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# Comfort Level Refinement of Military Tracked Vehicle Crew through Optimal Control Study

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**ABSTRACT**

Military tracked vehicle and crew are modelled together in this paper as integrated man-machine lumped parameter model, by integrating the simplified 5 degrees of freedom (DoF) tracked vehicle model, including seat and 4 DoF human bio-dynamic model, thus resulting in a 9 DoF simplified vehicle-occupant model. Then the natural frequency of major mass segment namely the chassis mass is obtained through simulation study, for a known road input. The value obtained is compared with that of an earlier research work, for validation of said man-machine model. Then focusing our study locally at crew seat location, parameters of crew seat suspension for ride comfort are optimised using the optimal digital state space controller designed for this purpose by implementing it in a 2 DoF occupant - seat suspension model and its Simulink model constructed. Simulation results illustrate the attainment of the goal by meeting the controller design requirements.

**Keywords:** Vibration responses; Vehicle-occupant system; Seat suspension system; Simulink model; Digital state space controller

## 1. INTRODUCTION

During the literature survey on current topic of discussion, it has been obviously found that vibration measurements carried out at vehicle occupant's seat level alone could not provide exact magnitudes of occupant's exposure to oscillations. Thus it is not a surprise that, such seat suspension designs done without taking the combined effect of the vehicle and crew into account, would not give much benefit. Other surveys done in this area also are in support of this view, which at end stress that combined biomechanical modelling and simulation of parameters of human body and occupant seat, together, is necessary[1]. Accordingly an earlier work was carried out on this by Patil and Palanichamy[1] for typical heavy vehicle viz. tractor, using man-machine integrated model, who finally demonstrated that maximum vibration responses occur only at the bodily parts of the vehicle occupant and not at vehicle level, thus insisting the need for vehicle-occupant composite model study. Accordingly in the current work, man-machine i.e military tracked vehicle – crew integrated model is formulated by integrating the simplified 5 degrees of freedom (DoF) tracked vehicle model, including seat and 4 DoF human bio-dynamic model, which results in a 9 DoF simplified vehicle-occupant model. Then governing equations of motion are obtained for all the lumped masses using Newton's law of motion. Then from the simulation and analysis study carried out, the natural frequency of major mass segment viz. chassis mass is obtained to compare it with the results of earlier research, for validating the model.

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Then continuing our study locally at the crew seat location, parameters of crew seat suspension for ride comfort are optimised using the optimal digital state space controller designed for this purpose by implementing it in a 2 DoF occupant-seat suspension model and its Simulink model constructed.

Lumped Parameter Model (LPM) approach is used in the first part of the current work, because of the fact that the LPM still exists as the choice of many researchers in this area as its validity is established in low frequency range applications i.e below 100 Hz. It is established that an off-road vehicle is subjected to a ride vibration environment, which is mostly of low frequency and large amplitude[2] in nature. As its application is popular in one direction analysis, it has been used in current work, as the present goal is to analyse mainly the vertical vibrations, which is considered to be more harmful to vehicle occupant.

Optimisation of nonlinear quarter car suspension-seat-driver model is carried out by Nagarkar[3], *et al.*, but the study considered only single DoF of vehicle. Stationary response of a military tracked vehicle was analysed by Ravishankar and Sujatha[4]. But bio-dynamic model was not included in the said study. In the work reported by Ramamurthy and Patil[5], for the assessment of crew comfort of military vehicle including development of a seat suspension system, parameter values for said suspension system were adopted from an earlier research work output.

Development of mathematical models, simulating vibration control of tracked vehicle weapon dynamics was carried out by Jakati[6], *et al.* But the developed model has not included occupant's bio-dynamic model. The task of state-space modelling of a mechanical system in Matlab/Simulink was undertaken by Sivak[7], *et al.* But the work discusses about 2 DOF model only.

Optimisation of passive vehicle suspension system by genetic algorithm was carried out by Mitra[8], *et al.* Though the study involves human model, only 2 DoF of vehicle was considered in the study. Modelling, analysis and PID controller implementation on double wishbone suspension using Simmechanics and Simulink was done by Tandel[9], *et al.* But the study does not consider bio-dynamic model.

Investigation of vibration and ride characteristics of 5 DoF vehicle suspension system was undertaken by Rahman[10], *et al.* But the work does not discuss on military vehicles. Semi-active suspension to improve both ride comfort and handling feel was discussed in the work by Takayoshi Fujita[11], et al. But the work was not carried out for military vehicles. Modelling, simulation and control of semi active suspension system for automobiles was done in Matlab Simulink environment by Rao[12], but with PID controller and not with digital state space controller, as done in the current work.

## 2. FORMULATION OF VEHICLE-CREW INTEGRATED LP MODEL

The biodynamic lumped (human body parts) model of 4 DoF, suggested by Nagarkar[3], *et al.* is adopted here and integrated with the simplified 5 DoF half plane vehicle model including crew seat to arrive at the man-machine i.e vehicle occupant integrated Lumped Parameter (LP) model of Fig. 1.

The nomenclature to illustrate what each individual part of biodynamic portion of integrated model of Fig. 1 represents, is given as follows. In the original biodynamic model proposed by Nagarkar[3], et al., four separate masses make the human body. One set of spring and damper connects the set of two body parts. Thus totally five spring – damper pairs have been considered in the said model. However, the notations of said 4 DoF biodynamic model are slightly changed in the current work. As per the revised notations of current model, $m1$ corresponds to the human body part *viz.,* 'head' and $m2$ denotes the body part *viz.,* 'chest combined with torso-upper portion'. Similarly, $m3$ corresponds to 'torso-lower portion' and $m4$, to 'thigh combined with pelvis'.

Now, $K1$ denotes stiffness of $m1$ and $C1$ denotes the damping coefficient of $m1$. Similarly, $K2$, $K3$, and $K4$ respectively denote the stiffness values of masses $m2$, $m3$, and $m4$ and $C2$, $C3$, and $C4$ denote, respectively, the damping coefficients of masses. The vertical displacements experienced by body parts *viz.,* $m1$, $m2$, $m3$, and $m4$, about the c.g of chassis (sprung) mass are given by $Y1$, $Y2$, $Y3$, and $Y4$, respectively. In the Fig. 1, $a1$ and $a2$ refer to the road input vertical displacement of virtual wheels 1 and 2 ($m$), which are defined in art 3.

Now, the nomenclature to illustrate what the individual parts of vehicle portion of integrated model of Fig. 1 represent alongwith the description of variables is as given in Table 1.

While formulating the said half plane vehicle model, virtual front and rear wheel stations are visualised by arriving at the equivalent suspension parameter values from those of the actual seven wheel stations of vehicle. Now, the parameters of biomechanical model of occupant are given in Table 2.

From the values illustrated in Table 2, it may be found that the total mass of seated Crew body parts is 55.20 kg. Table 3 illustrates the parameters of the AFV model obtained from literature and reports[13,14].

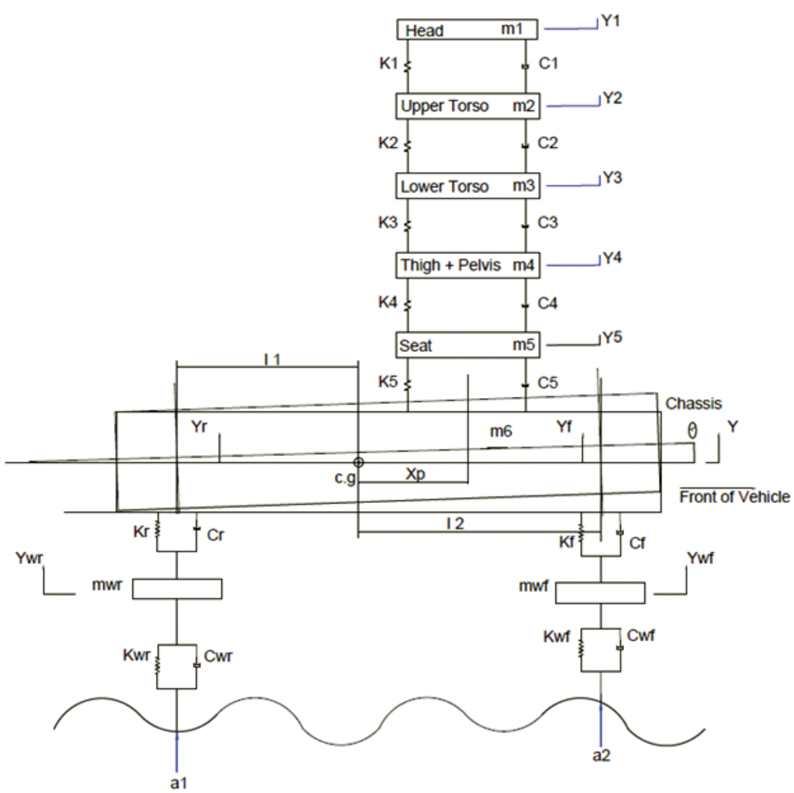


**Figure 1. Vehicle-Crew integrated two-wheel station half plane LP model.**

Governing equations for the various masses comprise of different sets of terms. One set of these terms corresponds to the inertial forces of the springs and dashpots. Other set of terms denote the forces exerted by these connecting members on various masses, due to the motion of one mass relative to other one.

## 3. GOVERNING EQUATIONS OF MOTION

Now, the governing equations for various masses of the integrated model are:

**Head**

$$\left(m1.\ddot{Y}1\right)+\left[K1.\left(Y1-Y2\right)\right]+\left[C1.\left(\dot{Y}1-\dot{Y}2\right)\right]=0 \quad (1)$$

**Upper Torso**

$$\left(m2.\ddot{Y}2\right)+\left[K1.\left(Y2-Y1\right)\right]+\left[C1.\left(\dot{Y}2-\dot{Y}1\right)\right] + \left[K2\left(Y2-Y3\right)\right]+\left[C2.\left(\dot{Y}2-\dot{Y}3\right)\right]=0 \quad (2)$$

**Lower Torso**

$$\left(m3.\ddot{Y}3\right)+\left[K2.\left(Y3-Y2\right)\right]+\left[C2.\left(\dot{Y}3-\dot{Y}2\right)\right]+ \left[K3\left(Y3-Y4\right)\right]+\left[C3.\left(\dot{Y}3-\dot{Y}4\right)\right]=0 \quad (3)$$

**Table 1. Description of variables of half plane vehicle model**

| Symbol | Description |
|---|---|
| *m5* | Crew Seat mass (kg) |
| *K*5 | Spring const. of Crew Seat suspension (N/m) |
| *C*5 | Damping coeff. of Crew Seat suspension (N.s/m) |
| *m*6 | Sprung mass of Chassis (kg) |
| *Kf* | Spring const. for virtual suspension unit (front) (kg) |
| *Cf* | Damping coeff. of virtual suspension unit (front) (N.s/m) |
| *Kr* | Spring const. for virtual suspension unit (rear) (N/m) |
| *Cr* | Damping coeff. of virtual suspension unit (rear) (N.s/m) |
| *mwf* | Mass of virtual road wheel (front) with track pad (kg) |
| *Kwf* | Spring const. of virtual road wheel (front) rubber with track pad (N/m) |
| *Cwf* | Damping coeff. of virtual road wheel (front) rubber with track pad (N.s/m) |
| *mwr* | Mass of virtual road wheel(rear) with track pad (kg) |
| *Kwr* | Spring const. of virtual road wheel (front) rubber with track pad (N/m) |
| *Cwr* | Damping coeff. of virtual road wheel (rear) rubber with track pad (N.s/m) |
| *Y*1, *Y*2, *Y*3 &*Y*4 | Vertical displacement of human body parts viz. head, chest & upper torso, lower torso and thigh & pelvis, respectively (m) |
| *Y*5 | Vertical displacement of Crew Seat (m) |
| *Y* | Vertical displacement of c.g (m) |
| *Yf* and *Yr* | Vertical displacement of chassis mass directly above the virtual road wheels front & rear respectively (m) |
| *Ywf* and *Ywr* | Vertical displacement of wheels at virtual road wheel stations front & rear respectively (m) |
| *a*1 and a*2* | Road input displacement to virtual wheels front & rear respectively (m) |
| *g* | Acceleration due to gravity ($m/s_2$) |
| *Iy* | Mass M.I of Chassis mass about pitch axis for half plane mass ($kg.m^2$) |
| θ | Pitch displacement of chassis (sprung) mass about vehicle c.g (rad.) |
| *l*1 and *l*2 | Distance from vertical C.L of virtual wheels rear & front respectively to the c.g of chassis (sprung) mass (m) |
| *Xp* | Horizontal distance between centre of seat and c.g of chassis (m) |

**Table 2. The biomechanical parameters of occupant model, as adopted from the model of Nagarkar[3], *et al.***

| Mass (kg) | Damping coeff. (N.s/m) | Spring constant (N/m) | Mass (kg) | Damping coeff. (N.s/m) | Spring constant (N/m) |
|---|---|---|---|---|---|
| m1 = 5.31 | C1 = 400 | K1 = 310000 | m3 = 8.62 | C3 = 330 | K3 = 162800 |
| m2 = 28.49 | C2 = 200 | K2 = 183000 | m4 =12.78 | C4 = 909.1 | K4 = 90000 |

**Table 3. Parameters of Half plane vehicle model as adopted from AMCP[13] & Banerjee[14], *et al.***

| Parameter | Value |
|---|---|
| *m6* | 18473 (kg) |
| *m5* | 10 (kg) |
| *mwf* = *mwr* | 2012.5 (kg) |
| *Kf* = *Kr* | 0.51905 x $10^5$ (N/m) |
| *K5* | 18000 (N/m) |
| *Cf* = *Cr* | 1.043658 x $10^6$ (N.S/m) |
| *C5* | 200 (N.S/m) |
| *Kwf* = *Kwr* | 15750000 (N/m) |
| *Cwf* = *Cwr* | 20275.5 (N.S/m) |
| *Xp* | 0 |
| *Iy* | 1.94x$10^5$ ($kg.m^2$) |
| *l*1 | 2.5 (m) |
| *l2* | 2.5 (m) |
| *g* | 9.81 ($m/s^2$) |

**Thigh + Pelvis**

$$\left(m4.\ddot{Y}4\right)+\left[K3.\left(Y4-Y3\right)\right]+\left[C3.\left(\dot{Y}4-\dot{Y}3\right)\right] +\left[K4.\left(Y4-Y5\right)\right]+\left[C4.\left(\dot{Y}4-\dot{Y}5\right)\right]=0 \tag{4}$$

**Crew Seat**

$$m5.\ddot{Y}5+\{K5.[Y5-Y-(Xp.\theta)]\} + \{C5.[\dot{Y}5-\dot{Y}-(Xp.\dot{\theta})]\} +[K4.(Y5-Y4)]+C4.\left(\dot{Y}5-\dot{Y}4\right)=0 \tag{5}$$

**Chassis (Bounce)**

$$m6.\ddot{Y}+[Kr.(Y+l1.\theta-Ywr)]+[Cr.(\dot{Y}+l1.\dot{\theta}-\dot{Y}wr)] +[Kf.(Y-l2.\theta-Ywf)]+[Cf.(\dot{Y}-l2.\dot{\theta}-\dot{Y}wf)] +[K5.(Y-Y5-Xp.\theta)]+[C5.(\dot{Y}-\dot{Y}5-Xp.\dot{\theta})]=0 \tag{6}$$

**Chassis (Pitch)**

$$Iy.\ddot{\theta}+[l1.Kr.(Y+l1.\theta-Ywr)]+[l1.Cr.(\dot{Y}+l1.\dot{\theta}-\dot{Y}wr)] +[l2.Kf.(Y-l2.\theta-Ywf)]+[l2.Cf.(\dot{Y}-l2.\dot{\theta}-\dot{Y}wf)] -[K5.Xp.(Y-Y5-Xp.\theta)]-[C5.Xp.(\dot{Y}-\dot{Y}5-Xp.\dot{\theta})]=0 \tag{7}$$

**Vertical displacement of virtual wheels**

**Virtual wheel rear**

$$mwr.\ddot{Y}wr+[Kwr.(Ywr-a1)]+[Cwr.(\dot{Y}wr-\dot{a}1)]+[Kr.(Ywr-Yr)] +[Cr.(\dot{Y}wr-\dot{Y}r)]=Kwr.A.\sin\omega t+Cwr.A.\omega.\cos\omega t \tag{8}$$

**Virtual wheel front**

$$(mwf\ .\ddot{Y}wf)+\left[Kwrf.(Ywf\ -a2)\right]+\left[Cwf\ .(\dot{Y}wf\ -\dot{a}2)\right]+\left[Kf\ .(Ywf\ -Yf)\right]$$
$$+\left[Cf\ .(\dot{Y}wf\ -\dot{Y}f)\right]=Kwf.A.\sin(\omega(t-\alpha))+Cwf.A.\omega.\cos(\omega(t-\alpha)) \quad (9)$$

The set of Auxiliary equations for chassis & virtual wheels (used while deriving their bounce and pitch equations above), for small values of angle $'\theta'$ are

$$Yr = Y + l1.\sin\theta = Y + l1.\theta \text{ and}$$
$$\dot{Y}r = \dot{Y} + (l1.\cos\theta).\dot{\theta} = \dot{Y} + l1.\dot{\theta}$$
$$Yf = Y - l2.\sin\theta = Y - l2.\theta \text{ and}$$
$$\dot{Y}f = \dot{Y} + (l2.\cos\theta).\dot{\theta} = \dot{Y} - l2.\dot{\theta} \quad (10)$$

Following state variables have been used

$$Y1 = X1; \dot{Y}1 = X2; Y2 = X3; \dot{Y}2 = X4;$$
$$Y3 = X5; \dot{Y}3 = X6; Y4 = X7; \dot{Y}4 = X8;$$
$$Y5 = X9; \dot{Y}5 = X10; Y = X11; \dot{Y} = X12;$$
$$\theta = X13; \dot{\theta} = X14; Ywr = X15; \dot{Y}wr = X16;$$
$$Ywf = X17; \dot{Y}wf = X18;$$

Above variables are now substituted in the governing Eqns. (1) to (10). Then the equations are written in state space form as

$$\dot{X} = A.X + B.Q \quad (11)$$

where system matrix

$$A = [A1\ A2\ A3\ A4\ A5\ A6\ A7\ A8\ A9\ A10$$
$$A11\ A12\ A13\ A14\ A15\ A16\ A17\ A18]^T$$

State matrix

$$X = [X1\ X2\ X3\ X4\ X5\ X6\ X7\ X8\ X9\ X10$$
$$X11\ X12\ X13\ X14\ X15\ X16\ X17\ X18]^T$$

Input matrix $B = [B1\ B2]$

where as

$$B1 = \left[0\,0\,0\,0\,0\,0\,0\,0\,0\,0\,0\,0\,0\,0\,0\,(Kwr/mwr)\,0\,0\right]^T$$

$$B2 = \left[0\,0\,0\,0\,0\,0\,0\,0\,0\,0\,0\,0\,0\,0\,0\,0\,0\,(Kwf/mwf)\right]^T$$

Input (Scalar) $Q = [a1\ a2]^T$

where $a1$ & $a2$ refer to the road input vertical displacement of virtual wheels 1 and 2 (m), as mentioned earlier, which are defined such that

$$a1 = A'.\sin(\omega t); a2 = A'.\sin(\omega(t-\alpha));$$

Here $A'$ represents the amplitude of sinusoidal road profile considered for simulation.

Now the sinusoidal road profile considered by Shirahatt[15], *et al.*, is updated here, with the amplitude and pitch values of typical sinusoidal profile used for military vehicle simulations, as in the current work, with amplitude $A'$ = 0.15 m and pitch $\lambda$ = 4 m as shown below. A vehicle velocity of $V$ = 11.11 m/s. is also used for a vehicle speed of 40 km/hr. Here, $\omega$ refers to the circular frequency of the displacement applied (rad / sec.) and $\alpha$ denotes the time lag between the front and rear road wheels to reach the same road profile. Thus, the rear virtual wheel crosses the same road profile of the first virtual wheel after a time delay of $\alpha$ which may be defined,

$$\alpha = \frac{(l1+l2)}{V} = \frac{(2.5+2.5)}{11.11} = 0.45 \text{ sec.}$$

Circular frequency of the displacement force applied i.e $\omega$ may be derived, for the above vehicle velocity as

$$\omega = 2.\pi.f = 2.\pi.\left(\frac{V}{\lambda}\right) = 2.\pi.(11.11/4) = 17.45 \text{ rad/s.}$$

Now, for typical sinusoidal road profile conditions of current case, displacement of front and rear wheels, as function of time will become[15]

$$a1(t) = \begin{cases} \frac{A}{2}(1-Cos(\omega t), \textit{ if } 0 \le t \le \frac{2\lambda}{V} \textit{ and} \\ 0, \textit{ otherwise} \end{cases}$$

$$a2(t) = \begin{cases} \frac{A}{2}(1-Cos(\omega(t-\alpha)), \textit{ if } \alpha \le t \le \alpha \pm 2\lambda/V \\ 0, \textit{ otherwise} \end{cases}$$

Now the system matrices may be defined from the governing equations formed, as below

$$A1 = [01\,0\,0\,0\,0\,0\,0\,0\,0\,0\,0\,0\,0\,0\,0\,0\,0]$$

$$A2 = [(-K1/m1)(-C1/m1)(K1/m1)$$
$$(C1/m1)0\,0\,0\,0\,0\,0\,0\,0\,0\,0\,0\,0\,0\,0]^T$$

$$A3 = [00\,01\,0\,0\,0\,0\,0\,0\,0\,0\,0\,0\,0\,0\,0\,0]$$

$$A4 = [(K1/m2)(C1/m2)((-K1-K2)/m2)$$
$$((-C1-C2)/m2)(K2/m2)$$
$$(C2/m2)0\,0\,0\,0\,0\,0\,0\,0\,0\,0\,0\,0]$$

$$A5 = [0\,0\,0\,0\,0\,1\,0\,0\,0\,0\,0\,0\,0\,0\,0\,0\,0\,0]$$

$$A6 = [0\,0\,(K2/m3)(C2/m3)((-K2-K3)/m3)$$
$$((-C2-C3)/m3)(K3/m3)$$
$$(C3/m3)0\,0\,0\,0\,0\,0\,0\,0\,0\,0]$$

$$A7 = [0\,0\,0\,0\,0\,0\,0\,1\,0\,0\,0\,0\,0\,0\,0\,0\,0\,0]$$

$$A8 = [0\,0\,0\,0\,(K3/m4)(C3/m4)((-K3-K4)/m4)$$
$$((-C3-C4)/m4)(K4/m4)(C4/m4)0\,0\,0\,0\,0\,0\,0\,0]$$

$$A9 = [0\,0\,0\,0\,0\,0\,0\,0\,0\,1\,0\,0\,0\,0\,0\,0\,0\,0]$$

$$A10 = [0\,0\,0\,0\,0\,0\,(K4/m5)(C4/m5)$$
$$((-K4-K5/m5)((-C4-C5/m5)$$
$$(K5/m5)(C5/m5)(K5.Xp/m5)$$
$$(C5.Xp/m5)0\,0\,0\,0]$$

$$A11 = [0\,0\,0\,0\,0\,0\,0\,0\,0\,0\,0\,0\,1\,0\,0\,0\,0\,0\,0]$$

$$A12 = [0\,0\,0\,0\,0\,0\,0\,0\,(K5/m6)(C5/m6)$$
$$((\text{-}Kf - Kr - K5)/m6)((\text{-}Cf - Cr - C5)/m6)$$
$$((K5*Xp - Kr*l1 + Kf*l2)/m6)$$
$$((C5*Xp - Cr*l1 + Cf*l2)/m6)\,(Kr/m6)$$
$$(Cr/m6)(Kf/m6)(Cf/m6)]$$

$$A13 = [0\,0\,0\,0\,0\,0\,0\,0\,0\,0\,0\,0\,0\,1\,0\,0\,0\,0]$$

$$A14 = [0\,0\,0\,0\,0\,0\,0\,0\,(\text{-}K5*Xp/Iy)(\text{-}C5*Xp/Iy)$$
$$((K5*Xp - Kr*l1 - Kf*l2)/Iy)$$
$$((C5*Xp - Cr*l1 - Cf*l2)/Iy)$$
$$((\text{-}Kr*l1*l1 + Kf*l2*l2 - K5*Xp*Xp)/Iy)$$
$$((\text{-}Cr*l1*l1 + Cf*l2*l2 - C5*Xp*Xp)/Iy)$$
$$(Kr*l1/Iy)(Cr*l1/Iy)(Kf*l2/Iy)(Cf*l2/Iy)]$$

$$A15 = [0\,0\,0\,0\,0\,0\,0\,0\,0\,0\,0\,0\,0\,0\,0\,1\,0\,0]$$

$$A16 = [0\,0\,0\,0\,0\,0\,0\,0\,0\,0\,(Kr/mwr)(Cr/mwr)$$
$$(Kr.l1/mwr)(Cr.l1/mwr)$$
$$((\text{-}Kr - Kwr)/mwr)((\text{-}Cr - Cwr)/mwr)0\,0]$$

$$A17 = [0\,0\,0\,0\,0\,0\,0\,0\,0\,0\,0\,0\,0\,0\,0\,0\,0\,1]$$

$$A18 = [0\,0\,0\,0\,0\,0\,0\,0\,0\,0\,(Kf/mwf)(Cf/mwf)$$
$$(\text{-}Kf.l2/mwf)(\text{-}Cf.l2/mwf)0\quad 0$$
$$((\text{-}Kf - Kwf)/mwf)((\text{-}Cf - Cwf)/mwf)]$$

## 4. NUMERICAL SIMULATION, ANALYSIS AND VALIDATION

Integrated model of Fig. 1 is subjected to idealised road inputs representing the ground reaction forces, during simulation. To validate the formulated model, the value of natural frequency of selected lumped mass viz. chassis is obtained, through analysis, from the governing equations formed, using Matlab 2014a[16] for comparison with the value established in an earlier research work. During the Matlab analysis phase, the approach suggested by Michael[17] is followed, wherein the transfer functions are used which are obtained from the state space variables formed based on the governing equations of motion (1) to (10) of the integrated model. The natural frequency of selected major lumped mass viz. the sprung mass i.e chassis of the tracked military vehicle thus obtained is compared with earlier research finding, to validate the model, as explained in results section.

## 5. OPTIMAL CONTROL OF CREW SEAT SUSPENSION SYSTEM

Now, let us focus our study on crew seat location. A 2 DoF seat and occupant model is formulated as shown in Fig. 2.

With reference to Fig. 2, the terms *ms, mus, Ks, Kus, Cs,* and *Cus* correspond to the sprung (seated occupant) mass, un sprung (crew seat mass), stiffness of sprung mass, stiffness of un sprung mass, damping coefficient of sprung mass and that of un sprung mass respectively. *u* refers to the control force. *U1, U2,* and *U* refer to the displacements of sprung & un sprung masses and the road disturbance, respectively.

First the Simulink model is constructed to represent the above system as shown in Fig. 3.

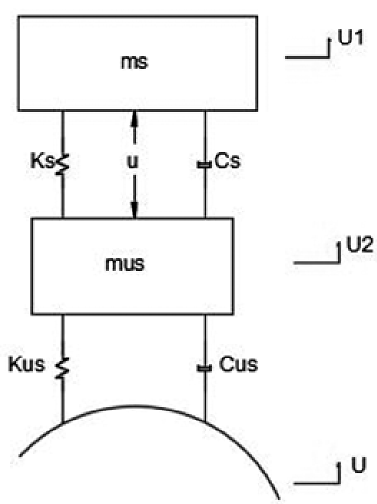


**Figure 2. Two DoF crew – seat suspension model.**

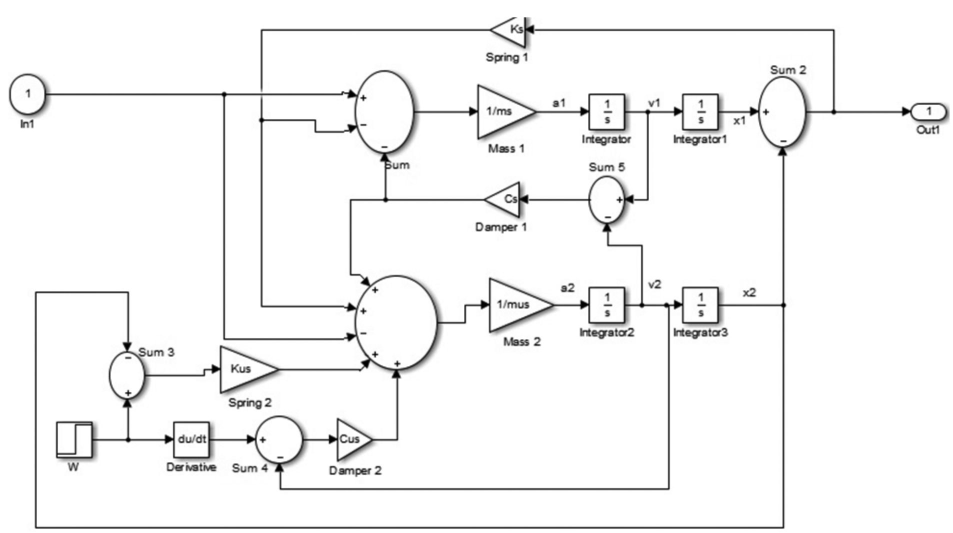


**Figure 3. Simulink model for the 2 DoF crew-seat suspension system.**

Then the Simulink model is extracted into Matlab using suitable code. Now to formulate the digital controller to optimally control the crew seat suspension parameters, the design requirements are to be set as follows: output *U1 – U2* is to have an overshoot such that it is below 5 per cent and also the settling time is to be shorter than 5 s, for a typical Step input road disturbance of *U*. This means that if simulation is done such that the seat suspension mass is considered as un sprung mass and runs on to a step with height of 0.01 *m*, then it may be observed that the said unsprung mass will experience to and fro motion in the range of + 0.005 *m* to -0.005 *m*. Also it may be noted that the crew seat and hence the crew will come back to the comfort zone in a time span of just 5 s[18]. The nominal values of seat suspension parameters are provided from earlier studies as

*ms* = 61 kg; *mus* = 10 kg; *Ks* = 49340 N/m; *Kus* = 90000 N/m; *Cs* = 2475 N.s/m, *Cus* = 400 N.s/m

Now as part of said state space digital controller design, a sampling time of 0.0005 s is set. Then continuous to discrete conversion is done using the appropriate Matlab function. Then the integral control is added and finally the controller design is completed for the current application.

## 6. RESULTS AND DISCUSSION

### 6.1 Validation of Mathematical Model Formulated

Value of natural frequency *fn* obtained through Matlab analysis from the governing equations formed, for sprung mass *m6* i.e Chassis of vehicle using the formulated half plane vehicle - Crew integrated model is 1.6731 Hz. The value of natural frequency of vehicle Chassis established by the earlier research work[19] is 1.2449 Hz. From the reasonable agreement between the two results, it is observed that validity of formulated model is established.

### 6.2 Optimal control of crew seat suspension system

The result of simulating the closed loop response using the designed optimal digital state space controller is given in Fig. 4. Now, by looking into graphical output, we can find that less than 0.004 *m* is the overshoot. Also it may be noted that within a time span of 5 s itself, the response comes to a settle, for the selected gain values. These meet our design requirements.

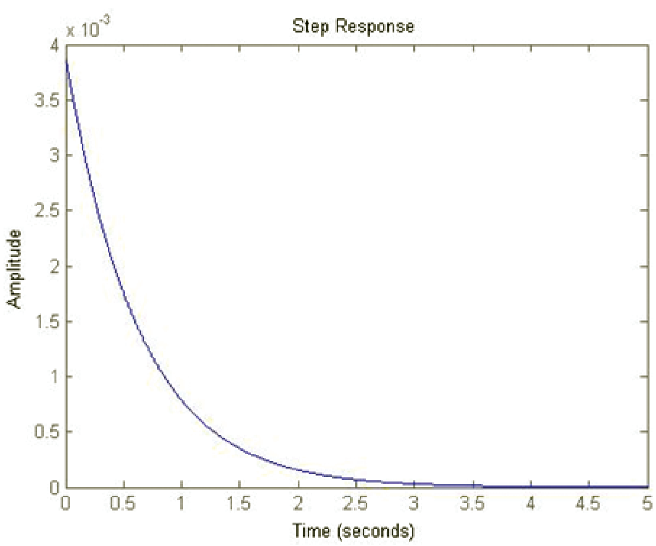


**Figure 4. Results of simulating closed-loop response.**

The screen shot showing the chosen gain values and closed-loop poles, is as given in Fig. 5.

```
d =
      u1  u2
   y1  0   0

Sample time: 0.0005 seconds
Discrete-time state-space model.


z =

  -0.9477 + 0.0000i
   0.9984 + 0.0177i
   0.9984 - 0.0177i


K =

   1.0e+07 *

    0.1414    0.0000    2.5219    0.0033    5.7996

>>
```

**Figure 5. Screen shot showing gain values for closed-loop response simulation.**

## 7. CONCLUSIONS AND FUTURE SCOPE

In the current work, first a two-wheel station simplified man-machine model is formulated by integrating the bio-dynamic model of crew and the simplified two-wheel station half plane model of tracked military vehicle. Though the simplified two-wheel station tracked vehicle model resembles the on-road vehicle model, in configuration, it gives reasonably reliable results, once the parameters are updated with that of former one. This is in support of Pazooki[20], *et al.*, who suggest that suspension systems developed for road vehicles can be adapted to off-road vehicles with appropriate modifications. Said simplified two wheel station model of military tracked vehicle is visualised, also based on the work by Banerjee[21], *et al.*, who have disclosed the feasibility of single station modelling of multiple stationed tracked vehicle. During model formulation, energy dissipating dampers are represented following the established procedures suggested by Silva[22]. Then using the formulated model, natural frequency of major mass segment viz. chassis mass is obtained through the current approach involving simulation and analysis, and compared with the result established by an earlier research work[19]. From the reasonable agreement between the two results, validity of formulated model is established. However to get more precise results, more detailed model involving more number of DoF may be considered for future works.

Then by focusing our study on the crew seat suspension system, a two DoF model is formulated and seat suspension parameters for ride comfort are optimised using the designed optimal digital state space controller i.e the study is focused here on crew seat location and the seat suspension parameters are optimised using optimal control study approach. Such a cost-efficient solution of locally reducing the vibration response i.e at seat level for comfort level improvement is also supported by Gan[23], *et al.*, who stipulate that, cancelling the vibration at seat itself is gaining impetus now-a-days, due to the reasons that: (i) seat is the part to which the vehicle occupant makes direct contacts and (ii) local reduction of occupant's vibration level in vehicles involves lesser difficulty and energy consumption, when compared to the strategy of controlling the level of vibration globally, in such vehicles.

The optimal seat suspension parameter values obtained in current work, may be impelemented in the design of seat suspension system of Crew of military tracked vehicle, to achieve further refinement in his comfort level. Aso the optimal control study strategy may be extended to optimise other parameter values by formulating more detailed model and validating the same through case study, as future work.

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