# Engineering Notes

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## Attenuation of Vortex Shedding Noise of Circular Cylinders Behind a Ducted Rotor

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### Introduction

IN recent years aircraft have been fitted with high bypass ratio engines having large mass flow compressors. Hence the compressor noise is likely to predominate. In an attempt to design a low-noise compressor, a ducted rotor-stator stage was designed[1] with the rotor made up of conventional blades and the stator comprised of circulation controlled circular cylinders. The circulation on each stator blade was produced by blowing air through tangential slots. Cheeseman[2] found from two-dimensional tests on a circular cylinder that lift coefficients up to 16 can be produced by blowing air through tangential slots. Therefore, the number of "lifting cylinders" required to replace a conventional bladed stator in an axial flow compressor is small. Thus, such a rotor-stator combination may be expected to be less noisy compared to a conventional rotor-stator combination.

Further, it has been found[1] that the lift fluctuations on a circulation controlled cylinder are fairly insensitive to the velocity fluctuations in the freestream flow. Pease[3] found that a helicopter rotor made up of a circulation controlled cylinder is 10 dB quieter than a conventional rotor for the same duty.

The use of lifting cylinders as a set of low-noise stator blades behind a conventional bladed rotor raises the obvious question of discrete vortex shedding and the associated noise one encounters in the case of plain cylinders in a range of Reynolds numbers. This note presents some experimental observations made on the vortex shedding noise of slotted circular cylinders behind a ducted rotor and the effect of slot blowing on the vortex shedding noise. Noise measurements were confined to the front arc only.

### Experimental Setup

The experimental setup consisted of a low-speed (pressure ratio of 1.06) axial flow compressor (Fig. 1) in an anechoic room. A set of six slotted brass cylinders was located behind the conventional bladed rotor. Each cylinder had three slots on the surface connected to a common axial passage through which compressed air was supplied to the slots. The width of each slot was 0.762 mm. The other details of the cylinder are given in Fig. 2. The outlet of each slot was so designed that the air leaves tangential to the surface, thus ensuring a good boundary layer control system. All the axial passages of the cylinders were connected by means of plastic tubings to a common plenum chamber. This plenum chamber was in turn connected to a portable air supply compressor installed outside the anechoic room. The cylinders were held by a split ring (S in Fig. 1) at the root and an outer ring at the tip. Provision was made for setting the slots to any desired position with respect to the mean flow direction.

In order to study the effect of rotor-cylinder separation, the outer ring, cylinders, and split ring could be mounted in any axial position A, B, or C. The split ring S and the other two spacer rings D and E were bolted to the bearing housing H. A wedge-type multihead pitot tube was used to measure the flow velocities and angles. The axial flow compressor was run by a 180 hp dc motor installed outside the anechoic room. The shaft speed was measured by a fibre optic revolution sensor coupled to a digital counter. Noise measurements were made by a 1.27 cm B & K microphone in a horizontal plane containing the axis of the compressor. The microphone output was recorded on a tape recorder.

#### Experiments and Analysis of Results

The ducted rotor was run at 3000 rpm. The flow velocity and direction were measured behind the rotor using the wedge probe. The cylinders were mounted in the axial position B such that the slot 1 was at 90 deg to the mean flow direction. The rotor was run again at 3000 rpm. Front arc noise was measured at 2.54 m from the rotor plane under blown and unblown conditions of the cylinders. Blowing through the slots was sufficient to produce enough lift on each cylinder so that the flow behind the cylinders was truly axial. This was confirmed by a flow survey downstream of the cylinders. This test sequence was repeated with the cylinders in position C. Figures 3–5 show typical noise spectra measured on the axis along with the bandwidth of the analysis.

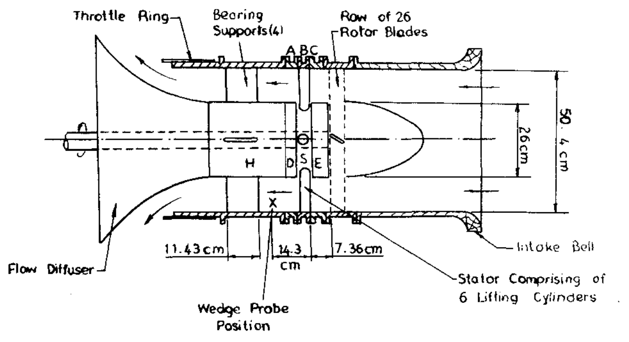


**Fig. 1 Illustrative view of the ducted fan.**

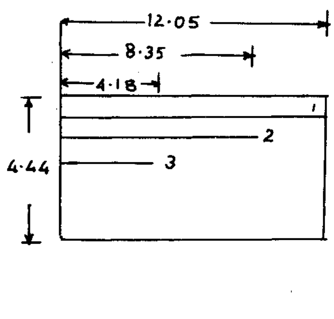


**Fig. 2 Slotted cylinder details. (All dimensions are in centimeters.)**

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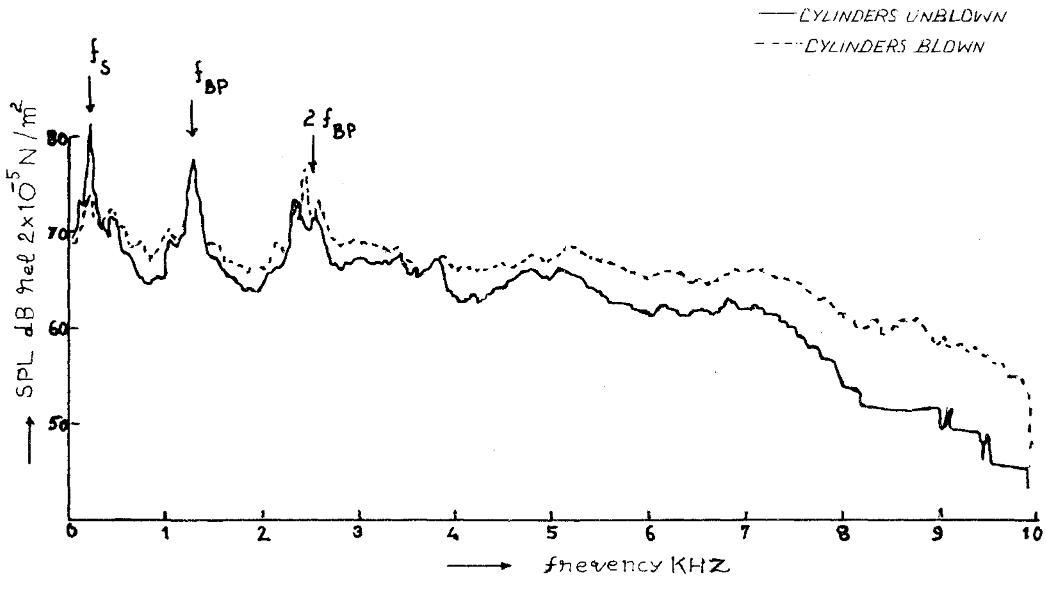


**Fig. 3 Rotor cylinder combination noise spectrum measured on axis. 30 Hz band width analysis. Cylinders in position B.**

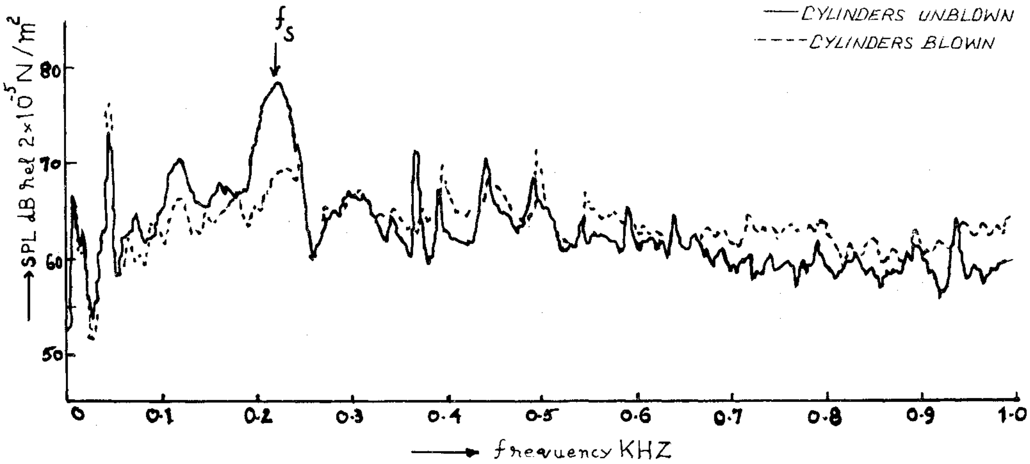


**Fig. 4 Rotor-cylinder combination noise spectrum measured on axis. 3 Hz bandwidth analysis. Cylinders in position B.**

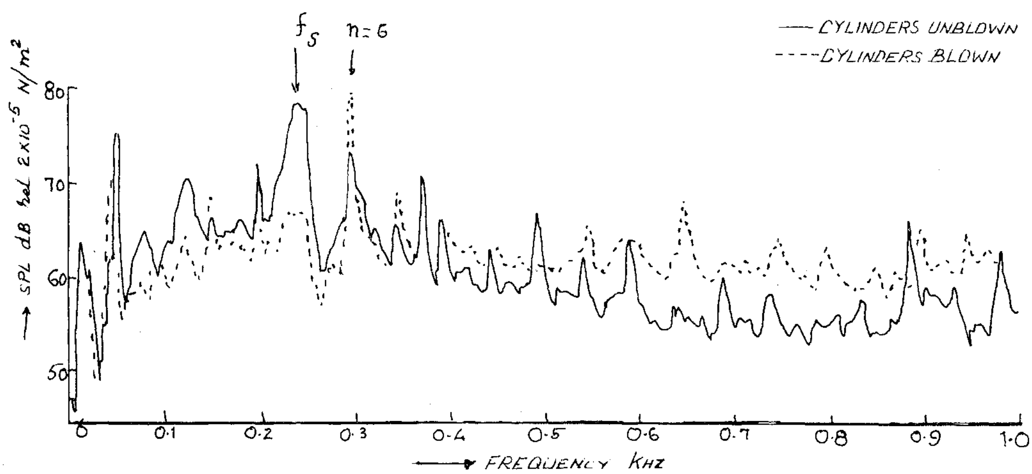


**Fig. 5 Rotor-cylinder combination noise spectrum measured on axis. 3 Hz bandwidth analysis. Cylinders in position C.**

## Discussion of Results

Figure 3 shows noise spectra of the rotor-cylinder combination measured on the forward axis with the blown and unblown cylinders in position B. In addition to the familiar blade passing frequency ($f_{BP}$) and harmonies, one can recognize the Strouhal number related vortex shedding frequency ($f_s$) noise. Slot blowing has strongly attenuated the vortex shedding noise. Figure 4 shows the details of the vortex shedding noise part of the spectrum. Figure 5 shows the effect of slot blowing on the vortex shedding noise with the cylinders in the axial position C. The effect of slot blowing is again the same, that is, to attenuate the vortex shedding noise. $n=6$ is the sixth harmonic of the shaft speed.

The phenomenon of periodic vortex shedding by plain circular cylinders and the associated force fluctuations on the cylinder and the radiated dipole noise centered at the vortex shedding frequency is well known. In order to suppress the wake-induced vibrations of tall chimneys, helical strakes have been successfully used.[4] A concentric perforated shroud enclosing a plain cylinder has also been found to be effective in suppressing the wake-induced vibrations.[5]

In order to understand the mechanism of suppression of the periodic vortices of a circular cylinder, Naumann et al.[6] have conducted a series of experiments. On a plain cylinder, periodic vortex shedding was found to be accompanied by a straight separation line. However, they observed that a crisscross arrangement of wires along the span of a cylinder always inhibited discrete vortex shedding and the separation line was found to be irregular. The suppression of the periodic vortex shedding was therefore attributed to the introduction of strong three-dimensional disturbances into the wake through the crisscross arrangement of wires forcing a nonlinear separation line on the cylinder.

In the present experimental arrangement, the circulation controlled cylinders are functioning as a set of stator blades by producing lift varying along the span of each cylinder. The spanwise variation of circulation on each cylinder was achieved by staggered slots as in Fig. 2. This arrangement of slots, when blown, forces the flow to separate at different angles, i.e., the separation line will be irregular. Therefore, the observed attenuation of the discrete vortex shedding noise with slot blowing is most probably due to the same mechanism of introducing strong three-dimensional disturbances into the wake, breaking up the coherent shedding of discrete vortices.

## Conclusions

The present investigation demonstrates clearly that the vortex shedding and the associated noise of circular cylinders behind a ducted rotor can be attenuated by producing differential circulation along the span of the cylinders. The attenuation is probably accompanied by an irregular separation line on the axis of each cylinder. The slotted circular cylinders used as a set of stator blades in an axial flow compressor will be almost free from the discrete vortex shedding noise.

## Acknowledgment

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## References

[1] Keshavan, N.R., "Noise Studies on Circulation Controlled Cylinders in an Axial Flow Compressor," Ph. D. thesis, University of Southampton, U.K., 1977.

[2] Cheeseman, I.C., "Circulation Control and Its Application to Stopped Rotor Aircraft," *The Aeronautical Journal,* Vol. 72, July 1968, pp. 635-646.

[3] Pease, C.B., "Preliminary Noise Measurements on a Circulation Controlled Rotor," NGTE, Powered Lift and Noise Note No. 53, 1969.

[4] Scruton, C., "Note on a Device for the Suppression of the Vortex Excited Oscillations of Flexible Structures of Circular or Near Circular Section with Special Reference to Its Application to Tall Stacks," National Physical Laboratory, Aero Note 1012, 1963.

[5] Price, P., "Suppression of the Fluid-Induced Vibration of Circular Cylinders," Proceedings of the American Society of Civil Engineers, *Journal of Engineering Mechanics Division,* Vol. 82, July 1956, pp. 1-20.

[6] Naumann, A., Morsbach, M., and Kramer, C., "The Conditions of Separation and Vortex Formation Past Cylinders," AGARD CP, No. 4, Part 2, May 1966, pp. 539-574.

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# Wind-Tunnel Balance Based on Piezoelectric Quartz Force Transducers

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## Introduction

THE test section of the subsonic wind tunnel at the ETH in Zurich has a cross section of 3×2.1 m, and the maximum dynamic pressure is 2300 N/m². It was decided to replace the mechanical compensation balances where the models were attached with wires by a new balance based on Kistler force transducers. The requirements were as follows: one balance for all kind of measurements, measurement of all six components, accuracy better than 1%, interference-free measurement of the components (cross talk below 0.1%), easy calibration at any time, and nonsteady measurements.

Drag forces to be measured down to 0.2 N. The general requirements called for a universally applicable single balance, simple mounting procedures of the models, and a direct readout data aquisition system. The whole arrangement is described in detail in Ref. 2.

## Layout

### Concept of Force Measurement

The requirements led to the location of the balance below the floor of the test section.

Disadvantages with the measurement of the pitch moment when a model is tested in the center of the section have to be taken into account by separate measurement at the model itself. The requirement of interference-free and nonsteady measurements led to a high stiffness of the design including force transducers, which could not be satisfied by a strain-gage balance. Furthermore, the range of a strain-gage balance would have been too small for all planned measurements.

Therefore, it was decided to use piezoelectric quartz force transducers, which offer a very wide measuring range and a high rigidity, yielding practically interference-free resolution of the components.

### Arrangement and Selection of Transducers

In order to resolve six components (three forces and three moments), at least six force transducers must be arranged spatially (Fig. 1). Disposed in the *x-y* plane are three transducers ($S_1$,$S_2$,$W$) of the type Kistler 9203 (±500 N), measuring the drag ($X$), lateral force ($Y$), and yawing moment ($M_z$). Arranged vertically are the transducers ($A_1$,$A_2$,$A_3$) of the type Kistler 9321, with the greater measuring range ±10,000 N. These elements measure the lift ($Z$) and the longitudinal and rolling moments ($M_y$ and $M_x$).

The great difference of maximum loads between the two types of transducers is explained by the fact that only the 9203-type offers the high sensitivity of 48 pC/N (which is needed in the x-y plane), and the high Z loads ask for the more rigid 9321 with a sensitivity of 4 pC/N.

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