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POTENTIAL OF DIESEL ENGINE, 1979 SUMMARY SOURCE DOCUMENT

T. Trella

U.S. DEPARTMENT OF TRANSPORTATION
Research and Special Programs Administration
Transportation Systems Center
Cambridge MA 02142

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FINAL REPORT

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16. Abstract

This document assesses the fuel economy potential of diesel engines in future passenger cars and light trucks. The primary technologies evaluated include: (1) engine control strategy and implementation, (2) the engine design variables, (3) emissions and noise, (4) fuels, (5) lubricants, (6) vehicle-engine matching, and (7) the effects of vehicle characteristics. The major findings are summarized.

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PREFACE

This report, DOT-TSC-NHTSA-79-38, "Potential of Diesel Engine, 1979 Summary Source Document," summarizes an assessment (as of the end of fiscal year 1979) of the fuel economy potential of diesel engines in future passenger cars and light trucks.

This Summary Source Document is supported by four companion reports. The are:

"Potential of Spark Ignition and Diesel Engine, Engine Catalog and Performance Analysis," by J, Kidd and J. Rogowicz, U.S. Department of Transportation, Transportation Systems Center Report No. DOT-TSC-NHTSA-79-39, March, 1980

"Potential of Diesel Engine, Engine Emission Technology," by J. Sturm and T. Trella, U.S. Department of Transportation, Transportation Systems Center Report No. DOT-TSC-NHTSA-79-40, March, 1980

"Potential of Diesel Engine, Design Concepts, Control Strategy and Implementation," by T. Shen and T. Trella, U.S. Department of Transportation, Transportation Systems Center Report No. DOT-TSC-NHTSA-79-41, March, 1980

"Potential of Diesel Engine, Fuels and Lubrication Technology," by G. Cornetti, Fiat Central Research, U.S. Department of Transportation, Transportation Systems Center Report No. DOT-TSC-NHTSA-79-42, March, 1980

The Summary Source Document and its companion reports comprise a series deliverable under PPA HS-027, "Support for Research and Analysis in Auto Fuel Economy and Related Areas."

METRIC CONVERSION FACTORS

Approximate Conversions to Metric Measures

Symbol	When You Know	Multiply by	To Find	Symbol
		LENGTH		
in	inches	2.5	centimeters	cm
ft	feet	30	centimeters	cm
yd	yards	0.9	meters	m
mi	miles	1.6	kilometers	km
		AREA		
in^2	square inches	6.5	square centimeters	cm^2
ft^2	square feet	0.09	square meters	m^2
yd^2	square yards	0.8	square meters	m^2
mi^2	square miles	2.6	square kilometers	km^2
	acres	0.4	hectares	ha
		MASS (weight)		
oz	ounces	28	grams	g
lb	pounds	0.45	kilograms	kg
	short tons (2000 lb)	0.9	tonnes	t
		VOLUME		
tsp	teaspoons	5	milliliters	ml
Tbsp	tablespoons	15	milliliters	ml
fl oz	fluid ounces	30	milliliters	ml
c	cups	0.24	liters	l
pt	pints	0.47	liters	l
qt	quarts	0.95	liters	l
gal	gallons	3.8	liters	l
ft^3	cubic feet	0.03	cubic meters	m^3
yd^3	cubic yards	0.76	cubic meters	m^3
		TEMPERATURE (exact)		
°F	Fahrenheit temperature	5/9 (after subtracting 32)	Celsius temperature	°C

Approximate Conversions from Metric Measures

Symbol	When You Know	Multiply by	To Find	Symbol
		LENGTH		
mm	millimeters	0.04	inches	in
cm	centimeters	0.4	inches	in
m	meters	3.3	feet	ft
m	meters	1.1	yards	yd
km	kilometers	0.6	miles	mi
		AREA		
cm^2	square centimeters	0.16	square inches	in^2
m^2	square meters	1.2	square yards	yd^2
km^2	square kilometers	0.4	square miles	mi^2
ha	hectares (10,000 m^2)	2.5	acres	
		MASS (weight)		
g	grams	0.035	ounces	oz
kg	kilograms	2.2	pounds	lb
t	tonnes (1000 kg)	1.1	short tons	
		VOLUME		
ml	milliliters	0.03	fluid ounces	fl oz
l	liters	2.1	pints	pt
l	liters	1.06	quarts	qt
l	liters	0.26	gallons	gal
m^3	cubic meters	35	cubic feet	ft^3
m^3	cubic meters	1.3	cubic yards	yd^3
		TEMPERATURE (exact)		
°C	Celsius temperature	9/5 (then add 32)	Fahrenheit temperature	°F

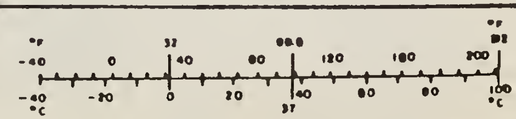


TABLE OF CONTENTS

TABLE OF CONTENTS (CONT'D)

LIST OF ILLUSTRATIONS

LIST OF ILLUSTRATIONS (CONT'D)

LIST OF ILLUSTRATIONS (CONT'D)

LIST OF TABLES

LIST OF TABLES (CONT'D)

1. INTRODUCTION

1.1 PURPOSE

Diesel engines are increasingly being used as a power source for passenger automobiles, light trucks and vans because of their fuel economy advantage. Current production vehicles powered by diesels offer a significant fuel consumption advantage over gasoline engines of equal performance. Although diesels are not commonly used in the U.S. for such applications, production and use are considerable in Europe and Japan.

The future of the diesel engine as an automotive power plant in light duty vehicles will ultimately have to be decided in the market place. However, before that can occur, the diesel-powered vehicles must meet the Federally mandated emission standards. Increasingly stringent standards have created much technical discussion regarding the diesel's future. For instance, emission control technology is in its infancy, and although several schemes have promise, control strategy is not nearly as mature as for the spark ignition systems.

This report assesses the potentials for the diesel engine for passenger cars and light trucks. The assessment is related to diesel engines produced in 1978. The findings are presented in terms of miles per gallon for the engines installed in typical vehicles. The results presented are an attempt to quantify fuel - economy improvements that could be accomplished; they are not intended to predict any manufacturer's actual plans. The emissions constraints maintained throughout the report are those established by the Clean Air Act Amendment of 1977 ($HC/CO/NO_x$) 0.41/3.4/1.0 gms/mile and the requirement for establishing particulate emission standards. There are, currently, no standards governing particulate emissions. However, a 0.6 gm/mile particulate emission regulation is proposed by EPA in 1981 for diesel-powered cars and light trucks. A more stringent particulate standard of 0.2 gms/mile is proposed for 1983 vehicles.

1.2 MAJOR FINDINGS

Some of the findings of this report are:

1) When normalized to 1978 average spark ignition engine (SI) horsepower-to-vehicle intertia weight and rear axle ratio, currently produced diesel passenger cars and light trucks and vans show fuel economy advantages ranging anywhere from 20 to 50 percent. Diesels currently in production are naturally aspirated and indirect injection engines.

2) The ability of light weight diesel passenger cars and light trucks to meet the 1981 federal emission standard of 0.41/3.4/1.0 ($HC/CO/NO_x$) grams/mile and future proposed particulate levels is dependent on the vehicle inertia weight class. Currently available technology indicates that for low mileage vehicles less than and/or equal to 2500 lbs, the 0.41/3.4/1.0 ($HC/CO/NO_x$) emission standard can be achieved along with particulate levels of about 0.3 grams/mile without compromising fuel economy through injection retard and turbocharging. This technology will also permit vehicles in the 2500 to 3500 lbs inertia weight class to achieve an emission standard of 0.41/3.4/1.5 ($HC/CO/NO_x$) and a particulate level of about 0.3 to 0.4 grams/mile. Finally, present day light-weight diesel passenger cars and light trucks in a vehicle weight class greater than 3500 lbs cannot meet the 1981 emission standard (1.0 gm/mi NO_x) but can achieve 2.0 gm/mi NO_x along with the proposed 1981 particulate level of 0.6 grams/mile by using currenly available technology. Particulate traps will be required to lower particulate emissions. As of today, traps have not demonstrated their long-term effectiveness in reducing diesel particulates since they must be replaced every 5000 miles.

3) There are a number of vehicle characteristics which affect the diesel's fuel economy. These include aerodynamics, rear axle ratio, tires and transmissions. The diesel's fuel economy sensitivities to these vehicle characteristics are different from those of spark ignition engines.

a) Fuel economy gains achieved by reduction of vehicle aerodynamic coefficient are influenced by the vehicle's HP/WT. For naturally aspirated vehicles (vehicles powered by lower HP/WT), reducing the aerodynamic drag by 10 percent results in a fuel economy gain of approximately 1.5 percent. Appropriate modifications to the vehicle's driveline result in a fuel economy gain of at least 3.5 percent. For turbocharged vehicles, which have a higher HP/WT, a 10 percent reduction in aerodynamic drag yields a fuel economy gain of at least 2 percent, and the further modifications to the driveline result in a gain of 4 percent.

b) Reducing the tire friction losses also improves vehicle fuel economy. Larger vehicles have greater fuel economy gains than small size vehicles when similar technological advancements in tire design are made since frictional losses are a function of vehicle weight.

A ten percent reduction of tire rolling resistance on 1978 tires would improve fuel economy by approximately 1 to 1.2 percent in naturally aspirated diesel cars. For a turbocharged diesel car the savings is 1.2 to 1.8 percent.

Reducing tire rolling losses also results in an improvement in the vehicle's performance. Vehicle performance should be kept constant by adjusting vehicle gear ratios in order to achieve the full fuel economy benefits of reducing the rolling resistance. If these driveline modifications are made, then a ten percent reduction in rolling resistance would yield a fuel economy gain of approximately 1.8 to 2.5 percent for the naturally aspirated diesel and 2.0 to 3.0 for the turbocharged diesel car.

c) A 3 to 5 percent improvement in fuel economy can be achieved by adding an extra gear to the transmission (overdrive). Fuel economy gains to the order of 14 percent have been reported when the manual shift points of the transmission are changed, as well as with the addition of an extra gear

4) Turbocharged vehicles offer a five to ten percent improvement in fuel economy when the driveline is reoptimized to take advantage of the corresponding improvement in horsepower to inertia weight. Turbocharging without attendant changes in the vehicle driveline gives improved performance but shows a 5 percent reduction in fuel economy.

5) Direct Injection engines hold promise to increase fuel economy upwards to 10 to 15 percent above current light duty IDI diesel engines. Limited data show that these engines potentially produce lower HC, CO and particulate emissions. The introduction of these engines in the 1980 through 1990 time frame depends on the development of cost effective fuel injection systems.

6) Various advanced technologies which hold promise for further improvements in diesel fuel economy, emissions reduction and performance include:

- adiabatic/turbocompounding
- intercooling
- variable compression ratio
- water emulsion
- alternative fuels
- combustion chamber modifications
- fuel injection system (electronic control of injection timing)

2. BACKGROUND

2.1 GENERAL

The diesel engine dates back to 1892 when Dr. Rudolph Diesel filed a series of patent applications on an engine that was intended to burn coal dust. Dr. Diesel originally formulated the concept when he realized that an ideal heat engine could be created by coordinating the rate of fuel injection with the movement of the piston in such a manner that the heat of combustion would be liberated at constant temperature, thus approaching the thermal efficiency of the Carnot Cycle.

Needless to say, the diesel engine as recognized today in passenger and light truck, and van vehicles did not immediately emerge in this period of time. In earlier diesel versions, a blast injection system was used (referred to as air-blast injection) in which highly compressed air was used to atomize and force the fuel into the combustion chamber: Diesel engines equipped with these injection systems were used primarily for power plant applications. Because the air-blast injection system did not offer flexible control of fuel timing and rates, and due to the engine's large bulk resulting from the high compression ratio required to achieve fuel ignition, application to automotive use was precluded.

It was not until 1926, when Robert Bosch introduced a scroll type pump, that fuel could be injected under high pressures. This method of injection soon became widely used. It was recognized, however, that if diesels were to succeed in the automotive field, their design concepts would have to break away from the previous steam engine design traditions. Interestingly, it was during this period that the first diesel engine, a 2.1 liter Acro engine, was installed in a passenger car. It operated until 1929 and drove the car about 25,000 kilometers.

By 1933, the air blast injection system was essentially completely superceded by the Bosch System. High speed diesel engines emerged that were entirely dependent on fuel injection. Targets were established to replace gasoline engines by diesels in existing chassis. However, the attainment of this goal was slow, since the extra weight of the engine required stronger front springs, more massive mountings, larger clutches and modification to other transmission items.

It was not until the early 1940's when full attention was given to methods of increasing diesel engine speeds, reducing their specific weights and improving their thermal efficiency that a diesel engine concept for a passenger car began to emerge. The work conducted at the British Royal Aircraft Establishment, under H. B. Taylor, demonstrated that the diesel engine could approach within 20 percent the power performance of gasoline engines.(1) Injection system limitations were overcome by implementing mechanical systems to control fuel metering and timing of fuel injection to the combustion chamber. Major emphasis was then placed on the development of diesel engines comparable to spark ignition engines for passenger car application. Two diesel engines (Ricardo Consulting Engineers and Mercedes-Benz) emerged; they featured good engine torque characteristics, increased power for a given cylinder capacity, acceptable specific weight and exhaust smoke limits, and control of cylinder pressure rise necessary for the engines to be acceptable for passenger car usage. Derivatives of these engine designs are found in production today.

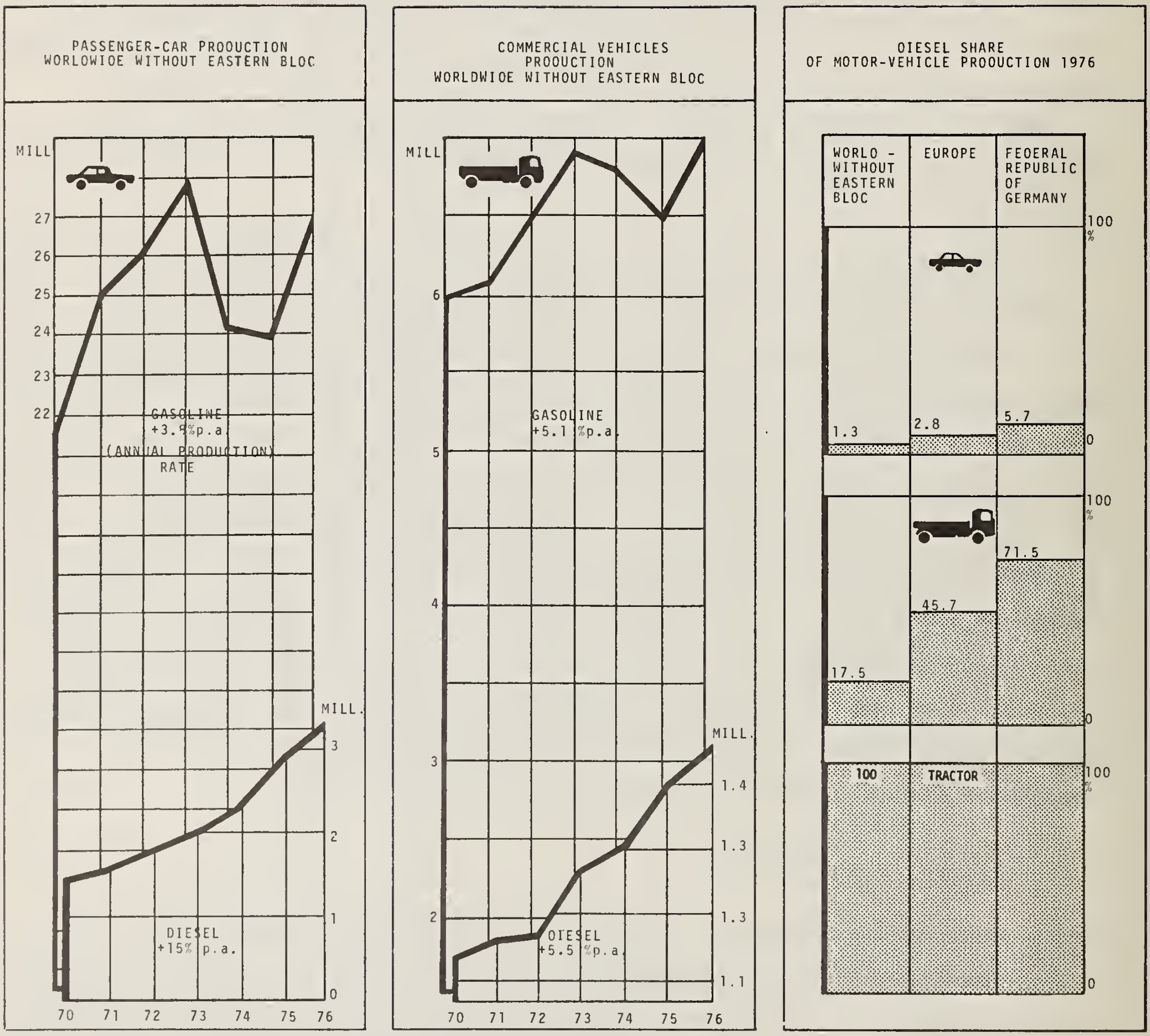
Daimler-Benz initiated the production of a 260 diesel model in 1936. This was a 6/7 passenger sedan with a 4-cylinder engine. In 1948, the 170 D vehicle was introduced paralleling the 170 gasoline powered vehicle. Both diesel and gasoline engines were manufactured on the same assembly line. Most recently, Volkswagen demonstrated the use of existing gasoline engine blocks for high speed diesel automobile engines and the effects of turbocharging.

Prior to the early 1970's, the diesel engine was used primarily in the U.S. automotive fleet for heavy duty vehicle applications. Some exceptions, however, were made for light duty applications and a relatively small volume of European diesel vehicles were imported for passenger car and taxi usage. A small number of retrofit diesel engines appeared during this time. The light duty diesel has maintained a significant market in foreign countries, in particular Western Europe and Japan, where a number of them have been widely used for taxis and vans. However, no significant penetration was made in the passenger car market.

The annual production of light duty diesels saw an upswing during the 1970's because of their fuel economy advantage. Today, the yearly worldwide rate is approximately 4 million with 18 percent in the light truck and van market and 1.2 percent penetration into passenger car production, (see Figure 2-1). Diesel power accounts for approximately 4 percent of the light trucks produced in the United States today. Passenger car diesels share a significantly smaller percentage. Current manufacturers marketing diesels in the U.S. include Mercedes Benz, Peugeot, Volkswagen and General Motors. General Motors entered the diesel market quite recently with their production of a 350 CID Oldsmobile diesel and is the first domestic manufacturer to produce diesel engines specifically for use in passenger cars.

2.2 DIESEL PRODUCTION

The passenger car and light truck diesels have not, as yet, significantly penetrated the U. S. market; however, a growing trend of increasing penetration exists. (See Tables 2-1 and 2-2). Recently, Mercedes Benz announced that its sale of diesel passenger cars has reached 50 percent of those sold in 1977 and that it plans to introduce, in 1978, a coupe powered by the naturally aspirated 5-cylinder diesel that already exists in the 300D sedan. Various American manufacturers also plan to provide diesel-powered vehicles in the 1980's. As an example, General Motors is currently evaluating its gasoline Chevy 1.6 liter L4 engine for dieselization.



SOURCE: Reference 2. (References listed at end of each section)

FIGURE 2-1. PASSENGER CAR AND LIGHT TRUCK WORLDWIDE PRODUCTION AND SHARE

TABLE 2-1. CURRENT OR UNDER DEVELOPMENT WORLDWIDE DIESEL ENGINES

United States

Manufacturer	Engine Displacement (liters)
Chrysler	2.068 (4 cylinder)
	2.375 (6 cylinder)
	3.59 (6 cylinder)
General Motors	1.6 (4 cylinder)
	5.3 (8 cylinder)
	4.1-4.2 (8 cylinder)

Japan

Manufacturer	Engine Displacement (liters)
Isuzu C190	1.951 (4 cylinder)
Nissan SD22	2.164 (4 cylinder)
SD33	3.246 (6 cylinder)
Mitsubishi 4DR50	2.659 (4 cylinder)
6DR50	3.988 (6 cylinder)

TABLE 2-1. CURRENT OR UNDER DEVELOPMENT WORLDWIDE DIESEL ENGINES (CONTINUED)

Western Europe

Manufacturer	Engine Displacement (liters)
British Leyland	1.50 (4 cylinders)
	1.80 (4 cylinders)
Citroen CRD 90	2.175 (4 cylinders)
Fiat 8144.65	1.995 (4 cylinders)
8140.61	2.446 (4 cylinders)
8160.61	3.668 (6 cylinders)
Ford York	2.358 (4 cylinders)
	2.540 (6 cylinders)
Land Rover 2 1/4	2.286 (4 cylinders)
Rover/MSA	3.431 (6 cylinders)

TABLE 2-1. CURRENT OR UNDER DEVELOPMENT WORLDWIDE DIESEL ENGINES (CONTINUED)

Western Europe (Continued)

Manufacturer		Engine Displacement (liters)
Mercedes Benz	220D	2.197 (4 cylinders)
	240D	2.404 (4 cylinders)
	300D	3.000 (5 cylinders)
Opel 2100D		2.068 (4 cylinders)
Perkins	4.108	1.760 (4 cylinders)
	4.154	2.522 (4 cylinders)
	6.247	4.052 (6 cylinders)
Peugeot XLD		1.357 (4 cylinders)
	XDP 4.83	1.946 (4 cylinders)
	XDP 4.90	2.112 (4 cylinders)
	6.90	3.168 (6 cylinders)
Volkswagen		1.5 (4 cylinders)
		1.838 (6 cylinders)
		2.205 (6 cylinders)

Reference: DOT-TSC-Contract 1545; "The Investigation of Compression Ignition Power Plant Parametric Tradeoffs."

TABLE 2-2A. CURRENT AND PROPOSED UTILIZATION OF DIESEL IN PASSENGER CARS

	Mini Compact Car	Sub Compact Car	Compact Car	Inter-mediate Car	Standard Car	Luxury Car	Station Wagon	Passenger Car
U.S. MANUFACTURERS								
General Motors		P*	P*	P	C	C	C	C
GMC								
Chrysler	P		P*	P*	P*			P
Dodge	P		P*	P*				P
Ford								P**
International Harvester								
U.S. IMPORTS								
Volkswagen		C				C		C
Daimler-Benz							C	C
Peugot			C					
Fiat	P							
WESTERN EUROPE MANUFACTURERS								
Volkswagen		C						C
Daimler-Benz						C		C
General Motors (Opel)		C						C
Ford (Germany)			C					C
Fiat		C						P*
Renault								C**
Alfa-Romeo								C**
Citroen								P
Volvo								
JAPAN								C
Nissan		C					C	
Isuzu							C	
Toyota								P*
Mitsubishi		P						P*
Honda								P*
Toyo Kyoga								P*

NOTES: P = Proposed
C = Current

* Vehicle class size determined from vehicle engine data with EPA vehicle size class listing
** Vehicle class size not specified

TABLE 2-2B. CURRENT AND PROPOSED UTILIZATION OF DIESEL IN LIGHT TRUCKS

	Jeep	Pickup	Light Truck
U.S. MANUFACTURERS			
General Motors		C	C
GMC		C	C
Chrysler			
Dodge		P	C
Ford		P	P
International Harvester	C		C

American Motors Coporation will offer a diesel engine Jeep CJ model in "International markets" in 1979 but not in the U.S., pending clarification of diesel emission standards by the U.S. government. General Motors also has plans to introduce more diesel engines into its automobiles. Pontiac and Buick, for example, are considering tooling programs for diesels though they have not as yet taken any steps; Cadillac is tooling up its production for a small V-8 diesel (4.2 or 4.1 liter), and a V-8 diesel is in the preproduction stage at Chevrolet. The Chrysler Corporation has a dieselized gasoline 225 CID slant 6 engine under development.

Diesel engine production in Western Europe is also on an increase and a number of companies in addition to Mercedes Benz and Peugeot are planning to introduce diesels as imports to the U.S. Fiat, Volvo and Renault are among them. While Nissan and Isuzu have produced diesel vehicles in Japan, predominantly for taxis, other major Japanese automotive firms are expected to enter the diesel passenger car market soon. However, the Japanese firms have not indicated whether they propose to export diesels to the U.S. at this time.

2.3 DIESEL ADVANTAGES AND DISADVANTAGES

On the basis of total energy consumed, the diesel engine shows a substantial advantage over the spark ignition engine. As an example,[3] the General Motors Oldsmobile 350 diesel uses about 700 gallons less fuel during 100,000 miles than the Oldsmobile 260 spark ignition engine, and over 1200 gallons less than the Oldsmobile 350 gasoline. Table 2-3 shows some diesel engine fuel economy comparisons based on the manufacturers' equivalent spark ignition engine.

TABLE 2-3. COMPARISON OF DIESEL AND GASOLINE EQUIVALENT

MANUFACTURERS	INERTIA WEIGHT (LBS)	CID (IN^3)	HORSEPOWER	HP/IW	TRANSMISSION	REAR AXLE RATIO	FUEL ECONOMY		% DIFFERENCE* (NORMALIZED)
							COMPOSITE (MPG)	% DIFFERENCE (COMPARISON)	
VOLKSWAGEN									
(DIESEL)	2250	90	48	0.0213	M4	3.9	44	42	23.3
(GASOLINE)	2250	89	62	0.0275	M4	3.9	31		
PEUGEOT 504									
(DIESEL)	3500	141	71	0.020	M4	3.79	31	55	41.5
(GASOLINE)	3500	120	81	0.023	M4	3.89	20		
MERCEDES BENZ									
(DIESEL 240)	3500	147	62	0.017	A4	3.69	27	42	23
(GASOLINE 230)	3500	141	81	0.023	A4	3.69	19		
GENERAL MOTORS									
350 (DIESEL)	4500	350	120	0.026	A4	2.41	24	41	16.5**
OLDS 98 (GASOLINE)	4500	350	170	0.037	A3	2.41	17		
INTERNATIONAL HARVESTER									
PICKUP-PERKINS (DIESEL)	4500	247	105	0.023	M4	3.75	24	58	50
GM PICKUP (GASOLINE)	4500	250	115	0.025	M4	3.73	17		
GMC PICKUP									
(DIESEL)	4500	350	120	0.026	A3	2.76	24	50	25.9
(GASOLINE)	4500	350	165	0.036	A3	2.76	16		
GMC PICKUP									
(DIESEL)	5000	350	120	0.024	A3	2.76	22	47	24
(GASOLINE)	5000	350	165	0.033	A3	2.73	15		

* Normalized for gasoline HP/IW and Rear Axle Ratio.

Normalization Factor: $FE = F_{E_1} \left[\frac{HP_1}{HP_2}\right]^{0.55}$; $FE = F_{E_1} \left[\frac{RAR_1}{RAR_2}\right]^{0.7}$

** Corrections for transmission not included.

Part of the increased trend of diesel market penetration has been stimulated by the increased energy consciousness of the buying public and a growing demand for transportation capacity. The diesel shows definite advantages over the spark ignition passenger car engine and provides improvements in fuel economy ranging anywhere from 22 to 52 percent for constant horsepower-to-weight and rear axle ratio over currently produced spark ignition engines. See Table 2-4. Most diesel fuel economy advantage is attributable to the absence of a throttle, better fuel economy performance during cold start operation (since the diesel does not require mixture enrichment), principle of burning with the excess air, presence of a regulator which automatically cuts out injection during deceleration, the diesel's lower specific output (horsepower-to-vehicle weight ratio), and higher compression ratio.

Besides providing better fuel economy, present day diesel engines offer[3] a) better reliability in terms of the probability of manufacturing and assembling a properly functioning vehicle, b) better serviceability relative to simpler maintenance schedules and parts to be serviced, and c) better durability and driveability.

However, present day diesels are not without their problems and do exhibit disadvantageous characteristics. Some of the major disadvantages include: a) poor cold starting (diesel engines presently require glow plugs), b) higher particulate emissions (roughly 10:1 over current gasoline engines), c) visible exhaust (including black and white smoke), d) exhaust odor and irritancy, e) higher noise and vibration levels, especially during idle, after cold starts and during cold weather conditions, f) cost, and g) engine weight and size.

Odor and particulates are important because of the pressure of public opinion. The "dirtiness" is visible and aesthetically

TABLE 2-4. 1978 FUEL ECONOMY SUMMARY OF DOMESTIC AND FOREIGN DIESEL PASSENGER CARS, LIGHT TRUCKS AND VANS (SHEET 1)

MANUFACTURER	MOOEL	ENGINE / VEHICLE SPECIFICATIONS: CIO	HORSEPOWER	INERTIA WEIGHT	TRANSMISSIONS	REAR AXLE RATIO	EMISSIONS (GMS/MILE) (URBAN): HC	CO	NO_x	FUEL ECONOMY: COMPOSITE (MPG)	% DIFFERENCE*	HP/WT % DIFFERENCE*	FUEL ECONOMY (c) % DIFFERENCE NORMALIZEO FOR HP/WT	FUEL ECONOMY (d) % OIFFERENCE NORMALIZED FOR HP/WT AND RAR
PASSENGER CARS (a)														
VOLKSWAGEN	RABBIT	90	48	2250	M4	3.30	0.78	1.00	0.61	55	83	-28	+53	+39
	RABBIT	90	48	2250	M4	3.90	0.30	1.00	1.05	44	47	-28	+23	+25
PEUGEOT	204	83	51	2500	M4	4.06	1.11	1.71	0.68	40	48	-34	+18	+23
FORD	PINTO	132	61	2750	M4	2.8	0.24	1.21	0.76	49	96	-37	+53	+26
POSTAL VAN	PERKINS	165	-	3000	A3	3.07	0.14	1.47	2.54	33	50	-	-	-
PEUGEOT	504	141	71	3500	A3	3.78	0.66	1.20	1.21	28	47	-38	+13	+40
	504	141	71	3500	A3	3.78	0.52	1.20	1 11	28	47	-38	+13	+40
	504WA	141	71	3500	A3	4.11	0.60	1.30	1.05	28	47	-38	+13	+48
	504WA	141	71	3500	A3	4.11	0.55	1.30	1.17	27	43	-38	+ 9	+42
	504	141	71	3500	M4	3.70	0.89	1.60	1.01	31	63	-38	+25	+52
	504	141	71	3500	M4	3.70	0.96	2.10	1.03	30	58	-38	+21	+47
	504WA	141	71	3500	M4	4.11	0.91	2.00	.96	30	58	-38	+21	+58
	504WA	141	71	3500	M4	4.11	0.70	1.80	1.02	29	53	-38	+17	+53
MERCEOES BENZ	240	147	62	3500	A4	3.69	0.11	1.0	1.74	27	42	-46	+ 2	+23
	240	147	62	3500	A4	3.69	0.15	0.8	1.75	27	42	-46	+ 2	+23
	240	147	62	3500	A4	3.69	0.10	1.77	1.77	28	47	-46	+ 5	+28
	240	147	62	3500	M4	3.69	0.19	1.4	1.63	29	53	-46	+ 9	+32
	240	147	62	3500	M4	3.69	0.14	0.8	1.44	30	58	-46	+13	+36
	300	183	110	4000	A4	3.07	0.17	0.8	2.04	26	53	-23	+33	+43
	300	183	77	4000	A4	3.46	0.10	1.2	1.86	25	47	-46	+ 5	+22
GENERAL MOTORS	DELTA	350	120	4500	A3	2.41	0.64	1.50	1.62	24	60	-27	+41	+27
	CUSTO	350	120	5000	A3	2.73	1.08	1.80	1.60	22	69	-36	+33	+33

NOTES ON SHEET 2

TABLE 2-4. 1978 FUEL ECONOMY SUMMARY OF DOMESTIC AND FOREIGN DIESEL PASSENGER CARS, LIGHT TRUCKS AND VANS (SHEET 2)

MANUFACTURER	MODEL	ENGINE / VEHICLE SPECIFICATIONS					EMISSIONS (GMS/MILE) (URBAN)		
		CID	HORSEPOWER	INERTIA	TRANSMISSIONS	REAR AXLE RATIO	HC	CO	NO_x
LIGHT DUTY TRUCKS (b)									
INTERNATIONAL HARVESTER	PICKUP	247	105	4500	M4	3.7	0.72	0.97	1.50
GENERAL MOTORS	PICKUP	350	120	4500	A3	2.76	0.88	1.80	1.56
	PICKUP	350	120	5000	A3	2.76	0.80	1.70	1.55
	PICKUP	350	120	5000	A3	3.40	0.76	1.60	1.79

MANUFACTURER	MODEL	FUEL ECONOMY		HP/WT	FUEL ECONOMY (c)	FUEL ECONOMY (d)
		COMPOSITE (MPG)	% DIFFERENCE*	% DIFFERENCE*	% DIFFERENCE NORMALIZED FOR HP/WT	% DIFFERENCE NORMALIZED FOR HP/WT AND RAR
LIGHT DUTY TRUCKS						
INTERNATIONAL HARVESTER	PICKUP	27	80	-28	+51	+83
GENERAL MOTORS	PICKUP	24	60	-17	+34	+25
	PICKUP	22	47	-27	+23	+14
	PICKUP	22	47	-27	+23	+37

NOTES:
(a) Percent differences based on 1978 EPA 49-state average passenger spark ignition cars (Domestic and Import).
(b) Percent differences based on 1978 EPA 49-state average spark ignition cars (Domestic and Imports).
(c) Normalization Factor = 0.55 (% change in Fuel Economy per % change in HP/IW).
(d) Normalization Factor = 0.7 (% change in Fuel Economy per % change in HP/IW and Rear Axle Ratio).
* 1978 Gasoline Average

offensive. Both odor and particulates may present potential health hazards, and this could limit or even eliminate the number of diesel passenger cars and trucks in the future automotive field. The effect of particulates on health is not known. Futher research is underway in this area at EPA.

The most immediate concern about the diesel engine today is the general objections to its exhaust, which under prolonged idling and light load conditions emits odor and noxious irritants, the blue and white smoke emitted under cold engine operating conditions, and the ability to start the engine under generally extreme cold weather conditions.

Even with these disadvantages the diesel engine still offers the best potential for improvement of the fuel economy. Most diesels today have not been fully developed to overcome these deficiencies, and they potentially merit further attention. The following sections address and quantify areas in which fuel economy improvement can be accomplished.

REFERENCES FOR SECTION 2

1. Arthur W. Judge, High Speed Diesel Engines: With Special Reference to Automotive, Stationary, and Marine Types., Chapman and Hall Ltd., London, 1967, 6th Edition.

2. Diesel Report, Publisher Robert Bosch Gmbit Postpach 50,700 Stuttgart 1, West Germany, 1977.

3. "Comparison of Diesel and Gasoline Engines for Passenger Car Usage," General Motors Corporation, presented at the Workshop on Regulated Diesel Emissions and Their Potential Health Effects, Washington, April 27-28, 1978.

3. FACTORS AFFECTING FUEL ECONOMY

The factors affecting the fuel economy of passenger automobiles, light trucks and vans powered by diesel engines are:

1. the engine efficiency,
2. the matching of the engine's power/efficiency characteristics to a given vehicle,
3. the characteristics of the vehicle, and
4. the driving cycle to which the vehicle is subjected.

The driving cycle has been fixed by the Energy Policy and Conservation Act of 1975 for purposes of Federal regulation of fuel economy. It is a composite of the Urban and Highway driving cycles of the Federal Test Procedure (FTP). The vehicle characteristics of importance are weight, aerodynamic drag, rolling resistance, and accessories. The matching of the appropriate engine to the vehicle is most critical and has a first-order effect on fuel economy, acceleration performance, and emissions. An engine that produces high horsepower for a given vehicle weight gives high acceleration performance but generally poor fuel economy. Here, matching is taken in its broadest sense, to include not only the initial selection of the engine but also the devices that determine an engine's operating schedule. Thus, the match for a given engine-vehicle combination is also governed by transmissions, axles and turbochargers.

Engine efficiency characteristics can be improved in a number of ways which are categorized as follows:

a) Reduction in engine friction primarily through control of low viscosity lubricants, as well as reduction of the engine's compression ratio.

b) Improvements in injection and combustion system designs to distribute the energy released during combustion without delaying the process of combustion. This can be accomplished by control of mixing and injection system characteristics such as timing, duration, shaping and atomization.

c) Determination of optimal control strategies and feedback controls for engine control variables (injection timing, and rates, exhaust gas recirculation, turbocharging, and etc.).

d) Optimization of turbocharging and recovery of wasted exhaust energy along with minimization of heat heat loss.

In the preliminary study presented below the fuel economies were expressed mathematically in the form:

$$FE = A(IW)^{a}(RAR)^{b}(CID)^{c}$$

where

A = coefficient

CID = engine displacement

IW = inertia weight

RAR = rear axle ratio

a,b,c = sensitivity coefficients, due to IW, RAR and CID

Both passenger car and light truck and van data were treated separately, and the following sensitivities were deduced:

SENSITIVITIES

	a	b	c
passenger car	-0.77	-0.735	-0.325
light truck & van	-0.83	-0.2	-0.338

Note: the sensitivity b (percent change in fuel economy to percent change in rear axle ratio) is to be taken with caution because of the limited data.

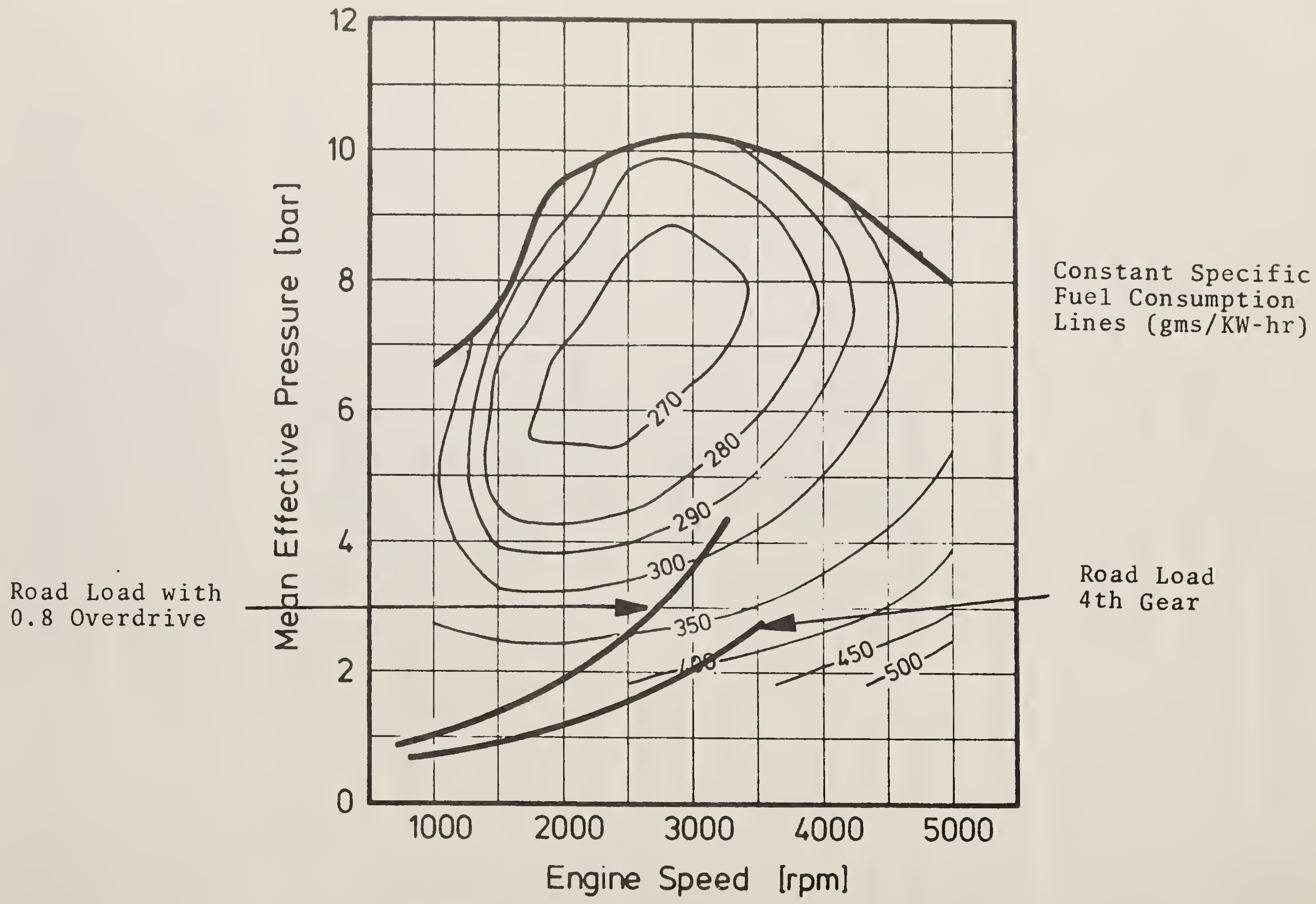
The sections that follow deal with many of these items in detail and attempt to quantify the fuel economy potential to the extent data is at hand.

3.1 ENGINE VEHICLE MATCHING

Diesel engines exhibit different levels of efficiency over their operating load-speed range. The optimum value of efficiency occurs at a particular speed and horsepower close to full throttle. The efficiency of the engine at other engine running conditions (i.e. load-speed points) is lower and is influenced by heat transfer at low speeds and friction at high speeds. Engine design and settings, such as injection and valve timing, combustion characteristics, air swirl and turbulence, manifolding and accessories (i.e. water and fuel pump and etc.) also have an influence on the efficiency of the engine. There are various means of improving a vehicle's fuel economy. The first is by utilizing a calibrated engine efficiently through changes in the vehicle and power train and the second is through systematic utilization of energy through changes in design and operating characteristics in the engine itself. This section is devoted to the first requirement.

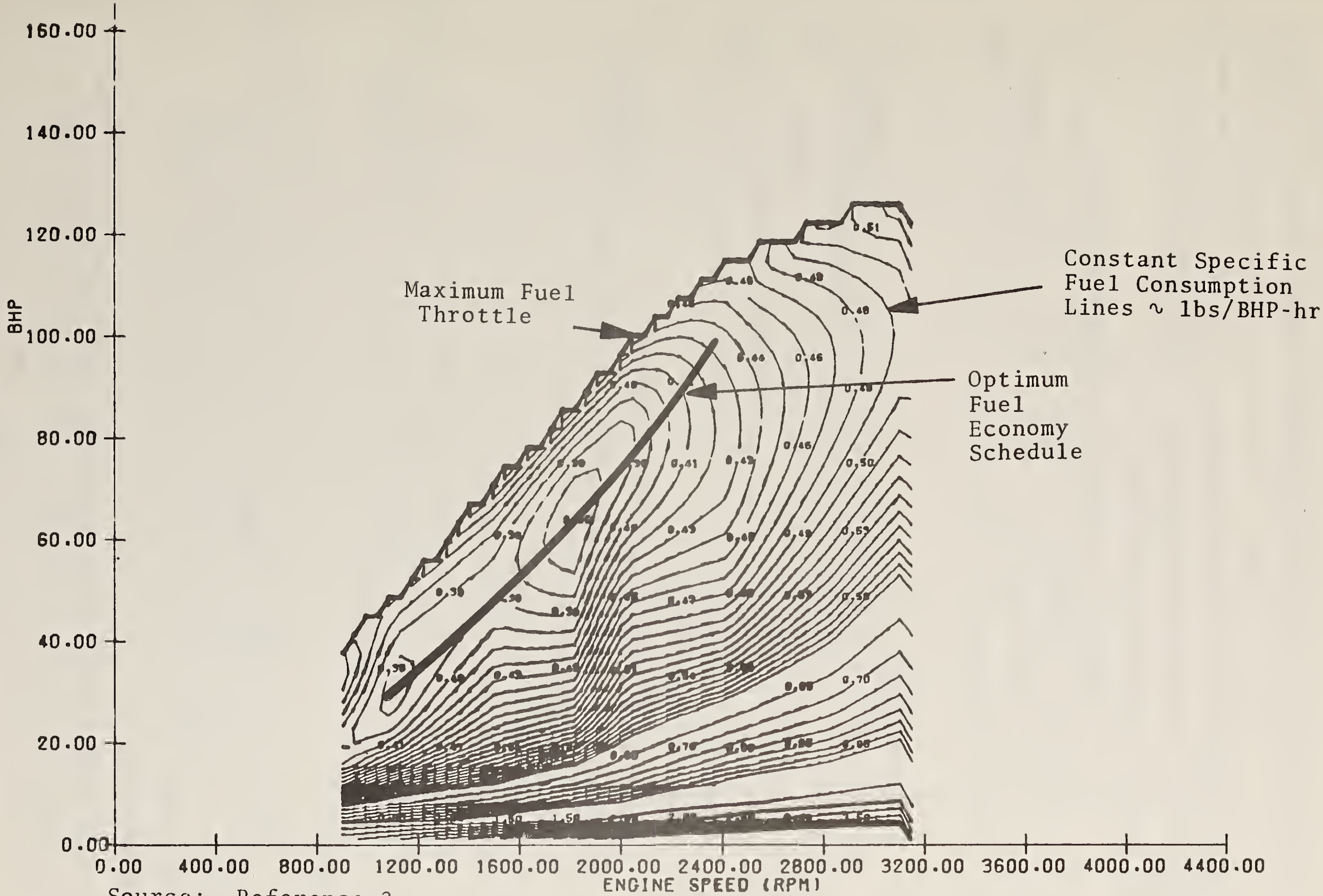
The power required from the engine is dictated by the vehicle weight and speed, the terrain and accessories. The speed at which the engine satisfies the power demand is determined by the transmission, axle ratio, and tire size. If the engine were operated closer to its peak efficiency more of the time during driving, greater improvements in fuel economy could be realized. The selection of proper engine, axle ratio, and transmission to provide optimum balance between fuel economy and acceleration performance is here termed vehicle-engine matching.

Figures 3.1-1 and 3.1-2 aid in visualizing the vehicle-engine matching concept. A typical diesel specific fuel consumption map is shown. The line labeled "ROAD LOAD" represents the schedule demanded on the engine by a vehicle travelling along a level road at increasing speeds in top gear (DIRECT DRIVE). The labeled "ROAD LOAD WITH OVERDRIVE" is the schedule if an 0.8 RATIO OVERDRIVE is added to the vehicle. Note that the overdrive schedule enables the calibrated engine to operate at lower specific fuel consumption (higher efficiency),



Source: Reference 1, (Note: References are listed at end of applicable sections).

FIGURE 3.1-1. DIESEL ENGINE FUEL CONSUMPTION MAP



Source: Reference 2.

FIGURE 3.1-2. TYPICAL DIESEL ENGINE FUEL CONSUMPTION MAP but the margin of power between the full throttle curve and the road load curve is less at any given speed. Thus, the instantaneous power margin available for acceleration is diminished and the fuel economy is increased with the 0.8 ratio. In other words, the new driveline ratio re-matches the engine operating condition to the vehicle operating condition.

The following subsection deals with engine-vehicle matching in detail, especially as affected by transmissions, axle ratios, and turbochargers. The fuel economy and acceleration sensitivities are quantified to the extent data is available.

3.1.1 Transmissions

The speeds at which engines deliver power are controlled by transmissions. A single schedule of power/speed points exists for any calibrated engine that is optimum for fuel economy. It is the path that travels through islands of minimum specific fuel consumption. Figure 3.1-2 illustrates the optimum schedule on a typical engine map.

Fuel economy improvements with transmissions results from (1) controlling engine speed closer to the optimum schedule and (2) higher transmission efficiencies. The transmission characteristics most significant to engine speed control are overall ratio range and transmission ratio control.

The optimum schedule can be approached by multi-speed, step ratio transmissions. By increasing the overall transmission ratio, the average operating condition is moved towards the optimum schedule.

Table 3.1-1 shows test results for a 4- and 5-speed manual transmission on a 2250 lb inertia weight Volkswagen turbocharged Rabbit diesel obtained at EPA.[1] The corresponding gear ratios were 3.45, 1.54, 1.32 and 0.97 for the four-speed transmission.

The speed logic for the 4-speed manual was 1-2 @ 18 mph, 2-3 @ 33 mph, and 3-4 @ 47 mph. The shift logic for the

TABLE 3.1-1. EFFECT OF TRANSMISSION ON FUEL ECONOMY AND EMISSION

	Transmission Type	Emissions grams/mile			Composite Fuel economy (MPG)
		HC	CO	NO_x	
IW = 2250 lbs.	4-speed (manual) shift logic	0.4	0.99	1.15	49.0
	5-speed (manual) shift logic	0.28	0.7	1.3	56.2
	Percentage Difference	-30%	-29%	+13%	+14%
General Motors IW = 3000 lbs.	3-speed automatic	0.55	--	0.25	19.6
	4-speed automatic with lockup	0.41	--	0.3	23.8
	Percentage Difference	-25%	--	+20%	+21%

5-speed manual was 1-2 @ 9 mph; 2-3 @ 20 mph; 3-4 @ 30 mph, 4-5 @ 40 mph. The test data indicates that by adding an extra gear, the HC and CO emission decreased, and NOx increased, providing a 14 percent increase in fuel economy. General Motors[3] predicted a similar increase in fuel economy for their Vehicle Simulation studies on their advanced concept low NOx producing diesel engine operated at a compression ratio of 29:1 (see Table 3.1-1). Part of the fuel economy gain was achieved under torque converter lock up conditions.

To eliminate the effect of shift logic on the fuel economy gains noted in Table 3.1-1 some additional Vehicle Simulation studies were conducted by DOT/TSC, Table 3.1-2, for the transmission shift points specified in Title 40 of the code of Federal Regulations, (1st-to-2nd at 15 mph, 2nd-to-3rd at 25 mph, 3rd-to-4th at 40 mph and 4th-to-5th at 50 mph). In this study two standard vehicles and engines were selected, a Rabbit passenger car vehicle in the 2250 lb inertia weight class with a turbocharged engine, and a Fiat 131 passenger car in the 3500 lb inertia class with a naturally aspirated engine. Simulation results on the Fiat turbocharged engines were also included for fuel economy comparisons. The Vehicle Simulation results show that a 3 to 5 percent gain in fuel economy is achieved with overdrive.

Further Vehicle Simulation studes, Table 3.1-3, were conducted for the turbocharged Rabbit diesel. These tests examined the effect of transmission shift points on fuel economy. For the sake of comparison, the EPA transmission shift points (1-2 @ 15 mph, 2-3 @ 25 mph, and 3-4 @ 40 mph) and the Volkswagen shift points (1-2 @ 18 mph, 2-3 @ 33 mph, 3-4 @ 47 mph) were used for the four-speed manual and the EPA transmission shift points (1-2 @ 15 mph, 2-3 @ 25 mph, 3-4 @ 40 mph and 4-5 @ 50 mph) and the Volkswagen shift points (1-2 @ 9 mph, 2-3 @ 20 mph, 3-4 @ 30 mph, and 4-5 @ 40 mph) were also used for the five-speed manual transmission. The Vehicle Simulation results show a 3.5 percent gain in fuel economy for the EPA transmission shift points and a 12 percent gain for the VW transmission shift points

TABLE 3.1-2. EFFECT OF TRANSMISSION ON FUEL ECONOMY AND ACCELERATION (SIMULATION RESULTS)

Inertia (lbs.)	Engine HP	Rear Axle Ratio	Transmission Type	Acceleration 0-60 mph (sec.)	Acceleration 40 mph to 60 mph (sec.)	Standing Start 5 sec. acceleration (distance)	Composite Mpg.	Percentage differences in Fuel Economy
2250	70 TC*	3.88	4 speed manual	14.69	7.18	121.44	44.10	5.14%
		3.88	5 speed manual	14.52	7.08	121.44	46.37	
	70 NA	3.0	4 speed manual	18.03	9.34	105.6	33.63	4.46%
		3.0	5 speed manual	19.4	9.71	105.6	35.13	
3500	90 TC	2.6	4 speed manual	14.47	7.5	132.0	33.98	3.5%
		2.6	5 speed manual	14.47	7.5	132.0	35.15	

*Note: Acceleration times indicated for comparison reasons.

1) acceleration start from idle
2) 0.55 sec. used for shift times

TABLE 3.1-3. EFFECT ON SHIFT LOGIC AND TRANSMISSION ON FUEL ECONOMY (VW TURBOCHARGED ENGINE, IW=2250 lbs)

DATA SOURCE		Fuel Economy Urban (mpg)	Highway (mpg)	Composite (mpg)
4 Speed Transmission	VEHSIM (VW-SHIFT POINT)	42.8	49.23	45.47
	VEHISM (EPA-SHIFT POINT)	45.51	50.07	47.45
5 Speed Transmission	VEHISM (VW-SHIFT POINT)	47.53	55.43	50.79
	VEHISM (EPA-SHIFT POINT)	45.63	54.29	49.16

when comparing the five to four speed transmissions.

Additional fuel economy gains can be achieved (Table 3.1-4) by cutting off the fuel supplied during the vehicle's deceleration phase and during the idle phase. When fuel is cut off during deceleration an additional 10 percent fuel economy gain is achieved, with larger gains obtained during highway rather than urban driving (approximately 16 percent compared to 7 percent). An additional 4 percent gain can be obtained in fuel economy when the fuel is shut off during idle in the urban driving mode.

3.1.2 Axle Ratios

The fuel economy senstitivity to axle ratio is:

$$FE = FE_1 \left[\frac{RAR_1}{RAR}\right]^b$$

where RAR = axle ratio

b = senstitivity

An average sensitivity determined from statistical inputs of 22 domestic and foreign NATURALLY ASPIRATED diesel passenger cars is -0.735 with a multiple r-square of 0.97. The average sensitivity does not necessarily apply to individual vehicles of course. Tables 3.1-5 and 3.1-6 show the effect of rear axle ratio on fuel economy, emissions and also acceleration performance as a function of inertia weight for a Fiat 8144.81, 70 horsepower naturally aspirated engine and a Fiat 8140.81, 90 horsepower turbocharged engine. Note that the hydrocarbon, carbon monoxide and nitrogen oxide emissions increase at an average rate of approximately 0.05, 0.0, and 0.1 grams/mile per tenth change in rear axle ratio respectively for the turbocharged vehicle and 0.05, 0.05, 0.025 grams/mile per tenth change in rear axle ratio respectively for this naturally aspirated vehicle over the rear axle ratios considered. Note the faster accelerations provided by the turbocharged vehicle version since the 0 to 60 mph and flying starting maneuvers are primarily a function of peak power to vehicle inertia weight. Standing start

TABLE 3.1-4. EFFECT OF FUEL SHUT OFF - SIMULATED RESULTS (VW-RABBIT TURBOCHARGED ENGINE)

			Fuel Economy	
		Urban (mpg)	Highway (mpg)	Composite (mpg)
4-speed transmission	Base Line	42.79	49.23	45.47
	Fuel Shut Off During Deceleration	46.09 (7.7%)*	57.03 (15.8%)	50.4 (10.8%)
	Fuel Shut Off During Deceleration and Idle	48.6 (13.58%)	57.1 (15.9%)	52.1 (14.58%)
5-speed transmission	Base Line	47.53	55.43	50.79
	Fuel Shut Off During Deceleration	50.98 (7.25%)	64.22 (15.8%)	56.2 (10.65%)
	Fuel Shut Off During Deceleration and Idle	54.04 (13.7%)	64.27 (15.9%)	58.21 (14.6%)

*Note: Bracketed values denote percent increase in fuel economy over baseline engine configuration without fuel shut off

TABLE 3.1-5. EFFECT OF REAR AXLE RATIO ON EMISSIONS AND FUEL ECONOMY (Part 1 of 2)

TURBOCHARGED FIAT ENGINE

Inertia Weight (lbs.)	Rear Axle Ratio	Emissions HC/CO/NOx gms/mile	Composite FTP-Cycle Fuel Economy
	2.4	0.27/1.3/1.3	39.8 (5.6%)
3000	2.6	0.28/1.3/1.5	37.7 -
	2.8	0.29/1.3/1.6	35.7 (-5.3%)
	2.4	0.26/1.3/1.4	38.2 (+6%)
3500	2.6	0.28/1.3/1.5	36.0 -
	2.8	0.28/1.3/1.7	33.9 (-5.8%)
	2.4	0.26/1.3/1.5	36.2 (±5.2%)
4000	2.6	0.27/1.3/1.6	34.4 -
	2.8	0.27/1.3/1.7	32.4 (-5.8%)

TABLE 3.1-5. EFFECT OF REAR AXLE RATIO ON EMISSIONS AND FUEL ECONOMY (Part 2 of 2)

NATURALLY ASPIRATED FIAT ENGINE

Inertia Weight (lbs.)	Rear Axle Ratio	Emissions HC/CO/NOx gms/mile	Composite FTP-Cycle Fuel Economy
	2.8	0.29/1.5/1.4	40.1 (+3.6%)
2500	3.0	0.3/1.6/1.4	38.6 -
	3.2	0.31/1.8/1.5	37.0 (-4.1%)
	2.8	0.29/1.5/1.5	37.9 (+3.8)
3000	3.0	0.3/1.6/1.5	36.5 -
	3.2	0.31/1.7/1.6	35.0 (-4.12)
	2.8	0.28/1.5/1.6	35.7 (+3.8%
3500	3.0	0.3/1.6/1.7	34.4 -
	3.2	0.31/1.7/1.7	33.2 (-3.5%)

Reference: DOT-TSC-Contract 1424; "Light Weight Automotive Diesel Power Plant Data Base"

Note: The static injection timing represents approximately equal emission performance. It does not represent the manufacturers' optimized engine.

Fiat Turbocharged Research Engine; Station Injection Timing = +5°ATDC

Fiat Naturally Aspirated Research Engine; Static Injection Timing = +1° BTDC

TABLE 3.1-6. EFFECT OF REAR AXLE RATIO ON VEHICLE ACCELERATION PERFORMANCE

Inertia Weight (lbs)	Rear Axle Ratio	Top Speed (Km/hr)	Engine Speed (RPM)	0→60mph (SEC)	starting (1000 meters) (SEC)	Flying start 18.6 mph (top gear) 1000 meters (SEC)
TURBOCHARGED FIAT ENGINE						
2750	2.4	172	3800	12.3	34.6	40.7
	2.6	172	4130	12.2	34.5	38.8
	2.8	165	4270	12.1	34.4	37.3
	3.0	155	4290	12.0	34.4	36.2
	3.2	146	4300	12.0	34.6	35.5
	3.4	140	4380	11.9	35.1	35.3
NATURALLY ASPIRATED FIAT ENGINE						
2750	2.8	158	4085	16.8	38.7	42.4
	3.2	146	4300	16.5	38.4	39.8
*	-	172		11.4	33.8	30.9

*Maximum Achieveable Performance (Constant Power) 2750

Note: Fiat Turbocharged 8140.81 Research Engine; Static Injection Timing = +5° BTDC
Fiat Naturally Aspriated 8144.01 Research Engine; Static Injection Timing = +1° BTDC

Reference: DOT-TSC-Contract 1424; "Light Weight Automotive Diesel Power Plant Data Base" accelerations are not strongly influenced by changes in rear axle ratios. There is even less influence on rear axle ratio on acceleration for the turbocharged diesel vehicle because the engine is designed for maximum torque at as low an engine speed as possible, followed by a constant horsepower at higher speeds (i.e., torque curve decreases in a hyperbolic way as engine speed increases). Data is also presented in Table 3.1-6 to demonstrate the best performance achieved with constant engine power output over its operating range (the ideal case is an electric motor), compared with the turbocharged engine power output at high speeds. The turbocharged engine approaches the ideal case when its power output is constant in the 3000 to 4000 rpm range without increasing fuel consumption.

The fuel economy sensitivities to rear axle ratio change are displayed in Table 3.1-7. It is noted that the TURBOCHARGED vehicle exhibits higher fuel economy sensitivity than the NATURALLY ASPIRATED vehicle. Generally the sensitivity decreases with an increase in inertia weight and decreases in rear axle ratio.

3.1.3 Horsepower-To-Weight Sensitivity

The fuel economy sensitivity to horsepower-to-weight is

$$FE = FE_1 \left\{ \left(\frac{HP}{IW}\right)_1 \Big/ \left(\frac{HP}{IW}\right) \right\}^b$$

where HP = engine horsepower

IW = inertia weight in pounds.

Figure 3.1-3 represents a calculated sensitivity for two naturally aspirated vehicles. These sensitivities are of interest because vehicles equipped with naturally aspirated engines have lower HP/weight ratios ($\sim$0.02 to 0.025). Note that by and large the average sensitivity is approximately -0.55 for vehicles with inertia weights ranging between 2500 and 4000 lbs. The VW data show a lower sensitivity of -0.27 for a 2000 lb. inertia weight vehicle.

TABLE 3.1-7. FUEL ECONOMY SENSITIVITIES TO REAR AXLE RATIO (FIAT ENGINE)*

Turbocharged Fiat Engine

Inertia Weight (lbs.)	HP/Wt.	Rear Axle ratio		
		2.4	2.6	2.8
3000	0.030	-0.63	-0.71	-0.78
3500	0.026	-0.67	-0.77	-0.88
4000	0.0225	-0.60	-0.72	-0.86

Naturally Aspirated Fiat Engine

Inertia Weight (lbs)	HP/Wt.	Rear Axle Ratio		
		2.8	3.0	3.2
2500	0.028	-0.52	-0.60	-0.69
3000	0.023	-0.52	-0.59	-0.68
3500	0.020	-0.51	-0.54	-0.58

*Sensitivies based on data presented in Table 3.1-3

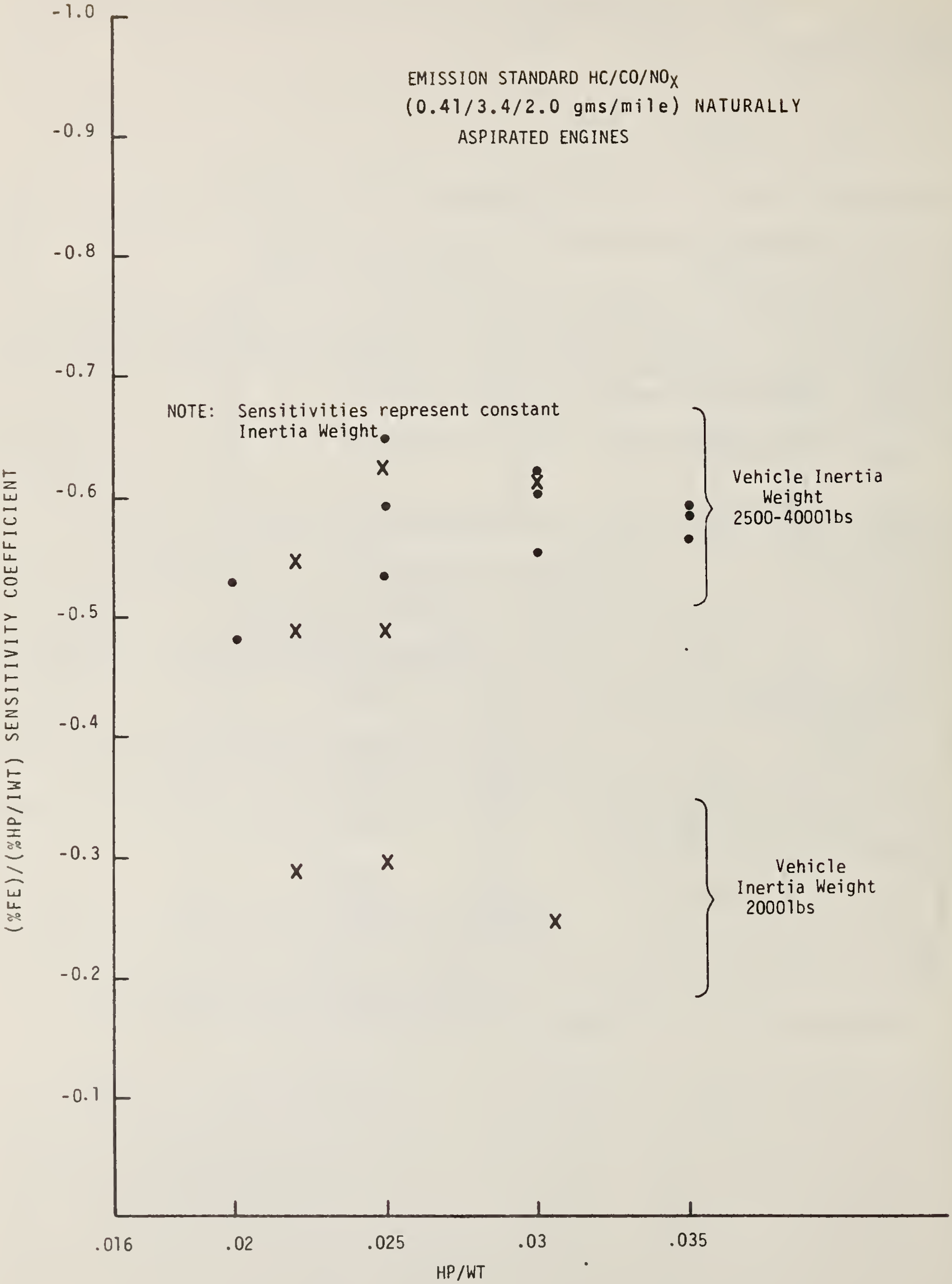


FIGURE 3.1-3. FUEL ECONOMY SENSITIVITY TO HORSEPOWER TO WEIGHT (APPLICATION)

3.1.4 Turbocharging and Compounding

Turbocharging is a technological process that could be used extensively in the 1980's. It offers considerable advantages for weight reduction, improved fuel economy and acceleration performance. The primary benefit of turbocharging is that the same engine displacement gives higher power outputs. A wastegate is employed with a turbocharger to limit the air flow at high engine speeds. This is coupled with a fuel pump pressure adjustment by sensing the inlet manifold pressure to prevent overfueling and black smoke.

Data from a DOT contract with Volkswagen(4) and tests performed by EPA, Table 3.1-8, show that the FUEL ECONOMY can be improved by at least 8 percent when a 50 horsepower naturally aspirated diesel engine's power in a 2250 lb. inertia weight vehicle is increased to 70 horsepower by turbocharging a swirl chamber engine under the current emission standard. Fuel economy improvements greater than 20 percent were predicted by Volkswagen by turbocharging diesel in vehicles in the 2750 and 3000 lb. inertia class. The improvement in fuel economy of the 2250 inertia weight vehicle becomes 29 percent by turbocharging when both naturally aspirated and turbocharged vehicles have the same HP-to-inertia weight performance. Here the acceleration performance of the naturally aspirated vehicle is brought up to the turbocharged vehicle by assuming an increase in the number of cylinders. Volkswagen also demonstrated that a 10 percent fuel economy advantage is retained with turbocharging under a more stringent emission standard, 0.41/3.4/1.0; HC/CO/NOx gms/mile for a 2250 lb inertia weight vehicle.

These fuel economy advantages of the turbocharging process are further supported by data collected by Mercedes-Benz. Representatives claim that the specific fuel consumption of their turbocharged diesel engines can be improved by ten percent in comparison with naturally aspirated vehicles. According to Mercedes-Benz, part of this improvement is reflected by a 40 percent improvement under part load performance.

TABLE 3.1-8. COMPARISON OF NATURALLY ASPIRATED AND TURBOCHARGED DIESEL ENGINE FUEL ECONOMIES(3)

Emission Level	Automobile Inertia Wt. Class (lbs.)	Naturally Aspirated Fuel Economy (mpg)	Turbocharged Fuel Economy (mpg)	Fuel Economy Difference Percent (%)
0.41/3.4/2.0 HC/CO/NO_x gms/mile	2250	41.0	45.0	+8
		44.0*	47.3	+7.5
	2750	34.0	42.0	+23.5
	3000	33.0	40.0	+21.2
0.41/3.4/1.0 HC/CO/NO_x gms/mile	2250	39.0	43.0	+10.3

*Note: 1) data based on EPA measurements.
2) a: Naturally Aspirated Engine; 50 HP
b: Turbocharged Engine; 70 HP
c: Fuel Economy Difference not corrected for equal HP/IWT

Besides providing better fuel economies, turbocharging also has the potential to lower the emission levels of HC/CO/NOx. This is illustrated in Figures 3.1-4 through 3.1-6, which provide summaries of tests conducted by Volkswagen. Without use of EGR as a means to reduce NOx emissions, turbocharging is effective in further reducing particulate emissions. However, with EGR, significantly higher particulate levels have resulted. These conclusions are based on limited test data.

Turbocharging is not without problems. The response time of turbochargers during low speed accelerations is still a major problem which limits the use of the engine's full throttle torque capability. Various schemes are under development to reduce this disadvantage, including a) reduction of compressor and turbine rotor inertia, b) use of a pelton wheel or burners, and c) electronic feedback systems. The other problem is that the fuel delivery must be controlled by sensing the inlet-manifold pressure in order to maintain a prescribed air/fuel ratio during initial operation of the turbocharger to eliminate over-fueling (black smoke). Such a system is available on rotary and inline pumps. Also, turbocharging may cause higher thermal loadings on this piston, and higher combustion chamber pressures. These may be attended by piston cooling, water cooling and wastegate.

Presently, turbochargers are becoming available for engines approximately as small as 1.5 liter[5] in swept volume, a size that is suitable for diesels in light duty applications.

A DOT contract* is in progress with Fiat to provide further experimental data for a turbocharged 2.44 liter engine with studies particularly aimed at defining the limiting maximum torque over a wide speed range. The preliminary data, Table 3.1-9, shows that the fuel economy of their turbocharged engine is better than ten percent in the urban cycle of the FTP while the highway fuel economy remains the same when compared to a naturally aspirated engine. Fiat attributes most of this fuel economy gain to the lower numeric rear axle ratio of the turbocharged engine/vehicle.

*DOT-TSC-1424

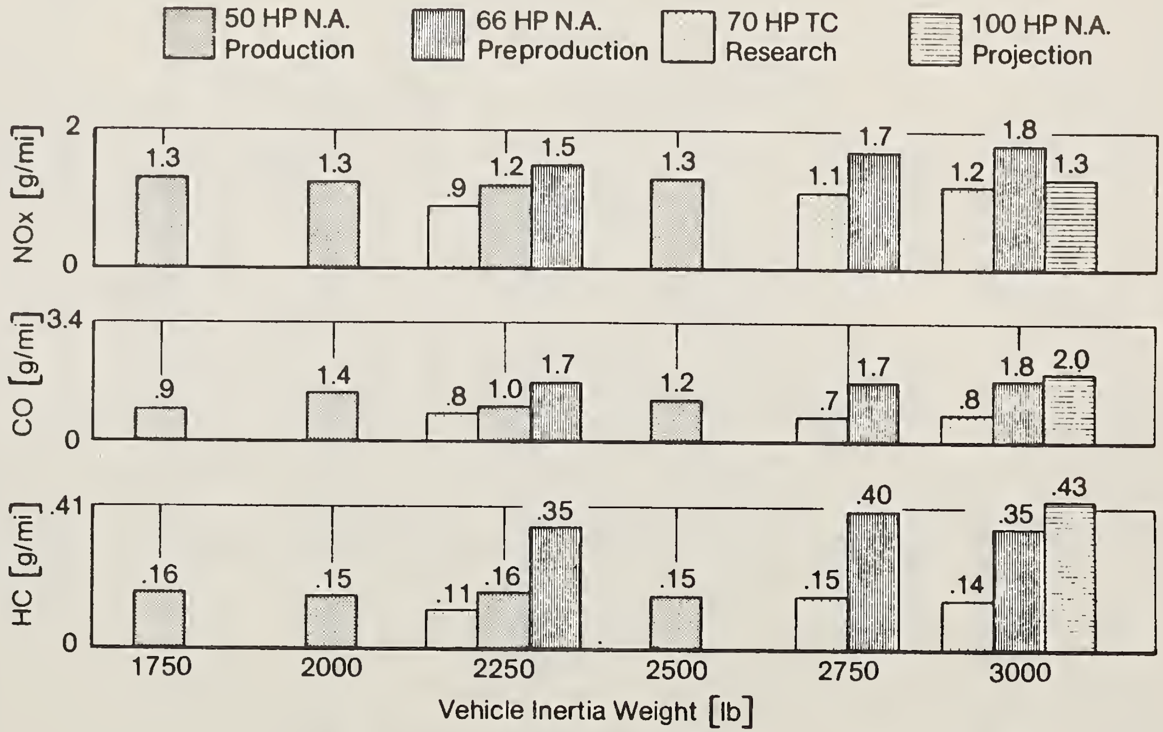


FIGURE 3.1-4. REGULATED EXHAUST EMISSIONS FROM VARIOUS DIESEL ENGINES (HC/CO/NOx = .41/3.4/1.0 GMS/MILE)
DETERIORATION FACTORS AND VARIANCES DUE TO MANUFACTURING (30% OF THE FIGURES INDICATED) ARE NOT TAKEN INTO ACCOUNT.

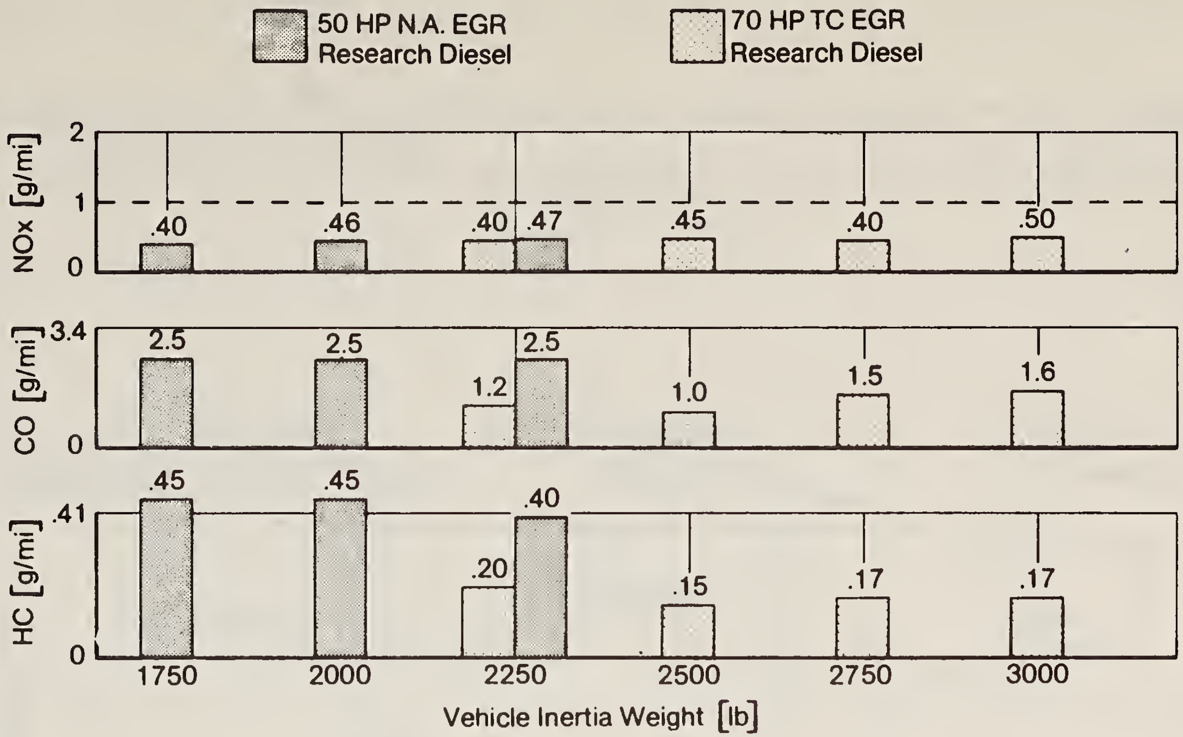


FIGURE 3.1-5. REGULATED EXHAUST EMISSION OF TWO DIESEL ENGINES WITH CONTROLLED EXHAUST GAS RECIRCULATION (PRELIMINARY DATA); (HC/CO/NOx = 0.41/3.4/1.0 GMS/MILE)
THE TRADE-OFF BETWEEN LOW NOx, AND ACCEPTABLE HC AND CO IS A FUNCTION OF THE AMOUNT OF EXHAUST GAS RECIRCULATED (EGR).
NOTE THE INCREASED SMOKE AND ODOR LEVELS WHICH ARE DUE TO EGR.

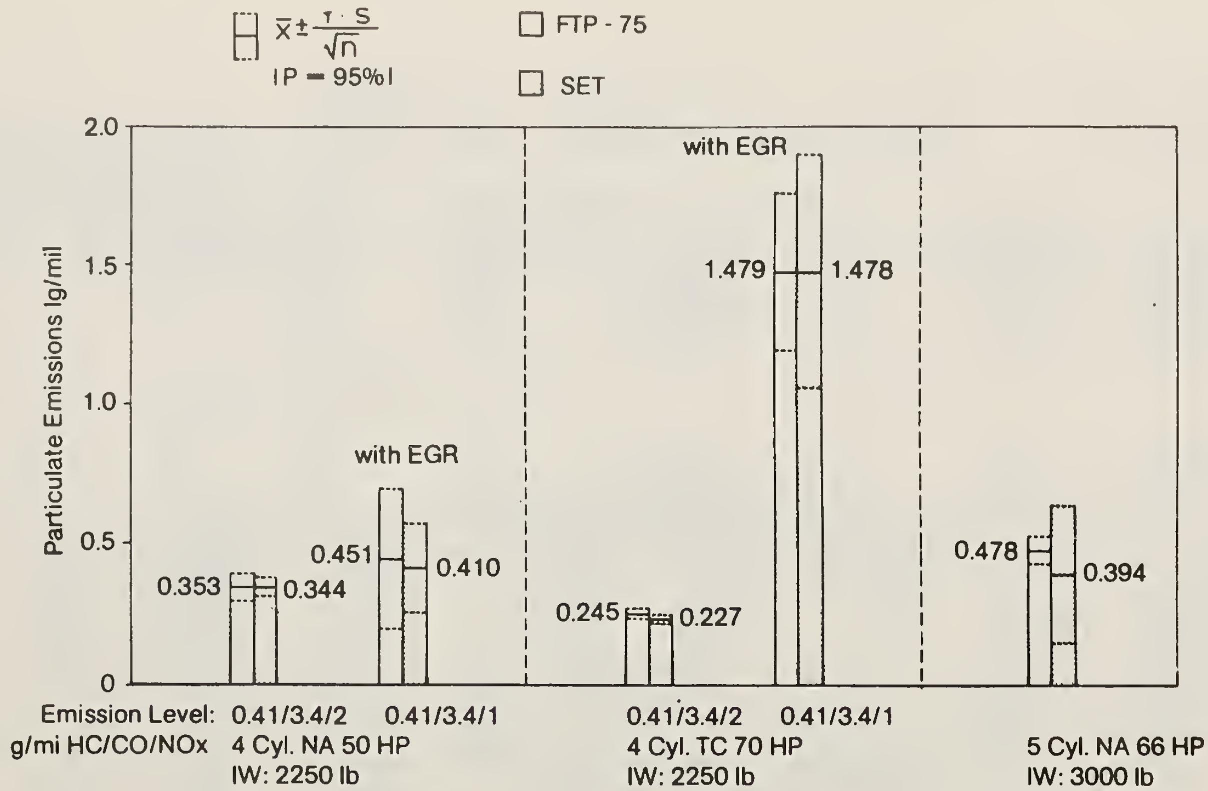


FIGURE 3.1-6. PARTICULATE EMISSIONS TESTS RESULTS FOR CURRENT DIESEL ENGINES AS MEASURED BY EPA RANGES FROM .30 TO .62 G/MI.
(SUMMARY REPORT ON THE EVALUATION OF LIGHT DUTY DIESEL VEHICLES, EPA, MARCH 1975, 75-21)

TABLE 3.1-9. COMPARISONS OF TURBOCHARGED AND NATURALLY ASPIRATED ENGINE FUEL ECONOMY AND EMISSION

I.W.	Engine	Rear Axle Ratio	HP/WT	Emission Comparison HC/CO/NOx gm/mi	Composite FE MPG	FE % difference
3000	NA	2.8	.023	0.29/1.5/1.5	37.9	---
	TC	2.8	.03	0.29/1.5/1.6	35.7	-6%
	TC	2.4	.03	0.27/1.3/1.3	39.0	+3%
3500	NA	2.8	.02	0.28/1.5/1.6	35.7	---
	TC	2.8	.026	0.28/1.3/1.7	33.9	-5%
	TC	2.4	.026	0.26/1.3/1.4	38.2	+7%

Reference: DOT-TSC-Contract 1424; "Light Weight Automotive Diesel Power Plant Data Base"

The ratio is 2.8 for the naturally aspirated engine and 2.4 for the turbocharged engine. Fuel economy comparisons for the same vehicle's rear axle ratio show the naturally aspirated engine is better by 5 to 6 percent, though the HP/IW ratio differs significantly. Fiat also showed that turbocharging is useful in reducing the exhaust emissions, particularly carbon monoxide and nitrogen oxide, which were reduced by ten percent with no increase in hydrocarbon levels.

Additional tests conducted by Fiat demonstrated that turbocharging has a dramatic effect on particulates. A reduction of 35 percent was achieved with turbocharging (0.55 gms/mile compared to 0.36 gms/mile for an engine in the 3000 lb interia weight class).

In addition to turbocharging, other technologies are currently being examined which may provide improvements in fuel economy and emissions as well as in performance. These technologies are still in early stages of development, and only limited data are available to determine their potential for passenger cars and light trucks. These technologies include intercooling, comprex system, turbocompounding, and the hyperbar concept.

a) Intercooling is a technology used simultaneously with turbocharging. It provides more power and improves fuel economy, with attendant reduction in nitrogen oxide emissions.[6] An air-to-air heat exchanger enables the intake charge to be cooled at the same time that it increases the air charge to the engine. Intercooling is used primarily in heavy duty truck applications and has not yet received attention for passenger car and light duty truck diesels.

b) Comprex is a supercharging technology whose primary benefit consists in increasing the power-to-weight ratio of diesel engines. This process provides a wider air charging application over the entire load-speed range of the engine, and it also provides a faster response to engine load variations than turbocharging does because of its direct coupling and higher torque capability at low engine speeds. The classic problems with comprex are the added complexity, bulk, weight and cost.

c) Turbocompounding and Rankine Bottoming cycle are two methods of extracting currently lost exhaust energy. Two-thirds of the input diesel energy is wasted in current engines through the coolant or by the outgoing exhaust. Both of these concepts are currently under development for heavy duty truck diesels but neither has received attention for passenger cars and light duty trucks. The power-producing mechanism of turbocompounding is quite similar to that of a turbocharged, after-cooled engine. As far as application is concerned, a low-pressure, free-power turbine is used to extract additional power from the high-temperature exhaust gas by transfering it to the crankshaft via reduction gears and a torsional isolator. Turbocompounding a turbocharged diesel offers improvement of 2 to 3 seconds in the response time of the engine under low speed, high torque accelerations. Recent data from a TARADCOM contract with Cummins Engine Company[7] also show an improvement in BSFC of 9 percent at maximum rated speed and load without insulating the engine's combustion chamber and manifolding. The Bottoming Cycle[8] also extracts energy from the exhaust but by a separate closed loop Rankine cycle. By adopting the Rankine cycle and incorporating insulated walls, the turbocompounding engine can achieve a 3.0 percent improvement in fuel economy at maximum rated speed and load.

d) The hyperbar combustor is a high pressure supercharging turbo-compressor for diesel engines which incorporates a self-sustaining power-augmented turboblower. Its advantages are that it can be independently operated under cold engine starting conditions; it permits diesel to be designed for lower compression ratios; it allows supercharging pressure to be adjusted independent of speed, and it allows flexible delivery of high torques at low speed when this is required. To date it has been limited to larger engines and requires further improvements to offset fuel economy disadvantages.

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2. Kidd, J., and Rogowicz, J., "Potential of Spark Ignition and Diesel Engines, Engine Catalog and Performance Analysis," Report No. DOT-TSC-NHTSA-79-39, October, 1979.
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5. French, C.C.J., "The Diesel is "the" Engine for High Annual Mileage Light Duty Vehicles," Ricardo Consulting Engineers, Report D.P. 77/943.
6. Parker, R.F., and Walker, J.W., "Exhaust Emission Control in Medium Swirl Direct Injection Diesel Engines," SAE 720755, 1972.
7. Kamo, R., Bryzik, W.W., "Adiabatic Turbocompound Engine Performance Production," SAE 780068, 1978.
8. Thermo Electron Corporation, "Status Report on Diesel Organic - Rankine Compound Engine for Long-Haul Trucks," material presented for Highway Vehicle Systems Contractors Coordination, DOE, Troy, Michigan, May, 1978.

3.2 VEHICLE CHARACTERISTICS

3.2.1 Weight

The relationship of fuel economy and vehicle inertia weight has been represented as follows

$$FE = FE_1 \ (IW_1/IW)^a$$

where FE = miles per gallon

IW = inertia weight in pounds

a = sensitivity

An average sensitivity "a" was determined by application of regression techniques to 22 domestic and foreign naturally aspirated diesel passenger cars, of which some constituted the 1978 EPA certification fleet. The value "a" was -0.77 (percent change in mpg per percent change in inertia weight). Although this is an estimate of the average sensitivity of the diesel fleet, it should be taken cautiously, given the samll number of data available for analysis.

The average sensitivity does not apply to individual vehicles; the following analysis of Fiat and VW data provides further insights into the effect of weight on fuel economy for single vehicles using naturally aspirated or turbocharged engines. The results are given in Figure 3.2-1. The trend in fuel economy sensitivity to weight change is less at lower vehicle weights. The Fiat turbocharged information follows consistently with the trend in sensitivity displayed by the naturally aspirated vehicles. The VW turbocharged vehicle exhibits a higher sensitivity relationship compared to vehicles with naturally aspirated engines with a twofold change at the higher inertia weight.

The VW turbocharged sensitivity data is also slightly influenced by the emission control strategy where slightly higher values of sensitivities are exhibited for the lower emission standard of 0.41/3.4/1.0 HC/CO/NOx gms/mile.

3.2.2 Aerodynamic Drag

The aerodynamic drag force, which resists the movement of a

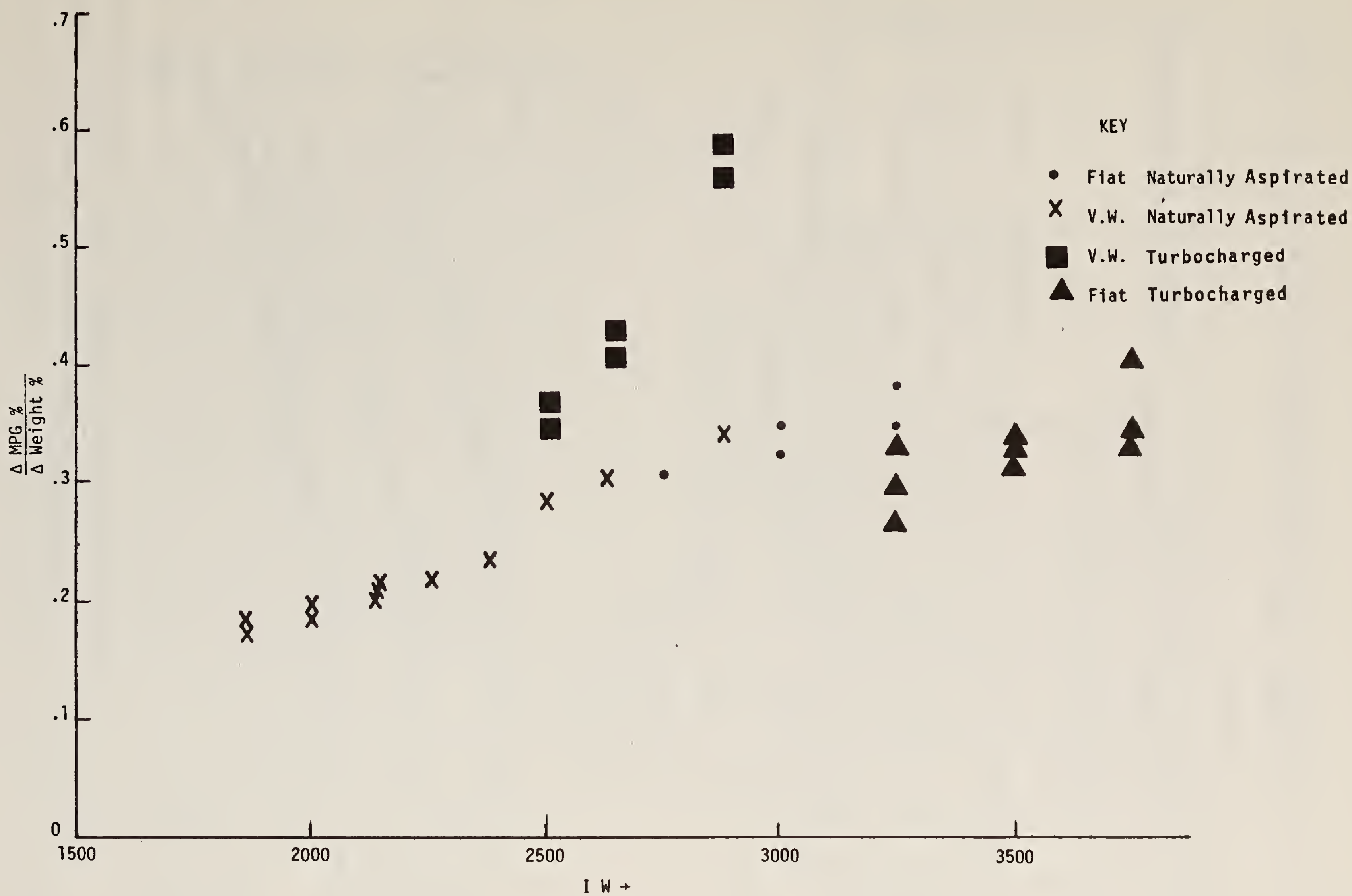


FIGURE 3.2-1. SELECTED LIGHT DUTY VEHICLE FUEL ECONOMY SENSITIVITY TO INERTIA WEIGHT

vehicle through still air, is described by the following relationship:

$$F = \text{aerodynamic drag force} = 1/2\ \rho\ V^2\ C_D\ A,$$

where ρ = air density,

V = velocity of the vehicle,

C_D = aerodynamic drag coefficient, which is a function of body shape and surface; and

A = vehicle frontal area.

The aerodynamic drag force is the sum of component forces resulting from a) the variation of pressure around the vehicle body, from b) the viscous shear of the air near the vehicle surface (friction or surface drag), and from c) the resistance of air passing through the vehicle (internal flow drag).

An approximate distribution of the drag component contributions to total drag for a typical sedan is:[1]

Pressure (form)	55%,
Induced (lift)	7%
Interference	17%,
Friction	9%, and
Internal	12%

The increase in fuel economy resulting from a reduction in drag coefficient depends upon vehicle characteristics, engine characteristics, and the drivetrain. To take full advantage of the drag reduction, the drivetrain of the vehicle should be optimized to the reduced vehicle power requirements.

Analytical studies performed by VW[2] showed that sensitivities (the percent improvement in fuel economy relative to the percentage of improvement in drag coefficient; (%ΔFE/%ΔCD) were approximately 0.35, 0.30 and 0.25 for a subcompact vehicle for aerodynamic drag coefficients of 0.45, 0.40 and 0.35 respectively.

Table 3.2-1 shows the composite cycle fuel economy/aerodynamic sensitivities for two passenger car vehicles in inertia weight class of 2250, and 3500 respectively. These figures were acquired

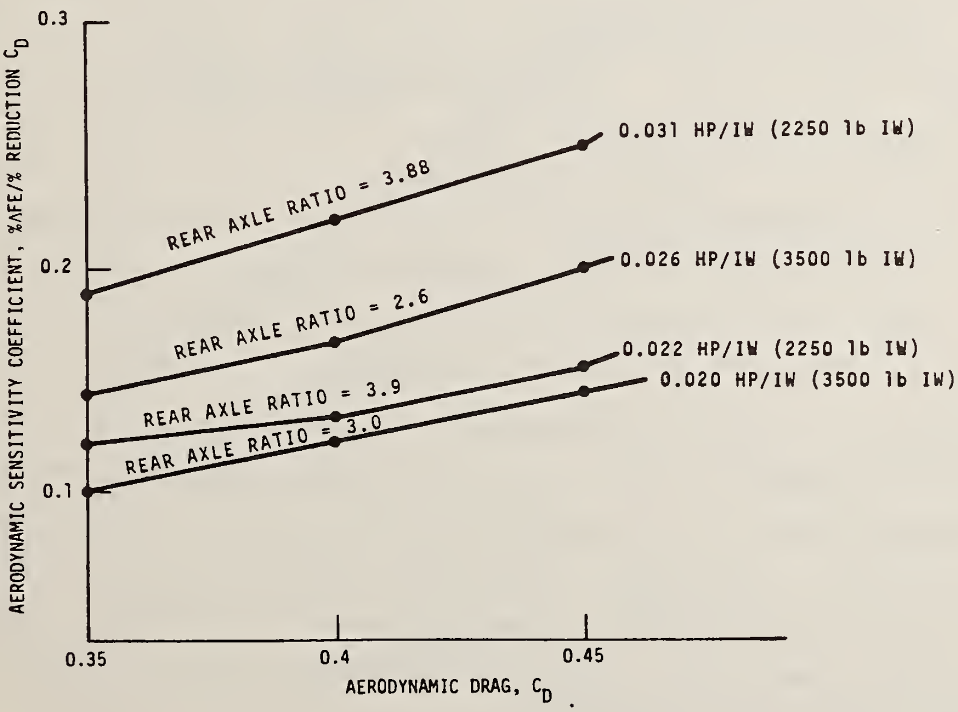
TABLE 3.2-1. FUEL ECONOMY SENTITIVITY TO AERODYNAMIC DRAG (%ΔFE/% REDUCTION C_D) CONSTANT REAR AXLE RATIO*

Vehicle Inertia Weight (lbs)	Engine Type	HP/IW	FUEL ECONOMY SENSITIVITIES		
			0.45 C_D	0.4 C_D	0.35 C_D
2250	NA**	0.022	0.16	0.14	0.13
	TC**	0.031	0.25	0.22	0.19
3500	NA**	0.020	0.15	0.13	0.11
	TC**	0.026	0.2	0.17	0.15

*Information on chart corresponds with tabular data

**Note: TC - Turbocharged

NA - Naturally Aspirated



using the DOT/TSC computer simulation program, VEHSIM for three assumed aerodynamic drag coefficients. Table 3.2-2 shows similar data when the rear axle ratios were modified to match the vehicle's road load at 50 mph for equal acceleration performance. The baseline vehicle drag coefficients were 0.46 for the 2250 lb IW vehicle, and 0.44 for the 3500 lb. IW vehicle.

Results of the Simulation study show that the sensitivities are approximately 0.16 - 0.25 for a 2250 lb inertia weight vehicle and 0.15 - 0.2 for a 3500 lb inertia weight vehicle when the aerodynamic drag coefficient is 0.45. The sensitivities decrease as the drag coefficient decreases, and they correlate with the horsepower-to-inertia weight. Those sensitivities also increase (Table 3.2.2) when the rear axle ratio is modified to match the vehicle's road load at 50 MPH. The increase is from 0.47 to 0.48 for the 2250 lb inertia weight vehicle and 0.39 - 0.43 for the 3500 lb inertia weight vehicle for an aerodynamic drag coefficient of 0.45.

3.2.3 Tire Rolling Resistance

The transition from bias to radial tires has taken place to a great extent in the passenger car fleet. It is now underway for the light truck fleet. The change lowers tire energy consumption without major sacrifices in riding comfort and handling. Further reductions in tire rolling resistance occur by altering tire geometry and increasing inflation pressures. Together, these changes promise improvements in fuel economy of 2 to 5 percent above radial tires or 5 to 8 percent above bias-belted tires, still in widespread use in the replacement market.

Higher inflation pressures is another approach to reducing rolling resistance. The trend is already apparent for high-performance cars. The Porsche 928 tires, for example, are inflated to 36 psi. The higher pressures affect ride quality, cornering, acceleration, and braking. Manufacturers of domestic cars may require five years for vehicle development and testing to accommodate the higher pressure tires.

TABLE 3.2-2. FUEL ECONOMY SENSITIVITY TO AERODYNAMIC DRAG (%ΔFE/% REDUCTION C_D)* MODIFIED REAR AXLE RATIO

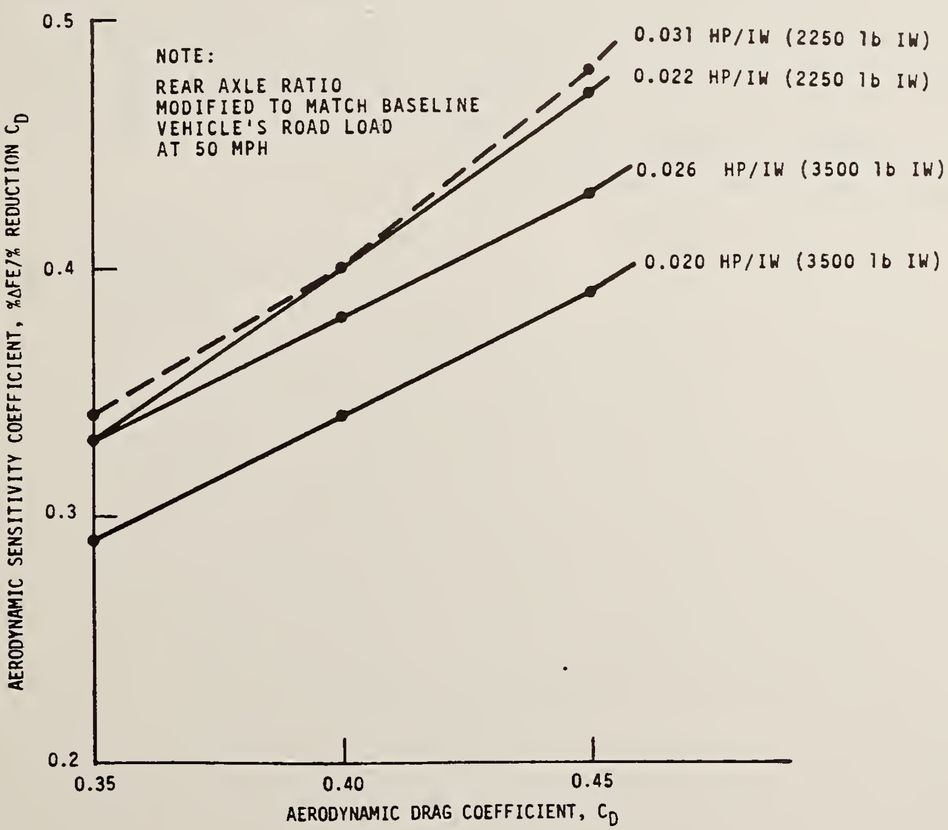
Vehicle Inertia Weight (lbs)	Engine Type	HP/IW	FUEL ECONOMY SENSITIVITIES		
			$0.45C_D$	$0.40C_D$	$0.35C_D$
2250	NA**	0.022	0.47	0.40	0.33
	TC**	0.031	0.48	0.40	0.34
3500	NA	0.020	0.39	0.34	0.3
	TC	0.026	0.43	0.38	0.33

*Information on chart corresponds with tabular data

**Note: TC - Turbocharged

NA - Naturally Aspirated

Fuel Economy Sensitivity to Aerodynamics represents values when modified to driveline to match vehicle's road load at 50 mph.



The elliptical tire is a widely advertised example of a geometric change that offers low rolling resistance. The elliptical tire requires few concessions in present suspension design. It requires a novel rim design, however, and poses interchangeability problems. A low-aspect ratio tire having conventional rim design can also offer low rolling resistance. The concept may prevail over the elliptical design. Table 3.2-3 gives the rolling resistance and estimated fuel economy improvement for the various size types.

Analytical simulation studies show that the sensitivities (fuel economy improvements relative to rolling resistance improvement) are approximately 0.09 and 0.13 for a 2250 lb. inertia weight vehicle, and 0.12-0.19 for a 3500 lb. inertia weight vehicle, at a tire rolling resistance coefficient of 10.0 lbs/1000 lbs (Table 3.2-4). These sensitivities increase (Table 3.2-5) when the rear axle ratio is modified to match the vehicle's road load at 50 mph to 0.19-0.22. and 0.26-0.32, for a 2250 lb., and 3500 lb. inertia weight vehicle. The simulation calculations also bear out that small vehicles are less sensitive to changes in tire rolling resistance and the larger vehicles exhibit a larger change. The turbocharged vehicles exhibit slightly larger sensitivities than the naturally aspirated vehicle because of their increased horsepower-to-inertia weight.

TABLE 3.2-3. TIRE ROLLING RESISTANCE

Type and Inflation Pressure	Rolling Resistance lb/K lb	Resistance Decrease Percent	*Estimated Fuel Saving Percent
Bias Ply - 24 psi	14	Base	Base
Bias Belted - 24 psi	13	7	1.6
Radial - 24 psi	12	14	4.2
Radial - 38 psi	11	21	4.7
Future Low Loss Tires - 38 psi	8	43	9.6

*Combined EPA driving cycles.

TABLE 3.2-4. FUEL ECONOMY SENSITIVITY TO TIRE ROLLING RESISTANCE* (%ΔFE/% REDUCTION ROLL)

Vehicle Inertia Weight (lbs)	Engine Type	HP/IW	FUEL ECONOMY SENSITIVITIES		
			8.0***	10.0***	12.0***
2250	NA**	0.022	0.07	0.09	0.11
	TC**	0.031	0.10	0.13	0.15
3500	NA	0.020	0.09	0.12	0.16
	TC	0.026	0.15	0.19	0.24

*Information on chart corresponds with tabular data

**Note: NA - Naturally Aspirated

TC - Turbocharged

***Tire Rolling Resistance (lbs/1000lbs)

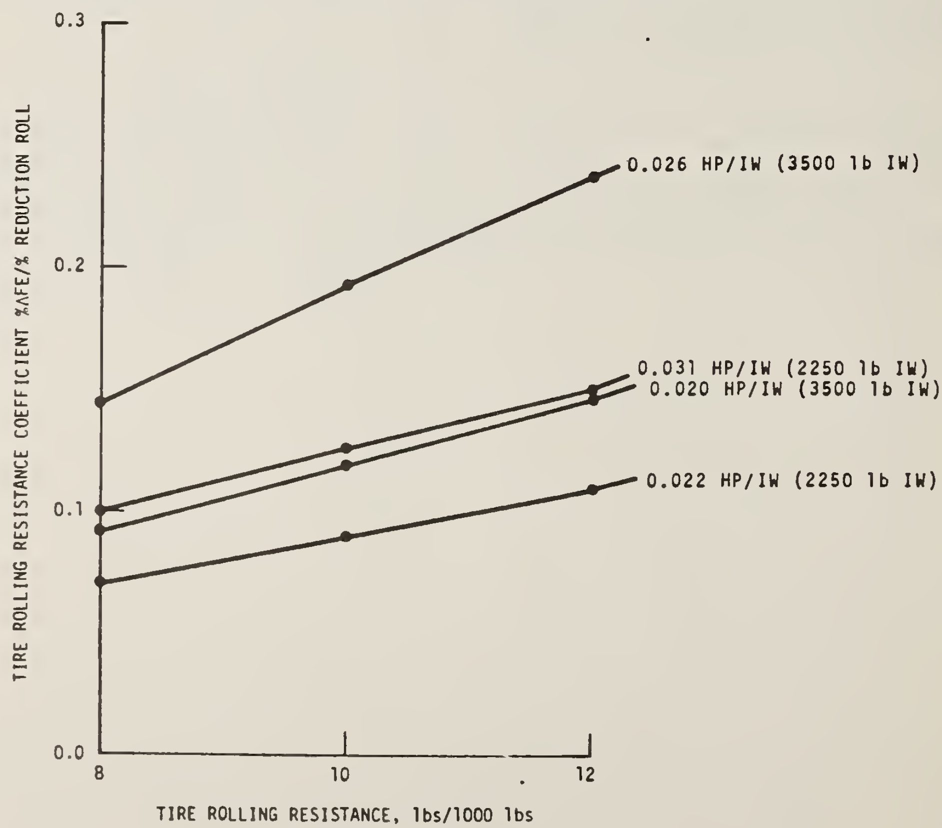


TABLE 3.2-5. FUEL ECONOMY SENSITIVITY TO TIRE ROLLING RESISTANCE (MODIFIED REAR AXLE RATIO)*

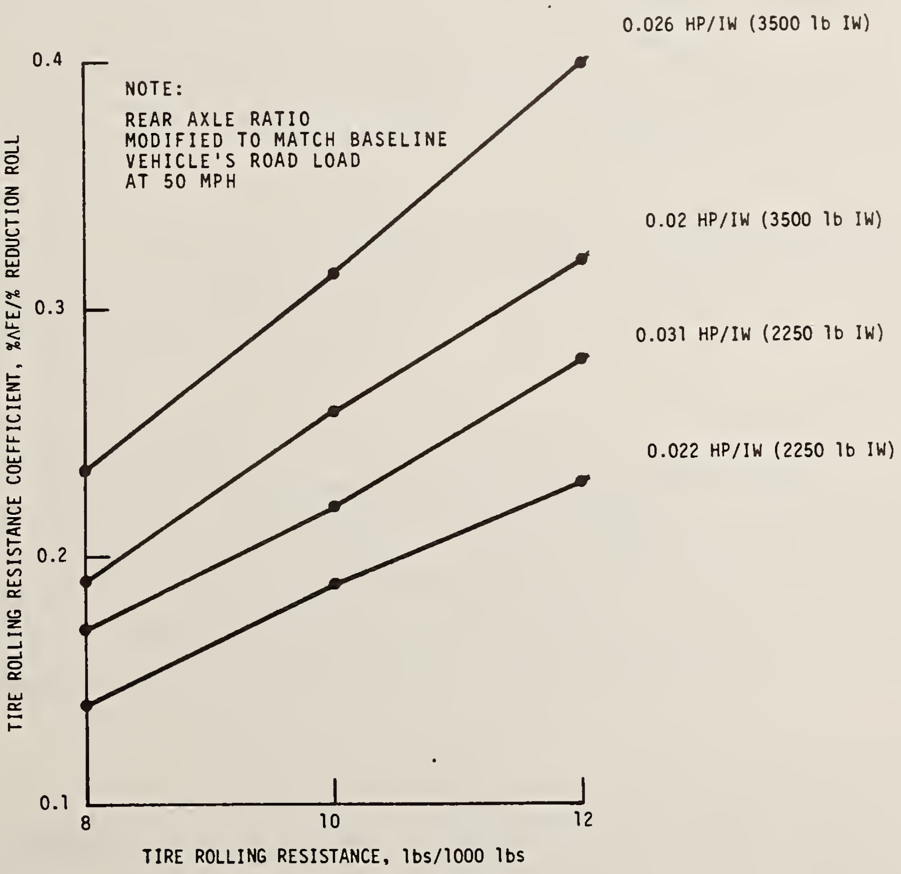
Inertia Weight (lbs.)	Engine Type	HP/IW	FUEL ECONOMY SENSITIVITIES		
			8***	10***	12***
2250	NA**	0.022	0.14	0.19	0.23
2250	TC**	0.031	0.17	0.22	0.28
3500	NA	0.020	0.19	0.26	0.32
3500	TC	0.026	0.24	0.32	0.4

*Information on chart corresponds with tabular data.

**Note: NA = Naturally aspirated

TC = Turbocharged

***Tire Rolling Resistance (lbs/1000 lbs)



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3.3 ENGINE FUELS

The bulk of the diesel fuel is utilized in trucks and buses, with a small percentage consumed by passenger cars and light duty trucks since the latter has not penetrated the U.S. market significantly. Diesel fuel is chiefly obtained by cracking the heavier fraction of crude oil. Currently the diesel engine manufacturers require fuel characteristics to be as consistent as possible. However, this requirement may change in the future because of:

a) Requirements to meet future regulated and non-regulated emissions standards, especially particulates, and to allow use of anti-pollution devices. This requirement is greater for passenger car and light duty truck diesels than for heavy duty trucks and buses because of the flexibility in the operations of the engine under their driving conditions.

b) requirement to decrease the gasoline/diesel fuel ratio to improve the present day efficiency of crude oil refining.

c) future requirement to consider alternate fuels such as alcohols, broad-cut derived from coal, vegetable oils or other resources.

The following discussion focuses on some critical factors and requirements under consideration for diesel fuels for passenger cars and light duty trucks.

3.3.1 Requirement/Quality

Currently, the ASTM standards D975[1] apply for type 1D and 2D diesel fuel. Table 3.3-1. Diesel fuel properties specified under the standard include: specific gravity, distillation characteristics, sulphur, aromatic and olefin content, flash point, viscosity, cetane number, pour point, water, sediment and ash content.

a) Viscosity influences the injection characteristics, pump wear and fuel system of an engine. Lower viscous fuels may provide a means to reduce particulates although there is not

TABLE 3.3-1. ASTM STANDARDS

	ASTM D 975	
Test	I-D Fuel	2-D Fuel
Appearance	Clear and Bright	Clear and Bright
Distillation I.B.P. (^{o}C)	166 - 199	171 - 210
10%	188 - 221	204 - 243
50%	210 - 249	243 - 282
90%	238 - 271	288 - 321
E P	260 - 293	304 - 349
Spec. Gravity (at 15.6 ^{o}C) gr/cm^3	.826 - .806	.860 - .840
Total Sulphur % wt	.05 - .2	.2 - .5
Aromatics % vol.	8 - 15	27 min.
Paraffins, Naphthenes and Olefins % vol.	Remainder	Remainder
Flash Point	48.9 C min.	54.4 C min.
Paraffins Viscosity (at 37.8 °C) CST	1.6 - 2.0	2.0 - 3.2
Cetane Number	48 - 54	42 - 50
Conradson number on 10% residue (% wt max)	.15	.35
Water content % vol. max	trace	.05
Sediment % wt max	trace	.05
Ash % wt max	.01	.01
Pour point max	5.55 C below	ambient
Copper corrosion max	3	3

sufficient evidence to support this. However, lowering viscosity contributes to wear thus impacting the operation of the injection system and engine itself.

b) Volatility is connected with the initial boiling point characteristics of the fuel and affects the quantity of fuel vaporized prior to the start of combustion. Thus, it influences the nitrogen oxide and hydrocarbon emissions produced by an engine as well as noise, odor, fuel consumption and cold start performance of an engine. Because the great influence volatility has on fuel evaporation, the type of diesel combustion system utilized plays an important role.

Higher volatility fuels have generally shown an ability to reduce smoke, but because of their lower specific gravity, they lower the power that is developed by the engine, which slightly decreases fuel consumption. Some limited studies to date have shown that an increase in volatility contributes to increased nitrogen oxides and hydrocarbons.

c) Cetane number is one of the most fundamental single fuel parameters affecting diesel emission, noise and startability because of its great influence on the ignition characteristics of the engine. Since the beginning of the 1960's the trend in cetane numbers has been downward (2 cetane numbers in ten years) and this trend is expected to continue because of the need for high octane unleaded gasoline which requires hydrocracking or cat-cracking technology.[2] One of the most important properties of a diesel fuel is its autoignition quality. The autoignition quality of the fuel is identified by its cetane number which is measured in the CFR (cooperative fuel research) engine. It is related to the compression ratio which would result in an ignition delay (pressure rise time) equal to 13° C.A., at an engine speed of 900 RPM, under specified inlet air and cooling water temperatures. This compression ratio is bracketed between that of two blends of primary reference fuels. The current primary reference fuels are n-cetane, $C_{16}H_{34}$, (C.N. = 100) and heptamethylnonane $C_9H_{13}(CH_3)_7$ (C.N. = 15). This means that the lower limit of the current cetane scale is C.N. = 15. Work[23] done to modify the scale and extent it to C.N. = 0 has been recently published. The ignition delay of the fuel is one of the most important parameters in the design of many diesel engines, particularly the direct injection type. It is shorter for higher cetane number fuels. The ignition delay decreases with the increase in the air pressure and temperature as illustrated in Figure 3.3-1 for a pressure of 600 psia, and Figure 3.3-2 for 400 psia.

Low cetane fuel generally leads to higher nitrogen oxide and hydrocarbon emission levels, higher noise levels and cold smoke. It is also speculated that a lower cetane number fuel may result in higher particulate emissions when corresponding changes are not introduced in the engine to allow it to accept a low cetane fuel. Naturally aspirated engines are more sensitive to cetane number variations than turbocharged engines, and indirect injection engines are less sensitive than direct injection engines. Recent data collected by Peugeot (Table 3.3-2) from tests on their 504 engine show that the hydrocarbon emission levels are decreased when the fuel cetane number was increased from 42 to 60. No data was supplied by Peugeot regarding the effect of cetane number on particulate emissions. It should be noted that most European engines are optimally designed for a higher cetane fuel (approximately 50). No significant effect on fuel economy was observed. Similar findings were reported by Volkswagen under contract to DOT/TSC.* For performance and fuel consumption purposes, passenger car and light truck engines may require a cetane number greater than 40 to avoid delayed combustion. The variance in cetane number should be maintained at a minimum, a range of no more than 6 cetane numbers for optimum engine design for best fuel economy and lowest emissions.

*DOT-TSC-1193.

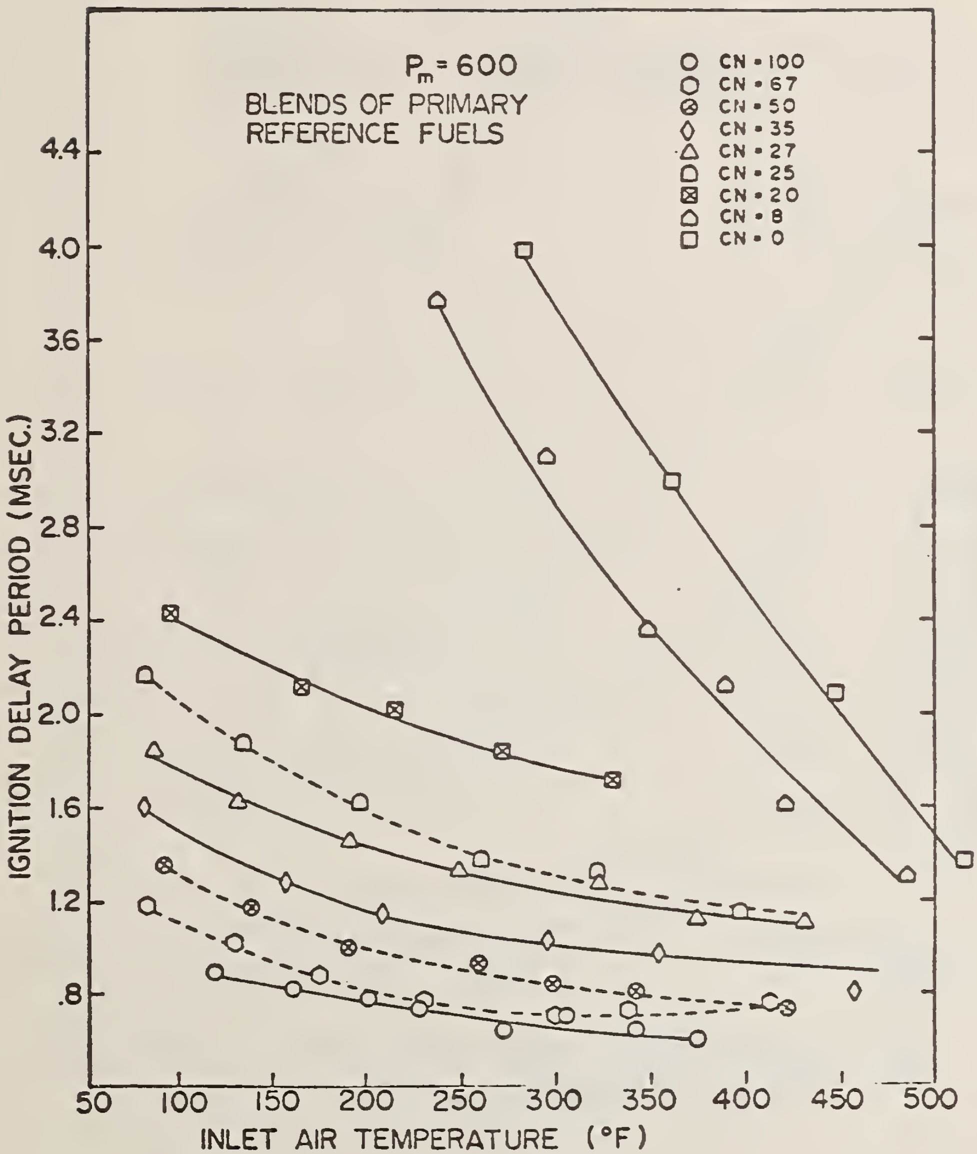

FIGURE 3.3-1. EFFECT OF INLET AIR TEMPERATURE ON IGNITION DELAY FOR BLENDS OF PRIMARY REFERENCE FUELS AT P_m = 600 psia

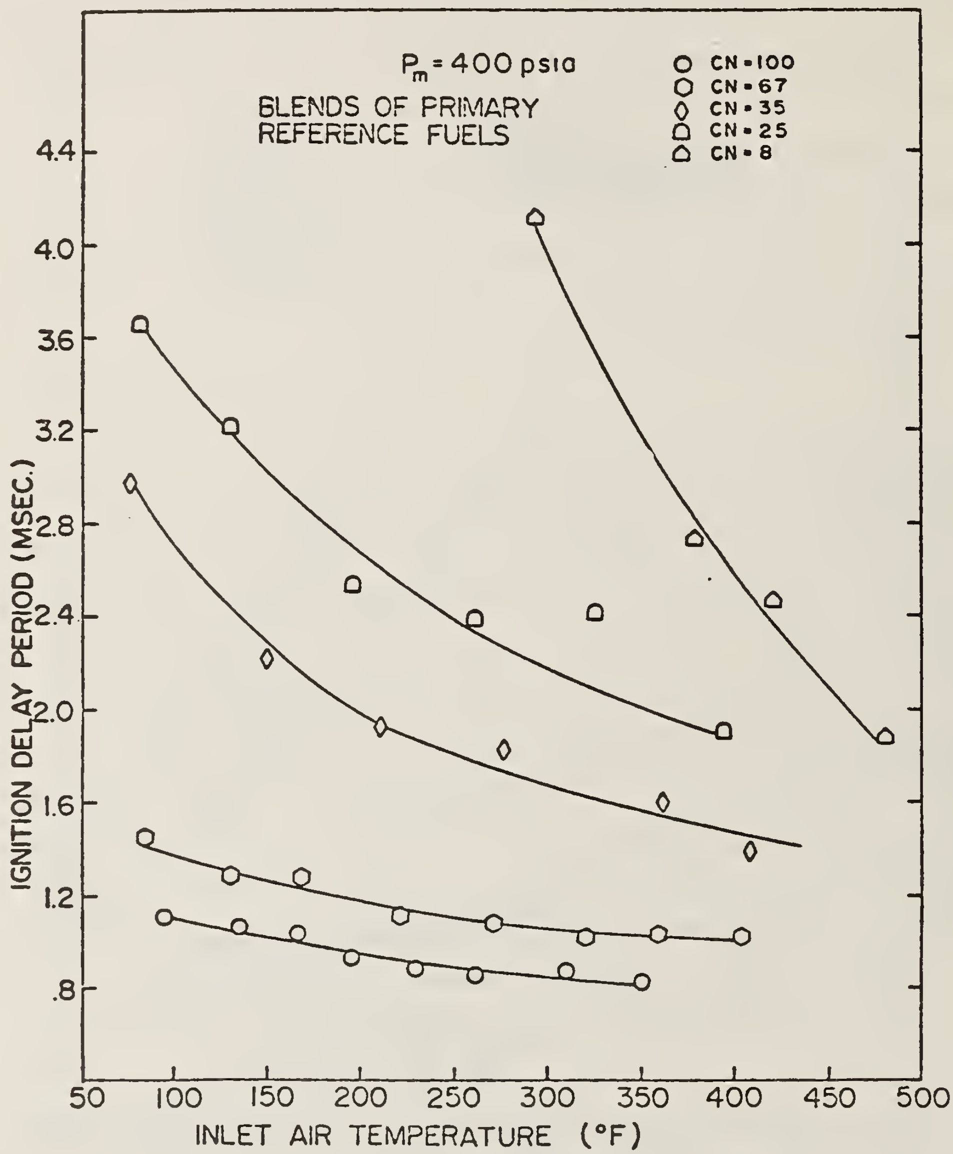

FIGURE 3.3-2. EFFECT OF INLET AIR TEMPERATURE ON IGNITION DELAY FOR BLENDS OF PRIMARY REFERENCE FUELS AT P_m = 400 psia

TABLE 3.3-2. EFFECT OF CETANE NUMBER ON EMISSIONS AND URBAN FUEL ECONOMY

504 DIESEL MODEL YEAR 7 - CERTIFICATION VEHICLE 4" 612

Laboratory		Amoco-Fuel (42 Cetane #) FTP Urban - Cycle (LA4) HC/CO/NOX/FE Grams/mile/mpg	Howell-Fuel (50 Cetane #) FTP Urban Cycle (LA4) HC/CO/NOX/FE Grams/mile/mpg
		0.35/1.34/1.09/32.33	0.245/1.024/1.12/ 30.03
Peugeot Paris	I		
	II	0.37/1.17/1.07/33.5	0.27/1.03/1.13/28.8
Daimler Benz	I	0.47/1.41/0.96/30.54	0.16/0.94/1.01/29.89
Stuttgart	II	0.46/1.3/0.99/30.74	0.25/0.98/9.99/30.4
Volkswagen	I	0.36/1.54/1.0/27.96	0.22/1.15/1.11/27.44
Wolfsburg	II	0.49/1.68/1.05/27.54	0.24/1.31/1.06/28.73
Ricardo	I	0.41/1.29/1.09/28.3	0.26/0.98/1.07/28.48
Shoreham	II	0.45/1.47/1.03/28.75	0.28/1.05/1.11/29.0
Peugeot Belchamp	I	0.45/2.2/0.98/29.02	0.27/1.76/1.00/28.56
EPA	I	0.540/1.50/1.18/28.3	0.380/1.20/1.14/29.5
		0.680/1.70/1.14/28.8	0.370/1.20/1.15/29.7

d) Reducing the Sulfur Content - Reducing the sulfur content in the refinery will reduce the sulfur oxide emissions and the related sulfate particulate emissions. Typical 2-D diesel fuel contains about eight times the sulfur levels of gasoline.

e) Effect of Fuel Composition on Particulate Emissions - Tests made by Hare[3] on a four-stroke Heavy Duty Truck Cummins NTC-290 and a 2-stroke Detroit Diesel engine indicated that diesel #1 (IBP = 177°C, 10% point = 199°C, 90% point = 263°C) produced less particulate than diesel #2 (IBP = 189°C, 10% point = 219°C, and 90% point = 302°C). The $\frac{C}{H}$ mass ratio in the particulates was 6.8 for the Cummins as compared to 15 for Detroit Diesel engine. This indicates that the particulates are carbonaceous soot-like material for the NTC-290 engine and hydrocarbon-like material for the 2-stroke Detroit Diesel engine. Braddock and Gabele[4] tested a Peugeot 504D, on Federal Test Procedure, highway fuel economy test (HWFET) and Sulfate Emission Test (SET) cycle using different fuels. Jet A fuel (IBP = 324°F, 10% point = 358°F, 90% point = 460°F) produced the least particulate emissions. The other fuels tested such as diesel #1 and diesel #2 were less volatile. However, diesel #2 emitted less particulates than #1, which does not agree with results of the heavy truck tests.[5] This lack of agreement might be caused by the difference in the aromatic contents of the fuel or the difference in the combustion process. In one case[6], the combustion chamber was of the D.I. type while in the other it was of the I.D.I. Swirl type. This lack of agreement indicates the need for a research program to study the mechanism of particulate formation in diesel engines. The mutagenicity[7] of particulate emissions is influenced by fuel. Both the Benzo (a) pyrene content and the mutagenic activity of emissions were the highest when the minimum quality fuel was used. This fuel has the lowest cetane value, highest aromatic content and highest nitrogen content of the five fuels examined. An increase of aromatic content can increase smoke and odors[8]. Recent tests carried out by G.M.[9] demonstrated that smoke and oxides of nitrogen increase when kerosene was used instead of number 2 diesel fuel. However, particulate emissions were much lower (about 50%) on kerosene fuel.

The differences are believed to be due to the combined effects of increased volatility and lower aromatics. Fuel hydrocarbon composition influences other unregulated emissions like aldehydes.

3.3.2 Additives

Additives are used to modify physical and/or chemical characteristics of fuels. A list of the main types of additives includes: Anti-smoke: Organic salts, usually barium. These additives help to reduce smoke emission but tend to increase in total particulates since barium and carbon particulates are obtained instead of carbon particles alone. Barium can combine with fuel sulfur to form $BaSO_4$ which does not represent a health hazard or to form $BaCO_3$ which is toxic.(10) Ignition Improvers: Organic compounds, such as peroxides, nitrates or others have been examined. Particularly iso-propyl nitrate, iso-amyl nitrate and cyclohexyl nitrate have been studied. Unfortunately, these additives contain nitrogen that willcontribute to the formation of nitrogen oxides.

The effectiveness of ignition improvers varies greatly with fuel type and origin. It may increase the black smoking propensity of fuels to which they are added.

3.3.3 Alternative Fuels

Various alternative fuels which includes distillates such as broad-cut and variable-composition fuels based on petroleum, coal, oil shale, biomass, etc., are under consideration today. Alcohols are also a major subject of interest. These include methanol, ethanol, higher alcohols, blends of these with hydrocarbons, etc. Vegetable oils, for example, peanut or soybean oil, can be very interesting for diesel application.

Broad-cut fuels

Several hypotheses have been put forth (11, 12) of broad-cut fuels obtainable by simple crude distillation to maximize refining efficiency. One idea[(13)] is to increase the final boiling point of middle distillates from 360° to 390°C, resulting in approximately 10 percent more middle distillates and correspondingly less residual fuel oil. An attempt was also made to decrease the initial boiling point. Diesel fuels having an Initial Boiling Point of about 50°C were examined (14), and more recently broad-cut distillates with an Initial Boiling Point of 40°C and a Final Boiling Point of up to 500°C have been under study. The Cetane Number of such broad-cut fuels ranges from 40 to 50. These fuels also exhibit good characteristics except for the cold flow properties, where the cloud point is about 7°C and the pour point is +6°C.

For the fuel having a distillation range from 50°C to 400°C (Cetane Number = 49), some light duty diesel engine powered cars (Opel Rekord 1998, Mercedes 220) and an 8140 Fiat engine were examined. Tests to date show no substantial differences in performances and smoke as the broad-cut fuel was used instead of a commercial diesel fuel having CN = 56.5. A slight decrease in fuel consumption was found when 50/400°C fuel was employed.

Coal derived Fuels

Fuels derived from coal or from oil shale, or fuels from synthetic crude[15,16,17,18] are considered today as possible replacements for petroleum-based gasoline and diesel fuel. Oil shale and coal are being considered for the production of (synthetic) crude. Lignites, sub-bituminous coals, can also be used. Synthetic diesel fuel derived from coal was commercialized over 40 years ago in Germany[19]. More recently, the South African Company SASOL produced synthetic diesel fuel from the Fischer Tropsch synthesis. Main features of the SASOL fuel are:

- Unusually high Cetane Number - around 70/75.
- Virtually sulphur free.
- Poor stability which led to formation of lacquer deposits in injectors-stuck needles etc.

Performance of this product was not entirely satisfactory in vehicles because of the high olefinic content which leads to problems with injector fouling and needle lacquering in some engines. On the positive side though, the feedstock value for chemical production from this material is extremely high due to the olefin content.

SASOL will produce diesel fuel in the future from coal in a new plant coming onstream some time in 1982. This plant will employ hydrogenation to decrease unsaturation and the diesel fuel produced is expected to have cetane numbers in the range of 50 to 55 and will be sulphur free.

Alcohols

Methanol is the most promising alcohol from a global point of view because of the availability of coal or oil shales in many countries.

Ethanol is as strong a candidate as alternative fuel, in largely agricultural countries like India and Brazil, for example.

Since alcohols exhibit very low ignition quality, limited studies have been conducted on heavy duty truck diesel engines. However, investigations[20] carried out concentrated on[21] additives to increase the Cetane rating of alcohols and forced ignition (dual fuel engine). In the case of methanol[22] a Swedish author, E. Holmes, of AB Volvo, concludes that a Cetane rating of 35 has been reached by using a 20 percent additive (Cetanox) which is sufficient to run the tested engine with 15 compression ratio at an ambient temperature of 25°C. Direct injection of methanol into the combustion chamber was also examined after injecting diesel fuel to avoid the methanol cooling off the charge. Good performance and low smoke emissions, HC and noise were attained. The design calls for a complete and separate diesel fuel system with tank, feed pump, injection pump and separate injectors.

Vegetable Oils

Peanut and soybean oils appear to be more interesting than alcohol for use in diesel engines. Cold properties such as the pour point (-1° to -5°C for peanut oil and less than -5°C for soybean oil), 8650 K Cal/Kg for peanut and 8420 K Cal/Kg for soybean oil as well as high cetane number (estimated at over 50 in engine tests) make them interesting.

Fiat Research Center sponsored by the Industrial Vehicles Corporation (IVECO) has tested vegetable oils in a 4-cylinder in-line 3.5 liter displacement direct injection diesel engine. Limited studies[23] have shown that when peanut or soybean oil was mixed with commercial diesel fuel in the proportion of 30/70 percent by volume to give equal power, a slight increase in thermal efficiency was observed and NO_X emissions were decreased by 20 percent. Furthermore, marked reduction of combustion noise and less objectionable odor were observed as compared with diesel fuel.

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3.4 ENGINE LUBRICANTS

Modern engine oils consist of the base stock, either mineral or synthetic, and an additive package. This additive package accounts for approximately one-fifth of the contents, by volume, of the oil and one half its cost. The additives are viscosity index improvers, friction modifiers, dispersants, detergents, and oxidation corrosion inhibitors. In the past, oils were primarily formulated to meet the temperature, wear, and oxidation problems encountered in engine operation. However, because of recent fuel economy standards, attention is being focused on the development of "fuel efficient" engine lubricants. Recent improvements for fuel efficiency in engine oils have been in the following areas:

a. Upgrading of conventional mineral oils through improved refining and development of a wide range of additives.
b. Development of synthetic base stocks or blends of synthetics and mineral oils.
c. Utilization of lubricating solids as a colloidal suspension and in conventional oil base stocks.

In all cases, lubricant fuel efficiency comes about through improved viscosity characteristics or friction modifications. Viscosity improvement can be obtained with viscosity enhancing additives or by the use of synthetic base stocks. The synthetics exhibit reduced low temperature viscosities without adverse effects on other lubricant properties. These improved viscosity characteristics at low temperatures are especially important to fuel economy during cold starts. Anti-friction modifiers can be either soluble additives or colloidal suspensions. The soluble additive is usually a fatty end group whereas the colloids may be graphite, teflon, or molybdenum disulfide.

Presently there appear no readily accessible data to quantify improvements in diesel fuel economy over the FTP cycle with fuel efficient lubricants.

Specification concerning lubricant performance and viscosity as well as oil and filter change intervals are all influenced by the combustion process, oil change and thermal piston design. In addition, the design of the combustion chamber is important because the soot emanating from the combustion of diesel fuel can be absorbed by the oil film on the combustion chamber wall, which may be transported to the oil pump, thus contributing to engine wear. This type of oil contamination is more of a problem with the IDI (indirect injection) engines used in passenger car and light duty truck diesels. Thus, the oil used in these engines requires a higher detergent capacity.

3.5 ENGINE CONTROL STRATEGY AND IMPLEMENTATION

The control strategies selected by an automobile manufacturer and the manner in which it is implemented have an effect on both fuel economy and emissions. To this date, very few controls have been introduced in the passenger car and light duty truck diesel because of the diesel's ability to meet the current required emission goals of 1.5/15/2.0 gms/mile of HC/CO/NO_x respectively. By and large, injection time retard and fuel enrichment under starting conditions are universally employed to reduce NO_x, HC and noise, and to minimize the warm-up time of the engine under all dynamic engine conditions concurrent with engine and fuel injection improvements.

The further lowering of the emission standard to 0.41/3.4/1.0 gms/mile of HC/CO/NO_x in addition to the proposed particulate standard, and other consumer nuisances such as irritants, and starting may require future control technologies.

Under a DOT/TSC contract, Ricardo Consulting Engineers* is investigating means of achieving fuel economy improvement from a diesel powered light duty vehicle. Under this contract, re-

*DOT-TSC-1545.

searchers will examine the effect of numerous engine parameters which potentially offer the best control of regulated and unregulated emissions and good starting characteristics which have a minimum impact on fuel economy. Parameteric tests will be conducted on a gasoline modified swirl chamber Chrysler 225 slant six diesel engine and its single cylinder counterpart. Various control strategies will be examined under this contract. Engine parameters to be examined will include:

a) inlet temperature
b) exhaust back pressure
c) injection timing
c) exhaust gas recirculation
e) fuel sprays (type and direction)
f) cetane number and aromatic content of fuels
g) injection rates
h) wall temperature (insulation)
i) catalyst material in chamber
j) valve timing and overlap
k) water/fuel emulsions
l) compression ratio
m) swirl chamber to main chamber volume ratio
n) swirl chamber throat size
o) coolant temperature
p) glow and plasma plugs
q) turbocharging and advanced fuel injection equipment.

3.5.1 Control Strategies

The task of calibrating a given engine/powertrain configuration is simply an iterative procedure of testing, adjusting the initial settings and then retesting. As the number of control variables grows, and the sophistication and complexity of emission control systems increase, the task becomes overwhelming. The engine/powertrain/aftertreatment mechanization and all of the other emission subsystems have to be treated as a total system and a methodical approach is needed in order to define an optimum control strategy for a given emission constraint that would yield minimum fuel consumption.

Various investigators [1,2,3] are addressing the system-wide control problem by utilizing the following procedure:

1. Define a given engine/vehicle configuration (horsepower, inertia weight, transmission and rear axle ratio). Via a vehicle simulation, determine the speed-load trajectory required to accomplish the FTP schedule. The total time at each speed-load point is also an output.

2. Based on the time profiles, determine a small set (approximately 10) of engine speed-load points which approximates the FTP schedule.

3. Given this set of points, run an exhaustive parametric, steady-state engine test collecting data on emissions, fuel consumption, and all engine state variables while varying all engine control variables.

4. Analyze this data for the optimum control strategy as a function of engine state variables.

3.5.2 Implementation

3.5.2.1 Electronic Control Systems - Use of on-board electronics for engine control and for the implementation of engine calibrations is likely in the 1980's. Future multivariable requirements will necessitate digital microprocessor control techniques. The advantages over analog circuitry lie in the microprocessor's ability for accuracy and its repeatability for the complex or shared logic applications. In addition, it has the ability to follow sophisticated optimal control trajectories with excellent dynamic response and to accommodate rapid changes in engine calibrations late in the certification cycle. The basic parts of an engine control system are:

1. Sensors for measuring "key" engine state variables,
2. Actuators for initiating control over control variables,
3. The electronic control system hardware (microcomputer), and
4. the control strategy and calibration program (software).

The flexibility of this type of electronic control system permits easy comparison of several engine calibrations, which allows for quick fuel economy and emission trade-offs to be made and gives deeper insight into the interaction of each control parameter.

Figure 3.5-1 presents a total integrated engine/powertrain control system that can be anticipated for the 1980-1990 time frame. Table 3.5-1 outlines in detail some of the mechanical hardware, sensors and the software/microprocessor implementation strategy that will be required to produce the engine/powertrain control system in the 1980's.

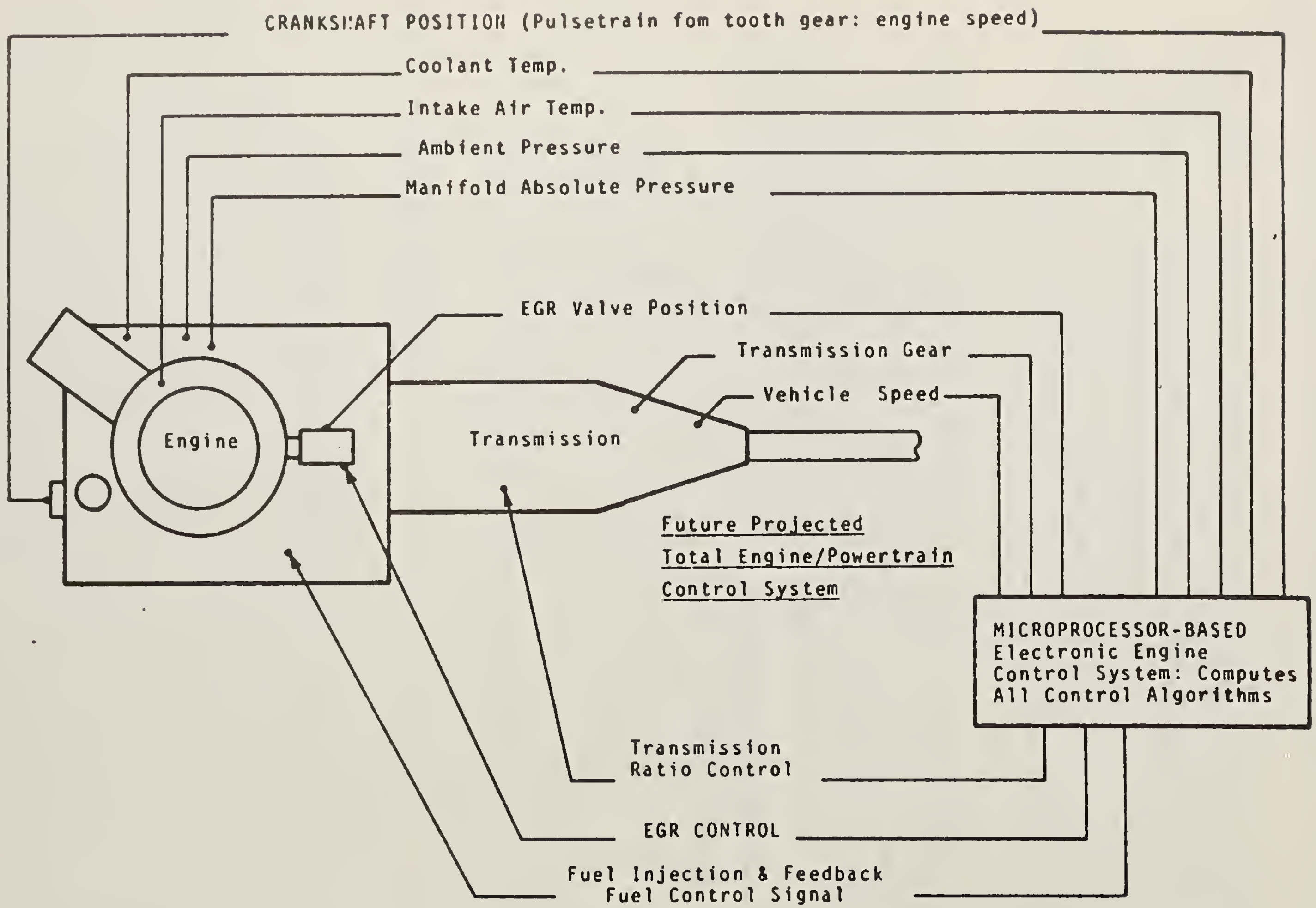


FIGURE 3.5-1. BASIC PARTS OF ENGINE CONTROL SYSTEM

TABLE 3.5-1. ADVANCED TECHNOLOGY REQUIRED FOR ELECTRONIC CONTROL OF SIGNIFICANT POWERTRAIN VARIABLES (1980-1990 TIME FRAME)

POWERTRAIN CONTROL VARIABLE	MECHANIZATION HARDWARE	SENSORS	STRATEGY OR CONTROL CONCEPT IMPLEMENTATION
Injection Timing	Microprocessor-based Electronic injection Control Transducer Systems	Sensor requirements will include manifold absolute pressure or engine brake torque, engine speed, inlet air temperature, coolant temperature, Barometric pressure and Piston Position.	Initial Software program contains control strategy as defined by dynamometer data and off-line computer generated opitmized engine calibration for fuel economy and emissions. Requires extremely flexible hardware.
EGR MASS FLOW	Single point entry sonic EGR valves will replace present systems. Multipoint entry variable-valve timing concept may be developed to improve EGR distribution to the engine cylinders.	Accurate control of EGR mass rate will require EGR, throttle position as well as	Simple control of EGR versus manifold vacuum will be replaced by microprocessor based optimized software strategy developed as defined under injection timing
FEEDFORWARD FUEL MASS FLOW MANAGEMENT	Electronic Fuel injection systems for direct and indirect injection engines.	Air mass flow meter-based EFI systems.	Speed-density control with EGR flow correction will be computed by microprocessor-based EFI control module. Initial system is calibrated to empricially derive engine volumetric efficiency expression
FEEDBACK CONTROL CLOSED EGR MASS FLOW LOOP	Used on all engines requiring NO_x abatement via EGR dilution.	For closed-loop EGR valve control a EGR pintle position sensor is used.	Closed-loop EGR, control will correct for initial manufacturing variance and for system degradation with mileage accumulation. The closed-loop control laws and logic will be part of the software program stored in the microprocessor system, which is performing the open-loop strategy computations.
SMOKE PARTICLES	Soot particle charge pick up on an insulated cone position in the exhaust pipe	Electronic soot sensor	Control of fueling rate to the engine under acceleration to enable better utilization of the engine power.
LOCK-UP TORQUE CONVERTERS AND CONTINUOUSLY VARIABLE TRANSMISSION RATIOS	Present Transmission designs will be replaced by o 4 forward speeds including overdrive with lock-up torque convertor o Continuously variable Transmission ratio design (CVT)	For both designs the control logic will require engine speed, vehicle speed, load, transmission gear and throttle position inputs.	In both designs the control logic will be computed by a microprocessor-based electronic module with the CVT control software being more sophisticated.

3.5.3 Diesel Exhaust Aftertreatment Devices

During the 1980-90 time interval, some automobile and light duty truck diesel engines may require clean-up devices to reduce particulates from the exhaust gas. Existing techniques for exhaust clean-up include oxidation catalysts, agglomerators and separators. All these technologies individually require further development and subsequently integration with the engine to establish the most effective application for removal of diesel particulates from the exhaust stream.

The work of the Bureau of Mines, Bartlesville (Oklahoma) Energy Research Center of DOE(4) showed that the platinum-based catalyst was more efficient than the nickel-based catalyst in reducing HC and CO, the pellet platinum catalyst was more effective than the monolyth platinum catalyst in reducing CO, HC and odor intensity. The catalyst efficiency was poor (less than 50%) at exhaust temperatures lower than 400°F. The increase in SO_3 and H_2SO_4 emissions with the use of the catalyst represents a serious problem in the catalytic treatment of the exhaust. This is caused by the much higher concentration of the fuel sulfur in the diesel fuel in comparison with the gasoline fuel.

The agglomeration and separation method was applied by Southwest Research Inst. to a Mercedes Benz 300 D(5). The engine operation modes consisted of 42 cold starts, 335 hot starts and 9654 km (6000) miles. The particulate removal efficiency was about 68 percent when the system was new. It decreased with time, and resulted in an increase in the engine back pressure. The relatively low time average exhaust temperature during the engine operation (374°F) could not start the combustion of the collected particulates. The increase in the exhaust temperature at near full loads would improve the possibility for burning the accumulated particulates.

Ricardo Consulting Engineers(6,7) have been very active in studying exhaust control devices, ranging from initial through-flow filters and a scrubber filter-like device. Originally Ricardo experimented with straightforward through-flow filter.

These filters proved to be effective in reducing particulates but displayed very high back pressure. To overcome the high back pressures, Ricardo attempted a scrubber. See Figure 3.5-1. In this design, the exhaust is allowed to flow around a spirial where zirconia wool is enclosed by a wire mesh. Ricardo found with this design that during light load engine operating conditions, black soot particulates solidified and were not burned until the higher engine loads were reached. The after treatment device reduced the particulates by 20 percent with negligible amount of particulate material retained in the device. Other design improvements were incorporated but were found difficult to optimize for back pressure. Because of this problem, Ricardo experimented with a new filter scrubber type device made of a series concentric cylinders of expanded perforated metal, with gaps in between. An aluminum filter filler was substituted for the zirconia wool. The gap width was optimized to give the correct amount of gas flowing through the gaps and filter for minimum back pressure.

Tests conducted by Ricardo showed that there was a 40 percent reduction in soot level. However, the soot collected during light load engine conditions tends to light off unpredictably and melt the collection matrix. This problem has not been overcome to this date and it seems due to the hydrocarbons adsorbed on the soot. To eliminate the hydrocarbons before they reach the soot filter, Ricardo has recently suggested combining the Filter scrubber with an oxidation catalyst consisting of platinized Fecroloy matrix liners positioned in the exhaust port. Preliminary results to date indicate that this design can reduce the hydrocarbons by 60 percent[5].

3.5.4 Fuel Injection Systems

The fuel injection system (fuel pump and injectors) for high speed diesel engines is important in the design operations and performance of the engine. The fuel injection equipment must break up the fuel into small droplets or "atomize" the fuel by the injection process and spray the fuel within the combustion chamber in a penetration pattern which is dependent on the size of the injector holes and their orientation. All these requirements placed on the injection equipment must be consistent with performance and emission objectives.

Presently there are four basic fuel injection systems, and modifications thereof, in use on high-speed diesels which are basically of the hydraulic/mechanical type. They include (a) pressure-time fuel systems, (b) multiple pump system, (c) unit injector system, (d) distributor system. The latter system is commonly used on high-speed indirect injection, swirl chamber engines.

The pressure-time fuel system, (a) above, uses injectors that meter and inject the fuel into the chamber. Metering is based on a pressure-time principle. The pressure at the injector is supplied by a low pressure fuel feed pump. Fuel metering is determined by how long the injector metering orifice remains open, an interval determined by the engine rotational speed via the camshaft-driven injector plunger which forces fuel into the cylinder. This system permits

(a) A simple injector to accomplish all metering and injection functions,

(b) Fuel to be injected and ignited at approximately the same piston position at all engine rotational speeds,

(c) Fuel to be injected as an extremely fine spray, permitting use of wide range of fuel,

(d) A low pressure, common rail injection system to eliminate the complication of high pressure lines running from fuel pump to each engine cylinder.

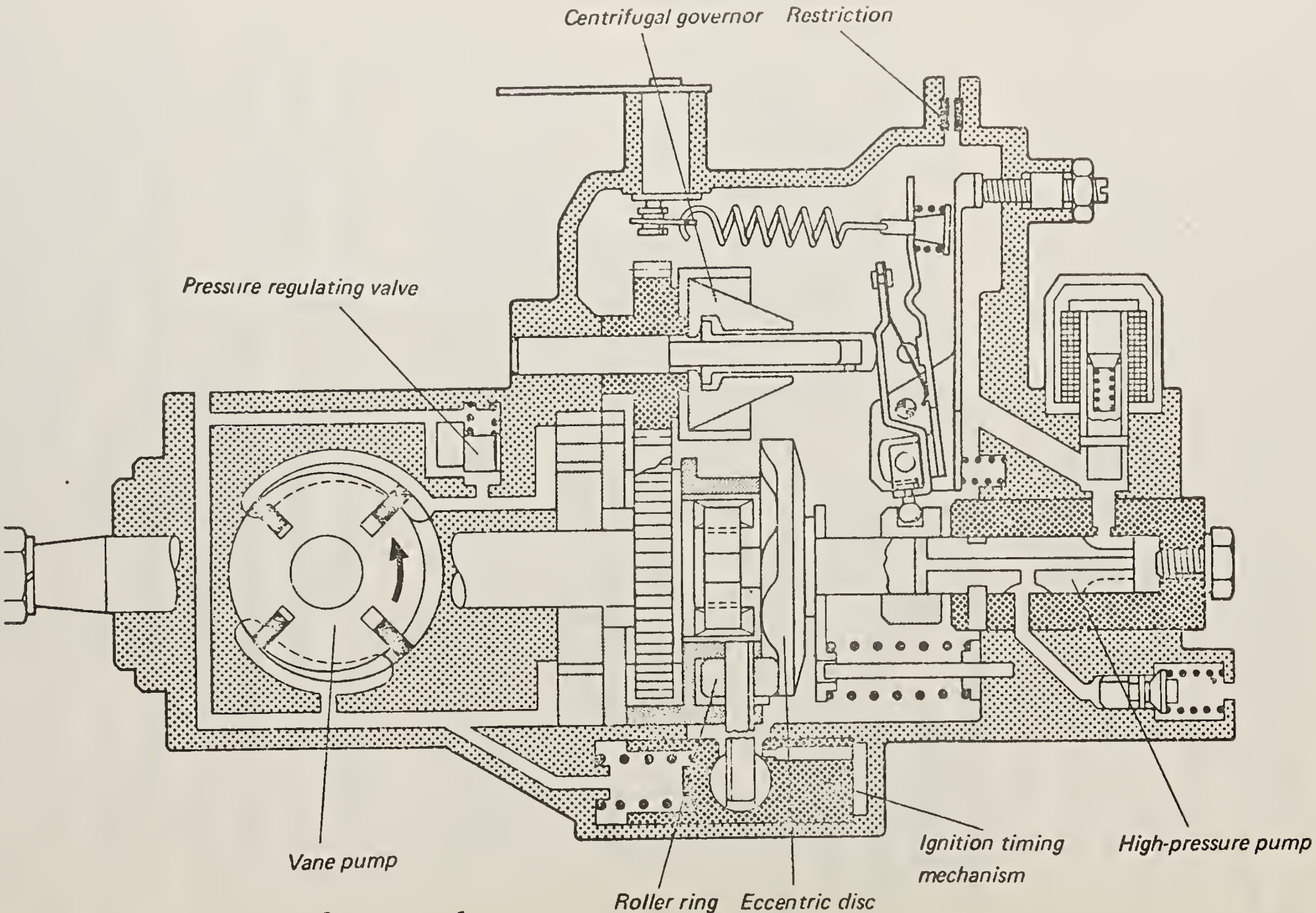
The multiple pump system, (b) above, uses individual plungers for metering and injecting fuel to each cylinder. These pumps are remotely mounted and are connected to the injectors which are located in the cylinder head by long high-pressure fuel lines. The advantage of the multipump system lies in its compactness, since all functions are combined in a separate pump for each cylinder. However, the cost is high because each pump in a multicylinder unit must precisely match its companions at all operating speeds and loads. Timing governors usually are supplied on some engines to reduce variation in injection timing.

The unit injector system, (c) above, locates both the metering and injection functions in the injector. The plunger pumps are similar to those of the multiple pump system but are positioned in the individual injectors, thus eliminating the use of high pressure fuel lines. Rotation of the plunger by the fuel rack controls metering.

The distributor system, (d) above, uses a single-plunger high pressure metering pump to meter the fuel for all cylinders. See Figure 3.5-2. The governor and throttle control the pump stroke length and thus the amount of fuel delivered on each stroke. Distribution to the cylinders is controlled by a rotating disc or shaft with holes located to synchronize the metering pump delivery stroke with the various cylinders. High pressure wall lines supply fuel to the injectors.

The fixed relationship and dependence of injector actuation and timing on the mechanical design of camshaft lobes and gear drive mechanisms require the use of complex control mechanisms. These adjust the supply of fuel for variable load accommodation at fixed engine speed, for acceleration of coupled masses to operating speeds, for control of overrun or overshoot, and for fuel-air enrichment needed under starting conditions. Figure 3.5-3. Adjustments for variation in fuel density or for operation at altitudes are usually provided also.

The foregoing mechanical design arrangements and control mechanisms in use today represent a compromise between the ideal of instant and precise response to load demand variation for each cylinder under all operating conditions, and the practical considerations of injection equipment cost, serviceability, and maintainability of precision equipment. Wear of mechanical fuel injection equipment, including cams, roller followers, plungers



Source: Reference 1

FIGURE 3.5-2. CROSS SECTION DISTRIBUTOR PUMP

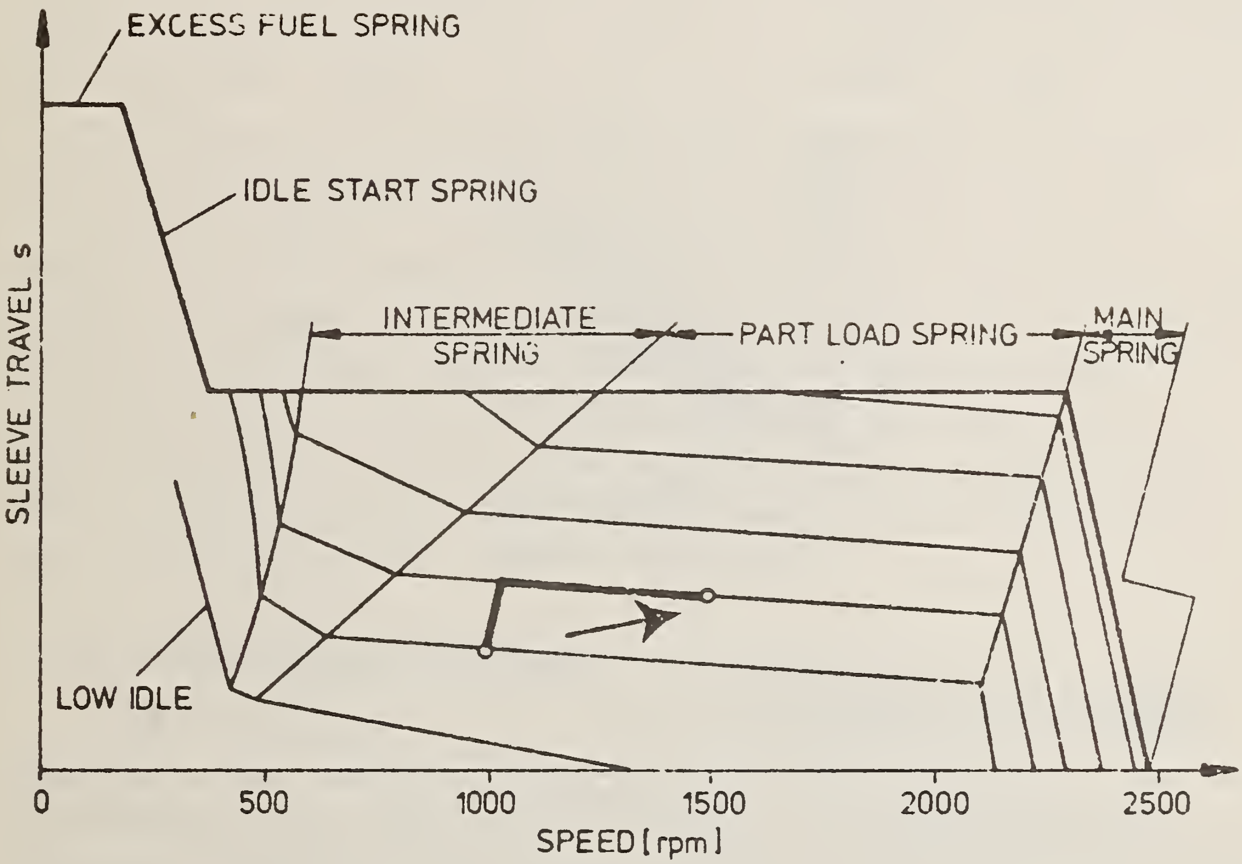


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FIGURE 3.5-3. OPERATION OF THE CAR APPLICATION SPEED GOVERNOR and barrels, governor linkages, and control rods and bearings, all contribute to the need for regular repair and maintenance to avoid engine damage, performance loss, and excessive soot emissions.

Requirements on the injection equipment of the diesel engine are mainly dependent on combustion system type. This is illustrated in Figure 3.5-4 where typical fuel injection pump injection rates are shown as a function of peak injection pressure for various diesel engines in the power range between 50 and 500 horsepower.[8]

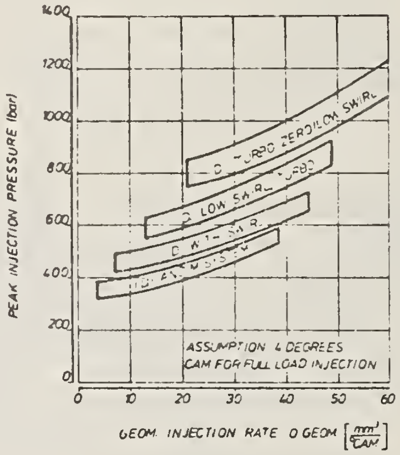

FIGURE 3.5-4. RELATIONSHIP OF COMBUSTION SYSTEM, INJECTION PRESSURE AND INJECTION RATE

The requirements for fuel injection equipment vary when peak pressures are concerned. The IDI-engines and M-type combustion chambers have the lowest demands during peak pressures. The DI-engines with swirl are next since they have a "soft combustion" and so the pressure gradients in the combustion chamber are relatively small after ignition. Above these, appear the DI-engines with low swirl. These are often turbocharged. The transitions between the various bands are certainly continuous. The engine designers[8] in the United States prefer relatively high injection pressures in the combustion system. Without considering exceptions the Europeans however favor DI-combustion systems with more swirl and lower peak pressures in the injection system. Nevertheless all diesel engine manufacturers[8] are moving towards higher injection rates in correlation with higher injection pressures.

A very important component of the injection equipment is the injection valve, the so-called nozzle. The injection valve should inject fuel within a short time, should not open too fast, should properly atomize, distribute and close very rapidly without secondary injection. Direct injection engines employ a number of holes in their fuel injector when a wide speed range has to be covered, with the fuel line pressure varying as the square of the fuel velocity and hence of the engine speed; pressures will either be excessive at high speed or too low at low speeds, depending on the orifice size chosen. As a result these is considerable difficulty in getting good engine performance over a wide speed range.[9] Single hole nozzles with a lifting obturator are employed in indirect injection engines. This provides an opportunity to have a smaller orifice area at lower speeds, thus enabling good combustion to be achieved over a wide speed range.

3.5.4.1 UFIS Injection System

The UFIS (universal fuel injection system) fuel injection system[10] uses unit injectors which are electronically controlled and operate on the common rail principle. The development of this system originated at American Bosch in 1968 in cooperation with the TARADCOM to develop a fuel injection system which would duplicate the injection characteristics required of any of the known Diesel combustion configurations such as open chamber, prechamber, MAN, and be adaptable to advanced combustion studies. It features positive displacement fuel metering, hydraulic pressure amplification and nozzle valve control. See Figure 3.5-5. The system is highly flexible and can achieve injection characteristics beyond the capability of conventional systems to provide the injection quantities, characteristics, and timing variations needed for control of gaseous emissions, noise and fuel economy. Figure 3.5-6

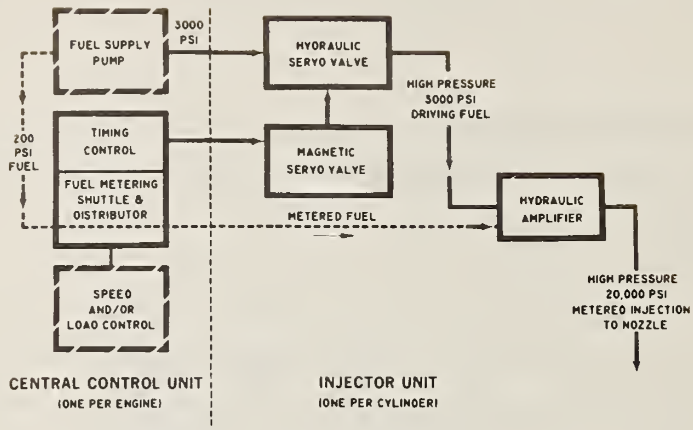

FIGURE 3.5-5. BLOCK DIAGRAM OF UFIS SYSTEM

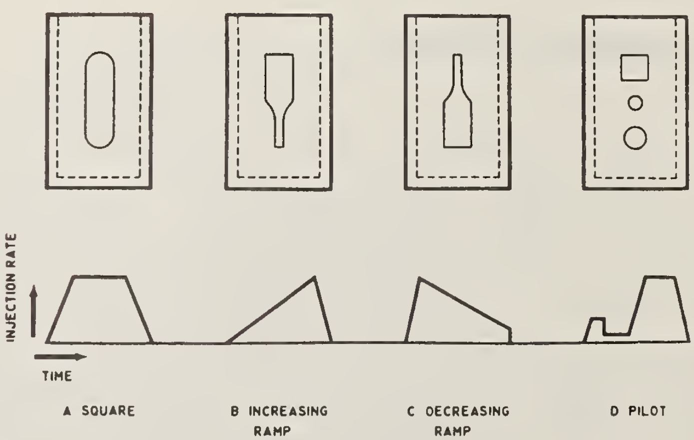

FIGURE 3.5-6. INJECTION RATES

illustrates some various injection rate profiles possible with this system.

Presently a self-contained engine driven system is being developed with the following features:

- -- 20,000 psi injection pressure capability - idle to rated speed.
- -- Fast cut off from the maximum rate of injection.
- -- Control of injection pressure as a function of engine speed.
- -- Control of injection rate as a function of fuel quantity.
- -- Control of:
  - Precise injection timing vs. engine speed.
  - Injection timing vs. load independent of engine speed.
  - Injection timing vs. temperature.
  - Injection timing vs. manifold pressure.
  - Injection quantity vs. manifold pressure.

3.5.4.2 Advanced Fuel Injection System

New approaches are currently being explored[8] to develop injection systems which will provide greater flexibility of fuel injection to improve fuel economy, emissions, smoke limited power, low speed torque and fuel tolerance of diesel engines which may lead to unconventional injection systems. Prototype high pressure injection systems are under development at Robert Bosch which offer, independent from each other:

- shaping of the injection rate
- variation of peak pressure and timing as a function of speed and load
- automatic excess fuel, plus/minus-torque control and consideration of additional signals such as boost pressure etc.

These sophisticated systems include unit injectors, electronical governing and hydraulic circuits with low and high pressures. Further studies must show before implementing on vehicles:

- what is technically feasible
- which advantages are achievable
- what are the costs
- which other consequences are connected with these measures.

Recently Robert Bosch has developed a central, computer controlled governor for diesel passenger cars. It required electrical activators as well as transmitters within injection pump. This pump is not in production as of yet but is being experimentally tested.

3.5.5 Exhaust Gas Recirculation (EGR) Systems

EGR is a technique which dilutes the incoming air charge in the inlet manifold of the engine with exhaust gas expelled from the previous cycle. Hot or cold exhaust gas can be introduced into the inlet manifold. An external ducting arrangement is employed to transfer the exhaust gas to the inlet manifold. There is usually a single point of entry in the inlet manifold. Where it is homogeneously mixed with air, EGR reduces the combustion temperature and therefore the NOx formation rates. Before the external EGR valve was introduced, charge dilution was accomplished via modifications of intake/exhaust valve timing. Valve overlap (inlet and exhaust valve opening times overlap) was an effective way of introducing EGR dilution. Unfortunately, the amount of valve overlap permitted in a diesel engine is restricted because of its higher compression ratio. Excessive valve overlap degrades overall volumetric (breathing) efficiency low-speed torque and idle characteristics. These reasons, along with the fact that NOx reduction is only required when the engine is under load, indicate that a system which can modulate charge dilution selectively better serves the requirements of NOx abatement. The external EGR valve is such a system.

EGR is a new technology for diesel powered light duty vehicles because presently the diesels do not require EGR to meet current emission standards of 1.5/15/2.0; HC/CO/NOx gms/mile since it is done through the simpler means of late injection of fuel into the combustion chamber. The 1981 emission standards will possibly require EGR. Being a new technology, its mechanization differs from gasoline practices. There have been only a limited number of systems developed.

Figure 3.5-7 shows a diagrammatic representation of a hydro-mechanically operated exhaust gas recirculation system for diesel powered vehicles.[11] In this system EGR is modulated roughly in proportion to engine load and consists of two principal components:

a) a throttle valve positioned at the end of an exhaust gas recirculation pipe in the intake manifold, hydraulically operated by an actuator which is controlled indirectly by the position at the accelerator pedal via the fuel injection system.

b) a mixture control unit consisting of air and fuel sensor for precise control and regulation of air-to-fuel ratio.

Precise control of EGR is required in diesel engines because, unfortunately, too high EGR rates increase emissions of hydrocarbons, carbon momoxide, odor and soot. Also exhaust gas recirculation is not without its problems, such as the contamination of the intake passage of the engine and of the lubricating oil. Driveability also deteriorates under large quantities of EGR in production diesel engines. Most EGR systems incorporate an engine coolout temperature lock-out mechanism which eliminates EGR for improved vehicle driveability during engine warmup when NO_x formation rates are low.

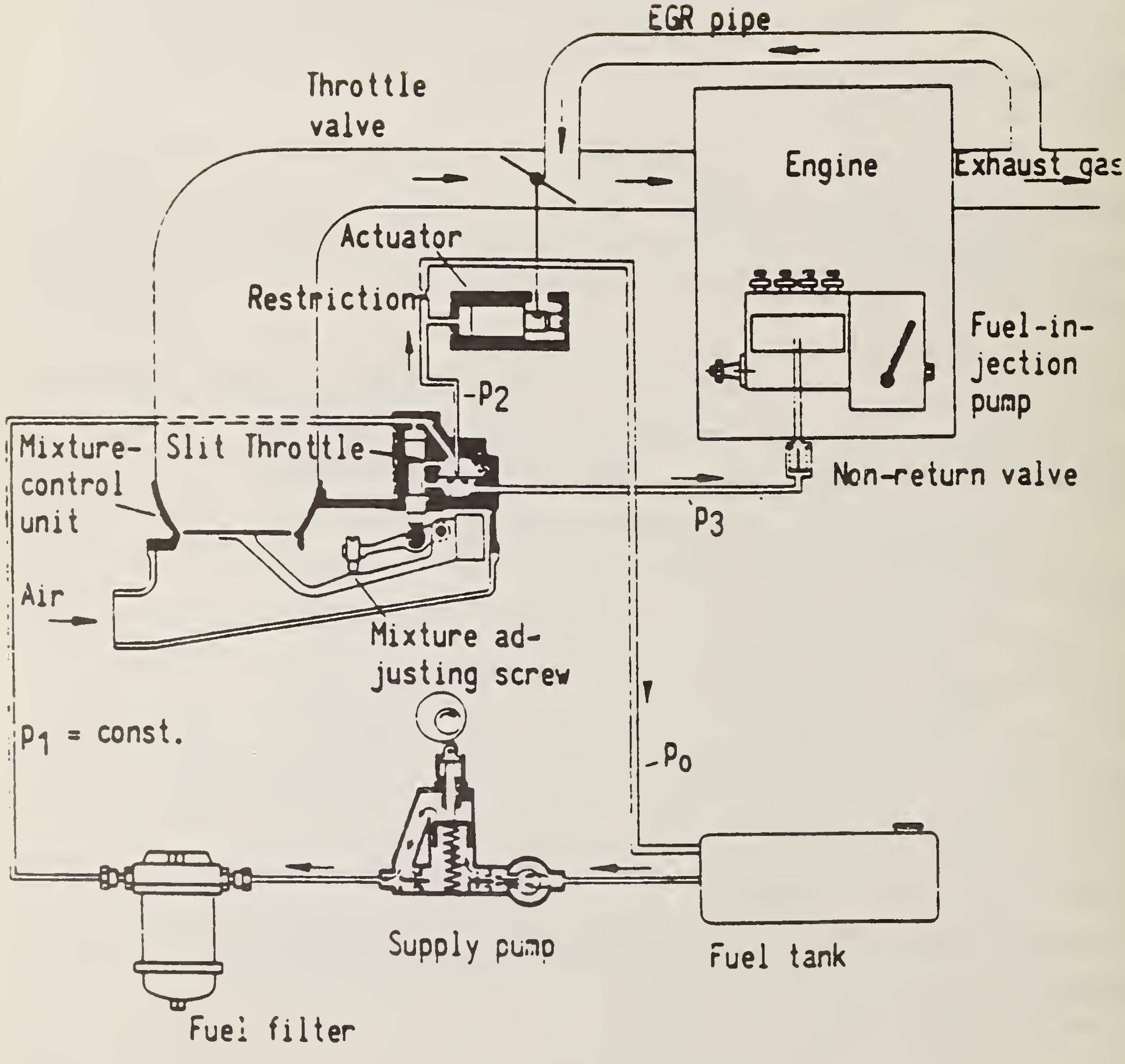

FIGURE 3.5-7. EXHAUST GAS RECIRCULATION SYSTEM

REFERENCES FOR SECTION 3.5

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3.6 ENGINE DESIGN CONCEPTS

Automobile diesel engines are a special family of stratified charge internal combustion engines which use the heat of compression for ignition. They are set to operate over a load-speed range usually under lean mixture conditions. The fuel in a diesel engine is injected directly, or indirectly via an antechamber, to the main cylinder chamber during the engine's compression stroke. Diesel combustion thus takes place in one chamber (direct injection or open chamber) or in a divided chamber (indirect injection or prechamber).

The combustion chamber is the heart of the engine; it produces power as well as the unwanted emission substances.

Combustion occurs in three phases during the fuel injection period: (1) lag or delay, (2) rapid combustion period, (3) combustion of remaining fuel. Initial lag is the most important factor in diesel engine combustion. It is the period between injection and ignition. When ignition occurs in the diesel engine, the premixed portion of the fuel burns with a non-luminous flame. The initial lag depends upon the temperature of the air intake charge, the degree of induced turbulence and engine speed. It also depends on engine operating parameters such as injection retard or advance, type of injectors, compression ratios, fuel-cetane rating and supercharging. Generally, the minimum compression ratio of diesel engines depends on the cetane number of the fuel as well as on the cetane requirement for cold starting. The second, or rapid combustion phase, is the spread of the flame to the main body of the combustion chamber and is dependent on turbulence. In this phase, fuel burns as a diffusion flame. The third and final combustion phase during the fuel injection period is the combustion of the fuel remaining once the flame has progressed throughout the chamber. During this time, the pressure and temperature in the cylinder chamber are so high that the fuel begins to burn as it leaves the injector nozzles.

The design parameters of an engine include the parts of the engine which are cylinder related, for example, compression ratio, degree of heat transfer through walls, geometrical configuration and the parts of the engine which provide a necessary function for its operation. These include the injection systems, valving and auxiliaries such as the water pump and cooling system.

3.6.1 Combustion Systems

General design and performance features and operating characteristics of three fundamentally different diesel combustion systems are discussed. These include a) the indirect injection system, b) the direct injection system and c) the MAN-system. The first is currently the preferred manufacturers' system for passenger cars and light duty trucks. The latter two systems hold potential applications since they currently exhibit independently good fuel economy and/or wide fuel capability.

3.6.1.1 High Speed Indirect Injection Diesel

The indirect injection engine was an outcome of requirements to improve performance by matching the higher rotational speeds of the gasoline engine and is widely used in light duty engines with cylinders under 900 cc.[1] The combustion chamber of the indirect injection diesel consists of an antechamber (prechamber), located above the cylinder head, which communicates with the main cylinder by means of a narrow-throat orifice. There are two types of prechambered diesel engines in automotive use, the quiescent and the swirl. See Figure 3.6-1. The quiescent indirect-injection engines, i.e., Daimler-Benz, are designed for antechamber volume to total combustion chamber volume ratios of approximately 25%. The antechamber is centrally located above the piston chamber. The swirl chamber type employs larger volume ratios (50%) which represents a compromise for best power at rated speed torque. In both types, fuel is injected into the antechamber, where ignition occurs. Generally in swirl chamber engines, the injected spray is directed towards the trailing edge of the orifice for best performance and smoke. Fuel injectors are usually of the single or two-hole type, using relatively low injection pressures since, because of the confined space, spray penetration is not so critical for combustion. Combustion occurs in two stages, the primary in the antechamber of the fuel-rich mixture followed by final combustion which occurs under fuel-lean conditions in the main cylinder chamber, once the expanding gases build up pressure in the antechamber and expell the partially burned fuel mixture through the orifice into the main chamber. The swirl combustion antechamber can be spherical or near-spherical. The throat of this chamber is arranged tangentially to permit a fast air swirl or vortex to be formed in the antechamber. This air swirl promotes fast mixing of fuel and air. The velocity of air swirl is considerably greater than that of the fuel and at all times proportional to engine speed, being of the order of 4 to 12 times the rotational speed of the crank shaft. Heat release rates are much slower with this design and peak pressure and rates of pressure change are reduced. This lowers noise and structural requirements. Time for combustion is no longer constrained by piston motion, permitting this diesel type to operate at reduced timings and at higher rotational speeds with increased power outputs. Since main chamber combustion is controlled by the fuel distribution from the precombustion chamber, peak cylinder pressure is slightly above the compression pressure. This permits pressure charging of the inducted air without excessive cylinder pressure development.

The indirect-injection chamber engines generally exhibit higher heat rejection due to cooling water and exhaust, and higher pumping losses due to throat passage friction, which result in unfavorable fuel economy. They also exhibit higher local thermal stresses caused by the impingement of the high velocity gases discharged from the auxiliary chambers into the main chamber. Thermal efficiency is generally higher for the indirect injection quiescent chamber engine than for the swirl chamber engine, since in the latter design the boundary layer is thinned by the scrubbing action of the swirl, thus causing greater heat to transfer to the coolant. But because of the lower pumping losses between the swirl chamber and the main chamber, the swirl chamber exhibits[2] about a 5 percent advantage in fuel economy over amescent type pre-chamber engines.

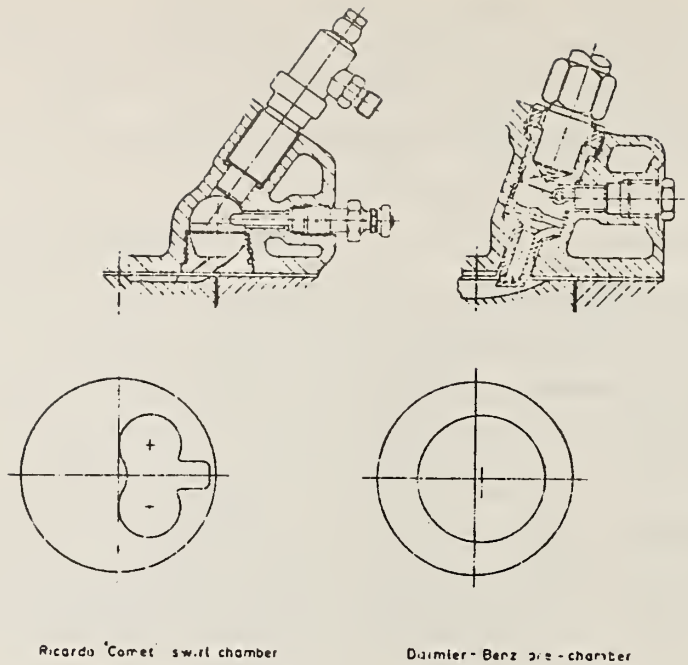


FIGURE 3.6-1. RICARDO SWIRL CHAMBER AND DAIMLER-BENZ PRE-CHAMBER.

3.6.1.2 High Speed Direct Injection Diesel

Direct injection diesels are currently used almost exclusively in all types of automotive applications apart from the light duty automobiles and trucks. The direct injection engine has a single compact combustion chamber, formed in the piston crown. Fuel is injected directly into the space above the piston. The injection system is considered a critical component of the direct injection diesel since it must optimize fuel atomization and penetration into the cylinder chamber. To speed up the mixture formation and increase the engine's performance a swirling motion is introduced during the charging process which is accomplished by using directional or helical intake ports, utilizing piston motion and masking inlet valves.

Direct injection offers better starting, lower heat losses, lower thermal loadings, better smoke characteristics, and fuel economy with an advantage of 8-10% compared to indirect injection.[3] Presently they are limited in speed, which restricts the wide speed range engine requirements to match the transmission of a conventionally engined vehicle and power output without excessive engine size. Better mixing technology is required to improve the DI wide speed range needs, emission (regulated and unregulated) and noise characteristics. This involves improvements in combustion by intricate fuel injection systems, air swirl and micro turbulence configurations. The injection equipment, however, prohibits operation of the direct injection diesel below a 1 liter/cylinder displacement because the size of the injection nozzle becomes too small to give reliable operation.

Efforts are currently in progress at the Anstalt fur Verbrennungsmotoren (AVL) to develop advanced concept (DI) diesel engines for passenger automobiles. Development targets include: a) improvement in fuel economy by 15-20 percent over current automobile diesels, b) engine noise reduction of 15 to 20 dB(A), comparable to gasoline engine levels, c) regulated exhaust emissions; HC 0.41 grams/mile, NOx 1.5 gms/mile (primary), NOx 1.0 gms/mile (secondary), d) unaided cold starting down to 0°F and immediate load pick-up after cold starts, e) reduction of odor and exhaust gas irritancy, f) particulate emissions less than 0.2 grams/mile, g) invisible exhaust smoke at any operating condition, h) engine size and weight to be competitive with current gasoline engines, i) increased high speed capability; rated speed 4500 RPM, high idle speed 5000 rpm, and j) turbocharging.

Table 3.6-1 shows some results of their effort with comparisons of naturally aspirated direct injection and indirect injection combustion systems under equivalent vehicle performance. The DI conversion 2.2 liter naturally aspirated diesel engine in the Mercedes Benz 220D vehicle shows a 20 percent improvement in fuel economy. The DI also exhibits lower particulate, hydrocarbon and carbon monoxide emissions at a designed equivalent NOx level of 1.4 gms/mile. Lower exhaust odor and exhaust gas irritance also were obtained for the DI under instant unaided cold starting conditions. AVL are of the opinion that further improvements

TABLE 3.6-1. COMPARISON OF IDI PRODUCTION ENGINE WITH DI SUBSTITUTE

VEHICLE	ENGINE TYPE	INERTIA WEIGHT (lbs)	CID (in^3)	HYDROCARBON (gms/mile)	CARBON MONOXIDE (gms/mile)	OXIDES OF NITROGEN (gms/mile)	PARTI CULATES (gms/ mile)	COM- POSITE FTP- CYCLE (mpg)	PERCENT DIF- FERENCE DI/IDI%
Opel*1	IDI	3000	126.2	0.57	1.25	1.45	-	32.9	8.5
Record	IDI	3000	126.2	0.39	1.21	1.29	-	26.8	33.0
2100D	DI	3000	134.0	(0.19)*4	(0.94)	(1.58)	(0.31)	(35.7)	-
Mercedes Benz*2	IDI	3500	183.4	0.23	1.43	1.55	0.40	26.8	17
300D	DI	3500	183.4	(0.25)	(1.10)	(1.80)	(0.25)	(31.4)	-
Mercedes Benz*3	IDI	3500	134	0.27	1.25	1.40	0.46	27.6	20
220D	DI	3500	134	0.22	1.0	1.40	0.34	33.2	-

*1 Acceleration Performance 0 to 50 mph, 15.5 sec.
0 to 62 mph, 23.5 sec.

*2 Acceleration Performance 0 to 50 mph, 11 sec.
0 to 62 mph, 19 sec.

*3 Acceleration Performance 0 to 50 mph, 18 sec.
0 to 62 mph, 27 sec.

*4 Bracketed Numbers Denotes Calculated Values

SOURCE: Reference 12.

could be obtained with the DI by incorporating unit injectors and unit-casting of the cylinder head and block to improve volumetric efficiency for better performance at high engine speeds.

3.6.1.3 MAN-Combustion System

The MAN combustion system, sometimes called the M system, was developed by Maschinenfabrik Angsburg Nuremberg. Its design differs in principle from both the direct and indirect injection and utilizes heat from hot walls to disperse and evaporate fuel during earlier stages of diesel combustion. Basically the engine features (Figure 3.6-2) a bowl-in-piston design with fuel injection through an orifice. Approximately 95% of the fuel injected into the bowl is purposefully impinged on the hot chamber walls to form a thin film where it evaporates, decomposes and mixes with high speed air swirl.

From the limited results[4] available on this design, a turbocharged version incorporating the MAN-system has demonstrated good cold starting characteristics, without sacrifice of power, good fuel economy (BSFC .37 lb/BHP-hr) and smoke levels. The latter is controlled by the spray-impinged direction.

This engine concept is used on heavy trucks but has not yet received attention for automobile and light truck applications.

3.6.2 Engine Configuration

The Combustion Chamber geometry of diesel engines plays an important role in the rate of mixing of fuel and air, and thus the performance and emission characteristics of engines. Because of manufacturers' design preferences, the performance optimizations vary from one engine to another and must be treated separately. Results of two recent studies are discussed below.

3.6.2.1 Squish Lip

Perkins Engine Group Ltd in England[5] conducted a series of tests to optimize their direct injection re-entry combustion cham-

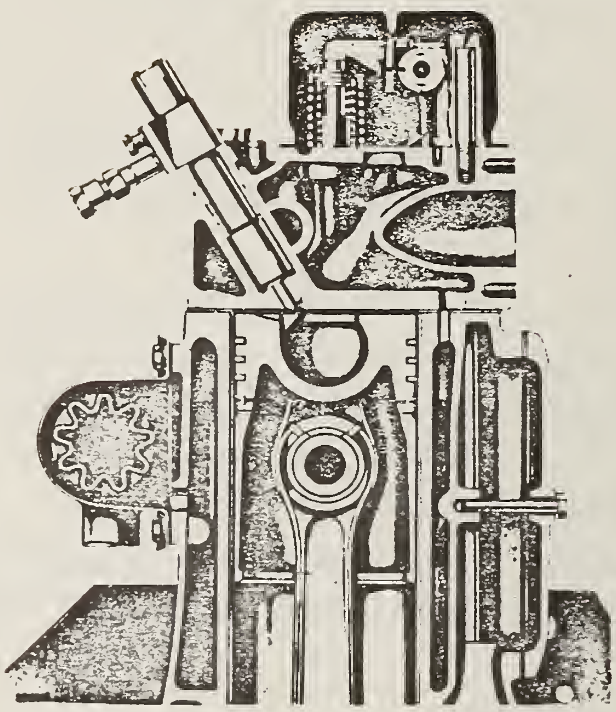


FIGURE 3.6-2. CROSS-SECTION OF 6-CYL. 4.4 x 5.5 IN. PRODUCTION ENGINE. 200 HP MAX. OUTPUT (SUPERCHARGED) AT 2300 RPM

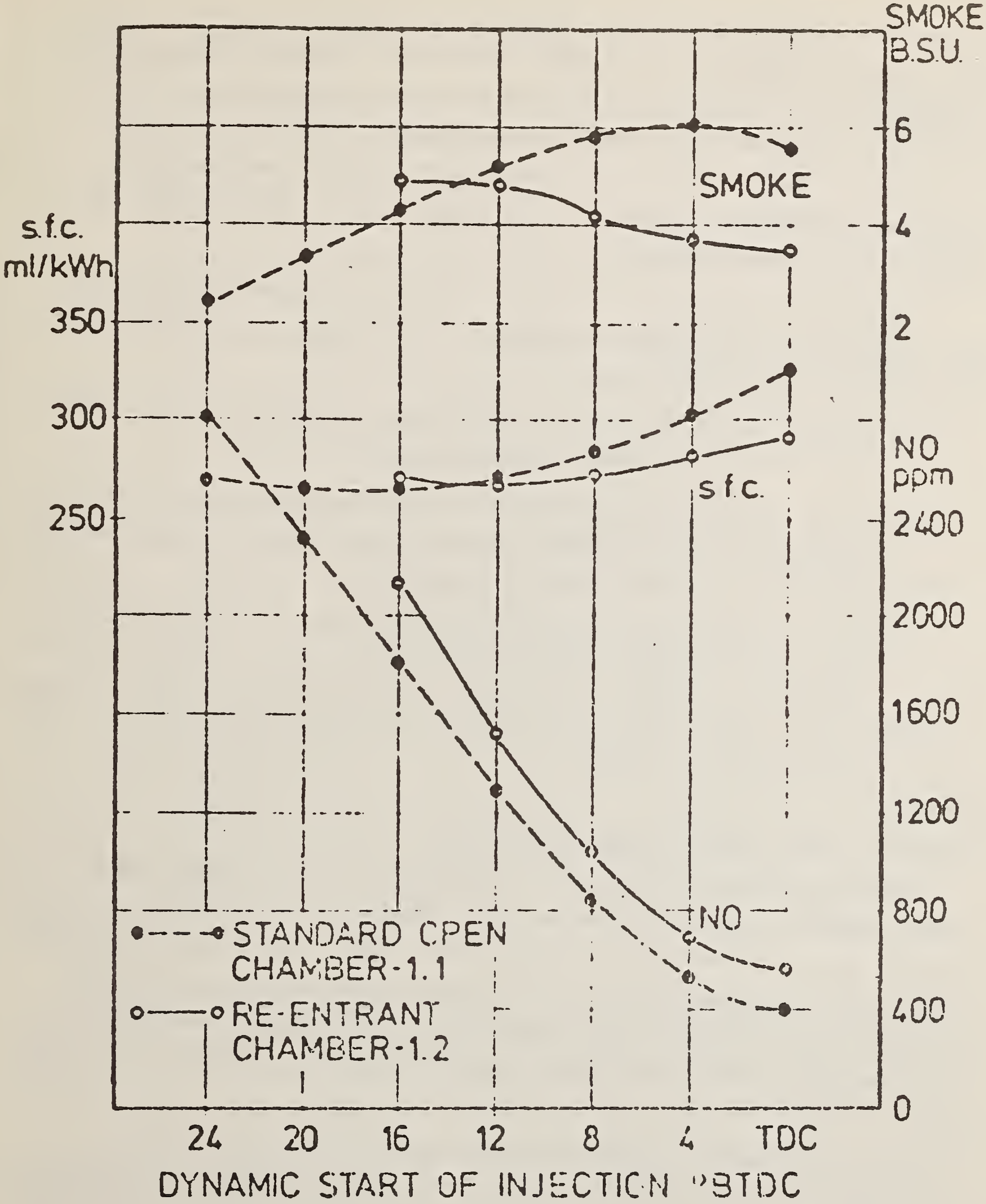


FIGURE 3.6-3. EFFECT OF INJECTION TIMING, STANDARD OPEN CHAMBER, AND RE-ENTRANT CHAMBER, ENGINE SPEED 1400 REV/MIN, EQUIVALENCE RATIO, .83

ber for low emission without compromise in fuel economy. This study showed (Figure 3.6-3) that with the re-entry chamber (bowl in piston, i.e. flash type) having a throat diameter less than the bowl's largest diameter, maximum thermal efficiency occurs at a more retarded injection timing than for the currently designed open chamber engines. This contributes to lowering the NO_X and smoke emissions.

Fourteen combustion configurations were tested, including bowl diameter, bowl throat diameter, flank angles, lip shape and bowl shape. They found that:

a) higher smoke levels, specific fuel consumption and lower NOx result for smaller bowl volumes

b) Lower smoke and higher NOx result for smaller bowl throat diameters with specific fuel consumption a minimum at an intermediate bowl throat diameter

c) For the lip shapes tested, small effects on performance and emissions were observed

d) Flank angle tests indicated that the optimum angle ranged from 2-to-40 degrees.

3.6.2.2 Swirl Chamber Configuration

Recently General Motors(6) undertook a development program on swirl chamber engines to reduce the high NOx levels observed during initial diesel conversion attempts for their baseline Oldsmobile 350. General Motors also observed that two other significant problems were encountered. They were: prechamber durability at full load and excessive combustion noise during warmup and under medium loads.

The problem of durability was not attended to, but GM believes that this could be solved through prechamber development. Combustion noise could be reduced to acceptable levels by using fuels of higher cetane number. Nearly 300 combinations of prechambers and injection nozzles were evaluated to obtain the necessary improvements.

The revised swirl chamber is shown in Figure 3.6-4.

1. Piston heating was corrected by a side outlet prechamber into the valve pockets rather than directly on the piston crown.

2. HC and smoke were reduced by using a symmetrical swirl cavity with low swirl velocity, and multiple hole, fixed orifice nozzles.

3. NOx and noise were reduced by locating the injecting nozzle to the inboard side of the prechamber with the fuel injected in the direction of swirl. A lower injection rate also helped reduce noise.

The General Motors study also revealed that the position of the injection in the swirl chamber had a pronounced effect on emissions. The lowest NOx, power, and noise were achieved with the injection nozzle located inboard of the center of the prechamber and the fuel longitudinally directed into the swirl. However, this configuration produced the highest HC value. It was also found that a) when the fuel was directed across the prechamber, with the injection located inboard of the center of the prechamber, a power change and HC emission, accompanied by higher NOx and noise levels, resulted, and b) The highest power with lowest HC, with higher levels of noise and NOx resulted when fuel was directed across the prechamber with the injector located at the top end.

Further studies showed, Figure 3.6-5, that both HC and smoke could be decreased by changing the direction of the spray from the injector located inboard of the center of the prechamber. However, this has an adverse effect on NOx and noise.

The emission characteristics of an engine are influenced by the number of holes configured in an injector nozzle. Figure 3.6-6, HC, NOx emissions were reduced by incorporating a three hole nozzle but adversely affect idle smoke.

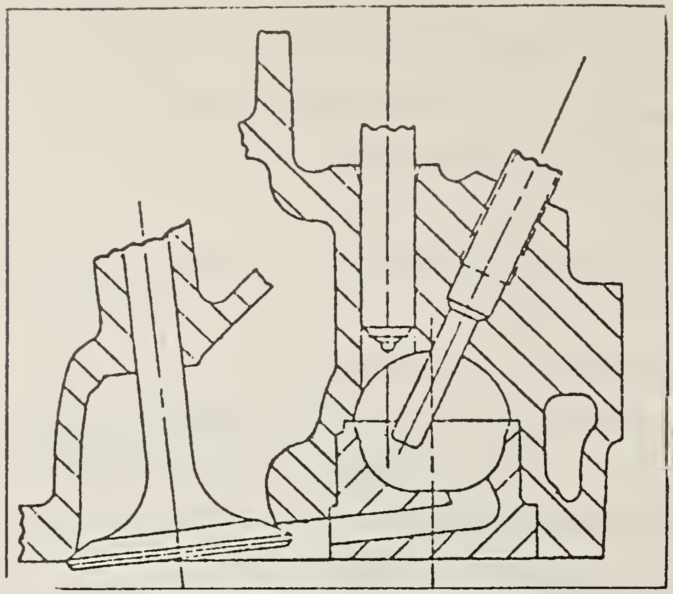


FIGURE 3.6-4. REVISED SWIRL CHAMBER DESIGN

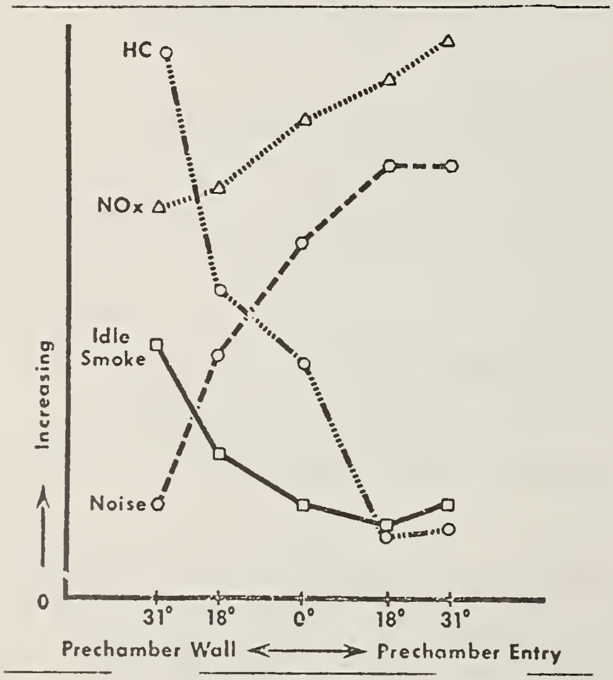


FIGURE 3.6-5. EFFECT OF NOZZLE HOLE ANGLE

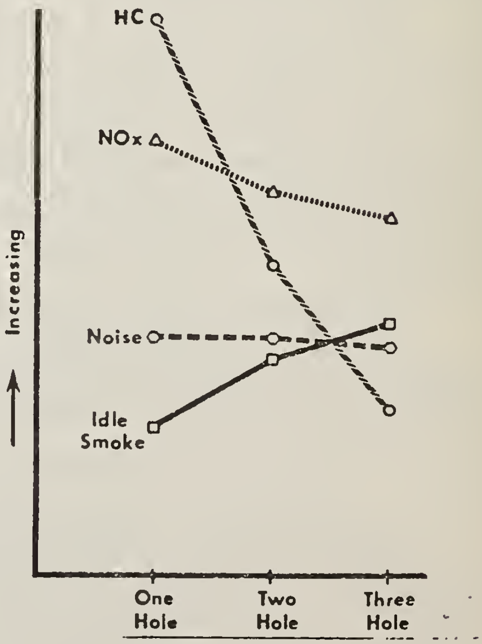


FIGURE 3.6-6. EFFECT OF MULTIPLE HOLES (CONSTANT TOTAL FLOW AREA.)

3.6.2.3 Engine Sizing

Diesel engines are designed for piston diameters ranging from 2 to 37 inches, and speed ranging from 100 to 4,400 rpm while delivering from 1.5 to 33,400 bhp on a single crankshaft.

In general,[7] engine size can correlate with many parameters. In practice, bmep., piston linear speed, specific output (bhp/in^2 of piston area) and ignition delay time appear to tend to fall slowly as cylinder bore increases. Relative to engine performance, except very small engines, as the bore increases, the engine indicated thermal efficiency increases slightly, so with the specific fuel consumption. Cylinders of less than 2 inches bore diameter usually have very poor brake thermal efficiency due to the relatively large loss of heat and friction.

The effects of cylinder size are as follows:

(1) Stresses due to gas pressure and inertia of the cylinder assembly are the same at the same crank angle, provided (a) mean piston speed is the same, (b) indicated diagrams are the same and (c) there is no increased vibration in the engine structure.

(2) When inlet and exhaust conditions and fuel-air ratio are the same, similar cylinders will have the same indicator diagrams at the same piston speed and the same friction mean effective pressure. Under these conditions brake power is proportional to bore squared or to piston area.

(3) Since the weight of a cylinder is proportional to the bore cubed or to the total piston displacement, when the mean pressure and piston speed are the same, the weight per horsepower increases directly with the bore.

(4) The temperature of the parts exposed to hot gas will increase as cylinder size increases.

(5) In diesel engines, as cylinder bore increases, because of reduced speed of revolution it becomes easier to control maximum cylinder pressures and maximum rates of pressure rise. Consequently, fuels of lower ignition quality can be used.

(6) As cylinder bore increases, wear damage in a given period of time decreases; that is, the engine lasts longer between overhauls or parts replacement.

(7) With the same fuel, fuel-air ratio, and compression ratio, efficiency tends to increase with increasing cylinder size due to reduced direct heat loss.

Various scaling parameters of an engine, for example, stroke-to-bore ratio, connecting rod/crank arm radius, exhaust valve and inlet valve lift/diameter ratio, exhaust valve and inlet valve/bore ratio etc. have an important role on the characteristic shapes of the maximum torque curve of an engine with consequent effect on vehicle fuel economy and performance.

An example[8] is illustrated in Figure 3.6-7 of the effect of stroke/bore ratio on performance, BMEP and fuel consumption where a stroke/bore ratio of 1:2 emphasizes low end torque with improvement in fuel economy. The smaller stroke/bore ratio emphasizes top end performance and fuel economy.

3.6.2.4 Diesel Rotary Engine

Rotary diesel engines have as many unique advantages as spark ignition engines. Yet, in using a compression ignition engine, some of those advantages become disadvantages. Some examples are:

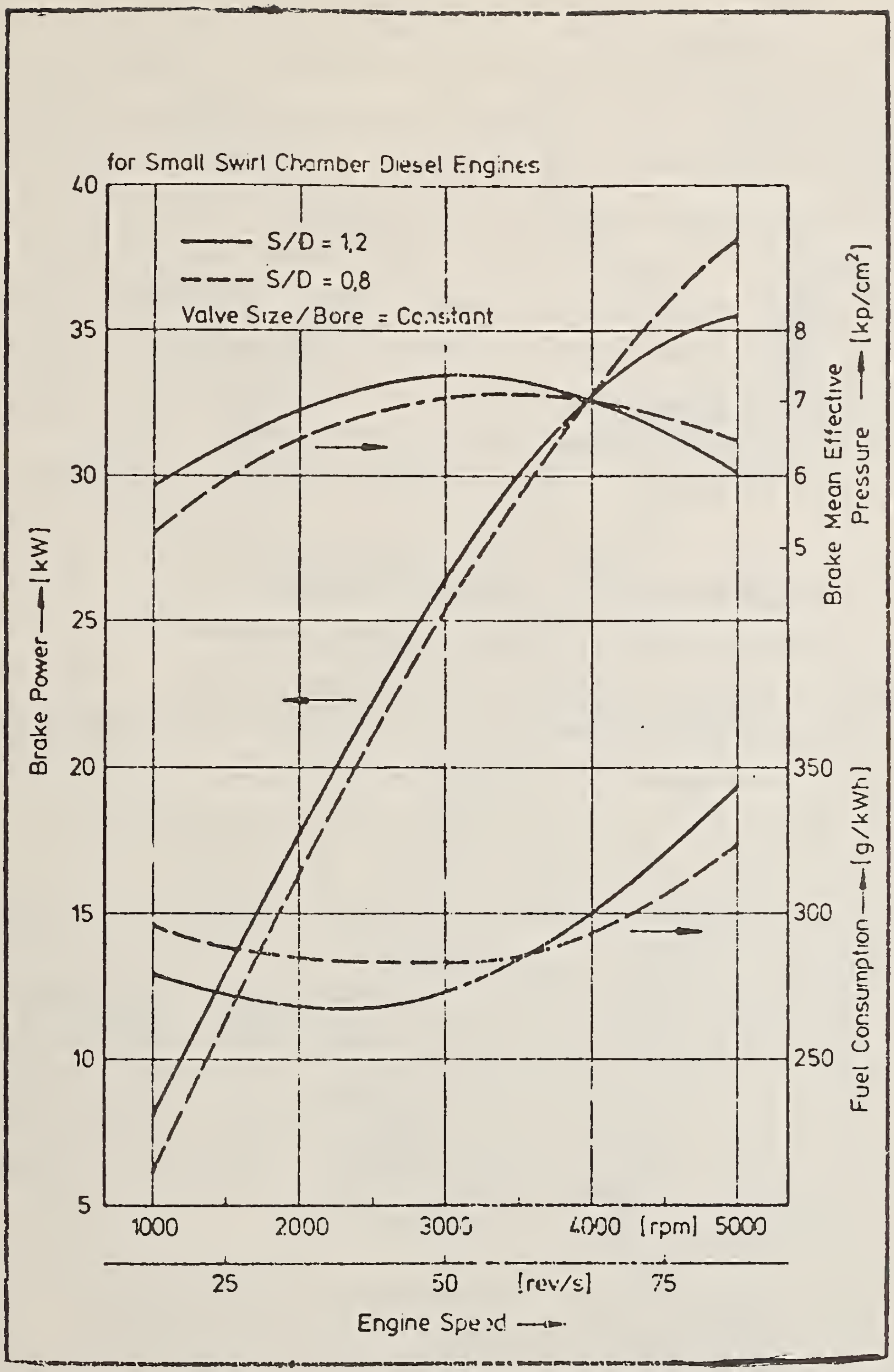
(1) the difficulty of obtaining a high enough compression ratio,

(2) a high surface/volume ratio at TDC,

(3) a shallow and elongated combustion space,

(4) a special gas sealing system, which contains single elements with line contact.

To overcome these difficulties, one of the methods is to use a two stage rotary engine. To demonstrate its practical feasibility, Research Rotary Engines, Rolls-Royce, Ltd.,[9] has conducted research programs since 1964. The major achievement of this development program was the development of an engine with tremendous



Reference: "Data Base for Light-Weight Automotive Diesel Power Plants" Volkswagen. Contract DOT-TSC-1193

FIGURE 3.6-7. ESTIMATED EFFECT OF STROKE/BORE RATIO ON PERFORMANCE, BMEP AND FUEL CONSUMPTION compactness. Figure 3.6-8 shows a schematic diagram of the typical two-stage rotating engine. The basic principle is to compress and expunge the gas medium in two successive rotating cylinders connected by passages. Gas intake and exhaust occur at low pressure cylinder and burn in high pressure cylinder. Different fuel injector positions have been tested. Optimized injector position is found as shown in Figure 3.6-9. The combustion chamber is constructed in such a way that air swirl can be induced. Apex seal had been studied extensively. A set of stepped seals, Figure 3.6-10, having the advantage of reduced mass, was developed and the desired operation of the seal was achieved. Figure 3.6-11 shows some typical results. No emission data is available.

3.6.2.5 Novel Materials for Engine Structure

For comparable swept volume a diesel engine[10] is heavier than a gasoline engine, as shown in Figure 3.6-12. The components which contribute most to the diesel's higher weight are the (1) fuel injection equipment (2) starter motor, (3) the cylinder head. Some of the difference is also made up of the running gear (pistons and connecting rods) and timing drive. The diesel also requires a larger capacity battery, which adds to its weight. A larger radiator may also be required to dissipate the diesel's swirl and quiescent indirect chamber heat. Without resorting to light alloys or novel forms of construction the possible areas of weight reduction are minimal. Iron accounts for about 50 percent of the total engine weight; substituting aluminum would reduce engine weight by about 25 percent. Some components are already in aluminum on a number of diesel engines.

The majority of engine components are produced from castings and forgings, with ferrous castings being predominant. In most cases, the components were over-designed in terms of contained metal because of manufacturing and reliability considerations. A significant weight reduction is already being obtained by redesigning the engine components.

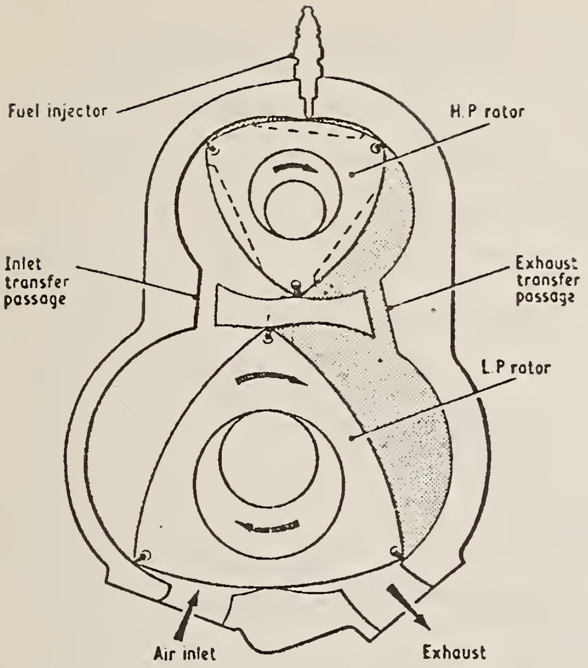


FIGURE 3.6-8. 2-STAGE DESIGN WITH TWO ROTORS

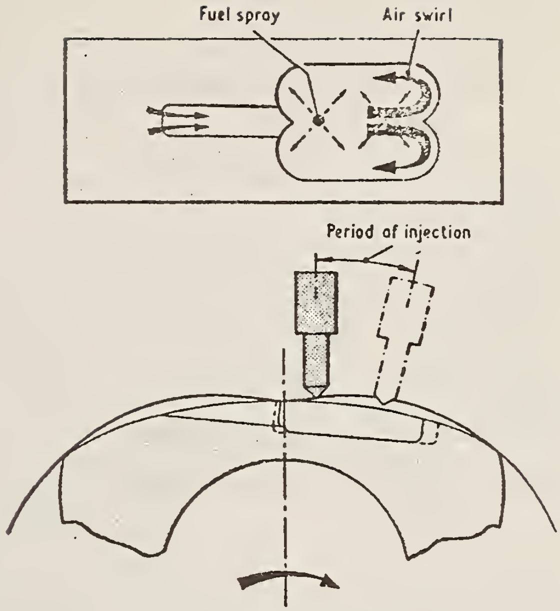


FIGURE 3.6-9. PREFERRED COMBUSTION SYSTEM

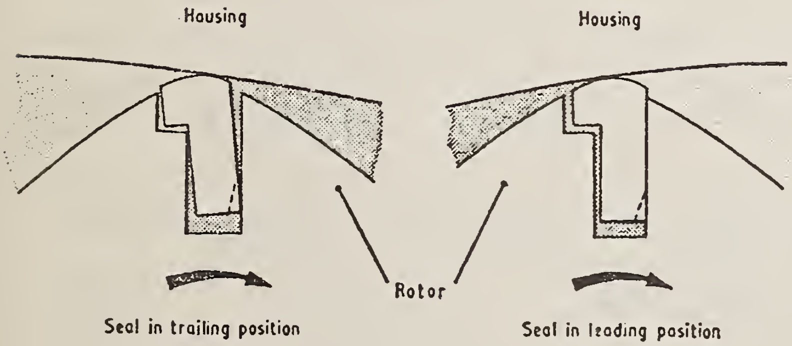


FIGURE 3.6-10. STEPPED APEX SEAL

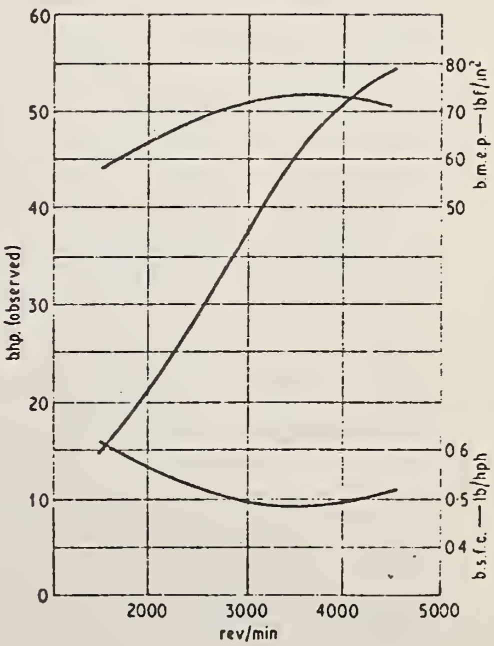
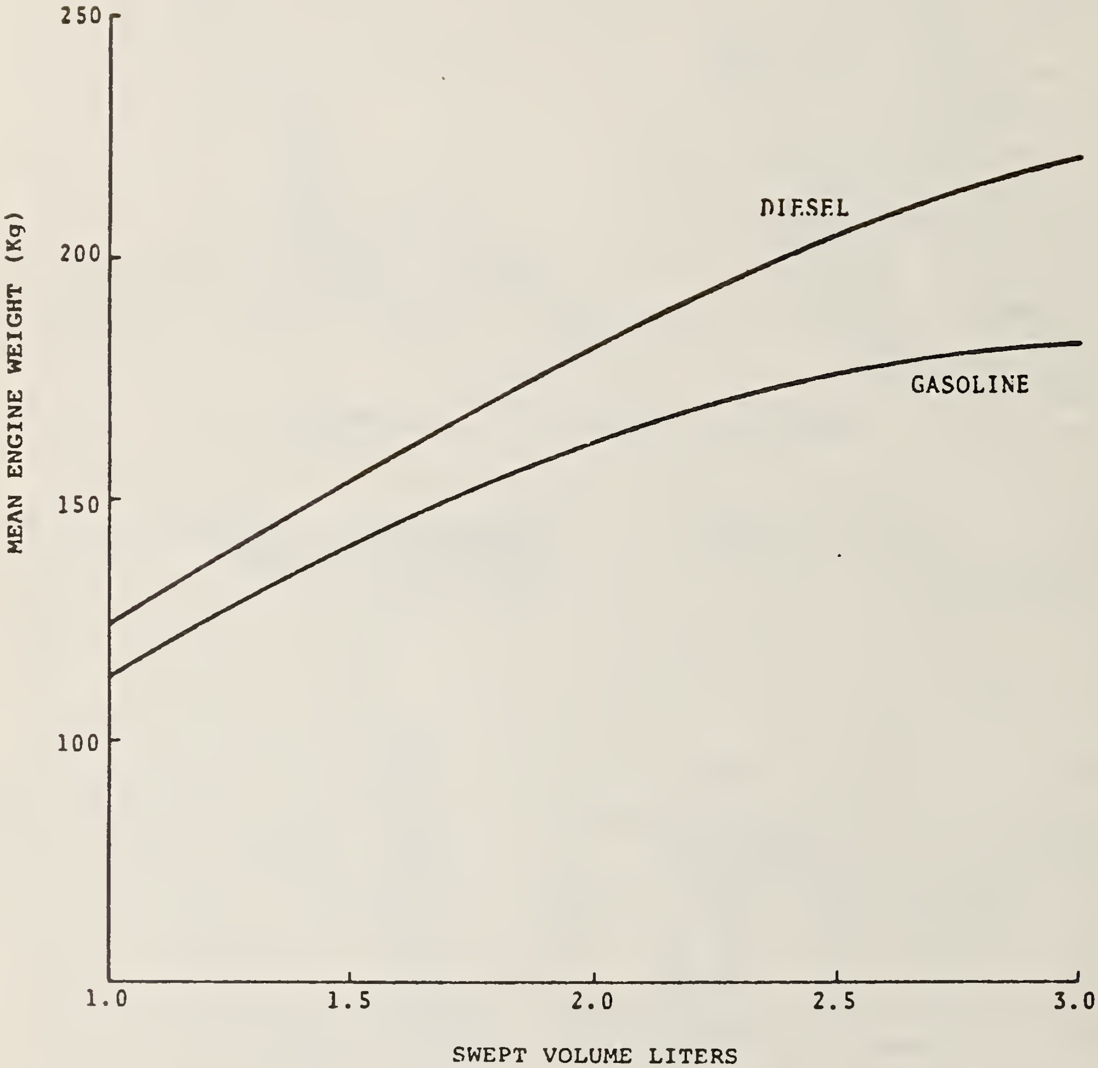


FIGURE 3.6-11. PERFORMANCE OF AN RIE ENGINE WITH AN AIR/FUEL RATIO OF 30:1



Source: Reference 10.

FIGURE 3.6-12. MEAN DIESEL AND GASOLINE ENGINE WEIGHTS

Beyond these efforts, weight reduction of engine components may be obtained by materials substitution. In the nearer term, this entails the replacement of ferrous castings by light metal alloy castings, principally aluminum, or by molded plastics. Cast aluminum is now replacing cast iron in cylinder heads.

Most of these developments are directed towards those improvements of an engine that operate under the more benign environmental conditions that exist under the hood of an automobile. Ferrous materials are still the materials of choice for those components that are exposed to the more severe environmental conditions, particularly higher temperatures. These components include crankshafts, camshafts, connecting rods, etc.

In the longer term, the requirements of high rigidity, stability at elevated temperatures (up to 500°F), environmental resistance, and low weight, may be met by some novel composite materials.

Graphite fiber reinforced epoxy and polyester resins are currently being explored as lightweight structural materials for body and chassis applications. They have limited use in engine applications, because of the temperature limitations of the plastic matrix materials (up to 350°F for selected epoxy resins). A number of plastics have been introduced into commerce that have distortion temperatures well in excess of 500°F. These include theromosetting resins such as Thermid 600 (Gulf Specialty Chemicles), 18 polyamide and thermoplastic resins such as Torlon (Amoco), a theromoplastic poly (amid-imid). Reinforcing resins such as these with graphite should result in materials that can be used in functional parts currently limited to ferrous materials.

These would include push rods which could be 70 percent lighter than the equivalent metal push rods. The composite push rod should also increase engine efficiency and reduce vibrational noise. Using graphite fiber reinforced composites in other engine compounds--such as connecting rods, wrist pins, and rocker arms--makes possible a reduction in counterweight on the crankshaft. The total effect would be to increase engine horsepower per unit volume of displacement and thereby allow a reduction in engine size.

Short fiber metal matrix composites could potentially be used in those applications also. In this case one would be dealing with aluminum or magnesium reinforced with either alumina or silicon carbide fibers. The presence of 15 percent reinforcing fibers in the metal matrix significantly increases the strength and the stiffness of the matrix metal without seriously affecting the metal density, nor the ability of the metal to be worked by standard fabrication techniques.[10]

The engine block, currently made of cast iron, is the heaviest component in current internal combustion engines. Material substitution in this area offers the promise of significant weight reduction. Successful use of die cast aluminum engine blocks has required the use of iron liners, either inserted at the casting stage, or later in machinery and assembly. These have been costly because of resulting slower fabrication and additional machining. An aluminum material is required with good wear and scuff characteristics to obviate the need for the cast iron liners. This was nearly accomplished with the hyperentectoid aluminum silicon alloy used to make the block of the Chevrolet Vega. The Vega aluminum engine is no longer in production and has been replaced with a cast iron block because of operational problems in the field.

A radical approach that has been proposed in the literature,[10] and that could lead to significant reduction in power plant weight and increased fuel efficiency, is the operation of an engine under adiabetic conditions, so as to eliminate the need for any cooling system. Adiabatic operation would result in higher operating temperatures, particularly at the crown of the piston. This temperature would increase to 1300°F so that aluminum pistons could no longer be used. The use of ceramics, in particular silicon nitride, has demonstrated that a monolithic silicon nitride piston can be operated in a modern diesel engine environment.[10] Further significant efforts are required before a ceramic engine reaches commercial return.

3.6.3 Other Design Variables

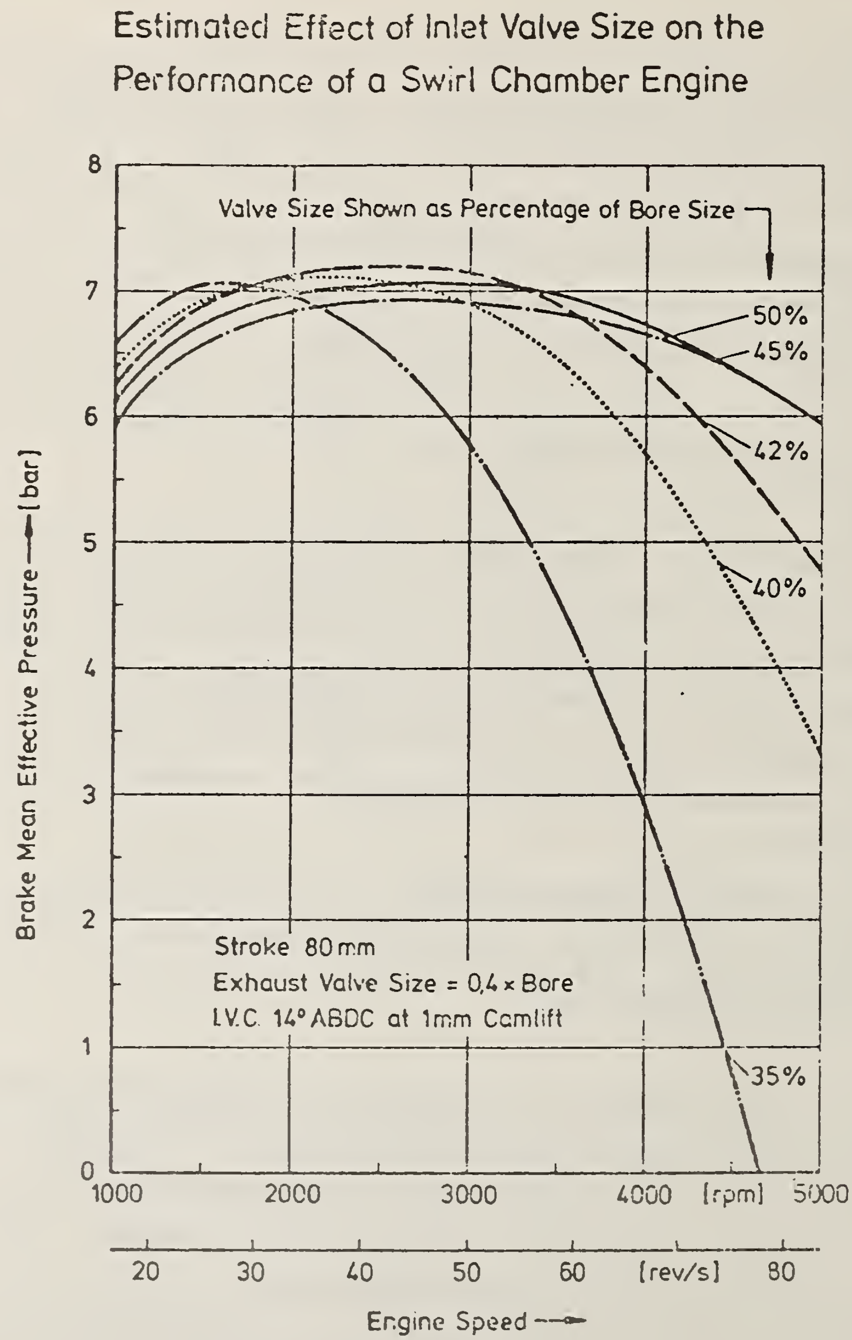
3.6.3.1 Valving

Valves control the inlet and exhaust gas of the engine. Valve geometry, timing, duration, and lift all influence engine performance. Basically, the valve geometry together with the inlet duct and cylinder design relate to intake swirl turbulence, their patterns and levels. Valve timing, duration and lift have a direct effect on the engine's volumetric efficiency. In addition, certain operations of valves, such as overlap, will also influence air compositions and physical parameters, for example, temperature and pressure.

The poppet valve is universally used in 4-cycle engines. There are other types of valves such as sleeve, piston and rotary but these valves are not currently as acceptable as the poppet valve. The poppet valve possesses the following features(7): a) excellent flow coefficients, b) low manufacturing cost, c) very little friction, requiring less lubrication, d) needs no cooling on exhaust valves.

Poppet valve design must achieve satisfactory results in respect to amount of gas flow, cooling and heating flow, structural strength, lubrication and wear.

Parametric study of valve operations on volumetric efficiency has been pursued rigorously and extensively.(7) Figure 3.6-13 shows the effect of inlet valve size on the performance of a swirl engine for an exhaust valve size equal to 0.4 x Bore. Smaller valve size emphasizes low end performance and larger valve size emphasizes high end performance with a corresponding flatter torque curve. Figure 3.6-14 presents some of Ricardo's test data on valve closings and the effect on volumetric efficiency. As can be seen, there is an optimized engine speed for maximum volumetric efficiency and inlet valve closing time has a significant effect on volumetric efficiency at either high or low engine speeds.



Reference: "Data Base for Light Duty Automotive Diesel Power Plants"
Volkswagen Contract DOT-TSC-1193

FIGURE 3.6-13. ESTIMATED EFFECT OF INLET VALVE SIZE ON THE PERFORMANCE OF A SWIRL CHAMBER ENGINE

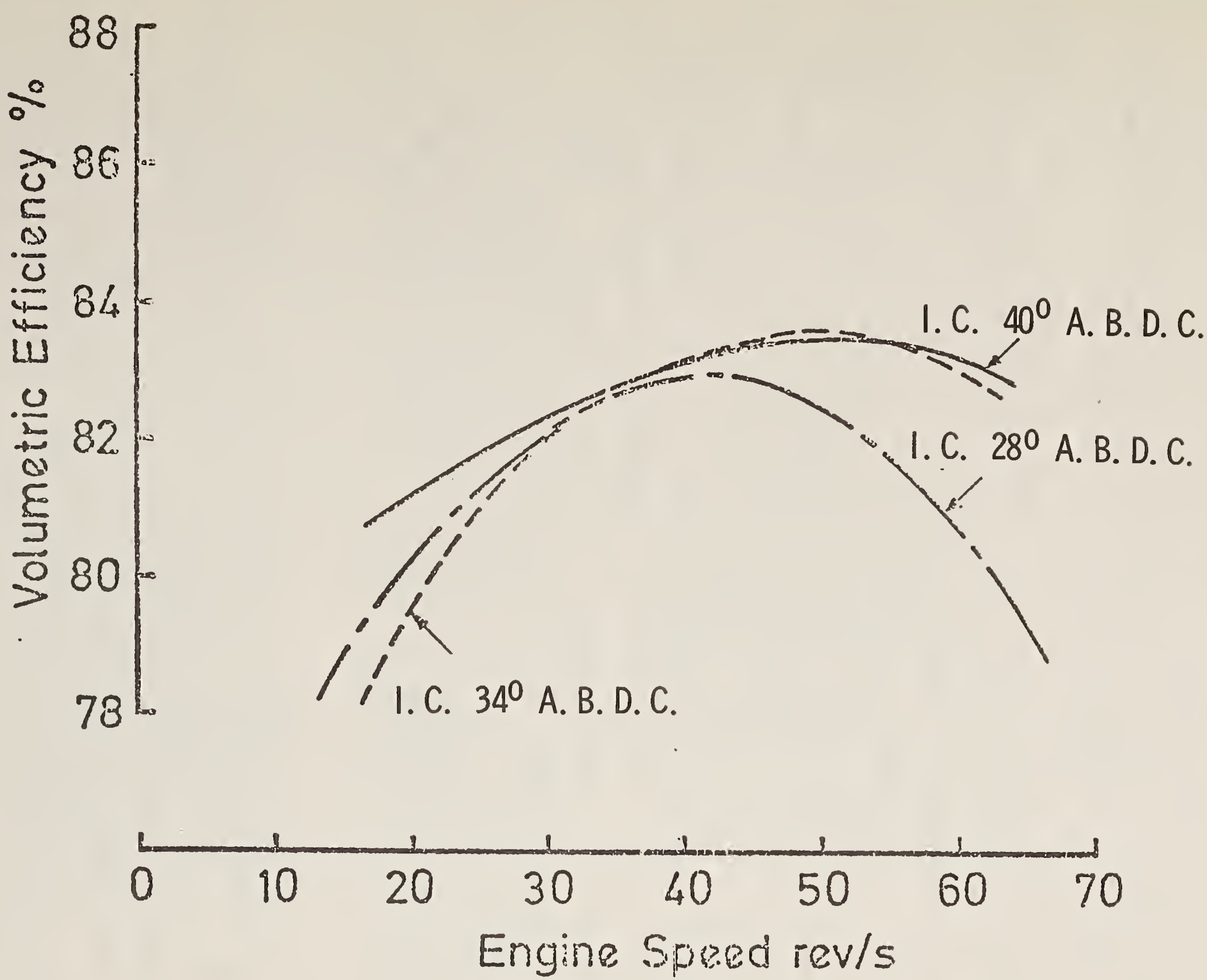


FIGURE 3.6-14. THE EFFECT ON VOLUMETRIC EFFICIENCY OF VARIOUS INLET VALVE CLOSINGS ON A 3.6 LITRE 'COMET' ENGINE

As mentioned above, exhaust valve cooling is a very important problem. Ordinarily, valves are made of austenitic steels, EV3 to EV11. Because of the cooling requirements, a relatively large stem diameter, plenty of material in the valve head, minimum exposure of stem to hot gases, and coolant passages in all seats and stems are all considered. For valves of about 2 inches in internal diameter, cooling becomes necessary. Sodium is frequently used as an internal cooling material.

Engine valve deactivation has shown an improvement(11) in fuel economy for a gasoline powered passenger car of approximately 30 percent. This concept provides deactivation of the intake and exhaust valve of certain cylinders during the vehicle's operation to allow the engine to operate at more favorable brake mean effective pressures. For spark ignition engines, this minimizes the part throttle losses and provides improvements in fuel consumption.

Presently there is no available test data showing the potential of this concept in a diesel engine. It is speculated that valve deactivation may provide only marginal improvements in fuel economy since the diesel operates without a throttle. Besides its effect on fuel economy, the valve deactivator concept may not be readily implementable in a diesel engine since a control logic must be integrated in the fuel injection system to prevent fueling during the deactivation mode. Other performance and operating characteristics of the engine may also be affected by this concept.

3.6.3.2 Compression Ratio

From a theromodynamic point of view, higher compression ratio results in higher thermal efficiencies in general. Diesel engines usually run at higher compression ratio and this is one reason that diesel engines have better fuel economy compared to gasoline engines. For a direct injection naturally aspirated diesel engine, the compression ratio is between 16:1 and 18:1. Swirl chamber indirect injection engines are designed for higher compression ratios of about 20:1 to 23:1. The minimum compression ratio of diesel engines depends on the cetane number of the fuel used as well as on the cetane requirement of the engine for cold starting (design compression ratio at a given cetane number). This limitation is related basically to the auto-ignitability of the compressed fuel-air medium in a diesel cylinder. Cetane number is the index which characterizes the auto-ignitability.

The diesel design problem is to reduce mechanical and thermal stresses and NO_X emissions by reducing the compression ratio, to increase the mean effective pressure by cutting down the combustion chamber dead volume, and to improve the power-to-weight ratio. Increased ignition delay, cold-starting and warm-up difficulties, excessive pressure rises and peak pressures result from the reduction of the diesel's compression ratio, as well as intolerable combustion noise, delayed and incomplete combustion. Higher ratios on the other hand are generally chosen for their lower fuel consumption, lower smoke levels and better ignition characteristics. However, there are some limitations, Figure 3.6-15, since higher friction losses[12] and reduced combustion chamber turbulence levels[10] tend to cancel these gains and reduce the effective speed range of the engine.

Tests on various compression ratios in IDI engines[1] showed that by increasing the compression ratio, lower HC/CO/NOx emission levels and noise resulted. Similar tests[1] on a DI single cylinder engine showed that NO was reduced by reducing the compression ratio but smoke increased. Because of these effects studies are in progress utilizing the variable compression ratio (VCR) concept to optimize engine operations.

Teledyne[10] examined the VCR concept as a means to reduce the peak transient smoke bursts to acceptable levels. By varying the compression ratio of a turbocharged engine from 16 to 9 over load (no load to full load), actual thermal efficiencies were observed higher than initially assumed for an engine employing a 9:1 compression ratio. With optimized components, the test engine smoke level was reduced compared to the standard engine, Figure 3.6-16.

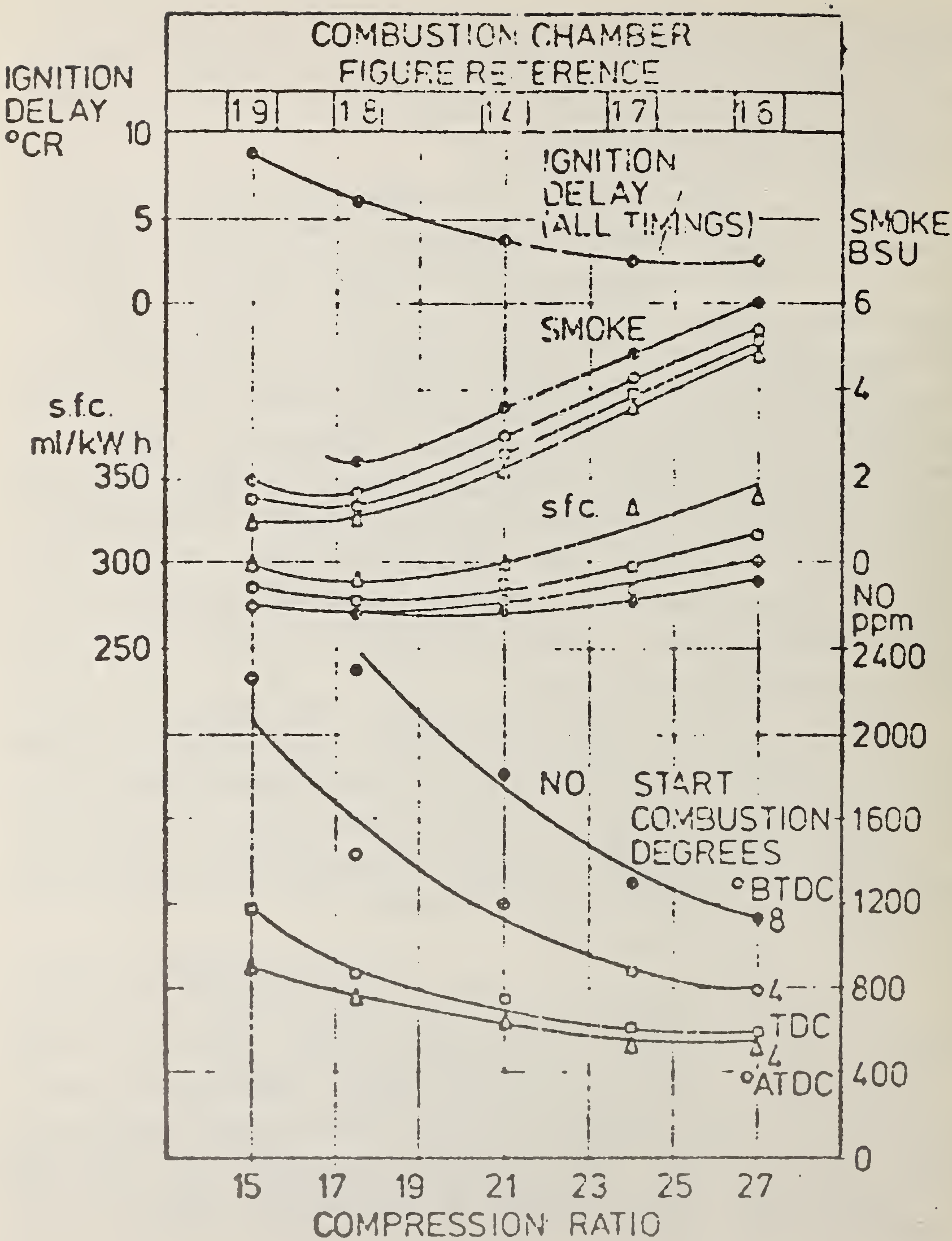


FIGURE 3.6-15. EFFECT OF COMPRESSION RATIO ON SMOKE, SFC, AND NO EMISSIONS, ENGINE SPEED 1400 REV/MIN, EQUIVALENCE RATIO, .83

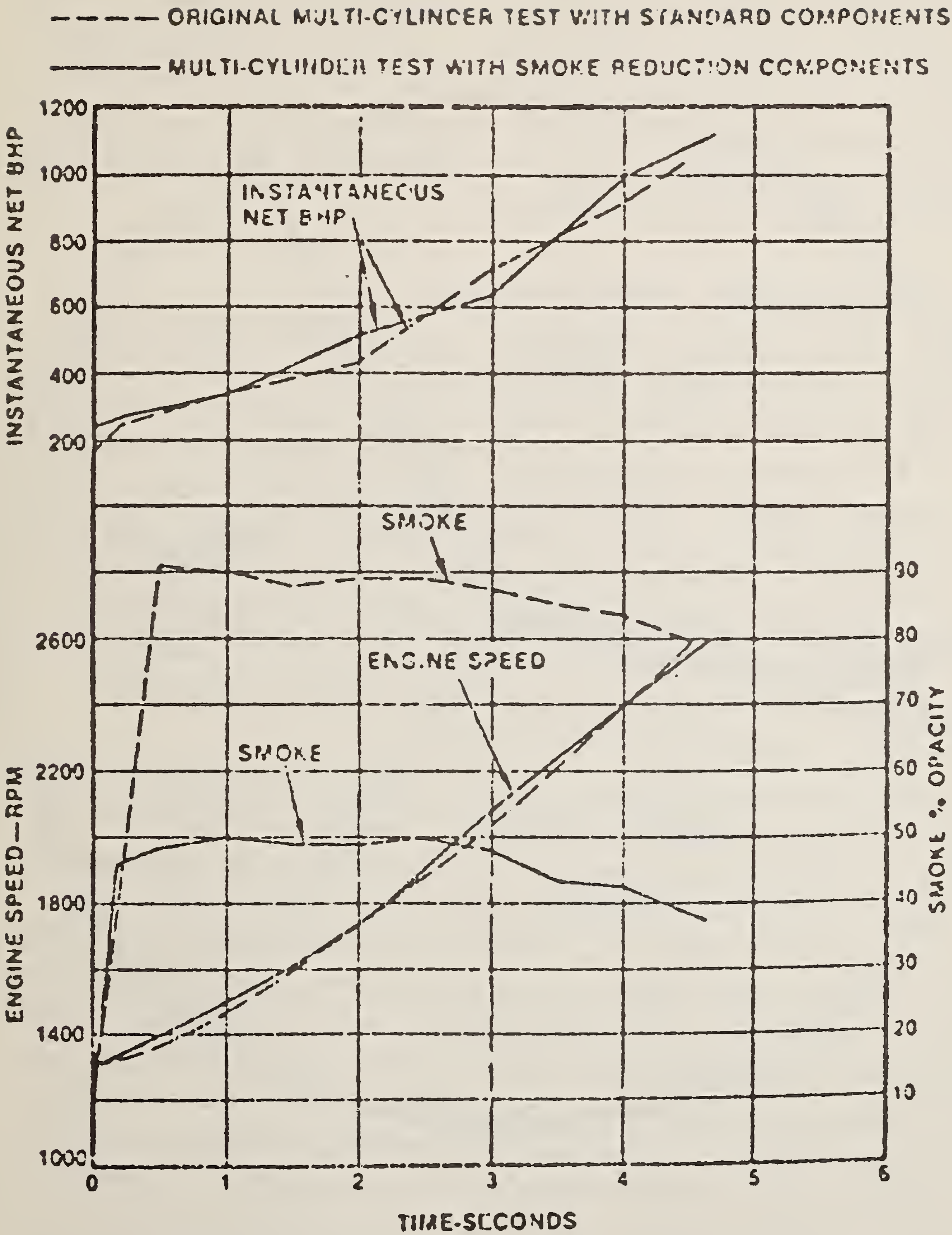


FIGURE 3.6-16. MULTI-CYLINDER ENGINE TRANSIENT ACCELERATION SMOKE TEST

The response times of VCR pistons is still a major factor. Earlier larger versions of VCR pistons exhibited response time of approximately 5 seconds at idle with 2 seconds at highway speeds. The impact of the VCR pistons response and the associated potential for brief periods of excessive cylinder pressure on consumer acceptability is presently unclear. Integrating VCR pistons with turbocharging has yet to be resolved.

3.6.3.3 Engine Losses

The indicated power developed by burning fuel and air in a diesel does not all appear as brake power since a part is lost in overcoming friction of the bearings, pistons and other mechanical parts of the engine, in induction of the air charge and in delivery of the exhaust gases. Friction power loss appears as heat dissipated by the radiator and oil cooling system. Ideally, the total loss is derivable by subtracting the brake power from the indicated power measured from a cylinder pressure diagram. Figure 3.6-17 shows an energy balance from a typical diesel engine.[12] Energy losses to the exhaust and cooling system, represented approximately two-thirds of the energy.

Generally, heat transfer from the combustion gases is dependent on the temperature, pressure and velocity of the gases and, in the case of the diesel engine, on radiation from carbon particles in the flame which, at high loads plays an important part. Overall, the heat losses of an indirect injection diesel at high load are greater than those of a gasoline engine of the same power output. At part load, radiation plays a much smaller part. The diesel's mean gas temperatures fall rapidly and as a result, the diesel engine has a lower heat rejection at low loads than does the gasoline engine. It is well to remember that the coolant load reflects heat transfer from all of the process including friction. Losses arising from heat transfer can be reduced by developing combustion systems that require the minimum of air movement consistent with performance objectives, exhaust limitations, noise, or other required or desired characteristics. A low heat loss chamber will also improve starting characteristics and fuel economy.

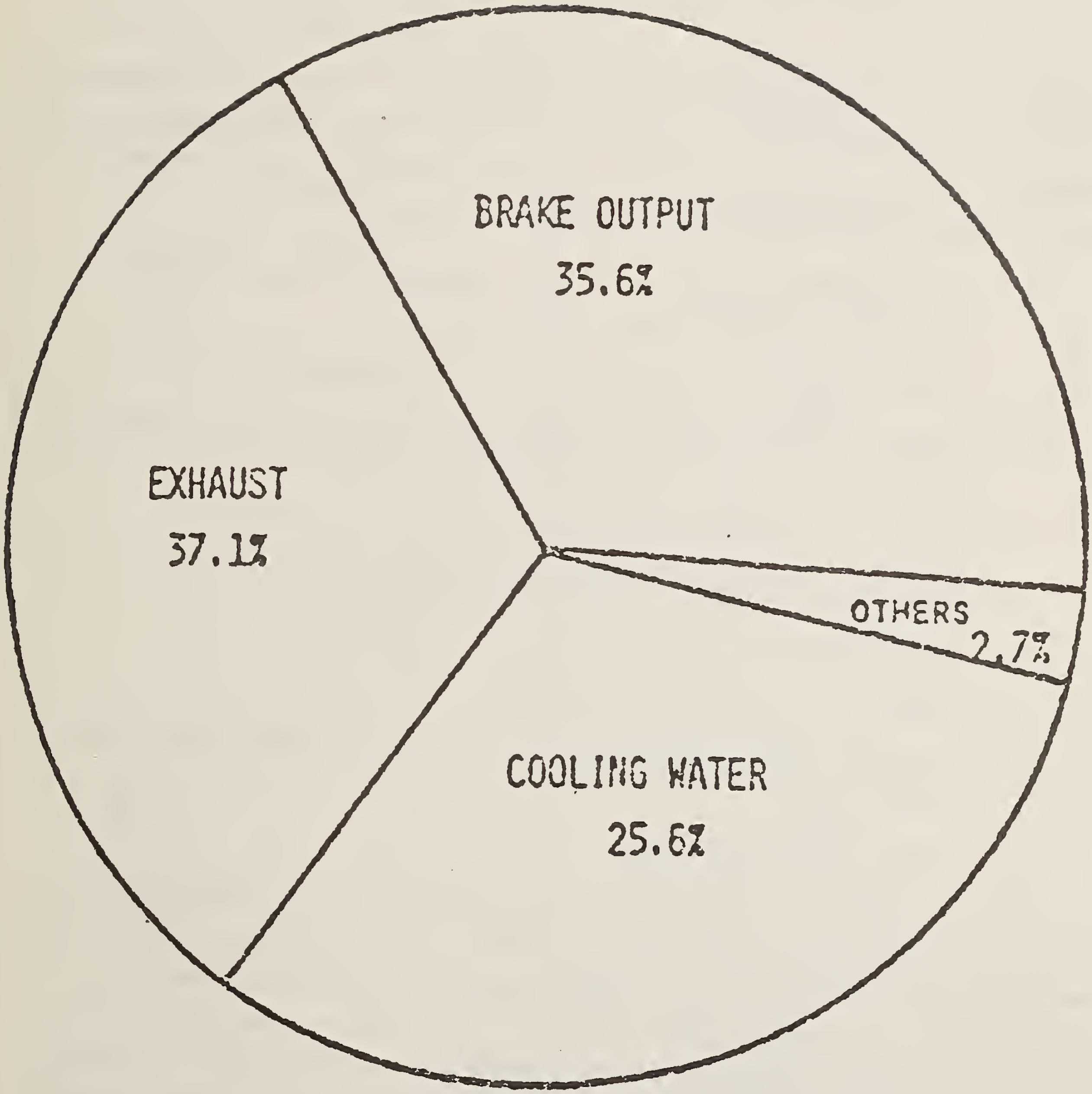


FIGURE 3.6-17. TYPICAL ENERGY BALANCE OF DIESEL ENGINE

Friction losses result from the mechanical friction between lubricated surfaces, as distinct from losses arising from pumping air in and out of the cylinder. Decreasing frictional losses increases the maximum power potential, first because of the direct increase in brake torque and secondly because the engine can be operated at higher speeds. The piston assembly[13] significantly contributes to mechanical losses which are attributed to rubbing. Table 3.6-2. These losses can be reduced by piston assembly design changes. This should result in a lower specific fuel consumption. Other design considerations suggested include[14]:

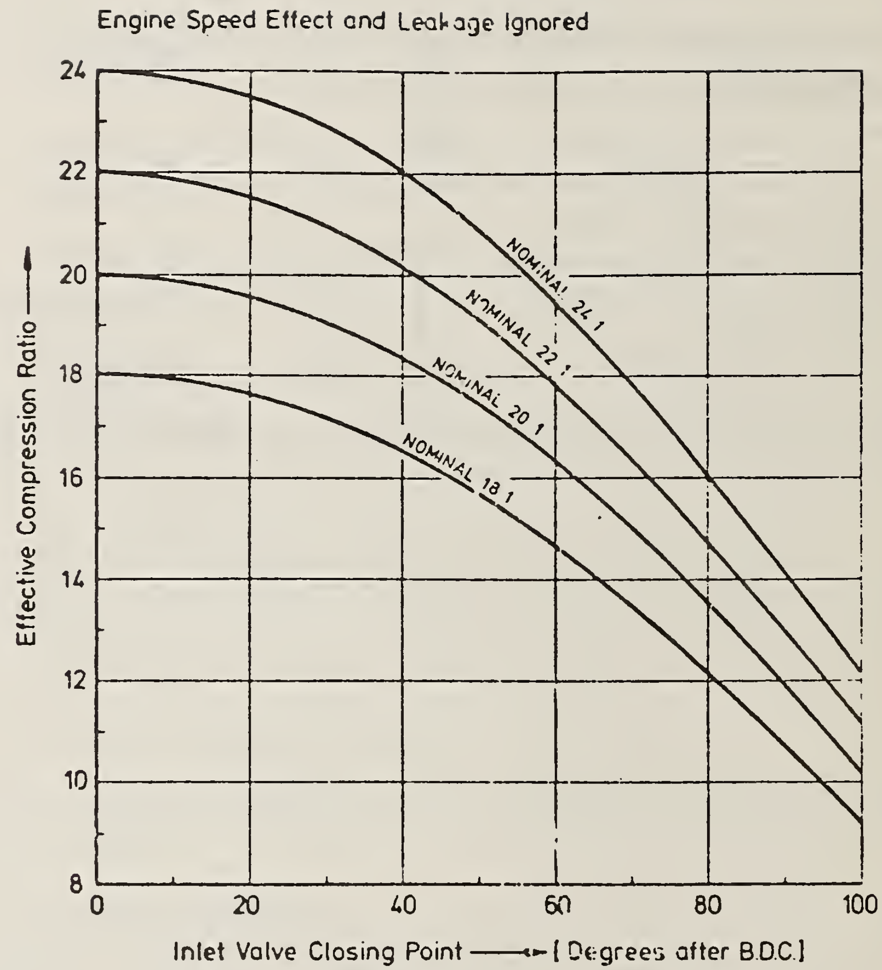
1. Use minimum possible number of low tension, barreled periphery piston rings.
2. Design pistons as light-weight as possible.
3. Reduce piston skirt area by relieving surfaces, keeping relieved surfaces well drained of oil.
4. Keep the compression ratio as low as possible.
5. Use large valves for minimum pumping losses.
6. Use light oils for low viscous losses.
7. Keep bearing diameters small.
8. Minimize power demand of oil and water pumps and other auxiliaries.
9. Inlet valve control.*

Swirl chamber engines generate higher losses than the direct injection engines at comparable speeds. The increased losses are partially due to the swirl chamber and to additional pumping, compression, and valve gear losses. The higher compression loss for swirl chamber engines is an indication of the great increase in heat loss that is caused by firing conditions at high compression ratios.

*Note: See Figure 3.6-18 for an example of the effect of inlet valve closing on effective compression ratio.

TABLE 3.6-2. BREAK-DOWN OF 100% OF MECHANICAL LOSSES

COMPONENTS	PERCENT
FRICTION PISTON	40 to 45
FRICTION BEARINGS	25 to 30
VALVE TRAIN AND GEARS	14 to 17
WATER AND OIL PUMPS	9 to 10
INJECTION PUMP	6 to 8
	~ 100



Reference: "Data Base for Light-Weight Automotive Diesel Power Plants" Volkswagen, Contract DOT-TSC-1193

FIGURE 3.6-18. VARIATION OF EFFECTIVE COMPRESSION RATIO WITH INLET VALVE CLOSING POINT

3.6.4 Engine Startability

The ability of diesel engines to be started in cold weather is presently an immediate critical concern which may bear a large impact on the consumers acceptance of a diesel powered car or light duty truck in the 1980 through 1990 time frame. The other immediate concern is the odor they produce.

Since there are many different types of diesel engines, each has its own unique temperatures and weather condition range at which it can be started. Lower outdoor temperatures require a much longer starting time.

Cold weather starting difficulties of current diesel passenger cars and light duty trucks generally may be overcome by: a) designing the diesel for a higher compression ratio at low loads, b) using an ether spray, c) preheating the coolant water, perhaps requiring a higher capacity battery, d) using a higher cetane diesel fuel, e) low viscosity lubricant, f) variable valve lift; g) plasma plug etc.

A number of startability tests, Figure (1) were conducted by Fiat[15] with a winter diesel fuel of the European market to establish, a) the environmental temperature required to start the engine in 15 seconds under different lubricants and unaided starting conditions using a 12 volt battery, b) the time required to warm up their glow plug before cranking the engine at different environmental temperatures and c) the lowest temperature which the engine could be started in 15 seconds by preheating the engine cooling water. From these tests it was concluded that with an SAEW-20 oil and a lower weight SAEW-10 oil, the engine could be started in 15 seconds at -16°C and 21°C respectively, with a standard battery and a Bosch 11 V Preheater plug in the swirl chamber. The lowest temperature limit for starting an engine in 15 seconds is -35°C with a heater installed in the block.

For cold weather starting, General Motors[6] advises their customers to switch from number (2D) two diesel fuel to number (1D) one diesel fuel or a winterized 2D fuel when ambient temperatures are below 20°F. They have achieved a start in temperature conditions as low as -40°F by using a block heater, warm battery and number one diesel fuel.

Once an engine is started, the warm-up period also has an adverse effect on hydrocarbon, particulate and smoke emissions, noise and odor because the engine is cold and takes a finite time to warm up. This is a greater problem with passenger car diesels than with heavy duty truck diesel, and much research work is required to minimize this effect.

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3.7 Engine Emissions/Noise Control

The exhaust emissions from light-duty diesel vehicles are an important consideration and may have a dramatic role in the penetration of the diesel in the 1980-1990 automotive fleet. Presently, diesel engines are able to comply with the 1979 Federal emission standards (1.5/15.0/2.0 HC/CO/NOx gms/mile, See Table 3.7-1) with minimum impact on fuel economy. Improvements in the emission characteristics of the engine have been achieved with minimal control changes or modifications through injection timing and injector nozzle sac volume alterations. These techniques are not sufficient, however, to meet the 0.43/3.4/1.0 HC/CO/NOx emissions standard in the 1980's and standards proposed beginning in 1981 to reduce the particulate emissions and future desires to reduce other currently non-regulated emissions which are specific to the diesel, for example, odors, irritants, sulfates and aldehydes. However, the evolving advanced control techniques such as an exhaust catalyst and particulate filters, EGR, turbocharging, electronics and engine designs which promote appropriate mixing and turbulence, and advance fuel injection systems are expected to allow the diesel to achieve the exhaust emission levels in the 1980-1990 time frame. Totally integrated, these techniques are not expected to compromise fuel economy. However, their potential for improvement in fuel economy is presently unknown because of the many complex interrelations. These advances can only be made by continuous development of engine components and system technology.

Fiat data summarized in Table 3.7-2 focuses on the current state of the art technology. Different emission goals are summarized along with the corresponding fuel economy changes expected for different engine and vehicle modifications for a vehicle of 3000 lb inertia weight. The lower particulate emission levels (0.2 gm/mi) and NO_x level (1.0 gm/mi) intended for the 1980-1990 time frame cannot be achieved with turbocharging, fuel injection system optimization and EGR. A potential of 10 to 15 percent increase in fuel economy may be achieved under the 1981 Federal emission standard with waiver of the NO_x level from 1.0 to 1.5 gms/mile along with particulates greater than 0.2 gm/mi

TABLE 3.7-1. EMISSION LEVELS OF DOMESTIC AND FOREIGN DIESEL POWERED PASSENGER CARS, LIGHT TRUCKS AND VANS

MANUFACTURER	ENGINE/VEHICLE SPECIFICATIONS						EMISSIONS GMS/MILE			FUEL ECONOMY MILES/GALLON		
PASSENGER CARS #1	MODEL	CID	HORSEPOWER	INTERTIA WEIGHT	XMISSION	REAR AXLE RATIO	HC	CO	NOX	URBAN	HIGHWAY	COMPOSITE
VOLKSWAGEN	RABBIT	90	48	2250	M4	3.30	0.78	1.00	0.61	50	64	55
	RABBIT	90	48	2250	M4	3.90	0.30	1.00	1.05	39	52	44
	RABBIT TC	90	70	2250	-	-	0.29	0.97	1.03	42.5		
	RABBIT TC	90	70	2250	M4	-	0.4	0.99	1.15	44.5	55.8	49
	RABBIT TC	90	70	2250	M5	-	0.28	0.7	1.3	51.4	63.4	56.2
	RABBIT TC	90	70	2750	M4	-	0.34	0.95	1.39	39.6	48.5	43.2
	RABBIT TC	90	70	3000	M4	-	0.37	0.95	1.29	39.4	47.3	42.6
PEUGEOT	204	83	51	2500	M4	4.06	1.11	1.71	0.68	36	44	40
FORD	PINTO	132	61	2750	M4	2.8	0.24	1.21	0.76	46	55	49
POSTAL VAN	PERKINS	165	-	3000	A3	3.07	0.14	1.47	2.54	30	38	33
PEUGEOT	504	141	71	3500	A3	3.78	0.66	1.20	1.21	25	32	28
	504	141	71	3500	A3	3.78	0.52	1.20	1.11	26	30	28
	504WA	141	71	3500	A3	4.11	0.60	1.30	1.05	26	31	28
	504WA	141	71	3500	A3	4.11	0.55	1.30	1.17	24	30	27
	504	141	71	3500	M4	3.70	0.89	1.60	1.01	28	35	31
	504	141	71	3500	M4	3.70	0.96	2.10	1.03	28	34	30
	504WA	141	71	3500	M4	4.11	0.91	2.00	.96	27	35	30
	504WA	141	71	3500	M4	4.11	0.70	1.80	1.02	27	32	29
MERCEDES BENZ	240	147	62	3500	A4	3.69	0.11	1.0	1.74	24	31	27
	240	147	62	3500	M4	3.69	0.19	1.4	1.63	26	34	29
							0.14	0.8	1.44	27	34	30
	300D	183	-	3500	M4	-	0.23	1.43	1.55	-	-	26.8
	220D	134	-	3500	M4	-	0.22	1.0	1.4	-	-	27.6
	220D (DI) conversion	134	-	3500	M4	-	0.27	1.25	1.4	-	-	33.2
OPEL	Record	126.2	-	3000	M4	-	0.57	1.25	1.45	-	-	32.9
	Record	126.2	-	3000	M4	-	0.39	1.21	1.29	-	-	26.8
DATSON		122.4	-	3500	M4	-	0.35	1.89	1.96	23.3	-	-
NISSAN	22C	132	70	3500	M4	3.91	0.25	1.1	1.32	24	27	29
MERCEDES BENZ	300	183	110	4000	A4	3.07	0.17	0.8	2.04	24	29	26
		183	77	4000	A4	3.46	0.10	1.2	1.86	22	28	25
GENERAL MOTORS	OLDSMOBILE (1)	350	-	4000	-	-	0.45	1.54	1.72	18.9	-	-
	DELTA	350	120	4500	A3	2.41	0.64	1.50	1.62	21	30	24
	CUSTO	350	120	5000	A3	2.73	1.08	1.80	1.60	19	27	22
<u>LIGHT TRUCKS</u>												
INTERNATIONAL HARVESTER	PICKUP	247	105	4500	M4	3.7	0.72	0.97	1.50	26	28	27
GENERAL MOTORS	PICKUP	350	120	4500	A3	2.76	0.88	1.80	1.56	21	28	24
	PICKUP	350	120	5000	A3	2.76	0.80	1.70	1.55	20	27	22
	PICKUP	350	120	5000	A3	3.40	0.76	1.60	1.79	20	25	22

TC = Turbocharged

Source: Reference 1

TABLE 3.7-2. CURRENT STATE OF ART TECHNOLOGY IMPACT ON EXHAUST EMISSION LEVELS

Engine	HC	CO	NO_x	Particulate	Estimated Relative Fuel Economy Changes
N.A. Present solution	>0.41	<3.4	>2.0	<0.6	-
Fuel Injection Optimization (F.10)	<0.41	<3.4	<2.0	<0.6	0
N.A. + F.I.O. + Vehicle Modification (V.M.)	"	"	"	"	+10%
N.A. + F.I.O. + V.M. + E.G.R.	>0.41	<3.4	<1.0* <1.5**	>0.6	+6%
Turbocharged (T.C.) + F.I.O + V.M.	<0.41	<3.4	<2.0	<0.20	+15 to 20%
T.C. + F.I.O + E.G.R.	<0.41	<3.4	<1.0* <1.5**	>0.20	+10 to 15%

*For Inertia Weight less than or equal to 2,500 lbs.
**For Inertia Weight ranging from 2750 to 3,500 lbs.

SOURCE: Reference 5.

by proper adjustments based on electronic control of turbocharging, EGR, injection timing and fuel injection system optimization (the effect of EGR on the durability of the engine is unknown).

One study, Figure 3.7-1, conducted for EPA by Ricardo Consulting Engineers, shows that a 10 percent loss in fuel economy is incurred in meeting the 0.41/4.4/1.0 HC/CO/NOx emission standard when injection retard in combination with modulated cold exhaust gas recirculation is used as a control technology on a 300 D turbocharged Mercedes diesel. The study also concluded that EGR increases particulates, here by 200 percent when the engine's injection timing was not changed. The turbocharger in this study was of low boost and was used principally to compensate for the loss of air caused by the diluted exhaust stream to 300 D naturally aspirated engine. Similar fuel economy losses were measured by Volkswagen in its study for DOT.

The diesel engine also emits a number of higher molecular weight hydrocarbon gaseous species and liquid organic materials which are bonded to the carbon-like particles of the particulate matter. Some of these components are suspected carcinogens. They are a major health concern.

3.7.1 Regulated Emissions (HC/CO/NOx)

Three classes of light-duty vehicle exhaust emissions are currently under regulation. These include oxides of nitrogen (primarily NO and NO_2) designated as NO_x; carbon monoxide (CO); and unburned or partially burned hydrocarbons (HC). Both CO and NO_x are extremely toxic gases. The major reason for regulation of exhaust NO_x, and HC is the role each plays, even after dilution in ambient air, in the development of photochemical smog.

Current diesels emit relatively small quantities of carbon

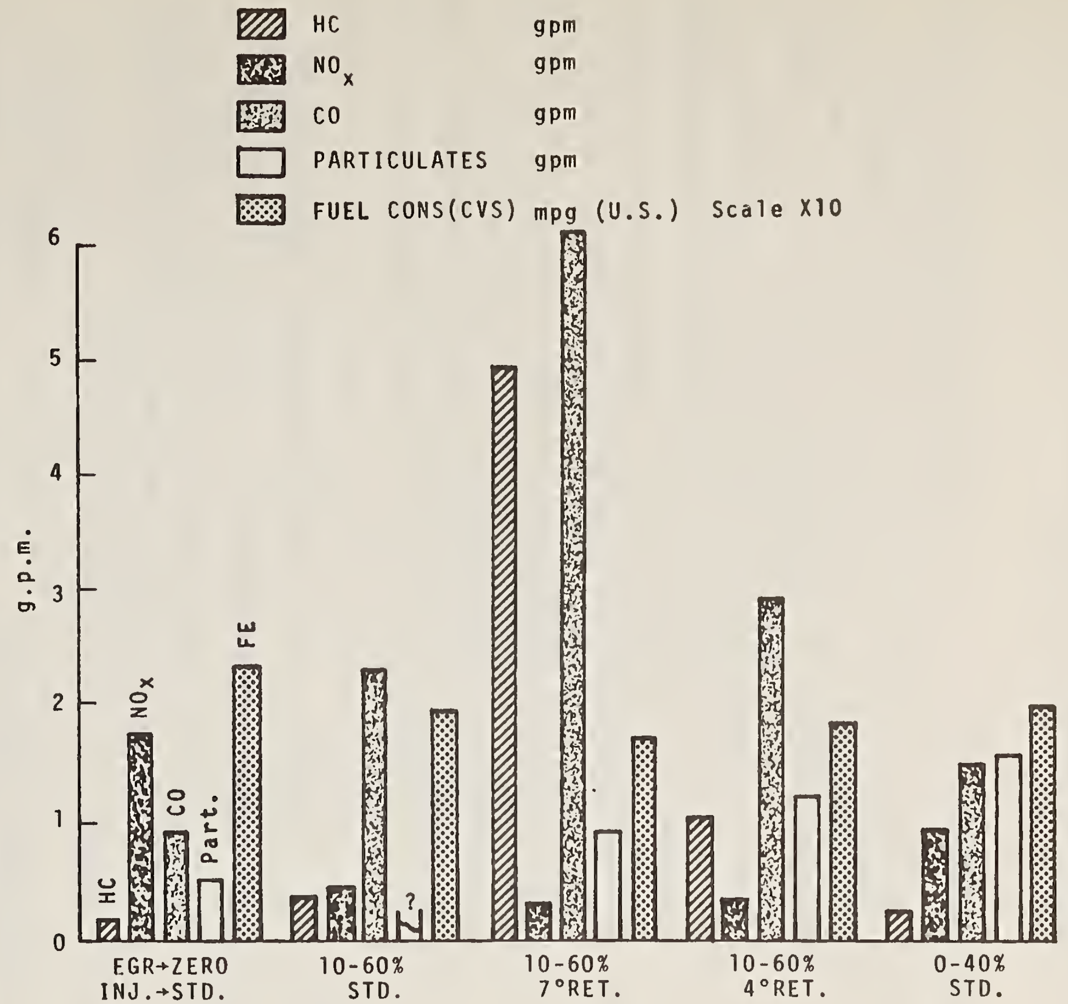

FIGURE 3.7-1. MERCEDES 300D T/C EGR RESULTS

monoxide and hydrocarbons. These emissions are achieved without exhaust after-treatment devices as used in gasoline powered vehicles. Nitrogen oxide emissions levels range from 0.61 to 2 gram/mile with lighter weight vehicles emitting lower NO_x levels. Variations do exist which are mainly attributed to vehicle and engine design characteristics.

3.7.1.1 Oxides of Nitrogen

Most of the nitrogen oxides in the exhaust are in the form of NO. On the average, about 10 percent of the NO_x appears as NO_2.[2] However, under some operating conditions, especially during idle, larger amounts of nitrogen dioxide may be emitted. Human exposure to NO_2 can result in respiratory irritations and nausea. The peak NO_x concentration in diesel engine emissions is lower than that for gasoline engines, mainly because of the lower mass average temperatures reached in the diesel combustion process.[2] Also, it has been suggested that nitrogen bonded in the fuel can contribute to the formation of nitrogen oxides.[2] Since the rate of formation of nitric oxide is dependent on temperature, reduced peak cycle temperatures, during the diesel's combustion phase, reduce the nitrogen oxide (NO_x) levels. Reduction in NO_x formation can also be achieved by lower oxygen concentrations or reduced residence times. Unfortunately, due to subsequent in-omplete combustion, these reduce engine efficiency. Thus, it is extremely difficult to reduce NO_x emissions to very low levels in the engine without sacrificing engine power.

3.7.1.2 Carbon Monoxide, CO

Diesel carbon monoxide levels [2] are low since carbon monoxide is usually formed as an intermediate product of a multiphase combustion process which is not favored in the diesel under normal running conditions. As combustion proceeds to completion, oxidation of CO to CO_2 occurs through recombination reactions between CO and the different oxidants.

If these recombination reactions are incomplete because of a lack of oxidants or low gas temperatures, CO will result.

Carbon monoxide emissions in indirect injection engines (I.D.I.) are less, in general, than in direct injection engines (D.I.).

3.7.1.3 Hydrocarbon Emission, HC

Hydrocarbon concentrations in current automobile diesel emissions are low but might be increased by measures to reduce NO_x[2]. The unburnt hydrocarbons in diesel emissions are caused primarily by incomplete combustion of the fuel near relatively cooler surfaces consisting of either original or decomposed fuel molecules or recombined intermediate compounds. A portion of these hydrocarbons originate from the lubricating oil. HC emissions increase[2] under conditions of cold start, severe retardation of injection timing, idle and light load operation. Under those circumstances misfire may occur, resulting in larger quantities of unburnt fuel exhausting from the main chamber.

Hydrocarbon emissions[2] in I.D.I. engines have been reported lower than in D.I. engines because of the more effective mixing which results from the flow of the combustion products from the prechamber to the main chamber.

Hydrocarbon emissions problems can be partially solved by catalytic exhaust treatment which also provides reduction in CO, aldehydes, and odor intensity. Presumably, catalysts are more efficient at high temperature conditions occurring under heavy engine load operations. Prolonged idle and light load operations lead to cooler exhaust temperature, resulting in ineffective catalytic treatment. Platinum monolithic exhaust catalysts tend to increase the sulfate emission.

3.7.2 Unregulated Emissions

There are diesel exhaust emissions that are not currently regulated but which may be potentially harmful from a health and/or esthetic standpoint, and control thereof may become essential. These emissions include visible smoke and fine carbon particles, sulfur oxides, aldehydes, and polyneuclear aromatics (PNA). Of these, the polyneuclear aromatics are of major concern because of suspected carcinogenicity. The bulk of PNA from diesels is believed to be adsorbed in the carbon particles. Another concern is the general nastiness of diesel exhaust related basically to vapor irritants and odor.

3.7.2.1 Particulates

Diesel exhaust particulates are considered anything that can be collected on Type A glass fiber filtering media, at a temperature not to exceed 125°F, excluding condensed water.

Diesel exhaust particulate is a complex mixture composed of carbon particles and associated organic and inorganic compounds such as partially oxidized and non-oxidized fuel particles and inorganics.

Diesel particulates are a major concern today because:

a. these particles contain compounds which are detrimental to human health

b. their size distribution is such that it enables them to reach the inner regions of the human lungs.

c. it has high specific surface, enabling it to carry adsorbed molecules into respiratory tissue at many orders of magnitude higher than ambient concentration.

Different types of particulates are emitted from diesel engines under different modes and operating conditions. These particulates can be divided into the following:

a. Liquid particulates appear as white/blue clouds of vapor emitted under cold starting, idling, and low loads. These consist mainly of fuel and a small portion of lubricating oil[2] emitted without combustion; they may be accompanied by partial oxidation products. The blue/white clouds disappear as the load is increased and the cylinder walls become warmer.

b. Soot or black smoke is emitted as a product of the incomplete combustion process, particularly at high load conditions.

c. Other particulates include lubricating oil and fuel additives.

All current production diesel engines emit about twenty-five times the quantity of particulates as gasoline vehicles operating on unleaded fuel equipped with catalysts. (See Tables 3.7-3). The percentage of organic solubles in the particulate (Table 3.7-4) shows values for the two prototype diesels, Cutlass and Rabbit, ranging from 12 to 22 percent. This indicates relatively dry exhaust particulate, free from excessive unburned fuel, oil and aerosol-like matter. The turbocharged Rabbit diesel exhibits lower particulate levels compared to the naturally aspirated version.

Adding EGR increases particulates levels as well as smoke during transient operations.

Numerous techniques are under discussion in regard to reduction of particulates. (See Table 3.7-5). Of these, a continuous filtering after-burning system may appear as one practical means to reduce particulate emissions (by 30 percent). More novel approaches have been discussed but appear at this time to be too premature to speculate about. (A plasma gun which prevents soot particles from coagulating during early stages of combustion by introducing ionized nitrogen and/or argon to electrically charge the particles.)

Although smoke is a regulated emission for heavy-duty diesel engines, there are no regulations covering smoke from light-duty diesel engines. Currently, passenger car diesel engines are set at a maximum fuel throttle for Bosch smoke levels in the range of

TABLE 3.7-3. PARTICULATE EMISSION RATES OF LIGHT-DUTY DIESEL VEHICLES

Vehicle/Model	Engines Disp. (CID)	Inertia wt (lbs)	Test Lab	Particulates g/mile		
				FTP	SET	HFET
1. Mercedes 220D	134	3500	SwRI	0.60	0.43	0.38
2. Mercedes 240D	146	3500	SwRI	0.48	0.36	0.31
3. Mercedes 300D	183	4000	SwRI	0.49	0.37	0.39
4. Peugeot 204D	83	2500	SwRI	0.38	0.24	0.30
5. Perkins 6-247	247	4500	SwRI	0.81	0.49	0.54
6. VW Diesel	90	2250	SwRI	0.29	0.26	0.25
6a. VW Gasoline*	?	?	SwRI	0.007	0.002	0.003
7. Oldsmobile Diesel	350	4500	SwRI	0.92	0.58	0.48
7a. Oldsmobile Gasoline*	260	4500	SwRI	0.009	0.016	0.021
8. Pinto Diesel	?	2750	AA	0.35	0.38	0.28
9. Postal Van Diesel	165	3000	AA	0.47	0.27	0.24
10. VW Rabbit Diesel	90	2250	AA	0.32	0.23	0.26
11. Chrysler Diesel	200	4500	AA	0.19	0.17	0.17
12. VW Rabbit Diesel	90	2250	AA	0.31	0.20	0.20
13. Mercedes 300D	183	?	AA	0.43	0.27	0.25
14. Oldmobile Diesel	350	4500	AA	?	0.42	0.39
15. Nissan Diesel	122	3500	RTP	0.3	?	0.33
16. Peugeot 504	129	3000	RTP	0.51	.31	0.40
17. Mercedes 220D DI Conversion	134	3500	PUL	?	?	0.34

*Gasoline car included for comparison to diesel counterpart

TABLE 3.7-4. PARTICULATE ORGANIC SOLUBLES

VEHICLE	TEST	PARTICULATES (grams/mile)	ORGANIC SOLUBLES[e] % OF PARTICULATE
Cutlass Diesel[c]	1975 FTP (Urban Drive Cycle)	0.92	13.3
	FTP COLD	1.01	12.4
	FTP HOT	0.84	14.0
	SET[a]	0.58	18.2
	FET[b]	0.48	19.0
	1975 FTP[d]	0.92 $\pm$0.09	39.0[e]
Cutlass Gasoline[c]	FET	0.022	10.7
Rabbit Diesel[c]	1975 FTP	0.29	21.4
	FTP COLD	0.33	20.2
	FTP HOT	0.27	22.3
	SET	0.26	15.4
	FET	0.25	13.0
Turbo-[d] Rabbit	1975 FTP	0.18 $\pm$ 0.01	16.9[e]
Rabbit Gasoline[c]	FTP HOT	0.0025	42.3
Nissan[d]	FTP	0.39$\pm$0.02	14.6[e]

(a) Sulfate emission test by congested freeway cycle

(b) Federal 1975 highway fuel economy test

(c) Source: Reference 1

(d) Values represent cold filter extractible, % of THC (sum of THC HOT FID and the hot filter extractibles hydrocarbon) Source: Reference 4

(e) Extraction methodology and solvent system not explicitly defined in References c and d.

TABLE 3.7-5. METHODS TO REDUCE DIESEL EMISSIONS

Fuel Modification

a. Fuel-water emulsion decreases NO_x 50%, decreases particulates 30% but increases hydrocarbons 50%.

b. Water injection decreases NO_x 10 to 20%.

c. Reduction of fuel sulfur content, NO_x reduced pro rata with sulfur.

d. Fuel additives--small effect.

e. Methanol aspiration - NO_x increases 10%, CO increases 30%, particulates/hydrocarbons decrease 50%.

f. Methanol emulsification increases NO_x 50%, increases hydrocarbons 50%, but reduces particulates 80%.

Engine Modification

a. Turbocharging (Comprex, Sulzer-Switzerland system) reduces particulates 50%.

b. Control of injection-spray quality and timing provides improvements in all emissions.

c. Thermal isolation of cylinder walls and prechambers reduces PNAs, hydrocarbon odors, irritants and other oxidizable compounds.

d. Catalytic in-cylinder combustion reduces all pollutants other than NO_x.

e. Thin-metal catalysts as integral parts of cylinder head and exhaust ports reduces CO 95%, reduces HC 90% (better performance at light loads.)

Exhaust Treatment

a. Use of standard catalysts (monoliths and pellets) reduces CO 90%, reduces hydrocarbons 80%.

*b. Basic water scrubber (optimized) using pumped water sprayers reduces SOx 95%, reduces particulates less than 30%.

c. Water scrubber with Venturi pressure reduction (mist reduction water retention) reduces SO_x 95%, reduces particulates less than 30%.

d. Same as c above but with manifold water injection for condensation scrubbing reduces SO_x 95%, carbon monoxide 90%, hydrocarbon 90%, particulates 80% and NO_x 20%.

e. Particulate control by hot filter (e.g., spiral manifold reactors with or without catalyst loading) reduces CO 95%, HC 95%, and particulates 30%.

*f. Control by cold filters (fabric or fiber with a form of plug-in cartridge for 1 shift use) reduces particulates 90%, HC 50%.

*g. Same as f above but with continuous water irrigation for self-cleaning nonreplaceable elements reduces particulates 90%, HC 30%.

h. Agglomerations of thermal precipitators with induced soot breakaway and cyclone collection reduces particulates 50%, HC 30%.

i. Emission control by exhaust feedback sensing (e.g., soot by opacity or electronic sensing, CO by infrared sensing or O_x by Zirconia dector): useful for malfunction/warning detection and transients.

j. Exhaust Gas Recirculation (EGR) uncooled reduces NO_x 50%.

k. EGR, cooled, reduces NO_x 50%, particulates +7, HC +7.

l. Catalytic conversion of NO to NO_2 with water scrubbing using alkaline solutions or organic agents, limestone, etc., reduces NO_x 25%.

m. Ozone oxidation of NO to NO_2 and scrubbing in m above reduces NO_x 50%.

n. Electrostatic precipitators and filters composites: unknown efficiency and durability.

*Note: These and some of the other methods of emission reduction mentioned in this list would probably not be practical or economic for light surface vehicle use.

SOURCE: Reference 2.

3.0 to 3.5 under steady state operating conditions which are just visible. Typical smoke opacities comparable to the heavy duty smoke test range from 5.7 to 27.0 on the acceleration mode, 3.3 to 37.8 on the lug down mode and 9.3 to 29.8 on the peak mode.

Figure 3.7-2 summarizes resulting smoke-NO_x tradeoffs acquired under single cylinder tests for two diesel combustion systems (D1 and IDI). Of the fourteen parameters examined, IDI and water injection exhibited similar trends in control of smoke and NO_x, while smoke typically increased when NO_x was reduced. At present, the heavy duty truck smoke limits are 20, 15 and 50 percent opacity on the lug down, acceleration and peak mode cycles, respectively. While aesthetically unpleasant, smoke remains unquantified as to its health effects and relationship to other particulate emissions.

3.7.2.2 Sulfates

Sulfate emissions from automobiles equipped with diesel engines are within the range of sulfate emissions from automobiles equipped with gasoline engines. Typical diesel fuel contains about eight times the sulfur level of typical gasoline. The sulfate emissions from diesels correspond to only one to two percent of the diesels fuel.[7] Because sulfate emissions tend to be in proportion to fuel sulfur level a reduction in the fuel sulfur level of diesel fuel would reduce sulfate emissions.

SO_2 is almost totally derived from the fuel since a little of it comes from the oil. The amount of SO_2 can be calculated from the sulfur content of the fuel. Virtually all the fuel sulfur is oxidized to SO_2 when fuel is burned in an internal combustion engine.[7]

3.7.2.3 Benzene

Benzene emissions from automobiles equipped with diesel engines are about the same as the benzene emissions from automobiles equipped with gasoline engines. Recent studies of the potential carcinogenic effect of benzene have caused a renewed interest in benzene emissions from automobiles, although the need for control of benzene emissions from automobiles has not yet been determined.

CAUTION

- Test engine not optimized, results not strictly representative of production engines.
- % changes are relative to arbitrarily selected baseline --this allows numerical values, of ΔY/Y to exceed 100%.
- Emissions variations were non-linear in most cases; a linear fit was imposed.
- Most coefficients averaged over several cases involving distinct combinations of speed, load. etc.

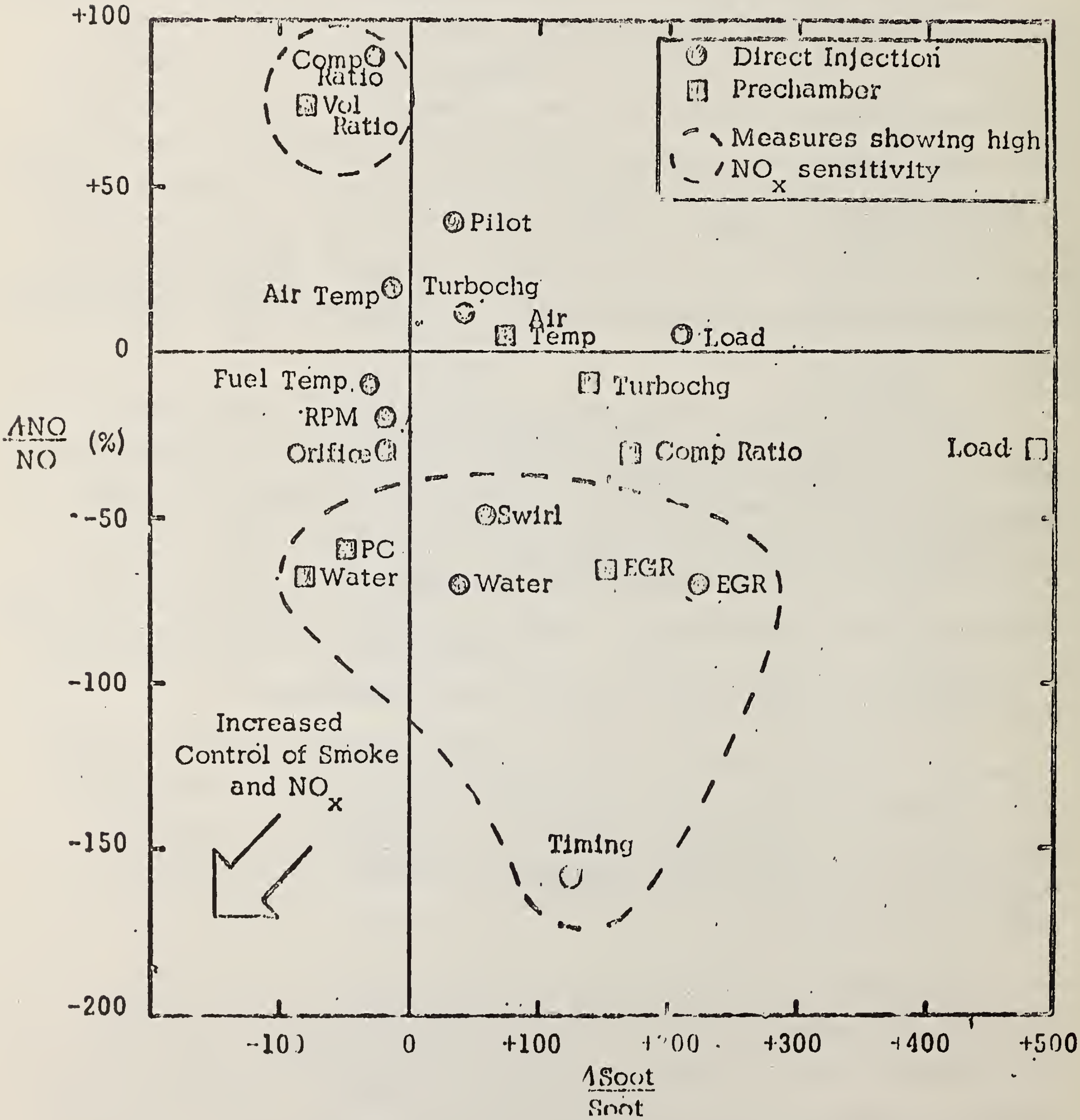

FIGURE 3.7-2. EMISSIONS SUMMARY ILLUSTRATING THE SOOT-NO TRADEOFF

3.7.2.4 Odor

Diesel exhaust[3] contains very small amounts of a large number of unburned partially oxidized HC compounds. Some of these compounds have strong odors even at low concentrations. Odors from these, enhanced by odors from other materials, combine to produce the "typical" diesel exhaust odor. Two types of odors are important. One is an "oil-kerosene" odor which appears to be caused by certain classes of aromatic hydrocarbons. The second is a "smoky-burnt" odor which appears to be caused by partially oxidized material, which may also have an aromatic structure.

Diesel engine exhaust generally has noticeable odor and, like smoke, is considered a nuisance pollutant. Definitive relatioships between auto exhaust odor and health effects have not been determined nor has the impact of increased diesel usage on odor in urban areas been forecast.

Odor intensity is comparatively low when the engine is warm and is greatest at medium temperatures occurring during warm-up. The most promising means of odor reduction are reduction of HC emitted by the engine.

With detailed developments, diesel odor level could be significantly reduced, even to that of gasoline engines. The ultimate solution, however, will probably come from some kind of exhaust conditioner.

Fuel odor may also arise as a result of bad housekeeping since diesel fuel does not readily evaporate. It is essential that there be no leaks from the fuel system and no spillage when re-filling fuel tanks if odor from this source is to be avoided.

3.7.2.5 Polynuclear Aromatic Hydrocarbons (PNA)

PNA emissions are though to be related to the combustion process and to some extent to the fuel. Some PNA's are carcinogenic and consequently there is some concern from a health standpoint, for example, BaP.

3.7.2.6 Aldehydes

Aldehyde emission levels are somewhat higher in exhaust from diesel automobiles than in exhaust from modern, catalyst equipped gasoline automobiles. Aldehydes may participate in photochemical reactions to form smog, and some have distinctive odors.

The aldehydes are partially oxidized hydrocarbons and consist mainly of formaldehyde, higher aliphatic aldehydes, and aliphatic ketones. A controlled inlet burner is one means of reducing aldehyde emissions, in addition to an after burner treatment device.

3.7.2.7 Noise

Diesel powered vehicles characteristically produce higher exterior noise levels during idle than carbureted gasoline engines by approximately 5dB(A) as indicated in Table 3.7-6 through 3.7-8. This difference is due primarily to the diesel form of combustion and consequently pressure loading where the lower and mid range frequencies of noise are emphasized. This results in higher levels of engine radiated noise. External passby noise levels are slightly higher compared to those of the gasoline powered vehicles. (The difference is larger compared to U.S. produced gasoline engines and smaller compared to European gasoline engines, since the latter gasoline engines are generally of the high speed type.) At road speed, the noise from both gasoline and diesel powered vehicles may be dominated by tire noise. However, studies conducted by Ricardo Consulting Engineers* showed that the major source of noise for a Puegeot 504D vehicle at a constant road speed of 30 miles and during maximum acceleration is caused by the engine itself. Other noise sources caused by the fan and exhaust system were also identified but they contributed a lesser degree. The latter noise sources were found to be dependent on the driving mode of the vehicle. Passenger compartment noise levels in diesel powered vehicles are comparable to those of vehicles powered by gasoline for the three operating modes. Noise transmitted by the diesel engine to the passenger compartments can be attenuated by

*DOT-TSC-1242.

TABLE 3.7-6. DIESEL VEHICLE EXTERIOR NOISE LEVELS*

VEHICLE	LABORATORY	NOISE LEVEL dB(A) (RE: 0.0002 MICRO-BARS)
PEUGEOT 504	RICARDO[6]	77.7 (LHS)** (FAN OFF) 77.0 (RHS) (FAN OFF)
PEUGEOT 504	EPA	70.8
CUTLASS OLDSMOBILE	SWRI[4]	73.8 (1st GEAR ACCELERATION)
CUTLASS OLDSMOBILE (GASOLINE)	SWRI	68.8 (1st GEAR ACCELERATION)
VOLKSWAGEN RABBIT	SWRI	71.5
VOLKSWAGEN RABBIT (GASOLINE)	SWRI	71.0
50 HP (NAUTRALLY ASPIRATED)	VOLKSWAGEN	75.0
50 HP (NA WITH EGR)	VOLKSWAGEN	71.0
66 HP NATURALLY ASPIRATED)	VOLKSWAGEN	76.5
70 HP (TURBOCHARGED)	VOLKSWAGEN	73.5
70 HP (TURBOCHARGED WITH EGR)	VOLKSWAGEN	69.0
MERCEDES BENZ	EPA	77.0
OPEL RECORD	EPA	68.0

*Noise Levels Based on SAE J986a Pass-by Acceleration Measurement Procedure

**LHS Means Left Hand Side
RHS Means Right Hand Side

TABLE 3.7-7. DIESEL VEHICLE INTERIOR NOISE LEVEL

VEHICLE	LABORATORY	NOISE LEVEL dB(A) RE: (0.0002 MICRO-BARS)
PEUGEOT 504	RICARDO[6]	
PEUGEOT 504	EPA	78.0
CUTLASS OLDSMOBILE	SWRI[4]	74.2 (BLOWER ON) 70.5 (BLOWER OFF)
CUTLASS OLDSMOBILE (GASOLINE)	SWRI	71.5 (BLOWER ON) 70.5 (BLOWER OFF)
VOLKSWAGEN RABBIT	SWRI	71.8 (BLOWER ON) 68.0 (BLOWER OFF)
VOLKSWAGEN RABBIT (GASOLINE)	SWRI	73.5 (BLOWER ON) 70.5 (BLOWER OFF)
VOLKSWAGEN 50 HP (NATURALLY ASPIRATED)	VOLKSWAGEN	75.0
50 HP (NA WITH EGR)	VOLKSWAGEN	72.5
66 HP (NATURALLY ASPIRATED)	VOLKSWAGEN	72.0
70 HP (TURBOCHARGED)	VOLKSWAGEN	72.5
70 HP (TURBOCHARGED WITH EGR)	VOLKSWAGEN	70.5
MERCEDES BENZ	EPA	74.0
OPEL RECORD	EPA	73.0

TABLE 3.7-8. DIESEL VEHICLE IDLE NOISE LEVEL

VEHICLE	LABORATORY	INTERIOR NOISE LEVELS (RE: 0.0002 MICRO-BARS)	EXTERIOR NOISE LEVELS (RE: 0.0002 MICRO-BARS)
PEUGEOT 504	RICARDO [6]		
PEUGEOT 504	EPA	52.0	68.0
CUTLASS OLDSMOBILE	SWRI [4]	71.0 (BLOWER ON) 51.5 (BLOWER OFF)	70.0 (AT 3.05 METERS)
CUTLASS OLDSMOBILE (GASOLINE)	SWRI	71.5 (BLOWER ON) 48.5 (BLOWER OFF)	64.5 (AT 3.05 METERS)
VOLKSWAGEN RABBIT	SWRI	69.5 (BLOWER ON) 58.0 (BLOWER OFF)	67.0 (AT 5.05 METERS)
VOLKSWAGEN RABBIT (GASOLINE)	SWRI	69.5 (BLOWER ON) 58.0 (BLOWER OFF)	72.5 (AT 3.05 METERS)
50 HP (NATURALLY ASPIRATED)	VOLKSWAGEN	-	78.5 (AT 0.5 METERS)
50 HP (NA WITH EGR)	VOLKSWAGEN	-	73.0 (AT 0.5 METERS)
66 HP (NA)	VOLKSWAGEN	-	75.0 (AT 0.5 METERS)
70 HP (NATURALLY ASPIRATED WITH EGR)	VOLKSWAGEN	-	75.0 (AT 0.5 METERS)
70 HP (TURBOCHARGED WITH EGR)	VOLKSWAGEN	-	76.0 (AT 0.5 METERS)
MERCEDES BENZ	EPA	50.0	66.0
OPEL RECORD	EPA	53.0	72.0

proper acoustic materials. In general, noise treatment of vehicles is presently designed to have no direct effect on fuel economy or emissions. For noise levels more stringent than 70 dB (A), barriers, mats, insulators and larger silencers may be required which contribute directly to the weight of the vehicle and hence its fuel economy.

The Calspan Corporation under contract to DOT-TSC* will provide data on changes in weight, penalty in fuel economy and cost of noise control systems as a function of noise reduction. Other techniques, for example, recirculation of exhaust gases and turbocharging, have shown that a 2 to 3 dB(A) reduction of the noise level can be achieved.

Presently, there are no federal noise standards in effect for light duty passenger cars or trucks. The EPA is currently developing an urban acceleration noise test procedure and conducting studies directed towards the identification of noise levels. This procedure consists of simulating a community driving situation which entails part throttle low speed accelerations under drive away conditions for an acceleration of 0.15 g's or steady state speed of 25 mph over a distance of 10 feet.

Most manufacturers operate on a voluntary noise standard and acoustically design their vehicles to meet the wide open throttle passby (SAE-J986a) noise levels 2 to 3 dB(A) below the noise standards enacted by such cities as Chicago, Illinois, and Los Angeles, California. The most stringent standard presently calls for light duty vehicle passby noise levels of 77 - 78 dB(A) with progressing of values in time to a future level of 70 dB(A).

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*DOT-TSC-1242.

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