External Methods for Evaluating Shock Absorbers for Road Roughness Measurements

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Two new experimental methods are described for the selection of shock absorbers for response-type road roughness meters. The methods allow for verification of the acceptability of the shock absorbers before they are mounted in the road roughness measuring vehicle. In the first method, a programmable shaker table is used to obtain the time response of the shock absorber. In the second method, a simple scotch yoke mechanism is used to produce the frequency response of the shock absorber. In both methods, the test results are compared with the corresponding time domain or frequency domain acceptability limits, which are determined from computer simulation and based on the relevant ASTM standards.

During the development of ASTM E 1082, *Test for Measurement of Vehicular Response to Traveled Surface Roughness (1)*, and of the proposed *Standard Specification for Trailers Used for Measuring Vehicle Response to Roughness (2)*, it became apparent that a method was needed to externally evaluate shock absorbers for cars and trailers used for measuring vehicle response to road roughness. Gillespie et al. *(3)* originally proposed an in-car test conducted on a fabricated calibration surface, but this test is essentially a trial-and-error method that allows an evaluation to insure that the shock absorber remains within specifications. In a companion paper, Kulakowski et al. *(4)* describe a greatly enhanced method for use in calibration of trailers.

The most important improvement of the latter method is that it does not require a specially fabricated calibration surface and can be applied on any road section for which the profile data have been acquired. (It is desirable that the selected test surface have a relatively high level of roughness uniformly distributed along its length.) A computer simulation program is used to calculate maximum and minimum acceptable values of the trailer suspension travel produced when it is towed over the test surface. If the measured suspension travel falls within the acceptable range for a given hysteresis of the axle-body displacement transducer, the shock absorber on board the trailer is then acceptable for roughness testing. If not, a new or different shock absorber would have to be mounted and the testing procedure repeated.

The method presented in this paper allows for testing the shock absorber before it is mounted so that the trial-and-error procedure can be avoided. This method can be used in selecting shock absorbers for both cars and trailers used in measuring road roughness.

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THE EXTERNAL PROCEDURE

In the following external method for the selection of the shock absorber for the road roughness measuring vehicle, the shock absorber is tested outside the vehicle and is mounted only if found acceptable. The time and effort involved in this procedure can thus be considerably reduced in comparison with the internal method described by Kulakowski et al. *(4)*.

The proposed procedure requires the mounting of the shock absorber in a programmable shaker table or in a semiportable scotch yoke mechanism to test its dynamic characteristics. Either test apparatus must meet two specifications. First, the shock absorber should be subjected to forces of amplitudes and frequencies varying over a broad spectrum so that nonlinear effects are averaged out. Second, the particular range of amplitudes and frequencies of the excitation signal should be the same as would have been experienced by the shock absorber if it were mounted in the trailer and run over a calibration surface.

The schematic of the test apparatus is shown in Figure 1a, where C_s represents the damping coefficient of the tested shock absorber. The conditions experienced by the shock absorber in the system shown in Figure 1a should be similar those imposed on the shock absorber mounted in the vehicle represented by a standard quarter-vehicle model shown in Figure 1b. The dynamics of the two systems shown in Figure 1 are compared, first with the test apparatus driven by the shaker table and then by the scotch yoke mechanism.

PROGRAMMABLE SHAKER TABLE TESTING PROCEDURE

The acceptability limits for shock absorbers used in road roughness measurements have been defined in terms of the relative damping coefficient $\gamma = C_s/M_s$, where M_s is a sprung mass of a quarter-car or half-trailer *(1, 2)*. According to ASTM standards, the shock absorber is acceptable if its relative damping coefficient falls within specified limits, that is, if

$$\gamma_{min} \le \gamma \le \gamma_{max} \tag{1}$$

where, for a standard quarter-car,

$$\gamma_{min} = 4.0 \text{ sec}^{-1} \text{ and } \gamma_{max} = 9.5 \text{ sec}^{-1} \tag{2}$$

and, for a standard half-trailer,

$$\gamma_{min} = 5.5 \text{ sec}^{-1} \text{ and } \gamma_{max} = 13 \text{ sec}^{-1} \quad (3)$$

The standard values of γ are 6.0 sec^{-1} and 8.0 sec^{-1} for a quarter-car and a half-trailer model, respectively.

For the external testing procedure with a programmable shaker table, it is better to formulate an equivalent acceptability criterion in the time domain rather than in the domain of the relative damping coefficient. In order to transpose the criterion of Equation 1 to the time domain, assume that a specified calibration profile $x_p(t)$ is used as the input to the quarter-vehicle model shown in Figure 1b. Displacement of the sprung mass $Z_s(t)$ is considered as the output signal. The shock absorber having relative damping coefficient γ is considered acceptable if the quarter-vehicle time response $Z_s(t)$ satisfies the following condition:

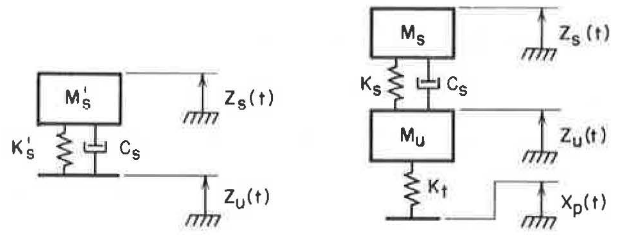

FIGURE 1 Schematic of the test apparatus (a) and of the general quarter-vehicle model (b).

$$Z_s'(t) \leq Z_s(t) \leq Z_s''(t), \text{ for } t_1 \leq t \leq t_2 \quad (4)$$

where

$Z_s'(t)$ = response with $\gamma = \gamma_{max}$,
$Z_s''(t)$ = response with $\gamma = \gamma_{min}$,
t_1 = time at beginning of the test, and
t_2 = time at end of the test.

Figure 2 presents time responses of the half-trailer model *(1)* for γ = 5.5, 8.0, and 13.0 sec^{-1} subjected to the ABAB pattern calibration profile proposed by Gillespie et al. *(3)*. The plots shown in Figure 2 were obtained using the computer simulation program CARTRA developed by Chapman *(5)*. It can be seen that the response of the sprung mass $Z_s(t)$ for γ = 8.0 sec^{-1} lies between the responses obtained for γ_{min} = 5.5 sec^{-1} and γ_{max} = 13.0 sec^{-1}.

In the testing procedure proposed here, the shock absorber is mounted in the test apparatus shown in Figure 1a. In order to subject the shock absorber to the same testing conditions as those generated in the quarter-vehicle model (Figure 1b), the displacement of the unsprung mass $Z_u(t)$, recorded when the standard trailer model (γ = 8.0 sec^{-1}) was run over the ABAB pattern, is used as the input signal to the test apparatus. If the mass and spring constant of the test apparatus are the same as the sprung mass and suspension spring constant ($M_s' = M_s$ and $K_s' = K_s$, respectively), the time domain acceptability criterion

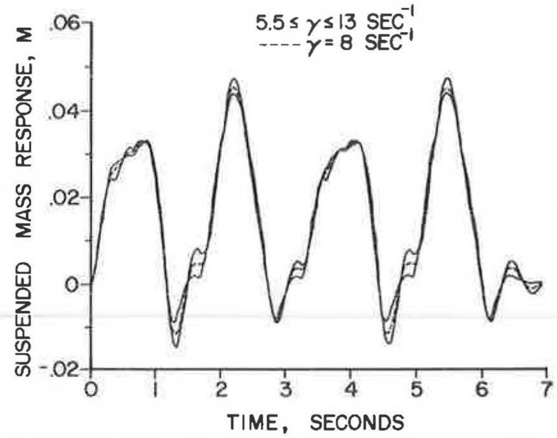

FIGURE 2 Trailer-suspended mass response to ABAB profile.

(Equation 4) can be applied to determine whether the shock absorber is acceptable. Similarly, any road profile $X_p(t)$, not necessarily the ABAB pattern, can be used as the input to the computer simulation program to obtain time responses $Z_u(t)$ and $Z_s(t)$. A programmable shaker table is then used to reconstruct $Z_u(t)$ as the input signal to the test apparatus shown in Figure 1a. In summary, the following step-by-step procedure is proposed.

Step 1: Simulate a standard quarter-vehicle model shown in Figure 1b subjected to a specified input profile $X_p(t)$, and record the response of the unsprung mass $Z_u(t)$. The computer program CARTRA *(5)* can be used.

Step 2: Simulate the quarter-vehicle model again, but now use the limiting values of the relative damping coefficient γ_{min} and γ_{max} and record the resulting displacements of the sprung mass, $Z_s'(t)$ and $Z_s''(t)$, respectively.

Step 3: Use a programmable shaker table to reconstruct $Z_u(t)$ recorded in Step 1.

Step 4: Apply the shaker table signal $Z_u(t)$ to the test apparatus with the tested shock absorber mounted as shown in Figure 1a and record $Z_s(t)$.

Step 5: Compare $Z_s(t)$ recorded in Step 4 with $Z_s'(t)$ and $Z_s''(t)$ obtained in Step 2. The shock absorber acceptability condition is given by Equation 4.

One difficulty with this testing procedure is that the displacements $Z_s(t)$ in the test apparatus and in the quarter-vehicle model can only be compared if the mass and spring constant of the test apparatus are the same as the sprung mass and the suspension spring constant of the vehicle model ($M_s' = M_s$ and $K_s' = K_s$, respectively). The sprung mass is 483.3 kg (1,063 lb) for the standard quarter-car and 304 kg (670 lb) for the standard half-trailer model. A programmable shaker table for such large masses is generally impractical, so a procedure using a reduced mass M_s' was developed for use in the test apparatus. For trailers, Chapman *(5)* proposes the following values of the parameters: M_s' = 22.7 kg (60 lb) and K_s' = 470,000 N/m (1,055 lb/in.). The system with these parameters has the same damping ratio as the suspension of the original trailer, but the resulting natural frequency ω_n' is approximately 13 times higher than the original. Thus the system does not respond the same as the original system in terms of frequency or time. In fact, the small mass M_s' produces a small $Z_s(t)$ if the original $Z_u(t)$ is used to drive the test apparatus.

In order to remedy this problem, a time-scaled $Z_u'(t)$ must be obtained so that the input causes the test apparatus to respond in the manner in which the trailer did. In other words, if the original $Z_u(t)$ has harmonic components of frequencies near ω_n for the original trailer, these same corresponding portions of the new signal $Z_u'(t)$ must cause the test apparatus to respond in the same manner at frequencies near the new ω_n'. This calls for a frequency shift of the original $Z_u(t)$ signal, which can be accomplished through time scaling or time compression. If the old $Z_u(t)$ is compressed in time, all frequencies in the signal will be increased. With the time scaling determined by the compression factor set to the ratio (of 13) of the ω_n divided by the ω_n', the resulting $Z_u'(t)$ and the test apparatus simulate the original system well. Figure 3 shows the time-domain-based acceptability criterion in compressed time scale.

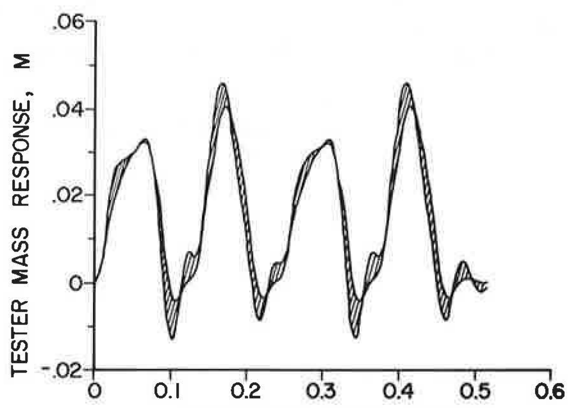

FIGURE 3 The external time-based acceptability criterion for a trailer.

A similar time scaling procedure should be applied for a quarter-car model. The acceptability criterion for shock absorbers mounted in cars is illustrated in Figure 4. The same five-step procedures are followed with the time compression.

Scotch Yoke Mechanism Testing Procedure

A programmable shaker table may not be available for testing shock absorbers as described in the previous section. A simple scotch yoke mechanism can then be used in a frequency-based test. This mechanism can be obtained from the Vehicle-Surface Interaction Program, Pennsylvania Transportation Institute, Pennsylvania State University, University Park, Pennsylvania.

The quarter-vehicle model (Figure 1b) is linear; therefore, if a sinusoidal input signal of frequency ω is applied, the displacements Z_s and Z_u of the sprung and unsprung masses are also sinusoidal of the same frequency, and of amplitudes $|Z_s(j\omega)|$ and $|Z_u(j\omega)|$, respectively. The ratio of the two amplitudes (the magnification factor) is a function of frequency and can be represented by the magnitude of the sinusoidal transfer function

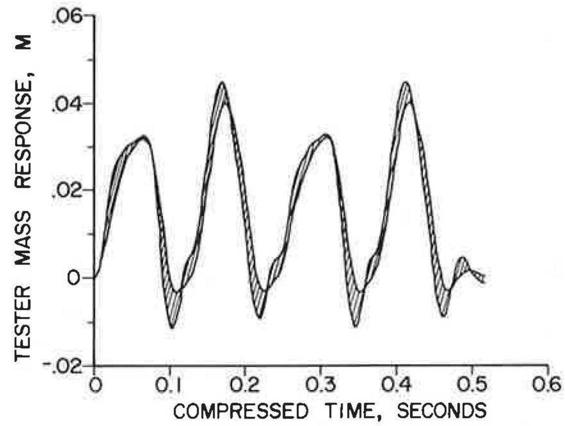

FIGURE 4 The external time-based acceptability criterion for a car.

$$M(\omega) = \frac{|Z_s(j\omega)|}{|Z_u(j\omega)|} \qquad (5)$$

For the quarter-vehicle model the expression for $M(\omega)$ takes the form

$$M(\omega) = \left[\frac{\omega^2\gamma^2 + \omega_n^2}{\omega^2\gamma^2 + (\omega_n^2 - \omega^2)}\right]^{1/2} \qquad (6)$$

where the natural frequency ω_n is

$$\omega_n = \left(\frac{K_s}{M_s}\right)^{1/2} \qquad (7)$$

By substituting $\gamma = \gamma_{min}$ and $\gamma = \gamma_{max}$ into Equation 6, minimum and maximum magnification factor curves $M_{min}(\omega)$ and $M_{max}(\omega)$, respectively, are obtained. The curves calculated for a half-trailer are plotted in Figure 5, and the curves for a quarter-car are shown in Figure 6.

A scotch yoke mechanism can be used to apply a sinusoidal

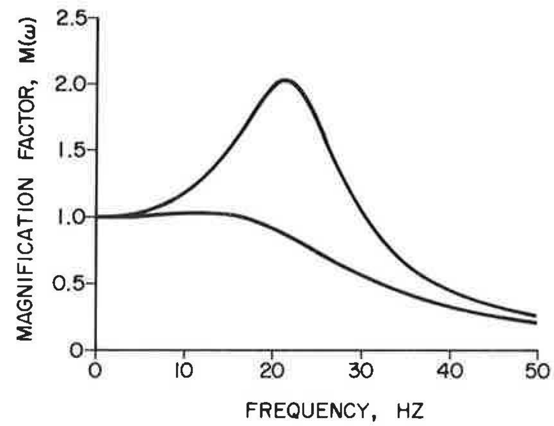

FIGURE 5 Trailer shock absorber acceptability criterion.

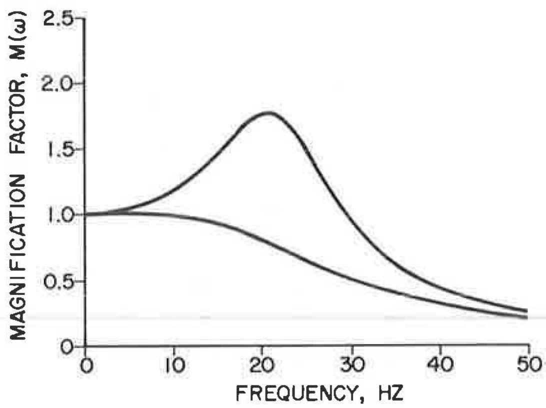

FIGURE 6 Car shock absorber acceptability criterion.

signal of amplitude $|Z_u|$ and frequency ω to the test apparatus shown in Figure 1a. If the amplitude of the steady state sinusoidal displacement of the mass $|Z_s|$ is such that the ratio $|Z_s|/|Z_u|$ lies within the minimum and maximum magnification factor curves shown in Figure 5 or 6, the shock absorber is acceptable. Mathematically, the acceptability condition in frequency domain can be written in the following form:

$$M_{\min}(\omega) \leq M(\omega) \leq M_{\max}(\omega) \text{ for } 0 \leq \omega \leq \omega_{\max} \quad (8)$$

This form is equivalent to the original acceptability condition (Equation 1) as well as to the time domain criterion (Equation 4).

CONCLUSIONS

Two experimental methods for verification of acceptability of shock absorbers for use in road roughness measuring systems were described. In the two approaches, the general acceptability criteria for shock absorbers used in roughness-measuring cars and trailers stated in the ASTM Standards *(1, 2)* are transposed to the time and frequency domains. The testing procedures are simple and, more important, allow for the determination of shock absorber acceptability before it is mounted in a car or trailer. Also, the procedures are sufficiently flexible to be customized according to the type of equipment available and the specific needs of the user.

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