EFFECTS OF FLOW UNSTEADINESS ON THE TIME-MEAN RESPONSE OF A KIEL-TYPE TOTAL PRESSURE PROBE : PRELIMINARY INVESTIGATION

by

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Abstract

CERL measure total pressures in steam turbines with a Kiel-type probe because of its insensitivity to flow incidence. However there is evidence in the literature that in unsteady flow the time-mean pressure recorded by this type of probe may be in error. The CERL probe has been tested in an unsteady flow generator which was developed to simulate turbine-type unsteadiness. Initial results showed a marked difference at 30% turbulence level between the mean responses of Kiel and pitot probes. At a level of 5 to 20%, more typical of the flows in which the CERL probe is used, the effect was smaller but the results were not sufficiently accurate to give a true picture. Further experiments are desirable to obtain higher accuracy and to determine which probe, if either, gives correct readings.

1. INTRODUCTION

The Kiel-type of stagnation pressure probe, incorporating a concentric open-ended cylinder around the head of the impact tube has been considered particularly suitable for turbomachinery applications because of its insensitivity to the angle of the incident flow, giving it a useful range of two to three times that of the simple pitot tube. For this reason it was adopted by CERL for turbine traversing (Walters, Moore and Langford, 1971) (Fig. 1). However, there is evidence in the literature (e.g. Samoilovich and Yablokov, 1970) that in unsteady flows the Kiel probe indicates higher mean pressures than the pitot probe.

At the University of Sussex an unsteady flow generator was developed as a flow simulator for heat transfer measurements in gas turbine cascades (Bayley and Priddy, 1980). The flow unsteadiness is generated by a rotating cage situated in the throat of a convergent nozzle, to which the cascade test section is usually attached.

For the present tests the unsteady flow generator was operated in open-jet mode to enable the mean response of Kiel and conventional pitot probes to be compared. Thus it was intended to discover whether the presence of the hood seriously modified the time-mean pressure recorded in a flow which incorporated flow fluctuations of the character obtaining in a turbine. The tests were planned as a preliminary experiment which would indicate the desirability of a more detailed investigation.

2. FLOW FLUCTUATIONS IN A STEAM TURBINE

To judge the adequacy of modelling the turbine flow unsteadiness by the test facility we may consider the steam turbine measurements of Wood (1973). Figure 2 shows the last two stages of the 500 MW steam turbine in which hot wire measurements were made, together with a summary of the results obtained. From this it may be seen that the r.m.s. velocity fluctuation deduced from the hot wire measurements varied from 4.8% downstream from the hub region of a fixed blade to 17% downstream from the tip region of the moving blades. The power spectral density of the hot wire signal obtained downstream from the final moving blade tip region is shown in Fig. 3. The features to note are the broad bandwidth of the random fluctuations and the various discrete frequencies. The blade passing frequency was 4.6 kHz of which both the primary and second harmonics are clearly visible. Other lower discrete frequencies are likely to result from blade vibrations and acoustic emissions. The reduction in power spectral density at high frequency is a function of the tape recorder and hot wire frequency responses. The former had a frequency response which was flat to approximately 20 kHz whilst the hot wire system frequency response was flat to approximately 10 kHz (Wood, 1975a) as shown in Fig. 4. Consideration of the blade boundary layers suggests that frequencies up to the order of 2 MHz could be present and recordings made with a tape recorder having a maximum frequency capability of 300 kHz showed evidence of enhanced signal component at this frequency (Wood, 1975b).

3. THE UNIVERSITY OF SUSSEX FACILITY

The wind tunnel at Sussex is capable of exhausting up to 1 kg/s of dry air through a turbulence grid, resulting in a bulk velocity of 170 m/s, Reynolds number of 82 × 10^5 per metre and Mach number of 0.45.

The air stream through the wind tunnel is supplied by a six stage, centrifugal blower at approximately 90°C and 1.6 bar maximum pressure. A Dall tube-type Venturi incorporated in the 150 mm supply piping meters the flow, which then passes through a 2 m long diffuser section. This is followed by a 380 mm square settling chamber, incorporating two gauzes as flow straighteners and coarse filters. Finally the air exhausts at atmospheric pressure from the nozzle of a 16:1 contraction. The turbulence level of the exhaust jet, in the absence of grids, is approximately 0.5% and pitot traverses have shown the velocity profile to be uniform across the entire outlet area (79 mm × 74 mm), except in the regions of the wall boundary layer.

4. FLOW FLUCTUATIONS IN THE TEST FACILITY

The squirrel cage (Priddy, 1980), consists of thirty equi-spaced 2.4 mm diameter carbon fibre bars placed on a pitch circle of 58.6 mm diameter. It can be driven up to speeds of 20,000 rpm corresponding to a bar passing frequency of 10 kHz, about the horizontal axis transverse to the flow. A photograph of the rotatable cage is shown in Plate 1.

Traverses transverse to the flow have shown the turbulence intensity to be uniform across the central plane where probes can be tested. The horizontal mean flow profile is also flat, but in the vertical plane the cage induces some distortion. This distortion is most severe at the highest cage speed and lowest mass flows.

For comparison with the turbine results some power spectral densities of the flow from the University of Sussex facility are shown in Fig. 5 from which it can be seen that the squirrel cage gives a broad band random fluctuation with usually one main discrete peak (10 kHz at 20,000 rpm). The variation of rms turbulence level with distance downstream from the nozzle exit is shown in Fig. 6, demonstrating that the required fluctuation levels were available.

5. TEST PROCEDURE

The test probe consisted of an internal pitot tube of 1 mm outside diameter (od) set in a cylindrical shroud of 6 mm od. The shroud was internally profiled, venturi-like, to a minimum internal diameter of 3 mm. The probe was mounted on a test stand in the open jet downstream from the rotating squirrel cage. The probe head was connected via 1 m of 3 mm internal diameter (id) PVC tubing to a SE Laboratories 180 variable reluctance pressure transducer. This was a transducer intended for steady readings and of unspecified frequency response.

Having taken a reading from the Kiel probe, the latter was replaced by the reference probe at the same position in the flow. The reference probe was a simple pitot tube of 1 mm od and 0.5 mm id. Some additional checks were made with a "CERL pitot tube" of 1.2 mm od and 1 mm id.

To check whether errors in position between the Kiel probe and reference probe could give different results some vertical and horizontal traverses were made, but no significant error was found to arise from this cause. When used in a turbine, the Kiel probe is connected to the pressure transducer via a combination of stainless steel hypodermic tubing and PVC tubing. This was simulated to test for the effect of connecting tube length on the time-mean response, but no effect was observed.

The pressure transducer was calibrated against a mercury manometer and barometer combination.

6. PRELIMINARY RESULTS

Fig. 7 summarizes the results of the initial tests in the form of ratio of pressure recorded by the Kiel probe to that recorded by the pitot probe plotted versus turbulence level. The three curves drawn through the results are for Mach numbers of approximately 0.2, 0.33 and 0.45 reading from lowest to highest. The most striking feature of these results is the large ratio of Kiel/Pitot pressures at the highest turbulence level (30%), representing a difference in the deduced mean velocity of approximately 5% for all three Mach numbers.

In the case of the range of interest, 5 to 17%, the results are less certain. There was considerable scatter (∿0.5%) in the comparison between Kiel and reference probes obtained at 0.5% turbulence level and this is of similar order to the discrepancy apparent in the range up to 20% turbulence. However, some comparisons between the Kiel probe and the "CERL pitot probe" gave exact correspondence at 0.5% turbulence but showed also significant differences at 30% turbulence as shown in the figure.

It is clear from these initial results that there are differences between the mean response of a Kiel probe and a pitot probe in unsteady flow. However, the tests reported were unable to indicate which probe, if any, read the true time-mean stagnation pressure. If there were a large flow angle fluctuation associated with the unsteadiness it would be expected that the Kiel probe would give the more accurate reading, being relatively insensitive to flow angle.

An error in measured pitot pressure will lead to an error in the mass flow obtained by integration of the flow quantities across the annulus of the turbomachine. The error enters through the velocity deduced from the pressure measurements and it may be shown that the error in velocity is given approximately by:

$$\frac{dV}{V} \cong \frac{1 + \frac{\gamma-1}{2} M^2}{\gamma M^2} \left(\frac{dp_T}{p_T} - \frac{dp}{p} \right) \qquad \ldots (1)$$

where M is "frozen" Mach number and d( ) represents the error quantity.*

Taking the results shown in Fig. 7 as representing errors in measured total pressure and relating the errors to velocity via eq. (1) we obtain the following table:

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* This result is for wet steam and depends on the approximation $d\rho/\rho \cong dp/p$.

	Unsteadiness (%)			
	10		20	
M	dp_T/p_T	dV/V	dp_T/p_T	dV/V
0.2	0.0005	0.0095	0.0010	0.019
0.35	0.0007	0.0043	0.0015	0.0093
0.45	0.0035 (max.)	0.0102	0.0050	0.0145

Therefore, dependent on Mach number, the error in velocity, and hence in mass flow, could be up to 1% at 10% unsteadiness and up to 2% at 20% unsteadiness. These errors are comparable with some of the other errors inherent in making flow measurements in rotating machinery and, providing their presence is recognized, need not cause major problems. However, as part of the programme to minimize measurement error the present tests should be extended to clarify their significance.

A further point which should be considered when measuring pressure in unsteady flow is the form taken by the Bernoulli equation. This is:

$$\frac{1}{\rho}\frac{\partial p}{\partial s} + \frac{\partial}{\partial s}\left(\frac{1}{2}V^2\right) + \frac{\partial V}{\partial t} = 0 \qquad \ldots (2)$$

where p is pressure
s is distance
t is time
V is velocity
and ρ is density

Integrating this equation with respect to s for compressible flow gives:

$$\frac{\gamma}{\gamma - 1}\frac{p}{\rho} + \frac{1}{2}V^2 + \int \frac{\partial V}{\partial t}\, ds = \text{const.} \qquad \ldots (3)$$

The last term on the LHS involves both time and distance and indicates that there could be an interaction between the scale of the unsteadiness and the scale of the probe.

7. CONCLUSIONS

Preliminary experiments to measure the time-mean response of a Kiel probe in unsteady flow have revealed a difference between the response of the Kiel probe and a pitot probe. Further experiments are necessary:-

(i) to improve the accuracy of results in the range of turbulence level previously measured in the LP cylinder of a steam turbine (5 to 20%).

(ii) to determine which, if either, type of probe reads the correct time-mean stagnation pressure.

8. REFERENCES

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9. ACKNOWLEDGEMENTS

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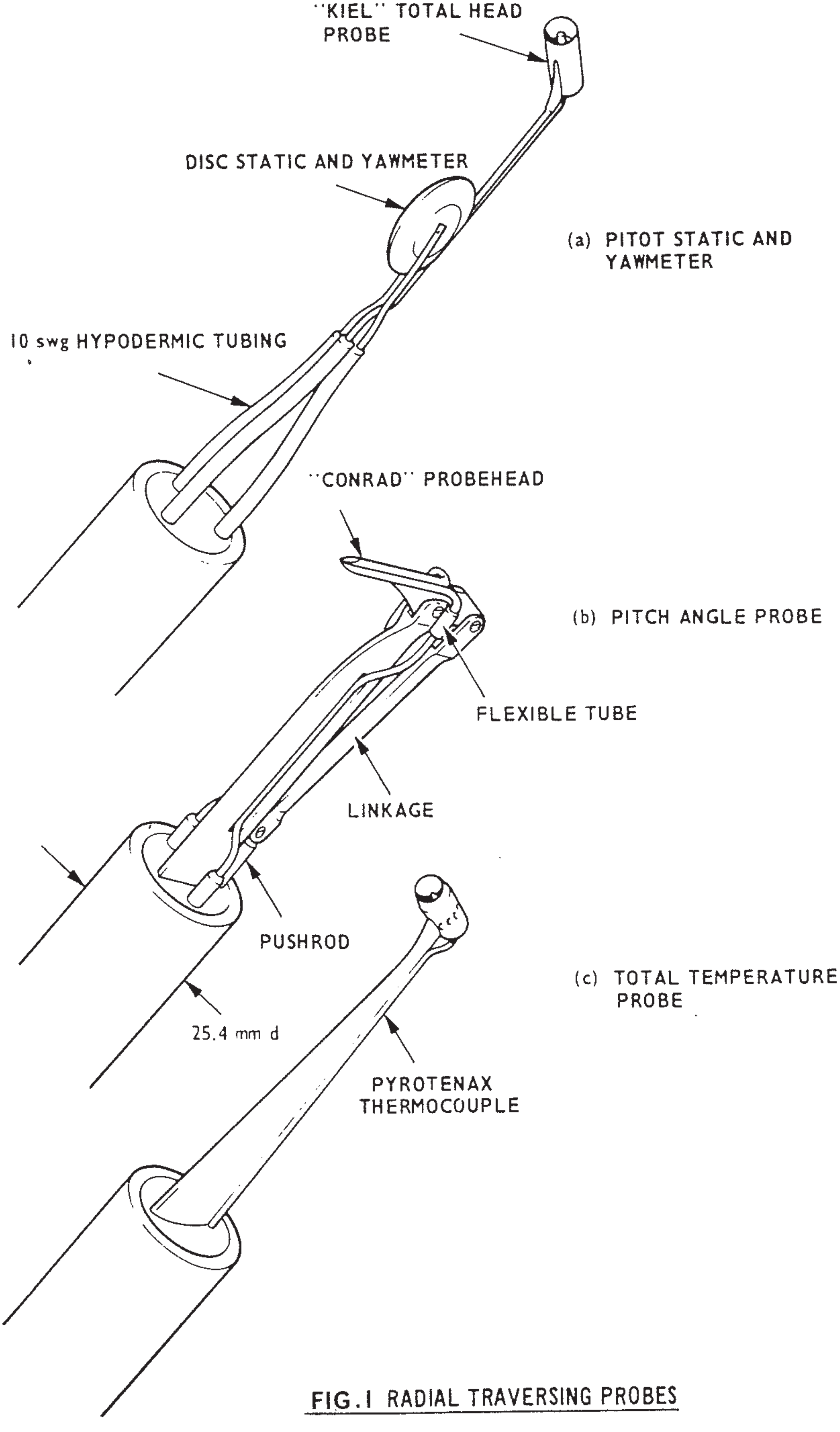
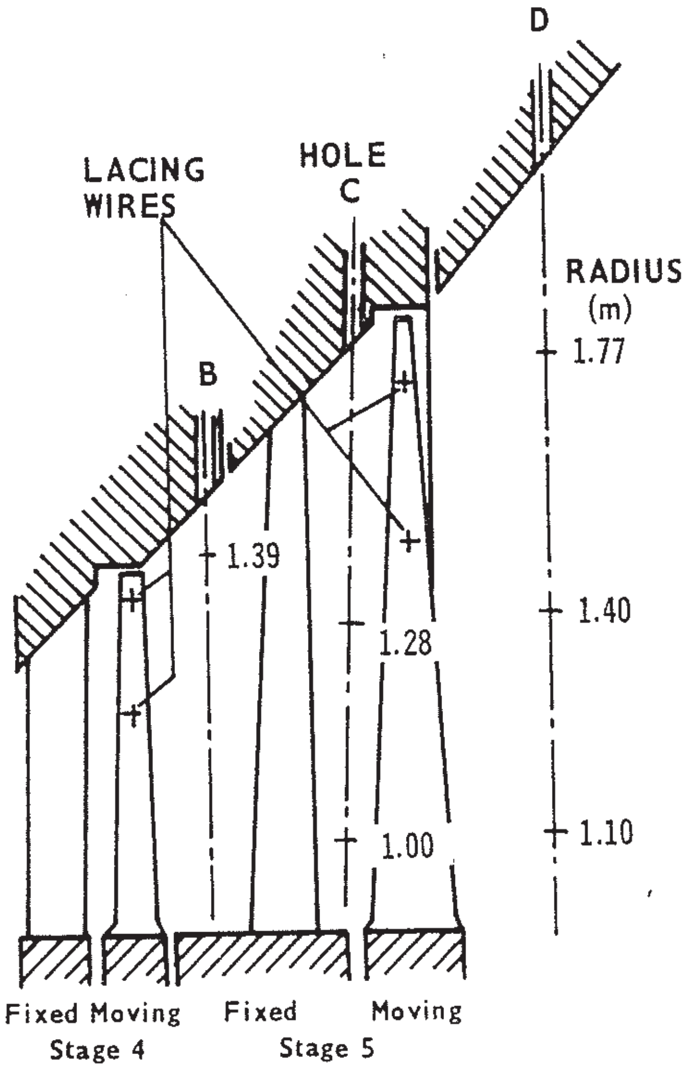


FIG.1 RADIAL TRAVERSING PROBES



FINAL TWO STAGES OF THE LOW-PRESSURE TURBINE SHOWING PROBE POSITIONS

HOLE	RADIUS (m)	m′	T′	R_{mT}	u′	T_S′
B	1.39	0.17	0.045	0	0.17	0.045
C	1.28	0.065	0.072	0	0.088	0.068
C	1.00	0.040	0.035	0	0.048	0.033
D	1.77	0.18	0.080	0	0.17	0.079
D	1.40	0.071	0.046	0	0.081	0.045
D	1.10	0.080	0.029	0	0.084	0.029

FIG. 2 MEASURED FLOW FLUCTUATIONS IN A 500 MW TURBINE

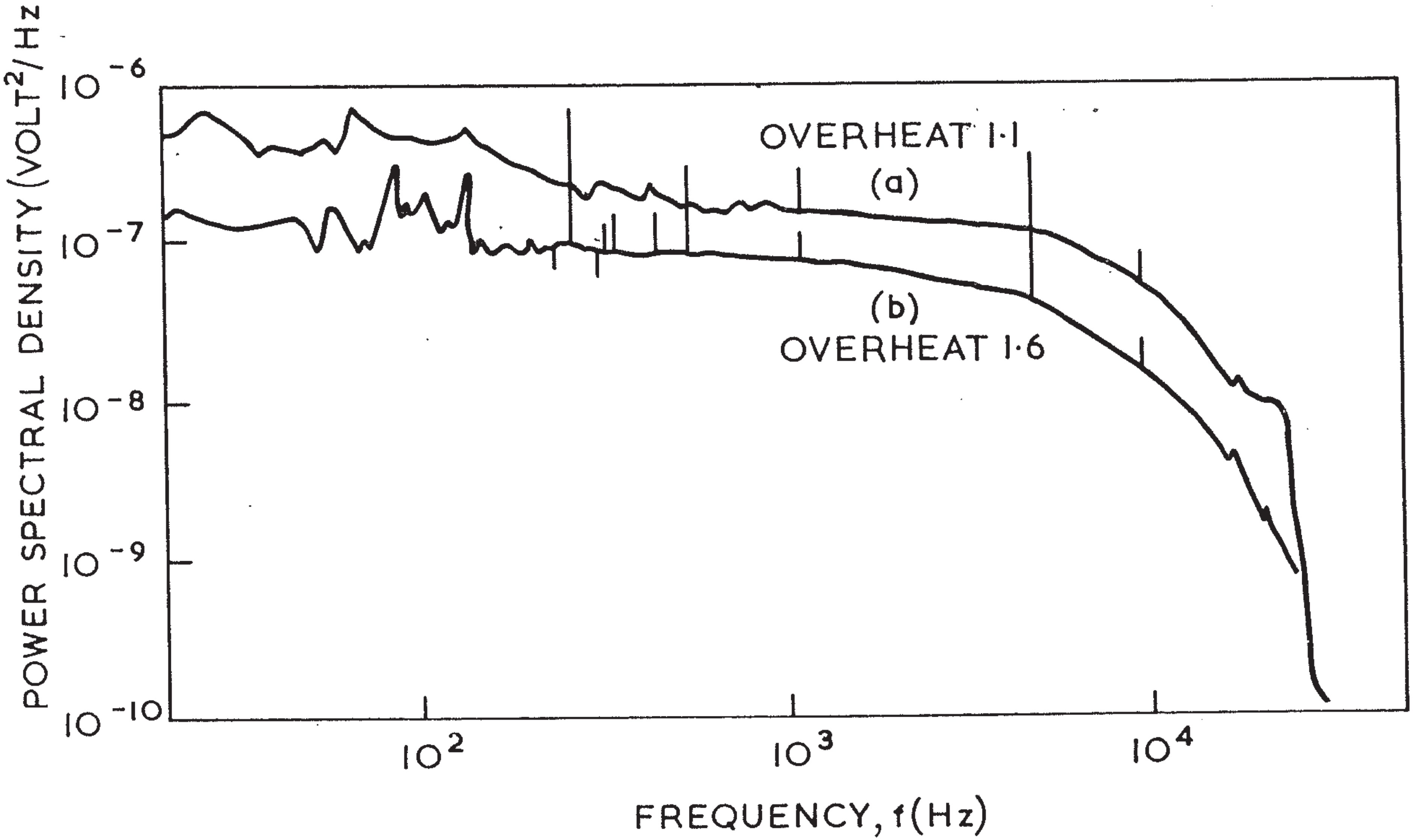


FIG.3 POWER SPECTRAL DENSITIES OF ELECTRICAL SIGNALS DOWN-STREAM OF ROTOR TIP

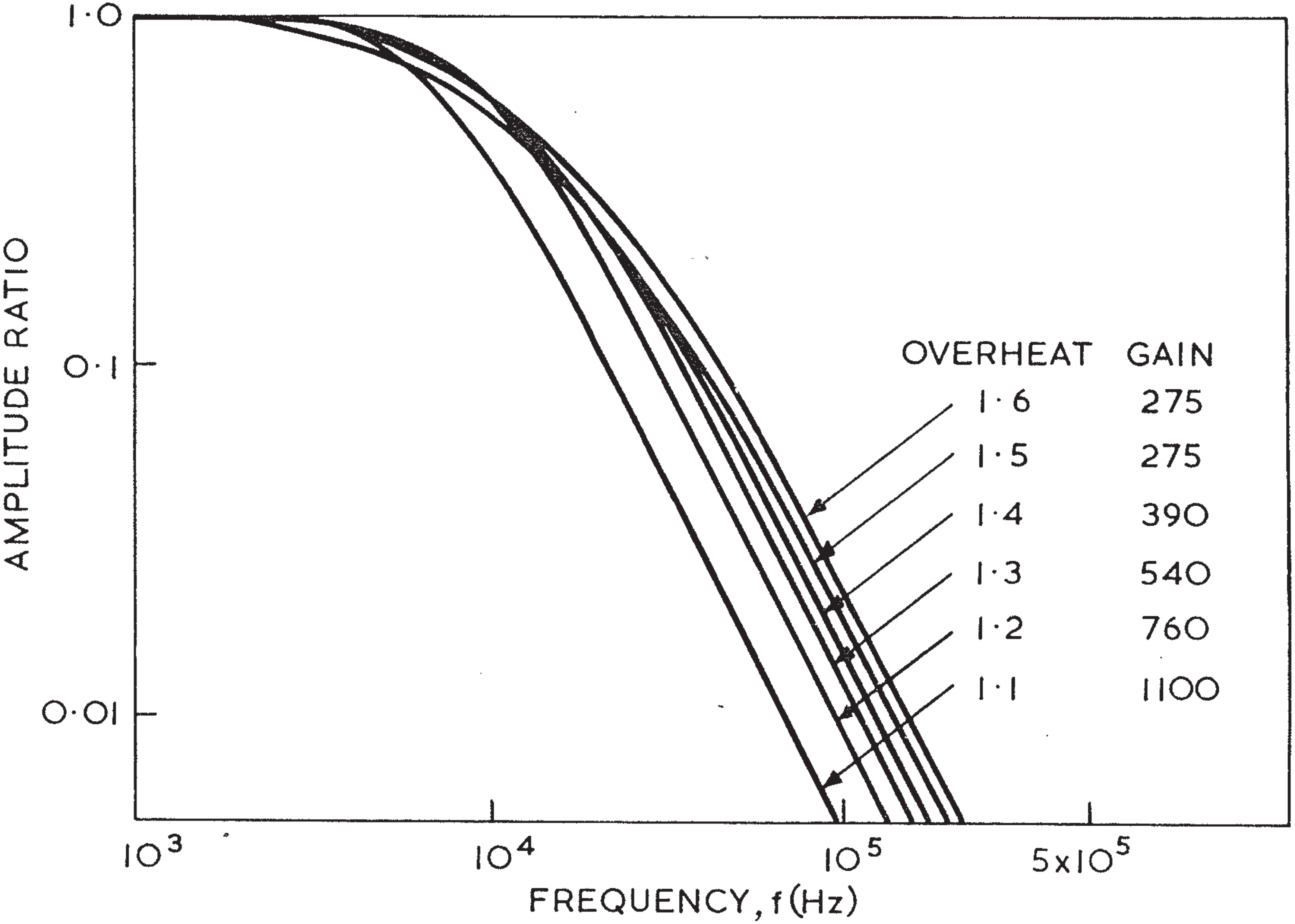
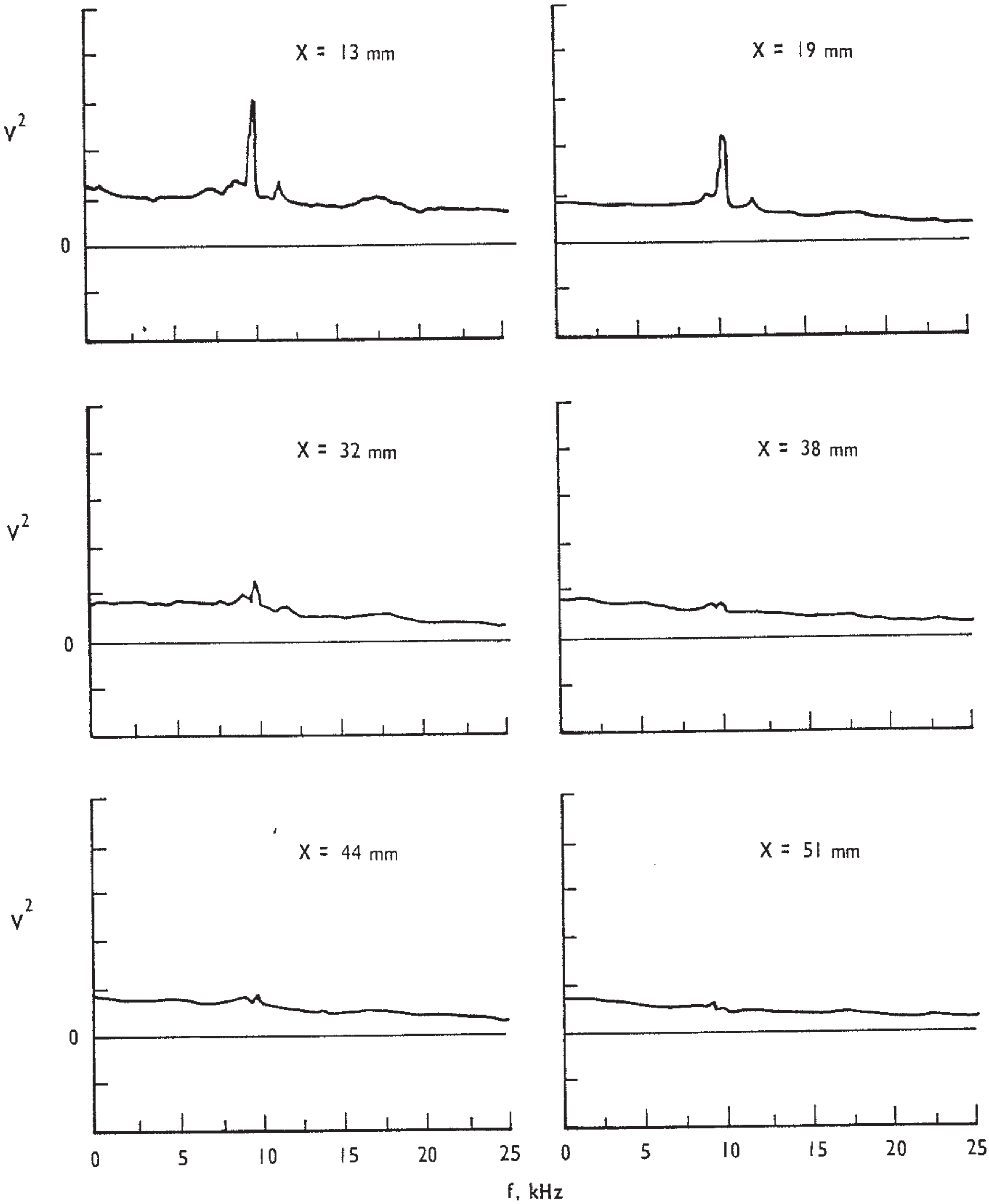


FIG.4 FREQUENCY RESPONSE OF HOT-WIRE ANEMOMETER



CFRP CAGE N = 20,000 rpm, m = 0.64 kg/s

FIG. 5 POWER SPECTRAL DENSITIES OF HOT WIRE SIGNALS IN UNIVERSITY OF SUSSEX FACILITY AT DIFFERENT AXIAL DISTANCES

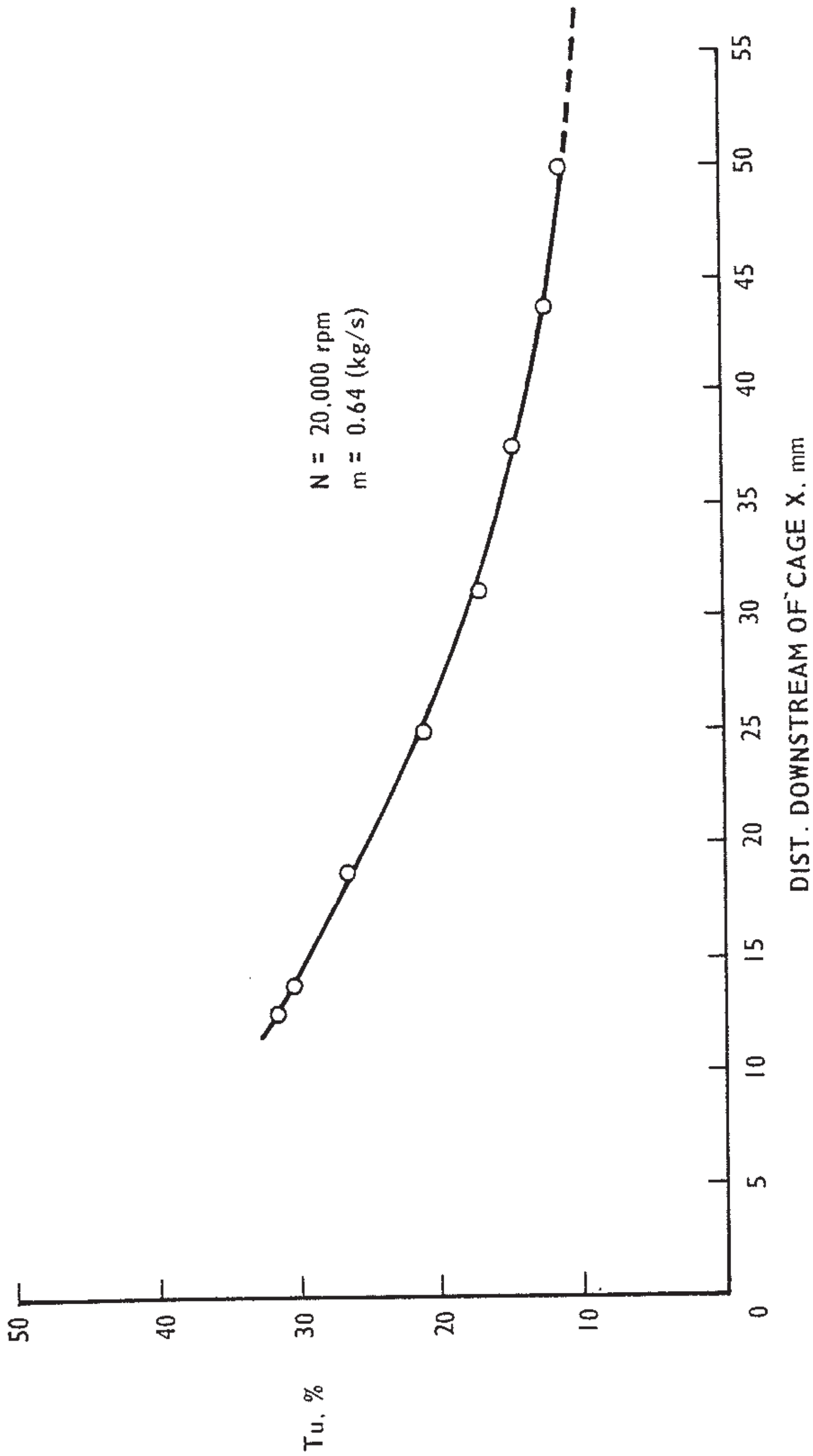
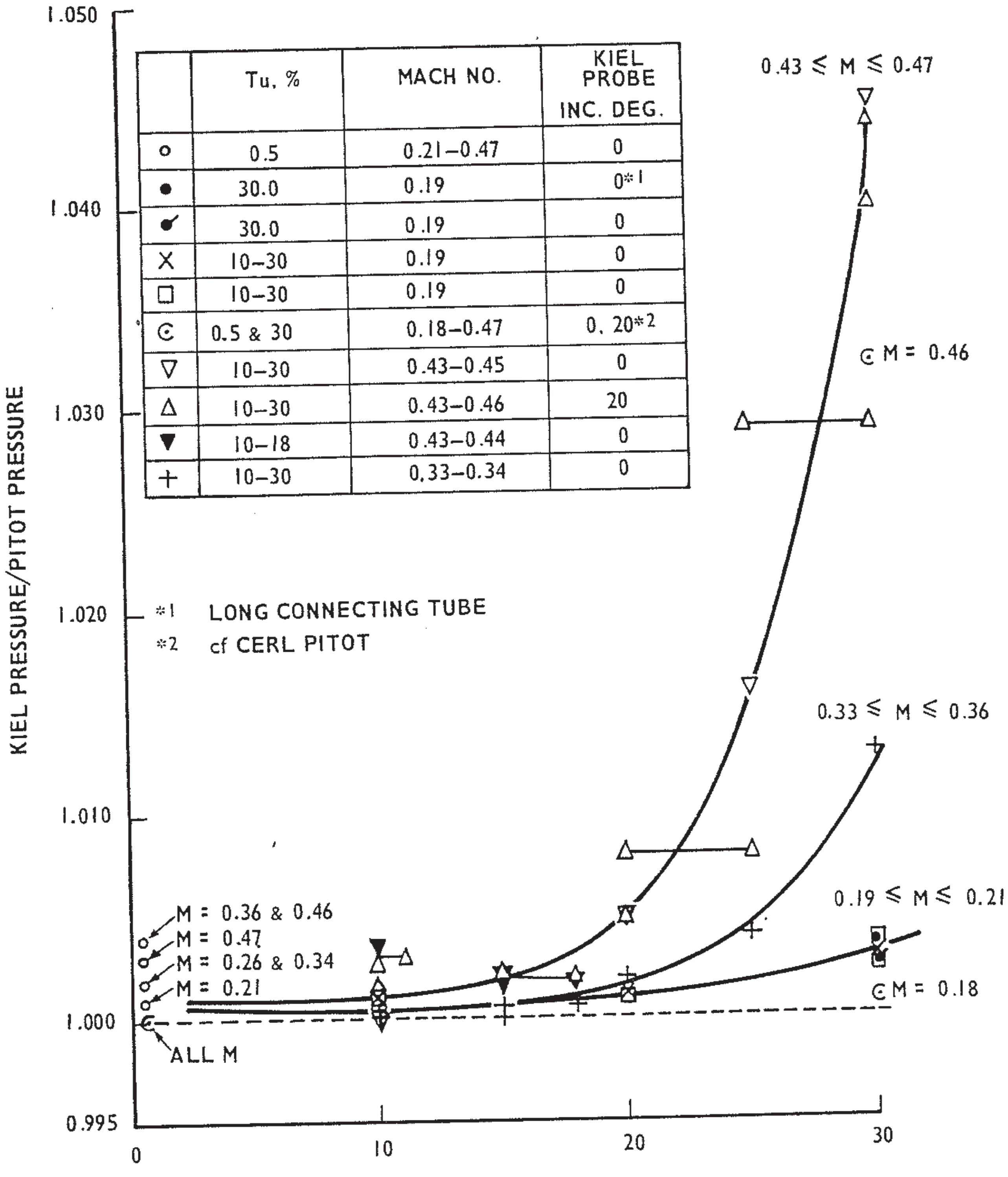


FIG. 6 VARIATION OF RMS TURBULENCE INTENSITY WITH AXIAL DISTANCE IN UNIVERSITY OF SUSSEX FACILITY



	Tu, %	MACH NO.	KIEL PROBE INC. DEG.
○	0.5	0.21–0.47	0
●	30.0	0.19	0*1
●⁄	30.0	0.19	0
×	10–30	0.19	0
□	10–30	0.19	0
⊂	0.5 & 30	0.18–0.47	0, 20*2
▽	10–30	0.43–0.45	0
△	10–30	0.43–0.46	20
▼	10–18	0.43–0.44	0
+	10–30	0.33–0.34	0

FIG. 7 INITIAL RESULTS: RATIO OF PRESSURES MEASURED BY KIEL PROBE AND 1 mm OD PITOT PROBE. CAGE SPEED: 10,000 rpm

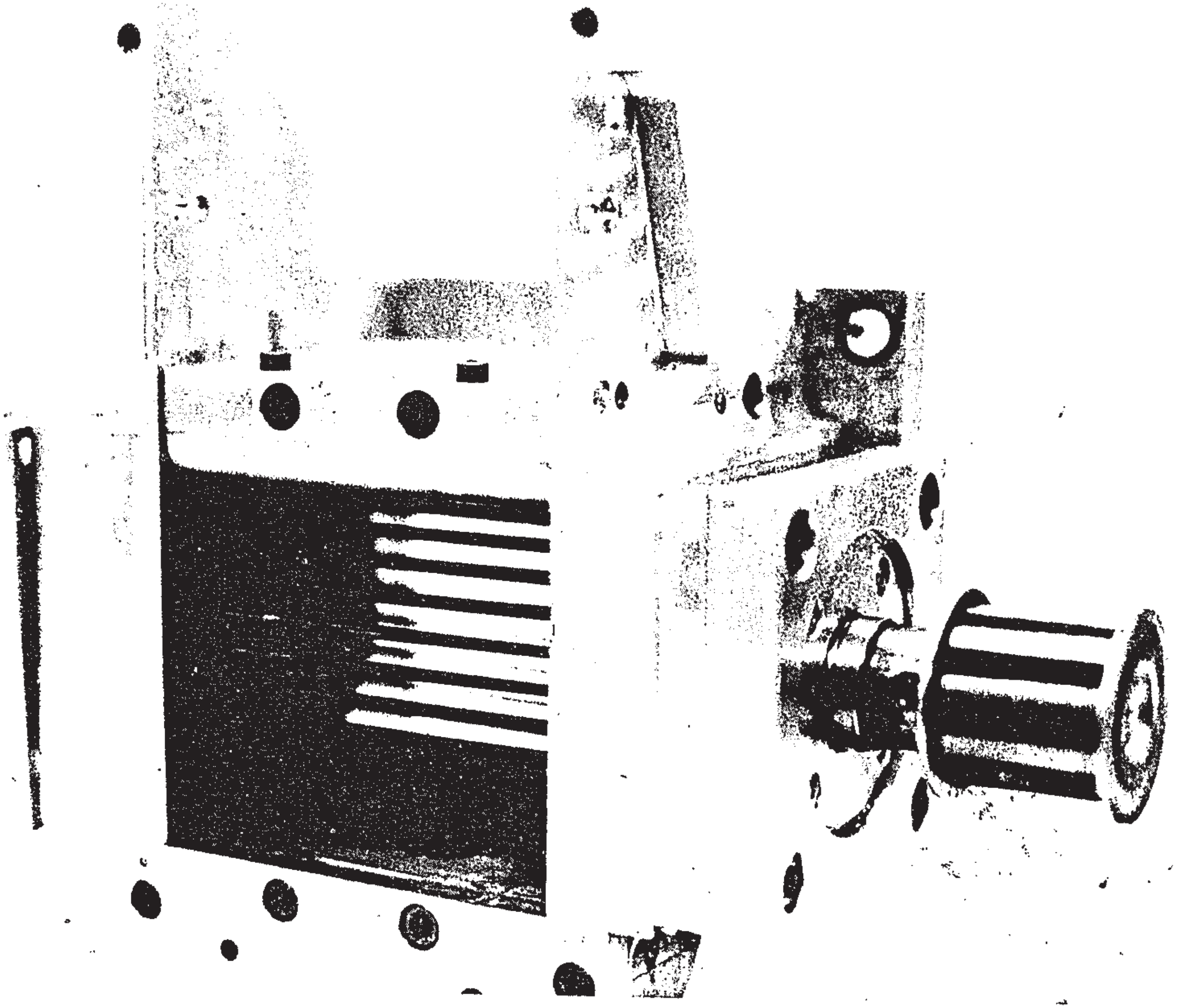


PLATE 1. UNIVERSITY OF SUSSEX TURBULENCE GENERATOR