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POTENTIAL OF SPARK IGNITION ENGINE, 1979 SUMMARY SOURCE DOCUMENT

by

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16. Abstract

This report provides an assessment of the potential for spark ignition engines passenger cars and light trucks. The assessment includes:

- o tradeoffs between fuel economy and emissions
- o improvements in spark ignition engine efficiency
- o improvements in engine parasitics
- o improvements due to transmissions
- o effect of aerodynamic drag and tire rolling resistance on fuel economy
- o effect on performance and fuel economy of weight and axle ratio
- o lubricant improvements
- o impact of fuels
- o noise considerations.

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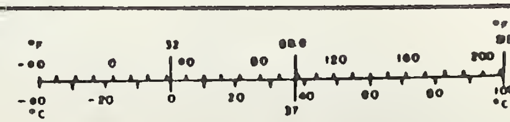
METRIC CONVERSION FACTORS

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



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1. INTRODUCTION

1.1 GENERAL

This report summarizes an assessment of the potential for spark ignition engines in passenger cars and light trucks. It contains test data derived from support contractors ans independent laboratory tests. It also contains engineering analyses utilizing engine control optimization analyses and vehicle simulations.

The information presented is compatible with the methodology employed by NHTSA's fuel economy projection program (FUEL 11). Required input to that program includes estimated percentage fuel economy improvements or reductions resulting from the following:

- o emissions/test procedures
- o engine technology
- o parasitics
- o transmissions
- o aerodynamics
- o tire rolling resistance
- o performance reduction
- o lubricants

To that end, the information required is included in this report as follows:

- o tradeoffs between fuel economy and emissions (Section 2)
- o improvements in spark ignition engine efficiency (Section 3)
- o improvements in engine parasitics (Section 3)

- o effect of aerodynamic drag and tire rolling resistance on fuel economy (Section 4)
- o effect on performance and fuel economy of weight and axle ratios (Section 4)
- o lubricant improvements (Section 5)

Additional information in this report not currently required for fuel economy projection, but useful as background for related technical questions are:

- o impact of fuels (Section 5)
- o noise considerations (Section 6).

1.2 SOME MAJOR FINDINGS

o Passenger car fuel economy at the 1981 emission standard of 0.41/3.4/1.0 ($HC/CO/NO_x$) grams/mile depends on vehicle inertia weight, engine configuration, warm-up response, control system strategy, and exhaust aftertreatment system efficiency. Current carbureted engines incorporating electronic control of EGR and spark timing along with a three-way catalyst in vehicles less than or equal to 3500 lbs. provide a potential increase in fuel economy up to 5 percent. This improvement diminishes to 0 percent for heavier vehicles. Present day production engines optimally calibrated for a oxidation catalyst do not provide the improvements obtained by engines operated with three-way catalyst systems. Further lowering of the emission standard to 0.41/3.4/0.4 ($HC/CO/NO_x$) gms/mile, provides a 4 percent loss in fuel economy for production engines with three-way catalyst systems.

o Light truck engines optimally calibrated for EGR, spark timing, and air-to-fuel ratio can attain a 10 to 20 percent improvement in fuel economy under the current emission standard of 1.7/18/2.3 ($HC/CO/NO_x$) gms/mile. This fuel economy improvement is maintained for light trucks equal to and below an inertia weight of 3000 lbs. at the proposed 1983 emission standard of

0.8/10/2.3 ($HC/CO/NO_x$) gms/mile. When compared to the current vehicles at current emission standard, light trucks of inertia weight greater than 3000 lbs. will suffer no penalty in fuel economy under the proposed 1983 emission standard. When compared to optimal possible fuel economies at current emissions standard, the optimum potential fuel economy of trucks heavier than 3500 lb. inertia weight decreases by about 10 percent. Part of this lower potential fuel economy is attributed to the inability of present oxidation catalysts to reduce HC emissions.

o Turbocharging offers a potential to increase fuel economy and provide improvements in emission characteristics. It provides two basic functions: the use of smaller engines to minimize throttling losses, and better preparation of the fuel/air charge. A number of concepts, which integrate turbocharging along with advanced engine and vehicle technologies, are under development. One prototype concept has demonstrated that a fuel economy of 33 miles per-gallon is technically feasible for a 3000 lb. inertia weight vehicle at the 1981 passenger car emission standard. Presently designed turbocharging is limited to spark ignition engines of 1.6 liters and larger.

o Approximately 18-20 percent of the energy supplied by the fuel to spark ignition engine is available to propel and overcome the inefficiencies of the vehicle. Over 50 percent is lost due to the engines basic cycle, incomplete combustion and real gas effects. An additional 26 percent is lost due to heat transfer, lower heating value of fuel, mechanical friction, pumping, and finite combustion. Some methods to reduce these losses include:

- increased turbulence of air-fuel mixture,
- use of leaner air-fuel mixtures,
- increased compression ratio,
- improved materials and oils,
- elimination of the throttle,

- closer tolerances on engine parts, and
- rolling contact bearings and rolling cam followers.

o Lean burn, high compression ratio, compact chamber spark-ignition engines hold promise to improve fuel economy at least 10 percent above currently employed U.S. traditional wedge chamber spark ignition engines. Limited data show that these engines require higher octane fuels and can meet the 1981 passenger car emission standard with an improved oxidation catalyst. Presently, no information is available on the potential of these engines to operate on a lower quality fuel, alternative fuels, or blends. Other lean burn spark-ignition combustion systems, frequently mentioned, include:

- bathtub or bowl-in-piston, and
- prechamber designs with torch ignition and/or dual ignition systems.

These configurations currently require further development to overcome the higher HC and fuel consumed during idle as well as improvements to reduce their power penalty and increase fuel consumption under wide open throttle conditions.

o Vehicle simulation studies utilizing prototype direct injection rotary stratified charge engine data have indicated a potential for fuel economy improvements over production homogeneous charge engines. One design, with preliminary test data, has shown about 20 percent improvement potential on the simulation.

o Various efforts are in progress to improve the lubrication efficiency in engines, notably by; a) upgrading conventional mineral oils, b) use of synthetic oils, and c) use of lubricating solids. On an average, 2-5 percent improvement in fuel economy can be anticiapted in the 1982-1985 time frame.

o Reduction of parasitic losses arising from a) air-conditioning; b) fans; c) alternators, and d) pumps, by means of constant speed accessory drive systems, presently holds promise to provide economy gains or about 4 percent.

2. ENGINE SYSTEMS OPTIMIZATION

2.1 BACKGROUND

The engine control strategy selected by an automobile manufacturer and the manner in which it is implemented have a powerful effect on both fuel economy and emissions.

Prior to 1975, much emphasis was placed by the manufacturers on meeting emission standards. During this period, emission control strategies were implemented which caused the fewest modifications to the existing hardware. These strategies included the use of: a) excessive retardation of spark timing, under lean mixtures to control hydrocarbon (HC) and carbon monoxide (CO) emissions, and b) engines of low compression ratios with high levels of valve overlap and exhaust gas recirculation (EGR). EGR provides increased dilution for the control of oxides of nitrogen (NOx). The result was a decrease in engine efficiency and performance. Subsequent activities were directed towards: a) improved ignition systems, b) improved mechanisms for advancing spark timing at higher engine rotational speeds (vacuum advance, centrifugal advance), and c) temperature switches to activate spark timing at idle under cold or over-heated engine conditions to improve emission characteristics.

In 1975, fuel economy was improved with the introduction of oxidation catalysts (COC-Conventional Oxidation Catalyst), to reduce hydrocarbons and carbon monoxide emissions, and with further development of the exhaust gas recirculation (EGR) systems. Recently, more advanced spark timing logic calibrations have been incorporated which monitor several engine variables, including engine speed, derivatives of engine speed, throttle position, derivatives of throttle speed, intake air temperature, EGR rates, barometric pressure, and A/F ratio in a central control module. These spark timing logics permit a finer control of emissions under closed, part and wide open throttle, idle, and cold starting conditions. Typical systems which accomplish this function are the

Chrysler Lean-Burn System, General Motors' MISAR system, and the Ford EEC-I system. Spark timing in these systems is under an open loop control.

Present efforts are focused on the introduction of dynamic closed loop control features for fuel metering systems, and secondary air control for precise monitoring of engine characteristics and their emissions. In addition, the three-way catalyst (TWC), which is capable of reducing all three regulated emissions, is being introduced and will find widespread use in 1981. The TWC system requires the air-fuel ratio range of an engine to be stoichiometric (i.e., about 14.5:1). Further reduction of HC and CO emission can be accomplished with a COC catalyst downstream of the TWC with secondary air injection between the TWC and COC.

The next significant technology introduced has been electronic engine control systems. The importance of this technology to emissions and fuel economy has been its ability to control "key" engine variables (spark, EGR, A/F ratio, etc.) with a greater degree of accuracy, repeatability and speed. This improved control enables the fuel efficiency to be maximized for a given engine configuration and emission constraint.

Efforts to date have been confined to optimal control of three engine variables (spark timing, exhaust gas recirculation and air-to-fuel ratio). Various treatments have appeared which utilize mathematical formulation of a constrained optimization problem based on engine speed-load requirements for a given vehicle system. These approaches involve the use of steady state dynamometer data. One major drawback of these treatments is that they do not account for the variety of transient effects, e.g., cold start and catalyst light-off. Studies[1] are just beginning to appear that attempt to establish approaches to these transient effects under optimal control. Optimum speed-load control (i.e. transmission shifting and ratios) are key variables for future electronic controls.

2.2 PASSENGER CAR CONTROL STRATEGY

Tables 2-1 and 2-2 show the calculated improvements in fuel economy anticipated with electronic engine control systems for two cases, a) unconstrained for emission, and b) for the 1981 passenger car emission standard of 0.41HC/3.4CO/1.0NO_x gms/mile.

The results are reported for two control strategies. The first involves 3-parameter control, where air-to-fuel, spark timing and EGR are optimally established over the engine's operating range. A COC catalyst is employed with this strategy. The second control strategy involves the control of 2-parameters (EGR and spark timing). It incorporates a TWC for clean-up of the three regulated emissions, and, if required, a COC for further clean-up of HC and CO emissions.

The computations shown in Tables 2-1 and 2-2 were performed at TSC using the Engine Optimization Model reported in SAE-790179.[4] The experimental data for the 318 and 225 CID engines were acquired under contract with Chrysler.[2] The data in Table 2-2 showing the potential in fuel economy of 132 engine is a 2750 lb inertia weight vehicle was obtained under a contract with Fiat.[3] Fuel economy comparison shown in these tables was based on the composite EPA cycle with the exception of those reported by Chrysler which were evaluated for the EPA-urban cycle.

For the unconstrained emissions condition, the analysis predicts improvements greater than 10 percent in fuel economy with optimum 3-parameter control. For stoichiometric controlled engines (2-parameter control), the improvement is less. This is demonstrated by the +3 to +5 percent improvements shown in Table 2-1. It appears that these optimally calibrated engines do not, in general, exhibit large variations in improvement across the inertia weights considered. (The Chrysler data obtained under vehicle testing shows that the 318 and 225 engines controlled with a three way catalyst can provide optimally a 10 to 20 percent increase in hot cycle, urban fuel economy.) When the 1981 emission standard is imposed, optimal fuel economy is reduced as compared with the unconstrained emission condition. This is seen by

TABLE 2-1. OPTIMIZED FUEL ECONOMY (UNCONSTRAINED EMISSIONS)

	VEHICLE INERTIA WEIGHT (LBS)	ENGINE DISPLACEMENT (CID)	ENGINE CONTROL STRATEGY	BASELINE COMPOSITE FUEL ECONOMY (MPG)	FUEL ECONOMY (MPG)	UNCONSTRAINED ENGINE-OUT EMISSIONS HC/CO/NOx GMS/MILE	IMPROVEMENT (%)
CALCULATED					COMPOSITE		
	3000	121	3-parameter	29.0	32.5	1.01/1.84/5.65	+11
			2-parameter		30.4	1.57/5.87/1.96	+ 5
	3500	318	3-parameter	18.3	20.0	2.74/1.28/4.43	+9.5
			2-parameter		18.9	2.29/8.14/6.05	+2.7
	4500	305	3-parameter	18.2	20.6	4.0/3.75/3.3	+13
			2-parameter		18.75	2.6/41.0/3.2	+ 3
EXPERIMENTAL					WARM CYCLE URBAN		
	4000*	318	2-parameter	15.3	18.4	3.0/12.5/3.5	+19
	3500*	225	2-parameter	19.4	21.3	3.3/13.4/4.0	+10
				19.4	20.62	3.44/7.49/2.5	+ 6

*Source: Reference 2.

TABLE 2-2. OPTIMIZED FUEL ECONOMY (EMISSION STANDARD: 0.41/3.4/1.0 HC/CO/NO_x; GMS/MILE)

VEHICLE INERTIA (LBS)	ENGINE DISPLACE-MENT (CID)	REAR AXLE RATIO	ENGINE CONTROL STRATEGY	BASELINE COMPOSITE FUEL ECONOMY (MPG)	OPTIMAL COMPOSITE FUEL ECONOMY (MPG)	FUEL ECONOMY IMPROVEMENT (%)
2750*	132	3.58	2-parameter (3-way cat)	26.0	31.1	+20
3000	121	3.31	2-parameter (3-way cat)	29.0	30.3	+4.5
3000	121	3.31	3-parameter (3-way +O_x cat)	29.0	30.4	+5
3000	121	3.31	3-parameter (O_x cat)	29.0	29.7	+2
3500	318	2.47	2-parameter (3-way cat)	18.35	**	-
3500	318	2.47	2-parameter (3-way + O_x cat)	18.35	18.74	+2.1
3500	318	2.47	3-parameter (O_x cat)	18.35	**	-
4500	305	2.56	2-parameter (3-way + O_x cat)	18.2	16.7	-8
4500	305	2.56	3-parameter (O_x cat)	18.2	**	-

*Source: Reference 3
**Could not meet one or more of the emission standards.

examining the 3-parameter control calculations of Tables 2-1 and 2-2. If only 2-parameter control and lighter weight vehicles, (as shown in the 3000 and 3500 lb IW case), are considered, optimal fuel economy at the 1981 Federal Emission standard is the same as optimal fuel economy with unconstrained emissions. The 2-parameter control fuel economies for the heavier weight cars (4500 lb or greater) with present day catalysts are lower at the 1981 emissions standard as compared with the unconstrained emissions case. Note a 20 percent improvement in fuel economy is predicted for the 132 engine in a 2750 lb inertia weight vehicle. This calculation was performed by Fiat.[3] Part of this large increase is attributed to the poor control strategy of their base line vehicle.

These findings are supported by recent experiments conducted by General Motors with their electronically programmed engine control system (EPEC) which showed a 1 percent improvement in fuel economy with 2-parameter control for a 260 V8 engine in a 4000 lb IW vehicle calibrated for the 1977 emission standard.[5] The study showed that a 1.5 percent penalty is incurred over the baseline configuration when the engine is optionally calibrated for the 1981 emission standard of 0.41 HC/3.4 CO/1.0 NO_x gms/mile as shown in Table 2-3. General Motors attributes the penalty in fuel economy for the 1981 emissions standard to required spark retard, increased idle speeds and the addition of port air injection during the first 2 minutes. The increased air injection reduced the time required for the catalytic converter to reach light-off (260°F) from 220 seconds to 140 seconds. General Motors attempted to regain the loss in fuel economy resulting from the 1981 emission standard. One strategy provided a 2.5 percent improvement by recalibrating to allow the vehicle to operate at its best highway fuel economy mode. This strategy, however, caused the highway NO_x emissions to increase by approximately 90 percent. This exceeded the highway NO_x criteria of 1.22 times urban NO_x stipulated under the recent EPA guidelines.[7]

TABLE 2-3. EXPERIMENTAL OPTIMIZED PASSENGER CAR FUEL ECONOMY (EMISSION STANDARD: 0.41/3.4/1.0 HC/CO/NO_x; GMS/MILE)

VEHICLE INERTIA (LBS)	ENGINE DISPLACEMENT (CID)	REAR AXLE RATIO	ENGINE CONTROL STRATEGY	BASELINE COMPOSITE FUEL ECONOMY (MPG)	OPTIMAL COMPOSITE FUEL ECONOMY (MPG)	FUEL ECONOMY IMPROVEMENT (%)
4000*	260 V8	-	2-parameter	-	19.5	-
4000**	260 V8FI	2.56:1	2-parameter	-	21.0	-1.5
			Best Economy Mode***	-	21.5	+1.0

*Source: Reference 6

**Source: Reference 5

***The engine control strategy includes:
EFI Electronic Fuel Injection
EST Electronic Spark Timing
EEGR Electronic Exhaust Gas Recirculation
ISC Idle Speed Control.

The highway mode strategy includes A/F = 20:1 and 0% EGR.

The effect of a 0.4 NO_x standard on fuel economy is shown in Table 2-4 where predicted fuel economies are shown for NO_x constrained to 1.0 and 0.4 gms/mile under optimal controls. For the two cases analyzed, a 3 and 4 percent loss in fuel economy was calculated for the 318 and 232 CID engines respectively in 2750 and 3500 lb inertia weight vehicles.

Some calculated and vehicle tested fuel economy comparisons made by Chrysler under 2-parameter control are summarized in Tables 2-5 and 2-6 for the 318 CID-V8 in a 4000 lb car, and the 225 CID L6 in a 3500 lb car. Both the calculations and vehicle tests considered only the warmed-up portion of the EPA urban drive cycle. Four control strategies were reported: (1) the 1978 production engine strategy, (2) the minimum fuel consumption strategy, (3) constant #1, represented by reduction in NO_x emissions produced by the engine calibrated for minimum fuel by a factor of 2, and (4) emission constant #2 represented by a reduction in NO_x emissions of the minimum-fuel strategy by a factor of 3. Vehicle level test for the 318 CID in a 4000 lb IW vehicle shows that a 19 percent improvement in urban fuel economy is achieved over the Chrysler 1978 production engine when the engine was calibrated for minimum fuel. A lower percentage improvement in urban fuel economy, 9.8 percent, was tested for the 225 CID engine in a 3500 lb IW vehicle. The urban fuel economies are generally shown to decrease when the more rigorous emission strategies were considered.

Figure 2-1 presents steady state warm engine fuel economy and emission predictions made by Fiat for their 132 CID engine when a) spark advance is controlled, and b) when both spark advance and EGR are controlled along with a three-way catalyst. These data show the trend in urban cycle fuel economy for various constraints of hydrocarbon and oxide of nitrogen emissions. The positive influence of EGR on fuel economy is seen by comparing the solid with the dotted lines. It is believed that EGR is helpful in three ways. First, it reduces the requirements of throttling, secondly, it helps to increase the efficiency of the engine because of the higher specific heat content of the exhaust gases, and thirdly,

TABLE 2-4. EFFECT OF 0.4 GMS/MILE NO_x EMISSION STANDARD ON OPTIMIZED FUEL ECONOMY

	VEHICLE INERTIA (LBS)	ENGINE DISPLACEMENT (CID)	FUEL ECONOMY FOR NO_x EMISSION STANDARD		PERCENT DIFFERENCE FUEL ECONOMY DUE TO EMISSION STANDARD
			1.0 GM/MI	0.40 GM/MI	
2-Parameter	2750	232	31.1	30.0	-4%
	3500	318	18.7	18.0	-3.7%
3-Parameter	3500	318	18.8	*	*

*Could not meet NO_x emissions standard

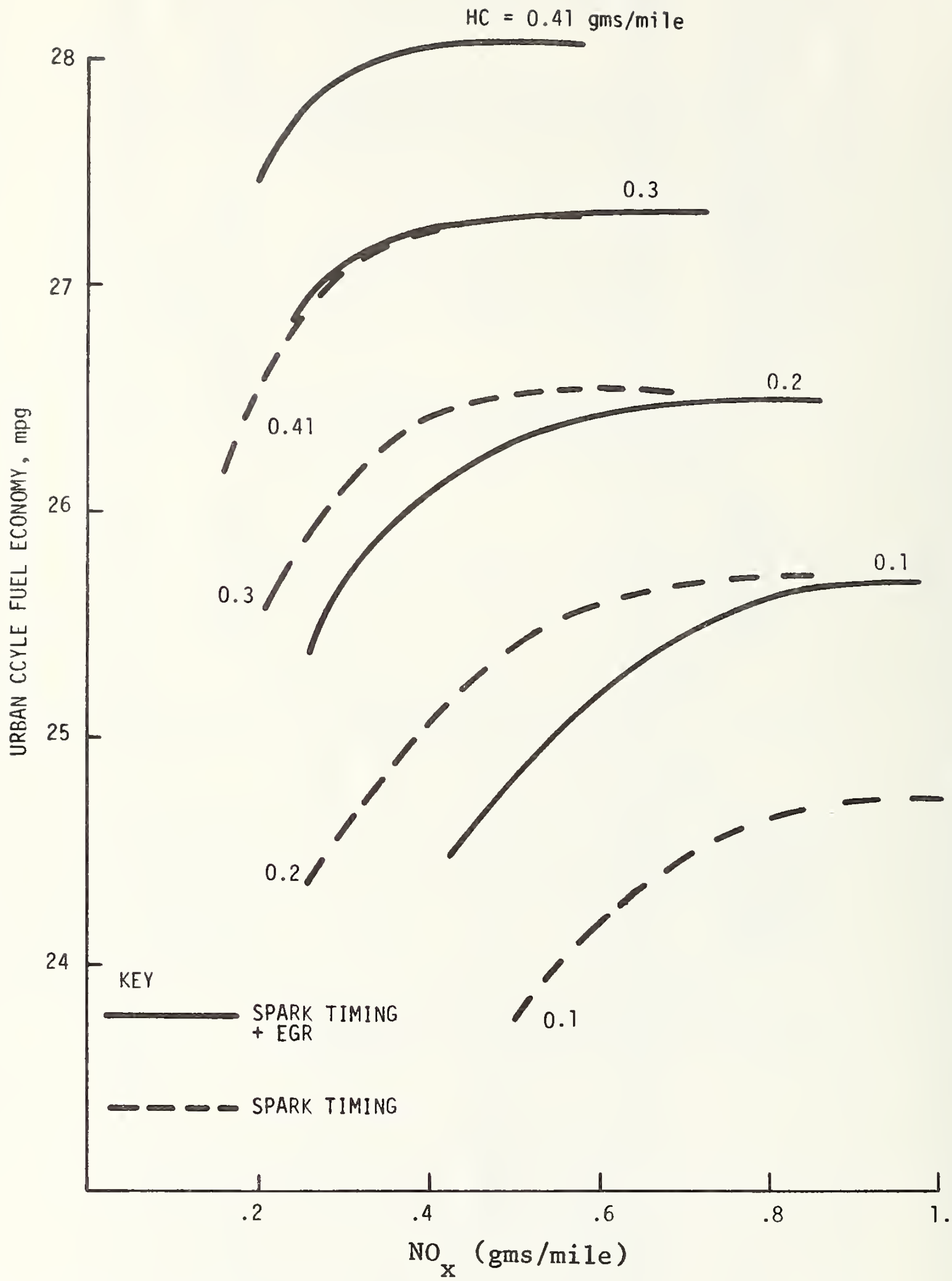
TABLE 2-5. EXPERIMENTAL RESULTS SHOWING IMPACT OF 2-PARAMETER CONTROL STRATEGIES ON FUEL ECONOMY AND EMISSIONS FOR 318 CID V8

318 CID V-8, 4000 LB IW, 2.47 REAR AXLE RATIO

STRATEGY	UNTREATED ENGINE EMISSIONS			TAILPIPE EMISSIONS			WARM URBAN CYCLE FUEL ECONOMY	% DIFFERENCE FUEL ECONOMY
	HC	CO	NO_x	HC	CO	NO_x		
'78 PRODUCTION	2.7	22.6	1.5	.3	4.6	1.4	15.5	BASE
MINIMUM FUEL CONSUMPTION	3.0	12.5	3.5	.2	2.3	2.0	18.4	+18.7
EMISSIONS CONSTANT #1	2.6	10.9	2.1	.2	1.7	1.3	17.1	+10.3

TABLE 2-6. COMPARISON OF CALCULATED AND EXPERIMENTAL RESULTS FOR 2-PARAMETER CONTROL OF 225 CID L6

225 CID L6, 3500 LB IW, 2.71 REAR AXLE RATIO										
	STRATEGY	UNTREATED ENGINE EMISSIONS			TAILPIPE EMISSIONS			WARM URBAN CYCLE FUEL ECONOMY	% DIFFERENCE FUEL ECONOMY
		HC	CO	NO_x	HC	CO	NO_x		
VEHICLE TEST	'78 Production	Not Available			.4	7.9	1.4	19.4	BASE
	Minimum Fuel Consumption	3.3	13.4	4.0	.4	3.1	2.2	21.3	+ 9.8
	Emissions Constant #1	3.3	13.3	3.0	.3	3.2	.7	20.8	+ 7.2
	Emissions Constant #2	3.9	14.7	1.6	.3	3.8	.4	21.4	+10.3
CALCULATED	Minimum Fuel Consumption	3.4	7.5	2.5	-	-	-	20.6	+ 6.2
	Emissions Constant #1	3.2	7.4	1.3	-	-	-	18.9	- 2.6
	Emissions Constant #2	3.3	9.7	0.9	-	-	-	18.0	- 7.2



Source: Reference 3

FIGURE 2-1. FIAT PROJECTED OPTIMAL FUEL ECONOMY it leads to an increase in inlet manifold temperature which promotes better atomization and homogeneity of the charge mixture. Although the effect of EGR closely demonstrates increase in fuel economy on the whole, it presently limits the true minimum NO_x levels which can be achieved (i.e., EGR tends to shift the curves to the left hand side in the mpg vs. NO_x plane).

2.3 LIGHT TRUCK CONTROL STRATEGY

Table 2-7 shows the calculated improvement in fuel economy anticipated for light trucks calibrated for the 1983 emission standard of 0.8/10.0/2.3 (HC/CO/NO_x) gms/mile under optimal electronic control of spark timing, air-to-fuel ratio and EGR.

Under the current emission standard, 1.7/18.0/2.3; HC/CO/NO_x; gms/mile, a 10 to 20 percent improvement in fuel economy is realized for current light truck and van vehicles equipped with current production spark ignition engines. The range of this improvement depends on the choice of engine and the type of options (rear-axle ratio, transmissions, tires, and bodies) packaged in a vehicle.

For the most part, the increase in fuel economy considered under an optimal electronic engine control strategy and current emission standard is that which is achieved when the engine is optimally calibrated without considerations of an emission standard. However, a slight loss in optimal fuel economy is incurred for the heavier vehicles.

When the emission standard is lowered to the 1983 levels, the 10 through 20 percent improvement in fuel economy achieved by optimally calibrating the engine under the current emission standard can no longer be retained for the heavier pickup and van, and shrinks to an improvement of 1 to 6 percent. Most of this decrease in improvement is due to the inability of the oxidation catalyst to reduce the hydrocarbon emissions efficiently.

If the total burden of emission reduction could be handled by the catalyst, the improvements in fuel economy shown under the present standard could be retained. Table 2-8 shows the sensitivity

TABLE 2-7. OPTIMIZED FUEL ECONOMIES FOR LIGHT TRUCKS

VEHICLE	INERTIA WEIGHT (LBS)	ENGINE DISPLACEMENT (in^3)	REAR AXLE RATIO	VEHICLE FRONTAL AREA (ft^2)	COMPOSITE FUEL ECONOMY (c)(d)			
					BASELINE	OPTIMAL UNCONSTRAINED EMISSION	CURRENT EMISSION STANDARD (a)	1983 EMISSION STANDARD (b)
Mini Pick-Up	2500	121	3.08	24.4	28.3	34.1 (+20.5%)	33.8 (+19.5%)	33.8 (+19.5%)
Mini Pick-Up	3000	121	3.31	20.6	29.0	32.5 (+11%)	32.0 (+10%)	32.0 (+10%)
Regular Pick-Up	3800	305	3.54	32.3	18.7	21.5 (+15%)	21.4 (+14%)	19.9 (+ 6%)
Van	4500	305	2.56	38.0	18.2	20.7 (+14%)	20.2 (+11%)	18.3 (+ 1%)

(a) Current Emission standard; 1.7/18/2.3; HC/CO/NO_x; gms/mile

(b) 1983 Emission standard; 0.8/10/2.3; HC/CO/NO_x; gms/mile

(c) Assumed

(a) 0.66 Engineering factor
(b) Catalyst through-put ratio & factor for cold start, (HC=0.4; CO-0.35)

(d) Bracketed numbers denote the percent difference in fuel economy compared to the baseline vehicle.

TABLE 2-8. LIGHT TRUCK OPTIMIZED COMPOSITE FUEL ECONOMIES AS A FUNCTION OF OVERALL HYDROCARBON THROUGH-PUT RATIO (PROPOSED 1983 LT EMISSION STANDARD 0.8/10/2.3; HC/CO/NO_x GMS/MILE)

	HC OVERALL CATALYST THROUGH-PUT RATIO			
LIGHT TRUCK CLASSIFICATION	0.4	0.3	0.2	0.1
Mini Pick-Up (121 CID)	32.0 (+10%)*	32.0 (+10%)	32.0 (+10%)	32 (+10%)
Regular Pick-Up (305 CID)	19.9 (+ 6%)	21.0 (+12%)	21.4 (+14%)	21.4 (+14%)
Van (305 CID)	18.3 (+ 1%)	19.1 (+ 5%)	20.0 (+10%)	20.2 (+11%)

*Numbers in brackets denote percent difference in fuel economy between baseline and optimized engine.

of fuel economy to four levels of hydrocarbon (HC) catalyst through-put ratio for the 1983 light truck emission standard. The 1 through 6 percent improvement is for representative overall HC through-put ratio of 0.4. This overall improvement doubles when the HC through-put ratio is halved to 0.2.

2.4 ELECTRONIC CONTROL SYSTEMS

All projections indicate an increasing use of on-board electronics for engine control and for the implementation of engine calibrations. Some introductory electronic modules were composed of analog circuitry (Ford's ECU-A Feedback Carburetor controller, and Chrysler's LEAN BURN spark advance controller). They were primarily single function or single variable controllers. Future multivariable requirements will necessitate digital microprocessor control techniques. The advantages over analog circuitry lie in the microprocessor's ability for accuracy and repeatability, and for the complex or shared logic applications. In addition, the microprocessor 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 2-2 presents the total integrated engine/powertrain control system that is anticipated for the 1980-1990 time frame. Table 2-9 outlines, in detail, the mechanical hardware, sensors,

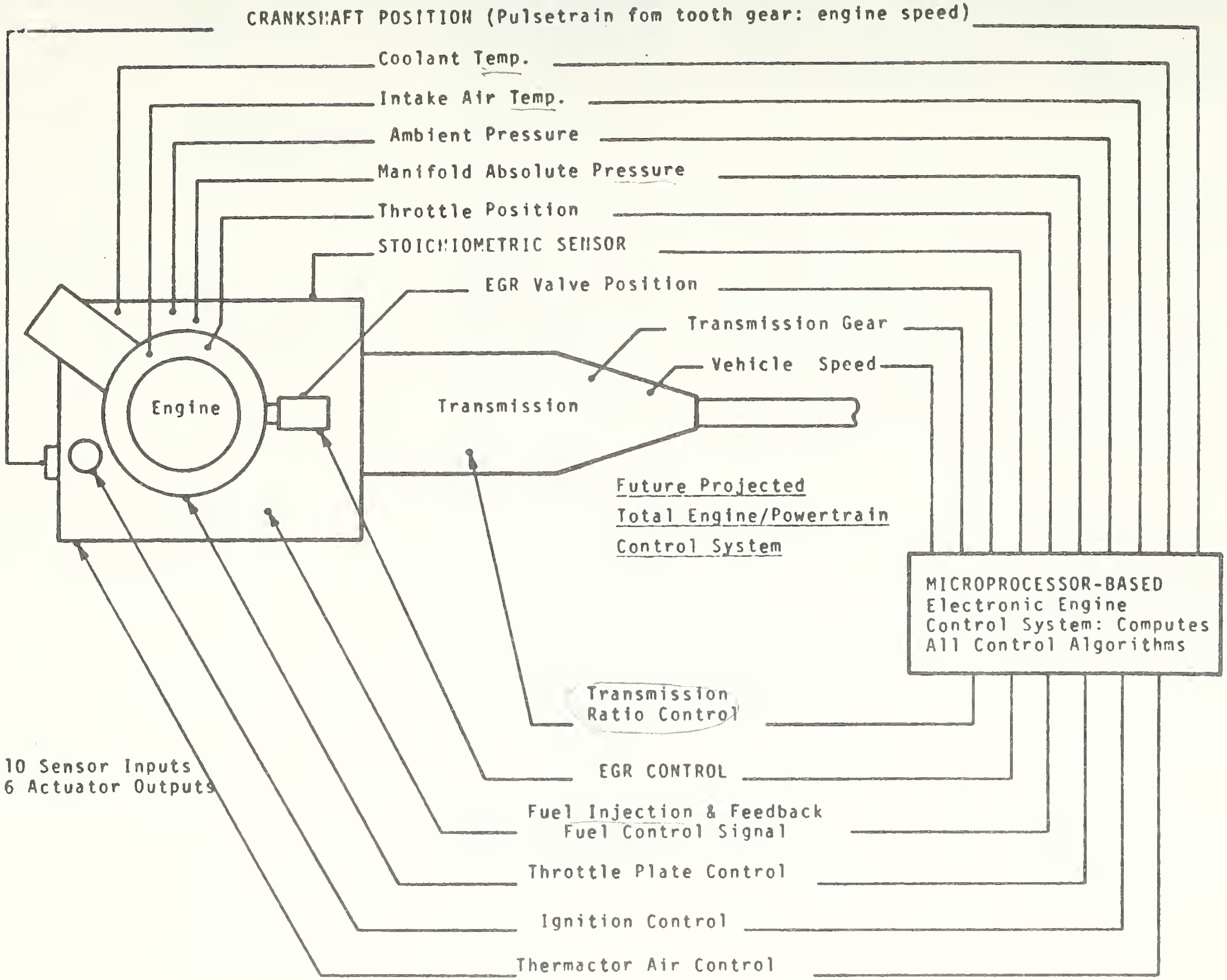


FIGURE 2-2. BASIC PARTS OF ENGINE CONTROL SYSTEM

TABLE 2-9. ADVANCED TECHNOLOGY REQUIRED FOR ELECTRONIC CONTROL OF SIGNIFICANT POWERTRAIN VARIABLES (1980-1990 TIME FRAME)

POWERTRAIN CONTROL VARIABLE	Mechanization Hardware	Sensors	Strategy of Control Concept Implementation
SPARK Advance	Mechanical and Vacuum Advance mechanisms will be totally replaced by microprocessor-based Electronic Spark Control Systems including Electronic Ignition modules for improved spark quality and reliability.	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 optimized 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 pintle position, throttle position as well as sensor listed under spark advance.	Simple control of EGR versus manifold vacuum will be replaced by microprocessor based optimized software strategy developed as defined under spark advance.
FEEDFORWARD FUEL MASS FLOW Management	Carburetors (calibrated as volumetric flow devices) will be replaced by Electronic Fuel injection systems. Throttle-body injection, intake-port injection or direct cylinder-stratified charge injection systems will be used, purpose being improved cylinder distribution and reduced interactions with intake manifold.	Except for air mass flow meter-based EFI systems, all others are speed-density based requiring all sensors listed under spark advance and EGR in order to compute control law.	Speed-density control with EGR flow correction will be computed by microprocessor-based EFI control module. Initial system is calibrated to empirically derive engine volumetric efficiency expression.
FEEDBACK CONTROL Closed Loop: A/F Ratio, Spark Advance, EGR MASS FLOW	o Stoichiometric A/F set-point - used with TWC catalyst o Lean A/F setpoint - used with COC catalyst and Stratified Charge - PROCO TYPE engine systems o Knock Spark Control - used on all engines requiring NO_x abatement via EGR dilution.	o For Stoichiometric control a Zirconia dioxide sensor is required o For Lean A/F control a linearized Cobalt oxide or Titanium dioxide is used o For Knock Control an accelerometer is used o For closed-loop EGR valve control a EGR pintle position sensor is used.	Closed-loop EGR, SPARK and A/F Ratio 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.
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 micro-processor-based electronic module with the CVT control software being more sophisticated.
Dual Displacement Engines	Under low load conditions for V-8 or V-6 engines half the cylinders may be turned off via the Eaton valve deactivator/Solenoid System	Sensor requirements will be engine speed, vehicle speed, load, transmission gear and throttle position.	The ON/OFF logic may be computed by boolean logic digital circuits but a centralized computer may already have the sensory inputs required.
Thermactor Air Control	Requires presently available thermactor air pump, switching thermactor air valve and electrically actuated solenoids.	Sensor requirements include engine coolant, throttle position, catalyst temperature and inlet air temperature.	Microprocessor-based control strategy will switch thermactor air to atmosphere, upstream or mid-bed of a TWC/COC catalyst system.

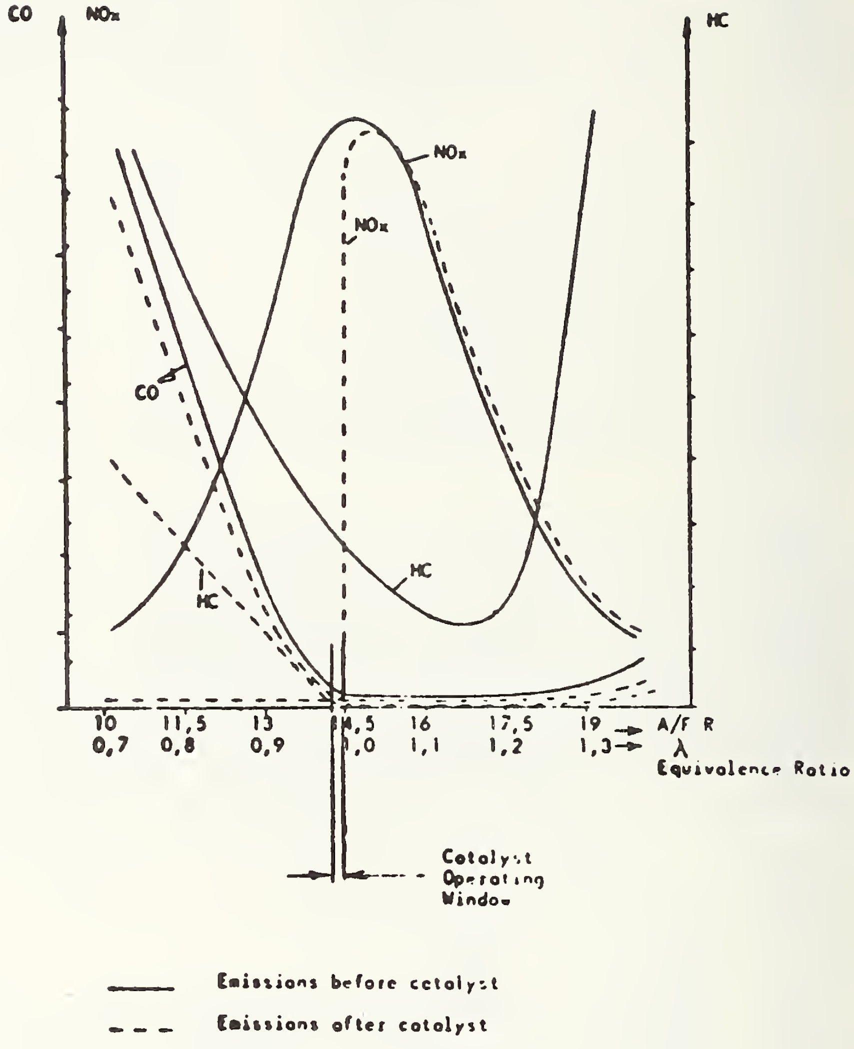
and the software/microprocessor implementation strategy that will be required to produce the engine/powertrain control system for the 1980's.

2.5 EXHAUST AFTERTREATMENT DEVICES

There have basically been two types of catalysts used in production to date, the conventional oxidizing catalyst (COC), and the three-way catalyst (TWC). Since the COC canister is an oxidation catalyst only, it is usually implemented with a secondary air injection system to insure sufficient free oxygen for complete oxidation of HC and CO species. The COC system is also applicable to future engine/emission systems which are designed to operate at lean air/fuel ratios. However, if accuracy of A/F control is not good enough to allow very lean operations, additional NO_x control systems such as EGR or spark retard will be required.

The TWC system is capable of conversion of all three regulated emissions specified (oxidation of HC and CO, and reduction of NO_x) and, therefore, requires precise operation near the stoichiometric A/F ratio. The conventional catalyst can be used in conjunction with a three-way catalyst for additional oxidation of HC and CO. In this configuration, secondary air injection occurs between the TWC canister and the COC canister. Emphasis is given here to the TWC, because the requirements it places on the engine control system will have a significant impact upon the use of electronics.

The conversion efficiencies of catalysts are particularly sensitive to temperature, space velocity, mean A/F ratio, and variations of A/F ratio with time. Figure 2-3 shows the typical conversion efficiencies for a three-way catalyst. Catalytic control of HC and CO emissions requires an oxidizing atmosphere in the exhaust which is typically available with lean A/F ratio operation or secondary air injection. Catalytic control of NO_x emissions requires a reducing atmosphere in the exhaust which is typically available with rich A/F ratio operation. However, at the chemically correct mixture (stoichiometry) a very narrow A/F



Source: Reference 9

FIGURE 2-3. EXHAUST EMISSIONS WITH THREE-WAY CATALYST ratio window exists where the conversion efficiencies for HC, CO and NO_x simultaneously exceed 80 percent. This implies that the design requirement for TWC-based emission control systems is that the A/F ratio is controlled to within this window (± 1 percent of stoichiometry).

Previously, this window was thought to define the required accuracy of the vehicles' fuel control system. However, several factors relax this picture significantly. Time-varying A/F ratio affects catalyst performance, and the A/F ratio of an engine change in a manner which favors conversion of the pollution component which is most important at the moment (A/F enrichment during power modes tends to favor NO_x conversion at the times NO_x is being produced in the greatest quantities).

SECTION 2 - REFERENCES

1. Dohner, A.R., "Transient System Optimization of an Experimental Engine Control System over the Federal Emissions Driving Schedule," SAE 780286.

2. Chrysler Progress Report, Contract DOT-TSC-1421.

3. Fiat Contract DOT-TSC-1422.

4. Trella, T.J., "Spark Ignition Engine Fuel Economy Control Optimization - Techniques and Procedures," SAE 790179.

5. Lockhart, "A Fuel Economy Development Vehicle with Electronic Programmed Engine Control (EPEC), SAE 790231.

6. Nicholson, R. C., "Emission and Fuel Economy Interaction," (Closed Loop Fuel Control, 3-Way Catalyst), SAE 780016.

7. EPA Advisory Circular, Optional Objective Criterion, BC-24-2.

8. Oser, P., "Catalyst Systems with an Emphasis on Three-Way Conversion and Novel Concepts," Volkswagenwerk AB, SAE 790306.

3. ENGINE DESIGN CONCEPTS

3.1 GENERAL

A vehicle's fuel economy is determined by several major factors, i.e., the efficiency of the engine, the driveline matching the engine's power/efficiency characteristics to a given vehicle, the characteristics of the vehicle, and the driving cycle to which the vehicle is subjected. This section is concerned with engine design concepts and characteristics, including: a) the breakdown of the energy losses that occur in operating an engine based on the energy supplied by the fuel), b) engine combustion systems and configurations and their impacts of fuel economy, and c) other design technologies and their potential for improving the efficiency of engines.

3.2 ENERGY LOSSES IN THE ENGINE

An illustration of the process by which the energy of the fuel is distributed among the various chemical and mechanical functions of a typical gasoline engine during urban and steady state driving is shown in Table 3-1. In this example, it is observed that 20 percent of the fuel energy is available for the operation of the vehicular system on the road. This system includes the accessories, transmission, rolling resistance, aerodynamics, and braking.

The losses listed in Table 3-1 interact with each other, and the means by which they interact are quite complicated. For example, an increase in the compression ratio would decrease the air-standard Otto cycle inefficiency and might increase heat transfer from the working fluid, mechanical friction, and dissociation and losses due to changing specific heats. Other losses, particularly those due to incomplete combustion, might decrease slightly. A brief description of the losses presented above is given in the following paragraphs. In addition, Table 3-2 lists some methods of reducting the energy losses. Although the information in the table was developed in 1966,[2] it still is a good representation of unsolved problems areas.

TABLE 3-1. ENGINE FUEL ENERGY DISTRIBUTION

ITEM	ENERGY DISTRIBUTION AT 50 MPH (% OF FUEL HIGHER HEATING VALUE)
LOSSES	
Otto Cycle Inefficiency (basic cycle)	35
Incomplete Combustion	6
Dissociation and Changing Specific Heats (real gas)	9
Heat Transfer from Cylinder	10
Lower Heating Value	6
Mechanical Friction	6
Pumping	6
Finite Combustion	2
USEFUL WORK	
BHP Available for Vehicle System	20
	100%

Source: Reference 1

TABLE 3-2. REDUCTION OF ENERGY LOSSES IN THE ENGINE

Energy Conversion Loss	Possibility of Reduction	Relative Size of Possible Reduction	Methods of Reduction	Disadvantages of Methods of Reduction
Uncondensed water vapor in products of combustion	None	--	--	--
Incomplete combustion	Good	Moderate	1. Increase turbulence of fuel-air mixture 2. Lean fuel-air mixtures	1. Increases heat transfer from working fluid 2. Requires improvements in carburetion and manifolds
Dissociation and changing specific heats	Good	Moderate	Leaner fuel-air mixtures resulting in lower temperatures inside cylinder	Requires improvements in carburetion and manifolding and/or fuel injection
Air-standard Otto cycle inefficiency	Good	Small	Increasing compression ratio	Increase engine friction, octane requirement, and mechanical stress
Finite combustion	Fair	Small	More closely approximating constant volume combustion	Creates higher rate of pressure rise and higher maximum pressure thus placing higher stresses on engine. Also increases engine roughness
Heat transfer from working fluid	Fair	Small	Improved materials and oils making possible higher metal temperatures	Expense, lower volumetric efficiency and lower compression ratio of higher octane fuel
Pumping	Good	Small to Moderate	unthrottled engine	Requires another method of load control
Mechanical friction	Fair	Small to Moderate	1. Closer tolerances on engine parts 2. Rolling contact bearings and rolling cam followers	1. Expense 2. Expense

Source: Reference 2.

a) Lower Heating Value

This loss is attributed to the uncondensed water vapor in the product of combustion.

b) Incomplete Combustion

Not all of the fuel is burned to complete combustion of products in a spark ignition engine. This loss accounts for the presence of CO and unburned hydrocarbons in the exhaust.

c) Dissociation and Changing Specific Heats

Gases in an engine exhibit variable specific heats as the composition of the gas and temperature change. During combustion, dissociation takes place. Dissociation results in less energy available to do work and, thus, represents a loss. Dissociation occurs under high temperatures and pressures. The dissociation losses decrease as leaner mixtures are used. However, under these conditions, the power output of an engine decreases. Thus, a tradeoff exists.

d) Otto Cycle Inefficiencies

This inefficiency represents a significant amount of the fuel lost and is a function of the compression ratio. The ideal Otto cycle efficiency is customarily calculated using the air standard cycle, and cycle inefficiencies are referenced to that value. In the case of turbocharging, the compression ratio is established by both the turbocharger boost pressure ratio and the basic engine's compression ratio. Generally, if the basic engine's compression ratio is decreased, and the turbocharger boost is increased, more work is transferred from the reciprocator to the turbine. The loss is increased since the turbine is generally less efficient than the reciprocator.

e) Finite Combustion

This loss is attributed to the duration of burning over the combustion volume space. The basic cycle losses are based on the assumption that energy is released instantaneously at constant volume. Finite combustion losses can also be thought of as "long" or "late" combustion losses. These energy losses are relatively small as indicated in Table 3-3. However, the effect of finite combustion on emissions is significant. The pollutants which form during the combustion process affect emission characteristics directly and fuel economy indirectly. Some factors which influence finite combustion (extent of combustion) include flame speeds, macro/micro turbulence, mixture strength distribution, and combustion space size.

f) Heat Transfer from Cylinder

Table 3-3 shows an estimated distribution of heat transfer of parts of the cylinder and the ports.[3] This estimate was generated for a diesel engine and is given here only to indicate general trends for reciprocating, internal combustion engines. Roughly one-quarter of the heat transfer loss is in the intake and exhaust ports, and three-quarters is in the cylinder. Most of the heat losses occur over a small portion of the engine cycle, near the top dead center of piston motion, during the combustion phase. The heat transfer inside the cylinder during the compression and expansion stroke is less significant.

Cylinder heat transfer predominates at low engine speeds where sufficient time is available to promote this process. Generally, the reduction of cylinder heat transfer affects a) the cooling requirement of the engine which, when reduced, provides a great reduction in accessory losses (fan and radiator), and b) the combustion process itself. In the latter case, the primary gain in fuel economy will not come directly from reduced heat transfer, but from efficient utilization of the increased exhaust temperature by extracting useful energy from the exhaust. A cylinder could be built to meet structural requirements at temperatures as high as 1800°F or 1900°F. However, the lubrication requirements cannot be met, and this is the limiting item.

g) Pumping

Pumping is a direct loss which accounts for approximately 4 to 6 percent of the energy supplied by the fuel. In a conventional carbureted spark ignition engine, part of this loss is attributed

TABLE 3-3. DISTRIBUTION OF HEAT TRANSFER LOSSES FOR CYLINDER AND PORTS

ENGINE PART	% HEAT TRANSFER LOSS
Cylinder	
Piston	36
Cylinder Head	19
Cylinder Sleeve	13
Intake Valve	5
Exhaust Valve	2
	75%
Ports	
Exhaust	25.2
Intake	-0.2
	25%

Source: Reference 3.

to a) valving, b) manifolding, c) the throttle, d) the flow through the carburetor, and e) the state of the gas media itself. These losses directly affect the volumetric efficiency of an engine. A typical breakdown of the mechanical and pumping losses is illustrated in Figure 3-1 as a function of speed. The pumping components are the valve pumping and throttle losses. Part of these losses can be reduced by better intake/exhaust valve and porting layouts to provide good flow coefficients, by maximizing the acoustical characteristics of inlet and exhaust manifold, and by optimizing the time of inlet/exhaust valve opening, closure, phasing, and lift rates. Pumping loss has no significant effect on exhaust gas temperature. Some of the techniques listed above to reduce its effect do, however, have an indirect effect on the emission characteristics.

h) <u>Mechanical Loss</u>

The indicated power developed by a spark ignition engine does not all appear as break power since a part is lost in overcoming friction of the bearings, pistons, and other mechanical parts of the engine. This power loss is referred to as mechanical friction loss.

A typical breakdown of mechanical and pumping friction losses were computed by DOT/TSC and is shown as a function of engine speed in Figure 3-1. This breakdown uses simulation algorithms developed by I.W. Bishop[4] for seven major friction components: a) crankcase mechanical friction, b) ring gas pressure losses, c) static ring tension losses, d) viscous piston losses, e) blow-by losses, e) valve pumping loop losses, and g) exhaust and inlet throttling losses. The crankcase mechanical friction was further broken down into bearing friction, valve gear friction, pump, and miscellaneous. The piston mechanical friction was divided into two main categories, viscous and non-viscous. The non-viscous components include the static ring tension and tension resulting from gas pressure forces behind the ring.

Both the crankcase and piston mechanical losses constitute over 75 percent of the loss due to mechanical friction. These losses are emphasized by an increase in speed, see Table 3-4.

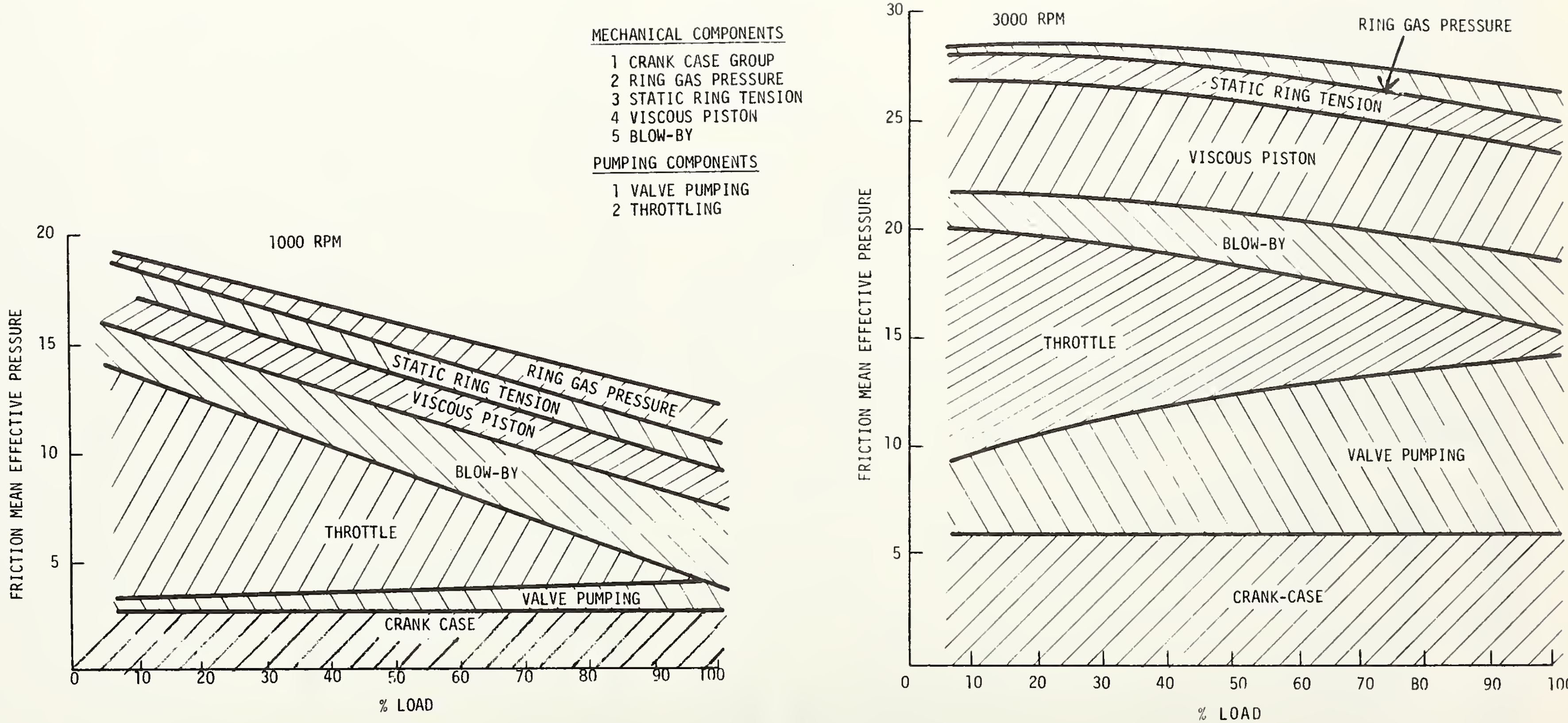


FIGURE 3-1. MECHANICAL AND PUMPING LOSSES AS A FUNCTION OF SPEED

TABLE 3-4. MECHANICAL FRICTION LOSS VERSUS SPEED

FRICTION CONPONENT	% of Mechanical Friction Loss	
	1000 RPM/40% load	3000 RPM/40% load
Crankcase	30	40
Piston Mechanical	42	45
Blow-by	28	15
	100	100

The mechanical piston friction losses result from the mechanical friction between lubricated surfaces. It appears as heat dissipated to the radiator and oil cooler system. Decreasing these losses increases the maximum power potential, first because of the direct increase in brake torque and second because the engine can be operated at higher speeds. Mechanical friction losses can be reduced by piston assembly design changes. This should result in a better specific fuel consumption. Design considerations suggested by Millington and Hartles[5] include:

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, and keeping relieved surfaces well drained of oil.
4) Use light oils for low viscous losses.
5) Keep bearing diameters small.
6) Minimize power demand of oil and water pumps and other auxiliaries.

3.3 COMBUSTION SYSTEMS

One of the fundamental design considerations of passenger car and light truck spark ignition engines is combustion chamber design. To date, most spark ignition engines produced by U.S.

manufacturers include wedge shaped combustion chamber designs. General Motors features one engine design of the open chamber type. Various types of combustion chamber designs are in production by foreign manufacturers. Their types include a) disc-in-cylinder-head, b) bowl-in-piston, c) hemispherical, d) bathtub, and e) wedge.

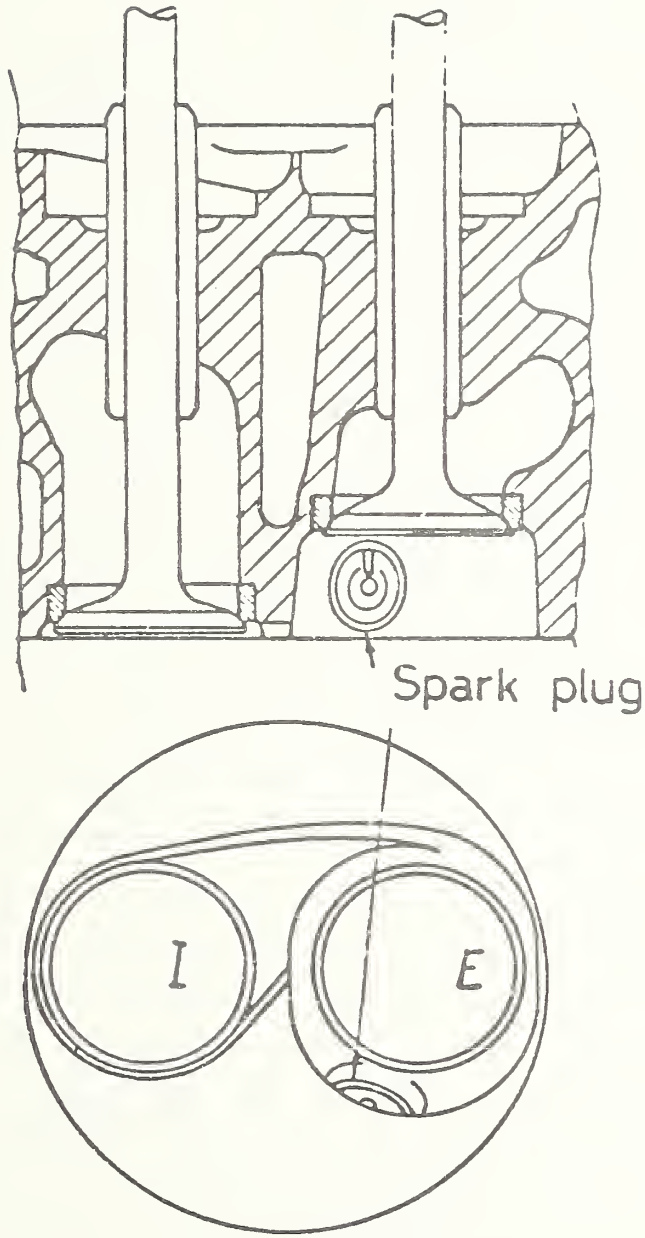
The benefits derived from combustion chamber redesigns are plentiful. Recent studies[6] conducted by Ricardo concluded that there was a strong correlation between compact chambers (hemispherical or bowl-in-piston) and good fuel economy. In this study, 28 production engines were evaluated.

An approach being pursued to improve the fuel economy and emission characteristics of vehicles is the lean burn, homogeneous charge combustion system. An advantage is its ability to operate at a lean air/fuel ratio to control emissions rather than using three-way catalysis. The stratified charge engine also shows positive fuel economy advantages with potentially good emission characteristics. The following paragraphs summarize the state-of-art of these combustion chambers and their potentials for use.

3.3.1 May Combustion Chambers

The May engine, commonly known as the May-Fireball engine, is a lean burn engine (homogenous charge), which emphasizes squish and turbulence to increase flame speeds in a high compression ratio compact chamber. The primary objective of this design is to reduce the part-load specific fuel consumption by promoting more complete and faster oxidation of lean mixtures under increased compression ratios, reduced internal residual gas fractions, and knock-limited conditions. To date, prototype design compression ratios ranging from 13.5 to 14.6:1 have been used for 97 RON fuels.

The most recent engine configuration is shown in Figure 3-2. The combustion chamber is located primarily below the exhaust valve. Thus, the combustion gases do not communicate with both



SOURCE: Reference 7

FIGURE 3-2. HIGH COMPRESSION LEAN MIXTURE HOMOGENEOUS CHARGE COMBUSTION CHAMBER the hot exhaust and cool inlet valves, maintaining more uniform gas temperature. A primary objective of this combustion system is to avoid hot spots and, thus, allow higher compression ratios without knock. The location of the spark plug is critical for optimal combustion. The design features a channel-like recess leading from the inlet valve area to the combustion chamber, thereby causing the flow to enter the combustion chamber recess in a tangential or swirling manner. Squish is provided by compressing the mixture under the inlet valve into the combustion chamber. Flat head pistons are normally used.

Prototypes have been produced and tested in a limited number of vehicles. Most of the evaluations have been made under steady state operating conditions or for the European drive cycle. A 97 Ron fuel was used in all cases. Limited data are available for the EPA drive cycle. Some emission data were presented for this engine in a Passat car. However, no fuel economy data were reported. Most recently, however, EPA drive cycle test data were presented for a 2500 lb. inertia weight vehicle equipped with the May engine where a composite economy of 25.6 mpg was reported. Comparing the May engine with the EPA average fleet fuel economy data under equal performance, the 2500 lb. vehicle with this engine shows a 12.5 percent improvement in fuel economy. (See Table 3-5.) The May engine also exhibits low NO_x and CO emissions without any exhaust emission aftertreatment or EGR.

The May engine is currently in the prototype stage of development. There are, of course, specific design areas which need attention before a production engine will be available. Some of the current issues which require resolution include:

1. Currently the engine exhibits a higher surface area/volume ratio than conventional gasoline engines which causes high HC emissions. The engine can be improved by tighter clearances, and by locating the piston rings closer to the top end of the piston. This design feature

TABLE 3-5. COMPARISON OF MAY COMBUSTION SYSTEM WITH AVERAGE 1978 PRODUCTION ENGINE PERFORMANCE FOR 2500 lb CARS.

	INERTIA WEIGHT (lbs)	CID (in^3)	HP	HP/IW	ENGINE OUT EMISSION $HC/CO/NO_x$	FUEL ECONOMY COMPOSITE (mpg)	FUEL ECONOMY CORRECTED FOR HR/WT	% DIFFERENCE
May (a)	2500	122	115	0.046	3.3/2.9/0.8	25.6	31.1	+12.5%
Average 1978 Certification Data (b)	2500	101	-	0.027	—	27.6	-	BASE

(a) 14.6 CR, 97 RON, Source: Reference 7.

(b) Source: Reference 8.

can also be incorporated for control of HC emissions.

2. The engine exhibits low exhaust gas temperatures and CO emissions. There are doubts that an oxidation catalyst could be employed under these conditions since present-day catalysts require higher temperatures to reduce HC.

3. There is some thermal penalty at WOT since the engine must be set at retarded timings under rich mixture conditions to avoid knock.

4. At this time, the engine requires a high voltage ignition system for damp weather starting, accurate timing, and, perhaps, a fuel injection system for good fuel distribution to each cylinder.

5. At this time, the pumping losses are higher since squish and swirl are features necessary to maintain turbulence to sustain lean operation. Successive tradeoffs in designs may be required to re-optimize for pumping losses.

The advantages include:

1. The engine is not susceptible to knock even up to 100,000 miles, because the tendency to form deposits on the combustion chamber is minimized under lean mixture conditions.

2. The cylinder and exhaust areas require less cooling because of lean mixtures.

3. The engine can be operated with a three-way catalyst system for heavier vehicles. Engine knock, however, may be a problem.

4. A compression ratio of 12.5:1 is feasible with U.S. 91 RON gasoline.

Currently, most of the information on the May combustion engine was generated in Europe where generally higher octane rated fuels are used. Some work is presently in progress at the Ricardo company (prototype level) on variations in design of combustion chamber geometries to promote better efficiencies under extended, lean limits. Presently, no information is available on the potential of these designs to operate on lower quality fuels (i.e., 91 RON). There are also current interests to examine the potential use of methanol as a fuel for this engine. Sophisticated controls will be required since the engine is highly sensitive to variability in spark timing. Unlike the conventional spark ignition engine, spark timings are usually set a few degrees before top dead center at high rotational engine speed. Finally, the May chamber must be machined because of its unique design features; this is a disadvantage when compared with conventional engines.

3.3.2 Lean Combustion Chambers

One example of a chamber designed for lean combustion, the May concept, was discussed in Section 3.3.1. In this section, the potential of other lean burn chambers are discussed. A comprehensive treatment on the current state of art of lean burn engines is available.[9]

For spark ignition engines, the desired air-fuel ratio is 18 to 20 to control emissions without a three way catalyst and to operate efficiently. Unfortunately, under these lean ratios, slow burning of the mixture results and causes a loss of performance and driveability. To overcome this problem, it is necessary to increase the flame speed. One means to increase flame speed is to create sustained turbulence in the cylinder chamber. Several alternative methods are currently under investigation. The two most prevalent methods include: a) increasing the squish areas in the combustion chamber, and b) adopting shrouded inlet valves to obtain more swirl. The first method is the most promising means to increase the flame speed since the turbulence is generated during the combustion phase and in the vicinity of the flame front. An analogous result could be achieved by means of short flame travel or with a more compact combustion chamber and higher compression ratios.

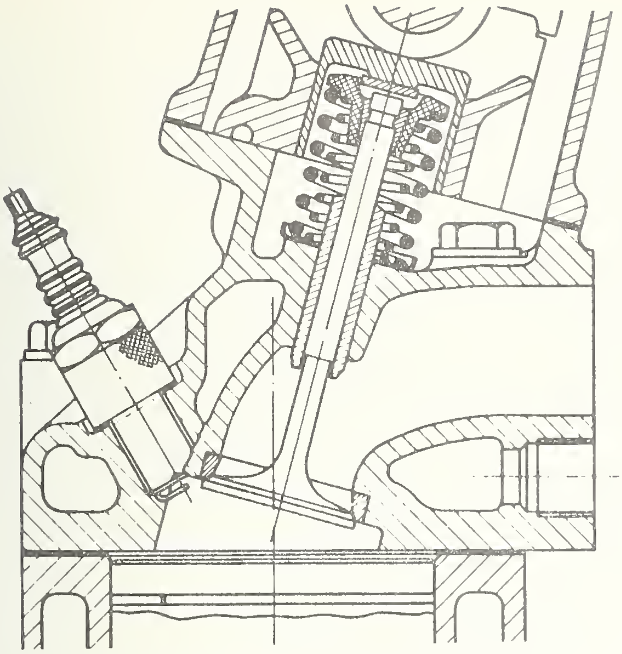
Combustion chamber configurations that hold promise as lean combustion designs include:

1. High squish area, bathtub, or bowl-in-piston type.
2. Prechamber designs with torch ignition and/or dual ignition system.

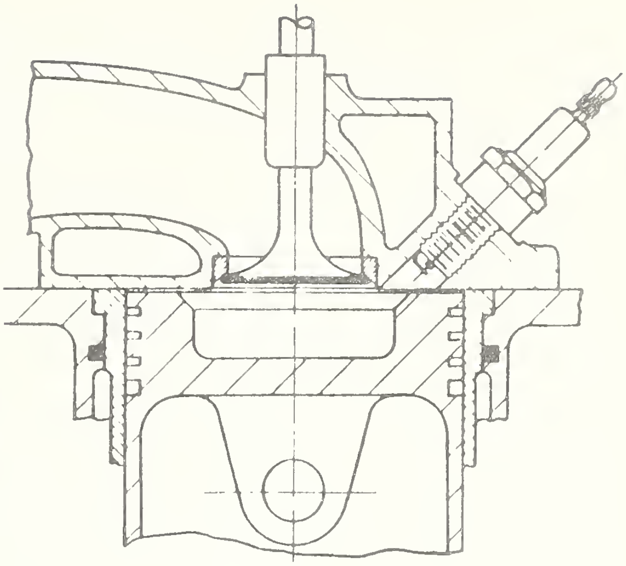
3.3.2.1 High Squish Area Lean Burn Engine - Some typical combustion chambers are shown in Figure 3-3. A study[6] presented test comparisons on the three combustion systems pictured, namely bathtub, bowl-in-piston, and quiescent disc modified with a light energy gap spark plug and a perfect mixture generator for fueling. The study concluded that the quiescent chamber gave the leanest limit at 26:1 air-fuel ratio, but needed a very large spark advance. The bowl-in-piston arrangement gave better economy. However, this arrangement could not operate at the lean mixture ratios observed for the quiescent chamber configuration. Finally, the bathtub design gave the best compromise at an air-fuel ratio of 24:1 with the best economy. The lower lean limit of the bowl-in-piston design was attributed to the greater loss of heat in the turbulent process.

Data furnished by Fiat[11] indicates that the bathtub design (see Figures 3-4 and 3-5) exhibits characteristically lower NO_x and HC exhaust emissions compared with their standard hemispherical chamber at air-fuel ratios of 15:1 and 18:1. However, the bathtub design exhibited fuel economy penalties. (See Figure 3-6 and Table 3-6.) The idle test results show that the idle fuel consumptions are slightly higher when the engine settings are calibrated for optimum spark timing and HC emissions. The wide open throttle test results show some loss in power with an increase in fuel consumption.

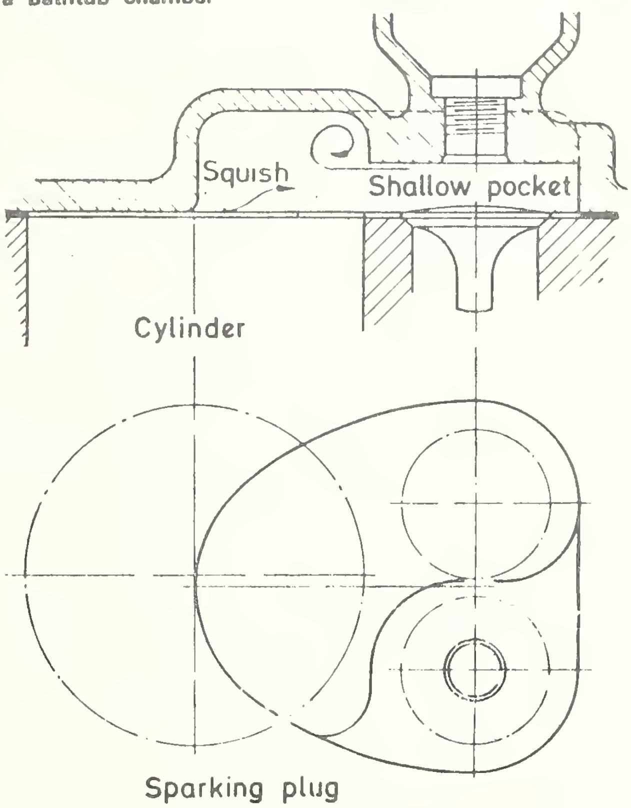
3.3.2.2 Prechamber Lean Burn Configurations - The desirability to maintain turbulence during all stages of combustion has focused interest on a new generation of prechamber (torch ignition) homogeneous charge spark ignition engines. Various prechamber design



A Ricardo design of a 'bathtub' chamber



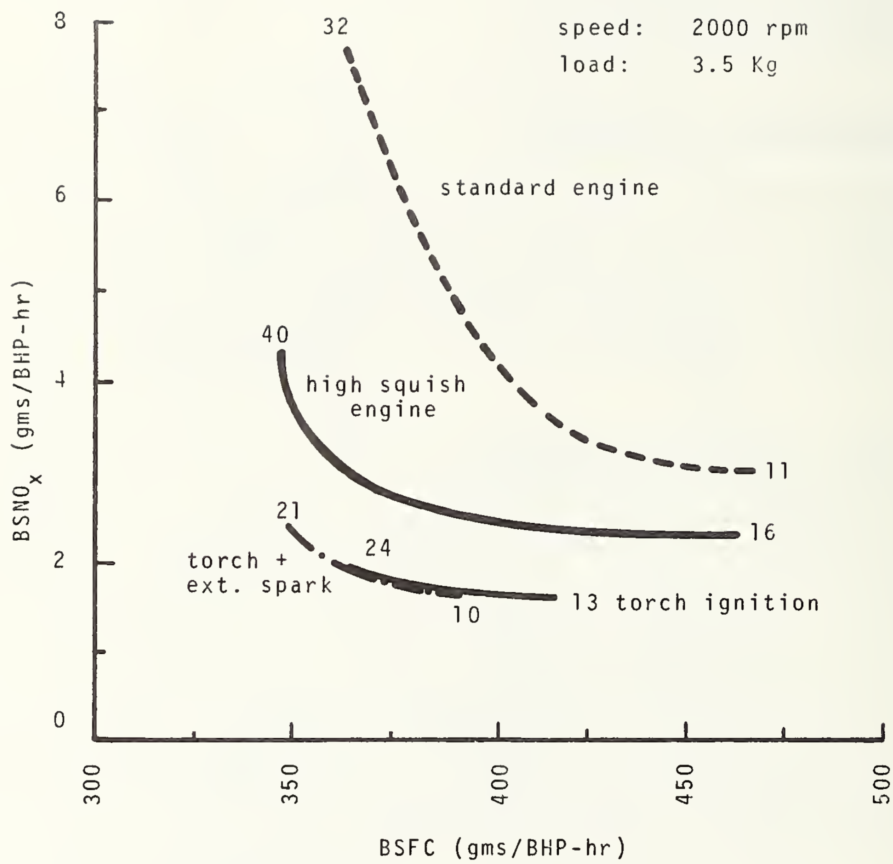
A Heron or bowl-in-piston combustion chamber



Shock absorber version of the turbulent h

Source: Reference 10, p. 13

FIGURE 3-3. TYPICAL TURBULENT LEAN BURN CHAMBERS



Source: Reference 11

FIGURE 3-4. BSNO$_x$ VS BSFC AT NO$_x$ PEAK FOR EACH SPARK ADVANCE (A/F ≃ 15)

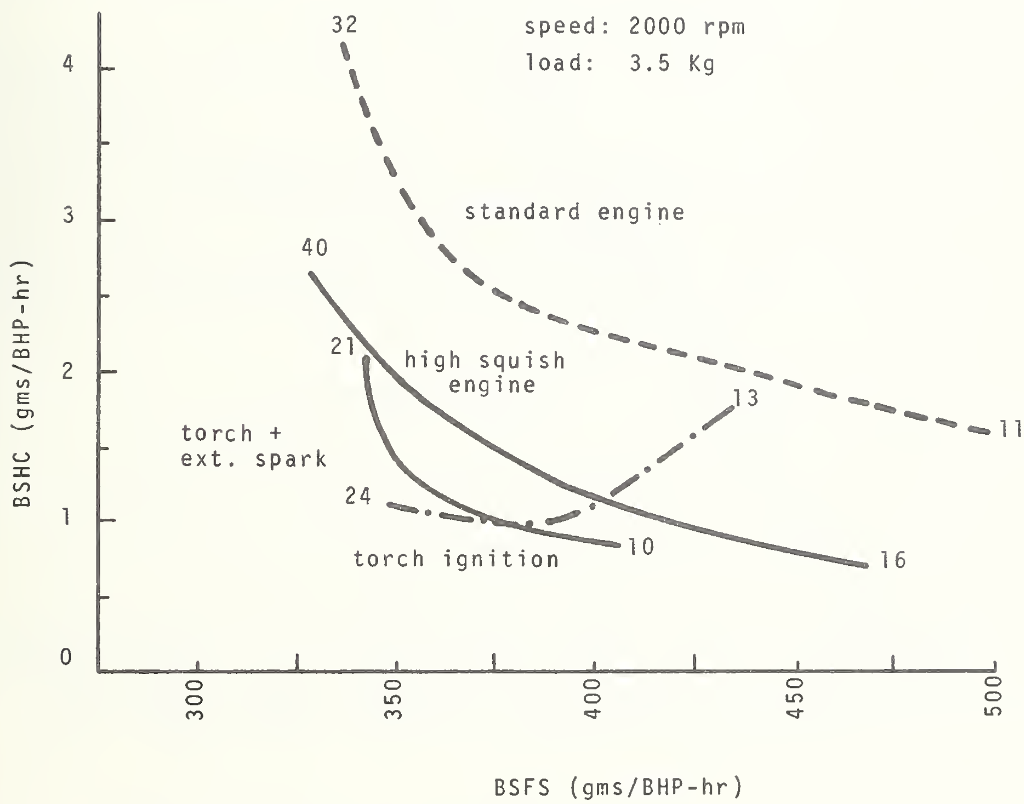


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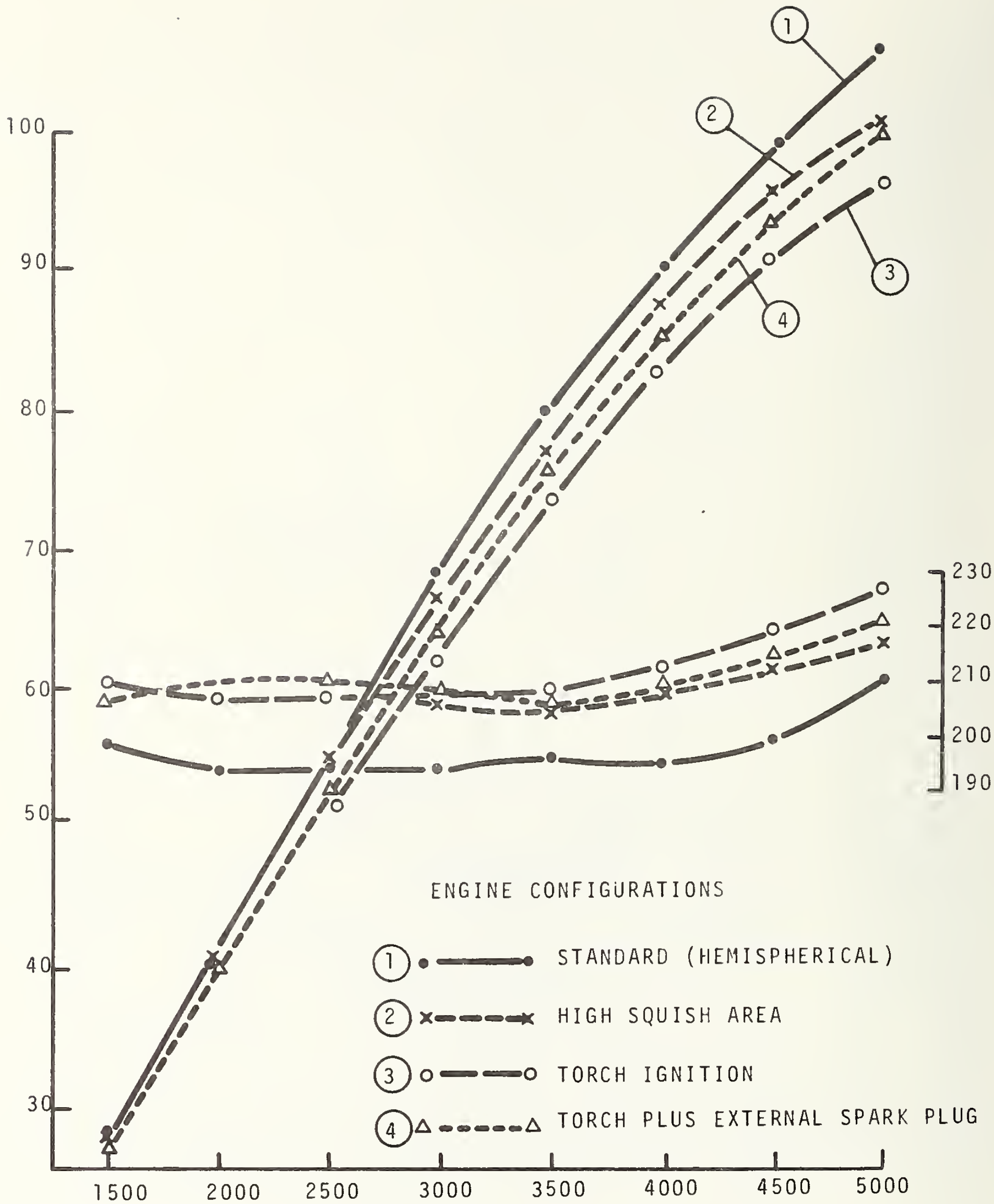
FIGURE 3-5. BSHC VS BSFC AT CONSTANT A/F = 18



Source: Reference 11

FIGURE 3-6. COMPARISON OF VARIOUS COMBUSTION SYSTEMS AT WOT FOR FIAT 132

TABLE 3-6. COMPARISON OF VARIOUS COMBUSTION SYSTEMS AT IDLE FOR FIAT 132

COMBUSTION CHAMBER CONFIGURATIONS	FUEL CONSUMPTION gr./hr.	HC (p.p.m.)
Standard (Hemispherical)	830	2500
High squish area (Bathtub)	900	3000
Torch ignition	1050	6500
Torch ignition plus external spark plug	920	3200

Source: Reference 11

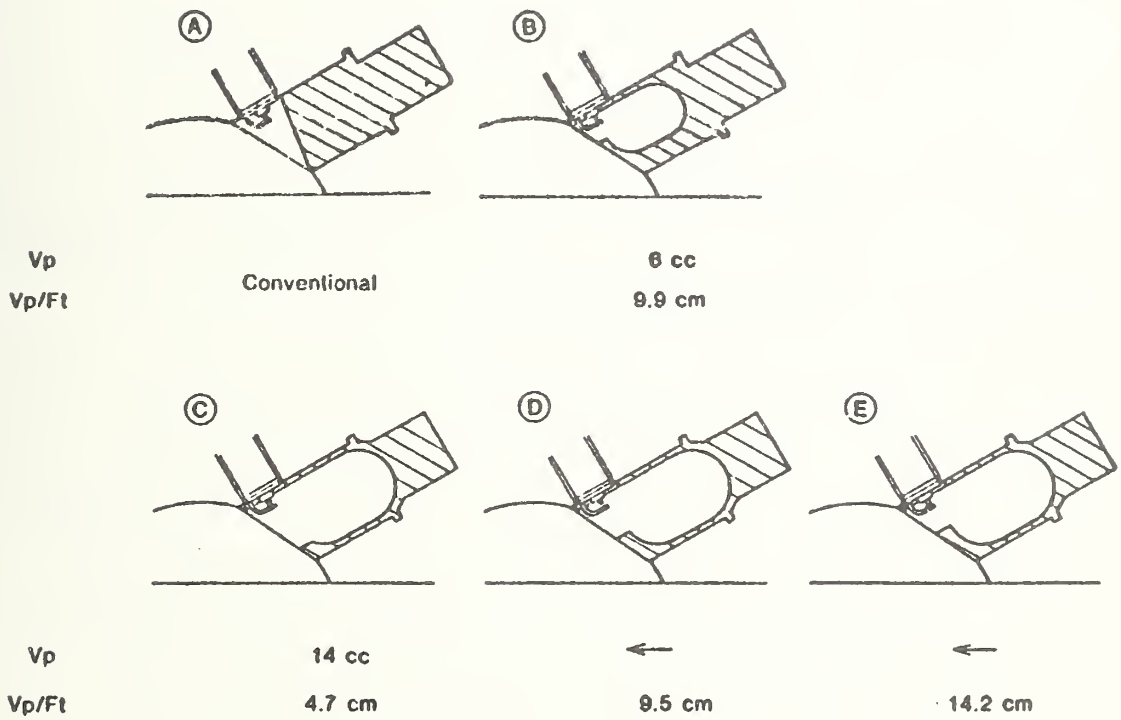
Running with the external spark plug only configurations are evolving with particular emphasis directed towards increasing combustion efficiencies and extending lean limits. Part of this activity involves a) positioning of the spark plug at critical locations in the prechamber, and b) the use of two plugs, one in the pre-chamber and the other in the main chamber. This lean burn concept does not have the problem of high squish open chamber engines (see Section 3.3.2.1) which do not maintain adequate turbulence throughout the entire period of combustion. The turbulence generated by squish in open-chambered engines decays rapidly after combustion commences.

Figure 3-7 shows typical prechamber designs under development at Toyota.[12] In these configurations, the prechamber does not contain a fuel injection nozzle or an additional intake valve as is the case of the Honda CVCC-stratified charge. It is charged by the homogenous mixtures in the main chamber during the compression.

Fiat has also conducted initial investigations (DOT/TSC-4222) of a small prechamber mounted on their spherical combustion chamber engines which serves as a torch ignition system. The main feature of this design is an ability to provide a decrease in NO_x emissions. At this time, fuel consumption and HC emissions are higher for these engines at idle compared with conventional engines, along with power penalties at wide open throttle. An additional spark plug in the main chamber shows promise to eliminate these disadvantages. See Figure 3-6 and Table 3-6. Limited data at two A/F ratios, Figures 3-4 and 3-5, show that the prechamber designs provide potentially higher optimum fuel economies under lower NO_x and HC emission levels. Current efforts are in progress at Fiat to assess the potential of these designs under various engine control strategies.

3.3.3 Stratified Charge Engines

One of the incentives behind the development of the early stratified charge engines was the reduction of pumping losses to improve the fuel economy at part loads. Another incentive was



Source: Reference 12

FIGURE 3-7. TYPICAL PRECHAMBER CONFIGURATIONS the elimination of the sensitivity of the engine to the fuel octane number. Also, the fuel economy of the stratified charge engine can be better than the homogeneous charge engine because higher compression ratios can be used.

A large number of stratified charge engines have been built and tested, but most of them are in the prototype stage. Table 3-7 presents a list of some representative types of stratified charge engines. They include: a) the single chamber type (open-chamber) in which combustion takes place in one chamber, initiated by a spark plug in a stratified mixture, and b) the dual chamber type (pre-chamber) which utilizes two chambers, one with a rich mixture and the other with a lean mixture, with spark ignition of the rich and torch ignition of the lean mixture in the main chamber. With the exception of the Honda CVCC, which uses a conventional carbureted system along with a third valve to introduce a rich mixture into the pre-chamber, fuel is generally introduced into both types of systems by means of a fuel injection system. The Honda CVCC engine, developed mainly for reasons of emission control, is the only production stratified charge engine currently in use in the United States.

In the stratified charge engines of the direct injection type, the overall fuel-air mixture is maintained at a leaner than stoichiometric level at wide open throttle to avoid smoking. Their power density is, therefore, less than that of the homogeneous charge engine which can operate rich at wide open throttle. These engines characteristically exhibit low NO_x emission levels and good fuel economy. The ability to provide good fuel economy is a result of the combined design features of lean-burn and high compression ratio without throttling. However, these engines are not without their problems. The hydrocarbon emissions are higher, which result from wall quenching and incomplete flame propagation in zones with fuel/air ratio below the lean flammability limit. At part loads, the hydrocarbons are not effectively oxidized during the expansion and exhaust processes because of the low mass average temperatures of the combustion products. To

TABLE 3-7. STRATIFIED CHARGE ENGINES BUILT AND TESTED

SINGLE CHAMBER (OPEN-CHAMBER)			
(1)	Ford PROCO	(5)	Texaco TCCS
(2)	Hesselman	(6)	Curtis Wright
(3)	Mitsubishi (MCP)	(7)	Deutz
(4)	Witsky	(8)	MAN-FM

DUAL CHAMBER (PRE-CHAMBER)			
(1)	Porsche	(4)	Pischinger
(2)	VW Combustion	(5)	Honda CVCC
(3)	Broderson	(6)	Ricardo Comet Mt. VB

overcome this problem, a throttle valve is used to reduce the part load hydrocarbons which, in turn, reduces part-load fuel economy. In addition, an oxidizing catalytic converter is used to reduce hydrocarbons. Three way catalysts are not used to reduce NO_x because of the high concentration of oxygen in the exhaust. There are questions concerning whether the engines 1) can be manufactured cost-effectively, 2) can operate on the FTP cycle with acceptable emissions and acceptable driveability, and (3) can operate with acceptable particulates and other non-regulated emissions.

The emissions and fuel economy potential is illustrated in Table 3-8 and Figure 3-8 for a 141 CID Texaco TCCS engine in a 2750 lb M-151 military vehicle and a 2500 lb car. Data is shown for a 6.6 liter PROCO in a 5000 lb vehicle (Table 3-9). For comparison, the average sales-weighted composite fuel economy and engine displacement for 1979 2500 lb and 2750 lb model cars are 27.6 and 24.0 mpg and 125 CID, respectively.

The relationship between emissions and fuel economy presented in Figure 3-8 shows a fuel economy penalty in the urban cycle of approximately 26 percent between unconstrained emissions and 0.41/3.4/0.4 ($HC/CO/NO_x$) grams/mile. This penalty reflects the use of linear, mechanical control of spark retard and EGR. The fuel economy penalty with electronic controls may be less.

The PROCO Combustion System (see Figure 3-9) includes:

a) a combustion bowl in the piston, an intake port designed to impart swirl to the inducted air, a fuel injector nozzle and two spark plugs;

b) a unique injector nozzle design which provides acceptable atomization with only 250 psi pressure drop;

c) dual spark ignition which facilitates operation with lean air/fuel ratios and high EGR rates which result in NO_x control capability coupled with limited HC and CO control;

TABLE 3-8. EMISSIONS AND FUEL ECONOMY OF THE NATURALLY-ASPIRATED TCCS ENGINE IN A 2750 lb. M-151 VEHICLE

LAB AND ENGINE CONFIGURATION	MASS EMISSIONS - gms/mi			EPA-COMPOSITE CYCLE FUEL ECONOMY - MPG
	HC	CO	NO_x	
Texaco - Full EGR	0.36	0.61	0.31	16.2
EPA - Full EGR	0.37	0.24	0.31	15.8
Texaco - Reduced EGR	0.48	0.57	0.45	17.6
EPA - Reduced EGR	0.50	0.14	0.70	21.9

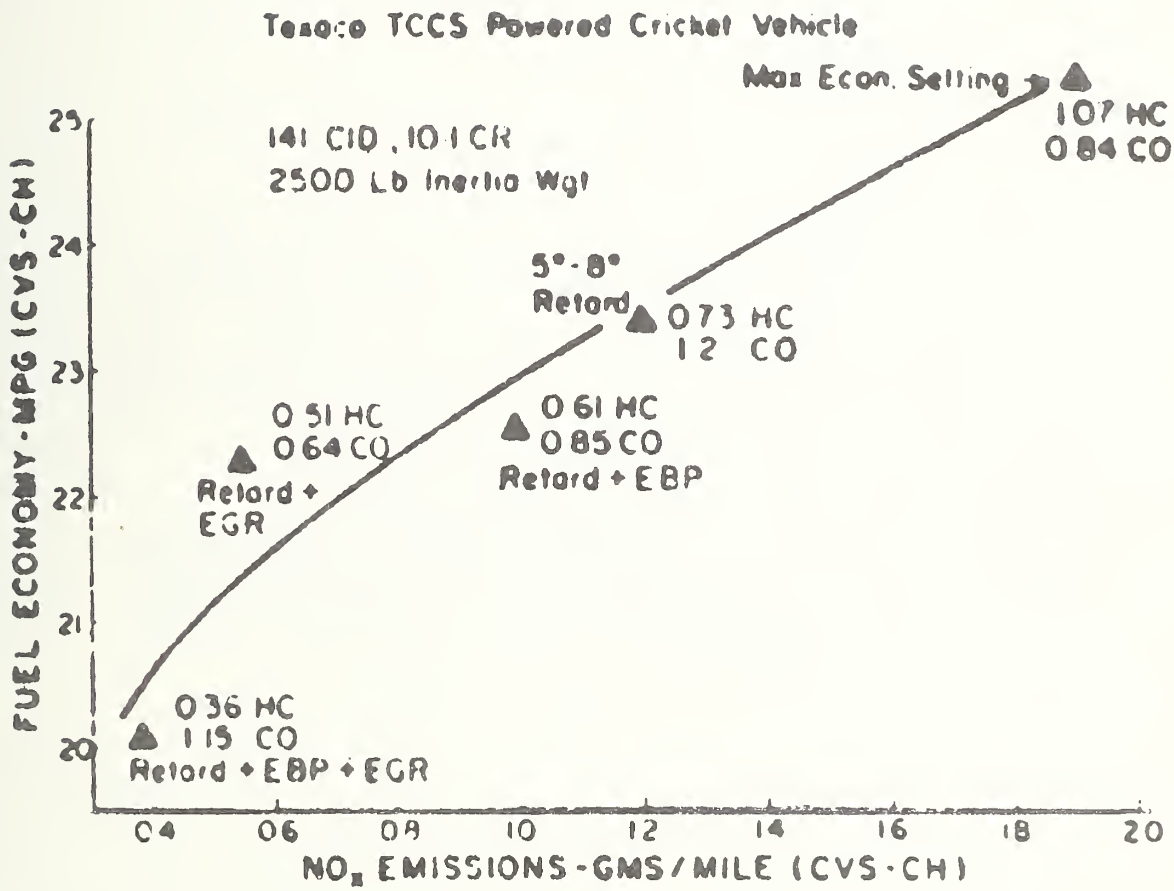
Source: Reference 13

TABLE 3-9. EMISSIONS AND FUEL ECONOMY TEST RESULTS FOR A 6.6 LITER PROCO ENGINE*

	INERTIA WEIGHT (lbs.)	RAR	DYNO HP	EPA-URBAN CYCLE EMISSIONS HC/CO/NO_x (gms/mile)	FUEL ECONOMY (mpg) URBAN/HWY/COMP.
PROCO	5000**	2.75	14.7	0.23/0.2/0.76	15.9/21.1/17.9
EPA*** average-1979	5000	-	-	-	12.3/17.7/14.2

Fuel Economy Percent Difference **** +29/ +19/ +26

* Source: Reference 15
** Note: Average of three cars
*** Source: Reference 8
**** Not corrected for HP/Weight - (The EPA 1979 average engine displacement equals 393 CID while the Ford PROCO displacement is 413 CID.)



Source: Reference 14

FIGURE 3-8. INFLUENCE OF NO_x EMISSIONS CONTROL ON FUEL ECONOMY FOR TEXACO TCCS SYSTEM

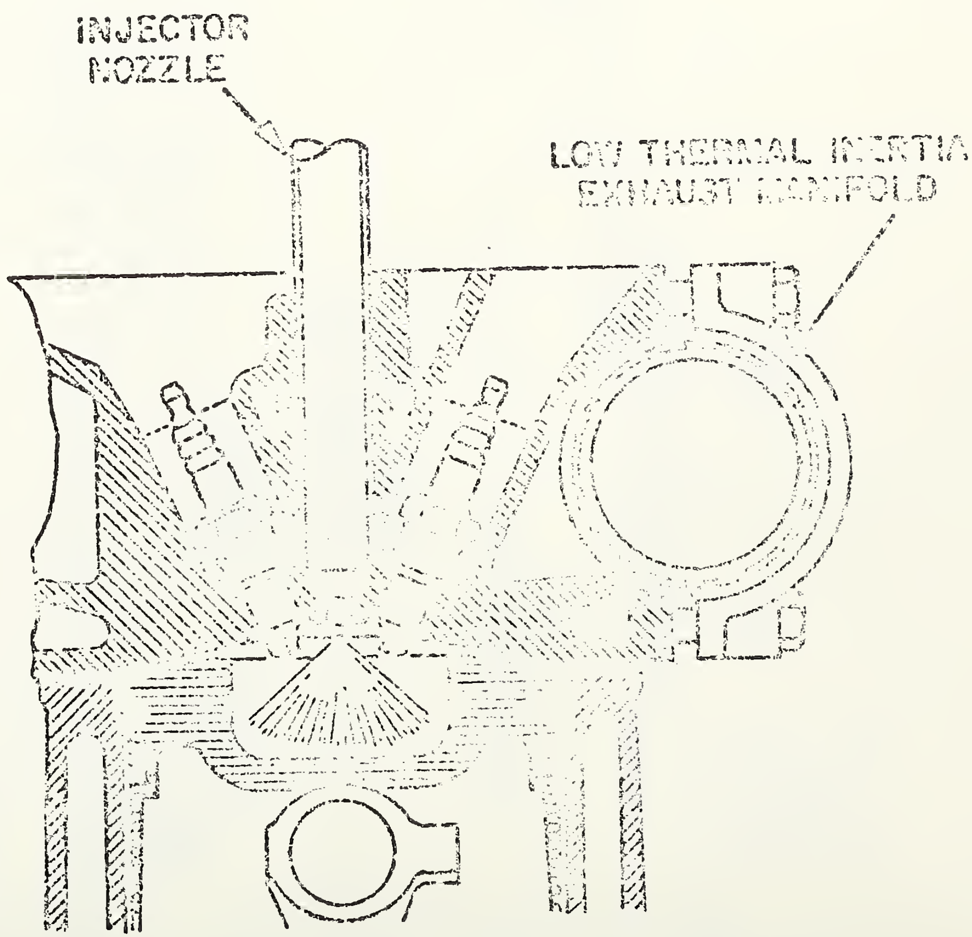


FIGURE 3-9. CROSS SECTION OF A TYPICAL PROCO COMBUSTION CHAMBER

d) an oxidation catalyst;

e) an experimental control system which features a modified speed density concept with variable air/fuel ratio control. The unique fuel spray and injection timing schedule results in full air utilization to provide maximum power capability comparable to a homogeneous charge engine;

f) a compression ratio of 11:1;

g) Intake Charge Control providing for intake charge throttling and for EGR preparation; and

h) exhaust manifolds of special tubular design for low thermal inertia.

The Ford Motor Company also has under development research prototype PROCO engines in the 5.0 to 5.8 liter range with current objectives to develop feasible production engines. Claims have been made that the research prototype shows a potential +20 percent improvement in fuel economy and still meets the 1981 Federal emission standard of 0.41/3.4/1.0; $HC/CO/NO_x$ gms/mile compared with 1978 conventional spark ignition engines calibrated to a 1.5/15.0/2.0 $HC/CO/NO_x$ emission standard. Claims are additionally made by the Ford Motor Co. that some of the design features included above are not feasible for volume production.

Those engines which use a third valve in conjunction with carburetion (Honda CVCC is an example) show a marginal potential for improvement in fuel economy, ranging from -10 percent to +16 percent under the 1979 Federal emission standards of 1.5/15.0/2.0 $HC/CO/NO_x$ gms/mile. (See Table 3-10.) Through continued research on combustion systems, Honda[16] has reported recently that a promising system called the "Branched Conduit System" was able to lower the HC and NO_x emissions further with no effect on fuel economy. No vehicle data is presently available to evaluate the extent to which the emissions can be lowered.

TABLE 3-10. 1979-EPA EMISSIONS AND FUEL ECONOMY FOR HONDA VEHICLE WITH CVCC's

	INERTIA WEIGHT (lbs)	CID cm^3	REAR AXLE RATIO	DYNO HP	URBAN CYCLE EMISSIONS $HC/CO/NO_x$ gms/mile	FUEL ECONOMY 1975 FTP URBAN/HWY./COMP.	COMPOSITE FUEL ECONOMY % DIFFERENCE*
	2000	91	3.88	8.6	1.13/5.7/1.7.	33/41/36	+10
	2000	91	3.88	8.6	1.05/4.9/1.79	33/45/38	+16
	2000	91	4.12	8.6	0.43/4.0/1.83	28/32/30	- 8
	2250	91	4.12	8.9	0.88/5.1/1.69	28/37/32	- 3
	2250	91	4.12	8.9	0.39/4.0/1.84	27/30/28	-15
	2500	107	4.38	9.1	0.87/5.0/1.59	26/34/29	+ 5
	2500	107	4.38	8.3	0.59/4.1/1.65	26/35/30	+ 8
	2500	107	4.12	9.6	0.51/7.6/1.70	24/28/25	-10
	2500	107	4.12	8.7	0.37/4.6/1.72	24/29/26	- 6
EPA AVERAGE	2000	84	-	-	-	28.97/38.87/32.72**	BASE
	2250	90	-	-	-	29.02/38.87/32.91**	"
	2500	101	-	-	-	24.56/32.71/27.66**	"

*Not corrected for rear axle ratio, dynamometer setting or engine size.

**Reference 8

3.4 ROTARY ENGINES

3.4.1 <u>General</u>

Rotary engines are high power density machines, because the ratio of working volume to total power section volume is high and the kinematics permit high speed. Rotary engines operate on the four-stroke cycle, intake, compression, combustion, and exhaust. Unlike the piston engine in which all operating cycles occur in the same area separated only by time, the rotary's operating cycles are spaced out according to the gas flow path (i.e., moving combustion chamber).

Mechanically, the rotary engine differs significantly from the piston engine. The cylinders and pistons are replaced by working chambers and rotors that are encased in a stationary housing. As a result, phases replace strokes, rotor seals perform the job of piston rings, and ports take the place of valves. Instead of a crankshaft, there is a mainshaft, the job of the crankpin being performed by eccentrics.

Rotary engines offer a high speed advantage and weight and volume savings of 10-20 percent and 30-40 percent, respectively, compared with conventional reciprocating engines. Regardless of maximum power output, the rotary engine has a far superior weight per horsepower ratio when compared with conventional reciprocating engines. The rotary engine is also far less complicated than the reciprocating engine, does not contain a valvetrain, since the valves are replaced by ports, and contains a fewer number of parts. This provides greater simplicity and lower cost. Due to the small number of moving parts, the required maintenance is limited, mechanical reliability is higher, and service life is extended. Other advantages include a) less power loss due to internal friction; b) no vibration balance of primary and secondary forces is required as it is in the reciprocating engine; c) no unbalanced inertia forces which impose a limit on the rotational speeds of reciprocating engines; d) reduced need for lubrication; e) wide fuel usage; and, finally, f) higher volumetric efficiencies and smoother power output since the engine produces positive torque for about two-thirds of the operating cycle as opposed to one quarter or less of the cycle in a four-stroke engine.

Despite all of these advantages, until recently, these engines (homogeneous charge) have not demonstrated fuel economies comparable to vehicles with piston engines. This is, in part, attributed to the emission requirement and to the rotary engine's combustion chamber configuration. Rotaries have a higher surface-to-volume ratio and require two spark plugs to provide faster flame propagation throughout the volume space, especially at the trailing edge where non-uniformities of the air-fuel mixture promote quenching. This quenching phenomenon also creates end gas problems just as with many reciprocating engines. Sealing between the rotor and trochoid surface and sides still requires attention at low speeds and low loads. Finally, the rotaries are set by their design to operate at fixed compression ratios; this feature restricts the consideration of state of art "variable stroke pistons," which are under development for reciprocating engines.

3.4.2 Homogeneous Charge Rotary Engines

A few passenger car, light truck, and van manufacturers have continuing programs on rotary engines. Since 1967, Mazda has mass-produced a 9.4 CR/125 HP single rotor engine in a 2750 lb. passenger car. Presently, the fuel economy of these vehicles (see Table 3-11) is below the sales-weighted average 1979 fuel economy for 2750 lb cars. Most recently, Mazda has reported that 23 mpg has been achieved on a 12A rotary engine in the company's RX7 sports car. According to Mazda, an additional 20 to 30 percent improvement is possible on these engines through better catalytic converter designs in conjunction with thermal reactors which enable these engines to be calibrated for leaner mixtures. To date, such measures would not place the rotary in a competitive fuel economy position.

If significant increases in fuel economy are to be made in the homogeneous charge rotary, considerable advancements will have

TABLE 3-11. MAZDA SINGLE ROTOR ENGINE FUEL ECONOMY

	INERTIA WEIGHT	REAR AXLE RATIO	TRANS-MISSION	EMISSIONS (gms/mile) $HC/CO/NO_x$	FUEL ECONOMY-mpg CITY/HWY./COMP.
Mazda	2750	3.91	M3	1.04/13.2/1.69	18/25/20
		3.91	M4	0.6/8.3/1.64	17/26/20
		3.91	M5	0.62/7.4/1.82	17/28/21
EPA 1979 Sales Weighted Ave.	2750	-	-		20.62/27.82/23.95
% Difference* (composite EPA Cycle)					-12.5% thru -17%

*Not corrected for rear axle ratio, dynamometer setting or engine size.

to be made in problem areas such as gas sealing, combustion chamber shape, thermal management, and port timings to improve their fuel economy and emission characteristics.

Some studies to date have been concerned with a) promoting faster combustion, b) reducing friction losses, c) leaner mixture operation, d) reducing heat losses, and e) modification of exhaust emission after-treatment controls to burn lean mixtures, to improve their characteristics.

A present disadvantage of rotary engines is still the higher emission of unburned hydrocarbons. Figure 3-10 compares the HC emissions of a 2-liter, 4-cylinder reciprocating engine with a rotary engine as a function of BMEP. After-treatment devices, such as a thermal reactor or a catalytic converter are employed to reduce HC. Wall temperatures of the rotor and the housings, and the gas sealing performance of the engine are factors which affect HC emission. The wall temperatures of the rotor and the housings, in particular, are related to HC emissions at the time of cold engine starting; it is very important to improve the warming-up characteristics of the engine.

3.4.3 Stratified Charge Rotary Engines

A few organizations have been evaluating prototype stratified charge rotaries for possible use in passenger cars, light trucks, and vans. The most noteworthy include Toyota Motor Company and Curtiss-Wright Corporation. The Toyota rotary prototype is a carbureted stratified charge rotary engine (SCRE) with two rotors of 600 cc each (2.4 liters on a reciprocating equivalent basis). This prototype develops 130 HP at 1000 rpm and a torque of 130-137 at 2000 rpm. Both spark plugs and glow plugs are required in the combustion chamber. The glow plug is located at the leading position in the chamber and the spark plug at the trailing position to improve the low-speed/low-load combustion. Since these engines are in the prototype stages of development, no data are available to quantify their potential for fuel economy. However, Toyota claims that a 15 to 20 percent increase in fuel economy can be anticipated over conventional gasoline engines for equal vehicles

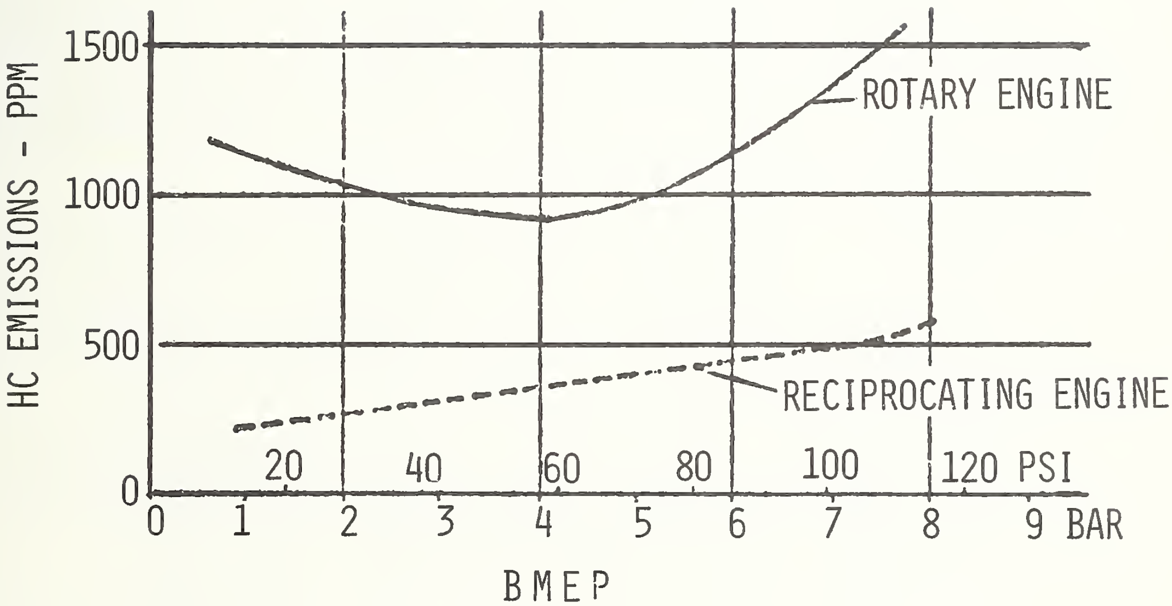


FIGURE 3-10. HC EMISSIONS OF ROTARY AND RECIPROCATING ENGINES AT 2000 RPM

under the Japanese 10 mode test and at the Japanese 1978 emission standard. The emission technology includes EGR, pellet oxidation catalyst, and air pump.

Curtis-Wright Corporation reported on its development effort on a direct injection, unthrottled stratified charge rotary engine.[17] They claim that unlike the rotary carbureted engine, the stratified rotary does not suffer at low-power/low-speed, and exhibits better breathing characteristics. These claims are based on steady state dynamometer measurements which showed specific fuel consumption equal to or better than pre-chamber diesels. In addition, Curtiss-Wright indicated that the stratified charge rotary has low emissions and multi-fuel capability.

Some preliminary computer vehicle simulation (VEHSIM) evaluations based on steady state engine dynamometer data of the latest Curtiss-Wright (CW) stratified charge rotary engine designs (RC1-60) were performed by DOT/TSC (See Figure 3-11). In Figure 3-11, the CW prototypes are compared to a turbocharged V-W prechamber diesel in a 2250 lb. Rabbit, a carbureted 140 CID/88 HP Ford engine in a 3000 lb. Pinto, and a 305 CID Chevrolet V-8 engine in a 4000 lb. vehicle. The three CW prototype engines denote various technologies:

a) the low performance engine at 8.5:1 compression ratio;

b) the mid-performance engine including an improved high speed fuel injection nozzle with an extended speed range from 4400 to 5000 rpm; and

c) the high performance engine representing further improvements (high trochoid surface temperature, etc.) achievable within the state-of-the-art.

In the evaluation, fuel economy potentials relative to three production engines cited in Figure 3-11 were computed by scaling the CW to the size of these three engines to produce equivalent vehicle acceleration times. Final computed fuel economies were corrected for differences due to fuel density effects. No auxiliary loads were assumed for the CW rotaries.

CW RC1-60 STRATIFIED CHARGED ROTARY
(Computer Evaluation Results Based on Dynamometer Data)

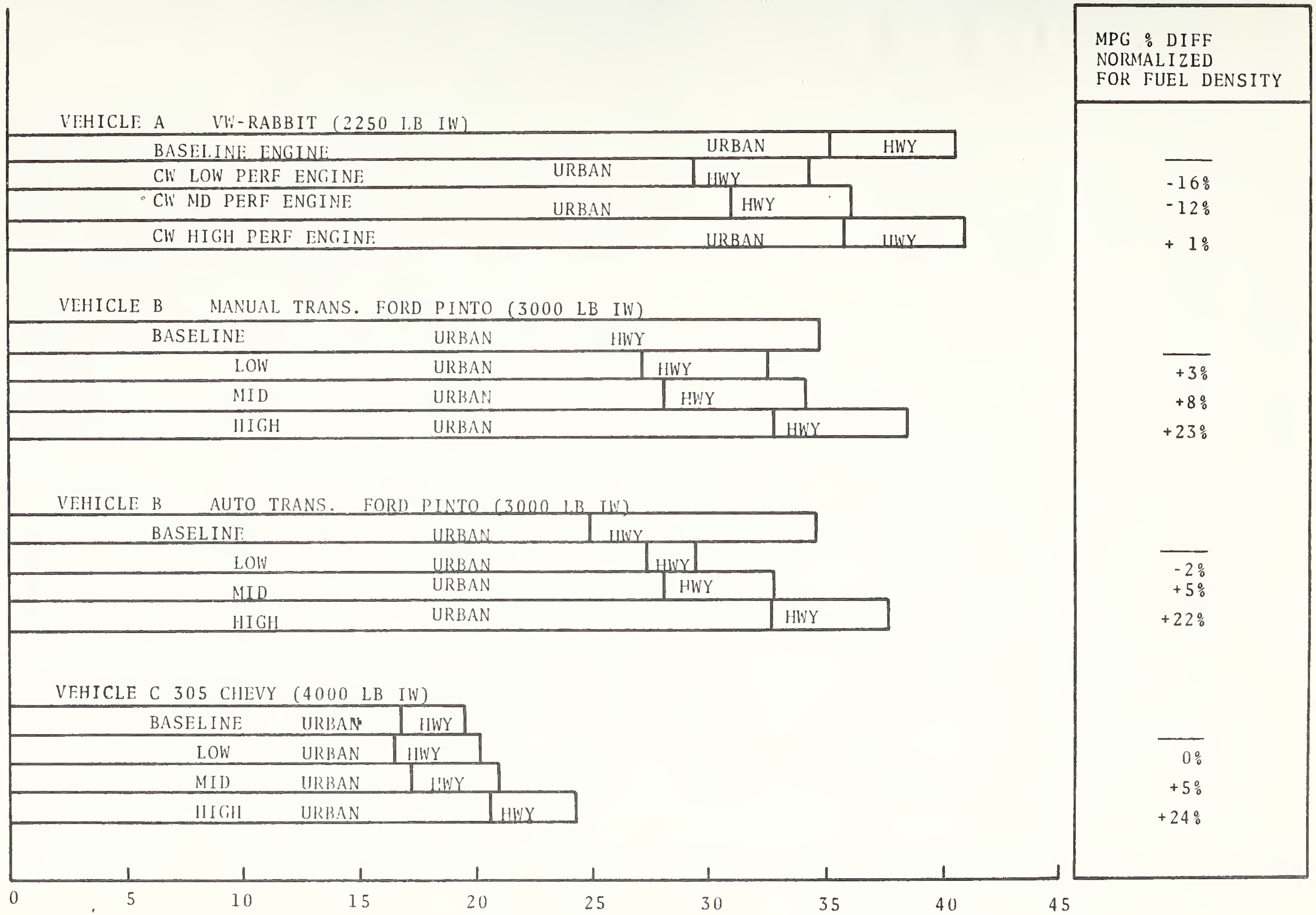


FIGURE 3-11. URBAN AND HIGHWAY CYCLE FUEL ECONOMY (NORMALIZED TO 638 LB/GAL GASOLINE)

The potential improvement in fuel economy of the stratified rotary for automotive use remains to be confirmed on experimental prototypes. The development of the rotary is still in its infancy, and much research effort is required to achieve an optimal configuration for automotive applications. Some areas which deserve immediate attention include a) optimum housing designs, b) nozzle orientation and spray patterns, c) spark plug type and orientation and rotor pocket system, d) apex sealing, wall temperature, etc., to consistently enable good efficiencies over the operating speed range of the engine and high torque stability at low speed. To date, no effort has been made to show their potentials under optimal electronic control schedules, nor have these engines been examined under conditions of turbocharging for automotive applications.

3.5 TURBOCHARGING

3.5.1 General

Turbocharging is a technology which will gain popularity in spark ignition and diesel engines in the 1980's. The demand for turbocharging results from a need to increase fuel economy and to improve emission characteristics of naturally aspirated engines. Current naturally aspirated engines operate at low loads. When smaller engines of smaller output are turbocharged, they can be operated at higher loads, generally under lower mechanical and pumping loss. Turbocharging is well developed for diesel engines, but the technology is less mature for automotive spark ignition engines. The main difference is that spark ignition engines operate with a throttle. Thus, air capacity varies widely over load at a fixed operating engine speed condition.

The turbocharging designs and the matching concepts for automotive spark ignition engines are continually evolving. For best tradeoffs regarding efficient and economical designs, the current approaches include a) wet turbocharging with wastegate control of the turbine in combination with a suction carburetor, and b) dry turbocharging with wastegate control and a single port or multi-port synchronized fuel injection system with a throttle positioned upstream of the compressor for improved engine responsiveness under increasing load operations.

Generally, a constant boost pressure is maintained by waste-gating and throttling over a large segment of load. At extremely low loads, the wastegate is closed, and boost pressure increases to maximum where it is maintained constant. (See Figure 3-12). In addition, the position of the turbine relative to the exhaust manifold is critical to provide a responsive engine and performance. For quick response and efficiency, the turbine should be mounted as close as possible to the exhaust manifold to minimize heat loss from the exhaust gas before reaching the turbine. However, this may not be possible since present day turbines are limited by the temperature of the exhaust stream. Currently, an improved material (Inconel) has been used by Volkswagen[18] which has lessened the thermal constraint.

Alternative concepts being investigated consist of utilizing a pressure release valve upstream of the compressor to regulate the amount of air going into the engine. With this concept, single point fuel injection is employed. A counter flow nozzle is incorporated in the turbine assembly providing braking to prevent over-run when the turbine is unloaded. The use of a variable nozzle turbine is also being examined as an alternative to waste-gating to allow the compressor to operate at its most efficient design point. This gives better fuel consumption, torque, and response behavior of the turbocharged engine.[19]

Most turbocharged spark ignition engines are limited to an effective boost ratio of 1.5, because at high effective compression ratios, knock occurs at the wide open throttle position. Knock sensors may be employed to retard spark, which lowers the effective compression ratio at the onset of knock. Another technology which may be used to minimize the influence of engine knock entails the cooling of the inlet charge with a charge air cooling system.

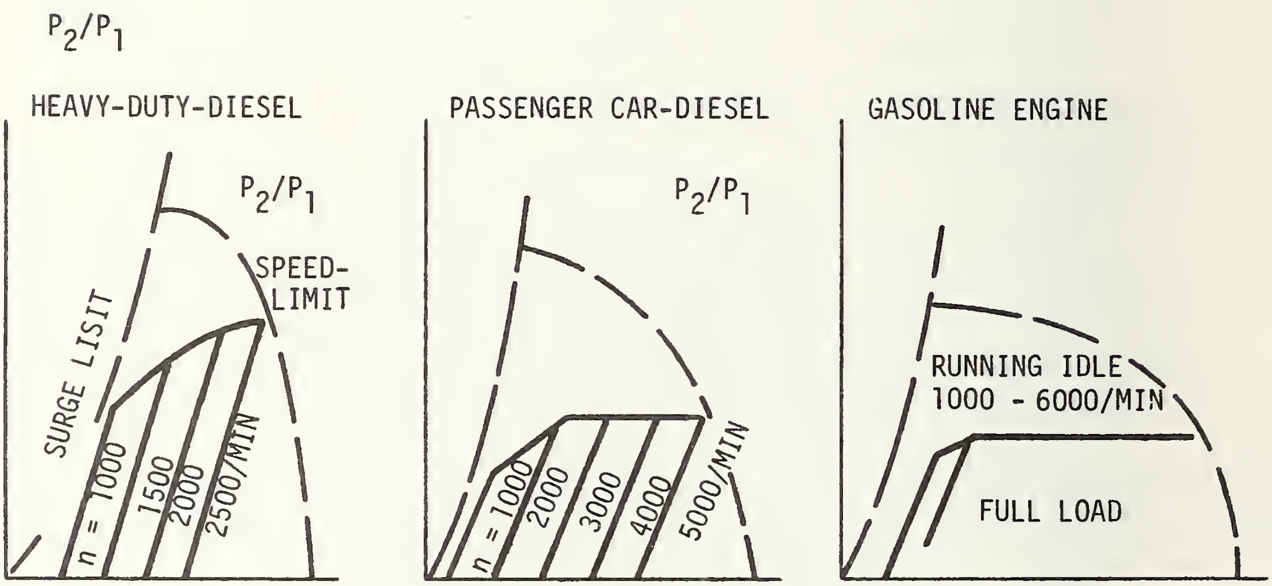


FIGURE 3-12. COMPRESSOR-MAPS OF A TURBOCHARGED HEAVY-DUTY-DIESEL-ENGINE, A DIESEL-PASSENGER CAR-ENGINE, AND A GASOLINE-ENGINE

In the following sections, vehicle data are given to demonstrate the potential of turbocharging to improve fuel economy. Then, the effects of several engine design parameters which influence knock are discussed. These include compression ratio, mixture condition, and spark timing.

3.5.2 Vehicle Data

Data[18] (Table 3-12) from a current DOT contract with Volkswagen show the fuel economy potential of turbocharging spark ignition engines under stoichiometric engine control. A 1.6 liter engine was investigated in a 3000 lb vehicle equipped with a 92.5/7.5 percent platinum/rhodium 3-way catalyst. Volkswagen's objective was to meet the 1981 passenger car emissions standard of 0.41/3.4/1.0 gms/mile of HC/CO/NO_x.

TABLE 3-12. FUEL ECONOMY AND EMISSIONS OF A TURBOCHARGED 1.6 LITER ENGINE IN A 3000 lb VEHICLE

ENGINE	URBAN CYCLE EMISSIONS (HC/CO/NO_x gms/mile)	FUEL ECONOMY (COMPOSITE)
Volkswagen - Experimental	0.19/3.27/0.71	33.0 mpg
1979 (Average)[8]	1.5/15/2.0 (std)	21.5 mpg
VW - Audi 5000	1.17/7.10/1.99	19.0 mpg

The Volkswagen 1.6 liter naturally aspirated engine yields 75 HP. The turbocharged version, equipped with the Garrett T-3 turbocharger, yields 100 HP with a boost ratio of 1.3, and a compression ratio of 8:1. The engine includes a two-stage carburetor located at the suction side (causing fuel wetting of the compressor), a three-way catalyst with feedback control for monitoring exhaust oxygen concentration, and a wastegate to by-pass the exhaust proportionately. The advantages of this sytem include improved cylinder-to-cylinder mixture control for CO distribution control and, to some degree, for favorable brake-specific fuel consumption at low speeds compared to the naturally aspirated version. Lower idle speed (lower fuel rates) was accomplished with the turbocharged version, attributed partially to better mixing of fuel and air caused by the compressor unit.

Efforts are underway at the Chrysler Corporation[20] to provide data on turbocharging their 225 CID engine under wastegate control. A Garrett T3-turbocharger is used in these investigations. Engine mapping data are being collected to determine the optimum boost pressure, the effects of fuel specification (91 RON and 98 RON fuels), knock limited timing, and the effect of cool versus hot EGR. Chrysler will use maximum cooled EGR to suppress knock, thus allowing a more advanced spark schedule under lower exhaust temperature conditions. (Garrett recommends a maximum exhaust temperature of approximately 1600°F.)

3.5.3 Knock

Engine knock can be reduced by lowering the compression ratio (Figure 3-13). This lowers the compression pressure and temperature in the cylinder. However, the engine's part load efficiency is also lowered.

Both the octane rating of the fuel and the air-to-fuel ratio affect the extent of pressure charging. Figure 3-14 shows that the knock-limited pressure charging can be increased by using a leaner mixture (fuel/air equivalence ratio $\lambda = 0.9$) and higher octane fuel (RON = 100) for various inlet charged temperatures.

Advanced spark timings, as shown in Figure 3-15, are limited by engine knock and boost ratio. Retarded spark timing reduces the effective compression ratio and runs counter to increased boost pressure.

3.6 COMPOUNDING

3.6.1 General

Compounding holds considerable potential for improved fuel economy. Loosely defined, compound engines are composed of two or more power producing systems. Examples of this technology include variable displacement systems, and exhaust energy recovery systems.

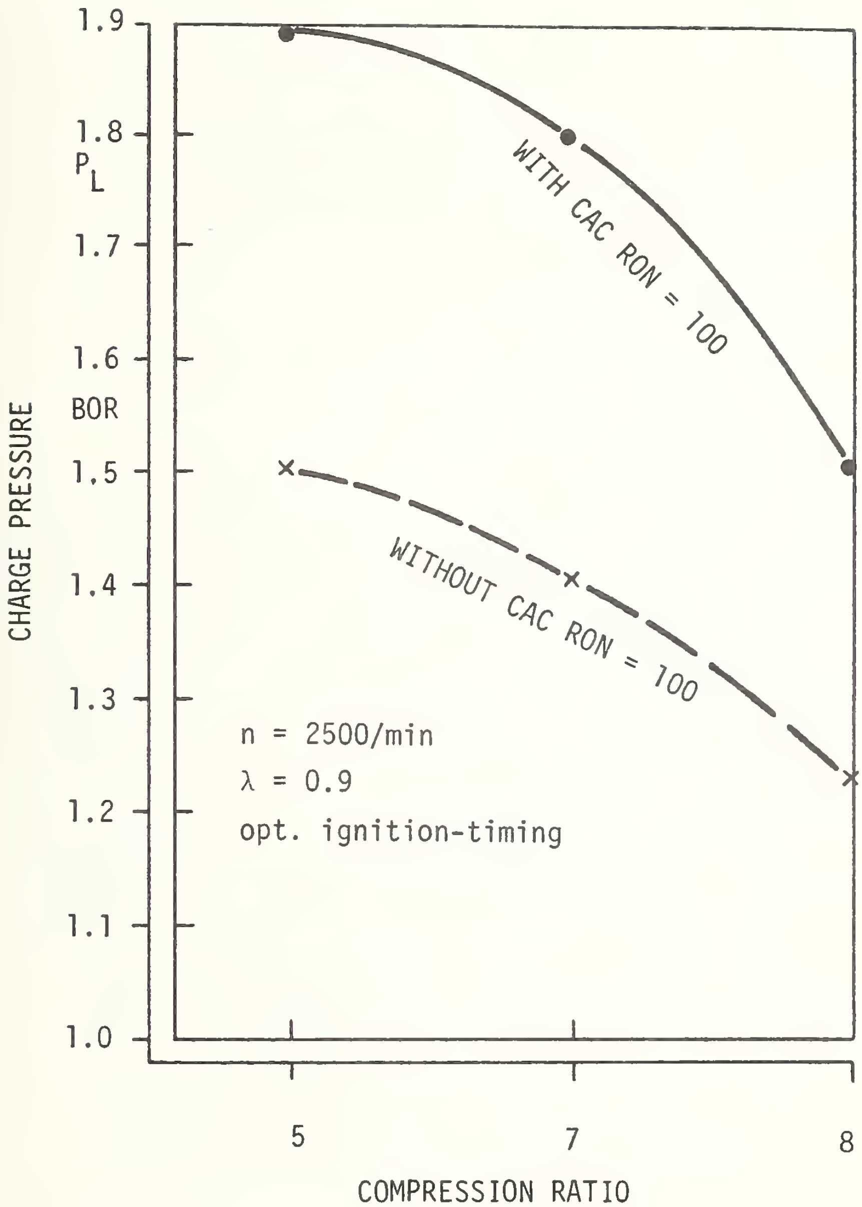


FIGURE 3-13. CHARGE-AIR-PRESSURE AS FUNCTION OF THE COMPRESSION-RATIO WITH AND WITHOUT CAC (CHARGE-AIR-COOLING)

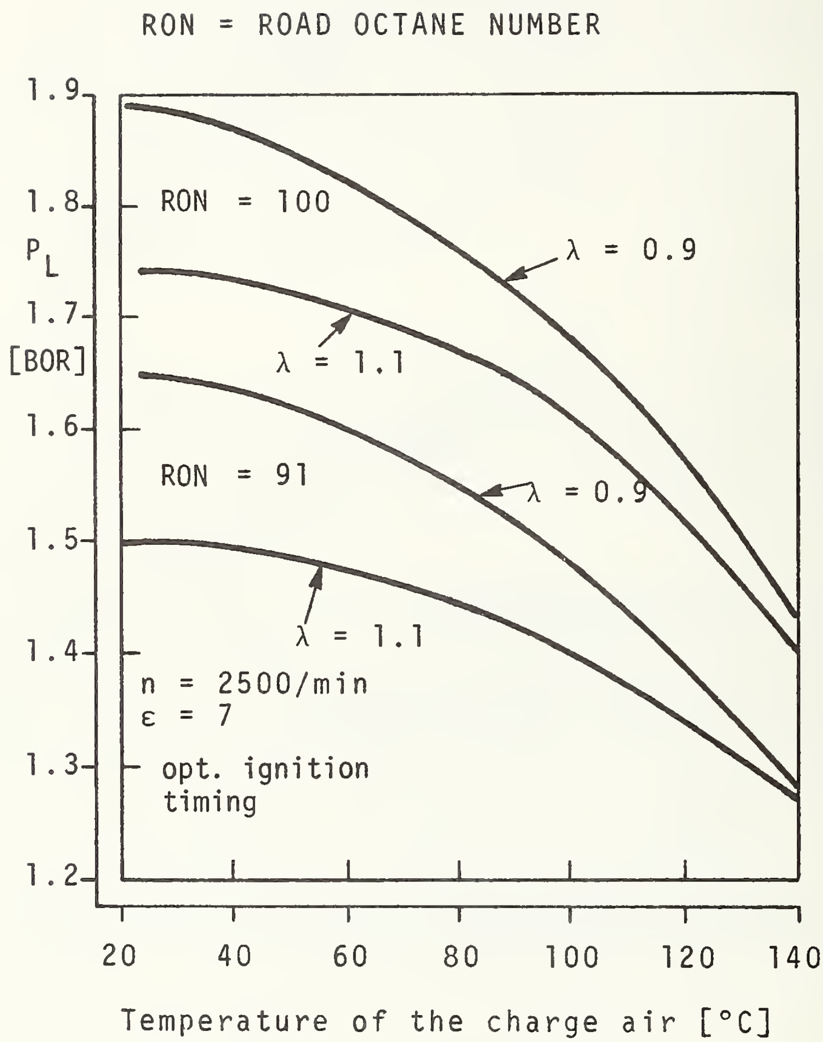


FIGURE 3-14. INFLUENCE OF CHARGE TEMPERATURE ON CHARGE PRESSURE (KNOCK-LIMIT) WITH DIFFERENT A/F RATIOS AND FUEL-QUALITIES

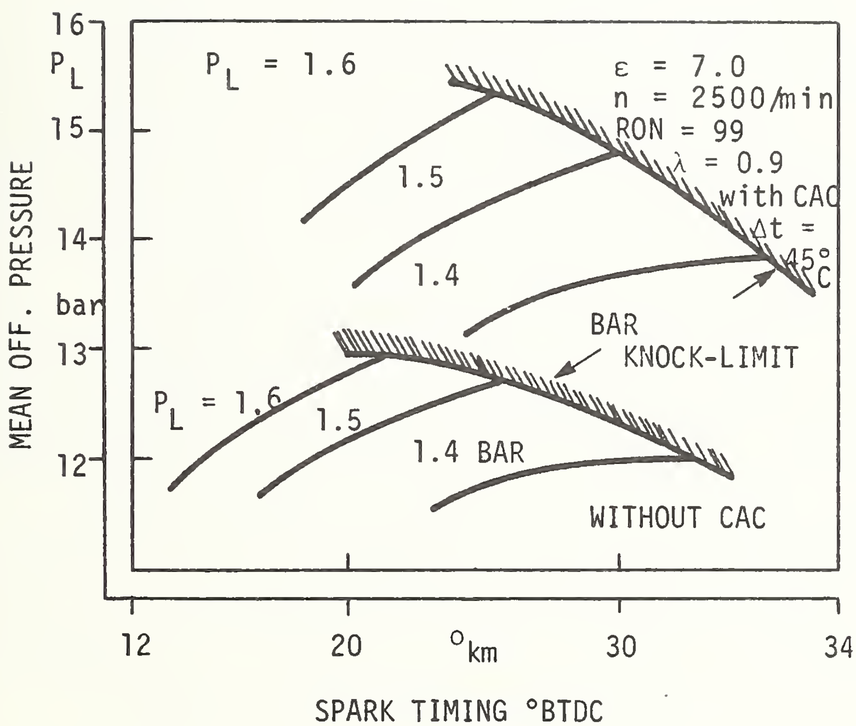


FIGURE 3-15. MEAN EFFECTIVE PRESSURE AS FUNCTION OF THE CHARGE-PRESSURE AND THE IGNITION-TIMING WITH AND WITHOUT CHARGE-AIR COOLING

3.6.2 Variable Displacement Engine

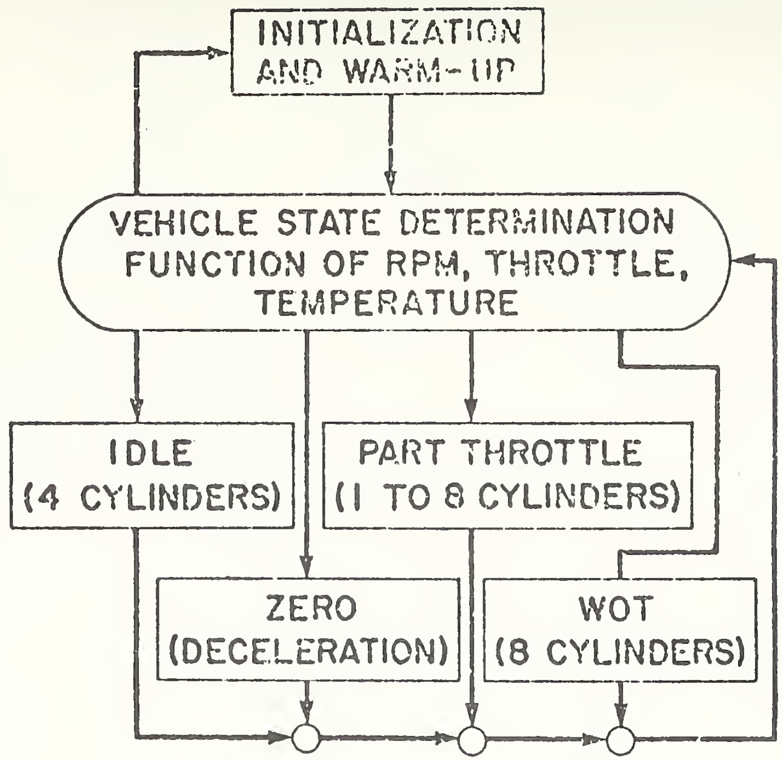
This concept can be accomplished in a number of ways. One approach currently under investigation involves the deactivation of certain cylinders during a vehicle's operation. The purpose of the deactivation is to cause the firing cylinders to operate under a more loaded condition to reduce throttling losses. Thus, with cylinder deactivation, the engine acts as a "small" engine under appropriate driving situations.

The deactivation of cylinders may be accomplished in a number of ways. One concept preferred by Ford Motor Company uses valve selectors to disable both intake and exhaust valves. This completely stops the fuel/air mixture from entering the cylinders. The cylinders that are shutoff are motored by the operating cylinders.

A system that eliminates the friction loss due to motoring inactive pistons includes a clutch mechanism in the drive shaft to remove the pistons totally from the operation of the engine. In this case, it is preferred that a single clutch be used to deactivate a group of cylinders. Although more complex to mechanize, this approach has the added advantage of reducing engine friction. Another approach would be to use two engines of different displacement coupled with a clutch mechanism.

A control strategy must be implemented for the deactivation of cylinders. One such strategy reported for an eight cylinder engine[21] consists of five major stages (see Figure 3-16). These include the initialization and warm-up phase where all cylinders are activated, an idle stage with half the cylinders activated, and a part throttle acceleration phase using all cylinders. Generally implemented strategy[22] deactivates a bank of cylinders when the torque value is 10 to 20 percent below the maximum torque that can be sustained by the lower number of cylinders.

Initial studies performed by Ford on a modified 5 liter engine operated on four cylinders showed that a fuel savings of 6 percent was achieved on the EPA Simulation Cycle for a 4.9 liter 16 truck engine.



SOURCE: Reference 21

FIGURE 3-16. CONTROL STRATEGY FLOW CHART

The effect of emissions on the fuel economy of this concept is unknown. This is particularly true in the case of HC emissions because of the on-again-off-again switching of the cylinders. Additional vibration and driveability problems have been observed by Ford, particularly at highway speeds.

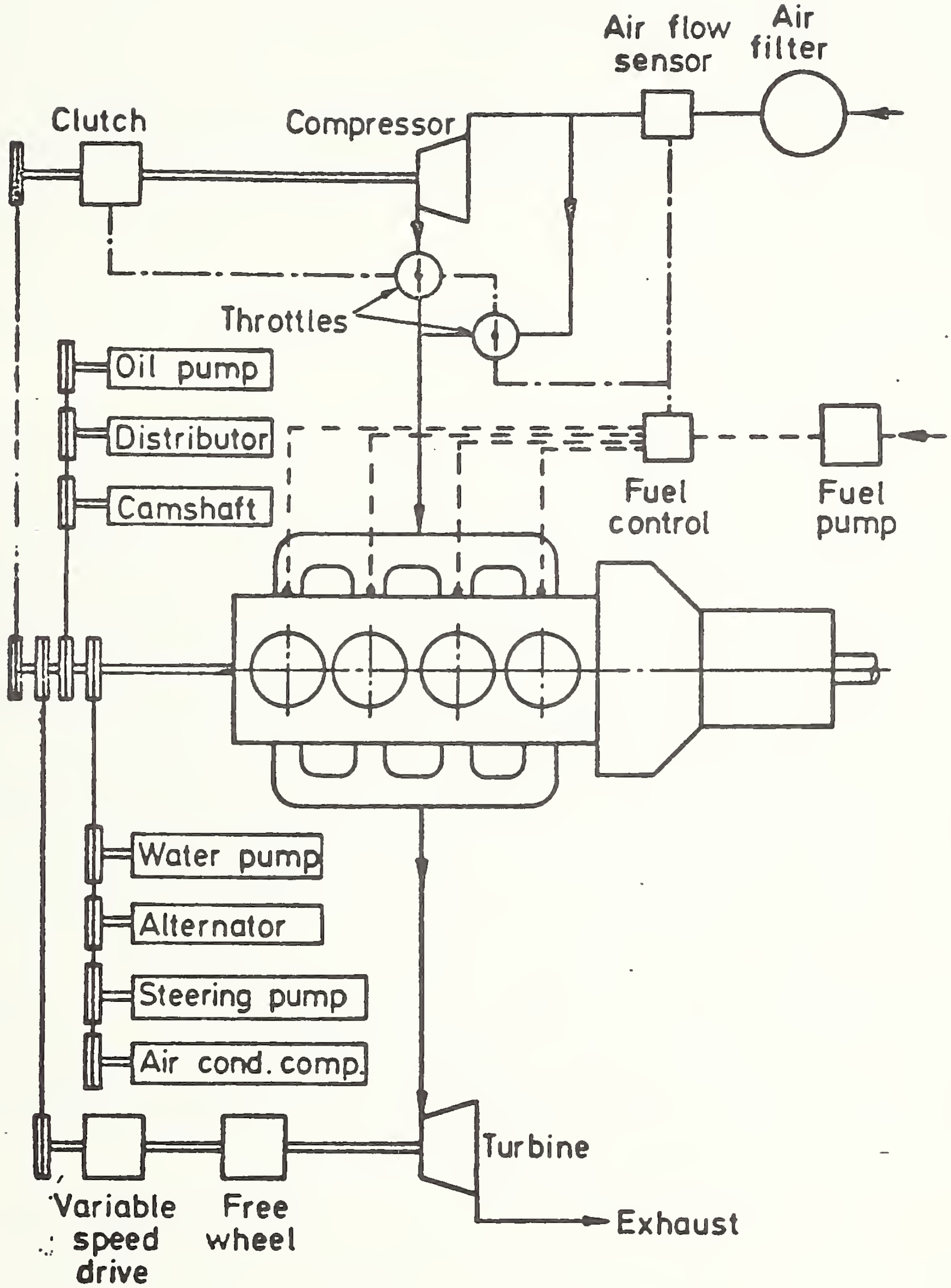
The Sandia Laboratories[23] have also conducted studies on a prototype variable displacement concept. The Sandia engine incorporates a linkage mechanism in the crankshaft which enables the stroke of the engine to be varied, and, therefore, the displacement of the engine as well. The linkage concept causes the clearance volume to be changed in proportion to the stroke. Thus, the compression ratio can remain constant or be varied. Preliminary computations using dynamometer data gathered for a 5-cylinder engine placed in a vehicle with a 2.76 rear axle ratio, 3-speed automatic transmission and a 3500 lb inertia weight have validated EPA urban fuel economies of about 24.0 mpg. The 1979 fleet average urban fuel economy for 3500 lb was 17.7 mpg.

3.6.3 Exhaust Energy Recovery Systems

The basic thrust of this concept is efficient use of the exhaust energy, and number of schemes have been proposed. The most noteworthy are the Rankine bottoming cycle[24], and turbocompounding. Both schemes are for turbocharged engines.

Limited studies have been performed on the bottoming cycle concept for diesel engines in heavy duty applications. However, this concept has yet to be evaluated for passenger cars with spark ignition engines. Design studies have been conducted on turbocompounding spark ignition engines. Figure 3-17 shows a schematic of such a concept.[25] In this system, energy is extracted directly from the exhaust turbine and applied to the main drive shaft via a variable reduction speed device. The net effect should be higher brake thermal efficiency and higher power output.

The main advantages of this concept include the use of retarded-timing for the use of NO_x control since exhaust energy can be recovered, and a reduction in lag time of the turbocharger



Source: Reference 25

FIGURE 3-17. COMPOUND ENGINE SCHEMATIC

during accelerations because of the additional coupling (the lag time can be reduced by 2 to 3 seconds). There still remain major obstacles which include the development of a practical, high ratio gear reduction system which is much larger for smaller engines than that required for larger engines, and the development of efficient exhaust turbines to improve the overall efficiency of the engine/ compound system.

Turbocompounding is still in its early stage of development. No practical system has yet been devised that can be tested to evaluate its potential for fuel economy with spark ignition engines.

3.7 VALVING

3.7.1 General

One of the fundamental design compromises of spark ignition engines is air handling or management. Currently, all spark ignition engines are designed for fixed valve timing, dwell, and lift, and they use a carburetor throttle plate to limit the engine fuel-air charge over the operating range. Valve events can be altered to change the characteristic shape of the torque curve. For example, by delaying the closing of the intake valve, the torque curve can be tailored to reduce the low speed torque with a corresponding increase in torque in the higher speed range. Also, the quality of fuel plays an important part. In an engine with all other factors remaining constant, a reduction in torque by change in valve timing results in a reduction in octane requirements. Since the low speed range determines the octane requirements of engines, future engines designed to operate at low speed and high torque will (for efficiency) require engine and fuel design to be tailored to this condition.

Since valve timing is fixed, present engines are generally optimized at a mid-speed performance which sacrifices both low and high speed performance. Complex transmissions help to rectify these sacrifices by reducing the requirement for off-design speed operation. However, they introduce problems of their own.

Another deficiency deals with the requirement to throttle the intake fuel-air mixture to achieve part load operation. The throttling process requires pumping work to be performed in raising the intake mixture from intake manifold vacuum. Thus, the overall pumping work increases as manifold vacuum increases. This results in greatly reduced part load efficiency over the engine's operating speed range. The efficiency reduction is also further aggravated by throttle enrichment requirements when operating at speeds below the valve-timing design speed, where overlap exceeds that dictated by the cycle flow dynamics.

Chrysler,[20] is currently investigating a valve-timing design which optimizes low speed engine performance and fuel economy.

3.7.2 Variable Valve Timing

Considerable improvements in performance and fuel economy of a four stroke engine would be expected if the valve timing were continually varied at each running condition. This can be accomplished by a variable cam shaft arrangement. The experimental data (Figure 3-18) accumulated for such an arrangement[26] for a variable inlet and exhaust camshaft installed in a Fiat 124 AC 1438 cm^3 engine illustrates the improvement in brake specific fuel consumption and torque achievable when the camshaft originally set for high speed performance is reoptimized at each engine speed.

A variable valve camshaft device also has a large impact on the exhaust emissions. This is illustrated in Figures 3-19 through 3-22 under hot cycle testing for various settings of the camshafts, which show that decreasing valve overlap decreases the emissions of hydrocarbons and carbon monoxide, but increases emissions of oxides of nitrogen.

3.7.3 Valve Sizing

Valve sizing is important to engine design and directly affects the performance and emission characteristics of an engine. Currently, engine valve sizing practices are based on measures of exhaust-to-inlet valve capacity and a Mach Index

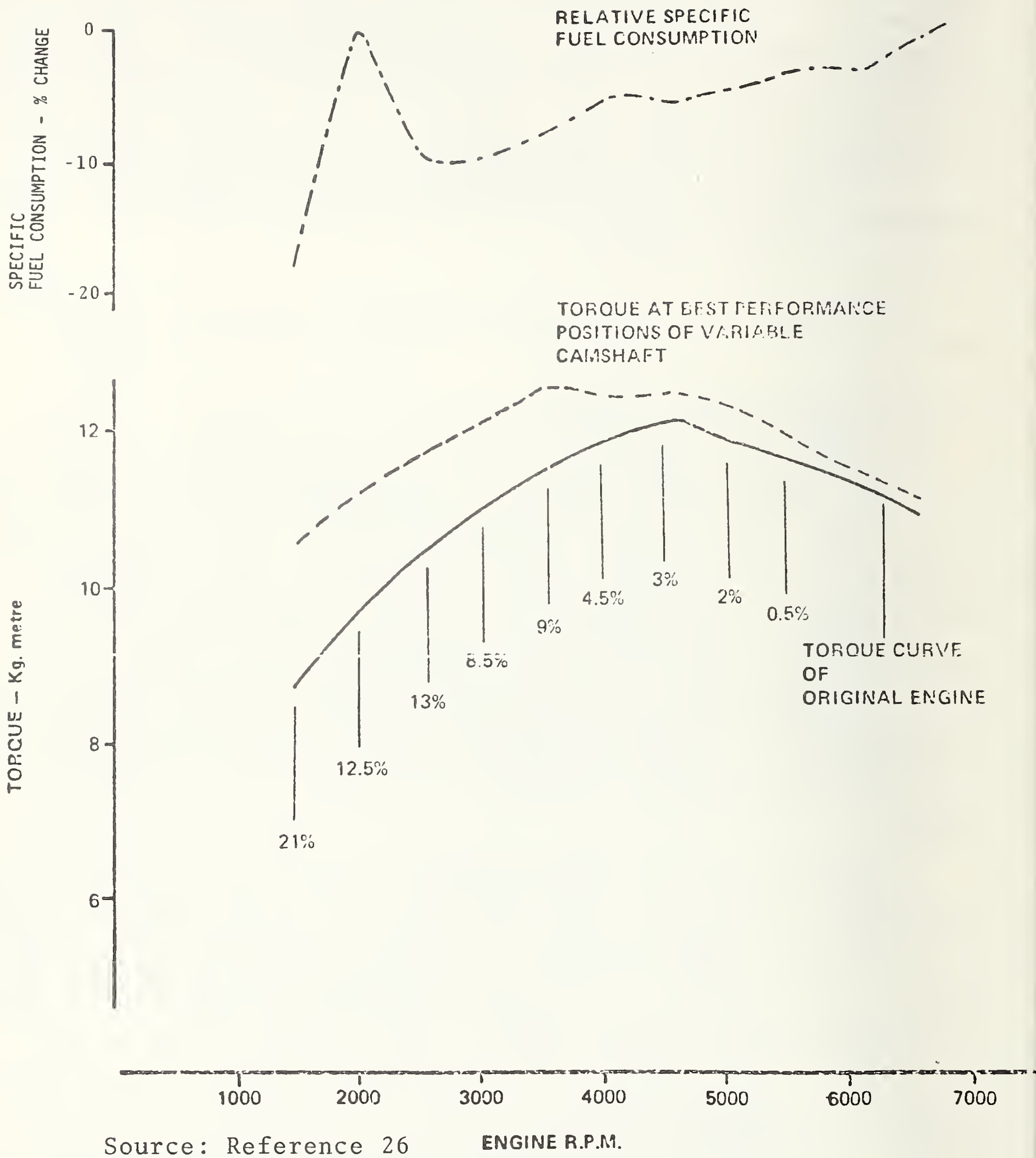


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FIGURE 3-18. IMPROVEMENTS IN TORQUE & FUEL CONSUMPTION WITH VARIABLE VALVE TIMING

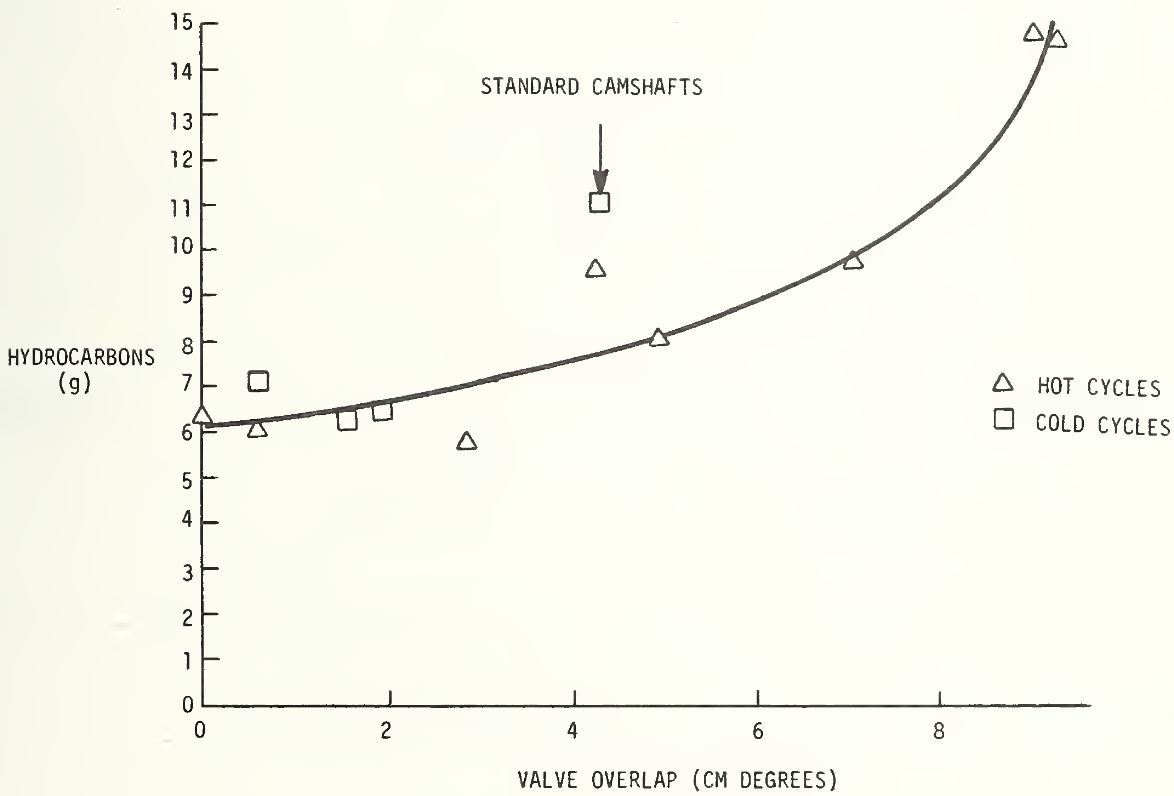

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FIGURE 3-19. EFFECT OF VALVE OVERLAP ON HYDROCARBON EMISSIONS DURING EUROPEAN CYCLE EMISSIONS TESTS

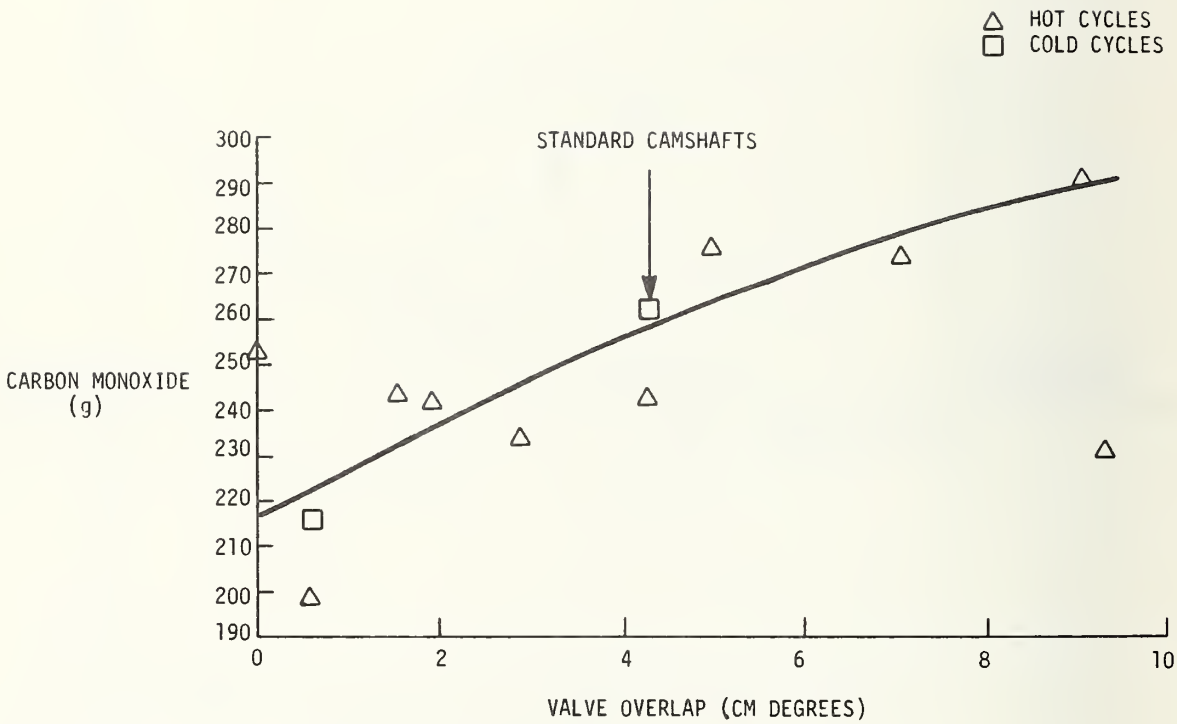

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FIGURE 3-20. EFFECT OF VALVE OVERLAP ON CARBON MONOXIDE EMISSIONS DURING EUROPEAN CYCLE EMISSIONS TESTS

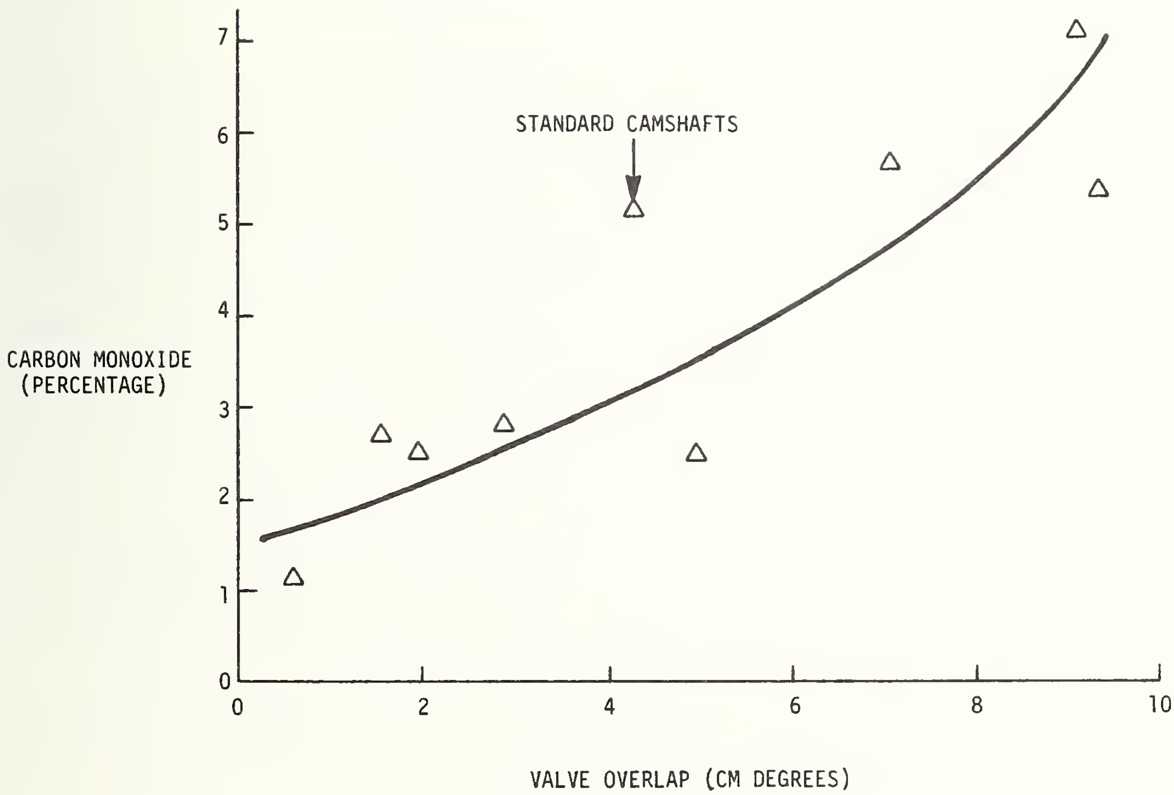

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FIGURE 3-21. EFFECT OF VALVE OVERLAP ON CARBON MONOXIDE EMISSIONS AT IDLING SPEED DURING EUROPEAN CYCLE EMISSIONS TEST

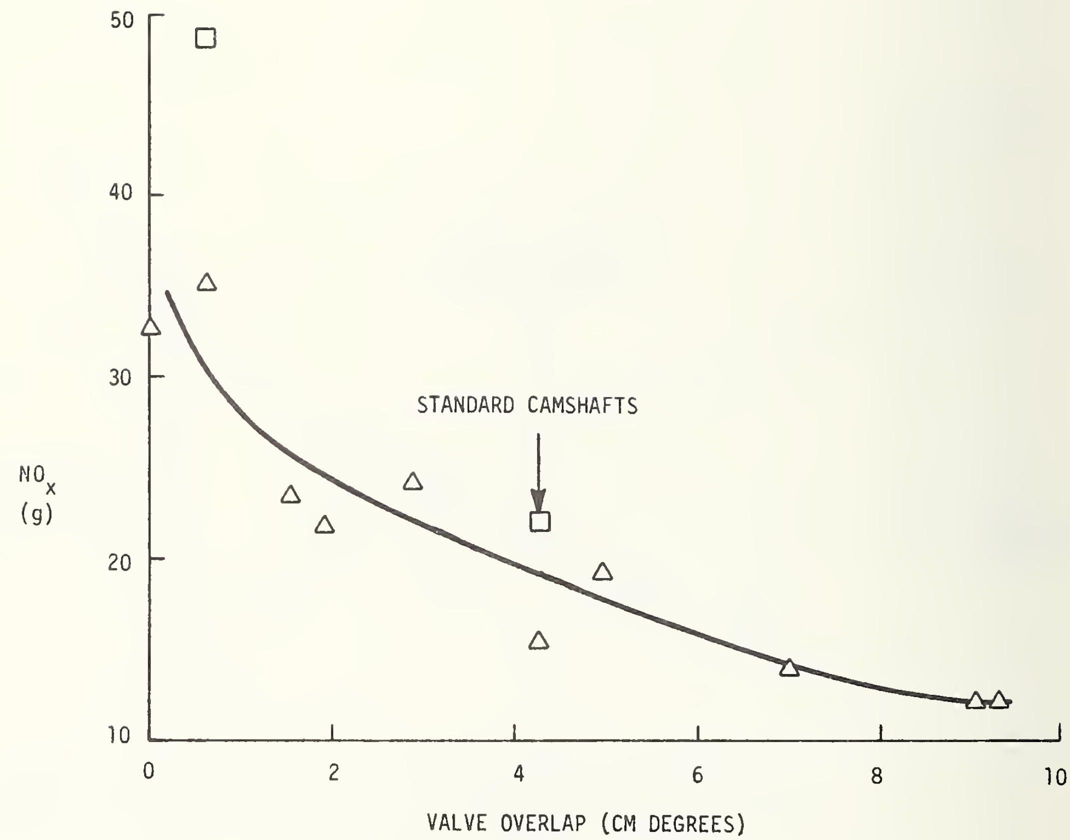

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FIGURE 3-22. EFFECT OF VALVE OVERLAP ON OXIDES OF NITROGEN EMISSIONS DURING EUROPEAN CYCLE EMISSIONS TEST number for an acceptable range of volumetric efficiency. Usually, exhaust-to-inlet valve capacity factors which range from 0.6 to 1 are considered. These result in exhaust valve sizes less than those required for the inlet valve. A recent experimental study shows[27] that a reduction in fuel consumption and emissions, accompanied by no loss of power and by the ability of the engine to run on lower than normal octane fuels, can be achieved with exhaust valves significantly larger than inlet valves, (i.e., exhaust-to-inlet valve capacity greater than 1). The mechanism is thought to depend upon an improved removal of combustion products from the combustion chamber, leaving a lower concentration of residuals during the compression stroke. It is also possible that the large exhaust valves introduce turbulence into the combustion chambers which influences the subsequent induction and combustion processes.

3.7.4 Inlet Valve Throttling

One of the fundamental inefficiencies of the spark ignition engine is caused by the use of an air throttle. Control of the engine output power is accomplished by controlling the amount of fuel-air charge entering into the cylinder during each engine cycle. The device commonly used to limit the engine fuel-air charge is the air throttle, better known as the carburetor throttle plate. The carburetor throttle plate restricts the flow of air-fuel mixture into the engine by acting as a flow restricter at the entrance of the intake manifold. Thus, a charge of reduced density is drawn into the cylinder during the intake stroke. This induction procedure results in lost work that is not recovered.

The most efficient way of controlling the fuel-air charge would be an induction process without a pressure drop. In practice, this can be approached by closing the intake valve when a sufficient fuel-air charge has been inducted into the cylinder (control of dwell), which requires a variable valve opening and dwell mechanism.

A related design concept known as inlet valve throttling[28] accomplishes throttling by control of the inlet valve lift. The pressure drop losses that occur with inlet valve throttling are reduced when compared with the conventional throttle plate. At the onset of the intake process, the pressure upstream of the intake valve is near atmospheric rather than at a manifold vacuum condition. This reduces pumping work during initial stages of the intake process. Moreover, the location of the throttling and its associated flow turbulence is at the inlet valve. The pressure drop at the inlet valve sets up high speed turbulent flow. This has been shown to enhance vaporization, fuel atomization, and distribution of the mixture in the cylinder chamber. To date, satisfactory spark ignition engine operation has been demonstrated at fractional loads with extremely lean mixture ratios (air-fuel ratios in excess of 20:1) where cyclically uniform and repeatable characteristics have been obtained. Other benefits included reduction of carbon monoxide emissions.

3.8 ENGINE SIZING

3.8.1 Engine Scaling Effects

Engine scaling effects have been extensively developed by C.F. Taylor.[29] The general conclusion reached was that engines of geometrically similar design have the same performance in terms of brake specific fuel consumption and brake horsepower per unit piston area as a function of load and speed, when load is defined as brake mean effective pressure, and speed as average piston speed.[30] Table 3-13 presents some scaling relationships and Table 3-14 shows the engine dimensions for a six cylinder engine scaled to an eight cylinder engine. As indicated, a six cylinder engine rescaled to eight cylinders with the same total piston area (and therefore the same power output) will require only 0.71 as much displacement, and, in a "straight eight" configuration, would have only 0.87 as much weight and volume. In a V-8 configuration, the weight and volume comparison would be more favorable to the eight cylinder engine, perhaps a ratio of about 0.8. This amounts to a weight savings of about 100 lbs. in the engine alone, based upon a bare engine weight of 500 lbs. for a six cylinder engine. As seen generally, engine size and weight decreases as the number of cylinders increases as long as geometric similitude in design

TABLE 3-13. ENGINE SCALING RELATIONSHIPS WITH GEOMETRIC SIMILITUDE

BORE	$B = B_o / \sqrt{X}$
STROKE	$S = S_o / \sqrt{X}$
DISPLACEMENT	$CID = CID_o / \sqrt{X}$
SPEED	$RPM = RPM_o \quad \sqrt{X}$
SURFACE/VOLUME RATIO	$S/V = (S/V)_o \quad \sqrt{X}$

X = Ratio of Number of Cylinders

TABLE 3-14. TYPICAL ENGINE DIMENSIONS

ENGINE DIMENSIONS	NUMBER OF CYLINDERS 6	8
BORE: INCHES	3.4	2.94
STROKE: INCHES	4.125	3.57
DISPLACEMENT: $INCHES^3$	225	195
SPEED: (MAX)	3600	4150
SURFACE/VOLUME: $INCHES^{-1}$	2.35	2.71

is maintained and providing that engine speed is not restricted. However, it becomes impractical to maintain exact similitude for very small cylinder sizes. The improvement in weight diminishes when the cylinder bore drops below two (2) inches in size.

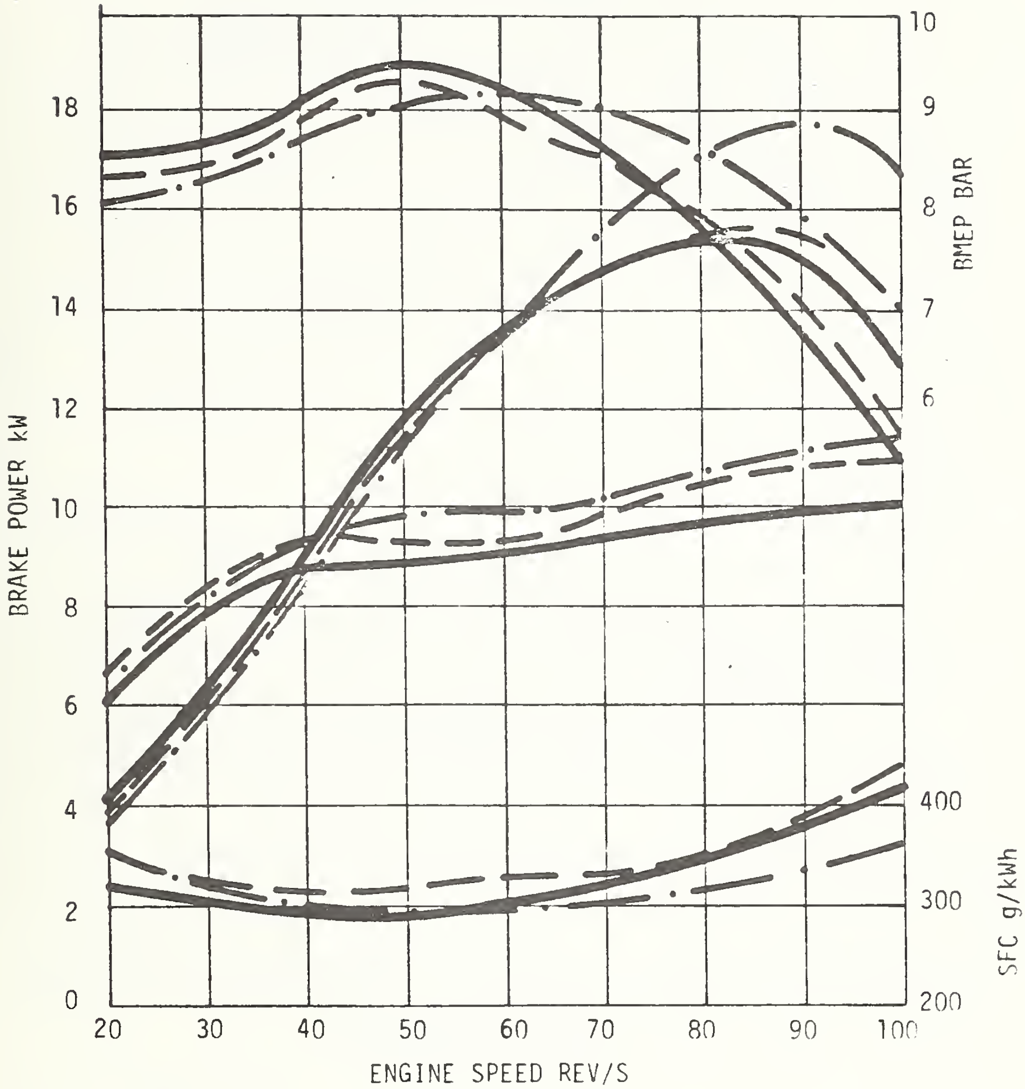
An engine rescaled from six to eight cylinders benefits from improved driveability and smoother operation. Taylor also shows, and it has been well demonstrated elsewhere, that smaller cylinders can tolerate a higher compression ratio without knocking due to the shorter flame travel in the cylinder.

Engines with a larger number of small cylinders must operate at higher engine speeds to obtain these advantages. Vehicle design characteristics also play an important role when an engine is rescaled to one with a greater number of cylinders. To maintain the same acceleration performance, transmission and rear axle ratio changes may be required to facilitate transfer of the higher engine speeds to those speeds required at the wheels. Such changes will affect the fuel economy.

There are, of course, disadvantages to repackaging engines to a greater number of cylinders. As cylinders become smaller, more complex details in engine design, reduction-gears, and valve gear, as well as materials of higher quality are required. The HC and CO emissions trade-offs are probably not favorable owing to the increased surface-to-volume ratio of the engines. The increased surface to volume ratio will also reduce engine thermal efficiency. Higher engine speeds will create more-noise, and wear rates will be greater. The importance of these disadvantages to the overall engine design considerations is not known. But, it should be pointed out that an engine size and weight advantage can be gained through the repackaging of engines into configurations with greater numbers of cylinders.

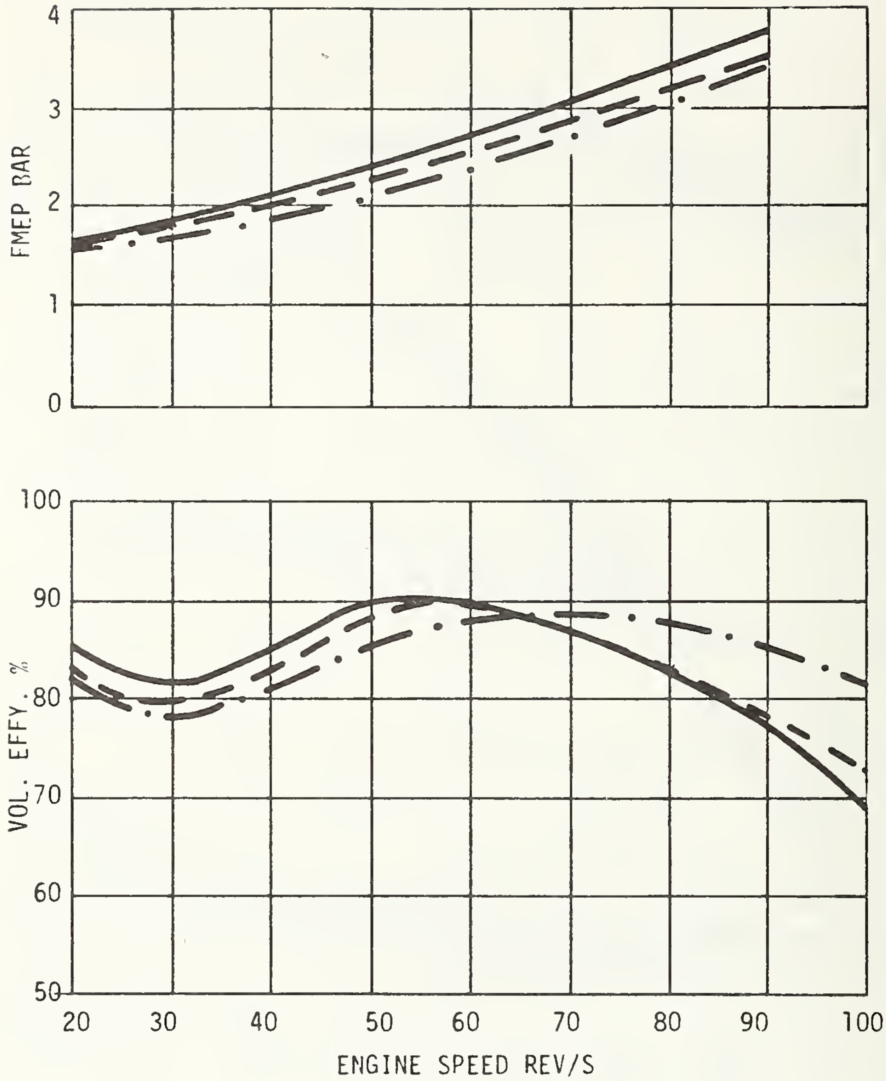
3.8.2 Bore/Stroke Ratio

The influence of two engine scaling parameters, bore and



BORE/STROKE		CONFIGURATION
———	0.84	80.95 mm x 96.5 mm Build (32 mm ∅ Choke Throttle Body, Small Valve Head)
– – – –	1.3	93.65 mm x 72 mm Build (32 mm ∅ Choke Throttle Body, Small Valve Head)
—·—	1.3	93.65 mm x 72 mm Build (36 mm ∅ Choke Throttle Body, Large Valve Head)

FIGURE 3-23. COMPARISON OF FULL LOAD PERFORMANCE



BORE/STROKE		CONFIGURATION
———	0.84	80.95 mm x 96.5 mm Build (32 mm Ø Choke Throttle Body, Small Valve Head)
– – – –	1.3	93.65 mm x 72 mm Build (32 mm Ø Choke Throttle Body, Small Valve Head)
—·—	1.3	93.65 mm x 72 mm Build (36 mm Ø Choke Throttle Body, Large Valve Head)

FIGURE 3-24. COMPARISON OF VOLUMETRIC EFFICIENCY AND MOTORING LOSSES

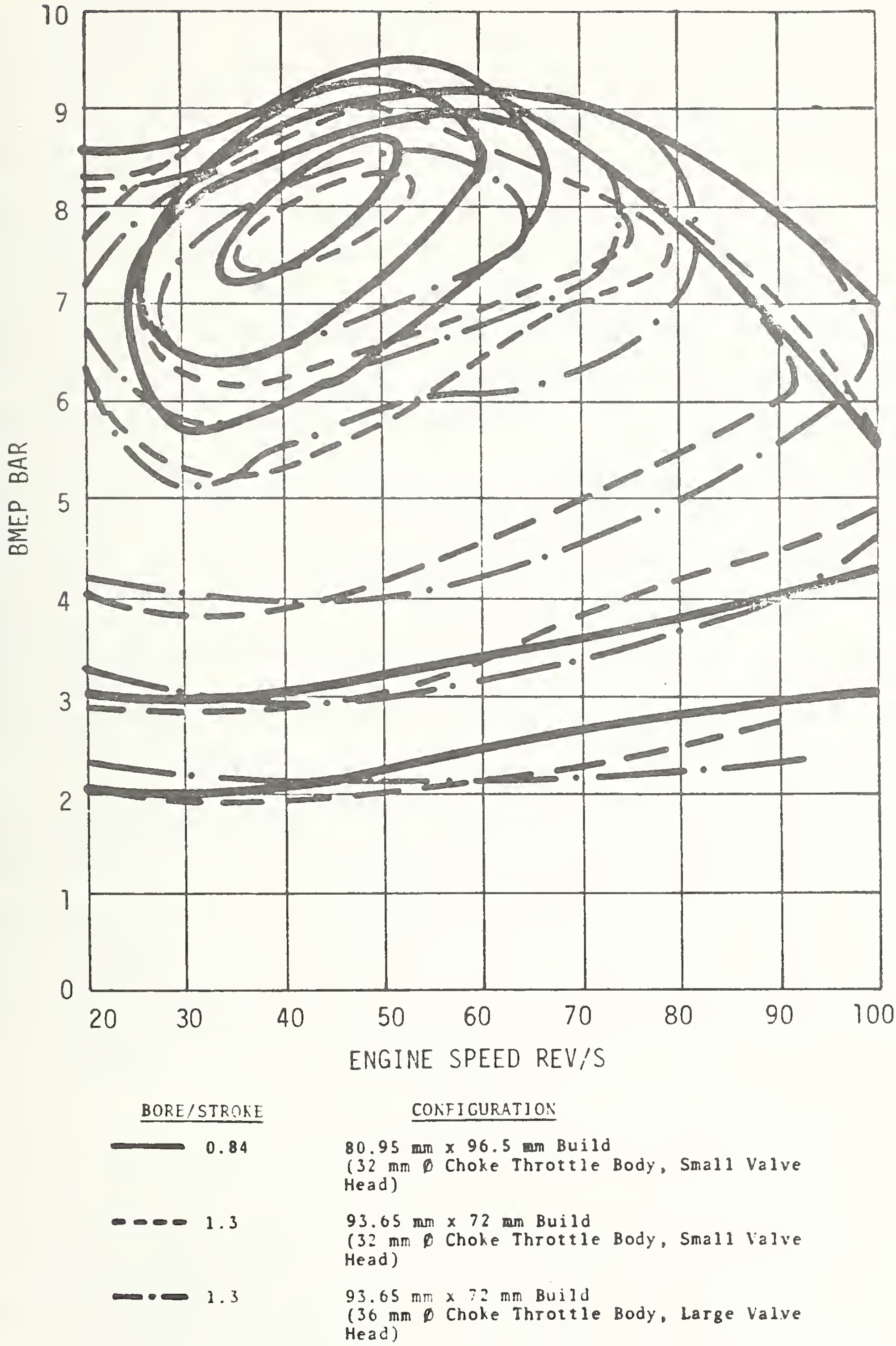


FIGURE 3-25. COMPARISON OF SPECIFIC FUEL CONSUMPTION CONTOURS: POINTS OF MINIMUM SPECIFIC FUEL CONSUMPTION g/kW.h

stroke, on engine performance is illustrated in Figures 3-23 through 3-25, where brake power and mean effective pressure, specific fuel consumption, engine friction, and volumetric efficiency are shown for single cylinder tests at ignition timings set for minimum spark advance for best torque (MBT). Two bore/stroke ratios are considered, the first at 1.3 and the second at 0.84. Data are also shown which consider a larger valve incorporated in the larger bore/stroke designs. The effect on wide open throttle performance closely demonstrates that the smaller bore-stroke design provides better fuel economy and increased power at lower engine speeds. This reversal in trend occurs for the large bore/stroke designs. Part of the smaller bore/stroke designs' poorer performance at high engine speed is attributed to increased friction and pumping losses. The latter effect is illustrated by the sharp drop off in volumetric efficiency with an increase in speed which is illustrated in Figure 3-24. The placement of larger valves provides increase in power, lower BSFC and increased efficiency at higher rotational engine speeds. Finally, Figure 3-25 illustrates the characteristic contours, fuel island, for the operating range of the single cylinder engine.

The fueling features of large bore/stroke configuration with the larger valves clearly illustrate that the fuel islands are rotated in the BMEP-plane to provide a flatter or more fuel efficient engine at higher rotational speeds. In fact, this characteristic is also evidenced to a lesser degree by a larger bore/stroke configuration with the smaller valves compared with the smaller bore/stroke configuration.

3.9 MIXTURE PREPARATION

Air-fuel mixtures are needed in all spark ignition engines. The mixtures are prepared either by carburetors or by direct or indirect fuel injectors. When a carburetor is used, the mixture is formed outside the cylinder. In the direct injection method, the mixture is formed inside the cylinder after the injection process starts. The port indirect injection method partially prepares the mixture outside the cylinder, and the process continues while the intake charge is drawn into the cylinder.

3.9.1 Carbureted Systems

The carburetor is undoubtedly one of the best known engine fuel delivery components. This device, in conjunction with its auxiliary subsystems and devices, is an extremely complicated mechanical marvel of springs, levers, bellows, linkages, solenoids and diaphragms. The primary function of a carburetor is to mix the air and fuel as homogeneously as possible and to provide a means for regulating engine power (via throttle plates). It must accomplish these tasks over a wide range of engine speeds and vehicle operating modes such as cold-start-warmup, idle, acceleration, deceleration, cruise, high altitude, and wide open throttle.

Current production carburetion systems provide relatively poor A/F ratio control, in the vicinity of ± 10 percent of the desired value, and as the system wears with mileage, the performance continues to degrade (± 15 percent of desired value). The problem lies primarily in four areas:

1. manufacturing tolerances and mechanical system wearability,
2. carburetor Venturi system meters fuel in proportion to volumetric air flow, not mass air flow,
3. induction thermodynamics in the intake manifold-plenum system, and
4. slow dynamic response of the carburetor Venturi system.

Problem areas 1) and 2) have been addressed through the implementation of electronic closed-loop feedback control. Problem areas 3) and 4), however, are extremely difficult to correct.

The function of the intake manifold or plenum is to take the fuel/air mixture delivered by the carburetor and distribute it uniformly to all cylinders. Unfortunately, a large amount of fuel in a liquid state adheres to the inner surface of the intake manifold, a problem known as manifold wetting. The occurrence is most prominent during cold engine warm-up modes. As the engine is throttled, and the manifold vacuum signal varies, the amount of fuel which vaporizes (fuel vapor density) from the manifold surface also varies, resulting in an extremely oscillatory A/F ratio being delivered to the combustion chamber. In addition to this, the poor dynamic response of carburetors to acceleration transients further complicates matters. The acceleration pump is a device which squirts raw fuel into the manifold in response to rapid throttle movement. Typically, fast throttle accelerations, especially from initial positions of nearly closed throttle, will result in a rapid inrush of air. The inability of the carburetor to follow this air signal with the proper amount of fuel causes a lean hold or engine hesitation. The acceleration pump attempts to cover this problem, usually with an A/F ratio disturbance on the order of ± 1.5 ratios. The net effect of problem areas 1) and 2) dynamically interacting is the ± 10 percent accuracy number stated earlier.

The potential for electronic fuel control is not great for carbureted systems. Even the electronic closed-loop feedback control system has only the capability of correcting relatively slow changes in system calibration caused by changes in fuel, temperature, atmospheric pressure and aging of components. Electronic feed-forward fuel metering is more readily applied to fuel injection systems, and it is believed that as the fuel economy standards become more stringent, these systems will eventually eliminate carburetor-based fuel delivery systems.

Dispersion, evaporation, and mixing are the three steps for any element of liquid fuel and gaseous air to combine. Sometimes, these steps either occur simultaneously or they overlap. Ordinarily, good dispersion helps to shorten the evaporation process and to increase the mixing rate. For good mixture preparation, these processes should occur as fast as possible. In ordinary carburetors, with the aid of a large exhaust heated "hot spot," good dispersion and evaporation become possible. In injection methods, high injection pressure improves the atomization. Shell is investigating an approach to improve atomization in which the fuel is heated to a gaseous state and then it is mixed with air. The Vapipe, as it is called, uses an aerodynamically designed hot tube bundle to give high heat transfer to the fuel, but low heat transfer to the air. Under certain conditions, the mixture consists of a finely dispersed fog of gasoline droplets. Another way to improve the atomization is to electrostatically charge the fuel to cancel out the surface tension effect when the droplets fly apart by charge repulsion.

A means by which fuel can be atomized is the introduction of the air/fuel mixture under sonic velocities to the induction system. A typical example of this method of fuel system control is the Dresser Industries' sonic carburetor. This system consists of a fuel injector which introduces fuel upstream of the throat of an inductor element under intense mixing conditions. Basically, the inductor consists of a variable convergent/divergent nozzle (or a critical flow venturi) which enables it to maintain sonic velocities at its throat over a large range of engine running conditions. This type of fuel control system eliminates the use of a throttle plate.

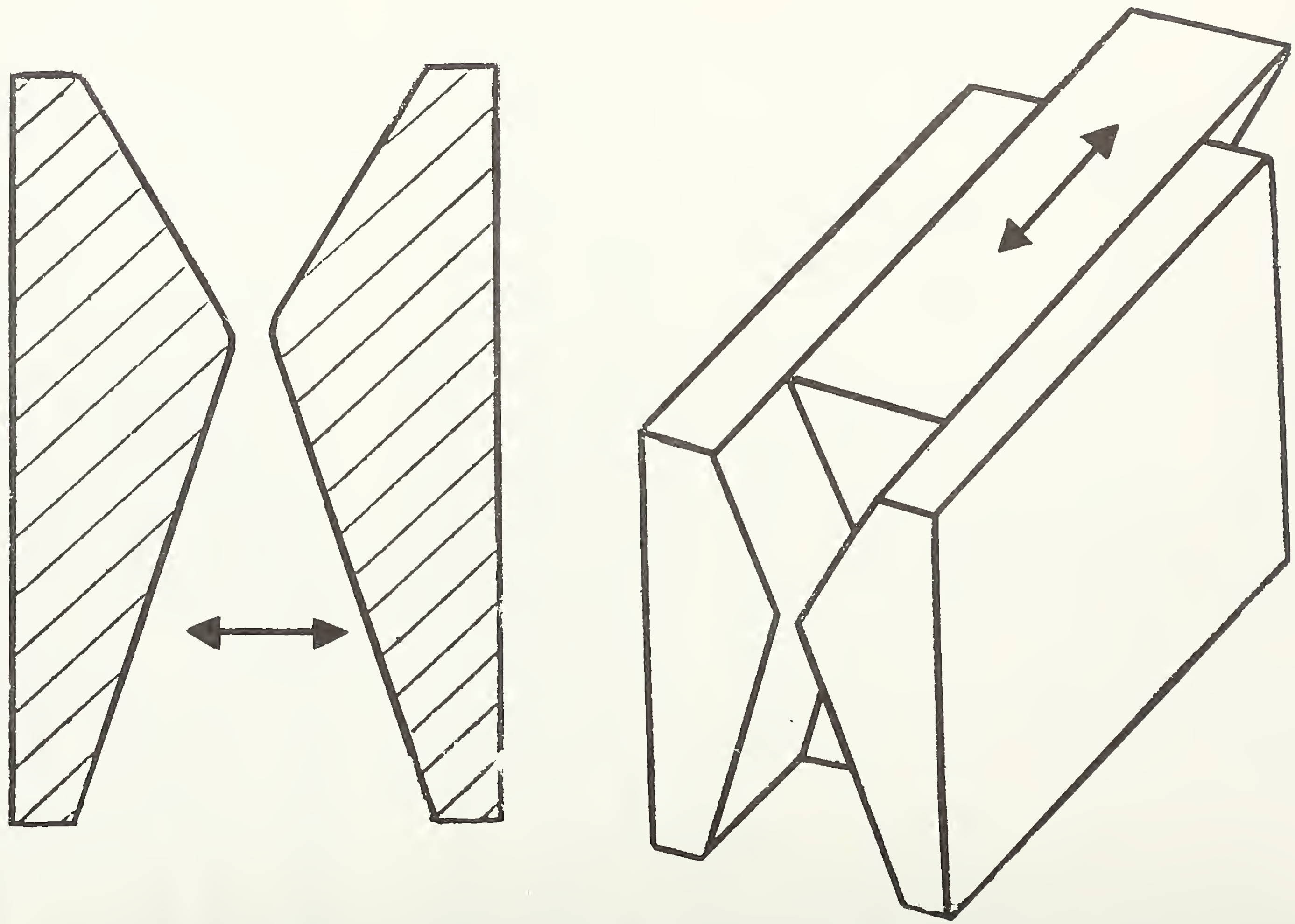
Figure 3-26 shows two inductor designs developed by Dresser Industries. The first (Model II) consists of two shaped jaws which are modulated by sliding apart or by pivoting around a top pivot point. The second (Model III), utilizes two fixed jaws with a slide in between to modulate the throat. Both models can be mated to a variety of metering systems.

3.9.2 Fuel Injection Systems

Fuel injection systems can be incorporated either in the manifolding as a replacement for the carburetors or directly into the combustion chamber. For the conventional spark ignition engine, the fuel injector system is positioned in the manifold whereas stratified charge engines utilize injectors positioned directly in the combustion chamber. The injector is preferred over carburetion because of its ability, via electronics, to control the air-fuel mixture ratio which enhances the reduction of emissions and improvement of fuel economy and driveability.

MODEL II - SLIDING TYPE

MODEL III - RECTANGULAR TYPE



Source: Reference 31

FIGURE 3-26. DRESSERATOR INDUCTORS

Typical examples include: a) single point or central fuel injection throttle-body type where one or two injectors are placed upstream of the throttle plate, and b) multi-point fuel injection which provides an injector for each cylinder positioned upstream of the inlet valve. Besides these two basic types, further characterization is based on its fueling function. Typical examples include, pulse modulation, synchronization, etc. These examples are synonymous with a multi-point fueling injection system.

Most existing fuel injection systems for spark ignition engines are compromise designs. These are low pressure port injection types that deposit a pool of fuel on the inlet port and valve. These systems are untimed in that either several injectors fire at the same instant or the injector flow is continuous.

These types of fuel injection systems achieve a well controlled cylinder-to-cylinder air-fuel ratio, but do not enhance fuel-air mixing. Good mixing is essential to consistent, complete and rapid combustion, particularly at part-throttle. One means to accomplish this is to position injectors in the ports of the inlet valve. Fuel sprayed behind the inlet valve in this manner gets well mixed with the air when it passes through the inlet valve opening and flows into the cylinder with the incoming air. The chances of fuel rich or lean pockets in the chamber are reduced by the common entry of the fuel and air.

A more optimum fuel injection scheme for spark ignition engines would employ timed injection and high pressure fuel delivery. This would improve the atomization and vaporization of the fuel and facilitate good fuel-air mixing.

Mixing time can be increased by moving the injectors upstream in the inlet manifold runner. With injection timing delay compensation if necessary, the more desirable course would be to atomize the fuel sufficiently at the time of injection to obviate the need for increased mixing time.

One advantage of the fuel injection system is to improve the part-load fuel economy due to better part-load combustion. This is particularly the case if the fuel injection scheme were combined with a good turbulence generating scheme. These two features combined would make the engine tolerant of dilute mixtures. Dilute mixtures mean improved engine efficiency and reduced emissions, especially NO_x emissions. Better mixed and more uniform fuel-air mixtures also mean reduced HC and CO emissions. The big HC and CO emissions payoff for this system is during cold starting.

The first two minutes of the FTP account for up to 20 percent of the total urban cycle HC and CO emissions. The principle causes of these high emission rates are excess fuel and incomplete combustion. Both of these factors are mainly related to the fuel induction system. A cold carburetor and inlet manifold simply cannot effectively vaporize fuel. The simple expedient is to use excessive fuel and reduce manifold pressure with the choke plate to secure a sufficient amount of fuel vapor to get a combustible mixture. The extra fuel passes out of the engine partially burned or unburned. The thermal causative factors of cold engine emissions are caused much more at the induction system than in the combustion chamber itself. Combustion chamber temperatures rise far more rapidly than does the temperature of the inlet manifold (which is usually heated by exhaust gas). One solution to the cold start emissions problem is a fuel induction system which does not rely upon high temperatures to vaporize the fuel. High pressure fuel injection may be one choice. Note that if low start-up emissions can be attained with the system, the emissions benefits can be traded-off against improved fuel economy and greater emissions elsewhere in the engine design or calibration. There are other means which can be employed to reduce cold start emissions. These include the use of separate induction systems designed specifically for cold start operation only, and the use of a secondary fuel, such as propane, for cold start augmentation.

3.10 COMPRESSION RATIO AND KNOCK

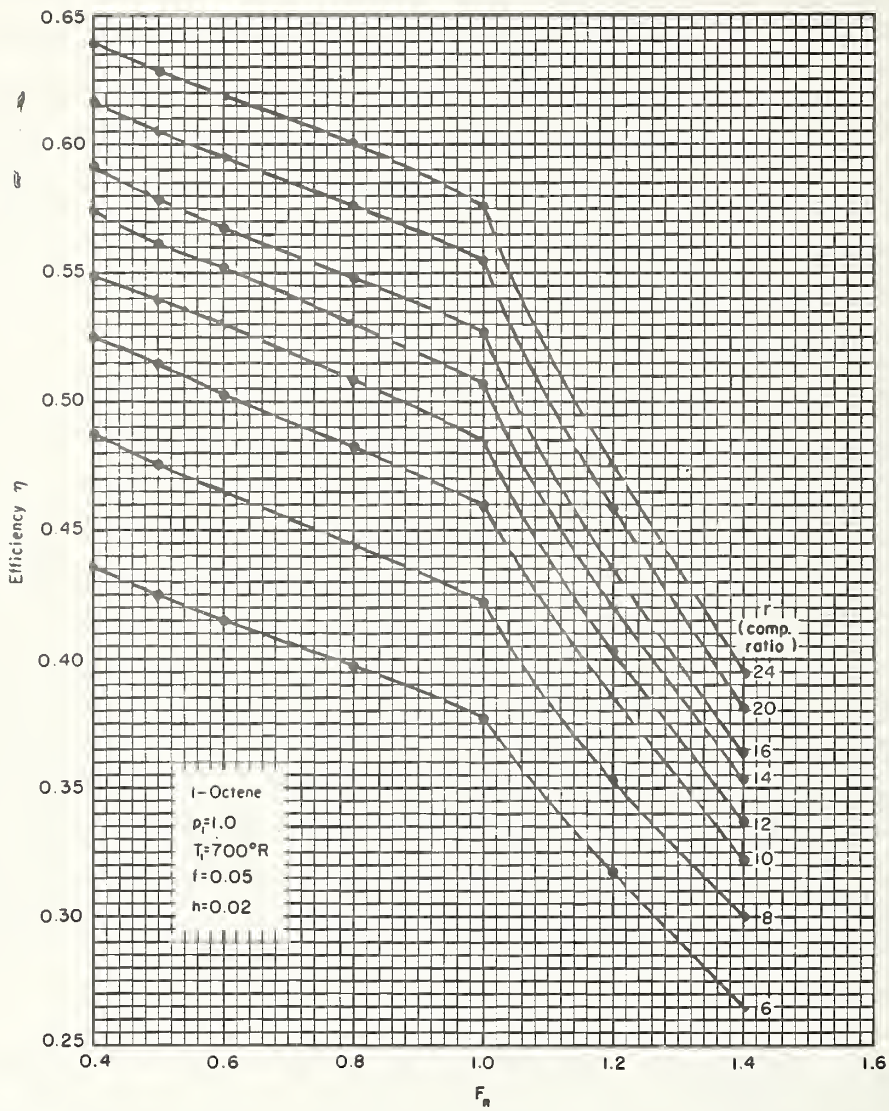
3.10.1 Compression Ratio

From a thermodynamic point of view, higher compression ratios result in higher thermal efficiencies (see Figure 3-27), thus, higher fuel economies. The British Technical Council reports that, on the average, the fuel economy increases by 7.6 percent per unit increase in compression ratio.[32] Thus, an engine's thermal efficiency is a direct function of the compression ratio. However, compression ratio is limited by engine knock. Today's spark ignition homogeneous charge engines operate at lower compression ratios, generally between 7 and 9, to meet the knock limited performance at wide open throttle.

The spark ignition problem is simultaneously to increase the compression ratio and to reduce the swept volume of the engine in order to operate the engine in the low speed, high torque regime under more favorable fuel economies. Tests on various compression ratios in spark ignition engines verify that by increasing the compression ratio, lower fuel consumptions result. (See Figure 3-28.) In this example, the engine displacement is maintained constant and the horsepower increases with increase in compression ratio under MBT spark settings. The increase in fuel economy is more dramatic at part load than at full load because of the decrease in internal residual gases remaining in the cylinder which offset the adverse effect of increased pumping and friction losses.

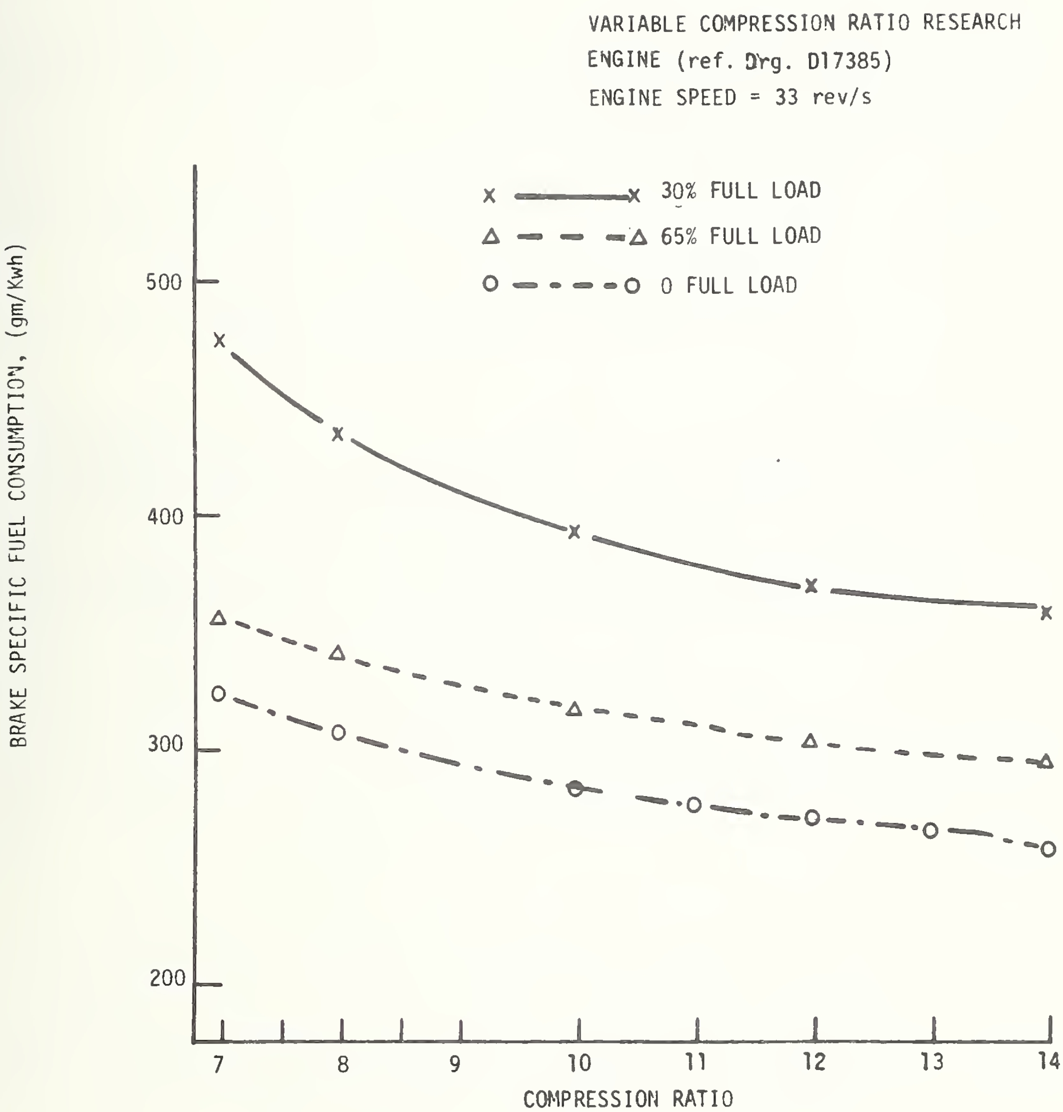
3.10.2 Engine Knock

The significant parameters in the design of engines and fuels which permit the compression ratio to be increased are the chemical and mechanical octane numbers. The chemical octane number (built into the fuel at the refinery) is influenced by the fuel structure and chemical additives in the fuel. Mechanical octane numbers (built into the engines by the engine designers) are considered to be improvements in the design of the engine; for example, ignition control, combustion chamber design, valve timing,



SOURCE: Reference 32

FIGURE 3-27. EFFICIENCY VERSUS FUEL/AIR EQUIVALENCE RATIO (F_R) FOR THE CONSTANT-VOLUME FUEL-AIR CYCLE



Source: Reference 33

FIGURE 3-28. EFFECT OF COMPRESSION RATIO ON SPARK IGNITED GASOLINE ENGINE FUEL CONSUMPTION carburetion, and injection design. Techniques such as spark retard at low speeds and full throttle (where the tendency to knock is the greatest) is an example of making feasible the use of higher compression ratio.

The fuel anti-knock quality is measured by ASTM Research and Motor Methods in a CFR unthrottled engine under specified running and boiling coolant conditions. The RON (research octane number) method compares the knock intensity of fuels to a reference iso-octane fuel under research conditions. The MON (motor octane number) method compares a fuel to that of the reference iso-octane fuel in a CFR engine under running conditions typical of actual engines in passenger car and light truck operations. The reference fuel is iso-octane under both methods. The sensitivity of the fuel is determined by subtracting the motor octane number from the research octane number.

Knock (detonation) and pre-ignition are two distinct types of reaction processes which occur in the spark ignition engine. Knock is associated with the "end gases," i.e., unburned gases compressed by virtue of the expanding flame front once combustion has been initiated. Pre-ignition occurs during the compression stroke prior to engine firing. Pre-ignition reactions occur principally at hot spots within the combustion chamber. They are thought to be unrelated, generally, to fuel quality, which differentiates it from knock which is associated both with engine design and fuel quality.

Figures 3-29 and 3-30 show the effect of speed and throttle opening for a typical engine on octane number requirements. Illustrations are shown for both a temperature sensitive and insensitive fuel. Two characteristics are immediately apparent for the temperature insensitive fuel (a) at any throttle opening, the tendency to knock decreases rapidly with an increase in speed, and (b) at any speed, the tendency to knock decreases rapidly with reduction in throttle opening. The temperature sensitive fuel displays, essentially, a constant octane number requirement over the speed range of the engine.

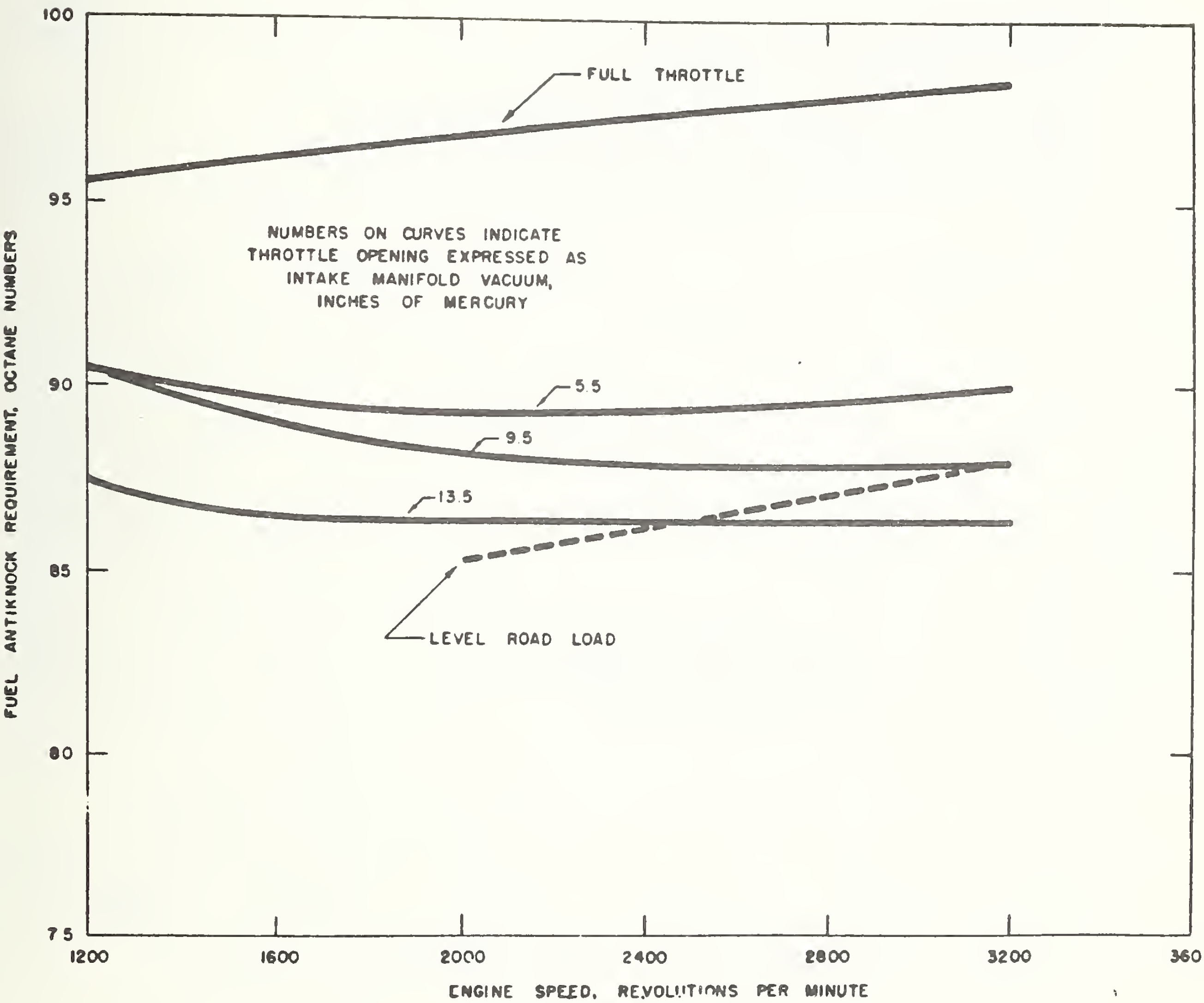


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FIGURE 3-29. EFFECT OF SPEED AND THROTTLE OPENING ON OCTANE-NUMBER REQUIREMENT WITH SENSITIVE FUELS

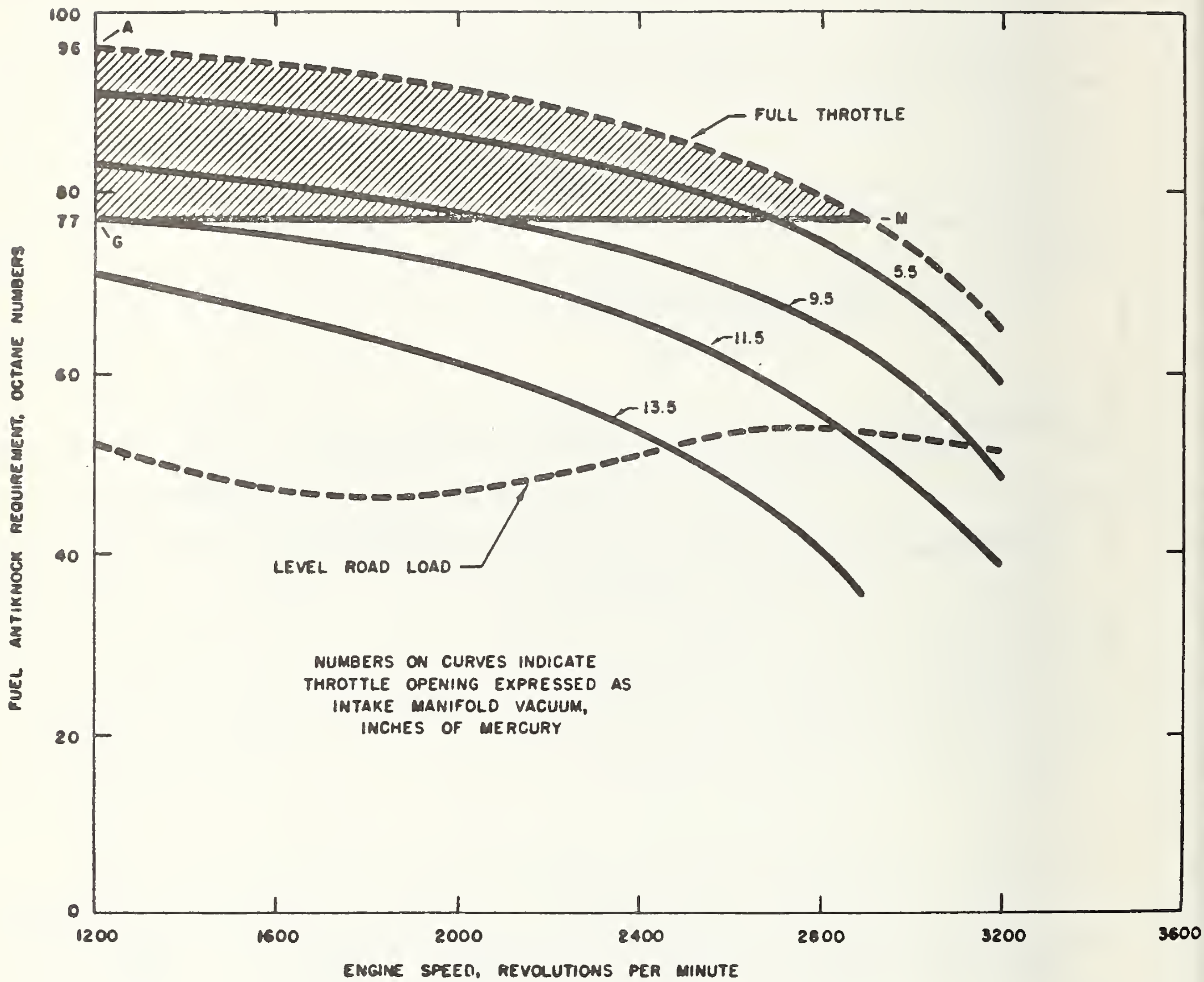

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FIGURE 3-30. EFFECT OF SPEED AND THROTTLE OPENING ON OCTANE-NUMBER REQUIREMENT WITH INSENSITIVE FUELS

It is well accepted that knock is due to the surface bound end-gas having less turbulence than the bulk charge. Therefore, it is important that the flame arrives before the zone can auto-ignite leading to knock. Combustion chambers which induce turbulence of the charge (a) permit the use of higher compression ratios of a given fuel, (b) enable engines to make relatively better use of sensitive rather than insensitive fuels, and (c) yield potentially greater fuel economy at part-throttle operating conditions.

Combustion chamber deposits also contribute to engine knock. Engine oils should be used which minimize the tendency to deposit. Engines should be designed to reduce their surface to volume ratios.

As an example of the tradeoffs considered during the development phases of an engine, two engine system parameters were investigated by Volkswagen to establish the effect of knock on work done for DOT/TSC on the potential of turbo charging a spark ignition engine. In this study (see Figure 3-31) a series of compression ratios were analyzed, and a combination of maximum boost compressor ratios were identified under MBT stoichiometric control, and engine running conditions at WOT. The characteristic trend shows that maximum power under knock limited conditions is obtained with the lower compression ratio and higher boost ratio. Volkswagen also showed that the best fuel economy was obtained for the high compression ratio, low boost configuration.

The British Technical Council[32] estimates that a unit increase of octane number results in a 1.3 percent increase in fuel economy.

3.11 PARASITIC LOSSES

Parasitic losses arise from a) air conditioning, b) fans, c) alternator, and d) pumps (water, power steering, oil and air pumps). Parasitic losses can be reduced with a controlled speed accessory drive system (CSAD).[36] This system consists of a pair of centrifugally controlled variable pitch belt pulleys mounted on the crankshaft and water pump of the engine. The water pump pulley serves as the output drive and as the power take-off for

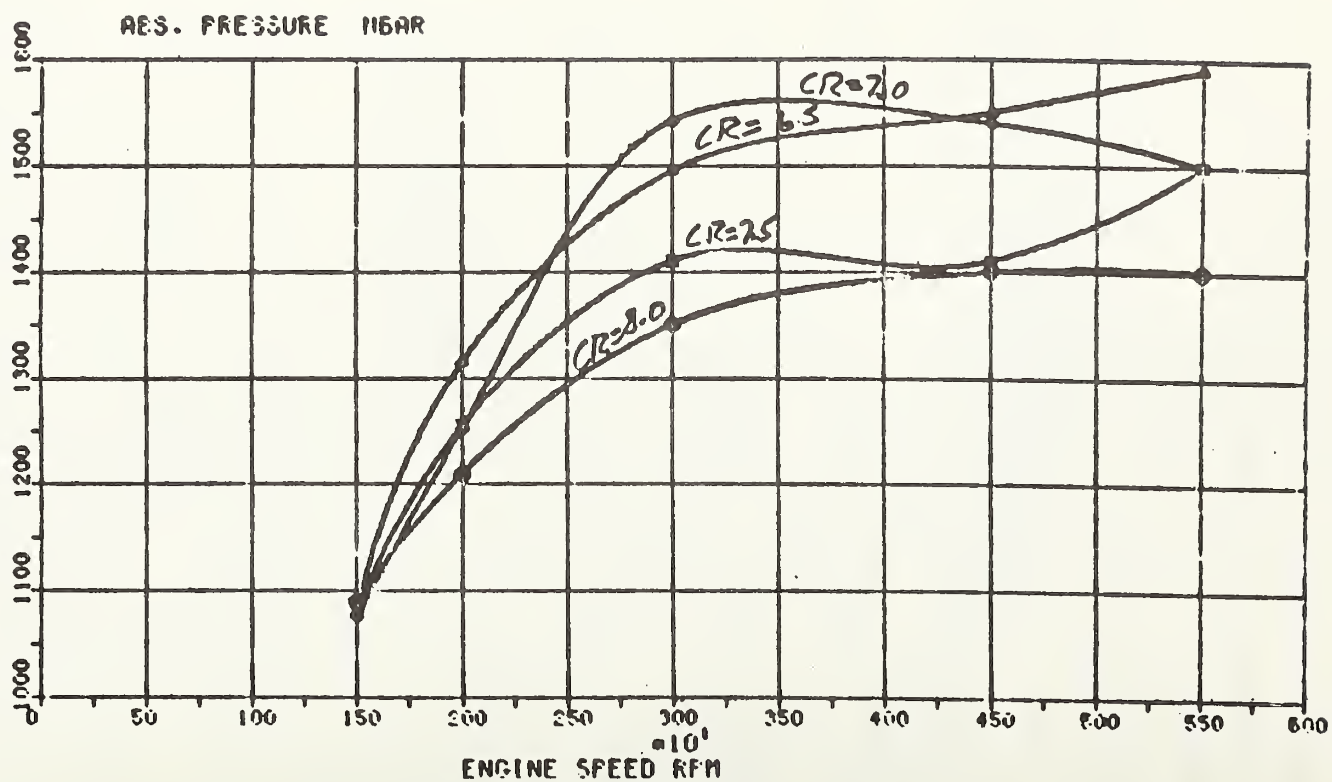

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FIGURE 3-31. FULL-LOAD-CURVE PRESSURE RATIO the engine accessories. As the engine speed increases, the drive ratio decreases, resulting in nearly constant accessory speed, which reduces parasitic losses and improves the fuel economy. Limited test data show that standard vehicles equipped with a CSAD unit obtain a 2 and 6 percent improvement in fuel economy for urban and highway driving respectively. Gains of 6 and 8 percent in fuel economy have also been measured for the urban and highway cycle, respectively, when both the air conditioner and lights were powered by the unit.

When the fan is directly coupled to the engine, the power required to operate the fan increases with engine speed. This results in an increase in fuel consumption. The primary function of the cooling fan is to draw cool air in through the radiator and blow it back over the engine during periods when there is insufficient ram air supplied by the forward motion of the vehicle. At low speeds, the fan is needed to provide adequate cooling. At speeds above approximately 50 mph there is sufficient ram pressure to provide the necessary air coolant flow across the radiator, and the fan is actually not needed. The power consumed by the fan above approximately 2000 rpm represents excessive power consumption that provides a potential area for increased fuel economy. Reduction of fan power at high engine speeds can be accomplished by:[37,38]

- use of variable speed accessory drive,
- use of variable controlled blades which decamber with increasing engine speeds,
- use of a viscous clutch fan drive which provides maximum fan performance at low speeds and reduces the power consumed at higher engine speeds by lowering the fan-to-engine speed ratio, and
- use of an electrically driven, thermostatically-controlled fan.

REFERENCES FOR SECTION 3

1. "Potential for Motor Vehicle Fuel Economy Improvement," Report to Congress, U.S. Department of Transportation and U.S. Environmental Protection Agency, October 24, 1974.

2. "An Installed Engine Efficiency Analysis for Steady and Cyclic Operations," Ford Motor Company, Product Research Office, Technical Memorandum, PRM-66-27, September 21, 1966.

3. Meyers, P. "Heavy Duty Diesel Truck Fuel Economy Research," presented at the Department of Transportation/Transportation Systems Center Diesel Truck Fuel Economy Optimization Conference, September 6 - 9, 1977, Cambridge MA.

4. Bishop, I.W., SAE paper 812A presented January, 1964 at the SAE Automotive Engineering Congress.

5. Millington, B.W., and Hartles, E.R., "Friction Losses in Diesel Engines," SAE Paper 680590, 1968.

6. Overington, M.T., et al. "A Literature Study of 28 Production Engines and the Interrelated Effects of Engine Design, Tuning, and Vehicle Environment on Fuel Economy and Exhaust Emissions," Ricardo Consulting Engineers, DP 76/235, April 1976.

7. "High Compression Gas Mixture, Homogeneous Charge Combustion," SAE Paper 790386.

8. Marrell, J.P., "Light Duty Automotive Fuel Economy - Trends Through 1979," SAE Paper 790225.

9. Peters, B., and Quader, A., "Whetting the Appetite of Spark Ignition Engines for Lean Combustion," SAE Paper 780234.

10. Wheeler, R.W., "Gasoline Combustion - Past, Present and Future," Ricardo Consulting Engineers, Report No. DP78/212, March 1978.

11. United States Department of Transportation/Transportation Systems Center, Contract DOT-TSC-1422 with Fiat Central Research, "Passenger Car Spark Ignition Data Base."

12. Koneshi, M., et al., "Effects of Prechambers on NO_x Formation Processes in the Spark Ignition Engine," SAE Paper 790389.

13. Mitchell, E., "Texaco's Stratified Charge Concept - A Fuel Efficient Approach to Exhaust Emission Control," An Oral Presentation to the International Stratified Charge Engine Conference, SAE, Troy MI, October 30 - November 1, 1974.

14. Newhall, H.K., "Combustion Process Fundamentals and Combustion Chamber Design for Low Emissions," SAE Paper 751001.

15. Scussel, A.J., "The Ford PROCO Engine Update," SAE Paper 780699.

16. Yagi, S., "A New Combustion System in the Three-Valve Stratefied Charge Engine," SAE Paper No. 790439, Feb. 1979.

17. Jones, C., "A Review of Curtiss-Wright Rotary Engine Developments with Respect to General Aviation Potential," SAE Paper 790621.

18. Emmenthal, K.D., "Automotive Spark Ignition Engine Data Base," presented at the Department of Transportation/National Highway Traffic Safety Administration Contractors Review Meeting, December 1978.

19. Hiereth, H., and Withalm, G., "Some Special Features of the Turbocharged Gasoline Engine," SAE Paper 790207.

20. Department of Transportation/Transportation Systems Center, Contract DOT-TSC-1421 with Chrysler Corporation.

21. Bates, B., Dosdal, J.M. and Smith, D.H., "Variable Displacement by Engine Valve Control," SAE Paper 780145.

22. Pouliot, H.N., et al., "A Variable Displacement Spark Ignition Engine," SAE Paper 770114.

23. Ibid.

24. "Feasibility Test on Compounding the Internal Combustion Engine for Automobile Vehicles," by Thermo Electron Corporation ERDA Contract Number E (11-1)-2690.

25. Palmer, R.M., "Turbo-Compounding the Spark Ignition Engine," Ricardo Consulting Engineers, presented at the May 6-8, 1973 ERDA Contractors Coordination Meeting.

26. Parker, D.A., and Kendruk, M., "B-1-11-A Camshaft with Variable Lift-Rotation Characteristics, Theoretical Properties and Application to the Valve Gear of a Multi Cylinder Piston Engine."

27. DOE contractors meeting, Washington DC, Spring 1977.

28. Stivender, D.L., "Intake Valve Throttling (I.V.T.) - A Sonic Throttling Intake Valve Engine," SAE Paper 680399.

29. Taylor, C.F., "The Internal Combustion Engine in Theory and Practice," Vol. II of II, MIT Press, 1968.

30. Trella, T.T., "Breathing Cycle of the Four-Stroke Automotive Engine," Simulation Council Proceedings, Series, Vol. 7, Chapter 12, No. 2, December 1977.

31. Berriman, L.P., "Advanced Emission Control - The Dresserator System," Special Report prepared by Dresser Advanced Technology Center, CA, 92713, June 15, 1979.

32. British Technical Council Report, MPE/761005.

33. U.S. Department of Transportation/Transportation Systems Center Contract DOT-TSC-1242 with Calspan Corp., "High Speed Engines and Their Noise Characteristics."

34. Bartholemew, E., "Mechanical Octane Numbers," Ethyl Corporation, Research Laboratories, Detroit MI, 1951.

35. U.S. Department of Transportation, Transportation Systems Center Contract DOT-TSC-1420 with Volkswagenwerk, "Passenger Car Spark Ignition Data Base."

36. U.S. Department of Energy Contractors' Coordination Meeting, "Controlled Speed Accessory Drive," April 24-26, 1979, Dearborn, MI.

37. Hunter, D.A., et al., "A Study of Technological Improvement in Automobile Fuel Consumption," Arthur D. Little, Inc., Cambridge MA, Department of Transportation/Transportation Systems Center Report No. DOT-TSC-OST-40.II, December 1974.

38. Stephenson, R.R., "Should We Have a New Engine?," Vol. II Technical Report, Jet Propulsion Laboratory, California Institute of Technology, August, 1975.

4. EFFECT OF VEHICLE DESIGN VARIABLES ON FUEL ECONOMY, GRADEABILITY, ACCELERATION AND TOP SPEED

4.1 GENERAL

With the increasing demand for improvements in fuel economy, the compromise between fuel economy and performance must be examined. Typically, performance has been loosely defined as the ability to accelerate and climb hills. Some standard tests used to characterize performance include 0-60 mph time and maximum gradeability. The fuel economy tests, of course, have been established by the EPA.

Vehicle design variables can be modified to improve fuel economy with a subsequent change in performance. The ratio of the change in fuel economy or performance to the change in a vehicle design variable is defined as a sensitivity. By establishing sensitivities for a range of vehicle design parameters for a cross section of vehicle weight classes, a better understanding of the influence of vehicle design parameters on fuel economy and performance can be obtained. The results establish fuel economy and performance trends for a given vehicle modification. The sensitivities created from these trends can be used to predict fuel economy and performance changes for future vehicle modifications.

4.2 BASELINE VEHICLES

The baseline vehicles selected for this study[1] represent a cross section of automobiles and light trucks. The engines include four, six and eight cylinders. Two engines of about the same horsepower were investigated to obtain an indication of sensitivity of the results to engine variations. The vehicles and their engine displacements are shown in Table 4-1. The independent (design) variables examined and the dependent (performance) variables are shown in Table 4-2.

TABLE 4-1. BASELINE VEHICLES

VEHICLE	WEIGHT (lbm)	ENGINE (CID)*	RAR	TRANSMISSION	HP/WT*
AUTOMOBILE	2000	98/98	3.58	M4	.034/.034
AUTOMOBILE	3000	130/151	3.91	M5	.030/.030
AUTOMOBILE	3500	231/225	2.56	A3	.027/.029
AUTOMOBILE	4500	318/301	2.71	A3	.030/.028
LIGHT TRUCK	3000	140/170	3.08	M4	.032/.032
LIGHT TRUCK	4000	351/350	3.54	M4	.040/.042

*BASELINE/ALTERNATE

A3 Automatic 3-speed

M4 Manual 4-speed

M5 Manual 5-speed

TABLE 4-2. VEHICLE VARIABLES

INDEPENDENT VARIABLES
• Weight
• Driveline
• Aerodynamic Drag
• Rolling Resistance
• Engine
DEPENDENT VARIABLES
• Acceleration
0-5 sec distance
0-60 mph time
40-60 mph time
• Top Speed
• Gradeability
• Fuel Economy

4.3 ACCELERATION

The acceleration of the baseline vehicles is characterized by three drive schedules: 0 to 60 mph time, 40 to 60 mph time, and the distance travelled over the first 5 seconds from a standing start. The majority of energy expended for these drive cycles is attributed to the mass as shown in Table 4-3. Changes in the rolling resistance and aerodynamic drag have a smaller impact on acceleration and are not evaluated in this report. The effect of weight reduction on the 0-60 mph time can be seen in Figure 4-1. The sensitivities associated with the weight change are also included on the figure. Acceleration is primarily a function of vehicle horsepower to weight ratio as is shown on Figure 4-2.

The influence of the rear axle ratio on the three acceleration drive schedules is presented in Figures 4-3, 4-4 and 4-5. The inflection points in the curves are caused by the shift logic selected. For example, in Figure 4-3 for the 98 CID vehicle, the acceleration time for a RAR of 3.8 is 13.2 sec, while that for a RAR of 4.0 is 13.7 sec. In the former case, the vehicle reaches 60 mph in 3rd gear and in the latter case, it is 4th gear. Another example can be taken from Figure 4-4. For the 140 CID vehicle, the acceleration time for a RAR of 2.7 is 6.05 sec, while that for a RAR of 3.0 is 6.73 sec. With a 3.0 RAR, the vehicle shifts into 3rd gear before 60 mph accounting for the additional 0.68 seconds.

4.4 TOP SPEED

The top speed of a vehicle will be determined by the intersection of the engine WOT horsepower and road load curves. By examining Figure 4-6, it can be seen that as a vehicle approaches its top speed, approximately 87 percent of its output energy is used to overcome aerodynamic drag. Therefore, any weight change or rolling resistance change will not significantly affect the top speed. The relationship among top speed, maximum engine horsepower and effective area of the vehicle (frontal area x drag coefficient) is shown in Figure 4-7. Since the aerodynamic drag is the predominant energy consumer at the top speed, it is obvious

TABLE 4-3. ENERGY REQUIREMENTS FOR WOT ACCELERATION DRIVE SCHEDULES

DRIVING SCHEDULE (WOT)	VEHICLE WT (lb)	ENERGY REQUIREMENTS (% of Total)		
		Mass	Aero	Rolling
0-5 sec	2000	79	-	4
	4000	75	1	3
0-60 mph	2000	67	8	6
	4000	68	7	6
40-60 mph	2000	64	20	9
	4000	63	13	9

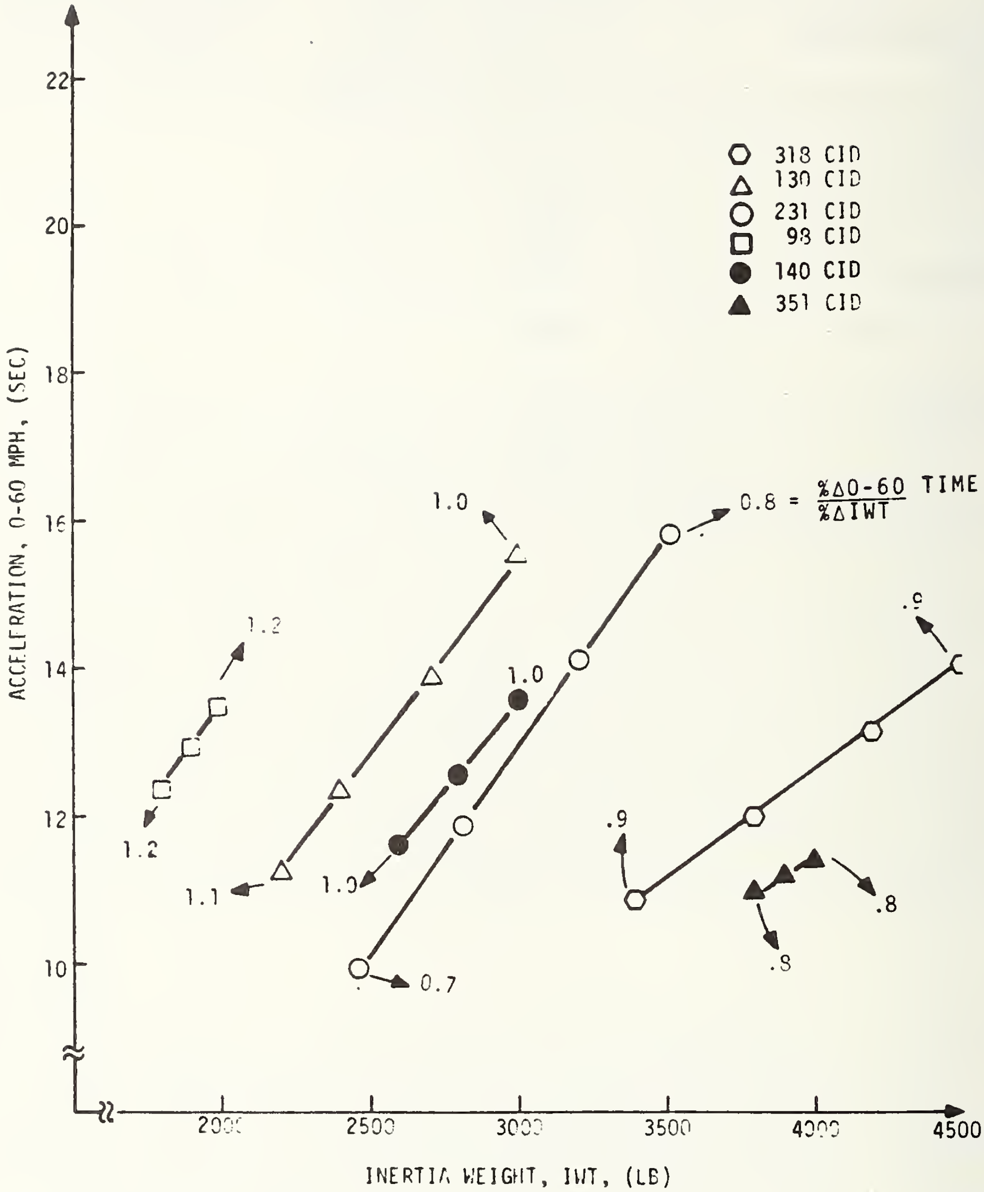


FIGURE 4-1. EFFECT OF WEIGHT ON ACCELERATION (0-60 mph)

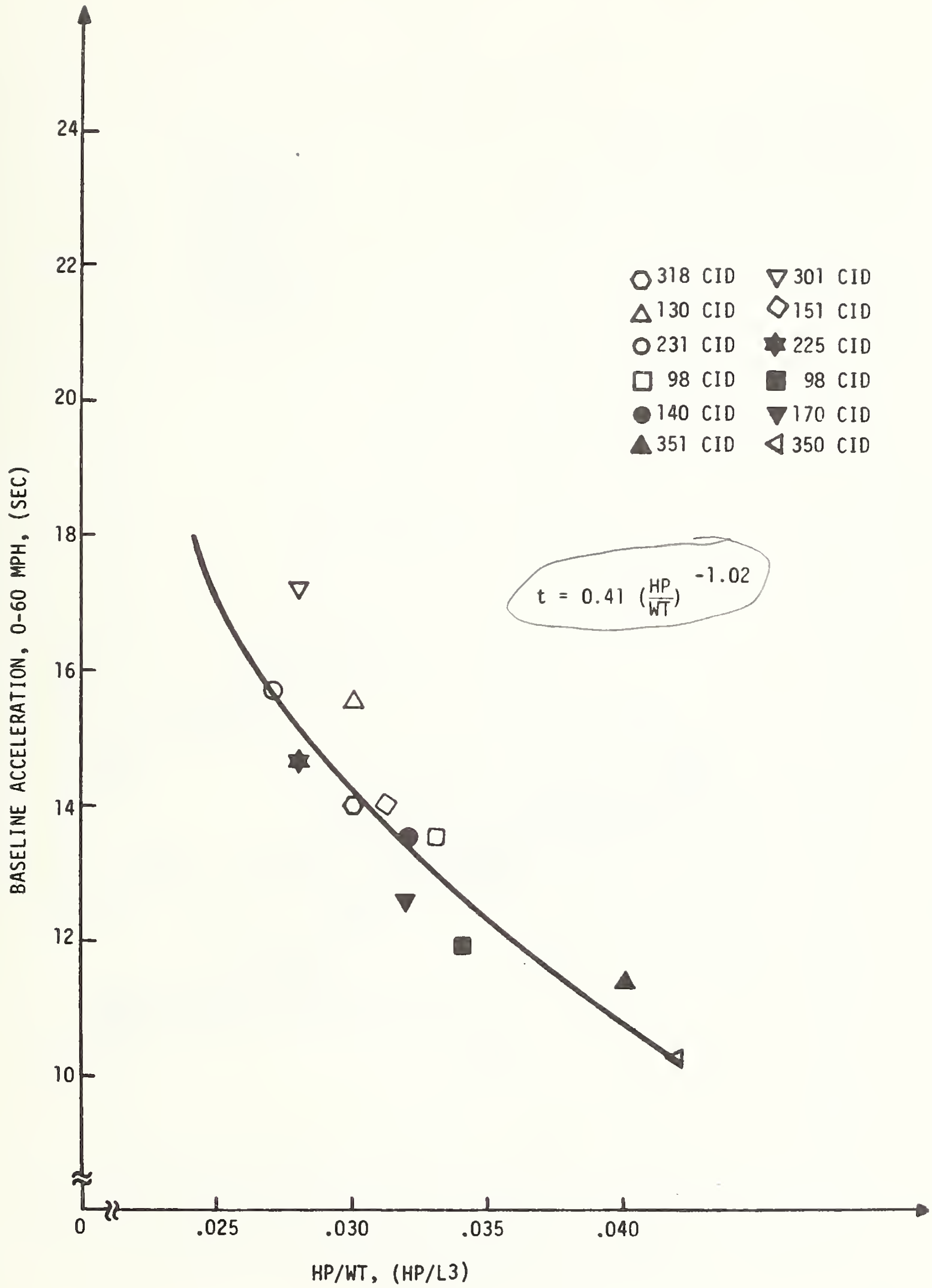


FIGURE 4-2. ACCELERATION AS A FUNCTION OF HP/WT FOR ALL VEHICLES SIMULATED

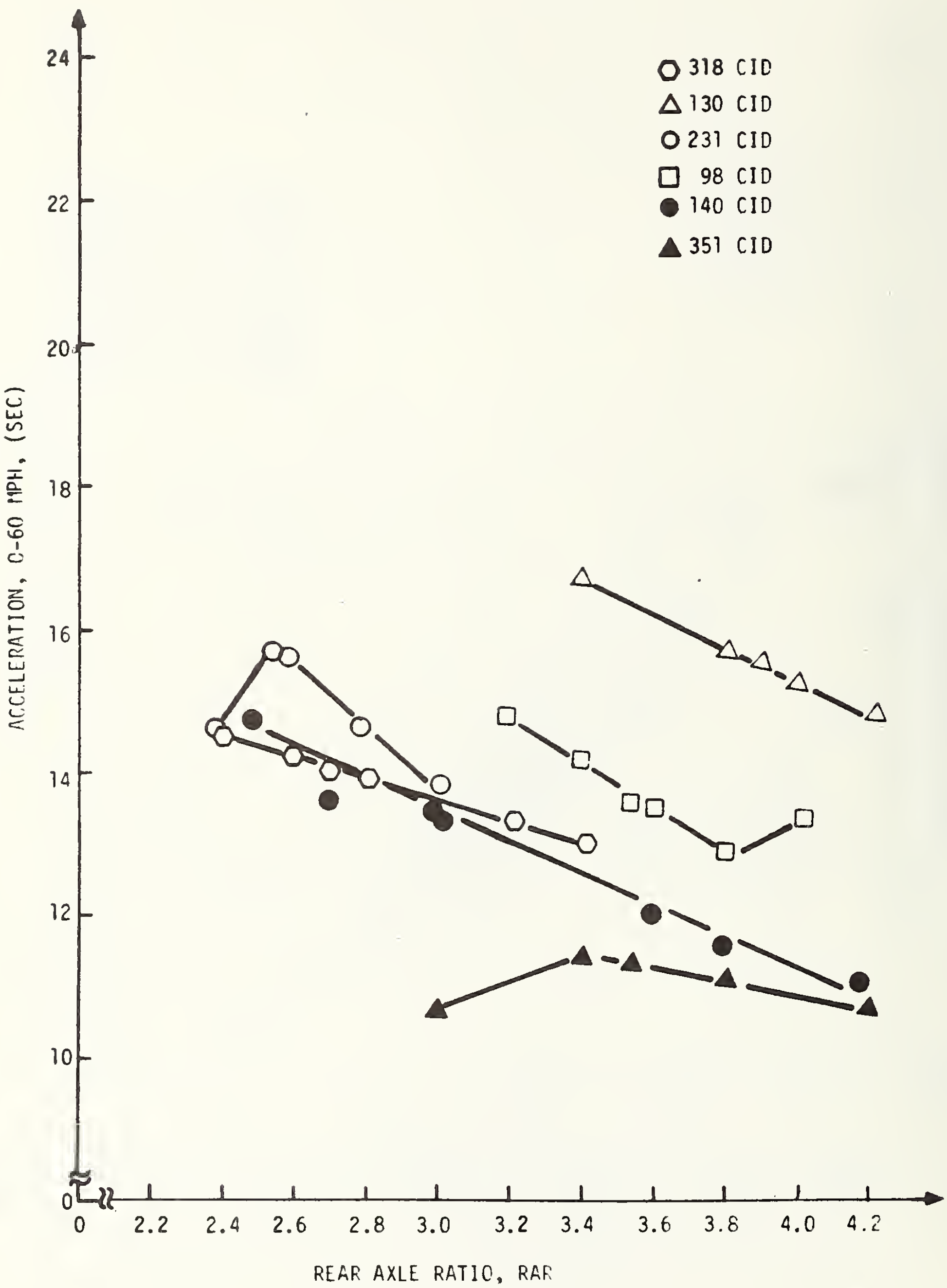


FIGURE 4-3. ACCELERATION (0-60 mph) TIME AFFECTED BY REAR AXLE RATIO

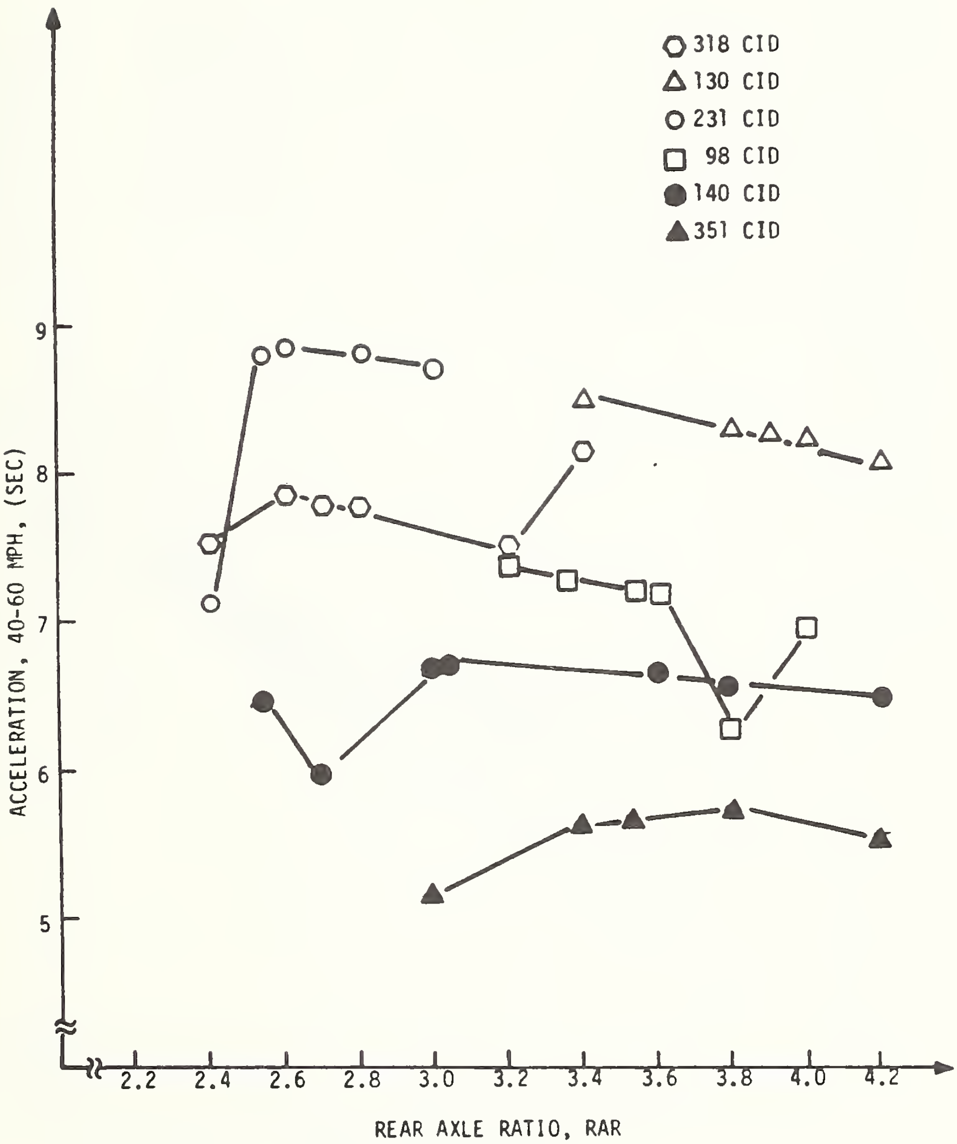


FIGURE 4-4. PASSING ACCELERATION (40-60 mph) TIME AS A FUNCTION OF REAR AXLE RATIO

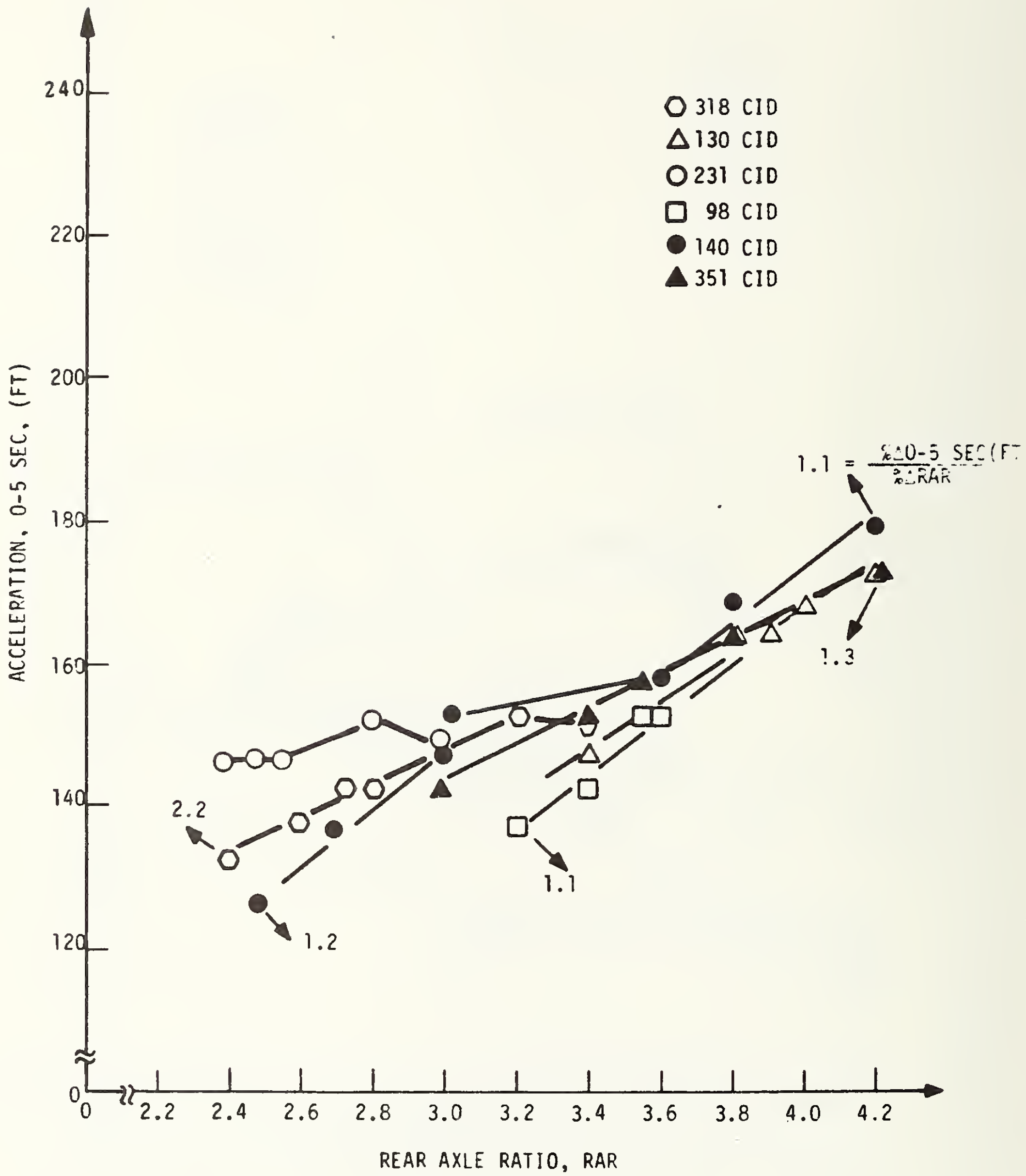


FIGURE 4-5. ACCELERATION DISTANCE (0-5 sec) AS A FUNCTION OF REAR AXLE RATIO

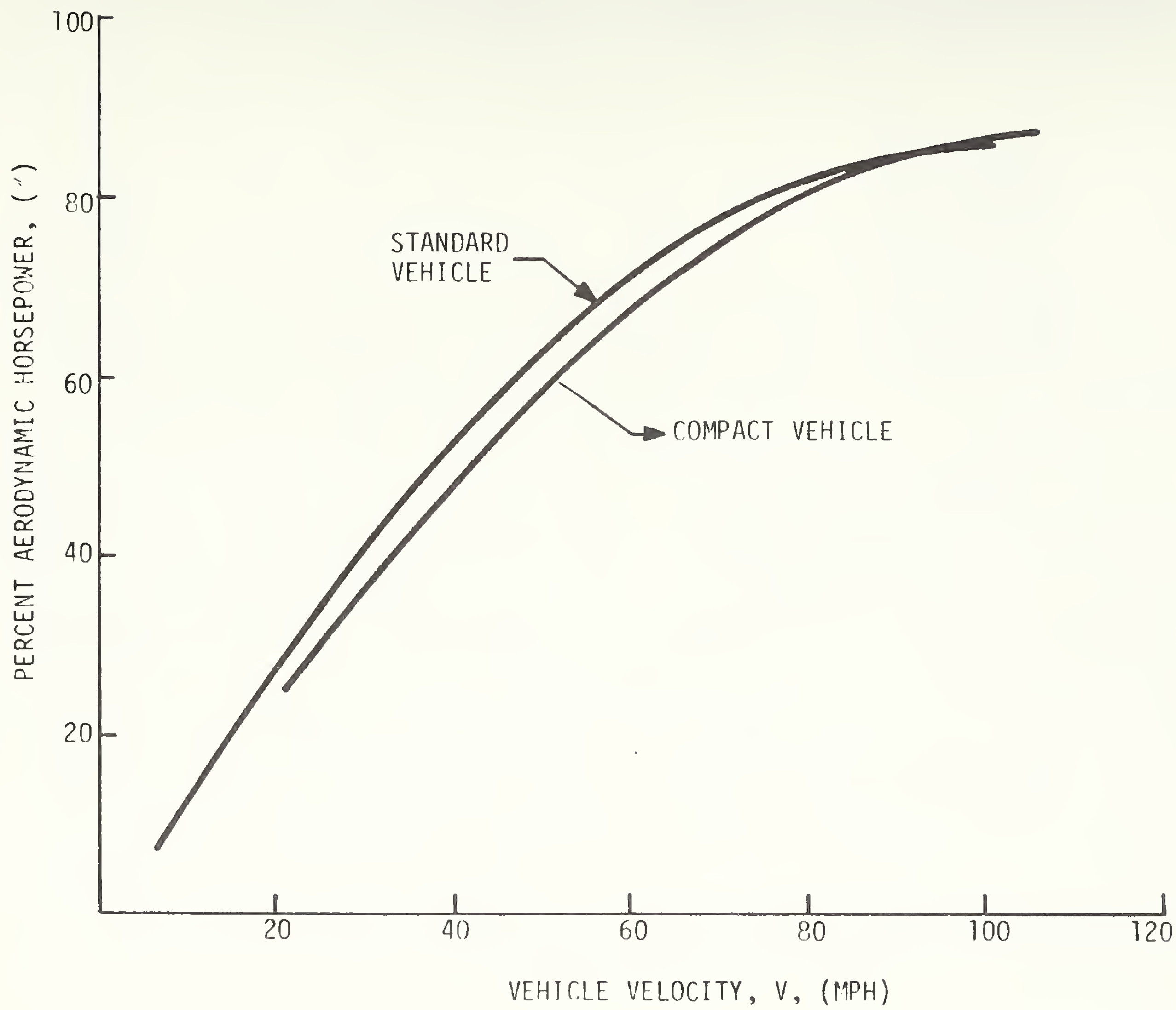


FIGURE 4-6. AERODYNAMIC HORSEPOWER AS A FUNCTION OF VEHICLE VELOCITY

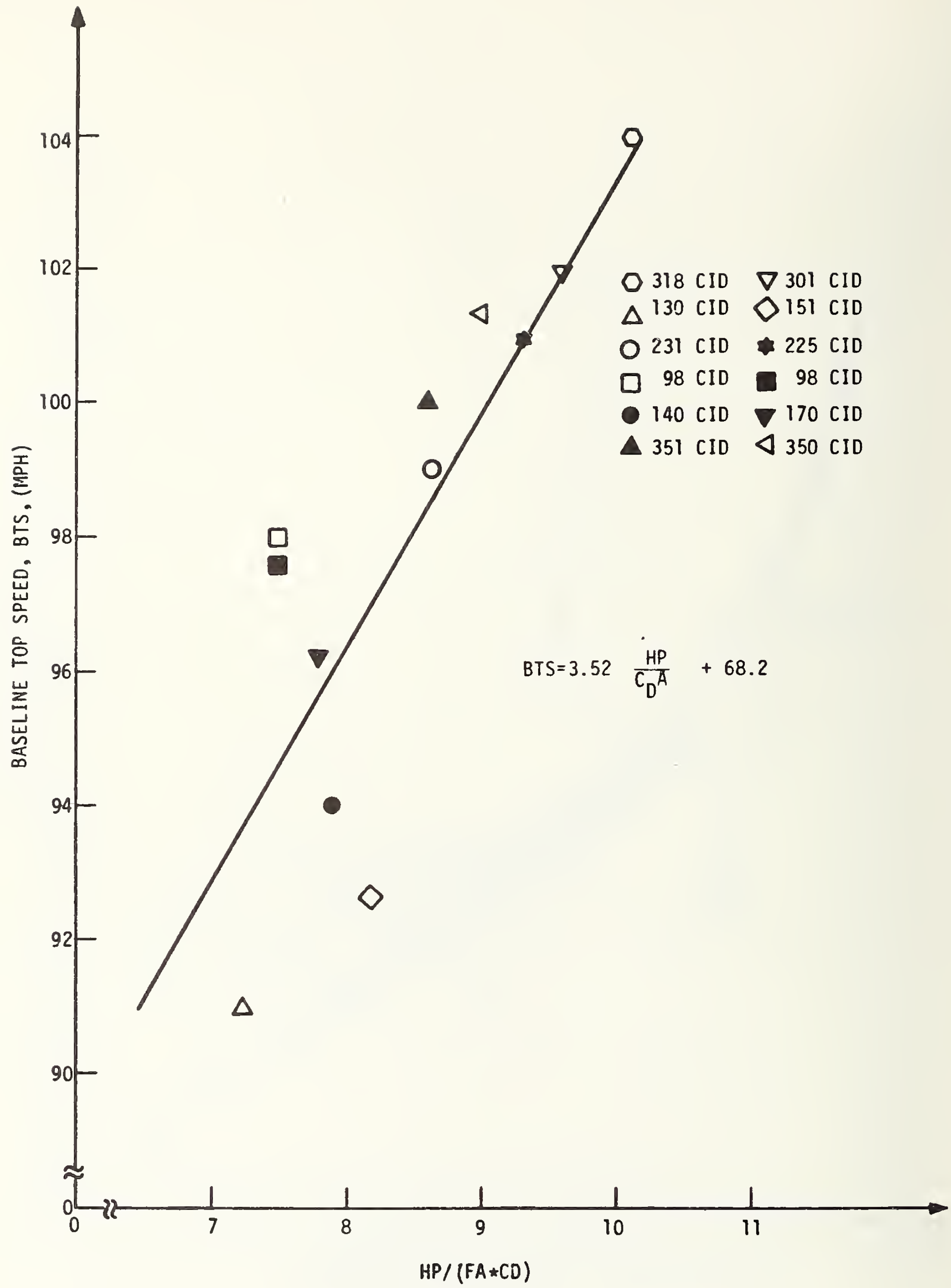


FIGURE 4-7. TOP SPEED AS A FUNCTION OF HORSEPOWER TO EFFECTIVE AREA FOR ALL VEHICLES SIMULATED

that a drag coefficient reduction will increase the top speed. Figure 4-8 illustrates this result along with the various sensitivities.

The effect of a rear axle ratio change on top speed can be best described qualitatively using Figure 4-9. A rear axle ratio change simply repositions the road load curve. A higher numerical ratio moves the curve to the right and a lower numerical ratio moves it to the left. Already it is apparent that the top speed change will be dependent upon just how much the road load curve moves, and by the shape of the horsepower curve. For this particular vehicle, the baseline rear axle ratio produces a road load curve which intersects the WOT engine horsepower curve to the right of its maximum. Therefore, a decrease in rear axle ratio will increase the top speed. The quantitative effect of axle ratio on top speed is given in Figure 4-10 for each of the base vehicles.

4.4 GRADEABILITY

Gradeability is defined as the maximum grade a vehicle can negotiate at a given speed. The results for each of the baseline vehicles are presented in Figures 4-11 - 4-16. When calculating the gradeability of a vehicle in first gear, it is important to consider three variables. The tires, the transmission and the clutch, when applicable, influence the gradeability at start and low vehicle velocities. It can be seen from observing the results that the gradeability is significantly affected by tire slippage. A value of 0.8 was used as the coefficient between the road and tire surface. This value is most likely different for the 351 CID light duty truck. Therefore, the upper limit for this vehicle was indicated as a range. Also, it is obvious that if four-wheel drive were implemented, the gradeability would be significantly increased.

The tractive force is a function of the product of the engine torque and the torque ratio. The maximum gradeability for a vehicle equipped with a manual transmission should occur at maximum engine torque, although the startup gradeability will be influenced by the clutch dynamics engagement. The maximum gradeability for

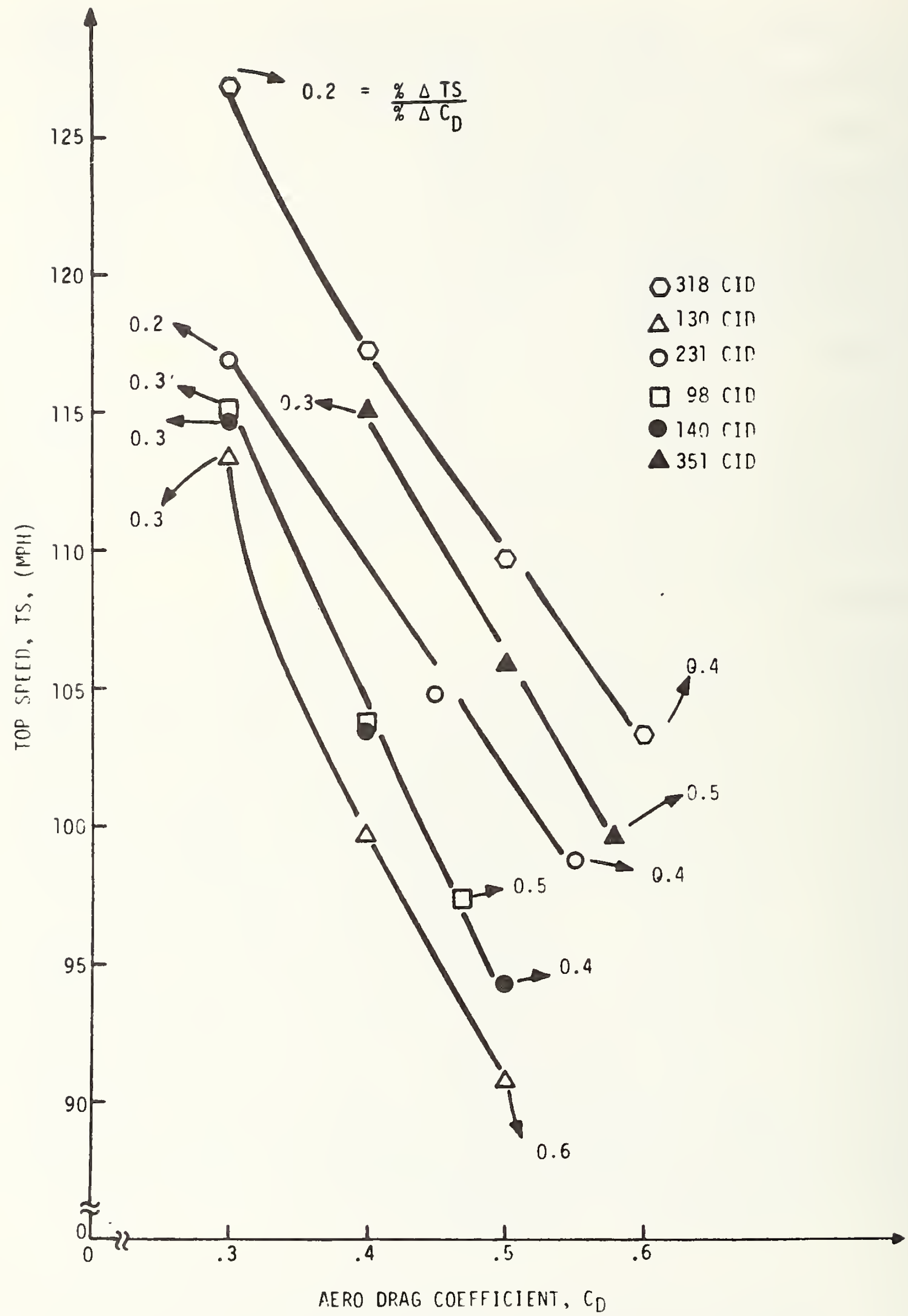


FIGURE 4-8. TOP SPEED AS A FUNCTION OF AERODYNAMIC DRAG

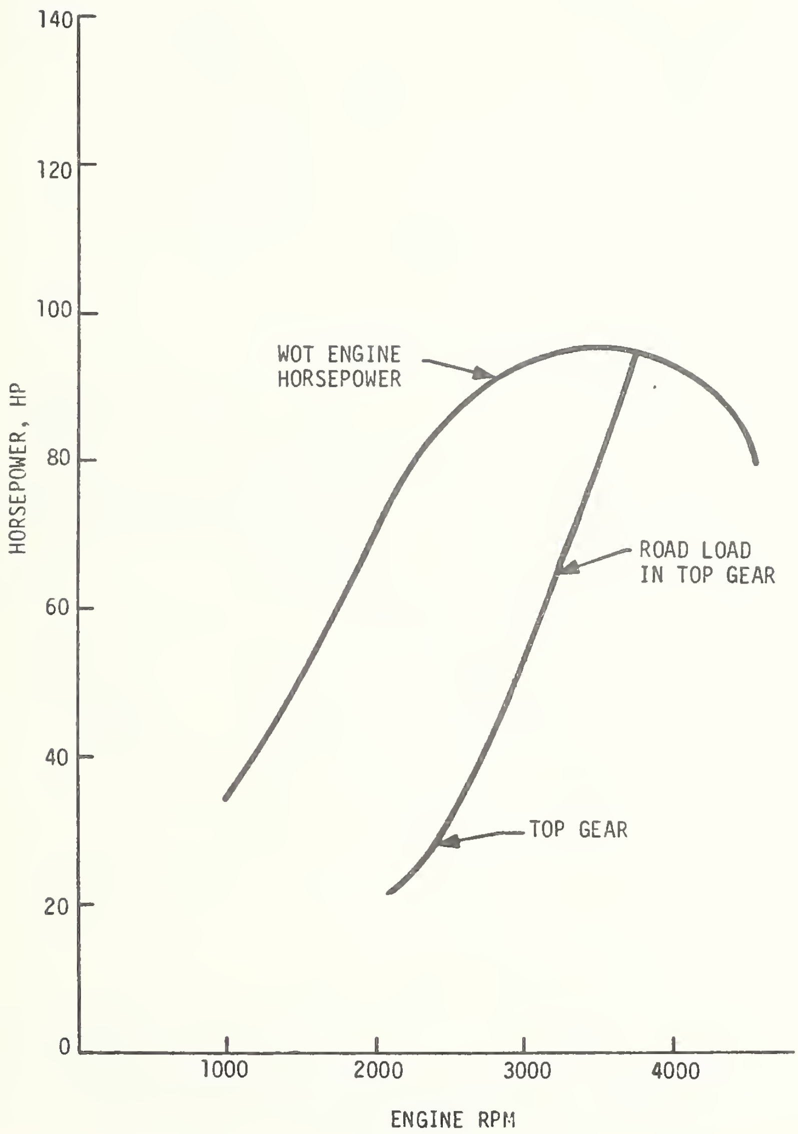


FIGURE 4-9. ROAD LOAD REQUIREMENTS FOR 3500 lb 231 CID AUTOMOBILE

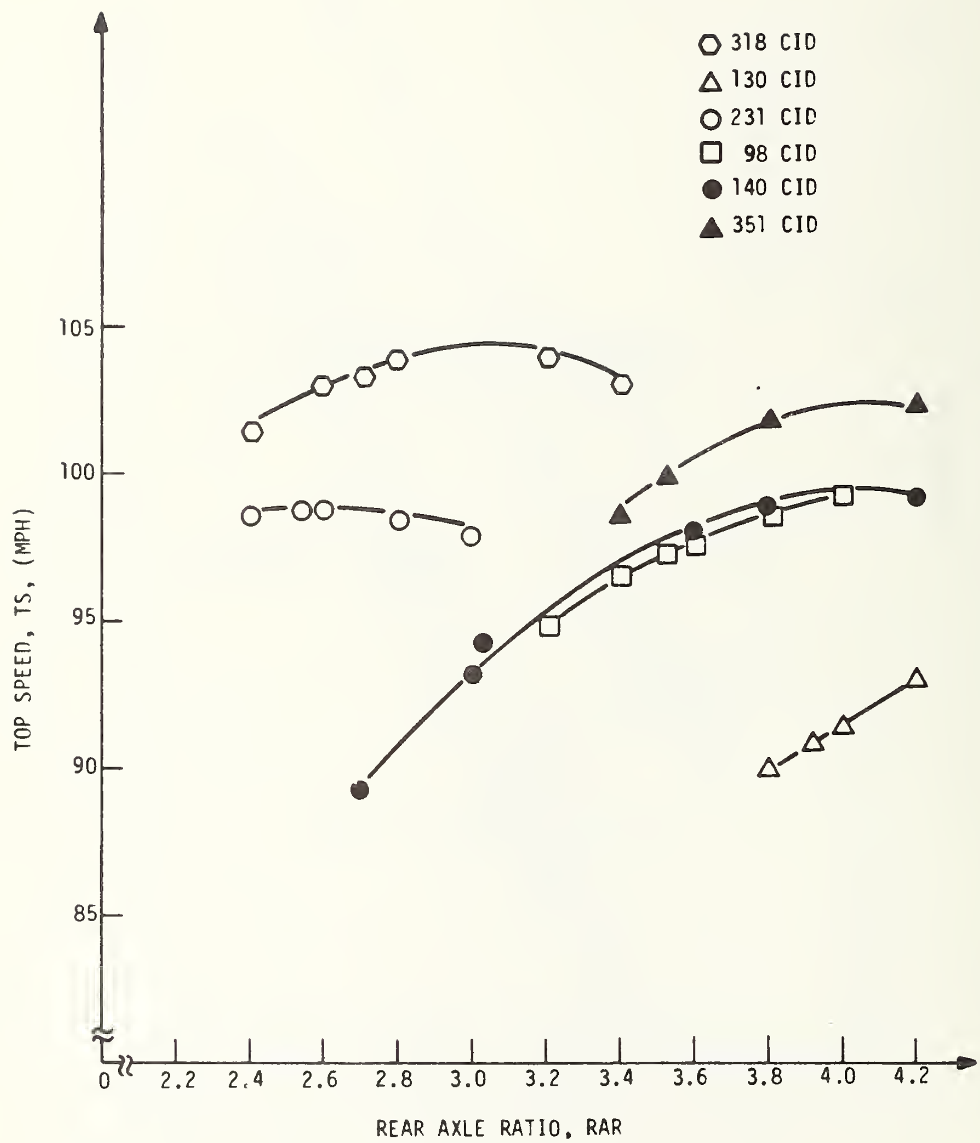


FIGURE 4-10. TOP SPEED AS A FUNCTION OF REAR AXLE RATIO

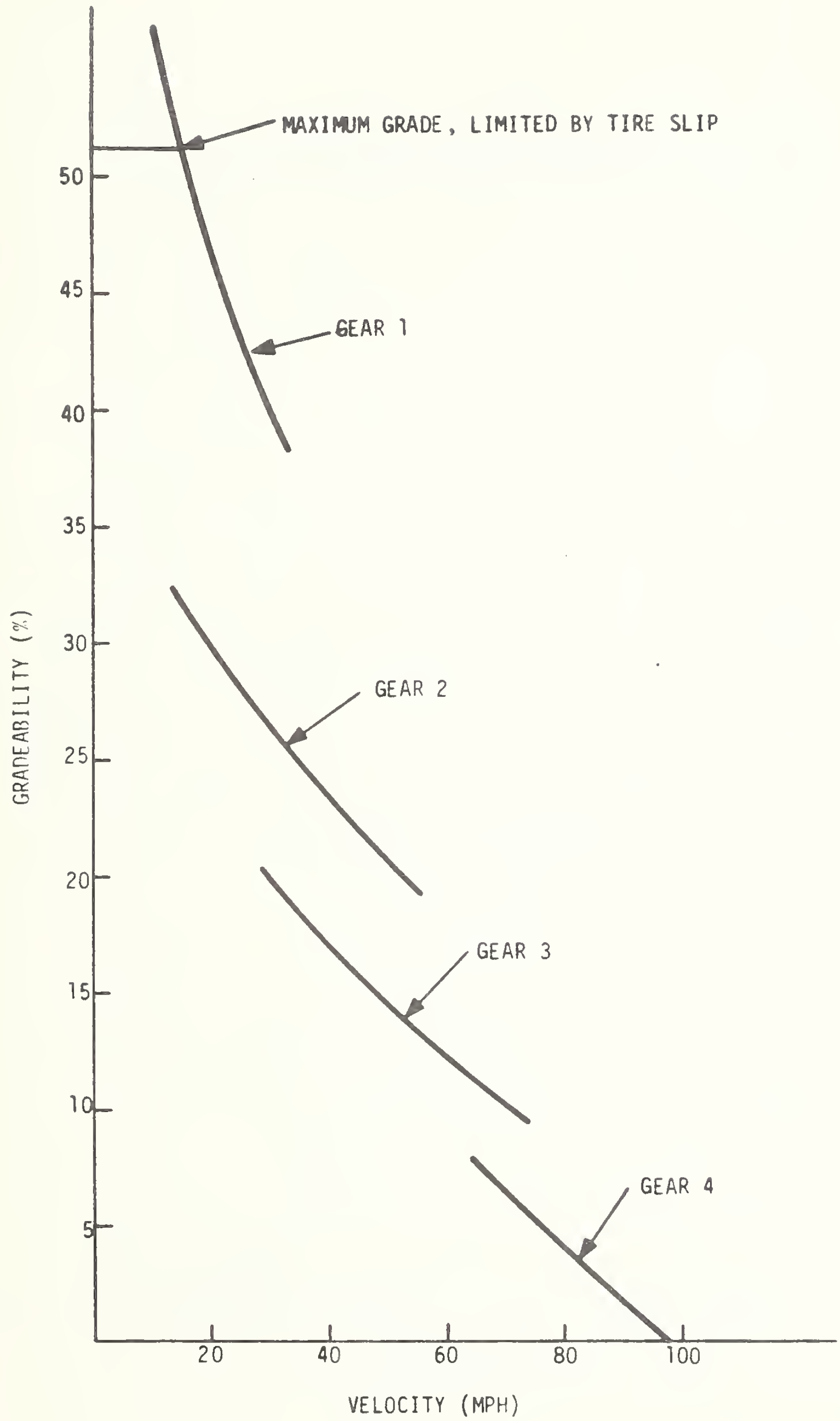


FIGURE 4-11. GRADEABILITY FOR A 2000 lb 98 CID AUTOMOBILE

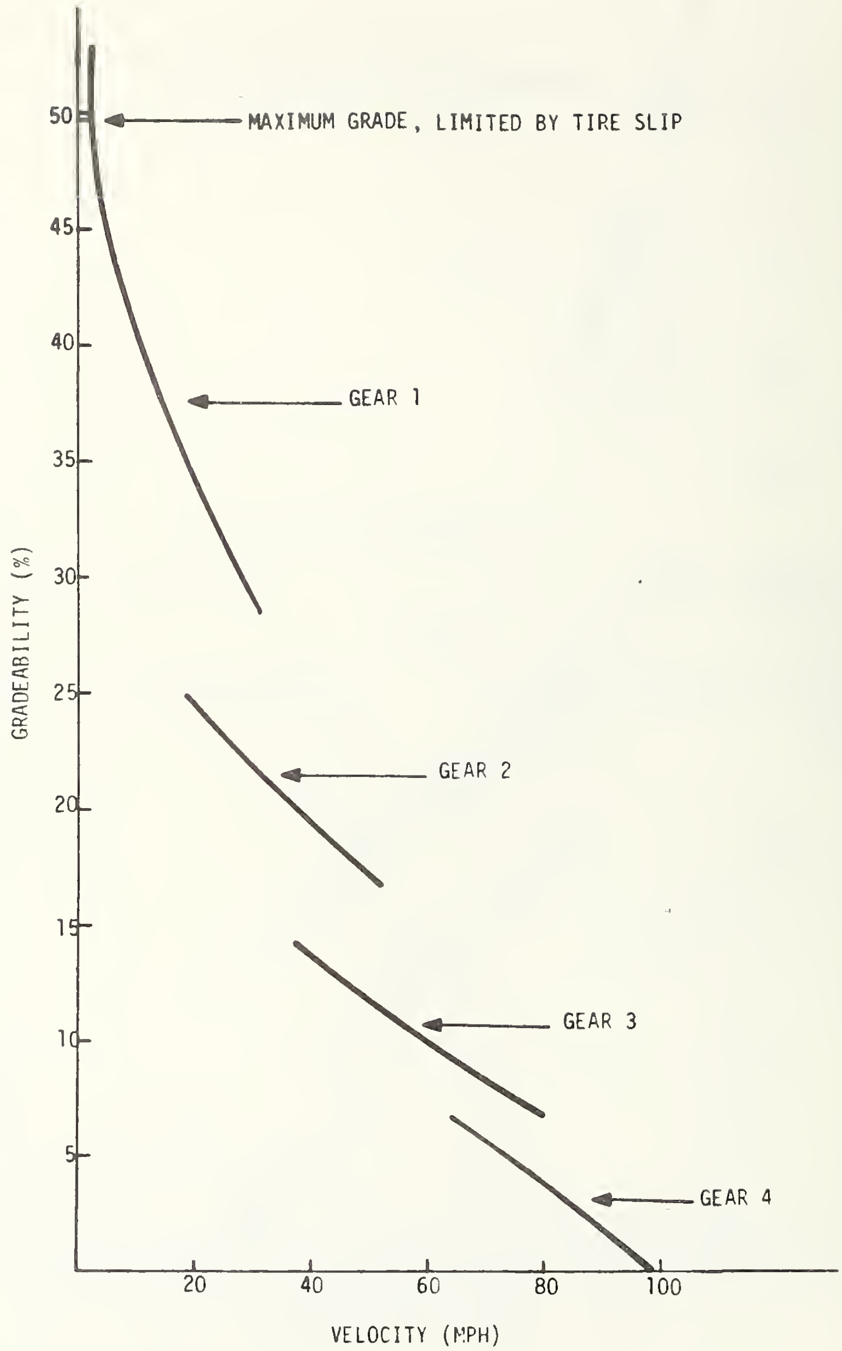


FIGURE 4-12. GRADEABILITY FOR A 3000 lb 130 CID AUTOMOBILE

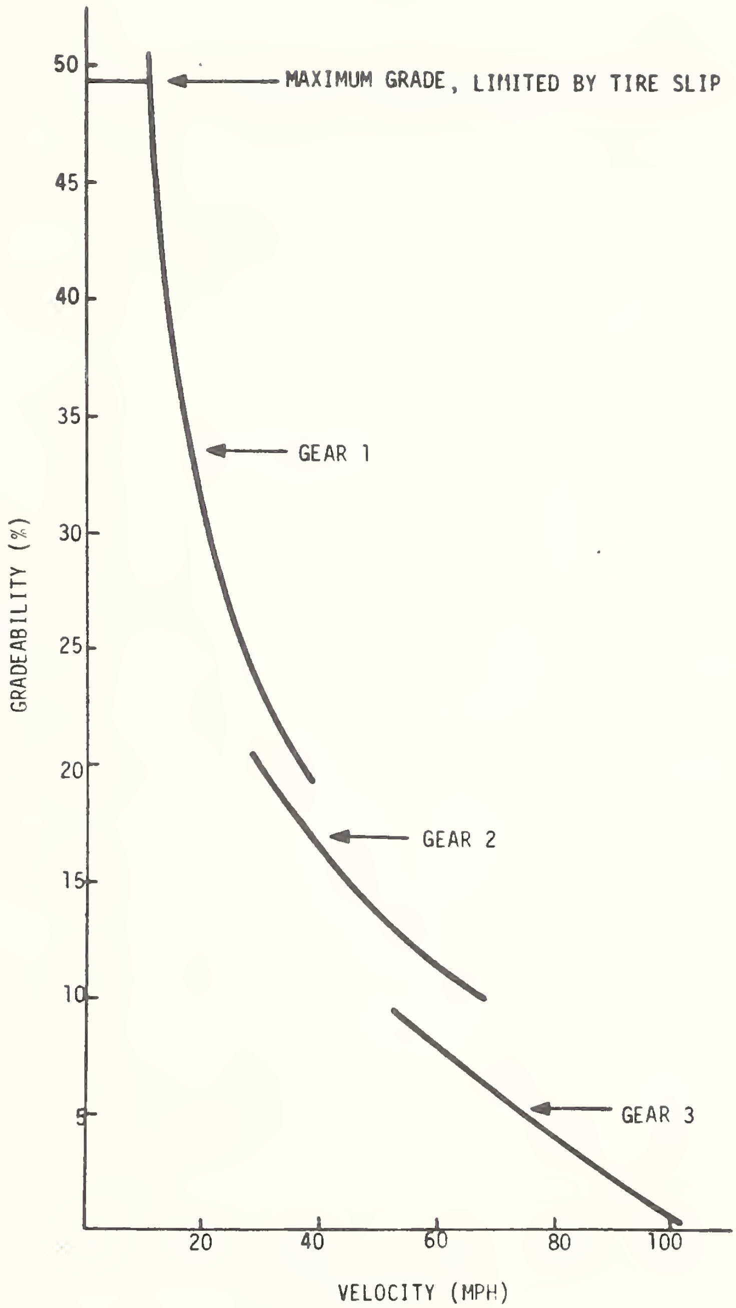


FIGURE 4-13. GRADEABILITY FOR A 3500 lb 231 CID AUTOMOBILE

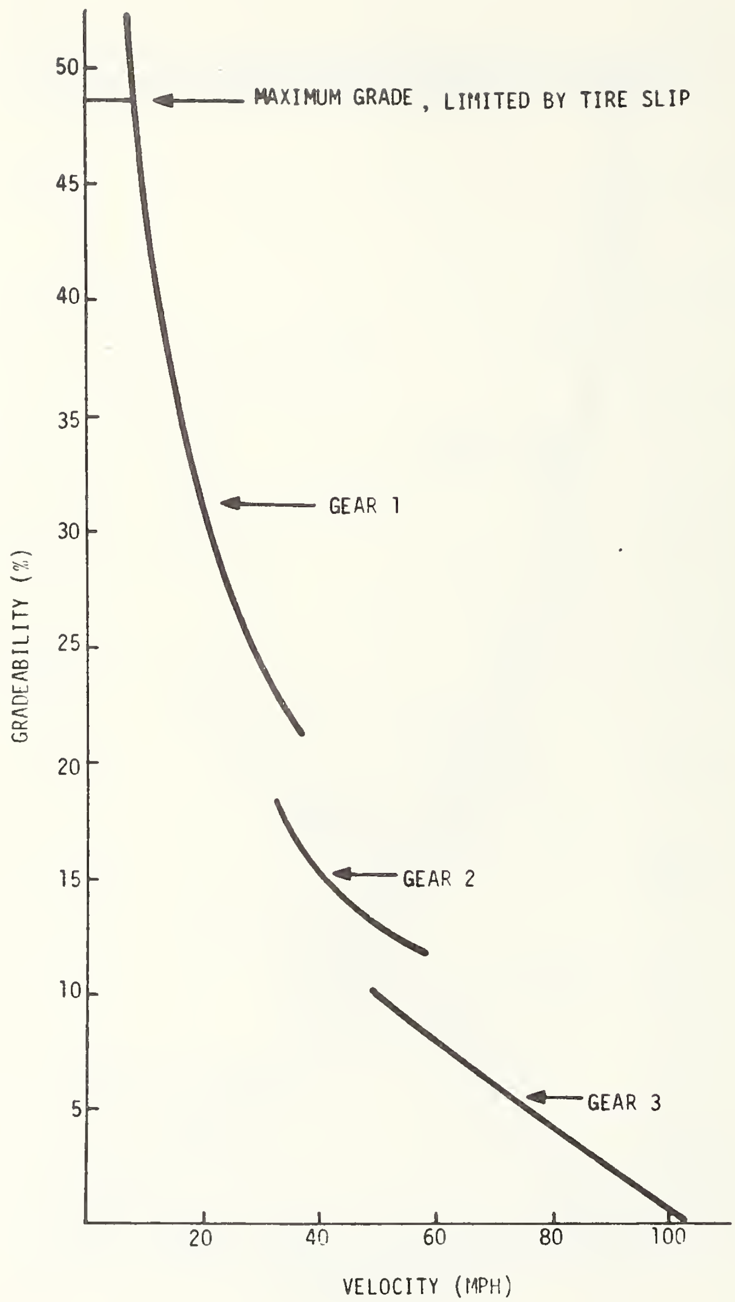


FIGURE 4-14. GRADEABILITY FOR A 4500 lb 318 CID AUTOMOBILE

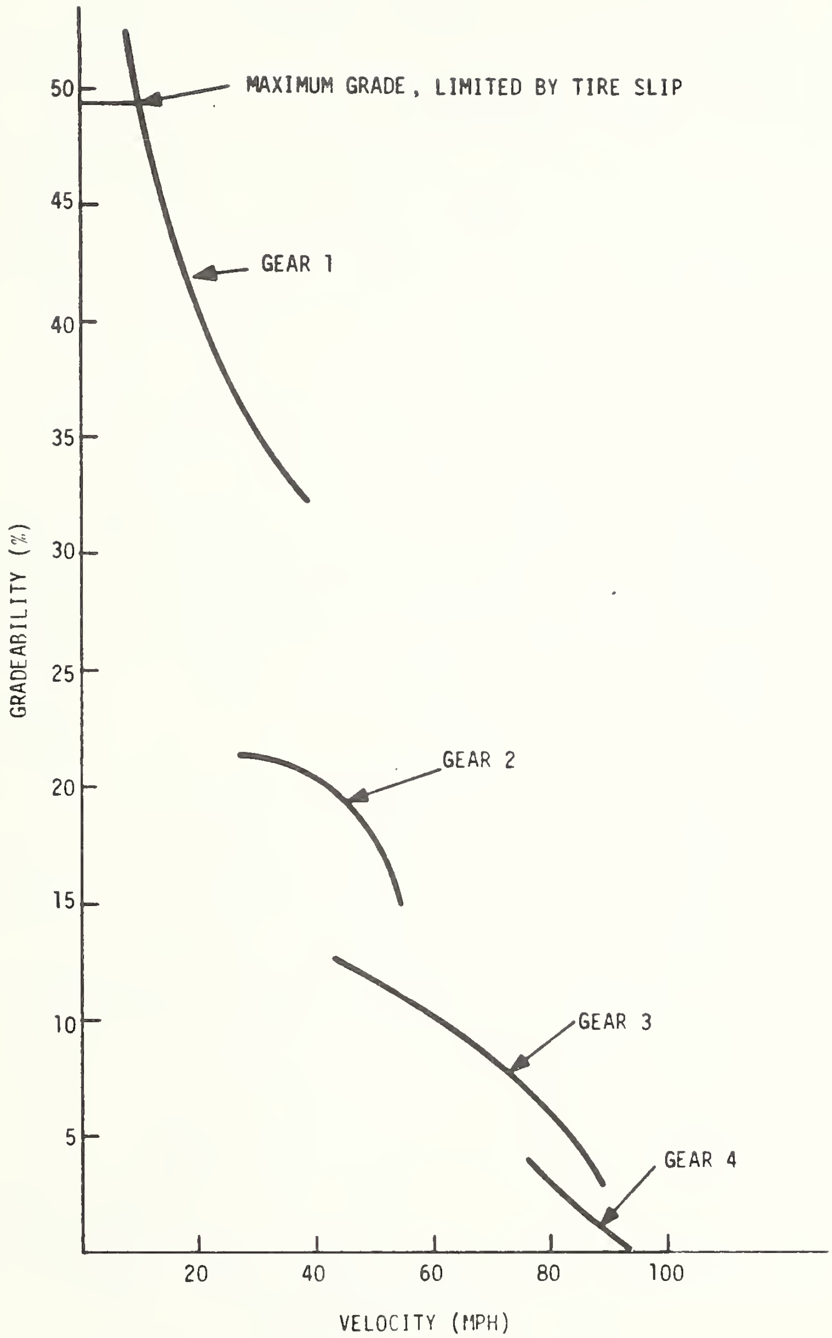


FIGURE 4-15. GRADEABILITY FOR A 3000 lb 140 CID MINI-PICKUP

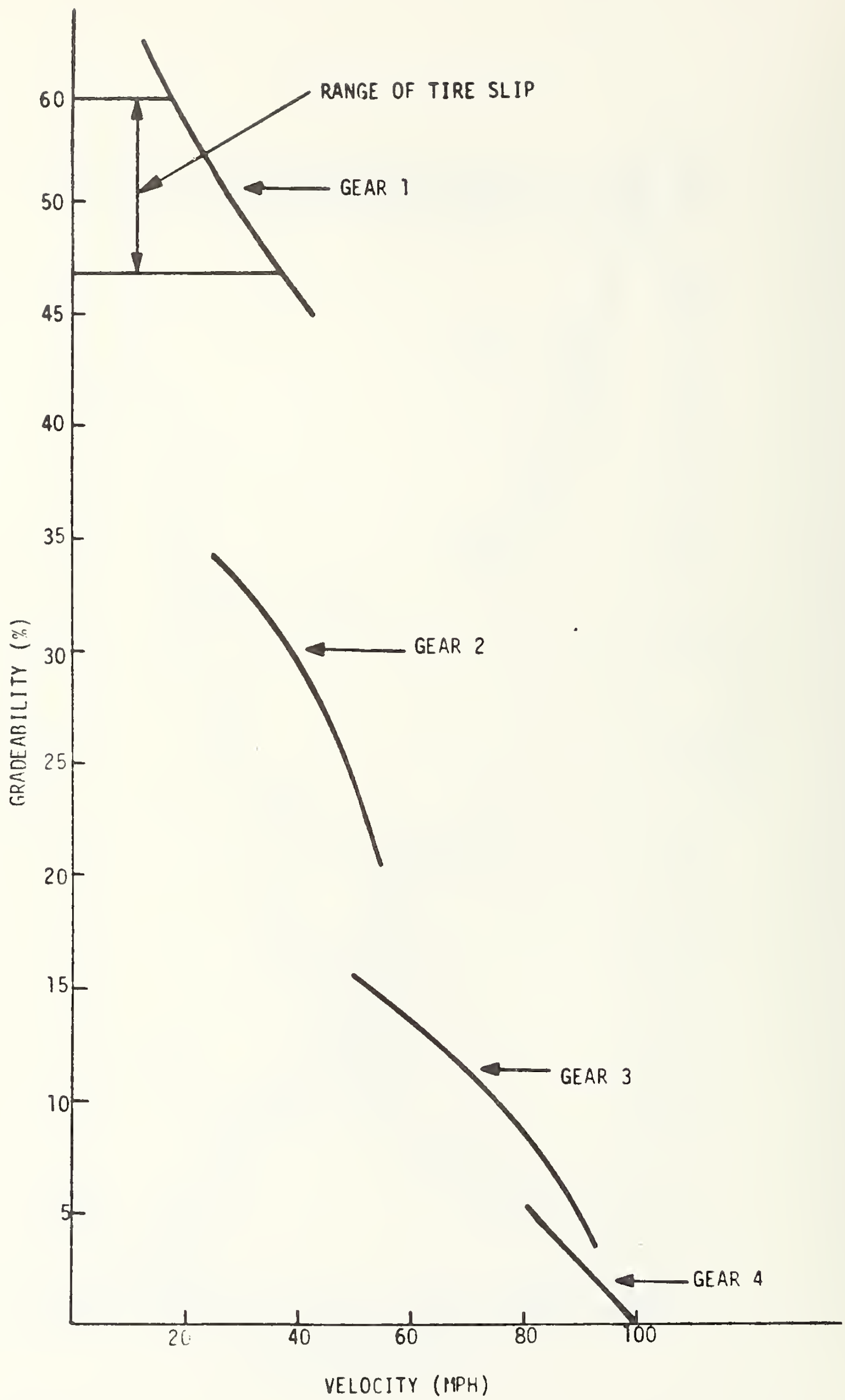


FIGURE 4-16. GRADEABILITY FOR A 4000 lb 351 CID LIGHT DUTY TRUCK

an automatic transmission will occur near the stall speed as the product of the torque and torque ratio will be a maximum at or very near the point.

The effect of a weight change on gradeability is shown in Table 4-4. At ten mph for most cases, tire slippage remains a counteracting factor. The gradeability at 55 mph should increase with a weight reduction. This increase is based partially upon the energy expenditure of a vehicle at 55 mpg. For example, because of the relatively large frontal area of the 351 CID vehicle, approximately 70 percent of the energy expended is used to overcome wind resistance. Therefore, a 5 percent reduction in weight is not going to affect the gradeability.

The gradeability change due to rear axle ratio variation is illustrated in Table 4-5. The higher rear axle ratio shifts the road load curve enabling the vehicle to negotiate a higher grade at the same velocity. The gradeability at ten mph for a rear axle change was omitted because tire slippage is limiting and thus a RAR change has no effect on gradeability for this case.

4.6 FUEL ECONOMY

The effect of aerodynamic drag, and tire rolling resistance on fuel economy is examined below. The fuel economy change connected with weight reduction is shown in Figures 4-17 and 4-18. Each improvement in vehicle road load loss not only has the direct effect of increasing the fuel economy, but also has the indirect result of allowing the vehicle to be equipped with a smaller engine. Therefore, each vehicle was equipped with a smaller engine geometrically downsized to maintain equal performance. An example of the fuel economy increase accomplished by maintaining equal performance is shown in Figure 4-19. The increase in fuel economy due to 1 percent weight reduction is 0.2 to 0.4 percent. The fuel economy gain, utilizing a geometrically smaller engine along with weight reduction at constant performance is shown in Table 4-6. It can be seen that the sensitivities of the vehicles are about 0.7 to 1.1, indicating that the percent change in weight

TABLE 4-4. GRADEABILITY RESPONSE TO VEHICLE WEIGHT CHANGE

BASE ENGINE/ VEHICLE	WT CHANGE (lb)	GRADEABILITY (% GRADE) 10 MPH	55 MPH
98 CID/PA	2000 → 1800	52[1] → 52[1]	14 → 15
130 CID/PA	3000 → 2200	41 → 50	11 → 15
231 CID/PA	3500 → 2450	50 → 50[1]	12 → 19
318 CID/PA	4500 → 3400	45 → 49[1]	9 → 12
140 CID/LT	3000 → 2600	50[1] → 50[1]	11 → 13
351 CID/LT	4000 → 3800	47/60[2,3] → 47/70[2,3]	15 → 15

1 Limited by tire slip

2 Road tire coefficient different for light-truck

3 For 4 WD gradeability will approach 80%.

TABLE 4-5. GRADEABILITY RESPONSE TO REAR AXLE RATIO CHANGE

BASE ENGINE/ VEHICLE	REAR AXLE RATIO	GRADEABILITY (% GRADE 55 MPH)
98 CID/PA	3.58 → 4.0	14 → 15
130 CID/PA	3.91 → 4.2	11 → 12
231 CID/PA	2.56 → 3.0	12 → 13
318 CID/PA	2.71 → 3.4	9 → 11
140 CID/LT	3.08 → 4.2	11 → 15
351 CID/LT	3.54 → 4.2	15 → 17

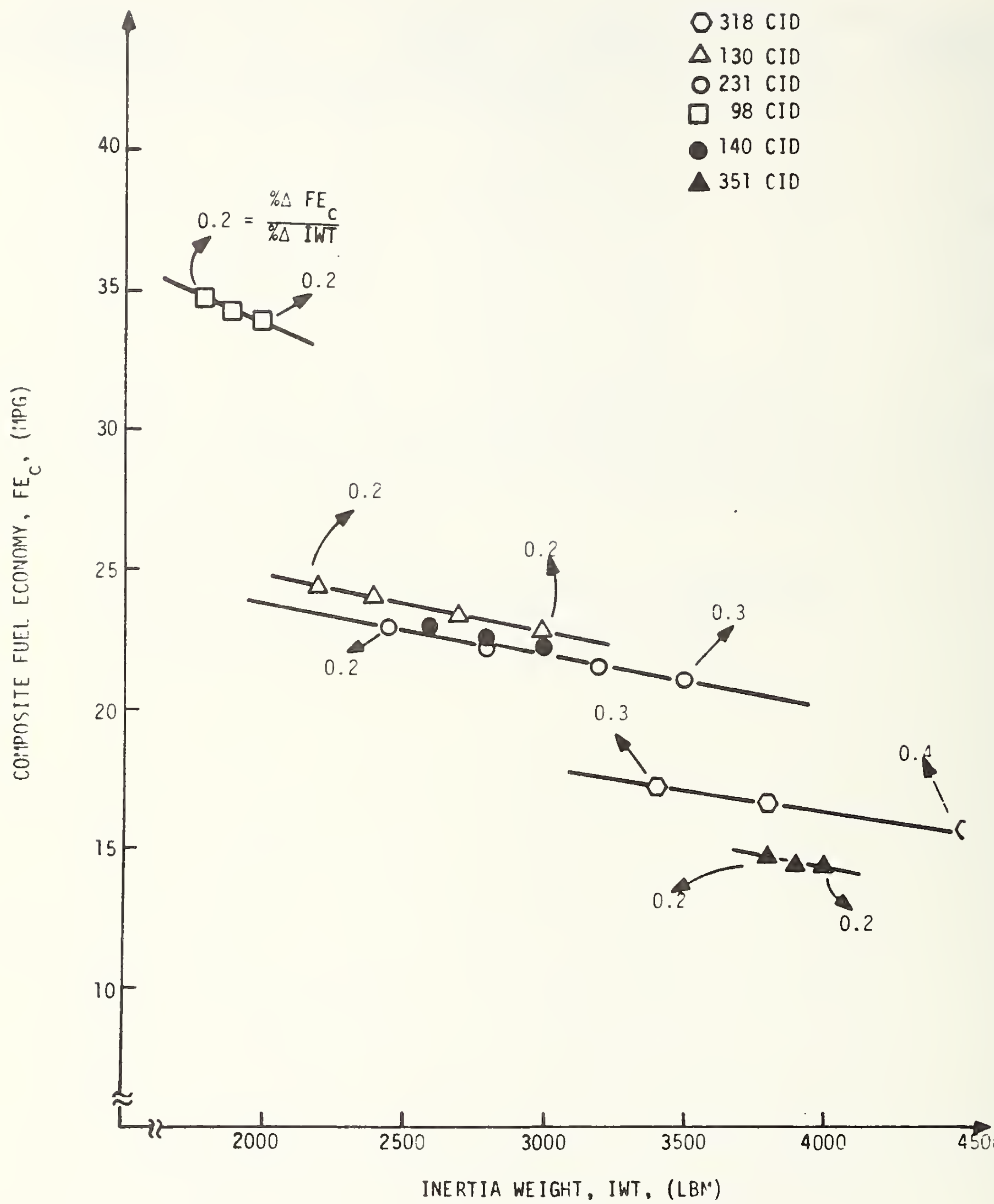


FIGURE 4-17. FUEL ECONOMY AS A FUNCTION OF WEIGHT FOR BASE ENGINES

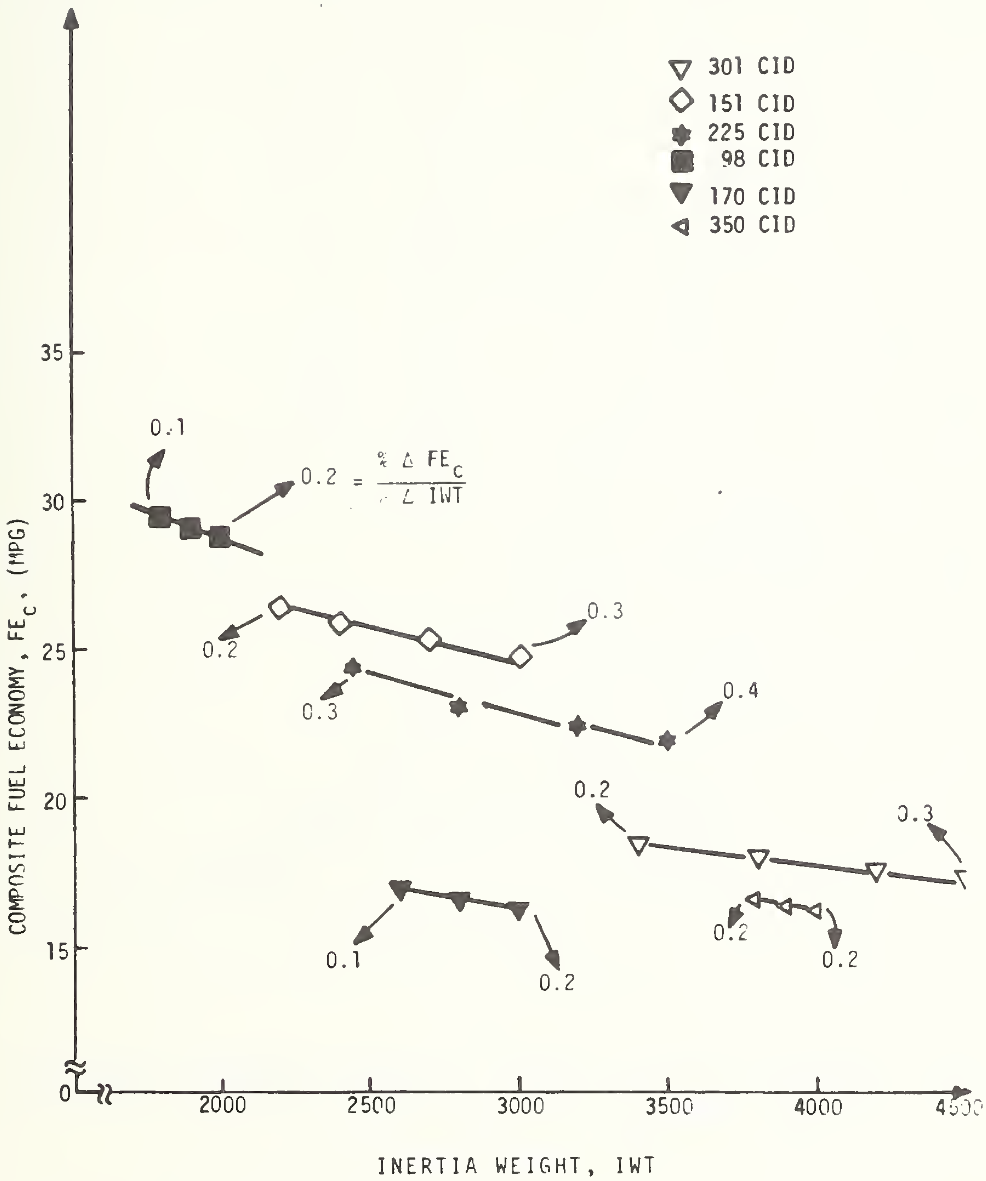


FIGURE 4-18. FUEL ECONOMY AS A FUNCTION OF WEIGHT FOR ALTERNATE ENGINES

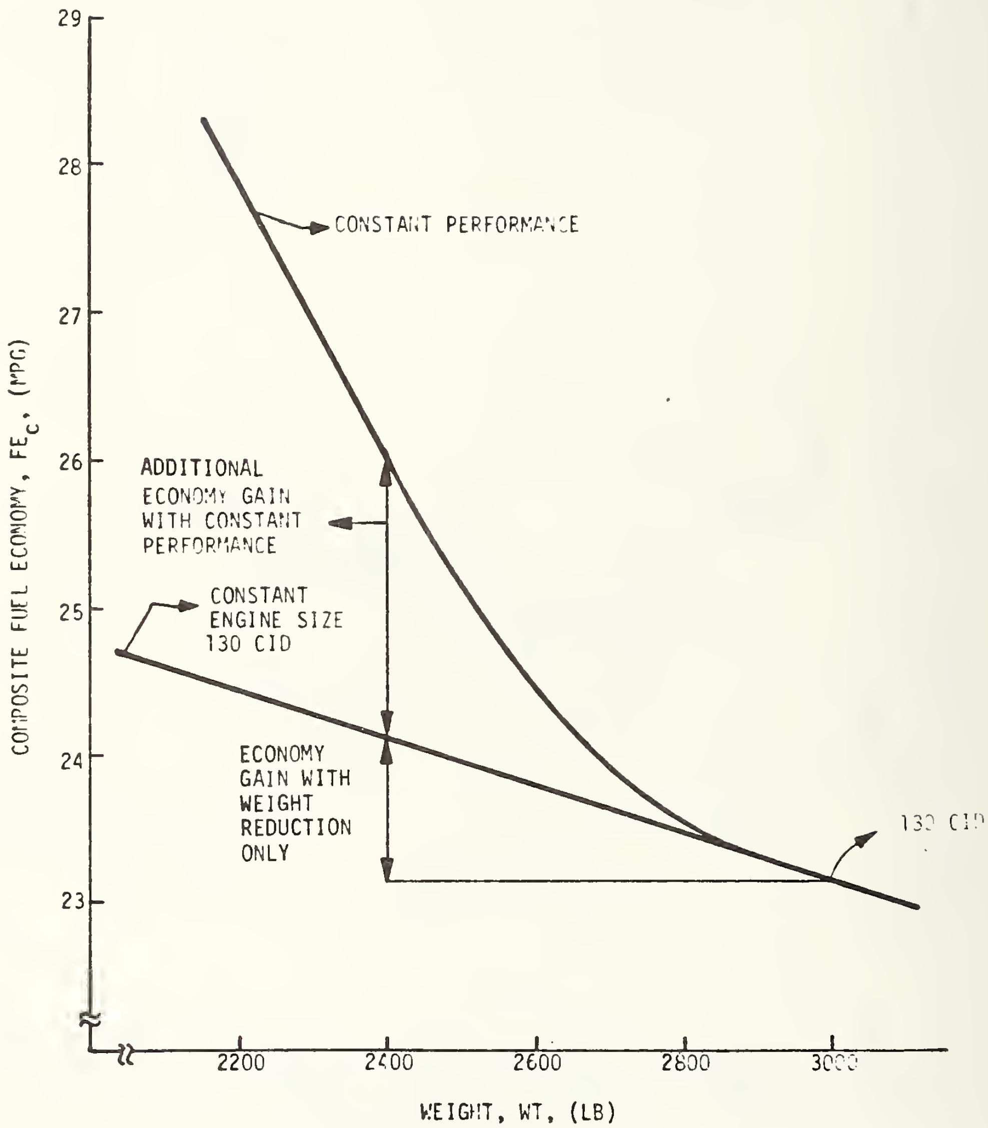


FIGURE 4-19. EFFECT OF WEIGHT ON FUEL ECONOMY FOR 130 CID 3000 lb VEHICLE

TABLE 4-6. EFFECT OF WEIGHT REDUCTION ON FUEL ECONOMY FOR CONSTANT PERFORMANCE

ENGINE MODIFICATION (CID)	WEIGHT REDUCTION (lb)	FE_C INCREASE (%)	SENSITIVITY $\frac{\%\Delta FE_C}{\%\Delta WT}$
318 → 240	4500 → 3400	25.3	1.04
130 → 96	3000 → 2200	26.4	1.00
231 → 160	3500 → 2450	32.4	1.09
98 → 90	2000 → 1800	7.1	0.71
140 → 120	3000 → 2600	14.2	1.07
351 → 331	4000 → 3800	4.7	0.91

reduction is accompanied by a nearly equivalent percentage change in fuel economy.

The effect of aerodynamic drag reduction on fuel economy is shown in Figure 4-20. The effect of the rolling resistance coefficient on fuel economy is presented in Figure 4-21. The sensitivities for the extremes are also included. In order to apply the full potential of a C_D (aerodynamic drag coefficient) and C_1 (rolling resistance coefficient) reduction, the rear axle ratio is changed to bring the new road load curve into alignment with the original curve.[2] Because the C_1 and C_D reduction provides a relatively smaller improvement in road load losses, it is more realistic to change the rear axle ratio rather than scale the engine as was done in the case of the weight reduction studies. An example of a drag coefficient reduction with equal performance is shown in Figure 4-22. For a 1 percent drag coefficient reduction, the fuel economy improvement is 0.2 percent. By reducing the rear axle ratio, this gain is increased to 0.3 percent.

The increase in fuel economy as a result of a rear axle ratio reduction is shown in Figures 4-23 and 4-24.

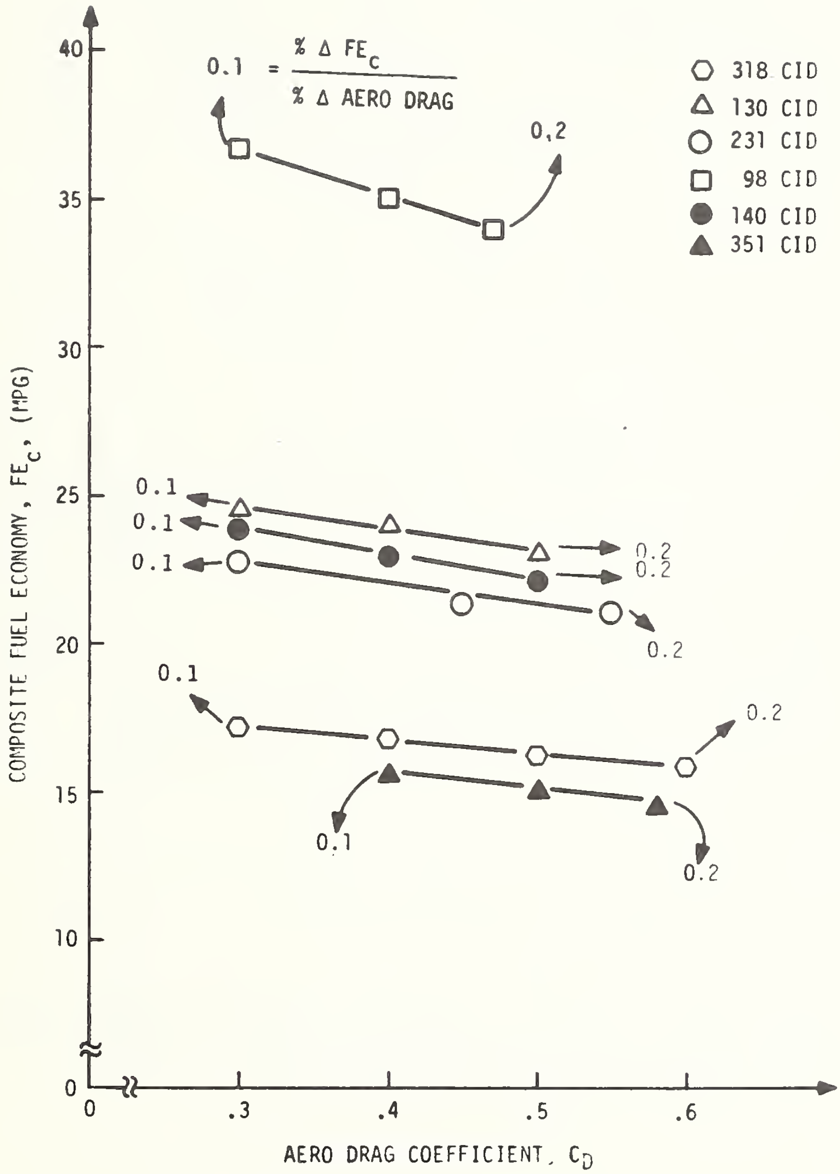


FIGURE 4-20. EFFECT OF AERODYNAMIC DRAG REDUCTION ON FUEL ECONOMY

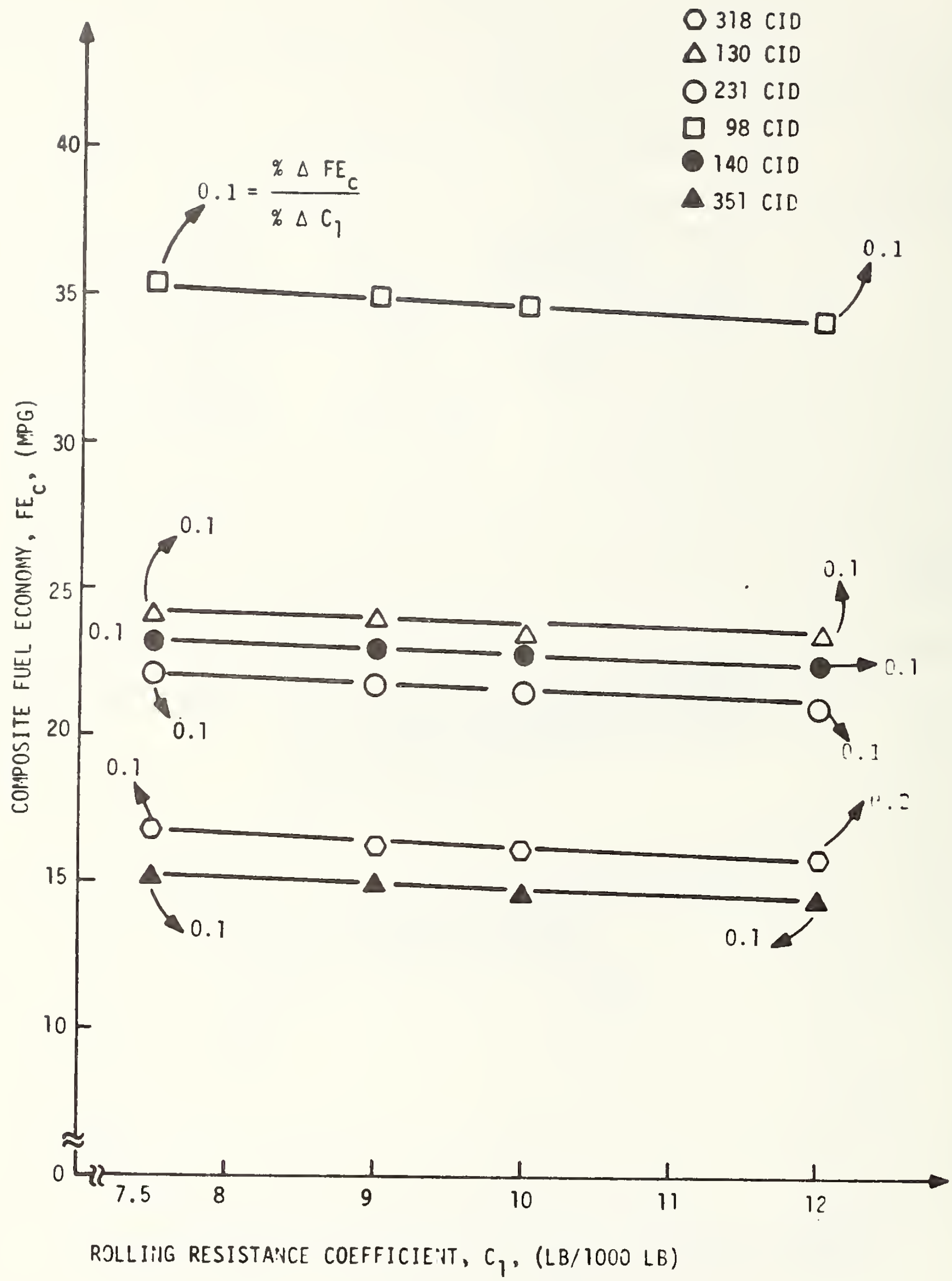


FIGURE 4-21. EFFECT OF ROLLING RESISTANCE COEFFICIENT ON FUEL ECONOMY

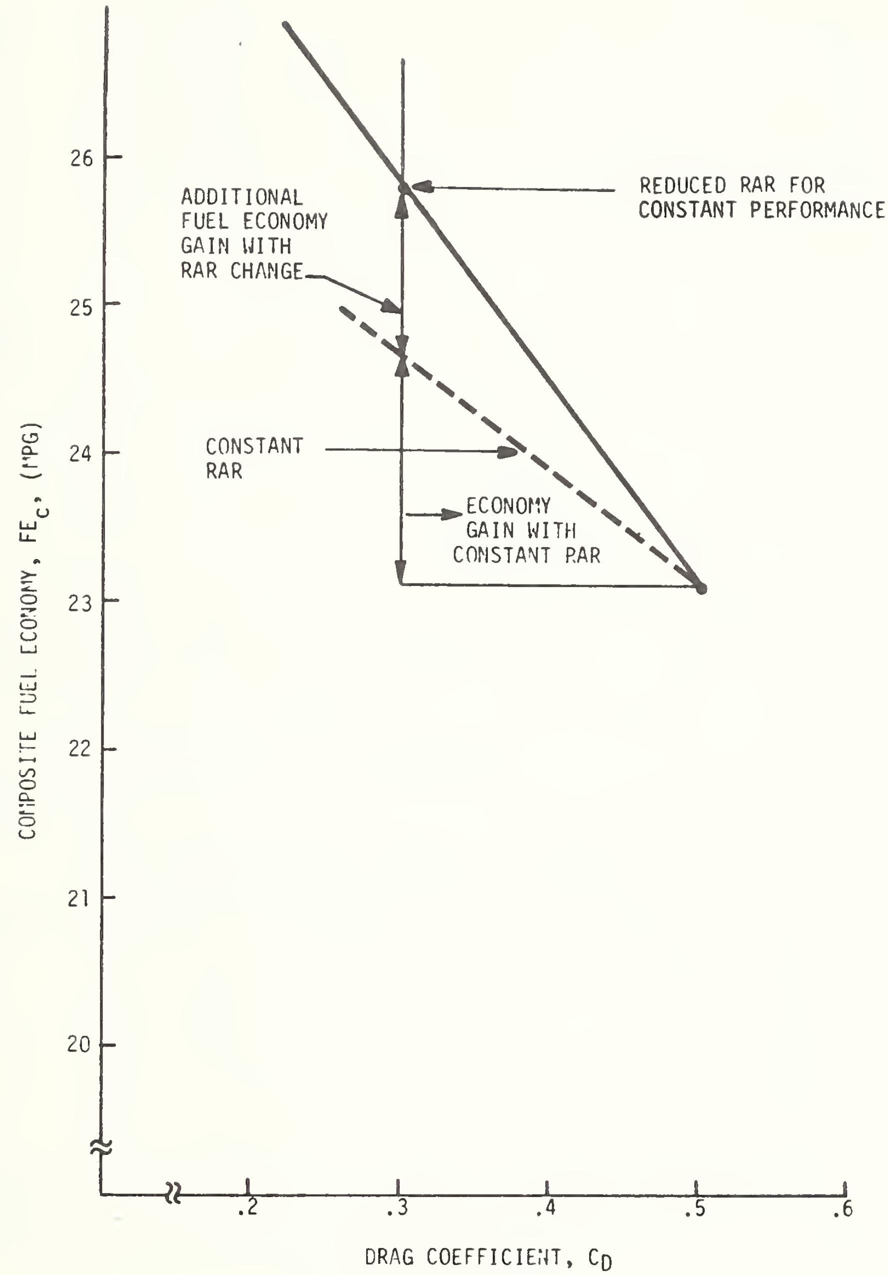


FIGURE 4-22. FUEL ECONOMY AS A FUNCTION OF C_D FOR DIFFERENT REAR AXLE RATIOS FOR 130 CID 3000 lb VEHICLE

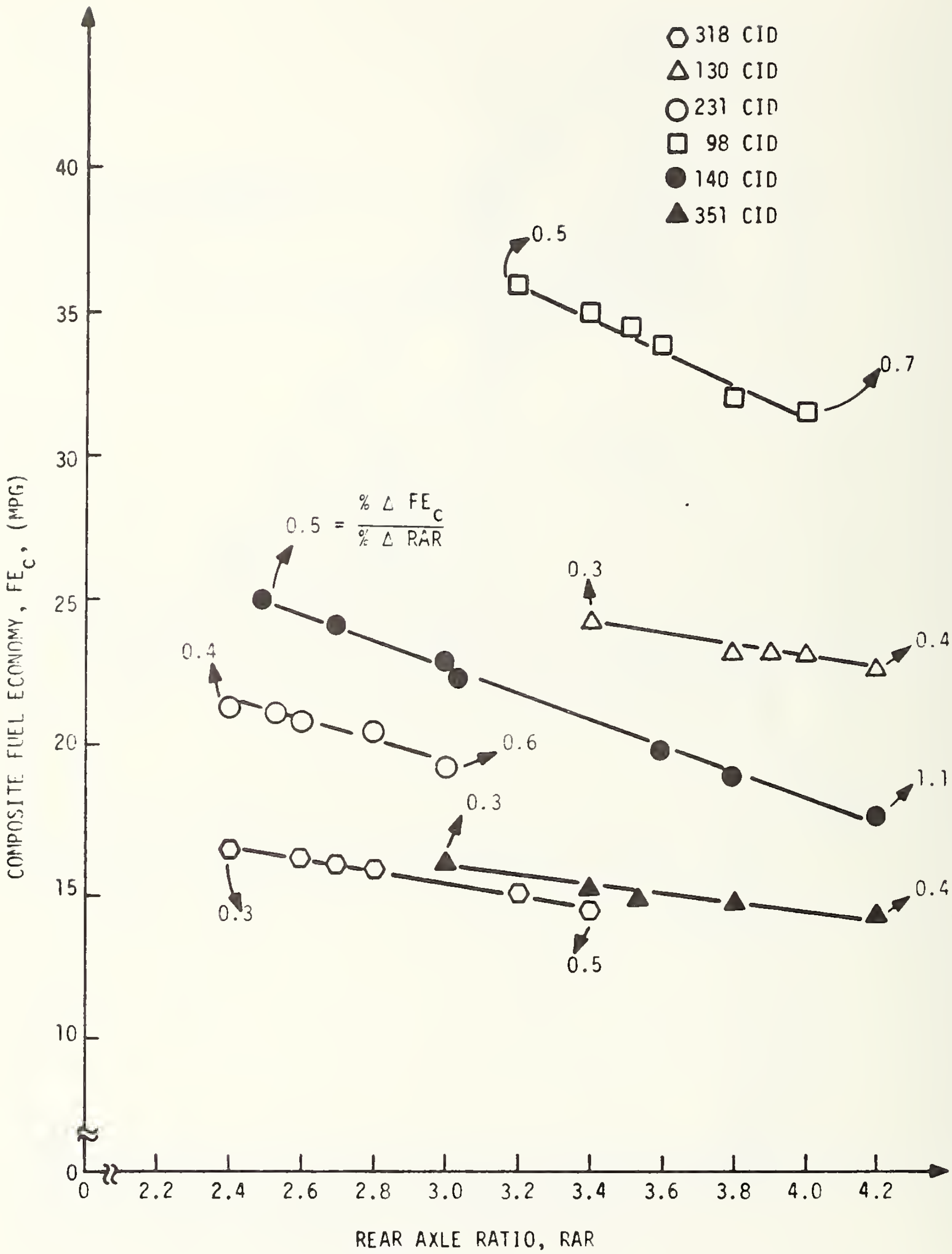


FIGURE 4-23. EFFECT OF REAR AXLE RATIO ON FUEL ECONOMY FOR BASE ENGINES

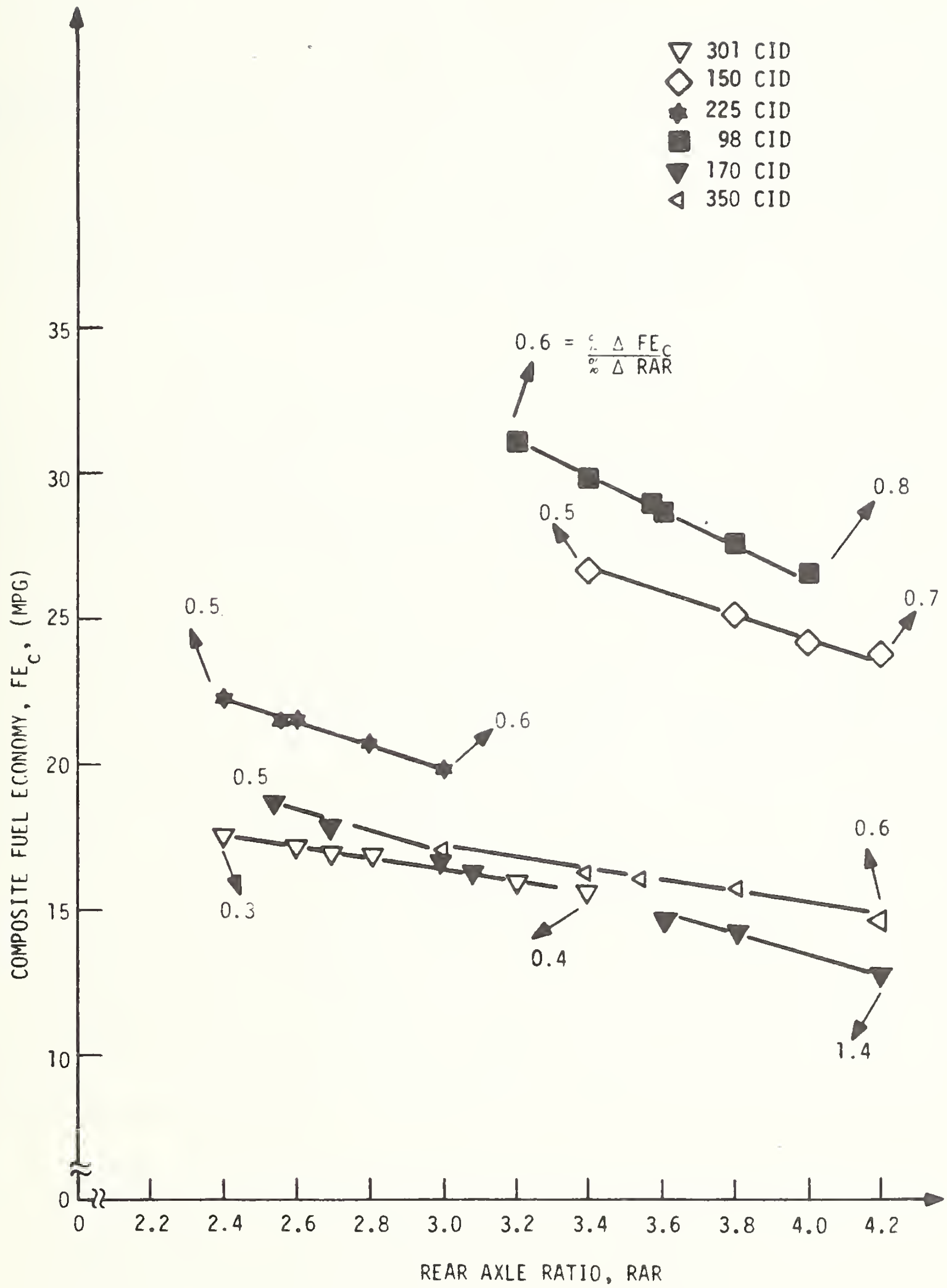


FIGURE 4-24. FUEL ECONOMY RESPONSE TO REAR AXLE RATIO CHANGE FOR ALTERNATE ENGINES

REFERENCES FOR SECTION 4

1. Zub, R.W. et al., "Effect of Vehicle Design Variables on Top Speed Performance, and Fuel Economy," U.S. Department of Transportation, Research and Special Programs Administration, Transportation Systems Center, Final Report, DOT-TSC-NHTSA-79-53, October 1979.

2. Jassen, L.J. and Emmelmann, H.J., "Aerodynamic Improvements - A great Potential for Better Fuel Economy," SAE Paper 780265, 1978.

3. Hsia H. and Kidd, J.A., "The Weight Reduction Potential of Automobiles and Light Trucks," U.S. Department of Transportation, Research and Special Programs Administration, Transportation Systems Center, Final Report, DOT-TSC-NHTSA-79-54, October 1979.

4. Burke, Carl E., et al., "Where Does All The Power Go?" SAE Symposium Meeting, Atlantic City, June 6, 1956.

5. FUELS/LUBRICANTS

5.1 ENGINE FUELS

5.1.1 General

Gasoline today consists of blends of hydrocarbons derived from petroleum. Gasoline properties are influenced by refinery practices and the nature of the crude from which they are produced. Agents are also blended with gasoline to ensure that the final gasoline product satisfied the requirements for:

a) antiknock quality,

b) distillation characteristics,

c) vapor pressure,

d) sulfur content,

e) oxidation stability, and

f) anticorrosion behavior.

Additives are used to provide or enhance specific performance features. Finished gasolines encompass a boiling range of about 85°F through 437°F. Historically, automotive gasolines have been classified on the basis of their antiknock quality.

Currently, the engine manufacturers require the gasoline to meet specifications on research octane number and certain properties which enable the engine to maintain reliability and durability over its life cycle. Some of these requirements may change, and others may be added because of:

a) emphasis to achieve increased engine efficiencies, and performance and requirements to meet future regulated and non-regulated emission standards. This must be in-line with overall requirements for efficient processing and refining procedures with consequent improvement in volumetric yield per barrel of crude at minimum cost.

b) greater dependence on residual fuel oil which must be hydrocracked to produce gasolines comparable to present day gasolines. Because of the cost of this process, greater use of broad-cut fuels is being urged in spark ignition engines today. These fuels will require specific examination.

c) requirements to consider alternative fuels with:

   1) emphasis on synfuels to displace the current gasoline pool. These fuels are chiefly derived from a non-petroleum source. Shale and coal are typical fuels.

   2) encouragement to use alcohols either for extenders of present day gasolines or by direct use in engines. Methanol and ethanol are typical alcohols under consideration.

The following discussion focus on some critical factors and requirements under consideration for gasoline fuels for passenger cars and light trucks.

5.1.2 Requirements/Quality

Currently, the fuel specifications shown in Table 5-1 apply to gasoline for exhaust and evaporative emission testing.[1]

5.1.2.1 Antiknock Quality - The antiknock quality of a spark ignition engine is important, and is highly interrelated with engine design and operating conditions. It is measured by several methods, which employ single-cylinder laboratory engines (research method ASTM D 2690) and multi-cylinders in cars (motored method ASTM D 2700). Both of these test procedures employ a variable-compression engine. The Motor method operates at a higher speed and inlet mixture temperature than the Research method; thus, it is more typical of driving. The Research Octane number (RON) is, in general, the better indicator of antiknock quality for engines operating at full throttle, low engine speeds. Motor Octane number (MON) is the better indicator at full throttle, high

TABLE 5-1. GASOLINE SPECIFICATIONS

Item Designation	ASTM	Leaded	Unleaded
Octane, research, minimum	D 2699	100	96
Pb. (organic), grams/U.S. gallon.....................	......	1.4 min.	0.0-0.5
Distillation range:			
IBP[a] °F	D86...	75-95	75-95
10 percent point, °F	D86...	120-135	120-135
50 percent point, °F	D86...	200-230	200-230
90 percent point, °F	D86...	300-325	300-325
EP, °F (maximum)	D86...	415	415
Sulfur, weight percent, maximum....................	D1266.	0.10	0.10
Phosphorus, grams/U.S. gallon, maximum............	.....	.01	.005
RVP[b][c] pounds..............	D323..	8.7-9.2	8.7-9.2
Hydrocarbon composition:			
Olefins, percent, maximum...	D1319.	10	
Aromatics, percent, maximum.	D1319.	35	35
Saturates..................	D1319	Remainder	Remainder

(a) For testing at altitudes above 1,219 meters (4,000 feet) the specified range is 75-105.

(b) For testing which is unrelated to fuel evaporative emission control, the specified range is 8-9.2.

(c) For testing at altitudes above 1,219 meters (4,000 feet) the specified range is 7.9-9.2.

Source: Reference 1

engine speeds, and part throttle, low and high engine speeds. The average of Research (RON) and Motor (MON), that is (RON + MON)/2, is considered to be a reasonable guide to a gasoline's road octane performance. The ASTM and Federal gasoline specifications contain (RON + MON)/2 minimums for various grades. The antiknock requirements for gasolines contained in SAE J 312d are listed in Table 5-2.[2]

5.1.2.2 Volatility - Volatility is connected with the initial boiling point characteristic of the fuel: it affects the quality of fuel vaporized prior to the start of combustion. This property is of prime importance with respect to driveability which relates, in addition, to starting, vaporlock, stalling and warmup, and acceleration performance of the vehicle. Since gasoline is a mixture of many hydrocarbons, it does not have a single boiling point. Thus, the temperatures at which 10, 50 and 90 percent have evaporated, generally, are used to characterize the volatility of gasolines. Volatility characteristics of automobile gasolines, defined by ASTM D 439 "Standard Specifications for Automobile Gasoline," are specified for areas and seasons. Table 5-3 shows the ASTM specifications which provide five volatility classes.[3] Varying limits are placed on the maximum or minimum vaporization tendency of the fuel adjusted for seasonal and geographical changes in temperature, and for the altitudes at which the gasoline is to be used. These volatility characteristics have been established on the basis of broad experience and cooperation between gasoline suppliers and manufacturers of automotive vehicles.

5.1.2.3 Additives - Gasoline additives are used to enhance or provide various performance features related to the satisfactory operation of engines, as well as to minimize gasoline handling and storage problems. These compounds complement refinery processes in attaining the desired level of product quality.

TABLE 5-2. ANTIKNOCK REQUIREMENTS FOR GASOLINE

Antiknock Index (RON + MON)/2 Minimum[a]	Application
85	For cars with low antiknock needs.
87[b]	Meets antiknock needs of most 1971 and later model cars.
89	For most 1970 and prior model cars designed to operate on "regular" gasoline, and for 1971 and later model cars that have higher antiknock requirements.
91	An antiknock level which meets the lower requirements of some cars designed for "premium" and the higher requirements of some cars designed for "regular" gasoline.
93 95	For most 1970 and prior model cars designed for "premium" gasolines and for later model cars with high antiknock requirements.
97	For cars designed for "premium" but having higher antiknock requirements than above.

[a] Reductions for altitude are allowed in accordance with Table 5-1.

[b] In addition, gasolines with an antiknock index of 87-88.9 must have a Motor Octane number not less than 82.0

Source: Reference 2

The amount and variety of additives used in gasolines have grown rapidly. For many years, antiknocks were almost the only additives used; however, the list now also includes combustion chamber deposit modifiers, antioxidants, metal deactivators, corrosion or rust inhibitors, carburetor anti-icing additives, fuel line antifreeze agents, gasoline detergents, gasoline dispersants, and identifying dyes.

Gasoline additives currently available are shown by class and function in Table 5-4.

TABLE 5-3. VOLATILITY REQUIREMENTS

Gasoline Volatility Class	Distallation Temperatures °C(°F), at Precent Evaporated										Dist. Residue Max. %	Vapor/Liquid Ratio[a]				
	max 10[a]		min 50[a]		max 50[a]		max 90[a]		End Point			Test Temperature		V/L max	Reid Vapor Pressure, Max	
	°C	°F	°C	°F	°C	°F	°C	°F	°C	°F		°C	°F		kPa	(lb)
A	70	158	77	170	121	250	190	374	225	437	2	60	140	20	62	(9.0)
B	65	149	77	170	118	245	190	374	225	437	2	56	133	20	69	(10.0)
C	60	140	77	170	116	240	185	365	225	437	2	51	124	20	79	(11.5)
D	55	131	77	170	113	235	185	365	225	437	2	47	116	20	93	(13.5)
E	50	122	77	170	110	230	185	365	225	437	2	41	105	20	103	(15.0)

(a) At 760 mm Hg pressure (101.3 kPa).

SOURCE: Reference 3

TABLE 5-4. COMMERICAL GASOLINE ADDITIVES - FUNCTION AND TYPE

Class or Function	Common Additive Type
1. Antiknock Compounds—to improve Research, Motor and road octane quality.	Lead alkyls, such as tetraethyllead, tetramethylead, and physical or reacted mixtures. Organo-manganese compounds, such as methyl cyclopentodenyl manganese tricabonyl (MMT).
2. Combustion Deposit Modifiers—to minimize surface ignition, rumble, preignition, and spark plyg fouling.	Organic or organo-metallic compounds usually containing phosphorus.
3. Antioxidants—to minimize oxidation and gum formation in gasoline and to improve handling and storage characteristics.	Phenylene diamine, phenol, and aminophenol compounds.
4. Metal Deactivators—to deactivate small traces of copper and other metal ions that are powerful oxidation catalysts.	Diamine and aminophenol compounds.
5. Corrosion or Rust Inhibitors—to minimize corrosion and rusting of fuel system and storage handling facilities.	Derivatives of carboxylic, sulfonic, or phosphoric acids, many of which have surface-active properties.
6. Carburetor Anti-Icing Additives—to minimize engine stalling due to ice accumulation of the throttle.	Derivatives of carboxylic, sulfonic, or phosphoric acid that have surface-active properties. Freeze point depressants such as alcohols and glycots.
7. Gasoline Detergents—to remove and or minimize the accumulation of deposits in the throttle section of the carburetor, which adversely affects the metering characteristics	Amines and derivatives of carboxylic, sulfonic, or phosphoric acid having surface-active properties, some of which are polymeric
8. Gasoline Dispersants—to extend PCV valve life, reduce engine sludge, and remove and or minimize the accumulation of deposits in the carburetor, intake manifold, intake ports, and underside of the intake valves.	Amines and low molecular weight synthetic polymers. Specific tractions of special oils.
9. Dyes-to identify gasoline blends.	Oil Soluble solid and liquid dyes.
Note: Some materials may also be marketed as multifunctional or multipurpose additives, performing more than one of the functions described in items 5, 6, 7, and 8.	

Source: Reference 3

5.1.3 Alternative Fuels

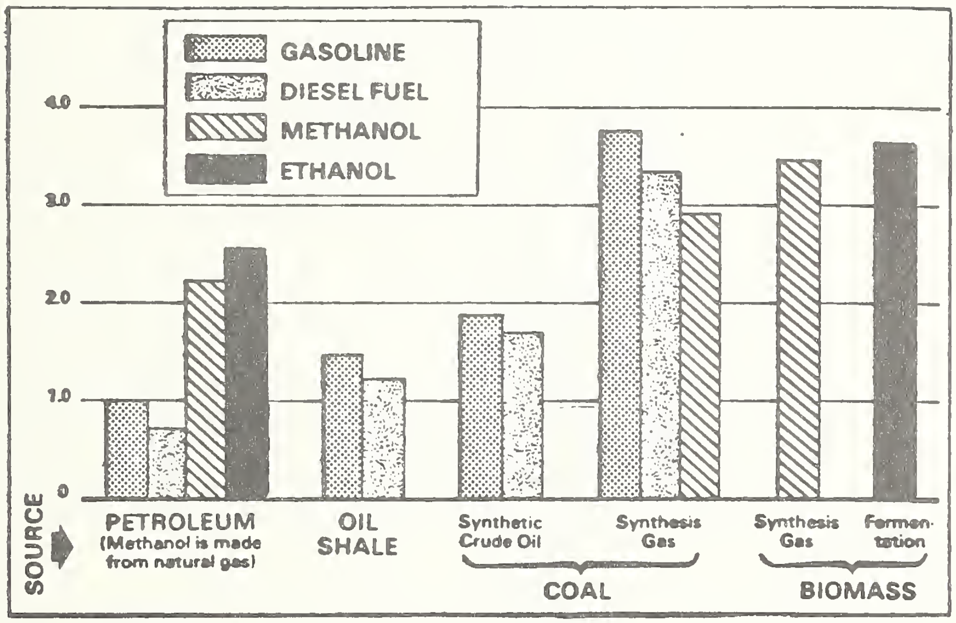
5.1.3.1 General - Various alternative fuels which include distillates such as broad-cut and variable composition fuels based on petroleum, coal, oil shale, biomass, etc., as well as transition fuels, are under consideration today. Alcohols are also a major subject of interest. These include methanol, ethanol, higher alcohols, and blends of these with hydrocarbons.

These fuels are unattractive because of the cost to produce them relative to cost of present day gasoline (See Figure 5-1). It is thought that some of these fuels can be economically produced at a cross over point when imported crude oil approaches $30./bbl.[4]

5.1.3.2 Oil Shale Derived Fuels - Fuels synthetically derived from oil shale are considered today as possible replacements for petroleum-based fuels for spark ignition engines.

Technologies for extracting petroleum-type liquids from oil shale have been in development for many years. Several of these technologies which are in various stages of development basically rely on a retorting (pyrolysis) procedure using mined shale. In-situ retorting procedures, which involve the in-place heating of an underground shale formation, are also under experimental development to produce crude shale oil. However, these in-situ procedures have not progressed in development to the extent that the above ground retorting procedures have. During retort, a waxy substance called kerogen begins to release crude shale oil and gas when the shale nears 900°F. Two small scale commerical plants are being discussed today for operation by 1982, which will have a capacity to deliver 10,000 bbl/day. Presently, both processes consume large amounts of water. As a result, harmful waste materials can be leached into aquafers and rivers presenting serious environmental problems.

Currently, the U.S. Western oil shales yield, approximately, an average of 30 gallons/ton. The most developed retorting processes include:



Source: Reference 4

FIGURE 5-1. RELATIVE COSTS OF ALTERNATIVE FUELS COMPARED TO PETROLEUM BASED ENERGY

a) Indirect-heated types

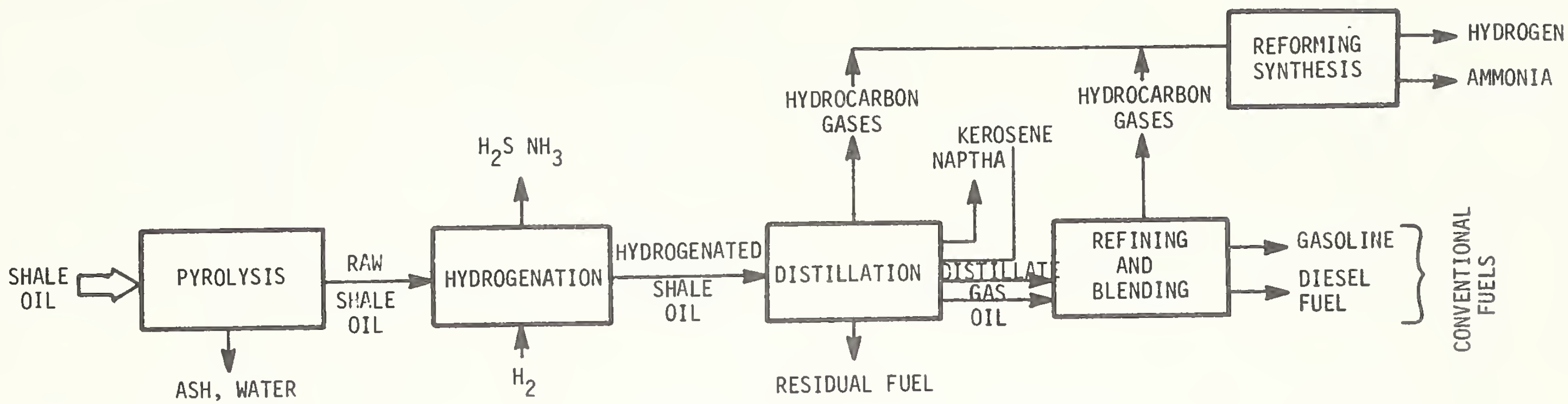
- o Union Oil B
- o Tosco II
- o Petrosix
- o USSR Kiviter and Galoter
- o Lurgi/Ruhrgas
- o Paraho Indirect

b) Direct-Heated types

- o Gas combustion
- o Union Oil Company
- o Paraho-direct.

Direct-heated processes rely on internal combustion of fuel with air or oxygen within the bed of shale in the retort to produce the crude shale oil. Indirect-heated processes utilize a separate furnace for heating solid or gaseous heat-carrier media which are injected, whole hot, into the shale in the retort. The shale oil processes listed above produce an essentially identical product. Generally the crude shale oil has little value as a fuel since it is composed of approximately half hydrocarbon compounds and half hetero (containing sulfur, nitrogen, and oxygen) compounds and requires upgrading using a hydrotreating process. (See Figure 5-2.) Currently, the effects of the hetero compounds on emissions are unknown. Means to control them have not been developed.

In general, the sulfur and nitrogen contained in the oil shale crude varies from two to five percent of the total weight. Present day specifications, Table 5-1, require the sulfur content to be 0.1 percent of the total weight. There are no specifications on the nitrogen content of gasolines. Extensive upgrading by cracking is necessary to yield a product equal to a good petroleum fuel.



Source: Reference 4

FIGURE 5-2. SYNTHETIC CRUDE SHALE OIL PROCESS

Tables 5-5 through 5-8 show the total boiling point distillation range and properties of light, medium and heavy distillates of crude shale oil obtained for the Paraho Indirect retort process stated above.[5] Typically, by weight, the crude contains, for the most part, a heavier distillate fraction which, if used in combination with the light distillate fraction to produce gasoline, requires hydrocracking treatment or a light distillate fraction obtained from other sources since only a small fraction of light distillates are produced from shale oil, (Table 5-5). The average H/C ratio content by weight is approximately 1.5 or 1.8 on the atomic basis, which is similar to gasoline (i.e., approximately 1.7). Total nitrogen by weight, especially in the middle and heavier fractions (See Table 5-8), is higher than gasoline as well as its olefinic content. Oil shale may yield gasolines with relatively low octane numbers. Research on octane boosting additives and techniques to decrease engine octane requirements may be required.

5.1.3.3 <u>Coal Derived Fuels</u> - In addition to shale oils, hydrocarbon type fuels derived from coal are also being considered today as possible replacement for petroleum-based gasolines. Coal is a solid fuel and of vegetable origin. Coal exists in many forms. The form in which coal exists depends upon what geographical changes it has undergone. The percentage of carbon increases and the oxygen content decreases with each succeeding change. The constituents of coal are carbon, hydrogen, oxygen, nitrogen, moisture and ash. Its composition and characteristics also depend upon the impurities that have entered into it during the different transformations. There are several basis which are used to classify coal, such as carbon content, the ratio of carbon to hydrogen content, the ratio of volatile matter to fixed carbon and the rank of the coal. The United States Geological Society uses the term "rank" to designate the present stage of transition in the formation of coal. The rank signifies the completeness of the changes from the first stage, which is peat, to the final

TABLE 5-5. TOTAL BOILING POINT DISTILLATION OF CRUDE (SHALE OIL CRUDE)

Cut	° API	Specific Gravity	Wt %	Vol %	Cumulative Vol %
Gas	–	0.5674	0.10	0.17	0.17
55-165	–	–	0.00	–	0.17
165-220	35.6	0.8468	0.88	0.97	1.14
220-280	38.0	0.8348	0.83	0.93	2.07
280-340	38.1	0.8343	0.97	1.09	3.16
340-380	34.9	0.8504	0.88	0.97	4.13
380-480	31.7	0.8670	7.28	7.87	12.00
480-520	28.9	0.8822	3.95	4.20	16.20
520-600	27.4	0.8905	8.96	9.43	25.63
600-650	23.6	0.9123	5.14	5.28	30.91
650-700	22.2	0.9206	7.09	7.22	38.13
700-750	19.7	0.9358	7.09	7.10	45.23
750-800	19.0	0.9402	10.28	10.25	55.48
800-840	17.7	0.9484	11.83	11.70	67.18
840+ BTMS	11.2	0.9916	34.10-34.72[a]	32.25-32.82[a]	100.00[a]
Loss			0.62	0.57	
Total			100.00-100.00	100.00-100.00	

[a]Corrected for loss.

Source: Reference 5

TABLE 5-6. PROPERTIES OF NAPHTHAS (SHALE-OIL CRUDE)

	165-200	220-280	280-340	340-380
Cumulative Mid Volume %	0.66	1.61	2.62	3.65
Vol % of Crude	0.97	0.93	1.09	0.97
Cumulative vol % of Crude	1.14	2.07	3.16	4.13
Gravity, ° API	35.6	38.0	38.1	34.9
Specific Gravity at 60° F	0.8468	0.8348	0.8343	0.8504
ASTM Distillation, D-86, ° F				
IBP	175	200	180	330
10%	196	235	293	348
30%	206	256	315	352
50%	218	273	322	356
70%	236	288	330	360
90%	270	306	340	370
EP	296	324	364	388
% Recovered	96.0	97.0	97.0	96.0
% Residual	4.0	3.0	3.0	4.0
Total Sulfur, wt %	1.93	1.67	1.45	1.15
Mercaptan Sulfur, ppm	140	10	100	20
Total Nitrogen, wt %	0.10	0.57	1.24	1.62
UOP "K" Factor	10.37	10.80	11.04	10.99
Aniline Point, ° F	<60	<60	<60	<60
Cetane Index	–	–	–	22.0
Smoke Point, mm	–	–	–	19
PNA by NMR, wt %				
Paraffins	9.1	14.0	30.1	24.0
Naphthenes	4.9	13.0	8.9	17.0
Aromatics	72.0	51.0	39.0	33.0
Olefins	14.0	22.0	22.0	26.0

Source: Reference 5

TABLE 5-7. PROPERTIES OF THE MIDDLE DISTILLATES (SHALE-OIL CRUDE)

	380-480	480-520	520-600	600-650
Cumulative Mid Volume %	8.07	14.10	20.92	28.27
Vol % of Crude	7.87	4.20	9.43	5.28
Cumulative Vol % of Crude	12.00	16.20	25.63	30.91
Gravity, °API	31.7	28.9	27.4	23.6
Specific Gravity at 60° F	0.8670	0.8822	0.8905	0.9123
ASTM Distillation, D-86, °F				
IBP	398	462	522	562
10%	412	477	530	589
30%	420	482	540	595
80%	426	487	546	601
70%	436	491	551	605
90%	452	499	564	616
EP	472	517	580	628
% Recovered	99.0	99.0	99.0	99.0
% Residual	1.0	1.0	1.0	1.0
Vanadium, ppm	–	–	<0.01	<0.01
Nickel, ppm	–	–	0.04	0.04
Iron, ppm	–	–	2.7	2.7
Viscosity, cSt at 100° F	–	–	6.07	11.09
Viscosity, SUS at 100° F	–	–	45.8	62.7
Viscosity, cSt at 210° F	–	–	1.67	2.38
Viscosity, SUS at 210° F	–	–	–	34.1
Total Sulfur, wt %	0.96	0.91	0.75	0.78
Mercaptan Sulfur, ppm	60	40	60	30
Total Nitrogen, wt %	1.15	1.46	1.71	1.80
UOP "K" Factor	11.08	11.13	11.25	11.18
Paraffins, wt %	30.4	33.4	36.3	31.2
Naphthenes, wt %	16.6	18.6	19.7	20.8
Aromatics, wt %	34.0	29.0	28.0	31.0
Olefins, wt %	19.0	19.0	16.0	17.0
Aniline Point, °F	<60	<60	<60	60
Pour Point, °F	<–40	–15	10	35
Freezing Point, °F	Too dark	Too dark	Too dark	Too dark
Thermo Viscosity at 60° F	Too dark	Too dark	Too dark	Too dark
Ring Number	Too dark	Too dark	Too dark	Too dark
Cetane Index	28.4	33.3	38.9	38.8
Smoke Point, mm	26	22	–	–
Diesel Index	–	–	–	–
Total Acid No., mg KOH/g	–	–	2.341	1.872
RI at 140° F	–	–	Too dark	Too dark

Source: Reference 5

TABLE 5-8. PROPERTIES OF THE HEAVY DISTILLATES (SHALE-OIL CRUDE)

	650-700	700-750	750-800	800-840
Cumulative Mid Volume %	34.52	41.68	50.36	61.33
Vol % of Crude	7.22	7.10	10.25	11.70
Cumulative Vol % of Crude	38.13	45.23	55.48	67.18
Gravity, °API	22.2	19.7	19.0	17.7
Specific Gravity at 60° F	0.9206	0.9358	0.9402	0.9484
ASTM Distribution D-1160, °F				
IBP	608[a]	561	556	554
10%	640	694	769	809
30%	642	707	782	822
50%	646	716	785	828
70%	650	722	791	835
90%	664	728	799	841
EP	688	734	806	849
% Recovered	99.0	95.0	98.0	98.0
% Residual	1.0	5.0	2.0	2.0
Vanadium, ppm	<0.01	<0.01	<0.01	<0.01
Nickel, ppm	0.03	0.04	0.19	0.24
Iron, ppm	0.71	1.3	9.7	21.0
Viscosity, cSt at 100° F	23.03	41.2[b]	110[c]	360[d]
Viscosity, SUS at 100° F	110.9	192.2	510	1668
Viscosity, cSt at 210° F	3.42	4.88	6.50	7.37
Viscosity, SUS at 210° F	37.6	42.3	47.5	50.3
Total Sulfur, wt %	0.73	0.72	0.64	0.55
Mercaptan Sulfur, ppm	19	36	52	68
Total Nitrogen, wt %	2.00	2.25	2.14	1.89
UOP "K" Factor	11.23	11.28	11.44	11.47
Paraffins, wt %	33.8	26.8	24.1	19.7
Naphthenes, wt %	15.2	16.2	27.9	37.3
Aromatics, wt %	33.0	35.0	32.0	28.0
Olefins, wt %	18.0	22.0	16.0	15.0
Aniline Point, °F	89	91	107	113
Pour Point, °F	45	55	75	95
Cetane Index	41.3	44.2	49.8	51.3
Total Acid No., mg KOH/g	0.710	0.720	1.310	0.731
RI at 140° F	Too dark	Too dark	Too dark	Too dark

[a]D-86 Distillation
[b]Plotted from 140° viscosity of 15.44 cSt
[c]Plotted from 140° viscosity of 28.89 cSt
[d]Plotted from 140° viscosity of 52.41 cSt

Source: Reference 5

stage which is almost pure carbon. The classification according to rank in ascending order is:

a) peat

b) lignite

c) sub-bituminous

d) bituminous

e) semi-bituminous

f) semi-anthracite

g) anthracite.

There are four major regions of deposits of coal in the U.S. for gasolines:

a) High volatile A Bituminous coal from the Eastern region,

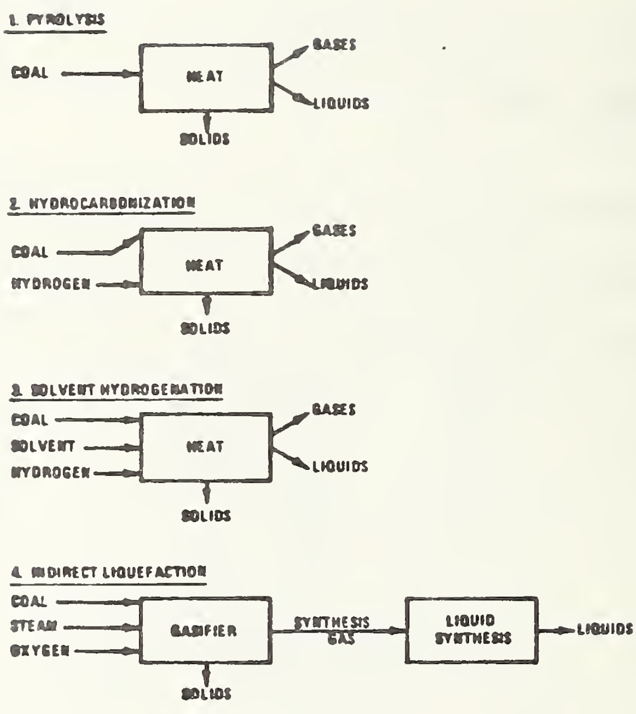
b) High volatile B Bituminous coal from the Eastern interior basin,

c) Subbituminous C coal from the northern Great Plains, and

d) Lignite also from the northern Great Plains.

The high volatile bituminous coals are the favored feedstock for synfuels. The synthetic crudes are obtained from coal liquefaction processes. There are four general classes proposed to produce liquid hydrocarbon (See Figure 5-3) which include: a) pyrolysis, in which heat is the only input required in the transformation of coal to a liquid hydrocarbon, b) hydrocarbonization, in which hydrogen is added during the heating operation, c) solvent hydrogenation which involves the operations of coal/solvent pasting, heating, hydrogenation, solid liquid separation, and hydrogen recycling, and lastly d) indirect liquefaction, wherein the coal is first completely converted to gaseous products which are then catalytically transformed to liquids.[5]

All four classifications of coal liquidification processes will produce syncrudes with significantly different characteristics. Various commercialized processes are shown in Tables 5-9 through 5-11 for the first three classifications. The South



Source: Reference 5

FIGURE 5-3. CLASSIFICATION OF COAL LIQUEFACTION PROCESSES

TABLE 5-9. PYROLYSIS PROCESSES

Pyrolysis Processes	Reactor Type	Method of Heat Addition	Temperatu. °F	Pressure psi	Maximum Size Demonstrated	Current Status
COED	Multiple Fluidized Beds	Hot gas	600-1500	5-10	36 TPD	Dismantled
Toscoai	Rotating drum	Solid-Solid	800-1000 (Ceramic balls)	atm	25 TPD	Inactive
Garrett	Entrained flow	Solid-Solid	1100 air (hot char)	atm	4 TPD	Active
Lurgi-Ruhrgas	Screw-mixer	Solid-Solid	1100 (hot char)	atm	850 TPD	Commercial
Clean Coke (Pyroiysis section)	Fluidized bed 2 stage	Hot gas	800-1300	100	1/4 TPD	100 TPD pilot plant designed

Source: Reference 5

TABLE 5-10. HYDROCARBONIZATION PROCESSES

Hydrocarbonization Processes	Reactor Type	Temperature °F	Pressure psi	Maximum Size Demonstrated	Current Status
COALCON-Hydro-carbonization	Fluidized bed	1040	555	10 TPD	2600 TPD Demonstration Planned
Cities Service Flash Hydropyrolysis	Free fall	1100-1800		2 lb/hr	Laboratory Development
CUNY	Fast fluidized bed	1450	1500		Laboratory Development
Rocketdyne	Entrained flow		1000	1 TPH	Laboratory Development
Schroeder	Catalytic	900			Laboratory Development

Source: Reference 5

TABLE 5-11. SOLVENT HYDROGENATION PROCESS

Solvent Hydrogenation Process	Reactor Type	Catalyst	Solvent	Temperature °F	Pressure psi	Maximum Size Demonstrated	Current Status
Solvent-Refined Coal						50 TPD	
H-Coal	Ebullated bed		Recycle Oil	850	2700	3 TPD	600 TPD Plant being constructed
Synthoil	Fixed bed	Harshow 0402T	Recycle Oil	840	3000	1/2 TPD	10 TPD unit being constructed
Exxon Donor Solvent			Recycle Oil	800	2000	1/2 TPD	200 TPD Plant being designed
Consol Synthetic Fuel (Project Gasoline)	Multi-stage reactor-dissolver		Recycle Oil	800	100-400	20 TPD	Being renovated
Bergius		Iron sulfate sodium sulfide	Recycle oil	800	10,500	1000 TPD	
Pott-Broche			Recycle oil	800	2200		

Source: Reference 5

African company, SASOL, produces synthetic gasoline fuel from the Fischer Tropsch synthesis (indirect liquefaction process), which is the only commerical plant operation to produce liquid hydrocarbons. Presently, these current technologies possess serious environmental problems since large quantities of carbon dioxide and carcinogens are released into the atmosphere and water.

When selecting a process for producing synthetic crude oil from coal, a much more serious situation is confronted than is the case for shale oil. Because of its lower hydrogen-to-carbon ratio, coal cannot be converted to liquid products by pyrolysis as efficiently as oil shale can. The process of converting to liquid or gaseous fuels is basically one of adding hydrogen. The more hydrogen added, the more liquids or gases are produced. An additional complexity is the great variability of coal properties and corresponding variabilities in the response of coals to different processing techniques. Some processes may be very successful with one type of coal feed and either uneconomical or inoperable with a different coal.

Coal syncrudes may be better used in gasoline production because of their high aromatic content. However, the effects of high aromaticity is presently unknown in engines. It may result in combustion chamber wall cooling problems because of the flame's high luminosity. Also, high NO_x emissions may result.

5.1.3.4 Alcohols - Alcohols offer a potential to increase the volume of fuels for spark ignition engines by direct use as an automotive fuel or by blending with gasoline (extenders). Gasohol is an example of the latter (a blend consisting of 90 percent gasoline and 10 percent ethanol). The alcohol blends in gasoline change the fuel's front end volatility characteristics. Alcohols are a partial oxidation product of petroleum. The compounds are saturated with a chain structure of the general formula R·OH. Both methanol and ethanol have been emphasized as alcohols for spark ignition engines. Some properties of ethanol and methanol and their blends are presented in Table 5-12.[6]

TABLE 5-12. ALCOHOL PROPERTIES

SELECTED LIQUID FUEL CHARACTERISTICS

	Gasoline	Ethanol	Methanol	10% Methanol Gasoline Blend	10 % Ethanol Gasoline Blend
SOURCE	Petroleum	Art. Products Petroleum	Nat. Gas, Coal, Wood Municipal Solid Waste	—	—
FORMULA	C_4- C_{12} Mix.	C_2H_5OH	OH_3OH	—	—
STOICHOMETRIC A/F RATIO	4.2-14.8	9.0	6.5	13.8	14.0
FLAMMABILITY LIMITS (% BY VOL IN AIR)	1.4-7.6	4.3-19.0	6.7-10.0	—	—
SPECIFIC GRAVITY	0.70-0.78	0.79	0.79	0.75	0.75
LOWER HEATING VALUE					
BTU LB	18.950 (AVG)	11.500	8.570	17.870	18.160
BTU GAL	115.400 (AVG)	75.670	56.560	109.520	111.400
OCTANE					
RESEARCH	91-105	106-111	106-112	Between Gasoline and Methanol	Between Gasoline and Ethanol
PUMP $\frac{(R + M)}{2}$	37-98	98-102	99-102		
HEAT OF VAPORIZATION					
BTU LB	150	396	306	188	175
BTU GAS	900	2.378	3.340	1.144	1.047
VAPOR PRESS (PSIA @ 100°F)	7.15	2.5	4.6	+3PSI	+1PSI
BOILING POINT (°F)	80-440	173	149	—	—
AUTO EMISSIONS					
UNBURNED HC	Ref. Case	Between Gasoline and MeOH*	Same*	Comparable	Comparable
CO	Ref. Case		Same*	Comparable	Comparable
NOx	Ref. Case		25% Less**	Comparable	Comp rable
FUEL ECONOMY (Equal BTU) STOICHIOMETRIC A/F RATIO	Ref. Case	+ 10 to 30%	+10 to 30%	Comparable (Vol. Basis)	Comparable (Vol. Basis)
COST (PLANT GATE)					
PER GAL.	\$0.55	\$1.25	\$0.50	\$0.55-0.60**	\$0.62-0.69**
PER CAL. GAS EQUIV.	\$0.55	\$1.90	\$1.02	\$0.57	\$0.64
PER MM BTU	\$4.75	\$6.45	\$6.85	\$5.16	\$5.92

*Will burn leaner than gasoline and thereby reduces an emission further.
**Lower figure is 10% volume blend in gasoline, higher figure in energy compared to a gallon of gasoline.

Source: Reference 6

Alcohols are characterized by their good antiknock quality as shown by their higher research octane rating, lower heating value, higher latent heat, lower A/F ratio, and lower boiling temperature revelant to gasoline. Unlike gasoline, the volatility of neat alcohol is characterized by a single distillation temperature. Alcohols can be manufactured from renewable and/or preferably indigenous sources. For example, ethanol can be manufactured from sugar cane, sugar beets, corn, straw, etc., and methanol can be manufactured from coal via gasification process, natural gas, biomass, etc. Presently, large scale proven technology exists to produce methanol and ethanol fuels. World-wide, there are 42 methanol plants from which methanol is derived mainly via synthesis processes. Also, there are 16 ethanol plants. Ethanol here is derived by seasonal fermentation. Estimates of cost of alcohol fuels vary and depend on feedstock and plant capacity. The displacement value* of methanol based on a gasoline priced at $1.00/gal at the pump is 42¢/gal (gasoline equivalent). To date, an SRI[7] study concluded that it costs 64¢/gal to 90¢/gal to produce coal-to-methanol fuel. Presently, the cost of ethanol is two to five times per gallon greater than methanol.

At a recent conference in Asilamer, California,[8] it was concluded that, in general, alcohols will provide no short-term impact as a transportation fuel. However, as a medium-term alternative, methanol derived from coal and/or natural gas is expected to provide an impact. The influence of ethanol was considered too small to make a major impact on transportation. The extent of use of alcohols in the future is unknown at this time. Quotations between 10 and 15 percent have been mentioned, but it is not presently known whether they will appear as extenders or octane boosters in gasolines or be burned directly in engines.

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*Price required to produce methanol at the equivalent gasoline pump price.

The potential use of alcohols in engines is unknown, largely because most studies have been conducted on engines not optimized for its use. Some fuel economy comparisons of present day vehicles operated on alcohol blends with vehicles operated on pure gasoline show that the alcohol blends produce lower fuel economies (Table 5-13). The fuel economy generally increases with an increase in percent alcohol blended with gasoline. Note that these fuel economy differences show a positive gain when comparisons are made on a gasoline equivalent energy basis.

TABLE 5-13. ALCOHOL BLEND FUEL-ECONOMY COMPARISONS

ALCOHOL BLEND %	FUEL ECONOMY (% Difference)
10% ethanol	-2%
16% ethanol	-1%
10% methanol	-5%
50% methanol	-3%

Some limited data show that engines designed with high compression ratios can provide a fuel economy increase of 7.5 percent when fueled with neat methanol and 5.0 percent when fueled with ethanol over engines designed with low compression ratios.

Present information, in general, supports that E-20 or M-15 (gasoline plus alcohol percentage) fuel blends can be adapted to gasoline engines without a big change. However, the durability and, in particular, engine life must be determined because of the corrosive action of alcohols on engine parts, for example, the fuel pump and carburetor since these engine parts are presently lined with zinc and aluminum. There are indications that corrosion occurs at a later age with 20 percent ethanol blends. Heavy corrosion occurs, on the other hand, when the engine is run on neat ethanol.

Vapor lock as well as higher evaporative emissions have been cited for vehicles served by ethanol/gasoline blends because of the higher Reid vapor pressure. Methanol powered vehicles also appear not to be without their problems. They include: 1) phase separation under cold weather starting, 2) driveability, due to high engine speed pre-ignition. Manifold heating and the use of finer atomization of the fuel or by adding isopentane to methanol are current technologies available to eliminate partially the adverse effects of cold starting. MBTE (methyl-tertiary butyl ether), made from methanol and isobutylene, added to alcohols was found to provide the same energy density as gasolines and to eliminate the adverse effect of vapor pressure and distillation properties. Methyl-fuel, which contains higher alcohols, is also a useful addition to gasolines.

In general, the solution to the problems stated above does not present any major barriers. However, since an optimized engine/fuel system does not exist either in performance or fuel economy, future development studies on engines which can operate on both gasoline and alcohol with good fuel economy and performance are required.

There are environmental concerns: Aldehydes, attributed to formaldehydes found in the exhaust, are higher when pure alcohols or alcohol blends are used. The current emission aftertreatment systems (oxidation catalyst) cannot be used successfully to eliminate it. Small amounts of amilines (fuel bounded nitrogen) can suppress aldehydes, but lead to increased oxides of nitrogen. Vehicles powered by engines fueled with alcohols also emit higher HC emissions. The present FID hydrocarbon measurement techniques do not respond to alcohols, and currently, non-regulated emission of unburned methanol and ethanol may become critical. Finally, recent studies[9] on methanol/gasoline blends, in particular an M-15 blend applied to human skin, have concluded that the blend is dangerous and presents an occupational hazard.

Alcohols offer the potential of ultralean burn, because of their extended lean misfire limit. Presently, the degree to which the lean limit can be extended is not known. It is of interest in order to minimize oxides of nitrogen emission. In addition, studies are required on surface ignition and its control in order to take advantage of the high octane of methanol by increasing compression ratio. Further studies are required to overcome the cold start problems with alcohol.

5.2 ENGINE LUBRICANTS

5.2.1 General

In a spark ignition engine, most of the friction loss which adversely affects fuel economy is caused by:

1. hydraulic shear of the oil in the distribution passages, shafts and sleeves, clearances and other moving parts,
2. rubbing contact caused by the motion of pistons/piston rings and valve trains.
3. starting and low temperature operation of an engine, and
4. blowby and throttling.

The lubrication characteristics of oil is a parameter which plays an important role in engine friction.

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 improvers, friction modifiers (oil soluble chemicals or solid lubricants), dispersants, detergents, and oxidation inhibitors. In the past, oils were primarily formulated to meet the temperature, wear, and oxidation problems encountered in engine operation. Recent improvements for fuel efficiency in engine oils have been in the following areas:

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

In all cases, lubricant efficiency comes about through improved viscosity characteristics or friction modifications. The effect of a combination of friction modifiers and reduced viscosity appears to be additive since they act via different mechanisms. Friction modifiers function when boundary layer conditions are present, while the lower viscosity results in reduced viscous drag in engines, especially under low temperature conditions. Viscosity improvement can be obtained with viscosity improver additives or by the use of synthetic base stocks. The synthetics exhibit reduced low temperature viscosities without adverse effects on other lubricant properties. 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.

Table 5-14 summarizes the results reported in the literature[10,11,12] over the FTP cycle with fuel efficient lubricants.

TABLE 5-14. FTP IMPROVEMENTS WITH FUEL EFFICIENT LUBRICANTS

Oil Type Improved Conventional Oil	Avg. % Improvement
- Friction Modified	5
- Low Viscosity	2
Synthetics:	3
Colloidal Suspension	4-5 (est.)

Source: References 10, 11, 12

5.2.2 Upgraded Conventional Oils

The upgrading of a conventional mineral oil through viscosity improvers and/or soluble friction modifiers has been the approach taken by Exxon Corporation to develop its new Uniflo engine oil. As part of its testing, six 1975-76 model year cars were tested over the FTP. The average results for these tests showed a fuel economy improvement of 3.9 percent on the urban cycle and 9.0 percent on the highway cycle, yielding a composite average fuel economy improvement of 5.5 percent.[12] The range in the composite results were from 1.5 to 12.0 percent. GM reported on results with lower viscosity mineral oils in which it compared 10W engine oil to a 10W-30 along with a 75W vs. 90W rear axle lubricant. GM stated that two-thirds of the improvements measured were due to the engine oil. On the FTP with a compact car, a 1.4 percent improvement was measured on the urban cycle, with a 3.0 percent improvement in the highway cycle, yielding a composite 2.3 percent improvement. In the urban part of the cycle, the largest improvement, 3 percent, was measured during the cold-start portion, indicating the importance of viscosity on cold-start fuel economy.[10] Further tests by GM on a pick-up truck compared 5W-30 to 10W-30 engine oil and 75W to 90W rear axle lubricant.[10] Similar results were obtained giving an average composite fuel economy improvement of 1.7 percent. GM is also testing friction modified oils, and results to date indicate improvements in the 1.5 to 4 percent range.[10]

5.2.3 Synthetic Oils

As with lower viscosity oils, the major improvements with pure synthetics are in the cold-start portion of the driving cycle. Mobil Oil Company tested its pure synthetic, Mobile 1, with 13 vehicles on the FTP.[10] The synthetic is a 5W-20, and it was compared to a 10W-40. An average 4 percent improvement was measured on the urban part of the cycle and a 2 percent average improvement on the highway cycle, yielding a 3.1 percent composite fuel economy. The measured improvements in the urban cycle ranged from

-1 to 9 percent. In the highway cycle, the improvements ranged from -2 to 6 percent. Mobil also compared this oil to the 10W-40 oil in 10 foreign cars. The EPA urban cycle cold start gave an average improvement of 6 percent, ranging from 2 to 11 percent. The urban cycle hot start gave an average 4 percent, ranging from 1 to 10 percent. Many other tests are reported in the literature on synthetic oils with various dynamometer and fleet tests. These results appear to substantiate the fact that synthetics can yield a 1 to 4 percent improvement in fuel economy. Synthetics also have better high temperature properties to resist oxidation and have the potential for extended drain intervals.

5.2.4 Colloidal Suspensions in Mineral Oil

The use of colloidal graphite as a friction modifier is the approach taken by Arco for its Arcographite. Extensive testing was performed by Arco, but none of this testing was according to the FTP.[10] However, Arco's data from dynamometer and fleet testing indicates that results with Arcographite should be comparable to that obtained with new friction modified Exxon Uniflo, that is, in the 4 to 5 percent improvement range.

5.2.5 CRC Study

Most recently, the Lubricants Group of the Coordinating Research Council (CRC) conducted a survey dealing with the influence of lubricants on light duty vehicle fuel economy (Table 5-15).[13] In the CRC study all fuel economy gains were relative to the fuel economies obtained with 1977 conventional lubricants. The higher gains expected in the 1982-1985 time frame take into account that the more efficient low viscosity lubricants and friction modifiers will prevail and that technological developments will improve the fuel efficient characteristics of the lubricants and their additives.

TABLE 5-15. FUEL ECONOMY GAINS FOR ENGINE OILS

	Presently Reported (%)	Estimated 1980 (%)	Estimated 1982-1985 (%)
EPA city/highway cycle	0-5.5	1-5.2	2-3
Field Service*	1-6.5	3-3.5	4-5

*The higher gains are partially attributed to a) difference in driving patterns, b) ambient temperatures.

Source: Reference 13

REFERENCES FOR SECTION 5

1. Federal Register, Vol. 40. No. 126, June 30, 1975, "Motor Vehicles," pg. 27448.
2. SAE Information Report 1978, Automotive Gasoline, SAE J 312d.
3. SAE Information Report, 1978, Automotive Gasoline, SAE J 312c.
4. Bidwell, J.B. et al., "Automotive Fuels - Outlooks for the Future," GMR-2733.
5. Department of Energy, "Identification of Probable Automotive Fuels Composition: 1985-2000." Executive Summary, HCP/W36 84-01/2, May 1978.
6. Table provided to participants at the Third International Symposium on Alcohol Fuels Technology, Asilmar, CA, May 28-31, 1979.
7. Jones, J.L. et al., "A Comparative Economic Analysis of Alcohol Fuels Production Options," presented at the Third International Symposium on Alcohol Fuels Technology, Vol. II, Asilmar, CA, May 28-31, 1979.
8. Third International Symposium on Alcohol Fuels Technology, Asilmar, CA, May 28-31, 1979..
9. Ferry, D. "Toxicity of Methane/Petrol Mixtures," presented at the Third International Symposium on Alcohol Fuels Technology, Vol. III, Asilmar, CA, May 28-31, 1979.
10. Harriott, W. "Report on Vehicle-In-Use," U.S. Department of Transportation, Transportation Systems Center, available upon request from R. Walter.
11. Southcoast Technology Center, Inc., "Augmentation of Research and Analysis for Timely Support of Automobile Fuel Economy Studies," Draft Final Report, March 29, 1978.
12. Department of Transportation/National Highway Traffic Safety Administration, Public Hearing on Fuel Economy in Reference to 1980-81 Standards, January 16, 1978.

REFERENCES SECTION 5 (CONT'D)

13. "Study of Lubricant Influence on Light-Duty Vehicle Fuel Economy," CRC Report No. 502, December 1978.

6. NOISE

Presently, there are no federal noise standards in effect for passenger cars and light trucks. Recently, the EPA has published in the Federal Register a draft test procedure to measure noise from light vehicles under urban driving.[1] This procedure consists of simulating a community driving situation which entails part-throttle, accelerations under drive away conditions for an acceleration of 0.15 g's up to steady state speed of 25 mph. Noise level measurements are made at a distance of 7.2 meters from the moving vehicle.

There are currently two test procedures used to measure exterior noise levels from vehicles.

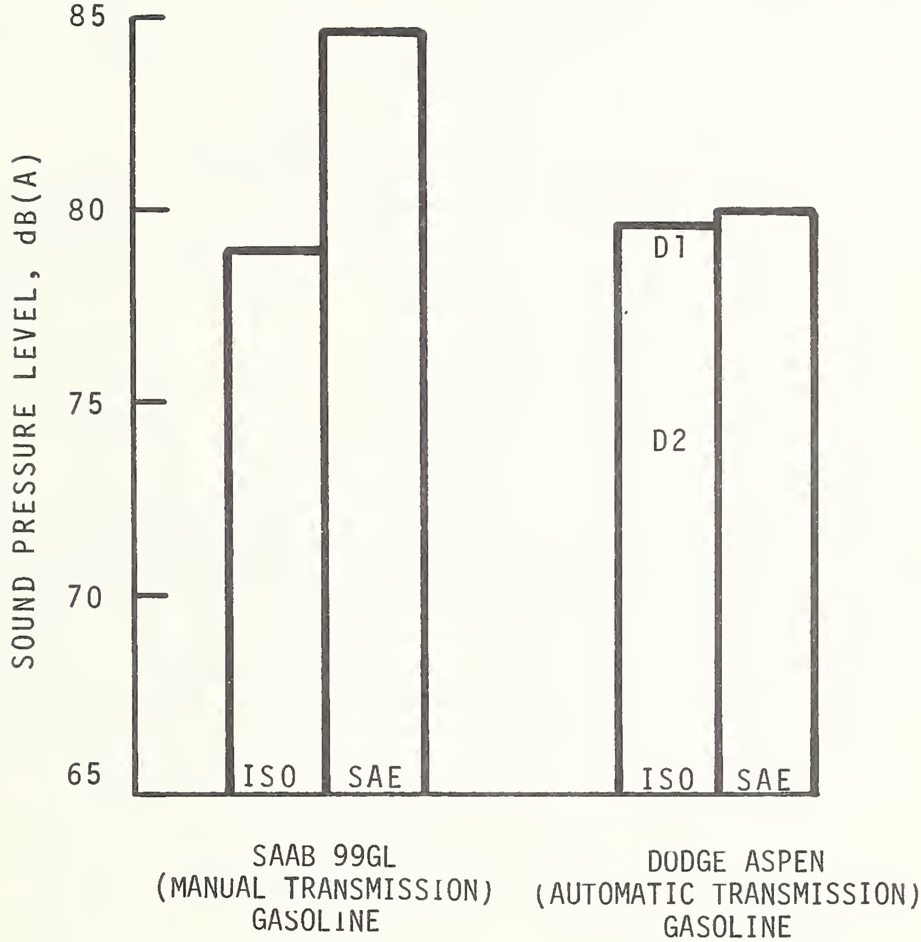
The first is the SAE J986b vehicle sound level measurement procedure which is a U.S. industry accepted procedure. The second is the ISO recommendation R362 which is the basis for noise measurements in European countries. The SAE J986b procedure is designed to reproduce the noise level of vehicles operated under high speed, maximum loaded engine conditions. Under this test procedure, the vehicle is driven over a test section of 15 meters. The vehicle enters the test section at 30 mph and accelerates under a wide open throttle condition. The highest numerical ratio gear is used such that the vehicle reaches rated engine speed at or beyond the end of the test zone in a region called the end zone which extends 30.5 meters beyond the test zone. Sound pressure level measurements are made 15 meters from the test track perpendicular to its centerline.

The ISO R362 measurement procedure is somewhat similar to the SAE J386b measurement procedure. Here, the vehicle is accelerated at wide open throttle over a test section of 20 meters at a vehicle entrance speed of 31 mph, when the engine speed is equal to or below 3/4 of its rated speeds. For engine speeds which exceed the 3/4 value when the vehicle enters at 31 mph, the 3/4 rated speed is used and the vehicle enters at a speed less than

31 mph. Vehicles with four-speed manuals are driven always in second gear, and vehicles with five speed manuals are driven in third gear. In the case of automatics, the highest gear is used, provided there is no kick down to a lower gear as the vehicle traverses over the test track. Sound pressure level measurements are acquired at 7.5 meters perpendicular and away from the centerline of the test track. The most accepted procedure to measure noise radiated from the engine itself is to use SAE J1074 recommended practices.

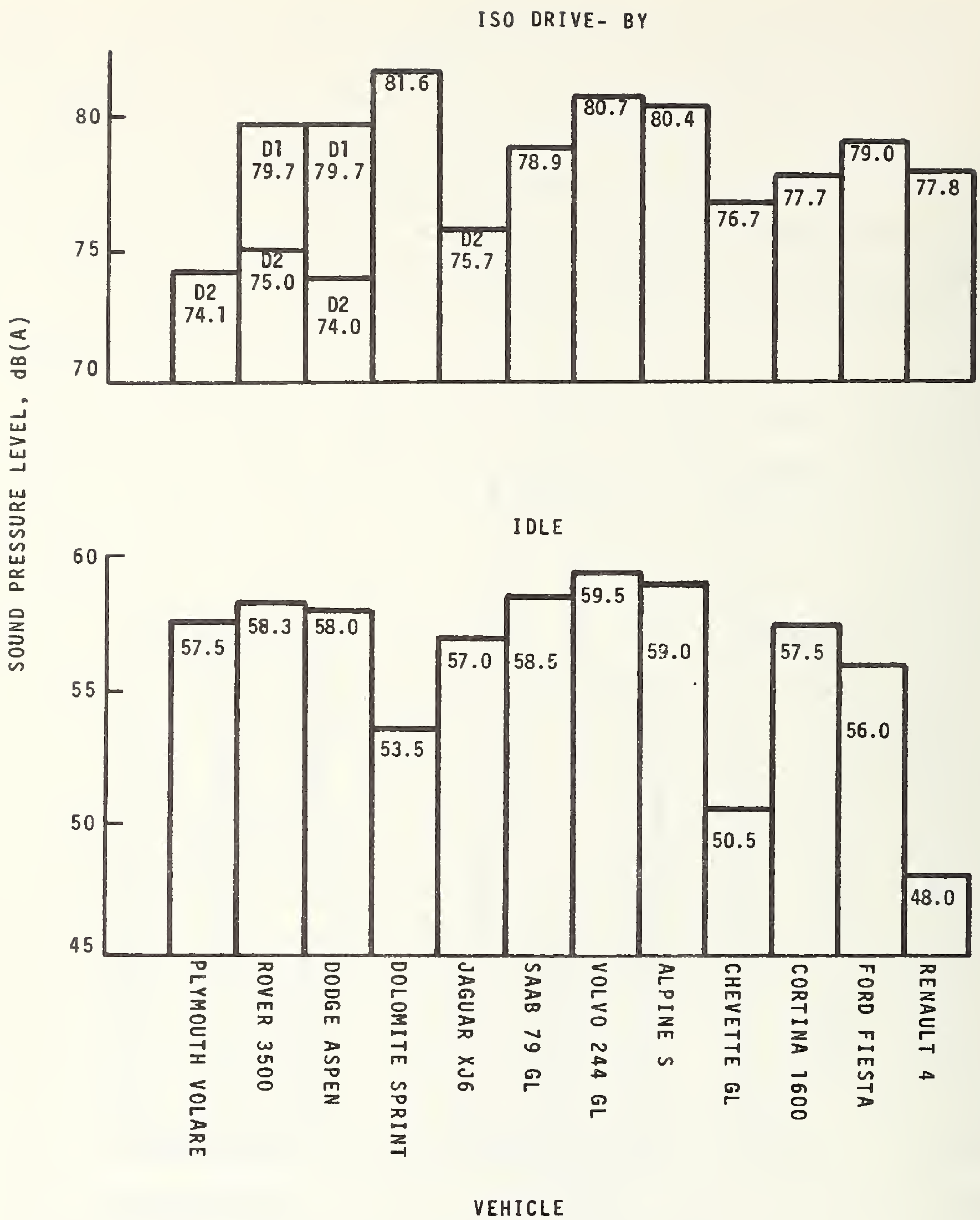
Figure 6-1 shows the noise level of two spark ignition vehicles, one with a manual transmission and the other with an automatic transmisison, for two measurement procedures.[2] A corrected factor of 6 dB(A) was used to project the noise levels measured under the SAE J986b to those at the location used for measurement of the noise level under the ISO R362 vehicle noise measurement procedure. These data show that there is no reliable correlation between the two test procedures. This lack of correlation is primarily due to gearing effects and the difference in engine speed conditions independent of gearing or vehicle characteristics. For example, the SAE test for the SAAB 99 GL requires first gear to be used which results in rated engine speed to be reached before the end zone. In the ISO R362 test, the second gear is always used for vehicles with a four speed transmission which results in lower engine speeds. Because of the gearing factor, noise level measurement acquired under the ISO R362 procedure, generally produces lower levels for those cars with high second gear ratio, in particular for automatic transmission ratios. In some isolated cases, however, where high gearing is accomplished by very high power/weight ratio, the gearing offset can be outweighed by the acceleration performance; then, high ISO R362 levels can be produced.

Spark ignition powered vehicles produce exterior and interior noise levels approximately 20 dB(A) lower during idling than during pass-by. See Figures 6-2 and 6-3 where typical exterior and interior noise levels are shown for selected European type vehicles.[2]



Source: Reference 2

FIGURE 6-1. COMPARISON OF SAE J986b AND ISO R362

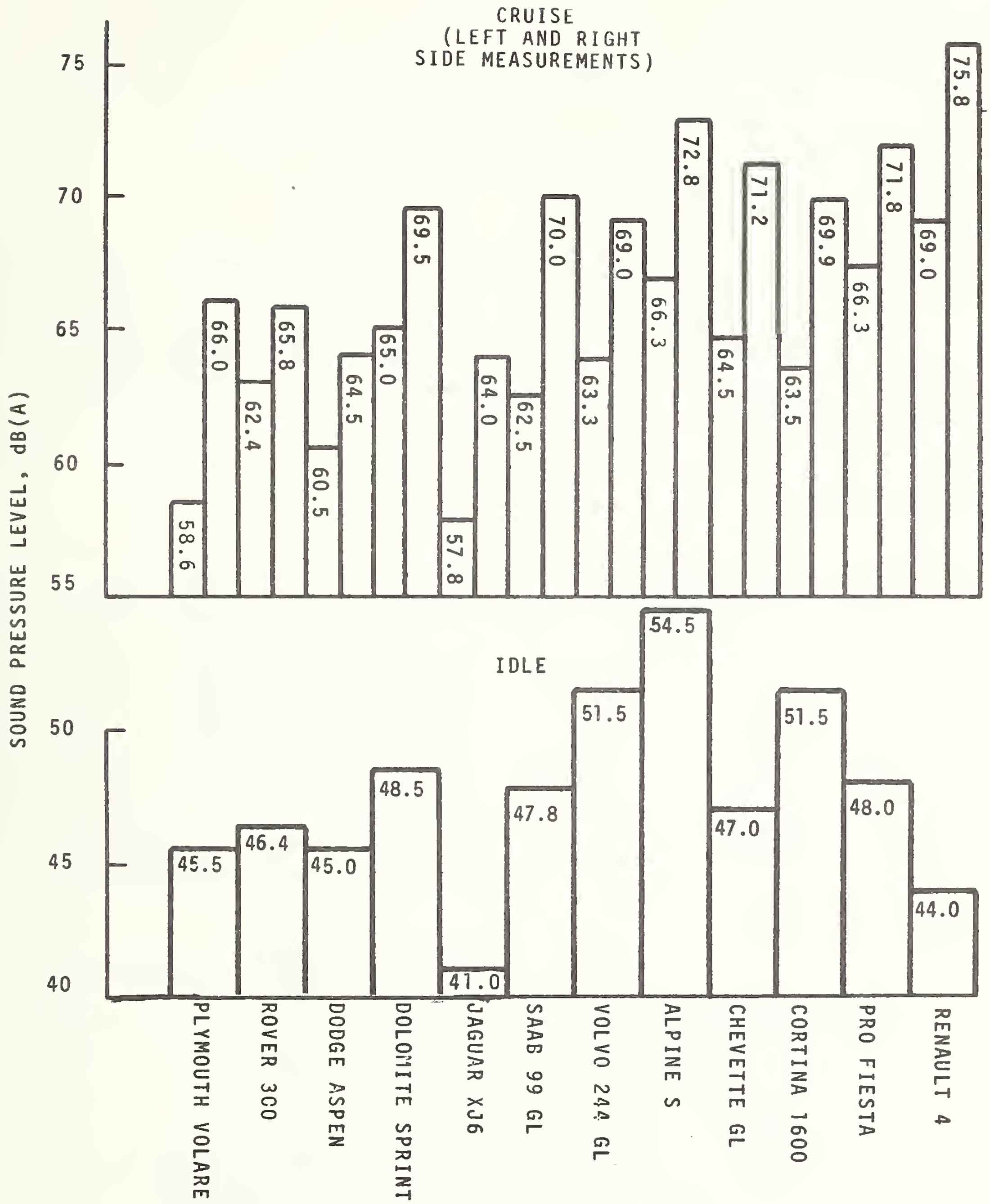


D_1 Denotes low gear

D_2 Denotes next to lowest gear.

Source: Reference 2

FIGURE 6-2. EXTERIOR VEHICLE NOISE LEVEL FOR SELECTED EUROPEAN VEHICLES POWERED BY GASOLINE ENGINES



Source: Reference 2

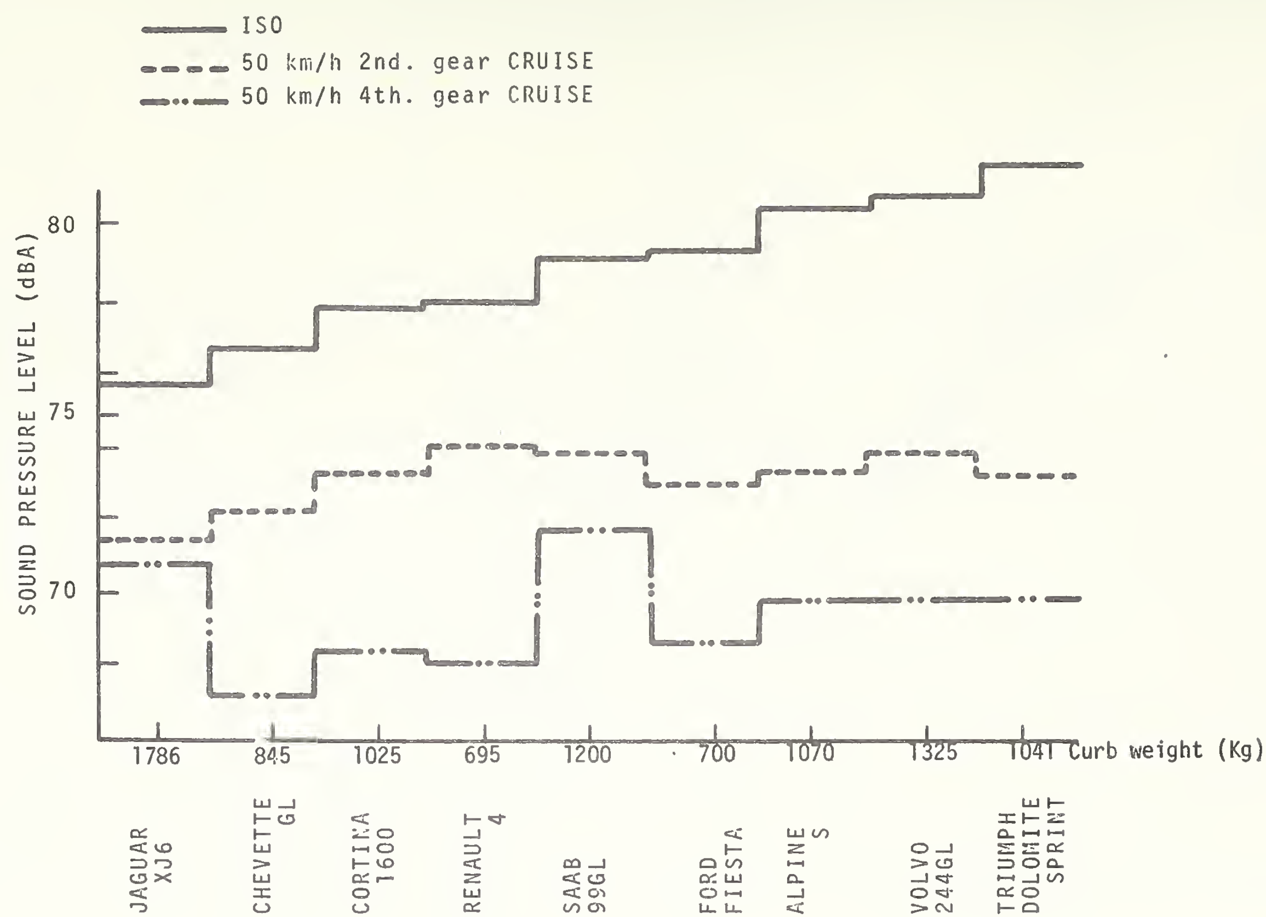
FIGURE 6-3. INTERIOR NOISE LEVELS OF SELECTED EUROPEAN PASSENGER CARS POWERED BY GASOLINE ENGINES

Based on ISO drive-by noise level measurements, the overall average noise level was 77.5 dB(A) with a standard deviation of ± 2.7. The average interior idle and cruise noise levels were 47.6 dB(A) and 63.5 dB(A) with standard deviations of 3.6 dB(A) and 3.4 dB(A), respectively. The average exterior idle noise level was 56.1 dB(A) with standard deviation of a 3.6 dB(A), respectively. Typical cruise (steady state) noise levels are displayed in Figure 6-4 for two gears, second and fourth.

Generally, the drive-by noise levels associated with the larger, low rated speed gasoline engines typical of the current full size U.S. cars, are, on an average, approximately 2.5 dB(A) less than European passenger cars with "high speed" engine (on the basis of ISO R362 tests). Part of this is explained by the high gearing associated with universally used automatic transmissions.

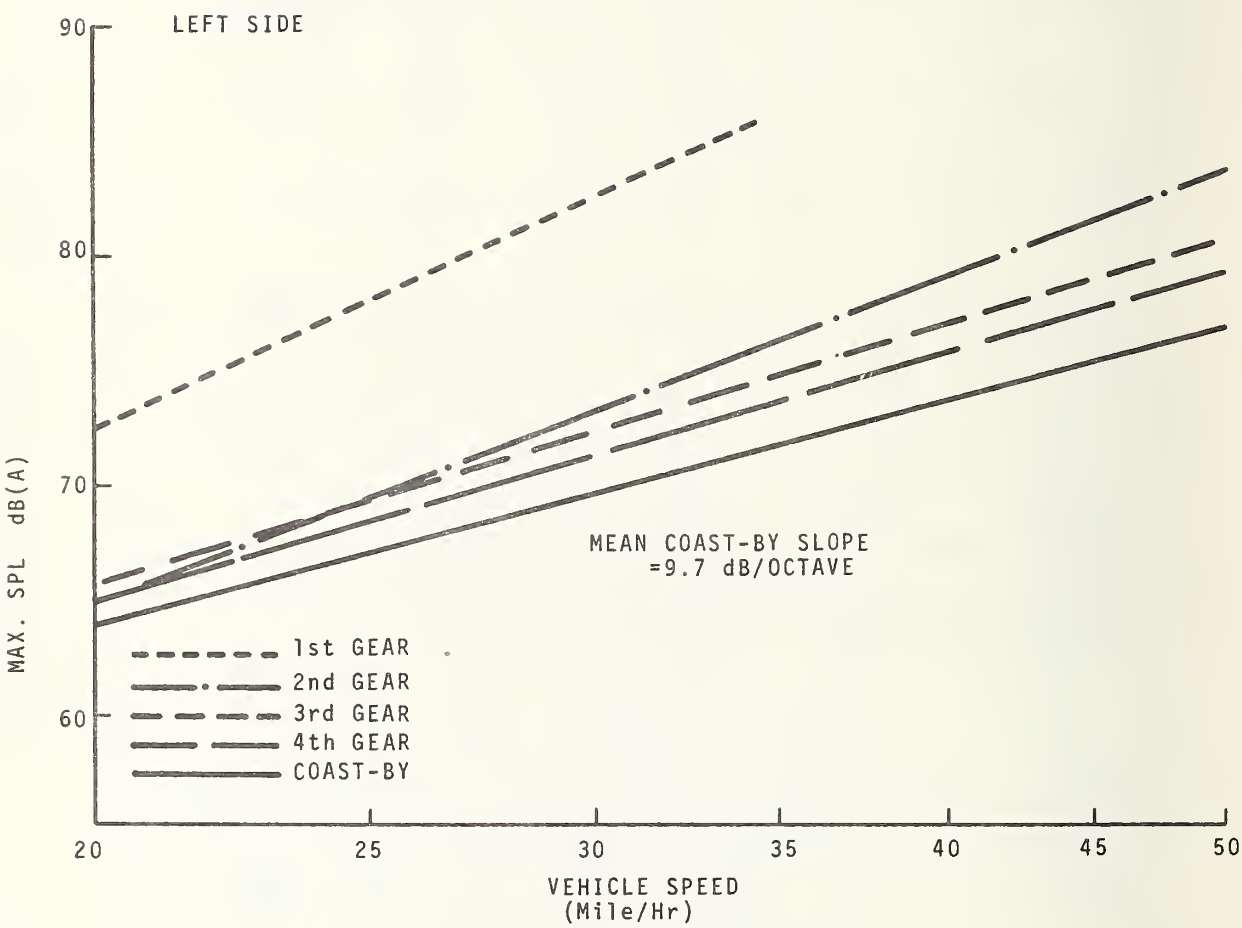
Ricardo also performed a number of noise tests on the Saab 99GL vehicle to characterize the acoustic signature under selected controlled drivings.[3] These included both steady state tests and accelerating the vehicle at a constant rate over the test track. Items which were examined included the effect of driving gear, and horsepower-to-weight ratio. Vehicle noise directivity patterns were evaluated as well as identification of major vehicle noise sources. The following conclusions were drawn:

a) The gear ratio tests concluded that the Saab exhibited a large sensitivity to load. See Figures 6-5 and 6-6. In third and top gears the overall noise is dominated by rolling noise and in second gear vehicle noise was roughly of the same order as rolling noise, the overall SPL measured in first gear was significantly higher on the order of 10-15 dB(A) above rolling noise at high



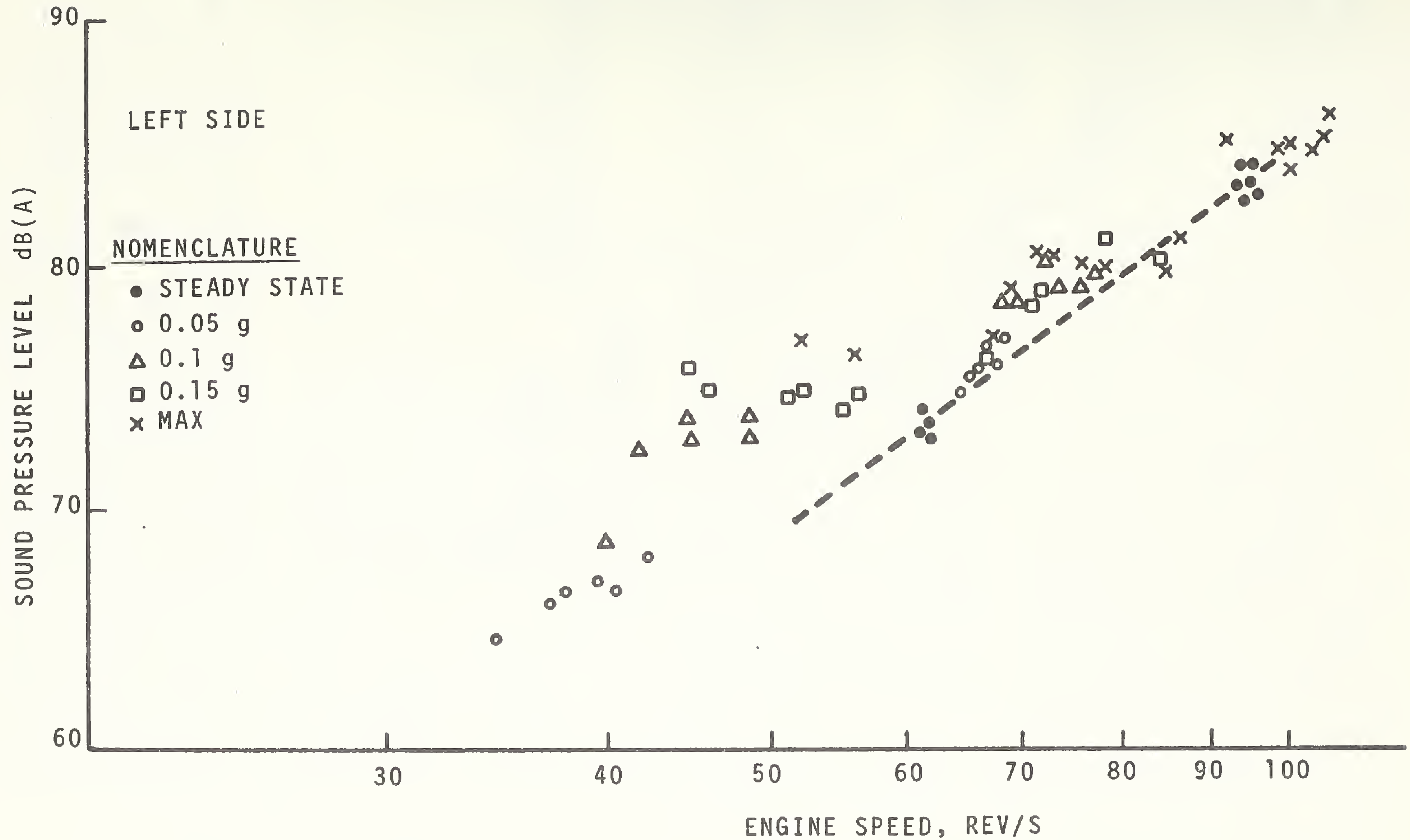
Source: Reference 2

FIGURE 6-4. NOISE LEVELS AT 7.5m FOR VARIOUS DRIVE-BY CONDITIONS (AVERAGE OF LEFT AND RIGHT HAND SIDES)



Source: Reference 3

FIGURE 6-5. EFFECT OF VEHICLE SPEED AND GEAR RATIO ON MAXIMUM DRIVE-BY SOUND PRESSURE LEVEL (SPL) FOR STEADY STATE TESTS (INCLUDING COAST-BY)



Source: Reference 3

FIGURE 6-6. SOUND PRESSURE LEVEL VERSUS ENGINE SPEED AT VARIOUS ACCELERATION RATES (VEHICLE OPERATED UNDER FIRST GEAR) engine speeds. Noise data for first gear also showed a considerable effect of load on Sound Pressure Level at the lower end of the engine speed range which was largely attributed to the effect of load on speed. In addition, the Saab vehicle exhaust system was dominated by resonances (Figure 6-7) at lower engine speeds.

b) The horsepower-to-weight ratio tests concluded that vehicle noise increases for a given engine speed when engine load is increased by decreasing the HP/WT ratio of the vehicle.

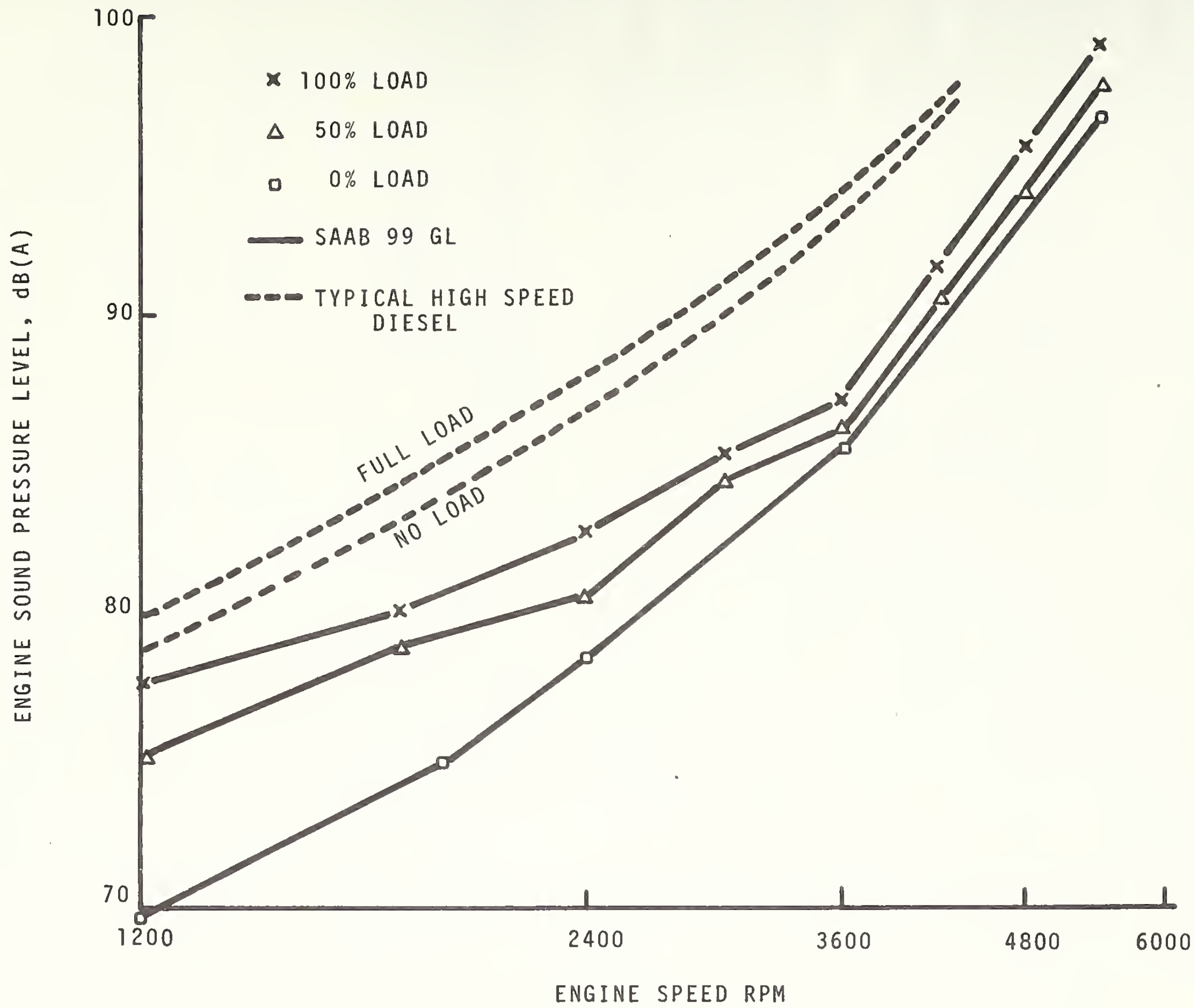
c) Measured noise directivities showed increased noise caused by the exhaust system. Other than this, the noise field is symmetric for this vehicle.

d) The major noise sources were identified as those due to the contribution of the engine exhaust and tires evaluated for two vehicular driving modes of a) steady state and b) under maximum acceleration with the transmission engaged in second gear under an entrance speed of 30 miles/hr.

		Driving Conditions			
Vehicle	Noise Source	Steady State % Overall SPL LHS*	RHS*	Maximum Acceleration (J986a Test Procedures) % Overall SPL LHS	RHS
Saab 99E	Engine	38	49	33	56
	Exhaust	30	13	56	33
	Tires	32	38	11	11

*Note: Left and Right-hand side.

e) Variation of engine noise levels with speed and load showed that the Saab engine levels were generally about 8 dB(A)'s lower than that of a typical high speed indirect injection diesel engine. Unlike the diesel, the variations in noise levels are higher over load, and are much greater in the lower speed range, generally on the



Source: Reference 3

FIGURE 6-7. SOUND PRESSURE LEVEL VS. ENGINE SPEED order of 10 dB(A). The increased noise of the gasoline engine at higher engine speeds, characterized by the lower variability due to load, is caused by piston slap. This is highlighted by the higher slope of engine noise/decade RPM.

Tables 6-1[4] and 6-2[5] show summaries for six noise control measures as a function of percent change in fuel economy, vehicle weight and cost. The emission levels were referenced to a Vehicle Emission Standard of 0.41/3.4/1.0 gms/mile $HC/CO/NO_x$. Noise levels were referenced to the wide-open throttle, maximum acceleration drive-by SAE J986b vehicle sound level measurement procedure. A baseline vehicle of 2500 lb inertia weight was maintained throughout the study. The noise control measures included: 1) reduced engine speed by 10 percent, 2) engine size and configuration, 3) combustion processes, 4) engine structures, 5) gas flow and 6) others. The implications of the first three noise control measures are shown for the spark ignition vehicle in Table 6-1. The later three measures are considered independent of engine type, thus, single entry data is presented which is applicable to both engines.

Redesign of major engine structural components is the single most important noise control measure. It bears no relationship to vehicle weight or fuel economy; however, it contributes to a cost increase of approximately 5 percent.

TABLE 6-1. SPARK IGNITION NOISE REDUCTION MEASURES

Noise Reduction Measure	Noise Reduction (dBA)	Vehicle Weight Change (%)	Fuel Economy Change (mile/US gal)	Cost Increase (%)
Reduce engine speed by 10%				
a) by changing final drive ratio	1 to 2	0	<+1	0
b) by changing intermediate ratio	1 to 2	0	<+1	0
c) raise final drive ratio and increase engine swept volume	0	+1	~0	<+1
d) raise final drive ratio and turbocharge	1	+2 to +3	<+1	~+7
Engine size/configuration change				
a) Reduce bore/stroke ratio	1 to 2	<+1	<+1	0
b) Reduce bore and stroke - increase cylinders	1 to 2	+2	-1	+7
c) Reduce swept volume and turbocharge	1 to 2	0 to -1	+2	~+5
Combustion process changes				
a) retard ignition at high speed	0 to 1	0	0	0
b) increase compression ratio (c.r.) to 9.5	(<1 increase)	0	+1 to +2	0
c) increase c.r. to 12	(<1 increase)	0	+2	<+1
d) increase c.r. to 12, reduce swept volume	~1	1	+2-1/2	<+1
e) increase spark energy	0	0	0 or <+1	<+1
f) increase number of spark plugs to 2	0	0	<+1	<+1
g) turbocharge	(increase by 1 to 2)	+2 to +3	-1 to -2	~+7
h) EGR (10% at WOT)	<1	<+1	0	<+1
i) A/F ratio (less at high speed,	<1	0	0	0

Source: Reference 4

TABLE 6-2. DIESEL AND SPARK IGNITION NOISE CONTROL MEASURES (COMMON TO BOTH ENGINE TYPES)

Noise Reduction Measure	Noise Reduction 9dBA)	Vehicle Weight Change (%)	Fuel Economy Change (mile/US gal)	Cost Increase (%)
Engine Structures				
Damping/Mass Loading components	0 to 2	<+1	0	<+1
Isolation of engine components	0 to 2	<+1	0	<+1
Comprehensive shielding	1 to 3	+1 to +2	~ -1/2	+1 to +2
Enclosure (tunnel type)	4 to 6	+1 to +3	~ -1/2	+2 to +3
Major engine structure design	4 to 8	0	0	~ +5
Gas Flow				
Improved Exhaust Muffler	1-2	+1 to +2	-1 to -2	+1 to +2
Improved Intake Muffler	0.1	~ +1	~ -1	~ +1
Other				
re-design piston (close clearance)	1 to 2	0	0	0 to 1

Source: Reference 5

REFERENCES FOR SECTION 6

1. Federal Register, Vol. 44, No. 99, May 21, 1979, "Emission Standard for Suitable Transportation Equipment for Light Vehicles."
2. DOT-TSC-1242 - Special Report, "Noise Tests on a Selection of European Passenger Cars," Ricardo Report No. DP79/685.
3. DOT-TSC-1242 - Special Report, "Summary Report Covering Test Work on the Saab 99B and Peugot 504D Vehicles", Ricardo Report No. DP77/1158.
4. DOT-TSC-1242 - Special Report "Tests on a Saab B.1.2 Liter Gasoline Engine," Ricardo Report No. DP79/820.
5. DOT-TSC-1242 - Special Report, "Passenger Car Noise Control Measures and Their Effects on Fuel Economy, Cost, and Vehicle Weight," Ricardo Report No. DP79/594.

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