

Real-time Tire Imbalance Detection Using ABS Wheel Speed Sensors

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ABSTRACT

This paper proposes an approach to use ABS wheel speed sensor signals together with other vehicle state information from a brake control module to detect an unbalanced tire or tires in real-time. The proposed approach consists of two-stage algorithms that mix a qualitative method using band-pass filtering with a quantitative parameter identification using conditional least squares. This two-stage approach can improve the robustness of tire imbalance or imbalances. The proposed approach is verified through vehicle testing and the test results show the effectiveness of the approach.

INTRODUCTION

As the need for fuel economy increases, the percentage of light weight vehicles over the total vehicle population is increasing. Light weight vehicles are generally more sensitive to vibration disturbances than their heavier counter parts such as light trucks. Many sources can induce such vibration disturbances which can cause occupants' uneasiness and potential mechanical problem to vehicle parts and subsystems.

One of the disturbance sources is the one generated from the rotation of tire and wheel assemblies such as tire imbalances. Proper tire balancing is as critical as the smooth operation of the other subsystems, especially in attaining good ride quality. While a well balanced tire has a uniform distribution of mass around its rotational axle, an imbalanced tire has an un-even mass distribution that can generate unwanted periodic motion such as forced vibrations to degrade the ride quality, shorten the life of suspension components, and annoy

occupants. The vibration from the imbalance can be felt by occupants through vehicle shaking. The driver can also feel the steering wheel shake and nibble. If significant tire imbalances are neglected for a long period of time, excessive tire wear and suspension performance deterioration can happen. The induced vibrations from tire imbalances are dependent on vehicle's travel speed and they are usually noticeable beyond, for example, 60mph.

Besides ride quality degradation and potential subsystem problems, tire imbalance can influence the longitudinal motion behavior of the vehicle. The driving dynamics and performance of a vehicle are related to the traction capabilities of its tires. Tire imbalances cause fluctuations in the tire normal loadings that can lead to traction fluctuation such that the average tractions used to move the vehicle are reduced. Hence the imbalance could theoretically affect a vehicle's fuel economy. The fluctuation of the normal loadings can also affect the braking capability and brake efficiency of the vehicle.

Tire imbalance can be corrected by using expensive and heavy computerized tire balancer such as a dynamic high speed spin balancing machine used in many tire centers and stores. While it can be detected during regular tire service, a real-time determination through on-board sensors is much more appealing for reliable detection since tire imbalances might occur in between tire services. The real-time detection can also generate signals that can be used by electronic control systems.

Theoretically tire imbalance can be detected in real-time using spindle-mounted accelerometers or force transducers.

Known methods for detecting out-of-roundness of tires, under-inflated tires, defective front-end alignment or off-center loads on are disclosed, for example, in [1,2]. A monitoring system for measuring tire imbalance using acceleration signals from accelerometers on wheels is described in [3]. In [4], the relatively cost effective method of using ABS wheel speed sensors to detect imbalances is disclosed. The approach in [4] is a pure software solution that does not need additional hardware. Due to the popularity of ABS in modern vehicles, such an approach can potentially be incorporated in many vehicles. This paper provides an extension of the approaches disclosed in [4].

The paper is organized as in the following. We first discuss modeling the forced wheel hop motion due to the imbalance in a tire and connecting it with the wheel rotational speed and the dynamic rolling radius of the tire. A simple description about the conditional least squares method is discussed next. Then we discuss real-time implementation of the proposed approach using mixed quantitative and qualitative solutions. The test results are discussed next. The last section concludes the paper.

FORCED WHEEL HOP MOTION DUE TO IMBALANCED TIRE

In order to detect a tire imbalance, we need to understand the vibration behavior caused by the asymmetric wheel and tire assembly. Let's use one corner of a vehicle to illustrate this. Such corner includes an independent suspension, a tire and wheel assembly, and a quarter of the vehicle body which can be depicted in [Figure 1](#).

Let z_s denote the body vertical displacement, z_w the wheel vertical displacement, w the vertical road profile and F_{susp} the suspension force if a controllable suspension is used ($F_{susp} = 0$ if the suspension is passive). Denote the sprung mass as M_s , the unsprung mass as M_w , the passive suspension stiffness as K_s , the passive suspension damping as C_s , the tire vertical stiffness as K_t , and the tire vertical damping as C_t . Then the equation of motion of this corner model can be expressed as in the following

$$\begin{aligned} M_s \ddot{z}_s &= -K_s z_{sh} - C_s \dot{z}_{sh} + F_{susp} \\ M_w \ddot{z}_w &= K_s z_{sh} + C_s \dot{z}_{sh} - F_{susp} - K_t z_{td} - C_t \dot{z}_{td} + F_z \end{aligned} \quad (1)$$

where $z_{sh} = z_s - z_w$ is the suspension height, $z_{td} = z_w - w$ is the tire deflection, F_z is the tire imbalance induced disturbance force projected in the wheel hop (or vertical) direction.

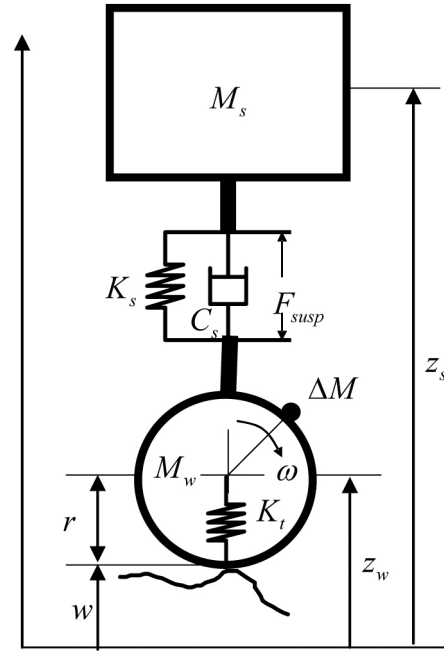


Figure 1. A quarter car model with an imbalanced tire

Assume the rotational velocity of the tire is ω and F_z can thus be expressed as in the following

$$F_z = M_w \alpha^T \begin{bmatrix} \omega^2 \sin \Omega + \dot{\omega} \cos \Omega \\ \omega^2 \cos \Omega - \dot{\omega} \sin \Omega \end{bmatrix} \quad (2)$$

where Ω is the angle traveled by the vector connect the imbalance mass and the rotation center of the tire. Ω can be computed through the integration of the rotational speed ω

$$\Omega = \int_0^t \omega(\tau) d\tau \quad (3)$$

and α is a vector of dimension 2.

If we denote θ_0 as the initial angle between the vector connecting imbalance mass and the rotation center and the vertical direction, then α can be explicitly expressed as:

$$\begin{aligned} \alpha_1 &= \frac{\Delta M}{M_w} \rho \cos \theta_0 \\ \alpha_2 &= \frac{\Delta M}{M_w} \rho \sin \theta_0 \end{aligned} \quad (4)$$

where ΔM is the added mass due to the tire imbalance and ρ is the distance between the center of ΔM and the rotational center of the tire.

If α can be computed, then the imbalance mass ΔM can be determined from α as in the following:

$$\Delta M = \frac{\sqrt{\alpha_1^2 + \alpha_2^2}}{\rho} M_w \quad (5)$$

Since the vertical loading applied to the tire can affect its rolling radius, the tire deflection can be related to the dynamic rolling radius as

$$z_{td} = r - r_0 \quad (6)$$

where r_0 is the static radius of the tire under static vehicle loading. If the tire does not have any slip, the tire rolling radius can be further related to the linear velocity of the rotational center of the tire

$$v_x = \omega r \quad (7)$$

Notice that v_x is different from the ABS wheel speed sensor signal that is usually defined as

$$v_{wss} = \omega r_{\text{nominal}} \quad (8)$$

with a nominal rolling radius r_{nominal} which is usually set to the static radius r_0 .

Using the tire rolling radius to replace z_{sh} and z_{td} and adding the two equations of (1) together lead to

$$F_z = M_s \ddot{z}_s + M_w (\ddot{r} + \ddot{w}) + K_t (r - r_0) + C_t \dot{r} \quad (9)$$

Plugging (9) into (2) leads to

$$\alpha^T U = y \quad (10)$$

with

$$U = \begin{bmatrix} \Phi_t \{ \omega^2 \sin \Omega + \dot{\omega} \cos \Omega \} \\ \Phi_t \{ \omega^2 \cos \Omega - \dot{\omega} \sin \Omega \} \end{bmatrix}$$

$$y = r + \Phi_t \left\{ \frac{M_s}{M_w} \ddot{z}_s + \ddot{w} - 2\eta_t \omega_t r_0 \right\} \quad (11)$$

where Φ_t is a filter of the following transfer function

$$\Phi_t(s) = \frac{1}{s^2 + 2\eta_t \omega_t s + \omega_t^2} \quad (12)$$

and the coefficients in the filter can be determined from the inertial parameters of the corner model as

$$\omega_t = \sqrt{K_t / M_w}$$

$$\eta_t = \frac{C_t}{2\sqrt{K_t M_w}} \quad (13)$$

Considering that the road disturbance w is usually a wideband noise and filter Φ_t in (12) is a low-pass filter, y in (11) can be further simplified as

$$y \approx r + \Phi_t \left\{ \frac{M_s}{M_w} \ddot{z}_s - 2\eta_t \omega_t r_0 \right\} \quad (14)$$

Since the vehicle heave motion z_s usually has frequency contents around 1 Hz, hence there exists a high-pass filter Φ such that

$$\Phi \left\{ \Phi_t \left\{ \frac{M_s}{M_w} \ddot{z}_s - 2\eta_t \omega_t r_0 \right\} \right\} \approx 0 \quad (15)$$

Applying filtering Φ to (10) and using condition (14) and (15) leads to

$$\alpha^T \Phi \{U\} = \Phi \{r\} \quad (16)$$

which implies that α can be related to a function of the wheel speed and the filtered tire dynamic rolling radius.

If the vehicle is driven on a constant speed or the wheel speed is constant, then ω terms in the expression for U are zero such that

$$U = \omega^2 \Phi_t \begin{bmatrix} \sin(\omega t) \\ \cos(\omega t) \end{bmatrix} \quad (17)$$

The magnitude of the tire rolling radius due to the tire imbalance can now be expressed as the following function of

the wheel rotation speed ω since the inputs to the transfer function Φ_t are $\sin(\omega t)$ and $\cos(\omega t)$

$$|r| = \frac{|\alpha| \omega^2}{\sqrt{(\omega^2 - \omega_t^2) + (2\eta_t \omega \omega_t)^2}} \quad (18)$$

Notice that (18) also theoretically proves that the rotational vibration frequency of the imbalance is the wheel rotational speed ω which matches the common sense.

The peak magnitude of (18) happens when $\omega = \omega_t$. Hence the critical vehicle speed when a driver can feel the significant vibration due to tire imbalance can be computed as

$$v_{xc} = \omega_t r_0 = r_0 \sqrt{\frac{K_t}{M_w}} \quad (19)$$

The critical speed can be verified through vehicle testing. Figure 2 shows a test run with wide open throttle to drive the vehicle from zero speed to around 130 kph. The wheel speed of the front left and right tires are shown on the top row of the Figure, while the bottom row should the computed dynamical rolling radius (dotted line in blue color) of a tire with imbalance and of a tire without an imbalance (solid line in

red color, computed from the wheels without imbalance). The value for the imbalanced tire (in dotted blue color) starts to diverge from the value of the normal tire (solid line in red color) as vehicle speed increases and the divergence achieves its peak around the vehicle's critical speed.

LEAST SQUARES IDENTIFICATION

Equation (16) indicates that the two unknowns need to be determined from one equation. This can be conducted off-line or in real time through the so-called least squares method [5]. One of the necessary conditions for the least squares to work well is that the input output signals have to be well excited. This corresponds to conditions where the forced vibration due to tire imbalance can be excited.

Since not all the vehicle speeds can excite the wheel hop vibration due to tire imbalance, the general least squares in [5] will be conducted when there is enough evidence that the vibration is excited. A conditional least squares method is used here.

Let I_{excit} be a flag which can be active when the wheel hop due to the imbalance is excited, the conditional least squares can be expressed as in the following

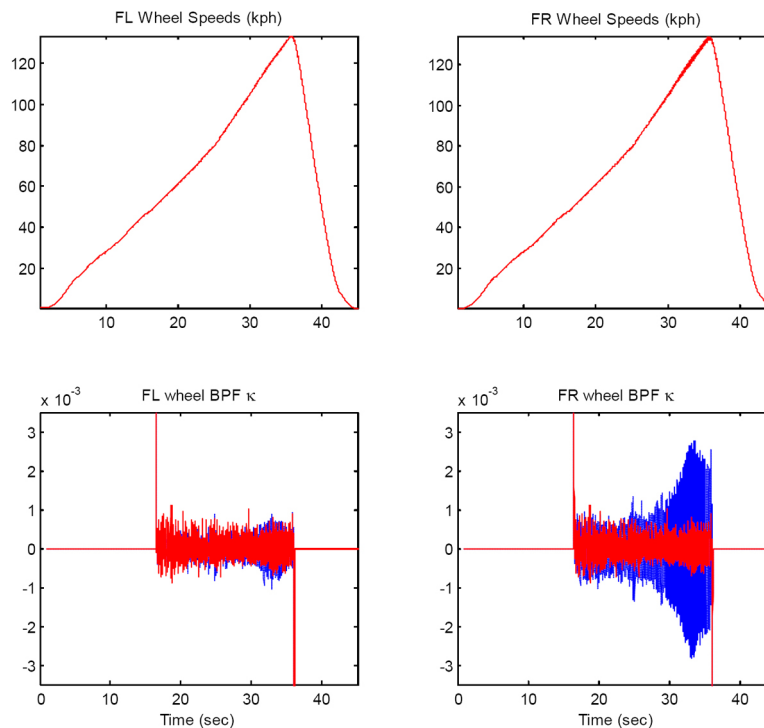


Figure 2. Top: front-left and front-right wheel speeds; Bottom: the front-left and front-right wheel band-pass filtered dynamic rolling radius (in blue and dotted line) vs. the band-pass filtered normal wheel dynamic rolling radius (in red and solid line). An imbalance mass was installed in the rim of the front-right wheel.