CONTROL THEORY APPROACH FOR ON-BOARD ESTIMATION AND MONITORING OF HEAVY DUTY VEHICLE DYNAMICAL TYRE FORCES




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Abstract
Heavy good vehicle (HGV) is the dominant freight transport mode, carrying more than 75% of the payload in most countries of the European Union. Moreover, HGVs proportion is increasing in the traffic flow and often reaches 15 to 20%. Therefore, to meet the new objectives of sustainable development and infrastructures, a particular attention shall be paid to assess and limit the wheel and axle dynamic loads on pavements, for a better durability. To maintain HGV's safety and efficiency, a continuous on-board wheel load monitoring could be a useful part of any anti-rollover systems or other stability program. With the advance in the control theory, it seems to be interesting to apply the existent techniques to the on-board WIM techniques. The aim is to develop smart systems to estimate the impacts of heavy vehicles on pavements and to develop active control strategies to reduce the maximum dynamic effects.
Keywords: heavy good vehicles, wheel and axle loads, impact forces, on-board weigh-in-motion, sliding mode observers, control theory.

Résumé
Le mode routier est dominant pour le transport de marchandises, avec plus de 75% du fret dans la plupart des pays de l'Union européenne. En outre, la proportion de poids lourds croît dans le trafic routier et atteint souvent 15 à 20%. Par conséquent, pour atteindre les nouveaux objectifs de développement durable et pour les infrastructures, une attention particulière doit être portée pour évaluer et limiter les charges dynamiques des roues et essieux sur les chaussées, pour garantir leur durabilité. La sécurité et l'efficacité des poids lourds peuvent être améliorées par une surveillance continue embarquée des charges de roues, servant à des systèmes anti-renversement ou de stabilisation. Les progrès de la théorie du contrôle fournissent des techniques intéressantes pour le pesage en marche embarqué. L'idée est de développer des systèmes intelligents d'estimation des forces d'impacts des poids lourds sur les chaussées et un contrôle actif pour réduire les effets dynamiques maximaux.
Mots clefs: poids lourds, charges de roue et d'essieu, forces d'impact, pesage en marche embarqué, observateurs à mode glissant, théorie du contrôle.

1. Introduction

There are different ways to measure impact forces. One way consists of using an instrumented wheel hub. This provides accurate measurements but is a high-cost solution. Another common and rather easy mean, used by many researchers, uses strain gauges on the axles (Davis, 2008). This works well but only on straight path, and requires a complex strain gauge installation and calibration. One can also use optic sensors, such as (Blanksby, 2008), which measures the distance between the wheel axis and the road surface, thereby giving a measurement of the tire deflection and therefore tire forces. The principal drawbacks are the errors due to concentricity of the sensor and road surface irregularities. Recently, (Tuotonen, 2009) proposed a solution to measure tire frame deformation using built-in sensors, but the solution is complex to implement. The literature proposes estimation methods (Siegrist, 2003), (Bouteldja, 2005), (Imine, 2008), based on assumptions on the forces behavior or requiring the knowledge of vehicle and tire characteristics usually difficult to access. We propose here a practical solution easy to implement on modern trucks and to be coupled with active control systems. This method uses sensors available in the vehicle and connected to the CAN bus, such as ABS and ESP sensors. Furthermore, a low cost gyro, pressure sensors and accelerometers, easy to install and to calibrate before installation, constitute an additional instrumentation. A model of vehicle dynamics was developed to account for the tractor-trailer motion in the yaw plane, and the vertical axle hop and rolling motion. The lateral tire forces are taken as inputs and the vehicle accelerations as outputs for the model. While the accelerations are either measured or estimated by a numerical differentiation with a second order sliding mode algorithm (a super twisting algorithm – robust differentiator), the model will be used in a reverse way to evaluate the unknown lateral forces. These estimated forces are then used in addition with the suspension forces (measured by a pressure sensor on the air springs), to evaluate the vertical wheel forces from the axle model.

This method (Khemoudj, 2009), (Khemoudj, 2010), is easy to implement using an optimized sensor configuration, but for real-time operating conditions, the matrix inversion can saturate the CPU. The forces can then be estimated by classical robust state observers. Using the estimated forces, a methodology is proposed to reduce the effects of dynamic forces on pavement. A control strategy based on hybrid approach is introduced to act on the vehicle by the steering angle. The method is adjustable to different driving scenarios (straight line, curve…) and various road profiles may be accounted for. This paper is divided into four sections: (i) the vehicle model, (ii) the validation by simulator, (iii) the estimation results, (iv) the hybrid control strategy with some results, and finally (v) a conclusion.

2. Modeling and Estimating Tire Forces

To develop observers and to validate our approach, we use an observable non linear yaw plane model (Figure 1). The behavior of an articulated vehicle is related to the forces applied by pavement on the tires. In the yaw plane model, two axles of the tractor are considered and the axle-group of the semi-trailer is represented by a single equivalent axle. The articulated vehicle dynamics is represented by the tractor and the semi-trailer apart. The internal hitch force appears as an external force for both the tractor and the semi-trailer. Moreover, the hitch force is represented by two components: a longitudinal component denoted X_h and a lateral component denoted Y_h.

F_{fx} and F_{fy} are respectively the tractor front longitudinal and lateral tire forces, F_{rx} and F_{ry} the tractor rear longitudinal and lateral tire forces and F_{tx} and F_{ty} the total forces applied on the semi-trailer axle. $\ddot{x}$ and $\ddot{y}$ are respectively the longitudinal and lateral tractor acceleration. l_f and l_r the distances between the centre of gravity of the tractor and respectively the front and the rear axles. Moreover l_{ft} and l_{rt} correspond to the distances between the centre of gravity of the semi-trailer and respectively the hitch and rear axle. δ is the steering angle and θ is the relative yaw angle between the tractor and the semi-trailer. The equations of motion of the tractor in the yaw plane are given in the longitudinal axis:

$$F_{fy}\sin(\delta)+F_{fx}\cos(\delta)+F_{rx}-X_h=M_1\ddot{x} \quad (1)$$

and for the lateral direction:

$$F_{fy}\cos(\delta)-F_{fx}\sin(\delta)+F_{ry}-Y_h=M_1\ddot{y} \quad (2)$$

The yaw moment of the tractor at the hitch point is:

$$(l_f+l_h)(F_{fy}\cos(\delta)-F_{fx}\sin(\delta))-(l_r-l_h)F_{ry}=I_{z1}\ddot{\alpha}_1 \quad (3)$$

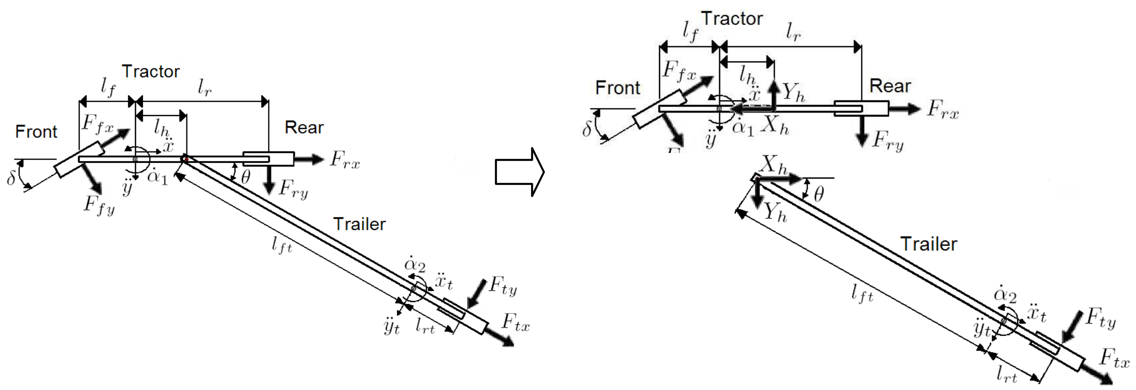

Figure 1 - Extended semi-trailer bicycle model

Remark 1. I_{z1} is the moment of inertia of the tractor around a vertical axis crossing the hitch. It is obtained by the Huygens theorem: $I_{z1}=I_{z1cog}+M_1 l_h^2$, where I_{z1cog} is the inertia moment according to the vertical axis crossing the centre of gravity of the tractor. M_1 is the total mass of the tractor. l_h is the distance between the centre of gravity of the tractor and the hitch. The semi-trailer dynamics is governed by the following equations, in the longitudinal direction:

$$X_h+F_{tx}\cos(\theta)-F_{ty}\sin(\theta)=M_2(\ddot{x}_t\cos(\theta)-\ddot{y}_t\sin(\theta)) \quad (4)$$

and in the lateral direction:

$$Y_h+F_{tx}\sin(\theta)+F_{ty}\cos(\theta)=M_2(\ddot{x}_t\sin(\theta)+\ddot{y}_t\cos(\theta)) \quad (5)$$

and the yaw dynamics:

$$(Y_h\cos(\theta)-X_h\sin(\theta))l_{ft}-F_{ty}l_{rt}=I_{z2}\ddot{\alpha}_2 \quad (6)$$

where M_2 and I_{z2} are respectively the mass and the moment of inertia of the semi-trailer according to the vertical axis. $\ddot{x}_t$ and $\ddot{y}_t$ are respectively the longitudinal and lateral accelerations of the semi-trailer. The state space vector consists of measured velocities. It is composed of the tractor yaw rate $\dot{\alpha}_1$, the front and rear tractor wheel rotational velocities denoted respectively ω_{fw} and ω_{rw}, the semi-trailer yaw rate $\dot{\alpha}_2$ and the semi-trailer wheel rotational velocity noted ω_{tw}. It is given by:

$$X_v = \begin{bmatrix} \dot{\alpha}_1 & \omega_{fw} & \omega_{rw} & \dot{\alpha}_2 & \omega_{tw} \end{bmatrix}^T \tag{7}$$

The output vector Y consists of the tractor longitudinal and lateral accelerations and the semi-trailer longitudinal and lateral accelerations considered in the tractor frame and denoted respectively $\ddot{x}_t^*$ and $\ddot{y}_t^*$.

$$Y = \begin{bmatrix} \ddot{x} = \frac{1}{M_1}(F_{rx} + F_{fx}\cos(\delta) + F_{fy}\sin(\delta) - X_h) \\ \ddot{y} = \frac{1}{M_1}(F_{ry} - F_{fx}\sin(\delta) + F_{fy}\cos(\delta) - Y_h) \\ \ddot{x}_t^* = \frac{1}{M_2}(F_{tx}\cos(\theta) - F_{ty}\sin(\theta) + X_h) \\ \ddot{y}_t^* = \frac{1}{M_2}(F_{tx}\sin(\theta) - F_{ty}\cos(\theta) + Y_h) \end{bmatrix} \tag{8}$$

with:

$$\begin{cases} \ddot{x}_t^* = \ddot{x}_t \cos(\theta) - \ddot{y}_t \sin(\theta) \\ \ddot{y}_T^* = \ddot{x}_t \sin(\theta) - \ddot{y}_t \cos(\theta) \end{cases} \tag{9}$$

and:

$$\begin{cases} I_{fw}\dot{\omega}_{fw} = T_{fw} + r_w F_{fx} \\ I_{rw}\dot{\omega}_{rw} = T_{rw} + r_w F_{rx} \\ I_{tw}\dot{\omega}_{tw} = T_{tw} + r_w F_{tx} \end{cases} \tag{10}$$

The state-space model can be written in this form:

$$\begin{cases} \dot{X}_v = Bu + W_1(u)u_F \\ Y = W_2(u)u_F \end{cases} \tag{11}$$

where u is the known input vector composed of the tractor front and rear wheel torques respectively T_{fw} and T_{rw}, the semi-trailer wheel torque T_{tw}, the steering angle δ and the relative yaw angle θ.

$$u = \begin{bmatrix} T_{fw} & T_{rw} & T_{tw} & \delta & \theta \end{bmatrix}^T \tag{12}$$

Remark 2. In practice, steering angle, yaw rate, wheel rotational velocity, the lateral acceleration and the engine torques are signals available on the CAN-bus of the vehicle. The braking torque can be deduced from the brakes pressure which is also available at the CAN-bus. In our application, we do not consider braking situations. The relative yaw angle θ between the tractor and the trailer can be obtained by integrating the difference between the measured tractor yaw rate $\dot{\alpha}_1$ and the measured trailer yaw rate $\dot{\alpha}_2$.The unknown input forces u_F

$$u_F = \begin{bmatrix} F_{fx} & F_{rx} & F_{tx} & F_{fy} & F_{ry} & F_{ty} & X_h & Y_h \end{bmatrix}^T \tag{13}$$

The matrix $B \in R^{5\times5}$ is given by:

$$B = \begin{bmatrix} 0 & 0 & 0 & 0 & 0 \\ \frac{1}{I_{fw}} & 0 & 0 & 0 & 0 \\ 0 & \frac{1}{I_{rw}} & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & \frac{1}{I_{tw}} & 0 & 0 \end{bmatrix} \qquad (14)$$

By considering $l_{fh} = l_f + l_h$ and $l_{rh} = l_r - l_h$. The matrix $W_1(u) \in R^{5\times 8}$ is given by:

$$W_1(u) = \begin{bmatrix} \frac{l_{fh}}{I_{z1}}\sin(\delta) & 0 & 0 & \frac{l_{fh}}{I_{z1}}\cos(\delta) & -\frac{l_{rh}}{I_{z1}} & 0 & 0 & 0 \\ \frac{r_w}{I_{fw}} & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & \frac{r_w}{I_{rw}} & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & -\frac{l_{rt}}{I_{z2}} & -\frac{l_{ft}}{I_{z2}}\sin(\theta) & \frac{l_{ft}}{I_{z2}}\cos(\theta) \\ 0 & 0 & \frac{r_w}{I_{tw}} & 0 & 0 & 0 & 0 & 0 \end{bmatrix} \qquad (15)$$

and $W_2(u) \in R^{4\times 8}$ is given by:

$$W_2(u) = \begin{bmatrix} \frac{1}{M_1}\cos(\delta) & \frac{1}{M_1} & 0 & \frac{1}{M_1}\sin(\delta) & 0 & 0 & -\frac{1}{M_1} & 0 \\ -\frac{1}{M_1}\sin(\delta) & 0 & 0 & \frac{1}{M_1}\cos(\delta) & \frac{1}{M_1} & 0 & 0 & -\frac{1}{M_1} \\ 0 & 0 & \frac{1}{M_2}\cos(\theta) & 0 & 0 & -\frac{1}{M_2}\sin(\theta) & \frac{1}{M_2} & 0 \\ 0 & 0 & \frac{1}{M_2}\sin(\theta) & 0 & 0 & \frac{1}{M_2}\cos(\theta) & 0 & \frac{1}{M_2} \end{bmatrix} \qquad (16)$$

with I_{fw}, I_{rw} and I_{tw} are respectively the rotational inertias of the front and rear tractor wheels and semi-trailer wheels. r_w is the wheels radius supposed constant for all wheels. To evaluate the vertical forces, an axle model is introduced. The forces applied to the axle are shown in Figure 2.

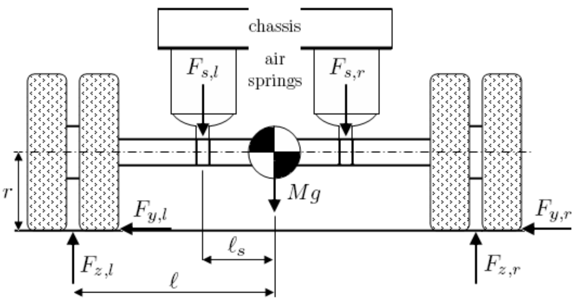

Figure 2 - Axle model

We can set the equations of motion of the free body in the vertical plane as:

$$F_{z,l} + F_{z,r} = F_{s,l} + F_{s,r} + M_{ax} g + M_{ax} a_{ax} \tag{17}$$

and for the equation of moments:

$$(F_{z,l} - F_{z,r})\ell = (F_{s,l} - F_{s,r})\ell_s + F_y r_w + I_{ax} a_\Phi \tag{18}$$

with gravity acceleration g = 9.81 m/s². $F_{z,l}$ and $F_{z,r}$ are respectively the vertical wheel forces at the left and at the right of the axle, $F_{s,l}$ and $F_{s,r}$ represent respectively the suspension forces at the left and the right of the axle, F_y is the resultant lateral force i.e., the sum of the left $F_{y,l}$ and right $F_{y,r}$ lateral tire force applied to the axle, M_{ax} the mass of the axle, I_{ax} the moment of inertia of the axle around its roll axis, a_{ax} the axle-hop acceleration. a_Φ the axle-roll acceleration. ℓ is the distance between the application point of the tire force and the centre of gravity of the axle. ℓ_s is the distance between the application point of the suspension force and the centre of gravity of the axle and finally r is the distance between the ground and the centre of gravity of the axle. In order to simplify the problem, some assumptions have been taken:

1. The roll centre and the centre of gravity of the axle are the same.
2. The axle is rigid and perfectly symmetrical.
3. The distance r is constant and is equal to wheel radius.

Remark 3. The forces between chassis and axles can be measured by the use of a pressure transducer because of the proportionality between the force and pressure in air-springs.

3. Model validation

The validation of both yaw plane chassis model and vertical plane axle model is done by the simulation software PROSPER (Delanne, 2003). A double lane maneuver is simulated to excite lateral and vertical dynamics of the vehicle and also a straight line maneuver with real acquisition of an uneven road profile. The results for the yaw plane model are given in Figure 3 and those for the axle model in Figure 4. More results can be found in (Khemoudj, 2010).

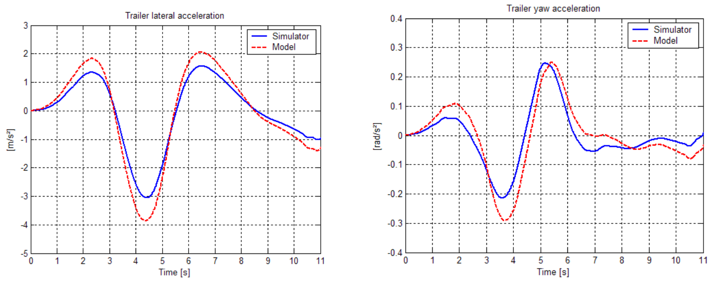

Figure 3 - Simulator (solid) and model (dashed) trailer accelerations (left: lateral, right: yaw)

Figures 3, 4 and 5 show that the model correctly follows the dynamics given by simulation. For the yaw plane model, the semi-trailer accelerations are reconstructed (Figure 3) with small errors which are likely due to the simplification considering a lumped axle group for the semi-trailer axles. Using the axle model, we first simulated a straight line maneuver with an irregular road profile to excite axle hop. In this case, the model follows the simulation and the vertical force peak is correctly reconstructed (Figure 4). A double lane change maneuver (Figure 5) shows the lateral load transfer. The model also tracks correctly the simulation with an acceptable error at the maximum load transfer (at 4 seconds) of 1000 N less that 5% of the total force 25000 N. One can notice that the behavior of the proposed model is close to PROSPER simulator. It also consists of limited and generally known parameters which make it suitable for real-time on-board use. In the next section, estimation methods are developed in order to reconstruct dynamic tire forces.

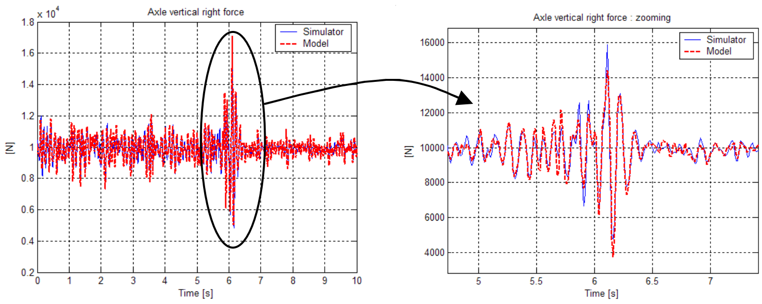

Figure 4 - Simulator (solid) and model (dashed) axle vertical forces for straight line maneuver

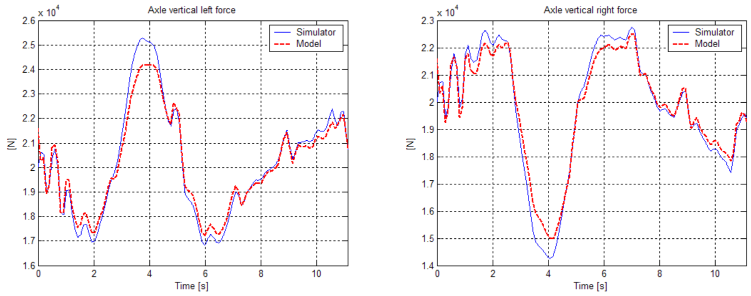

Figure 5 - Simulator (solid) and model (dashed) axle vertical forces for double lane maneuver

4. Vertical Force Estimation

In this section, two methods for vertical force reconstruction are presented

4.1 Inverse Model Method

In the previous section, the validation of the proposed model was shown. In this section, sliding mode observers are developed to reconstruct the contact forces, in two steps: (i) estimation of the lateral forces applied on the axles, using the vehicle yaw plane model; then (ii) evaluation of the vertical forces using the estimated lateral forces.
The state space model of equation (12) can be rewritten as:

$$\begin{bmatrix} \dot{X}_v - Bu \\ Y \end{bmatrix} = \begin{bmatrix} W_1(u) \\ W_2(u) \end{bmatrix} u_F \tag{19}$$

Remark 4. A direct method would use direct root mean square estimate of u_F, however, it can be shown that $W_2^T W_2$ in the RMS formula $u_F = W_2^T (W_2^T W_2)^{-1} Y$ is not invertible. Consequently, the above method cannot be applied, and an alternative method is needed to derive u_F.
To determine the unknown forces u_F, two conditions must be satisfied: (i) the matrix $W = \begin{bmatrix} W_1(u) \\ W_2(u) \end{bmatrix}$ must be of full rank, and (ii) $\dot{X}_v$ must be known. Using a symbolic computation

software, it is verified that W is of full rank, therefore its pseudo-inverse can be calculated numerically at each step. Furthermore, the wheels and yaw accelerations are derived from the measured velocities in X_v, using an exact differentiator. The robust differentiator is based on the sliding mode theory (Imine and Dolcemascolo, 2008) and is given by the set of equations:

$$\begin{cases} \dot{\hat{X}}_{v,i} = Z_i + \lambda_{ii} \left| X_{v,i} - \hat{X}_{v,i} \right|^{\frac{1}{2}} sign(X_{v,i} - \hat{X}_{v,i}) \\ \dot{Z}_i = \alpha_{ii}\, sign(X_{v,i} - \hat{X}_{v,i}) \end{cases} \tag{20}$$

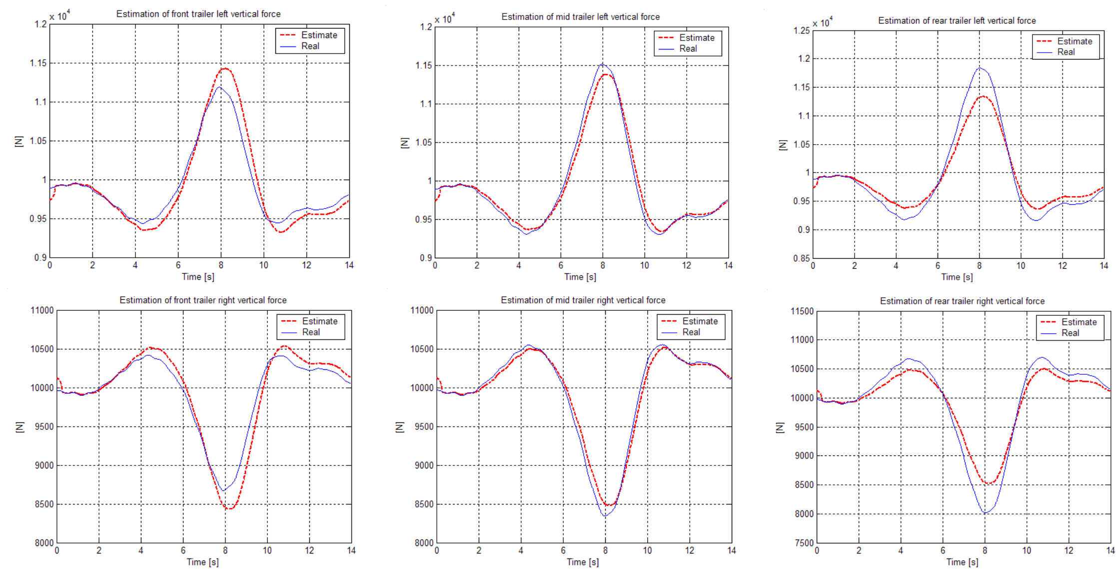

Figure 6 - Estimation of trailer vertical force (from left to right: front, mid and rear trailer axle; up to down: left and right wheels) using inverse model method

Considering the known locally bounded velocities, it exists a choice of α and λ such that the observer error $e_i = \dot{\hat{X}}_{v,i} - \dot{X}_{v,i}$ tends to zero in finite time. The complete proof of this theorem can be found in (Saadaoui et al., 2006). These conditions concern the gain diagonal matrices $\alpha \in R^{5\times5}$ and $\lambda \in R^{5\times5}$. The gains are tuned such that for all $i = 1,\ldots,5$, we have:

$$\alpha_{ii} > \sigma \text{ and } \lambda_{ii} > (\sigma + \alpha_{ii})\sqrt{\frac{2}{\alpha_{ii} - \sigma}} \tag{21}$$

The parameter σ is an upper bound of the second derivative. With the right choice of the gains, the differentiator converges in finite time, moreover, the advantage to use this differentiator that others (Euler approximation for example) is that the exact differentiator is robust to noise and does not create discrepancy between the real and the estimated derivatives. The simulation results (Figure 6) show the effectiveness of the method in estimating vertical forces, the differences at the picks of forces in mainly due to modeling errors as shown in previous section.

4.2 Adaptive Observer Method

The first method works well but in some cases, it can result in ill-conditioned matrices so that the pseudo-inverse results in errors. The idea is to estimate the unknown forces in the yaw plane without using an inverse model. For that, a modification of the state space model is done:

$$\begin{cases} \dot{X}_1 = W_1(u)X_2 + Bu \\ \dot{X}_2 = \mu \\ X_2 = u_F \end{cases} \qquad (22)$$

where μ is an unknown input. The estimation method is based on two successive steps: (i) to estimate the quantity vector $W_1(u)X_2$, and once this estimation is achieved, (ii) to introduce a classic estimator to evaluate the unknown forces contained in X_2.

Step 1. the quantity $W_1(u)X_2$ can either be estimated by a first order observer or by a differentiator; the use of a differentiator is preferred because it is not imposed to use a filter for chattering. The differentiator of equation (20) is used, after convergence, one can obtain: $W_1(u)X_2 = \dot{\hat{X}}_1 - Bu$ with $\dot{X}_1 - \dot{\hat{X}}_1 \to 0$ in finite time noted t_1

Step 2. after convergence of the differentiator in step 1, the vector $W_1(u)X_2$ is a known. We can therefore propose the first order observer: $\dot{\hat{X}}_{2,i} = \hat{\mu}_i + K_{ii} sign\left([W^*(u)\tilde{X}_2]_i\right)$ for $i = 1,...,8$ with $\hat{\mu}_i$ an estimation of μ_i, given that this parameter is in reality unknown, one can give an arbitrary constant value $\hat{\mu}_i = 0$. K is the sliding mode observer gain, a diagonal matrix of dimension 8×8. $\tilde{X}_2$ is the observation error: $\tilde{X}_2 = X_2 - \hat{X}_2$ and $W^*(u)$ is an 8×8 matrix composed of rows from $W_1(u)$ and $W_2(u)$ disposed in such a manner that the diagonal elements of $W^*(u)$ are different from zero.

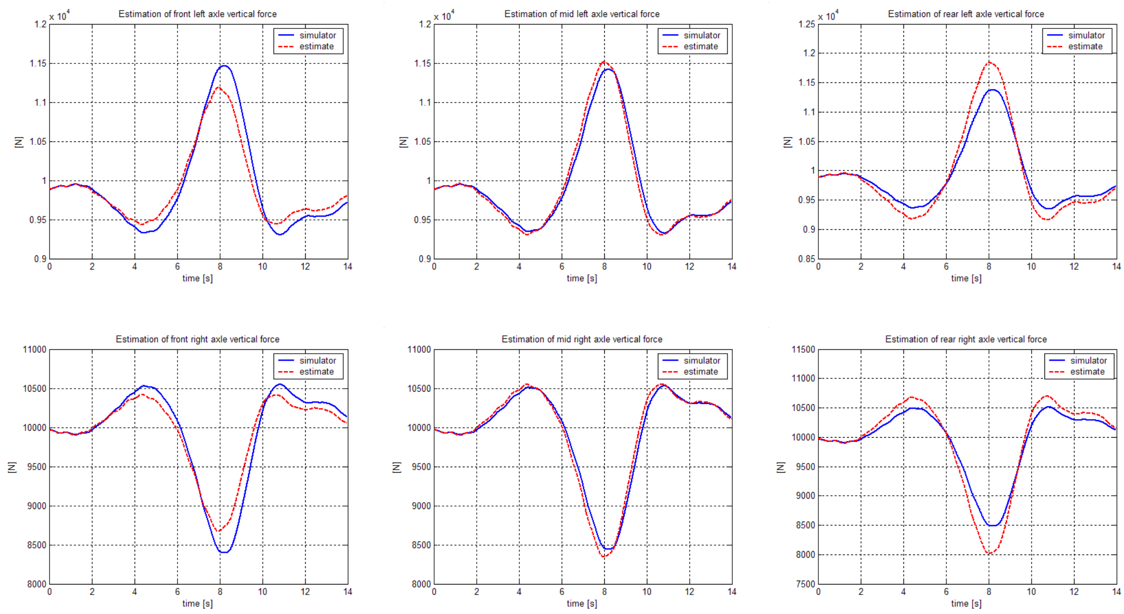

Figure 7 - Estimation of trailer vertical forces (from left to right: front, mid and reear trailer axle; up to down: left and right wheels) using adaptive observer method

To ensure the convergence, a variable adaptive gain is used for the sliding observer K so that the observer converges to zero. The adaptive gain is given by: $K = W^T(u)K^*$. The proof of the convergence can be shown by choosing the Lyapunov function $V = \frac{1}{2}\tilde{X}_2^T \tilde{X}_2$. The gain K^* is constant diagonal matrix $K^* = diag(K_i^*)$ with $i = 1,...,8$ satisfying the condition

$K_i^* >> \max\left\{ abs\left\{ \tilde{\mu}_i \Big/ \left\{ W^{*T}(u) \right\}_{ii} \right\} \right\}$. It is noticed that the estimation results (Figure 7) are similar to the previous method with the advantage of avoiding inverting the matrix model.

5. Hybrid Active Control

Impact forces being estimated, it is now possible to use them as indicator for applying a control strategy. The aim of the control is to have the best trajectory of the vehicle with respect to load transfer. To assist the driver in turning corners or lane changing, a hybrid control strategy is introduced. The originality is that the controller is switched to 'on' only on critical situation identified by both the load transfer ratio evaluated continuously from estimated forces. The hybrid control strategy is given by the diagram of Figure 8.

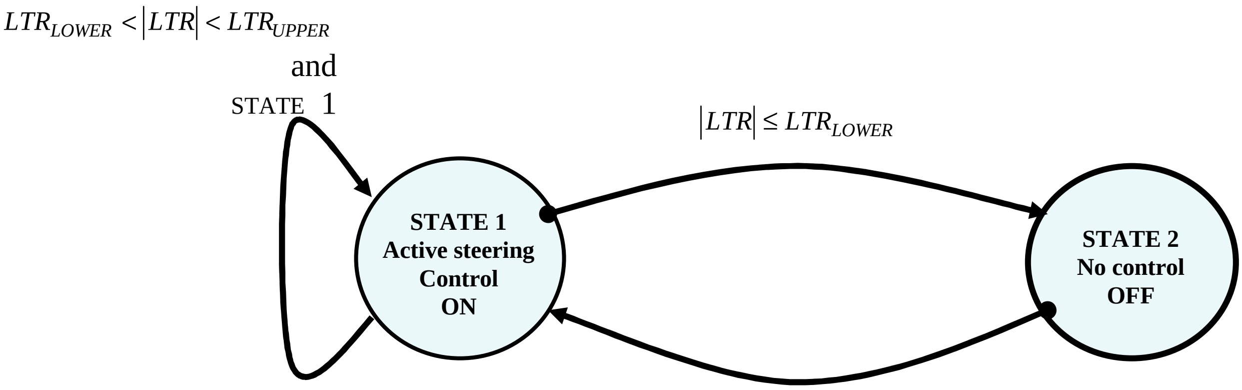

Figure 8 – Hybrid control scheme

There are two different cases: (i) when the controller is 'off', it means that only the driver acts on the steering of the front wheels; this can be considered as the nominal situation when no instability or danger is identified in the vehicle; (ii) when the controller is 'on', which occurs when important load transfers are identified in the semi-trailer. The transition between both cases is based on the load transfer ratio, introducing two thresholds: an upper threshold denoted LTR_{UPPER} and a lower threshold denoted LTR_{LOWER}. The transitions are made as follows:

- if $|LTR| \leq LTR_{UPPER}$, then the controller is '**off**', only the driver acts on the vehicle;
- when $|LTR| > LTR_{UPPER}$ and the controller is '**off**', it is put '**on**';
- if $LTR_{LOWER} < |LTR| < LTR_{UPPER}$ and the controller is '**on**', it is kept active until $|LTR| \leq LTR_{UPPER}$

The hybrid control strategy is summarized in the diagram of Figure 9.

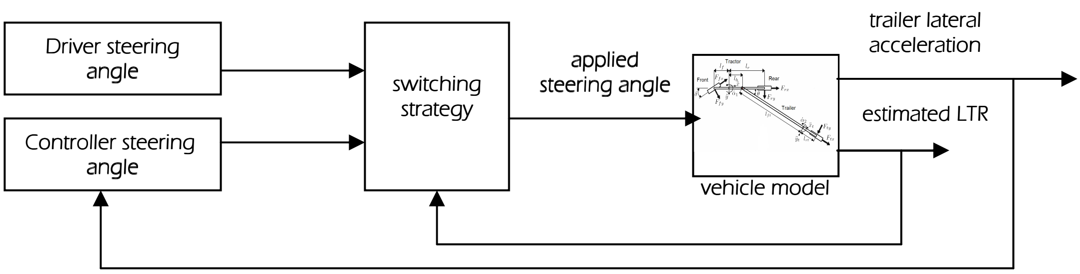

Figure 9 – Steering control loop

When the controller is active, it acts on the steering angle. The control algorithm aims to stabilize the semi-trailer lateral acceleration to zero. A yaw-model of the articulated vehicle, validated by PROSPER, is used. The controller is based on PI regulator. More sophisticated and robust control methods will be considered in future as we are interested in testing the effectiveness of the hybrid control strategy.

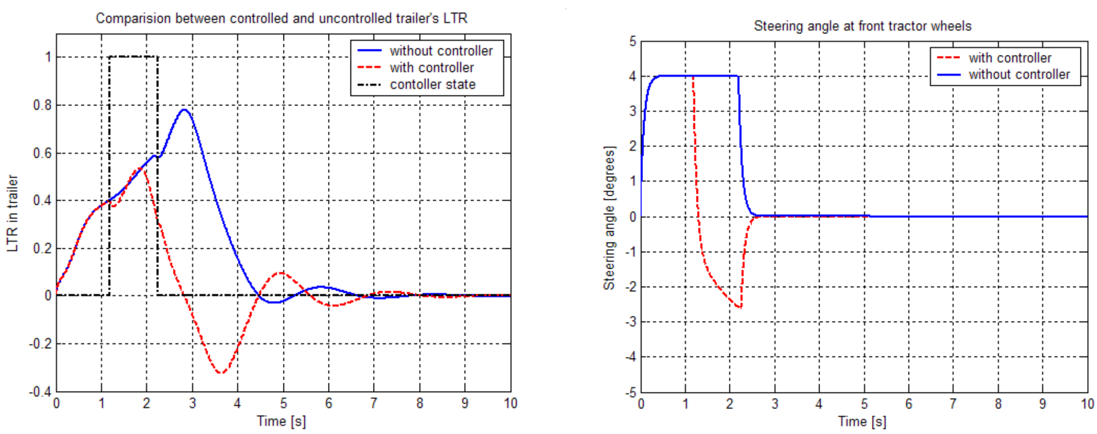

Figure 10 – Left: the total LTR in the trailer with and without controller, right: the steering angle with and without controller

For simulation requirement, we have chosen the upper LTR limit : LTR_{UPPER} =0.40 and the lower LTR limit: LTR_{UPPER} =0.30. Figure 9 (left) shows the semi-trailer total LTR with and without the use of the controller and only with driver. The controller state is equal to 1 when it is '**on**' and 0 when '**off**'. Figure 10 (right) shows the steering applied by the driver (without the controller) and the steering applied to the vehicle when the hybrid control runs. The controller is '**on**' when the LTR is over 0.4 and the controller is switched to '**off**' when the LTR is under 0.3, the lower value. When the maneuver is done without controller, the LTR reaches a high value of 0.8 which can result in a rollover. The controller stabilizes the vehicle at an early stage, helping to get a more secure driving and causing less damage to the pavement.

6. Conclusion

Control theory tools are used as state observers and hybrid control in order to estimate and stabilize impact forces of heavy good vehicles. Estimation and control techniques are presented for a better operation of heavy good vehicles. The techniques can be used for commercial use, while they are based on optimized, low cost and easy to install sensors. Some additional control techniques, such as active suspension, would be added to the hybrid configuration in order to increase vehicle safety and to minimize infrastructure damage due to large impact forces variations.

7. References

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