Abstract—This paper proposes the application of a new optimal based controller for vehicle handling and stability system based on well known model matching structure. The controller is known as the Composite Nonlinear Feedback (CNF) controller which is expected could further improve the conventional optimal based controller (i.e. Linear Quadratic Regulator (LQR)). This is due to the inclusion of an additional nonlinear term in its control law which further reduces the feedback error of vehicle system with the constraint of input saturation. The controller has been applied to the direct yaw moment compensation system as the correction input to enhance the vehicle dynamic behavior. Two selected maneuvers have been numerically simulated on the nonlinear vehicle model using MATLAB/Simulink to examine its effectiveness. Results show improvements in the yaw tracking capability, hence reduces the effect of oversteer or understeer during critical maneuvers.

Keywords—direct yaw moment; composite nonlinear feedback; independent-wheel-drive; electric vehicle;

I. INTRODUCTION

Recent development in EV industry, the introduction of in-wheel-motor has offered numerous advantages to today’s vehicle. Consequently, the implementation of supreme vehicle control is practically feasible. In recent technology, the in-wheel-motor has been introduced with the capability to supply the driving propulsion directly at the wheels. This will reduce the mechanical losses, hence made the EV dynamic control becomes faster and more precise. A large number of researches in EV conducted so far, concerned on the performance of the motor characteristics and battery capabilities. However, a few studies are dedicated to explore the potential of IWD-EV’s advantage in terms of traction and handling. This is due to its notable difficulties dealing with torque allocations of each wheel, typically for situations such as maneuvering in a bad road conditions [1].

Various types of control strategies had been introduced aiming to improve vehicle dynamic behavior during such maneuvers. The vehicle dynamic behavior can be considered as a highly nonlinear system environments. Model matching approach is one of proven control strategies which effectively could handle the dynamic response of the vehicle. With such control structure approach, control theory such as simple PID, optimal based theory, surface sliding theory and etc, could be applied into the compensation system in order to mitigate the dynamic error response. Hence, it improves the vehicle behavior towards its targeted response. In the past decade, a new nonlinear control technique had been introduced by [2] which is called the Composite Nonlinear Feedback (CNF) control that improves the tracking control objective. The technique had initially been applied for a higher order MIMO system in the applications such as hard disk servo system and servo position system with disturbance [2],[3].

Recently, such control technique also had been applied in vehicle dynamic system. In [4], this tracking control technique had been applied for the active front steering system to improve the vehicle yaw rate tracking. Results in [4], show that this new control approach could offer great potential in other nonlinear applications, particularly in vehicle dynamic behavior improvements. Nevertheless, such control approach has not yet being applied in the torque distribution system for vehicle handling and stability improvements. Therefore, in this paper, the potential of CNF control theory is investigated to be as the feedback control technique for the purpose of vehicle dynamic behavior improvements via direct yaw
moment compensations.

II. NONLINEAR VEHICLE AND TIRE MODEL

In this section, the modeling of the nonlinear vehicle model and its associate model are briefly being described.

A. Nonlinear vehicle model

The vehicle model used in the simulations consist of 8-DOF nonlinear vehicle model as shown in Fig. 1. These well-known motions equations are based on the Newton’s Law of motion and have comprehensively been described in literatures. However, for this particular simulations, the response of heave, roll and pitch motion are neglected for analysis. A quasi-static ride model is used to generate the load transfer between the front and rear axle, while the handling model is used to generate the resultant vehicle longitudinal, lateral and yaw motion. The wheel model is also included, to simulate the tire forces and the wheel rotational response. Details descriptions of the vehicle modeling is as found in [5].

B. Tire model

The adhesion coefficient, \( \mu \) can be estimated based on various established models either using semi-empirical methods or linear piecewise relationships. The adhesion coefficient is a nonlinear function, based on the road pavement as well as vehicle longitudinal and lateral slip. In this paper, \( \mu \) is calculated based on a simplified Calspan tire model which could be able to describe the vehicle behavior in any driving scenario [6]. The tire chamber effect is assumed negligible and the detail descriptions could be found in [6].

III. CONTROLLER DESIGN

A. Model matching controller

In this paper a model matching controller (MMC) as proposed by [7], [8], has been adopted as the key approach of designing the control structure. The MMC is based on a 2-DOF linear vehicle model system, consists of lateral and yaw motion of the vehicle. The governing equations of the model can be expressed as follows:

Lateral motion:

\[
 mv(\dot{\beta} + r) = F_{yFL} + F_{yFR} + F_{yRL} + F_{yRR}
\]

Yaw motion:

\[
 I_z \dot{r} = l_F(F_{yFL} + F_{yFR}) + I_R(F_{yRL} + F_{yRR}) + M_{zc}
\]

\[
 M_{zc} = T/(2(F_{xFL} + F_{xRL}) - (F_{xFR} + F_{xRR}))
\]

where \( F_{xFL}, F_{yFL} \) denotes as the longitudinal and lateral force generated at each wheel, \( M_{zc} \) denotes as the corrective yaw moment, while \( \beta \) and \( r \) denotes as the vehicle sideslip and yaw rate response respectively.

From (1)~(2), the state space representation of the linear vehicle model can be written as:

\[
 \dot{X} = AX + BU + E\delta_P
\]

where,

\[
 X = \begin{bmatrix} \beta \\ r \end{bmatrix}; \quad U = \begin{bmatrix} M_{zc} \end{bmatrix}; \quad B = \begin{bmatrix} 0 \\ \frac{2C_F}{I_z} \end{bmatrix}; \quad E = \begin{bmatrix} 0 \\ \frac{2C_F}{I_z} \end{bmatrix};
\]

and,

\[
 A = \begin{bmatrix} -\frac{2(C_F+C_R)}{mv} & -\frac{2(l_F C_F - l_R C_R)}{I_z} \\ -\frac{2(l_F C_F - l_R C_R)}{I_z} & -\frac{2(l_F C_F - l_R C_R)}{I_z} \end{bmatrix} - 1
\]

On the other hand, to improve the handling and stability of the vehicle, two variables that are the sideslip and yaw rate response must follow its targeted value. The desired sideslip response is designed to be zero in steady state at the center of gravity, while the desired yaw rate response is represented by the first order lag. Both desired parameters are written as follow;

\[
 X_d = \begin{bmatrix} \beta_d \\ r_d \end{bmatrix} = \begin{bmatrix} 0 \\ \frac{K_{yd}}{1 + \tau_{yd}s} \end{bmatrix} \cdot \delta_P
\]

\[
 K_{yd} = \frac{vL}{1 + K_{us} \cdot v^2}
\]

where \( K_{us} \) and \( \tau_{yd} \) are the stability factor and the desired time constant respectively.

Figure 1. Vehicle model
B. Control structure

The control structure proposed in this paper is as depicted in Fig. 2. The structure consists of two compensation strategies that are the feed forward and feedback compensations. The yaw rate and the sideslip response are selected as the control variables to guarantee the performance of vehicle handling and stability respectively. A fix steering input is applied only at the front wheels with the correction of the vehicle behavior only assisted by the torque of the corrective yaw moment.

C. Feed forward compensation

The main objective of this module is to suppress the sideslip angle to its desired value. The purpose of the feed forward compensation is to complement the response of the feedback compensation in the system. Basically, this module represent the relationship between two main inputs of the vehicle (i.e. steering input and yaw moment) which allow the inclusion of the steering input in terms of the designed control input as in the following expression;

\[ M_{zc}(s) = U_{ff} \cdot \delta_F(s) \]  

(7)

where \( U_{ff} \) is the proportional gain of the feed forward compensation. Considering the vehicle is in a steady state condition (i.e. constant sideslip and yaw rate values), via applying a Laplace transform to the lateral and yaw rate motion equations, the feed forward gain can be determined as (8).

\[ U_{ff} = -l_F \cdot C_F \cdot \left( \frac{l_R \cdot C_R}{m^2} \right) \frac{(1 + l_R) - \frac{m^2 y}{2}}{2(l_F \cdot C_F - l_R \cdot C_R)} \]  

(8)

D. Feedback compensation

Principally, the feedback compensation could overcome the limitation of the feed forward correction strategy. It works as the error suppression module which mitigates the error caused by any external disturbance or dynamic changes in the system. In [8], the feedback compensation is designed based on optimal theory, in which only a linear term dictated the gain response of the controller. The feedback compensation adopted in this paper is an extension of the optimal based controller in which it consists of the linear and nonlinear law. Considering the derivative of error response of the system can be is written as;

\[ \dot{E} = \dot{X} - \dot{X}_d \]  

(9)

By substituting the actual and desired state equation into (9), the reduced form of error equation can be written as follows;

\[ \dot{E} = AE + BU_{fb} + D_e \]  

(10)

where \( D_e \) is the sum of the third and fourth terms of (9) which is considered as a steering input dependent lumped disturbances.

E. Composite Nonlinear Feedback, (CNF) controller

Assuming that the disturbance terms to be neglected, the CNF controller feedback control law can be described as in (11). The control input consists of a combination of linear and nonlinear elements of the feedback system.

\[ U_{fb} = U_{fb(linear)} + U_{fb(nonlinear)} \]  

(11)

To design the CNF controller, a system is considered as follows;

\[ \dot{x} = Ax + Bu \]

\[ y = Cx \]  

(12)

where \( A, B, \) and \( C \) are a constant system matrices, \( x \) is the system state variable, \( y \) is the system output, and \( u = U_{fb} \) is the control input to the system.

The systems matrices must first being considered in compliance with the following assumptions;

(A, B) is controllable/stabilizable.

(A, C) is observable/detectable.

(A, B, and C) is invertable and has no zero at \( s = 0 \).

In general, the linear term of the CNF feedback law consist of two important matrices described as follows,

\[ U_{fb(linear)} = FX + Gr \]  

(13)

where \( r \) is step input reference, \( G \) is a scalar matrix given by (14), and \( F \) is the feedback matrix which can be determined via various optimal based method such as \( H_2, H_\infty \), or pole placement. In this paper, LQR theory is adopted to assign the \( F \) matrix value.

\[ G = -[C(A + BF)^{-1}B]^{-1} \]  

(14)

On the other hand, the nonlinear feedback law can be described as in the following equations:

\[ U_{fb(nonlinear)} = \rho(r, y)B^TP(x + (A + BF)^{-1}BGr) \]  

(15)

where \( \rho(r, y) \) is a non-positive function locally Lipschits, in that \( y \) is used to modify the close-loop system output approaches the desired command. \( P \) is a positive matrix resolved by the solution of Lyapunov equation. Details descriptions can be found in [2], [3], [4].
F. Torque distribution module

The purpose of this module is to distribute the torque required by each motor. In this paper, all wheels of the vehicle are assumed installed with an in-wheel-motor. The corrective torque is evenly distributed for each of the vehicle propulsion mechanisms via the following expression [9], [10]:

$$T_i = \frac{2M_{zc}R_w}{W}$$

where $i$ denotes the front/rear and left/right wheel of the vehicle.

IV. NUMERICAL SIMULATION

The evaluation of the proposed control scheme has been conducted via numerical simulations using MATLAB/Simulink as the platform. All associate modules have been modeled accordingly and the simulation parameters are as tabulated in Table I. Simulations have been conducted based on two critical high speed maneuvers. The simulated road condition is considered on a wet asphalt surface (i.e. $\mu = 0.6$). Steering input for both maneuvers are as depict in Fig. 3. The initial vehicle velocity is assumed at 20 m/s. The vehicle’s sideslip and yaw rate dynamic behavior are investigated and the proposed control scheme is compared with well known optimal based method (i.e. LQR) to evaluate its effectiveness.

A. Lane change maneuver

Simulation results of lane change maneuver are as depict in Fig. 4, and Fig. 5. The results show that with the conventional LQR control method the vehicle maintains its handling and stability performance due to its ability to track the desired yaw rate while maintaining the sideslip angle within the stability margin. However, a slight enhancement can be accomplished as for CNF controller. The yaw rate tracking objective is improved and the vehicle successfully follow the desired path.

B. J-curve maneuver

On the other hand, in J-curve maneuver, the proposed controller also shows a slight improvement in both investigated...
dynamic behaviors. The yaw rate response is improved, hence reduced the potential of oversteer effect. The results are as depict in Fig. 6, and Fig. 7. Results analysis are as described in the following sub-section.

C. Results analysis

The performance of the proposed controller has been evaluated based on two criterions. The sideslip of the vehicle must comply with a stability margin as in (17) to secure its handling and stability performance [9]. Meanwhile, the vehicle path tracking capability is evaluated based on the dynamic response of vehicle yaw rate to track the targeted value.

\[ |2.4979\hat{\beta} + 9.549\beta| < 1 \]  \hfill (17)

The results analysis for both maneuvers are as tabulated in Table II and Table III. Based on the tables, the proposed CNF controller shows an improvement in both designated maneuvers. Fig. 8 and Fig. 9 depict the error reductions of the yaw rate tracking. Performance improvements up to 90 percents can be achieved when compared with a conventional optimal based method. The proposed CNF controller shows good potential in order to improve vehicle tracking objective via direct yaw moment approach.

V. Conclusion

In this paper, a new technique based on optimal based method is introduced for vehicle stability and handling per-
formance improvements via wheel torque distributions. The technique is based on the model matching controller method with the application of CNF control theory as the feedback compensation tool. The torque at each wheel is evenly distributed based on the vehicle desired corrective yaw moment during maneuvers. From the simulation results assessment, the proposed CNF controller shows preeminent performance with respect to conventional optimal based technique. The tracking error can be further reduced while maintaining the vehicle dynamic stability within its stable margin. It should be noted that the controller only appropriately modify the torque distribution of the IWD-EV with no active steering correction strategy applied in the system. With the inclusion of such active steering system, the dynamic response of the vehicle and its tracking capability performance are expected will offer a promising improvement. The direction of future works will concentrate on the tuning strategies of the controller.

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