VEHICLE SKIDDING ASSESSMENT THROUGH MAXIMUM ATTAINABLE CONSTANT SPEED INVESTIGATION

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ABSTRACT

In current road design practice, vehicle dynamics during cornering are addressed through the simplified point mass model where steady state cornering conditions are assumed. An existing vehicle dynamics model was utilized to define design parameters up to which steady state cornering conditions apply and consequently lift the restrictions of the point mass model.

Besides a passenger car, the motion of a two-axle truck was examined as well for both loaded and unloaded conditions, in order to quantify more accurately the potential safety hazard.

A range of design speed values paired with control design elements from AASHTO 2011 Design Guidelines as well as three values of peak friction coefficients (0.35, 0.50 and 0.65) were utilized in order to assess critical safety concerns in terms of vehicle skidding.

Many interesting findings in terms of design arrangements are reported for each vehicle type. However the unloaded truck was found to be the most critical vehicle in terms of reaching a maximum constant speed, termed as *safe speed*.

Furthermore, through a proposed statistical analysis, the authors provide a tool for practitioners in order to concurrently assess the impact of the horizontal alignment, grade and friction in terms of defining the vehicle's safe speed, and consequently take certain actions. These actions include the adoption of acceptable arrangements for the above values regarding new alignments, posted speed management for existing but also scheduling friction improvement programmes more accurately for both cases.

INTRODUCTION AND PROBLEM STATEMENT

It is well known that design speed, adopted by many Design Policies (e.g.1-4), is regarded as a key factor during the determination process of critical geometric elements. However, in terms of curve negotiation, a simplified approach has prevailed according to which steady state cornering conditions are assumed throughout the cornering process.

Moreover, the vehicle path is represented as a point mass, following the curve centerline, where the centrifugal force (responsible for the outward drift of the vehicle) is counterbalanced by the vehicle weight component related to the roadway's cross-slope, and the friction developed from the tire – road interaction. The fundamental equation representing the vehicle motion on a curve from which the minimum horizontal radius is derived is (Equation 1):

$$R_{min} = \frac{V^2}{127(f_{R,perm} + e_{max})} \qquad (1)$$

where R_{min} : minimum radius of curve (m)

V : vehicle speed – usually design speed- (km/h)

$f_{R,perm}$: available side friction factor

e_{max} : maximum superelevation rate (%/100)

In terms of potential safety violations, the above generalization actually fails to assess the impact of more critical road – vehicle parameters with special emphasis on their interactions. The most important deficiencies of the point mass model are summarized as follows:

- the acceleration effect during cornering is ignored
- key parameters of the vehicle such as type, mass and position of gravity (mass) center, loading, driving configuration, horse-power supply are disregarded as well which means that heavy vehicles dynamics are not taken under consideration
- the vehicle's motion is examined independently in the tangential and lateral direction of travel, although the respective friction components interact
- the utilized lateral friction is not an outcome of the actual demand but instead based on empirical vehicle accident considerations, and assumed as a fixed portion of the relevant peak (40%-50%)
- the roadway environment in terms of longitudinal design is assumed flat

Despite these simplifications, Harwood and Mason (5) evaluated the geometric design policy for horizontal curves in the 1990 AASHTO *Green Book* and concluded that regarding passenger vehicles, existing design policy provides adequate margins of safety against both skidding and rollover.

From the road design point of view, regarding the point mass sufficiency, many researchers have pointed out the necessity of more sophisticated models to simulate vehicle's cornering process (6-10) especially in cases where steep grades are present (11-15).

As far as the tractive mode is concerned, research studies (e.g. 11, 15-16) revealed that steep upgrades reduce the margin of safety. More specifically, one of these (15) performed on a roadway with combined sharp horizontal curves and steep longitudinal grades, revealed that steep upgrade road segments engage greater friction portions in the longitudinal direction of travel where as a result less friction is available in the lateral direction. The above finding combined with Equation 1, suggests that steep upgrade sections are more critical in terms of horizontal radii requirements.

The objective of the present paper is to assess the suggested by AASHTO 2011 horizontal design values during vehicle's motion in tractive mode on steep upgrades. For this purpose an existing vehicle dynamics model was utilized in order to define the parameters up to which steady state cornering conditions apply and consequently lift the restrictions of the point mass model stated above.

Moreover, besides a passenger car, the motion of a two-axle truck was examined as well in order to quantify the potential safety hazard for both loaded and unloaded conditions.

METHODOLOGY

A previous vehicle dynamics model developed by the authors (17-19) was utilized according to which the motion of any vehicle can be analyzed in three linear movements: longitudinal, lateral, and vertical, as well as three rotational movements: yaw, roll, and pitch.

The basic assumption of the present study is that vehicle motion is considered on a road surface, following the curve centerline, in which all three geometric parameters remain constant; namely, grade s, cross slope e, and horizontal radius R. All forces and moments applied to the vehicle are analyzed into a moving three dimensional coordinate system, coinciding at the vehicle gravity center and formed by the vehicle's longitudinal (X), lateral (Y) and vertical (Z) axis respectively. Through these axes, the influence of certain vehicle technical characteristics, road geometry and tire friction were expressed, such as vehicle speed/ wheel drive/ sprung and unsprung mass and it's position of gravity center/ aerodynamic drag/ vertical lift/ track width/ wheel-base/ roll center/ suspension roll stiffness/ cornering stiffness/ grade/ superelevation rate/ rolling resistance tire-road adhesion values and horse-power supply.

Thus with respect to the laws of mechanics, and after slight simplifications the following formulas express the equilibrium around each axis accordingly:

$$\sum X = 0$$

$$m\frac{dv}{dt} = \sum U_i - \sum S_i\theta_i + \frac{mv^2}{R}\beta - mgs - A_d \qquad (2)$$

$$\sum Y = 0$$

$$m\frac{dv}{dt}\beta = \sum U_i\theta_i + \sum S_i - \frac{mv^2}{R} + mge \qquad (3)$$

$$\sum Z = 0$$

$$\sum P_i = mg + \frac{mv^2}{R}e - A_n \qquad (4)$$

where (f=front, r=rear) :
dv/dt: vehicle's acceleration rate (positive value) (m/sec^2)
U_f , U_r: driving forces acting to front and rear axle respectively (Nt)
S_f , S_r : lateral forces acting to front and rear axle respectively (Nt)
P_f , P_r : vertical forces acting to front and rear axle respectively (Nt)
m : vehicle mass (kgr)
v : speed (m/sec)
A_n,A_d : air resistance forces acting vertically and on the frontal vehicle area respectively (Nt)
s : grade (%/100)
e : superelevation rate (%/100)
R: curve radius (m)
β : sideslip angle (rad)
θ : steer angle (rad)

The variables for the sideslip angle and the steer angle were taken from the literature (20). Furthermore the model takes into account the actual wheel load due to the lateral load transfer and the corresponding alteration of the lateral force on each wheel thus creating a four-wheel vehicle dynamics modelling (20-22).

In order to assess the ability of the vehicle to negotiate a curve in steady state cornering conditions and moreover define this maximum speed value, certain considerations should be further clarified. The following paragraphs provide a brief discussion on how these concerns were addressed. Further details are available through references (14) and (17-18).

The available tractive effort of the vehicle (driving force minus rolling resistance) acting on the front or rear axle (depending on the driving configuration) should be associated to the vehicle's speed as well the net power available at the driving wheels. Since a vehicle cannot always be driven at 100% of its available horse-power rate, the horse-power utilization factor (n), was utilized through Equation (5) as follows:

$$F_x = 745.6\frac{P}{v}n \qquad (5)$$

where :
F_x : tractive force (Nt)
P : net engine horse-power available at the driven axle (hp)
V : Vehicle speed (m/sec)
n : Horse-power utilization factor (%/100)

Taking moments about the front and rear vehicle axle and by using Equation (2) and Equation (5), the vehicle's longitudinal acceleration can be expressed as four degree polynomial equation, for which the parameters A through E are expressed as functions of vehicle technical characteristics and road geometry values as follows:

$$A\left(\frac{dv}{dt}\right)^4 + B\left(\frac{dv}{dt}\right)^3 + C\left(\frac{dv}{dt}\right)^2 + D\left(\frac{dv}{dt}\right) + E = 0 \qquad (6)$$

On the other hand, the expression according to which the pavement friction reserves are distributed to the longitudinal and lateral direction of travel is introduced by Krempel (23). During a curve negotiation, the portion of friction experienced in the longitudinal direction, is engaged by the friction demanded laterally and the following equation applies, the upper of which is known as impending skid conditions:

$$(\frac{f_T}{f_{T,max}})^2 + (\frac{f_R}{f_{R,max}})^2 \leq 1 \qquad (7)$$

where f_T : longitudinal friction demand
$f_{T,max}$: maximum longitudinal friction factor
f_R : side friction factor
$f_{R,max}$: maximum side friction factor

Finally by setting an increment rate for speed (0.25km/h in the present analysis) and adapting each time the horse-power utilization factor ‘n’ from Equation 5 at impending skid conditions (Equation 7), there is a certain value of speed which eliminates the vehicle’s acceleration impact as given through Equation 6 (dv/dt=0). This is the point where the vehicle’s maximum attainable constant speed is reached since it refers to impending skid conditions. However, it must be stressed that under the term “impeding skid conditions”, the model delivers data for the critical wheel, since not all wheels skid at the same time. This means that not necessarily vehicle skidding will occur; but instead a transition to an unstable vehicle motion is evidenced, which is in every case undesirable.

The model’s outputs were correlated against the known data derived by two other distinct cases: the final climbing speed of a truck travelling on a grade (14) and the output data from the well-known CARSIM Simulation Software (18). Both cases revealed a satisfying match.

PARAMETERS DESCRIPTION

The potential safety violation assessment for AASHTO 2011 design guidelines, in terms of horizontal design values were performed for a C-class mid-sized passenger car and a two-axle truck of 19t GVW, where at least from the vehicles’ dimensions points of view real cases are represented (KIA Proceed and Volvo FL7 - GVW 19.7t). Although an effort was made to provide the utilized vehicles’ parameters from the vehicle industry, most of them were taken from the literature (21) and (24) regarding the passenger car ant the two-axle truck respectively.

The two-axle truck was selected on the basis of the 120kg/kW climbing performance weight to horse power ratio (200lb/hp), as adopted in the AASHTO 2011 guidelines. Moreover the investigation involved both loaded and unloaded conditions.

The parameters from the vehicles point of view inserted in the model are shown in Table 1.

TABLE 1 Vehicles’ Parameters Inserted to the Model

	19t unloaded	19t loaded	C class pass. car	
L (m)	3.800	3.800	2,650	wheelbase
t_f (m)	2.012	2.012	1,538	front track width
t_r (m)	1.804	1.804	1,536	rear track width
m (kgr)	5855	19700	1300	vehicle mass
l_f (m)	1.226	2.508	1,161	position of gc from front axle
h (m)	1.200	2.013	0,620	position of gc from surface
$K_{\varphi f}$ (Nm/rad)	453711	453711	27502	suspension roll stiffness (front)
$K_{\varphi r}$ (Nm/rad)	453711	453711	14324	suspension roll stiffness (rear)
C_{af} (kp/rad)	13634.1	23026.0	2295.7	cornering coef. (front)
C_{ar} (kp/rad)	3247.0	22348.8	2120.7	cornering coef. (rear)
m_{uf} (kgr)	425	425	92	unsprung mass (front)
m_{ur} (kgr)	341	341	120	unsprung mass (rear)
h_{Rf} (m)	0,530	0,530	0.020	roll center height (front)
h_{Rr} (m)	0,530	0,530	0.410	roll center height (rear)
r_{dyn} (m)	0,500	0,500	0.290	dynamic radius (tire)
A_f (m^2)	6.188	6.188	1.850	frontal area
c_N	0,360	0,360	0.280	lift drag
c_d	0,900	0,900	0.360	aerodynamic drag
P (hp)	216.2	216.2	100	hp available on the wheels

The examined design speed values ranged over 50km/h, since lower design speeds result in limited performance especially for the truck case and concurrently sharper horizontal curves are combined with steeper grade values. The authors believe that such a research should be performed separately.

The control grade value for each examined design speed was in line with the roadways' functional classification as adopted in the Green Book. Table 2 illustrates the critical grades for each case. Finally, the superelevation rate was set to 6% for all the examined cases.

As far as pavement friction values are concerned, highway agencies in general perform measurements by means of locked wheel skid tests with a "standard" tire (25). These tests determine a value equivalent to the coefficient of sliding. The results of these tests are often multiplied by 100 and referred to as skid numbers rather than sliding coefficients of friction. Although skid numbers are usually determined at specific speed values [e.g. 65km/h (40 mph)], a procedure is available to determine the skid number at any speed (26-28). In general, the peak coefficients of friction exceed the sliding friction by 10%-45% varying with tire and pavement types (29). However in highway design the available side friction, utilized in Equation 1 for the R_{min} determination is considered to be a portion (40%-50%) of the related sliding coefficient in order sufficient friction to be present in the longitudinal direction of travel for any desired or undesired maneuvers. The above mentioned friction values referring to the AASHTO 2011 Design Guidelines are shown in Table 3.

Moreover, it is evident that the sliding friction coefficient and consequently the relevant peak value are subject to marginal variations in terms of wet-dry pavement conditions as well. For this reason, in the present study 3 values of peak friction coefficients were examined for all the design speed values; 0.35, 0.50 and 0.65 in order to assess pavements with poor friction performance under both wet (0.35) and dry (0.65) pavement conditions.

TABLE 2 Maximum Grade Values and R_{min} based on Road Type and Design Speed Values

Functional Classification	V_{design} (km/h)	R_{min} (e=6%) (m)	max grade (%)
Local Rural	50	79	14
	60	123	13
	70	184	12
	80	252	10

	90	336	10
Urban Collector	100	437	9
Rural Arterial	110	560	5
	120	756	5
	130	951	5

TABLE 3 Available Side Friction as well as Range of Peak Friction for Passenger Cars during Cornering based on AASHTO Design Guidelines.
Note: f_S : sliding friction coef., f_{Rperm} : available side friction coef.

V (km/h)	$f_{R,perm}$	Peak (unfavorable pavement) $f_{R,max}=1.10f_S$	Peak (favorable pavement) $f_{R,max}=1.45f_S$
60	0.17	0.37	0.49
80	0.14	0.34	0.45
100	0.12	0.33	0.44
120	0.09	0.31	0.41

MODEL'S OUTPUTS

Based on AASHTO 2011 Design Guidelines, for each pair of design speed value and the corresponding R_{min}, the maximum attainable constant speed value was determined, termed as *safe speed* for a range of grade values. The upper limits of the grade values utilized per pair of V_{design}, R_{min} are shown in Table 2, where the respective lower limits (per pair of V_{design} - R_{min}) where set on the basis of delivering a safety margin of 10km/h in the V_{safe} value ($V_{safe} \approx V_{design} + 10$). This process was performed for all 3 of the examined peak friction coefficients (0.35, 0.50 and 0.65). Since design speed values more than 90km/h didn't raise safety concerns for passenger cars, control alignments corresponding up to 90km/h were examined. As far as trucks are concerned, design speed values up to 90km/h were adequate since the truck speed limit is confined to 88km/h (55mph).

Passenger Car

Figure 1 illustrates this V_{safe} variation for the examined passenger car referring to poor friction pavement (f_{max}=0.35). More specifically the horizontal axis of Figure 1 is divided in 5 parts, where each one corresponds to a pair of control horizontal radius (R_{min}) and the related design speed value (V_d), marked with light orange. For each part the smaller bars indicate the V_{safe} values for the corresponding curve but for different grade values. Since for certain grade values these V_{safe} values are less than the respective design speeds, cases of safety concerns are raised which are marked with red. For example it can be seen that the vehicle while negotiating the control curve for the design speed of 50km/h cannot retain this speed for grade values greater than 11%. In other words, for the examined poor friction pavement (f_{max}=0.35), steady state cornering at 50km/h for the corresponding control radius (R=79m) is possible for grade values up to 11%, although the control grade value referring to local rural roads is 14%.

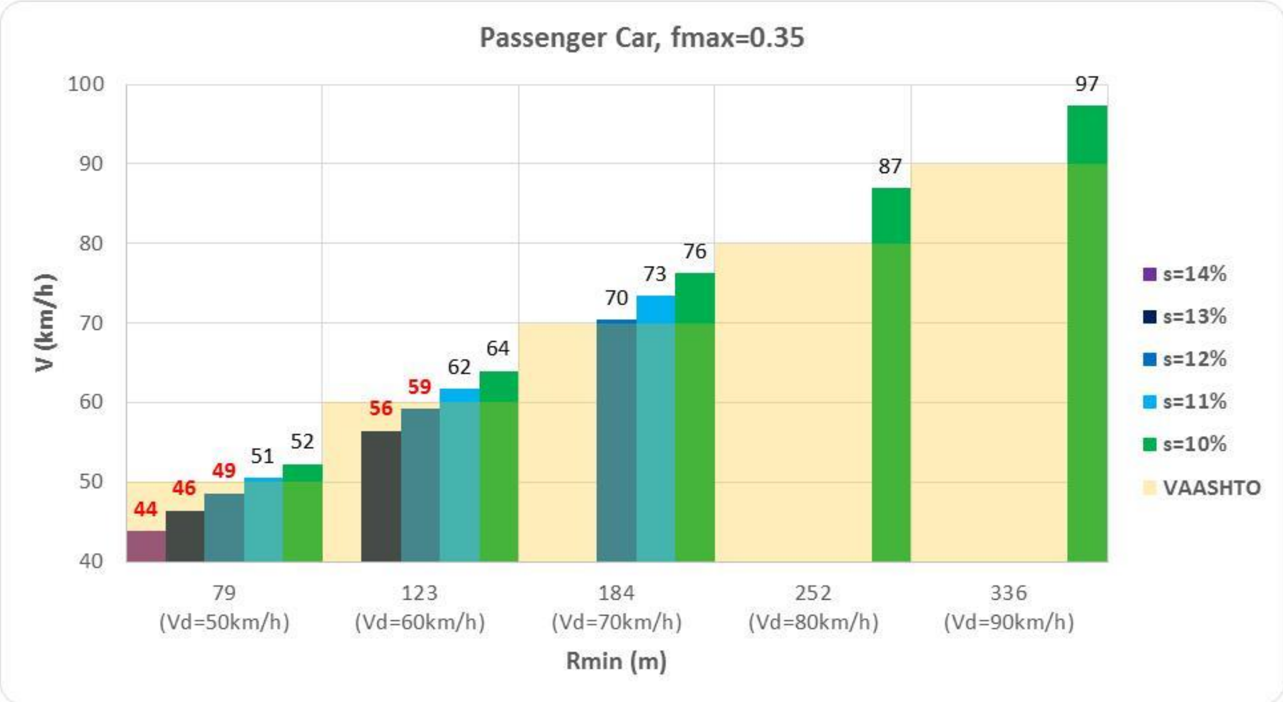


FIGURE 1 V_{safe} Variation for Passenger Car on Poor Friction Pavement (f_{max}=0.35).

The safety concerns, for the poor friction pavement (f_{max}=0.35), stated above are addressed through Figure 2, where the case of V_d=60km/h is further described. The V_{safe} values at the examined grade values are associated also to the relevant horse power utilization rates (n). It can be seen that up to 11% grade the vehicle may retain the constant speed value of 60km/h even if only 40.9% of the available horse power rate is utilized. However on curved alignments of R=123m and over 11% grade values, as the red line enters the shaded area, steady state cornering at 60km/h cannot be reached. If the driver attempts so, the vehicle will skid when the horse power utilization rate exceeds 41%.

Therefore such cases should be treated very cautiously either through posted speed management or friction improvement. However, it was found that regarding passenger cars, such cases are rather rare since only control alignments with grade values over 11% combined with poor friction pavements (f_{max}=0.35) are affected. In general for f_{MAX}=0.40, the current AASHTO 2011 control values as illustrated in Table 2 don't seem to raise any critical safety concerns.

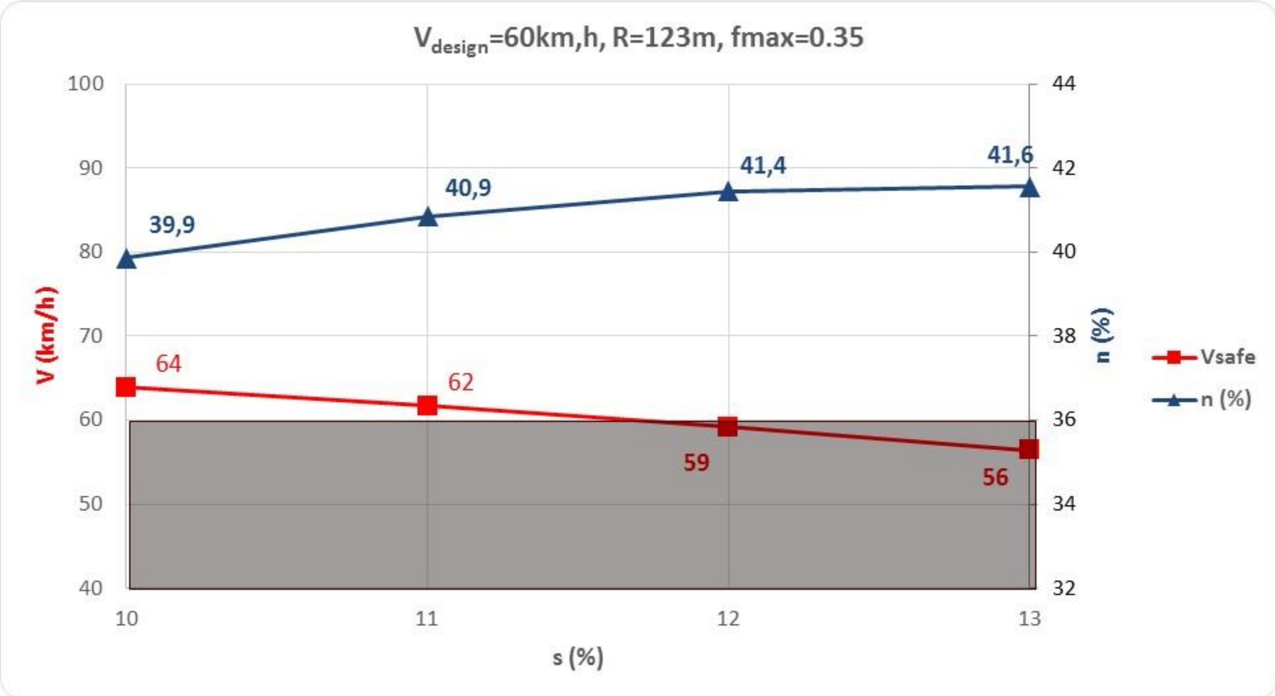


FIGURE 2 V_{safe} **Variation vs Grade Values for** V_d**=60km/h, R=123m, (**f_{max}**=0.35).**

Two-Axle Truck

As far as the two-axle truck investigation is concerned, rather different findings are reported. The same process was followed also in this case, and, as already stated through the parameters of Table 1, the safe motion of the truck was examined under both loaded and unloaded conditions.

Research on friction values for trucks (5) in the early 90's, has shown that truck tires can generate only about 70 percent of the friction of passenger car tires. However, since then, significant improvement has been achieved in this field from the tire industries. Therefore, the examined peak friction factors were once again set to 0.35, 0.50 and 0.65 respectively, but it should be underlined that these values refer only for trucks and may differ from the relevant values experienced by passenger cars.

Figure 3a shows the V_{safe} variation for the unloaded truck case assuming a moderate friction pavement of f_{max}=0.50. It can be seen that the utilized vehicle, for control alignments based on AASHTO 2011 Design Guidelines, seems to be insufficient in terms of maintaining the design speed for grade values greater than 9%. For example, for the control alignment corresponding to the design speed of 70km/h (R=184m), at 11% grade the maximum attainable safe speed is 68km/h (marked with red).

Another interesting finding in Figure 3a, for the peak friction coefficient of f_{max}=0.50, is that cases where the vehicle outperforms in terms of the horse power utilization rate are reported (n=100%). This is seen especially at high grade values combined with increased speed where the necessity for tractive force is more essential. Such cases are labeled with light blue and refer to the V_{safe} value for which the vehicle utilizes the total amount of the available horse power rate (P=216.2hp). From these speed values, the ones positioned below the orange area, which indicates the pair of control horizontal radius (R_{min}) and the corresponding design speed value (V_d), do not raise safety concerns at least in terms of vehicle skidding, since the vehicle cannot reach a greater constant speed.

In cases where the poor friction pavement (f_{max}=0.35) is utilized, the horse power utilization rate was always found to be below 100% (n<100%). The most important outcome for such cases is that safety, in terms of V_{safe} determination, is violated (V_{safe}<V_d) for all the control alignments regarding grade values over 4%.

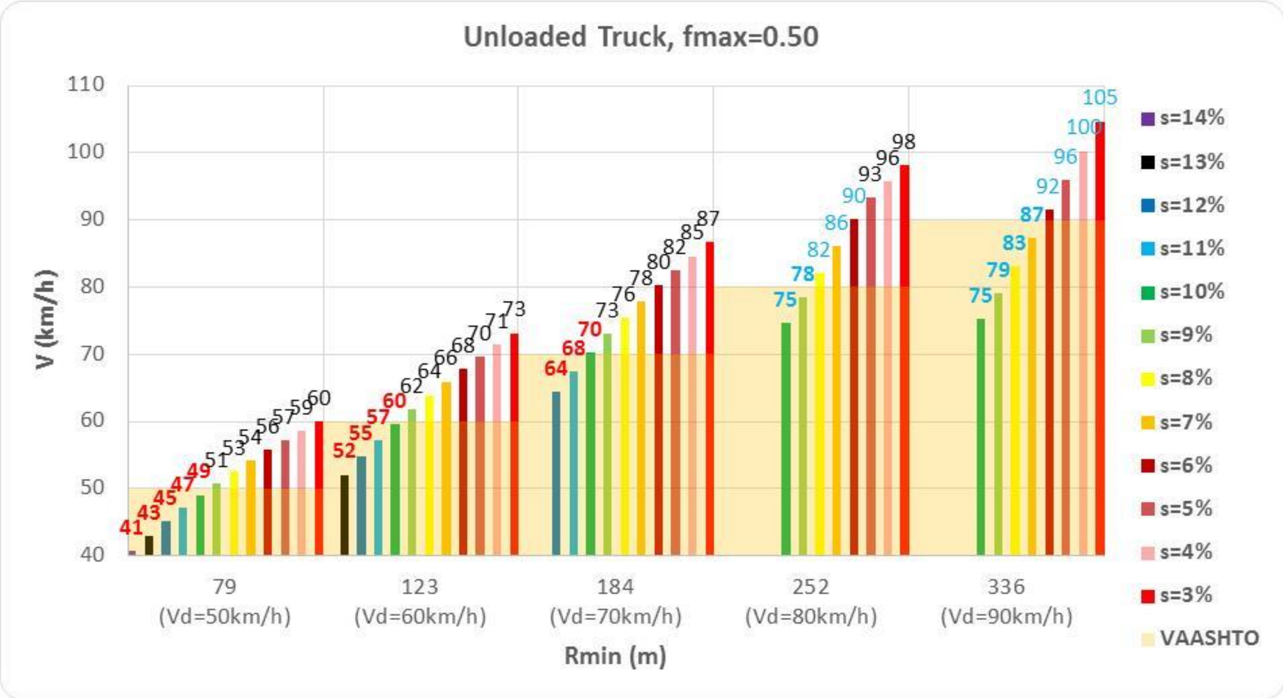
For pavement friction set to f_{max}=0.65, it was seen that no critical issues rise since even the control alignment suggested by AASHTO for V_d=50km/h, delivers V_{safe} speed value over 50km/h for the peak grade of 14% as well.

Regarding the loaded truck case, it was found that in general the vehicle utilizes 100% of the available horse power rate. Therefore, the pavement friction seems not critical since even for the poor friction case (f_{max}=0.35), the vehicle's available horse power utilization rate reaches n=100% in most cases.

More specifically for peak pavement friction values of 0.35 and 0.50, the same alignment arrangements where n=100% are delivered. In Figure 3b, the safe speed values for the alignments where n=100% are marked with light blue. Consequently, as stated again previously, for control alignments where the maximum attainable constant speed of the vehicle falls below the design speed (orange bars), no critical safety concerns rise at least in terms of vehicle skidding. However these cases are subject to further analysis from the operational point of view (e.g. define critical length of grade, etc.).

Another interesting finding is that grade values over approximately 4% actually dominate the loaded truck's performance in terms of defining the maximum attainable constant speed, which seems independent not only to friction but horizontal geometry as well. The explanation to this finding is closely linked to the fact stated above, according to which the vehicle's available horse power utilization rate reaches n=100%.

On the other hand, sharp horizontal curves, where the loaded vehicle's horse power utilization rate is below 100% (n<100%), were found to influence the speed of loaded heavy vehicles significantly on mild upgrades (s<4%) and as expected be dependent on friction.



(a) Unloaded Truck

(b) Loaded Truck

NOTE: Truck speed limit 88km/h (55mph).

FIGURE 3 (a,b) V_{safe} Variation for Unloaded and Loaded Two-Axle Truck (f_{max}=0.50).

STATISTICAL ANALYSIS

In current design practice, grade is not considered a dynamic design element during the determination of the minimum horizontal radius; instead an inferior parameter that affects road safety mainly from the operational point of view. Furthermore, road's available friction is another critical parameter that impacts greatly vehicle stability during cornering.

Regression analysis is carried out to interpolate the relationship of vehicle speed with certain control design parameters, and provide related formulae to practitioners for vehicle skidding assessments. The authors, besides the restrictions imposed from the horizontal alignment, aim to evaluate the interaction of grade and friction effect in terms of determining vehicle's safe speed. As a result, acceptable arrangements of certain values and/or the scheduling the friction improvement programmes can be more accurately defined.

On the basis of the above exploratory analysis, statistical models were developed associating the vehicle's maximum attainable constant speed (safe speed) with the utilized peak friction values (0.35, 0.50, 0.65), and certain control design elements associated to design speed values and ranging from 50km/h – 90km/h; namely, the longitudinal grade (s) and the curve radius (R). The statistical analysis was performed for all three types of the examined vehicles: passenger car, unloaded truck and loaded truck.

Data were generated by examining a number of alignment arrangements and combining the above variables. In this respect, it is underlined that in order to comply with geometric design principles, care should be taken to using meaningful combinations of values within the ranges applicable in each case, as shown in Table 3

As a first step, the distributions of the values of vehicle speed were examined, in order to assume correct statistical properties. It was found that speed of passenger cars and loaded trucks conforms to a log-normal distribution (i.e. the natural logarithm of speeds conforms to a normal distribution) - the conformity of the speed distribution of loaded trucks being somewhat less satisfactory - while speed of unloaded trucks conforms to normal distribution (Figure 4).

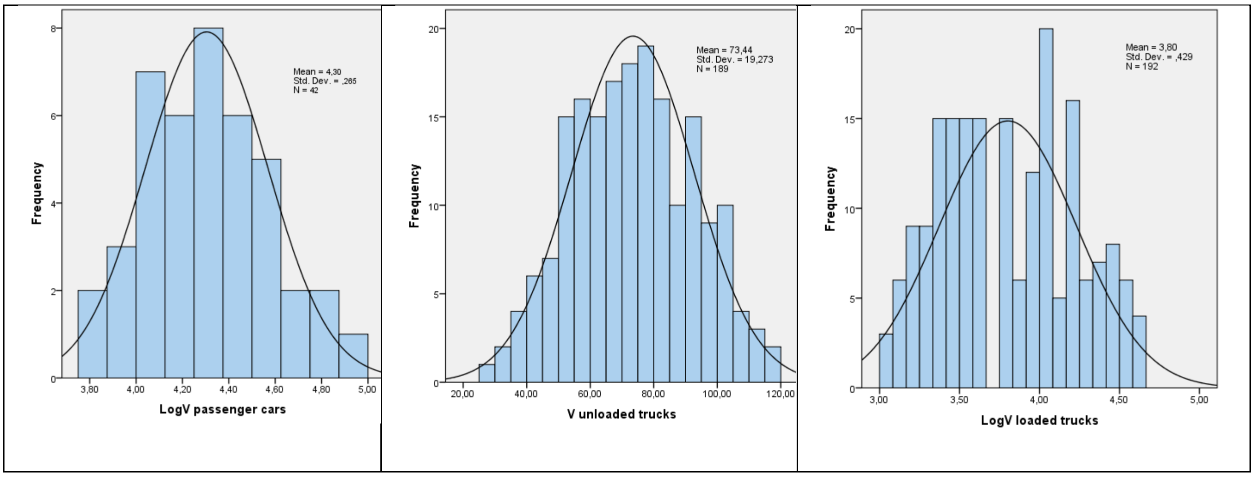


FIGURE 4. Histograms of vehicle speed frequencies vs. normal distribution curves - Passenger car (left panel), unloaded truck (middle panel), and loaded truck (right panel)

Consequently, log-normal regression analysis was implemented in the former case (in which the dependent variable speed is log-transformed), and regression analysis in the latter case. The functional form of the regression model to be developed is:

$$Y_i = \beta_0 + \beta_1 x_1 + \beta_1 x_1 + \ldots + \beta_1 x_1 + \varepsilon_i \qquad (8)$$

With Yi the dependent variable, βi parameters to be estimated and εi the random error term (assumed to be normally distributed with mean 0 and variance σ^2). When a log-transformation of the dependent variable is applied, the model becomes:

$$Y_i = \exp\left(\beta_0 + \beta_1 x_1 + \beta_1 x_1 + \ldots + \beta_1 x_1\right) + \varepsilon_i \qquad (9)$$

The modelling results are summarized in Table 4. The results of the log-normal regression model for passenger car speed suggest that, as would be expected, safe speed increases when friction increases and when curve radius increases, and it decreases when grade increases. The parameter estimates of the regression model for unloaded trucks and the log-normal regression model for loaded trucks have similar signs but different magnitude. For instance, there is a higher effect of grade on loaded truck safe speed compared to passenger car, while there is a higher effect of curve radius on passenger car safe speed compared to loaded truck. However, the effect of friction on loaded trucks' safe speed is non-significant (as expected from the vehicle dynamics analysis) at 95% confidence level.

All three models present very satisfactory fit, with adjusted R^2 higher than 0.95 and statistically significant F-test, suggesting that the largest part of the variation is explained. This was expected, since the models are in fact an interpolation among data points defined on the basis of known relationships. The model residuals were tested and were found to conform to the normal distribution hypotheses (i.e. normally distributed and homoscedastic). The interest in this approach lies on the ability to use the models formulae for directly assessing combinations of design parameters in terms of speed without performing the analytical calculations.

TABLE 4 Parameter estimates and models performance

	Passenger cars			**Unloaded trucks**			**Loaded trucks**		
	B	T-test	p-value	B	T-test	p-value	B	T-test	p-value
Constant	3.7069	32.7	<0.001	34.783	16.2	<0.001	4.3426	151.3	<0.001
f_{max}	1.1161	14.8	<0.001	57.435	16.6	<0.001	0.0337	0.7	0.465*
R	0.0026	18.1	<0.001	0.137	28.9	<0.001	0.0005	7.1	<0.001
s	-0.0300	-3.6	.001	-2.652	-23.3	<0.001	-0.1067	-71.3	<0.001
Dependent	LnV			V			LnV		
Adj. R^2	0.949			0.909			0.967		
F-test	254.4		<0.001	630.7		<0.001	1853.53		<0.001

* non significant at 95% confidence level

On the basis of the above, the following equations are proposed for the assessment of speed:

Passenger cars: $V = \exp(3.7069 + 1.1161 * f_{max} + 0.0026 * R - 0.03 * s)$

Unloaded trucks: $V = 34.783 + 57.435 * f_{max} + 0.137 * R - 2.652 * s$

Loaded trucks: $V = \exp(4.3426 + 0.00045 * R - 0.1067 * s)$

CONCLUSIONS

The present paper investigated the determination of the maximum attainable constant speed value at impending skid conditions, termed as *safe speed* during vehicle's motion in tractive mode on upgrades. Besides a passenger car (C-class mid-sized), the motion of a two-axle truck was examined as well in order to quantify the potential safety hazard for both loaded and unloaded conditions. The two-axle truck was selected on the basis of the 120kg/kW climbing performance weight to horse power ratio (200lb/hp), as adopted in the AASHTO 2011 guidelines.

A range of design speed values paired with control design elements from AASHTO 2011 Design Guidelines as well as three values of peak friction coefficients (0.35, 0.50 and 0.65) were utilized in order to assess critical safety concerns in terms of vehicle skidding.

Regarding the passenger car it was found that control alignments referring to grade values over 11% combined with poor friction pavements (f_{max}=0.35) are critical. However, for f_{MAX}=0.40, the current AASHTO 2011 control values don't raise any critical safety concerns in terms of vehicle skidding.

The unloaded truck was found to be the most critical vehicle since skidding is avoided only for pavements with peak friction set to 0.65. For pavement of peak friction 0.50, vehicle skidding occurs when the vehicle travels on control horizontal alignments that correspond to design speed values up to 80km/h, for grades over 9%. Mostly critical is the case where peak friction is set to 0.35, since safety is violated for all the examined design speed values where grade is over 4%.

The loaded truck in general was found to outperform in terms of available horse power utilization (n=100%), for every examined design speed value and even for the poor friction pavement (f_{max}=0.35). As a result, this vehicle type, especially on steep upgrades, while travelling on control horizontal alignments, although reaches far lower safe speed values compared to their corresponding design speeds, raises no critical safety concerns at least in terms of vehicle skidding. Grade values over 4% actually were found to dominate the loaded truck's performance in terms of defining V_{safe}, where for lower grade values, the impact of pavement friction was noticeable mainly on sharper control horizontal curves.

Furthermore, through a proposed statistical analysis, the authors intend to provide a tool for practitioners in order to concurrently assess the impact of the horizontal alignment, grade and friction in terms of defining the vehicle's safe speed, and consequently take certain actions. These actions include the adoption of acceptable arrangements for the above values regarding new alignments, posted speed management for existing but also scheduling friction improvement programmes more accurately for both cases.

However, since only certain types of vehicles were examined, further investigation for the entire vehicle fleet (SUVs, etc.) is required. Moreover, research with respect to the interaction between driver and vehicle especially on sharp curves is essential before final decisions in current practice can be reached.

REFERENCES

1. American Association of State Highway and Transportation Officials (AASHTO). *A Policy on Geometric Design of Highways and Streets*, Fifth Edition. Washington, DC., 2011
2. Ed.German Road and Transportation Research Association, Committee, Geometric Design Standards. *Guidelines for the Design of Freeways, (RAA)*, Germany, 2008.
3. Ministry of Environment, Regional Planning and Public Works. *Guidelines for the Design of Road Projects, Part 3, Alignment (OMOE-X)*, Greece, 2001.
4. Ministerio de Fomento. *Instrucción de Carreteras, Norma 3.1 – IC "Trazado",* Spain, 2000.
5. Harwood, D. W. and J. M. Mason. Horizontal Curve Design for Passenger Cars and Trucks. *Transportation Research Record 1445*, Transportation Research Board, Washington, DC., 1994, pp. 22-33.
6. Hassan, Y., Easa, S.M., and Abd El Halim, A. State of the Art of Three-Dimensional Highway Geometric Design. Canadian *Journal of Civil Engineering*, 1998, 25(3), pp.500–511.
7. Kontaratos, M.,Psarianos, B., and Yiotis, A. Minimum Horizontal Curve Radius as a Function of Grade Incurred by Vehicle Motion in Driving Mode. Journal of *Transportation Research. Rec*., 1994, pp. 86-93.
8. Macadam C.C., Fancher P.S. and Segal L. Side Friction for Superelevation on Horizontal Curves. *Final Technical Report, DTFH61-85-C-00019, Federal Highway Administration*, Washington DC, August 1985.
9. Chang, T.H. Effect of vehicles suspension on highway horizontal curve design. Journal of *Transportation Engineering*., 127(1), 2001, pp. 89-91.
10. Bonneson, J.A. A Kinematic Approach to Horizontal Curve Transition Design. *Transportation Research Board*, 1999, Paper No: 00-0590.
11. Psarianos, B. Kontaratos, M., and Katsios, D. Influence of Vehicle Parameters on Horizontal Curve Design of Rural Highways. *Transportation Research Circular E-C003*, 22:1-22:10. 1998.
12. Varunjikar, T. Design of horizontal Curves with Downgrades Using Low-Order Vehicle Dynamics Models. *Master of Science Thesis, The Pennsylvania State University*, Pennsylvania. 2011.
13. Eck, R.W., and French, L.J. Effective Superelevation for Large Trucks on Sharp Curves and Steep Grades. *West Virginia University, Report 153*. 2002.
14. Mavromatis, S., and Psarianos, B. Analytical Model to Determine the Influence of Horizontal Alignment of Two-Axle Heavy Vehicles on Upgrades. *Journal of Transportation Engineering,* 129(6), 2003, pp. 583-589.
15. Mavromatis S., B. Psarianos, P. Tsekos, G. Kleioutis, Investigation of Vehicle Motion on Sharp Horizontal Curves Combined with Steep Longitudinal Grades, *Transportation Letters,* DOI: 10.1080/19427867. 2015.1114748, 2016.
16. Bonneson, J.A. Superelevation Distribution Methods and Transition Designs. *NCHRP Report 439.:* Transportation Research Board, Washington, D.C., 2000.
17. Mavromatis S., B. Psarianos and C. Spentzas. Influence of the Vehicle Acceleration on the Road Minimum Horizontal Curve Radius. Paper presented and published on the *32nd International Symposium on Automotive Technology and Automation (ISATA),* pp.93-101, Vienna Austria, 1999.
18. Mavromatis S, B.Psarianos, M., D'Apuzzo and V. Nicolosi. Design Speed Ranges to Accommodate a Safe Highway Geometric Design for Heavy Vehicles. *Transportation*

Research Board. 2nd International Symposium on Highway Geometric Design, Mainz Germany 14th-17th June 2000, pp.339-351.
19. Mavromatis S., B. Psarianos and E. Kasapi. Computational Determination of Passenger Cars' Braking Distances Equipped with Anti-Block Brake Systems. *Transportation Research Board. 3rd International Symposium on Highway Geometric Design*, Chicago USA, 2005.
20. Gillespie T.D. Fundamentals of Vehicle Dynamics. *Society of Mining Metallurgy and Exploration* Inc.1992.
21. Dixon J.C., Tires, Suspension and Handling. Second Edition. *Society of Autimotive Engineers, Inc Warrendale,* Pa., United Kingdom 1996.
22. Heisler H. Advanced Vehicle Technology. *Edward Arnold. A Division of Hobber & Stoughton*, Germany 1993.
23. Krempel G. *Experimenteller Beitrag zu Untersuchungen an Kraftfahrzeugreifen.* Dissertation. Karlsruhe 1965.
24. Harwood D.W., Mason J.M., Glauz W.D., Kulakowski B.T and Fitzpatrick K. Truck Characteristics for Use in Highway Design and Operation. *Volume I, II. US. Department of Transportation. Federal Highway Administration. Publication No.FHWA-rd-89-226,* August 1990.
25. Harwood D., W. Glauz and L. Eleftheriadou. Review of Truck Characteristics as Factors in Roadway Design. *NCHRP Report 505.:* Transportation Research Board, Washington, D.C., 2003.
26. Olson, P. L., D. E. Cleveland, P. S. Fancher, L. P. Kostyniuk, and L. W. Schneider, Parameters Affecting Stopping Sight Distance, *NCHRP Report 270, National Cooperative Highway Research Program,* Transportation Research Board, 1984.
27. Saito, K., J. J. Henry, and R. R. Blackburn, Development and Application of Predictor Models for Seasonal Variations in Skid Resistance. *Proceedings of Australian Road Research Board*, Vol. 13, 1986.
28. Wambold, J. C., Obtaining Skid Number at Any Speed from a Test at a Single Speed, *Presented at the 67th Annual Meeting of the Transportation Research Board*, 1988.
29. Gauss F. Skid Resistance Properties of Tires and their Influence on Vehicle Control In Skidding Accidents: *Transportation Research Board 621*, TRB, National Research Council, Washington D.C., 1976, pp.8-18.