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Observer based road-tire friction estimation for slip control of braking system

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Abstract

In this paper the friction coefficient at contact surface of the road and tire of ground vehicle has been estimated for the anti-lock braking system. The non-linear Dugoff's friction model has been taken here to characterize the tire forces. The main aim is to update the controller with current information about the friction so that the vehicle can be manoeuvred by anti-lock braking controller in an emergency braking situation. A robust non-linear sliding mode observer has been suggested to observe the tire force and the longitudinal velocity of the vehicle. Simulation is carried out for different road conditions for a quarter car model and the results are evaluated.

Keywords-anti-lock braking systemt; Dugoff's friction model; quarter car model; sliding mode observer; sliding mode controller.

1. Introduction

In recent years newly developed sophisticated cars are well equipped with the electronic stability controller for passenger's safety and comfort. The controllers are made for maneuvering the vehicle in an emergency situation to avoid road accidents. Such controllers are called anti-lock brake controller or traction force controller. In an emergency braking situation wheel of the vehicle become locked if driver presses brake pedal very hard to stop the car immediately. During this braking situation, the locked wheel leads the vehicle to skid and to go out of driver's control. So the development of the anti-lock braking system or anti-skid controller is associated with the application of smooth controlled brake torque on the

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wheel so that no accident due to skidding is ensured. The brake force applied to the wheel strongly depends on the tire force which comes from the contact surface of road and tire [9]. Therefore the tire model plays an important role to transmit the exact tire force from contact surface to the wheel efficiently. The magnitude of this tire force depends on the normal load of the vehicle and the current friction coefficient of road surface. For emergency maneuvering, it is important to deliver maximum braking torque on the wheel to stop the vehicle within short distance. Thus maximum tire force should be captured with the current information of friction coefficient. Due to the unavailability of economical commercial sensor for online measurement of tire force or road friction coefficient, efficient estimation technique is strongly recommended for this purpose.

The characteristics of the tire force are highly non-linear and model dependent. Several well known tire models such as Lugre tire model [8], Burkhard tire model, Dugoff's tire model [17, 4] are there in the literature to establish the relationship between tire force or friction coefficient and the slip ratio. In this paper Dugoff's tire model is taken to model the system dynamics as this model holds necessary characteristics of the tire forces. Among the above said models the Dugoff's tire model has uniform distribution of pressure and has more realistic approach. Estimation schemes of the frictional force are suggested in literature by many authors [3, 5, 6, 7 and 8] for Anti-lock Braking System. In this paper the estimation of the friction coefficient has been done with the velocity observer. The sliding mode observer [4, 8, 13 and 16] is designed and implemented in the ABS with non-linear sliding mode controller [12, 31, 15 and 16] to deal with non-linearity and uncertainty.

Nomenclature

M	Total mass of the quarter vehicle
F_x	Longitudinal tire force
V	Longitudinal velocity of the vehicle
ω	Wheel angular velocity
J	Total moment of inertia of the whee
r	Radius of wheel
T_{brake}	Braking torque
m_{vs}	The vehicle sprung mass
m_w	Mass of the wheel
g	Gravitational acceleration
F_L	Dynamic load transfer
h_{cg}	Height of the sprung mass c.g.
$\ddot{x}$	Longitudinal acceleration
l	Wheel base
C_i	Tire longitudinal stiffness
C_α	Tire cornering stiffness
λ	Slip ratio
μ	Road friction coefficient
F_z	Normal load
ε_r	Road adhesion reduction factor

2. System Dynamics

2.1. Car model

A quarter car model [2] characterizes the major characteristics of the whole vehicle has been taken here for the ease of application. Free body diagram is shown in Fig. 1.

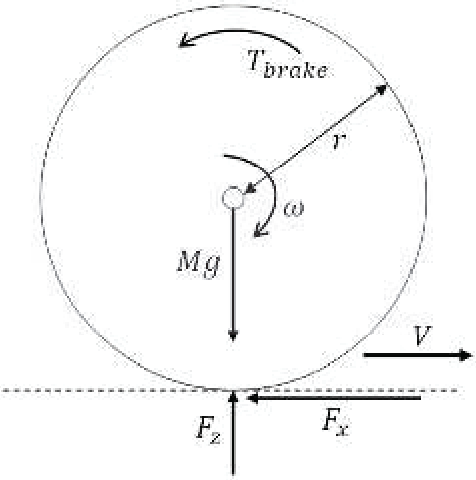

Fig. 1. Free body diagram of the quarter car.

Equation of motion of the vehicle can be represented as

$$\dot{V} = \frac{-F_z\mu(\lambda)}{M} \tag{1}$$

and $\dot{\omega} = \frac{1}{J}(rF_x - T_{brake})$ (2)

Where

$$F_x = F_z\mu(\lambda) \tag{3}$$

The total mass of the vehicle is given by

$$M = \frac{1}{4}m_{vs} + m_w \tag{4}$$

Normal force of the tire is given by

$$F_z = Mg - F_L \tag{5}$$

Where $F_L = \frac{m_{vs}h_{cg}}{2l}\ddot{x}$ (6)

2.2 Dugoff's friction model

Friction model plays an important role to characterize the tire forces. The Dugoff's friction model [2, 4, and 17] shows the direct relationship between tire force and the slip ratio. The typical characteristics curve of the friction coefficient is shown in Fig. 2.The non-linear relation of the longitudinal force of the tire is given as follows

$$F_x = \frac{C_i}{1-\lambda}f(s) \tag{7}$$

Where,

$$f(s) = \begin{cases} s(2-s), if\ s < 1 \\ 1, if\ s > 1 \end{cases} \tag{8}$$

and $$S = \frac{\mu F_z\left(1-\varepsilon_r V\sqrt{\lambda^2+\tan^2\alpha}\right)(1-\lambda)}{2\sqrt{C_i^2\lambda^2+C_\alpha^2\tan^2\alpha}} \tag{9}$$

2.3Modified Dugoff's friction model

The modified Dugoff's tire model [3] exhibits more simple and realistic approach. The tire force is given as

$$F_x = f(s)C_i\lambda \tag{10}$$

Where $f(s)$ is a piecewise function

$$f(s) = \begin{cases} (2-s)s, & s < 1 \\ 1, & s \quad 1 \end{cases} \tag{11}$$

and $$s = \frac{\mu F_z}{2|C_i\lambda|} \tag{12}$$

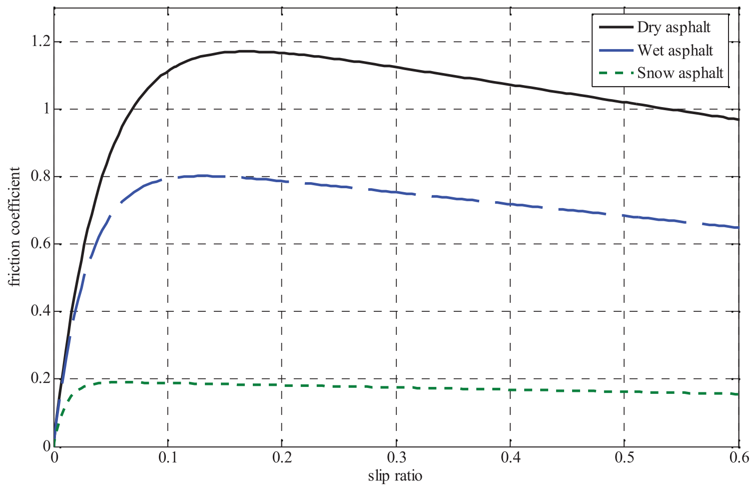

Fig. 2. Typical friction characteristics of the road-tire contact surface.

3. Wheel-slip controller design for ABS

As the dynamics of the system is highly non-linear and the vehicle parameters such as vehicle mass, position of centre of gravity are uncertain, the robust non-linear sliding mode controller has been designed for slip control. Here the main objective is to track the wheel-slip ratio to the desired sip ratio so that the vehicle stopping without skidding is ensured. The slip is defined as

$$\lambda = \frac{V - r\omega}{V} \tag{13}$$

The maximum brake force can be obtained at the peak of the $-\lambda$ curve. As per Dugoff's tire model, the location of the peak lies within the value 0.1 to 0.2. The desired dynamics [2] of the slip ratio is given as

$$\lambda_d(t) = \lambda_{opt} - \lambda_{opt}e^{-\alpha t} \tag{14}$$

Where the optimum slip ratio $\lambda_{opt} = 0.15$ and α=20 is the time constant. The response is shown in Fig. 4.

The sliding surface [1] is taken as

$$s = k\lambda_e + \dot{\lambda}_e = 0 \tag{15}$$

Where $\lambda_e = \lambda - \lambda_d$ and k is a positive constant.

Applying first order sliding mode control law the input braking torque is defined as

$$T_{brake} = \frac{1}{\hat{h}}[-\varphi sign(k\lambda_e + \dot{\lambda}_e) + \dot{\lambda}_d - \hat{f}] \tag{16}$$

Where $\hat{f} = -\frac{1}{\hat{V}}\left[\frac{\hat{F}_x}{M}(1-\lambda) + \frac{r^2\hat{F}_x}{J}\right]$

and $\hat{h} = \frac{r}{\hat{V}J}$

The switching gain φ is chosen such that the slip tracking error approaches to zero satisfying the condition $s(\lambda, t)\dot{s}(\lambda, t) < 0$.

The above control law (16) is based on following two assumptions

Assumption 1: the wheel speed ω and the vehicle speed V are measurable.

Assumption 2: the tire-road friction coefficient μ is known.

Assumption 1 is reasonable as the wheel speed can easily be obtained from the easily available sensors and the vehicle velocity can be measured by inertial sensor or by estimation. In assumption 2 the tire-road friction μ can be estimated. However vehicle velocity is estimated by sliding mode observer and the friction coefficient estimation scheme for modified Dugoff's friction model is suggested in the next section.

4. Sliding mode observer design

For full order state feedback ABS controller it is necessary to measure all the states which are necessary for the controller. From the previous section it is to be noted that vehicle velocity, wheel velocity and the friction coefficient are to be measured. However the friction coefficient and the vehicle velocity are estimated here due to the difficulties of direct measurement. Sliding mode observer for the vehicle velocity as in Fig. 3 is suggested here as it exhibits good robustness to the model uncertainties and finite convergence. The modified Dugoff's friction model is considered for ease of computation.

The non-linear zone i.e. $f(S) = (2 - S)S$ is considered here.

The tire force can be represent as

$$F_x = \left(2 - \frac{\mu F_z}{2|C_i\lambda|}\right)\frac{\mu F_z}{2|C_i\lambda|}C_i\lambda \tag{17}$$

It can be represented as second order algebraic equation of the friction coefficient

$$\mu^2(F_z{}^2C_i\lambda) + \mu(-4C_i\lambda\, F_z|C_i\lambda|^2) + F_x4|C_i\lambda|^2 = 0 \tag{18}$$

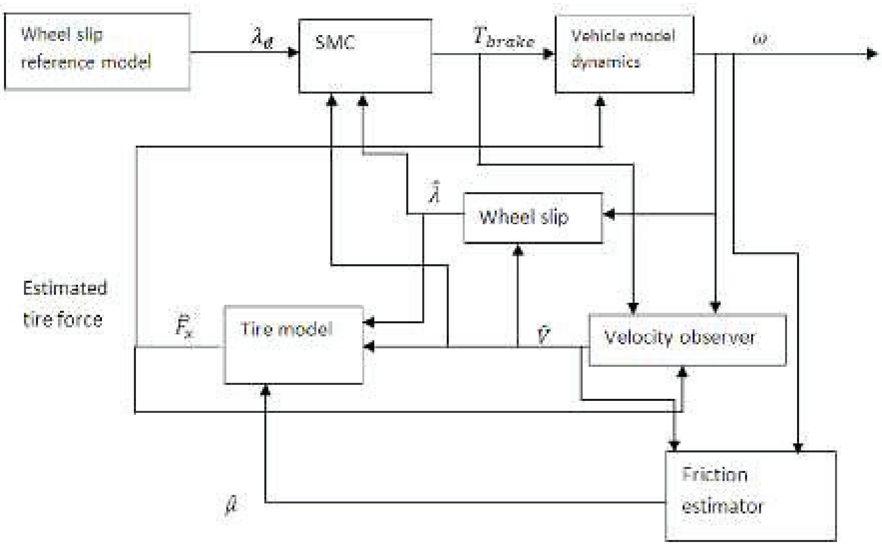

Fig. 3. The friction estimation scheme.

The solutions of the friction coefficient are

$$\mu = \frac{4C_i^2\lambda^2 F_z \pm \sqrt{(-4C_i^2\lambda^2 F_z)^2 - 4(F_z^2 C_i\lambda)(F_x 4|C_i\lambda|^2)}}{2F_z^2 C_i\lambda} \tag{19}$$

The estimated friction coefficient ($\hat{\mu}$) can be represented as follow

$$\hat{\mu} = \frac{4C_i^2\hat{\lambda}^2 F_z \pm \sqrt{(-4C_i^2\hat{\lambda}^2 F_z)^2 - 4(F_z^2 C_i\hat{\lambda})(\hat{F}_x 4|C_i\hat{\lambda}|^2)}}{2F_z^2 C_i\hat{\lambda}} \tag{20}$$

Where $\hat{\lambda} = \frac{\hat{V} - r\hat{\omega}}{\hat{V}}$ and $\hat{F}_x = -\dot{\hat{V}} M$

From the above equations it is seen that the friction coefficient can be estimated with the help of estimated vehicle longitudinal velocity profile. The observed vehicle velocity and wheel velocity are

$$\dot{\hat{\omega}} = \frac{1}{J} r F_x(\hat{\lambda}) - \frac{1}{J} T_{brake} + L_1 sign(\omega - \hat{\omega}) \tag{21}$$

$$\dot{\hat{V}} = \frac{-F_x(\hat{\lambda})}{M} + L_2 sign(\omega - \hat{\omega}) \tag{22}$$

Where the observer gains are L_1 and L_2. The sliding variable $\tilde{\omega} = (\omega - \hat{\omega})$

The estimation error is $\tilde{V} = V - \hat{V}$. The error dynamics of the sliding mode observer of ABS can be written as

$$\dot{\tilde{\omega}} = \frac{1}{J} r F_z [\mu(\lambda) - \mu(\hat{\lambda})] - L_1 sign(\tilde{\omega}) \tag{23}$$

$$\dot{\tilde{V}} = \frac{-F_z\mu(\hat{\lambda})}{M} + \frac{F_z\mu(\lambda)}{M} - L_2 sign(\tilde{\omega}) \tag{24}$$

Now the tire-road friction coefficient can be estimated as

$$\hat{\mu} = \frac{4C_l^2\hat{\lambda}^2 F_z \pm \sqrt{(-4C_l^2\hat{\lambda}^2 F_z)^2 - 4(F_z^2 C_l\hat{\lambda})(-[\frac{-F_3}{M} + L_2 sign(\omega-\hat{\omega})]M4|C_l\hat{\lambda}|^2)}}{2F_z^2 C_l\hat{\lambda}} \tag{25}$$

5. Simulation results

The simulations have been performed under the straight braking condition on a flat road with initial velocity taken as 20m/s. The parameters are taken as given in the table I. As this method is for online estimation of the road-tire friction, the test conditions of the road have been given with three different asphalts. Initially the estimation policy of the vehicle velocity has been performed with a road surface of 0.6 friction coefficient. The performance of the sliding mode controller (16) has been shown in Fig. 5. The switching gain φ is chosen as 740 and the 'sign' function is approximated as 'sat' function. The controller tracks desired slip dynamics to keep the optimum value of slip ensuring the braking without skid. Fig. 5 shows the slip error converges to zero within 0.4 second. Estimated longitudinal velocity of the vehicle is shown in Fig. 6. The friction coefficient is estimated for different road condition as shown in Fig. 7. The plot of estimation error of the friction has been shown in Fig. 8 which shows good robustness. The braking torque profile and the vehicle longitudinal velocity profile for the variation of road condition are shown in Fig. 9 and Fig. 10 respectively.

Table 1. Simulation parameters

Parameters	Nominal value
Wheel radius, r	0.326 m
Wheel base, l	2.5 m
Center of gravity height, h_{cg}	0.5 m
Wheel mass, m_w	40 kg
¼ of vehicle sprung mass, $\frac{1}{4}m_{vs}$	415 kg
Total moment of inertia of the wheel, J	1.7 kg m^2
Tire longitudinal stiffness, C_l	17349.8 N
Tire cornering stiffness, C_α	2720.55422(N/rad)
Road adhesion reduction factor, ε_r	0.01099

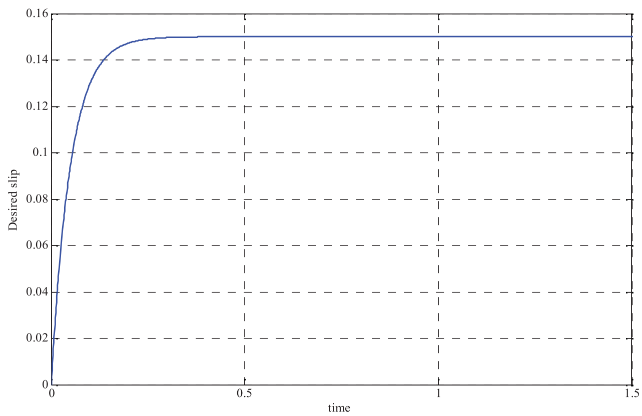

Fig. 4. Desired slip dynamic.

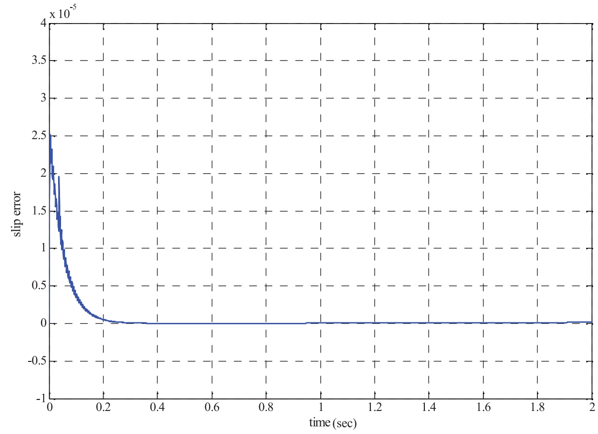

Fig. 5. Slip error.

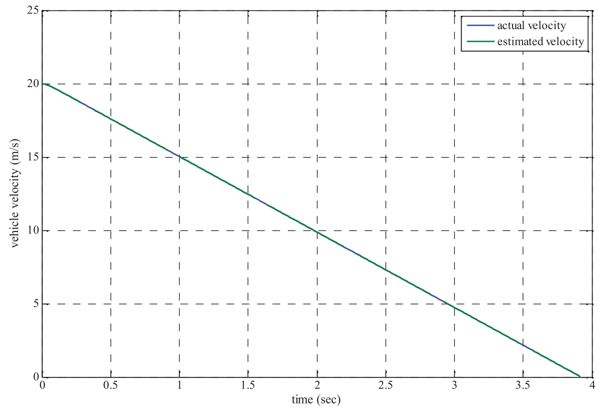

Fig. 6. Estimation of the vehicle longitudinal velocity.

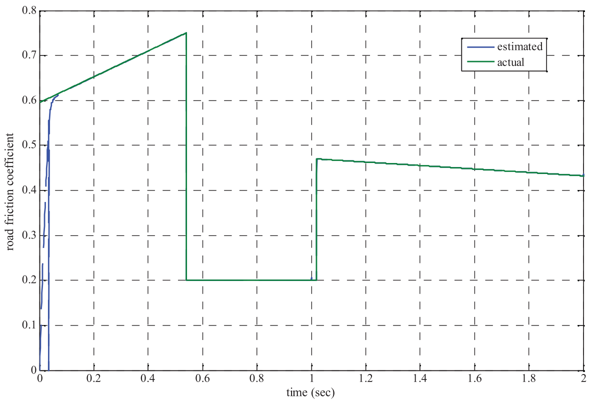

Fig. 7. Estimation of the friction coefficient in different asphalts.

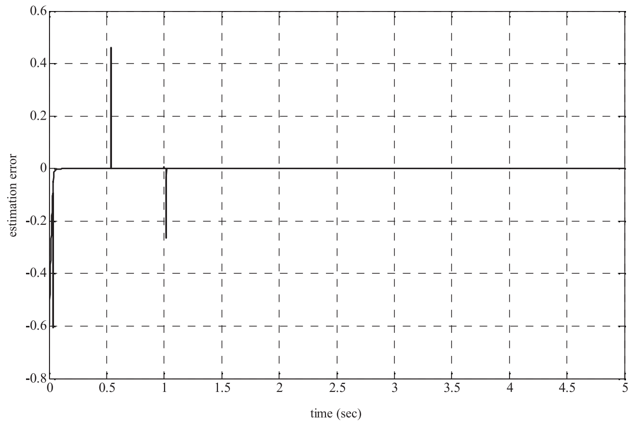

Fig. 8. Observation error.

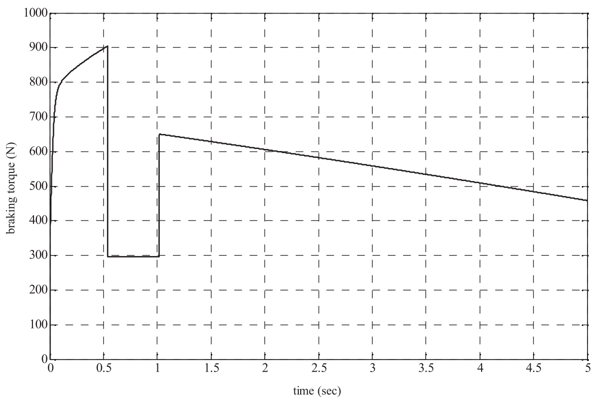

Fig. 9. Braking torque profile.

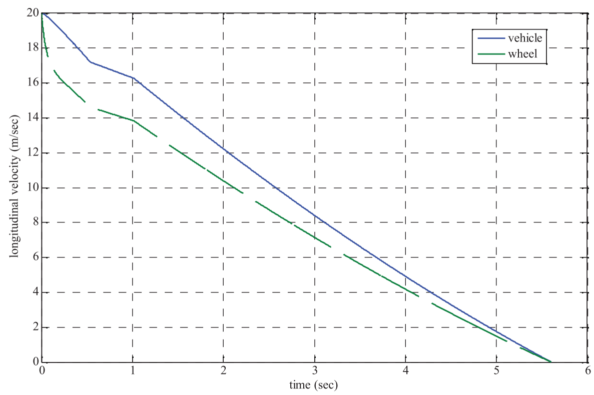

Fig.10. Longitudinal velocity of the vehicle and wheel in different asphalt.

6. Conclusion

The proposed road-tire friction estimation technique with sliding mode controller for the non-linear Dugoff's tire model is implemented in ABS. The sliding mode observer has shown good robustness against the variation of the road condition. The tracking performance of the controller is good with change in friction coefficient from ice to dry asphalt i.e. for $0.2 \quad \mu \quad 0.8$.

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