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APPLICATION OF ON-HIGHWAY EMISSIONS REDUCTION TECHNOLOGY TO AN OFF-HIGHWAY ENGINE

FINAL REPORT

VOLUME I

SwRI Project No. 03-3354-200

Prepared for:

Santa Barbara County Air Pollution Control District
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1. ACKNOWLEDGEMENTS

Southwest Research Institute gratefully acknowledges the assistance and contributions made by the following companies and organizations in the successful completion of this project.

Caterpillar Inc.
EXXON Company, USA
Quinn Engine Systems
Santa Barbara County Air Pollution Control District
Santa Barbara County Department of Public Works

2. EXECUTIVE SUMMARY

An investigaton was performed to determine the effects of applying on-highway heavy-duty diesel engine emissions reduction technology to an off-highway version of the engine. Special attention was paid to the typical constraints of fuel consumption, heat rejection, packaging and cost-effectiveness.

The primary focus of the effort was NO_x reduction while hopefully not worsening other gaseous and particulate emissions. Hardware changes were limited to "bolt-on" items, thus excluding piston and combustion chamber modifications.

In the final configuration, NO_x was improved by 28 percent, particulates by 58 percent, CO and HC were also better and the fuel economy penalty was limited to under 4 percent.

The engine was installed in a Caterpillar 637E scraper in a landfill site in Santa Barbara, California. This project is a part of a major Low NO_x Demonstration Program being conducted by the Air Pollution Control District and EXXON. It covers mobile sources, construction equipment and offshore equipment.

Observations are made about the effectiveness of various individual and combined strategies, and potential problems are identified.

3. RECOMMENDATIONS

Clearly, the application of on-highway engine experience and hardware can be used to accelerate the development process of improving emissions from off-highway engines. How far up the learning curve one can begin the process depends on how similar the base engines are and how much of a technology gap exists between them.

For the 3306 DITA scraper engine that was the subject of this investigation, three modifications are most influential.

- **Change to the high pressure injection system** - This consists of new injectors with fewer holes in the nozzle tip which improves both NO_x and particulates substantially. To work properly, the system requires new injection fuel lines and a higher pressure injection pump as well.

- **Retard injection timing by 2 degrees** (referenced at crankshaft) - This can be accomplished at the same time the injection system modifications are made. While further retard continues to improve NO_x emissions, penalties in fuel economy and engine heat rejection to coolant gets worse quite rapidly.

- **Change to reformulated diesel fuel** - The substitution of a modified diesel fuel (in this case, Chevron "Special Diesel") improved HC emissions by about 16 percent. The test fuel also had a higher cetane number, which is directionally right for good combustion, although no significant changes in any other emissions or performance parameter were measured.

An overall summary of these changes, individually and in combination, is shown in Table 3-1.

4. PROGRAM DEVELOPMENT

4.1 Introduction

As on-highway engines have become "cleaner" to meet increasingly stringent emissions requirements, the remaining off-highway emissions have become a relatively more significant contributor to non-attainment of air quality goals. For this reason, and to improve air quality even further, it is appropriate to turn attention to off-highway sources.

Fortunately, the work accomplished to date to reduce emissions from on-highway engines is in large part transferrable to off-highway engines, subject to certain limitations explored below.

The purpose of this program was to apply such technology to a Caterpillar 3306B DITA engine used in a Model 637E scraper to improve primarily, NO_X and particulate emissions.

4.2 Background

Emissions, both gaseous and particulate, from on-highway engines have been regulated for some time, the most recent tightening of regulations taking effect for the 1991 model year. Assessment of emissions for on-highway heavy-duty engines is accomplished using a transient test cycle as described in Federal Test Procedure Title 40, Code of Federal Regulations, Part 86, Subpart N - "Emissions Regulations for new Otto cycle and diesel heavy-duty engines; gaseous and particulate exhaust test procedures." Hence, these tests are often called "FTP transients." However, for off-highway engines, CARB has proposed an 8-Mode test procedure outlined later in this section. Along with the proposal for a steady-state test procedure, was a set of emission requirements which also differed somewhat from those for FTP on-highway use, both because equivalency between the 8-mode and the FTP had not (still has not to any significant degree) 17 been established. Table 4.2-1 shows the requirements for 1995 and 2000 and beyond.

TABLE 3-1 SUMMARY OF CHANGES

Base: Injectors	8N7005 (9 hole)	Base Emissions (g/bhp-hr):	NO_x	6.61	Base Fuel Consumption	219 g/kW-hr
JWAC	88°C manifold temperature		NMHC	1.56		
Timing	14° BTDC (spill port)		PM_{10}	0.27	Rated Power	201 kW
					Speed	2200 rpm
			CO	1.62	Peak Torque @ Speed	509 Nm 1400 rpm

	NO_x				Impact on Other Pollutant Emissions			
NO_x Control Technology	g/bhp-hr	% Reduction	Hardware/Labor Cost	BSFC g/kW-hr	HC g/bhp-hr	CO g/bhp-hr	Part g/bhp-hr	Comments
Individual								
High pressure injection (on-highway injector 7W7038)	5.8	12	(a)	229	0.72	1.04	0.14	Std. JWAC, std. timing
Charge air cooling (from 88°C - 43°C),	6.1	8		218	0.95	1.68	0.21	Std. JWAC, std. injector
Injection timing retard (2 deg)(b)								Std. inj., std. timing
Truck turbocharger(c)	6.7	-1		217	1.62	1.57	(e)	Std. inj. std. timing, std. injectors
Fuel composition (Chevron, special diesel, high cetane 48.6)(c)	6.6	(d)		219	1.26	1.59	(e)	Std. inj., std. timing, std. inj.
Combination								
Hp inj. (7W7038) + inj. timing retard (2°)	4.8	27		228	0.45	0.84	0.11	Std. JWAC
HP inj (7W7038) + inj. timing retard (2°) + inj (special diesel (48.6))(c)	4.8	27		228	0.37	0.86	0.11	Std. JWAC
HP Inj. + inj. timing retard + charge cooling(c)	3.7	41		227	(f)	0.81	0.12	

(a) High pressure injection pump, hardened injection lines, injectors
(b) Not tested with base injectors
(c) Predicted, from test with truck injectors
(d) Difference not statistically significant
(e) Not measured
(f) Insufficient data to predict

equivalency between the 8-mode and the FTP had not (still has not to any significant degree) been established. Table 4.2-1 shows the requirements for 1995 and 2000 and beyond.

TABLE 4.2-1 CARB PROPOSED OFF-HIGHWAY STANDARDS

Year	HC g/hp-hr	CO g/hp-hr	NO_x g/hp-hr	PM g/hp-hr
1995 - 1999	1.0	(1)	6.9	0.40
2000 →	1.0	(1)	5.8	0.16
(1) Not specified, presumed to remain at on-highway: 15.5				

The purpose of this project was to demonstrate the application of on-highway technology to an off-highway rating of a CAT 3306B DITA diesel engine with the primary goal in mind of meeting requirements of 5 g/bhp-hr NO_X and 0.25 g/bhp-hr particulates. Note that these differ from both of the finally adopted sets of regulations listed in Table 4.2-1. They are in fact the on-highway requirements for 1991-1994 vehicles.

4.2.1. Transient Test Cycle

Transient engine operation is performed in accordance with the 1984 Transient Federal Test Procedure for Heavy-Duty Diesel Engines. The procedure specifies a transient engine exercise of variable speed and load, depending on the power output capabilities of the test engine. The cycle requires relatively rapid dynamometer control, capable of loading the engine one moment and motoring it the next. The system consists of a motoring/absorbing dynamometer with a suitable control system. The test operator's control station contains computer, operator keyboard, analog recorder, and constant volume sampling (CVS) control panel.

The 1984 Transient cycle is described in the Federal Register by means of percent maximum torque and percent rated speed for each one second interval, for a test cycle of 1199 seconds duration. The 20-minute transient cycle, developed from heavy-duty truck data, is composed of four five-minute segments. The four segments are described below.

To generate the transient cycle for an engine, the engine's full power curve is obtained from low idle rpm to maximum no load engine speed. Data from this "power curve," or engine map, is used in conjunction with the specified speed and load percentages to form the transient cycle.

Transient Cycle Segment	Time (sec)
New York Non-Freeway (NYNF)	297
Los Angeles Non-Freeway (LANF)	300
Los Angeles Freeway (LAF)	305
New York Non-Freeway (NYNF)	297

A graphic presentation of speed and torque commands which constitute an FTP transient cycle for a particular 250 hp diesel engine is given in Figure 4.2.1-1 for illustration purposes. Another way of showing this is the relative time spent at various conditions as displayed in

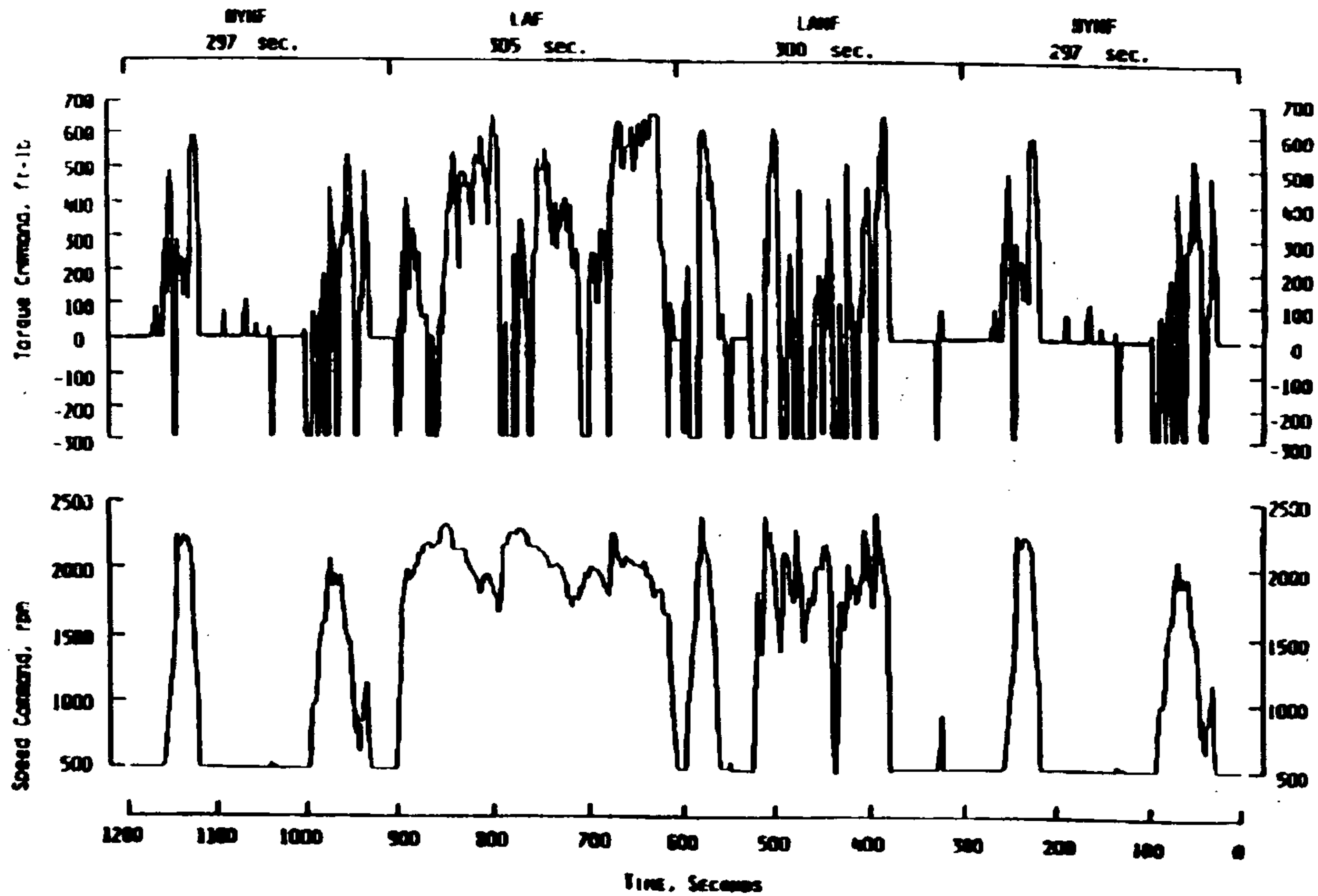


FIGURE 4.2.1-1 GRAPHIC REPRESENTATION OF TORQUE AND SPEED COMMANDS FOR THE 1984 TRANSIENT FTP CYCLE FOR A 250 hp AT 2400 rpm DIESEL ENGINE

Figure 4.2.1-2. For this example, the resulting cycle work was 15.66 hp-hr (11.68 kW-hr), based on a peak torque of 650 ft-lbs (880 N m) and a rated speed of 2200 rpm. The relatively large negative torque commands shown in the figure are to insure that the "throttle," or rack control, goes closed for motoring operation.

A "Transient FTP Test" consists of a cold-start transient cycle and a hot-start transient cycle. The same engine control or command cycle is used in both cases. For the cold-start, the engine is operated over a "prep" or practice cycle, then allowed to stand overnight in an ambient soak temperature of 20 to 30°C (68 to 86°F). The cold-start transient cycle begins when the engine is cranked for cold start-up. Upon completion of the cold-start transient cycle, the engine is shut down and allowed to stand for 20 minutes. After this hot soak period, the hot-start cycle begins with engine cranking. Transient composite results from cold- and hot-start transient are computed by the following:

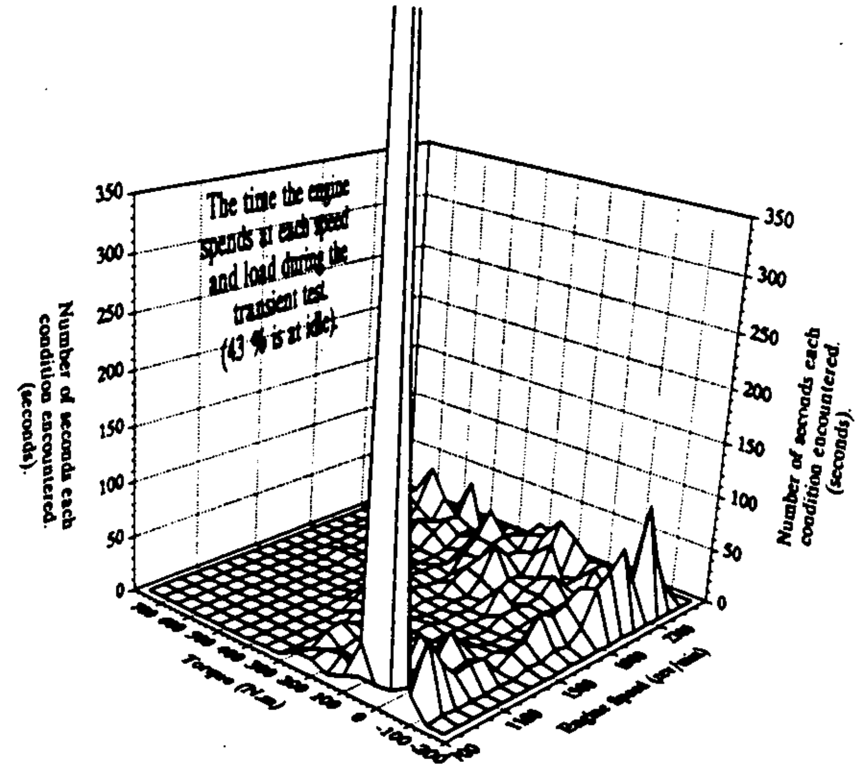


FIGURE 4.2.1-2 RELATIVE TIME SPENT AT MODES IN TRANSIENT CYCLE

$$\text{Brake Specific Emissions} = \frac{\frac{1}{7}\,(\text{Mass Emissions, Cold}) + \frac{6}{7}\,(\text{Mass Emissions, Hot})}{\frac{1}{7}\,(\text{Cycle Work, Cold}) + \frac{6}{7}\,(\text{Cycle Work, Hot})}$$

4.2.2. CARB 8-Mode Test Cycle

In California Air Resources Board Mail-out 90-62, issued in October 1990, it was first proposed to use a steady-state modal analysis technique to quantify gaseous and particulate emissions from heavy-duty off-highway vehicles. The test modes and weighting factors were determined after studying input from sources such as EEA (Energy and Environmental Analysis) and EMA (Engine Manufacturers Association). The resulting procedure has to date, only been defined as shown in Table 4.2.2-1. It consists of eight engine operating conditions ranging from low idle to full power, and includes a mode (the last) at 10 percent load at engine rated speed which approximates a light load condition such as a high speed transition to "overrun" or "trailing throttle".

TABLE 4.2.2-1 CARB 8-MODE OFF-HIGHWAY TEST MATRIX

Mode	Speed	Percent Load	Weighting Factor
1	Idle	None	0.15
2	Torque Peak	50	0.10
3	Torque Peak	75	0.10
4	Torque Peak	100	0.10
5	Rated Power	100	0.15
6	Rated Power	75	0.15
7	Rated Power	50	0.15
8	Rated Power	10	0.10

The proposed cycle is identical to the ISO-8178 Cycle B being proposed in Europe for use in off-highway emissions assessment as well.

The modes and weighting factors shown in the table were used for the steady-state testing reported herein. No standards have been published pertaining to engine temperatures, dwell times at mode, reference fuels etc., so for the tests performed in this project, simple good engineering judgement and laboratory practices were used. Particulates were sampled using a "splitter tunnel" as shown in Figure 4.2.2-1.

4.3. Engine Description

The 3306B, chosen for this emissions reduction program, powers the scraper axle of the tandem-powered 637E Wheel Tractor Scraper. The demonstration vehicle is equipped with laminated thermo shields for waste disposal applications. The laminated thermo shields cover high temperature engine componetns and reduce their surface temperatures to below the ignition temperatures for common combustibles. A 3408 DI-TA Jacketwater Aftercooled engine rated at 473 Gross HP at 2000 RPM powers the 673 Tractor. The 112,000 pound 637E has a 31 cubic yard heaped capacity and a top speed of 30 miles per hour.

Development of the direct-injected 3306B began in the mid 1970's. The 3306B began replacing the precombustion chamber 3306 engine in Caterpillar earthmoving vehicles in 1980. In 1982 Caterpillar introduced the jacketwater aftercooled 3306B in the heavy-duty on-highway truck market. Naturally-aspirated, turbocharged and aftercooled versions of the 3306B power 21 percent of all Caterpillar earthmoving vehicles. The 10.45 liter 3306B is a four-cycle engine with a quiescent combustion system. The engine has a scroll metering-type fuel system, three-ring aluminum alloy pistons, and a two-valve cylinder head with integral cast iron intake manifold.

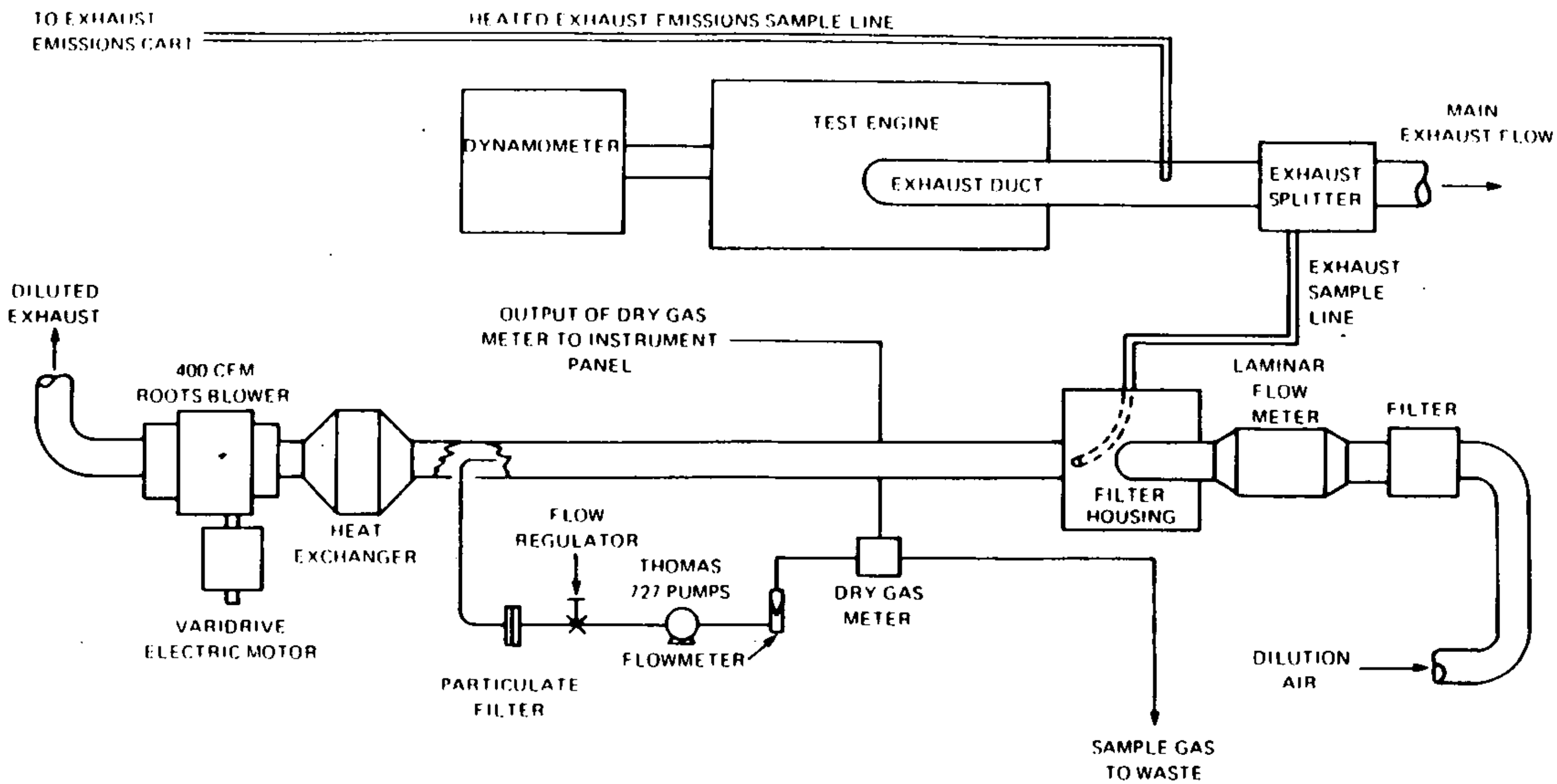


FIGURE 4.2.2-1 TYPICAL TRANSIENT CELL INSTRUMENTATION

TABLE 4.3-1 3306 DITA SPECIFICATIONS

Output Power	201 kW (270 hp) at 2200 rpm
Peak Torque	509 N.m (829 lb/ft) at 1400 RPM
Brake Specific Fuel Consumption at Rated Speed	219 g/kW-hr (360 lb/hp-hr)
Injection Timing	14°BTDC (small port)
Injector Type	Caterpillar pencil - 8N7005
Hole Geometry	9 holes, 0.205 mm diameter
Sac and Hole Volume	2.33 mm^3
Inlet Manifold Temperature at Rated	88°C (191°F) JWAC
Maximum Inlet Restriction	3.4 kPa (1" Hg) clean
Maximum Exhaust Restriction	6.7 kPa (1 psi)

4.4. Test Program

The test program will be briefly described in terms of phases that evolved, then more detailed discussion of the test results themselves will be covered in Section 5.

4.4.1 Engine Installation/Instrumentation

As received, the engine was configured exactly as it would be for a replacement in a CAT Model 637E scraper. Since this vehicle uses a torque converter in the transmission driveline, the engine flywheel was of a special configuration and did not match any driveshaft adapters in the laboratory. After consultation with CAT stress engineers, allowable modifications to the flywheel were detailed and machining modifications undertaken. While this was being done, installation of more routine instrumentation was also accomplished. Figure 4.4.1-1 shows the locations of pressure, temperature and airflow sensing instrumentation on the engine. It was also intended to include measurement of cylinder pressure, injection line pressure and injector needle lift, but since these modifications were of a longer lead time nature, engine break-in was initiated while modifications to a spare cylinder head were made. A close-up of the engine and a wider view of the test cell are shown in Appendix F.

4.4.2 Engine Break-In

To ensure that lingering improvements in oil consumption as the engine acquired hours of running time did not distort the particulate measurements, a careful engine break-in spanning 80 hours was undertaken. The test procedure was in accordance with CAT practice as defined in ET29. Following the break-in, the cylinder head was removed and exchanged with the spare which now had a cylinder pressure transducer installed (Kistler model 6121A2). This also afforded the opportunity to examine the cylinders in detail and ensure that no unknown damage might have occurred during break-in that would later obscure the emissions test results. Upon

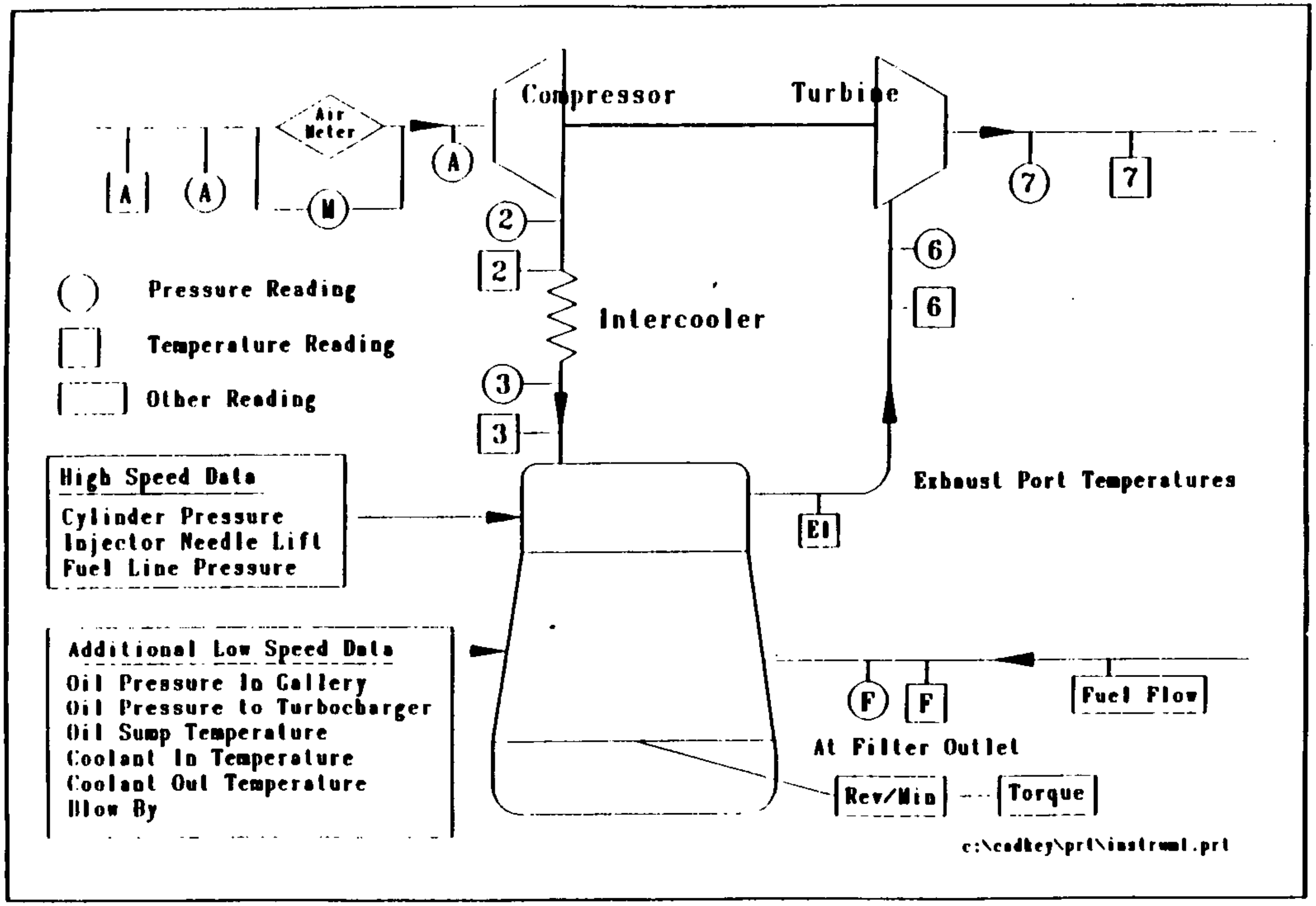


FIGURE 4.4.1-1 ENGINE INSTRUMENTATION

disassembly, it was discovered that the engine had a bent push tube for the number 6 intake valve. A replacement was installed and the engine readied for the subsequent baseline testing.

4.4.3 Baseline Tests

The first tests performed were to establish the starting point for emissions using the 8-mode steady-state test procedure. The targets to meet were as follows:

TABLE 4.4.3-1 TARGET AND BASELINE EMISSION RESULTS

	HC	CO	NO_x	PM
Target[2]	1.30	15.5	5.0	0.25
Baseline[1]	1.56	1.62	6.6	0.27

[1] Average of three runs; first run HC and CO were uncharacteristically high
[2] Patterned after on-highway truck

The actual results are printed below the requirements, and it can be seen that the HC, NO_X and particulates (PM) all exceeded the requirement. The high HC number is somewhat surprising and may be due to slight oil leakage down the valve stems during the first hour of running (recall that while the cylinder kits had been broken in, the new cylinder head had new stem seals). In subsequent retests of the baseline configuration, the HC number stabilized at about 1.35 g/hp-hr.

4.4.4 Parametric Tests

After establishing a baseline from which to make comparisons, the bulk of the real testing began, which consisted of systematic changes to engine hardware (and sometimes operating conditions) to achieve the program goals, while not violating certain limits in engine operation from a durability standpoint. These included limits in allowable sea level turbocharger speed, peak cylinder pressure and exhaust gas temperature.

Periodically, repeats of previous testing was done to ensure repeatability of both data and engine condition. The data was of a uniformly good nature, and no engine problems were encountered.

The results of the parametric testing will be discussed, not in chronological order, but in order of decreasing effect on emissions in Section 5.

4.4.5 Transient/Steady-State Cell Comparisons

As discussed above, the dilution tunnel used to measure particulate emissions was a "splitter" type, and a comparison with the "full dilution" type used for FTP transient testing of on-highway engines was felt to be of value. Also, a comparison between FTP and 8-mode emissions results was planned. To do this, the engine was moved from the steady-state test cell to a transient emission test cell, and an 8-mode test sequence was performed there as well.

It was then intended to set the Fuel/Air Ratio Controller (FARC) to three different levels of stringency in acceleration smoke control and perform a series of FTP transient emissions tests for each to establish correlations between steady-state and transient testing. This was thwarted by both an inconsistently performing FARC and limitations in the mechanical governor itself. This will be discussed in more detail in Section 5.10.

4.4.6 Final Engine Mapping

In order that the mechanics in Santa Barbara could duplicate the engine configuration ultimately developed during this program, it was necessary to measure the final governor settings so that this information could be communicated to them for use in the engine conversion in the 637E scraper.

As part of this activity, engine part-load "hooks" were run to acquire data on turbocharger speed, injection line pressure, both at the injector and pump end, and exhaust temperature. This information is necessary to make assessments of the degree to which the techniques developed during this project can be applied to CAT 3306 engines of other configurations and power levels. For higher power levels, and operation at altitude, excessive turbocharger speed or exhaust temperature may preclude transfer of this technology.

5. DISCUSSION OF RESULTS

For the discussions below, the text has been organized by the engine or test variable, rather than as a strict chronology of events. This makes comparisons of the effects of hardware or test condition changes more concise. They are also organized roughly in decreasing order of influence on emissions; the most important, such as injector tip geometry and injection timing, being discussed first, and the very weak effects, such as injector protrusion coming later.

5.1 Effect of Injector Configuration

One of the most influential variables affecting not just NO_X, but all measured emissions, was the physical configuration of the fuel injector nozzle. The variables and emissions results are shown in Table 5.1-1 and detail of a typical diesel fuel injector nozzle tip is shown in Figure 5.1-1.

Referring to Figure 5.1-1, in a typical sac or minisac nozzle, once injection line pressure has become high enough to unseat the needle, fuel flows past the needle and seat into the sac, where it then

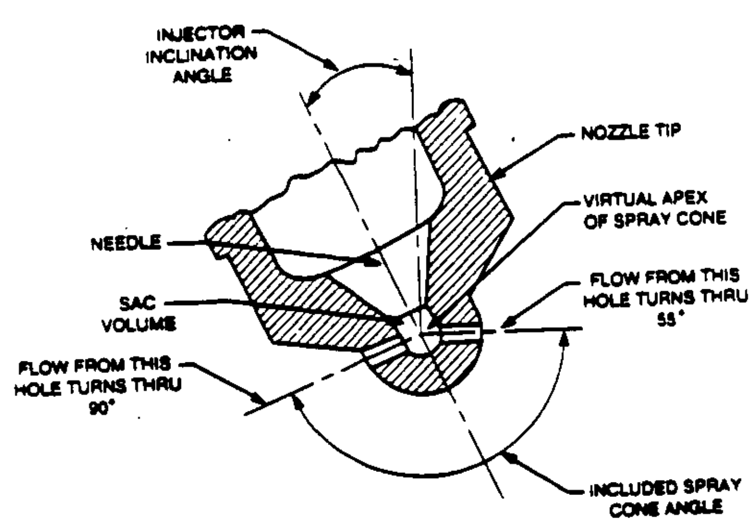


FIGURE 5.1-1 TYPICAL DIESEL INJECTOR TIP CONFIGURATION

distributes itself among the holes and finally enters the engine combustion chamber. The volume of the sac and the number and size of the holes has an effect on emissions due to a number of mechanisms.

For example, once the injection event is completed and the needle reseats, a small amount of residual fuel remains in the sac and holes. If this dribbles out in an uncontrolled fashion late in the combustion event when the piston has moved a considerable distance down-bore, successful low-emission combustion of this fuel is unlikely, leading to high HC levels. This effect can be seen in the change from the 8N7005 to the W3X1954 injector where the reduction in sac volume improved HC emissions by about 60 percent.

TABLE 5.1-1 EFFECT OF INJECTOR TYPE

Injector P/N	Sac Volume[1]	Hole Geometry[2]	Charge Cooling	BSFC[3]	HC[4]	CO	NO_x	PM
8N7005	2.33	9X.205	A-A[5]	.359	0.95	1.68	6.08	0.21
W3X0829	1.00	6X.254	A-A	.360	0.77	1.87	4.69	0.30
7W7038	0.93	6X.217	A-A	.365	N/A	0.94	4.71	0.16
W3X1954[6]	0.95	6X.229	JWAC[7]	.379	0.54	0.75	4.70	0.22
7W70386[6]	0.93	6X.217	JWAC	.375	0.45	0.84	4.78	0.11

1 = mm^3 2 = No. X diameter 3 = g/kW-hr 4 = g/hp-hr

5 = Air-to-air 6 = 12° timing 7 = Jacket water aftercooled

The most successful injector configuration was that used on the highway truck versions of the 3306 engine. It has 27 percent less flow area than the 8N7005 injector nozzle with a corresponding increase in injection pressure, which as shown in the table had a beneficial effect on both NO_X and PM. This pressure was enough higher though, that higher strength injection lines are necessary to ensure adequate durability in the field.

5.2 Effect of Injection Timing

As shown in Table 5.2-1 and Figure 5.2-1, the effect of injection timing also had a pronounced effect on emissions, especially NO_X, an effect which has been known for some time. Balanced against this however, is an increasing fuel economy penalty associated with retarded timing. This not only has the obvious effect of more fuel burned, but also heat rejection to the coolant, since the power of the engine must be maintained to satisfy the vehicle requirements. This in turn means more fuelling to the engine to offset the poorer fuel economy, and since the fraction of fuel energy going to the coolant is relatively constant, the coolant has to carry more heat away to the fixed size radiator, so the temperature rises. This is probably a bigger concern than fuel burned in a vehicle like the scraper, where changes to the cooling system involve very expensive modifications. Interestingly, this engine is what is called a "non-responder", meaning that changes in timing do not affect particulate emissions much. HC was only weakly affected

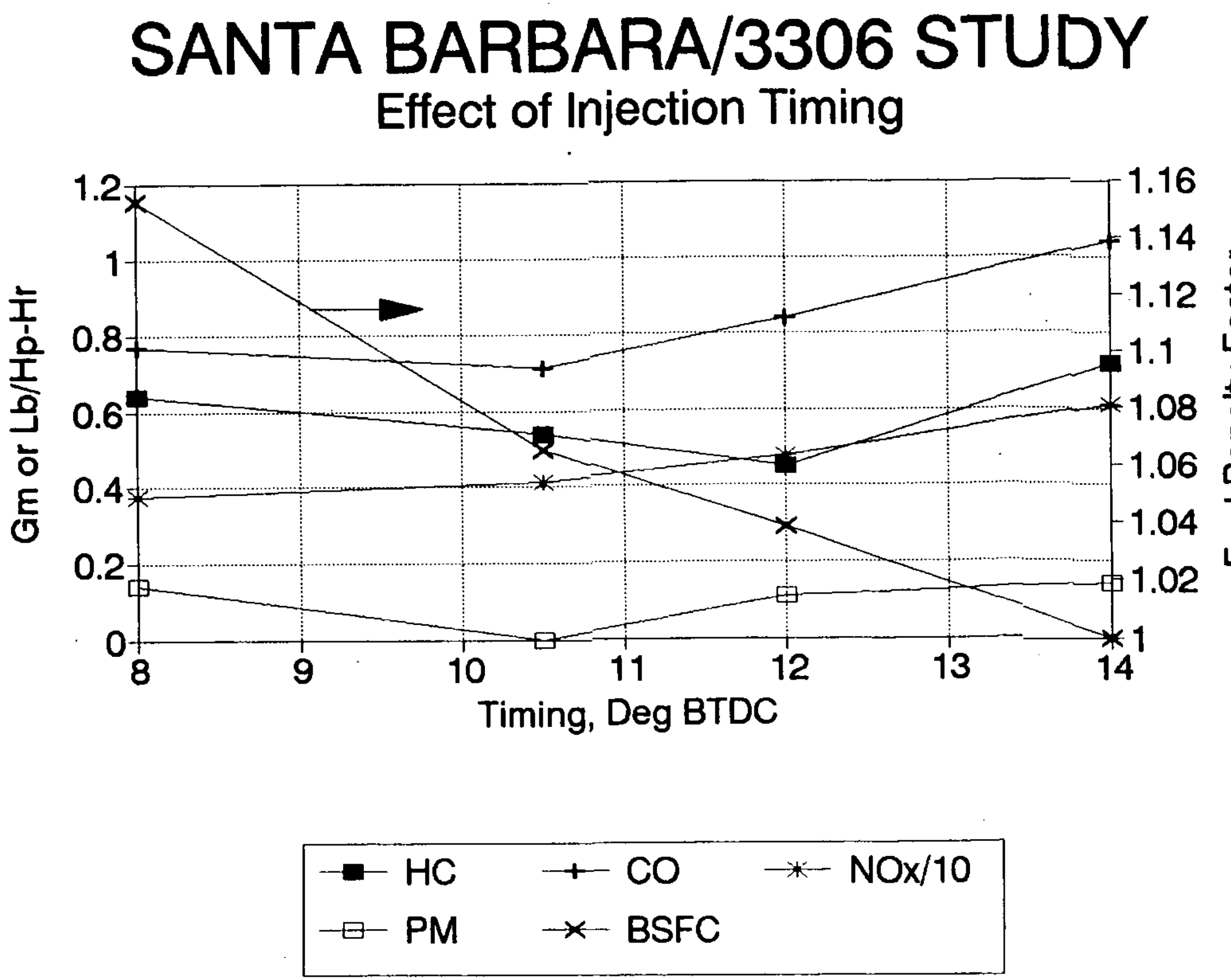


FIGURE 5.2-1 EFFECT OF INJECTION TIMING

by timing and CO slightly more so. Both were improved with retarded timing, but not by enough to justify the fuel economy penalty. Reduced sac volume is a better strategy for HC reduction and CO is very low in diesel engines anyway.

TABLE 5.2-1 EFFECT OF INJECTION TIMING

Parameters: Injectors - 7W7038
Cooling - Std. JWAC

Injection Timing °BTDC	HC	CO	NO_x	PM	BSFC Penalty %
14[1]	0.72	1.04	5.8	0.14	----
12	0.45	0.84	4.8	0.11	3.9
10.5	0.54	0.71	4.1	n/a	6.6
8	0.64	0.77	3.8	0.14	15.4

(1) Base engine timing

5.3 Effect of Charge Temperature

It was known from experience with the truck version of the 3306 engine that reduced manifold temperature has a beneficial effect on NO_x emissions. The scraper version of the engine responds in similar fashion as shown in Table 5.3-1. By way of explanation, the table contains two forms of emissions data. Those under the "8-Mode" heading are the result of full 8-Mode tests. In some cases, in order to acquire data more cost- effectively, especially sensitivity sweeps, data were taken at the engine rated power point only. These are shown under the "Rated" heading. Good correlation between rated power and full 8-Mode was established (Figure 5.3-1), especially for percentage changes in a variable.

By reducing manifold air temperature from the typical value of 88°C achieved with jacket water aftercooling (JWAC), to the 43°C more typical of air-to-air aftercooled truck units, an 8 to 19 percent reduction in NO_x was achieved. There were changes in particulate emissions depending on which injector was used, and fuel economy was only slightly, and not consistently, affected.

Because NO_x improvements of the same magnitude were gained by modest timing changes (5.B. above), and because the engineering efforts associated with air-to-air aftercooling were substantial, all testing beyond sequence number 17 (there were 36 total test sequences) was done with JWAC values of inlet manifold air temperature.

5.4 Effect of Turbocharger Change

Improvements in turbocharger efficiency can have a beneficial effect on emissions, but in a mostly indirect way. For example, an increase in compressor peak efficiency of four per cent (from a typical 0.75 to a better 0.78) means either a 5°C reduction in air delivered to the charge cooler, or an increase in boost pressure ratio from a typical 2.6 to 2.69. Even if the

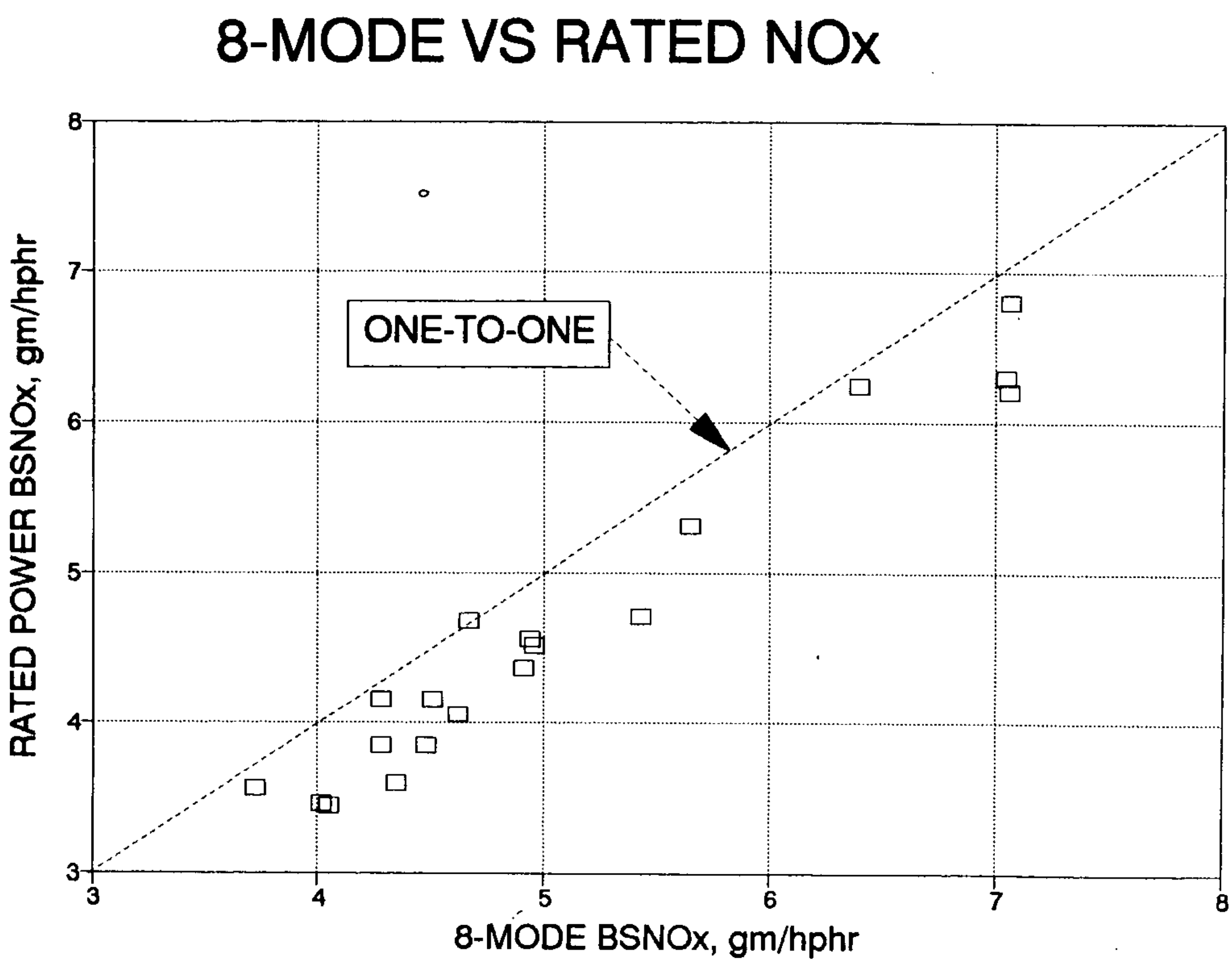


FIGURE 5.3-1 8-MODE VS RATED NO_X CORRELATION

temperature reduction was passed through the charge cooler, the effect on emissions would be quite small, but the improved air-fuel ratio associated with the higher boost pressure ratio does two things. It lowers the exhaust temperature slightly, which reduces the turbocharger speed, while the increased efficiency means that the speed reduction does not result in a loss in boost pressure. The reason that turbocharger speed is so important is that it often sets the limit on allowable boost pressure for the engine.

TABLE 5.3-1 EFFECT OF CHARGE COOLING

Parameters: **Timing - 14° BTDC**
Cooling - Std. JWAC
Fuel - Std

		8-Mode (g/hp-hr)				Rated		
Manifold Temperature, °F	Injector P/N	HC	CO	NO_x	PM	HC	NO_x	BSFC g/kW-hr
192	7W7038					0.79	5.8	.337
157	7W7038					0.78	5.2	.376
109	7W7038					0.76	4.4	.373
192	7W7038	0.72	1.04	5.8	0.14			.380
110	7W7038	N/A	0.94	4.7	0.16			.370
192	8N7005	1.56	1.62	6.6	0.27			.361
110	8N7005	0.75	1.68	6.1	0.21			.360

For these reasons then, it was planned to change from the "stock" turbocharger used in the scraper application to one like that used on the truck versions of the engine. As shown in Figure 5.D.1 the increase in "peak island" efficiency is about two per cent, but this only affects the higher power modes at torque peak speed. Because the engine in the scraper application runs at 2200 rpm rather than 2100 rpm, it runs "deeper in choke" on the compressor where the efficiencies have fallen off by a greater amount from the peak, so the net effect is no discernable difference in turbocharger performance at rated power speeds. While turbochargers with even greater peak efficiency than that used on the truck application exist, they cannot be readily fitted to the rather specialized water-cooled bearing housing necessary to meet the durability requirements of a machine like the scraper.

It comes as no surprise then, that given the small effective differences between the available turbochargers, no statistically meaningful change in emissions was seen (see Table 5.4-1 and Figure 5.4-1).

5.5 Effect of Fuel Composition

In an effort to assess the influence of fuel composition on emissions, a test was performed using a blend of fuel from Chevron called "Special Diesel". Table 5.5-1 compares some fuel

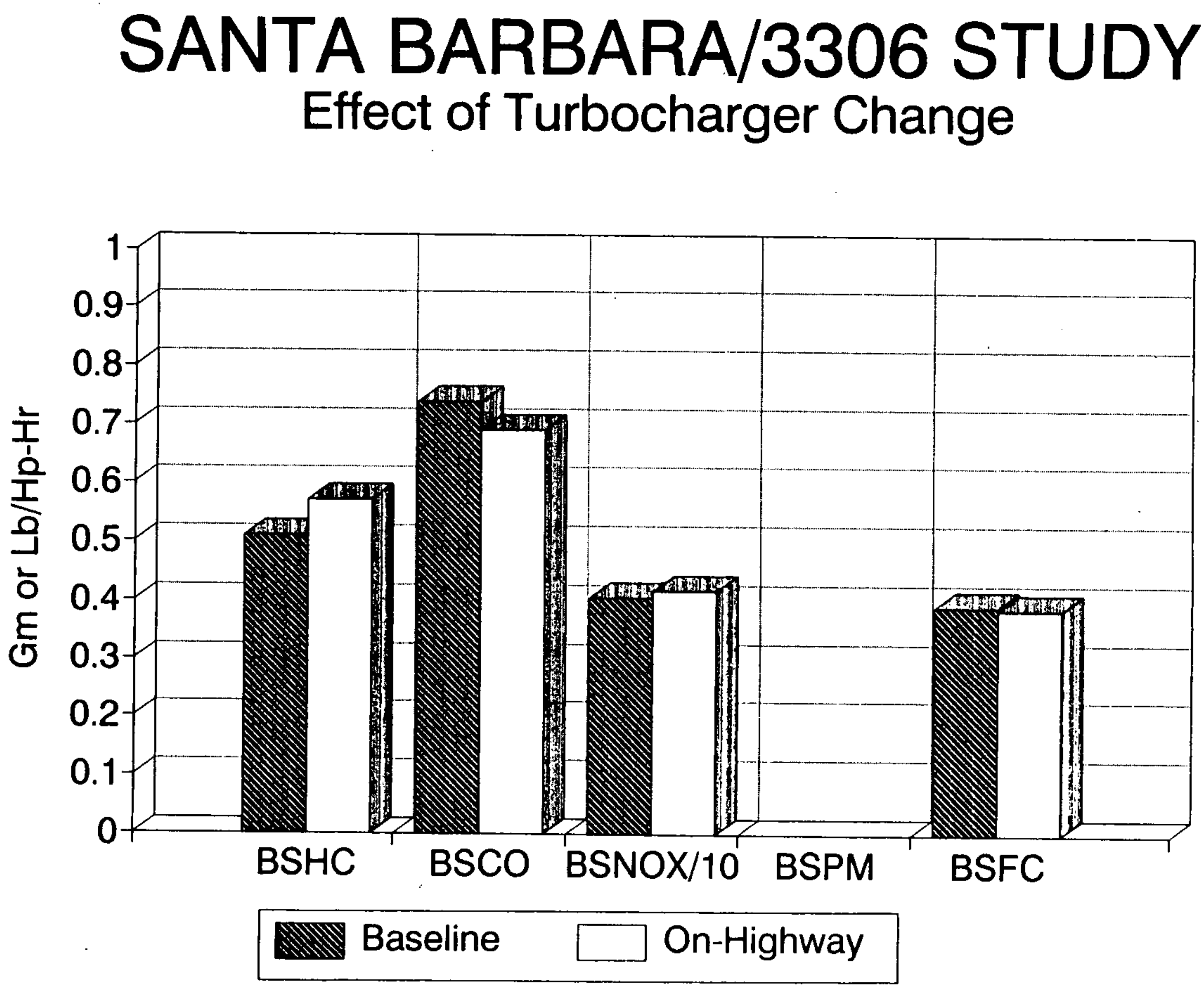


FIGURE 5.4-1 EFFECT OF TURBOCHARGER CHANGE

TABLE 5.4-1 EFFECT OF TURBOCHARGER CHANGE

Parameters: Injectors - 7W7038
Timing - 10.5° BTDC
Cooling - Std JWAC

Turbo Type	HC g/hp-hr	CO g/hp-hr	NO_x g/hp-hr	PM g/hp-hr	BSFC g/kW-hr
Scraper	0.51	0.74	4.0	(1)	237
Highway Truck	0.57	0.69	4.2		235

(1) Not measured

characteristics between the baseline fuel (low sulfur DF2) with the Special Diesel fuel. Both fuels have extremely low sulfur levels representative of fuels of the not-too-distant future in California, but the primary difference is in the aromatics. In Figure 5.5-1 the major difference in emissions is a 16 percent reduction in HC, with no statistically meaningful difference in any other category.

TABLE 5.5-1 FUEL COMPARISONS

Parameters: Injector - 7W7038
Timing - 10.5° BTDC
Cooling - JWAC - Std

Determination	Test Method	Baseline Fuel	"Special Diesel"
Cetane Number	D613	42.8	48.6
Sulfur (m%)		0.010	0.016
Hydrocarbon Type			
Aromatics %		29.8	34.7
Olefins %		2.9	2.1
Saturates %		67.3	63.2

5.6 Sensitivity Tests

Realizing that engine-to-engine build variations, daily, seasonal and long term changes in the operating environment can have an effect on emissions production, a number of sensitivity tests were performed to quantify these variables. For a number of these tests, for the same reasons as outlined above, abbreviated test sequences at rated power rather than a full 8-Mode test sequence were performed.

5.6.1 Charge Cooler Pressure Drop

In charge air coolers (sometimes called intercoolers or aftercoolers) both jacket water and air-to-air, the air that has been compressed by the turbocharger is passed through a matrix of metal fins. By virtue of being themselves attached to a "sink" at lower temperature (most of the

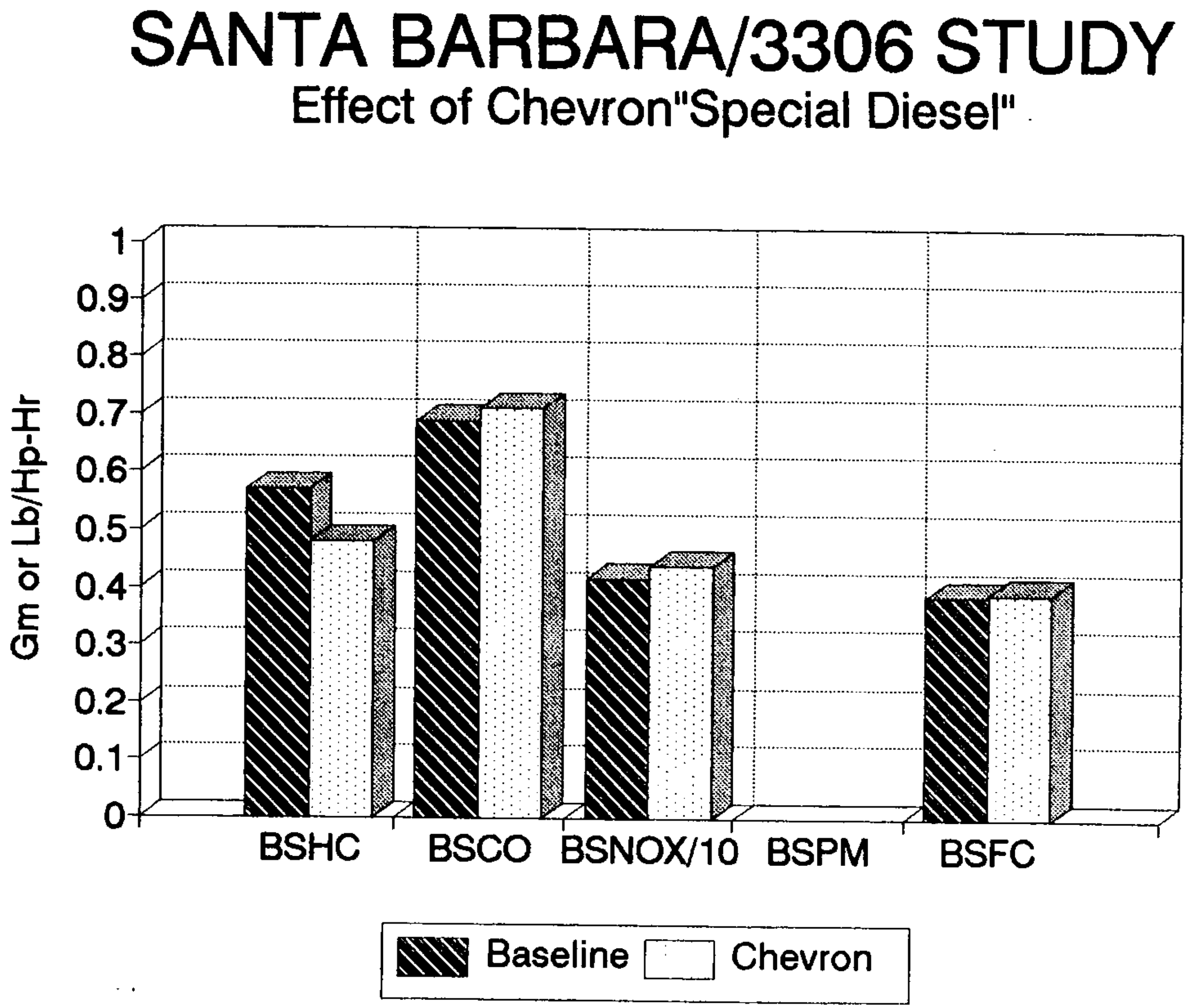


FIGURE 5.5-1 EFFECT OF CHEVRON "SPECIAL DIESEL" FUEL

time) than the air, heat of compression is removed from the air through these fins to the benefit of both charge density and some emissions.

Over time, these fins can accumulate a coating of dirt which reduces their effectiveness as a heat transfer medium, and reduces the passage width of the core through which the air passes. The effect of reduced heat transfer effectiveness has already been simulated in the manifold air temperature study described in Section 5.3 above. The consequence of this gradual "plugging" of the cooler core is increased resistance to airflow through the engine which has a detrimental effect on power.

The first section of Table 5.6-1 shows the influence on rated power HC and NO_X emissions.

TABLE 5.6-1 SENSITIVITY TESTS

Parameters: Injectors - 7W7038
Cooling - Std JWAC
Fuel - Std.

Test Variable	Value	HC g/hp-hr	CO g/hp-hr	NO_x g/hp-hr	PM g/hp-hr	BSFC g/kW-hr
Charge cooler pressure drop, kPa	4.1	0.86		4.6	(1)	222
	14.5	0.93		4.6		223
	22	0.67		4.7		224
Injector protrusion, mm	0.5 deeper	0.69	0.98	3.8		252
	Baseline	0.64	0.78	3.8		248
	0.5 shallower	0.63	0.77	3.7		248
Inlet and exhaust restriction	Baseline	0.63	0.77	3.7		246
	Maximum	0.60	0.85	3.6		249

(1) Not measured

Interestingly, despite rather remarkable increases in pressure drop, the effect on HC emissions is rather slight (the value of 0.67 lb/hphr is believed to be erroneous as it falls well outside the pattern of other observations at similar operating conditions) and the change in NO_X and fuel economy very slight. This is partly because of a self-compensating effect of the turbocharger. The increased restriction cuts down on the airflow, richening the fuel air ratio (because for these tests, power was held constant, so fuel was adjusted to whatever was necessary to satisfy the power demand) which raises the exhaust temperature, giving more energy to the turbocharger turbine. The turbocharger rotor thus "spools up" to a higher speed, creating more boost, and this helps overcome the increased restriction in the flowpath due to the cooler plugging.

5.6.2 Injector Protrusion

Injector protrusion refers to the distance into the combustion chamber that the fuel injector tip projects axially from the surface of the cylinder head (sometimes also called the "fire deck" and almost always flat in diesel engines). Since diesel engines have very high compression ratios, the clearance between the piston and the cylinder head is necessarily very small when the piston is at top dead center (TDC). In the 3306 engine the fuel is injected into a shallow "bowl" in the top (crown) of the piston and it can be appreciated that even small changes in injector protrusion could sufficiently alter the placement of the fuel plumes in the bowl to have an effect on emissions.

However, as shown in the second section of Table 5.6-1 and in Figure 5.6.2-1, the influence of protrusion was negligible for this engine (the high CO value for the baseline configuration is an anomaly in the data unsupported by other similar runs).

Notice should be taken of the very low NO_X readings for all three protrusions; these tests were performed at the most retarded timing. While this had a very beneficial effect on NO_X emissions, as discussed above, it was at the expense of severe fuel economy penalties.

5.6.3 Engine Intake and Exhaust Restrictions

For the majority of the testing described in this report, the inlet and exhaust restrictions were fairly minimal in comparison to the values suggested as maximum allowable guidelines by the engine manufacturer. This is because a conscious effort was made to model the engine as it would actually be installed in the scraper. A typical large diesel engine air filter and exhaust muffler were installed, and the ducting lengths approximate those of the vehicle.

To assess the effect of long term degradation in inlet and exhaust restrictions, a test was performed with levels set at the maximum allowed by the manufacturer. The results are shown in the last section of Table 5.6-1 and it can be seen that, as with charge cooler pressure drop, the emissions sensitivity is minimal. Not surprisingly fuel economy does suffer; the energy to move the air through the higher restrictions has to come from somewhere.

5.6.4 Air Filter Inlet Temperature

For the majority of the tests described in this report the engine was installed in a test cell which did not have conditioned inlet air. Since the tests spanned the period from November through March, considerable variation in inlet air temperatures to the engine were experienced as shown in Figure 5.6.4-1. Two of the most important variables in the test sequence; NO_X and BSFC were found to be sufficiently sensitive to inlet temperature that corrections had to be applied lest some degree of corruption of the relationships stemming from the data would have resulted.

A test sequence, specifically to ascertain the influence on BSFC was performed near the end of the program, and by judicious choice of similar hardware configurations, a NO_X correlation was also obtained. These are shown in Figures 5.6.4-2 and 5.6.4-3. The slopes of the regression lines through the uncorrected data quantify the correction necessary to be applied to the data. This was done for all full 8-Mode test results reported herein.

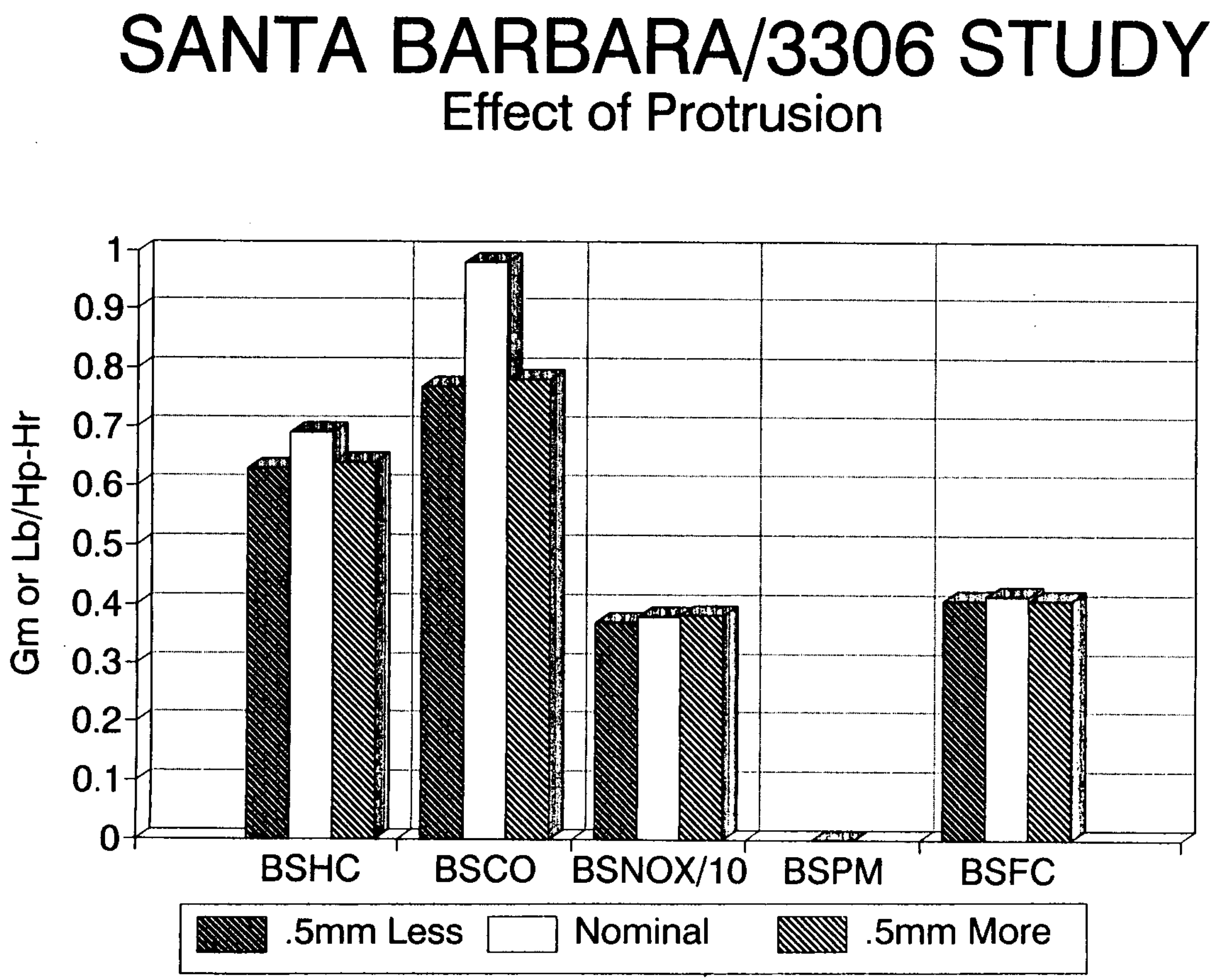


FIGURE 5.6.2-1 EFFECT OF INJECTOR PROTRUSION

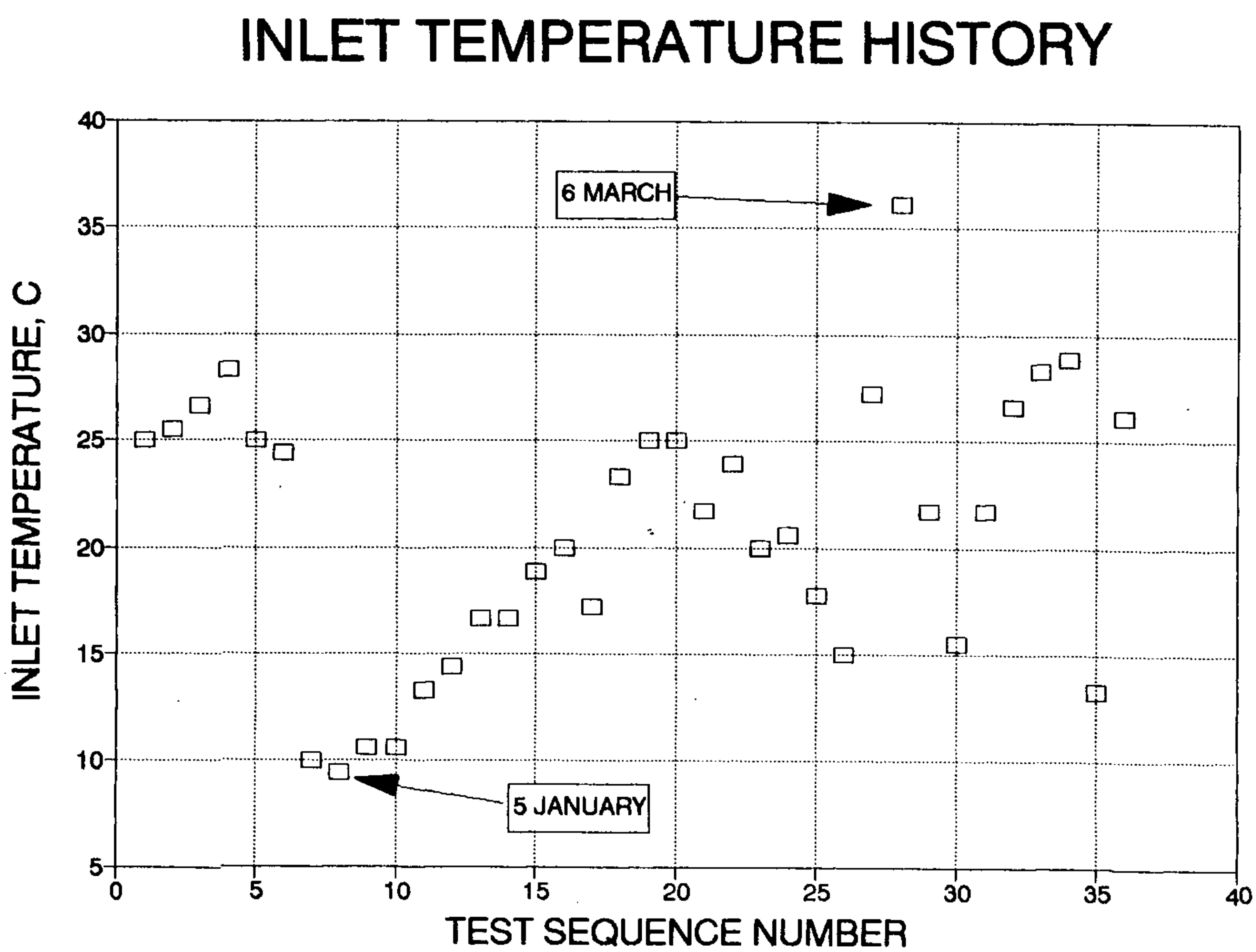


FIGURE 5.6.4-1 INLET TEMPERATURE HISTORY

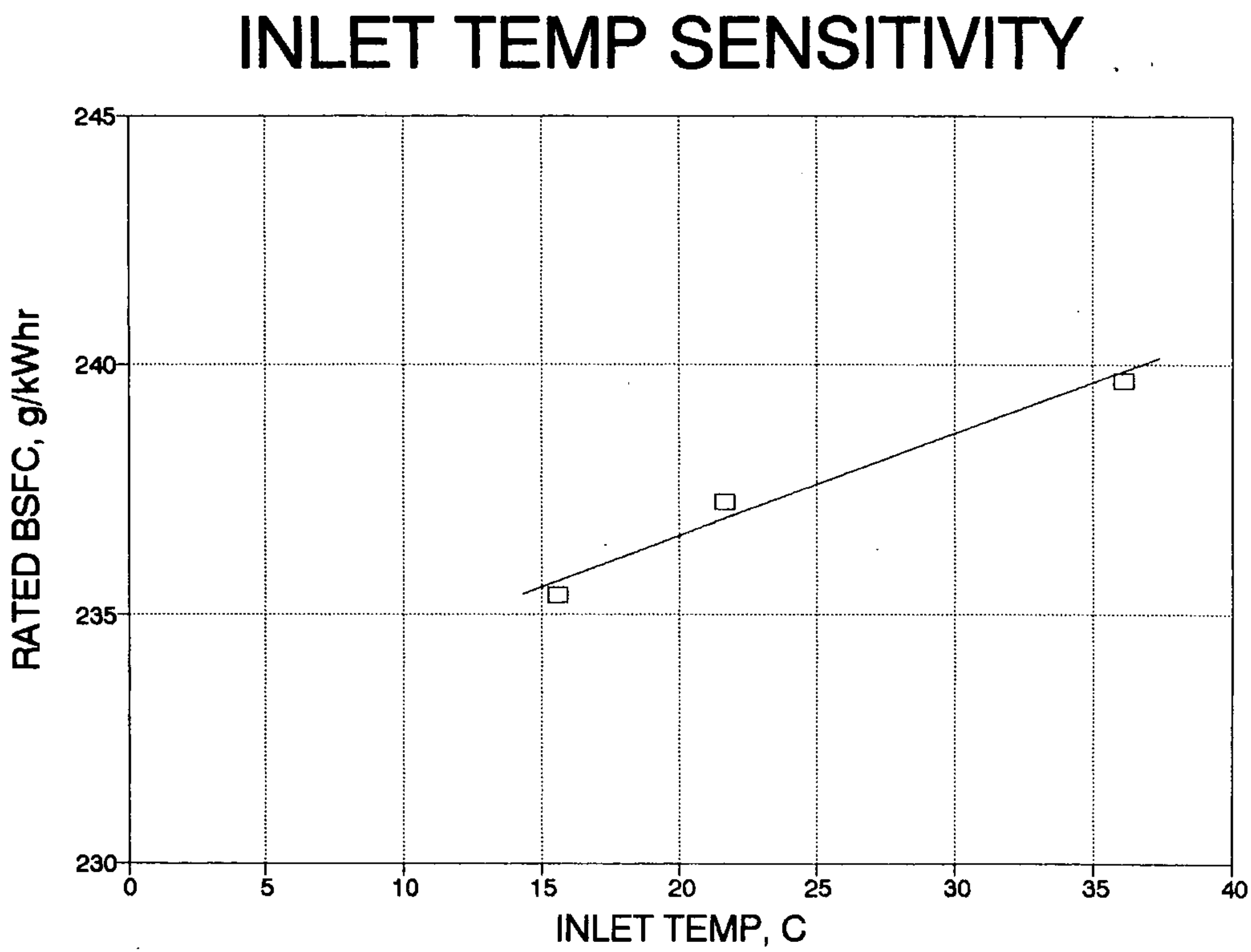


FIGURE 5.6.4-2 INLET TEMPERATURE SENSITIVITY, BSFC

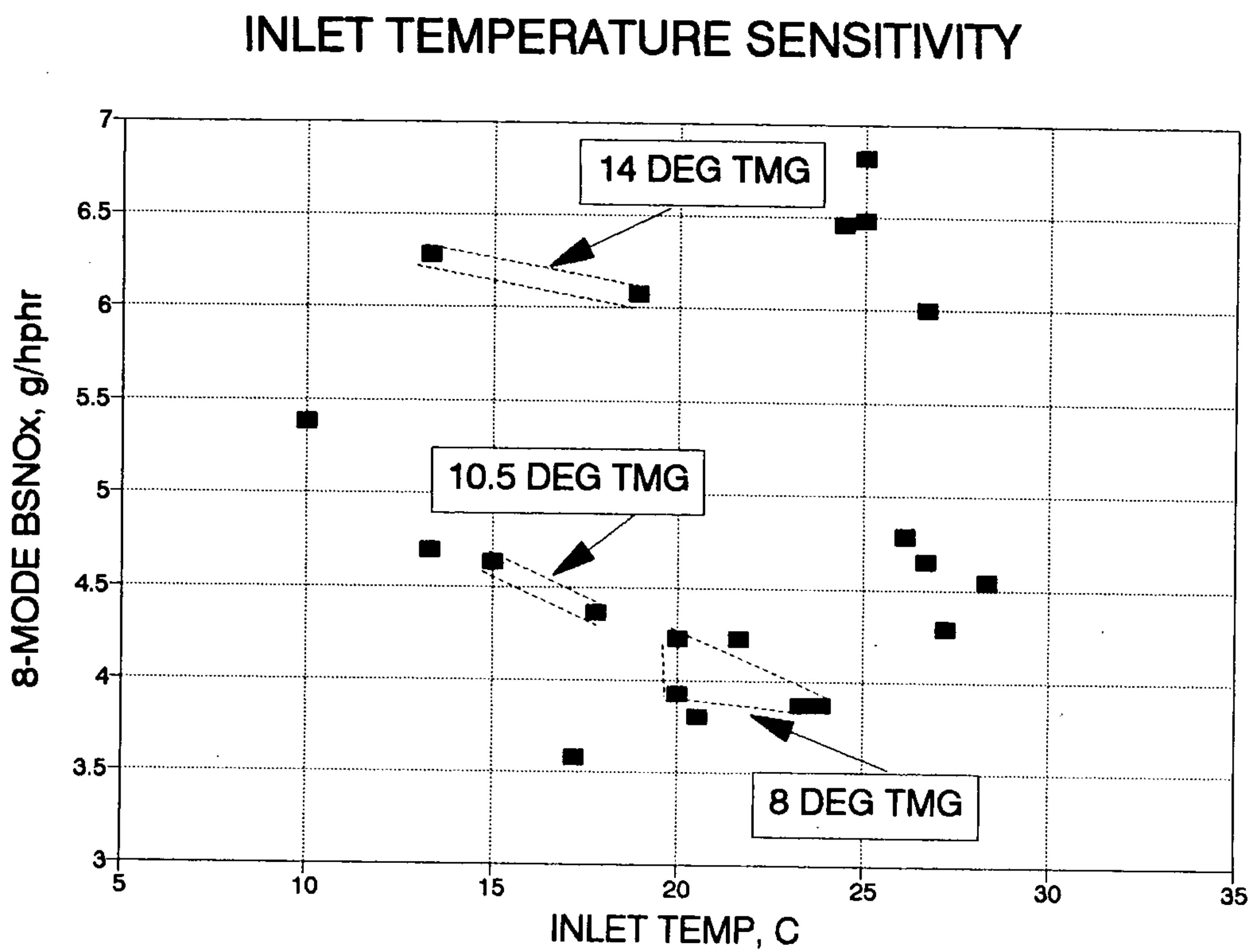


FIGURE 5.6.4-3 INLET TEMPERATURE SENSITIVITY, NO_X

5.7 Repeat/Confirmation Tests

In a test series as long as the one reported here (over 230 individual tests were run) it is prudent to periodically re-establish confidence that the baseline engine performance is not shifting. For this reason a number of repeat tests were performed. The engine was not returned to the as-received condition for each one, as this would have involved costly and time consuming conversion, but careful choice of comparison criteria allowed confidence to be established early in the program. Figures 5.7-1 and 5.7-2 show, with the exception of the 8° timing 7W7038 BSHC, very good repeatability (compare sets of contiguous bars).

5.8 Special Tests

Two "special" tests were performed during the test program; one at the request of the engine manufacturer, and the other at the request of the program sponsor. These are described below.

5.8.1 Turbocharger Speed and Fuel Line Pressure Assessment

The changes ultimately made to the engine to reduce NO_x emissions, including timing retard and injector changes, are in a direction that subjects the engine to additional stress.

Timing retard, because it degrades fuel economy, forces the use of additional fuel in order to maintain constant power. It also pushes the burning of the fuel later in the expansion stroke of the engine, and both of these raise the exhaust temperature. This in turn, has a threefold effect. More heat is rejected to the cooling water, which reduces boiling margin, higher metal temperatures result in many parts of the engine, especially the exhaust valves and exhaust manifolding, and the turbocharger speed increases.

The engine manufacturer had sufficient experience with the engine to feel confident about margins for the first two effects (as long as timing is not retarded beyond the two degrees finally settled on), but the turbocharger speed at sea level, rated power is difficult to predict when a number of interrelated variables are changed, as was the case with this program.

When the aggregate hole area of the injector tip is reduced, as was the case with the 7W7038 injectors chosen for the final engine configuration in this project, the fuel line pressure increases dramatically, and it was thought that this might be beyond the capabilities of the production fuel injection lines to withstand from a long term durability standpoint. Also, the forces at the injection pump plunger/camshaft interface increase as well, and an assessment of how much margin remained was felt prudent.

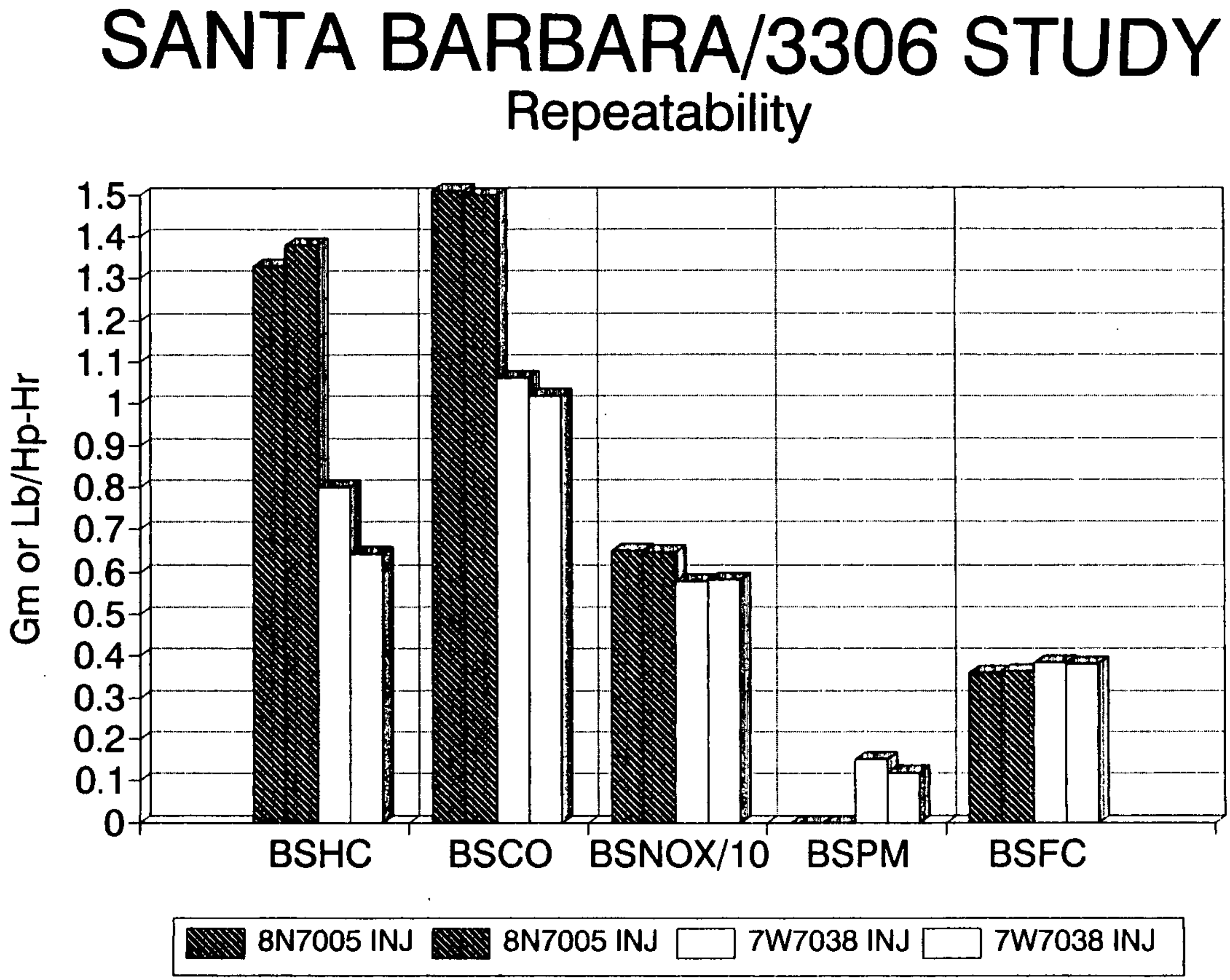


FIGURE 5.7-1 REPEATABILITY

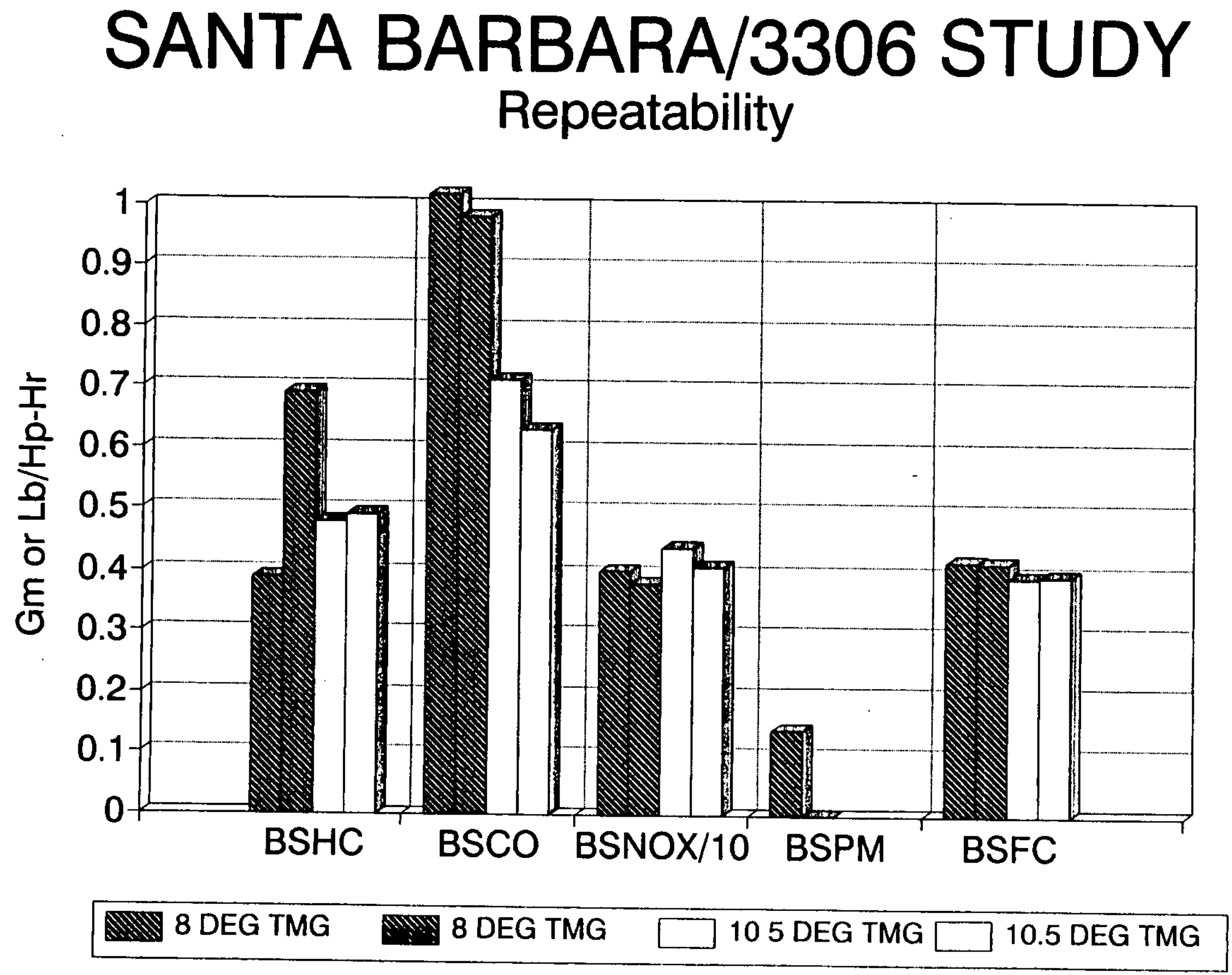


FIGURE 5.7-2 REPEATABILITY

To address these concerns; turbocharger speed and fuel line pressures, a separate matrix of speeds and loads was run. The results are presented in Table 5.8.1-1.

TABLE 5.8.1-1 TURBO SPEED AND FUEL LINE PRESSURE DATA

Point	Speed	Power kW	Turbo rpm	Inj Ln P psi	Cyl P psi	Inj Ln P mpa	Cyl P kPA
ss9	2200	276	96960	16850	1440	116.10	9922
s10	2200	213	84480	15150	1230	104.38	8475
ss11	2200	153	70620	12150	1055	83.71	7269
ss12	2200	75	52680	7050	870	48.57	5994
ss13	2200	26	42480	4300	695	29.63	4789
ss14	2000	270	92820	15600	1425	107.48	9818
ss15	2000	210	78900	14075	1225	96.98	8440
ss16	2000	148	64500	11675	1050	80.44	7235
ss17	2000	74	46440	6900	850	47.54	5857
ss18	2000	28	37740	4300	725	29.63	4995
ss19	1800	260	88380	13825	1400	95.25	9646
ss20	1800	215	77100	12925	1215	89.05	8371
ss21	1800	144	59280	11050	1000	76.13	6890
ss22	1800	69	40500	6525	860	44.96	5925
ss23	1800	29	33360	4150	710	28.59	4892
ss24	1400	233	78840	10225	1440	70.45	9922
ss25	1400	190	67080	10300	1225	70.97	8440
ss26	1400	102	41280	8400	930	57.88	6408
ss27	1400	51	28380	5425	810	37.38	5581
ss28	1400	21	24120	3725	695	25.67	4789

Using these data, the engine manufacturer was able to make the following determinations:

1. The turbocharger speed would not be excessive for use in vehicle operations near sea level in Santa Barbara County

2. The injection lines need to be changed to a higher strength design, the lines are now in the process of being production released.

3. The plunger/cam stresses were not excessive.

5.8.2 "Fuel Enhancer" Test

The Fuel Enhancer was a device, which when fitted into the fuel supply line of the engine, it was claimed, would improve combustion, fuel economy and emissions. Over the 8-Mode test cycle used for this project, no such improvements were observed. A more complete interim test report was prepared in January 1991.

5.9 Transient Test Cell/Steady-State Test Cell Comparison

As an addition to the original scope of work of the program, it was decided to make a comparison between the 8-Mode emissions test results and those obtained using the Federal Heavy-Duty Test procedure (often just called "FTP" or "transient"). But, since differences can exist between different test cells, especially in emissions measuring equipment, it was decided to perform a steady-state 8-Mode test sequence in the transient test cell to ensure that cell-to-cell variations did not obscure the comparison. This afforded the opportunity to make a comparison between the two particulate tunnels as well since these are the most obvious difference between the test cells.

Table 5.9-1 summarizes the contrast between the emissions results for the two test cells.

TABLE 5.9-1 COMPARISON OF 8-MODE RESULTS BETWEEN STEADY-STATE AND TRANSIENT TEST CELLS

Cell	HC g/hp-hr	CO g/hp-hr	NO_x g/hp-hr	PM g/hp-hr
Steady-State	0.45	0.84	4.8	0.11
Transient	0.46	0.69	5.1	0.17

Agreement for HC is very good, NO_x numbers are within 7 percent of each other, while CO and PM differ by wider margins. The PM differences may be due in part to the use of a "splitter tunnel" in the steady-state cell and a "full dilution tunnel" in the transient cell. It was beyond the scope of this project to pursue these differences, but it is an area for potential future work.

5.10 Fuel/Air Ratio Controller Setting

For most of the steady-state development testing, the Fuel/Air Ratio Controller (FARC) was removed from the engine to ensure that it did not interfere with full load operation. However, at the completion of the development testing, and before meaningful transient tests could be run, new settings for the FARC had to be determined. This information is also necessary to complete the conversion of engines presently installed in 637E scrapers in the field.

The function of the FARC is to reduce engine acceleration smoke by inhibiting full fuel rack travel until sufficient air is available to ensure complete combustion (not necessarily totally smoke-free, but an improvement over unrestrained fuelling). It does this on the test engine by using an oil pressure "precocked" mechanism to prevent full rack motion. When adequate turbocharger boost pressure has built up to ensure that the engine air-fuel ratio is high enough, the rod tension withholding full rack travel is relaxed. This is a dynamic process accomplished by a hydromechanical device which is sensitive to boost pressure, engine speed and load demand.

Setting the FARC is largely a trial-and-error process when significant changes have been made to the engine as was the case in this program.

This was attempted while the engine was still in the transient test cell following the 8-Mode mapping, but persistent problems with the FARC made long term experimentation in that facility cost-prohibitive. Consequently the engine was returned to the steady-state cell where continued work could be accomplished under less pressure. It was also during this return trip that the turbocharger speed mapping described above was concluded.

As mentioned in Section 4.4.5, it was planned to make three setting determinations; one using the criteria currently applied to engines installed in off-highway vehicles, another using highway truck criteria, and an intermediate setting. Then FTP hot starts were to be performed at each setting and the results compared. However, the inordinate sensitivity of the mechanical governor mechanisms to temperature, forced a change in plans.

The off-highway vehicle criteria is 45 percent opacity (briefly) during a no-load sudden acceleration of the engine. This was fairly easy to attain (strip-chart recordings are included in Appendix E).

The highway truck criteria involves a more complicated procedure pertaining to "load acceptance" within a prescribed time period. This proved impossible to attain since the sensitivity of the governor mechanism to recent temperature history resulted in very inconsistent run-to-run variation. Therefore, it was decided to test only the off-highway vehicle setting in the transient test cell.

5.11 Transient Test Comparison

After satisfactory compliance with the opacity criteria had been demonstrated, the engine was returned to the transient test cell. Here it underwent one "cold FTP" and three "hot FTP" tests. The results are summarized in Table 5.11-1 where a comparison with the 8-Mode results from the transient cell are also presented.

TABLE 5.11-1 COMPARISON OF 8-MODE AND TRANSIENT TEST RESULTS

Parameters: Injectors - 7W7038
Cooling - JWAC Std.
Fuel - DF-2
FARC Setting - Final

Test	HC g/hp-hr	CO g/hp-hr	NO_x g/hp-hr	PM g/hp-hr
Transient[1]	0.53	3.2	5.7	0.40
Transient[2]	0.52	3.1	5.7	0.40
8-Mode[3]	0.46	0.69	5.1	0.17
8-Mode[4]	0.45	0.84	4.8	0.11

1 Cold Start/7 + Hot Start x 6/7   2 Hot Starts Only   3 In Transient Cell   4 In Steady-State Cell

It should be noted that the fuel used for all the transient test cell work was not the same very low sulfur fuel used for the steady-state development, but a fuel more representative of that used for the body of past test data with which these results are likely to be compared.

6. CONCLUSIONS

6.1 Summary of Results

A direct-injection turbocharged, aftercooled diesel engine was modified to improve gaseous and particulate emissions as assessed using a steady-state 8-Mode test procedure.

The three most influential variables were fuel injector tip geometry, injection timing and charge air temperature. Each by itself had the potential to improve NO_x (the primary emphasis of this study) by 20 to 40 percent when exercised through the range of values that were investigated in this program. However, in some cases undesirable side effects, such as substantial increases in fuel consumption resulted. The final results were achieved through a combination of techniques.

NO_x was improved by over 27 percent and particulate emissions by over 58 percent. HC and CO were also improved, all at a modest cost of 3.8 percent in fuel economy.

Other variables such as fuel composition, charge cooler pressure drop, injector protrusion and intake and exhaust restrictions had very little effect on emissions.

Clearly, significant improvements in the emissions of diesel engines in off-highway service can be made by appropriate application of on-highway technology. In particular, engines like the one tested in this program respond to changes in injection timing, injector tip geometry and high pressure injection pumps.

Improvements of over 27 percent in NO_x and almost 60 percent in particulate emissions were attained using only hardware that can be bolted on to the engine without materially affecting it's outline dimensions.

This last point is an important one because many off-road engines are very tightly integrated into their vehicle packages, making technical solutions which impact outline very undesirable. A case in point is air-to-air charge cooling. As documented in Table 5.3-1 reductions in NO_x from 6.6-5.8 down to 4.7-4.4 g/hphr (an average 25 percent improvement) could be obtained by lowering charge air temperatures from 190°F to 110°F.

Unfortunately, the only way to achieve these reductions, using hardware readily available today, is by changing from jacket water aftercooling to air-to-air aftercooling. Aside from the obvious repercussions on packaging (the cooler is almost universally placed outboard of the radiator requiring additional grill clearance and ducting), added stress is placed on the cooling system because of the higher "cold-side" air temperature resulting from the pass through the cooler first.

Using the techniques applied to the engine in this study, improvements of equal or better magnitude can be had with far less negative impact on the vehicle.

Care should be exercised however in stretching the techniques beyond the ranges bounded by this study. For example, Table 5.2-1 presents tantalizing reductions in NO_X emissions using injection timing retard, but as shown in the last column, and illustrated even more dramatically in Figure 6.1-1, fuel economy penalties mount quickly when too much injection retard is used. Too, the particular engine tested was what is termed a "non-responder", meaning that as timing is retarded to improve NO_X , particulates do not get appreciably worse. In many diesel engines this is not the case.

6.2 Applicability to Other Similar Engines

The 3306B engine family is comprised of a variety of different configurations of the same basic "core" engine. Extrapolation of results from this study to others in the family must be limited to the direct-injection turbocharged aftercooled versions, but even here there are ample variations.

Using the data acquired primarily in the final engine mapping (Sections 4.4.6 and 5..8.1) part of the program, it is possible to interpolate with significant, and extrapolate with reasonable confidence what the emissions results would be for alternate ratings as contrasted with the 270 hp 2200 rpm specimen tested in this program.

This work is being performed by Caterpillar engineering in a fairly comprehensive manner, but was not completed as this report was published. The results will be issued as a supplement when available.

However, to get a rough estimate of the probable results, a simple spreadsheet analysis of NO_x emissions for a 300 hp 2000 rpm version of the engine was made (recall that the test engine produced 270 hp at 2200 rpm). Cycle weighted NO_x was shown to reduce by 5%. This small difference is close to the ability to measure repeatably, but in broad terms, NO_x emissions are not expected to vary appreciably among the various ratings of the 3306 direct injected engine family.

The recommendations made at the outset of this report are directionally appropriate to reduce emissions in these more distantly related engines as well. Injection retard (until heat rejection or fuel economy penalty become too severe), higher pressure injection systems, and reformulated fuel would help to reduce emissions from any of the diesel engines mentioned above.

Predictions about other engine families, such as the older 3306 indirect injection types, 3200 series and 3400 series engines simply cannot be made without extensive test data. This is because these engines differ so significantly in areas such as combustion chamber shape and even number of valves, that extrapolation of 3306 data cannot be used. Efforts similar in scope to that described in this report would be necessary for each engine.

BSFC/NOX TRADE-OFF
INLET TEMP CORRECTED

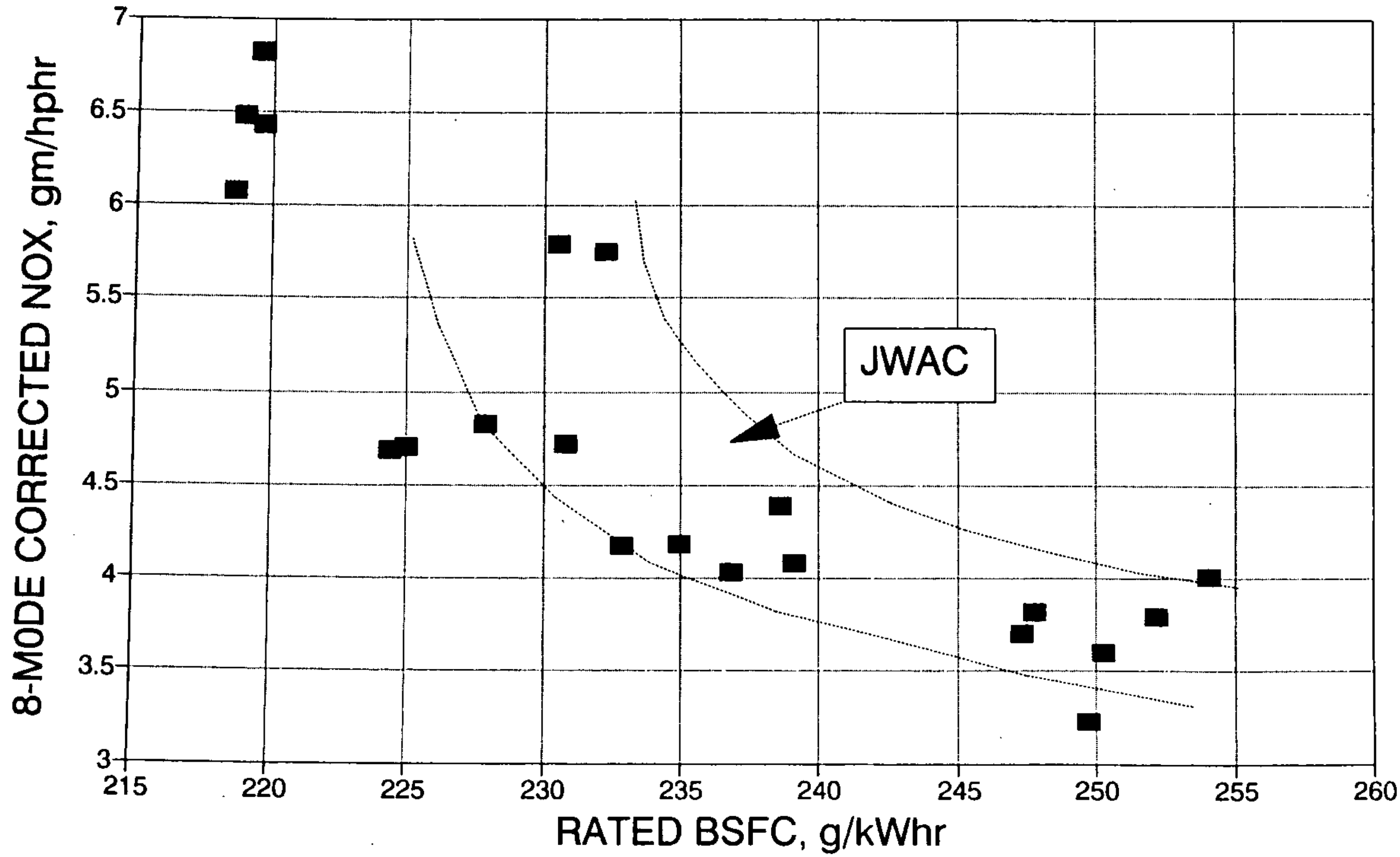
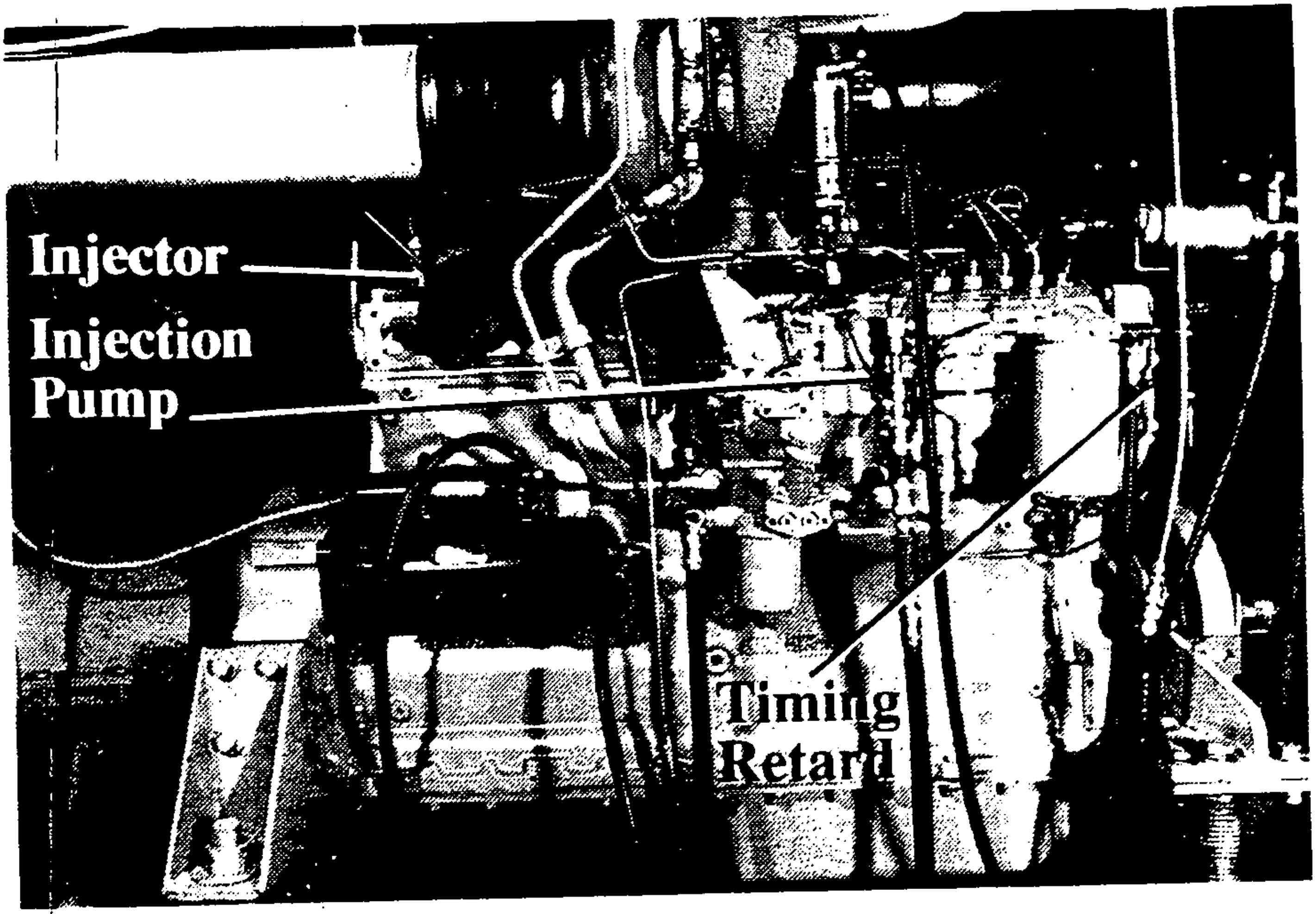
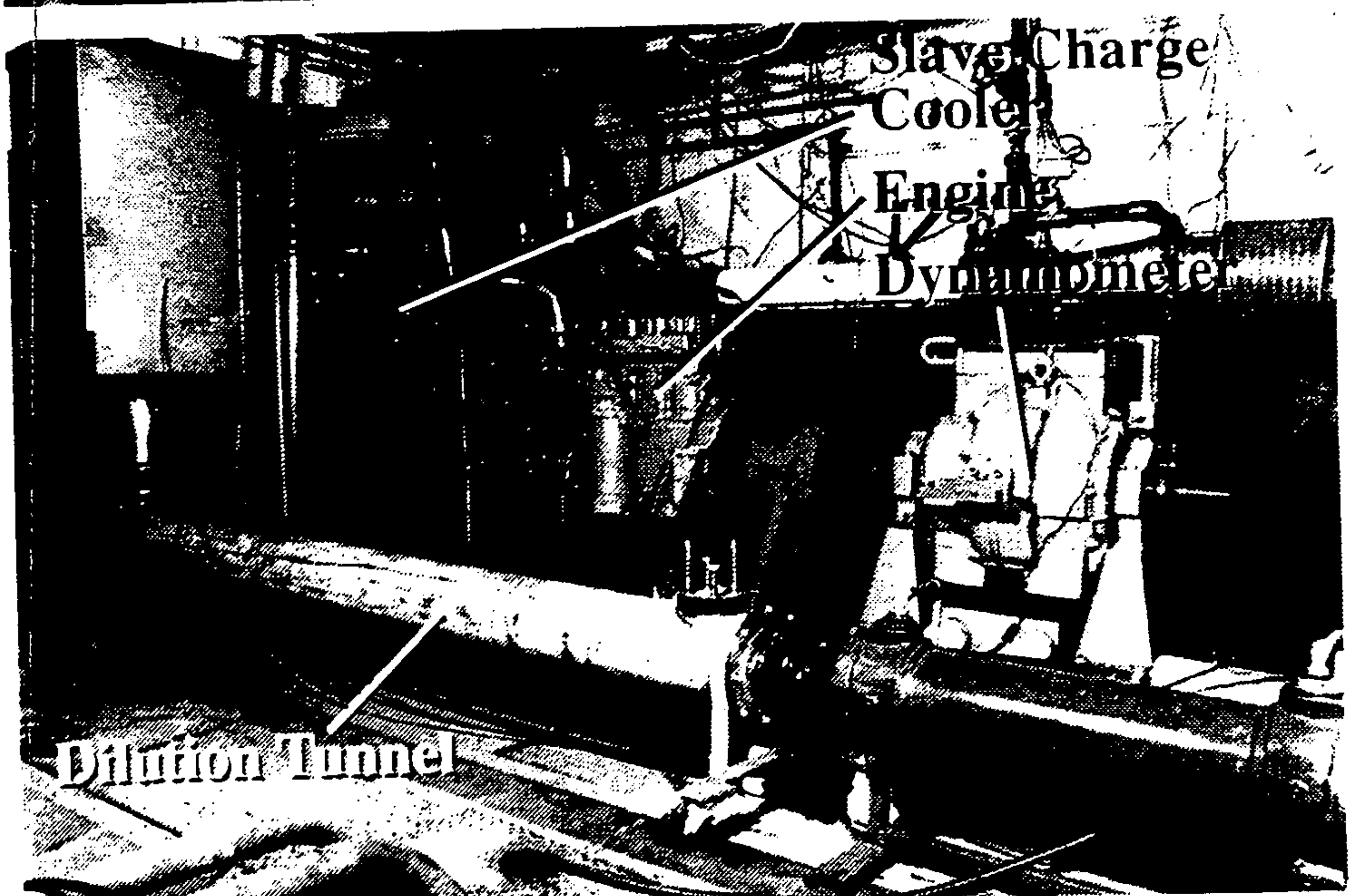


FIGURE 6.1-1 BSFC/NO_X TRADE-OFF

APPENDIX A

PHOTOGRAPHS


Injector
Injection Pump
Timing Retard


Slave Charge Cooler
Engine
Dynamometer
Dilution Tunnel