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THE USE OF PASSIVE ABSORPTION TO PREVENT ACOUSTIC RESONANCES IN FLOW SYSTEMS

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ABSTRACT

Acoustic absorbers were fitted to three different test rigs in which flow induced acoustic resonances normally reached levels of over 150 dB to establish whether or not passive absorption could be used to reduce the amplitudes of the resonances. Each absorber comprised a perforated plate facing sheet over sections of honeycomb material with solid backing. The rigs were (a) a rectangular passage spanned by a single plate, (b) an annular passage with a cascade of 15 flat plates and (c) a single stage axial-flow compressor rig. In every case the resonances were completely eliminated when the honeycomb depth and the position of the absorber were optimised.

1. INTRODUCTION

There are many situations in which fluid flow induces some form of oscillation which may be mechanical, acoustic or, in many cases, a combination of both. Apart from musical instruments, such oscillations are generally highly undesirable and engineers go to considerable trouble to eliminate these vibrations. It is naturally desirable that unwanted sources of sound and/or vibration should be eliminated at the design stage, to do this the possible sources of excitation must be identified and all the frequencies and amplitude distributions of possible resonant modes must be calculated. It is not surprising that there are many cases in which problems are encountered during prototype testing or plant commissioning and, when this happens, it is necessary to investigate the source of excitation and the nature of the resonances involved before deciding on appropriate remedial measures.

In some cases the modifications required are relatively simple. For example, when flow over plates or vanes results in vortex shedding which excites acoustic or mechanical resonances, the problem can often be cured by modification of the trailing edge profile (for example providing an asymmetrical chamfered section) to alter the vortex shedding characteristics of the plate and so eliminate the source.

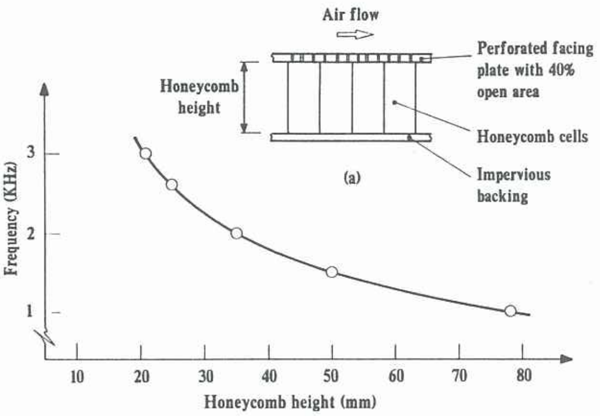
In other cases solutions are not so easy, particularly where modification of the components involved in the excitation of resonances would prejudice the performance of a machine. For example, acoustic resonances have been found to be a source of blade vibration in axial-flow compressors. Compressor blades are designed to achieve a specified amount of turning with the minimum possible loss and modification of the blade profiles to eliminate a source of excitation, while possibly being necessary in the interests of machine integrity, may impair the performance. It has also been found that, in a compressor, vortices shed from one row of blades interact with other blades downstream and form a second source of acoustic or vibrational energy. In circumstances such as these, it is logical to examine other possible remedial measures.

Flow induced oscillation results from an unstable feedback loop and it is well known that the phase and gain of the loop are the factors influencing which oscillations will build up and which will decay. Elimination of an unwanted oscillation can be achieved by adjustment of the characteristics of the loop and one obvious possibility is the provision of some form of energy absorption or damping. When dealing with flow induced vibration problems, the role played by the acoustic characteristics of the enclosure surrounding vibrating components is often underestimated although, in the authors' experience, it is often a dominating factor. This has been shown to be particularly true in systems involving flow through enclosed passages which naturally includes axial flow compressors. In systems of this sort, it is possible to provide passive acoustic absorption in the passage walls without intrusion into the flow or modification of other components, this offers a possible way of modifying the feed back loop and therefore solving many flow induced vibration problems without prejudice to machine performance.

This paper presents the conclusions of a series of experiments involving the application of passive acoustic absorption in three different flow systems and shows how effective it can be in eliminating flow induced sound and vibration.

2. THE ACOUSTIC ABSORBER

The actual design of an acoustic absorber depends on the application involved but, to establish the potential of absorbers for eliminating flow induced acoustic resonances, it was decided to use a type of absorber commonly used with aero-engines and similar applications known as a "laminar liner". The construction of the liner is illustrated in Figure 1(a). The holes in the perforated facing plate and the cells of the honeycomb act as a series of resonant cavities which function as tuned absorbers. Figure 1(b) is a theoretical curve showing the relation between the depth of the honeycomb and the resonant frequency of the liner, that is the frequency at which the maximum absorption coefficient would normally be expected.



(b) Variation of tuned frequency with height of honeycomb

FIGURE 1 THE ACOUSTIC ABSORBER

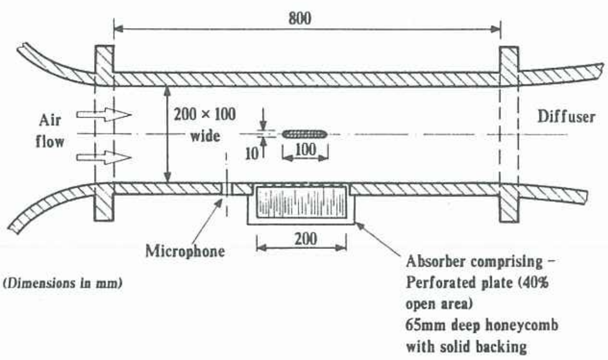


FIGURE 2 THE WIND TUNNEL WORKING SECTION WITH FLAT PLATE AND ACOUSTIC ABSORBER

3. TEST CONFIGURATIONS AND RESULTS

3.1 A Rectangular Passage Spanned By A Flat Plate

Figure 2 illustrates the test facility which was a small wind tunnel designed for an investigation of the vortex shedding characteristics of plates and interaction of vortices with a downstream plate. Initially, the acoustic absorber shown in the Figure was replaced by a solid plug and, with a single plate the first transverse mode (commonly referred to as the β mode) (Parker 1967) was excited at velocities from 28 m/s up to the maximum tunnel speed of 40 m/s giving a peak amplitude at the microphone position of over 150 dB. When the absorber was in place no evidence of a resonance could be found at any velocity.

3.2 An Annular Flow Passage with a Cascade of Flat Plates

Figure 3 shows the general arrangement of this rig which was also designed for investigation of vortex shedding and interaction phenomena. The acoustic absorber was a circular ring which could be inserted at various axial positions providing absorption in the outer annulus wall. The facing plate of the absorber had 40% open area and was 25 mm in axial length, backed by honeycomb 41 mm deep. For each configuration the flow velocity was varied over the available range and, when a resonance was found, readings of acoustic pressure were taken at a series of positions along the length of the outer annulus wall. The highest amplitudes recorded for each resonance at the various absorber positions are listed in Table 1. The absorber position is defined as the distance of the centre of the absorber downstream of the mid chord of the plates. The modes are described as α modes when there is a node in the mid chord plane and β modes when there is an antinode and the associated figures define the number of circumferential lobes.

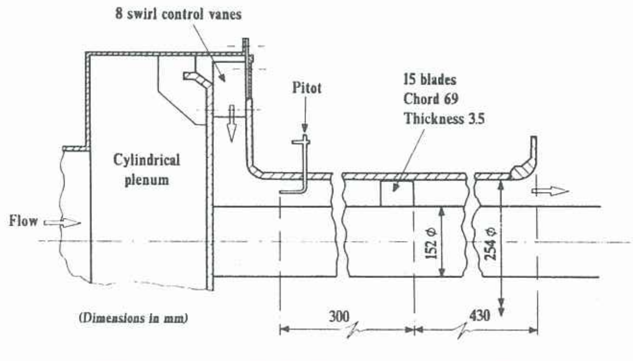


FIGURE 3 HALF SECTION OF THE ANNULAR TEST RIG

With the hard wall (no absorber) several modes were excited with amplitudes of up to 158 dB but, when the absorber was placed over the leading or trailing edges of the cascade (-34.5 and +34.5 mm positions), the principal modes were eliminated and only the $\alpha 7$ mode was found, at relatively low levels of 130 dB. and 128 dB respectively. As the absorber was moved away from the cascade (either upstream or downstream) its effectiveness decreased, at ± 79 mm, all the modes found with the hard wall, with the exception of modes $\beta 3$ and $\beta 4$ (the lowest modes), could be identified though, in every case, at reduced amplitude. At 123.5 mm, mode $\beta 4$ was found at a level 10 dB down and, at 168 mm, mode $\beta 3$ was found at a level 27 dB down, moving the absorber further away resulted in gradual increases in amplitude until, at 344 mm, the attenuation of every mode was less than 10 dB.

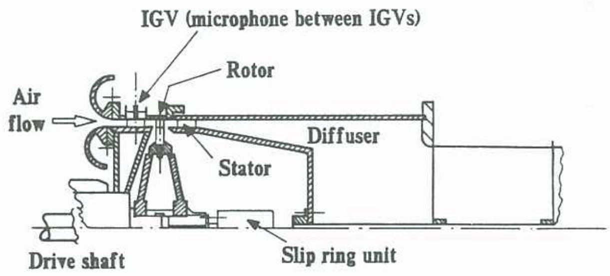
Table 1. The effect of the position of the absorber on the peak amplitudes recorded in the annular test rig

Distance of absorber centre line downstream of blade mid-chord	Maximum SPL, dB. (* indicates mode not found)					
	Mode β3 1350 Hz	Mode β4 1630 Hz	Mode β5 1800 Hz	Mode β6 1960 Hz	Mode α6 3150 Hz	Mode α7 3460 Hz
No absorber	150	158	156	148	147	*
- 79	*	*	144	150	*	146
- 34.5	*	*	*	*	*	130
+ 34.5	*	*	*	*	*	128
+ 79	*	*	144	145	136	137
+ 123.5	*	148	146	149	145	141
+ 168	123	147	147	150	146	142
+ 212	139	149	148	151	146	139
+ 257	138	148	148	150	148	141
+ 301	140	148	148	150	148	141
+ 345	144	149	150	151	148	141

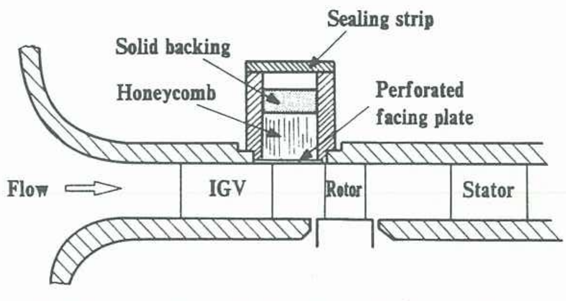
3.3 Single-Stage Compressor Rig

Figure 4(a) shows the main features of this test rig which, together with the data recording and processing system, was described in some detail by Parker and Stoneman (1985 and 1987). Figure 4(b) shows the details of the absorber which was fitted in six segments each covering an arc of 43° with solid spacers in between, thereby covering 72% of the circumference. The perforated facing plates were replaced by solid plates for the hard wall case. Figure 5 presents the results with and without absorber in the form of "Z plots" (Parker and Stoneman 1985 and 1987). Figure 5(a) shows that, with the hard wall, there are several separate resonances in the frequency range from 1.5 to 2.6 kHz. As the speed increases through each resonance the frequency increases slightly and then jumps to the next resonance. At higher speeds, further peaks appear at frequencies up to a little over 4 kHz. Figure 5(b) shows that, with the absorber, the separate resonances can no longer be identified and the peaks merge together to give the impression of a continuous line. The amplitude of the largest peak in the spectrum recorded at each speed is plotted in Figure 6 for both conditions. The progressive increase in amplitude to nearly 160 dB between 35 and 40 revs/sec with the hard wall is completely eliminated by the absorber and, at all speeds, the peaks are below 115 dB.

This result was obtained after a systematic optimisation process. Initially the absorber was placed about 17 mm downstream of the final position. Variation of the depth of the honeycomb showed the optimum to be about 22 mm but the resonances were not eliminated although the maximum level was reduced by almost 30 dB.



(a) Half-section of rig with hard wall



(b) Schematic of acoustic absorber in the final position

FIGURE 4 COMPRESSOR TEST RIG

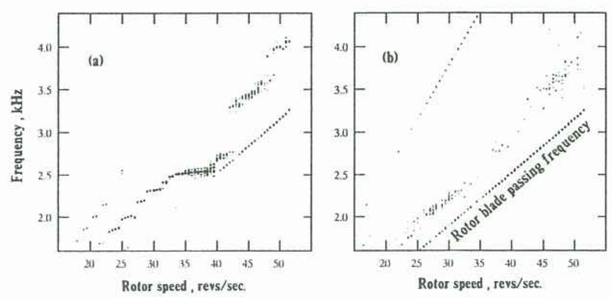


FIGURE 5 COMPRESSOR RIG : RESULTS
WITH (a) Hard wall
(b) Absorbent liner

The results obtained with the annular rig showed that the position of the absorber was fairly critical and confirmed that it was most effective when it was at or near the position of maximum amplitude of the resonances excited with the hard wall. Traverses of the rotating rig showed that the maximum amplitudes occurred between the IGVs and the rig was therefore re-designed to bring the absorber as far forward as was structurally possible and a series of tests was carried out to optimise the honeycomb depth. This was found to be 26 mm and the results shown if Figures 5 and 6 were obtained with the optimised absorber in the forward position (as shown in Figure 4b).

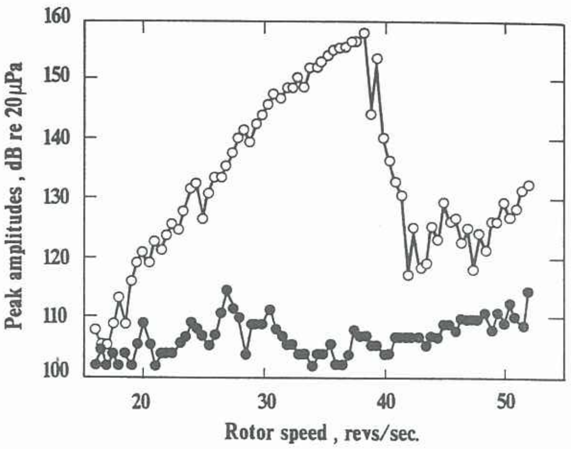


FIGURE 6 COMPRESSOR RIG : THE LARGEST PEAKS RECORDED AT EACH SPEED
—o— with hard wall
—●— with acoustic absorber with 26mm honeycomb depth

4. DISCUSSION AND CONCLUSIONS

The results presented in this paper were selected to illustrate the fact that flow induced vibrations of the nature encountered in the three test facilities discussed can be effectively eliminated by fitting passive absorbers in the walls of the flow ducts. The only design requirement is that the absorber should be located in the areas of maximum amplitude associated with the resonant modes and a reasonable area of absorption should be provided in those regions. The frequency of a "tuned" type of absorber should be in the band of frequencies to be absorbed and generally as close as possible to that of the most powerful resonance. For example, the optimum depth of honeycomb found for the rotating rig was 26 mm which, from the curve in Figure 1, corresponds to 2.5 kHz, close to the frequencies of the largest recorded amplitudes and Figure 5(b) shows that, not only were the peaks in the region of 2.5 kHz completely eliminated, but at all frequencies from 1.5 to 4 kHz the peaks were much reduced.

5. ACKNOWLEDGEMENTS

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