Vibration Control of Mobile Robot Vehicle by Dynamic Vibration Absorber*

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The paper considers a planar vibrational model of mobile robot vehicle equipped with dynamic vibration absorber. The purpose is first, to determine the dynamic vibration absorber parameters optimal adjustment giving the lowest vibrational displacement of the vehicle body at resonance, when the vehicle is subjected to ground roughness modeled as sinusoidal or periodic rectangular spaced pulse disturbances. Second, the purpose is to determine disturbance frequencies at which the dynamic vibration absorber parameters optimal adjustment fails to ensure the vibration stability of the mobile robot vehicle equipped with the dynamic vibration absorber, and to recommend measures to guarantee the vibration stability at these frequencies.

Key Words: Mobile Robot Vehicle, Periodic Ground Disturbances, Vibration Control, Dynamic Vibration Absorber, Absorber Efficiency, Absorber Reliability

1. General Introduction

Vibration isolation of mobile robot vehicle is conventionally realized by vibration suspensions. To use better the vehicle geometry, specially the empty space and the accessories mass into the vehicle body, dynamic vibration absorber seems to be more convenient to vibration control of the vehicle body than vibration suspensions.

The mobile robot vehicle, called the system, is modeled according to the standardization and to the common modeling⁽¹⁾. Indeed, the vehicle's sprung and unsprung masses are modeled as a pure mass, the vehicle's elasticity is modeled as a pure linear spring, and the vehicle's damping is assumed to be negligible. On the other hand, the dynamic vibration absorber, called the absorber, is modeled as a pure mass to be attached by means of a pure linear spring in parallel with a pure linear viscous damper. Therefore, the mobile robot vehicle equipped with the dynamic vibra-

tion absorber, called the whole system, is modeled as a pure mass supported symmetrically by two identical pure linear springs, and to the mass is attached an other pure mass by means of a pure linear spring in parallel with a pure linear viscous damper.

Due to the geometry of the whole system, the masses can only exhibit planar motion, the system springs can only exhibit vertical motion or/and deformation, and the absorber damper and spring can only exhibit perpendicular motion or/and deformation to the masses. On the other hand, the vehicle velocity is assumed to be constant during the vehicle motion, and vibration stimulation caused by any other sources than the ground roughness is assumed to be of negligible effect on the whole system vibration.

The ground roughness on which the vehicle moves, is modeled first as a sinusoidal disturbance, and second as a periodic rectangular spaced pulse disturbance. Each roughness model has constant parameters, and may be at stable or unstable frequency.

Figure 1 shows the whole system planar vibrational model as described previously, for which the following notations are used.

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Fig. 1 Planar vibrational model of mobile robot vehicle equipped with dynamic vibration absorber

- m_1 : the system mass
- k_1 : the system stiffness coefficient
- l_1 : the system stiffness initial length
- m_2 : the absorber mass
- c : the absorber damping coefficient
- k_2 : the absorber stiffness coefficient
- l_2 : the absorber stiffness initial length
- G_1 : the system mass center
- G_2 : the absorber mass center
- G_3 : the front excitation application point
- G_4 : the rear excitation application point
- x_1 : the system mass vertical displacement
- x_2 : the absorber mass vertical displacement
- y_1 : the system mass horizontal displacement
- y_2 : the absorber mass horizontal displacement
- θ : the system and absorber masses angular rotation
- v_1 : the front excitation
- v_2 : the rear excitation
- h : the system mass length parameter

In the following, the masses angular rotation and horizontal displacements are assumed to be negligible. Their negligibility may be realized by proper design of the whole system⁽²⁾. On the other hand, the system parameters are considered as constant relatively to the absorber parameters.

The paper contains two main sections: the absorber efficiency and the absorber reliability. The absorber efficiency analysis has to determine the absorber parameters optimal adjustment giving the lowest vibrational displacement amplitude of the system mass at resonance, when the system is subjected to ground roughness modeled as sinusoidal or periodic rectangular spaced pulse disturbances. The absorber parameters optimal adjustment, called the absorber optimal adjustment, corresponds to the absorber best efficiency. On the other hand, the absorber reliability analysis has to determine the disturbance frequencies at which the absorber optimal adjustment fails to ensure the whole system vibration stability, and to recommend measures to guarantee the vibration stability at these frequencies. The realization of the

whole system vibration stability corresponds to the absorber best reliability.

2. Absorber Efficiency

2.1 Sinusoidal disturbance

2.1.1 Excitations expressions The sinusoidal disturbance is defined as:

$$v = V \sin(\omega t + \phi) \quad (2.1)$$

where, V , ω , and ϕ are respectively, the disturbance amplitude, angular frequency, and phase angle. The angular frequency is defined as: $\omega = 2\pi\lambda/L$; where λ is the whole system motion velocity, defined as: $\lambda = L/T$, L and T are respectively the disturbance space and time periods.

The front and rear excitations are respectively defined as:

$$v_1 = V \sin(\omega t + \phi_1) \quad (2.2)$$

$$v_2 = V \sin(\omega t + \phi_2) \quad (2.3)$$

where, ϕ_1 is the front excitation phase angle, defined as: $\phi_1 = \phi + h\omega/\lambda$; and ϕ_2 is the rear excitation phase angle, defined as: $\phi_2 = \phi - h\omega/\lambda$.

2.1.2 Displacements expressions The system and absorber masses vertical displacements are respectively defined as:

$$x_1 = UX_1 \sin(\omega t + \phi + \varphi_1) + x_{10} \quad (2.4)$$

$$x_2 = UX_2 \sin(\omega t + \phi + \varphi_2) + x_{20} \quad (2.5)$$

where,

$$U = V(2(1 + \cos(\phi_2 - \phi_1)))^{1/2};$$

$$\varphi = \tan^{-1} \left(\frac{\sin \phi_2 + \sin \phi_1}{\cos \phi_2 + \cos \phi_1} \right);$$

$$X_1 = \left(\frac{A^2 + B^2}{C^2 + B^2 D^2} \right)^{1/2}; \varphi_1 = \tan^{-1} \left(\frac{B(AD + C)}{AC - B^2 D} \right);$$

$$X_2 = \left(\frac{f^4 + B^2}{C^2 + B^2 D^2} \right)^{1/2}; \varphi_2 = \tan^{-1} \left(\frac{B(f^2 D + C)}{f^2 C - B^2 D} \right);$$

$$A = f^2 - s^2; B = 2fds;$$

$$C = s^4 - (\mu f^2 + f^2 + 2)s^2 + 2f^2; D = (\mu + 1)s^2 - 2;$$

$$\mu = \frac{m_2}{m_1}; f = \frac{\omega_2}{\omega_1}; d = \frac{c}{2m_2\omega_2}; s = \frac{\omega}{\omega_1};$$

$$\omega_1^2 = \frac{k_1}{m_1}; \omega_2^2 = \frac{k_2}{m_2}; x_{10} = -g \frac{m_1 + m_2}{2k_1} + l_1;$$

$$\text{and } x_{20} = -g \left(\frac{m_1 + m_2}{2k_1} + \frac{m_2}{k_2} \right) + l_1 - l_2.$$

U and φ are respectively the equivalent excitation amplitude and phase angle; X_1 , φ_1 , X_2 , and φ_2 are respectively, the system amplitude coefficient and phase angle, and the absorber amplitude coefficient and phase angle; while x_{10} and x_{20} are respectively the system and absorber masses initial displacements. Also, μ , k , d , and s are respectively the mass, stiffness, damping, and frequency ratios, where k is defined as: $k = k_2/k_1 = \mu f^2$.

2.1.3 Numerical results To determine the absorber optimal adjustment, we considered three values; small, medium, and large; of each of the mass, stiffness, and damping ratios. The whole values

Fig. 2.1.1 Response curves for the first value sets of the damped absorber parameters

Fig. 2.1.5 Response curves for the fifth value sets of the damped absorber parameters

Fig. 2.1.2 Response curves for the second value sets of the damped absorber parameters

Fig. 2.1.6 Response curves for the sixth value sets of the damped absorber parameters

Fig. 2.1.3 Response curves for the third value sets of the damped absorber parameters

Fig. 2.1.7 Response curves for the seventh value sets of the damped absorber parameters

Fig. 2.1.4 Response curves for the fourth value sets of the damped absorber parameters

Fig. 2.1.8 Response curves for the eighth value sets of the damped absorber parameters

constitute twenty seven sets of the mass, stiffness, and damping ratios, for each of them, the response curves; the system and absorber amplitude coefficients function of the disturbance frequency; are obtained, and are shown in Figs.2.1. In addition, the response curves for the absorber optimal adjustment, numerically found in the limits of the considered ratios values, are obtained, and are shown in Fig. 2.2. The same previous value sets are again considered with a damping ratio identical to zero, and the response curves are shown in Figs. 2.3.

Figures 2.1 and 2.2 show the absorber best effect is when the absorber mass is far heavier than the system mass, the absorber stiffness is far weaker than

Fig. 2.1.9 Response curves for the ninth value sets of the damped absorber parameters

the system stiffness, and the damping ratio is of a specific value under one and dependent on the absorber mass and stiffness values. Besides, the absorber vibrational displacement amplitude is low at the

Fig. 2.2 Response curves for the numerically found absorber optimal adjustment; $\mu=10.0$, $k=0.1$, $d=0.9$

Fig. 2.3.1 Response curves for the first value sets of the undamped absorber parameters

Fig. 2.3.2 Response curves for the second value sets of the undamped absorber parameters

Fig. 2.3.3 Response curves for the third value sets of the undamped absorber parameters

absorber best effect. Figures 2.3 show the system vibrational displacement amplitude is identical to zero when the disturbance frequency is equal to the absorber undamped natural frequency.

2.1.4 Discussion When it is damped, an absorber having a heavier mass and a weaker stiffness than the system, helps the disturbance energy to move from the system to the absorber due to the latter lower resistance to vibration creation under the above described conditions. This disturbance energy behavior, called immigration phenomena, reflects the fact that disturbance energy prefers to act in sites where the resistance to create vibration is the lowest. On the

other hand, a value of the absorber damping ratio under one; the absorber is under-damped; allows the absorber to exhibit oscillatory motion, which favors disturbance energy dissipation by fluid friction. Moreover, the best disturbance energy dissipation is when the fluid resistance is high, without leading to a critically damped or an over-damped absorber. Therefore, a specific fluid resistance is to be set, and is the optimum of a dilemma: keeping the absorber oscillatory motion while having the maximum fluid resistance. Thus, the absorber mass vibrational displacement amplitude is low at the absorber best effect.

When the absorber is undamped, a perfect vibration isolation of the system mass can be realized with tuning the absorber undamped natural frequency to the disturbance frequency. Indeed, when the disturbance is at the absorber undamped natural frequency, the absorber presents no resistance to vibration creation, therefore all the disturbance energy moves from the system to the absorber.

2.2 Periodic rectangular spaced pulse disturbance

2.2.1 Excitations expressions The periodic rectangular spaced pulse disturbance is defined as:

$$\begin{aligned} v &= V, t \in [\alpha T, (\alpha + \beta)T] \text{ \& } \\ v &= 0, t \in [0, \alpha T \cup (\alpha + \beta)T, T] \end{aligned} \quad (2.6)$$

where, V and T are respectively the disturbance pulse amplitude and time period; α is the disturbance phase coefficient, defined as: $\alpha \in [0, 1]$; β is the disturbance pulse width coefficient, defined as: $\beta \in]\alpha, 1]$, which has to be enough large to make sure the pulse does not become an impulse; and t is a real defining time.

The front and rear excitations are respectively defined as:

$$\begin{aligned} v_1 &= V, t \in [(\alpha - \alpha_1)T, (\alpha - \alpha_1 + \beta)T] \text{ \& } v_1 = 0, \\ t &\in [-\alpha_1 T, (\alpha - \alpha_1)T \cup (\alpha - \alpha_1 + \beta)T, (1 - \alpha_1)T] \end{aligned} \quad (2.7)$$

$$\begin{aligned} v_2 &= V, t \in [(\alpha - \alpha_2)T, (\alpha - \alpha_2 + \beta)T] \text{ \& } v_2 = 0, \\ t &\in [-\alpha_2 T, (\alpha - \alpha_2)T \cup (\alpha - \alpha_2 + \beta)T, (1 - \alpha_2)T] \end{aligned} \quad (2.8)$$

where, α_1 is the front excitation phase coefficient, defined as: $\alpha_1 = \text{decimal}(h/\lambda T)$; and α_2 is the rear excitation phase coefficient, defined as: $\alpha_2 = \text{decimal}(-h/\lambda T)$; where, λ is the whole system motion velocity, defined as: $\lambda = L/T$, L is the disturbance space period.

2.2.2 Displacements expressions Using Fourier series, the system and absorber masses vertical displacements are respectively defined as:

$$x_1 = x_{10} + \beta V + \sum_{n=1}^{\infty} U_n X_{1n} \sin(n\omega t + \varphi_n + \varphi_{1n}) \quad (2.9)$$

$$x_2 = x_{20} + \beta V + \sum_{n=1}^{\infty} U_n X_{2n} \sin(n\omega t + \varphi_n + \varphi_{2n}) \quad (2.10)$$

where,

$$U_n = V_n (2(1 + \cos(\phi_{2n} - \phi_{1n})))^{1/2};$$

$$\varphi_n = \tan^{-1} \left(\frac{\sin \phi_{2n} + \sin \phi_{1n}}{\cos \phi_{2n} + \cos \phi_{1n}} \right);$$

$$V_n = \frac{V}{\pi n} (2(1 - \cos 2\pi n \beta))^{1/2};$$

$$\phi_{1n} = \tan^{-1} \left(\frac{\sin 2\pi n(\alpha - \alpha_1 + \beta) - \sin 2\pi n(\alpha - \alpha_1)}{\cos 2\pi n(\alpha - \alpha_1) - \cos 2\pi n(\alpha - \alpha_1 + \beta)} \right);$$

$$\phi_{2n} = \tan^{-1} \left(\frac{\sin 2\pi n(\alpha - \alpha_2 + \beta) - \sin 2\pi n(\alpha - \alpha_2)}{\cos 2\pi n(\alpha - \alpha_2) - \cos 2\pi n(\alpha - \alpha_2 + \beta)} \right);$$

$$X_{1n} = \left(\frac{A_n^2 + B_n^2}{C_n^2 + B_n^2 D_n^2} \right)^{1/2};$$

$$\varphi_{1n} = \tan^{-1} \left(\frac{B_n(A_n D_n + C_n)}{A_n C_n - B_n^2 D_n} \right);$$

$$X_{2n} = \left(\frac{f^4 + B_n^2}{C_n^2 + B_n^2 D_n^2} \right)^{1/2};$$

$$\varphi_{2n} = \tan^{-1} \left(\frac{B_n(f^2 D_n + C_n)}{f^2 C_n - B_n^2 D_n} \right);$$

$$A_n = f^2 - n^2 s^2; \quad B_n = 2fdns;$$

$$C_n = n^4 s^4 - (\mu f^2 + f^2 + 1)n^2 s^2 + f^2;$$

$$D_n = (\mu + 1)n^2 s^2 - 1;$$

$$\mu = \frac{m_2}{m_1}; \quad f = \frac{\omega_2}{\omega_1}; \quad d = \frac{c}{2m_2\omega_2}; \quad s = \frac{\omega}{\omega_1};$$

$$\omega_1^2 = \frac{k_1}{m_1}; \quad \omega_2^2 = \frac{k_2}{m_2}; \quad \omega = \frac{2\pi\lambda}{L}; \quad x_{10} = -g \frac{m_1 + m_2}{2k_1} + l_1;$$

$$\text{and } x_{20} = -g \left(\frac{m_1 + m_2}{2k_1} + \frac{m_2}{k_2} \right) + l_1 - l_2.$$

U_n and φ_n are respectively the elementary equivalent excitation amplitude and phase angle; X_{1n} , φ_{1n} , X_{2n} , and φ_{2n} are respectively, the system elementary amplitude coefficient and elementary phase angle, and the absorber elementary amplitude coefficient and elementary phase angle; while x_{10} and x_{20} are respectively the system and absorber masses initial displacements. Also, μ , k , d , and s are as defined for the sinusoidal disturbance, ω is the angular frequency, and n is a positive integer.

2.2.3 Numerical results To examine the positive integer influence on the elementary response curves; the system and absorber masses elementary amplitude coefficients function of the disturbance frequency; the elementary response curves are considered for some values of the positive integer, and are presented in Figs. 2.4.

Figures 2.4 show the positive integer variation has no effect on the shapes of the elementary response curves; the system elementary response curves shape, and the absorber elementary response curves shape. In addition, each of the system and absorber elementary response curves shapes is identical to its respective response curve shape found for the sinusoidal disturbance.

2.2.4 Discussion If the frequency ratio is re-

Fig. 2.4.1 Influence of the positive integer on the system elementary response curves

Fig. 2.4.2 Influence of the positive integer on the absorber elementary response curves

placed by the product of the frequency ratio by the positive integer for the elementary response curves, only two response curves will be obtained; one for each of the system and the absorber; having respectively the same shapes found for the sinusoidal disturbance. Therefore, any variation of the positive integer for the elementary response curves, leads only to a different distribution on the disturbance frequency scale without affecting the curves shapes. Thus, the absorber parameters optimization results found for the sinusoidal disturbance are also valid for the periodic rectangular spaced pulse disturbance.

2.3 Conclusion

The work done in this section leads to the following conclusions:

(1) If the disturbance is at stable frequency, then an undamped absorber, called tuned absorber, may be used. The best vibration isolation in displacement of the system at resonance, is when the absorber undamped natural frequency is tuned to the disturbance frequency, and is perfect. This deduction may be extended to disturbance frequencies variable in narrow bands, but the vibration isolation is not perfect.

(2) If the disturbance is at unstable frequency, or

at more than one frequency due to Fourier series use, then a damped absorber, called untuned absorber, is to be used. The best vibration isolation in displacement of the system at resonance, is when the absorber has a far heavier mass and a far weaker stiffness than the system, and the absorber damping ratio is of a specific value under one and dependant on the absorber mass and stiffness values.

(3) The absorber parameters optimization results found for the sinusoidal disturbance, are not only valid for the periodic rectangular spaced pulse disturbance, but for any disturbance which can be expressed by Fourier series, especially periodic disturbances. In addition, the disturbance may depend on any parameters independent of the absorber parameters.

3. Absorber Reliability

The whole system vibration stability analysis may constitute the originality of the paper. Indeed, although it is indispensable to make sure the absorber reliability, the vibration stability analysis is not considered in many works such as Refs. (3), (4), and (5).

The whole system vibration stability is of two parts: the transient stability and the permanent stability. The transient stability corresponds to the disturbance transient effect, and is realized when the vibrational system is positively damped. On the other hand, the permanent stability corresponds to the disturbance steady effect, and is realized when the whole system energy balance per disturbance cycle is identical to zero. Indeed, we found for both ground roughness models treated in the previous section, that the energy per disturbance cycle for each of the whole system masses and springs is equal to zero⁽²⁾. Thus, each of the masses and springs regains its respective initial position after a disturbance cycle. Then, no disturbance energy is left in the whole system after a disturbance cycle. Therefore, the whole system external energy per disturbance cycle is totally dissipated by damping, within a disturbance cycle. Thus, the whole system external and dissipated energies per disturbance cycle are equal, and the whole system energy balance per disturbance cycle is identical to zero, which leads to the whole system permanent stability expression, called the whole system vibration stability expression. In contrast, if the whole system external and dissipated energies per disturbance cycle are different after a disturbance cycle, then each of the whole system masses and springs does not regain its respective initial position after the disturbance cycle ends. Thus, each of the masses and springs energies per disturbance cycle is different of zero. Therefore, the whole system dynamic equilibrium is not realized, then the whole system vibration is unsta-

ble.

3.1 Sinusoidal disturbance

3.1.1 Whole system vibration stability expression The whole system external energy per disturbance cycle is defined as:

$$W_e = \pi k_1 U^2 X_1 \sin \varphi_1 \quad (3.1)$$

The whole system dissipated energy per disturbance cycle is defined as:

$$W_d = \pi c U^2 W^2 \omega \quad (3.2)$$

where, $W = (X_2^2 + X_1^2 - 2X_2X_1 \cos(\varphi_2 - \varphi_1))^{1/2}$

Therefore, the whole system vibration stability expression is defined as:

$$U^2(X_1 \sin \varphi_1 - 2\mu f d W^2 s) = 0 \quad (3.3)$$

The whole system vibration stability is obtained when at least one of the following cases is realized.

$$(U^2=0) \text{ or } (X_1 \sin \varphi_1 - 2\mu f d W^2 s = 0)$$

3.1.2 Numerical results The case of $U^2=0$, corresponds to situations where the whole system vibration stability is independent of the absorber parameters. The vibration stability is realized when the disturbance frequency is equal to one of the absorber independent vibration stability frequencies defined by the above mentioned expression.

The case of $X_1 \sin \varphi_1 - 2\mu f d W^2 s = 0$, corresponds to situations where the whole system vibration stability is dependent on the absorber parameters. To show the influence of the absorber parameters; mass, stiffness, and damping ratios; on the whole system

Fig. 3.1 Influence of the mass ratio on the whole system vibration stability for the sinusoidal disturbance

Fig. 3.2 Influence of the stiffness ratio on the whole system vibration stability for the sinusoidal disturbance

Fig. 3.3 Influence of the damping ratio on the whole system vibration stability for the sinusoidal disturbance

vibration stability, Figs. 3.1, 3.2, and 3.3 are presented. In these Figures, black spaces indicate the whole system vibration stability is realized, while blank ones indicate the vibration stability is not realized.

Figures 3.1, 3.2 and 3.3 show the absorber optimal adjustment, defined in the previous section, does not ensure the whole system vibration stability for all possible disturbance frequencies. In addition, these Figures show the difficulty to realize the vibration stability at low disturbance frequencies. Besides, Figure 3.3 shows the whole system vibration stability is not realized when the absorber is undamped.

3.2 Periodic rectangular spaced pulse disturbance

3.2.1 Whole system vibration stability expression The whole system external energy per disturbance cycle is defined as:

$$W_e = \pi k_1 \sum_{n=1}^{\infty} n U_n^2 X_{1n} \sin \varphi_{1n} \quad (3.4)$$

The whole system dissipated energy per disturbance cycle is defined as:

$$W_d = \pi c \omega \sum_{n=1}^{\infty} U_n^2 W_n^2 n^2 \quad (3.5)$$

where, $W_n = (X_{2n}^2 + X_{1n}^2 - 2X_{2n}X_{1n} \cos(\varphi_{2n} - \varphi_{1n}))^{1/2}$. Therefore, the whole system vibration stability expression is defined as:

$$\sum_{n=1}^{\infty} U_n^2 n (X_{1n} \sin \varphi_{1n} - 2\mu f d W_n^2 n s) = 0 \quad (3.6)$$

The whole system vibration stability is obtained when at least one of the following cases is realized.

$$(U_n^2 = 0) \text{ or } \left(\sum_{n=1}^{\infty} U_n^2 n (X_{1n} \sin \varphi_{1n} - 2\mu f d W_n^2 n s) = 0 \right)$$

3.2.2 Numerical results The case of $U_n^2 = 0$, corresponds to situations where the whole system vibration stability is independent of the absorber parameters. The vibration stability is realized when the disturbance frequency matches one of the absorber independent vibration stability frequencies defined by the above mentioned expression.

The case of $\sum_{n=1}^{\infty} U_n^2 n (X_{1n} \sin \varphi_{1n} - 2\mu f d W_n^2 n s) = 0$, corresponds to situations where the whole system vibration stability is dependent on the absorber parameters. As for the sinusoidal disturbance, to show the influence of the absorber parameters; mass, stiffness, and damping ratios; on the whole system vibration stability, Figs. 3.4, 3.5, and 3.6 are presented. In these Figures, black spaces indicate the whole system vibration stability is realized, and blank ones indicate the vibration stability is not realized. Figures 3.4, 3.5 and 3.6 show the absorber optimal adjustment, defined in the previous section, does not ensure the whole system vibration stability for all possible disturbance frequencies. Also, these Figures show the

Fig. 3.4 Influence of the mass ratio on the whole system vibration stability for the periodic rectangular spaced pulse disturbance

Fig. 3.5 Influence of the stiffness ratio on the whole system vibration stability for the periodic rectangular spaced pulse disturbance

Fig. 3.6 Influence of the damping ratio on the whole system vibration stability for the periodic rectangular spaced pulse disturbance

difficulty to realize the vibration stability at low disturbance frequencies. Besides, Figure 3.6 shows the whole system vibration stability is not realized when the absorber is undamped.

3.3 Conclusion

The absorber efficiency analysis led to define the absorber optimal adjustment giving the lowest vibrational displacement amplitude of the system mass at resonance, when the system is subjected to ground roughness modeled as sinusoidal or periodic rectangular spaced pulse disturbances. Unfortunately, this absorber optimal adjustment does not ensure the whole system vibration stability for all possible disturbance frequencies. Therefore, the following is to be taken into account to guarantee the whole system vibration stability.

(1) If the disturbance is at stable frequency, then a passive absorber may be used, and the whole system vibration stability is realized when at least one of three cases is present. First, when the absorber optimal adjustment guarantee the vibration stability. Second, when the disturbance frequency matches one of the absorber independent vibration stability frequencies. Third, when at least one of the absorber parameters values is changed, closely to its optimum, to ensure the vibration stability.

(2) If the disturbance is at unstable frequency,

then the whole system vibration stability is realized when at least one of two cases is present. First, when the absorber optimal adjustment guarantee the vibration stability, therefore a passive absorber may be used. Second, when at least one of the absorber parameters is variable, closely to its optimal value, to ensure the vibration stability where the absorber optimal adjustment fails to guarantee it, therefore a semi-active or an active absorber is to be used.

On the other hand, the following conclusions can also be made.

(1) The tuned absorber can not be used for the system vibration isolation since it does not guarantee the whole system vibration stability.

(2) The system vibration isolation is difficult to realize at low disturbance frequencies since the whole system vibration stability is not easy to obtain at these frequencies.

(3) If the whole system vibration stability is realized, then the whole system residual vibration per disturbance cycle is identical to zero. Therefore, the used energy-based stability criterion is sufficient, thus no need for a time-based stability analysis.

4. General Conclusion

The absorber efficiency and reliability analysis done for a planar vibrational model of mobile robot vehicle equipped with dynamic vibration absorber, when the vehicle is subjected to ground roughness modeled as sinusoidal or periodic rectangular spaced pulse disturbances, led to:

(1) Determine the absorber optimal adjustment, leading to the absorber best efficiency, for any ground roughness model which can be expressed by Fourier series, in particular periodic disturbances. Also, the ground roughness model may dependent on any parameters independent of the absorber parameters, and may be of stable or unstable frequency.

(2) Determine the frequencies at which the absorber optimal adjustment fails to guarantee the whole sys-

tem vibration stability, and recommend the necessary and sufficient measures to ensure the vibration stability at these frequencies, leading to the absorber best reliability.

(3) Demonstrate first, the tuned absorber can not be used for the system vibration isolation, thus only untuned absorber may be used. Second, the passive absorber can not be used for all possible disturbance cases, thus semi-active or active absorbers are to be used when required. Third, the system vibration isolation is hard to realize at low disturbance frequencies.

(4) The results of the absorber efficiency and reliability analysis may be used for any ground vehicle for which the treated vibrational and roughness models are valid.

Dedication

The paper is dedicated to the memory of Professor Sakiichi Okabe, who passed away in January 27, 1998.

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