Journal of Sound and Vibration (2002) **256**(1), 179–188
doi:10.1006/jsvi.2001.4177, available online at http://www.idealibrary.com on IDEAL®

EXPERIMENTS WITH TUNED ABSORBER—IMPACT DAMPER COMBINATION

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(*Received* 12 *July* 2000, *and in final form* 18 *October* 2001)

1. INTRODUCTION

Generically, a secondary system is under the influence of a more significant primary system. The disturbance is applied on the primary system. The dynamic response of the secondary system is the consequence of its unavoidable interaction with the larger primary system. Examples of such systems are encountered as light flexible components mounted on heavier machines, piping in buildings and in transport of flexible cargo. If a secondary system is poorly damped, large amplitude oscillations may need to be endured [1–5]. Results of an experimental investigation are presented here to suppress the dynamic response of a poorly damped light secondary system making use of two conventional passive controllers.

Tuned vibration absorbers have been used to control the excessive vibrations of the resonant systems [5, 6]. With these absorbers, control may be very effective in restraining vibration amplitudes at the tuning frequency. However, effectiveness of a conventional tuned absorber deteriorates rapidly as the frequency of oscillations differs from this critical tuning frequency or in case of transient disturbances. Addition of proper damping has been demonstrated to clearly improve the performance of tuned absorbers [5, 6]. However, this improvement comes with some practical difficulties as damping elements may require maintenance due to the rather extreme working conditions of their components. Hence, alternative means are desirable to improve the performance of the conventional tuned absorber.

An impact damper, another passive controller, is used in this study to enhance the performance of a virtually undamped tuned absorber. Impact damper is placed in the tuned absorber. The purpose is to create a momentum opposition and to provide energy dissipation through intentional collisions between the impact damper and the tuned absorber. Using these two passive controllers together has been introduced earlier [7, 8]. Therefore, the purpose here is to provide additional design data.

2. EXPERIMENTS

Majority of the experiments were performed by using a three-degree-of-freedom (3-d.o.f.) model shown in Figure 1(a). The second model shown in Figure 1(b) will be discussed in

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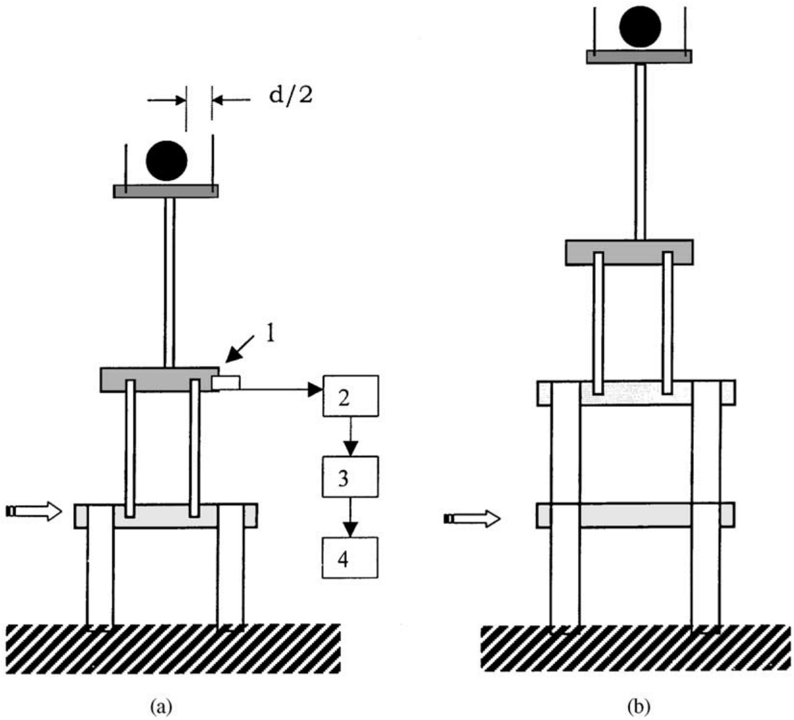

Figure 1. Schematic of the experimental setup with (a) S-d.o.f. primary system and (b) 2-d.o.f. primary system. 1: PCB 357, B01 accelerometer; 2: noise and vibration measurement systems amplifier; 3: data acquisition board; 4: personal computer.

section 4. The model consists of three mechanical oscillators. The primary structure was a rigid plate cantilevered from a fixed base using thin strips of aluminum which acted as the resilient elements as well as contributing to the equivalent mass. The secondary structure was another oscillator mounted on the primary structure using similar thin strips. The equivalent mass of the secondary structure was approximately ten times smaller than the primary's. Finally, the tuned absorber, a third oscillator, was mounted on the secondary structure whose mass was approximately 10 times smaller than that of the secondary structure. The impact damper was suspended into the cavity of the absorber by using it as the mass of a simple pendulum. Hence, the damper's motion was free of all external forces between contacts. The radial clearance, $d/2$, between the damper and the absorber is indicated in Figure 1(a).

The dynamic response of the secondary system depends upon the level of interaction between it and the primary system. The parameters of the system shown in Figure 1 were chosen so that the resulting combination would produce the largest response of the secondary system. Affecting the dynamic response of a light secondary system, are the two critical system parameters, namely the resonant frequencies of the primary system and the tuned absorber. Previous work has demonstrated that the largest response of the secondary system is produced when the two natural frequencies are the same [9]. Again, the most effective control is obtained when the absorber is tuned at the natural frequency of the secondary system. Hence, the natural frequencies of all three systems in Figure 1, are picked to be the same (within experimental variations) when they are tested as single degree-of-freedom oscillators one at a time. Table 1 summarizes the system parameters.

Table 1

System parameters

	m_{eq}(kg)	f_n(Hz)	ξ_{eq}
Primary sys.	8·080	7·9 ± 0·2	0·020 ± 0·005
Secondary	0·804	8·5 ± 0·2	0·010 ± 0·005
Absorber	0·080	9·0 ± 0·2	0·008 ± 0·005
Damper	0·020	—	—

Note: f_n and ξ_{eq} represent the natural frequency and equivalent viscous damping ratio. m_{eq} were obtained with an accuracy of 1 g.

Experimental procedure consisted of exciting the primary system by striking it with a pendulum (indicated with a block arrow) released at a predetermined distance. This simple excitation has been found to be quite reliable to produce a consistently repeatable transient disturbance. Care was taken to avoid multiple strikes. Response was measured by using an accelerometer, item 1 in Figure 1(a). The output of the accelerometer was first amplified and then digitized and transferred to a personal computer for further processing. The procedure described above was repeated with and without the presence of the impact damper for each different level of excitation. The coefficient of restitution of 0·34 ± 0·05, between neoprene rubber lined walls of the absorber and mild steel impact damper, has been reported earlier [7].

3. RESULTS

Histories of the acceleration at different levels of excitation are shown in Figures 2(a), 2(b) and 2(c), in the order of decreasing amplitude of excitation. Decreasing excitation may also be interpreted as increasing "effective" clearance of the impact damper as it will be discussed in relation to Figure 4. The three cases in Figure 2, are marked as points A, B and C in Figure 4. Each frame in Figure 2 has two traces: one corresponding to response of the secondary mass with (——), and one without (······) the impact damper.

The general trend in Figure 2 is the same for all three frames. Due to poor structural damping, transient oscillations take a long time to decay without impacts. With impacts, oscillation amplitudes are quite comparable to those without impacts at the start. However, once an effective pattern of collisions is established, decay rate is large, leaving quite minimal amplitudes for all three cases after the first 5 s. Furthermore, effectiveness of the control brought by the impact damper increases as the effective gap increases from Figure 2(a) to 2(c). The best case in Figure 2(c) produces a rapid decay within the first 2·5–3 s and quite insignificant remainder amplitudes after this period. Further demonstration of effectiveness will be from this particular case.

In Figure 3(a), the fast Fourier transformation of the acceleration of the secondary system is shown for the case in Figure 2(c), and for both with and without the impact damper. Without the impact damper, two large spectral peaks could be identified clearly at approximately 6·2 and 11·3Hz, and a smaller third peak at 9·3 Hz. Presence of collisions reduce the spectral amplitudes of the two large peaks quite significantly. The peak response at 9·3 Hz is not much affected due to not having a large enough amplitude even before the impact damper is introduced.

The same trend in Figure 3(a) may also be seen in Figure 3(b), this time for the amplitude distribution of the acceleration of the secondary system. Secondary system with the impact

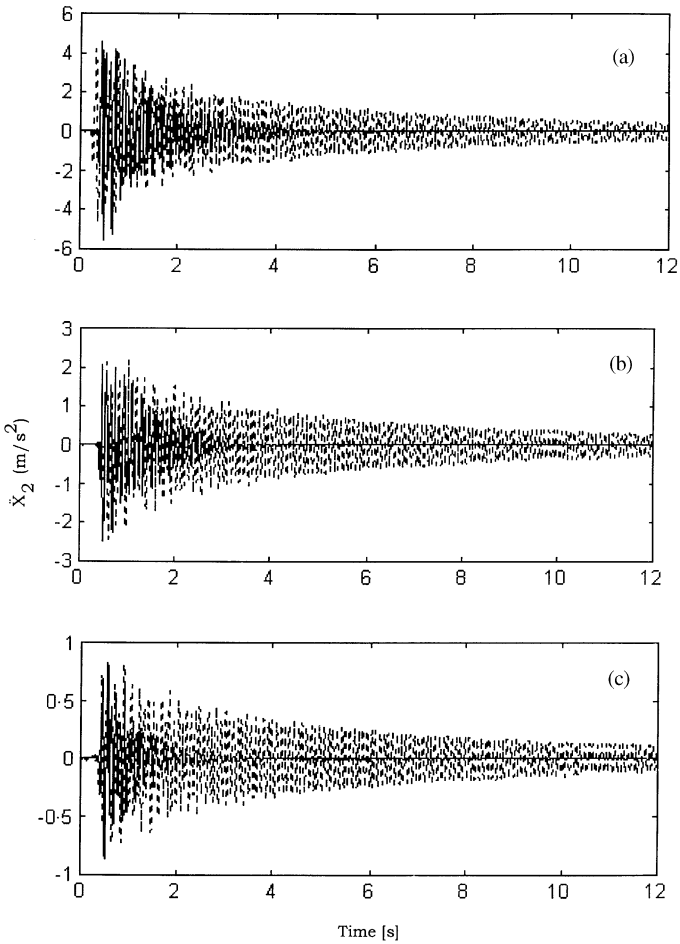

Figure 2. Acceleration histories of the secondary system at points A, B and C in Figure 4.

damper produces accelerations more clustered around zero acceleration with a much narrower spread of its peaks as compared to the case without the impact damper.

Results in Figure 4 correspond to the ratio of the settling times, t/t_0, recorded with (t) and without (t_0) the impact damper. Two settling time ratios are shown. A 10% settling time corresponded to the duration required for the acceleration of the secondary mass to decay to 10% of its peak amplitude without the impact damper. A 5% settling time measured the duration for half the acceleration amplitude of that of 10%. Whenever impacts are effective the ratio t/t_0 is smaller than unity where unity represents no change. The horizontal axis shows the non-dimensional clearance $d/(X_{20}/\omega_3^2)$ where d is the total clearance, X_{20} is the peak acceleration of the secondary system without impacts and ω_3 is the frequency of the largest spectral peak in the acceleration of the secondary system without impacts. The choice of ω_3 is rather arbitrary. The primary concern in picking this particular group of parameters in the denominator, is to non-dimensionalize the clearance with a displacement of the linear system without impacts. In the experiments, clearance was kept constant whereas the level of excitation was changed which produced a proportional change in X_{20}.

Results in Figure 4 suggest minimum reductions in the order of 75 and 80% (for the 10 and 5% settling times respectively) corresponding to the largest possible excitation

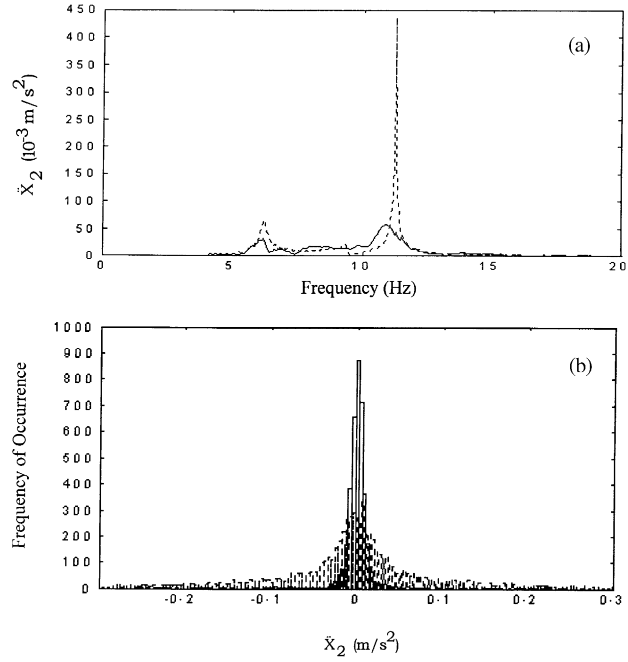

Figure 3. Acceleration (a) spectrum and (b) histogram of the secondary system with and without impacts at Level C in Figure 4.

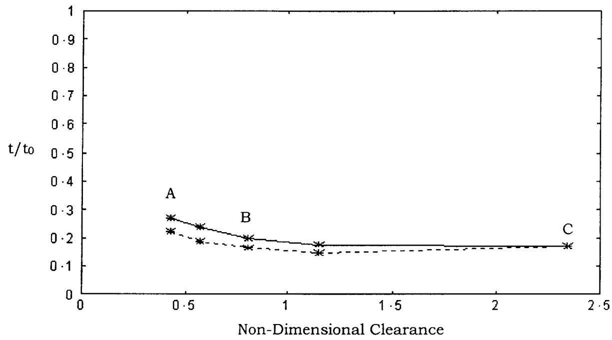

Figure 4. Variation of 5% (– – –) and 10% (——) settling time ratios with non-dimensional clearance.

amplitude which produced a non-dimensional clearance of approximately 0·4. All clearances larger than 0·75 produced at least 80% reduction of both settling times. Slope of the settling time ratios level off almost perfectly and remain leveled for a non-dimensional clearance of about 2·4. This value corresponded to the smallest level of excitation possible in the experiments. If it were possible to observe larger clearances, this settling time ratio would be expected to deteriorate toward larger values. Large clearances would produce too few collisions making them less effective for control. The histories presented earlier in Figure 2 are from the non-dimensional clearances marked A, B and C in this figure.

The reason for effectiveness of the impact damper may be related to the narrow band character of the uncontrolled system demonstrated in Figure 3(a). The linear system has its preferred frequencies marked clearly by the corresponding spectral peaks in this figure. Left

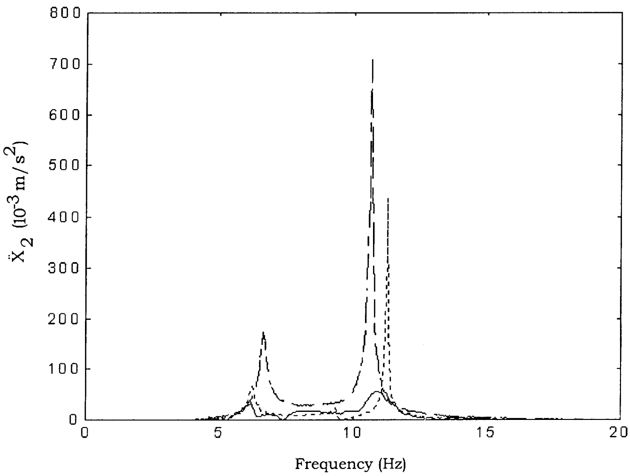

Figure 5. Acceleration spectra of the secondary system alone (– – –), with the tuned absorber (......) and with the impact damper (—) at Level C in Figure 4.

alone, the linear system naturally organizes itself to respond at these frequencies with large amplitudes due to light structural damping. The discontinuities introduced by the collision of the impact damper severely interferes with this natural process, and does not let the system develop nearly as large amplitudes. Resulting oscillations have a richer spectral distribution with the impact damper, as shown in Figure 3(a). For the most effective clearances of the impact damper, interference is maximized. For too small or too large clearances, however, control effectiveness is reduced due to changing collision patterns.

In Figure 5, the frequency spectra of the secondary system's acceleration are shown for three cases: when the secondary system alone is mounted on the primary system (– – –); secondary system with the tuned absorber (.....); and with the impact damper (——). The largest spectral amplitudes are for the 2-d.o.f. system. The spectral peaks are somewhat further separated with smaller amplitudes when the tuned absorber is added. However, the most dramatic change occurs with the addition of the impact damper where all spectral amplitudes are significantly reduced due to impacts.

Figure 6 is similar to Figure 5, except now the acceleration of the primary system is presented. For these measurements, the accelerometer was mounted on the first level of the structure in Figure 1(a). Four different configurations are given in this figure: S-d.o.f. primary system alone (–·–·); 2-d.o.f. primary and the secondary systems together (– – –); 3-d.o.f. system with the tuned absorber (.....); and with the impact damper (——). Table 2 summarizes the settling times of these four cases.

Addition of the secondary system on the primary system exaggerates its response to its largest amplitudes among the four cases shown here. For this 2-d.o.f. case, both 5 and 10% settling times of the primary system are more than three times longer than the S-d.o.f. primary system alone. Tuned absorber is somewhat helpful to alleviate the problem caused by the secondary system. However, the primary system is still worse off with the absorber than the case when it was alone. When the impact damper is included, both settling times improve significantly. When compared to the case with the tuned absorber, these improvements are in the order of 75–80% which are quite comparable to those of the secondary system shown in Figure 4.

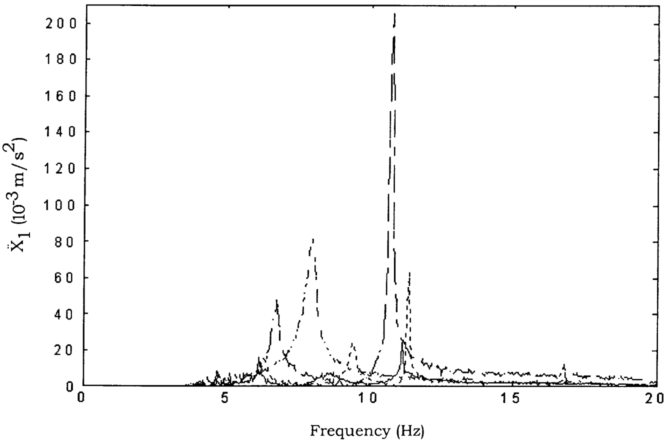

Figure 6. Acceleration spectra of the primary system alone (–·–·), with the secondary system (– – –); with the tuned absorber (......) and with the impact damper (——) at Level C in Figure 4.

TABLE 2

Settling times of the primary system

	t/t_0 – 5%	t/t_0 – 10%
Primary sys. alone	1	1
Prim. + secondary	3·87	3·37
Prim. + sec. + absorber	2·63	1·50
Prim. + sec. + absorber + impact damper	0·53	0·37

4. RESULTS WITH A LARGER PRIMARY SYSTEM

In the first part, the natural frequencies of the primary and the secondary systems were chosen to be coincident in order to induce the largest dynamic response from the secondary system. Then the absorber was tuned to the same frequency to ensure its strong interaction with the secondary system. In this section, additional data is offered after doubling the size of the primary system by adding another level to it. The two structures used for the preceding sections and for this section are shown in Figures 1(a) and 1(b) respectively. Parameters of the second level of the primary system are identical to those given in Table 1.

Having a two-degree-of-freedom (2-d.o.f.) primary structure presents the opportunity to investigate the effects of different tuning frequencies for the absorber. With the addition of the second level in the primary system, the combination of primary–secondary structures now has three natural frequencies at approximately 4·9, 9·3 and 16·5 Hz. It is interesting to note that the second resonance corresponds to the tuning frequency of the vibration absorber in the earlier study. Hence, with the doubled primary system, tuning the absorber at any of these three frequencies should ensure strong interaction between the absorber and the secondary system.

Results in Figure 7 give the ratio of the 10% settling times of the secondary system with and without the impact damper for different values of the non-dimensional clearance, $d/(X_{20}/\omega_3^2)$. Here, d is the total clearance, X_{20} is the peak value of the acceleration of the secondary system without impacts, and ω_3 is the frequency of the third spectral peak of the

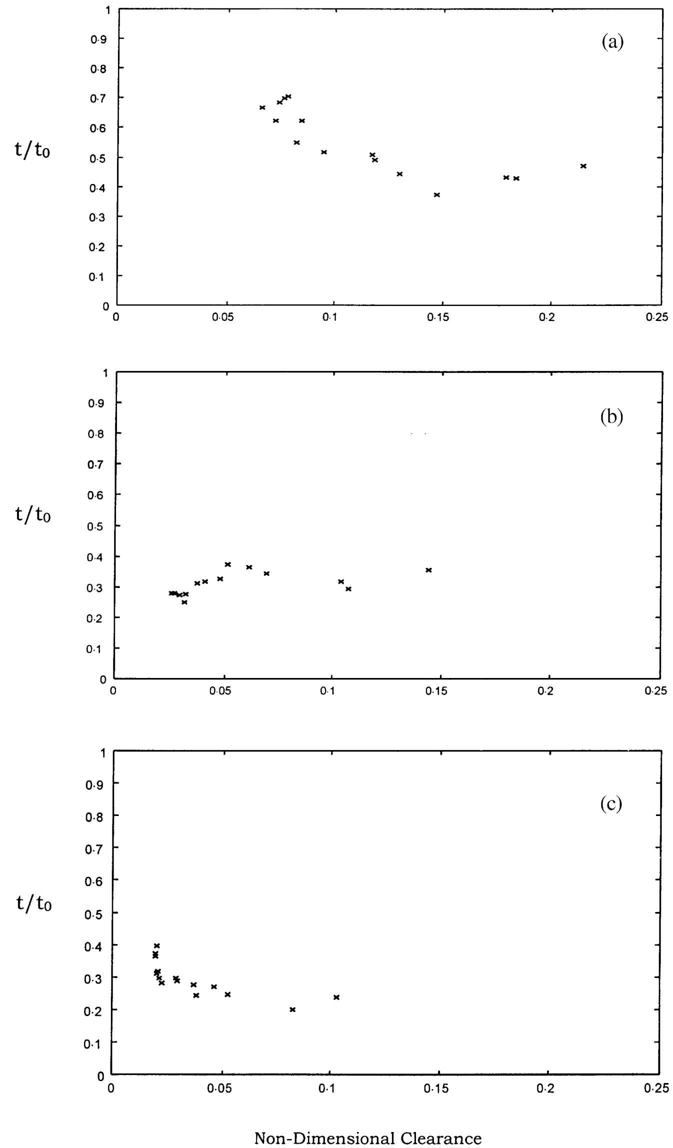

Figure 7. Variation of the settling time ratios for (a) the low, (b) original and (c) high tuning frequencies.

acceleration of the secondary system without impacts. As briefly discussed earlier, the choice of ω_3 is quite arbitrary and any change in the level of disturbance causes a proportional change in the denominator of the non-dimensional clearance.

In all three frames of Figure 7, the smallest and the largest values of non-dimensional clearances represent the largest and the smallest practically possible disturbances in the experiments respectively. Settling time ratios in Figure 7(a) correspond to the case where the absorber is tuned at the first natural frequency at 4·9 Hz. In Figure 7(b), the same absorber is used as in the earlier study with a natural frequency of 9·3 Hz. This is the second natural frequency of the 3-d.o.f. structure without the absorber. In Figure 7(c), the absorber is tuned at 16·5 Hz, the third natural frequency. The tuning of the absorber was changed by varying the length of the thin aluminum strips in Figure 1.

For the low tuning frequency, control is relatively ineffective producing reductions in the order of 60% for clearances of 0·13–0·20. Outside these clearances, control effect deteriorates quite significantly. For the original tuning frequency in Figure 3(b), control is significantly more effective. Reductions in the settling time range between 65 and 75% for virtually all clearances. For the high tuning frequency, control effect is improved further. Clearances larger than 0·03, ensure a reduction of 75–80%. However, there is rapid deterioration for smaller clearances to reductions as low as 60%.

For brevity, acceleration histories and frequency spectra similar to those in Figures 2 and 3 are not presented here. They may be obtained from the first author. One trend which is worth mentioning here is that of the high tuning frequency. Tuning the absorber at the third resonance produces a rapid decay of accelerations initially, taking out the spectral contributions of the high frequencies quickly. But this rapid decay is followed by a slow decaying period which detrimentally affects the overall performance of the impact damper. Hence, addition of a second impact damper whose clearance is designed to target the low frequency contributions may significantly improve the performance of this case from that presented in Figure 7(c).

5. CONCLUSIONS

Experimental observations are presented to demonstrate the effect of adding an impact damper to a tuned vibration absorber. Measurements were taken for the response of a light secondary system in transient oscillations. The suggested impact damper is a simple addition for those cases where a tuned absorber already exists.

When an impact damper, with a mass 25% of that of the absorber and a coefficient of restitution of about 0·35, is added to the tuned absorber the settling times of the combined system improve by the order of 80%. This improvement is relatively insensitive to different non-dimensional clearances. In addition, impact damper also seems to control the response of the primary system quite effectively. Considering that the impact damper is significantly smaller than the primary system, this last trend may be of some practical importance.

Additional experiments conducted using a two-level primary system suggest that approximately 70% improvement of control is still attainable at the same tuning frequency used in the first part. In addition, the level of attenuation is virtually independent of the level of external disturbance. Low and high tuning frequencies are not as practical due to the sensitivity of the resulting control on the level of disturbance. However, high tuning frequency may hold more promise than the low frequency. Although no results are presented in this paper, observations clearly show that the control effect of the impacts on the primary system is lost with the two-level primary system.

ACKNOWLEDGMENTS

Support from The Danish Research Council (STVF) to finance the travel and accommodation expenses of the second author is gratefully acknowledged.

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