Safety Assessment of Control Design Parameters through Vehicle Dynamics Model

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Abstract

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. Aiming to assess critical safety concerns in terms of vehicle skidding, the motion of a passenger car was examined over a range of design speed values paired with control design elements from AASHTO 2011 Design Guidelines as well as certain values of poor pavement friction coefficients.

Two distinct cases were investigated; the determination of the maximum attainable constant speed (termed as safe speed) at impending skid conditions as well as the case of comfortable curve negotiation where lower constant speed values were utilized. The overall objective was to define the safety margins for each examined case.

From the interaction between road geometry, pavement friction and vehicle characteristics, many interesting findings are reported, where some of them are beyond the confined field of road geometry parameters; such as demanded longitudinal and lateral friction values and horse-power utilization rates. From the road geometry point of view it was found that control alignments on steep upgrades consisting of low design speed values and combined with poor friction pavements are critical in terms of safety.

Such cases should be treated very cautiously through 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.

Keywords

Vehicle dynamics model, safe speed, friction, road geometry, passenger car

1. Introduction

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. In existing practice, during vehicle's cornering process, a simplified approach has prevailed (point mass model), from which the minimum horizontal radius is derived.

This 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:

- steady state cornering is assumed and the acceleration effect 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
- 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 the longitudinal design is assumed flat

Despite these simplifications, Harwood and Mason [5] concluded that regarding passenger cars, existing design policy provides adequate margins of safety against both skidding and rollover. However, many researchers have pointed out the necessity for more sophisticated models to simulate vehicle's cornering process [e.g. 6-9] especially in cases where steep grades are present [e.g. 10-14].

Regarding tractive mode, research studies [8, 11 and 14] revealed that steep upgrades reduce the margin of safety. More specifically, one of these [14] performed on a roadway with combined sharp horizontal curves and steep longitudinal grades, showed that steep upgrade road sections are more critical in terms of horizontal radii requirements.

In current road design practice, steady state cornering is assumed during vehicle curve negotiation. However the ability of a vehicle to negotiate a curved section with a certain speed and tire-road friction hasn't been an issue of deeper concern. The control values of critical road geometric parameters associated with the design speed are based mostly on experience but also limitations from the operational point of view (e.g. control grade values) and are not an outcome of a safety assessment.

The paper aims to assess the ability of a typical passenger car to maintain common design speed values for the corresponding control design parameters, including critical upgrade values and various tire – road friction values, and consequently lift the restrictions of the point mass model. Therefore the assessment of the steady state cornering process is addressed by investigating the vehicle's maximum attainable constant speed (termed as safe speed) at impending skid conditions as well as by examining the case where lower constant speed values can be utilized for various control road geometry and tire friction parameters. The overall objective is to define the safety margins for each examined case.

2. Methodology

A previous vehicle dynamics model developed by the authors [15, 16] was utilized where all forces and moments applied to the vehicle were 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:

$$\sum X = 0$$

$$m\frac{dV}{dt} = \sum U_i - \sum S_i\theta_i + \frac{mV^2}{R}\beta - mgs - A_d \qquad (1)$$

$$\sum Y = 0$$

$$m\frac{dV}{dt}\beta = \sum U_i\,\theta_i + \sum S_i - \frac{mV^2}{R} + mge \qquad (2)$$

$$\sum Z = 0$$

$$\sum P_i = mg + \frac{mV^2}{R}e - A_n \qquad (3)$$

where (f=front, r=rear) :

dv/dt: vehicle's acceleration (positive value) (m/sec2)

U_f , U_r: driving forces acting to front and rear axle respectively (N)

S_f , S_r : lateral forces acting to front and rear axle respectively (N)

P_f , P_r : vertical forces acting to front and rear axle respectively (N)

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 [17]. 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 [17 - 19].

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 [13] and [15 - 16].

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 (4) as follows:

$$F_x = 745.6\frac{P}{V}n \qquad (4)$$

where :
Fx : tractive force (N)
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 1 and Equation 4, 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:

$$A\left(\frac{dV}{dt}\right)^4 + B\left(\frac{dV}{dt}\right)^3 + C\left(\frac{dV}{dt}\right)^2 + D\left(\frac{dV}{dt}\right) + E = 0 \qquad (5)$$

On the other hand, according to [20], pavement friction reserves are distributed to the longitudinal and lateral direction of travel. 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:

$$\left(\frac{f_T}{f_{T,max}}\right)^2 + \left(\frac{f_R}{f_{R,max}}\right)^2 \leq 1 \qquad (6)$$

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

Assuming an initial speed value referring to certain vehicle, road and friction parameters, from Equation 4, the software calculates the horse-power utilization factor (n) for impending skid conditions (Equation 6), where at the same time the critical wheel is pointed out. By setting an increment rate for speed (0.25km/h in the present analysis) and adapting each time the horse-power utilization factor, always at impending skid conditions, as the acceleration decreases, there is a certain value of speed which eliminates the vehicle's acceleration impact as given through Equation 5 (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 [13] and the output data from well-known multi-body vehicle simulation software (CARSIM) [16], the contribution of which is mostly valued in the automotive industry for higher-reliability vehicle stability prediction. Both cases revealed a satisfying match. It must be stressed that although the present vehicle dynamics model utilizes a number of vehicle technical parameters, it is focused mainly in associating vehicle speed to critical arrangements of road geometry parameters. Therefore the proposed model is solely oriented in assessing vehicle safety from the highway engineering point of view.

The potential safety violation assessment for AASHTO 2011 design guidelines, in terms of horizontal design values was performed for a C-class mid-sized, front wheel drive (FWD) passenger car, where at least from the vehicles' dimensions points of view, a real case is represented (KIA Proceed). Although an effort was made to provide the utilized vehicles' parameters from the vehicle industry, most of them were taken from the literature [19]. The vehicle parameters inserted in the model are shown in Table 1.

Design speed values starting from 50km/h were examined. The higher grade utilized 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 (most unfavourable) paired with the control horizontal radii for each design speed value. Finally, the superelevation rate was set to 6% for all 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 [21]. These tests determine a value equivalent to the coefficient of sliding. In general, the peak coefficients of friction exceed the sliding friction by 10%-45% varying with tire and pavement types [20]. However in highway design the available side friction, utilized for the minimum horizontal radius determination, which is based in driver's comfort [1], is considered as 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 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 wheels

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

In order to assess vehicle safety during curve negotiation, for every wheel the demanded friction was calculated in both longitudinal and lateral directions of travel and by utilizing Equation 7, both the vehicle's maximum attainable constant speed as well as the critical wheel at impending skid conditions were defined.

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

3. Analysis and Results

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. This process was performed for all 3 of the examined peak friction coefficients (0.35, 0.50 and 0.65).

The V_{safe} calculation was performed on the basis of defining the most critical vehicle wheel which in every case was the front inner to the curve wheel; front due to the FWD configuration of the vehicle and inner to the curve since the inner wheels towards the curve experience more increased friction values due to the lateral load transfer.

Figure 1a shows the grade impact during the V_{safe} determination (maximum attainable constant speed at impending skid conditions) for control design values referring to design speed of 70km/h and assuming poor friction pavement (f_{max}=0.35). As seen through the transparent columns in the primary vertical axis (left), the design speed of 70km/h can be maintained even for the steepest grade value of 12% (V_{safe}=70.2km/h). The secondary vertical axis (right) shows the demanded longitudinal and side friction demand for the vehicle's most critical wheel (front inner to the curve wheel). It can be seen that the lateral friction decreases when the grade rises, while at the same time increasing values of friction in the longitudinal direction are demanded on steep grades during the vehicle's effort to generate more intense tractive force. For every pair of the longitudinal and lateral friction, the vector sum is equal to the peak friction value utilized (0.35). In the same figure, the accuracy of the point mass model in delivering the lateral friction was examined as well through the well-known R_{min} determination formula as follows:

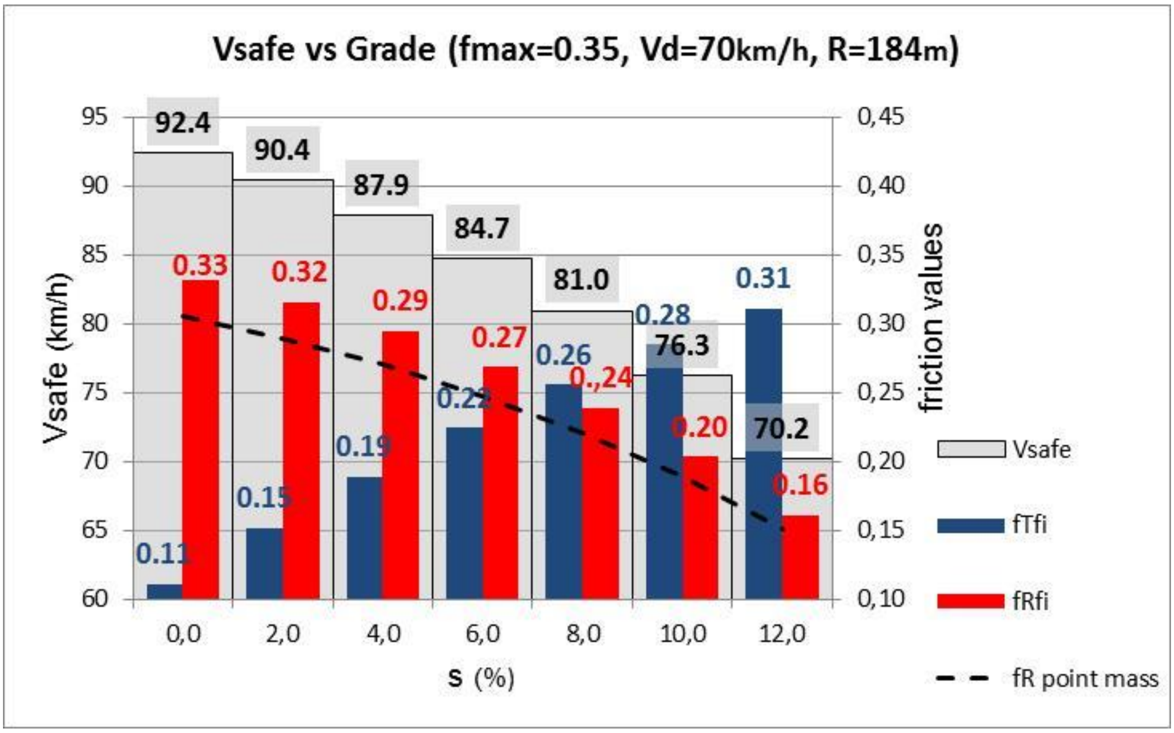
$$f_R = \frac{V^2}{127\mathrm{R}} - e \qquad (7)$$

where :

f_R : lateral friction based on the point mass model
V (km/h) : vehicle's speed
R (m) : radius of curve
e (%/100) : cross slope value

The results of Equation 7, shown through a black dashed line, compared to the demanded lateral friction values (f_{Rfi}), are in line with previous research findings [14, 22] according to which the point mass model somehow underestimates the actual demanded side friction values.

The safety concerns for the poor friction pavement (f_{max}=0.35), stated above are addressed through Figure 1b, where the case of V_{design}=70km/h is further analyzed. 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, as expected from above, greater grade values come along with increased horse-power utilization rates, but only up to a certain rate (e.g. up to 51% can be utilized for grade values of s=12%). If the driver attempts to utilize more horse power percentage compared to the values shown in Figure 2b, the vehicle will skid. As a result, a vehicle's safety performance may result in critical situations when entering a curve.



Note: f_{Tfi}, f_{Rfi}: longitudinal and lateral friction factors of the front inner to the curve wheel

Figure 1a: Grade impact during V_{safe} determination.

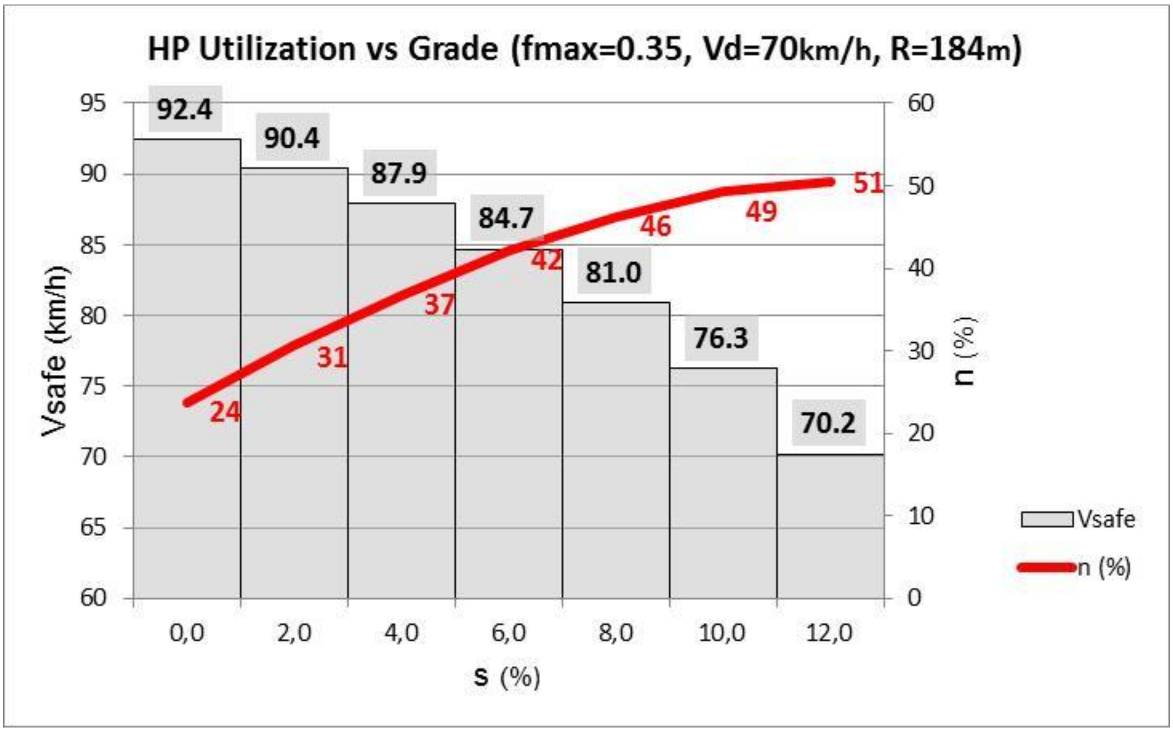


Figure 1b: Horse power utilization rates during V_{safe} determination.

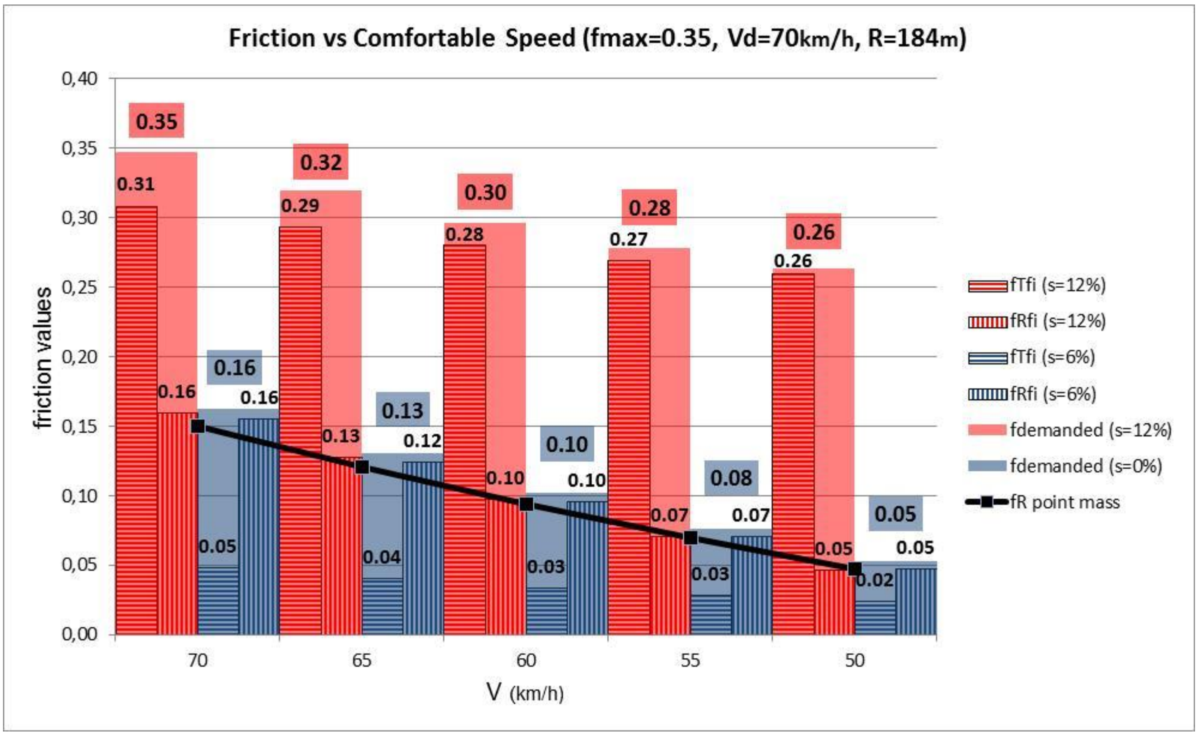
The above figures correspond to impending skid conditions. It is also interesting to define the safety margins in terms of the utilized longitudinal and lateral friction factors for cases where the vehicle negotiates a curve with less speed compared to the relevant V_{safe} value.

This assessment is performed through Figure 2 for poor friction pavement and control design values corresponding once again to V_{design}=70km/h, where a number of speed values related to more or less "comfortable driving" are utilized. Moreover, in Figure 2 two extreme cases in terms of road grade are shown; the most unfavorable grade value of 12% (red columns) and the mild grade value of 0% (blue columns). The horizontally and vertically patterned columns refer to the longitudinal and lateral friction respectively of the front inner to the curve wheel (critical wheel). Finally the transparent columns illustrate the demanded friction in vehicle's both directions of travel (vector sum of longitudinal and lateral friction values).

The general conclusion during comfortable curve negotiation is that solely the demanded longitudinal friction is grade dependent. On the other hand, the demanded lateral friction components seem to maintain values equivalent to the values extracted from the point mass model. This finding was confirmed for all the examined design speed values, as well as the utilized pavement friction values.

From the above analysis it is evident that safety margins expressed through friction and/or horse power utilization rates, although interesting in understanding the interaction between vehicle and road geometric parameters are not at all practical in terms of guiding practitioners. Therefore, since speed values are always appreciated, as already mentioned, for each pair of V_{design}, R_{min} and examined peak friction coefficients, an overall assessment of the vehicle's capability to maintain the design speed for a range of grade values was carried out. It is obvious that such an investigation refers to vehicle motion under impending skid conditions. The upper limits of the grade values utilized 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 ($V_{safe} \approx V_{design}+10$). For the examined vehicle, design speed values corresponding to control alignments of up to 90km/h were examined, since more excessive speed values didn't raise safety concerns.

Figure 3a 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 3a 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_{design}), marked with light grey. 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 in bold. It can be seen that the vehicle while negotiating control horizontal curvatures of up to 60km/h design speed, cannot maintain the design speed value for grades greater than 11%. In other words, for the examined poor friction pavement (f_{max}=0.35), steady state cornering at 50km/h and 60km/h for the corresponding control radii (R=79m and R=123m respectively) is possible for grade values up to 11%, although the maximum grade values referring to local rural roads is 14% and 13% respectively. Therefore such cases should be treated very cautiously either through posted speed management or friction improvement programmes.



Note: f_{Tfi}, f_{Rfi}: longitudinal and lateral friction factors of the front inner to the curve wheel

Figure 2: Friction safety margins for speed values less than V_{safe} (V_{design}=70km/h).

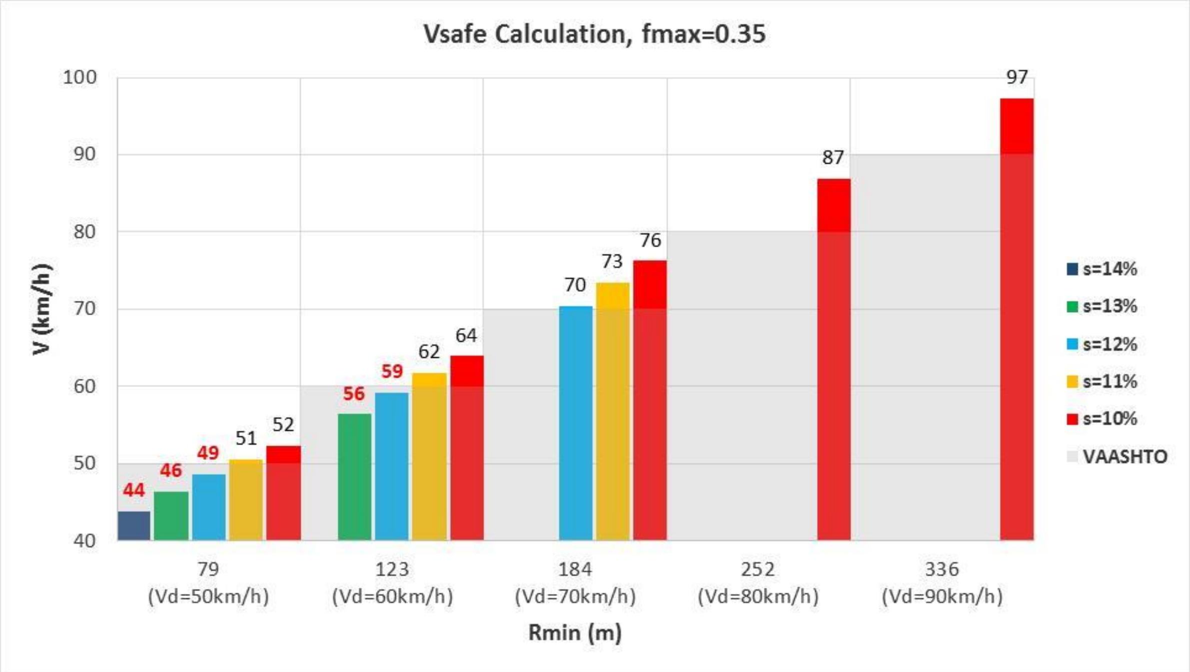


Figure 3a: V_{safe} variation for control design values and poor friction pavement (f_{max}=0.35).

Finally, since the case of V_{design}=50km/h was found to be the most critical case, a further assessment regarding all the utilized pavement friction values was carried out in Figure 3b. It was confirmed that safety concerns exist only for control alignments with grade values over 11% combined with poor friction pavements (f_{max}=0.35). 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.

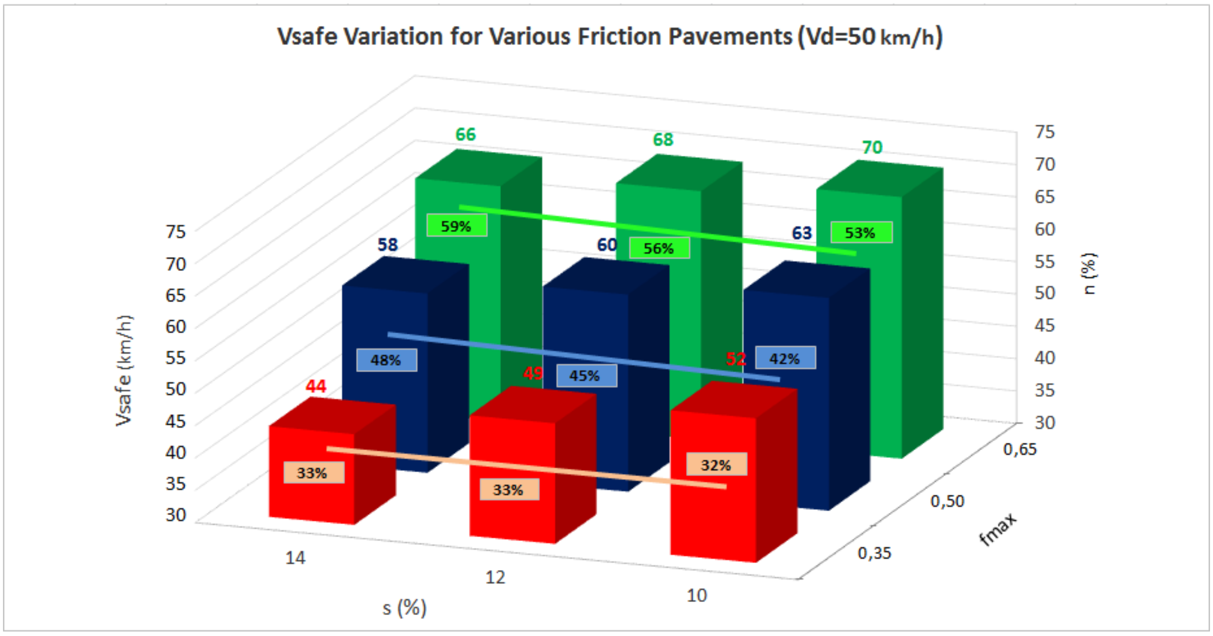


Figure 3b: V_{safe} variation for control design values referring to V_{design}=50km/h.

Furthermore in Figure 3b, besides the V_{safe} values, the horse-power utilization factor for each case can also be seen (values marked with "%"). It is evident that in cases of friction pavement less than f_{max}=0.40 and upgrade values over 11%, the driver during his effort to retain the speed of 50km/h may deal with unexpected critical situations, since not only the speed value of 50km/h cannot be reached, but the vehicle will skid when the horse power utilization rate exceeds 30%.

Generalizing this finding, the authors believe that vehicles equipped with excessive amounts of horse power rates must be driven very conservatively in roads with poor friction pavement.

4. Conclusions

The present paper, for various tire – road friction values, investigated the ability of a typical (C-class mid-sized) passenger car on steep upgrades to maintain common design speed values based on the corresponding control design parameters. A range of design speed values paired with control design elements based on 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 pavements with poor friction performance under both wet (0.35) and dry (0.65) pavement conditions and investigate potential critical safety concerns in terms of vehicle skidding

Two distinct cases were investigated; the determination of the maximum attainable constant speed (termed as safe speed) at impending skid conditions as well as the case of comfortable curve negotiation where lower constant speed values were utilized. The overall objective was to define the safety margins for each examined case.

The comfortable curve negotiation case was assessed through the demanded friction coefficients in both lateral and longitudinal directions of travel as well as by comparing their overall value (vector sum) to the available friction. The research, for the utilized peak friction coefficients and control design parameters imposed by the respective design speed values, revealed that contrary to the friction demand in the lateral direction, the longitudinal friction is grade dependent where increasing values come along with steeper grades. On the other hand, the demanded lateral friction components seem to maintain values equivalent to the values extracted from the point mass model.

Regarding the vehicle motion at impending skid conditions, it was found that lateral friction demand is mostly critical on mild grades. At the same time, increasing values of friction in the longitudinal direction are demanded as grade becomes steeper, due to the vehicle's effort to generate more intense tractive force. The findings related to the point mass model revealed that the delivered side friction values are somehow below the actual demanded.

Moreover, the research aiming to provide practical guidance to practitioners revealed that under certain circumstances, steady state cornering with the design speed on respective control alignments is not always feasible on poor friction pavements (f_{max}=0.35). More specifically vehicle motion with design speed values equivalent to 50km/h and 60km/h on the respective control alignments is feasible only for grades up to 11%. In general for f_{MAX}=0.40, the current AASHTO 2011 control values don't raise any critical safety concerns in terms of vehicle skidding. Another interesting finding is that since rather low horse power rates were found to be utilized, vehicles equipped with excessive amounts of horse power rates must be driven very conservatively in roads with poor friction pavement.

The authors believe that such cases should be treated very cautiously through 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 a certain passenger car type was examined which definitely is not a representative of the passenger car fleet, further investigation towards the entire vehicle fleet (SUVs, heavy vehicles etc.) is required before reaching to final decisions and implementing such actions in current practice. Moreover, research with respect to the interaction between driver and vehicle especially on sharp curves is essential as well.

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