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OPTIMAL PREVIEW CONTROL OF TRACKED VEHICLE SUSPENSION SYSTEMS

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ABSTRACT−In this paper, an optimal suspension system with preview of the road input is synthesized for a half tracked vehicle. The main goal of this research is to improve the ride comfort characteristics of a fast moving tracked vehicle in order to maintain the driver's driving capability. Several different kinds of preview control algorithms are evaluated with active or semi-active suspension systems. The road information estimated from the motion of the 1st road-wheel is adequate to make the best use of the preview control algorithm for tracked vehicles. The ride-comfort characteristics of the tracked vehicle are more dependent on pitching angular acceleration than heaving acceleration. The pitching motion is reduced by the suspension system with hard outer suspensions and soft inner suspensions. Simulation results show that the performance of sky-hook algorithms for ride comfort nearly follow that of full state feedback algorithms.

KEY WORDS : Tracked vehicle, Preview information, Vibratory characteristic evaluation, Frequency response characteristics

NOMENCLATURE

a_i : distance from mass center to suspension, m
I : moment of inertia, kg·m^2
k_i : i-th suspension stiffness, N/m
M : body mass, kg
m : operator's mass, kg
u_i : control force of i-th actuator, N
v : vehicle speed, m/sec
V : vertical velocity, m/sec
v_i : i-th variable damping coefficient, N·sec/m
z_c : vertical position of mass center, m
z_{0i} : vertical position of i-th road-wheel, m
θ : pitch angle about mass center, rad
ρ_i : weighting factors in performance index
τ_{i-1} : delayed time from 1st to i-th road-wheel, sec

1. INTRODUCTION

In recent years, active or semi-active suspensions with preview of road disturbances have been studied to make use of this type of technology in practice (Thompson and Pearce, 1998). Early work on prview suspension includes Bender (1968), Tomizuka (1976) and Hac (1992). It has been demonstrated that the presence of preview information can simultaneously improve all aspects of performance (Youn, 1992). The preview information about the road disturbance can be obtained easily in case of tracked vehicles by measuring the motions of front road-wheel or suspension because there is no tire deflection to measure. The approaching road disturbance recognized as preview information will be applied to the road wheels in order from the second to the last wheel. Early research on tracked vehicle suspensions evaluated the performance simulated by semi-active control algorithms (Youn *et al.*, 2001).

Military tracked vehicles like tanks or armored personnel carriers, which traverse over rough off-road terrain, demand strong driving force, maneuverability, durability, ride comfort and the like. Mathematical modeling of track-wheel-terrain interaction was studied to compute the track tension and the normal and shear forces at the track-terrain interface for the dynamic simulation of tracked vehicles (Ma and Perkins, 2002). In this paper, the fast moving tracked vehicle is studied from the viewpoint of the improvement of ride comfort to ensure satisfactory driving environment and to protect electric devices from vibration. To analyze and evaluate the dynamic behavior of a tracked vehicle, the circumstances and conditions for performance evaluation may be considered separately for the following two cases. The first case concerns the effect that occurs at impact with the

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ground after the vehicle track drifts away from the surface. In this case, modeling and estimation of track tension is important because the dynamic characteristics are influenced considerably by track tension. In the second case, the track is in contact with the ground. In this case, it is not necessary to consider the track tension in the model to evaluiate vibratory characteristics in the vertical direction. This research focuses on the second case scenario only.

This paper deals with the dynamic performance evaluation of a half tracked vehicle model with 6 active or 6 semi-active suspension units operated by an optimal finite preview controller. The control algorithms for suspension systems, such as full state feedback active, full state feedback semi-active, sky-hook active, and sky-hook semi-active suspension systems, are evaluated and analyzed with respect to ride comfort. The proposed controller utilizes information about approaching road disturbances estimated from the dynamic motion of the front wheel. The dynamic performances of the vehicle with ride comfort preference are evaluated with performance indices, vibratory characteristic evaluation curves and frequency characteristic curves obtained from Matlab simulations.

2. PROBLEM FORMULATION

In this section an optimal preview control problem for a half tracked vehicle model with active suspensions is formulated. The vehicle system model is shown in Figure 1.

Dynamics of the system in Figure 1 is described by

$$M\ddot{z}_c = \sum_{i=1}^{6} R_i \tag{1a}$$

$$I\ddot{\theta} = \sum_{i=1}^{3} a_i R_i - \sum_{j=4}^{6} a_j R_j \tag{1b}$$

where

$$\begin{aligned} R_i &= k_i[z_{0i} - z_c + a_i\theta] + u_i, \quad i = 1, 2, 3 \\ R_i &= k_i[z_{0i} - z_c - a_i\theta] + u_i, \quad i = 4, 5, 6 \end{aligned} \tag{2}$$

where all variables are as defined at the beginning under nomenclature.

The system states and road velocity disturbances are as follows:

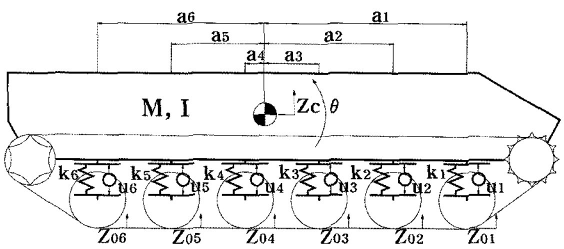


Figure 1. A half tracked vehicle model with 6 suspension units.

Table 1. Parameter values of a half vehicle model used for simulation.

Parameters	Values	Units
M	12420	kg
I	36500.6495	kg·m^2
$k_1, k_2, k_3, k_4, k_5, k_6$	250,000	N/m
a_i	|(3.656-i)*0.8|	m
$v_{i\,\min}$	1000	N·s/m
$v_{i\,\max}$	100000	N·s/m
$v_{i\,\min} = v_{i\,\max}$ (Passive)	10000	N·s/m

$$\begin{aligned} x_1 &= z_c + a_1\theta - z_{01} & x_5 &= z_c + a_4\theta - z_{04} \\ x_2 &= \dot{z}_c + a_1\dot{\theta} & x_6 &= z_c + a_5\theta - z_{05} \\ x_3 &= z_c + a_2\theta - z_{02} & x_7 &= z_c + a_6\theta - z_{06} \\ x_4 &= z_c + a_3\theta - z_{03} & x_8 &= \dot{z}_c + a_6\dot{\theta} \end{aligned} \tag{3}$$

$$w_i = \dot{z}_{0i} = w_1(t - \tau_{i-1}) \qquad i = 1, \ldots, 6 \tag{4}$$

where τ_{i-1} is the time delay of the road disturbance input applied to the i-th road-wheel relative to the fore-most wheel and v is the vehicle speed.

$$\tau_{i-1} = \begin{bmatrix} \dfrac{a_1 - a_i}{v} & \text{for } i = 1, 2, 3 \\ \dfrac{a_1 + a_i}{v} & \text{for } i = 4, 5, 6 \end{bmatrix} \tag{5}$$

x_1, x_3, x_4, x_5, x_6 and x_7 represent the suspension deflections in order from the first to the 6th one. x_2 and x_8 stand for the vertical velocities of the first and the last suspension mounting points on the body, respectively.

The equations of motion (1) can be expressed as a system of first order differential equations which can be written as the following vector-matrix state space equation:

$$\dot{x} = Ax + Bu + Dw \tag{6}$$

The synthesized optimal suspension force minimizes a performance index that compromises measures of ride comfort, suspension rattle space and control effort:

$$\begin{aligned} J &= \lim_{T\to\infty}\frac{1}{2T}\int_0^T \Big[\rho_1\ddot{z}_c^2 + \rho_2\ddot{\theta}^2 + \rho_3 x_1^2 \\ &\qquad + \sum_{i=3}^{7}\rho_{i+1}x_i^2 + \sum_{i=1}^{6}\rho_{i+8}u_i^2\Big]dt \\ &= \lim_{T\to\infty}\frac{1}{2T}\int_0^T [x^TQx + 2x^TNu + u^TRu]dt \end{aligned} \tag{7}$$

where Q and R are symmetric matrices, Q is positive semidefinite and R is positive definite. The weighting factors are determined by the control designer. In this investigation, two sets of weighting facctors in Table 2 are considered. The simulation results will indicate that a much better performance can be achieved when the

Table 2. Two types of weighting factors used in the performance indice.

Weighting factor	Targets	1st set	2nd set
ρ_1	Heaving acc.	1	1
ρ_2	Pitching ang. acc.	1	1
ρ_3, ρ_4	Suspension defl.	1000	2000
ρ_4, ρ_7	Suspension defl.	1000	1000
ρ_5, ρ_6	Suspension defl.	1000	200
ρ_9, ρ_{10}, ρ_{11}, ρ_{12}, ρ_{13}, ρ_{14}	Control force	10^{-7}	10^{-7}

weighting factors on suspension deflections are varied (2nd set); to be more, if the outside suspensions are designed to be hard by letting the weighting factors, ρ_3 and ρ_8, large and the inner suspensions to be soft by letting the weighting factors, ρ_5 and ρ_6, small, the dynamic behavior is better than the case of the same weighting value of 1000 for all suspension deflections. Outside and inside are defined as seen from the side of the vehicle.

3. DESIGN OF PREVIEW CONTROLLER

The control law that minimizes the performance index (7) for the system (6) is given by

$$\begin{aligned} u(t) &= -R^{-1}[(N^T + B^T P)x(t) + B^T r(t)] \\ &= -Kx(t) - R^{-1}B^T r(t),\ K = R^{-1}(N^T + B^T P) \end{aligned} \tag{8}$$

where P is obtained by solving the Riccati equation

$$Q_n + A_n^T P + PA_n - PBR^{-1}B^T P = 0 \tag{9}$$

where

$$Q_n = Q - NR^{-1}N^T \text{ and } A_n = A - BR^{-1}N^T$$

and K is the feedback control gain designed for the preference of ride comfort in this work and produces feedback control force by multiplying the state vector. $r(t)$ is computed by preview information about the approaching road surface:

$$\int_0^{t_p+\tau_5} e^{A_c^T \sigma} PD \begin{bmatrix} H(t_p - \sigma)w_1(t+\sigma) \\ H(t_p + \tau_1 - \sigma)w_1(t+\sigma-\tau_1) \\ H(t_p + \tau_2 - \sigma)w_1(t+\sigma-\tau_2) \\ H(t_p + \tau_3 - \sigma)w_1(t+\sigma-\tau_3) \\ H(t_p + \tau_4 - \sigma)w_1(t+\sigma-\tau_4) \\ H(t_p + \tau_5 - \sigma)w_1(t+\sigma-\tau_5) \end{bmatrix} d\sigma \tag{10}$$

where A_c^T is the closed system matrix defined by $A_c^T = (A - BR^{-1}N^T)^T - PBR^{-1}B^T$, and t_p in the above equation is the preview time about the road information approaching to the first road-wheel. In this study t_p is zero and τ_i is as defined by equation (5). The Heaviside function (unit step function) satisfies

$$H(t_p + \tau_i - \sigma)\begin{bmatrix} 1 \text{ for } \sigma \le t_p + \tau_i \\ 0 \text{ for } \sigma > t_p + \tau_i \end{bmatrix} \tag{11}$$

Notice that the first two terms in the performance index are related to the riding comfort, the third and the fourth the rattle space and the fifth the control effort. If the preview information is unreliable, $r(t)$ may be set to zero. The sky-hook active control force is computed by multiplying of state variables to represent absolute velocities (x_2 and x_8) and corresponding control gain elements. Therefore the sky-hook active system is relatively easy to implement because only a few absolute velocity measurement sensors are required. The sky-hook active preview control force is obtained by adding of the above sky-hook active control force and the feed-forward part, $-R^{-1}B^T r(t)$.

4. CONTROLLER DESIGN FOR SEMI-ACTIVE SUSPENSION SYSTEM

The variable damping coefficient $v_i(t)$ in the semi-active system changes in a range between $v_{i\max}$ and $v_{i\min}$. The i-th variable damping coefficient is determined by the following conditions:

$$v_i(t) = \begin{Bmatrix} v_{i\min} & \text{if} & -u_{0i}\dot{x}_j \le v_{i\min}\dot{x}_j^2 \\ v_{i\max} & \text{if} & -u_{0i}\dot{x}_j \ge v_{i\max}\dot{x}_j^2 \\ -u_{0i}/\dot{x}_j & & \text{otherwise} \end{Bmatrix}. \tag{12}$$

where (i, j) = (1, 1), (2, 3), (3, 4), (4, 5), (5, 6), (6, 7). Equation (12) implies that the relative velocities of suspension deflections are required to calculate the variable damping coefficients. The absorbed forces of the variable dampers may be computed by multiplying the variable damping coefficients and the relative velocities of suspensions. These absorbed forces can be expressed by the variable damping coefficient vector, state vector and road velocity vector as follows:

$$u(t) = -(\text{diag } v)[Ex + Fw] \tag{13}$$

where

$$E = \begin{bmatrix} 0 & 1 & 0\,0\,0\,0\,0 & 0 \\ 0 & \frac{(a_6+a_2)}{(a_1+a_6)} & 0\,0\,0\,0\,0 & \frac{(a_1-a_2)}{(a_1+a_6)} \\ 0 & \frac{(a_6+a_3)}{(a_1+a_6)} & 0\,0\,0\,0\,0 & \frac{(a_6-a_3)}{(a_1+a_6)} \\ 0 & \frac{(a_6-a_4)}{(a_1+a_6)} & 0\,0\,0\,0\,0 & \frac{(a_6+a_4)}{(a_1+a_6)} \\ 0 & \frac{(a_6-a_5)}{(a_1+a_6)} & 0\,0\,0\,0\,0 & \frac{(a_6+a_5)}{(a_1+a_6)} \\ 0 & 0 & 0\,0\,0\,0\,0 & 1 \end{bmatrix}$$

and $F = -I$.

The sky-hook semi-active suspension is able to produce only the absorbed forces that the sky-hook active suspension generates.

5. BLOCK DIAGRAM FOR PERFORMANCE EVALUATION

The block diagram constructed to simulate the dynamic performance of the system in Matlab Simulink software is shown in Figure 2. The road surface data are filtered by a low pass filter and differentiated to apply to the road-wheel. The delay times between the first and other road-wheels are obtained by the distance divided by the vehicle speed. The same road velocity input is applied to all the road-wheels after taking the delay time into account. The control forces are composed of the feed-forward term using preview information and the feedback term computed by the present states of the vehicle.

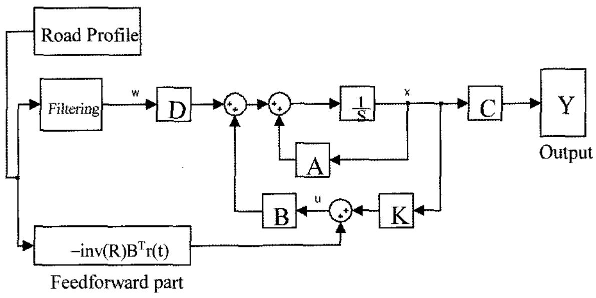


Figure 2. Main block diagram for the simulation.

6. SIMULATION RESULTS

6.1. Comparative Evaluation of Performance Indices

It is assumed that the driver's seat is at 1.5235 m ahead of the mass center. The RMS values of acceleration at the driver, heaving acceleration of mass center, pitching angular acceleration and relative velocities of suspensions that occur when the vehicle travels on the road with the roughness of RMS 2.62 cm are given in Tables 3 and 4. The performance of the passive system is considered as 100%, and the relative performance values of active and semi-active systems, sky-hook active and sky-hook semi-active systems, and a system with preview and a system without preview are comparatively evaluated. The performances in terms of the driver's vertical acceleration (driv. acc.) and total performances (J_{avg}'s) in Table 4 are

Table 3. Components of performance indices obtained after traveling on the RMS 2.62 road with 1st weighting set.

Suspension systems	Driv Acc	$E(\ddot{z}_c^2)$	$E(\ddot{\theta}^2)$	$E(x_1^2)$	$E(x_2^2)$	$E(x_3^2)$	$E(x_4^2)$	$E(x_5^2)$	$E(x_6^2)$	J_{avg}
Passive	100.00	100.00	100.00	100.00	100.00	100.00	100.00	100.00	100.00	100.00
Active	58.88	74.47	63.01	80.16	96.54	207.94	162.29	117.46	88.57	84.72
Active w/preview	41.24	93.02	28.09	52.57	52.26	79.72	91.95	71.13	43.02	50.98
Sky-hook active	59.80	71.59	58.75	81.88	97.48	188.90	145.53	108.85	81.70	81.37
Sky-hook act. w/preview	39.41	93.64	25.82	53.76	55.88	83.71	89.59	71.47	44.28	51.19
Semi-active	73.77	90.52	73.86	87.40	97.13	152.60	125.99	102.63	84.53	87.65
Semi-act. w/preview	57.57	146.52	33.92	56.24	51.17	58.45	86.86	71.21	45.01	58.21
Sky-hook semi-act.	74.23	88.76	71.15	88.61	98.38	146.30	119.40	99.12	81.17	86.12
Sky-hook semi-act. w/pre.	54.71	141.49	32.09	56.58	53.31	59.55	83.36	70.57	45.01	57.36

Table 4. Components of performance indices obtained after traveling on the RMS 2.62 road with 2nd weighting set.

Suspension systems	Driv Acc	$E(\ddot{z}_c^2)$	$E(\ddot{\theta}^2)$	$E(x_1^2)$	$E(x_2^2)$	$E(x_3^2)$	$E(x_4^2)$	$E(x_5^2)$	$E(x_6^2)$	J_{avg}
Passive	100.00	100.00	100.00	100.00	100.00	100.00	100.00	100.00	100.00	100.00
Active	47.16	70.79	50.67	67.77	80.25	192.00	158.11	105.55	70.61	68.50
Active w/preview	35.78	90.14	21.02	45.78	46.48	83.48	86.49	60.83	33.30	42.23
Sky-hook active	50.85	72.15	46.17	75.48	93.33	195.79	138.46	94.53	63.88	69.55
Sky-hook act. w/preview	33.46	83.63	17.79	50.51	58.37	99.89	77.29	56.93	33.44	43.56
Semi-active	61.89	88.77	62.14	75.15	81.32	141.15	124.80	92.28	67.94	73.63
Semi-act. w/preview	51.75	132.35	27.22	49.77	46.36	61.52	77.15	58.26	34.42	47.91
Sky-hook semi-act.	64.62	90.63	59.12	80.80	91.07	149.50	115.57	86.71	64.20	74.93
Sky-hook semi-act. w/pre.	47.96	122.90	24.16	51.77	52.70	69.74	70.76	55.76	34.42	47.69

much better than those in Table 3. This means that the suspension system with the soft inner suspension and hard outer suspension can provide better ride-comfort characteristics than that with identical suspensions. Tables 3 and 4 show that the performances of sky-hook systems almost exactly follow those of nonsky-hook systems as long as the designer prefers ride-comfort. The sky-hook system is practical to implement because only a few sensors are required. The heave acceleration of the system with preview is generally worse than that of the system without preview, while the pitching acceleration of the system with preview is better than that of the system without preview. It should be noted that the pitching motion is important for ride-comfort. It is shown that the preview road information inferred from the motion of the first road-wheel is effective for better dynamic performance of the tracked vehicle. For the next simulations, the second weighting set is used.

6.2. Evaluation of Vibratory Characteristics

The average power absorbed by a driver during the travel of a tracked vehicle is expressed by

$$P_{av} = \lim_{T \to \infty} \int_0^T F(t)V(t)dt \tag{14}$$

where $F(t) = ma(t)$, m is the common mass of a driver weighing 70 kg, $a(t)$ is vertical acceleration and $V(t)$ is vertical velocity at the driver's position. 6 Watt is the maximum absorbed power in order that the driver may be able to keep driving capability when the vehicle is traveling fast on a rough road. In this investigation, test roads with variances of RMS 1.15 cm and RMS 3.05 cm are used. 4 types of suspension systems are tested with or without preview information. Figures 3 and 4 show the absorbed power as a function of the vehicle speed for two types of roads. The figures show that the passive vehicle is able to traverse over the road with RMS 3.05 cm up to a speed of 23.8 KPH, while the active system without preview and the active system with preview can run up to 29 KPH and 31.9 KPH respectively, and the semi-active system without preview and the semi-active system with preview can run up to 26 KPH and 27.3 KPH respectively. The performances of sky-hook systems are almost equal to those of nonsky-hook systems in Figure 3 and 4. The allowable maximum speed of the tracked vehicle for every type of suspension system and the roughness of the road are shown in Figure 5.

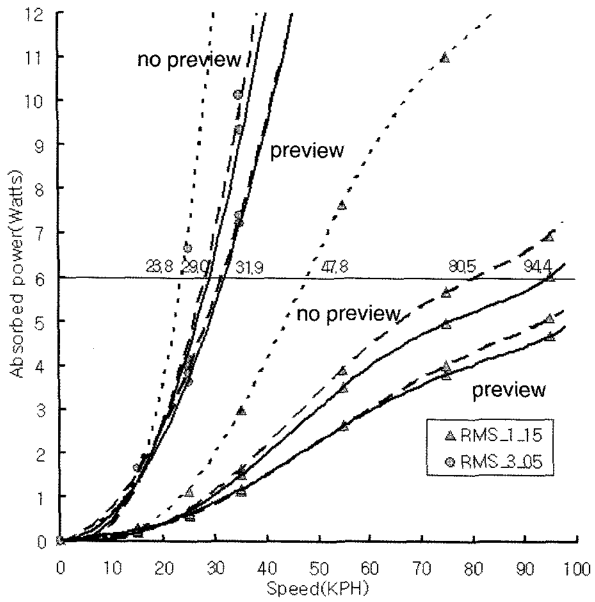


Figure 3. Vibratory performance evaluation curves for: a) passive suspension system: ················; b) full state feedback active suspension system with and without preview: ————; c) sky-hook damper active suspension system with and without preview: – – – – – –.

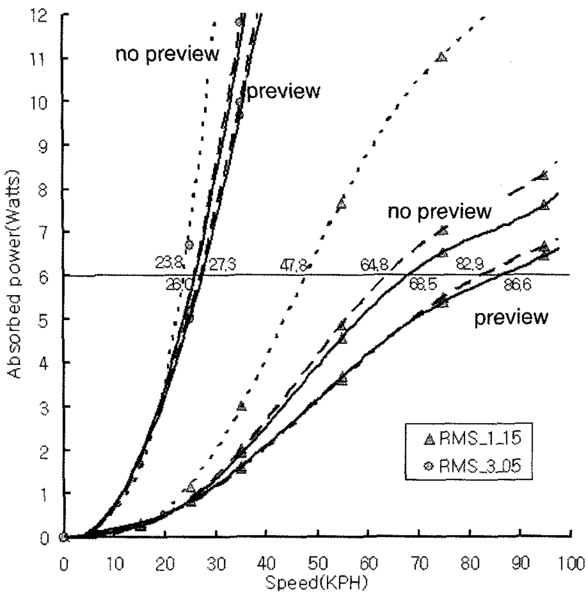
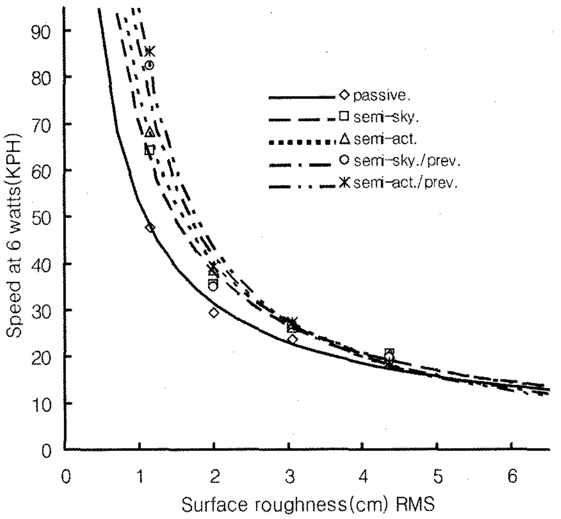


Figure 4. Vibratory performance evaluation curves for: a) passive suspension system: ················; b) full state feedback semi-active suspension system with and without preview: ————; c) sky-hook damper semi-active suspendsion system with and without preview: – – – – – –.

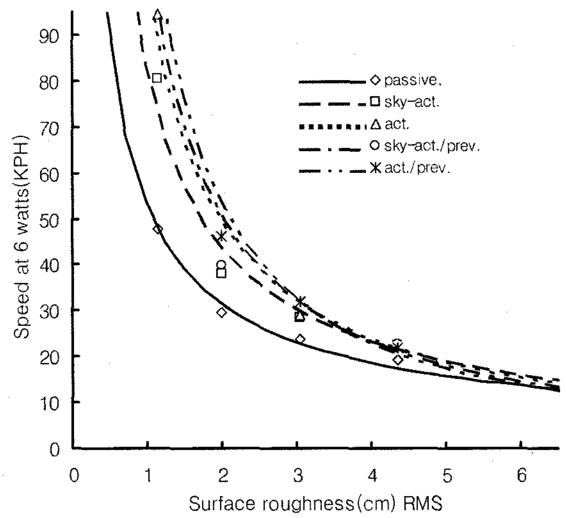
6.3. Frequency Characteristics

Two kinds of visible road profiles applied to 6 road-wheels are shown in Figure 6-a and 6-b to exaggerate the heaving and pitching motion of a tracked vehicle. The frequency varies according to the vehicle speed.

The inputs applied to all wheels are in phase. The same phase input is for heave motion as shown in Figure 6-a, while the difference between the phase angle of the input



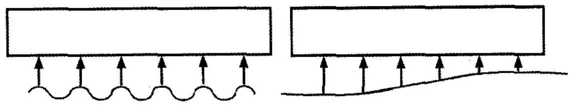
(a)



(b)

Figure 5. Relationship of allowable maximum vehicle speed & road RMS values with respect to: (a) sky-hook semi-active, semi-active, sky-hook semi-active with preview, semi-active with preview, and passive suspension systems; (b) sky-hook active, active, sky-hook active with preview, active with preview, and passive suspension systems.

for the first wheel and that to the last wheel is 180 degree out of phase for pitching motion. The preview information for the i-th road-wheel concerns the road between the 1st road-wheel and the i-th road-wheel. The frequency response characteristics for the linear systems such as active systems can be obtained by the closed loop transfer function for the given system. The absolute values of $|X(j\omega)/w_1(j\omega)|$ are the amplitude ratios of the states and the road velocity:



(a) Heaving input (b) Pitching input

Figure 6. Two types of road disturbances to obtain frequency characteristics.

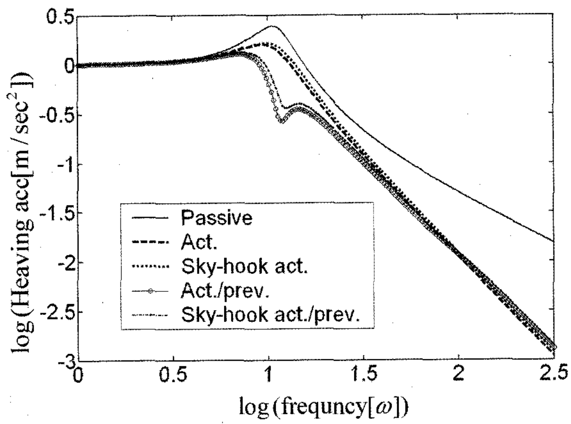


Figure 7. Frequency characteristics with respect to the heaving acceleration response of a body mass center to heaving input.

$$\frac{X(j\omega)}{w_1(j\omega)} = (j\omega I - A_c)^{-1}\{D\cdot[1e^{-j\omega\tau_1}e^{-j\omega\tau_2}e^{-j\omega\tau_3}e^{-j\omega\tau_4}e^{-j\omega\tau_5}]^T$$

$$-BR^{-1}B^T\int_0^{t_p+\tau_5} e^{A_c^T\sigma}PD\begin{bmatrix} H(t_p-\sigma) \\ H(t_p+\tau_1-\sigma)e^{-j\omega\tau_1} \\ H(t_p+\tau_2-\sigma)e^{-j\omega\tau_2} \\ H(t_p+\tau_3-\sigma)e^{-j\omega\tau_3} \\ H(t_p+\tau_4-\sigma)e^{-j\omega\tau_4} \\ H(t_p+\tau_5-\sigma)e^{-j\omega\tau_5} \end{bmatrix} e^{j\omega\sigma}d\sigma\} \tag{15}$$

In the case of heaving input, $e^{-j\omega\tau_i} = 1$ and $\tau_i = 2\pi i/\omega$. On the other hand, $e^{-j\omega\tau_i} = e^{-j\frac{i}{5}\pi}$ and $\tau_i = i\pi/5\omega$ for pitching input where $i = 1, 2, 3, 4, 5$ and τ_i is the delay time between the 1st road-wheel and the (i+1)-th road-wheel which changes according to the frequency or the vehicle speed. The frequency is related to vehicle speed by $2\pi v/0.8=\omega$ for heaving input and $\pi v/4=\omega$ for pitching input. The frequency response curves in Figures 7 and 8 are achieved by equation (15) and verified by the amplitude ratio between input and output at steady states in time domain. Acceleration frequency responses of a body mass center to heaving input in Figure 7 state that the preview control of the tracked vehicle using time delayed road information has great benefits around the vehicle's natural frequency. Frequency response charac-

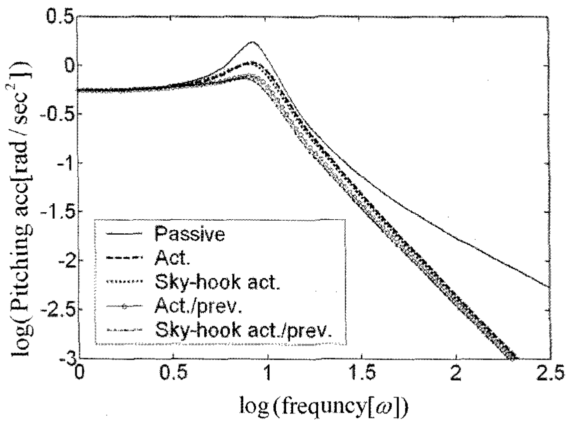


Figure 8. Frequency characteristics with respect to the pitching angular acceleration response of a body to pitching input.

teristics of a body to pitching input with respect to pitching angular acceleration also show considerable improvement in Figure 8. It is clearly shown in Figures 7 and 8 that the performances of sky-hook systems are almost equal to those of nonsky-hook systems.

7. CONCLUSIONS

It is clear in the simulation results that the performances of the sky-hook suspension systems are virtually identical to those of full state feedback suspension systems. The merit of the sky-hook system is to make considerable improvements on ride-comfort characteristics with the output feedback controller using only a few measurements about body accelerations with the exception of suspension deflections. For the vehicle simulated in this paper, the best effect of preview control is accomplished with about 0.15 sec preview information. The road information estimated from the motion of the first road-wheel is adequate to improve the performance of the tracked vehicle because it moves slowly in contrast to wheeled vehicles. The performance of the tracked vehicle with respect to ride-comfort or safety is more dependent on pitch motion than heave motion. Therefore, to reduce the pitch motion, the suspension system should be designed to make the outer suspensions hard and inner soft. Simulations showed that 20–30% performance improvement is achieved by an active system compared with a passive system. Active systems with a preview controller provide 20–30% better performance than active systems without a preview controller. Since the road-wheels almost trace the profiles of the road surface as long as the track does not depart from the ground, the preview information can be obtained by measuring only the absolute position or velocity of the first road-wheel.

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