EXPERIMENT FOR TYRE VERTICAL FORCES ESTIMATION ON THE ACCELERATED LOADING FACILITY EQUIPMENT

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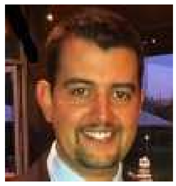

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Abstract

In this paper, an experiment for vehicle forces estimation is presented. The purpose of this experiment is to provide understanding and validation for future experiments on a heavy duty vehicle. The main points discussed in this paper concern the study of evaluating the wheel vehicle forces using either suspension air cushions pressure sensor or suspension deflection measuring. The identification of parameters of stiffness and damping of the suspension and the use of an exact differentiator based on sliding mode theory to assess the vertical wheel force are presented. Strain gauges are used as reference measurements. The validation tests were carried out on an instrumented test bench with wheels rolling at various speeds and road longitudinal profiles.

Keywords: ALF system, tyre forces, sliding mode observer, estimation, identification.

1. Introduction

Heavy duty vehicle, while rolling, apply important dynamic wheel loads on the pavement due to road infrastructure and load transfers coming from driving maneuvers. These dynamic loads can cause damage for the pavement and be the source of dangerous accidents. It is therefore important to quantify the magnitude of these forces and stabilize them in order to enhance the security of the vehicle.

Multiple studies are made for evaluation of the vertical forces, we can cite among them methods based on direct instrumentation and measurment of contact forces like using strain gauges on the axles and wheel hubs or the use of a laser sensor (Blanksby *et al.*, 2008) to measure dynamically the tyre deflection and deduce the applying dynamic load. We can also cite (Tuononen, 2009) also used a laser sensor consisting in optically measuring carcass displacement of the tyre using a particular optical sensor installed on the tyre rim. These techniques involve complex installation and calibration of the sensor and often require using a test bench.

Other additional researches suggest introducing control theory tools to estimate the contact forces based on more usual and easy to install sensors. These methods are based on the modelling of the dynamics of the vehicle in order to evaluate the contact forces using estimation algorithms. We can cite (Siegrist, 2003) who uses a method based on Kalman filter to estimate the contact forces in an off-highway mining truck. We can also cite the studies of (Khemoudj *et al.*, 2010a) and (Khemoudj *et al*., 2010b) who uses some estimation algorithms based on sliding mode thoery for heavy vehicle tractor-trailer model.

The experiment in this paper aims to provide some comparison between a method based on estimation algorithm using a suspension deflection sensor and another using the air pressure of suspension cushions, both methods are compared to the reference measurmeent provided by strain gauges.

This paper is divided into four main sections. In the first section, we describe the test bench and the used instrumentation. In the second section, we present the used model for the wheels dynamics modelling. In the third section, we present the developed estimation technique and present and discuss in the fourth section the experimental results. We finally conclude with some perspectives in the last section.

2. Testbench description ans instrumentation

The experiment is conducted in partnership with the Australian Road Research Board. It uses the Accelerated Loading Facility equipment. The ALF simulates heavy vehicle rolling wheels through a loaded half axle with dual or single tyres. The loading wheels are driven in one direction to replicate a real trafficking of a heavy duty vehicle. The Figures 1 and 2 show views of the equipment.

To proceed to the experiment, number of sensors are installed on the ALF, these sensors are:

- strain gauges on the axle for getting a reference measurement on the vertical forces.
- suspension deflection sensors LVDT to measure the deflection of suspension.
- pressure sensors mounted to measure the pressure of the air suspension cushions.
- accelerometers to measure the wheels hop acceleration.

Figure 1- Lateral view of ALF

Figure 2- Tandem on ALF

Due to unavailability, there is no use of extra dynamometric wheel during this experiment. The setup of the different sensors are shown in Figures 3 to 6. The gauges are placed near to the wheel hub. They can measure the deformation due to shear force and hence the vertical force. This method provides a baseline measurement to evaluate later the estimation of vertical forces.

Figure 3- Air cushions suspension similar to HDV

Figure 4- Air pressure transducer (APT)

Figure 5- Accelerometers on wheel hub

Figure 6- Strain gauges on wheel axle

3. Rolling semi-axle model

As described above, the test bench contains two tandems. Each is equivalent to a half axle.

Let us note F_z the vertical force, F_s the suspension force, a_z the vertical accelereation, m_w the wheel mass, z_r is the longitudinal road profile and K_t is tyre stiffness. The vertical dynamics of the wheel yields to the equation:

$$m_w \ddot{z}_w = F_s + K_t(z_w - z_r) \tag{1}$$

The vertical force is then given by:

$$F_z = K_t(z_w - z_r) \tag{2}$$

The aim is to estimate the vertical forces F_z and compare the estimated forces to the reference strain gauges measurments.

4. Vertical forces on wheel estimation

There are two ways to evaluate the suspension force F_s, either from pressure measurements or from measurements of deflection. This experiment evaluates the performance of both types of measurements in terms of the resuting estimated vertical force F_z. The first method is to assume a proportionality between the suspension force and the pressure at the air cushion. This technique is also used in the so-called on-board static weighing, in evaluating the carried load of heavy duty vehicles. The second method is based on the use of the suspension deflection and some estimation algorithms. The first step will be to identify the suspension parameters in terms of stiffness and damping.

4.1 Identification of suspension parameters

The first method used to evaluate the vertical force is based on the air pressure sensors, we assume that the suspension force F_s is proportional to the pressure measured at the air cushions.

$$F_s = K_p P \tag{3}$$

Where K_p is the coefficient of proportionnality and P is the pressure in the suspension cushions. To calculate the coefficient of proportionality, a number of measurements are made statically. These static tests consists in measuring the total vertical forces under the wheel while not rolling. To the measured total force is subtracted the unsprung weight which includes the weight of the wheels, levers, air cushions and shock absorbers. We find a gain $K_p = 817N / psi$.

The second method allows the calculation of vertical forces by the use of suspension deflection. In this case, the suspension force is given by:

$$F_s = K_s d_s + C_s \dot{d}_s + F_{s,static} \tag{4}$$

Where d_s is the measured deflection, $\dot{d}_s$ is the deflection variation velocity (derivative of the deflection), $F_{s,static}$ is the static suspension forces when wheel is not rolling, K_s and C_s are respectively the unknown siffness and damping coefficients of the suspension we need to identify.

The stiffness coefficient K_s can be calculated from stactic tests. It is obtained from the slope defining the variation of suspension force relative to the suspension deflection mesured by the LVDT. Note that the LVDT measures the distance between the chassis and suspension arms, we use two separate tests to evaluate the stiffness.

To evaluate the damping coefficient, we rely on a dynamical test and a graphical method. Knowing that the suspension system can be considered as a damped second order system, we first evaluates the natural frequency $\omega_n = \sqrt{\dfrac{K_s}{M_s}}$ where K_s is the stiffness previously calculated and M_s the sprung mass. Then, from dynamic tests, we evaluate the pseudo-period noted T_p which is the time-distance between two successive peaks of a damped response as shown in Figure 7.

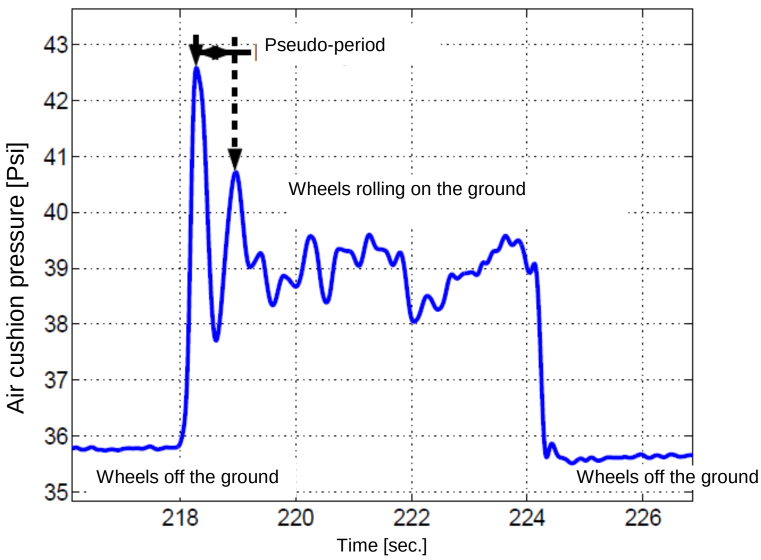


Figure 7- Graphical determination of damping coefficient

From the pseudo-period T_p ,we can calculate the pseudo-frequency $\omega_p = \dfrac{2\pi}{T_p}$.

The next step is then to calculate damping ratio ζ from the formula $\omega_p = \omega_n \sqrt{1-\zeta^2}$ and finally the damping coefficient of the suspension system is given by: $C_s = 2\zeta\sqrt{K_s M_s}$. According to the dynamic tests, we find $C_s = 61021 N/ms$.

4.2 Determination of vertical forces at wheel

To estimate vertical forces, we need to evaluate the part due to the damping. To do so, we introduce the exact differentiator which allows to estimate the suspension deflection velocity from the measured delection provided by the LVDT. The robust differentiator is based on the sliding mode theory (Davila, 2006) and is given by:

$$\begin{cases} \hat{\dot{d}}_s = Z + \lambda \left|\tilde{d}_s\right|^{\frac{1}{2}} sign\left(\tilde{d}_s\right) \\ \dot{Z} = \alpha\, sign\left(\tilde{d}_s\right) \end{cases} \tag{5}$$

with $\tilde{d}_s$ the estimation error: $\tilde{d}_s = d_s - \hat{d}_s$, $\hat{d}_s$ the estimate of d_s and $\hat{\dot{d}}_s$ the estimate of $\dot{d}_s$. α and λ are the observer gains. Given that the suspension deflection is bounded, there exists values for α and λ such that the observer error $e = \hat{\dot{d}}_s - \dot{d}_s$ tends to 0 in finite time t_0 (Saadaoui, 2006). Sufficient conditions for convergence of the differentiator are given on gains α and λ. These gains are tuned such that:

$$\alpha > \sigma \tag{6}$$

and

$$\lambda > (\sigma + \alpha)\sqrt{\frac{2}{\alpha - \sigma}} \tag{7}$$

σ is an upper bound for the second derivative of d_s : $\sigma = \max\{\ddot{d}_s\}$.

With the correct tuning of the gains, the differentiator converges in finite time, moreover, the advantage to use this differentiator than 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.

5. Experimental results

The ALF device is remotely controlled using a servo system by choosing a reference velocity. The latter is reached after 6 to 7 successive passings. Several tests were performed. Each test consists of a number of passings along a 12-meters track. For a passage to go, the wheels are in contact to the ground and for a back passage, the wheels are off the ground.

Wood boards are placed 5 meters from the starting point of the wheels on the track. Two dimensions were used, a first board of dimension 26x29 (*width* x *height*) and a second of dimension 140mm x 18mm.

In this section, we compare the results of the reconstruction of impact forces to the reference measurements from strain gauges. We recall that the dual wheels on the device ALF are limited to a straight line movement, no yaw dynamics are involved. The use of gauges provides an information about the shape of the contact forces. We call "reference" the measurement given by the gauges to distinguish it from the estimated forces.

Note that the goal is not to exactly match the estimated forces on the reference given by the strain gauges as the gauges also provide measurement errors in regards to the real contact forces. The most reliable validation tool remains the use of dynamometric wheels which provide the closest measurement to the real force values.

From the Figures 8 to 11, we observe that the shapes of impact forces from the model and those given by the reference (measurement of strain gauges) are close to each other. The presence of a disturbance on the track allows to excite the vertical dynamic load. From the use of deviation histogramms, we note that the vertical forces used from the APT sensors remain closer the reference (deviations between 0 to 10%) than the estimated forces from LVDT (deviations between 0 to 20%). Compared to the LVDT, the forces obtained by using the pressure sensor have better similarity with the reference. By assuming that the strain gauges are closer to real forces at low speeds and in strait line, the APT are in this case more convenient, nevertheless; some addition tests on higher speeds and in a vehicle will allow to provide more understanding and a deeper comparision between the estimations and mesurement methods.

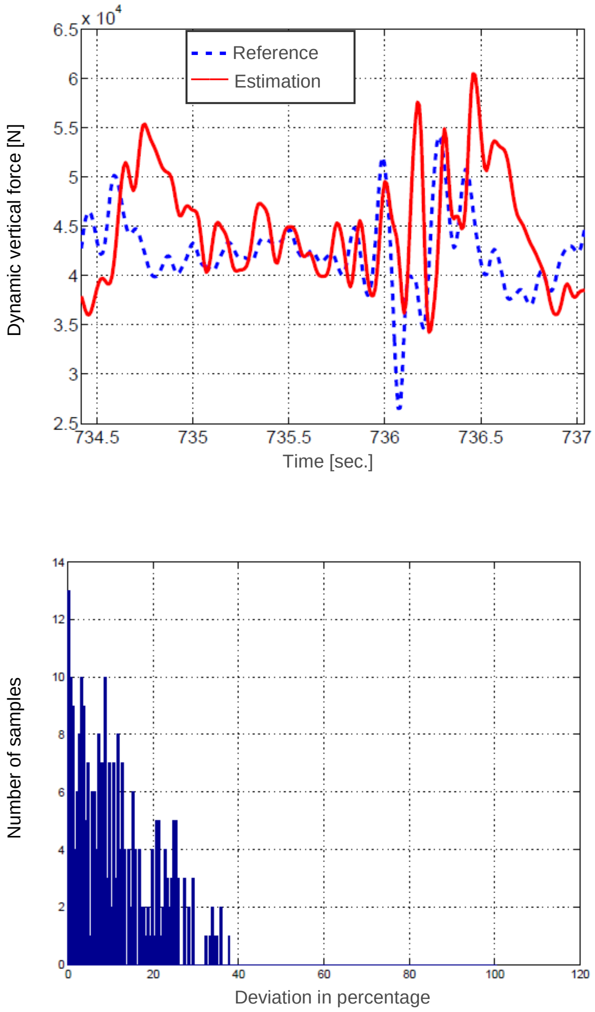


Figure 8 – Estimation based on LVDT and differentiator – passing No.1

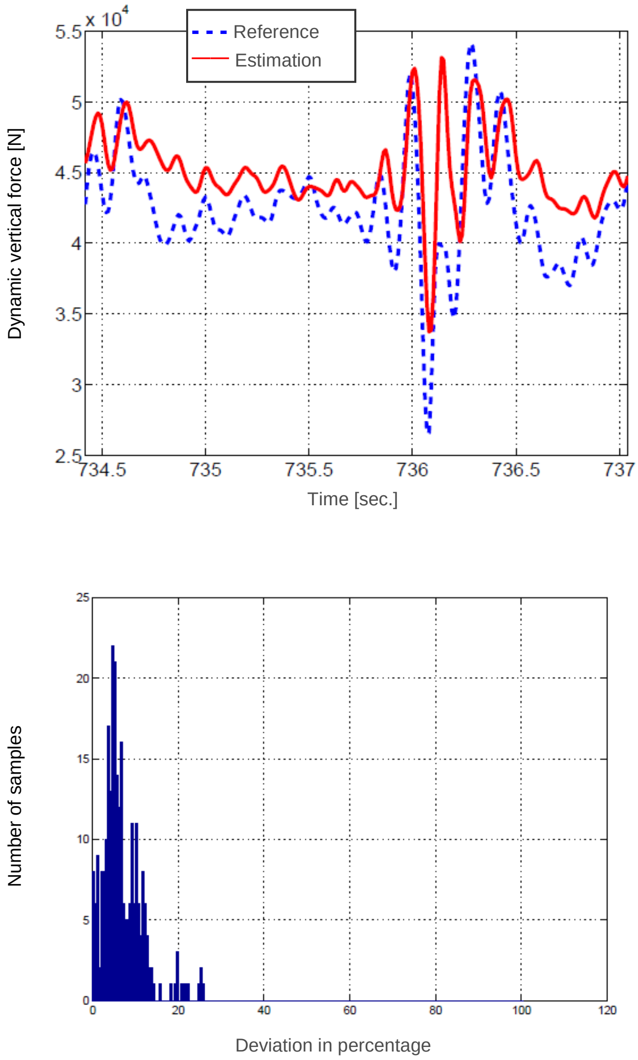


Figure 9 - Estimation based on APT – passing No.1

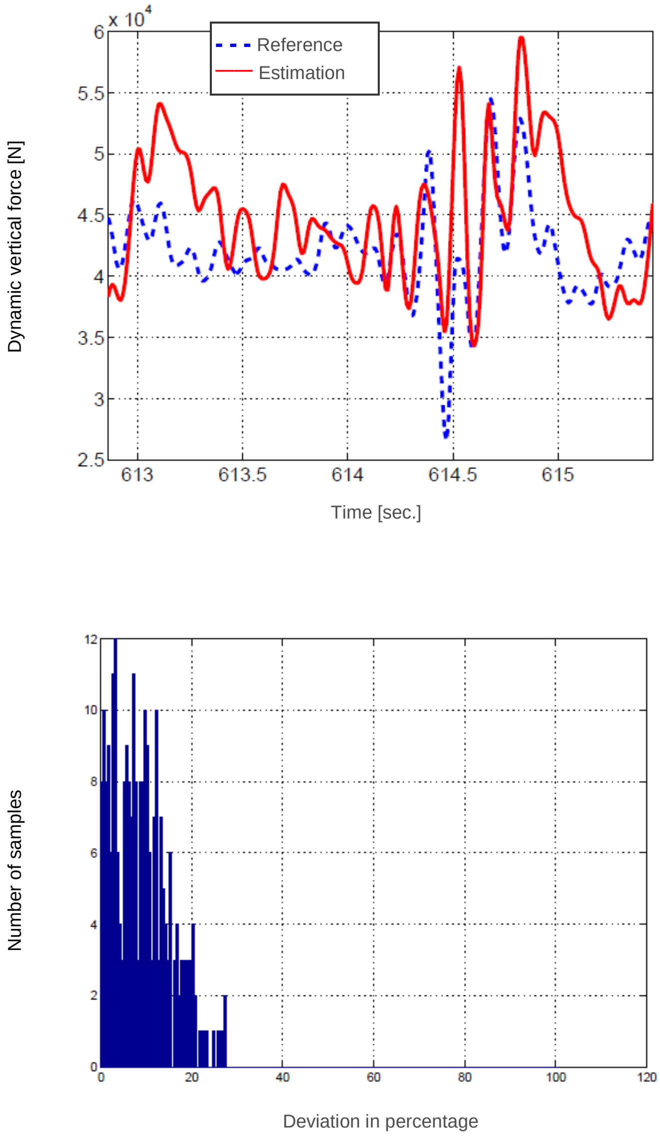
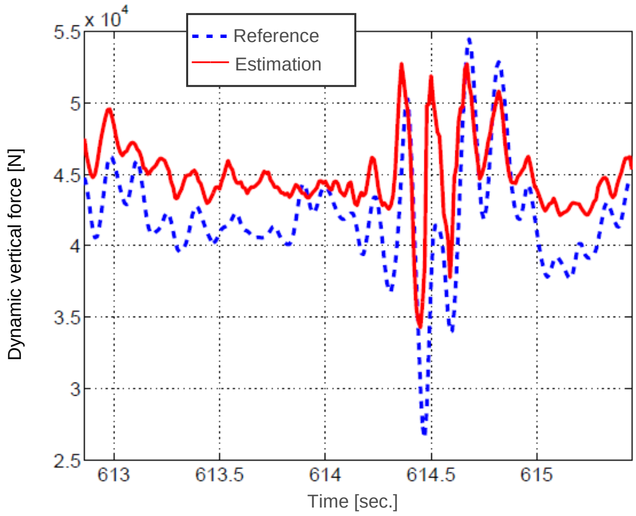


Figure 10 – Estimation based on LVDT and exact differentiator – passing No.2



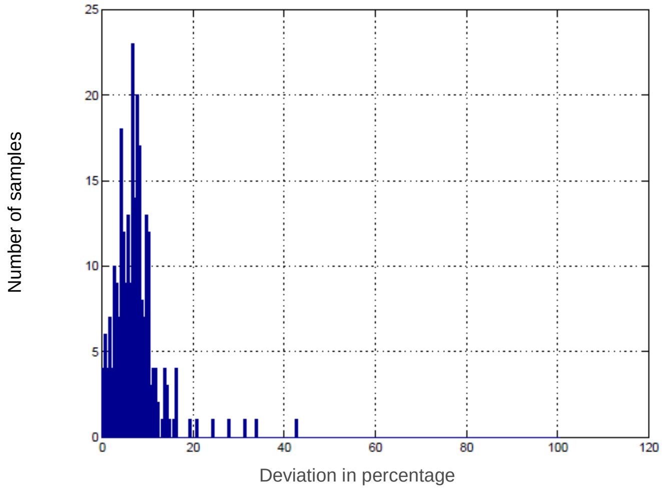


Figure 11 – Estimation based on APT – passing No.2

6. Conclusion

In this paper, an experiment on the accelerated loading facility is presented. The aim is to provide some preliminary tests to get a better understanding on the sensors and methods to use for the estimation of the vertical contact forces in heavy vehicles. Air pressure sensors and LVDT were both used to reconstruct the vertical forces which are compared to the reference forces provided by the strain gauges installed on the wheel hubs. The parameters of the suspension are identified using a graphical approach and the obtained shapes of the forces are quite similar. The shape of the forces issued from the air pressure sensor are closer to the reference than those provided by the LVDT in this case (low speed and straight line rolling).

It is however expected that the LVDT react faster than the air pressure sensors and the vertical forces can be better estimated in important load transfers in the vehicle. This can be more deeply investigated by testing on a complete vehicle.

7. Acknowledgement

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8. References

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