Journal of Sound and Vibration (1997) **208**(5), 869–875

PARTIAL COVERAGE AIR FILM DAMPING OF CANTILEVER PLATES

D. I. G. Jones
D/Tech Systems, 2343 Lantern Hill Road, Dayton, OH 45459, U.S.A.

and

T. Lewis and C. Michael
Damping Technologies Inc., 1329 Stanley Avenue, Suite 2, Dayton, OH 45404, U.S.A.

(*Received 14 May 1997, and in final form 21 July 1997*)

1. INTRODUCTION

Currently recognized means of increasing the modal damping of structures include dry friction and viscoelastic layers or coatings. Friction damping has been quite widely investigated and many technical papers and patents are found in the literature, but applications have not been very widespread because of several factors, such as the non-linear nature of the phenomena involved and practical matters such as fretting of the sliding interfaces. Viscoelastic layers have become better understood in recent years but some applications have not been successful because of factors such as temperature ranges required in the proposed application and complications involved in allowing for the effects of temperature and frequency in analytical predictions. Few other means of introducing damping appear to be widely recognized.

Recently, another, hitherto not widely recognized, mechanism for introducing damping into vibrating structures has been examined, namely *air film damping*. A conceptual view of a simple air film damper is illustrated in Figure 1. The air gap is created by an appropriate manufacturing process. The damping mechanism appears to be the relative transverse motion between the base plate and the added platelet, which occurs as the system resonates in any specific mode, and which cyclically forces the entrained air along the extremely thin gap between the plates, as illustrated in Figure 2. The flow is probably viscous and results in high internal cyclic pressure differences, acting in opposite phase to the modal velocity, much as occurs with squeeze film dampers in rotating turbomachinery. The difference is that no rotation is involved here, and no special means are needed to maintain the film.

References [1] to [6] are representative of prior studies of the phenomenon of fluid film damping. Ungar [1], Ungar and Carbonell [2] and Maidanik [3] concluded that the main mechanism of damping in built-up, riveted, skin–stringer structures typical of the type used in aircraft construction is pumping of the air film between the overlapping sheets of metal at the structure boundaries. Fox and Whitton [4] conclude that the air film created when a flat sheet of metal is placed against a flat surface of a structural element provides the main mechanism of damping. Chow and Pinnington [5, 6] compare a passive air film damper, created when two flat sheets of elastic material are placed parallel to each other with a thin air film between, to a squeeze-film damper. They show that relative transverse motion between the parallel plates drives the air or fluid film along the gap and this flow, assumed to be viscous, is the source of the damping.

0022–460X/97/500869 + 07 $25.00/0/sv971236

2. SUGGESTED PHYSICAL MECHANISM

A relatively complicated system such as that illustrated in Figures 1 and 2 requires finite element analysis to fully quantify the effect of the partial coverage air film on the damping in each mode of vibration. The work of Fox and Whitton [4] and of Chow and Pinnington [5, 6] may provide some guidance in performing such an analysis; however, we do not know of any work which shows that this has been accomplished. A first step in the process of developing a valid and accurate model would be to use discrete or finite element analysis to predict the normal modes of vibration of the system, including the air film gap itself, *in vacuo*. With the damping mechanism absent, the modes may be determined by relatively conventional means. This structure is not two-dimensional, as is often assumed for a cantilever beam or plate, even when performing finite element analysis, but must be assumed to be three-dimensional. Therefore, the mode shape vector at each point within the three-dimensional body of the metallic (or solid) parts of the system will exist at all points and may be slightly different over the "upper" and "lower" surfaces. It certainly will be different along the "upper" surface of the air gap, relative to the "lower" surface, as illustrated in Figure 3. It is this difference which drives the air in the air film back and forth parallel to the upper and lower restraining surfaces. When this general concept of mode shape is used, it may be possible to calculate the oscillating flow velocities for a chosen mode amplitude, and hence the oscillating pressures over these restraining surfaces caused by the air flow. These pressures will be in-phase with the modal velocity, and will do work as the surfaces move cyclically relative to each other, thereby producing the damping.

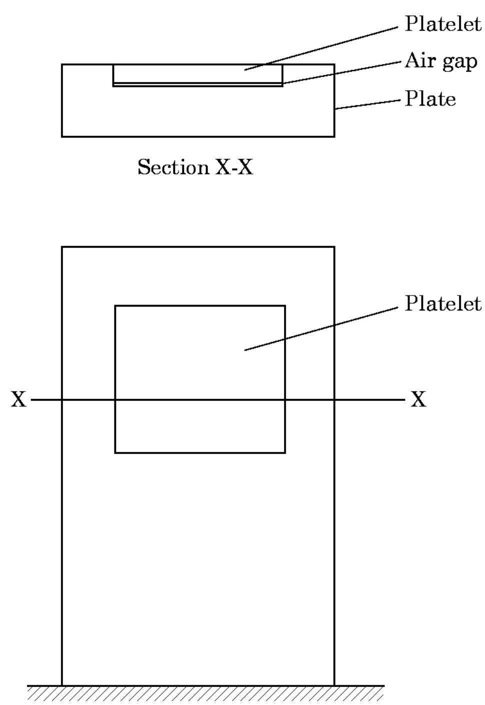

Figure 1. Conceptual view of air film damper.

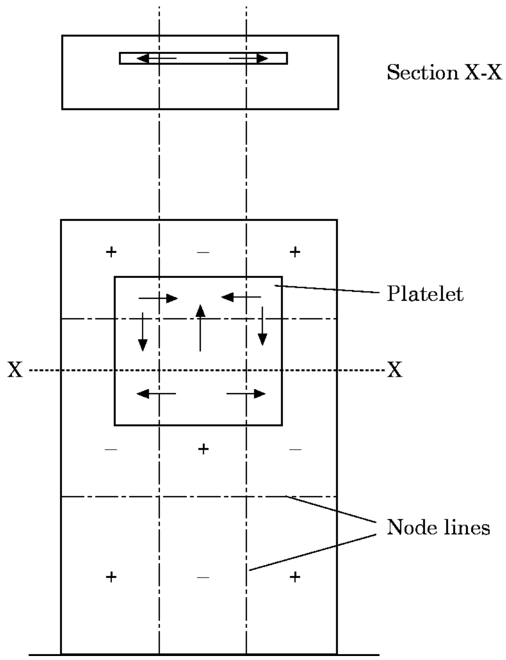

Figure 2. Hypothetical air flow directions.

3. PRELIMINARY TEST RESULTS

A limited number of laboratory vibration tests have been recently conducted, using the test system illustrated in Figure 4, to measure the dynamic response and modal damping of a flat cantilever plate having an additional flat platelet applied to the surface, both unbonded to the surface and bonded around the edges. The specimen configuration that was tested is illustrated in Figure 5. The cantilever plate was 4·826 mm thick, 101·6 mm wide and 203·2 mm long, made of an aluminum alloy (probably 7075-T6 but this could not be determined with certainty). The platelet was 66.0 mm wide by 98·4 mm long by 1·02 mm thick, and was placed about 12·7 mm away from the outer edge of the cantilever

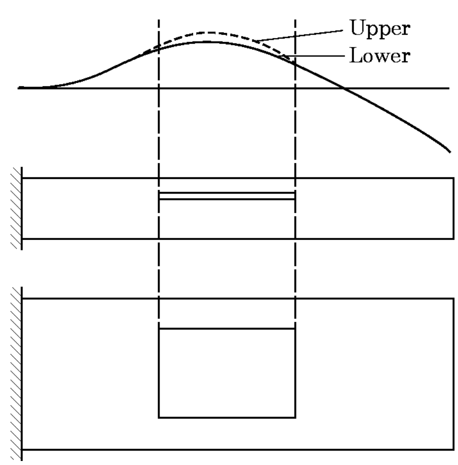

Figure 3. Hypothetical mode shape of split cantilever.

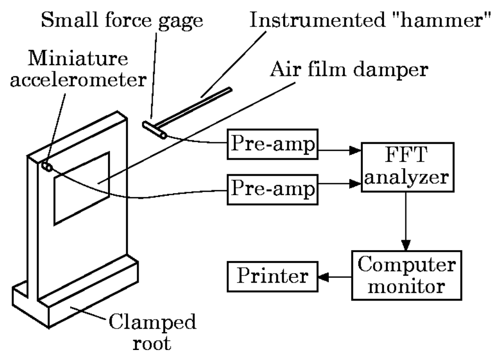

Figure 4. Vibration test system.

plate. The air gap was nominally zero, but slight non-flatness of the plate surface probably created a gap of mean thickness about 0·025 to 0·05 mm. This is somewhat below the optimum gap size noted by Fox and Whitton [4], which was about 0·1 mm, for a full coverage air film damping system. It was found that such a partial coverage damper significantly increased the damping in several modes over a wide frequency range. Figure 6 illustrates some typical measured plots of acceleration per unit force (g/N) versus frequency (Hz) for the damped and undamped system, driven at the outer corner, with the response measured at the same point in each case, and with the platelet resting on the cantilever plate near the tip (Configuration A) or near the base (Configuration B). Figure 7 shows the same plots for pickup and excitation at the outer center. Figure 8 shows the response when the platelet was placed near the tip of the cantilever plate (Configuration A) and bonded around the edges, using a quick drying glue, with excitation and pickup at the outer corner. Figure 9 shows the same plot for excitation and pickup at the outer center. It is seen that considerable amounts of damping can be introduced, especially in some of the higher order modes. At the present time, experiments to vary the important parameters of air film damping have not been completed, and the above results, while interesting, must be considered preliminary. Similar results were obtained for a plate of width 127·0 mm. The results show that appreciable damping is achieved in some modes but not in others,

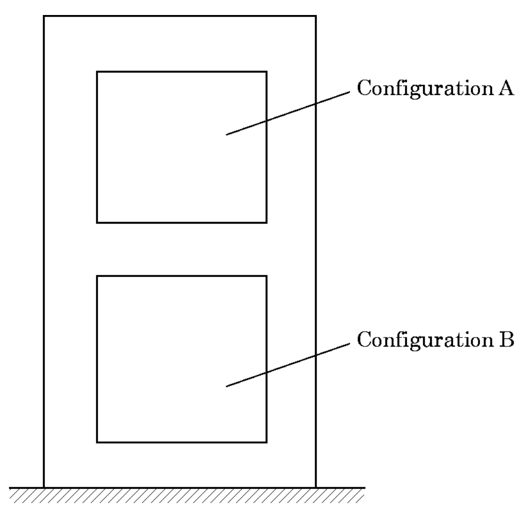

Figure 5. Test article configurations.

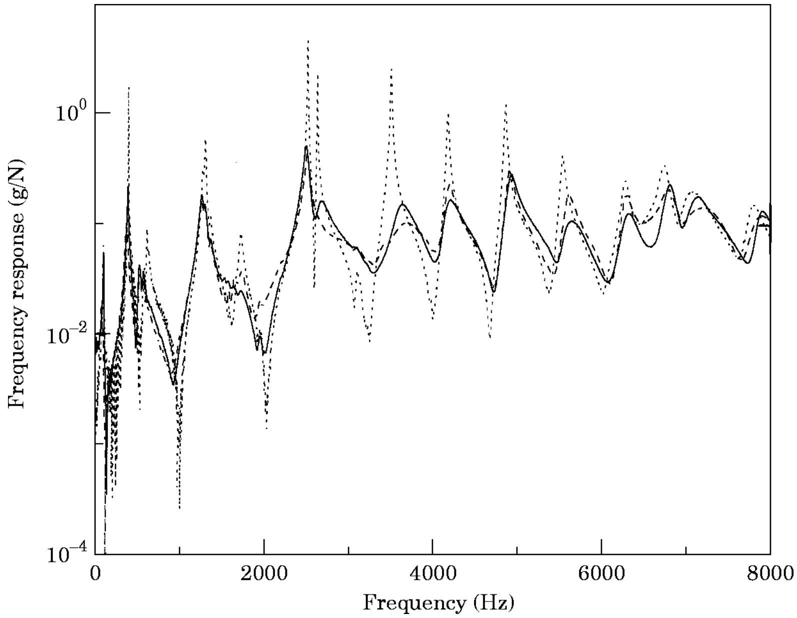

Figure 6. Measured response of damped and undamped systems with unbonded platelet and with pickup and impact at outer corner: · · · · , baseline; ----, with air film (configuration A); ——, with air film (configuration B).

and this is most probably associated with the position of the platelet with respect to the nodal lines for the various modes. For example, when the node lines are close to the platelet boundaries, one may expect relatively high damping, but when the node lines cross the platelet, the damping can be expected to diminish.

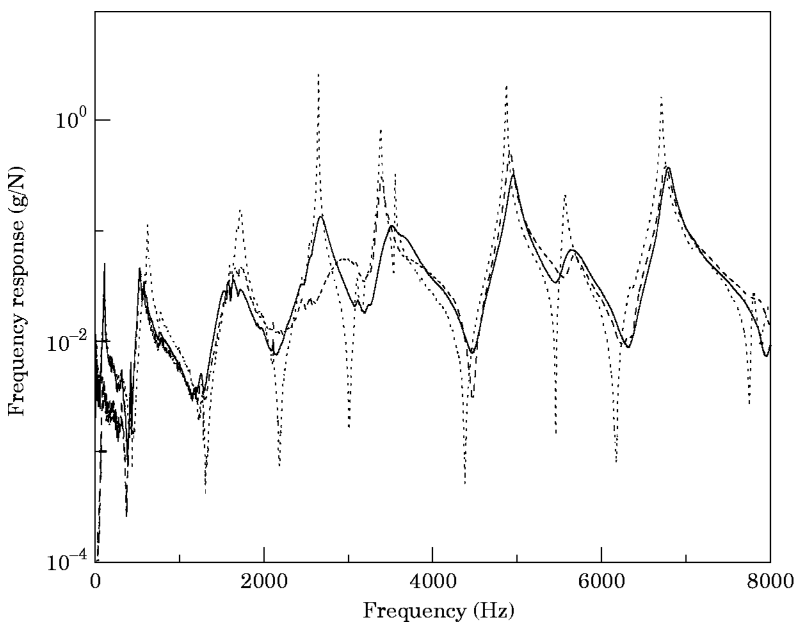

Figure 7. Measured response of damped and undamped systems with unbonded platelet and with pickup and impact at outer center: · · · · , baseline; ----, with air film (configuration A); ——, with air film (configuration B).

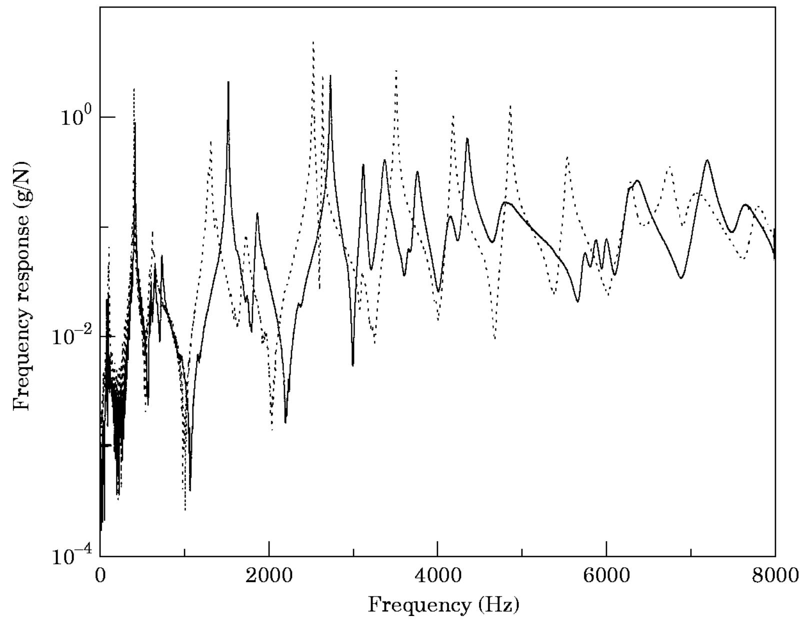

Figure 8. Measured response of damped and undamped systems with bonded platelet and with pickup and impact at outer corner: · · · ·, baseline; ——, with air film.

4. SUGGESTED ADDITIONAL RESEARCH

The investigations reported in this brief article are indicative of what may be achievable by application of partial-coverage air film damping as a practical approach for vibration control. Much research remains to be accomplished to fully understand the precise details

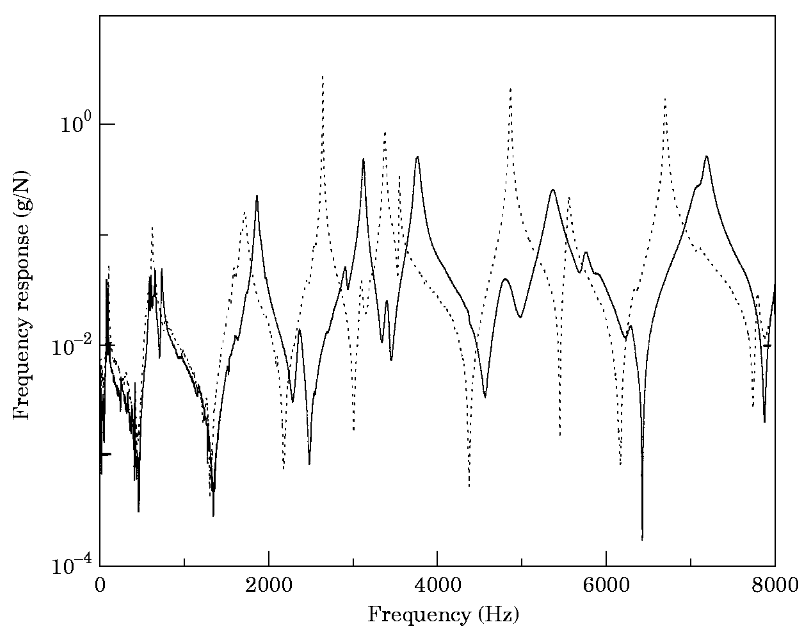

Figure 9. Measured response of damped and undamped systems with bonded platelet and with pickup and impact at outer center: · · · ·, baseline; ——, with air film.

of the damping mechanism for a partial coverage treatment, including theoretical and experimental investigations of the effects of structure and platelet dimensions and boundary conditions, platelet position relative to each modal pattern, air gap thickness, air viscosity and so forth. The work of Fox and Whitton [4], in particular, provides considerable insight concerning the effect of some of these parameters. The authors intend to pursue some of these aspects in future work, but the purpose of this particular note is limited to bringing the approach back to the attention of the technical community.

5. CONCLUSIONS

The phenomenon of air film damping has been investigated experimentally for a more complicated structure than has been studied hitherto, namely a cantilever plate with partial coverage, and it has been shown that significant amounts of added passive modal damping can be introduced. The proposed damping mechanism is discussed, and possibilities for analytical studies are outlined. The mechanism of air or fluid film damping is considered to have many application possibilities in the future, especially for situations where frictional, viscoelastic or other damping mechanisms are not usable.

REFERENCES

1. E. E. UNGAR 1964 *Shock and Vibration Bulletin* **33**(4), 189–199. Shock and Vibration Information Center, Vibrational energy losses at joints in metal structures.
2. E. E. UNGAR and J. R. CARBONELL 1966 *American Institute of Aeronautics and Astronautics Journal* **4**, 1385–1390. On panel vibration damping due to structural joints.
3. G. MAIDANIK 1966 *Journal of the Acoustical Society of America* **40**, 1064–1072. Energy dissipation associated with gas pumping in structural joints.
4. M. J. H. FOX and P. N. WHITTON 1980 *Journal of Sound and Vibration* **73**, 279–295. The damping of structural vibration by thin gas films.
5. L. C. CHOW and R. J. PINNINGTON 1987 *Journal of Sound and Vibration* **118**, 123–139. Practical industrial method of increasing structural damping in machinery, I: Squeeze-film damping with air.
6. L. C. CHOW and R. J. PENNINGTON 1989 *Journal of Sound and Vibration* **128**, 333–347. Practical industrial method of increasing structural damping in machinery, II: squeeze-film damping with liquids.