VARIABLE POWER VEHICLE DYNAMICS MODEL FOR ESTIMATING TRUCK ACCELERATIONS

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ABSTRACT

The paper introduces the concept of a linearly increasing variable power to the basic vehicle dynamics model that has been proposed in the literature. The paper also calibrates the model to typical truck classifications 8 through 10 that travel along interstate highways in the US. The proposed enhancement is demonstrated to result in significant improvements in vehicle performance curves especially at low speeds when vehicles are engaged in gearshifts.

Key words: Vehicle dynamics, traffic modeling, truck modeling, traffic flow theory, vehicle acceleration.

INTRODUCTION

State-of-practice vehicle dynamics models assume vehicle power to be constant. This assumption results in an overestimation of vehicle acceleration levels at low speeds as will be demonstrated in the paper. Consequently, this research effort introduces the concept of variable power in order to enhance current state-of-the-art vehicle dynamics models in order to capture the build-up of power as a vehicle engages in gearshifts at low travel speeds. The proposed enhancement to the current state-of-practice vehicle dynamics model allows the model to reflect typical vehicle acceleration behavior more accurately.

Objectives of Research

The objectives of this paper are three-fold. First, the paper enhances the state-of-the-art vehicle dynamics model by introducing the concept of variable power. Second, the paper characterizes typical trucks along interstate highways in the US. Finally, the paper validates the proposed vehicle dynamics model for the typical US trucks.

Significance of Research

The significance of this research effort lies in the fact that it develops a simple vehicle dynamics model that captures vehicle acceleration behavior. This model can be incorporated within microscopic simulation software to capture vehicle acceleration behavior, which is critical in the modeling of environmental impacts of traffic. In addition, the model can be utilized to update the Highway Capacity Manual (HCM) truck performance curves in order to reflect advances in truck engines since the curves were developed.

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Paper Layout

The paper first describes the state-of-the-art vehicle dynamics model that was presented and validated by Rakha et al. (2001). The shortcoming of the current state-of-the-art model is presented and a proposed variable power model is introduced. The logic of the proposed enhancement and the derivation of the parameters of the model are described. Typical trucks along US interstate roadways are characterized, and the proposed model is validated using a combination of truck and load configurations that are typical of US interstate roadways. Finally, the findings and conclusions of the study are presented.

STATE-OF-PRACTICE VEHICLE DYNAMICS MODEL

This section first describes the state-of-practice vehicle dynamics model that was presented and validated by Rakha et al. (2001) before describing the proposed enhancement to the model in the subsequent section.

The model computes the maximum acceleration based on the resultant force, as indicated in Equation 1. Given that acceleration is the second derivative of distance with respect to time, Equation 1 resolves to a second-order Ordinary Differential Equation (ODE) of the form indicated in Equation 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 non-continuous function. Consequently, a first-order solution technique is inevitable, as will be described subsequently in the paper.

$$a = \frac{F - R}{M} \qquad (1)$$

$$\ddot{x} = f(\dot{x}, x) \qquad (2)$$

Tractive Force

The state-of-practice vehicle dynamics model estimates the vehicle tractive effort using Equation 3 with a maximum value based on Equation 4, as demonstrated in Equation 5. Equation 4 accounts for the friction between the tires of the vehicle's tractive axle and the roadway surface. The use of Equation 5 ensures that the tractive effort does not approach infinity at low vehicle speeds.

Equation 3 indicates that the tractive force F_t is a function of the ratio between the vehicle speed v and the engine power P. The model assumes the vehicle power to be constant and equal to the maximum potential power. The model considers 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 the fan, generator, water pump, magneto, distributor, fuel pump and compressor. The second source of power loss occurs in the transmission system. Typical transmission efficiencies are assumed to be constant and range from 0.89 to 0.94 depending on the type of transmission (SAE J2188, 1996). The maximum tractive force is a function of the proportion of the vehicle mass on the tractive axle. Typical axle mass distributions for different truck types and typical axle mass distributions were presented by Rakha et al. (2001).

$$F_t = 3600\,\eta\,\frac{P}{v} \qquad (3)$$

$$F_{\max} = 9.8066\,M_{ta}\,\mu \qquad (4)$$

$$F = \min(F_t, F_{max}) \quad (5)$$

Resistance Forces

State-of-the-art vehicle dynamics models consider three major types of resistance forces, including aerodynamic, rolling, and grade resistance as suggested in the literature (Mannering and Kilareski, 1990; Fitch, 1994; Archilla and De Cieza, 1999; Rakha et al., 2001). The total resistance force is computed as the sum of the three resistance components, as summarized in Equation 6:

$$R = R_a + R_r + R_g \quad (6)$$

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 Equations 7 and 8. The constant c_1 accounts for the air density at sea level at a temperature of 15°C (59ºF). Typical values of vehicle frontal areas for different truck and bus types and typical drag coefficients are provided in the literature (Rakha et al., 2001). Equation 8 is a linear approximation that was 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 to 5000 m (0 – 16400 ft).

$$R_a = c_1 C_d C_h A v^2 \quad (7)$$

$$C_h = 1 - 8.5 \times 10^{-5} H \quad (8)$$

Rolling Resistance

The rolling resistance is a linear function of the vehicle speed and mass, as indicated in Equation 9. Typical values for rolling coefficients (*C_r, c_1, and c_2*), as a function of the road surface type, condition, and vehicle tires, are provided in the literature (Rakha et al., 2001). Generally, radial tires provide a resistance that is 25 percent less than that for bias ply tires.

$$R_r = 9.8066\, C_r\, (c_2 v + c_3) \frac{M}{1000} \quad (9)$$

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 Equation 10. The grade resistance accounts for the proportion of the vehicle weight that resists the movement of the vehicle:

$$R_g = 9.8066\, M\, i \quad (10)$$

Model Application

The vehicle dynamics model was validated using a 1990 truck that was equipped with a Cummins NTC-350 engine with an engine power rating of 261 kW (350 hp) at an engine speed of 2100 rpm (Rakha et al., 2001). The test vehicle included a single trailer with a total of 10 axles; thus it would be classified as vehicle class 10 using the Federal Highway Administration (FHWA) classification. This section describes the validation effort in detail because a similar effort was made to validate the proposed enhancement to the model.

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 percent at the leftmost end to 2.8 percent at the rightmost end.

Test Run Execution

The vehicle dynamics model was validated by Rakha et al. (2001) using the parameters described in the literature with the test truck that was driven along the Smart Road test facility. 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/s). 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 ten 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 2400 kg (5300 lb). Axle weights were recorded prior to conducting the test runs using General Electrodynamics Corporation (GEC) weigh scales with an advertised accuracy of 98 percent. A minimum of 10 repetitions was executed for each load configuration in order to provide a sufficient sample size for the validation analysis.

Model Results

The ODE was solved to estimate the truck speed as it traveled along the study section. Given that the ODE presented in Equation 11 is a second-order ODE, it can be recast as a system of two first-order equations (an n^{th}-order equation reduces to a set of n 1^{st}-order equations), as demonstrated in Equation 12. These ODEs are solved using a first-order Euler approximation, as demonstrated in Equations 13 and 14. 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 two seconds 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 Equations 3, 7, and 9. Using the initial condition of distance equal to zero ($x(t_0) = 0$), the grade was estimated using a polynomial grade function of the test section and computed using Equation 10. The maximum acceleration was then computed using Equation 11.

At i=1, the speed of the vehicle and location of the vehicle were estimated using Equations 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 second of travel. The acceleration at i=1 was then estimated using Equation 11.

The final speed profile from the model was superimposed on the field collected GPS second-by-second speed measurements, as shown in FIG. 1 illustrates the variation in vehicle speed as a function of traveled distance for load configuration 1, which corresponds to a gross vehicle mass of 25,120 kg (55,380 lb). The fit indicates a good fit past the initial 400 m of travel. 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 (data points on plot) are also due to the shifting of gears, as there is virtually no power transmission to the tractive axles while the clutch is activated. The following section will describe a modification to the model to capture the buildup of power as a vehicle accelerates from a complete stop.

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

$$\text{Where : } \quad t_i = t_0 + i\Delta t \quad \text{ for } i = 1,2,...,n$$

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

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

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

PROPOSED MODEL ENHANCEMENT

This section describes the proposed enhancement to the vehicle dynamics model in order to capture the buildup of power as a vehicle engages in gearshifts. Initially, vehicle engine characteristics are presented in order to provide a background before describing the proposed enhancement.

Vehicle Engine Characteristics

Although a complete description of engine design is beyond the scope of this paper, an understanding of how engine output is measured and applied is important to the study of vehicle performance, which relates to the proposed enhancement to state-of-practice vehicle dynamics models. The two most commonly used measures of engine output are torque and power (Mannering and Kilareski, 1990). Specifically, Mannering and Kilareski define torque as "*the work generated by the engine (the twisting movement) and is expressed in units of newton-meters (Nm)*." In addition, they define power as "*the rate of engine work, usually expressed in kilowatts (kW) for engines*." This is related to the engine's torque by Equation 15.

$$P = \frac{2\pi TN}{1000} \quad (15)$$

FIG. 2 illustrates a sample torque-power diagram for a typical gasoline vehicle (Oldsmobile Aurora) and a typical diesel engine truck (NTC-350 engine). The figure clearly demonstrates the differences between an automobile and truck engine. The first difference is the higher power and torque curves that are associated with truck versus automobile engines. The second difference is the smaller range of engine speeds for truck versus automobile engines. Specifically, the engine speed for the Oldsmobile Aurora extends up to 6200 rpm while the engine speed for the truck extends to 2100 rpm. This difference signifies the fact that the larger the engine is, the slower the engine speed of operation, in order to reduce the significant stresses that build up in the reciprocating engine parts at high engine speeds. Therefore, most large engines (i.e., engines having a large bore and stroke) are governed or restricted to a lower engine speed than smaller bore engines. The third observation from the engine performance curves is that truck engine curves are typically reported to begin at an engine speed of 1000 to 1200 rpm. The slowest engine speed at which the engine will operate without stalling or fouling up averages about 500 to 800 rpm (Fitch, 1994). Typically, an idling engine operates in the 500 to 800 rpm speed range, at which the engine power is zero. Noteworthy from the figure is that the idle engine speed for the Oldsmobile is approximately 700 rpm with a power that is approximately 1 percent the maximum power.

The literature also indicates that gear shifting may occur during high idling, which is illustrated in the figure. The high idling typically occurs at an engine speed that is 100 to 200 rpm above the rated speed and results in an engine power of zero.

Because tractive effort for acceptable vehicle performance is typically higher at lower vehicle speeds, and because maximum engine torque is developed at fairly high engine speeds, the use of gasoline-powered engines requires some form of gear reduction. This gear reduction provides the mechanical advantage necessary for acceptable vehicle acceleration. With gear reductions, two factors determine the amount of tractive effort reaching the driving wheels. First, the mechanical efficiency of the driveline (i.e., the gear reduction devices including the transmission and differential) must be considered, which typically range between 5 and 25 percent. Second is the overall gear reduction ratio, which refers to the relationship between the revolutions of the engine's crankshaft and the revolutions of the road wheels. For example, an overall gear ratio of 4 to 1 means that the engine's crankshaft turns four revolutions for every one turn of the road wheel. In order to illustrate the concept of gear shifting, FIG. 3 illustrates the torque and power relationships for the four gears of the Oldsmobile Aurora. Gear shifting typically occurs at maximum torque, which does not necessarily coincide with the maximum power, as demonstrated in FIG. 3. Consequently, during gearshifts the vehicle operates at sub-optimal power.

Variable Power Adjustment Factor

While the proposed model does not include gear shifting, it does account for the major behavioral characteristics that result from gear shifting, namely the reductions of power as gearshifts are being engaged. Specifically, the approach uses a simple vehicle dynamics model that captures the more complicated gear shifting behavior while accounting for the buildup of power as a vehicle accelerates from a complete stop.

The modification that is proposed uses a variable power efficiency factor that is dependent on the vehicle speed, as opposed to the constant factor that is currently utilized in the model. The factor is a linear relation of vehicle speed with an intercept of $1/v_0$ and a maximum value of 1.0 at a speed v_0, as demonstrated in Equation 16. The intercept guarantees that the vehicle has enough power to accelerate from a complete stop. The adjustment factor is then multiplied by the vehicle power and incorporated in Equation 3 to compute the tractive force, as demonstrated in Equation 17.

The estimation of the variable power factor "β"requires the calibration two parameters, namely, the minimum power and the speed at optimum power. The proposed model assumes the minimum power to be a function of the optimum speed, as demonstrated in Equation 16. As will be demonstrated in a forthcoming section of the paper, these parameters were calibrated using four trucks with vehicle rated powers ranging from 260 to 375 kW (350 to 500 hp), each involving ten weight configurations. The calibration demonstrated that higher weight-to-power ratios required a lower optimum speed and a higher minimum power. The proposed power lower bound addresses this need for a variable parameter while maintaining a single calibration parameter.

FIG. 4 illustrates the variation in the vehicle power and the resulting vehicle acceleration by incorporating the power adjustment factor that is presented in Equation 16. As illustrated in the figure, the modification reduces the vehicle acceleration levels at the lower speeds while not altering acceleration behavior at higher speeds very slightly.

The calibration of the variable power factor involves calibrating the speed at which the vehicle power reaches its maximum (termed the optimum speed). The optimum speed was found to vary as a function of the weight-to-power ratio, as demonstrated in FIG. 5. The details of how this relationship was derived are discussed in the model calibration section; however, it is sufficient to note at this point that the relationship is a power relationship, as demonstrated by Equation 18.

$$\beta = \frac{1}{v_0}\left[1 + \min(v, v_0)\left(1 - \frac{1}{v_0}\right)\right] \quad (16)$$

$$F_t = 3600\ \beta\ \eta \frac{P}{v} \quad (17)$$

$$v_0 = 1164 w^{-0.75} \quad (18)$$

The linearly increasing power relationship as a function of vehicle speed results in a constant acceleration as a function of vehicle speed given that force (product of the vehicle mass and acceleration) is the first derivative of power with respect to speed and that the vehicle mass is a constant.

FIG. 6 illustrates the field observed and model predicted vehicle acceleration levels for the load configuration that were illustrated earlier. The figure clearly demonstrates fluctuations in vehicle accelerations between positive and negative values as the vehicle engages in initial gearshifts. The trend, however, involves fewer fluctuations as the vehicle speed increases as a result of the fewer gearshifts that are required. The model predictions appear to reflect the average acceleration behavior during the initial gear shifting and after the vehicle attains its maximum power.

The acceleration profile as a function of distance also demonstrates consistency between field data and model predictions. The vehicle typically attained maximum power within the initial 200 m (656 ft) of travel.

CHARACTERIZATION OF TRUCKS ON US INTERSTATE HIGHWAYS

In order to test the proposed enhanced vehicle dynamics model it was necessary to calibrate the optimum speed parameter using trucks that cover a typical range of weight-to-power ratios that are observed in the field. This section describes the characterization of in-field trucks that was necessary for the calibration effort.

The characterization of trucks along interstate highways was achieved by conducting a one-day survey at the Troutville weigh station along Interstate 81. Specifically, a random sample of 200 trucks was included in the survey, of which 157 were classified as vehicle class 9. FIG. 7 clearly demonstrates that the distribution of truck weights for the sample that was included in the survey was very similar to typical truck weight distributions along I-81. It should be noted that a full characterization of the trucks along I-81 is beyond the scope of this paper; however, a forthcoming paper will deal specifically with the characterization of the trucks.

In conducting the survey, the weight of the sample trucks was obtained from the static scales at the weigh station. In addition, information on the engine power, type of tires, aerodynamic features, and type of transmission were obtained through a questionnaire that was conducted with the truck drivers.

The survey demonstrated that the majority of class 9 trucks had conventional cabs (96 percent), 97 percent of the trucks had radial tires (the remainder 3 percent surveyed were not sure what tires they had), and 96 percent of the trucks were manual with only 4 percent being automatic. In terms of aerodynamic features, 55 percent of the trucks had full aerodynamic features, 15 percent had partial aerodynamic features, while the remaining 29 percent had no aerodynamic features. The weight of the trucks ranged from 6800 kg (10000 lbs) to 38000kg (90000 lbs), as illustrated in FIG. 7. The truck powers ranged from a minimum of 150 kW (200 hp) to a maximum of 450 kW (600 hp) with the majority of trucks falling in the 350hp to 550hp range (82 percent of the truck sample). The average truck power was 435 hp with a standard deviation of 70 hp. Consequently, the 95 percent confidence limits ranged from 300hp to a 575hp. The weight-to-power ratio for the sample ranged from a minimum of 25 kg/kW (44 lb/hp) to a maximum of 150 kg/kW (240 lb/hp) with a mean of 80 kg/kW (130 lb/hp) and a standard deviation of 25 kg/kW (45 lb/hp). Consequently, the 95 percent confidence limits ranged from approximately 30 to 130 kg/kW (40 to 220 lb/hp).

MODEL CALIBRATION

Having characterized the trucks along I-81, the next step was to calibrate the proposed enhanced model for trucks that are consistent with field characteristics. This section describes how the optimum speed was calibrated using four trucks with different weight configurations. The characteristics of the trucks are summarized in TABLE 1. Each of these four trucks was tested for at least ten weight configurations along the Smart Road test facility, as summarized in TABLE 2. In addition, two of the trucks were tested without a trailer in order to cover the fairly low weight-to-power ratios that were observed in the field. The test scenarios covered a range of weight-to-power ratios from 24 kg/kW to 171 kg/kW (39 lb/hp to 282 lb/hp), which covers a wider range than was observed in the field. The truck powers ranged from a minimum of 260 kW (350 hp) to a maximum of 375 kW (500 hp), which does not cover the full range of the sample that was observed in the field but does cover a substantial range (85 percent of the field observations). It should also be noted that the trucks included engines that were developed by different engine manufacturers for different model years. All trucks were equipped with radial tires and all trucks were manual, which is consistent with the survey data that were gathered.

The execution of the test runs was identical to what was described previously in the paper where drivers accelerated from a full stop at the maximum possible rate as they traveled up the grade along the test roadway. The model parameters were selected based on recommendations by Rakha et al. (2001) to reflect the characteristics of the trucks in terms of their aerodynamic features, tire characteristics, and pavement conditions. Multiple repetitions were made for each weight-to-power ratio combination in order to ensure that the runs were not biased by a single observation. The test scenarios included a total of 42 weight-to-power combinations, as summarized in TABLE 2. For each scenario, the system of two first-order ODEs was solved using a first-order Euler approximation, as presented in Equations 13 and 14.

The estimation of the optimum speed for each weight-to-power scenario can be formulated as a constrained non-linear optimization problem, as demonstrated in Equation 19. The objective function is to minimize the sum of squared error between the estimated and observed speeds for each observation *i*. The estimated vehicle accelerations are computed using the variable power vehicle dynamics model and solving for the optimum speed. The constraints ensure that the estimated speeds and distances traveled satisfy the system of first-order ODEs in addition to the non-negativity constraints. Using the formulation of Equation 19, the optimum vehicle speed was calibrated as a function of the truck weight-to-power ratio by minimizing the speed error between the model estimates and the average observed speed estimates, as summarized TABLE 2.

The calibrated optimum speed was found to decrease as a function of the weight-to-power ratio in a consistent fashion for the various trucks, as illustrated in FIG. 5. A regression model was fit to the 42 observations to generate the relationship between the optimum speed and the weight-to-power ratio, as indicated in Equation 18.

FIG. 1 illustrates for the same sample runs, that the introduction of the variable power concept improves the accuracy of the vehicle dynamics model in estimating the truck speed profile. The overall average speed Root Mean Squared Error (RMSE) was significantly enhanced as a result of introducing the concept of variable power into the traditional vehicle dynamics model. Specifically, the average RMSE was reduced from 39.5 km/h to 7.5 km/h.

$$\begin{aligned} \min \quad & E = \sum_{i=1}^{n} \left(\tilde{v}_{\tilde{x}(t_i)} - v_{\tilde{x}(t_i)} \right)^2 \\ \text{S.T.} \quad & \tilde{v}(t_i) = \tilde{v}(t_{i-1}) + \tilde{a}(t_{i-1})\Delta t \\ & \tilde{x}(t_i) = \tilde{x}(t_{i-1}) + \tilde{v}(t_{i-1})\Delta t \\ & \tilde{x}(t_i), \tilde{v}(t_i) \geq 0 \end{aligned} \tag{19}$$

CONCLUSIONS AND RECCOMENDATIONS

In summary, the paper introduced the concept of a linearly increasing variable power to the basic vehicle dynamics model that has been proposed in the literature. The paper also calibrated the model to typical truck classifications 8 through 10 that travel along interstate highways in the US. The proposed enhancement resulted in significant improvements in vehicle performance curves especially at low speeds when vehicles are engaged in gearshifts.

It is recommended that further research be conducted in calibrating similar power relationships for other vehicle classes including automobiles, buses, and smaller trucks.

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APPENDIX NOTATION

The following symbols are used in this paper:

A	=	Vehicle frontal area (m2);
a	=	The maximum vehicle acceleration (m/s^2);
$a(t_i)$	=	Vehicle acceleration at instant t_i;
Cd	=	Vehicle drag coefficient;
Ch	=	Altitude coefficient;
Cr	=	Rolling coefficient;
c1	=	Constant (0.047285);
c2, c3	=	Rolling resistance coefficients;
F	=	Tractive effort (N);

F	=	Tractive effort effectively acting on truck (N);
F(t) $F(t_i)$	=	Total resistance force at instant t_i;
Fmax	=	Maximum tractive force;
Ft	=	Tractive effort (N);
H	=	Altitude (m);
i	=	Percent grade (m/100 m);
M	=	Vehicle mass (kg);
M	=	Vehicle mass (kg);
Mta	=	Vehicle mass on tractive axle (kg);
N	=	Engine speed or speed of the crankshaft (revolutions per second);
P	=	Engine power (kW);
P	=	Engine power (kW);
R	=	Total resistance force, which is the sum of the aerodynamic, rolling, and grade resistance forces (N);
R	=	Total resistance force (N);
Ra	=	Air drag or aerodynamic resistance (N);
Rg	=	Grade resistance (N);
Rr	=	Rolling resistance (N);
T	=	Engine torque (Nm);
V	=	Vehicle speed (km/h);
v	=	Truck speed (km/h);
v	=	Vehicle speed (km/h);
$v(t_i)$	=	Vehicle speed at instant t_i;
v0	=	Speed at which vehicle attains maximum power (km/h);
$v_{\tilde{x}(t_i)}$	=	Observed speed at the distance location of observation i;
$\tilde{v}_{\tilde{x}(t_i)}$	=	Estimated speed at the distance location of observation i;
w	=	Vehicle weight-to-power ratio;
x	=	Distance traveled by vehicle (m);
$\dot{x}$	=	First derivative of distance: speed (m/s);
$\ddot{x}$	=	Second derivative of distance: acceleration (m/s^2); and
$x(t_i)$	=	Vehicle location along test section at instant t_i.
α	=	Calibrated parameter that tends to zero (recommended value 0.01);
β	=	Variable power factor;
Δt	=	Duration of time interval used for solving the ODE (in this case 1-second duration);
η	=	Transmission efficiency;
μ	=	Coefficient of friction between tires and pavement;

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TABLE 1. Test Truck Characteristics

Engine power (kW/hp) (1)	Vehicle year (2)	Engine Manufacturer (3)	Engine RPM (4)	Number of gears (5)	Aerodynamic features (6)
260/350	1990	Cummins	2,100	9	None
320/430	1998	Detroit Diesel	1,800	10	Full
350/470	1997	Detroit Diesel	2,100	10	Full
375/500	1997	Detroit Diesel	2,100	10	Full

TABLE 2. Computed Optimum Speed as a Function of Truck Weight-to-power Ratio

Load Configuration	Test Truck Power (kW/hp)							
	260/350		320/430		350/470		375/500	
	W/P (kg/kW)	u0 (km/h)	W/P (kg/kW)	u0 (km/h)	W/P (kg/kW)	u0 (km/h)	W/P (kg/kW)	u0 (km/h)
(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)
No Trailer			27	100	24	110		
0	87	42	64	51	64	49	61	50
1	96	38	78	43	72	48	67	48
2	106	36	86	41	80	44	74	45
3	115	33	92	40	88	42	80	42
4	125	30	100	37	92	40	87	41
5	134	29	108	35	101	37	94	39
6	143	27	112	33	106	36	100	37
7	153	25	121	32	112	35	107	36
8	162	25	127	31	118	35	114	33
9	171	24	134	31	126	33	120	32

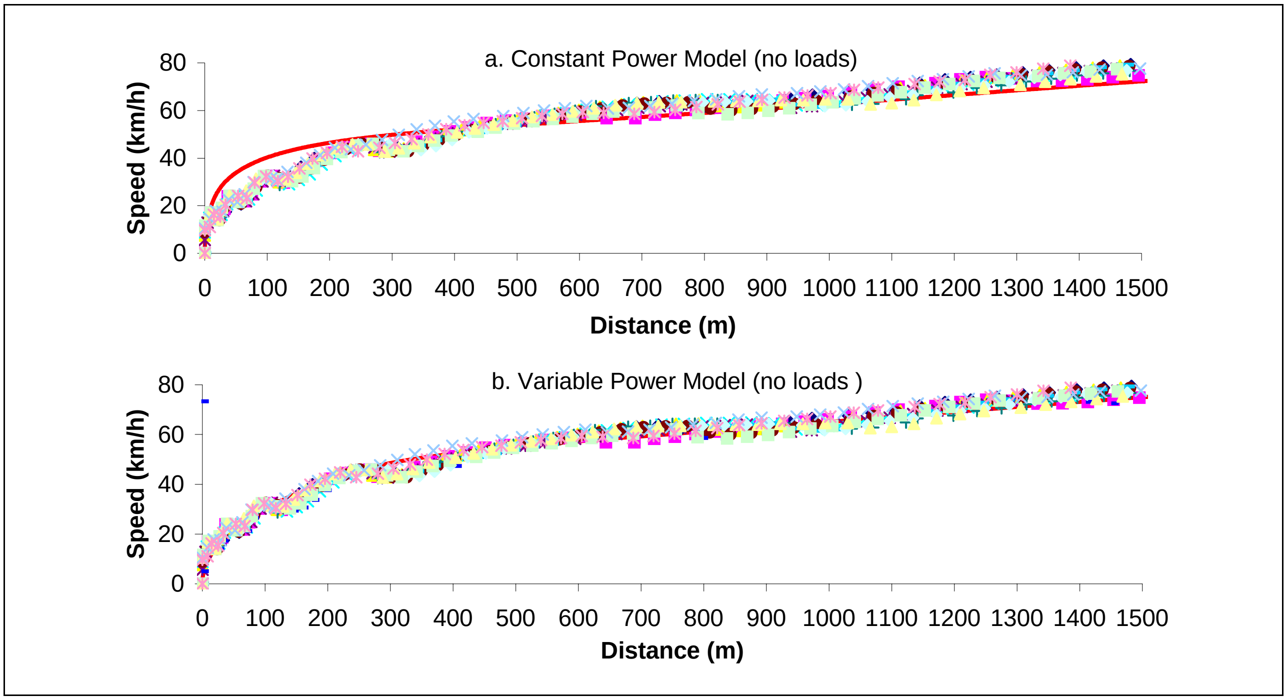


FIG. 1. Sample Speed Profile Validation Using the Constant Power Model (NTC-350 Engine)

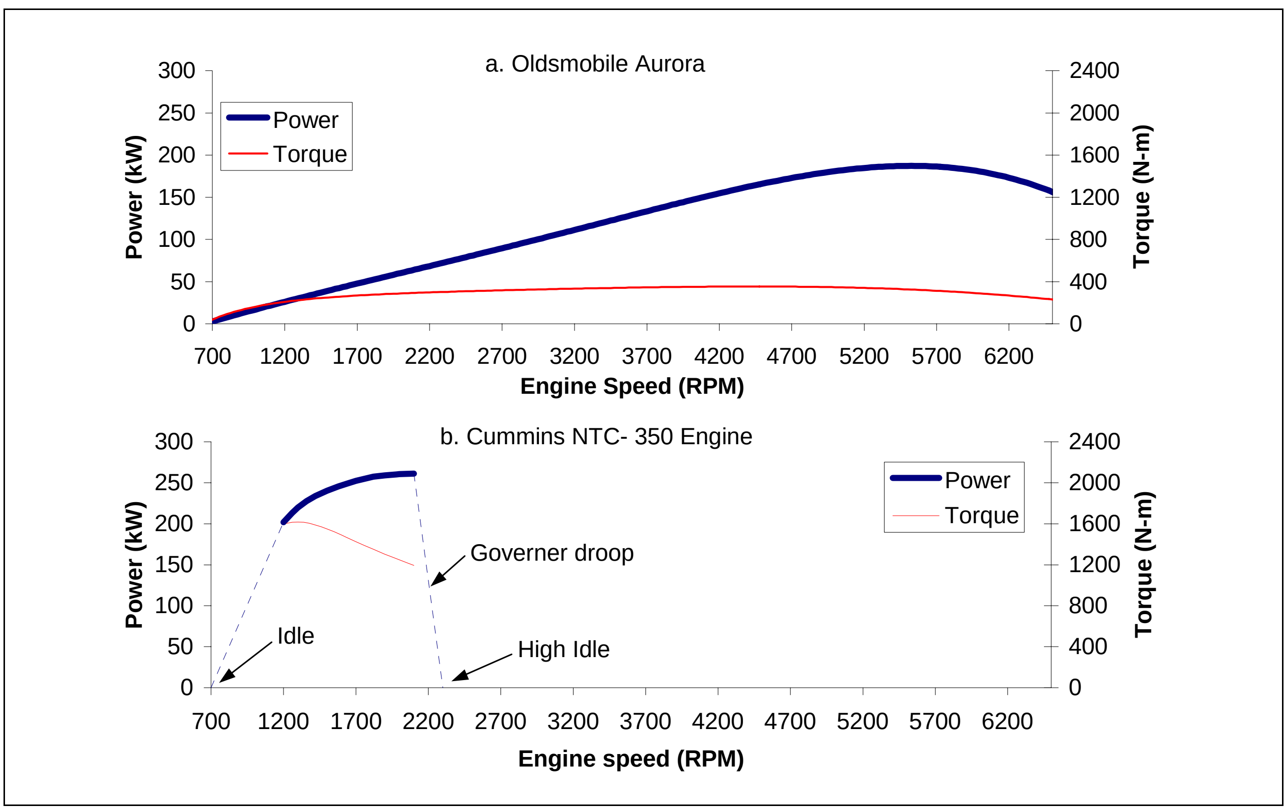


FIG. 2. Sample Torque-Power Curves for a Gasoline Powered Vehicle and a Truck

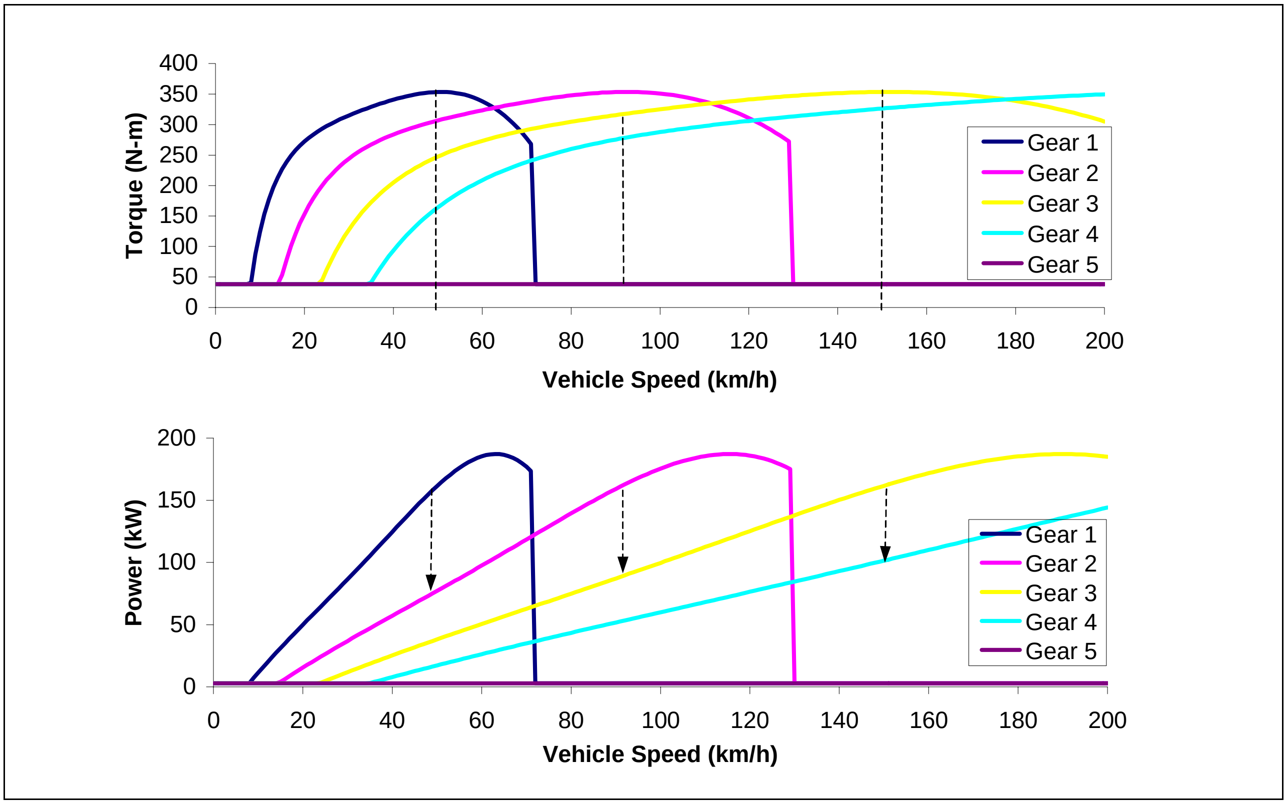


FIG. 3. Sample Torque and Power Curves for Different Gears (Oldsmobile Aurora)

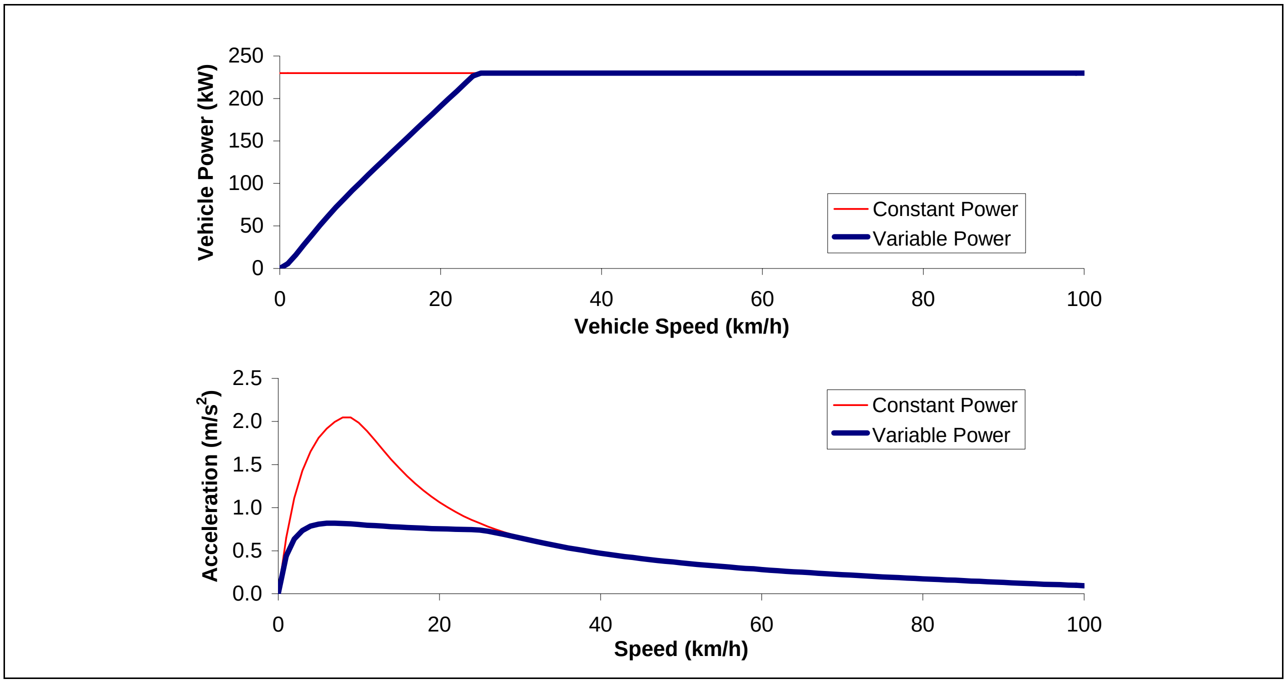


FIG. 4. Variation in Vehicle Power and Maximum Acceleration as a Function of Vehicle Speed

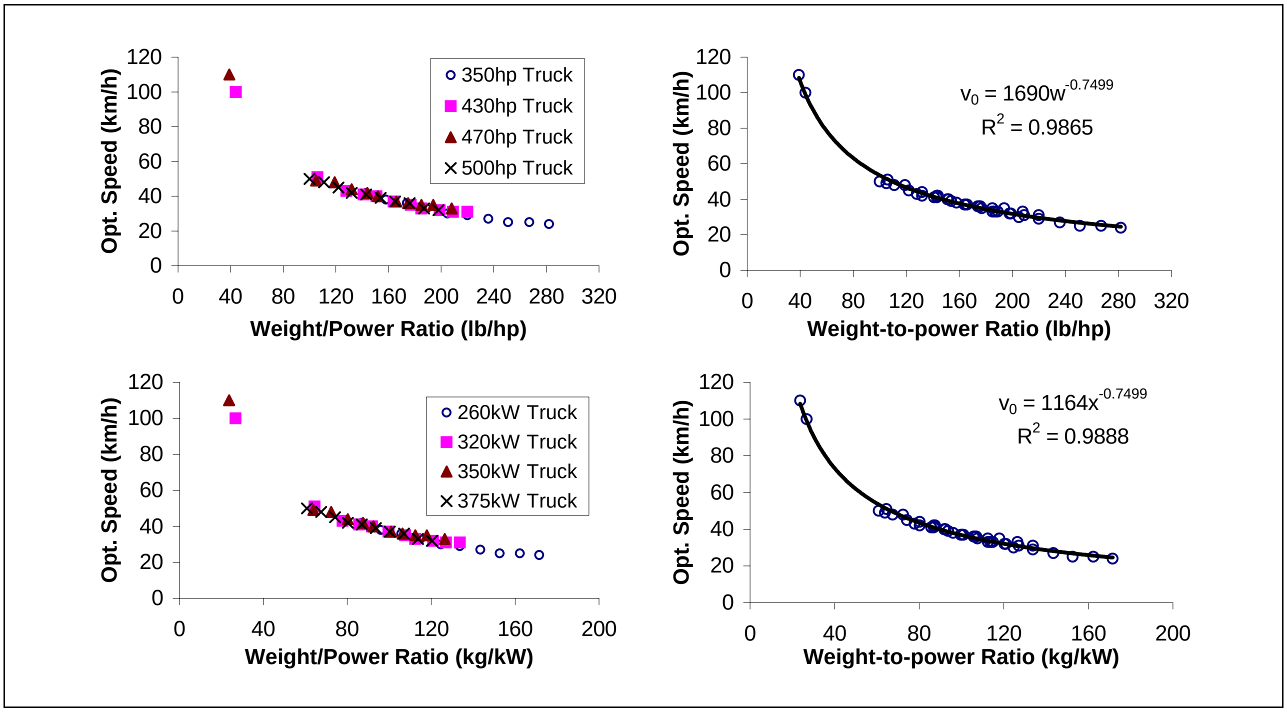


FIG. 5. Relationship between Speed at Optimum Power and Weight-to-Power Ratio

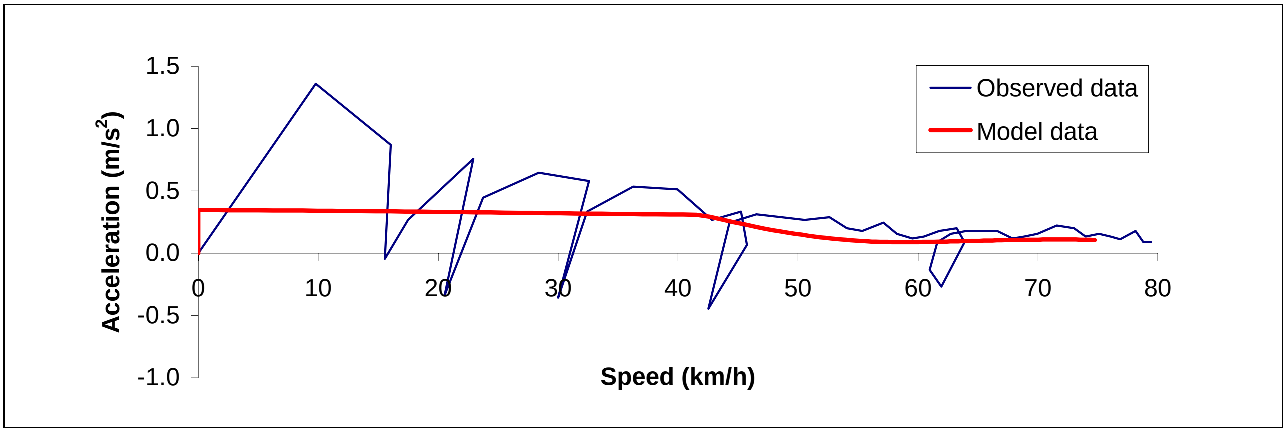
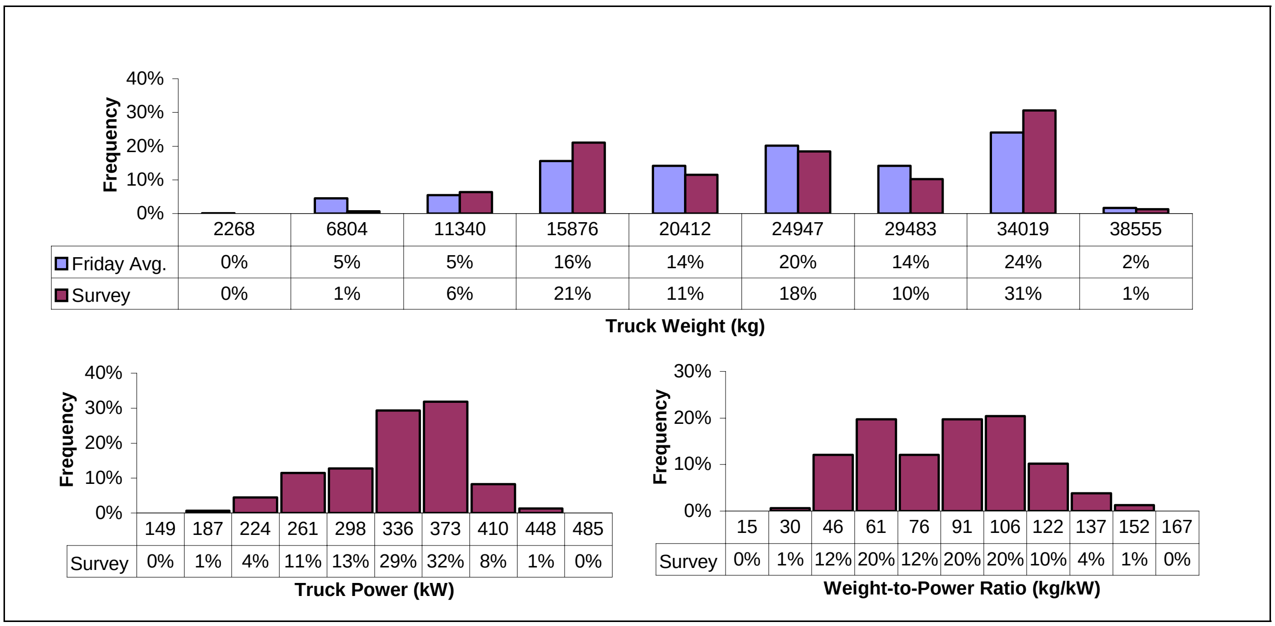


FIG. 6. Sample Observed Vehicle Acceleration versus Speed for 260kW (350hp) Truck



Truck Weight (kg)	2268	6804	11340	15876	20412	24947	29483	34019	38555
Friday Avg.	0%	5%	5%	16%	14%	20%	14%	24%	2%
Survey	0%	1%	6%	21%	11%	18%	10%	31%	1%

Truck Power (kW)	149	187	224	261	298	336	373	410	448	485
Survey	0%	1%	4%	11%	13%	29%	32%	8%	1%	0%

Weight-to-Power Ratio (kg/kW)	15	30	46	61	76	91	106	122	137	152	167
Survey	0%	1%	12%	20%	12%	20%	20%	10%	4%	1%	0%

FIG. 7. Truck Characterization at I-81 Troutville Weigh Station