Buletin Teknik Elektro dan Informatika
Vol. x, No. x, 2018, pp. 1 ∼ 12
ISSN: 2089-3191

Newly Developed Nonlinear Vehicle Model for an Active Anti-roll Bar System

Noraishikin Zulkarnain[1*,2], Hairi Zamzuri[4], Sarah 'Atifah Saruchi[4], Mohd Marzuki Mustafa [1], Nurbaiti Wahid[4,5], and Alias Jedi[2,3]
[1]Centre for Integrated Systems Engineering and Advanced Technologies (Integra)
[2]Centre of Research in Engineering Education and Built Environment (PEKA)
[3]Centre of Integrated Design for Advanced Mechanical Design (PRISMA, Faculty of Engineering and Built Environment, Universiti Kebangsaan Malaysia,43600 Bangi, Selangor
[4]Vehicle System Engineering, Malaysia-Japan International Institute of Technology, Universiti Teknologi Malaysia, 54100 Kuala Lumpur, Malaysia
[5]Faculty of Electrical Engineering, Universiti Teknologi MARA, Universiti Teknologi MARA, 23000 Dungun, Terengganu, Malaysia
[*]corresponding author, e-mail: shikinzulkarnain@ukm.edu.my

Abstract

This paper presents the development of a simplified linear vehicle model consists of half car modelling with single track bicycle model, verification and validation of the model. As for the verification of simplified vehicle model, a newly developed nonlinear vehicle model is used in the validation process of the vehicle model. The parameters chosen in a newly developed vehicle model is developed based on Carsim vehicle model by using non-dominated sorting genetic algorithm version II (NSGA-II) optimization method. The ride comfort and handling performances have been one of the main objective in order to fulfil the expectation of customers in the vehicle development. Full nonlinear vehicle model which consists of ride, handling and Magic tyre subsystems has been derived and developed in MATLAB/Simulink. Then, optimum values of the full nonlinear vehicle parameters are investigated by using NSGA-II. The two objective functions are established based on RMS error between simulation and benchmark system. A stiffer suspension provides good stability and handling during manoeuvres while softer suspension give better ride quality. The final results indicated that the newly developed nonlinear vehicle model is behaving accurately with input ride and manoeuvre. The outputs trend are successfully replicated.

Keywords: *vehicle model, NSGA-II,optimization, anti-roll bar, ride and handling*

1. Introduction

Vehicle suspension system is used to keep apart the vehicle from uncomfortable vibrations transmitted from the road excitation on the tyres and the driver. During a manoeuvre, it can keep the vehicle under control as it transmits the control forces back to the tyres [1], [2], [3], [4], [5]. The ride comfort and handling performances have been the major development objectives of vehicles in order to fulfil the expectation of customers in the development of mechanical and electronics vehicle technology. Thus, designing a suitable suspension system is always an important research issue for attaining the desired vehicle quality. Currently, there are so many researchers in active suspension system field around the world that tackle these issues. However, some of them prefer to use active suspension others prefer to focus on the anti-roll bar itself.

Currently, an automotive engineer encounter many challenges in trying to have a good ride and handing whilst at the same time in cornering or straight line. Thus, this research proposed to investigate a new control design in order to provide the required active roll control of the anti-roll bar system for a ground vehicle. At the same time, to be able to find an improvement in handling capabilities without sacrificing the ride comfort.

Due to the growing market share of light duty vehicle, automotive engineers have been prompted to examine body roll minimization strategies. During driving manoeuvres in high centre of gravity vehicles due to its roll-over, dangerous operating scenarios may induce drivers to drive in an aggressive way such as high lateral acceleration, rapid tire dilation and emergence lane change in [6]. Commercially, to counteract the roll movement, the most used topology is by using

Received May 9, 201x; Revised August 3, 201x; Accepted August 16, 201x

anti-roll bar. This bar can be implemented in a passive as well in active topology , where some of the most important parameters are again performance, energy consumption and costs [7].

From the previous researches, most researchers used genetic algorithm to optimise the system parameter such as routing in networks, job shop scheduling problem and isothermal liquid-phase kinetic sequence [8], [9], [10]. Moreover, the Non-dominated Sorting Genetic Algorithm II(NSGA-II) is proving to be a robust optimization algorithm for a wide range of multi-objective problems. The NSGA II Pareto ranking algorithm is an elitist system and maintains an external archive of the Pareto solutions [11], [12], [8], [13], [14], [15].

Trade-off analysis is the one of the principles of active anti-roll bar system. It can help decision makers achieve the ride and handling performance requirements at the same time. However, the trade-off analysis in an active ant-roll bar system for tilt control design using NSGA-II are yet available in the literature [16]. Therefore, in this paper, a general methodology is proposed for conducting trade-off analysis for the selection problem of multi-objective of ride and handling for ground vehicle. A general trade-off based on multi-objective optimization framework for an active anti-roll bar system by optimising the novel control strategy parameters are established in this research [17], [18].

Therefore, in this paper the newly developed nonlinear vehicle model for an active anti-roll bar system has been used NSGA-II optimization method for the future used of the controller design. This paper is organized as follows. In the next section, the full nonlinear vehicle model and vehicle parameters are presented. Then, the following section describes the the newly developed nonlinear vehicle model. The fourth section shows the verification of newly developed vehicle model with Carsim Model. Finally, the conclusion and future works are being concluded and suggested in the last section.

2. Simplified Model

The simplified model consists of the front view of a half car model and the single linear bicycle model as shown in Figures 1 and 2. Four degrees of freedom (DOF) are included to model an independent suspension instead of three DOF commonly used in the literature where the right and left unsprung masses are mode led by one axle only. The model consists of vertical, lateral, yaw and roll motion. The simplified vehicle suspension model is based on a half car model from front view and a single track model with roll dynamics developed from the bicycle model.

2.1. Half Car Model

The front view of a half car model is illustrated in Figure 1. The half car model explains the relationship between body bounce, body roll angle, left and right wheels hop and road excitations.

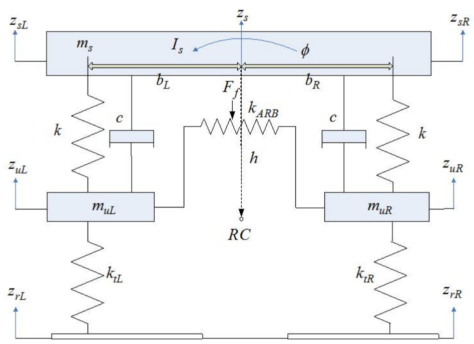


Figure 1 Half car model of a vehicle from front view

Then, the equations of motions for this model are combined with the single track model with roll dynamics to design the four degree of freedoms vehicle dynamic model as follow:
Body vertical acceleration:

$$\ddot{z}_s = -\frac{2c\dot{z}_s}{m_s} + \frac{c\dot{z}_{uL}}{m_s} + \frac{c\dot{z}_{uR}}{m_s} - \frac{c(b_L - b_R)}{m_s}\dot{\phi} - \frac{2kz_s}{m_s} + \frac{kz_{uL}}{m_s} + \frac{kz_{uR}}{m_s} - \frac{k(b_L - b_R)}{m_s}\phi \tag{1}$$

Left wheel vertical acceleration:

$$\ddot{z}_{uL} = \frac{kz_s}{m_{uL}} - \frac{(k + k_{tL})}{m_{uL}}z_{uL} + \frac{c\dot{z}_s}{m_{uL}} - \frac{c\dot{z}_{uL}}{m_{uL}} + \frac{kb_L}{m_{uL}}\phi + \frac{cb_L}{m_{uL}}\dot{\phi} + \frac{k_{tL}}{m_{uL}}z_{rL} + F_L \tag{2}$$

Right wheel vertical acceleration:

$$\ddot{z}_{uR} = \frac{kz_s}{m_{uR}} - \frac{(k + k_{tR})}{m_{uR}}z_{uR} + \frac{c\dot{z}_s}{m_{uR}} - \frac{c\dot{z}_{uR}}{m_{uR}} - \frac{kb_R}{m_{uR}}\phi - \frac{cb_R}{m_{uR}}\dot{\phi} + \frac{k_{tR}}{m_{uR}}z_{rR} - F_R \tag{3}$$

Roll acceleration of half car model:

$$\ddot{\phi} = -\frac{(b_L - b_R)kz_s}{I_s} + \frac{(b_L k z_{uL})}{I_s} - \frac{(b_R k z_{uR})}{I_s} - \frac{c(b_L - b_R)\dot{z}_s}{I_s} + \frac{b_L c\dot{z}_{uL}}{I_s} - \frac{b_R c\dot{z}_{uR}}{I_s} - \frac{(kb_L^2 + kb_R^2)}{I_s}\phi - \frac{c(b_L^2 + b_R^2)}{I_s}\dot{\phi} + eF_f \tag{4}$$

2.2. The Linear Single Track Model with Roll Dynamics

A linear single track model is the simplest vehicle model used in control research area. This model is also known as the bicycle model, which is obtained by approximating the front and rear pairs of wheels as single wheels [19]. Roll dynamics and non-constant longitudinal velocity can be incorporated into a single-track model. The model is illustrated in Figure 2.

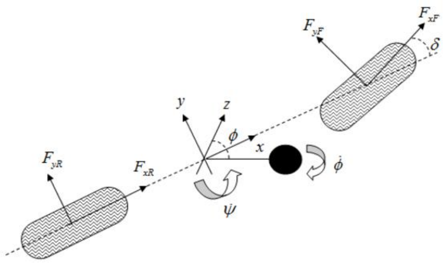


Figure 2 Single-track model, showing the combined front and rear tyre forces, the steering angle , the yaw rate, and the vehicle sideslip angle

Assuming that the steering angle is small, the equations of motion are given by [20]. The equations of motion are augmented by a torque balance around the x axis. The equations are:: Lateral motion:

$$m(\dot{V}_y + \dot{V}_x\dot{\psi}) \quad = \quad F_{yF} + F_{yR} \tag{5}$$

where, $\dot{V}_y$ is lateral acceleration, $\dot{V}_x$ is longitudinal acceleration, F_{yF} is front lateral tyre force, F_{yR} is rear lateral tyre force and $\dot{\psi}$ is yaw rate.

Yaw motion:

$$I_{zz}\ddot{\psi} = aF_{yF} - bF_{yR} \tag{6}$$

$$I_s\ddot{\phi} + C_\phi\dot{\phi} + K_\phi\phi = mh(\dot{V}_y + V_x\dot{\phi}) \tag{7}$$

where,

$$F_{yF} \approx C_f\alpha_F \tag{8}$$

$$F_{yR} \approx C_r\alpha_R \tag{9}$$

(10)

Where F_{yF} and F_{yR} are the combined front and rear lateral tyre forces, C_f is the front cornering stiffness and C_r is the rear cornering stiffness, I_{zz} is the moment of inertia around the z-axis, a and b are the distances from the front and rear wheels to the centre of gravity, C_ϕ and K_ϕ are roll damping and spring coefficients respectively and $\dot{\phi}$ is the roll rate. The slip angles of the front and rear wheels α_F and α_R can be approximated as

$$\alpha_F \approx \delta - \frac{1}{V_x}(V_y + \alpha\dot{\psi}) \tag{11}$$

$$\alpha_R \approx -\frac{1}{V_x}(V_y - b\dot{\psi}) \tag{12}$$

The input to the system is steering angle, δ , while the outputs of the system are lateral velocity, V_y , yaw rate, $\dot{\psi}$, and roll rate, $\dot{\phi}$. The equations single track model with roll dynamics system is written as follows:
Lateral acceleration:

$$\dot{V}_y = \frac{C_f}{m}[\delta - \frac{1}{V_x}(V_y + a\dot{\psi})] + \frac{C_r}{m}[-\frac{1}{V_x}(V_y - b\dot{\psi})] - \dot{V}_x\dot{\psi} \tag{13}$$

Yaw acceleration:

$$\ddot{\psi} = \frac{aC_f}{I_{zz}}[\delta - \frac{1}{V_x}(V_y + a\dot{\psi})] - \frac{bC_r}{I_{zz}}[-\frac{1}{V_x}(V_y - b\dot{\psi})] \tag{14}$$

Roll acceleration of single track model:

$$\ddot{\phi} = \frac{mh}{I_s}(\dot{V}_y + V_x\dot{\psi}) - \frac{C_\phi\dot{\phi}}{I_s} - \frac{K_\phi\phi}{I_s} \tag{15}$$

3. Full Nonlinear Vehicle Model

A full nonlinear vehicle model described by using mathematical equations was developed as a MATLAB/Simulink model and was validated by comparing results from the double lane change test with speed 70 km/h. The steering input was used as the input to the simulation steering wheel angle and divided by a gain to represent the steer angle to the wheel in the vehicle model. The ride test was performed with CARSIM's input which are roll bump signal, pitch bump signal and combination pitch and roll bump signal with 20 km/h. The whole vehicle model consists of five main sub-models i.e. ride model, tyre model, handling model, side slip angle model and longitudinal slip ratio model as illustrated in Figure 3.

4. Newly Developed Nonlinear Vehicle Model

A new methodology is proposed to optimise the parameters of full nonlinear vehicle model based on ride and handling performances. The objective is to find the optimum values of the full

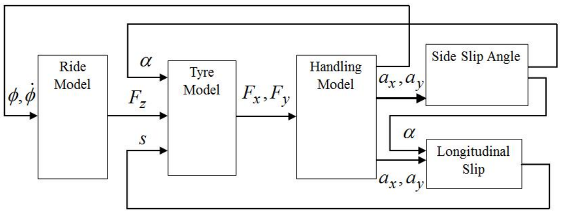


Figure 3 Full vehicle model

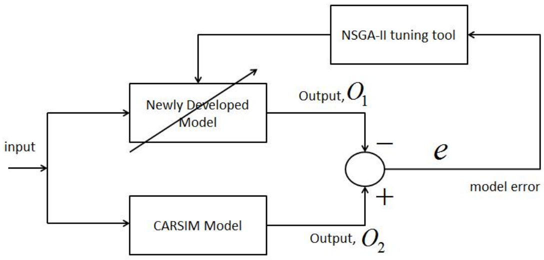


Figure 4 Newly developed vehicle model block diagram configuration

nonlinear vehicle parameters. The block diagram configuration of methodology on development of nonlinear vehicle model is illustrated in Figure 4.

The two objective functions are established based on RMS error between simulation and benchmark system. This is done by optimizing the nonlinear vehicle parameters by comparing with output response based on commercial available vehicle dynamic software (CARSIM). The vehicle dynamic software is used as a benchmark in order to provide similar results in vehicle model. Newly developed nonlinear vehicle model must fulfil some conflicting area. Among those is ride comfort which is attained by minimizing the error of body roll angle in speed bump test. While, a good handling of a vehicle is a desirable property which is achieved by minimizing the error of body roll angle in steering test.

Objective 1 in the following form, which is consisted of RMS error value of the vehicle body roll angle for the handling performance means when the test input is handled in steering part as described in Equation 16. A small value of f_1 indicate good ride performance. Objective function 2 is built also using the mathematical formula of RMS error. RMS error value of the vehicle body roll angle for the ride performance means when the test input is handled in speed bump test as declared in Equation 18. A small value of f_2 indicate good ride performance.

The objective functions considered are minimization of (i)R.M.S error value of the vehicle body roll angle, which is for the handling performance on steering input test and (ii)R.M.S error value of the vehicle body roll angle, which is for the ride quality on the Speed Bump test. The following relations express these two objectives:

(i.) $Obj1$, f_1 = R.M.S error value of the vehicle body roll angle, which is for the handling performance on steering input test (Double Lane Change)

$$e_{rms1} = \sqrt{\frac{1}{T}\int_0^T O_1^2(t)\,dt} - \sqrt{\frac{1}{T}\int_0^T O_2^2(t)\,dt} \tag{16}$$

$$f_1 = |e_{rms1}| \tag{17}$$

where T is the transition time and the constraint function,

$$w_1 = \begin{cases} f_1 & if \quad f_1 \geq 0 \\ 0 & . \quad Otherwise \end{cases}$$

(ii.) $Obj2$, f_2 = R.M.S error value of the vehicle body roll angle, which is for the ride quality on the Speed Bump test

$$e_{rms2} = \sqrt{\frac{1}{T}\int_0^T O_1^2\,(t)\,dt} - \sqrt{\frac{1}{T}\int_0^T O_2^2\,(t)\,dt} \tag{18}$$

$$f_2 = |e_{rms2}| \tag{19}$$

where the constraint function is given by:

$$w_2 = \begin{cases} f_2 & if \quad f_2 \geq 0 \\ 0 & . \quad Otherwise \end{cases}$$

Therefore the objective function with constraint is given by:

$$F(x) \quad = \quad f_x + R_x w_x \tag{20}$$

Where R_x is the penalty parameter which is for R_1 equal to 0.5 and R_2 also is 0.5 respectively. The value of penalty function is equal for both objectives in order to get equal trade-off between both objectives. In order to get the optimal solutions of parameter design for a newly developed nonlinear vehicle model, the RMS error of roll angle value is chosen as an objective function. From literature, the main goal of using anti-roll bar is to reduce the body roll [21], [22]. For that reason only the roll angle output is considered in optimization process.

The newly developed nonlinear vehicle model be able to fine tuned by choosing the suitable value of parameters are listed in Table 1. The eleven parameters chosen to optimise are roll axis moment inertia, I_r, spring stiffness at the front right, K_{sfr}, spring stiffness at the front left, K_{sfl}, spring stiffness at the rear right, K_{srr}, spring stiffness at the rear left, K_{srl}, damping stiffness at the front right, C_{sfr}, damping stiffness at the front left, C_{sfl}, damping stiffness at the rear right, C_{srr}, damping stiffness at the rear left, C_{srl}, roll spring stiffness, K_ϕ and roll damping stiffness, C_ϕ. Table 1 shows the design variables, their bounds for newly developed nonlinear vehicle model and also initial design referring from work done by Hudha et. al. [23]. In this study, the value of I_r, K_{sfr}, K_{sfl}, K_{srr}, K_{srl}, C_{sfr}, C_{sfl}, C_{srr}, C_{srl}, C_ϕ and K_ϕ are observed as eleven design variables to be optimally found based on multi-objective optimization of two different objective functions [24]. The optimization problem is solved using Elitist Non-Dominated Sorting Genetic Algorithm (NSGA-II) [25], [26]. Since it is a multi-objective optimization, a number of optimal solutions have been obtained and it is shown in the Figure 5 in which every solution is non-dominated by other solutions.

These objective functions are considered in a Pareto optimization process to obtain some important trade-offs among the conflicting objectives, simultaneously. The evolutionary process of multi-objective optimization is accomplished with a population size of 30 which has been chosen with probability of crossover and probability of mutation as 0.8 and 0.1, respectively. These parameter setting of NSGA-II optimization method are shown in Table 2. A total number of 30 non-dominated optimum design points have been obtained. In order to analyse Pareto set and select good solutions, it is widely accepted that visualization tools are valuable to provide the decision maker in meaningful way. It is normally easy to make an accurate graphical analysis of the Pareto set points for a two dimensional problem but for higher dimensions it becomes more difficult [27], [28]. According to Pareto chart in Figure 5, a solution in central region of the trade-off is chosen for analysis. Therefore, the choice of parameters optimization for nonlinear model are based on the trade-off between both situations, and gives the result in the Table 3.

Table 1: Design variables and Lower-Upper bounds for newly developed model

Design Variable	lower-upper	Initial Design
I_r	100-1000	700
$K_{sfr}, K_{sfl}, K_{srr}, K_{srl}$	500 - 1000	750
$C_{sfr}, C_{sfl}, C_{srr}, C_{srl}$	20000 - 40000	30000
C_ϕ	100 - 4000	3495.7
K_ϕ	100 - 60000	56957

Table 2: NSGA-II user defined parameters

Parameter setting	Value
Number of generation	500
Population Size	30
Probability of Crossover	0.8
Probability of Mutation	0.1
Distribution index in SBX	20
Distribution index in polynomial mutation	20

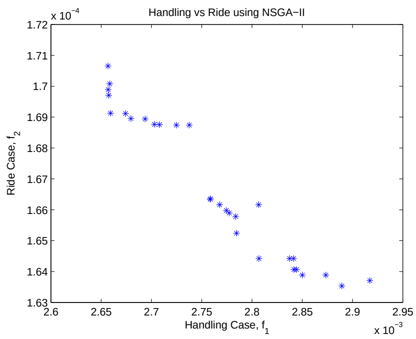


Figure 5 RMS error roll angle in handling vs RMS error roll angle in ride: optimal front

Table 3: The best parameter optimization of newly developed nonlinear model by NSGA-II

Parameter	Value	Parameter	Value
I_r	825	K_{sfr}	829
K_{sfl}	798	K_{srr}	813
K_{srl}	802	C_{sfr}	35019
C_{sfl}	34560	C_{srr}	35009
C_{srl}	36008	C_ϕ	3905
K_ϕ	57240		

4.1. Verification of Newly Developed Vehicle Model with Carsim Model

The newly developed nonlinear vehicle model using NSGA-II optimization method is verified with CarSimEd version 4.51, a well-known vehicle dynamics software in order to determine its output response effectiveness. The dynamics behaviour of a vehicle obtained from CarSimEd is used as a benchmark for the 14 DOF nonlinear vehicle model under the same input conditions. Therefore, a performance comparison with benchmark vehicle is required. As a benchmark for validation purpose, a vehicle dynamic analysis software, Carsim is used. The comparison between the full vehicle model with Carsim benchmark is done by comparing the vehicle output response trend.

This study is focused on both ride and handling parts of vehicle dynamic. Thus, the selected appropriate vehicle handling output responses are selected including roll angle and roll rate for validation is preferable. While, the chosen suitable vehicle ride output components also used roll angle and roll rate taken in consideration respectively. In the validation process, a comparison of the performance output trend of newly nonlinear vehicle model with Carsim benchmark vehicle requires proper handling and ride test procedure. As such, two types of analysis procedures are used for the validation purpose which are roll mode bump and double lane change manoeuvre.

4.1.1. Ride Analysis Result

As for the ride test analysis, a rolling mode test is used. For this, only one side of the vehicle hits the bump i.e. front left side and rear left side, to create rolling effect in the rolling mode test. The bumps were arranged in a way that it created the rolling effect. The dimension of the bumps used in this simulation is height at 0.075 m, weight at 0.11 m and length of 2.44 m. The vehicle was driven at a constant speed of 20 km/h. Figure 6 shows the illustrations of the test bumps. In the roll mode test, only the left side of the vehicle will hop on and hop down from the bump in order to model this condition, the front left road input Z_{rfl} and the rear left road input Z_{rrl} be given that the road input for the right hand side will be considered as zero. Figure 7 shows the simulation and the benchmark results for this mode indicating good correlation in terms of similar trends.

The body roll angle of Figure 7 shows that the peak magnitudes from the full nonlinear vehicle model is almost identical with the Carsim model response. The peak magnitudes at t = 5.396 s is about 0.1676 m/s^2 and t = 5.851 s is about 0.1564 m/s^2 when the front left tyre hit the bump and the peak magnitudes at about 7.585 s is about 0.1529 m/s^2 when the rear left tyre hit the bump. The roll rate graph that is generated from the simulation of full nonlinear vehicle model with Carsim model results show a good correlation. The data from 0 to 5 second can be ignored as the vehicle is accelerating in order to achieve a constant speed 20 km/h before hitting the bump.

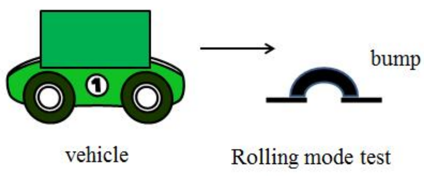
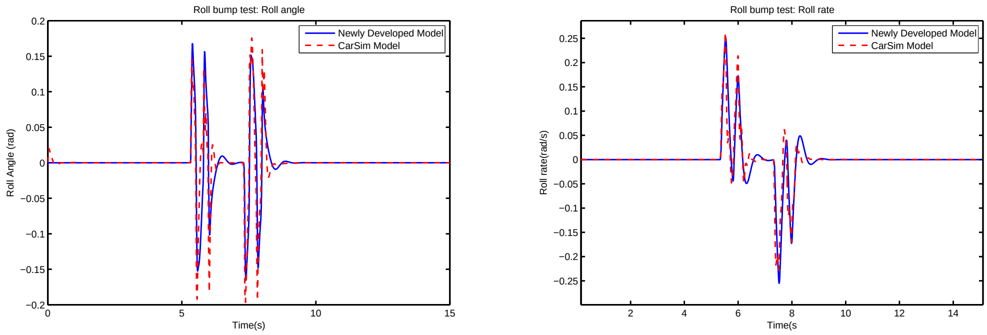


Figure 6 Rolling mode test

4.1.2. Handling Analysis Result

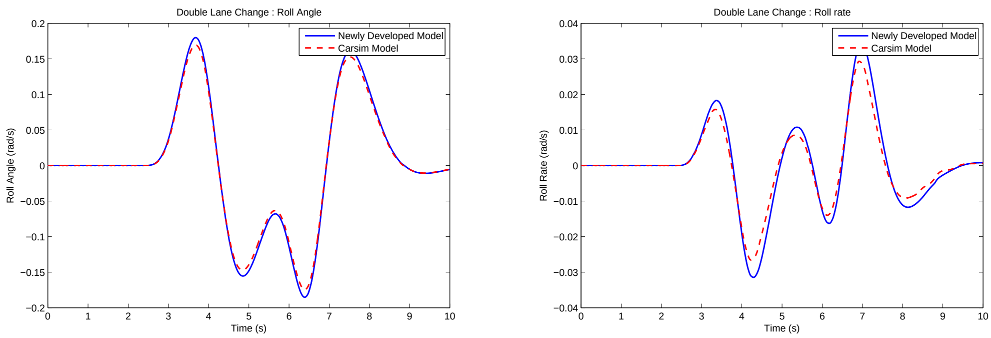
Next, in the handling analysis, a double lane change manoeuvre test is used. As for the double lane change test, it is taken from Carsim where the steering wheel angle input for Carsim model is exported to newly developed nonlinear vehicle model, thus both models were operated



(a) (b)

Figure 7 Roll bump test (a) roll angle response; (b) roll rate response

with the similar input condition. This manoeuvre test is operated in speed condition at 70 km/h. The full vehicle model performance with speed of 70 km/h is shown in Figure 8. The newly developed nonlinear vehicle model outputs trend are successfully replicated. However, there are some error in magnitude for roll angle and roll rate. This error in magnitude occurs because of nonlinearity properties of tyre model. The results indicated that the newly developed nonlinear vehicle model is behaving accurately with input manoeuvre.

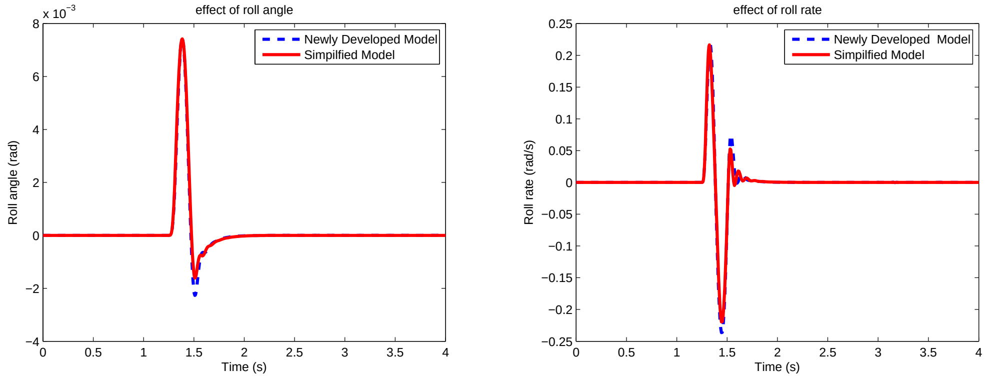


(a) (b)

Figure 8 Double lane change (a) roll angle response; (b) (c) roll rate response

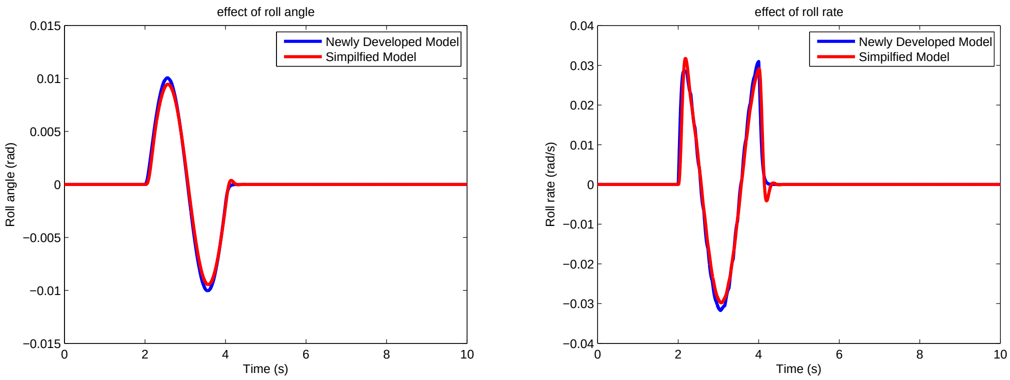
5. Verification of Simplified Model

The 4 DOF half car model including roll motion is verified with 14 DOF full nonlinear vehicle model developed by Kadir et al. [29] that has been verified with a well-known vehicle dynamics software for ride and handling tests. In this research, a similar model has been modified by including the function of an active anti-roll bar. From Figure 9, the dynamic responses consist of roll angle and roll rate for half car model are verified with full nonlinear vehicle model by using Speed Bump test. While, by using Single Lane Change test, the results of the roll angle and roll rate for half car model are verified with full nonlinear vehicle model has shown in Figure 10. From the results shown in figures, the trend of dynamic responses for simplified vehicle model as compared with a full nonlinear vehicle model are similar and quite exchangeable for the magnitudes of this two models. All the simulation parameters used for simplified model are similar with the parameters of full nonlinear vehicle model. Therefore, the simplified model is verified and can be used in this study.



(a) (b)

Figure 9 The dynamic responses verified using ride test (a) roll angle response; (b) roll rate response



(a) (b)

Figure 10 The dynamic responses verified using handling test (a) roll angle response; (b) roll rate response

6. Conclusion

A newly developed nonlinear vehicle model for an active anti-roll bar system using NSGA-II optimization method is successfully presented in this paper. This methodology is proposed to optimise the parameters of full nonlinear vehicle model based on ride and handling performances. The newly developed nonlinear vehicle model using NSGA-II optimization method has been verified with CarSimEd version 4.51 that is a well-known vehicle dynamics software in order to determine its output response effectiveness. In the validation process, a comparison of the performance output trend of newly nonlinear vehicle model with Carsim benchmark vehicle requires proper handling and ride test procedure. As such, two types of analysis procedures are used for the validation purpose which are roll mode bump and double lane change manoeuvre. The roll rate graph that is generated from the simulation of full nonlinear vehicle model with Carsim model results show a good correlation in ride analysis part. However, there are some error in magnitude for roll angle and roll rate in handling analysis performance. This error in magnitude occurs because of nonlinearity properties of tyre model. For the conclusion, the results indicated that the newly developed nonlinear vehicle model is behaving accurately with input manoeuvre.

Acknowledgement

The work presented in this study is funded by Universiti Kebangsaan Malaysia under GGPM Grant (vote no: GGPM-2017-087).

References

[1] P. H. Cronjé and P. S. Els, "Improving off-road vehicle handling using an active anti-roll bar," *Journal of Terramechanics*, vol. 47, no. 3, pp. 179–189, June 2010.

[2] Z. H. Saruchi, Sarah 'Atifah, N. Zulkarnain, N. Wahid, M. H. M. Ariff *et al.*, "Composite nonlinear feedback with disturbance observer for active front steering," *Indonesian Journal of Electrical Engineering and Computer Science*, vol. 7, no. 2, pp. 434–441, 2017.

[3] R. Wiryadinata, A. S. Nurliany, I. Muttakin, and T. Firmansyah, "Design of a low cost remotly operated vehicle with 3 dof navigation," *Bulletin of Electrical Engineering and Informatics*, vol. 6, no. 1, pp. 13–23, 2017.

[4] N. Zulkarnain, H. Zamzuri, Y. Sam, S. Mazlan, and S. Zainal, "Improving vehicle ride and handling using lqg cnf fusion control strategy for an active antiroll bar system," in *Abstract and Applied Analysis*, vol. 2014. Hindawi Publishing Corporation, 2014.

[5] S. A. Saruchi, H. Zamzuri, S. A. Mazlan, S. M. H. Fahami, and N. Zulkarnain, "Wheel synchronization control in steer-by-wire using composite nonlinear feedback," in *Applied Mechanics and Materials*, vol. 575. Trans Tech Publication, 2014, pp. 762–765.

[6] D. Cimba, J. Wagner, and A. Baviskar, "Investigation of active torsion bar actuator configurations to reduce vehicle body roll," *Vehicle System Dynamics*, vol. 44, no. 9, pp. 719–736, 2006.

[7] N. K. Cooperrider, T. M. Thomas, and S. A. Hammoud, "Testing and analysis of vehicle rollover behavior," SAE Technical Paper, Tech. Rep., 1990.

[8] M. Moza and S. Kumar, "Routing in networks using genetic algorithm," *Bulletin of Electrical Engineering and Informatics*, vol. 6, no. 1, pp. 88–98, 2017.

[9] S. Zamiri, S. Balochian, and H. Baloochian, "Multi objective optimization of multi component isothermal liquid-phase kinetic sequence using multivariable pi control," *Bulletin of Electrical Engineering and Informatics*, vol. 3, no. 4, pp. 277–284, 2014.

[10] S. Kalantari and M. SanieeAbadeh, "An effective multi-population based hybrid genetic algorithm for job shop scheduling problem," *Bulletin of Electrical Engineering and Informatics*, vol. 2, no. 1, pp. 59–64, 2013.

[11] K. Deb, A. Pratap, S. Agarwal, and T. Meyarivan, "A fast and elitist multiobjective genetic algorithm: Nsga-ii," *IEEE Transactions on Evolutionary Computation*, vol. 6, no. 2, pp. 182–197, 2002.

[12] D. Mokeddem and A. Khellaf, "Multicriteria optimization of multiproduct batch chemical process using genetic algorithm," *Journal of Food Process Engineering*, vol. 33, no. 6, pp. 979–991, 2010.

[13] A. W. Pratama, A. Buono, R. Hidayat, and H. Harsa, "Estimating parameter of nonlinear bias correction method using nsga-ii in daily precipitation data," *TELKOMNIKA*, vol. 16, no. 1, pp. 241–249, 2018.

[14] L. Zhang and Q. Wang, "A community detection algorithm based on nsga-ii," *TELKOMNIKA (Telecommunication Computing Electronics and Control)*, vol. 14, no. 3A, pp. 288–296, 2016.

[15] Y. Arkeman, A. Yusuf, M. Mushthofa, G. FitriLaxmi, and K. B. Seminar, "The formation of optimal portfolio of mutual shares funds using multi-objective genetic algorithm," *TELKOMNIKA (Telecommunication Computing Electronics and Control)*, vol. 11, no. 3, pp. 625–636, 2013.

[16] J. T. Pearson, R. M. Goodall, and I. Pratt, “Control system studies of an active anti-roll bar tilt system for railway vehicles,” *Proceedings of the Institution of Mechanical Engineers, Part F: Journal of Rail and Rapid Transit*, vol. 212, no. 1, pp. 43–60, Jan. 1998.

[17] Q. Bai, “Trade-off analysis in multiobjective optimization for transportation asset management,” Ph.D. dissertation, Purdue University, 2012.

[18] T. Shi, S. Chen, D. Wang, and J. Chen, “Performance optimization of vehicle handling and stability using multiple surrogate models,” *Journal of Testing and Evaluation*, vol. 44, no. 6, pp. 2100–2113, 2015.

[19] M. Hiemer, “Model based detection and reconstruction of road traffic accidents,” Ph.D. dissertation, University of Karlsruhe, Germany, 2004.

[20] H. B. Pacejka, *Tyre and Vehice Dynamics*. Butterworth Heine-mann, 2002.

[21] K. Caliskan, “Automated design analysis of anti-roll bars,” Ph.D. dissertation, The Middle East Technical University, 2003.

[22] M. P. Bharane, M. K. Tanpure, and M. G. Kerkal, “Optimization of anti-roll bar using ansys parametric design language (apdl),” *International Journal of Engineering Research and General Sciece*, vol. 2, no. 5, 2014.

[23] K. Hudha, H. Jamaluddin, P. M. Samin, and R. A. Rahman, “Semi active roll control suspension (sarcs) system on a new modified half car model,” SAE Technical Paper, Tech. Rep., 2003.

[24] 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, no. S1, pp. 535–547, 2008.

[25] K. Deb, S. Agrawal, A. Pratap, and T. Meyarivan, “A fast elitist non-dominated sorting genetic algorithm for multi-objective optimization: Nsga-ii,” in *International Conference on Parallel Problem Solving From Nature*. Springer, 2000, pp. 849–858.

[26] K. Deb, A. Pratap, S. Agarwal, and T. Meyarivan, “A fast and elitist multiobjective genetic algorithm: Nsga-ii,” *IEEE Transactions on Evolutionary Computation*, vol. 6, no. 2, pp. 182–197, 2002.

[27] M. Sharifi, B. Shahriari, and A. Bagheri, “Optimization of sliding mode control for a vehicle suspension system via multi-objective genetic algorithm with uncertainty,” *Journal of Basic and Applied Scientific Research*, vol. 2, no. 7, pp. 6724–6729, 2012.

[28] J. Herrero, X. Blasco, M. Martínez, C. Ramos, and J. Sanchis, “Non-linear robust identification of a greenhouse model using multi-objective evolutionary algorithms,” *Biosystems Engineering*, vol. 98, no. 3, pp. 335–346, 2007.

[29] Z. Kadir, K. Hudha, F. Ahmad, M. F. Abdullah, A. Norwazan, M. Mohd Fazli, A. Khalid, and M. Gunasilan, “Verification of 14dof full vehicle model based on steering wheel input,” in *Applied Mechanics and Materials*, vol. 165. Trans Tech Publication, 2012, pp. 109–113.