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Application of Impact Dampers in Vibration Control of Flexible Structures

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TABLE 4.- MULTIPLE REGRESSION ANALYSIS OF THE RESULTS FOR THE FIRST MODE

Multiple Regression Y_1:DAMPING (%) 3 X variables

DF:	R:	R-squared:	Adj. R-squared:	Std. Error:
19	.938	.881	.858	.07

Analysis of Variance Table

Source	DF:	Sum Squares:	Mean Square:	F-test:
REGRESSION	3	.581	.194	39.399
RESIDUAL	16	.079	.005	p = .0001
TOTAL	19	.66		

Residual Information Table

SS[e(i)-e(i-1)]:	e ≥ 0:	e < 0:	DW test:
.147	11	9	1.863

1

Multiple Regression Y_1:DAMPING (%) 3 X variables

Beta Coefficient Table

Parameter:	Value:	Std. Err.:	Std. Value:	t-Value:	Probability:
INTERCEPT	.315				
Im(H)/FRQ	-.017	.007	-.337	2.5	.0237
MASS*MS^2	7.362	.974	.813	7.555	.0001
GAP (mm)	.02	.005	.495	4.277	.0006

2

Multiple Regression Y_1:DAMPING (%) 3 X variables

Confidence Intervals and Partial F Table

Parameter:	95% Lower:	95% Upper:	90% Lower:	90% Upper:	Partial F:
INTERCEPT					
Im(H)/FRQ	-.032	-.003	-.029	-.005	6.248
MASS*MS^2	5.296	9.428	5.66	9.063	57.072
GAP (mm)	.01	.03	.012	.028	18.289

3

TABLE 3.- DAMPING ESTIMATES FOR THE THIRST MODE

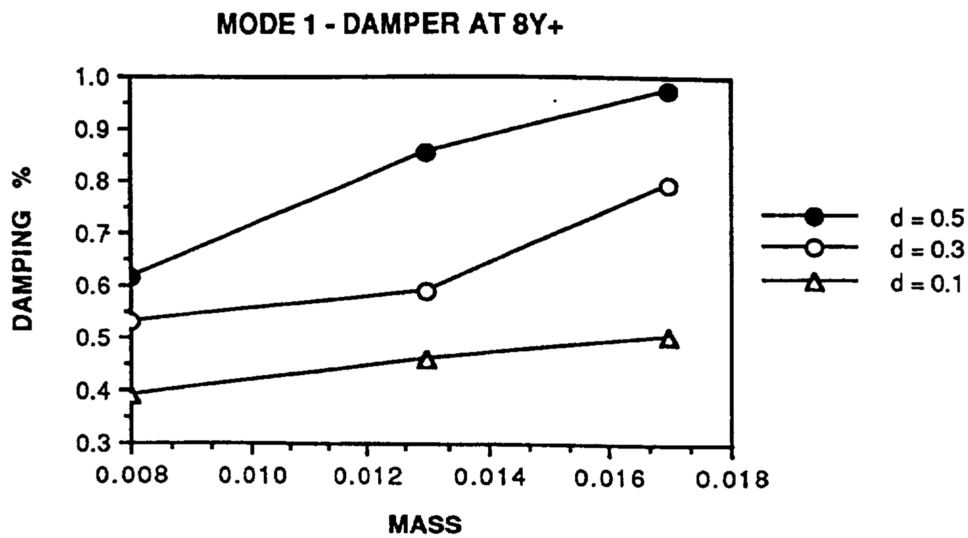
Test File	Damping %	Mass (kg)	Gap (m)	Im(FRF) ($m/s_2/N$)	Freq. (Hz)	Φ_d
TA_11110F3	0.356	0.017	0	188.9	22.12	1.436
TB_11111F3	0.692	0.017	0.013	100.4	22.2	1.436
TB_11112F3	0.384	0.017	0.008	134.66	22.2	1.436
TA_11113F3	1.27	0.017	0.003	35.82	22.2	1.436
TB_11121F3	0.693	0.013	0.013	99.22	22.16	1.436
TB_11122F3	0.473	0.013	0.008	164.28	22.16	1.436
TA_11123F3	1.535	0.013	0.003	41.54	22.16	1.436
TA_11131F3	0.461	0.008	0.013	145.48	22.38	1.436
TB_11132F3	0.638	0.008	0.008	86.48	22.38	1.436
TA_11133F3	0.82	0.008	0.003	83.17	22.38	1.436
TD_11210F3	0.318	0.017	0	101.61	24.32	0.837
TD_11211F3	0.408	0.017	0.013	84.71	24.38	0.837
TD_11212F3	0.446	0.017	0.008	70.64	24.38	0.837
TD_11213F3	0.504	0.017	0.003	73.66	24.38	0.837
TC_11221F3	0.357	0.013	0.013	95.23	24.38	0.837
TD_11222F3	0.392	0.013	0.008	81.41	24.38	0.837
TD_11223F3	1.086	0.013	0.003	35.35	24.38	0.837
TD_11231F3	0.371	0.008	0.013	91.95	24.38	0.837
TD_11232F3	0.341	0.008	0.008	98.01	24.38	0.837
TD_11233F3	0.559	0.008	0.003	62.49	24.38	0.837

TABLE 2.- DAMPING ESTIMATES FOR THE SECOND MODE

Test File	Damping %	Mass (kg)	Gap (m)	Im(FRF) ($m/s_2/N$)	Freq. (Hz)	Φ_d
TA_11110F2	0.37	0.017	0	170.86	13.05	1.393
TA_11111F2	0.566	0.017	0.013	113.2	13.07	1.393
TA_11112F2	0.728	0.017	0.008	92.94	13.07	1.393
TA_11113F2	0.62	0.017	0.003	99.33	13.07	1.393
TA_11121F2	1.97	0.013	0.013	32.77	13.07	1.393
TA_11122F2	1.19	0.013	0.008	53.49	13.07	1.393
TA_11123F2	0.765	0.013	0.003	87.67	13.07	1.393
TA_11131F2	0.593	0.008	0.013	112.06	13.19	1.393
TA_11132F2	1.08	0.008	0.008	60.38	13.19	1.393
TA_11133F2	0.621	0.008	0.003	104.19	13.19	1.393
TA_11510F2	0.447	0.017	0	127.11	13.61	0.988
TA_11511F2	0.684	0.017	0.013	82.82	13.62	0.988
TA_11512F2	0.652	0.017	0.008	85.96	13.62	0.988
TA_11513F2	0.828	0.017	0.003	70.39	13.62	0.988
TA_11521F2	0.602	0.013	0.013	93.04	13.7	0.988
TA_11522F2	0.63	0.013	0.008	90.05	13.7	0.988
TA_11523F2	0.608	0.013	0.003	92.73	13.7	0.988
TA_11531F2	0.592	0.008	0.013	94.83	13.67	0.988
TA_11532F2	0.601	0.008	0.008	92.8	13.67	0.988
TA_11533F2	0.683	0.008	0.003	83.09	13.67	0.988

TABLE 1.- DAMPING ESTIMATES FOR THE FIRST MODE

Test File	Damping %	Mass (kg)	Gap (m)	Im(FRF) ($m/s_2/N$)	Freq.	Φ_d
TN_21000	0.336	0.017	0	72.34	4.44	1.857
TM_21111	0.976	0.017	0.013	25.66	4.44	1.857
TM_21112	0.797	0.017	0.008	33.11	4.44	1.857
TM_21113	0.503	0.017	0.003	54.83	4.44	1.857
TM_21121	0.855	0.013	0.013	29.56	4.467	1.857
TM_21122	0.59	0.013	0.008	49.86	4.467	1.857
TN_21123	0.461	0.013	0.003	63.16	4.467	1.857
TN_21131	0.617	0.008	0.013	34.68	4.5	1.857
TN_21132	0.532	0.008	0.008	54.8	4.5	1.857
TN_21133	0.391	0.008	0.003	70.81	4.5	1.857
TA_21510F1	0.31	0.017	0	41.06	5.535	0.869
TA_21511F1	0.482	0.017	0.013	30.06	5.54	0.869
TA_21512F1	0.359	0.017	0.008	37.41	5.54	0.869
TA_21513F1	0.344	0.017	0.003	38.29	5.54	0.869
TA_21521F1	0.53	0.013	0.013	24.84	5.55	0.869
TA_21522F1	0.456	0.013	0.008	30.89	5.55	0.869
TA_21523F1	0.356	0.013	0.003	39.7	5.55	0.869
TA_21531F1	0.452	0.008	0.013	30.23	5.565	0.869
TA_21532F1	0.383	0.008	0.008	34.61	5.565	0.869
TA_21533F1	0.311	0.008	0.003	43.48	5.565	0.869



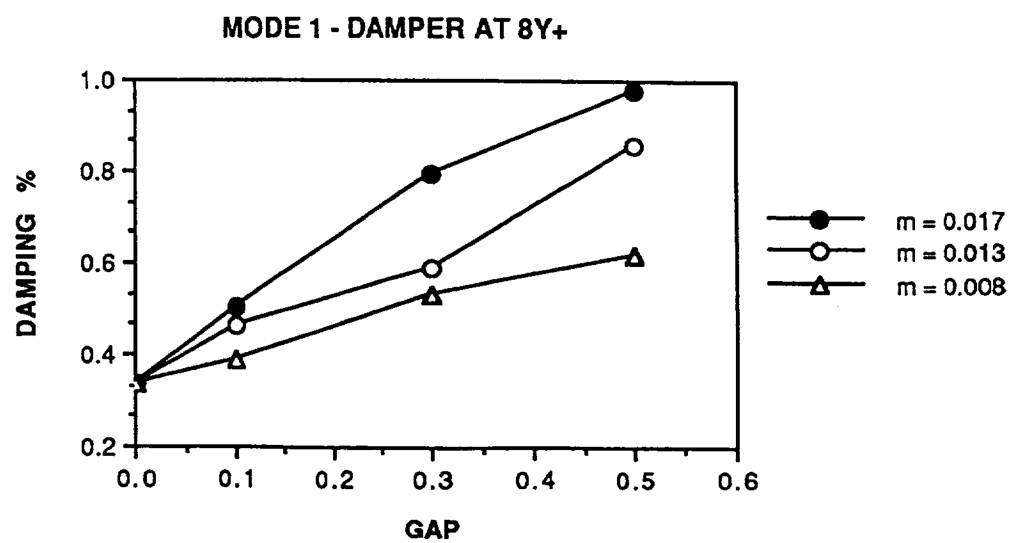
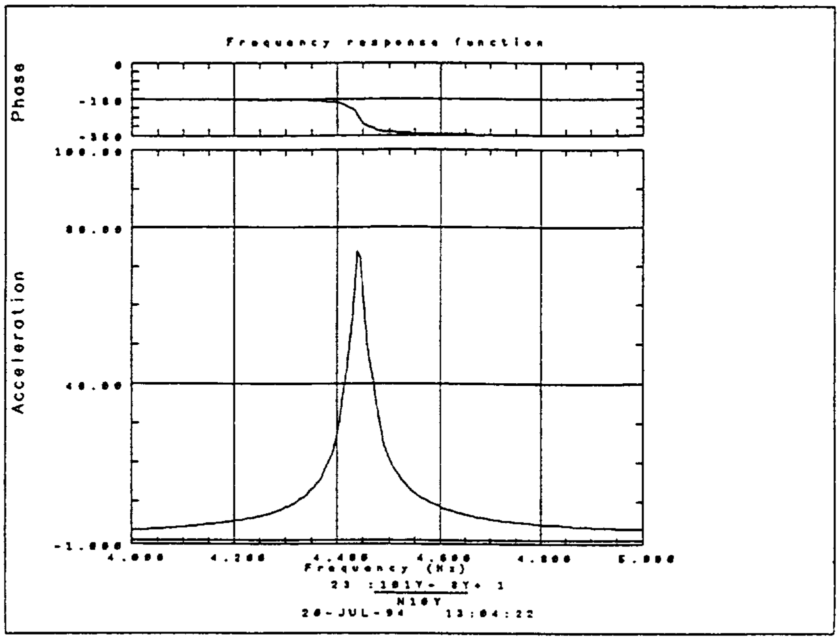


Figure 4.- Variation of Damping with Mass and Gap



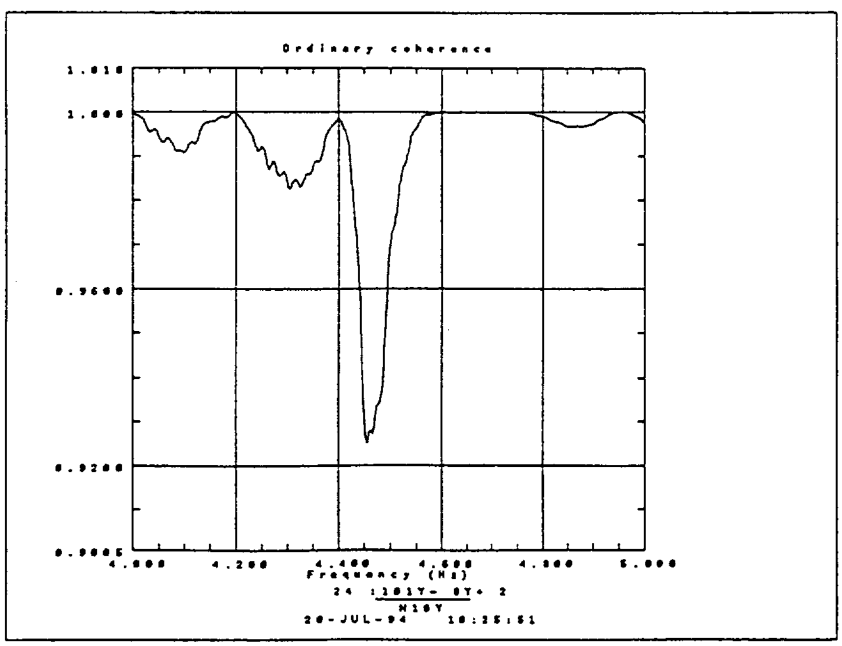
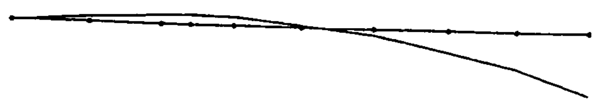
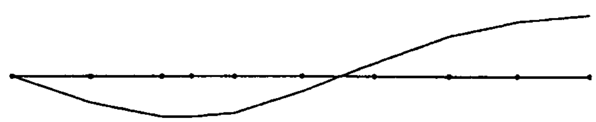


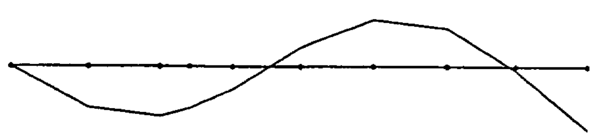
Figure 3.- Frequency Response Function and Coherence Plots for the First Mode



Mode 1 (f_1 = 5.53 Hz) - Spring at 81.5 in



Mode 2 (f_2 = 13.04 Hz) - Spring at 121.25 in



Mode 3(f_3 = 22.12 Hz) - Spring at 121.25 in

Figure 2.- Modes Shapes of the Test Structure

to broaden the scope and the range of the parameters governing the behavior of flexible structures. Numerical simulation of the behavior of the problem would be needed to predict the behavior of the flexible structure under operating conditions in space.

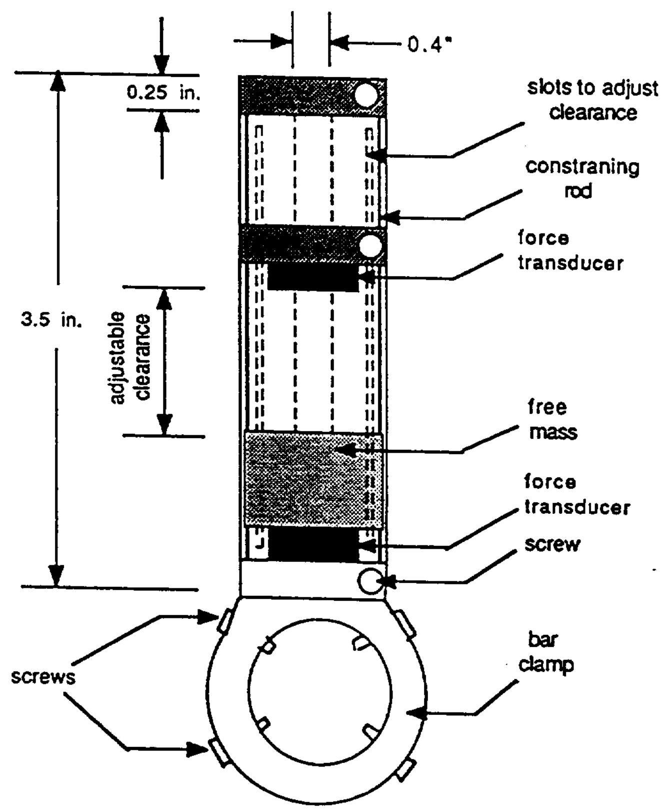


Figure 1.- Details of the Impact Damper

as follows.

$$\int_l v_i(x)\, dm + \int F_s\, dt + F_d\, (\Delta T) = \int_l v_f(x)\, dm$$

Where:

$v_i(x),\ v_{f(x)}$	the initial and final velocities, respectively.
l	the length of the test structure.
dm	an infinitesimal mass of the test structure.
F_s	the force supplied by the shaker.
$F_d(\Delta T)$	the impulse exerted by the damper.

The increase in the damping ratio of the test structure may be attributed to the presence of the third term in the L.H.S in the impulse and momentum equation above. The momentum of the beam is expected to be in the same direction before and after the impact. Therefore if the impulse is in the opposite direction to the momentum of the test article, it is then expected that the resulting momentum at the end of the half-cycle duration will be less due to the contribution of the impulse from the impact damper.

Figures 4 shows a plot of the variation of the damping ratio as a function of the moving mass of the damper and the gap. It is somewhat difficult to draw general conclusions about the behavior of the test structure form these figures.

A multiple linear regression analysis was performed using the *StatView* software. We observed, from the numerous tests we conducted, that in order for the damper to be active, it is necessary to drive the test structure at a certain level of excitation necessary to overcome friction between the damper and the guide rods on which it travels. We further hypothesize that the effectiveness of the damper depends on:

1. The velocity of the impact mass {X1 = Gap}
2. The mass multiplied by the square of the amplitude of the mode shape at the location of the damper, assuming a unit modal mass for the test structure {X2 = $m.\Phi^2_d$}, where Φ_d is the ordinate of the mode shape at the location of the damper.
3. The velocity of the test structure at the location of the damper {X3 = Im(FRF)/Freq}, where Im(FRF) is the imaginary part of the inertance frequency response function.

A multiple regression analysis based on the above hypothesis is presented in Table 4. From the statistical point view, the results indicate that the hypothesis is valid. For the first mode, the statistical analysis indicates that 88% of the variation in the damping ratio can be attributed to the independent variables {X1, X2, X3}.

CONCLUSIONS

Impact dampers can be effective in increasing the damping ratio of lightly damped flexible structures. The increase is damping is attributed to contribution of the impulse of the damper to the impulse and momentum equation of the test structure. Statistical analysis of the data obtained offers a degree of confidence in the multiple regression analysis performed in this study. It is believed that additional analysis of the data obtained can offer some insight in the behavior of the test structure. Additional experimental work on earth and in space is necessary

2. Using the multiple regression method to correlate the data.

INSTRUMENTATION

A 50-lb electromagnetic shaker was placed at a distance of 40.5 in from the center of the metal bracket. Eight accelerometers were placed approximately equally spaced along the length of the test structure (Halpin 1994). In addition a ninth accelerometer was used as a reference accelerometer at the point of excitation. A force transducer was attached to the test article at the location of the drive point to measure the input force. In addition, two force transducers were used to measure the impulsive force of the impact damper. Each steel billet was also instrumented with four accelerometers. Data acquisition was accomplished with the ZONIC 7000 48-channel data acquisition system.

TEST RESULTS AND ANALYSIS

Initial tests conducted on the test structure with impact occurring between the bare metal of the test article and the mass was recorded on magnetic tape and later digitized at rate of 10^6 samples per cycle. The results indicated that impact occurs at two impact per cycle at frequencies of 5, 10 and 20 Hz. It also showed that the impact force has the general form of an impulse with a duration of about 0.15 ms. A cushioning tape layer of about 0.1 in was used at both ends of the moving mass to widen the duration of the impact. This was necessary, first to eliminate the ringing effect of the noisy impact, and second to allow the capture of the impulse within the sampling limitations of the ZONIC 7000 system of 12,800 samples per second. The introduction of the cushioning tapes at both ends of the travelling mass resulted in impulse duration of about 1.5 ms, well within the limits of the ZONIC system.

To determine the damping ratios, twenty sine-sweep tests were performed for each of the first three natural frequencies for a total of sixty tests. Tables 1,2 and 3 show the results obtained. Six tests were conducted with the impact damper inactive, ie. the mass is restrained from moving. These tests form the baseline for comparison. It is noted that the intrinsic damping ratios ζ for the test structure are (0.336% and 0.31% for the first mode, 0.37% and 0.447% for the mode second, and 0.356% and 0.318% for third mode of vibration. Figure 2 shows the mode shapes for the test structure used in this investigation.

In performing the data acquisition, the maximum allowed block size of 4096 samples under triggered-continuous condition with 50% overlap was used. The Nyquist frequencies used were equal to 8, 16, 32 Hz for the first, second and third mode, respectively. The linear sweep rate recommended by Ewins (1983) was used to achieve accuracy.

The circle fit method was used to obtain an estimate of the damping ratio for each mode. Figure 2 shows a sample FRF and Coherence Functions. The results indicates that the parameters which was utilized in the acquisition and analysis of the data produced a good quality fit and hence an accurate estimate of the damping ratios. It is observed that the impact damper increases the damping ratio by as much as five folds.

Consider a mechanical system consisting of the test structure acted upon by the force supplied by the shaker and two impulses per cycle from the impact damper. The equation of impulse and momentum in the y-direction can be written covering a duration of one half cycle

TEST SETUP AND CONFIGURATION

The test structure consists of a 122 inch long brass tube, weighing 3.043 lb. It is simply supported at one end using a metal bracket and a universal join. A linear helical spring is used to support the test structure at a variable distance from the simple support end. The stiffness of the spring was determined in the laboratory to be equal to 15 lb/in. The impact damper consists of a PVC cylindrical tube housing two aluminum discs at the top and bottom (Figure 1). The bottom disc is secured to a ring which fits around the brass tube with 4 screws. A sliding aluminum disc is used to adjust the distance in which the impacting mass is allowed to travel. The reader is advised to refer to Halpin (1994) for additional information related to the test setup and configuration.

The test structure was installed on two 2,000 pound steel billets. The billets were placed on steel supports to raise their levels high enough for shaker installation. The test structure was installed to the long sides of the billets for testing in the horizontal direction (Y+). One billet was used to attach the metal bracket providing the simple support for the test structure. It remained stationary throughout the test. The spring support was attached to the other billet which was moved as needed to vary the span between the two supports.

The test configurations can be summarized by the following conditions:

a. Two spring support locations.
b. Three impact damper locations.
c. Three gaps.

TEST AND ANALYSIS PROCEDURES

To achieve the desired objectives of this investigation, the following test procedures were observed:

1. Check the quality of input forcing functions and driving point responses.
2. Confirm requested frequency range of 3 to 30 Hz is sufficient to obtain the desired flexural modes of test article.
3. Perform three tests at frequencies of 5, 10 and 20 Hz. Record the data on magnetic tape.
4. Digitize the data at a rate of one million per second and investigate the characteristics of the impact force at the top and bottom of the damper.
5. Determine optimum block size for data acquisition to have proper resolution for lowest test article flexural mode.
6. Find optimal input force level.
7. Document linearity and reciprocity of test structure (Halpin 1994).
8. Document repeatability of test structure from test to test.
9. Document modal characteristics of test structure.
10. Perform data quality review.
11. Perform sine sweeps of the test structure and record data.
12. Record time histories of all channels for selected tests an at selected frequencies.

Analysis of collected data was completed in two phases:

1. Using the SDRC-IDEAS software to obtain an estimate of the damping ratio using the circle fit method. The same software was used to obtain the mode shapes and natural frequencies.

INTRODUCTION

Attenuation of excessive vibration is an important design consideration in structural systems exposed to dynamic loads during their service. This can be achieved by actively or passively controlling the dynamic behavior of the structure. Methods of passive vibration control are quite often successful in achieving their objective at a relatively lower cost. They are often preferred over comparable active methods of vibration control if the structure will be used in a hostile operating environment, such as in outer space, where maintenance or monitoring is expensive.

An impact damper belongs to the category of passive vibration control devices. It consists of a free mass constrained to travel between two container walls. It produces substantial amount of damping in the structure it is attached to through momentum transfer.

Since the 1930's, numerous analytical and experimental studies have been conducted on SDOF and MDOF impact damped systems. However the experimental investigation of impact damped continuous systems was not undertaken until the early seventies.

In 1973 Masri (1973) performed experiments on a class of nonlinear dissipative cantilever and simply supported beams subjected to external sinusoidal excitation. He concluded that a heavier mass at an optimal clearance added to the systems damping. In addition the damper was more effective when located away from the node of the mode shape. In 1975 Roy and others (1975) produced similar results for fixed-fixed and simply supported beams subjected to base excitation.

Yousef and Akl (1987) performed a study of the free and forced vibration response of a vertical cantilever steel stack. Sliding and pendulum impact dampers were used in that investigation. A condition of two impacts per cycle was observed at frequencies higher than the fundamental frequency of the system. In addition it was concluded that an optimal clearance was associated with each impactor's mass.

Recently the second mode of a cantilever beam was studied by Chalmers and Semercigil (1992). They performed experiments with and without a rubber lining inside the boundaries of the damper's container. It was concluded that the rubber produced a characteristic behavior which was quite insensitive to the changes in particle clearance. Maximum damping was obtained by locating two dampers each at the locations of the largest amplitude.

OBJECTIVES

The objectives of this study are:

1. to investigate the effect of an impact damper on the dynamic behavior of a flexible structure where the effect of gravity is minimized.
2. to perform a number of tests on the flexible structure to study the effect of the mass, gap and location of the impact damper on the damping in the first three modes of vibration.
3. to analyze the data obtained in order to gain a better understanding of the behavior of the impact damper.

ABSTRACT

Impact dampers belong to the category of passive vibration devices used to attenuate the vibration of discrete and continuous systems. An Impact damper generally consists of a mass which is allowed to travel freely between two defined stops. Under the right conditions, the vibration of the structure to which the impact damper is attached will cause the mass of the impact damper to strike the structure. Previous analytical and experimental research work on the effect of impact dampers in attenuating the vibration of discrete and continuous systems have demonstrated their effectiveness. It has been shown in this study that impact dampers can increase the intrinsic damping of a lightly-damped flexible structure. The test structure consists of a slender flexible beam supported by a pin-type support at one end and supported by a linear helical flexible spring at another location. Sinusoidal excitation spanning the first three natural frequencies was applied in the horizontal plane. The orientation of the excitation and the test structure in the horizontal plane minimizes the effect of gravity on the behavior of the test structure. The excitation was applied using a linear sine sweep technique. The span of the test structure, the mass of the impact damper, the distance of travel, and the location of the impact damper along the span of the test structure were varied. The damping ratio are estimated for sixty test configurations. The results show that the impact damper significantly increases the damping ratio of the test structure. Statistical analysis of the results using the method of multiple linear regression indicates that a reasonable fit has been accomplished. It is concluded that additional experimental analysis of flexible structures in microgravity environment is needed in order to achieve a better understanding of the behavior of impact damper under conditions of microgravity. Numerical solution of the behavior of flexible structures equipped with impact dampers is also needed to predict stresses and deformations under operating conditions of microgravity in space applications.