ACOUSTIC PROPERTIES OF PLATES IN TANDEM

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Introduction

Acoustic resonances in turbomachinery have been known for a number of years [1]. Latterly, in collaboration with Rolls-Royce, Parker and Stoneman have been investigating the nature of acoustic resonances in a single stage, axial flow compressor test rig to identify the parameters which are a major influence on the generation of circumferentially propagating acoustic waves. The collaboration arose from the identification by Rolls-Royce of unacceptably high rotor blade stresses in a research compressor for the RB-211 aero-engine.

This paper briefly restates the main conclusions of the above work [2 to 5] and reports the results of wind tunnel experiments conducted at the Division of Energy Technology, CSIRO Melbourne to investigate the acoustic properties of tandem plate configurations, of varying axial spacing, which was intended to model the Inlet Guide Vane and rotor blade geometry of the turbomachine.

Turbomachinery Test Rig and Results, University College of Swansea

Figure 1 is a half sectional elevation of the single stage axial flow compressor test rig at the Dept. of Mechanical Engineering, University College Swansea. Figure 2 is a partially assembled view of the test rig showing the major components.

Figure 3 shows the blade geometries for the two main phases of the work, i.e. vortex shedding from (a) 33 zero stagger, slab sided, rounded trailing edged IGV's and (b) 33 and 66 zero stagger, airfoil sectioned IGV's subjected to incident flow produced by an upstream row of pre-swirl vanes. The results from the slab sided test are representative of all the results obtained and the following is therefore limited to this geometry.

Figure 4 shows the frequency/flow velocity relationship for a stationary microphone upstream of and between two IGV's. These results are for an IGV/rotor axial spacing of 33 mm which, non-dimensionalised in terms of the thickness of the vortex shedding IGV's (5 mm), is a space to thickness ratio of 6.6 . It can be seen that the resonances manifest themselves as a series of locally approximately constant frequency lines over small ranges of velocity, with the mode number varying from 7 to 16, propagating sometimes with and sometimes against the rotor direction of rotation. The range of Strouhal numbers over which resonances are generated being 0.229 to 0.292.

Figure 5 shows the relationship for a space to thickness ratio of 1.04 where the long series of resonances has been replaced by just two resonances, modes 16 and 15 which are frequency locked over a very large velocity range, corresponding to a range of Strouhal numbers from 0.252 to 0.335.

When the space to thickness ratio was an intermediate value of 2.68 (Figure 6) two series of resonances were generated. As the flow velocity was decreased the modes changed in a series of steps from 16 to 12 whereupon the mode number jumped to 16 again, decreasing to mode 8 with decreasing velocity. The Strouhal number ranges corresponding to these series of resonances were

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0.212 to 0.260 for the high velocity series and 0.316 to 0.363 for the low velocity series. It was to provide an explanantion of the means by which two very different resonances can be excited at similar flow velocities as a result of changing the axial blade spacing that an experimental programme was undertaken in collaboration with CSIRO.

Tandem Plates in a Wind Tunnel, CSIRO

To simulate the geometry of a vortex shedding IGV positioned upstream of a rotor row, two plates were mounted on the axial centreline of a wind tunnel in a tandem configuration (Figure 7) such that the axial spacing between the two plates could be varied from zero to 200 mm (space to thickness ratios of 0 to 25). The vortex shedding and the acoustic field were monitored by a probe microphone located in the wake region of the upstream plate and a 3582A Hewlett Packard spectrum analyser.

Figure 8 shows the now familiar result for a single plate in a wind tunnel where over a limited speed range, the vortex shedding excites a duct acoustic resonance which in turn becomes locked to the acoustic resonant frequency. Figure 9 shows the results obtained with a second plate installed 30 mm downstream of the first plate (space to thickness ratio of 3.75) where it can be seen that when the vortex shedding frequency is near to the acoustic resonant frequency the vortex shedding is locked to it, as with a single plate. Over and above this, there were a number of other flow velocities at which a resonance was excited which were found to occur when the vortex shedding frequency was an integer sub-multiple of the acoustic resonant frequency, e.g. 8:9, 7:5, 5:2 and 2:1. The particular value of the integer sub-multiple was a function of the axial spacing between the plates. Work is in hand at CSIRO to mathematically model the transfer of energy from integer sub-multiple vortex shedding to an acoustic field and initial results indicate that the essential character of the phenomenon can be predicted.

When the flow velocity was set at a constant 22 m/s (such that in the absence of the second plate a strong acoustic resonance would have been generated) then varying the plate spacing from zero to 200 mm caused the peak sound pressure level to vary as shown in Figure 11. The peak sound pressure levels obtained corresponded approximately to those found in the absence of a second plate. However, at the intermediate positions the acoustic resonances were effectively destroyed being some 30 to 50 dB lower than the peaks.

Discussion

It is not known at this time whether the low velocity series of resonances in the compressor (Figure 6) is integer sub-multiple vortex shedding. The frequency step from mode 12 to mode 16 is an integer ratio of 5:6 but no evidence has yet been found to indicate that the IGV vortex shedding is not locked to the acoustic field. This may not be necessary since in the turbomachine the SPL's are much higher and may lock the shedding once the resonance is established.

An explanation of the variation of SPL with plate spacing is that energy can be transferred from the flow to the acoustic field when there is a net positive imbalance in the summation of the individual values of the vector triple product of the Howe formula associated with each vortex as it traverses the acoustic field [6,7]. The position of the second plate inlfuences the total number of generating vortices in relation to the total number of absorbing vortices since when a vortex traverses the second plate its net effect on the acoustic field must be zero since the convection and acoustic velocities (as terms in the vector triple product) are parallel.

The IGV/rotor spacings at which the turbomachinery results were obtained are indicated in Figure 11 and it is significant that the spacing at which the double series of resonances was obtained (Figure 6) corresponds to one of the spacings at which the tandem plate resonance was effectively destroyed. This may indicate that the phase relationship of the vortex arrival at the rotor favoured the excitation of the sub-multiple resonances at the low flow velocities over those at the higher flow velocities.

Conclusions

1. The results of turbomachinery experiments and those from tandem plates in a wind tunnel show encouraging similarities suggesting that modelling the acoustic properties of turbomachines with stationary plates in a wind tunnel has a contribution to make in expediting investigations.

2. Two plates in tandem in a wind tunnel can excite the an acoustic resonance when vortex shedding frequency is an integer sub-multiple of the acoustic resonant frequency.

3. The presence of a second plate in tandem with a vortex shedding plate in a wind tunnel can destroy the presence of the acoustic resonance normally associated with vortex shedding at the acoustic resonant frequency.

References

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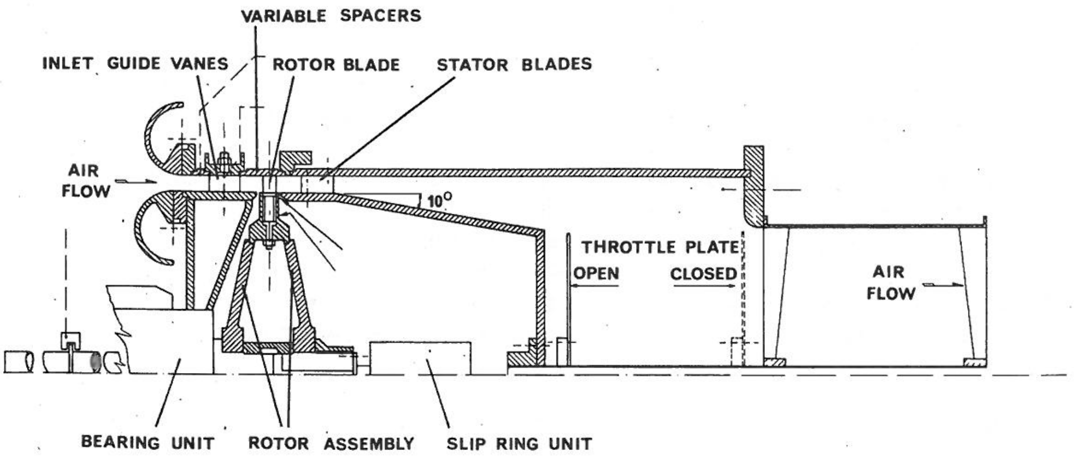

Fig. 1 Half sectional elevation of axial flow compressor test rig.

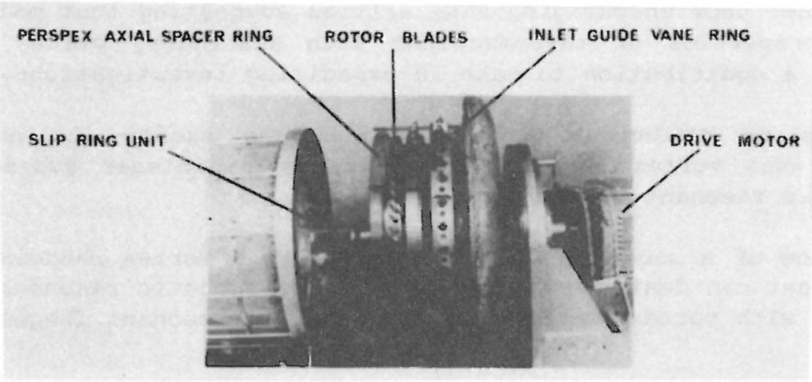

Fig. 2 Partially assembled view of test rig.

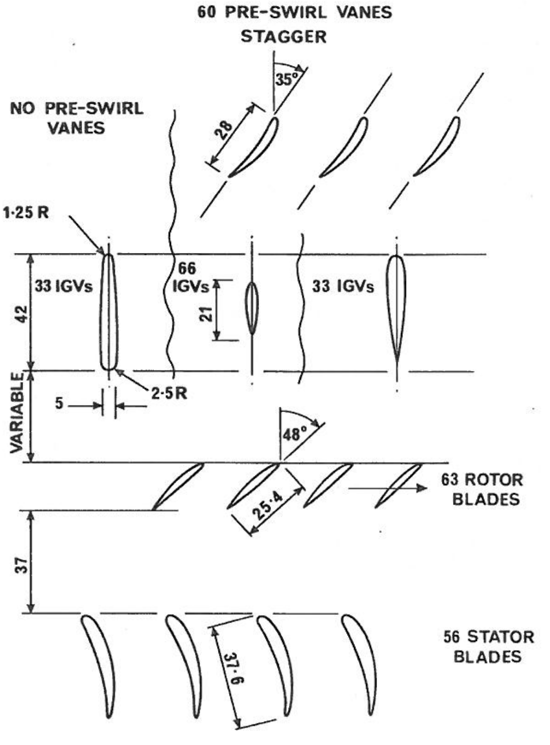

Fig. 3 Compressor blade geometries.

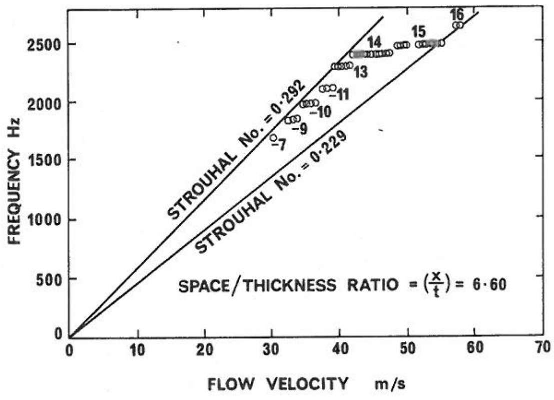

Fig. 4 Frequency/velocity relationship
x/t = 6.6

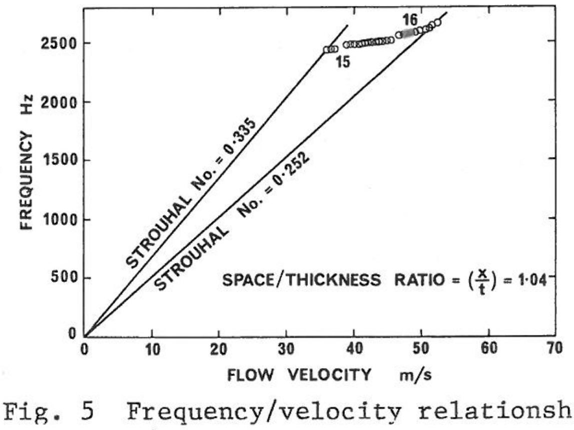

Fig. 5 Frequency/velocity relationship
x/t = 1.04

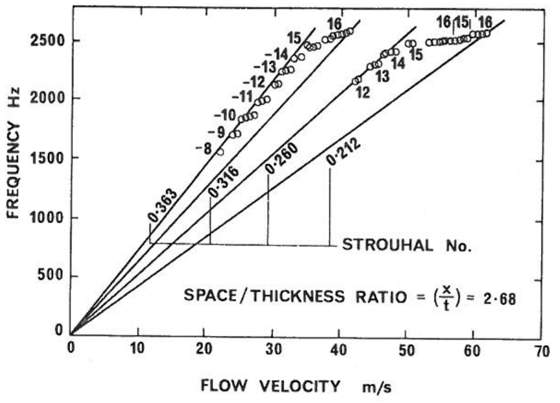

Fig. 6 Frequency/velocity relationship x/t = 2.68

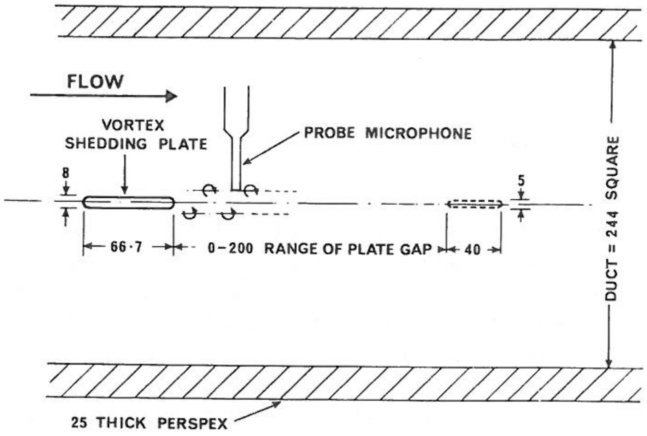

Fig. 7 Variable spacing tandem plates in wind tunnel.

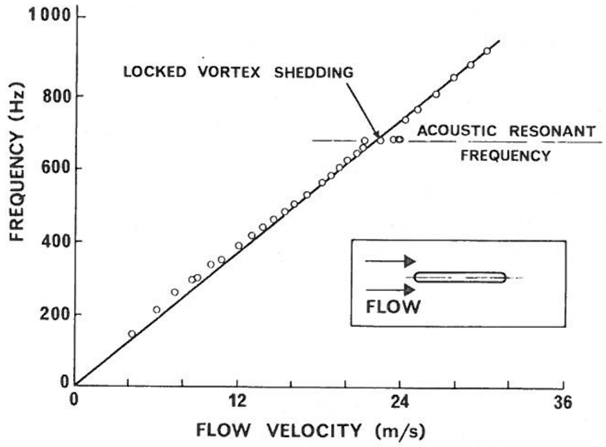

Fig. 8 Frequency/velocity relationship - single plate.

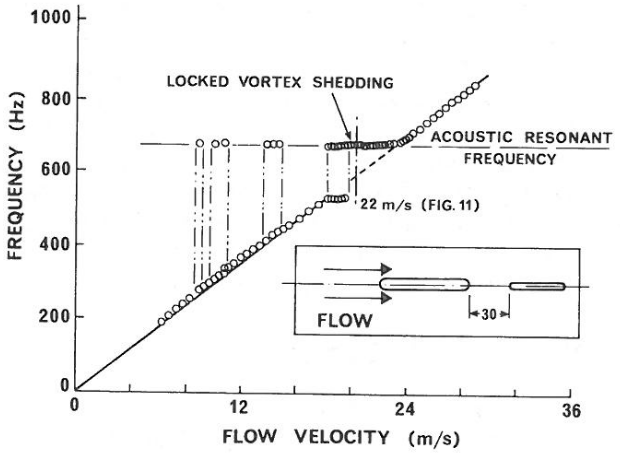

Fig. 9 Frequency/velocity relationship - single plate.

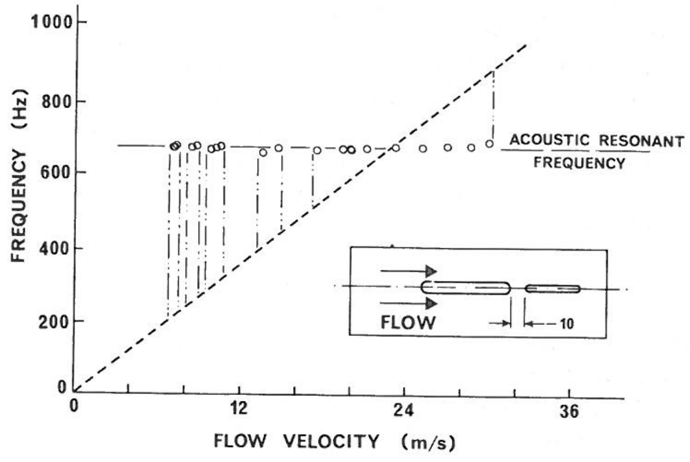

Fig. 10 Frequency/velocity relationship - single plate.

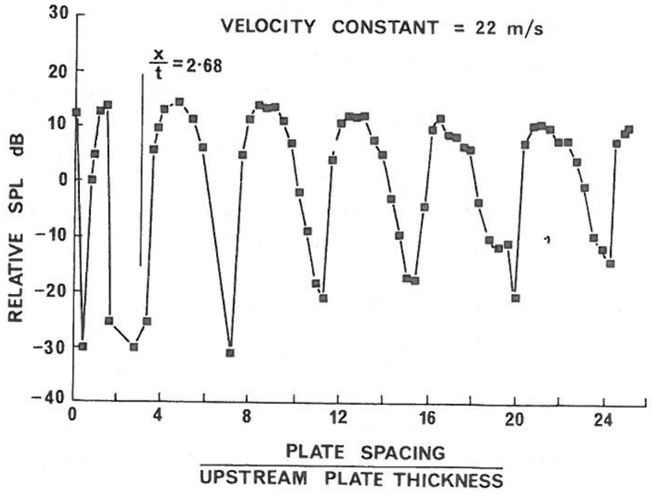

Fig. 11 Variation of peak SPL with plate axial spacing at constant flow velocity.