Anti-Lock and Anti-Slip Braking System, Using Fuzzy Logic and Sliding Mode Controllers

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***Abstract*—Anti-lock and also Anti-slip braking system controller designed for stability enhancement of vehicle during braking and turning circumstances is presented. Using available signals, a novel structure is proposed for vehicle stability improvement encountering critical braking conditions such as braking on slippery or μ-split road surfaces. In conventional vehicles, undesired lane changes may occur due to equal dispatch of braking torques to all wheels simultaneously. Also, intensive pressure on brake pedal can bring about wheels lockup which results in vehicle instability and undesired lane changes. The Anti-Lock Braking System (ALBS) and also Anti-Slip Braking System (ASBS) can serve as a driver assist system in vehicle path correction facing critical driving conditions during braking and turning round on different road surfaces. For these purposes, at first, a reference yaw angle is used for predicting of stable vehicle path. Then, slip of each wheel will be controlled by an Anti-Slip fuzzy controller which has been designed for each wheel. Also, a sliding mode controller has been designed so as to control the yaw angle concerning the yaw error where the yaw error is resulted from the difference between the reference value and actual yaw angle. Then, the difference between the left and the right wheels braking torques is used by the sliding mode controller in order to decrease the yaw error. Considering a model of Three-Degree-of-Freedom for chassis modeling and One-Degree-of-Freedom Dugoff's tire model for each wheel, a series of Matlab/Simulink simulation results will be presented in order to validate the effectiveness of the proposed controller.**

Keywords— Fuzzy; Slip of wheel; Sliding mode; SUGINO form.

NOMENCLATORE

a_x	Longitudinal acceleration of vehicle
a_y	Lateral acceleration of vehicle
CG	Corresponding to vehicle centre of gravity
C_x	Longitudinal stiffness of tire (N)
C_y	Lateral stiffness of tire (N/rad)
F_x	Longitudinal force of wheel (N)
F_y	Lateral force of wheel (N)
F_R	Rolling resistance force of wheel (N)
F_a	Aerodynamic drag force (N)
F_z	Vertical force of wheel (N)
h_{cg}	Height of CG
I_z	Vehicle moment of inertia around z axis (kg.m^2)
I_w	Wheel's moment of inertia (kg.m.^{2})
L_f	Distance of CG from front axle
L_r	Distance of CG from rear axle
M_t	Total mass of vehicle (kg)
R_w	Radius of wheel (m)
r	Yaw rate of vehicle
T_a	Length of vehicle axles
T	Applied torque to wheel
u	Longitudinal velocity of CG
v	Lateral velocity of CG
V_u	Circumference speed of wheel
X,Y	Denotation to static reference frame (m)
x,y	Denotation to moving reference frame (m)
X_{bp}	Braking pedal displacement
δ	Steering angle (rad)
α	Slip angle of wheel
γ	Yaw angle of vehicle
δ	Steering angle
λ	Longitudinal slip of wheel
μ_{peak-i}	Friction coefficient for ith wheel
ω	Angular speed of wheel
τ	Braking torque demand for each wheel
τ_R	Rolling resistance torque of wheel
τ_{Left}	Applied torque to left wheels
τ_{Right}	Applied torque to right wheels

I. INTRODUCTION

Anti-Lock-Anti-Slip Braking system (ALASB) comprising electrical sensors and intelligent controllers, is a new method for safety braking which considered to be a driver assist system. In ALASB system, braking force is initiated by driver while applied separately to wheels according to wheels' undesired yaw angles and slips for the enhancement vehicle stability. Such aim has recently invoked a lot of interest from both industry and academic sectors, worldwide. Hence, a great deal of new research woks has been performed for better control and higher safety achievement in vehicles. As some samples, the steering control is relatively an old method which has been proposed in [1]-[2]. In [1], a control-oriented model has been proposed that decouples the tire forces which facilitate control algorithm development. A yaw control system via steering, utilizing a fuzzy controller has been introduced in [2] for a 4WD vehicle. The idea of differential braking was first proposed as a steering intervention in [3] which has been used in [4] for yaw stability control. The Brake-By-Wire (BBW) system for undesired lane changes control and yaw angle control has been proposed in [5]. In the latter, a fuzzy controller has been used to obtain the difference between the left and the right wheels braking torques where, in the case of lateral deviation, a large yaw angle, or yaw instability, a yaw moment is generated by differential braking to help the vehicle turn back to the lane and restore its stability. A noticeable weakness of [5], is the likelihood wheels saturation due to extra applied braking torque, meaning that Anti-Lock braking has not been considered in the mentioned paper. In [6], sliding mode controller has been exploited for yaw rate control via electrical traction system installed on the rear wheels of an electrical vehicle and, also has been used in [7] for a 4WD hybrid vehicle. While, in [8], instead of sliding mode controller, a fuzzy controller has been utilized for this purpose in a 4WD hybrid vehicle. In all [6]-[8], which have been proposed by the authors of this paper, the wheel slip has not been controlled nor has differential braking

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been investigated. Differential Braking outperforms steering method and can be easily realized through providing optimal braking torques on different wheels for Anti-Slip and also Anti-Lock operation.

II. ALASB System Architecture and Aims

The ALASB architecture is illustrated in Fig.1 which consists of four Anti-Slip fuzzy controllers in SUGINO form (ALBS). Each module is relevant to each wheel and Braking force applied to each one will be computed based on the input braking force, wheel slip and its changes. When driver breaks on slippery road, during turning circumstances or on µ-split road surface, it is needed the vehicle behaves similarly as it operating on a normal road surface, thus in this method, braking force will be decreased by this controller to avoid wheel lockup, and moreover braking force input from the left and the right controllers will be computed by another sliding mode controller (ASBS) in order to avoid vehicle slipping and undesired lane change during braking. To meet this purpose a reference yaw angle could be used from stable turning theory. This reference value has been used as an input for sliding mode controller.

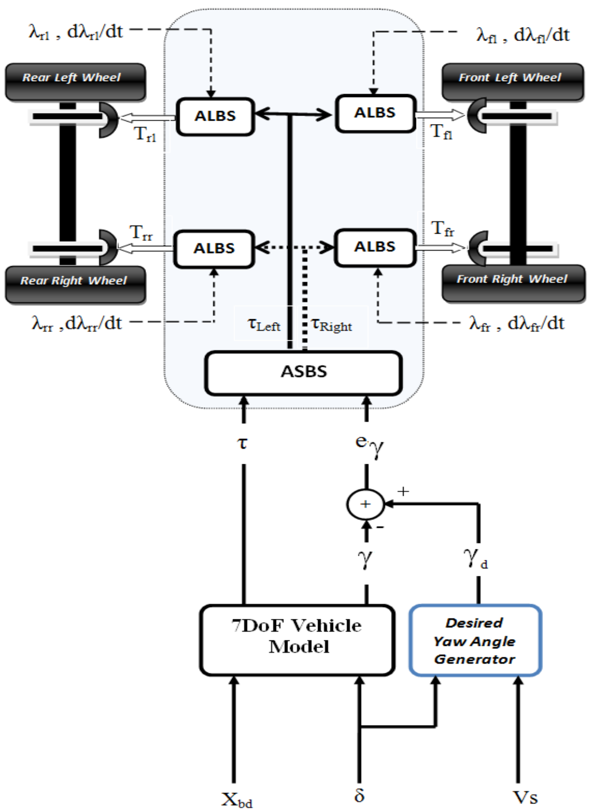

Fig.1 Proposed structure for ALASB system

III. Vehicle and Tire Modeling

A model of seven degree-of-freedoms is used for simulation, where Three degrees are pertinent to the chassis dynamics and four degrees are relevant to wheels angular speed. And also, fl, fr, rl and rr are denoting front left, front right, rear left and rear right respectively.

III.1 Vehicle Body Modeling

In this model, regarding the delineated parameters in the nomenclature, the system dynamics can be described as follows:

$$M_t(\dot{u}-rv)=F_{xfl}\cos\delta-F_{yfl}\sin\delta+F_{xfr}\cos\delta+F_{yfr}\sin\delta+F_{xrl}+F_{xrr}-F_{ax} \tag{1}$$

$$M_t(\dot{v}+ru)=F_{xfl}\sin\delta-F_{yfl}\cos\delta+F_{xfr}\sin\delta+F_{yfr}\cos\delta+F_{yrl}+F_{yrr}-F_{ay} \tag{2}$$

$$I_z\dot{r}=L_f[F_{xfl}\sin\delta+F_{yfl}\cos\delta+F_{xfr}\sin\delta+F_{yfr}\cos\delta]-L_r[F_{yrl}+F_{yrr}]+ T_a/2[F_{xfl}\cos\delta-F_{yfl}\sin\delta+F_{xfr}\cos\delta+F_{yfr}\cos\delta+F_{xrl}-F_{xrr}] \tag{3}$$

III. 2. Tire Model

Tire Modeling is one of the most important and rather sophisticated parts of a vehicle modeling. Applying revolving torque (T_i) on the wheel, the rotation can be expressed as follows:

$$I_{wi}\dot{\omega_i} = T_i - R_w F_{xi} - \tau_{Ri} \quad for \quad i: fl, fr, rl, rr \tag{4}$$

Where, I_w and R_w are wheel's moment of inertia and wheel's radius respectively, ω is wheel's angular speed, and τ_R is wheel's rolling resistance torque which is an important factor in fuel consumption computing.

$$\tau_R = (C_0 + C_1|V_w|^2)R_w.F_z \tag{5}$$

Where, V_w is wheel's linear speed and usually $0.04 \le C_0 \le 0.2$, $C1 << C_0$. The Well known Dugoff's model has been utilized for longitudinal and lateral forces modeling in this article [5]-[8]. F_z is the vertical force on the tire, considering the effects of vehicle longitudinal and lateral accelerations which can be obtained via the definite formulas given in [6].

$$\mu_i = \mu_{peak,i}\sqrt{1 - A_s.R_w(\lambda_i + tan(\alpha_i)} \tag{6}$$

$$H_i = \sqrt{\left[(\frac{C_x\lambda_i}{\mu_i F_{zi}(1-\lambda_i)})^2 + (\frac{C_y Tan(\alpha_i)}{\mu_i F_{zi}(1-\lambda_i)})^2\right]} \tag{7}$$

$$F_{xi} = \begin{cases} \dfrac{C_x\lambda_i}{1-\lambda_i} & for \quad H_i < 0.5 \\ \dfrac{C_x\lambda_i}{1-\lambda_i}\left(\dfrac{1}{H_i} - \dfrac{1}{4H_i^2}\right) & for \quad H_i \ge 0.5 \end{cases} \tag{8}$$

$$F_{yi} = \begin{cases} \dfrac{C_y Tan(\alpha_i)}{1-\lambda_i} & for \quad H_i < 0.5 \\ \dfrac{C_y Tan(\alpha_i)}{1-\lambda_i}\left(\dfrac{1}{H_i} - \dfrac{1}{4H_i^2}\right) & for \quad H_i \ge 0.5 \end{cases} \tag{9}$$

Figure 2 shows sample curves for various values of λ and α which can be obtained from (6)-(9). According to these curves, when the longitudinal slip λ and slip angle α are small, the coupling between F_x and F_y is so little, which can be ignored. Whereas, when the longitudinal slip is large, F_y decreases for all values of α, indicating un-steer ability. This case might occur due to high λ which could happen during takeoff and hard braking situations.

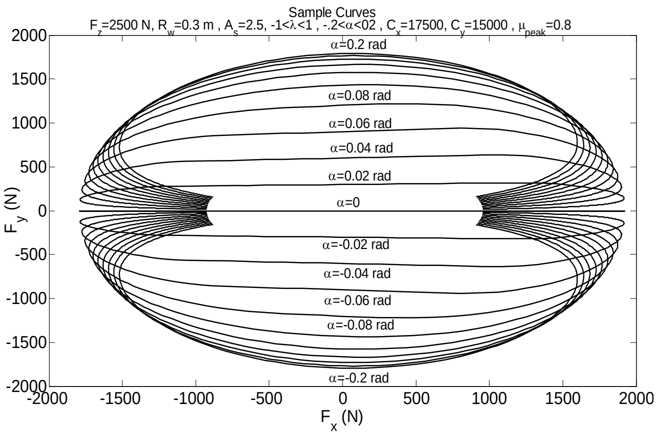

Fig.2 Sample curves of tire

Equations expressing the vertical forces on the tires are given as follows:

$$F_{zfl} = \frac{M_t}{(L_f+L_r)}[g.\frac{L_r}{2} + a_x.\frac{h_{cg}}{2} + a_y.L_r.\frac{h_{cg}}{T_a}] \quad (10)$$

$$F_{zfr} = \frac{M_t}{(L_f+L_r)}[g.\frac{L_r}{2} - a_x.\frac{h_{cg}}{2} - a_y.L_r.\frac{h_{cg}}{T_a}] \quad (11)$$

$$F_{zrl} = \frac{M_t}{(L_f+L_r)}[g.\frac{L_f}{2} + a_x.\frac{h_{cg}}{2} + a_y.L_f.\frac{h_{cg}}{T_a}] \quad (12)$$

$$F_{zrr} = \frac{M_t}{(L_f+L_r)}[g.\frac{L_f}{2} - a_x.\frac{h_{cg}}{2} - a_y.L_f.\frac{h_{cg}}{T_a}] \quad (13)$$

Where, $a_x = u^{\cdot} - rv$, and $a_y = v^{\cdot} + ru$ denote the longitudinal and lateral acceleration of CG, respectively. The longitudinal slip of a wheel can be obtained from:

$$\lambda_i = \frac{R_w.\omega_i - |V_{wi}|\,cos(\alpha_i)}{|V_{wi}|\,cos(\alpha_i)} \quad (14)$$

To complete the vehicle modeling, the linear speeds and lateral slips of wheels are stated as:

$$V_{wfl} = \left(u - \frac{T_a}{2}r\right)i + \left(v + L_f.r\right)j \quad (15)$$

$$V_{wfr} = \left(u + \frac{T_a}{2}r\right)i + \left(v + L_f.r\right)j \quad (16)$$

$$V_{wrl} = \left(u - \frac{T_a}{2}r\right)i + (v - L_r.r)\,j \quad (17)$$

$$V_{wrr} = \left(u + \frac{T_a}{2}r\right)i + (v - L_r.r)\,j \quad (18)$$

$$\alpha_{fl} = \delta - tg^{-1}\left[\frac{v + L_f.r}{u - \frac{T_a}{2}.r}\right] \quad (19)$$

$$\alpha_{fr} = \delta - tg^{-1}\left[\frac{v + L_f.r}{u + \frac{T_a}{2}.r}\right] \quad (20)$$

$$\alpha_{rl} = -tg^{-1}\left[\frac{v - L_r.r}{u - \frac{T_a}{2}.r}\right] \quad (21)$$

$$\alpha_{rr} = -tg^{-1}\left[\frac{v - L_r.r}{u + \frac{T_a}{2}.r}\right] \quad (22)$$

Using the above equations, a conventional vehicle can be modeled. Henceforth, the vehicle longitudinal dynamics and also lateral dynamics can be achieved. Regarding the applied torque τ on the wheels, the motion of vehicle can be expressed as follows.

$$V_s = \sqrt{(u^2 + v^2)} \quad \text{(m/s)} \quad (23)$$

$$X = \int V_s.\cos(\gamma)\,dt \quad \text{(m)} \quad (24)$$

$$Y = \int V_s.\sin(\gamma)\,dt \quad \text{(m)} \quad (25)$$

III.3. Brake and Accelerator Pedals Modeling

A simple PID controller has been designed simulating driver behaviors in utilizing Accelerator/Brake pedals.

$$T_i = \left(K_p + \frac{1}{K_i.s} + K_d.s\right)\left(V_{ref} - V_s\right) for\ i{:}\ fl, fr, rl, rr \quad (26)$$

$T_i < 0 \rightarrow.\ T_i = T_{fl} = T_{fr} = T_{rl} = T_{rr}$

$T_i > 0 \rightarrow T_i = T_{fl} = T_{fr},\ T_{rl} = T_{rr} = 0$

IV. Desired Yaw Angle

From the steady state cornering theory of bicycle model, one may anticipate that the vehicle velocity and yaw rate error satisfy the following equations [5-8].

$$r_d = \frac{1}{1+AV^2}.\frac{V}{L}.\delta\ ,\ A = \frac{m}{2L^2}\frac{L_r C_{y,r} - L_f C_{y,f}}{C_{y,f} C_{y,r}} \quad (27)$$

$$\gamma = \int_{t_{start}}^{t_{end}} r\,.dt \quad , \quad \gamma_d = \int_{t_{start}}^{t_{end}} r_d\,.dt \quad (28)$$

In (27), $C_{y,r}$ and $C_{y,f}$ are lateral stiffness of the rear and front tires respectively. In (28), t_{start} and t_{end}, are times of start and end of braking respectively.

V. Controler Design

Regarding the proposed architecture discussed in section 2, there are two main aims in this paper as follows:

A. Avoiding wheels lockup during braking due to operating on slippery road surface or hard braking torque. For this purpose, a fuzzy controller in SUGINO form has been designed for each wheel.

B. Keeping vehicle on its desired lane during braking, and for this purpose a sliding mode controller has been designed to achieve separate braking torques for each of the left and right wheels.

V. 1. Sliding Mode Controller for ASBS design.

In order to generate differential braking torque to meet $e_\gamma = \gamma_d - \gamma = 0$, the sliding mode control is used, where sliding surface is defined as:

$$s = m.e_\gamma(t + \Delta t) + \frac{de_\gamma(t+\Delta t)}{dt}.\Delta t \quad (29)$$

Where, m is an arbitrary positive constant and $e_\gamma(t+\Delta t)$ is the estimated yaw error which can be obtained via the following simple linear estimator.

$$e_\gamma(t+\Delta t) = e_\gamma(t) + \frac{de_\gamma(t)}{dt}\Delta t \tag{30}$$

So, using the sliding surface (29), equation (31) can be expressed as:

$$\Delta\tau_p(t+\Delta t) = \tau(t).\frac{s}{|s|} \tag{31}$$

Where, $\Delta\tau_p$ is the difference between the left and the right braking torques to generate the direct yaw moment, and $\tau(t)$ is braking torque demand which is requested from driver. Since $\Delta\tau_p(t+\Delta t)$ might Oscillate with high frequencies on the sign of the S, so the following low pas filter ought to be used.

$$\tau_c.\Delta\dot{\tau_p}(t+\Delta t) + \Delta(t+\Delta t) = \Delta\tau(t+\Delta t) \tag{32}$$

Where, τ_c is the time constant of the first order low pas filter and $\Delta\tau$ is the filtered torque which is the difference between the braking torques. Now, the left and the right braking torques can be obtained as follows.

$$\begin{aligned}\tau_{Right}(t+\Delta t) &= \frac{\tau(t)+\Delta\tau(t+\Delta t)}{2}\\ \tau_{Left}(t+\Delta t) &= \frac{\tau(t)-\Delta\tau(t+\Delta t)}{2}\end{aligned} \tag{33}$$

V.2. FUZZY CONTROLLER FOR ALBS DESIGN

For the design of Anti-Lock braking controller a fuzzy controller with SUGINO, in constant form, has been devised. According to the tire behaviors, when the longitudinal slip is high, the longitudinal force and lateral force decrease which reduces the vehicle's steer ability. Therefore, the ALBS controller is designed utilizing the following memberships and rule base.

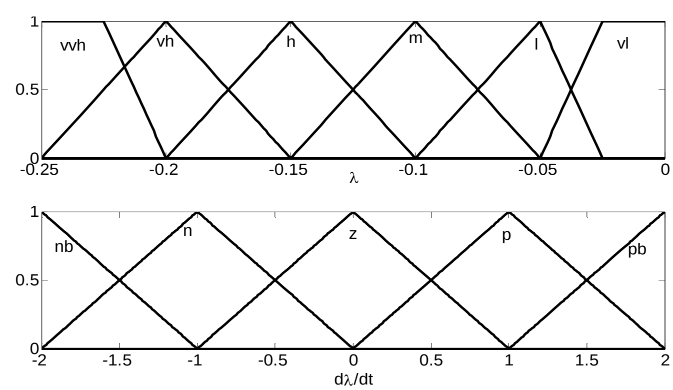

Fig.3 Fuzzy membership functions for input of controllers

Table1 Fuzzy rule base of ALBS controller

		λ					
		vvh	vh	h	m	l	vl
dλ/dt	nb	0	0.125	0.25	0.375	0.5	0.75
	n	0	025	0.375	0.5	0.625	0.875
	z	0	0.375	0.5	0.625	0.75	1
	p	0	0.5	0.625	0.75	0.875	1
	pb	0	0.625	0.75	0.875	1	1

Defining K_i as the output of ALBS controller, the braking torque applied to each wheel can be expressed as follows:

$$T_{fl} = K_{fl}.\tau_{Left}\ ,\ T_{fr} = K_{fr}.\tau_{Right}\ ,\ T_{rl} = K_{rl}.\tau_{Left}\ ,\ T_{rr} = K_{rr}.\tau_{Right} \tag{34}$$

V.3. NEEDED SENSORS FOR PRACTICABILITY

For experimental case, there are need to some sensors and microprocessor base computer unit, for reading of needed data of vehicle behavior and applying of input signals to overall controller. Regarding to figure 1 and also dynamical equations, there are need to some data include of longitudinal slip of wheels. Since in braking case and stable condition, linear speed of each wheel is greater than vehicle speed, equation (14) can be written as following to simplifying of measurement and decreasing of needed sensors.

$$\lambda_i = \frac{R_w.\omega_i - |V_{wi}|\cos(\alpha_i)}{|V_{wi}|\cos(\alpha_i)} \cong -1 + \frac{V_{ui}}{V_s} \tag{35}$$

Where, V_{ui} and V_s, are circumference speed of i^{th} wheel and vehicle speed respectively. So, the needed sensors for experimental case are tabulated in table 2. Also, the overall controller, has been show in figure 4 and 5.

Table 2 Needed sensors for experimental case

Signal	Symbol	Unit
Circumference speed of "fl" wheel	V_{ufl}	m/s
Circumference speed of "fr" wheel	V_{ufr}	m/s
Circumference speed of "rl" wheel	V_{url}	m/s
Circumference speed of "rr" wheel	V_{urr}	m/s
Speed Of Vehicle	V_s	m/s
Braking pedal displacement	X_{bd}	%
Yaw Angle of vehicle	γ	rad
Steering Angle of Vehicle	δ	rad

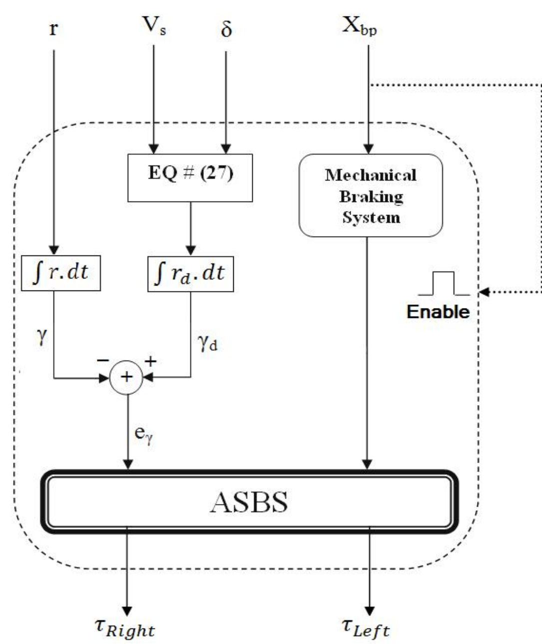

Fig.4 Structure of ASBS controller and relative inputs

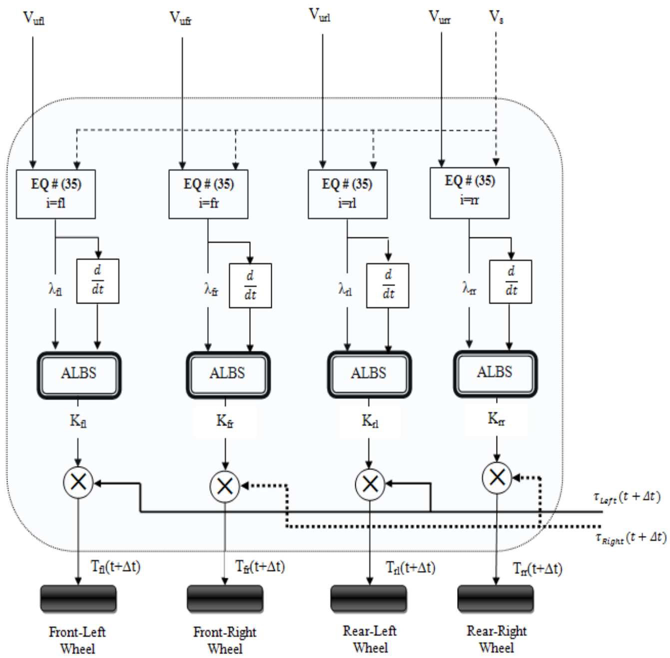

Fig.5 Structure of ALBS controller and relative inputs

VI. Simulation Results

To evaluate the effectiveness of the proposed controller (ALASB), the numerical values of the vehicle model [5] are listed in table 3. These parameters corresponded to a mid-size passenger car. In addition, m and τ_c which are used in the sliding mode controller and are set to 1 and 0.05 values respectively.

Table 3 Vehicle Properties

Parameter	Symbol	Unit	Value
Vehicle total mass	M_t	Kg	850
Distance from front axle to CG	L_f	m	1.147
Distance from rear axle to CG	L_r	m	1.197
Track width	T_a	m	1.4
Drag coefficient	C_d	$N.s^2/m^2$	0.41
Frontal area	A_F	m^2	1.8
Lateral area	A_L	m^2	4.5
Vehicle inertia about z axis	I_z	Kgm^2	7809
Wheel's longitudinal stiffness	C_x	N	17500
Wheel's lateral stiffness	C_y	N/rad	15000
Wheel's radius	R_w	m	0.275
Wheel's inertia	I_w	Kgm^2	3.625

VI.1. *Braking on μ-Split Road*

In this section a braking at 110 km/h on a μ-split road, comprising dry pavement, μ_{peak}=0.95, on the right side and unpacked snow, μ_{peak}=0.45, on the left side, has been simulated where the steer angle is assumed to be zero. The vehicle speed reduction and its lane are depicted in figure 6, which show that the ALASB system has a good stability during braking and undesired lane change are so little which can be neglected. Figures 7, 8 and 9 show applied braking torques and slip of wheels, respectively.

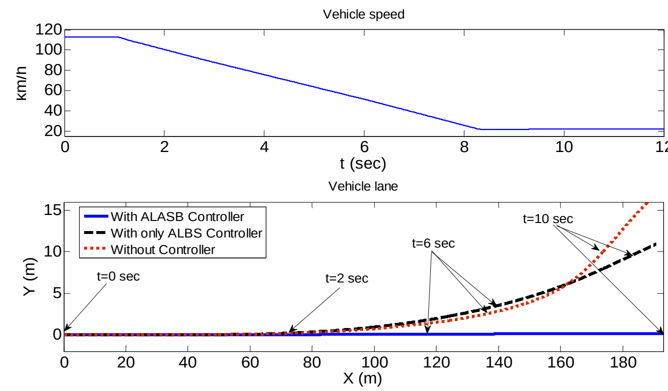

Fig.6 Vehicle speed and its lane during braking on μ-Split Road

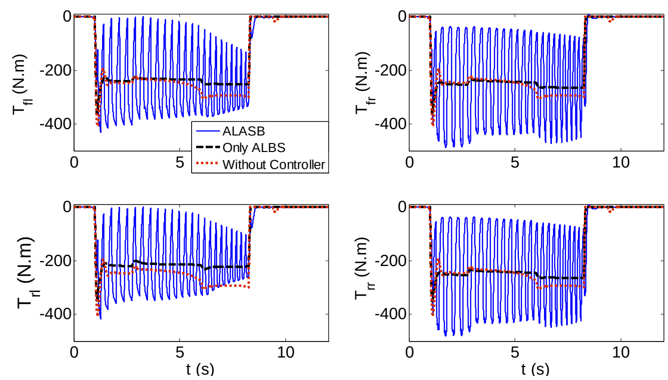

Fig. 7 Longitudinal slip of wheels during braking on μ-Split Road

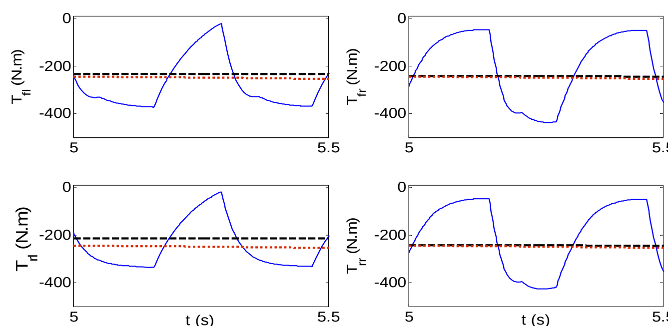

Fig. 8 Zoomed of figure 7.a for 5<t<5.5 s for better illustration of differential braking

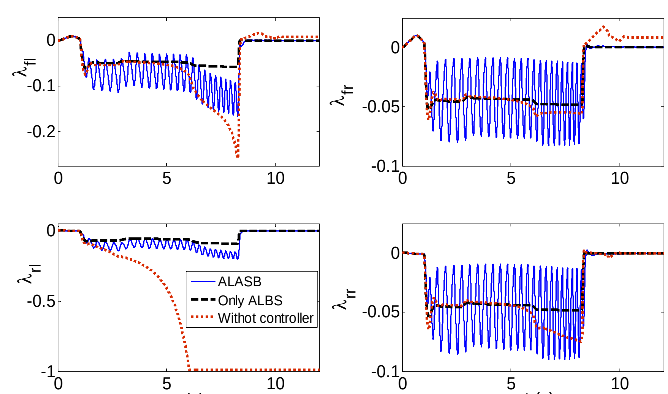

Fig.9 Longitudinal slip of wheels during braking on μ-Split Road

VI.2. BRAKING AND TURNING ROUND ON SLIPPERY ROAD SURFACE

In this section, synchronous braking and tuning has been simulated and Figure 10 shows the vehicle speed and steering angle during the simulation. The simulation performed concerning the following condition:

A. Driving on a road surface with μ_{peak}=0.95 for the right wheels and μ_{peak}=0.45 for the left wheels, without any controller.

B. Driving on a road surface with μ_{peak}=0.95 for the right wheels and μ_{peak}=0.45 for the left wheels, associated with merely an ALBS controller.

C. Driving on a road with μ_{peak}=0.95 for right wheels and μ_{peak}=0.45 for left wheels, associated with an ALASB controller.

Throughout conditions A-D, it is assumed that the vehicle speed and steering angle have been requested as in figure 10. Regarding figure 11, the vehicle has a good dynamical behavior on slippery road. In fact, the ALASB controller is a driver assist system for stability enhancement of vehicle during braking, indicating that the vehicle has the same behavior in operating on slippery road surface as does it have on normal ones. Figures 12 and 13 show the longitudinal slips and the applied braking torques.

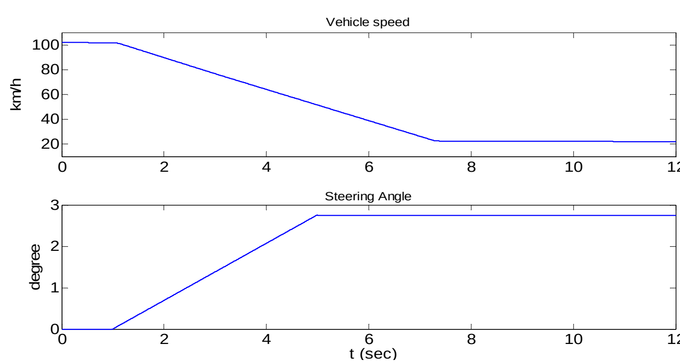

Fig. 10 Vehicle speed and steering angle during braking and turning round on µ-Split Road

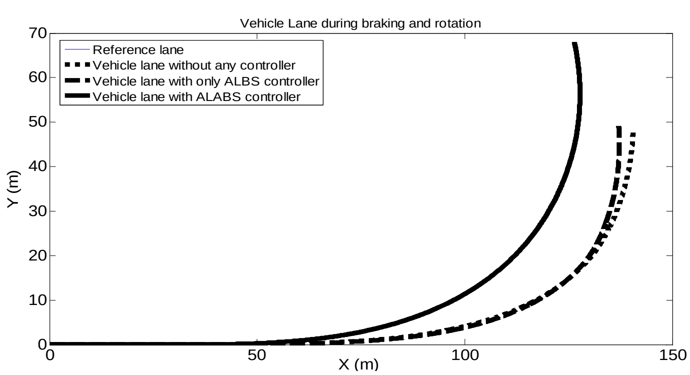

Fig. 11 Vehicle lane during braking and turning round on µ-Split Road and Comparison

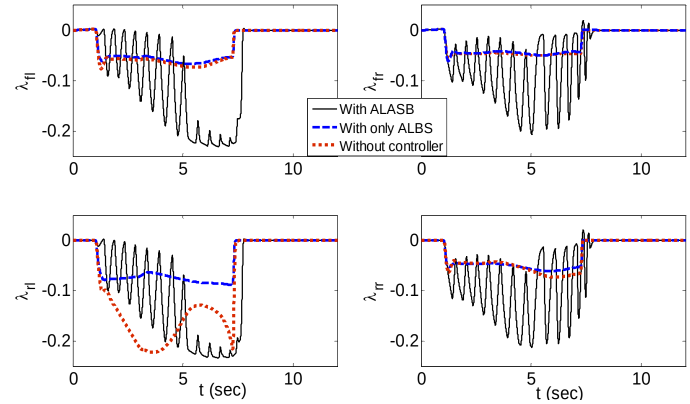

Fig.12 Longitudinal slips during braking and turning round on µ-Split Road and comparison

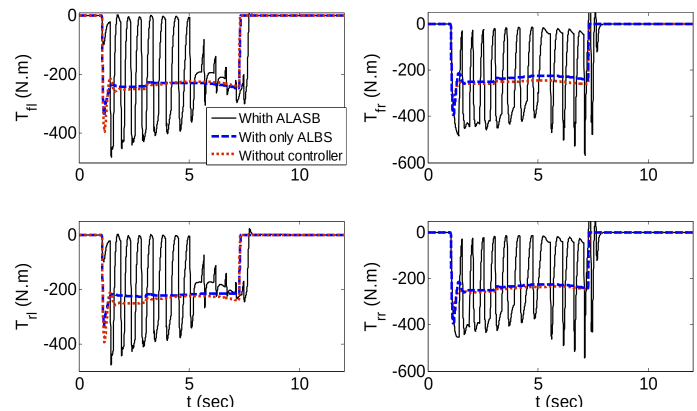

Fig. 13 Torques applied during braking and turning round on µ-Split Road and comparison

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