2500 RPM and 8 DBTDC

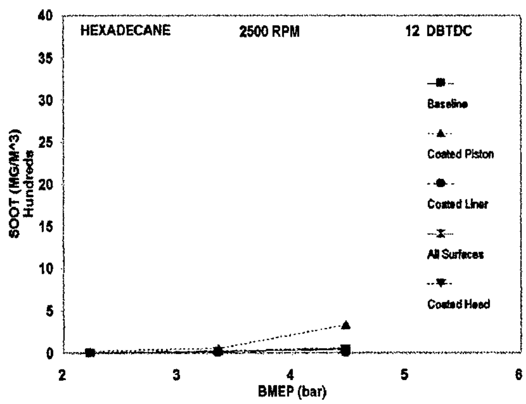

Figure D.14 - Soot Vs Load for Hexadecane at 2500 RPM and 12 DBTDC

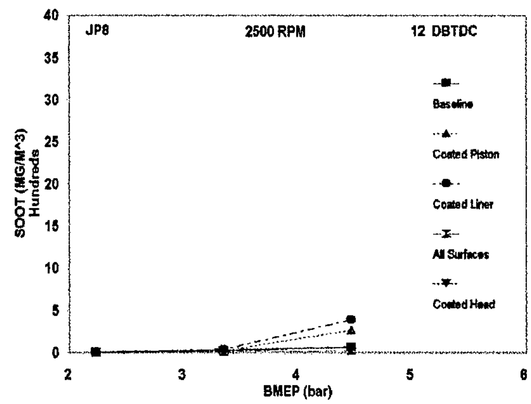

Figure D.17 - Soot Vs Load for JP-8 at 2500 RPM and 12 DBTDC

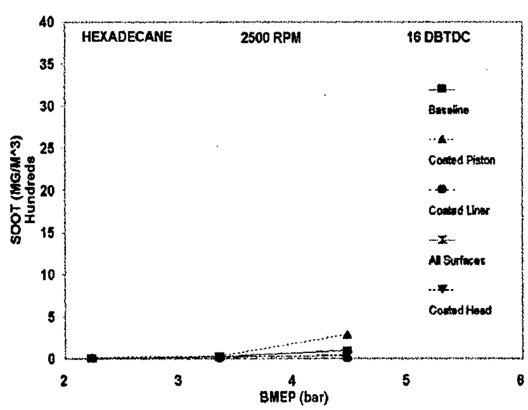

Figure D.15 - Soot Vs Load for Hexadecane at 2500 RPM and 16 DBTDC

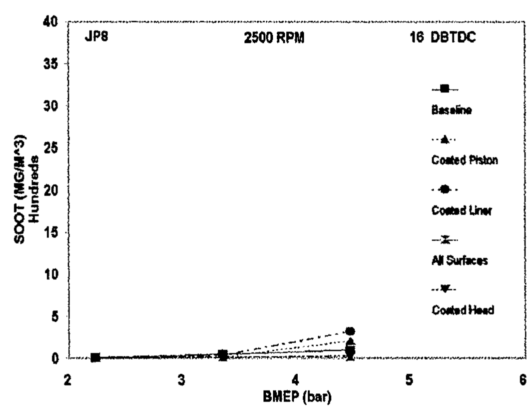

Figure D.18 - Soot Vs Load for JP-8 at 2500 RPM and 16 DBTDC