



# Driving/Regeneration and Stability Enhancement of a 4WD Hybrid Vehicles Using Multi-Stage Fuzzy Controller

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## ABSTRACT

In front wheels driven vehicles, fuel economy can be obtained by summing torques applied to rear wheels. On the other hand, unequal torques applied to rear wheels provides enhanced safety. In this paper, a model with seven degrees of freedom is considered for the vehicle body. Thereafter, power-train subsystems are modeled. Considering an electrical machine on each rear wheel, a fuzzy controller is designed for each driving, braking, and stability conditions. Another fuzzy controller recognizes the vehicle requirements between the driving/regeneration and stability modes. The simulations performed in MATLAB/Simulink environment show that the proposed structure can effectively enhance vehicle performance in different modes.

## 1. INTRODUCTION

One of the significant qualitative factors of the vehicle behavior is its stability in critical driving conditions such as braking on  $\mu$ -Split road and rotation with high speed. Recent research results of fuel economy in vehicles have led to invention of hybrid vehicles. In these vehicles, driver power demand is provided by gasoline engine and electrical machine. In most research works the goal of control strategy is only based on fuel economy. In [1], a model based on the real time road control strategy was offered for parallel hybrid vehicles. An optimal control strategy that chose power split between the engine and electrical machine was presented in [2] to minimize fuel consumption in parallel hybrid vehicles. In [3], fuel economy was improved using field oriented control of a permanent magnet motor and its belt coupling with crankshaft. A simulation program was given in [4] to simulate behavior of various components of hybrid vehicles. Some studies have focused on the vehicles' stability. A driver-assist stability system and stability enhancement for all-wheel-drive electric vehicles has

been introduced in [5-7]. This system was proposed in [8] for two-motor-drive electric vehicle to enhance safety using a fuzzy logic based controller. In [9], using an electrical machine on front and rear axles, stability enhancement and regenerative braking were provided. Direct yaw rate control with road condition estimation and anti-slip control have been proposed in [10]. In [11], for an electrical vehicle, a new estimation method of slip-rate has been presented.

## 2. PROPOSED STRUCTURE

In a front differential vehicle, unequal torques applied to rear wheels will bring vehicle dynamic control to path correction. On the other hand, summation of torques is an essential factor in power managing among engine and electrical machines. Figure 1 shows the proposed structure. Beside battery state of charge (SoC), other input signals are necessary which will be illustrated in the following parts.

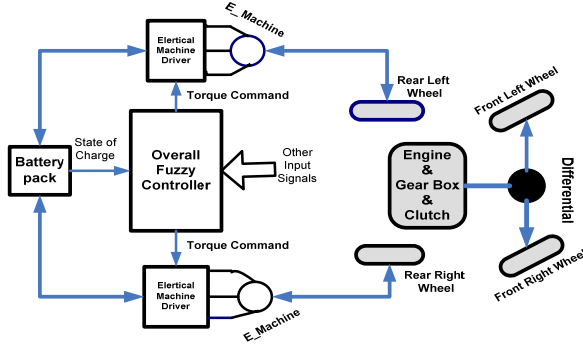


Figure 1: The proposed structure and controller

### 3. VEHICLE MODELLING

#### A. Body modeling and tire forces

A model with seven degrees of freedom was used for simulation. In this model, the system dynamic can be described as follows [6]:

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

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

$$I_z r = L_f [F_{xfl} \sin \delta + F_{yfl} \cos \delta + F_{xfr} \sin \delta + F_{yfr} \cos \delta] - L_r [F_{yrl} + F_{yrr}] + \frac{T}{2} [F_{xfl} \cos \delta - F_{yfl} \sin \delta - F_{xfr} \cos \delta + F_{yfr} \cos \delta + F_{xrl} - F_{xrr}] + M_{zfl} + M_{zfr} + M_{zrl} + M_{zrr} \quad (3)$$

Three degrees were devoted to the chassis' motion and four degrees were assigned to wheels' angular speed. fl, fr, rl and rr denoted front left, front right, rear left and rear right, respectively.

| Symbol   | Definition                           |
|----------|--------------------------------------|
| $F_x$    | Wheel's longitudinal force           |
| $F_y$    | Wheel's lateral force                |
| $F_R$    | Wheel's rolling resistance force     |
| $F_a$    | Aerodynamic drag force               |
| $X, Y$   | Denotation to static reference frame |
| $x, y$   | Denotation to moving reference frame |
| $\delta$ | Steer angle                          |
| CG       | Corresponding to center of gravity   |
| $L_f$    | Distance from CG to front axle       |
| $L_r$    | Distance from CG to rear axle        |
| $T$      | Long of vehicle axle                 |
| $u$      | Longitudinal velocity of CG          |
| $v$      | Lateral velocity of CG               |
| $r$      | Vehicle yaw rate                     |
| $M_z$    | Wheel's self-aligning torque         |
| $M_t$    | Vehicle total mass                   |

#### B. Tire modeling

Tire modeling is one of the most important and ambiguous parts of vehicle modeling. By applying mover torque ( $\tau_w$ ) to the wheel, the rotation can be described as follows:

$$I_w \dot{\omega}_i = \tau_{wi} - R_w F_{xi} - \tau_{Ri} \quad \text{for } i : fl, fr, rl, rr \quad (4)$$

where  $I_w$  and  $R_w$  are wheel's moment of inertia and wheel's radius, respectively,  $\omega$  is wheel's angular speed, and  $\tau_R$  is wheel's rolling resistance torque, which is an important factor in computing fuel consumption.

$$\tau_R = C_0 F_z + C_1 |V_w|^2 \quad (5)$$

$V_w$  is wheel's linear speed and usually  $0.04 \leq C_0 \leq 0.2$ ,  $C_1 < C_0$ . Well known Dugoff's model for longitudinal and lateral forces is used in this article [12].  $F_z$  is vertical force on the tire considering effects of vehicle longitudinal and lateral accelerations and can be obtained by the known formulas mentioned in [6].

#### C. Power transmission system modeling.

Transmission subsystem includes engine, gear box, clutch, brake, and differential. The output engine power is transferred to driven wheels via clutch, gear box, and differential. Braking torque is transferred to all wheels directly by the brake pedal command. On account of the equality between input and output power in gear box and differential systems, modeling of these subsystems can be performed by assuming a constant coefficient for each of them. Engine and gear box speed equivalency is assumed for simulation purposes. Since this regulation would be violated in some cases such as low speed motion or driving by improper gear, the engine power will be wasted in clutch subsystem. Figure 2 shows transmission modeling. Figure 3 shows clutch power transmission curve which is utilized in this work for simulation. The clutch is simulated by two surfaces. One of them is connected to engine shaft and another is jointed to gear box input.

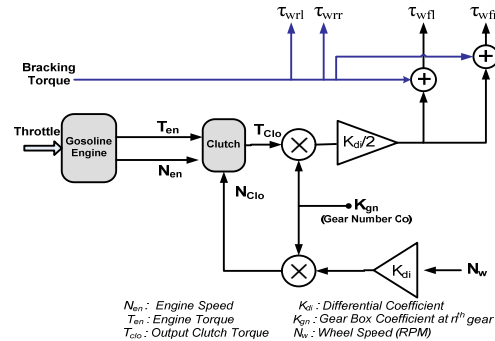


Figure 2: Mechanical power flow modeling

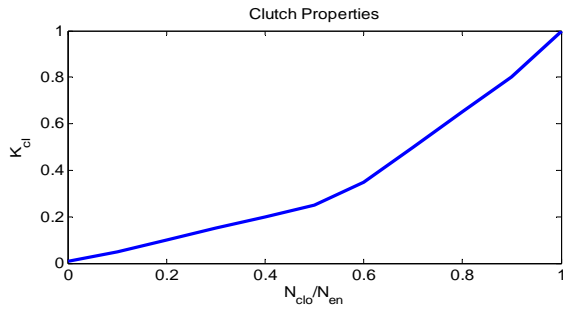


Figure 3: Clutch model curve

In Figures 2 and 3:

| SYMBOL                     | DEFINITION                       |
|----------------------------|----------------------------------|
| $K_{GN}$                   | The nth gear coefficient         |
| $K_{DI}$                   | Differential gear coefficient    |
| $N_{EN}$                   | Engine shaft speed               |
| $T_{EN}$                   | Engine output torque             |
| $N_{CLO}$                  | Gear box speed at the nth gear   |
| $T_{CLO}$                  | Output torque of gear (nth gear) |
| $K_{CL} = T_{CLO}/T_{CLI}$ | Clutch coefficient               |

Engine torque and fuel consumption will be computed regarding engine maps for modeling purposes. One of these maps computes shaft torque based on throttle opening and shaft speed. The engine fuel consumption is determined according to shaft speed and shaft torque. Due to previous discussion, the shaft speed will be related to the vehicle speed in usual conditions. Also, driver power demand will be implemented by throttle and brake pedals in positive and negative accelerations, respectively. Figures 4 and 5 show the engine maps employed for simulation taken from 'ADVISOR' simulation program [13]. Considering Figure 4, engine output torque is negative in some cases. This case happens due to engine power shortage compared with wheels' power in downhill driving condition, for example. This is a key note for power regeneration simulation when there is no pressure on brake pedal.

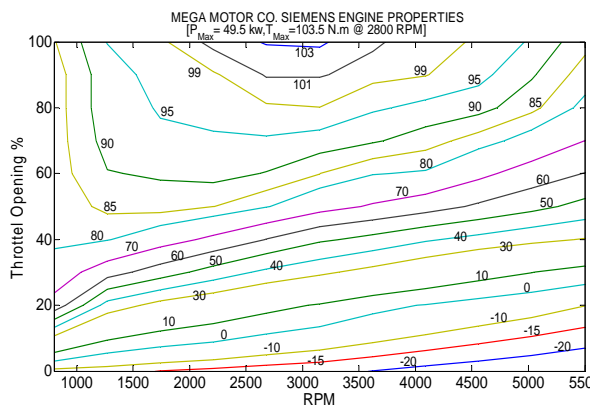


Figure 4: Engine's torque map

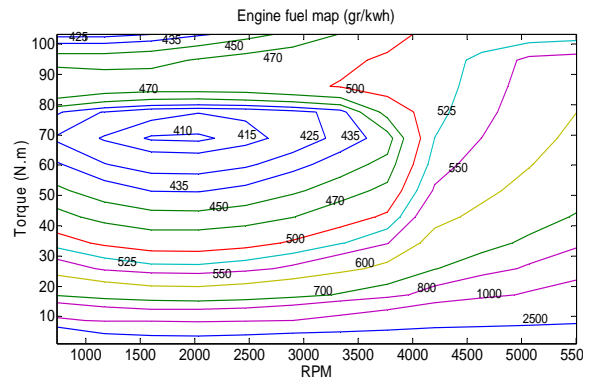


Figure 5: Engine's fuel consumption map

#### 4. MODELLING OF ELECTRICAL COMPONENTS

Electrical subsystems, used in this article, consist of AC/DC converter, electrical machine, batteries, and power electronic components. Because of fast dynamicity of these subsystems in comparison with vehicle dynamics, only the battery's dynamic model is taken into account. In this way, the 'ADVISOR' statistical model is used for Inverter/Electrical machine modeling [13]. The models demonstrated above are utilized in simulation part.

##### 4.1. Inverter/Electrical machine modeling

In electrical machine and connected inverter models, efficiency and maximum rotor torque are available. Figure 6 shows Inverter/Electrical machine maps for modeling.

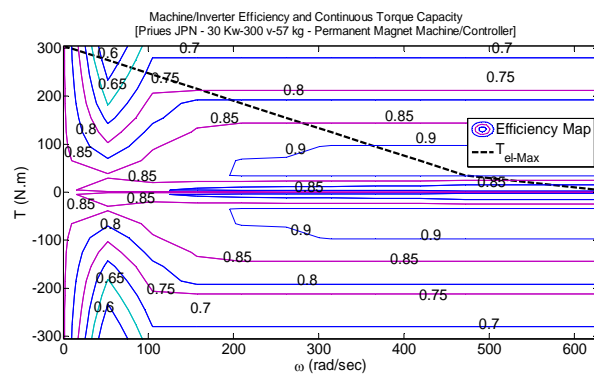


Figure 6: Electrical machine curves

##### 4.2. Battery modeling

Battery state of charge (SoC) is the most important control signal in the hybrid vehicle. In this paper, one of the well-known battery models is employed. This model is based on variable voltage source and internal variable resistance depending on SoC. Figures 7 and 8

show this model and typical values of its parameters [13]. One of the simple and well known formulas for SoC calculation is given below: where SoC(0) is initial state of charge, Ah<sub>cap</sub> and Ah<sub>used</sub> are maximum and used battery Amper×hour, respectively, and I<sub>b</sub> is instantaneous battery current.

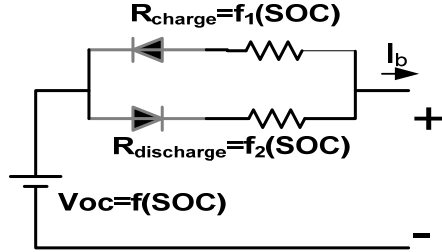


Figure 7: The 'R\_Internal' battery model

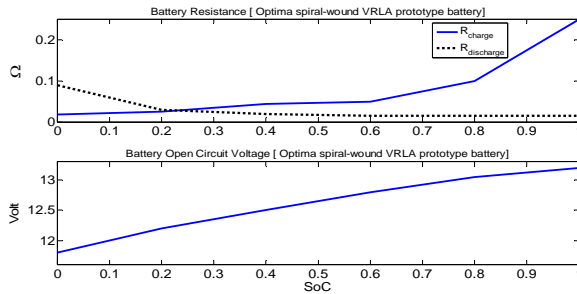


Figure 8: internal parameters of Battery

$$SoC = \frac{(Ah_{cap} - Ah_{used})}{Ah_{cap}} \quad (10)$$

$$Ah_{used} = Ah_{cap}(1 - SoC_{(0)}) + \int_0^t \frac{I_b}{3600} dt \quad (11)$$

### 5. DRIVER MODELING

A simple PID controller is used for driver behavior simulation in 'Throttle/Brake pedals' pressure. In addition, for gear changing simulation, it is assumed that the change established upon throttle opening and vehicle speed experimental data.

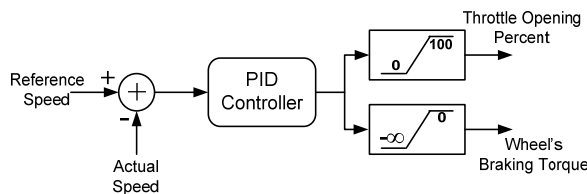


Figure 9: Simple PID controller for pedals' pressure simulation

### 6. CONTROLLER STRUCTURE AND STRATEGY

Regarding (3), the yaw rate can be directly controlled by applying a differential input torque to non-driven wheels (F<sub>xrl</sub>-F<sub>xrr</sub>). According to the steady state cornering theory of bicycle model, it is known that the vehicle velocity and yaw rate error satisfy the following equations [6].

$$r_d = \left(\frac{V_s}{L_f + L_r}\right)\delta \quad (12)$$

$$e_r = r_d - r \quad (13)$$

Moreover, Driving/Regeneration braking can be obtained by summing torques applied to rear wheels which are called assistant forces (F<sub>xrl</sub>+F<sub>xrr</sub>). Overall controller consists of four sub-controllers, as shown in Figure 10. As could be seen in this figure, the braking mode control will be activated by brake pedal pressure.

Furthermore, in the driving condition, the assistant torque is evaluated on account of battery state of charge, vehicle speed, and mechanical torque enforced on front wheels. The Assistant/Regenerative torque generated by electrical machines may be near maximum torque capacity. In this case, if the yaw rate error exists, applying the computed torque by the stability mode sub-controller will be impossible. In order to solve this problem, the above mentioned goals are weighted by goal management sub-controller. Additionally, overall control operation will be activated only when the speed of vehicle is non-zero.

Controller structure, fuzzy membership's functions and rule bases are presented as follows. The symbols and definitions used in Figure 10 are noted in the following table.

| Symbol                  | Definition                                    |
|-------------------------|---|
| Gn                      | Gear number                                   |
| V <sub>s</sub>          | Normalized vehicle speed                      |
| X <sub>bp</sub>         | Normalized brake pedal displacement           |
| Th%                     | Throttle opening percent                      |
| T <sub>Rotor_l</sub>    | Left machine torque                           |
| T <sub>Rotor_r</sub>    | Right machine torque                          |
| T <sub>el_max</sub>     | Maximum torque capacity of electrical machine |
| T <sub>mech_drive</sub> | Total of applied torque to front wheels       |
| T <sub>com</sub>        | Electrical machines torque command            |

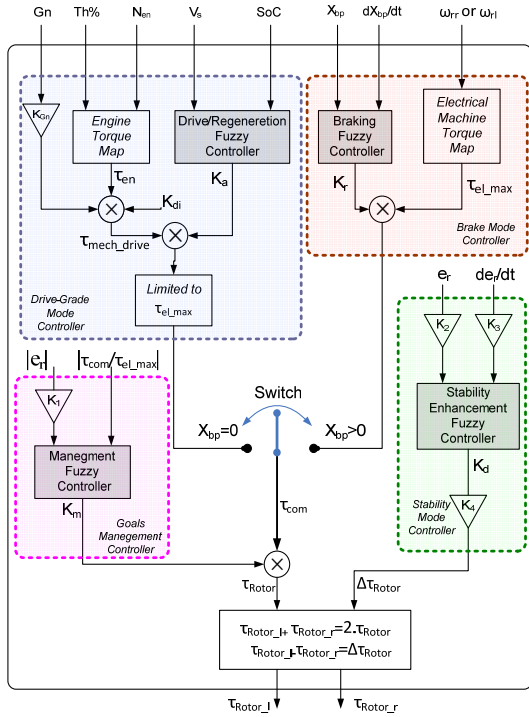


Figure 10: Controller structure

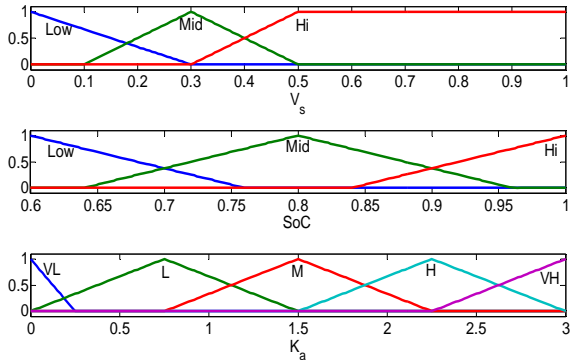


Figure 11: Drive-Grad fuzzy controller memberships

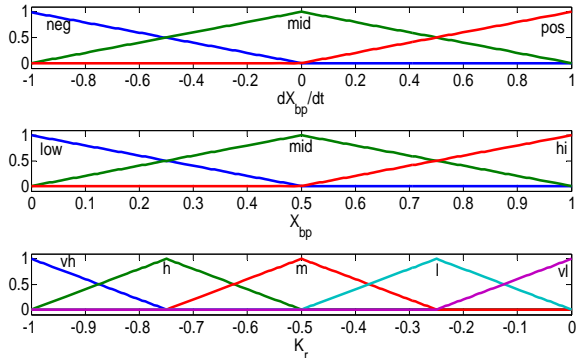


Figure 12: Braking fuzzy controller memberships

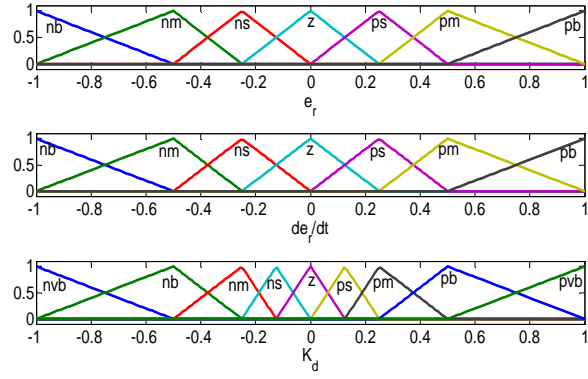


Figure 13: Stability fuzzy controller memberships

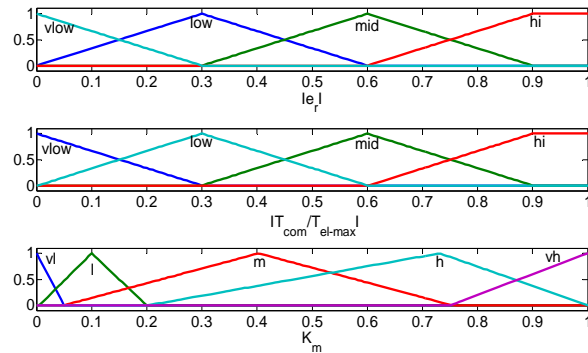


Figure 14: Goals management controller memberships

TABLE 1  
FUZZY ROLL BASE OF EACH SUB-CONTROLLER

| STABILITY MODE RULE BASE | e <sub>r</sub> |                 |                 |                 |    |                 |                 |                 |
|--------------------------|----------------|-----------------|-----------------|-----------------|----|-----------------|-----------------|-----------------|
|                          | NB             | NM              | NS              | Z               | PS | PM              | PB              |                 |
| de <sub>r</sub> /dt      | NB             | NV <sub>B</sub> | NV <sub>B</sub> | NV <sub>B</sub> | NB | NM              | NS              | Z               |
|                          | NM             | NV <sub>B</sub> | NV <sub>B</sub> | NB              | NM | NS              | Z               | PS              |
|                          | NS             | NV <sub>B</sub> | NB              | NM              | NS | Z               | PS              | PM              |
|                          | Z              | NB              | NM              | NS              | Z  | PS              | PM              | PB              |
|                          | PS             | NM              | NS              | Z               | PS | PM              | PB              | PV <sub>B</sub> |
|                          | PM             | NS              | Z               | PS              | PM | PB              | PV <sub>B</sub> | PV <sub>B</sub> |
|                          | PB             | Z               | PS              | PM              | PB | PV <sub>B</sub> | PV <sub>B</sub> | PV <sub>B</sub> |
| DRIVE MODE RULE BASE     |                | V <sub>s</sub>  |                 |                 |    |                 |                 |                 |
| SoC                      | Low            | VL              | VL              | VL              |    |                 |                 |                 |
|                          | Mid            | H               | M               | L               |    |                 |                 |                 |
|                          | Hi             | VH              | H               | M               |    |                 |                 |                 |

| BRAKE MODE      |      | dx <sub>bp</sub> /dt                 |     |     |    |
|-----------------|------|--------------------------------------|-----|-----|----|
| RULE BASE       |      | Pos                                  | mid | neg |    |
| X <sub>bp</sub> | hi   | Vh                                   | h   | m   |    |
|                 | mid  | H                                    | m   | l   |    |
|                 | low  | M                                    | l   | vl  |    |
| GOAL MANAGMENT  |      | τ <sub>com</sub> /τ <sub>e,max</sub> |     |     |    |
| RULE BASE       |      | vlow                                 | low | mid | hi |
| e <sub>c</sub>  | vlow | Vh                                   | vh  | vh  | vh |
|                 | low  | Vh                                   | vh  | h   | m  |
|                 | mid  | Vh                                   | h   | m   | l  |
|                 | hi   | Vh                                   | m   | l   | vl |

7. SIMULATION RESULTS

Certain parameters of an automobile called 'PRIDE' are tabulated below for the simulation

TABLE2  
GEAR BOX DATA

| Gear # | Coefficient | Symbol          |
|--------|-------------|-----------------|
| 1      | 3.454       | K <sub>g1</sub> |
| 2      | 1.944       | K <sub>g2</sub> |
| 3      | 1.275       | K <sub>g3</sub> |
| 4      | 0.861       | K <sub>g4</sub> |
| 5      | 0.6920      | K <sub>g5</sub> |

TABLE 3  
APPROXIMATELY CHANGED PARAMETERS

| Parameter                      | Symbol         | Unit | Value |
|--------------------------------|----------------|------|-------|
| Vehicle total mass             | M <sub>t</sub> | Kg   | 1460  |
| Distance from front axle to CG | L <sub>f</sub> | m    | 1.397 |
| Distance from rear axle to CG  | L <sub>r</sub> | m    | 0.947 |

TABLE 4  
BODY, WHEELS AND DIFFERENTIAL PROPERTIES

| Parameter                      | Symbol         | Unit                             | Value |
|--------------------------------|----------------|----------------------------------|-------|
| Vehicle total mass             | M <sub>t</sub> | Kg                               | 1160  |
| Distance from front axle- CG   | L <sub>f</sub> | m                                | 1.297 |
| Distance from rear axle- CG    | L <sub>r</sub> | m                                | 1.047 |
| Track width                    | T              | m                                | 1.4   |
| Drag coefficient               | C <sub>d</sub> | N.s <sup>2</sup> /m <sup>2</sup> | 0.41  |
| Frontal area                   | A <sub>F</sub> | m <sup>2</sup>                   | 1.8   |
| Lateral area                   | A <sub>L</sub> | m <sup>2</sup>                   | 4.5   |
| Vehicle inertia about z axis   | I <sub>z</sub> | Kgm <sup>2</sup>                 | 1809  |
| Differential coefficient       | K <sub>d</sub> | No unit                          | 3.78  |
| Wheel's longitudinal stiffness | C <sub>x</sub> | N                                | 17500 |
| Wheel's lateral stiffness      | C <sub>y</sub> | N/rad                            | 15000 |
| Wheel's radius                 | R <sub>w</sub> | m                                | 0.272 |
| Wheel's inertia                | I <sub>w</sub> | Kgm <sup>2</sup>                 | 3.264 |

After installing electrical components which change vehicle into a hybrid one, some parameters will change, as shown in Table 3. In the next step, various scenarios will be simulated and the comparison will

be done in order to evaluate the proposed structure's performance.

7.1. Power management examination in civic driving cycles

Three standard driving cycles which are shown in Figure 15 were used for simulations [13]. Engine behavior, battery operation, electrical machines torques, and braking torque for 'INDIA' driving cycle are shown in Figures 16-18. Fuel consumption of all of these three driving cycles is given in Table 5.

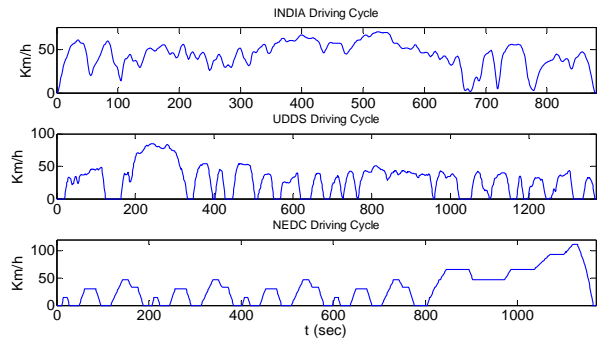


Figure 15: Civic driving cycles

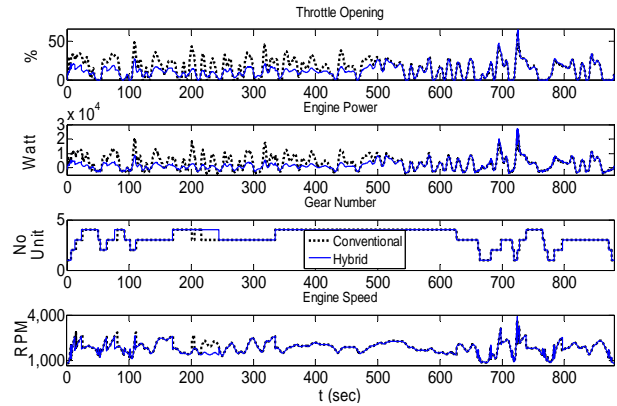


Figure 16: Engine behavior and gear changed in the 'INDIA' cycle



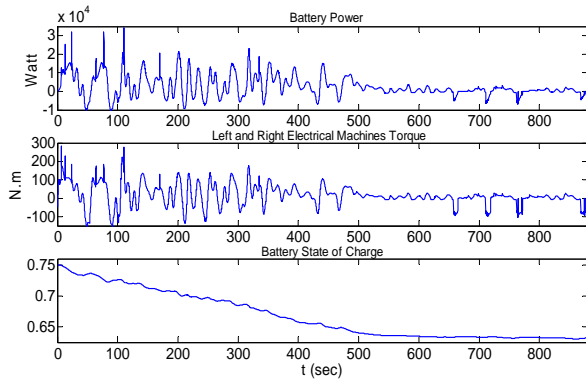


Figure 17: Battery behavior and electrical torques for the 'INDIA' cycle

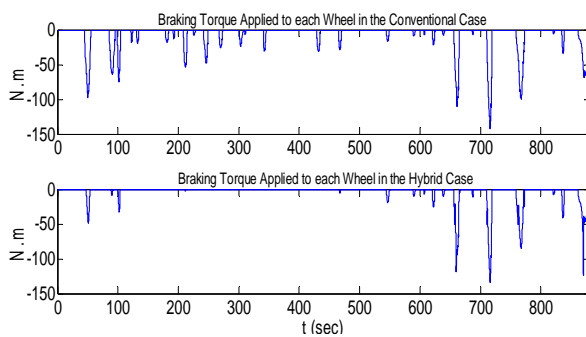


Figure 18: Braking torque applied in the 'INDIA' cycle

TABLE 5  
FUEL CONSUMPTION

| Drivin<br>g<br>Cycle | CONVENTIONA<br>L<br>Litre/(100 Km) | HYBRID<br>Litre/(100Km<br>) | FUEL<br>ECONOM<br>Y |
|----------------------|------------------------------------|-----------------------------|---------------------|
| INDIA                | 6.33                               | 4.31                        | 31.9%               |
| UDDS                 | 8.28                               | 6.33                        | 23.5%               |
| NEDC                 | 8.19                               | 5.11                        | 37.6%               |

According to Figure 16, the engine output power in hybrid case is lower than the conventional one. Also, noticing the vehicle speed and battery output power, shown in Figure 17, when the vehicle speed increases between 350 to 600 sec, participation of electrical machines in power demand will be decreased by the controller. As seen in Figure 18, the vehicle in hybrid case has lower enforced braking torque which is corresponding to regenerative braking condition.

### 7.2. Lane change with high speed

Lane change maneuver results at 70 Km/h on normal road are illustrated as follows. It was assumed that driver applied the same steering effort in both conventional and hybrid vehicles. As depicted in Figure 19, the vehicle in hybrid case is more stable and the yaw rate is very close to the reference value.

### 7.3. Braking on $\mu$ -split road

In this section, braking at 110 km/h on a  $\mu$ -split road (corresponding to dry pavement,  $\mu=0.95$ , on the right side and unpacked snow,  $\mu=0.35$ , on the left) was simulated. During the simulation, the steer angle was assumed to be zero. The vehicle speed reduction and simulation results are depicted in Figure 20. This figure shows that the hybrid vehicle has better stability during braking and the undesired lane change is also lower than the conventional one.

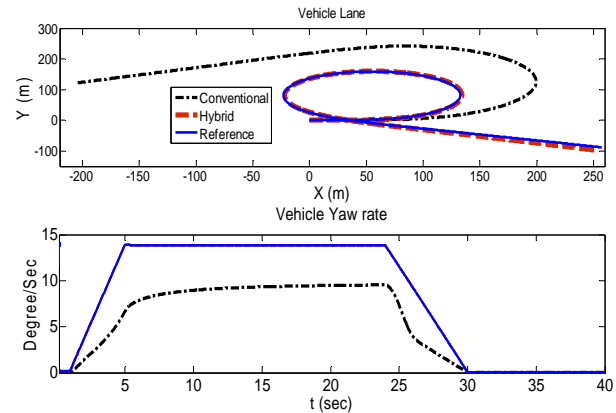


Figure 19: Yaw rate and vehicle lane during of lane changing

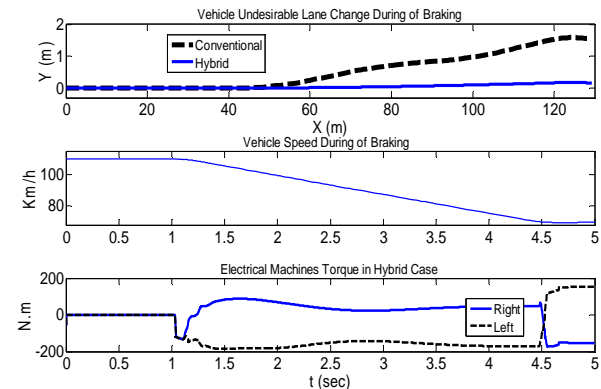


Figure 20: Vehicle speed, vehicle lane and electrical machines torque during of braking

## 8. CONCLUSION

In this paper, a driver-assistant stability system with Driving/Regeneration braking was introduced for a front differential vehicle using electrical traction system on rear wheels. Intelligent performance of the overall control system in making electrical machines' torque commands based on the driving necessities was the main advantage of the proposed controller. The simulation results showed intelligent performance of the proposed control system in various driving environments such as slippery roads.

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## BIOGRAPHIES



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