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USAAVLABS TECHNICAL REPORT 69-20

FEASIBILITY STUDY OF NON-NEWTONIAN GELLED FLUIDS AS A SHOCK- AND VIBRATION-ISOLATION MEDIUM

By

E. D. Griffith

April 1969

U. S. ARMY AVIATION MATERIEL LABORATORIES
FORT EUSTIS, VIRGINIA

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THE WESTERN COMPANY
RESEARCH DIVISION
RICHARDSON, TEXAS

[redacted]

U. S. ARMY MATERIEL COMMAND

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DEPARTMENT OF THE ARMY
U. S. ARMY AVIATION MATERIEL LABORATORIES
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This report presents the test and analytical results of the application of non-Newtonian (gelled) fluids as damping media for isolation systems. Both water and oil were modified with chemical additives to alter the rheological properties of these fluids to improve their damping characteristics.

This report is published for the exchange of information and the stimulation of ideas.

Task IF162204A14608
Contract DAAJ02-67-C-0039
USAAVLABS Technical Report 69-20
April 1969

FEASIBILITY STUDY OF NON-NEWTONIAN GELLED FLUIDS AS A SHOCK- AND VIBRATION-ISOLATION MEDIUM

Final Report

By

E. D. Griffith

Prepared by

The Western Company
Research Division
Richardson, Texas

for

U. S. ARMY AVIATION MATERIEL LABORATORIES
FORT EUSTIS, VIRGINIA

[redacted]

SUMMARY

A research and development program has been conducted for the purpose of developing methods of improving the operating characteristics of passive vibration and shock isolators. To achieve such improvement, rheological properties of base fluids in viscous dampers were altered for alternating flow conditions. Newtonian flow properties of both water and MIL-H-5606A hydraulic fluid were changed with chemical additives to yield pseudoplastic flow properties having an initial yield strength at low shear rates and "shear thinning" characteristics at high shear rates. When such modified fluids were used in the fluid dampers of the tested isolation systems, damping resulted. The advantages of this damping in limiting amplication at low-frequency resonance, while providing effective isolation at higher frequencies, were realized and recorded during system tests.

Both analytical and dynamic situations were checked. Laboratory experiments included tests with a special test damper and with a Government-furnished (GFE) damper. These devices were both tested as a part of isolation systems. Significant changes in operating characteristics were achieved.

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LIST OF SYMBOLS

C	Damping Coefficient
C_{EQ}	Equivalent Viscous Damping Coefficient To Be Used as Approximate Representation of Coulomb Damping
D	Hydraulic Conduit Diameter (in.)
F	Input Force
F_d	Damping Force (=K)
F_f	Coulomb Damping, Equal to a Constant Independent of Velocity
f_o	Applied Force (F_o, Maximum Value)
f_t	Transmitted Force (F_t, Maximum Value)
K	Spring Constant
L	Length of Conduit (in.)
M	System Mass
N	One-Half Times the Ratio of Damping Coefficient to Isolated Mass
$\bar{n}$	Ratio of Damping Force to Spring Force
P_K	The Pressure Losses Due to Kinetic Energy Dissipation
P_V	The Viscous Pressure Losses in a Conduit Due to Fluid Flow
r	Radius
R	Hydraulic Conduit Radius (in.)
R_e	Reynolds Number
S	Cross-Sectional Area of Conduit (in.^{2})
T	Transmissibility
T_A	Absolute Transmissibility
T_R	Relative Transmissibility
t	Time

LIST OF SYMBOLS - Continued

u	Displacement of Massless Exciting Support
$\dot{u}$	Velocity of Massless Exciting Support
$\ddot{u}$	Acceleration of Massless Exciting Support
U_o	Maximum Displacement of Massless Exciting Support
x	Displacement (X, Maximum Value)
$\dot{x}$	Velocity ($\dot{X}$, Maximum Value)
$\ddot{x}$	Acceleration ($\ddot{X}$, Maximum Value)
X_o	Maximum Displacement of Isolated Mass
δ	$x-u$
δ_o	X_o-U_o
θ	Displacement Angle
μ	Absolute Viscosity (lb-sec/in.^{2})
υ	Mean Velocity of Fluid Flow (in./sec)
ξ	Fraction of Critical Damping
ρ	Fluid Density (lb-sec^2/in.^{4})
V	$\frac{dX}{dt}$
ω	Forcing Frequency
ω_N	Natural or Fundamental Resonant Frequency of System

INTRODUCTION

Army aircraft, particularly helicopters, commonly experience motor shock and vibration problems. There is a requirement for improvement of the operating characteristics of isolation devices. The capability to decrease amplification of these dynamic environments in passive systems at low-frequency resonance is required, but isolation effectiveness at higher frequencies also must be maintained.

This program involved a study of the feasibility of using non-Newtonian gelled fluids to achieve these improved characteristics in present passive isolation devices using fluid as the damping medium.

DISCUSSION

LABORATORY SETUP

Several tasks were required to prepare the laboratory apparatus setup and perform the initial damper experiments. These tasks included

Design and fabrication of test fixtures.

Design and fabrication of test dampers.

Preliminary experiments with fabricated and purchased (GFE) dampers.

Test Fixtures

A massive test fixture was required to provide a rigid base for experiments throughout the 5- to 100-Hz frequency range. This fixture was constructed from reinforced concrete, isolated from the laboratory workbench by elastic pads. Figure 1 provides details of this fixture.

Test Dampers

Funding limitations precluded a formal documented analysis and experimental program to produce a test damper for use on this program. Appendix I presents the design criteria generated by analysis and illustrates one unsuccessful design that was tested and found to be inadequate. Other damper designs were tested and abandoned.

The fabricated test damper shown in Figure 2 was tested and empirically found to be suitable for conducting the tests required by contract using the "high viscosity" fluids. No suitable design was found for testing the drag reducing fluids.

The original GFE damper was tested and found to be completely unsuitable for experiments required on this program. A second damper (Figure 3) supplied by the Government was used.

Late in the program the contract was modified to specify only tests of "high viscosity" fluids in the two dampers shown in these photographs.

Experimental Procedures

Two types of laboratory experiments were conducted. The first type utilized the operating setup shown in Figure 4. The GFE damper in Figure 4 was originally planned for use in the tests, but it proved to be too "stiff" for the associated vibrator. The smaller unit shown in Figure 3 was ultimately chosen.

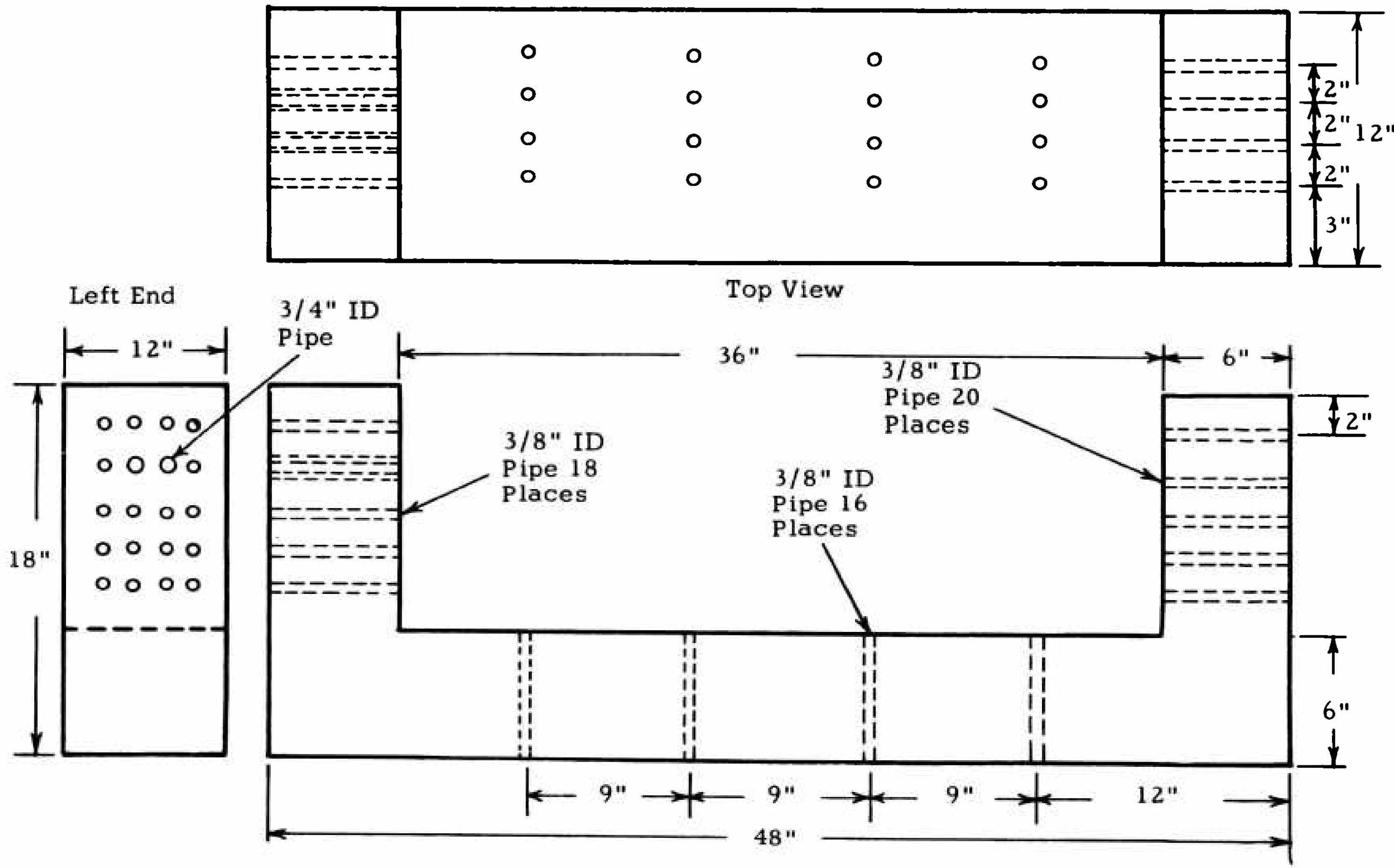
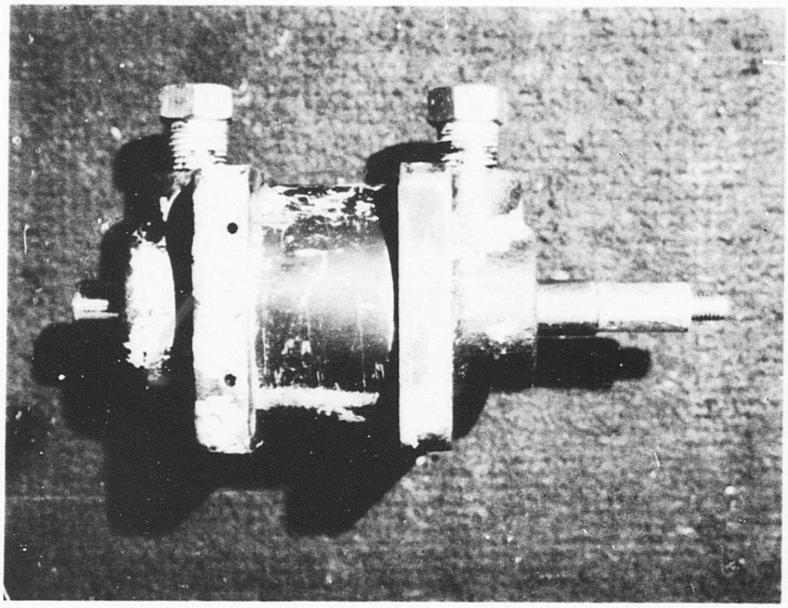


Figure 1. Test Fixture (Not to Scale).



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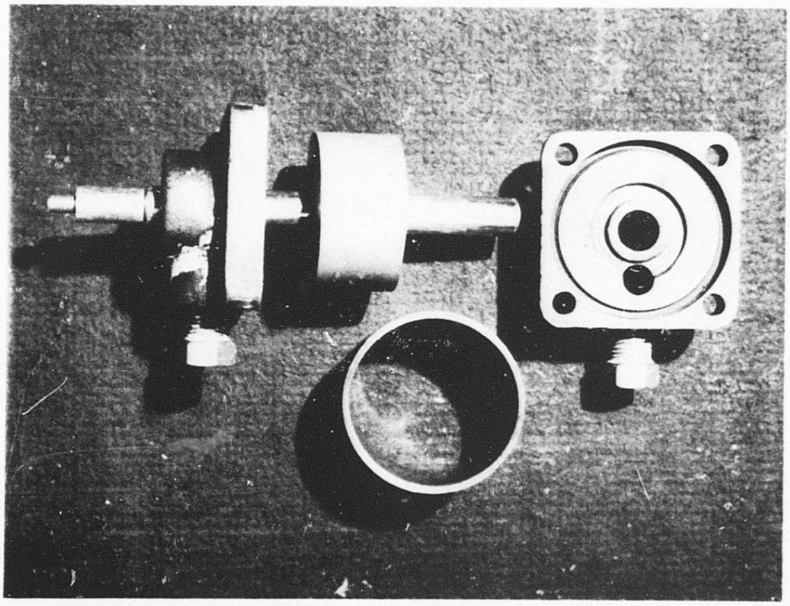
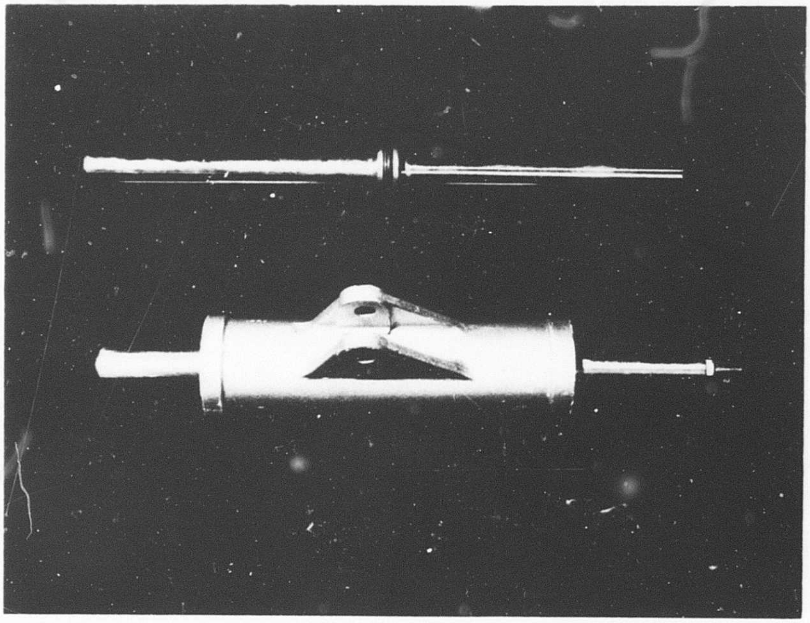


Disassembled

Figure 2. Fabricated Test Damper.



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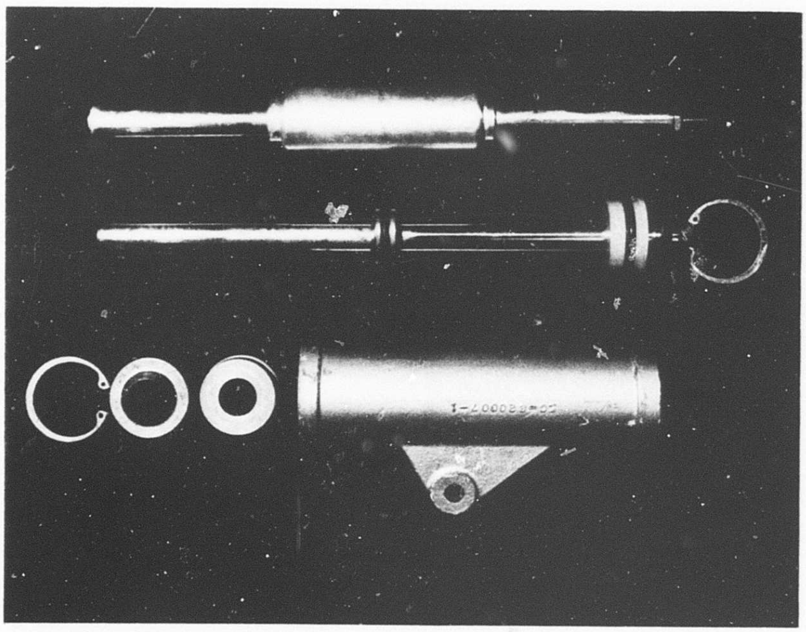


Disassembled

Figure 3. Government-Furnished (GFE) Damper.

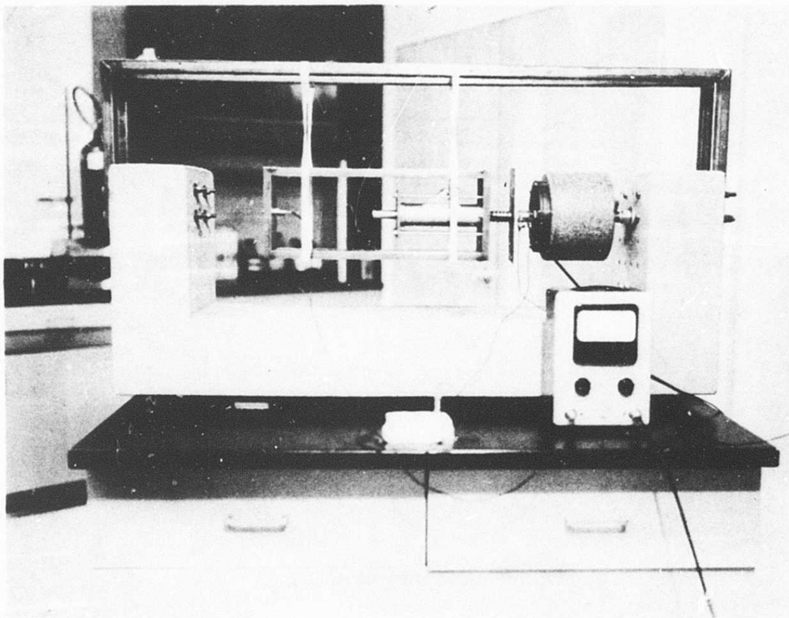


Figure 4. Vibrator and Damper in Rigid Fixture.

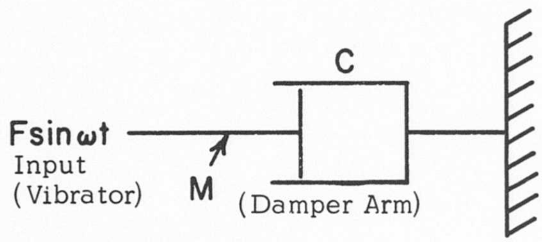


Figure 5. Schematic Representation of the Experimental System.

In the first mode, both the vibrator and damper were rigidly positioned on the concrete fixture as illustrated in Figure 5. This setup was used to measure the damping function produced by the actual GFE or test damper. The equation

$$\frac{F}{\dot{x}} = \sqrt{M\omega^2 + C^2} \tag{1}$$

(discussed in Appendix II) was used to evaluate system viscous damping. In the test procedure, magnitudes of input force (F), piston velocity ($\dot{x}$), and frequency were measured.

In the second operating mode, the damper was suspended as shown in Figure 6.

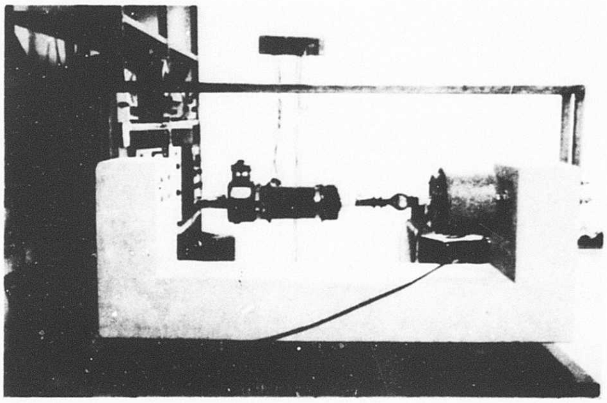


Figure 6. Suspended Damper.

This was the operating mode required by the statement of work. In later tests, 20- and 40-pound masses were attached to the end of the damping device, as illustrated in Figure 7.

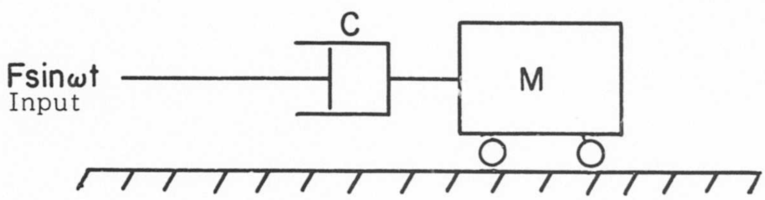


Figure 7. Isolated Mass System.

Figure 8 shows the fixture holding the damper unit and vibrator as well as the control and data monitoring electronics.

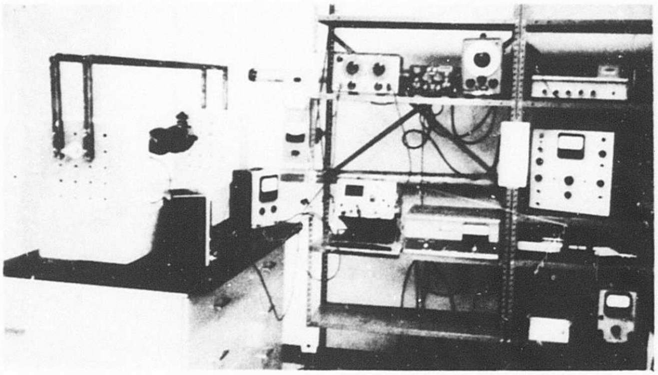


Figure 8. Electronic Control and Monitor Instrumentation.

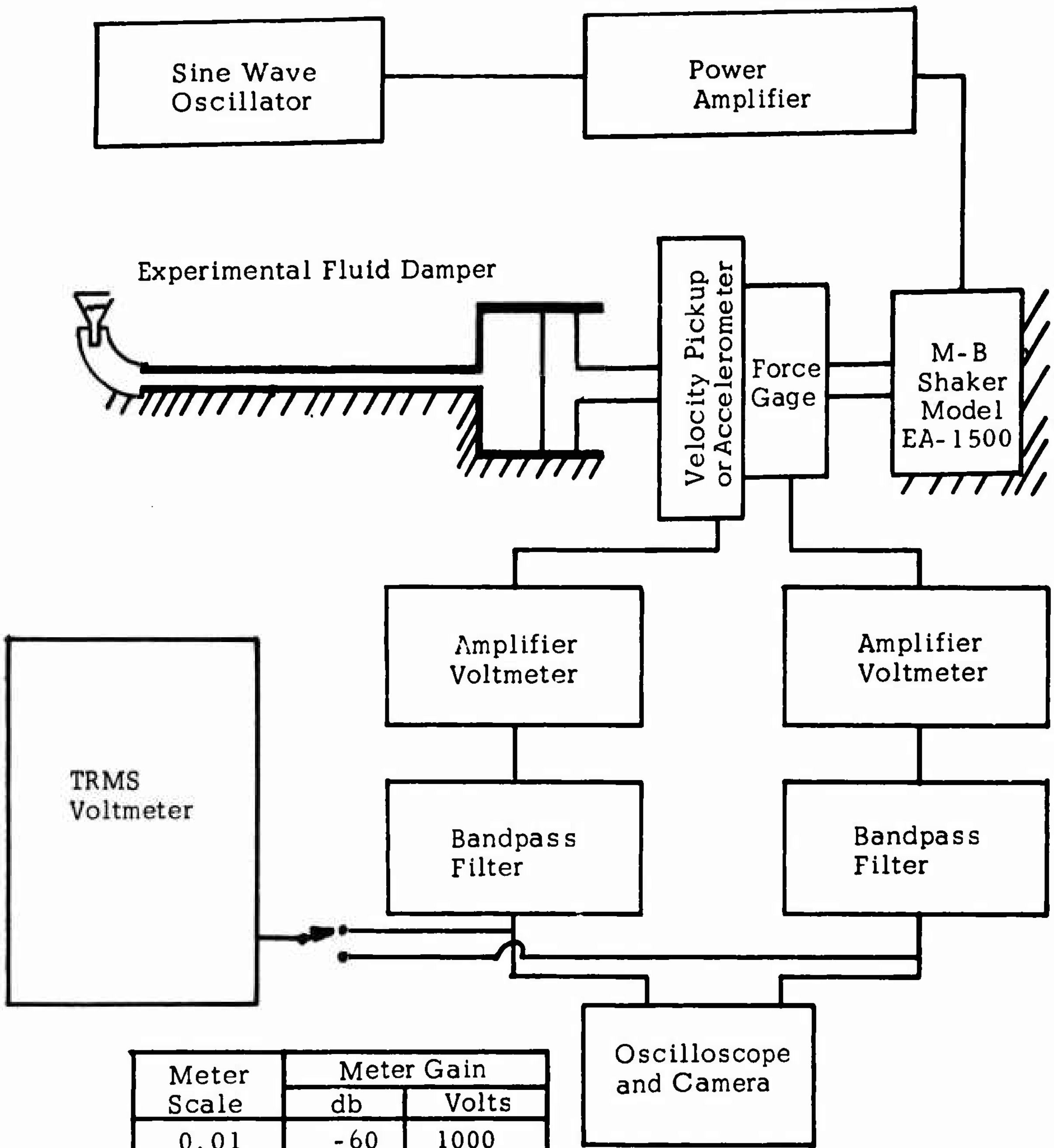
INITIAL EXPERIMENTS

Type One Procedure

Initially, experiments were conducted with the rigidly mounted vibrator and damper (Figure 5) and a test damper first filled with water and then with a high-viscosity fluid. Figure 9 illustrates the test configuration, and typical results of these tests are shown in Figure 10.

Notice that the curve for water (damping force versus flow velocity) indicates viscous damping, since the force is a linear function of velocity; however, the high-viscosity fluid produced the completely different function predicted by theory. At velocities approaching zero, the damper has a resistance to flow equivalent to a nearly solid material. Above a given yield strength, however, the damper will move and, as force is increased, greater velocities are achieved. This has been described as a "shear thinning" rheological property. Such phenomena are well known in unidirectional flow devices, but it is believed that this is the first demonstration of these effects in fluid dampers undergoing alternating excitation at high forcing frequencies.

Many experiments were conducted with this same setup, utilizing higher force levels to produce velocities great enough to produce turbulent flow; however, no recognizable turbulent flow could be established in these tests. As a result of the tests and the noted lack of turbulent flow, this part of the program was abandoned.



Meter Scale	Meter Gain	
	db	Volts
0.01	-60	1000
0.03	-50	316
0.10	-40	100
0.30	-30	31.6
1.00	-20	10
3.00	-10	3.2
10.00	0	0.0

Force Gage:
2.06 lbs/volt
Accelerometer:
0.0506 volt/g
Velocity Pickup:
0.0936 volt in./sec

Figure 9. Setup for Water Versus High-Viscosity Fluid Test.

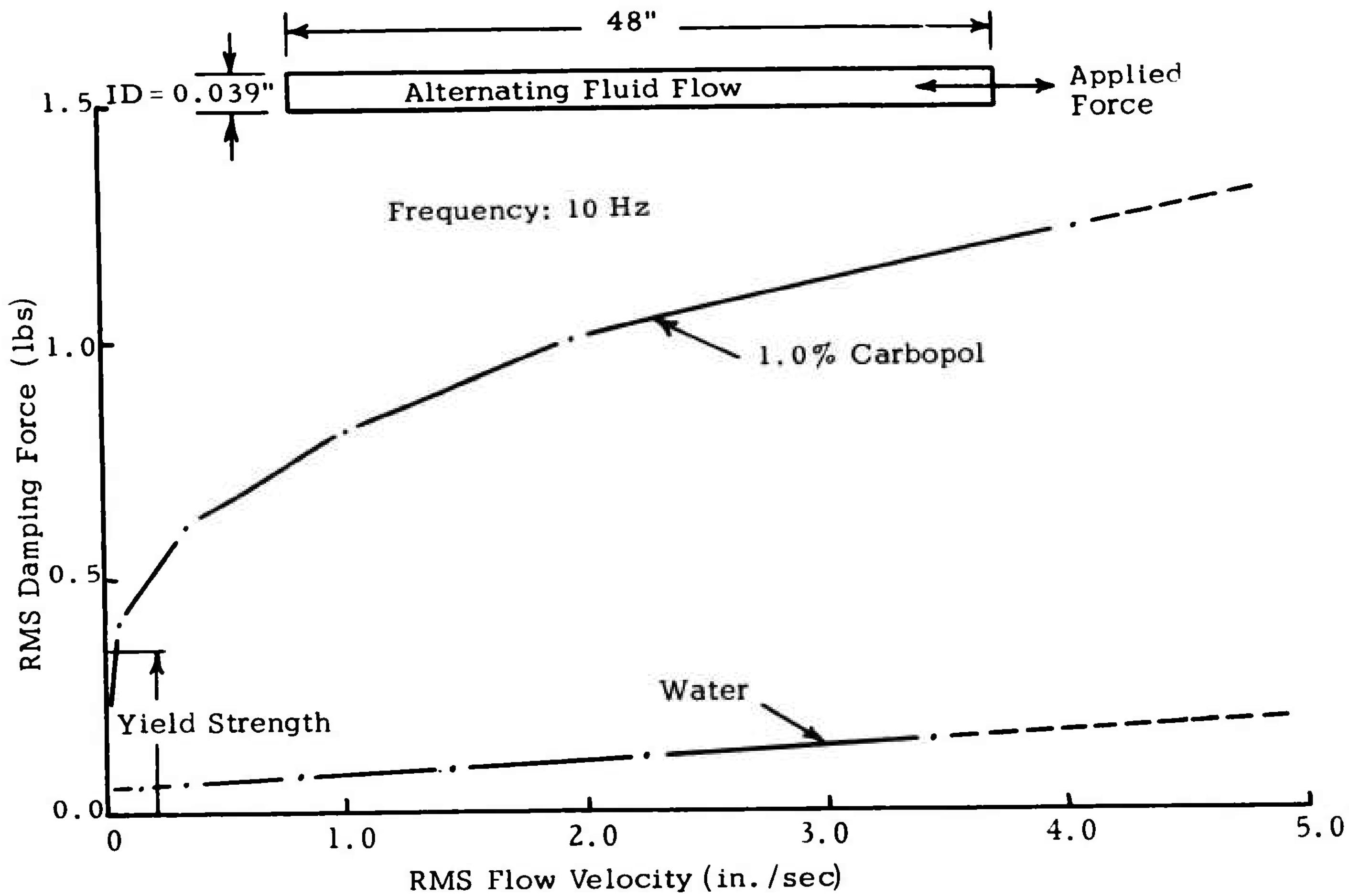


Figure 10. Experimental Results Obtained With Fabricated Damper.

Transmissibility of Viscous Damped System[1]

x = displacement

$\dot{x}$ = velocity

$\ddot{x}$ = acceleration

f_o = applied force

f_t = transmitted force

$X, \dot{X}, \ddot{X}, F_o, F_t$ = maximum values of these parameters

ω = forcing frequency

ω_N = natural system frequency

ξ = fraction of critical damping

ω/ω_N = normalized frequency

Transmissibility(T) = F_t/F_o = Transmitted Force/Applied Force

$$T = \sqrt{\frac{1 + (2\xi\,\omega/\omega_N)^2}{(1 + \omega^2/\omega_N^2)^2 + (2\xi\,\omega/\omega_N)^2}} \qquad (2)$$

When transmissibility(T) is plotted against normalized frequency (ω/ω_N) for a system utilizing spring viscous-damped isolators, the curves in Figure 11 result.

Type Two Procedure

Laboratory experiments were also performed with isolated masses mounted on spring viscous-damped isolated systems. These tests are those specifically required by contract.

Initially, data were taken with either water or oil as the base fluid in the damping device. This produced base-line data of a conventional spring-mass system with viscous damping. Then tests were performed with certain chemical additives that altered the rheological characteristics of the base fluids in the damping devices. After redirection of the program to eliminate tests of drag-reducing fluids, all tests were concerned with the so-called high viscosity fluids, which have an initial yield strength at velocities near zero and a decreasing apparent viscosity as the velocity of fluid flow increases.

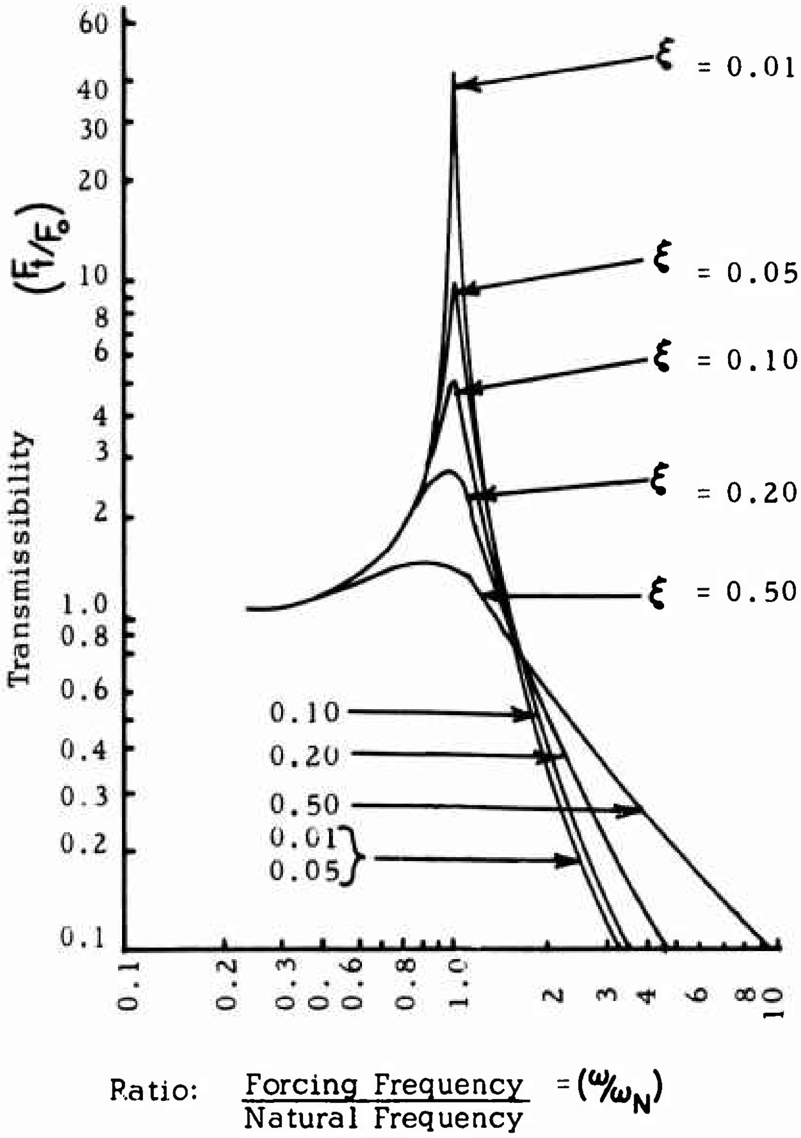


Figure 11. Spring Viscous Damped Isolators.

In these isolated mass systems with viscous damping, the mass was free to move horizontally with negligible sliding friction. The isolator was constructed from a spring in conjunction with either the fabricated test damper or the GFE damper. An electrodynamic vibrator was arranged to drive the mass through the isolator.

Experiments were performed with sinusoidal excitation and with transient shock excitations. The frequency range covered was from 5- to 100-Hz at 1 g. Under shock excitation, data were filtered to exclude data outside this range. Level of excitation was 1.0 g, except during continuous sinusoidal excitation response tests,when additional experiments were conducted at high levels in order to illustrate the improvement accrued by using the base fluid additives. In illustrations of test results, these different test levels of 1.0 and 10.0 g's are referred to as responses A and B, respectively.

Theory

The theoretical situation is illustrated by the schematic diagram in Figure 12. A discussion of the response of such a system follows.

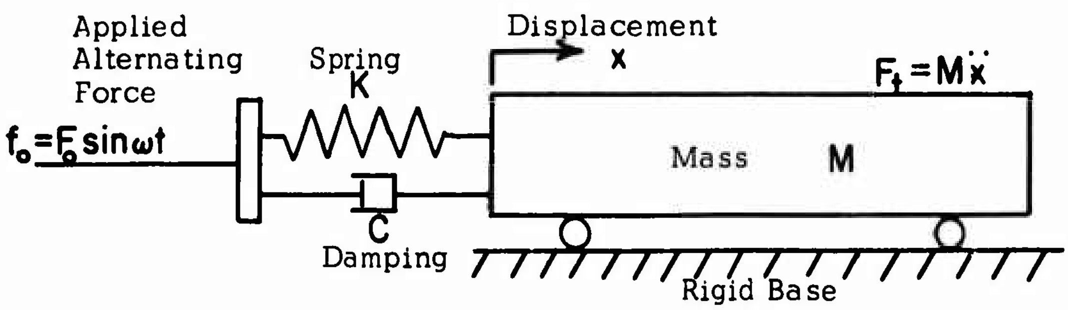


Figure 12. Schematic Diagram of an Isolated Mass With Viscous Damping.

The variable parameter in Figure 11 is the fraction of critical damping (ξ). Percent of critical viscous damping is this value multiplied by 100.

Vibration isolation occurs above the frequency region near resonance given by

$$\text{Isolation Frequency} \qquad > 1.41\ (\omega/\omega_N) \qquad (3)$$

Below this frequency, amplification occurs near and at the system resonance. A good isolator provides adequate isolation at the higher frequencies and minimizes low-frequency amplification near resonance.

SPECIFIC TESTS WITH FABRICATED TEST DAMPER

A series of experiments using the fabricated test damper was conducted with high-viscosity fluids. These tests were conducted with procedure type two (isolated masses), using the setup shown in Figure 13. Masses of 0, 20, and 40 pounds were used. Tests were conducted with both water and MIL-5606 hydraulic oil as the base fluid. Carbopol (B.F. Goodrich Chemical Company) and J-2 (Stein Hall Chemical Company) additives were tested in water, and Cabosil (Cabot Corporation) and G-8 (The Western Company) additives were tested in oil. Three concentrations were tested for each additive. Table I shows base fluid, mass chemical additive, and concentration for these experiments.

SPECIFIC TESTS WITH GFE DAMPER

A second series of experiments was conducted with the GFE damper, using the same procedures used with the fabricated test damper. It was necessary to modify the GFE damper for use in this program. Therefore, base-line tests were conducted on the original damper and two modified configurations. Table II indicates configuration, base fluid, additive, and concentration. All tests were performed with a 40-pound mass.

SIMULATED LANDING VELOCITY SHOCK TESTS

Obtained from the GFE test series as a integral part of the test program were simulated landing velocity shock data. Specific excitations of 5 ft/sec and 10 ft/sec were utilized. Tests and data obtained were coincidental with tests described above.

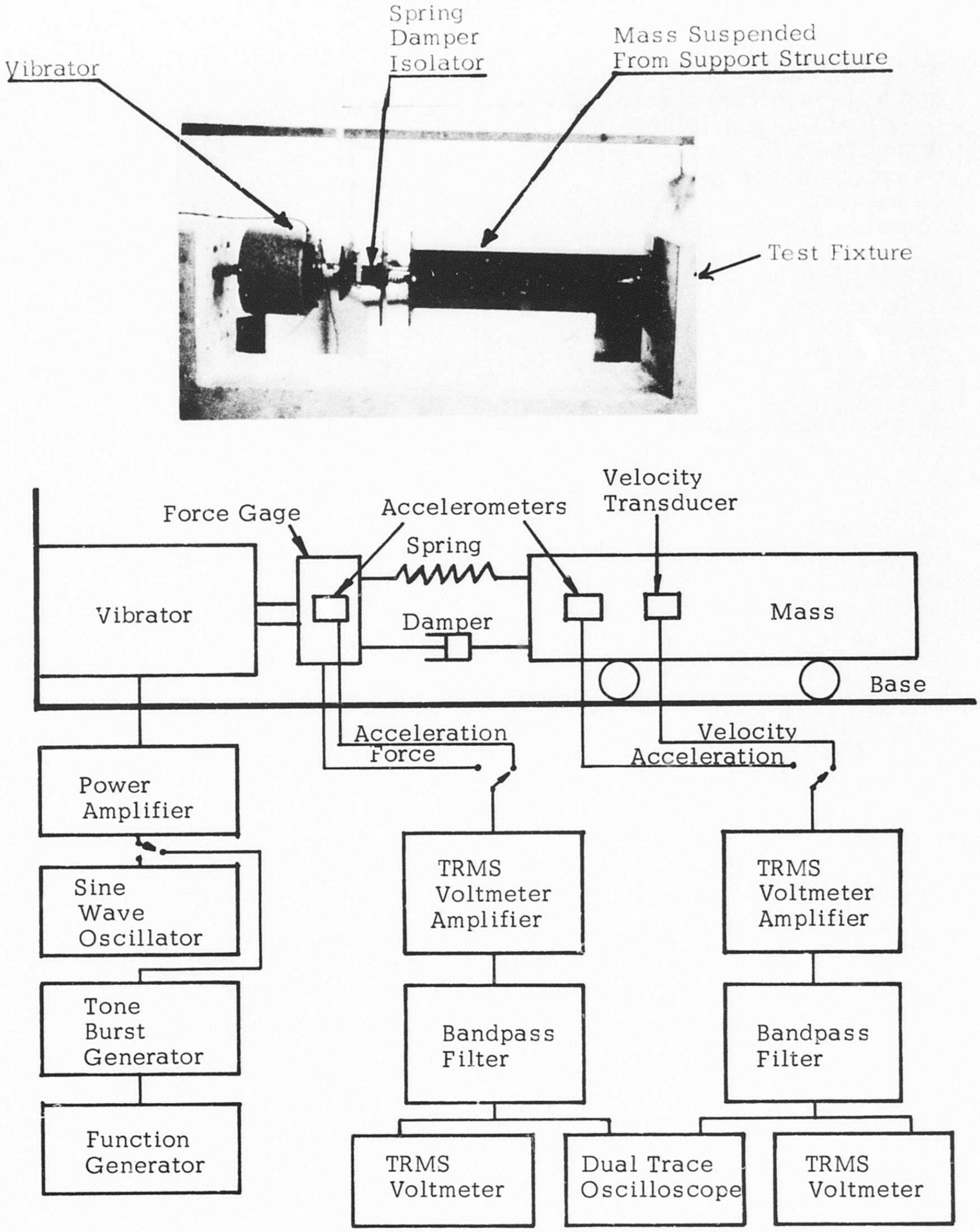


Figure 13. Diagram of Isolated Mass Experiment.

TABLE I. CONFIGURATIONS TESTED WITH FABRICATED TEST DAMPER			
Base Fluid	Additive	Concentration (%)	Mass (lb)
Water	J-2	0.5	0, 20, 40
Water	J-2	2	0, 20, 40
Water	J-2	3	0, 20, 40
Water	Carbopol	0.5	0, 20, 40
Water	Carbopol	1	0, 20, 40
Water	Carbopol	0.75	0, 20
Water	Carbopol	2	40
Water	-	-	0, 20, 40
MIL-5606 Oil	-	-	0, 20, 40
MIL-5606 Oil	G-8	6	0, 20, 40
MIL-5606 Oil	G-8	10	0, 20, 40
MIL-5606 Oil	G-8	20	0, 20, 40
MIL-5606 Oil	Cabosil	0.5	0, 20, 40
MIL-5606 Oil	Cabosil	1	0, 20, 40
MIL-5606 Oil	Cabosil	3	0, 20, 40

NOTE: Both sinusoidal and shock pulse experiments were conducted.

TABLE II. CONFIGURATIONS TESTED WITH GFE DAMPER				
Configuration	Base Fluid	Chemical Additive	Concentration (%)	Mass (lb)
Original	Water	-	-	40
	MIL-5606 Oil	-	-	40
1.443" Dia. Piston	Water	-	-	40
	Water	Carbopol	0.5	40
	Water	Carbopol	1.0	40
	Water	J-2	0.5	40
	Water	J-2	2.0	40
	Oil	-	-	40
	Oil	G-8	6.0	40
	Oil	G-8	10.0	40
	Oil	Cabosil	0.5	40
	Oil	Cabosil	1.0	40
1.440" Dia. Piston	Water	-	-	40
	Water	Carbopol	0.5	40
	Water	Carbopol	1.0	40
	Water	J-2	0.5	40
	Water	J-2	2.0	40
	Oil	-	-	40
	Oil	G-8	6.0	40
	Oil	G-8	10.0	40
	Oil	Cabosil	0.5	40
	Oil	Cabosil	1.0	40

EVALUATION OF RESULTS

Typical data from experiments with the fabricated test damper and the GFE damper have been selected for evaluation, as discussed in the following paragraphs.

FABRICATED TEST DAMPER

Figure 14 shows the results of sinusoidal tests in terms of transmissibility plotted against normalized frequency, using a 40-pound mass and a water base fluid with 0.5-percent Carbopol as an additive. Excitation frequencies in the 5- to 100-Hz region were applied, at a g level of 1. At resonance and throughout the range of high frequencies in the isolation region, the theoretical plot and the actual data (taken with water in the damper) coincide. Below resonance, these curves differ slightly (probably because of resonance effects in the holding fixture).

Significant results of these data are the 65-percent reduction in low-frequency amplification at resonance and the maintenance of isolation efficiency at higher frequencies.

Figure 15 shows the results of testing the same 40-pound mass with 1-percent Carbopol additive in the water. In this case, the amplification at resonance was approximately equal while isolation was increased.

Finally, Figure 16 shows the results of testing a 20-pound mass with a 0.75-percent Carbopol additive in the water. Here the low-frequency amplification has been decreased by 23 percent and the isolation has been increased by 22 percent.

Typical input and output wave forms from pulse or shock tests are shown in Figures 17 and 18. These data were taken with the test damper. During performance of the specific tests, both input and output accelerations were monitored and a transmissibility ratio was computed. Typical results are shown in Table III. The excitation level was 1.0 g. Insufficient background data is available for interpertation of this type of experimental information.

GFE DAMPER

Figure 19 shows the results of sinusoidal tests in terms of transmissibility versus normalized frequency using the GFE damper and the 40-pound mass. The damper was filled with MIL-5606 hydraulic oil. These data indicate that the system had a large amount of damping, over 0.5 of critical. Figure 20 shows the transmissibility function for the modified damper. In this case, the diameter of the damper piston has been reduced to 1.443 inches. Since there is greater distance between the moving elements of the damper, the system shows less damping (about 0.2 of critical).

In Figure 21, a 6-percent G-8 additive has been added to the base fluid. Because the resulting fluid is thicker and has greater apparent viscosity, the damping in the system, for low frequencies, has been increased to about 0.5 of critical. Low-frequency amplification is minimized; however, isolation is also achieved at higher frequencies because of the "shear thinning" effects in the high-viscosity fluid that provide less damping at high excitation levels. Improved isolator effectiveness, therefore, has been achieved for large vibration or shock excitations. (Refer to curves A and B in Figure 21).

Figure 22 shows base-line data with oil as the base fluid for the damper when the piston diameter was decreased to 1.400 inches. Again, the damping has decreased to about 0.10 of critical, because of the greater distance between piston and cylinder.

Addition of 6-percent G-8 had marginal effects in this case, as shown in Figure 23. This amount of additive was insufficient to produce increased damping at the resonant condition.

Table IV shows typical data obtained with the GFE damper subjected to shock-wave excitation as required by contract. These data cannot be evaluated in terms of information available at this time.

SIMULATED SHOCK LANDING TESTS

Simulated landing velocity shock data did not lend itself to interpretation beyond the fact that apparently those configurations yielding a sine wave or square wave pulse of less than 1.0 (Table V) provided improved damping.

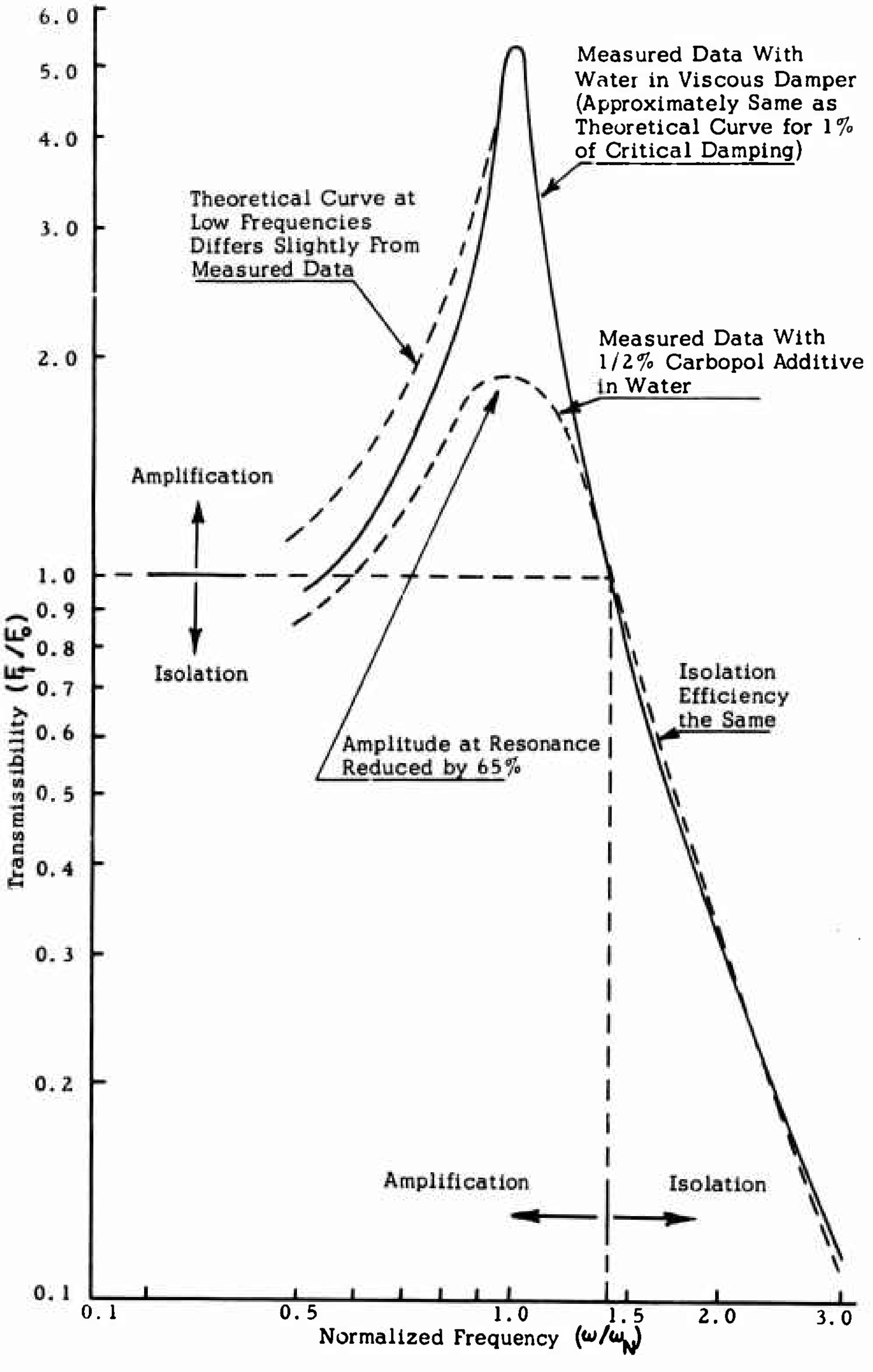


Figure 14. Isolator System With 40-Pound Mass, Using the Fabricated Test Damper and 0.5 Percent Carbopol in Water.

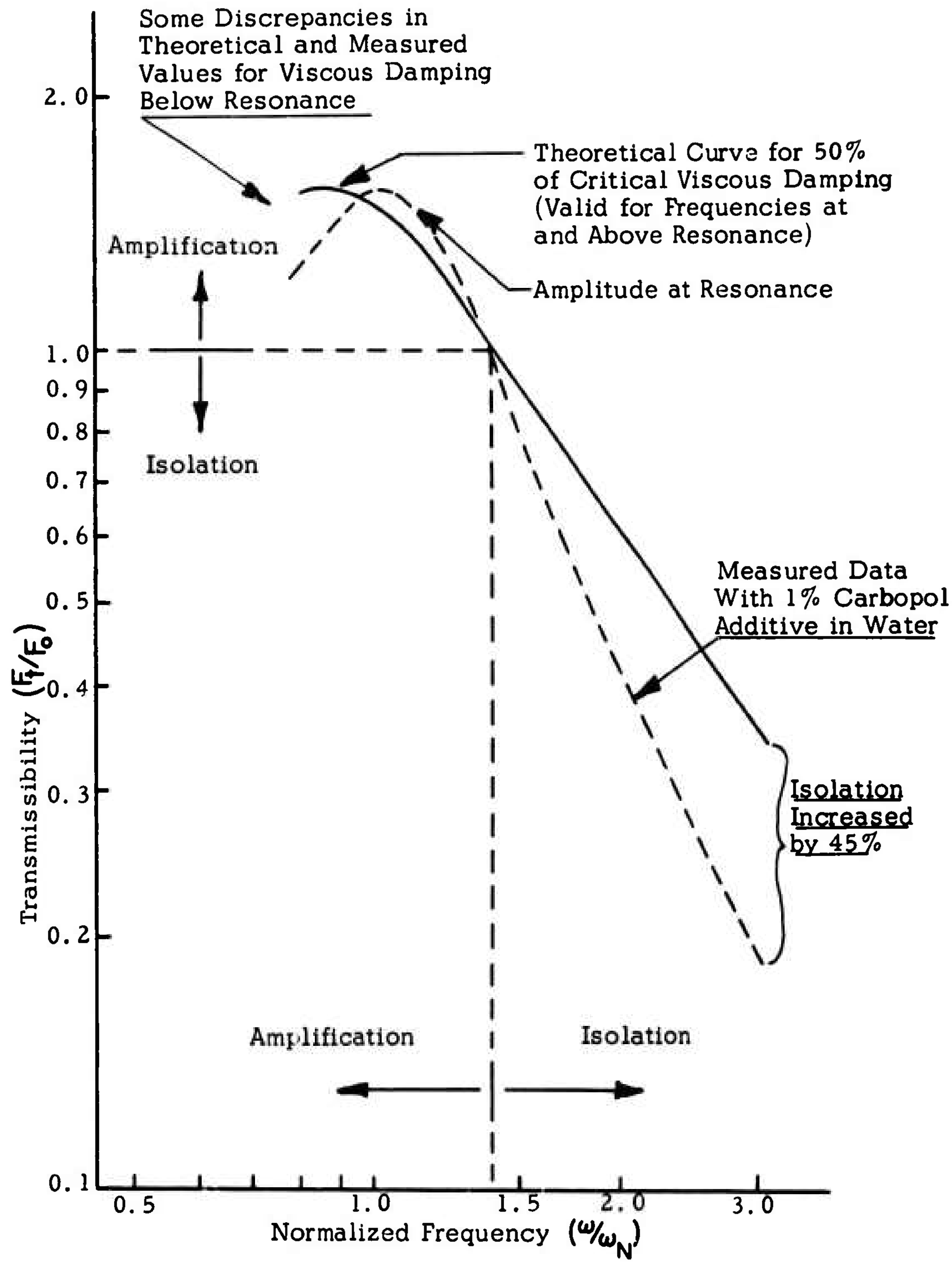


Figure 15. Isolator System With 40-Pound Mass, Using the Fabricated Test Damper and 1.0 Percent Carbopol Additive in Water.

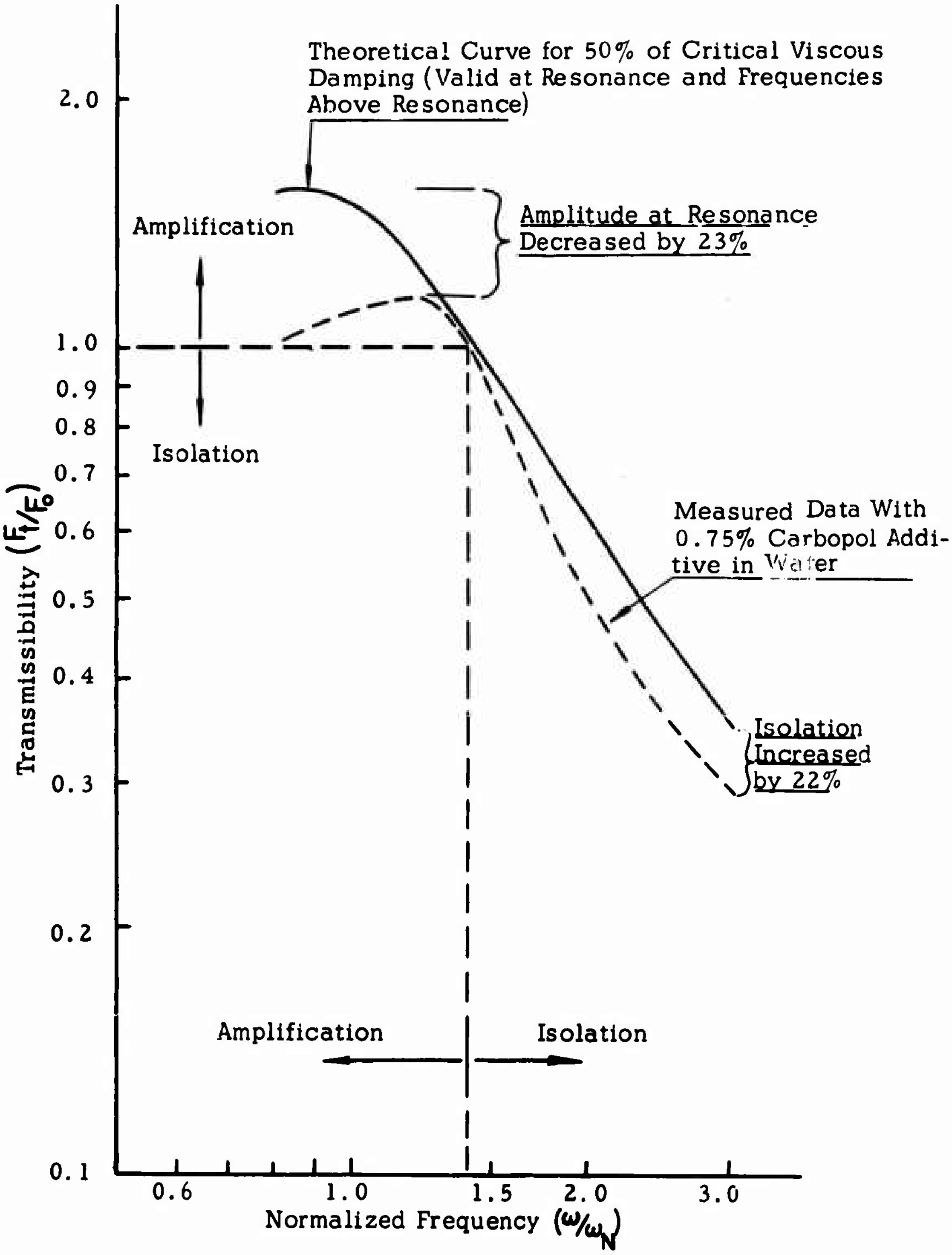
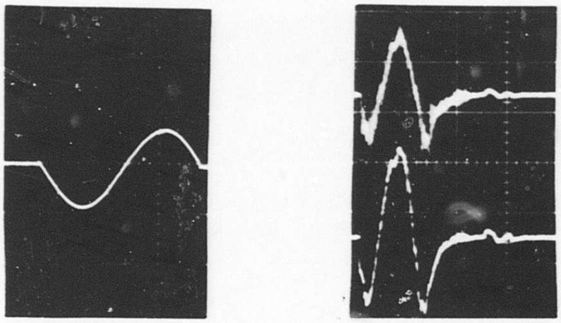


Figure 16. Isolator System With 20-Pound Mass, Using the Fabricated Test Damper and 0.75 Percent Carbopol in Water.

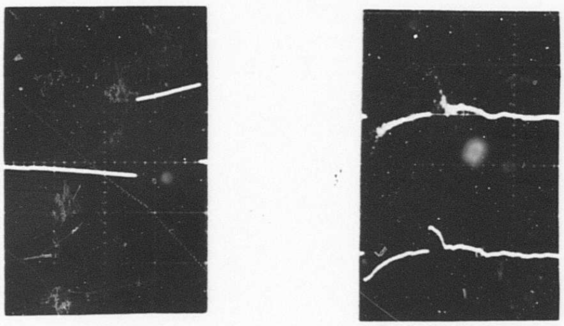


Vibrator Excitation

Isolator Response

Voltage Signal

One-Cycle Sine Wave Pulse Experiment

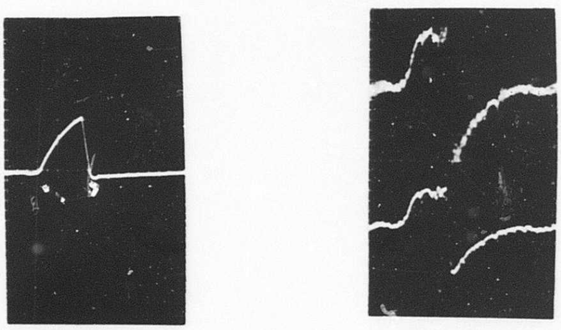


Vibrator Excitation

Isolator Response

Voltage Signal

Square Wave Pulse Experiment



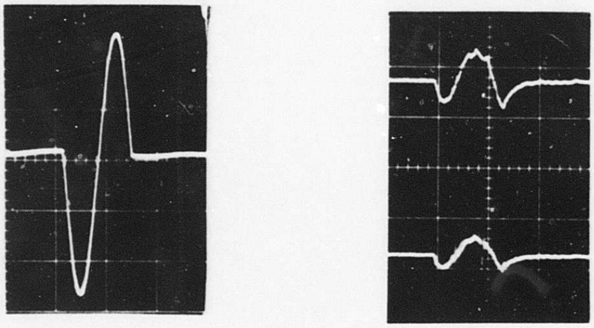
Vibrator Excitation

Isolator Response

Voltage Signal

Triangular Wave Pulse Experiment

Figure 17. Examples of Pulse Excitation Wave Forms, Using the Fabricated Test Damper, 0.5 Percent Carbopol in Water, and a 40-Pound Mass.

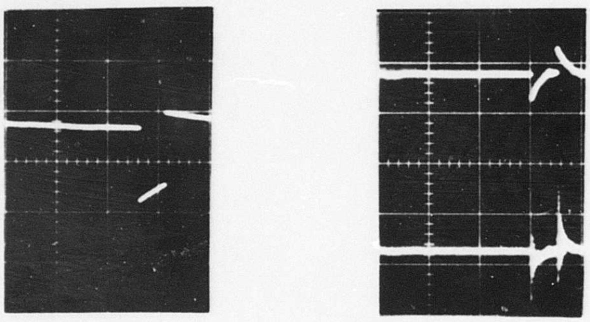


Vibrator Excitation

Isolator Response

Voltage Signal

One-Cycle Sine Wave Pulse Experiment

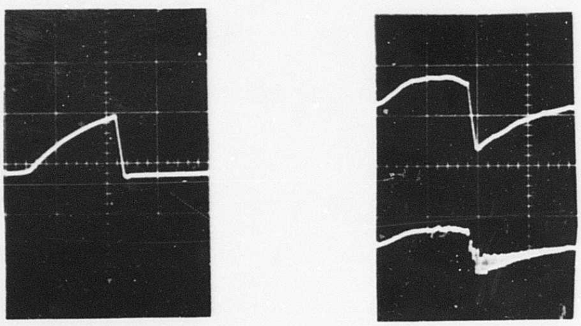


Vibrator Excitation

Isolator Response

Voltage Signal

Square Wave Pulse Experiment



Vibrator Excitation

Isolator Response

Voltage Signal

Triangular Wave Pulse Experiment

Figure 18. Examples of Pulse Excitation Wave Forms, Using the Fabricated Test Damper, 1 Percent Carbopol in Water, and a 40-Pound Mass.

TABLE III. TRANSMISSIBILITY FOR THE FABRICATED TEST DAMPER			
Experimental Setup With Test Damper, 40-Pound Mass, and Carbopol-Water Mixture	Pulse Shape		
	Sine	Square	Sawtooth
0.5 percent Carbopol	0.81	0.91	0.77
1 percent Carbopol	1.36	1.37	1.55

TABLE IV. TRANSMISSIBILITY FOR THE GFE DAMPER			
Experimental Setup With GFE Damper and 40-Pound Mass	Pulse Shape		
	Sine	Square	Sawtooth
Original configuration (oil only)	44.50	19.80	2.40
1.443" Piston (oil only)	1.71	2.80	2.50
1.443" Piston (oil, 6 percent G-8)	0.34	0.56	0.23
1.440" Piston (oil only)	4.60	5.12	0.75
1.440" Piston (oil, 6 percent G-8)	4.10	4.30	2.56

TABLE V. SIMULATED SHOCK LANDING RESULTS			
	Pulse Shape		
	Sine	Square	Sawtooth
Experimental Setup With Test Damper, 40-Pound Mass, and 0.5 Percent Carbopol-Water Mixture	0.81	0.91	0.77
Experimental Setup With GFE Damper, 40-Pound Mass, and 1.443" Piston (oil with 6 percent G-8)	0.34	0.56	0.23

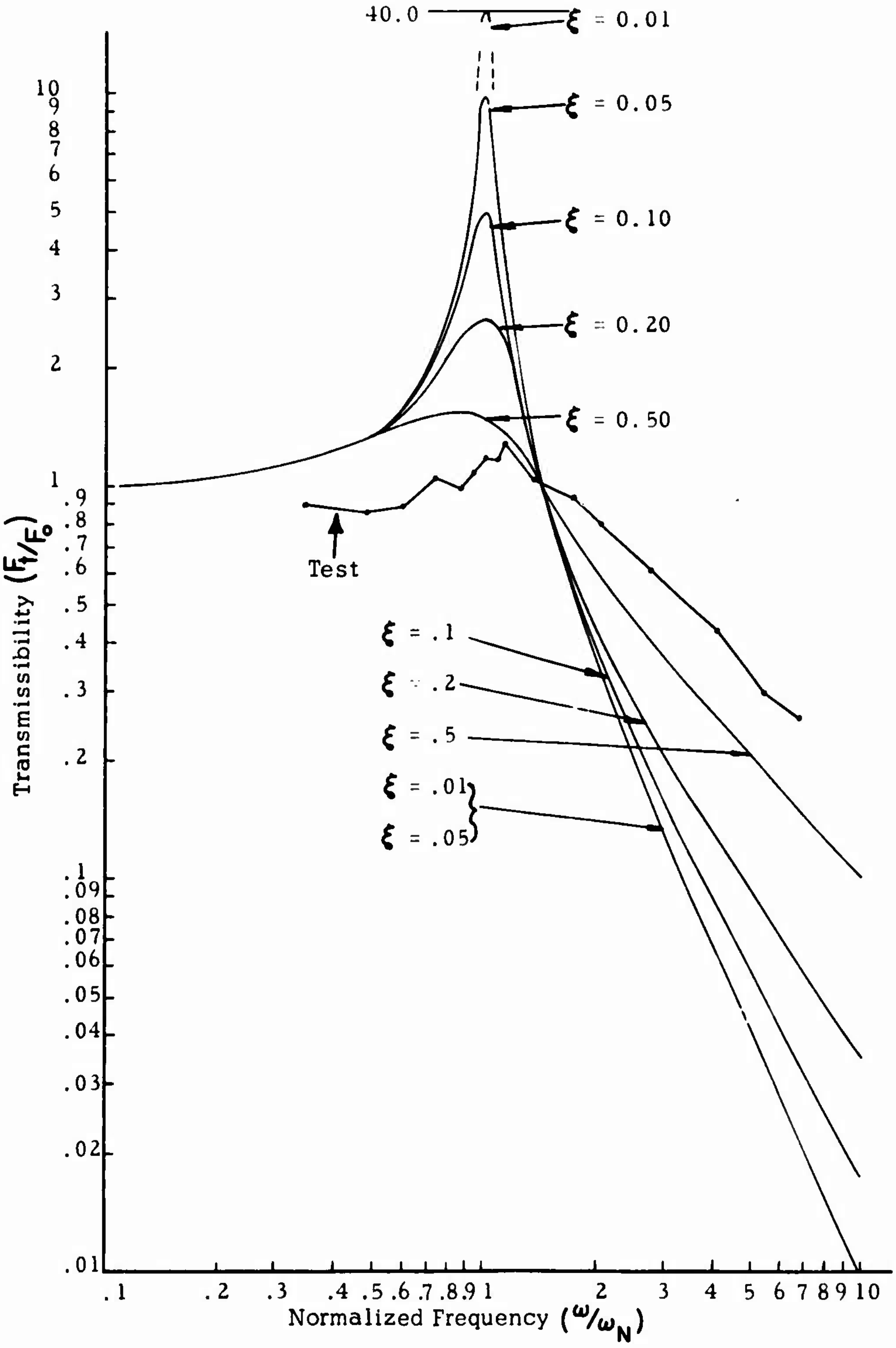


Figure 19. Sinusoidal Response of Isolated 40-Pound Mass, Using the Original GFE Damper and MIL-5606 Base Fluid.

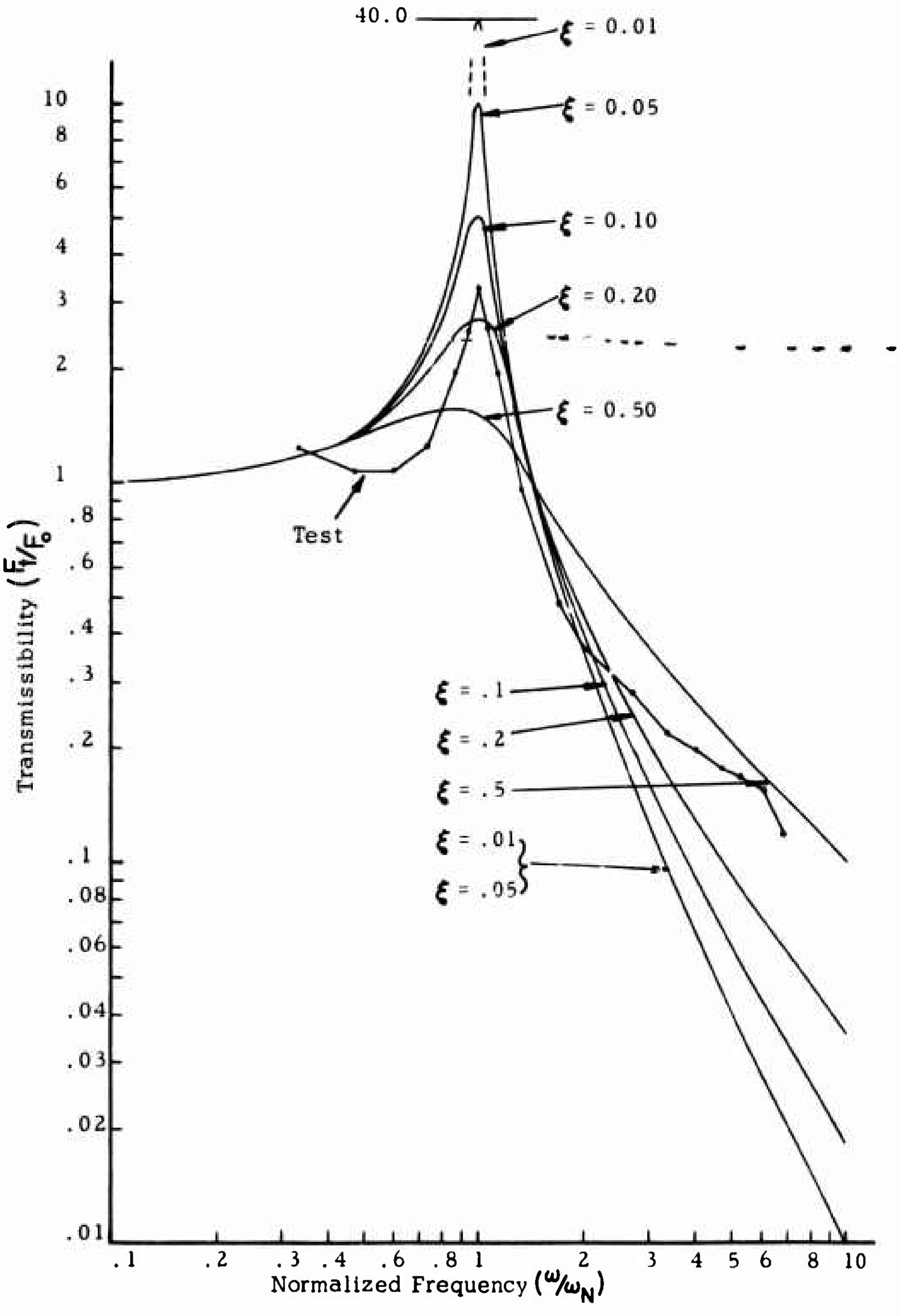


Figure 20. Sinusoidal Response of Isolated 40-Pound Mass, Using the Modified (1.443-Inch Piston) GFE Damper and MIL-5606 Base Fluid.

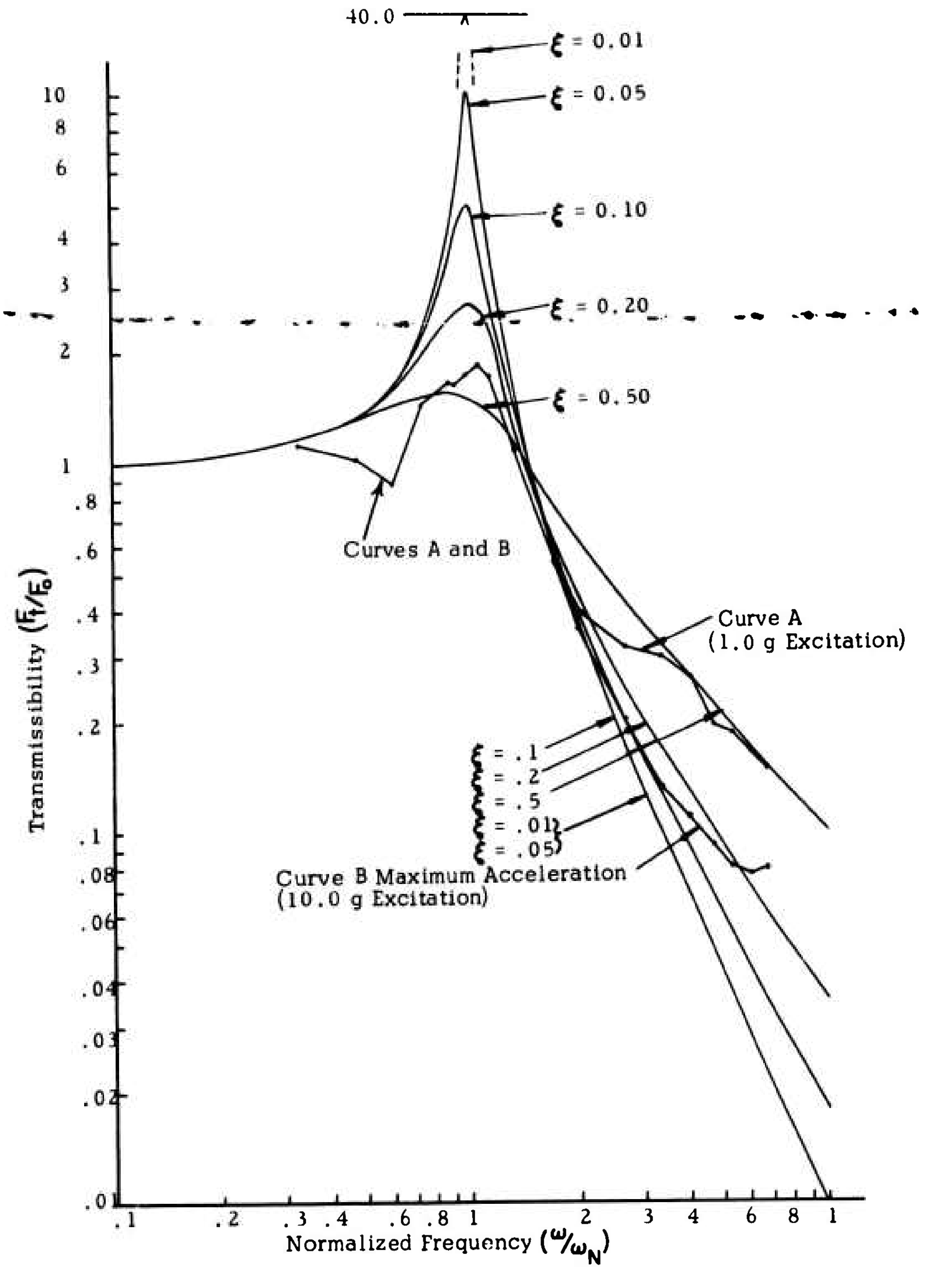


Figure 21. Sinusoidal Response of Isolated 40-Pound Mass, Using the Modified (1.443-Inch Piston) GFE Damper and MIL-5606 Base Fluid With 6 Percent G-8 Additive.

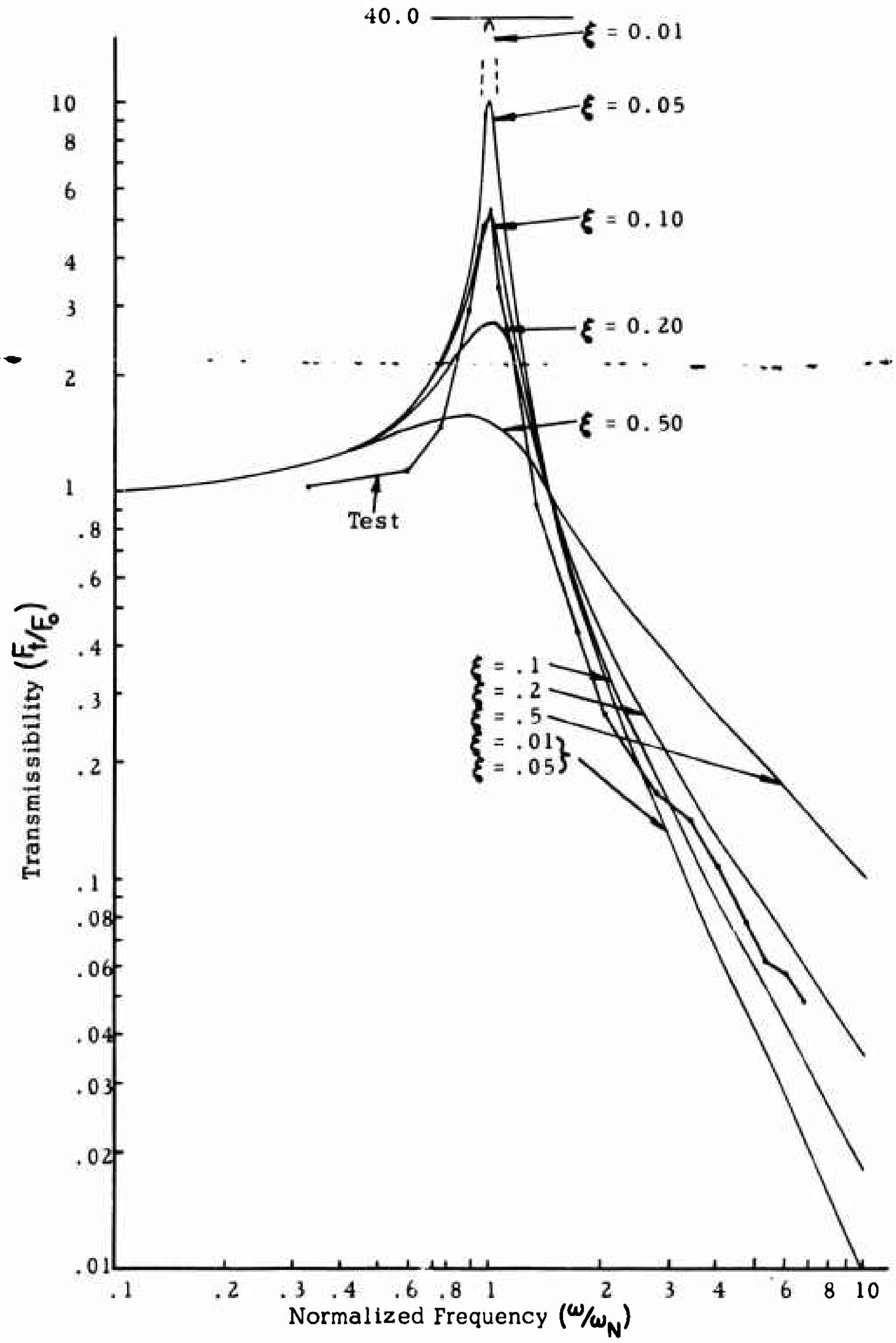


Figure 22. Sinusoidal Response of Isolated 40-Pound Mass, Using the Modified (1.400-Inch Piston) GFE Damper and MIL-5606 Base Fluid.

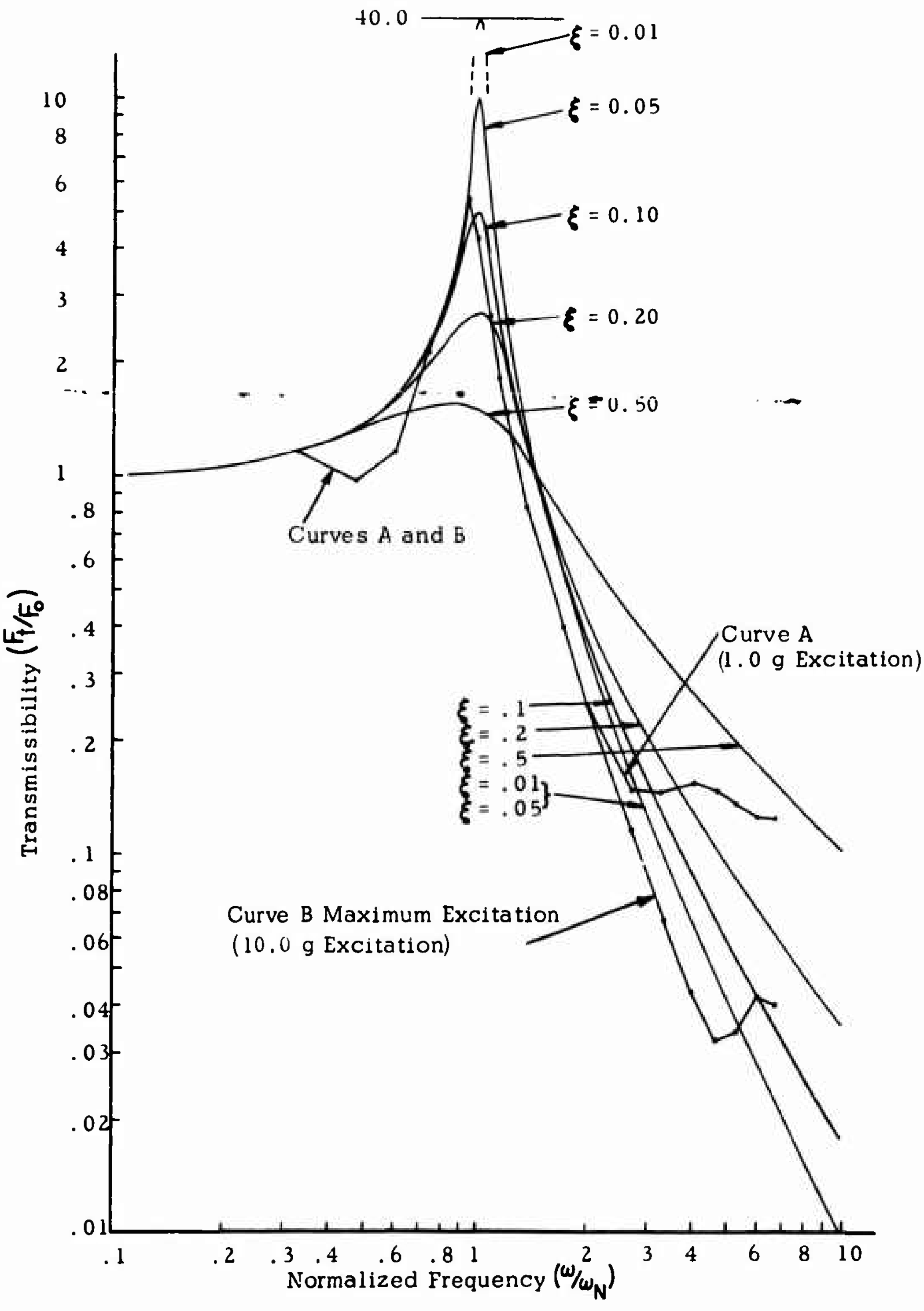


Figure 23. Sinusoidal Response of Isolated 40-Pound Mass, Using the Modified (1.400-Inch Piston) GFE Damper and MIL-5606 Base Fluid With 6 Percent G-8 Additive.

CONCLUSIONS

These experiments have demonstrated the feasibility of improving the operation of passive fluid dampers by introducing chemical additives into the base fluid. Several significant benefits can be realized with the high-viscosity fluids when the damper is used as part of an isolation system:

> The capability of decreasing the low-frequency amplification of the isolator near resonance can be very important. Most packages utilizing vibration-isolation devices have space limitations. Damage often results when low-frequency displacement limitations are exceeded. A large decrease in transmissibility in this region can be a major aid for design of such shock- and vibration-isolation systems.
>
> The capability of increasing vibration effectiveness can be of great value in reducing vibration and shock loads on helicopters and other aircraft. Such reduction of dynamic environments can increase structure fatigue life, prevent instrument malfunction, and reduce annoyance to flight personnel.
>
> The capability of adjusting the transmissibility function and improving, to some extent, both high- and low-frequency performance can be a useful design tool in a wide variety of situations.

Figure 24 is a comparison of results shown in Figure 14 and in the Standard Military Specification for vibration isolators. The reduction of low-frequency amplification can be of real help in meeting and exceeding this specification. By using the methods developed under this program, future specifications may be written to reflect these advantages in the state of the art.

Although the data from experiments on this program are impressive and show potentially useful methods for application to vibration and shock problems, there is no evidence that maximum limits have been reached. Future work may achieve greater benefits from modified systems.

Experiments with drag-reducing additives in water (J-2) and oil (G-8) were disappointing. (The data indicate that turbulence in alternating fluid flow may occur at much greater velocities than is predicted on the basis of unidirectional flow theory.) Thus, additive concentrations necessary to achieve results applicable to this program would have to be much higher than those tested.

The theoretical analysis presented in Appendix I is limited by complex mathematical problems. Since the change in damping function produces a differential equation to which there is no specific solution, the usefulness of analyses in this program was minimized.

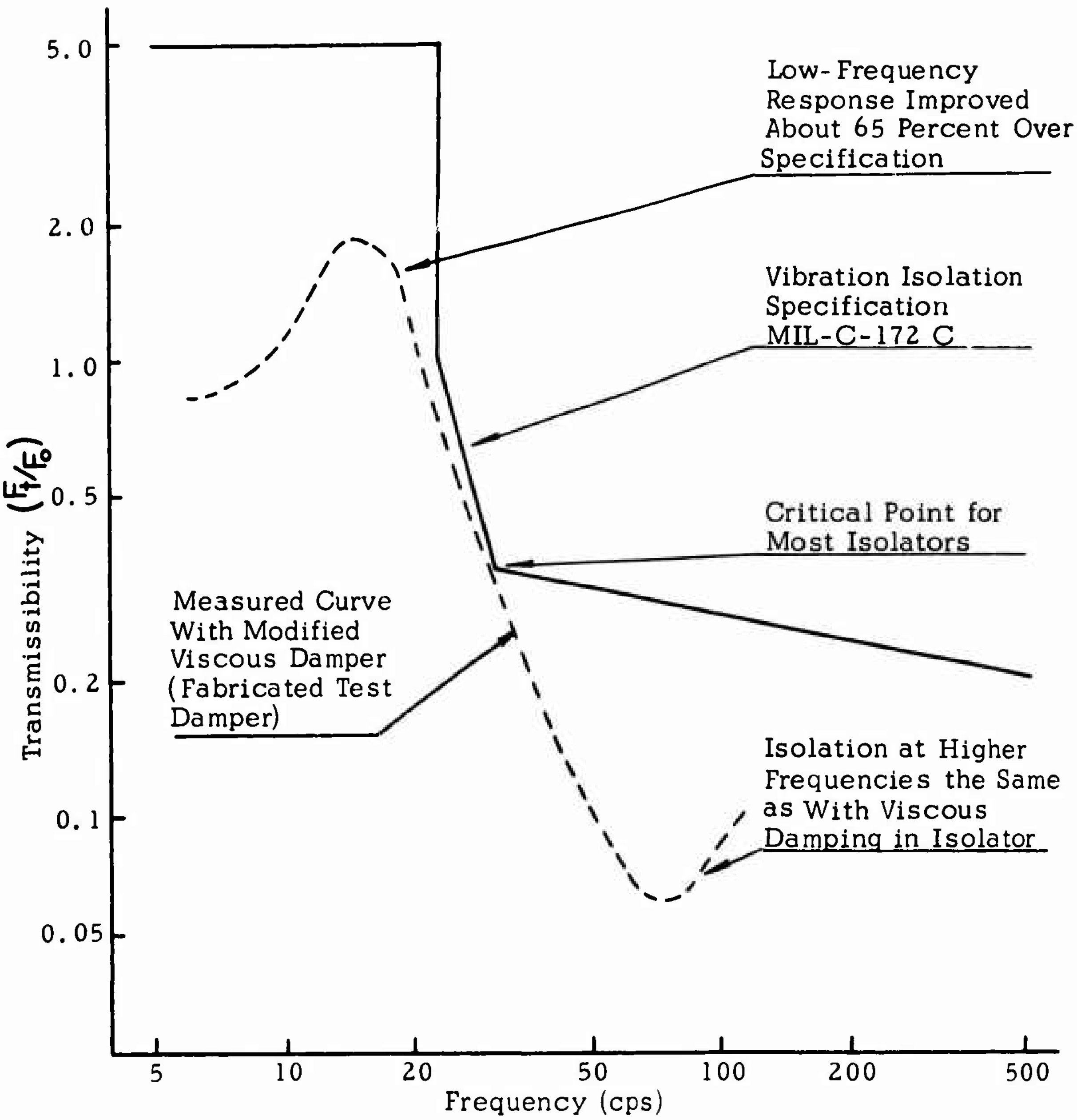


Figure 24. Comparison of Isolator Effectiveness With MIL Specification, Using the Damper.

Changing the mass from 0 or 20 pounds to 40 pounds varies the amount of damping required to yield a given percent of critical, assuming a given spring constant. No unusual effect other than this was noted in this program.

The amount of concentration of the additive must be adjusted to produce the desired effect. This program demonstrated that no analytical prediction exists and that, for a given application, empirical methods must be utilized to achieve a given result.

For a given shock-pulse excitation, the deflection decreased in most cases where transmissibility curves indicated improved isolation; however, some data did not reflect this decrease.

Comparable results were obtained for the fabricated test damper and the GFE damper; however, because of different design parameters, these results were not achieved with the same additives and concentrations. Further study of the reheological properties of gelled fluids could provide an understanding of some of the seemingly unrelated phenomena revealed during the test program.

RECOMMENDATIONS

It is recommended that the evaluated capabilities of gelled fluids be applied to existing problem areas in Army aircraft. Specifically, three major factors should be investigated for valid applications:

Low-frequency attenuation.

Increased isolation system effectiveness in higher frequency ranges.

Transmissibility function adjustment.

A study should be performed to identify problem areas on existing and planned aircraft that can be helped by these new capabilities. When these problem areas are known, individual research programs covering specific vibration and shock problems should be initiated.

Since more data are needed on these effects, a research and development program designed to determine the maximum benefits that can be achieved in isolators should be conducted. Such a program should also provide more design data and permit greater theoretical effort than was possible within the limited scope of this program.

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APPENDIX I
DESIGN OF TEST DAMPER

In designing the test damper, the following criteria were used:

> A high proportion of viscous damping to kinetic pressure drop and frequency-dependent effects.
>
> A design that used readily available components, eliminating most machine work.
>
> A transition from laminar to turbulent flow within a feasible operating range.

The second of these criteria led to the use of the damping effect of the flow of a viscous fluid through a conduit, rather than shearing between solid members having relative velocity. The damping is transmitted to a solid piston using an elastic seal around the edge. The device is shown in Figure 25.

The resistance offered by a viscous fluid in flowing through a conduit is defined in terms of the pressure in the fluid:

$$P_V = \frac{8\pi\mu L\upsilon}{S} = \frac{\left(\frac{\text{lbs-sec}}{\text{in.}^2}\right)(\text{in.})\left(\frac{\text{in.}}{\text{sec}}\right)}{\text{in.}^2} = \frac{\text{lbs}}{\text{in.}^2} \qquad (4)$$

where μ is the absolute viscosity in units of lb-sec/in.^{2}, L is the length of the conduit in inches, υ is the mean velocity of fluid flow in in./sec, and S is the cross-sectional area of the conduit in square inches. Equation (4) should be used only where the Reynolds number is lower than approximately 2000. For greater values, the flow is turbulent. The limit is expressed as follows for a conduit of circular cross section:

$$\frac{\upsilon D\rho}{\mu} < 2000 \qquad (5)$$

where D is the diameter of the conduit in inches, υ is the velocity of flow in in./sec, and ρ is the mass density of the liquid in lb-sec^2/in.^{4}

In addition to the viscous resistance, there is a kinetic resistance corresponding to the pressure drop through an orifice given by

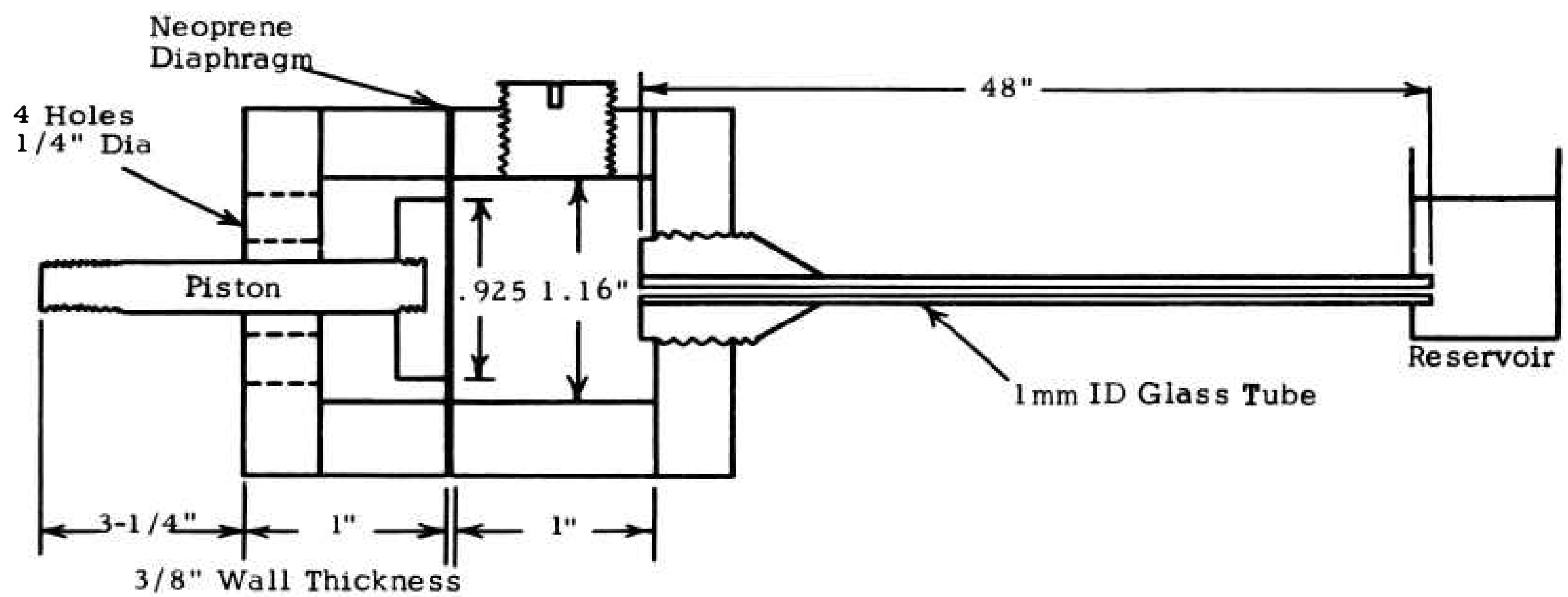


Figure 25. Fabricated Test Damper, Outline Dimensions.

$$P_K = \frac{v^2\rho}{2} \text{ lb/in.}^2 \tag{6}$$

There is also a limitation on frequency defined for a circular tube:

$$\sqrt{\frac{R^2\rho\omega}{2}} < 2 \tag{7}$$

If the parameter exceeds a value of approximately two, the coefficient given by Equation (4) is modified by a frequency-dependent factor.

It is desired to maximize the fraction

$$\frac{P_V}{P_K} = \frac{\text{viscous pressure}}{\text{kinetic pressure}} \tag{8}$$

$$\frac{P_V}{P_K} = \frac{8\pi\mu L v(2)}{\pi r^2 v^2 \rho} = \frac{16\mu L}{r^2 v\rho}$$

Setting $v = \frac{2000\mu}{D\rho}$, $\frac{P_V}{P_K} = \frac{16\mu L}{r^2\left(\frac{2000\mu}{D\rho}\right)\rho} = \frac{16\mu L D\rho}{r^2\, 2000\mu\rho} = \frac{32L}{2000r}$. (9)

Since ρ is determined by the properties of the liquid, $\frac{P_V}{P_K}$ is determined by the fraction $\frac{L}{r}$.

If $\sqrt{\frac{R^2\rho\omega}{2}} \leq 2$, then R_{MAX} can be determined if ω_{MAX} is known. Set $\omega_{MAX} = 30$ hz.

Then $$\frac{R^2(1\times10^{-4})(30)}{2(1.45\times10^{-7})} \leq 4$$

$$\frac{R^2(3)(10^{-3})(10^7)}{2.9} \leq 4$$

$$R^2 \leq \frac{4(2.9)}{3(10^4)} \qquad (10)$$

$$R^2 = 3.87\times10^{-4}$$

$$R \leq 1.97\times10^{-2} \text{ inches}$$

$$\therefore D \cong 1\text{mm}$$

Determination of transition from "soft" to "hard" condition for water:

$$\frac{vD\rho}{\mu} = 2000 \qquad (11)$$

where $$\mu = 1.45\times10^{-7}$$

$$\rho = 1.\times10^{-4}$$

$$v = \frac{2000\mu}{D\rho}$$

then $$v = \frac{2\times10^3(1.45)(10^{-7})}{3.94\times10^{-2}(1\times10^{-4})}$$

$$v = 73.6\ \frac{\text{in.}}{\text{sec}} \qquad (12)$$

Viscous pressure at this point using a 48-inch-long tube is calculated by

$$P = \frac{8\pi(1.45\times10^{-7})(48)(73.6)}{\pi(0197)^2} \qquad (13)$$

$$P = 10.56 \text{ psi}$$

Since the effective area of the piston is approximately 0.785 in^2., the transmitted force is 8.3 lbs.

The piston velocity corresponding to 73.6-in./sec velocity of liquid in the tube is

$$73.6\left(\frac{S_{tube}}{S_{piston}}\right) = 73.6\left[\frac{\pi(.039)^2}{.785}\right] \cong .45\ \frac{\text{in.}}{\text{sec}} \tag{14}$$

Piston displacement $\cong$ .001 inch to .006 inch in a frequency range of from 5 to 30 Hz.

The ratio of viscous pressure to kinetic pressure is

$$\frac{P_V}{P_K} = \frac{32(48)(1 \times 10^{-4})}{2000(1.97 \times 10^{-2})} \cong 39$$

APPENDIX II

THEORETICAL ANALYSIS OF FLUID DAMPERS

1. Motion Equations for Fluid Dampers Operating With and Without Chemical Additives.

The goal of this research program was to investigate the change in fluid damper design characteristics when the rheological properties of the fluid are altered with certain chemical additives. Two different concepts were studied and are discussed in the following paragraphs.

The first concept involves the use of additives to reduce turbulent drag and to extend the useful range of viscous damping. Figure 26 illustrates this phenomenon.

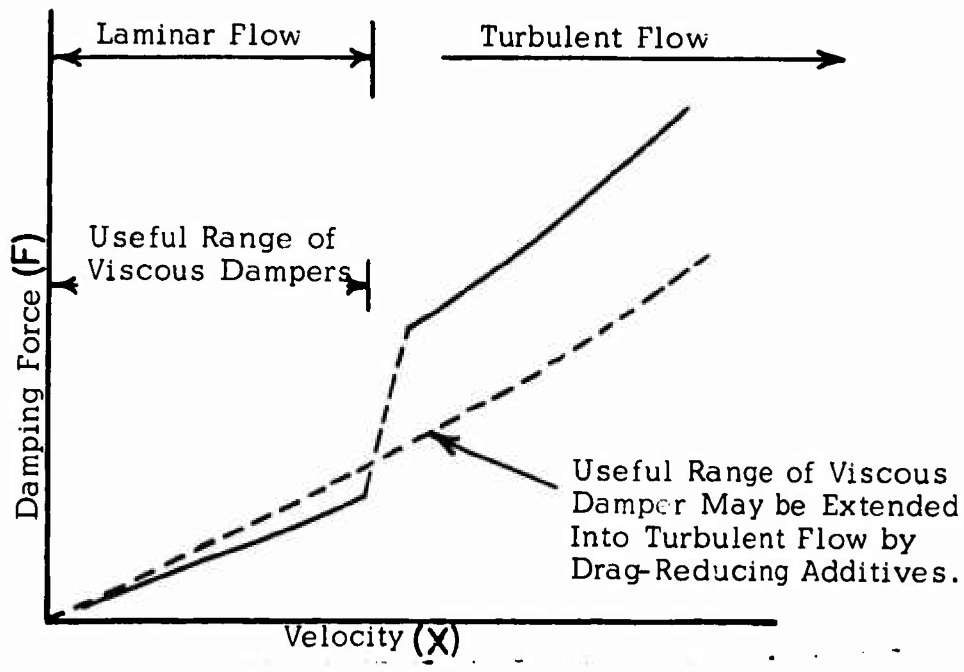


Figure 26. Drag-Reducing Additives.

If we demonstrate that these drag-reducing effects exist in alternating flow, the extension of the useful dynamic range of velocities into the turbulent flow region of fluid dampers can be shown to be feasible.

For this initial concept, the analysis of the physical situation with the mass, spring, and viscous damper is discussed in many textbooks. The differential equation for an isolated mass system and its solution are discussed below.

Viscous Damping

Consider the system shown in Figure 27.

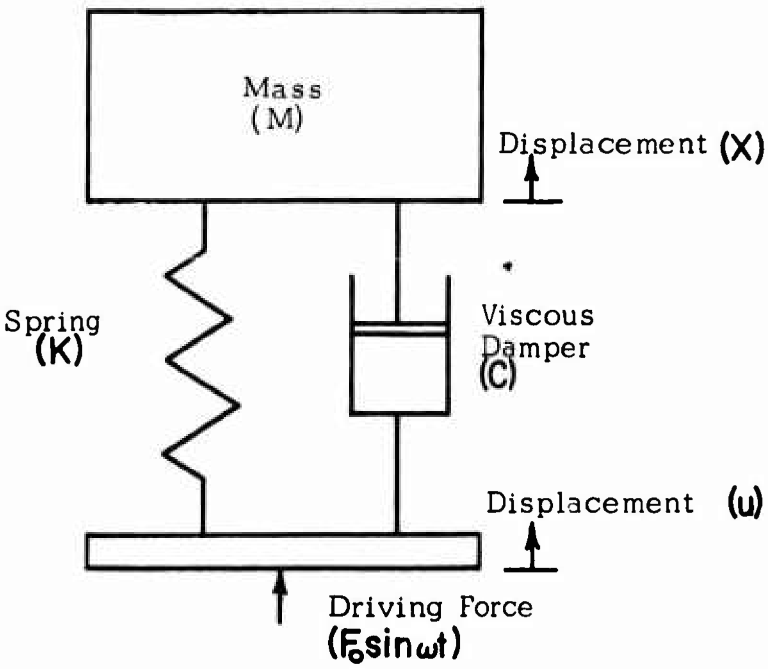


Figure 27. Mass, Spring, and Viscous Damper System With Driving Force.

The differential equation describing this system can be set up by equating the driving force to the sum of the inertial, spring, and damping forces:

$$MX - C(\dot{X} - \dot{u}) - K(X - u) = F_o \sin \omega t \tag{15}$$

where M is the isolated mass, X is the displacement of M, C is the coefficient of viscous damping, u is the displacement of the massless base and plunger arm of the damper, K is the spring constant, and F_o is the maximum amplitude of the driving force.

Solution of this differential equation involves the assumption of the solution and proving the validity of this form. Reference 2 contains an excellent discussion of this proof. The steady-state solution is of this harmonic form:

$$X = X_o \sin(\omega t + \theta) \tag{16}$$

or

$$\delta = \delta_o \sin(\omega t + \theta) \tag{17}$$

where $\delta = x - u$. The resonant frequency of the system is

$$\omega_N = \sqrt{K/M} \tag{18}$$

if $C = 0$. Critical damping is given by

$$N_C = \omega_N = 2\sqrt{K/M} \tag{19}$$

The fraction of critical damping is

$$\xi = \frac{N}{N_C} = \frac{C}{C_C} \tag{20}$$

For the purposes of this program, these equations will suffice. If the turbulent drag-reducing concept proves to be valid, application to a greater range of values is a simple operation.

The second concept to be studied involves changing the rheological properties to produce a "plastic" or "pseudoplastic" material. Such a material has an initial yield strength but flows as velocity increases. It is expected that a viscous damping function will be changed by these drag-increasing additives into an approximation of Coulomb-type damping as illustrated in Figure 28. In many applications this type of damping function is useful.

Coulomb Damping

Consider the spring, mass, damper system as before with Coulomb-type damping, Figure 29. The differential equation for the isolated mass system shown in Figure 29 is

$$M\ddot{X} + K(X - U_o) \pm F_f = F \sin \omega t \tag{21}$$

where F_f is a constant that is independent of velocity.

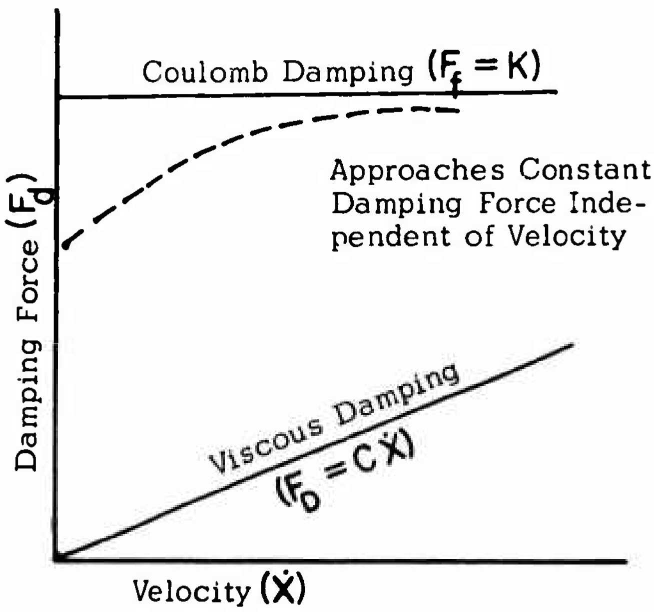


Figure 28. Drag-Increasing Additives.

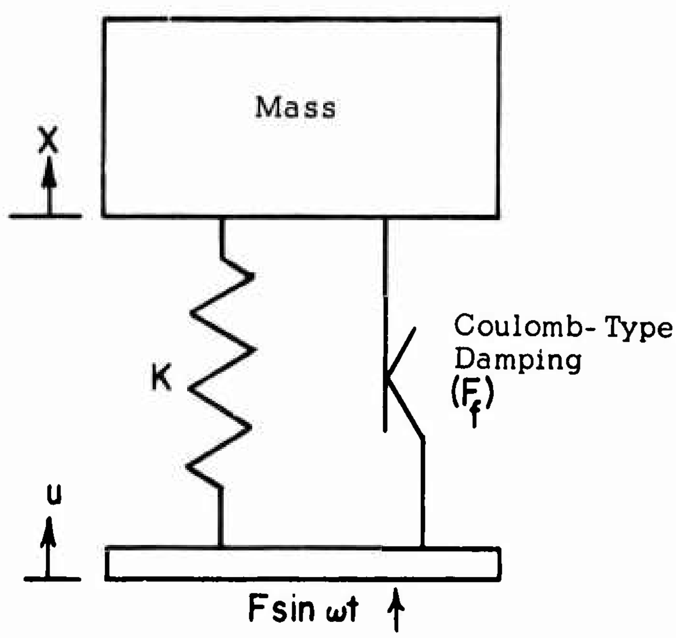


Figure 29. System With Coulomb-Type Damping.

Unfortunately, this differential equation does not have a harmonic solution. At values of input force less than F_f, the damper provides a rigid connection. A discontinuity occurs as the sign of velocity changes each half cycle. This equation has been studied by Drs. Hartog and Jacobsen, and results of approximate step-by-step solutions are reported in References 3 and 4, respectively. Crede summarizes this work in Reference 5 and arrives at a solution based upon equivalence of energy dissipated per cycle, compared to a viscous-damped system. His equation is

$$\pi C \omega \delta_o^2 = 4 F_f \delta_o \tag{22}$$

where the left side refers to the viscous-damped system and the right side refers to the Coulomb-damped system. Solving for C gives

$$C_{EQ} = \frac{4F_f}{\pi \omega \delta_o} \tag{23}$$

where C_{EQ} is the equivalent viscous-damping coefficient. For the purposes of this program, this approximate solution for the damping coefficient will suffice.

Viscous Damping Without Spring

In the experimental program, the spring forces involved may be small. Consider the case illustrated in Figure 30.

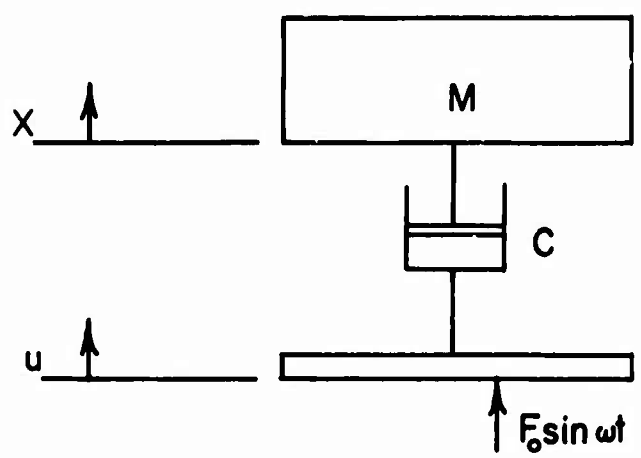


Figure 30. System With Only Damper in Use.

Experimentally, the displacement (u) can be measured by

$$u = U_o \sin \omega t \tag{24}$$

and

$$\dot{u} = \omega U_o \cos \omega t \tag{25}$$

or

$$C\dot{u} = C\omega U_o \cos\omega t \tag{26}$$

In this case, the differential equations are

$$M\ddot{X} + C\dot{X} = C\dot{u} \tag{27}$$

and

$$M\ddot{X} + C\dot{X} = C\omega v \cos\omega t$$

or

$$\ddot{X} + \frac{C}{M}\dot{X} = \frac{C\omega v}{M}\cos\omega t \tag{28}$$

or

$$\frac{d^2X}{dt^2} + \frac{C}{M}\frac{dX}{dt} = \frac{C\omega v \cos\omega t}{M}$$

Let

$$v = \frac{dX}{dt} \tag{29}$$

then

$$\frac{dv}{dt} + \frac{C}{M}v = \frac{CU_o\omega\cos\omega t}{M} \tag{30}$$

and

$$e^{\frac{C}{M}t}\frac{dv}{dt} + \frac{C}{M}e^{\frac{C}{M}t}v = e^{\frac{C}{M}t}\frac{CV}{M}\omega\sin\omega t \tag{31}$$

Integrating both sides of the equation yields

$$ve^{\frac{C}{M}t} = e^{\frac{C}{M}t}\frac{U_o\omega C}{M}\left(\frac{\frac{C}{M}\cos\omega t + \omega\sin\omega t}{\frac{C^2}{M^2} + \omega^2}\right) + K \tag{32}$$

or
$$\frac{dX}{dt} = v = \frac{u\omega C}{M}\left[\frac{1}{\frac{C^2}{M^2}+\omega^2}\right]\left(\frac{C}{M}\cos\omega t + \omega\sin\omega t\right) + K e^{-\frac{Ct}{M}} \tag{33}$$

or
$$X = \frac{v\omega C}{M}\left[\frac{1}{\frac{C^2}{M^2}+\omega^2}\right]\left(\frac{C}{M\omega}\sin\omega t - \cos\omega t\right) - \frac{K}{C}e^{-\frac{Ct}{M}} + A \tag{34}$$

$$A=0$$

Evaluating $\frac{dX}{dt}$ and X at $t=0$ yields

$$K = \frac{-u\omega C}{C^2+\omega^2 M^2} \tag{35}$$

and

$$X = \frac{u\omega C}{M}\left[\frac{\frac{C}{M\omega}\sin\omega t - \cos\omega t}{\frac{C^2}{M^2}+\omega^2}\right] - \frac{KM}{C}e^{-\frac{Ct}{M}} \tag{36}$$

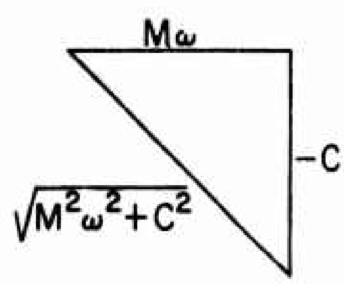
If t is large and $\frac{dX}{dt} \longrightarrow 0$, then

$$\frac{C}{M}\cos\omega t + \omega\sin\omega t = 0$$

or
$$\arctan\left(\frac{-C}{M\omega}\right) = \omega t \tag{37}$$

or
$$\frac{1}{\omega}\arctan\left(\frac{-C}{M\omega}\right) = t$$

i.e.,



Then $X(t)$ is an extreme and, by substitution,

$$X = \frac{-uC}{\sqrt{C^2+\omega^2 M^2}} \tag{38}$$

2. Analytical Determination of Relative Transmissibility.

Transmissibility equations for viscous and Coulomb damping are given in equations (37) and (38), respectively. Curves for these equations can be found in Reference 1.

<u>Viscous Damping</u>

$$T_r = \frac{\delta_o}{U_o} = \sqrt{\frac{(\omega/\omega_o)^4}{(1-\omega^2/\omega_o^2)^2+(2\zeta\,\omega/\omega_o)^2}} \tag{39}$$

<u>Coulomb Damping</u>

$$T_r = \frac{\delta_o}{U_o} = \sqrt{\frac{(\omega/\omega_o)^4-(4/\pi-\bar{n})^2}{(1-\omega^2/\omega_o^2)^2}} \tag{40}$$

where

$$\bar{n} = \frac{F_f}{KU_o}$$

3. Analytical Determination of Deflection for Isolated Masses.

Deflection of isolated masses can be obtained from absolute displacement transmissibility equations. Curves for these equations can also be found in Reference 1.

Viscous Damping

$$T_A = \frac{X_o}{U_o} = \frac{F_T}{F_o} = \frac{1+(2\xi\,\omega/\omega_o)^2}{(1-\omega^2/\omega_o^2)^2+(2\xi\,\omega/\omega_o)^2} \tag{41}$$

Coulomb Damping

$$T_A = \frac{X_o}{U_o} = \sqrt{\frac{(1+4/\pi\,\bar{n})^2(1-2\,\omega_o^2/\omega^2)}{(1-\omega^2/\omega_o^2)^2}} \tag{42}$$

For the case of a mass and viscous damper without a spring, the relative transmissibility is

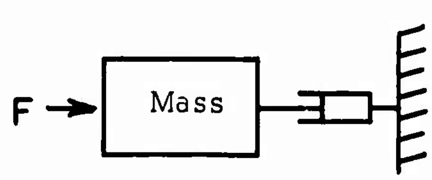
$$T_r = \left| \frac{C}{\sqrt{C+\omega^2 M^2}} - 1 \right| \tag{43}$$

4. Analytical Determination of the Effect That Varying Additive Percentage Has on Damping Characteristics of Damper.

Findings of this program indicated that under alternating flow conditions, the effects of varying the percentage of additives were very complex. Analytical determination of the effects is considered to be beyond the scope of this program.

Additional Analysis

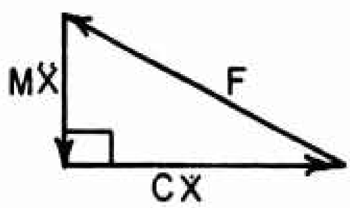
It became apparent that the damping characteristics of both the GFE damper and the test damper must be measured in the laboratory. The diagram below shows how these laboratory tests were set up.



The differential equation for this system is

$$F_o \sin \omega t = M\ddot{X} + C\dot{X} \tag{44}$$

where $F_o \sin \omega t$ is the impressed forcing function, M is the mass of the plunger arm of the damper under test, and C is the coefficient of damping of the damper. In the laboratory, it is convenient to monitor magnitudes of F_o , $\dot{X}$, and $\ddot{X}$, and, since $\dot{X}$ and $\ddot{X}$ are 90 degrees out of phase, this force diagram of force magnitudes can be assumed to be



Magnitudes of these forces are related by

$$(F)^2 = (M\ddot{X})^2 + (C\dot{X})^2 \tag{45}$$

or

$$F = \sqrt{(M\ddot{X})^2 + (C\dot{X})^2} \tag{46}$$

Since X, $\dot{X}$, and $\ddot{X}$ are related in the manner

$$x = X \sin \omega t$$

and

$$\dot{x} = X \omega \cos \omega t \tag{47}$$

or

$$\ddot{x} = \omega^2 X \sin \omega t$$

then

$$\ddot{x} = \omega X \tag{48}$$

then

$$F = \sqrt{(M\omega\dot{X})^2 + (C\dot{X})^2} \tag{49}$$

or

$$\frac{F}{\dot{X}} = \sqrt{(M\omega)^2 + (C)^2} \tag{50}$$

If, in the laboratory, F and X are determined for various frequencies, the coefficient of damping C can be determined by solving simultaneous equations of this form produced from experimental data.

APPENDIX III
TEST DATA AVAILABILITY

The quantity of available test data precludes its inclusion in this document. Data for specific tests as required for evaluation may be obtained from the contracting agency shown on the title page of this document.

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11. SUPPLEMENTARY NOTES	12. SPONSORING MILITARY ACTIVITY U. S. Army Aviation Materiel Laboratories Fort Eustis, Virginia

13. ABSTRACT

A research and development program has been conducted for the purpose of developing methods of improving the operating characteristics of passive vibration and shock isolators. To achieve such improvement, rheological properties of base fluids in viscous dampers were altered for alternating flow conditions. Newtonian flow properties of both water and MIL-H-5606A hydraulic fluid were changed with chemical additives to yield pseudoplastic flow properties having an initial yield strength at low shear rates and "shear thinning" characteristics at high shear rates. When such modified fluids were used in the fluid dampers of the tested isolation systems, damping resulted. The advantages of this damping in limiting amplification at low-frequency resonance, while providing effective isolation at higher frequencies, were realized and recorded during system tests.

Both analytical and dynamic situations were checked. Laboratory experiments included tests with a special test damper and with a Government-furnished (GFE) damper. These devices were both tested as a part of isolation systems. Significant changes in operating characteristics were achieved.

DD FORM 1 NOV 65 1473 REPLACES DD FORM 1473, 1 JAN 64, WHICH IS OBSOLETE FOR ARMY USE.

Unclassified
Security Classification

14. KEY WORDS	LINK A		LINK B		LINK C	
	ROLE	WT	ROLE	WT	ROLE	WT
Non-Newtonian						
Gelled Fluids						
Shock Isolation						
Vibration Isolation						
Rheology						
Fluid Properties Alteration						
Fluid Damping						