Stability Analysis and Rollover Scenario Prediction
For Tractor Semi-Trailer.
Mohamed Bouteldja 1,2, Nacer K. M’Sirdi1
1Laboratoire de Robotique de Versailles, CNRS/FRE 2659
10 avenue de l’Europe, 78140 VÉLIZY, FRANCE
2Laboratoire Central des Ponts et Chaussées
58 Boulevard Lefèvre, 75014 PARIS, FRANCE
mohamed.bouteldja@lcpc.fr

Résumé—An 5 DOF nonlinear dynamic simulation model for tractor semi-trailers is proposed. The simulation model is derived by applying Lagrange’s equation in conjunction with appropriate coordinate transformation techniques so that the simulation model can be easily handled and extended by using Matlab/Simulink and Maple. The objective of this work consists in the design of an estimator for relevant parameters for the prediction of movement of inversion of a heavy vehicles. The proposed model is used to validate the prediction of rollover threshold accelerations derived based on quasi-static analysis. The predicted threshold accelerations are compared with the threshold accelerations obtained by quasi-static analysis. The study consist in the following stages: vehicle modelling, presentation of two parameters and stats estimators (one off line and the other on line), synthesis of a sliding modes observers (necessary for the on line method), presentation of the simulation results.

Mots-clés—Dynamic Modelling of Heavy Vehicle, Robust observers, Estimation of gravity center, Rollover Prediction and Detection.

I. Introduction
The improvement of safety of the roads : estimation and control for heavy vehicles

II. Heavy Vehicle Model
A. Coordinate System
The type of heavy vehicle considered in this work is a tractor-semi-trailer with 5-axels (2-axels for tractor and 3-axels for semi-trailer). In order to presented the essentials dynamics in cornering manoeuvr of vehicle. We adopt a configuration of vehicle representing a tractor with two axles with one chassis body and semi-trailer with one wheel axle and one chassis [REF]. In this configuration, we consider only the dynamics of the two bodies (i.e. tractor and semi-trailer’s chassis). The roll angle is considered around the tractor roll axis. The pitch and bounce will be neglected in the model.

Four coordinate systems are used to describe the motions of the vehicle sprung and unsprung masses as shown in figure (??).

To develop the dynamics equations of the model we consider the motion of the two sprung masses in the coordinate system (?). \( X_E Y_E Z_E \) is the earth-fixed coordinate system. \( X_t Y_t Z_t \) and \( X_s Y_s Z_s \) are respectively the tractor and semi-trailer’s sprung masses coordinate systems fixed at the center of gravity of each body. \( X_s Y_s Z_s \) is the tractor’s unsprung mass coordinate defined at center plane of the front axle with \( Z_s \) is parallel to \( Z_E \). The relative motion of \( X_s Y_s Z_u \) 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 \( Z_E \) axis. The roll motion is described by motion of coordinate \( X_s Y_s Z_t \) relative to the coordinate \( X_s Y_s Z_u \). The articulation angle between the tractor and trailer can be described by relative motion of the coordinate \( X_s Y_s Z_t \) with respect to the coordinate \( X_s Y_s Z_u \).

With this description of the coordinate systems and their relative motion, we consider the following generalized coordinates:

\( x_E \) : position of the tractor C.G. in \( X \) direction of the earth-fixed coordinate system,
\( y_E \) : position of the tractor C.G. in \( Y \) direction of the earth-fixed coordinate system,
\( \psi \) : yaw angle of the tractor,
\( \phi \) : roll angle,
\( \psi_f \) : angle between tractor and trailer at fifth wheel.

The previous description of the vehicle motion allows the calculation of the translational and rotational velocities of each body-mass at C.G. by study of the kinematics with respect to different references frames. Then the total kinetic energy \( (E_K) \) and potential energy \( (E_P) \) are calculated in the inertial reference frame \( X_E Y_E Z_E \). The expressions of kinetic energy and potential energy are used to derive the vehicle model based on the Lagrangian mechanics approach.

B. Suspension model and Tyre model
By far the majority of heavy vehicle suspensions use the leaf spring as the vertically compliant element. For the sake
of simplicity, we assume in this paper that the suspension is modeled as the combination of a nonlinear spring and a damper element. As shown in (??), the vertical force acting on the vehicle sprung mass through the suspension system is equal to the static equilibrium force plus the variation force, which is denoted as \( F_{zi} \), from the spring equilibrium point.

The suspension variation forces \( F_{si} \), are obtained of by the function of suspension deflection and deflection rate as [Tomi] :

\[
F_{si} = \begin{cases} 
 k_f e_1 + k_f^2 e_1^3 + D_f \dot{e}_i & \text{for } i = 1, 2 \\
 k_r e_3 + k_r^2 e_3^3 + D_r \dot{e}_i & \text{for } i = 3, 4 \\
 k_t e_5 + k_t^2 e_5^3 + D_t \dot{e}_i & \text{for } i = 5, 6 
\end{cases}
\]

(1)

where \( k_f \) and \( k_r \) are parameters of the tractor front spring, \( k_r \) and \( k_r \) are parameters of the tractor rear spring, \( k_t \) and \( k_t \) are parameters of the trailer spring, \( D_f \), \( D_r \), and \( D_t \) are parameters for dampers, and e is the deflection of the \( i^{th} \) spring from its equilibrium position and is given as

\[
\begin{align*}
e_1 &= -\frac{T_1}{2} \phi \\
e_2 &= \frac{T_1}{2} \phi \\
e_3 &= -\frac{T_2}{2} \phi \\
e_4 &= \frac{T_2}{2} \phi \\
e_5 &= -(\frac{T_2}{2} \phi) \cos \psi_f + (l_3 \phi) \sin \psi_f \\
e_6 &= (\frac{T_2}{2} \phi) \cos \psi_f + (l_3 \phi) \sin \psi_f
\end{align*}
\]

(2)

III. SABILITY ANALYSIS OF HEAVY VEHICLE

A. Parameters Influencing

The influence of the various parameters will be then studied.

A.1 Steering angle

steering angle was

A.2 Adherence coefficient

Adhesion was one of the most important parameters in the rollover accident.

Figure (5) represents the stable and unstable domain. When adhesion decreases, the heavy vehicle is stable. The rollover principal characteristic i.e. the strong mobilization of the pneumatic contact is represented : when adhesion is lower than 0.45, which corresponds to a strongly wet roadway, the heavy vehicle does not turn over. However, other problems intervent like trajectory loss while slipping.

A.3 Mass and height of the gravity center

A.4 Other factors influencing roll stability

Suspension lash, present in the leaf spring suspension systems commonly used on heavy vehicles, degrades the roll-over threshold by reducing the effective roll stiffness of the suspensions.

Torsional compliance of the vehicle frames also reduces the roll-over threshold. For example, a flexible trailer frame rolls to a greater angle under the influence of lateral acceleration, thus increasing the magnitude of the destabilizing lateral displacement moment. Furthermore torsional compliance of the tractor frame reduces the ability of the tractor steer axle to provide a stabilizing moment to resist the roll motion of the payload.
Torsional compliance of the vehicle couplings reduces roll stability in a similar way. Note that the roll stiffness of a conventional fifth wheel coupling decreases with articulation angle, and the roll moment that can be transmitted through a fifth wheel coupling saturates at an included angle of around 2%.

IV. Prediction of Rollover

A. Prediction of Rollover Threshold Acceleration

The rollover evolution can be presented in case of heavy vehicle by two steps:

1. (a) i. evolution of roll angle of the sprung mass before the lift-off of the wheels,
   ii. décollage des roues due au variation des forces appliquée sur la masse nonsuspendue.

Figure (7,8) montrent les différentes forces et moments appliquée sur la masse suspendue et non-suspendue de tracteur et vis versa pour le semi-remorque.

\[
\begin{align*}
\sum M_{O,A} &= 0 \quad (3) \\
-M_1 - M_2 + (F_{1z} + F_{2z}) \frac{S}{2} + P_1 (y_0 + h_2 \phi) + \frac{P_1}{g} a_y b_2 &= 0 \quad (4)
\end{align*}
\]

where the \(a_y\) is the lateral acceleration. Then, using \(M_i = k_{\phi,i} \phi\), \(k_{\phi,i}\) is the roll stiffness of suspension and the roll angle of the sprung mass, \(\phi\) can be expressed in terms of the lateral acceleration as:

\[
\phi = \left( \frac{P_1 y_0}{2k_0 - P_1 h_2} \right) + \left( \frac{P_1 h_2}{g(2k_0 - P_1 h_2)} \right) a_y \quad (5)
\]

we observe that equation (5) is in the form \(\phi = Aa_y + B\).

Figure (8) shows the free body diagram for unsprung mass in a left turn. The roll moment equilibrium equation as:

\[
M_1 + M_2 + (F_{1y} + F_{2y}) h_1 - \Delta F_z T = 0 \quad (6)
\]

Tableau A.2 : Valeurs des simulations.

<table>
<thead>
<tr>
<th>Paramètres</th>
<th>Valeurs</th>
<th>Unités</th>
</tr>
</thead>
<tbody>
<tr>
<td>(C_f)</td>
<td>582 kN/rad</td>
<td></td>
</tr>
<tr>
<td>(C_r)</td>
<td>783 kN/rad</td>
<td></td>
</tr>
<tr>
<td>(C)</td>
<td>457 kN/rad</td>
<td></td>
</tr>
<tr>
<td>(d)</td>
<td>100 kN/rad</td>
<td></td>
</tr>
<tr>
<td>(g)</td>
<td>9.81 m/s²</td>
<td></td>
</tr>
<tr>
<td>(h_R)</td>
<td>0.68 M</td>
<td></td>
</tr>
<tr>
<td>(h)</td>
<td>1.15 M</td>
<td></td>
</tr>
<tr>
<td>(I_x)</td>
<td>24201 N m²</td>
<td></td>
</tr>
<tr>
<td>(I_{zz})</td>
<td>0 N m²</td>
<td></td>
</tr>
<tr>
<td>(I_z)</td>
<td>34917 N m²</td>
<td></td>
</tr>
<tr>
<td>(L_f)</td>
<td>1.95 M</td>
<td></td>
</tr>
<tr>
<td>(L_r)</td>
<td>1.54 M</td>
<td></td>
</tr>
<tr>
<td>(m)</td>
<td>14300 Kg</td>
<td></td>
</tr>
<tr>
<td>(m_2)</td>
<td>12487 Kg</td>
<td></td>
</tr>
<tr>
<td>(T)</td>
<td>1.860 M</td>
<td></td>
</tr>
</tbody>
</table>

V. Conclusion

Le comité d’organisation souhaite que vous ne rencontriez pas de problèmes lors de l’utilisation de la classe \texttt{cifa.cls}, et vous remercie d’avoir lu cet exemple. En cas de problème, envoyez un courrier à \texttt{webmaster_cifa2004@ec-lille.fr}, et nous tenterons de trouver une solution pour l’envoi des articles définitifs.

\[
\begin{align*}
\delta_f & \quad \text{Angle de braquage des roues avant} \\
\beta & \quad \text{Angle de dérive total du véhicule} \\
\phi & \quad \text{Angle de roulis} \\
\psi & \quad \text{Angle de lacet} \\
r & \quad \text{Vitesse du lacet} \\
v & \quad \text{Vitesse du véhicule} \\
C_f & \quad \text{coefficients de raideur des pneus avant} \\
C_r & \quad \text{coefficients de raideur des pneus arrière} \\
C & \quad \text{raideur de la barre anti-roulis totale} \\
d & \quad \text{atténuation de roulis de la suspension passive} \\
g & \quad \text{accélération de la gravité} \\
h_R & \quad \text{hauteur du centre de roulis} \\
h & \quad \text{hauteur de centre de gravité CG2, au dessus de h}_R. \\
I_{zz} & \quad \text{Inertie du moment de roulis, de la masse suspendue} \\
I_{zz} & \quad \text{moment d’inertie croisé} \\
I_z & \quad \text{inertie globale du moment de lacet} \\
L_f & \quad \text{distance du centre de gravité CG1 à l’essieu avant} \\
L_r & \quad \text{distance du centre de gravité à l’essieu arrière} \\
\text{m} & \quad \text{la masse totale du véhicule} \\
\text{m}_2 & \quad \text{La masse suspendue du véhicule} \\
\text{ coefficient d’adhérence de la route} \\
T & \quad \text{longueur des essieux}
\end{align*}
\]

Tableau I.1 : Nomenclature de véhicule
Références