DYNAMIC MODELING AND CONTROL OF OFF-ROAD TRUCK USING BOND GRAPHS

Ashish Gupta[1], Vikas Rastogi[2]

[1] Assistant professor in Mechanical Engineering department
[2] Professor in Mechanical Engineering department
[1]Krisha Engineering College, Mohan Nagar, Ghaziabad
[2]Delhi Technical University, New Delhi

ashish.j001@gmail.com, rastogivikas@yahoo.com

ABSTRACT
The truck vibration problems have increase in recent years due to the rapid industrlization and demand for the modern machines and product to transfer from one place to the other. Vibration in modern off-road road trucks can affect driver comfort, cargo safety, equipment wear and productivity. This work presents the modeling of off road truck using bond graphs. The vehicle system is comprised of cab, cargo, base frame, suspension system and pneumatic tires. In this work, dynamic behavior of the truck has been presented using bond graphs, which include the wheel/tires, axle/suspension and frame of the vehicle. The vehicle is modeled with the integration of rigid bodies that are allowed to move the dimensional space subject to forces and moments. This works also incorporating Proportional Integrated Derivative (PID) control in the suspension system of truck. These controllers are frequently designed in the frequency domain, and on the hypothesis of linearity. These control systems reduce the deflection of suspension system and increase comfort level of driver.

Keywords: bond graph modeling, ride comfort, proportional control.

1. INTRODUCTION

Modeling and simulation has an increasing importance in the development of complex, large mechanical systems. In areas like road vehicles (Demic and Lukic 2002; Filippini 2004; Pacejka 1975), rail vehicles, high speed mechanisms, industrial robots and machine tools (Paul 1975), simulation is an inexpensive way to experiment with the system and to design an appropriate control system. Generally, modeling is an in expensive way to experiment with different system design concepts and to aid the design development of an appropriate control system. There are different approaches to model a vehicle.1) Ignoring body flexibility by using a lumped mass model, 2) Modeling the frame as a regular free-free beam and calculating, estimating or measuring modal masses and stiffness's, and 3) Modeling the entire vehicle using the finite element method **(**Ibrahim 1996; Goodarzi and Jalali 2006; Yi 2000; Cao 2005). As in most other dynamic systems, the analyst is faced with a spectrum of possible model complexity. Traditionally, linear models have been used, based on the well-known partial differential equation for transverse vibration $w(x, t)$ of a free-free beam subject to separation of variables (Karnopp 2006). Bond graph models of a bus and tractor-trailer were created in ref. (Margolis and Edeal 1989), in which small vertical vibration motions were assumed.

Vibration in modern over-the-road trucks can affect driver comfort, cargo safety, equipment wear, productivity etc. Among all the problems, vibration may have a number of different sources and its effective solution due to various causes should be accurately determined. The truck vibration problems have increased in recent years due to the rapid industrialization and demand for the modern machines and product to transfer from one place to the other. The dynamic behavior of truck also depends on load on cargo-body and the mechanical system such as springs, dampers etc, which interact with the wheels/tires, the truck base, cab and cargo. Ride Comfort evaluation is one of the most critical factors to evaluate the vehicle performance and has been an interesting topic for researchers for many years. Generally, two methods are prevalent to investigate the ride comforts. The first method is the use of computer simulation, and the second is the road experiment. Generally, the road experiment found to be very costly and complex process finding out the results. This paper deals with the multi-rigid body theory through the bond graph technique (Ronald 2003; Albert 1986; Inhee and Changsoo 1998; Erial, Stein and Louca 2004; Cellier) for modeling of a complex off-road truck. Primarily, bond graphs (BG) represent elementary energy-related phenomena (generation, storage, dissipation, power exchange) using a small set of ideal elements that can be coupled together through external ports representing power flow. With the aid of bondgraphs, hierarchical modeling becomes possible through coupling of component or subsystems models through their connecting ports. Besides these physical features capturing energy exchange phenomena, it is also possible to code on the graph the mathematical structure of the physical system to show the causal relationships (in a computational sense) among its signals (Ronald 2003).The conjunction of all these features make the bond graphs techniques a physically based, object oriented, graphical language, which is most suitable for dynamic modeling, analysis and simulation of complex engineering systems involving mixed physical and technical domains in their constitution (Cellier).

This paper also incorporates the PID control system (which is combined form of proportional, integrated and derivative controller) to the vehicle suspension system with an objective to reduce vehicle jerks and increase comfort level for the driver. This controller determines the value of controlled variable, compares the desired value, determines the deviation and produces a control signal that will reduce the deviation to zero or to a smallest possible value.

2. DETAILED DESCRIPTION OF THE ELEMENTS OF THE OFF-ROAD TRUCKS

Truck structure can be disassembled into the following parts, a) Cab b) Cargo c) Base frame, d) Suspension system e) Pneumatic tires. The schematic view of whole truck structure is shown in Fig.1. A structure of the vehicle is composed of components such base frame, suspension system, cab, cargo and so-forth. When dynamic system is connected to these components, one must interconnect rotating and translating inertial elements with axial and rotational spring and dampers and also appropriately account for the system structure. Bond graphs are well suited for this task. The next section consists of modeling of different sub system of a truck-car vehicle. In deriving the bond graph models, the vehicle velocities in upward direction are assumed to be positive and springs and dampers are all assumed positive in compression. The base frame is modeled as a rigid body, which is allowed to move for pitching and rolling. The various elements of off-road trucks are presented in the following subsections.

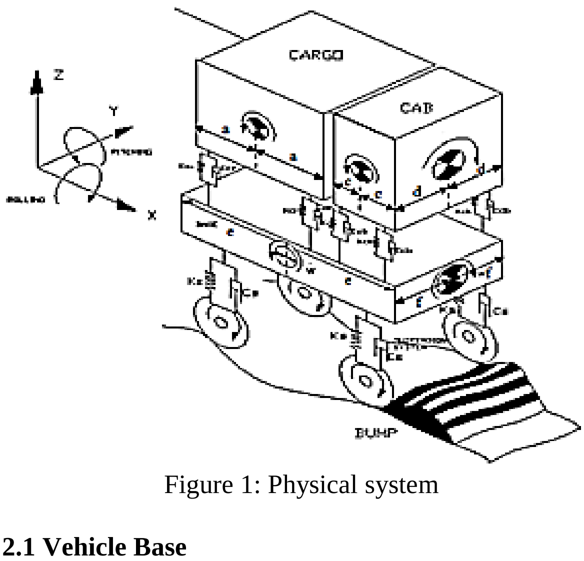

Figure 1: Physical system

2.1 Vehicle Base

The vehicle base is modeled as a rigid body with a local coordinate reference frame (x, y) attached to the center of mass and aligned with the inertia principal axes as is shown in Fig.1. It has mass m_b, and the following principal inertia moments: roll (J_{bx}) respect to the body x-axis, pitch (J_{by}) respect to the body y-axis. It is also connected with suspension system.

2.2 Cab body

The cab body is also designed as a rigid body. In this part of vehicle, have a driver seat and the other space of sitting. It has mass m_{cb}, and the rolling inertia J_{cbx} respect to the X-axis or (J_{cby}) pitching inertia moments respect to the axis. It has a four spring-damper systems which connected the cab-body to the base. The cab-body is also shown in Fig.1.

2.3 Cargo mass

The cargo mass is designed as a rigid body. These masses are lifted by the base frame. The cargo is assembled with truck for carrying a mass from one place to another. This mass also includes the mass of frame structure of the truck.

2.4 Suspension system

Suspension is the term given to the system of springs, shock absorbers and linkages that connects a vehicle to its wheels. This assembly is used to support weight, absorb and damped road shock, which helps to maintain the tire contact as well as proper wheel-to-base relationship. Without being a restriction for future extensions of the overall vehicle model, only static suspension systems are considered in this research work, which may or may not have non-linearly.

2.5 PID control system

A structure of PID control is shown in fig.2, where it can be seen that in a PID controller, the error signal e(t) is used to generate the proportional, integral and derivative actions, with the resulting signal weighting and summed to control signal u(t) applied to a plant model. A mathematical description of the PID controller is,

$$u(t) = K_p e(t) + \frac{1}{G_i}\int_0^t e(t) + G_d \frac{de(t)}{d(t)}$$

Where, u(t) is the input signal to the plant model, the error signal e(t) is defined,

e(t) = r(t) – y(t)

A proportional controller (G_p) will have the effect of reducing the rise time and will reduce, but never eliminate, the steady-state error. An integral control (G_i) will have the effect of eliminating the steady-state error, but it may make the transient response worse. A derivative control (mu) will have the effect of increasing the stability of the system, reducing the overshoot, and improving the transient response. The combination of proportional, integral, and derivative control is called PID control system. In the fig 2, G_p presents proportional gain, G_I presents integrated gain and μ presents derivative gain.

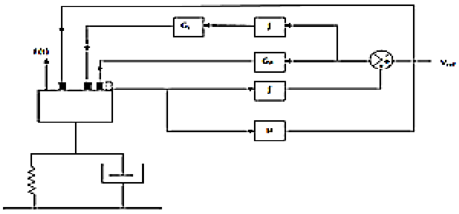

Figure 2: Proportional control

3. BOND GRAPH MODELING

This section will elucidate the bond graph modeling of various elements and the integrated bond graph model of off-road truck.

3.1 Cab body

The cab body modeled as a rigid body with centre of mass located at distance ‘c’ from the front and rear spring damper & at a distance ‘d’ from the left and right spring damper system. The cab body is characterized by the centre of mass velocity and angular velocity, so I elements for the mass, M_{cb} and moment of inertia, J_{cbx} and J_{cby}, are attached to the appropriate 1-junctions. The weight of the body is an effort source attached to the centre of mass velocity. Positive power is directed into the source owing to the velocity convention (positive upward).

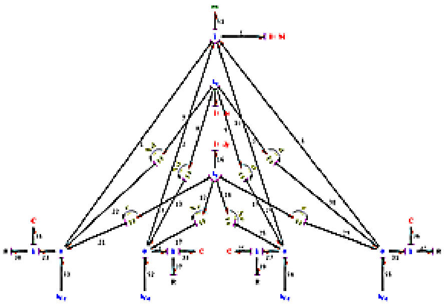

Figure 3: Bond graph model of cab-body

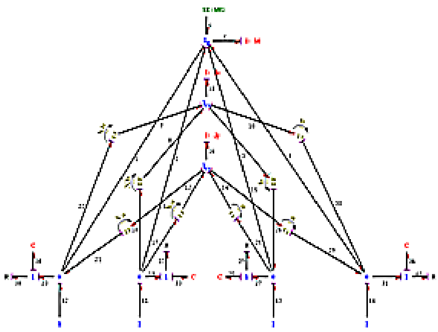

Figure 4: Bond graph model of cargo-body

Kinematic constraints must be enforced correctly for constructing the correct model. Kinematic constraints are always having relationship among the flow variables. To add the flow according to the constraints, 0-junctions are used. The transformers are used to convert the angular velocity (ω) into velocity components ω×c and ω×d. On the transformer the moduli are appended according to the definition of the transformer. In Fig. 3, the constraints of Eq. (1) to Eq. (8) are summed up using 0- junctions.

$$V_{cb1} = V_{Gcb} + \omega_{ycb} \times c \tag{1}$$

$$V_{cb3} = V_{Gcb} - \omega_{ycb} \times c \tag{2}$$

$$V_{cb2} = V_{Gcb} + \omega_{ycb} \times c \tag{3}$$

$$V_{cb4} = V_{Gcb} - \omega_{ycb} \times c \tag{4}$$

$$V_{cb1} = V_{Gcb} + \omega_{xcb} \times d \tag{5}$$

$$V_{cb2} = V_{Gcb} - \omega_{xcb} \times d \tag{6}$$

$$V_{cb3} = V_{Gcb} + \omega_{xcb} \times d \tag{7}$$

$$V_{cb4} = V_{Gcb} - \omega_{xcb} \times d \tag{8}$$

3.2 Cargo Body

The cargo body is also modeled as a rigid body with center of mass located at distance of ‘a’ from the front and rear spring damper system ‘b’ from the left and right spring-damper system. The cargo body is characterized by the centre of mass, velocity and angular velocity, so I-elements for the mass Mcr and moments of inertia, Jcrx and Jcby are attached to the appropriate 1-junction. The weight of the body is an effort source attached to the center of mass velocity. Positive power is directed in to the source owing to the velocity convention (positive upward).

On the transformer the moduli are appended. 0-junctionis used again to add the flow to the kinematic constraints. In Fig 4, the constraints of Eq (9-16) are summed up using 0 junctions.

$$V_{cr1} = V_{Gcr} + \omega_{ycr} \times a \tag{9}$$

$$V_{cr3} = V_{Gcr} - \omega_{ycr} \times a \tag{10}$$

$$V_{cr2} = V_{Gcr} + \omega_{ycr} \times a \tag{11}$$

$$V_{cr4} = V_{Gcr} - \omega_{ycr} \times a \tag{12}$$

$$V_{cr1} = V_{Gcr} + \omega_{xcr} \times b \tag{13}$$

$$V_{cr2} = V_{Gcr} - \omega_{xcr} \times b \tag{14}$$

$$V_{cr3} = V_{Gcr} + \omega_{xcr} \times b \tag{15}$$

$$V_{cr4} = V_{Gcr} - \omega_{xcr} \times b \tag{16}$$

3.3 Base frame

The Base frame also modeled as a rigid body with center of mass located at distance of ‘e’ from the front and rear spring damper system and ‘f’ from the left and right spring-damper system. The cargo body is characterized by the centre of mass, velocity and angular velocity, so I-elements for the mass, M_{bs} and moments of inertia Jbsx and Jbsy are attached to the appropriate 1-junction. The weight of the body is an effort source attached to the center of mass velocity. Positive power is directed in to the source owing to the velocity convention (positive upward).One may obtain the similar equations presented in Eqs (17-24) are summed up using 0 junctions. However, due to page limitation, the base frame model is shown in the integrated model.

$$V_{bs1} = V_{Gbs} + \omega_{ybs} \times e \quad (17)$$
$$V_{bs2} = V_{Gbs} + \omega_{ybs} \times e \quad (18)$$
$$V_{bs3} = V_{Gbs} - \omega_{ybs} \times e \quad (19)$$
$$V_{bs4} = V_{Gbs} - \omega_{ybs} \times e \quad (20)$$
$$V_{bs1} = V_{Gbs} + \omega_{xbs} \times f \quad (21)$$
$$V_{bs2} = V_{Gbs} - \omega_{xbs} \times f \quad (22)$$
$$V_{bs3} = V_{Gbs} + \omega_{xbs} \times f \quad (23)$$
$$V_{bs4} = V_{Gbs} - \omega_{ybs} \times f \quad (24)$$

3.4 Suspension System

The spring damper in general reacts to the relative velocity across them. To properly add the velocity component at each end of these elements, the spring and damper are assumed to be positive in compression. The modeling of suspension system and pneumatic tire has been presented in the integrated bond graph model of the system.

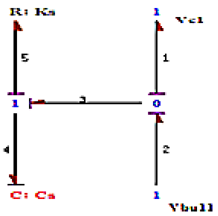

Figure 5: Bond graph model of Suspension system

3.5 PID control system

The bond graph model of PID control system is shown in Fig 6. A system is applying on a quarter suspension system. In the present model, 'mu' is representing a derivative gain, G_p is proportional gain which modulated on a Transfer Function (TF) and G_i is an integral gain.

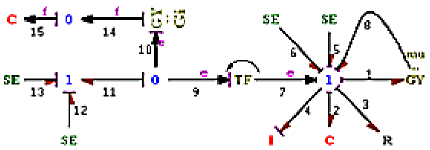

Figure 6. Bond graph model of PID control system

3. 6 Integrated Bond Graph model of complete off-road truck without control system

The bond graph model of road-truck is constructed by adding the models of different components of vehicle as shown in Fig. 6.

4. SIMULATION STUDIES

The bond graph model of the vehicle is simulated for 10 sec to obtain different output responses. Total 1024 records are used in the simulation and error of the order of 5.0 x 10-4 is considered. Symbols shakti software is used to carried out for simulation work. Fifth-order Runge-Kutta method, is used in simulation work.

The dynamic wheel loads generated by a moving vehicle are mainly due to various wheel/road imperfections. These imperfections are considered as the primary source of dynamic track input to the road vehicles. In actual practice different type of periodic, periodic or random road irregularities may exist on the track. However, but in the present study bump type of irregularity is considered as shown in Fig.8 (Mukherjee and Karmakar 2000). For simplicity, the shape of irregularity is assumed to be of same nature on left and right wheel. However different shape of irregularity may be attempted in near future. Velocity inputs at different wheels are calculated by using equations from Eq. (25) and Eq. (26).

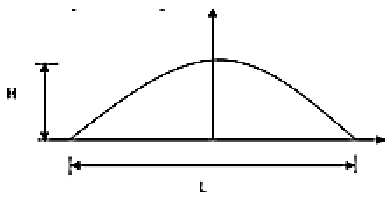

Figure 8: Bump type surface irregularities

The bump excitation of the front wheels is,

$$V = H\sin(\pi \frac{V}{L} t) \qquad For \qquad 0 \le t \le \frac{L}{V}$$

$$V = 0 \qquad For \qquad t \ge \frac{L}{V}$$

The bump excitation of the rear wheels is,

$$V = H\sin(\pi \frac{V}{L}(t - \frac{A}{V})) \qquad For \frac{A}{V} \le t \le \frac{A+L}{V}$$

$$V = 0 \qquad For \qquad t \ge \frac{A+L}{V}$$

In the present study H is taken as 0.03 m and L is taken as 1m.

Following output parameters are obtained in the simulation of the bond graph model of the complete truck model:

a) Suspension displacement with PID and proportional control system for quarter car model

b) Vertical effort on suspension system

(a) Suspension displacement with PID and proportional control system for quarter car model

In a quarter truck model, PID control system incorporates in left rear suspension system and Proportional control system incorporates in right rear system therefore front suspensions are not connected by these systems. The deflection of suspension at various speeds is shown in Figs. The effect of PID and proportional control systems in suspensions of both rear wheels are shown in Figs 8-12. Since irregularities on both the wheel are assume to be same shape, therefore the deflection of suspension springs of both rear suspension will not be same, which is also apparent from all the above simulated curves.

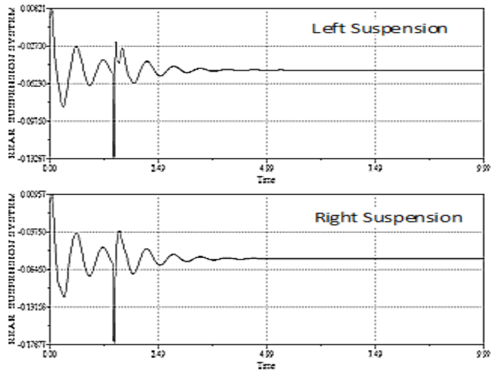

Figure 8: Deflection of suspension at 20 km/hr

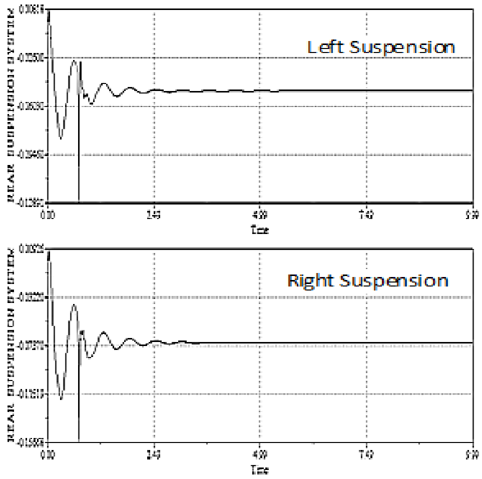

Figure 9: Deflection of suspension at 40 km/hr

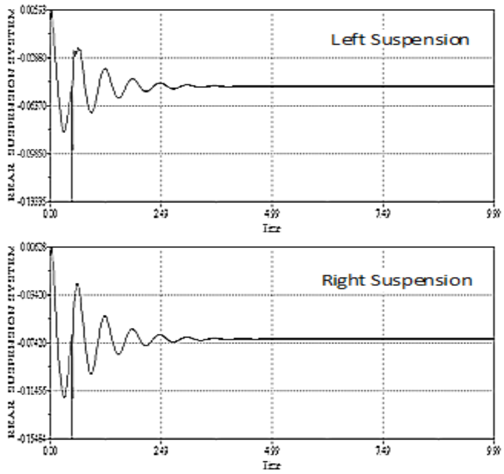

Figure 10: Deflection of suspension at 60 km/hr

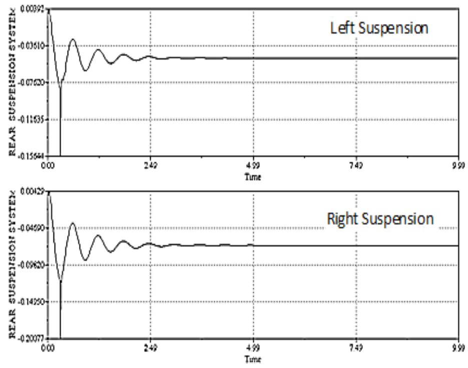

Figure 12: Deflection of suspension at 100 km/hr

b) Vertical effort on suspension system

The vertical effort on left rear suspension with respect to time are presentented in Figs 13-17, whether suspension system is incorporates with Proportional Integrated Derivative control system.

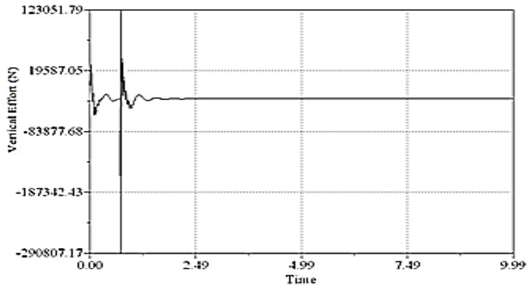

Figure 13: Vertical Effort on Suspension system at 20 Km/hr

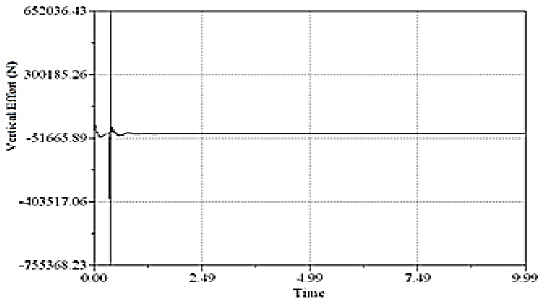

Figure 16: Vertical Effort on Suspension system at 80 Km/hr

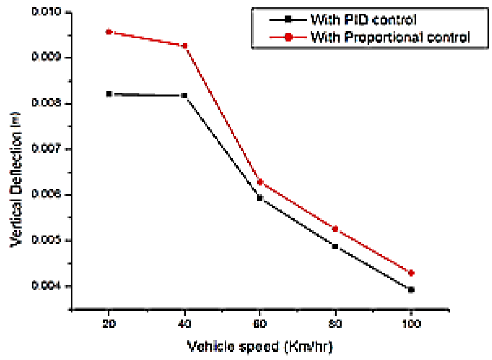

Figure 18: Suspension deflection of truck model at various speeds

It is evident from Fig.18 that the magnitude of displacement of suspension is comparatively lower, when connected by PID control system. At speed 60 km/hr, PID control connected suspension gives lower magnitude, whereas when the suspension is connected with proportional controller gives comparatively higher magnitude of amplitude of vibration.

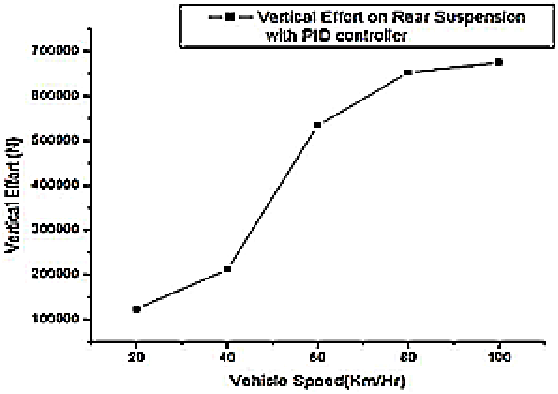

Figure 19: Vertical effort on suspension system

Fig.19 shows a vertical effort of rear suspension system, when connected by a Proportional Integrated Controller. It is evident from plot, that a value of vertical effort is continuously increased with the vehicle speed. It is resulted from both figures that PID control system gives satisfactory result at speed limit 40-60 km/hr.

5. CONCLUSIONS

The dynamic model of off-road truck of cab/cargo/base had been constructed through bond graph technique. Vertical dynamics has been carried out for off-road truck model. A 6-degree of freedom model is used for the analysis. Velocity input at the entire tire has been given by considering same shape bump irregularity at both right and left tire. The PID control system is attached to a suspension system, which provides a good analysis to analyze the vibration of the system.

REFERENCES

Demic, M., Lukic, J., 2002. Some aspects of the investigation of random vibration influence on ride comfort, *Journal of Sound Vibration*. 253 (1), 109–129.

Filippini, G., 2004, *Nigro. N. and Junco.S.,* Dinámica *Vehicular mediante bond graphs.* Proyecto final de carrera de grado. Escuela de Ingeniería Mecánica, Universidad Nacional de Rosario.

Pacejka, H. B., 1975. Principles of plane motion of automobiles, *IUTAM symposium on the dynamics of automobiles*, Delft Univ. of Tech. pp. 33- 59.

Paul, B., 1975. Analytical dynamics of mechanisms-a computer oriented overview, *Mechanism and Machine Theory*, vol 10, pp. 481-507, 1975.

Ibrahim, I.M., 1996. Effect of Frame Flexibility on the Ride, *Vibration of Heavy Trucks*, 58(4), pp. 709-713. Elsevier Science Ltd.

Goodarzi, A., and Jalali, A., 2006. An Investigation of Body Flexibility Effects on the Ride Comfort of Long Vehicles. Proc. CSME *Canadian Congress of Applied Mechanics*, (*CANCAM 2006).*

Yi, T.Y., 2000. Vehicle Dynamic Simulations Based on Flexible and Rigid Multi-body Models, SAE Paper 2000-01-0114. *Society of Automotive Engineers*, Warrendale, PA.

Cao, C., 2005. Approaches to Reduce Truck Beaming, SAE Paper 2005-01-0829. *Society of Automotive Engineers*, Warrendale, PA.

Margolis, D. and Edeal, D., Modeling and Control of Large Flexible Frame Vehicles Using Bond Graphs SAE Paper 892488. *Society of Automotive Engineers*, Warrendale, PA, 1989.

Karnopp, D. et al., 2006. System Dynamics – Modeling and Simulation of Mechatronic Systems (4th ed.) *John Wiley and Sons, Inc.*

Rosenberg, R.C., Margolis, R.L., Karnopp, D.C., 2003. Modeling and Simulation of Mechatronic Systems, *A Wiley-Interscience Publication*, New York.

Albert M. Bos,1986 . *Modeling Multibody Systems in terms of Multi-bond Graphs, with application to a motorcycle*, PhD thesis at Twente Univ.

Jang, I., Changsoo H Edeal , 1998. Proposition of a Modeling Method for Constrained Mechanical Systems Based on the Vector Bond Graph, *Journal of the Franklin Institute*. Vol. 335B, No 3, pp. 451 469..

Erial,T., Stein, J., Louca, L., 2004. A Bond Graphs Based Modular Modeling Approach towards an Automated Modeling Environment for Reconfigurable Machine Tools. *IMAACA 2004*.

Cellier, F., Hierachical non-linear bond graph: A unified methodology for modelling complex physical systems, *Simulation*, Vol 58, No. 4, pp. 230-248.

Mukherjee, A., and Karmakar, R., Modelling and Simulation of engineering System through Bond Graph, *Narosa publishing House*, New Delhi, reprinted by CRC press for North America and by Alpha Science for Europe, 2000.

Mukherjee, A., and Samantaray, A. K., *SYMBOLS-Shakti User's Manual*, High-Tech Consultants, STEP Indian Institute of Technology, Kharagpur, India, 2006.

Authors Biography

First author of this paper, is working as an Assistant Professor in Mechanical Engineering Department in Krishna Engineering College, India. He has completed his Master degree from Sant Longowal Institute of Engineering & Technology, Estd by Govt. of India. His interested areas of research are system dynamics, rotor dynamics and vehicle dynamics.

second author of the paper is working as Professor in Mechanical Engineering Department in Delhi

Technological University. He got his doctor of philosophy from Indian Institute of Technology, Kharagpur. His main areas of research are rotor dynamics, vehicle dynamics, bio-mechanics and bond graph modeling.

APPENDIX

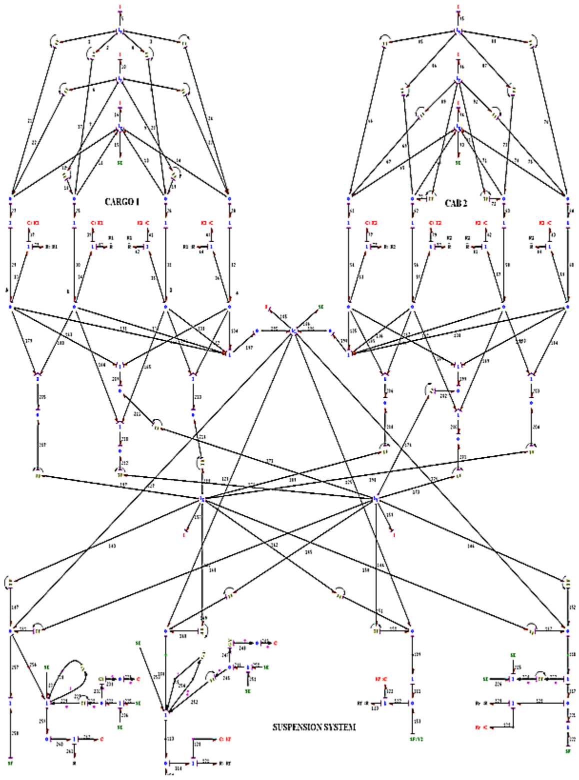

Figure 7: Bond graph model of integrated system

Table 1: Main data

Cab/Engine Mass *m*	2500 kg
Cargo	10,000 kg
Base mass	4250 kg
Mass of tire	500 Kg
Front_suspension_stiff	2.67e+006Ns/m^2
Rear_suspension_stiff	1.15e+006 Ns/m^2
Front_suspension_resistanse	31895 Ns/m
Rear_suspension_resistanse	33884 Ns/m
Length of ground excitation	0.3m
Distance_between_suspension	8m
Base_angular_inertia_X-direction	24000Kg-m^2
Base_angular_inertia_Y-direction	24000Kg-m^2
Cab_angular_inertia_X-direction	2000 Kg-m^2
Cab_angular_inertia_Y-direction	2000 Kg-m^2
Cargo_angular_inertia_X-direction	4500 Kg-m^2
Cargo_angular_inertia_Y-direction	4500 Kg-m^2
Cargo_front_stiffness	1.67e+07 Ns/m^2
Cargo_rear_stiffness	1.67e+07 Ns/m^2
Cargo_rear_resistance	8124 Ns/m
Cargo_front_resistance	8124 Ns/m
Cab_front_stiffness	1e+08 Ns/m^2
Cab_rear_stiffness	1e+08 Ns/m^2
Cab_rear_resistance	1e+06 Ns/m
Cab_front_resistance	1e+06Ns/m
Left/Right suspension distance to the centre of the base	2.28
Left/Right spring-damper system distance to the centre of the cargo	2.28
Front /Rear spring-damper distance to the centre of cargo	3
Front /Rear spring-damper distance to the centre of base	4