S-806-047
SC-NHTSA-81-22

The Effect of Light Truck Design Variables on Top Speed, Performance, and Fuel Economy, 1981

Russell W. Zub
Richard P. Meisner

Transportation Systems Center
Cambridge MA 02142

November 1981
Final Report

This document is available to the public through the National Technical Information Service, Springfield, Virginia 22161.

U.S. Department of Transportation
National Highway Traffic Safety Administration

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.

Technical Report Documentation Page

1. Report No.	2. Government Accession No.	3. Recipient's Catalog No.
DOT-HS-806-047		

4. Title and Subtitle	5. Report Date
THE EFFECT OF LIGHT TRUCK DESIGN VARIABLES ON TOP SPEED, PERFORMANCE, AND FUEL ECONOMY, 1981	November 1981
	6. Performing Organization Code: DTS-323
7. Author/s: R.W. Zub, R.P. Meisner	8. Performing Organization Report No.: DOT-TSC-NHTSA-81-22
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): HS273/R2410
	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, October 1980 - September 1981
	14. Sponsoring Agency Code: NRD-13

15. Supplementary Notes

16. Abstract

The effect of vehicle weight, rolling resistance, aerodynamic drag, and driveline configuration on fuel economy and performance for light duty trucks is examined. The effect of lockup and extended gear ratio range is also investigated. The assessment of these vehicle variables on fuel economy and performance is determined by using the Transportation Systems Center's vehicle simulation program, VEHSIM, which predicts fuel economy and performance for vehicle parameter changes. The results indicate fuel economy and performance trends which can be used to project future improvements.

17. Key Words	18. Distribution Statement
Acceleration, Top Speed, Fuel Economy, Gradeability, Light Truck, Tire Pressure, Accessory	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	112	

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

PREFACE

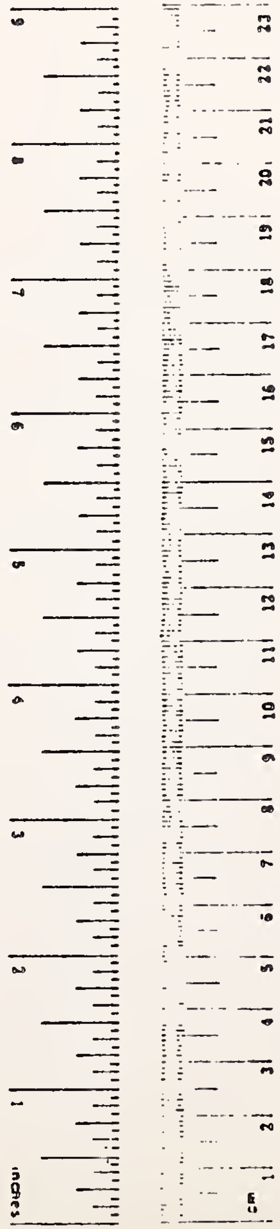
The Energy Technology Branch, Transportation Systems Center, would like to thank Anne Yang, MIT summer intern, for help in the preparation of this document.

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

*1 in = 2.54 (exactly). For other exact conversions and more detailed tables, see NBS Misc. Publ. 286, Units of Weights and Measures, Price $2.25, SD Catalog No. C13.10:286.



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

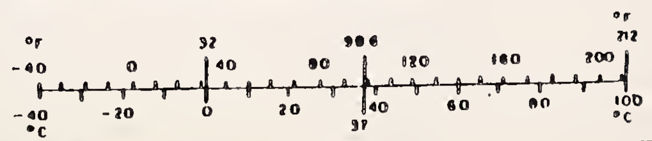


TABLE OF CONTENTS

LIST OF ILLUSTRATIONS

LIST OF TABLES

LIST OF TABLES (CONT.)

NOMENCLATURE

A:	Frontal area (ft^2)
AR:	Axle ratio
BSFC:	Brake specific fuel consumption (lb/hp-hr)
BTS:	Baseline top speed (mph)
C_D:	Aerodynamic drag coefficient
CID:	Displacement ($in.^3$)
C_p:	Pressure sensitivity
C_1:	Rolling resistance coefficient (lb/1000 lb)
F_E:	Composite fuel economy (mpg)
F_r:	Rolling resistance force (lbf)
F_z:	Load on tire (lbf)
grr:	Gear ratio range
GVM:	Gross vehicle weight (lb)
GVWR:	Gross vehicle weight rating (lb)
HP:	Maximum engine horsepower (hp)
IWT:	Inertia weight (lb)
K_{out}:	Output capacity factor
ℓ:	Displacement (liter)
MPH:	Speed (mph)
N/V:	Engine speed/vehicle speed (rpm/mph)
P:	Inflation pressure ($lb/in.^2$)
PAU:	Power absorption unit (hp)
PSI:	Pressure ($lb/in.^2$)
RAR:	Rear axle ratio
RPM_{out}:	Output speed (rpm)

$\overline{S}$:	Average speed (mph)
t:	Time (sec)
T_{qout}:	Output torque (lb-ft)
T_{qw}:	Torque at drive wheels (lb-ft)
TS:	Top speed (mph)
WOT:	Wide open throttle
WT:	Vehicle weight (lb)
V:	Vehicle velocity (mph)
ρf:	Fuel density (lbm/ft^3).

EXECUTIVE SUMMARY

Based on the vehicles simulated for this report the following summary findings are made. Because of the wide range of some of the sensitivities presented, the reader is advised to check the details of the particular vehicles simulated and find the final and reference vehicle variable before using an average sensitivity value. For example, because tire pressure is not linear with fuel economy, the fuel economy increase from a 10 percent increase in tire pressure is not equivalent (percent) to a fuel economy decrease resulting from a 10 percent decrease in tire pressure. Also, all references made to fuel economy are implied to be composite fuel economy.

1. Fuel economy and performance sensitivities for light trucks are as follows:

   - The effect, $\frac{\%\Delta FE}{\%\Delta IWT}$, of a weight increase (to maximum payload) on fuel economy is 0.32 to 0.63

   - The effect, $\frac{\%\Delta FE}{\%\Delta AR}$, of axle ratio on fuel economy is 0.30 to 0.47

   - The sensitivity, $\frac{\%\Delta FE}{\%\Delta PAU}$, of fuel economy to dynamometer power absorption unit (PAU) is 0.23 to 0.27.

   - The sensitivity, $\frac{\%\Delta FE}{\%\Delta C_1}$, of the rolling resistance coefficient on fuel economy is 0.06 to 0.14.

   - Modifying the drivetrain to maintain equal performance at large aerodynamic and rolling resistance reductions can increase the fuel economy by

     13 to 16% for aerodynamic reduction

     3 to 7% for rolling resistance reduction.

2. The performance (0-60 mph time) of the light trucks studied herein closely follows the equation $t = 0.41 \left(\frac{HP}{WT}\right)^{-1.02}$ presented earlier for automobiles.

3. Fuel economy can be improved through the use of transmission lockup and by extending the gear ratio range. Results are presented graphically (Section 4) which illustrate the potential improvement using these two techniques.

4. The effect of vehicle variables on fuel economy at a constant 55 mph was studied. The resultant range of sensitivities for a cross section of vehicles is:

   - Axle ratio – 0.10 to 0.56, $\frac{\%\Delta FE}{\%\Delta AR}$
   - Weight – 0.14 to 0.32, $\frac{\%\Delta FE}{\%\Delta IWT}$
   - Aero drag – 0.37 to 0.75, $\frac{\%\Delta FE}{\%\Delta C_D}$
   - Displacement – 0.24 to 0.52, $\frac{\%\Delta FE}{\%\Delta CID}$
   - Tire pressure – 0.03 to 0.08, $\frac{\%\Delta FE}{\%\Delta PSI}$
   - The fuel economy decrease from accessory use is

     7-18% for 100% duty cycle of air conditioner

     1.4-2.5% for nighttime use of lights

   - The effect of various drive schedules is presented in Section 5.8. The fuel economy change ranges from +5 to -22 percent.

5. The effect of accessories on fuel economy for a cross section of vehicles was studied. The effect of air conditioning on fuel economy over the EPA urban cycle is 1.6 to 2.2 percent.

6. Fuel economy sensitivity values for two diesel vehicles are presented in Section 7. The values are:

- N/V change: 0.37 to 0.46, $\frac{\%\Delta FE}{\%\Delta N/V}$.
- PAU change: 0.20 to 0.25, $\frac{\%\Delta FE}{\%\Delta PAU}$
- Weight change: 0.27 to 0.36, $\frac{\%\Delta FE}{\%\Delta WT}$
- Engine displacement change: 0.36 to 0.46, $\frac{\%\Delta FE}{\%\Delta CID}$.

7. The effect of tire pressure on rolling resistance and fuel economy for a light truck is presented graphically (Section 8). The sensitivity for seven automobiles and light trucks is

$$0.03 \text{ to } 0.05 \ \frac{\%\Delta FE}{\%\Delta PSI} .$$

1. INTRODUCTION

The purpose of this document is to examine the effects of vehicle design variables on top speed, performance, and fuel economy. This report is an update of a previous report[1*] published in 1979 and is not intended to replace but to supplement the earlier version. Because the analysis of vehicle systems on fuel economy and performance is fundamental and remains applicable for this updated document, the reader is advised to review the analysis presented last year to facilitate the comprehension of this years results.

The main emphasis of this document is to examine sensitivities of light duty trucks to fuel economy and performance. This analysis is presented in Section 3. The effect of a lockup torque converter and the extension of the gear ratio range is also studied and the results are given in Section 4. In order to improve fuel economy, the manufacturers have created various schemes of lockup torque converters for automatics and have extended the gear ratio range for manual transmissions.

Because of the national 55 mph speed limit, an analysis of vehicle design variables at a constant 55 mph is presented in Section 5. Other studies include the effect of nighttime alternator loads and air conditioning on fuel economy, the effect of increased tire pressure on fuel economy, and an examination of diesel fuel economy sensitivities.

*Superscripts indicate references listed in Section 9.

2. BASELINE SELECTION

The baseline vehicles used for the previous design variable report[1] and this report are listed in Table 2-1. The addition of the light duty trucks required three new baseline light duty trucks which are shown in Table 2-2. Unless indicated, such as in Sections 4 and 7, all the vehicles used for this study are from Table 2-1 and 2-2 and are identified by engine size or vehicle weight. The selection of the baseline vehicles is intended to represent a cross-section of current production vehicles and will be replaced by more current vehicles as new engines are tested for vehicle simulation application.

TABLE 2-1. BASELINE VEHICLES

VEHICLE	WEIGHT (lbm)	ENGINE (ℓ)	AXLE RATIO	TRANSMISSION	HP/WT
AUTOMOBILE	2000	1.6	3.58	M4	.034
AUTOMOBILE	3000	2.1	3.91	M5	.030
AUTOMOBILE	3500	3.8	2.56	A3	.027
AUTOMOBILE	4500	5.2	2.71	A3	.030
MINI-PICKUP	3000	2.3	3.08	M4	.032
LIGHT DUTY PICKUP	4000	5.8	3.54	M4	.040

NOTES: A3 Automatic 3-speed

M4 Manual 4-speed

M5 Manual 5-Speed

TABLE 2-2. BASELINE LIGHT DUTY TRUCKS

ENGINE (ℓ)	WEIGHT (TEST/GVW) (LB)	TRANSMISSION	PAU (HP)	AXLE RATIO
3.7	3900/5500	M3	16.5	3.55
4.1	4750/6400	M3	17.6	3.73
5.7	5250/7500	M4	17.0	3.54

3. LIGHT DUTY TRUCKS

3.1 ANALYSIS

The purpose of Section 3 is to examine the effect of weight, axle ratio, power absorption unit, and rolling resistance coefficient on performance and fuel economy for light trucks. The performance calculations are at gross vehicle weight rating (GVWR) and the fuel economy computations at inertia test weight (IW). The dependent performance variables, acceleration, top speed, and gradeability were examined extensively in a previous report[1] and will not be presented again since their fundamental explanations remain applicable. However, some additional technical explanations are presented to clarify any ambiguity associated with the original presentation. These explanations include gradeability calculations for a fully loaded light truck and a comparison of different gradeability equations.

The fuel economy and performance results present trends for a range of vehicle variables. These trends are quantified by sensitivities which are used to project the effect of vehicle variables on fuel economy and performance. Because of the generic nature of the results they can be used to project performance and fuel economy trends for a variety of vehicles. For example, if a vehicle being studied has a larger aerodynamic loading than a baseline vehicle presented in this report, then the Δmpg or Δperformance for the two vehicles can be calculated from the graphs of aerodynamic horsepower versus fuel economy and performance since the graphs include both values of aerodynamic loadings. This difference (Δ) is then added to the baseline fuel economy or performance to establish a new baseline fuel economy or performance for this particular vehicle. This methodology can also be used for the other vehicle variables examined.

3.2 ACCELERATION

The acceleration capability of the three light trucks is delineated by three drive schedules; the distance covered in 0 to 5 sec, the time from 0 to 60 mph and the time from 40 to 60 mph. All three schedules require the engine to operate at wide open throttle (WOT) to assure maximum performance. The mass of the vehicle is the most significant vehicle parameter when determining acceleration performance. The relationship between the weight of the vehicle and the 0 to 60 mph time is shown in Figure 3-1. The maximum weight of each vehicle is the gross vehicle weight rating (GVWR) and this is reduced to the inertia weight (IWT). The difference between the two weights is the payload capacity of the vehicle. The 0 to 60 mph time exhibits a linear relationship with weight reduction and the sensitivity of reducing the weight from GVWR to IWT with respect to 0 to 60 mph time is indicated in Figure 3-1. The inertia weight used here is the simulation weight input to VEHSIM.

The effect of axle ratio on the drive schedules is shown in Figures 3-2, 3-3 and 3-4. For all cases, as the axle ratio is reduced (numerically increased) the performance of the vehicle improves. This is due to more torque being delivered to the drive wheels because the axle ratio acts as a torque multiplier from the engine to the wheels. Figure 3-2 portrays a linear relationship of acceleration time and axle ratio. The sensitivities for an increase and decrease from the base case are also included. A further numerical decrease in axle ratio, although not shown. will begin to significantly decrease performance, eventually becoming asymptotic. This is due to the road load curve being shifted to the right of the WOT engine torque curve.

Sensitivities for the passing time and 0 to 5 sec distance were not included because of the discontinuity of the curves. These discontinuities are caused by the shift points, but independent of these shift points the overall trend is to increase the distance in 0 to 5 sec and decrease the time from 40 to 60 mph for numerical increases in axle ratio.

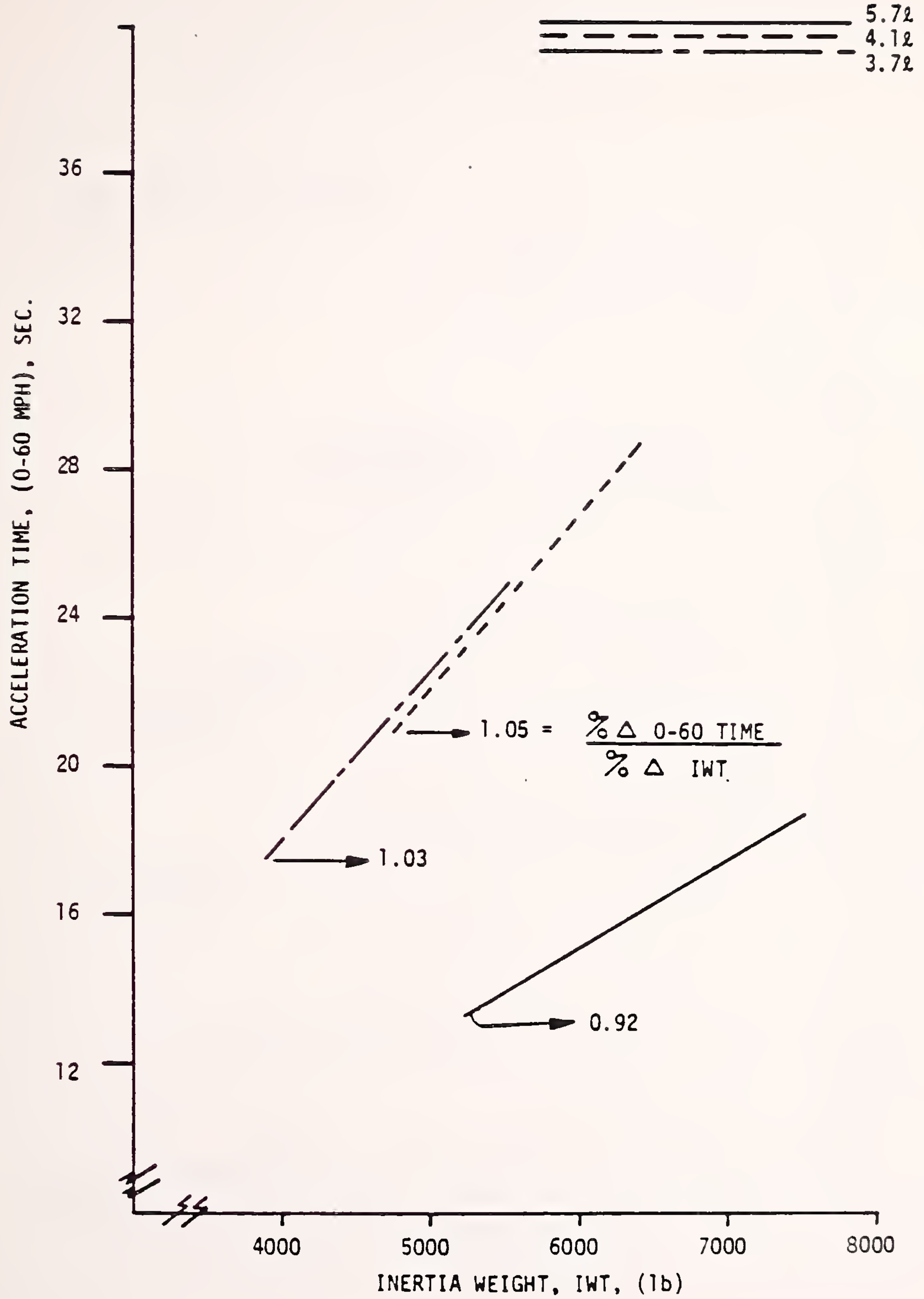


FIGURE 3-1. ACCELERATION TIME (0 TO 60 MPH) AS A FUNCTION OF INERTIA WEIGHT

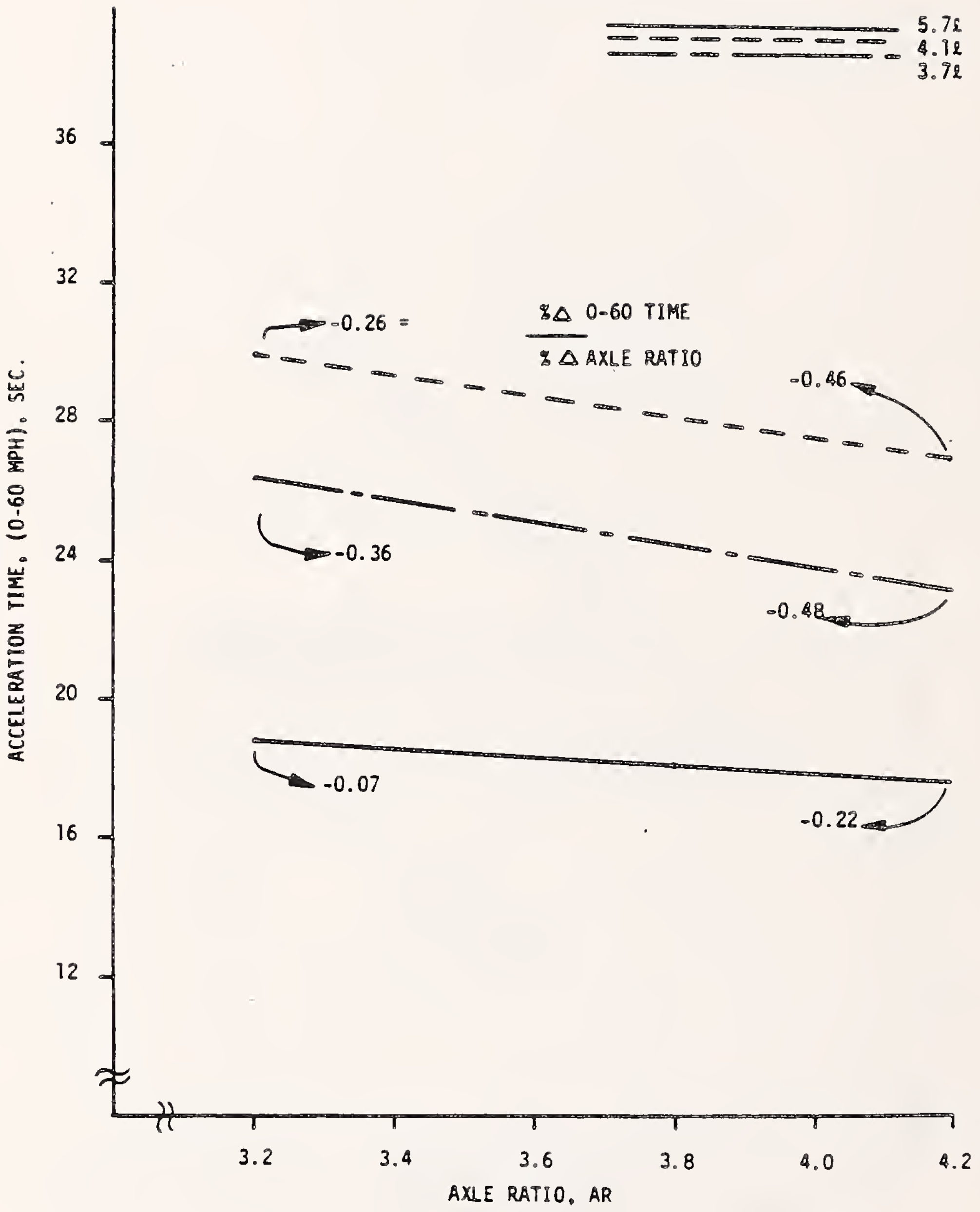


FIGURE 3-2. ACCELERATION TIME (0 TO 60 MPH) AS A FUNCTION OF AXLE RATIO

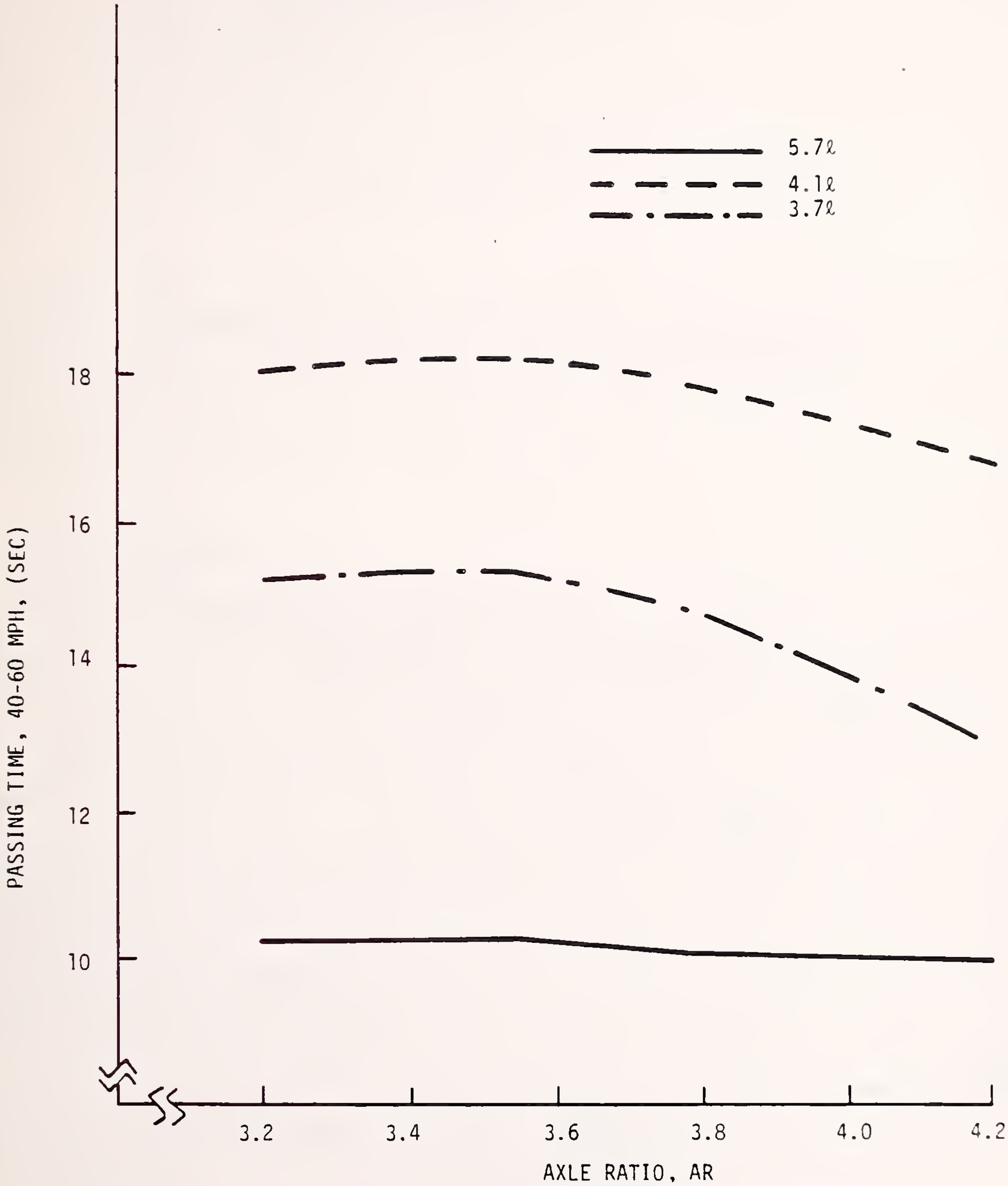


FIGURE 3-3. PASSING TIME (40 TO 60 MPH) AS A FUNCTION OF AXLE RATIO

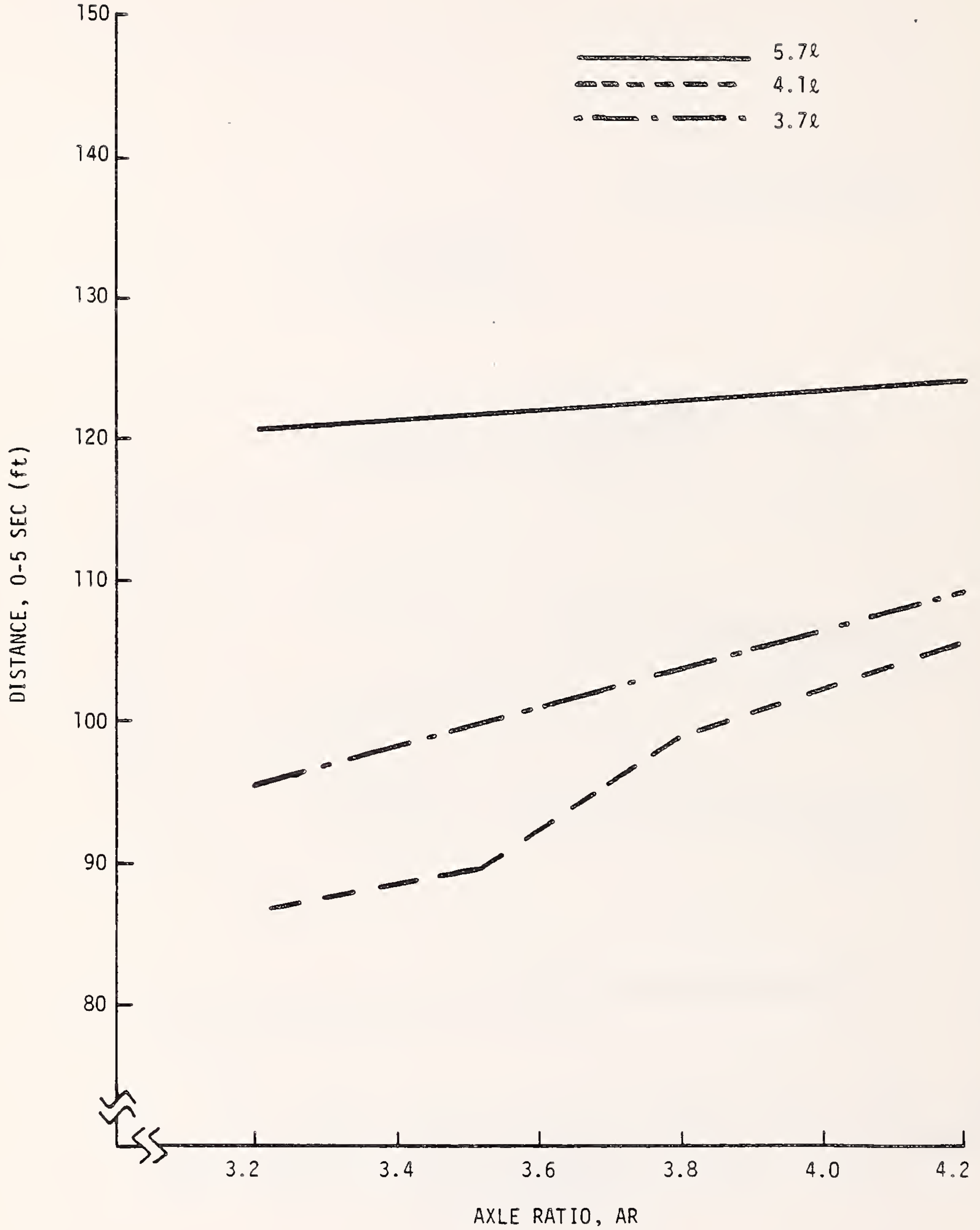


FIGURE 3-4. DISTANCE TRAVELED IN 0 TO 5 SECONDS, WOT, AS A FUNCTION OF AXLE RATIO

The 0 to 60 mph time can be estimated from the HP/WT ratio of the vehicle. A correlation previously presented was given by:[1]

$$t = 0.41 \left(\frac{HP}{WT}\right)^{-1.02} \qquad (3.2.1)$$

where

t = time (sec)
HP = maximum engine power (hp)
WT = GVWR (lb).

The three light truck points are superimposed on this curve as shown in Figure 3-5. The proximity of these points to equation (3.2.1) indicates that the 0 to 60 mph time can be estimated for a light duty truck with a full payload.

3.3 TOP SPEED

The top speed of a vehicle will be determined by the point at which the road load curve intersects the WOT engine horsepower curve. The road load is comprised of rolling resistance and aerodynamic horsepower and as a vehicle approaches its top speed the aerodynamic power dominates the road load as shown in Figure 3-6. This indicates that the aerodynamic loading will significantly effect top speed. The aerodynamic loading for a dynamometer simulation vehicle loading is characterized by the aerodynamic power at 50 mph. This loading is a function of the vehicle frontal area and drag coefficient and is given by:

$$C_D = \frac{HP}{0.81xA} \qquad (3.3.1)$$

where

C_D = Drag Coefficient
HP = Power (hp) at 50 mph
A = Area (ft^2).

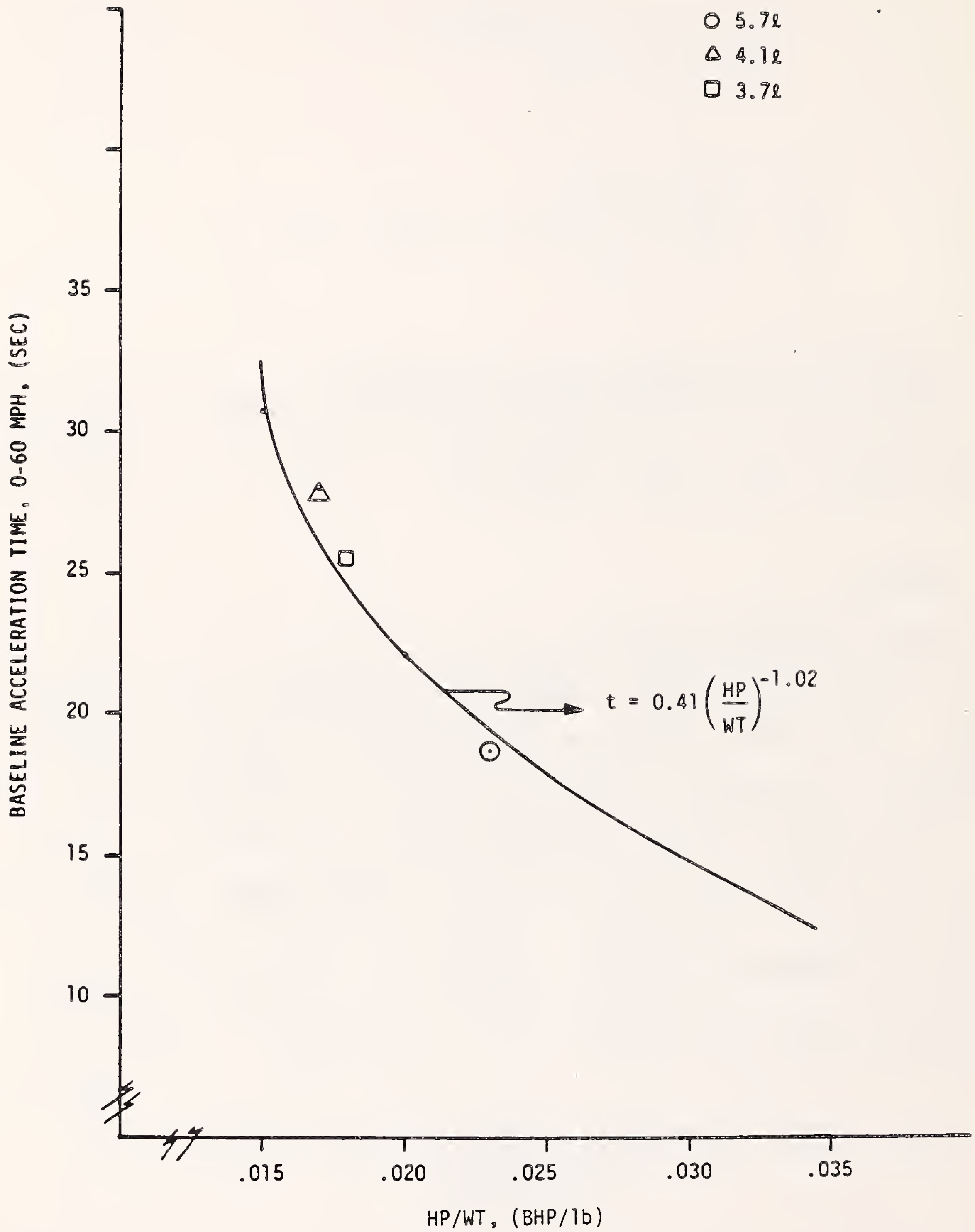


FIGURE 3-5. BASELINE ACCELERATION TIME (0 TO 60 MPH) AS A FUNCTION OF HP/WT

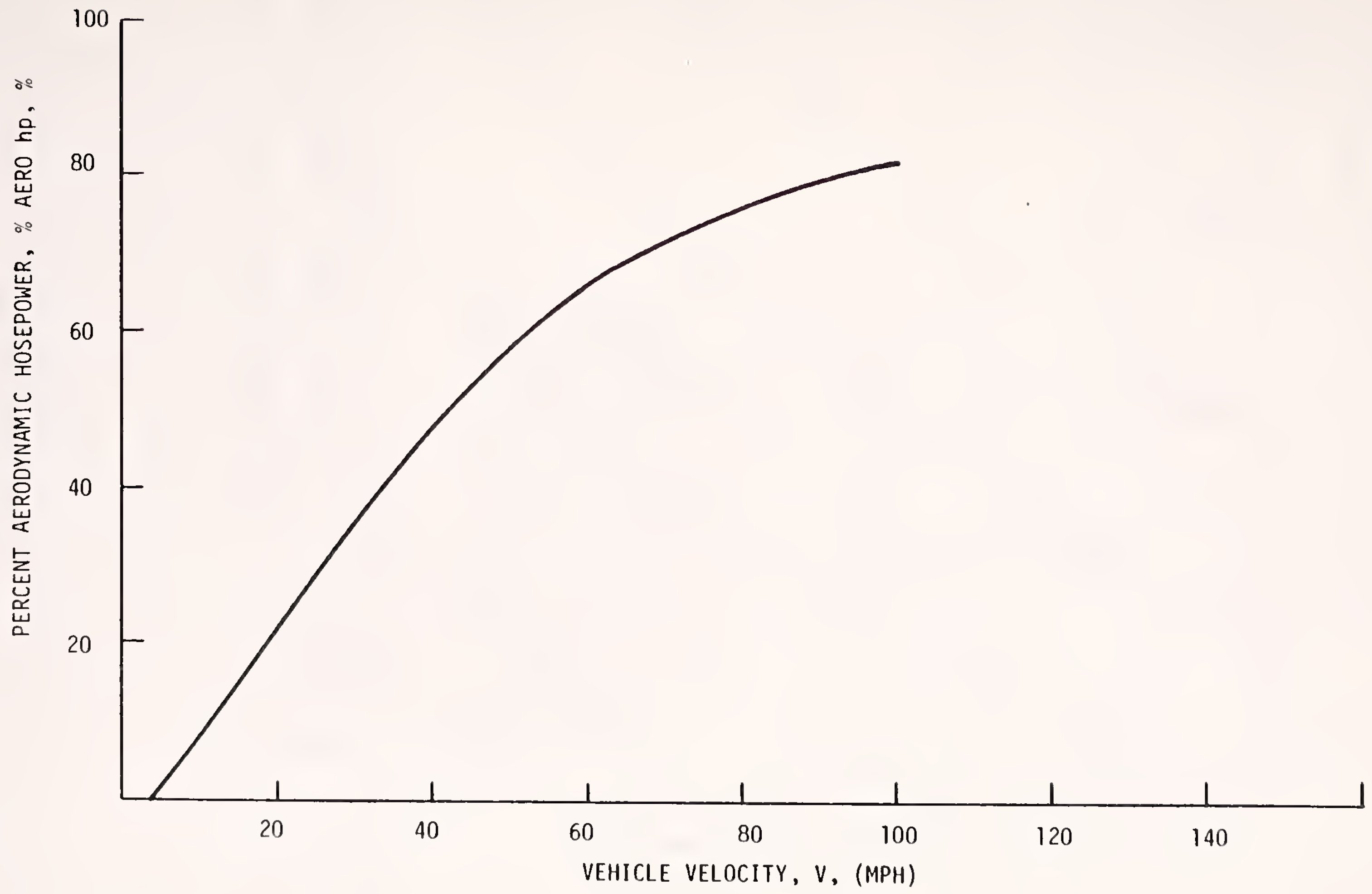


FIGURE 3-6. AERODYNAMIC HORSEPOWER AS A PERCENT OF ROAD LOAD HORSEPOWER AS A FUNCTION OF VEHICLE VELOCITY FOR A 7500-LB GVWR LIGHT TRUCK

This equation is illustrated graphically for lines of constant power in Figure 3-7. From this figure it can be seen that the aerodynamic loading can be reduced through either decreasing the drag coefficient or the frontal area. Typical pickup trucks have a C_D of about 0.58-0.65 and dynamometer loadings or power absorber units (PAU) of 16-18 hp. The increase in top speed as a function of PAU reduction is shown in Figure 3-8.

The effect of axle ratio on top speed is shown in Figure 3-9. Modifying this parameter shifts the intersection of road load curve and WOT engine horsepower curve subsequently altering the top speed. The top speed of three light trucks shown is relatively insensitive to axle ratio change.

The top speed of a vehicle was previously given by:[1]

$$TS = 3.52 \left[\frac{HP}{C_D * A_f}\right] + 68.2 \qquad (3.3.2)$$

where

TS = Top Speed (mph)
HP = Engine Power (hp)
C_D = Drag Coefficient
A_f = Frontal Area (ft^2).

The points of the three light trucks are superimposed on equation (3.3.2) as portrayed in Figure 3-10. Obviously this equation is only valid for a limited range as the equation becomes asymptotic at lower speeds.

3.4 GRADEABILITY

Gradeability is defined as the maximum grade a vehicle can achieve at a given velocity. The calculation of gradeability is taken from the following derivation. The forces on the vehicle are summed based on Figure 3-11.

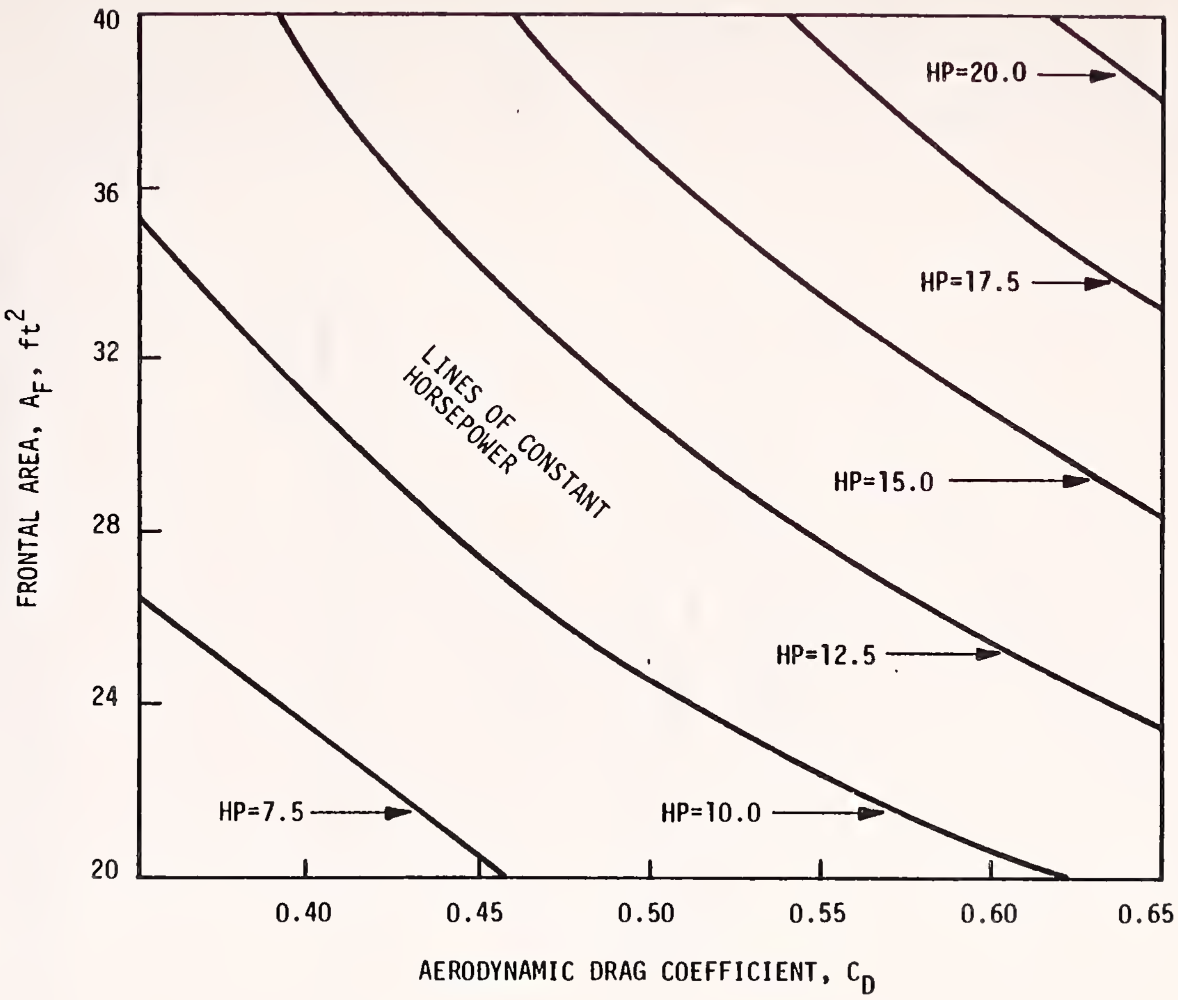


FIGURE 3-7. AERODYNAMIC ROAD LOAD HORSEPOWER REQUIREMENTS AT STEADY 50 MPH AS A FUNCTION OF FRONTAL AREA AND AERODYNAMIC DRAG COEFFICIENT

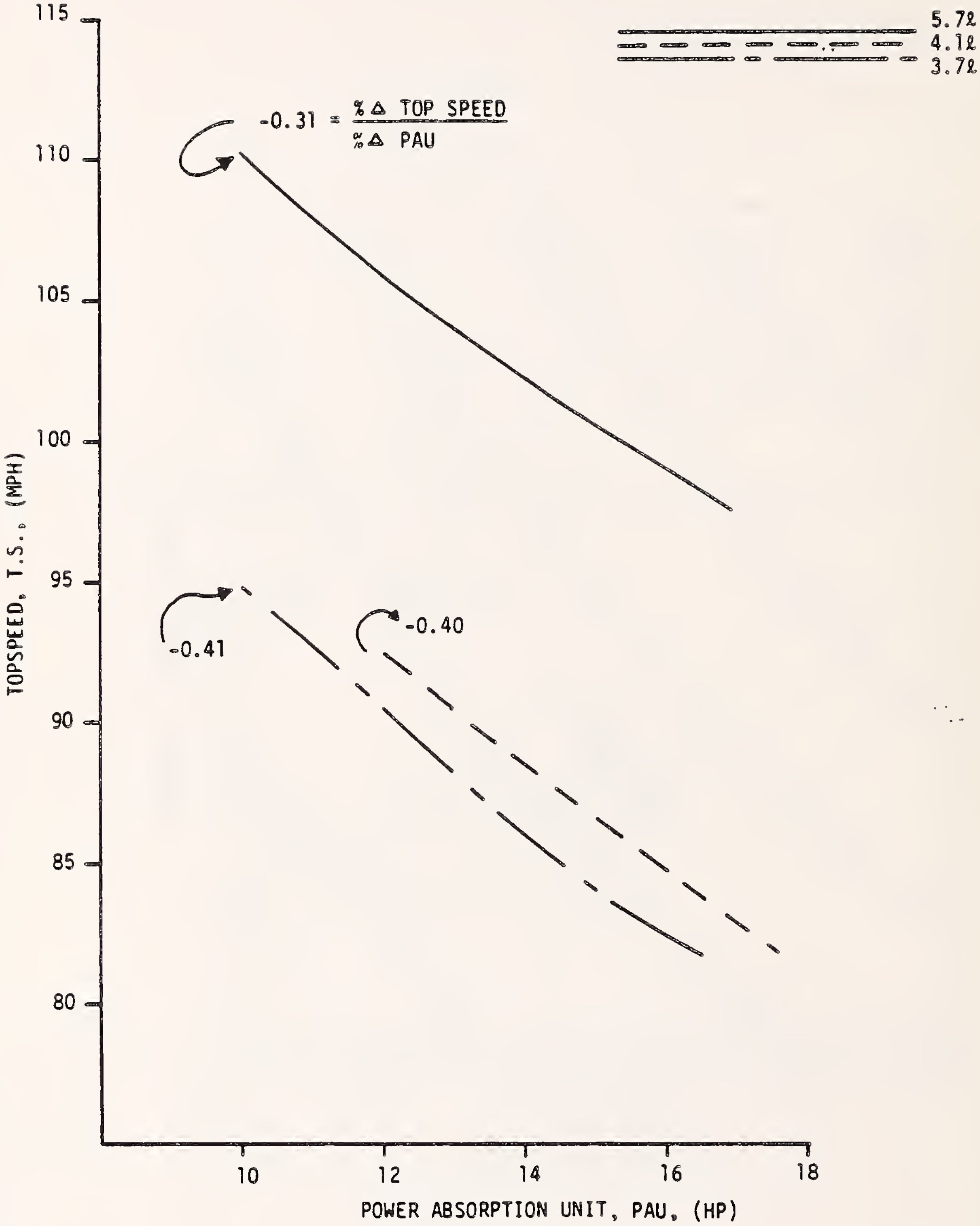


FIGURE 3-8. TOP SPEED AS A FUNCTION OF POWER ABSORPTION UNIT (PAU)

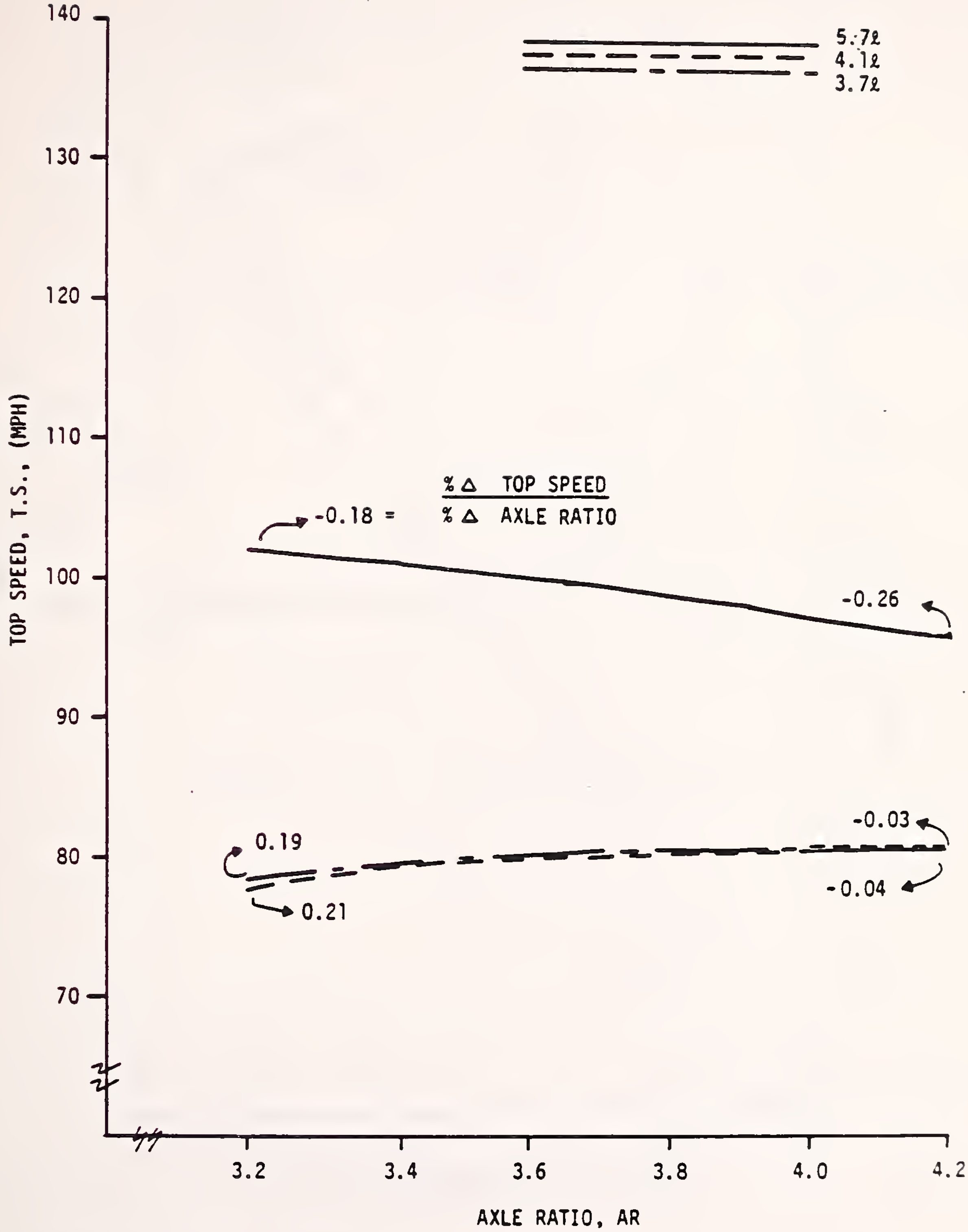


FIGURE 3-9. TOP SPEED AS A FUNCTION OF AXLE RATIO

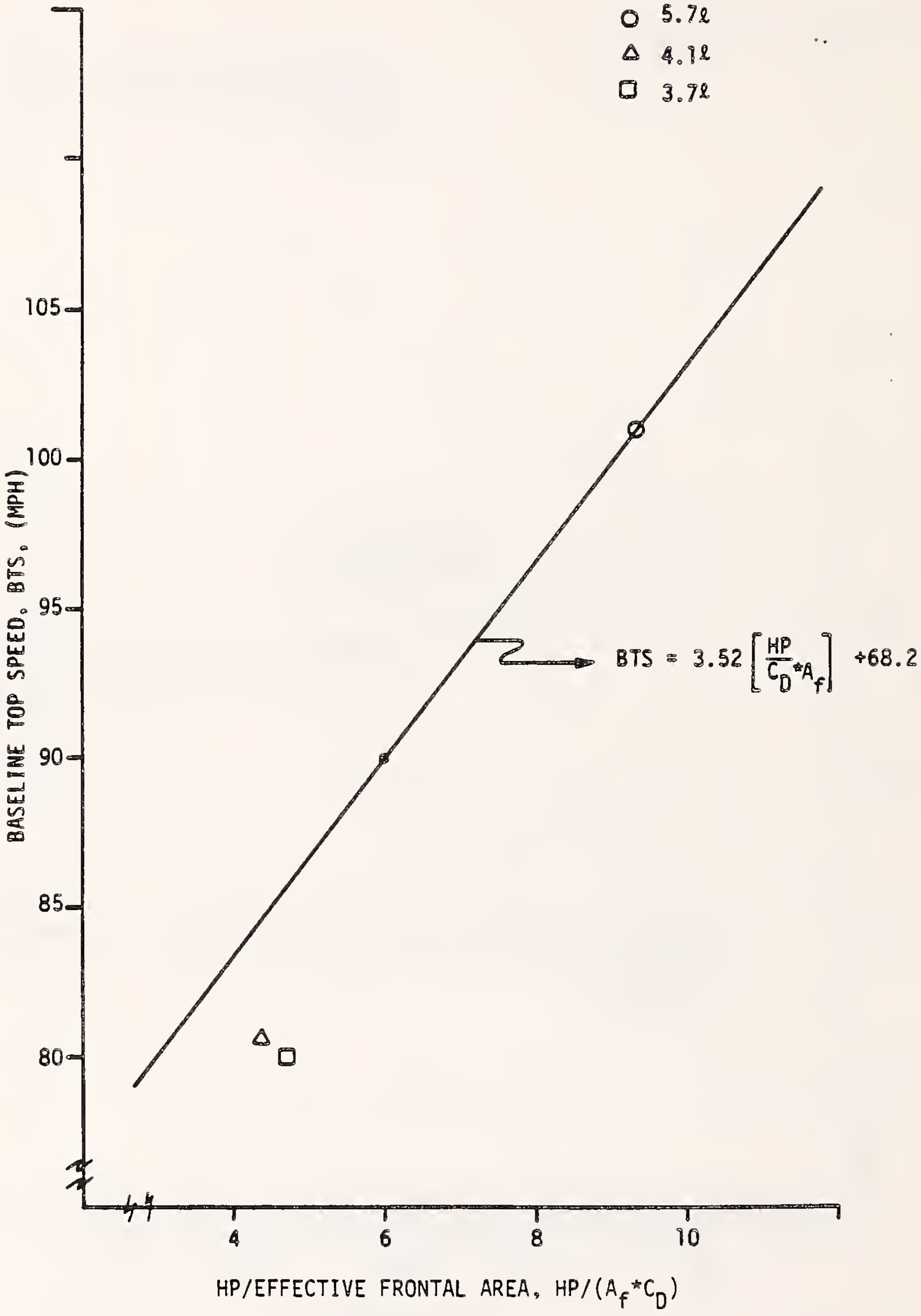


FIGURE 3-10. BASELINE TOP SPEED AS A FUNCTION OF HORSEPOWER TO EFFECTIVE FRONTAL AREA

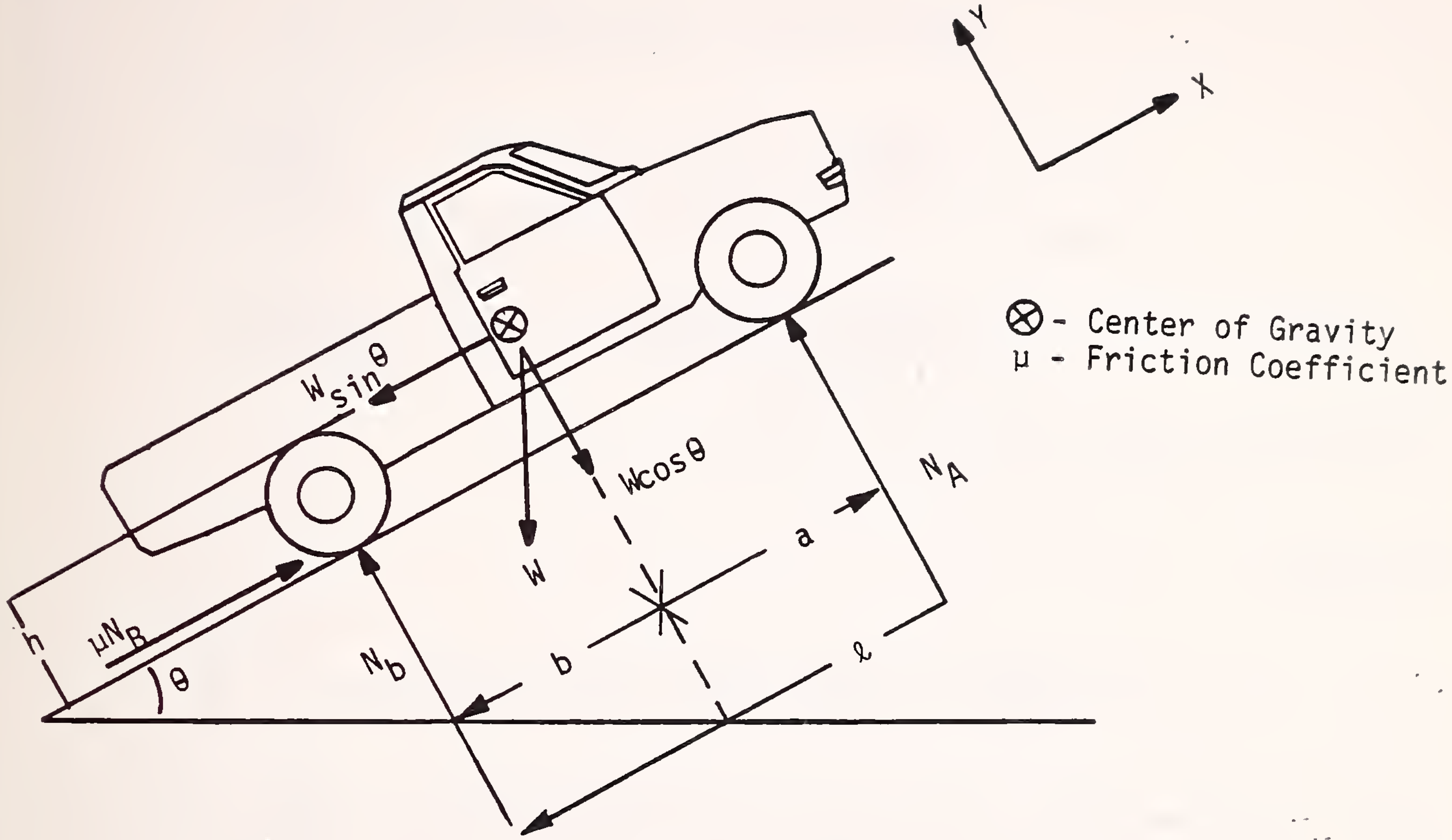


FIGURE 3-11. VEHICLE ON GRADE

$$\Sigma F_x = \mu N_B - W\sin\Theta = 0 \quad \therefore \quad N_B = \frac{W\sin\Theta}{\mu} \qquad (3.4.1)$$

$$\Sigma F_y = N_B + N_a - W\cos\Theta = 0 \qquad (3.4.2)$$

The moment about point A is

$$\Sigma M_A = -(a+b)N_B + aW\cos\Theta + hW\sin\Theta$$

$$= -(a+b)\sin\Theta + a\cos\Theta + h\sin\Theta = 0 \qquad (3.4.3)$$

$$\left(\frac{a+b-h}{\mu}\right) \sin\Theta = a\cos\Theta \qquad (3.4.4)$$

solving for G (gradeability)

$$\frac{a}{\frac{a+b-\mu h}{\mu}} = \tan\Theta = G \qquad (3.4.5)$$

$$\frac{\mu a}{(a+b)-\mu h} = \tan\Theta = G \qquad (3.4.6)$$

By definition, the wheelbase length = ℓ, $a+b=\ell$

Therefore,

$$G = \frac{\mu a}{\ell-\mu h} \text{ for rear wheel drive vehicle.}$$

If it is assumed that the payload weight is supported by the rear wheels, then the maximum gradeability for the fully loaded 5.7ℓ engine vehicle is calculated as 58 percent gradeability. This calculation, in addition to other gradeability equations, is presented in Appendix A. The gradeability curve for this particular vehicle is presented in Figure 3-12.

The effect of axle ratio on startup gradeability is shown in Figure 3-13. The startup gradeability is estimated to be 90 percent of the maximum gradeability due to clutch losses during engagement. Startup gradeability is shown to increase with a numerical increase in axle ratio. This is also true for 55 mph gradeability as shown in Figure 3-14.

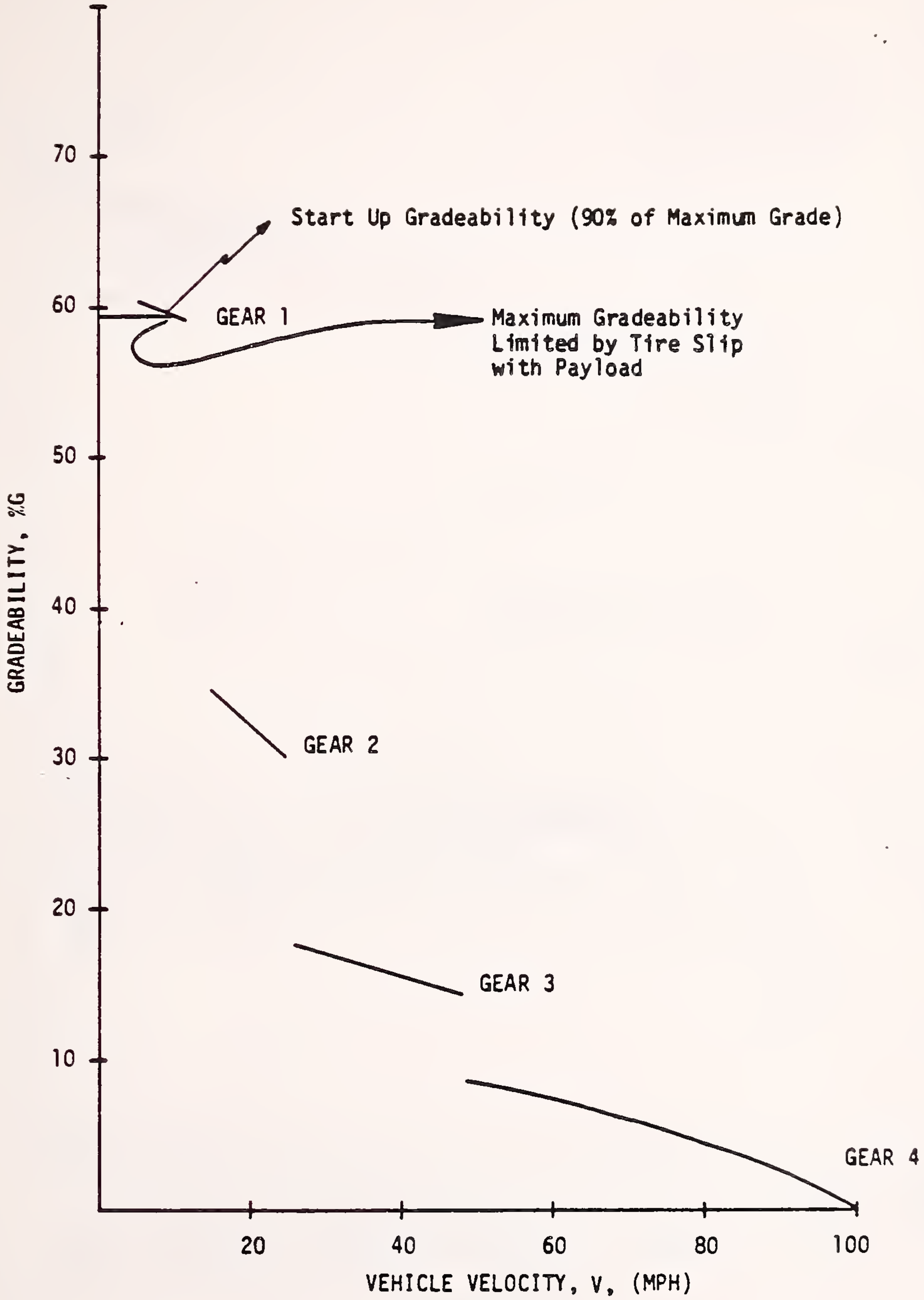


FIGURE 3-12. GRADEABILITY AS A FUNCTION OF VEHICLE VELOCITY OF A 7500-LB, 350-CID (5.7ℓ) LIGHT TRUCK

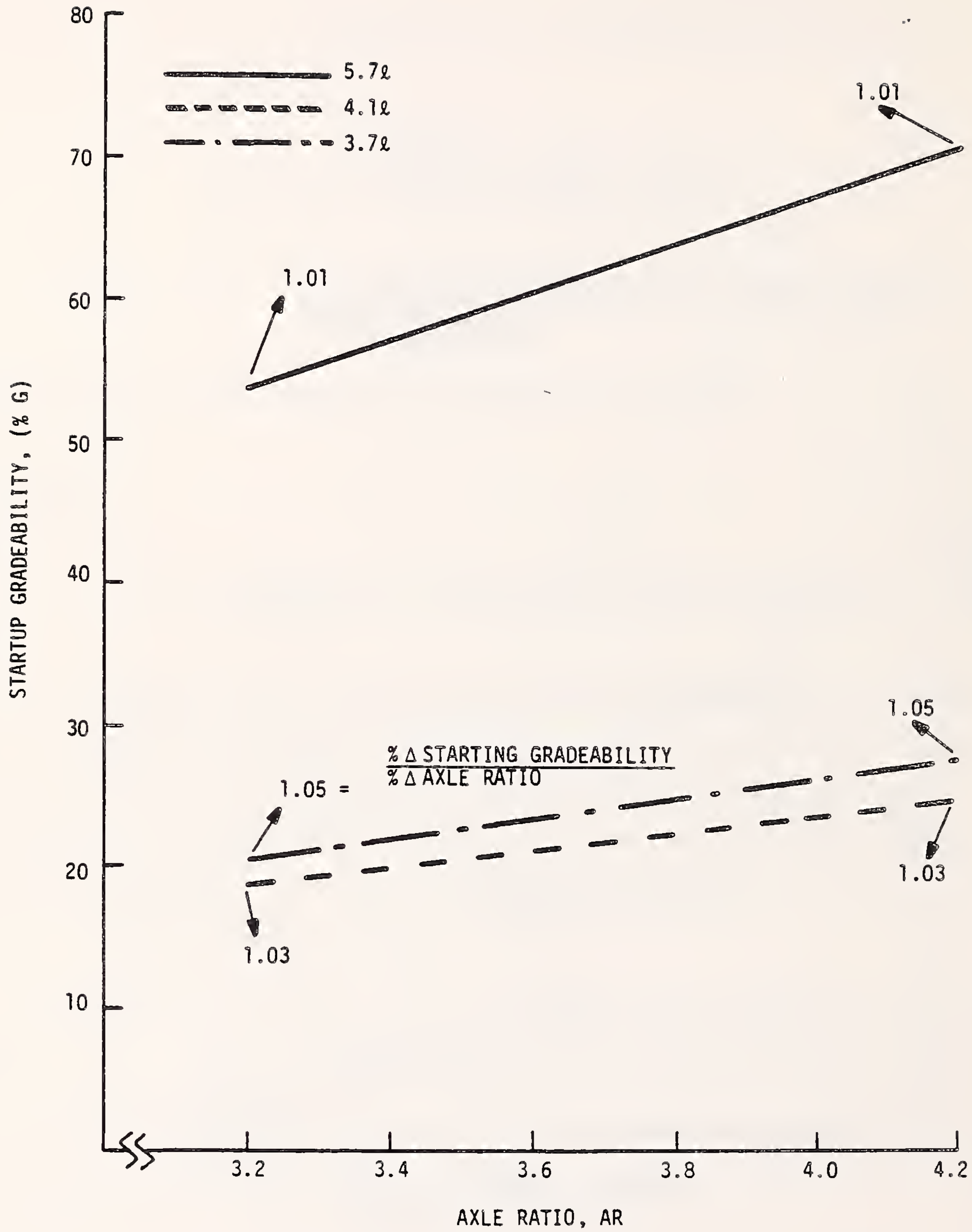


FIGURE 3-13. STARTUP GRADEABILITY AS A FUNCTION OF AXLE RATIO

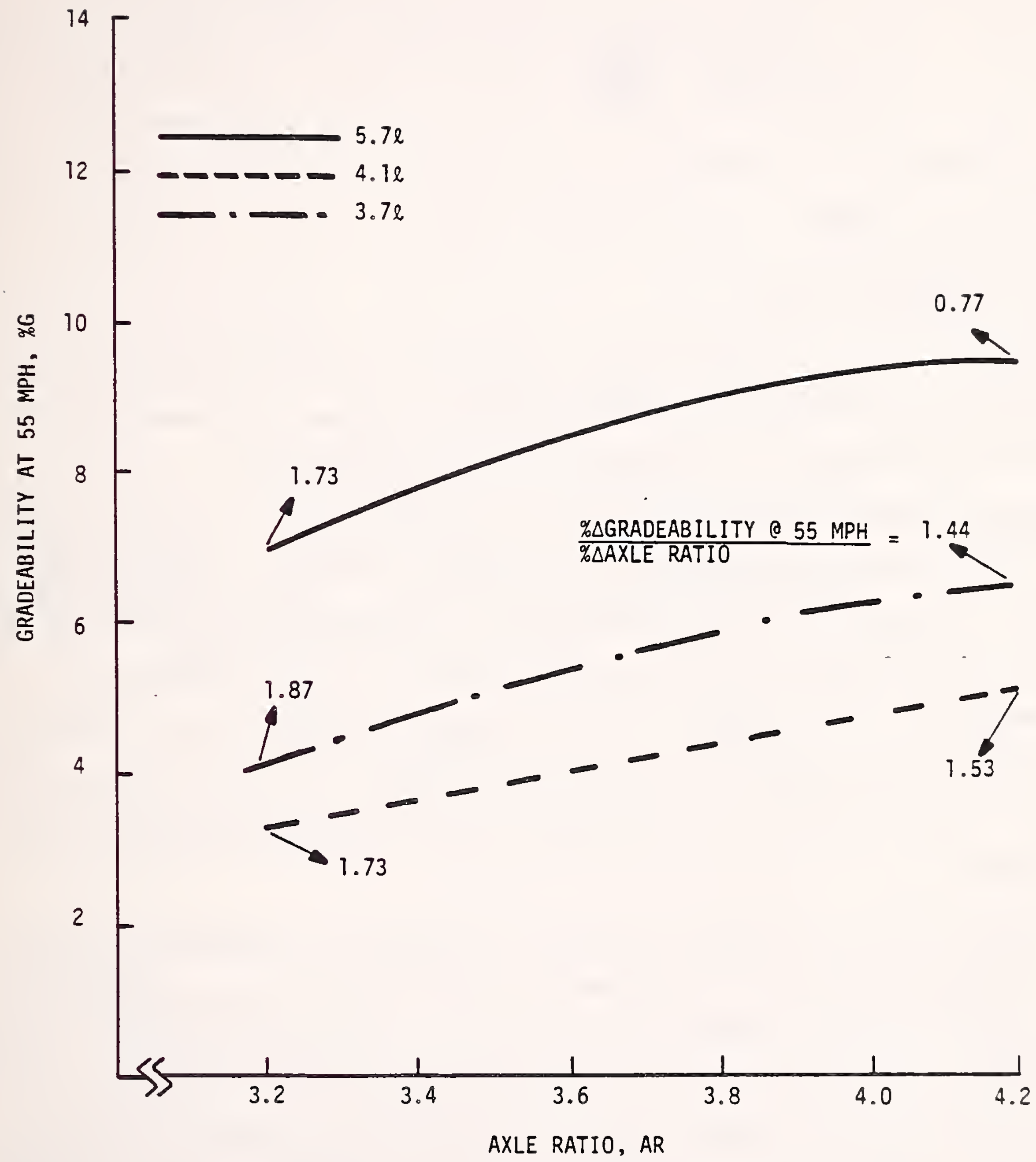


FIGURE 3-14. GRADEABILITY AT 55 MPH AS A FUNCTION OF AXLE RATIO

3.5 FUEL ECONOMY

The composite fuel economy for each vehicle is derived from the Environmental Protection Agency (EPA) urban and highway drive schedules as shown below:

$$FE_C = \frac{1}{\frac{.55}{URB} + \frac{.45}{HWY}}$$

where FE_C = Composite (mpg)

URB = Urban (mpg)

HWY = Highway (mpg).

The urban cycle is 7.5 miles long and takes 1371 seconds for an average speed of 20 mph while the highway cycle traverses 10.3 miles for 765 seconds at an average speed of 48 mph.

The fuel economy response due to weight reduction is shown in Figure 3-15. As the weight is reduced the road load decreases, subsequently increasing the performance. By maintaining constant performance, the engine can be scaled down in displacement as the weight is reduced. An example of the fuel economy gain achieved through scaling the engine is shown in Figure 3-16 for the 6400 lb GVW vehicle. The results of the remaining two light trucks are shown in Table 3-1.

The effect of axle ratio on fuel economy is shown in Figure 3-17. The increase in fuel economy for an axle ratio reduction is due to the shift of the road load curve to islands of lower brake specific fuel consumption (BSFC). The range of axle ratios examined here is 3.2 to 4.2 and the trend of the curve at greater extremes can be seen by the examples in Section 4.1. The effects of the axle ratios on driveability and emissions are not examined in this report.

The effect of vehicle aerodynamic drag and rolling resistance on fuel economy is illustrated in Figures 3-18 and 3-19 respectively. The aerodynamic drag is based on power absorption units (PAU) as explained in Section 3.3. The sensitivities shown are for a rolling resistance coefficient (C_1) decrease and increase from the baseline in Figure 3-19. The sensitivity for any given variable reduction can readily be calculated from the graphs.

The full potential of a PAU on C_1 reduction is accomplished by bringing the road load curve into alignment with the original curve with an axle ratio reduction.[1] An example of a PAU reduction with constant performance is shown in Figure 3-20. The baseline PAU which is 16.5 hp, is reduced to 11.0 hp resulting in a 7.9 percent fuel economy improvement. However, if the axle ratio is also reduced to maintain constant performance, the resulting fuel economy improvement is increased an additional 6.5 percent resulting in an overall improvement of 14.4 percent. The overall increase resulting from a road load (PAU or C_1) reduction <u>and</u> an axle ratio decrease is presented in Table 3-2. The fuel economy increase is for the extremes of road load reduction and any improvement between the baseline and extremes can be calculated from a graph similar to that of Figure 3-20.

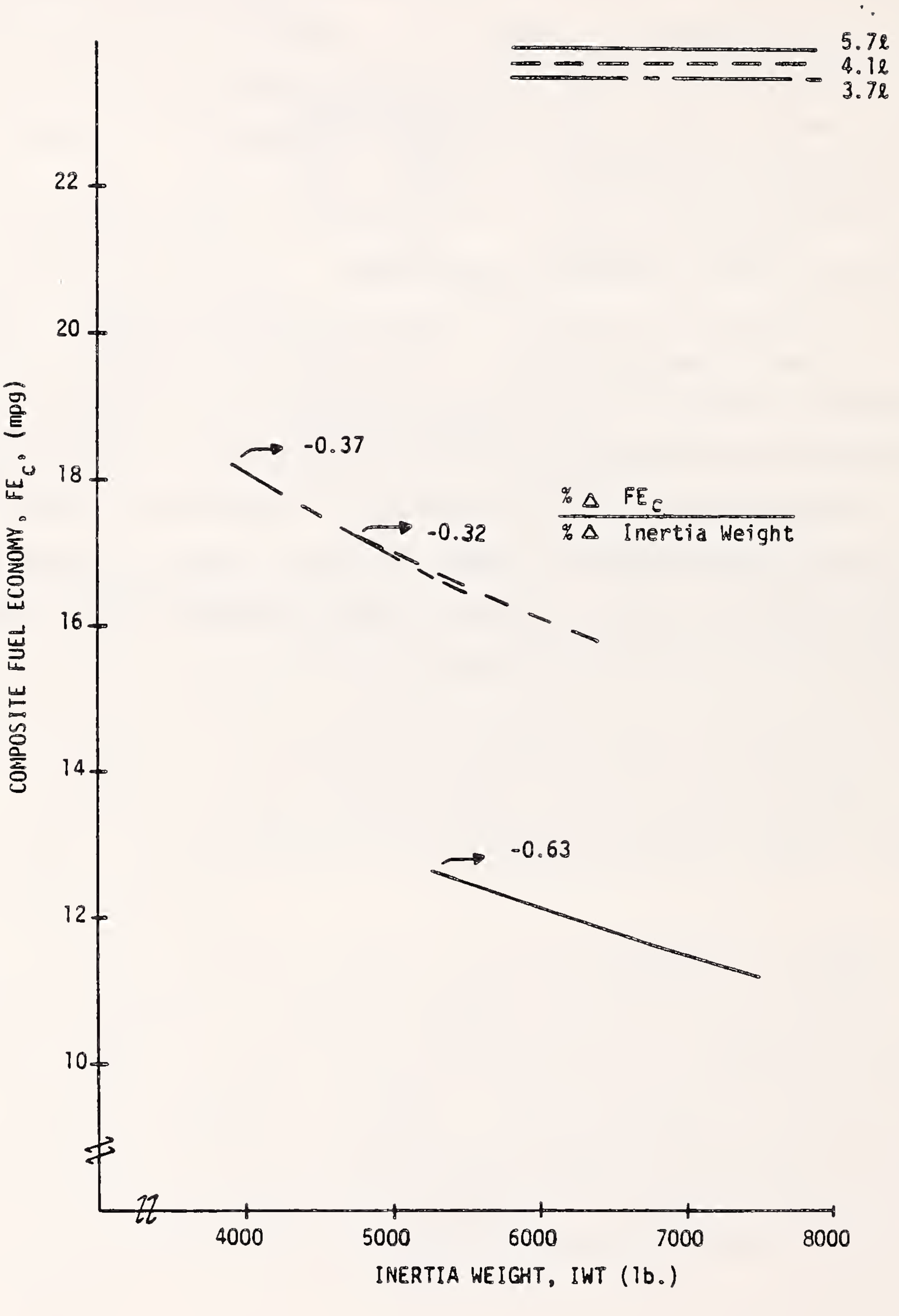


FIGURE 3-15. COMPOSITE FUEL ECONOMY AS A FUNCTION OF INERTIA WEIGHT

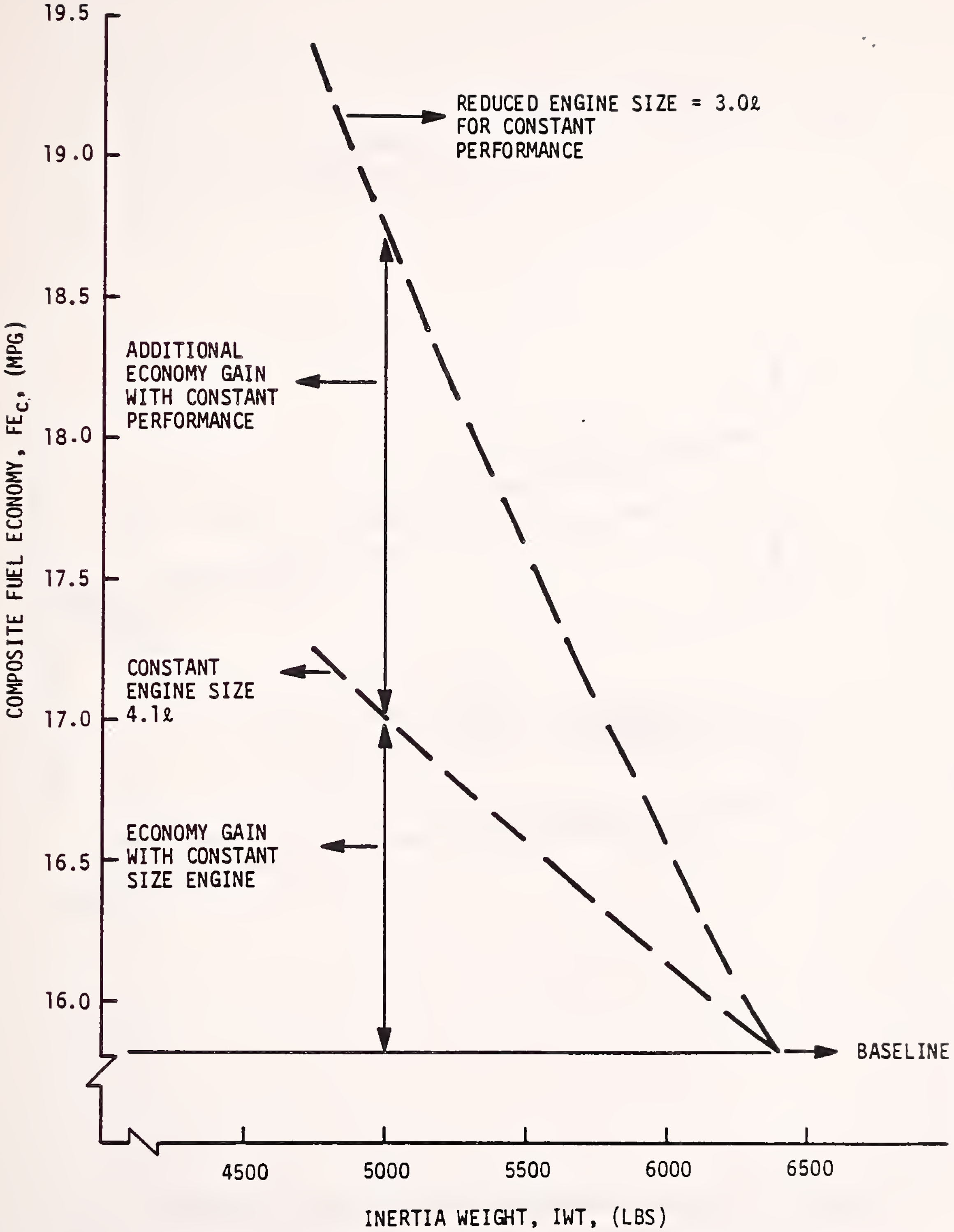


FIGURE 3-16. EFFECT OF WEIGHT REDUCTION AND CONSTANT PERFORMANCE ON FUEL ECONOMY FOR A 4.1ℓ ENGINE 6400-LB GVW LIGHT TRUCK

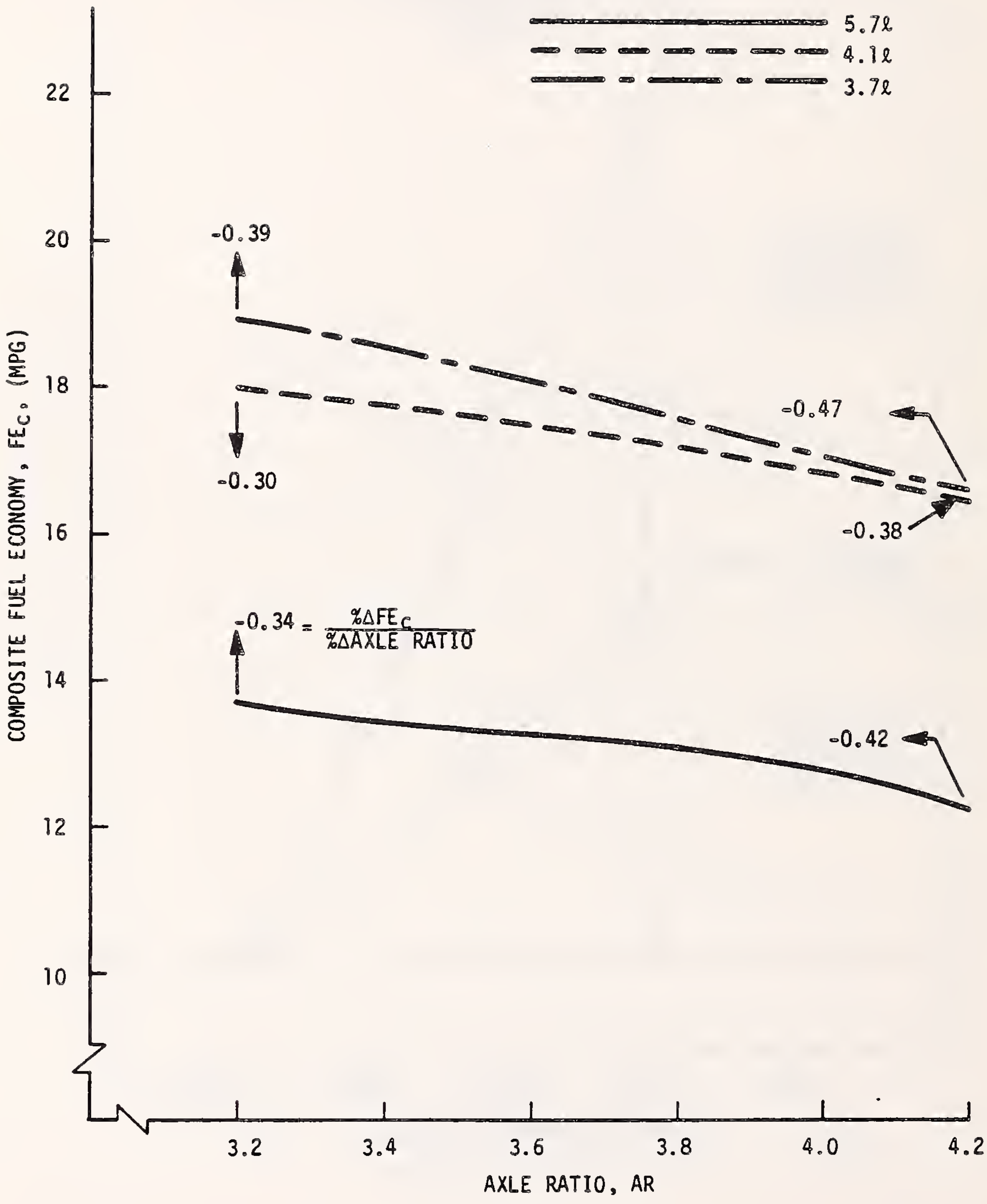


FIGURE 3-17. COMPOSITE FUEL ECONOMY AS A FUNCTION OF AXLE RATIO

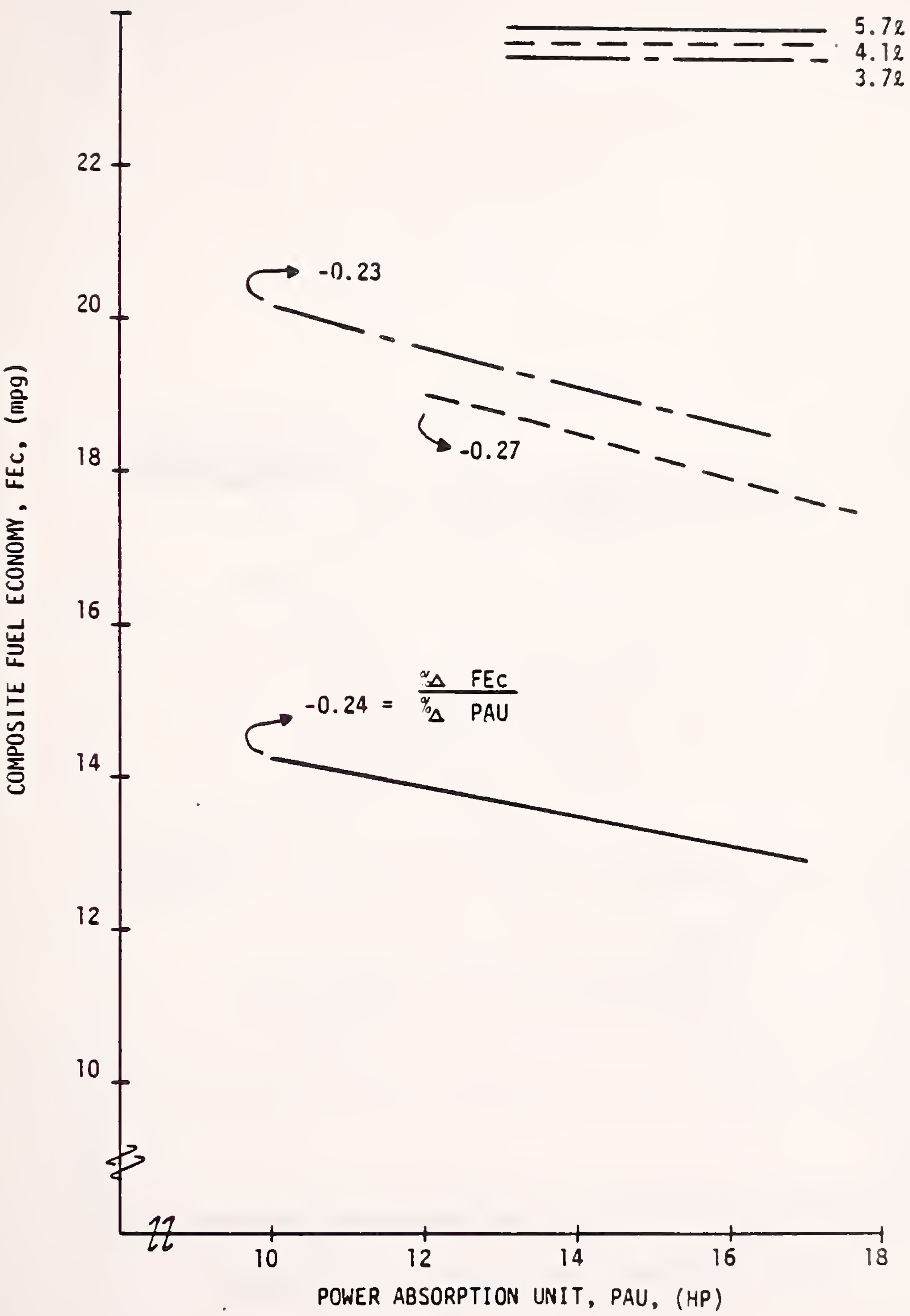


FIGURE 3-18. COMPOSITE FUEL ECONOMY AS A FUNCTION OF POWER ABSORPTION UNIT

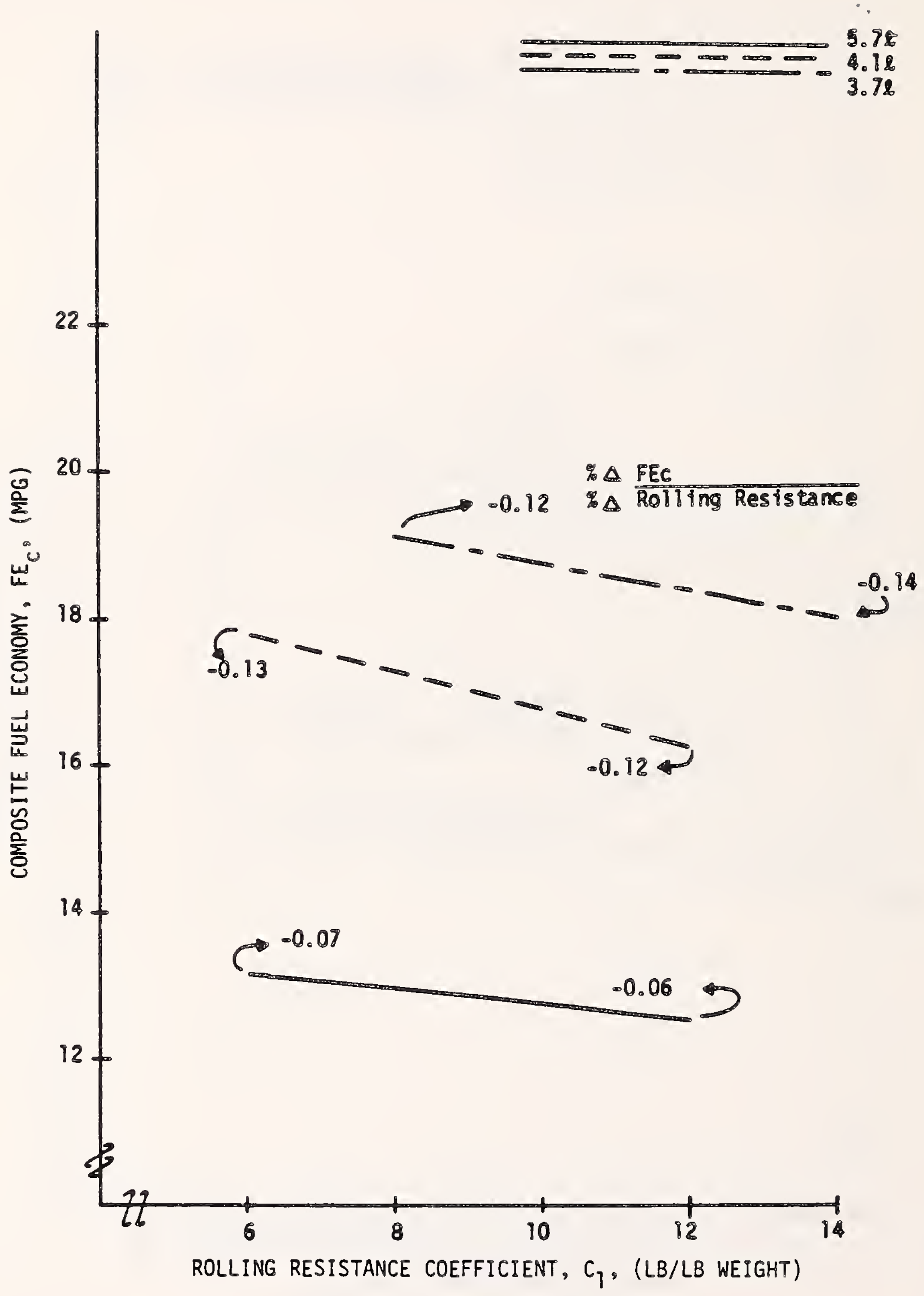


FIGURE 3-19. COMPOSITE FUEL ECONOMY AS A FUNCTION OF ROLLING RESISTANCE COEFFICIENT

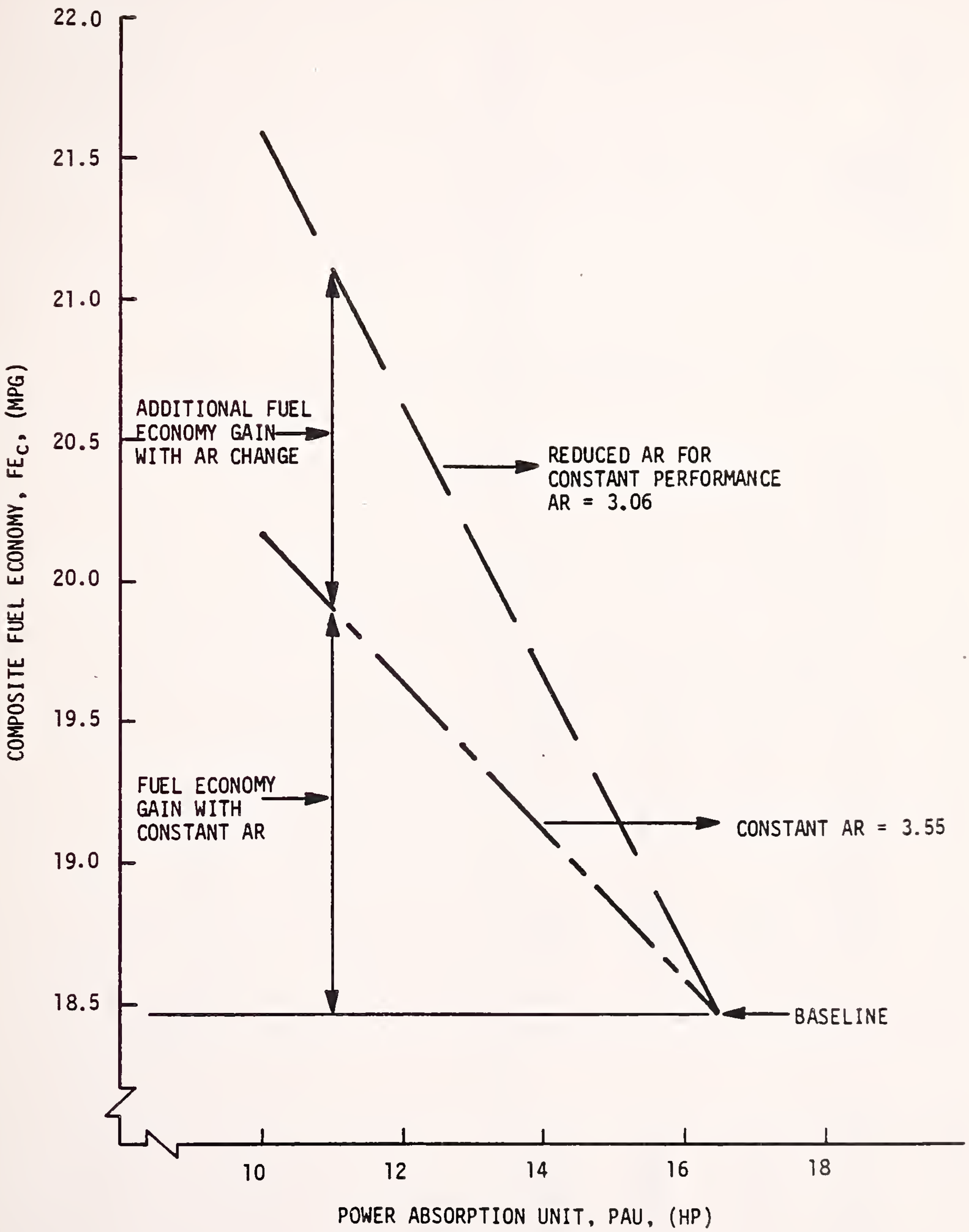


FIGURE 3-20. EFFECT OF POWER ABSORPTION UNITS WITH CONSTANT PERFORMANCE ON FUEL ECONOMY FOR A 3.7ℓ ENGINE 5500-LB GVW LIGHT TRUCK

TABLE 3-1. EFFECT OF WEIGHT REDUCTION ON FUEL ECONOMY FOR CONSTANT PERFORMANCE

ENGINE MODIFICATION (ℓ)	WEIGHT REDUCTION (lb)	FE_c INCREASE (%)	SENSITIVITY $\frac{\%\Delta FE_c}{\%\Delta WT}$
5.7 → 4.0	7500 → 5250	33.9	1.1
4.1 → 3.0	6400 → 4750	18.5	0.72
3.7 → 2.6	5800 → 3900	21.3	0.73

TABLE 3-2. FUEL ECONOMY INCREASE ATTRIBUTED TO EQUAL PERFORMANCE FOR AERODYNAMIC DRAG AND ROLLING RESISTANCE COEFFICIENT

Engine (ℓ)	PAU	AR	FE_C Increase %	C_1	AR	FE_C Increase %
	Baseline/Reduction			Baseline/Reduction		
5.7	17/10	3.54/3.21	14.4	8/6	3.54/3.44	3.4
4.1	17.6/12	3.73/3.31	13.4	8/6	3.73/3.61	5.6
3.7	16.5/10	3.55/3.06	16.8	13/8	3.55/3.41	6.7

4. DRIVELINE MODIFICATIONS

4.1 ANALYSIS

Transmission modifications can improve the fuel economy of a particular vehicle in two ways. First, the power requirements of the vehicle can be reduced through the use of more efficient transmission components. This is particularly significant with automatic transmissions since energy losses of the torque converter can be reduced by "tight" (low slip) converters or by bypassing the converter.

Another improvement in fuel economy can be achieved by extending the gear ratio range (grr = first gear ratio divided by final gear ratio) of the transmission. The high final gear ratio allows the engine to operate at a lower BSFC during cruise.

The purpose of this section is to investigate the potential of these two approaches. A set of eight transmissions, including five automatics and three manuals, was chosen for the study as shown in Table 4-1. For each transmission, the "optimal" performance versus fuel economy curve was defined using the methods introduced by H. Chana et al.[6] This method employs a systematic modification of axle ratio and engine displacement. For each transmission and performance level, a particular displacement-axle ratio combination defines the maximum fuel economy attainable. Fuel economy was defined as composite miles per gallon, while performance was measured by the 0 to 60 mph time in seconds. Standard EPA shift logic was used with the manual transmissions, while a typical speed versus load shift schedule controlled the automatics. The automatic three speed shift logic was identical to the four speed, except for the absence of 3-4 and 4-3 shift lines. The shift logics produced the gear time distributions shown in Table 4-2; gear time distributions for the automatic transmissions are averages, since vehicle speed at shift is determined by load. For example, a more fully loaded vehicle would shift later and therefore spend more time in the lower gears.

TABLE 4-1. TRANSMISSIONS STUDIED

Transmission Type*	Gear Ratio	Gear Ratio Range
A3 (baseline)	2.45/1.45/1.0	2.45
A3	2.74/1.54/1.0	2.74
A4	2.45/1.45/1.0/0.67	3.66
A4	2.74/1.54/1.0/0.60	4.57
A4	3.45/1.79/0.97/0.58	5.95
M3	3.0/1.7/1.0	3.00
M4	3.0/1.7/1.0/0.73	4.11
M5	3.0/1.94/1.42/1.0/0.73	4.11

*Automatics are run in both unlocked and locked (all gears but first) modes.

TABLE 4-2. GEAR TIME DISTRIBUTIONS

A. URBAN DISTRIBUTIONS (% OF TOTAL TIME)

Transmission Type	Gear: 1	2	3	4	5
A3	28	15	57		
A4	28	15	29	28	
M3	32	22	46		
M4	32	22	37	9	
M5	32	22	37	1	8

B. HIGHWAY DISTRIBUTIONS (% OF TOTAL TIME)

Transmission Type	Gear: 1	2	3	4	5
A3	2	1	97		
A4	2	1	2	95	
M3	2	1	97		
M4	2	1	7	90	
M5	2	1	7	6	84

The wide open throttle shift point for performance runs is determined by minimizing the 0 to 60 mph time.

For each transmission-engine displacement combination, graphs of fuel economy and of performance versus axle ratio were constructed as illustrated in Figure 4-1. A crossplot of the tradeoff between fuel economy and performance for a given displacement-transmission combination is shown in Figure 4-2. By scaling the engine displacement, a number of these curves were generated, Figure 4-3, and their envelope defines the optimal performance versus fuel economy tradeoff curve for each transmission. The transmissions were then compared on an equal performance level of 0 to 60 mph in 15 seconds. Since the tradeoff lines were nearly parallel, the choice of performance level had little effect on the results. Figures 4-1, 4-2, 4-6, which serve as examples and are not part of the comparative study, were generated with the vehicle parameters listed on those figures. The vehicle parameters used in the comparative study, and the resulting composite fuel economy for each transmission are listed in Appendix B.

This study does not account for the drivability or emissions effects of the transmission modifications studied, nor for the weight and rotating inertia increases they may entail. For that reason, the reported results represent the upper limit to the potential of the modifications studied.

4.2 LOCKUP TORQUE CONVERTERS

Torque converters are responsible for a significant fuel economy difference between automatic and manual transmissions. Converters allow engine idling in drive and improve shift smoothness, but at the cost of converter slip and the resulting energy loss caused by this slip. Converter efficiency varies over a wide range, and is a function of the capacity factor which is defined by:

$$K = \frac{RPM}{\sqrt{TQ}} \qquad (4.2.1)$$

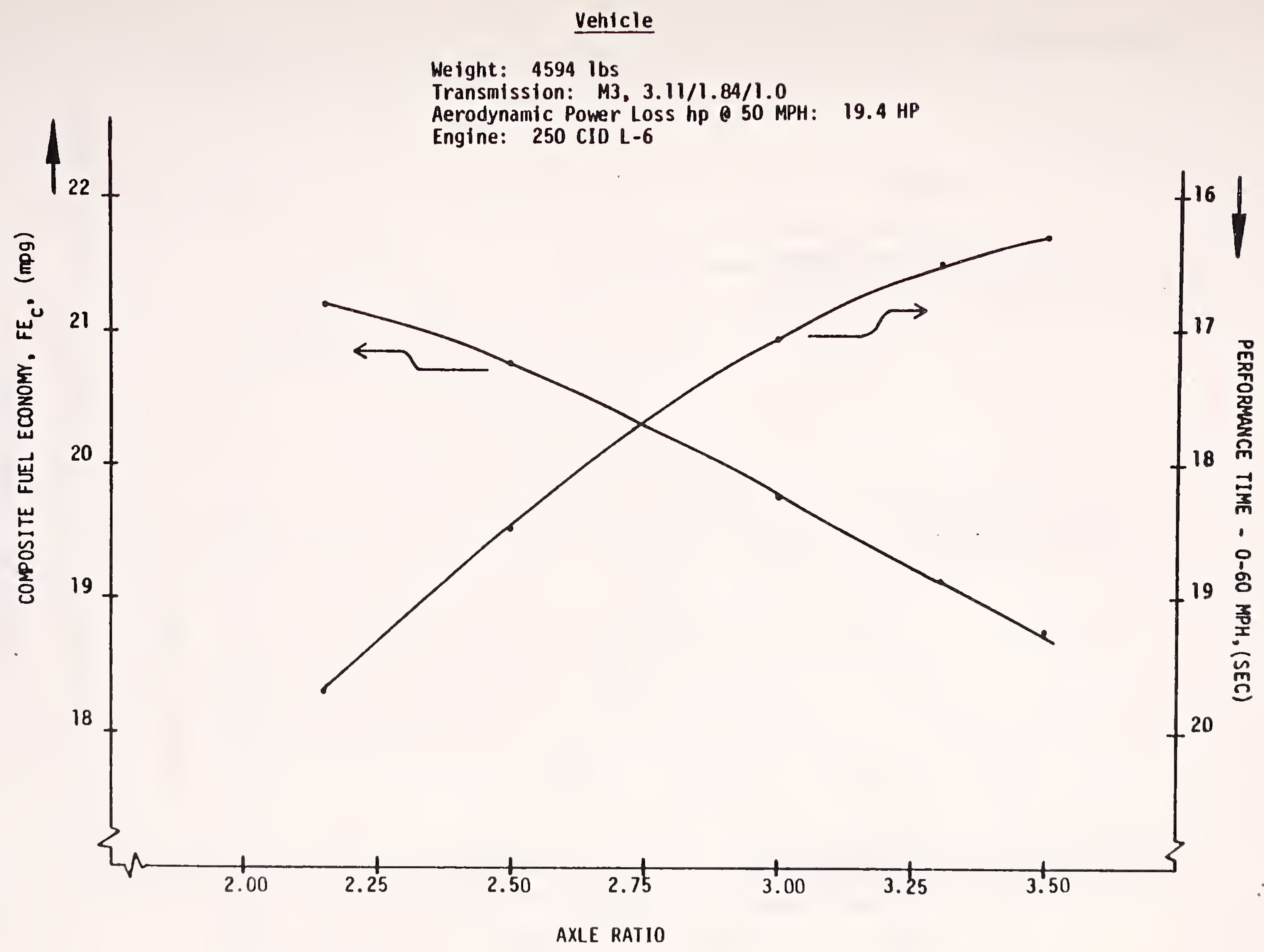


FIGURE 4-1. TYPICAL PERFORMANCE VERSUS AXLE RATIO AND FUEL ECONOMY VERSUS AXLE RATIO CURVES

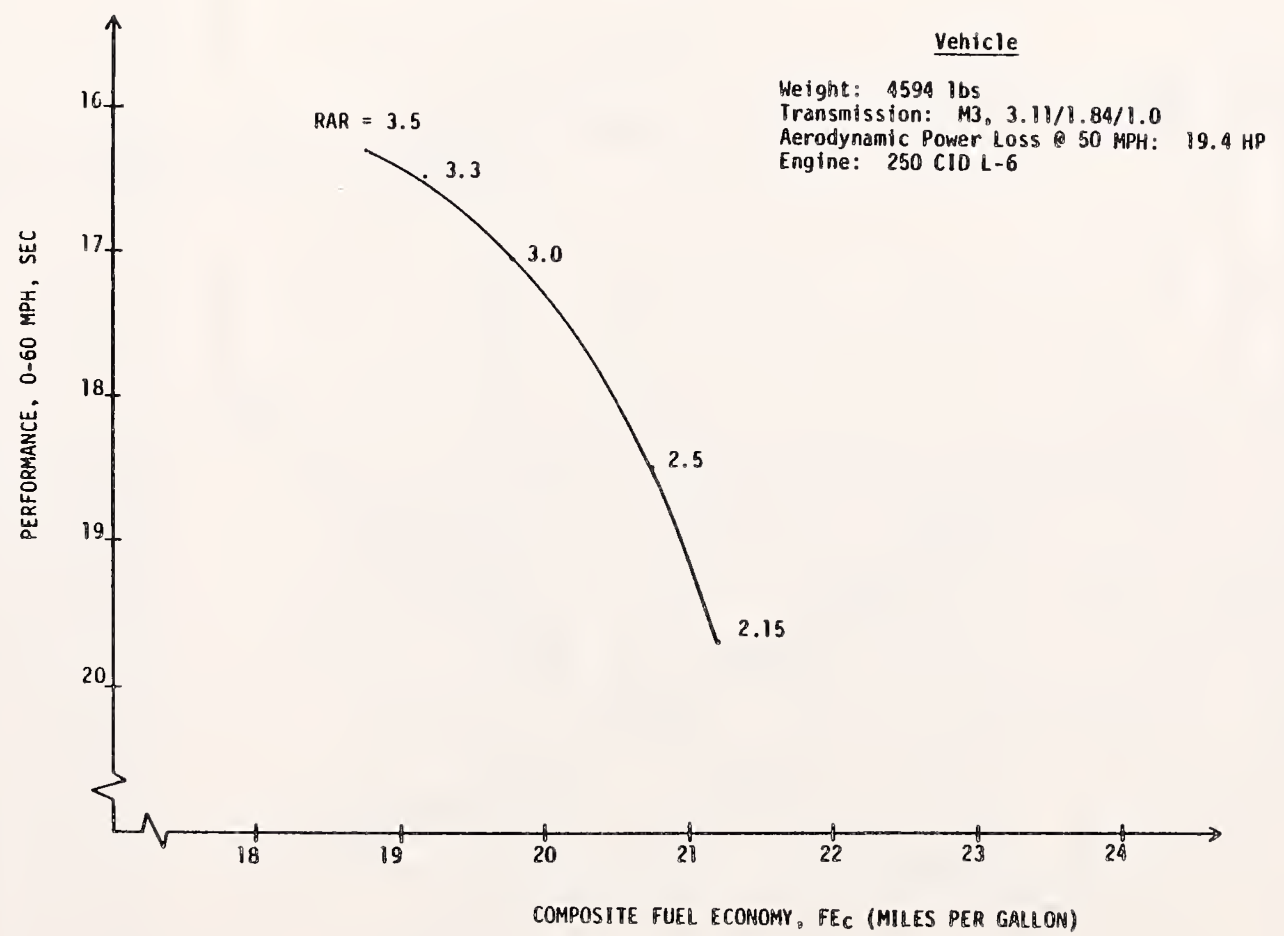


FIGURE 4-2. TYPICAL PERFORMANCE VERSUS FUEL ECONOMY CURVES

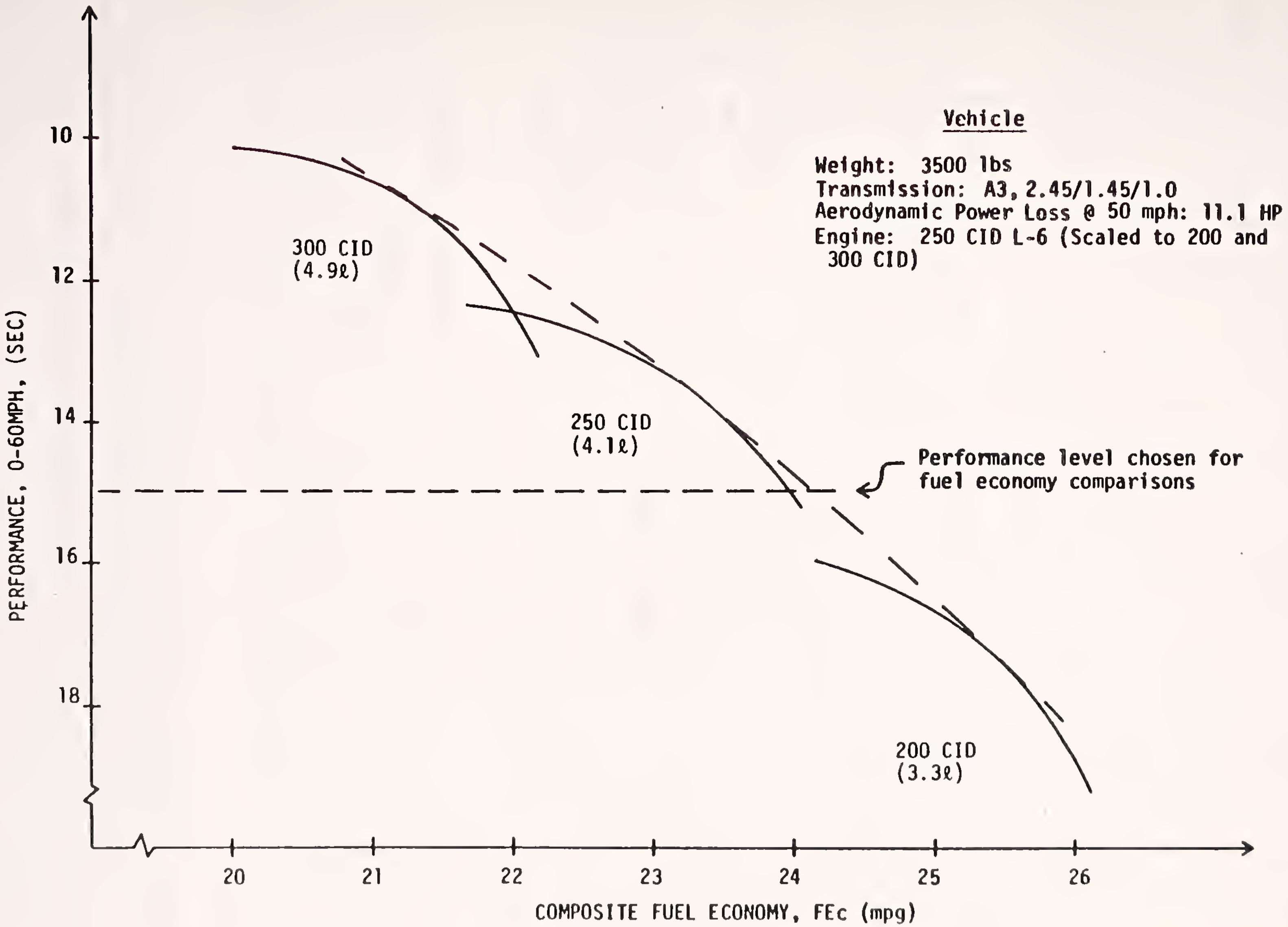


FIGURE 4-3. OPTIMAL PERFORMANCE VERSUS FUEL ECONOMY TRADEOFF CURVE FOR BASELINE TRANSMISSION (UNLOCKED)

where:

K = capacity factor and TQ = torque.

Capacity factor may be calculated using either input or output torque and rpm. The efficiency versus capacity factor profile for the torque converter used for this study is shown in Figures 4-4 (input capacity factor) and 4-5 (output capacity factor). In both figures, the efficiency uniformly increases with capacity factor except near a discontinuity denoted by "coupling point." Beyond this point, the torque converter acts as a fluid coupling and provides no torque multiplication.

Torque converters can be designed "tighter" (reduced slip loss) to reduce converter losses. Torque converter tightness is generally characterized by the input capacity factor at which the converter begins to transmit power. As seen from Figure 4-4, the capacity factor for this converter is 100. This capacity factor can be reduced, and the converter made tighter and more efficient when operating at low capacity factor, by increasing the converter diameter or by changing blade angles. The increased tightness would however incur a penalty in startup driveability, shift smoothness, and torque multiplication, and the efficiency in the coupled range is unimproved.

A more effective method of reducing torque converter losses is by bypassing the converter. In this way, torque converter losses during cruise can be eliminated, while low-speed torque multiplication and driveability are maintained. For this study, a locked up transmission was defined as one in which all gears except first were locked.

The fuel economy benefit of locking up a torque converter is dependent on a number of factors which include the torque converter efficiency profile, vehicle parameters, and the effect of lockup on engine operating BSFC. The torque converter efficiency profile determines the losses, and thus limits the potential of

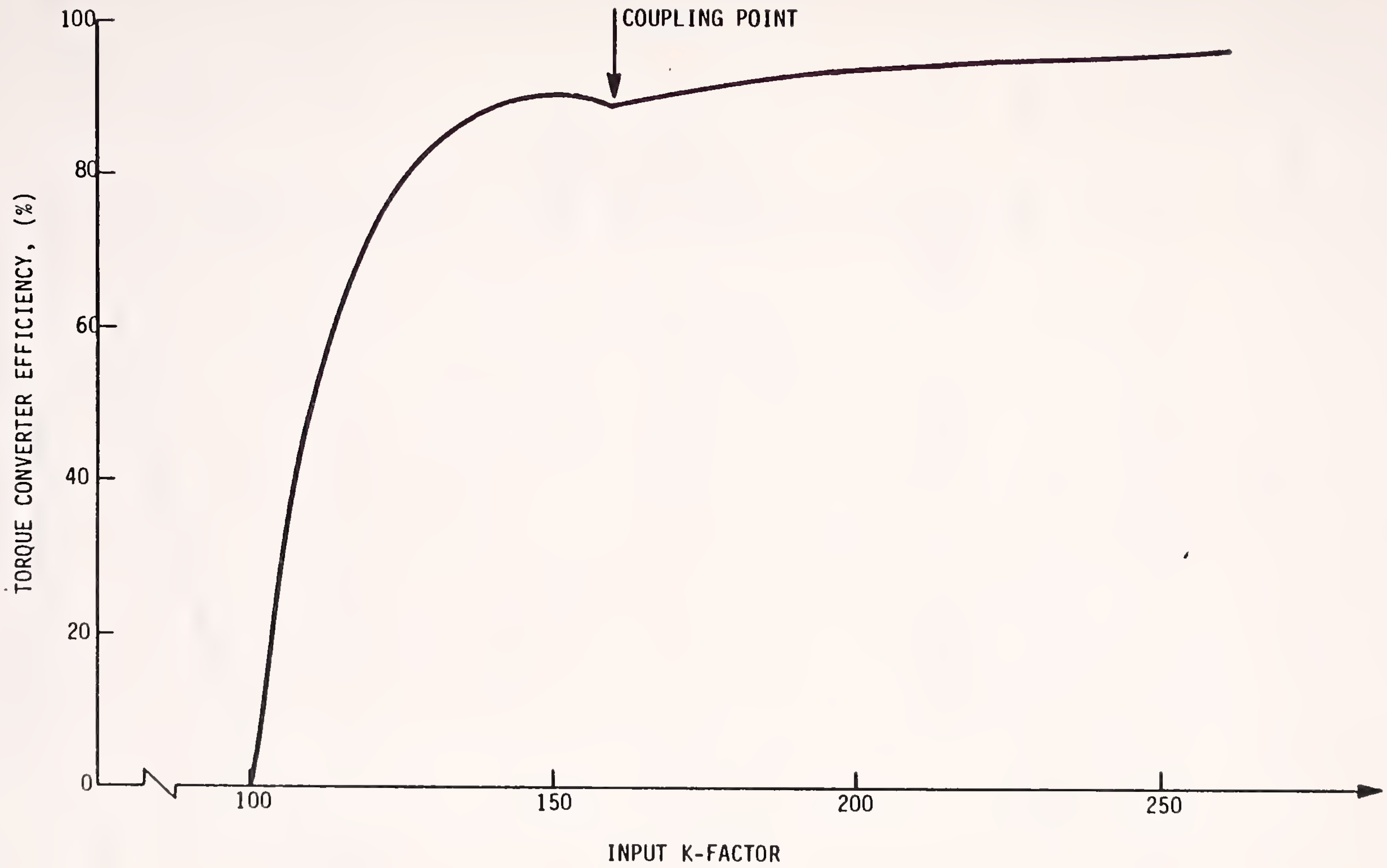


FIGURE 4-4. TORQUE CONVERTER EFFICIENCY VERSUS INPUT CAPACITY FACTOR

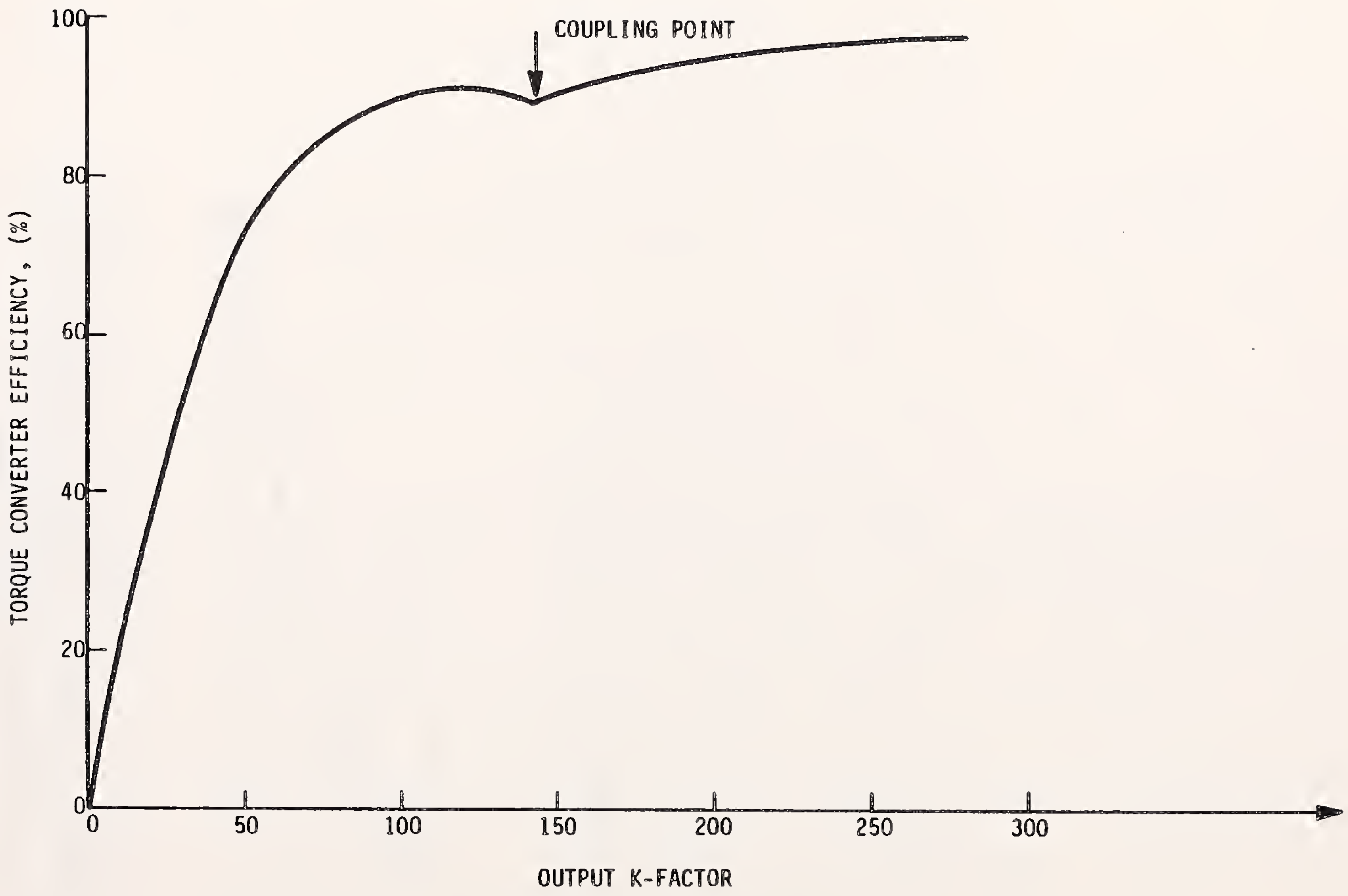


FIGURE 4-5. TORQUE CONVERTOR EFFICIENCY VERSUS OUTPUT K-FACTOR

eliminating those losses. Therefore, a "loose" torque converter will benefit more from lockup than a tight converter. Vehicle cruise and acceleration power demands, identified by vehicle weight, rolling resistance and aerodynamic drag losses affect torque converter efficiency by increasing torque demands and subsequently reducing output capacity factor as shown in equation (4.2.1).

Increasing the drive ratio (defined as the product of gear ratio and axle ratio) increases engine speed and reduces torque. The output capacity factor is given by equation (4.2.2), and is directly proportional to the axle ratio to the 3/2 power:

$$Kout = \frac{RPMout}{[Tqout]^{\frac{1}{2}}} = \frac{V \cdot AR}{[Tqw/AR]^{\frac{1}{2}}} = \frac{V \cdot AR^{3/2}}{[Tqw]^{\frac{1}{2}}} \qquad (4.2.2)$$

where:

Kout = output capacity factor

Tqout = output torque

RPMout = output rpm

V = vehicle velocity

AR = axle ratio, and

Tqw = torque at drive wheels.

The effect of vehicle weight and axle ratio on torque converter energy losses during the highway cycle is illustrated in Figure 4-6. Urban cycle torque converter losses consume 9 percent of engine output at 2.94 axle ratio, versus about 4.5 percent in the highway cycle for the same axle ratio.

4.3 GEAR RATIO RANGE

Extending the gear ratio range of a transmission can improve fuel economy at a particular performance level by allowing a high drive ratio in first and second gears (improved performance) with a low drive ratio in top gear (improved fuel economy). Small

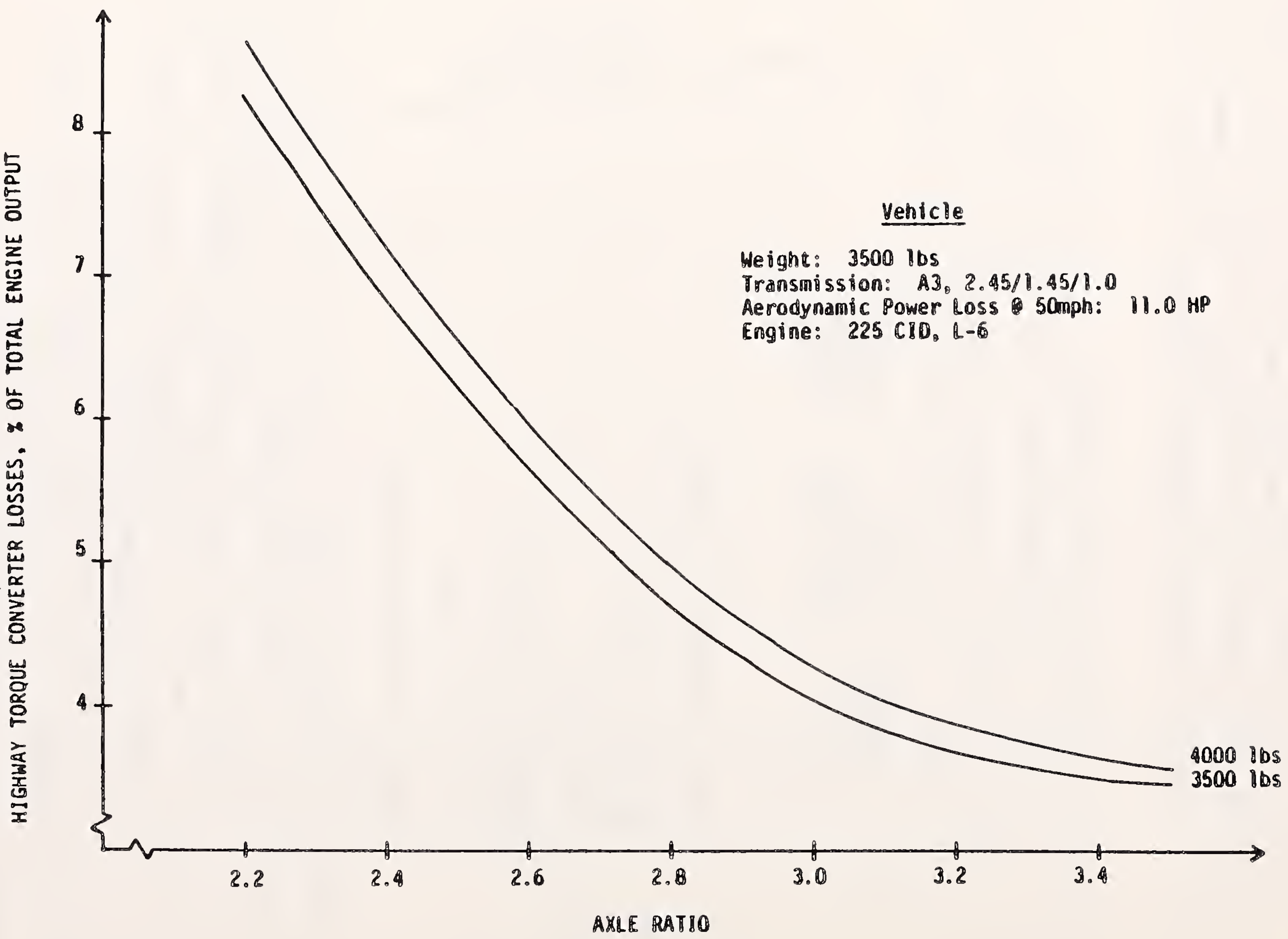


FIGURE 4-6. TORQUE CONVERTER ENERGY LOSSES VERSUS AXLE RATIO

increases in gear ratio range can be accomplished by the modification of gear ratios, but large increases can only be achieved by the addition of gears due to practical limitations of shift smoothness on gear ratio steps. Extended gear ratio range transmissions have been designed by adding an overdrive gear to a typical three speed gearbox. Mercedes[9] has taken a different approach, making top gear a direct drive for reduced noise and improved efficiency. Accordingly, Mercedes employs a low axle ratio and a high (3.68) first gear.

Figure 4-7 shows the fuel economy effects of extending the gear ratio range of an automatic transmission. The "locked" curve is based on a 2.45 gear ratio range three speed locked up, while the "unlocked" curve is based on the same transmission unlocked. The low engine speeds during cruise results in low capacity factor and low torque converter efficiency, so the unlocked case shows a lower improvement. The maximum improvement from extended gear ratio range gearsets is about 6.5 percent for the unlocked case and about 13.5 percent for the locked case.

Figure 4-8 illustrates the effect of lockup on fuel economy as a function of gear ratio range. For a conventional (2.45/1.45/1.0) transmission the improvement is about 7.8 percent; for an extended range (5.95) transmission, the improvement approaches 15 percent.

Figure 4-9 demonstrates the fuel economy effects of lockup and extended gear ratio range, based on a 2.45/1.45/1.0 unlocked transmission. The data produced by Chana[6] is also presented on Figure 4.9 and good agreement is noted between the two sets of data. Without lockup, the maximum improvement attainable through extended range is about 6.5 percent and with lockup, the potential improvement increases to 22 percent. This clearly illustrates the benefit of lockup for extended range transmissions because without lockup, the torque converter becomes the limiting factor on the transmission's effectiveness.

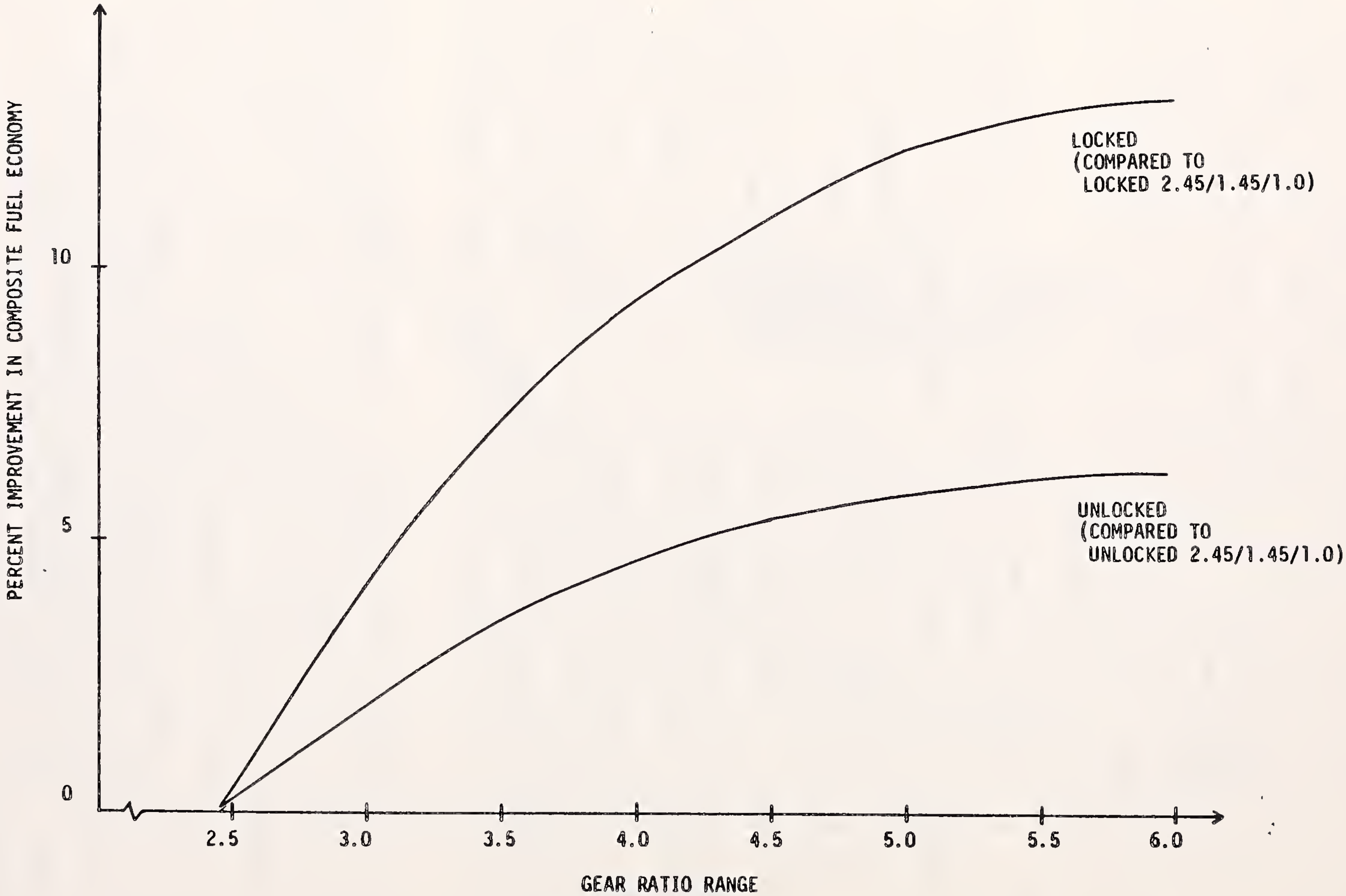


FIGURE 4-7. EFFECT OF GEAR RATIO RANGE ON FUEL ECONOMY

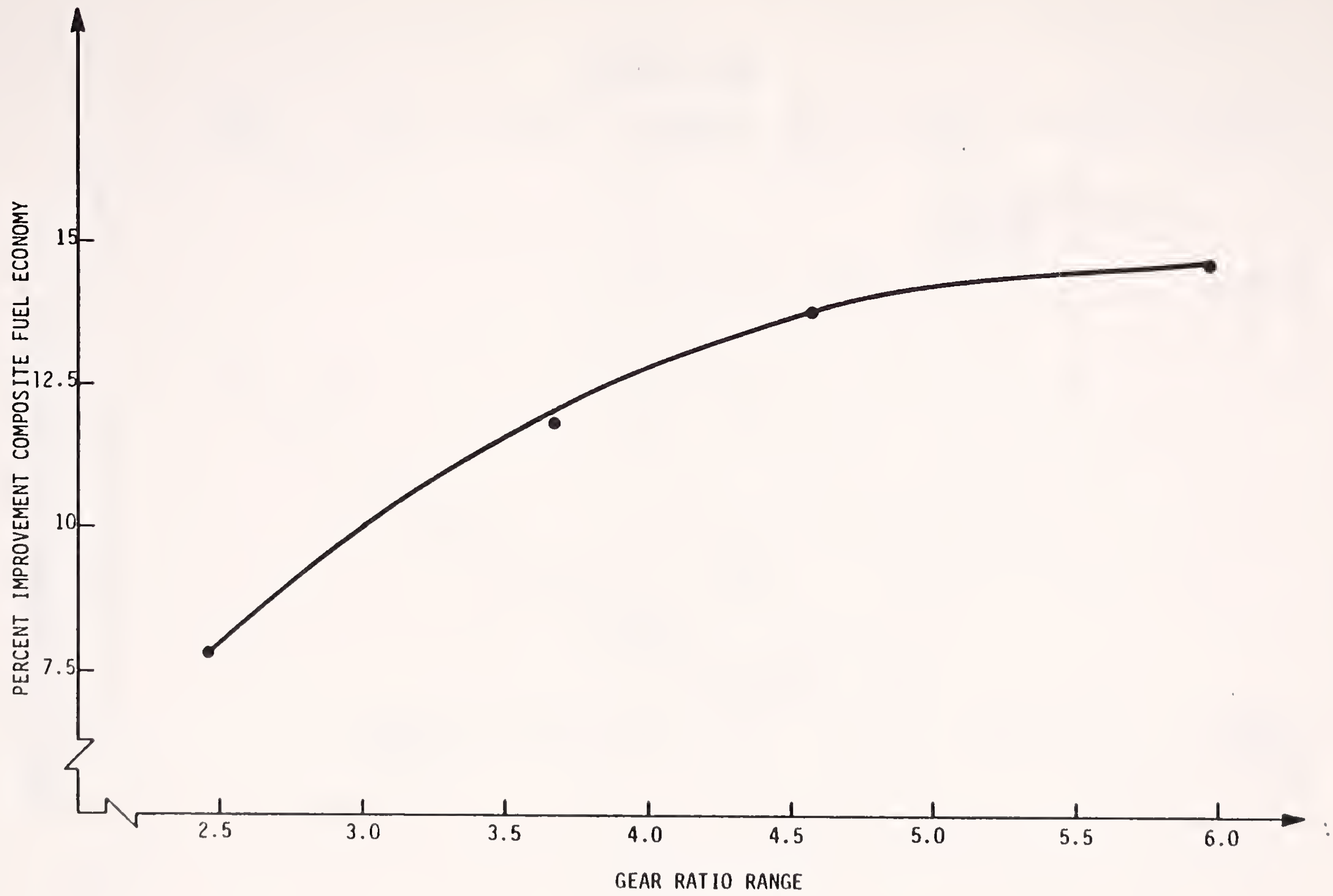


FIGURE 4-8. EFFECT OF LOCKUP ON FUEL ECONOMY AS A FUNCTION OF GEAR RATIO RANGE

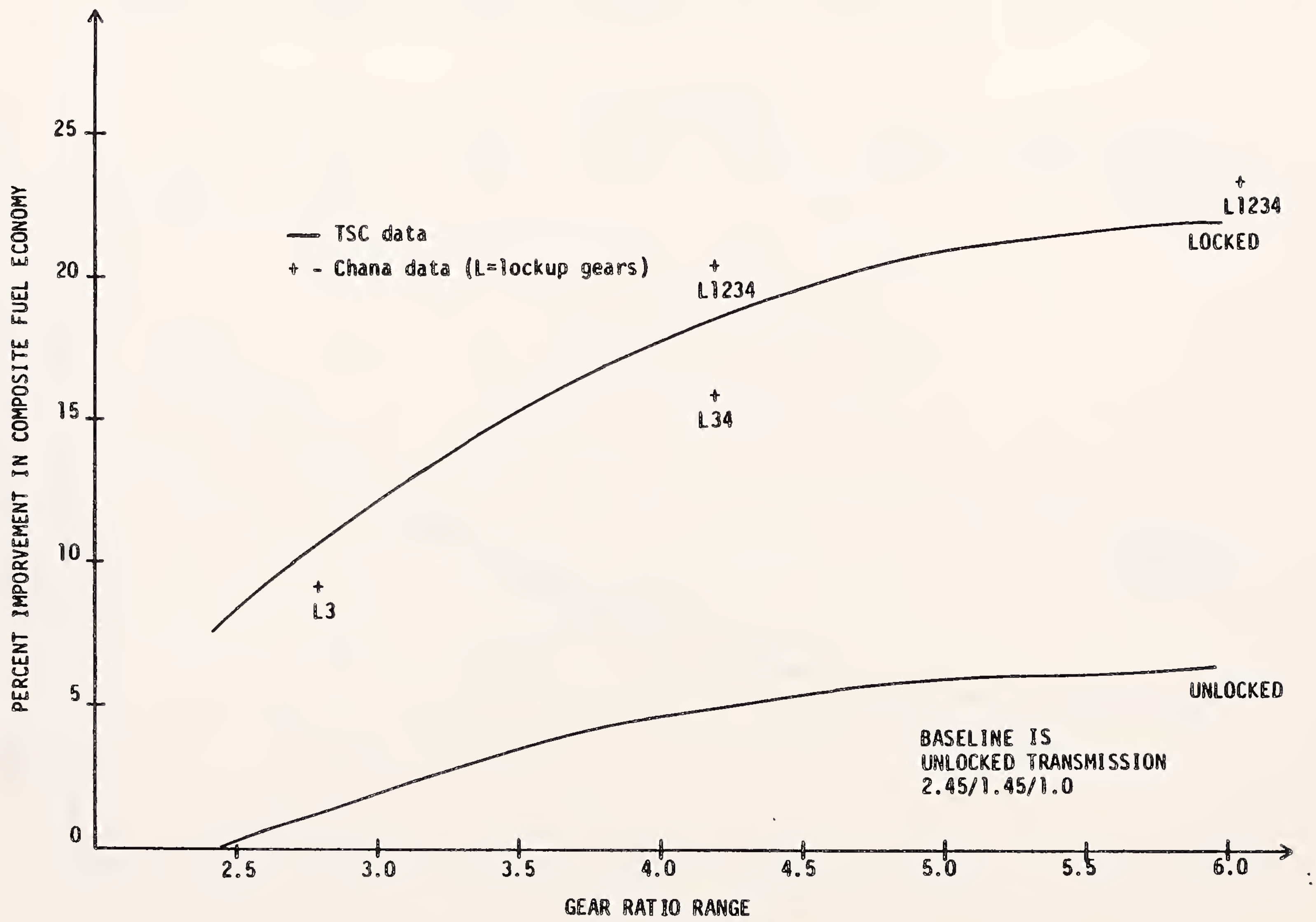


FIGURE 4-9. EFFECT OF GEAR RATIO RANGE AND LOCKUP ON FUEL ECONOMY

Extending the gear ratio range of a three speed manual (3.0/1.7/1.0) by the addition of a 0.73 overdrive produced a 4.2 percent improvement (250 CID engine, 3.0 axle ratio). When the same gear ratio range was spread over five gears (3.0/1.94/1.42/1.0/0.73), the improvement over the three speed was only 1.6 percent, or a fuel economy loss of 2.5 percent from the four speed. This surprising result (noted also by Bickerstaff[2]) is caused by the shift schedule used. The EPA five speed manual shift logic is identical to the four speed, with additional 4-5 and 5-4 shift lines. As Table 4-2 shows, this causes the five speed transmission to spend a lower percentage of its time in top gear. Optimally, the shifts should be compressed, so that the five speed shifts into overdrive when the four speed does. In that case, a slight improvement (0.3 percent) in fuel economy over the four speed of equal range could be expected.[6] For both manuals and automatics, the effect of increasing the number of gears (without increasing range) on fuel economy is mainly a function of the shift logics used. Even with optimal shifting, though, the improvement would be small. Ignoring driveability constraints, the dominant transmission parameter affecting fuel economy is gear ratio range.

5. CONSTANT FIFTY-FIVE-MPH SPEED

5.1 ANALYSIS

The National Highway Traffic Safety Administration (NHTSA) with support from the Federal Highway Administration (FHWA) has convened a Task Force to assess the 55 mph speed limit. NHTSA, in turn, has requested the Transportation Systems Center to promote three tasks to support NHTSA in this effort under TSC's Automotive Fuel Economy Research and Analysis Program (AFER). This section supports Task 3 which is to provide fuel economy sensitivity analyses of vehicle dependent or driver controllable functions for passenger cars and light trucks based on computer simulations.

The six vehicles used in this study are listed in Table 5-1. A baseline case was simulated for each vehicle to determine its baseline fuel economy prior to the analyses. Vehicle characteristics, drive schedules or routes were then modified as independent variables to determine the resulting fuel economy. The resulting fuel economy was then analyzed and compared with the baseline results. The only dependent variable in this section is fuel economy. The independent variables examined individually include vehicle axle ratio, weight, aerodynamic drag coefficient, engine displacement, tire inflation pressure, and certain accessory effects as well as drive cycle variation.

The constant speed baseline fuel economy for each vehicle is shown in Figure 5-1. In this figure and in subsequent tables and figures, vehicles are identified by their respective weights as in Table 5-1. For the indicated speeds (40 to 70 mph), it can be seen that lower constant speeds yield higher fuel economy. Apparent trend anomalies, such as the intersections of the curves for the 3000 lb auto, the 3000 lb truck, and the 4000 and 4500 lb autos, result from drivetrain optimization and engine fuel economy differences between different manufacturers' vehicles (i.e., drive ratios and road load/engine fuel economy optimization).

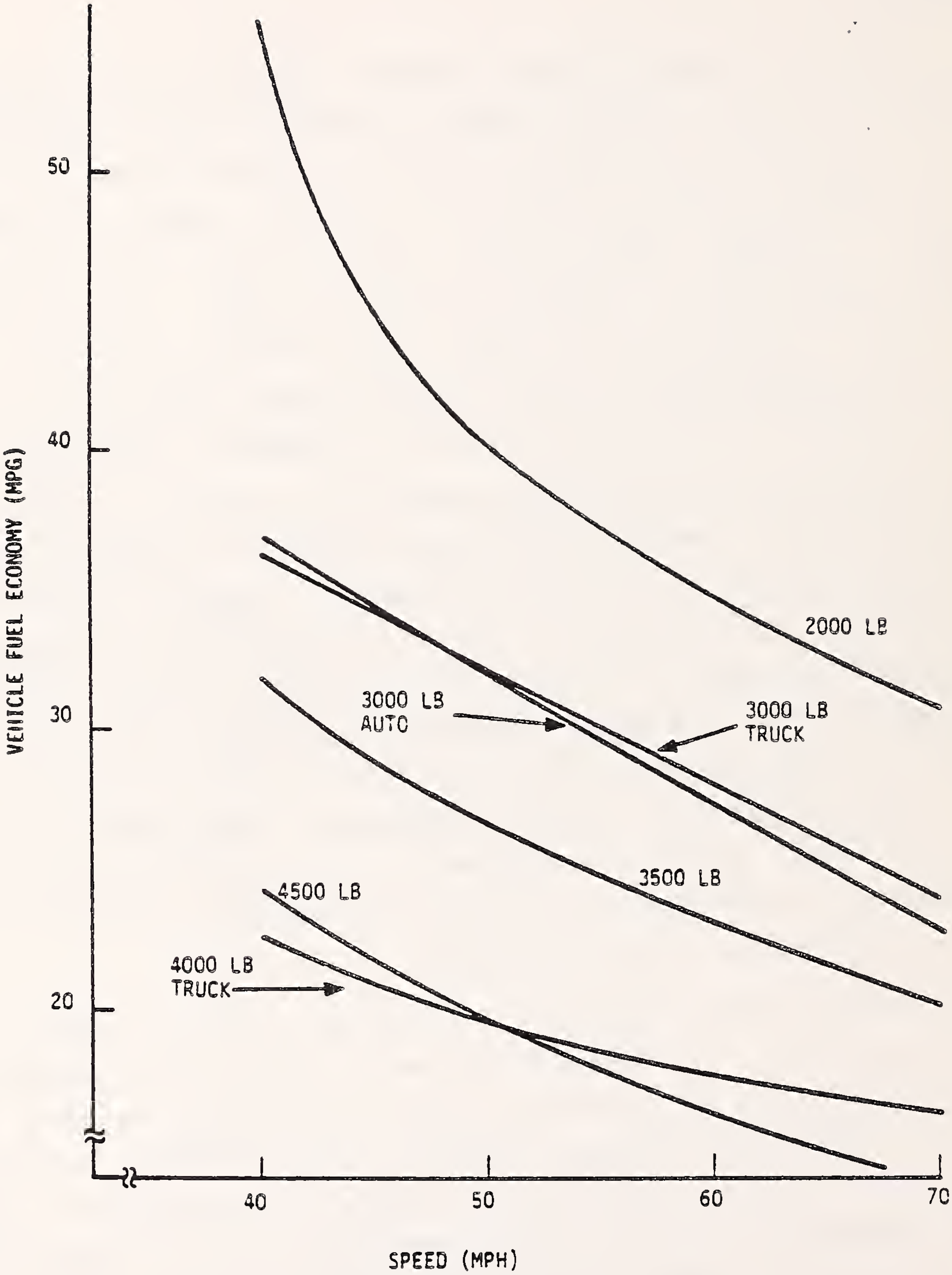


FIGURE 5-1. BASELINE VEHICLE, FUEL ECONOMY VERSUS SPEED

TABLE 5-1. AXLE RATIO CHANGE

FUEL ECONOMY SENSITIVITY TO AXLE RATIO AT 55 MPH

	VEHICLE WEIGHT LBS	BASELINE AXLE RATIO	SENSITIVITY %ΔFE/%ΔAR	
			DECREASE	INCREASE
AUTOS	2000	3.58	.52	.56
	3000	3.91	.35	.22
	3500	2.56	.40	.36
	4500	2.71	.37	.41
TRUCKS	3000	3.08	.37	.45
	4000	3.54	.10	.07

5.2 AXLE RATIO

The effect of axle ratio on fuel economy at 55 mph was determined by modifying the axle ratio 5 percent above and 5 percent below the baseline case. From the results, it is apparent that fuel economy is approximately linear with axle ratio. As shown in Table 5-1, a numerical increase in axle ratio will decrease the fuel economy, and a numerical decrease will increase the fuel economy. Because the effect of an axle ratio change is most noticeable in top gear, it would probably be more practical to modify the top gear ratio (i.e., overdrive) as the modification of the axle ratio will affect performance throughout all the gear ratios. If it is decided to modify the top gear ratio, these results remain applicable as the overall result is to modify the final drive ratio (rpm/mph) for either the axle ratio or top gear ratio.

5.3 WEIGHT CHANGE

For a vehicle traveling over a given drive cycle, force is required to overcome aerodynamic drag, accessory loads, and rolling resistance, and to provide acceleration. However, at constant speed, the weight only affects the rolling resistance, and, therefore, any change in fuel economy is attributable to rolling resistance only. Each vehicle was simulated at a weight 100 and 300 pounds greater than the baseline case. The results are shown in Table 5-2.

The fuel economy decrease is dependent on the energy consumption of a vehicle at 55 mph. For example, approximately 70 percent of the energy consumed is used to overcome air resistance for the 4000 lb truck. Therefore, a small weight increase, which has no effect on vehicle drag, will not significantly affect the fuel economy for this vehicle.

TABLE 5-2. WEIGHT CHANGE

SENSITIVITY OF FUEL ECONOMY TO WEIGHT REDUCTION AT 55 MPH

	BASELINE VEHICLE WEIGHT LBS	SENSITIVITY %ΔFE/%ΔWT
AUTOS	2000	0.14
	3000	0.22
	3500	0.21
	4500	0.32
TRUCKS	3000	0.19
	4000	0.14

5.4 AERODYNAMIC DRAG COEFFICIENT

For the vehicles studied, the percent of the total energy expended to overcome aerodynamic drag is shown in Table 5-3. The aerodynamic drag force acting on a vehicle is a function of air density, frontal area, drag coefficient, and the square of the vehicle speed. The drag coefficient is, basically, a function of the shape of the vehicle. The effect on fuel economy of an increase of 0.1 in the drag coefficient is shown in Table 5-4.

5.5 DISPLACEMENT

The effect of engine displacement on fuel economy was determined by scaling the baseline engines. The resulting scaled engines have fuel rate and torque characteristics similar to the baseline engines, except that the absolute values are different. The change in displacement has basically the same effect as modifying the axle ratio. Decreasing the displacement will increase the fuel economy; increasing it will decrease the fuel economy. Also, the acceleration performance subsequently will be affected, as in the case of an axle ratio change. The results are presented in Table 5-5.

5.6 TIRE PRESSURE

The rolling resistance of a vehicle is a function of the vehicle weight, rolling resistance coefficient, and vehicle speed. Assuming that the load of the vehicle is constant, the pressure effects can be related to the rolling resistance coefficient and then to fuel economy as shown in Table 5-6. Because each tire has its own characteristics, these results only apply to the range of conditions indicated.

5.7 ACCESSORIES

Two accessory loads were simulated for each vehicle. The first involves the effect of air conditioning on fuel economy as shown in Table 5-7. The fuel economy decrease assumes a 100 percent air conditioner duty cycle. The effect of ambient temperature

TABLE 5-3. AERODYNAMIC PERCENTAGE OF TOTAL ENERGY EXPENDED

VEHICLE WEIGHT LBS.		VEHICLE VELOCITY	
		55 MPH	70 MPH
AUTO	2000	61	70
	3000	46	68
	3500	45	63
	4500	53	63
TRUCK	3000	63	73
	4000	66	73

TABLE 5-4. FUEL ECONOMY RESPONSE TO AERODYNAMIC DRAG INCREASE

FUEL ECONOMY SENSITIVITY TO AERODYNAMIC DRAG

	VEHICLE WEIGHT LBS	BASELINE DRAG COEFFICIENT (C_D)	SENSITIVITY $\%\Delta FE/\%\Delta C_D$	
			55 MPH	70 MPH
AUTOS	2000	0.46	0.30	0.40
	3000	0.50	0.26	0.51
	3500	0.55	0.34	0.40
	4500	0.60	0.45	0.75
TRUCKS	3000	0.50	0.37	0.58
	4000	0.58	0.37	0.57

TABLE 5-5. ENGINE DISPLACEMENT CHANGE

SENSITIVITY OF FUEL ECONOMY AT 55 MPH TO ENGINE DISPLACEMENT

	VEHICLE WEIGHT LBS	BASELINE DISPLACEMENT (LITER)	SENSITIVITY %ΔFE/%ΔCID	
			-10% CID	+10% CID
AUTOS	2000	1.6	.52	.43
	3000	2.1	.29	.31
	3500	3.8	.36	.38
	4500	5.2	.25	.24
TRUCKS	3000	2.3	.40	.33
	4000	5.8	.50	.40

TABLE 5-6. EFFECT OF TIRE PRESSURE ON FUEL ECONOMY AT 55 MPH

SENSITIVITY OF FUEL ECONOMY AT 55 MPH TO TIRE PRESSURE

	VEHICLE WEIGHT LBS	BASELINE TIRE PRESSURE (PSI)	SENSITIVITY %ΔFE/%ΔPSI	
			-20% PSI	+10% PSI
AUTOS	2000	27	.085	.03
	3000	27	.065	.06
	3500	27	.12	.08
	4500	27	.13	.07
TRUCKS	3000	23	.09	.07
	4000	50	.035	.03

TABLE 5-7. EFFECT OF ACCESSORY LOADS ON FUEL ECONOMY AT 55 MPH

	VEHICLE WEIGHT LBS	FUEL ECONOMY DECREASE FROM MAXIMUM AIR CONDITIONING USE	FUEL ECONOMY DECREASE FROM LIGHTING LOAD
AUTOS	2000	18%	2.2%
	3000	17%	2.5%
	3500	12%	2.3%
	4500	12%	2.3%
TRUCKS	3000	14%	2.3%
	4000	7%	1.4%

conditions was not considered. The second accessory simulation is concerned with lighting load. The lighting load was determined by adding the power rating (watts) of all vehicle lamps used in night driving for an average sized vehicle. The night driving lighting load consists of low beam head lamps and side, parking, tail, and license lamps. The increase in power was then transferred to the alternator. The fuel economy decrease due to an increase in alternator load representative of night driving is presented in Table 5-7.

5.8 DRIVING SCHEDULE

A comparison was made between steady-state fuel economy at 55 mph and 70 mph and driving schedules that include speed changes. The schedules include the SAE interstate 55 and 70 mph driving schedules which are listed in Table 5-8 and modifications of these schedules. The interstate 55 mph cycle covers 4.7 miles in 308 seconds (average 55 mph), while the interstate 70 mph cycle transverses 4.7 miles in 242 seconds (average 70 mph). In each modified cycle, constant speed operation was eliminated and the first modification (M1) is thus the result with the constant speed segments excluded. In the second modification (M2), the acceleration rate was doubled. In the third modification (M3), the speed change interval was increased, from 10 mph to 20 mph, but the acceleration rate was identical to the SAE cycles. The fourth change (M4) was identical to the third except that the acceleration rate doubled. The purpose of these changes was to emphasize the importance of the drive cycle on fuel economy. Comparisons between constant speed fuel economy and these drive schedules are presented in Tables 5-9 and 5-10. From the results, it can be seen that there is a large range of fuel economy changes for the various drive schedules.

As can be observed from Table 5-9, small improvements in fuel economy are predicted for the 2000 lb passenger car and the 4000 lb truck. This, however, would not normally be experienced in on-road testing because of transient vehicle characteristics.

TABLE 5-8. SAE INTERSTATE DRIVING SCHEDULES

55-MPH CYCLE

DISTANCE (MILE)	OPERATION
0.0	APPROACH STARTING LINE AT 55 MPH AND PROCEED TO 0.2-MILE MARKER
0.2	ACCELERATE TO 60 MPH AT 1FT/SEC^{2*}. IMMEDIATELY DECELERATE TO 50 MPH AT 1FT/SEC^2. IMMEDIATELY ACCELERATE TO 55 MPH AT 1FT/SEC^2. PROCEED AT 55 MPH TO THE 1.2-MILE MARKER
1.2	REPEAT ACCELERATIONS AND DECELERATIONS AS AT 0.2-MILE MARKER 3 TIMES UNTIL 4.7-MILE MARKER IS REACHED

70-MPH CYCLE

DISTANCE (MILE)	OPERATION
0.0	APPROACH STARTING LINE AT 70 MPH AND PROCEED TO 0.2-MILE MARKER
0.2	ACCELERATE TO 75 MPH AT 1FT/SEC^2. IMMEDIATELY DECELERATE TO 65 MPH AT 1FT/SEC^2. IMMEDIATELY ACCELERATE TO 70 MPH AT 1FT/SEC^2. PROCEED AT 70 MPH TO THE 1.2-MILE MARKER.
1.2	REPEAT ACCELERATIONS AND DECELERATIONS AS AT 0.2 MILE MARKER UNTIL 4.7-MILE MARKER IS REACHED

*An acceleration of 1ft/sec^2 is equivalent to 0.68 mi/hr/sec or accelerating from 55 MPH to 60 MPH in 7.3 sec.

TABLE 5-9. COMPARISON OF 55-MPH FUEL ECONOMY TO DRIVE CYCLE FUEL ECONOMY

	VEHICLE WEIGHT LBS	CONSTANT SPEED FUEL ECONOMY MPG	DRIVE CYCLE % CHANGE IN FUEL ECONOMY FROM BASELINE				
			SAE 55	M1	M2	M3	M4
AUTOS	2000	35.9	+1.4	+4.2	+2.2	+2.8	-1.3
	3000	29.8	-2.3	-6.0	-10.7	-8.1	-12.0
	3500	24.7	-1.6	-3.6	-13.0	-4.0	-14.7
	4500	18.0	-3.3	-8.3	-21.7	-8.9	-22.2
TRUCK	3000	30.1	-1.7	-4.0	-15.9	-5.3	-17.3
	4000	17.4	+1.1	+4.6	+1.7	+1.7	-2.9

NOTE: For explanation of cycles MI-M4, see Section 5.8.

TABLE 5-10. COMPARISON OF 70-MPH FUEL ECONOMY TO DRIVE CYCLE FUEL ECONOMY

	VEHICLE WEIGHT LBS	CONSTANT SPEED FUEL ECONOMY MPG	DRIVE CYCLE % CHANGE IN FUEL ECONOMY FROM BASELINE				
			SAE 70	M1	M2	M3	M4
AUTOS	2000	30.1	-1.7	-3.6	-18.6	-5.0	-18.3
	3000	22.6	-1.3	-2.2	-4.4	-4.0	-6.2
	3500	20.1	0	0	-18.4	-2.5	-18.4
	4500	13.8	-1.0	-1.5	-10.1	-2.9	-10.9
TRUCK	3000	24.0	-3.8	-7.1	-15.4	-16.3	-16.3
	4000	15.1	-2.0	-4.6	-16.6	-6.7	-17.9

NOTE: For explanation of cycles MI-M4, see Section 5.8.

Because simulation results are obtained from steady-state engine performance measurements and engine efficiency is load dependent, these small inversions can sometimes occur even though the work performed by the vehicle is always greater under transient operations. The following calculation, comparing the SAE interstate 55 cycle with constant speed fuel economy for the 2000 lb vehicle, demonstrates this effect.

The instantaneous fuel economy is calculated by:

$$FE = \frac{1}{BSFC} \cdot \frac{1}{hp} \cdot \rho f \cdot \overline{S} \tag{5.8.1}$$

where

- FE = Fuel economy (mpg)
- BSFC = Brake specific fuel consumption (lb/hp-hr)
- hp = Engine power (hp)
- ρf = Fuel density
- $\overline{S}$ = Average speed.

In comparing the fuel economy of the 2000 lb automobile for constant 55 and SAE 55, it is assumed that the fuel density and distance are equal for both cases. The fuel economy is then calculated by finding the engine horsepower and brake specific fuel consumption for both cases. The engine horsepower will change proportionately as the road load horsepower changes, because the drivetrain and accessory losses will not vary significantly between 50 and 60 mph. The road load for the SAE 55 must be greater than the constant 55, because power is based on the cube of velocity for aerodynamic drag and the square of velocity for rolling resistance. This means that the SAE 55 case vehicle must operate at a lower average BSFC than the constant 55 case. A summation of this calculation is presented in Table 5-11.

TABLE 5-11. BRAKE SPECIFIC FUEL CONSUMPTION

CYCLE	BSFC	ENGINE hp	% TIME
Constant 55	0.58	15.8	100%
SAE 55	0.58	15.8	64%
	0.45	28.3	9%
	1.00	5.9	17%
	0.46	24.0	9%

Substituting the numbers from Table 5-11 into equation (5.8.1) yields,

$$\left(\frac{1}{0.58\ \frac{\text{lbs}}{\text{hp}\cdot\text{hr}}}\right)\left(\frac{1}{15.8_{\text{hp}}}\right)\ (55\ \text{mi/hr})\ \left(5.98\ \frac{\text{lbs}}{\text{GAL}}\right) = 35.9\ \text{mpg.} \tag{5.8.2}$$

$$FE_{\text{SAE } 55} = \left(\frac{1}{9.01\ \frac{\text{lbs*}}{\text{HR}}}\right)\ (55\ \text{mi/hr})\ \left(5.98\ \frac{\text{lb}}{\text{GAL}}\right)$$

$$= 36.5\ \text{mpg.} \tag{5.8.3}$$

This calculation indicates that, for this particular vehicle, it is possible to obtain better fuel economy over the SAE 55 than constant 55 mph. The word, particular, is emphasized since each engine has its own fuel consumption characteristics.

5.9 SYNERGETIC EFFECTS

The fuel economy influences of the individual design variables have been evaluated separately. The net effect of all the variables cannot be accurately estimated by adding the individual effects, because engine efficiency changes as a function of load. For small changes, however, the accuracy of the estimate is not seriously compromized. Therefore, both cases (adding individual effects in combination and simulating all changes simultaneously) were considered. The comparison of the two methods is shown in Table 5-12. The air conditioning and lighting load were not included because they are duty-cycle dependent. If the combined effect of these accessories is desired, an estimate can be made by adding their respective individual effects.

*The average fuel flow rate of 9.01 lbs/hr is obtained by a summation of the BSFC, engine hp and % time values shown for the SAE 55 cycle in Table 5-11.

TABLE 5-12. COMBINED EFFECT OF VEHICLE VARIABLES ON FUEL ECONOMY

	AUTO				LIGHT TRUCK	
VEHICLE WEIGHT (LB)	2000	3000	3500	4500	3000	4000
1. Increased aerodynamic Drag (C_D increased by 0.1)	6.5	8.1	6.8	7.5	7.4	6.4
2. Increased weight (3000 lbs)	2.2	2.2	1.9	2.1	1.9	1.0
3. Increased Displacement (10%)	4.3	3.1	3.8	2.4	3.3	4.0
4. Increase axle ratio (5%)	2.7	0.5	2.2	2.0	1.9	0.6
5. Decrease tire pressure (20%)	1.7	1.3	2.4	2.6	1.8	0.7
All, combined (1)	17.4	15.2	17.1	16.6	16.3	12.7
All, combined (2)	16.1	15.9	17.6	15.1	16.0	11.0
6. Air conditioning	Duty Cycle Dependent					
7. Lighting	2.2	2.5	2.3	2.3	2.3	1.4

NOTES: (1) Individual effects added.
(2) All changes simulated simultaneously.

6. ACCESSORIES

The simulated effect of accessory loading on fuel economy can be approached by different techniques. For example, the effect of air conditioning can be determined by cycling a fully loaded torque curve (vehicle simulation input) and an unloaded torque curve by means of a duty cycle. Alternatively, this effect can be determined by increasing the dynamometer power 10 percent up to a maximum of 1.4 hp to account for air conditioning.[15] The latter method was chosen as this approach eliminates experimental error and duty cycle variations. The analytical procedure involved utilizing graphs of aerodynamic drag coefficient or PAU setting versus fuel economy. The baseline aerodynamic drag is simply extrapolated 10 percent and the reduced fuel economy is compared to that of the baseline fuel economy as shown in Table 6-1. The 1.4 hp maximum increase only effects the light duty trucks as their baseline dynamometer horsepower settings are in the range of 17 hp. This is most likely due to the low application of air conditioning to light duty trucks. The results indicate the fuel economy penalty due to air conditioning using this method is approximately 1.9 percent.

The effect of alternator load on fuel economy was determined by using laboratory test data on a 1980 model year alternator used on a 2.8ℓ engine. The data was obtained by varying the output watts of the alternator and recording the corresponding torque. These torque curves were then input to VEHSIM to simulate different power loads required by the alternator. The results of the alternator loads on fuel economy are shown in Table 6-2. A typical vehicle load for night driving is approximately 130 watts and the maximum load of the alternator is 335 watts.

TABLE 6-1. COMPOSITE FUEL ECONOMY PENALTY DUE TO AIR CONDITIONING

VEHICLE	ENGINE (ℓ)	PAU (HP)	%Δ (MPG)
AUTO	1.6	7.4	2.1
AUTO	2.1	9.7	1.6
AUTO	3.8	9.3	1.8
AUTO	5.2	11.2	1.8
TRUCK	2.3	10.3	1.7
TRUCK	3.7	16.5	1.8
TRUCK	4.1	17.6	2.2
TRUCK	5.7	17.0	1.9

TABLE 6-2. FUEL ECONOMY FOR DIFFERENT ALTERNATOR LOADS

VEHICLE	ADDITIONAL ALTERNATOR LOAD ABOVE BASELINE (WATTS)	PERCENT CHANGE FROM BASELINE, $\frac{(FE)_C - (FE)_C^*}{(FE)_C^*} \times 100\%$
2000-lb AUTOMOBILE	*	-
	30	-0.59
	60	-1.33
	100	-2.27
	140	-3.11
	335	-4.47
3000-lb AUTOMOBILE	*	-
	30	-0.36
	60	-0.70
	100	-1.08
	140	-1.66
	335	-2.45
3500-lb AUTOMOBILE	*	-
	30	-0.28
	60	-0.56
	100	-0.96
	140	-1.28
	335	-2.15
4500-lb AUTOMOBILE	*	-
	30	-0.28
	60	-0.50
	100	-0.82
	140	-1.12
	335	-1.74
3000-lb MINI-PICKUP	*	-
	30	-0.40
	60	-0.77
	100	-1.37
	140	-1.84
	335	-2.93

* Baseline.

7. DIESEL SENSITIVITIES

All previous vehicles studied were equipped with spark ignition engines. Because of the increase in diesel engine applications in automobiles, a sensitivity analysis was conducted for vehicles with diesel engines. The baseline vehicles were chosen on the availability of steady-state diesel engine maps provided for VEHSIM use. Two vehicles were configured; one for a 90 CID (1.5ℓ)engine and one for a 183 CID (3.0ℓ) engine application as shown in Table 7-1. The fuel economy sensitivities were determined by varying the independent variables N/V, dynamometer horsepower, vehicle weight, and engine displacement. The N/V sensitivity for the 3.0ℓ engine vehicle was not included because the estimated shift logic influenced this number yielding unrepresentative results. However, the remaining sensitivities were calculated and are compared to previous spark ignition results as shown in Table 7-2.

The application of these simulation results is enhanced by comparing the simulated baseline fuel economy to that of the 1980 EPA fuel economy numbers. The EPA did not provide fuel economy estimates for the 3.0ℓ engine 4000 lb vehicle, but the 1.5ℓ engine 2375 lb vehicle compared favorably to the VEHSIM simulation result as shown in Table 7-3.

TABLE 7-1. VEHICLES SIMULATED WITH DIESEL ENGINES

VEHICLE	WEIGHT (LBm)	ENGINE (CID)	AR	TRANSMISSION	PAU (HP)
Mercedes	4000	183	3.46	A4	13.2
Volkswagon	2375	90	3.90	M5	6.8

TABLE 7-2. COMPARISON OF SPARK IGNITION AND DIESEL SENSITIVITIES

VEHICLE	VARIABLE				
	N/V	PAU	WT	CID	HP/WT
Diesel Powered					
3.0ℓ	-	.20-.25	.27-.36	.36	.58-.74
1.5ℓ	.37-.46	.22-.23	.29-.33	.43-.46	.62-.87
Spark Ignition Engine Powered (Previous calc.)					
o Regression	.40	.18	.40	.40	
o VEHSIM	.43	.17	.26	.63	

TABLE 7-3. COMPARISON OF SIMULATION WITH 1980 EPA DATA (2375 IWT - 1.5ℓ DIESEL)

CYCLE	VEHSIM	EPA	%Δ
URBAN	40.25	42	4.3
HIGHWAY	54.98	56	1.8
COMPOSITE	45.77	47	2.6

8. TIRE PRESSURE

By increasing tire inflation pressure, it is possible to reduce hysteretic losses and, consequently, tire rolling resistance. The sensitivity of rolling resistance to inflation pressure is usually quantified by C_p, which is a function of tire material and design. By using C_p rolling resistance can then be estimated from:

$$F_r = F_{ro}\left(\frac{F_z}{F_{zo}}\right)\left[1 + C_p\left(\frac{p_o}{p} - 1\right)\right] \tag{8.1}$$

where F_r = rolling resistance force
F_z = load on tire
C_p = pressure sensitivity
p = inflation pressure

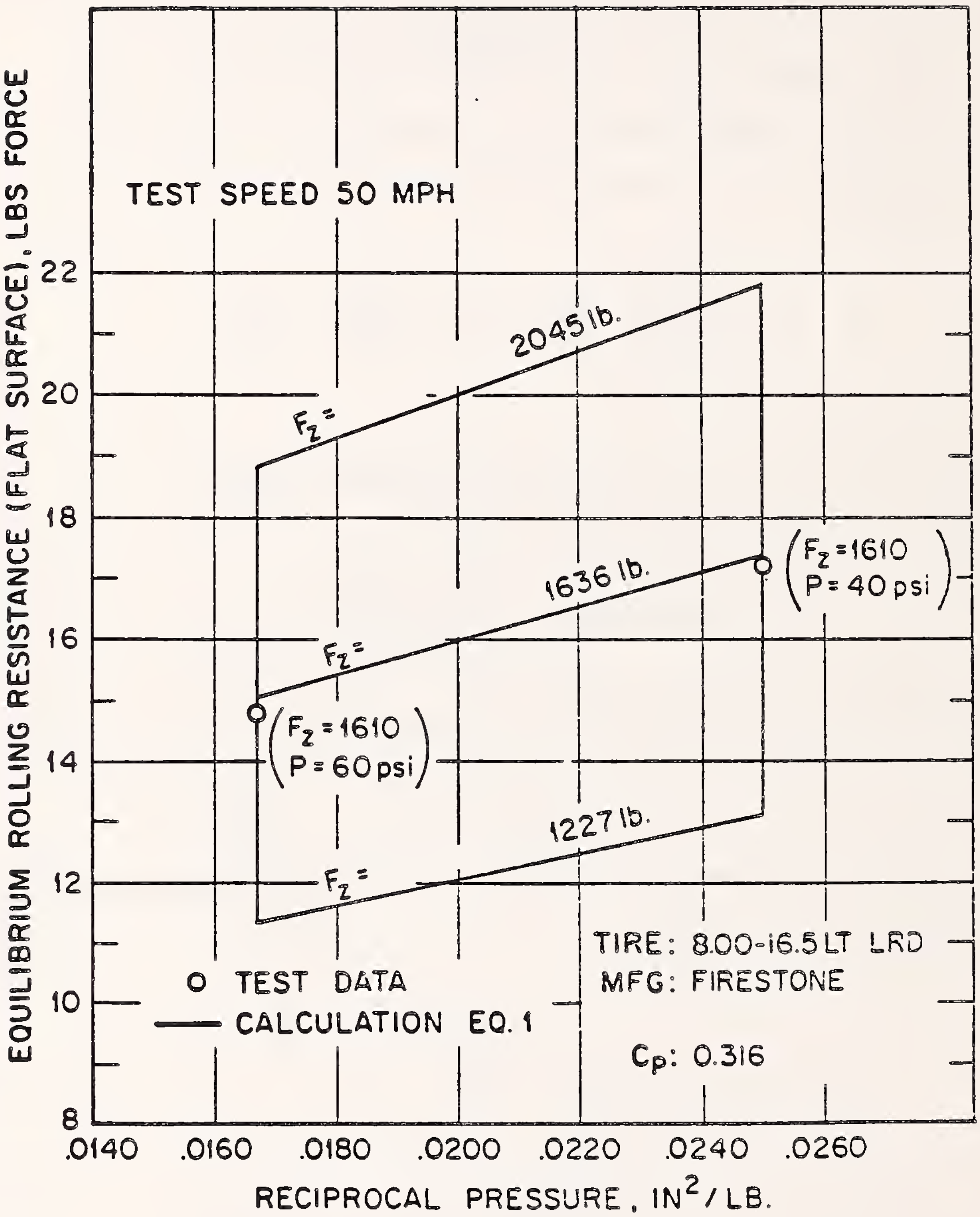
and F_{ro} is determined experimentally for a given p_o and F_{zo}.

Equation (8.1) can be rearranged to yield a coefficient of rolling resistance ($C_1 = \frac{F_r}{F_z}$),

$$C_1 = C_{1o}\left[1 + C_p\left(\frac{p_o}{p} - 1\right)\right] \tag{8.2}$$

where: $C_{1o} = \dfrac{F_{ro}}{F_{zo}}$.

Tire data is generally obtained by observing F_r for various combinations of inflation pressure and load. A "carpet plot" is then generated by plotting rolling resistance versus reciprocal inflation pressure for constant load lines, as shown in Figure 8-1.[13] Using the carpet plot generated for a given tire



Source: Reference 13.

FIGURE 8-1. EQUILIBRIUM ROLLING RESISTANCE (FLAT SURFACE) VERSUS LOAD AND INFLATION PRESSURE: FIRESTONE 8.00-16.5 LT LRD it is possible to evaluate C_p, and then, using equation (8.2), plot the coefficient of rolling resistance versus tire pressure. In Figure 8.2, this analysis has been performed for the tire of Figure 8-1.

Finally, the relationship of the coefficient of rolling resistance to composite fuel economy can be determined by utilizing previous VEHSIM runs (Section 3.5). A crossplot of fuel economy versus tire pressure will then reveal the relationship of composite fuel economy to tire inflation pressure, as shown in Figure 8-3.

The final results are strongly influenced by the initial calculation of C_p, which determines how strongly rolling resistance is affected by inflation pressure. The parameter C_p varies over a wide range 0.18 - 0.514 for the light truck tires listed;[13] therefore a few examples and the corresponding C_p are listed in Appendix C. A number of vehicles are also listed, each exhibiting a particular sensitivity of fuel economy to rolling resistance.

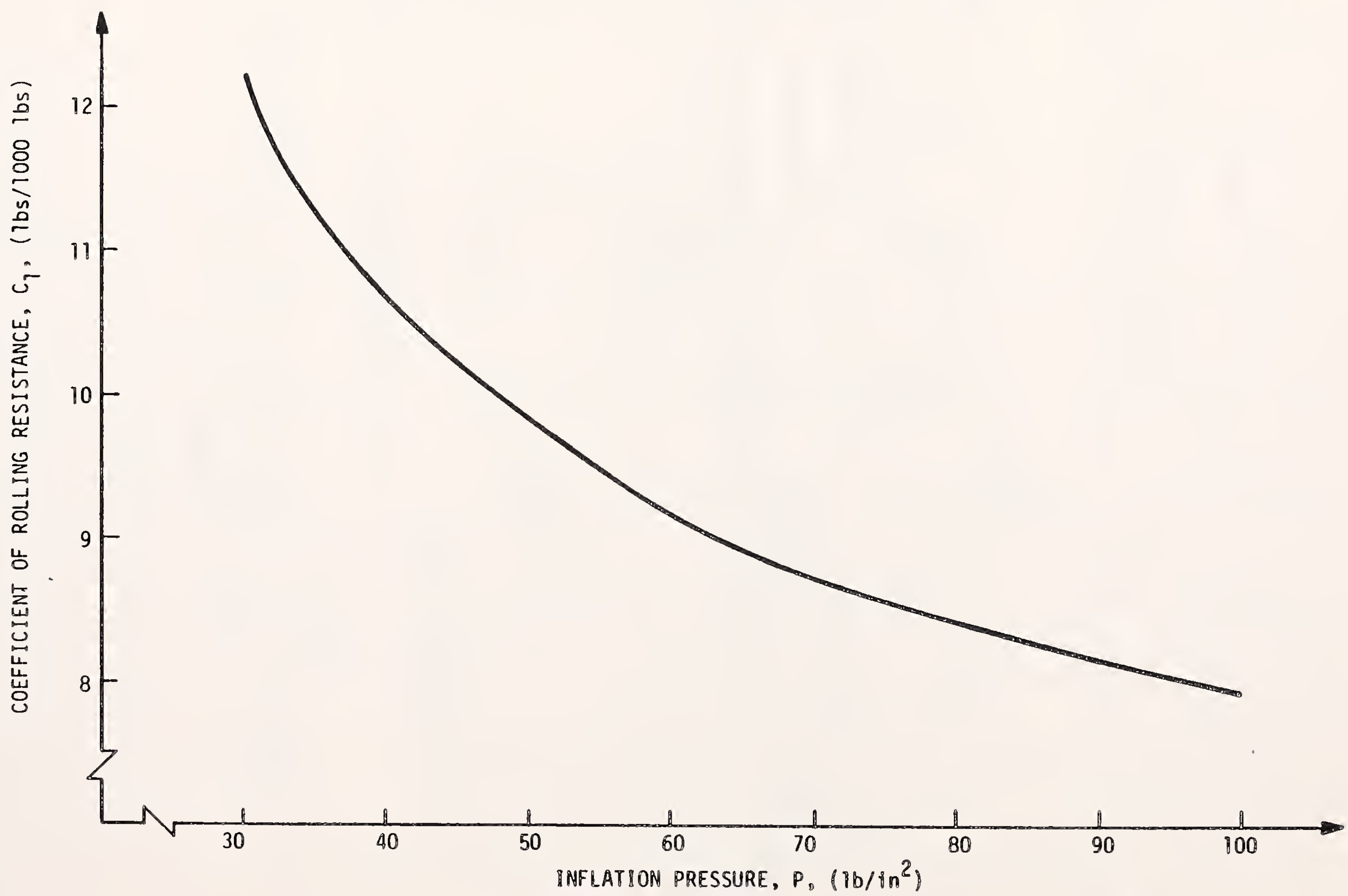


FIGURE 8-2. COEFFICIENT OF ROLLING RESISTANCE VERSUS INFLATION PRESSURE FIRESTONE 8.00-16.5 LT LRD

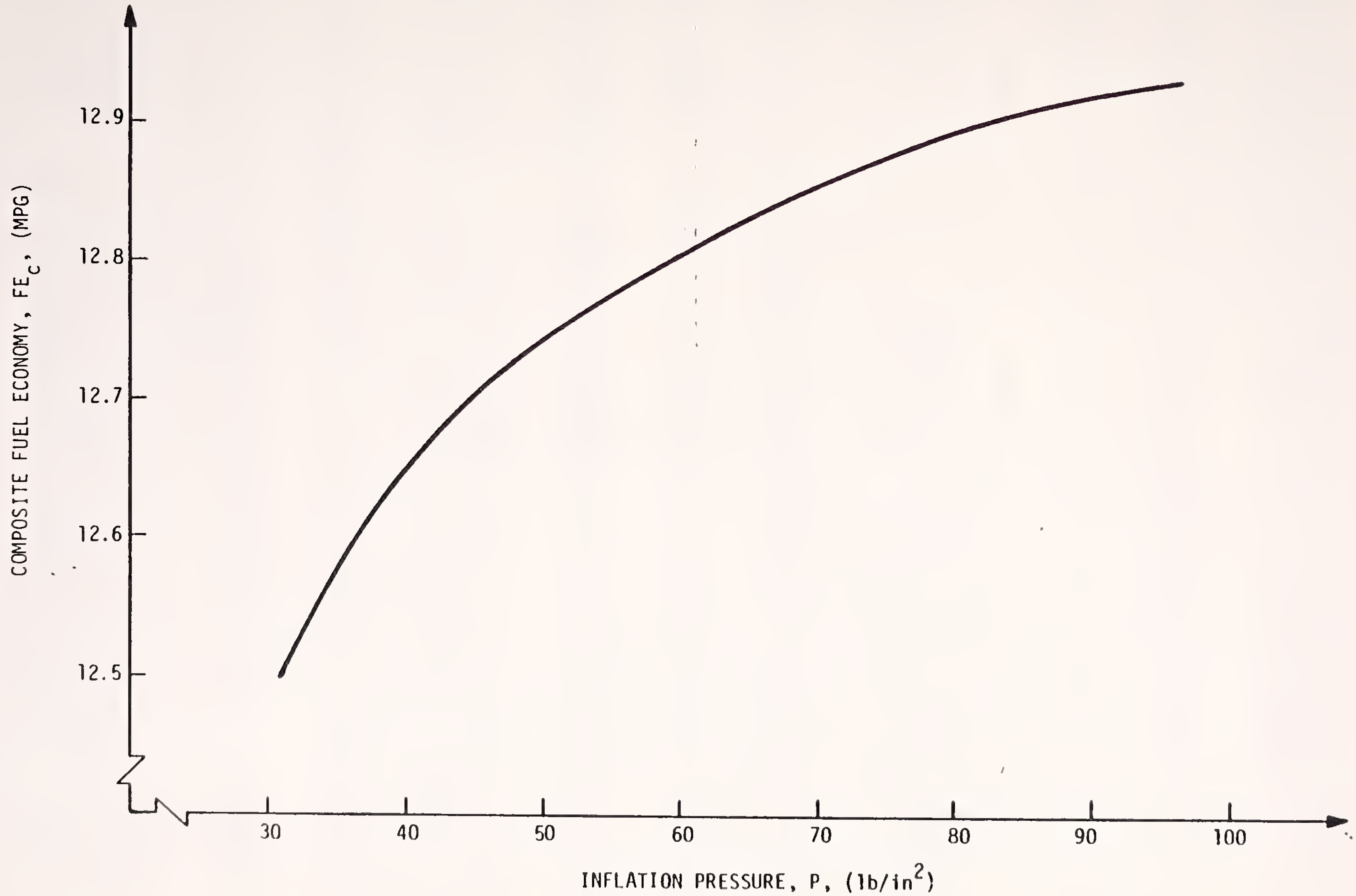


FIGURE 8-3. COMPOSITE FUEL ECONOMY VERSUS INFLATION PRESSURE FOR 4000-LB PICKUP

9. REFERENCES

1. Zub, R.W. et al, "Potential of Spark Ignition Engine, Effect of Vehicle Design Variables on Top Speed, Performance, and Fuel Economy," U.S. Department of Transportation, Transportation Systems Center, Report No. DOT-TSC-NHTSA-79-53, DOT-HS-805 133, March 1980.

2. Bickerstaff, D.J., "Light Truck Fuel Economy by Design Efficiency," SAE Paper 781063, February 1978.

3. Chevrolet Truck Data Book, Chevrolet Motor Division, September 1979.

4. Ford Truck Data Book, Ford Motor Company, February 1979.

5. Zub, R.W. et al.," Support to 1982-1985 Light Truck Fuel Economy Standards," U.S. Department of Transportation, Transportation Systems Center, Material on file, January 1980.

6. Chana, H.E., Fedewa, W.L., and Mahoney, J.E., "An Analytical Study of Transmission Modifications as Related to Vehicle Performance and Economy," SAE Paper 770418, March 1977.

7. 1978 Motor Vehicle Manufacture's Association Specifications, Ford LTD.

8. Numazawa, A., Kubo, S., Shindo, Y., and Moroto, S., "Toyota Four-Speed Automatic Transmission with Overdrive," SAE Paper 780097, March 1978.

9. Forster, H.J., "Ways of Improving Fuel Economy in Vehicles with Automatic Transmissions," presented at First International Fuel Economy Research Conference," Nov. 1, 1979.

10. Harmon, K.B. and Schaefer, R.H., "Shift Modulation for Allison Automatic and Powershift Transmissions," SAE Paper 730839, February 1973.

11. Colello, R.G., "Automobile Drivelines," U.S. Department of Transportation, Transportation Systems Center, Report No. DOT-TSC-OST-77-34, May 1977.

12. Hurter, D.A. and Gott, P.G., "Evaluation of Automobile Drivetrain Components to Improve Fuel Economy," U.S. Department of Transportation, Transportation Systems Center, Report No. DOT-TSC-NHTSA-79-12/DOT-HS-803 840, May 1979.

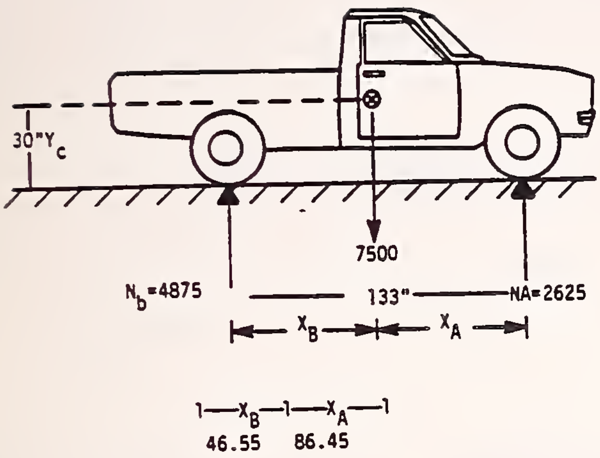
13. Clark, S.K., "Rolling Resistance of Light Truck Tires," U.S. Department of Transportation, Transportation Systems Center, Report No. DOT-TSC-NHTSA-80-30/DOT-HS-805 665, January, 1981.

14. Taylor, T. and Clark, S.K., "Effect of Automobile Tire Pressure Changes on Fuel Economy," Material on file at U.S. Department of Transportation, Transportation Systems Center, March 1980.

15. U.S. Environmental Protection Agency, "OMSAPC Advisory Circular," A/C No. 55B, December 1978.

APPENDIX A

GRADEABILITY CALCULATIONS



The calculation for the fully loaded 5.7ℓ engine vehicle is given below.

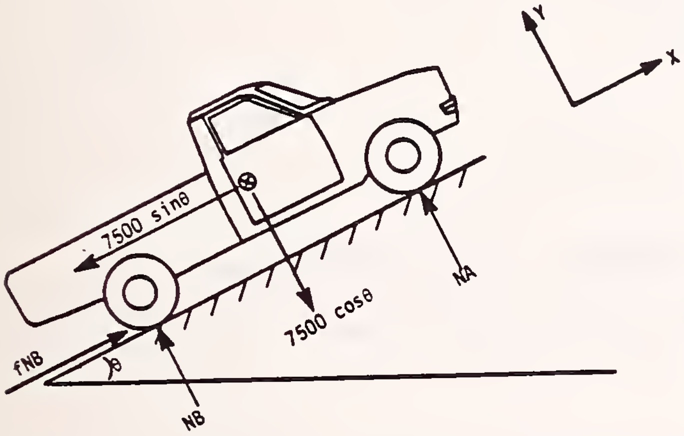
$f = 0.8$, wheelbase = 133 in.

To find X_A and X_B:

$\Sigma MA = 7500 \times X_A - 4875\ (133)$

$X_A = 86.45$ in.

$X_B = 46.55$ in.



For a Rear Wheel Drive Vehicle

Y_c = height of center of gravity - <u>assume</u> = 30"

Σf and $\Sigma M=0$ at incipient wheel slip.

$\Sigma f_x = fN_B - 7500 \sin\Theta = 0 \quad \rightarrow \quad .8N_B = 7500 \sin\Theta$

$\Sigma f_y = N_B + N_A - 7500 \cos\Theta = 0 \quad \rightarrow \quad N_B = 9375 \sin\Theta$

$\Sigma M_A = -133 \times N_B + 7500 \cos\Theta \times 86.45 + 7500 \sin\Theta \times 30$

$-1,246,875 \sin\Theta + 648,375 \cos\Theta + 225,000 \sin\Theta = 0$

$648,375 \cos\Theta = 1,021,875 \sin\Theta$

$.6345 = \tan\Theta$

$32.4° = \Theta$

58.31% Gradeability

This result can be compared to that using a GM[3] formula shown below.

$$\text{Startup Gradeability} = (\text{Max \% Grad}) \times 0.9 = \frac{100\ TE}{GW} - \frac{RR}{10}$$

TE = tractive effort = C x T x R x M

C = 0.00104

T = Max torque = 279.9 ft-lb at 1600 RPM

R = RAR x Gear Ratio = 3.54 x 6.55 = 23.187

$$M = \text{tire rev. per mile} = 5280\ \frac{ft}{mile} \times \frac{1}{2\pi r}\ \frac{rev}{ft} = 743.66\ \frac{rev}{mile}$$

r = tire radius = 1.13 ft.

GW = total weight of loaded vehicle = 7500 lbs.

RR = Rolling resistance = 8 lbs./1000 lbs. load

TE = 0.00104 x 279.9 ft-lb x 23.187 x 743.66 rev/mile

TE = 5019.45 ft-lb rev/mile

$$\text{Max \% Grade} = \frac{100\ (5019.45\ \text{ft-lb-rev/mile})}{7500\ \text{lbs.}} - \frac{8\ \text{lbs}}{10} = 66.13$$

Starting Grade = (Max % Grad) x 0.9 = 59.51% grade.

Ford[4] also has an equation for calculating gradeability.

$$\text{Est. Max. Gradeability} = \frac{KxMxRxT}{GW} - 1.0$$

K = 0.1011

$$M = \text{tire rev. per mile} = \frac{5280 \text{ ft.}}{\text{mile}} \times \frac{1}{2\pi r} \frac{\text{rev}}{\text{ft}} = 743.66 \text{ rev/mile}$$

R = Max Gear Reduction = 6.55 x 3.54 = 23.187

T = 279.9 ft-lb @ 1600 RPM

GW = Wt. = 7500 lb.

$$\text{Max. Gradeability} = \frac{0.1011 \times 743.66 \text{ rev/mile} \times 23.187 \times 279.9 \text{ ft-lb}}{7500 \text{ lb}}$$

$$- 1.0 = 64.06$$

Starting Gradeability = 57.65 .

The results of the three calculations for the 5.7ℓ engine 7500 lb GVWR truck are illustrated below.

SOURCE	STARTUP GRADEABILITY
TSC	58.3
G.M.	59.5
Ford	57.7

From these results it can be seen that any of the above equations are a good approximation for calculating MAXIMUM and STARTUP gradeability. This result supports previous work[5] performed at TSC.

APPENDIX B
SIMULATED FUEL ECONOMY

B.1 VEHICLE SIMULATION RESULTS

Transmission Type	Gear Ratio Range	Composite Fuel Economy* (mpg)
A3	2.45	24.16
A3 (locked up)	2.45	26.04
A3	2.74	24.35
A4	3.66	25.10
A4 (locked up)	3.66	28.08
A4	4.57	25.44
A4 (locked up)	4.57	28.95
A4	5.95	25.68
A4 (locked up)	5.95	29.44
M3	3.0	26.40
M4	4.11	27.51
M5	4.11	26.82

B.2 VEHICLE PARAMETERS

Weight: 3500 lbs

Aerodynamic power loss @ 50 mph: 11.1 hp

Rolling resistance coefficient: 0.10

Engine: 250 CID L-6 (scaled to 200 and 300 CID)

*Optimum fuel economy at 0-60 mph in 15 sec.

APPENDIX C
FUEL ECONOMY SENSITIVITY TO TIRE INFLATION PRESSURE

Vehicle	WT	C_p*	Inflation Pressure (psi gauge)	C_1	$\Delta\%FE/\Delta\%p$
1. Passenger Cars	2000	0.50	27 → 35	10 → 8.9	0.04
	3000	0.50	27 → 32	10 → 9.2	0.04
	3500	0.50	24 → 30	10.6→9.5	0.04
	4000	0.50	27 → 30	10 → 9.5	0.05
2. Light Trucks	5500	0.27	45 → 60	9.8 → 9.0	0.03
	6400	0.35	43 → 75	10 → 8.0	0.03
	7500	0.32	48 → 63	10 → 9.0	0.03

*See Equation 8.1 in Section 8. For passenger cars, a representative value of 0.50 has been assumed.

185 copies

U.S. Department
of Transportation

search and
ecial Programs
lministration

idall Square
nbridge, Massachusetts 02142

00347991

Official Business
Penalty for Private Use $300

Postage and Fees Paid
Research and Special
Programs Administration
DOT 513

U.S. MAIL