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An Analysis of Speed Changes for Large Transport Trucks

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By means of a force and momentum balance a method was devised for calculating the speed vs distance history of large trucks traversing various types of vertical highway curves at wide-open throttle. The equations that resulted were solved with the aid of electronic computing machines over the following ranges of values of vehicle and highway properties:

Vehicle wt = 30,000 to 72,000 lb;
$\frac{\text{Wt}}{\text{Horsepower}}$ = 200 to 400;
Vehicle speed = 10 to 50 mph;
Highway grade = -8 to + 8 percent;
Vertical curve radius = 2,500 to 20,000 ft; and
Uniform grades.

The results of the calculations are presented as charts relating vehicle speed to distance along the vertical highway curve. A comparison of calculated and experimental values showed satisfactory agreement. Various possible methods of utilizing the charts for highway design or vehicle selection purposes are discussed.

• OF ALL VEHICLES operating on our highways the large transport trucks have the lowest engine power relative to their weight. Hence these vehicles are generally the slowest on upgrades and require the longest distances to accelerate. Realistic design of highway grades and acceleration lanes should be based on the performance of these particular vehicles inasmuch as all others can perform better. This study was undertaken to develop a means of calculating the speed and acceleration properties of large transport trucks on various grades and acceleration lanes.

NOMENCLATURE

The following nomenclature is used:

S = slope of highway, radians;
G = grade of highway, percent;
G_o = grade at the start of a section of highway, percent;
L = horizontal distance along a highway section, ft;
r = radius of curvature of a highway in a vertical plane, ft;
t = time, sec;
t_s = average time required to shift between gears, sec;

g	= gravitational constant, ft per sec per sec;
F_O	= net initial force acting on a truck at the start of a highway section, lb;
F_R	= total rolling resistance force of a truck, lb (includes both tire resistance and air resistance);
F_T	= thrust force on a truck due to engine torque, lb;
v	= truck speed, ft per sec;
v_O	= truck speed at entry to a section of highway, ft per sec;
v_{max}	= maximum truck speed attainable in a particular gear setting as limited by maximum useable engine rpm, ft per sec;
mph	= truck speed, mph; per hour;
(GVW)	= gross vehicle weight, lb;
(BHPW)	= horsepower delivered by the engine to the clutch at wide open throttle
NE	= engine rpm;
NEW	= engine rpm at which BHPW was measured;
R_1	= main transmission gear ratio $= \frac{\text{Engine RPM}}{\text{Transmission output shaft RPM}}$;
R_2	= auxiliary transmission gear ratio $= \frac{\text{Transmission output shaft RPM}}{\text{Auxiliary transmission output shaft RPM}}$;
R_3	= rear axle gear ratio $= \frac{\text{Auxiliary transmission output shaft RPM}}{\text{Drive wheel RPM}}$; and
TF	= tire factor* $= \frac{\text{Drive wheel RPM}}{\text{Truck speed, mph}}$.

FORCE AND MOMENTUM BALANCE

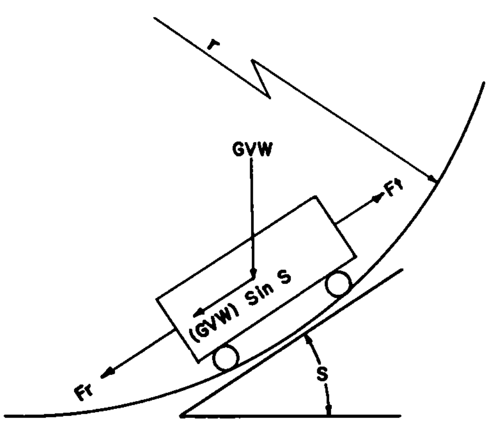

Figure 1. Force diagram of a vehicle on a general section of highway.

Figure 1 shows a vehicle on a general section of highway of slope S and radius of vertical curvature r together with the several forces acting on the vehicle. In general the engine thrust force, F_T, will not always equal the sum of the rolling resistance force, F_R, and the vehicle weight force, (GVW) sin S, and the vehicle will either accelerate if F_T exceeds this sum or decelerate if F_T is less than this sum. From Newton's law the acceleration may be calculated:

$$\frac{(GVW)(dv)}{g\,(dt)} = F_T - F_R - (GVW) \sin S \qquad (1)$$

A solution to this equation provides the desired relation between truck performance, highway geometry, and truck properties.

*For values of TF for various tire sizes, see (1).

The engine thrust force, F_T, must be determined for (a) clutch engaged and engine at wide open throttle, and (b) clutch disengaged while shifting gears. Engine operation at part throttle is not considered here because this means that driver choice, rather than highway geometry, is determining the vehicle performance which is thus indeterminate. Hence the results presented herein are the maximum attainable speed characteristics of a vehicle. A skillful driver will generally achieve these maximum attainable speeds except as limited by traffic congestion, legal speed restrictions, or safe-operating speed limitations.

With the clutch disengaged; F_T is clearly zero. At wide open throttle, F_T may be calculated when the following are known:

1. The horsepower delivered by the engine to the clutch at wide open throttle, BHPW, and the engine RPM at this power output, NEW.
2. The operating gear ratios of the main transmission, R_1, the auxiliary transmission, R_2, and the rear axle, R_3.
3. The drive wheel tire size and tire factor, TF.

Road test methods for measuring the BHPW and NEW of a vehicle are described by Sawhill and Firey (2). The gear ratios and tire size are known for a vehicle or can usually be obtained from the manufacturer.

The value of F_T can then be calculated from the following equation:

$$F_T = \frac{(BHPW)}{(NEW)} (R_1)(R_2)(R_3)(TF)(375) \tag{2a}$$

Implicit in the use of this equation to calculate F_T is the assumption that engine torque at wide-open throttle is constant over the operating speed range of the engine. Though not strictly correct, this assumption is a reasonable approximation for most unsupercharged commercial truck engines.

The vehicle rolling resistance force, F_R, may be calculated by means of the following equation from (2):

$$F_R = \frac{GVW}{148.5} + 195 \tag{2b}$$

This total rolling resistance force equation was based on coasting tests of several large transport trucks and, as discussed in (2), is subject to the limitations of these experimental data. For significant upgrades generally the precision of Eq. 2 for F_R is not too important because the major resistance to vehicle motion is then the vehicle weight force, (GVW) sin S.

On the general section of highway shown in Figure 1 the slope, S, of the highway may vary with the distance, L, along the grade; and thus, S may be a function of time, t. For circular vertical highway curves the simple approximate relation, shown on Figure 2, exists between curve radius, r, distance, L, and slope, S.

$$S = S_O + \frac{L}{r} = \sin S \tag{3}$$

The approximation that sin S equals S is accurate to within 0.2 percent for highway grades of 10 percent or less. The sign convention for Eq. 3 is as follows:

1. Distance, L, is positive in the direction of vehicle motion.
2. For concave upward curves (i.e., sag curves), r is positive and S increases with L.

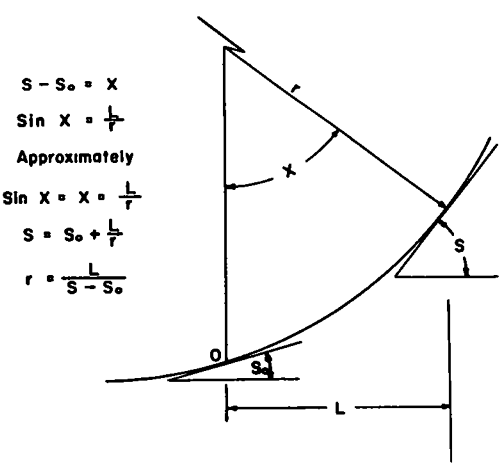

Figure 2. Relation between slope and distance for circular vertical highway curves.

3. For convex upward curves (i.e., summit curves), r is negative and S decreases with L.

4. Slope, S, is positive for upgrades and negative for downgrades.

Real vertical highway curves are not simple circles but can be adequately approximated as such without introducing appreciable error for highways of less than about 10 percent maximum grade. To determine the equivalent radius of a real vertical curve only the entering and leaving slopes and the horizontal distance between them need be known, as shown in Figure 2. The following equations are approximately correct for small angles.

$$r = \frac{L_o}{S_1 - S_2} \tag{4}$$

In highway design, slope is most commonly expressed as percent grade, G, rather than in radians. The following equations are expressed in terms of G:

$$G = 100S \tag{5a}$$

$$G = G_o + \frac{100L}{r} \tag{5b}$$

$$r = \frac{100L_o}{G_1 - G_2} \tag{5c}$$

$$(GVW)\sin S = (GVW)\,S = \frac{(GVW)G_o}{100} + \frac{(GVW)\,L}{r} \tag{5d}$$

Distance, L, along the highway is related to vehicle velocity, v, and time, t, as follows:

$$v = \frac{dL}{dt} \quad \frac{dv}{dt} = \frac{d^2L}{dt^2} \tag{6}$$

The force and momentum balance equation for a vehicle operating on a general section of highway with the clutch engaged and the throttle wide open now becomes:

$$\frac{GVW(d^2L)}{g\ (dt^2)} = \frac{(BHPW)}{(NEW)}(R_1)(R_2)(R_3)(TF)(375) - \frac{GVW}{148.5} - 195 - \frac{(GVW)G_o}{100} - \frac{(GVW)L}{r} \tag{7}$$

With the clutch disengaged during gear shifting the equation becomes

$$\frac{GVW(d^2L)}{g\ (dt^2)} = -\frac{GVW}{148.5} - 195 - \frac{(GVW)G_o}{100} - \frac{(GVW)L}{r} \tag{8}$$

These differential equations provide a mathematical relation between distance and time inasmuch as all other factors are constant for a particular vehicle in a particular gear and on a particular vertical curve. The vehicle may be presumed to enter the section of highway at initial velocity v_o when L = O and t = 0. Under these conditions the solution of Eq. 5, 6, 7, and 8 takes the forms of

$$L = \frac{(F_o r)}{(GVW)} - \frac{(F_o r)}{(GVW)}\cos\sqrt{\frac{g}{r}}\,t + \frac{v_o}{\sqrt{g/r}}\sin\sqrt{\frac{g}{r}}\,t \tag{9}$$

$$V = V_o \text{Cos} - \sqrt{\frac{g}{r}}\,t + \frac{(F_o r)}{(GVW)}\sqrt{\frac{g}{r}}\sin\sqrt{\frac{g}{r}}\,t \tag{10}$$

The net initial force, F_o, acting on the vehicle at the start of the highway section is defined by the following relations. At wide open throttle:

$$F_o = \frac{(BHPW)}{(NEW)}(R_1)(R_2)(R_3)(TF)(375) - \frac{GVW}{148.5} - 195 - \frac{(GVW)G_O}{100} \tag{11}$$

With the clutch disengaged:

$$F_o = -\frac{GVW}{148.5} - 195 - \frac{(GVW)G_O}{100} \tag{12}$$

METHOD OF CALCULATION

The preceding vehicle motion relations are not handy to use for highway design inasmuch as many different motion equations apply even to a single section of highway. These many equations result from the wide variation in truck design and operation. To put the force and momentum balance equations into a potentially useful form, certain assumptions were found necessary.

On many vertical highway curves, the driver of a large transport truck will change gears several times. With each gear change the value of F_O at wide-open throttle changes and the equation of vehicle motion is altered. On a general section of highway the value of G_O may also change for each gear setting. Thus, on a vertical curve requiring five gear changes, ten different vehicle motion equations may apply to a particular vehicle.

Available to the purchaser of a large transport truck are a wide selection of main transmissions, auxiliary transmissions, and rear axles, and an even wider selection of gear ratios and number of gear settings. The result has been that there is no such thing as "typical" or "standard" gearing in these trucks, each being geared for the needs of the purchaser at the time of ordering. Hence for a selected vertical highway curve, each truck will have its own set of vehicle motion equations.

The weight of a particular truck may vary between the empty weight and the maximum legal weight depending on the type of cargo being carried and the availability of cargo for particular sections of highway.

The total number of equations to be considered by the highway designer has a minimum value equal to the sum, for all vertical curves on the highway, of the product of the number of large transport trucks using the highway and about twice the number of gear changes required. With so many equations, useful calculations become tedious.

To reduce the complexity introduced by these variations the following assumptions were made:

1. Only those trucks were considered whose properties lay within the following ranges:

$$\frac{GVW}{BHPW} = 200 \text{ to } 400;$$

$GVW = 30{,}000$ to $70{,}000$ lb; and

mph $= 10$ to 50.

2. The ratio of minimum to maximum useable engine rpm was assumed to be 0.80.
3. Only those sections of highway were considered whose geometry lay within the following ranges:

$G = -8$ to $+8$ percent; and

$r = 2{,}500$ to $20{,}000$ ft or r = infinity.

The vehicle properties limitations do not appear serious because the majority of large transport trucks are included except when empty or very lightly loaded. In this latter condition these trucks are not usually a highway design limitation. Although it would be desirable to include speeds up to 60 or 70 mph, the rolling resistance data available did not extend beyond 50 mph. In general, large transport trucks reach speeds in excess of 50 mph only on nearly level or downhill sections of highway where the principal force opposing vehicle motion is the rolling resistance force, F_R. Hence, to obtain useful results in this higher speed range, reliable data on F_R above 50 mph are required.

The useable engine speed ratio of 0.80 was selected as the average value for the several vehicles described in (1) for which this ratio varied from 0.67 to 0.86.

The highway geometry limitations do not appear serious because modern highways generally lie well within the indicated ranges.

The equation for the engine thrust force, F_T, was modified to eliminate the specific vehicle gear ratios and tire factor:

$$v = \frac{1,465(NE)}{R_1 R_2 R_3 (TF)} \tag{13a}$$

$$v_{max} = \frac{1,465\ (NEW)}{R_1 R_2 R_3 (TF)} = \text{maximum vehicle speed attainable in the gear setting, } R_1,\ R_2,\ R_3. \tag{13b}$$

$$F_T = \frac{(BHPW)}{(v_{max})} 550 \tag{13c}$$

The detailed steps to calculate the velocity vs distance history of a selected vehicle on a selected vertical curve were as follows:

1. Specific values of the following were assumed; GVW, $\frac{GVW}{BHPW}$, initial G_o, initial v_o, r.

2. On vertical curves causing the vehicle to slow down (deceleration curves) v_{max} equals v_o. On vertical curves causing the vehicle to speed up (acceleration curves) v_{max} was assumed equal to $(v_o/0.8)$.

3. The vehicle motion equations for wide-open throttle were solved for values of distance, L, and velocity, v, as a function of time, t, from which v could be determined as a function of L.
4. On deceleration curves the first gear shift was assumed to occur when v reached the value 0.8 v_o. On acceleration curves the first gear shift was assumed to occur when v reached the value $v_o/0.8$.
5. The truck was assumed to follow the vehicle motion equations for clutch disengaged during the gear shift time interval, t_s. An average measured value for t_s of 2 sec was obtained from (1). The values of v_o and G_o for the gear shift interval were the terminal values for the preceding wide-open throttle period (step 4).
6. Steps 2 and 3 were repeated using the vehicle motion equations for clutch disengaged for the time interval t_s.
7. For the second wide-open throttle period steps 2, 3, and 4 were repeated with values of v_o and G_o obtained from the terminal condition of the preceding clutch disengaged period (step 6).
8. The foregoing calculations were repeated through each gear until v reached the value of 10 mph on deceleration curves or the value of 50 mph on acceleration curves, or the end of the curve (G = ± 8 percent) was reached.
9. For the special case of uniform grades (r = infinity) the vehicle motion equations become

$$L = v_o t + \frac{F_o g}{GVW} \frac{t^2}{2} \tag{14a}$$

$$v = v_o + \frac{F_o g}{GVW} t \tag{14b}$$

10. For uniform grades steps 1 through 8 were carried out with the additional limit of reaching maximum sustained speed, v_{max}. Maximum sustained speeds occur only on upgrades when the net initial force acting on the vehicle, F_o becomes zero. Thereafter the vehicle velocity cannot change on a uniform grade.

As the calculations progressed it was found possible to make a further simplification by using only an average value of vehicle weight, GVW = 50,000 lb. For a particular value of vehicle weight to horsepower ratio, $\frac{GVW}{BHPW}$, the value of $\frac{F_0}{GVW}$ was found to vary only slightly with GVW within the range of GVW of 30,000 to 70,000 lb. As an indication of the magnitude of error introduced by this assumption maximum sustained speeds on grades in excess of 2 percent were within less than 6 percent of the approximate value within the preceding vehicle weight range.

An actual graph of vehicle velocity vs distance along an upgrade will have steps due to the shifting of gears, as shown in Figure 3. In these calculations the assumed usable engine speed ratio of 0.80 determined the locations of the shift points which may not correspond to any real vehicle. Accordingly only the smoothed curves, rather than the stepped curves, were used.

The detailed calculations were carried out with the aid of an IBM 610 computing machine which greatly reduced the tedium of the work.

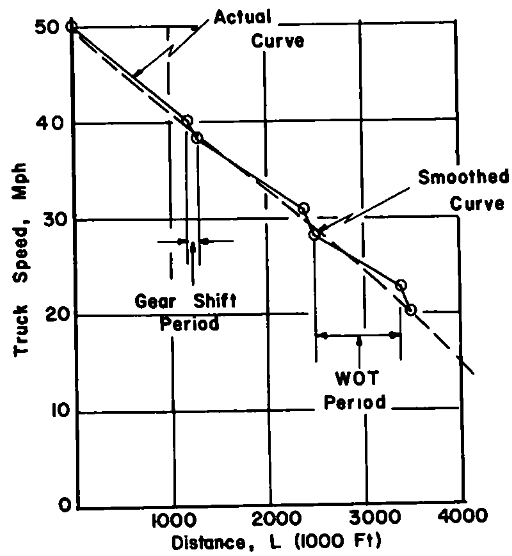

Figure 3. Example of truck speed vs distance history on an upgrade.

RESULTS AND COMPARISON WITH EXPERIMENT

The results of the foregoing calculations are shown in chart form in Figures 4 through 28 as follows:

1. Figures 4 through 6 inclusive show vehicle velocity vs distance on uniform upgrades for values of $\frac{GVW}{BHPW}$ of 400, 300, and 200 for an entering velocity of 50 mph.
2. Figures 7 through 9 inclusive show vehicle velocity vs distance on uniform downgrades for values of $\frac{GVW}{BHPW}$ of 400, 300, and 200 for an entering velocity of 10 mph.

For uniform grades when the entering velocity is other than 10 or 50 mph, the curves of items 1 or 2 are used by shifting the zero distance point to the actual entering velocity.

3. Figures 10 through 12 inclusive show vehicle velocity vs distance on circular vertical sag curves for values of $\frac{GVW}{BHPW}$ of 400, 300, and 200 for entering velocities of 30, 40, and 50 mph, and for curve radii of 2,500, 5,000, 10,000, 15,000, and 20,000 ft.
4. Figures 13 through 15 inclusive show vehicle velocity vs distance on circular vertical summit curves of radii, 2,500, 5,000, 10,000, 15,000 and 20,000 ft for a value of $\frac{GVW}{BHPW}$ of 400, and for entering velocities of 10, 20, 30, 40, and 50 mph.
5. Figures 16 through 18 inclusive are identical to item 4 except the value of $\frac{GVW}{BHPW}$ is 300.
6. Figures 19 through 21 inclusive are identical to item 4 except the value of $\frac{GVW}{BHPW}$ is 200.
7. Figures 22 through 24 inclusive show vehicle travel time vs distance on uniform upgrades for values of $\frac{GVW}{BHPW}$ of 400, 300, and 200 for an entering velocity of 50 mph.

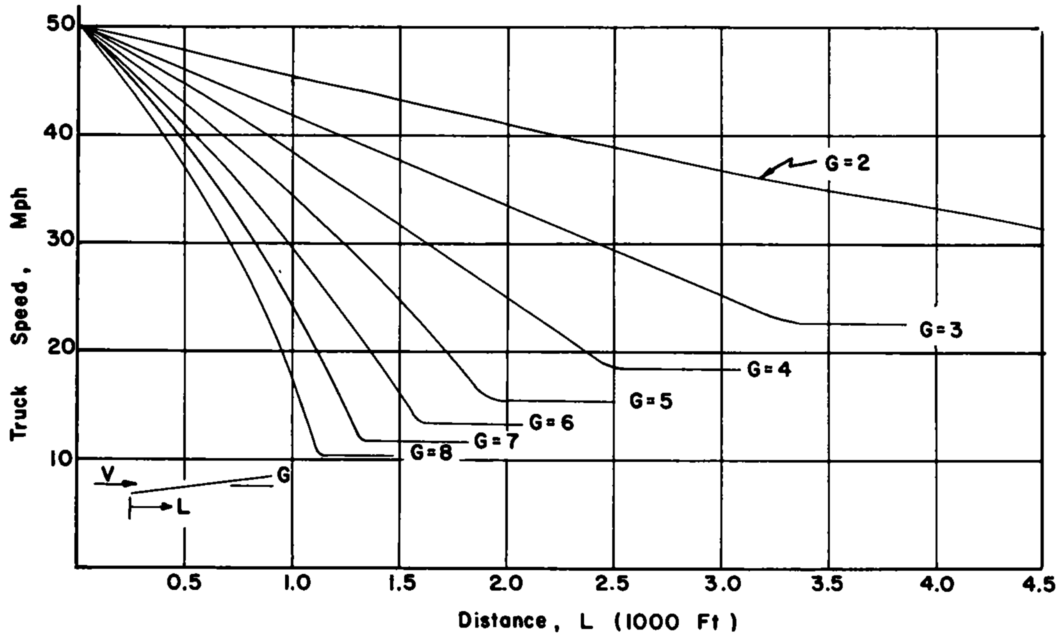

Figure 4. Velocity vs distance chart on uniform upgrades for trucks with GVW/BHPW = 400 and an entering speed of 50 mph.

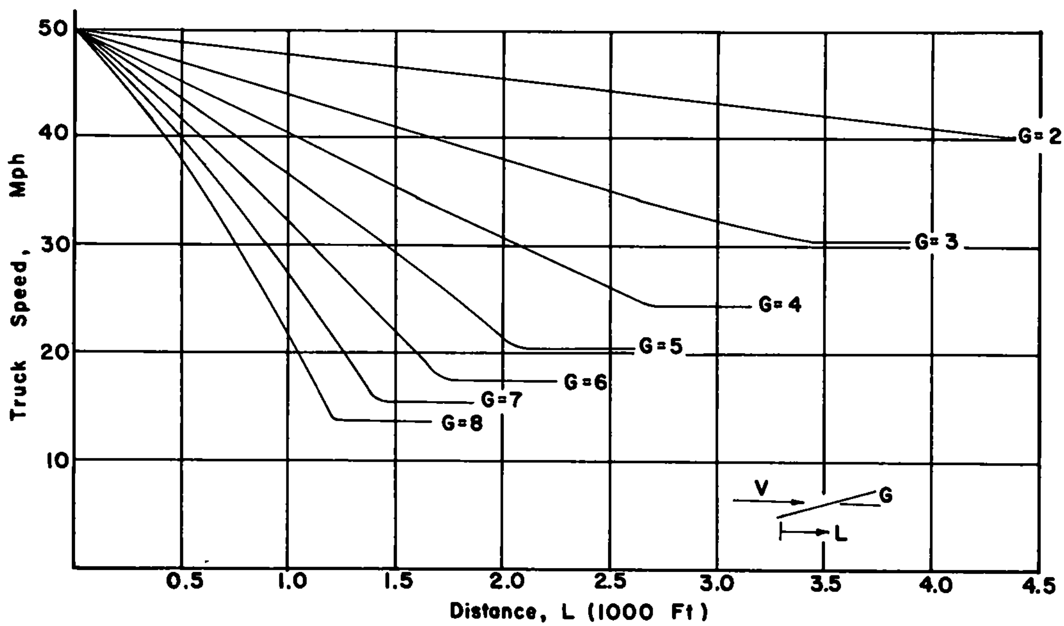

Figure 5. Velocity vs distance chart on uniform upgrades for trucks with GVW/BHPW = 300 and an entering speed of 50 mph.

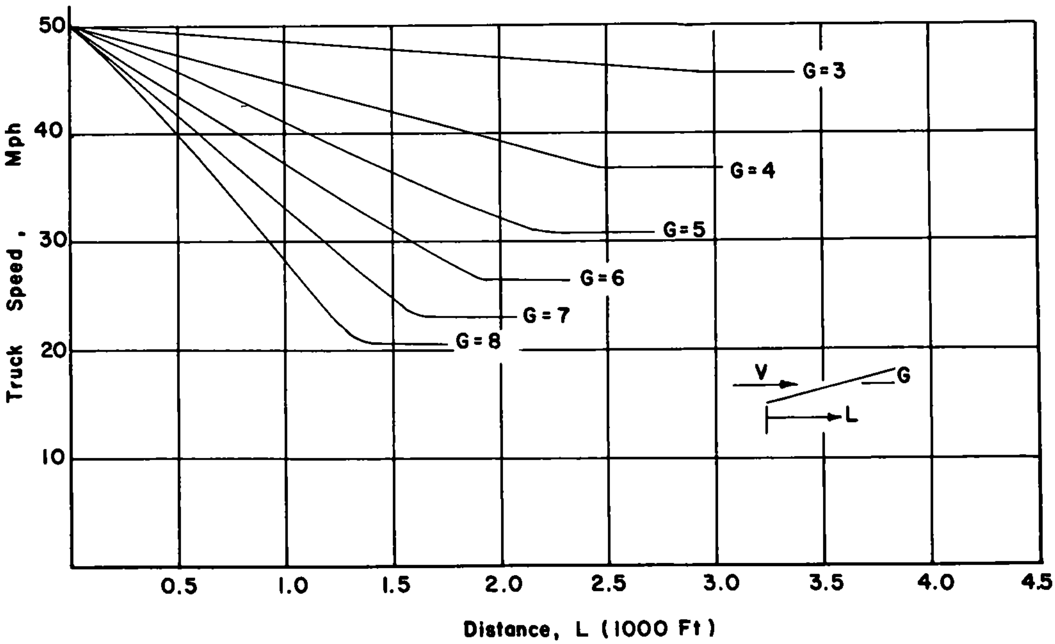

Figure 6. Velocity vs distance chart on uniform upgrades for trucks with GVW/BHPW = 200 and an entering speed of 50 mph.

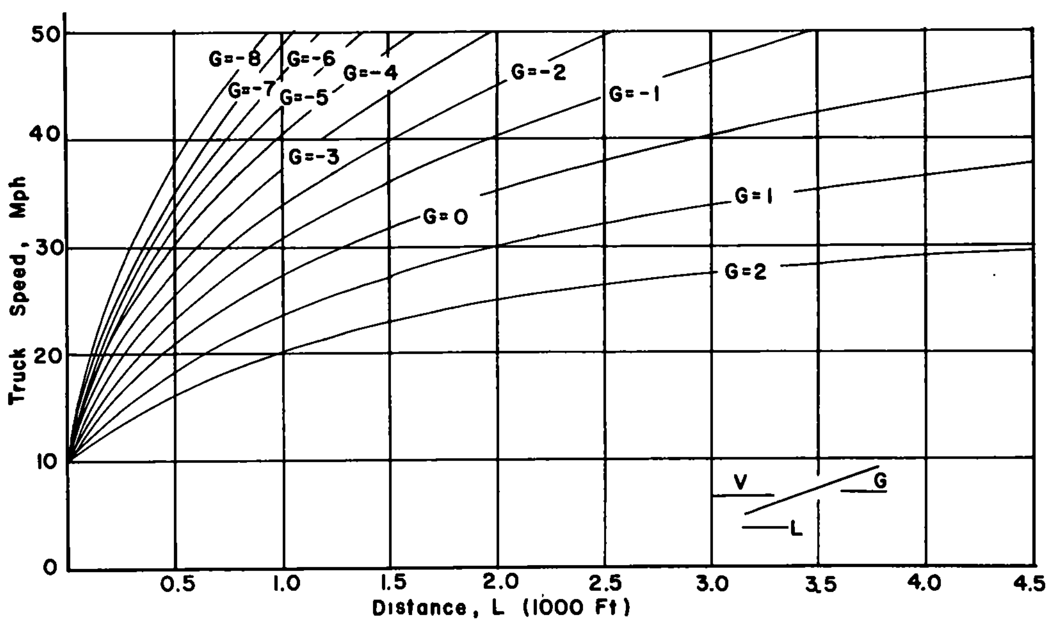

Figure 7. Velocity vs distance chart on uniform downgrades for trucks with GVW/BHPW = 400 and an entering speed of 10 mph.

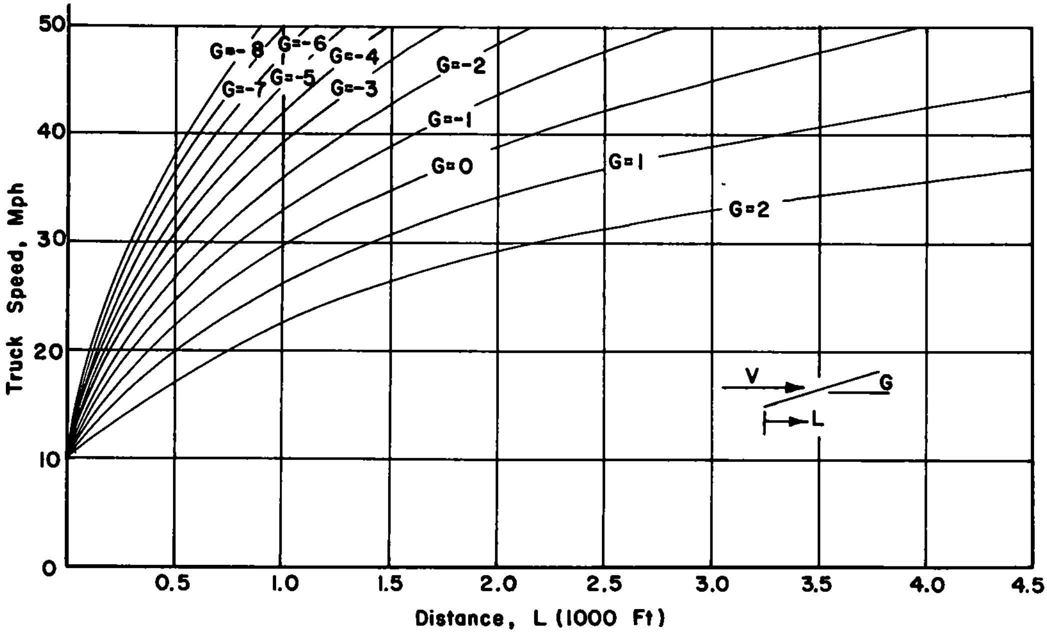

Figure 8. Velocity vs distance chart on uniform downgrades for trucks with GVW/BHPW = 300 and an entering speed of 10 mph.

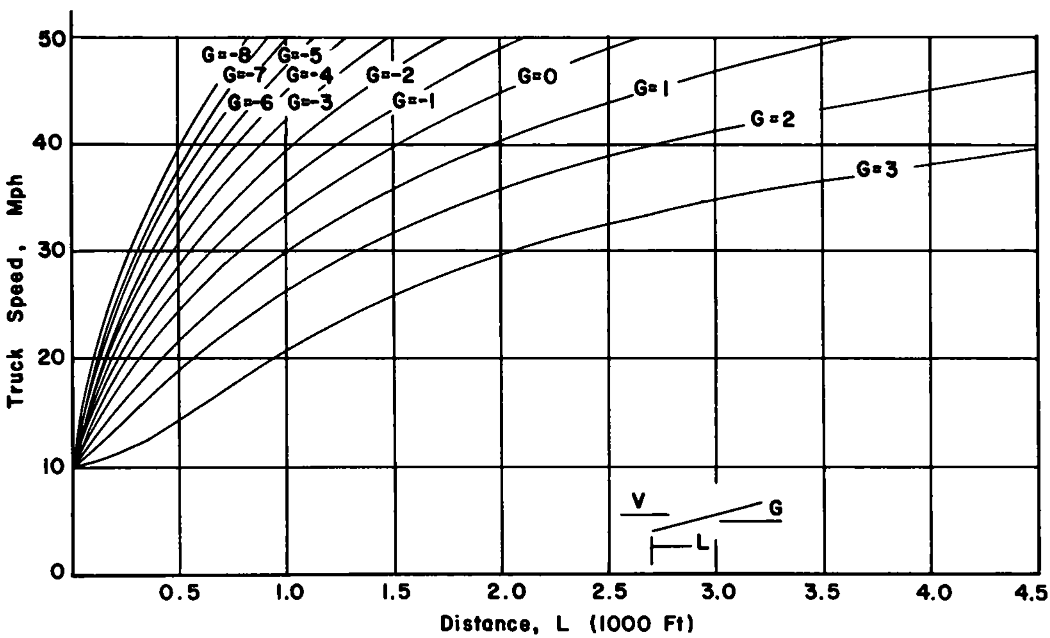

Figure 9. Velocity vs distance chart on uniform downgrades for trucks with GVW/BHPW = 200 and an entering speed of 10 mph.

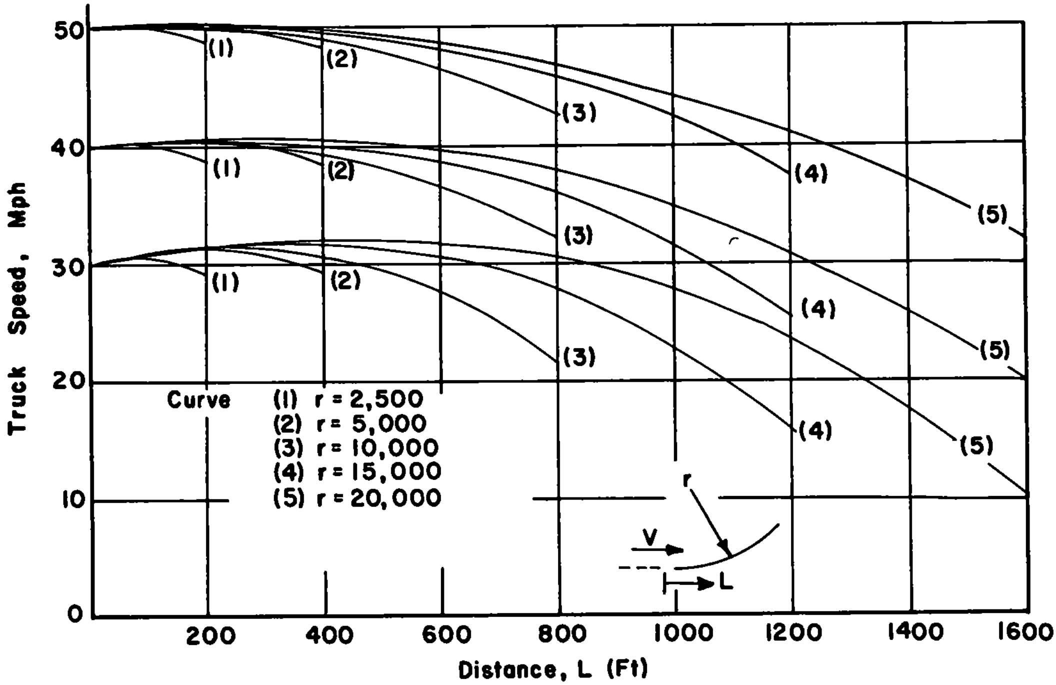

Figure 10. Velocity vs distance chart on vertical sag curves for trucks with GVW/BHPW = 400.

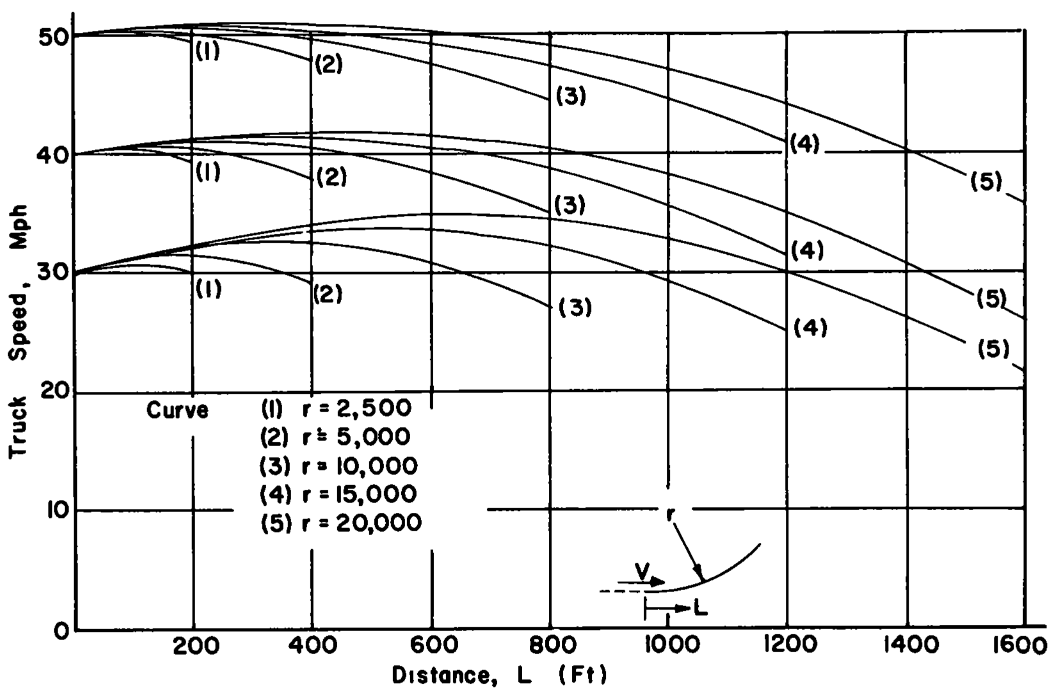

Figure 11. Velocity vs distance chart on vertical sag curves for trucks with GVW/BHPW = 300.

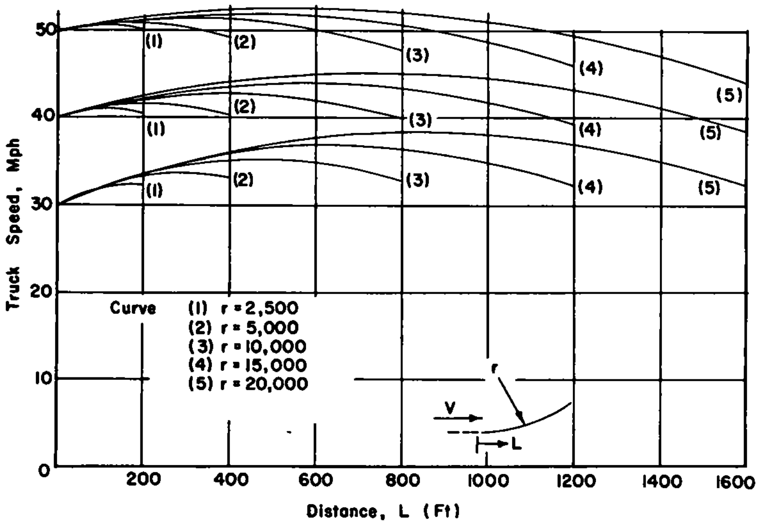

Figure 12. Velocity vs distance chart on vertical sag curves for trucks with GVW/BHPW = 200.

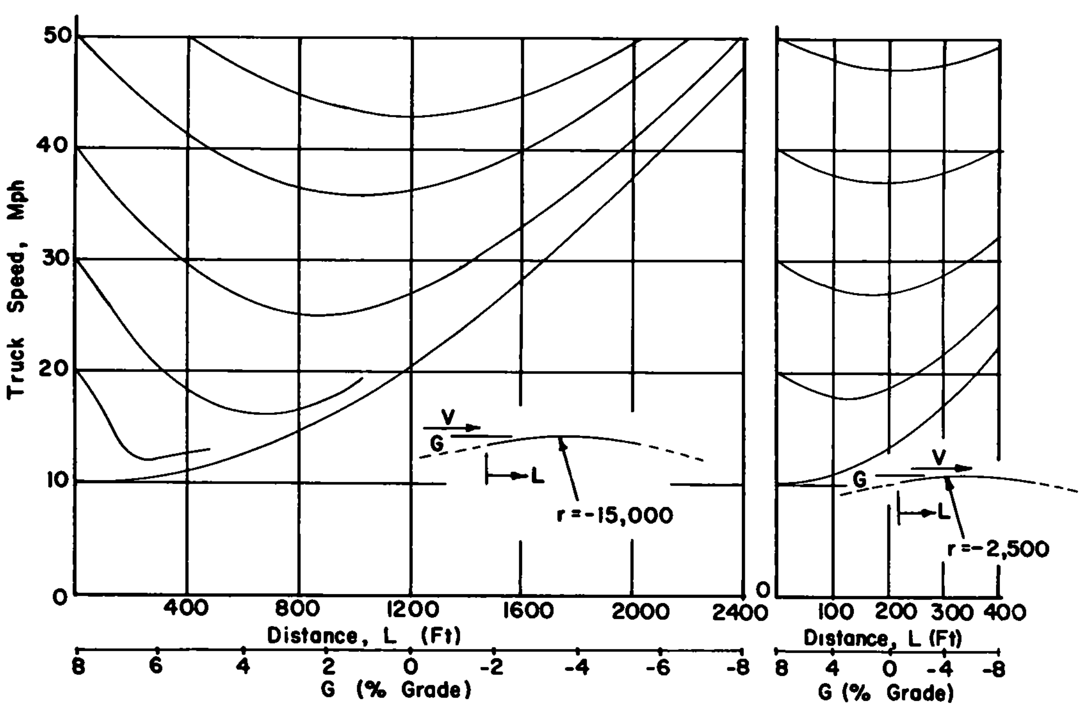

Figure 13. Velocity vs distance chart on vertical summit curves for trucks with GVW/BHPW = 400 and curve radii of 2,500 and 15,000 ft.

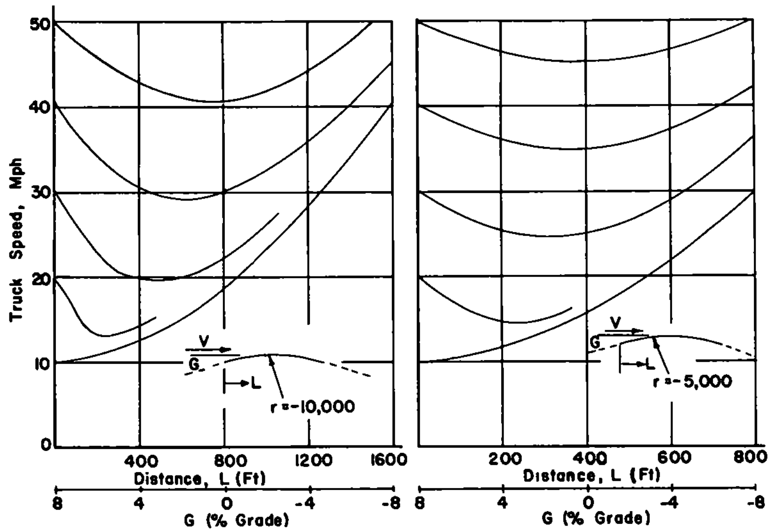

Figure 14. Velocity vs distance chart on vertical summit curves for trucks with GVW/BHPW = 400 and curve radii of 5,000 and 10,000 ft.

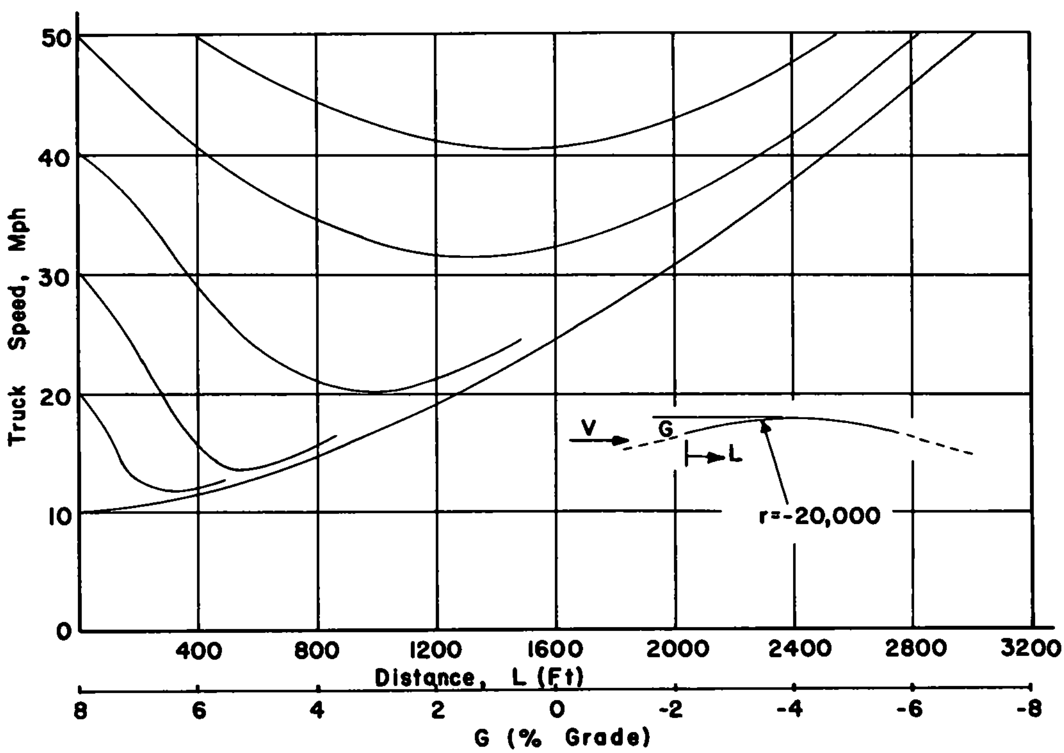

Figure 15. Velocity vs distance chart on vertical summit curves for trucks with GVW/BHPW = 400 and curve radius of 20,000 ft.

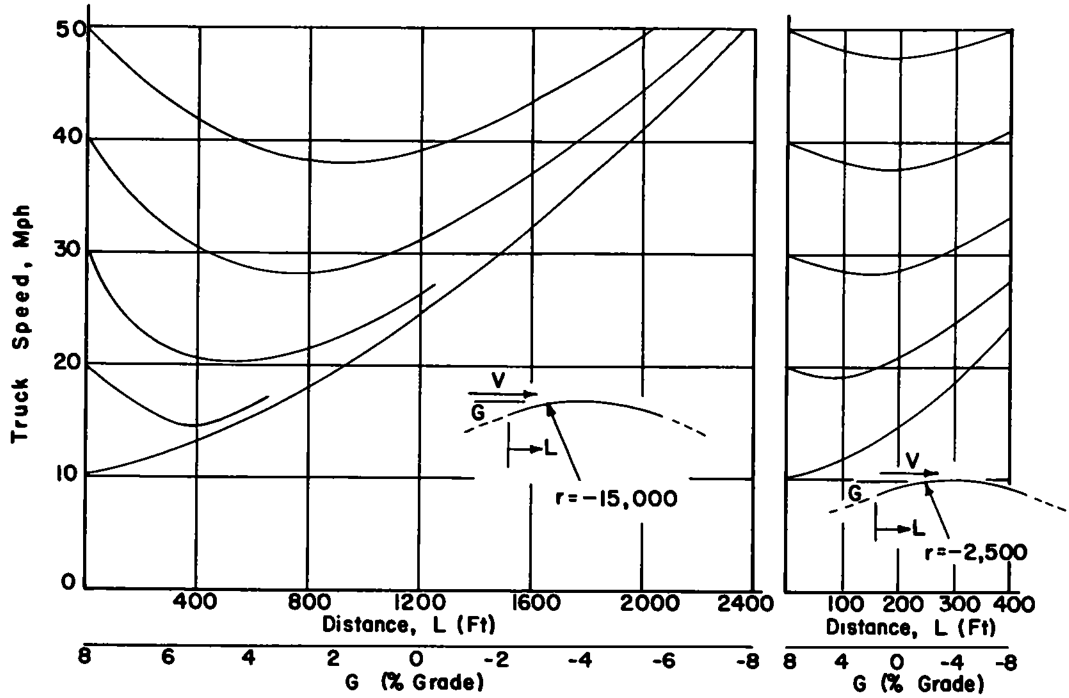

Figure 16. Velocity vs distance chart on vertical summit curves for trucks with GVW/BHPW = 300 and curve radii of 2,500 and 15,000 ft.

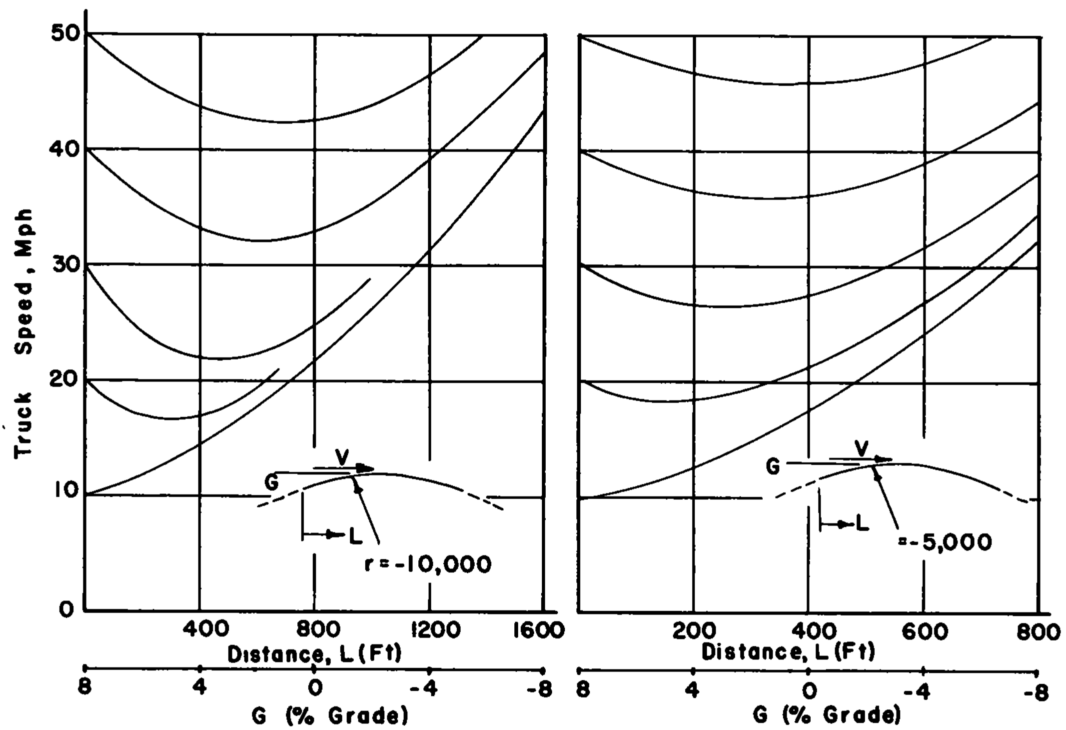

Figure 17. Velocity vs distance chart on vertical summit curves for trucks with GVW/BHPW = 300 and curve radii of 5,000 and 10,000 ft.

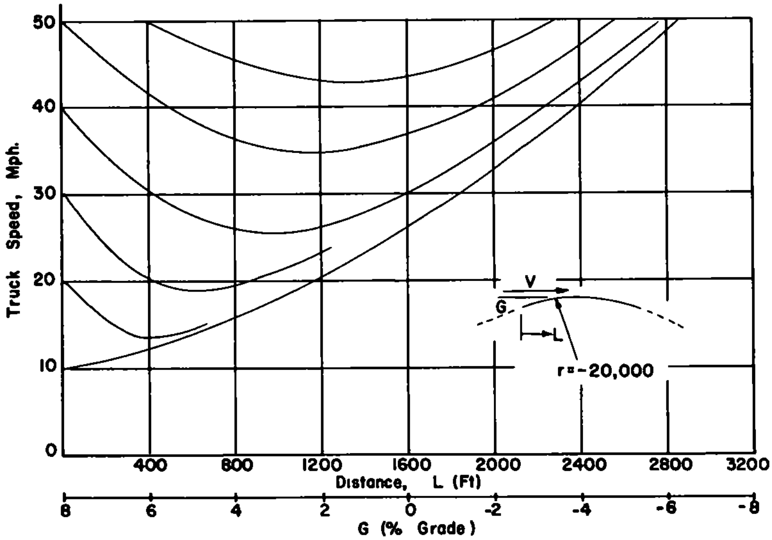

Figure 18. Velocity vs distance chart on vertical summit curves for trucks with GVW/BHPW = 300 and a curve radius of 20,000 ft.

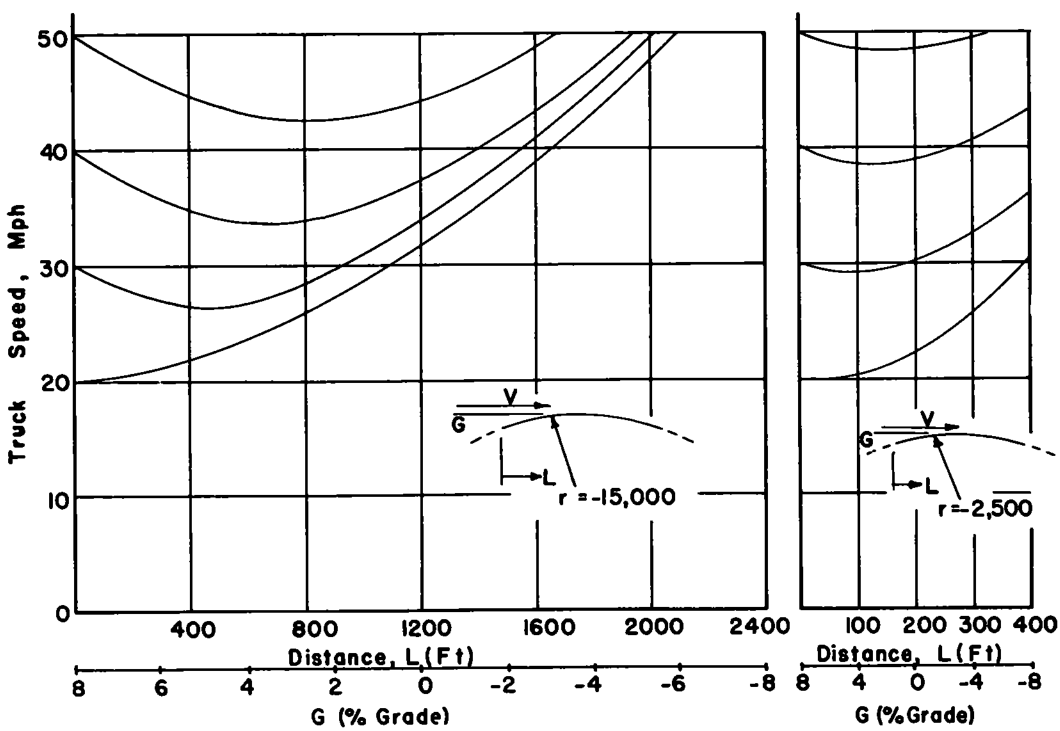

Figure 19. Velocity vs distance chart on vertical summit curves for trucks with GVW/BHPW = 200 and curve radii of 2,500 and 15,000 ft.

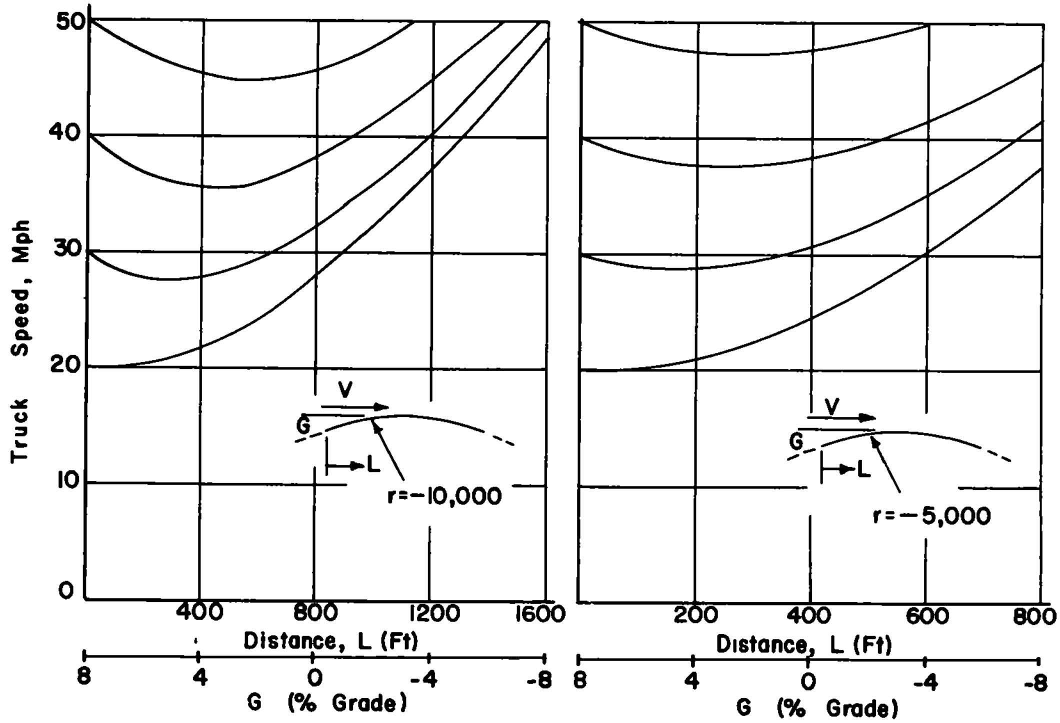

Figure 20. Velocity vs distance chart on vertical summit curves for trucks with GVW/BHPW = 200 and curve radii of 5,000 and 10,000 ft.

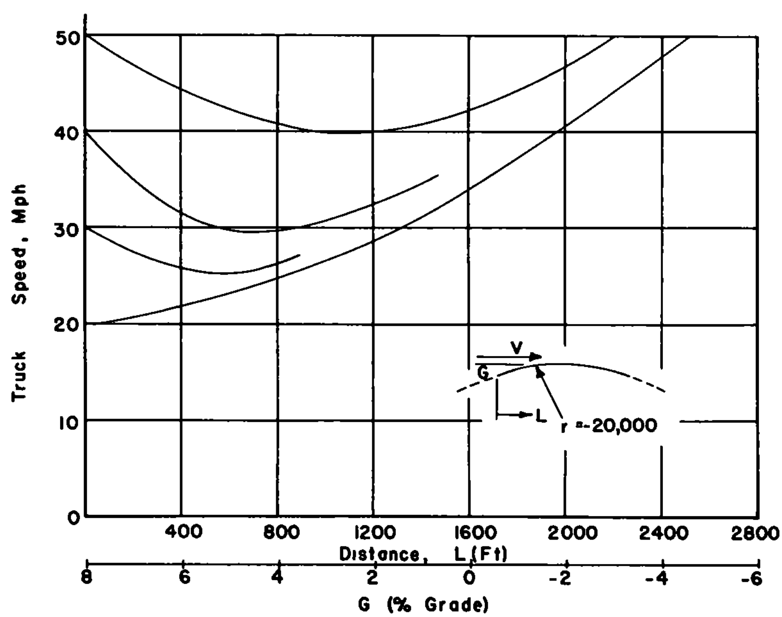

Figure 21. Velocity vs distance chart on vertical summit curves for trucks with GVW/BHPW = 200 and a curve radius of 20,000 ft.

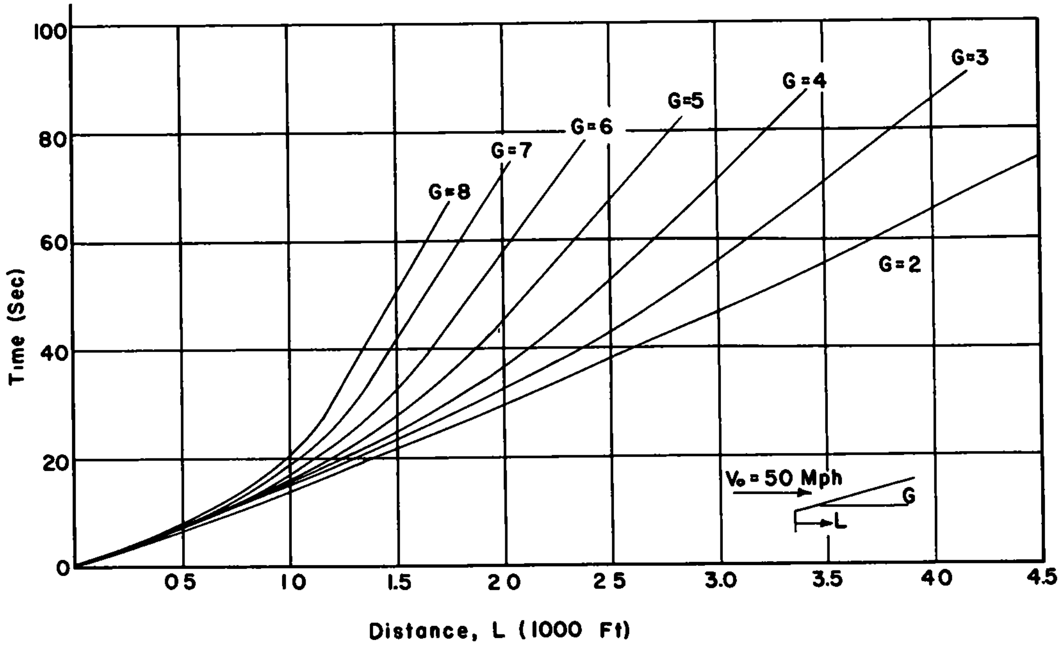

Figure 22. Travel time vs distance chart on uniform upgrades for trucks with GVW/BHPW = 400.

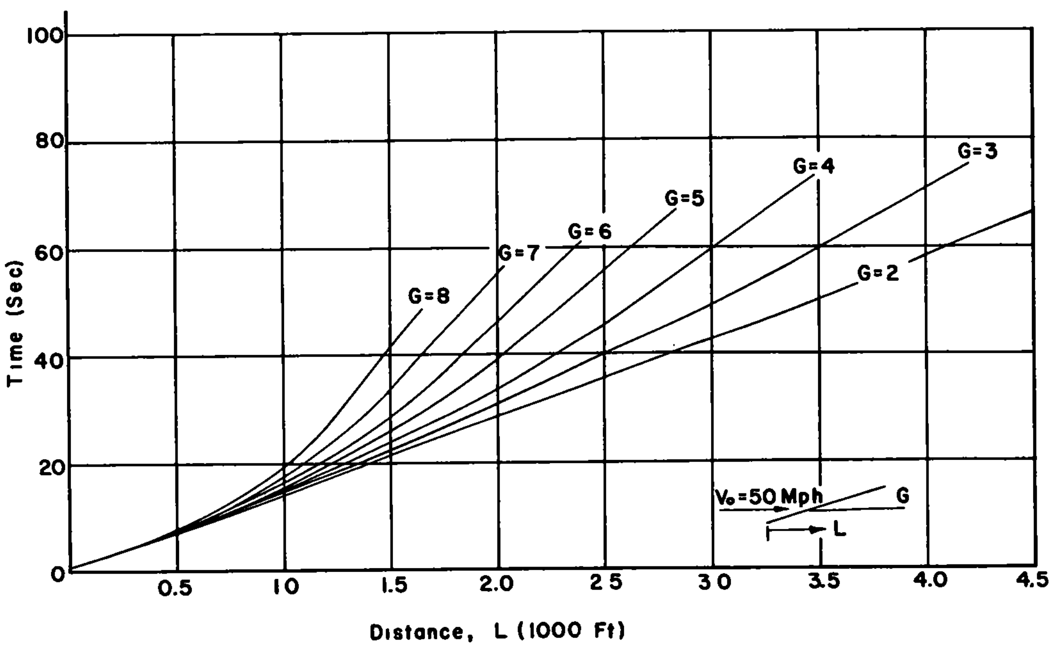

Figure 23. Travel time vs distance chart on uniform upgrades for trucks with GVW/BHPW = 300.

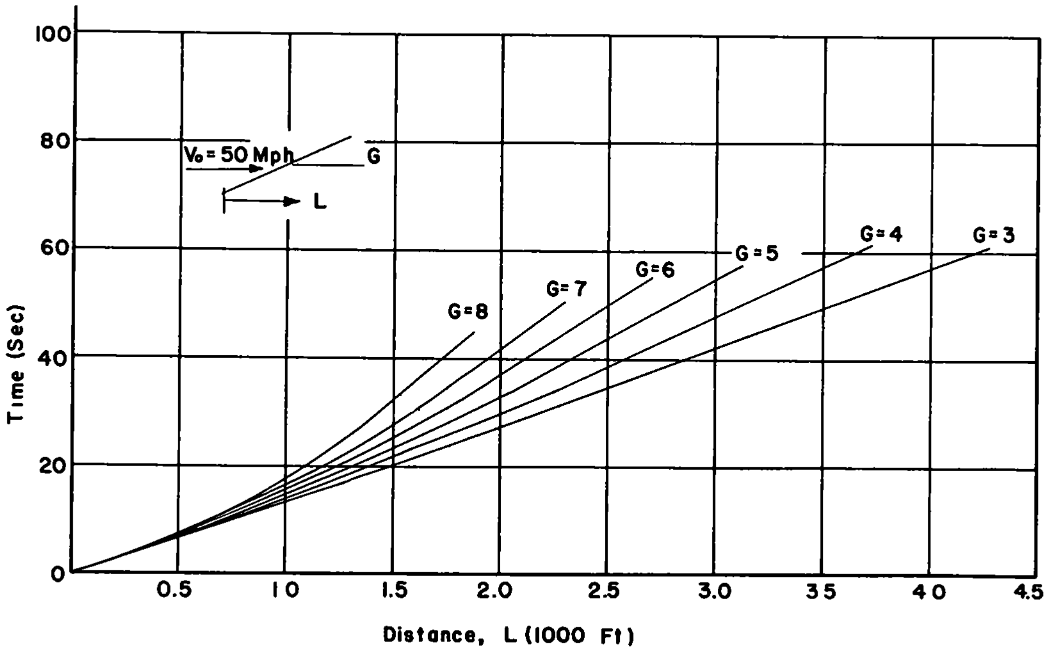

Figure 24. Travel time vs distance chart on uniform upgrades for trucks with GVW/BHPW = 200.

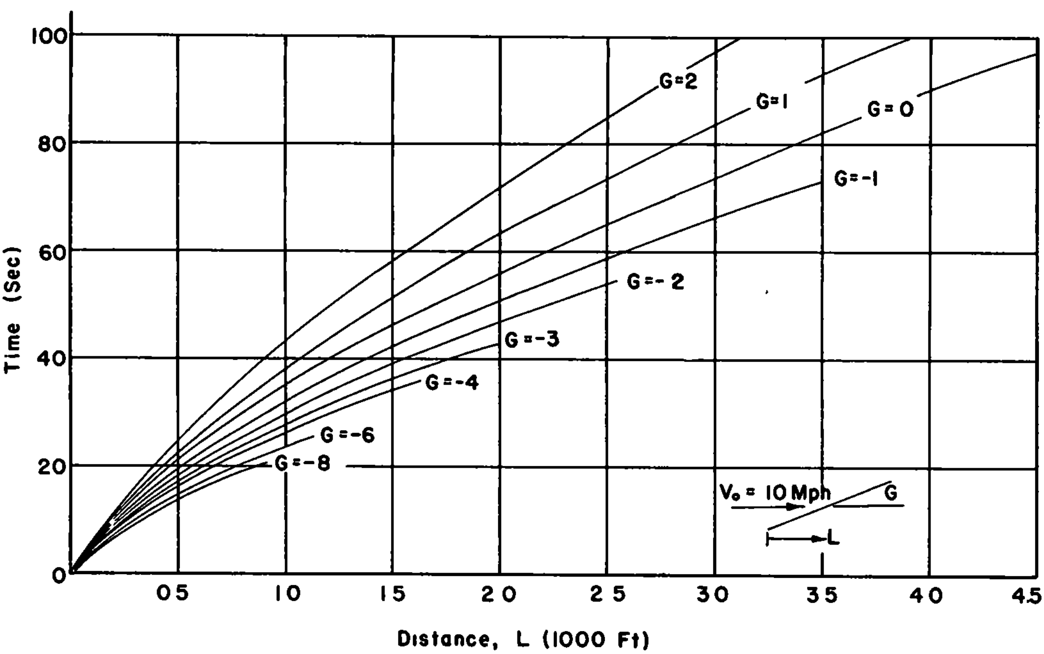

Figure 25. Travel time vs distance chart on uniform downgrades for trucks with GVW/BHPW = 400.

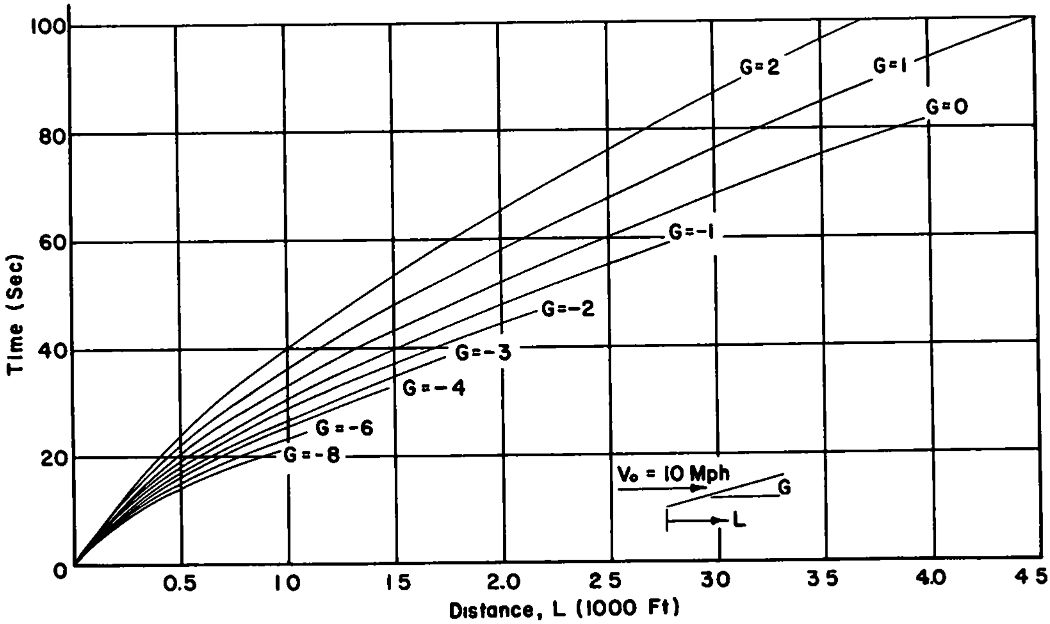

Figure 26. Travel time vs distance chart on uniform downgrades for trucks with GVW/BHPW = 300.

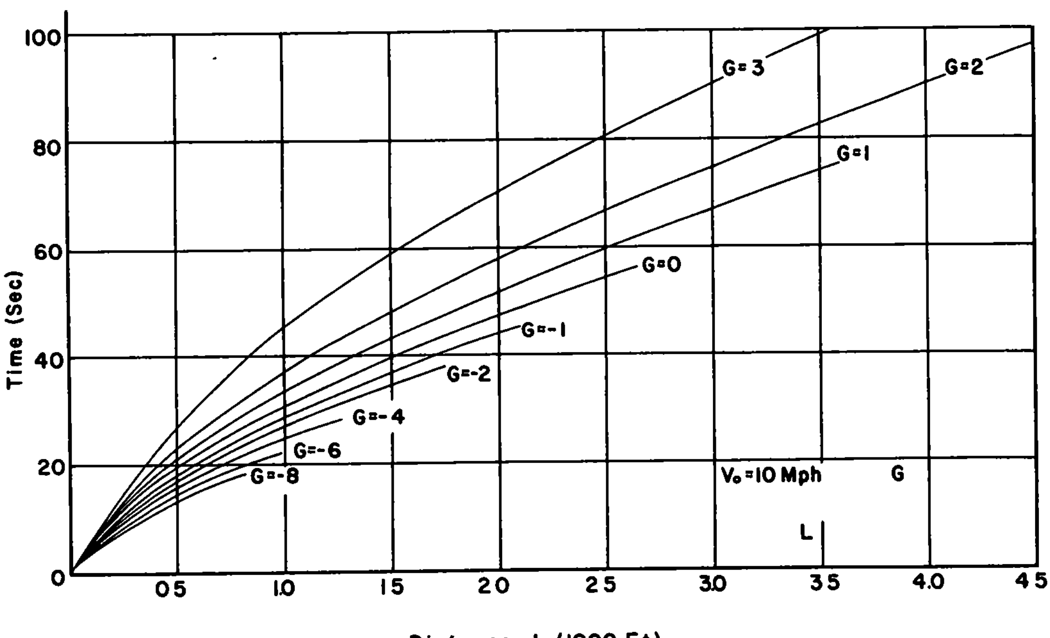

Figure 27. Travel time vs distance chart on uniform downgrades for trucks with GVW/BHPW = 200.

8. Figures 25 through 27 inclusive show vehicle travel time vs distance on uniform downgrades for values of $\frac{GVW}{BHPW}$ of 400, 300, and 200 for an entering velocity of 10 mph.

For uniform grades when the entering velocity is other than 10 or 50 mph the curves of items 7 and 8 are used by shifting the zero distance and time point to the distance for the actual entering velocity as obtained from the curves of items 1 or 2.

9. Figure 28 shows theoretical maximum sustained speed of a vehicle vs highway grade for vàlues of $\frac{GVW}{BHPW}$ or 400, 300, and 200.

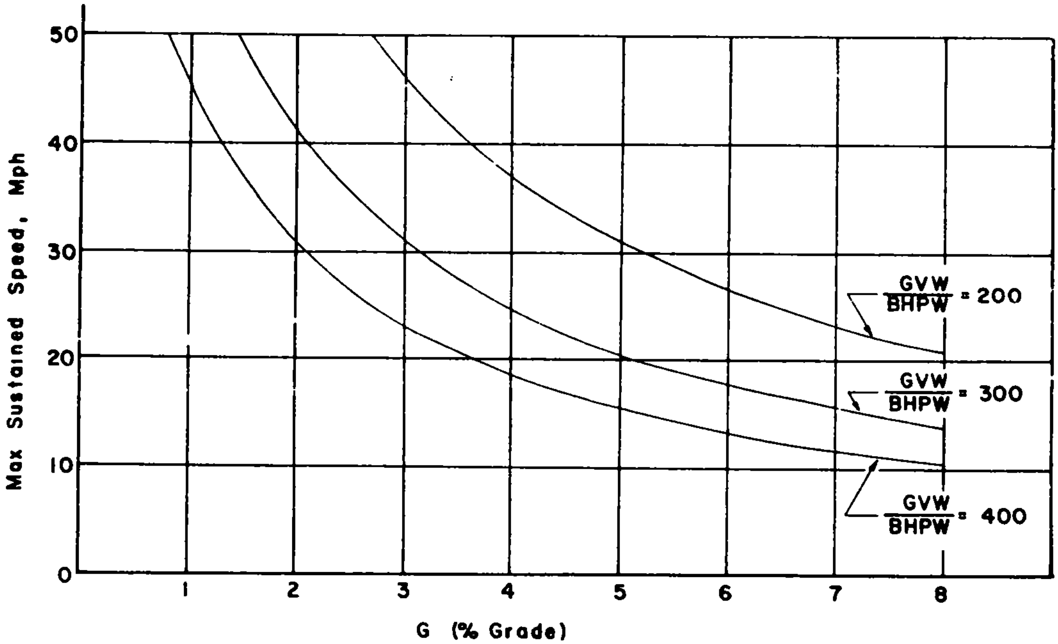

Figure 28. Effect of uniform grade on theoretical maximum sustained speeds of trucks.

To investigate whether the preceding calculated results bear any relation to the actual behavior of real trucks on real highways, several trucks were run over vertical highway curves and the speed vs time and distance history recorded. The properties of the truck were measured by the methods described in (1). The geometry of the vertical highway curve was obtained from surveys of the Washington State Highway Department. A comparison was then made between these measured results and the corresponding calculated results. An example of such a comparison is shown in Figure 29 for a sag curve and in Figure 30 for a summit curve. Comparisons of this type were carried out within the following ranges of values of vehicle and highway properties:

$\frac{GVW}{BHPW}$	= 200 to 400;
GVW	= 27,000 to 66,000 lb;
mph	= 10 to 50;
G	= + 6 to -6 percent;
r	= 15,000 to 18,333 ft; and
r	= infinity.

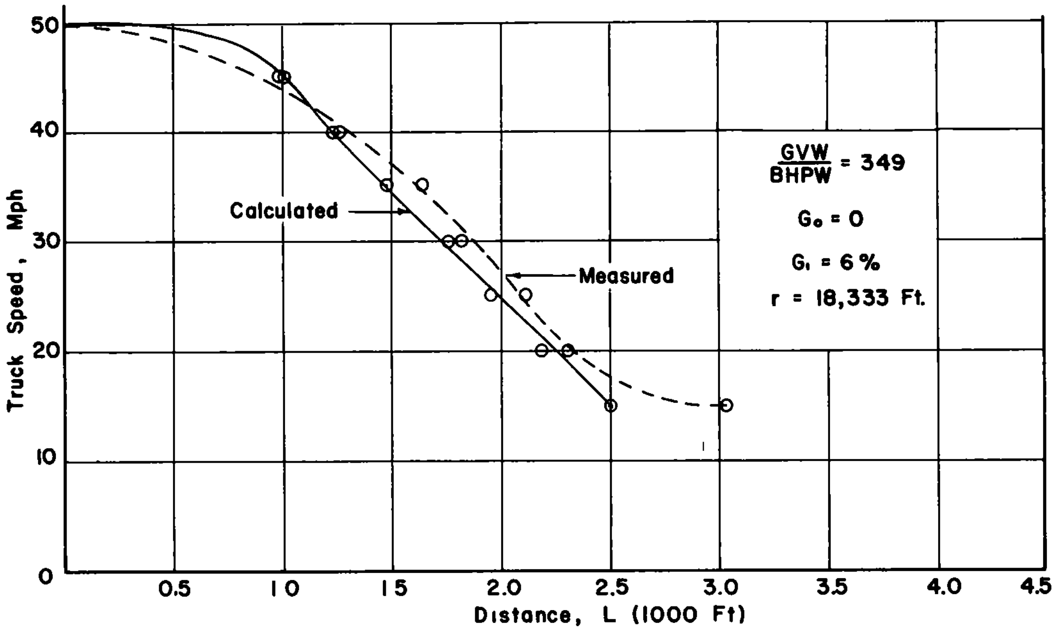

Figure 29. Comparison of calculated and measured truck velocity profiles on a sag curve.

The details of these experiments are given by Sawhill and Firey (3).

The results of these comparisons are summarized in Figure 31 showing the relation between calculated vehicle velocity and measured vehicle velocity. The calculated values are seen to agree satisfactorily with the measured values. In every case where the calculated and measured values disagreed by more than 2 mph the cause was traced to one of the following circumstances:

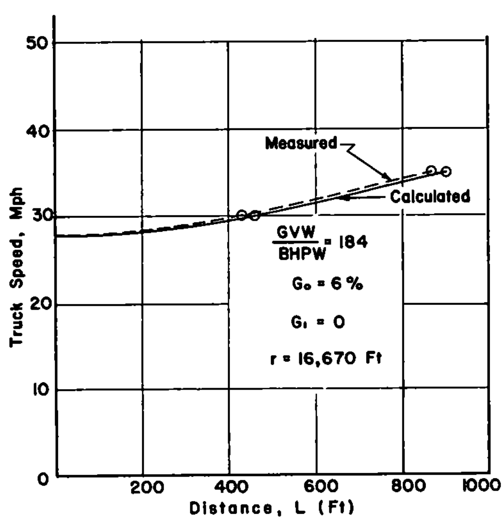

Figure 30. Comparison of calculated and measured truck velocity profiles on a summit curve.

1. In some of the sag curve experiments, the driver did not open the throttle wide until discernible deceleration occurred. The calculations necessarily assumed wide-open throttle to exist at the start of the curve, and in many cases the vehicle would then accelerate briefly until an appreciable grade exists.

2. In the experiments on uniform upgrades the theoretical maximum sustained speed is rarely achieved because the driver has available only a finite number of gear settings. As the maximum sustained speed is approached the driver will find himself either in too low a gear with F_0 slightly positive and the vehicle capable of being accelerated or in too high a gear with F_0 slightly negative and the vehicle slowing down. Most commonly the driver selects the next lower gear and operates the vehicle at maximum usable engine rpm, with slightly less than wide open throttle and at slightly less than theoretical maximum sustained speed.

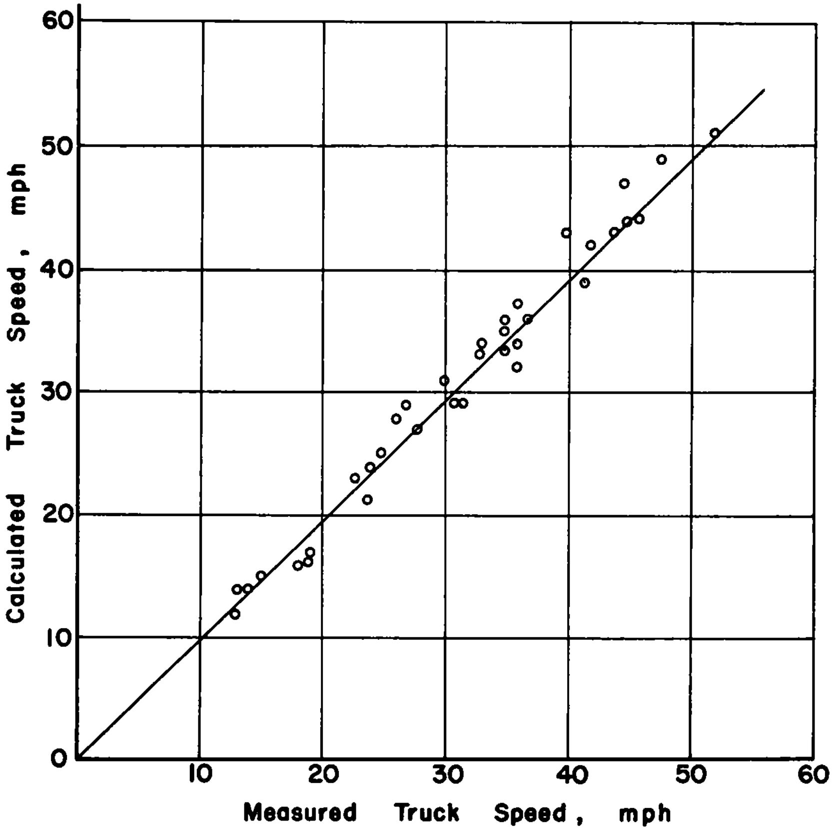

Figure 31. - Comparison of calculated and measured truck speeds on various vertical curves.

LIMITATIONS

The truck performance charts presented herein are not only limited to the ranges of truck and highway properties described previously but also subject to the following additional limitations:

1. Engine torque must be reasonably constant over the useable range of engine rpm at wide-open throttle.
2. The transmission characteristics must be such that the ratio of engine speed to vehicle speed is nearly constant in any one setting.
3. The highway surface must be of concrete or asphalt.
4. The rolling resistance characteristics of the truck must be reasonably close to the assumed relation for F_R.

Most naturally, aspirated truck engines and some supercharged engines produce nearly constant torque at wide-open throttle over the useable range of engine rpm. On supercharged engines, however, it is possible to achieve anything from a rising torque with increasing engine speed to a rising torque with decreasing engine speed. The charts presented herein do not apply to truck engines with these characteristics. The calculation method used can be applied to trucks equipped with such engines provided that wide-open throttle torque can be expressed as a function of engine rpm.

The usual geared transmission with friction clutch satisfies limitation 2 because engine speed and vehicle speed are directly related in any one gear setting. It is the hydraulic coupling and/or hydraulic torque converter transmission to which these charts are inapplicable.

Gravel or dirt surfaced roads greatly increase truck rolling resistance compared to asphalt or concrete surfaced roads which appear to be nearly equal in this respect. Hence the charts are not applicable to gravel or dirt surfaced roads.

The assumed relation for the rolling resistance force indicates the vehicle weight to be the only variable that suggests that tire hysteresis is a principal source of rolling resistance. As shown in (4), tire hysteresis losses vary markedly with tire construction and tire temperature as well as with tire loading. Hence, the assumed relation in F_R cannot be completely general. The extent to which the charts become inapplicable on this account is difficult to assess because quantitative data on the various factors influencing rolling resistance are not at present available. For this reason the charts should be used with reservations on level or nearly level roads where the rolling resistance force significantly affects truck performance.

APPLICATIONS

The charts of truck performance on highways may prove useful to highway designers and truck operators. A few examples of the use of the charts are presented here to illustrate possible applications.

Freeway Approach Ramp Example. — A large truck approaches the ramp shown in Figure 32 at an initial speed of 30 mph from an industrial area. This truck is eventually to merge with freeway traffic moving at 40 mph. It is desired to know how long an acceleration lane should be provided alongside the freeway so that the truck can reach a speed of 40 mph before merging into the freeway traffic. The design limiting truck is heavily loaded, therefore, a value of 400 for the ratio GVW/BHPW is used. Minimum vertical curve length to transition from the level industrial street to the 6 percent upramp to the freeway is 200 ft for a speed limit of 30 mph.

Reference to the scale at the bottom of Figures 13 and 14 show that to change from G = 0 to G = 6 percent the following are required:

r	L
2,500	150
5,000	300
10,000	600

For this example the radius of a vertical curve of 5,000 ft and length of 300 ft is used so that the design minimum of 200 is exceeded.

The length of constant 6 percent grade is 400 ft but the length of transition to the level freeway grade is 300 ft of 5,000-ft vertical curve radius to meet design standards. Curve 2 of Figure 10 shows truck speed to be 31 mph at point 2. Curve G = 6 of Figure 4 shows truck speed to be 20 mph at point 3. Figure 14 shows truck speed to be 15 mph at point 4. Figure 7 shows that the truck can reach a speed of 40 mph on the level freeway within a distance of 2,550 ft. This is the desired length of acceleration lane to be provided beyond point 4.

Truck Travel Time Example. — A truck operator wishes to ascertain the effect of engine power on truck travel time between points 1 and 9 on the 4-mi section of highway shown in Figure 33. The legal speed limit is 50 mph.

Figures 14 and 13 are used to determine the highway distances, which are found to be

1 to 2 = 300 ft
2 to 4 = 9,560 ft
4 to 5 = 900 ft
5 to 6 = 500 ft
6 to 8 = 9,760 ft
8 to 9 = 200 ft

The first analysis will assume an engine of low power and assume $\frac{GVW}{BHPW}$ to be 400.

Figure 10 shows truck speed to be 50 mph at point 2. Hence the travel time from point 1 to 2 becomes

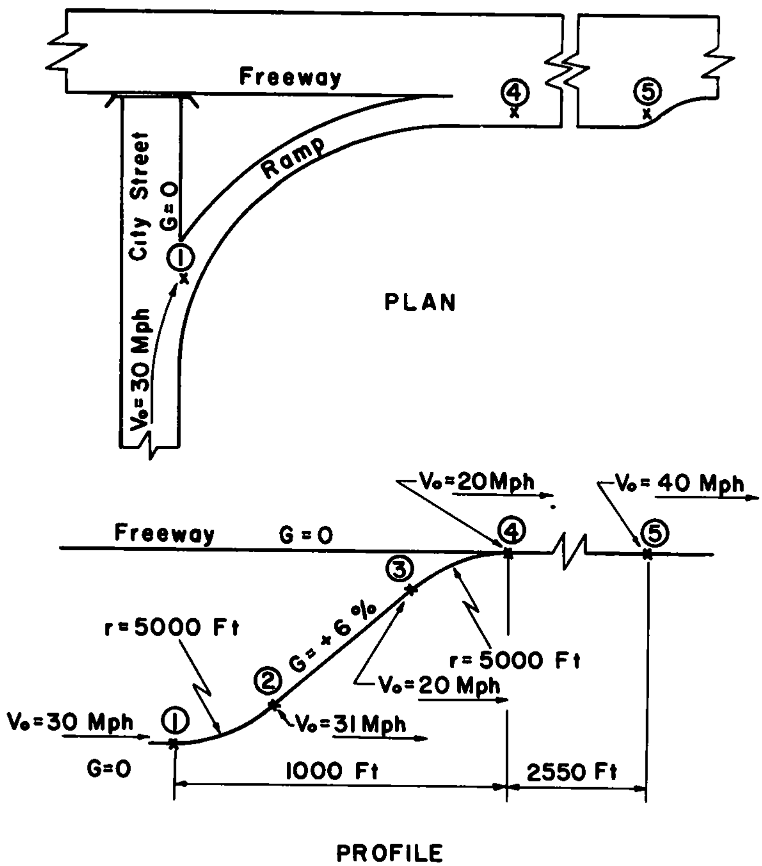

Figure 32. Highway profile for freeway approach ramp example.

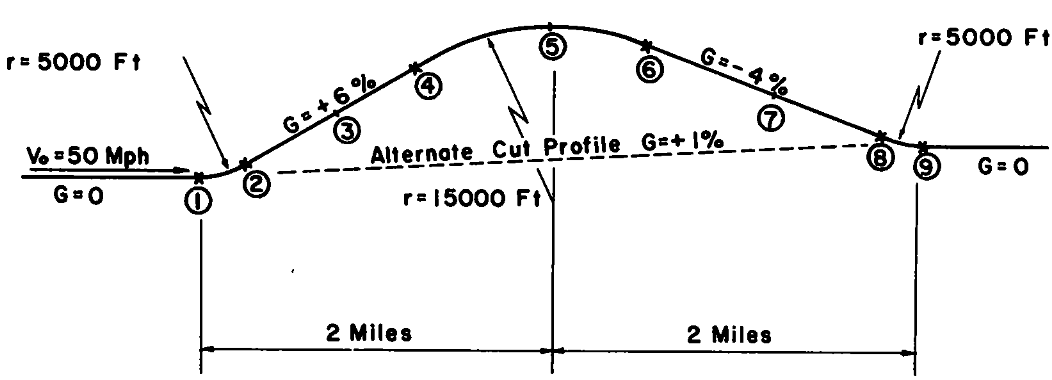

Figure 33. Highway profile for truck travel time and highway cut examples.

$$t_{1-2} = \frac{300(3,600)}{(50)(5,280)} = 4.1 \text{ sec}$$

Figure 4 shows that the truck will reach a maximum sustained speed of 13 mph at point 3 in a distance of 1,600 ft beyond point 2. Figure 22 shows that 32 sec will elapse between points 2 and 3.

The truck will continue at 13 mph up the uniform 6 percent grade from point 3 to 4 and the elapsed time becomes

$$t_{3-4} = \frac{(9,560 - 1,600)(3,600)}{(13)(5,280)} = 418 \text{ sec}$$

Figure 13 shows that truck speed will be 21 mph at point 5 and 34 mph at point 6. The elapsed time may be adequately approximated by assuming a linear variation of velocity between points 4 and 5 and between points 5 and 6:

$$t_{4-6} = \frac{(900)(3,600)(2)}{(13+21)(5,280)} + \frac{(500)(3,600)(2)}{(21+34)(5,280)} = 65 \text{ sec}$$

Figure 7 shows that the truck will reach a speed of 50 mph at point 7 after traveling a distance of 1,000 ft for an elapsed time calculated as

$$t_{6-7} = \frac{(1,000)(2)(3,600)}{(34+50)(5,280)} = 16.3 \text{ sec}$$

Presumably the truck will travel from point 7 to 9 at the legal speed limit of 50 mph and the elapsed time becomes

$$t_{7-9} = \frac{(9,760-1,000+200)(3,600)}{(50)\ (5,280)} = 123 \text{ sec}$$

The total elapsed time between points 1 and 9 is the sum of the elapsed times over the individual sections.

$$t_{1-9} = 658 \text{ sec for } \frac{GVW}{BHPW} = 400$$

Repeating the preceding steps for an engine of high power with $\frac{GVW}{BHPW}$ equal to 200,

$$t_{1-9} = 399 \text{ sec for } \frac{GVW}{BHPW} = 200$$

By comparing the cost of higher engine power with the value of shorter travel time a truck operator can select the most economic engine for trucks running over the hypothetical highway section of Figure 33.

Highway Cut Example. — The preceding truck travel time example can be extended to illustrate the use of the charts to estimate vehicle travel time savings resulting from highway cuts. What would the travel time be if the hill in Figure 33 were replaced with a cut of uniform grade?

The grade between points 1 and 9 turns out to be 1 percent on which trucks with GVW/BHPW equal to 200 can maintain the legal speed limit as shown in Figure 28, and the elapsed time becomes

$$t_{1-9} = \frac{4}{50} \times 3,600 = 288 \text{ sec for } \frac{GVW}{BHPW} = 200$$

For a value of $\frac{GVW}{BHPW}$ of 400, Figure 28 shows a maximum sustained speed of 45 mph on a 1 percent grade. The elapsed time then becomes approximately

$$t_{1-9} = \frac{4}{45} \times 3,600 = 320 \text{ sec for } \frac{GVW}{BHPW} = 400$$

The travel times for various smaller cuts can be calculated as in the preceding example and a comparison then made between the value of reduced vehicle travel time and the cost of highway cuts to determine the most economic depth of cut.

These examples represent rather simple highway sections for purposes of illustration only. The charts may also be used for several other types of calculation.

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1. "Truck Ability Prediction Procedure." Soc. Automotive Engineers, Recommended Practices TR82 (Aug. 1957).
2. Sawhill, R.B., and Firey, J.C., "Motor Transport Fuel Consumption Rates and Travel Time." HRB Bull. 276, 35-91 (1960).
3. Sawhill, R.B., and Firey, J.C., "Predicting Fuel Consumption and Travel Time of Motor Transport Vehicles." HRB Bull. 334, pp. 27-46 (1962).
4. Stiehler, R.D., Steel, M.N., Richey, G.G., Mandel, J., and Hobbs, R.H., "Power Loss and Operating Temperature of Tires." Jour. Res., Nat. Bur. Standards, C, 64C: No. 1 (Jan.-March 1960).

Predicting Fuel Consumption and Travel Time Of Motor Transport Vehicles

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Extensive data on fuel consumption and travel time of heavier trucks have been collected by research engineers at the University of Washington during the past two years under research contracts with the Bureau of Public Roads. Additional field measurements were supplied and the total information used to assist in identifying the factors to be considered in developing a formula for predicting fuel consumption or travel time. In the development of these equations the first approach was to investigate a theoretical work and energy method for making the prediction. By making a comparison of the theoretical values of fuel consumption and travel time with the actual measured values, it was possible to understand the factors involved and then to make an empirical mathematical fit of the data to an equation. The prediction of fuel consumption depends on the vehicle characteristics of gross vehicle weight and brake horsepower at wide-open throttle as well as the road characteristics of the distance traversed, the length of downhill distance, and the amount of rise in the highway profile. The prediction of travel time likewise depends on these vehicle and road characteristics in the cases of relatively rolling or mountainous terrain. However, in relatively flat topography and free-moving traffic the travel time is a function of the properly posted speed limit.

The formulas presented in this paper will be beneficial for future highway programming and planning purposes in evaluating benefits to be derived from various alternate highway locations. In addition, this information can be used for determining the economics of hauling commodities in large trucks on one route vs another, for commercial operation or construction purposes.

• THE MEASUREMENT of fuel consumption and travel time is not a new endeavor — literature is available dating back to shortly after the advent of the automobile. The major reasons for continued measurement of fuel consumption and travel time are to keep abreast of modifications in the vehicle performances and to improve the preciseness of instrumentation for making necessary measurements.

The measurement of these two characteristics is not as prevalent for the heavier-weight commercial vehicle as for passenger cars. The first major contribution to evaluating the gasoline consumption of trucks was performed (1) by the Oregon State Highway Department in the late 1930's. The Oregon study was comprehensive regarding the measurement of fuel consumption relative to surface type and alignment of the highway. It was, however, more complete in analyzing passenger car operation and was somewhat limited in the amount of information collected on the commercial vehicle.

Not until 1948 was a study concentrating on the fuel consumption and travel time of the heavier weight vehicles undertaken. The results of this study, conducted under the direction of Saal, have been published (2). The study is considered one of the first steps forward in putting the fuel consumption and travel time of the commercial vehicles into a form intended to predict or evaluate the travel time and the fuel consumption of gasoline-powered vehicles in traversing a section of highway. This report is believed to be the first of its kind to utilize the parameters of rise and fall of the highway profile in making the predictions.

In 1958, under the sponsorship of the Bureau of Public Roads, fuel consumption and travel time studies were performed by two agencies throughout the United States for the measurement of fuel consumption and travel time of commercial vehicles in normal operation (3). Information was collected on the fuel consumption and travel time of these commercial vehicles operating over conditions of varied roadway types and traffic volume characteristics. The rise and fall of the highway profile was evaluated for the routes by use of altimeters. The route over which the commercial vehicles operated was subdivided into sections of sufficient length to obtain a large enough quantity for accurate measurement of the fuel consumption while the change in roadway or traffic volume conditions was also kept in mind. This series of tests stimulated engineers at the University of Washington to measuring fuel consumption and travel time of commercial vehicles; and this was the agency that collected the much-needed data on the fuel consumption of diesel powered vehicles. With the development of such a fuel meter by the University of Washington engineers, a subsequent study was performed in the summer of 1959 that evaluated the fuel consumption and travel time of gasoline- and diesel-powered vehicles in operating under a variety of conditions of speed, grade, load, surface, and traffic operating conditions (4).

During 1959 and 1960 one of the research test truck drivers became interested in the rise and fall concept of predicting fuel consumption and travel time. As a research assistant, this graduate student (5) evaluated a new concept of the rise and fall consideration that would eliminate one or two steps necessary in the use of the data presented in HRB Bulletin 9A. This thesis led to the more thorough investigation and measurement of fuel consumption and travel time for a range of values of rise and fall.

During the summer of 1960 and 1961 the University of Washington conducted additional fuel consumption studies that in part evaluated the more precise measurement of the fuel consumption of diesel- and gasoline-powered vehicles in traversing a wide range of rise and fall sections of four-lane divided highway relatively free of traffic interference.

There is a definite need for the evaluation of the operating characteristics of commercial vehicles particularly of fuel consumption and travel time requirements in negotiating the various highway profiles. The highway engineer has been only partly cognizant of the limitations of these vehicles in the design of highways. Evidence indicates that the most beneficial solution to the economics of motor-vehicle transportation is a compromise between vehicle and highway design. The highway engineer should design future highways with consideration to the operating characteristics of existing late model commercial vehicles. Such vehicles, unlike passenger cars, represent a major investment that cannot be amortized over a short period of time.

One of the primary purposes of the present study was to collect reliable information on up-to-date motor transport vehicles operating over a variety of highway profile conditions. From this data the study may possibly be expanded to include nonexisting vehicle types relative to weight and horsepower as well as percent of highway grades that are not now presently allowed under interstate or Federal highway construction standards.

Most highway engineers are becoming acutely aware of the need to evaluate the benefits derived by a highway relocation or improvement and to substantiate this by factual information for purposes of design, budget, and programming, as well as for public acceptance.

The previous studies of this type have been extremely well conceived and conducted. However, as in the case of HRB Bulletin 9A, the vehicles used for testing were some years older than those used for the present study. The level of control of the previous studies is not precisely known, but it is doubtful that the accuracy of measurements exceeded that of the University of Washington study. Fuel was measured to the nearest 5 ml and travel time to the nearest 0.01 min. HRB Bulletin 9A was a contribution to the evaluation of these two benefits derived from highway improvements. The method of analysis was an attempt to evaluate the fuel consumption and also the travel time by the manipulation of the apparently obvious parameters of rise and fall of the highway profile. However, no attempt is made in the report to rationalize the method of interpolating the data for arriving at a solution to the amount of fuel or travel time consumed in traversing a section of highway.

One contribution by the University study has been the inclusion of fuel consumption of diesel-powered vehicles in addition to that of gasoline-powered vehicles. The study also attempts to rationalize the method of predicting the fuel consumption and travel time and to verify or revise the concept published in HRB Bulletin 9A regarding the relation of fuel consumption to vehicle weight, as well as travel time as a function of weight to horsepower.

DATA COLLECTION

The measurement of fuel was undertaken by using the same method as in previous studies by the University (4). The system of burettes was slightly modified by the installation of a small-diameter standpipe tube beside each of the burettes. Behind each tube was mounted a graduated scale allowing greater precision as well as allowing a reading to be taken with the vehicle in motion. The day tank installation for metering diesel fuel was also improved by providing a more continuous flow of fuel to the day tank and also by reducing the slight error introduced by the operation of the vehicle on steep grades.

The measurement of time for the enroute tests as well as all other tests was made with stop watches calibrated to 0.01 min.

TEST SECTION AND VEHICLE

The test section utilized for the series of tests reported in this study is a portion of US 10 between Seattle and Cle Elum over the Snoqualmie Pass Highway. This same section was used in the study of 1958, which established control points for various types of highway or traffic conditions. Some of the same check points were used in this subsequent study, and additional check points were selected where deemed necessary to provide sufficient fuel consumed to obtain a reliable measure of the quantity used. In all cases the section of highway was four-laned and in general divided. Few of the sections operated at a traffic volume in excess of their practical capacity; and in cases where the driver observed infringement on his freedom of operation the observer would identify the particular section.

Table 1 gives geometric characteristics of the test section under study. The determination of the distance between check points was ascertained from actual construction plans and verified by the use of a calibrated fifth-wheel odometer, accurate to within $\pm 1/2$ percent. The elevations used for each check point was likewise determined from the actual construction plans.

The thesis that was the forerunner of this report considered diesel-powered tractors with semitrailers for fuel consumption and travel time to indicate a method of approach or the manipulation of the parameters for arriving at a more rapid evaluation of the fuel consumption and travel time. The test vehicles used for the study were a gasoline- and diesel-powered three-axled tractor with two-axle semitrailers, operated during the summer of 1960 and a gasoline- and diesel-powered two-axle tractor pulling a single-axle semitrailer and a two-axle full trailer operated during the summer of 1961. The test units used in 1961 and the diesel used in 1960 are shown in Figure 1.

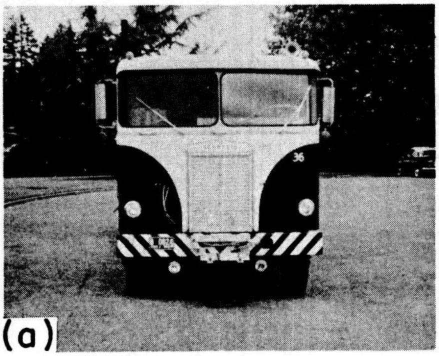
TABLE 1

TEST SECTION DESCRIPTIVE DATA

Check Point	Distance (ft)	Rise[a]		Fall[a]	
		PVI	POVC	PVI	POVC
Overpass SSH 2-D					
to					
Light at Issaquah	19,346	35.57	37.17	159.36	159.36
to					
East Fork Issaquah Creek	16,648	403.48	399.09	0	0
to					
North Bend signal light	55,482	653.37	616.68	694.31	656.45
to					
Edgewick Road	21,379	224.82	223.47	5.21	3.80
Marker east of Weigh Station					
to					
Middle crossing, Snoq. River	25,629	620.27	653.70	24.20	20.77
to					
Ollallie Creek	47,140	678.74	678.17	0.95	0.29
to					
Marker at West Summit	18,844	947.22	946.74	0	0
Marker at East Summit					
to					
Hyak Creek Bridge	12,899	0	0	393.44	391.40
to					
West end of snow shed	15,201	41.15	35.36	55.33	49.61
to					
Stampede Pass Overpass	26,236	20.50	14.01	148.38	144.22
to					
U-turn sign	25,001	281.69	273.94	52.90	45.53
to					
Easton	21,168	24.00	23.60	487.57	487.17
to					
U-turn sign	21,690	513.62	506.22	33.00	27.40
to					
Stampede Pass Under Crossing	25,038	52.48	44.50	280.85	271.11
to					
West End Snow Shed	26,352	199.20	186.42	57.20	55.96
to					
Hyak Creek Bridge[b]	16,152	82.19	69.63	66.69	49.52
	(15,201)	(55.33)	(49.61)	(41.15)	(35.36)
Hyak Creek Bridge					
to					
Marker at East Summit	12,139	442.94	442.94	0	0
Marker at West Summit					
to					
Ollallie Creek	18,844	0	0	941.06	941.06
to					
Middle crossing Snoq. River	47,346	53.30	36.40	771.53	733.41
to					
Edgewick Road	28,200	35.54	21.30	663.11	651.69
to					
Signal light North Bend	21,009	0.30	0.13	212.70	221.56
to					
East Fork Issaquah Creek	55,488	694.31	656.45	653.38	616.68
to					
Light at Issaquah	16,590	0	0	403.48	399.09
to					
Overpass SSH 2-D	19,309	159.36	159.36	35.57	37.17

[a]PVI—elevations were determined from highway profiles using intersection of grades; POVC—elevations were determined by using actual elevations of highway directly under intersection of grade lines.

[b]This section varied for summer of 1960; correct data for period given in parentheses.



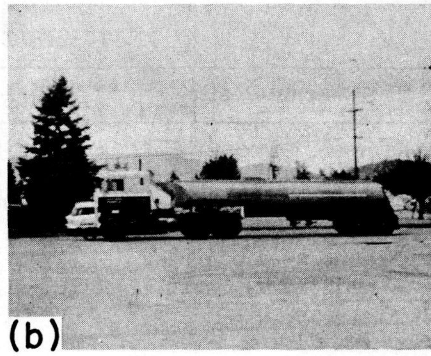
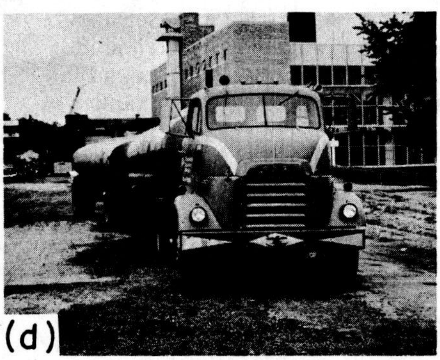
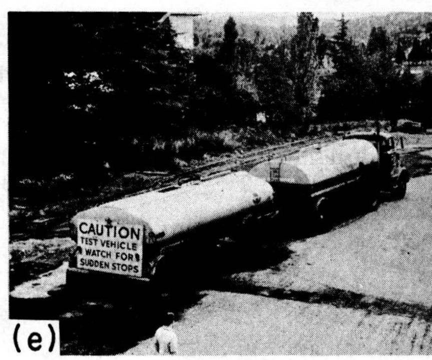


Figure 1. Visual description of three of the four test units used for precise measurement of fuel and travel time: (a) and (b) test unit 20-B (3-S2 diesel), summer 1960; (c) test unit 30-AB (2-S1-2 diesel), summer 1961; (d) and (e) test unit 31-AB (2-S1-2 gasoline), summer 1961.





The initial data, collected during the summer of 1960, was given preliminary evaluation before the other two test vehicles were operated during the summer of 1961. The latter study provided additional data, because it was found advantageous to increase the number of different gross vehicle weights from three to five. The characteristics of the vehicle and the various loading conditions are given in Table 2.

In addition, data were utilized from other previous studies to evaluate the procedure developed in the estimation of fuel consumption and travel time. Field data from the 1948 study conducted by Carl Saal was supplied to the University of Washington research group, and it was subjected to the same analysis as the detailed field data obtained for this study to verify or modify the methods developed in this report.

TABLE 2

TEST VEHICLE DESCRIPTIVE DATA

Characteristic	Test Vehicle			
	20 B	22 B	30 AB	31 AB
Axle class. of comb.	3-S2	3-S2	2-S1-2	2-S1-2
Power unit vehicle (tractor):				
Year of manufac.	1958	1955	1955	1956
Body type[a]	--	--	--	--
Frontal area of power unit (sq ft)			57.0	41.0
Wheelbase (ft):				
Axle 1 to 2		15.8	9.8	10.7
Axle 2 to 3		4.0		
Engine				
Fuel	Diesel	Gasoline	Diesel	Gasoline
No. of cylinders	6	6	6	6
Displacement (cu in.)	743	501	743	503
Mfgrs. net hp at rpm	200 at 2,100	184 at 2,600	220 at 2,100	214 at 3,200
Rear axle gear ratio	5.54	6.686	6.51/4.93	8.87/6.50
Transmission ratio:				
Main 1st	10.45	8.08	5.19	5.71
2nd	8.38	4.67	2.28	3.02
3rd	6.52	2.62	1.72	1.78
4th	5.23	1.38	1.00	1.34
5th	4.09[b]	1.00		1.00
Aux. 1st		1.29		
2nd		0.84		
3rd				
Tire size, power unit	11 × 24.5	10.0 × 20	11 × 24.5	10.3 × 20
First trailer:				
Type	Semi	Semi	Semi	Semi
Body type	Tanker	Tanker	Tanker	Tanker
Frontal area	Less than tanker	Less than tanker	Less than tanker	Less than tanker
Wheelbase (ft):				
Kingpin to axle	22.3	22.3	15.2	15.2
Axle 1 to 2	3.8	3.8		
Second trailer:				
Type			Full	Full
Body type			Tanker	Tanker
Frontal area			Less than tractor	Less than tractor
Wheelbase, axle 1 to 2 (ft)			19.5	19.5
Combination, over-all length (bumper to bumper) (ft)		47.8	61.2	62.1
Gross weight, empty (lb)	26,960	27,900	27,000	25,340

[a]None on tractors.
[b]Vehicle 20 B had 12 gear transmissions; gears 6 through 12 were 3.28, 2.55, 2.05, 1.59, 1.88, 1.00, and 0.80.

NOMENCLATURE

The following nomenclature is used:

HP = engine horsepower output;
HHV = fuel higher heating value, Btu per gallon;
n = engine thermal efficiency;

Samples taken from a continuous mixer at standard mix temperatures showed about the same amount of hardening as batch plants.

Mixes.—The results of three different mixes made at Plant F are shown in Figure 6. There is no significant difference in the hardening obtained with the different mixes. Data from other plants indicate no definite effect on hardening of mix composition. Another example is shown in Figure 7.

Asphalts.—The asphalts used in this project were from three different sources but thin film oven tests, viscosities, and other properties varied little. Table 8 gives some typical test results that were supplied by the producers for their 85-100 penetration grade asphalts, and Figure 1 shows typical viscosity-temperature relationships as supplied by the same producers.

There appears to be some difference in viscosity values at 275 F between the table values and the curves. This is brought about by variations in the asphalt from time to time as the tables or charts were compiled. Tests made on the viscosity of the asphalts indicate the curves in Figure 1 represent the material used in the mixes of this project.

Table 9 gives the results of the thin film oven tests compared with hardening obtained in the plant mixes at 325 F. This temperature probably approaches the maximum for acceptable operations in construction. The retained penetrations of the laboratory tests and of the field mixing are in excellent agreement.

The thin film oven tests and other laboratory tests of the asphalts do not vary greatly and it would not be expected that there would be any marked difference in the behavior of the asphalts when mixed in the field. The effect of asphalt type on hardening was somewhat difficult to evaluate because it was impossible to separate this variable completely from the effect of plants and aggregates. However, there appeared to be no great difference in hardening that could be attributed to type of asphalt. Had asphalts

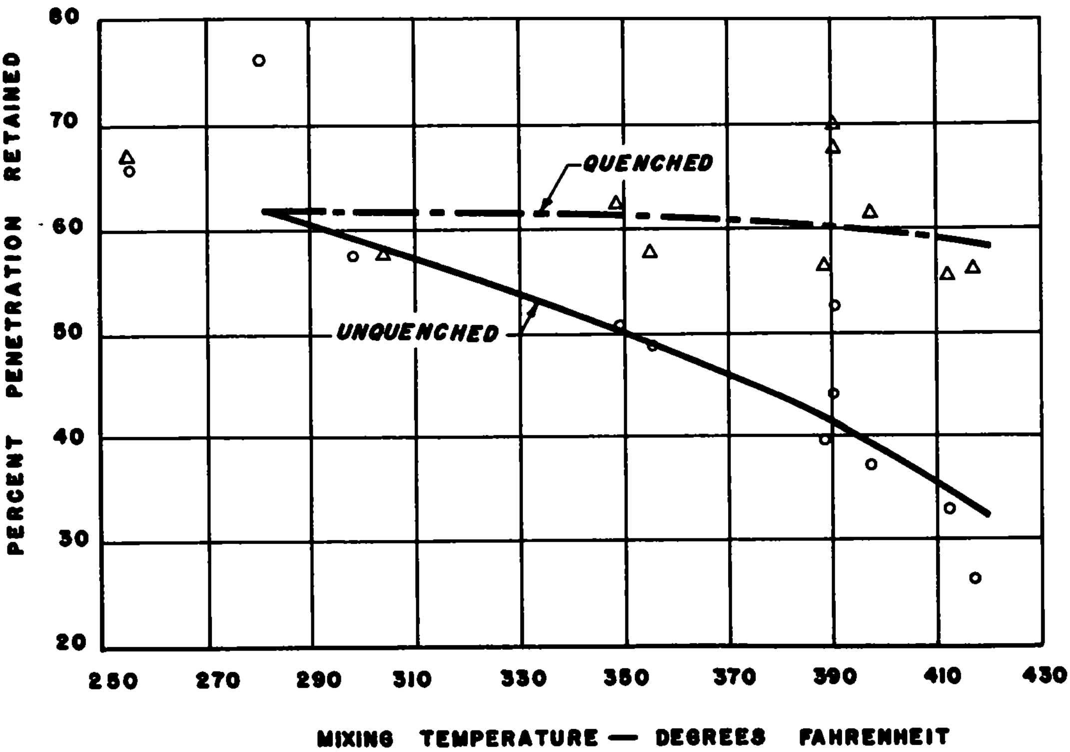


Figure 8. Comparison of percent penetration retained for all quenched and unquenched samples.

less resistant to hardening been used, it is quite likely that greater hardening would have occurred in the plant.

Rate of Cooling. —In measuring hardening in the mixer, a problem arose as to the proper procedure to use in cooling samples. Some investigators (19) have found that samples cooled quickly showed less penetration loss than those cooled slowly. This behavior was borne out by the present investigation. A study of Figures 8 and 9 shows this effect quite clearly.

The quenched samples show little change in hardening when the temperature was increased from about 300 to 400 F. Because large changes took place in the slowly cooled samples, indications are that the hardening is both a time- and temperature-controlled phenomenon. The mixes made at about 300 F show less difference between quenched and unquenched samples than do the mixes made at higher temperatures. This would mean that the hardening occurring after mixing is not serious with mixing temperatures normally used. Moreover, under present day practice, it is difficult to see what could be done about the situation to prevent such hardening. The finding is interesting from the theoretical aspect in that it shows appreciable hardening occurring long after the violent agitation of the mixture has ceased.

Some attempts were made to follow trucks enroute from the plant to the paver and take samples at intervals during the hauling period. Such results indicate some hardening enroute but results were spotty and it was difficult to draw conclusions.

None of the off-specification batches were laid or compacted, so taking road samples was impossible. Others (1, 11) found no difference between samples taken at the plant and those taken from the newly laid pavement. On this project, slowly cooled samples taken at the plant gave results similar to samples taken at the paving machine, so it is assumed that the slowly cooled samples approached the values that would have been obtained if paving samples had been available.

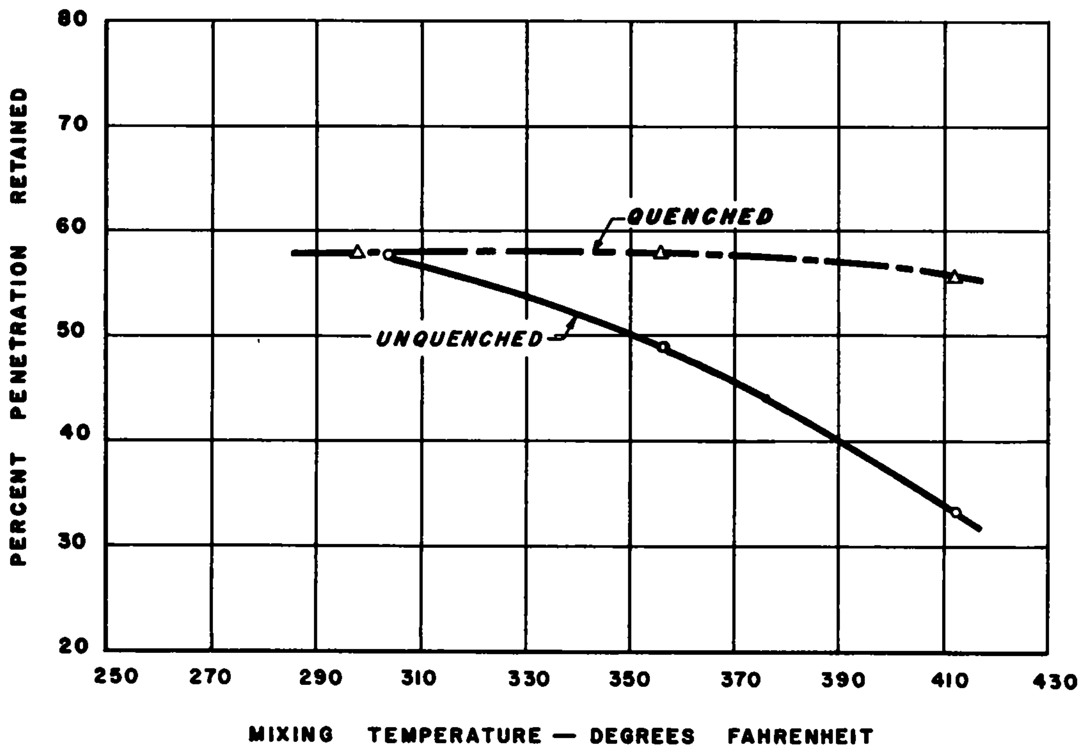


Figure 9. Comparison of percent penetration retained for quenched and unquenched samples within Group XVII.

Grade resistance can be calculated from the potential energy relation,

$$(PE) = \left(\frac{GVW}{37{,}500}\right) G_i \frac{d_i}{52.8} \tag{6}$$

in which (PE) is the potential energy acquired by a truck of weight GVW when traversing a highway section of grade G_i and length d_i expressed in horsepower-hours. The highway percent grade, G, is here taken positive on upgrades and negative on downgrades. To assist in the subsequent evaluation of energy dissipated at the brakes, grade energy is best evaluated separately for the uphill and downhill portions of a highway. For uphill sections:

$$\int_o^T (PHP)_u \, dT = \left(\frac{GVW}{37{,}500}\right) \sum_i G_i \frac{d_i}{52.8}$$

$$G_i d_i = R_i = \text{rise of the section i in feet}$$

$$\int_o^T (PHP)_u \, dT = \frac{GVW}{1.98 \times 10^6} \sum R_i \tag{7a}$$

Similarly, for downhill sections:

$$G_j d_j = F_j = \text{fall of the section j in feet}$$

$$\int_o^T (PHP)_d \, dT = \frac{GVW}{1.98 \times 10^6} \sum F_j \tag{7b}$$

The total grade energy then becomes

$$\int_o^T (PHP) \, dT = \frac{GVW}{1.98 \times 10^6} \sum R - \frac{GVW}{1.98 \times 10^6} \sum F \tag{7c}$$

Eq. 8 for truck kinetic energy is obtained from Beakey (1) and includes an approximate correction for rotational kinetic energy:

$$(KE) = \left(\frac{GVW}{29.6 \times 10^6}\right)\left(1 + 152 \frac{W+1}{GVW}\right) \frac{(mph)^2}{2} = K\,(mph)^2 \tag{8}$$

in which (KE) is the kinetic energy acquired by a truck of weight GVW when the speed is increased from zero to (mph) expressed in horsepower-hours. Because in loading and subsequently unloading cargo a truck must be stopped, the net change of truck kinetic energy is always zero on any useful run, even though wide variations of kinetic energy may occur during the run. Hence,

$$\int_o^T (AHP) \, dT = 0 \tag{9}$$

Nevertheless, changes of truck kinetic energy must be evaluated because in coming to a stop at the end of a run and in making stops during a run the energy dissipated at the brakes is most readily evaluated as being equal to a portion of the resulting loss of vehicle kinetic energy.

As discussed elsewhere (4), engine friction horsepower can be approximated by Eq. 10, provided the usable range of engine RPM is not too wide.

$$FHP = bN_E \tag{10a}$$

$$\int (FHP) \, dT = bN_E \, dT = \frac{b(M)}{60} \tag{10b}$$

The total number of engine revolutions, M, made while running over a section of highway can be approximated as the product of average engine rpm and the time taken to traverse the section.

$$M_d = N_{EA} T_d (60) \tag{11}$$

The time taken to traverse a section of highway of length d can be calculated as

$$T_d = \frac{d}{mph_d} = \frac{d}{52.8\ mph_d} \tag{12}$$

in which (mph_d) is the average truck speed over the section.

The engine friction energy term now becomes

$$\int_o^d (FHP)\, dT = \frac{bN_{EA}}{52.8} \frac{d}{(mph_d)} = f \frac{d}{(mph_d)} \tag{13}$$

The energy dissipated at the brakes is clearly indeterminate because the driver has a choice in the extent to which he may use the brakes, and widely varying traffic conditions may dictate in part the extent to which the brakes are used. An estimate of the energy dissipated at the brakes can, nevertheless, be made by use of the following assumptions:

1. The brakes are used only when stopping or on downhill sections.
2. On downhill runs, the truck potential energy is fully utilized to overcome engine friction, rolling resistance, and brake resistance only. Stated otherwise, the brakes are assumed to be used on downhill to keep truck speed constant.
3. When the brakes are being applied, the throttle is closed.

The horsepower balance for a downhill run then becomes

$$IHP_i = FHP + RHP + PHP + AHP + HPB_F \tag{14}$$

in which IHP_i is the indicated power developed by the engine at closed throttle. Over the downhill distance d_F, the work and energy balance becomes

$$\int_F (IHP_i)\, dT = \int_F (FHP)\, dT + \int_F (RHP)\, dT + \int_F (PHP)\, dT + \int_F (AHP)\, dT + \int_F (HPB_F)\, dT \tag{15a}$$

in which HPB_F is the power dissipated at the brakes during downhill running. Rearranging.

$$\int_F (HPB_F)\, dT = \int_F (IHP_i)\, dT - \int_F (FHP)\, dT - \int_F (RHP)\, dT - \int_F (PHP)\, dT - \int_F (AHP)\, dT \tag{15b}$$

The indicated work done by the engine at closed throttle downhill running is small. It can be adequately approximated by use of the fuel consumption relation.

$$(\text{Gal used downhill}) = \frac{2,545}{(nI)(HHV)} \int_F (IHP_i)\, dT = (Gal_F) \tag{16a}$$

$$\int_F (IHP_i)\, dT = \frac{(n)(HHV)(Gal_F)}{2,545} \tag{16b}$$

The total gallons of fuel used on downhill runs can be estimated as the product of the engine idle fuel flow rate, ghp_i, and the time spent on downhill runs, T_F.

$$(Gal_F) = (ghp_i)\, T_F \tag{17}$$

The time spent on downhill can be calculated as

$$T_F = \frac{d_F}{52.8\ mph_F} \tag{18}$$

in which mph_F is the average truck speed during downhill running that would normally be the legal or safe operating speed limit.

$$\int_F (IHP_i)\, dT = \frac{(nI)(HHV)(ghp_i) d_F}{(2,545)(mph_F)(52.8)} = e \frac{d_F}{(mph_F)} \tag{19a}$$

The remaining terms in the relation for downhill braking energy are evaluated as explained previously with the following results:

$$\int_F (FHP)\, dT = f \frac{d_F}{(mph_F)} \qquad (19b)$$

$$\int_F (RHP)\, dT = a \frac{d_F}{52.8} \qquad (19c)$$

$$\int_F (PHP)\, dT = -\frac{GVW}{1.98 \times 10^6} \sum F \qquad (19d)$$

$$\int_F (AHP)\, dT = 0 \qquad (19e)$$

After collecting all terms, the energy dissipated at the brakes during downhill running is evaluated as

$$(HPB_F)\, dT = e \frac{d_F}{(mph_F)} - f \frac{d_F}{(mph_F)} - \frac{ad_F}{52.8} + \frac{GVW}{1.98 \times 10^6} \sum F \qquad (20)$$

The horsepower balance when the brakes are being applied to stop the truck is

$$IHP_i = FHP + RHP + PHP + AHP + HPB_B \qquad (21)$$

Over the braking distance d_B, the work and energy balance becomes

$$\int_B (IHP_i)\, dT = \int_B (FHP)\, dT + \int_B (RHP)\, dT + \int_B (PHP)\, dT + \int_B (AHP)\, dT + \int_B (HPB_B)\, dT \qquad (22a)$$

By rearranging terms,

$$\int_B (HPB_B)\, dT = \int_B (IHP_i)\, dT - \int_B (FHP)\, dT - \int_B (RHP)\, dT - \int_B (PHP)\, dT - \int_B (AHP)\, dT \qquad (22b)$$

The time spent braking the truck to a stop is estimated as

$$T_B = \frac{2d_B}{52.8\ mph_t} \qquad (23a)$$

in which mph_t is the average truck speed at the start of braking. The various terms are evaluated as before with the following results:

$$\int_B (IHP_i)\, dT = e \frac{2d_B}{mph_t} \qquad (23b)$$

$$\int_B (FHP)\, dT = f \frac{2d_B}{mph_t} \qquad (23c)$$

$$\int_B (RHP)\, dT = \frac{ad_B}{52.8} \qquad (23d)$$

$$\int_B (PHP)\, dT = \frac{GVW}{1.98 \times 10^6} \sum R_B - \frac{GVW}{1.98 \times 10^6} \sum F_B \qquad (23e)$$

$$\int_B (AHP)\, dT = S\,K\,(mph_t)^2 \qquad (23f)$$

in which S is the total number of stops made from an average truck speed of (mph_t). By collecting all terms the energy dissipated at the brakes during stops is evaluated.

$$\int_B (HPB_B)\, dT = e\,\frac{2d_B}{(mph_t)} - f\,\frac{2d_B}{(mph_t)} - \frac{ad_B}{52.8} - \frac{GVW}{1.98 \times 10^6}\left(\sum R_B - \sum F_B\right) +$$

$$S\,K\,(mph_t)^2 \qquad (24)$$

Some simplification results if it is assumed that, on an average, as many stops are made going uphill as are made going downhill and that the net truck potential energy change during stops is zero.

$$\int_B (HPB_B)\, dT = e\,\frac{2d_B}{(mph_t)} - f\,\frac{2d_B}{(mph_t)} - \frac{ad_B}{52.8} + S\,K\,(mph_t)^2 \qquad (25)$$

Each of the original five integral terms in the work and energy balance relation has now been evaluated. When these are collected together, the following equations result for gasoline-powered trucks,

$$\int_D (IHP)\, dT = \frac{a(D - d_F - d_B)}{52.8} + \frac{GVW}{1.98 \times 10^6}\sum R +$$

$$f\left(\frac{D}{mph_D} - \frac{d_F}{mph_F} - \frac{2d_B}{mph_t}\right) + e\left(\frac{2d_B}{mph_F}\right) +$$

$$e\left(\frac{d_F}{mph_F} + \frac{2d_B}{mph_t}\right) + S\,K\,(mph_t)^2 \qquad (26a)$$

and for diesel-powered trucks,

$$\int_D (BHP)\, dT = \frac{a(D - d_F - d_B)}{52.8} + \frac{GVW}{1.98 \times 10^6}\sum R +$$

$$(e - f)\left(\frac{d_F}{mph_F} + \frac{2d_B}{mph_t}\right) + S\,K\,(mph_t)^2 \qquad (26b)$$

The fuel consumption relations now become for gasoline-powered trucks,

$$Gal = \frac{2,545\ a}{(nI)(HHV)(52.8)}(D - d_F - d_B) + \frac{2,545\ (GVW)}{(nI)(HHV)(1.98 \times 10^6)} R +$$

$$\frac{2,545\ f}{(nI)(HHV)}\left(\frac{D}{mph_D} - \frac{d_F}{mph_F} - \frac{2d_B}{mph_t}\right) +$$

$$\frac{2,545\ e}{(nI)(HHV)}\left(\frac{d_F}{mph_F} + \frac{2d_B}{mph_t}\right) + \frac{2,545\ S\,K\,(mph_t)^2}{(nI)(HHV)} \qquad (27a)$$

and for diesel-powered trucks,

$$Gal = \frac{2,545\ a}{(nB)(HHV)(52.8)}(D - d_F - d_B) + \frac{2,545\ (GVW)}{(nB)(HHV)(1.98 \times 10^6)}\sum R +$$

$$\frac{2,545\ (e-f)}{(nB)(HHV)}\left(\frac{d_F}{mph_F} + \frac{2d_B}{mph_t}\right) + \frac{2,545\ S\,K\,(mph_t)^2}{(nB)(HHV)} \qquad (27b)$$

These equations relate total fuel consumption to the properties of the vehicle and the geometry of the highway. In using them, the following truck properties must be known:

a $= \frac{RHP}{mph}$ = rolling resistance factor;

nI = average engine indicated thermal efficiency for gasoline engines;

nB	= average engine brake thermal efficiency for diesel engines;
HHV	= higher heating value of fuel in Btu per gallon;
GVW	= gross vehicle weight, pounds;
f	= engine friction factor;
e	= engine idle factor; and
K	= truck kinetic energy factor.

Methods of measuring these truck properties are described elsewhere (4). The following properties of the highway must also be known:

D	= total length in hundreds of feet;
d_F	= length of downhill highway in hundreds of feet; and
$\sum R$	= sum of rises of the highway in feet.

These highway properties may be determined from highway surveys or plans.

The following additional needed quantities are influenced by both the truck and the highway as well as by traffic conditions:

d_B	= total braking distance to stops in hundreds of feet;
(mph_t)	= average truck speed at start of braking to a stop;
S	= number of stops made;
mph_D	= average truck speed over the highway; and
mph_F	= average truck speed on downhill sections.

In general, these quantities are not readily determinable. For existing highways and trucks these could be measured experimentally. For planned highways and new truck designs they can only be estimated. The methods described by Crumbley (5) can be used to estimate the average truck speed over the highway. The average truck speed on downhill sections is presumably the legal speed limit or the safe operating speed limit. Especially difficult to estimate are the quantities involved in braking to a stop, because these depend primarily on traffic conditions. In the experiments described herein only one stop was made on a test run, and the energy lost in braking to a stop could be ignored.

THEORETICAL TRAVEL TIME ANALYSIS

Although the methods described by Crumbley (5) can be used to estimate the travel time of trucks over a highway, they become time consuming for highways of appreciable length. For longer highways an approximate method of estimating the travel time has been developed, described here.

Large transport trucks have low engine power relative to their weight. As a result their normal operation on a highway can be approximated as follows:

1. On level and nearly level as well as on downhill roads the truck travels at the legal speed limit.
2. On all other sections of highway, the engine is at wide-open throttle.

The travel time at the legal speed limit is calculated as

$$T_L = \frac{d_L}{52.8\ mph_L} \tag{28a}$$

in which d_L is the total length of level, nearly level, and downhill highway traveled at the speed limit (mph_L). Where several different legal speed limits apply, the relation becomes

$$T_L = \frac{d_{50}}{52.8(50)} + \frac{d_{45}}{52.8(45)} + \frac{d_{40}}{52.8(40)} + \text{etc.} \tag{28b}$$

To determine the sections of a highway on which the truck can operate at the legal speed limit, the chart of maximum sustained speed vs grade (6) can be utilized. If maximum sustained speed exceeds the legal speed limit, the truck is presumed to be at the speed limit. If maximum sustained speed is less than or equal to the legal speed limit, the engine is presumed to be at wide-open throttle.

At wide-open throttle the time rate of fuel consumption (gallons per hour) is approximately constant because truck engines operate within a narrow range of usable engine rpm. The following relation for the time rate of fuel consumption at wide-open throttle is obtained by rearranging the engine thermal efficiency equation.

$$\text{Gal per hour at WOT} = \text{ghpw} = \frac{2,545(\text{HPW})}{(n)(\text{HHV})} \tag{29}$$

The wide-open throttle engine power output, HPW, should be evaluated for the average engine RPM when at wide-open throttle.

The travel time on the wide-open throttle sections of the highway, T_W, can be calculated as the ratio of gallons of fuel used on these sections to the rate of fuel consumption at wide-open throttle.

$$T_W = \frac{\text{Gal at WOT}}{\text{gphw}} \tag{30}$$

The gallons of fuel used at wide-open throttle can be estimated by the methods of the preceding section.

The total travel time over a highway is the sum of the travel time at legal speed limit and the travel time at wide-open throttle. Ignored in this treatment of truck travel time are the following:

1. On level and downhill highway sections, a truck may run at less than the legal speed limit due to traffic conditions or the limits of safe operation.
2. The effect of stops and slowdowns due to traffic is not included.
3. Truck drivers often exceed the posted legal speed limits.

The influence of these factors is difficult to estimate. The travel time calculated by the methods described here can be considered as a "standard" travel time or a "minimum legal" travel time. Actual travel times may be greater or less than this "standard" travel time, depending on the effects of the foregoing factors.

COMPARISON OF THEORETICAL AND MEASURED FUEL CONSUMPTION AND TRAVEL TIME

Fuel consumptions and travel times calculated by the preceding theoretical method were found to agree reasonably well with the measured values for the test trucks. The comparison of theoretical and measured fuel consumption is shown on Figures 2 and 3. The comparison of theoretical and measured travel time is shown on Figure 4. These comparisons show a sufficiently good correlation to justify the assumptions made and the general form of the resulting theoretical fuel consumption and travel time relations.

Perhaps the greatest single source of error in the theoretical calculated results is the assumption of constant thermal efficiency of the engine. Dynamometer tests of engines as well as the truck road test data (4) clearly show engine thermal efficiency to vary over a moderate range of values. At idle (closed throttle), as on downhill runs the brake thermal efficiency of a diesel engine is clearly zero rather than the constant value used in the foregoing theoretical calculation; and this error led to negative calculated values of fuel consumption on downhill runs. On the other hand, failure to use this simplifying assumption leads to a very complex calculation procedure.

The negative fuel consumption values for diesel-powered trucks were eliminated and the correlation of theoretical to measured values improved by making the following assumptions:

1. At closed throttle and idle, the brake thermal efficiency of a diesel engine is zero.
2. At all other conditions the brake thermal efficiency of a diesel engine is constant.

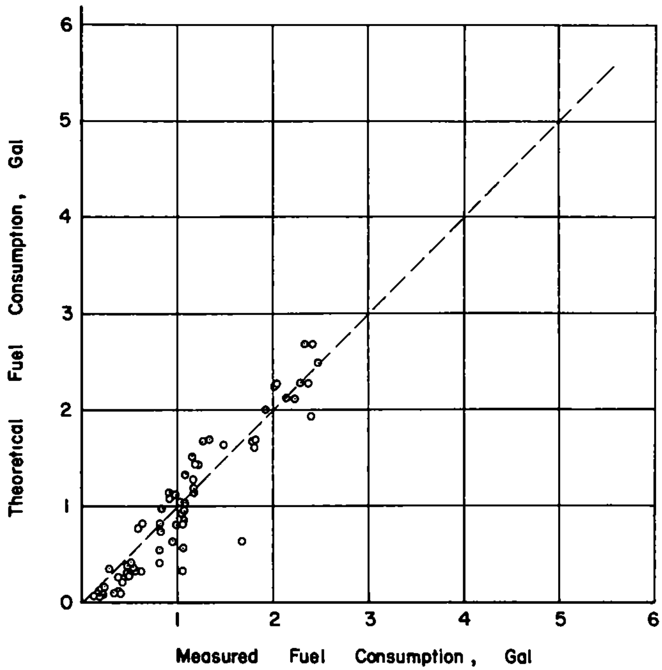


Figure 2. Comparison of theoretical calculated fuel consumption with measured fuel consumption of diesel-powered trucks.

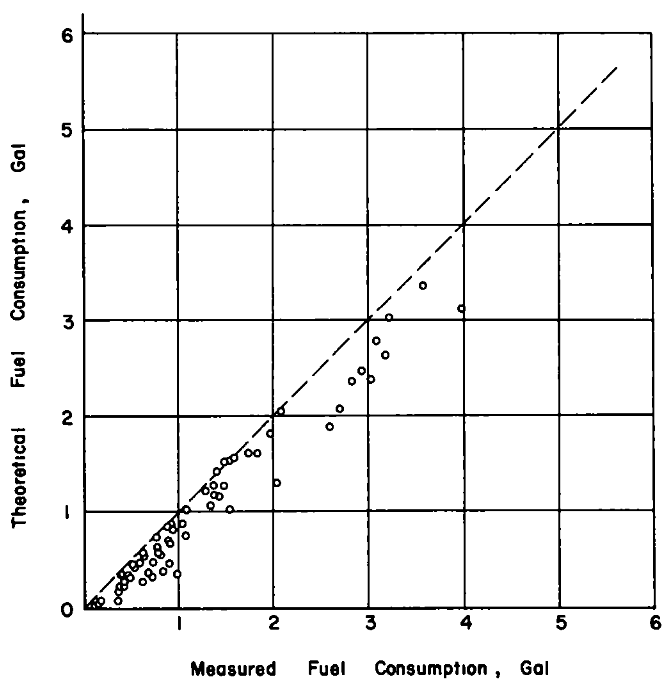


Figure 3. Comparison of theoretical calculated fuel consumption with measured fuel consumption of gasoline-powered trucks.

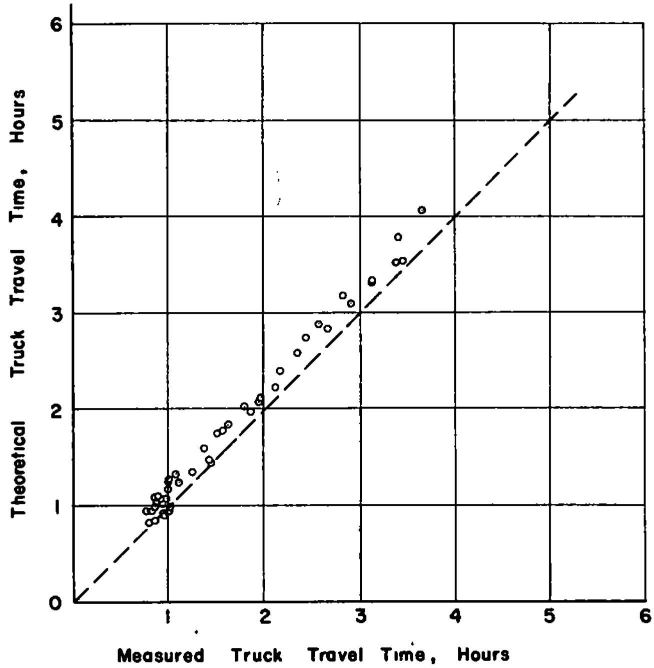


Figure 4. Comparison of theoretical calculated travel time and measured truck travel time.

With these assumptions the fuel consumption relation for diesel-powered trucks became

$$\text{Gal} = \frac{2,545\ a}{(nB)(HHV)}(D - d_F - d_B) + \frac{2,545\,(GVW)}{(nB)(HHV)(375)}\ R + (gph_i)\left(\frac{d_F}{mph_F} + \frac{2d_B}{mph_t}\right) + \frac{2,545\ S\ K\ (mph_t)^2}{(nB)(HHV)} \tag{31}$$

TABLE 3

TEST VEHICLE GROSS WEIGHTS

Test Vehicle	Loading Condition (lb)				
	1	2	3	4	5
3-S2 diesel (20B)	33,000		55,000	66,000	
3-S2 gas (22B)	33,000		55,000	66,000	
2-S1-2 diesel (30AB)	33,000	44,000	55,000	66,000	76,000
2-S1-2 gas (31AB)	32,100	42,600	53,500	64,000	76,000
2-S1 diesel (30A)[a]	18,000	33,000			
2-S1 gas (31A)[b]	16,300	33,000			

[a]Same as 30AB but trailer B dropped.
[b]Same as 31AB but trailer B dropped.

An alternate method of analyzing the results of this investigation is to assume the form of the fuel consumption and travel time equations to be that resulting from the theoretical analysis, but to determine the truck properties constants empirically so that a best fit is obtained to the measured data.

MATHEMATICAL FIT OF EMPIRICAL DATA

To develop an improved truck fuel consumption equation the experimental fuel consumption measurements were fitted to an equation of the same form as the theoretical equation. The vehicle factors were adjusted to fit the data to

$$Gal = K_0 + K_1 (D - d_F) + K_2 \sum R + K_3 d_F \tag{32}$$

The factors K_0, K_1, K_2, and K_3 are the empirical vehicle factors; the highway factors of rise and distance remained unchanged. Values of the empirical vehicle factors were calculated by applying a multivariant regression analysis to the experimental data through the use of an IBM 709 computing machine.

The regression analysis was applied to each test truck at each test weight. Observation showed that the factors K_0 and K_1 were nearly independent of vehicle weight: factor K_2 varied directly with vehicle weight, and factor K_3 varied nearly directly with maximum engine power output (or probably engine displacement and rpm). Eq. 32 was accordingly rewritten:

$$Gal = K_0 + K_1 (D - d_F) + S_2 (GVW) \sum R + S_3 (BHPW) d_F \tag{33}$$

Average values of these empirical vehicle factors, K_0, K_1, S_2, and S_3 were then determined for gasoline- and for diesel-powered trucks with the following results:
For gasoline-powered trucks,

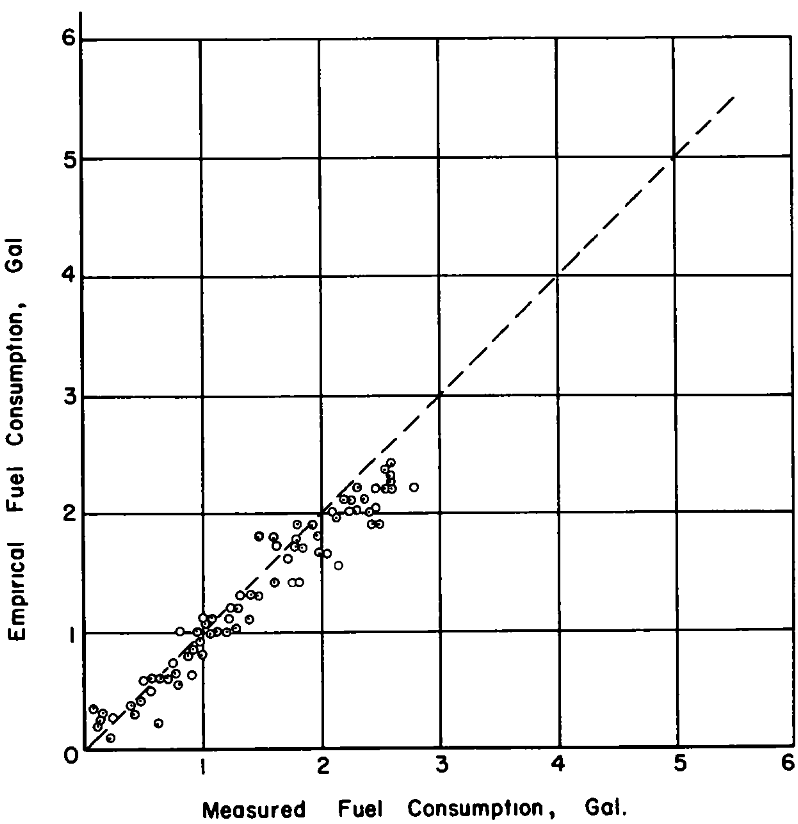


Figure 5. Comparison of empirical calculated fuel consumption with measured fuel consumption of diesel-powered trucks.

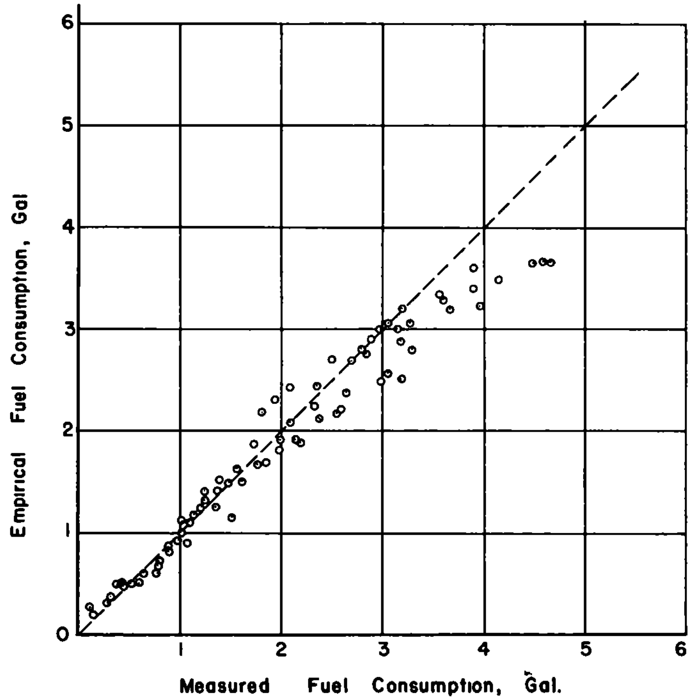


Figure 6. Comparison of empirical calculated fuel consumption with measured fuel consumption of gasoline-powered trucks.

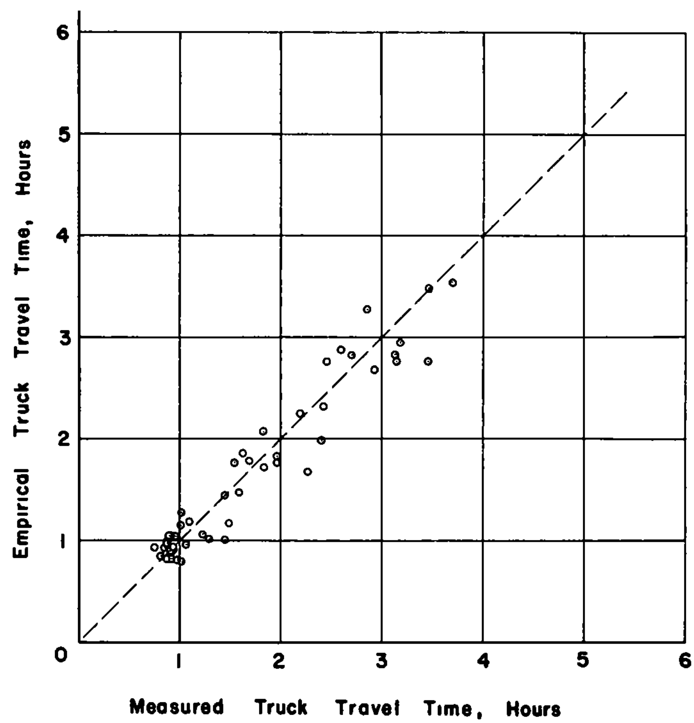


Figure 7. Comparison of empirical calculated travel time and measured truck travel time.

$$\text{Gal} = 0.0249 + 0.00314\,(D - d_F) + (41.7 \times 10^{-9})(\text{GVW}) \sum R +$$
$$(1.005 \times 10^{-5})\,(\text{BHPW})\, d_F \qquad (34a)$$

For diesel-powered trucks,

$$\text{Gal} = (0.0058) + 0.0026(D - d_F) + (22.0 \times 10^{-9})(\text{GVW}) \sum R +$$
$$(1.047 \times 10^{-5})(\text{BHPW})\, d_F \qquad (34b)$$

The degree to which these empirical fuel consumption equations fit the experimental data is shown in Figure 5 for gasoline-powered trucks and in Figure 6 for diesel-powered trucks. The fit is seen to be distinctly better than that obtained with the theoretical fuel consumption equations.

Table 4 compares the numerical values of the empirical vehicle factors to the corresponding theoretical vehicle factors.

TABLE 4

COMPARISON OF THEORETICAL AND EMPIRICAL VALUES OF VEHICLE FACTORS

Vehicle Factor	Vehicle Type	Value	
		Empirical	Theoretical
K_0	Diesel	0.0058	0
	Gasoline	0.0249	0
K_1	Diesel	0.0026	0.0016 to 0.0023
	Gasoline	0.00314	0.0021 to 0.0030
S_2	Diesel	22.0×10^{-9}	33.2×10^{-9}
	Gasoline	41.7×10^{-9}	35.8×10^{-9}
S_3	Diesel	1.047×10^{-5}	0.28×10^{-5}
	Gasoline	1.005×10^{-5}	0.31×10^{-5}

The empirical values agree well with the theoretical values, indicating that the basic form of the fuel consumption equation is probably satisfactory.

To develop an improved truck travel time equation, the theoretical equation was used, but the gallons of fuel used during wide-open throttle running were calculated by means of Eqs. 32, 33, and 34. A comparison of these empirically calculated travel times to the measured travel times, which appears in Figure 7, shows an improvement in correlation.

LIMITATIONS

The truck fuel consumption and travel time relations presented are probably usable only within the following ranges of truck and highway properties:

1. GVW between 25,000 and 75,000 lb.
2. $\frac{\text{GVW}}{\text{BHPW}}$ between 150 and 400.
3. Grades up to 8 percent.
4. Geared transmissions with friction clutches.
5. Hard surfaced highways (e.g., concrete and asphalt).
6. Naturally aspirated engines.

Most naturally aspirated truck engines develop nearly constant torque at wide-open throttle over the usable engine-speed range. The wide-open throttle torque of supercharged engines may, however, vary markedly with engine speed and, for this reason, the equations presented may not apply to such trucks.

Hydraulic couplings and torque converters do not hold a fixed ratio between vehicle speed and engine speed in any one gear setting. Hence the fuel consumption and travel time relations developed herein may not apply to trucks equipped with hydraulic couplings or torque converters.

Gravel- or dirt-surfaced roads greatly increase truck rolling resistance and hence increase fuel consumption and travel time compared to asphalt- or concrete-surfaced roads. Thus the equations presented are not applicable to gravel- or dirt-surfaced roads.

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Fuel Meter Model FM 200

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The motor benefits fuel meter is an accurate, wide flow range fuel-measuring device designed for use in fuel consumption tests of large gasoline- and diesel-powered highway vehicles. The meter is virtually uneffected by normal changes in flow rates, fuel pump pressures, electrical voltage, ambient temperatures, and vehicle ride.

•VEHICLE STUDIES at the University of Washington involving measurement of fuel have resorted to use of calibrated burettes placed in the cab of the test tractor. This arrangement, although satisfactory, has the undesirable features of large space requirements, excessive installation time, and the fire hazard of liquid fuel in the cab. This fuel meter is the outcome of numerous discussions for a more simplified method of measuring the fuel consumed by the test vehicles. Commercial fuel measuring devices are not available that have the features of covering the flow rates encountered and the accuracy desired, and that can be easily installed and carried in a test truck.

THE METER

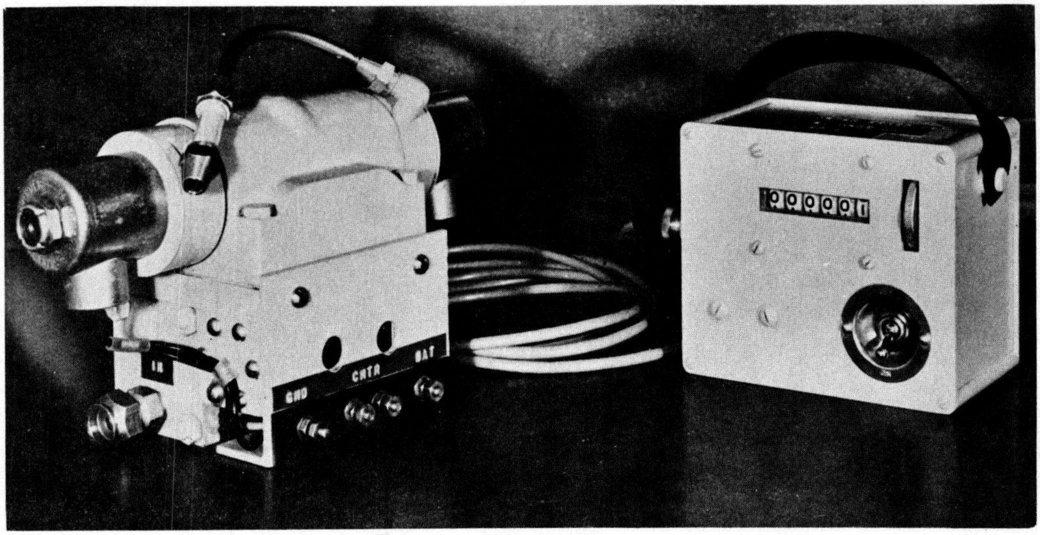

Figure 1. Fuel meter model FM 200 with counter model FM 201.

The meter (Fig. 1) is a volumetric fuel-measuring device consisting of a solenoid-actuated, air-operated valve and two measuring chambers. In operation, fuel enters a float-controlled vapor separator where liquid fuel is separated from vapor or air. The liquid fuel passes through the valve into one end of a double cylinder fitted with a free-floating piston. On entering the chamber at one end of the cylinder, the incoming fuel moves the piston which in turn discharges fuel from the second cylinder into a surge chamber and out into the fuel line. At the end of the piston travel, an electrical switch is actuated by the piston, causing the solenoid valve to interchange inlet and outlet ports of the two chambers. The measuring chambers are alternately filled and exhausted by actuation of the valve. A magnetic digital counter in series with the solenoids of the valve will record each actuation of the valve. By having a predetermined piston travel, the cylinder volume can be established; and hence, the quantity of fuel passing through the meter is known from cylinder volume and number of times the cylinder is filled. Piston travel is limited to give a cylinder volume of approximately 2.74 ml which gives approximately 1,380 counts per gal. This can be altered to give a greater or lesser number of counts per gallon by changing spacing of the electrical switches. With an increased number of counts per gallon, the operational period of the meter between servicing will become proportionally reduced. Because the electrical switches and O-rings are subject to much use, their life will determine the length of operation between normal replacement of these items. The period between meter servicing should be the time required to meter approximately 500 to 700 gal of fuel.

A fuel pressure drop through meter ranges from 1 to 1½ psi at flow rates corresponding to normal truck fuel use. A booster fuel pump is recommended in addition to the vehicle fuel pump to insure proper engine operation.

Installation of Meter

Reference to Figures 2 and 3 is made when making the meter installation.

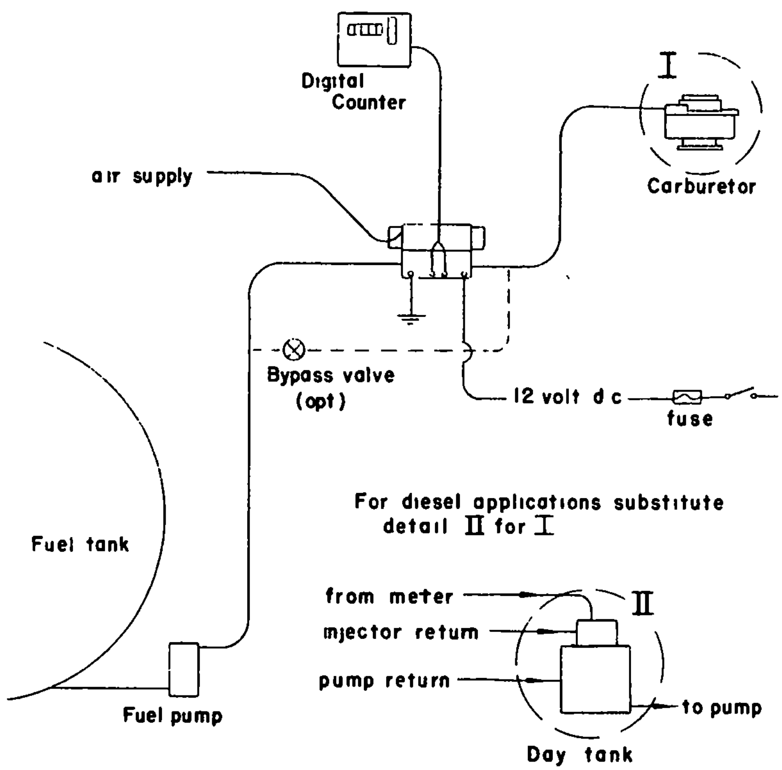

Figure 2. Installation diagram.

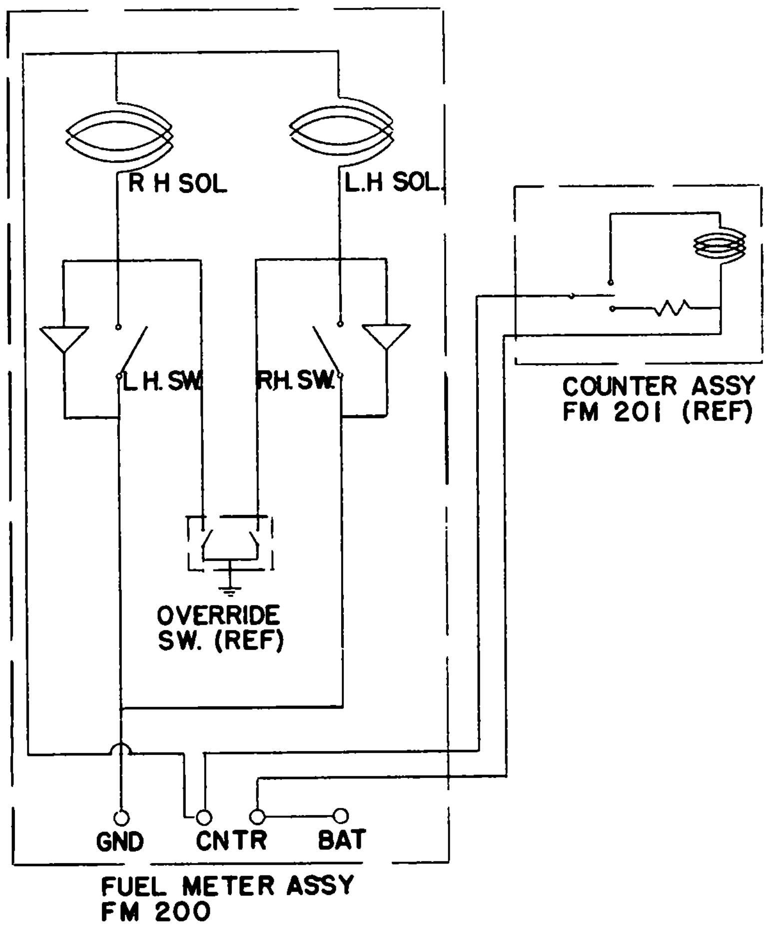

Figure 3. Wiring diagram.

The meter is installed in an upright position in the engine compartment of the vehicle being tested. The extreme temperature area of the exhaust manifold should be avoided. Mounting of the meter in other than an upright position will hinder operation of the float-controlled vapor separator and may permit fuel to pass directly from the vapor separator to the surge chamber without being measured.

Inasmuch as installations will vary so widely, no specific mounting directions are given but left to the discretion of the instrumentation mechanic. Two holes have been provided in the bottom of the cover plate for anchoring the meter to a horizontal shelf; and two holes, tapped 1/4-20 UNC, have been provided in the rear side of the base plate for attaching the meter to a vertical bracket or bulkhead. The meter may also be suspended by clamps placed around the solenoid covers. Suspension by attaching to the ¼-in. copper air line is not recommended.

Fuel lines are connected as indicated on the installation drawing. Copper fuel lines may be used in place of flexible lines if no relative movement exists between fuel pump, meter, and carburetor. Check to make sure the fuel flow is in the proper direction through the meter. A bypass valve is recommended to permit fuel to be diverted past the meter if meter failure occurs.

An air supply is connected to the ¼-in. tube tee on one solenoid. A flexible line should be used unless relative movement is small. On vehicles that normally have an air compressor (heavy trucks), the air connection is made directly to a constant pressure line from the air tank and reduced to the required pressure. If the vehicle is not equipped with a compressor, a nitrogen or air cylinder (2,000 to 3,000 psi) can be carried in the vehicle with necessary pressure regulating equipment. Air supply pressure is adjusted to 20 to 25 psi. When new O-rings are installed, the 20 to 25 psi may not be enough for operation of the valve and a slightly higher pressure may be required. "Pound out" of the valve may occur if excessive pressure is used.

Electrical connections are made using 18-gage wire or heavier. A small automotive fuse (9 or 14 SFE) should be used to safeguard equipment in case of an electrical short circuit. For convenience the meter and any electrical fuel pumps should be connected to the ignition switch. The electrical cord supplied with the magnetic counter (FM 201) has solderless terminals at one end for connecting to the meter. Although the wires are of different color, interchanging the leads makes no difference in meter operation. If the meter is used with a negative ground, the diodes should be reversed. The meter requires approximately 1 amp at 12 volts.

Calibration

Laboratory calibration (Figs. 4 and 5) of the flow meter was accomplished by recirculating fuel from a supply tank and through a series arrangement of an electric fuel pump, the meter, a needle valve, a three-way valve, and back to the supply tank. The needle valve was used in regulating the flow to permit calibration for any flow rate that may be encountered in the field. Fuel was recirculated until a calibration was to be made, at which time the three-way valve was used to divert fuel into a graduated burette while a predetermined number of counts were registered on the digital recorder. Fuel quantity, digital counter record, and time to meter the given quantity were all recorded for use in preparing a calibration chart. Fuel quantity was usually the quan-

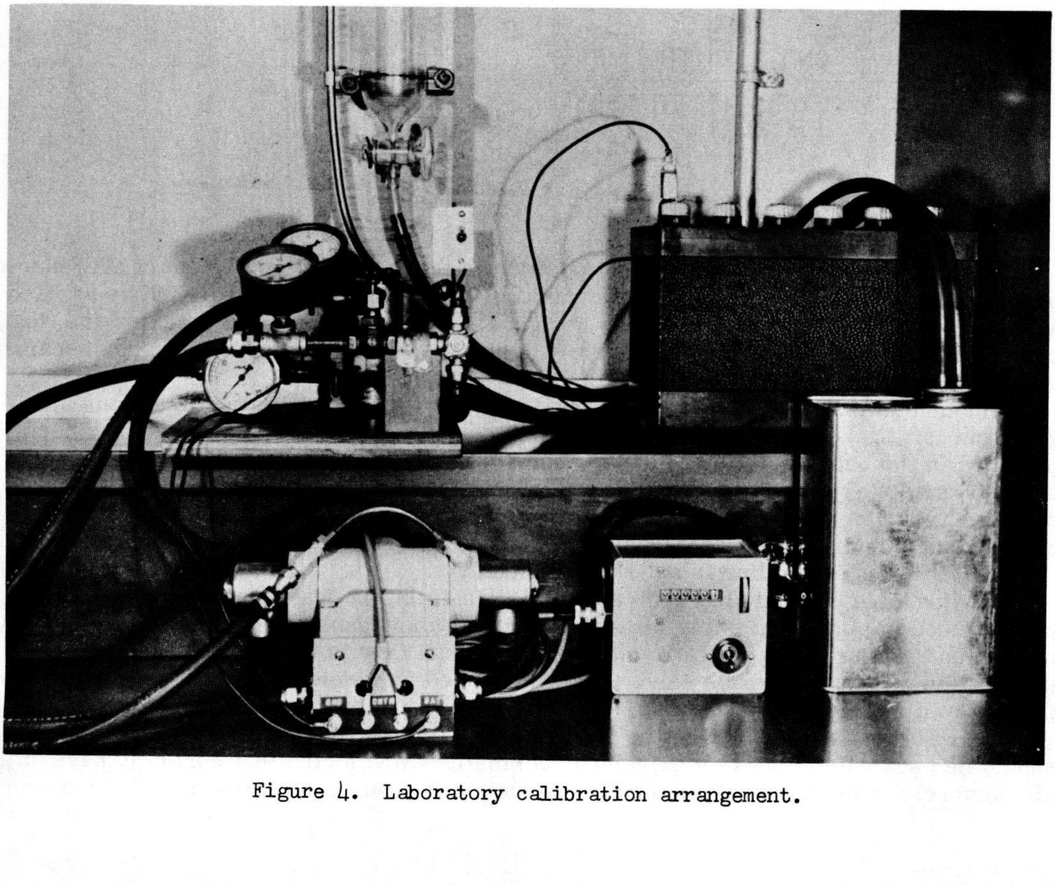

Figure 4. Laboratory calibration arrangement.

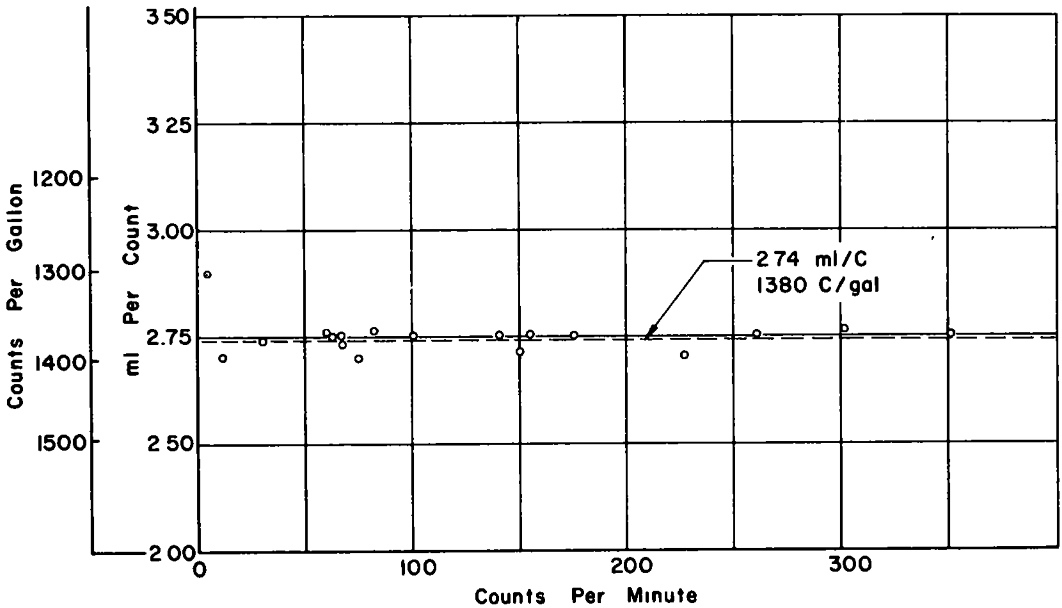

Figure 5. Calibration chart.

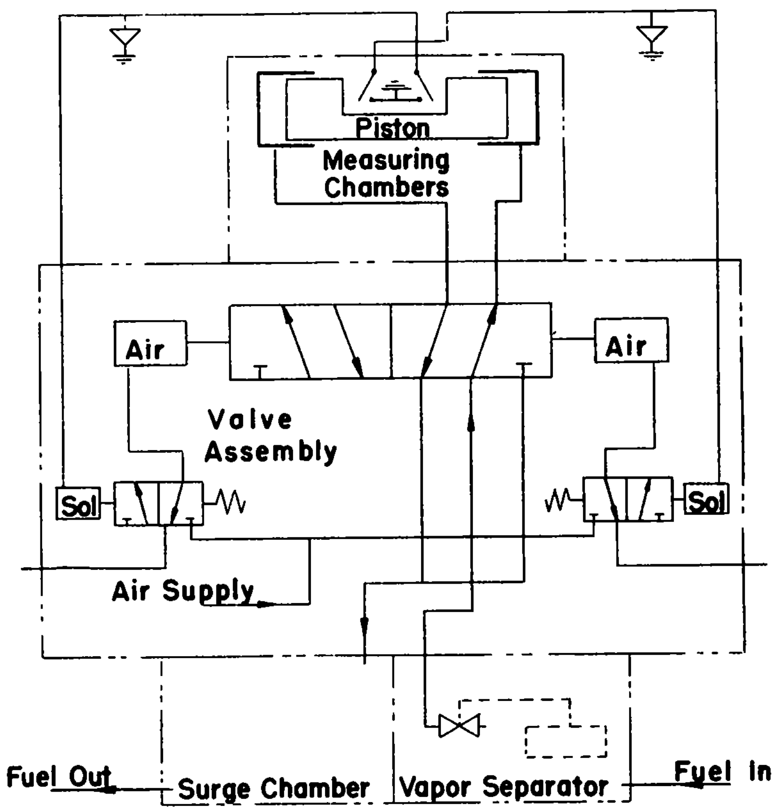

Figure 6. Schematic arrangement.

tity corresponding to 100 counts on the recorder. Fuel inlet and outlet pressures were also recorded every time a significant change was made in the flow rate.

Field calibration was also performed by installing the meter in series with the burette fuel measuring equipment in the test truck. Laboratory and field calibration were in very good agreement.

Troubleshooting

Erratic operation of the meter is probably due to air in the fuel line or a large ratio of vapor to fuel. The vapor separator will function properly with low volume of vapor in the fuel lines, but the meter will become erratic when air or vapor is admitted to the measuring chamber. Erratic operation should cease after a few minutes operation unless there is an air leak or restriction in the suction line. A restriction in the suction line will cause a portion of the fuel to vaporize because of the reduced pressure.

If meter fails to start, valve spool may not be positioned properly. With a screwdriver, one of the piston limit switches is actuated (or one of the two override switches depressed). If the meter has not started, air pressure, fuel pressure, and electrical connections should be checked. When actuating the electrical switches, there should be an audible exhaust from the solenoids. If air pressure is great enough, the spool can be heard moving in the valve when the switches are alternately actuated.

If the meter stops at low flow rates, it is probably because fuel is bypassing the O-rings. Two of the six O-rings are replaced on the valve spool, the second ring from each end of the spool. The ring and cylinder should fit snugly when the ring is properly installed on the spool. These two rings are critical items in the meter. Manufacturing tolerances for the rings are too wide to permit the use of all commercial O-rings for this application.

If the meter stops at any flow rate but can be started by actuation of override switches, then probably life expectancy of electrical switches has been shortened by permitting switch to become soaked in fuel. Both switches should be replaced.

If there is visible fuel solenoid exhaust, then end O-rings on valve spool should be replaced.

If the override switch does not actuate solenoid valve, then probably there is poor electrical connection, or electrical limit switch is in closed position due either to piston actuation or to defective switch.

If the meter is "dead", but all external electrical connections are good, poor electrical connection between meter and counter should be checked. This is a series electrical circuit and good electrical continuity must be maintained.

If meter calibration is nonlinear, fit of number 2 and 5 O-rings should be checked, and O-rings not having a sung fit in cylinder replaced.

If the meter does not recalibrate after changing electrical switches, then switches may not be tight against micarta block of switch assembly.

Applications

The fuel meter is intended to be used in vehicle testing and can adequately measure fuel at flow rates corresponding to all highway vehicles using gasoline or diesel fuels. The meter is not suitable for constant use on highway vehicles, because its initial cost and periodic servicing would make it impractical.

When using the meter to measure fuel consumption of diesel vehicles having a recirculating fuel system, a float-controlled "day tank" must also be used. Using the day tank, an equivalent volume of fuel is measured instead of the actual quantity entering the vehicles injectors.

Improvements

A direct acting solenoid valve replacement for the air-operated solenoid valve would permit a better application on test vehicles not equipped with a constant pressure air supply such as found on large trucks. Solenoid life and high current for the small electrical switches will be problems faced when redesigning for the direct acting solenoid valve.

Future fuel meters using the air-operated solenoid valves should be redesigned slightly to incorporate a valve built to tighter valve spool tolerances than the commercially available valves. This should not change the meter except in appearance.

The meter should be scaled down in future changes to provide for better operation at the extremely low flow rates. Presently the meter operates best at rates above idle fuel flow rates.

CONCLUSIONS

The Motor Benefits fuel meter will measure motor vehicle fuel consumption from idle to wide-open throttle operations with accuracies better than any commercially available meter. Although research personnel at the University of Washington have not used this meter exclusively for vehicle fuel consumption tests, they have had good correlation between laboratory tests and field tests and have used the meter to measure in excess of 700 gal of fuel between service periods without major difficulty or any meter component replacements.

ACKNOWLEDGMENT

This meter was developed as part of Research Contract CPR 11-7875 with the Bureau of Public Roads.

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