"THE INFLUENCE OF MOVING CAGES ON AIRFLOW IN MINE SHAFTS"

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A C K N O W L E D G E M E N T S

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PREFACE

This work contains the results of an experimental investigation into some of the effects produced on the airflow conditions in mine shafts by the movement of cages. The investigation was carried out in the Department of Mining, The Royal College of Science and Technology, Glasgow and took the form of a programme of tests on models. The work represents an extension of the experiments of Alex. Stevenson, Ph.D., who used this method to study the shaft pressure losses produced by stationary cages.

A brief outline of the background, scope and objects of the work and of the methods employed is given in the Introdution.

Sections 2 to 5 deal with the development of suitable apparatus for continuously recording low level fluctuating pressures. The methods available for recording fluctuating pressures are discussed, and tests with electric resistance strain gauge transducers and with variable capacitance type transducers are described. The factors affecting the response of the apparatus to fluctuating pressures are also considered.

Sections 6 to 9 contain a description of the model shaft in which the experiments were carried out and give details of the experimental work. The resistance of stationary cages, with and without straight-sided fairings, is examined further with particular reference to the effects produced when two cages are within the zone where their combined resistance is not independent of the distance between them. The work is then extended to include moving cages. The effects of the passing of two cages in midshaft on the shaft resistance and on conditions upstream and downstream from the passing place, and the effect of a cage passing a point in the shaft on conditions at that point, and the possibility of reducing these effects by using straight-sided fairings are covered.

C O N T E N T S

Section 1. <u>I N T R O D U C T I O N</u>

Adequate ventilation of the workings is essential for the safety of any underground mining project and in Great Britain minimum standards are laid down by statute (Ref. 1.1). It is also important that the ventilation be supplied as efficiently and as economically as possible.

The ventilation energy is supplied by the main mine fan supplemented by auxiliaries, boosters and natural ventilation effects. An air fan works most efficiently when it is handling its design volume against its design pressure. Any attempt to operate the fan under different conditions results in a decrease in efficiency and loss of power. When it is considered that the main fan motor in a modern mine may have to develop 1000 horse-power the importance of keeping the fan near to its design duty becomes apparent. In making his choice of fan the engineer must estimate the quantity of air required to ventilate the mine satisfactorily and the pressure required to drive this amount of air through the mine. The quantity is estimated from a knowledge of gas emission rates, number of working places and output and the pressure by estimating the resistance to air flow of the various parts of the workings.

The proportion of the total fan energy absorbed in the shafts can be considerable because they carry larger quantities of air than any other links in the network of underground roadways. In an estimate of the ventilation requirements of a reorganised colliery undertaken by the author (1.2) it was shown that with a fan drift W.G. of 5.2 inches the pit bottom W.G. was 3.5 inches. The shafts were less than 1300 feet deep, only one of them had rigid guides and the air quantity was about 250,000 c.f.m. and yet the shafts made by far the greatest individual demand on the fan. Modern mines are planned for more intensive production from deeper levels with higher rock temperatures and so require more air. It has been estimated (1.3) that at least 50% of the fan power will be absorbed in the shafts of such a mine. Accordingly, inaccuracies in the estimation of shaft resistances will have more serious effects than inaccuracies elsewhere and reductions in shaft resistances will be of greater value than comparable reductions elsewhere.

The resistance of a shaft can be estimated by

1) experiments "in situ"
2) analytical methods based on knowledge of the characteristics of other similar shafts.
3) experiments on a model of the shaft.

Tests "in situ", if carefully conducted, should give the best results and it is interesting to note that a team of investigators (1.4) claim to have estimated the resistance of a shaft to ± 1% using the "full volume - reduced volume" method. Analytical methods have limited application due to the wide variety of shafts in existence. In many cases "in situ" tests are not possible and in such cases results from experiments on models can be very useful. Scale model testing of mine airways has been carried out by many investigators (1.5, 1.6) and the suitability of the technique for shafts has been fully discussed (1.7). In cases where comparisons have been possible between full scale installations and models degrees of agreement varying from 5% to 15% have been found.

The most profitable field for model testing has been in what has been called "systematic" investigations (1.8) in which the effect of successive modifications to an airway are studied. Models are particularly suitable for this type of work since features can be easily and cheaply altered and since scale effects are unlikely to seriously affect the relative values of resistance produced by different designs. This type of work has been done on shafts for bunton spacing, patterns, and streamlining of buntons but apart from the work of Stevenson (1.9) the resistance produced by cages has received no attention. Following "in situ" tests on No.5 Shaft, City Deep Ltd., South Africa it was stated (1.4) that the effects produced by cages were negligible. However, this shaft is rectangular, rough-lined, heavily timbered and 5,849 feet deep and it is understandable that the cages produced an insignificant increase in the total resistance. In his work Stevenson investigated the factors controlling the resistance to air flow of cages by tests on stationary models. He also investigated the possibilities of reducing the cage resistances by means of streamlining.

The purpose of the work described here was to extend this work to include the effects produced by the cages in motion, to compare this with stationary cage tests, and to establish the relative importance of the cage effects and the rest of the shaft resistance. Tests were carried out on moving and stationary model cages in a horizontal wind tunnel equipped as a model shaft. The movement of the cages relative to each other and to the shaft produced fluctuating air pressures which were recorded and related to the position of the cages in the shaft by apparatus specially built for the purpose.

Most investigators have expressed their results on shaft resistances as "friction coefficients" (f or λ) as found in the well known D'Arcy formula which can be used to calculate the head lost in any length of shaft of any diameter. In the case of cages, however, the resistance is not dependent on shaft length and it is found more suitable to quote cage resistance as Pressure Drop Coefficient (k) which is defined as the ratio of the loss in pressure due to the cage to the mean approach velocity head. Like "f" and "λ", "k" is dimensionless and will have the same value for any two systems which are dynamically similar. To maintain uniformity in this work shaft resistances are also given as P.D.C.'s although this requires to be qualified by a statement of the length and diameter of shaft to which the quoted value refers.

Confident correlation of the results of tests on models with full scale installations has always proved difficult due to the very high Reynolds Numbers common in mine shafts. Great power would be required to produce high enough air velocities in a model to make the Reynolds Numbers equal and even if this were possible it is likely that air velocities would exceed the value above which compression effects can no longer be ignored (1.10). It has been discovered, however, that once the Reynolds Number is high enough to ensure that the air flow is fully turbulent very little further change in the values of friction coefficients takes place, and the results corresponding to the highest Reynolds

Numbers obtainable in models have frequently been extrapolated to Reynold's Numbers corresponding to full scale conditions.

It has been assumed that this extrapolation gives a higher value of "f" for a shaft than is actually the case. An investigation by Le Roux and Chasteau (1.11) on a large model with a powerful fan in which Reynolds Numbers up to 2,500,000 could be obtained gave experimental proof of this and showed decreases of between 2.5% and 5% depending on the total resistance. As an alternative solution Jones and Hinsley (1.8) have shown experimental results in which better correlation was obtained by operating geometrically similar systems at the same mean velocity rather than at different Reynolds Numbers and extrapolating over the gap.

The experiments in this work were done at Reynolds Numbers in the range 250,000 to 450,000 and there was no measurable difference in "k" within the range. Extrapolation to higher Reynolds Numbers may produce slight differences in resistance but relative values should be unchanged and comparisons made should be valid.

The work includes details of the construction of a suitable recording manometer and a theoretical section dealing with the factors affecting its response to oscillating pressures.

Section 2. THE RECORDING MANOMETER - GENERAL

2.1. The Need for a Recording Manometer

In the section of his thesis entitled "Suggestions for Future Work" Stevenson (1.9) pointed out the necessity of extending his work to include moving cages in order to study the resistance to air flow due to cages in mine shafts under conditions which approached full scale conditions more closely. Stevenson suggested that this could be done using model testing techniques similar to those used by him in the study of the effects produced by stationary cages. Before such a test programme could be carried out it was necessary to provide a means for the measuring and recording of the fluctuations in air pressures which the moving of the cages was expected to produce. The pressures in the wind tunnel were in the range 0 to 14 inches water gauge with superimposed fluctuations of unknown amplitude and frequency. The amplitude was expected to be fairly small - about one inch water gauge - but it was not possible to estimate the probable frequency.

At the time when the investigations were begun (1956) no manometer capable of recording such small pressure fluctuations was in commercial production. It was therefore necessary to build one before the proposed test programme could be started.

2.2. Desired Characteristics of a Recording Manometer.

Ignorance of the nature of the pressure effects requiring to be measured made it difficult to decide upon the characteristics required by the Recording Manometer. The important factors to be considered in the design are the response, the sensitivity, the accuracy and the utility. The response should be linear and undistorted to static or transient pressures. This can only be achieved by keeping the natural frequency of the instrument high. Any moving parts must therefore be light and any straining members must be stiff. The sensitivity should be such that a reasonable range of low pressures can be measured with a single gauge. The calibration should be unaffected by time and by external conditions such as vibration and temperature and the instrument errors must be kept to a minimum. It has been suggested (2.1) that the allowable variance should be 2% of the total range. Other desirable features which should be provided, if possible, are simplicity of manufacture, installation and operation, and the provision of means for indication or recording at stations remote from the point of installation.

2.3. Methods of Recording Fluctuating Air Pressures.

The recording of fluctuating air pressures requires the provision of a mechanism capable of converting pressure into a quantity suitable for input to a recorder. The methods (2.2) of doing this can be divided into two broad classes - mechanical and electrical.

In mechanical pressure recording the pressure acts upon an element producing a displacement which may be magnified by a system of levers before being recorded, usually by a pen, on moving paper. The nature of the method makes it more suitable for, and more generally applied to, the recording of high pressures in the low frequency range.

The most common example of its use is found in the spring operated engine indicator of the type first built by James Watt. The method has been applied (2.3, 2.4, 2.5) however to air pressures in the range generally found in ventilation practice. The technique employed is to use a light diaphragm which responds to the pressure changes and controls by its movement a mirror, which is part of an optical system which moves a spot of light over photographic film.

In electrical pressure recording the pressure sensitive element forms a component in an electrical circuit. The effect of the pressure is to alter the electrical properties of the element and so produce a change in the conditions in the circuit. The circuit is so arranged that this change will produce an electrical output suitable for use with a cathode ray oscilloscope or a galvanometer recorder. The electrical change produced by the pressure may be resistive, inductive, electrostatic or piezo-electric. Resistance type pressure transducers use wire strain gauges cemented to an elastic element which strains under the pressure (2.1) and are particularly suitable for measuring pressure pulsations, in pipe lines (2.6). Variable inductance type transducers operate by allowing the pressure to vary an air gap in a magnetic circuit, so varying the inductance of a coil. Variable capacitance type transducers utilise the change in capacitance which results when the spacing between the plate of a condenser is altered (2.7, 2.8) while the piezo-electric type utilise the electric polarities developed when crystals of certain minerals are subjected to pressure (2.9).

It has been suggested (2.10 - 1940) that electrostatic type gauges are more suitable for high frequency phenomena and electromagnetic types for lower frequency. Resistance types are a more recent development, resulting from the improvement in design and properties of strain gauges, which has taken place in recent years.

Mechanical types are generally the simpler but electrical methods have a wider application and are especially required when pressures have to be recorded at a point remote from the installation of the manometer.

2.4. Pressure Sensitive Elements

The most widely used element in instruments used for measuring pressures in the range with which this work is concerned, is the metallic diaghragm. German silver, beryllium copper, phosphor bronze and nickel alloys have all been used (2.11, 2.12). Diaphragms may be flat or they may have corrugations spun, pressed or stamped into them in order to improve their load bearing properties. Some instruments incorporate a pair of diaphragms forming a capsule and bellows (2.13) and Bourdon tubes have also been used.

Diaphragms have the advantages of being very compact, simple to manufacture and relatively cheap. By making a suitable choice of pattern, material, thickness and diameter it is possible to produce a diaphragm capable of measuring any required pressure. A variable capacitance type diaphragm manometer capable of measuring (but not recording) pressures of 0.001 mm. of mercury (about 2×10^{-5} lb/sq.in.) has been built (2.14), while several commercial manufacturers use diaphragm type gauges for pressures as high as 10,000 lb/sq.in.

A diaphragm can be made more sensitive by reducing its thickness and increasing its diameter. However, this lowers the natural frequency and restricts the frequency range in which the diaphragm can be used to record oscillating pressures. Due to their ease of manufacture and availability of the materials, flat brass diaphragms were used exclusively in our experimental work on the development of the recording manometer.

Section 3. THE RECORDING MANOMETER - THE STRAIN GAUGE DIAPHRAGM

3.1. Electric Resistance Strain.

The basic element in this type of transducer is the electrical resistance strain gauge (Plate 3.1) which consists of a continuous resistance wire bent to a suitable pattern and fixed to a paper backing. The gauge is cemented to the diaphragm or other sensitive element and the strain produced by the pressure is transmitted to the gauge. The effect of the strain on the wire is to increase its length and reduce its diameter causing an increase in its electrical resistance. The increase in resistance has been found to be slightly greater than would be expected from calculations of the change of dimensions, and it has been suggested (3.1) that this is due to a change in the specific resistance of the material when it is strained. The strain sensitivity factor of the gauge is defined as :-

$$f = \frac{\text{proportionate change in resistance}}{\text{mechanical strain}}$$

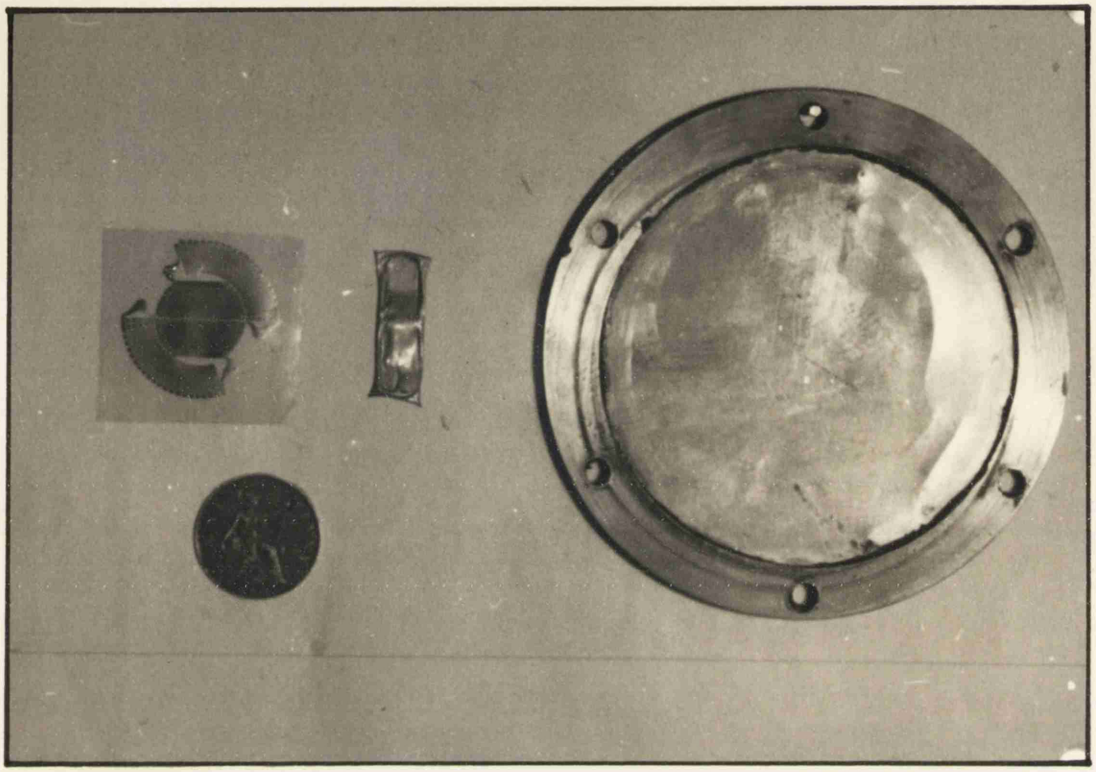


PLATE 3.1. STRAIN GAUGES AND BRASS DIAPHRAGM.

For most types of wire gauge its value is about 2.2.

The absolute change in resistance is very small and it is necessary to incorporate the gauge in a bridge circuit (Fig. 3.1). The simple one-to-one bridge (a) is more common. The dummy gauge is made identical to the active gauge and is cemented to a part of the assembly of about the same mass as the straining part, thereby giving automatic compensation for any changes in temperature. The sensitivity of the arrangement can be increased if the dummy gauge is cemented to a part of the member which experiences a strain opposite in sign to the active gauge, and by using further straining gauges as the ratio arms. The bridge sensitivity can be calculated (reference to part b of Fig. 3.1) as follows.

$$\text{Voltage Sensitivity} = e = \frac{S \times f \times E \times m}{(m + 1)^2}$$

$$\text{Current Sensitivity} = i = \frac{S \times f \times I \times m}{\frac{Rg}{R}(m + 1) + m(n + 1)}$$

where :-

R = resistance of unstrained gauge (ohms)

E = bridge voltage (volts)

m, n, = bridge constants

f = strain sensitivity factor

S = mechanical strain

I = gauge current amps.

e = bridge output (volts) on open circuit

i = bridge output (amps) through instrument of resistance Rg.

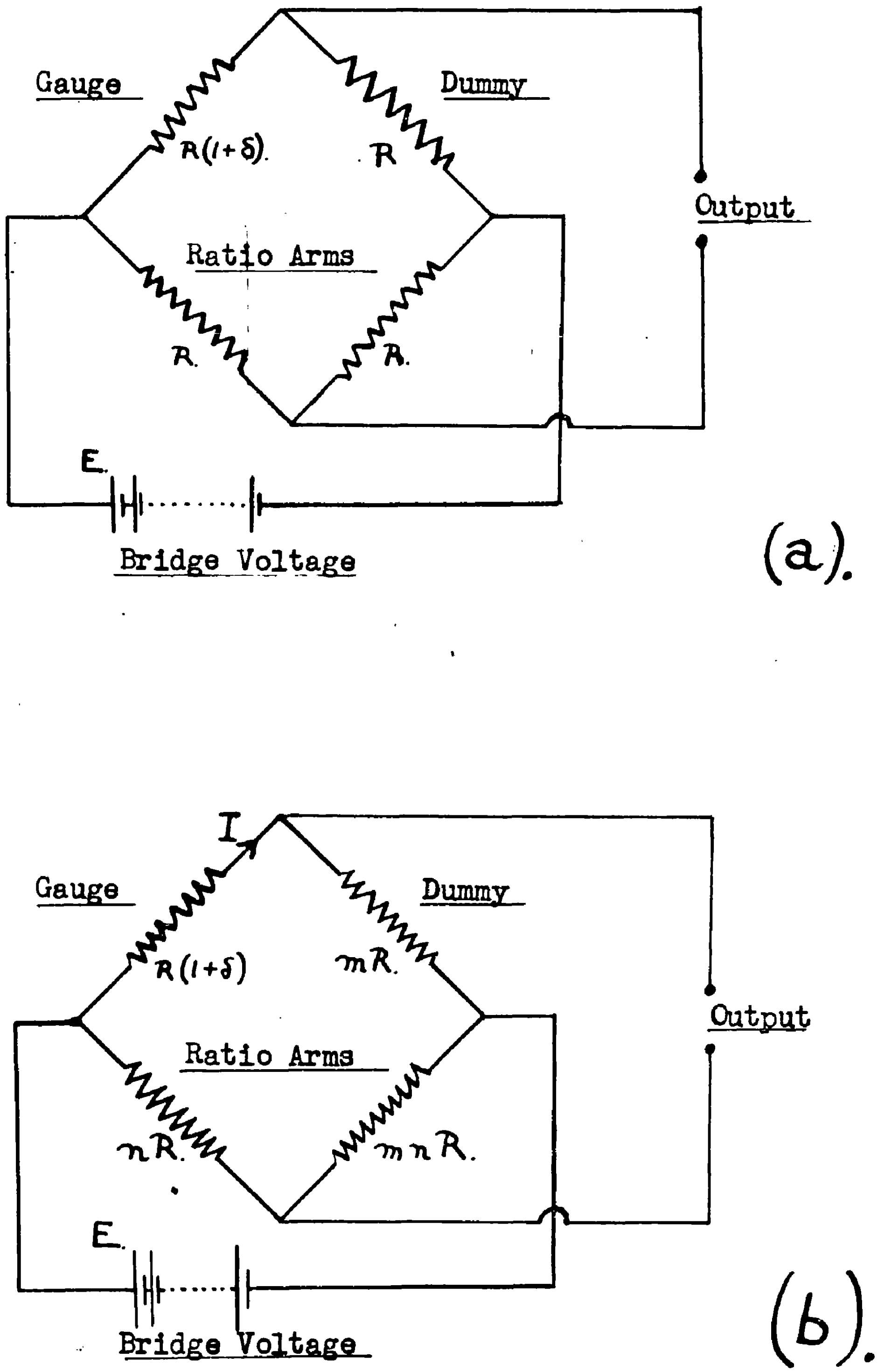


FIG. 3.1. STRAIN BRIDGES

Consideration of these expressions suggests that to obtain maximum current sensitivity (for a given gauge voltage) "m" should be large and "n" should be small but the limits to which this can be carried are strict, since such a bridge would require a dummy gauge and two ratio arms of high dissipating power, and would make a heavy power demand on the supply. Both these factors will adversely affect the steadiness of the bridge. If the strain bridge has to be used for recording dynamic strains then the external resistance presented to the galvanometer by the bridge has to be adjusted to arrange for suitable damping of the galvanometer (3.2). Such an arrangement may require resistors in series or parallel with the galvanometer reducing the sensitivity of the bridge.

3.2. Initial Work.

The initial work in the attempt to use a strain gauge diaphragm to measure fluctuating pressures was undertaken by Thomson (3.3), who designed and built two diaphragms. These were of brass 0.0015 inches thick and 4 inches in diameter clamped between steel rings using Araldite 103 as a fixative. Thomson experienced difficulty in making a tight flat diaphragm and so eliminating a sudden deflection of the diaphragm before the stress-strain relationship began. This sudden deflection he named the "Click Effect" and overcame it by assembling the diaphragm after heating the parts under an infra red lamp and allowing the differential expansion of the steel and the brass to tighten the diaphragm.

On one of these diaphragms Thomson mounted eight 2400 ohm

De Havilland one inch strain gauges. These were placed radially, four on each side at 90 degree intervals and were connected in pairs in parallel to form the four arms of a strain bridge. On the other he mounted four "home made" radial gauges covering the entire surface of both sides. Due to lack of time he did not attempt a calibration.

3.3. Further Development.

The output from the strain bridges on Thomson's diaphragms could be detected on a sensitive spot galvanometer, but it was much too low to give a satisfactory output on a cathode ray oscilloscope or a galvanometer recorder and an attempt to build a sufficiently stable direct current amplifier was not successful. At this stage it was doubtful if these diaphragms would give satisfactory results and a more theoretical line was attempted.

Theory of Diaphragm Straining

The theory for the straining of flat circular plates under uniform pressure was developed by Grashof (3.4) and has been established experimentally for central deflection (3.5) using a micrometer and for surface strain (3.6) using electric resistance strain gauges. Other experiments (3.7) showed that Grashof's theory was limited to cases where the central deflection is very small compared with the thickness. A more accurate (and much more difficult) theory has been given (3.8) and has been experimentally proved (3.9).

Grashof's Theory is inadequate in that it considers bending only and assumes that no strain exists on the neutral plane in the centre of the plate material at right angles to the load and that the stress varies symmetrically through the thickness of the plate from compression on the high pressure side to tension on the low pressure side. In the design of the strain gauge diaphragm it was considered very desirable that the output signal from the strain bridge should vary linearly with the pressure difference across the diaphragm. The strains produced by the bending of the diaphragm have this property but as the deflection increases other strains have to be considered. Grashof's theory shows that in the range where only bending is important the strain is inversely proportional to the square of the thickness. It follows that to obtain the best conditions the diaphragm should be as thin as possible without losing its lenear properties at its maximum working pressure. Different writers vary in their opinions as to the point at which Grashof's Theory breaks down and it was decided to carry out linearity tests on some selected diaphragms to find out their useful range.

<u>Tests.</u>

The diaphragms tested were made from brass foil clamped between steel rings 4 inches internal diameter (Plate 3.1). The brass was kept tight by assembling the diaphragms under heat. For foil 0.0015 inches thick the heat from an infra red lamp proved sufficient but for thicker foils it was necessary to use steam to provide a high enough temperature to ensure that the brass would tighten satisfactorily on cooling. Four diaphragms were tested - one of those built by Thomson, another also 0.0015 inches thick and two each 0.005 inches thick. One one-inch

600 ohm gauge was fixed (3.10) on each of the new diaphragms at varying radii. Differential pressures were applied to the diaphragm and simultaneously to a Betz Projection Manometer (See Plate 6.5). The resulting strain in the gauge was picked up either with a Phillip's Model GM5536 A.C. Strain Bridge or by wiring the gauge into a bridge with suitable 5 watt rated resistors and reading the out of balance current on a sensitive galvanometer. The pressure across the diaphragm could be varied from zero to about 12 inches W.G. and provision was made for regular checking of the strain bridge zero.

Results.

The tests showed that the linear relationship between the strain on the 0.0015 inch thick diaphragms broke down at pressures greater than 1 inch W.G. while on the 0.005 inch thick ones the breakdown occurred around 5 inches W.G. when the gauge was in compression and around 8 inches W.G. when in tension.

The tests also showed that at pressures higher than 1 inch W.G. the strain on both surfaces of Thomson's 0.0015 inch thick diaphragm was of the same sign and not opposite as he had supposed. This proved conclusively that in the case of such thin diaphragms the strains produced by stretching of the material were of a higher order than those produced by bending and that a linear pressure-strain relationship could not be obtained. In the case of the 0.005 inch thick diaphragms the bending strains seemed to predominate up to pressures sufficiently high to make a satisfactory pressure-strain relationship possible. No further thicknesses were tried.

The actual level of strain achieved in these tests was very low and while it could be picked up easily enough on a sensitive spot galvanometer the current output from the bridge was too low to operate a galvanometer recorder which is much less sensitive because its design (3.11) needs a smaller core and smaller number of turns on its moving coil to ensure a high natural freqency.

Calculations showed that if a deflection of 1 cm. was wanted on the available galvanometer recorder a strain of about 700×10^{-6} would be required at the gauge. If all four arms of the bridge could be made active then one quarter of this strain would do. According to Grashof's Theory the strains on the experimental diaphragms should have been about 100×10^{-6} per inch W.G. but the actual strains picked up were very much less, in most cases only about one tenth of this. The cause of the discrepancy is not certain but it seems that either 0.005 inches is still too thin to allow application of the theory to a 4 inch diameter diaphragm or else local strengthening of the thin brass foil by the gauge backing and the layer of bonding cement upset the normal pressure-strain relationship in a complex way.

From these tests it was concluded that strain gauges on the surface of these diaphragms would not give a high enough output to feed the available galvanometer recorder without electrical amplification. Stable amplification of the very small direct current output would prove very difficult and an alternative method was sought.

3.4. Unmounted Strain Gauges.

In an attempt to produce a higher level of strain on the gauge it was decided to investigate the case of a diaphragm straining a gauge stretched at right angles to the plane of the diaphragm between the diaphragm centre point and a rigid support bar.

Theory of Plate Loaded Uniformly with Central Support.

Considering the theory of this case it was felt that since the action of the gauge would reduce the central deflection of the diaphragm to a very small value then Grashof's Theory was likely to apply.

The theory gives :-

Free deflection of plate under uniform load $= W_p$

$$W_p = \frac{3}{16} \frac{m^2 - 1}{m^2} \frac{p r^4}{E t^3}. \text{ inches}$$

and Free deflection under concentrated load at centre $= W_t$

$$W_t = \frac{3}{4} \frac{m^2 - 1}{m^2} \frac{W r^2}{E t^3} \text{ inches}$$

where p = the uniform pressure (lbs. per sq.in.)

W = the concentrated load (lbs.)

$\frac{1}{m}$ = Poisson's Ratio

E = Young's Modulus (lbs. per sq.in.)

r = radius (ins.)

t = thickness (ins.)

For plates of any one material under a fixed pressure we get

$$W_p = A \frac{r^4}{t^3} \qquad \text{where } A = \text{constant}$$

$$W_t = B \frac{Wr^2}{t^3} \qquad \text{where } B = \text{constant}$$

The two cases were then combined as follows :-

If rigidity of the support = e inches per lb. then theoretically

$$W_p = W_t + We.$$

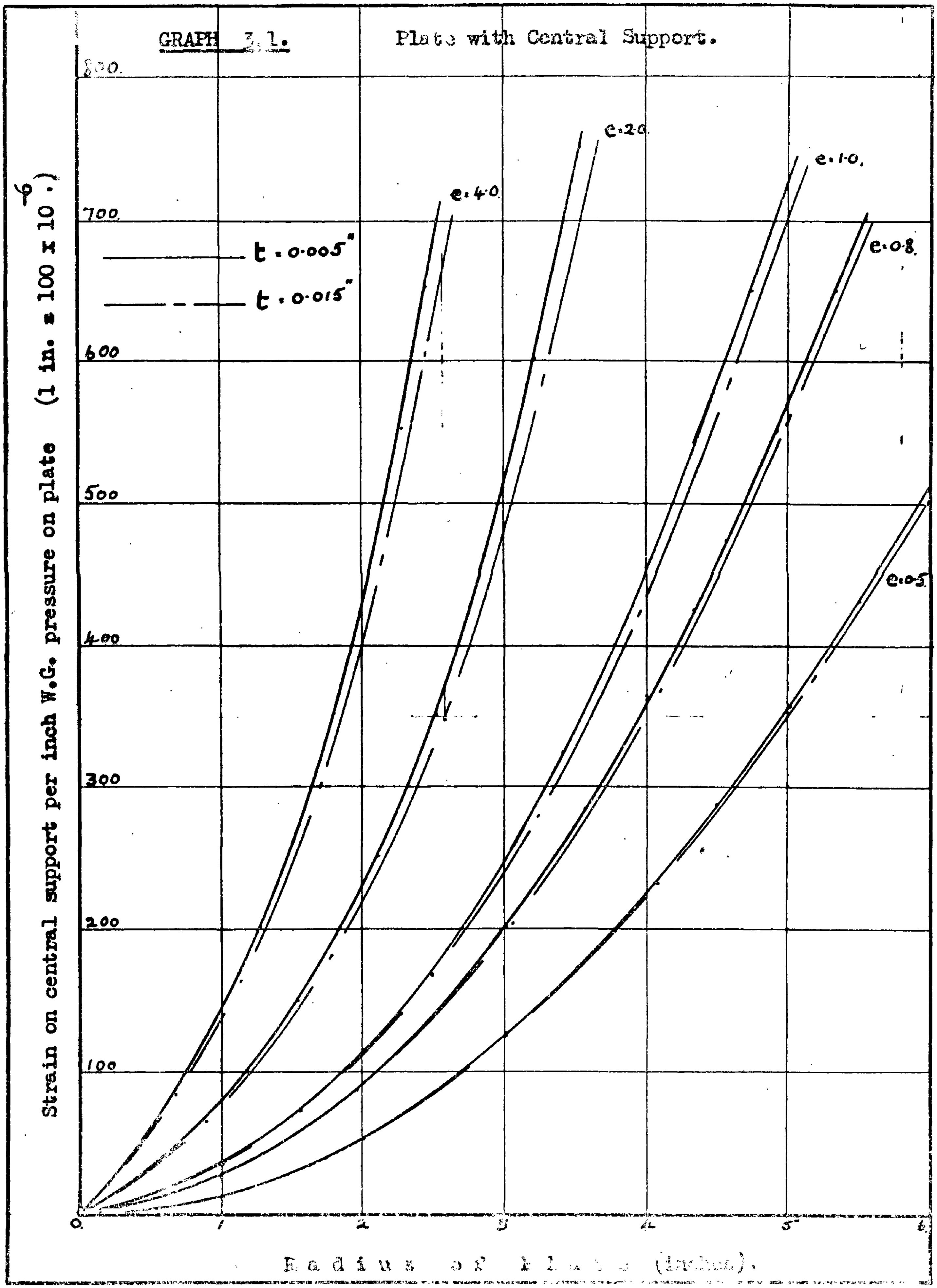
$$\therefore W = \frac{Ar^4}{Br^2 + et^3}$$

and Strain on gauge = $S = \frac{We}{L}$

where L = length of gauge in inches.

$$\therefore S = \frac{Ar^4 e}{(Br^2 + et^3)L}$$

The values of A and B for a brass diaphragm under a pressure of 1 inch W.G. were calculated and used to estimate the value of strain likely to be found on a one inch gauge fixed to the centre point of a diaphragm of varying radius and thickness. The results are shown on Graph 3.1 and indicate that the strain increases with radius for a given gauge strength and increases with a fall in gauge strength (i.e. a rise in "e") at a given radius. The most significant point is the very small effect of varying diaphragm thickness except with the weakest gauges. The first tests with perpendicular mounting were carried out on the same diaphragms as the surface strain tests. A light stirrup was fixed to the centre point of the diaphragm and a one inch Saunders Roe type foil gauge was stretched between the stirrup and a rigid bar placed above the diaphragm. Pressures were applied to the diaphragm and the strain



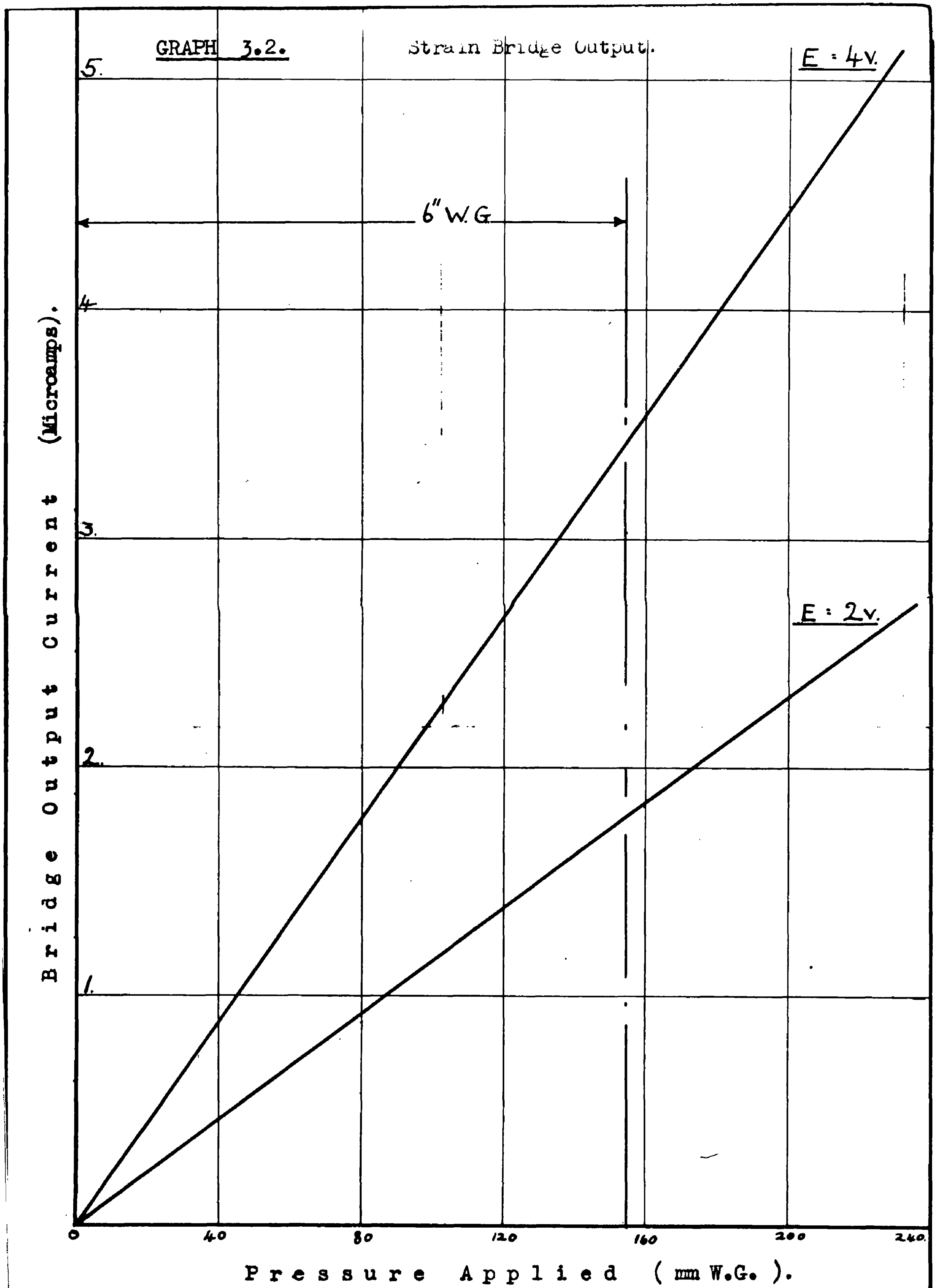
GRAPH 3.1.
Plate with Central Support.
800.
700.
600
500
400
300
200
100
0.
1
2
3
4
5
6.
e.4.0.
e.2.0.
e.1.0.
e.0.8.
e.0.5.
t.0.005"
t.0.015"
Strain on central support per inch W.G. pressure on plate (1 in. = 100 x 10⁻⁶.)
Radius of Plate (inches).

measured. The mechanical strength of the gauge (e) was found from a separate test by detatching the stirrup from the diaphragm and suspending weights on it. Graph 3.2 shows the values of bridge current picked up in the pressure range 0 to 11 inches W.G. Good linearity and stability was obtained. The mean value of strain picked up in these tests was 82×10^{-6} per 1 inch W.G. pressure. This compared very favourably with the theoretical value which was 90×10^{-6} per 1 inch W.G. and with about 10×10^{-6} per 1 inch W.G. which was the highest strain level measured on the diaphragm surface. It was still much less than the 700×10^{-6} per 1 inch W.G. which was required. From the theoretical curves it was obvious that a larger diameter diaphragm was needed. 12 inches was selected and the thickness was increased to 0.018 inches to maintain a high natural frequency and to ensure that the theory would be more strictly applicable. A slightly different type of strain gauge with strippable backing had to be used. The strain level produced was about 250×10^{-6} per 1 inch W.G. but the bridge was not very stable at the voltage which was required. This was probably due to the different thermal characteristics of the unmounted gauge and the mounted dummy.

At this point strain gauge tests were discontinued because better results were being obtained with a variable capacitance type transducer which was being developed simultaneously.

3.5. Conclusions.

1) The level of mechanical strain produced on the surfaces of the



GRAPH 3.2.
Strain Bridge Output.
E = 4v.
E = 2v.
6" W.G.
5.
4
3.
2.
1.
0
40
80
120
160
200
240.
Bridge Output Current (Microamps).
Pressure Applied (mm W.G.).

diaphragms tested by pressures of 0 to 8 inches W.G. is not sufficient to operate a simple direct current bridge and a recording galvanometer as a means of recording pressure fluctuations.

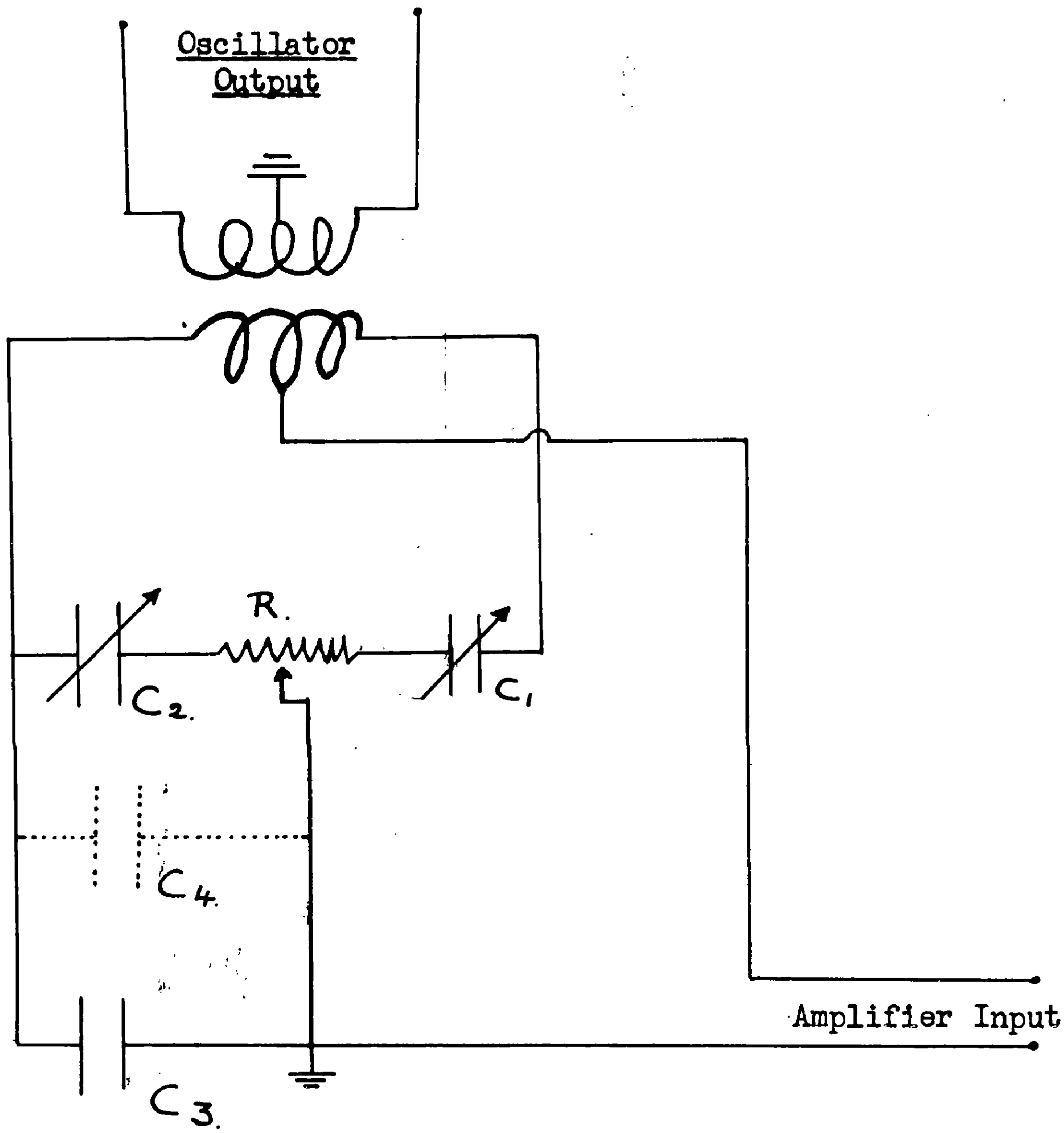
2) Amplification of the d.c. output may be possible but will require bulky and expensive equipment built to match both the strain bridge and the galvanometer and carefully designed to avoid interference.

3) The theory suggested for the strain on a gauge stretched at right angles to the plane of a diaphragm agrees well with practical results. This method gives much higher strain than is found on the diaphragm surface but requires great constructional skill to avoid drift and hysteresis. The potential sensitivity of this method is adversely affected by the difficulty of arranging for more than one active (i.e. straining) gauge.

Section 4. THE RECORDING MANOMETER - USE OF VARIABLE CAPACITANCE

4.1. Equipment Used.
The Fielden Proximity Meter. (Plate 4.1)

This instrument provides an electronic means of measuring very small mechanical displacements. It incorporates a mains rectifier circuit which feeds an R.F. oscillator which in turn supplies the A.C. bridge shown in Fig. 4.1. In this unit the variable internal components C_1, C_2 and R are combined with an external condenser C_3 to form a bridge which can be completely balanced (i.e. in voltage and in phase) by the slow motion drive controls. The oscillator output is split into two



C_1 - coarse balance control (100 pf.)

C_2 - fine balance control (5 pf.)

R - phase balance control (300 ohms)

C_3 - external variable capacitor

C_4 - external stray capacitance

FIG. 4.1. PROXIMITY METER BRIDGE UNIT CIRCUIT

earth free, equal and antiphase voltages at the transformer. These are fed to earth, one through each side of the bridge. When the bridge is balanced the voltage at the centre tap on the transformer is zero and there is no input to the amplifier. Any change hereafter in the value of the external condenser produces an out of balance voltage which is amplified to form the instrument output.

To operate the instrument it is necessary to build a condenser (C_3) whose capacitance is varied by the physical quantity requiring to be measured. This is usually done by varying the distance between two plates but variations in the effective area of the plates or in the dielectric properties of the matter between the plates can also be used. This condenser is arranged with one plate earthed and the other connected to the bridge through a double screened cable. The length of this cable has to be kept to a minimum to reduce external stray capacitance which would reduce the potential sensitivity of the bridge.

The output of the instrument varies linearly with the change in capacitance at the end of the screened cable. Full output is an alternating 10 volts R.M.S. across 10,000 ohms which can be recorded by a cathode ray oscilloscope across the output terminals. The current in the output resistance is full wave rectified and deflects the meter on the panel. A jack socket in the circuit makes it possible to connect a galvanometer recorder in series with this meter as a means of measuring the output. The output circuit is shown in Fig. 4.3.

Two models of this instrument, the P.M.2 and the P.M.4 are in commercial production. In the P.M.4 the measuring bridge unit

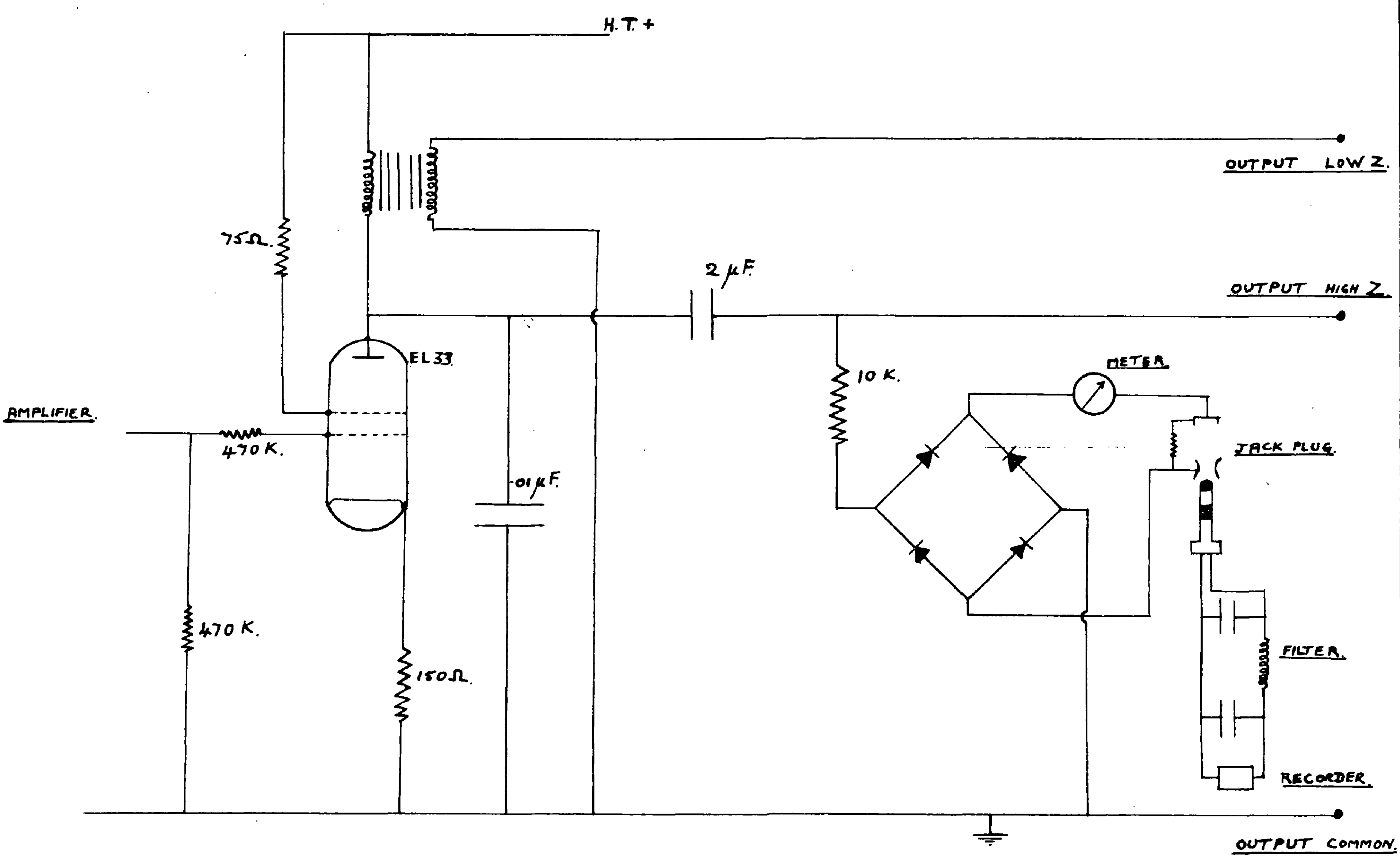


FIG. 4.3. PROXIMITY METER OUTPUT CIRCUIT

is housed in a box separate from the other components. This helps to reduce the temperature variation of the bridge components which is the principal cause of drift of the instrument zero.

The A.R.L. 12-Channel Galvanometer Recorder (Plate 4.2).

Developed at the Admiralty Research Laboratory at Teddington this instrument is designed to record up to 12 signals simultaneously. Light from a lamp filament is reflected from the mirrors of 12 galvanometer elements in a common magnet block and focussed on 70 mm wide film moving through the light gate. Film speeds of $\frac{5}{8}$, $1\frac{1}{4}$, $2\frac{1}{2}$, and 5 inches per second are available. There are two types of element whose sensitivities are 80 micro amps. per cm (natural frequency around 140 c/s) and 13 micro amps per cm (natural frequency around 60 c/s).

Cossor Double Beam Oscillograph (Plate 4.3).

This is a standard laboratory model. The screen is 4 inches diameter and shows the positions of the two halves of a split electron beam. The Y plates for each half of the beam are driven by separate amplifiers enabling two independent voltages to be measured simultaneously. The sweep of the time base can be varied from 0.05 to 1500 milliseconds. A camera is available which records the spot movements on 35 mm film moving at speeds from 0.05 to 25 inches per second.

Variable Capacitance Diaphragm Gauges.

The diaphragms used were of thin brass clamped tightly and flat between steel rings and were built by the same method as those used in the tests with strain gauges. Perspex cover plates with tappings were used to make the diaphragms sensitive to differential air pressures.

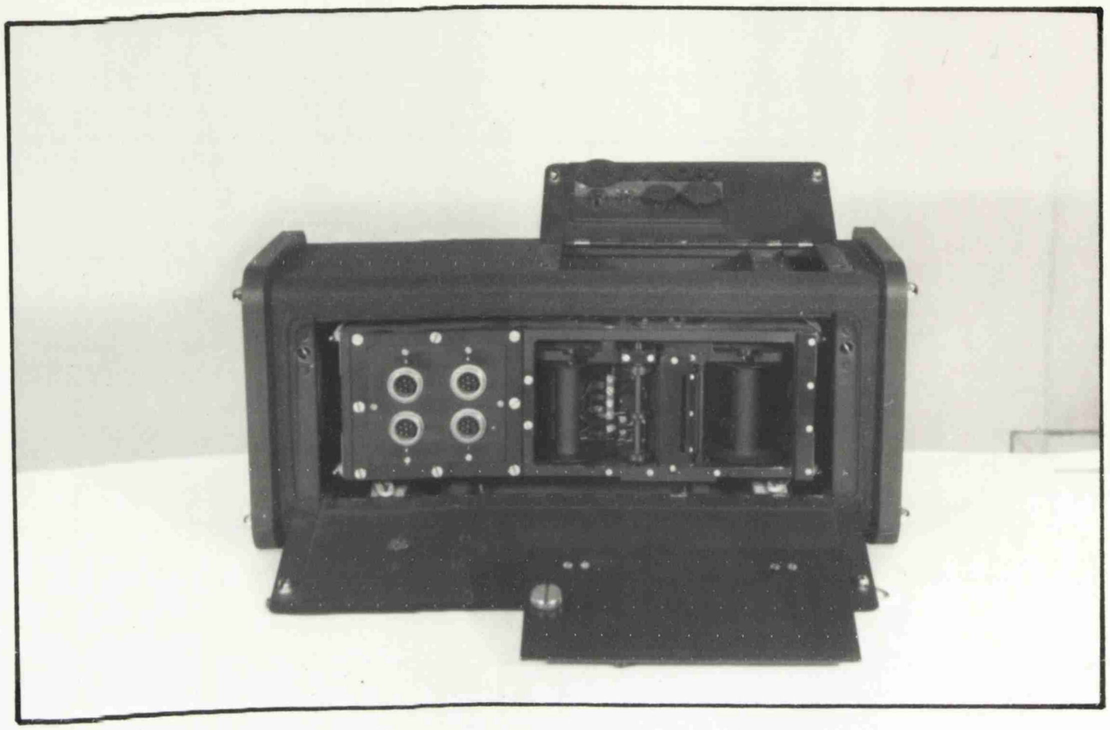


PLATE 4.2. A.R.L. GALVANOMETER RECORDER.

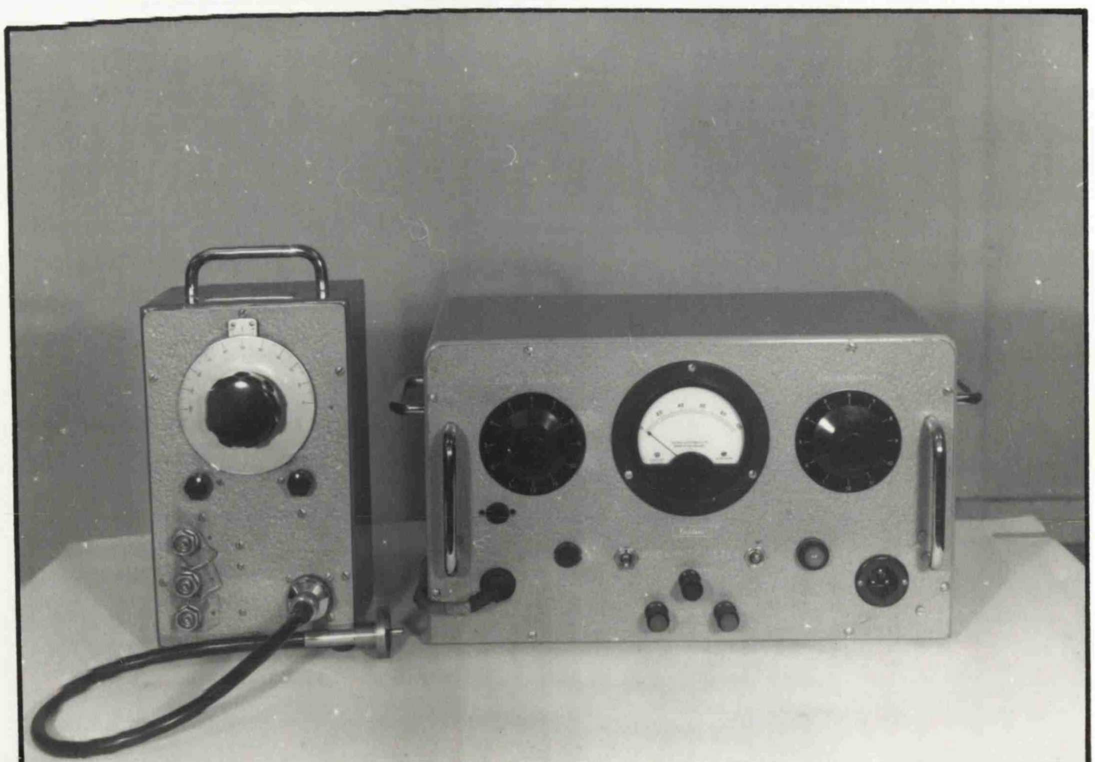


PLATE 4.1. FIELDEN PROXIMITY METER TYPE PM4.

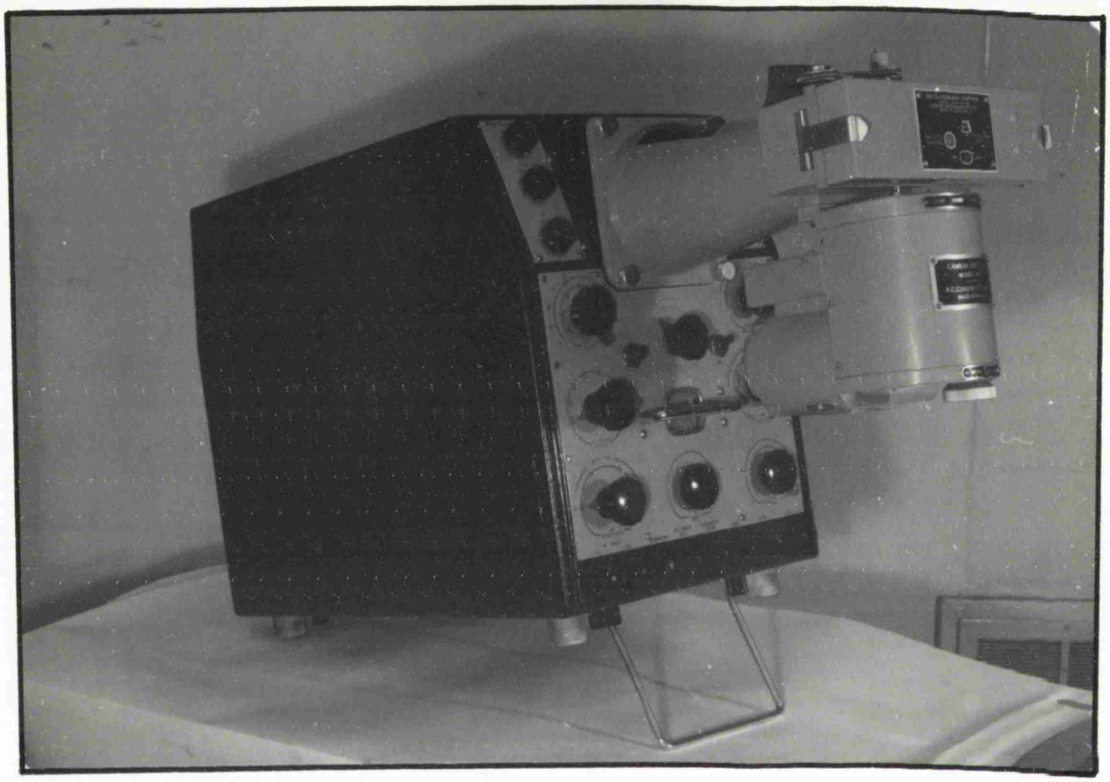


PLATE 4.3. COSSOR DOUBLE-BEAM OSCILLOGRAPH & CAMERA.

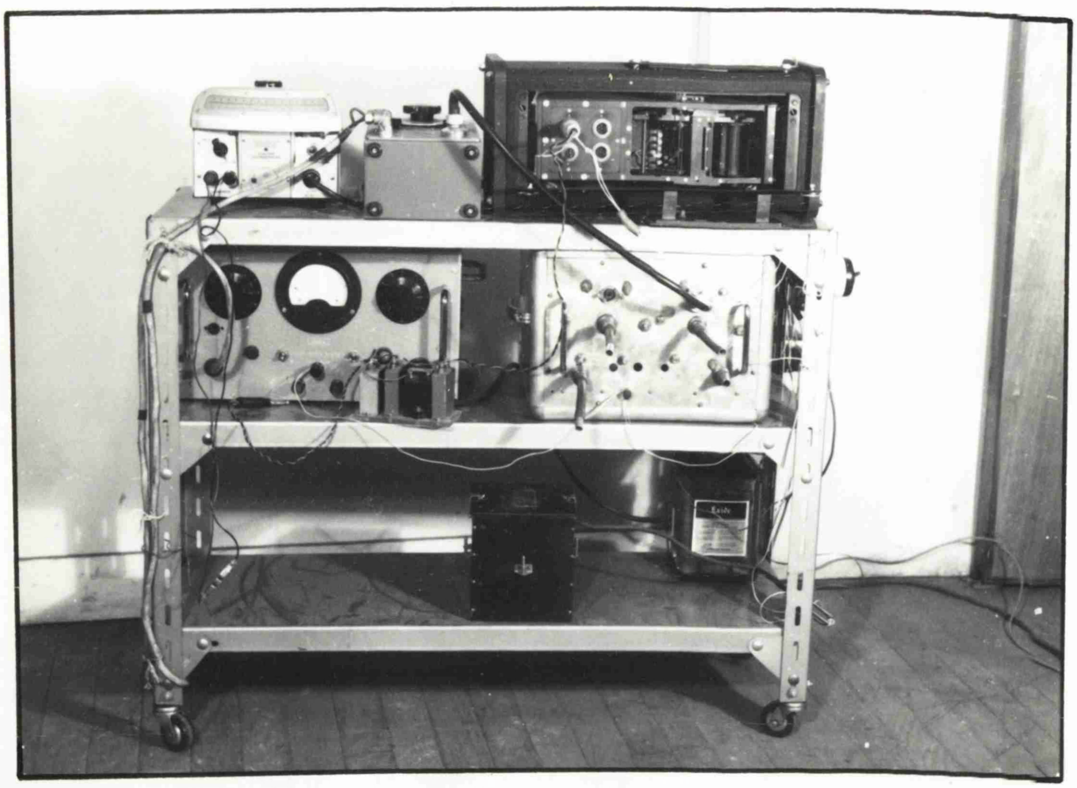


PLATE 4.4. INSTRUMENT TROLLEY.

In operation the diaphragm was earthed and a flat brass disc, concentric with the diaphragm and parallel to it was rigidly fixed a short distance away. This arrangement formed the variable condenser (C_3 Fig. 4.1) in the Proximity Meter bridge circuit. Under pressure the diaphragm is deflected towards the fixed plate producing the necessary change in capacitance. This method is particularly useful since the sensitivity of the gauge can be controlled within very wide limits by varying the design constants thus -

Let A = area of condenser plates

d = distance between plates

then capacitance $C_o = B \dfrac{AK}{d}$

where K = dielectric constant of the material separating the plates (= 1 for air).

B = constant.

if one plate moves distance "w" where $w < d$

then new capacitance $C_1 = B \dfrac{AK}{d-w}$

$\therefore$ Change in capacitance $= dC = C_o - C_1$

$$= C_o \frac{w}{d - w} \qquad (1)$$

The sensitivity can be increased by making "A" large, "d" small and "w" large. Limits are placed by the natural frequency required and by the necessary degree of linearity in change of capacitance with change of pressure. Natural frequency decreases as "A" is increased and as "w" is increased by reducing diaphragm thickness. From (1) it is seen that the change in capacitance varies hyperbolically with the deflection of the diaphragm. An approximation to linearity can be obtained by keeping "w" small compared to "d" and so working only on a small portion of the curve. The two curves on Graph 4.1 show the effect of limiting "w" to 0.5d and to 0.1d respectively. If the sensitivity requirements of a gauge are such that "w" has to be large then the linearity of the output can be improved by placing a layer of mica in the air gap between the plates. The exact thickness of mica required has to be found by trial and error.

4.2. Experimental Work.

The preliminary tests were carried out using a P.M.2 model feeding the oscilloscope. A diaphragm 4 inches in diameter and 0.005 inches thick was used in conjunction with circular plates of varying diameters fixed at different distances from it. Some of the results were plotted on Graph 4.2 and indicated that for a reasonably linear response over a pressure range of 6 inches W.G. the plate would have to be at least 5 mm. from the diaphragm. The most suitable place for the fixed plate was bolted on the inside of the perspex cover plate (See Fig. 4.2). The plate was then separated from the diaphragm by the thickness of the metal clamping ring and since it was cut slightly smaller than the inside diameter of the ring it was insulated from it by the perspex. The central core of the proximity meter cable was connected to the plate by a screened plug passing through the perspex. Interference from outside sources was

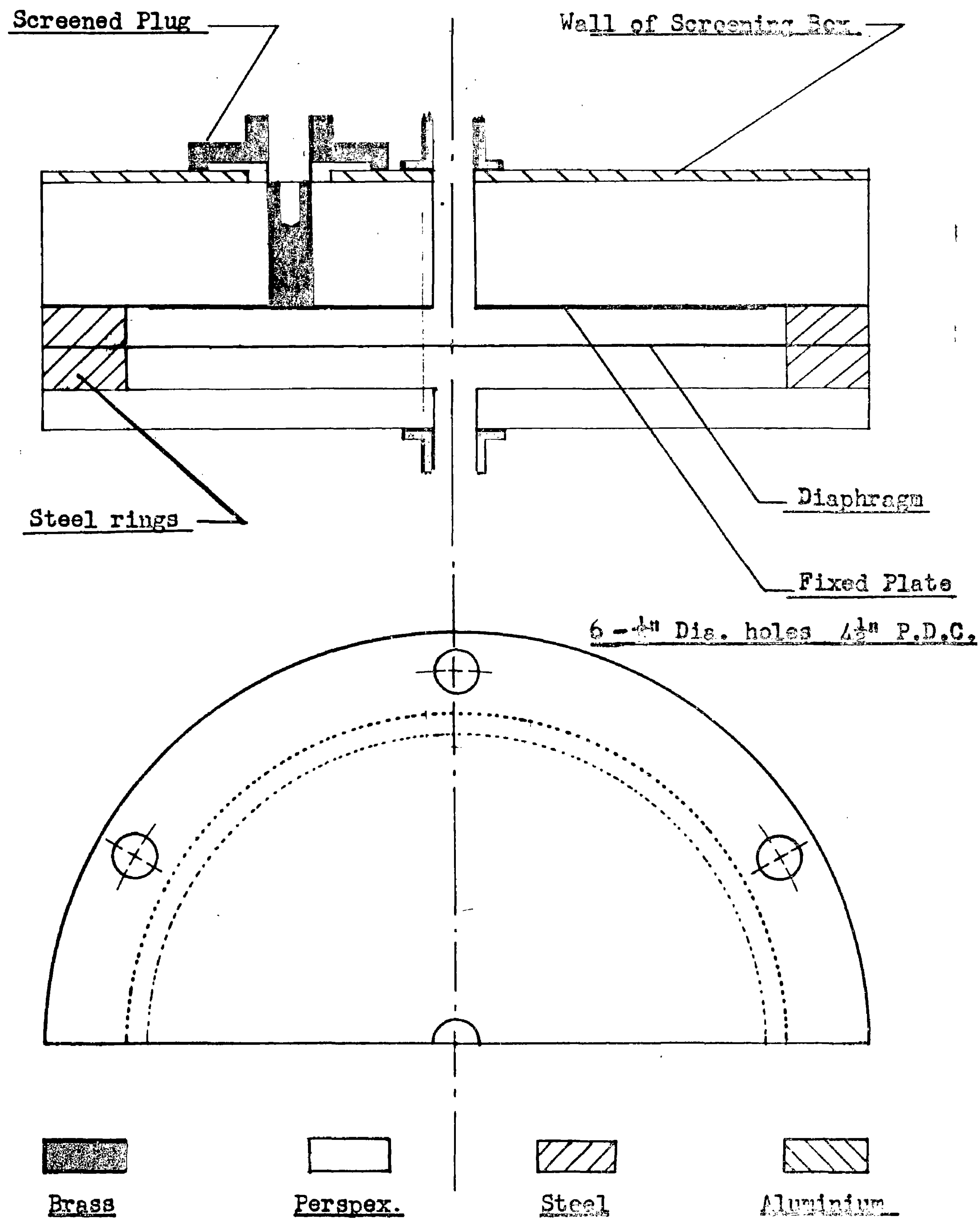
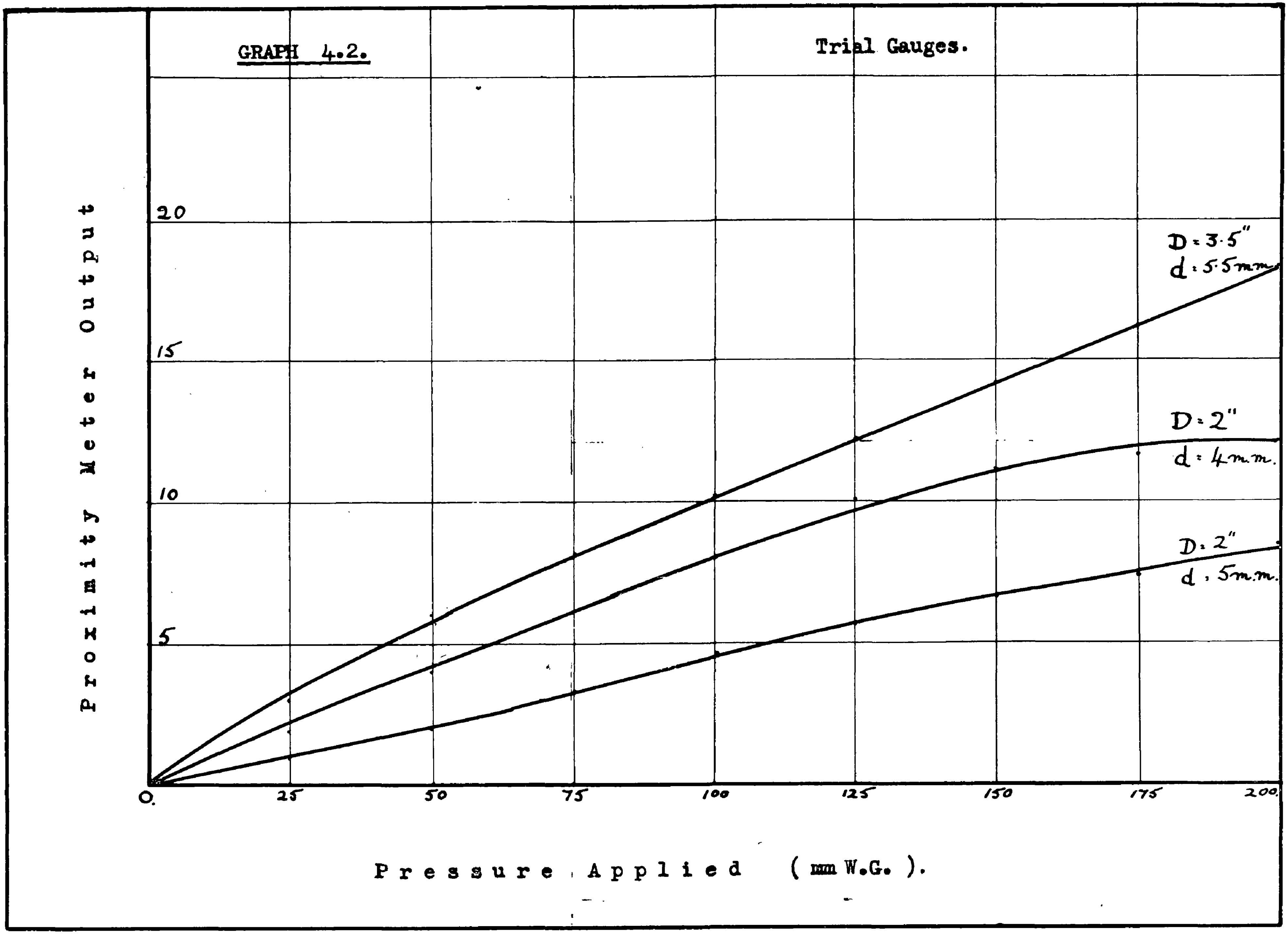


FIG. 4.2. DETAIL OF DIAPHRAGM GAUGE FULL SIZE


GRAPH 4.2.
Trial Gauges.
Proximity Meter Output
Pressure Applied (mm W.G.).
0.
25
50
75
100
125
150
175
200.
5
10
15
20
D = 3.5"
d = 5.5 mm.
D = 2"
d = 4 m.m.
D = 2"
d = 5 m.m.

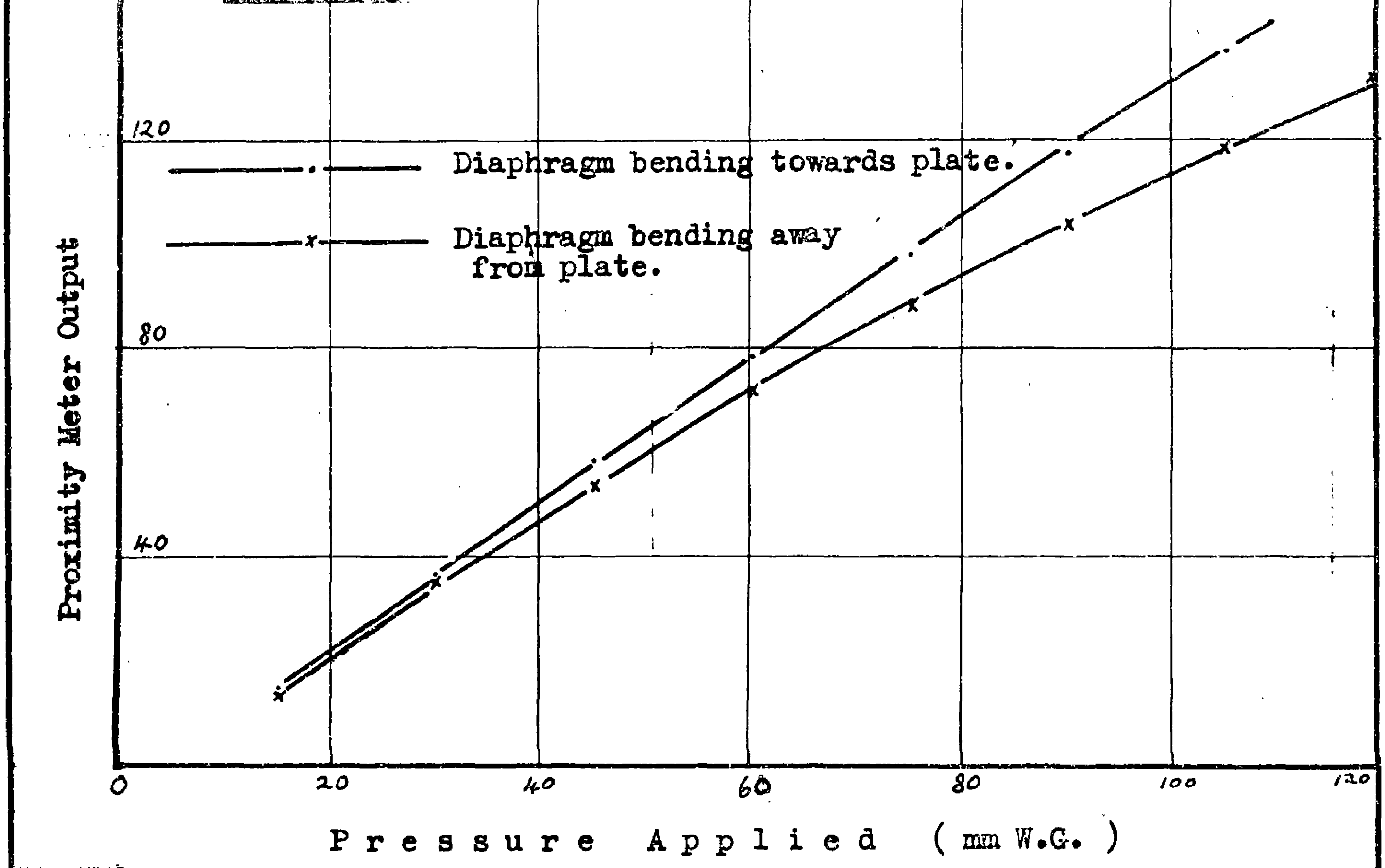
reduced to a minimum by enclosing the diaphragm in a screened box.

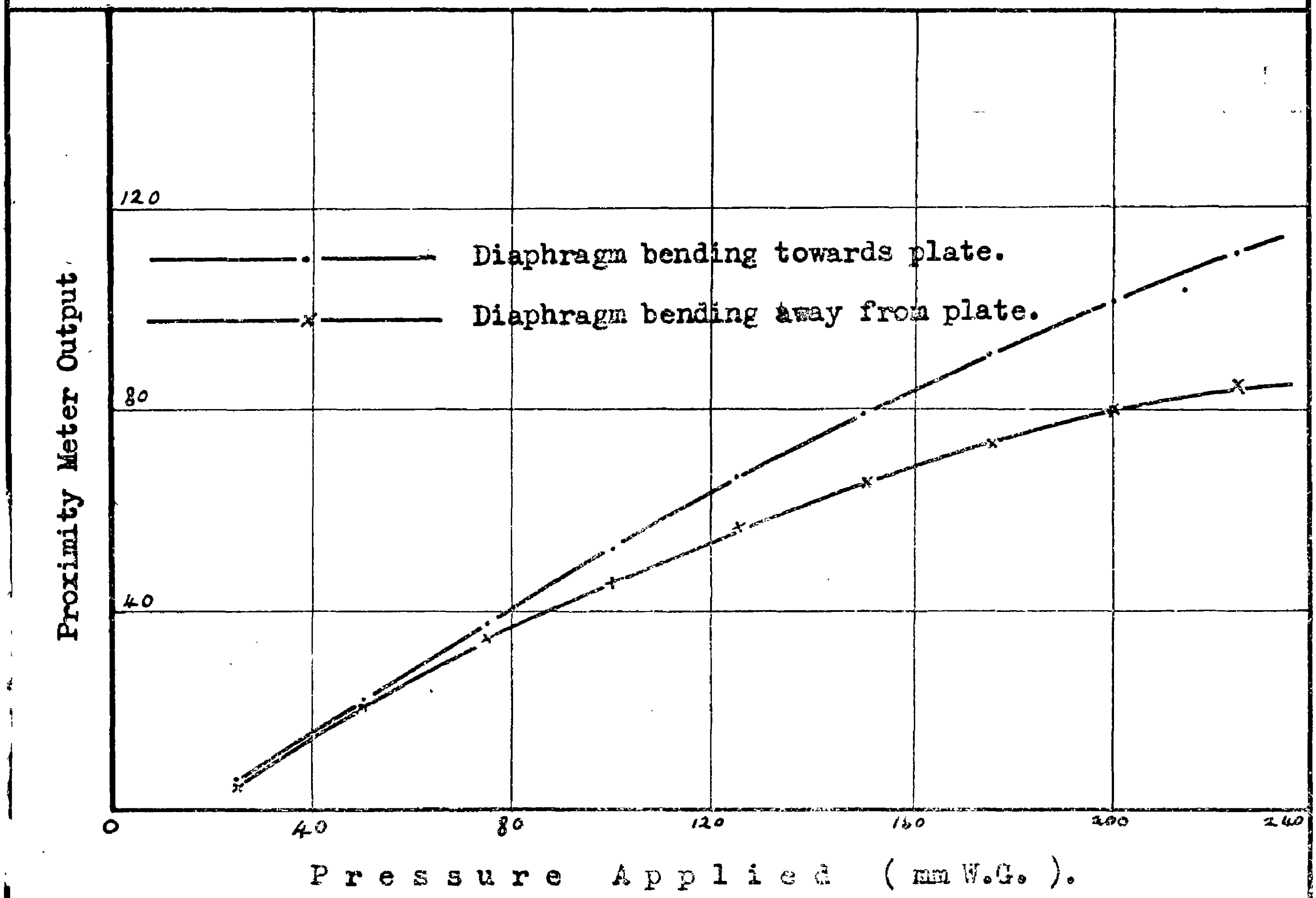
This arrangement was calibrated against steady pressures over two ranges from 0 to 2.2 inches W.G. and from 0 to 12 inches W.G. In each case the gain of the proximity meter amplifier was set to give full oscilloscope deflection for the maximum pressure used. The results showed a linear pressure - deflection relationship up to about 8 inches W.G. Repeated calibrations showed some discrepancies in sensitivity which may have been caused by variations in ambient temperature or by inaccurate resetting of the amplifier gain control which was a continuously variable potentiometer.

A P.M.4 model then became available and was used for all further tests. Since this instrument had a separate bridge unit and since the amplifier gain control was of the decade type, it was expected that the calibration would remain much more stable. It was decided to use the A.R.L. galvanometer recorder as a pick up with this instrument since the use of wider film offered greater possible accuracy and since more channels were available to put extra information on the records. When recorded on the galvanometer the rectified output showed considerable ripple and it was necessary to add a condenser input filter (4.1) to the output circuit. This removed all the ripple except a little 50 c/s leakage which was not serious. The complete output and recording circuit is shown in Fig. 4.3.

Trial calibrations were carried out, some of which are shown on Graph 4.3. These showed that better linearity was obtained when the diaphragm was moved towards the fixed plate than when it was moved away

GRAPH 4.3. Trial Calibrations.





from it. The limit of linear response was at about 5.5 inches W.G. - lower than in the earlier tests. This range was extended by building another 4 inch diameter diaphragm 0.015 inches thick and mounting it in the box alongside the original (see Plate 4.5). This diaphragm gave linear output for pressures up to 14 inches W.G. All the measuring equipment was permanently mounted on a trolley (see Plate 4.4) with all connections well secured before final calibration was carried out.

The equipment was then calibrated against the Betz manometer over three ranges. These were from 0 to 2.2 inches W.G. and 0 to 6 inches W.G. using the 0.005 inch thick diaphragm and from 0 to 14 inches W.G. using the 0.015 inch thick diaphragm. In each case the amplifier gain was set to give about 80% full scale deflection on the galvanometer recorder for the maximum pressure reached. The stability of this final arrangement, while still not perfect, was much better than the earlier arrangement. The calibrations were repeated several times on different days and mean calibrations were obtained. The average deviation of the individual readings from the means was about 2%, and the maximum deviation about 3%.

4.3. Conclusions.

1) By careful attention to the relevant theory it proved possible to use the Fielden Proximity Meter in conjunction with a diaphragm type gauge to record fluctuating air pressures at the required low level.
2) Even greater sensitivity than has been achieved here is possible by using a thinner diaphragm and reducing the plate spacing. The range

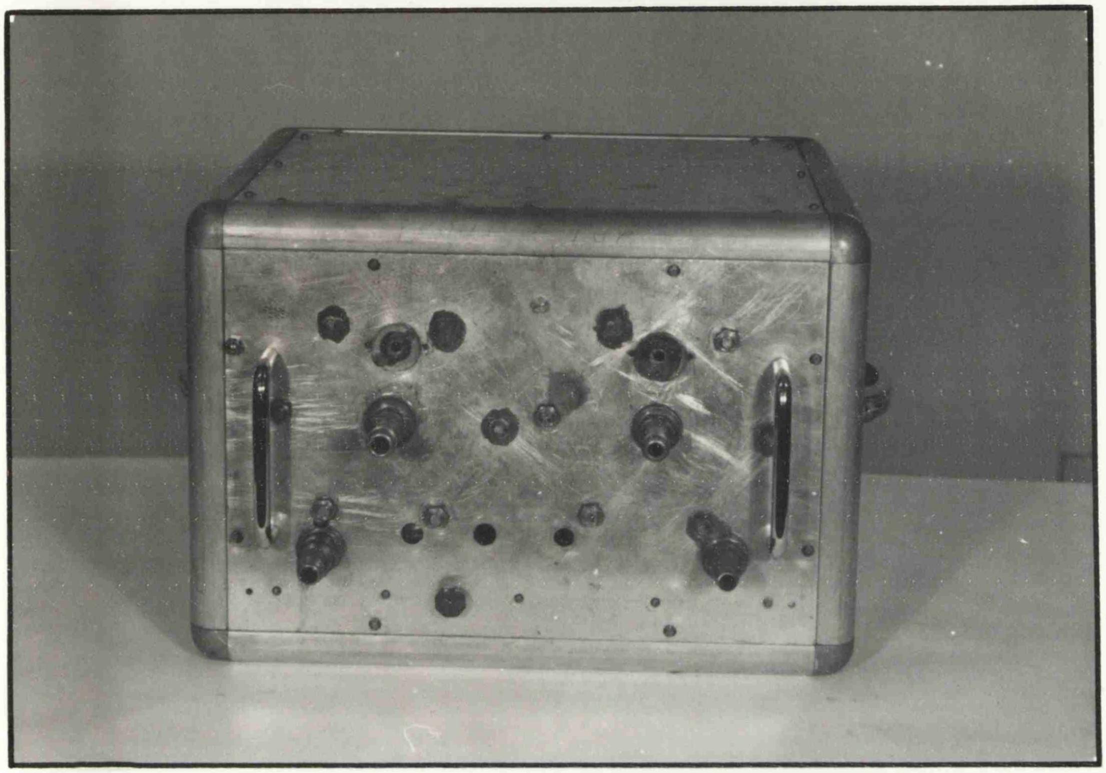


PLATE 4.5. TRANSDUCER BOX.

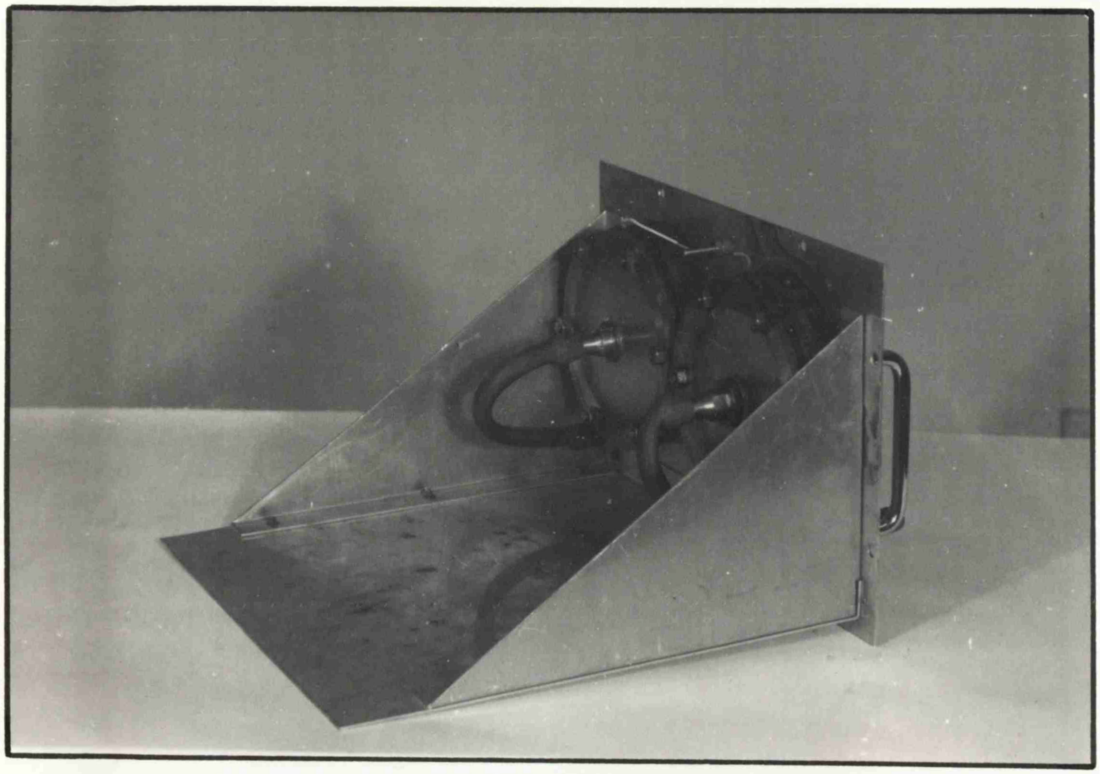


PLATE 4.6. TRANSDUCER BOX INTERIOR.

over which the output of such a gauge will be linear is restricted.

3) The equipment was not sufficiently stable to allow its use as an absolute standard, but since it was mainly required to record relatively small changes superimposed on a known steady pressure this proved to be no disadvantage.

Section 5 THE RECORDING MANOMETER - RESPONSE

5.1. Natural Frequency of the Diaphragm

A deflecting diaphragm can be expected to respond faithfully to a fluctuating pressure only if the forcing frequency of the applied pressure is substantially lower than the natural frequency of the diaphragm. It has been suggested (2.2) that the forcing frequency be limited to 0.3 of the natural frequency.

The natural frequency of a brass diaphragm 4 inches in diameter and 0.005 inches thick was estimated experimentally by allowing the diaphragm to vibrate in sympathy with a loudspeaker driven by a variable frequency oscillator. The lowest frequency at which the diaphragm would vibrate was around 150 cycles per second.

An attempt was also made to estimate the natural frequency of a diaphragm theoretically. A suitable formula (See Appendix) was found to be :-

$$p = \frac{10.21}{a^2} \sqrt{\frac{gD}{wt}} \qquad (1)$$

where a = radius of plate

t = thickness of plate

w = density of material

g = acceleration due to gravity

$$D = \frac{Et^3}{12(1 - S^2)}$$

E = Young's Modulus for the material

S = Poisson's Ratio for the material

$p = 2\pi f$

f = frequency

Using this formula the natural frequency of the diaphragm in question can be calculated to be 89 cycles per second. The formula, however, applies to a plate vibrating in a vacuum and (like Grashof's Theory for deflection) makes no allowance for stretching of the middle surface of the plate. As shown in the Appendix the effect of the mass of the air in which the plate vibrates is to reduce the natural frequency slightly, while the effect of the stretching of the middle surface in the case of a thin plate with a relatively large deflection is to increase the natural frequency. From this it would appear that the experimental value of 150 cycles per second is a reasonable figure for the natural frequency.

5.2. Response of the Measuring System

The natural frequency of the diaphragm is not the only factor to be considered in estimating the frequencies which can be confidently measured with a gauge of this type. The effect of the tubes used to connect the gauge to the points at which the pressure is to be recorded has also to be considered and it was pointed out (5.1) that this effect can result in the natural frequency of the system being lower than that of the diaphragm and consequently of greater importance. Theoretical and experimental investigation of this effect was carried out by Taback (5.2).

In the theory the propagation of pressure along a tube from an inlet to an instrument is governed by the general equations for a transmission line (5.3). Although the proof of these equations given in the reference applies to an electrical system, the method is general and can be applied equally well to an acoustical system, provided correctly analagous terms are used to describe the system parameters (5.4).

The general equations are :-

$$E_S = E_R \cosh \sqrt{ZY} \cdot \ell + I_R \frac{Z}{Y} \sinh \sqrt{ZY} \cdot \ell \qquad (1)$$

$$I_S = I_R \cosh \sqrt{ZY} \cdot \ell + E_R \frac{Y}{Z} \sinh \sqrt{ZY} \cdot \ell \qquad (2)$$

where E_S = Voltage at system inlet (or sending end)

E_R = Voltage at instrument (or receiving end)

I_S = current at system inlet

I_R = current at instrument

Z = series impedance of unit length of line

Y = shunt admittance of unit length of line

The quantities "Z" and "Y" are defined in terms of the system parameters as follows :-

$$Z = R + j w L$$

$Y = g + j\,w\,C$

where R = series resistance per unit length

L = inductance per unit length

g = shunt conductance per unit length

C = capacitance per unit length

$w = 2\,\pi\,f$

f = frequency in cycles per second

The quantity $\sqrt{ZY}$ is a complex number called the propagation constant and can be written.

$$\sqrt{ZY} = a + j\,b \tag{3}$$

where "a" is dependent on the decrease in amplitude per unit length of line and is called the "attenuation constant" and "b" is dependent on the phase shift per unit length of line and is called the "phase constant". The quantity $\sqrt{\frac{Z}{Y}}$ is called the "characteristic impedance" of the line and is designated by Z_o

$$\text{i.e.}\quad Z_o = \sqrt{\frac{Z}{Y}} \tag{4}$$

From equations (1), (3) and (4) we get

$$\frac{E_S}{E_R} = \cosh(a + j\,b)\ell + \frac{Z_o}{Z_R}\sinh(a + j\,b)\ell \tag{5}$$

where $Z_R = \frac{E_R}{I_R}$ = instrument impedance.

Equation (5) gives the ratio of the voltage (in the electrical case) or pressure (in the acoustical case) at the system inlet or "sending end" to the voltage or pressure at the instrument or "receiving end". The reciprocal of this ratio is called the Response and will be a maximum at the natural frequency of the system. Estimation of natural frequency by this method will be exceedingly difficult since "a" and "b" are both functions of "f" and $\frac{Z_o}{Z_R}$ is a function of (a + j b)

"b" can be calculated from

$$b = \frac{2\pi f}{V}$$

where V = velocity of propagation of pressure waves in the tube which is itself dependent on "f" (5.5)

the final form is $b = \dfrac{2\pi f}{C - \frac{D}{\sqrt{f}}}$ where C and D are constant for any one tube. The form of the expression for "a" is simpler since "a" is proportional to the square root of the frequency but any values used are not reliable since they are not independent of the amplitude of vibration.

The difficulty of the theoretical method made it necessary to check the response of the system by a series of experiments. These covered sinusoidally varying pressures of frequencies from 6 to 35 cycles per second in tubes of bore 2.8 to 11 mm. in lengths up to 20 feet and showed that for the commonly used 6 mm. bore tubing reasonable response could be expected at frequencies up to 25, 15, 10 and 5 cycles per second with tubes of length 1, 2, 4 and 10 feet respectively. The addition of a capillary restriction at the inlet to the system produces damping of the large pressure amplitudes at resonance frequencies but has very little effect away from the resonance. The small holes in the pitot-static tube have this effect and make it possible to increase these lengths without serious effects on the response.

Throughout the work on cages the tubes from the measuring point to the instrument were kept as short as possible and some checks were made by recording the same pressure effects using longer tubes. No differences significantly greater than the normal random fluctuations were noticed except with very long tubes. The pressure variations produced by the movement of the cages tended to be "square" rather than sinusoidal and any response effects which were present were easily recognised.

Section 6. THE MODEL SHAFT

6.1. Duct and Fan

An experimental wind tunnel equipped with model buntons, guides and cages was used to study how the presence and movement of cages affects the airflow in mine shafts. The layout is shown in Fig. 6.1. The wind tunnel was built up from sections of circular steel ducting each 6 feet long and 11¼ inches internal diameter. Six 4-foot long sections each made up of two half cylinders bolted together along longditudinal flanges were also used. The upper half of each of these sections was made of thick perspex with strong steel flanges. These sections were placed at convenient points along the length of assembled wind tunnel and provided very useful observation points. Removal of the perspex section was very simple and provided a means of access to the inside of the duct without dismantling. Tapped holes designed to allow the insertion of a pitot-static tube were provided at various points along the entire length of the duct which was 100 feet.

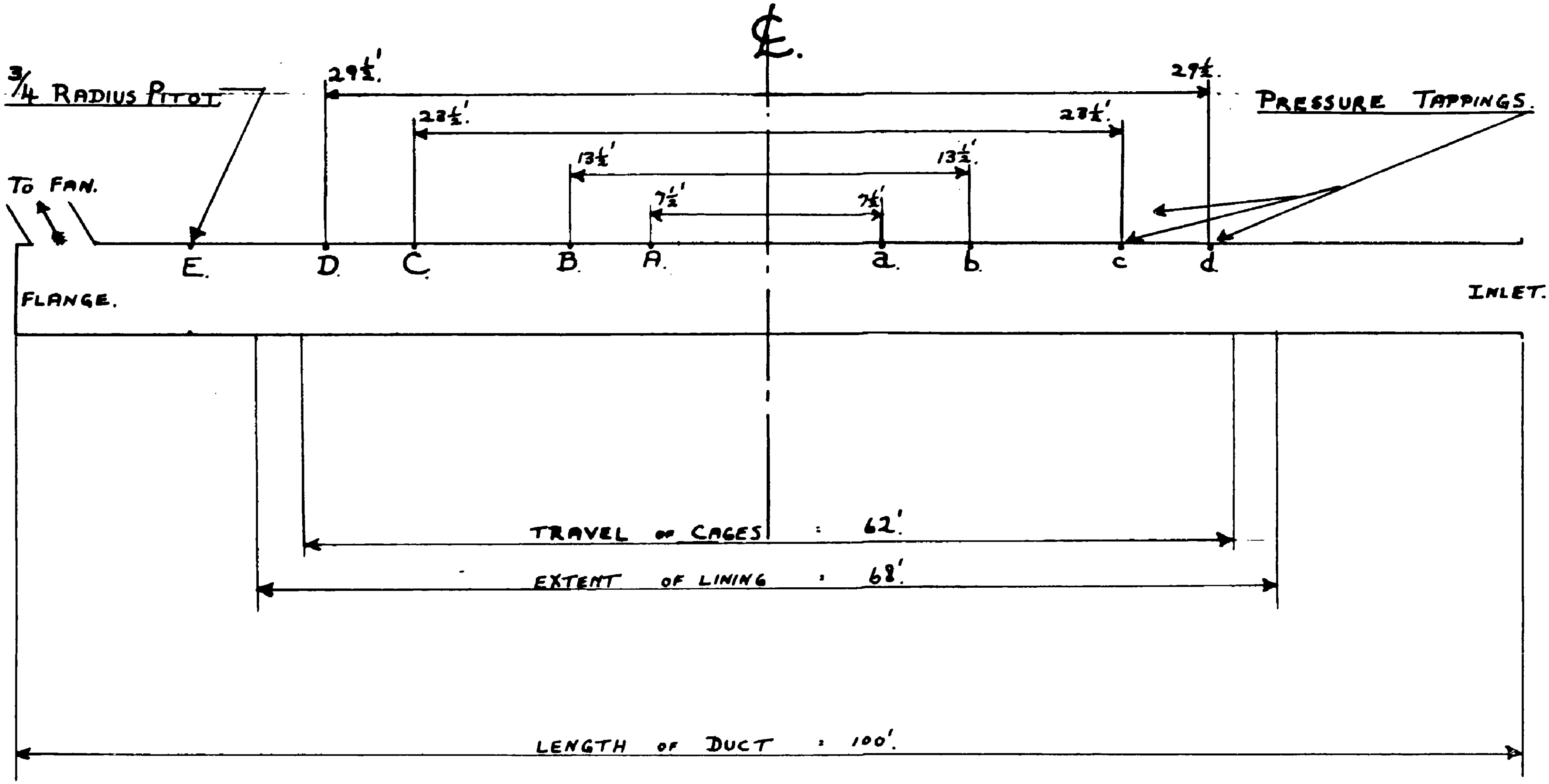


FIG. 6.1. LAYOUT OF WIND TUNNEL HORIZONTAL SCALE 1 in. - 12 ft.

Air was drawn through the wind tunnel by a $14\frac{1}{2}$ inch Howden centrifugal fan (Plate 6.1). The fan inlet was connected to a 45° Y - junction piece at the end of the duct by a length of 12 inch diameter Spiratube. This arrangement proved particularly suitable as it left a blank flange on the main line of the duct through which the driving ropes could pass to the cages without interfering with the fan. The fan was driven directly by a 20 h.p. A.C. motor running at 2850 R.P.M. No speed control was available and it was necessary to use throttling devices to control the quantity of air flowing in the duct. A butterfly type shutter (See Plate 6.1) was placed at the fan inlet and used to prevent overloading of the motor by allowing the fan to run up to full speed on no-load. The control exercised over the air quantity by this device was far too coarse to be of any use in experimental work and finer control was supplied by a conical shutter which could be screwed into the duct inlet on a finely threaded spindle. This proved very satisfactory during the early tests but had to be removed later as it interfered with the driving ropes. It was replaced by a sliding internal sleeve-type throttle at the Y-junction close to the fan (Plate 6.1). The removal of the throttling cone allowed the addition of a flared inlet to the duct (Plate 6.2). The presence of the driving ropes did not allow the use of any flow straightening devices in the main section of the duct but it was possible to insert a short length of impregnated paper honeycomb in the Y-piece. This helped to damp out any pulsations coming from the fan and also provided a useful safety device by preventing any material from reaching the fan inlet in the event of any of the shaft

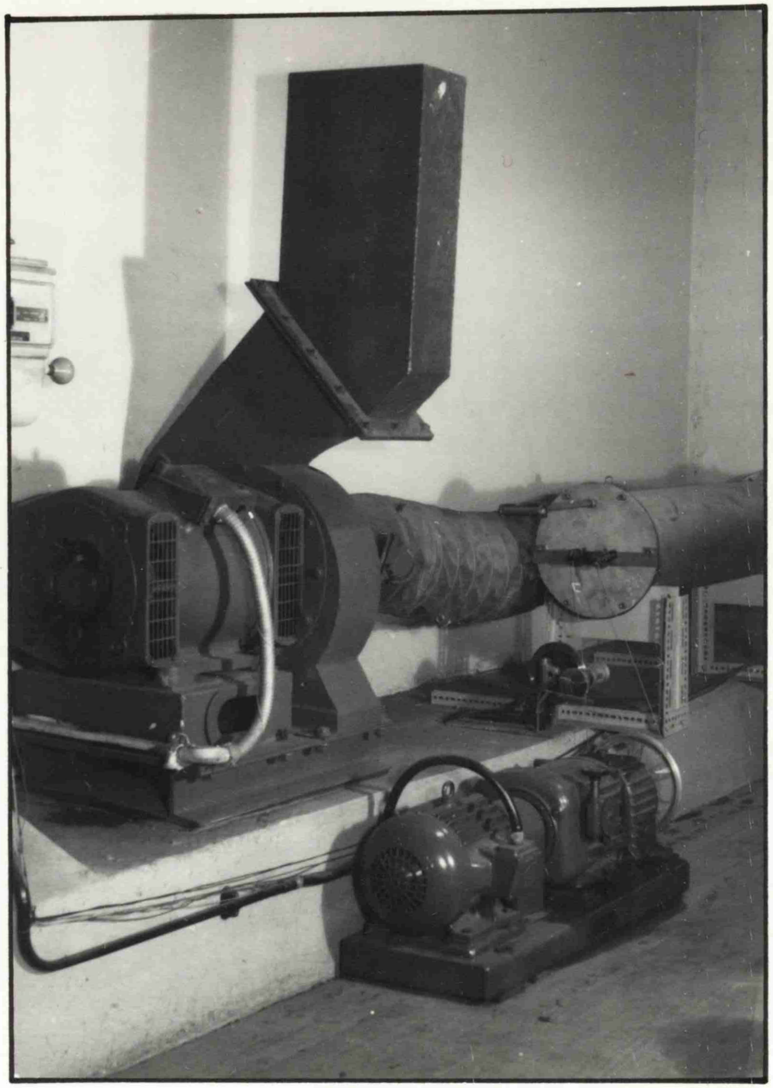


PLATE 6.1. FAN & CAGE DRIVING GEAR.

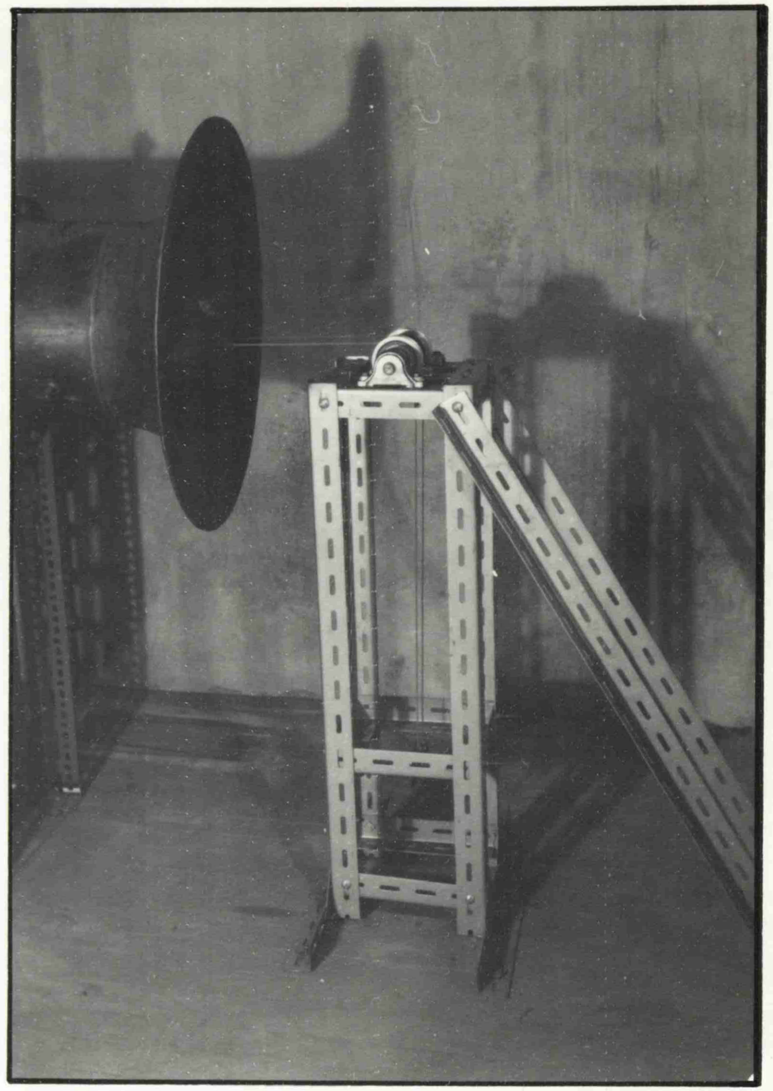


PLATE 6.2. DUCT INLET.

equipment breaking loose.

6.2. Shaft Lining.

Two patterns of guides and buntons were used in the tests, the design of the shaft layouts being based on information supplied by the National Coal Board (6.1). Since it was intended to simulate conditions in a shaft 24 feet in diameter in the duct the scale ratio was fixed at 25.6 to 1. The buntons were set at 4.7 inch centres corresponding to 10 feet in a full scale shaft. The lining was made up in 12 foot lengths by soldering the components together on a specially designed jig. In addition to the guides and buntons a strip of $\frac{1}{8}$ by $\frac{3}{8}$ inch brass was fitted along the ends of the buntons to give the necessary rigidity.

In the two-guide system (Fig. 6.2) in which tests were confined to stationary models the total length of lining installed was 24 feet. In the four-guide system (Fig. 6.3) however, the work was extended to include moving models. No information was available on the features which moving cages might exhibit and it was anticipated that a cage starting to move from rest might require to move some distance before steady conditions were achieved. In view of this it was decided to increase the length of the shaft lining. Conditions in the laboratory prevented the total length of the duct being increased above the existing 100 feet. The length of lining, however, was increased to 68 feet placed centrally in the 100 foot length of the duct. Since the lining was longer and was intended for use with moving cages greater rigidity and much smoother alignment at the guide joints were required than in any

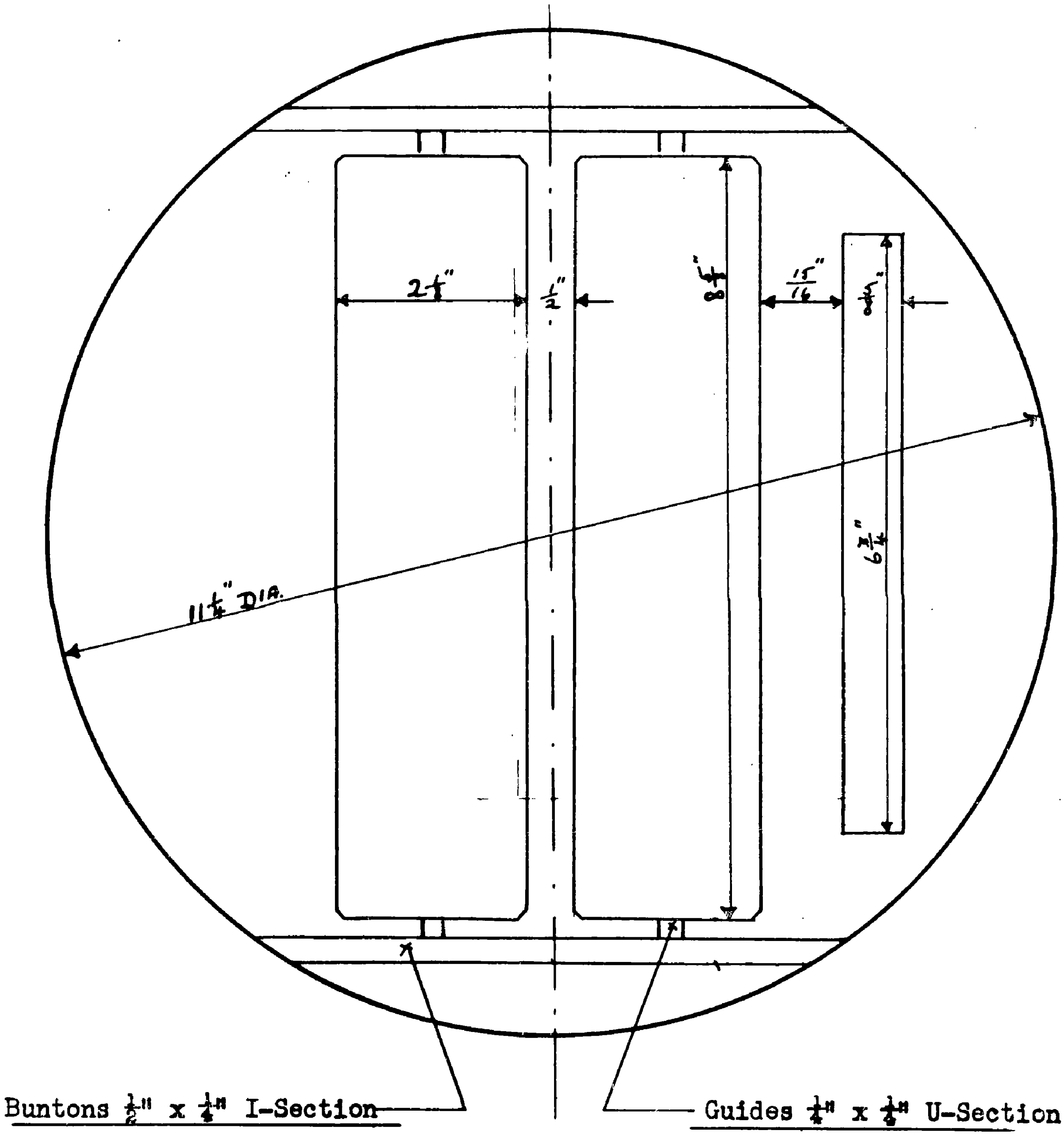


FIG. 6.2. TWO GUIDE SYSTEM HALF SIZE

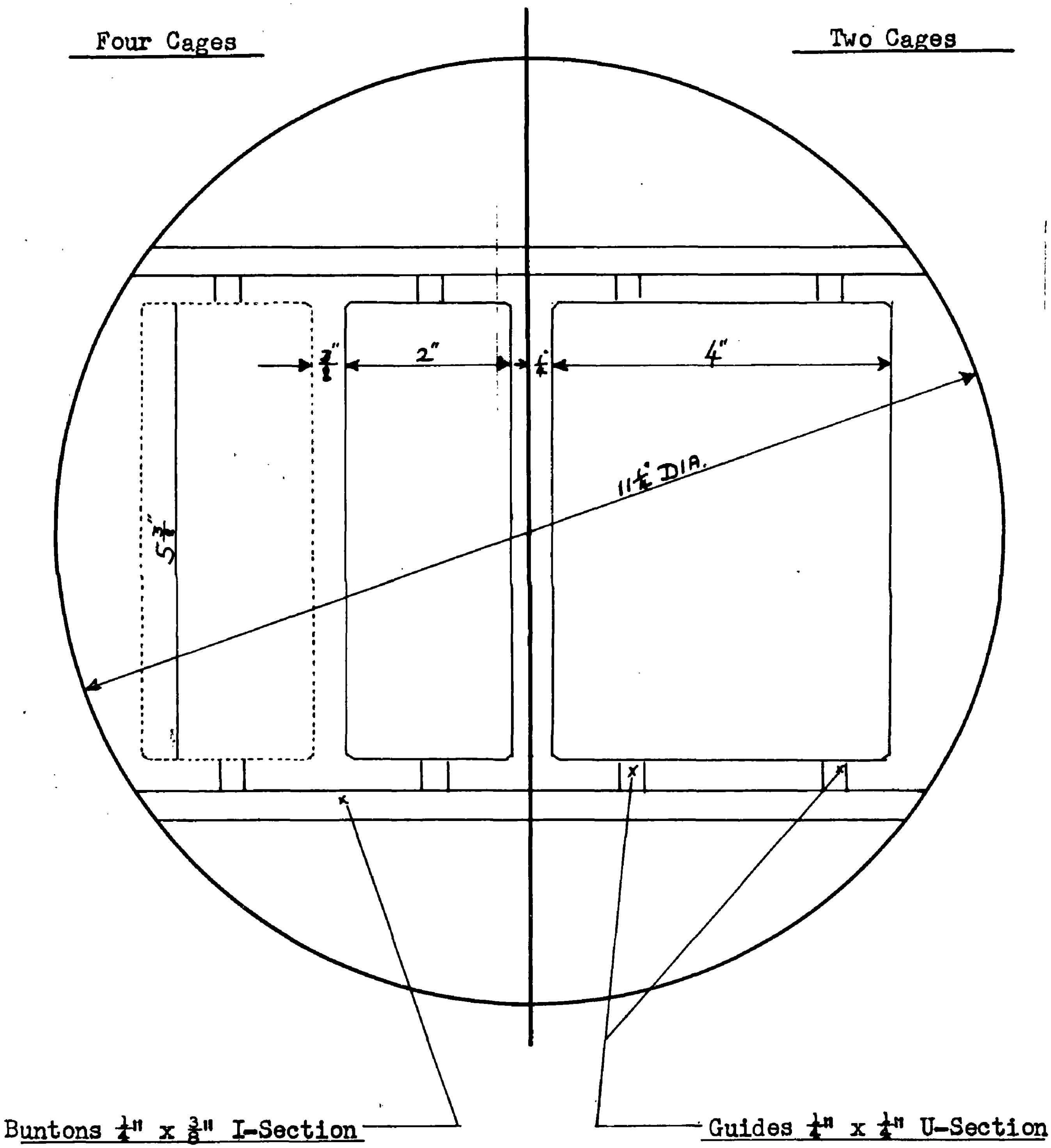


FIG. 6.3. FOUR GUIDE SYSTEM HALF SIZE

earlier work. The longer lining provided a longer test length and a settling length of lined duct which could be expected to improve the similarity between the model and a full scale installation. When scaled up in the ratio of the diameters the lined section represented a length of nearly 1750 feet of 24 foot diameter shaft.

6.3. Cages and Cage Control Gear.

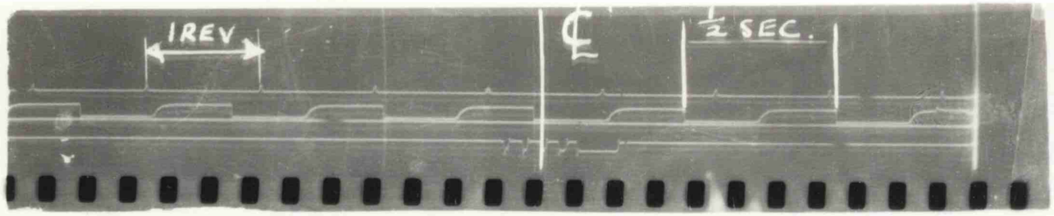
The cages were made of wood and were of very simple design. Several types of cage shoes were tried before satisfactory smooth running was achieved along the guides and past the joints. The final choice was short spring loaded rods running inside the channel section of the guides.

The cages were driven by a $1\frac{1}{2}$ h.p. 3 phase A.C. motor fitted with a continuously variable hydraulic reduction gear (Plate 6.1). The power was transmitted to the cages by friction drive on a $\frac{1}{4}$ inch circumference galvanised steel rope passing $1\frac{1}{2}$ times round a 10 inch diameter driving Vee-pulley. The motor unit was designed to give an output torque of 510 inch lb. at all speeds from 5 to 124 R.P.M. in either direction. This meant that a rope pull of 102 lbs. was available to drive the cages at any speed from $12\frac{1}{2}$ to 325 feet per minute. Suitable tension was maintained on the driving rope by the tail rope which passed from the cages round the return frame shown in Plate 6.2. (This plate also shows the flared inlet to the duct). The nominal breaking strength of the rope was 538 lbs. but it was fixed to the cages by easily broken safety links of soft wire. After the initial work in designing suitable cage shoes very little trouble was experienced. On the few occasions when a cage did stick the breaking of these links prevented any serious damage.

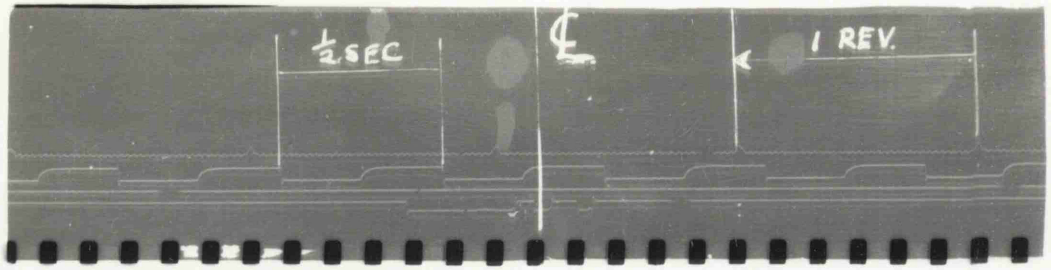
The driving motor was fitted with a reversing starter, and for convenient operation a set of remote control buttons for this starter was fixed on the trolley carrying the measuring instruments. The starter could also be operated automatically by the cages themselves by means of micro-switches with specially designed actuators fitted inside the duct at the end of the lining. These switches stopped the motor before starting it away in the opposite direction, thus preventing accidental over-run of the lining by the cages and allowing completely automatic operation if required.

6.4 Cage Position Indicating Equipment.

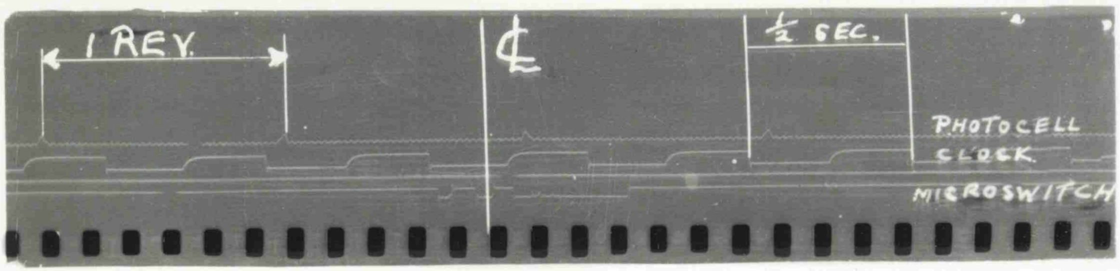
This equipment was required to relate the pressures being recorded on the galvanometer recorder to the position of the cages in the duct. Three channels on the recorder were used to record this information. The actual position of the cages was marked by a signal produced by the closing of a micro switch by each deck of a cage in turn. The distance which the cage had moved from this micro-switch was obtained by recording the signals from a photo-cell which was momentarily exposed to a lamp by the passing of a small hole in a light pulley driven by the main rope. Experience showed that the recording film did not always maintain its nominal speed through the light gate and so a signal from a clockwork operated switch was introduced to a third channel to give a square wave of $\frac{1}{2}$ second period. On very clear records it was also possible to check the film speed by counting the peaks on the 50 c/s interference which was sometimes picked



a) Series C/4/BN.
Cage moving downstream.



b) Series C/2/AN.
Cage moving upstream.



c) Series C/2/AN.
Cage moving downstream.

FIG. 6.4. OUTPUT OF CAGE POSITION INDICATING EQUIPMENT.

up. Fig. 6.4 shows typical film records of these signals.

6.5. Air Flow Measurement in the Wind Tunnel.

During the tests on the two-guide system the air flow was measured with a standard orifice plate (6.2). The greater length of the lining of the four-guide system, however, greatly increased the resistance to air flow of the duct as a whole and reduced the quantity of air which the fan was capable of delivering. The orifice plate absorbed a considerable proportion of the fan water gauge which could have been more usefully employed maintaining a higher overall air velocity if a suitable low-resistance flowmeter could be used. A centrally placed pitot tube would have given the lowest loss of all flowmeters but its calibration could only have applied to one particular velocity distribution, a quantity which is itself dependent on Reynold's Number. A Three-Quarter Radius Pitot Tube Flow Meter (6.3) was recommended (6.4). This consists of 4 pitot tubes at 90° intervals placed at three quarters of the duct radius and four static holes on the wall also 90° apart. (See Fig. 6.5 and Plate 6.3).

The advantages of placing the pitot tubes at this radius can be shown by the following theoretical considerations :-

The distribution of velocity, "u", across the section of a duct of radius R, can be expressed empirically as

$$u = U\left(\frac{y}{R}\right)^{\frac{1}{m}} \qquad (1)$$

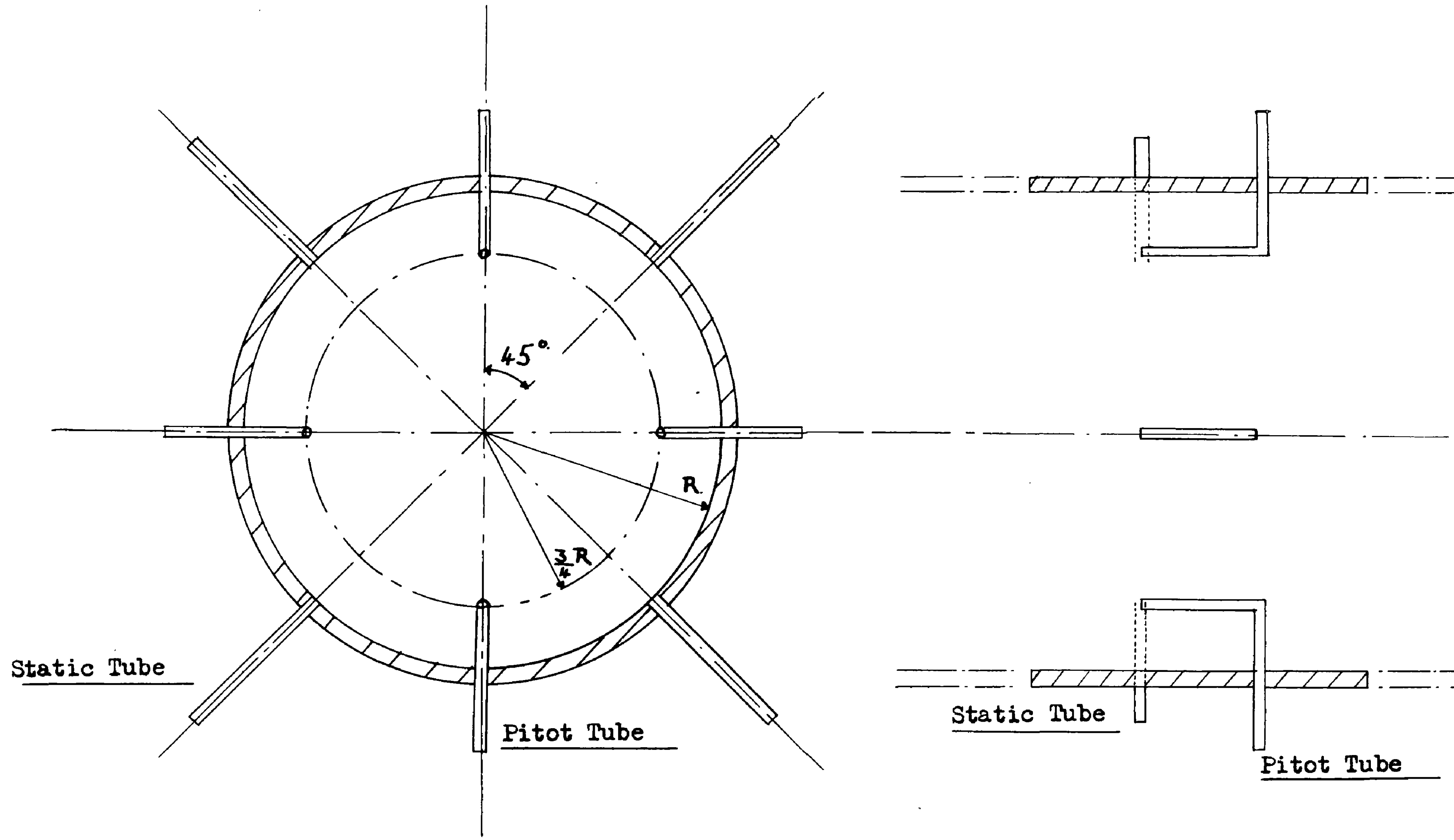


FIG. 6.5. THREE QUARTER RADIUS PITOT TUBE TYPE FLOWMETER. QUARTER SIZE

where U = velocity at centre

y = distance from the wall

i.e. $y = R - r$ where r = any radius

m = constant dependent on Reynolds Number

Now, if mean velocity $= \bar{u}$

then Quantity flowing $= Q = \pi R^2 \bar{u}$

$$= \int_0^R 2\pi \; u \, r \; dr.$$

$$= -\int_R^0 2\pi \; u(R - y)dy$$

$$= \int_0^R 2\pi U \left(\frac{y}{R}\right)^{\frac{1}{m}} (R - y)dy$$

$$\therefore \quad \pi R^2 \bar{u} = 2\pi U \int_0^R \left(\frac{y}{R}\right)^{\frac{1}{m}} (R - y)dy.$$

Integration and simplification gives

$$\frac{\bar{u}}{U} = \frac{2m^2}{(m + 1)(2m + 1)}$$

$\therefore$ Quantity flowing $= \pi R^2 \bar{u}$

$$= \pi R^2 \; U \; \frac{2m^2}{(m + 1)(2m + 1)}$$

Let Quantity indicated on pitot tube at $\frac{3}{4}$ radius $= Q_1$

$$\therefore \quad Q_1 = \pi R^2 u \quad \left(\frac{y}{R} = \frac{1}{4}\right)$$

From (1) $Q_1 = \pi R^2 \times U \left(\frac{1}{4}\right)^{\frac{1}{m}}$.

$$\therefore \quad \frac{Q_1}{Q} = \left(\frac{1}{4}\right)^{\frac{1}{m}} \frac{(m + 1)(2m + 1)}{2m^2} \tag{2}$$

As already stated the value of "m" depends on Reynolds Number. For medium Reynolds Numbers $m = 7$ is generally used. Tabulation of the values of (2) shows that for values of "m" between 4 and 10 the value of Q_1 / Q lies within the range 0.995 to 1.005. This means that the error in quantity reading caused by variations in velocity distribution at different Reynolds Numbers is less than 0.5% over this wide range. Q_1/Q is unity when $m = 5$ hence the calibration constant is seldom exactly equal to 1. A further advantage of the use of the three -quarter radius position is that it provides greater sensitivity to changes in quantity than the central position. The most sensitive position will be in the elementary annulus which makes the greatest contribution to the total quantity. This occurs at the point where the product "u.r" is a maximum.

Now $\quad u\,r = U \left(\frac{y}{R}\right)^{\frac{1}{m}} (R - y)$

$\therefore$ For a maximum value, $\left(\frac{y}{R}\right)^{\frac{1}{m}} (R - y)$ must be a maximum and analysis shows that this occurs at the point $\frac{y}{R} = \frac{1}{1 + m}$.

$\therefore$ For $m = 7$ the most sensitive spot is $\frac{y}{R} = \frac{1}{8}$ or at $\frac{7}{8}$ th radius.

Relative values are :-

$\frac{y}{R}$	$\frac{1}{8}$	$\frac{1}{4}$	$\frac{1}{2}$
r	100	94	69

The $\frac{3}{4}$ radius position is preferred because it gives a calibration constant very close to unity and because it gives a higher differential pressure to be measured with very little loss of sensitivity.

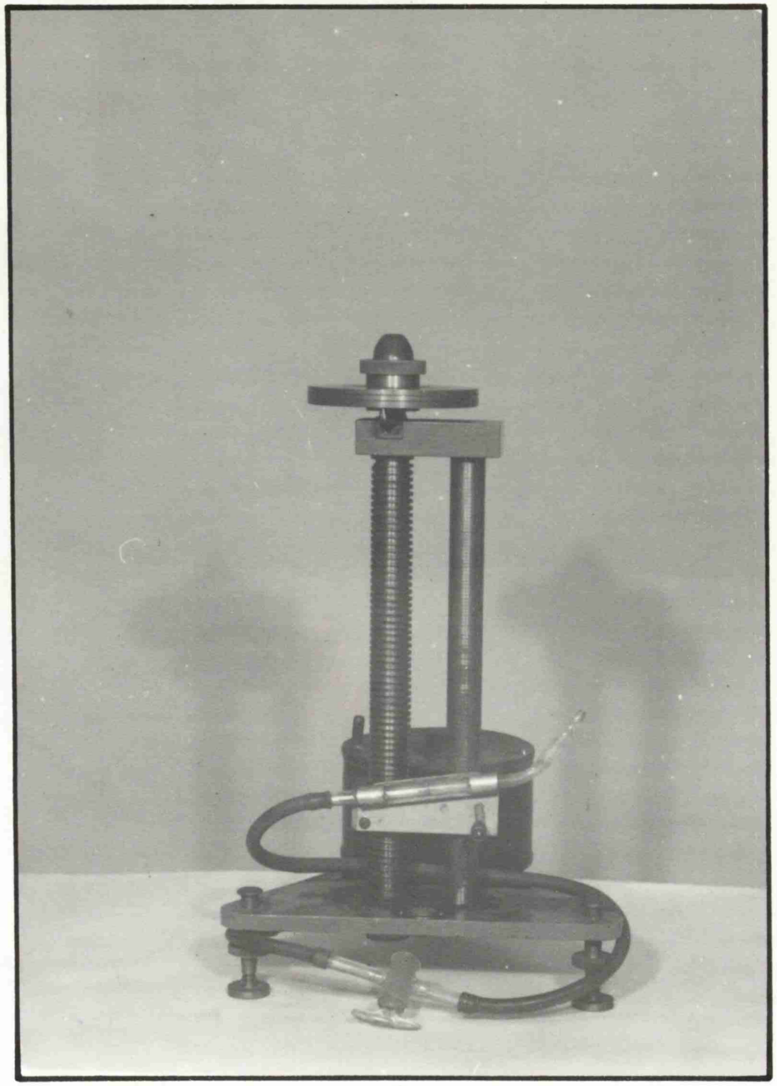
The presence of the four pitot tubes in the $\frac{3}{4}$ radius position helps to reduce the effect of any asymmetry in the velocity distribution.

While not as accurate as the orifice plate because the differential pressure head was lower and less steady, the three-quarter radius pitot tube flowmeter proved very useful and allowed the use of a much higher velocity in the duct than would otherwise have been possible.

6.6. Manometers.

During the tests fluctuating air pressures were measured with the recording equipment already described and static air pressures were measured with two water manometers described below.

The Direct-lift Micromanometer (See Plate 6.4) was built in the Mining Department of the University of Nottingham. Pressures are measured by returning the water level to a fixed mark on the inclined tube which is raised or lowered on a micrometer screw. The range is from 0 to 10 inches W.G. and the smallest division on the micrometer head represents 0.001 inch W.G. The instrument is easily read and can



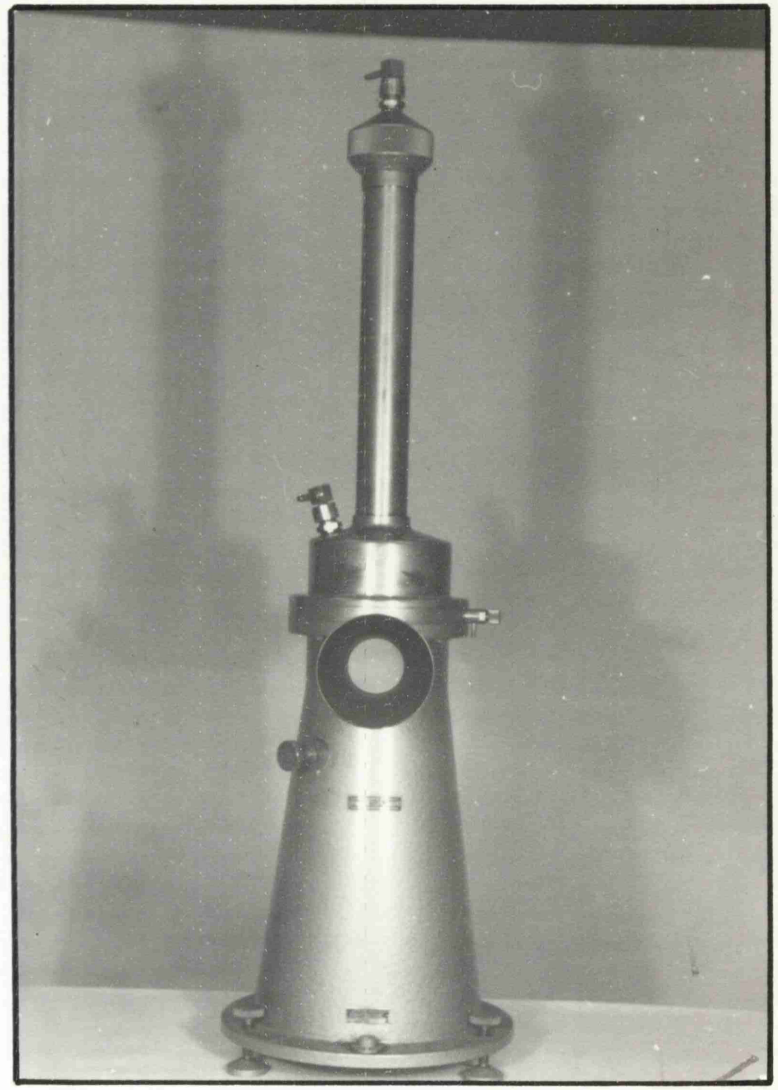


PLATE 6.4. DIRECT LIFT MICROMANOMETER.

PLATE 6.5. BETZ PROJECTION MANOMETER.

give a good estimation of the mean value of a pressure which is not perfectly steady.

The Betz Projection Manometer (See Plate 6.5) is of German design and manufacture. The application of pressure causes a change in water level inside the stem of the instrument. Accurate measurement of the change in water level is made by observing, through an optical system, a long finely divided scale which is suspended in the water from a float. The total range is 0 to 400 mm. (15.75 ins.) W.G. and the smallest scale division is 0.1 mm. (0.04 ins.) W.G.

Section 7 PRELIMINARY INVESTIGATIONS

7.1. Conditions in the Wind Tunnel.

Reference to Stevenson's work made it unnecessary to undertake a complete B.S. fan test but as some changes had been made in the layout since that time, it was considered necessary to make a check on the airflow conditions prevailing in the duct. This was done by making an 11-point pitot tube traverse across the section of the duct using the special pitot tube shown in Plate 7.1. The carriage of this pitot tube fits on to either of two bosses placed 90 degrees apart on one of the duct sections and the measuring head can therefore be racked across a horizontal diameter and a vertical diameter its distance from the centre being measured on a vernier scale. Horizontal and vertical traverses were undertaken at four different air velocities and the results are shown on Graph 7.1. A theoretical velocity distribution derived from

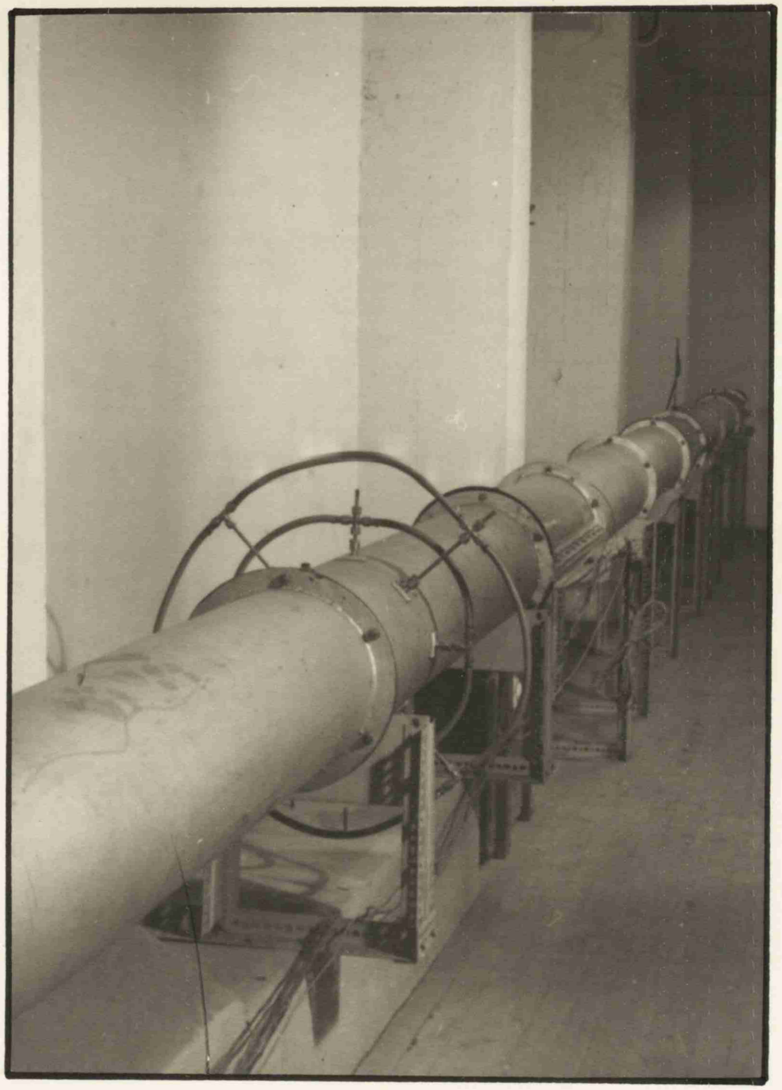


PLATE 6.3. VIEW OF DUCT SHOWING FLOWMETER.

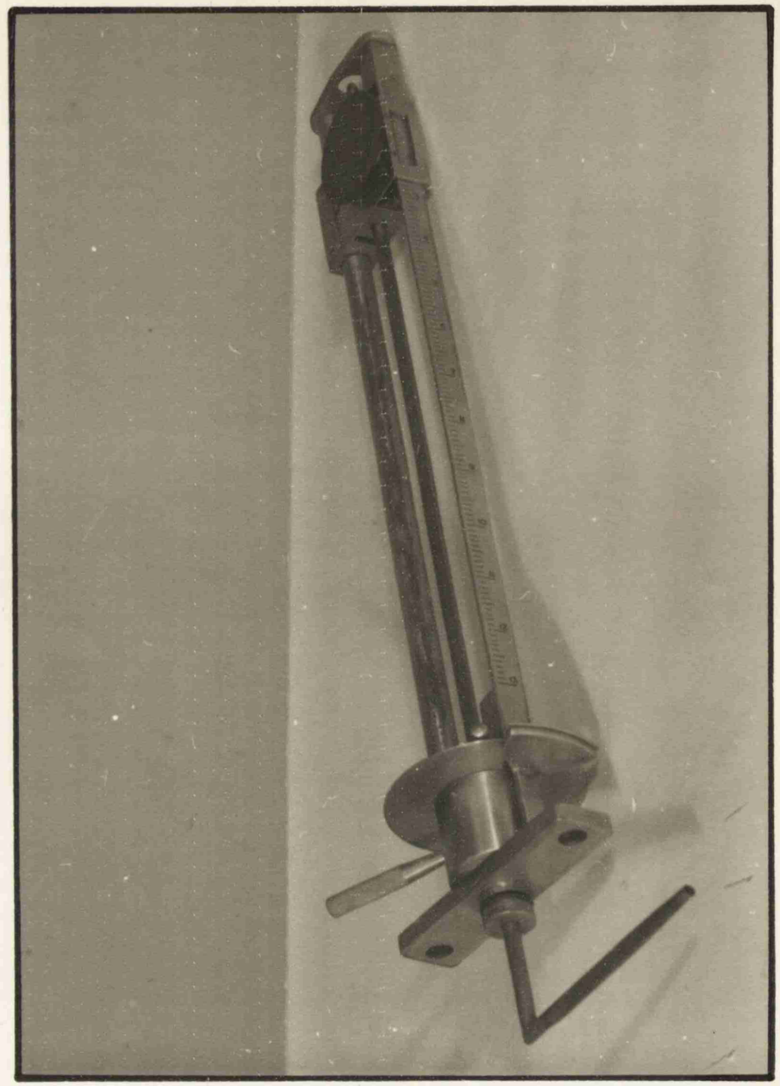
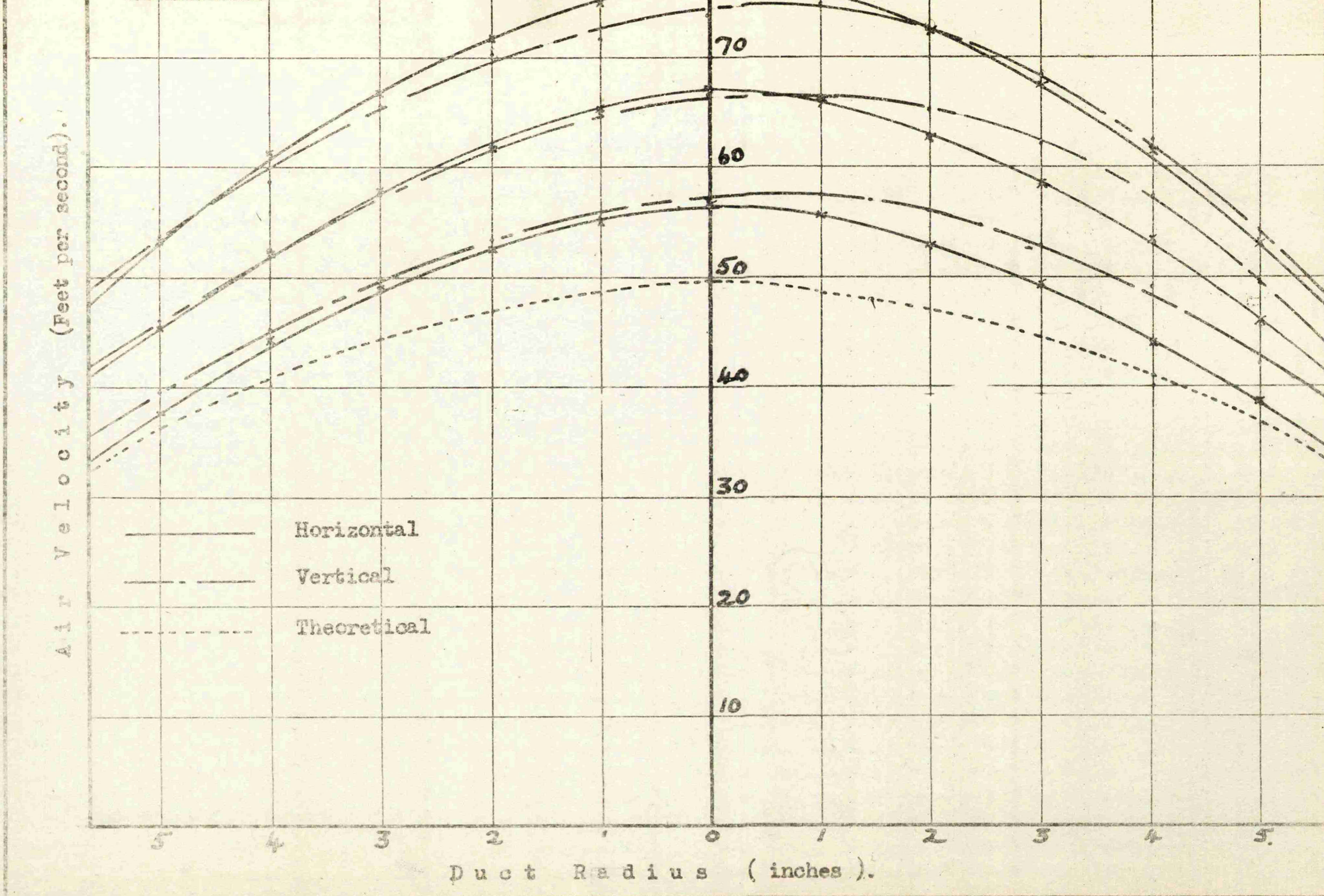


PLATE 7.1. TRAVERSING PITOT TUBE.

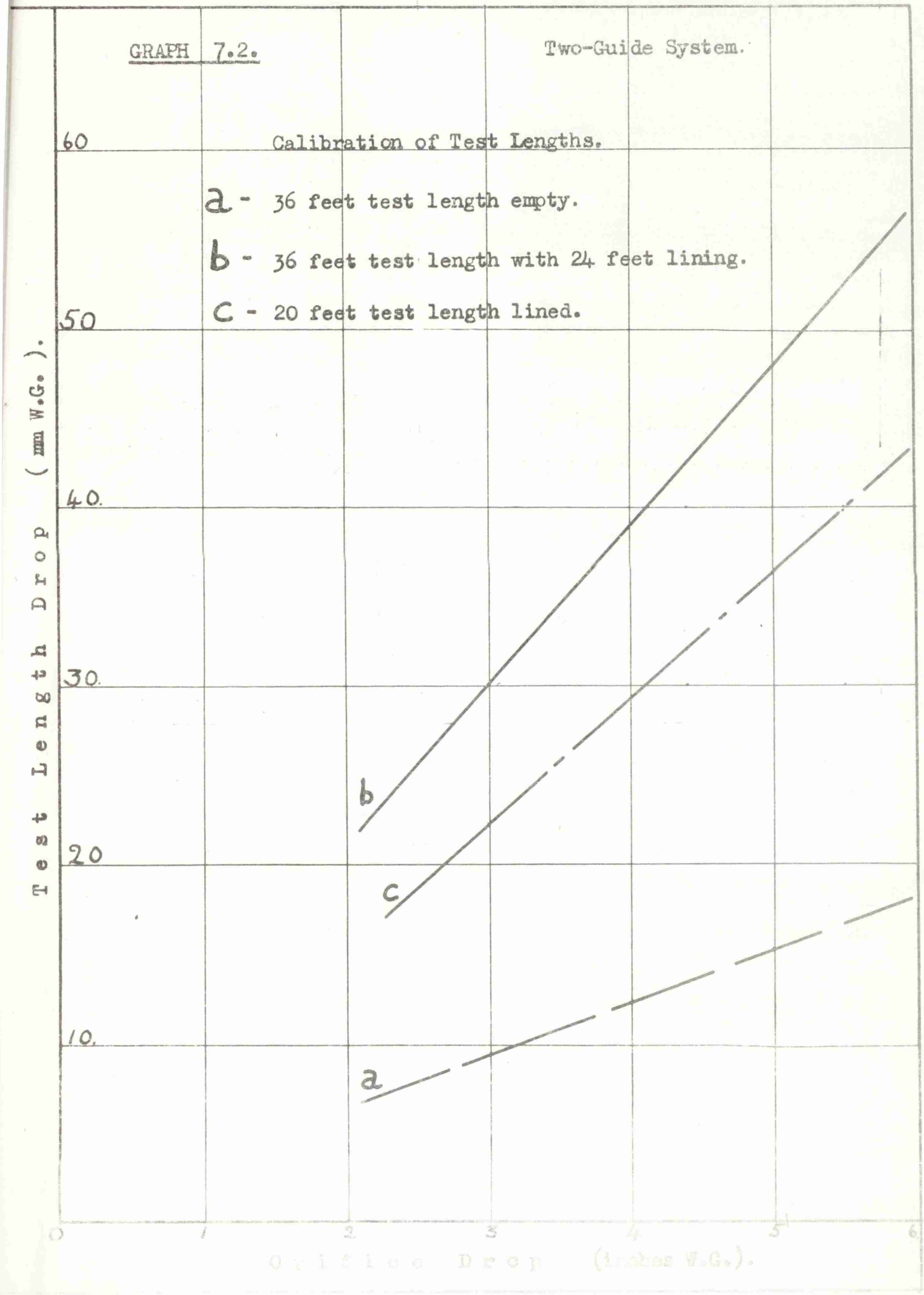
<u>GRAPH 7.1.</u> Velocity Distributions.



expression (1) (Section 6.5, page 37) is also plotted on this graph. The experimental velocity distributions appeared very satisfactory and were in much better agreement with the theoretical and with each other than a similar set shown by Stevenson. The most likely reason for this is that the fan was now working on the exhausting system whereas it had previously been forcing. The mean value for the centre constant obtained in these tests was 0.81 and excellent agreement existed between the four values. This is about the value expected for such a duct (7.1) and is more satisfactory than a value of 0.95 given by Stevenson.

7.2. Two Guide System.

The test length used for this system was fixed at 36 feet. Two pitot tubes were set on the axis of the duct this distance apart and a calibration relating the pressure drop across the test length (T.L.D.) to the pressure drop at the orifice (O.D.) was obtained. A 24 foot length of lining was then installed centrally in the test length and a further calibration obtained. This calibration was repeated three times on different days to get a representative value before proceeding to tests involving cages. The pitot tubes were then moved to 20 feet apart across the lining and a third calibration carried out in order to obtain the resistance of a length of lined shaft. The three calibrations are shown on Graph 7.2.


GRAPH 7.2.
Two-Guide System.
Calibration of Test Lengths.
a - 36 feet test length empty.
b - 36 feet test length with 24 feet lining.
c - 20 feet test length lined.
60
50
40.
30.
20
10.
b
c
a
0
1
2
3
4
5
6
Test Length Drop (mm W.G.).
Orifice Drop (inches W.G.).

7.3. Calibration of Flowmeter.

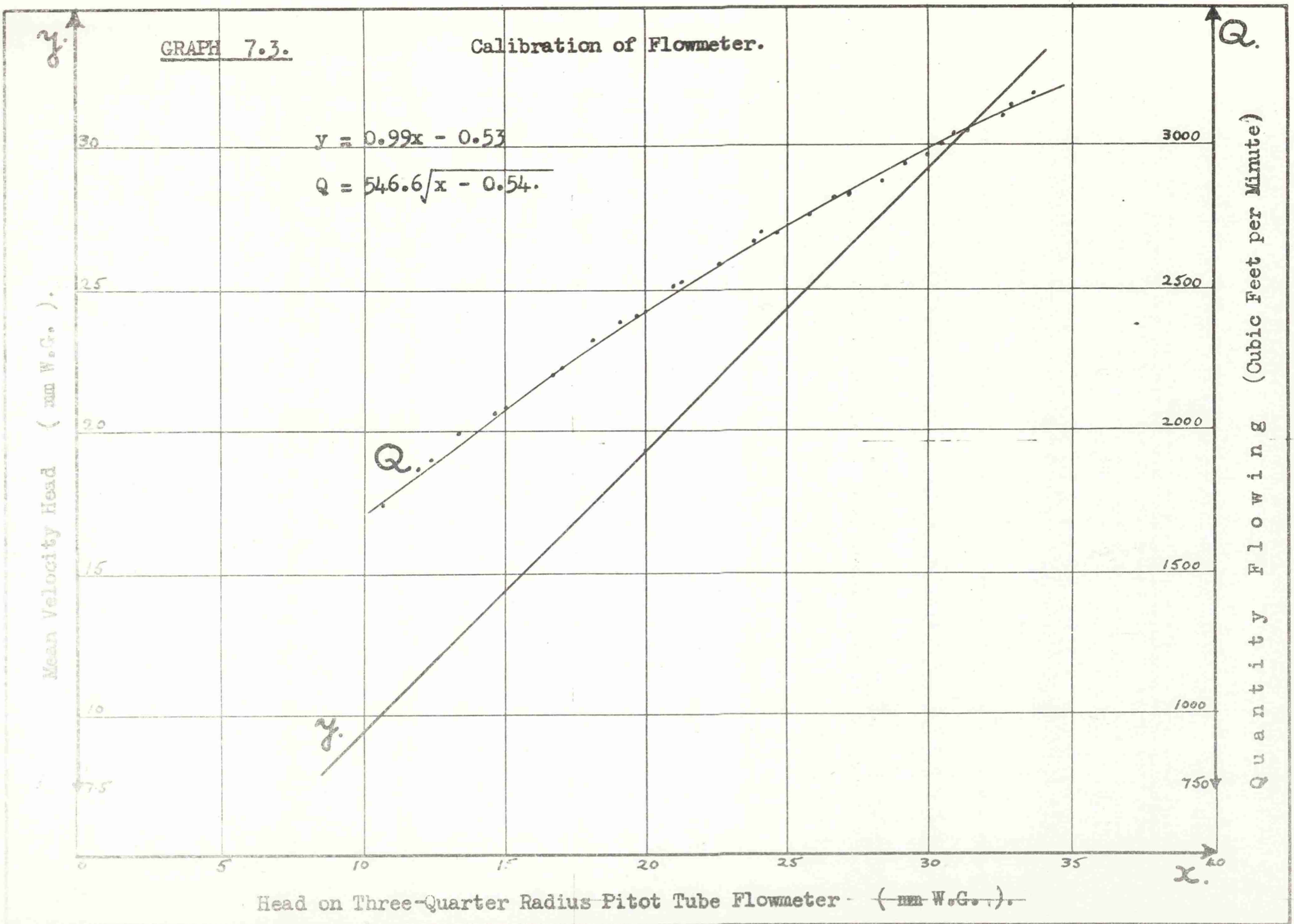
Before the four guide lining was built into the duct the calibration of the three quarter radius pitot tube flowmeter was carried out. This was done by taking simultaneous readings of the pressure drop across the orifice and the head on the flowmeter, the test being repeated on different days. Values of air quantity and mean velocity head were calculated from the orifice drop readings using the standard orifice formula (6.2) and the results were plotted (Graph 7.3). The plot showed that a linear relationship existed between the head on the flowmeter and the mean velocity head over the range covered. Since the purpose of the calibration was to obtain a method of estimating the Mean Velocity Head (y) from the corresponding flowmeter head (x) and since a large number of observations were available the results provided a very suitable subject for use of a regression line (7.2). The method of obtaining the equation of such a line is a common statistical practice and no attempt is made to describe it here. Its suitability for application in this particular instance can be shown as follows:-

If the relationship between two quantities "x" and "y" is significantly linear then the "regression of y upon x" gives the equation of that line from which the sum of the squares of the distances from all the known points to the line measured parallel to the y - axis is a minimum. The regression of "y" upon "x" therefore gives the "best straight line" for estimating the mean value of "y" (in this case the unknown mean velocity head) for any given value of "x" (in this case the observed flowmeter head).

The use of this technique supplied a simple algebraic relationship

GRAPH 7.3.

Calibration of Flowmeter.



between the two quantities by means of which the mean velocity head could be quickly calculated without continual reference to a calibration chart.

Thus :-

By calculation :-

$$y = 0.99\,x - 0.53 \qquad \text{where the units are mm. W.G.}$$

Further calculation incorporating the cross-sectional area of the duct gives

$$Q = 546.6\sqrt{x - 0.54}.$$

where Q = quantity flowing in cubic feet per minute.

The range of flowmeter readings covered in the experimental calibration was from 11 to 35 mm. W.G. The corresponding range of Reynolds Numbers referred to the duct diameter is from 250,000 to 450,000. Extrapolation of these formulae beyond this range is not advisable due to variation with Reynolds Number of a combined multiplier in the standard orifice formula on which they are based. All later experimental work was carried out within this range.

An attempt was also made to calibrate the flowmeter against a simplified pitot tube traversing method (7.3). The traverse used only four pitot tube positions on each diameter and is based on the assumption that the velocity distribution is a log-linear relationship of the form.

$$v = V + V_1 \log\left(\frac{y}{D}\right) + V_2\left(\frac{y}{D}\right)$$

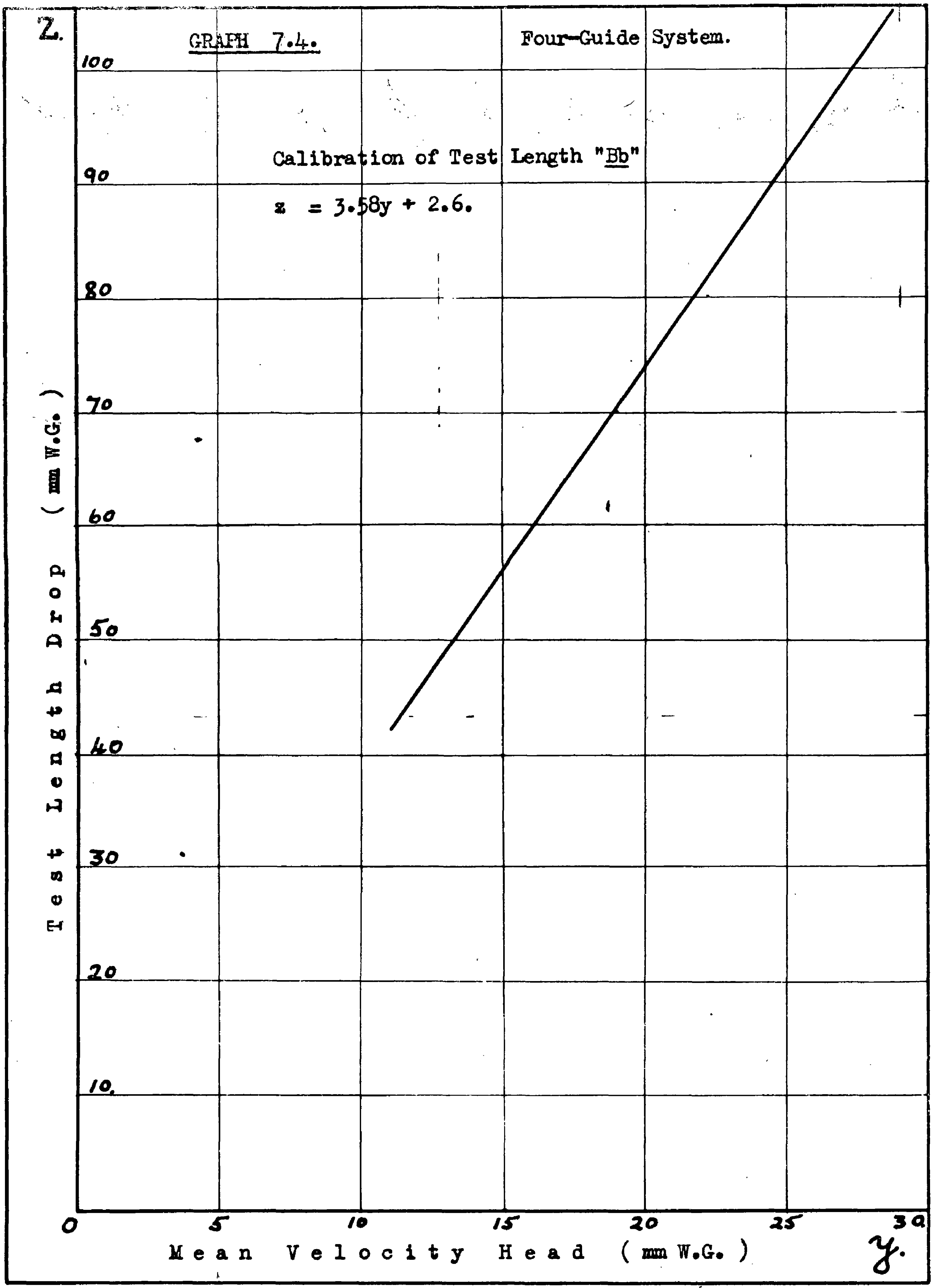
where D = diameter

y = distance from wall.

The results gave values of mean velocity head about 4% below the orifice plate calibration and were judged to be less reliable. This discrepancy could have been caused by some lack of symmetry in the velocity distribution. Other sources of inaccuracy are the difficulty in placing the pitot tubes at the exact measuring spots required and the possibility that the section of the duct was not exactly circular.

7.4. Four Guide System.

The test length used in this system was between the points "B" and "b" (See Fig. 6.1) and was 27 feet long. A calibration exactly similar to that on the two guide lining was carried out and is shown on Graph 7.4.



Z.
GRAPH 7.4.
Four-Guide System.
Calibration of Test Length "Bb"
z = 3.58y + 2.6.
100
90
80
70
60
50
40
30
20
10.
0
5
10
15
20
25
30
Test Length Drop (mm W.G.)
Mean Velocity Head (mm W.G.)
y.

Section 8 STATIONARY CAGE EXPERIMENTS

8.1. Scope

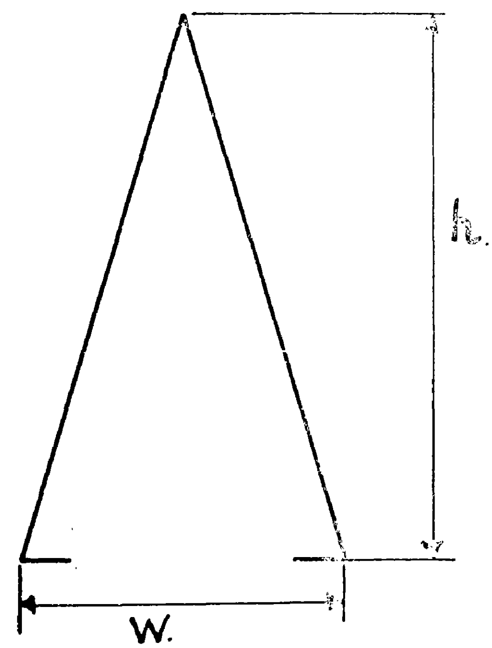
The purpose of the tests described in this section was to find the Pressure Drop Coefficients for several arrangements of cages and linings, and to study the effect on shaft resistance of the interaction of two cages of a winding system. Of particular interest was the length of the zones within which the combined resistance of two cages was dependent on the distance between the cages. Each test consisted of first finding the Pressure Drop Coefficient produced by the two cages side by side and then increasing the distance between the cages in stages until no further fall in Pressure Drop Coefficient took place. It was hoped that this information would be of use in predicting the conditions likely to be found in full scale installations as well as providing an interesting basis for comparison with later work on moving cages.

In the two-guide system (Fig. 6.2) the tests covered two-deck models, four-deck models, and a two deck model with an appropriately sized counterweight. In the four-guide system which can be operated either as a two cage or four cage winding installation tests were carried out on two sizes of cages as shown in Fig. 6.3. On the four cage winding system tests were confined to those with two-deck cages (series A/2) running on the two inside guides. These cages occupied only a small proportion of the available shaft area and only slightly affected the conditions as they passed. The larger cages of the two-cage winding system produced much more marked effects and work on them was extended to include two-deck cages (series C/2) and four-deck cages (series C/4).

The tests also covered cages of this type fitted with straight-sided nose and tail fairings. The outline of the two types of fairing used is shown in Fig. 8.1. The ratio of the fairing height (h) to the width of the base (w) is defined as the aspect ratio. For nose fairings it was fixed at 2 and for tail fairings at 3, the choice of values being based on earlier work by Stevenson. The right-angled fairings were fixed to the cages with the shorter side innermost so that two cages side by side presented a triangular profile to the air flow. The following table gives the details of the four guide systems tested with reference numbers.

Series	Cage Details				Fairing Details		Overall Height ins.
	Length ins.	Width ins.	No. of Decks	Total Height ins.	Nose	Tail	
A/2	$5\frac{3}{8}$	2	2	$6\frac{3}{8}$	None	None	$6\frac{3}{8}$
C/2	$5\frac{3}{8}$	4	2	$6\frac{3}{8}$	None	None	$6\frac{3}{8}$
C/2/AN	$5\frac{3}{8}$	4	2	$6\frac{3}{8}$	Type A	None	$14\frac{3}{8}$
C/2/ANT	$5\frac{3}{8}$	4	2	$6\frac{3}{8}$	Type A	Type A	$26\frac{3}{8}$
C/2/BN	$5\frac{3}{8}$	4	2	$6\frac{3}{8}$	Type B	None	$14\frac{3}{8}$
C/2/BNT	$5\frac{3}{8}$	4	2	$6\frac{3}{8}$	Type B	Type B	$26\frac{3}{8}$
C/4	$5\frac{3}{8}$	4	4	$12\frac{3}{8}$	None	None	$12\frac{3}{8}$
C/4/AN	$5\frac{3}{8}$	4	4	$12\frac{3}{8}$	Type A	None	$20\frac{3}{8}$
C/4/ANT	$5\frac{3}{8}$	4	4	$12\frac{3}{8}$	Type A	Type A	$32\frac{3}{8}$
C/4/BN	$5\frac{3}{8}$	4	4	$12\frac{3}{8}$	Type B	None	$20\frac{3}{8}$
C/4/BNT	$5\frac{3}{4}$	4	4	$12\frac{3}{8}$	Type B	Type B	$32\frac{3}{8}$

Type A Isosoceles



Type B Right-Angled

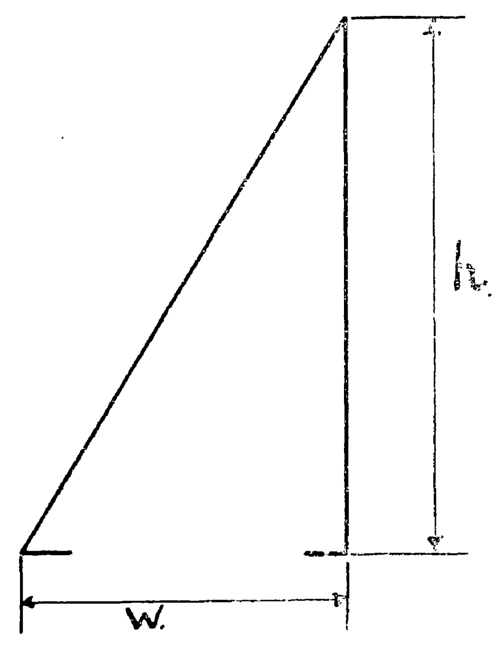


FIG. 8.1. OUTLINE OF CAGE FAIRINGS

8.2. Calculation of Pressure Drop Coefficient.

The Pressure Drop Coefficient (P.D.C.) of a cage is defined as the ratio of the loss in pressure produced by the cage to the mean velocity head in the shaft.

To measure the P.D.C. of any cage arrangement the cages were placed in the appropriate established test length (Sections 7.2 and 7.4) and simultaneous readings of head on the flowmeter and drop across the test length were taken at several Reynolds Numbers in the available range. The proportion of the experimental test length drop due to the lining was estimated from the calibration (Graph 7.2 or 7.4). The remainder of the measured drop was attributed to the cage(s) and was divided by the mean velocity head to give the cage P.D.C. Generally good agreement existed between the values of P.D.C. at different Reynolds Numbers and such variations as did occur were completely random. It was therefore possible to conclude that the value of P.D.C. was independent of Reynolds Number within the range covered.

8.3. Results

8.3.1) Shaft Linings.

From the calibrations of the test lengths (Section 7) the P.D.C.'s of lengths of empty and lined shafts were calculated. To make comparisons possible the values were reduced to those for 1,000 feet of 24-foot diameter shaft.

The values obtained were :-

Empty Shaft - 0.58

Two-Guide Lining - 2.81

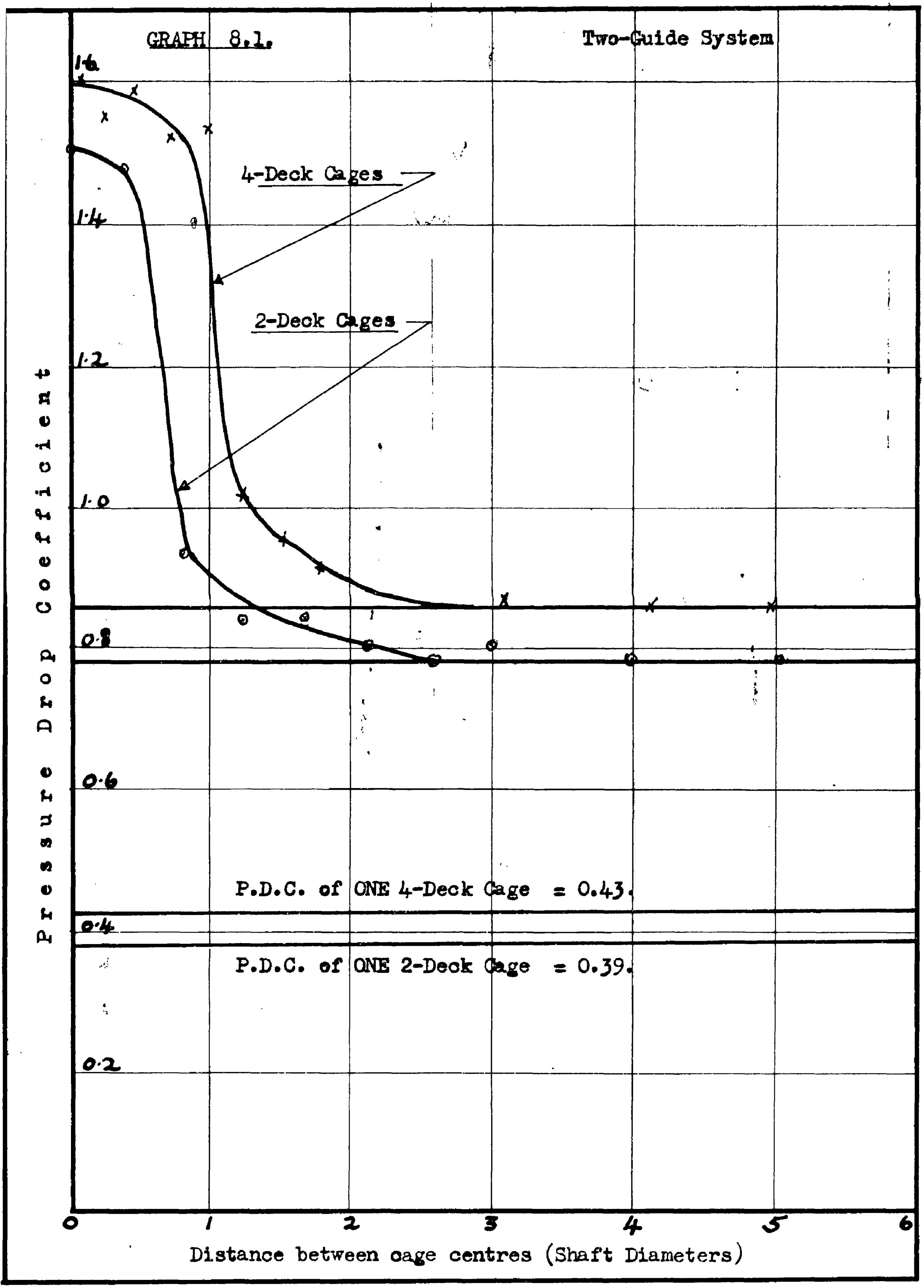
Four-Guide Lining - 5.32

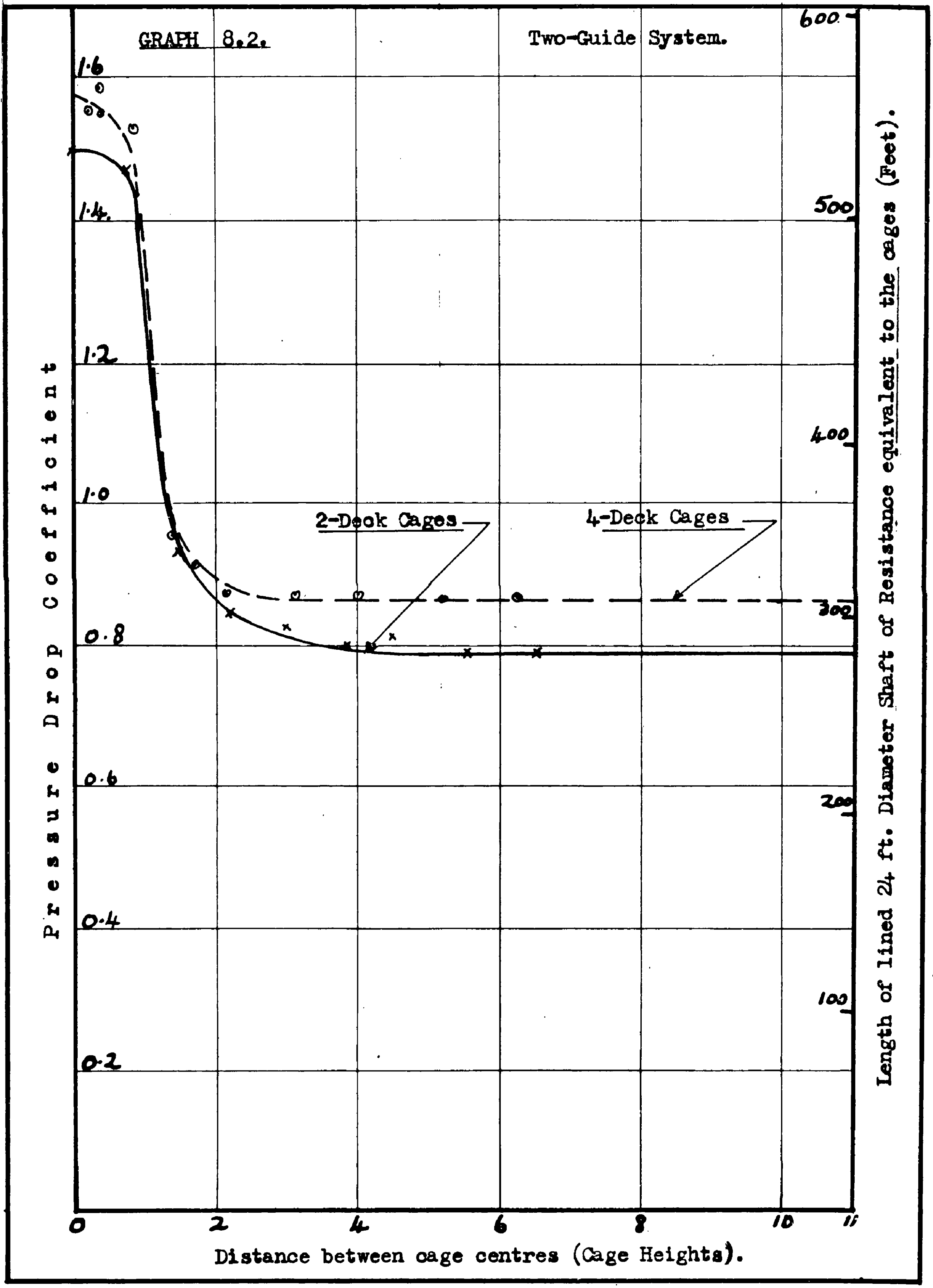
These figures show that the resistance to airflow of a mine shaft is greatly increased by the presence of buntons - an effect which has been established by several investigators. The figures also show how widely the effect can vary from shaft to shaft. In the two guide system the presence of buntons increases the shaft resistance by 4.85 times while in the four guide system the increase is 9.2 times. The difference can almost certainly be attributed to the change in bunton pattern and the effect of the guides can be assumed to be very small. The four guide system with its longer buntons placed more centrally in the shaft could be expected to have the higher resistance but that such an apparently small change in bunton pattern should almost double the shaft resistance underlines the problem faced by the ventilation planning engineer who tries to estimate this resistance. It is obvious that no simple solution to this problem exists. Attempts have been made (8.1, 8.2) to derive empirical formulae for the purpose with some success. The number of variables involved, however, limits the application of simpler formulae and makes more comprehensive ones unwieldy. In other cases (e.g. 8.3) an estimate of the resistance of a shaft has been made by building a model of it and testing in the laboratory. Such tests have given valuable information some of which has been satisfactorily checked from the results of surveys of the full scale shaft.

8.3.2. Two Guide System.

Before starting the tests a further check on the airflow conditions in the duct was made by measuring the P.D.C. of a single cage at various points in the test length. The results showed satisfactory consistency and tests involving dual cages were begun. Graph 8.1 shows the P.D.C. plotted on a base of distance between the centre lines of the two cages and shows that for both two-deck and four-deck models the total resistance to air flow of the two cages can be considered to be the sum of the resistances of the individual cages when the distance between the cage centre lines exceeds $2\frac{1}{2}$ shaft diameters. As the clearance between the cages becomes less than this the P.D.C. shows a gradual rise followed by a very rapid rise to approximately twice the steady value, followed again by a gradual rise to a maximum when the cages are side by side. Graph 8.2 shows the same information with the distance between the cages expressed in terms of the cage height. When plotted this way, the curves for two-deck models and four-deck models fall quite close together with the greatest rate of increase taking place when the distance between the centres is about equal to the height of a cage. This shows that for both types of cage the serious increase in shaft resistance takes place during the period when the cages overlap in the shaft. The maximum resistance presented during the passing of two two-deck cages is 3.9 times the resistance of a single cage. For four-deck cages this figure is 3.7 times, suggesting that the greater cage height affects the duration of the high resistance much more than its value.

In the case of the two-deck cage and counterweight system the


GRAPH 8.1.
Two-Guide System
4-Deck Cages
2-Deck Cages
P.D.C. of ONE 4-Deck Cage = 0.43.
P.D.C. of ONE 2-Deck Cage = 0.39.
Pressure Drop Coefficient
1·6
1·4
1·2
1·0
0·8
0·6
0·4
0·2
0
1
2
3
4
5
6
Distance between cage centres (Shaft Diameters)


GRAPH 8.2.
Two-Guide System.
Pressure Drop Coefficient
Length of lined 24 ft. Diameter Shaft of Resistance equivalent to the cages (Feet).
1.6
1.4
1.2
1.0
0.8
0.6
0.4
0.2
600
500
400
300
200
100
0
2
4
6
8
10
11
2-Deck Cages
4-Deck Cages
Distance between cage centres (Cage Heights).

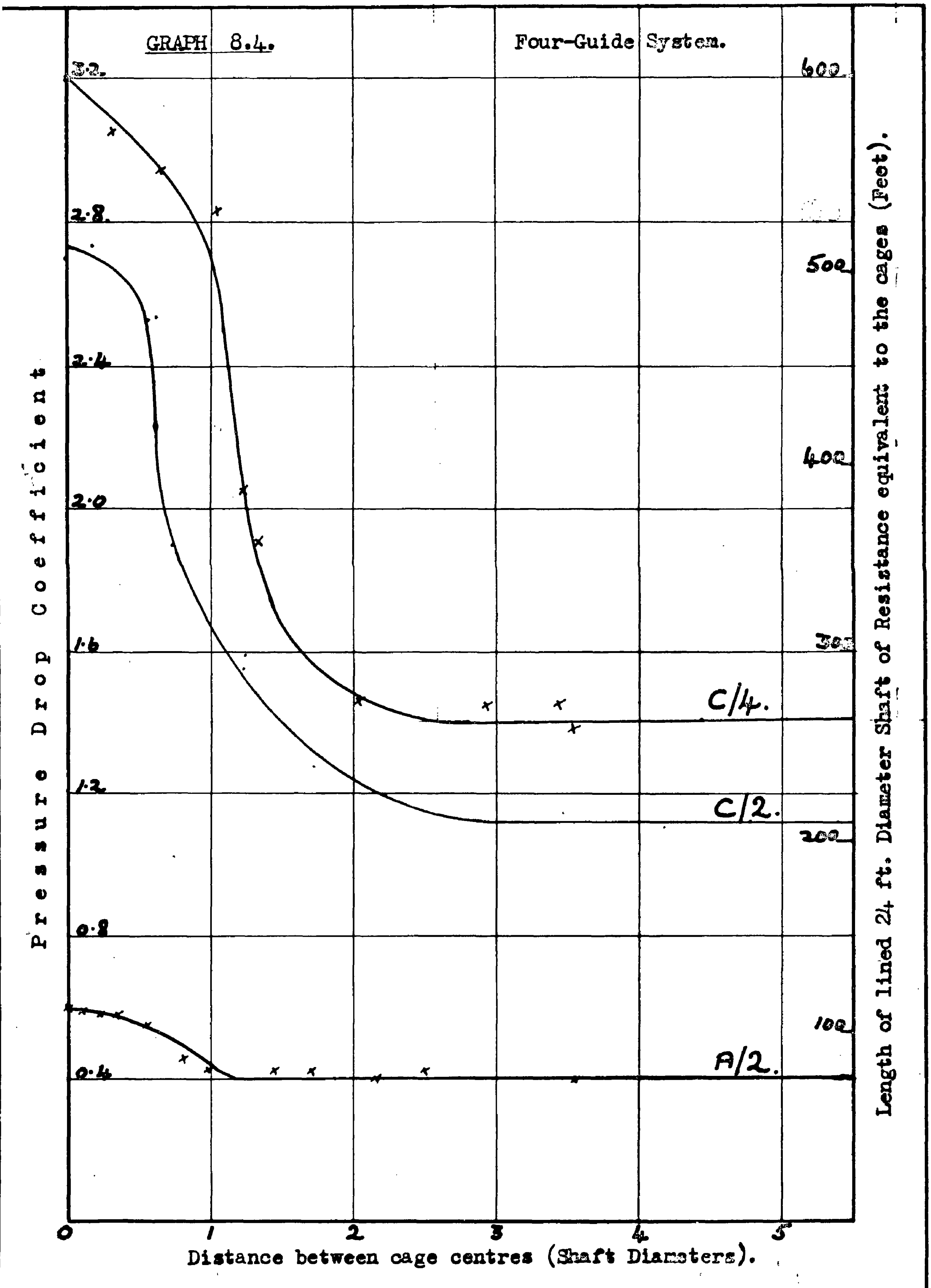
Length of lined 24 ft. Diameter Shaft of Resistance equivalent to the cages (Feet).
300
200
100
GRAPH 8.3.
Two-Guide System.
2-Deck Cage and Counterweight.
P.D.C. of C/wt. = 0.07.
P.D.C. of Cage = 0.39.
0.8
0.6
0.4
0.2
0
4
3
2
1
0
1
2
3
4
Distance between centres (Shaft Diameters)
Cage Upstream
Cage Downstream
Pressure Drop Coefficient

maximum P.D.C. occurs when the centre of the cage is 0.4 shaft diameters upstream from the centre of the counterweight. The increase in resistance as the cage approaches the counterweight from the upstream side is very sharp while the decrease to the normal value as the cage recedes downstream is more gradual. From this it would appear that the effect of the counterweight is at its greatest when the counterweight lies in the wake of the cage hindering the recovery of static pressure from the high velocity head of the air passing through the restricted space between the cage and the shaft wall. The resistance of the counterweight alone in the shaft is very small yet its presence in the vicinity of the cage increases the resistance due to the winding appliances by about 45% of the normal value. The total effect extends for about $3\frac{1}{2}$ shaft diameters in all compared with 5 shaft diameters in the case of passing cages.

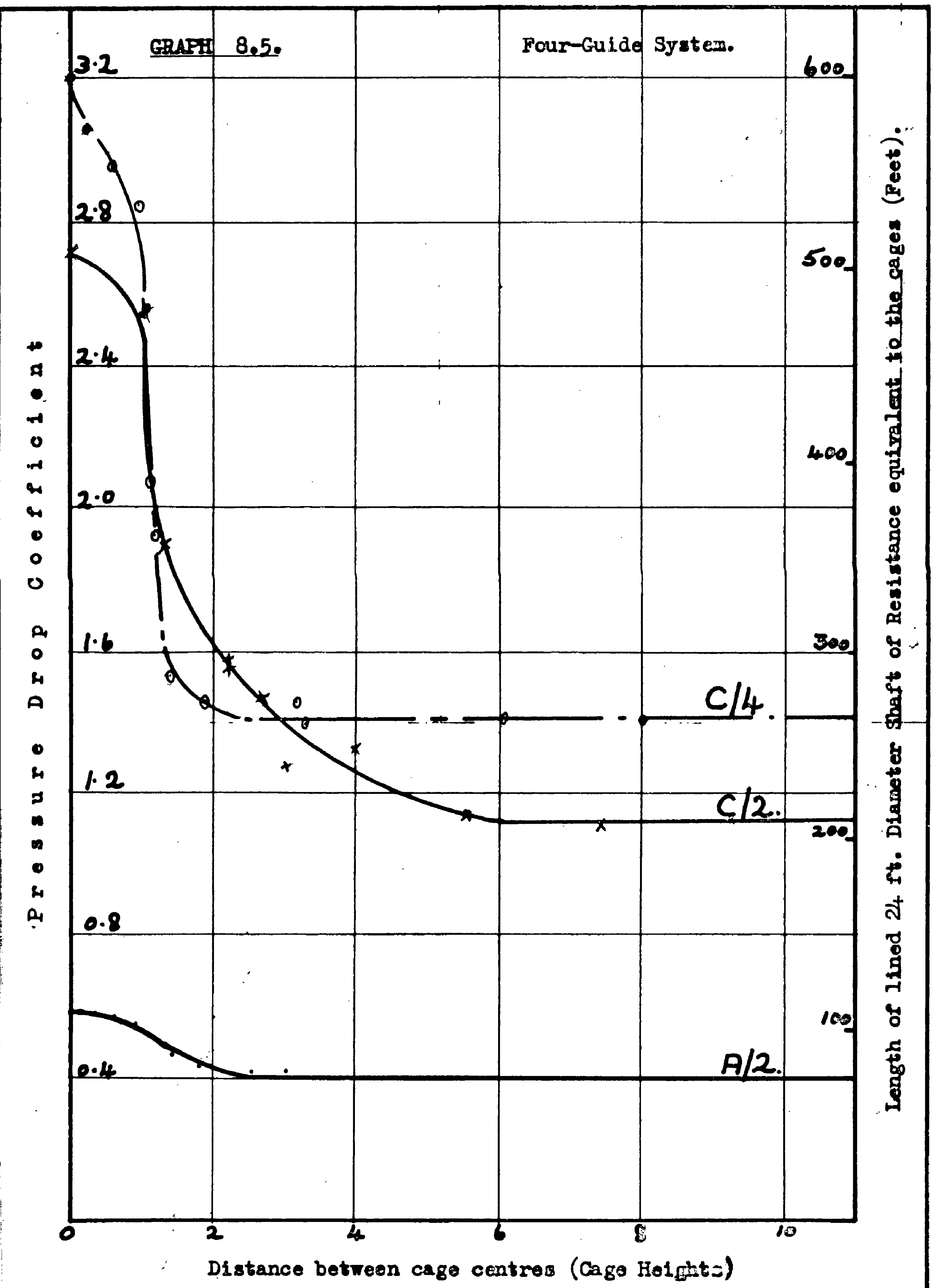
8.3.3. Four Guide System.

Cage Resistances (Graphs 8.4 and 8.5)

These graphs show the variation in P.D.C. as two cages pass in a shaft and are similar in shape to the corresponding graphs in the two guide system. The resistance due to the cages exceeds its normal value when the distance between the cage centres is less than 3 shaft diameters (Graph 8.4) and its greatest rate of increase takes place when the distance is approximately equal to the height of a cage (Graph 8.5). The maximum value of cage resistance is 2.4 times the normal value (i.e. 4.8 times the resistance of a single cage) for two-deck cages and 2.3 times the normal (i.e. 4.6 times a single cage) for four-deck cages.



GRAPH 8.4.
Four-Guide System.
Pressure Drop Coefficient
Length of lined 24 ft. Diameter Shaft of Resistance equivalent to the cages (Feet).
Distance between cage centres (Shaft Diameters).
3.2
2.8
2.4
2.0
1.6
1.2
0.8
0.4
600
500
400
300
200
100
0
1
2
3
4
5
C/4.
C/2.
A/2.


GRAPH 8.5.
Four-Guide System.
3·2
2·8
2·4
2·0
1·6
1·2
0·8
0·4
600
500
400
300
200
100
C/4.
C/2.
A/2.
0
2
4
6
8
10
Pressure Drop Coefficient
Length of lined 24 ft. Diameter Shaft of Resistance equivalent to the cages (Feet).
Distance between cage centres (Cage Heights)

These figures are higher than in the two-guide system due to the slightly greater frontal area and because the cages are short and wide rather than long and narrow and approach the sides of the duct more closely. The resistance produced by four-deck cages is only slightly greater than that produced by two-deck cages probably because shock and recovery losses are comparable irrespective of the number of decks and the friction loss is greater for four decks. The resistance of the smaller cages used for four cage winding (series A/2) is very much less than the others due to the much smaller area of shaft occupied.

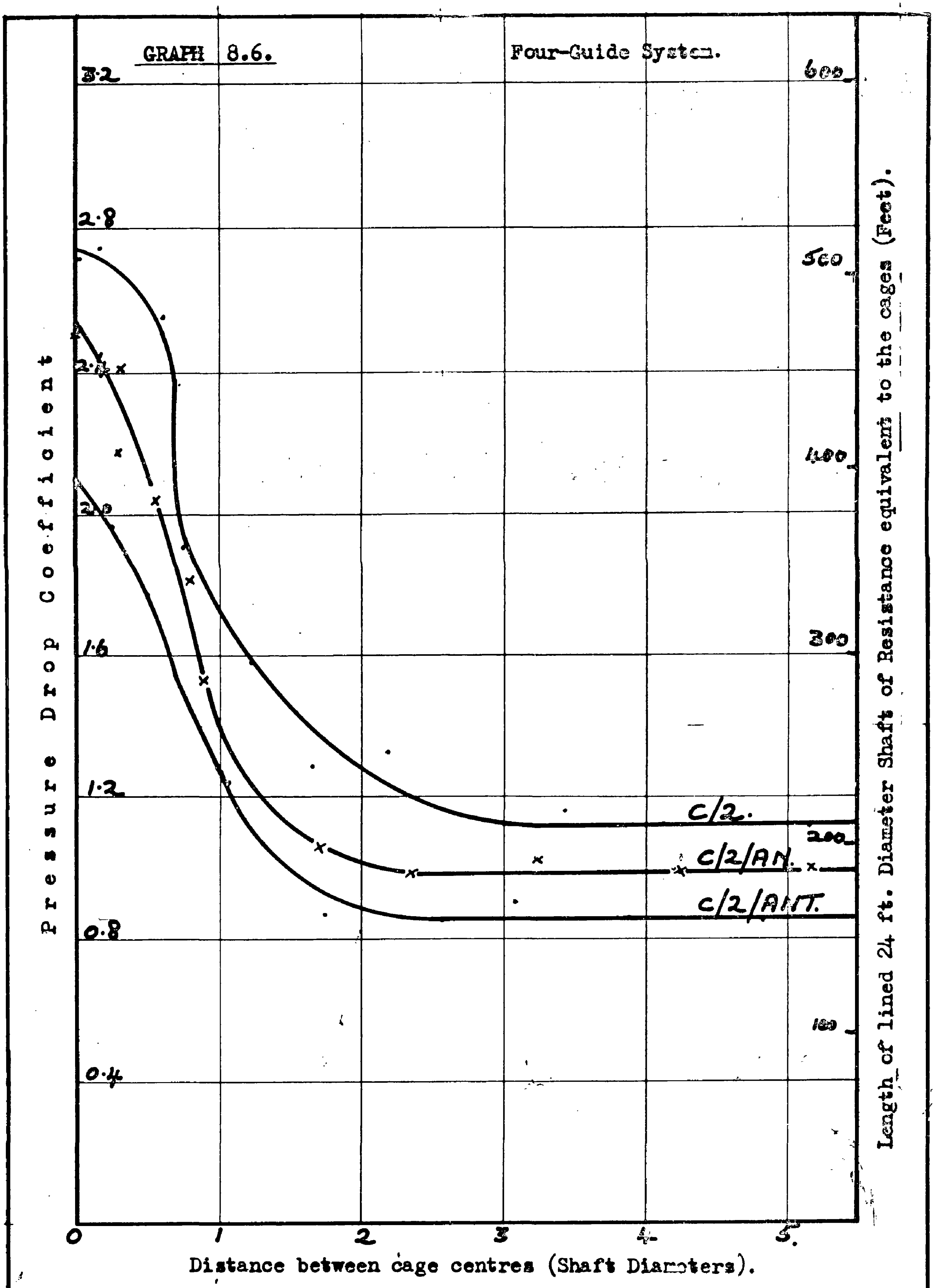
Comparison between the two guide system and the four guide system and consideration of the importance of the presence of cages to shaft resistance is best made by considering the cage resistances relative to the shaft resistance. For this purpose the vertical ordinate of the graphs is marked in terms of the length of lined 24-foot diameter shaft of resistance equal to the cages as well as in terms of P.D.C. This shows that while the absolute value of cage P.D.C. in the two-guide system is much lower than in the four-guide system the presence of cages in the two-guide system results in a slightly greater proportional increase in overall shaft resistance. For both systems the normal value of cage resistance is equivalent to between 200 and 300 feet of lined 24-foot diameter shaft and the maximum value is equivalent to between 500 and 600 feet of lined shaft. In the four-cage system the normal value of cage resistance can be considered equivalent to about 150 feet of lined shaft. An estimate of the value of cage resistance in terms of shaft resistance was made using the empirical methods of current practice

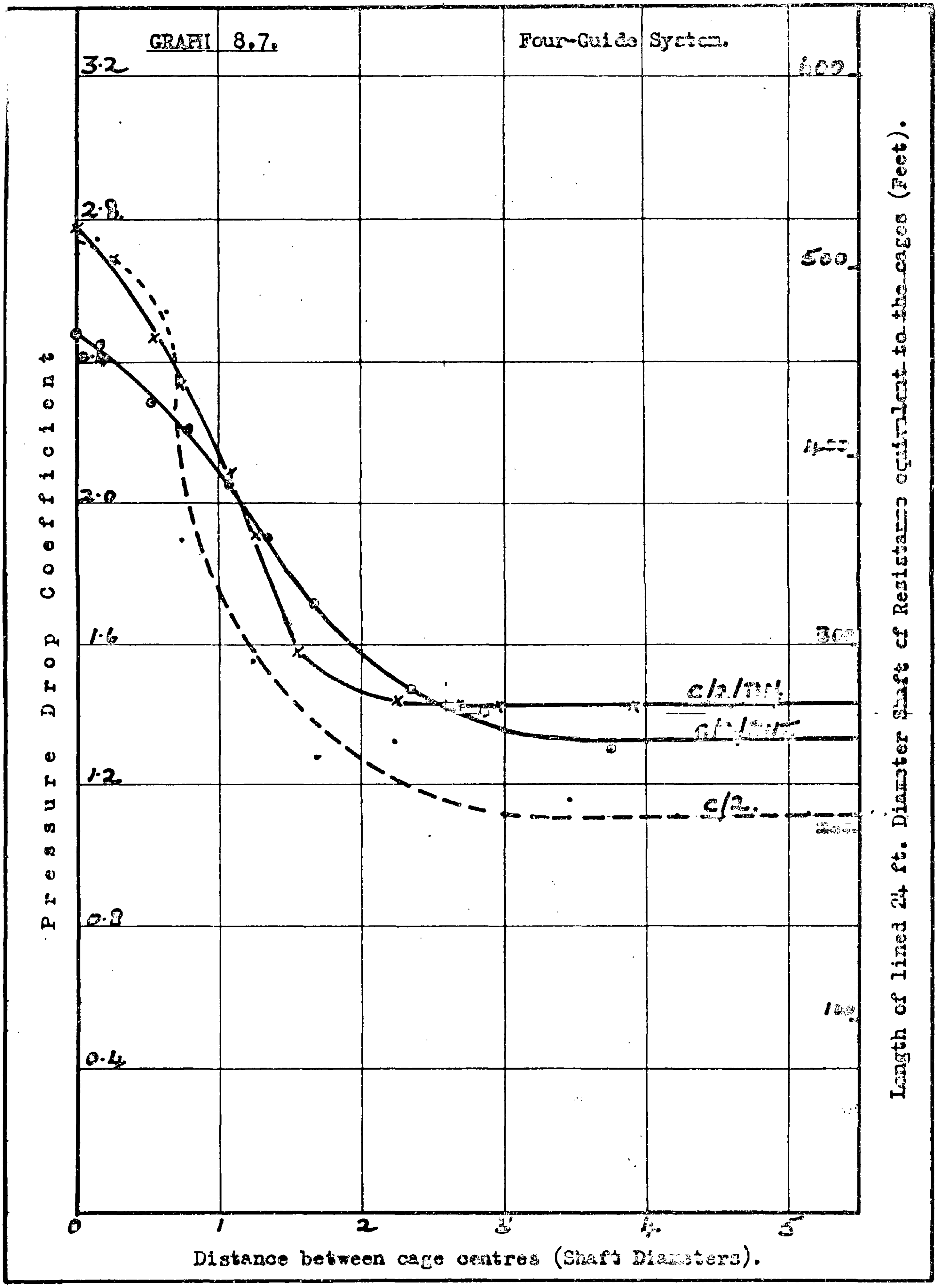
(8.4) and comparable results were obtained. In view of this it is suggested that the additional resistance produced by the presence of cages in shafts should not be ignored in ventilation planning unless the shaft depth exceeds 3,000 feet in the case of a two-cage winding system, and 1,500 feet in the case of a four-cage winding system.

<u>Cages with Fairings (Graphs 8.6, 8.7 and 8.8)</u>

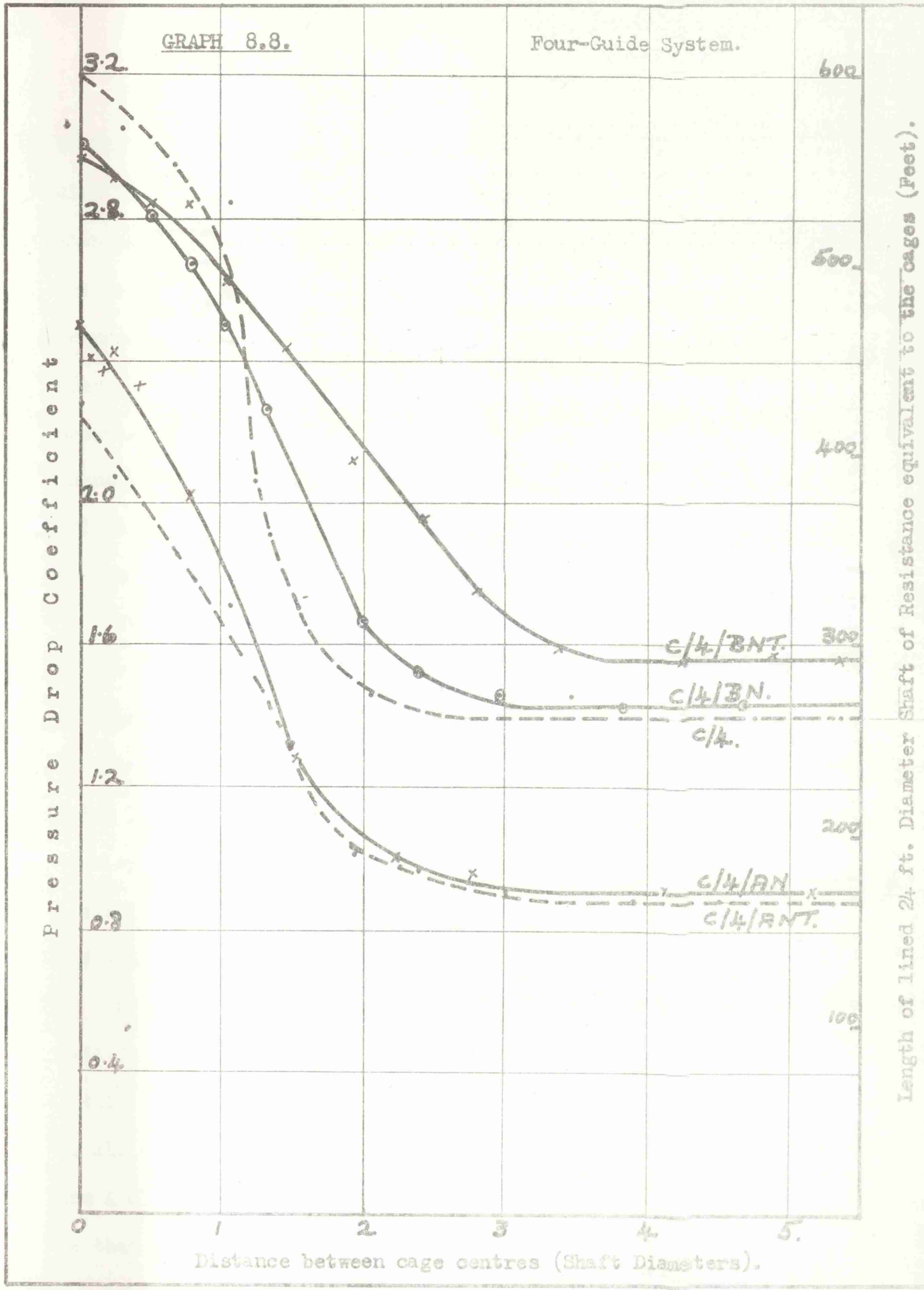
Relative values of the resistance of cages of the four-guide system with and without fairings are shown in the following table.

System	Normal Resistance			Maximum Resistance		
	P.D.C.	Ft. of Shaft Equivalent	Relative	P.D.C.	Ft. of Shaft Equivalent	Relative
C/2	1.12	210	100	2.74	515	100
C/2/AN	0.96	180	86	2.53	475	92
C/2/ANT	0.84	157	75	2.08	390	76
C/2/BN	1.42	266	127	2.78	520	101
C/2/BNT	1.30	244	116	2.48	465	90
C/4	1.40	262	100	3.20	600	100
C/4/AN	0.90	169	64	2.50	470	78
C/4/ANT	0.88	165	62	2.29	430	71
C/4/BN	1.44	270	103	3.00	560	94
C/4/BNT	1.56	292	112	2.96	555	92


GRAPH 8.6.
Four-Guide System.
3.2
2.8
2.4
2.0
1.6
1.2
0.8
0.4
Pressure Drop Coefficient
600
500
400
300
200
100
Length of lined 24 ft. Diameter Shaft of Resistance equivalent to the cages (Feet).
C/2.
C/2/AN.
C/2/ANT.
0
1
2
3
4
5.
Distance between cage centres (Shaft Diameters).



GRAPH 8.7.
Four-Guide System.
Pressure Drop Coefficient
Length of lined 24 ft. Diameter Shaft of Resistance equivalent to the cages (Feet).
3·2
2·8
2·0
1·6
1·2
0·8
0·4
500
c/2.
0
1
2
3
4
5
Distance between cage centres (Shaft Diameters).



GRAPH 8.8.
Four-Guide System.
3.2
2.8
2.0
1.6
1.2
0.8
0.4
600
500
400
300
200
100
Pressure Drop Coefficient
Length of lined 24 ft. Diameter Shaft of Resistance equivalent to the cages (Feet).
C/4/BNT.
C/4/BN.
C/4.
C/4/AN
C/4/ANT.
0
1
2
3
4
5.
Distance between cage centres (Shaft Diameters).

The use of fairings of type A considerably reduces the resistance of the cages. Their efficiency is greater with four-deck cages than with two-deck cages and is greater when the cages are spaced apart than when they are side by side. Fairings of type B produce a smaller reduction in resistance when the cages are side by side but they actually produce an increase in resistance when the cages are outside the passing zone. It would appear, therefore, that the reduction in shock loss produced by the use of assymetrical fairings does not compensate for the increased friction loss due to the greater rubbing surface area. The effect is more marked with two-deck cages where the proportional increase in rubbing surface is greater.

When compared with some earlier results on a system very similar to our two-guide system the reductions brought about by the use of fairings of type A give comparable figures. Stevenson did not examine fairings of type B on cages supported in rigid guides but in tests on a model rope guide layout he found that fairings of this type, while less efficient than symmetrical ones, still produced a reduction in the resistance of a single cage. The presence of buntons obviously nullifies this reduction and makes this type of fairing unsuitable for a rigid guide system.

From the graphs it is seen that except in the case where nose and tail fairings of type B are used the presence of the fairings makes very little addition to the length of the zone of above normal resistance. In all cases steady conditions are reached by the time the cage centres are 4 shaft diameters apart. The most useful property of the fairings is that they avoid the very sudden increases in resistance produced by the passing of cages alone. It is unfortunate that the fairings of type B give a more gentle curve than those of type A which are preferred on grounds of lower aerodynamic resistance.

Section 9 MOVING CAGE EXPERIMENTS

9.1. General

Experiments on the influence of the moving of cages in the model shaft were carried out on all the cage series listed in Section 8.1 (page 47) except C/2/BN and C/2/BNT and the same notation was used to identify the different series. The tests involved using the pressure recording apparatus to get a continuous record of the pressure at several points in the model shaft as the cages passed each other. The results can be conveniently divided into four sections as follows:-

a) Tests on the influence of passing cages on shaft resistance - done by recording the pressure drop across the test length "Bb" (Fig. 6.1).

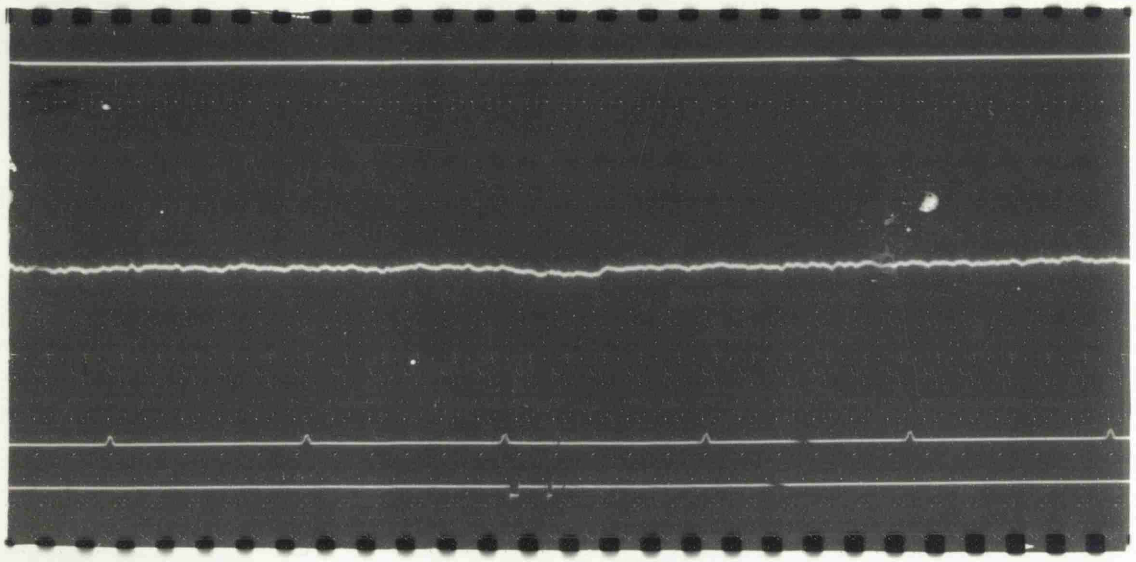
b) Tests on the influence of passing cages on the conditions upstream from the passing place - done by recording the static pressure at the points "a" and "b" (Fig. 6.1).

c) Tests on the influence of passing cages on the conditions downstream from the passing place - done by recording the static pressure at the points "A", "B" and "C" (Fig. 6.1).

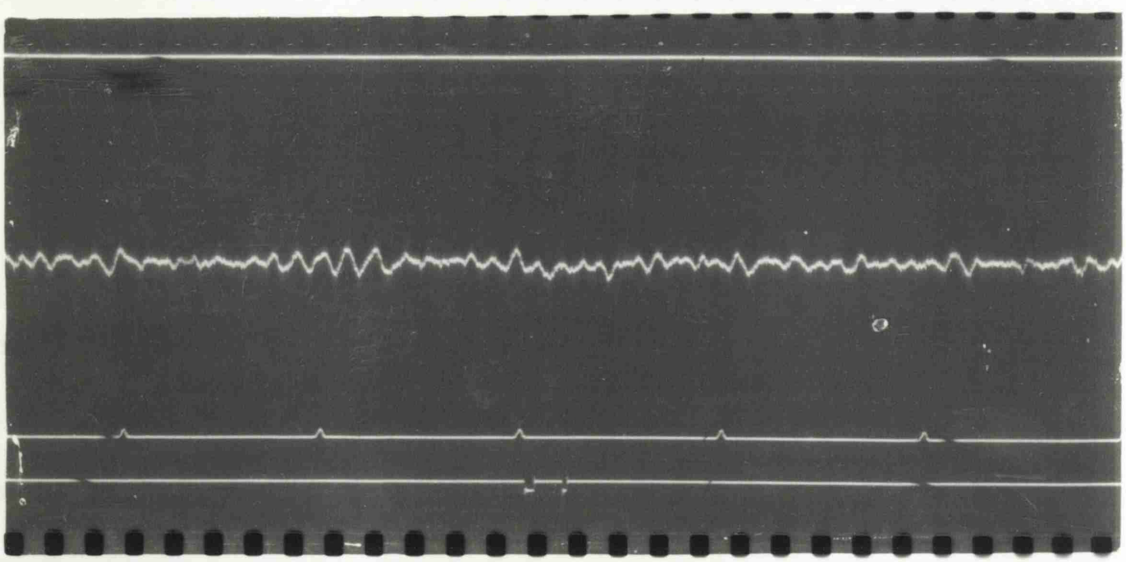
d) Tests on the influence of a cage passing a point on the conditions at that point - done by recording the static pressure at the point "A" as a cage passed "A".

In early tests the static pressures were taken with $\frac{1}{4}$ inch bore tubes inserted into the duct at right angles to the airstream. Part of a typical record obtained by this method is shown in Fig. 9.1 and shows the natural fluctuations in the pressure in the duct. The amplitude of these fluctuations is comparable with the variations in pressure produced by the movement of the cages and estimation of the mean pressure is very difficult. The frequency of the fluctuations, however, is much higher than the equivalent frequency of the variations produced by the cages and hence it was possible to damp them out by using an N.P.L. pitot-static tube to measure the static pressure. Records of the same pressure recorded by the two methods are shown in Fig. 9.1 and show clearly the superiority of the pitot-static tube for the purpose.

Records of pressure were taken at different Reynolds Numbers and with the cages moving in both directions. In each test recording was started with the cages sufficiently far apart to ensure that there was no interaction between them. Recording was continued while the cages approached, passed and receded until the pressure being measured returned to its original normal value. In order to make different tests properly comparable the pressure readings were plotted as a percentage of this normal value on a base of the distance between the cage centres expressed in terms of the shaft diameter. Graph 9.1 shows the plot of a typical set of tests conducted at different air speeds and with the cages travelling in both directions plotted in this way. Good agreement was obtained and it was therefore possible to reduce a series of tests to a single curve representing the variation in the pressure at any one point



a) Using Pitot-Static Tube.



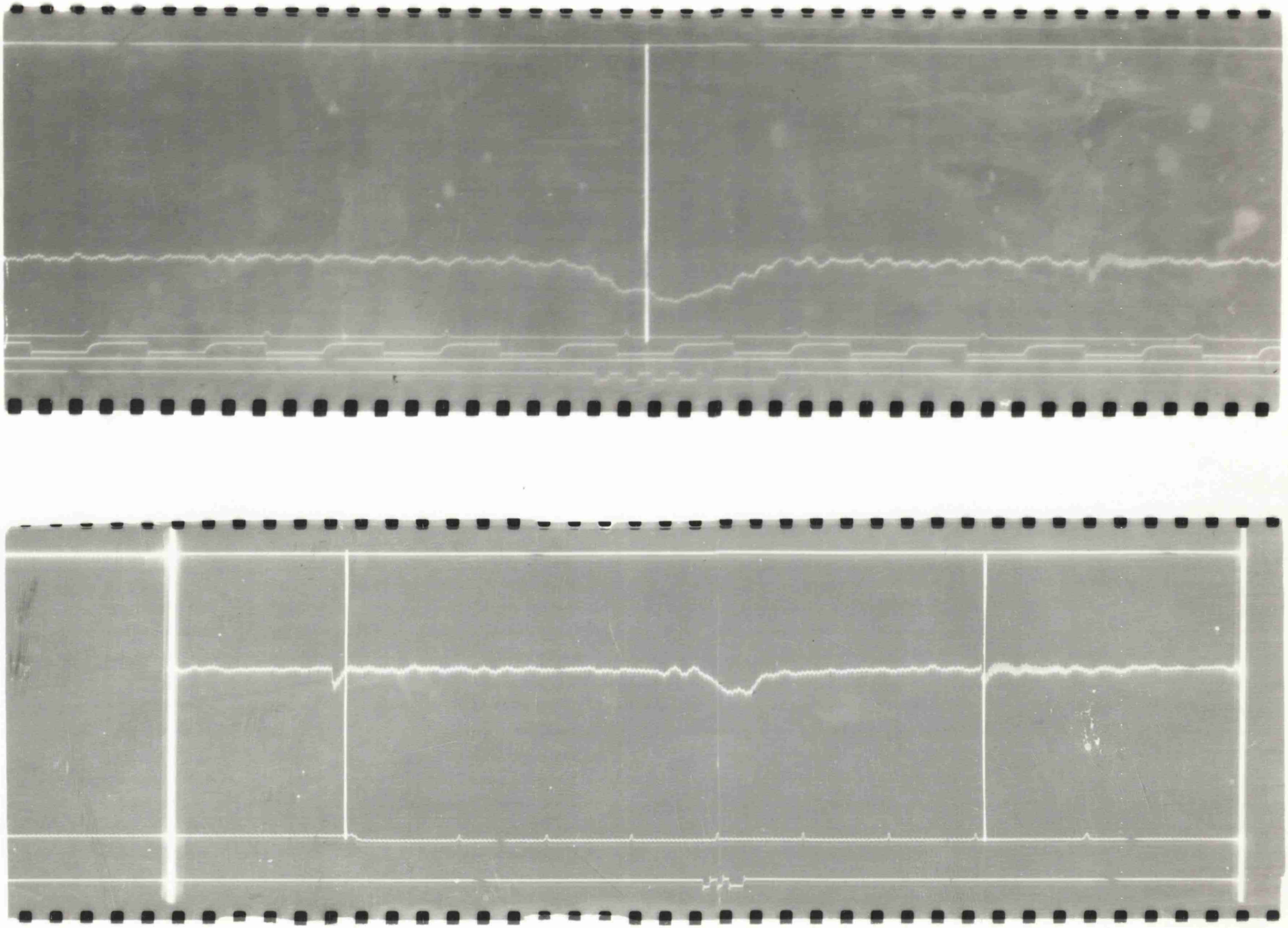
b) Using $\frac{1}{4}$ in. bore tube.

FIG.9.1. METHODS OF RECORDING PRESSURES.

as the cages passed at any one speed for all Reynolds Numbers in the range. A further advantage of this method of plotting is that the use of relative values of pressure relies only on the nature of the calibration of the pressure recording equipment and not on its absolute value. This avoided any difficulty arising from the slight instability of the equipment (referred to in Section 4) since the calibration could be relied upon to be a straight line passing through the origin even although the slope varied slightly from day to day depending on the conditions of temperature and mains voltage.

Fig. 9.2. shows some of the records their actual size. In the graphs the pressure scale is increased by about five times and the time scale is converted to distance and is increased by about twice at the lowest cage speeds and about four times at the highest cage speeds. This results in an apparent scatter of the experimental points about the mean line due to exaggerated measuring errors. The records also show that the normal pressure is subject to occasional random variations whose magnitudes are not negligible compared with the variation produced by the cages and which could not be separated from the cage effects if they occurred at the same time. This effect also produces some scatter. The graphs which appear in this section, however, are all the result of at least four tests under different conditions and can be considered to represent closely the pressure variations in the shaft due to the movement of the cages.

In the course of the experimental work on nine series of cages the variations in eight different quantities at four cage speeds were



F I G. 9.2. T Y P I C A L   R E C O R D S.

recorded and the results provided material for about 300 curves. A selection of these curves showing the characteristics of unstreamlined cages in full and the important trends in the characteristics of streamlined cages is shown in Graphs 9.2. to 9.21. In all these graphs the part to the left of the central ordinate ($y = 0$) represents the conditions while the cages are approaching each other and the part to the right represents the conditions after they have passed.

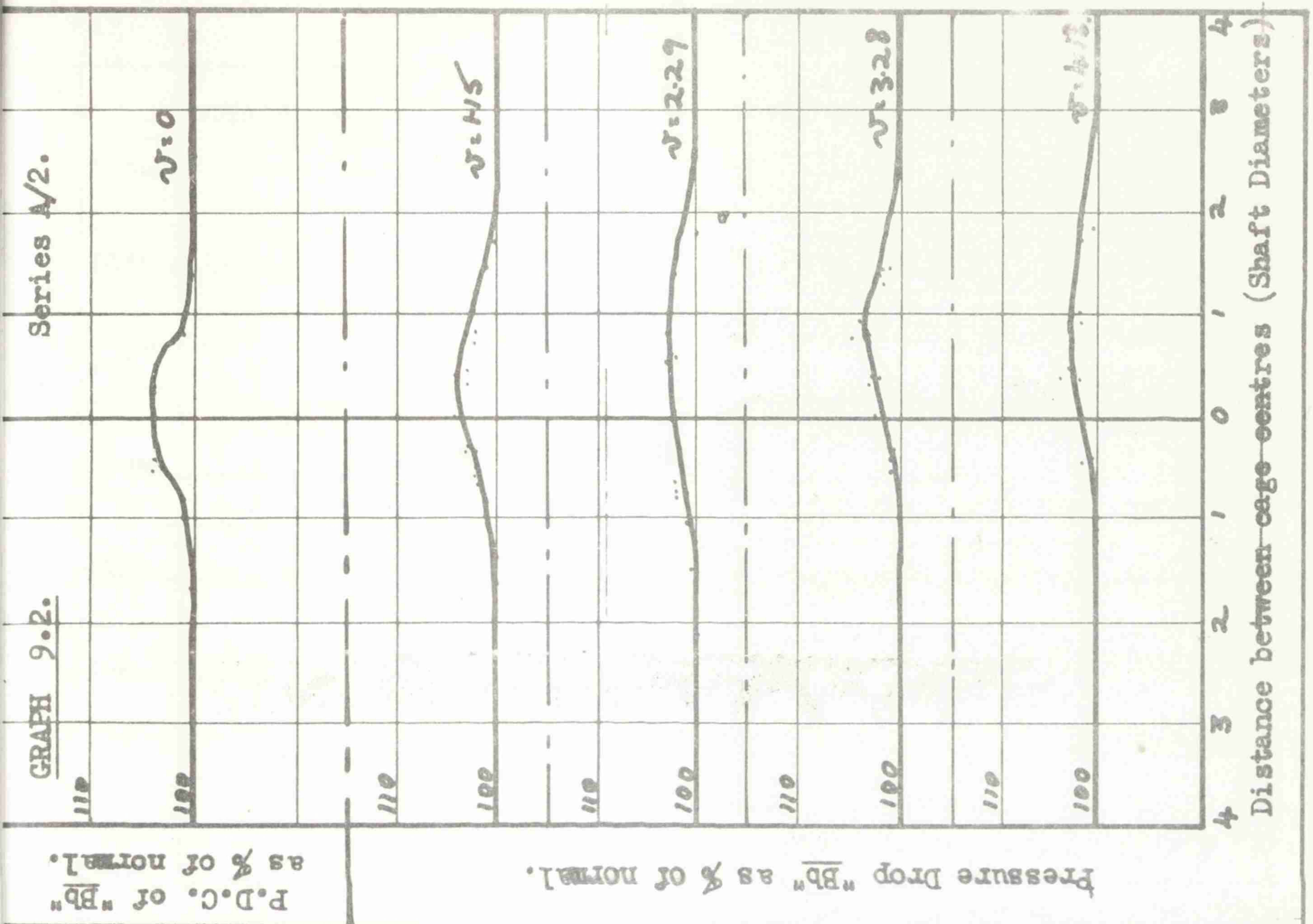
9.2. Effect of Passing Cages on Shaft Resistance (Graphs 9.2 to 9.6).

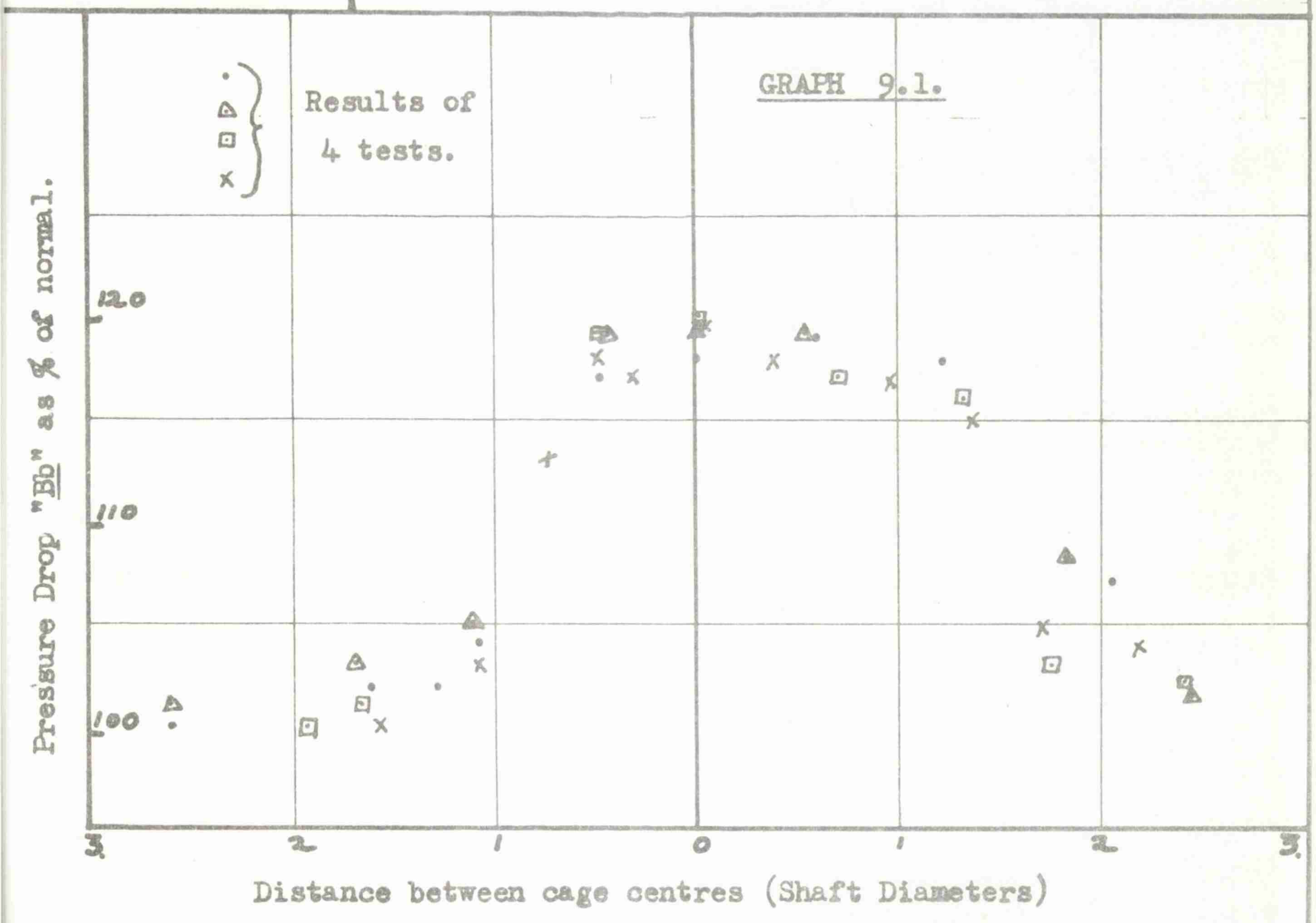
Graphs 9.2, 9.3 and 9.4 show the variation in the pressure drop across the test length "Bb" as cages pass at a range of speeds. The speeds vary from about half the maximum possible up to the maximum and are expressed in shaft diameters per second throughout. The variation in P.D.C. of the test length plus the cages was calculated from the results of the tests on stationary cages and is shown for comparison. The most marked effect is that while the curves for stationary cages are symmetrical about the centre line those for moving cages are not and that the degree of asymmetry increases with cage speed. Movement of the cages appears to increase the length of the zone within which there is interaction between the cages. Although the effect of increasing speed is to increase the length of the zone the rate of increase is less than the rate of increase in speed and the net effect is to reduce the duration of the period of abnormal pressure.

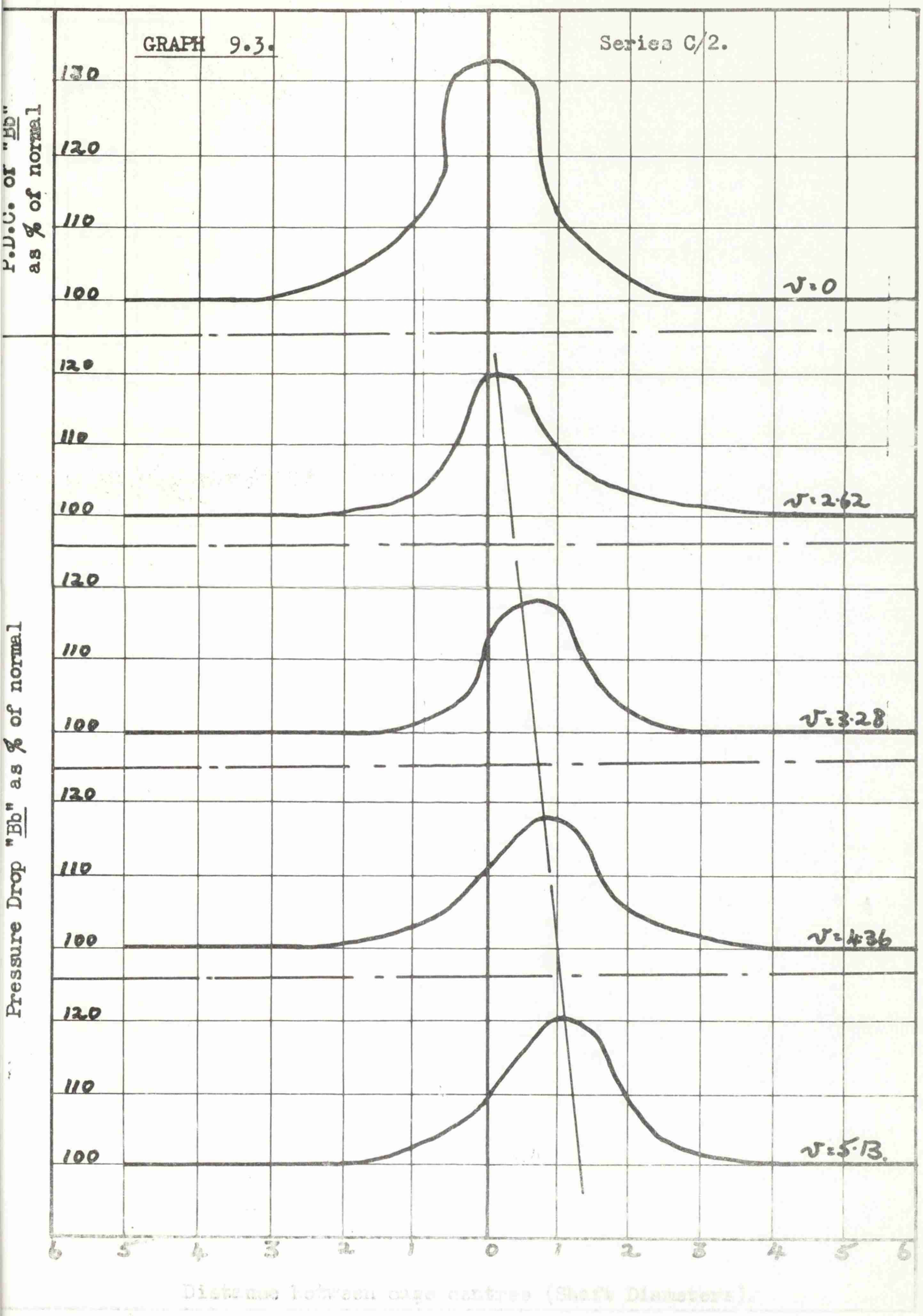
The passing of two cages of series A/2 produces a very small increase in pressure and appears unlikely to have any serious effect on the conditions in the shaft. More serious effects could be produced if three or all of the cages in a four-cage system came within the zone of interaction simultaneously. Since the four-cage system is usually used to wind with two pairs of cages from different levels the arrival of all the cages at one point will not be possible and serious effects should not occur.

When the larger cages of series C/2 and C/4 are used, however, a much more marked increase in the test length pressure drop is recorded. The shape of the curves is similar to that obtained for stationary cages in that for both types of cage the overall length of the zone of interaction is about the same, but the most serious increase occurs earlier and persists for longer in the case of four-deck cages. The maximum pressure drop recorded appears to be independent of cage speed within the range covered although the point at which it occurs is not. The effect of increasing speed is to make the rise and fall of pressure with distance more gentle without much effect on the maximum value.

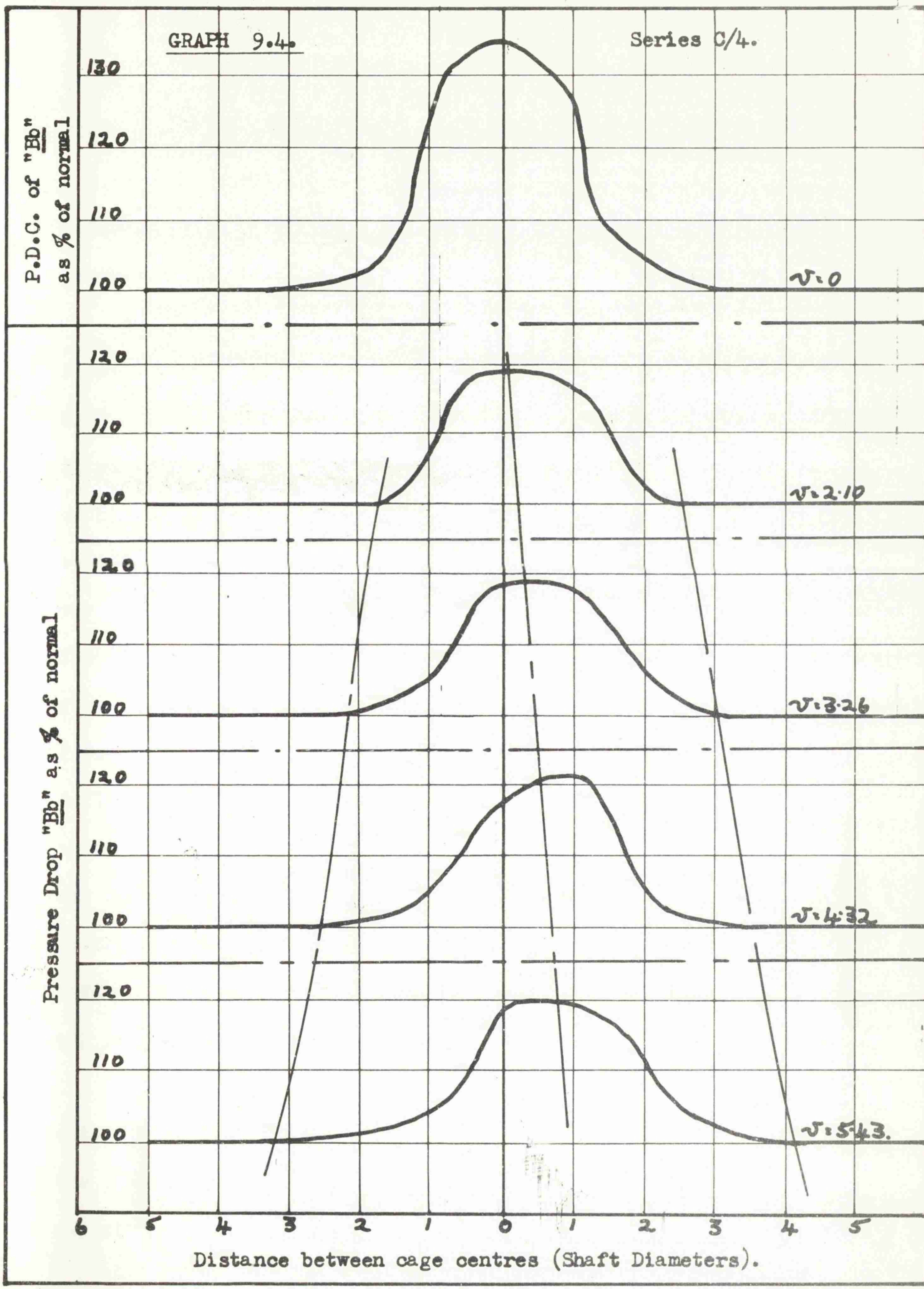
The increase in P.D.C. of the test length plus the stationary cages is about 50% greater than the increase in the test length drop produced by moving cages, but this does not mean that the resistance of the moving cages is lower than the resistance of the stationary cages since the curves are not strictly comparable. A curve of test length drop will be identical to a curve of P.D.C. only if the quantity of air flowing remains constant during the test. In the case of a mine shaft, however, the result of the passing of cages is a temporary increase in


GRAPH 9.2.
Series A/2.
v : 0
v : 1.15
v : 2.29
v : 3.28
Distance between cage centres (Shaft Diameters)
Pressure Drop "Bb" as % of normal.
P.D.C. of "Bb" as % of normal.


GRAPH 9.1.
Results of 4 tests.
Pressure Drop "Bb" as % of normal.
120
110
100
Distance between cage centres (Shaft Diameters)



GRAPH 9.3.
Series C/2.
P.D.C. of "Bb" as % of normal
Pressure Drop "Bb" as % of normal
130
120
110
100
v=0
v=2.62
v=3.28
v=4.36
v=5.13.
6 5 4 3 2 1 0 1 2 3 4 5 6
Distance between cage centres (Shaft Diameters).


GRAPH 9.4.
Series C/4.
P.D.C. of "Bb" as % of normal
Pressure Drop "Bb" as % of normal
130
120
110
100
v=0
v=2.10
v=3.26
v=4.32
v=5.43.
6 5 4 3 2 1 0 1 2 3 4 5
Distance between cage centres (Shaft Diameters).

the total resistance which is accompanied by a fall in the quantity flowing. The extent of the fall will depend on the fan characteristic and on the relative values of the cage resistance and the total circuit resistance. Such conditions vary widely from shaft to shaft and no general rules can be laid down.

An attempt was made to record the variation in the head on the flowmeter while the cages passed but no definite results were obtained. Fig. 9.3 shows the natural flowmeter head with its heavy fluctuations. It was possible to damp these out fairly satisfactorily by using lengths of narrow bore tubing in the connections but even then the method was not sufficiently accurate to give reliable quantitative results of the variation in flowmeter head. The graphs show that the test length drop reaches a maximum of 120% of its normal value. This increase combined with only a 7% drop in mean velocity head will result in a P.D.C. of 130% of the normal as found with the stationary cages. The deflection on the galvanometer produced by the flowmeter head was only about half an inch and it was not possible to measure a 7% change with any confidence. The decrease in quantity which could be detected on the records was very small showing that while the increase in resistance of moving cages is greater than the increase in test length drop it is not any greater than the increase in the resistance of stationary cages.

Examination of the records also showed that when the cages were outside the zone of interaction the test length drop was the same whether the cages were moving or not. Fig. 9.4. shows part of a record which was started before the cages began to move. The disturbance at the point

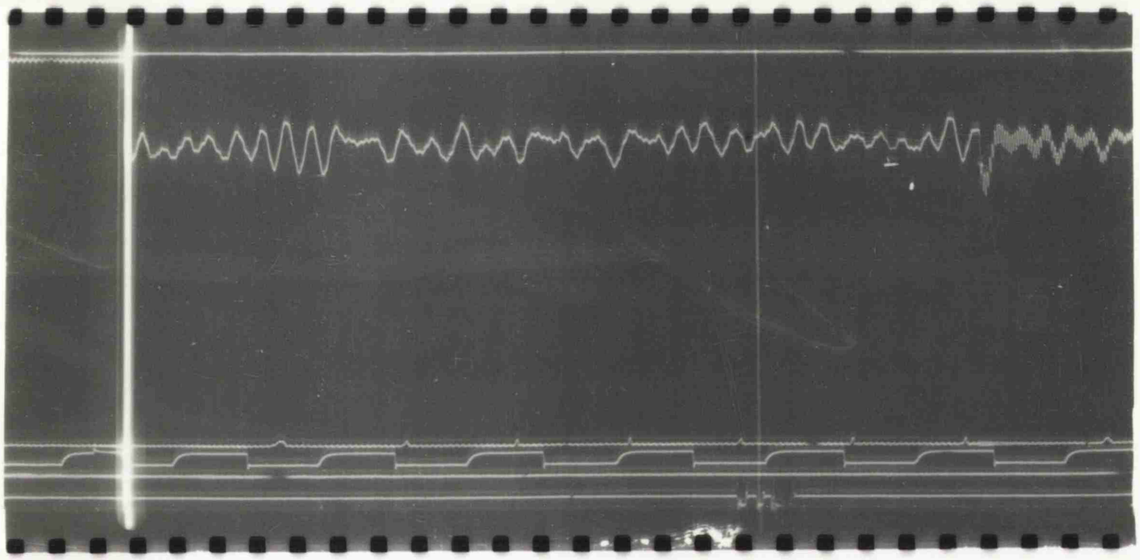
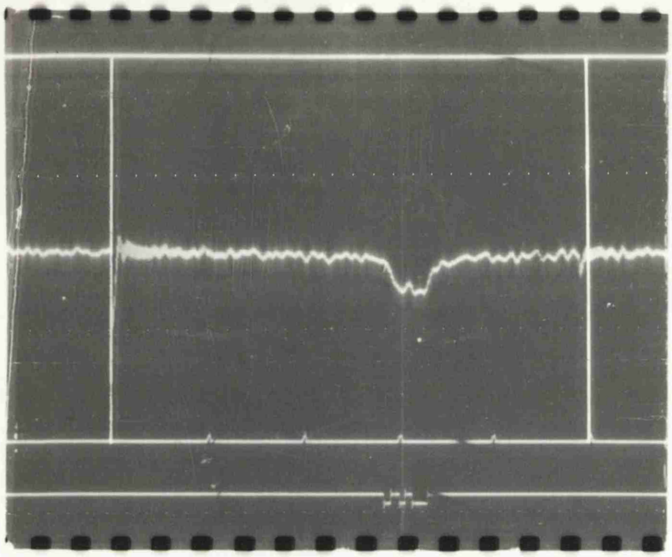
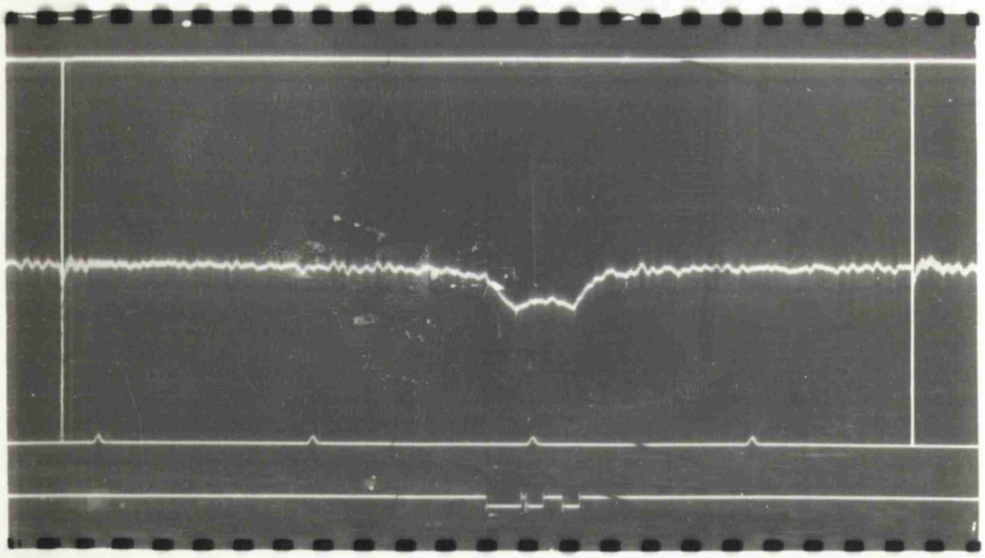


FIG. 9.3. HEAD ON 3/4 RADIUS PITOT TUBE FLOWMETER.



a) Cage Speed = 5.5 Shaft D/sec.



b) Cage Speed = 2.2 Shaft D/sec.

FIG. 9.4. PRESSURE DROP ACROSS TEST LENGTH Bb.

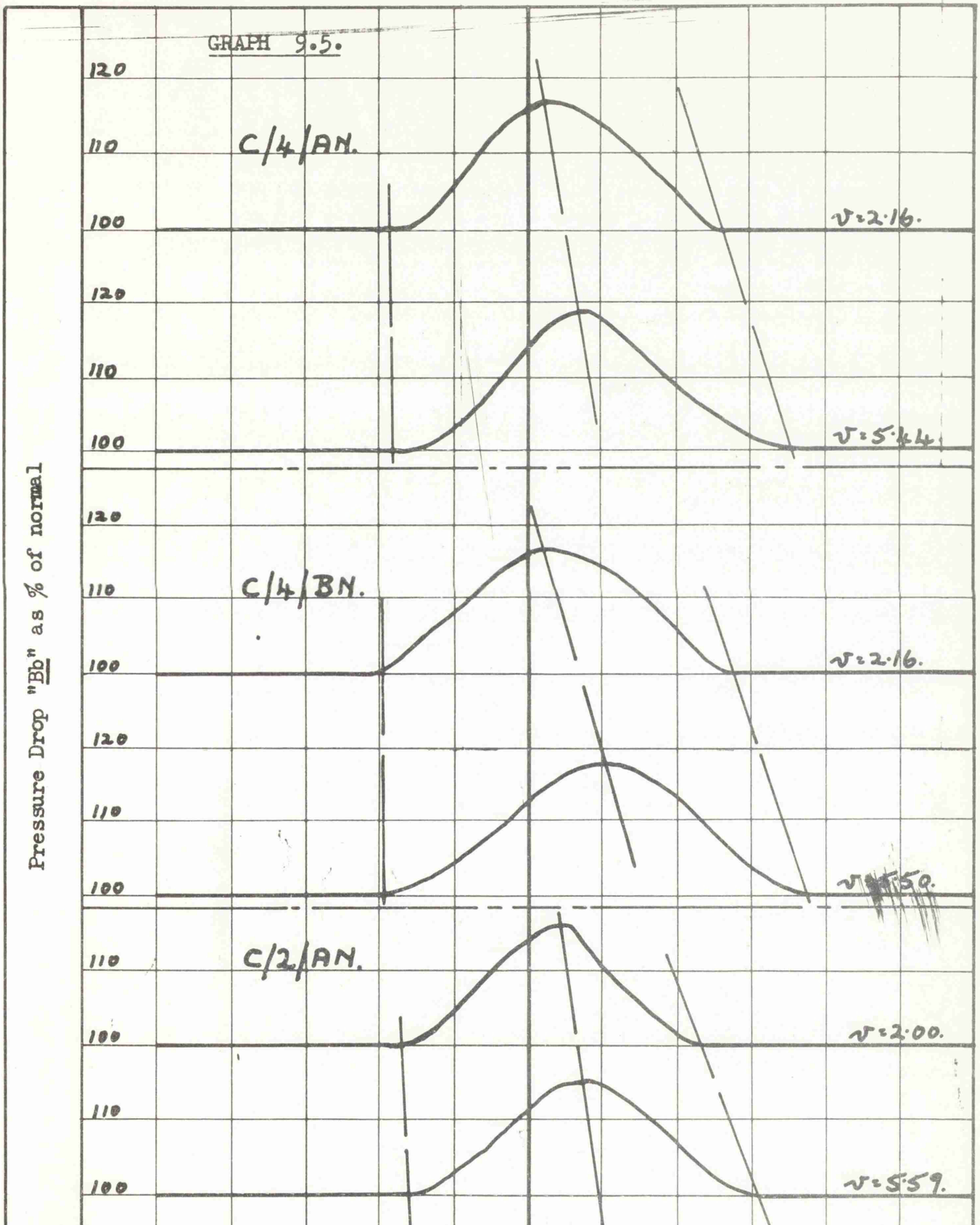
Note:- The cages are moving during the part of the record between the vertical lines.

where the cages start to move is due to electrical interference from the motor starter circuit. This suggests that any increase in pressure drop due to increased air velocity relative to one cage is balanced by the decrease in pressure drop at the other cage.

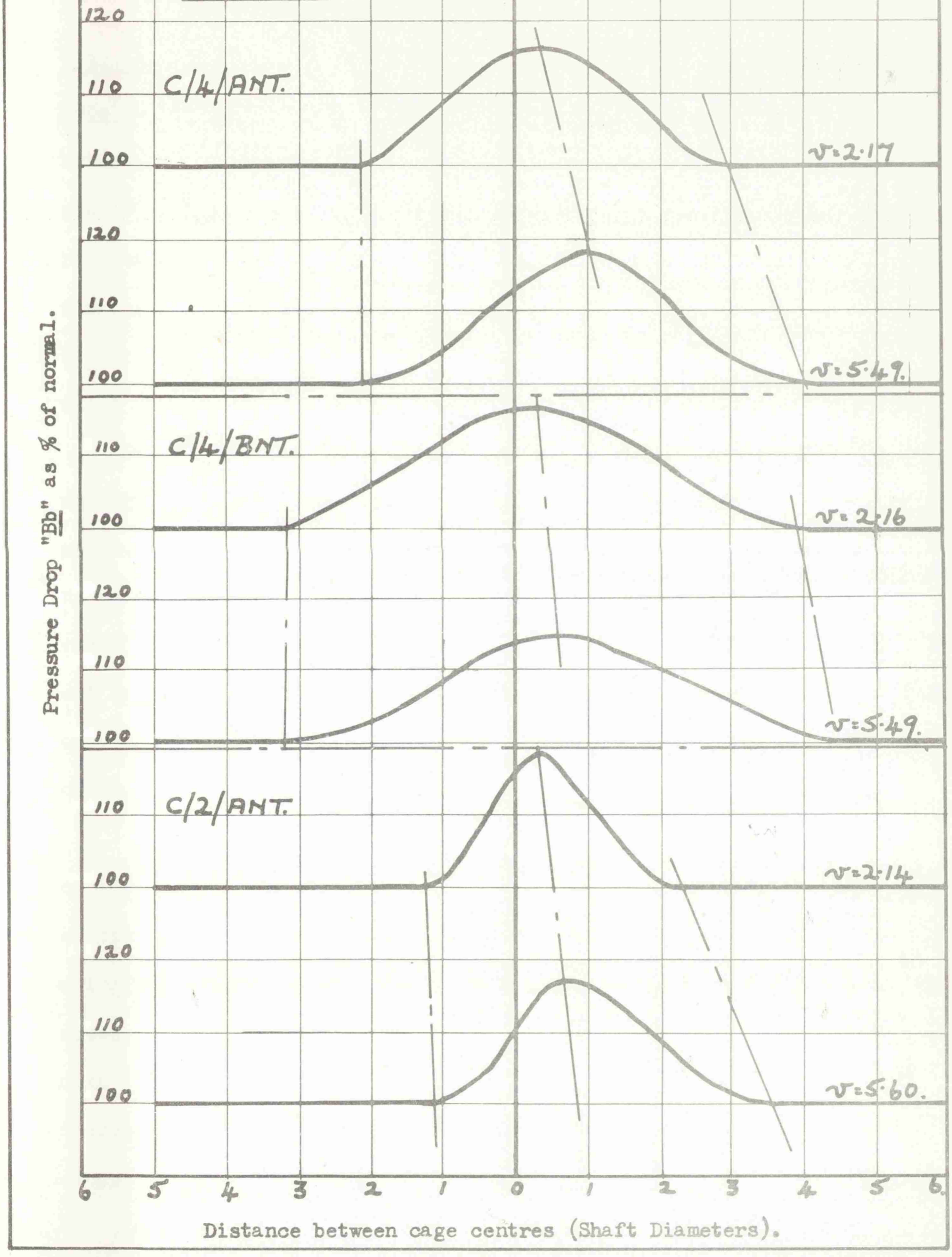
The effect of fairings on the resistance of moving cages is shown on Graphs 9.5 and 9.6. Only the highest and the lowest cage speeds are shown in full but it was possible to estimate the position of the peaks and ends of the intermediate curves and show the same trends as are seen in Graphs 9.2, 9.3 and 9.4.

The addition of fairings of type A to a cage lowers the maximum pressure drop and gives a smoother curve with an approximately constant rate of increase and decrease of pressure. There is very little change in the length of the zone of interaction with four-deck cages while with two-deck cages the zone is shortened even although the fairings greatly increase the length of the model.

The use of fairings of type B gives curves in which the rate of change of pressure is also approximately constant and is even less than in the curves using type A. This is done at the expense of an increase in the length of the zone of interaction. In this respect the results are similar to those obtained for the P.D.C.'s of stationary cages. Although the reductions achieved by the use of fairings do not appear to be very spectacular when plotted this way it must be remembered that the test length is equivalent to 692 feet of 24-foot diameter shaft and even a small percentage reduction represents a considerable saving.


GRAPH 9.5.
C/4/AN.
C/4/BN.
C/2/AN.
120
110
100
v=2·16.
v=5·44
v=2·16.
v=2·00.
v=5·59.
Pressure Drop "Bb" as % of normal
6 5 4 3 2 1 0 1 2 3 4 5 6
Distance between cage centres (Shaft Diameters).

GRAPH 9.6.



Pressure Drop "Bb" as % of normal.
120
110
100
C/4/ANT.
v=2.17
v=5.49.
C/4/BNT.
v=2.16
v=5.49.
C/2/ANT.
v=2.14
v=5.60.
6 5 4 3 2 1 0 1 2 3 4 5 6
Distance between cage centres (Shaft Diameters).

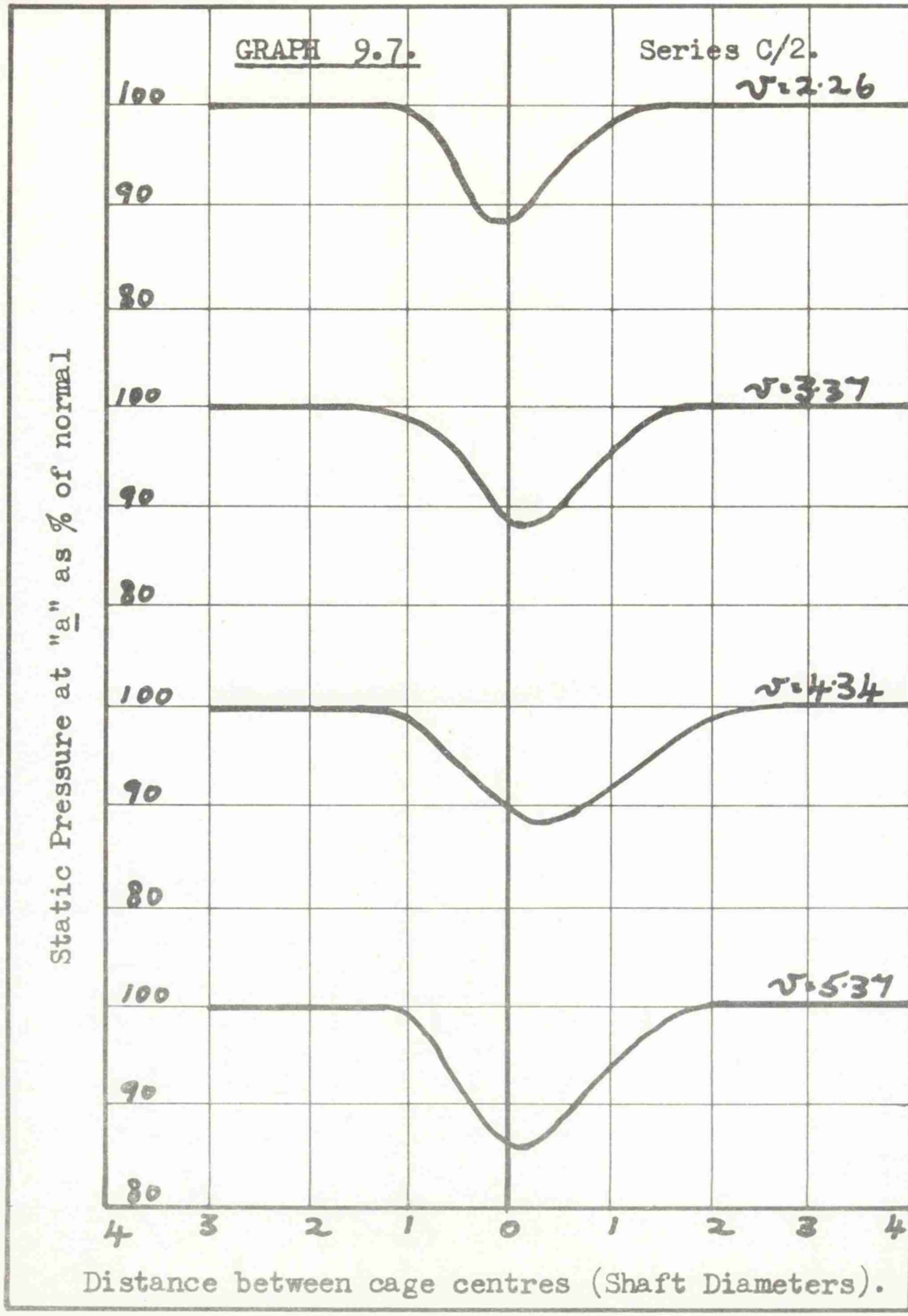
9.3. Effect of Passing Cages on Conditions elsewhere in the Shaft.

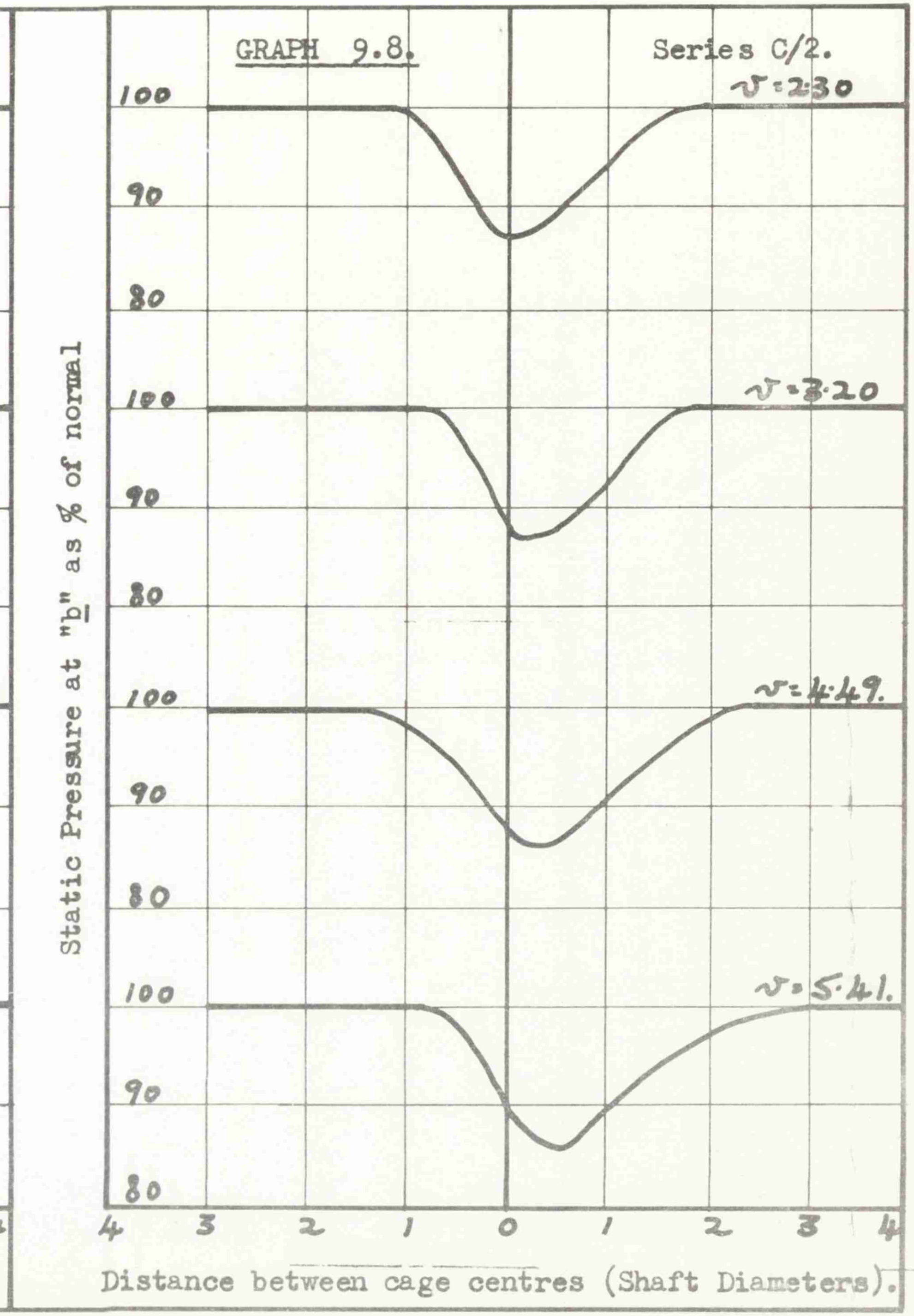
Note :- The pressures recorded in these tests represented the difference between the static pressure at certain points inside the duct and atmospheric pressure. Since the fan worked on the exhausting system the pressures plotted along the positive axis actually represent the depression at these points below atmospheric. The graphs can therefore be considered as records of the pressure loss along the duct from the inlet to the point concerned.

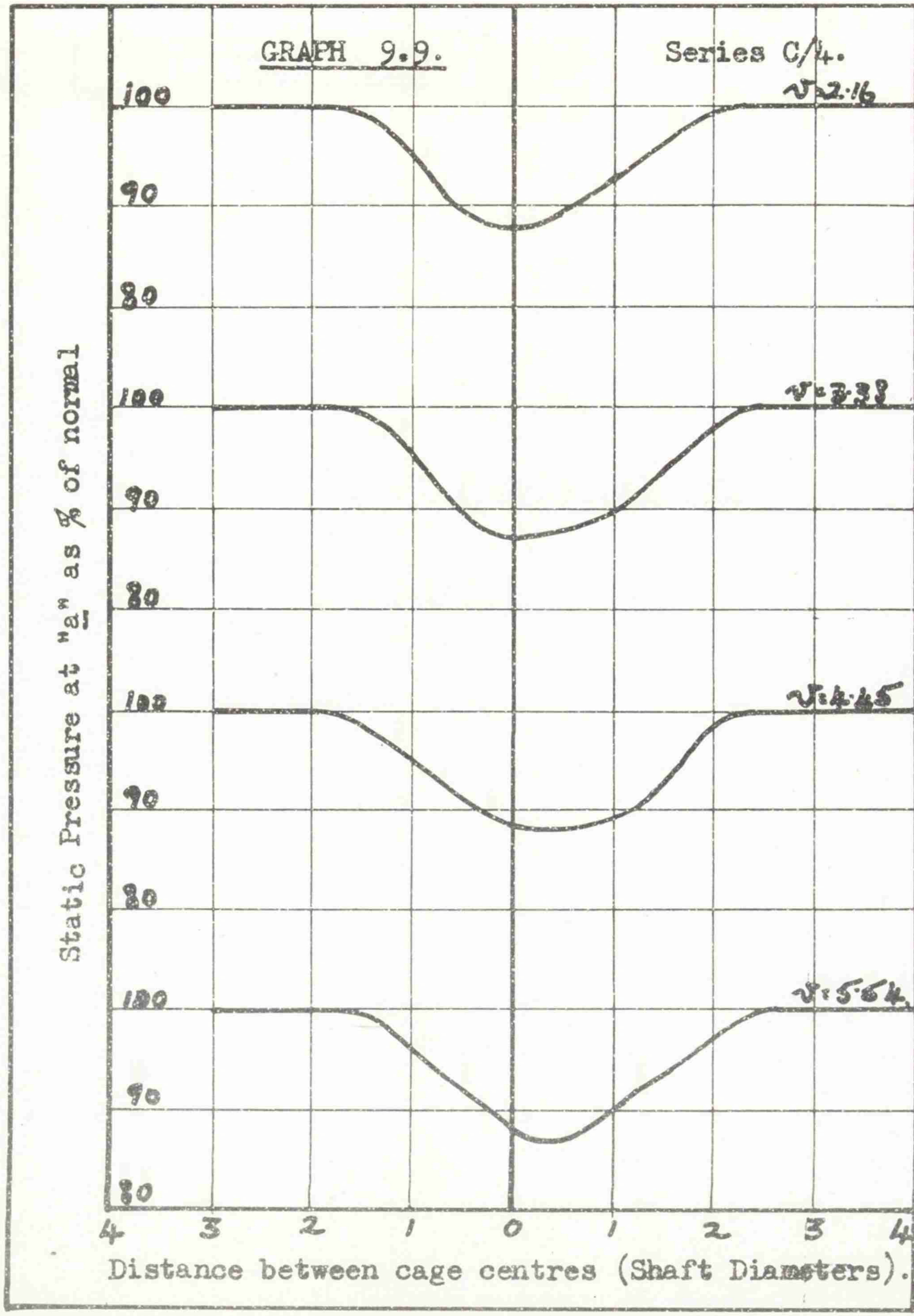
Graphs 9.7 to 9.11 show the variation in pressure at points "a" and "b" situated $7\frac{1}{2}$ and $13\frac{1}{2}$ feet respectively upstream from the passing place. The curves show that there is a reduction in the pressure at these points as the cages pass. The maximum value of the percentage reduction is approximately the same for two-deck and four-deck cages and seems to be slightly greater at point "b" than at point "a". The curves for two-deck cages appear more peaked due to the shorter duration of the overall effect and of the period of serious reduction.

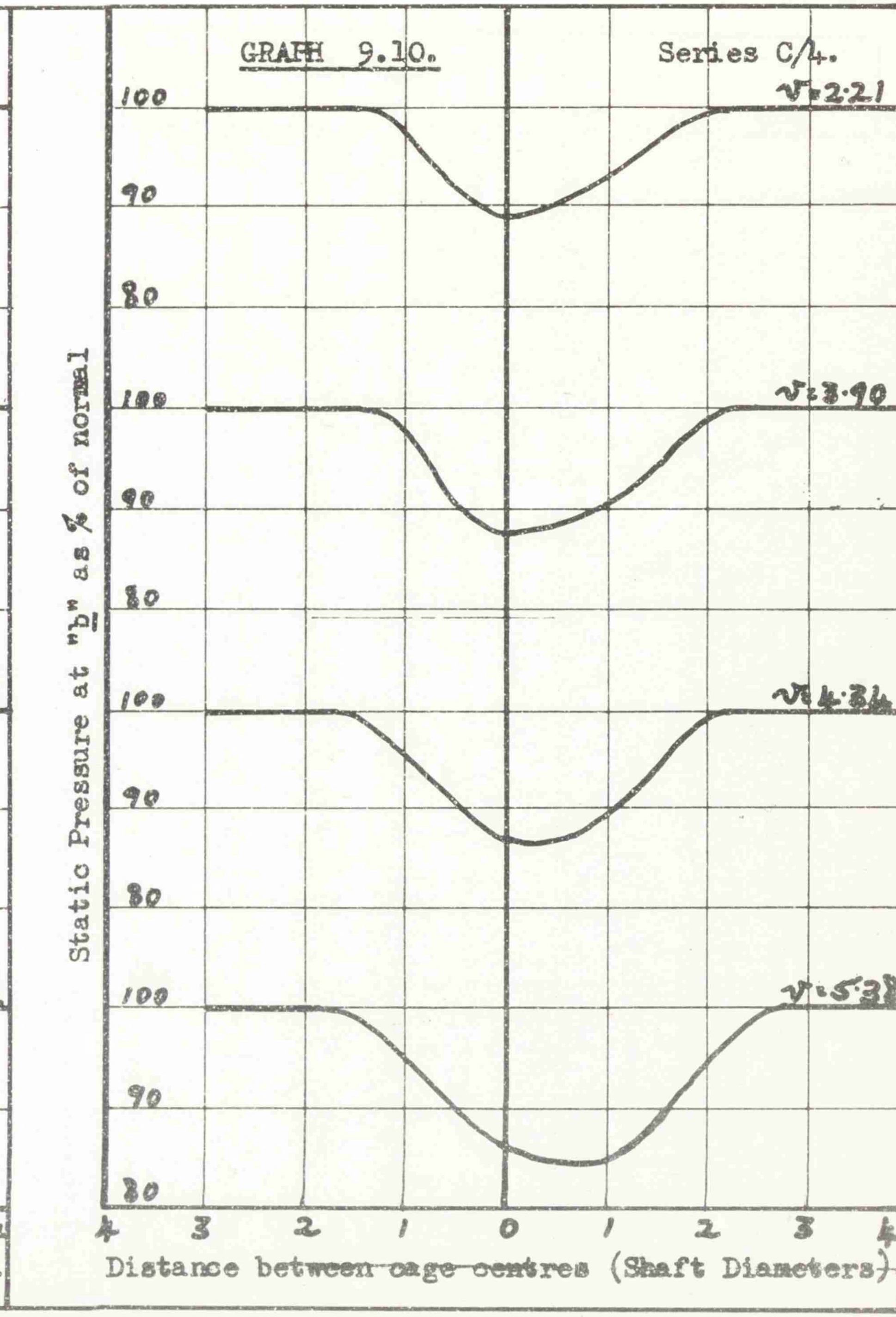
The effect of increasing speed is to slightly increase the extent of the effect giving smoother curves and also to increase the asymmetry of the curves about the centre line. Graph 9.11 shows the effect of using fairings on the variation in the pressure at point "a" at the top speed only. The fairings reduce the maximum change in pressure and make the changes in pressure more gradual. Fairings of type A do not increase the length of the disturbance appreciably but those of type B extend it a certain amount.

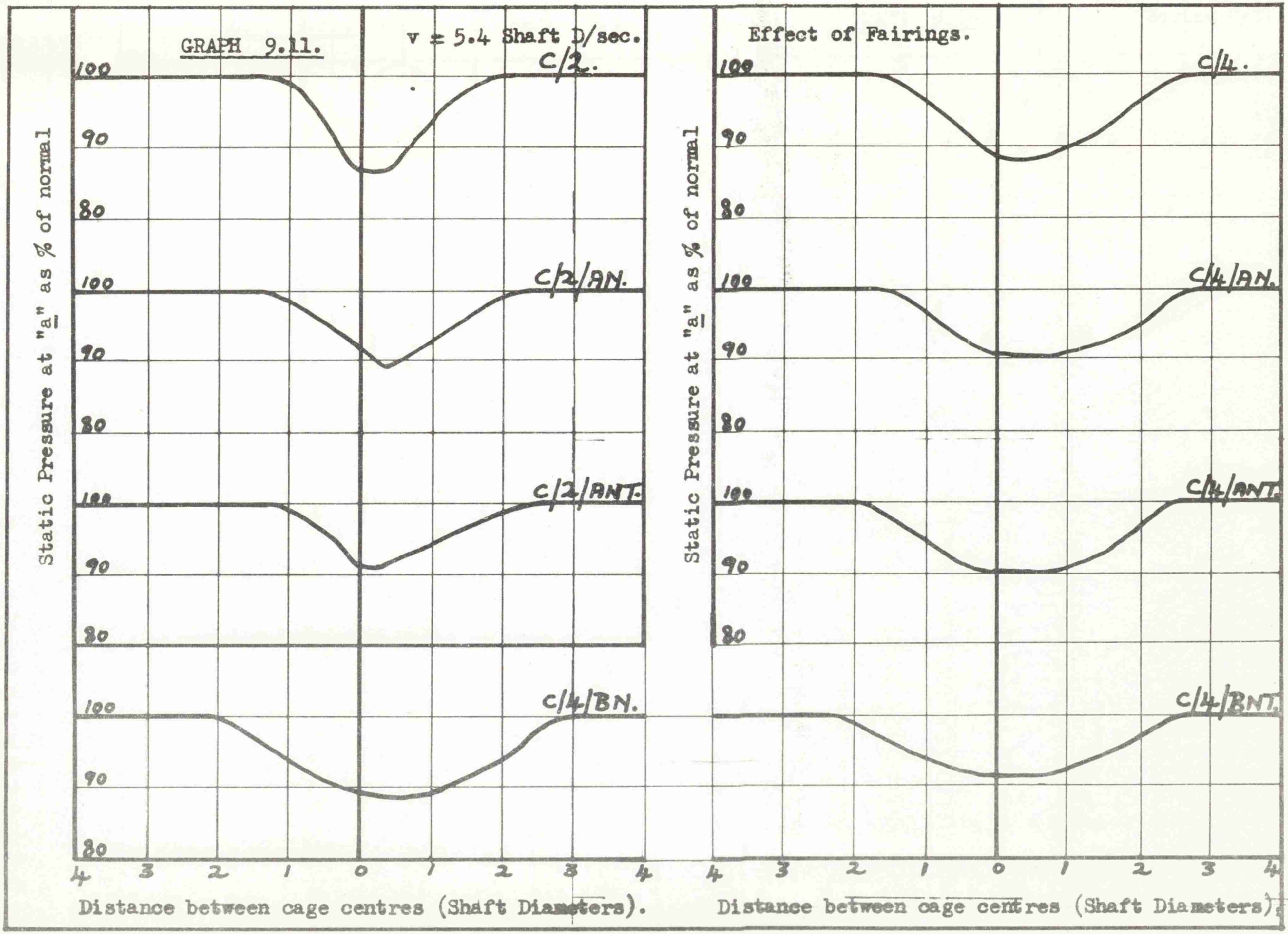
Graphs 9.12 to 9.17 show the effect of passing cages on the


GRAPH 9.7.
Series C/2.
v=2·26
v=3·37
v=4·34
v=5·37
100
90
80
4
3
2
1
0
Static Pressure at "a" as % of normal
Distance between cage centres (Shaft Diameters).

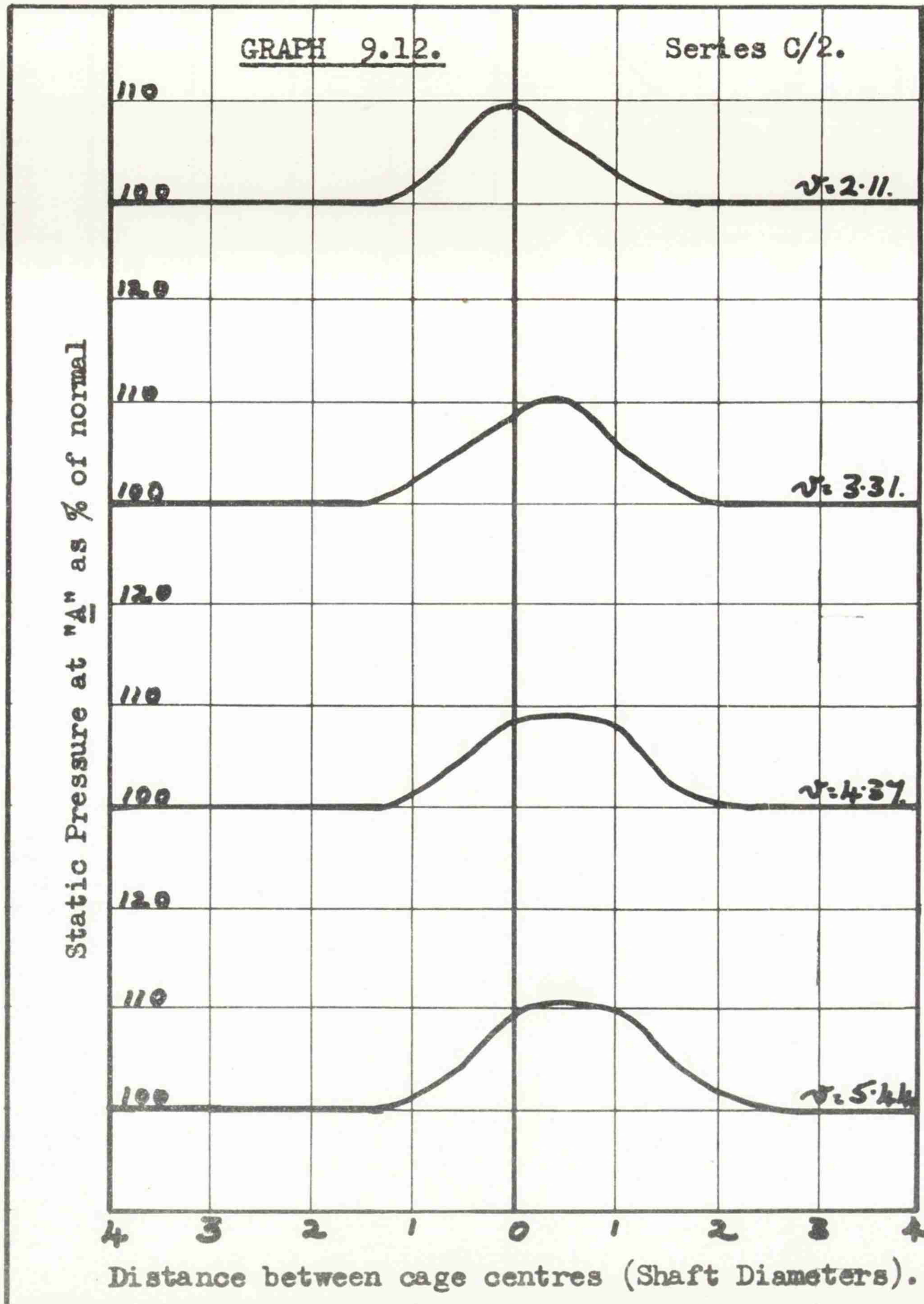

GRAPH 9.8.
Series C/2.
v=2·30
v=3·20
v=4·49.
v=5·41.
100
90
80
4
3
2
1
0
Static Pressure at "b" as % of normal
Distance between cage centres (Shaft Diameters).


GRAPH 9.9.
Series C/4.
v=2·16
v=3·38
v=4·45
v=5·64
100
90
80
Static Pressure at "a" as % of normal
4 3 2 1 0 1 2 3 4
Distance between cage centres (Shaft Diameters).

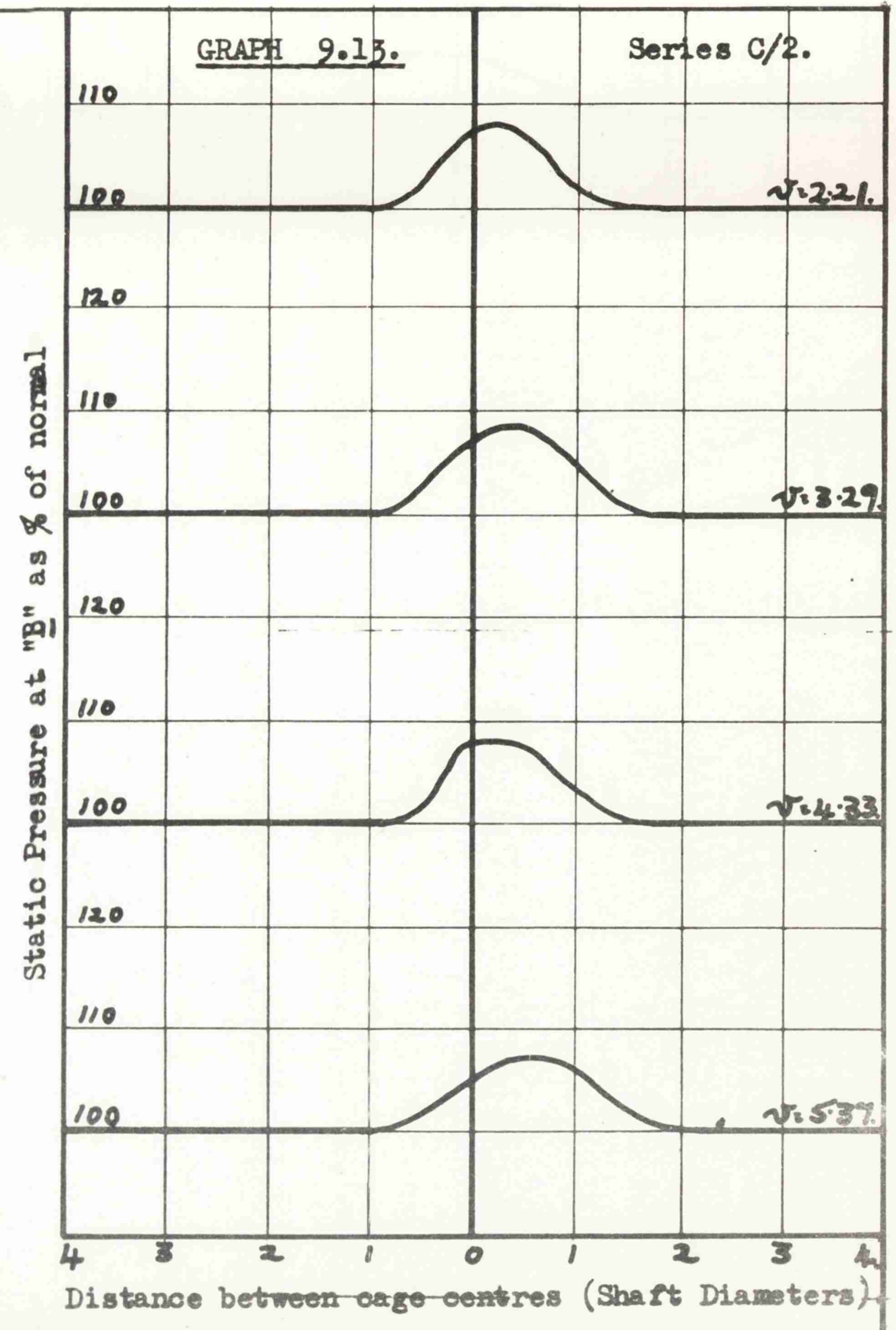

GRAPH 9.10.
Series C/4.
v=2·21
v=3·90
v=4·34
v=5·38
100
90
80
Static Pressure at "b" as % of normal
4 3 2 1 0 1 2 3 4
Distance between cage centres (Shaft Diameters)



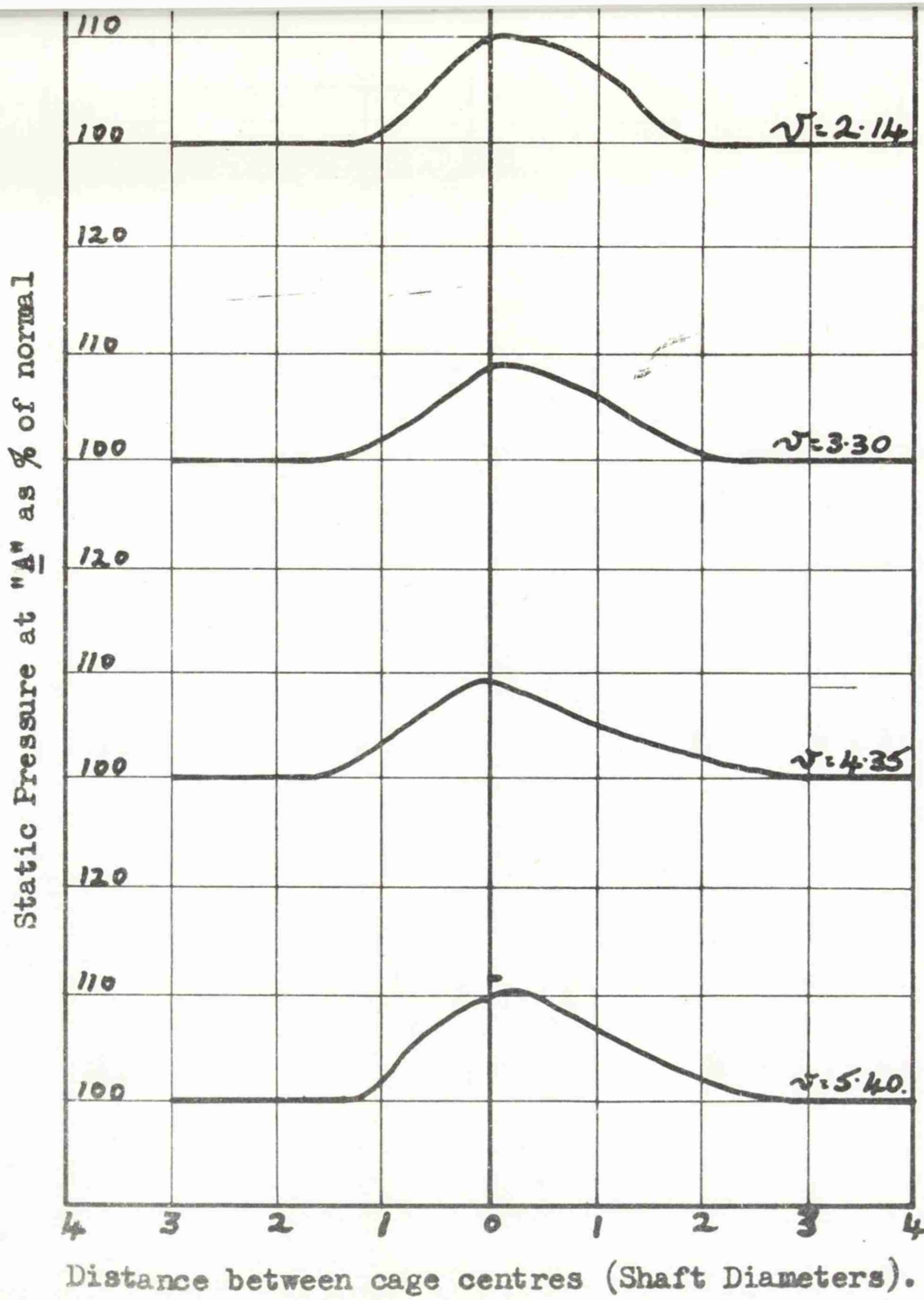
GRAPH 9.11.
v = 5.4 Shaft D/sec.
Effect of Fairings.
C/2.
C/2/AN.
C/2/ANT.
C/4/BN.
C/4.
C/4/AN.
C/4/ANT.
C/4/BNT.
100
90
80
4 3 2 1 0 1 2 3 4
Static Pressure at "a" as % of normal
Distance between cage centres (Shaft Diameters).

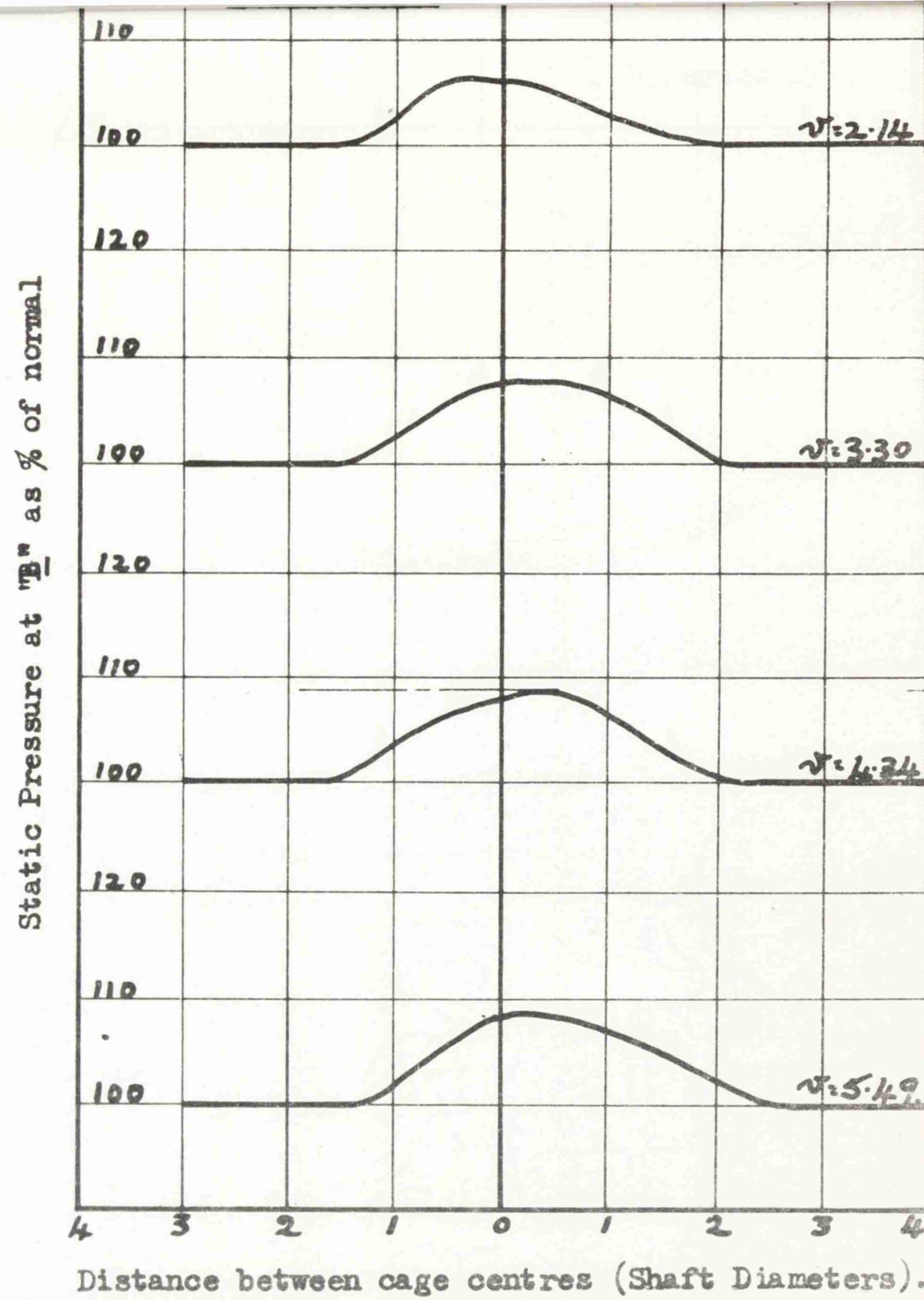


GRAPH 9.12.
Series C/2.
Static Pressure at "A" as % of normal
110
100
120
110
100
120
110
100
120
110
100
v=2·11.
v=3·31.
v=4·37
v=5·44
4 3 2 1 0 1 2 3 4.
Distance between cage centres (Shaft Diameters).

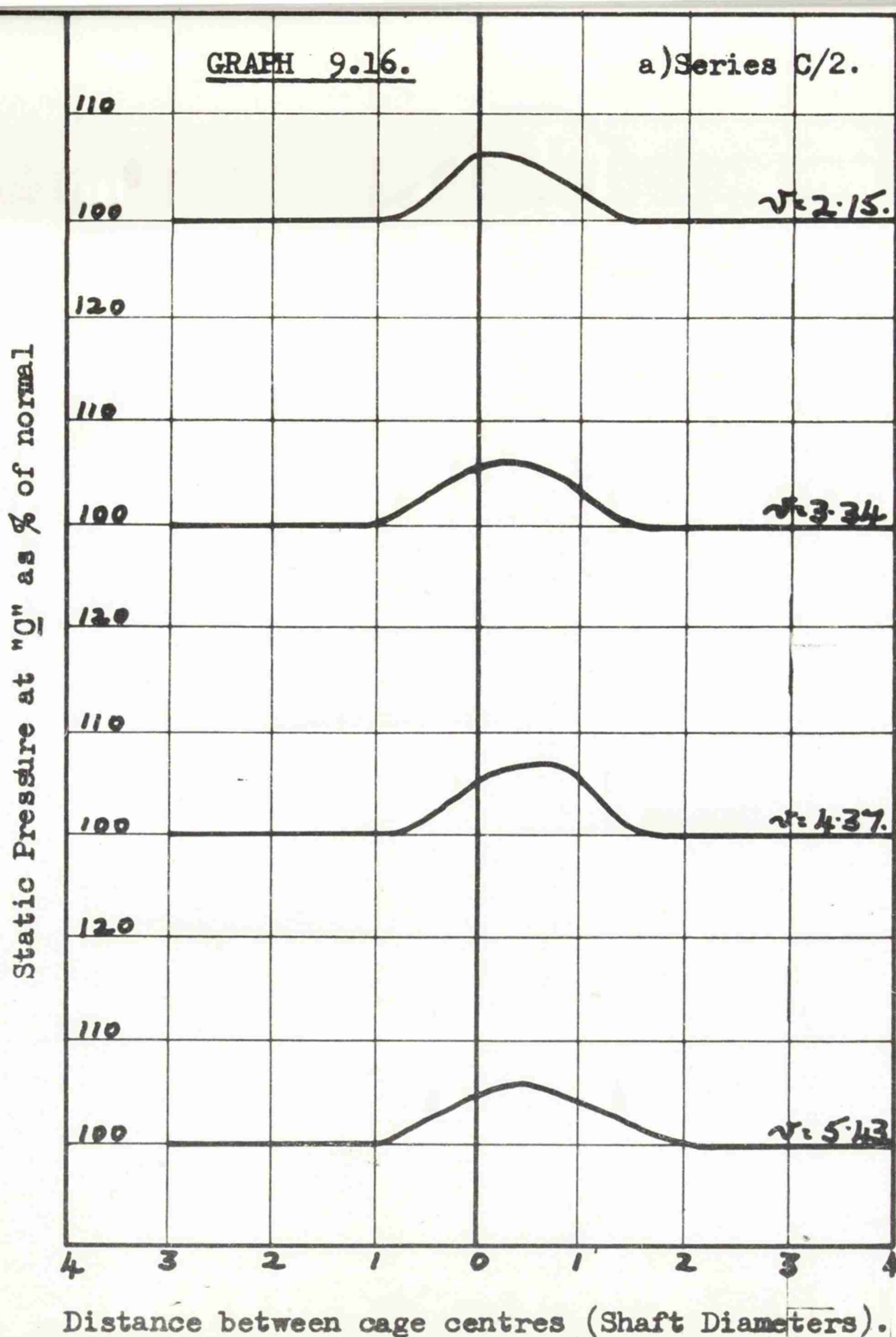


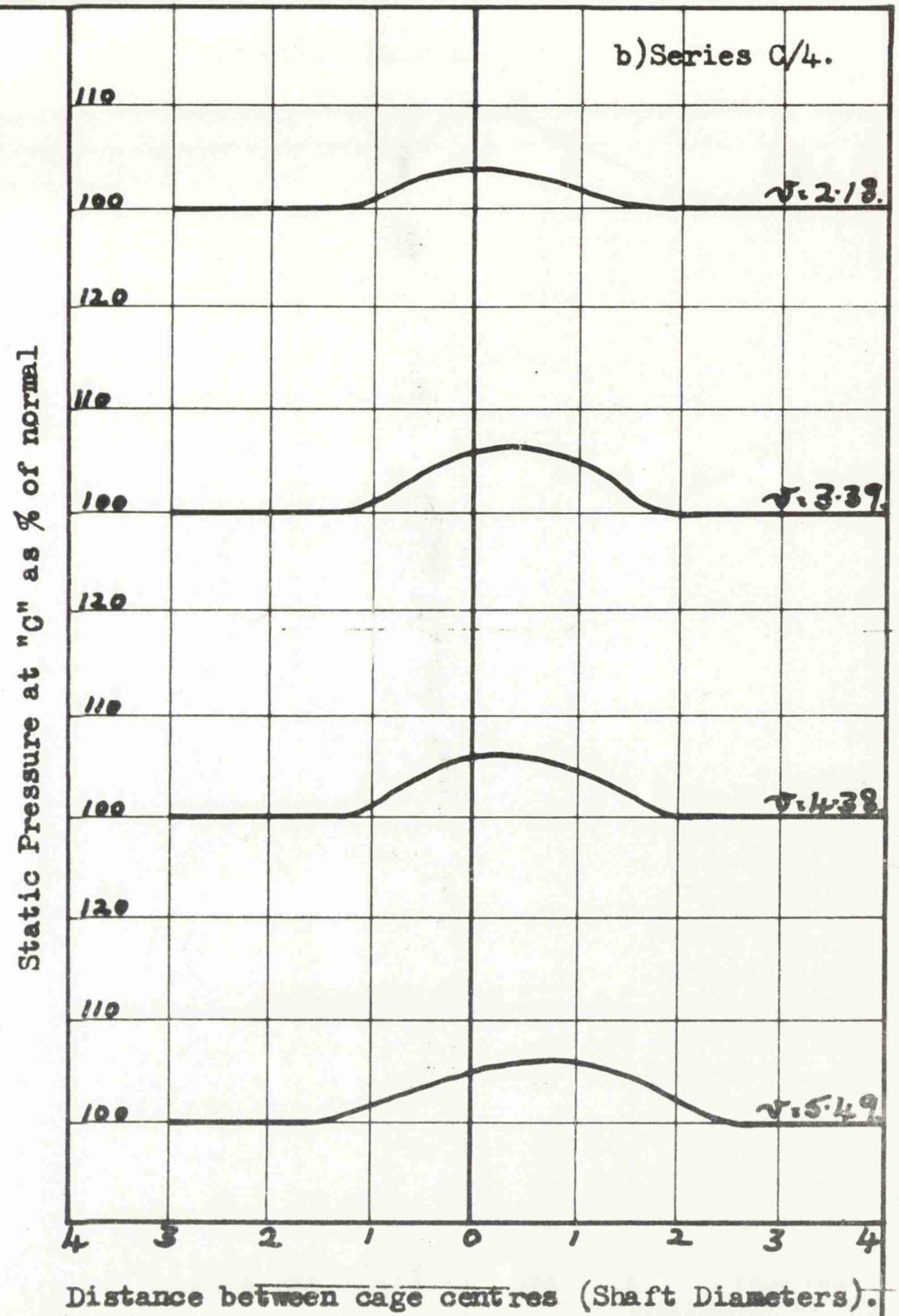
GRAPH 9.13.
Series C/2.
Static Pressure at "B" as % of normal
110
100
120
110
100
120
110
100
120
110
100
v=2·21.
v=3·29.
v=4·33
v=5·37.
4 3 2 1 0 1 2 3 4
Distance between cage centres (Shaft Diameters).

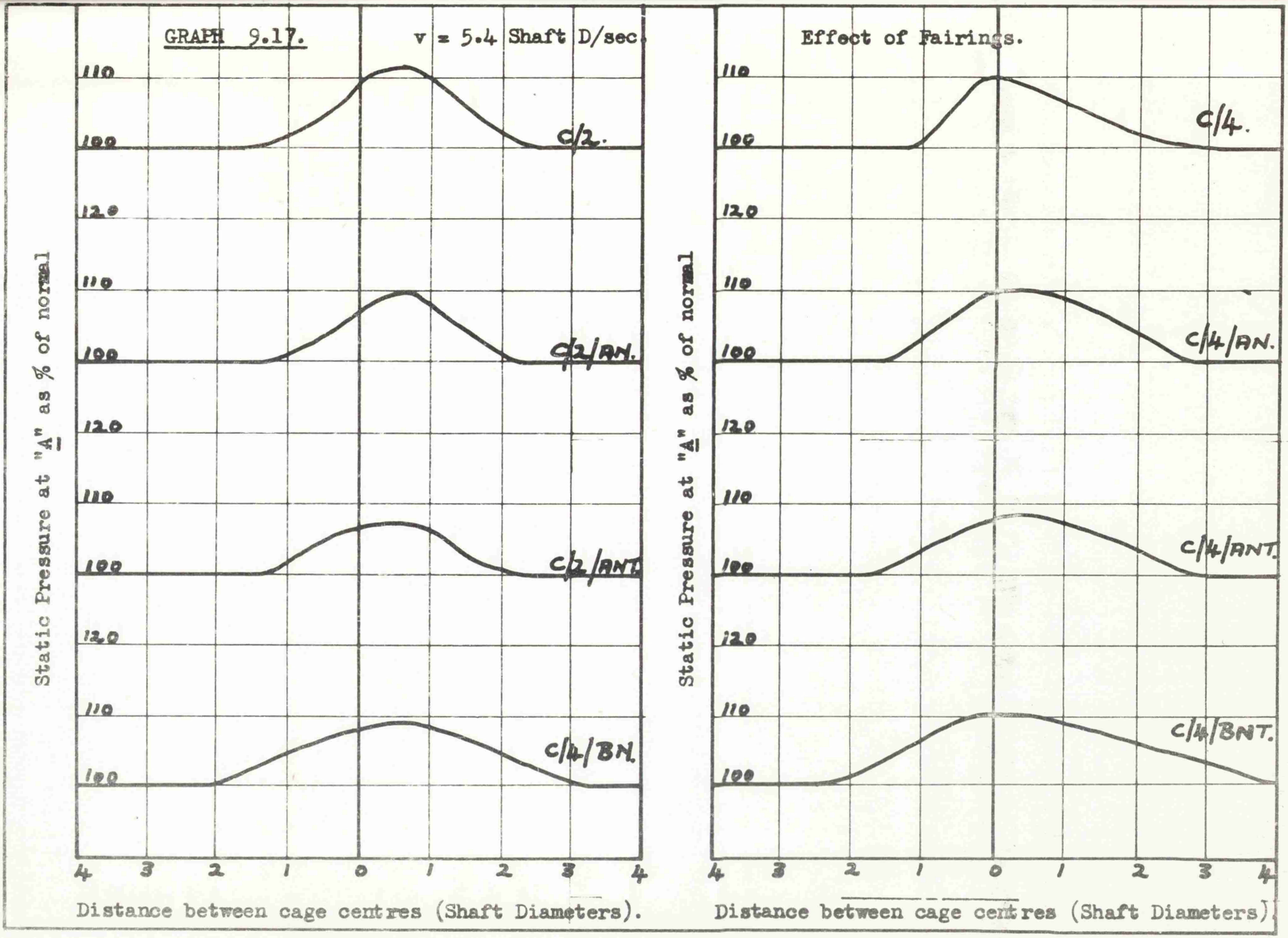

Static Pressure at "A" as % of normal
110
100
120
110
100
120
110
100
120
110
100
ν=2·14
ν=3·30
ν=4·35
ν=5·40
4 3 2 1 0 1 2 3 4
Distance between cage centres (Shaft Diameters).


Static Pressure at "B" as % of normal
110
100
120
110
100
120
110
100
120
110
100
ν=2·14
ν=3·30
ν=4·24
ν=5·49
4 3 2 1 0 1 2 3 4
Distance between cage centres (Shaft Diameters).

GRAPH 9.16.


a) Series C/2.
Static Pressure at "G" as % of normal
110
100
120
110
100
120
110
100
120
110
100
v: 2·15.
v: 3·34
v: 4·37.
v: 5·43
4 3 2 1 0 1 2 3 4
Distance between cage centres (Shaft Diameters).


b) Series C/4.
Static Pressure at "G" as % of normal
110
100
120
110
100
120
110
100
120
110
100
v: 2·18.
v: 3·39.
v: 4·38.
v: 5·49.
4 3 2 1 0 1 2 3 4
Distance between cage centres (Shaft Diameters).



GRAPH 9.17.
v = 5.4 Shaft D/sec.
Effect of Fairings.
Static Pressure at "A" as % of normal
110
100
120
C/2.
C/2/AN.
C/2/ANT.
C/4/BN.
C/4.
C/4/AN.
C/4/ANT.
C/4/BNT.
4 3 2 1 0 1 2 3 4
Distance between cage centres (Shaft Diameters).

pressure at points "A", "B" and "C" at $7\frac{1}{2}$, $13\frac{1}{2}$ and $23\frac{1}{2}$ feet respectively downstream from the passing place. The features are very similar to the previous set except that the passing of cages produces a rise in pressure at these points. The maximum percentage increase reached falls progressively from "A" through "B" to "C".

Records were also taken of the pressures at these points as the smaller cages of series A/2 passed. No significant changes in pressure could be noticed and it seems safe to assume that the movement of such small cages will not produce any drastic effects on the shaft conditions.

The effects just described are considered in the following theoretical analysis of the pressure changes produced by the variation in air quantity flowing in the duct. None of the records taken showed any signs of reflected waves travelling at sonic speeds and it is assumed that there was no compression and hence conditions of continuity prevailed.

Consider any point P on the model shaft distance "x" from the inlet.

At any time :-

Static pressure at P $= \dfrac{p}{w}$

Air velocity $= V$

Air acceleration $= \dfrac{dV}{dt}.$

Bernoulli's Equation between P and a point Q distance "δx" downstream from P gives

$$\frac{p}{w} + \frac{V^2}{2g} = \frac{p}{w} + \frac{V^2}{2g} + \frac{d}{dx}\left(\frac{p}{w} + \frac{V^2}{2g}\right)\delta x + \text{Work done by the air.} \qquad (1)$$

The work done by the air can be considered in two parts :-

a) Work done against friction $= \frac{4fV^2}{2gd} \cdot \delta x$

where f = friction factor

b) Work done accelerating the air

$$= \text{increase in kinetic energy}$$

$$= \frac{(V + \delta V)^2 - V^2}{2g}$$

$$= \frac{1}{g} \; V\delta V \qquad \text{neglecting } (\delta V)^2$$

$$= \frac{1}{g} \; \frac{V\delta V}{\delta t} \; \delta t \qquad \text{and } V.\delta t = \delta x$$

$$\therefore \text{ Work Done} = \frac{1}{g} \; \frac{dV}{dt} \; \delta x.$$

$\therefore$ Substituting in (1) we get

$$\frac{d}{dx}\left(\frac{p}{w} + \frac{V^2}{2g}\right)\delta x = -\frac{4fV^2}{2gd}\delta x - \frac{1}{g} \; \frac{dV}{dt} \; \delta x$$

$$\therefore \frac{p}{w} + \frac{V^2}{2g} = -\frac{4fV^2}{2gd} \int_0^x \delta x - \frac{1}{g} \; \frac{dV}{dt} \int_0^x \delta x + C$$

Since the pressures were recorded as the difference from atmospheric we assume

$$\frac{p}{w} = 0 \quad \text{at } x = 0$$

also $\int_0^x \delta x = 0$ at $x = 0$

$$\therefore \quad C = \frac{V^2}{2g}$$

$$\therefore \quad \frac{p}{w} = -\frac{4fV^2}{2gd}\,x - \frac{1}{g}\,\frac{dV}{dt}\,.\,x \qquad (2)$$

i.e. the pressure at any point in the model at any time is the algebraic sum of the friction loss between the inlet and that point (as calculated from the instantaneous velocity) and the acceleration head of the column of air between that point and the inlet.

At points in the duct upstream from the passing place the pressure is directly proportional to "x" and the percentage pressure change as plotted in the graphs should be the same for all such points. In a normal shaft this will be the case but in the model the lining does not extend to the inlet and the theoretical expression for "p" is of the form :-

$$\frac{p}{w} = -A\,y\,V^2 - B\,(x-y)V^2 - \frac{1}{g}\,\frac{dV}{dt}\,x$$

where y = distance from the inlet to the end of the lining.

and A, B = constants.

In the model, therefore, the ratio of the two pressures produced at a point by two sets of conditions is not independent of "x".

At points in the duct downstream from the passing place the term expressing the friction loss in (2) cannot be directly applied since the passing of the cages increases the resistance to airflow between the inlet and the point. If we assume for the moment that no reduction in air flow takes place then the effect of passing cages will be to increase the pressure at all points downstream by a constant amount. Since the pressure increases progressively downstream the percentage change in pressure at a point is reduced as we move further downstream. The reduction in quantity flowing will produce an effect similar to that at points upstream but the reduction it causes is not enough to compensate for the increased friction loss at the cages and hence the net effect is an increase in pressure with the proportional change becoming smaller as we move downstream.

9.4. Effect of a Cage passing a point on conditions at that point.

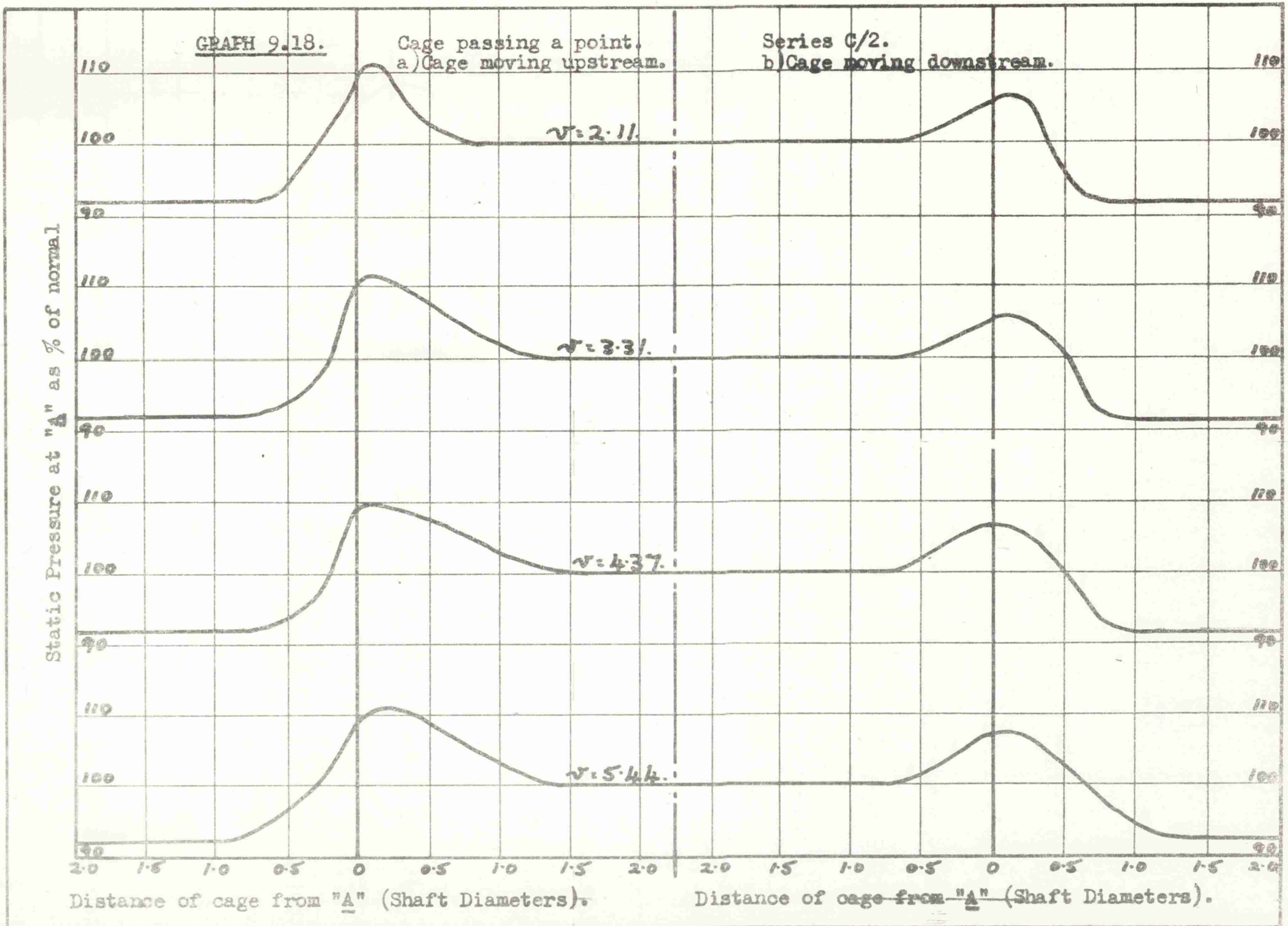
Graphs 9.18 to 9.21 show the variation in pressure at a point in the shaft as a cage passes that point. The pressures are plotted as percentages of the steady pressure when the cage is upstream from the point, and the difference in level between the ends represents the increased friction loss upstream due to the cage.

These graphs show that the pressure at the point increases to a value above its higher normal as the cage passes. Because of the method of measuring and plotting used this represents an increase in the depression below atmospheric and hence represents a lower absolute pressure in the duct corresponding to a higher velocity head due to the reduced area available for flow. The pressure waves formed by this forced increase in velocity will travel up and down the shaft along with the cages. The relative amount by which the passing of a cage will disturb the steady pressure at a point will depend on the ratio of the cage drop to the normal pressure at that point. The curves as shown for point "A" therefore are only typical and not general. They do,

GRAPH 9.18.

Cage passing a point.

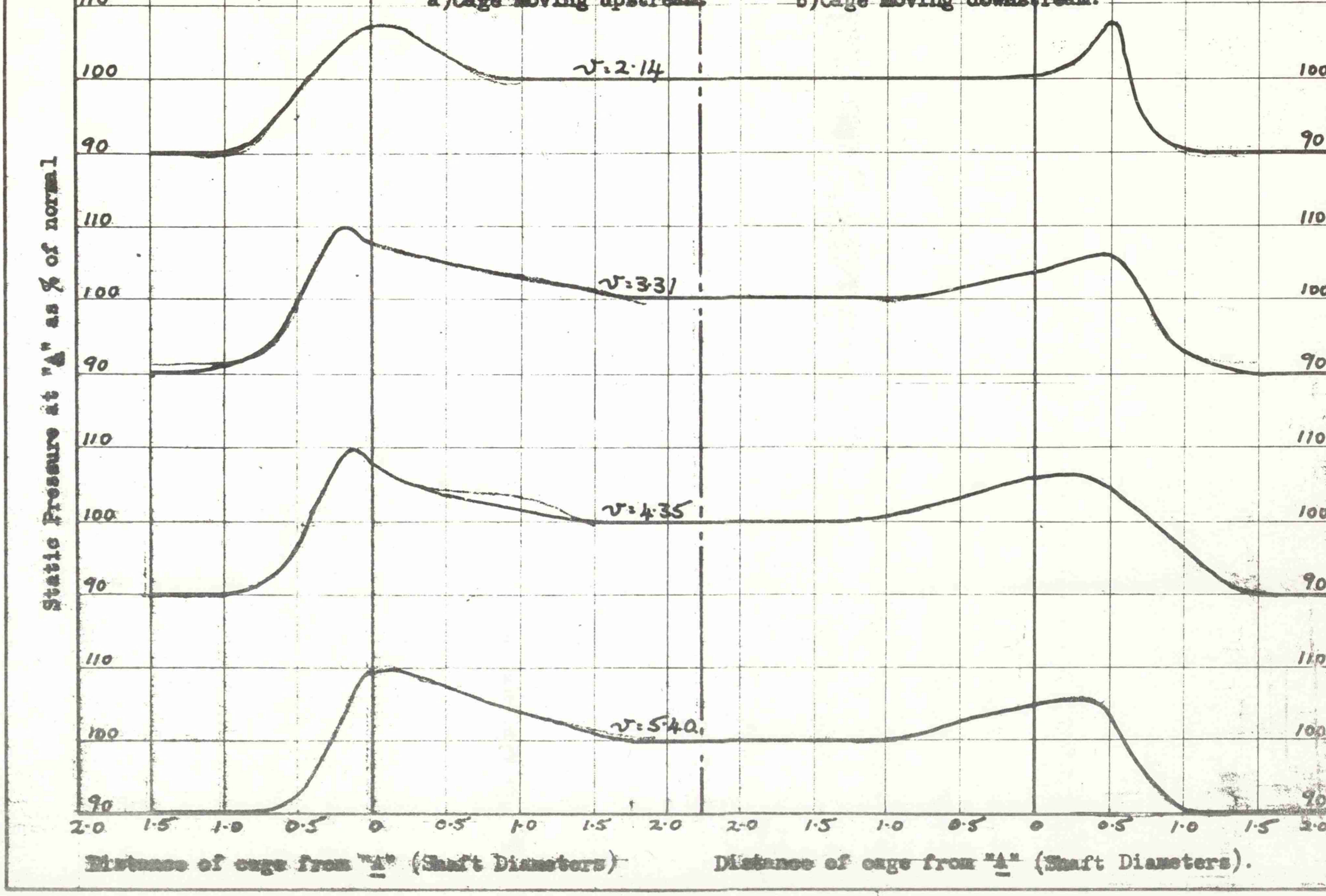
Series C/2.

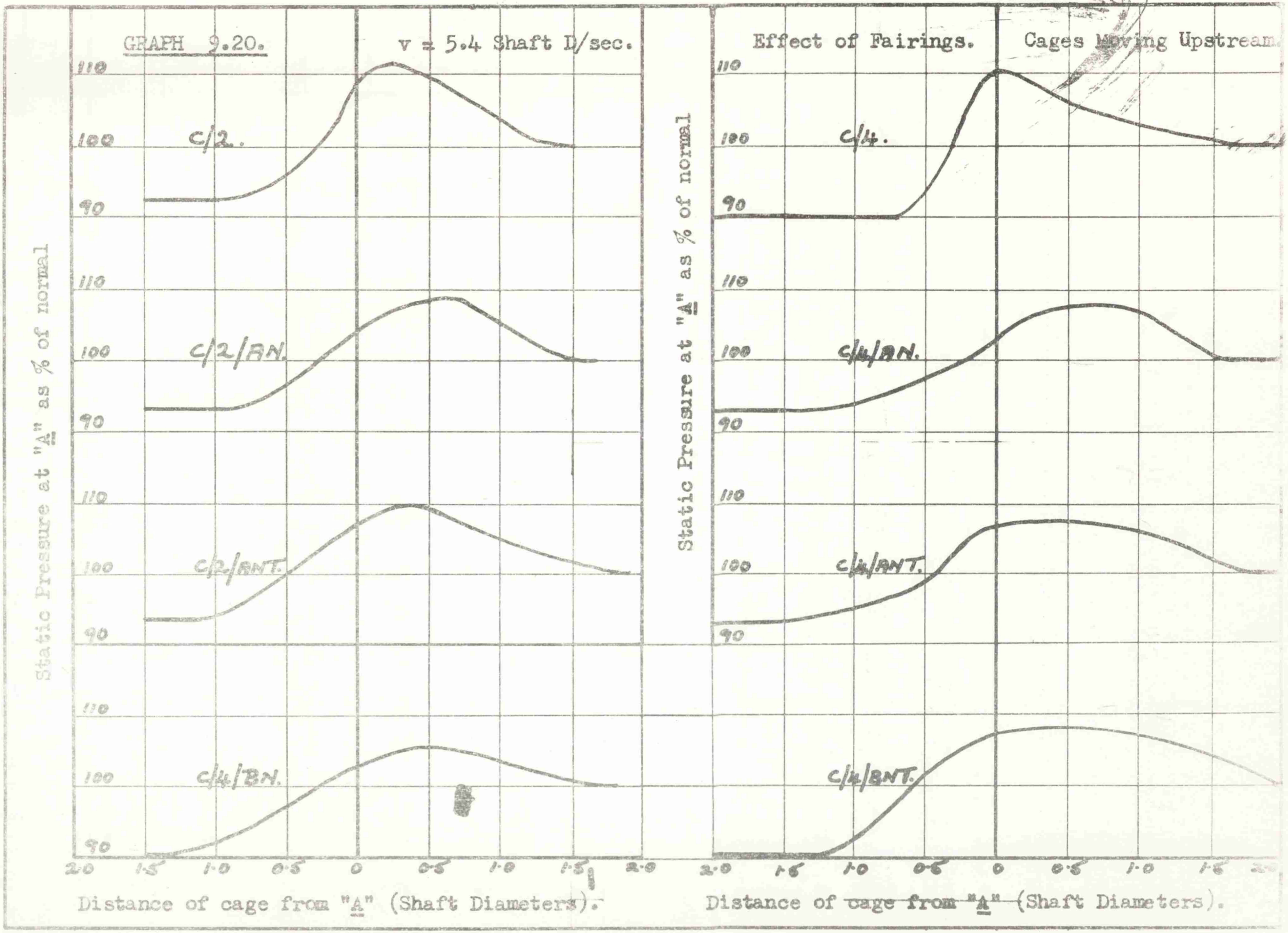


GRAPH 9.19.

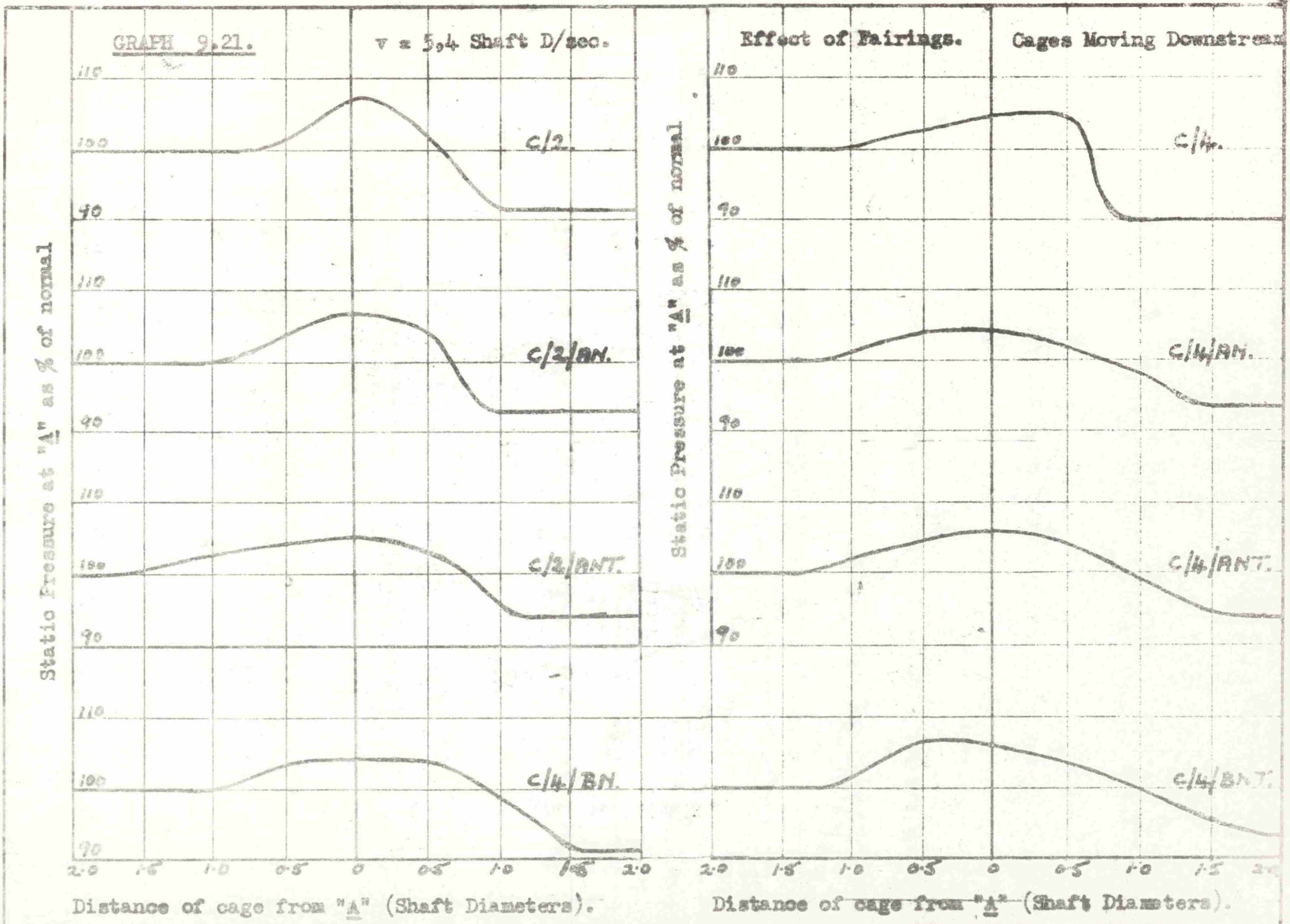
Cage passing a point.
a) Cage moving upstream.

Series C/4.
b) Cage moving downstream.




GRAPH 9.20.
v = 5.4 Shaft D/sec.
Effect of Fairings.
Cages Moving Upstream.
C/2.
C/2/AN.
C/2/ANT.
C/4/BN.
C/4.
C/4/AN.
C/4/ANT.
C/4/BNT.
110
100
90
2.0
1.5
1.0
0.5
0
Static Pressure at "A" as % of normal
Distance of cage from "A" (Shaft Diameters).
Distance of ~~cage from "A"~~ (Shaft Diameters).

GRAPH 9.21.


v = 5,4 Shaft D/sec.
Effect of Fairings.
Cages Moving Downstream
Static Pressure at "A" as % of normal
110
100
90
C/2.
C/2/BN.
C/2/BNT.
C/4/BN.
C/4.
C/4/BN.
C/4/BNT.
C/4/BNT.
2·0 1·5 1·0 0·5 0 0·5 1·0 1·5 2·0
Distance of cage from "A" (Shaft Diameters).

however, give a good indication of the effects.

One outstanding feature is how quickly steady conditions return after the passing of a cage showing that the lower air velocity is quickly restored as the air re-expands into the full area of the shaft. Increasing cage speed has very little effect on the pattern of the graphs apart from a slight smoothing of the curve, although the effect occupies a much shorter time at higher speeds and the rates of change of pressure are higher. This is probably because the air accelerations are only local and no great inertia exists. A cage moving against the airflow creates a slightly greater disturbance in the pressure at a point than a cage moving with the airflow.

Graphs 9.20 and 9.21 show the smoothing of the disturbance achieved by adding fairings to the cages. The curves shown are those for the highest cage speed used. The fairings of type A appear to be more successful since they reduce the cage drop making the permanent change in the pressure after a cage has passed smaller. Both types extend the duration of the abnormal pressure but allow the difference in pressure level to be established by less sudden changes.

9.5. General Conclusions.

The results give an indication of the features which moving cages can be expected to produce in a full scale shaft.

The passing of two cages increases the resistance of the shaft and results in a temporary decrease in the quantity flowing. This decrease itself is probably the most serious effect produced and it will have an effect on the variations in pressure which take place. The amount of the decrease will depend on

a) the shape of the fan characteristic,

b) the proportional increase in the total circuit resistance,

c) the inertia of the air in the circuit.

Factors b) and c) will probably cause the reduction in quantity to be less in a full scale shaft than in the model.

The curves obtained by plotting the fluctuating pressures in the model as percentages of their normal values should be applicable to full scale shafts if the ratios of cage resistance to shaft resistance are comparable. In deep heavily lined shafts the cages will have proportionately smaller effects. In the case of the records of varying pressure as a cage passes a point the increase in air velocity due to the cage and the corresponding change in static pressure will be the same at all points in the shaft, but obviously the proportional change will vary from point to point. Exactly how it will vary will depend on the pattern of the static pressure along the length of the shaft. This is governed by whether the shaft is an upcast or a downcast, whether the main fan is forcing or exhausting and by the natural increase in atmospheric pressure with depth. These factors will also affect the proportional change in pressure produced at various points in the shaft.

The tests on the use of fairings proved that cage resistances can be substantially reduced. The efficiencies obtained were sometimes less than the values given by Stevenson. The main reason for this is differences in shape and size of the cages used. The important factor seems to be the relative sizes of the cage and the shaft. A wider cage requires a longer fairing before the same aspect ratio is reached and increased friction losses in the space between the fairing and the shaft walls are probably responsible for the lower efficiency. The tests on moving cages showed that fairings are also useful for making the pressure changes in the shaft caused by the cages less violent. Type B fairings seem more effective than type A but are unlikely to be used because of their higher resistance.

In his thesis Stevenson mentioned the possibility of the use of fairings (particularly of type B) producing side thrusts on the cages due to aerodynamic forces. The system of guides and cage shoes used in this work was very successful but some trouble was encountered in series C/4/BNT due to the cage shoes jumping out of the guides. This may simply be due to the great overall length of the model but unbalanced air forces may have had some effect.

APPENDIX

Theory of Vibration of a Circular Plate clamped at the Edges

In connection with the work of Section 5 (pp.27 and 28) a formula suitable for estimating the natural frequency of vibration of a flat circular plate clamped at its edges was required. In a study of some of the relevant literature it was noticed that different authors gave formulae which were irreconcilable and some doubt existed as to which one ought to be used. Details of the formulae considered are contained in the table on the next page. The notation used is as follows :-

a = plate radius

$t = 2h$ = plate thickness

w = density of material of plate

E = Young's Modulus for the material

S = Poisson's Ratio for the material

g = acceleration due to gravity

$f = \frac{p}{2\pi}$ = natural frequency of vibration

Of the formulae given only (2) and (5) are identical. (1) is similar to these except that it quotes the result in cycles/sec. instead of radians/sec. In the text the author does not define the units of "f" but he calls it the "natural frequency" which most readers would surely assume to be cycles/sec.

Formula (3) becomes identical with (2) and (5) if "mass" density $\frac{w}{g}$ is used instead of "w". This correction is required to make the equation dimensionally balanced. In the text the author gives a worked example in which he uses a value of 7.8 for "w" for steel in the C.G.S. system and applies the formula without including "g". This is almost certainly in error.

Formula (4) is similar in form but numerically different from the others.

In cases (2) and (4) proofs are given. Both use the Rayliegh approximate method and yet the results disagree. The proofs were checked and an apparent error discovered in Temple & Bickley's method.

The basis of the methods used in the proofs is to equate a value of the strain energy in the plate to a corresponding value of the kinetic energy. Timoshenko uses maximum values whereas Temple & Bickley use mean values but since the maximum value can be shown to be twice the mean in each case the proofs do not differ in this point.

Kinetic Energies

Consider an element of the plate at radius "r", width radially "dr" and subtending angle "$d\theta$" at the centre.

Now.

Mass of element $= \dfrac{w}{g}$ tr dr $d\theta$.

Displacement of element from normal position $= Z_o \sin pt$

where Z_o = amplitude of vibration $= f(r)$

Velocity of element $= \dfrac{dZ_o}{dt} = Z_o\, p \cos pt$

$\therefore$ Kinetic Energy of element $= \dfrac{wt}{2g} Z_o^2 \; p^2 \; \cos^2 pt \; r \, dr \, d\theta$.

$$\text{Kinetic Energy of plate} = \frac{wt}{2g} p^2 \cos^2 pt \int_{\theta=0}^{2\pi} \int_{r=0}^{a} Z_o^2 \, r \, dr \, d\theta.$$

If the plate is symmetrical about the centre then Z_o is not a function of θ.

$$\therefore \quad \text{Kinetic Energy of plate} = \frac{wt}{2g} p^2 \cos^2 pt \; 2\pi \int_0^a Z_o^2 \, r \, dr.$$

For maximum value during the cycle $\cos^2 pt = 1$.

For mean value during the cycle we use the mean value of $\cos^2 pt$ which is :-

$$\frac{1}{2\pi} \int_0^{2\pi} \frac{1}{2} (1 + \cos 2pt) \; d(pt)$$

$$= \frac{1}{2}$$

Hence mean value $= \dfrac{1}{2}$ maximum value.

Strain Energies

Both proofs take the strain energy on the element of plate to be :-

$$d(\text{S.E.}) = \frac{Et^3}{24(1 - s^2)} \left\{ \frac{\partial^2 z}{\partial r^2} + \frac{1}{r} \frac{\partial z}{\partial r} \right\}^2 r \, dr \, d\theta.$$

For the whole plate :-

$$S.E. = \frac{Et^3}{24(1 - S^2)} \int_{\theta=0}^{2\pi} \int_{r=0}^{a} \left\{ \frac{\partial^2 Z}{\partial r^2} + \frac{1}{r} \frac{\partial Z}{\partial r} \right\}^2 r \, dr \, d\theta.$$

again if $Z = Z_0 \sin pt$ where "Z" is a function of "r" but not of "θ", and "sin pt" is not a function of "r" we get

$$S.E. = \frac{Et^3}{24(1 - S^2)} \; 2\pi \; \sin^2 wt \int_0^a \left\{ \frac{\partial^2 Z_0}{\partial r^2} + \frac{1}{r} \frac{\partial Z_0}{\partial r} \right\} r \, dr.$$

Up to this point the proofs are identical. The next step requires the choice of a function for "Z_0" in terms of "r" to enable the integration to be carried out. The chosen function must satisfy the boundary conditions and Rayliegh's Principle states that provided it gives a reasonable approximation to the deflected shape of the plate then the frequency as calculated from the expression will be close to the natural frequency and in any case not lower than it.

Timoshenko uses the series :-

$$Z_0 = a_1 \left(1 + \frac{r^2}{a^2}\right)^2 + a_2 \left(1 + \frac{r^2}{a^2}\right)^3 + a_3 \dots\dots\dots$$

and taking the first two terms and equating maximum K.E. to maximum S.E. he gets

$$p = 10.21 \frac{t}{a^2} \sqrt{\frac{gE}{12w(1 - S^2)}}$$

Temple and Bickley take the function :-

$$Z_o = C\left(a^3 - 3\,a\,r^2 + 2\,r^3\right)$$

and get

$$\text{mean K.E.} = \frac{wt}{2g} \quad p^2\pi\int_0^a C^2\left(a^3 - 3\,ar^2 + 2\,r^3\right)^2 r\,dr.$$

Expansion, integration and substitution gives :-

$$\text{Mean K.E.} = \frac{3}{70}\,\frac{wt}{g} \quad p^2\pi\;C^2\;a^8.$$

Also if :-

$$Z_o = C\left(a^3 - 3\,ar^2 + 2\,r^3\right)$$

we get

$$\frac{1}{r}\,\frac{\partial Z_o}{\partial r} = 6\;C\;(r - a)$$

and $$\frac{\partial^2 Z_o}{\partial r^2} = 6\;C\;(2r - a)$$

$$\therefore \quad \text{Mean S.E.} = \frac{1}{2}\,\frac{Et^3}{24(1 - S^2)} \quad 2\pi\int_0^a\Big\{6\;C(3r - 2a)\Big\}^2\;r\,dr.$$

Expansion, integration and substitution gives :-

$$\text{Mean S.E.} = 9\,\frac{Et^3}{24(1 - S^2)}\;\pi\;C^2\;a^4.$$

This does not agree with the value given by Temple and Bickley which is

$$\text{Mean S.E.} = \frac{90}{2}\,\frac{Et^3\pi\,C^2\;a^4}{24(1 - S^2)}$$

However, continuing with the values as calculated above we equate the mean values of strain and kinetic energies to get

$$\frac{3}{70}\;\frac{wt}{g}\;p^2\;a^8 = 9\;\frac{Et^3 \qquad a^4}{24(1-S^2)}$$

whence

$$p^2 = 210\;\frac{Et^3}{24(1-S^2)}\cdot\frac{g}{w\,t}\cdot\frac{1}{a^4} \qquad (6)$$

Note that

$$\frac{Et^3}{24(1-S^2)} = \frac{Eh^3}{3(1-S^2)} = D^1$$

and $\frac{wt}{g}$ = mass per unit area = m.

Temple and Bickley give :-

$$p^2 = \frac{105\;D^1}{m\;a^4}$$ however even

if the figures they give for mean K.E. and mean S.E. were correct they should get

$$p^2 = \frac{1050\;D^1}{m\;a^4}.$$

Reducing (6) to a form comparable with the other formulae gives

$$p = \sqrt{105}\;\frac{t}{a^2}\sqrt{\frac{Eg}{12w(1-S^2)}}$$

and since $\sqrt{105} \doteqdot 10.21.$ the formula obtained is identical with Timoshenko's apart from a very slight discrepancy due to the different choice of function for Z_o.

Strictly speaking this formula applies to a plate vibrating in a vacuum without stretching of its central plane. When the plate vibrates in a fluid some of the fluid vibrates along with the plate. This has the effect of increasing the inertia of the system (A.6) and reducing the natural frequency. The size of the reduction depends on the ratio of the plate density to the fluid density and when the fluid is air the reduction will be small. In the case of a thin plate where the strain energy is not the result of simple bending a more accurate theory consistent with the more accurate theory for deflection (3.8) has been developed. This gives a higher value for the natural frequency than Timoshenko's formula.

R E F E R E N C E S

1.1 H.M. STATIONERY OFFICE "Mines & Quarries Act 1954", Sect. 55.

1.2 WILKIE, A. "Some Applications of Mine Ventilation Network Theory". Mining Dept. Library, R.C.S.T., Glasgow. 1956.

1.3 SKOCHINSKY, A.A., et al. "Resistance of Concrete Mine Shafts with metal structures" Ugol, March 1951, pp. 4-9. N.C.B. Trans. A623/AB.

1.4 BARCZA, M., et al. "An Investigation into the Resistance to Airflow of No.5 Shaft, City Deep, Ltd." Jour. Chem. Metall. Min. Soc. S. Africa. Vol. 56, No.2, August 1955, p. 49.

1.5 SMITH, C.M. "Application of Model Tests to the Determination of Losses around a Shaft Bottom". Bull. Eng. Exp. Stat., Illinois. No.265, 1934.

1.6 TOMLIN, N. "The Investigation of Fan Drift Design by Means of Models". Trans. Inst. Min. Engrs. Vol. 113, 1953, p. 138.

1.7 KEMP, J.F. "Scale Model Testing in Aerodynamics with Special Reference to the Study of Mine Shaft Resistance". Jour. Mine Vent. Soc. S. Africa, Vol. 11, No.12, Dec. 1958. pp. 251-258.

1.8 JONES, J.D. and HINSLEY, F.B. "The Use of Models in the Prediction of Mine Airway Resistance" Trans. Inst. Min. Engrs. Vol. 118. No.11. August 1959, p.723.

1.9 STEVENSON, A. "Mine Ventilation Investigations" Mining Dept. Library, R.C.S.T., Glasgow, 1956.

1.10 SCHLICHTING, H. "Boundary Layer Theory", Pergammon Press Ltd., London, 1955. p. 10.

1.11 LE ROUX, W.L. and CHASTEAU, V.A.L. "Investigations into the Resistance to Airflow of the Pioneer Shafts" Buffelsfontein Gold Mining Co. Ltd. Jour. Mine Vent. Soc. S. Africa. Vol. 12, No. 6, June 1959, pp. 129 et seq.

2.1 WENK, E. "A Diaphragm Type Gauge for Measuring Low Pressures in Fluids". David Taylor Model Basin, Rep. 665. Jan. 1950.

2.2 WOLFE, M.O.W. "The Measurement of Fluctuating Fluid Pressures". Aircraft Engineering. Vol. 21, Dec. 1949, pp.368-377.

2.3 HINSLEY, F.B. and KILLORAN, J.R. "Pressure Fluctuations in Mine Airflow" Colliery Guardian, Vol. 195, Oct. 1957, p. 499.

2.4 HAGEN, H.F. "Pulsation of Air Flow from Fans and its effect on test procedure". Trans. Amer. Soc. Mech. Engrs., Vol. 55, Sect. 7, 1933, p. 105.

2.5 FECHHEIMER, C. J. "Ventilation of Revolving Field Salient Pole Alternators". Quar. Trans. A.I.E.E., April 1930, p. 548.

2.6 GROSS, N., and LANE, P.H.R. "An 'Accurate' Wire Resistance Method for the Measurement of Pulsating Pressures". Brit. Welding Res. Assoc. Comm. Rep., FE 12/23, May 1952.

2.7 BROOKS-SMITH, C.H.W. and COLLS, J.A. "The Measurement of Pressure and other Mechanical Quantities by Electrostatic Methods". Jour. Sci. Instru. Vol. 16, No. 12, Dec. 1939, pp. 361-366.

2.8 WILLIAMS, J.L. and EVESON, G.F. "A Vibrating Condenser Manometer". Jour. Sci. Instru. Vol. 35, No.3, March 1958, p.97.

2.9 THOMSON, SIR J.J. "Piezo-Electricity and its Applications". Engineering Vol. 107, 1919, p. 543.

2.10 SMITH, F.D. "Basic Principes in the Design of Cathode Ray Oscillograph engine indicators". Proc. Inst. Mech. Engrs., Vol. 143, April 1940, p. 48.

2.11 HERSEY, M.D. "Diaphragms for Aeronautic Instruments". N.A.C.A., Tech. Rep. 165, 1923.

2.12 WILDHACK, W.A. and GOERKE, V.H. "Corrugated Metal Diaphragms for Aircraft Pressure Measuring Instruments". N.A.C.A. Tech. Note 738, 1939.

2.13 JONES, E.B. "Instrument Technology". Butterworth's Scientific Publications, London, 1953, Vol. 1, p. 33 et seq.

2.14 COOK, D.B. and DANBY, C.J. "A Simple Diaphragm Micromanometer". Jour. Sci. Instru. Vol. 30, No. 7, July 1953, p. 238.

3.1 AUGHTIE, F. "Electrical Resistance Wire Strain Gauges". Proc. Inst. Mech. Engrs. Vol. 152, No.2, 1945, p. 213.

3.2 HARRIS, F.K. "Electrical Measurements". John Wiley & Sons Inc., New York, 1951, Ch. 3.

3.3 THOMSON, R.T.C. "The Initial Work on the Strain Gauge Diaphragm". Mining Dept. Library, R.C.S.T., Glasgow, 1955.

3.4 MORELY, A. "Strength of Materials". Longmans Green & Co. 11th Ed., 1954, p.457 et seq.

3.5 CRAWFORD, W.J. "The Elastic Strength of Flat Plates". Proc. Roy. Soc. Edin., Vol. 32, part 4, 1911-12, p. 348.

3.6 CARTER, B.C. et al. "Measurement of Surface Strains in Diaghragms". Engineering, Vol. 168, Dec. 1949, pp. 581-583.

3.7	STEINTHAL, P.T.	"Some Experimental Data on the Flexure of Flat Circular Plates within the elastic limits". Engineering, Vol. 91, May 1911, p. 677.
3.8	PRESCOTT, J.	"Applied Elasticity". Longmans Green & Co., 1924. Ch. 15.
3.9	CARRINGTON, H.	"The deflection of Flat Plates fixed at the Circumference". Engineering, Vol. 125, Jan. 1928, p. 31.
3.10	KOCH, J.J., et al.	"Strain Gauges Theory and Applications".
3.11	SENIOR, D.A.	"Recording Signals from Resistance Strain Gauges". The Engineer. Vol. 197, March 1954, pp. 410-412.
4.1	ARGUIMBAU, L.B.	"Vacuum Tube Circuits". John Wiley and Sons, New York, 5th Printing, 1952, p. 29 et seq.
5.1	TANFIELD, D.A.	N.C.B. Directorate of Extra-Mural Research. Private communication.
5.2	TABACK, I.	"The Response of Pressure Measuring Systems to Oscillating Pressures". N.A.C.A. Tech. Rep. 1819, 1949.
5.3	WILLIAMS, E.M. and WOODFORD, J.B.	"Transmission Circuits". The Macmillan Co., New York, Fifth Printing 1959, p. 9.
5.4	OLSON, H.F. and MASSA, F.	"Applied Acoustics". Constable & Co. Ltd., London. 2nd Ed. 1939, p. 17 et seq.
5.5	KAYE, G.W.C. and SHERRAT, G.G.	"The Velocity of Sound in Gases in Tubes". Proc. Roy. Soc., Vol. 141, 1933. p. 133.
6.1	NATIONAL COAL BOARD	Drawing No. 6023 F 97. Headquarters, London.
6.2	BRITISH STANDARDS INSTITUTION	"Flow Measurement". B.S. Code 1042, 1943.

6.3	PRESTON, J.H.	"The Three-Quarter Radius Pitot Tube Type Flowmeter". The Engineer, Vol. 190, 1950, p. 400.
6.4	WINTERNITZ, F.A.L.	Fluid Mechanics Division, M.E.R.L., East Kilbride. Private Communication.
7.1	OWER, E.	"Measurement of Air Flow". Chapmans Hall, London. 1949, p. 34.
7.2	BROWNLEE, K.A.	Industrial Experimentation. H.M.S.O. 4th Ed. 1957.
7.3	WINTERNITZ, F.A.L. and FISCHL, C.F.	"A Simplified Integration Technique for Pipe Flow Measurement". Water Power, Vol. 9, No.6, 1957, pp. 225 - 234.
8.1	SKOCHINSKY, A.A. et al.	"Air Resistance of Mine Shafts and Means of Lowering it". Ugol, July 1952, pp. 9 - 19. N.C.B. Trans. A622/AB.
8.2	ABRAMOV, F.A.	"Determination of Resistance to Ventilation of Round Concreted Mine Shafts". Ugol, 1953, pp. 28 - 31. N.C.B. Trans. A624/AB.
8.3	MARTINSON, M.J.	"Determining the Friction Factors of No.2 and No.3 Shafts, Harmony Gold Mining Co. by means of Scale Models". Jour. Mine Vent. Soc. S. Africa. Vol. 10, No.3, 1957, pp. 51 - 63.
8.4	NATIONAL COAL BOARD	"Planning the Ventilation of New or Reorganised Collieries". Information Bulletin No. 55/153, Hobart House, London.
A.1	HETENYI, M.	"Handbook of Experimental Stress Analysis". John Wiley & Sons, New York, 1950, p. 278.
A.2	TIMOSHENKO, S.	"Vibration Problems in Engineering". Constable & Co., London 1928, p. 315.
A.3	WOOD, A.B.	"Textbook of Sound". G. Bell & Son, London, 3rd Ed. 1955, p. 366.

A.4 TEMPLE, G. and BICKLEY, W.G. "Rayliegh's Principle". Dover Publications Inc., London 1933, p. 113. Reprinted 1956.

A.5 CARRINGTON, H. "The Frequencies of Vibration of Flat Circular Plates fixed at the Circumference". Phil., Mag., Vol. 50, No.3, Dec. 1925, p. 1261.

A.6 LAMB, H. "On the Vibrations of an Elastic Plate in Contact with Water". Proc. Roy. Soc., Vol. 98, 1921, p. 205.

SUMMARY

"THE INFLUENCE OF MOVING CAGES ON AIRFLOW IN MINE SHAFTS"

A thesis presented for the degree of Doctor of Philosophy of the University of Glasgow by Alan Wilkie, B.Sc., December, 1959.

This thesis contains the results of an experimental investigation into some of the effects produced on the airflow conditions in mine shafts by the movement of cages.

The investigation was carried out in the Mining Department and was supervised by Professor G. Hibberd.

The investigation took the form of a programme of tests on a model shaft fitted with buntons, guides and moving cages. The range of Reynold's Numbers available was from 250,000 to 450,000.

During the years 1953-56, A. Stevenson studied the factors influencing shaft pressure losses produced by stationary cages, and noted also the effect of streamlined cages in reducing these losses.

The present work is an extension of Stevenson's work and includes the effects produced by cages in motion. These effects are compared with those produced by stationary cages and the relative importance of the cage effects and the general shaft resistance is established.

The movement of the cages in the model shaft was expected to produce small fluctuations superimposed on air pressures in the range 0 to 14 inches W.G. (0 to 0.5 p.s.i.) which had to be continuously recorded. When the investigations were begun in 1956 no manometer capable of recording such small pressure fluctuation was in commercial production and it was necessary to build one.

The thesis contains details of the development of suitable pressure-recording apparatus. The methods available for recording fluctuating fluid pressures are discussed and trials with electric resistance strain gauge transducers and with variable capacitance type transducers are described. These show the latter type to be more suitable and a satisfactory prototype is eventually produced. The factors affecting the response of the apparatus to fluctuating pressures are also considered.

In the experimental work the tests first cover stationary two-deck and four-deck cages with and without straight-sided fairings supported in two different patterns of guides. Cage resistances are expressed as Pressure Drop Coefficients (P.D.C.) and particular attention is paid to the magnitude and extent of the increase in combined cage P.D.C. when two cages of a winding system are within the zone where their combined resistance varies with the distance between them. The length of this zone is shown to be less than 4 shaft diameters in all cases and the variations in P.D.C. when the cages are within it are shown in detail. The use of fairings is shown to produce some reductions in the cage P.D.C.

The work is then extended to note the effects of cages in motion. The tests cover a range of cage speeds and can be divided as follows :-

a) Tests on the influence of passing cages on shaft resistance,
b) Tests on the influence of passing cages on conditions upstream from the passing place,
c) Tests on the influence of passing cages on conditions downstream from the passing place,
d) Tests on the influence of a cage passing a point on conditions at that point.

The results show that the moving of cages can produce considerable disturbance and often very rapid changes in the airflow conditions in mine shafts. These effects, however, are usually of fairly short duration. The use of fairings is shown to reduce the disturbances and make the changes less rapid but this is usually done at the expense of an increase in the duration of the effects.