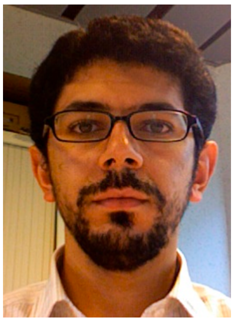
RELIABILITY ANALYSIS OF ROLLOVER AND JACKKNIFING RISK :



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Abstract

This paper presents a stochastic based approach to prevent rollover and jackknifing risk of articulated heavy vehicle. First, a specific 6-DOF articulated vehicle model is developed and risk criteria are used. The inputs and parameters of the vehicle are modeled by random variables or stochastic processes in order to take into account uncertainties. A sensitivity analysis by Sobol indices is presented to exhibit influent parameters towards risk and to reduce the number of random variables involved in the stochastic model. Then structural reliability methods are employed to assess the probability of risk using well known FORM/SORM methods and are compared to Monte-Carlo simulation.

Keywords:

Articulated heavy vehicle, rollover, jackknifing, reliability analysis, sensitivity analysis, probabilistic approach, stochastic process, FORM/SORM

1. Introduction

At present time, the truck transport is one of the most important activities of the country's economy. According to the French road safety statistics (*ONISR*) for year 2008 [1], accidents involving heavy vehicles have serious consequences for road users and incidents induce major congestions or damage to the environment or the infrastructure at disproportionate economic costs. The risk of having dead people in accidents involving trucks is multiplied by 2.4 in comparison to the same risk computed for accidents involving only light vehicles, mainly because of the important gross mass difference between light vehicles and trucks. Many real time active safety systems have been developed to control vehicle stability in dangerous conditions: rollover, jackknifing, roadway departure...

Nevertheless all these systems don't detect and warn drivers early enough for preventing risky situations. Consequently, the aim is to develop a driving support system, embedded in the heavy vehicle and communicate with on road equipment's, to warn about risk's level early enough.

Little work is available in the literature concerning combined probabilistic analysis and vehicle dynamics studies. It's mostly deals with suspensions analysis under uncertain characteristics and loading [2]. One Icelandic study [3] covers the wind-related accident issue with reliability analysis, but no warning system has been envisaged. More recently, classical reliability methods are deployed in [4], [5] to prevent rollover risk of single heavy vehicle and roadway departure risk of lightly vehicle.

2. Principle of the proposed approach

This paper proposes a probabilistic method for identifying dangerous situations of risk of articulated heavy vehicle. To this end, we develop a stochastic heavy vehicle model derived from a 6−DOF deterministic heavy vehicle model. Next, two safety criteria related to rollover and jackknifing risk are introduced. In practice, parameters involved in the model are not known with absolute precision. Thus, it's necessary to take into account the random nature of these parameters by modeling them as random variables or stochastic processes. A part of the work presented here aims to exhibit the relevant entry variables that must be known to be representative of the system uncertainties. So, we perform and compare two different sensitivity analysis methods in order to identify influent parameters and to reduce the number of stochastic parameters involved in the stochastic model. Finally, probability of failure is assessed through standard structural reliability methods by using the well-known FORM/SORM methods.

3. Deterministic articulated heavy vehicle model

This section presents the deterministic model of an articulated heavy vehicle and its validation. Then safety criteria, respect to rollover and jackknifing risks, are chosen to define the riskiness of a situation. Finally, a validation is conducted upon the safety criterion values by comparison with a complete heavy vehicle simulator Prosper [8].

3.1 Heavy vehicle dynamics model

The framework of this study is focused on an articulated heavy vehicle composed of a rigid tractor with 2 axles and a rigid semitrailer with 3 axles. These two components are linked at the fifth wheel as shown in figure 1. The tractor is the superposition of an unsprung mass (axles and tires) and a sprung mass (cabinet and chassis). The trailer is composed of an unsprung mass (axles and tires) and a sprung mass (chassis and trailer). Unsprung and sprung masses are connected together with suspensions. In this study, the following two basic assumptions are used:

1. The cornering maneuver realized at constant vehicle speed,
2. No breaking and acceleration actions by the driver.

Under these hypotheses, the considered dynamic component of the motion are both the yaw tractor, the translation of the vehicle in the horizontal plane, the tractor roll angle and the articulation angle between the tractor and the trailer. To construct the model of the heavy vehicle we complete the previous assumptions with the following ones:

3. The pitch and bounce dynamics motions are neglected,
4. Tractor and trailer frames are modeled as rigid bodies,
5. The total axles with axle suspension are reduced to roll suspension only.

To obtain the dynamics equations of simplified heavy vehicle, the classical Lagrangian formulation is used.

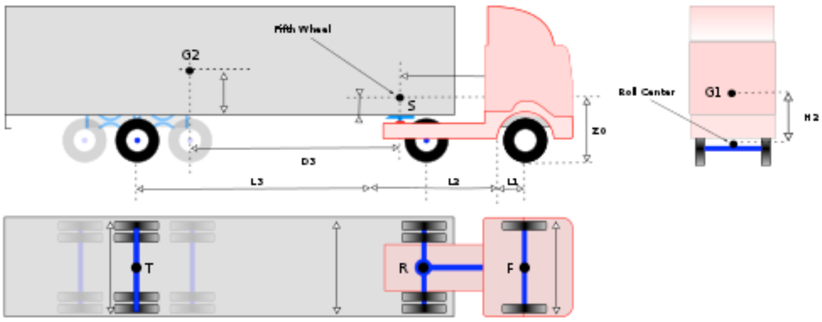


Figure 1 – Articulated heavy vehicle

The proposed model is based on 6 degrees of freedom (*DOF*) *yaw-roll* model derived from a 5−DOF [6], [7]. To get the dynamics equation of the model we consider the motion the two sprung masses in the coordinate system (see figure 2). (X_E,Y_E,Z_E) is the earth-fixed coordinate system, (X_t,Y_t,Z_t) and (X_{st},Y_{st},Z_{st}) are respectively the tractor and semi-trailers's sprung masses coordinate systems fixed at the center of gravity (C.G) of each body. (X_u,Y_u,Z_u) is the tractor's unpring mass coordinate defined at the center plane of the front axle with Z_u parallel to Z_E. The relative motion of (X_u,Y_u,Z_u) with respect to the earth fixed coordinate system (X_E,Y_E,Z_E) describes the translation motion of the tractor in the horizontal plane and its yaw motion along the Z_E-axis. The roll motion is described by motion of the coordinate system (X_t,Y_t,Z_t) associated with the coordinate system (X_u,Y_u,Z_u). The articulation angle between the tractor and trailer can be described by relative motion of the coordinate (X_t,Y_t,Z_t) with respect to the coordinate system (X_{st},Y_{st},Z_{st}).

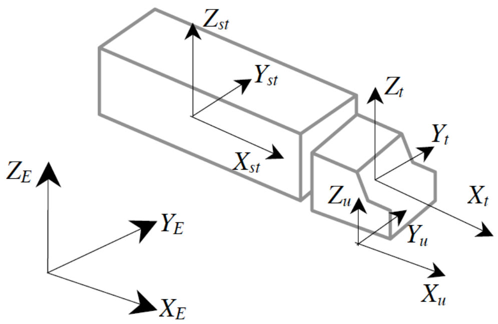


Figure 2 – Systems of coordinate

From these coordinate systems, we introduce the generalized coordinate system (x_E, y_E, z_E, ψ ,ϕ , ψ_f), defined as follows:

- x_E is the position of the tractor C.G. in the direction of the $X_E$axis,
- y_E is the position of the tractor C.G. in the direction of the Y_E axis,
- z_E is the position of the tractor C.G. in the direction of the Z_E axis,
- ψ is the yaw angle of the tractor,
- ϕ is the roll angle if the tractor,
- ψ_f is the relative yaw angle, *i.e.*, angle between tractor and trailer at fifth wheel.

The vector equation describing the dynamics of the heavy vehicle is obtained using the Lagragian formulation. It's of the form:

$$M(q(t),p)\ddot{q}(t)+C(q(t),\dot{q}(t),p)\dot{q}(t)+G(q(t),p)=F_g(p,\delta(t)), \quad t\geq 0 \qquad (1)$$

where $q=(x,y,z,\psi,\varphi,\psi_f)^t$ is the generalized coordinates vector in which we set to simplify : $x = x_E$, $y = y_E$ and $z = z_E$; F_g is the vector of generalized forces, M is the inertial matrix that is symmetric positive definite, $C(q,\dot{q})\dot{q}$ is the combined Coriolis and centrifugal forces and G is the gravity vector. The generalized forces F_g represents the effect of external forces acting on the vehicle body. These later result from the tire-road interface and suspensions defined in terms of longitudinal and lateral tire forces and vertical forces. Generalized forces depend on steering angle δ . We use a specific steering angle profile (hook maneuver), which allows observing rollover and jackknifing risks. All heavy vehicle's parameters, defined in Table 1, are gathered in the vector p of dimension n_p =25.

Table 1 – Main heavy vehicle characteristics

Parameter	Symbol	Unit	Value
Tractor's sprung mass	m_1^s	kg	4533
Tractor's unsprung mass	m_1^u	kg	1156
Semitrailer's sprung mass	m_2^s	kg	5833
Semitrailer's unsprung mass	m_2^u	kg	1469
Tractor's moment of inertia relative to axis x	I_{x1}	Kg.m^2	3115
Tractor's moment of inertia relative to axis y	I_{y1}	Kg.m^2	20000
Tractor's moment of inertia relative to axis z	I_{z1}	Kg.m^2	45000
Semitrailer's moment of inertia relative to axis x	I_{x2}	Kg.m^2	60000
Semitractor's moment of inertia relative to axis y	I_{y2}	Kg.m^2	71318
Semitractor's moment of inertia relative to axis z	I_{z2}	Kg.m^2	100000
Distance between tractor C.G. and front wheel axle	l_1	m	1.46
Distance between tractor C.G. and rear wheel axle	l_2	m	2.66
Distance between joint (fifth wheel) and trailer rear wheel axle	l_3	m	7.5
Relative x position between tractor's C.G. to fifth wheel	d_1	m	2.24
Relative y position between tractor's C.G. to fifth wheel	d_2	m	0.16
Relative x position between semitrailer's C.G. to fifth wheel	d_3	m	5.85
Relative y position between semitrailer's C.G. to fifth wheel	d_4	m	0.155
Tractor front axle track width	T_{w1}	m	2.04
Tractor rear axle track width	T_{w2}	m	1.78
Trailer rear axle track width	T_{w3}	m	2.15
Distance from tractor roll center to C.G.	h_r	m	0.3
Tractor's C.G. height	h_1	m	0.348
Semitrailer's C.G. height	h_2	m	0.256
Wheel radius	r	m	0.6
Adhence coefficient	μ	–	0.95
Initial velocity	v_0	m.s^{-1}	15
Gravity accelaration	g	m.s^{-2}	0.81

Then, we rewrite the vehicle model as a first order differential equations (ODE) system as:

$$\dot{u}(t) = f(t, u(t), \delta(t)), \quad t \geq 0 \qquad (2)$$

where $u = (q^t, \dot{q}^t)^t$ and f is a function from $\mathbb{R}_+ \times \mathbb{R}^{12} \times \mathbb{R}^{np} \times \mathbb{R}$ into $\mathbb{R}^{12}$. Using classical Runge-Kutta method of order 4 solves this equation. Such a model easily brings forth useful insights of dynamic phenomena (yaw-roll) with fast computation time, compared to mulibody approach. The system is simulated and validated in the MatLAB environment with a specific ODE Fortran solver. Parameters are obtained from Prosper truck simulator [8]. In practice, heavy vehicle simulation requires only 50ms compared to a quasi real-time simulator (Simulink, Prosper, TruckSim...). These improvements allow to embed our heavy vehicle simulator into sensitivity and reliability algorithms.

3.2 Safety criteria

The framework of this work is focused on two risk situations:

1. The rollover, which is a lateral instability due to a lateral load transfer.
2. The jackknifing, which is a loss of control that causes the rotation of the tractor with respect to the semitrailer.

For each risk we propose and choose a safety criterion to assess and detect dangerous situations. Rollover is a well-known phenomena and its analysis is spread enough in the literature [9]. Unfortunately, jackknifing risk is a more complicated phenomenon and there are few safety criteria.

Rollover criterion

Rollover is one of the most frequent accidents (20%) and causes significant damages to the vehicles and injuries to its driver and passengers. Several anti-rollover systems and rollover warnings systems were developed to assist and warn the driver [10], [11]. Several rollover indicators can be found in the literature : SRT (Static Rollover Threshold), RPER (Rollover Prevention Energy Reserve), CSV (Critical Sliding Velocity)...

In our study, rollover risk evaluation is based on the maximum of a rollover risk indicator, namely the load transfer ratio (LTR), which corresponds to the load transfer between the left and the right sides of the vehicle. The resulting expression of this indicator is defined as:

$$LTR = \frac{F_{z,l} - F_{z,r}}{F_{z,l} + F_{z,r}} \tag{3}$$

where $F_{z,l}$ and $F_{z,r}$ are respectively the left and the right normal forces. In practice, when LTR is equal to 0, the heavy vehicle has stable roll dynamics. The risk becomes high as this indicator goes towards ±1. Ackermann [11] simplified the LTR criterion, when overlooking unsprung mass, as follows:

$$LTR = \frac{2m_2^s}{m_2 T_{w,3} g}\left[\left(h_r + h_2\cos\phi\right)\left(\ddot{y} + v_0 r - h_2\ddot{\phi}\right) + g h_2 \sin\phi\right] \tag{4}$$

where $T_{w,3}$, h_r, v_0,h_2,m_{s2}, r, g are defined in Table 1 and $m_2 = m_2^s + m_2^u$. Kamnik [12] proposed an improved nonlinear rollover criterion for articulated heavy vehicle called LLT. This later is accurate and coincides to the Ackermann criterion when neglecting unsprung mass contribution.

Jackknifing criterion

Jackknifing is characterized by a loss of stability in the yaw motion of the articulated system. It occurs for several reasons:

1. When the rear wheels of the tractor are blocked,
2. When the vehicle applies in turn the brakes abruptly,
3. When the road is slipping (low adherence).

This phenomenon is more frequent when then trailer is empty or when the load is badly distributed in the trailer. Theoretically, jackknifing is detected the relative angle y_f is greater than $\pi/2$. Jackknifing is characterized by the friction indicator [13]:

$$\mu_{\min} = \frac{F_y}{F_z \cos\psi_f} \tag{5}$$

where F_y and F_z are respectively the lateral and normal road forces applied to the heavy vehicle. When $\mu_{\min}$ the heavy vehicle remains stable.

In our work, we use the criterion proposed in [14] and based on the estimation of the articulation yaw angle ψ_f. The jackknifing criterion is defined:

$$C_m = \dot{r}_1.(r_2 - r_1) = \dot{r}_1.r \quad (6)$$

where r_1 is the vector position of the tractor fifth wheel and r_2 is the vector position of rear trailer axle. The heavy vehicle is on jackknifing situation when $C_m \leq 0$.

4. Sensitivity analysis

Among all the parameters of the mechanical model, some present a marked random variability. Therefore it is crucial for the credibility of the application to take into account this reality via a suited stochastic modeling. The object of this section is to identify the minimal family of variables that must be considered as random using a sensitivity analysis. We investigate a global sensitivity method is performed by computing the Sobol indices.

4.1 Global sensitivity

The parameters are gathered in the vector $p \in \mathbb{R}^{n_p}$ modeled as a continuous $\mathbb{R}^{n_p}$-valued random variable denoted $\mathbf{P} = (P_1, ..., P_{np})$ for which the following hypotheses are made :

H1. its components are mutually independent,
H2. two distributions are alternatively considered for each of its components : a truncated Gaussian distribution and an uniform distribution,
H3. all the components of **P** follow simultaneously the same type of distribution,
H4. all the components of **P** have the same coefficient of variance.

Each random variable P_i verifies:

$$\begin{aligned} E\left[P_i\right] &= p_i \\ Var\left[P_i\right] &= k.p_i \end{aligned} \quad (10)$$

where Var[P_i] and k are respectively the standard deviation and the coefficient of variance of P_i. In the following, this later is taken equal to 1% (0.01). In order to take into account perturbations on steering angle δ, this one is modeled as bounded stochastic process of the form [5]:

$$\Delta(t) = \delta(t) + \Lambda(t) \quad (11)$$

with :

$$\Delta(t) = e.\sin\left(v.t + s.W(t) + 2\pi U\right), \quad t \geq 0 \quad (12)$$

where e, v, s are given real positive constants, W is a standard real Wiener process and U is a random variable uniformly distributed on [0,1] and independent of W. With these stochastic modeling, the response system **u** is now a vector random process **U** which depends on $(\mathbf{P}, \Delta)$ and the control variable r becomes a random process R depending on **P** and Δ:

$$R(t) = S(P, \Delta(t)) = S(P_1, ..., P_{n_p}, \Delta(t)) \quad (13)$$

The Sobol index s_i^R [18], [19] associated with the random variable P_i is a deterministic function of t defined from the conditional expectation E[R|P_i], according to the formula:

$$s_i^R = \frac{Var\left[E\left[R \mid P_i\right]\right]}{Var\left[R\right]} = \frac{V_i}{V} \tag{14}$$

where $Var(.)$ denotes the variance. From this definition, s_i^R ranges from 0 to 1. A small value means that uncertainty on *Pi* has few influence on the variability of *R* and consequently, in this case, *Pi* can be considered as a deterministic parameter. On the contrary if s_i^R is closed to 1, P_i must keep its status of random variable. Second order indices reflect the interaction between two parameters. They are defined

By :

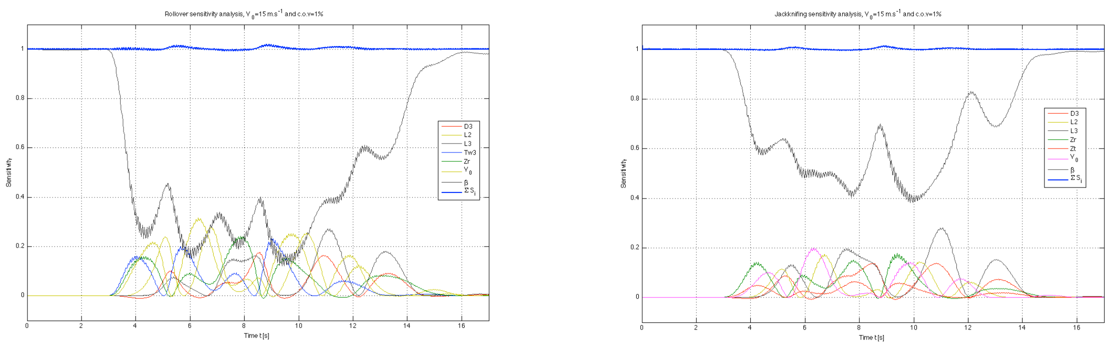
$$s_{i,j}^R = \frac{Var\left(E\left[R \mid P_i P_j\right]\right) - V_i - V_j}{Var\left(R\right)} = \frac{V_{i,j} - V_i - V_j}{V}, \quad i \neq j \tag{15}$$

Higher orders, until the n_p-th order, are defined in the same way. All these sensitivity indices satisfy the fundamental property:

$$\sum_i s_i^R + \sum_{i<j} s_{i,j}^R + \ldots + \sum_{i_1 < \ldots < i_{n_p}} s_{i_1,\ldots,i_{n_p}}^R = 1 \tag{16}$$

Hence, the analysis starts with the first order indices calculation. Then, if $\sum_i s_i^R \approx 1$ then all the higher orders are negligible. Otherwise, interactions exist in the safety criterion model, and further calculations are needed. The sensitivity of R with respect to Δ is estimated from the Iooss's work [20].

Sobol's indices are computed with a huge size sample 10^6 to obtain good first order indices. In fact sums of Sobol indices, plotted with blue line on figure (3-a) and (3-b), are closed to 1. Figures (3) show the obtained results for the rollover and jackknifing criteria, using the Sobol sensitivity analysis with perturbations on the steering angle. Results exhibit that only 6 parameters d_3, l_2, l_3, T_{w3}, h_r, v_0 are really influent on rollover and only 5 parameters d_3, l_2, l_3, h_r, h_2, v_0 on jackknifing.



(a) Rollover case

(b) Jackknifing case

Figure 3 – Evolution of first order Sobol indices

4.2 Conclusions

With both methods of sensitivity analysis, we get quite the same influent parameters. Global sensitivity gives relevant results and allows taking into account the perturbation on steering angle. However here, for simplicity and to apply standard reliability analysis, the steering angle perturbation is supposed is not taken into considerations. As a result, the only random parameters considered are at most the six random variables $P_1 = v_0$, $P_2 = h_r$, $P_3 = l_2$, $P_4 = l_3$, $P_5 = h_2$, $P_6 = T_{w,3}$.

5. Reliability analysis

5.1 General principle and statement of the problem

In the following the vector random parameter $\mathbf{P}=(P_1, ...,P_6)^T$ is assumed to be defined on the probability space (Ω,F,P), where Ω is a sample space, F is σ-algebra on Ω and P is a probability measure on F. According to (H1)-(H4) hypotheses (cf. section 4.2), the distribution is known and admits a probability density denoted in that follows f_p. Let Z be the safety margin associated with the control variable R, such that:

$$Z = r_0 - \max_{t \in [0,T]} |R(t)| \tag{17}$$

Where r_0 is a given limit value and T is the observation interval. This real random variable is a function of P:

$$Z = G(P) \tag{18}$$

Where G is the limit state function associated with the safety criterion chosen for the study. It's a measurable mapping from R^6 into R which defines two complementary subset of R^6, D_s and D_f, such that:

$$\begin{aligned} D_s &= \left\{ p \in \mathbb{R}^6 : G(p) > 0 \right\} \\ D_f &= \left\{ p \in \mathbb{R}^6 : G(p) \le 0 \right\} \end{aligned} \tag{19}$$

Called respectively the safety domain and the failure domain. Two events are associated with these domains, the safety event E_s and the failure event E_f, such that:

$$\begin{aligned} E_s &= \left\{ \omega \in \Omega : G(P(\omega)) > 0 \right\} \\ E_f &= \left\{ \omega \in \Omega : G(P(\omega)) \le 0 \right\} \end{aligned} \tag{20}$$

and which verify $E_s \cup E_f = \Omega \quad E_s \cap E_f = \varnothing$. Once known the pdf f_P and defined the events E_s and E_f, an import objective of the reliability analysis is then to evaluate the probabilities $P(E_s)$ and $P(E_f)$ (respectively called safety probability and failure probability) given by :

$$\begin{aligned} P_s &= P(E_s) = \int_{D_s} f_P(p)\,dp \\ P_f &= P(E_f) = \int_{D_d} f_P(p)\,dp \end{aligned} \tag{21}$$

And such that P_s=1-P_f. In practice, an exact calculation of P_f is not possible and a Monte-Carlo procedure must be used. In classical reliability approach, it is customary to transform the initial formulation into a standard formulation in which the vector of the random parameters follows a standard Gaussian distribution. This leads to construct a regular transformation T, with inverse T^{-1}, such the vector random variable P can be written $P=T^{-1}(T)$, where Y is a R^6-valued standard Gaussian random variable. Carrying out the change the change of variable y=T(p), probability of failure becomes :

$$P_f = \int_{\Delta_f} \varphi_6(y)dy \tag{22}$$

where φ_6 is the standard Gaussian pdf on R^6 and $\Delta_f = T(D_f)$.

FORM (First Order Reliability Method) approximation consists in replacing the failure domain Δ_f by a half-space Δ_f^L tangent to Δ_f at the design point M^* (see figure 4). Using this approximation, the failure probability P_f can be approximated by:

$$P_f \approx P_f^L = \int_{\Delta_f^L} \varphi_6(y)dy = \Phi(-\beta_{HL}) \tag{23}$$

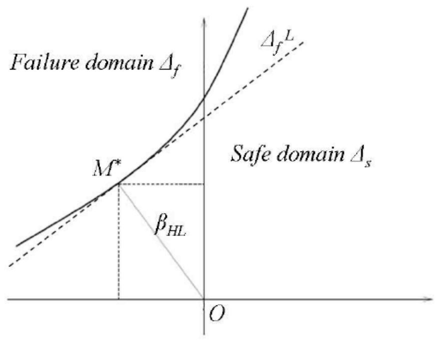


Figure 4 – Systems of coordinate

where Φ is the one-dimensional standard Gaussian distribution function and β_{HL} is the Hasofer-Lind index [15] defined by :

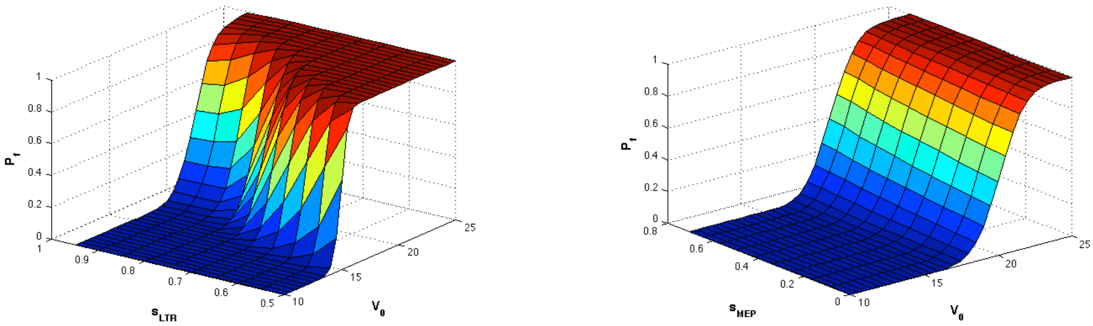
$$\beta_{HL} = \min_{M \in \Delta_f} \|OM\| = \|OM^*\| \tag{24}$$

In practice, β_{HL} is computed by the Rackwitz-Fiessler (iHLRF) algorithm [15]. It is possible to approximate the limit state function by a quadratic surface at the design point, which leads to the SORM (Second Order Reliability Method). In this case, the probability of failure can be approximated by the Hohenbichler formula:

$$P_f \approx P_f^Q = \int_{\Delta_f^Q} \varphi_6(y)dy = \Phi(-\beta_{HL}) \prod_i \sqrt{1 + \frac{\varphi(\beta_{HL})}{\Phi(-\beta_{HL})} \kappa_i} \tag{25}$$

Where κ_i denote the main curvatures of the limit-state function at M^*.

Monte-Carlo method is employed to estimate the "exact" probability of failure. A set of 10^6 simulated realizations of P is used for an initial velocity ranges from 20m.s^{-1} to 24m.s^{-1}. Each calculation requires about one hour on a standard PC. We use the MatLAB toolbox FERUM [21] to compute Pf^L and Pf^Q. Figures (5) compares the probabilities of failure given by Monte-Carlo procedure and the FORM/SORM methods.



(a) Rollover case (b) Jackknifing case

Figure 5 – Evolution of probability of w.r.t initial velocity and criterion threshold

6. Conclusion

An application of structural reliability methods to road safety domain is proposed, with a view to develop a heavy vehicle rollover and jackknifing warning systems. Such methods consider the uncertainties in the model input to deduce a failure probability (i.e. here the probability to violate a safety criterion). Hence it seems appropriate to handle with the random variability existing in the triptych driver-vehicle-infrastructure. Compared to other warning system studies that rely on deterministic safety criteria, the major advantage of the stochastic approach presented here consists in its direct computation of a risk index, since P_f represents the probability to violate the safety criterion. Thus the obtained result is the evolution of risk against speed, which would have been hard to define with a deterministic approach. This evolution of risk is valuable decision support tool for the alarm triggering.

Acknowledgments

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