Reduction of Acoustic Responses using Viscoelastic Damping Materials

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ABSTRACT

This paper presents the results of a development test conducted at GE/ASD for reduction of acoustic responses of honeycomb panels using viscoelastic damping material. The damping strip consists of viscoelastic damping material, honeycomb core, and graphite constraint layer. Acoustic tests were conducted on the undamped and damped panels. The comparison of the acoustic responses with and without damping strip are presented. Analytical prediction of the reduction factors and their comparison with the test results are also described.

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INTRODUCTION

Vibroacoustic damping material has been used successfully over the past 17 years at GE/ASD to reduce the vibroacoustic responses of spacecraft structures (References 1-5). The applicatons include printed circuit boards of electronic boxes and equipment panels. This paper discusses the analysis and test results of a damped honeycomb panel. Comparision of the analysis prediction and test results is also discussed.

TEST DESCRIPTION AND RESULTS

The acoustic test was conducted on the test panel with and without the damping strips. The panel is an aluminum honeycomb panel. Two components are attached to the panel to simulate a typical mass loaded panel for spacecraft applications. Thirteen accelerometers were installed on the panel to measure the responses. Figure 1 shows the test panel and accelerometer locations. The shaded line indicates the location of damping strips. The cross-section of the strip is shown in Figure 2. The panel is supported by a rigid frame mounted around the boundary of the panel. During the test, the panel was laid on the floor, supported by thick forms. Six microphones were located near the test panel to measure the sound pressure level.

The test panel was exposed to the protoflight acoustic level environment in the 10,000 cubic foot acoustic chamber. Two tests were conducted; one before and the other after the application of the damping material to the panel. Table 1 summarizes acoustic test levels. The first column identifies the 1/3 octave band and center frequencies and the second column gives the test specification.

The panel vibroacoustic responses which are of most significance for components mounted on the panel are those at component mounting locations. These are the responses that dictate component random vibration test requirements. Test measurements at component mounting locations for the damper and undamper panels are compared in Figure 3 for accelerometer 6Z. These responses have been scaled to a common acoustic test environment which corresponds to the test environment measured in the undamped panel acoustic test. Figure 3 shows that significant reductions of peak levels in the undamped panel random vibration due to damping.

ANALYSIS CORRELATION

Modal damping of the test panel due to viscoelastic damping is determined using modal strain energies and viscoelastic material damping properties. In this approach, the viscoelastic material is represented in a finite element model by the real part of the material shear modulus where the complex shear modulus is given by:

$$G = (1 + i\eta_v)G \qquad (1)$$

where

G = real part of complex sheet modulus

η_v = viscoelastic material composite loss factor

A NASTRAN model of the panel is used to compute the fraction of the modal strain energy in the viscoelastic material for each mode. The fraction of the modal strain energy for each mode is then multiplied by the viscoelastic material loss factor at the modal frequency and at the appropriate temperature.

The finite element model used for the damping prediction is shown in Figure 4. The model included the basic panel and the damper strips which consisted of viscoelastic damping material layer, aluminum honeycomb core, and graphite/epoxy constraining layer for each of the damper strips. The predicted damped panel modal frequencies and composite loss factor for all the panel modes up to 500 Hz are presented in Table 2 and in Figure 5. As shown in Table 2, some modes have modal loss factors which are relatively high while others are somewhat low.

To estimate the reduction of random vibration responses for the damped versus the undamped panel for the same acoustic environment, a reduction factor can be calculated based on the loss factor of undamped and damped panels. The reduction factor is defined as:

$$\text{REDUCTION FACTOR} = (QUD/QD)^2 \qquad (2)$$

where

QUD = undamped panel amplification factor

QD = 1/(.7 x LOSS FACTOR + (1/QUD))

and

LOSS FACTOR = the predicted modal composite loss factor due to viscoelastic damping only

Only 70 percent of the predicted modal loss factor due to viscoelastic damping was assumed to be effective based on previous experience with viscoelastic damped panels (Ref. 5).

Results for reduction factor based on prediction and test data are plotted in Figure 6. Six accelerometer responses at the component mounting locations were used in Figure 6. Test values are based on reductions in levels for significant peaks in the undamped panel random vibration responses. Figure 6 indicates that significant reductions in random vibration levels have been achieved and that predictions are generally conservative over most of the frequency range of interest.

SUMMARY

Comparison of predicted and test results for random vibration reductions due to viscoelastic damping for the test panel shows that:

1) Reduction of random vibration peaks seen in panel responses at component mounting points was generally greater than predicted. Reasonable agreement was obtained between analytical predictions and experimental results.

2) Significant reductions in test responses indicate that the damping design methodology is effective for honeycomb panels.

ACKNOWLEDGEMENT

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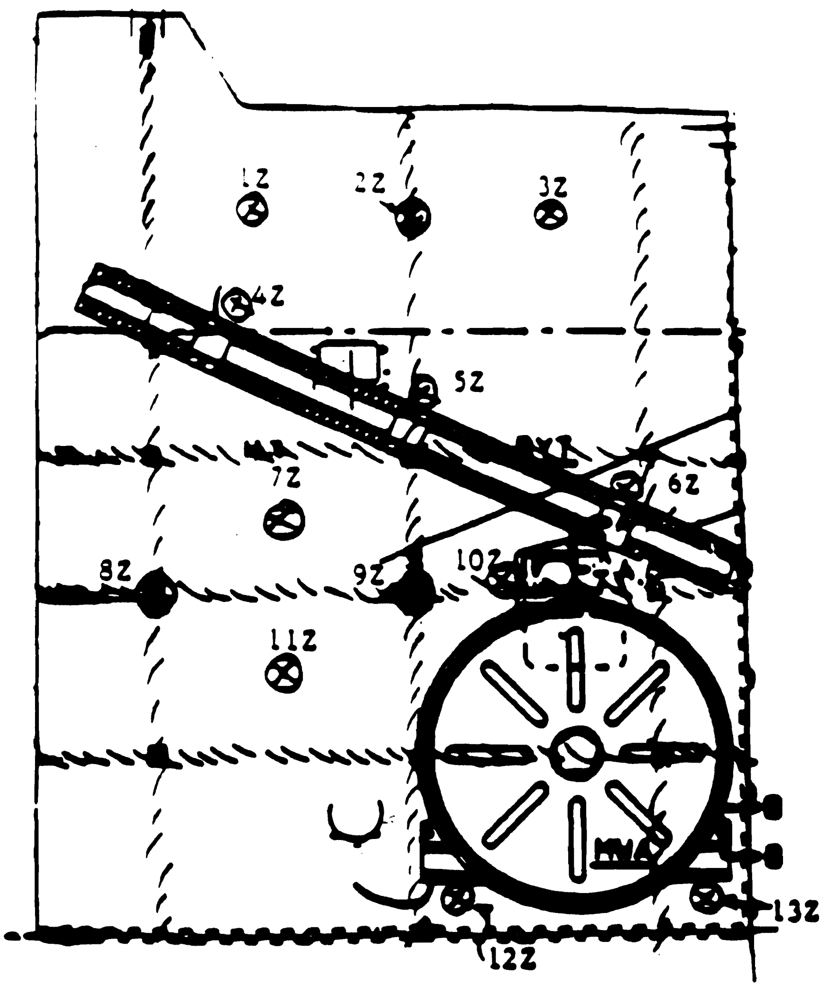

Figure 1. Test Panel and Accelerometer Locations

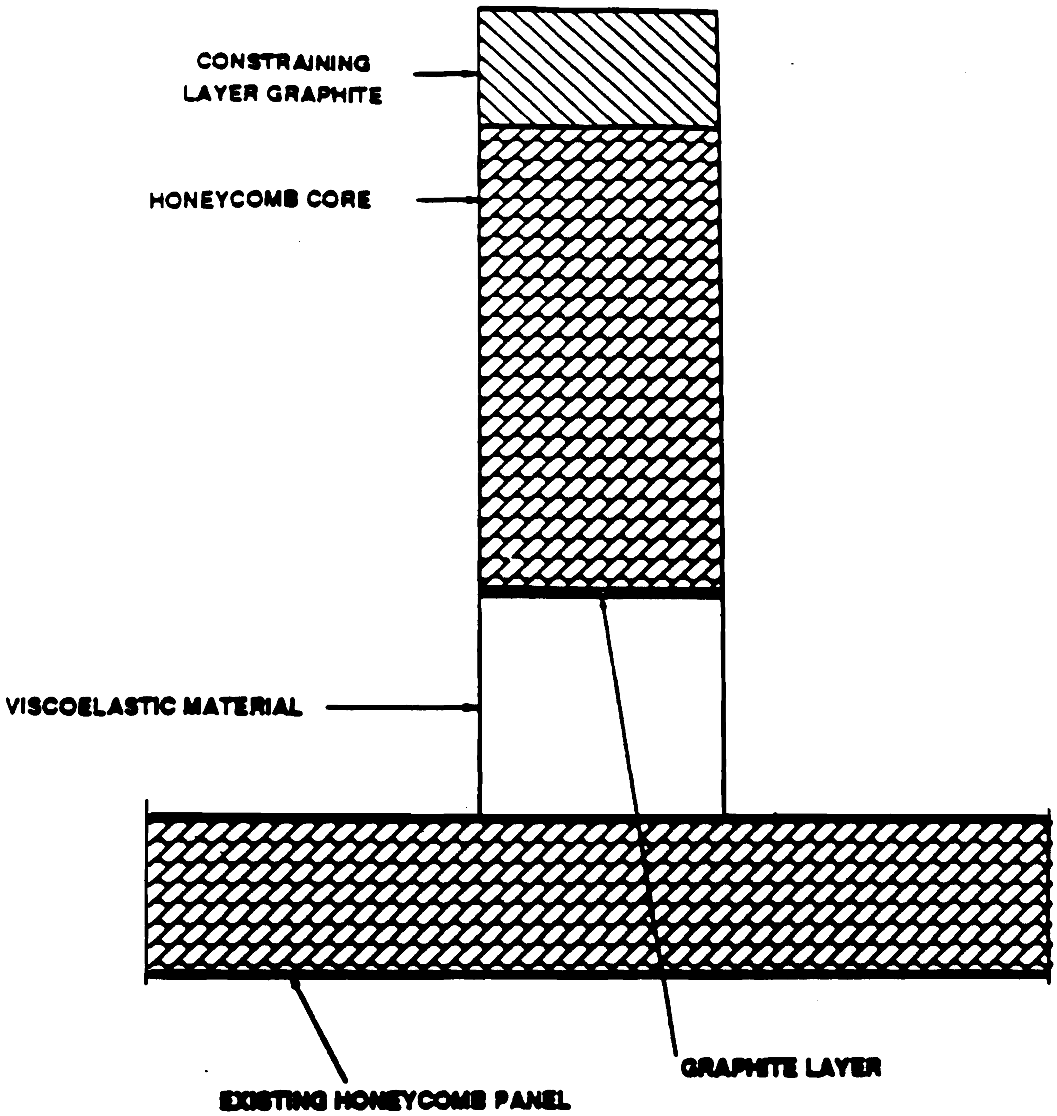

Figure 2. Damper Strip Cross-Section

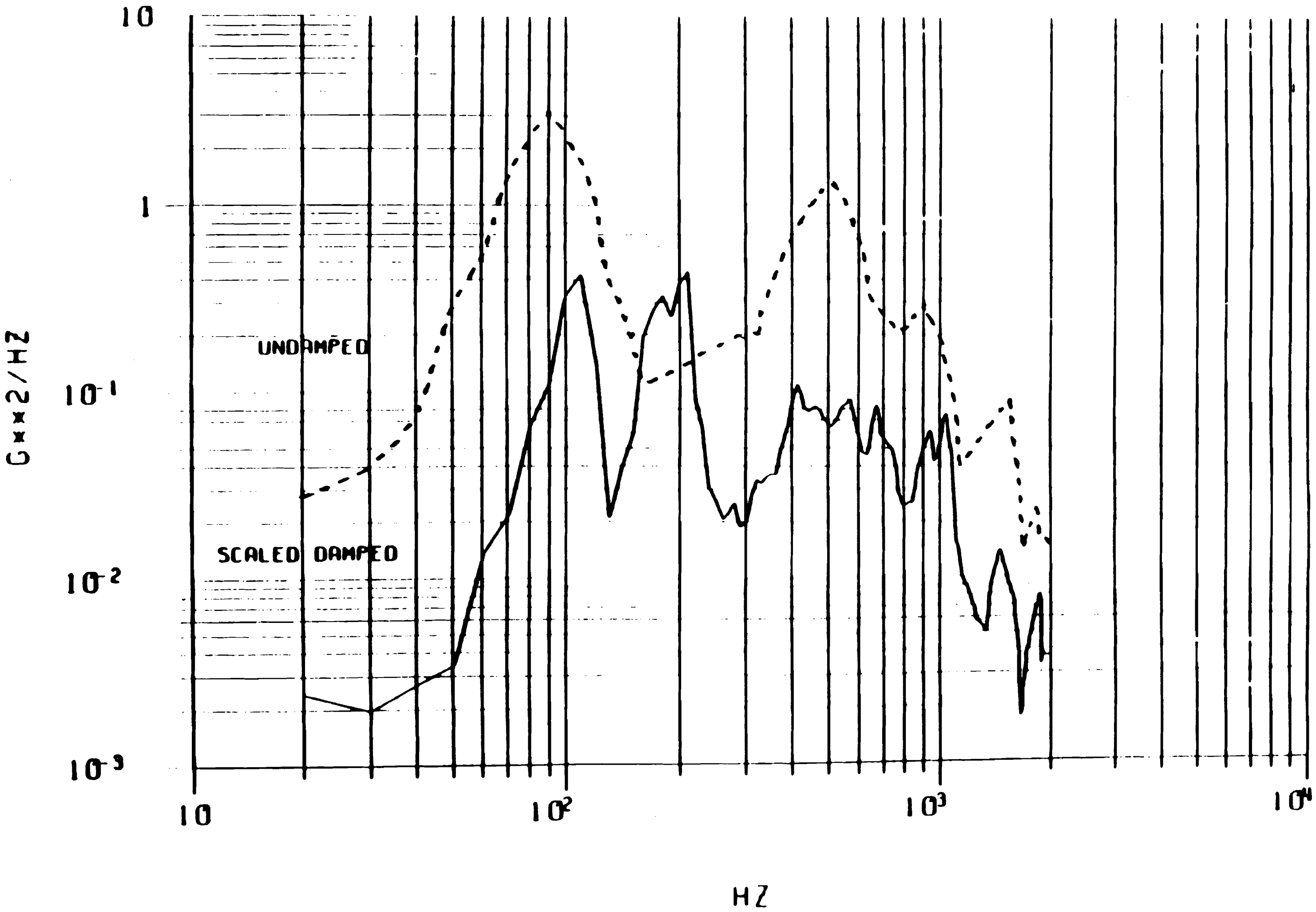

Figure 3. Damped and Undamped Panel Test Responses for Accelerometer #6Z

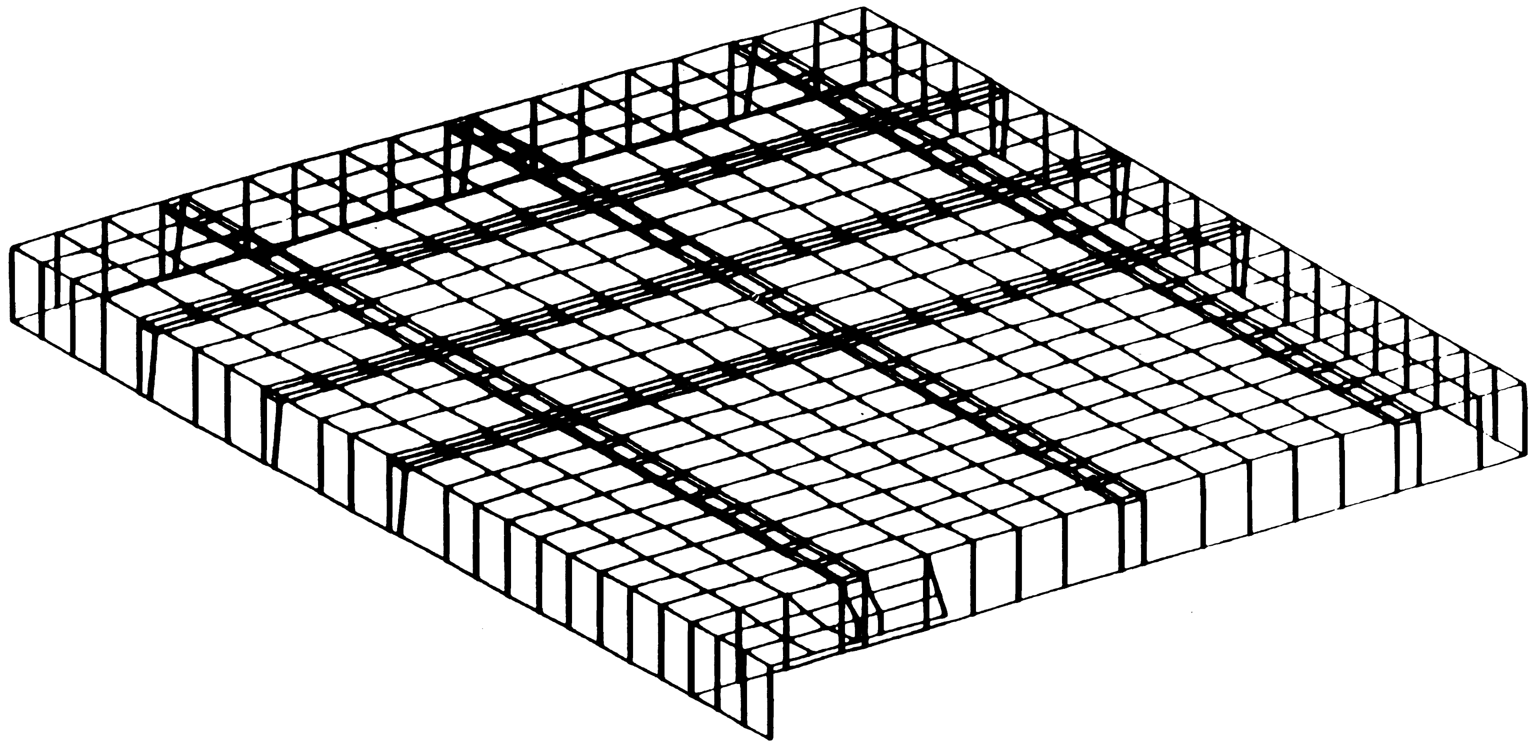

Figure 4. Damped Panel Finite Element Model

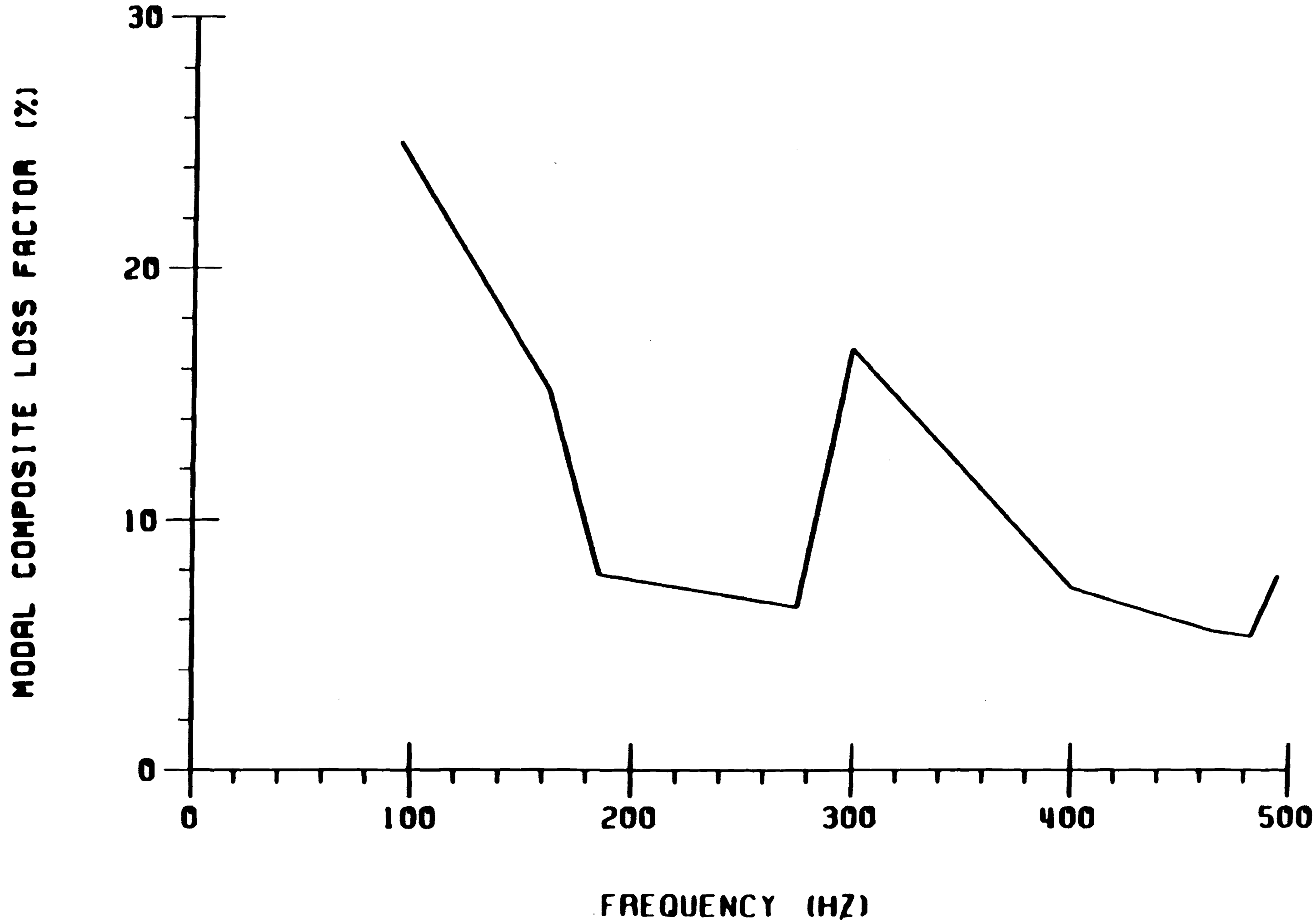

Figure 5. Predicted Modal Composite Loss Factor Vs Modal Frequency

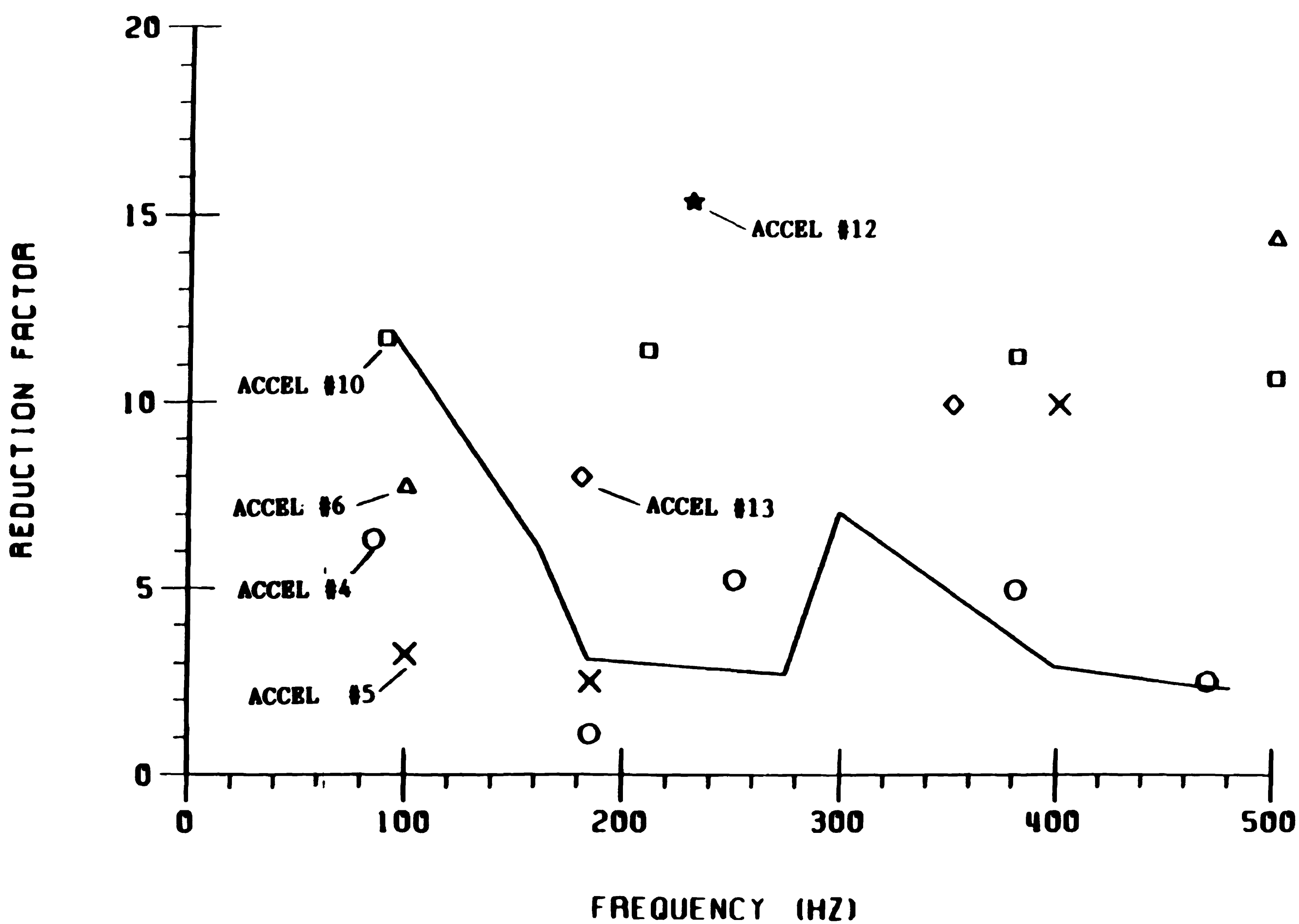

Figure 6. Reduction Factor—Prediction and Test

Table 1. Acoustic Test Level for Damping Material Test

FREQ. (Hz)	SPEC. (dB)
40	128.5
50	131.0
63	134.0
80	136.5
100	138.0
125	138.0
160	138.0
200	136.5
250	135.0
315	133.0
400	131.0
500	129.0
630	127.5
800	125.5
1000	124.0
1250	122.5
1600	121.0
2000	119.5
2500	117.5
3150	115.0
4000	113.0
5000	111.0
6300	109.0
8000	107.0
10000	105.0
Overall	146.2

Table 2. Damped Panel Predicted Modal Frequencies and Composite Loss Factors

MODE	UNDAMPED FREQ. (HZ)	FREQ. (HZ)	CLF (%)
1	57.9	92.6	25.03
2	121.1	160.7	15.12
3	155.3	184.2	7.74
4	259.2	273.8	6.51
5	269.0	297.8	16.87
6	371.9	399.1	7.28
7	441.4	463.8	5.56
8	468.6	481.4	5.35
9	481.8	493.4	7.82

AVE LOSS FACTOR = 10.8%
MAX LOSS FACTOR = 25.0%
MIN LOSS FACTOR = 5.4%