Simple Active Aerodynamic Suspension Systems for High-Speed Ground Vehicles

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The feasibility of using active aerodynamic surfaces to improve the vertical ride quality of a tracked air cushion vehicle is investigated analytically with a six-degree-of-freedom linear model. Two aerodynamic fins actively control the pitch and plunge steady-state acceleration response of the passenger compartment to harmonic guideway excitations. For speeds of 200 and 300 mph, the choice of simple acceleration feedback signals sensed at the passenger compartment c.g. results in an acceptable ride quality. This is attained by practicable angles of attack and reasonable torques for the aerodynamic surfaces.

Introduction

EXTREMELY high-speed ground transportation systems will play a major role in the nation's transportation program within the next decade. Nearly all high-speed ground transportation vehicles, however, moving at speeds approaching 150–200 mph demand unconventional suspension systems to achieve ride quality, and unique levitation features to significantly reduce resistance to motion. Reducing the resistance to motion has resulted in the evolution of the tracked air cushion vehicle, the magnetically levitated vehicle, and the ram wing vehicle. The achievement of ride quality has resulted in the application of the Controls Configures Vehicle (CCV) concept to rapid transit systems. Passive suspensions and manual operators are simply not up to the task, and the CCV concept must be adopted in vehicle design.[1] The basic CCV technology has been pioneered by the aircraft industry in the form of gust alleviation systems for improving ride quality in high-performance terrain following aircraft. It has been demonstrated to be a very effective way of improving over-all vehicle performance and fatigue life.

Borrowing some of the basic concepts from the aircraft industry, the present paper suggests a somewhat novel approach to the active suspension system design problem for rapid transit systems. An active aerodynamic suspension is proposed which appears to be simple in construction and operation. It has the advantage of reacting mainly with the airstream and not the guideway for response suppression. This avoids guideway-induced deformations and stresses which could be significant in guideway design and maintenance considerations. The proposed aerodynamic suspension consists of small control surfaces in the form of ring wings and V tails located near the ends of the vehicle as illustrated in Fig. 1. At high ground speeds where ride quality is needed these fins can exhibit all of the characteristics of a more conventional active suspension. That is, they exhibit apparent mass, damping, and stiffness characteristics which can be controlled.

The feasibility of employing active aerodynamic control fins to improve the ride quality of a tracked air cushion vehicle traveling over a rigid but wavy guideway is investigated theoretically by using the six-degree-of-freedom model. The functional diagram in Fig. 2 illustrates the interaction of vehicle dynamics with control dynamics as the vehicle responds to guideway and gust excitations. For various assumed control laws for each aerodynamic fin, the acceleration response of each end of the vehicle is de-

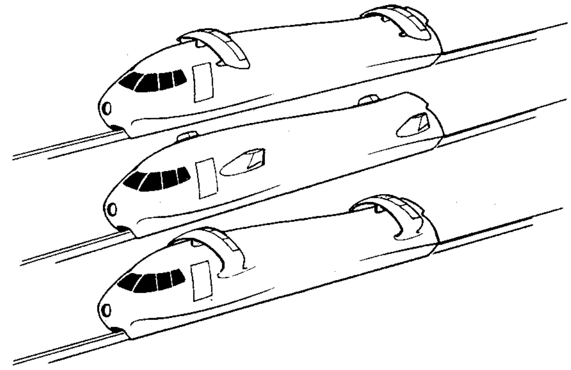

Fig. 1 Some active aerodynamic control configurations for high-speed ground vehicles.

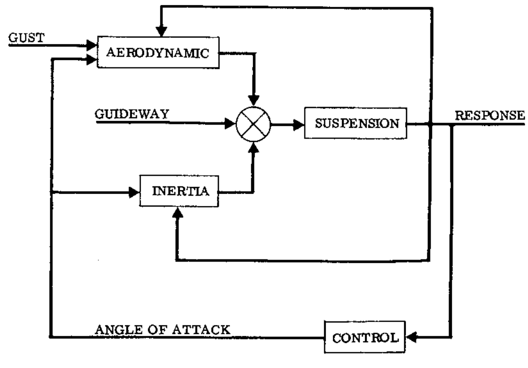

Fig. 2 A functional diagram of a tracked air-cushion vehicle system.

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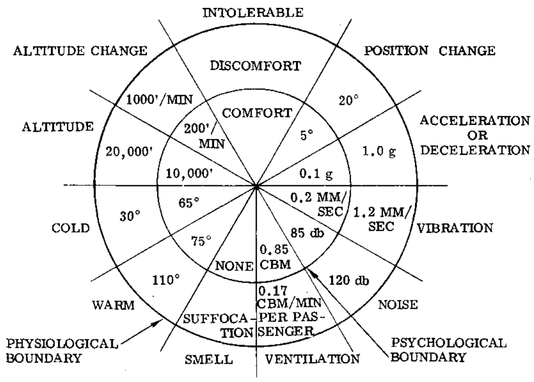

Fig. 3 Pirath comfort wheel.[2]

termined over a range of frequencies critical to ride comfort and compared with two ride comfort limits. Once an acceptable ride is achieved, the angle of attack and the required torque to activate the control fins are found to examine the feasibility of such a system.

Ride Comfort Limits

Convenience factors such as travel time and ride comfort determine to a large degree the acceptability by the traveling public of any public transportation system. A high-speed lightweight vehicle may decrease travel time but simultaneously deteriorate ride quality. The noise, vibrations, accelerations, pressure changes, and visual disturbances sensed by the passengers at high speeds can offset the advantage of decreased travel time. Limiting factors for passenger comfort are therefore necessary in the design of an acceptable transportation system.

A convenient means of qualitatively describing different levels of comfort is the Pirath Comfort Wheel (Fig. 3). On the Pirath Comfort Wheel, comfort is separated into three tolerance levels: intolerable, uncomfortable, and comfortable. The uncomfortable level is bounded by a physiological boundary above which the level of comfort is intolerable and at which physical harm is possible. Near this boundary the comfort of a passenger is directly related to body changes affecting the nervous system and causing pain, nausea, or loss of consciousness. The comfortable tolerance level is bounded by a psychological boundary at which a passenger may feel bodily comfort but notice unpleasant surroundings. Thus, the comfort of a passenger is dependent on his mental attitude as well as his physical feelings. While physiological and psychological feelings of passengers are useful in qualitatively determining ride comfort, quantitative data is necessary to establish useful ride comfort limits for design purposes.

A rapid change in the environment of a passenger appears to be the basic factor quantitatively affecting ride comfort criteria.[2] Critical to ride comfort are the acceleration and the rate of change of acceleration, jerk, that are related to the vibrations and accelerations of the vehicle. These factors place restrictions on the motion of the vehicle and form requirements on the performance of the suspension system, on the elastic deformation of the vehicle, and on the acceleration and braking ability of the vehicle. The limits of acceleration and vibration are given respectively in terms of g's per second, allowable jerk, and in terms of g's allowable acceleration. The limits vary with the orientation of the passenger, with the degrees of freedom that describe the motion, with the duration and intensity of the motion, and with the frequency and amplitude of the motion.

In particular, vibration limits are generally represented as plots of maximum acceleration in a cycle vs vibration frequency. The restriction of acceleration for a certain

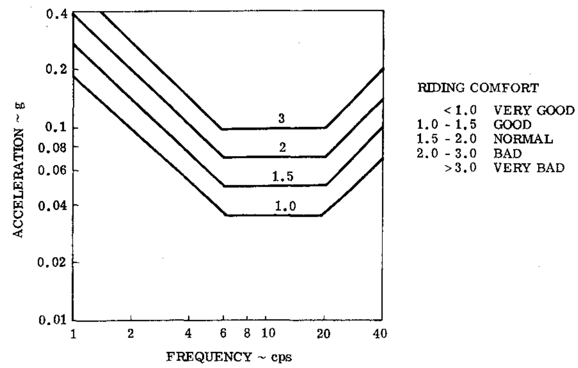

Fig. 4 The vertical vibration ride comfort limits used by the Japanese National Railroad.[3]

level of comfort varies with frequency and is a minimum from about 2–20 cps, particularly near 6 cps. Muscular fatigue is caused by vibration near 6 cps and is the result of the passenger tightening muscles to overcome the resonance of parts of his body. Several existing vibration limits for ride comfort are useful for the design of transportation systems. One limit for vertical ride comfort is used by the Japanese National Railroad (Fig. 4). This limit varies for differing degrees of comfort and has a critical frequency range of 6–20 cps. Another useful vertical ride comfort limit is one that is used by the British railroads and that is based on work done by Reiher and Meister (Fig. 5). This limit varies for differing degrees of comfort and has a critical frequency of 6 cps. The sensitivity of passengers to vibrations in a particular direction increases as the vibrations occur in the vertical, longitudinal, and lateral directions. The ride comfort limit in the lateral direction places the most severe limit on vibrations and is approximately 1.4 times lower than those limits for the vertical direction.

Sources of Excitation

Because tracked air-cushion vehicles are capable of reaching speeds of 300 mph or more the dynamic inputs usually encountered by ground vehicles are greatly magnified. The inputs arise from atmospheric conditions, from curves and grades in the guideway, and from the roughness or irregularities of the guideway. These excitations can be classified as either transient or steady-state aerodynamic forces, or as transient or steady-state guideway-induced excitations.

The effect of aerodynamic forces on the dynamics of tracked air-cushion vehicles become quite significant at speeds near 300 mph. For example, high-speed, jet-powered cars have been overturned by the aerodynamic pitch-

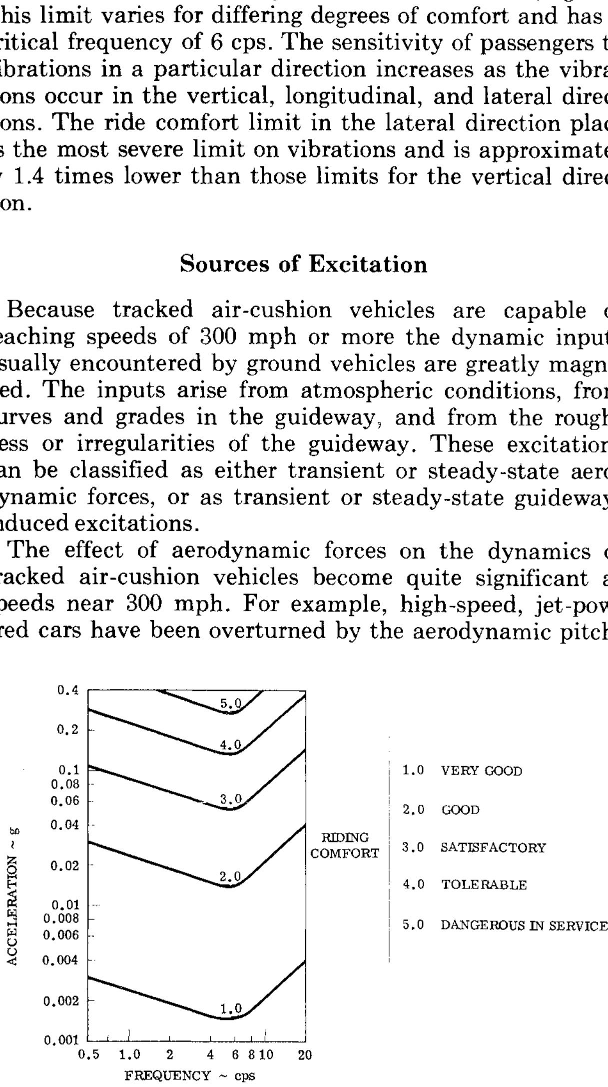

Fig. 5 The vertical vibration ride comfort limits developed by Reiher and Meister.[3]

ing moment and lift force caused by the forward motion of the car.[4] However, more significant than the forward-speed effects are the side wind disturbances. Atmospheric turbulence produces gusts that commonly have variations of 10 mph (rms), but that can have velocities up to 60 mph. These side winds produce noticeable, fluctuating rolling and yawing moments that influence the stability of the vehicle. In fact, aerodynamic loads as high as 20,000 lb lift or side force are estimated for proposed tracked air-cushion vehicle configurations traveling at 300 mph in a 60 mph side wind.[5] Sidewalls or sidefences to provide aerodynamic shielding are relatively ineffective in reducing the incremental lift and rolling moment due to the side winds, but the shields are quite effective in reducing the side force.[6] Karman vortices may also cause considerable yawing moments as the vortices separate from alternate sides of the vehicle.[4]

Guideway-induced excitations arise from the necessity of the vehicle to enter or leave a grade or to round a curve, and from the roughness and deformation of the guideway. The transition of the guideway into grades or into curves results in an acceleration of the vehicle. One possible means of alleviating the lateral acceleration of the vehicle rounding a curve is to partially bank the guideway and to allow the passenger compartment to be partially banked.[7] The guideway excitations created by guideway irrregularities are frequently classified into two categories: a discrete frequency sinusoidal excitation and a broadband sinusoidal excitation.[8] The discrete frequency excitation originates from the sag of an elevated guideway between support columns. The frequency of the excitation is determined from the ratio of the vehicle forward speed to the pier spacing which is the irregularity wavelength. The static displacements resulting from camber, thermal distortions, and flexibility of the elevated guideway, can combine with the dynamic response of the guideway to significantly affect the response of the vehicle traversing the span.

The broadband sinusoidal excitation is the basic approach employed in the present study. Here, it is assumed that the guideway exhibits a more nearly random unevenness which can be decomposed into its harmonic components. Typically, the amplitude of a particular irregularity wavelength is determined in a statistical manner from measurements of surfaces, such as airport runways that are thought to possess a similar roadway roughness as the guideway for a tracked air-cushion vehicle. The vertical roughness model illustrated in Fig. 6 was established in this manner and is used to determine the peak to zero sinusoidal amplitude of the guideway irregularities.[7] In the vertical roughness model (Fig. 6), wavelenghts less than the length of a single air cushion of 10 ft are not considered.

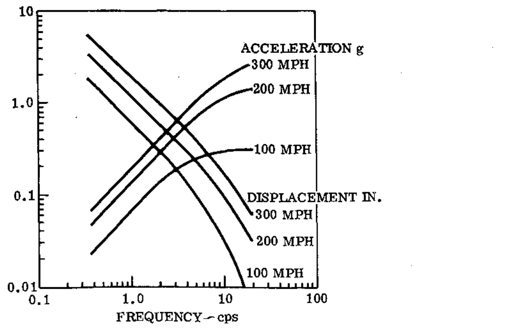

Fig. 7 Vertical excitation frequency spectra of the guideway irregularities.[7]

These shorter guideway irregularities are not transmitted through to the passenger compartment, but they are filtered out by the air cushion.

To study the response and ride quality of a tracked air cushion vehicle to harmonic guideway excitations, the vertical excitation spectrum in Fig. 6 is easily transformed to a more convenient plot of excitation amplitude and excitation acceleration vs excitation frequency (Fig. 7). A comparison of the guideway-induced accelerations with one of the ride comfort limits shows the need for a secondary suspension system to isolate the passengers from the guideway excitations, especially at high speeds.

Mathematical Model

A six-degree-of-freedom analytical model is used to investigate the feasibility of employing active aerodynamic control surfaces to improve the vertical ride quality of a tracked air cushion vehicle responding to harmonic guideway excitations (Fig. 8). The vehicle is represented as a rigid mass, the passenger compartment, which is actively controlled by an aerodynamic fin located at each end of the vehicle. A passive mechanical suspension in series with an air cushion supports this mass at each end above a rigid, but wavy guideway.

The primary suspension systems, two air cushions, are identical and each consists of a mass supported by a flexible cushion of air. The force-displacement relationship for the air cushions is represented by the linear, dynamic model developed by Richardson.[5]

$$\left(\frac{c}{k_b}\right)\dot{f}_{AC} + f_{AC} = k_a\left[\frac{c}{k_b}\left(\frac{k_a + k_b}{k_a}\right)(\dot{u}_a - \dot{x}_a) + (u_a - x_a)\right]$$

or

$$\tau_2 \dot{f}_{AC} + f_{AC} = k_a[\tau_1(\dot{u}_a - \dot{x}_a) + (u_a - x_a)]$$

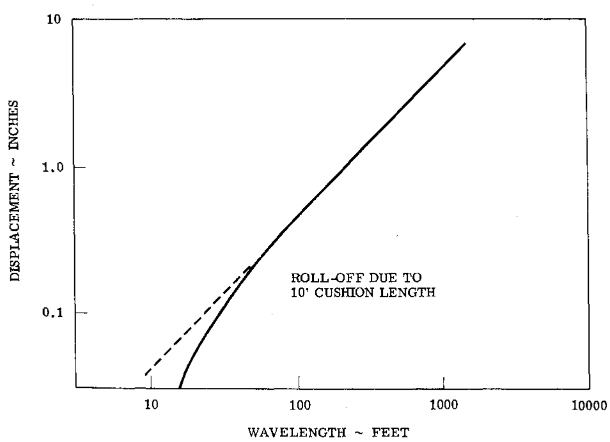

Fig. 6 Vertical roughness model of the guideway.[7]

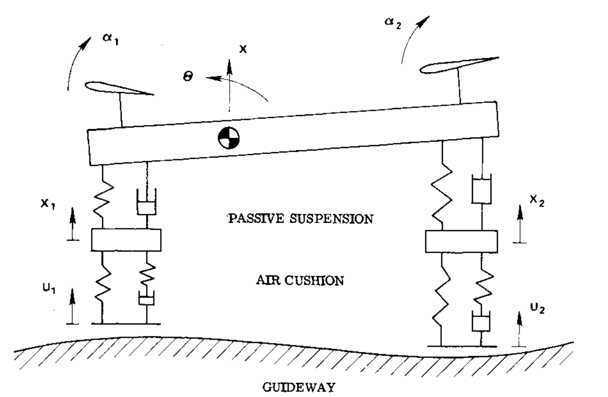

Fig. 8 Six-degree-of-freedom model of an actively controlled tracked air-cushion vehicle.

As illustrated in Fig. 9, the air cushions are too stiff to effectively isolate the passenger compartment from the guideway excitations. A secondary passive suspension is therefore placed between the passenger compartment and each air cushion. The elements of both passive suspensions, a linear spring and viscous damper, are similar. A set of physical parameters specifying the two air cushions, the passive suspensions, and the vehicle inertia characteristics are taken from a representative tracked air cushion vehicle proposed in a General Electric Co. report.[5] Two speeds, 200 mph and 300 mph, are chosen because the severity of the guideway excitations and the need for ride quality control are more apparent at these speeds.

The active suspension system is comprised of a symmetrically shaped aerodynamic fin located on the passenger compartment above each passive suspension. Each fin is rotated about its c.g. by an active torque to control the two-degree-of-freedom motion of the passenger compartment. A linear combination of acceleration, velocity, and displacement feedback signals of the pitch and plunge degrees of freedom of the passenger and of the front air cushion mass degree of freedom defines the angle of attack of the front airfoil. A similar blend of feedback signals, except that the feedback signals from the rear air cushion mass replace those of the front air cushion mass, defines the angle of attack of the rear airfoil. The inertia characteristics of each fin are calculated for a homogeneous solid with a density resembling semi-monocoque construction. The mass of each control surface is constrained such that the two fins are mass balanced about the c.g. of the passenger compartment. That is, if L_1 and L_2 are, respectively, the distance from front fin of mass M_1 to the passenger compartment c.g. and the distance from the rear fin of mass M_2 to the passenger compartment c.g., then $M_1L_1 = M_2L_2$. The aerodynamic forces produced by each control fin and the body of the moving vehicle is a sea level atmosphere are determined using a 2D quasi-steady aerodynamic approximation. A rectangular planform and NACA 0009 airfoil section are chosen for both fins. The planform areas of the two control surfaces is determined such that their combined maximum lift is much less than the weight of the vehicle. The aerodynamic coefficients for an assumed vehicle body configuration are taken from wind tunnel data for various model configurations near a simulated ground plane, as reported in a NASA Langley study.[9]

Method of Solution

A system of eight simultaneous, linear differential equations represents the vertical motion of the six-degree-of-freedom model for small pitch motion of the passenger compartment about its c.g. and the force-displacement relationship of the two air cushions. Three characteristic parameters of the vehicle, its mass, its length, and its forward speed, are used to nondimensionalize the system of equations. The systems of equations are then put in following matrix form:

$$\mathbf{M}\bar{X}'' + \mathbf{C}\bar{X}' + \mathbf{K}\bar{X} = \bar{F}$$

The control law for each control surface is substituted into the system of eight equations to produce a system of six equations in terms of only four degrees of freedom. Because the system of equations is linear, the steady-state response of the vehicle to harmonic guideway excitations is also harmonic with the same frequency, but the magnitude and phase angle of the response are different from those of the guideway excitation. By a simple conversion of the Laplace transformation of the system of equations to an exponential Fourier transformation, the magnitude and phase angle of the harmonic response can be found for various excitation frequencies.

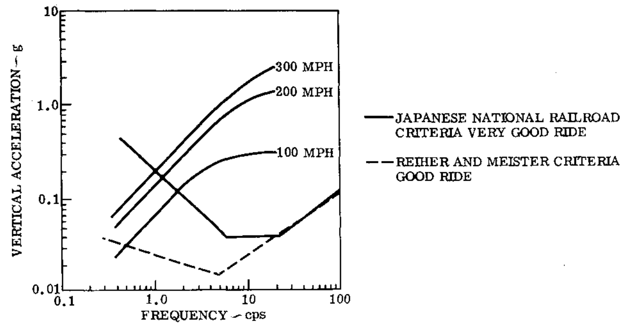

Fig. 9 A comparison of the expected guideway excitations with ride quality criteria demonstrating the need for a secondary suspension.

The Laplace tranformation of the system of equations produces a system of linear algebraic equations in terms of the nondimensional Laplace variable, S.

$$\mathbf{A}\bar{r} = \bar{R}$$

The system of algebraic equations is solved for the response of vector, **r**, whose elements are the ratio of each dependent variable to the amplitude of the guideway irregularity, to obtain a vector of transfer functions.

$$\bar{r} = \mathbf{A}^{-1}\bar{R}$$

The Fourier transformation of the vector of transfer functions is found by replacing the nondimensional Laplace variable with the complex variable $j\Omega$, in which Ω is the nondimensional excitation frequency. For a particular value of the frequency, the magnitude of each element of the response vector is determined by calculating the complex absolute value of the corresponding transfer function. However, of practical significance are the deflection of each passive suspension and the acceleration response of each end of the vehicle. The absolute and relative response of each end of the passenger compartment are easily found as the magnitudes of linear combinations of elements of the response vector. The acceleration response is then found by multiplying the absolute response by the square of the excitation frequency.

The frequency response of the vehicle was calculated over the range of frequencies 0.01 cps to 40 cps for a variety of feedback signals for each aerodynamic fin. Initially, the effect of each fin on the frequency response was examined separately. Various control laws were assumed using a trial-and-error approach. Once the types of feedback signals which were most effective in reducing the steady-state response were found, both fins were operated simultaneously for various combinations of these feedback signals. For each combination of feedback signals, the quality of the ride was evaluated from comparisons of the acceleration response of each end of the passenger compartment with the two ride comfort limits. In this way, a simple but effective control law for each aerodynamic fin was determined to achieve a desired ride quality of the vehicle. As soon as an acceptable level of ride quality was attained, the angle of attack and the required torque were calculated to determine the feasibility of such an active suspension.

Discussion of Results

The use of actively controlled aerodynamic fins appears to be an attractive means of achieving good ride quality for tracked air cushion vehicles. For vehicle speeds of 200 mph and 300 mph, the choice of simple acceleration feedback signals to each control fin results in the desired ride quality of the passenger compartment. The aerodynamic fins not only suppress the unwanted steady-state harmon-

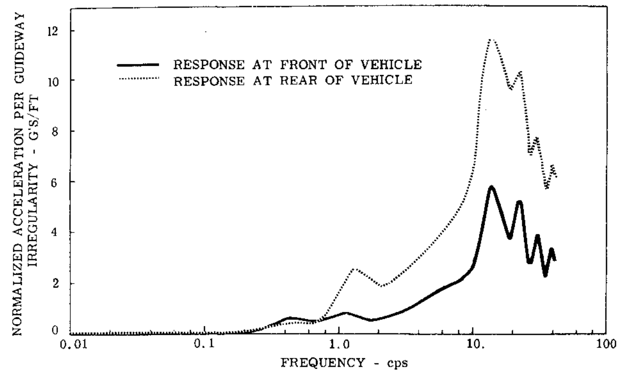

Fig. 10 Normalized acceleration response of the uncontrolled vehicle at 300 mph.

ic response, but the angle of attack and the active torques required are attainable.

Frequency Response at 300 mph

The acceleration reponse in g's of each end of the passenger compartment and the response of each passive suspension, both normalized by the amplitude of the guideway excitation, are plotted, respectively, in Fig. 10 and in Fig. 11 vs excitation frequency for the vehicle without any active control. Three peaks in the response curves occur at the two resonant frequencies corresponding to the two degrees of freedom of the passenger compartment and at the resonant frequency of the air cushions. The two resonant frequencies of the passive suspensions are approximately 0.4 cps and 1.2 cps. The mode at 0.4 cps is lightly damped and the passive suspensions respond significantly at this frequency. A deflection of 13 in. occurs in the front passive suspension and a deflection of 8 in. occurs in the rear passive suspension. At 1.2 cps, the response of the passive suspensions is less than 3.5 in., and at higher frequencies the response decreases to less than 1.0 in. As a result of the soft springs and light damping in the passive suspensions, both suspensions respond identically at frequencies above 6 cps.

The acceleration response of the passenger compartment appears to be greatest at the resonant frequency of the two air cushions, at approximately 14 cps. The acceleration response is especially large at the rear of the vehicle, the response being almost twice that at the front. In Fig. 12, the actual acceleration levels at the rear of the passenger compartment are compared with two ride comfort limits. The ride quality is acceptable near 0.4 cps. Between 1.2 cps and 14 cps, however, the level of acceleration is five times too high for good ride quality.

At certain other frequencies, the acceleration response of the passenger compartment has relative maximums and

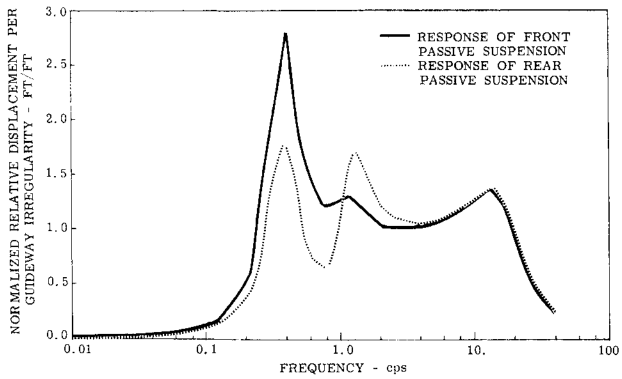

Fig. 11 Normalized passive suspension performance of the uncontrolled vehicle at 300 mph.

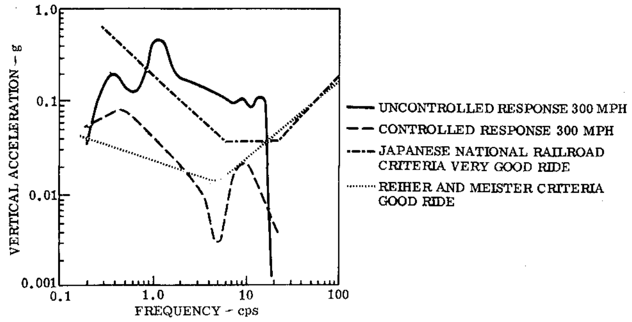

Fig. 12 Comparison of expected vehicle accelerations with ride quality criteria.

relative minimums due to an almost completely rigid body response of the vehicle. This type of response is influenced only by the motion of the two air cushions. An in-phase motion of the air cushions occurs at excitation frequencies at which the lengths of the air cushion spacing is an integral number of the corresponding irregularity wavelengths. A minimum amount of pitching motion of the passenger compartment is caused at the in-phase frequencies. This results in a relative minimum in the acceleration response. For a vehicle speed of 300 mph, the in-phase frequencies begin at 9 cps and occur at intervals of 9 cps. For those excitation frequencies at which the length of the air cushion spacing is an integral number of half of the corresponding irregularity wavelengths, the response of the two air cushions is 180 degrees out of phase. The acceleration response of the passenger compartment is a relative maximum at the out-of-phase frequencies because of the large amount of pitching motion of the passenger compartment. For a vehicle speed of 300 mph, the out-of-phase frequencies begin at 4.5 cps and occur at intervals of 9 cps. Both these types of motion cause the rear of the passenger compartment to respond more than the front because of the forward location of the passenger compartment c.g. which is used for aerodynamic stability.

From an examination of acceleration, velocity, and displacement types of feedback signals for each control fin, the acceleration feedback signals were found to have the most significant effect on the steady-state response of vehicle near the resonance of the air cushions. Pitch acceleration feedback signals are the most effective in reducing the response of the vehicle, and plunge acceleration feedback signals have a small, but beneficial effect on ride quality. However, acceleration, velocity, and displacement feedback signals sensed at the air cushion masses deteriorate the ride quality of the vehicle and the passive suspension performance. Pitch and plunge acceleration feedback reduce the residues of the two air cushion modes rather than increasing the damping on those modes to achieve ride quality. The damping and frequency of the two passive suspension modes are decreased with this type of feedback. But with the addition of pitch rate feedback, good ride quality is achieved without degrading passive suspension performance.

Shown in Figs. 13 and 14 are, respectively, the normalized acceleration response and the normalized passive suspension response at each end of the vehicle with the active aerodynamic fins operating. With the front fin controlling plunge acceleration and the rear fin controlling pitch acceleration and pitch rate of the passenger compartment, a desired level of ride comfort is attained. The control law has the following form:

$$\alpha_1 = -0.1\ddot{x} \quad \alpha_2 = -5.0\ddot{\theta} - 13.4\dot{\theta}$$

The vehicle is stable for both the aerodynamic and inertia portions of the feedback loops closed. The acceleration

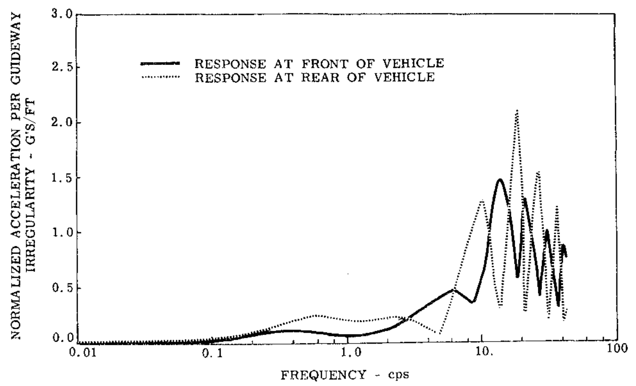

Fig. 13 Normalized acceleration response of the controlled vehicle at 300 mph.

response of the passenger compartment is reduced to approximately one fifth of its value for the uncontrolled vehicle. When the expected accelerations are compared with the two ride comfort curves, the ride quality of the vehicle is very good by Japanese National Railroad standards and except for the low-frequency range is good by the standards developed by Reiher and Meister (Fig. 12). Although passive suspension response and required control at the lowest passive suspension mode frequency is still significant, the angle of attack and active torque necessary for good ride quality at 300 mph are quite feasible. The maximum angle of attack of either fin is about five degrees in the frequency range of 1–40 cps. In this frequency range, the maximum torque is less than 2400 ft–lb.

Frequency Response at 200 mph

The normalized acceleration response and the normalized passive suspension performance of the vehicle traveling at 200 mph with no active control are plotted in Figs. 15 and 16 vs excitation frequency. The shapes of the curves are similar to the corresponding curves for a vehicle speed of 300 mph. At 200 mph, though, the magnitude of the guideway excitations is less severe and the actual response levels have decreased.

The two resonant frequencies of the passive suspension are changed by vehicle body aerodynamics. At 200 mph, these frequencies are approximately 0.44 cps and 1.1 cps. For a zero density airstream, the frequencies become 0.5 cps and 0.9 cps. Again, the greatest passive suspension response is at the lowest resonant frequency. At this frequency the front suspension deflects 9.0 in. and the rear suspension deflects 5.0 in. The passive suspensions respond less than 2.5 in. at the other passive suspension resonant frequency, and their responses decrease similarly at higher frequencies to less than 1 in.

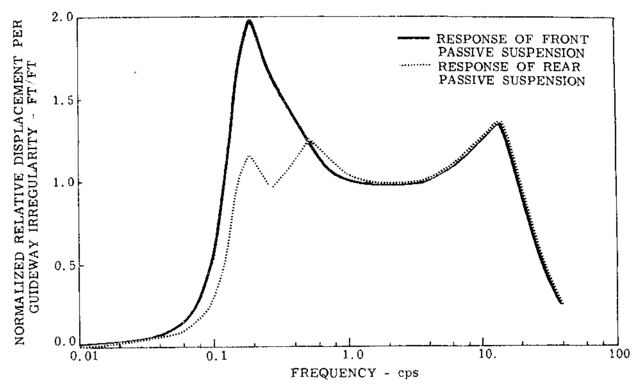

Fig. 14 Normalized passive suspension performance of the controlled vehicle at 300 mph.

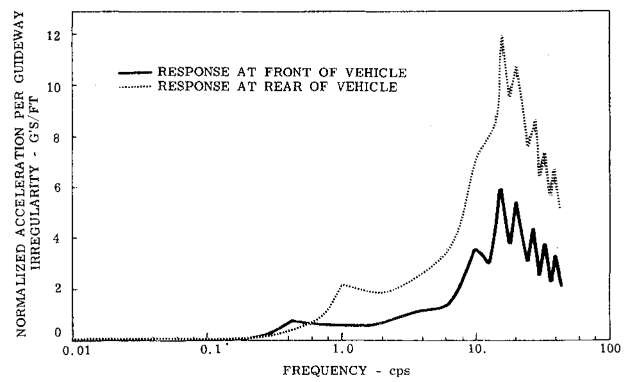

Fig. 15 Normalized acceleration response of the uncontrolled vehicle at 200 mph.

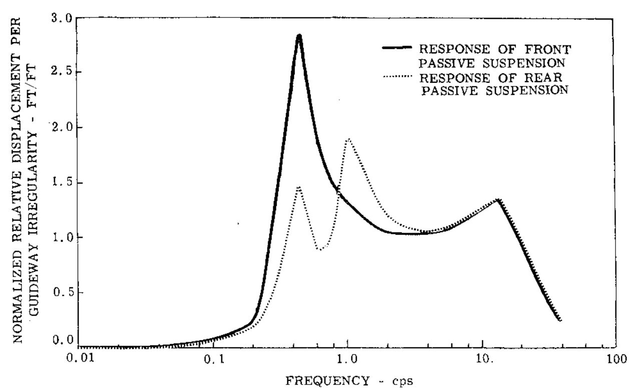

Fig. 16 Normalized passive suspension performance of the uncontrolled vehicle at 200 mph.

At the resonant frequency of the two air cushions, the normalized acceleration response of the passenger compartment is largest, particularly at the rear of the vehicle. In Fig. 17 the actual acceleration level at the rear of the passenger compartment is compared with the two ride comfort curves. The level of acceleration is about three times too high for good ride quality between 1 cps and 20 cps. In this frequency range, acceleration levels are determined primarily by resonance response and also influenced by the in-phase and out-of-phase motion of the air cushions. For a vehicle speed of 200 mph, the out-of-phase frequencies begin at 3 cps and are spaced 6 cps apart. The in-phase frequencies start at 6 cps and occur at 6 cps intervals. A combination of out-of-phase motion and resonance response of the air cushions causes an acceleration peak near 9 cps. At the resonant frequency of the air cushions the acceleration response is smoothed by an in-phase motion of the air cushion. Thus, the minimum amount of pitching motion at in-phase frequencies tends

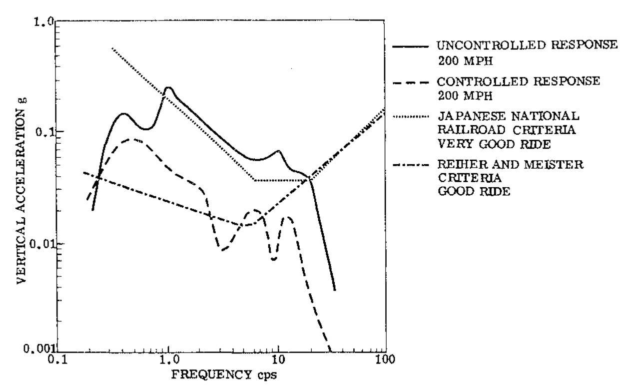

Fig. 17 Comparison of expected vehicle accelerations with ride quality criteria.

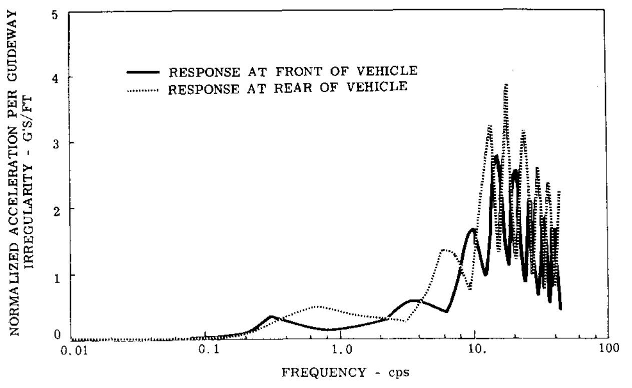

Fig. 18 Normalized acceleration response of the controlled vehicle at 200 mph.

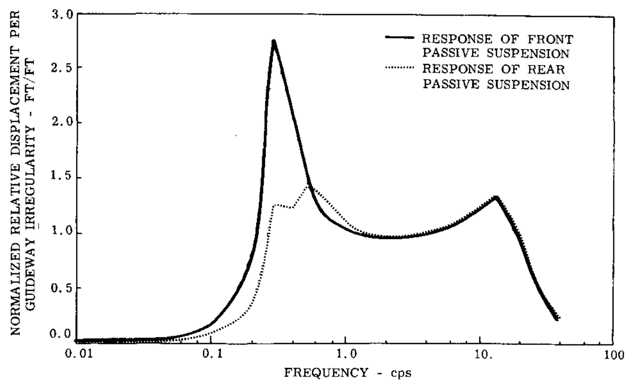

Fig. 19 Normalized passive suspension performance of the controlled vehicle at 200 mph.

to attenuate the resonance response, while the resonance response is accentuated by the pitching motion at out-of-phase frequencies.

The same acceleration control laws for both aerodynamic fins as used at 300 mph are used at 200 mph. The normalized acceleration response as shown in Fig. 18 is reduced by only one third of its value for the uncontrolled vehicle. The ability of the aerodynamic fins to control the steady-state harmonic response of the vehicle is lessened as the dynamic pressure of the airstream is decreased. However, a desired level of ride comfort is once more attained without degrading the passive suspension response (Fig. 19). The comparison of the expected accelerations with the two ride comfort curves in Fig. 17 illustrates the similarity of the ride quality of the controlled vehicle at 200 mph and at 300 mph. That is, the ride quality is very good by the standards of the Japanese National Railroad and is better than satisfactory by the standards of Reiher and Meister. The passive suspension response and required control are again significant at the lowest resonant frequency. In the frequency range of 1–40 cps, the maximum angle of attack of either fin is about six degrees. The maximum torque in this frequency range is less than 1400 1400 ft-lb.

Conclusions

A simple and practical means of achieving the desired ride quality of a tracked air cushion vehicle is the use of active aerodynamic fins to control the steady-state response of the vehicle to harmonic guideway excitations. At low speeds when the dynamic pressure of the airflow is small this type of active suspension may not be too effective. But, the vibrations and accelerations of the passenger compartment at these speeds would probably be imperceptible to passengers sitting in comfortable seats. As a result of the aerodynamic control system reacting with the atmosphere rather than reacting with the guideway, the response of the guideway and the stress levels in the guideway can be lessened. This reduction would be advantageous in the design and maintenance of the guideway. The concept of an active aerodynamic suspension is also applicable to aircraft with air cushion landing systems. In this application an active landing gear could be realized.

Though the proposed aerodynamic control system appears to be an acceptable active suspension system, further investigation and improved modeling of the tracked air cushion vehicle and the control system are warranted. The present study should be extended to include both the lateral as well as the vertical plane dynamics and the control of transient response such as horizontal gust response of the vehicle. A refined potential flow aerodynamic modeling such as the doublet lattice method could be useful. The complex modeling of the vehicle dynamics should also be accompanied by thorough study of possible optimal control systems.

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