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Investigation on Semi-active Suspension System for Multi-axle Armoured Vehicle using Co-simulation

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ABSTRACT

The objective of the study is to evaluate the performance of various semi-active suspension control strategies for 8x8 multi-axle armoured vehicles in terms of comparative analysis of ride quality and mobility parameters during negotiation of typical military obstacles. Since the cost, complexity and time precludes realisation of actual system, co-simulation technique has been effectively implemented for this investigation. Co-simulation combines advanced virtual prototyping and control technology which offers a novel approach to investigate the dynamics of such complex system. The simulations for the integrated control system along with multi body model of the vehicle are carried out for the control strategies, viz. continuous sky hook control, cascade loop control and cascade loop with ride control and compared with passive suspension system. The vehicle with 8x8 configuration is run on the real world obstacle profiles, viz. step, trench, trapezoidal bump and corrugated road and the effect of control strategies on ride comfort, wheel displacement and ground reaction is presented. It is observed that cascade loop with ride control in semi-active mode offers better vehicle ride comfort while crossing the said obstacles. The improved performance parameters are achieved through stabilisation of heave, pitch and roll motions of the vehicle through outer loop and isolation of vehicle level uneven disturbances through the fuzzy logic controller employed in inner loop.

Keywords: Semi-active suspension; Multi-body dynamics; Obstacle crossing; Co-simulation

1. INTRODUCTION

Military vehicles, both wheeled and tracked platforms, have to overcome different terrain conditions such as soft soil, cross country, snow area along with variety of natural and man-made obstacles. Worldwide tracked vehicles are being replaced by their wheeled counterpart for various combat and combat support roles due to their inherent advantages in terms of strategic mobility, maintainability and logistic footprint. However, military vehicle designers face challenges in providing ‘matching mobility’ and ‘adequate ride comfort’ for wheeled vehicles as that of tracked vehicles, especially in terms of negotiation of variety of terrains and manoeuvring various ground based obstacles either man-made or natural type.

Co-simulation has been significantly employed for study of variety of automotive sub systems including their overall interaction with the vehicle. The examples are an integrated simulation of electric power steering (EPS) and the dynamics behaviour of the vehicle[1], evolving vehicle stability control logic for four-wheel drive hybrid electric vehicle with fuzzy control[2], controlling a vehicle platooning system by interfacing automatic dynamic analysis of mechanical systems (ADAMS) and MATLAB[3], simulation of multi-body dynamic model of truck[4-7] and simulating the dynamics of robots[8].

Even though co-simulation has been quite extensively used for variety of systems, it is not exploited for study of dynamics of multi-axle armoured wheeled vehicles. Numerical research on the influence of movement conditions, viz. Velocity and various types of obstacles on the level of dynamic loads of the body shell and the vehicle crew was presented by co-simulation study[9]. Co-simulation technique was used where the multi-body virtual prototype of a sedan with sensor-less control methodology for semi-active controller for vehicle vibration control was tested under various load conditions in a near real environment[10]. An experiment was devised and executed to obtain both objective and subjective ride comfort values for the military vehicle under off-road conditions over typical terrain[11]. Vehicle mobility analysis was performed using NATO reference mobility models which considered only the input parameters pertaining to soil type, soil strength and terrain surface[12]. Parabolic and half sine wave shapes for obstacles were suggested as approximate functions where asymmetrical shapes of cosine and parabolic type are also discussed, but these shapes showed high sensitivity of different response variables[13]. Additionally, Weibull distribution function for generation of longitudinal road profiles with randomly distributed local obstacles was also used[14]. However, the severity of these profiles is not enough for military applications. In another case, a large rectangular obstacle for predicting the non-linear

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deformation and enveloping characteristic of the tire was used and the ride comfort was evaluated using co-simulation. For this, the size of obstacle considered is about 25 mm which is of very small order for multi-axle wheeled vehicle[15]. Vibration control strategies for 8x8 multi-axle platform using semi-active suspension control were presented with stochastic road inputs to represent cross country terrain profile[16]. It can be understood from the above studies that the co-simulations have not yet been attempted for the investigation of multi-axle vehicle, incorporating complex semi-active suspension control algorithms. Apart from this, the advance suspension systems are not assessed for their performance evaluation on actual obstacles to be encountered by the military vehicles.

The present work reports the application of co-simulation approach for performance evaluation of 8x8 multi-axle armoured platform with different semi-active suspension control schemes while negotiating actual field obstacles, viz. step climb, trench crossing, trapezoidal bump and corrugated track. The vehicle response in terms of ride comfort, wheel displacement and ground reaction is obtained. The performance of semi-active suspension control schemes, viz. Continuous skyhook, cascade loop and cascade loop with ride control over passive system is presented.

2. MULTI-BODY SEMI-ACTIVE DYNAMIC MODELLING OF 8X8 FULL VEHICLE

The investigation of performance of various semi-active controllers for 8x8 multi-axle wheeled platform presented in16 for random road inputs has been extended here to assess the performance of the controllers during negotiating the actual military terrains using co-simulation approach. The study considers three controllers, viz. continuous skyhook, cascade loop control and cascade loop with ride control which vary the damping force.

The continuous skyhook control provides the required damping coefficient based on the relationship expressed in Eqns. (1) and (2). The controller varies the damping coefficient between high state and low state damping forces:

If $\dot{Z}_S \times \dot{Z}_{rel} \geq 0$,

$$F_D = \max\left\{C_{\min}, \min\left[\left(C_{sky}\frac{\dot{Z}_S}{\dot{Z}_{rel}}\right), C_{sky}\right]\right\}\dot{Z}_{rel} \quad (1)$$

If $\dot{Z}_S \times \dot{Z}_{rel} < 0$, $\quad F_D = C_{\min}\dot{Z}_{rel} \quad (2)$

where $\dot{Z}_S$ is absolute sprung mass velocity, $\dot{Z}_{rel}$ is the relative velocity between the sprung and unsprung mass, C_{sky} and C_{min} are the skyhook damping coefficient and minimum damping coefficient, respectively and F_D is the desired damping force.

Figure 1 shows control scheme for cascade loop control. The cascade loop controller stabilises heave, pitch and roll motions of the sprung mass by linear control of gains.

The stabilising forces and moments are generated by the attitude control block which are then transferred to eight damping forces using input decoupling transformation. The input decoupling transformation blends the inner and outer loops. Figure 2 shows the control scheme for cascade loop with ride control. The ride control to isolate the vehicle body from wheel vibrations induced by road irregularities, load levelling and load distribution during vehicle manoeuvres is provided through the inner and outer loops. Unlike the cascade loop control system where the output force from input decoupling transformation is directly connected to the vehicle model, in cascade loop with ride control, the output force from input decoupling transformation is added to the force generated by ride control loop and connected to the vehicle model. Fuzzy logic has been implemented in the ride control loop. The fuzzy logic controller also varies the damping coefficient between high state and low state damping forces. The damping coefficients obtained from fuzzy controller are multiplied with feedback gain as presented in Eqn. (3). Twenty-five rules are used in the fuzzy controller[16]. Trapezoidal membership is found to be effective in this study and uses five values for input variable and seven values for output variable. This combination is found to achieve the best trade off performance. The input linguistic variables are classified into negative big (NB), negative small (NS), zero (Z), positive small (PS), positive big (PB), and output as small small (SS), small average (SAVG), small (S), medium (M), large (L), large average (LAVG), and large value (LV). Linguistic data to numerals transformation is done through the centroid method.

$$\text{Gain} = \left(\frac{\dot{Z}_S}{\dot{Z}_S - \dot{Z}_U}\right) \quad (3)$$

All kinematics and dynamical systems of 8x8 vehicle are implemented in a multi-body dynamics (MBD) environment using MSC-ADAMS. Mass and inertia properties are assigned to each component of the vehicle in the MBD model. The model has 233 degrees of freedom. Joints and constraints are

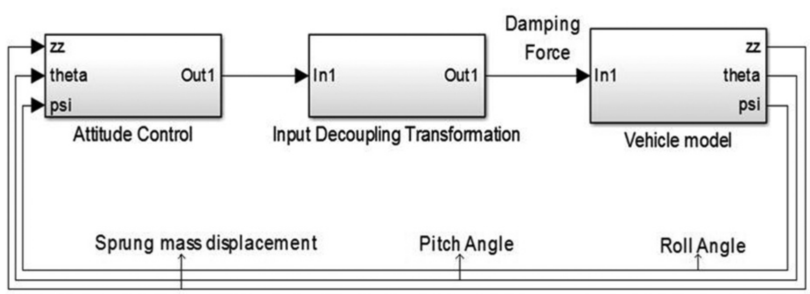

Figure 1. Control structure for cascade loop control[16].

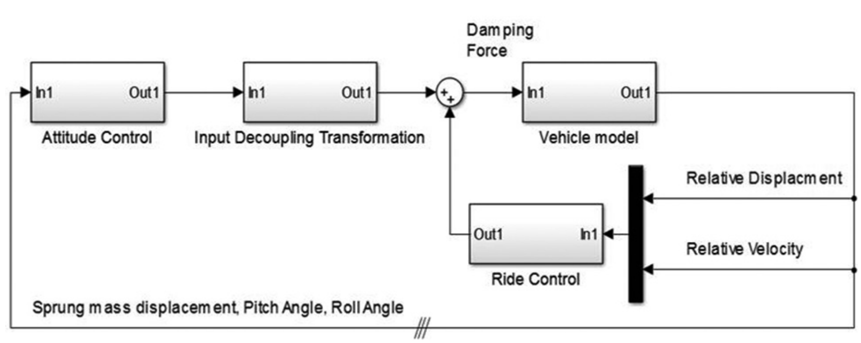

Figure 2. Control structure for cascade loop with ride control[16].

added between components. Pac 2002 tyre model is used for dynamic analysis. Road obstacles used for simulation, viz. step, trench, trapezoidal bumps and corrugated track are modelled in ADAMS. The topology map of suspension system is as shown in Fig. 3. It consists of vehicle body/hull, chassis (suspension mounting brackets), front and rear suspension system, steering system, road with obstacles. The vehicle has steerable double wishbone independent suspension in front two axles and non-steerable trailing arm independent suspension in rear two axles. The front axle steering knuckle (FAL_knuckle) is connected through spherical joints (S) to upper control arm (UCA) represented by FAL_UCA (front axle left UCA) and lower control arm (LCA) noted as FAL_LCA (front axle left LCA). These UCA and LCA are connected through revolute joints (R) to the chassis. Similarly, trailing arm knuckle left (TAL_knuckle) is fixed (F) to trailing arm (TAL_ARM) and trailing arm bracket shaft (TAL_BKT SHAFT). This trailing arm bearing bracket shaft is connected to the trailing arm bearing (TAL_bearing) fixed (F) on chassis through a revolute joint (R). Lower end of spring and damper assemblies in both front and rear are mounted on lower control arm (LCA) and trailing arm through revolute joints respectively. Higher end is connected to the chassis through revolute joint.

3. CO-SIMULATION

Two separate simulation programs are simultaneously used, viz. ADAMS for multi-body model of the vehicle and MATLAB/Simulink for semi-active damper control system. Both these platforms simulate the whole system by communicating with each other during run-time and thus exchange each other's output. A co-simulation is setup to run the vehicle model in ADAMS using the semi-active damper control model in Simulink. The ADAMS block contains complete vehicle information in terms of suspension geometry, type of joint and constraints and solves the mechanical system equations. The steering, road inputs and damping force for each wheel are defined as input state variable for ADAMS. The MATLAB/Simulink solves the damping control system equations by implementing various control schemes.

Steps involved in setting up a co-simulation between ADAMS and Simulink are :

i. Loading ADAMS/Controls
ii. Defining input and output variables
iii. Referencing input variables in the ADAMS model
iv. Exporting the ADAMS block
v. Connecting the ADAMS block and the semi-active damper control block in Simulink
vi. Running the co-simulation

Figure 4 presents the co-simulation flow chart. The co-simulation is carried out by considering sprung mass acceleration, ground reaction and wheel displacement as performance indicators.

On initiation of the simulation command, Simulink invokes ADAMS and runs the model in ADAMS/VIEW while the damper forces are calculated in Simulink and fed into ADAMS while the simulation is running.

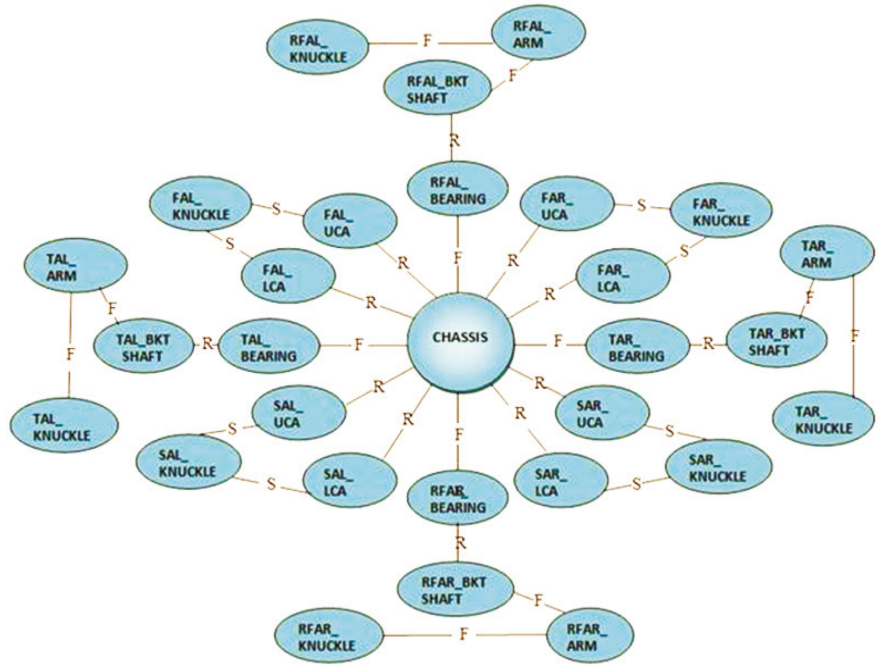

Figure 3. Topology map for suspension system.

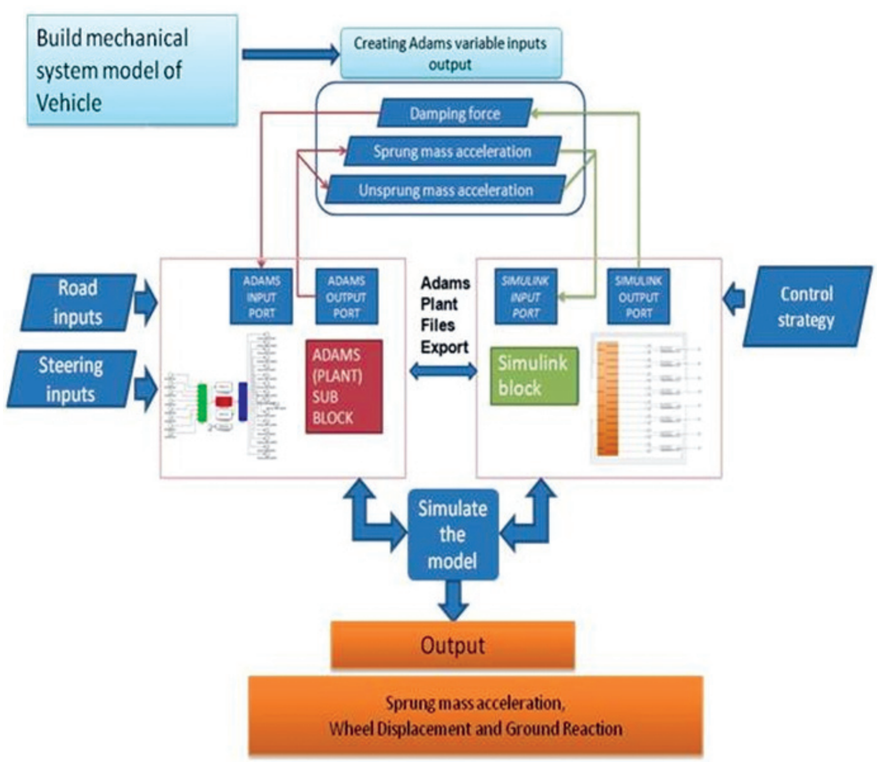

Figure 4. Co-simulation flow chart.

4. OBSTACLE CROSSING

Evaluation of various semi-active suspension control strategies for 8x8 multi axle armoured platform in terms of ride quality and mobility parameters during negotiation of typical military obstacles is carried out based on co-simulation results. Two types of obstacles encountered by military vehicles are considered in this study, viz. event based/transient and cyclic. Step and trench represent event based/transient obstacles and trapezoidal bump and corrugated track represent cyclic obstacles. Crawling speed of the vehicle is 5 km/h and the same is considered for step, trench and corrugated track. Constant vehicle speed is considered as 30 km/h for trapezoidal bump as per vehicle performance requirement. Sprung mass acceleration is a commonly used criterion to assess the dynamic behaviour of a suspension, as it is directly related to ride comfort. Lower sprung mass acceleration indicates superior ride performance in terms of better crew comfort and higher speeds while negotiating terrains. The simulations are performed for a period of 100 s using ode 45 (Dormand-Prince) solver with a time step of 0.02 s for 100 s for all road obstacles and results of passive system are compared with co-simulation results. The actually measured vehicle parameters of representative 8x8 armoured vehicle, as given in Table 1.

Table 1. Model parameters used for simulation

Parameter	Value
Mass of vehicle	24000 kg
Each unsprung mass	200 kg
Wheel base (front axles)	4.275 m
Wheel base (rear axle)	2.750 m
Track width	2.6 m
Spring stiffness (front two axles)	1100 x10^3 N/m
Passive damping coefficient (front two axles)	156666 Ns/m
Spring stiffness (rear two axles)	920 x 10^3 N/m
Passive damping coefficient (front two axles)	89586 Ns/m
Roll inertia (IXX)	30000 kgm^2
Pitch inertia (IYY)	90000 kgm^2
Location of CG from 1st axle	2.418 m
Location of CG from 2nd axle	0.893 m
Location of CG from 3rd axle	-1.106 m
Location of CG from 4th axle	-2.606 m
Tire stiffness	1.56 x 10^6 N/m
Tire damping	500 Ns/m

4.1 Event based /Transient Obstacle

4.1.1 Step Climbing

The simulation is carried out for a step of height 600 mm road input. The vehicle speed is kept constant at 5 km/h. The variations of sprung mass acceleration with time under passive and three controlled systems presented in Fig. 5. It is seen that the performance of both the passive and controlled systems are identical in the beginning. Suspension control in this case is implemented through velocity dependent damper. It can be clearly seen that controlled suspension is more effective in controlling the sprung mass acceleration peaks. A comparison amongst the controlled suspensions for peak sprung mass acceleration amplitudes and settling time reveals that cascade loop with ride control provides better performance than the other two control strategies.

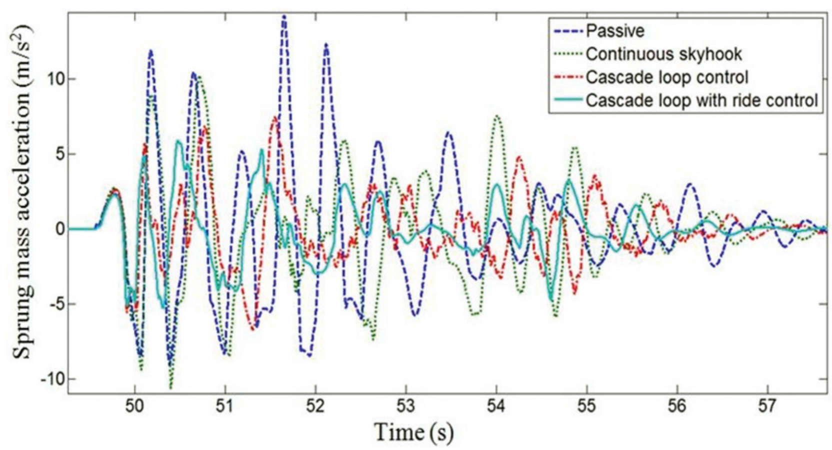

Figure 5. Comparison of the sprung mass acceleration without and with implementation of various control strategies during step climbing.

4.1.2 Trench Crossing

The simulation is carried out for a straight walled trench of width 1100 mm and depth 500 mm. For this obstacle, it is observed from Fig. 6 that the performance of cascade loop with ride control and cascade loop control are better than passive system. In the event of loss of ground contact, passive system performs better than continuous skyhook control since its control logic is dependent on relative velocity of sprung and unsprung mass.

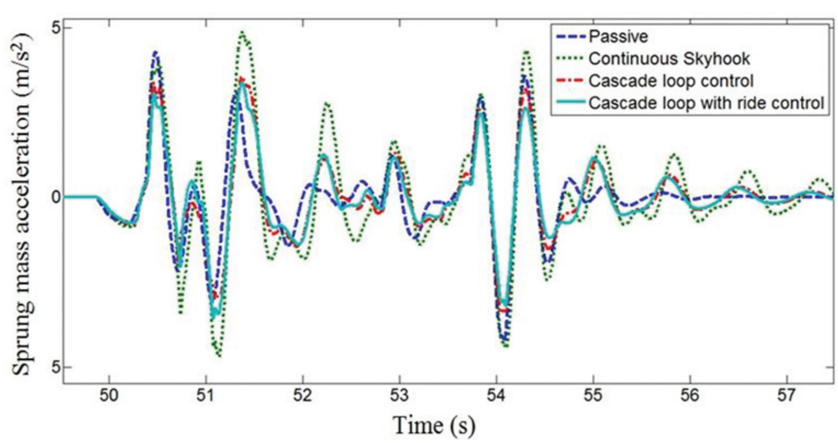

Figure 6. Comparison of the sprung mass acceleration without and with implementation of various control strategies during trench crossing.

4.2 Cyclic Obstacle

4.2.1 Trapezoidal Bumps

Two trapezoidal bumps of length at top and bottom of 410 mm and 1010 mm, respectively and height 100 mm. The distance between two trapezoidal bumps is 3000 mm. The simulation is carried out at a speed of 30 km/h considering real test conditions. From Fig. 7 it is observed that controlled systems perform better than the passive system. It is observed that cascade loop controlled system has not settled even after crossing of the obstacle. This can be attributed to the structure of cascade loop controller of which inputs are vertical displacement, pitch angle displacement and roll angle displacement. In this scheme the attitude control of sprung mass is achieved by linear control of gains alone.

4.2.2 Corrugated Track

The simulation is carried out while crossing a corrugated track of amplitude 100 mm and wavelength 1125 mm. The speed of the vehicle is kept at 5 km/h. Sprung mass acceleration for controlled and passive system is presented in Fig. 8. For this case also controlled systems perform better than the passive system.

4.2.3 Discussion

A comparison of suspension control effectiveness in obstacle crossing in terms of sprung mass acceleration, ground reaction and wheel displacement is done. It is desired that military vehicle travel through cross country at high speeds and overcome the obstacles to meet mission objectives. This leads to uncomfortable oscillating motion to the occupants. Ride comfort of the vehicle is in direct relation with sprung mass acceleration. This uncomfortable motion is measured in terms of sprung mass acceleration. However, achieving better ride comfort alone is not sufficient. Vehicle road holding ability while overcoming the obstacles also needs to be ensured. Two parameters, viz. the wheel displacement and ground reaction which are indicative of vehicle road holding ability are discussed. Wheel displacement is directly related to handling stability of automobile. Higher wheel displacement for a given road input implies better ground contact of the wheel at that instant. Also higher wheel displacement results lower sprung mass acceleration. Ground reaction or wheel load provides an indication of vehicle traction/braking and handling ability.

Co-simulation with control strategies, viz. continuous skyhook, cascade loop control and cascade loop with ride control is carried out and the results are compared with continuous skyhook control and passive system for benchmarking purpose. The sprung mass acceleration at the centre of gravity of vehicle, ground reaction and wheel displacement are obtained with these control strategies and compared.

It is observed from Fig. 9 that the sprung mass acceleration could be reduced with the implementation of suspension control compared to those with passive system. Performance of controlled system is better in terms of sprung mass acceleration. It is also observed that controlled suspension performs the best for step climbing. For trench crossing, sprung mass acceleration for controlled suspension is almost close to passive system except continuous skyhook control. This is attributed to the skyhook control logic's sole dependence on relative velocity. Vibration control, as such, is indicated by reduction in this parameter. Higher off-road speed with reduced fatigue to the occupants is ensured when the vehicle passes through rough terrains for prolonged duration. Thus, ride comfort is significantly improved by the semi-active control strategies.

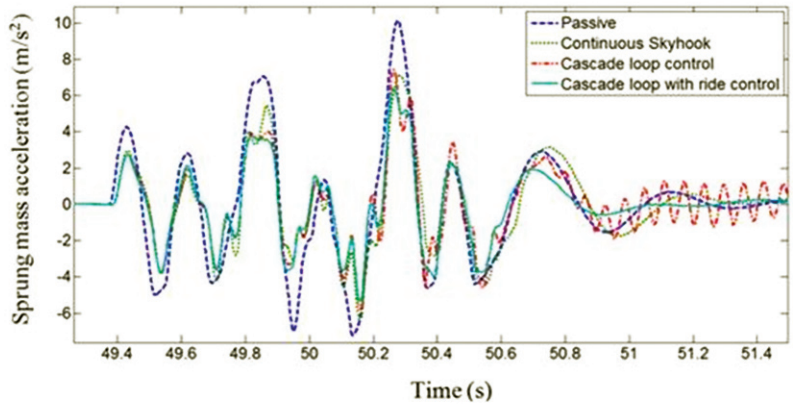

Figure 7. Comparison of the sprung mass acceleration without and with implementation of various control strategies during crossing trapezoidal bumps.

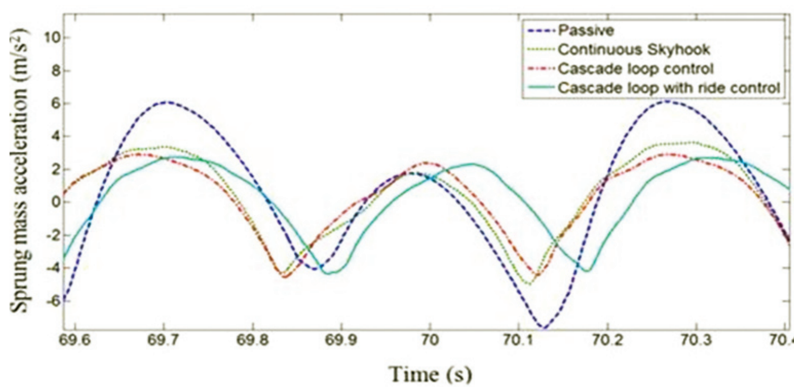

Figure 8. Comparison of the sprung mass acceleration without and with implementation of various control strategies during crossing corrugated track.

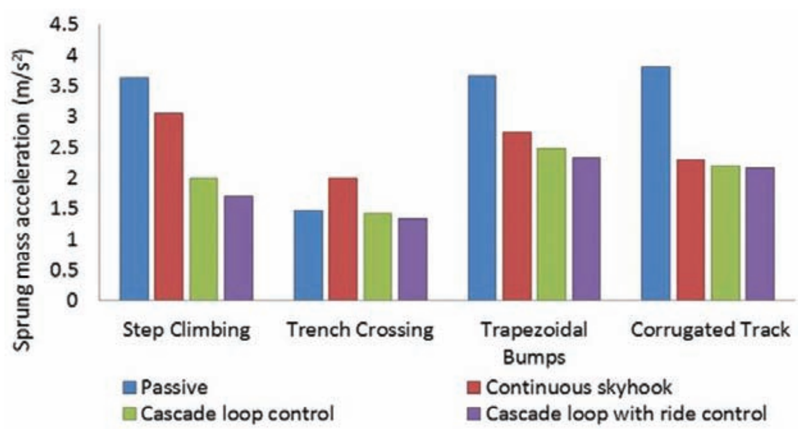

Figure 9. Comparison of RMS sprung mass acceleration (m/s^2) under various road inputs.

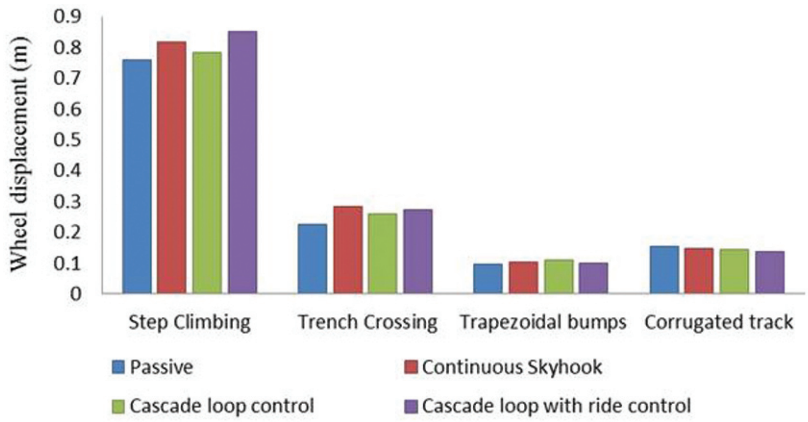

Figure 10. Comparison of actual wheel displacement (m) under various road inputs.

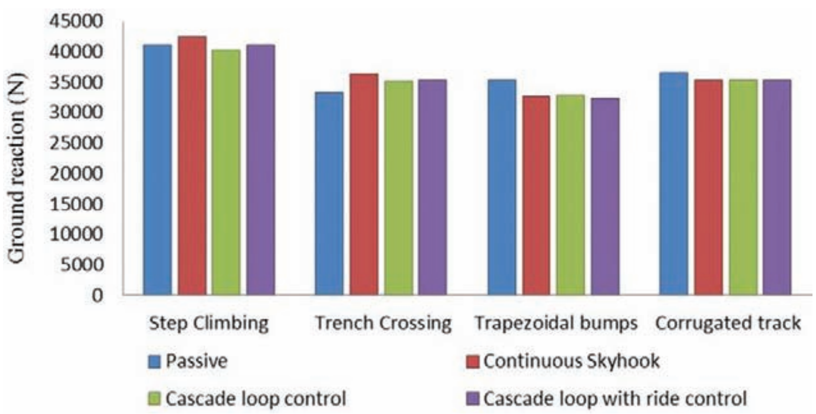

Figure 11. Comparison of RMS ground reaction (N) under various road inputs.

Wheel displacement and ground reaction for controlled and passive system are compared for the obstacles as given in Figs. 10 and 11, respectively. In step climbing it is observed that the wheel displacement for cascade loop with ride control is about 10 per cent higher than passive system but reduction in sprung mass acceleration is about 50 per cent. This means that significant amount of improvement in ride comfort is achieved by marginal increase in wheel displacement. For other obstacles, wheel displacement is very close to or marginally higher than passive system. Ground reaction gives an indication of road/ground holding. Variation of this parameter with respect to its static wheel load should be minimum to ensure sufficient traction/braking ability. Performance of controlled systems in respect of this parameter matches with passive system. This shows that semi-active suspension control with cascade loop with ride control can be effective in vibration control with road holding performance comparable to that of passive suspension system.

7. CONCLUSIONS

The present study demonstrates the use of co-simulation technique for performance evaluation of semi-active suspension parameters for 8x8 armoured vehicle which can be extended to variety of multi-axle configuration used in military ground mobile systems. This methodology would give *a priori* information for selection of a particular control strategy for suspension system of multi-axle vehicles for vibration control, which otherwise is not possible unless an arduous route of realising and testing the actual hardware is resorted to. The paper also demonstrates effective implementation of transient and cyclic obstacles for analysis of heavy off-road vehicles. The obstacles help to assess the vehicle as regards traction/braking ability and handling stability. The study also highlights the relative obstacle crossing performance of semi-active suspension control strategies, viz. continuous skyhook, cascade loop and cascade loop with ride control. The quantified data of response of these strategies to the obstacles in terms of ride comfort, ground reaction and wheel displacement is considered useful in military domain to predict the overall sustainability of occupants for prolonged missions as well as to decide upon the battlefield mobility of such complex platforms. Based on the investigation, it is inferred that the cascade loop with ride control is the most promising choice for semi-active suspension control for the multi-axle vehicles.

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Contribution in the current study, he did concept generation, outline of the research work, analysis of the results, compilation of paper.

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Contribution in the current study, he did topology of suspension system, dimensioning and modelling of the Terrain obstacles.

Dr Vasudevan Rajamohan obtained his PhD from Concordia University, Canada. Currently working as a Professor at Centre for Innovative Manufacturing Research (CIMR) at VIT University, Vellore, India. His research focuses on broad range of problems in Mechanics of composite structures, dynamics and control, smart materials and nano composites, structural health monitoring, with applications in aerospace and automotive industries.

Contribution in the current study, he guided the entire project, interpretation of results and correlation studies, guidance in paper writing and documentation.

Mr Pandhare Rakesh Sampat Rao is the Post Graduate student of VIT University, Vellore, India. His areas of interest include vibration, mechanics and materials.

Contribution in the current study, he executed of models, performing requisite iterations of co-simulation and reporting of results.