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ADVANCES IN TILT CONTROL DESIGN OF HIGH-SPEED RAILWAY VEHICLES: A STUDY ON FUZZY CONTROL METHODS

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Abstract. *The advantage of high speed trains is in the reduction of journey times between two places. Most countries developed new infrastructure to accommodate the need for high speed trains. However, this approach is rather costly. An alternative solution, which avoids new infrastructure costs and merely increases maintenance cost of current rail tracks, is to introduce tilting train technology. The main idea is tilting the vehicle body while on curved sections of the rail track. Current technologies in tilting railway vehicles use a 'precedence' control scheme; however, this increases complexity on the actual controller structure and inter-vehicle signal connections. Research on local sensor loop control strategies is still important to overcome such drawbacks. Work using conventional and modern control approaches has been investigated by previous researchers. In this paper, we propose a fuzzy correction mechanism acting as 'add-on' to enhance the capability of the controller response on curved track without compromising the effect from track irregularities on the vehicle. The fuzzy correction mechanism, as it is referred to, is applied in series with the nominal controller. Furthermore, the proposed control scheme is compared with a precedence type controller and a classical type controller to illustrate its effectiveness.*
Keywords: Control systems, Genetic algorithms, Fuzzy logic, Tilting railway vehicle, Tilt control

1. **Introduction.** Active tilting technology has been widely introduced for high speed trains over the last twenty years. An important task of high-speed trains is to shorten journey time between two places. There are two ways to accomplish this task: (1) to develop a new rail track infrastructure to accommodate increased train speed or (2) use existing curved track with a modified type of train vehicle. The second choice is most preferred due to avoiding cost of building new infrastructure. The actual way to do this is by tilting the car body when the train traverses a curved track.

Note that when a train traverses a curved section of the track, the passengers will experience a speed-dependent lateral acceleration (the faster the speed, the higher the acceleration). This may lead to feeling uncomfortable and also may experience nausea (due to roll velocity components during transients). In fact, the curved track is actually canted (elevated by an appropriate angle, given each country's rail standard practice) so as part of the gravitational force compensates for part of the centrifugal force. However, due to limited track elevation (for safety purposes while train halts on a curved track) tilting is the key point to further compensate the lateral forces as speed increases.

Control systems in early tilting trains utilised only local vehicle measurement, i.e., sensors mounted on the current vehicle. The signals from body-mounted accelerometers were feedback to adjust the tilt angle and hence minimising the passenger lateral acceleration. The advantage of using this strategy is that the signals are not significantly affected by track irregularities, since these are filtered by the secondary suspensions. However, it proved impossible at the time to get an appropriate combination of straight track and curving performance.

In Europe most systems now use the command-driven with precedence control [3] devised in the early 1980s as part of the Advanced Passenger Train development. In this scheme a bogie-mounted accelerometer from the vehicle in front is used to provide "precedence" information, carefully designed so that the delay introduced by the filter compensates for the preview time corresponding to a vehicle length. Although there has been some development of the concept, including the use of additional sensors, the overall principles remain the same. In addition, there are various developments based upon using curving information from track databases, either using direct train location data, or by synchronising the database information on the basis of the tilt sensor signals [5, 7]. Nevertheless achieving a satisfactory local tilt control strategy is still an important research target due to system simplifications and more straightforward failure detection.

Investigation using conventional PI controller in local-loop tilt control system has been reported by [11] and further using modern control approaches has been reported in [12]. The work of intelligent control focusing on fuzzy control method has been progressing in the past 20 years [1, 4]. Preliminary investigation using fuzzy control in tilt local feedback loop has been presented in [8] and optimisation in Fuzzy tilt control in [9]. In this paper, authors propose a fuzzy correction mechanism as 'add-up' factor to take advantages on the linguistic rules in fuzzy logic. The fuzzy correction mechanism is used in series with conventional controller to further improve the curved and straight track performance.

2. **Vehicle Model.**

Railway vehicles are dynamically complex systems characterised by significant coupling between the lateral and roll motion which is often referred to as the 'sway modes' (see Figure 1). The mathematical model of the system is based upon the end-view of a railway vehicle, to incorporate both the lateral and roll degrees of freedom for both the body and the bogie structures. A pair of airsprings represents the secondary suspension, whilst the primary suspension is modelled via pairs of parallel spring/damper combinations. The stiffness/damping of an anti-roll bar connected between the body and the bogie is also included. The model also contains an anti-roll bar connected between the body and the bogie frame, which provides active tilt using a rotational displacement actuator [6]. Since in this research, the focus is on the behaviour of the roll and lateral motion of the vehicle, it is not necessary to include the wheelset dynamics. Thus, the wheelset is assumed to be perfectly connected to the railway track. However, the effect of wheelset dynamic is incorporated by using a 2$^{\text{nd}}$ order Low Pass Filter with 20 Hz frequency cutoff and 20% damping.

The tilt model system can be represented in state space by

$$\begin{aligned} \dot{x} &= Ax + Bu + \Gamma w \\ y &= Cx \end{aligned} \tag{1}$$

where $A \in \mathbb{R}^{n\times n}$, $B \in \mathbb{R}^{n\times m}$ and $C \in \mathbb{R}^{p\times n}$ and n, m and p represent the number of states, control inputs and outputs respectively.

Consider ω as constant external disturbance matrix given by

$$w = \left[\ \tfrac{1}{R}\ \theta_o\ \dot{\theta}_o\ \ddot{\theta}_o\ y_o\ \dot{y}_o\right]^T \tag{2}$$

where

$\frac{1}{R}$	curvature
$\theta_o\ \dot{\theta}_o\ \ddot{\theta}_o$	elevation track components
$y_v\ \dot{y}_v$	lateral track irregularities components

and the state vector x,

$$x = [\ y_v\ \theta_v\ y_b\ \theta_b\ \dot{y}_v\ \dot{\theta}_v\ \dot{y}_b\ \dot{\theta}_b\ \theta_r]^T \tag{3}$$

and $u = [\delta_a]$. For the parameters see section Notation.

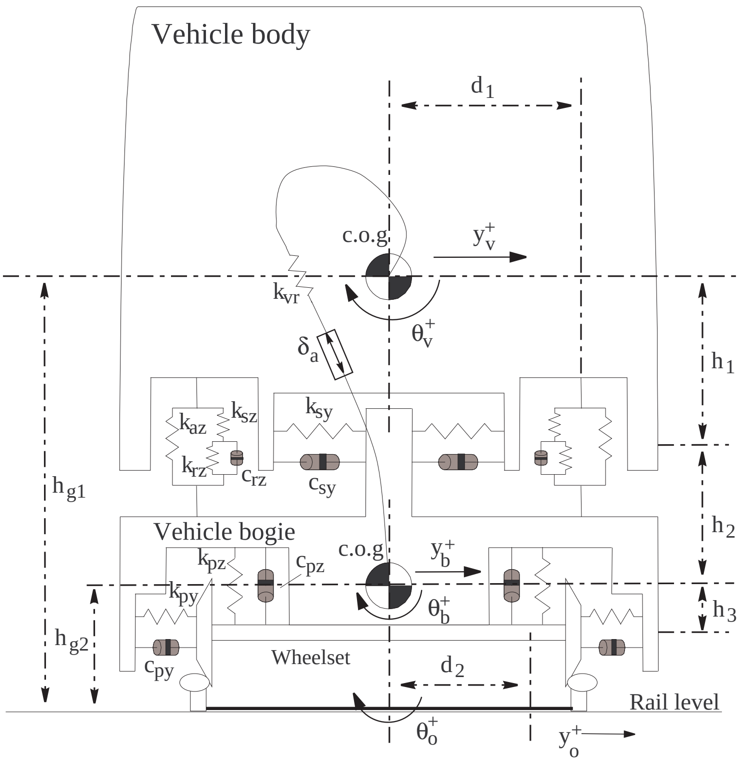


FIGURE 1. End-view of the vehicle model

Moreover, the variety of track inputs also contribute to the system complexity which can be categorized into deterministic and stochastic. Deterministic inputs refer to the curved track which is designed by civil engineers to meet the requirement, of the passenger comfort. The curved track is leaned inward (elevated) around 6°, rising linearly over a period of 2-3 seconds at the transition of the start and end of the curve. The stochastic track input represents the deviations of the actual track from the intended alignment, irregularities which occur in the vertical, lateral and cross-level directions. The secondary suspension of the vehicle is designed to reduce the effect of track irregularities, expressed in RMS acceleration levels in the body of the vehicle. In principal, the design of the tilt controller is to provide fast response related to the transition to and from the curves, but at the same time does not affect the responses on track irregularities, i.e., the ride quality on straight track.

3. **Performance Assessment.** Two main design criteria are concerned with tilting trains: (i) *deterministic criterion*: provide a fast response on curve track, (ii) *stochastic criterion*: maintain an acceptable ride quality in response to track irregularities on straight track segments. The performance assessment approach proposed in [10] was utilised for the purposes of this work.

The assessment on *curve transition* relates to the idea of "ideal tilting", i.e., whereby the tilt action follows the specified tilt compensation in an ideal manner according to the maximum tilt angle and cant deficiency compensation factor. Deviation from the "ideal tilting" response quantifies the additional dynamic effects caused by the suspension/controller dynamics during the transitions to and from the curves, and provides an objective measure which can be used to compare different strategies.

The *straight track* performance criterion is based on the RMS lateral ride quality in response to the effect of track irregularities, typically no more than 7.5% degradation. For proper comparison this is made at high speed for both the tilting and non-tilting cases.

However, any tilt control system directly controls the secondary suspension roll angle and not the vehicle lateral acceleration. Thus, a fundamental trade-off exists between the vehicle curve transition response and straight track performance. In addition, for reasons of human perception, designers employ *partial tilt compensation*, whereby the passenger still experiences a small amount of acceleration on steady curve in order to minimis motion sickness phenomena.

4. **Fuzzy Correction Mechanism in Tilting Nulling Control Scheme.** The design of fuzzy correction mechanism is mainly to overcome the difficulty to achieve simultaneously improvement both in stochastic and deterministic track performance. Work in [12, 14] studied the use of both classical and modern control approaches to improve curved track performance, however difficulty arise to achieve good ride quality on straight track without compromising the performance on curved track. The design idea of fuzzy correction mechanism is based on heuristic control, where the rule and the membership functions are designed mainly to:

- minimise effect of controller gain on straight track.
- fast response on curve transition.

Figure 2(a) shows the overall concept of the fuzzy correction mechanism control scheme. Two signals are used as inputs to the fuzzy mechanism: controller output, u', and measured body roll velocity, $\dot{\theta}_{vm}^{gyro}$. The roll rate signal is used to differentiate between the curved and straight track, while the controller output u' is to give accurate responses of the system. Based on these two signals, the membership functions and the rules based are developed.

The fuzzy correction mechanism, both inputs are shown in Figure 2(b), consist of three equally distributed Gaussian Membership Functions with 25% overlap for each signal. Furthermore, the *Center of Area* (COA) defuzzification procedure with well known max-min inference method.

The linguistic variables for each membership function represent the condition for each value. For example, the regulator output u' is represented by the linguistic variables *Neg*, *Zero* and *Pos*. For the body roll gyro input $\dot{\theta}_{v}^{gyro}$, the linguistic variable are also *Neg*, *Pos* and *zero*, while for the fuzzy correction output u'', the linguistic variables represent the tilting direction of the car body as *tilt the car body clockwise maximum* represented by *TiltClkwM*, and tilt car body medium anticlockwise represented by *TiltAclkwm*, etc. Clockwise and Anticlockwise characterize the direction of tilt based on the curve direction

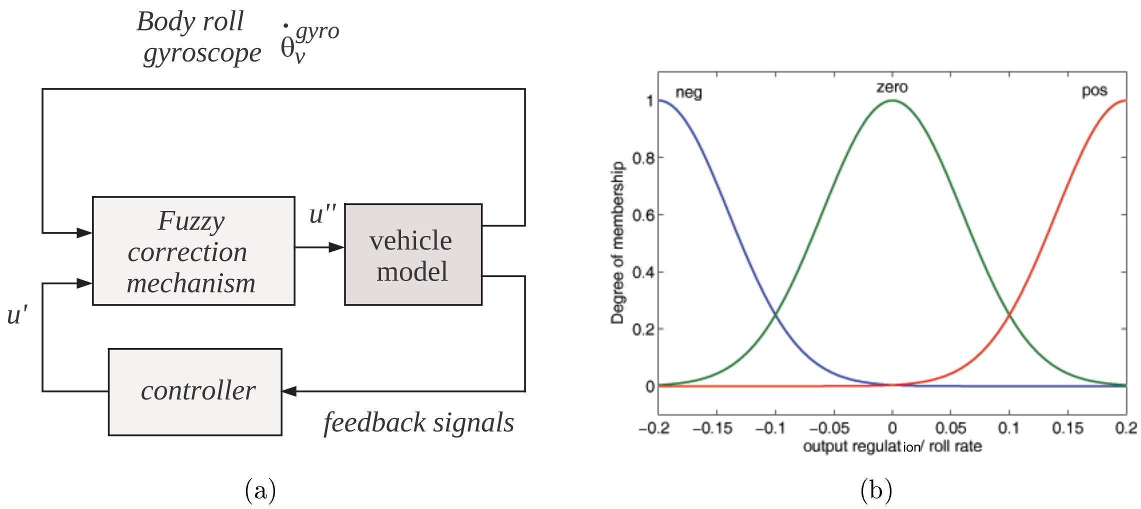


FIGURE 2. Add-on fuzzy mechanism: (a) the basic concept, and (b) fuzzy membership

(i.e., inwards and outwards of the curve respectively). Note that the membership function ranges represent the required operating range of the variables. Detail on the rules is shown in Table 1.

TABLE 1. Fuzzy correction rule base

$\dot{\theta}_v^{gyro}/u''$	Neg	Zero	Pos
Neg	TiltClkwm	TiltClkwm	TiltClkwM
zero	TiltAclkwM	NoChange	TiltClkwM
Pos	TiltAclkwM	TiltAclkwm	TiltAclkwm

The fuzzy correction mechanism is then applied in series with the controller as described later. Here the correction mechanism was applied PID and LQG control schemes to prove the effectiveness of utilised the proposed schem.

4.1. **PID controller with fuzzy correction.** This scheme involves the use of a conventional PID controller to provide fast tilt response in curved track, while the fuzzy correction increases damping and improves straight track performances. The PID controller utilise two signals (roll gyro and lateral acceleration) to give 60% tilt compensation. Two signals are used as inputs to the fuzzy correction: PID output (u) and measured body roll velocity ($\dot{\theta}_v^{gyro}$). The roll velocity is used to give extra information on straight and curved track. The control scheme can be seen in Figure 3.

For this scheme, the design of the controller can be divided into three stages:

1. design PID controller to give fast response on curved track, neglecting influences of track irregularities.
2. design fuzzy correction with the objective to minimis straight track irregularities and prevent overshoot and oscillation on curved track.
3. tune further as necessary.

Eight real-coded GA variables were used to optimise the PID parameters (K_p, K_I, K_D), the position and the width of the output fuzzy membership functions. The tuning process aiming to improvement tilt performance on curved track while maintaining below 7% ride quality degradation on straight track performance. The tuning process involved tuning

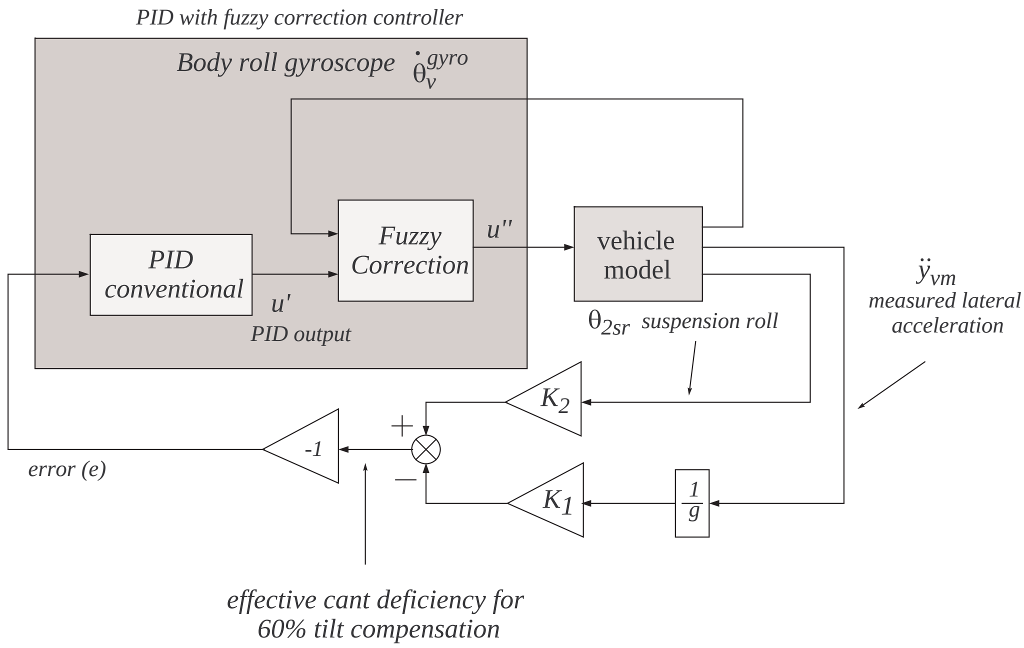


FIGURE 3. PID with fuzzy correction control scheme

the output membership functions while the scaling functions for the inputs membership functions were set to 0.215 and 0.2, related to the maximum expected control signal in rad and body roll gyro in rad/s^2. The output memberships consisted of three triangular and two trapezoidal membership functions located at each end of the fuzzy set. The width and position of the membership functions were transformed into chromosomes to search for the 'best location' in the fuzzy set. These chromosomes used real coded GAs to represent the actual values of the membership functions. There can be represented by mathematical equations as described below:

TiltAckwM *membership functions*. These membership functions located at the end of the universe of discourse used trapezoidal shape membership functions as follows:

$$\text{trapezoid}\,(x; p_1, w_1) = \begin{cases} 0, & x \le p_{\min} \\ 1, & p_{\min} \le x \le p_1 \\ \dfrac{w_1 + p_1 - x}{w_1}, & p_1 \le x \le w_1 + p_1 \\ 0, & w_1 + p_1 \le x \end{cases} \tag{4}$$

where p_1 and w_1 are the position and width of the membership functions, respectively. $p_{\min}$ is a constant variable representing the maximum allowance, and actuator, σ_a drive angle equals to -0.27. The '-ve' sign represents the body roll angle in the opposite direction to the current one.

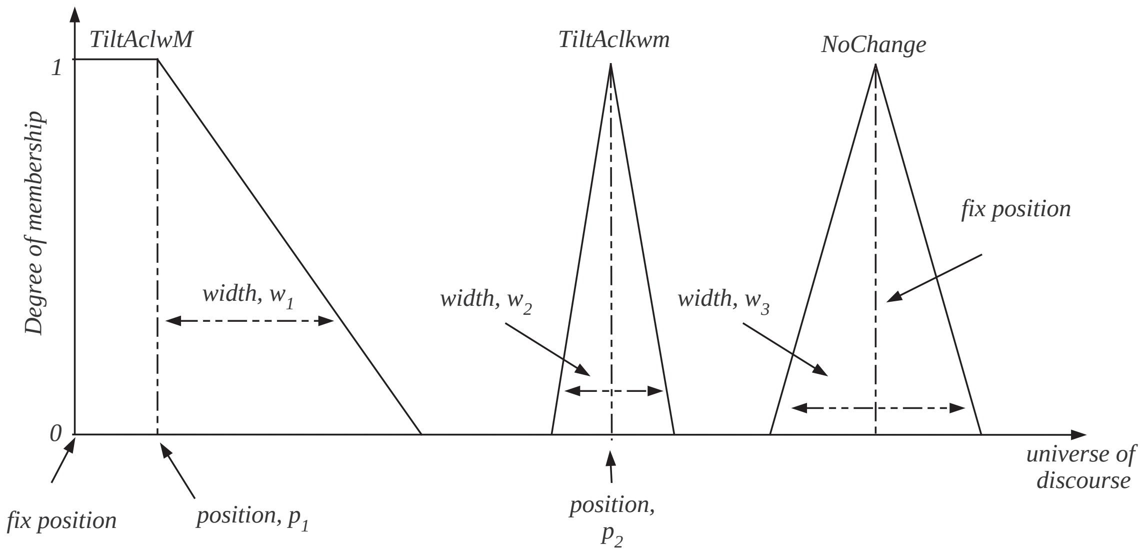


FIGURE 4. Coding membership functions

TiltAckwm *membership functions*. These functions used triangular membership functions and are specified by two (p_2, w_2) parameters as follows:

$$\text{triangle}\ (x; p_2, w_2) = \begin{cases} 0, & x \leq \dfrac{2p_2 + w_2}{2} \\ \dfrac{p_2 - w_2 + x}{2p_2 - w_2}, & \dfrac{w_2 - p_2}{2} \leq x \leq p_2 \\ \dfrac{2(p_2 - x) + w_2}{w_2}, & p_2 \leq x \leq \dfrac{2p_2 + w_2}{2} \\ 0, & \dfrac{w_2 + p_2}{2} \leq x \end{cases} \tag{5}$$

where p_2, w_2 are the position and width of the membership functions.

The membership functions mentioned above are symmetrical for both sides (to reduce complexity and fast convergence) except for the *NoChange* membership function. This function is located at the centre of the universe of discourse, therefore only the width w_3 is to be optimised. Figure 4 shows the overall concept of the coding membership functions.

Six objectives (f_x) with constraints (ω_x) as shown in Figure 5 were used:

1. Deterministic: settling time at steady state curve t_s, not more than 5 sec

$$f_1 = t_s \tag{6}$$

where the constraint function is given by

$$\omega_1 = \begin{cases} f_1 & \text{if } f_1 > 5 \text{ sec} \\ 0 & \text{otherwise} \end{cases}$$

2. Deterministic: optimise curved track response based on maximum peak lateral deviation error between ideal $\ddot{y}_{m_i}$ and actual $\ddot{y}_m$ lateral acceleration taken between 1 sec before and 3.6 sec after start and end of transition for both in(f_2) and out(f_3) of curved transition track, given by

$$f_{(2,3)} = |\ddot{y}_{mi(2,3)} - \ddot{y}_{m(2,3)}|_{peak} \tag{7}$$

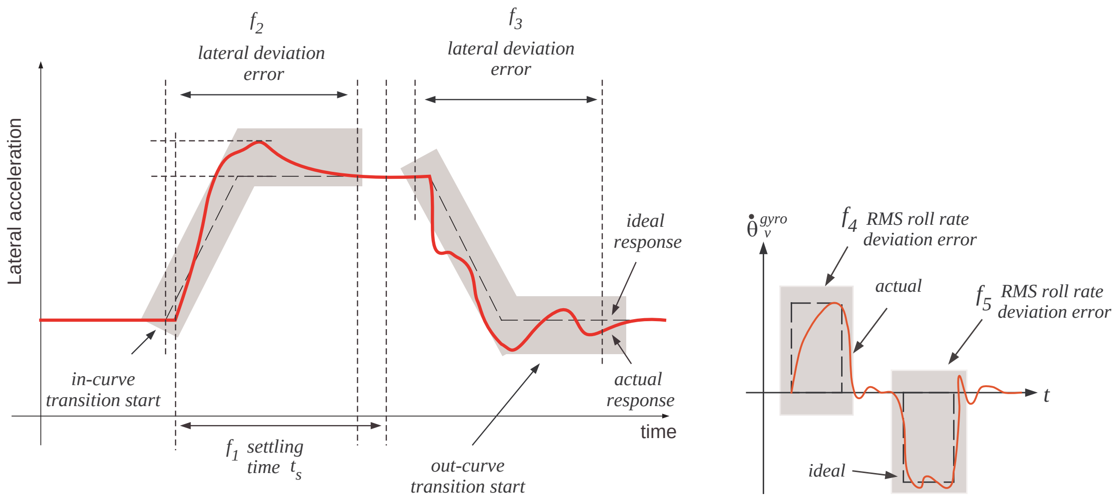


FIGURE 5. Training trajectory for (a) body lateral acceleration, and (b) roll rate angle

and the constraint function is given by

$$\omega_{(2,3)} = \begin{cases} f_{(2,3)} & \text{if } f_{(2,3)} > 5.5\ \%\text{g} \\ 0 & \text{otherwise} \end{cases}$$

3. Deterministic: optimise body roll rate response to system changes based on the RMS (root mean square) deviation error between the actual $\dot{\theta}_m$ and ideal roll rate $\dot{\theta}_{m_i}$ profile. The calculation is taken from 1 sec before until 3.6 sec after the start and at the end of curve transition for both in(f_4) and out(f_5) of the curved transition track:

$$f_{RMS(4,5)} = \sqrt{\frac{1}{T_{2(4,5)} - T_{1(4,5)}} \int_{T_{1(4,5)}}^{T_{2(4,5)}} |\dot{\theta}_{mi_{(4,5)}} - \dot{\theta}_{m_{(4,5)}}|^2 dt} \tag{8}$$

where T_1 and T_2 the respective curve transition times, and the constraint function are,

$$\omega_{(4,5)} = \begin{cases} f_{(4,5)} & \text{if } f_{4,5} > 0.03 \text{ rad/s} \\ 0 & \text{otherwise} \end{cases}$$

4. Stochastic f_6: constrain the degradation of the straight track ride quality within the allowed degradation of 7.5% taken between the active (with controller) and passive (without controller) system at high speed.

$$\omega_6 = \begin{cases} f_6 & \text{if } f_6 > 7.5\% \\ 0 & \text{otherwise} \end{cases}$$

The constraint violation functions are added together to give the overall constraint, given by

$$\Omega_x = \sum_{x=1}^{6} \omega_x$$

Therefore, the objective function with constraint is given by

$$F_m(x) = f_x + R_x \Omega_x$$

where R_x is the penalty parameter and $x = 1, 2, \ldots, 6$. Figure 5 shows the graphical responses of the overall calculations.

4.2. **LQG with fuzzy correction mechanism.** In fact, work in [13] has revealed the potential of using LQG with integral action to give an acceptable response of the tilt system. However, the authors enhance the LQG scheme with the addition of the fuzzy mechanism to further improve the design trade-off. The concept and design of fuzzy correction is similar to that used in PID fuzzy Correction control scheme in Section 4.1, but with the controller design using LQG control.

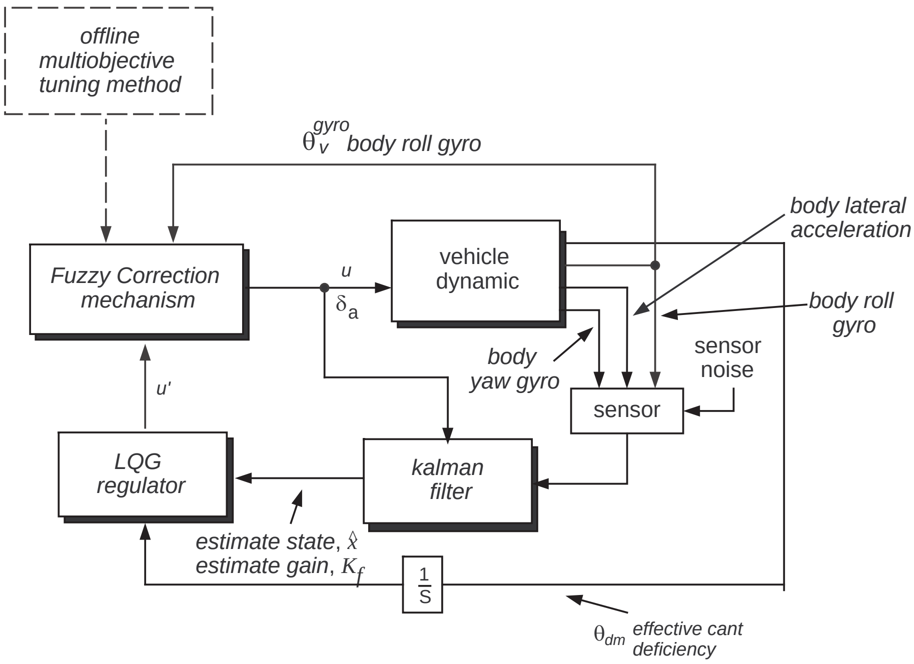


FIGURE 6. LQG controller scheme with fuzzy correction

Figure 6 shows the combination of the compensator (LQG) and fuzzy mechanism. The compensator consist of the regulator and Kalman filter estimator. This solution is based on the separation principle where the LQR and Kalman filter are designed independently and than combined to form the LQG compensator. An extra state, $\int \theta_{(e.c.d)}$ is include to ensure zero steady curved. The output regulator is then fed to the fuzzy correction mechanism to further improve the roll angle transition at curved track while minimize the effect of lateral straight track irregularities. A signal from the body roll gyro, θ_v^{gyro} is also fed to the fuzzy correction mechanism as additional decision-making variable. The design of the controller can be divided by into two stages:

1. design LQG controller with integral action
2. tune fuzzy correction mechanism using genetic algorithm optimization method aiming to further improve the transition on curved track while reducing the effect of lateral movement during straight track.

4.2.1. *LQR controller design.* Referring to states Equation (3) and Figure 6, the state model is augmented with the integral of the *effective cant deficiency* ($\int \theta_{dm}$) to enhance the controller for regulation purposes, i.e., achieve required tilt compensation with zero steady state errors. This gives an optimal PI controller [2]. The overall system model with $\int \theta_{dm}$ is,

$$\begin{bmatrix} \dot{x}' \\ \dot{x} \end{bmatrix} = \begin{bmatrix} C' & 0 \\ A & 0 \end{bmatrix} \begin{bmatrix} x' \\ x \end{bmatrix} + \begin{bmatrix} 0 \\ B \end{bmatrix} u \tag{9}$$

where $x' = \int \theta_{dm}$ and C' is the selector matrix for the output of *effective cant deficiency*, θ_{dm}. The control law is given by

$$u(t) = -K_I x'(t) - K_P x(t) \tag{10}$$

where the initial condition of integral is set to zero and the quadratic performance index for state regulator is given by

$$J = \int_0^\infty \left[\bar{x}(t)^T Q \bar{x}(t) + u^T(t) R u(t) \right] dt \tag{11}$$

where

$$\bar{x} = \left[\overbrace{\int \theta_{dm}}^{x'} \; \overbrace{y_v \; \theta_v \; y_b \; \theta_b \; \dot{y}_v \; \dot{\theta}_v \; \dot{y}_b \; \dot{\theta}_b \; \theta_r}^{x} \right]^T \tag{12}$$

and u is angle of the tilt actuator δ_a, while matrix Q is chosen as 10×10 diagonal matrix and R is the control weight matrix.

4.3. **Kalman-Bucy filter (observer) design.** Ideally, the original state space expression can be used for the design of the Kalman filter. However, to provide an accurate estimation model, the elements of curved track state should be included into the state vector rather than in the disturbance vector. Therefore, the extended model is given by

$$\dot{x}_k = A_k x_k + B_k u + \Gamma_k \omega_k \tag{13}$$

where

$$x_k = \begin{bmatrix} x & \tilde{w} \end{bmatrix}^T \tag{14}$$

and the output equation is given by

$$y_k = C_k x_k + D_k u + v \tag{15}$$

where v is sensor noise corruption and C_k, D_k are based on the relative rows of A_k and B_k. The reformulated state x_k becomes $[\, y_v \; \theta_v \; y_b \; \theta_b \; \dot{y}_v \; \dot{\theta}_v \; \dot{y}_b \; \dot{\theta}_b \; \theta_r \; \theta_o \; \dot{\theta}_o \; R^{-1}]^T$, where $\theta_o \; \dot{\theta}_o \; R^{-1}$ are the deterministic track states. While $w_k = \begin{bmatrix} R^{-1} & \ddot{\theta}_o \end{bmatrix}^T$.

The state estimate from the Kalman-Bucy filter can be calculated by the following differential equation,

$$\dot{x}_e = A_k x_e + B_k u + K_f (Y_k - C_k x_e D_k u) \tag{16}$$

where x_e is the state estimate vector of the re-formulated state and K_f is the Kalman filter gain. The performance of the Kalman-Bucy filter was tuned based on the variance matrix of noise track Q_{kf} where $Q_{kf} = \operatorname{diag}\left(Q^{kf}_{\ddot{\theta}_o}, Q^{kf}_{\frac{1}{R}} \right) = \operatorname{diag}(10^{-6}, 0.85^{-3})$. Figure 7 shows the passive system responses between actual and estimate output using the Kalman-Bucy filter.

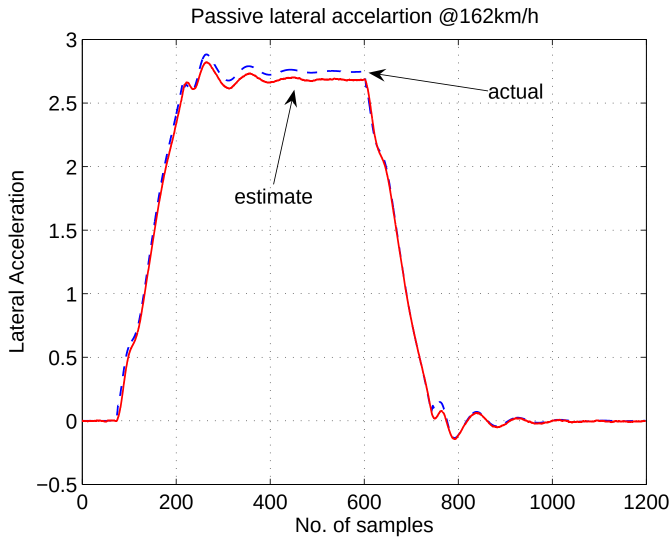


FIGURE 7. Kalman-Bucy filter estimate for passive lateral acceleration output responses

4.4. **Optimization and tuning process.** The optimisation process was done in the MATLAB® environment while the objective function used as mention in Section 4.1. The NSGA II method run for 350 iterations with 30 potential solutions at each iteration. The output responses for the body lateral accelerations and body roll gyro at the final stage of the optimisation process can be seen in Figure 8(a) which shows that the NSGA II gives multiple solutions with a diversity to each. The trade-off between the objectives can be easily seen in Figure 8(b), which indicates the difficulty in satisfying each objective.

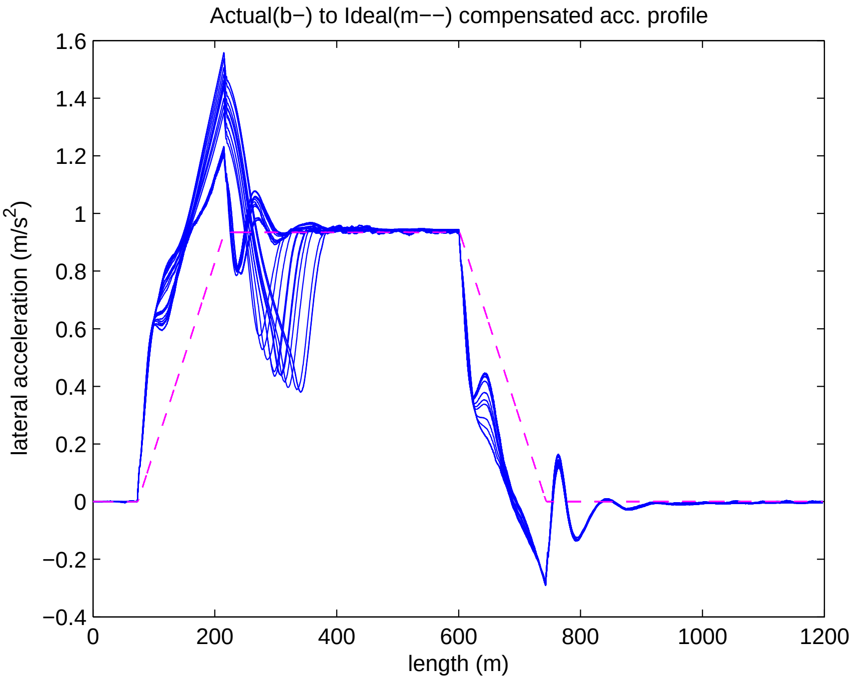
The solution was chosen by observing the responses for both the stochastic and deterministic outputs. By referring to Figure 8(a), the selection give a fast response on curved track while reducing the stochastic track irregularities to the system. Please note: the selection is also must considered the anti-tilt responses in body lateral acceleration in which influence the dynamic effect of the overall responses.

4.5. **System responses and analysis.** Figure 9 shows the overall output responses for both controllers. The figure also present a comparison between conventional (noted as conv. in the figures) and the controllers with fuzzy mechanism. In addition, Figures 9(a) and 9(c) show the body lateral acceleration responses. The figure clearly show the improvement on the output response due to the effect of the proposed fuzzy 'add-on' mechanism. The advantage of the fuzzy 'add-on' mechanism can be clearly seen in the body roll responses as well, i.e., refer to Figures 9(b) and 9(d).

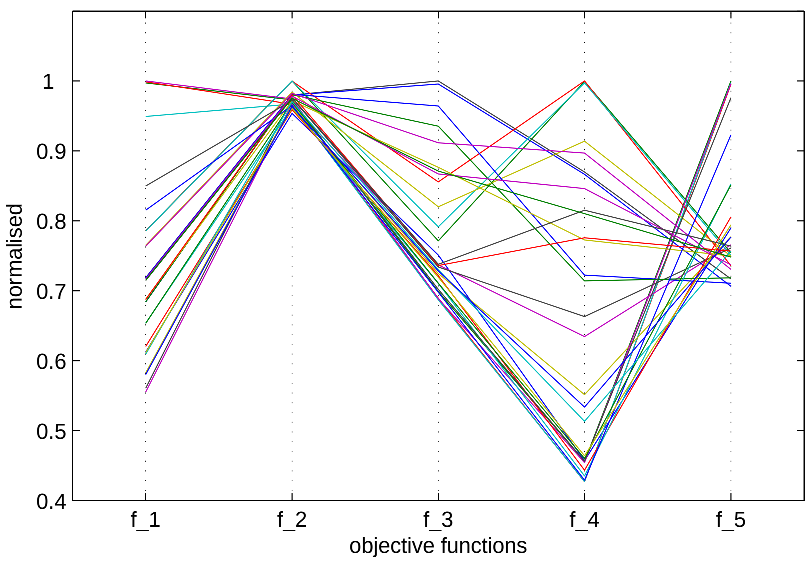
The performance of the controller design was analysed based on both stochastic and deterministic performance via the assessment method proposed in Section 3, taking account the lateral/roll dynamic of the system, and it is summarised in Table 2. The table presents the comparison between conventional controller (using PID and LQG) and conventional controller with fuzzy mechanism. It is clearly seen that the proposed schemes perform rather closely to the current precedence type approach, and in some cases the ride quality is further improved. Note that in the precedence scheme the filter delay should be designed appropriately because any correlation in the drive signals may have a detrimental effect in ride quality, i.e., on straight track.

5. **Conclusion.** The potential of using the fuzzy correction mechanism was investigated by integrating it with conventional classical and optimal control. The use of a conventional controller (in this case a PID and LQG controller) in series with fuzzy correction mechanism clearly shows improvement on both deterministic and stochastic performance.

However, the performance can be further improve by including more decision rules in the knowledge base, which of course will add another function to the input and output membership functions. This will give more precise decision which could provide a means of extra stability and robustness to the system.

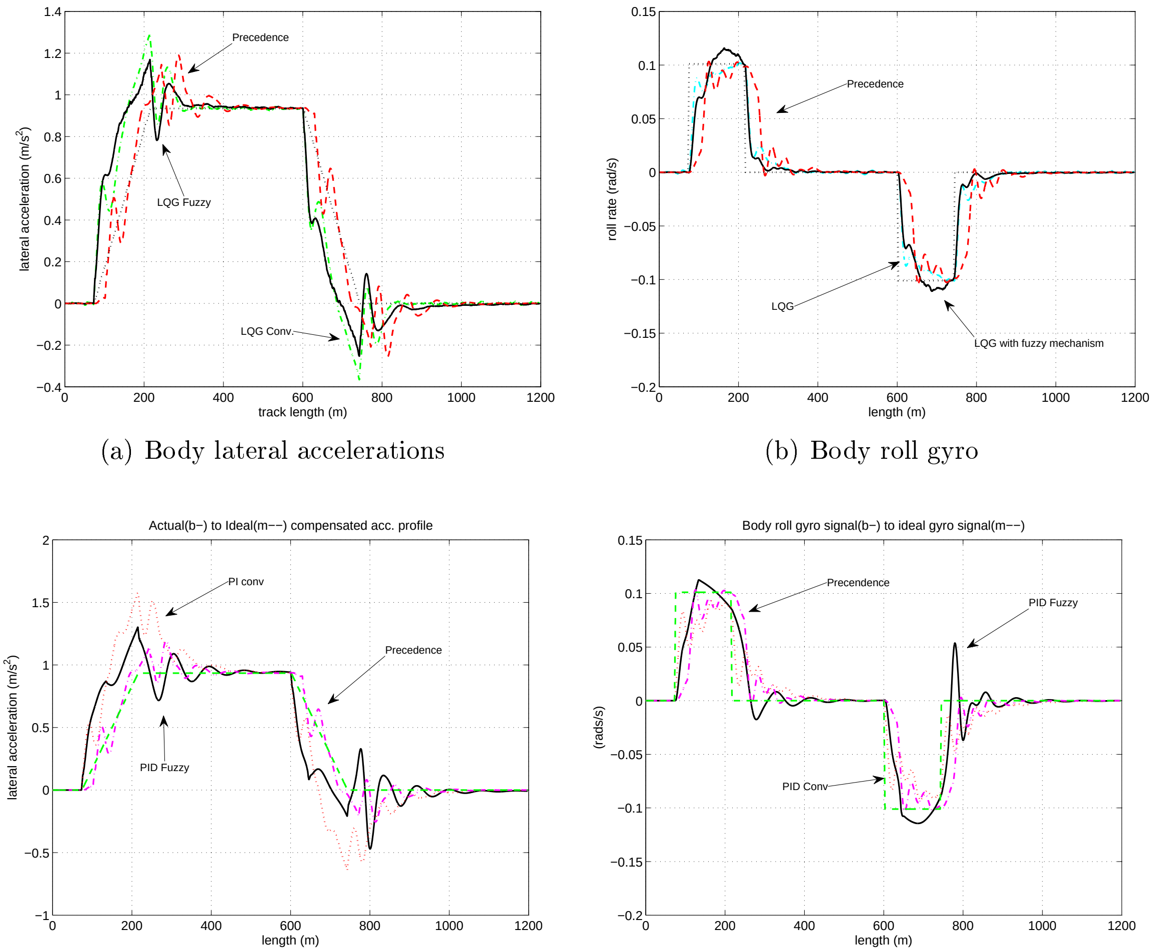


(a) Body lateral acceleration responses



(b) Trade-off between objectives at 350 generations

FIGURE 8. tuning and trade off using NSGA II



(a) Body lateral accelerations (b) Body roll gyro

(c) Body lateral accelerations output responses (d) Body roll gyro

FIGURE 9. Comparison between fuzzy mechanism, conventional controller and precedence tilt control strategies

The use of a multi-objective genetic algorithm gives advantages by providing a multiple solution to a problem. The use of GAs optimisation not only gives an extra degree of freedom in searching space, but more importantly, avoided the need for a trial and error approach to manual tuning which in general introduces a substantial constrain in finding an optimum solution. Further work relates to optimization of the input membership functions, in the tuning process (expecting to have a positive effect in the transient performance), as well as further study in stability and robustness of the designed system.

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REFERENCES

[1] E. P. Castillo and O. Melin, Intelligent systems with interval type-2 fuzzy logic, *International Journal of Innovative Computing, Information and Control*, vol.4, no.4, pp.771-783, 2008.
[2] P. Dorato, C. Abdallah and V. Cerone, *Linear-Quadratic Control: An Introduction*, Prentice-Hall, Englewood Cliffs, NJ, 1995.

TABLE 2. Comparison on ride quality using fuzzy mechanism in PID and LQG controller

	Deterministic	Current Tech.	LQG Conv.	LQG Fuzzy correction	PID conv	PID Fuzzy
Lat. accel.	steady state (%g)	9.5	9.5	9.5	9.5	9.5
	R.M.S deviation error (%g)	1.52	2.23	2.090	3.463	3.2
	peak value (%g)	12.14	13.106	12.468	16.07	13.3
Roll gyro	R.M.S jerk level (rad/s)	0.02	0.022	0.024	0.025	0.03
	peak value (rad/s)	0.103	0.110	0.097	0.09	0.110
P_{ct}	peak value level (%g/s)	6.72	6.527	7.418	9.158	8.6
	standing (% of passengers)	47.275	49.533	51.307	60.905	56.70
	seated (% of passengers)	13.334	13.993	14.524	18.764	16.5
	Stochastic					
Ride qual.	passive (%g)	0.381	0.381	0.381	0.381	0.381
	active tilt (%g)	0.388	0.383	0.364	0.536	0.388
	Ride qual. degradation (%)	–12.143	1.105	–4.398	39.00	1.02

[3] R. M. Goodall, Tilting trains and beyond – The future for active railway suspensions: Part 1. Improving passenger comfort, *Computing and Control Engineering Journal*, vol.10, no.4, pp.153-160, 1990.

[4] W.-G. Ma, Design of signal fuzzy controller of single intersection in intelligence transportation, *International Journal of Innovative Computing, Information and Control*, vol.3, no.4, pp.1023-1029, 2007.

[5] M. Miyamoto and Y. Suda, Recent research and development on advanced technologies of high-speed railways in Japan, *Vehicle System Dynamic*, vol.40, no.1-3, pp.55-99, 2003.

[6] J. T. Pearson and R. M. Goodall, Control system studies of an active anti-roll bar tilt system for railway vehicles, *Proc. of Institute Mechanical Engeenring (Part F)*, vol.212, pp.43-60, 1998.

[7] K. Sasaki, Position detection system using GPS for carbody tilt control, *QR of Railway Technical Research Institute*, vol.46, no.2, pp.73-77, 2005.

[8] H. Zamzuri, A. C. Zolotas and R. M. Goodall, Intelligent control approaches for tilting railway vehicles, *Vehicle System Dynamics*, vol.44(S1), pp.834-842, 2006.

[9] H. Zamzuri, A. C. Zolotas and R. M. Goodall, Tilt control design for high-speed trains: A study on multi-objective tuning approaches, *Vehicle System Dynamics*, vol.46, pp.535-547, 2008.

[10] A. C. Zolotas, R. M. Goodall and J. Evans, Assesment of the performance of tilt system controllers, *The Railway Conference at Railtex 2000*, UK, pp.21-23, 2000.

[11] A. C. Zolotas, R. M. Goodall and G. D. Halikias, New control strategies for tilting trains, *Vehicle System Dynamics*, vol.37, pp.171-182, 2003.

[12] A. C. Zolotas, G. D. Halikas and R. M. Goodall, A comparison of tilt control approaches for high speed railway vehicles, *Proc. of ICSE*, Conventry, UK, vol.2, pp.632-636, 2000.

[13] A. C. Zolotas and R. M. Goodall, Improving the tilt control performance of high-speed railway vehicles: An LQG approach, *Proc. of the 16th IFAC World Congress*, Prague, 2005.

[14] A. C. Zolotas, R. M. Goodall and G. D. Halikias, Recent results in tilt control design and assessment of high-speed railway vehicles, *Proc. of the Institution of Mechanical Engineers, Part F: Journal of Rail and Rapid Transit*, vol.221, no.2, pp.291, 2007.