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A Preliminary Experiment in Resonance Testing a Rotating Blade

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A PRELIMINARY EXPERIMENT IN RESONANCE TESTING A ROTATING BLADE

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SUMMARY

An experiment to excite the flapping modes of a two-blade non-articulated model rotor is described. The experiment was designed to develop a method of exciting a blade whilst it is rotating, and to measure the change of natural frequencies due to centrifugal effects.

Three of the first four modes of the blade were excited over a range of rotational speeds, and the variations of frequency and damping with rotational speed are given for each mode.

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1 INTRODUCTION

The theoretical treatment of many of the dynamic problems of rotorcraft involves the calculation of the modal characteristics of the blades, and it is desirable that techniques for measuring these characteristics should be available. It is obvious that the application of standard aircraft resonance test procedures to a rotor blade is complicated by having to apply excitation to, and measure the response of, a rotating structure. By using slip-rings, an experiment can be designed in which both the excitation and response measurement systems rotate with the blade. Such an experiment, even at model scale, is likely to demand a considerable effort in design and in the provision of hardware; it may also give rise to problems which cannot be solved without considerable development work being undertaken. On the excitation side there might be problems associated with slip-rings in the power supply lines, and with high lateral loads on the exciters arising from centrifugal forces. On the measurement side, the use of conventional motion transducers operating under steady acceleration conditions might prove difficult.

A method of measuring the modal frequencies and vector responses of a rotating blade, which avoids these difficulties, has been devised. It falls short of what would ideally be required in that it does not yield information on mode shape and it is restricted to single-point excitation. It does, however, enable mode frequencies and dampings to be found from a simple experiment using conventional resonance test equipment and techniques.

The experiment is described in the following sections, but it may be noted here that neither the exciter, nor the single response transducer that was used, rotated with the blade, and thus no slip-rings were necessary. The experiment was confined to the flapping modes of the blade.

The tests show that rotating blade mode frequencies in flap can be measured without the need for complex apparatus or techniques. This result is encouraging, in that it will simplify the testing of rotating aeroelastic blade models when the need is for frequency and damping measurements rather than for mode shapes. The result also suggests that torsion and lag-bending modes could probably be excited by similar methods without much increase in test complexity.

2 MODEL ROTOR AND DRIVING SYSTEM

The rotor consisted of a strip of aluminium alloy 25.4 mm wide, 1.2 mm thick and 1.372 m from tip to tip. It was clamped rigidly between two circular plates of 152 mm diameter which were then mounted on the shaft of a printed armature dc motor having a maximum continuous torque output of 0.215 Nm. The motor had a speed range from zero to 10 revolutions per second, and the rotational speed was measured by timing the period per revolution, using a rotary transducer attached to the motor shaft.

The arrangement is shown in Fig.1, from which it will be seen that each of the two blades is effectively a cantilever from a point 76 mm from the axis of rotation, the blade length being 0.61 m.

3 EXCITATION AND MEASUREMENT SYSTEM

Excitation was provided by a Pye-Ling 201A electromagnetic exciter. This was fixed in a frame with its axis coincident with the axis of rotation of the rotor. The rotor hub was used as a base to which the system shown in Fig.2 was attached. The vertical post AA' supports a horizontal rod BB' (which is stiff relative to the blade) on a flexure hinge at A; a pin BC links the rod to the blade, each end of BC being pointed and resting in a punch mark. A tension spring between the rod and the blade at BC maintains the pin in contact at both ends. The drive rod EF, with nylon cup bearings at each end, bears on fixed ball bearings at the exciter E and the point F which is on the axis of rotation of BB'. A piezo-electric accelerometer is attached to the exciter drive rod between E and the exciter body, and thus responds to motion at the excitation point C.

A sinusoidal exciting force can be applied at C by a sinusoidal current in the exciter provided the joints at E, F, B and C do not chatter or lose contact. The contacts at B and C can be adequately held by the spring already mentioned. A steady compression force across the bearings at E and F, greater than the peak oscillatory force, is provided by the tension springs between B' and D, and G and H. The spring between B' and D also serves to centralize the excitation system. As an alternative to the spring GH, a dc signal to the exciter can be used to provide a compressive force in the joints at E and F, and adjustment of the level of the dc signal enables the exciter coil to be centralized in its travel range with less difficulty than with a spring arrangement.

4 CONTROL AND RECORDING SYSTEM

Signals representing the force input to the blade, and the acceleration response, were taken to a transfer function analyser and thence to an xy plotter so that the vector response of the blade could be plotted.

A Lissajous display of the force and response signals was also arranged so that the typical ellipse degenerated into a straight line when force input and acceleration response were in quadrature.

For the tests in which mode frequencies were measured over a range of rotational speeds, the excitation frequency was varied until the Lissajous figure became a straight line, the frequency at this condition being taken as the resonance frequency. For the rather smaller number of tests in which damping as well as frequency was measured, a vector response curve was obtained on the xy plotter for each resonance.

5 TEST RESULTS

It had been hoped that the test arrangement described above would enable the behaviour of the first three or four flapping modes to be investigated. Two factors prevented this: the first was the high level of damping which developed in the fundamental mode as speed* was increased, and the second was the occurrence of blade flutter at the upper end of the speed range which affected the subcritical response of one of the other modes under investigation.

The level of damping in the fundamental mode, combined with the inboard position of the excitation and response measurement points made it virtually impossible to determine its modal characteristics except at very low speeds. Attention was therefore concentrated on the second, third and fourth modes. As a first exercise, the frequencies determined from a quadrature force-response relationship were obtained at speeds from zero to 6.5 revolutions per second. The results are shown in Figs.3, 4 and 5. It can be seen that there is an increase in frequency with speed, as would be expected from the action of the centrifugal forces. It is true, of course, that aerodynamic forces are also acting, but with purely flapping modes, and at the speeds tested, the effect of aerodynamic forces on mode frequency would be expected to be small. A crude theoretical estimate can be obtained using an expression developed by Southwell[1] for a lower bound for the frequency of the fundamental mode of a beam subjected to centrifugal loading; the values obtained by applying Southwell's expression to the overtone modes are shown as full lines in Figs.3, 4 and 5. The measured values are higher than estimated for the second and third modes but lower for the fourth mode. It should be noted, however, that Southwell's expression is strictly only applicable to the fundamental mode, and in any case, takes no account of aerodynamic forces.

* Throughout the paper 'speed' is used to describe 'rotational speed' except where this would cause confusion.

A further series of tests was then made over the same speed range, measuring the vector response for the three modes. In general, the resulting vector plots were satisfactory in giving nearly circular response loci which could readily be analysed to yield mode frequency and damping. Examples of typical vector plots for the third and fourth modes are shown in Fig.6; it will be seen not only that the curves are very nearly circular, but that the position of the circle in relation to the origin and the force reference conforms closely to that of a single degree of freedom system.

The corresponding curves for the second mode were less satisfactory for, although nearly circular in the region of resonance, were somewhat offset from the origin at the upper end of the speed range indicating a contribution to the response from other modes. It was realised that the supporting structure for the exciter might well have resonances within the frequency range of the test, and if this were so, one of these could account for the characteristics revealed in the vector plots. All the vector plots were analysed and the variation of damping with speed is shown in Fig.7. It is immediately obvious from this figure that whereas damping rises steadily with speed in the third and fourth modes, in the second mode it reaches a maximum at 4 revolutions per second and thereafter decreases. In fact, extrapolating the damping values, an instability at about 7 revolutions per second can be predicted. This was explored, with the results described later in this section. The modal frequencies obtained from the vector plots showed some discrepancies when compared with those obtained using the quadrature force-response criterion. There were several reasons for this. One was, of course, that the offset of the vector plot circles from the origin for the second mode at the higher speeds meant that the resonance frequency did not coincide with the frequency at which quadrature response occurred. To a lesser extent this was also the case with the third and fourth modes. Several modifications to the excitation system were made between the stages of the tests, resulting in small mass and stiffness changes in the system which may have been reflected in small changes of mode frequency.

Further investigation was then made into the fundamental flapping mode. A series of vector responses was measured for the mode, starting at very low speed and covering the frequency range up to the second mode. The response curves are shown in Fig.8 for a constant force input and the variation of damping with speed is shown in Fig.7. It can be seen that the damping rises very rapidly, reaching 0.11 of critical at only 1.87 revolutions per second. Above this speed the characteristic circle becomes too small for analysis, and the practical limitations of excitation and measurement prevented a solution to the difficulties being found with the test rig in its original form.

Fig.8 also suggests that one reason for the offset of the vector responses of the second mode is the contribution of the fundamental mode to the response, although the possibility of responses from supporting framework resonances cannot be ruled out.

At this stage the instability predicted from the damping plots of Fig.7 was explored. The speed was slowly increased (with no external excitation) until at 8 revolutions per second a self-maintained oscillation occurred. The speed was higher than expected and was close to the maximum attainable drive-motor speed for the allowable motor power dissipation. Because the oscillation resulted in an increase of blade drag, the onset of the oscillation caused a fall in speed to a region in which the blade was stable, whereupon the speed picked up again, and the cycle was repeated. The frequency of the oscillation was about 120 hertz - well above that of any of the mode frequencies previously investigated, and, at the critical speed, nearly four times that of the second flapping mode whose damping trend had predicted the existence of an instability.

Stroboscopic investigation showed that there was considerable twist of the blade tip, probably as much as ±30°, although this figure, which is only a visual estimate, could be erroneous since it was difficult to view the motion for more than a second or two at a time. An attempt was made to investigate the torsion modes of the blade by off-setting the pin BC (Fig.2) towards one edge of the blade; this system of excitation was not satisfactory - the pin tending to become dislodged - but it was sufficiently effective at zero speed to show that the blade had an overtone torsion mode with a natural frequency of about 120 hertz.

6 DISCUSSION AND CONCLUSIONS

The original purpose of the experiment was to demonstrate a simple technique for resonance testing a rotating blade. It is concluded that the technique is effective, and offers the possibility of obtaining a great deal of dynamic data from scale rotor models with conventional resonance test instrumentation.

The high level of aerodynamic damping in the fundamental flapping mode is not unexpected, although the relatively low values in the higher order modes do not seem altogether consistent with the behaviour of the fundamental. However, it is probably idle to speculate about the aerodynamics characteristics of the blade since no attention whatever was paid to its aerodynamic contours. What is significant, however, is the fact that the fundamental mode could not be effectively excited in the tests, and this is a problem that is likely to recur with rotor models. Improvements can certainly be made by applying the excitation further outboard along the blade (by extending the length of BF in Fig.2) and by increasing the force output of the exciter. Both these changes will, of course, increase the signal available from the transducer. But whereas the technique, as tried here, would seem to be applicable not only to small scale models but also to full scale rotors on, say, a rotor testing tower, it becomes a less attractive full-scale technique if a very long excitation arm is needed.

The instability described in section 5 may have been a simple flexure-torsion flutter or, more likely, a wake-induced flutter. The high angles of twist noticed at the tip might suggest a form of stall flutter, but if this were so then one would not expect to find the damping tending to zero in a bending mode. Nevertheless, the obvious explanation that the oscillation was a binary type of flutter involving the first overtone flapping mode (because the damping appeared to go to zero in this mode) and an overtone torsion mode (because such a mode was found with a frequency near to the flutter frequency) is not entirely satisfactory. It leaves unexplained the surprising fact that when flutter occurred it did not do so at the frequency of the mode whose damping was reducing and apparently tending to zero. Unfortunately, the speed control of the rig did not allow a detailed investigation of the mode characteristics to be made right up to the flutter speed. In fact, although the flutter was of considerable interest, it can hardly be regarded as of more than academic significance at the present time, since all full-scale rotor blades are massbalanced to avoid flutter, whereas the blade of the tests described here had inertial and flexural axes both at half chord. It may be that a trend towards reducing blade weight by reducing massbalance will make blade flutter a design consideration, but this does not appear to be the case at the moment.

7 FURTHER WORK

The main shortcoming of the experiment described is that it gives no information about the mode shape, and hence distribution of stress or bending moment, along the blade. The next step, therefore, is to incorporate a set of slip-rings in the rig which can handle the signal from blade transducers. This modification is being made, and the blade will be instrumented with strain gauges from which the bending moment distribution in each mode will be obtained.

The difficulties that have been encountered are mainly due to the presence of aerodynamic forces. It is likely that there will be a need, in model rotor dynamic investigations, to separate the effects of centrifugal forces from the effects of aerodynamic forces to facilitate comparison of experiment and theory. Tests are therefore planned in which the rotor will be installed in a decompression chamber and the blade dynamic characteristics measured at pressures down to 0.01 of an atmosphere. At the lowest pressures, the aerodynamic forces will be negligible and the test results will be compared with calculations based solely on the blade inertia, stiffness and centrifugal characteristics.

For any experimental investigation which includes aerodynamic effects, representative blade geometry will be necessary. Dynamic models of rotor blades having correct planforms and sections are under construction, and these will be used for all model rotor work once they are available.

The extension of the present technique to excite blade torsion and chordwise (lagwise) bending modes is also desirable and should present little difficulty. The possibility of mounting the experiment in a wind tunnel to investigate cyclic behaviour of blades is also under consideration.

REFERENCE

No.	Authors	Title, etc.
1	R.V. Southwell Barbara S. Gough	On the free transverse vibrations of airscrew blades. A.R.C. R & M 766 (1921)

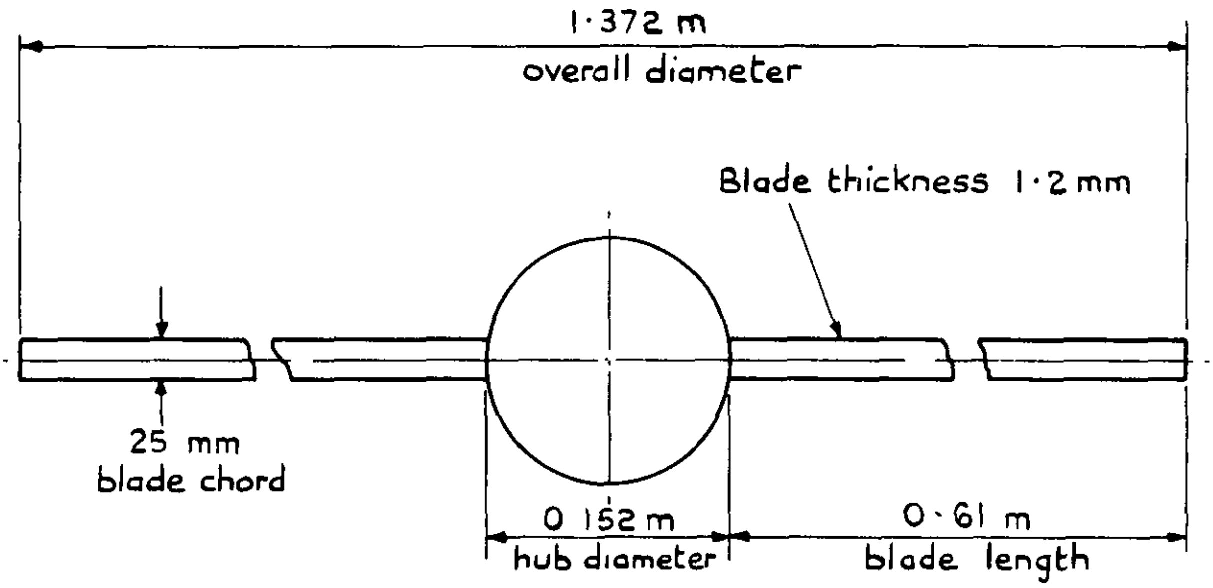

Fig. 1 Geometry of rotor

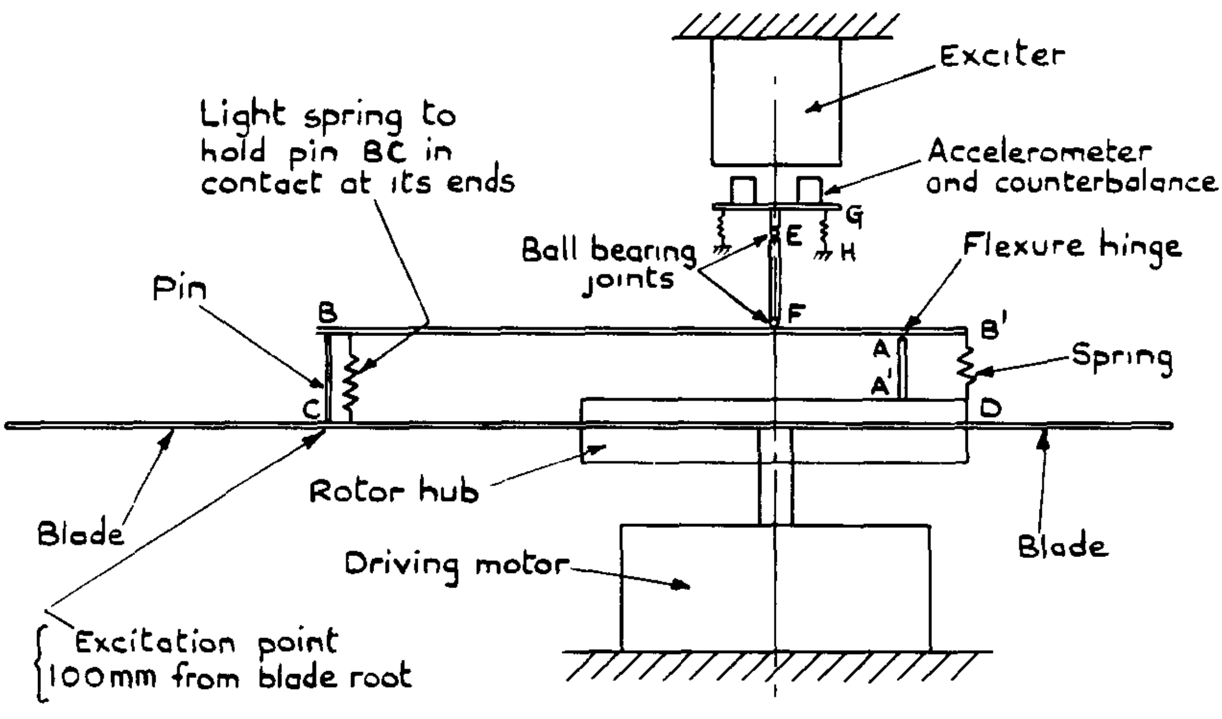

Fig. 2 Excitation and measurement system

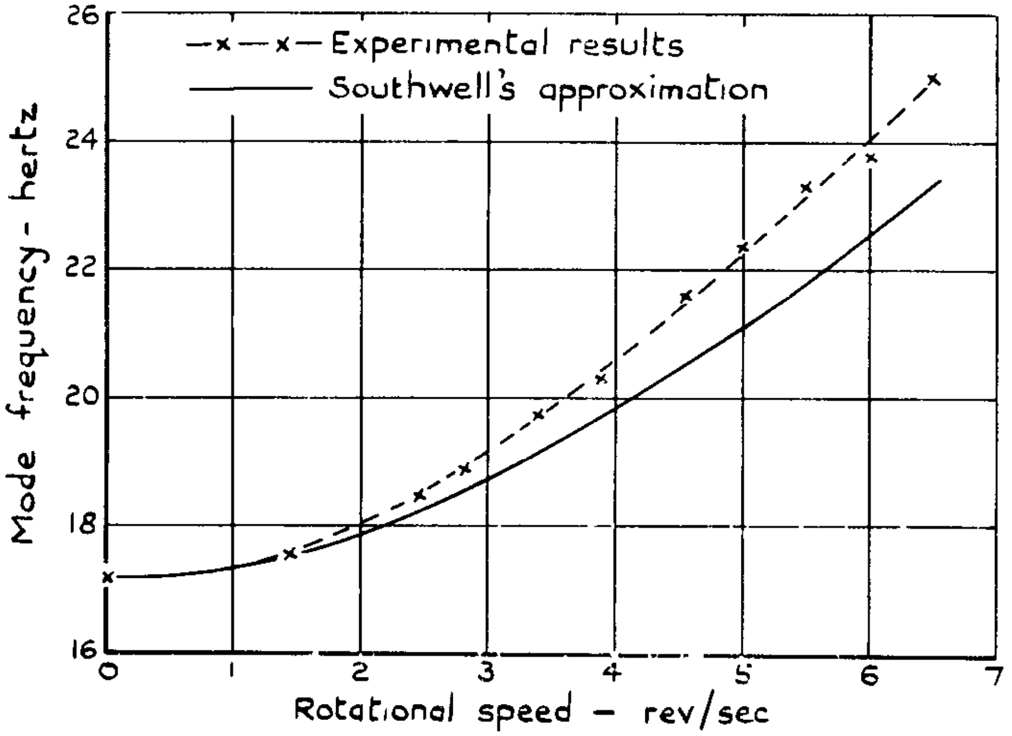

Fig. 3 Effect of rotation on frequency of second mode

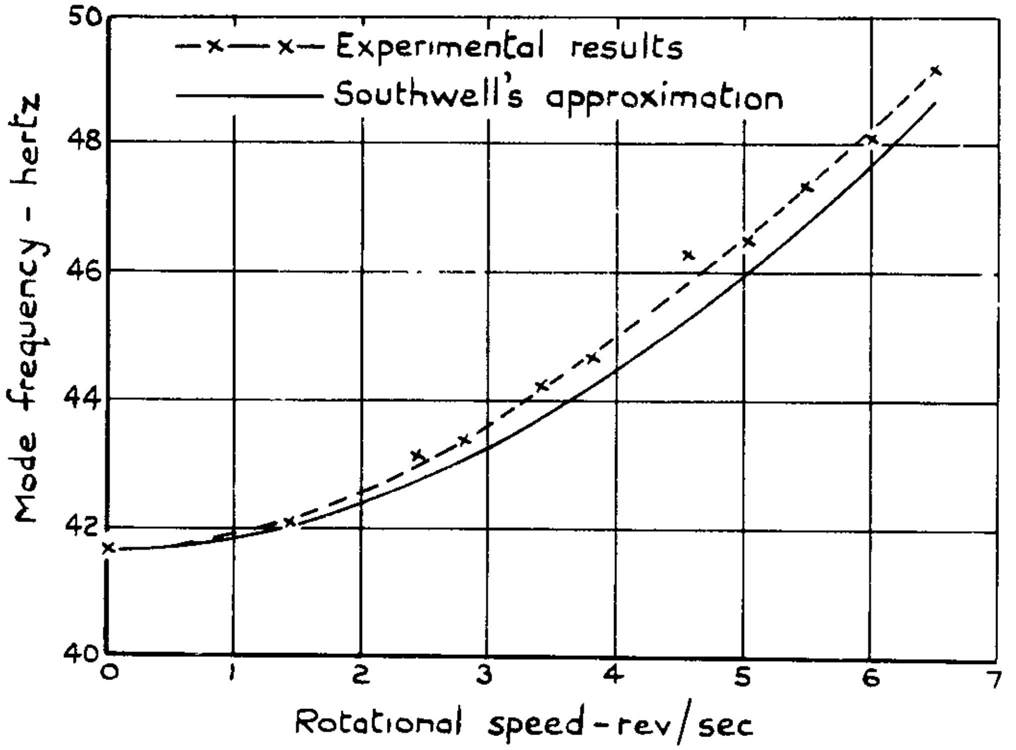

Fig. 4 Effect of rotation on frequency of third mode

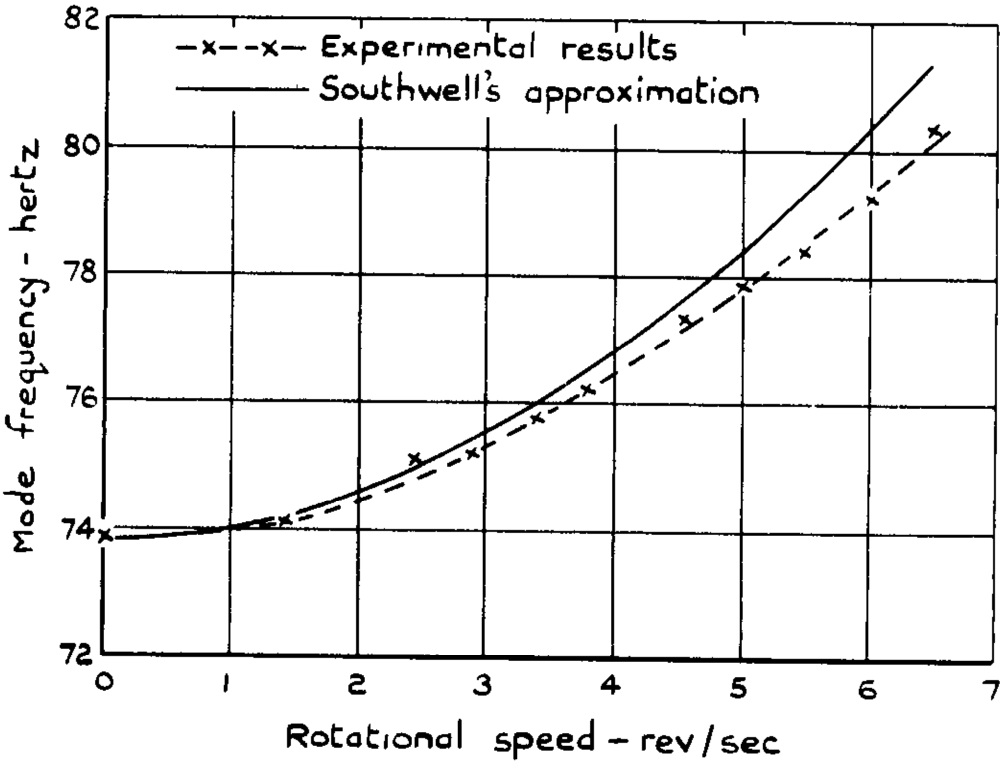

Fig.5 Effect of rotation on frequency of fourth mode

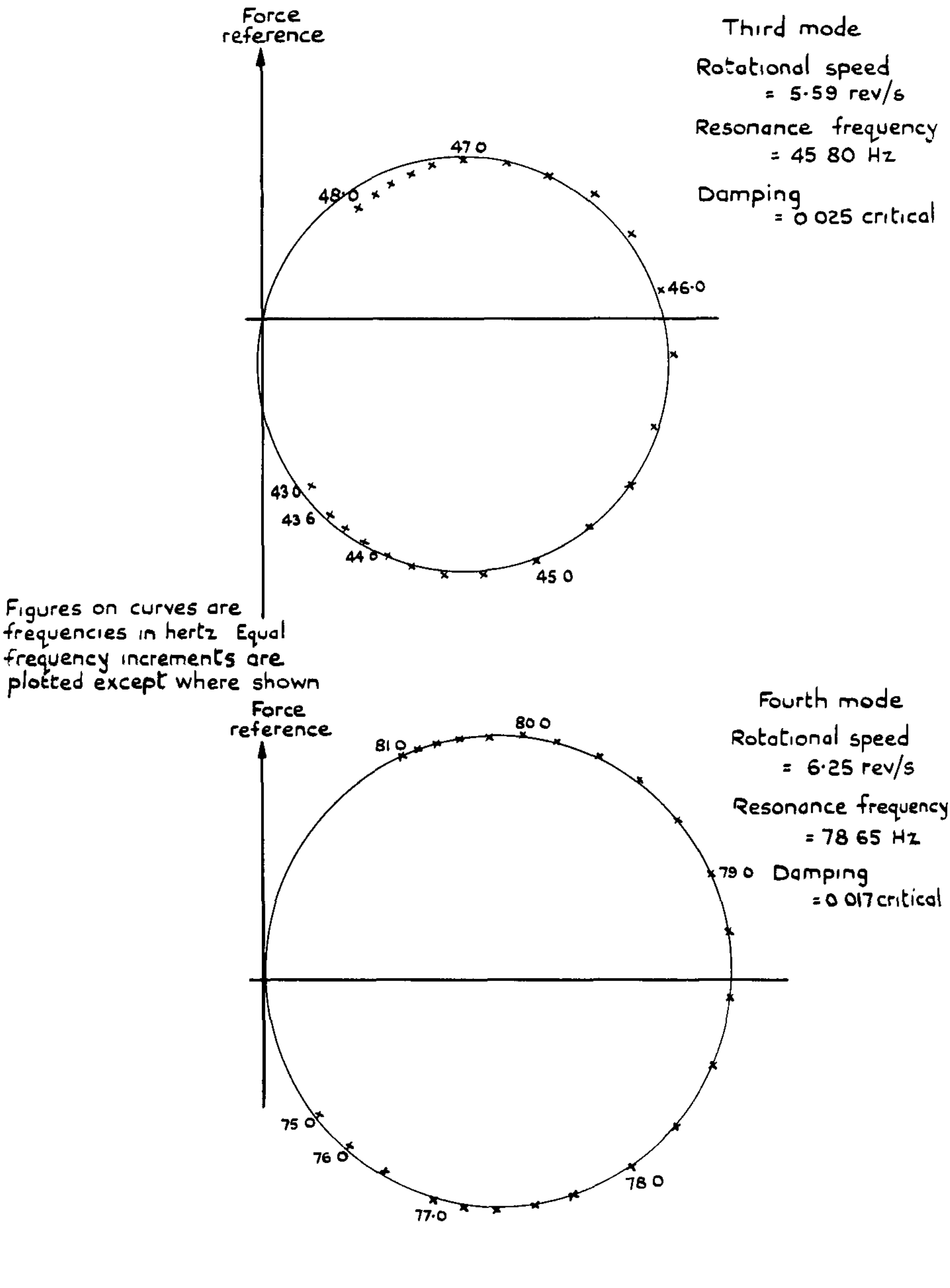

Fig.6 Typical vector plots for the third and fourth modes

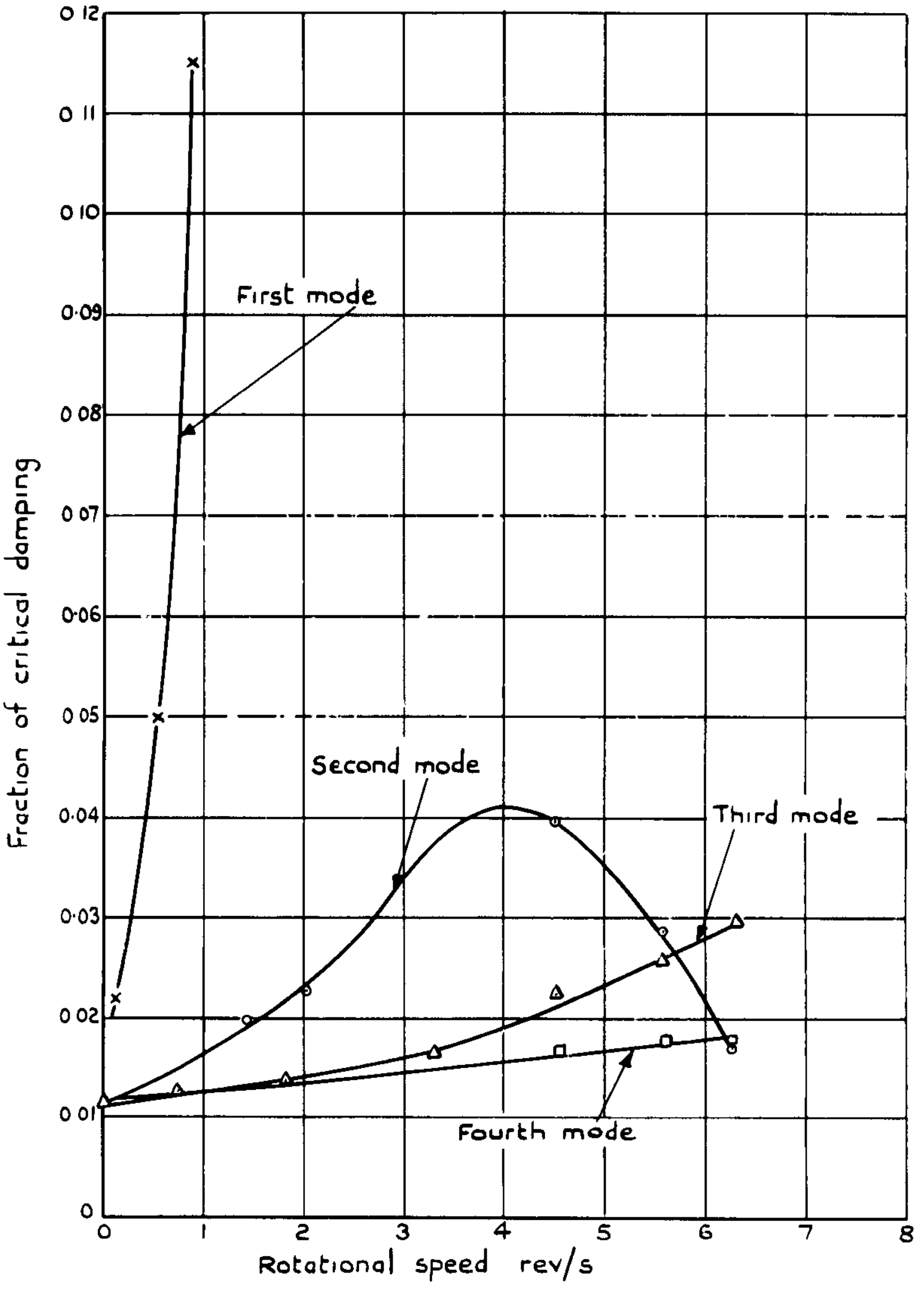

Fig.7 Effect of rotation on mode damping

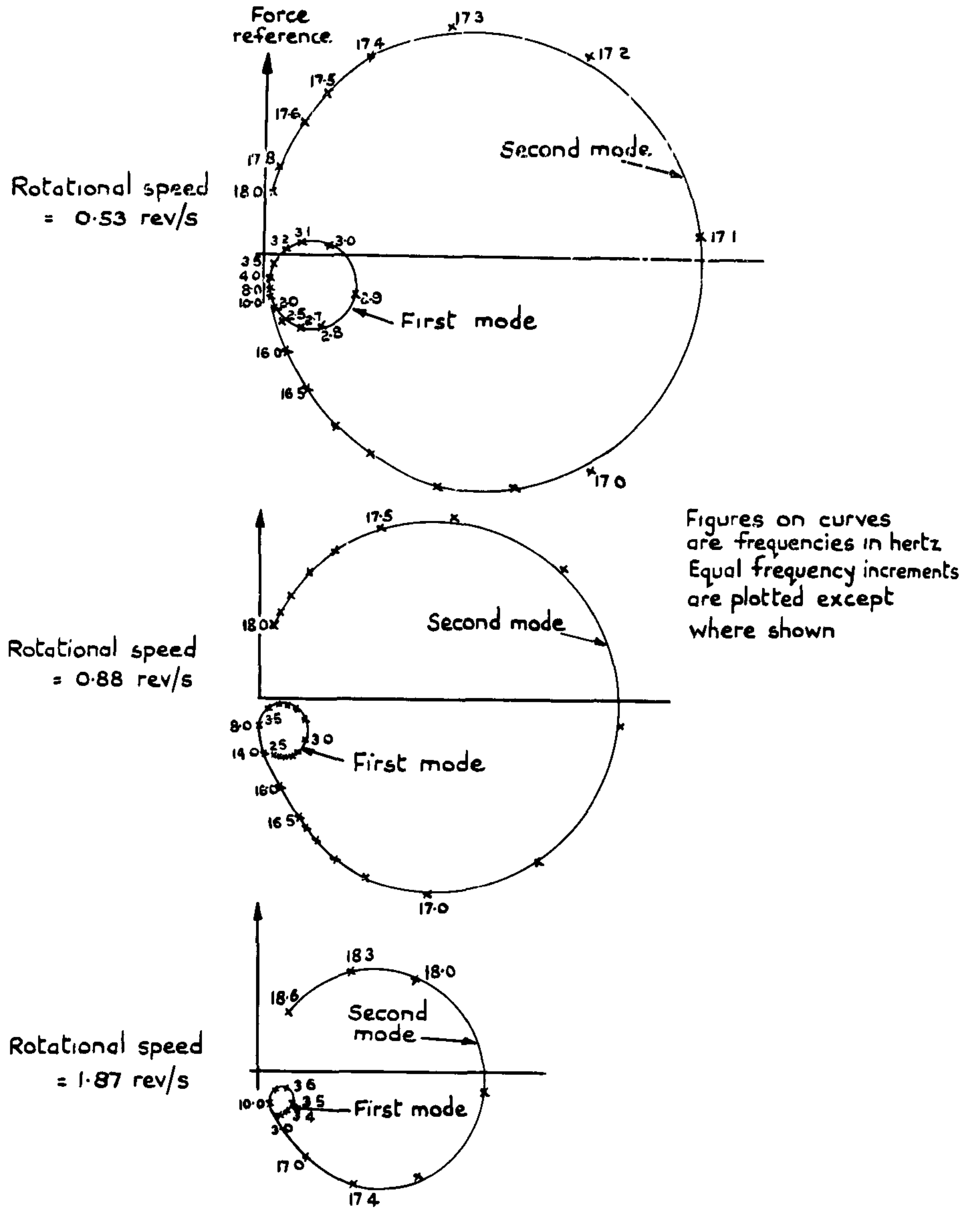

Fig.8 Vector responses for first and second modes under constant exciting force

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