Vehicle Dynamics Model for Predicting Maximum Truck Acceleration Levels

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Abstract: The paper presents a simple vehicle dynamics model for estimating maximum vehicle acceleration levels based on a vehicle's tractive effort and aerodynamic, rolling, and grade resistance forces. In addition, typical model input parameters for different vehicle, pavement, and tire characteristics are presented. The model parameters are calibrated/validated against field data that were collected along the Smart Road test facility at Virginia Tech utilizing a truck and trailer for 10 weight-to-power configurations, ranging from 85 kg/kW to 169 kg/kW (140 lb/hp to 280 lb/hp). The model was found to predict vehicle speeds at the conclusion of the travel along the section to within 5 km/h (3.1 mi/hr) of field measurements, thus demonstrating the validity and applicability of the model.

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INTRODUCTION

Truck performance along grade sections may have significant impacts on roadway throughput depending on the grade level, the truck characteristics, the percentage of trucks, and the level of congestion along the roadway section. Although the *Highway Capacity Manual* (HCM) provides curves for predicting vehicle speeds as a function of the distance traveled and the percentage grade along the section (TRB 1998), these curves were developed more than 20 years ago and thus may not be reflective of current trucks. For example, Fig. 1 illustrates that the maximum speed along a level roadway (0 percent grade) barely exceeds 90 km/h (55 mi/hr). Furthermore, the curves indicate different equilibrium speeds (crawl speeds) depending on whether a truck is accelerating or decelerating (grades 1, 2, and 3%). Specifically, differences in equilibrium speed estimates based on a vehicle's acceleration and deceleration behavior contradicts basic vehicle dynamics. It is not clear at this point if this difference in crawl speeds is a result of some flaw in the HCM procedures or that the equilibrium speeds occur outside the 5-km travel range.

The study first presents the proposed vehicle dynamics model and recommended parameters for the model. Next the paper describes how the model was calibrated/validated using field data collected along the Smart Road test facility. In order to demonstrate the applicability of the model, performance curves for a sample 120 kg/kW (200 lb/hp) truck are developed. These performance curves overcome the major shortcomings of the current performance curves in the HCM that were described earlier. Finally the conclusions of the paper and recommendations for further research are presented.

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STUDY OBJECTIVES

The objectives of this study are threefold. First, the study presents a simple model based on vehicle dynamics for the analysis of vehicle acceleration behavior. While similar models have been developed and described elsewhere [e.g., Mannering and Kilareski (1990); Fitch (1994); Archilla and De Cieza (1999)], the literature appears to lack a systematic approach for the application of these models. Consequently, the second objective of the study is to provide a feasible range of input parameters for the use in these models. Third, the study demonstrates the validity of these models using second-by-second speed measurements collected along the Virginia Tech Smart Road test facility. The validation effort is unique in two aspects. First, vehicle acceleration levels were collected in a controlled environment, thus isolating maximum vehicle acceleration behavior. Second, the model is systematically validated for different vehicle weight-to-power ratios.

VEHICLE DYNAMICS MODEL

Tractive Effort

The proposed model is similar to models presented by others. Specifically, the model constrains the maximum tractive force that is computed in (1) using (2), as demonstrated in (3). Eq. (2) accounts for the friction between the tires of the vehicle's tractive axle and the roadway surface. The use of (3) ensures that the tractive effort does not approach infinity at low vehicle speeds

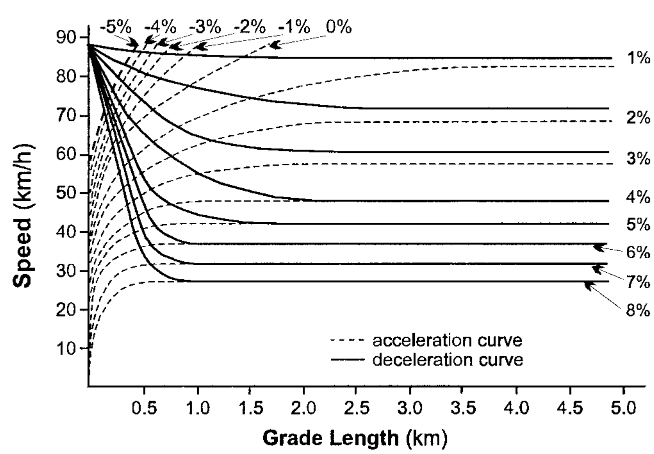

FIG. 1. Truck Performance Curves for an Average Truck (120 kg/kW or 200 lb/hp) (TRB 1998)

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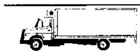
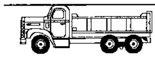
Truck configuration	Length (m)	Mass (10^3 kg) and percent mass per axle[1]						
		Total	Front axle		Tractive axle[2]		Trailer axles	
	6.86	13.6	4.5	33%	9.1	67.0%		
	7.92	19.3	3.9	20%	15.4	80.0%		
	7.92	22.7	8.2	36%	14.5	64.0%		
	15.24	28.6	4.3	15%	8.9	31.0%	15.4	54.0%
	16.07	35.4	4.6	13%	15.4	43.5%	15.4	43.5%
	19.80	36.3	3.5	10%	8.2[3]	22.5%	8.2[3]	22.5%

[1] Limits acording to the STAA 1982 assuming a maximum mass per single axle of 9.0 metric tons (20000 lb), a maximum mass per tandem axle of 15.4 metric tons (34000 lb), a maximum total mass of 36.3 metric tons (80000 lb), a maximum width of 2.6 m (8.5 ft), a maximum length for a single trailer of 14.6 m (48 ft), and a maximum length for a double trailer of 17.1 m (56 ft).
[2] Rear axle is the tractive axle.
[3] For each single axle.

FIG. 2. Length and Mass Distribution for Typical Trucks (Fitch 1994)

$$F_t = 3{,}600\eta \frac{P}{V} \quad (1)$$

$$F_{\max} = 9.8066 M_{ta}\mu \quad (2)$$

$$F = \min(F_t, F_{\max}) \quad (3)$$

where F_t = tractive effort (N); P = engine power (kW); V = truck speed (km/h); η = transmission efficiency; $F_{\max}$ = maximum tractive force (N); M_{ta} = vehicle mass on tractive axle (kg) such that $M_{ta} = M \cdot perc_{ta}$; $perc_{ta}$ = percent mass acting on tractive axle; μ = coefficient of friction between tires and pavement; and F = tractive effort effectively acting on truck (N).

Eq. (1) indicates that the tractive force F_t is a function of the ratio between the vehicle speed V and the engine power P. It should be noted that there are two main sources of power loss that degrade the tractive effort produced by the truck engine. The first source of power loss is caused by engine accessories including fan, generator, water pump, magneto, distributor, fuel pump, and compressor. The second source of power loss occurs in the transmission system. Typical transmission efficiencies range from 0.89 to 0.94 (SAE J2188 1996). In addition, Fitch (1994) recommends an additional 10% reduction caused by accessory losses.

It should be noted that the maximum tractive force is a function of the proportion of the vehicle mass on the tractive axle. A summary of typical axle mass distributions for different truck types is provided in Fig. 2 for use within the proposed model. Axle mass distributions shown in Fig. 2 are in accordance with the limits established in STAA 1982, assuming a maximum mass per single axle of 9,000 kg (20,000 lb), a maximum mass per tandem axle of 15,400 kg (34,000 lb), a maximum total mass of 36,300 kg (80,000 lb), a maximum width of 2.6 m (8.5 ft), a maximum length for a single trailer of 14.6 m (48 ft) and a maximum length for a double trailer of 17.1 m (56 ft).

Resistance Forces

The model considers three major types of resistance forces, including aerodynamic, rolling, and grade resistance as was proposed in previous models (Mannering and Kilareski 1990; Fitch 1994; Archilla and De Cieza 1999). The total resistance force is computed as the sum of the three resistance components, as summarized

$$R = R_a + R_r + R_g \quad (4)$$

where R = total resistance (N); R_a = air drag or aerodynamic resistance (N); R_r = rolling resistance (N); and R_g = grade resistance (N).

Aerodynamic Resistance

The aerodynamic resistance, or air drag, is a function of the vehicle frontal area, the altitude, the truck drag coefficient, and the square of speed of the truck, as indicated in (5) and (6).

$$R_a = c_1 C_d C_h A V^2 \quad (5)$$

$$C_h = 1 - 8.5 \times 10^{-5} H \quad (6)$$

where A = truck frontal area (m^2); V = truck speed (km/h); C_d = truck drag coefficient; C_h = altitude coefficient; c_1 = constant equals to 0.047285; and H = altitude (m).

The constant c_1 accounts for the air density at sea level at a temperature of 15°C (59°F). Typical values of vehicle frontal areas for different truck and bus types are provided in Table 1 while typical drag coefficients are provided in Table 2. Eq. (6) is a linear approximation that the writers derived from a more complex formulation (Watanada et al. 1987). The linear approximation was found to provide similar results to the more complex formulation for altitudes in the range of 0–5,000 m (0–16,400 ft).

TABLE 1. Typical Vehicle Frontal Areas (SAE 1996)

Vehicle type	Area (m^2)
Semi-trailer van type	10.0
Semi-trailer van type body (2.60 m wide)	10.7
Straight truck van type body	8.9
Tanker and flatbeds, conventional cab	7.0
Tanker and flatbeds (cab-over, high tilt)	7.9
Dump truck, conventional cab	6.8

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TABLE 2. Typical Vehicle Drag Coefficients

Truck type	C_d	Source
Single unit	0.70	Fitch (1994)
Tractor-semitrailer	0.70	Fitch (1994)
Car hauler—cattle hauler	0.96–1.10	SAE J2188
Garbage	0.95–1.05	SAE J2188
No aerodynamic aids	0.78	SAE J2188
Aerodynamic aids on roof	0.64	SAE J2188
Full aerodynamic treatment	0.58	SAE J2188

TABLE 3. Road Surface Coefficients [Derived from Fitch (1994)]

	Rolling Coefficients		
Pavement type	Pavement condition	C_r	Friction coefficients
Concrete	Excellent	1.00	0.80
Concrete	Good	1.50	0.70
Concrete	Poor	2.00	0.60
Asphalt	Good	1.25	0.60
Asphalt	Fair	1.75	0.50
Asphalt	Poor	2.25	0.40
Macadam	Good	1.50	0.55
Macadam	Fair	2.25	0.45
Macadam	Poor	3.75	0.35
Cobbles	Ordinary	5.50	0.50
Cobbles	Poor	8.50	0.40
Snow	5.08 cm (2 in.)	2.50	0.20
Snow	10.16 cm (4 in.)	3.75	0.15
Dirt	Smooth	2.50	0.30
Dirt	Sandy	3.75	0.20
Mud		3.75–15.0	0.15
Sand	Level soft	6.0–15.0	0.15
Sand	Dune	16.0–30.0	0.10

TABLE 4. Rolling Resistance Constants c_2 and c_3 [Derived from Fitch (1994)]

Tire type	c_2	c_3
Bias ply	0.0438	6.100
Radial	0.0328	4.575

Rolling Resistance

The rolling resistance is a linear function of the vehicle speed and mass, as indicated in (7). Typical values for rolling coefficients (C_r) as a function of the road surface type and condition are provided in Table 3. In addition, the rolling resistance coefficients (c_2 and c_3) vary as a function of the vehicle's tire type, as summarized in Table 4. Generally, radial tires provide a resistance that is 25% less than that for bias ply tires

$$R_r = 9.8066C_r(c_2V + c_3)\frac{M}{1{,}000} \tag{7}$$

where M = truck total mass (kg); C_r = rolling coefficient; and c_2, c_3 = rolling resistance coefficients.

Grade Resistance

The grade resistance is a constant that varies as a function of the vehicle's total mass and the percent grade that the vehicle travels along, as indicated in (8). The grade resistance accounts for the proportion of the vehicle weight that resists the movement of the vehicle

$$R_g = 9.8066Mi \tag{8}$$

where i = percent grade (m/100 m).

Maximum Vehicle Acceleration

The maximum acceleration is a function of the forces acting on the vehicle and can be computed using

$$a = \frac{F - R}{M} \tag{9}$$

where a = maximum truck acceleration (m/s^2); F = tractive effort (N); R = total resistance force (N); and M = vehicle total mass (kg).

Assuming a constant vehicle power and given that acceleration is the second derivative of distance with respect to time, (9) resolves to a second-order ordinary differential equation (ODE) of the form indicated in (10). The ODE is a function of the first derivative of distance (vehicle speed) because the tractive effort, the rolling resistance, and aerodynamic resistance forces are all functions of the vehicle speed. In addition, the ODE may be a function of the distance traveled if the roadway grade changes along the study section. It should be noted at this point that because the tractive effort includes a minimum operand, the derivative of acceleration becomes a noncontinuous function. Consequently, a first-order solution technique is inevitable, as will be described subsequently in the paper.

$$\ddot{x} = f(\dot{x}, x) \tag{10}$$

where x = distance traveled; $\dot{x}$ = speed; and $\ddot{x}$ = acceleration.

MODEL VALIDATION

In an effort to validate the proposed model together with the recommended input parameters, a number of field tests were conducted along a test roadway at the Virginia Tech Transportation Institute. This section first describes the test truck that was utilized to conduct the validation effort. Second, the study roadway section is described in detail prior to discussing the specifics of the validation effort. Finally, the model validation results are presented and discussed.

TEST TRUCK CHARACTERISTICS

The test truck that was utilized in the validation effort was a 1990 truck owned by the Virginia Department of Transportation (VDOT). The truck used a Cummins NTC-350 engine with an engine power rating of 261 kW (350 hp) at an engine speed of 2,100 rpm. The peak torque of 1,627 N·m (1,200 lbf-ft) occurred at an engine speed of 1,300 rpm. The engine was fairly large with a piston displacement of 14 L (850 cu in.); the engine power, however, would be considered fairly low, as typical truck engines currently range from 223 kW to 485 kW (300 hp to 650 hp).

The test vehicle and trailer was composed of a single trailer with six axles, and thus would be classified as vehicle class 10 using the Federal Highway Administration (FHwA) classification. The front axle was a single axle, the tractive axle was a dual axle, and the trailer axle was a triple axle. The truck did not have any aerodynamic fixtures and used radial tires.

STUDY SECTION DESCRIPTION

The study section that was considered included a 1.5-km (0.9-mi) section of the Smart Road test facility at the Virginia Tech Transportation Institute. Currently, the Smart Road is a 1.5-km (0.9-mi) roadway that will be expanded to a 3.2-km (2-mi) experimental highway in southwest Virginia that spans varied terrain, from in-town to mountain passes.

The horizontal layout of the test section is fairly straight with some minor horizontal curvature that does not impact vehicle speeds. The vertical layout of the section demonstrates a substantial upgrade that ranges from 6% at the leftmost end to 2.8% at the rightmost end, as illustrated in Fig. 3. In constructing the vertical profile of the test section the elevation of

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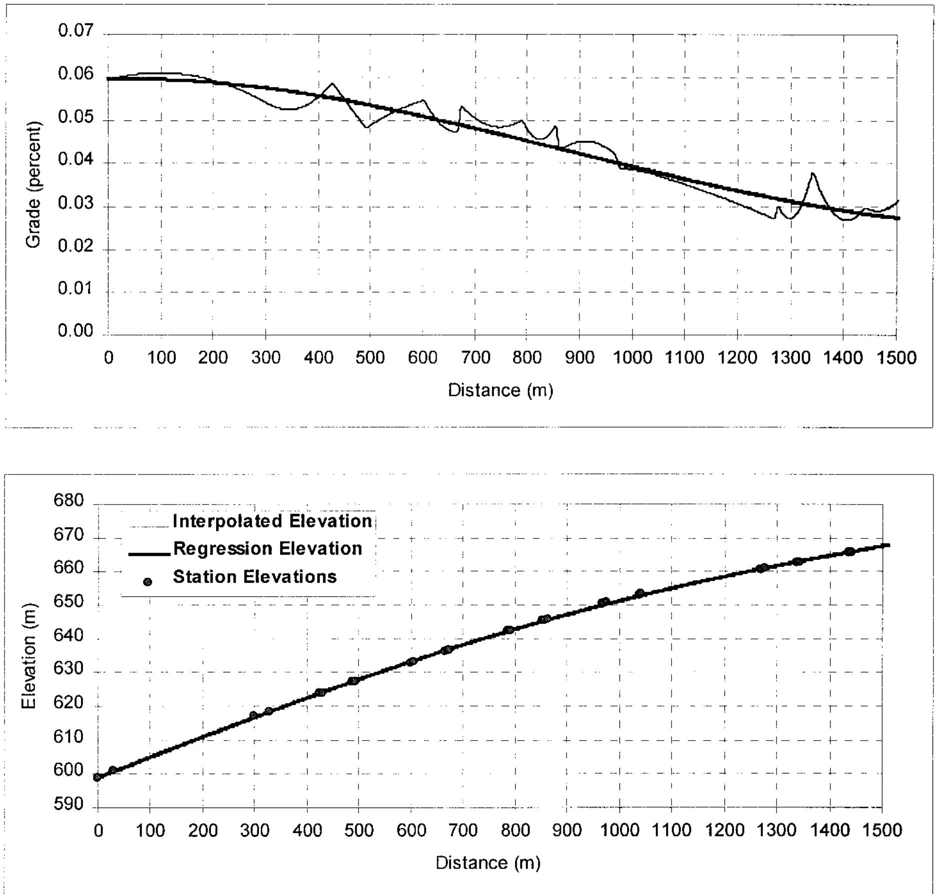

FIG. 3. Smart Road Test Section Longitudinal Profile

35 stations were surveyed, as indicated by the diamond symbols in Fig. 3. The vertical profile of the test section was then generated by interpolating between the station elevations using a cubic spline interpolation procedure at 1-m (3.28-ft) increments. The cubic spline interpolation ensures that the elevations, the slopes, and the rate of change in slopes are identical at the boundary conditions (in this case every meter). The grade was computed for each 1-m (3.28-ft) section and found to vary considerably, as illustrated in Fig. 3 (thin line). A polynomial regression relationship was fit to the grade data (R^2 of 0.951) for two reasons. First, to ensure a smooth transition in the roadway grade while maintaining the same vertical profile. Second, to facilitate the solution of the ODE because it ensures that the grade function is continuous. The modified grade and vertical elevation, which are illustrated in Fig. 3 (thick line), demonstrate an almost identical vertical profile with much smoother grade transitions when compared to the direct interpolation.

Apart from a 150-m (492-ft) segment of the roadway that was a rigid pavement, the entire roadway surface was asphalt. Consequently, a rolling resistance coefficient for asphalt pavement was only utilized. In addition, it should also be noted that the quality of the road surface was good when the test runs were conducted. These factors are important in identifying the road surface rolling resistance coefficients, as will be discussed later.

TEST RUN EXECUTION

In an attempt to validate the dynamics model and the parameters that have been proposed in the literature, a test truck was driven along the Smart Road test facility. This section describes how vehicle speeds were measured and the specifics of the test run execution.

Speed Measurement

The vehicle was equipped with a Global Positioning System (GPS) unit that measured the vehicle speed to an accuracy of 0.1 m/s (0.305 ft/sec). GPS is a worldwide, satellite-based radio-navigation system that can determine with certain accuracy the position and velocity of any object equipped with a GPS receiver.

Typical output data from GPS receivers include latitude, longitude, altitude, speed, heading, and time. GPS receivers can also typically update these parameters once every second. Nominal position accuracy is specified with a 25-m (82-ft) spherical error probability, while nominal velocity accuracy is specified with a 0.1 m/s (0.31 ft/sec) error probability. These inaccuracies are attributed to a number of sources of error. The majority of the errors are linked to the way the distance between a satellite and a GPS receiver is measured. Within the system, distances are measured by calculating the time it takes for a signal to travel between a satellite and a receiver. Consequently, any delay in the signal transmission then results in distance overestimation and inaccuracies in the estimated position of an object.

Test Run Description

The test runs involved accelerating at the maximum possible acceleration rate from a complete stop at the start of the test section. The acceleration continued until the end of the test section, over the entire 1.5-km (0.9-mi) section.

In an attempt to alter the mass-to-power ratio, a total of 10 mass configurations were analyzed using the same test truck. The truck and trailer mass was altered by progressively reducing the number of concrete blocks on the trailer from 9 to 0 blocks, with each block weighing approximately 2,400 kg

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TABLE 5. Characteristics of Load Configurations Analyzed

Load configuration	Axle Mass (kg)			Gross vehicle mass (kg)	Mass/power ratio		$perc_{ta}$
	Axle 1	Axle 2	Axle 3		(kg/kW)	(lb/hp)	
0	4,401	7,904	9,904	22,208	85.16	140	0.356
1	4,388	9,327	10,902	24,617	94.28	155	0.379
2	4,683	10,502	11,845	27,029	103.41	170	0.389
3	4,657	11,360	13,424	29,440	112.53	185	0.386
4	4,919	11,951	14,981	31,852	122.27	201	0.375
5	5,174	13,477	15,612	34,263	131.39	216	0.393
6	5,103	15,035	16,537	36,675	140.51	231	0.410
7	5,107	15,713	18,267	39,086	149.64	246	0.402
8	5,549	15,892	20,058	41,498	158.76	261	0.383
9	5,085	16,269	22,555	43,910	168.50	277	0.371

TABLE 6. Example Solution to ODE (Load Configuration 9)

Time (s)	Distance (m)	V (km/h)	A (m/s^2)	F (N)	Grade	R_a (N)	R_r (N)	R_g (N)	R (N)
0.0	0.00	0.00	1.64	56,001	0.0596	0.0	140.9	14,688.8	14,829.7
1.0	0.00	5.90	1.64	56,001	0.0596	13.0	156.1	14,688.8	14858.0
2.0	1.64	11.80	1.64	56,001	0.0596	52.1	171.4	14,690.2	14,913.7
3.0	4.92	17.69	1.26	46,772	0.0596	117.1	186.6	14,692.6	14996.4
4.0	9.83	22.24	0.88	37,194	0.0597	185.2	198.4	14,696.0	15,079.6
5.0	16.01	25.41	0.69	32,255	0.0597	241.8	206.6	14,699.5	15,147.9
6.0	23.06	27.90	0.57	29,644	0.0597	291.6	213.0	14,702.8	15,207.4
7.0	30.81	29.97	0.49	27,598	0.0597	336.5	218.4	14,705.3	15,260.1
8.0	39.14	31.74	0.43	26,061	0.0597	377.3	223.0	14,706.7	15,307.0
9.0	47.96	33.28	0.38	24,854	0.0597	414.9	227.0	14,706.9	15,348.8
10.0	57.20	34.64	0.34	23,877	0.0597	449.5	230.5	14,705.7	15,385.7

(5,300 lb). Axle weights were recorded prior to conducting the test runs using General Electrodynamics Corporation (GEC) weigh scales with an advertised accuracy of 98%. In summary, the study involved varying the mass of the truck and trailer from 22,200 to 43,900 kg (49,000 lb to 97,000 lb), as summarized in Table 5, with the mass-to-power ratio varying from 85 kg/kW to 168 kg/kW (140 lb/hp to 277 lb/hp).

In conducting the study, a minimum of 10 repetitions was executed for each load configuration in order to provide a sufficient sample size for the validation analysis.

MODEL VALIDATION PROCEDURES AND RESULTS

This section describes how the model and the proposed model parameters were validated and the results of the validation exercise.

Model Parameters

The proposed dynamics model was applied by setting the model parameters to reflect the truck, trailer, altitude, tire, and pavement conditions. Values for these parameters were obtained from the manufacturer's specifications or from the literature (Fitch 1994; SAE J2188 1996). These parameters included:

1. A horsepower of 261 kW (350 hp), assuming that the engine operates at maximum power over the entire section (to compute maximum acceleration levels).
2. The percentage mass on the tractive axle was computed as the percentage mass on axles 2 and 3 relative to the entire truck and trailer mass, as shown in Table 5.
3. The power efficiency (η) was set to 0.94 (single drive) as suggested in the literature.
4. The air drag coefficient (C_d) was set to 0.78 because no aerodynamic aids were on the truck roof.
5. The coefficient of friction (μ) was set to 0.6 and the rolling coefficient (C_r) was set to 1.25 to account for a good asphalt pavement surface.
6. The rolling resistance coefficients were set to 0.04375 and 4.575, respectively to account for radial tires on the truck and trailer.
7. The frontal area was set to 10.7 m^2 (115.2 sq ft) for a semi-trailer van type body whose width is 2.60 m (8.5 ft).

Model Results

Given that the ODE that is presented in (11) is a second-order ODE it can be recast as a system of two first-order equations (an nth-order equation reduces to a set of n first order equations), as demonstrated in (12). These ODEs are solved using a first-order Euler approximation, as demonstrated in (13) and (14). Table 6 summarizes the solution of the ODE system for the first 10 s of the trip. It should be noted at this point that a higher-order solution to the ODE was not feasible because the first derivative of the acceleration function was not continuous as explained earlier.

The procedures for solving the ODE are best described by illustrating how the various parameters were computed for the first 2 s of a test run. Using the initial condition of speed equal to zero [$v(t_0) = 0$], the tractive force, aerodynamic resistance, and rolling resistance were estimated using (1), (5), and (7). Using the initial condition of distance equal to zero [$x(t_0) = 0$] the grade was estimated using the polynomial grade function that was described earlier and the grade resistance was computed using (8). The maximum acceleration was then computed using (11).

At $i = 1$, the speed of the vehicle and location of the vehicle were estimated using (13) and (14), respectively using the acceleration at $i = 0$ ($t = t_0$). Again as was the case at $i = 0$, the tractive force, aerodynamic resistance, rolling resistance, and grade resistance forces were computed based on the speed and location after 1 s of travel. The acceleration at $i = 1$ was then estimated using

$$a(t_i) = \frac{F(t_i) - R(t_i)}{M} \quad (11)$$

where $t_i = t_0 + i\Delta t \quad \text{for } i = 1, 2, \ldots, n$

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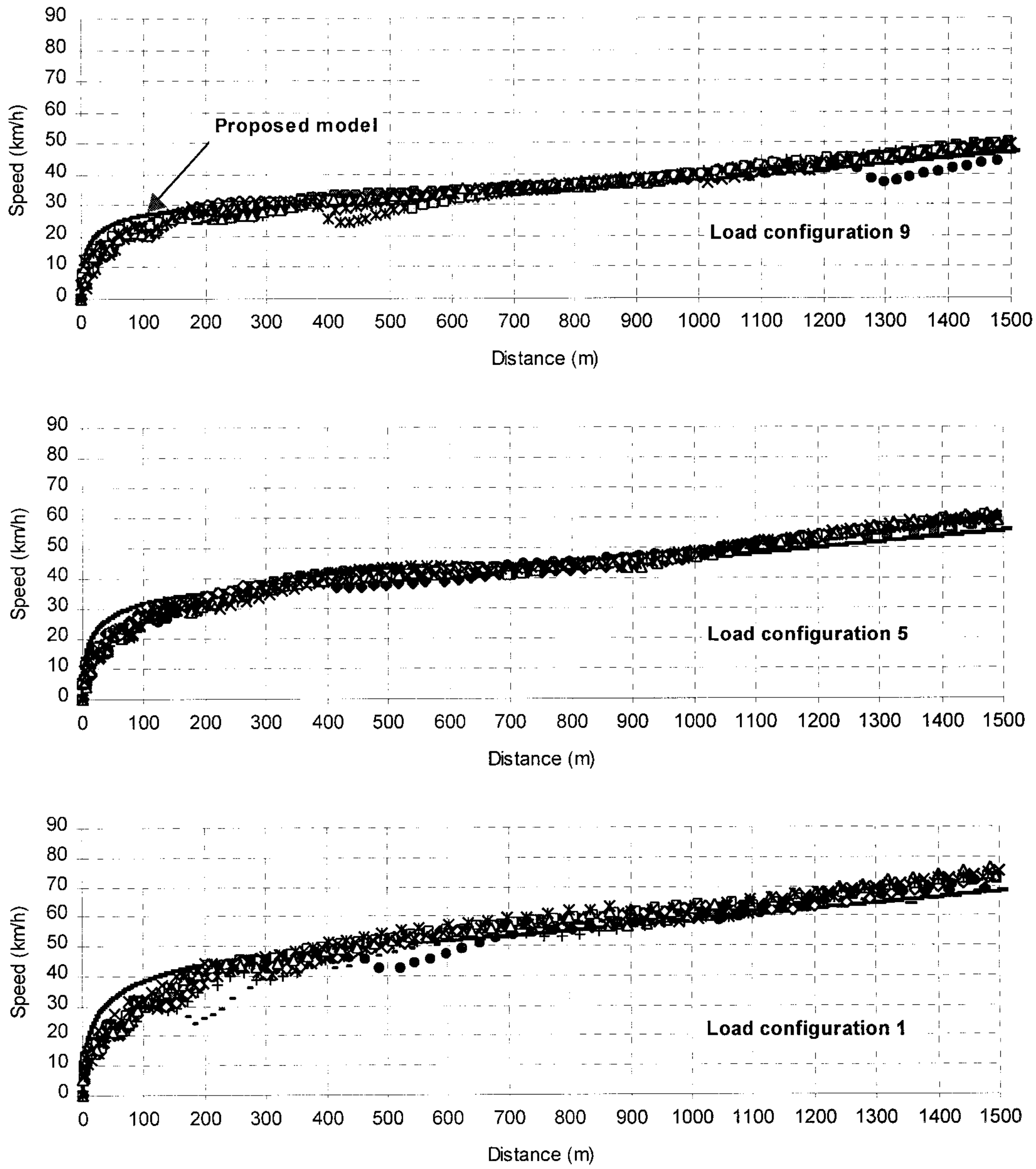

FIG. 4. Predicted and Observed Speed Profile for Selected Load Configurations

$$\begin{Bmatrix} \dot{v}(t_i) \\ \dot{x}(t_i) \end{Bmatrix} = \begin{Bmatrix} a(t_i) \\ v(t_i) \end{Bmatrix} \quad (12)$$

$$v(t_i) = v(t_{i-1}) + a(t_{i-1})\Delta t \quad (13)$$

$$x(t_i) = x(t_{i-1}) + v(t_{i-1})\Delta t \quad (14)$$

where Δt = duration of time interval used for solving the ODE (in this case 1-s duration); $F(t_i)$ = total resistance force at instant t_i; $a(t_i)$ = vehicle acceleration at instant t_i; $v(t_i)$ = vehicle speed at instant t_i; and $x(t_i)$ = vehicle location along test section at instant t_i.

The final speed profile from the model was superimposed on the field collected GPS second-by-second speed measurements, as illustrated in the upper graph of Fig. 4, which illustrates the variation in vehicle speed as a function of traveled distance for load configuration 9, which corresponds to a gross vehicle mass of 43,910 kg (96,800 lb). The fit indicates a maximum error in the range of 5 km/h (3.1 mi/hr) for the first 200 m (656 ft) with an error within the variability of the data for 200 m (656 ft) onwards.

The initial error as the truck accelerates from a speed of zero can be explained by the fact that while the truck initially accelerates the gear shifting behavior results in the vehicle operating at a power that is less than the maximum power of 261 kW (350 hp). The analytical model, on the other hand, assumes a constant power of 261 kW (350 hp) over the entire trip. The dips observed in the measured speed curves are also due to the shifting of gears, as there is virtually no power transmission to the tractive axles while the clutch is activated. Further enhancements to the model are being considered to capture the buildup of power as a vehicle accelerates from a complete stop and to include the effects of gear shifting.

Similar results are observed for other load configurations, however, only the results for the configuration 5 and 1 are presented due to lack of space. As illustrated in the other two graphs shown in Fig. 4, the error in the estimated speed versus the measured speed was found to be less than 10% at the conclusion of the 1.5-km (0.9-mi) section. These findings were consistent across the various load configuration tests.

UPDATED SAMPLE PERFORMANCE CURVES FOR DESIGN TRUCK

Having validated the proposed dynamics model, this section demonstrates the model applicability by developing performance curves for a 120 kg/kW (200 lb/hp) truck along grades ranging from 0 to 6%, as illustrated in Fig. 5. These curves

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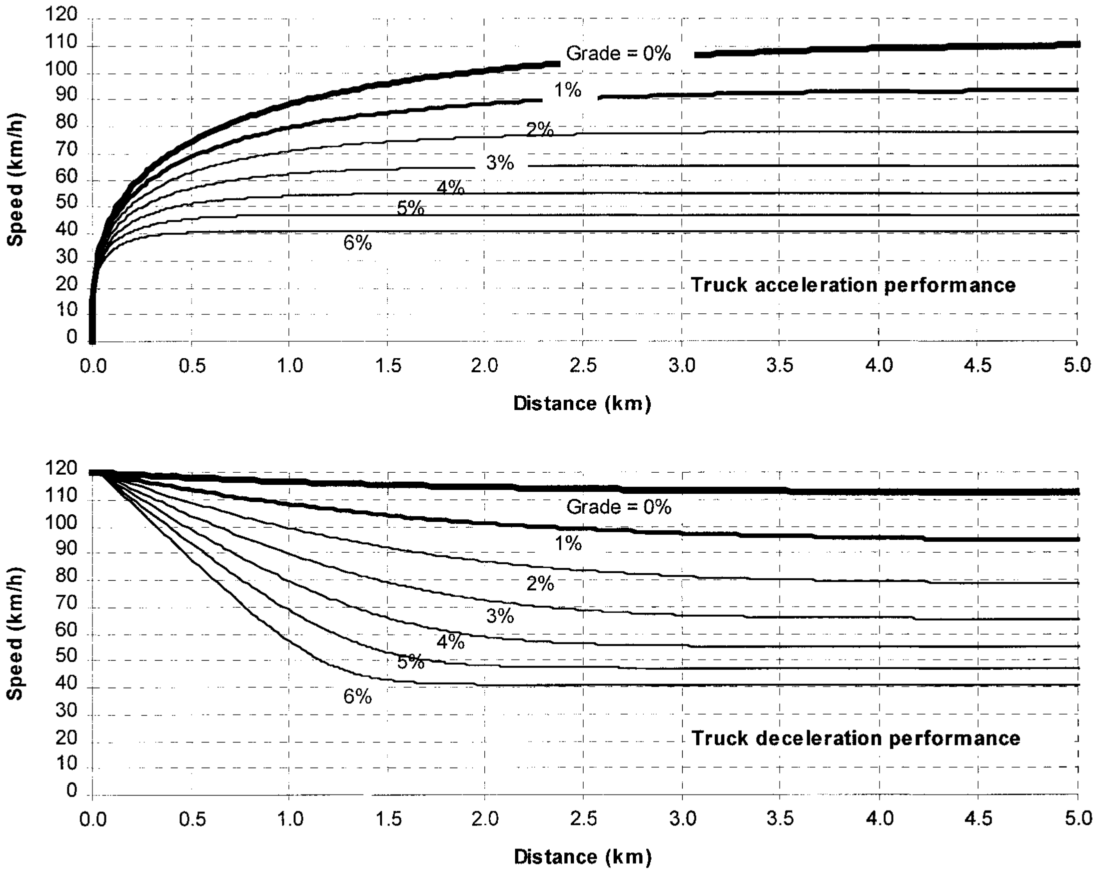

FIG. 5. Truck Performance Curves for NTC-350 Engine [120 kg/kW (200 lb/hp)]

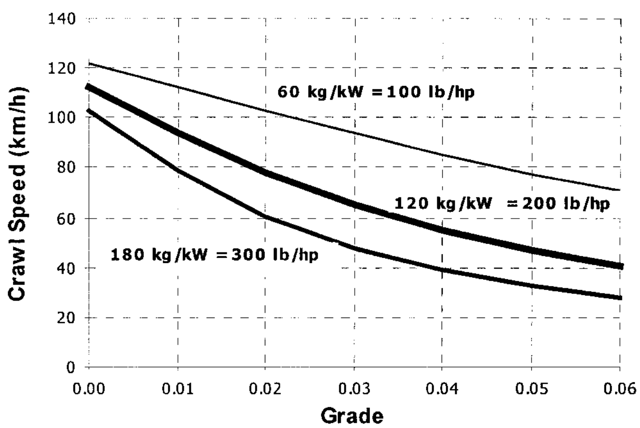

FIG. 6. Variation in Equilibrium Speed as a Function of Roadway Grade for NTC-350 Engine

can be used to update the truck performance curves currently used in the geometric design of highways, among other uses. It should be noted that the performance curves reflect a truck with no aerodynamic treatments using radial tires driving along a good surfaced asphalt pavement. Further research is currently under way to develop performance curves for different truck and roadway characteristics.

Fig. 5 demonstrates that the equilibrium speed for this design truck on a 0% grade is 112 km/h (70 mi/h). Furthermore, the equilibrium speed is very similar for both acceleration and deceleration curves along a 5-km (16,000 ft) section, and identical for longer sections when the equilibrium speed can be attained. Consequently, these curves overcome the first shortcoming of the HCM curves, namely the difference in equilibrium speeds depending on whether the vehicle is accelerating or decelerating. Identical equilibrium speeds is expected given that the equilibrium speed represents the speed at which the tractive effort equals the total resistance force, the equilibrium speed is independent of how the vehicle approaches the equilibrium speed (either accelerating or decelerating). The updated curves also provide higher equilibrium speeds than proposed by the HCM curves. For example, the 120 kg/kW (200 lb/hp) test truck has an equilibrium speed along a 1% upgrade slope of 98 km/h (61 mi/h) versus 88 km/h (55 mi/h) in the HCM. These higher equilibrium speeds are more reflective of the trucks in North America.

Finally, the model demonstrates a reduction in the equilibrium speed as the vehicle mass-to-power ratio increases and as the roadway grade increases, as illustrated in Fig. 6. Specifically, the equilibrium speed ranges from a high of 120 km/h (70 mi/h) to a low of only 30 km/h (19 mi/h) depending on the truck weight-to-power ratio and the roadway grade.

CONCLUSIONS AND RECOMMENDATIONS

Based on the field tests that were conducted on the Smart Road test facility it can be concluded that the proposed model and proposed model input parameters provide results that are consistent with field observations, presenting errors less than 10% and within 5 km/h (3.1 mi/h). The new performance curves could be used to update the curves that are currently used for the geometric design of highways and climbing lanes.

As in any research effort, further investigations are required to better establish the accuracy of the proposed models. These investigations will include:

1. Conduct similar field tests with alternative trucks to investigate differences in vehicle performance as a function of engine and truck characteristics.
2. Conduct similar field tests to establish the sensitivity of truck performance to type of pavement, to pavement con-

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dition, to type of vehicle tires, and to percentage weight on the tractive axle.
3. Establish a relationship that identifies the decay in vehicle power as it accelerates from a stop.
4. Develop curves similar to the HCM curves that reflect grade, truck, pavement, and tire conditions.

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NOTATION

The following symbols are used in this paper:

A = frontal area (m^2) (Table 1);
C_d = air drag coefficient (Table 2);
C_h = altitude coefficient;
C_r = rolling coefficient (Table 3);
c_1 = constant = 0.047285;
c_2, c_3 = rolling resistance constants (Table 4);
F = residual force acting on the truck (N);
F_{max} = maximum tractive force (N);
F_t = tractive force (N);
H = altitude (m);
i = grade magnitude (m/100 m);
M = vehicle mass on tractive axle; $M \times perc_{ta}$ (kg) (Fig. 2);
P = engine power (kW);
R = total resistance force (N);
R_a = air drag resistance (N);
R_g = grade resistance (N);
R_r = rolling resistance (N);
V = vehicle speed (km/h);
$perc_{ta}$ = percentage of vehicle mass on tractive axle (%) (Fig. 2);
η = power transmission efficiency (ranges from 0.89 to 0.94); and
μ = coefficient of friction between tires and pavement (Table 3).

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