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Dynamic Testing of Elastomer Mountings

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Synopsis

Elastomer mountings are used extensively for impact and shock absorption and for the control of noise and vibration. Dynamic testing has become increasingly important in the formulation, design and evaluation of effective mount materials. A servo-controlled hydraulic tester has been used for the forced vibration testing of various elastomer compounds. The essential dynamic properties of such materials can be obtained directly over a broad range of operating conditions. Frequency, amplitude, and preload can be independently controlled and different test temperatures can be obtained through the use of an insulated control chamber. A production type automotive engine mount was used for most of the tests. Experimental compounds were molded into test mounts and checked for dynamic spring rate and coefficient of damping. The effect of different operating conditions such as frequency, amplitude, preload, and temperature on the dynamic properties of butyl mounts is described. Data are also presented on the role of compounding in altering the properties of mounts. Some data are also presented on the characteristics of natural rubber, SBR, and EPR mounts. The dynamic properties of the mount determine its performance features in a mounting system. The transmission of forces in a simple mounting system can be calculated using dynamic property data and the appropriate transmissibility equation. This has been done for several of the mounts tested and the vibration isolation characteristics of each are discussed. The data illustrate the effect of mount properties on transmissibility.

INTRODUCTION

Elastomer mountings are extensively used for the control of noise and vibration. They offer many advantages over mechanical systems, perhaps the most important of which are simplicity of design, versatility of operation, and relatively low cost. With the almost continuous development of new and improved elastomers, a very broad range of operating characteristics and environmental resistance are available to the designer, and there is every indication that the effectiveness and usage of elastomer mountings will increase considerably in the years ahead.

Many of the problems encountered in using elastomeric mountings can be related to the general properties of elastomers. The dynamic properties of viscoelastic materials, which control mount performance, are not constants, but can vary widely due to changes in frequency, amplitude, temperature, and other operating conditions, and these conditions must be taken account of in any thorough evaluation of mount performance. A second source of problems is the wide variety of materials available for any given application. Not only are there many base elastomers being marketed, but there is a virtually limitless number of compounds which can be made from them through changes in filler systems, plasticizers, and cure systems.

In view of these problems, it is not surprising that dynamic testing of elastomer mounts has become an increasingly important tool in obtaining design, performance, and quality control data. This paper describes a test unit currently being used in all three capacities and presents some recent data obtained on mounts made from butyl rubber and some other elastomers.

TEST APPARATUS

The tester is a hydraulically-powered, servo-controlled, forced-vibration unit. Hydraulic power is supplied through a 3000 psi, 30 gpm pump driven by a 50 horsepower electric motor. The hydraulic power supply and the electronic console are remotely located from the test stand. The

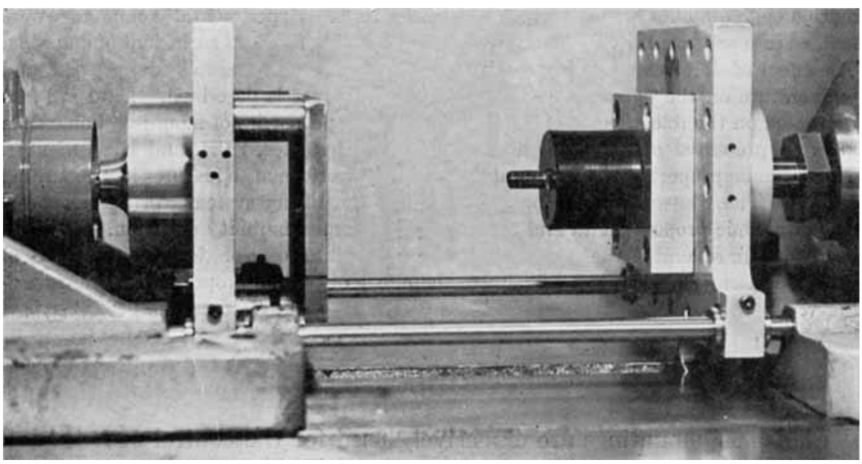


Fig. 1. Dynamic test apparatus,

electronic console contains an oscilloscope, power supply, feedback system, function generator, and amplifiers—all rack mounted. A photograph of the test stand is shown in Figure 1, and a drawing of the essential components in Figure 2. The position of the ram is monitored by a potentiometer. Two potentiometer signals are used—one for display and the other for feedback control. The load signal is generated by a strain gauge load cell mounted directly behind the test specimen. The load signal is also fed to an oscilloscope for display and recording. An insulated temperature control box is used for low and high temperature tests. The box completely encloses the sample, ram head, and backing plate, but does not cover the load cell. The box has a large transparent door for visual observation and manipulation of the test sample. Good temperature control has been obtained in the range of −50 to 250°F.

As it is now set up, the tester is displacement controlled, although with the necessary modifications, force, or velocity commands could be used. The absolute position of the ram plus its mode, amplitude, and frequency of oscillation are controlled by the system composed of the potentiometer, signal generator, servo-valve, and feedback amplifier.

The tester has been designed for a maximum of 10,000 lb. total force and a maximum ram velocity of 40 in./sec. Maximum ram stroke is 10 in.

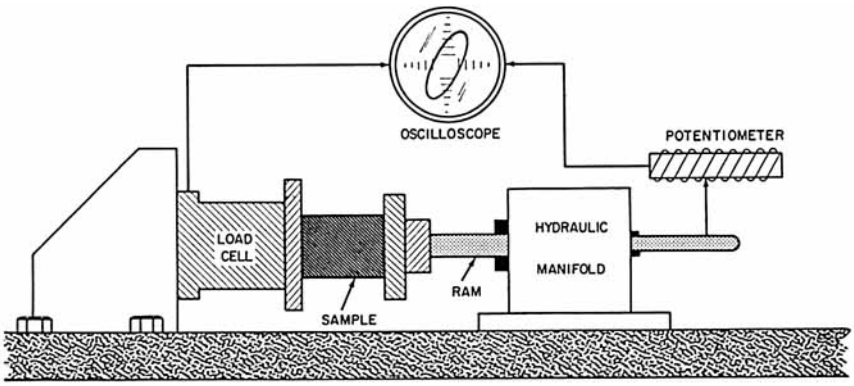


Fig. 2. Dynamic test apparatus.

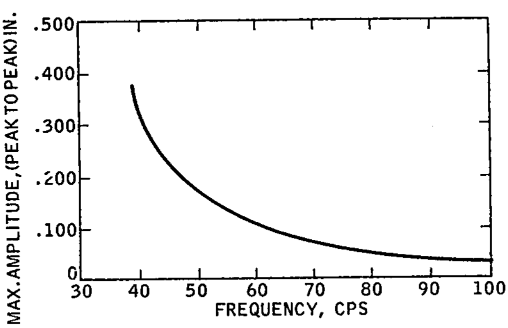


Fig. 3. Effect of frequency on maximum amplitude 500 lb. preload.

Within these limits, any part can be tested in tension or compression using sine, square, or triangular waves and employing frequencies of from 0.01 to 100 cps. Preload (or prestrain), amplitude, and frequency are independently controlled and can be easily and quickly changed during a test.

Figure 3 shows the maximum amplitude (sinusoidal) obtained on a test engine mount using various frequencies and a 500-lb. preload. Although there would appear to be plenty of capacity for low amplitude tests above 100 cps, control of ram position is poor above 50 cps using the present control system.

TEST METHOD AND RESULTS

Most of the tests were performed on automotive engine mountings molded from several elastomers and various compound formulations. A photograph of a test mount is shown in Figure 4. The part is two inches high and approximately 2.4 in. in diameter and represents a typical compression-type production auto mount. Metal end plates with mounting studs are bonded to the rubber during vulcanization. The parts were compression molded, the usual butyl cure being 40 min. at 320°F. A typical butyl engine mount formulation is shown in Table I. Physical property data obtained on test samples are also given. In use, these parts

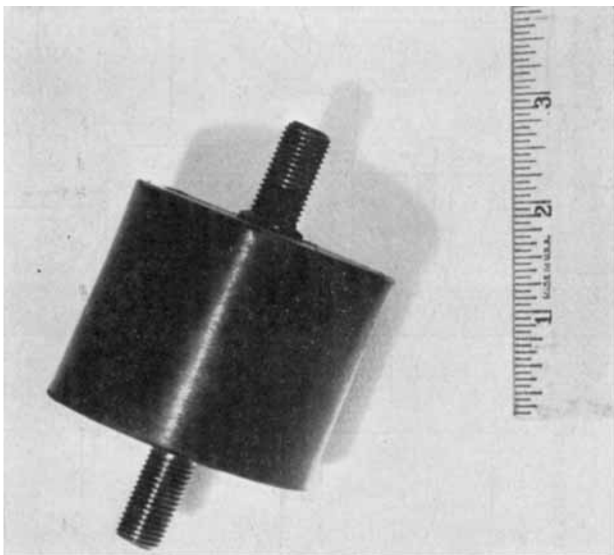


Fig. 4. Test mount.

are mounted vertically (parallel to the cylinder axis) and support a preload of about 250 lb. Although engine mounts perform many functions, their primary one is to suppress the transmission of vibration to the car frame and, ultimately, the passenger compartment.

The mounts were tested under a constant preload and fixed sinusoidal dynamic deflection as follows: (*1*) Mounts were conditioned for one hour at the test temperature; (*2*) Preload (usually 250 lb.) was applied by adjusting the static ram position; (*3*) A dynamic deflection of fixed frequency and amplitude was superimposed on the preload; (*4*) The part was conditioned under operating frequency for 10–15 min. Preload was maintained manually; (*5*) The oscilloscope trace of stress versus strain was photographed.

The photograph of the hysteresis loop was analyzed to obtain the desired dynamic property data. In general, two values are reported for each test.

TABLE I
Butyl Engine Mount Compound

	Parts by weight
Enjay butyl 268	100
HAF carbon black	50
Flexon 845 process oil	25
Zinc oxide	5.0
Cadmium diethyldithiocarbamate	2.0
Benzothiazyl disulfide	0.5
Sulfur	1.5
Original Physical Properties	
Cure 20 min. at 320°F.	
Shore A hardness	51
100% Modulus, psi	170
300% Modulus, psi	840
Tensile strength, psi	2020
% Ultimate elongation	580
Compression set B,[a] %	
22 hr. at 158°F.	17

[a] Pellets cured 35 min. at 320°F.

1. Dynamic Spring Rate, lb./in.

This is analogous to the complex dynamic modulus and contains both real and imaginary components.

2. Coefficient of Damping, lb.-sec./in.

The damping coefficient was used to measure the damping capacity or hysteresis of the mounts. The loss rate (analogous to the imaginary part of the complex modulus) can be obtained by multiplying the coefficient of damping by the frequency.

Details of the methods used in calculating these values from the hysteresis loop are given in the Appendix.

EFFECT OF OPERATING CONDITIONS ON DYNAMIC PROPERTIES

Figures 5 and 6 show the effect of tensile and compression preloads on the dynamic properties of a butyl mount. The mount was tested at room temperature under constant frequency (12 cps) and dynamic amplitude (0.100 in., peak-to-peak). Dynamic spring rate increases rapidly under increasing compression preload, since the mount is forced to operate at a steeper portion of the stress-strain curve. Dynamic rate does not change as rapidly with changing tensile preloads in the range studied. The decrease in dynamic rate and damping with increasing low tensile preloads has been noted previously in tests of this kind. At high tensile preloads, rate increases with increasing preload. In general, the dynamic rate

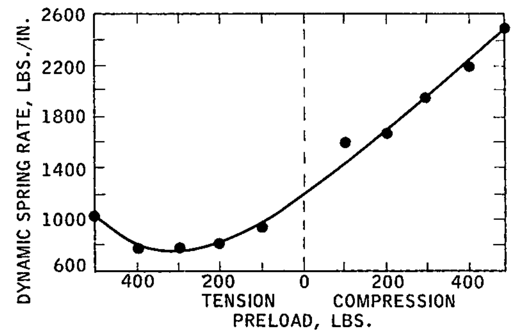


Fig. 5. Effect of preload on dynamic spring rate. Butyl mounts: 75°F., 12 cps.

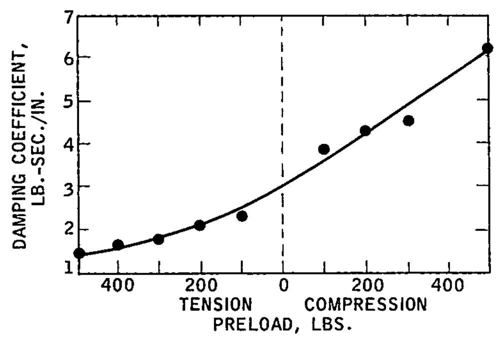


Fig. 6. Effect of preload on damping coefficient. Butyl mount: 75°F., 12 cps

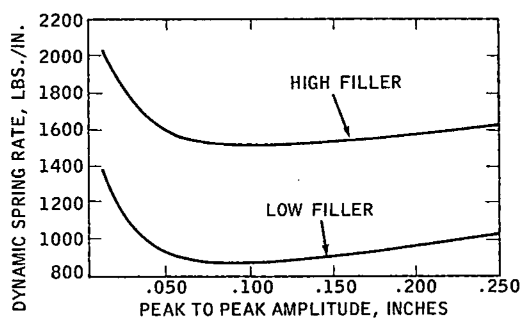


Fig. 7. Effect of amplitude on dynamic spring rate. Butyl mounts: 75°F., 250 lb., preload 12 cps.

versus preload curve follows the static stress-strain curve. Both of these effects, then, can be explained on the basis of the sigmoidal shaped static stress-strain curve for rubbers in tension. At low strains, the slope of the curve is high, but decreasing, as is the dynamic rate curve. At high strains, the curve is concave upward, and this also is reflected in the dynamic rate curve which increases with preload at high preloads.

Figure 7 shows the effect of dynamic amplitude on the dynamic spring rate of two butyl mounts containing low and high filler loadings (HAF carbon black). The pronounced decrease in spring rate with increasing amplitude at low amplitude is common in rubber vulcanizates. It is presumably due to filler effects including polymer to carbon black bonding and carbon black "structure." It is interesting to note that the magnitude of the effect is similar for low and high loadings of carbon black.

The effect of test frequency on the dynamic properties of a butyl mount is illustrated in Figure 8. The increase in elastic spring rate (the in-phase component of dynamic spring rate) with frequency is to be expected although the shape of the curve has probably been modified by the increase in running temperature of the sample. The rapid decrease in

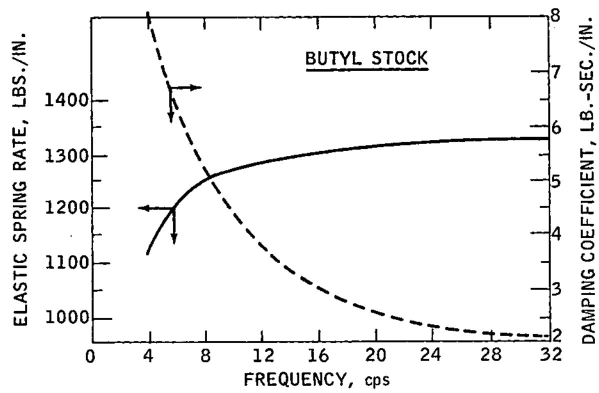


Fig. 8. Elastic spring rate and damping versus frequency: 75°F., 250 lb. preload, DA = 0.100 in.

damping coefficient is surprising, but can also be explained on the basis of heat build-up in the mount as frequency is increased (since the tests were performed at constant amplitude). It should be remembered that the values shown are for damping coefficient, not loss rate or damping force. Loss rate (imaginary component of dynamic spring rate) increased with increasing frequency as shown below.

	8 cps	12 cps	16 cps	32 cps
Damping coefficient, lb.-sec./in.	5.1	3.9	3.0	2.0
Loss rate, lb./in. (damping coefficient × frequency)	260	290	300	400

The change in mount properties with frequency is important in assessing the performance characteristics of a mount system. In many cases, elastic spring rate and damping coefficient are assumed constant over the entire operating frequency range of a system. Since they obviously are not, testing over a range of frequencies is desirable in estimating mount performance.

EFFECT OF COMPOUNDING ON DYNAMIC PROPERTIES

Figures 9 and 10 show the effect of carbon black content on the dynamic properties of butyl mounts at four different frequencies. The stocks tested were identical except for the amount of carbon black (furnace type, HAF) used in them. Increased carbon black loading leads to increased

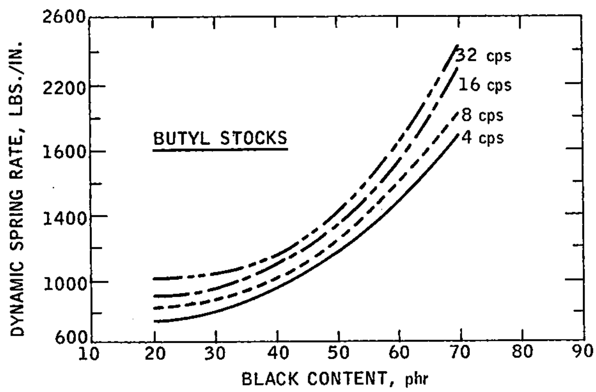


Fig. 9. Dynamic spring rate versus black content: 250 lb. preload, DA = 0.100 in., 75°F.

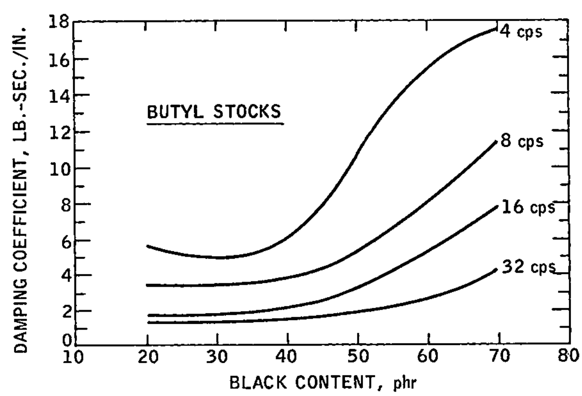


Fig. 10. Damping coefficient versus black content: 250 lb. preload, DA = 0.100 in., 75°F.

rate and damping at all frequencies. Figure 10 also indicates that changes in damping with frequency are greater at higher black loadings. Figure 11 shows the effect of temperature on the dynamic spring rate of three of these same black filled butyl stocks. The 40, 55, and 60 Shore A hardness stocks contain, respectively, 30, 60, and 70 phr carbon black. Ambient temperature is an important operating variable affecting mount performance. Rate increase due to decreasing temperature can shift the natural frequency to higher frequencies and thus cause poor performance. Al-

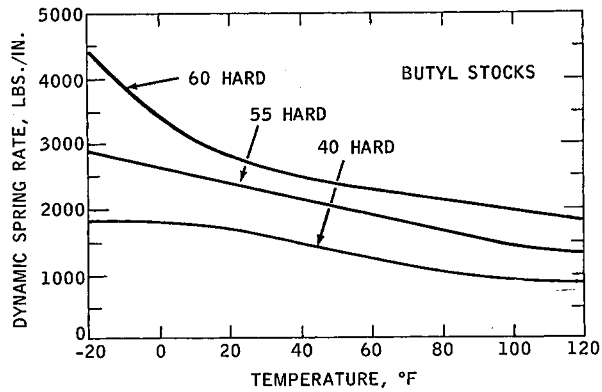


Fig. 11. Dynamic spring rate versus temperature: 12 cps, 250 lb. preload, DA = 0.100 in.

though all the stocks tested were affected by temperature changes, the effects are greatest at high carbon black loadings. The data in Figures 10 and 11 indicate that for butyl stocks, high carbon black loading levels tend to increase mount sensitivity to frequency and temperature changes. In the stocks tested, all other compounding ingredients, including plasticizer, were held constant. Increased black loading would have a less deleterious effect on frequency and temperature stability if plasticizer content was increased proportionately.

The effect of carbon black type on butyl mount properties is shown in Table II. The carbon blacks used in the formulation are listed in order of decreasing particle size. All three stocks were compounded to 50 Shore A hardness by using the different amounts of black shown.

The differences in reinforcing capabilities of each black is indicated by the amount necessary for 50 Shore A hardness. However, even at equal Shore A, the "soft" thermal black shows a lower dynamic spring rate and less damping than the more highly reinforcing furnace blacks.

TABLE II
Effect of Carbon Black Type on Mount Properties[a]

		Dynamic spring rate, lb./in.		Damping coefficient, lb.-sec./in.	
Black type	Amount	DA = 0.025	DA = 0.100	DA = 0.025	DA = 0.100
MT (medium thermal)	135 phr	2010	1510	5.23	4.14
HAF (high abrasion furnace)	50 phr	2200	1540	5.99	5.05
SAF (super abrasion furnace)	40 phr	2280	1670	8.08	5.82

[a] Mounts were tested at 75°F., 250 lb. preload, 12 cps and 0.025-in. and 0.100 in. double amplitudes.

The dynamic properties of rubber mounts are also sharply affected by plasticizer, or process oil loading levels. This is illustrated in Table III for a series of butyl compounds. The stocks are identical except for the process oil levels indicated. All compounds contain 50 phr carbon black.

TABLE III
Effect of Process Oil on Butyl's Dynamic Properties[a]

Process oil (phr)	Shore A hardness	Dynamic spring rate (lb./in.)	Damping coefficient, (lb.-sec./in.)
10	59	1920	6.6
15	54	1800	5.9
20	50	1500	5.1
25	48	1430	4.2
30	45	1370	3.6
35	43	1190	3.1
40	40	1050	2.3

[a] All stocks contain 50 phr HAF carbon black.

Because of its low unsaturation (hence limited functionality) the physical properties of butyl vulcanizates are not changed greatly by sulfur level. However, increasing sulfur does result in an increase in dynamic spring rate and damping coefficient as shown below for a series of mounts tested at 12 cps and 0.025 in. double amplitude using a 250 lb. preload.

Sulfur content	Dynamic spring rate, lb./in.	Damping coefficient, lb.-sec./in.
1.0 phr	1650	3.81
1.5 phr	1700	4.95
2.0 phr	1890	5.85

Chemical promoting agents have been used extensively in butyl compounding. These agents promote polymer to polymer and/or polymer to pigment bonding which can significantly affect dynamic properties. For instance, promoters are used in butyl tire carcass stocks to improve flex life and reduce heat build-up. The effect of a promoter on the dynamic stiffness and damping of two butyl mounts is clearly indicated below.

	Dynamic spring rate, lb./in.	Damping coefficient lb.-sec./in.
Butyl (non-promoted)	2850	6.91
Butyl (promoted[a])	2410	4.95

[a] Compound was treated with 1.0 phr Elastopar (*N*-methyl-*N*,4-dinitrosoaniline). Mounts tested at room temperature, 12 cps, 250 lb. preload, and 0.025 in. double amplitude.

Butyl rubbers are made with various levels of unsaturation. Within the relatively narrow range available, however, unsaturation has only a small effect on dynamic properties. In the data presented below the formulations were identical except for polymer type and both mounts were vulcanized under the same conditions.

% Unsaturation	Dynamic spring rate,[a] lb./in.	Damping coefficient, lb.-sec./in.
(2.1–2.5)[b]	2000	5.31
(1.5–2.0)[c]	1890	5.85

[a] Tested at RT, 12 cps, 250 lb. preload, 0.025 in. double amplitude.
[b] Enjay Butyl 325.
[c] Enjay Butyl 268.

Figure 12 shows the dynamic properties of typical butyl and natural rubber mounts over a range of ambient test temperatures. Damping for the butyl mount is significantly higher than the natural rubber part throughout the temperature range, −20 to 120°F. Dynamic spring rate is higher for the butyl mount but this is probably due to the difference in

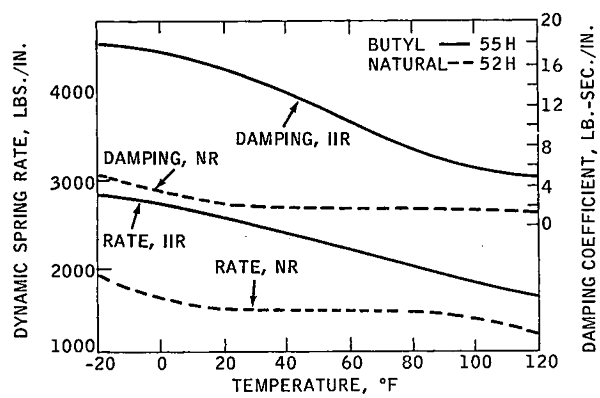


Fig. 12. Dynamic spring rate and damping versus temperature of butyl and natural rubber: 12 cps, DA = 0.100 in., 250 lb. preload.

Shore A hardness (55 for butyl, 52 for natural rubber). The butyl mount shows more sensitivity to temperature change than the natural rubber part with respect to dynamic spring rate and damping, although both mounts show relatively small changes for the broad range of temperatures involved. In both cases, compounding modifications can be used to further improve temperature stability.

Figure 13 shows similar data for EPR* and SBR engine mounts of about equal Shore A hardness. For the test conditions shown the EPR part showed higher resilience (less damping) and less temperature sensitivity

* Ethylene–propylene copolymer—Enjay EPR 404.

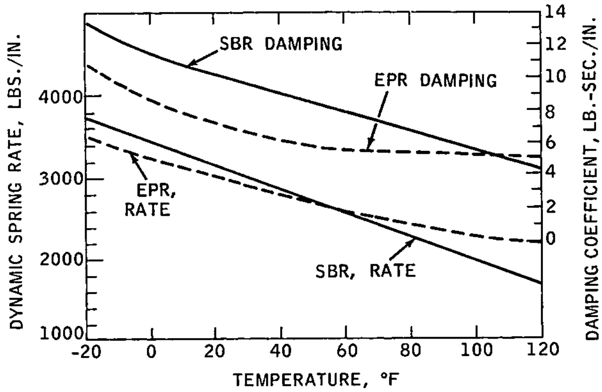


Fig. 13. Dynamic spring rate and damping versus temperature of EPR and SBR: 12 cps, DA = 0.100 in., 250 lb. preload.

than the SBR mount. Although different types of EP rubbers will very probably have different dynamic properties, these materials should find use in many mount applications where high damping is not needed and where excellent resistance to heat, age, and ozone are important factors.

PERFORMANCE OF ELASTOMER MOUNTINGS

The performance of mounts can be estimated by using the experimental dynamic property data and the appropriate transmissibility equation. In most cases, many simplifying assumptions must be made, since the actual application may be considerably complex. This is certainly true of automotive engine mount systems. Nevertheless, the importance of various factors can be illustrated in a general way by using calculated values for transmissibility.

In the examples shown, transmissibility was calculated for a simple single degree of freedom mounting system. The mass of the system was taken equal to the preload used in the experimental tests, and experimental values for elastic spring rate and coefficient of damping were used to characterize the spring system. Details of the calculations used can be found in the appendix. The equation for transmissibility used is shown below.

$$\text{Transmissibility} = \frac{\text{Force Output}}{\text{Force Input}}$$

$$= \sqrt{\frac{1 + \left[\frac{c\omega}{k}\right]^2}{\left[1 - \frac{\omega^2}{{\omega_n}^2}\right]^2 + \left[\frac{c\omega}{k}\right]^2}}$$

In a given mounting system, then, transmissibility depends on the following: ω is the frequency of impressed force; ω_n is the natural frequency;

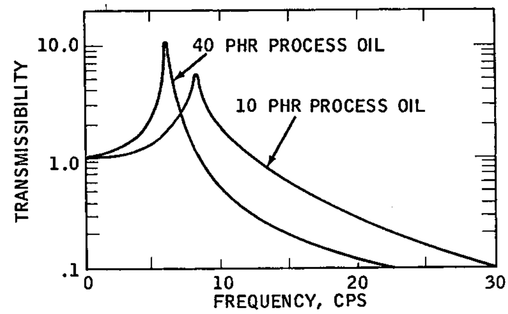


Fig. 14. Transmissibility—high and low oil loading.

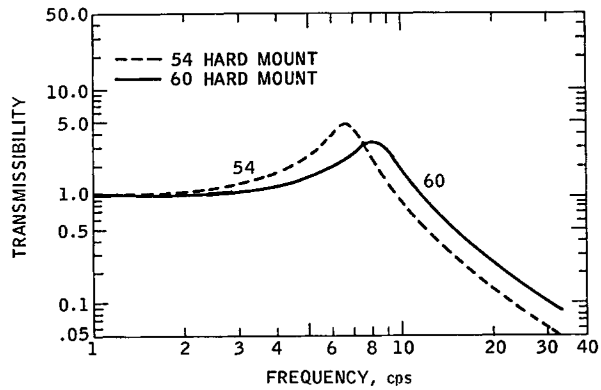


Fig. 15. Transmissibility—butyl mounts.

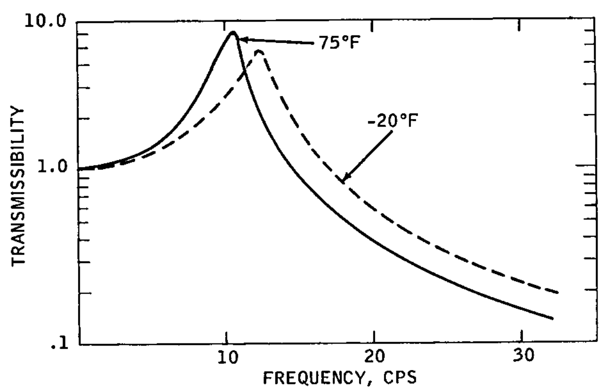


Fig. 16. Transmissibility—EPR mounts.

c is the coefficient of damping; and k is the elastic spring rate (real part of the dynamic spring rate). Usually, in calculating the transmissibility of a given spring/mass system ω_n, c, and k are taken as constants, and the response of the system depends only on ω, the frequency of the impressed force. Of course, the data already presented have shown that c and k

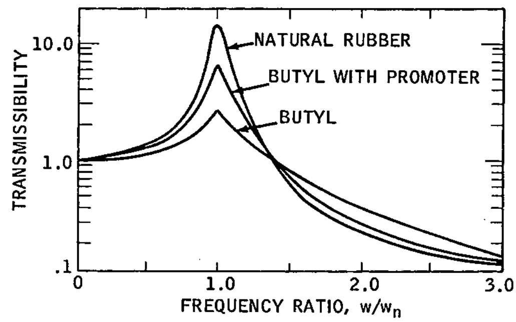


Fig. 17. Transmissibility—butyl and natural rubber mounts.

are not constants. For viscoelastic materials c and k are significantly influenced by preload, frequency, temperature and many other operating and compounding factors. Thus, in many cases, the calculated transmissibility of a system may represent only an approximation of its actual performance characteristics.

The effect of compounding changes on dynamic properties has already been discussed. Their influence on transmissibility is shown in Figures 14 and 15. Figure 14 shows the calculated transmissibility of two butyl mounts having different amounts of process oil in the formulation (the data are presented in Table III). An increase in process oil reduces dynamic spring rate and damping coefficient, which results in a lower resonant frequency and increased transmissibility at resonance. Figure 15 presents similar data for two higher damping butyl mounts of different Shore A hardness.

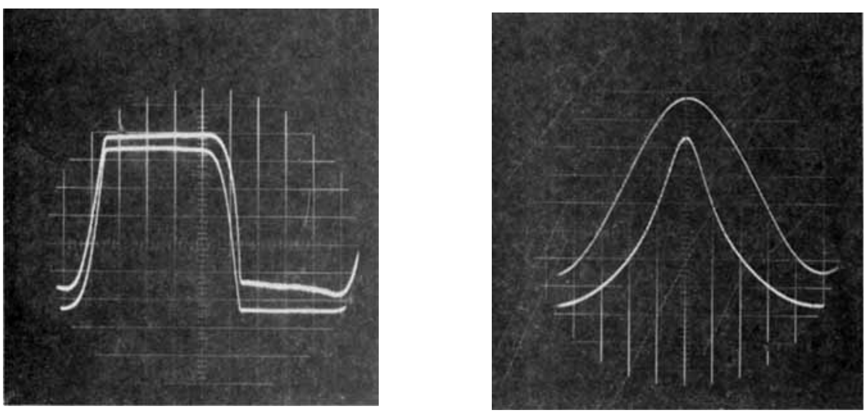
The transmissibility of an EPR mount at two different temperatures is shown in Figure 16. The effect of reduced temperature is to increase dynamic spring rate and damping. Considering the range of temperatures involved (−20 to 75°F) the shift in the transmissibility curves is quite small and represents good temperature stability on the part of this EPR stock.

Figure 17 illustrates the performance of three mounts tested at room temperature. The great difference between low damping (natural rubber) and high damping (butyl) systems is clearly shown. In general, high damping systems are much superior in transmissibility near the resonant frequency of the system. Mounts with low damping show less transmissibility at the higher frequencies. The effect of a chemical promoter is also shown. In this case, the addition of 1.0 phr promoter to the butyl formulation resulted in resonant properties falling roughly midway between natural rubber and the original butyl formulation.

SINUSOIDAL VERSUS SQUARE WAVE FORCING

Sinusoidal and square wave forcing was used in a series of tests on butyl rubber automotive suspension bumpers. A photograph of the test bumper

Fig. 18. Automotive suspension bumper.

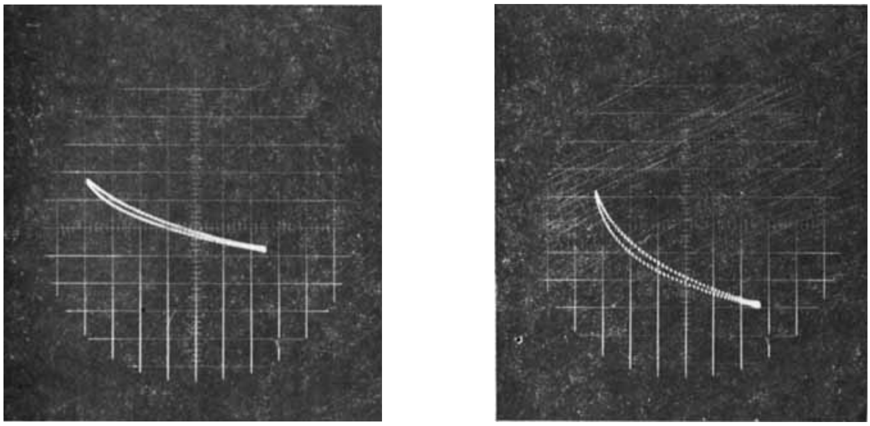


← TIME ← TIME

(a) (b)

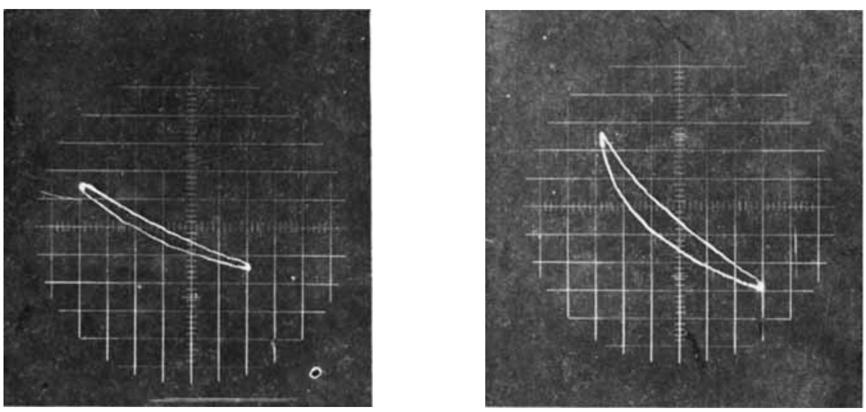
Fig. 19. Wave forms: (a) square wave forcing—upper curve is deflection, and lower is force; (b) sinusoidal forcing —upper curve is deflection and lower one is force

is shown in Figure 18. It is two inches square at the base and three inches high. In use, these parts limit the vertical deflection of the rear axle under jounce conditions and are subjected to fairly rapid impact loads. Gross deflections of 50–60% of original part height can be encountered under rough road conditions. The function of the bumper is to limit axle travel while at the same time transmitting the minimum amount of disturbing forces to the car frame and body.



(a) (b)

Fig. 20. Sinusoidal forcing—butyl suspension bumpers. Force versus deflection curves. y axis is force, x axis deflection. (a) low filler content; and (b) higher filler content.



(a) (b)

Fig. 21. Square wave forcing—butyl suspension bumpers. Force versus deflection curves using square wave forcing. y axis is force, x axis deflection. (a) low filler content; and (b) higher filler content.

It was felt that square wave forcing would more closely similate impact loading. Two butyl bumpers were tested at 30 cpm using a 1.5 in. peak-to-peak amplitude (50% of original part height). The load and deflection signals for both wave forms are shown in Figure 19. It is interesting to note that the sinusoidal deflection signal is near perfect, whereas the resultant force trace is considerably different in form. This is probably due to the geometry of the part. The paraboloid shape results in a very sharp rise in force as deflection increases. The deflection signal for the square wave is rounded slightly, but the initial rise is good. Time to peak deflection was about 200 milliseconds (for 1.5 in.).

Figures 20 and 21 show the hysteresis loops for both bumpers using each wave form. The softer bumper was made from a butyl compound containing a very low filler loading (10 phr carbon black), while the more highly loaded compound contained 75 phr carbon black and 25 phr process oil. The oscilloscope traces show very clearly the higher peak loads and greater energy absorption or damping obtained under square wave forcing. The data are summarized below.

	Sinusoidal		Square wave	
	Peak load, lb.	Damping[a]	Peak load, lb.	Damping[a]
Butyl—low filler	260	39	310	74
Butyl—high filler	420	58	580	100

[a] Arbitrary energy units/cycle.

CONCLUSIONS

The hydraulic test apparatus described presents an effective and versatile method for determining the dynamic properties of full size rubber parts over a very broad range of operating conditions. The data indicate that test amplitude, frequency, preload, and temperature have significant effects on the dynamic stiffness and damping of elastomer mountings. Of course, dynamic properties are also dependent on the base elastomer and the compounding techniques used in formulating a particular compound. All three factors—base elastomer, compounding, and operating conditions—are important in determining the actual performance characteristics of elastomer mounts.

APPENDIX

1. Details of Dynamic Calculations

A sketch of the oscilloscope trace photographed during the test is shown below. The y axis represents the force in pounds developed in the mount and measured by the load cell. The x axis represents deflection as measured by potentiometer.

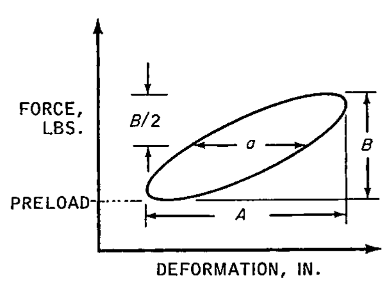


Fig. 22. Sketch of the oscilloscope trace photographed during test.

Dynamic Spring Rate, or the complex rate is measured by taking the slope of the major axis of the hysteresis loop:

$$\text{Dynamic Spring Rate} = \text{B/A (lb./in.)}$$

Coefficient of Damping (c) is calculated from the loss angle θ. The loss angle is defined here as the phase angle between the *dynamic spring rate* and the *elastic spring rate*. The phase diagram is shown below:

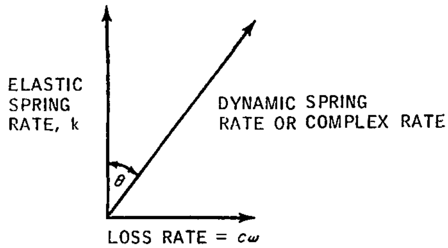


Fig. 23. Phase angle between the dynamic spring rate and the elastic spring rate.

The calculation used is:

$$\text{sine } \theta = \frac{a}{A}$$

$$\text{sine } \theta = \frac{a}{A} = \frac{\text{Loss Rate}}{\text{Dynamic Spring Rate}}$$

$$= \frac{c\omega}{\text{Dynamic Spring Rate}}$$

$$\text{Coefficient of Damping} = c$$

$$= \frac{\text{Dynamic Spring Rate}}{\omega}\left[\frac{a}{A}\right]$$

Units are lb.-sec./in.

Coefficient of damping can also be calculated directly from the area of the hysteresis loop based on considerations of the work done by the loss rate/cycle.

$$\text{Coefficient of Damping} = \frac{(\text{Area of Loop})}{\pi\omega\left[\frac{A}{2}\right]^2}$$

The *Elastic spring rate* (k) is calculated from the dynamic spring rate based on the relationship shown in the phase diagram.

$$\text{Elastic Spring Rate} = \sqrt{(\text{dynamic spring rate})^2 - (\text{loss rate})^2}$$

or

$$k = (\text{dynamic spring rate}) \cos \theta$$

Units are lb./in.

Résumé

Des supports en élastomère ont été beaucoup utilisés pour l'absorption de l'impact et du choc et pour le contrôle du bruit et des vibrations. Les essais dynamiques sont devenus de plus en plus importants pour la formulation, le but et l'évaluation des matériaux de support effectifs. Un appareil de test à servo-contrôle hydraulique a été employé pour l'essai de la vibration forcée de différents composés élastomères. Les propriétés dynamiques essentielles de ces matériaux peuvent être obtenues directement dans un large domaine de conditions opératoires; La fréquence, l'amplitude et la précharge peuvent être contrôlés indépendamment, et on peut obtenir différentes températures d'essai en employant une chambre de contrôle isolée. Un support du type production automatique a été employé pour la plupart des tests. Des composés expérimentaux ont été coulés dans des supports; ests et testés au point de vue de la vitesse d'élasticité dynamique et du coefficient d'amortissement. L'effet de différentes conditions opératoires telles que la fréquence, l'amplitude, la précharge et la température sur les prorietés dynamiques des supports en butyle est décrit. On présente également des données sur le contrôle du composant dans l'altérations des propriétés des supports. Certains résultats sont également présentés sur les caractéristiques des supports en caoutchouc naturel, en SBR et EPR. Les propriétés dynamiques du support déterminent ses performances dans un système du support. La transmission des forces dans un système du support simple peut être calculée en utilisant les données de propriétés dynamiques et l'équation de transmissibilité appropriée. Cela a été fait pour plusieurs de supports testés et les caractéristiques d'isolation vis-à-vis de la vibration ont été discutés pour chacun des supports. Les résultats illustrent l'effet des propriétés du support sur la transmissicibilité.

Zusammenfassung

Elastomermontierungen werden in grossem Masse für Stoss- und Schlagabsorption und für die Kontrolle von Lärm und Vibration verwendent. Dynamische Testmethoden haben für den Aufbau und den Entwurf und die Berechnung von wirkungsvollem Montierungsmaterial immer mehr an Bedeutung gewonnen. Ein hydraulischer Tester mit Servosystem wurde für die Testung verschiedener elastomerer Verbindungen mit erzwungener Schwingung benützt. Die wesentlichen dynamischen Eigenschaften solcher Stoffe können direkt über einen breiten Bereich von Arbeitsbedingungen erhalten werden. Frequenz, Amplitude und Vorlast können unabhängig kontrolliert und verschiedene Testtemperaturen durch Verwendung einer isolierten Kontrollkammer erhalten werden. Für die meisten Tests wurde eine technische Maschinenmontierung benützt. Die Versuchsverbindungen wurden zu Testmontierungen geformt und auf dynamische Federungsgeschwindigkeit und Dämpfungskoeffizienten überprüft. Der Einfluss verschiedener Arbeitsbedingungen, wie Frequenz, Amplitude, Vorlast und Temperatur auf die dynamischen Eigenschaften von Butylmontierungen wird beschrieben. Weiters werden Ergebnisse bezüglich der Rolle der Mischung für die Eigenschaften der Montierung und auch zur Charakterisierung von Montierungen aus Naturkautschuk, SBR und EPR vorgelegt. Die dynamischen Eigenschaften der Montierungen bestimmen ihr Beanspruchungsverhalten in einem Montierungssystem. Die Übertragung von Kräften in einem einfachen Montierungssystem kann aus dynamischen Daten und der geeigneten Transmissionsgleichung berechnet werden. Dies wurde für einige getestete Montierungen durchgeführt und die Schwingungsisolierungscharakteristik in jedem Fall diskutiert. Die Ergebnisse zeigen den Einfluss der Eigenschaften des Montierungsmaterials auf die Transmissionsfähigkeit.