NO. DOT-TSC-NHTSA-79-53 DOT-HS-805 133

POTENTIAL OF SPARK IGNITION ENGINE, EFFECT OF VEHICLE DESIGN VARIABLES ON TOP SPEED, PERFORMANCE, AND FUEL ECONOMY

by

Russell W. Zub
Carol M. Neckyfarow
William M. Lew
Ralph G. Colello

U.S. DEPARTMENT OF TRANSPORTATION
RESEARCH AND SPECIAL PROGRAMS ADMINISTRATION
Transportation Systems Center
Cambridge MA 02142

MARCH 1980
FINAL REPORT

U.S. DEPARTMENT OF TRANSPORTATION
NATIONAL HIGHWAY TRAFFIC SAFETY ADMINISTRATION
Office of Research & Development
Washington DC 20590

Technical Report Documentation Page

1. Report No. DOT-HS-805 133	2. Government Accession No.	3. Recipient's Catalog No.
4. Title and Subtitle POTENTIAL OF SPARK IGNITION ENGINE, EFFECT OF VEHICLE DESIGN VARIABLES ON TOP SPEED, PERFORMANCE AND FUEL ECONOMY		5. Report Date March 1980
		6. Performing Organization Code
7. Author(s) R.W. Zub, C.M. Neckyfarow, W.M. Lew, R.G. Colello		8. Performing Organization Report No. DOT-TSC-NHTSA-79-53
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 and Development Washington DC 20590		13. Type of Report and Period Covered Final Report
		14. Sponsoring Agency Code

15. Supplementary Notes

16. Abstract

The purpose of this report is to evaluate the effect of vehicle characteristics on vehicle performance and fuel economy. The studies were performed using the VEHSIM (vehicle simulation) program at the Transportation Systems Center.

The computer simulation offers repeatability and can predict minute changes in fuel economy based on relatively small vehicle alterations. The degree to which each vehicle parameter is modified is based upon projections presented in current literature. The results are assessed and an explanation of the interaction of the vehicle design characteristics on performance is presented.

17. Key Words Acceleration, Top speed, Fuel economy, Gradeability, Aerodynamic drag, Driveline configuration, Vehicle Weight	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	88	

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

PREFACE

This report, DOT-TSC-NHTSA-79-53, is one of a series of four companion reports to DOT-TSC-NHTSA-79-52 "Potential of Spark Ignition Engine, 1979 Summary Source Document."* It evaluates the effect of vehicle characteristics on vehicle performance and fuel economy based primarily on vehicle simulation studies.

This report is a deliverable under PPA HS-027 "Support for Research and Analysis in Auto Fuel Economy and Related Areas."

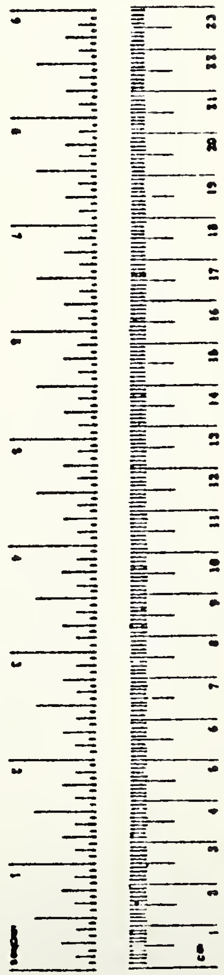
---

* "Potential of Spark Ignition Engine, 1979 Summary Source Document," by T. Trella, R. Zub and R. Colello, 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

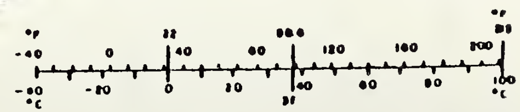


TABLE OF CONTENTS

LIST OF ILLUSTRATIONS

LIST OF TABLES

1. INTRODUCTION

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

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

2. ANALYSIS

2.1 SYSTEMS INTEGRATION

Many criteria have been used to evaluate the performance of automobiles and light duty trucks. Before examining these evaluation methods, it is important to realize that the performance, fuel economy and emissions of a vehicle are determined by the particular characteristics of the vehicle and its operating requirements. The motion of the vehicle is governed by Newton's Second Law of motion. The force required at the rear wheels is delivered by the engine through the transmission and rear axle. The forces at the rear wheels are given by

$$F_t - F_a - F_r - F_g = ma \tag{1}$$

where

F_t = Tractive force

F_a = Aerodynamic drag

F_r = Rolling resistance

F_g = Grade force

m = Vehicle mass

a = Acceleration.

The transmission and rear axle combination multiply the torque output of the engine depending upon the demand at the rear wheels as given in equation (1). The purpose of this report is to modify the variables in equation (1), and the driveline configuration and evaluate the results with respect to vehicle performance, fuel economy and top speed. The vehicle characteristics, weight, rolling resistance, aerodynamic drag and driveline are discussed in the following sections.

2.2 CHARACTERISTICS

2.2.1 Aerodynamic Drag

The aerodynamic drag force which resists the movement of a vehicle through still air is described by the following relationship:

$$F_a = \text{Aerodynamic drag force} = 1/2\ \rho\ V^2\ C_D\ A, \qquad (2)$$

where ρ = Air density

V = Velocity of the vehicle

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

A = Vehicle frontal area.

The aerodynamic drag force is the sum of component forces resulting from the variation of pressure around the vehicle body, from the viscous shear of the air near the vehicle surface (friction or surface drag), and from the resistance of air passing through the vehicle (internal flow drag). The drag from pressure distribution is a function of the basic vehicle shape and attachments on the vehicle (radio antennae, roof gutters, door handles, etc.). The pressure (form) drag and the induced (lift) drag are determined by the pressure distribution over the basic body shape. Interference drag results from perturbations in the flow field associated with projections (such as mirrors, doorhandles, etc.) which cause interaction between flow over basic body shape and flow about the projections. The drag resulting from the projections attached to a vehicle can be several times greater than that of the isolated component in free air flow. The skin friction (surface) drag is dependent upon the surface area exposed to the flow field. The internal drag results from resistance to air passing through openings in the vehicle body, i.e., engine cooling air flow and passenger compartment ventilation.

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

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

The measurement of aerodynamic efficiency is based upon the drag coefficient. The drag coefficient is a dimensionless quantity which has traditionally been determined by experimental methods. Some of the methods include full scale wind tunnel testing, model wind tunnel testing and coast down testing. By measuring various fluid and vehicle properties, the drag coefficient can be calculated by using one of the above techniques. Some investigators have suggested a method based on a statistical rating of given shape characteristics. Whichever method is implemented, it must be realized that each technique has its limitations and variation in results.

The reduction of the drag coefficient can be accomplished in two ways. The vehicle can be redesigned with a primary concern of aerodynamic efficiency instead of aesthetic appeal, or by adding drag reducing devices to the present configuration. One of the techniques posed for reducing the drag coefficient has been the adaptation of a spoiler. Ohanti[1*] of the Nissan Motor Company

[1*] Superscript numbers designate references listed in Section 6.

investigated the effect of changing the car shape and the principle of the rear spoiler. Both situations resulted in lowered drag coefficients and consequently improved fuel economy. Arthur D. Little[2] completed an evaluation of various feasible technologies to improve fuel economy. One of the findings involves changes in chassis and body design to lessen aerodynamic drag that would increase fuel economy six to seven percent. Analytic studies performed by VW investigated the effect of aerodynamics on fuel economy when drivetrain optimization is included.[3] These studies have concluded that the sensitivities (the fuel economy improvement relative to C_D improvement $(FE/FE_O)/(C_D/C_Do)$ are approximately 0.15 for the urban cycle, 0.5 for highway and 0.25 for the composite cycle. It appears that in the 1981 to 1990 time frame a 20 percent improvement in C_D is quite feasible.

2.2.2 Driveline Configuration

The driveline configuration provides the transfer mechanism between the engine output and the force required at the rear wheels. The transmission, rear axle and wheel multiply the torque output of the engine based on the demand at the rear wheels. The selection of the proper driveline configuration is usually made to provide a balance between fuel economy and performance.

With the emphasis on fuel economy, various transmissions have been developed which increase fuel economy. Fuel economy improvements with transmissions result by controlling the engine operating conditions closer to the optimum schedule and from higher transmission efficiencies. The optimum schedule with respect to fuel economy requires operation of the engine through the minimum specific fuel consumption points. A typical spark ignition specific fuel consumption map is shown in Figure 2.1. The road load curve represents the demand on the engine by a vehicle traveling along a level road at constant speed in top gear. By reducing the top gear ratio and/or the rear axle ratio, the road load curve will cross through islands of lower specific fuel consumption. An evaluation of drivetrain components to improve fuel economy is presented by Arthur D. Little.[4] The results indicate that by using wide ratio range automatic transmissions with lock-up torque converters, composite fuel economy improvements of up to 14 percent can be achieved with no change in acceleration, but a loss in driveability.

There are other schemes being discussed to upgrade transmissions, in particular those techniques which utilize extended use of electronic controls. One example is a discrete variable transmission, which when coupled with electronic controls, enables an engine to operate in a narrow region about the minimum fuel consumption line in the horsepower speed plane during light-to-medium acceleration demands. Under high accelerations, especially during urban and highway operations, the transmission is uncoupled from its electronics and assumes operation as a conventional transmission. A typical trajectory of a 5-speed discrete variable transmission is shown in Figure 2.2 during light-to-medium operations. The Fiat Company has conducted studies on the potential fuel economy gains which can be realized through an electronically controlled 4-speed transmission using a torque converter with lock-up in first gear. This study (see Table 2-1) indicates that a 13 percent increase in fuel economy is possible by means of electronics. By electronically con-

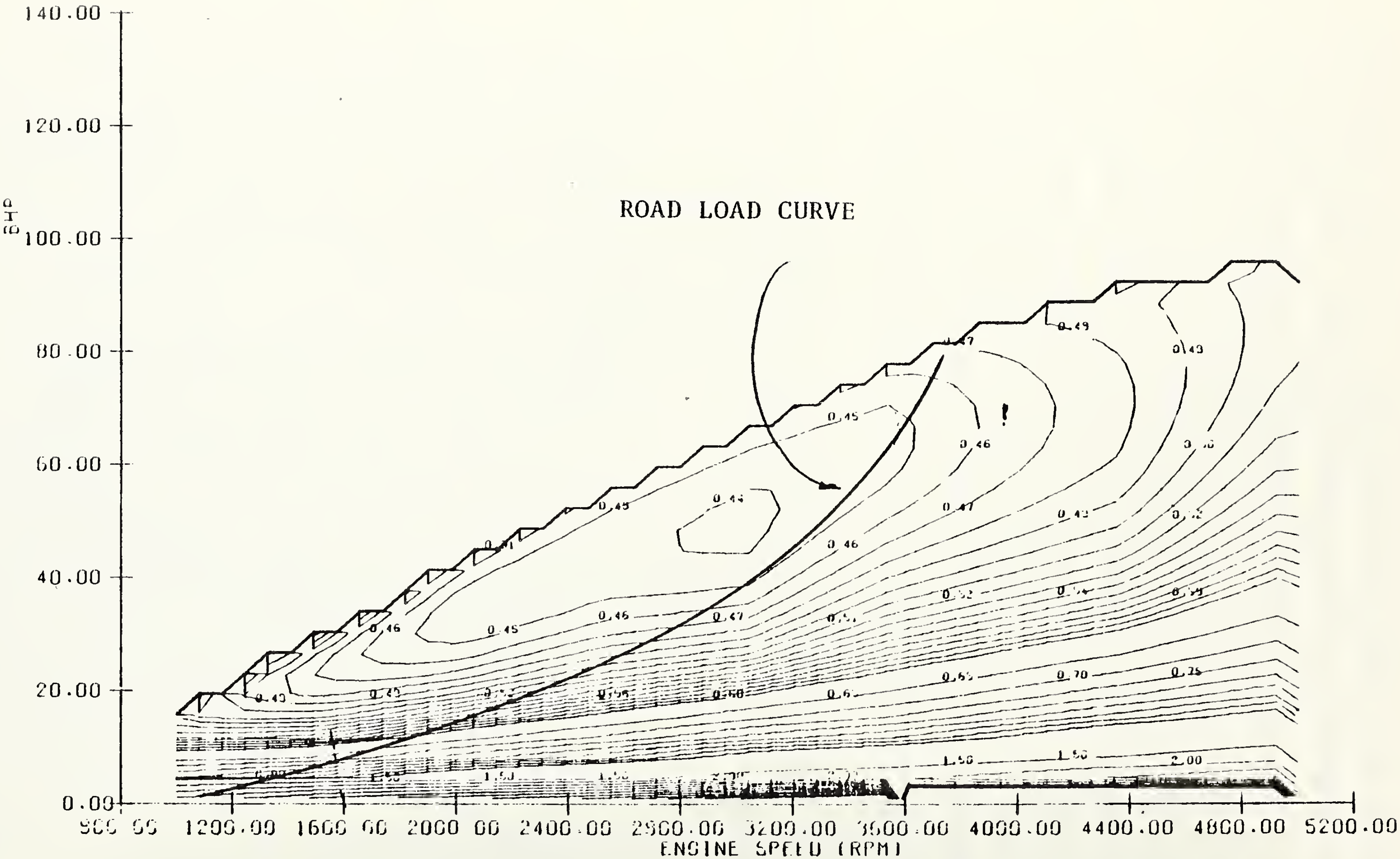


FIGURE 2-1. SPARK IGNITION BRAKE SPECIFIC FUEL CONSUMPTION MAP

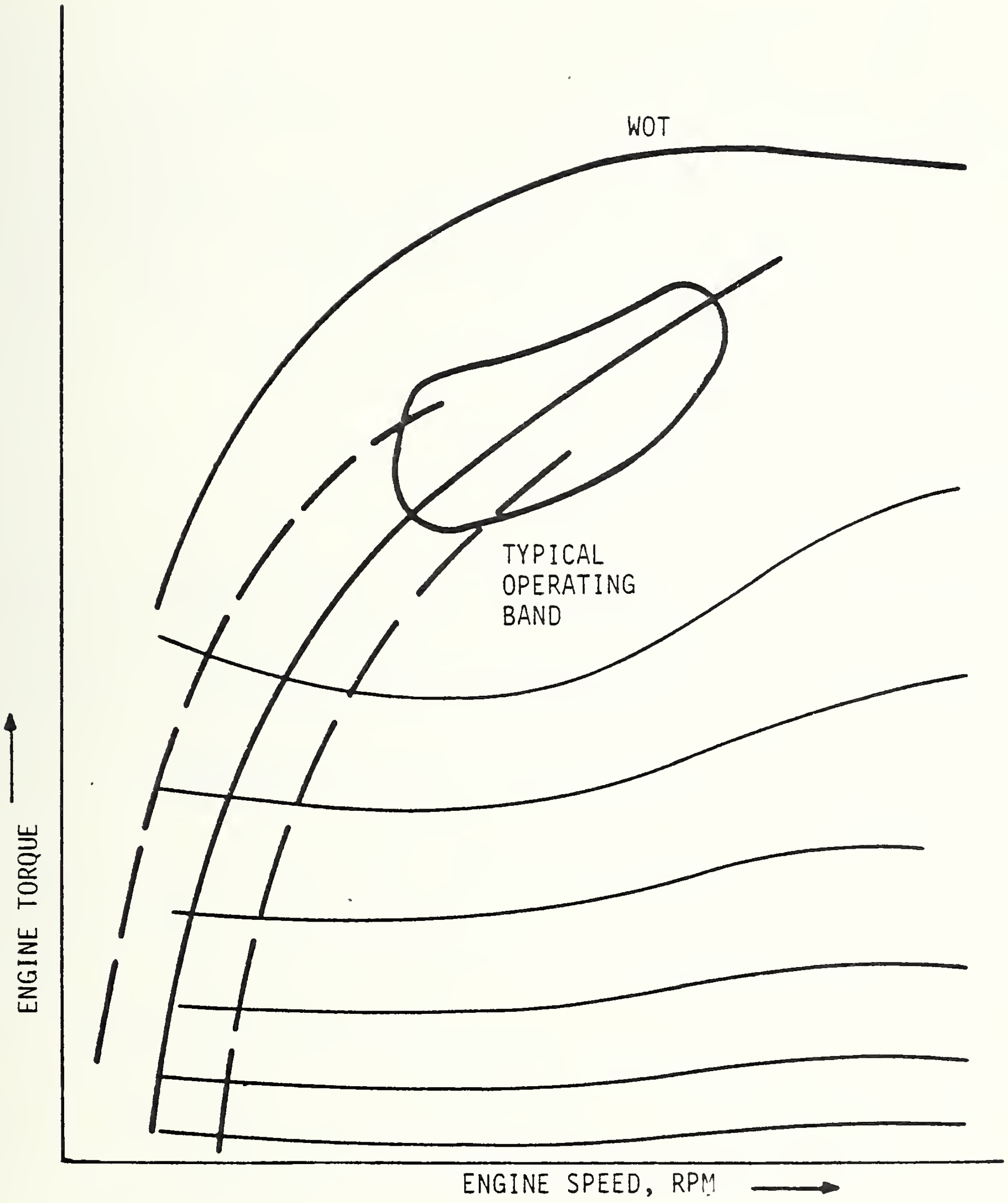


FIGURE 2-2. TYPICAL OPERATION OF DISCRETE VARIABLE TRANSMISSION

TABLE 2-1. POTENTIAL FUEL ECONOMY OF ELECTRONICALLY CONTROLLED TRANSMISSIONS

TRANSMISSION TYPE	FUEL ECONOMY MPG			% Fuel Economy Improvement (Composite cycle)
	Urban	Highway	Composite	
5-speed manual	21.6	29.0	24.4	Base
electronically controlled Discrete Transmission (4-speed + lock-up)	25.4	31.0	27.6	13%
electronically controlled (CVT)	27.3	33.4	29.7	22%

Fiat 132 engine in 2750 lb inertia weight European passenger car trolling a continously variable transmission, Fiat estimates that an additional nine percent increase in fuel economy is possible.

During the last four years, manufacturers have been lowering the rear axle ratio to reduce fuel consumption. In 1975, the lowest axle ratio available on a General Motors car was 2.56. One year later the ratio was lowered to 2.41. Since 1975, Ford has lowered the rear axle ratio of the Pinto/Bobcat by 12 percent. Chrysler has made lower axle ratios optional on more models. The relationship of fuel economy sensitivity to axle ratio is given by:[5]

$$FE \sim \frac{CID\ (RAR)^{b}}{IWT} \qquad (3)$$

where CID = Engine cubic inch displacement

RAR = Rear axle ratio

IWT = Inertia weight in pounds

b = Sensitivity.

An average sensitivity determined from statistical regression of the 1977 EPA Certification Fleet data is -0.4.

2.2.3 Rolling Resistance

Tire rolling resistance is the total force required to overcome the resistance of the four tires to forward motion at a given speed. The measurement of tire rolling loss is usually accomplished by utilizing a 67" drum, Clayton twin roll, or flat belt configuration. The correlation between these test facility results and the actual on road rolling resistance results is unacceptable and a facility for simulating real road test conditions in definitely needed. The rolling loss of a tire is made up of three parts:[6]

a) Friction or scrubbing between tire and roadway

b) Windage loss of the tire

d) Hysteretic losses of the tire materials due to cyclic stressing.

The majority of the tire loss is hysteretic which is influenced by inflation pressure, tire load, tread composition, carcass design and inflation temperature. By modifying these variables the hysteretic losses can be reduced. This reduction can be accomplished in many ways, such as utilizing a radial geometry design and increasing inflation pressures.

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

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

The elliptical tire is a widely advertised example of a geometric change that offers low rolling resistance. The elliptical tire requires few concessions in present suspension design. It requires a novel rim design, however, and poses interchangeability problems. A low-aspect ratio tire having conventional rim design can also offer low rolling resistance. The concept may prevail over the elliptical design. Table 2-2 gives the rolling resistance and estimated fuel economy improvement for the various tire types.

TABLE 2-2. TIRE ROLLING RESISTANCE

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

*Combined EPA driving cycles. **Based on Bias Ply Tires
Source: Reference 5

Recently, the Society of Automotive Engineers stated that with continued research efforts tire rolling resistance could be reduced to 7.5 lbs./1000 lbs. load[7] through continued research into:

- o Higher Inflation Pressure
- o Better Compounding
- o More Use of Radial Design
- o More Efficient Construction.

2.2.4 Weight

From equation (4) it can be seen that the tire rolling resistance and acceleration are both affected by the weight of the vehicle. A ten percent reduction in vehicle weight can result in an improvement in fuel economy of two to three percent to a significant eight to nine percent depending on the method of weight reduction. Three methods of weight reduction commonly used are downsizing, material substitution and optimum design. In 1976 General Motors downsized its full size Chevrolet as shown in Figure 2-3. The vehicle weight was reduced and a smaller engine replaced the original. The net affect was to maintain performance while reducing fuel consumption.

The fuel economy sensitivity to weight has been represented in a number of reports[5] as follows:

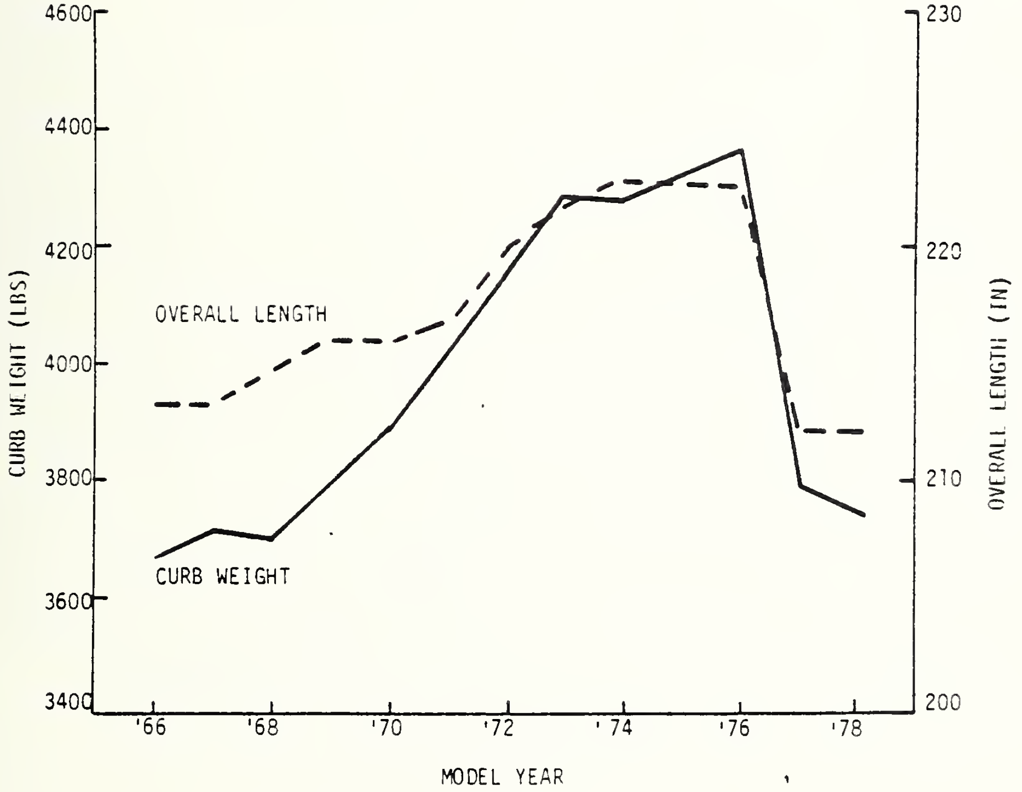
$$FE \sim W^a \qquad (4)$$

where FE = Miles per gallon

W = Inertia weight in pounds

a = Sensitivity.

An average sensitivity was determined when statistical regression techniques were applied to EPA certification fleets for various model years. The value of average sensitivity was -0.4 (percent mpg per percent change in weight) from investigation of 423 vehicles in the 1977 40 state certification fleet.



Source: Reference 8

FIGURE 2-3. CURB WEIGHT AND OVERALL LENGTH OF FULLSIZE CHEVROLET WITH V-8 ENGINE

2.3 PERFORMANCE EVALUATION

The criteria used to evaluate the driving attributes of the vehicles are acceleration, fuel economy, top speed and gradeability. Each of these criteria is examined individually below.

2.3.1 Acceleration

The acceleration capability of the vehicle was measured by means of 0-60 mph wide open throttle time, distance covered in five sec from a standing start and 40-60 mph passing time. The acceleration can be related to Newton's Second Law.

$$a = \frac{F}{m} \qquad (5)$$

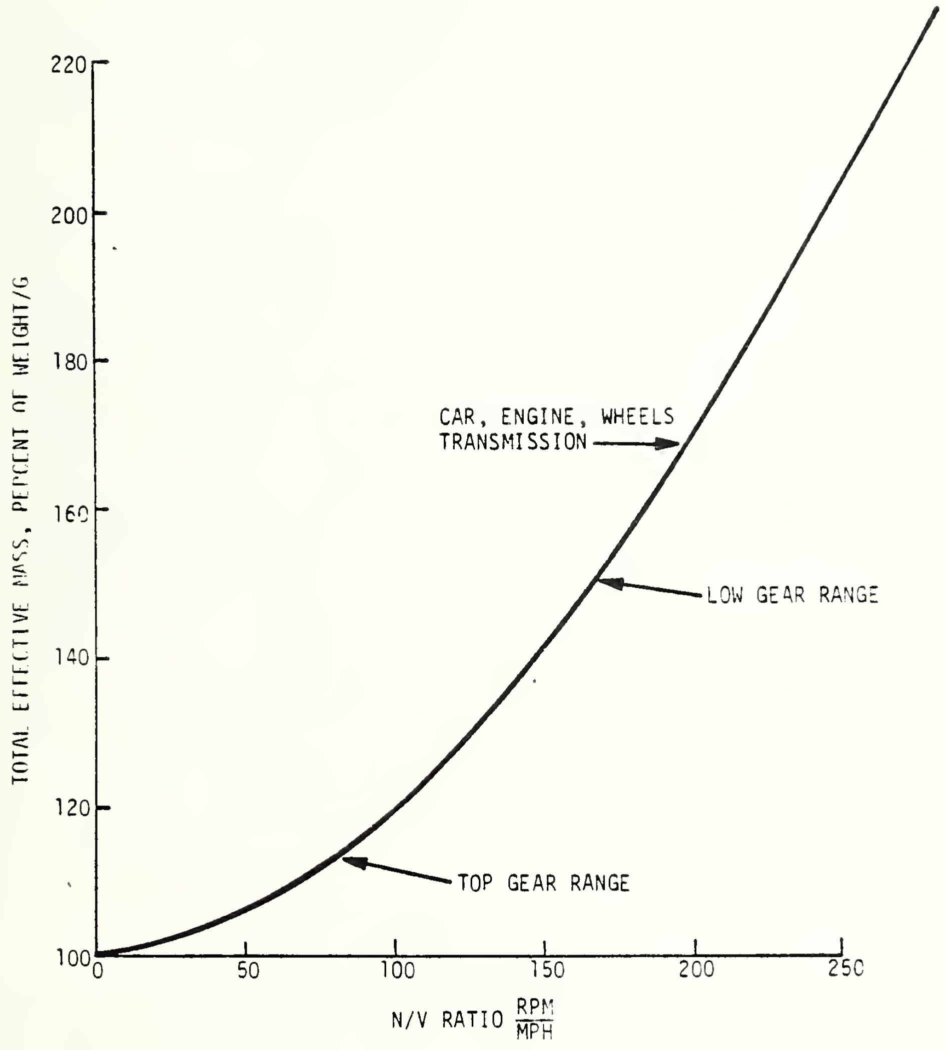
where

a = Acceleration

m = Vehicle mass

F - Rear wheel force.

This equation is not valid unless the rotational inertia of the wheels and driveline parts is considered. Traditionally, the rotational inertia is expressed in terms of linearly accelerated weight. Then the mass is called the total effective mass of the vehicle. This mass, with the lower gears having the high N/V ratios, increases with speed ratio, as shown in Figure 2-4. This curve is shown for illustrative purposes only and the actual effective mass for each vehicle will be different. The net accelerating force available can be calculated by knowing the rolling radius of the rear wheels, the rear axle/transmission gear reduction, the road load force and the WOT torque curve. A typical engine torque curve is shown in Figure 2-5. The maximum



Source: Reference 9

FIGURE 2-4. TOTAL EFFECTIVE MASS AS A FUNCTION OF N/V RATIO

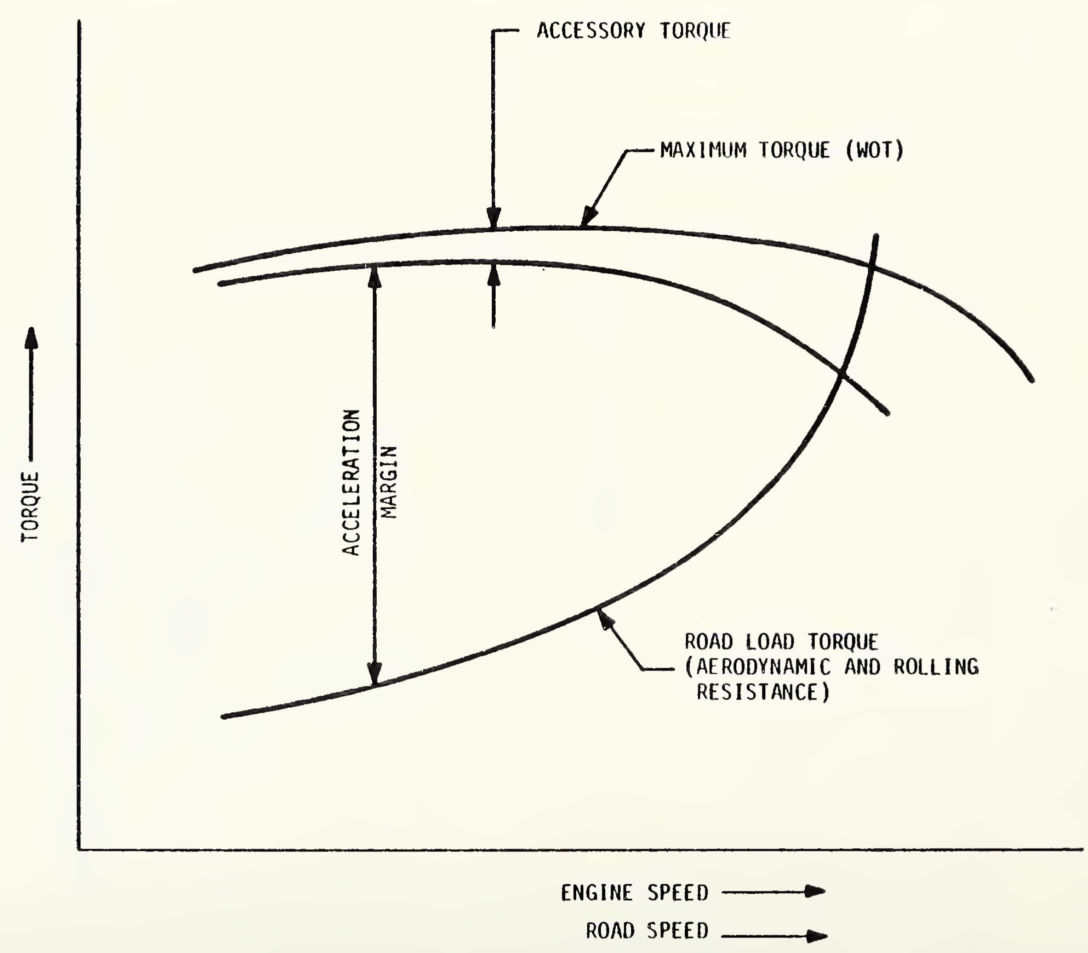


FIGURE 2-5. ENGINE TORQUE CURVE

acceleration is obtained where the difference between the road load torque and engine torque is greatest. It should be noted that the accessory torque reduces the torque available for acceleration. This acceleration capability is utilized through gear reduction by changing the gear ratios.

The utilization of the acceleration capability of the vehicle can be characterized over a given velocity by the 0-60 mph time, five sec distance from a standing start and 40-60 mph passing time. Very simply, the 0-60 mph time is an indication of the overall acceleration performance of the vehicle while the five sec distance delineates the starting acceleration of a vehicle, such as starting from a stop sign. The 40-60 mph time portrays the passing ability of the vehicle.

2.3.2 Top Speed

The theoretical top speed of a vehicle is determined by the intersection of the road load curve and engine horsepower curve. The maximum top speed will be obtained by intersecting the road load curve at the maximum engine horsepower output. Top speed has not been used as a design criteria but more as an end result based upon desired acceleration and fuel economy constraints. In the past, a typical vehicle had the road load curve intersecting the WOT engine curve slightly after the peak horsepower point.

At high vehicle speeds the centrifugal force acting on the tire increases the rolling radius which subsequently changes the top speed by several miles per hour. This effect is acknowledged but not considered an important factor in this study.

2.3.3 Gradeability

Gradeability is defined as the maximum grade that a vehicle can climb at a given speed in each gear. Gradeability and acceleration are directly related, except gradeability is a steady-state calculation and therefore the static weight of the vehicle is used. The basic calculation for gradeability is given by:

$$G = \tan \arcsin \frac{F_t - F_W}{W} \tag{6}$$

where G = Gradeability (% grade)

F_t = Tractive force

F_W = Road load force

W = Vehicle weight.

The maximum tractive rear wheel force can be calculated by:

$$F_t = T_{ENG} \times TR \times GR \times RAR \times n/RR \tag{7}$$

where T_{ENG} = Engine torque

T_R = Torque conversion

G_R = Gear ratio

RAR = Rear axle ratio

n = Driveline efficiency

RR = Rolling radius of wheel.

The road load force which is a function of the velocity is just the sum of the aerodynamic drag and rolling resistance. To simplify the analysis, these equations do not consider the traction capability of the drive wheels which is a function of the road surface, tire condition and weight distribution. However, tire slip can limit the maximum gradeability in first gear.

2.3.4 Fuel Economy

Fuel economy, which is usually measured in miles/gallon, is the end result of many complex interactive processes. Fuel economy is based on the amount of energy required to propel a vehicle and the fuel consumption characteristics of the engine BSFC (brake specific fuel consumption) map. A BSFC curve was shown in Figure 2-1 with road load curves. Modifications to a vehicle, such as lowering the rear axle ratio or reducing the frontal area, which cause the engine to operate at islands of lower BSFC, will increase the fuel economy, all other factors being equal.

The standard method of determining fuel economy is to simulate a vehicle over the EPA urban and highway drive schedule. The shift logics for the automatic transmissions are fixed by the manufacturer and the shift logic for the manual is based on that recommended by the EPA. However, the manufacturer may submit a shift schedule which is more representative for a particular vehicle. This schedule must meet with EPA approval.

3. BASELINE SELECTION

The baseline vehicles selected for this study represent a cross section of automobiles and light duty trucks. The engines include four, six and eight cylinders. The vehicles and their engine displacements are shown in Table 3-1. The approach used here is first to examine the performance criteria and run a baseline case for each vehicle. Then the independent variables will be modified to determine their effect on the dependent variables as shown in Table 3-2. The engine was included as an independent variable to evaluate the effect on performance between similar engines.

TABLE 3-1. BASELINE VEHICLES

VEHICLE	WEIGHT (lbm)	ENGINE (CID)	RAR	TRANSMISSION	HP/WT
AUTOMOBILE	2000	98	3.58	M4	.034
AUTOMOBILE	3000	130	3.91	M5	.030
AUTOMOBILE	3500	231	2.56	A3	.027
AUTOMOBILE	4500	318	2.71	A3	.030
MINI-PICKUP	3000	140	3.08	M4	.032
LIGHT DUTY PICKUP	4000	351	3.54	M4	.040

A3 Automatic 3-speed

M4 Manual 4-speed

M5 Manual 5-speed

TABLE 3-2. VEHICLE VARIABLES

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

4. RESULTS AND DISCUSSION

4.1 ACCELERATION

As shown in Equation (5), the vehicle acceleration is related to Newton's Second Law. Since the mass of the vehicle is known the acceleration can be derived once the rear wheel force is calculated. At WOT conditions the rear wheel force required is simply the difference between the road load force and that delivered to the rear wheels at WOT. The rear wheel force is given by Equation (7). The torque at the wheels is simply the product of the force and the rolling radius. Horsepower, which is the rate of work, is given by:

$$HP = \frac{T \times RPM}{5252} \tag{8}$$

where

HP = Horsepower (Hp)

RPM = Wheel revolutions (rpm)

T = Torque (lb-ft).

The acceleration of the baseline vehicles is characterized by three drive schedules. The majority of energy expended for these drive cycles is attributed to the mass as shown in Table 4-1. Therefore, changes in the rolling resistance and aerodynamic drag will not be evaluated.

The effect of weight reduction on the 0-60 mph time can be seen in Figure 4-1. The sensitivities associated with the weight change are also included. As the weight is reduced, the performance

TABLE 4-1. ENERGY REQUIREMENTS FOR WOT ACCELERATION DRIVE SCHEDULES

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

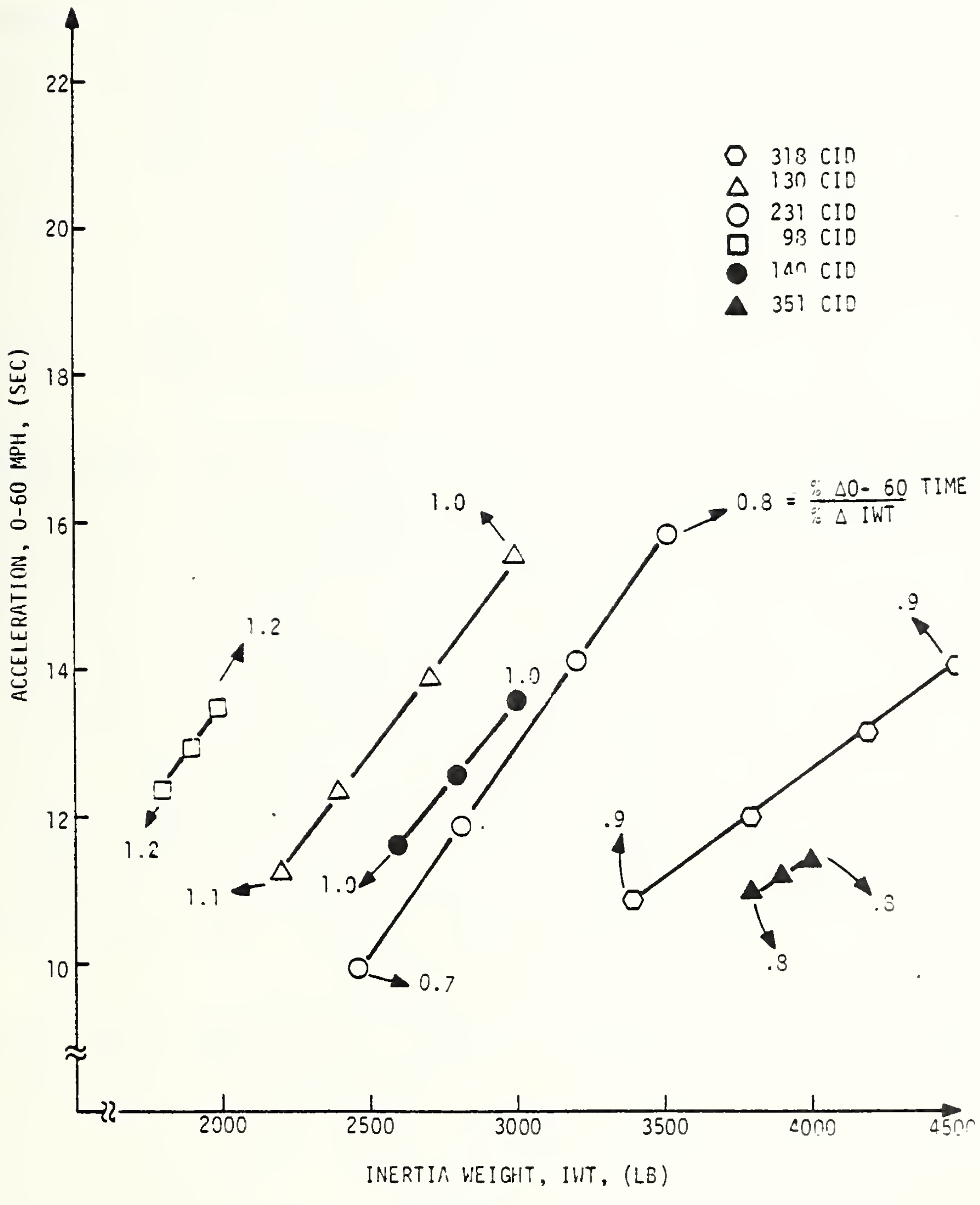


FIGURE 4-1. ACCELERATION (0-60 mph) INCREASE DUE TO WEIGHT REDUCTION

improves. An additional gain in fuel economy (discussed in Section 4-4) can be achieved by geometrically scaling the engine to maintain constant performance. This improvement in fuel economy can be made without significantly affecting the acceleration time as shown in Table 4-2.

The influence of the rear axle ratio on the three acceleration drive schedules is presented in Figures 4-2, 4-3 and 4-4. The inflection points in the curves are caused by the shift logic. For example, in Figure 4-2 for the 98 CID vehicle, the acceleration time for a RAR of 3.8 is 13.2 sec, while that for a RAR of 4.0 is 13.7 sec. In the former case, the vehicle reaches 60 mph in 3rd gear and in the latter case, it is 4th gear.

Another example can be taken from Figure 4-3. For the 140 CID vehicle, the acceleration time for a RAR of 2.7 is 6.05 sec, while that for a RAR of 3.0 is 6.73 sec. With a 3.0 RAR, the vehicle shifts into 3rd gear before 60 mph accounting for the additional 0.68 seconds. Therefore, when discussing acceleration times, it is apparent that shift logic does affect the time simply because the shift point is based upon RPM and not mph. With the exception of the shift points, it can be seen that the rear axle ratio does affect the acceleration.

An estimate of the 0-60 mph time can be taken from Figure 4-5.

TABLE 4-2. ACCELERATION TIME AFFECTED BY WEIGHT CHANGE FOR CONSTANT PERFORMANCE

VEHICLE WT CHANGE (lb)	ENGINE CHANGE (CID)	Δ0-60 TIME (sec)	SENSITIVITY $\frac{\%\Delta Time}{\%\Delta WT}$
2000 → 1800	98 → 90	14 → 14	-
3000 → 2200	130 → 96	16 → 17	0.2
3500 → 2450	231 → 160	16 → 19	0.6
4500 → 3400	318 → 240	14 → 15	0.3
3000 → 2600	140 → 120	14 → 14	-
4000 → 3800	351 → 331	11 → 12	1.7

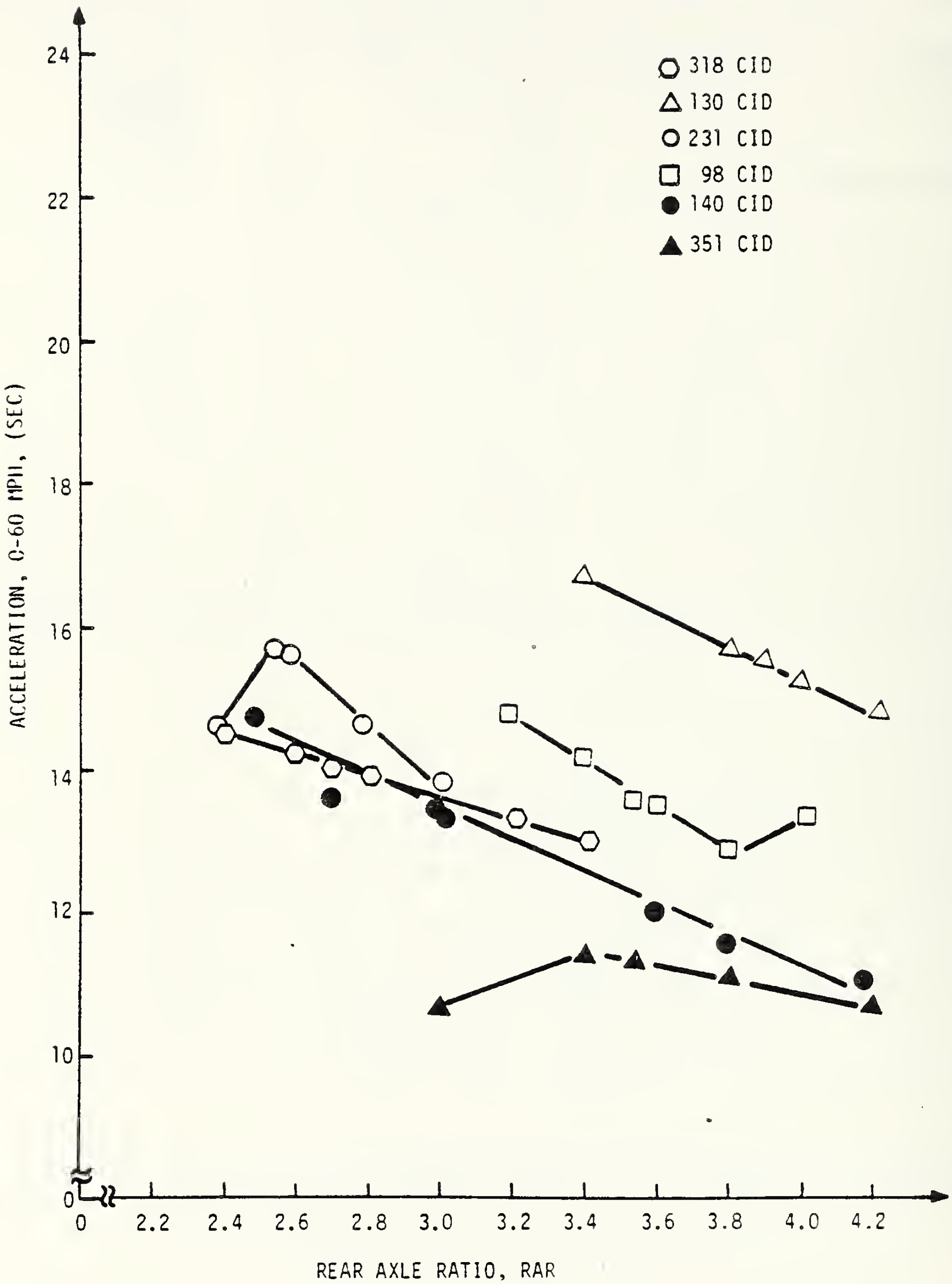


FIGURE 4-2. ACCELERATION (0-60 mph) TIME AFFECTED BY REAR AXLE RATIO

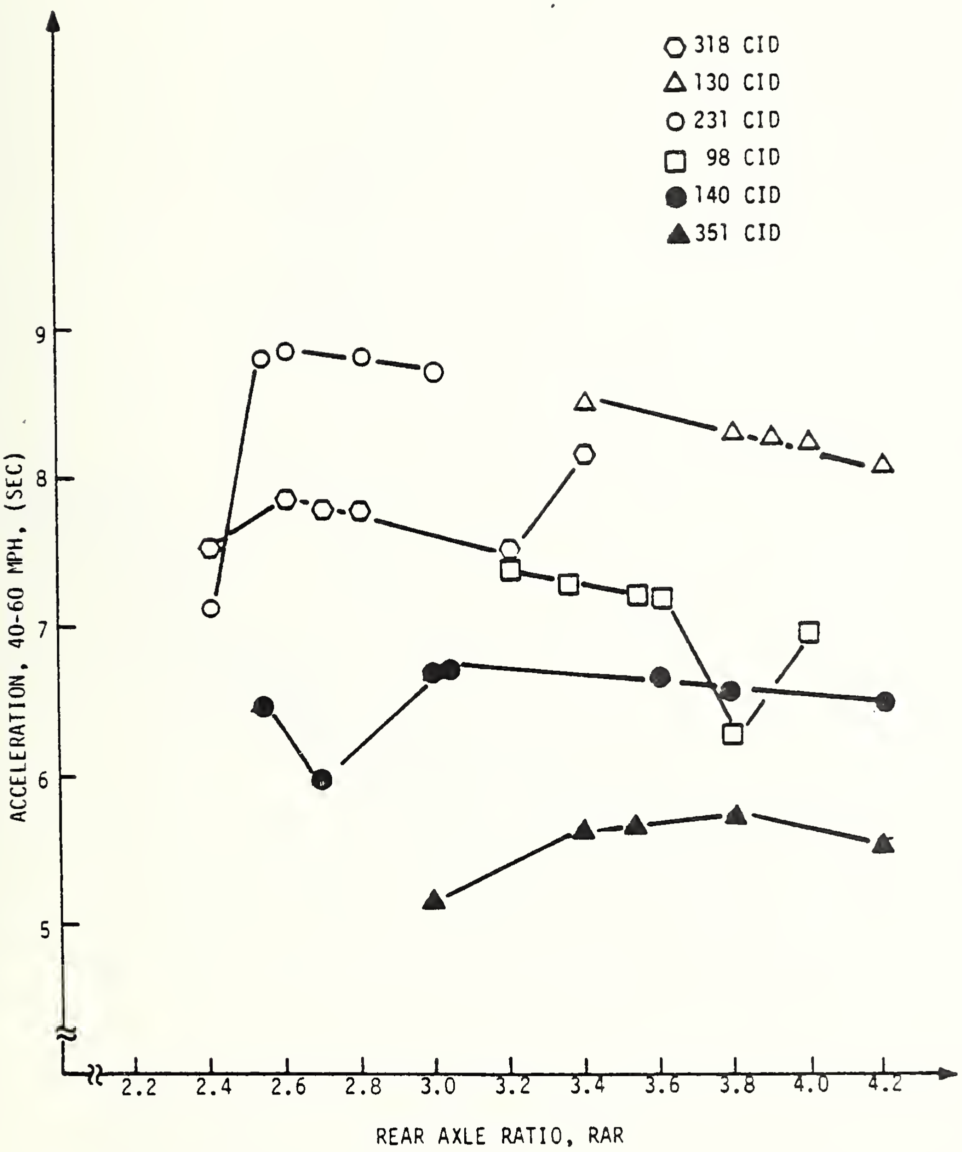


FIGURE 4-3. PASSING ACCELERATION (40-60 mph) TIME AS A FUNCTION OF REAR AXLE RATIO

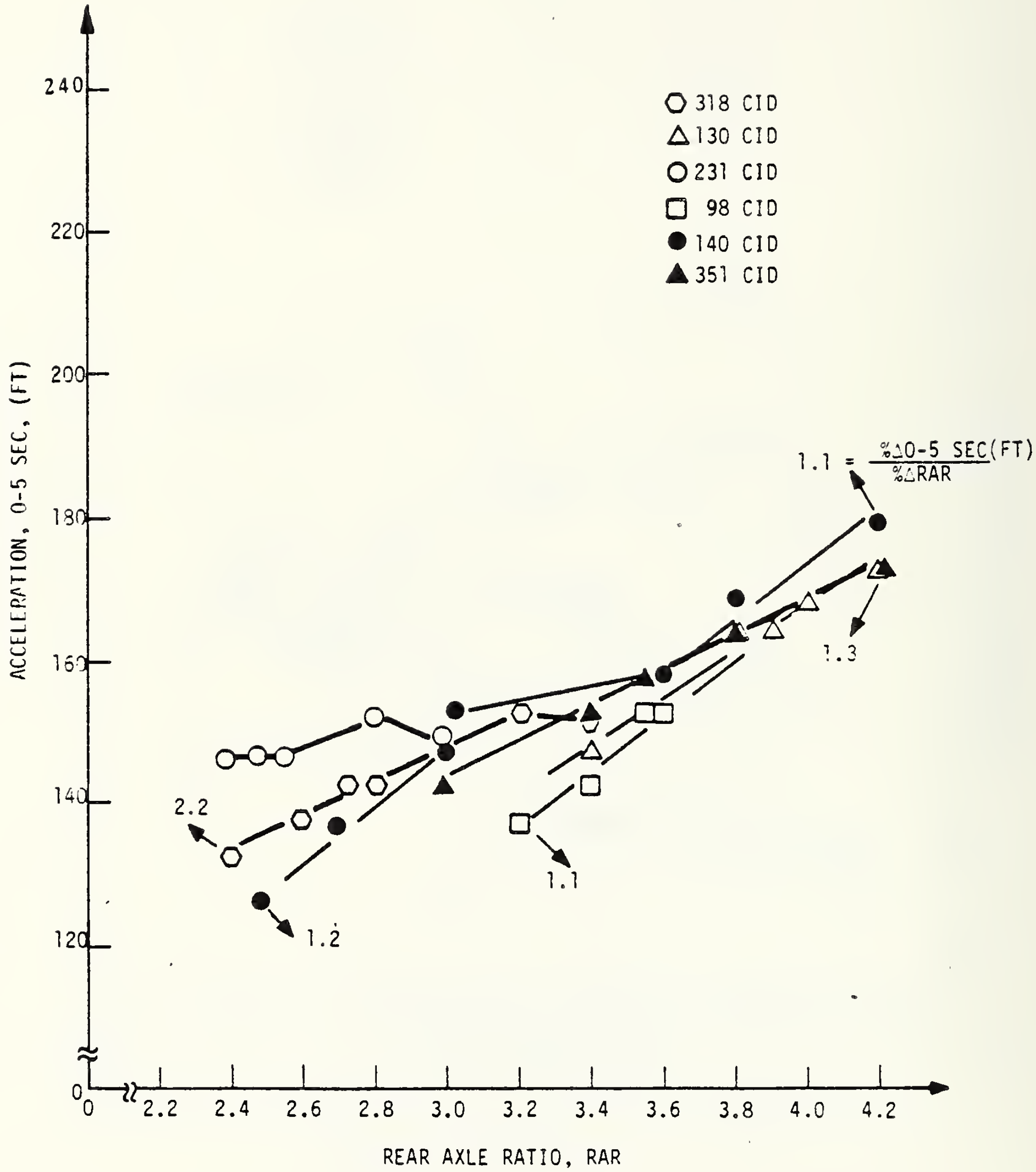


FIGURE 4-4. ACCELERATION DISTANCE (0-5 sec) AS A FUNCTION OF REAR AXLE RATIO

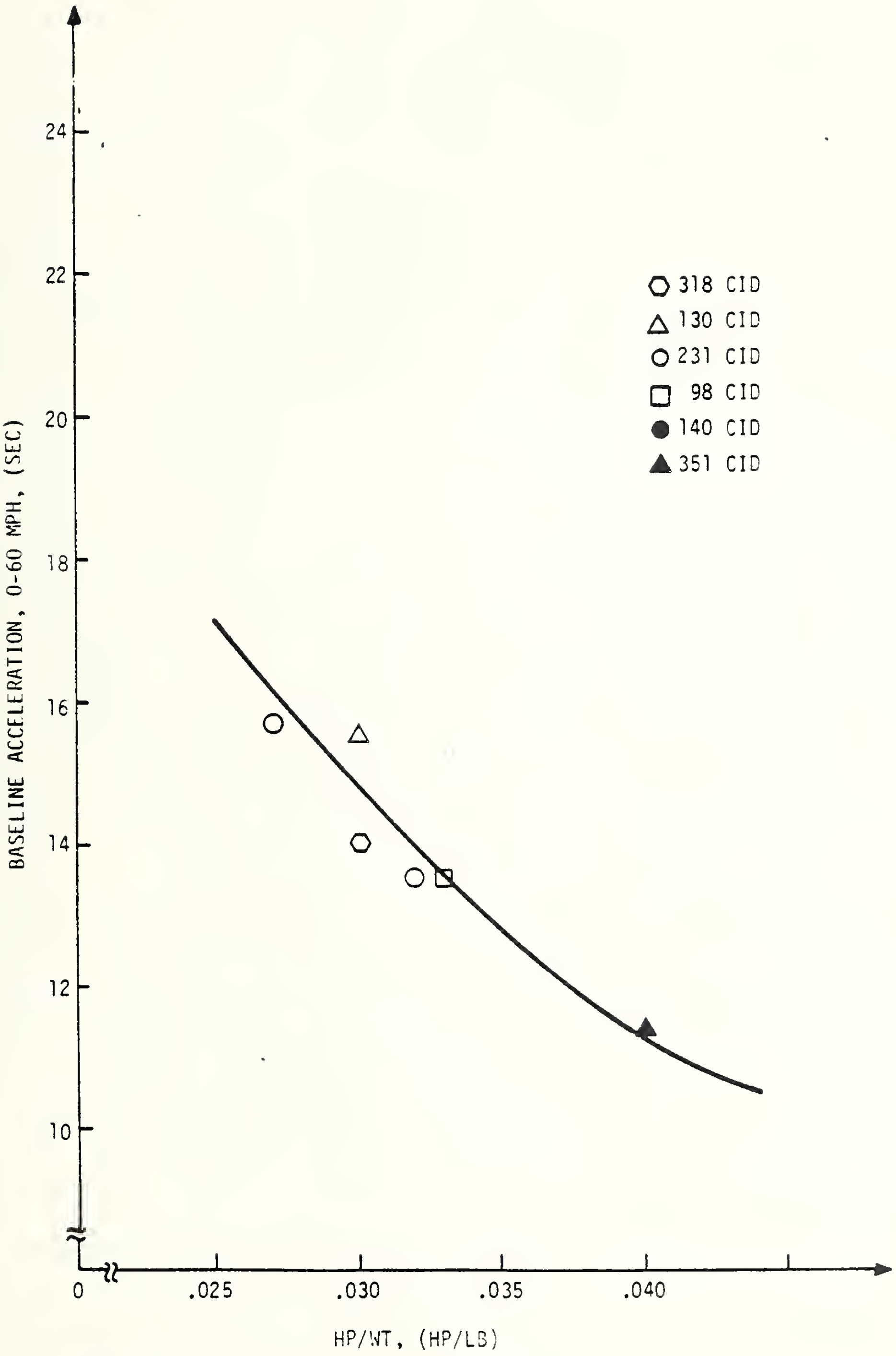


FIGURE 4-5. ACCELERATION TIME (0-60 mph) AS A FUNCTION OF HP/WT

4.2 TOP SPEED

The top speed of a vehicle will be determined by the intersection of the engine WOT horsepower and road load curves. By examining Figure 4-6, it can be seen that as a vehicle approaches its top speed, approximately 87 percent of its output energy is used to overcome aerodynamic drag. Therefore, any weight change or rolling resistance change will not significantly affect the top speed. However, a weight change accompanied with a scaled engine will decrease the top speed as shown in Table 4-3. This top speed decrease is simply caused by the decrease in engine horsepower. The relationship among top speed, maximum engine horsepower and effective area of the vehicle (frontal area x drag coefficient) is shown in Figure 4-7. Since the aerodynamic drag is the predominant energy consumer at the top speed, it is obvious that a drag coefficient reduction will increase the top speed. Figure 4-8 illustrates this result along with the various sensitivities.

The effect of a rear axle ratio change can be explained by understanding Figure 4-9. A rear axle ratio change simply repositions the road load curve. A higher numerical ratio moves the curve to the right and a lower numerical ratio moves it to the left. Already it is apparent that the top speed change will be dependent upon just how much the road load curve moves, and by the shape of the horsepower curve. For this particular vehicle, the baseline rear axle ratio produces a road load curve which intersects the WOT engine horsepower curve near its maximum. Therefore, either a decrease or increase in rear axle ratio will lower the top speed. If the baseline road load curve intersects the engine curve to the

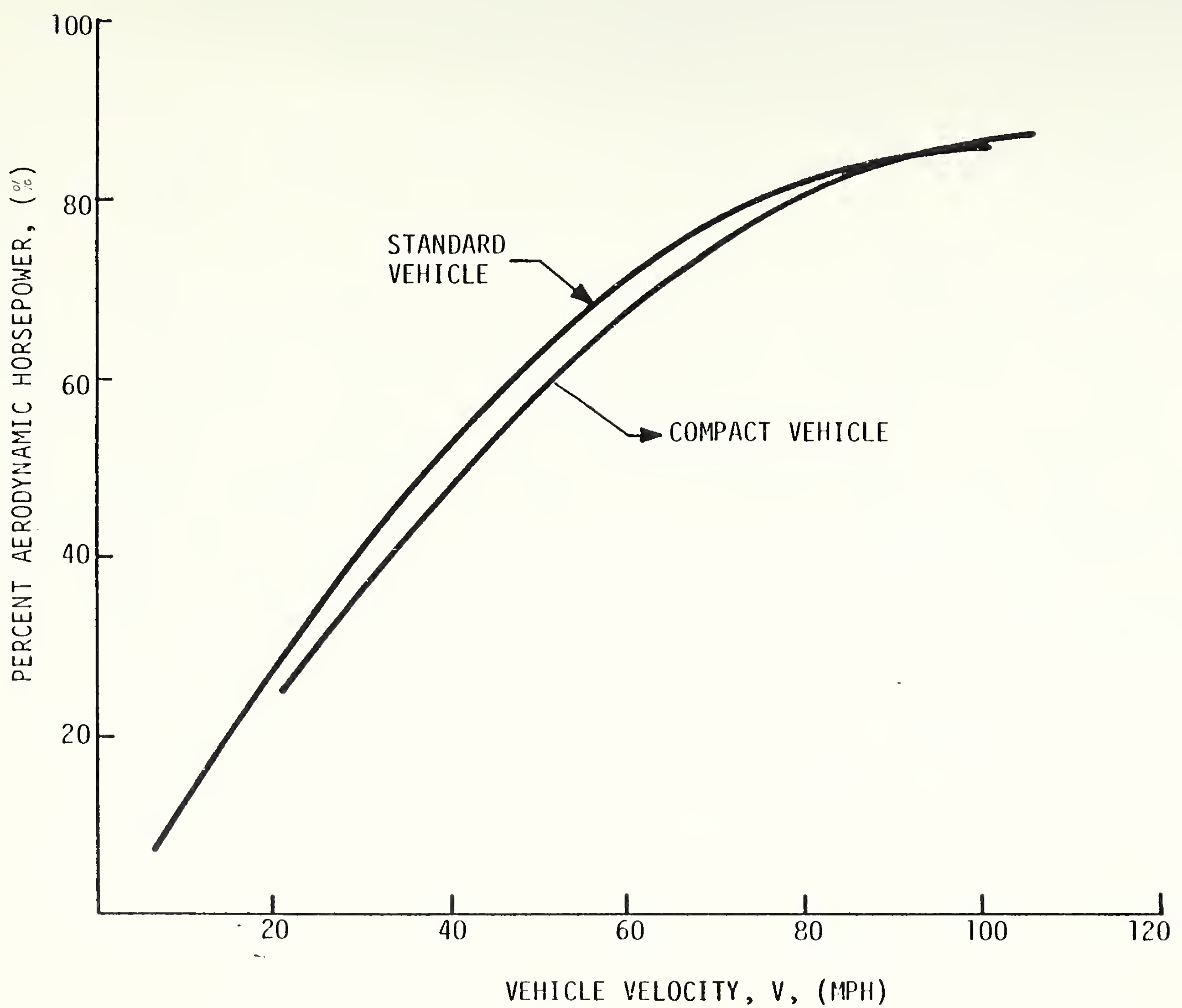


FIGURE 4-6. AERODYNAMIC HORSEPOWER AS A FUNCTION OF VEHICLE VELOCITY

TABLE 4-3. TOP SPEED VARIATION AS A FUNCTION OF WEIGHT REDUCTION FOR EQUAL PERFORMANCE ENGINES

Δ WEIGHT CHANGE (lb)	Δ ENGINE DISPLACEMENT (CID)	Δ TOP SPEED (MPH)
2000 → 1800	98 → 90	98[1] → 94
3000 → 2200	130 → 96	91[1] → 88
3500 → 2450	231 → 160	99[1] → 87
4500 → 3400	318 → 240	103[1] → 94
3000 → 2600	140 → 120	94[1] → 88
4000 → 3800	351 → 331	100[1] → 97

1 Baseline configuration

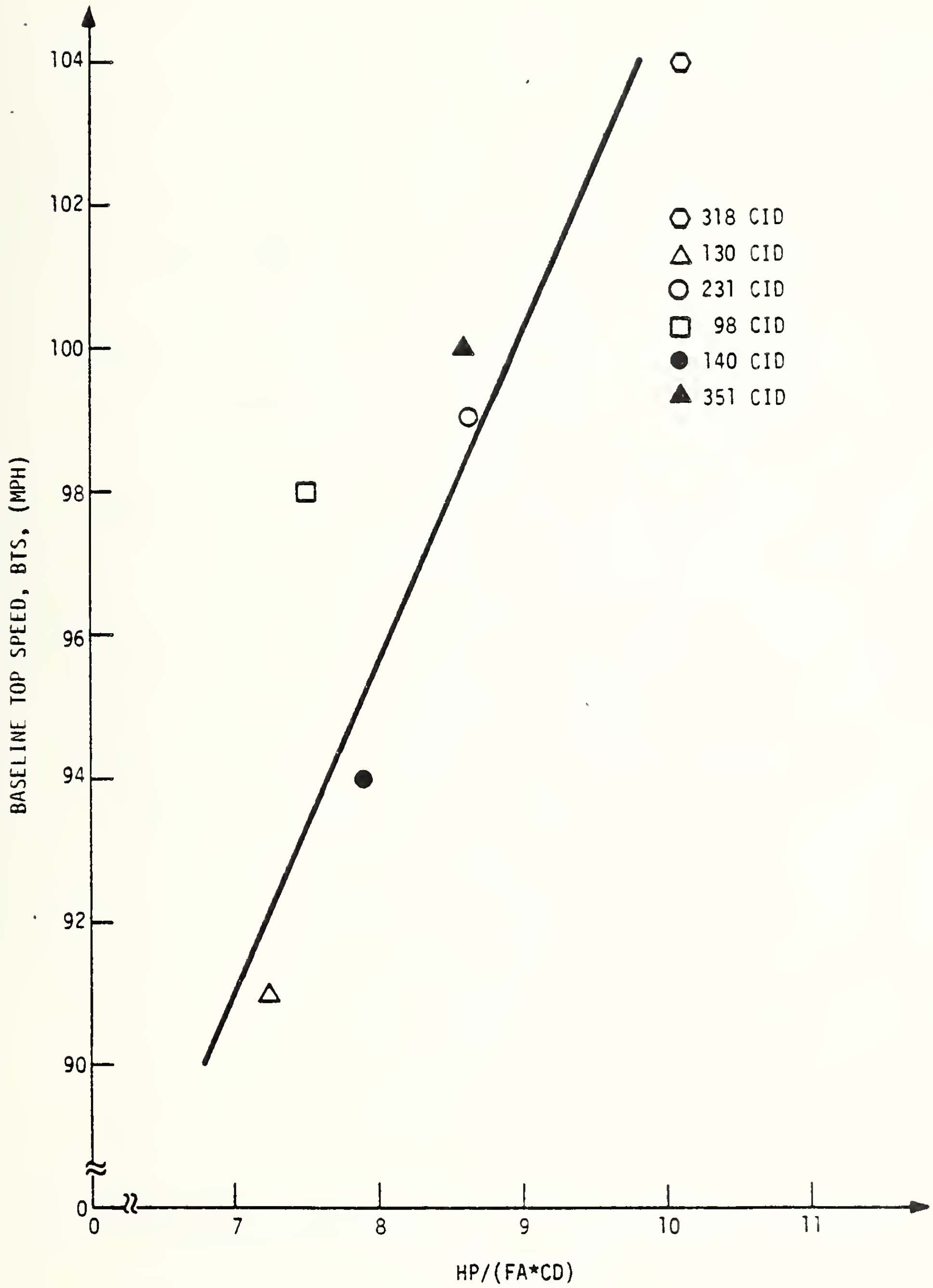


FIGURE 4-7. BASELINE TOP SPEED AS A FUNCTION OF EFFECTIVE FRONTAL AREA

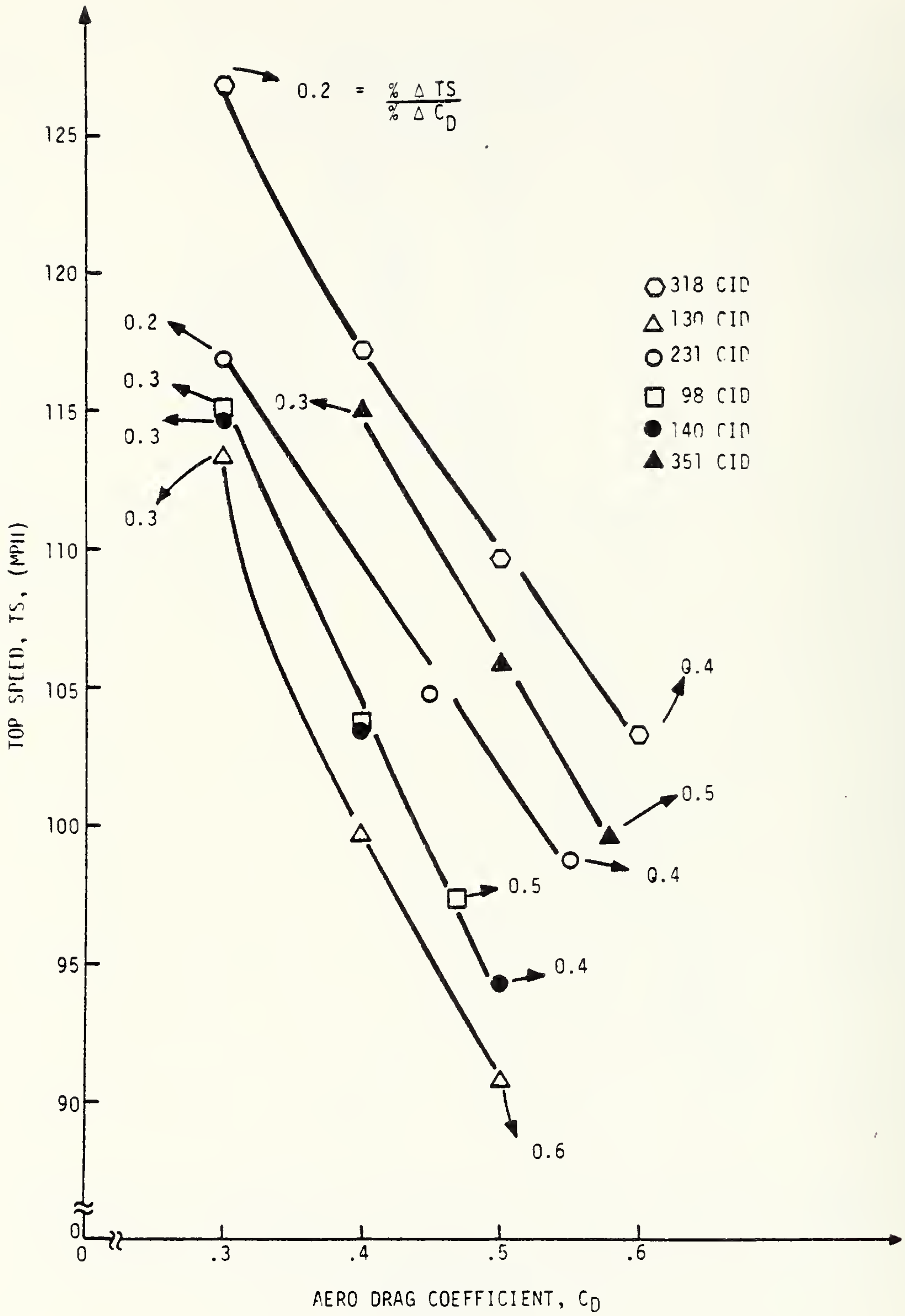


FIGURE 4-8. TOP SPEED AS A FUNCTION OF AERODYNAMIC DRAG

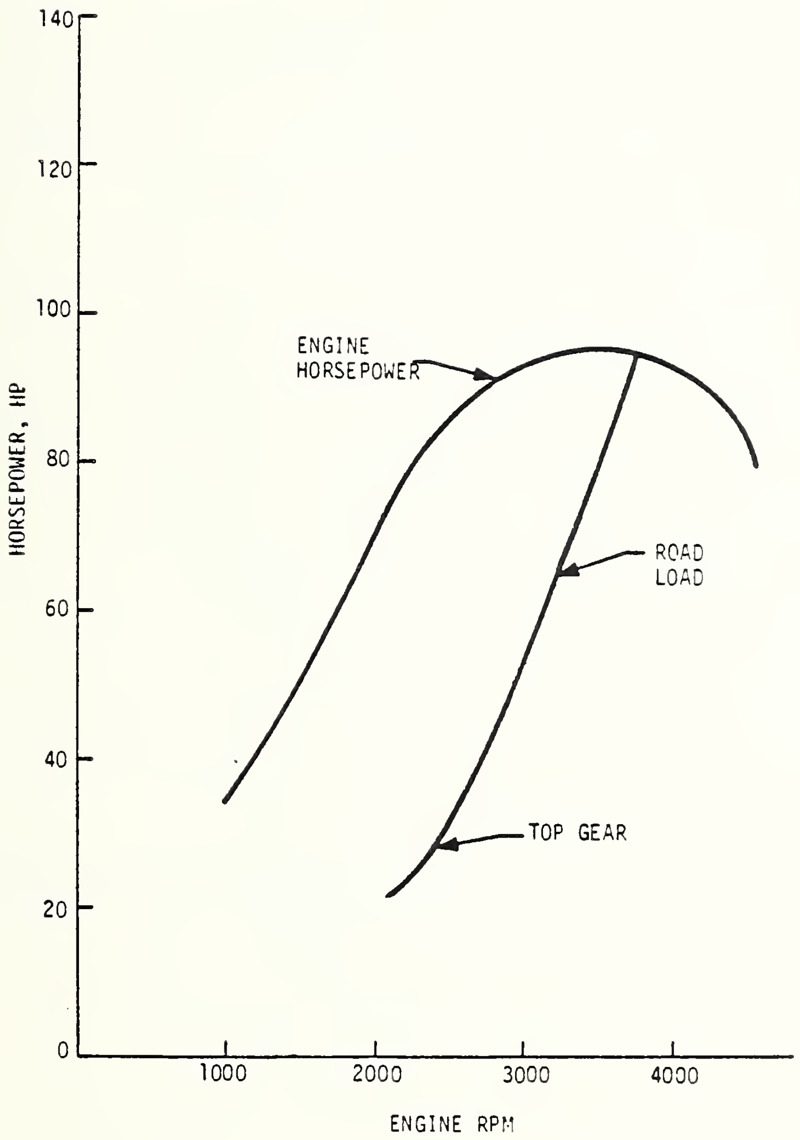


FIGURE 4-9. ROAD LOAD REQUIREMENTS FOR 3500 lb 231 CID AUTOMOBILE

left of the peak, then a numerical reduction in the rear axle ratio could force the vehicle to obtain a higher speed in a lower gear because the shift points are based on RPM and not speed. This is what occurs for the 351, 140, and 130 CID vehicles as indicated in Figure 4-10. If the road load curve is shifted to the right of the maximum engine horsepower, the top speed could be limited by the maximum RPM of the engine.

The torque will affect the top speed only by affecting the acceleration, which is the means to obtaining a top speed.

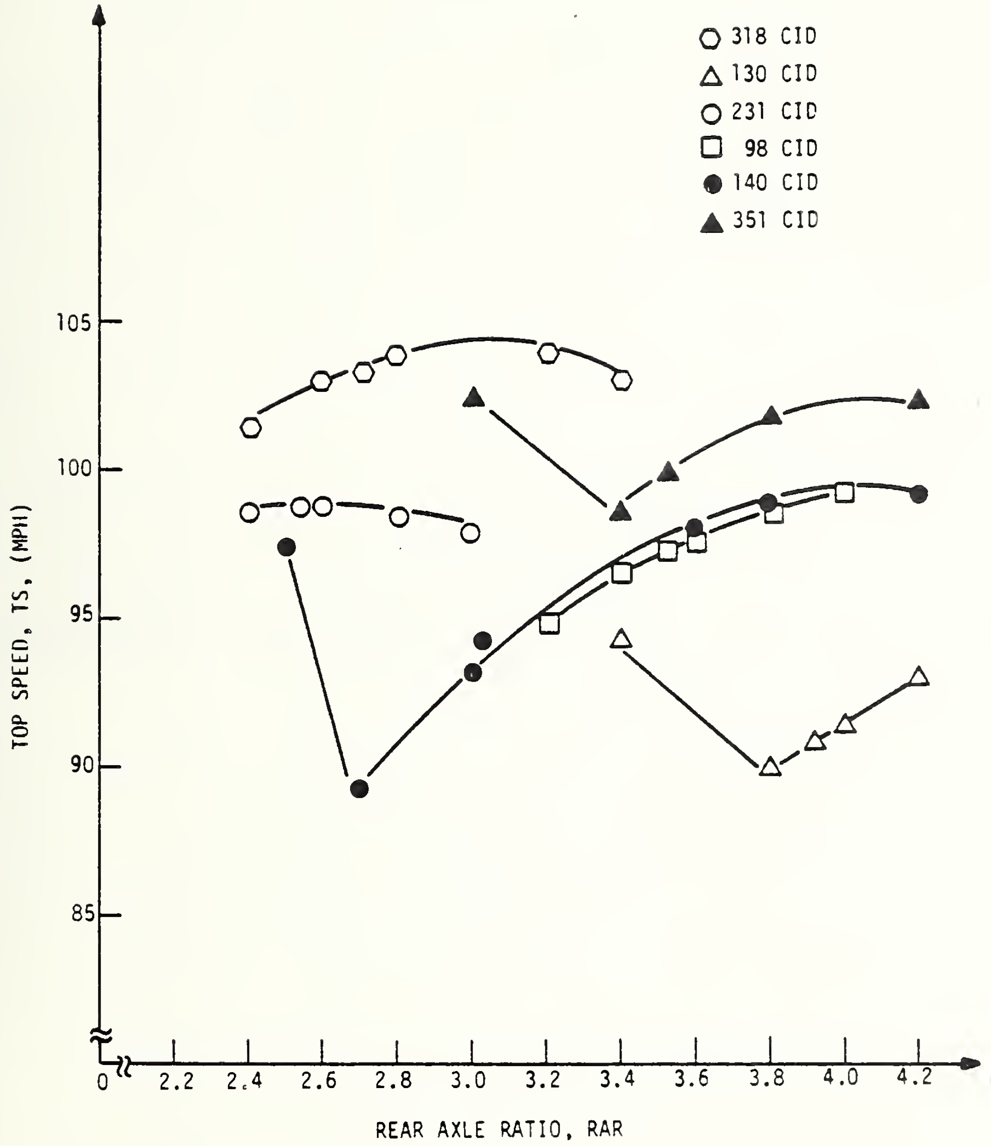


FIGURE 4-10. TOP SPEED AS A FUNCTION OF REAR AXLE RATIO

4.3 GRADEABILITY

Gradeability, which is defined as the maximum grade a vehicle can negotiate at a given speed, is calculated from Equation (6). The results for each of the baseline vehicles are presented in Figures 4-11 - 4-16. When calculating the gradeability of a vehicle in first gear, it is important to consider three variables. The tires, the transmission and the clutch, when applicable, influence the gradeability at start and low vehicle velocities. It can be seen from observing the results that the gradeability is affected by tire slippage. A value of 0.8 was used as the coefficient between the road and tire surface. This value is most likely different for the 351 CID light duty truck. Therefore, the upper limit for this vehicle was indicated as a range. Also, it is obvious that if four-wheel drive were implemented, the gradeability would be significantly increased.

As shown in Equation (7) the tractive force is a function of the product of the engine torque and the torque ratio. The maximum gradeability for a vehicle equipped with a manual transmission should occur at maximum engine torque, although the startup gradeability will be influenced by the clutch dynamics engagement. The maximum gradeability for an automatic transmission will occur near the stall speed as the product of the torque and torque ratio will be a maximum at or very near that point.

As stated before, both gradeability and acceleration are a measure of the capacity to accelerate a vehicle, the only difference being gradeability is a steady-state calculation. Therefore, at WOT conditions as the velocity increases, the

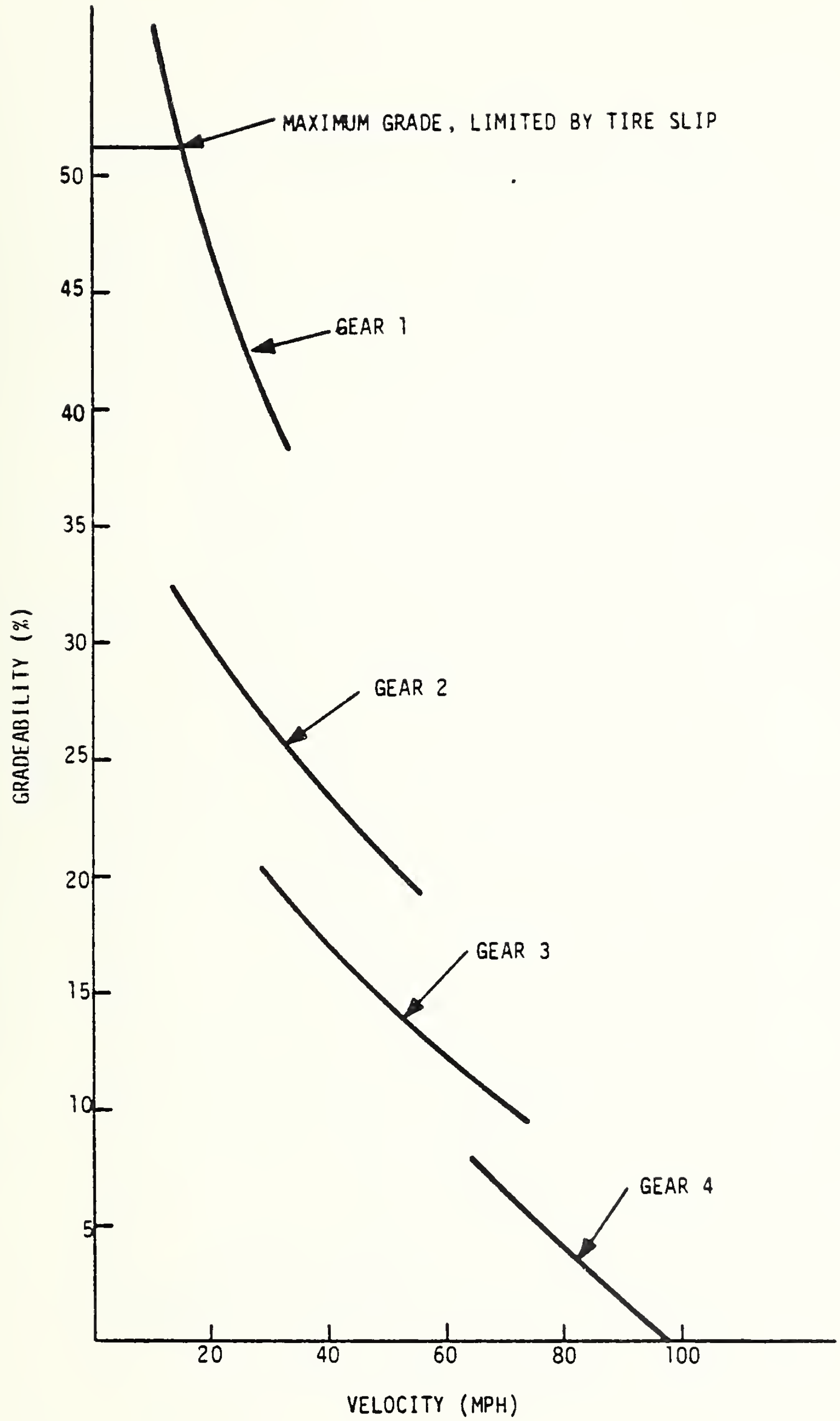


FIGURE 4-11. GRADEABILITY FOR A 2000 lb 98 CID AUTOMOBILE

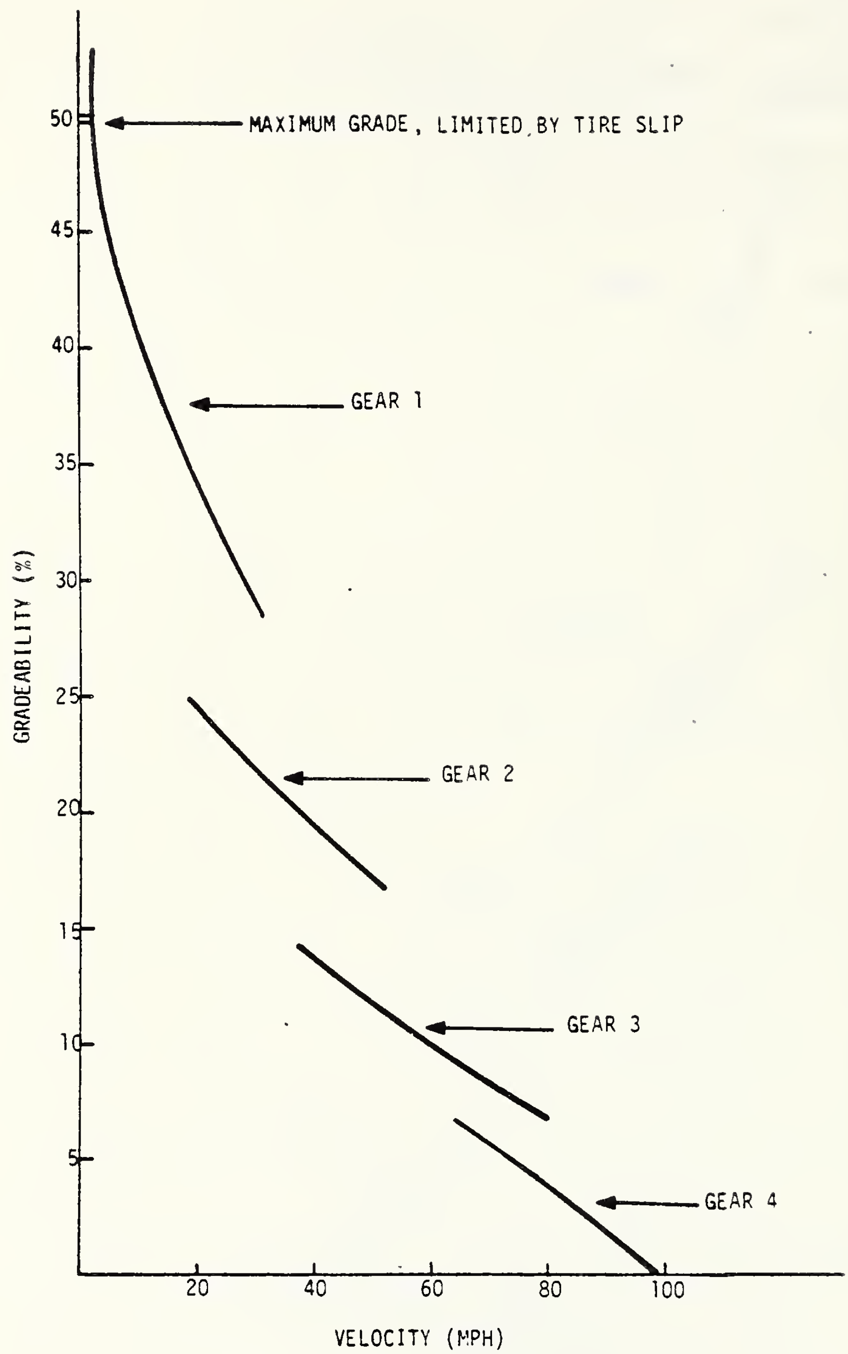


FIGURE 4-12. GRADEABILITY FOR A 3000 lb 130 CID AUTOMOBILE

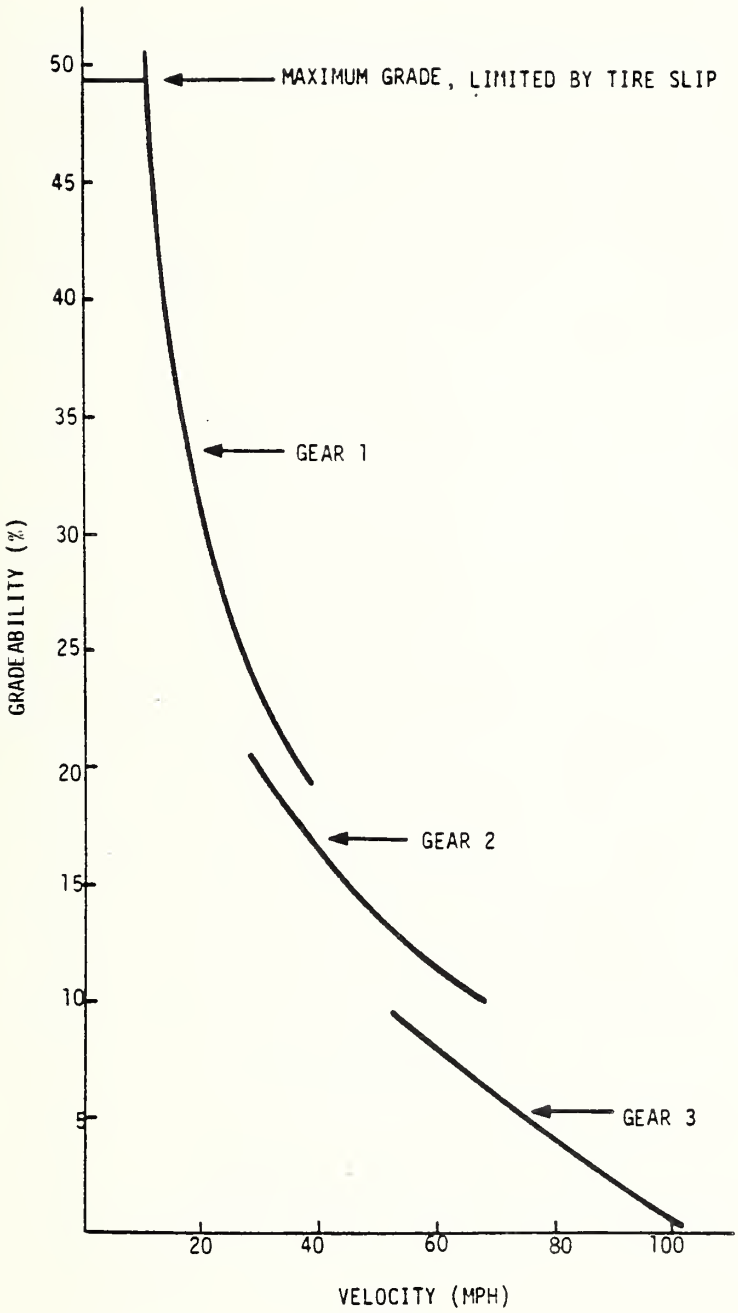


FIGURE 4-13. GRADEABILITY FOR A 3500 lb 231 CID AUTOMOBILE

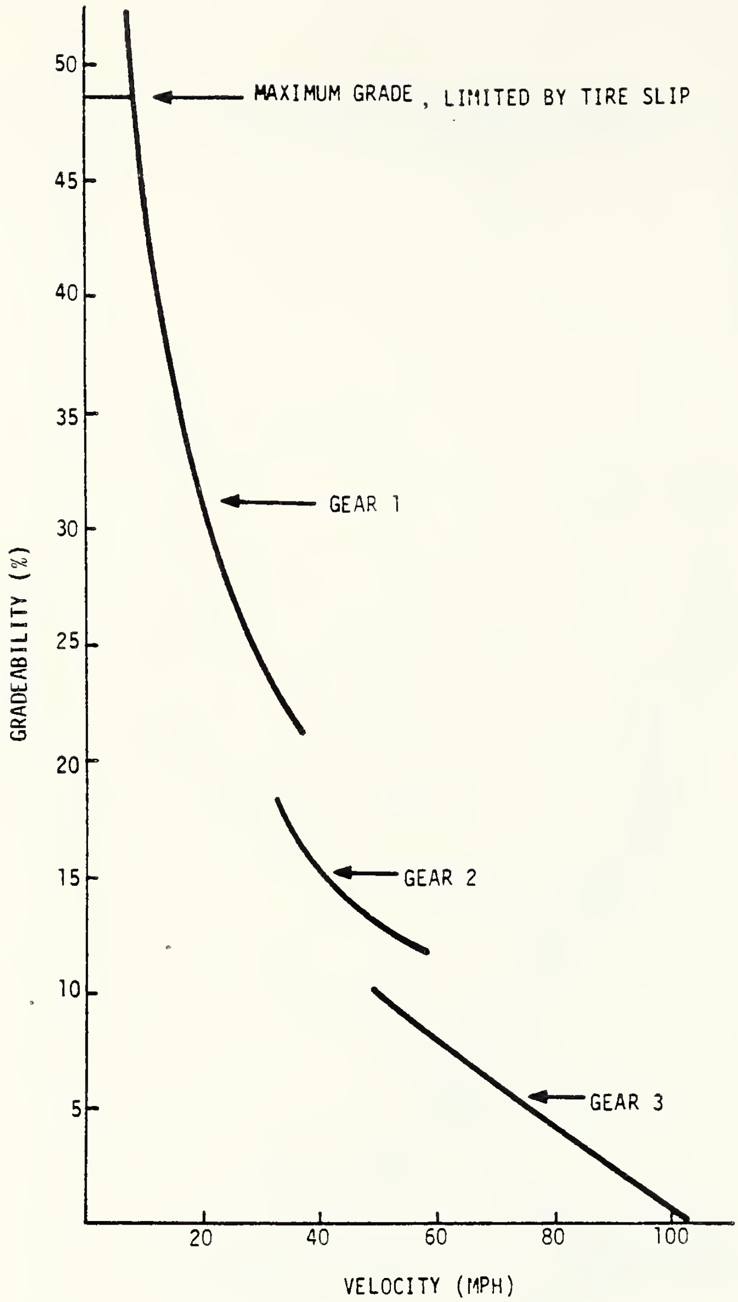


FIGURE 4-14. GRADEABILITY FOR A 4500 lb 318 CID AUTOMOBILE

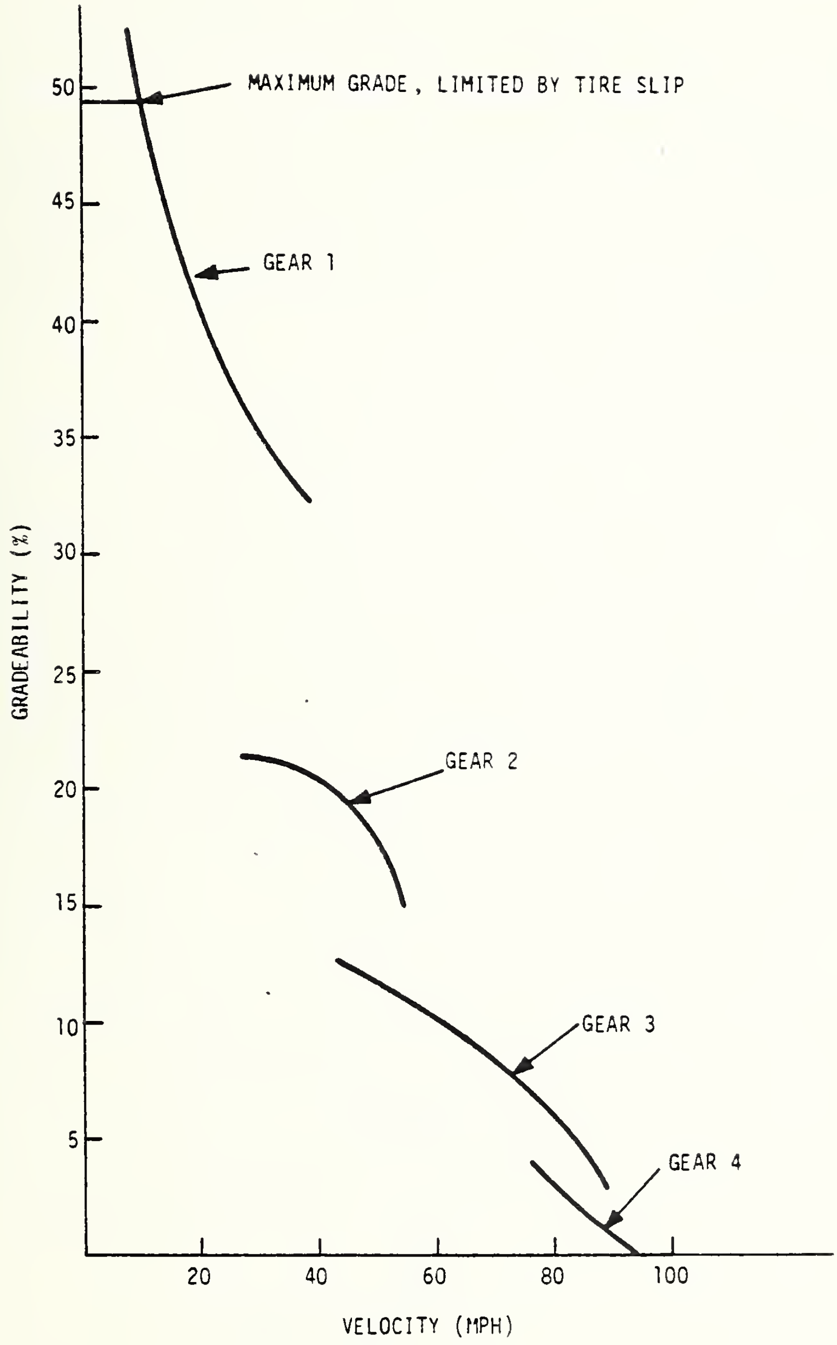


FIGURE 4-15. GRADEABILITY FOR A 3000 lb 140 CID MINI-PICKUP

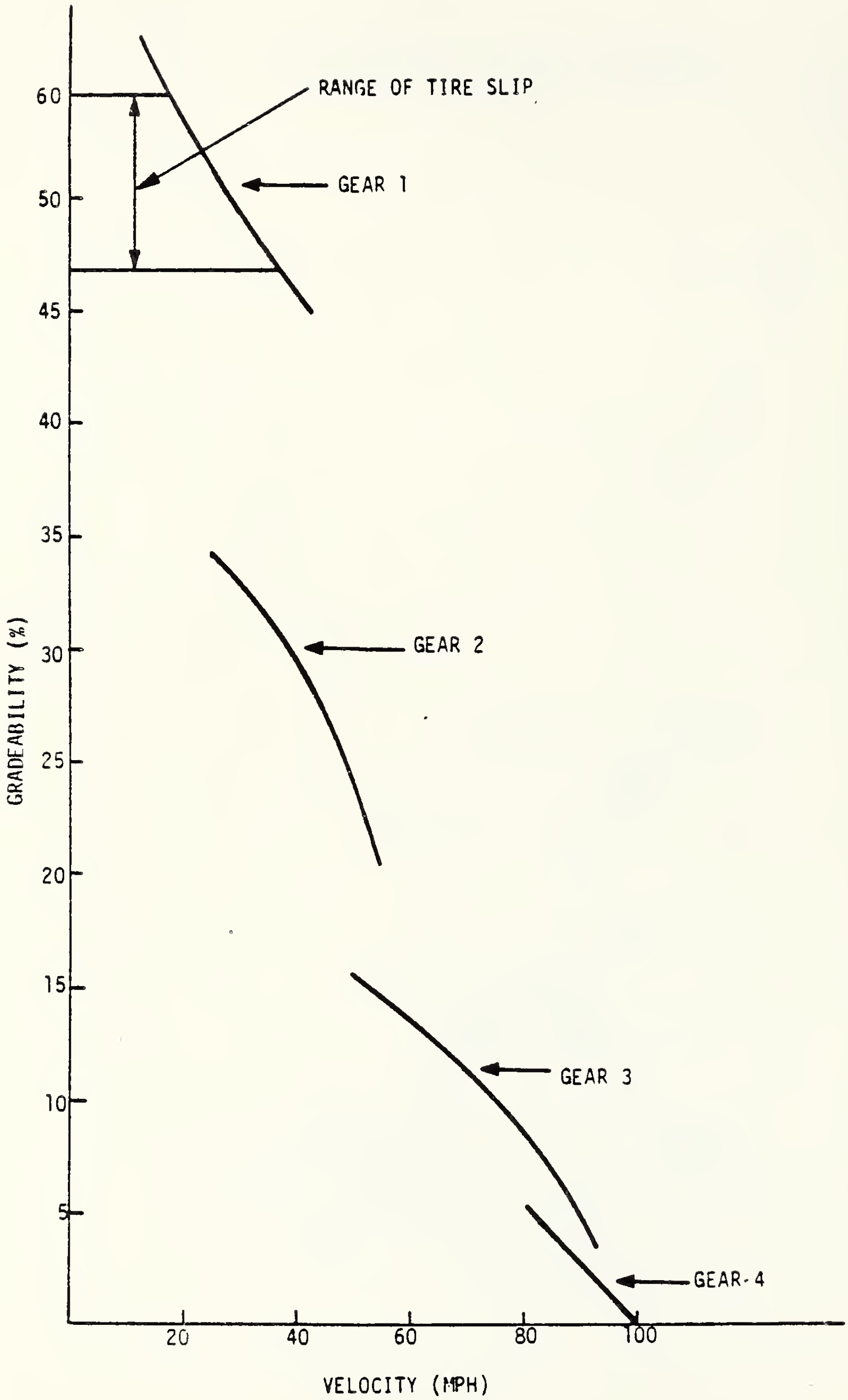


FIGURE 4-16. GRADEABILITY FOR A 4000 LB 351 CID LIGHT TRUCK

acceleration of a vehicle decreases approaching steady-state, minimizing the difference between acceleration and gradeability. This implies that if we are concerned with the gradeability of a vehicle at 55 mph, we can estimate it from the acceleration performance at 55 mph. From the acceleration results it can be seen that aerodynamic drag and rolling resistance reductions have a negligible effect on gradeability.

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

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

TABLE 4-4. GRADEABILITY RESPONSE TO VEHICLE WEIGHT CHANGE

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

1 Limited by tire slip

2 Road tire coefficient different for light-truck

3 For 4 WD gradeability will approach 80%.

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

VEHICLE	REAR AXLE RATIO	GRADEABILITY (% GRADE 55 MPH)
98 CID	3.58 → 4.0	14 → 15
130 CID	3.91 → 4.2	11 → 12
231 CID	2.56 → 3.0	12 → 13
318 CID	2.11 → 3.4	9 → 11
140 CID	3.08 → 4.2	11 → 15
351 CID	3.54 → 4.2	15 → 17

4.4 FUEL ECONOMY

The engine can be considered an energy conversion device which utilizes gasoline to propel a vehicle over a distance. Assuming that gasoline is the beginning of this energy conversion cycle, the energy distribution of a typical automobile[7] can be seen in Figure 4-17. An increase in fuel economy can be achieved by either reducing the energy consumption of the vehicle or by operating the engine more efficiently. The losses to be examined here are those ascribed to the energy consumption of the vehicle; specifically the weight of the vehicle, the aerodynamic drag and the rolling resistance of the tires.

The fuel economy change connected with weight reduction is shown in Figure 4-18. Each improvement in vehicle road load loss not only has the direct effect of increasing the fuel economy, but also has the indirect result of allowing the vehicle to be equipped with a smaller engine. Therefore, each vehicle was equipped with a smaller engine geometrically downsized to maintain equal performance. An example of the fuel economy increase accomplished by maintaining equal performance is shown in Figure 4-19. The increase in fuel economy due to 11 percent weight reduction is 2 percent. This fuel economy change can be increased to 11 percent by also reducing the engine size. The weight reduction of each vehicle is based upon projections for 1985 vehicles. The fuel economy gain, utilizing a geometrically smaller engine, as a result of the weight reduction projected for 1985 in shown in Table 4-6. It can be seen that the sensitivities of the vehicles are fairly constant and near unity indicating that the percent change in weight reduction is accompained by an equivalent change in fuel economy.

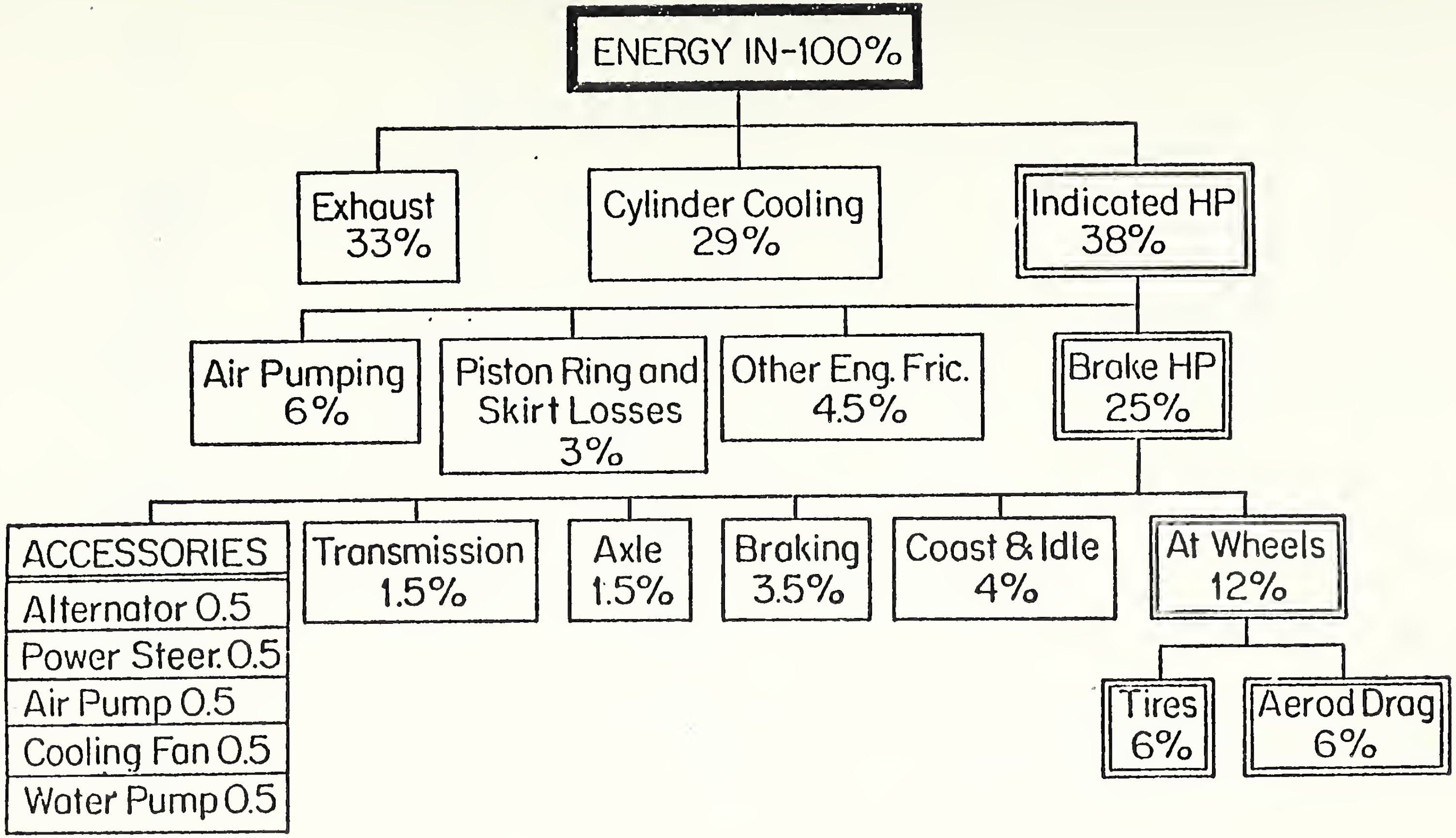


FIGURE 4-17. ENERGY DISTRIBUTION IN PASSENGER CAR DURING EPA CYCLE

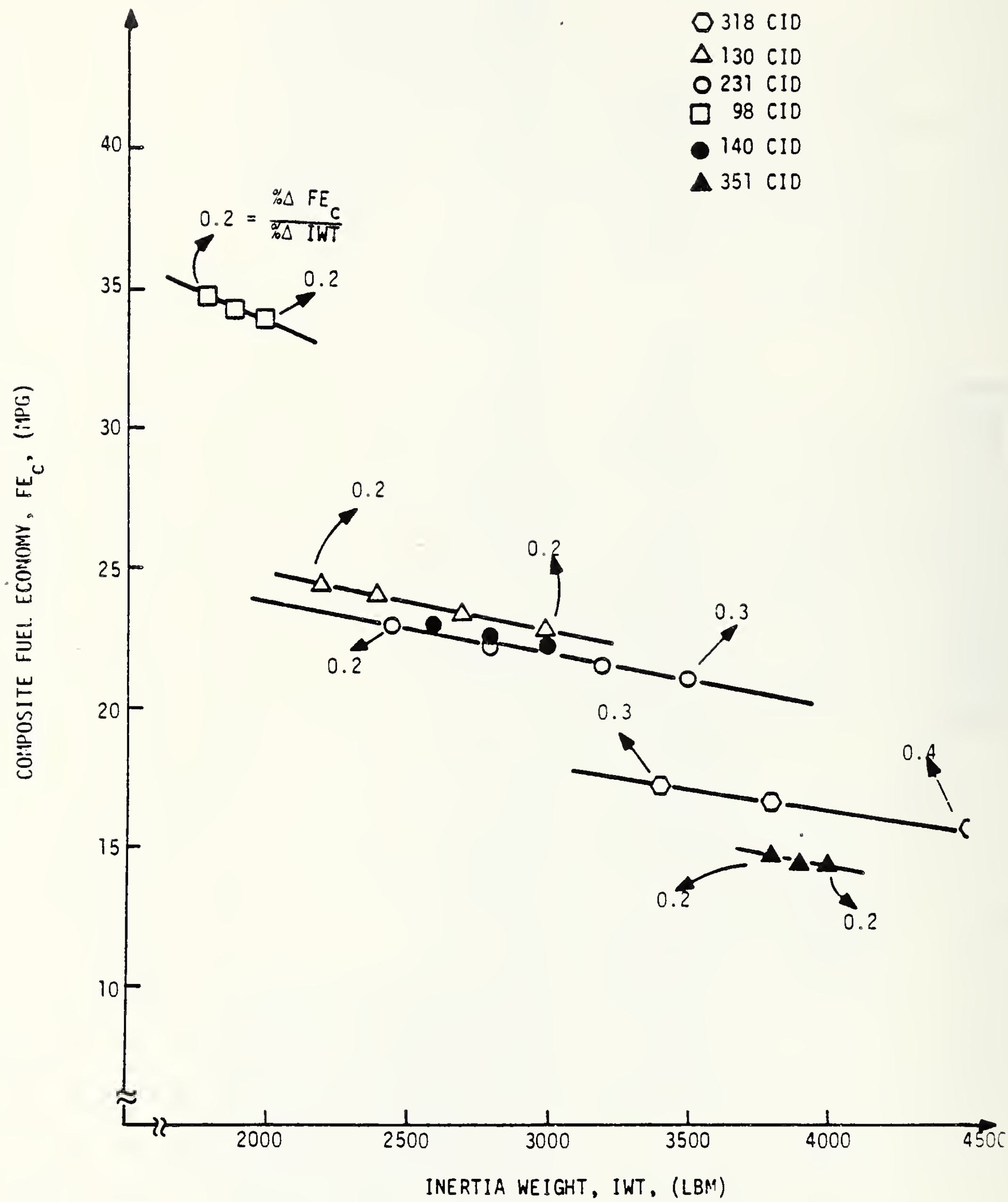


FIGURE 4-18. FUEL ECONOMY AS A FUNCTION OF WEIGHT REDUCTION

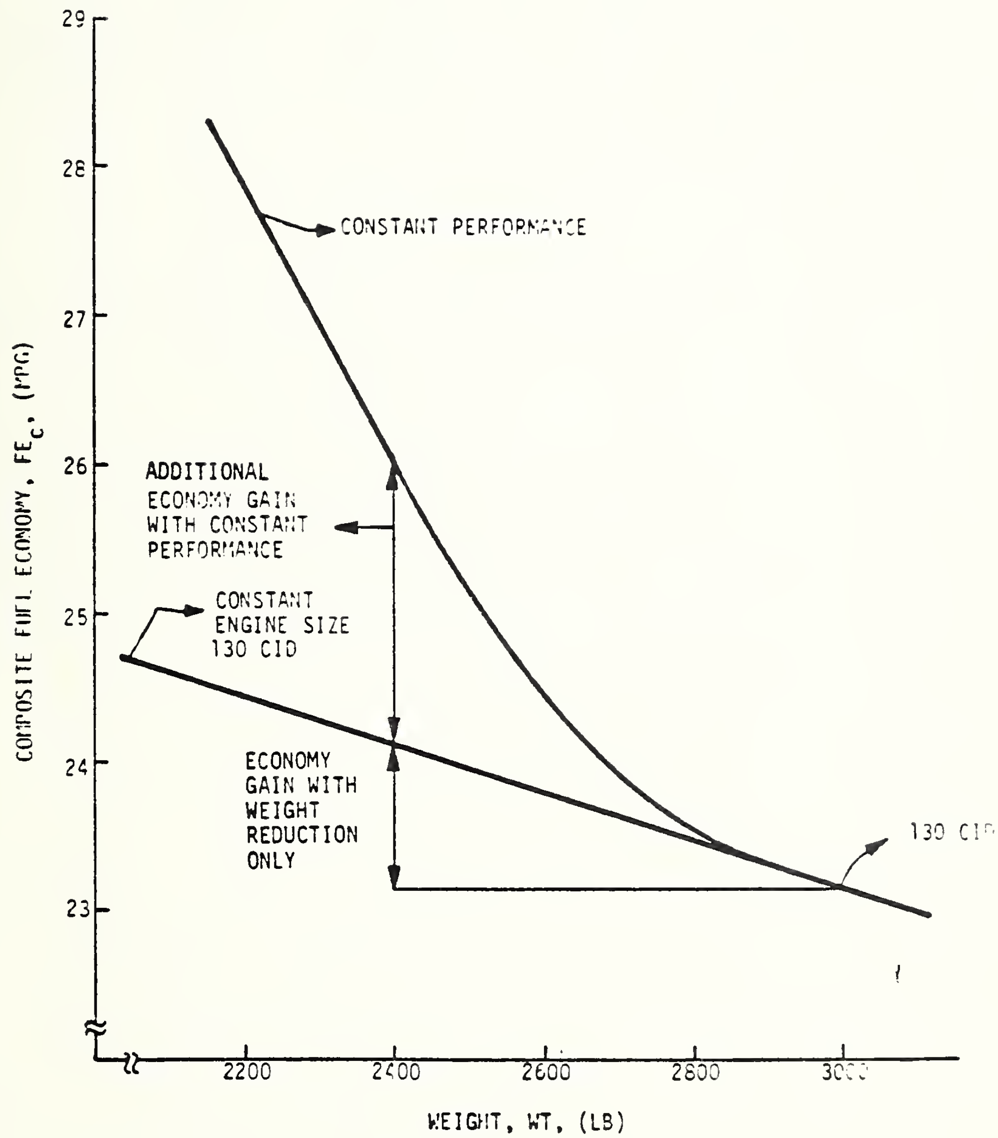


FIGURE 4-19. EFFECT OF WEIGHT REDUCTION ON FUEL ECONOMY FOR 130 CID 3000 lb VEHICLE

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

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

The effect of an aerodynamic drag and a rolling resistance reduction on fuel economy is also considered. Figure 4-20 portrays the fuel economy increase with drag coefficient reduction, while the effect of the rolling resistance coefficient on fuel economy is presented in Figure 4-21. The sensitivities for the extremes are also included. When evaluating these results it should be remembered that the composite fuel economy is based on the EPA urban and highway driving schedules. For example, although a reduction in the aerodynamic drag coefficient may effect the highway fuel economy, it will have a less significant impact on the composite fuel economy because of the urban cycle contribution. In order to apply the full potential of a C_D (aerodynamic drag coefficient) and C_1 (rolling resistance coefficient) reduction, the rear axle ratio is changed to bring the new road load curve into alignment with the original curve.[10] Because the C_1 and C_D reduction provides a relatively smaller improvement in road load losses, it is more realistic to change the rear axle ratio rather than scale the engine as was done in the case of the weight reduction studies. An example of a drag coefficient reduction with equal performance is shown in Figure 4-22. For a 40 percent drag coefficient reduction, the fuel economy improvement is 7.1 percent. By reducing the rear axle ratio, this gain is increased to 11.7 percent. The fuel economy change for a drag coefficient and for a rolling resistance coefficient reduction was shown in Figure 4-20 and Figure 4-21, respectively. The <u>additional</u> fuel economy increase due to a rear axle ratio change is presented in Table 4-7.

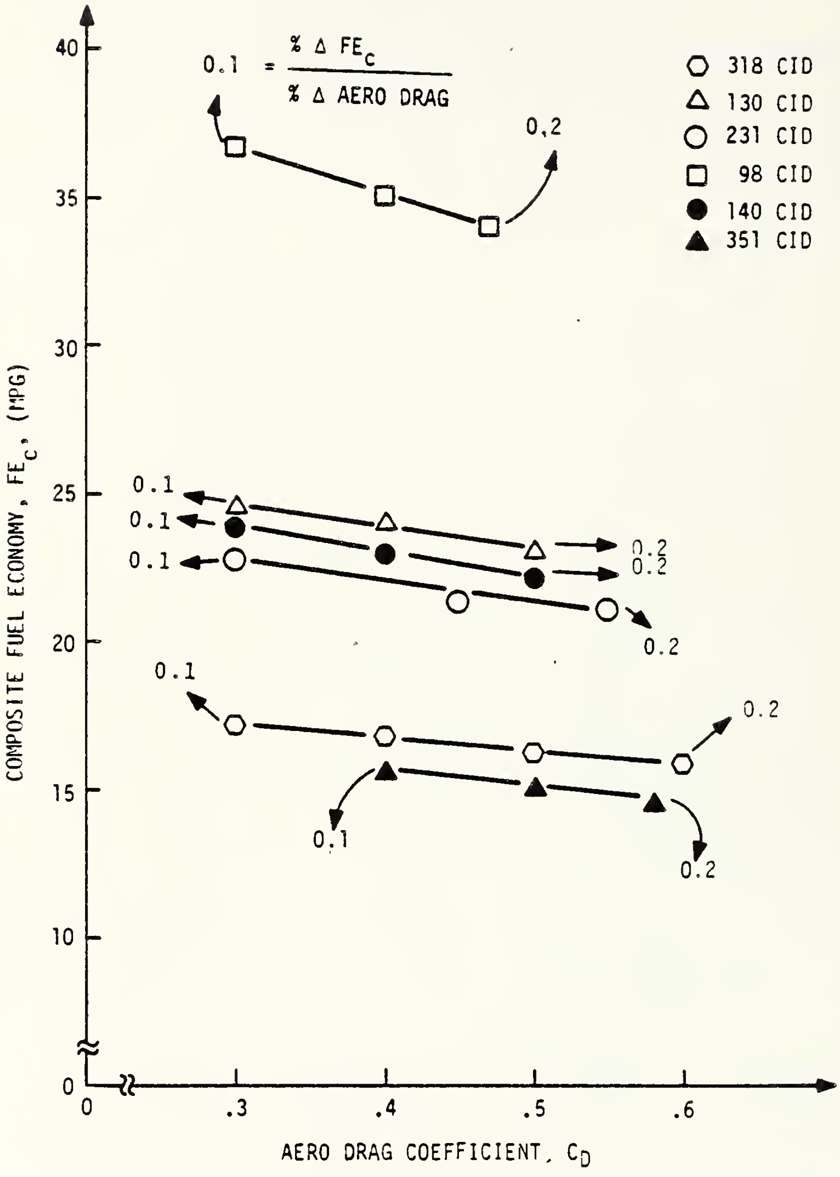


FIGURE 4-20. EFFECT OF AERODYNAMIC DRAG REDUCTION ON FUEL ECONOMY

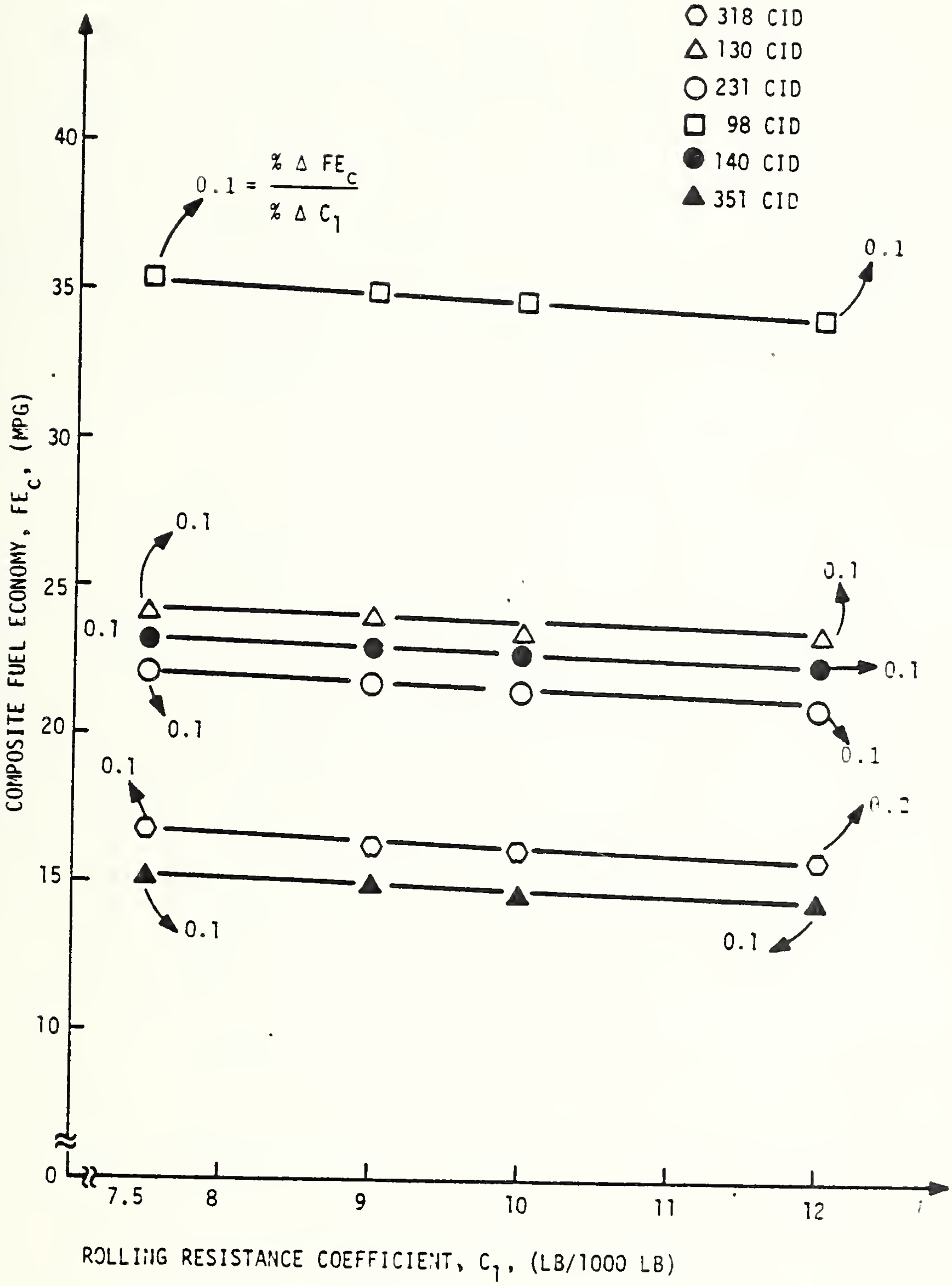


FIGURE 4-21. EFFECT OF ROLLING RESISTANCE COEFFICIENT ON FUEL ECONOMY

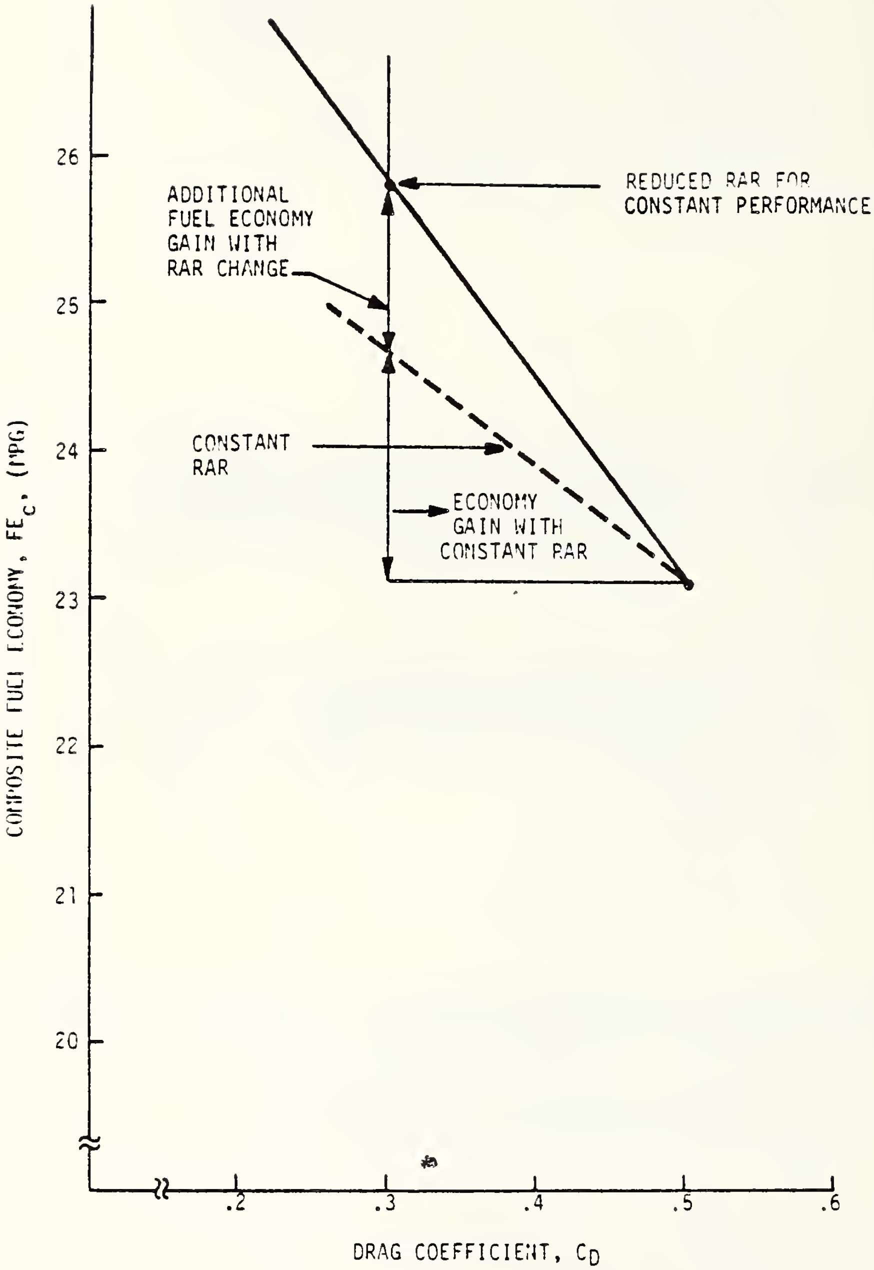


FIGURE 4-22. FUEL ECONOMY AS A FUNCTION OF C_D FOR DIFFERENT AXLE RATIOS FOR 130 CID 3000 lb VEHICLE

The increase in fuel economy as a result of a numerical rear axle ratio reduction is explained by examining the vehicle road load curve on an engine BSFC map. The BSFC is an indication of how efficiently the engine converts fuel into work. By reducing the rear axle ratio, the road load curve is forced to operate at islands of lower BSFC which reduces the fuel flow. The result of this effect is shown in Figure 4-23.

TABLE 4-7. ADDITIONAL FUEL ECONOMY INCREASE ATTRIBUTED TO EQUAL PERFORMANCE FOR AERODYNAMIC DRAG AND ROLLING RESISTANCE COEFFICIENT

ENGINE (CID)	DRAG COEFFICIENT C_D BASELINE/REDUCTION	% FE_c INCREASE	C_l[1] (lb/1000 lb)	% FE_c INCREASE
318	.60/.30	6.5	7.5	3.1
130	.50/.30	4.6	7.5	2.3
231	.55/.30	6.6	7.5	3.0
98	.47/.30	7.9	7.5	3.6
140	.50/.30	8.8	7.5	3.6
351	.50/.30	3.9	7.5	1.6

[1]Baseline C_l=12.0

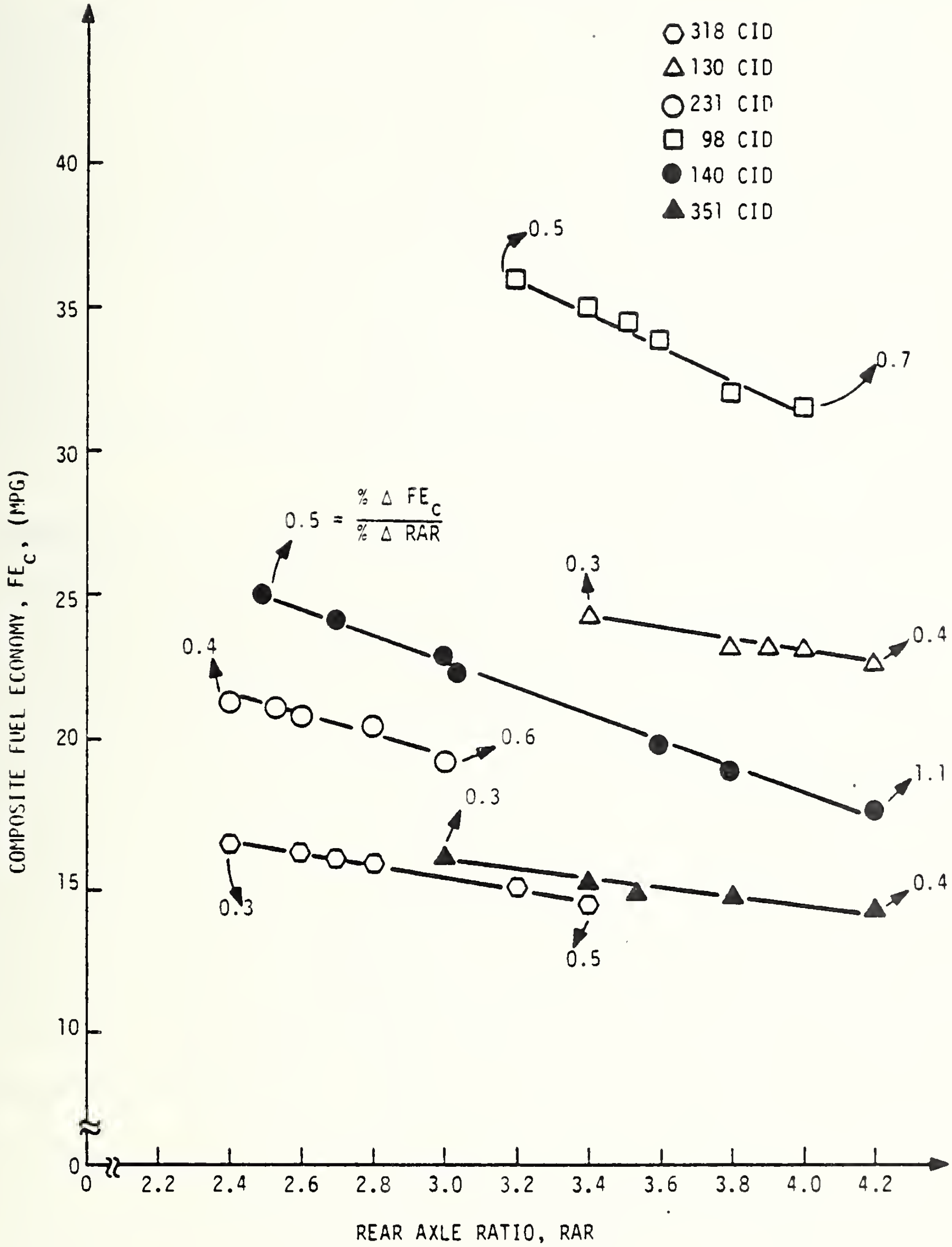


FIGURE 4-23. EFFECT OF REAR AXLE RATIO ON FUEL ECONOMY

4.5 ENGINE

The purpose of this section is to assess the effect of engine variation for a given vehicle weight classification. This has been accomplished by selecting six alternate engines, one for each original weight group as shown in Table 4-8. The only parameter that changes is the engine. The vehicle used for the alternate engine is identical to that used for the first engine.

The 0-60 mph time for the alternate vehicle configuration (identical vehicle - different engine) is superimposed on the original HP/WT vs. 0-60 mph time graph. From Figure 4-24, it can be seen that the curve is a relatively good approximation for estimating 0-60 mph time.

The acceleration (0-60 time) of a vehicle can be estimated by its HP/WT ratio. However, acceleration is not based solely upon the engine peak horsepower curve and the vehicle weight. The potential of a vehicle to accelerate can be estimated from the difference between the road load horsepower and the engine horsepower curves and the maximum acceleration that can be obtained is based upon the engine torque curve. The comparison between the 140 CID engine and the 170 CID engine in identical vehicles illustrates this point. Both have the same HP/WT ratio, yet the 140 CID vehicle has a 0-60 time of 13.7 seconds, which is one second slower than the 170 CID vehicle, as shown in 4-25. The 170 CID engine has a high peaked torque curve, while the 140 CID engine has a relatively low and flat torque curve. A comparison of a WOT acceleration is shown in Figure 4-26. This clearly indicates that the 170 CID engine produces a higher acceleration rate which

TABLE 4-8. COMPARISON OF BASELINE AND ALTERNATE ENGINE

VEHICLE WEIGHT	BASELINE ENGINE		ALTERNATE ENGINE	
	CID	HP/WT	CID	HP/WT
2000	98	.034	98	.034
3000	130	.030	151	.030
3500	231	.027	225	.029
4500	318	.030	301	.028
3000	140	.032	170	.032
4000	351	.040	350	.042

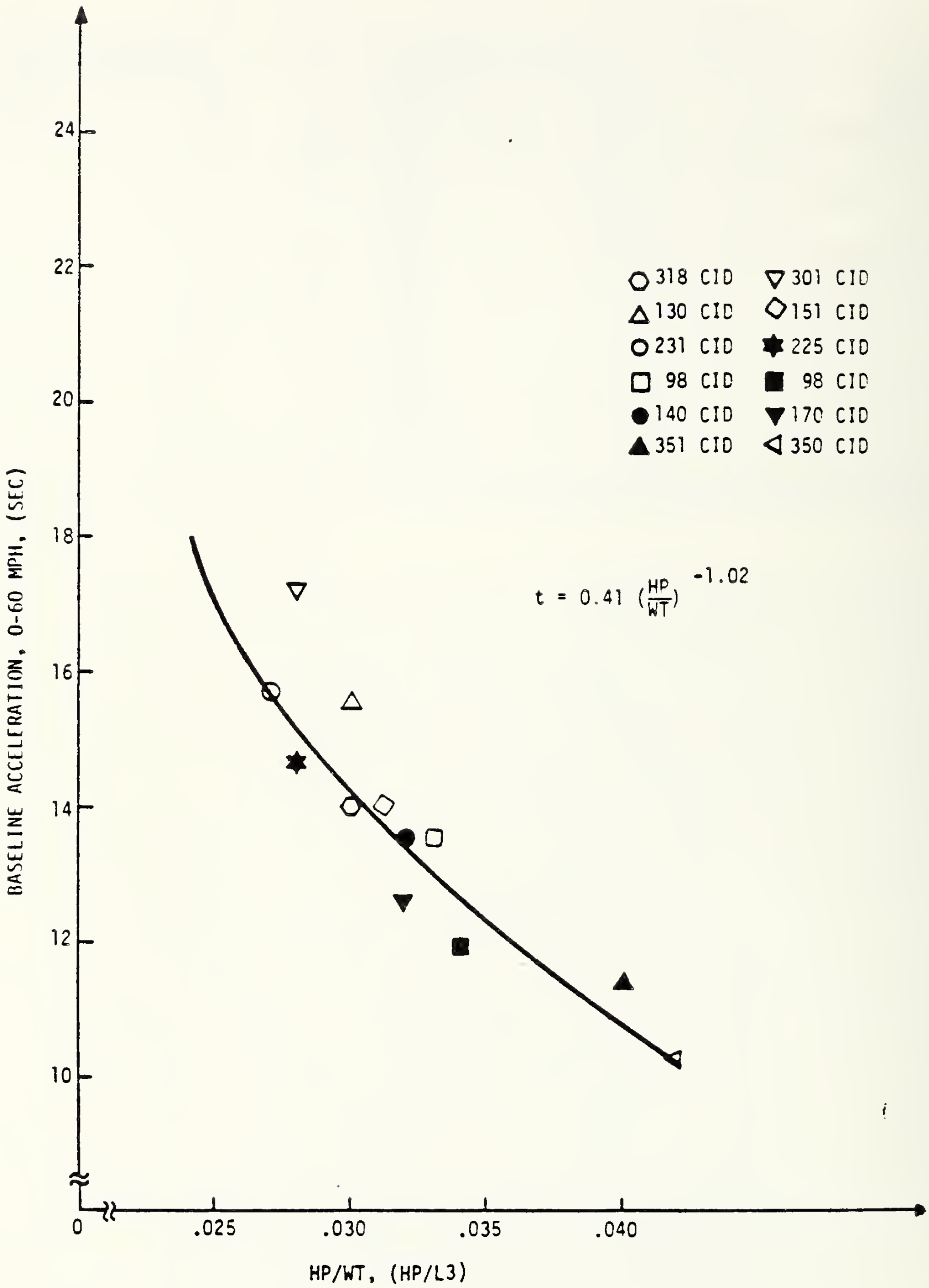


FIGURE 4-24. ACCELERATION AS A FUNCTION OF HP/WT FOR ALL VEHICLES SIMULATED

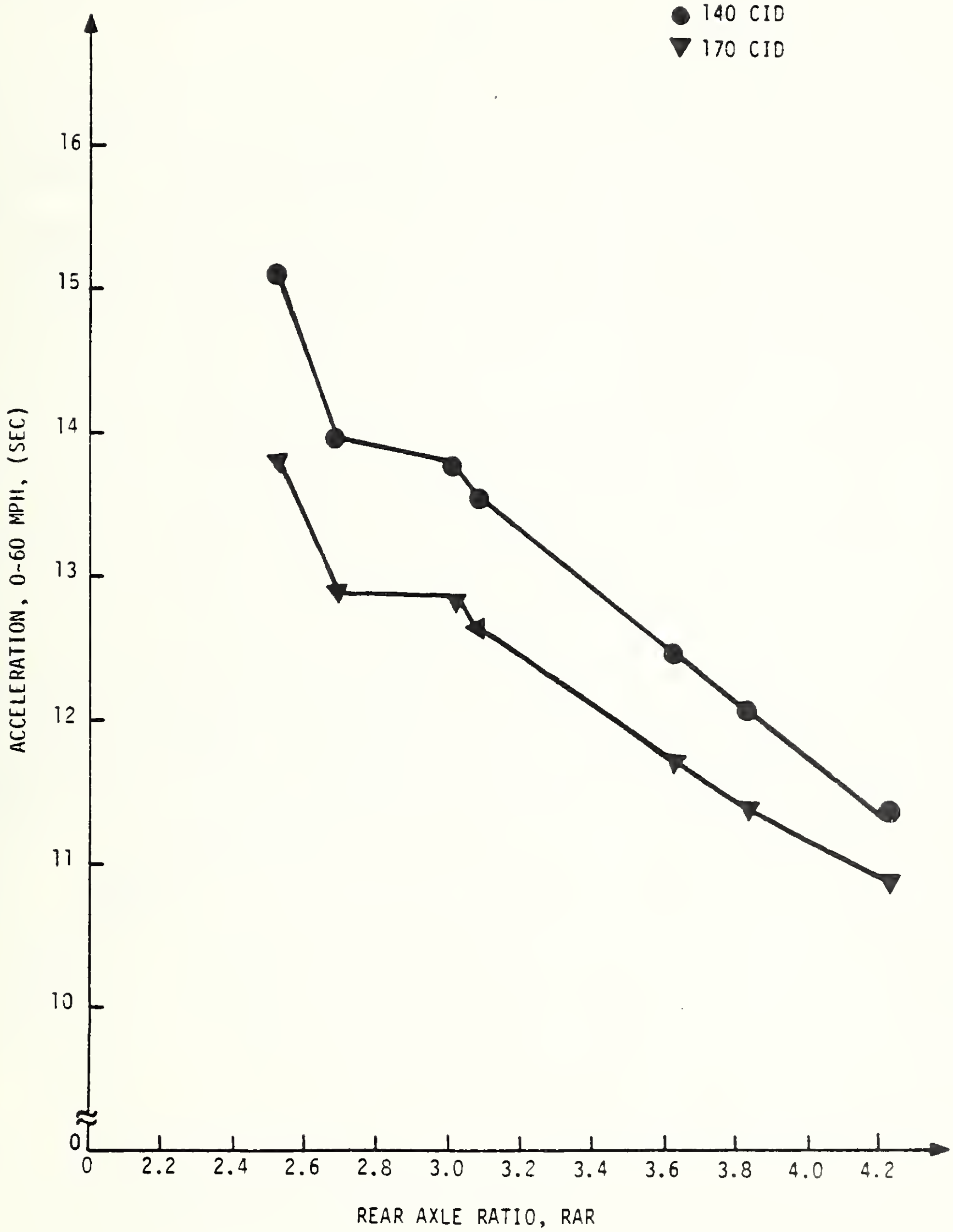


FIGURE 4-25. COMPARISON OF ACCELERATION TIMES FOR THE 170 CID AND 140 CID ENGINES

results in a lower acceleration time. Therefore, if the peak HP/WT ratio does not sufficiently explain the acceleration time, then an examination of the engine torque curve should clarify many questions pertaining to acceleration.

The baseline top speed data points for the alternate engines are superimposed on the original graph of baseline top speed vs. effective area as shown in Figure 4-27. The alternate engine's top speed corresponds to the trend of the original engine's. Because the top speed is based upon the intersection of the road load curve which is identical for identical vehicles and engine horsepower curve, the difference in top speed for identical vehicles is due to the characteristics of the engine curve (i.e., slope).

The fuel economy response to a RAR change and weight reduction is shown in Figure 4-28 and Figure 4-29, respectively. The trends in these figures support the original results presented in Section 4.4.

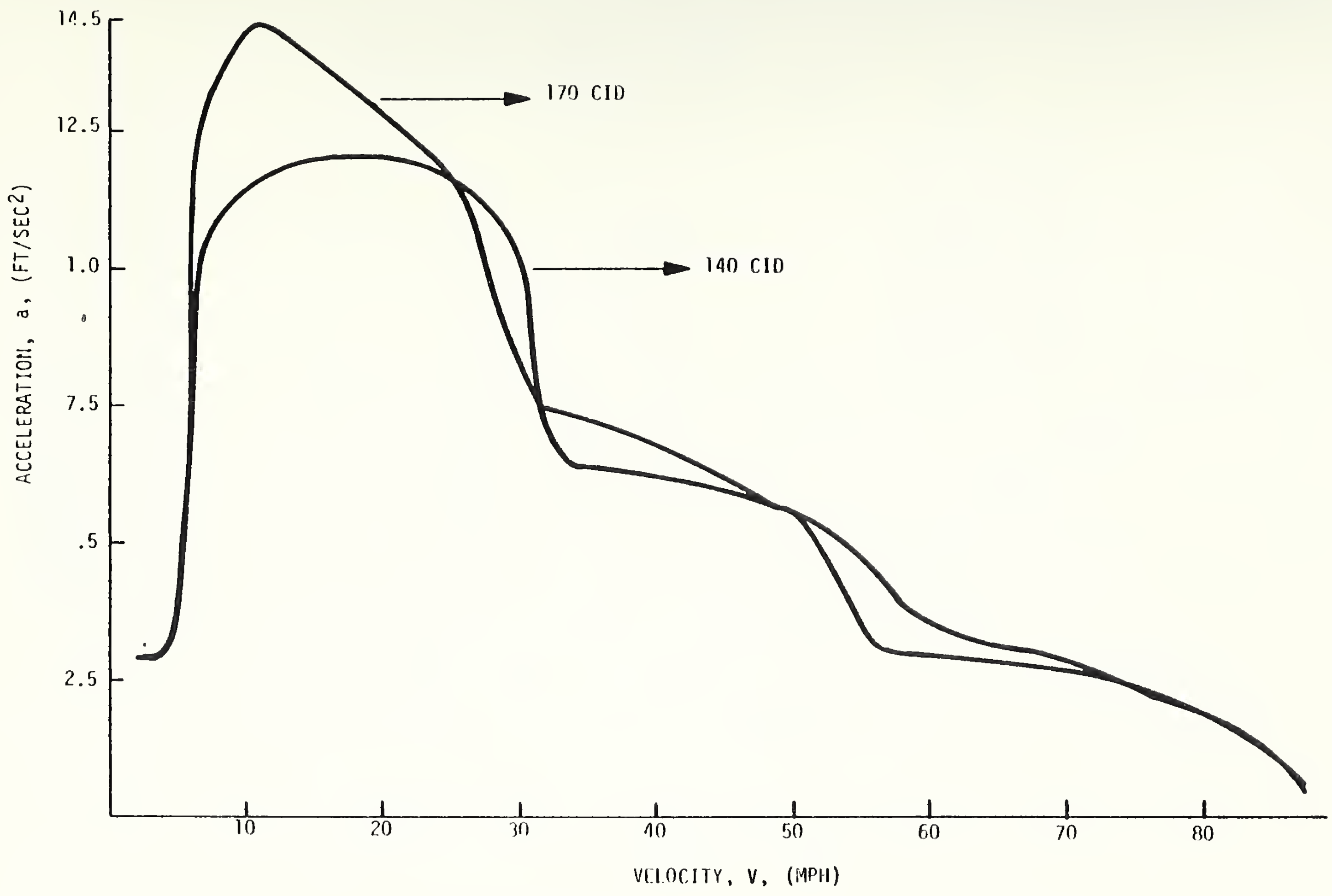


FIGURE 4-26. COMPARISON OF WOT ACCELERATION OF 170 CID AND 140 CID ENGINES IN IDENTICAL VEHICLES

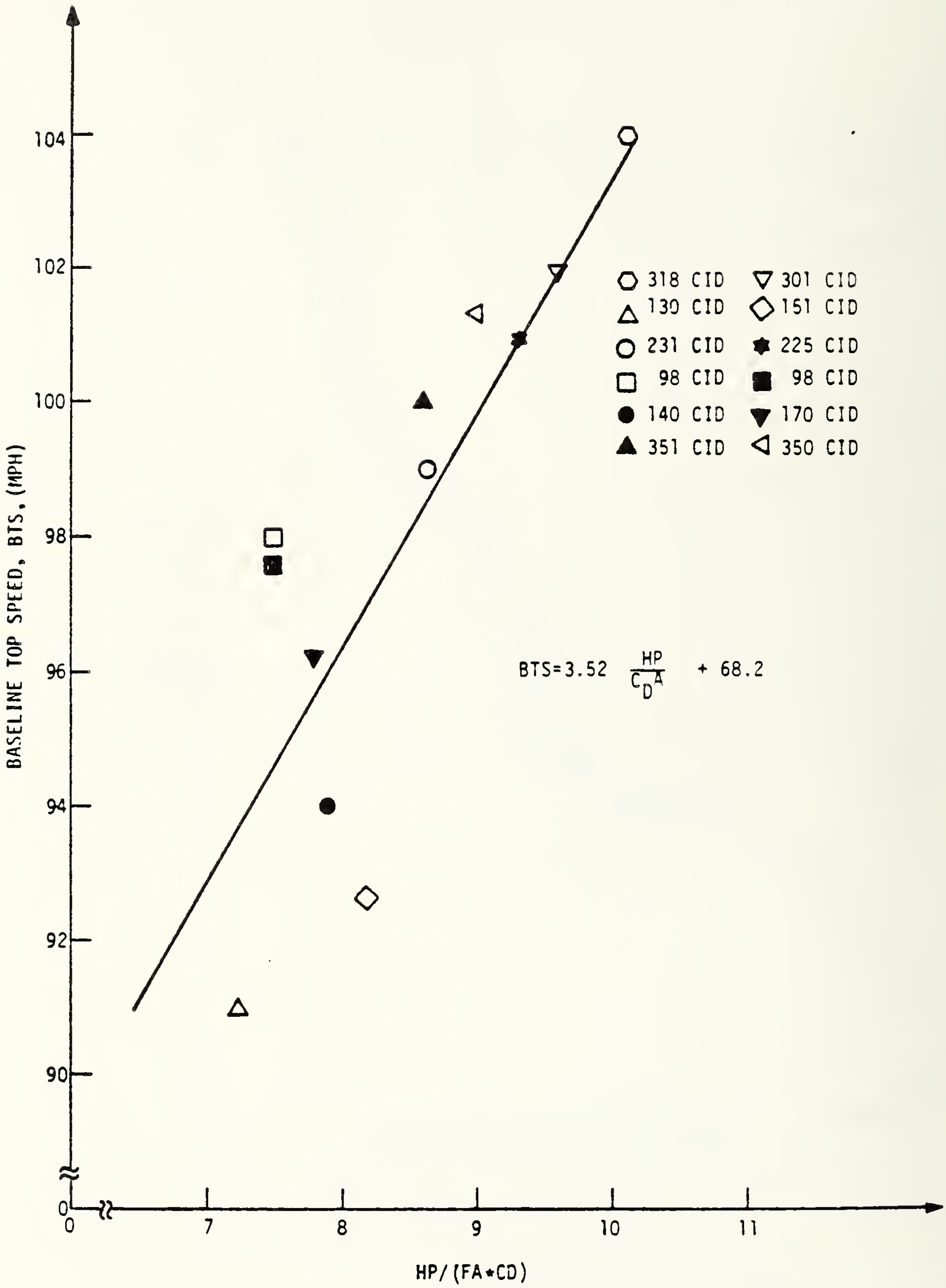


FIGURE 4-27. TOP SPEED AS A FUNCTION OF HORSEPOWER TO EFFECTIVE AREA FOR ALL VEHICLES SIMULATED

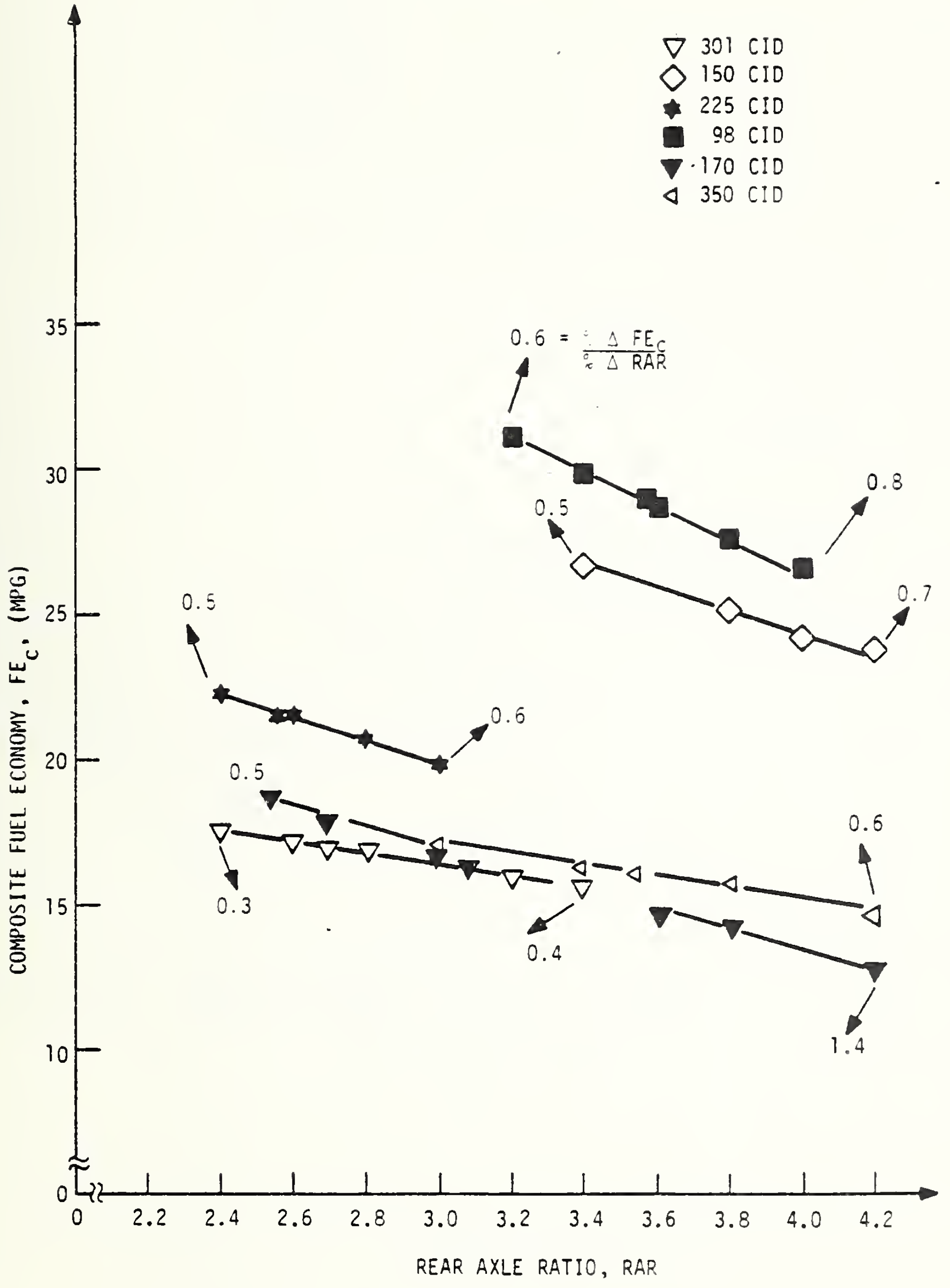


FIGURE 4-28. FUEL ECONOMY RESPONSE TO RAR CHANGE FOR ALTERNATE ENGINES.

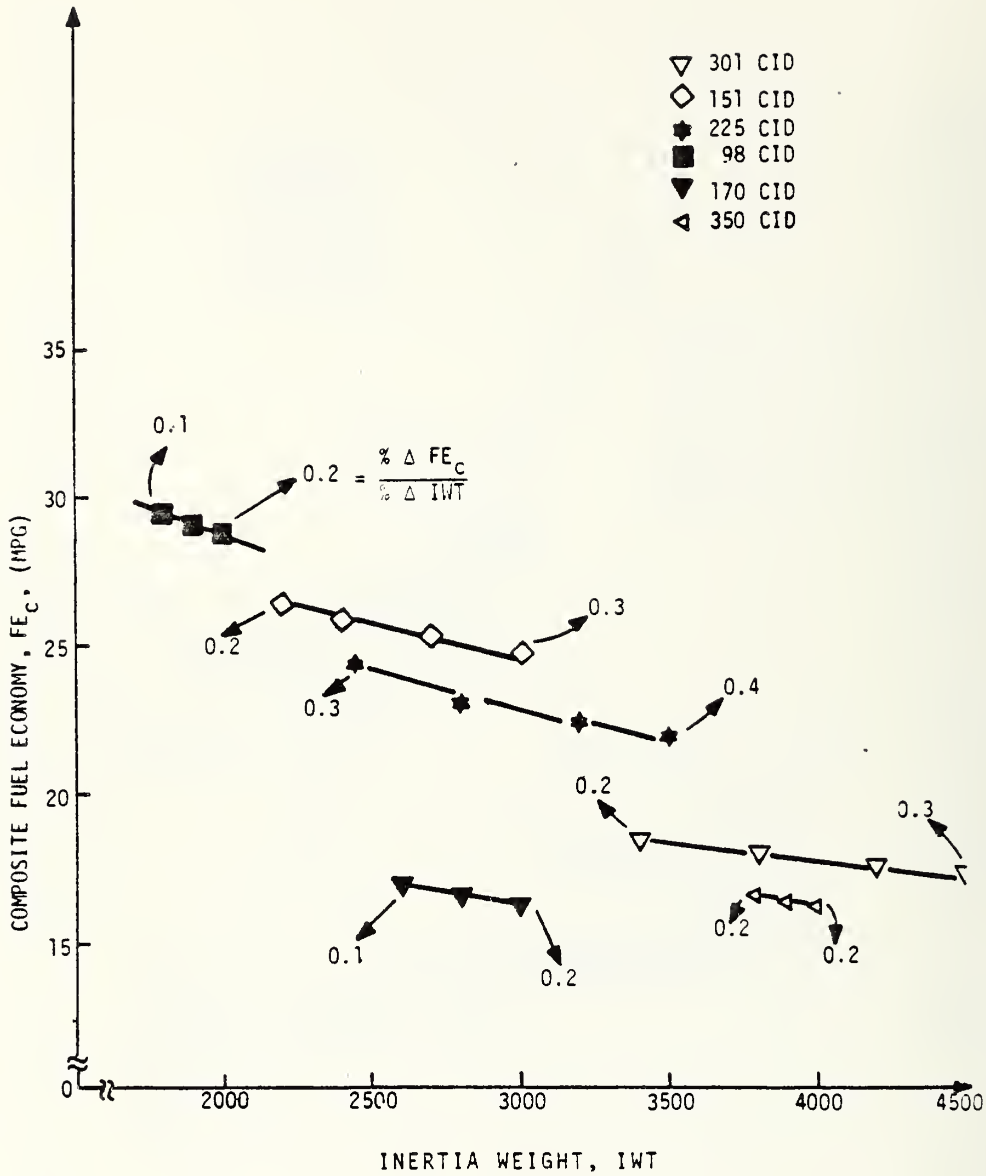


FIGURE 4-29. FUEL ECONOMY RESPONSE TO WEIGHT CHANGE FOR ALTERNATE ENGINES.

5. SUMMARY FINDINGS

Based upon the results presented in this report, the conclusions listed below can be made.

1. Approximately 63-79 percent of the energy expended for the three acceleration drive schedules is due to the mass of the vehicle. Because of this, a 38 percent reduction in rolling resistence or a 40 percent reduction in aerodynamic drag does not significantly affect the acceleration times for these drive schedules.
2. The sensitivity of the 0-60 MPH acceleration time to weight reduction is .7 to 1.2.
3. A numerical increase in the rear axle ratio will decrease the acceleration time. A sensitivity was not calculated because an increase can be caused by the shift points; but, aside from shift points the acceleration time will be reduced.
4. The acceleration time (0-60 mph) can be estimated from the HP/WT of a vehicle.
5. The acceleration time is based upon the difference between the road load and engine HP curves. The maximum acceleration will occur where the difference between the road load and engine torque curve is greatest. For vehicles with an identical HP/WT the difference in acceleration time (0-60 mph) can be explained by the engine torque curves.

6. The top speed is determined by the intersection of the road load and WOT engine HP curves.
7. As a vehicle approaches its top speed, approximately 87 percent of its output energy is used to overcome aerodynamic drag. A 1.6-5.4 percent reduction in the drag coefficient will increase the top speed 1 percent.
8. A rear axle ratio change may increase or decrease the top speed depending upon where the baseline road load curve intersects the engine HP curve. However, the change is not significant enough to use this as a means for reducing top speed.
9. The gradeability is based upon the tractive force delivered to the driving wheels.
10. Vehicle parameters that affect first gear gradeability, but were not examined, include automatic transmission characteristics, clutch dynamics and, most importantly, tire slippage.
11. Tire slippage is a limiting factor for start-up gradeability and changes in vehicle design variables may not affect the gradeability because of this factor.
12. The reduction in vehicle road load loss not only has the direct effect of increasing fuel economy but also the indirect result of allowing the vehicle to be equipped with a smaller engine or a numerically lower rear axle ratio.
13. The sensitivity of fuel economy to weight reduction is .2-.4. If the engine is modified to maintain equal

performance, the sensitivity changes to .7-1.1.

14. The sensitivity of fuel economy to an aerodynamic drag reduction is .1-.2. An additional gain in fuel economy of 3.9-8.8 percent can be made by altering the rear axle ratio.

15. The sensitivity of fuel economy to the rolling resistance coefficient is .1-.2. By changing the rear axle ratio, the fuel economy can be improved an additional 1.6-3.6 percent.

16. The sensitivity of fuel economy to the rear axle ratio is .3-1.1.

17. The effect of engine variation for a given vehicle on performance and fuel economy is not significant.

Although each vehicle should be evaluated separately, some generalized qualitative conclusions regarding vehicle design parameters can be made.

- o Weight reduction
    - reduce acceleration time
    - increase gradeability*
    - top speed not affected
    - fuel economy increase
- o Weight reduction with smaller engine
    - acceleration time remains constant
    - gradeability remains constant
    - top speed decrease
    - fuel economy increase

*provided no tire slip.

- o Numerical rear axle ratio increase
  - increase acceleration
  - top speed increase or decrease
  - increase gradeability*
  - fuel economy decrease
- o Aerodynamic drag reduction
  - acceleration and gradeability not affected
  - top speed increased
  - fuel economy increased
- o Rolling resistance reduction
  - acceleration and gradeability not affected
  - top speed remains constant
  - fuel economy increase.

From the above conclusions it can be seen that the design parameters have a wide impact on evaluation criteria and that there are certain compromises made. For example, in Figure 5-1 the compromise between performance and fuel economy for different rear axle ratios can be seen. Therefore, because of the complex interaction of the vehicle design variables on evaluation criteria the impact of any design change should be carefully appraised against all evaluation criteria.

---

*provided no tire slip.

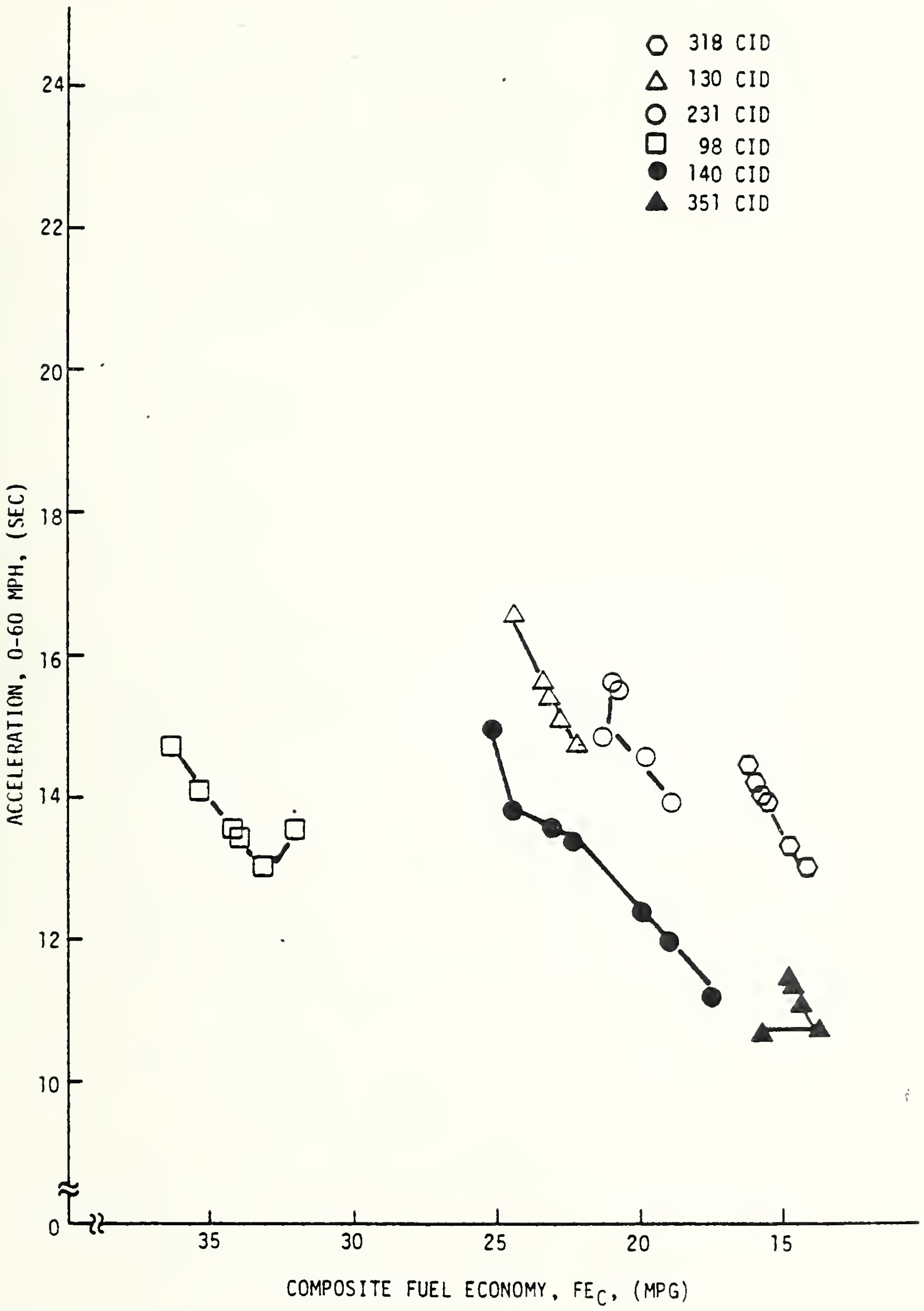


FIGURE 5-1. ACCELERATION (0-60 MPH) TIME VS. FUEL ECONOMY FOR DIFFERENT REAR AXLE RATIOS.

6. REFERENCES

1. Ohanti, X., Takei, M., and Sakamoto, H., "Nissan Full Scale Wind Tunnel - Its Application To Passenger Car Design," SAE Paper 720100, Detroit, MI, January 10-14, 1972.

2. Chemical Engineering, "Energy/Ecology Constraints Keep Triggering Automobile - Propulsion Development," December 23, 1974.

3. Hocho, W.H., et al., "The Optimization of Body Details - A Method For Reducing The Aerodynamic Drag Of Road Vehicles," Volkswagenwerk AG, SAE Paper 760185, February, 1976.

4. Hurter, D.A., and Gott, P.G., "Evaluation Of Automobile Drive-train Components to Improve Fuel Economy," Arthur D. Little, U.S. Department of Transportation, Transportation Systems Center, Report No. DOT-TSC-NHTSA-79-12 (DOT HS-803 840), March, 1979.

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

6. Clark, S.N., and Dodge, R.D., "A Handbook for the Rolling Resistance of Pneumatic Tires," University of Michigan, U.S. Department of Transportation, Transportation Systems Center, Report No. DOT-TSC-NHTSA-79-28, October, 1979.

7. Research Plan for Tire Rolling Resistance, prepared for The Society of Automotive Engineers, Washington, DC, March, 1979.

8. Hsia, H., and Kidd, J.A., "The Weight Reduction Potential of Automobiles and Light Trucks," U.S. Department of Transportation, Transportation Systems Center, Report No. DOT-TSC-NHTSA-79-54, October, 1979.

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

10. Jassen, L.J., and Emmelmann, H.J., "Aerodynamic Improvements, A Great Potential For Better Fuel Economy," SAE Paper 780265, 1978.

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