HE
18.5
.A34
no.
DOT-
TSC-
NHTSA-
79-56

TAD 491

ıRT NO. DOT-TSC-NHTSA-79-56

DOT-HS-805 238

POTENTIAL OF SPARK IGNITION ENGINE, ENGINE DESIGN CONCEPTS

DEPARTMENT OF
TRANSPORTATION
JUL 22 1980
LIBRARY

T. Trella
T. Shen
M. Koplow

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

MARCH 1980
FINAL REPORT

DOCUMENT IS AVAILABLE TO THE PUBLIC
THROUGH THE NATIONAL TECHNICAL
INFORMATION SERVICE, SPRINGFIELD,
VIRGINIA 22161

for

U.S. DEPARTMENT OF TRANSPORTATION
National Highway Traffic Safety Adminsitration
Office of Research and Development
Washington DC 20590

NOTICE

This document is disseminated under the sponsorship of the Department of Transportation in the interest of information exchange. The United States Government assumes no liability for its contents or use thereof.

NOTICE

The United States Government does not endorse products or manufacturers. Trade or manufacturer's names appear herein solely because they are considered essential to the object of this report.

NOTICE

The views and conclusions contained in the document are those of the author(s) and should not be interpreted as necessarily representing the official policies or opinions, either expressed or implied, of the Department of Transportation.

18.5
.A34
no.
DOT-TSC-NHTSA-
79-56

Technical Report Documentation Page

1. Report No. DOT-HS-805 238	2. Government Accession No.	3. Recipient's Catalog No.
4. Title and Subtitle POTENTIAL OF SPARK IGNITION ENGINE, ENGINE DESIGN CONCEPTS		5. Report Date March 1980
		6. Performing Organization Code
7. Author(s) T. Trella, T. Shen, M. Koplow		8. Performing Organization Report No. DOT-TSC-NHTSA-79-56
9. Performing Organization Name and Address U.S. Department of Transportation Research and Special Programs Administration Transportation Systems Center Cambridge MA 02142		10. Work Unit No. (TRAIS) HS027/R0404
		11. Contract or Grant No.
12. Sponsoring Agency Name and Address U.S. Department of Transportation National Highway Traffic Safety Administration Office of Research & Development Washington DC 20590		13. Type of Report and Period Covered FINAL REPORT July 1978 - October 1978
		14. Sponsoring Agency Code
15. Supplementary Notes		

16. Abstract

This report provides a review and assessment of potential improvements in fuel economy for a selected number of spark ignition engine design technologies for passenger cars and light trucks. The engine design technologies examined include:

a) optimal engine control strategy,
b) combustion system and engine configurations, and
c) valving.

DEPARTMENT OF TRANSPORTATION
JUL 22 1980
LIBRARY

17. Key Words Spark Ignition Engine, Engine Control Optimization, Combustion Systems, Engine Materials	18. Distribution Statement DOCUMENT IS AVAILABLE TO THE PUBLIC THROUGH THE NATIONAL TECHNICAL INFORMATION SERVICE, SPRINGFIELD, VIRGINIA 22161

19. Security Classif. (of this report)	20. Security Classif. (of this page)	21. No. of Pages	22. Price
Unclassified	Unclassified	98	

Form DOT F 1700.7 (8-72) Reproduction of completed page authorized

PREFACE

This report, DOT-TSC-NHTSA-79-56, is one of four companion reports to DOT-TSC-NHTSA-79-52 "Potential of Spark Ignition Engine, 1979 Summary Source Document."* It provides a review and assessment of potential improvements in fuel economy for a number of spark ignition engine design concepts for passenger cars and light trucks. This report is a deliverable under PPA HS-027 "Support for Research and Analysis in Auto Fuel Economy and Related Areas."

*"Potential of Spark Ignition Engine, 1979 Summary Source Document," by T. Trella, R. Zub, and R. Colello, U.S. Department of Transportation Systems Center, Report No. DOT-TSC-NHTSA-79-52, March, 1980.

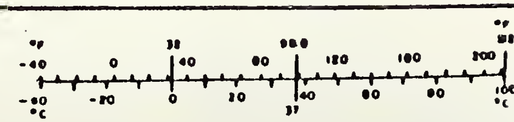
METRIC CONVERSION FACTORS

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



CONTENTS

CONTENTS (Cont'd)

LIST OF ILLUSTRATIONS

LIST OF ILLUSTRATIONS (Cont'd)

LIST OF TABLES

1. INTRODUCTION

In this document, several important engine design concepts are discussed with relevance to fuel economy improvement. These include: engine control, compression ratio, combustion systems, engine configurations, turbocharging and turbocompounding, and valve control. The engine control discussions describe the techniques and procedures utilized to calculate fuel economy achievable through engine control optimization. Data acquired from 305 CID and 121 CID engines are analyzed. Calculated fuel economy potential is presented. The other topics listed above are treated through presentation of data available in technical literature and through Department of Transportation spark ignition engine assessment contracts.

2. ENGINE CONTROL VARIABLES

2.1 GENERAL

With the coming of the energy crisis in 1974, it became of national importance to improve transportation efficiency, in general, and fuel efficiency of automobile engines in particular. Before 1968, the manufacturers of spark ignited (S.I.) engines have been attempting to meet the increasingly stringent, federally-mandated automotive emission standards. Initial attempts to meet the emission constraints cause sharp declines in fuel economy of a significant portion of the total automotive fleet. Since 1975, with the introduction of the exhaust oxidation catalyst, for reduction of hydrocarbon (HC) and carbon monoixde (CO), a significant reversal in the trend of decreasing fuel efficiency was accomplished. Today, the most significant emission control technology consists of electronic engine control systems. The improtance of this technology to emissions and fuel economy lies in its ability to control "key" engine variables (spark, EGR, A/F ratio, etc.) with a greater degree of accuracy, repeatability and speed.

Spark ignition internal combustion engines have numerous operating variables that must be coordinated to meet optimum fuel economy and power requirements at constrained exhaust emission levels. The obvious variables include air consumption, fuel consumption, and ignition timing. Techniques and engine parameters considered as controls for minimizing exhaust emissions include mixture strength, exhaust gas recirculation (EGR), turbulence, inlet and exhaust temperature and pressure, compression ratio, valve timing, combustion processes, and cylinder chambers. Attempts to control optimally three engine variables (spark timing, exhaust gas recirculation and air-to-fuel ratio) have been addressed.[1,2,3,4] Various treatments have appeared which utilize mathematical formulation of a constrained optimization problem based on measured or simulated engine-load requirements for a given vehicle system. These approaches involve the use of steady state dynamometer-based data to approximate the fuel economy and emissions over the Federal Test Procedure (FTP) driving schedules. One major drawback of these treatments is that they do not account for the variety of transient effects, notably due to cold start and exhaust aftertreatment, which prevail during vehicle operations. Studies[5] are just beginning to appear in attempts to establish approaches to the treatment of these transient effects under optimal control. In addition, there is an increasing requirement to optimize the fuel economy, taking into account the entire engine-vehicle-aftertreatment system. The transmission and vehicle drive components are key variables for future electronic controls. Also, engine variables such as valve timing, combustion chambers, etc. should be taken into consideration.

In this Section, an analytical structure based on current optimization treatments is presented and applied to demonstrate its usefulness in predicting optimal fuel economy under constrained emission levels.

2.2 OPTIMIZATION STUDIES

Optimization studies were performed to examine the optimal tradeoff of selected engine and vehicle parameters relative to fuel economy and emissions. The first study compared the fuel economy potential of two engine control strategies, namely optimal control of three parameters (spark advance, exhaust gas recirculation and air-to-fuel ratio) and two parameters (spark advance and exhaust gas recirculation at stoichiometric mixture conditions). The second study examined the effects of two vehicle/drivetrain parameters, namely inertia weight and rear axle ratio on optimal fuel economy.

The analyses were conducted on two baseline vehicles, the first one of 3000 lb inertia weight equipped with a 121 cubic inch displacement engine, and the second one of 4500 lb inertia weight equipped with a 305 cubic inch displacement engine. The rear axle ratios were 3.31 and 2.56 for the 3000 lb and 4500 lb vehicles respectively unless otherwise specified. A three-speed automatic transmission was selected for the 305 CID engine/vehicle configuration and a four-speed manual transmission for the 121 CID engine/vehicle configuration.

All numerical computations were also carried out at twelve-speed/torque points.

2.2.1 Two and Three Parameter Engine Control

Two engine control strategies were evaluated. The first strategy controls three engine variables by use of an electronic device and controls tailpipe HC and CO emissions by use of an oxidation catalyst. The second strategy controls exhaust gas reciruclation and spark advance at stoichiometric engine mixture conditions. A three-way catalyst is employed for the treatment of tailpipe HC, CO and NO_x. An oxidation catalyst can also be added to further reduce the HC and CO emissions.

Table 2-1 shows the calculated optimal fuel economies without constraint for emissions (i.e., optimal engine settings for best fuel economy). As can be seen, the three-parameter control strategy shows 6 to 10 percent higher fuel economies than the two-parameter control strategy. In relation to the baseline engine (i.e., control settings for an as-received engine), an 11 to 13 percent and 3 to 5 percent improvement were predicted for the 3-parameter and 2-parameter strategies respectively. This trend in fuel economy is to be expected. It should also be noted that the two-parameter control strategy shows higher hydrocarbons and carbon monoxides (measured before the catalyst) than the three-parameter control strategy for the 121 CID engine in a 3000 lb inertia weight vehicle. The 305 CID engine in a 4500 lb inertia weight vehicle, on the other hand, showed lower hydrocarbons for the 2-parameter strategy.

A further examination of the calculated engine control settings revealed that under the control of three engine parameters both engines obtained their best fuel economy at lean mixture conditions. Furthermore, on comparing the spark advance and EGR, the 121 CID engine under the control of two parameters favored higher EGR with

TABLE 2-1. TWO AND THREE PARAMETER OPTIMIZED FUEL ECONOMY

Vehicle Inertia Weight (lbs)	Engine Displacement (CID)	Engine Control Strategy	Baseline Composite Fuel Economy (mpg)	Optimal Composite Fuel Economy (mpg)	Unconstrained Engine-out Emissions HC/CO/NO_x gms/mile	Improvement (%)
3000	121	3-paremeter	29.0	32.5	1.01/1.84/5.65	+11
		2-paremeter		30.4	1.57/5.87/1.96	+5
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

no appreciable change in spark advance over the three-parameter control strategy. On the other hand, the 305 CID engine, under the control of two parameters, favored a retardation in spark relative to the three-parameter strategy. This may partially explain the difference in the observed trend in HC emissions.

The fuel economy/emission tradeoff characteristics are shown in Figures 2-1 through 2-4. The two-dimensional plots of optimal warm engine fuel economies are displayed as a function of the constrained hydrocarbon and oxides of nitrogen emissions for the two control strategies.

2.2.2 Effect of Engine Emission Levels

Sensitivity analyses were conducted to relate the percent difference in optimal warm engine fuel economy (relative to the baseline configuration, see Table 2-1) to various engine emission levels under two- and three-parameter control. For purposes of illustration, the two engines discussed in the previous section were analyzed. The results are shown in Tables 2-2 through 2-5.

TABLE 2-2. THREE-PARAMETER ENGINE CONTROL (121 CID ENGINE)*

WARM ENGINE HC EMISSION LEVEL (gms/mile)	ENGINE CO EMISSION LEVEL (gms/mile): 11.25	7.5
1.35	+5.6%**	+4.9%**
0.675	+3.0%**	+2.4%**

*Warm engine NO_x emission level = 0.66 gms/mile.
**(+) indicates fuel economy increase above baseline engine (see Table 2-1).

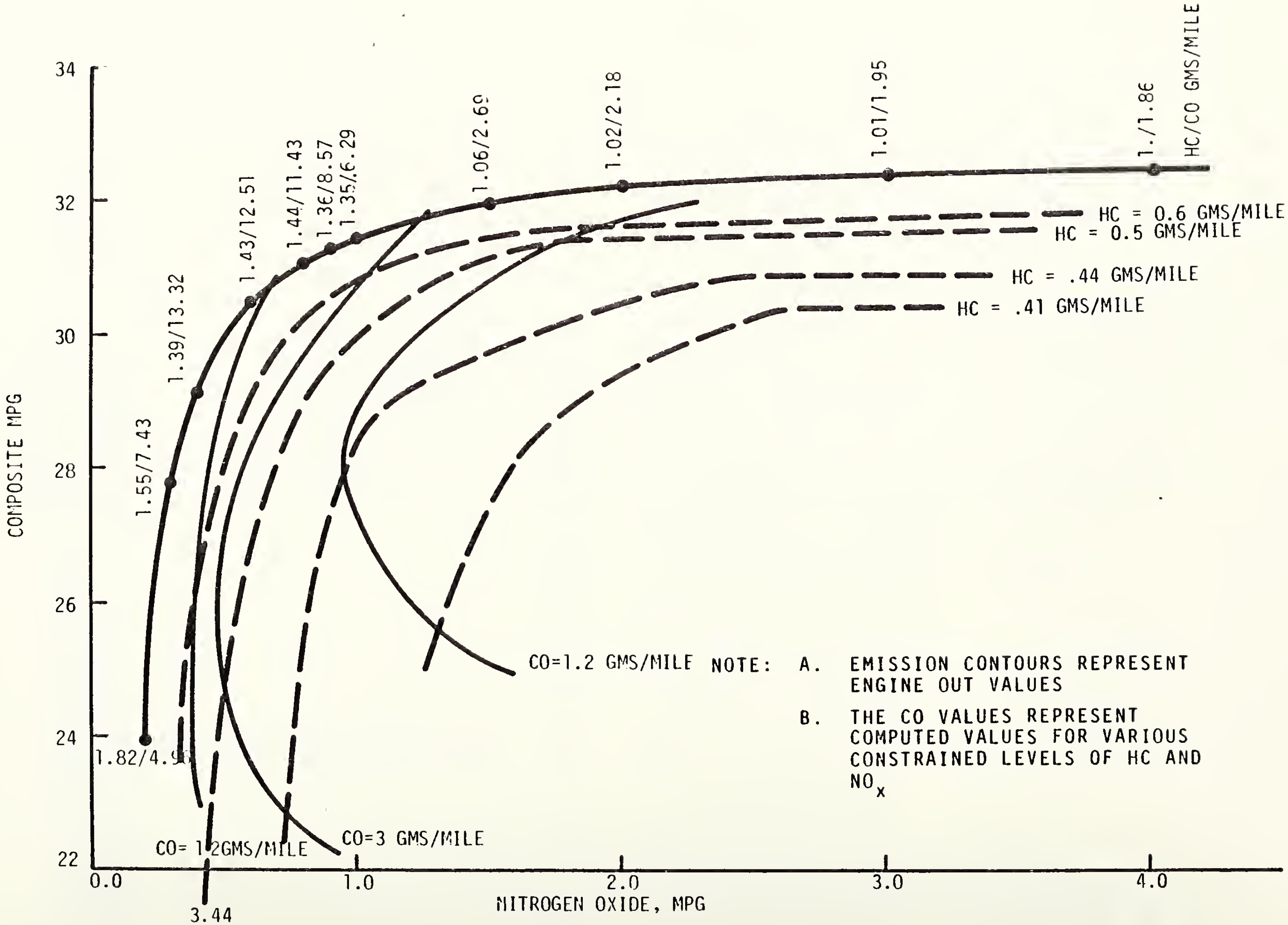


FIGURE 2-1. OPTIMAL FUEL ECONOMY MAP FOR THREE-PARAMETER ENGINE CONTROL (121 CID Engine)

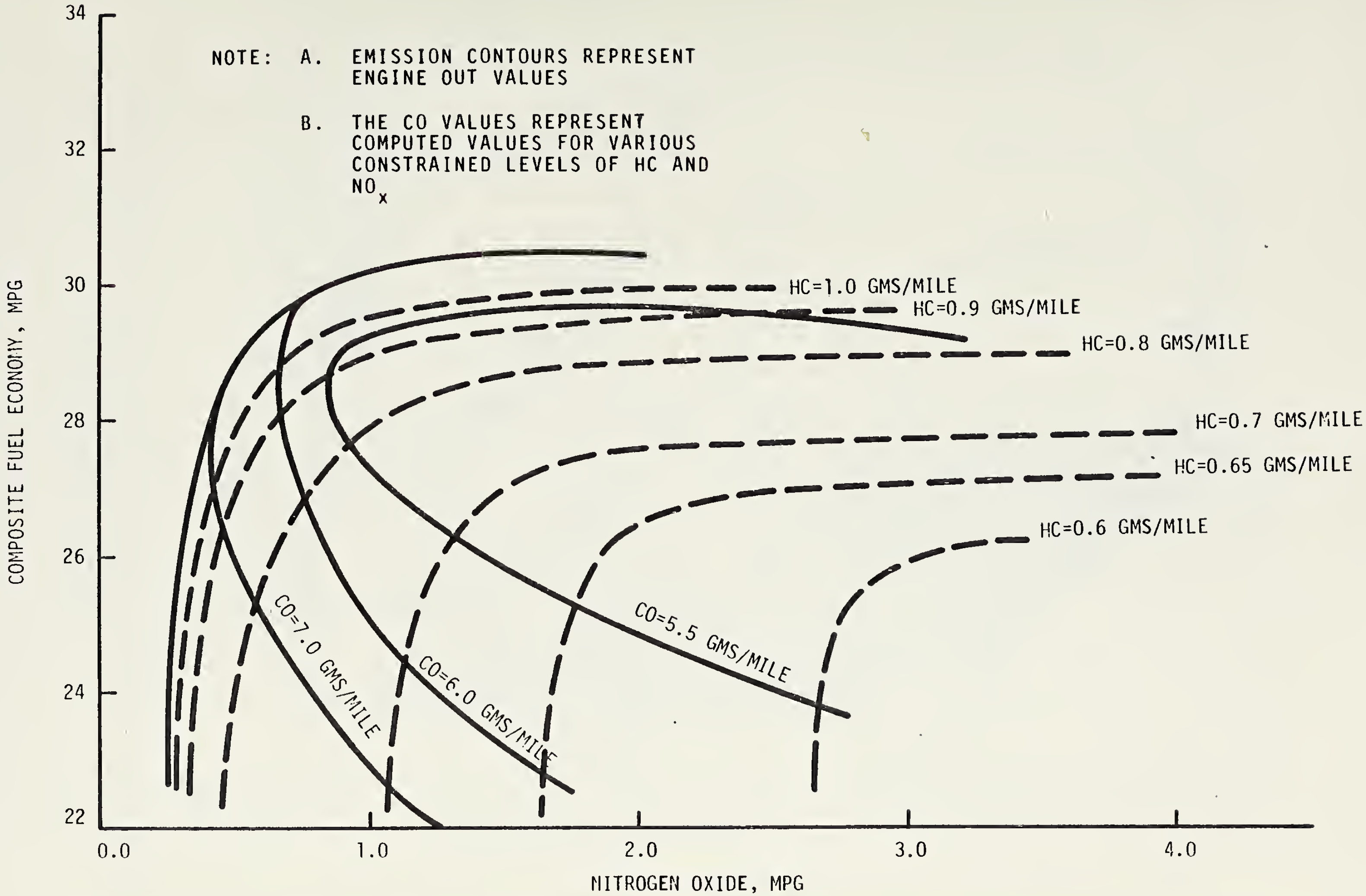


FIGURE 2-2. OPTIMAL FUEL ECONOMY MAP FOR TWO-PARAMETER ENGINE CONTROL (121 CID Engine)

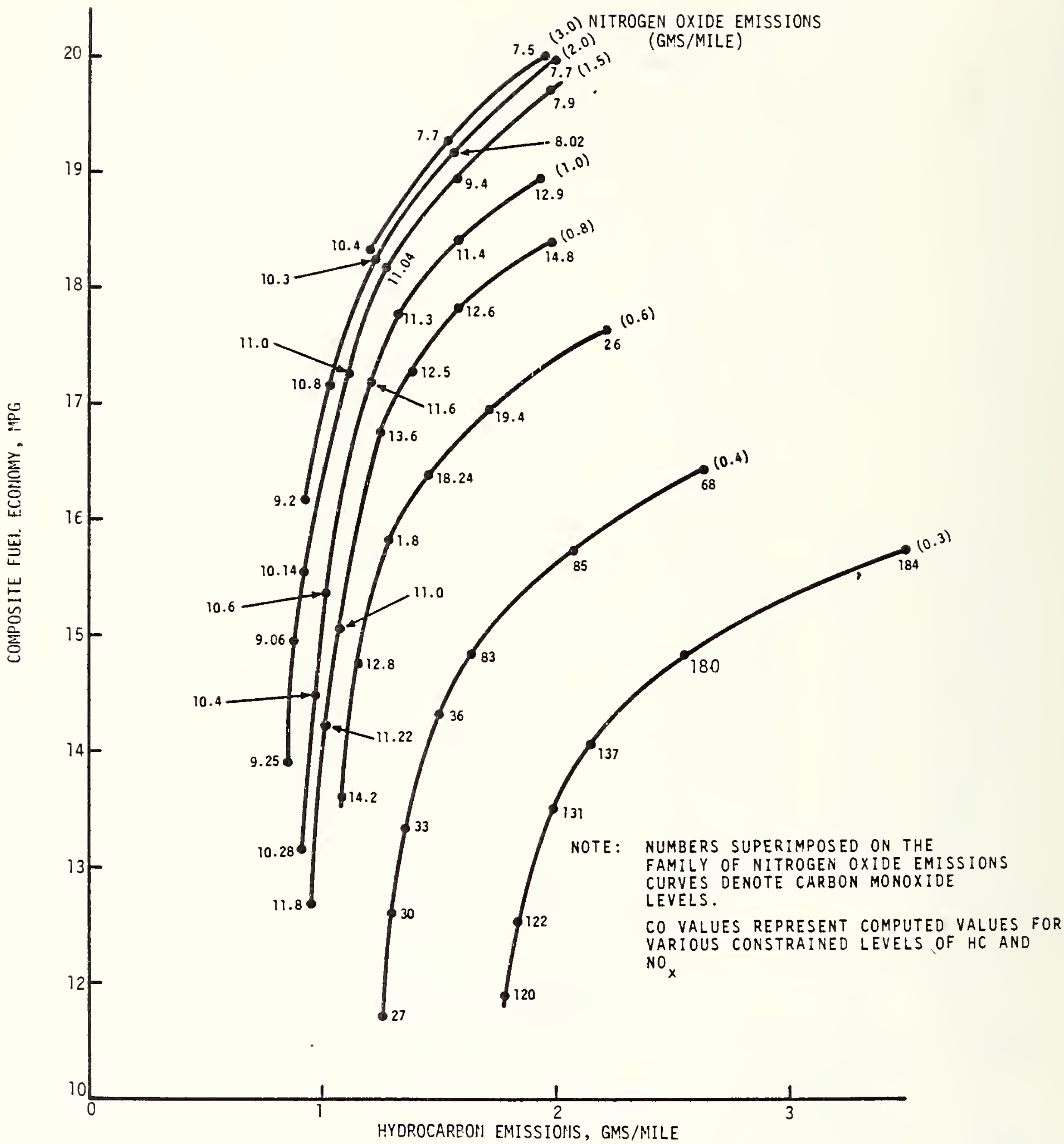


FIGURE 2-3. OPTIMAL FUEL ECONOMY MAP FOR THREE-PARAMETER ENGINE CONTROL (305 CID Engine)

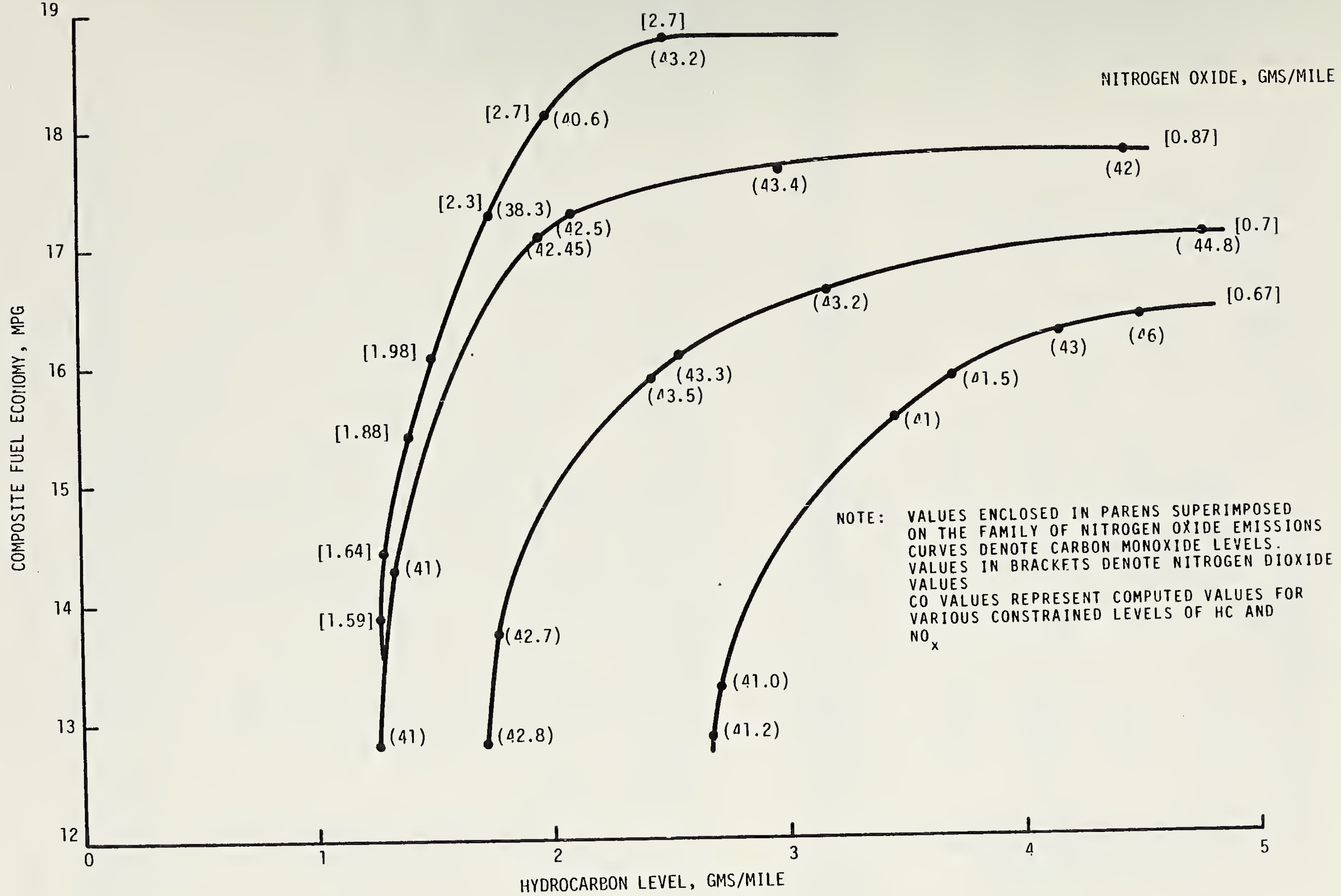


FIGURE 2-4. OPTIMAL FUEL ECONOMY MAP FOR TWO-PARAMETER ENGINE CONTROL (305 CID Engine)

TABLE 2-3. TWO-PARAMETER ENGINE CONTROL (121 CID ENGINE)*

WARM ENGINE HC EMISSION LEVEL (gms/mile)	WARM ENGINE NO_x EMISSION LEVEL (gms/mile) 3.3	2.2	1.65
1.35	+4.8%**	+4.8%**	+4.7%**
0.90	+2.1%**	+2.1%**	+1.9%**
0.675	-4.7%***	-6.0%***	-6.6%***

*Warm engine CO emission levels less than 11.25 gms/mile.
**(+) indicates fuel economy increase above baseline engine (see Table 2-1).
***(-) indicates fuel economy decrease below baseline engine (see Table 2-1).

TABLE 2-4. THREE-PARAMETER ENGINE CONTROL (305 CID ENGINE)*

WARM ENGINE HC EMISSION LEVEL (gms/mile)	22.5	11.25	7.5
5.40	+6.5%***	+6.5%***	+6.5%***
2.70	+2.8%***	+2.8%***	+2.8%***
1.35	-12.0****	-14.0%****	-20.0%****
0.90	**	**	**

*Warm engine NO_x emission level = 0.66 gms/mile.
**0.90 gms/mi engine HC emission not achievable.
***(+) indicates fuel economy increase above baseline engine (see Table 2-1).
****(-) indicates fuel economy decrease below baseline engine (see Table 2-1).

TABLE 2-5. TWO-PARAMETER ENGINE CONTROL (305 CID ENGINE)*

WARM ENGINE HC EMISSION LEVEL (gms/mile) \ WARM ENGINE CO EMISSION LEVEL (gms/mile)	22.5	11.25	7.5
2.7			
1.35		warm engine emission levels could not be met	
0.90			
0.675			

*Warm engine NO_x emission level = 2.2 gms/mile.

Under the control of three parameters, engine carbon monoxide and hydrocarbon levels ranged from 7.5 to 22.5 and 0.90 to 5.4 grams/mile respectively. Under the 2-parameter control, optimal fuel economies are illustrated as a function of carbon monoxide and oxides of nitrogen for the 121 CID engine and as a function of hydrocarbon and carbon monoxide emissions for the 305 CID engine.

The two vehicles exhibit different fuel economy sensitivities to the desired emission levels. The lighter weight vehicle under the control of three parameters shows a 2.4 percent improvement in fuel economy for an assumed engine HC and CO level of 0.675 and 7.50 gms/mile respectively, whereas the heavier vehicle could not be optimally calibrated (from a fuel economy point of view) to meet the hydrocarbon engine emission level at the constrained NO_x emission level of 0.66 grams/mile. See Table 2-5.

An improvement in fuel economy of 4.8 percent is shown for the light-weight vehicle under control of two parameters for engine emission levels of 1.35/11.25/2.2, HC/CO/NO_x; gms/mile respectively.

This percentage improvement in fuel economy approaches that noted in the previous section under no emission constraints (4.8 percent compared to 5 percent). On the other hand, the heavier vehicle could not meet the engine emission levels as indicated by Table 2-5 because of the high CO emission issuing under the 2-parameter control strategy.

2.2.3 Effect of Engine/Vehicle Configuration

Table 2-6 compares the baseline and 3-parameter optimized 305 CID engine fuel economies and their relative differences for four engine/vehicle configurations. The designated fuel economies apply to a 2.8/40/1.5, $HC/CO/NO_x$, grams/mile warm engine emission level. In general, the analysis showed that the improvement in fuel economy did not vary to a large degree when the vehicle's weight was increased from 4500 lbs to 5000 lbs for the rear axle ratio and vehicle frontal area of 3.08 and 39 ft^2 respectively. Reduction in rear axle ratio showed an improvement in fuel economy from 7 to 11 percent. However, a lower inertia weight vehicle of reduced frontal area showed the least improvement.

2.3 OPTIMIZATION STRUCTURE

Numerous authors[1,2,3,4] have presented theoretical treatments and generalized analyses on optimizing fuel economy of automobile vehicles when a number of engine operating control parameters are constrained by a specified emission level. All these approaches use steady state engine data to approximate fuel economy and emissions over a preselected driving schedule. In this case, the 1975 FTP-Federal drive cycle was considered.

Of these treatments, a popular one[1] consists of an on-line approach in which a computer interface (either a microprocessor or minicomputer) programmed with an optimization algorithm acquires engine mapping data. Simultaneously, it seeks out the minimum values of a Performance Index Function using a selected Lagrange multiplier to establish the optimal fuel economy and control settings for a specific emission level. A second approach[4] makes

TABLE 2-6. BASELINE/OPTIMIZED ENGINE FUEL ECONOMY

VEHICLE INERTIA WEIGHT (lbs)	ENGINE DISPLACEMENT (in^3)	REAR AXLE RATIO	VEHICLE FRONTAL AREA (ft^2)	COMPOSITE FUEL ECONOMY (mpg)		OPTIMIZATION PERCENTAGE INCREASE (%)
				BASELINE	OPTIMIZED	
4000	305	3.08	17.1	18.0	18.9	+ 5
4500	305	2.56	38.0	18.2	20.2	+11
4500	305	3.08	38.0	17.6	18.8	+ 7
5000	305	3.08	38.0	17.2	18.5	+ 7.5

use of off-line procedures. Briefly, this approach considers two tasks. First, engine parametric data (i.e., fuel consumption, emission rates, etc.) are experimentally collected as functions of the engine operating variables (i.e., spark advance, exhaust gas recirculation, etc.), and analytical expressions are generated to relate the first set of variables to a second set over the engine operating range. It is popular, in most instances, to employ multiple polynomial regression procedures to generate such expressions. The second task includes the optimization analysis itself. Two techniques have been generally employed in the past. The first is mentioned above under the on-line approach, and the second uses dynamic programming procedures to address a multi-dimensional allocations process.[6] Both optimization approaches yield the same results. The off-line approach was preferred in our case, because it makes it possible to incorporate the vehicle driveline characteristics and to use the engine parametric data base over the full range of variability, without additional testing. Figure 2-5 presents a flow chart of the major steps considered as part of the optimization procedures. These steps appear in sequence and constitute, a) vehicle driving simulation analysis, b) determination of time weighting coefficients, c) acquisition of engine experimental mapping data and multiple regression analyses, and d) fuel economy/emission optimization analyses. Mathematically, the optimization procedures can be formulated in the following manner:

A. <u>Optimization Function</u>

$$\text{Minimum Fuel Flow (lbs/hr)} = A \sum_{1}^{N} T_i^u \text{ (Fuel Rate)}_i$$

$$+B \sum_{1}^{N} T_i^H \text{ (Fuel Rate)}_i$$

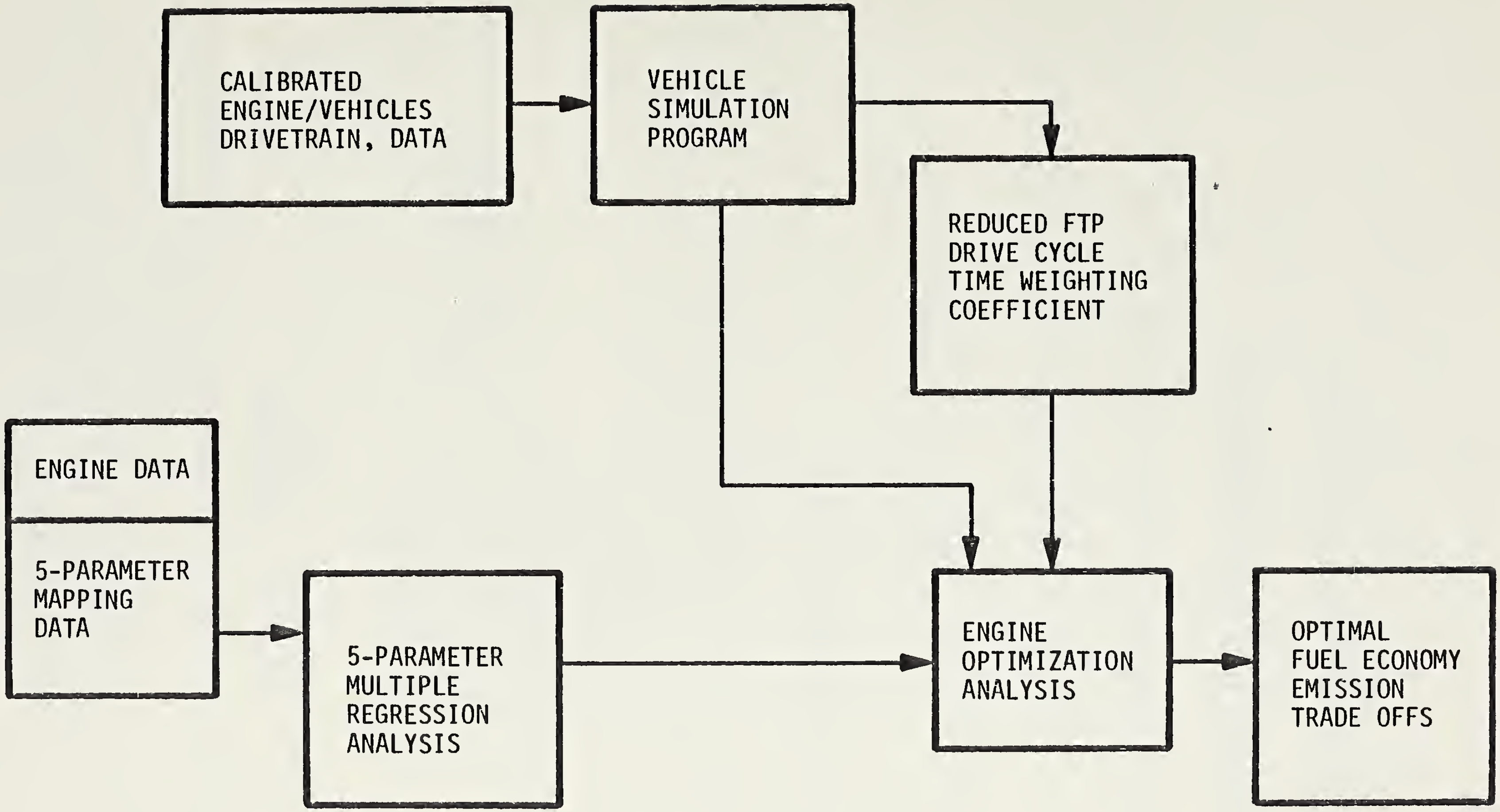


FIGURE 2-5. ENGINE OPTIMIZATION ANALYSIS STRUCTURE

where the constants are defined by

i = index representing speed/torque point

$$A = \frac{\text{Urban Time for 55 miles } (T^u)}{\text{Composite time for 100 miles}}$$

$$B = \frac{\text{Highway Time for 45 miles } (T^H)}{\text{Composite time for 100 miles}}$$

T_i^u; T_i^H = urban and highway percentage time weightings

T^u; T^H = FTP urban and highway driving cycle time.

$$\text{Composite time (100 miles)} = \frac{55\ T^u}{\text{Distance}^u} + \frac{45\ T^H}{\text{Distance}^H} .$$

B. Constraint Functions

$$\text{Emissions Flow (gms/hr) } (NO_x,\ CO,\ HC) = \sum_i T_i^u \ (\text{Emission Rate})_i .$$

C. Fuel Economy/Emissions

Fuel Consumed (miles/gal)

$$= (\text{Fuel Density/Fuel Flow}) \ \frac{100 \text{ miles}}{\text{Composite Time}}$$

Emissions (gms/mile)

$$= (\text{Emission Flow}) \ \frac{T^u}{\text{Distance}^u}$$

Distance^u = distance traversed during urban driving

Distance^H = distance traversed during highway driving.

The procedure is designed to approximate the vehicles' driving schedule with a set of points in the engine's load-speed plane. A vehicle simulation computer model is used in selecting the engine load-speed points.

Usually, the time weighting coefficients are obtained from vehicle simulation analysis and are distributed over a large set of speed/torque points. In most instances, numerical compuation must be carried over a finite number of speed and torque points. Generally, eight to twelve points are considered, but the analysis may be performed over an expanded set of points. Various authors have suggested schemes to select a reduced number of engine speed/torque points to cover the operating range of the engine for a particular driving schedule of the vehicle. The procedure adopted here follows that presented in Reference 1. The selection procedure consists of locating points on the road load curve, below the curve to represent deceleration and above the curve for part-throttle acceleration. A final speed/torque point is chosen at idle.

The optimization strategy, in conjunction with the powertrain simulation, required tables of discrete values of fuel and emission flows as function of the control variables at any speed/load point. Multivariate regression techniques applied to discrete data can provide analytical expression for fuel flow and pollutant flows as a function of any number of the experimental independent engine variables. Five critical independent variables were selected, namely spark advance, air-to-fuel ratio, exhaust gas recirculation, engine speed, and torque.

The engine maps available consisted of data accumulated at constant speed and torque, the data being a set of measures of engine performance in terms of fuel flow and pollutant flows, and engine operating conditions in terms of spark advance, exhaust gas recirculation and air-to-fuel ratio.

Multivariate least-square regression techniques[7] provide an efficient and accurate tool for screening, smoothing, and interpolating data bases when its application is tailored to the specific nature of the data base under study.

The size of the data base affects the degree of the polynomial chosen to fit the dependent variable in order to assure reliability of the regression coefficients. The physical nature of the dependent variable also affects the degree of the fitting polynomial since a fast varying variable can be better represented by a higher order polynomial than a slow varying variable. Also, the order of the polynomial should be as high as necessary to give a robust representation of the variable, but not too high to include questionable, experimental points. Use of higher order polynomials also affects the computation of the dependent variable for a new set of the independent variables; it is important that the interpolation should be restricted only to the range of the independent variables in which the actual data limits points were accumulated. Attempts to compute the dependent variable for values of the independent variables lying outside of the original experimental range involve extrapolation and may yield meaningless results. Variables, either dependent or independent, can benefit from transformations. In engine data maps, logarithmic representation, decimal or natural, of pollutant flows is with respect to the control variables.

Also, close inspection of the regression coefficients is essential in order to reduce the fitting polynomial to the relevant terms, therefore improving the degree of overdetermination. Generally, two quantities, R^2 (Multiple R-square) and R (Multiple R), provided by multiple regression analysis, can be considered as indicators of the average goodness of fit.

2.4 MULTIPLE REGRESSION ANALYSIS

Multiple polynomial regression was applied to five parameter engine maps in order to express experimental fuel and emission rates as functions of load, speed and control settings, (i.e., ignition timing, air-to-fuel ratio, and percent exhaust recirculated gas). A carbureted 121 CID[9] and 305 CID engine[8] were analyzed by this procedure.

2.4.1 121 CID Engine Regression Analysis

965 measurements of engine performance and control setting accumulated at 21 speed/load points were available for analysis: the engine performance in terms of fuel and emission flows, and

the control setting in terms of spark advance (SA), exhaust gas recirculation (EGR), and air-to-fuel ratio (A/F). The number of measurements for each speed/torque point varied from a minimum of 4 to a maximum of 64, and the torque (TQ) varied from a minimum of 0 TQ to a maximum of 70 Torque, and the speed (SP) varied from 900 to 3000 RPM. A natural logarithmic transformation was applied to the emission flows; torque was converted to 10^{-1} BMEP and speed to 10^{-3} RPM.

The range of the engine variables are illustrated in Table 2-7. In the subsequent analysis, fuel flow (FC) and the emission flows are the dependent variables and the independent variables include SA, EGR, AFR, 10^{-3}RPM, 10^{-1}BMEP and the combination thereof.

The requirements of the optimization algorithm in conjunction with the powertrain simulation, and the lack of sufficient number of observed data at each speed/torque point, demanded a five independent variables model. This approach has its advantages in the sense that it makes use of all observations in the data base, so allowing greater flexibility in the order of fitting polynomials and assuring greater reliability in the regression coefficients. The outcome in terms of R and R^2 is summarized in Table 2-8. The B-weights or regression coefficients are listed in Table 2-9, with the variable to which they apply. A close examination of the values of dependent variables predicted by the regression equations supplies some relevant information. If for each measured value the percentage error is computed as

$$\% \text{ ERR} = \frac{(\text{OBSERV} - \text{PREDICT})}{\text{OBSERV}} * 100$$

it is observed that the most misrepresented regressed values fall in the interval $TQ \leq 20$. (See Table 2-10). In fact, if all the values are flaged, for which the %ERR falls outside the range mean $\pm$ 2.5 SD DEV, the following result is obtained.

Given 183 observations with $TQ \leq 20$
742 observations with $TQ \geq 30$.

TABLE 2-7. AUDI ENGINE DATA, N = 965

	Minimum	Maximum
RPM	900.	3000.
TQ	0.	70.
SA	-2.	58.
EGR	0.33	15.4
AFR	11.34	26.77
FUEL FLOW (lbs/hr)	1.78	21.45
HC (g/hr)	2.97	168.72
NO_x (g/hr)	0.33	517.52
CO (g/hr)	31.82	5662.26

TABLE 2-8. AUDI ENGINE DATA, FUEL QUADRATIC POLYNOMIAL REGRESSION N = 965

Dependent Variable	FC	$Log_e(HC)$	$Log_e(NO_x)$	$Log_e(CO)$
R	.99	.90	.96	.91
R^2	.98	.81	.93	.84

TABLE 2-9. REGRESSION COEFFICIENTS

Independent Variables	FC	Dependent Variables: $\log_e(NO_x)$	$\log_e(HC)$	$\log_e(CO)$
AFR	0.65500000E-02	-0.15136600E+00	-0.28609000E+00	-0.[illegible]E+00
EGR	-0.40210000E+00	0.13851000E+00	-0.18413000E+00	0.17316000E+00
SA	0.38910000E-01	-0.12870000E-01	0.66900000E-02	-0.12148000E+00
EGR*SA	-0.44200000E-02	-0.93000000E-03	-0.67000000E-03	0.22000000E-03
10^{-3}RPM	0.20434500E+01	0.17473000E+01	-0.68062000E+00	0.24500500E+01
10^{-1}BMEP	0.82904000E+00	0.39403000E+00	0.39622000E+00	0.16472600E+01
$(10^{-3}RPM)^2$	0.27923000E+00	-0.30700000E-02	0.12441000E+00	-0.24216000E+00
$(10^{-1}BMEP)^2$	-0.26670000E-01	-0.49880000E-01	0.34900000E-02	0.59100000E-02
AFR^2	0.21800000E-02	0.58000000E-02	0.43900000E-02	0.15890000E-01
EGR^2	0.51200000E-02	-0.15000000E-02	0.27000000E-02	-0.39000000E-03
SA^2	0.28600000E-02	-0.58000000E-03	0.15000000E-03	-0.40000000E-03
$(10^{-3}RPM)*(10^{-1}BMEP)$	0.64199000E+00	-0.46460000E-01	0.30990000E-01	0.66300000E-02
10^{-3}RPM*AFR	-0.13310000E-01	-0.62600000E-01	0.16960000E-01	-0.66500000E-01
10^{-3}RPM*EGR	0.60190000E-01	-0.12710000E-01	-0.65200000E-02	0.13000000E-03
10^{-3}RPM*SA	-0.64060000E-01	0.26100000E-02	-0.14100000E-02	0.67200000E-02
10^{-1}BMEP*AFR	-0.44610000E-01	0.36880000E-01	-0.27230000E-01	-0.11242000E+00
10^{-1}BMEP*EGR	-0.58700000E-02	-0.23610000E-01	-0.71900000E-02	-0.58000000E-03
10^{-1}BMEP*SA	-0.49200000E-02	0.66700000E-02	0.27900000E-02	-0.12900000E-02
AFR*EGR	0.28600000E-01	-0.63500000E-02	0.15960000E-01	-0.10250000E-01
AFR*SA	-0.59900000E-02	-0.38000000E-03	-0.61000000E-03	0.81700000E-02
COST	-0.13190000E-01	0.28469000E+00	0.60441400E+01	0.12657770E+02

TABLE 2-10. REGRESSED VALUES VS. PERCENTAGE ERROR

		# points flagged	% points flagged	range for %FRR
FC	TQ≤20	18	9.8%	-24.82, 20.53
	TQ≥30	1	.13%	
$\log_e$(HC)	TQ≤20	16	8.8%	-30.515, 29.845
	TQ≥30	5	.65%	
$\log_e(NO_x)$	TQ≤20	12	6.6%	
	TQ≥30	0	0%	-308.142, 311.368
$\log_e$(CO)	TQ≤20	4	2.2%	-27.184, 25.496
	TQ≥30	4	.52%	

In view of these results, it was decided that a possible worthwhile approach would be to find a fitting polynomial at $TQ \leq 20$ and a fitting polynomial at $TQ \geq 30$. Unfortunately, the small number of measurements at $TQ \leq 20$ could not allow great flexibility in the order of the fitting polynomial for the dependent variables. Thus, it was decided that it be limited to quadratic equations in the five independent variables, 10^{-3}RPM, 10^{-1}BMEP, SA, EGR and AFR for FC, $\log_e(NO_x)$, $\log_e(CO)$, at both $TQ \leq 20$ and $TQ \geq 30$.

The analytical representation of $\log_e(HC)$ was proceeded in a different way. A cubic fit across all SP/TQ points was used instead of the quadratic fit since previous experience indicated a better fit using a full cubic in five independent variables and since the number of coefficients involved (55) requires a large number of observations.

The results of the new regression analysis on FC, $\log_e(NO_x)$, $\log_e(CO)$ and $\log_e(HC)$ are illustrated in Tables 2-11 and 2-12 in terms of R and R^2. In each case the equations are reduced to essential terms through a replication of regression analyses that eliminated coefficients affected by unacceptable standard errors and those whose contribution was negligible.

In Table 2-13, the regression weights are listed along with the variables to which they apply.

Finally, comparing Table 2-10 with Table 2-14, it can be concluded that the fitting of the engine data map in two regions does not effect in average the fitting of FC, improves the fitting of log (CO) in both regions, and improves the fitting of $\log_e(NO_x)$ in the region $TQ \geq 30$. Table 2-11 indicates a definite improvement of $\log_e(HC)$ using a cubic fit instead of a quadratic fit.

The range of %ERR for the dependent variables predicted by the new polynomials is illustrated below. (Table 2-14.)

It can be concluded after comparing the %ERR presented in Table 2-14 with those in Table 2-10 that the fitting curves on

TABLE 2-11. AUDI ENGINE DATA
QUADRATIC POLYNOMIAL REGRESSION
2 REGIONS: $TQ \leq 20$; $TQ \geq 30$

Dependent Variable		FC	Log (NO_x)	Log (CO)	
TQ<20	R	.995	.97	.90	
	R^2	.990	.95	.81	N = 183
TQ>30	R	.99	.95	.96	
	R^2	.98	.90	.93	N = 742

TABLE 2-12. AUDI ENGINE DATA HCL - N = 965

Cubic Fit		Quadratic Fit	Δ % Variance
R	.95	.90	
R	.89	.81	8%

TABLE 2-13. AUDI ENGINE DATA

QUADRATIC POLYNOMIAL REGRESSION

Torque ≤ 20 ft-lbs

Regression Coefficients Reduced To Essential Terms

	FC	$\log_e$(NOX)	$\log_e$(CO)
AFR	0.00000000E+00	0.00000000E+00	-0.41227000E+00
EGR	-0.38400000E+00	0.00000000E+00	-0.31020000E+00
SA	0.00000000E+00	0.00000000E+00	-0.62700000E-02
RPM 10^{-3}	0.00000000E+00	0.00000000E+00	0.00000000E+00
BMEP 10^{-1}	0.00000000E+00	0.00000000E+00	0.00000000E+00
(RPM $10^{-3})^2$	0.21058000E+00	0.00000000E+00	-0.77235000E+00
(BMEP $10^{-1})^2$	0.00000000E+00	0.00000000E+00	0.20853400E+01
AFR^2	0.63000000E-03	0.00000000E+00	-0.51000000E-03
EGR^2	0.36800000E-02	0.00000000E+00	0.25700000E-02
SA^2	0.15000000E-02	0.40000000E-03	-0.17000000E-03
10^{-4}RPM*BMEP	0.13337200E+01	0.37135000E+00	0.45221000E+00
10^{-3}RPM*AFR	0.00000000E+00	0.00000000E+00	0.16113000E+00
10^{-3}RPM*FGR	0.22890000E-01	0.00000000E+00	0.19400000E-02
10^{-3}RPM*SA	-0.22250000E-01	0.00000000E+00	0.30700000E-02
10^{-1}BMEP*AFR	-0.35580000E-01	0.24750000E-01	-0.38404000E+00
10^{-1}BMEP*EGR	0.27490000E-01	-0.29020000E-01	0.11060000E-01
10^{-1}BMEP*SA	-0.19340000E-01	0.72900000E-02	-0.79200000E-02
AFR*EGR	0.17920000E-01	0.00000000E+00	0.19470000E-01
AFR*SA	-0.19300000E-02	0.00000000E+00	0.17500000E-02
EGR*SA	-0.22500000E-02	-0.14100000E-02	-0.73000000E-03
COST	0.21316200E+01	0.14667000E+00	0.89255000E+01

Torque ≥ 30 ft-lbs

	FC	$\log_e$(NOX)	$\log_e$(CO)
AFR	-0.17340500E+01	0.29316100E+01	-0.85363400E+01
EGR	-0.47216000E+00	0.18321000E+00	-0.20017000E+00
SA	0.15064000E+00	-0.36000000E-01	-0.13220000E-01
RPM 10^{-3}	0.41131100E+01	0.17135200E+01	0.75293000E+00
BMEP 10^{-1}	0.19626200E+01	0.14849000E+00	-0.10116000E+00
(RPM $10^{-3})^2$	0.00000000E+00	-0.32590000E-01	-0.10079000E+00
(BMEP $10^{-1})^2$	-0.21040000E-01	-0.23850000E-01	-0.51900000E-02
AFR^2	0.76200000E-01	-0.98650000E-01	0.24485000E+00
EGR^2	0.70400000E-02	-0.24400000E-02	0.80000000E-03
SA^2	0.33900000E-02	0.37000000E-03	0.20000000E-04
10^{-4}RPM*BMEP	0.48186000E+00	-0.78880000E-01	0.77320000E-01
10^{-3}RPM*AFR	0.00000000E+00	-0.39780000E-01	-0.22200000E-01
10^{-3}RPM*EGR	0.55560000E-01	-0.46600000E-02	0.37900000E-02
10^{-3}RPM*SA	-0.69660000E-01	-0.79000000E-03	0.66500000E-02
10^{-1}BMEP*AFR	-0.10305000E+00	0.35920000E-01	0.82200000E-02
10^{-1}BMEP*EGR	0.00000000E+00	-0.11030000E-01	-0.13700000E-02
10^{-1}BMEP*SA	-0.46400000E-02	0.38700000E-02	-0.27200000E-02
AFR*EGR	0.32510000E-01	-0.15180000E-01	0.15470000E-01
AFR*SA	-0.13910000E-01	0.30200000E-02	0.67000000E-03
EGR*SA	-0.57900000E-02	-0.22000000E-03	-0.11100000E-02
COST	0.60728100E+01	-0.21798000E+02	0.76986610E+02

TABLE 2-13. AUDI ENGINE DATA (CONTINUED)

(Cubic Polynomial Regression)

Regression Equation Reduced to Essential Terms

$$\log_e(HC) = -1.95941\ AFR + .50623\ EGR - .07447\ SA - 1.48832\ BMEP\ 10^{-1} +$$

$$1.14057\ (RPM\ 10^{-3})^2 + .12027\ (BMEP\ 10^{-1})^2 + .08003\ AFR^2 -$$

$$.03380\ EGR^2 - .00207\ SA^2 + .41890\ RPM*BMEP\ 10^{-4} - .41997\ RPM\ 10^{-3}*AFR$$

$$+ .03847\ RPM\ 10^{-3}*EGR + .00061\ RPM\ 10^{-3}*SA + .06759\ BMEP*AFR\ 10^{-1} +$$

$$.03990\ BMEP\ 10^{-1}*EGR + .0011\ BMEP\ 10^{-1}*SA - .08121\ AFR*EGR +$$

$$.01638\ AFR*SA + .00711\ EGR*SA - .00063\ SA*EGR*AFR - .16907\ (RPM\ 10^{-3})$$

$$- .00535\ (BMEP\ 10^{-1})^3 - .00001\ SA^3 + .00025\ EGR^3 - .00142\ AFR^3 +$$

$$[-.08544 * BMEP\ 10^{-1} + .00506 *SA - .00234 *EGR + .01475 *AFR]$$

$$*(RPM\ 10^{-3})^2 + [-.00951 * RPM\ 10^{-3} -.00028 *SA + .00129 *EGR -$$

$$.0013 *AFR] *(BMEP\ 10^{-1})^2 + [.00016 *RPM\ 10^{-3} - .00004 *BMEP\ 10^{-1} +$$

$$.00005 *EGR + .00019 *AFR] *SA^2 + [-.01773 *RPM\ 10^{-3} -$$

$$.00041 *BMEP\ 10^{-1} - .00013 *SA + .00257 *AFR] *EGR^2 + [.01626 *RPM\ 10^{-}$$

$$- .00058 *BMEP\ 10^{-1} - .00052 *SA + .00369 *EGR] *AFR^2 +$$

$$[.00184 *SA + .00785 *EGR - .00453 *AFR] * (RPM\ 10^{-3} * BMEP\ 10^{-1}) +$$

$$[.00044 *EGR - .004433 *AFR] * (RPM\ 10^{-3} *SA) - .00488\ RPM\ 10^{-3}$$

$$* EGR * AFR - .00006 * BMEP\ 10^{-1} + SA * EGR - .00451 *BMEP\ 10^{-1}$$

$$* EGR * AFR + 19.98492.$$

TABLE 2-14. RANGE % ERR

FC	TQ≤20 TQ≥30	-13.69, 13.78 -9.27, 9.16
$\log_e(NO_x)$	TQ≤20 TQ≥30	-366.14, 412.08 -19.81, 18.73
$\log_e(CO)$	TQ≤20 TQ≥30	-23.96, 22.73 -17.81, 16.72
$\log_e(HC)$	cubic fit	-24.11, 26.45

which Table 2-14 is based, give a better representation of the observed data, with the exception of NO_x at $TQ \leq 20$; this does not seem particularly critical as NO_x is a high torque pollutant.

It should be noted that all of these attempts have been purely experimental and definitively limited by the small number of measurements available, especially at low TQ values. Only repeated analyses of extensive engine data bases could develop a more precise opinion about the feasibility of using Multiple Regression Techniques to reproduce and predict engine performances.

Figures 2-6 through 2-12 illustrate the relationship between observed and predicted values and their cluster around the "regression line". The extreme values of both y and x axis correspond to -5 and +5 standard deviations from the y/x mean; the length of the interval is one standard deviation.

2.4.2 305 CID Engine Regression Analysis

826 engine measurements were available for the regression analysis. The independent or control variables were in the form of SA, EGR, and AFR, and the dependent variables were in the form of BSFC, BSHC, $BSNO_x$, and BSCO. The number of measurements for each speed/torque varied from a minimum of 32 to a maximum of 70, and the TQ from a minimum of 0.32 to a maximum of 152. The number of measurements at TQ<20 was 62, at constant 800 RPM.

A full quadratic polynomial regression in five independent variables (RPM/1000, BMEP.10, SA, EGR, AFR) was performed using

1) FC, HC, NO_x, CO as dependent variables,
2) $\log_{10}(HC)$, $\log_{10}(NO_x)$, $\log_{10}(CO)$ as dependent variables,
3) $\log_e(HC)$, $\log_e(NO_x)$, $\log_e(CO)$ as dependent variables,

in order to assess the effect of logarithmic transformation, either decimal or natural, applied to the emission rates. Elementary statistics on the original variables on their transforms are

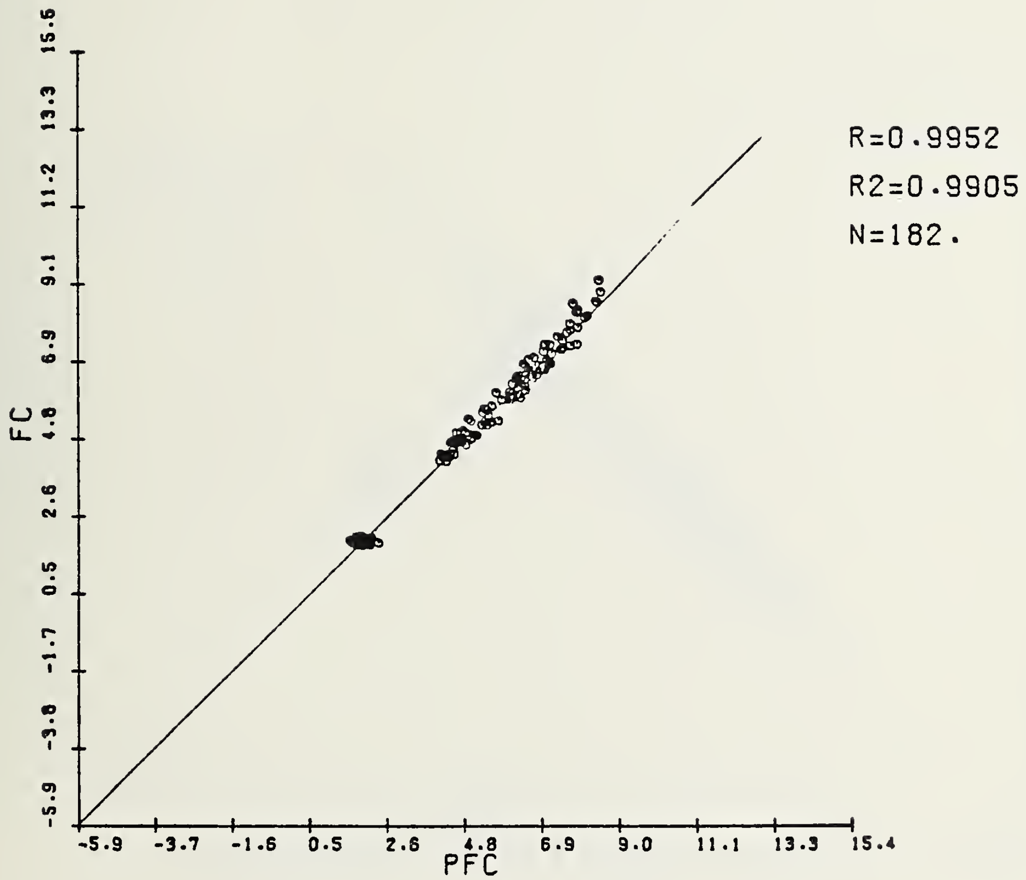


FIGURE 2-6. AUDI ENGINE: OBS VS. PRED FC, TQL≦20

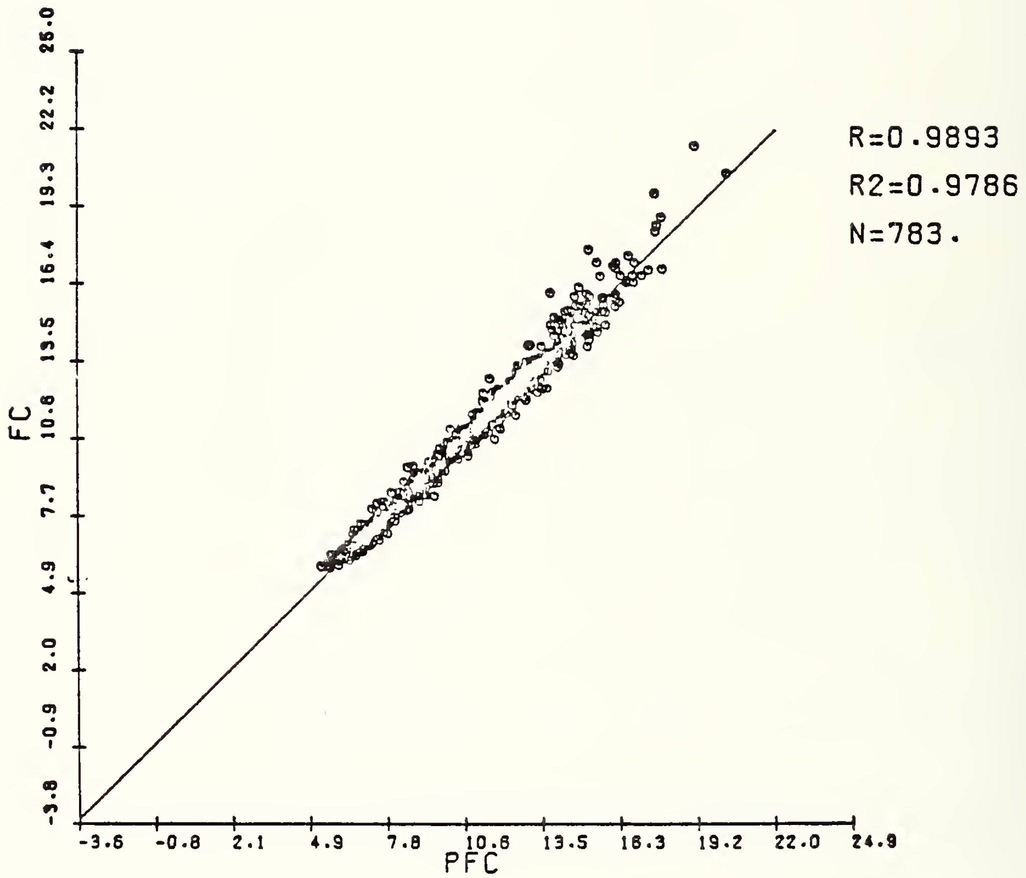


FIGURE 2-7. AUDI ENGINE: OBS VS. PRED FC TQ>20

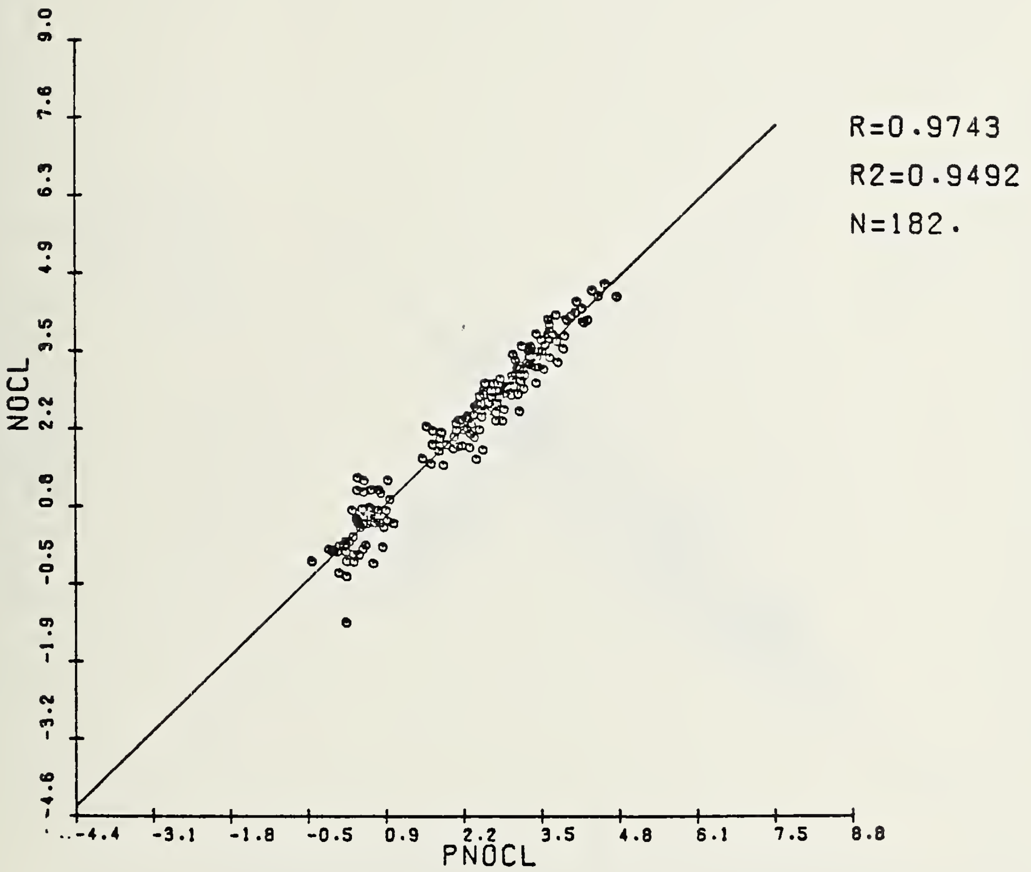


FIGURE 2-8. AUDI ENGINE: OBS VS. PRED NOCL TQ<=20

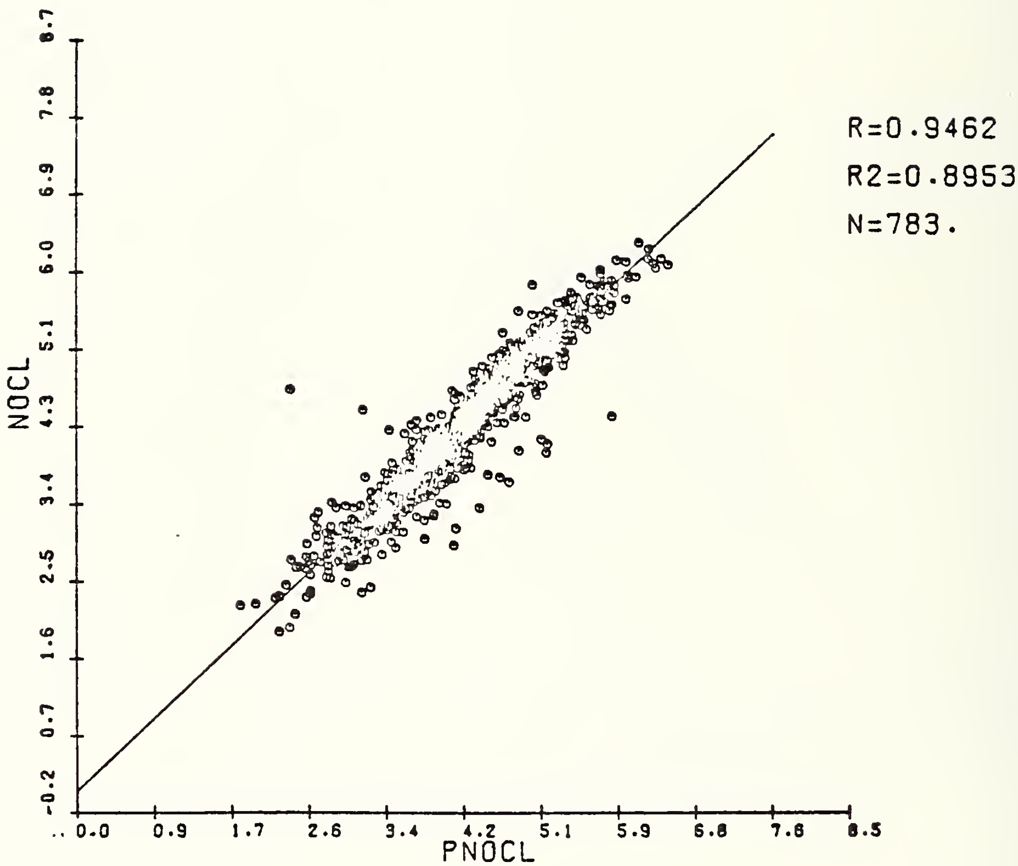


FIGURE 2-9. AUDI ENGINE: OBS VS. PRED NOCL TQ>20

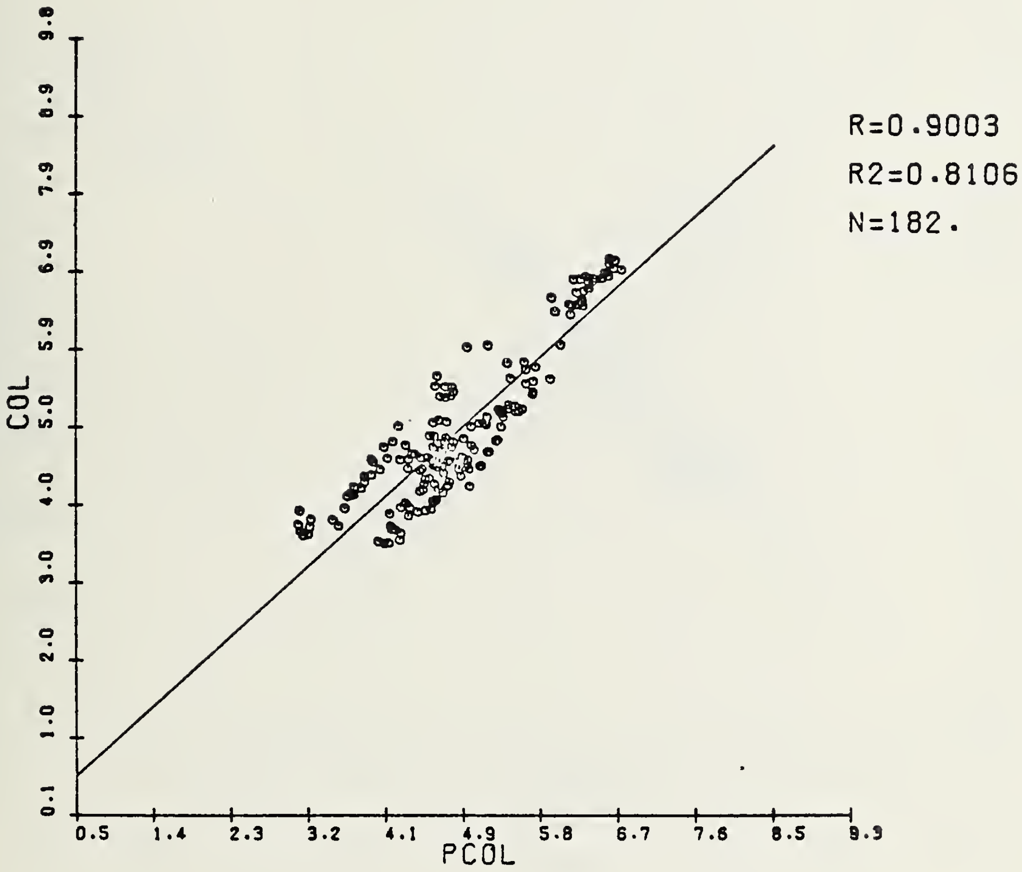


FIGURE 2-10. AUDI ENGINE: OBS VS. PRED COL TQ<=20

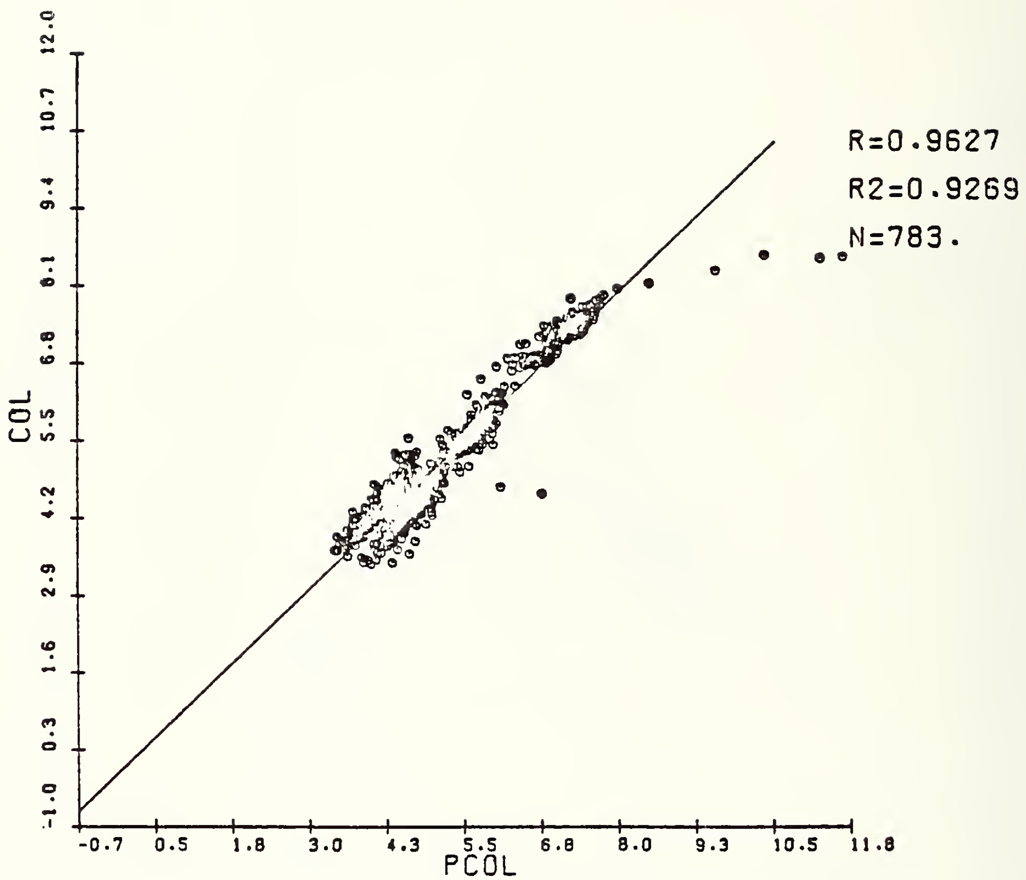


FIGURE 2-11. AUDI ENGINE: OBS VS. PRED COL TQ>20

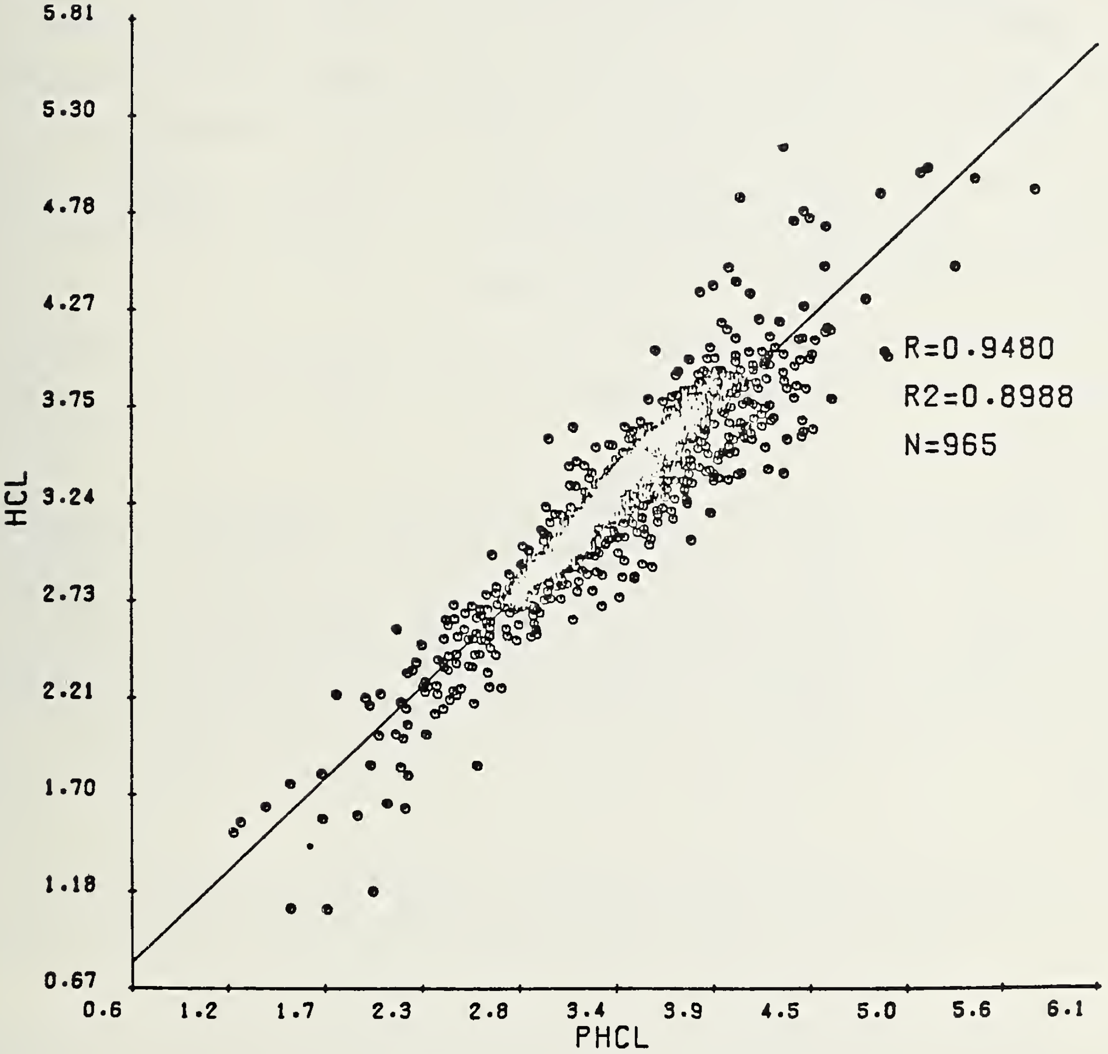


FIGURE 2-12. AUDI ENGINE: OBS VS. PRED HCL CUBIC FIT

reported in Table 2-15. The outcome of the regression analysis performed is reported in Table 2-16 in terms of R and R^2.

Inspection of Table 2-16 offers some suggestions:

a) Fuel flow rate seems very well modelled without performing any transformation even with the inclusion of very low TQ points.

b) Nitrogen oxide flow rate is better modelled by a logarithmic transformation and the inclusion data of Torque below 25 ft-lbs does not affect the fitting task, probably because the nitrogen emission is essentially a steady state pheonmenon.

c) Hydrocarbon and Carbon Monoxide are better modelled by a logarithmimic transformation, although the fitting is not very satisfactory, especially for Hydrocarbon.

d) Decimal and natural logarithm transformations for emissions are equally effective, therefore, the common use of natural log is acceptable.

The regression weights for FC, $\log_e(NO_x)$, $\log_e(HC)$, $\log_e(CO)$, are listed in Table 2-18, with the variables to which they apply.

In order to evaluate the effect of the inclusion of the torque data below 25 ft/lbs on the HC and CO flow rates, the overall regression was repeated for both emissions and their log transforms for all points with $TQ \geq 25$, excluding the 800 RPM, $TQ \simeq 1$ point.

The outcome in terms of R and R^2 is reported in Table 2-17. Inspections of Table 2-18 and comparison with Table 2-17 indicates that:

a) log transformation is effective and there is no difference between decimal and natural log;

b) the fitting of HC and CO emission rates or their log transforms is affected in a negative way (smaller R^2) by the inclusion of data below 25 ft-lbs of Torque, the impact being particularly large for HC.

TABLE 2-15. 305 ENGINE ELEMENTARY STATISTICS N=826

	minimum	maximum
RPM	594.20	2019.0
TQ	0.37	152.0
10^{-3}RPM	0.59	2.02
10^{-1}BMEP	0.02	7.52
SA	0.0	60.0
EGR	0.74	31.71
AFR	12.37	22.04
FC	3.15	32.26
HC	1.34	663.21
NO_x	0.61	1427.72
CO	27.66	4423.34

TABLE 2-16. 305 ENGINE QUADRATIC POLYNOMIAL REGRESSION

N=826

	No Transformation		log$_{10}$ (emissions)			log$_e$ (emissions)		
	R^2	R		R^2	R		R^2	R
FC	.99	.997						
HC	.56	.75	$\log_{10}$(HC)	.58	.76	$\log_e$(HC)	.58	.76
NOX	.93	.96	$\log_{10}$(NOX)	.98	.99	$\log_e$(NOX)	.98	.99
CO	.77	.88	$\log_{10}$(CO)	.85	.92	$\log_e$(CO)	.84	.92

TABLE 2-17. QUADRATIC POLYNOMIAL REGRESSION: TQ$\geq$25

N=764

	No Transformation		log$_{10}$			log$_e$		
	R^2	R		R^2	R		R^2	R
HC	.60	.77	$\log_{10}$(HC)	.65	.81	$\log_e$(HC)	.65	.81
CO	.79	.89	$\log_{10}$(CO)	.85	.92	$\log_e$(CO)	.85	.92

TABLE 2-18. 305 ENGINE QUADRATIC POLYNOMIAL REGRESSION

Regression Weights N = 826

	FC	$Log_e(NO_x)$	$Log_e(HC)$	$Log_e(CO)$
10^{-3}RPM	0.27029100E+01	0.44466800E+01	-0.73485000E+00	-0.25900000E+01
10^{-1}BMEP	0.60260000E+00	0.15133500E+01	0.43345000E+00	0.36381000E+00
SA	0.51840000E-01	-0.89400000E-02	0.46030000E-01	0.75480000E-01
EGR	-0.25440000E-01	-0.47030000E-01	0.48180000E-01	-0.10730000E+00
AFR	-0.24309300E+01	0.23551700E+01	-0.29072300E+01	-0.58328200E+01
EGR*AFR	0.40600000E-02	0.49000000E-03	0.53500000E-02	0.45000000E-02
$(10^{-3}RPM)^2$	0.65229000E+00	-0.48221000E+00	-0.28739000E+00	-0.13804000E+00
$(10^{-1}BMEP)^2$	0.13760000E-01	-0.58530000E-01	0.78000000E-03	-0.11000000E-02
SA^2	0.27000000E-02	0.43000000E-03	-0.10000000E-04	0.11000000E-03
EGR^2	0.73000000E-03	0.82000000E-03	-0.31600000E-02	0.18500000E-02
AFR^2	0.78290000E-01	-0.67380000E-01	0.93790000E-01	0.15864000E+00
10^{-3}RPM*10^{-1}BMEP	0.11768400E+01	-0.10861000E+00	0.12532000E+00	0.82870000E-01
10^{-3}RPM*SA	-0.90520000E-01	-0.71200000E-02	0.16570000E-01	0.14500000E-02
10^{-3}RPM*EGR	0.57330000E-01	0.90300000E-02	-0.20100000E-02	0.12760000E-01
10^{-3}RPM*AFR	0.16439000E+00	-0.99600000E-01	0.11760000E-01	0.19449000E+00
10^{-1}BMEP*SA	0.42000000E-03	0.19200000E-02	-0.21200000E-02	-0.21500000E-02
10^{-1}BMEP*EGR	-0.78600000E-02	-0.77100000E-02	-0.91000000E-02	0.12800000E-02
10^{-1}BMEP*AFR	-0.38720000E-01	-0.20850000E-01	-0.27990000E-01	-0.21940000E-01
SA*EGR	-0.19200000E-02	-0.96000000E-03	-0.30000000E-03	-0.21000000E-03
SA*AFR	-0.96700001E-02	0.13200000E-02	-0.24100000E-02	-0.48400000E-02
COST	0.19683870E+02	-0.22354500E+02	0.26041050E+02	0.57291690E+02

2.5 DYNAMIC MULTI-DIMENSIONAL ALLOCATION TECHNIQUE

Optimization of fuel economy for a given driving cycle while meeting specific emission constraints can be formulated as a resource allocation problem.[6] Basically, this procedure[4] considers the allocation of resources (emissions) among competing activities (speed/torque points) so as to maximize the total return (fuel economy). The optimization procedure is as follows: First, the best obtainable fuel economy at each speed/torque point is tabulated as a function of specified emissions over an evenly spaced grid of emission values. The size of this grid must be small enough in order to obtain accurate results. A multi-stage procedure is then employed which makes use of the best fuel economy data tabulated for two speed/torque points to obtain a tabulation of the best combined fuel economy over a range of grid values for the two points. The procedure is then repeated using the tabulated data at the other speed/torque points to obtain a final tabulation giving the optimal fuel economies for the entire set of speed/torque points. Similar techniques coupled with the use of Lagrange multipliers enable consideration of more than one emission constraint.

In the application of these techniques, caution must be used in selecting a grid increment (the difference between tabulated emission levels) so that residual error is minimized. In our analysis, an emission grid increment of 0.01 grams/mile was generally adequate. Occassionally, a smaller grid increment was used when fuel economies were calculated at low emission levels. Also, the residual error was reduced by summing the actual emissions rather than grid levels at each stage of the process.

Finally, a discrete data base was constructed by permuting the engine control variables through appropriate ranges at each speed/torque point. Spark advance was varied in <u>one</u> degree increments, exhaust gas recirculation in increments of 0.5 percent, and air-to-fuel ratio in increments of 0.25. This results in approximately 30,000 combinations of engine control variables at each speed/torque point.

The limited number (N=62) of points at TQ<25 is not sufficient to measure the severity of the problem of finding a common analytical model to provide separate expressions for 2-fitting regions.

2.6 COMPRESSION RATIO

A potential area for improvement in gasoline engines is the use of high compression ratios, Figure 2-13. The highest value of compressor ratio for optimum brake thermal efficiency depends on engine design and operating conditions. In general, the optimum compression ratio of gasoline engines is probably not higher than around 17 or 17:1.[10] Values up to 15:1 with 98 octane gasoline are currently under investigation with somewhat lower values for low octane fuels.[11] Examples of combustion chamber configurations for high compression ratio operation include, the Heron and the M-May chambers. Both of these systems operate under lean mixtures, and their chambers are configured to enhance high turbulance. The Heron Chamber utilizes a bowl-in-piston combustion chamber and high energy ignition with port fuel injection. The M-May system incorporates port injection upstream of the inlet valve, and maintains combustion which allows the engine to operate under extended lean limit conditions at part load. Optimum configurations have not been fully realized for both designs.

A major limitation of operating engines under higher compression ratios is engine "knock". Knock in a gasoline engine is caused by the sudden auto-ignition of the mixture in the end gas regions. A short flame travel (i.e. compact chambers) high flame speed, and lean mixtures are well known factors which tend to reduce knock. A retarded engine spark setting also contributes towards reducing knock as well as dilution with recirculated exhaust gas. However, recirculated exhaust gas in the main cylinder chamber does not preclude misfire. Turbulence on the other hand plays a dual role. First, it promotes more complete mixing of the fuel and air within the bulk gas, thus enables a faster moving flame front to be established which comes in contact

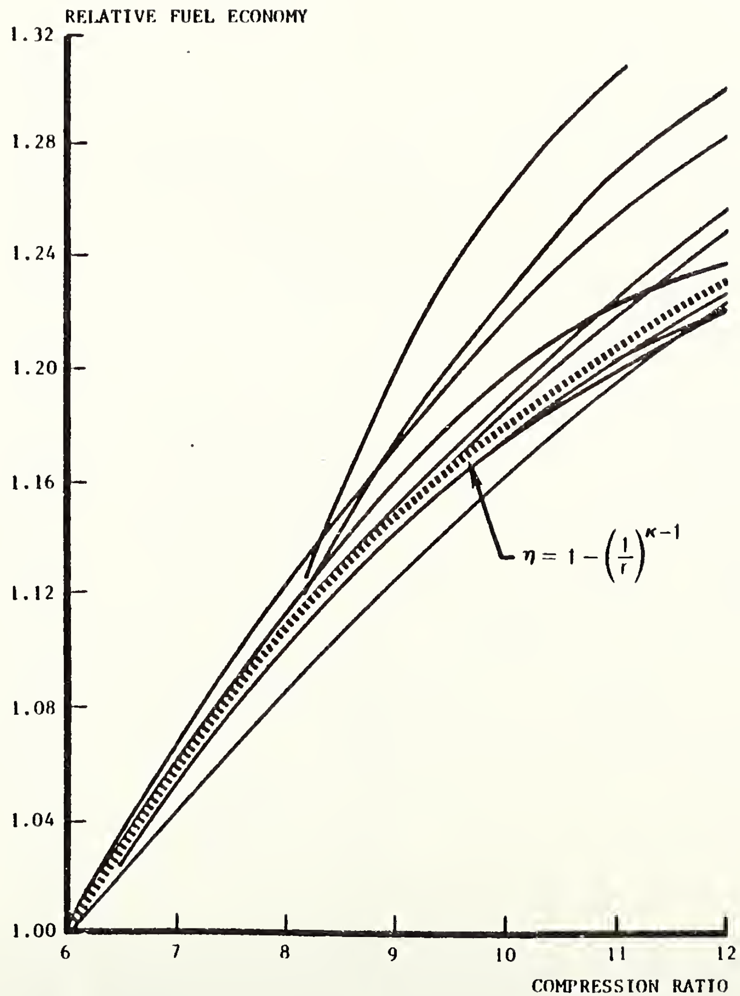


FIGURE 2-13. RELATIVE FUEL ECONOMY VERSUS COMPRESSION RATIO

with the quench layer (surface end gas) before auto-ignition can occur. Secondly, turbulence tends to reduce the thickness of laminar sub layers, resulting in an increase in heat loss and subsequent cooling of the air-fuel charge upstream of the flame front. This action prevents knock. The M-May system is an interesting design since it prevents the existence of the end-gas under normal engine operations by introducing very high squish to scour the walls of the cavity.

Engine knock is inversely related to the octane rating of the gasoline. The octane requirement of an engine is a complex factor of many design and operating factors.[12] Chief among these factors are compression ratios, spark timing, intake pressure and temperature, cylinder size and engine use due to deposit accumulation. The latter causes the octane requirements to increase. The influence of spark timing on the octane requirement of an engine is illustrated in Figures 2-14 and 2-15. Note that a fuel of higher octane number is required under advanced engine timing conditions. Turbocharging has a negative influence on knock at wide open throttle conditions and requires lowering the compression ratio, Figure 2-16. Use of gasoline of higher octane rating than the one required by the engine does not improve the efficiency of the engine. Generally, changes in engine design, for example, increased compression ratios, advance in spark timing, and turbocharging, unattended by changes to other combustion system design and operating parameters, may require fuels with antiknock, or octane ratings higher than those of the current unleaded fuels, 90 RON. Otherwise, the engines on present fuels may well have to be designed for a lower compression ratio and, thus, their fuel economy will no longer be better than those of current engines.

A "running on" condition sometimes encountered after the ignition is switched off, may be more prevelant with high compression ratio engines. This condition is caused by high under the hood temperatures and lower octane fuels.

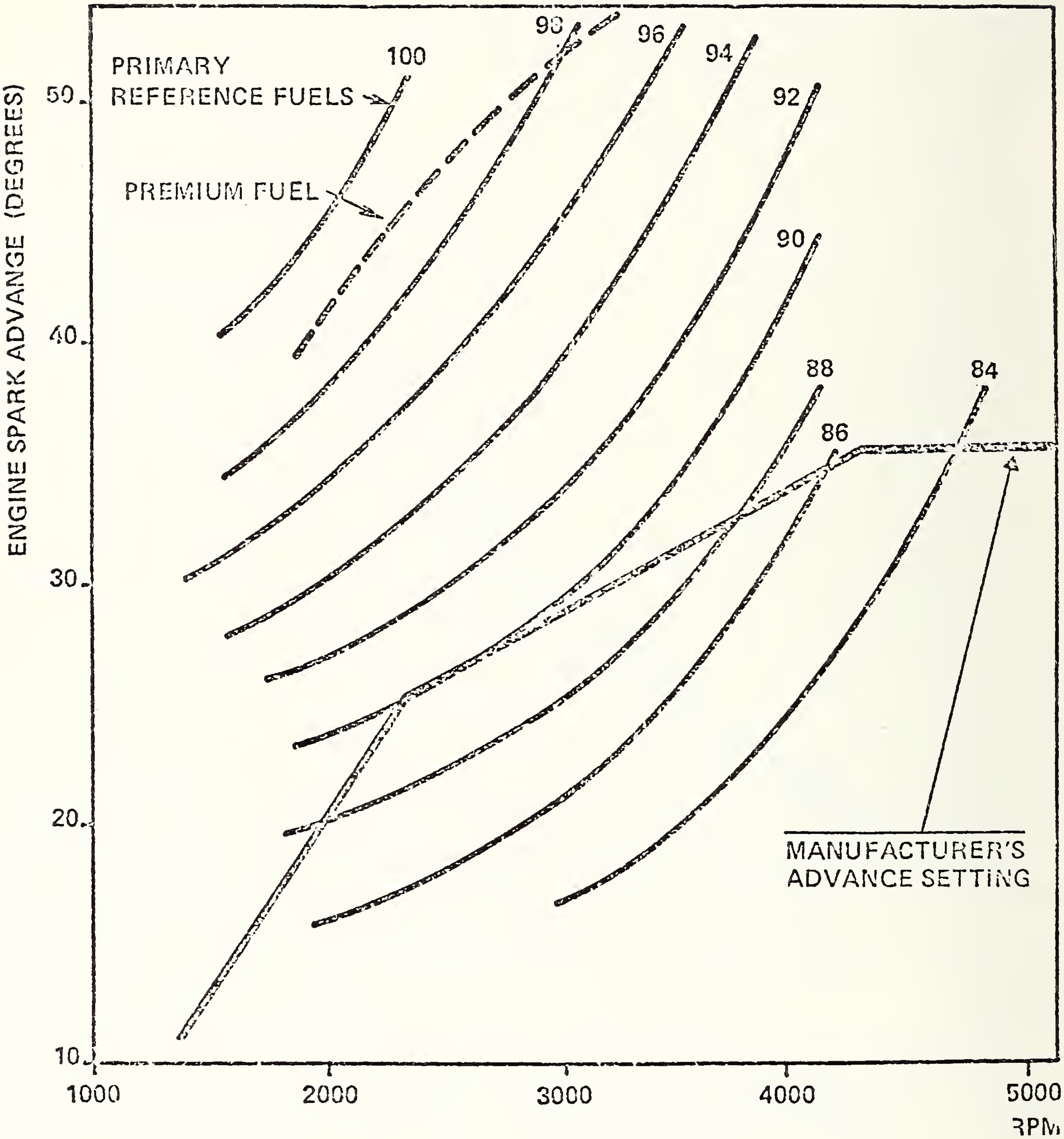


FIGURE 2-14. RPM/ADVANCE LIMIT CURVES

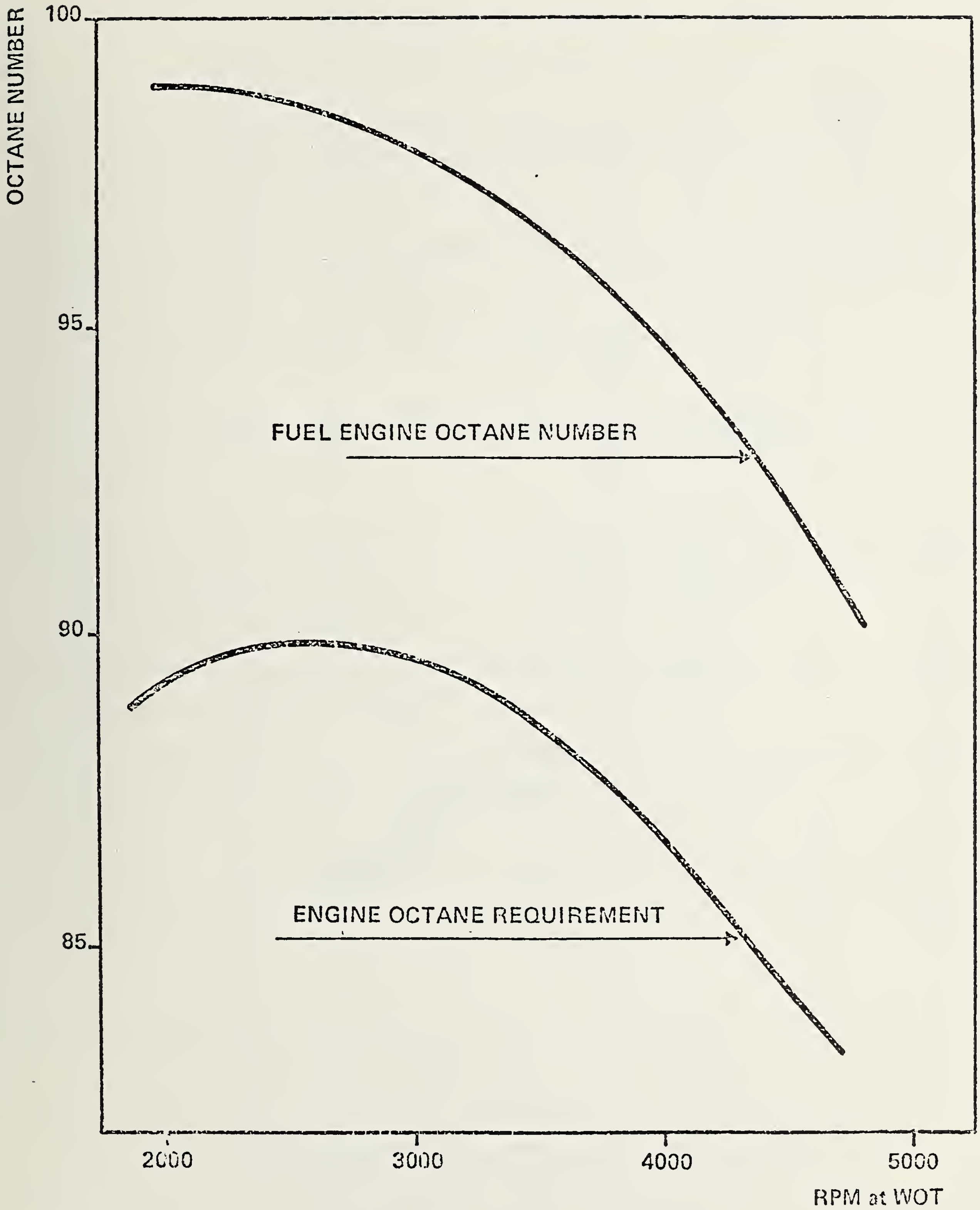


FIGURE 2-15. ENGINE OCTANE REQUIREMENT/NUMBER

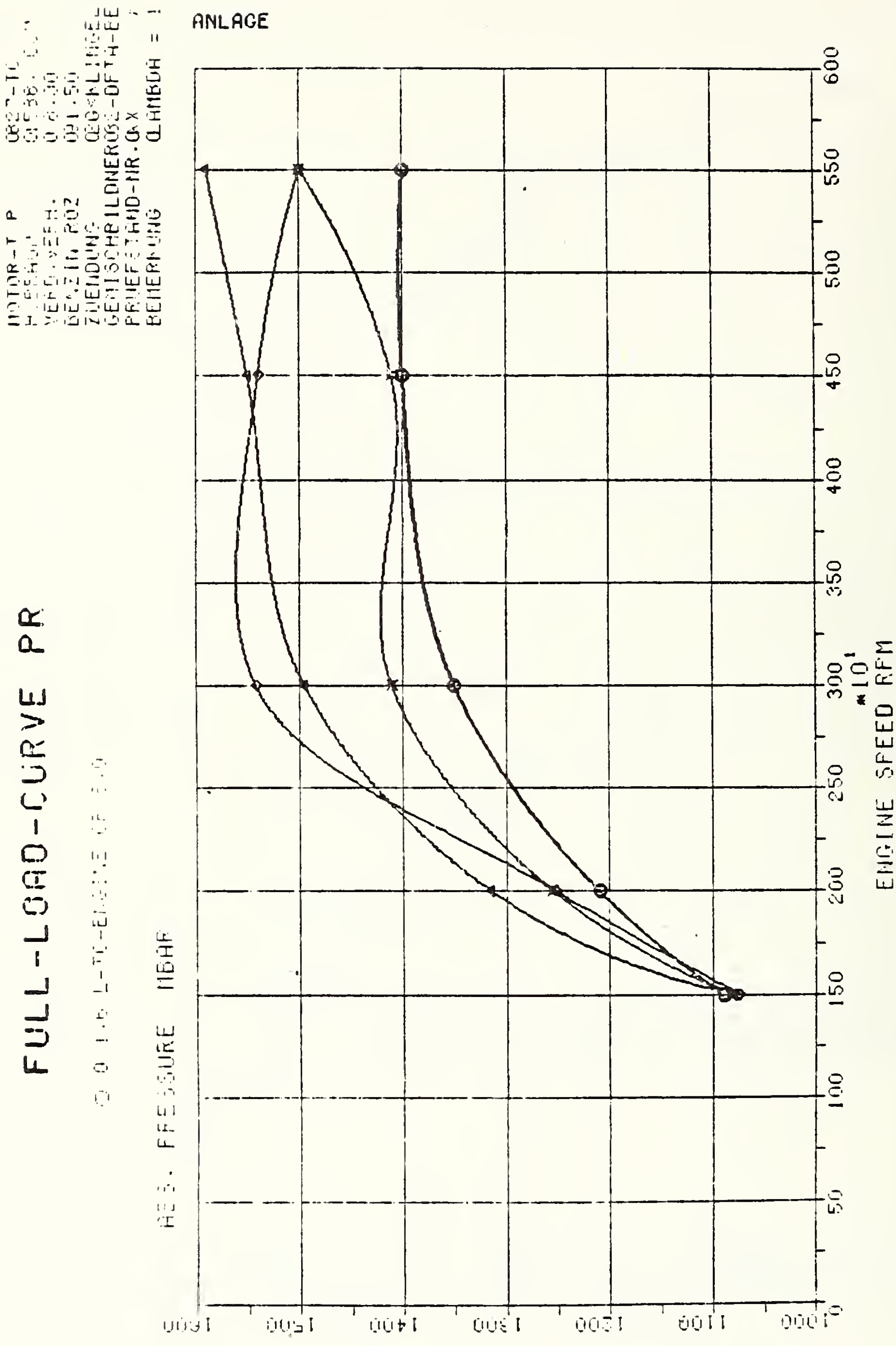


FIGURE 2-16. TURBOCHARGING EFFECT ON KNOCK

REFERENCES FOR SECTION 2

1. J.F. Cassidy, "A Computerized On-Line Approach to Calculating Optimum Engine Calibrations," General Motors Research Laboratories Report, SAE 770078.

2. L. Vora, "Computerized Five Parameter Engine Mapping," General Motors Research Laboratories Report, GMR-2291R.

3. E.A. Rishavy, et al., "Engine Economy with Emission Constraints," SAE 770075.

4. J.F. Auiler, et al., "Optimization of Automotive Engine Calibration for Better Fuel Economy - Methods and Applications," SAE 770076.

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

6. Bellman and Drayfus, "Applied Dynamic Programming," Princeton University Press, 1962.

7. Z. Mencik and P.N. Blumberg, "Representation of Engine Data by Multi-Variate Least Squares Regression," SAE 780288.

8. Performance Characteristics of Automotive Engines in the United States, U.S. Department of Transportation, Transportation Systems Center, Report No. DOT-TSC-NHTSA-78-3, April 1978.

9. American Motors Corporation Data Source.

10. Bascumance, J.L., and Stahman, R.C., "Impact of Gasoline Characteristics on Fuel Economy," SAE 780628.

11. Wheeler, R.W., "Gasoline Combustion--Past, Present and Future," Ricardo Consulting Engineers Report, March 1978.

12. Taylor, C.F., "The Internal Combustion Engine in Theory and Practice," Volume II, Chapter 2, MIT Press, 1968.

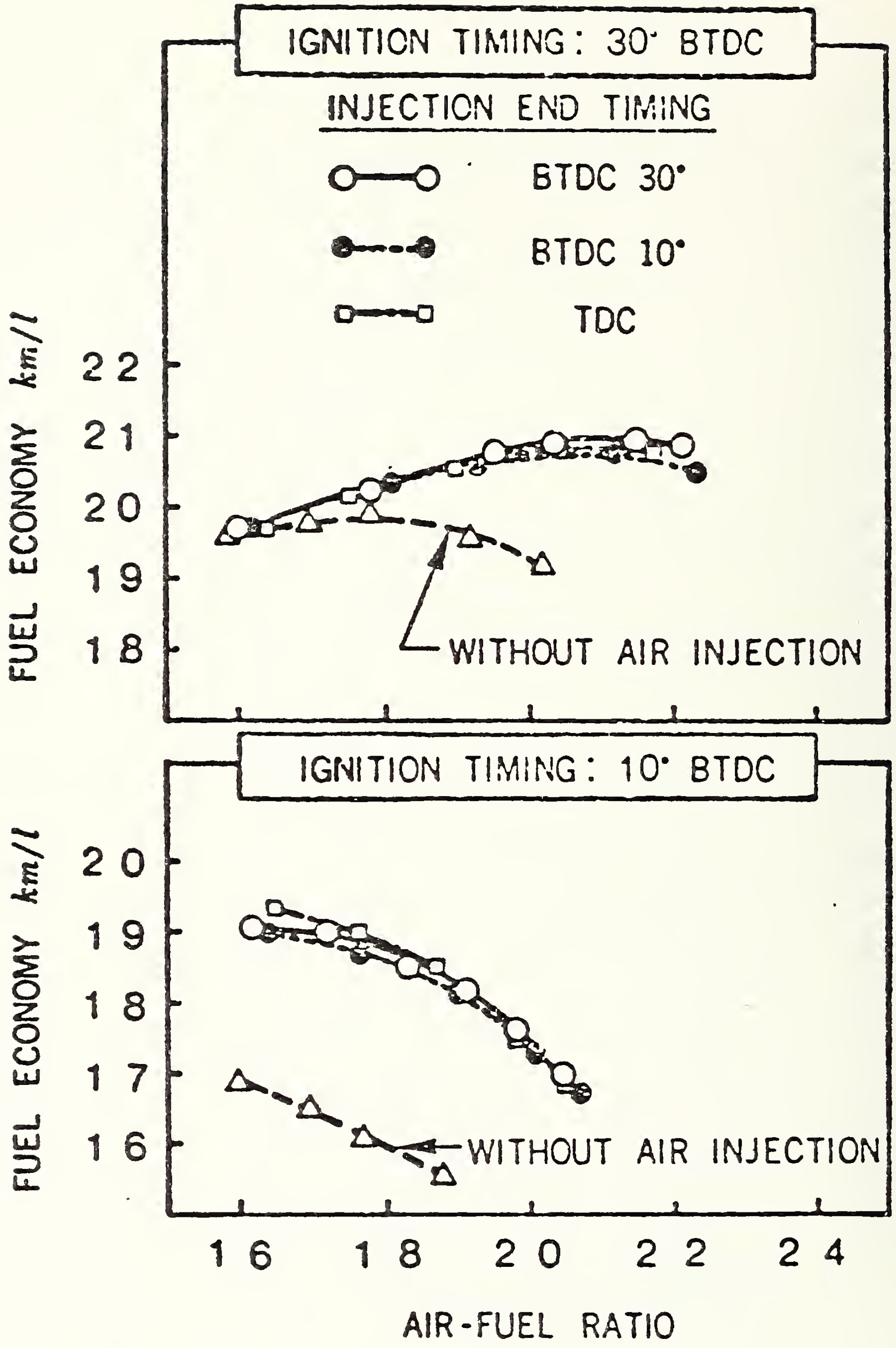
3. COMBUSTION SYSTEMS

3.1 COMBUSTION WITH JET STIRRING

To improve the combustion efficiency in terms of both fuel economy and emission control, one method is to increase the turbulence level of the fluid in the cylinder. Investigators from Mitsubishi Motors Corp. designed a "jet-stirrer" to achieve the above goals,[1] emphasizing in particular, the fuel economy. Their basic idea was induced from some preliminary results about the effects of auxilliary air jets on combustion efficiency (reflected by improvement of fuel economy) and combustion stabilization (reflected by the evolution of stable combustion limit). Figure 3-1 shows some preliminary results, performed on a 4 cylinder jet piston engine (Figure 3-2) with swept volume, 1600 cc., and compression ratio of 8.5 to 1. The results in Figure 3-1 clearly indicate the improved fuel economy and extended stable combustion limit towards the lean air-fuel ratio. The injected air is not of a rich air-fuel mixture, which, therefore, leads them to believe that the jet air gives turbulence instead of rich mixture stratification to the main cylinder mixture and, thus, affects the combustion. Based upon these preliminary results, Mitsubishi built their MCA-JET combustion system, as shown in Figure 3-3. With this engine, they conducted a systematic investigation. The test independent variables included ignition timing, idling conditions, EGR, jet air flow rate, jet orifice diameter, and jet flow direction. Figure 3-4 shows results of two different sets of ignition timing. The early ignition results (injection ended at 30° BTDC) appear to take greater advantage of the injection-induced turbulence effect.

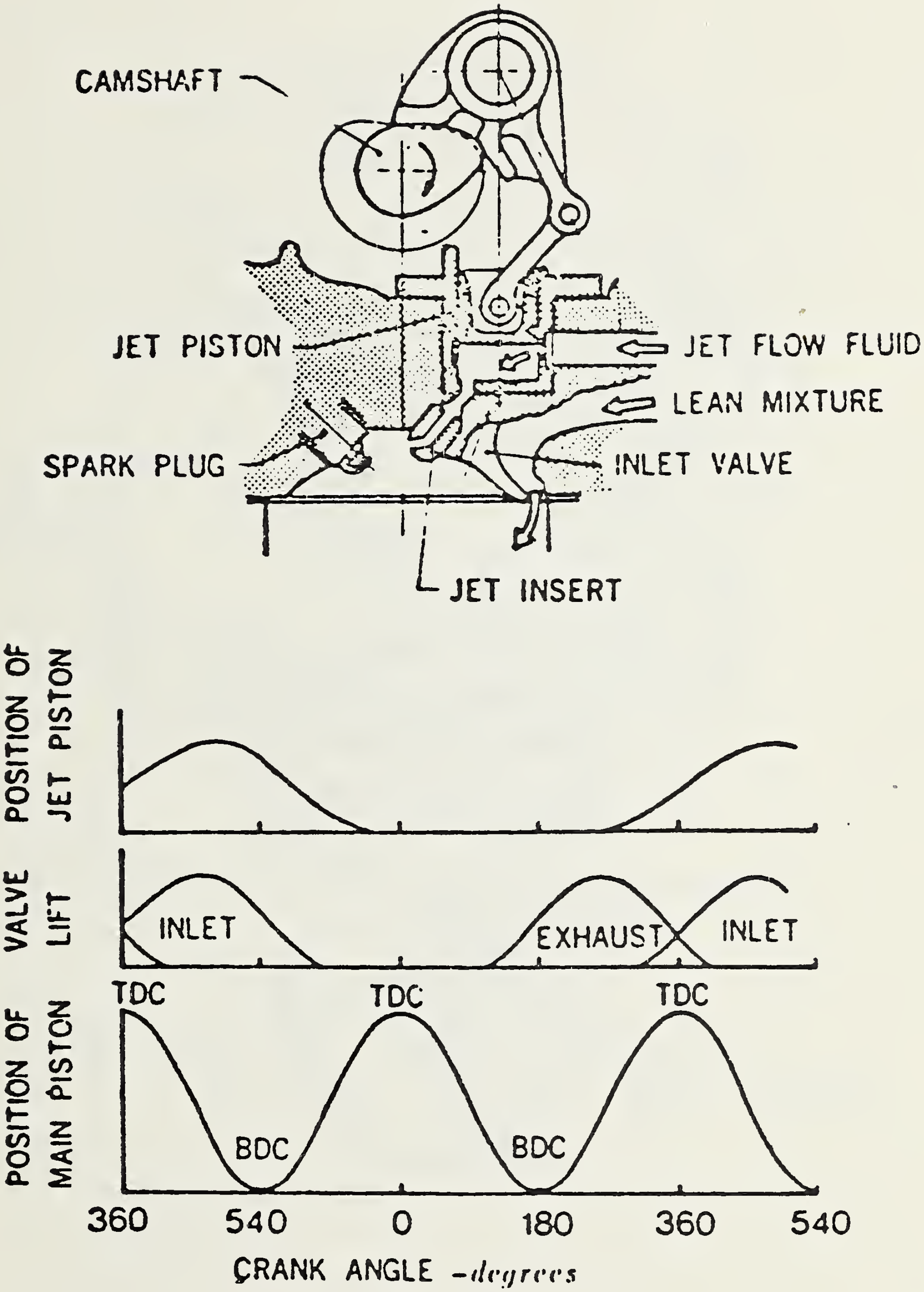
Figure 3-5 shows the fuel consumption results at two different sets of idling conditions. It is clear that at high idling condition, the benefits of the air jet effect are greater.

Figure 3-6 shows the effects of jet air flow rate on fuel economy with and without EGR. As it is expected, as air jet flow rate increased, the improvement of fuel economy is greater because of the increasing level of turbulence.



Source: Reference 1

FIGURE 3-1. FUEL ECONOMY AS A FUNCTION OF JET INJECTION TIMING AT 40 km/h ROAD LOAD (4.3 PS/1430 rpm) AND A JET AIR FLOW RATE OF 1.0 L/s



Source: Reference 1

FIGURE 3-2. CONCEPT OF THE JET PISTON ENGINE

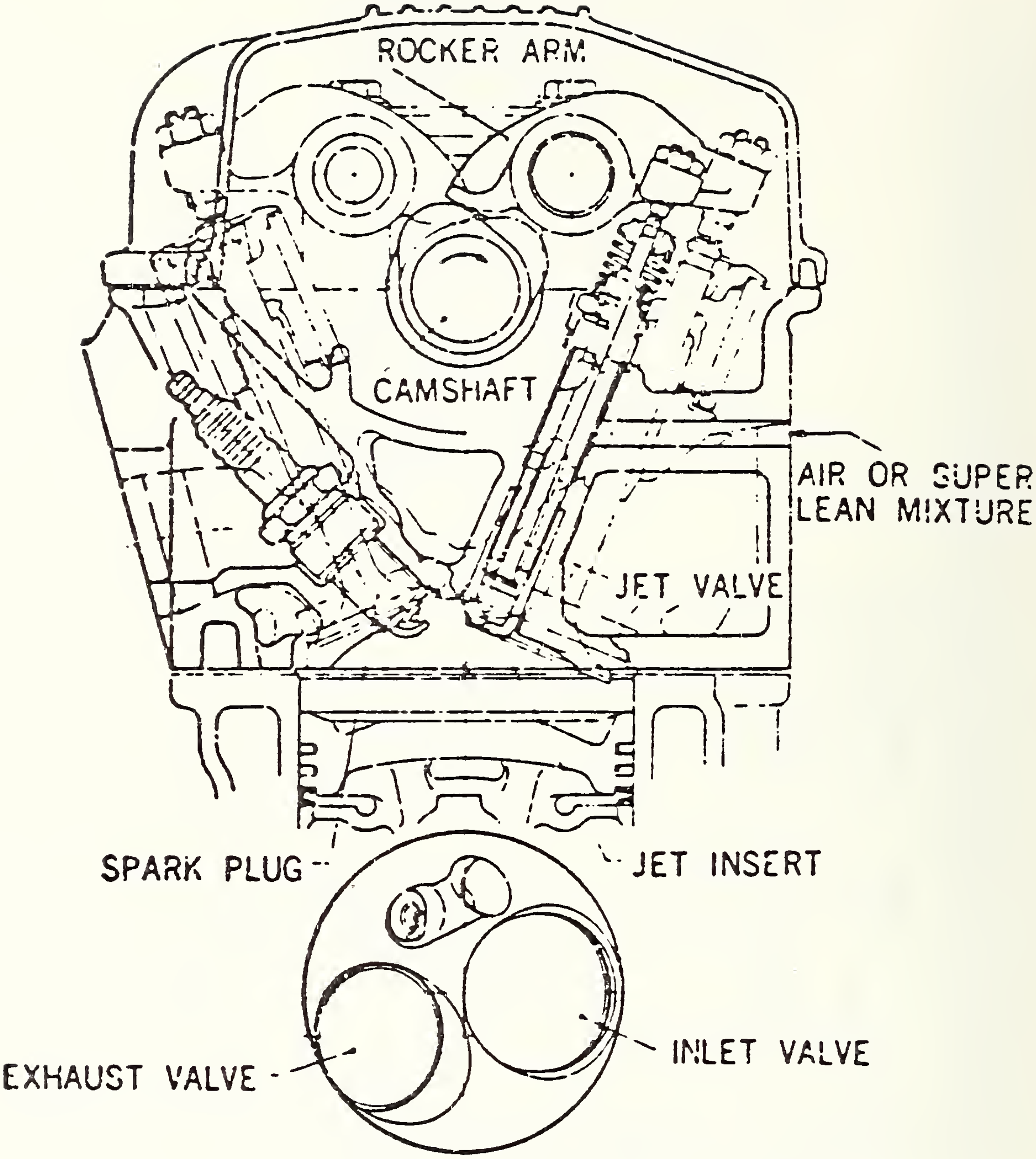
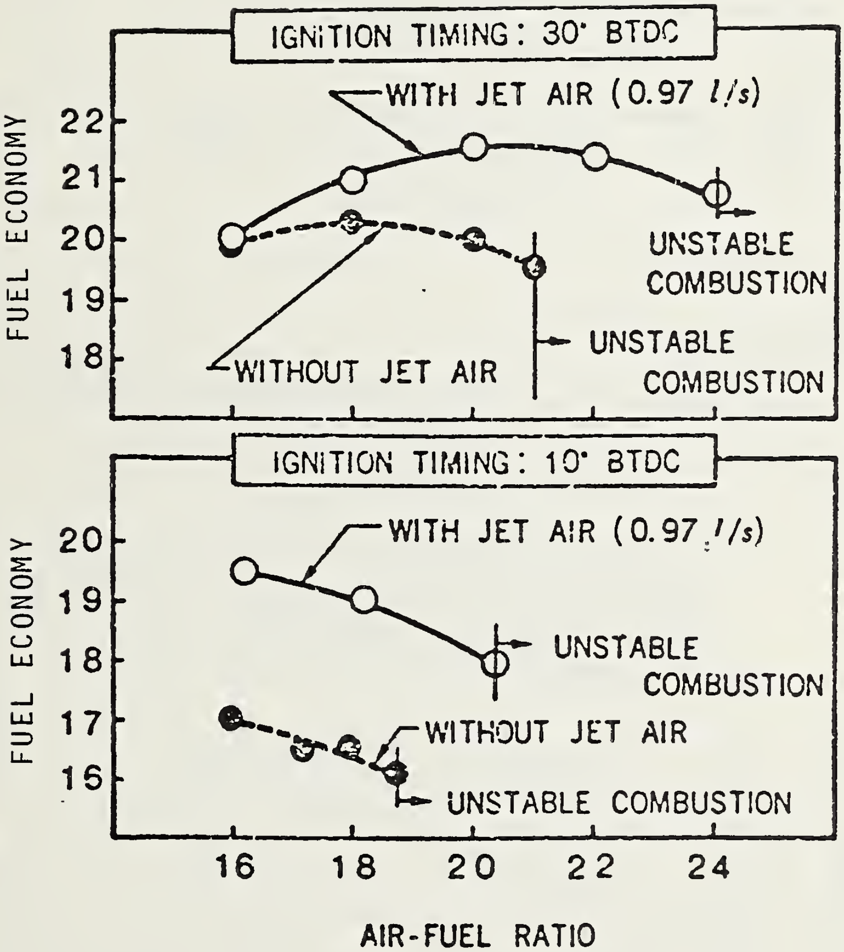
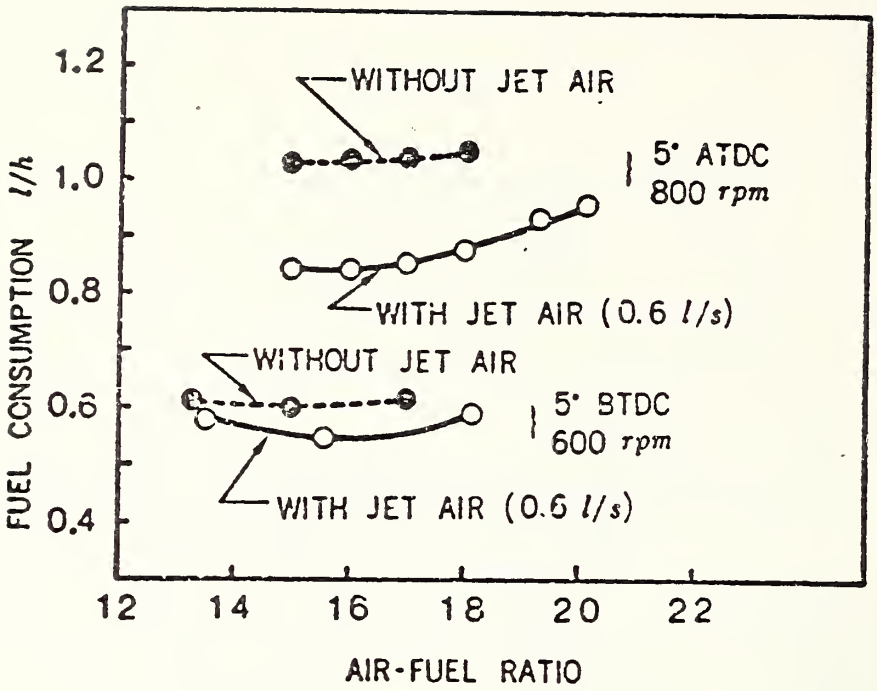


FIGURE 3-3. CYLINDER HEAD CONFIGURATION FOR THE MCA-JET COMBUSTION SYSTEM



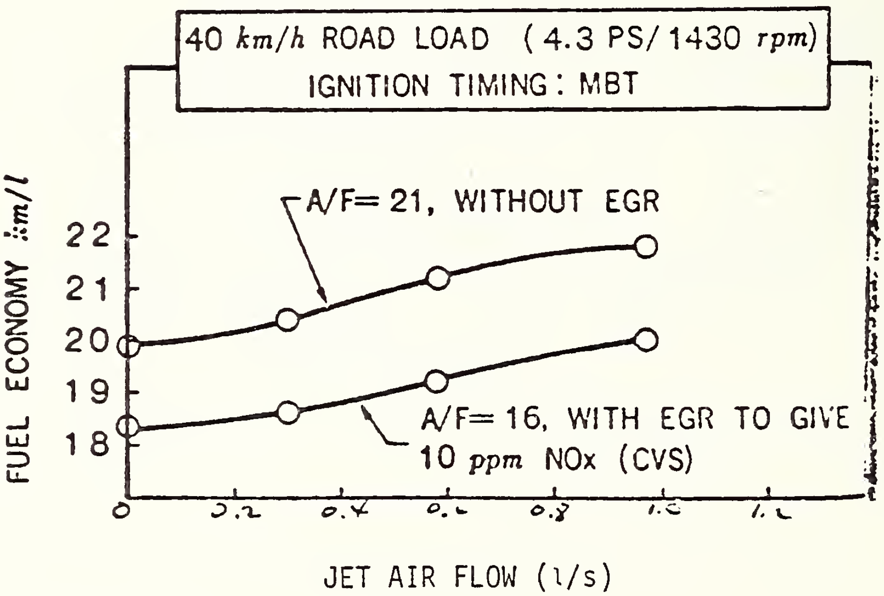
Source: Reference 1

FIGURE 3-4. FUEL ECONOMY AT 40 km/h ROAD LOAD (4.3 S/1430 rpm)



Source: Reference 1

FIGURE 3-5. FUEL CONSUMPTION AT IDLING



Source: Reference 1

FIGURE 3-6. EFFECTS OF EGR ON FUEL ECONOMY

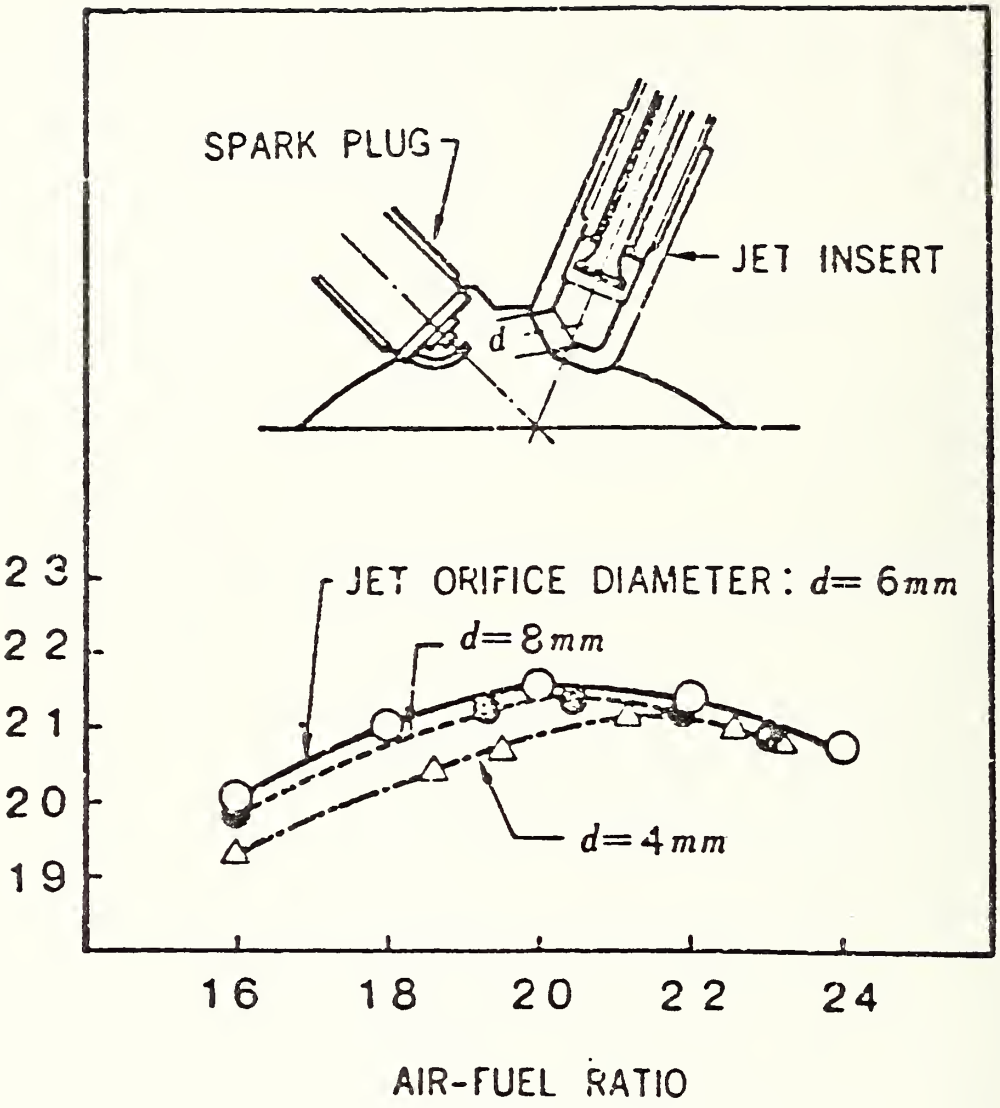
The effect of jet orifice diameter is shown in Figure 3-7. It is observed that there is an optimized orifice diameter (for this set of data, it is 6 mm), at which the fuel economy improves most. It may be correlated with the maximum effective injecting momentum.

Figure 3-8 shows the effect of jet flow direction on fuel consumption. It can be observed that for the jet direction nearest towards the wall, as expected, the improvement of fuel economy is least. For that direction, the jet momentum may have consumed the most.

In summary, Mitsubishi demonstrated an idea: By using the auxilliary air jet concept, turbulence may be stirred up near the spark plug region, so the fuel economy and combustion stability near a lean limit can be improved. Yet, it has also been observed that the absolute NOx level will be increased to a certain extent. The unwanted by-effect may be overcome by using converters, or, because of the special definitions of fuel economy and emission requirements, significant improvement of fuel economy may overcome the slight increase of absolute emission levels and, thus, cause the total emission levels of the vehicle to decrease.

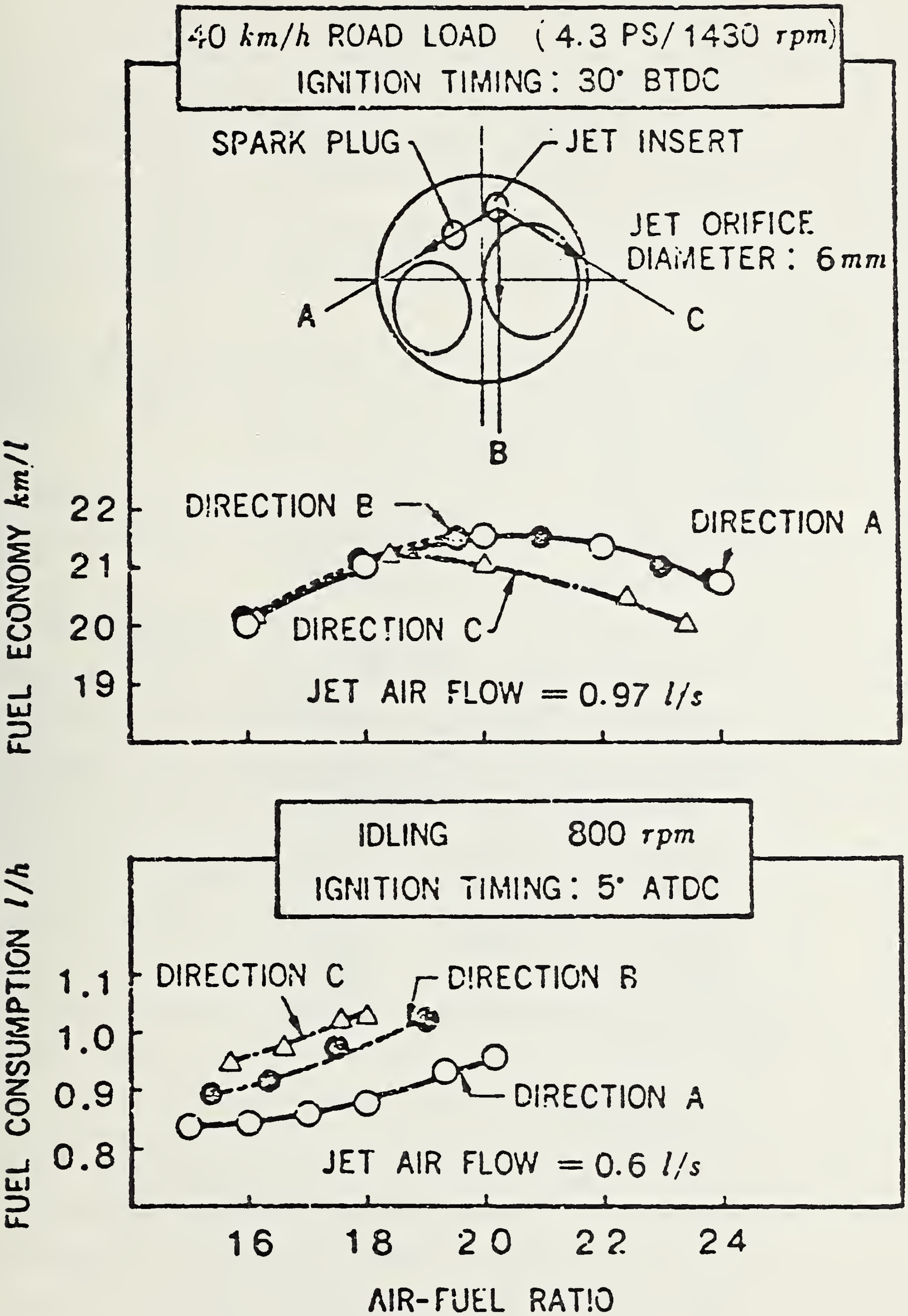
3.2 STRATIFIED CHARGE COMBUSTION

Under the demands of better fuel economy, lower emission levels, and relatively few modifications to the existing automobile engines, the stratified-charge (SC) engine demonstrates high promises toward these goals. The stratified-charge engine differs from the conventinal, homogeneous charge, spark ignition engine in that the former provides a controlled non-uniform distributed air-fuel mixture with rich mixture in the region of spark plug to facilitate the ignition of the compressed mixture in the cylinder. The overall air-fuel ratio of the SC engine may be leaner than the corresponding conventional SI engine, although this is not necessarily the case[2]. Because of the non-uniform distributed charge and system lean feature, better fuel economy and lower emission levels are ordinarily obtained. The efficiency gains may result



Source: Reference 1

FIGURE 3-7. EFFECT OF JET ORIFICE DIAMETER ON FUEL ECONOMY AT 40 km/h ROAD LOAD (4.3 PS/1430 rpm), AN IGNITION TIMING OF 30 DEG BTDC AND A JET AIR FLOW RATE OF 0.97 L/s



Source: Reference 1

FIGURE 3-8. EFFECT OF JET FLOW DIRECTION ON FUEL CONSUMPTION from the improved completeness of combustion. The emission reduction results from minimizing the simultaneous presentation of high temperature and excess oxygen in the cylinder and the scarcity of fuel gas presented in the boundary layer region. Furthermore, due to the special stratified distribution of fuel, the fuel concentration in the end-gas is often low, which greatly reduced the knock tendency in the end-gas region. Therefore, many SC engines possess multifuel capability.

Currently, two types of stratified-charge engines are under development, the open chamber and divided chamber types. A comprehensive, descriptive type report is available.[3]

The open chamber engines use fuel injection and air swirl to stratify the air fuel mixture. The most developed examples are the Ford PROCO program and the Texaco TCCS program.

A typical PROCO engine cylinder is sketched in Figure 3-9. A bowled piston with a cavity on its face and a flat cylinder head are used. A fuel injector is seated in the center of the cylinder head. Fuel is injected in a controlled wide angle, low penetrating, conical spray during the compression stroke, and is timed to permit sufficient vaporization and pre-ignition mixing. The spark plug is positioned with its gap just above the spray, near the bore center line. The air swirl is induced as it passes through the intake port. The swirl rate is about 3-5 times the engine speed. Qualitatively, the engine processes may be percieved as follows: As fuel is injected into the center of the chamber, during the pre-ignition period, the relatively heavier fuel substance is carried away, both circumferentially and radially, by the fast swirl air stream to accelerate the evaporation and mixing processes. However, after combustion starts in the central region of the combustion chamber, the strong buoyancy effect retards the rate of spreading of the combustion wave towards the cylinder wall region. Test data reveal improved fuel economy and lower emission. Table 3-1 shows some of their PROCO Vehicle test data and the comparison with the conventional vehicle test results.

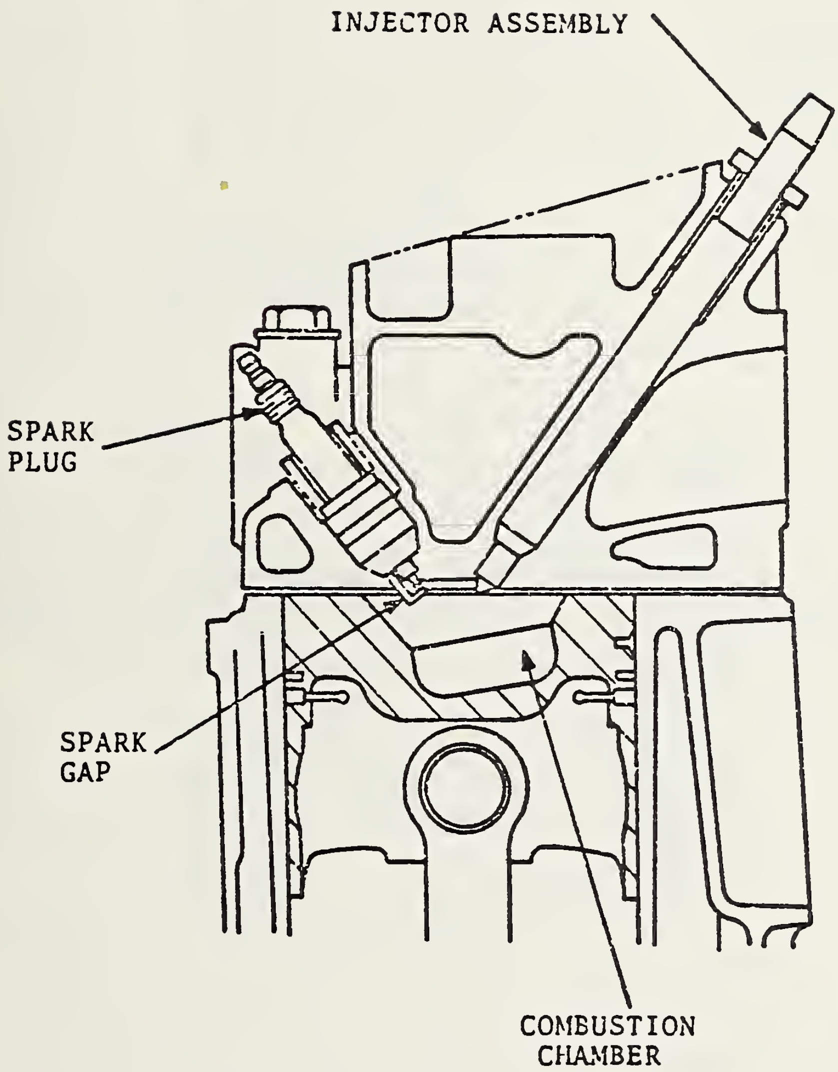


FIGURE 3-9. TYPICAL FORD PROCO ENGINE CYLINDER

TABLE 3-1. FORD VEHICLE TEST DATA

M-151 Vehicle	EMISSIONS gm/mile			Fuel Economy MPG	No. of Tests
	HC	CO	NO_x		
Conventional	4.55	41.6	4.4	17.2	3
PROCO, W/o Catalyst	2.60	13.45	0.32	21.7	1
PROCO, With Catalyst	0.35	1.01	0.35	21.3	2
PROCO, With Catalyst (EPA)	0.37	0.93	0.33	-	14

Smoke emission during the single cylinder engine tests showed that smoke levels were an order of magnitude below that of a diesel engine. However, higher smoke concentrations were reported at high loads and high EGR rates.

Figure 3-10 shows a sketch of Texaco TCCS engine cylinder.[4] The TCCS system differs mainly from the PROCO system in that both the injector nozzle aims towards and the spark plug is located at the periphery of the cylinder head. Also, the fuel jet of the TCCS system is oriented in such a way that its horizontal velocity component is in the same direction as the local air swirl. The TCCS piston cavity has a cusp in the center to facilitate the mixing. Because of such difference in injector arrangement and combustion chamber design, and mixing and combustion become completely different between these two engines. In TCCS systems, the fuel jet entrains air into it due to the spray momentum, the mixture is carried away circumferentially by the swirling air, and diffuses inward. Before ignition, due to the density difference between fuel and air, the mixing of the fuel in the periphery region with air in the central region is suppressed by the centrifignl force, so, the pre-ignition mixing is not as strong as PROCO systems. Yet, as combustion starts, the combustion wave spreads faster due to the acceleration caused by the buoyancy force, which results in better fuel economy. In addition, because of the low pre-ignition mixing level, there is hardly any fuel gas in front of the main combustion wave, so the pre-ignition in the end-gas zone is greatly reduced, which results in high knock-resistance and multi fuel

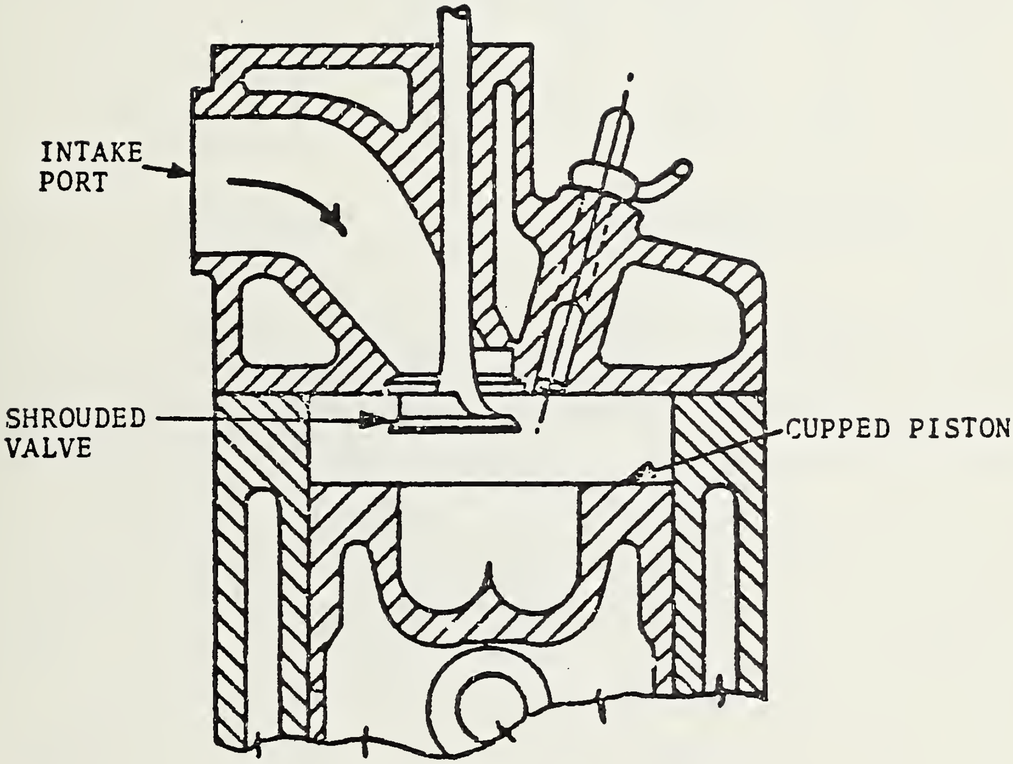


FIGURE 3-10. TEXACO TCCS ENGINE

capability.

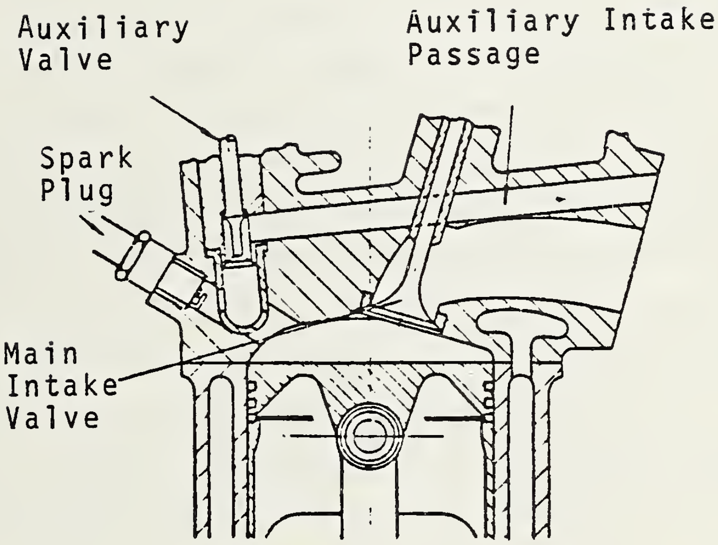
Table 3-2 shows the excellent performance data for both fuel economy and emission levels. These tests are conducted by EPA by using TCSS naturally aspirated M-151 vehicle (3,000 lb. inertia weight) using a catalytic converter with the 1975 Federal Test Procedure.

TABLE 3-2. EMISSION OF A NATURALLY ASPIRATED TCCS L-141 ENGINE

Lab	HC(gm/mi)	CO (gm/mi)	NO_x (gm/Mi)	Fuel Gas (MPG)
EPA	0.4	0.26	0.30	15.3
EPA	0.33	0.15	0.31	16.1
EPA	0.37	0.30	0.31	15.8

The divided chamber concept can be represented by Honda CVCC system.[5] It is the only commercial unit which uses stratified charge principles. The engine, as shown in Figure 3-11, contains a prechamber and a main cylinder. Rich and lean air-fuel mixtures are introduced into the prechamber and main cylinder through separate intake valves during the intake process. The rich mixture is ignited by a spark plug in the prechamber during the end of compression process, and the combustion gas is injected into the main cylinder to ignite the lean mixture through a connecting nozzle between the prechamber and the main cylinder. Honda had conducted a systematic test program for the CVCC system. The emission results are consistently lower than the corresponding conventional results. From those studies, it is also found that the fuel split, expressed by the ratio of prechamber fuel supply to the total fuel supply, is an important parameter. In general, the engine emission levels and fuel economy are both optimized as the split ration is in the range of 25 to 40 percent.

The reason for the emission concentrations of the stratified charge CVCC engine being lower than those of the conventional spark ignition engine is because of the two-chamber charge stratification. Total HC and CO mass concentrations are reduced because the overall mixture ratio is lean, usually leaner than can be



Source: Reference 5.

FIGURE 3-11. HONDA CVCC CYLINDER

readily achieved in the conventional engine. Ordinarily provided that the fuel-air mixture burns to completion without excessive quenching (extinguishing of the flame front near the relatively cold combustion chamber walls), lower HC and CO will result. Complete combustion is enhanced by the turbulence generated as the burning prechamber contents eject into the main chamber.

NO_x emissions are reduced by burning the fuel-air charge at local mixture ratios not favorable to NO_x formation: mixtures richer than stoichiometry in the prechamber create a chemically reducing environment in which nitrogen competes less successfully than carbon and hydrogen for the limited supply of oxygen; lean or dilute mixtures in the main chamber lower the peak combustion temperature to levels less conducive to NO_x formation.

It should be clear that because the total mass of emissions are measured by the FTP, and engine demand is in terms of instantaneous power, the proper emissions criteria for an engine is not specific emissions (i.e. grams/horsepower-hr) alone. An engine which has low specific emissions but low efficiency as well, will not produce low mass emissions because of its greater fuel usage.

To continue a little further in this degression, note also that a vehicle which consumes less power in the FTP (i.e. a smaller vehicle) will, all other things being equal, produce fewer emissions. This rather obvious point is often unintentionally (or intentionally) omitted or glossed over in statements concerning the ability of a particular engine to meet emission standards.

The above points are particularly germane to the Honda CVCC engine. It is acknowledged in principle to be less efficient than the conventional open chamber spark ignition engine. It is also acknowledged to have lower emission mass concentrations. What is not clear is whether there are configurations of this engine which are superior to the conventional engine (with its complement of emission control devices), and, if so, what emission levels and in what size vehicle. Evaluation of published data does not lead to unambiguous conclusions.

For example, the 1978 EPA Buyers Guide data shows the CVCC powered vehicles to be better than most, but not all, conventionally powered vehicles in their respective inertia weight classes. Also, attempts to scale up CVCC engines into larger V8 engines has produced lower fuel economy at comparable performance and emission levels than has been attained with conventional engines. Certainly, the way of "scale up" is worth a lot of careful study.

If there were more experimental data comparing the two types of engines it might be possible to demonstrate by example the relative superiority of one over the other. But, from an assessment viewpoint, such evidence would only apply to the actual hardware tested and not the underlying principles of the engine design. Certainly, the design of the hardware may well be guided by the optimized design criterion obtained by careful study of the basic principles. A few generalizations may have use in lending perspective to the issue: with no emission constraints, the conventional engine is more efficient; two chamber stratified charge engines can burn dilute mixtures better than open chamber homogeneous charge engines and can burn a given mixture ratio with less NO_x emissions than conventional engines. Thus, the more stringent the emissions requirement, (i.e. the lower the emission standard and the greater the vehicle weight) the better, by comparison, will be the stratified charge engine. Note that emissions requirements imposed on the engine may be substantially reduced by exhaust after-treatment (e.g. catalysts).

REFERENCES FOR SECTION 3

1. Nakamura, H., et al, "Development of a New Combustion System in Gasoline Engines," SAE Paper 78000.

2. Stephenson, R.R., "Should We have A New Engine?," Vol. II, Technical Reports, Jet Propulsion Lab., California Institute of Technology, 1975.

3. Withjack, Eric M., "Stratified Charge Engines," Final Report, January 1976, Report No. DOT-TSC-OST-75-56.

4. Mitchell, E., Alperstein, M., "Texas-Controlled-Combustion System-Multifuel, Efficient, Clean and Practical," Combustion Science and Technology, 1973, Vol. 8, pp 39-49.

5. Inone, K., Ukawa, H., Fujll, I., "Fuel Economy and Exhaust Emissions of Honda CVCC Engine," Combustion Science and Technology, 1976, Vol 12, pp 11-27.

4. ENGINE CONFIGURATIONS

4.1 ROTARY ENGINES

Although rotary engines offer a high speed advantage and weight and volume savings of 10-20 percent and 30-40 percent respectively,[1] compared to conventional piston engines, production vehicles with these engines until recently have not demostrated fuel economies comparable to vehicles with piston engines. This is attributed in part to the emission requirement and the rotary engine's combustion chamber design, and configuration.

Recent improvements in fuel economy have been achieved, however, on a number of production engines.[1] These improvements, are the result of a) improved gas sealing designs, modification of the combustion chamber and spark plug, b) port timing, c) the use of leaner air-to-fuel ratios with increase in the spark advance, and d) design improvements in thermal reactors.

Studies are in progress to improve fuel economy and the emission characteristics of the rotary engines. These studies are directed towards:

1. promoting faster combustion
2. reducing friction losses
3. leaner mixture operation
4. reducing heat losses
5. modifying exhaust emission after-treatment controls to burn lean mixtures.

4.1.1 Spark Plug and Combustion Chamber

The arrangement of the spark plugs and the shape and location of the combustion chamber recess are important to improve the engine's thermal efficiency. In theory, it is desirable to have a plural number of spark plugs disposed along the rotational direction since the combustion chamber moves in the rotational direction and is long and narrow. In practice generally, however, two spark plugs disposed at the trailing side and the leading side can be used. Some improvements in fuel economy were obtained when the leading spark plug was used at part load and both spark plugs were used at high load and high speed, under lean engine operation in combination with an after-treatment device such as a thermal reactor or a catalytic converter.

4.1.2 Improvement of Misfire Characteristics

In the rotary engine, the ignitability of the mixture is more different since the spark plug electrode does not protrude into the combustion chamber. The ignitability or misfire can be attended to by widening the spark plug gap, setting the spark plug closer to the trochoidal surface and increasing the ignition energy.

4.1.3 "Port Fuel Injection"

It is well known that further gains in fuel economy can be achieved by improving the vaporization and distribution of the air-fuel mixture in the manifold to extend the lean limit over the operating range of the engine.[2] Figure 4-1 shows this improvement when a carburetor was replaced with a Bosch K-Jetronic manifold port fuel injection system.

4.1.4 Exhaust Emission

1. MEASURES TO REDUCE NO_x

EGR is used primarily to reduce NO_x. Systems are presently under development to minimize the amount of EGR required at various torque-speed points as a means to control NO_x levels most effectively.

2. MEASURES TO REDUCE HC EMISSIONS

The present disadvantage of rotary engine is still the higher emission of unburned hydrocarbons. Figure 4-2 compares the HC emissions of a 2-liter, 4-cyl. reciprocating engine and a rotary engine as a function of BMEP. While the reciprocating engine

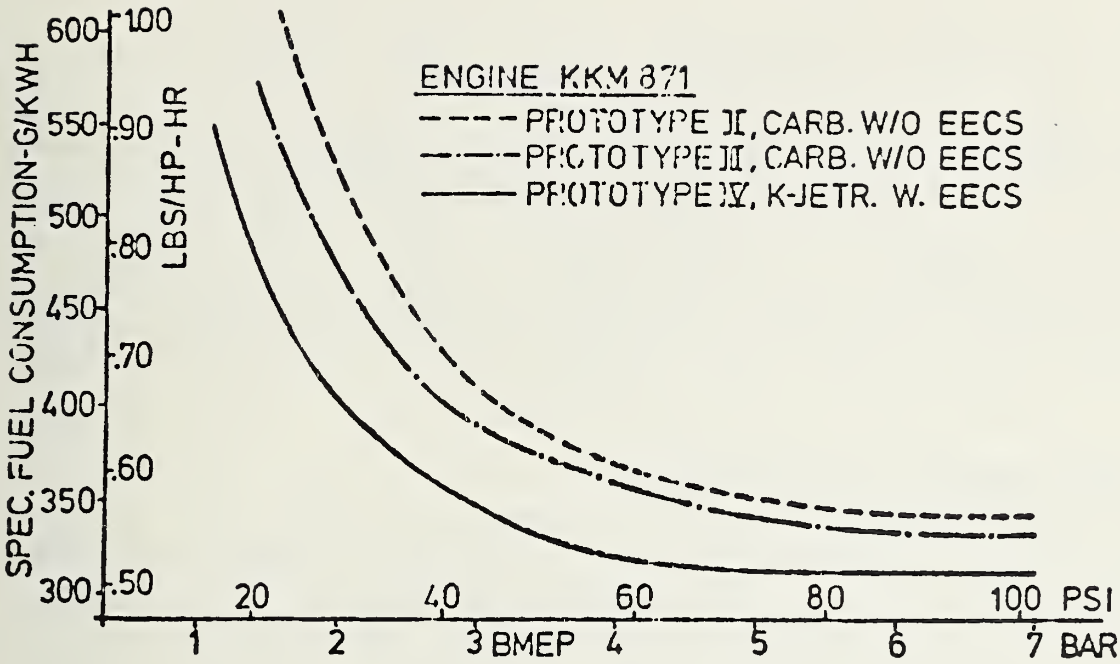


FIGURE 4-1. SPECIFIC FUEL CONSUMPTION OF DIFFERENT KKM 871-PROTOTYPES AT 2000 RPM

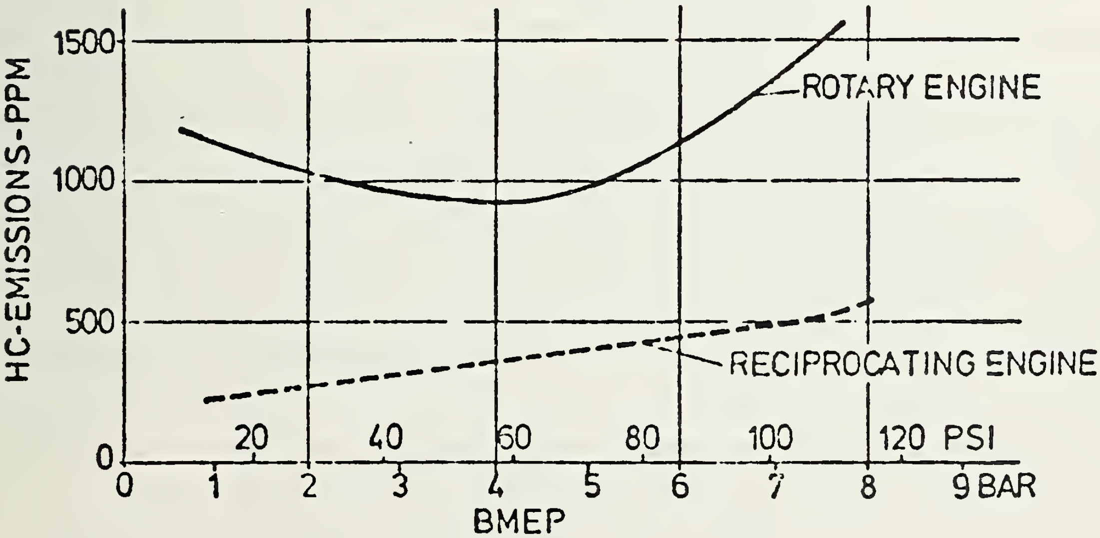


FIGURE 4-2. CHARACTERISTIC OF HC-BASE-EMISSION OF ROTARY AND RECIPROCATING ENGINES DEPENDING ON BMEP AT 2000 RPM

shows a steady increase of HC over BMEP the rotary engine has a definite minimum at the end of medium load range.

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 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.

4.1.5 Improvement in Fuel Economy

Potential for improvement in fuel economy of rotary vehicles include the areas of fuel supply systems, such as (a) EFI and variable venturi carburetor with appropriate induction porting systems; (b) new types of spark plugs and their locations with suitable combustion recesses; (c) methods of injecting secondary air to control exhaust emissions; (d) matching the vehicle with its engine size, selection of the basic engine specifications and the number of rotors.

4.2 ENGINE SIZING

C. F. Taylor developed and demonstrated rules for engine scaling effects. He showed that engines of geometrically similar design had the same performance in terms of BMEP versus piston speed. The implication of his scaling relationship is that multi-cylinder (geometrically similar) engines of the same piston area will have the same power output at the same piston speed. A four cylinder engine rescaled to eight cylinders with the same piston area will require only 0.71 as much displacement and in a "straight eight" configuration would have only 0.71 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 0.6. This amounts to a weight savings of about 100 lbs. in the engine alone, based upon a bare engine weight of 300 lbs. for a four cylinder engine.

An engine rescaled from four cylinders to eight obviously benefits from improved driveability, smoother operation, ease of starting and faster warm-up due to the decreased thermal and mechanical inertia and the greater number of cylinders. 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.

There are, of course, disadvantages to repacking engines to a greater number of cylinders. Obvious ones are increased cost and that scaling beyond eight cylinders gets increasingly complex and less attractive. The HC and CO emissions trade-offs are probably not favorable owing to the increased surface-to-volume ratio of the engines. This will also reduce engine thermal efficiency, off-setting some of the increased efficiency gained from the higher compression ratio. Higher engine speeds will create more noise and wear rates will, in general, be greater, and the effect of a given amount of wear will also be greater. How serious these disadvantages weigh in the overall engine design considerations is not known. But it should be clear that a substantial engine size and weight advantage can be gained through the repacking of engines into configurations with greater numbers of cylinders.

Repackaging of the conventional reciprocating engine into another engine configuration such as the Wankle rotary engine also offers weight and volume advantages. These savings would be similar or greater in magnitude than those as discussed above. However, in the case of the rotary engine, weight and volumetric savings are secondary considerations, unless overall engine efficiency and emissions are at least comparable to those of the conventional engine.

5. TURBOCHARGING AND COMPOUNDING ENGINES

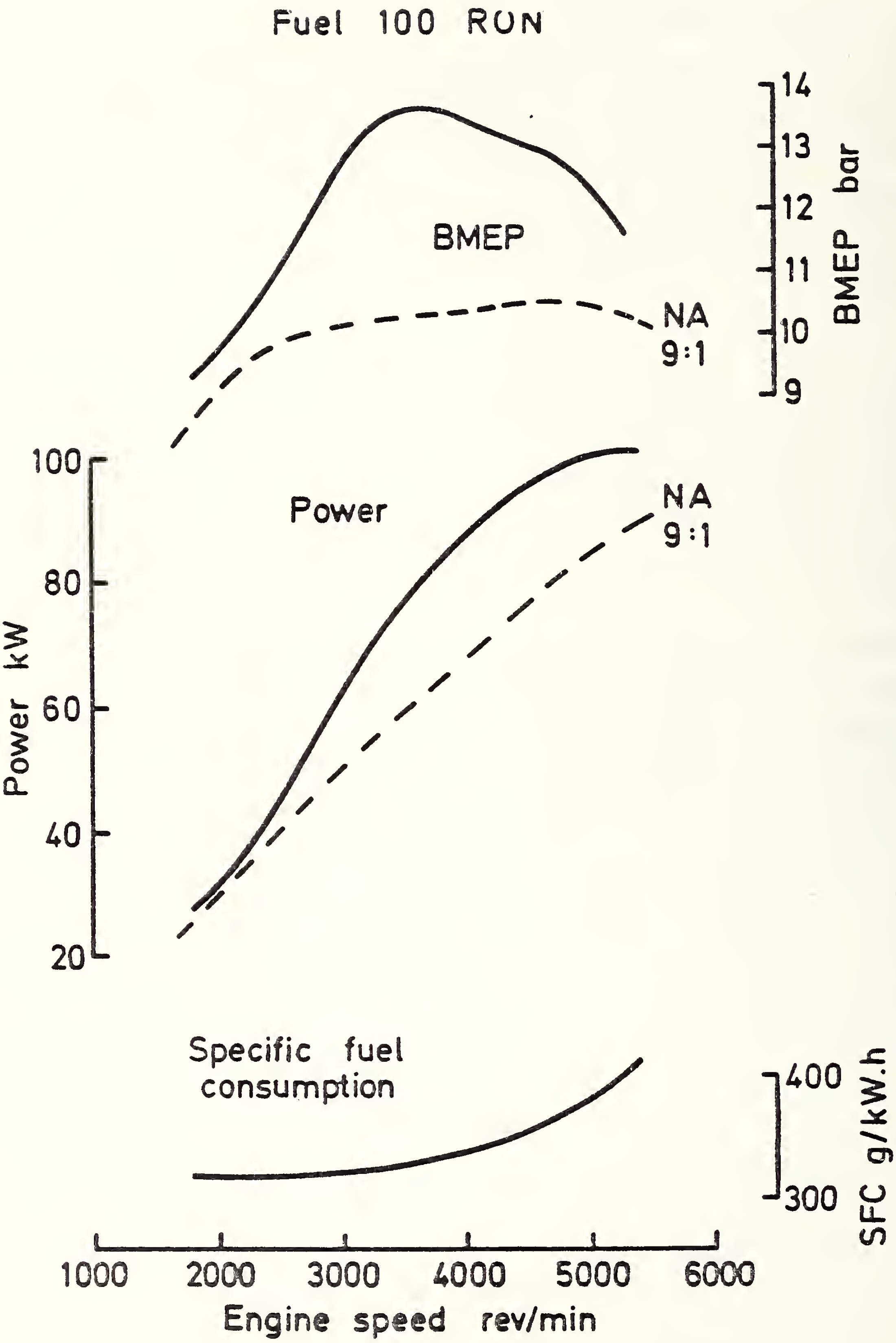
5.1 TURBOCHARGING ENGINES

Turbocharging offers considerable advantages for weight reduction since it is capable of boosting the engine' maximum torque and power, thus permitting the use of a smaller engine for a given application. (See Figures 5-1.) Increased fuel economy results because of lower throttling losses under more highly loaded engine conditions. Small turbochargers are now becoming available for engines down to approximately 1.5 liters in swept volume.

For automotive applications, the turbocharger must be matched to the engine at as low an engine speed as possible, and a waste gate must be employed to spill the exhaust gas from before the turbine to maintain a constant boost pressure at high engine speeds. If a waste gate is not employed excessive boost pressures will be created, resulting in an excessive mechanical loading of the engine.

Problems may arise from turbocharging a gasoline engine, requiring either the use of a higher Octane fuel or the reduction of the compression ratio of the engine to prevent detonation. The latter requirement results in some loss in fuel economy at part throttle operations. Higher compression ratios could be used, provided that auto ignition be suppressed by spark timing retard and the use of weak mixtures when approaching wide open throttle operations or by avoiding wide open throttle operation entirely.

Another problem encountered with turbocharge engines is the response time of the turbocharger under vehicle acceleration demands. Generally a 5 to 7 second lag is experienced during full throttle acceleration (i.e., the time difference when the throttle is depressed and the engine maintains power). Various schemes have been suggested to minimize the time lag. These include: a) reduction of compressor and turbine rotor inertia, b) use of a pelton wheel or burners.



Source: Reference 2.

FIGURE 5-1. TURBOCHARGED ALFA ROMEO 2000 ENGINE W.O.T. PERFORMANCE

Boost limited to 0.5 bar by waste gate

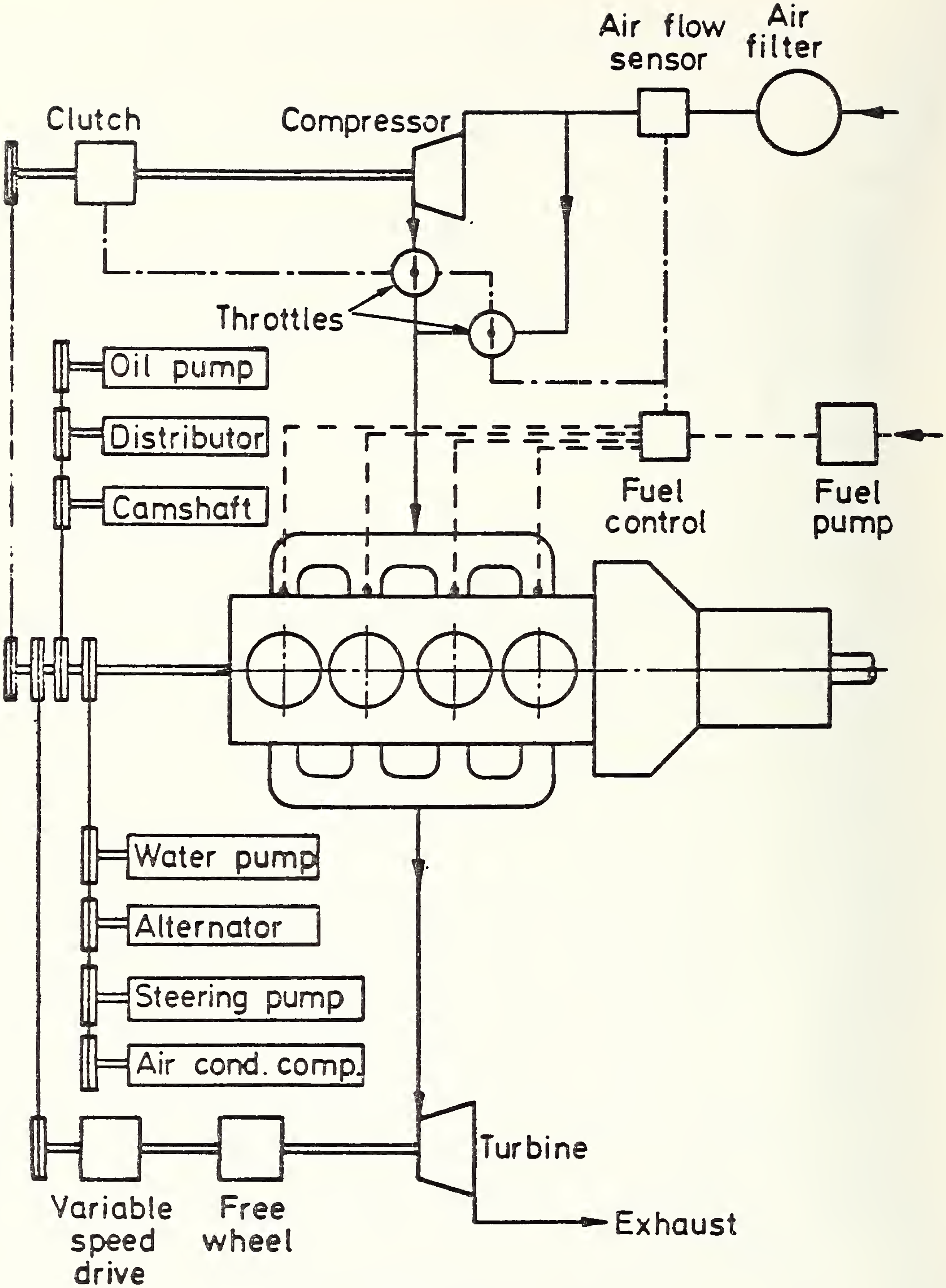
Compression ratio 7.5:1

The 1978 Buick turbocharged V-6 engine is a production example of this technique. This engine uses a knock sensing feedback device to retard ignition timing whenever incipient knock is detected. This largely overcomes the traditional problem of requiring a lower compression ratio with a turbocharger.

In total, when the turbocharged engine is compared to a naturally aspirated engine of equivalent performance the turbocharged engine will be smaller and lighter, more tolerant of EGR or lean mixtures, produce lower HC, CO and NO_x emissions, and provide greater fuel economy.

5.2 COMPOUNDING ENGINES

Compounding is another engine technology which holds considerable potentials for improvements in fuel economy. The basic thrust of compounding places emphasis on the efficient use of the exhaust energy. A number of schemes have been proposed. Of these, the most noteworthy are the Rankine bottoming cycle[1] and the plan in which exhaust energy is directly transfered by mechanical means to the engine drive shaft (turbocompounding). Some limited studies have been performed on the bottoming cycle concept in conjunction with diesel engines for heavy duty applications. However, as of today, this concept has yet to be evaluated for passenger car applications. Basically, the concept makes use of an Organic Rankine-Cycle System (ORCS) consisting of a vapor generator which abstracts and transfers heat from the exhaust of an engine to an organic fluid (Fluornal-50), and a turbine which transfers the removed heat energy to the main drive shaft via a mechanical gear reduction system. On the other hand, some limited design studies have been conducted with turbocompounding a gasoline engine. Figure 5-2 shows a schematic of the concept of turbocompounding. In this system energy is extracted directly from the exhaust turbine and redirected or "inputted" to the main drive shaft via a variable reduction speed drive device. The net effect should enable lowering of the brake thermal efficiency to enable higher power outputs and lower BSFC's to be achieved. However, this does not occur in practice since the efficiency of the exhaust turbine



Source: Reference 2.

FIGURE 5-2. COMPOUND ENGINE SCHEMATIC

is generally lower than the efficiency of the engine. An immediate advantage is also recognized in the turbocompound gasoline engine which includes retarded-timing for NO_x control since exhaust energy can be recovered. Compounding a turbocharged engine also provides a reduction in response time (lag) of the turbocharger during accelerations because of the additional coupling which is provided. Generally this lag time can be reduced by 2 to 3 seconds since the maximum engine torque provided under first throttle accelerations is increased at low engine speeds. One major obstacle of turbocompounding is the development of a high ratio gear reduction system. For smaller engines this ratio becomes larger than that required for larger engines. Although there have been a number of designs proposed, they are in an early stage of development and as yet no practical system has been developed which utilizes this concept. The degree and extent on the design of the gear reduction system and its material requirement.

REFERENCES FOR SECTION 5

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

2. Palmer, R.M., "Turbo-compounding the Spark Ignition Engines," Prepared for Energy Research and Development Administration, Contractor's Coordination Meeting, Ann Arbor, MI. May 6-8, 1973.

6. ENGINE VALVING

One of the fundamental designs compromises of light duty spark ignition engines is air handling or management. Currently, all spark ignition engines are designed for fixed valve timing dwell, and lift, and use a carburetor throttle plate to limit the engine fuel-air charge over their operating range.

Such a design presently brings out two key deficiencies. First, since valve timing is fixed, it is compromised at a mid-speed point which sacrifices both low and high speed performance. Engines in which the valve timing has been optimized for speed applications at either low or high speed, yield a considerable degree of these sacrifices. Complex transmissions help to rectify these sacrifices by reducing the requirement for off-design speed operations. However, they introduce other problems of their own in regard to efficiency, and driveability.

The second deficiency deals with the requirement to throttle the intake fuel-air mixture in order to achieve part load performance. In one instance, the throttling process causes pumping work to be performed in raising the intake mixture from intake manifold pressure; thus, the overall pumping work increases in proportion to the output of the engine 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.

6.1 VARIABLE VALVE CAM

Considerable improvements in performance of a four stroke engine would be expected if the characteristic of the valve timing were to be continually varied at each running condition. This can be accomplished by a variable cam shaft arrangement. The experimental data accumulated for such an arrangement[1] for a variable inlet and exhaust camshaft installed in a Fiat 124 AC 1438 cm^3

engine illustrates the extent of improvement in brake specific fuel and power which can be achieved when the camshaft settings are optimized for maximum engine torque and breathing at wide open throttle.

A variable valve camshaft device also has a large impact on the exhaust emissions, in particular, the reduction of hydrocarbons and carbon monoxides by promoting more complete combustion. This is illustrated in Figures 7-1 through 7-4 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 those of oxides of nitrogen.

6.2 VALVE SIZE

Valve sizing is an important engine design parameter which 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 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. Recent experimental study shows[1] 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 on improved removal of combustion products from the combustion chamber, leaving a lower concentration of residuals during the compression stroke and an intake of air/fuel mixture which, in turn, produces a better combustion efficiency and a lower level of exhaust emissions. It is also possible that the large exhaust valves introduce turbulence into the combustion chambers which influences the subsequent induction and combustion.

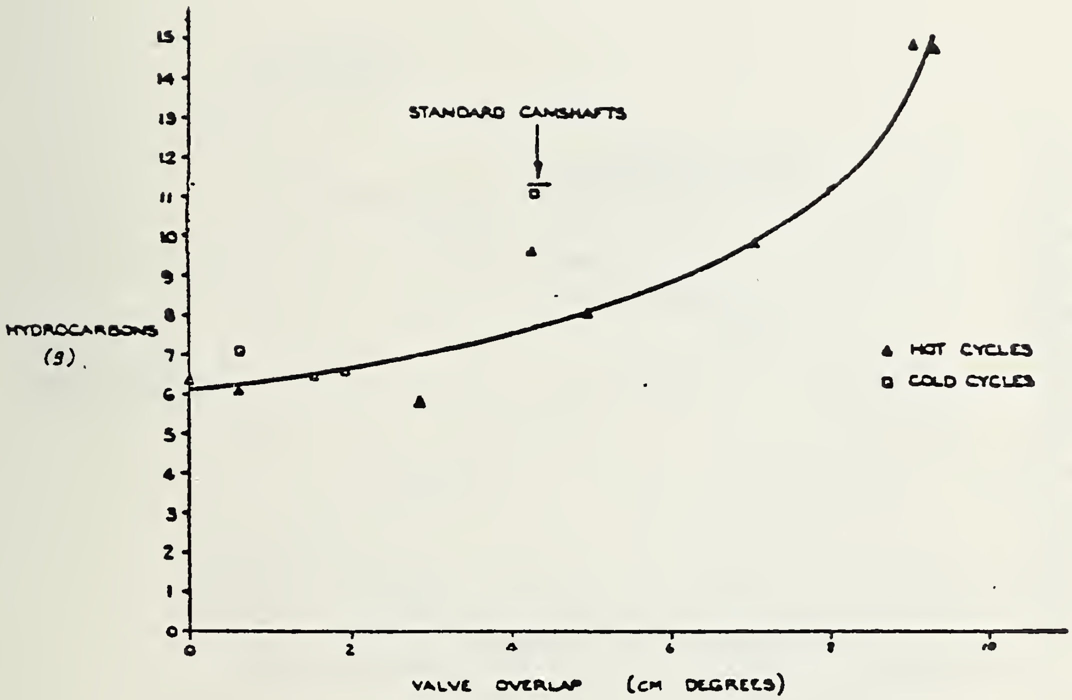


FIGURE 6-1. EFFECT OF VALVE OVERLAP ON HYDROCARBON EMISSIONS DURING EUROPEAN CYCLE TESTS

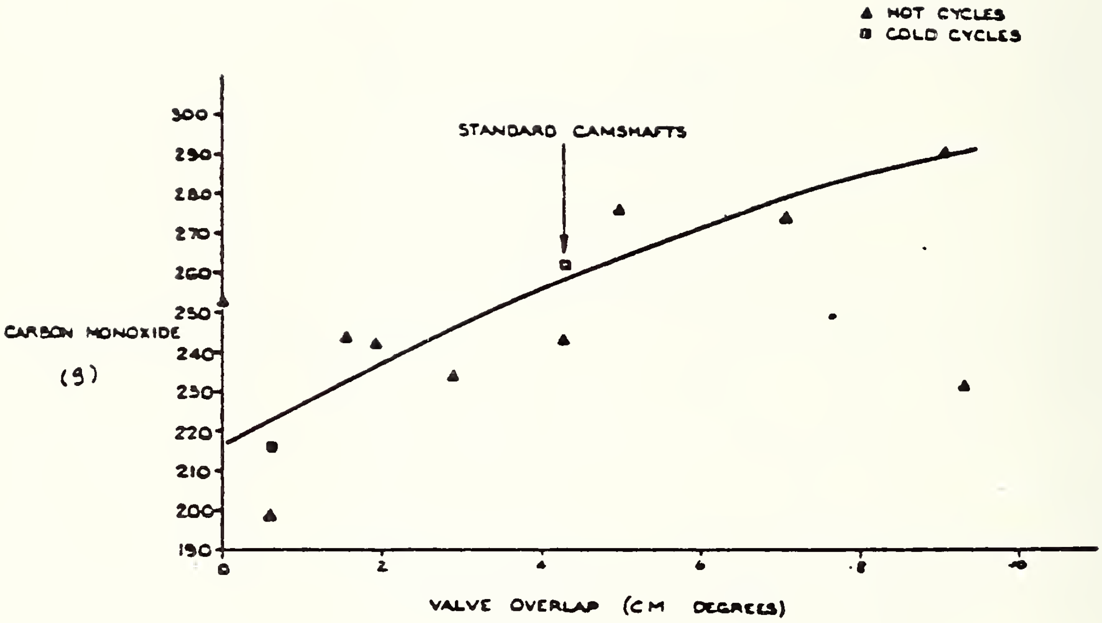


FIGURE 6-2. EFFECT OF VALVE OVERLAP ON CARBON MONOXIDE EMISSIONS DURING EUROPEAN CYCLE TESTS

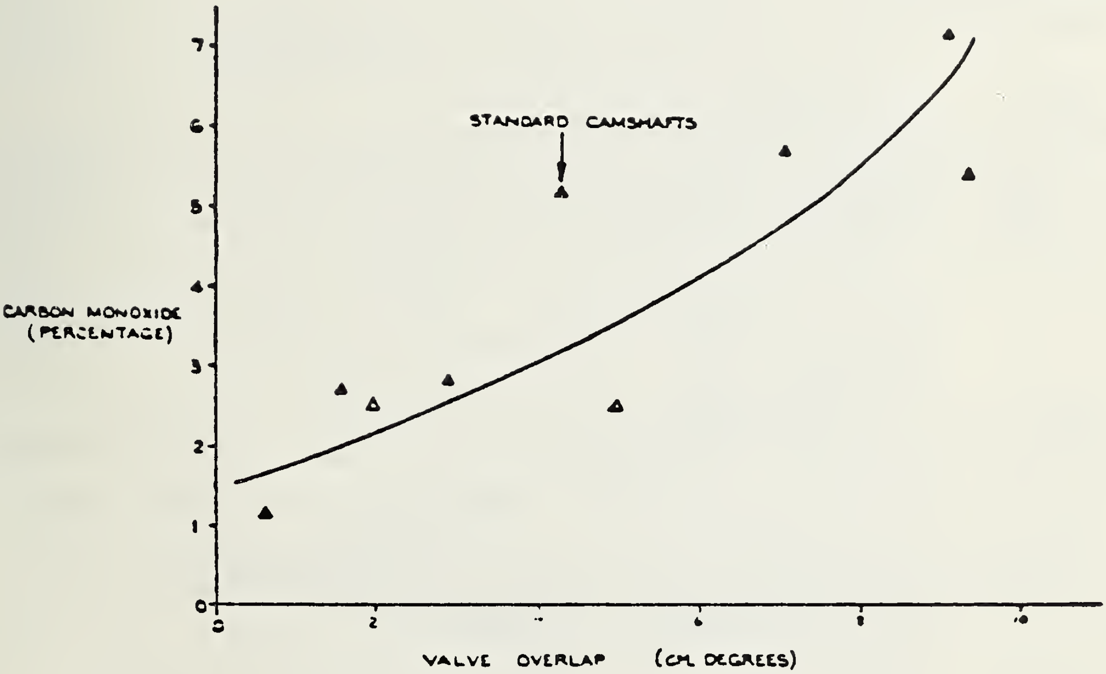


FIGURE 6-3. EFFECT OF VALVE OVERLAP ON CARBON MONOXIDE EMISSIONS AT IDLING SPEED DURING EUROPEAN CYCLE TESTS

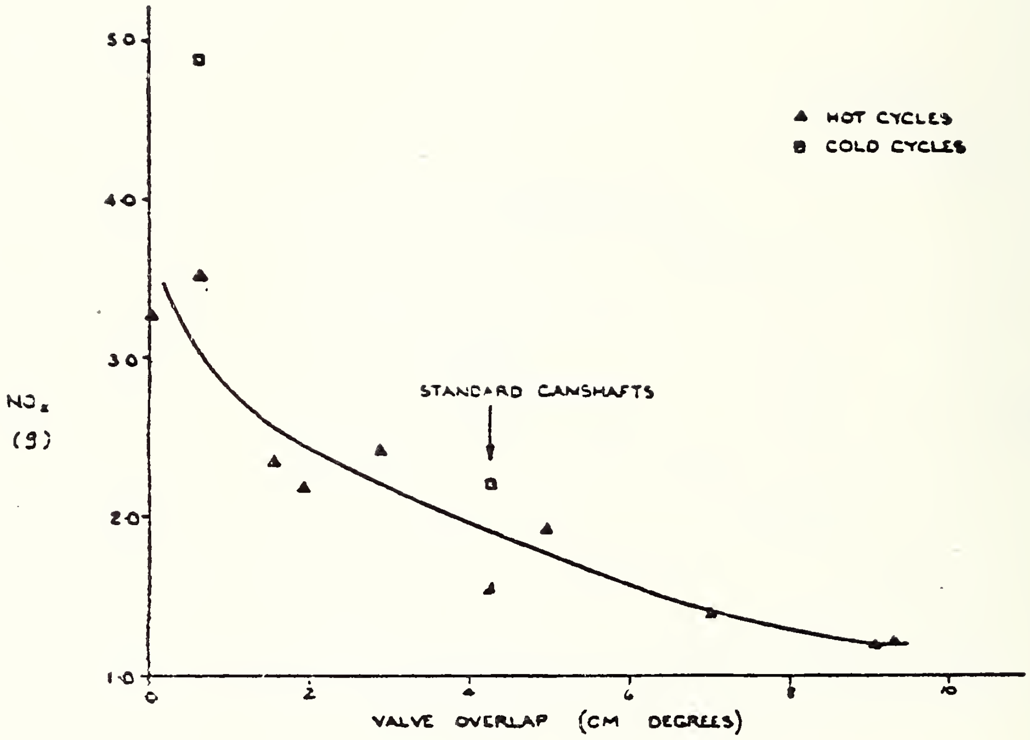


FIGURE 6-4. EFFECT OF VALVE OVERLAP ON OXIDES OF NITROGEN EMISSIONS DURING EUROPEAN CYCLE

6.3 VARIABLE DISPLACEMENT ENGINE

This compact provides the deactivation of the intake and exhaust valves of certain cylinders during a vehicle's operation as a potential means of appreciating fuel economy. The deactivation purpose of valves is to eliminate selected cylinders to enable the engine to operate under more loaded conditions and to reduce the effect of throttling. Any number of valves may be deactivated, provided that sufficient torque is available to power the vehicle.

Generally a control strategy must be implemented for the deactivation of valves, hence cylinders, for a driving mode. One such strategy reported for an eight cylinder engine[2] consists of five major stages. (See Figure 7-5.) These include the initialization and warm-up phase where all cylinders are activated, an idle stage, a part throttle acceleration using all cylinders. The implemented strategy sets the deactivated torque value at 10 percent to 20 percent below the torque value that could be sustained by the next lower number of cylingers.

Steady state tests performed by the Ford Advanced Engine Engineering on a modified 5 liter engine operated on four cylinders show that a fuel economy improvement of 30 percent over the base line vehicle was achieved.

6.4 ENGINE INLET VALVE CONTROL

One of the fundamental design compromises of the light duty spark ignition engine is air throttle. Control of the engine output power is accomplished by limiting the rate of fuel-air charge entering speed, controlling the amount of fuel-air charge entering into the cylinder during each engine cycle or a combination of both. In practice, more engine control is accomplished by limiting the fuel-air charge per cycle then by limiting engine speed. (The latter approach implies a continuously variable tranmission application.)

The device commonly used to limit the engine fuel-air charge is the air throttle more commonly, the carburetor throttle plate. The carburetor throttle plate restricts the flow of air-fuel mixture

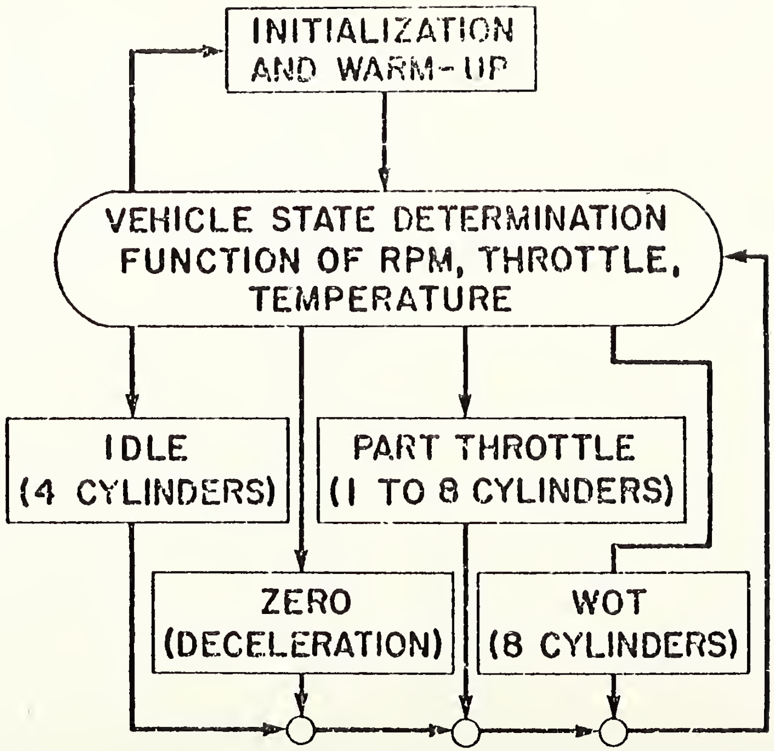


FIGURE 6-5. CONTROL STRATEGY FLOW CHART

into the engine simply by acting as a flow restricter at the entrance of the intake manifold so that a charge of reduced density is drawn into the cylinder during the intake stroke. This induction procedure results in lost work. Typically, a 10 to 20 psi reduction in mean effective pressure occurs at light loads with the carburetor throttle means. Thus, the elimination of this light load throttling loss can, under some conditions, double engine output.

The most efficient way of controlling the fuel-air charge would be an induction process without a pressure drop. In practice, this can be accomplished by opening the intake valve when a sufficient fuel-air charge has been inducted into the cylinder (control of dwell). This can be accomplished by means of a variable valve opening and dwell mechanism. Under full engine power, the inlet valve would stay open as presently required by the carburetor-throttled engines.

A related design concept, known as inlet valve throttling[3] accomplishes throttling by control of the inlet valve under sonic inflow velocities. In this concept the lift of the inlet valve is varied rather than constant over an engine's load speed range. The same pressure drop losses occur in inlet valve throttling as with the conventional throttle plate, but 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 thus 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 where cyclically uniform and repeatable characteristics at air-fuel ratios in excess of 20:1 have been obtained.
Other benefits included reduction of carbon monoxide emissions except during the power enrichment phase. This valving concept can be integrated with a fully developed port injection system to further reduce the emissions of carbon monoxides since enrichment beyond the stoichiometric air-to-fuel ratio is not required for stable engine operations.

REFERENCES FOR SECTION 6

1. 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."

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

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

HE 18.5 .A3
NHTSA- 7
Trella, Th

Potential
engine :

Form DOT F 172
FORMERLY FORM D

00347529

U.S. DEPARTMENT OF TRANSPORTATION
RESEARCH AND SPECIAL PROGRAMS ADMINISTRATION
TRANSPORTATION SYSTEMS CENTER
KENDALL SQUARE, CAMBRIDGE, MA. 02142

OFFICIAL BUSINESS
PENALTY FOR PRIVATE USE, $300

POSTAGE AND FEES PAID
U.S. DEPARTMENT OF TRANSPORTATION
613

U.S. MAIL