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VEHICLE ACCELERATION CHARACTERISTICS INFLUENCING HIGHWAY DESIGN

INTERIM REPORT

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16. Abstract The acceleration characteristics of current vehicles (as reported in the literature) are compared with those acceleration characteristics currently used in the geometric design of highways. The specific topics addressed are (1) climbing lane criteria; heavy trucks and recreational vehicles, (2) accelerating time versus distance traveled during acceleration; passenger cars and heavy trucks, (3) length of acceleration lanes; passenger cars, and (4) passing on two-lane highways; passenger cars.

The preliminary findings of this study indicate that the highway design policy as drafted by AASHTO [1] would be improved if the acceleration versus velocity characteristics of the design vehicles were included in the design policy. Nevertheless, the proposed requirements for climbing lanes are found to be conservative with respect to the acceleration capabilities of the current heavy truck fleet. With regard to recreational vehicles the design policy is based on a vehicle of "average" acceleration capability rather than a low capability vehicle. Policy matters, related to the normal acceleration characteristics of passenger cars as operated on the highway, are shown to be in conflict with various sources of information available in the literature and, therefore, appear to be subjects requiring further investigation.

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1.0 INTRODUCTION

This report provides a summary of preliminary findings concerning the interrelationships between the acceleration characteristics of highway vehicles and highway design policies. The report has been prepared by the University of Michigan Transportation Research Institute (UMTRI) as a step towards presenting a discussion on acceleration and deceleration characteristics to be included in a section or appendix of the final report for Project 15-8 conducted under the auspices of the National Cooperative Highway Research Program (NCHRP).

The main thrust of Project 15-8 addresses stopping sight distance; and, in that regard, ongoing work is examining the influences of vehicle deceleration characteristics on braking distance. The work on braking distance will provide the foundation for discussing deceleration characteristics at a later time.

The discussion of acceleration characteristics may be conveniently separated from a discussion of deceleration characteristics, because acceleration and deceleration are achieved using entirely different mechanical devices, namely the engine and the brakes. Nevertheless, both of these devices are controlled by drivers, thereby adding elements of driver skill and "taste" to the performance of the driver/vehicle system--the system of interest to the highway engineer. In some situations, the performance of the driver-vehicle system is limited primarily by vehicle and highway characteristics, for example, climbing a steep grade with a heavy truck. In contrast, driver comfort (or taste) determines driver/vehicle system performance in acceleration maneuvers that do not challenge the performance limits imposed by the mechanical properties of vehicle components or the friction available at the tire-road interface. Furthermore, traffic conditions influence how a particular vehicle is operated. Within this discussion of the acceleration subject, an attempt is made to distinguish situations in which driver and/or vehicle factors contribute to the findings presented.

Part of the material presented pertains to vehicle or component performance. This performance can be predicted more accurately than the performance of the driver/vehicle system. Generally representative observations of the performance of the driver-vehicle system may be difficult to make and they frequently involve uncertainties that require statistical evaluation. The approach taken here has been to apply (1) principles of physics, and (2) data from component measurements to predict vehicle performance. Where possible, these predictions are compared (and sometimes "calibrated") using the results of full scale vehicle tests or observations of vehicles in use.

The subjects discussed with respect to acceleration characteristics are:

(1) the basic physical factors influencing acceleration performance;

(2) those aspects of the proposed AASHTO (American Association of State Highway and Transportation Officials) geometric design policy [1] involving acceleration (propulsion);

(3) comparisons between the numerical values used in the design policy and numerical values based on the acceleration characteristics of the current vehicle fleet, and

(4) suggestions or insights to be considered in developing new design charts.

2.0 ACCELERATION

2.1 Basic Factors Influencing Acceleration Performance

The acceleration performance of pneumatic-tired vehicles depends upon the difference between the power available from the engine and the power required to overcome resistance to motion. So called, "natural" sources of retardation include rolling resistance in the tires, aerodynamic drag, and rolling resistance or inefficiency in the driveline (i.e., chassis friction). In addition to overcoming natural retardation, the engine supplies the power needed to increase velocity and/or climb hills, that is, the power needed to increase the the kinetic and/or potential energy of the vehicle.

The engines employed in highway vehicles may be roughly considered as nearly constant torque devices when operated at typical ranges of engine speed. As illustrated in the following simplified development, an interpretation of the implications of constant torque (or, an upper bound on power) is fundamental to understanding the relationship between vehicle speed and acceleration capability.

To first approximation engine power is the product of propulsive force and speed, i.e.,

$$P_e = F_p V \tag{1}$$

where P_e is power

F_p is propulsive force

V is forward velocity

Second, force equals mass times acceleration,

$$F = MA \tag{2}$$

where F is force

M is mass

A is acceleration

At the beginning of this development, ignore any natural retardation and grade influences such that $F = F_p$. (Natural retardation and grade influences will be considered later, after basic notions concerning power to weight ratio have been presented.) By combining equations (1) and (2) with $F_p = F$, the following result is obtained:

$$A = \frac{1}{V} \left(\frac{P}{M}\right) \tag{3}$$

As illustrated in Figure 1, equation 3 shows that the maximum upper bound on acceleration falls off (decreases) in a manner that is inversely proportional to the forward speed of a vehicle of a specified mass equipped with an engine with a given power capability.

For a particular vehicle, the power-to-mass ratio "scales" the acceleration-to-velocity relationship (see Figure 1). This power-to-mass scale factor (often referred to as a horsepower-to-weight ratio) provides a first order indication of the relative acceleration capabilities of various highway vehicles.

Clearly, the actual acceleration performance of a vehicle depends upon its natural retardation. The net propulsive force, F_p, is opposed by rolling resistance, F_r,

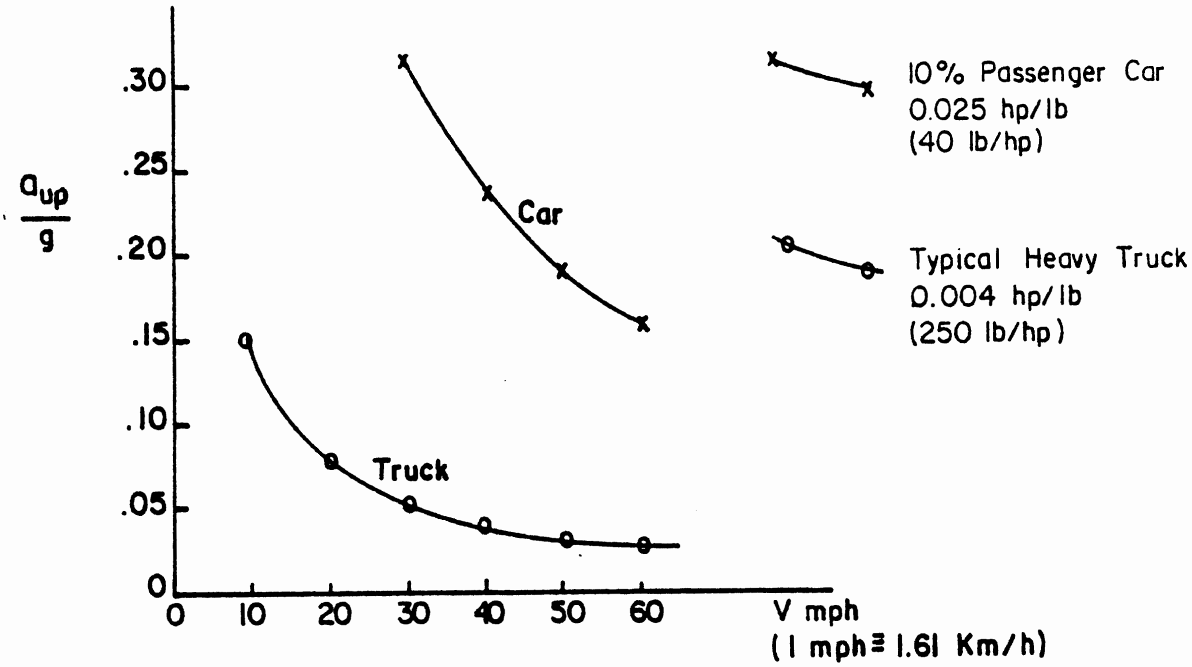


Figure 1. The influence of velocity on acceleration as determined by power-to-weight ratio.

and aerodynamic drag, F_a, in straightline motion on a level roadway (see Figure 2.).

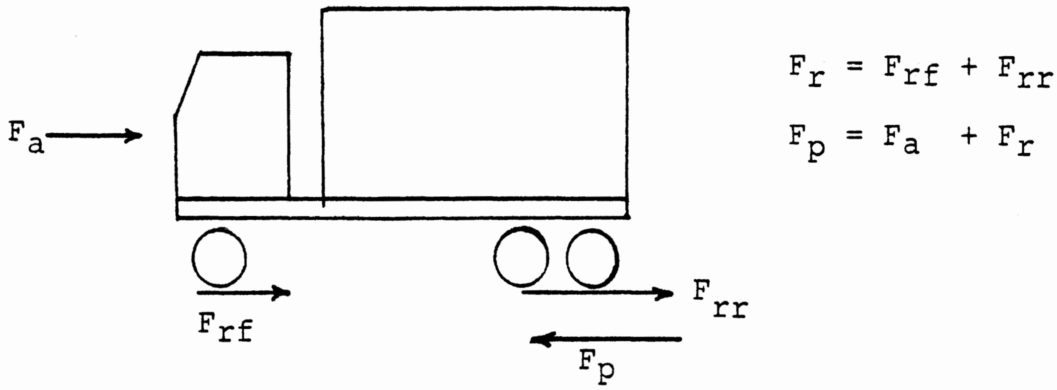


Figure 2. Force balance for sustained speed.

For zero acceleration (the condition for sustaining speed), the net force on the vehicle is zero, that is,

$$F_p = F_a + F_r \tag{4}$$

Figure 3 presents curves showing how F_p, F_a, and F_r vary with velocity. The point at which the total drag ($F_d = F_a + F_r$) equals the net propulsive force, F_p, determines the maximum sustained speed, V_s (see Figure 3).

In Figure 3 natural retardation has been broken down into 3 components:

(1) F_{rt}, a tire rolling-resistance force that depends upon vertical load but is independent of velocity

(2) F_{rc}, a chassis friction term that is conventionally represented as a linear function of velocity, and

(3) F_a, an aerodynamic drag force that depends upon the velocity squared.

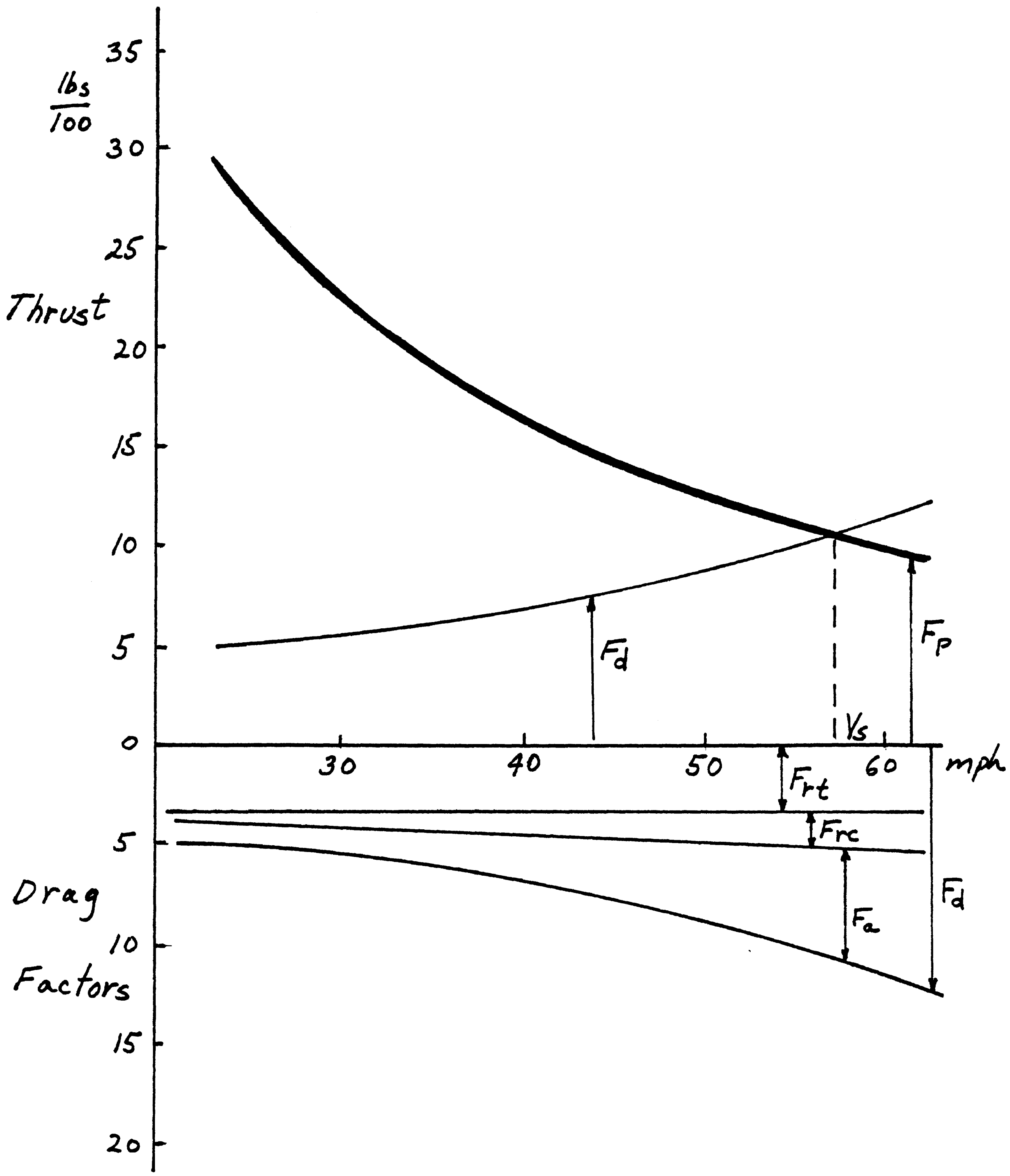


Figure 3. Influence of velocity on force components.

In addition to these components, upgrades produce another drag force that depends upon vehicle weight but is independent of velocity. Obviously, the maximum sustained speed on an upgrade is less than that on a level road; however, the amount of speed reduction caused by an upgrade varies (to first order) in a manner that is inversely proportional to the levels of velocity involved because of the relationship between propulsive force and velocity.

When the forces are balanced a vehicle sustains speed, neither accelerating or decelerating, however, when the forces are unbalanced, the vehicle will accelerate by an amount depending upon the inertias involved. There are two types of inertia to consider: (1) the mass of the vehicle, and (2) the rotational inertia of the drive system. In acceleration analyses, these two types of inertia are often combined into a single effective mass, m_e (or an equivalent weight, W_e). At highway speeds the contribution of the rotational inertia to m_e may be approximately 3 to 4% of the total; while at low speed the effective mass may be on the order of 1.5 times the actual mass. Due to the high gear ratios associated with low gears (that is, high effective mass), the low speed acceleration capability of a vehicle may be much less than that implied by the power-to-mass ratio.

For the purposes of the highway engineer, acceleration performance is often described through graphs or tables of velocity versus distance. The velocity versus distance performance of a vehicle can be derived from acceleration versus velocity information as follows:

(a) Use Newton's laws of motion to find acceleration; viz.,

$$A = \frac{dV}{dt} = \frac{F}{m_e} \tag{5}$$

where A is acceleration (or deceleration if $A<0$)

V is velocity

F is the net force which is a function of velocity

m_e is the effective mass

and d/dt represents the time rate of change of a variable

(b) Solve (5) to obtain velocity as a function of time. (This can be done theoretically, if F is a simple function of velocity, or it can be done numerically.)

(c) Since

$$V = \frac{d}{dt}\ (d) \tag{6}$$

that is, velocity is the time rate of change of distance, d, integrate (6) to obtain distance as a function of time.

(d) Using time to find corresponding pairs of speed-distance points, construct curves of velocity versus distance.

In summary, once the acceleration versus velocity characteristic of a vehicle has been determined, knowledge of elementary calculus can be used to obtain velocity and/or distance information.

Although the acceleration capability of a vehicle changes with velocity, first order estimates of acceleration performance for small speed changes are often made using a "constant" acceleration analysis. In this type of analysis an average acceleration is used to approximate the portion of the acceleration function existing between two speeds. The equations resulting from this type of analysis are readily derived from simple integrations with respect to time. The results are:

(1) with respect to elapsed time, T,

$$T = \left(\frac{V_f - V_i}{A}\right) \tag{7}$$

where V_f is the final velocity

V_i is the initial velocity

A is a constant level of acceleration (or deceleration)

(2) with respect to total distance, d

$$d = V_i\ T + 1/2\ A\ T^2 \tag{8}$$

$$\text{or } d = \frac{V_f^2 - V_i^2}{2A} \tag{9}$$

(Equation 9 can be interpreted and derived from a work/ energy balance, i.e.,

$\text{Work} = Fd = MAd = \frac{mV_f^2}{2} - \frac{mV_i^2}{2}$ = change in kinetic energy.)

In closing this general discussion two observations aid in providing a perspective with regard to (a) driver controlled acceleration performance and (b) braking performance. First, clearly the constant acceleration analysis applies to situations in which the driver chooses to use something less than the acceleration capability of the vehicle. Provided information on "normal" acceleration is available, the performance of the driver-vehicle system can be analyzed per equations (7), (8), and (9). In particular, the acceleration used from a standing start is usually chosen by the driver. In this case, the relationship between distance traveled and time elapsed is given by the following version of (8):

$$d = 1/2\ AT^2$$

where A is the driver's acceleration characteristic for the type of vehicle involved.

Second, the foundation brake used in motor vehicles is, to first order, a constant torque (or brake force) device when a brake line pressure is applied. Hence, a constant acceleration analysis is often used in estimating braking performance. A version of (9), commonly employed in estimating stopping distance, is as follows:

$$d = \frac{V_i^2}{2D} \tag{10}$$

where d is the stopping distance

V_i is the initial velocity

D is deceleration (D = -A)

2.2 Those Aspects of Geometric Design Policy Influenced by Acceleration Characteristics

The 1981 AASHTO Policy draft [1] has been reviewed to identify road designer needs for acceleration and deceleration data. (This work was performed by Prof. D.E. Cleveland of the Civil Engineering Department at the University of Michigan [2].) The "standard" applications, in which acceleration characteristics are used, include (a) enhancing the uniformity of vehicle operating speeds on grades, (b) determining the length of acceleration lanes for entrance terminals, (c) providing adequate sight distance for accelerating across intersections, and (d) providing adequate sight distance for passing on two-lane highways.

Different types of acceleration data are employed in these applications. For studying vehicle-operating characteristics on grades, design curves relating (a) speed, (b) magnitude of grade, and (c) length of grade are presented for recreational vehicles and trucks (see Figures III-26, III-27A and B, III-30, and III-31 from reference [1]). (The results for significant upgrades apply to situations in which the vehicles are actually decelerating because of a lack of power. Nevertheless, we have chosen to include these cases under the heading of acceleration since the driver is using the engine in an attempt to increase or, at least, maintain speed.) The acceleration characteristics for passenger cars are not required in this application since cars are believed to have enough power to readily negotiate grades as steep as 7 or 8 percent (see [1] page III-96).

With regard to the length of acceleration lanes, acceleration characteristics for passenger cars are used [2] (see Figure II-13 page II-17 of reference [1]). These characteristics are intended to represent the normal acceleration performance for a low horsepower passenger car. In practice this "normal" acceleration characteristic is more related to driver preferences than to vehicle capability. Even so, in certain situations, for example, at tight interchanges with grade separation, heavy trucks may not be able to accelerate to within 5 mph (8 km/h) of typical running speeds in a distance determined by normal passenger car acceleration levels.

In the case of accelerating across an intersection, the design policy provides information on three design vehicles (Figure IX-15 page IX-48), a passenger car, a straight truck, and a tractor-semitrailer vehicle. The information is given in the form of curves of accelerating time versus distance traveled for each type of vehicle. Unpublished data have been used to determine the assumed relationships for straight trucks and tractor-semitrailers [1].

Finally with regard to passing maneuvers, average acceleration levels are given for various speed ranges. (See Table III-4 page III-15 [1].) These values of acceleration are based on observations of traffic and, apparently, they represent passenger car performance.

2.3 Characteristics of the Current Vehicle Fleet Applicable to the Design Policy

The characteristics of the vehicle fleet are continually changing. An emphasis on fuel economy has brought about lighter and less powerful passenger cars. In heavy trucks, the trend has been towards heavier vehicles with more powerful engines. Vehicles now have more efficient aerodynamic shapes, tires with less rolling resistance, and more efficient engines and drivelines than they had 5 to 10 years ago. For trucks these changes in retardation are approximately equivalent to a 1% change in grade [3]. That is, for heavy trucks, 3% downgrades are now effectively 4%, and 3% upgrades are now effectively 2% - clearly an acceleration advantage and, as it has turned out, a braking problem. The purpose of this section is to compare the acceleration characteristics of the current vehicle fleet with those used in the design policy.

2.3.1 Heavy Truck Acceleration; Climbing Lane Criteria. In this study emphasis has been placed on the acceleration characteristics of the heavy truck. Early on, an attempt was made to acquire relevant data from various manufacturers. Although cooperation was obtained, the information received did not represent a comprehensive assessment of the nation's truck-fleet. In order to develop a uniform method for assessing truck performance, a review was made of the methods available for predicting the acceleration performance of heavy trucks. That review is included in this report (see Appendix A). Based on the findings of the review, we have concluded that (1) suitable models of the acceleration performance of trucks are available, and (2) a large body of pertinent information on heavy trucks is contained in the 1977 Truck Inventory and Use (TIU) Survey conducted by the Department of Commerce [4].

The draft design policy presents a set of curves showing a decreasing trend in weight-to-horsepower ratio for the vehicle fleet from 1949 to 1973. Data from the TIU survey have been superimposed on the information presented in the design policy (see Figure 4). The "1977" curve in Figure 4 was obtained from computerized files of the TIU information gathered from over 96,000 trucks. The weight used in the 1977 data is the maximum weight carried during 1977 as reported by the vehicle operator. In this regard the weight-to-power estimates represent the average of the performance capability of the fleet in a heavily loaded condition (that is, the weight-to-power properties of vehicles when they are operating empty or partially loaded are not included here). Nevertheless, the 1977 curve falls well below the other curves, thereby continuing the trend towards lower weight-to-power ratio (that is, higher power-to-weight ratio and greater acceleration capability).

With respect to the "300 lb/hp" vehicle used in the design policy, the results from the TIU survey indicate that the average loaded truck in the 60 to 80 thousand pound weight class has an engine with an average horsepower of 282 with an estimated standard deviation of 51 hp. These figures correspond to an average weight-to-horsepower ratio of 248 lbs/hp, with a 303 lb/hp vehicle being one standard

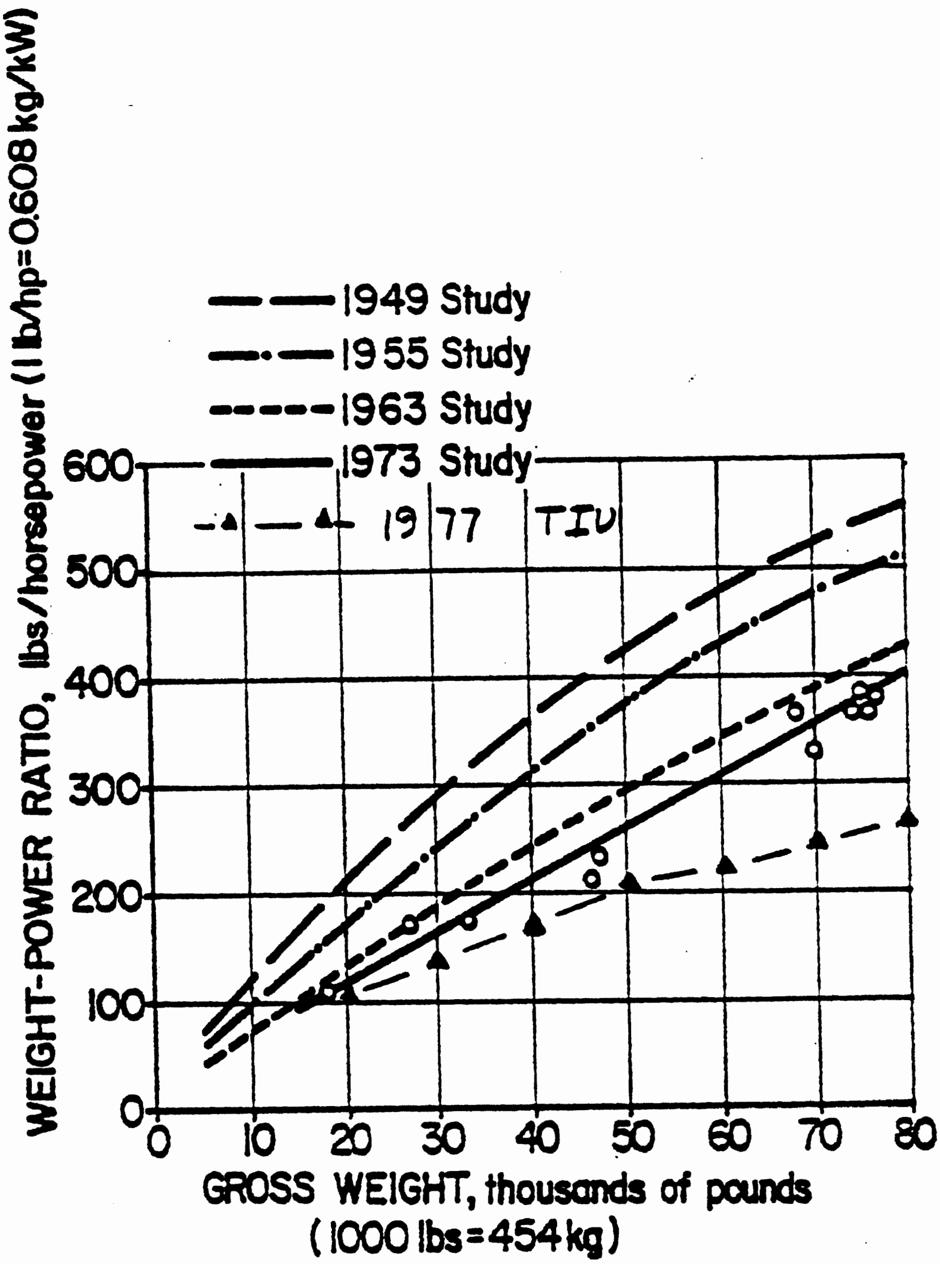


Figure 4. Trend in weight-power ratios from 1949 to 1977 [5].

deviation less powerful than average. Hence, according to the TIU data, a 300 lb/hp design vehicle might be characterized as "substantially below average" rather than as "typical" as qualitatively referred to in the draft design policy. Nonetheless, one could argue that a 300 lb/hp vehicle represents a reasonable vehicle to use in designing highways and establishing the need for climbing lanes.

The relative importance of the power-to-weight ratio (the inverse of the weight-to-horsepower ratio) may be understood by comparing the upper bound on propulsive thrust to the influences of driveline efficiency, rolling resistance, and aerodynamic drag on net thrust; for example, see Figure 5 representing a typical heavy truck similar to the one analyzed in [5] and subsequently used to develop Figure III-31 page 107 of the draft design policy. In this case, we have employed retardation (drag) factors that are derived from (a) our literature review and (b) contacts with manufacturers. Table 1 summarizes the equations, relationships, and coefficient values employed in this analysis.

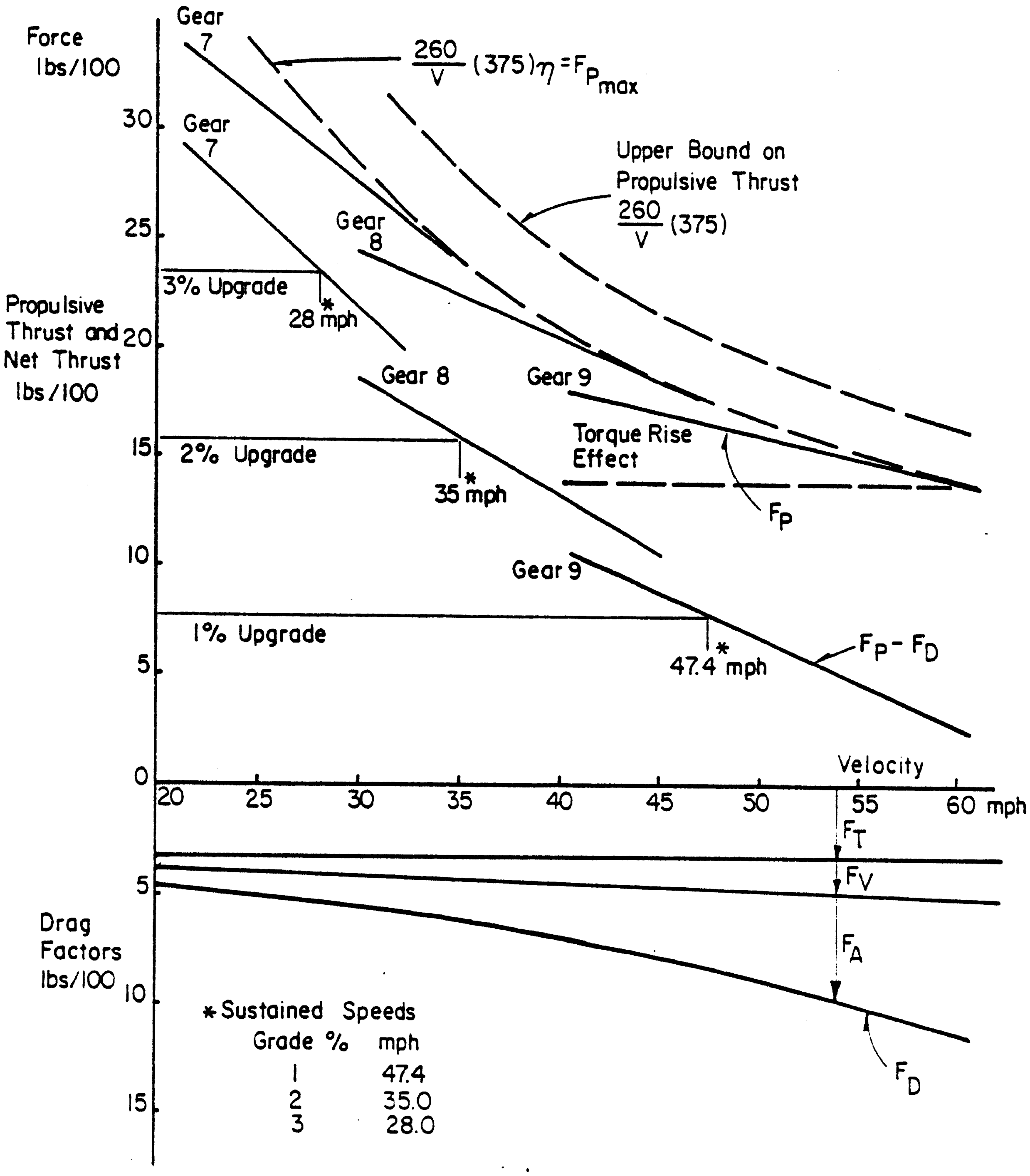


Figure 5. Force analysis of a design heavy truck operating in 7th, 8th, or 9th gear.

TABLE 1

ANALYSIS FACTORS FOR A 300 lb/hp TRUCK

V (forward velocity) = independent variable, mph

GVW (gross weight) = 78000 lbs

NHP (net engine horsepower) = 260 hp at 0 to 500'

(elevation correction factor, C_e = 1-4 (10^{-5}) E, where E = elevation e.g., C_e = 0.6 at 10,000 ft)

η (driveline efficiency) = 0.86 for tandem drive axles (dim'less)

(note: for a single drive axle η = 0.9)

F_r (rolling resistance, radial tires) = (GVW/1000) (4.1 + 0.041V) C_r

where

C_r is a factor defining the quality of the road surface Typical values of C_r are 1.0 for a smooth concrete road, 1.2 for worn concrete road or a cold black top road, and 1.5 for a hot black top road.

Notes: (1) For bias tires

$$F_r = (GVW/1000)\ (6.6 + 0.046V)\ C_r$$

(2) The source of the velocity dependent term in the rolling resistance equations may not be dependent upon tire properties but rather on friction in rotating parts.

F_a (aerodynamic drag) = $C_a(A)(0.0024)\ V^2\ C_p$

where

TABLE 1. (continued)

C_a is a drag factor depending upon vehicle shape. (Typical values of C_a are 0.9 for highway tractors without aerodynamic aids and 0.7 for tractors with aerodynamic shields.)

A (frontal area) = 102 ft^2

C_p (elevation factor) = 1.0 at sea level
0.86 at 5,000'
0.74 at 10,000'

GR (overall gear ratio, including a rear axle ratio of 4.11)

Gear Number	Ratio
1	48.62
2	32.47
3	23.80
4	17.76
5	13.15
6	10.15
7	7.74
8	5.54
9	4.11

Tire factor (rpm/mph) = 8.4 for a 10 x 20 truck tire (504 rev./mile)

Engine Power and Torque Characteristics
(see attached graphs from [5])

These data are characterized by a torque at 1400 rpm that is approximately 1.3 times the torque at 2100 rpm, the rated speed at which maximum power (260 hp) is delivered. In other words, the torque increases by approximately 30% going from rated speed down to the speed at which shifting is expected to occur.

$$W_e \text{ (total equivalent)} = W + g/R_t^2 \, (I_e \, G_r^2 + I_t)$$

where

I_e = 2.58 ft lb sec^2 for a typical engine

I_t = 170 ft lb sec^2 for 18 10x20 tires

g = 32.2 ft/sec^2

R_t = 1.667 ft for a 10x20 tire

W = gross vehicle weight

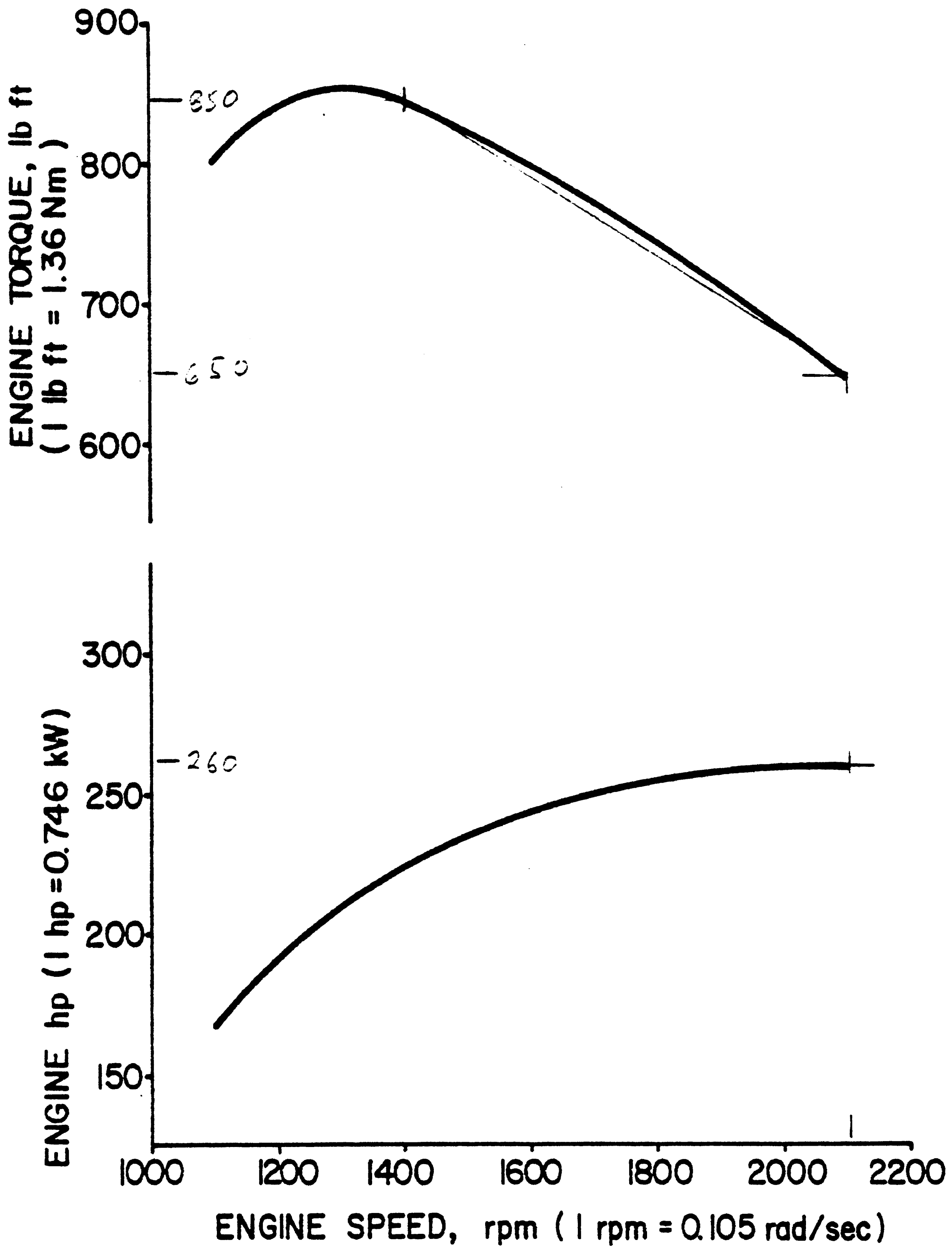


Figure 6. Engine torque and hp characteristics used in the truck simulation [5].

Although a driveline efficiency is used here in place of chassis friction, the thrust and drag forces are nearly equal to those employed in [5]. However, in computing acceleration (see Figure 7) an equivalent weight is employed to account for the inertia of rotating components, and a tire factor of 8.4 rpm/mph corresponding to a typical 10 x 20 tire, was selected. The analysis in [5] used 8.55 rpm/mph which corresponds to a smaller 9 x 20 tire. These seemingly small changes in tire factor (1.75%) and weight (3.5% in 9th gear) cause a significant change in the critical length of grade on slight upgrades (see Figure 8). On a 2% upgrade, for example, the critical length of grade is approximately 2400 feet for a vehicle with an 8.55 rpm/mph tire and a weight of 78,000 lb compared to approximately 2900' for a tire with 8.4 rpm/mph and an equivalent weight of 80,710 lbs.

On steeper grades (for example, 4 and 6%), the level road acceleration capability of the vehicle is a smaller fraction of the existing acceleration (deceleration). Hence, variations in vehicle parameters, such as the tire factor and equivalent weight, have less influence on acceleration performance on steep grades than they do on moderate grades.

This review of the climbing lane criteria, given in [5] and subsequently incorporated in [1], shows the criteria to be representative of a relatively low powered loaded-heavy-vehicle by 1977 standards (something like 84% of the

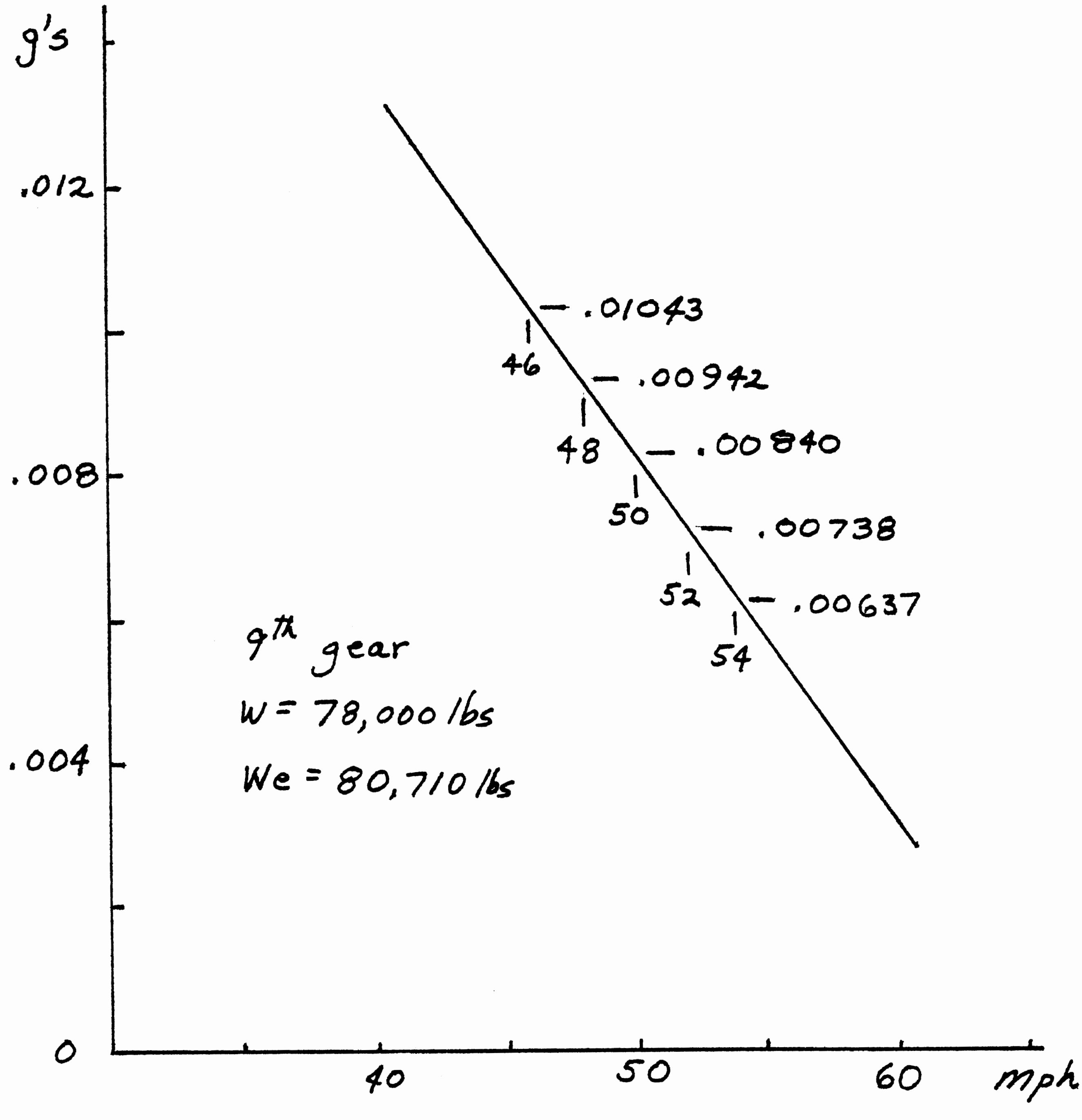


Figure 7. Level-road-acceleration versus velocity for a design heavy truck in 9th gear.

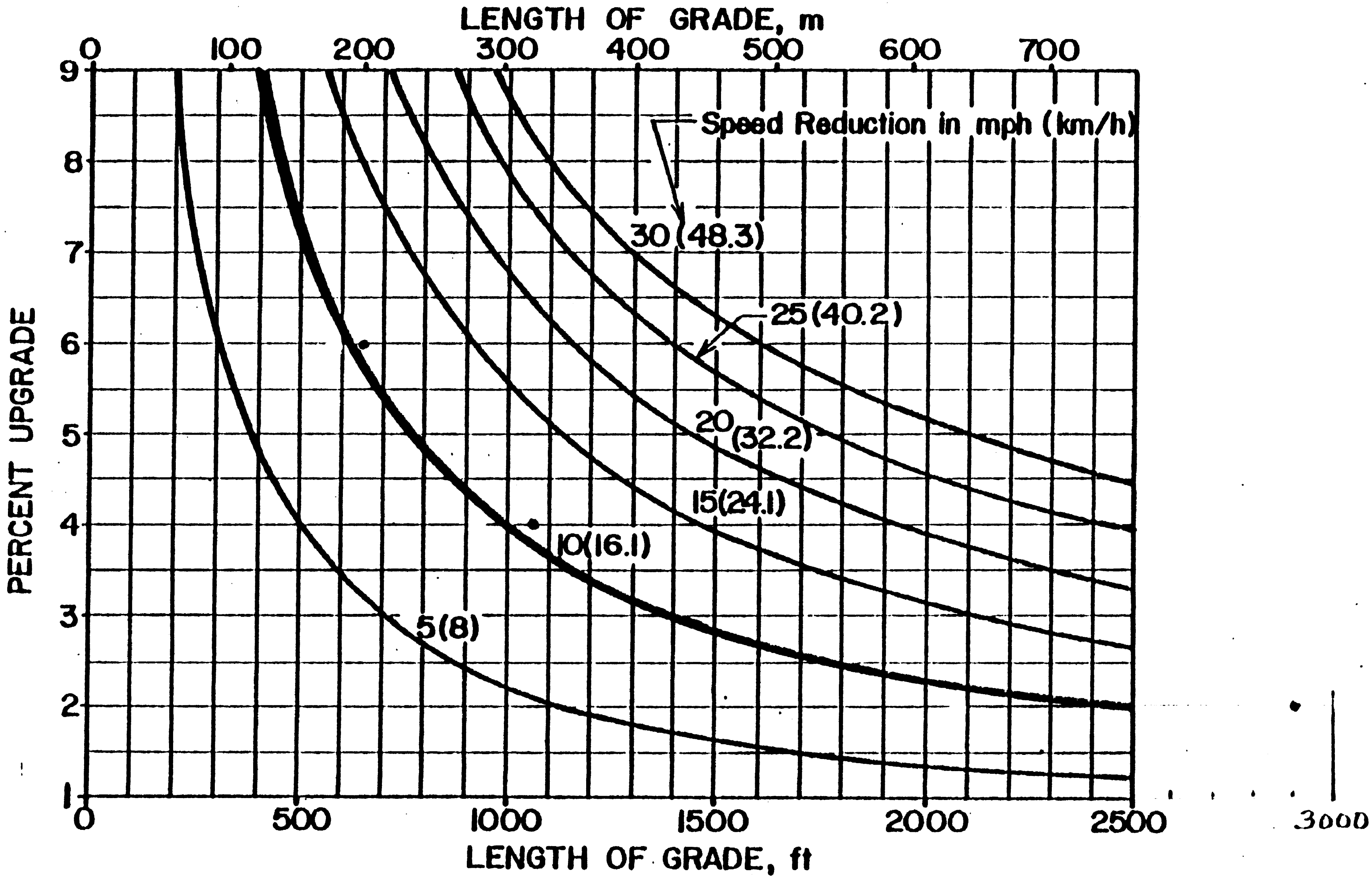


Figure 8. Critical lengths of grade for design, assumed heavy truck of 300 lb/hp (182.5 kg/kW), entering speed = 55 mph (88.5 km/h).

vehicles weighing between 60 and 80 thousand pounds had greater power to weight ratios). Even though the calculation procedure given in [5] does not include the effective mass of the vehicle, the results appear to be representative of heavy vehicle performance on steep upgrades in the speed range from 55 to approximately 30 mph. At low speeds and on mild upgrades the influence of effective mass should be included in the calculations.

During 1983, the Department of Commerce will conduct a Truck Inventory and Use Survey pertaining to vehicles operated in 1982. This 1982 data could be analyzed, using the procedures employed by manufacturers and the highway research community, to obtain an updated set of curves to be used in evaluating the need for climbing lanes.

2.3.2 <u>Heavy Truck Acceleration; Accelerating Time Versus Distance Traveled During Acceleration</u>. Data on acceleration from a stop is used in determining sight distance at intersections (see Figure 9 which includes Figure IX-15 page IX-48 of [1]). As indicated in the figure, two heavy vehicles, referred to as "WB-50" and "SU," and a passenger car, "P," have been assumed for design purposes.

The WB-50 design-vehicle is intended to represent a large tractor-semitrailer combination. Assuming that a heavy vehicle similar to the one used in the climbing lane application (see Section 2.2) is a suitable design vehicle of the WB-50 class, the acceleration performance of a

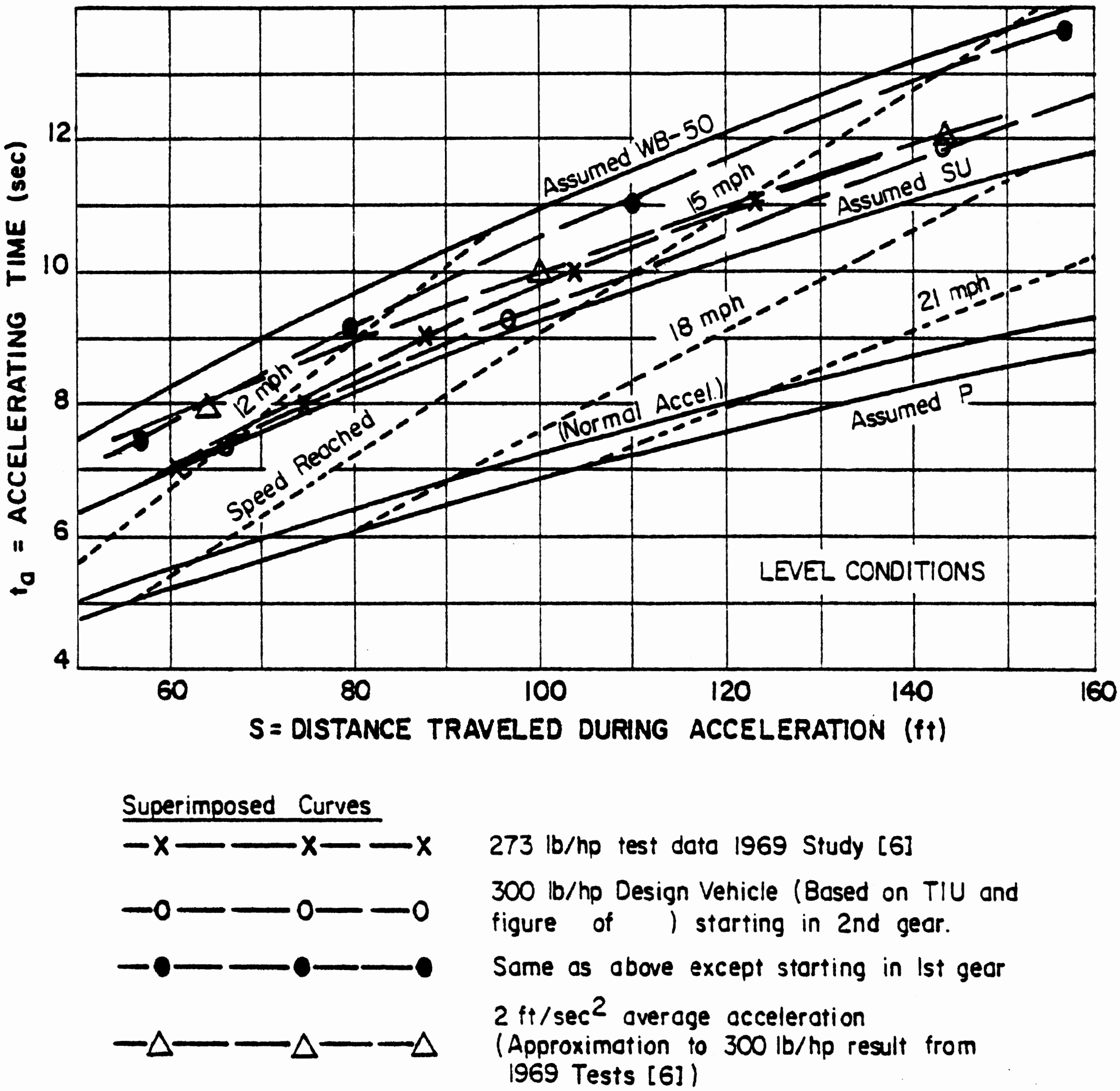


Figure 9. Information on acceleration from a stop.

300 lb/hp heavy truck can be used here to make a comparison with the data given Figure IX-15 of [1] (see Figure 9).

Note that in Figure 9, 4 curves are superimposed on the graphs presented in [1]. Two of these 4 curves represent the calculated performance of the 300 lb/hp vehicle described in Section 2.2. In one of these cases the vehicle is started in first gear and in the other car the vehicle is started in second gear. As shown, significantly better performance is obtained by starting in second gear (a fact that is well known to truck drivers).

Curves based on tests of heavy trucks [6] are also added to Figure 9. These curves correspond to (1) a 273 lb/hp truck and (2) an <u>average</u> acceleration level of 2 ft/sec^2 approximating a typical truck with 300 lb/hp, operating in 1969 [6]. These curves agree with the calculated results for the design vehicle when started in second year (the conventional gear selection for starting on the level).

The assumed WB-50 curve given in the design policy illustrates poorer performance than any of the 4 curves superimposed in Figure 9. In this sense, the WB-50 curve represents a conservative design policy, especially as long as the trend is towards vehicles with higher horsepower-to-weight ratios, less aerodynamic drag, and less rolling resistance.

The assumed SU curve represents a straight truck with a 20 foot wheelbase. A great variety of vehicles fit within this description. For example, a truck with a 12,000 lb front axle and a 34,000 lb tandem rear axle-set is a possible candidate for a design vehicle of this class. Example predictions of the acceleration performance of this type of vehicle fall near the SU curve given in [1]. However, in this case the assumed SU curve does not appear to be as conservative as the assumed WB-50 curve. Given current vehicle characteristics, certain fully loaded straight trucks, that satisfy the bridge formula, may require more time to accelerate across an intersection than that shown in Figure IX-15 of [1]. A possible method for resolving this situation would be to provide a more complete definition of the SU design vehicle.

2.3.3 Passenger Car Acceleration; Accelerating Time Versus Distance Traveled During Acceleration. In contrast to the situation with heavy trucks (as discussed in the previous section), passenger cars seldom accelerate at maximum performance so knowledge of the maximum performance capability of the vehicle is not as useful as it is for trucks. That is, an experienced driver uses the maximum performance of a truck while a prudent driver does not challenge the capabilities of the passenger car engine (unless he/she wishes to spin wheels) in accelerating to cross an intersection.

Possibly due to difficulties in determining "normal" acceleration, the results given in Figure II-13 of [1] differ from those given in Table 6.47 of [7]. In studies of ramps and speed change lanes [8], investigators have found the tables given in [7] to be more representative of vehicle performance on ramps than the information given in Figure II-13. Based on calculations of acceleration derived from the curve representing the "assumed P" vehicle in Figure IX-15, the average acceleration of the design passenger car is, approximately 2.86 mph/sec compared to a normal acceleration of 3.3 mph/sec given in the Transportation and Traffic Engineering Handbook [7] for speed changes from 0 to 30 mph. Or, as illustrated in Figure 10, the calculated accelerating-time-versus-distance curve (representing a normal acceleration of 3.3 mph/sec) indicates shorter acceleration times than those required by the "assumed P" vehicle. The AASHTO design policy is conservative in that acceleration levels corresponding to those normally chosen by passenger car drivers produce acceleration times that are considerably shorter than those given by AASHTO.

Hearne and Clark [9] have recently studied passenger car data reported by Consumer Reports, Motor Trend, and Car and Driver magazines for two acceleration maneuvers, specifically, (1) the time to accelerate from 45 to 65 mph, and (2) the time to accelerate from 0 to 60 mph. That study examined the trends in these measures of new vehicle performance over the period from 1971 to 1979. The resulting acceleration characteristics for the 1970's are compared with the acceleration performance criteria used in the "AASHO Blue Book" [10]. The following findings from [9]

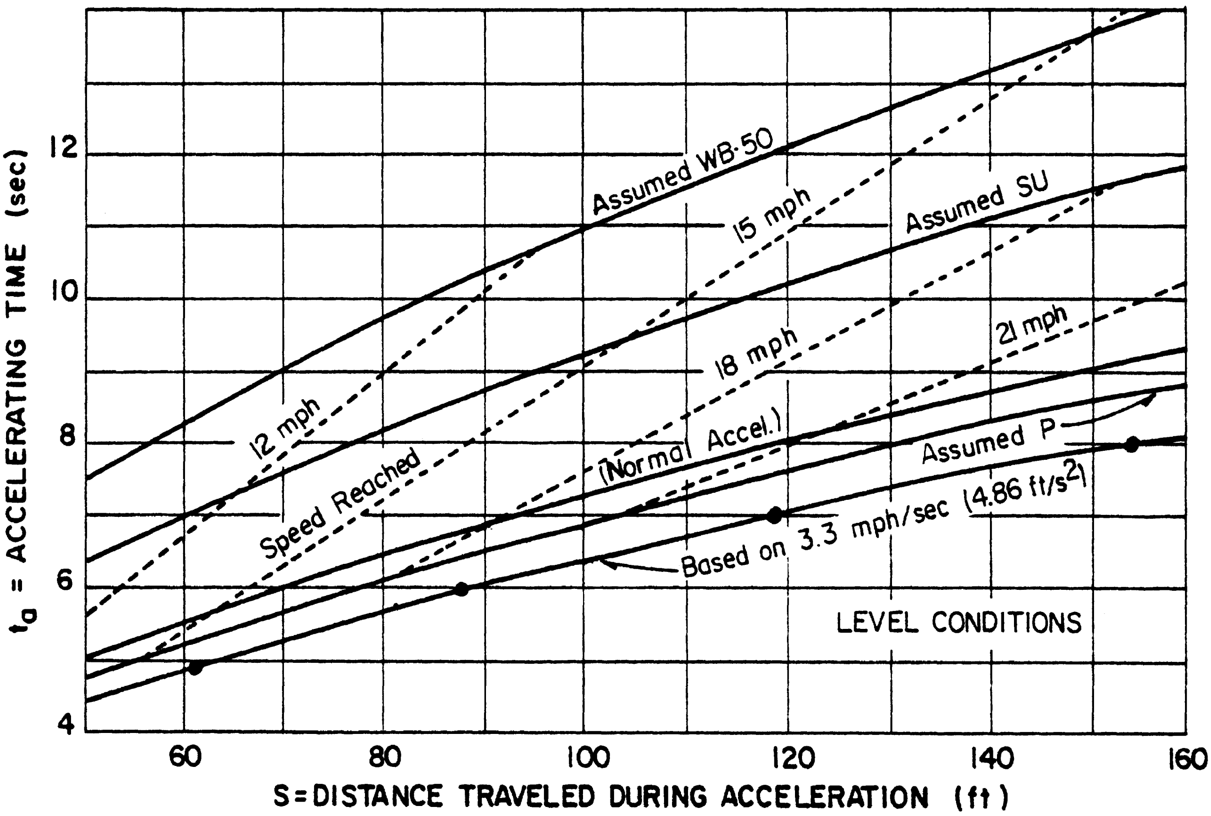


Figure 10. Comparison of AASHTO and ITE information on acceleration performance.

indicate that, even though passenger-vehicle acceleration-performance has been decreasing since approximately 1958, the acceleration performance of late model cars exceeds the criteria employed in the AASHO Blue Book. (The AASHO criteria are based on tests performed in 1937.) Between 1970 and 1980, the typical standard-sized car changed from approximately 4000 lbs with a 350 in^3 engine to approximately 3300 lbs with less than 250 in^3 of engine displacement [9]. The implications of these changes are illustrated in the times required to accelerate from 45 to 65 mph and 0 to 60 mph as shown in Tables 2 and 3 for model years 1971, 1973, 1975, 1977, and 1979. The times given in Tables 2 and 3 represent the car population for each model year since they are obtained by weighting the performance of each vehicle model in accordance with its annual sales volume. The average weighted acceleration time from 0 to 60 mph increased from 12.7 sec. in 1971 to 15.5 sec. in 1979 (see Table 2) indicating a decline in automotive performance during the 70's.

Clearly, overall acceleration performance has decreased during the decade of the 70's.

Nevertheless, the performance of 1979 and 1981 vehicles exceeds the AASHO criteria based on studies performed in 1937 (see Figures 11 and 12). Assuming that (1) normal acceleration performance is primarily determined by driver "taste" rather than by vehicle characteristics, and (2) drivers continue to prefer the same normal acceleration

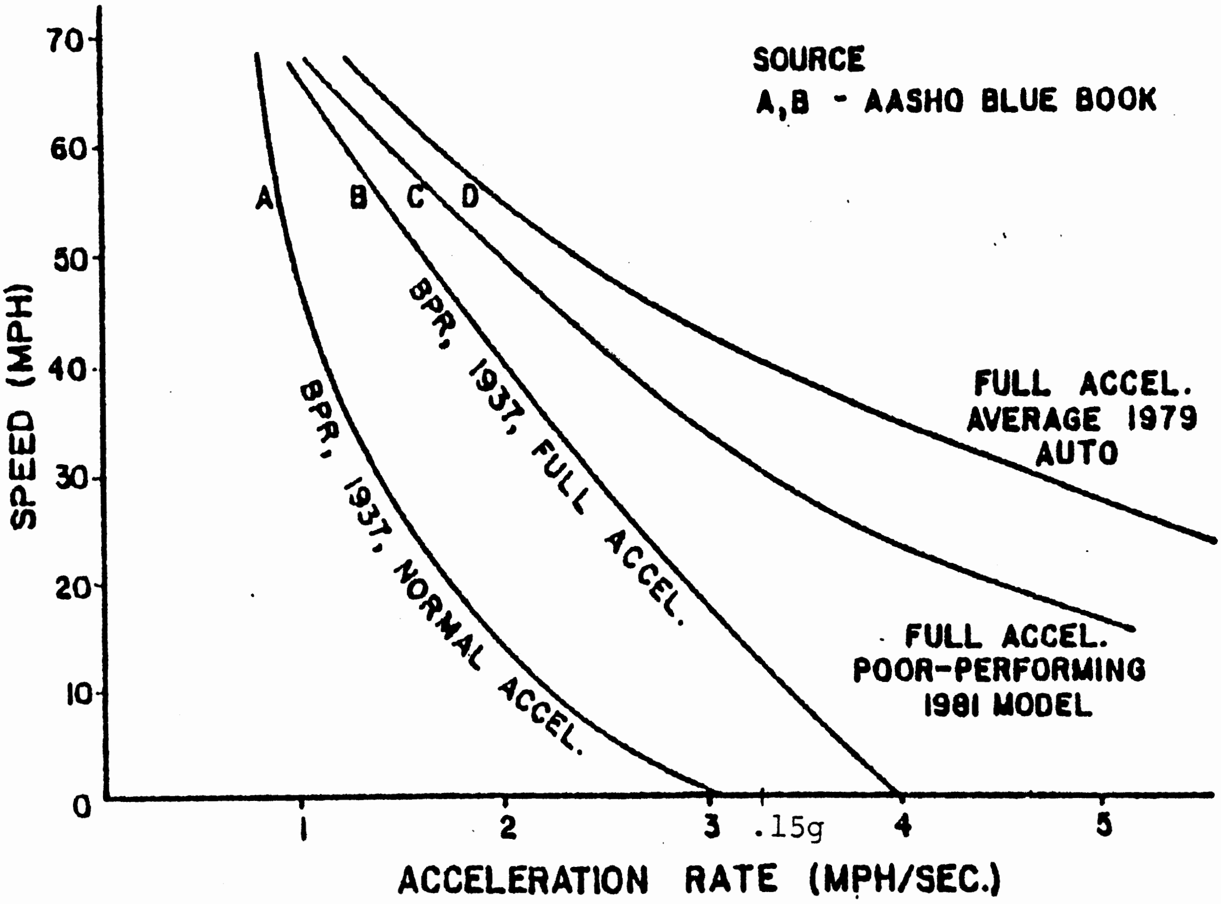


Figure 11. Normal and full acceleration rates for a range of speeds [9].

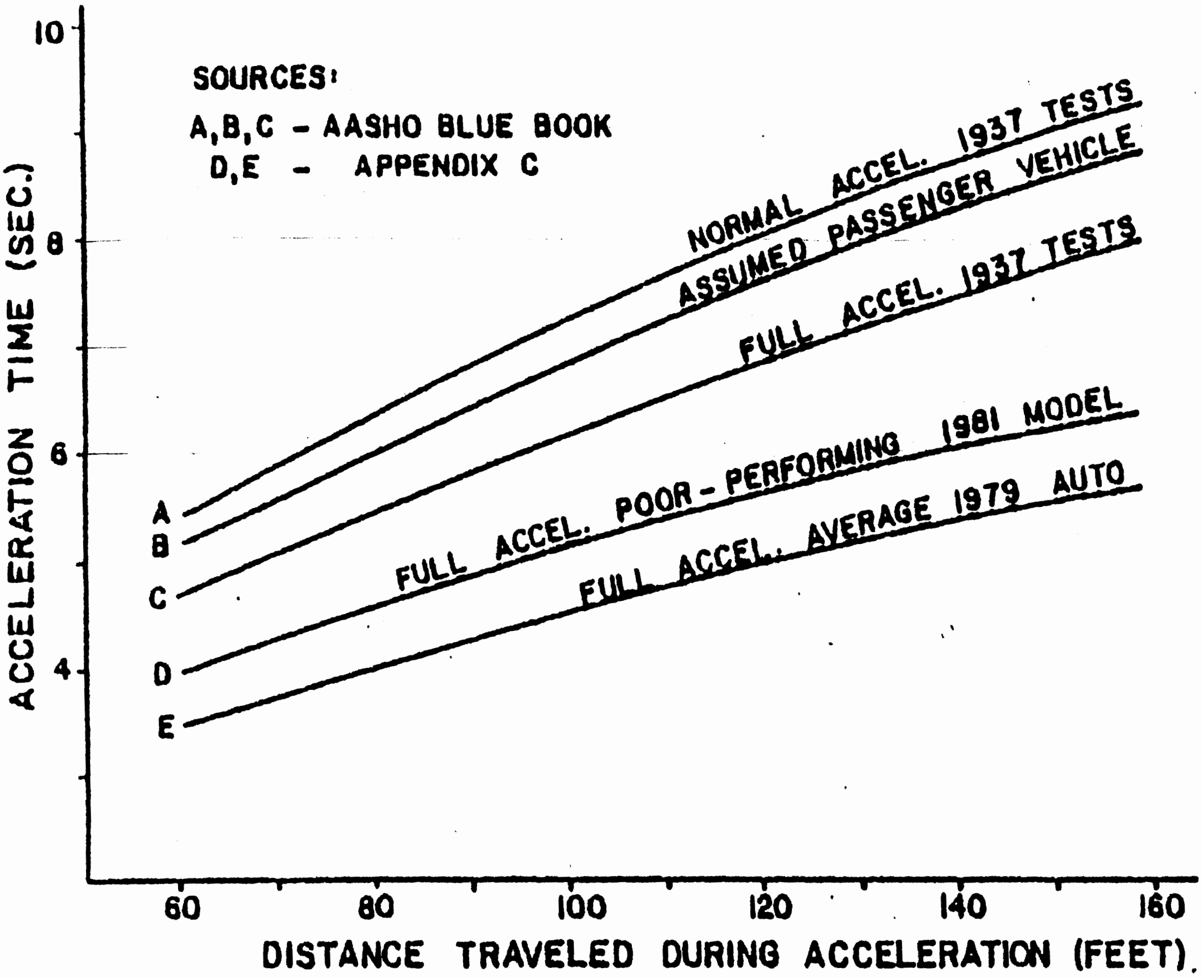


Figure 12. Relationship between acceleration time and distance traveled during normal and full acceleration [9].

TABLE 2

TIME REQUIRED FOR ACCELERATION FROM 0 to 60 MILES PER HOUR FOR SELECTED MODEL YEARS [9]

Year	Time (sec.)
1971	12.7
1973	14.1
1975	14.8
1977	15.5
1979	15.5

TABLE 3

TIME REQUIRED FOR ACCELERATION FROM 45 to 65 MILES PER HOUR [9]

Year	Time (sec.)	Average Accel. Rate (fps)
1971	8.4	3.49
1973	9.4	3.12
1975	9.4	3.12
1977	10.1	2.90
1979	9.9	2.96

performance as they did in 1937, then the difference between normal acceleration (1937) and the full acceleration capability of a vehicle may be interpreted as a margin of safety for situations in which the driver finds a need for accelerations that are greater than he/she would prefer to use under normal circumstances. In [9] it is found that the average weighted performance of 1979 models and a poorly performing 1981 model exceed (by a wide margin) the normal acceleration performance determined in 1937 (see Figures 11 and 12).

2.3.4 Passenger Car Acceleration; Length of Acceleration Lanes. As stated in [2], "The length of acceleration ramp required for an entrance is governed by the difference between the running speed of the last ramp curve or other constraint and that of the freeway. The policy is that the length provided should be sufficient for the motorist to reach a speed five miles per hour less than the average running speed of the freeway by the time the merge into the through lane is completed."

The acceleration characteristics of passenger cars are used to determine the ramp length required. As illustrated in Table 4 (Table X-5 from [1]), acceleration performance from low to high initial speeds, V_2', and from low to high entrance speeds, V_2, is considered in this application. In other words, knowledge of the full range of acceleration characteristics is needed.

The performance information presented by AASHTO has been summarized in graphs of speed reached versus distance traveled for initial speeds ranging from 0 to 50 mph in increments of 5 mph (see Figure 13). Although Figure 13 is a convenient form in which to display results, the curves

TABLE 4

MINIMUM ACCELERATION LENGTHS FOR ENTRANCE TERMINALS
WITH FLAT GRADES OF 2 PERCENT OR LESS

Highway		L = Acceleration Length (ft)								
		For Entrance Curve Design Speed (mph)								
Design Speed (mph)	Speed Reached (V_2) (mph)	Stop Condition	15	20	25	30	35	40	45	50
		And Initial Speed (V_2') (mph)								
		0	14	18	22	26	30	36	40	44
30	23	190	--	--	--	--	--	--	--	--
40	31	380	320	250	220	140	--	--	--	--
50	39	760	700	630	580	500	380	160	--	--
60	47	1,170	1,120	1,070	1,000	910	800	590	400	170
70	53	1,590	1,540	1,500	1,410	1,330	1,230	1,010	830	580

provided by AASHTO are based on relatively low levels of acceleration such as those given by the "BPR, 1937, normal accel." curve included in Figure 11. More recent information on normal acceleration is given in Table 5 as presented in reference [7].

For comparison purposes, the AASHTO data [1] and the ITE information [7] are presented on the same graph with full acceleration curves for (1) a "40 watt/kg" car representing a poorly performing vehicle for any year from 1967 to 1995 [12], (2) an average 1979 auto [9], and (3) a poor performing 1981 car [9] (see Figure 14). In addition Figure 14 contains a curve representing a so called "design" car. Examination of Figure 14 shows that the ITE information for normal acceleration, based on a 1971 study, falls between the full acceleration curves for the average 1979 auto and the "poor-performing" '81 model for accelerations less than 0.15g. Compared to the reported performance capabilities of current vehicles, the normal acceleration curve based on ITE information [7] appears to be unreasonably high near the limit of vehicle performance. Possibly, the decline in performance capability during the period from 1960 to 1980 accounts for the situation in which maximum performance in 1979 is close to the normal acceleration performance determined in a study reported in 1971.

Transportation Research Record 772 [12], published in 1980, contains projections of the make up of the passenger

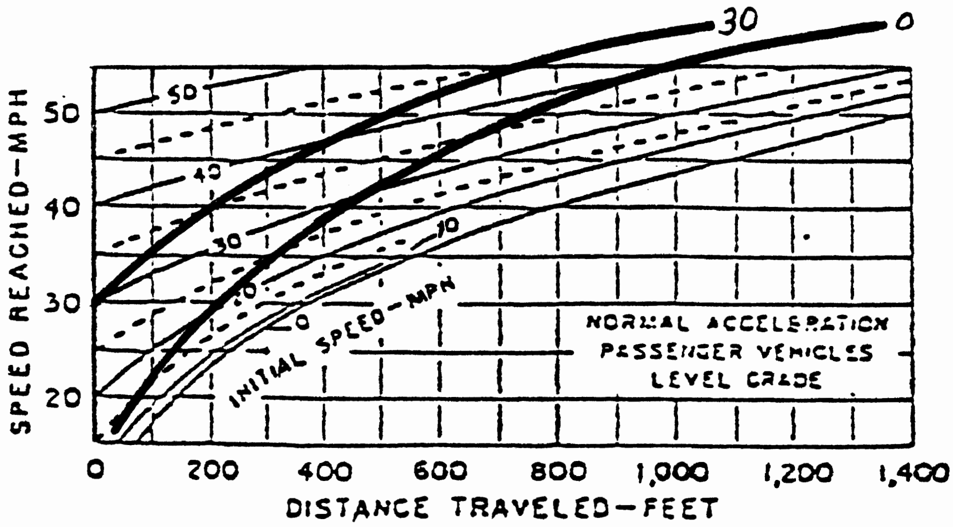


Figure 13. Normal acceleration from [1] plus superimposed "design" curves at initial velocities of 0 and 30 mph.

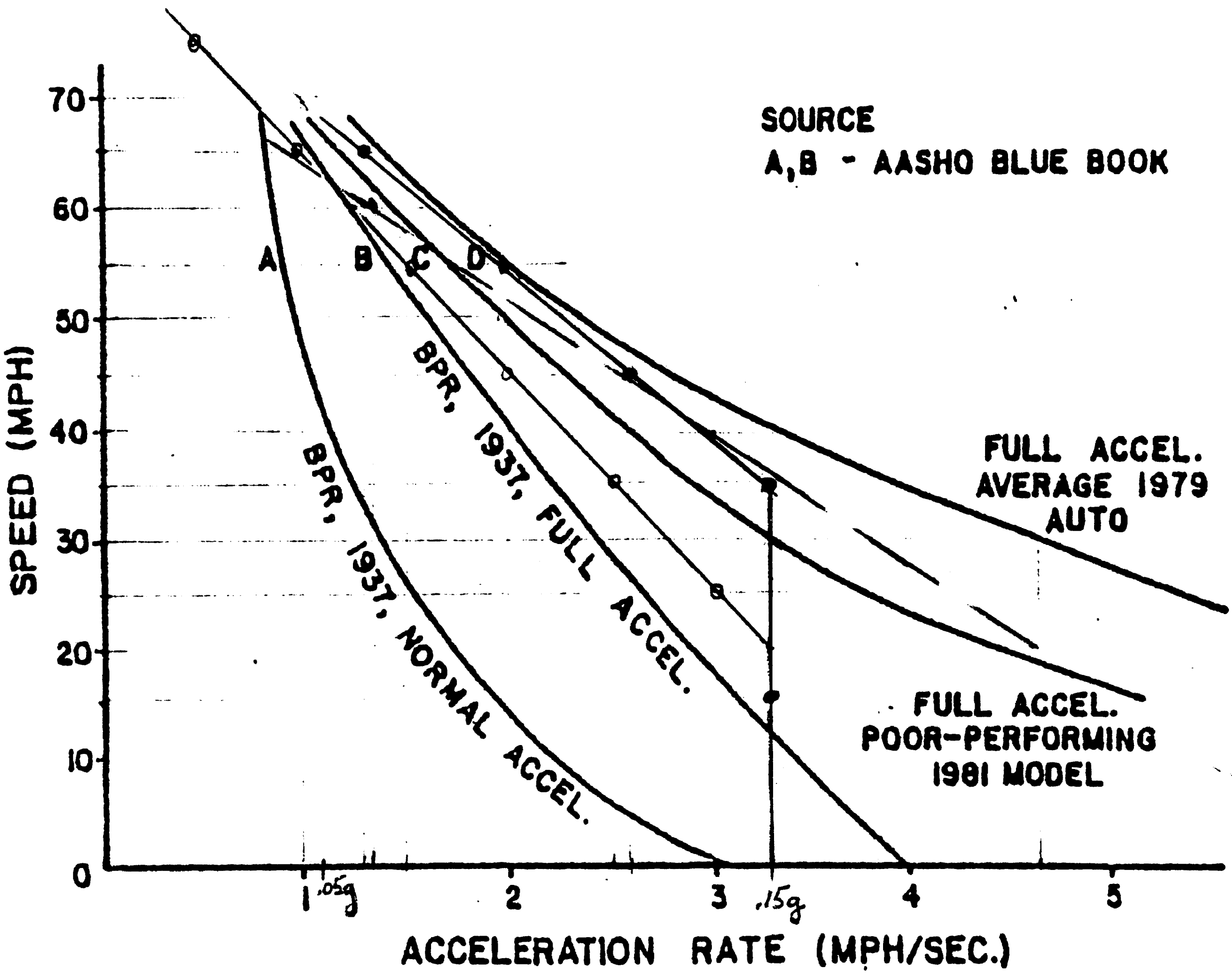
TABLE 5

NORMAL ACCELERATION OF PASSENGER CARS

Speed Change, mph	Acceleration mph/s
0-15	3.3
0-30	3.3
30-40	3.3
40-50	2.6
50-60	2.0
60-70	1.3

Source: NCHRP Project 2-5A, 1971 [11]

vehicle fleet in 1981, 1985, and 1995. These projections are reasoned extrapolations from vehicle data for the years 1967 through 1978. Based on data for 1967 and 1978 and projections for 1981, 1985, and 1995 (see reference [12], Figure 2) less than 10% of the passenger vehicles sold will have, or, have had power capabilities less than 40 watts/kg (approximately 0.025 hp/lb). An assessment of the acceleration performance of a 40 watt/kg car is given in [12] and the results have been used to construct the 40 watt/kg curve presented in Figure 14. This curve falls between the data reported in [9] representing the average '79 and poor '80 vehicles. This agreement lends credence to the proposition that a vehicle similar to the poor '81 vehicle or the 40 watt/kg vehicle approximately represents



SUPERIMPOSED CURVES

———●———	ITE Handbook [7]
— — —	40 Watt/kg [12]
———o———	"Design" car

Figure 14. Comparisons of velocity versus acceleration characteristics of passenger cars.

the acceleration capability of the low performance vehicles of the future.

The question remains as to what curves should be used for designing acceleration lanes. A possible choice is to select a speed-versus-acceleration-rate-characteristic that is deemed to be a rational mixture of driver preferences and vehicle capabilities. For example, the curve labeled "design" car in Figure 14 represents one such choice. In this case an upper bound on acceleration is set at 0.15g per the ITE information [7] at low speed. At speeds greater than 20 mph, the acceleration rate decreases from roughly 70% of a poor vehicle's capability at 20 mph to an amount that corresponds to almost all of that vehicle's capability at 70 mph. Specifically, the average acceleration capabilities for various speed ranges for this "design" car are given in the following table:

TABLE 6

"DESIGN" CAR ACCELERATIONS

mph	Average Acceleration
0-20	.150
20-30	.137
30-40	.114
40-50	.091
50-60	.068

Based on the accelerations provided in Table 6, the two curves superimposed in Figure 13 are obtained for the distance required to accelerate to various speeds from initial speeds of 0 and 30 mph.

Examination of the curves presented in Figure 13 indicates that drastic changes in the lengths of acceleration lanes are implied by the use of the design car concept. For example, the original AASHTO curve indicate that a distance of 1000 ft is required to accelerate from 30 to 50 mph, while, according to the curves for the design car, a distance of 535 ft would be required to accelerate from 30 to 50 mph. Clearly, such drastic changes should be examined critically.

Possibly, the longer lengths are needed for entering vehicles to find a gap in the traffic stream. Or, heavy trucks (with nowhere near the acceleration capabilities needed to match passenger cars) are a limiting factor. Nevertheless, if design policy is to be based on the acceleration of the car/driver system, the AASHTO curves should be re-examined and updated.

2.3.5 Passenger Cars; Passing on Two-Lane Highways. For passing on two lane highways, the design policy [1] specifies the sight distances needed for one vehicle to pass another before encountering oncoming traffic. The total passing sight distance specified in [1] is divided into 4 parts: (1) initial acceleration distance, (2) distance traveled in the left lane, (3) clearance safety margin, and

(4) distance traveled by the opposing vehicle. Vehicle acceleration performance is involved only in the first of these four items.

The design policy provides the following information (see Table 7) concerning the acceleration performance of the passing vehicle during the initial maneuver.

TABLE 7

INITIAL MANEUVERING CHARACTERISTICS

Speed Group, mph (passing vehicle)	30-40	40-50	50-60	60-70
Average passing speed (mph)	34.9	43.8	52.6	62.0
Average acceleration (mph/s)	1.40	1.43	1.47	1.50
Time (seconds)	3.6	4.0	4.3	4.5
Initial maneuver distance (ft.)	145	215	290	370

As observed in [9], the acceleration rates given in the design policy can be compared with vehicle capabilities to provide an indication of the "adequacy of the design values." Referring to Figure 11 in Section 2.3.3, an average acceleration of 1.5 mph/sec. can be exceeded up to a maximum velocity that depends upon vehicle characteristics; viz.,

Vehicle	Max. Velocity for 1.5 mph/sec. Acceleration
1937, BPR	52 mph
Poor-performing 1981 model	60 mph
Average 1979 auto	64 mph

Examination of these data and Figure 11 indicate that the design policy and car acceleration capabilities start to approach each other in the 60-70 mph speed group.

However, the fact that the design values are close to the capabilities of low powered vehicles may not be of great significance. The contribution of the acceleration part to the total passing sight distance is small, approximately 15% of the total. In addition, drivers of low-powered vehicles may be expected to refrain from attempting high-speed passing-maneuvers, or at least, to fall back once they observe that they do not have adequate power.

2.3.6 Recreational Vehicles; Hill Climbing. The power-to-mass ratio of recreational vehicles (RVs) are approximately equal to one-half of the power to mass ratios of passenger cars (possibly, because many of RVs are composed of a car and a trailer of nearly equal weight). As a conservative estimate of the lower bound of the 1978 RV population, Glauz et al. [12] determined a power to mass ratio of 19.7 w/kg (0.012 hp/lb). They predicted that this lower bound on power-to-mass ratio would apply into the future through 1995. Their estimate of the acceleration performance of the lowest 10% performance of RVs is summarized in the following graph.

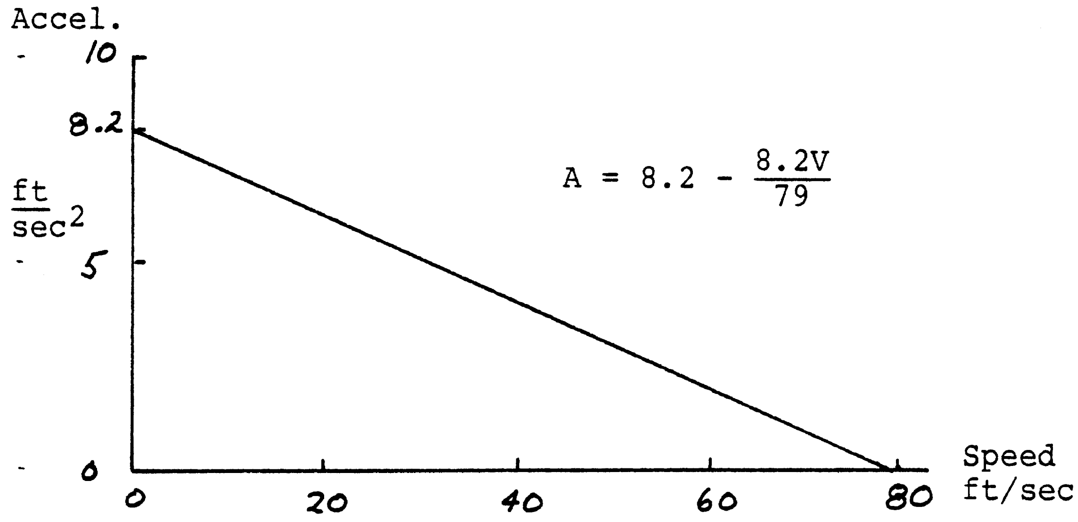


Figure 15. Acceleration performance of RV's, 1978-95, [12].

Although the estimated acceleration characteristic given in Figure 15 is a rough approximation to performance capability near V = 79 ft/sec. (approximately 54 mph), we have used that acceleration characteristic (Figure 15) to computed distances for 10 mph speed reductions for upgrades entered at 55 mph. Example results for a 10 mph reduction in velocity are 1300 ft. on a 5% upgrade, 900 ft. on a 6% upgrade, and 700 ft on a 7% upgrade. These points are superimposed on Figure 16 which also contains (a) calculated results obtained in [5], and (b) measured data that is used in AASHTO design policy [1]. Examination of Figure 16 indicates that the critical length of grade determined for a

0.012 hp/lb recreational vehicle is considerably less than that specified by AASHTO (the dashed line).

This finding is not unexpected given the difference in hp/wt. ratios involved (i.e., 0.022 versus 0.012 hp/lb). However, measured results show that drivers of recreational vehicles do not use all of the power available to them [5]. Hence the difference between (a) measured performance for a 0.022 hp/lb vehicle, and (b) calculated performance for a 0.012 hp/lb vehicle is not as large as it would be if drivers used almost all of the available power.

No comprehensive source of information on the acceleration performance of recreational vehicles has been identified in this study. Nevertheless a large body of sustained speed data has been obtained and processed in California by Ching and Rooney [13]. For vehicle/travel-trailer combinations, they [13] have observed that sustained speeds of 43 mph on 3% upgrades and 30 mph on 6% upgrades correspond to 12.5 percentile vehicles. By assuming an acceleration characteristic of the following form, we have used Ching and Rooney's data to add two more points to Figure 16; viz,

$$\bar{A} = - RR - AV^2 + B/V$$

where

$\bar{A}$ is the average acceleration in g's

RR is a rolling resistance factor equal to 0.02

A is an aerodynamic drag factor equal to 4.76 (10^{-6}) (chosen to match the data in [13])

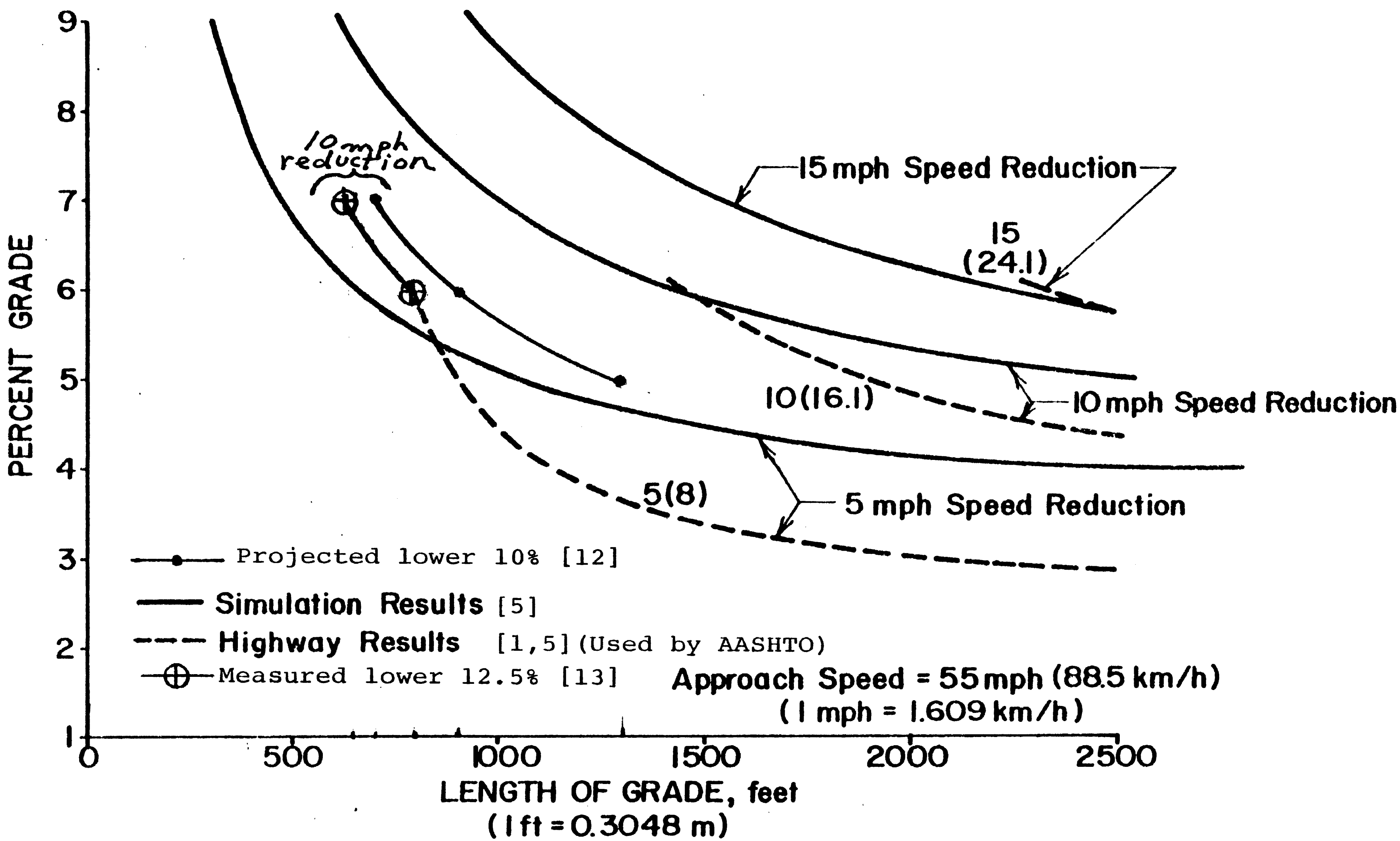


Figure 16. Critical lengths of grade for design, assumed recreational vehicles and field test data. [5]

B is a thrust factor equal to 2.53 (chosen to match the data in [13])

and V is forward velocity in mph.

The results extrapolated from the sustained speed information given in [13] indicate that the AASHTO values of critical length of grade are much longer than those corresponding to a low powered recreational vehicle. For example, on a 6% upgrade the critical length is 1500 ft according to the AASHTO curve while it is 800 ft for a low powered recreational vehicle operated by a driver that only uses approximately 0.007 hp/lb (i.e., 143 lb/hp). This result does not appear to be unreasonable if the low-powered vehicle had a capability of 0.012 hp/lb.

Even though only a few sources of data are available, the approach used in extrapolating from Ching and Rooney's results can be employed to develop design curves for critical length of grade for both average and 12.5 percentile recreational vehicles. In the future we will pursue the development of these design curves.

3.0 SUMMARY OF SUGGESTIONS (PRELIMINARY RECOMMENDATIONS)

This section summarizes the suggestions presented in various sections of this report and provides additional insights to be considered in developing new design charts.

The following list of suggestions is based on comparing current vehicle characteristics with the curves proposed for the AASHTO design policy [1]:

1. The design curves, proposed in [1] for determining the critical length of grade relating to the need for climbing lanes, are representative of the acceleration performance of low-powered heavy commercial vehicles currently in use. The design vehicle has a weight-to-power ratio of 300 lb/hp, which is higher than that applicable to approximately 84% of the heavy vehicle fleet (60,000 to 80,000 lb range) in 1977. Once data from the TIU survey of the 1982 truck fleet are available, the design curves should be reevaluated and, if necessary, recalculated.

2. The design curves, used to determine accelerating time in connection with sight distance at intersections, appear to be conservative with respect to both the tractor-semitrailer (WB-50) and the passenger car (P) design vehicle. That is, even low-powered cars and trucks in the current vehicle fleet are expected to be able to accelerate faster than the design curves imply. The design curve for the straight truck (SU) is difficult to evaluate because this type of design vehicle is not well defined in the AASHTO policy. From a vehicle characteristics standpoint,

the design policy could be improved if the acceleration versus velocity characteristics of the design vehicles were specified.

3. The recommended lengths of acceleration lanes are based on estimates of the normal acceleration rates of passenger cars. A discrepancy exists between the normal acceleration performance presented in the AASHTO design policy and the normal acceleration rates presented in the ITE handbook. The preliminary findings of this study appear to indicate that (a) the ITE values are higher than appropriate for the current vehicle fleet, and (b) the AASHTO values are considerably lower than needed. We suggest that this matter requires further investigation and, as suggested earlier, the presentation of acceleration versus velocity characteristics for design vehicles would aid in clarifying the basis for the speed-distance curves used in highway design. Possibly, detailed results from [8] will aid in defining appropriate levels of normal acceleration at various operating speeds.

4. The design values of the average acceleration during the initial maneuvering phase of passing on a two-lane road appear to be reasonable, based on the characteristics of an average 1979 auto and a poor-performing 1981 model.

5. The design policy includes information on the acceleration characteristics of recreational vehicles. These characteristics are used for evaluating the need for

climbing lanes in situations where RV's are likely to impede traffic. In this case the design policy is not conservative, in that the acceleration characteristics of the design vehicle are representative of an average car-travel-trailer combination rather than a low-percentile (10 or 12.5%) vehicle. If deemed necessary, the critical length of grade curves for an average RV could be augmented with curves for a low-powered RV. (The approach suggested in Section 2.3.6 could be applied here.)

From a vehicle dynamicist's point of view, the most difficult part of understanding the AASHTO design policy (as it pertains to acceleration performance) derives from a lack of acceleration versus velocity information defining design vehicles and normal acceleration performance. This matter has already been alluded to in the list of suggestions just presented. In the future we anticipate that greater reliance will be placed on computerized methods in performing design analyses. Models of vehicle acceleration performance that are not unduly complex appear to be suitable for use in highway design. Given appropriate computational capabilities, specifications of acceleration performance in terms of acceleration capability at various speeds can be used to calculate results for a wide range of situations. Changes in vehicle characteristics either due to evolution over time or to idiosyncracies of a local vehicle fleet can be readily accounted for, if suitable (generally accepted) models are available. The dependence upon specialized design charts may shift to the use of fundamental information that can be used in a variety of important applications. At some time in the future, both fundamental information and particularly important results based on that fundamental information could be presented in a manner consistent with current models of acceleration performance.

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APPENDIX A

LITERATURE REVIEW:
ACCELERATION PERFORMANCE OF HEAVY TRUCKS

A.1 The Critical Length of Grade

The "critical length of grade" is defined [18]* as "the maximum length of a designated upgrade upon which a loaded truck can operate without an unreasonable reduction in speed." This definition is given numerical significance by (1) assessing the acceleration performance of a typical vehicle to be used for design purposes and (2) establishing a design "bogie" for what constitutes a reasonable reducttion in speed. In 1965, the AASHTO Policy [3] used performance calculations based on a heavy truck with a weight-to-horsepower ratio of 400 lb/hp and an acceptable loss of speed of 15 mph for a vehicle entering the grade in question at 47 mph. A recent draft revision of this Policy [18] employs calculations for a 300 lb/hp truck and recommends a speed loss of 10 mph. An entering speed of 55 mph is used in the recent calculations. The reasons for these changes are (1) to be more representative of the performance of the current heavy truck fleet and (2) to avoid the sudden increase in accident involvement rate that is reported to occur when the difference between truck speed and the average running speed exceeds approximately 10 mph [20].

Graphs of critical length of grade may be constructed from speed-distance curves depicting the acceleration (deceleration) performance of a heavy truck operating on upgrades of various amounts (for example, see Figure 1). Figure 1 can be cross-plotted to construct lines of constant speed reduction as shown in Figure 2. As illustrated in Figure 2, a highway designer can enter this graph at a designated grade (on the vertical axis) and read the critical length of grade (from the horizontal axis) for a selected reasonable reduction in speed.

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*References in this appendix refer to the bibliography given on page 87.

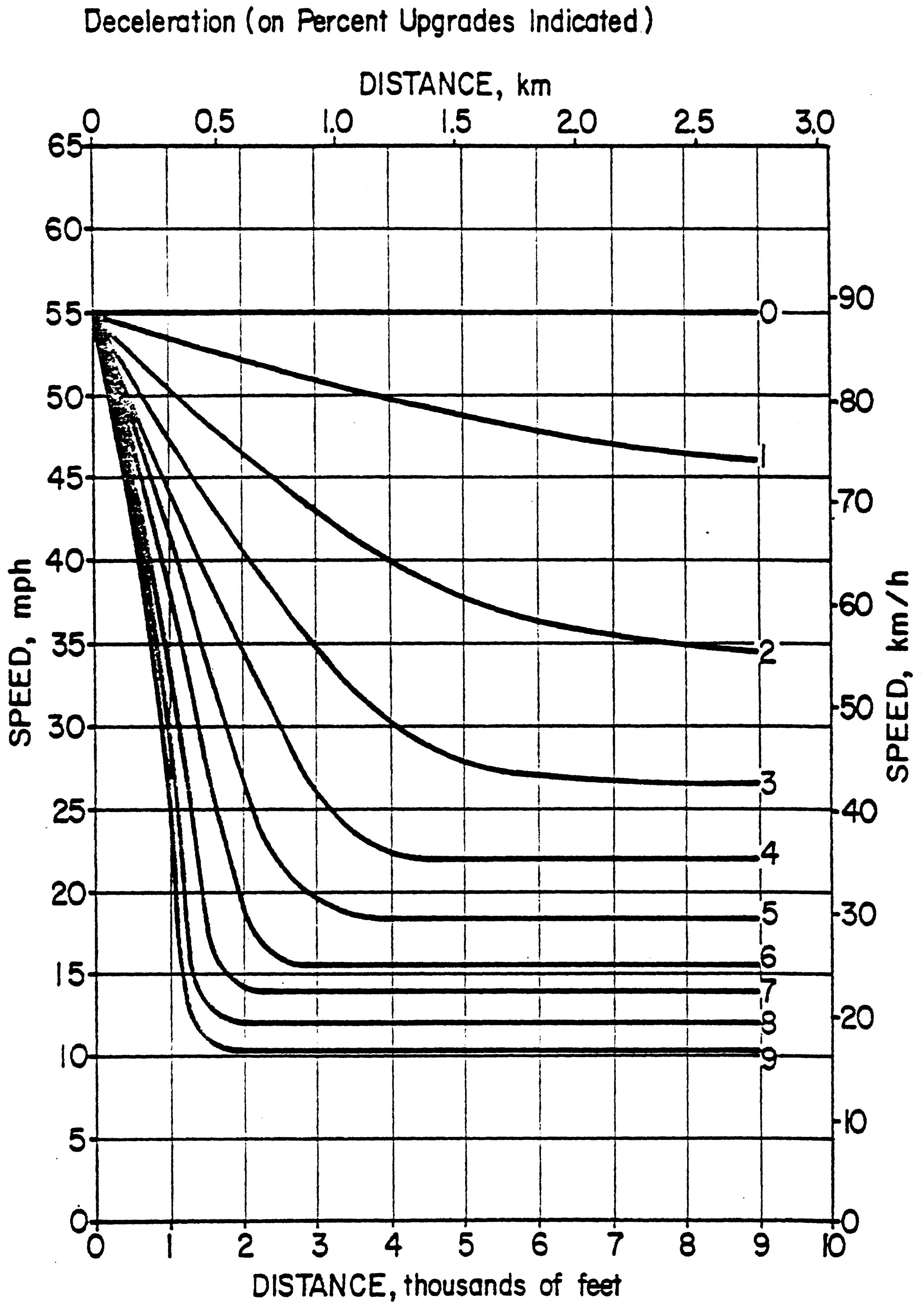


Figure 1. Speed-Distance Curves for a Typical Heavy Truck of 300 lb/hp (182.5 kg/kW) [19].

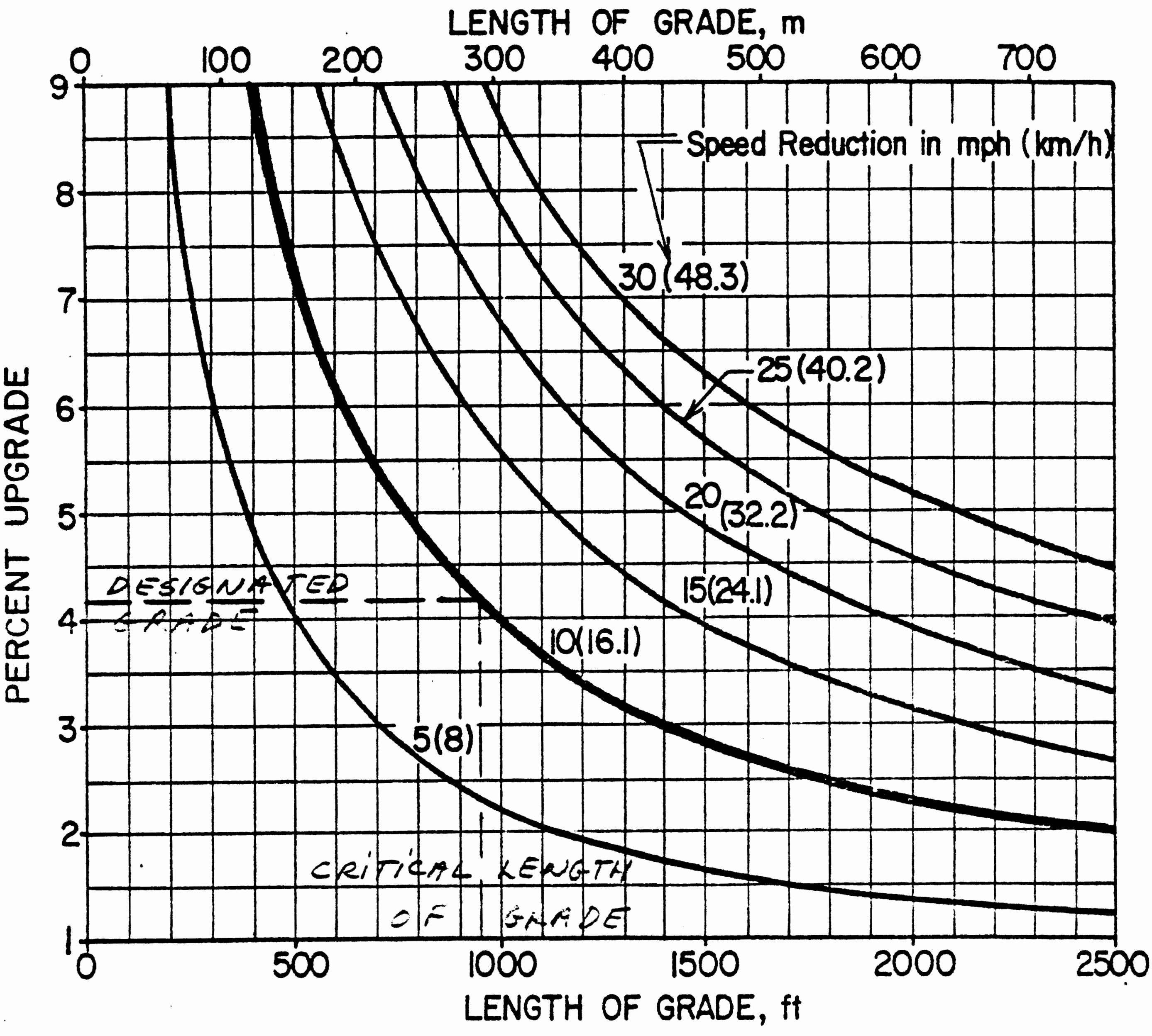


Figure 2. Critical Lengths of Grade for Design, Assumed Typical Heavy Truck of 300 lb/hp (182.5 kg/kW), Entering Speed = 55 mph (88.5 km/h) [19].

A.2 The Origins of AASHTO Design Policy and Subsequent Developments with Respect to the Prediction of Truck Deceleration in Climbing a Grade

This section reviews the state of the art in predicting the acceleration performance of heavy vehicles.

In 1955, Huff and Scrivner [1] presented a simplified theory for describing the motion of heavy vehicles on grades. They employed a very simple vehicle model based on the following expression of Newton's laws of motion:

$$\frac{1}{g}\left(\frac{dV}{dt}\right) = \frac{P}{W} - \sin\theta \qquad (1)$$

where

- W is the gross weight of the vehicle
- V is the forward velocity
- t is time
- g is the gravitational constant
- P is the net driving force
- θ is the grade angle

For the purposes of developing a simple model, they neglected the inertial resistance due to the angular acceleration of the rotating wheels, transmission, engine, etc., and lumped together the influences of rolling resistance, aerodynamic drag, transmission (driveline) efficiency, engine characteristics, and driver shifting performance.

The heart of their approach was a graph of P/W (net driving force divided by weight) versus velocity (see Figure 3). As indicated in the figure, maximum sustained speeds for various levels of grade were used to construct the graph of P/W versus V.

To find velocity information, Huff and Scrivner integrated the equation of motion (Eq. (1)) with P/W represented by the function of velocity that they developed from field data. A further integration of the velocity results produced distance information that was used in constructing speed-distance curves.

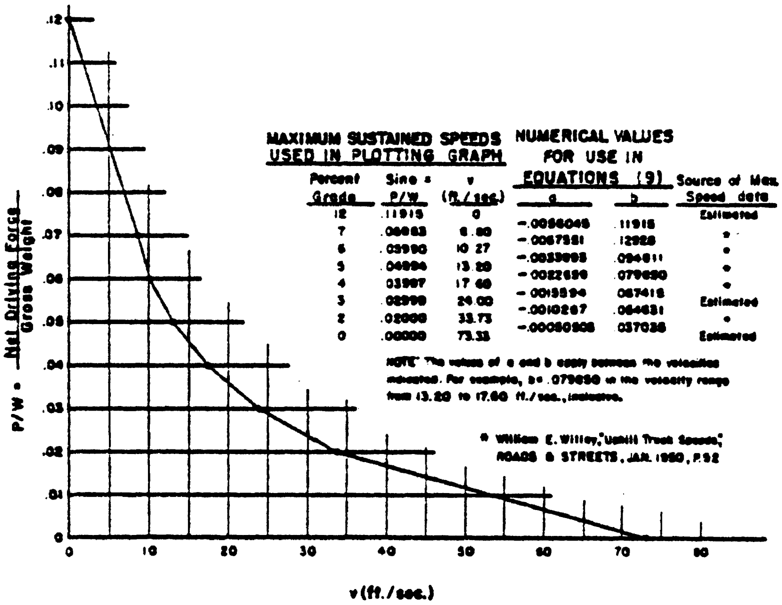


Figure 3. P/W Versus V [1].

In order to check or correct the data presented in Figure 3, road tests were performed in 1953 using a 57,180-lb test vehicle. By analyzing the test results, a new version of the "P/W versus V" curve was developed (see Figure 4).

The authors of [1] concluded that (1) even under controlled conditions with a skilled driver, the speed-distance relationship is not always consistent, (2) the speed-distance curves computed by the simplified theory correspond to the test data and are sufficiently accurate for use in the design of climbing lanes, and (3) the SAE procedure (in 1955) for computing

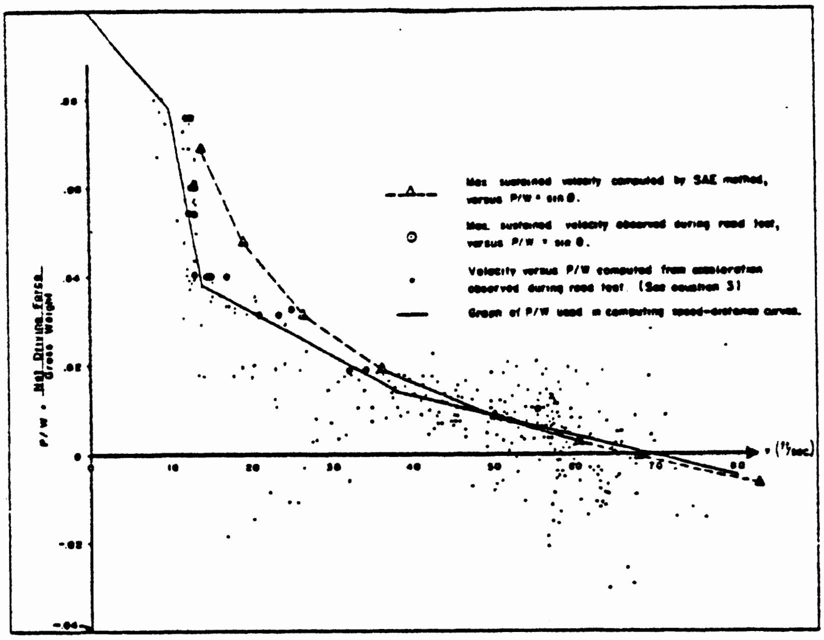


Figure 4. Graph of P/W Versus V from 1953 road-test data (solid line) and from SAE "Truck Ability Prediction Procedure" (dashed line). [1].

maximum sustained speeds for any weight-to-horsepower ratio provides values that could be used in conjunction with the simplified analysis to predict truck motion, at least approximately, without resorting to full-scale tests.

Although more sophisticated analytical methods are currently employed leading one to challenge the conclusions of this initial study, nevertheless, the basic elements of developing a method for predicting truck acceleration characteristics on grades are illustrated in Reference [1]. Specifically, these basic elements consist of (1) selecting a model of both sufficient accuracy and simplicity, (2) developing methods for obtaining parametric values and empirical functions that are suitable for (a) use in the model and (b) representative of the performance of vehicles on highway grades, and (3) evaluating the practicality of using a limited set of basic information on vehicles to minimize the need for resorting to full-scale tests.

In 1962, Firey and Peterson [2] devised a means for calculating the speed versus distance history of large trucks traversing vertical curves at wide-open throttle. They employed a model of the vehicle that treated the net driving force in more detail than was done in [1]. In [2], the driving force was divided into two components—the thrust force due to engine torque, F_T, and a "total rolling resistance" force including both tire and air resistance, F_R. The total resistance force, F_R, was calculated by means of an empirical equation based on coasting tests of several large commercial vehicles. The engine thrust force, F_T, was based on the brake horsepower of the engine, rated engine speed, a tire size factor (effective rolling radius), and the gear ratios of the transmission, auxiliary transmission, and the drive axle.

In addition, the authors [2] analyzed both constant grades and vertical curves, and they observed that separate calculations applied when (1) the throttle was wide open and (2) the clutch was disengaged and the driver was shifting gears. They noted that "there is no such thing as 'typical' or 'standard' gearing in these trucks,..." and adopted a calculation procedure based on a "usable engine speed ratio" of 0.8. (For example, during deceleration, a downshift is assumed to occur when the velocity reaches 0.8 times the maximum vehicle speed attainable in the assumed gearing arrangement.) Although they made separate calculations for the wide-open-throttle (WOT) and gear-shift periods, they smoothed their results because the assumed shift points did not correspond to any real vehicle (see Figure 5).

Comparisons between calculated and measured speeds are presented in [2] for a sag and a summit curve and for numerous cases on various vertical curves. The agreement shown between test and calculation appears to be very good and it is stated that in cases where the difference between calculated and measured values is more than 2 mph, the driver either did not operate the throttle as assumed (i.e., wide open), or the gearing was such that the driver had difficulty achieving the theoretical maximum sustained speed.

Clearly, References [1 and 2] do not constitute an exhaustive literature search, but they are representative of the state of the art prior to 1965 and the publication by the AASHO (currently AASHTO) of "A Policy on Geometric Design of Rural Highways" [3]. For example, graphs from Huff and Scrivner [1] are presented on page 197 of the AASHO policy [3].

Since 1965, numerous organizations have developed calculation procedures for predicting the acceleration/deceleration performance of trucks on grades. Vehicle, transmission, and engine manufacturers have developed computer programs for aiding customers in selecting vehicle components (engine, transmission, rear axles, etc.) that will provide satisfactory service in the operating environment anticipated by the customer (for example, see References [4 through 6]). These computer programs are based on the mechanics (physics) of vehicle motion. The parameters used in these programs are selected to represent specific vehicles, their components, the grades the vehicles operate on, and the manner in which the driver shifts gears and controls the throttle.

Although the individual computer programs currently used in the trucking field may differ in implementation details and computational algorithms, the basic physical factors included in these programs are well described in an SAE Buckendale lecture presented by G. Smith [7]. Smith states that

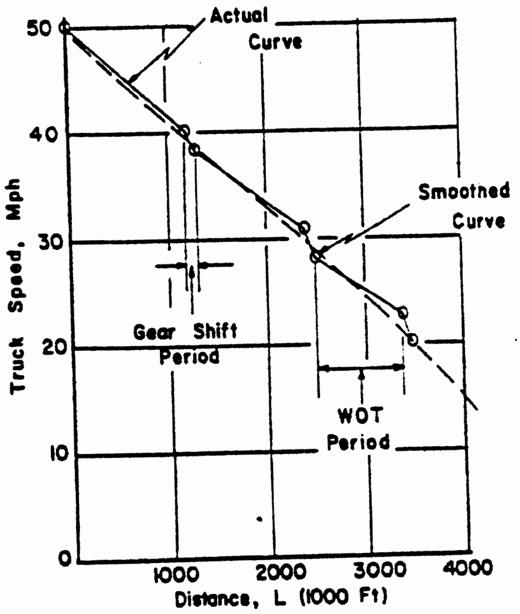
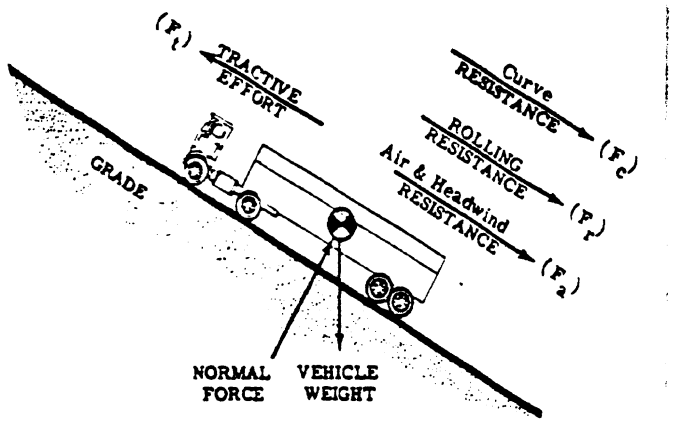


Figure 5. Example of Truck Speed vs. Distance History on an Upgrade [2].

"for performance considerations, the important forces acting on a vehicle are the driving forces from the power plant, tire rolling and cornering forces, aerodynamics, and the weight of the vehicle itself" (see Figure 6). The curve resistance force, F_c, deriving from tire cornering force, is, obviously, not important for straight-line motion, but it can be important for slowly moving vehicles negotiating curves on an upgrade. (This is a factor which might be overlooked in selecting sites for field studies or in estimating vehicle speeds on winding roads.) The other forces shown in Figure 6 (i.e., F_t, F_r, F_a, and $W \sin \theta$) are the basic factors utilized in computerized models. The rolling resistance force, F_r, is primarily dependent upon vertical load, but a small term proportional to velocity is sometimes included. The aerodynamic drag, F_a, is proportional to velocity squared, thereby making it unimportant at low speeds, but important at high speed.

Since 1970 (when [7] was presented), the increased emphasis on fuel economy has increased the use of radial tires and aerodynamic shields (aero-aids) on trucks. For approximating the influence of radial tires, rolling resistance forces have been reduced to approximately 0.7 times the rolling resistance used for bias tires [8]. Similarly, for approximating the influence of aerodynamic shields, aerodynamic forces have been reduced to approximately 0.9 to 0.7 times the aerodynamic drag of a comparable vehicle without drag reduction devices [8]. With respect to these "natural" sources of retardation, all four possible combinations (radial or non-radial tires and with or without aerodynamic improvements) exist in the vehicle fleet and these differences have a noticeable effect on performance. It has been estimated that without aero-aids and radial tires, the sum of F_r and F_a is approximately equivalent to the influence of a 2% grade for a heavy truck traveling at 50 mph [7,8]. With aero-aids and radial tires, this natural retardation is reduced to being approximately equivalent to a 1.2% grade.

Note that the equation for vehicle acceleration presented in Figure 6 employs an equivalent weight factor, W_e, which includes the influence of the rotating components. For example, if a tractor-semitrailer had (1) 18 10x20 tires, (2) an engine/clutch inertia of 2 ft-lb-sec^2, (3) a transmission ratio of 12.5 in the lowest gear and 1.0 in the highest gear, and (4) a



$$\text{Acceleration } (\ddot{x}) = \frac{g}{W_e}\left[F_t - (F_r + F_a + F_c + W\sin\Theta)\right]$$

Note: W_e is the total vehicle weight plus the translational equivalent of the rotating masses.

Figure 6. Vehicle Free-Body Diagram. [7].

rear axle ratio of 4, then calculations show that the additional weight factor, $\Delta W = W_e - W$, is 58,800 lbs in the lowest gear and 1,800 lbs in the highest gear. For an 80,000-lb vehicle operating at high speeds (that is, in the highest gear), the influence of omitting ΔW would amount to approximately a 2 to 3% error in acceleration capability. However, at very low speeds, even for an 80,000-lb vehicle, W_e is 1.735W, that is, the influence of omitting ΔW amounts to approximately a 60% over-estimate of the acceleration of the vehicle. Clearly, the importance of the influence of the rotating components ranges from minor at high speed to major at crawl speeds.

The primary difference between sustained speed performance and performance during acceleration relates to the difference between W_e and W. At sustained speed neither the rotating components nor the vehicle are accelerating and the required thrust force balances the drag forces, viz.:

$$F_t = F_r + F_a + W \sin \theta + F_c$$

When a vehicle is losing speed on an upgrade, the energy stored in the rotating components tends to aid the engine torque in attempting to maintain speed. If the vehicle is increasing speed, part of the engine thrust is required to accelerate the rotating components, thereby reducing the forward acceleration of the vehicle.

The influence of elevation is significant in predicting acceleration performance. Following the development presented in [7], the corrected horsepower, hpc, is related to elevation by the following pair of equations:

$$hpc = CF\ hpo$$

where CF is an elevation correction factor and hpo is the net engine horsepower including the influence of the power required by accessories, and

$$CF = (1 - \frac{0.04E'}{1000})$$

where E' = elevation (ft) above the stated elevation on the given horsepower curve (hpo)

For example, if E' is 5,000 ft, then CF = 0.8, that is, at a mile above sea level the net horsepower available for generating thrust, F_t, is approximately 20% less than that stated for a standard elevation (typically 500 ft or sea level).

(Incidentally, aerodynamic drag also decreases with altitude because the density of the air decreases.)

For a detailed analysis and also for understanding the performance of a particular vehicle, the relationships between engine power and torque and engine speed are required. Engine power increases with engine speed, reaches a maximum near rated speed, and falls off rapidly above rated speed due to governor control (see Figure 7 and Reference [9]). Clearly, the torque at full throttle varies with engine speed in a manner that is related to the manner in which power varies with engine speed; for example, compare the full throttle curves shown in Figures 7 and 8. In this case, the engine torque versus speed functions are characterized by maximum torques that occur at approximately 5/7 of the rated engine speed, even though maximum power is obtained at rated engine speed. Typically, for normal line-haul operation an upshift is attempted when the engine speed is greater than the rated engine speed [9]. After completing the gear shift, the engine speed will be below the rated speed by an amount that depends upon (1) the gear "splits," i.e., the decreasing ratios between gears involved, and (2) the amount the vehicle slows down during the shifting period. The transmission, drive axle ratio, and tire radius may be arranged so that the initial engine speed, attained after shifting gears, is approximately equal to the maximum torque speed. Even though the engine power is initially below rated power when a new gear is selected, the engine torque may be higher than the torque at rated engine speed. (This phenomenon is sometimes quantified by a term called "torque rise.")

Although the vehicle's thrust force depends upon the combination of engine torque, gear reductions, driveline losses (chassis friction), and tire radius, the maximum sustained capability of the vehicle is limited by

1 Hp ≡ 0.7457 kW

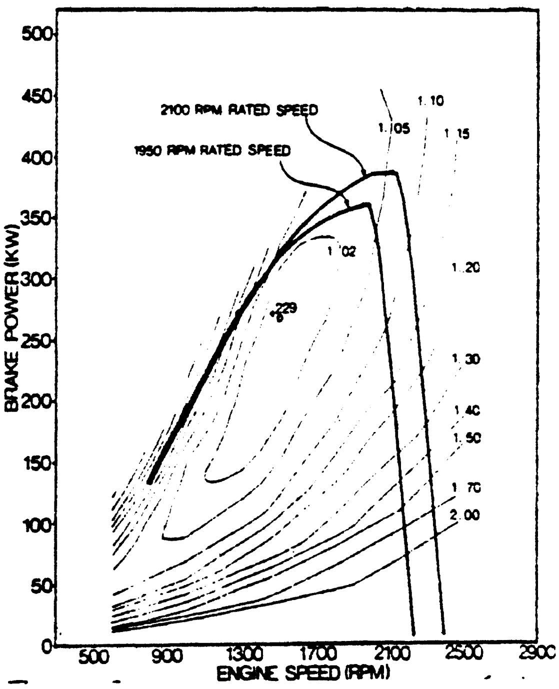


Figure 7. Power Versus Speed (relative fuel islands with lines of constant specific fuel consumption at the indicated ratios above the minimum 0.229 Kg/hphr) [9].

/ Nm ≡ 0.7376 ft. lb.

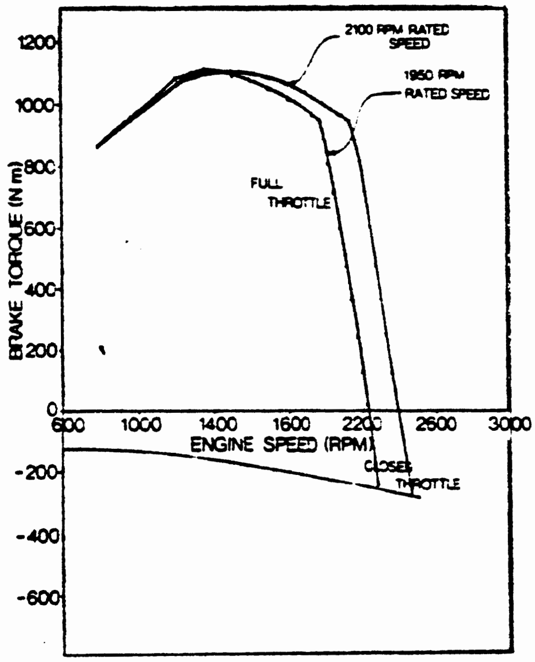


Figure 8. Full Throttle and Closed Throttle Engine Torque Curves [9].

the rated horsepower of the engine. For a given horsepower limit, the drive thrust at the wheels will be large at low vehicle speeds corresponding to low gears (high gear ratios) and low at high vehicle speeds corresponding to high gears (low gear ratios).

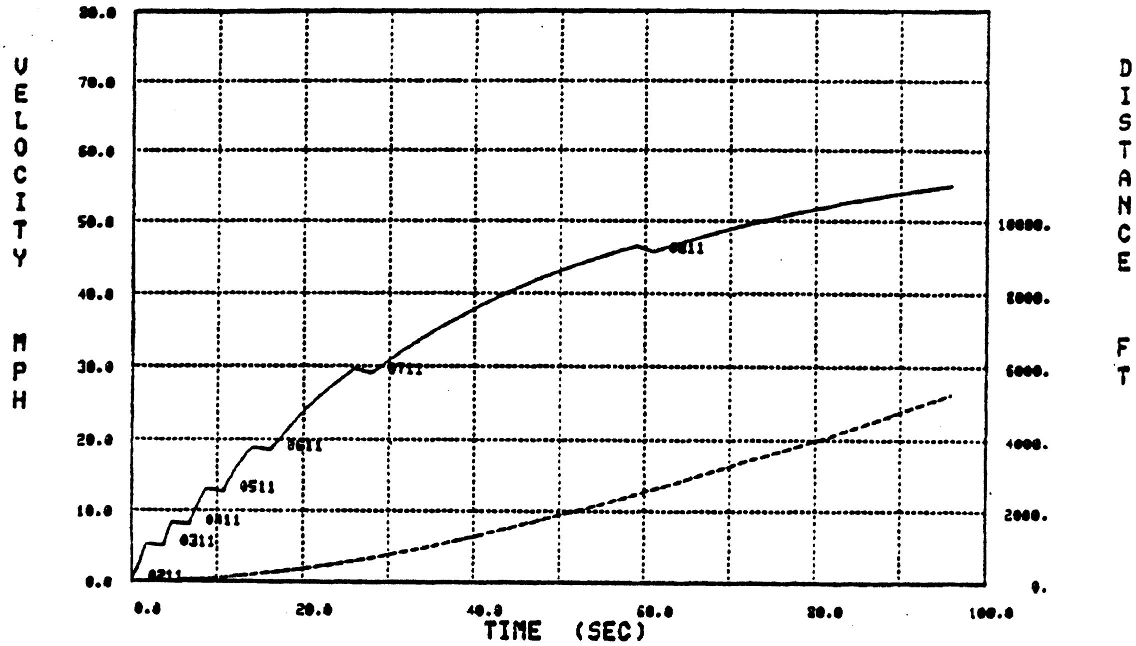
The level of drive thrust, F_t, actually available at the road wheels depends upon the throttle setting and the driveline losses. By modulating the throttle, the driver can obtain any engine torque between the closed and full throttle capability of the engine (see Figure 8). However, drivers of large trucks usually accelerate through the gears at full throttle [7] ("they put the pedal to the metal" so to speak). Hence, it is common practice to make calculations at full throttle. The driveline losses can be represented by an efficiency factor and/or a viscous loss factor. For typical transmissions and drive axles, the efficiency is nearly constant between full throttle and approximately 30% throttle [7]. When the entire driveline is considered, its efficiency tends to be nearly constant for all gears [7]. Current computational procedures employ driveline efficiencies ranging from 0.94 for highway vehicles with automatic transmissions and single drive axles to 0.86 for 6x4 highway tractors with manual transmissions [4-7]. The driveline efficiency relates directly to the drive thrust available at the road wheels and it represents a significant loss in the force available at the drive wheels. For example, for a 70,000-lb vehicle that is traveling at 37.5 mph with a 250 hp engine, a driveline efficiency of 0.86 represents a loss of 35 hp which is approximately equivalent to a change in grade of 0.5%, that is,

$$\Delta\theta = \frac{(1-.86)250(375)}{(70,000)(37.5)} = .005 \text{ rad} \equiv 0.5\%$$

Computational procedures based on the physical factors described by Smith [7] can produce very accurate results for well-defined vehicles operating on accurately described highway routes [7,9]. In order to make accurate predictions of fuel consumption, realistic representations of driver control practices are needed. This requirement means that manufacturers, who are attempting to aid customers in selecting vehicles to purchase, are faced with the same difficulty as highway engineers, who are trying to predict how vehicles will be driven. Specifically, both manufacturers and highway engineers need to know how drivers will operate vehicles in service. In this regard, rules for when to shift gears and for modulating the throttle are given in [9] and shifting periods are said to be from 1 to 2 seconds for vehicles with manual transmissions [7].

Nevertheless, there is a dichotomy between a manufacturer's and a highway engineer's perspective on predicting acceleration performance. The manufacturer combines detailed and specific information pertaining to a particular vehicle in order to predict acceleration performance (see Figure 9, for example). In this case, the calculation procedure contains detailed information on the characteristics of the specified engine, transmission, rear axle, and engine accessories listed in Figure 9. The vehicle is started in second gear because there is enough torque available. In the second gear, as indicated in Figure 9, it appears that the acceleration level may be limited by tire/road friction. Although the descriptive factors (printed below the graphs in Figure 9) do not contain driver characteristics, the plotted results indicate that a two-second period was allowed for shifting and probably the simulated driver started to up-shift when the engine speed exceeded rates speed. Also, the throttle was most likely wide open. It is our understanding that driver representation is an area where manufacturing organizations may feel that they have an edge over their competition. In contrast to the manufacturer, the highway engineer may not have detailed information on the vehicle he wishes to analyze. In fact, the highway engineer may wish to analyze the average or the 15th percentile driver-vehicle combination. It appears that the highway engineer would not object to using the manufacturer's methods, if parametric values were readily available.

The highway engineer is interested in how vehicles normally operate on the road. In service, trucks may not be in an optimal condition and driver skill may influence acceleration performance. In a carefully controlled study [10], Hutton, with aid from the Western Highway Institute and the Oregon State Police, examined the acceleration of heavy trucks that were engaged in performing their goods-carrying mission as driven by regularly assigned drivers. According to Hutton, "Driver skill was



ENGINE	123170	PT-240 CUMMINS
TRANSMISSION	23306	FULLER RTO-11607
AUXILLARY TRANSMISSION	NONE	
REAR AXLE	14472	RA-472
REAR AXLE RATIOS	5.38 - 1	
TIRE DESCRIPTION		
TIRE ROLLING RADIUS	20.06 INCHES	

AIR RESISTANCE FACTOR	0.0021990
FRICTION LOSS FACTOR	2.07
ROLLING RESIST FACTORS	7.210 - 1 0.000 - 2 0.000 - 3

GROSS COMBINATION WEIGHT	50000.
WEIGHT ON DRIVE WHEELS	34000.
TOTAL WHEEL WEIGHT	3146.
FRONTAL AREA	85.00
ENGINE ACCESSORY CODE	332211
FAN TYPE	VISCOUS
GOVERNOR CUTOFF RPM	2100.
CLUTCHING RPM	800.

AMBIENT CONDITIONS
TEMPERATURE 85 BAROMETER 29.38
ELEV AT START 0. GRADE 0.00 %
WIND SPEED 0. MPH @ 0. DEGREES

TIME TO REACH SPEED 55.0 MPH IS 95.9 SECONDS.

DISTANCE TRAVELED IS 5239. FEET

SOLID LINE - VELOCITY VS TIME

DASHED LINE - DISTANCE VS TIME

Figure 9. Computed velocity and distance time histories [4].

probably one of the largest of all the many variables that account for the wide band of data scatter." A comprehensive list of possible sources of variations is presented in [10] (see Figure 10). Due to the amounts of variation occurring in practice, the results presented in [10] show that a 300 lb/hp vehicle operating under favorable conditions can perform as well as a 200 lb/hp vehicle operating under unfavorable conditions (see Figure 11, for example). The best of the group of vehicles close to 300 lb/hp (in Fig. 11) and the worst of the group of vehicles close to 200 lb/hp attained just over 30 mph in 1000 feet. Other findings indicate that, when passing a vehicle traveling at 20 mph on a level roadway, a worst-case 100 lb/hp vehicle would have the same time and distance performance as that of a best-case 300 lb/hp vehicle. Although weight-to-horsepower ratio is the primary factor determining vehicle performance, the findings of [10] demonstrate that other driver- and vehicle-related factors do combine to influence acceleration performance to an extent equivalent to at least 100 lb/hp.

In an NCHRP study [11,12], St. John and Kobett develop a computational method that (1) is based on the physical factors discussed by Smith [7] and (2) incorporates many of the ideas used by Firey and Peterson [2] to produce a procedure and results tailored to the needs of the highway engineer. In [11,12], the authors describe a calculation procedure that, to some extent, is designed to reduce the burden imposed by having to acquire a large amount of detailed parametric data to describe a vehicle. Nevertheless, they see fit to include individual factors for rolling resistance, aerodynamic drag, and driveline losses (chassis friction). They provide for the possibility of using net engine horsepower versus RPM curves, not just net engine power at rated speed. The influences of elevation are taken into account. They do not require detailed transmission data, but they do "design" a "typical" transmission arrangement. The effects of engine inertia are included. The computer program has built in rules and parameters for shifting gears and the time required to shift gears. The results from this model are shown to produce acceleration values that compare favorably with acceleration characteristics provided by the Western Highway Institute [13] and the Road Research Laboratory in England [14].

Variables Possibly Causing Scatter of Results —

A. **Road Variables**
1. Coefficient of friction
2. Variations in grade
3. Road rolling resistance

B. **Atmospheric variables**
1. Temperature
2. Barometric pressure
3. Humidity
4. Altitude
5. Wind velocity and direction

C. **Engine variables**
1. Difference in tune
2. Difference in accessories
3. Fuel pump and injector settings.
4. Maximum governed speed
5. Fuel quality
6. State of wear
7. Internal friction
8. Torque curve
9. Air intake restrictions (when was air cleaner element changed?)
10. Exhaust restrictions.
11. 4-cycle vs. 2-cycle
12. Naturally aspirated vs. turbocharged

D. **Vehicle variations**
1. Transmission
   a. type of shift control
   b. number of speeds
   c. lubricant
2. Axle ratio, lubricant, friction
3. Tire size, type, pressure and temperature
4. Frontal configuration, shape and area
5. Type of bodies
   a. truck body
   b. semi or trailer body
6. Type of cargo
   a. liquid
   b. general commodities
   c. livestock
7. Gross vehicle weight
8. Internal friction of wheel bearings, seals, etc.
9. Coefficient of air resistance
10. Number of axles
11. Inertia of rotating components

E. **Drivers' skill**
1. Proficiency at shifting, maintaining maximum throttle
2. Familiarity with vehicle

Figure 10. Possible Sources of Scatter [10]

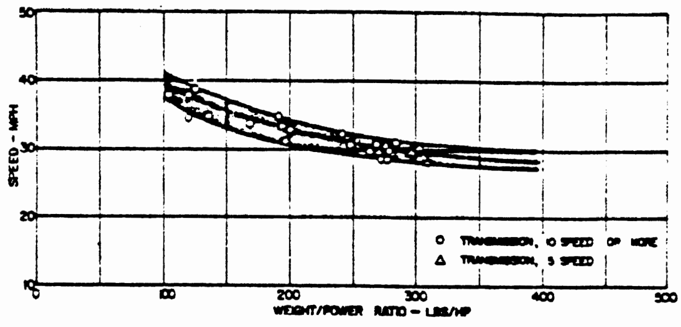


Figure 11. Vehicle Speed Attained While Accelerating from a Dead Stop on a Level Road Surface at a Distance of 1000 Feet [10].

In their work, St. John and Kobett discovered that the coasting period during shifting had an important influence that varied with the level of grade involved. They also found that the coefficients for determining rolling resistance and aerodynamic drag, as given in SAE Recommended Practice J688, were too large. For general calculations, they recommended using their procedure which employed shift delays and adjusted coefficients for rolling resistance and aerodynamic drag.

Recently, Abbas and May [15] utilized the model presented in [11,12] to "develop acceleration performance curves for five-axle trucks along grades of a gradient from -7 to +7 percent on straight sections of roadway under free flow conditions." In this case (Reference [15]), data from vehicle speed measurements, made by CALTRANS, were used to infer vehicle characteristics (specifically, weight-to-horsepower ratios). The data available for inferring weight-to-horsepower ratios consisted of (1) acceleration characteristics measured on a nearly level section of roadway following a weigh station and (2) sustained speeds on grades of 1.78, 3, 4, 5, 6, and 7 percent. Using the weight-to-horsepower ratios determined from the available measurements, the authors were able to use the model developed in [11,12] to estimate (predict) acceleration performance for the average and 12-1/2 percentile "vehicles" on grades from -7 to +7 percent. Even though this process seems somewhat circular, it does provide a means for extrapolating from measured results to unmeasured conditions using physically justifiable rules.

(If deemed necessary, direct assessments of the weight-to-horsepower ratios of vehicles operating on the grades analyzed could be made to verify further the authenticity of the prediction scheme used in [15].)

With regard to estimating the weight-to-horsepower ratio of this particular truck population [15], the authors obtained the results displayed in the following table. The results for upgrades (2 to 7%) are based on matching sustained speed data. The results for grades from +1 to -7% are based on matching the acceleration performance on a nearly level roadway. Reasons for the differences between weight/horsepower ratios on various upgrades and downgrades are not clear. Feasible explanations might

Table 1

Entries in lb/Net Hp							
	Grades %						
Five-Axle Trucks	2	3	4	5	6	7	1, 0, and -1 to -7
Average	160	160	150	140	135	130	230
12-1/2 Percentile	240	250	270	260	260	270	320

include the possibility that the magnitude of a grade influences the population of vehicles that operate on that grade. The results do indicate that a 300 lb/hp vehicle would be well below average, bordering on a very low acceleration capability for this vehicle population.

The approach taken in [15] employs weight-to-horsepower ratios derived from observations of vehicles in use. However, there are other sources of information on weight to horsepower. The Census Bureau, a part of the U.S. Department of Commerce, has been taking economic censuses at five-year intervals. The latest results of these censuses include the Truck Inventory and Use (TIU) survey [16] for the year 1977. (A TIU survey for the year 1982 will be conducted during 1983.) The TIU survey provides data on the physical and operational characteristics of the national truck population. These data are based on a probability sample of the private and commercial trucks registered in each state in 1977. Included in the information gathered in the TIU survey are the maximum gross weight carried in the past 12 months and the horsepower rating of the engine. In addition, information on the use of radial tires, drag reduction devices, and other fuel conservation equipment is contained in the TIU survey form (see Figure 11, Section C - Physical Characteristics). The data derived from the TIU survey provides much of the information needed to estimate the acceleration performance capability of the trucks in each state or in the entire nation.

APPENDIX A. Survey Form

DUE DATE 15 DAYS AFTER RECEIPT OF FORM — Form Approved: O.M.B. Number 41-S76075

Form TC-200 (1-26-77) — U.S. DEPARTMENT OF COMMERCE, BUREAU OF THE CENSUS

TRUCK INVENTORY AND USE SURVEY

1977 CENSUS OF TRANSPORTATION

NOTICE – Response to this inquiry is required by law (title 13, U.S. Code). By the same law your report to the Census Bureau is confidential. It may be seen only by sworn Census employees and may be used only for statistical purposes. The law also provides that copies retained in your files are immune from legal process.

In correspondence pertaining to this report, please refer to this control number

INSTRUCTIONS

In correspondence pertaining to this report, please include State and license number and the control number shown in the address label.

Return the form in the enclosed preaddressed envelope not later than 15 days after receipt.

RETURN TO — Bureau of the Census, 1201 East Tenth Street, Jeffersonville, Indiana 47132

(Please correct any error in name and address including ZIP code)

Section A – VEHICLE IDENTIFICATION AND USE

Item 1 – VEHICLE IDENTIFICATION *(Please correct any errors or omissions in the identification of the vehicle)*

Make	Year of model	Registered weight or capacity	State	License number	Vehicle identification number

NOTE: Please complete this form whether or not you are still the owner of the vehicle identified in item 1.

Item 2 – OWNERSHIP OF VEHICLE

Are you still the owner (or license holder) or lessee of this vehicle?

1 ☐ Yes

2 ☐ No → When did you sell, trade, or otherwise dispose of it? → Month | Year

Item 3 – ACQUISITION OF VEHICLE

a. How did you acquire this vehicle?

1 ☐ Purchased new

2 ☐ Purchased used

3 ☐ Leased from someone else

b. When did you acquire this vehicle? → Month | Year

c. During past 12 months was this vehicle leased or rented to others?

1 ☐ Yes

2 ☐ No

d. How was this vehicle leased or rented?

1 ☐ Without driver

2 ☐ With driver (or as owner operator)

Item 4 – LEASE CHARACTERISTICS

a. Was the lessee –

1 ☐ Private (non-government)?

2 ☐ Unit of government?

b. What is the length of lease or rental agreement?

1 ☐ Less than 30 days

2 ☐ 30 days to 1 year

3 ☐ 1–3 years

4 ☐ More than 3 years

c. Does your agreement include–

1 ☐ Financing?

2 ☐ Maintenance?

3 ☐ Procurement and sale?

Item 5 – CLASSIFICATION OF OPERATOR

Mark (X) the box which is the most appropriate for your type of operation

Not for hire –

1 ☐ Private owner or an individual, or company which just transports its own materials or merchandise. Includes an individual or a business such as a bakery, oil company, or soft drink bottler.

For hire –

2 ☐ Interstate – exempt carrier (not required to have an I.C.C. certificate because only exempt commodities are transported, such as: fresh agricultural products, fish, newspapers, or air freight haulage)

3 ☐ Interstate – I.C.C. certified contract carrier (carrying the goods of other than the owner by individual contract or agreement)

4 ☐ Interstate – I.C.C. certified common carrier (offering service to the general public, usually operating a regularly scheduled service between established terminals over a more or less regular route)

5 ☐ Intrastate – operating only within the State of registration (including local cartage, hauling between locations in the same town, city or suburb)

Item 6 – MAJOR USE OF THE TRUCK OR COMBINATION

How was the vehicle mostly used during the past 12 months? *Mark (X) ONE box*

If the vehicle was leased to someone else mark (X) ONE box that describes the business of the person or company to whom you leased the vehicle the longest time.

01 ☐ On farm or ranch, or other agricultural activity

02 ☐ In forestry or lumbering

03 ☐ In mining or quarrying

04 ☐ In construction – buildings, or roads

05 ☐ In manufacturing, refining, or processing

06 ☐ In wholesale trade

07 ☐ In retail trade

08 ☐ For hire transportation – mixed or general cargo

09 ☐ In utilities – telephone, electric, gas, etc.

10 ☐ In services – hotel, automobile repair, laundry, funeral services, advertising, plumbing, refuse collection, repair, etc.

11 ☐ Daily rental or short term lease without driver

12 ☐ For personal transportation – used in place of an automobile to go from home to work, for outdoor recreation (camping, etc.)

13 ☐ Other – *If none of the above applies to the use you made of the vehicle, describe the main use of the vehicle here.*

Item 7 – PRODUCTS CARRIED

a. Principal products carried during past 12 months

Mark (X) ONE box which indicates products usually carried by this vehicle

01 ☐ Farm products (crops and fruits, raw milk, etc.)

02 ☐ Live animals (horses, livestock, poultry or other animals)

03 ☐ Mining products

04 ☐ Logs and other forest products

05 ☐ Processed foods (dressed meat, beverages, dairy products, etc.) or tobacco

06 ☐ Textile mill products including apparel and leather goods, etc.

07 ☐ Building materials (lumber, millwork, sand, gravel, glass, concrete, etc.)

08 ☐ Household goods (moving)

09 ☐ Furniture or hardware (not including household goods moving)

10 ☐ Paper products, including printing and publishing products

11 ☐ Chemicals or related products, including drugs, paints, fertilizers, etc.

12 ☐ Petroleum or petroleum products

13 ☐ Primary metal products (ingots, billets, pipes, sheets, etc.)

14 ☐ Fabricated metal products (except machinery and transportation equipment)

15 ☐ Machinery, except electrical

16 ☐ Electrical machinery, equipment, and supplies, including household appliances

17 ☐ Transportation equipment (motor vehicles, trailers, boats, motorcycles, etc.)

18 ☐ Scrap, refuse, or garbage

19 ☐ Mixed cargoes

20 ☐ Craftsman's vehicle, such as plumbers, carpenters, "traveling workshops," etc.

21 ☐ Special equipment such as a crane, compressor, winch, drilling rigs, etc.

22 ☐ No products carried (personal transportation)

23 ☐ Other – *Describe*

b. Secondary product carried (If applicable) — Code No.

Of the list above, what would you consider to be the secondary product most carried by this vehicle? →

Item 8 – HAZARDOUS MATERIALS

a. Was this truck (or combination) used to haul hazardous materials during the past 12 months in quantities large enough to require a placard under the Code of Federal Regulations, Title 49, Transportation?

2 ☐ No – *SKIP to item 9*

1 ☐ Yes – *Continue with b*

b. Approximately what percent of the time was this unit used to haul hazardous materials?

1 ☐ Less than 25%

2 ☐ 25–49%

3 ☐ 50–74%

4 ☐ 75–100%

Section B – OPERATIONAL CHARACTERISTICS

Item 9 – BASE OF OPERATION

a. What was the principal place from which this vehicle was operated?

City or town	
County	State

b. What percentage of the miles traveled by this vehicle was within the State named in item 9a?

1 ☐ Below 25% 3 ☐ 50–74%

2 ☐ 25–49% 4 ☐ 75–100%

Item 10 – NUMBER OF TRUCKS, TRUCK-TRACTORS AND TRAILERS OPERATED FROM "BASE OF OPERATION"

How many trucks, truck-tractors and trailers are you operating from base named in item 9a?

	Total number
a. Pickups, vans (panels), multi-stops	
b. Other straight trucks	
c. Truck-tractors	
d. Trailers (semi- and full trailers)	

CONTINUE ON REVERSE SIDE

Figure 12. TIU Data Sheet

APPENDIX A—Continued

Section B – OPERATIONAL CHARACTERISTICS – Con.

Item 11 – AREA OF OPERATION

Where was this vehicle mostly operated?
Mark (X) ONE box only

1 ☐ Mostly in the local area (in or around the city and suburbs or within a short distance of the farm, factory, mine, or place vehicle is stationed)
2 ☐ Mostly over-the-road (beyond the local area) but not usually more than 200 miles one way to the most distant stop from the place vehicle is stationed
3 ☐ Mostly over-the-road trips that are usually more than 200 miles one way to the most distant stop from the place the vehicle is stationed
4 ☐ Mostly off-the-road operations as is usually associated with construction and farming operations

Item 12 – VEHICLE MILES AND MILES PER GALLON

ANNUAL MILES

a. What are the total miles this vehicle was driven during the past 12 months? *(If vehicle was idle for the year, enter "None." If owned less than 12 months, estimate probable miles for a year.)* ___ Miles

LIFETIME MILES

b. What are the total miles this vehicle has been driven since new? *(Give speedometer (odometer) reading or, if not indicated by speedometer, give your best estimate.)* ___ Miles

MILES PER GALLON

c. What was the average miles per gallon you received from this vehicle? ___ Miles per gallon

d. Is the figure entered in 12c above measured or estimated?
1 ☐ Measured from records 2 ☐ Estimated

Item 13 – MAINTENANCE

a. Was major maintenance (nonroutine) performed on the following equipment of this vehicle during the past 12 months?
1 ☐ Engine
2 ☐ Transmission
3 ☐ Braking system
4 ☐ Rear axle and differential
5 ☐ None of the above

b. By whom was this major maintenance performed –
1 ☐ Yourself or own repair shop (set up specifically for maintenance)?
2 ☐ Truck dealer?
3 ☐ Factory branch?
4 ☐ Leasing company?
5 ☐ Independent garage?
6 ☐ Other – *Describe*

Section C – PHYSICAL CHARACTERISTICS

Item 14 – GROSS WEIGHT

Mark (X) ONE box that is nearest the maximum gross weight in pounds (empty weight of vehicle plus carried load) at which this truck or combination was operated during the past 12 months. (If straight truck report GVW, if combination, report GCW.)

01 ☐ 6,000 or less	08 ☐ 33,001 to 40,000
02 ☐ 6,001 to 10,000	09 ☐ 40,001 to 50,000
03 ☐ 10,001 to 14,000	10 ☐ 50,001 to 60,000
04 ☐ 14,001 to 16,000	11 ☐ 60,001 to 80,000
05 ☐ 16,001 to 19,500	12 ☐ 80,001 to 100,000
06 ☐ 19,501 to 26,000	13 ☐ 100,001 to 130,000
07 ☐ 26,001 to 33,000	14 ☐ 130,001 and over

Item 15 – TYPE AND SIZE OF ENGINE

a. Type of engine
Mark (X) ONE box that describes the type of engine used in this vehicle.
1 ☐ Gasoline 2 ☐ Diesel 3 ☐ LPG or other

b. Size of engine
Mark (X) ONE box that describes the number of cylinders in the engine used in this vehicle.
1 ☐ Four 2 ☐ Six 3 ☐ Eight 4 ☐ Other

c. What is the displacement of the engine in cubic inches? → ___ Cubic inches

d. What is the horsepower rating of your engine? → ___ Horsepower

Item 16 – TYPE OF TRANSMISSION

Mark (X) ONE box that describes the type of transmission used in this vehicle.
1 ☐ Manual 2 ☐ Automatic 3 ☐ Semiautomatic

Item 17 – TYPE OF BRAKING SYSTEM

a. *Mark (X) ONE box that describes the type of braking system used in this vehicle*
1 ☐ Hydraulic 2 ☐ Air 3 ☐ Other

b. Does this system also include the new anti-wheel lock device?
1 ☐ Yes 2 ☐ No

Item 18 – POWER STEERING

Does this vehicle have power steering?
1 ☐ Yes 2 ☐ No

Item 19 – FUEL CONSERVATION EQUIPMENT

Does this vehicle have the following equipment?
Mark (X) ALL applicable items
1 ☐ Radial tires
2 ☐ Drag reduction device on top of cab
3 ☐ Variable speed fan (clutch type)
4 ☐ Fuel efficient engine RPM reduction, etc.
5 ☐ Axle or drive ratio change

Item 20 – AIR CONDITIONING

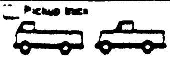
Is this vehicle air conditioned?
1 ☐ Yes 2 ☐ No

Section C – PHYSICAL CHARACTERISTICS – Con.

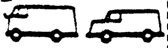
Item 21 – TYPE AND SIZE OF BODY
Indicate both body type and body size

BODY TYPE (a)	LENGTH OF LOAD SPACE OR CAPACITY (b)
Mark (X) ONE box to describe the type of the truck or combination. If the power unit is a truck-tractor, report body type of the combination most frequently used with the power unit.	*Mark (X) ONE box to indicate length of load space or capacity. If two or more trailing units, mark (X) for combined length or capacity.*

01 ☐ Pickup truck



02 ☐ Panel truck or van



03 ☐ Multi-stop or walk-in

Length of load space (Feet)
01 ☐ Less than 7 feet
02 ☐ 7 and less than 10

a. Does this pickup, panel, multi-stop or walk-in truck have 4-wheel drive?
1 ☐ Yes 2 ☐ No

b. What is the number of axles on vehicle?
1 ☐ Two 2 ☐ Three

c. Is this pickup, panel, multi-stop or walk-in truck equipped with a . . .
1 ☐ Slide in camper?
2 ☐ Pickup shell cover?
3 ☐ Camper body?
4 ☐ None of above
Mark (X) Col. (b) and SKIP to item 28

03 ☐ 10 and less than 13
04 ☐ 13 and less than 16
05 ☐ 16 and less than 20
06 ☐ 20 and less than 28
07 ☐ 28 and less than 36
08 ☐ 36 and less than 41

04 ☐ Platform with added devices such as spreaders, dumpers, lifts, etc.
05 ☐ Low boy or depressed center
06 ☐ Other platform – including grain flatbed, high bed, stake
07 ☐ Cattle rack (hogs, calves, and other livestock)
08 ☐ Insulated non-refrigerated van
09 ☐ Insulated refrigerated van
10 ☐ Furniture van
11 ☐ Open top van
12 ☐ All other enclosed vans
13 ☐ Beverage
14 ☐ Utility (body equipped for mobile repair and service, e.g., telephone line truck, electric utility, etc.)
15 ☐ Winch or crane other than wrecker (including roll on, roll off)
16 ☐ Wrecker
17 ☐ Pole or logging
18 ☐ Auto transport
19 ☐ Boat transport
20 ☐ Mobile home pullers

01 ☐ Less than 7 feet
02 ☐ 7 and less than 10
03 ☐ 10 and less than 13
04 ☐ 13 and less than 16
05 ☐ 16 and less than 20
06 ☐ 20 and less than 28
07 ☐ 28 and less than 36
08 ☐ 36 and less than 41
09 ☐ 41 and less than 45
10 ☐ 45 and less than 73
11 ☐ 73 or more

30 ☐ Garbage or refuse hauler
31 ☐ Front loader
32 ☐ Rear loader packer
33 ☐ Roll off

Capacity size (Cubic yds.)
31 ☐ Less than 20
32 ☐ 20 to 25
33 ☐ 26 and over

40 ☐ Dump truck or combination

Capacity of dump (water level without side boards) (Cubic yards)
41 ☐ Under 5
42 ☐ 5 to 6.9
43 ☐ 7 to 9.9
44 ☐ 10 to 11.9
45 ☐ 12 to 14.9
46 ☐ 15 to 17.9
47 ☐ 18 to 19.9
48 ☐ 20 to 29.9
49 ☐ 30 or more

50 ☐ Tank truck or combination (for liquids)

Liquid capacity of tank (Gallons)
51 ☐ Less than 1,000
52 ☐ 1,000 to 1,999
53 ☐ 2,000 to 2,999
54 ☐ 3,000 to 3,999
55 ☐ 4,000 to 5,999
56 ☐ 6,000 to 7,999
57 ☐ 8,000 to 11,999
58 ☐ 12,000 or more

60 ☐ Tank truck or combination (for dry bulk)

Dry bulk capacity (Cubic feet)
61 ☐ Less than 300
62 ☐ 300 to 599
63 ☐ 600 to 899
64 ☐ 900 to 1,199
65 ☐ 1,200 to 1,499
66 ☐ 1,500 or more

70 ☐ Concrete mixer
71 ☐ Front discharger
72 ☐ Rear discharger

Capacity of mixer (Cubic yards)
71 ☐ Less than 6
72 ☐ 6 to 6.9
73 ☐ 7 to 7.9
74 ☐ 8 to 8.9
75 ☐ 9 to 9.5
76 ☐ 10 to 10.9
77 ☐ 11 to 11.9
78 ☐ 12 or over

80 ☐ Other body types – *If the above descriptions do not satisfactorily describe your vehicle, enter identifying body type and size of capacity.*

Section C – PHYSICAL CHARACTERISTICS – Con.

Item 22 – POWERED AXLES

How many driving (powered) axles does this vehicle have?
Report powered tandem drives as two axles.
1 ☐ One 2 ☐ Two 3 ☐ Three 4 ☐ Four

Item 23 – VEHICLE TYPE

Mark (X) ONE box which best describes your vehicle

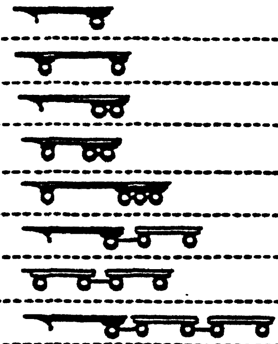
Single unit truck
1 ☐ Two axle
2 ☐ Three axle
3 ☐ Other
} *SKIP to item 25*

Truck tractor
4 ☐ Two axle
5 ☐ Three axle
6 ☐ Other
} *Continue with item 24*

Item 24 – AXLE ARRANGEMENT OF TRAILER UNITS

Mark (X) ONE box that illustrates the axle arrangement of the trailer unit most frequently used with the power unit.

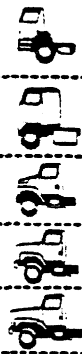
1 ☐
2 ☐
3 ☐
4 ☐
5 ☐
6 ☐
7 ☐
8 ☐



9 ☐ If none of the above applies, please indicate total number of axles and trailing units → | Total axles ___ | Total trailing units ___ |

Item 25 – CAB TYPE

a. *Mark (X) ONE box that illustrates the cab type of the power unit.*
1 ☐ Cab forward of engine
2 ☐ Cab over engine
3 ☐ Short hood conventional
4 ☐ Medium hood conventional
5 ☐ Long hood conventional
6 ☐ Other – *Describe*



b. Is this cab equipped with a sleeper unit?
1 ☐ Yes 2 ☐ No

Remarks

Item 26 – PERSON TO CONTACT REGARDING THIS REPORT

Name

Address *(number and street, city, State, ZIP code)*

Telephone → | Area code | Number | Extension |

Fleet number of vehicle

Item 27 – CERTIFICATION

This report is substantially accurate and has been prepared in accordance with instructions.

Signature

Title | Date

Figure 12. (Cont.)

The TIU data have been acquired and stored in computer files [17], thereby making it practical to examine these data for over 96,000 trucks. Example results for the horsepower-to-weight ratio (hp/wt) across weight classes for all trucks indicate that heavy trucks have much lower hp/wt ratios than small vehicles (see Figure 13).* For vehicles in the 60 to 80 thousand pound range, the average horsepower is 282 with an estimated standard deviation of 51 hp. Based on these results, the average weight-to-horsepower ratio for heavy trucks is 248 lbs/hp and, for vehicles that are one standard deviation less powerful than average, the wt/hp ratio is 303 lbs/hp.

These results from the TIU survey compare reasonably well with the findings of [15] (see Table 1 presented previously)—particularly the results of [15] that were obtained from acceleration performance on the level. The entries in Table 1 that were derived from sustained speed information on upgrades of from 2 to 7% appear to indicate very low levels of wt/hp. Perhaps, it is difficult to determine whether a vehicle has reached sustained speed, or, possibly, there happened to be a number of unloaded vehicles operating on these grades, thereby influencing the average results. In any event, both of these data sources [15 and 16] indicate that roadway designs based on a 300 lb/hp heavy truck would be conservative compared to the performance of an average heavy truck.

Currently, the AASHTO "Blue Book," <u>A Policy on Geometric Design of Rural Highways</u>, is being revised and review drafts of the new book have been assembled (see Reference [18]). With respect to acceleration performance, the work of Hayhoe and Grundmann [19] is used in [18] to update the work of Huff and Scrivner [1]. In [19] speed-distance curves for a representative heavy truck are developed. These curves are based on numerical calculations for a 300 lb/hp vehicle entering various levels of grade at 55 mph. The calculations performed in [19] employ representative parametric values describing a typical vehicle to compute performance curves. An

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*Horsepower-to-weight ratio is presented in Figure 13 because horsepower is a continuous quantity but maximum gross weight (during the last 12 months) is recorded in certain ranges rather than continuously (see Fig. 12, item 14 - Gross Weight). Also, hp/wt is directly proportional to acceleration performance rather than inversely proportional as wt/hp is.

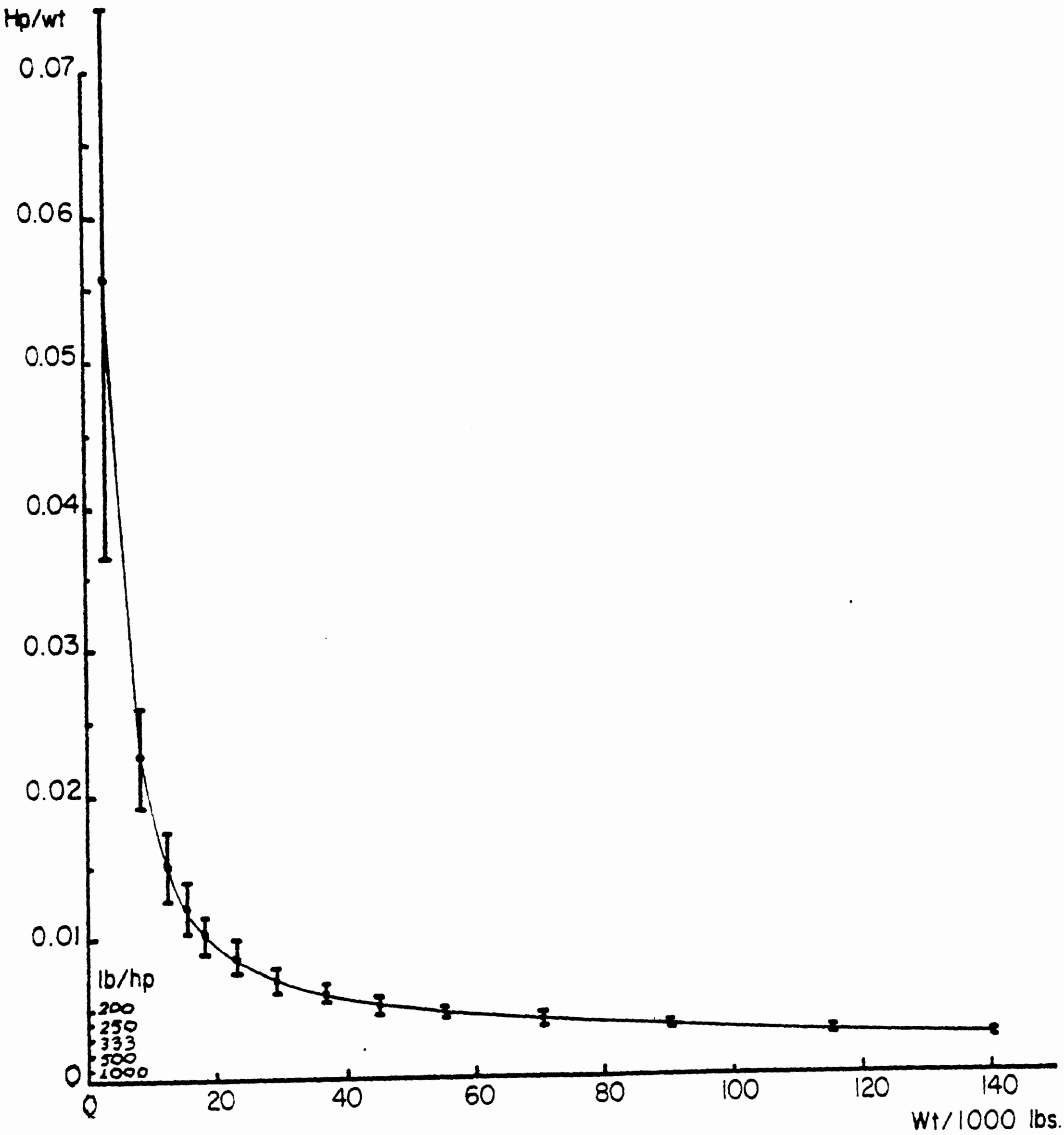

Figure 13. Hp/Wt across weight classes for all trucks.

initial speed of 55 mph is used because current trucks typically achieve this speed on level grades. The choice of a 300 lb/hp vehicle is based on (1) examining the trend of wt/hp estimates (see Figure 14) and (2) private communications with an engine manufacturer and a large trucking organization. Given that the state of the art in predicting acceleration performance has advanced to the point where accurate predictions can be reliably made, the procedures used in [19] should produce satisfactory results, if suitable and appropriate parametric values are used in the calculations.

The average results from the TIU survey [16] have been superimposed in Figure 14 by taking the inverse of the hp/wt ratio curve presented in Figure 13. Clearly, all the curves presented in Figure 14 indicate the continued trend towards lower wt/hp ratios, particularly for heavy vehicles.

Based on this review of the state of the art, the following factors have an important influence on the acceleration performance of heavy trucks:

	hp/wt,	net horsepower-to-weight ratio
	W,	gross weight
	W_e,	equivalent weight including rotating components
	h,	elevation
	F_r,	rolling resistance of the tires
	F_a,	aerodynamic drag
	θ,	grade
	η,	driveline efficiency (chassis friction factor)
	t_s,	the shifting time of the driver
and	Δrpm,	the engine speed change resulting from shifting from one gear to another

In addition, the torque-speed characteristic of the engine (torque rise), the radius of the tires, and the number of gears and their ratios are included in the detailed calculations. To meet the needs of the highway engineer, the above information is combined to produce acceleration versus velocity relationships for design vehicles operating on various magnitudes of grade. The acceleration versus velocity information is used to compute (through numerical integration) speed-distance graphs that the highway engineer can readily employ in developing or evaluating the geometric design of up- and downgrade sections of roadway.

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