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MINE VENTILATION INVESTIGATIONS:

A. ANEMOMETER TRAVERSING ACCURACY,

B. REGULATOR PRESSURE LOSSES,

C. SHAFT PRESSURE LOSSES DUE TO CAGES.

by

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A Thesis presented for the degree of Doctor of Philosophy of Glasgow University.

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

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P R E F A C E

The Thesis embodies the results of investigations into some of the problems met with in the ventilation of mines. There are three main sections.

Section A deals with some of the factors affecting the accuracy with which the mean velocity across an underground roadway cross-section can be determined by traversing with an anemometer of the wind-mill type. The accuracy of traversing with the operator out of the air-way cross-section is first of all examined. Then the effect of the operator's position during traversing is investigated, and also the influence of disturbed flow conditions produced by a bend.

In Section B factors affecting the pressure loss produced by a regulator are examined, viz., the ratio of area of regulator opening to area of wind-tunnel; the influence of the shape of the regulator opening; and the location of the regulator opening in the wind-tunnel cross-section.

The resistance to air-flow of cages is investigated in Section C, by experiments on stationary models. The factors examined are:
the resistances of cages suitable for single and double-cage winding in rope-guide installations; and for double and quadruple winding in rigid-guide installations; the relationship between resistance and the number of decks; the effect of loading a cage with tubs; the reduction in resistance brought about by using Joukowski Symmetrical Aerofoil nose and tail-fairings; the relationship between resistance and the ratio

of the frontal area of the cage to the cross-sectional area of the shaft; the result of changing the frontal shape of the cage; the effect of symmetrical straight-sided fairings on resistance, and also of asymmetrical straight-sided fairings designed to offset any "suction" produced when the cages pass each other; and the resistance of shaft linings, comprising rigid end-guides and buntons, suitable for two and four-cage installations.

The three sections are for the most part independent of each other. However, there is some correlation and similarity between the measuring techniques used throughout the investigations. As the research work progressed, satisfactory methods of estimating mean air quantities were developed. In addition instruments suitable for the measurement of fluctuating air pressures were eventually obtained.

C O N T E N T S

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SECTION A

ANEMOMETER TRAVERSING ACCURACY

INTRODUCTION

In recent years considerable progress has been made in the solution of ventilation network problems by machine. It has been stressed (1), however, that the full advantage of these developments will not be realised unless the accuracy of the basic experimental readings can be increased. One of the biggest sources of error lies in the determination of the amount of air flowing in a roadway; measurements taken by different operators and instruments can differ widely and at a roadway junction the several quantities are usually irreconcileable. In order to find the air-flow it is necessary to determine the mean velocity existing across a known area and this is normally accomplished by traversing the cross-section with an anemometer of the windmill type. This section deals with the investigation of some of the factors affecting the accuracy of this method of determining mean velocity.

Such problems as the effect of change in air conditions from the time of calibration, the acceleration of the gears when the mechanism recording the air speed has been engaged, and the effect of yawing have already been carefully investigated. Another important factor which has been found (2) to influence the instrument readings is the position of the operator. In recent work into this aspect (3) the influence of the position of the operator upon the readings of an anemometer maintained in a fixed position were thoroughly examined;

also it was shown that, using anemometers in both cases, the traversing method compared favourably with the more precise system of sub-dividing the cross-section.

The purpose of the investigations here described was to examine the influence of the operator's position during traversing, and also to try to establish the fundamental accuracy of this method of determining mean velocity. The mean velocity obtained by anemometer traverse was compared with the value derived from an independent measurement of the quantity passing. The quantity passing was obtained indirectly from pitot-static tube readings taken at a specially constructed measuring station. Also examined was the influence of disturbed flow conditions. All the tests were carried out in an experimental mine where suitable conditions were readily obtained.

The section is concluded with a discussion of the results obtained, followed by conclusions drawn therefrom.

GLASGOW UNIVERSITY MINE

Recently, when preparing the foundations for an extension to the University the uncharted workings of an old mine were discovered. The system of underpinning adopted allowed the formation of a small experimental mine.

The layout of the mine is shown in Fig. A1. The doors and regulators permit various air circuits to be obtained and the distribution of the air to be closely controlled. The sides of each roadway are brick lined while the roof is of sandstone. They are rectangular in cross-section and the maximum area available is about 35 square feet. Adequate electric lighting is provided in all parts of the mine except the coal face.

The mine is ventilated by a 42 inch diameter single-stage axial flow fan capable of producing 2.5 inches water gauge. For experimental purposes the fan speed may be regulated by means of a ten-notch rheostat. The controls for the fan and the distribution of electricity are located in the fan-house (see Plate IA).

Two separate stretches of the main roadway have been lined, one with circle-arch girders, the other with round timber, since the time of the investigations (see Plates IIA and IIIA). The main roadway is paved with rock chips and maintained dry by a drainage channel. Any water present in the mine gravitates to the sump from which it is pumped to the surface. The mine is entered through a three-door airlock.

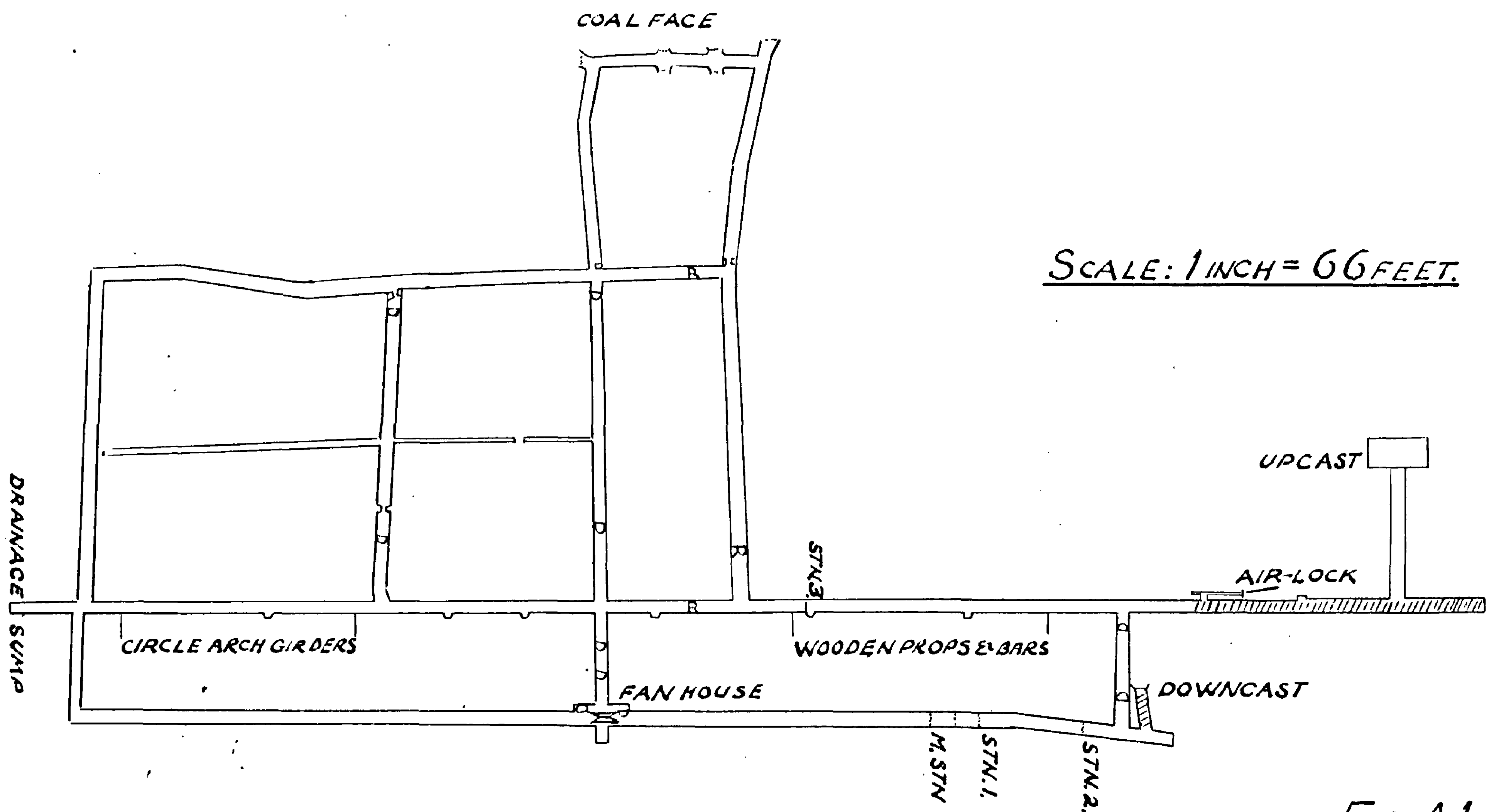


FIG. A1.

GLASGOW UNIVERSITY MINE.

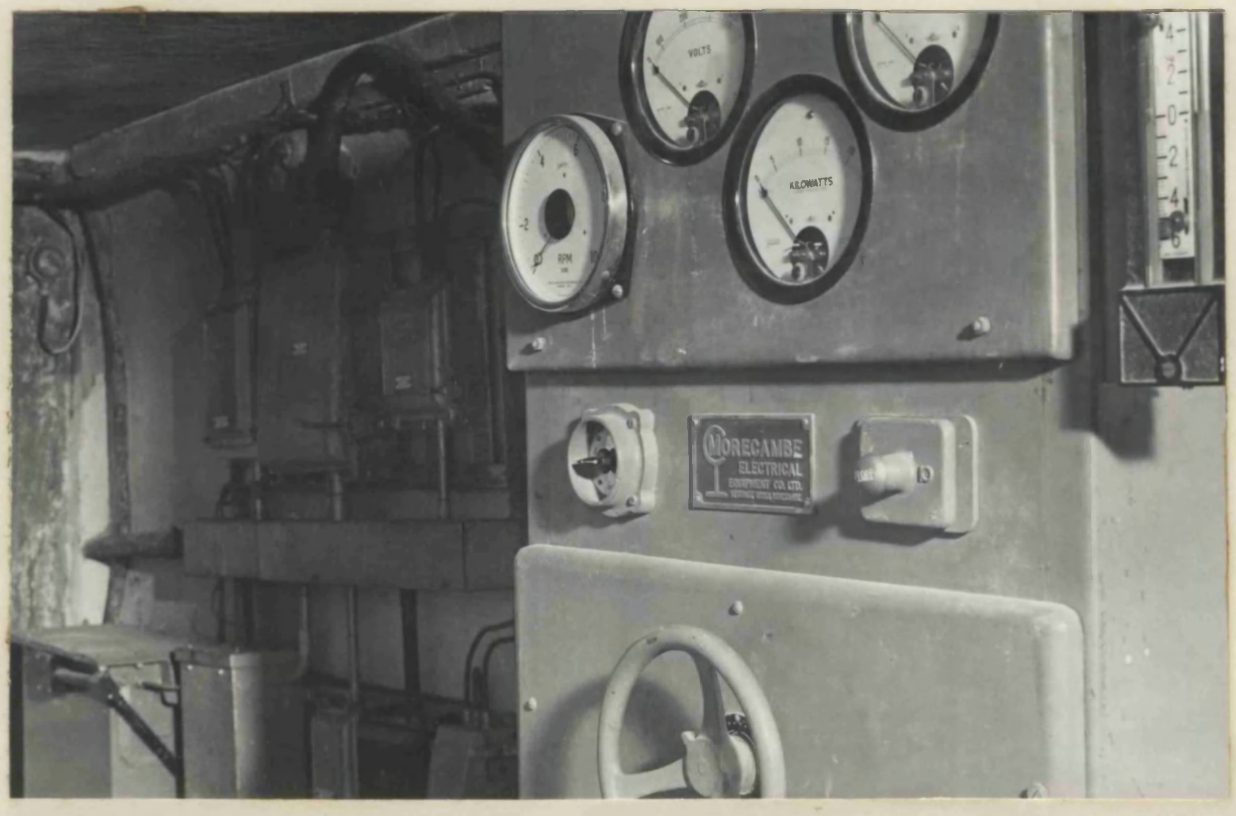


PLATE IA: Fan House - University Mine

PLATE IIA: Circle-Arch Girdered Section - University Mine

PLATE IIIA: Timber Supported Section - University Mine

PLATE IVA: Measuring Station - University Mine

UNDERGROUND PREPARATIONS

Measuring Station (see Plate IVA)

Since it was proposed to obtain an accurate measurement of the quantity of air flowing in the mine by means of a pitot-static tube traverse, it was necessary to construct a smooth-sided Measuring Station whose area could be determined within one or two per cent. This method of traversing has already been used satisfactorily underground in an unadapted air-way (4) and at a specially constructed measuring station (5). A suitable site was obtained 122 feet from the fan and 82 feet from the downcast air-pit.

The station was made six feet long, the floor and sides being given a smooth cement finish. A pipe at one side kept the station free from ground water. The roof consisted of expanded metal plastering screen supported on a wooden frame and covered by a layer of cement-hair mixture. The profile of the station was maintained as rectangular as possible.

The arrangements were completed by the construction of a vertical carrier for the pitot-static tube which could be clamped between two horizontal steel guides situated at the end of the station nearer the fan. By suitably marking the guides and carrier the pitot-static tube could be located at the desired positions in the airway cross-section. So that its characteristics would be uninfluenced by the proximity of the vertical support the pitot tube was clamped with its end eighteen inches from it. The cross-section whose mean velocity

velocity would be ascertained was determined and the outline marked with paint.

Experimental Stations

Stations 1, 2 and 3 were prepared by removing any loose rubble and painting a one inch line on the sides, roof and floor. At Station 3 the presence of a 2 feet gauge single track railway necessitated the levelling off of the floor.

EQUIPMENT

Askania Minimeter (see Plate VA)

The principle of operation of this type of instrument is well known so that no explanation is necessary. In the instrument used the least scale division represented 0.01 mm. of water column and the range was 0 - 150mm.

Psychrometer

This consisted of two N.P.L. calibrated thermometers graduated in degrees from 15 - 110 degrees Fahrenheit with their bulbs ventilated by a small clock-work fan.

Barometer

A Short and Mason aneroid barometer reading to 0.02 inch over the range 27 - 32 inches of Mercury was used.

Anemometers

The anemometers used were of the following types:-

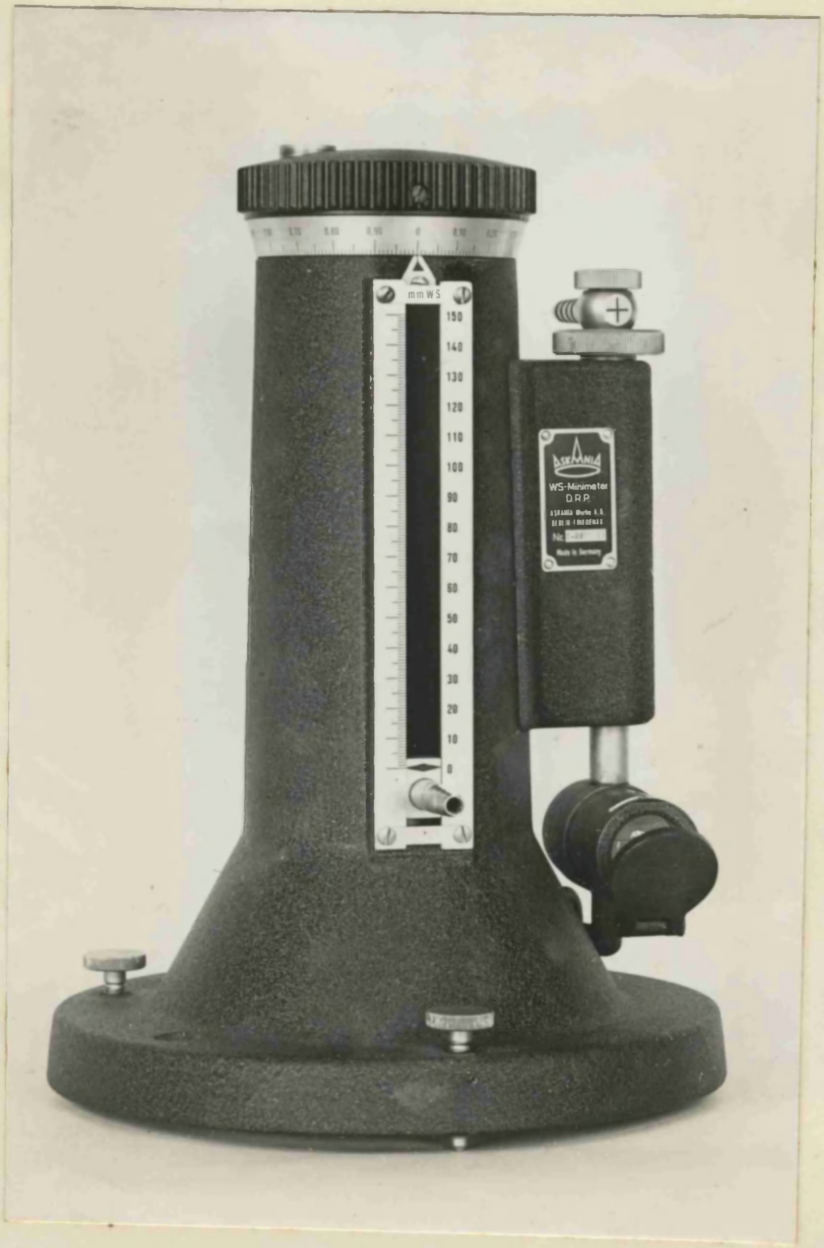


PLATE VA: Askania Minimeter

(a) Davis Biram (B125) which can be fitted with a short hand-grip or attached to the end of a 3 feet rod. In each case the instrument can be started and stopped by a press-button located on the handle but must always be used to the side of the operator.

(b) No.K9534 equipped with a trigger release. This anemometer can be used for traversing at the end of a three feet rod with the operator either in the same cross-section as the instrument or downstream from it.

Whirling Arm

The anemometers were calibrated by mounting them at $5\frac{1}{2}$ feet radius on a whirling arm whose speed could be varied from 5 - 70 revolutions per minute. Remote control of the instruments was provided by a solenoid arrangement.

MEASUREMENT OF AIR-FLOW PAST THE MEASURING STATION

The mean quantity of flow past the Measuring Station was obtained as follows:-

Determination of Centre Constants

The object of this experiment was to find the variation in the ratio of mean velocity to centre velocity at the measuring cross-section for the range of air speeds to be used. Then the mean velocity at the measuring station could be readily evaluated from a single velocity measurement.

The Askania Minimeter was installed in the fan-house and connected to the pitot-tube at the measuring station by two lengths of rubber tubing.

The psychrometer and Aneroid barometer were placed on the downstream side of the measuring station. From the readings of these two instruments the air density was calculated using the formula given by McElroy (6) for mining work, viz. -

$$w = \frac{1.325}{459 + t} (B - 0.378f)$$

where,

w = air density in lb. per cubic foot,

t = dry bulb temperature, $^{\circ}$F,

B = barometric pressure in inches of mercury,

f = vapour pressure at Dew Point in inches of mercury.

Suitable quantities in the range 7,000 - 22,000 cubic feet per minute were arranged to flow past the measuring station. For each quantity the mean velocity was determined by dividing the cross-section into sixteen parts of equal area geometrically similar to the whole section (7), and measuring the velocity head at the centre of each sub-section by pitot-static tube. The system of numbers and letters illustrated in Fig.B4 (follows page **34.**) was used to define the location of the pitot-static tube. The intersection of the "2" axis with the vertical centre line was taken as the "centre position" thereby slightly reducing the time required for a complete traverse.

The mean velocity of flow Vm is defined as that velocity which, when multiplied by the area of the cross-section being considered, gives the volume flowing past the section. It can be readily shown (8) that if the measuring station is divided into n equal areas in which the velocities are V_1, V_2, V_3 etc. then

$$Vm = \frac{1}{n}(V_1 + V_2 + \ldots\ldots. V_n)$$

Now for constant density conditions

$$V = kh^{\frac{1}{2}}$$

where,

k = constant

h = velocity head in height of manometric fluid.

Therefore,

$$Vm = \frac{k}{n}\left(h_1^{\frac{1}{2}} + h_2^{\frac{1}{2}} + \ldots\ldots. h_n^{\frac{1}{2}}\right)$$

and hence

$$\text{Centre constant} = \frac{Vm}{Vc}$$

$$= \frac{\left(h_1^{\frac{1}{2}} + h_2^{\frac{1}{2}} + \ldots . . h_n^{\frac{1}{2}}\right)}{h_c^{\frac{1}{2}}}$$

From these experiments the relationship between the Centre Constant and the velocity at the "centre position" was obtained for the maximum possible range of air quantity. The results are shown in Table No.1.

TABLE No.1

Traverse Number	1	2	3	4	5	6	7	8	9
Centre Velocity ft./sec.	6.75	7.75	8.40	9.10	10.1	11.4	13.4	16.5	18.9
Centre Constant	0.92	0.92	0.92	0.92	0.93	0.91	0.91	0.91	0.92

The range of Reynolds Number respresented by Table No.1 is 170,000 - 470,000, and results obtained by Nikuradse (9) show that between these two values the rate of change of the Centre Constant can be expected to be slight. Therefore from the figures shown in the Table it would appear reasonable to take the Centre Constant as 0.92 for the range of velocities measured.

Accuracy of Mean Velocity

The accuracy of the mean velocity will be determined by the accuracy with which the centre velocity can be ascertained and the accuracy of the Centre Constant. It was necessary therefore to establish a method of calculating the error involved in the measurement of the centre velocity. In this and future experiments the estimation of errors was performed using an expression obtained by Briggs (10).

In considering a number of observations of a certain quantity he states that the Mean Error of the Arithmetic Mean may be derived from the relation

$$Em = \pm \left\{ \frac{S.r^2}{n(n-1)} \right\}^{\frac{1}{2}}$$

where,

S = summation

r = residual

= observed value - arithmetic mean

n = number of observations.

The above formula is derived on the assumption that the distribution of the errors about a mean value is "normal" and Briggs argues that for small numbers of observations this supposition is hardly justified. He advocates the use of the Average Error of the Arithmetic Mean given by

$$Ea = \pm \frac{S.r}{n}$$

Since the number of observations of any one quantity usually had to be curtailed because of the time available, Average Errors have been used throughout this section of the investigations. The formula is much easier to handle and it will be noted that for n greater than two the Average Errors are always larger than the corresponding Mean Errors. Therefore some allowance is made for the deficiency in the number of observations by a decrease in the estimated accuracy of the arithmetic mean.

For each of four different air speeds twenty readings of the velocity head at the centre position were taken and the error of the arithmetic mean calculated. The velocities were determined from the relation

$$V = 3.63 \left(\frac{h}{w}\right)^{\frac{1}{2}}$$

where,

V = velocity in feet per second,

h = velocity head in m.m. water column,

w = air density in lb. per cubic foot.

Using other expressions derived by Briggs for the propagation of error the accuracy of each mean velocity was evaluated. It was discovered that due to the air density determinations being relatively much more accurate than the readings of velocity head, their influence upon the error involved in the calculated values of V was negligible. The results of this short experiment are shown in Table No.2.

TABLE No.2

Test Number	1	2	3	4
Velocity Head h mm.	0.37	0.65	1.17	1.68
Average Error in h_{mean}	±0.011	±0.012	±0.015	±0.010
Centre Velocity (ft.per sec.)	7.40	9.80	12.85	17.04
Error in Centre Velocity	±0.12	±0.10	±0.09	±0.05

From the tabulated figures it will be noted that as the velocity head being measured increases so does the accuracy in the estimation of the centre velocity. The results were considered satisfactory and it was decided that in future experiments the centre velocity and its accuracy could be determined from at least twenty readings of velocity head.

The error attributable to the Centre Constant could not be ascertained so precisely. Although 0.92 is a fair average of the values shown in Table No.1 a full analysis of the velocities in the sub-sections at the corners and sides of the measuring station could not be made. To allow for these omissions the Centre Constant was taken as 0.92 ± 0.01.

The accuracy of the mean velocity is obtained from the errors relating to the values of Centre Constant and centre velocity.

Cross-Sectional Area of Measuring Station

The final requirement for the determination of the mean quantity of air flowing past the measuring station is an estimate of the cross-sectional area. The four sides and one diagonal of the rectangular cross-section denoted by the painted line were measured to the nearest 0.5 inches. Fig. A2 shows the outline of the station looking in the direction of the fan and the dimensions obtained were:

AB	=	a	=	72.0 inches
BC	=	b	=	41.0 inches
CD	=	c	=	71.5 inches
DA	=	d	=	42.0 inches
AC	=	e	=	81.5 inches

From these measurements the area of triangles ABC and ADC was calculated and the area of the measuring station found to be 20.66 square feet.

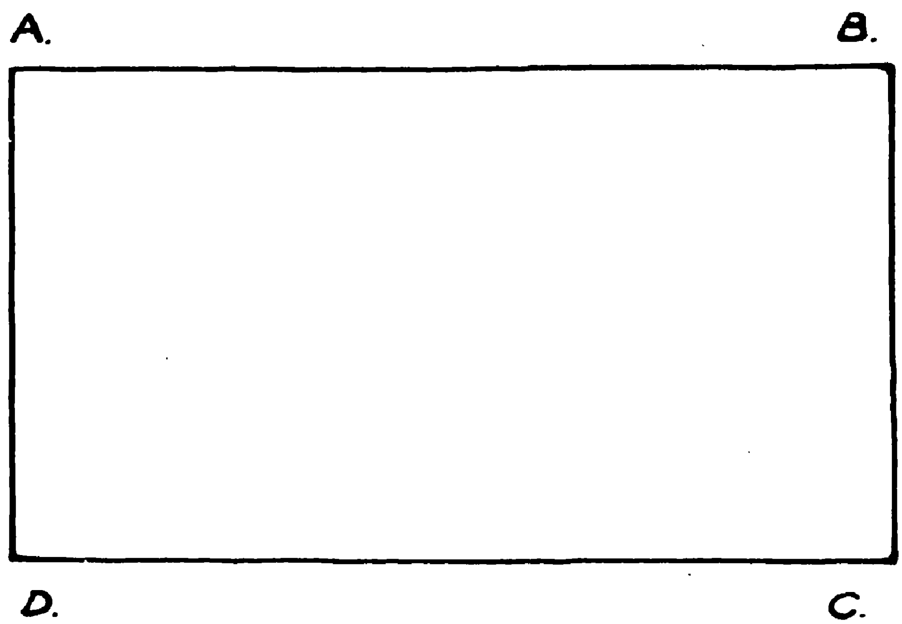


FIG. A2.

MEASURING STATION OUTLINE.

The accuracy of this result was estimated from the expressions already referred to (10). The maximum error incurred in the direct measurements was assumed to be ± 0.25 inches.

For triangle ABC

$$\text{Area} = F$$

$$= \left((S - a)(S - b)(S - e)\ S \right)^{\frac{1}{2}}$$

where

$$a = 72.0 \pm 0.25$$

$$b = 41.0 \pm 0.25$$

$$e = 81.5 \pm 0.25$$

$$2S = a + b + e$$

$$= 194.5 \pm (3 \times 0.25^2)^{\frac{1}{2}}$$

Considering 2S as a sum, with respect to S, rather than a product gives the acceptable result

$$S = 97.25 \pm 0.306$$

The error in (S - a), (S - b) or (S - e) is then

$$= \pm (0.306^2 + 0.25^2)^{\frac{1}{2}}$$

$$= \pm 0.391$$

If (S - a), (S - b), (S - e) are denoted respectively by A, B and E, then

$$F = (A \times B \times E \times S)^{\frac{1}{2}}$$

where

$$A = 25.25 \pm 0.391$$

$$B = 56.25 \pm 0.391$$

$$E = 15.75 \pm 0.391$$

The accuracy of the value of F is derived from

$$I_F = \pm\left((dF/dA)^2 I_A^{\,2} + (dF/dB)^2 I_B^{\,2} + (dF/dE)^2 I_E^{\,2} + (dF/dS)^2 I_S^{\,2} \right)^{\frac{1}{2}}$$

where,

I_F = error involved in F

d = partial derivative

Evaluating the above expression and repeating the procedure for triangle ADC the total error in the estimation of the area of the cross-section was found to be ± 0.22 square feet or 1.06 per cent.

The accuracy of the calculated quantity of air flowing past the Measuring Station is estimated from the errors relating to each of the factors involved.

MEASUREMENT OF ROADWAY AREAS

A photographic method evolved by Clarke (11) and later improved by Potts (12) was used to determine the cross-sectional area of stations 1, 2 and 3. Briefly the procedure comprised tracing out the periphery of the cross-section by means of a small light source, while a camera with shutter open was focussed on the plane of the station. In order to obtain the scale ratio the light was shone through small holes drilled a known distance apart in a length of wood.

In the initial stages the source of light was the bulb from a miner's cap lamp connected to a 2-volt accumulator. The trace on the photographic plate using this device was too thick, however, so a small torch was constructed which was operated by a dry battery. A conical solid glass lens was attached to the torch and partly shrouded with rubber tubing so that a pin-point source of light could be obtained.

A half-plate camera with focussing screen was used and very fast photographic plates were required. As a result of the size of plate used it was possible to ascertain the area by planimeter from a contact print thereby eliminating the time required for enlarging. Glass plates were used in preference to cut film so that there would be no possibility of distortion.

It was found possible in some cases to get a better estimate of the scale ratio by determining the true distance between easily recognisable points in the outline.

Before proceeding with the measurement of the areas of stations 1, 2 and 3, the reliability of the method was checked and found to be satisfactory by comparing the area of the Measuring Station obtained photographically with the value derived from direct measurements. The error involved in the determination of each station area was estimated as follows.

From twelve planimeter traverses the mean area of each photograph was obtained and the average error evaluated. Linear distances were measured to the nearest 0.01 inch. The calculations involved in checking the reliability and estimating the accuracy of the method are shown below.

Area of Measuring Station

From Plate VIA.

Area enclosed by trace = 4.78 square inches

Now it is known that the length of the diagonal from the top left-hand corner is 81.5 inches. On the photograph this is represented by 3.28 inches.

Therefore the scale is,

$$1 \text{ inch} = \frac{81.5}{12 \times 3.28}$$

$$= 2.07 \text{ feet}$$

Hence

$$\text{Area of Station} = 4.78 \times 2.07^2$$

$$= 20.49 \text{ square feet}$$

Errors involved:-

Planimeter traverses:

$$\text{Average Error} = \pm 0.05$$

Scale: $$1 \text{ inch} = \frac{81.5 \pm 0.25}{12 \times 3.28 \pm 0.02}$$

$$\text{error} = \pm\left((1/12 \times 3.28)^2 0.25^2 + (81.5/12 \times 3.28^2)^2 0.02^2 \right)^{\frac{1}{2}}$$

$$= \pm 0.014$$

Area of Station:

$$\text{Area} = 4.78 \pm 0.05 \times (2.07 \pm 0.014)^2$$

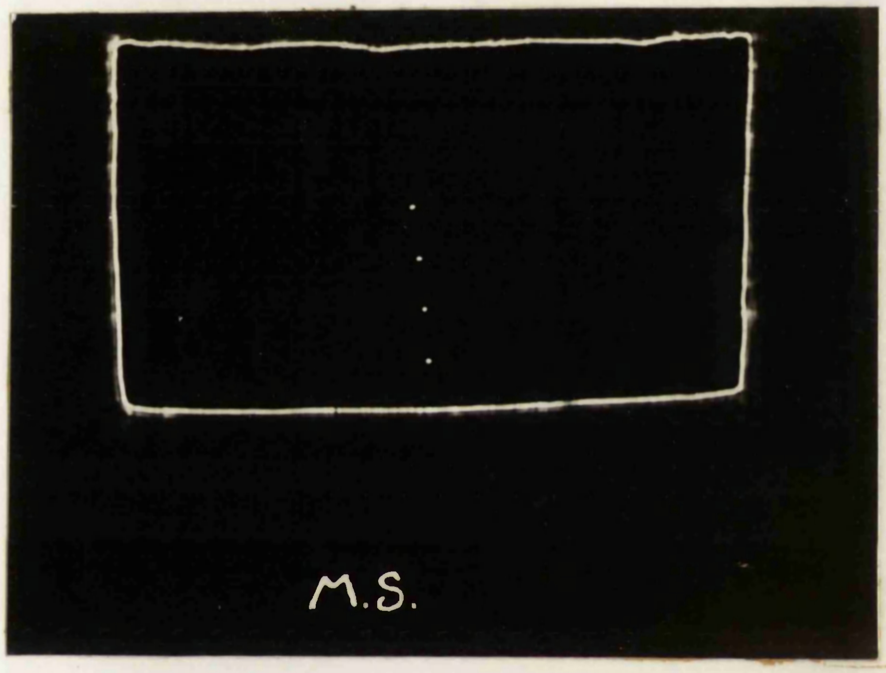


PLATE VIA: Outline Trace - Measuring Station

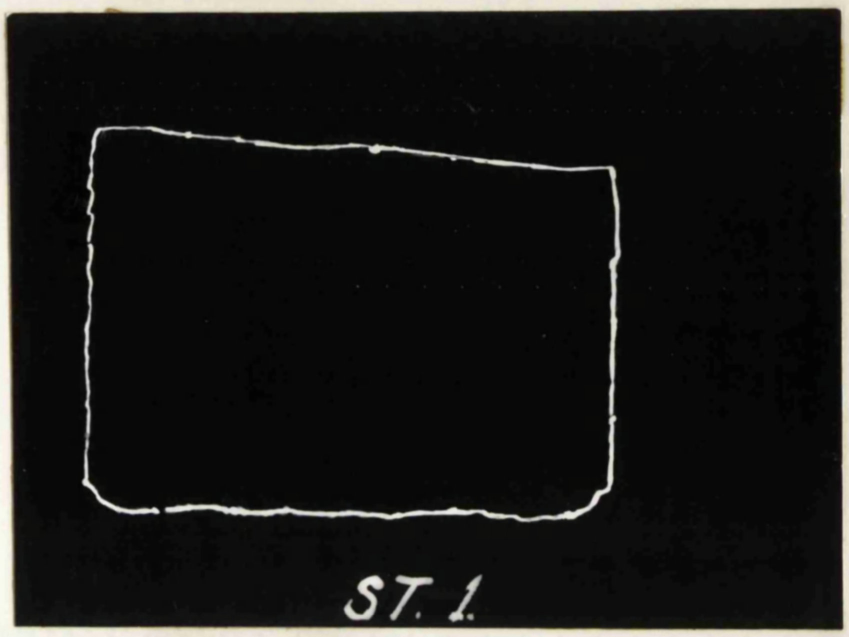


PLATE VIIA: Outline Trace - Station 1

$$\text{error} = \pm\left(2.07^2 \times 0.05^2 + (4.78 \times 2 \times 2.07)^2 0.014^2\right)^{\frac{1}{2}}$$

$$= \pm 0.30$$

$$\text{Percentage error} = \pm \frac{0.30 \times 100}{20.49} = \pm 1.46$$

From the results it will be noted that the photographic method gives the area of the measuring station to within 1% of the value calculated from direct measurements. Therefore this procedure was adjudged completely reliable for the determination of the areas of the remaining stations.

The main factor influencing the accuracy of the method was the error involved in taking measurements from the photograph. Especially with the thicker traces it was difficult to estimate the outside edge of the outline. Allowance was made by assuming such measurements to be correct to ± 0.02 inches.

Area of Station 1

From Plate VIIA:

Area enclosed by trace = 3.89 ± 0.05 square inches

Scale: 1 inch = 2.69 ± 0.025 feet

Therefore

Area of Station = 28.1 ± 0.38 square feet

Percentage error = ± 1.35

Area of Station 2

From Plate VIIIA:

Area enclosed by trace = 10.35 ± 0.05 square inches

Scale: 1 inch = 1.72 ± 0.013 feet

Therefore

PLATE VIIIA: Outline Trace - Station 2

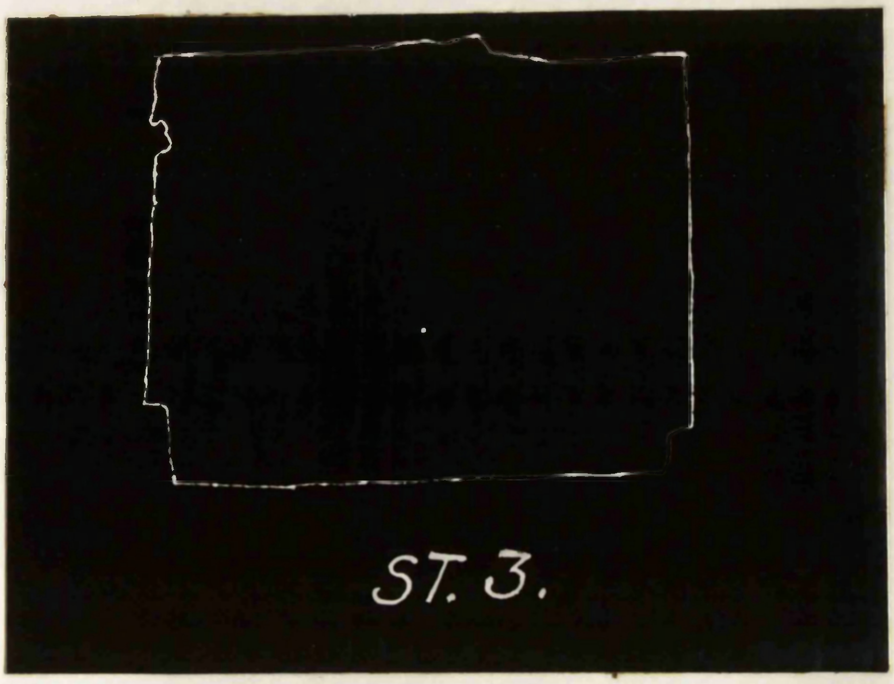


PLATE IXA: Outline Trace - Station 3

Area of Station = 30.4 ± 0.48 square feet

Percentage error = ± 1.57

Area of Station 3

From Plate IX A:

Area enclosed by trace = 4.64 ± 0.05 square inches

Scale: 1 inch = 2.54 ± 0.18 feet

Therefore

Area of Station = 29.9 ± 0.42 square feet

Percentage error = 1.40

CALIBRATION OF ANEMOMETERS BY WHIRLING ARM

Anemometers have been calibrated by a Whirling Arm method for some considerable time, but although this method has been accepted as being accurate by some workers (13), others (14) have disregarded it on account of the swirl set up by the rotating arm. It was therefore necessary to establish whether anemometers could be satisfactorily calibrated in this way. This was accomplished by examining the distribution of the observed values of air velocity in relation to the actual values. For each instrument used the relationship appeared, from a plot of the results, to be linear and this was assumed to be the case. If the best straight line through the experimental points could be obtained then the dispersion of the observed values about this line would confirm or refute the basic assumption of linearity. Furthermore, if the relationship were proved linear, the dispersion of the observed values would also indicate the accuracy of the method.

Theoretically the best straight line through a number of experimental points is the one which makes the sum of the squares, of the perpendicular distances from the observed values to the line, a minimum. However the algebra involved precludes any simple method of deriving the coefficients of such a line. Therefore it was decided to obtain a "regression line" (15) of the observed values.

By taking the regression of x upon y the straight line

$$y = a + bx$$

is obtained, such that the sum of the squares of the distances from the experimental points to the line in the direction parallel to the y-axis is a minimum. If the anemometer readings, being the observed values, are denoted by X and the actual values by Y then the coefficients a and b are obtained from the equations

$$S(Y) - an - bS(X) = 0$$

$$S(YX) - aS(X) - bS(X^2) = 0$$

where

$$S = \text{summation}$$

$$n = \text{number of observations}$$

Let the angle which the line of regression makes with the horizontal be denoted by θ. Then

$$\tan\theta = b$$

If the distance, in the direction of the y-axis, from any observed value to the line of regression is represented by y and the distance perpendicular to the straight line by s, then

$$s = y \cos \theta.$$

The results obtained during the calibration of the two anemometers are shown in Table No.3 and No.4. The regression lines are:

$$\text{Davis-Biram} - \quad y = 66 + 0.944x$$

$$\text{K9534} - \quad y = 35 + 0.985x$$

where the units of x and y are feet per minute.

TABLE No.3

Davis-Biram

Y ft./min.	X ft./min.	YX	X^2	66 + 0.944X = Y'	Y' - Y	$\frac{(Y'-Y)\cos\theta \times 100}{X}$
252	188	47376	35344	243	-9	-3.45
262	204	53448	41616	258	-4	-1.45
273	212	57876	44944	266	-7	-2.35
288	230	66240	52900	283	-5	-1.05
298	238	70924	56644	290	-8	-2.50
321	251	80571	63001	303	-18	-5.20
396	358	141768	128164	414	+18	+3.65
425	393	167025	154449	436	+11	+2.05
461	427	196847	182329	469	+ 8	+1.40
494	467	228728	214369	506	+12	+1.80
502	477	234434	218089	516	+14	+2.00
543	513	279645	265225	550	+ 7	+1.00
634	613	388642	375769	654	+20	+2.35
648	623	403704	388129	653	+ 5	+0.55
666	644	428904	414736	672	+ 6	+0.70
681	647	460607	418609	677	- 4	-0.45
716	702	432432	492804	727	+11	+1.15
830	818	678940	669124	837	+ 7	+0.60
847	842	713174	708964	860	+13	+1.15
865	856	740440	732736	873	+ 8	+0.70
903	897	809991	804609	912	+ 9	+0.75
989	977	966253	954529	987	- 2	-0.15
1065	1068	1137420	1140624	1073	+ 8	+0.55
1153	1150	1325950	1322500	1150	- 3	-0.15
1221	1204	1470084	1449616	1203	-18	-1.10
1243	1239	1540077	1535121	1234	- 9	-0.55
1258	1251	1573758	1565001	1246	-12	-0.70
1274	1263	1609062	1595169	1257	-17	-1.00
1298	1283	1665334	1653796	1276	-22	-1.25
1339	1316	1662124	1731856	1306	-33	-1.85

TABLE No.4

K.9534

Y ft./min.	X ft./min.	YX	X^2	35 + 0.985X = Y'	Y' - Y	(Y'-Y) cos θ x 100 / X
303	273	82719	74529	304	+ 1	+0.26
314	282	88548	79524	313	- 1	-0.25
318	288	91584	82944	319	+ 1	+0.25
328	297	97416	88209	327	- 1	-0.24
335	302	101170	91204	333	- 2	-0.47
340	305	103700	93025	335	- 5	-1.17
371	336	124656	112896	366	- 5	-1.06
384	353	135552	124609	373	-11	-2.22
384	351	134784	123201	381	- 3	-0.61
399	362	144438	131044	392	- 7	-1.38
546	537	293202	288369	565	+19	+2.52
577	550	317350	302500	576	- 1	-0.13
670	646	432820	417316	671	+ 1	+0.11
692	668	462256	446224	693	+ 1	+0.11
716	697	499052	485809	721	+ 5	+0.51
741	716	530556	512656	740	- 1	-0.10
741	720	533520	518400	744	+ 3	+0.30
783	759	594297	576081	782	- 1	-0.09
798	771	618450	600625	794	- 4	-0.37
830	816	678940	665856	839	+ 9	+0.79
830	806	677280	669124	829	- 1	-0.09
830	818	668980	649636	841	+11	+0.96
847	823	697081	677329	845	- 2	-0.17
847	825	698775	680625	847	0	0
903	879	793737	772641	900	- 3	-0.24
924	907	838068	822649	928	+ 4	+0.32
989	971	960319	942841	992	+ 3	+0.22
1013	986	998818	972196	1007	- 6	-0.43
1038	1020	1058760	1040400	1040	+ 2	+0.14
1038	1011	1049418	1022121	1031	- 7	-0.49

Discussion of Results

The criterion determining whether a suitable relationship between the actual and observed values has been found is the dispersion of the experimental points about the derived line. For both anemometers the deviation of the observed values in the direction perpendicular to the line of regression was calculated. A good approximation to the true significance of each deviation was obtained by expressing the perpendicular distance as a percentage of the observed value.

Therefore from the final columns in Tables 3 and 4 it may be concluded that:-

(a) The basic assumption that the relationship between the actual and observed values was linear has been comfirmed. In the case of the Davis-Biram the dispersion of the experimental points suggests a slight curvature. However the average deviation of 1.45% is comparable with the accuracies already obtained in this section of the investigations. For anemometer K.9534 the average deviation is only 0.53% which is probably due to the true relationship being nearly linear.

(b) The regression of x upon y gave a suitable straight line. Therefore this procedure may be recommended where it is desired to obtain the best straight line through a number of points. However, it must be remembered that the true standard of judgment is the sum of the squares of the perpendicular distances from the observed values to the line; and in some instances a closer approach to the minimum condition may be obtained by taking the regression of y upon x.

(c) For the range of air velocities considered the calibration of anemometers by whirling arm is probably as reliable as any other method. The swirl induced by the rotation of the arm does not appear to have had any appreciable effect upon either of the anemometers. It is almost certain that any effects due to swirl would have been irregular and thus easily discernable.

ANEMOMETER EXPERIMENTS

In these experiments different methods of traversing with an anemometer were investigated. The methods adopted may be summarised thus:-

Station 1: Davis-Biram fixed to hand-grip : operator in station cross-section.

K.9534 attached to the end of 3-feet rod : operator downstream.

Station 2: As for Station 1.

Station 3: Davis-Biram supported on 3-feet rod: operator in manhole at the side of the station.

K.9534 as for Davis-Biram.

From the readings obtained at Station 3 it was hoped to establish the basic accuracy of the anemometer traversing method in the determination of mean velocity. Then the effect of the position of the operator and the influence of disturbed flow conditions could be examined.

A suitable range of air velocities was examined. For each quantity of air the mean velocities at Stations 1, 2 and 3 were found by the methods detailed above; the "centre velocity" at the measuring station was also determined and checked periodically during each experiment.

Two operators, weighing 11 and $12\frac{1}{2}$ stones respectively, performed the traverses. The size of the operator did not exert any appreciable effect, however, so the mean of the figures obtained by both investigators for each traverse was calculated. For each air speed the mean quantity flowing past the measuring station was evaluated and the estimated true mean velocity at Stations 1, 2 and 3 determined. The accuracy of this estimation was derived from the errors involved in the measurement of the "centre velocity", centre constant and the relevant area.

The results are shown in Table No.5, No.6 and No.7. The difference between the mean velocity obtained by anemometer traverse and the estimated true mean velocity has been expressed as a percentage of the latter.

TABLE No.5

Station 1.

Test Number	Mean Quantity ft.^{3}/min.	True Mean Velocity: Estimated Value ft./min.	True Mean Velocity: Percentage Accuracy	Mean Velocity by Anemometer Traverse: Davis-Biram ft./min.	Mean Velocity by Anemometer Traverse: K.9534 ft./min.	Percentage Difference: Davis-Biram	Percentage Difference: K.9534
1	8300	295	± 2.5	390	305	-32.2	-3.4
2	10080	358	± 2.5	450	352	-25.7	+1.7
3	11800	420	± 2.3	513	413	-22.2	+1.7
4	13080	465	± 2.2	590	467	-26.9	-1.8
5	13830	492	± 2.2	596	483	-21.1	+1.8
6	16510	588	± 2.1	747	597	-10.0	-1.5
7	17010	605	± 2.1	763	599	- 9.6	+1.0
8	19270	685	± 2.1	865	687	-26.3	-0.3

TABLE No.6

Station 2.

Test Number	Mean Quantity ft.^{3}/min.	True Mean Velocity: Estimated Value ft./min.	True Mean Velocity: Percentage Accuracy	Mean Velocity by Anemometer Traverse: Davis-Biram ft./min.	Mean Velocity by Anemometer Traverse: K.9534	Percentage Difference: Davis-Biram	Percentage Difference: K.9534
1	8300	273	± 2.7	335	270	-22.7	+1.1
2	10080	332	± 2.5	381	311	-14.8	+6.3
3	11800	388	± 2.5	506	362	-30.4	+6.7
4	13080	430	± 2.4	513	415	-19.3	+3.5
5	13830	455	± 2.4	514	425	-12.8	+6.6
6	16510	544	± 2.3	670	537	-23.2	+1.3
7	17010	560	± 2.3	673	565	-20.2	-0.9
8	19270	634	± 2.3	775	626	-22.2	+1.3

TABLE No.7

Station 3.

Test Number	Mean Quantity ft.^{3}/min.	True Mean Velocity Estimated Value ft./min.	True Mean Velocity Percentage Accuracy	Mean Velocity by Anemometer Traverse Davis-Biram ft./min.	Mean Velocity by Anemometer Traverse K.9534 ft./min.	Percentage Difference Davis-Biram	Percentage Difference K.9534
1	8300	278	± 2.6	272	270	+ 2.2	+2.9
2	10080	337	± 2.4	323	326	+ 4.1	+3.3
3	11800	395	± 2.4	410	404	- 3.8	-2.3
4	13080	437	± 2.3	415	428	+ 5.0	+2.2
6	16510	552	± 2.2	558	575	- 1.1	-4.2
7	17010	570	± 2.1	560	586	+ 1.8	-2.8
8	19270	644	± 2.1	656	671	- 1.9	-4.2

Discussion of Results

From the tabulated results it may be noted that:-

(a) The presence of the operator in the cross-section being traversed causes a considerable increase in the mean velocity. The effect is very irregular, however, probably as a result of the operator having to move across the area being traversed. His position will affect the velocity distribution and it will be well-nigh impossible for him to reproduce the same movements during each traverse.

(b) The best results were obtained at Station 1 when the operator was in the downstream position. The distance from the anemometer to the operator was about four feet and this would appear sufficient to obviate any effect due to his presence.

(c) The results obtained at Station 2 with the operator in the downstrwam position indicate that the junction 12 feet upstream (see Fig. A1) is affecting the flow conditions. However the effect becomes less pronounced as the mean velocity increases.

(d) The accuracy of the results gained with the two anemometers at Station 3 is not outstanding. By this experiment it really was hoped to establish the reliability of the traversing method of determining mean velocity. Unfortunately the manhole sheltering the operator and his unavoidable presence in the cross-section for a short time appear to have produced an adverse effect. Due to the operator's movements the blocking-off of the manhole was impracticable.

CONCLUSIONS

From the results obtained during this series of investigations, it may be concluded that in the determination of mean velocity by anemometer traverse in underground roadways whose areas are comparable with those used in these experiments:-

(1) The operator should never be present in the cross-section being traversed. Even if the area of the operator's profile could be estimated fairly well the irregularity of the results shown in Tables No.5 and No.6 preclude any reliance being put on this method.

(2) Satisfactory results will be obtained if the operator is at least four feet downstream from the anemometer. No appreciable effect is produced by having the operating switch located on the instrument.

(3) The section being traversed should be removed as far as possible from regions likely to upset the air flow such as bends, junctions, manholes, etc.

The investigations just completed serve to illustrate the difficulties involved in establishing the fundamental accuracy of the anemometer traversing method. The technique used to determine the mean quantity is the generally accepted standard procedure. The error involved in these determinations is appreciable although it is felt that in every case the estimate of the accuracy is conservative. From the results, therefore, it would appear justifiable to claim the anemometer traverse method of measuring mean velocity to have been proved correct to at least ± 2.5%.

Most of the circumstances affecting the measurement of air velocity by anemometer can be examined, as some have already been examined (3) without determining the true air speed. However a better estimate of the basic accuracy of this method of traversing cannot be obtained unless the true mean velocity can be much more precisely determined. This would only appear possible by using much bigger areas. It is rather improbable that large areas of simple outline where the quantity passing could be finely controlled are to be found in practice. Therefore it would appear impractical to try to determine the basic accuracy of the traverse method beyond ±2.5%.

SECTION B

REGULATOR PRESSURE LOSSES

INTRODUCTION

In Ventilation Engineering, as applied to mining, there has always been the need for some contrivance whereby the resistance of certain parts of the air circuit can be readily increased. The device used since the rudiments of ventilation theory were first appreciated has been the regulator in some form or other. However, although the regulator has found widespread application in the control of the distribution of air in mines, only in recent years has any attempt been made to study the principles involved in the flow of air through it. It may be argued, of course, that a regulator is after all just an orifice, the theory of which has been extensively investigated; but there are important differences which must be kept in mind. A regulator, for example, is not usually circular but rectangular and its shape may vary considerably. Furthermore it is not always placed centrally in the airway the profile of which itself may be quite irregular. In any case it is the loss in pressure produced by the regulator that is of most concern and in orifice investigations this aspect is not usually considered.

As long ago as 1895 Halbaum (16) examined the flow of air through a regulator. He assumed that the "vena contracta" effect was independent of the ratio of the airway area to the regulator area and also, apparently unknown to himself, that there was no recovery of velocity energy downstream from the regulator. In consequence it was possible to derive a simple relationship between the resistance produced by a regulator and its cross-sectional area, which has been accepted right up to the present day.

During some recent Ventilation Survey investigations undertaken by the author (17) it was discovered that using Halbaum's formula in an attempt to design a regulator to increase the resistance of a length of airway by a desired amount resulted in an area greater than that of the roadway itself. It was apparent that Halbaum's formula had a limited application and further work on regulation was necessary.

Evidently Weeks (18) in 1928 was the first authority to expose the deficiencies in the generally accepted expression although his opinions have been, in this country at any rate, for the most part ignored. He assumed that the total loss of energy caused by a regulator could be attributed to the shock loss which results when a quickly moving stream of fluid enters a slowly moving stream. The formula for this loss is of course well known (19) but it should be remembered that it is derived with reference to the flow of fluid from a region of small area into a region of large area and consequently when applied to regulators really only accounts for the losses which occur on the downstream side of the opening.

Weeks experimented with a series of square-profile regulators located centrally in a square duct, and having ratios of duct area to regulator area ranging from 1 to 10. Using the shock loss formula he determined the Coefficient of Contraction (contracted area/regulator area) of each opening for two quantities of air. In a written discussion on Week's paper (18) it was stated by McElroy that, working from the shock loss formula, results he had obtained (20) using four circular openings and presumably only one air quantity for each gave similar values for the Coefficient of Contraction. On the strength of this correlation he developed a complicated nomograph for the determination of the size of a regulator to produce a desired pressure drop when inserted in an airway of known area and passing a given quantity. The tests carried out by both these workers was of an extremely limited nature and no indication of the values of Reynold's Number is given. Entry losses to and within the regulator were neglected and the effect of the thickness of the regulator upon the "vena contracta" completely ignored. More exact work on orifices (21) has shown that the values of Coefficient of Contraction they obtained were about 6 - 7% too high.

The object of the following experiments was to attempt first of all, by applying the Buckingham "Pi" Theorem, a new approach to the loss in energy produced by the flow of air through a regulator. The theory was used as a guide to the factors most likely to affect the pressure loss and the effects of these were then investigated. The tests were carried out in a small wind-tunnel where values of Reynolds Number up to 150,000 could be obtained, which it was hoped would allow the results to be correlated to actual mining conditions. For this purpose a nomographical method has been developed from which can be determined the area of a regulator to produce a desired pressure loss when inserted in an airway of a given cross-section, if a known quantity of air is flowing.

This section of the thesis consists firstly of a description of the preliminary tests carried out to familiarise the author with the flow conditions encountered in the wind tunnels available, with a view to establishing a suitable technique for the determination of mean velocity. Then follow details of the investigations into the factors affecting the pressure drop in regulators. Lastly are recorded the conclusions derived from the results of the various experiments.

EQUIPMENT

Wind Tunnels

The wind-tunnels (see Fig. B1 and B2) are constructed of 5 feet long rectangular detachable mild-steel sections of internal dimensions 12 inches by 16 inches. A contracting length precedes the 12 inches square measuring section which is followed by a similarly shaped diffuser. The 5-feet sections have slotted flanges at each end and can be rapidly fitted together by means of hinged bolts and wing-nuts. Rubber gaskets provide an airtight seal at every joint. The entry to both fans is circular and in each case the tunnels are fixed to the fan-units by a transition-section.

At each measuring station an airtight gland with free lateral movement allows a pitot-tube to be located at any point in the cross-section. Rigid pitot-tube tappings are provided in several of the 5 feet sections. A flared inlet attached to each duct minimises the entry losses.

Fans and Motors

(a) Keith Blackman Unit: A 2.6 B.H.P. shunt motor having a maximum speed of 1500 R.P.M. is coupled directly to a Keith-Blackman centrifugal fan. The motor is of the torque-reaction type which enables the power output from the shaft to be determined by means of a balance arm.

(b) Aeroto Unit: An Aeroto axial flow fan is driven directly by a 2 B.H.P. compound motor with a maximum speed of 2000 R.P.M.

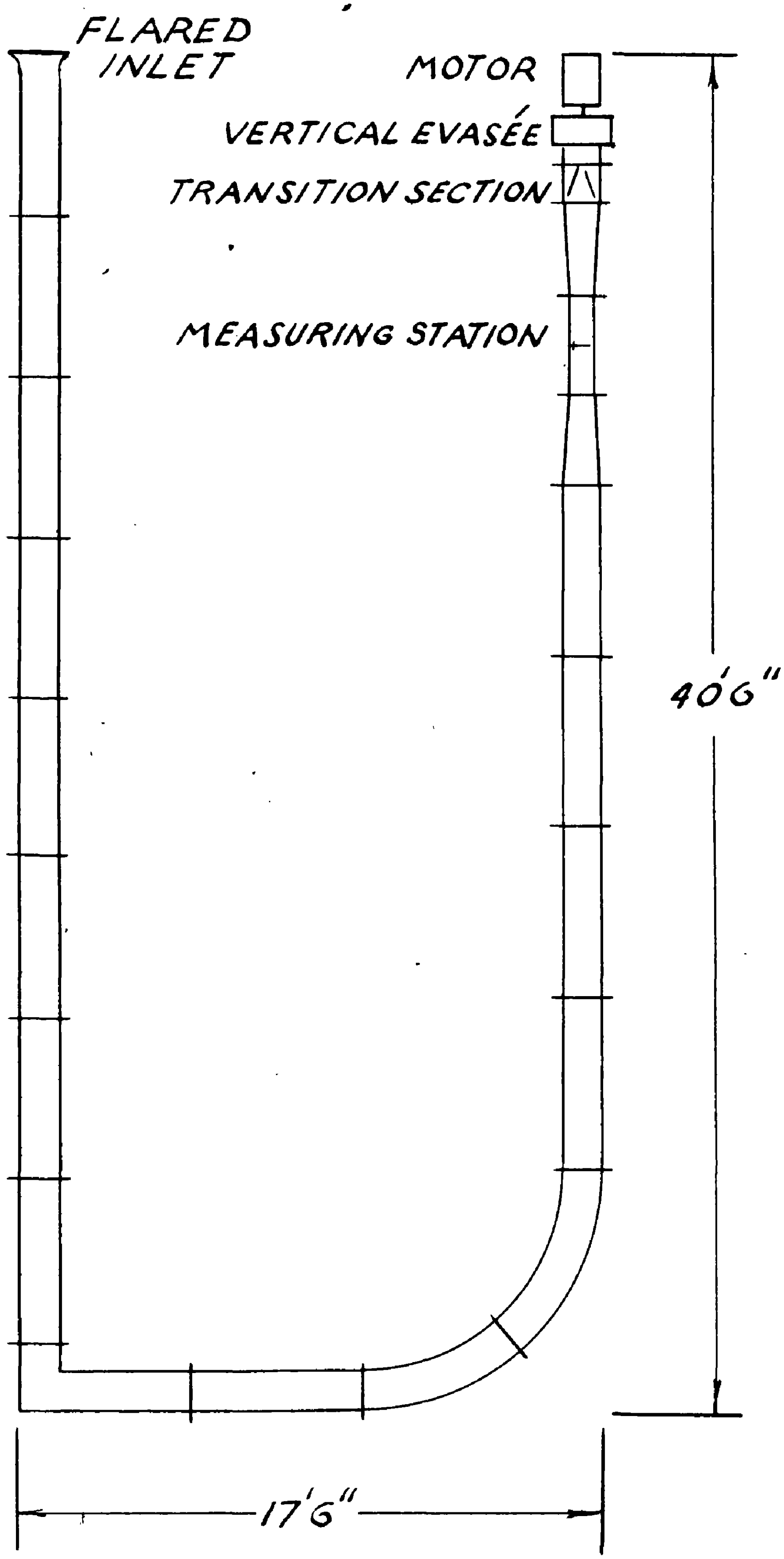
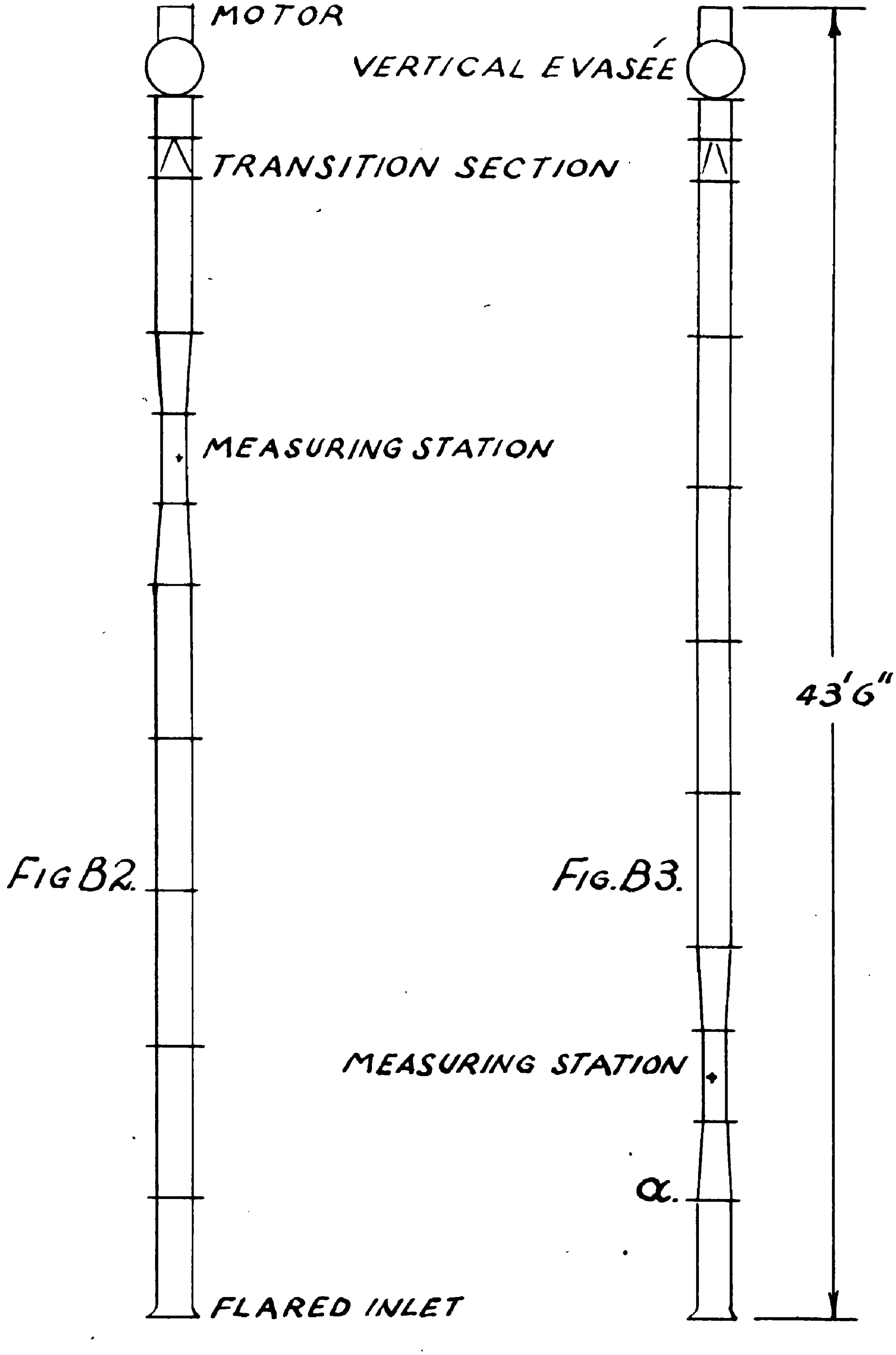


FIG. B1.

KEITH BLACKMAN WIND TUNNEL.



AEROTO WIND TUNNEL.

Both fans are capable of producing a maximum total head of just over 2.5 inches water gauge.

(c) <u>Speed Control Unit</u>: The speed of each motor can be varied from zero to maximum by a Single-Phase Thyratron Speed Control Unit. This apparatus is designed to give full-range speed control of a separately excited direct current, shunt or compound motor, with a constant torque characteristic. Any desired speed within the range of the motor may be selected. With constant torque a speed regulation of not more than ±0.1% of maximum is obtained at any load or speed. Due to the thermal characteristics of the gas discharge tubes a speed drift is experienced but this is extremely slow and amounts to less than 1% of the setting. A control panel contains "start" and "stop" push-buttons, the speed control knob, meters indicating armature voltage and current, and a tachometer activated by a small generator mounted on the motor shaft. A reversing switch enables the direction of rotation of the Aeroto fan to be altered.

<u>Instruments</u>

<u>Askania Minimeter</u>: As described on page (5).

<u>Casella Vernier Reading Manometer</u> (see Plate IB): This instrument is essentially a U-tube in which the water level in one of the limbs can be accurately observed. The reference level is obtained when a steel point and its image, totally reflected internally by the water, appear to touch. The range of the instrument is 0 - 50

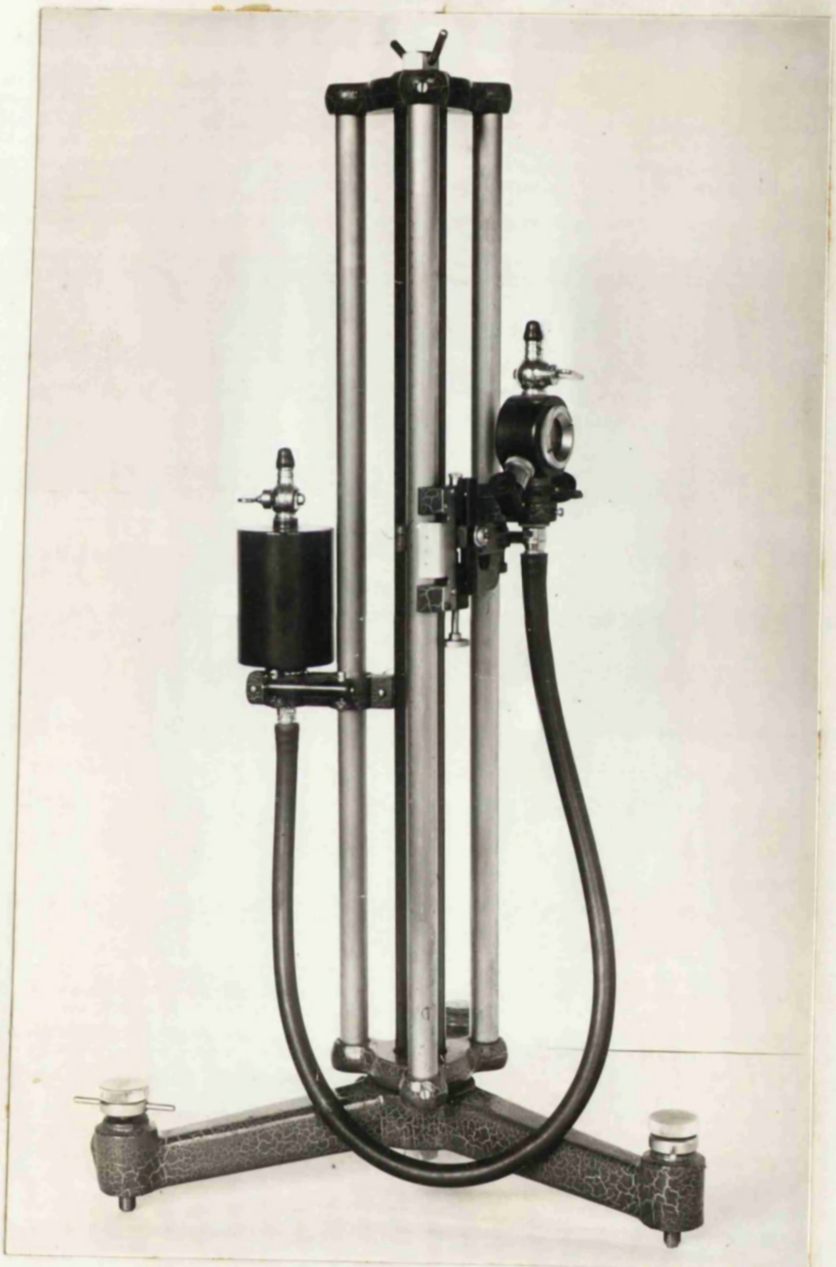


PLATE IB: Casella Manometer

centimetres and a vernier enables the height of water column to be measured to 0.005 centimetres.

Pitot-tubes: Pitot-static tubes of N.P.L. design were used throughout the investigations.

Psychrometer: As described on page (5).

Barometer: A mine-type barometer reading to 0.01 inches of mercury was used.

PRELIMINARY EXPERIMENTS

These tests were carried out in the Aeroto wind-tunnel to investigate ways of reducing the effect of fluctuations in total and velocity head, so that accurate measurements could be made. Since the experiments were rather elementary in nature only a brief description of the procedure and theory involved is given in each case.

Damping Effect of Capillary Tube

In this experiment the damping effects of different lengths of 1/16" bore capillary tube (22) were compared. A pitot-static tube was located at the centre of the measuring station cross-section, and a piece of capillary tube inserted in the total head lead to the Askania. The mean air velocity at the station was kept constant at approximately 40 feet per second.

For each length of tube ten readings of total, static and velocity head were obtained. The velocity head was also calculated from the algebraic difference of the total and static heads. The Average Error of the Arithmetic Mean (see page (10)) of each series of observations allowed the damping effects of the various lengths to be compared.

Accuracy of Mean Velocity at Measuring Station

A standard method (7) was used to determine the mean velocity. The measuring station was divided as shown in Fig. B4 and with the fan running at constant speed the velocity pressure at the centre of each of the sixteen sub-divisions was obtained. At each position four readings were taken and for the reasons given on page (10) the Average Error of the Arithmetic Mean calculated. Applying the Briggs' expressions (10) the accuracy of the mean velocity was found.

The mean velocity Vm was obtained (see page (8)) from the formula:

$$Vm = (k/n)(h_1^{\frac{1}{2}} + h_2^{\frac{1}{2}} + \ldots\ldots\ldots h_n^{\frac{1}{2}})$$

If,

Ea = average error of the arithmetic mean (h) of the four readings taken at any position

and,

Ea' = the error in the square root ($h^{\frac{1}{2}}$) of the mean value

then,

$$Ea' = \pm Ea/2h^{\frac{1}{2}}$$

The expression for the mean velocity of flow thus becomes

$$Vm = (k/n)\left((h_1^{\frac{1}{2}} \pm Ea'_1) + (h_2^{\frac{1}{2}} \pm Ea'_2) + \ldots(h_n^{\frac{1}{2}} \pm Ea'_n)\right)$$

$$= (k/n)(h_t^{\frac{1}{2}} \pm Ea'_t)$$

where,

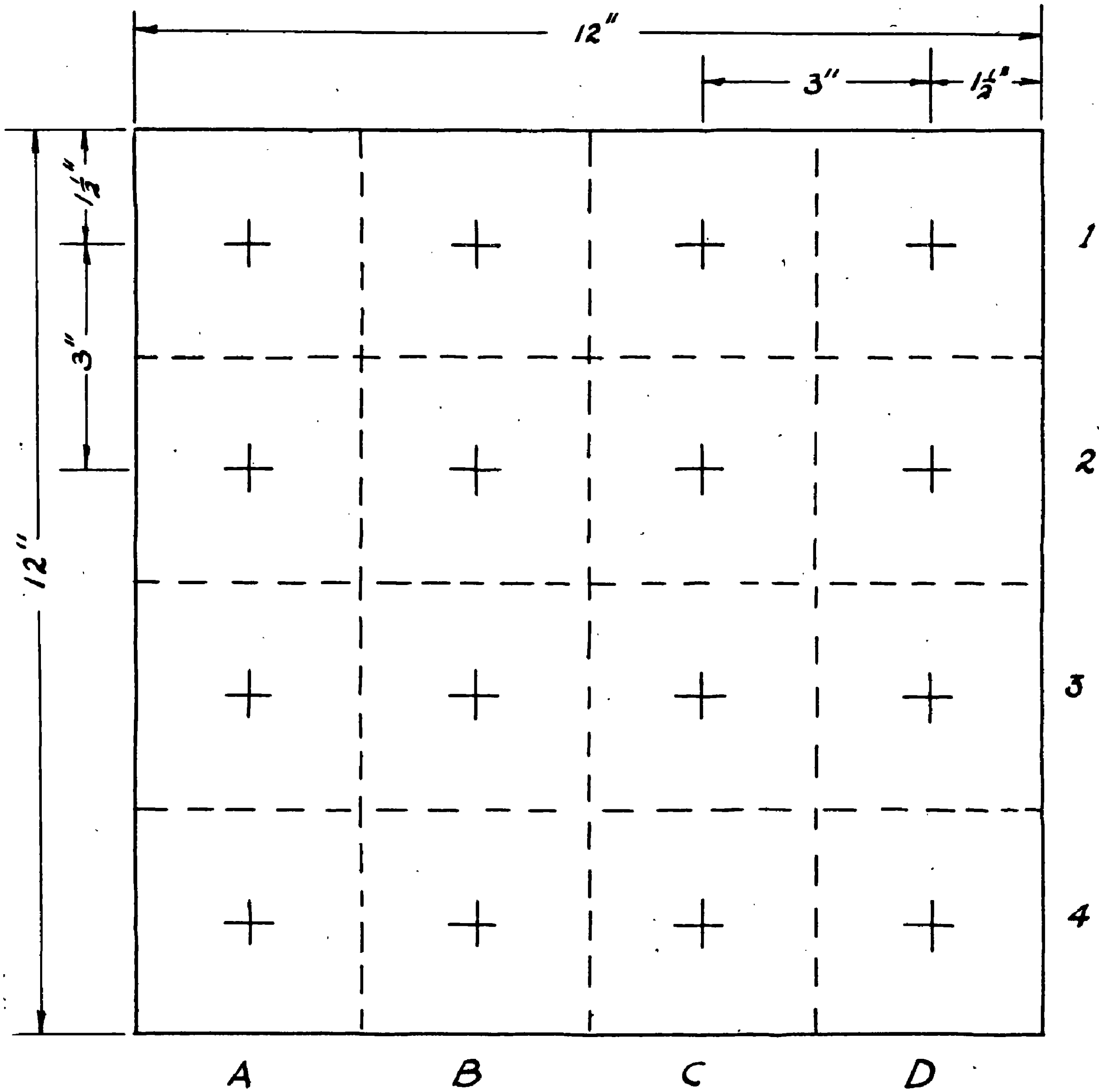


FIG. B4.

SUBDIVISION OF MEASURING STATION.

$$h_t^{\frac{1}{2}} = (h_1^{\frac{1}{2}} + h_2^{\frac{1}{2}} + \ldots\ldots\ldots h_n^{\frac{1}{2}})$$

and,

$$Ea'_t = \pm (Ea'^2_1 + Ea'^2_2 + \ldots\ldots Ea'^2_n)^{\frac{1}{2}}$$

Wind Tunnel Modifications

With the fan speed kept constant the velocity fluctuations at different points along the duct were compared. It was hoped to find whether possible inlet disturbances at the fan impellor affected the flow conditions and also if the flared inlet produced a stable region near the entrance to the wind tunnel. The results obtained were inconclusive, however, so the ducting was re-arranged, (see Fig. B3), in the form of an N.P.L. open circuit wind tunnel (23). The effects of flow straighteners and damping screens on the flow conditions, along with the pressure losses produced by these devices, were then investigated.

Flow Straighteners: These devices remove swirl from the air and are normally made of paper or metal honeycombing. They are placed with their axes parallel to the direction of flow, usually a short distance from the flared inlet where the radial component of inflow has become small. They tend to perpetuate non-uniform velocity distributions. The Pressure Drop Coefficient, defined as the ratio of the pressure drop through the straightener to the approach dynamic head, is normally small.

Damping Screens: From extensive experiments (24, 25, 26) the following points of interest emerge:-

(a) Because a screen presents greater resistance to flow where the velocity is high, the asymmetry of the velocity distribution is reduced.

(b) The turbulence of an air-stream may be moderated or increased by means of a suitable screen. Gauze wire diameter and mesh porosity are the controlling factors. The effects are usually localised.

Since the pressure loss produced by a close-mesh screen may be quite appreciable the variation of the loss with increase in approach velocity was investigated. A method developed by Annand (27) allows the Pressure Drop Coefficient (k) at any velocity of approach to be calculated from the gauze porosity and Reynolds Number (Rn) referred to the mesh wire diameter. The expression given is:

$$k = a(1 - B^2)/B^2$$

where,

a = coefficient depending on Rn

B = porosity of the gauze.

The porosity is obtained from the area of the holes projected orthogonally on to a parallel plane (see Fig. B5), viz. -

$$B = (p_1 - d)(p_2 - d)/p_1 p_2$$

where,

$p_1 p_2$ = pitches in the two directions of weave

d = mean wire diameter.

For a square mesh and a dry air density of 0.0764 pounds per cubic foot (30 inches mercury, 60° F):

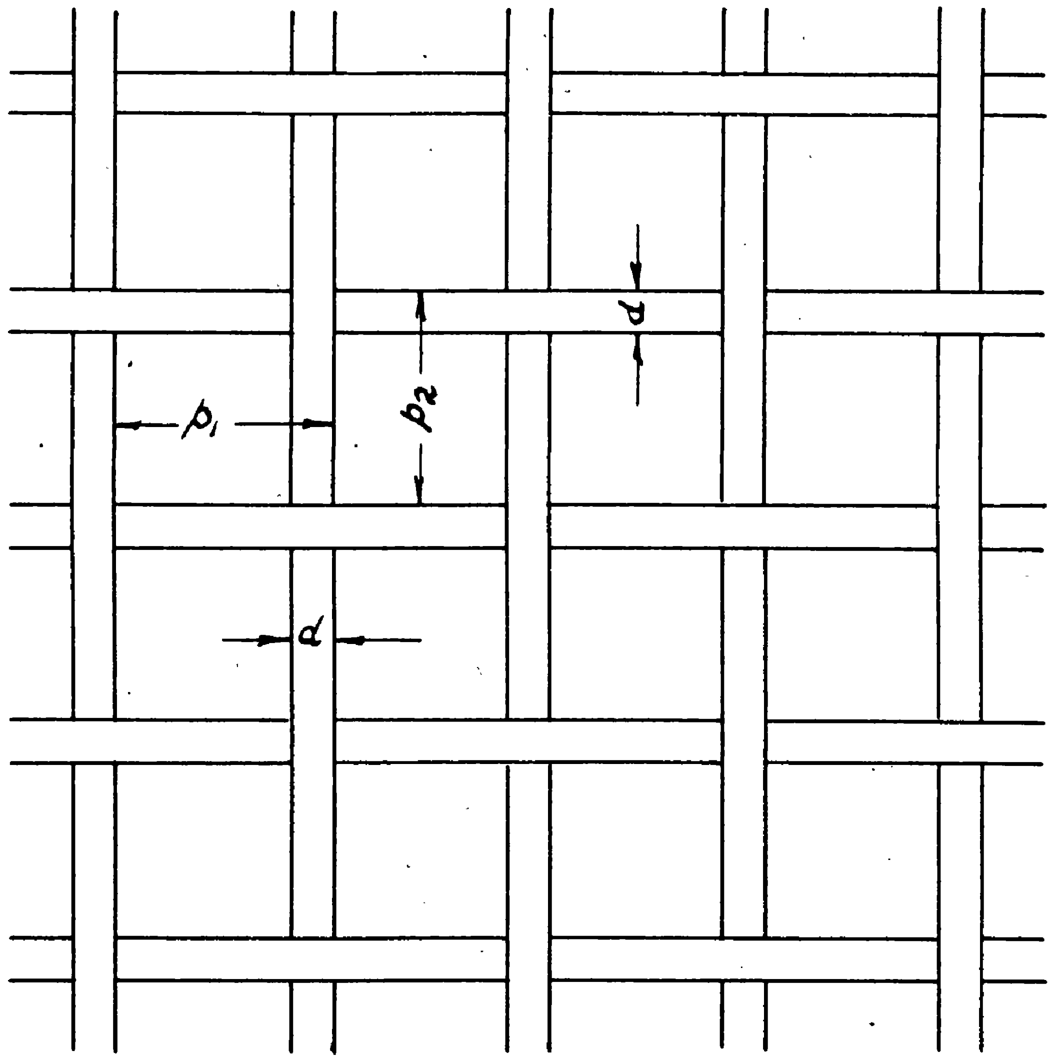


FIG. B 5.

MESH DIMENSIONS.

$$B = (1 - d/p)^2$$

and,

$$h = aV^2(1 - B^2)/4383B^2$$

where,

h = pressure drop through mesh in inches water gauge

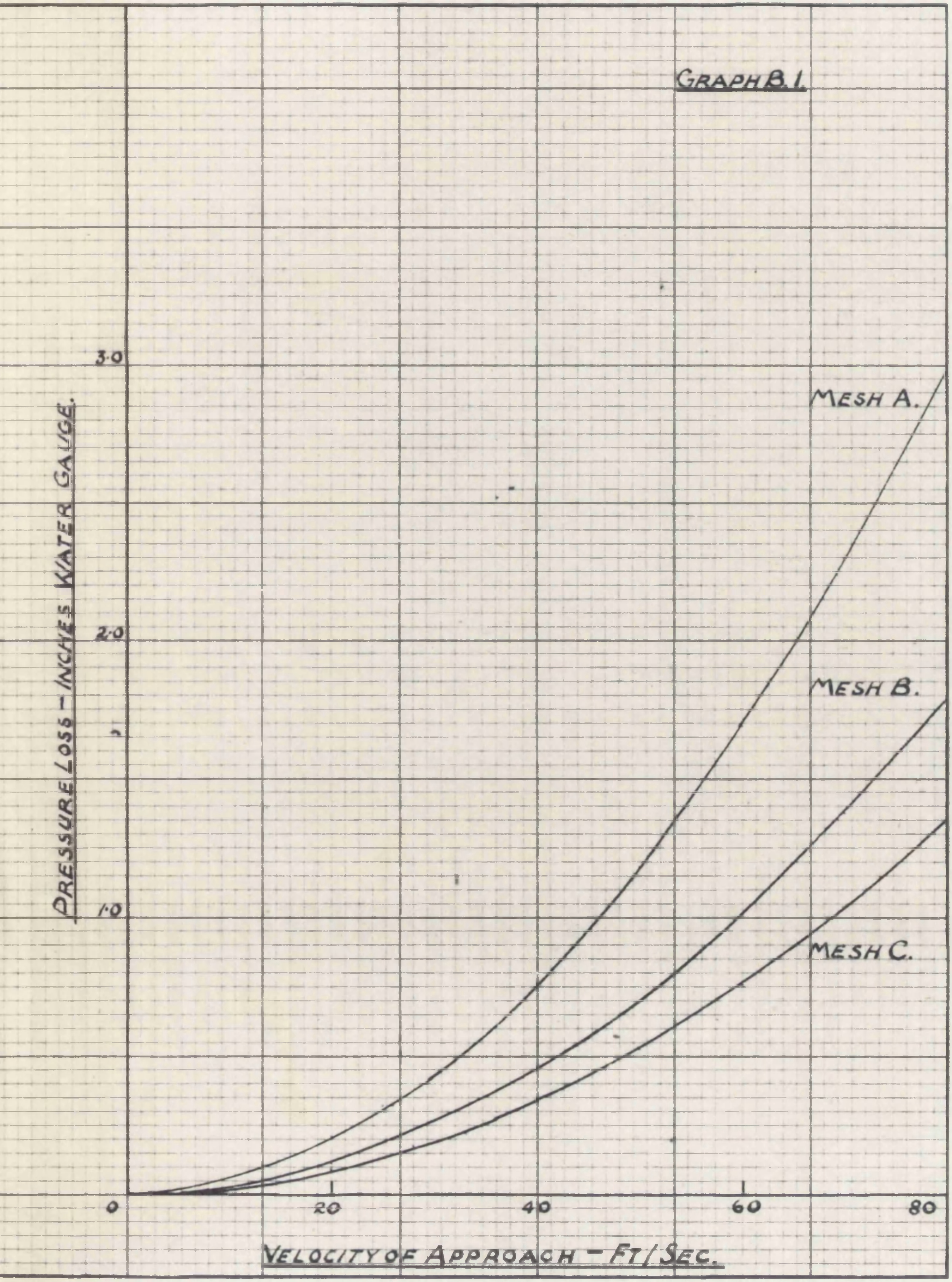
V = mean velocity of approach in feet per second

TABLE No.1

Mesh	Wires per inch	Wire Diameter d in.	Pitch p in.	$(1 - \frac{d}{p})^2$ = B
A	20	0.016	1/20	0.463
B	14	0.018	1/14	0.560
C	8	0.027	1/8	0.614

Values of "a" were obtained from a nomograph (27) and Graph B1 constructed. These curves were derived to enable a suitable gauze to be selected which would improve flow conditions in this or future wind-tunnel arrangements without causing too great a pressure loss.

The mean velocity at the measuring station (see Fig. B3) was adjusted to approximately 40 feet per second. Fluctuations made readings of velocity pressure by Askania Minimeter impossible. An expanded metal flow-straightener was inserted at the flared inlet but produced too great a throttling effect and had to be removed. Each mesh, stretched as tightly as possible for maximum efficiency (28),


GRAPH B.1.
MESH A.
MESH B.
MESH C.
PRESSURE LOSS - INCHES WATER GAUGE.
3·0
2·0
1·0
0
20
40
60
80
VELOCITY OF APPROACH - FT/SEC.

was in turn located at the flared inlet and then at joint a.

CONCLUSIONS

The following is a summary of the more important observations made from these preliminary experiments.

(a) The insertion of a length of capillary tube in the total-head connection from a pitot-static tube reduces the fluctuations in the level of the liquid contained in a manometer of the Askania type. The longer the tube the better is the effect. Too severe damping induces a sluggish response to deliberate alterations in velocity, however, and it must be appreciated that if the air-flow pulsations are large then the "damped" reading may be grossly in error (29).

(b) At any fan speed the electronic equipment maintains a constant flow of air in the duct. Fluctuations in air velocity are due to eddying which is intensified with increase in air speed. Small changes in the fan speed are possible but the rotor inertia will limit the rapidity with which these occur. The periodicity of the changes being long enough to be detected by pitot tube precludes the effects being attributed to true turbulence (30).

(c) The accuracy of the traverse method of estimating mean velocity was found to be completely satisfactory. However the time required makes the method impracticable where a large number of traverses have to be performed.

(d) Screens were of little use in damping out eddies in the air flow. Their other properties (24), however will probably be utilised in future work.

(e) Instruments employing the same principle as the Askania Minimeter to determine the reference level are totally unsuitable for the measurement of fluctuating pressures. Estimation of the mean position from the movements of the needle and its image is impossible.

THE BUCKINGHAM "Pi" THEOREM

The "Pi" theorem deduced by Buckingham (31) was applied to the flow of air through a plate orifice in order to determine the phenomena affecting the pressure loss. In this way the factors requiring investigation were indicated.

The complete proof is contained in the reference and only a brief summary will be given here. The development of the argument is as follows:-

1. If a number of physical quantities of different kinds, Q_1, Q_2 ..... Q_n are sufficient to describe absolutely a physical phenomenon, then the value of any one of them is completely determined when the values of the others are given. This mutual dependence may be stated symbolically in the physical equation

$$f_1 (Q_1, Q_2 \ldots\ldots Q_n) = 0$$

where,

$$f_1 = \text{function.}$$

2. To measure n kinds of quantity we require n kinds of units but these need not all be adopted arbitrarily because they can generally be defined in terms of some smaller number of fundamental units, m say (e.g. force, length, time. etc.). From theoretical reasoning Buckingham deduced that any complete physical equation which describes a relation subsisting among quantities of n different kinds, of which m are independent and not derivable from one another, is reducible to the form

$$f_2(I_1, I_2 \ldots\ldots I_{n-m}, r_1, r_2 \ldots\ldots\ldots)$$

The terms I_1, I_2, etc. represent all the independent dimensionless products that can be formed from the quantities involved. If several quantities of any one kind are involved in the relationship they may all be specified by the value of any one and the ratios r_1, r_2 etc. of the others to that one. Dimensional considerations do not reveal the manner in which these dimensionless ratios appear in the equation describing the relationship, but their possible influence must be indicated. This is best done in an entirely general way by introducing them as additional independent arguments of the unknown function f_2.

3. The values I_1, I_2. .....I_{n-m} are determinable from dimensional equations of the form

$$I_1 = (Q_1^{\,a} \; Q_2^{\,b} \ldots\ldots Q_m^{\,c} \; S_1) = \text{modulus } 1$$

where Q_1, Q_2, ...Q_m are a set, one at least always being present of physical quantities which are independent of each other. The term S_1 is one of the remaining m - n quantities. Indices a, b, ...c are determined by dimensional analysis.

Application to Flow through Regulators

The pressure drop P produced by air passing through a thin regulator inserted in a uniform duct is assumed to depend on:-

A = the cross-sectional area of the regulator

d = the aerodynamical diameter of the regulator

$= \dfrac{\text{four times the cross-sectional area}}{\text{the perimeter of the regulator}}$

p = the density of the air

u = the viscosity of the air

V = the mean velocity of the air flowing in the duct

and the independent ratios

$$r_1 = \frac{\text{cross-sectional area of the duct}}{\text{cross-sectional area of the regulator}}$$

$$r_2 = \frac{\text{aerodynamical diameter of the duct}}{\text{aerodynamical diameter of the regulator}}$$

This latter ratio takes into consideration the possible influence of the difference in shape between the regulator and the duct.

The "Pi" theorem gives the relationship

$$f_1(P, A, d, p, u, V, r_1, r_2) = 0 \text{ ....(1)}$$

and then

$$f_2(I_1, I_2, I_3, r_1, r_2) = 0 \text{ ....(2)}$$

where the I's are dimensionless products of the quantities listed in equation (1).

The only fundamental units which occur in the relationship are - Force (F). Length (L), and Time (T). All the physical quantities given in equation (1) may be written in terms of these, viz. -

A	=	area	L^2
d	=	length	L
p	=	mass/volume	FT^2/L^4
u'	=	force/(area X rate of shear)	FT/L^2
V	=	length/time	L/T
P	=	force/area	F/L^2

Three quantities are then selected which are independent of each other, e.g. d, V and p.

The I's of equation (2) are evaluated in the following manner:

Firstly,

$$\text{let } I_1 = d^a V^b p^c P$$

Substituting the units of the different quantities gives

$$I_1 = L^a \, L^b \, T^{-b} \, F^c \, T^{2c} \, L^{-4c} \, FL^{-2}$$

For I_1 to be dimensionless then

$$a + b - 4c - 2 = 0$$

$$-b + 2c = 0$$

$$c + 1 = 0$$

whence,

$$a = 0, \quad b = -2, \quad c = -1$$

giving,

$$I_1 = P/pV^2$$

Similarly,

$$\text{let } I_2 = d^e\ V^f\ p^g\ A$$

whence,

$$e = -2,\quad f = 0,\quad g = 0$$

giving,

$$I_2 = A/d^2$$

Finally,

$$\text{let } I_3 = d^h\ V^i\ p^j\ u$$

whence,

$$h = -1,\quad i = -1,\quad j = -1$$

giving,

$$I_3 = u/dVp$$

Substituting these values in equation (2) gives

$$f_2\ (P/pV^2,\ A/d^2,\ u/dVp,\ r_1,\ r_2) = 0 \ldots\ldots\ldots\ldots(3)$$

This may be rewritten as

$$P/pV^2 = f_3(A/d^2,\ u/dVp,\ r_1,\ r_2) \ldots\ldots\ldots\ldots\ldots(4)$$

or

$$P = pV^2\ f_3(A/d^2,\ u/dVp,\ r_1,\ r_2) \ldots\ldots\ldots\ldots\ldots(5)$$

Expression u/dVp will be recognised as the inverse of Reynolds Number and if it can be shown that, as a result of the air flow being turbulent, the pressure drop approaches proportionality to the square of the velocity then the expression degenerates into a mere constant c_1 and equation (5) becomes

$$P \quad = \quad c_1 pV^2 \, f_3 \, (A/d^2, r_1, r_2) \quad \ldots\ldots\ldots\ldots\ldots\ldots (6)$$

Equation (6) can now be used in the planning of the investigations. Due to the occurrence of a ratio between the cross-sectional area of the regulator and its aerodynamical diameter the influence of shape is suggested. If the ratios r_1 and r_2 were thought to be unimportant then equation (6) would become

$$P/pV^2 \quad = \quad c_1 \, f_3 \, (A/d^2) \quad \ldots\ldots\ldots\ldots\ldots\ldots (7)$$

A plot of observed values of P/pV^2 against simultaneous values of A/d^2 would soon confirm or refute this assumption, since if equation (7) were correct a single continous relationship would be obtained.

<u>Application to Flow in Ducts</u>

The pressure drop P resulting from the flow of air in a smooth, unlined uniform duct is assumed to depend on:-

A	=	the cross-sectional area of the duct
d	=	the aerodynamical diameter of the duct
l	=	the length of the duct
p	=	the density of the air
u	=	the viscosity of the air
V	=	the mean velocity of the air flowing in the duct.

The "Pi" theorem gives the relationship

$$f_4 \, (P/pV^2, A/d^2, u/dVp, l/d) \quad = \quad 0$$

If the values of Reynolds Number are high enough to exert a constant influence the above equation reduces to

$$P = c_2 pV^2 f_5 (A/d^2, 1/d)$$

For a given length of uniform duct this becomes

$$P = c_3 pV^2$$

Therefore if all the assumptions are correct a straight line relationship will exist between the pressure drop in the duct and the product of the air density and the square of the mean velocity of flow.

Application of Bernoulli's Theorem

In a horizontal duct of uniform cross-sectional area, consider two positions 1 and 3 where the velocity distribution is identical. These are spaced far enough apart to permit the insertion of a thin orifice plate at position 2 somewhere between them so that their velocity distributions still remain alike. Application of the well-known Bernoulli's Theorem enables an energy balance between points 1 and 3 to be derived. The following symbols are used:-

P_1 = the static pressure at position 1

V_1 = the mean velocity at position 1

p = the density of the air flowing in the duct

c = a constant

The relationship is,

$$P_1 + \tfrac{1}{2}pV_1^2 = P_3 + \tfrac{1}{2}pV_3^2 + \text{friction loss } 1 - 3 + \text{loss due to the orifice.}$$

From the preceding sections:

Friction loss before insertion of orifice = $c_1 pV_1^2$

Additional loss due to insertion of orifice = $c_2 pV_1^2 f(A/d^2, r_1, r_2)$

Substituting and transposing, the final equation becomes

$$c_2 p V_1^{\ 2} f(A/d^2, r_1, r_2) \quad = \quad P_1 - P_3 - c_1 p V_1^{\ 2}$$

Hence by measuring the drop in static pressure between positions 1 and 3 and knowing the loss attributable to friction, the value of the expression for the loss produced by the orifice can be determined.

CONSTRUCTION OF THE REGULATORS

The regulators were constructed from ¼" plywood in such a way that they could be clamped between the flanges of the wind-tunnel (see Fig. B6). For each orifice, square edges were obtained by using a wood-filler. The sides of the various openings were varnished to make them uniformly smooth.

WIND-TUNNEL LAYOUT

Since it afforded the longest test length the Keith Blackman wind-tunnel was chosen for the investigations. However, preliminary tests showed that the fan was not powerful enough to give satisfactory readings with the smaller regulators in position. Consequently it was decided to operate the two fans in series as shown in Fig. B7.

The Aeroto fan casing was turned through ninety degrees and the evasee bolted to it to form a horizontal flared inlet. In place of the flared entry a rectangular bend containing guide vanes was attached to the Keith Blackman tunnel by a short wooden length of duct. This

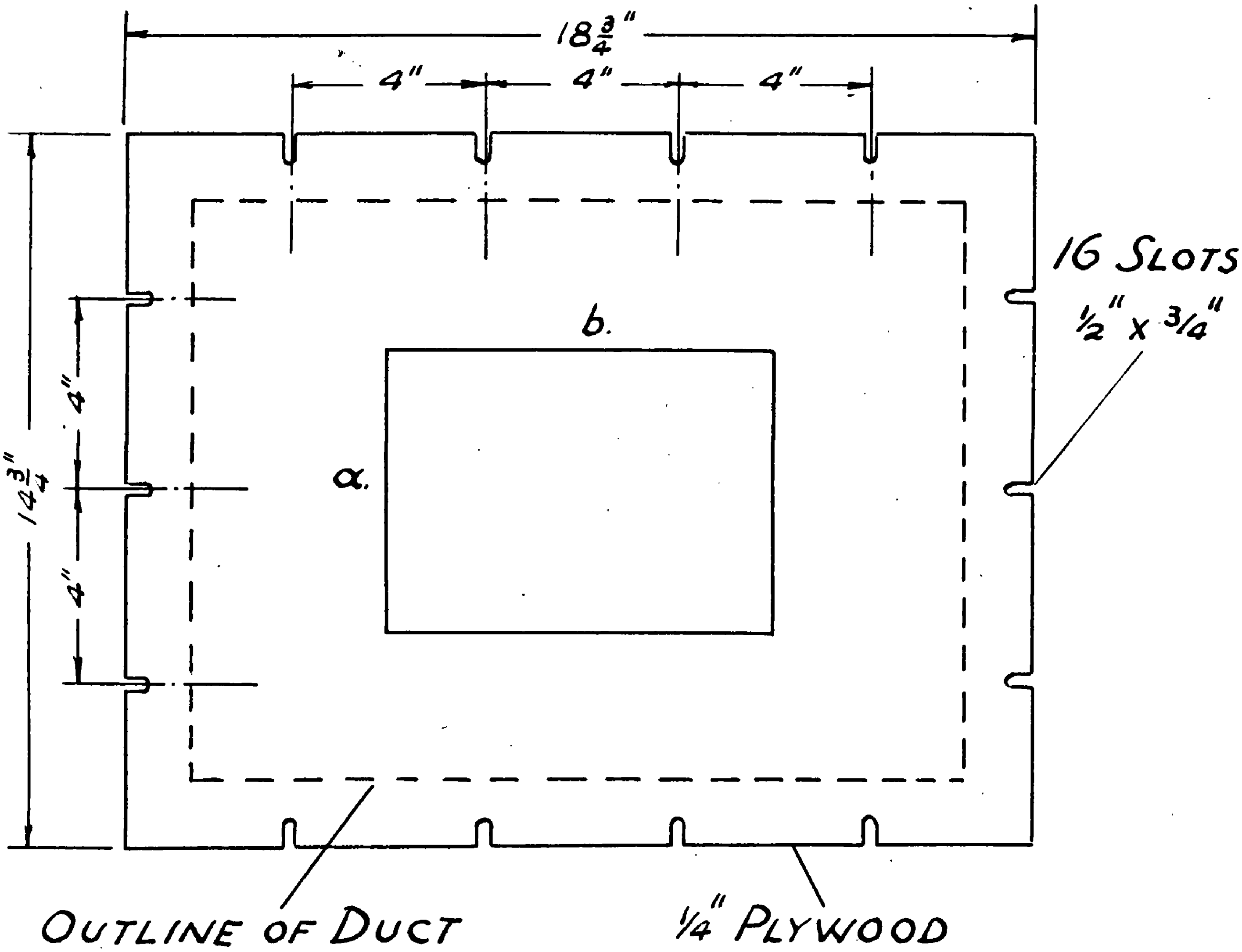


FIG. B.6.

REGULATOR DIMENSIONS.

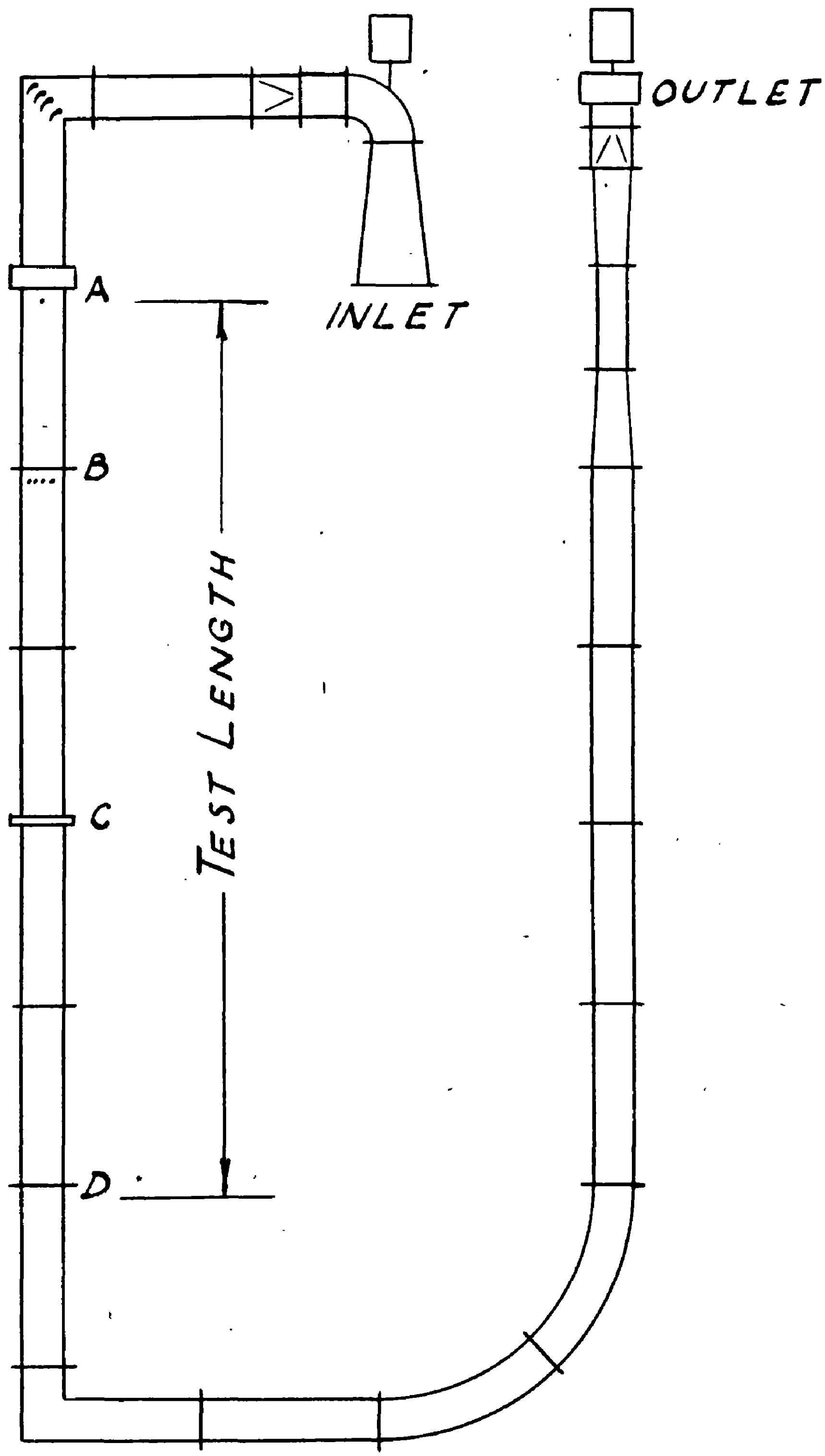


FIG. B 7.

COMBINED WIND TUNNEL.

bend was connected to the Aeroto fan by a section consisting of plywood sides screwed to welded "Dexion" flanges.

CALIBRATION OF TEST LENGTH

With the wind-tunnel layout just described it was possible to obtain a test length (A - D) of nearly 25 feet. It was proposed to insert the regulators at C so that they would not interfere with the flow conditions at positions B or D (32). The mean velocity at B was to be measured separately from the test length drop between A and D. Position B was located as far downstream as possible so that the effect of the rectangular bend on the velocity distribution would be minimised. The pressure drop in the test length was then calibrated against the mean dynamic pressure existing at B.

Firstly, since the time taken to complete the normal 16-point traverse had been shown to be prohibitively long if reasonable accuracy was desired, a new technique for the determination of mean velocity had to be devised. The method adopted was to measure the difference between the velocity pressure at the centre of the cross-section and the velocity pressure at each of the sub-sections in turn. Since the surges in pressure at all positions were very nearly in phase with one another, the fluctuations in the manometer were considerably reduced. Two pitot tubes were used: one placed at the centre of the cross-section and the second moved from one sub-section to another. The total head connection of each pitot tube was coupled to the Askania

Minimeter and the difference in total pressure determined. Previous traverses had proved the static pressure to be constant across the measuring section for all air speeds. Therefore the difference in total pressure between any two points at Station B is also the difference in velocity pressure. From a complete traverse the Centre Constant Cc, defined as the ratio of the mean velocity to the centre velocity, was evaluated.

The proof is as follows:-

Let Vc_{A1} = centre velocity at the instant when the difference between the velocity pressures at the centre and at sub-section A1 is being measured,

and Vs_{A1} = velocity at centre of sub-section A1.

It is assumed that the density is constant at a mean value of p during the traverse.

Therefore,

$$\tfrac{1}{2}pVc_{A1}^{2} - \tfrac{1}{2}pVs_{A1}^{2} = wh_{A1}$$

where h_{A1} = vertical height of water column

w = specific weight of water.

It is assumed that

$$Vc_{A1} = Vc_{B1} = Vc_{C1} \text{ etc.} = Vc$$

where Vc = the mean centre velocity.

It has already been shown that if the cross-section is divided into n sub-sections of equal area then

$$Vm = (1/n)S\ Vs$$

where Vm = mean velocity across the section

S = summation.

From the original equation the following general relationship may be obtained

$$\tfrac{1}{2}pVs^2 = \tfrac{1}{2}pVc^2 - wh$$
$$= whc - wh$$

Hence,

$$Vs = (2whc/p)^{\frac{1}{2}} (1 - h/hc)^{\frac{1}{2}}.$$

The Binomial expansion can be applied if it can be assumed that modulus h/hc is never greater than unity. This was verified during the experimental work and the smallness of the values of h relative to hc enabled powers greater than the first to be neglected. Consequently

$$Vs = (2whc/p)^{\frac{1}{2}} (1 - h/2hc + \ .....)$$

Substituting gives,

$$Vm = (1/n)\ (2whc/p)^{\frac{1}{2}}\ S(1 - h/2hc)$$
$$= (2whc/p)^{\frac{1}{2}} (1 - Sh/2nhc)$$

Now,

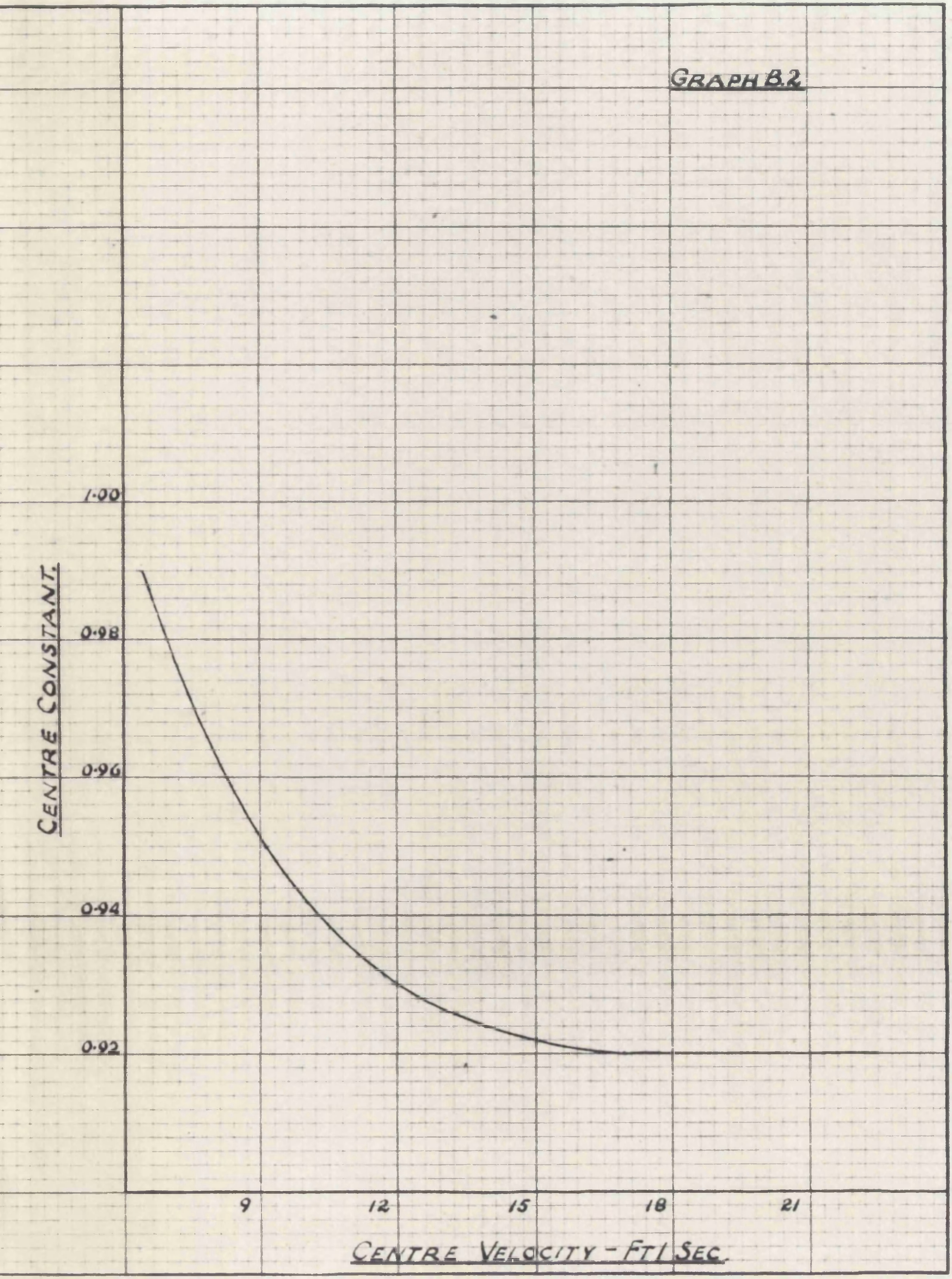
$$(Vm/Vc)^2 = (2whc/pVc^2)\ (1 - Sh/2nhc)^2$$
$$= (1 - Sh/2nhc)^2 \text{ (since } \tfrac{1}{2}pVc^2 = whc)$$

Therefore,

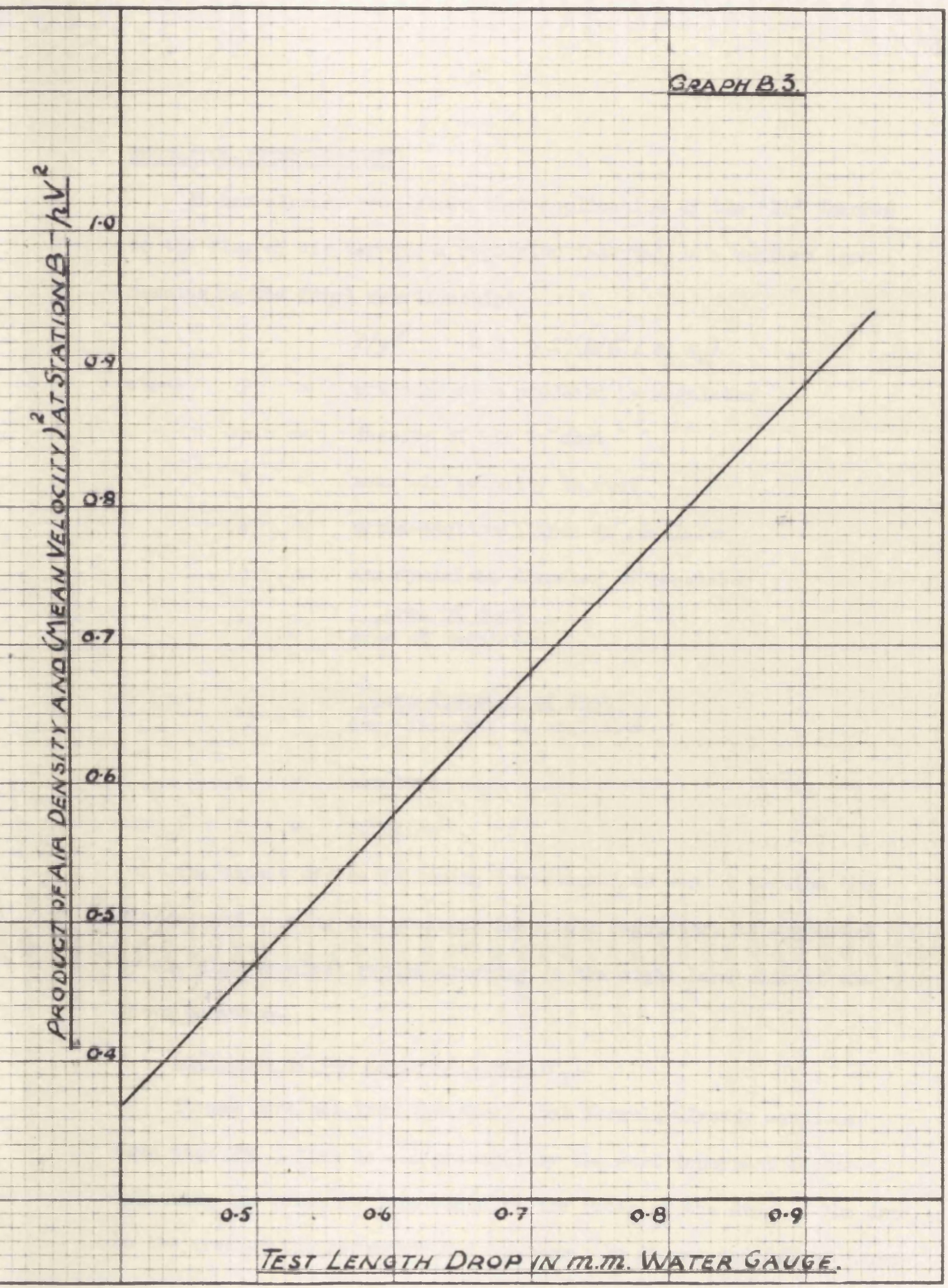
$$Cc = Vm/Vc = (1 - Sh/2nhc)$$

A number of traverses were performed at station B to cover a suitable range of velocities. For each differential observation, taken by Askania Minimeter, the velocity pressure at the centre of the cross-section was found. A T-piece manifold (see Plate VC following page 79.) allowed the connections to be rapidly switched without interfering with the instrument. To check that the velocity remained constant during each traverse the test length pressure drop was observed by means of the Casella Vernier Reading Manometer. The Centre Constants were found from the derived expression and plotted against the centre velocity at B (Graph B2). It will be seen that the Centre Constant is nearly invariable over the test range. Although the increase in value at the lower air speeds does not agree with the results of other workers (33) the experimental conditions are widely different and the rectangular bend will very probably affect the velocity distribution at position B.

The test length was calibrated by measuring the pressure drop for any particular centre velocity at station B. The two water manometers were used separately. From the results the pressure loss in millimetres of vertical water column was plotted against the product of the air density and the square of the mean velocity at B (Graph B3). The resulting straight line indicates that turbulent flow is fully developed and the frictional pressure drop is independent of Reynolds Number.



GRAPH B.2
CENTRE CONSTANT.
1·00
0·98
0·96
0·94
0·92
9
12
15
18
21
CENTRE VELOCITY - FT/SEC.


GRAPH B.3.
PRODUCT OF AIR DENSITY AND (MEAN VELOCITY)^2 AT STATION B - ρV^2
1·0
0·9
0·8
0·7
0·6
0·5
0·4
0·5
0·6
0·7
0·8
0·9
TEST LENGTH DROP IN m.m. WATER GAUGE.

REGULATOR INVESTIGATIONS

It has already been shown that application of the "Pi" Theorem to the flow of air through a regulator inserted in a uniform duct results in the final relationship:

$$P/pV^2 = c\ f\ (A/d^2,\ r_1\ r_2)$$

where P = pressure drop produced by regulator

p = density of air in duct

V = mean air velocity in duct

A = cross-sectional area of regulator

d = aerodynamical diameter of regulator

$r_1 = \frac{\text{area of duct}}{\text{area of regulator}}$

$r_2 = \frac{\text{aero-diameter of duct}}{\text{aero-diameter of regulator}}$

c = constant

f = function

The object of the following investigations was to examine the factors influencing the pressure loss in a regulator, as suggested by the dimensionless groups occuring in the right hand side of the above equation.

1. Influence of the Ratio of Areas (r_1)

It was surmised from the shock loss formula already mentioned (see page 28.) that in all probability the most important factor affecting the pressure drop would be the ratio of the area of the duct to the area of the regulator and consequently it was decided to investigate this aspect first of all. Referring again to the three right hand side dimensionless groups of the equation it will be readily appreciated that it is impossible to maintain any two constant while examining the third. However, it can be easily shown that the ratio A/d^2 will remain constant so long as the profile of the opening is unchanged. In addition, the ratio r_2 was really only introduced to take into account any possible influence deriving from the difference between the shape of the orifice and the shape of the tunnel, and hence, if the profiles of the regulators are identical, it is more than likely that the effect of variation in r_2 will be small. Therefore in this experiment the shape of the openings was kept constant so that an attempt could be made to establish the relationship between P/pV^2 and the ratio of the duct and regulator areas.

A number of regulators (series A) were constructed (see Appendix B). The openings were of different cross-sectional area but were all geometrically similar to the profile of the wind-tunnel. The minimum area was determined by the smallest size of opening which allowed the pressure drop to be measured over a fairly wide range of air velocities; the maximum area corresponded to the maximum area in a subsequent series B. Each regulator was in turn bolted in position at joint C. By careful alteration of the speeds of the two fans a range of air velocities was obtained. For each setting of the fan controls the centre velocity at station B was measured by pitot-static tube and Askania Minimeter, and the loss in static pressure over the test length ascertained by means of two pitot tubes connected to the Casella manometer. From Graph B2 the mean velocity at B was obtained, and, the air density having been calculated, the loss due to friction determined from Graph B3. Hence the loss in pressure caused by the regulator was established.

It will be noticed that P/pV^2 is exactly half of the Pressure Drop Coefficient of the regulator. Therefore since this factor, hereafter referred to as the Loss Coefficient (l), gives a true comparison of the resistances of the orifices, it was determined for each. For every regulator the values of l over the test range were obtained by dividing the pressure loss produced by the orifice by the product of the air density and the square of the mean velocity of approach.

During the derivation of the equation, it will be remembered, it had been assumed that the value of Reynolds Number (Rn) was high enough for the influence of this factor to be neglected. However, it has been shown (34) that for square-edged orifices the "vena contracta" effect, and hence the effective ratio of the areas, is not stabilised even above Rn values, referred to the aerodynamic diameter of the opening, of 10,000. The main point illustrated is that while for area ratios greater than two the rate of change is really very small, the stable condition is not reached until Rn is nearly 100,000. The maximum Reynolds Number it was possible to achieve with the smallest opening (A,10) was in the region of 43,000. In all the tests a slight tendency for the Loss Coefficient to increase with increase in Rn indicated that the stable region had not been attained.

The main source of inaccuracy was incurred during the measurement of the centre velocity at B. Although the Askania Minimeter is graduated to 0.01 mm., in actual practice the least scale division is more likely to be 0.05 mm. As a direct result of this the greatest accuracy obtainable was about 2%. This figure was reached during the reading of the highest velocity encountered and the accuracy decreased with reduction in air speed.

The calculated values are shown in Table No.2 The Loss Coefficient (l) was taken as the average of the values within 5% of the result at maximum Rn.

TABLE No.2

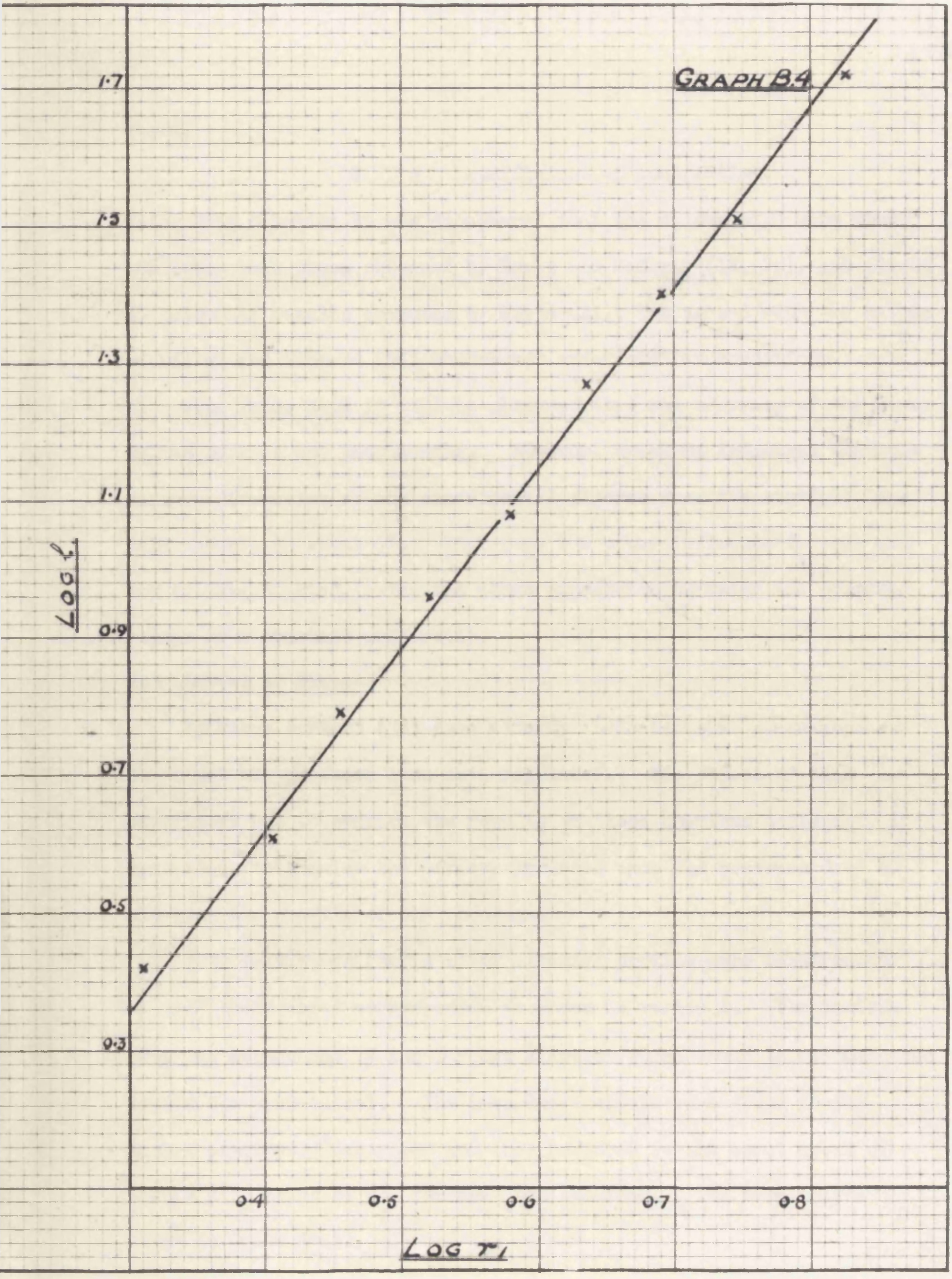
Regulator	A1	A2	A3	A4	A5	A6	A7	A8	A9	A10
Ratio of Areas, r_1	1.77	2.05	2.55	2.84	3.46	3.81	4.30	4.91	5.52	6.70
Loss Coefficient, l	1.2	2.6	4.1	6.1	9.1	11.9	18.7	25.0	32.5	52.0

An empirical relationship was sought between the Loss Coefficient and the ratio of the area of the duct to the area of the regulator. This was eventually achieved by plotting log l against log r_1 (Graph B4) from which the following expression was derived

$$l = 0.363\, r_1^{2.64}$$

Discussion of Results: From the shock loss formula it can easily be shown that

$$l = 0.5(r_1/C - 1)^2$$



GRAPH B.4
1·7
1·5
1·3
1·1
0·9
0·7
0·5
0·3
LOG ℓ
0·4
0·5
0·6
0·7
0·8
LOG r₁

where,

C = coefficient of contraction.

Using this relation it was discovered that the values of C were about 10% lower than those obtained by Weeks and McElroy (18, 20), and almost 3% below the results computed by Witte (21), but agreed with the values of Miller and Hibberd (35) who worked under similar conditions.

From Table No.2 it will be observed that the accuracy of the lower values of l is not outstanding. However, Graph B4 indicates that the maximum dispersion of the experimental figures from the straight line only amounts to about 5%. Therefore the other influence factors can now be investigated, using the above expression to refer the observed values to a constant area ratio.

2. Effect of Shape

Previous workers (35) have already indicated that shape appears to affect the pressure loss in a regulator. The object of this experiment was to confirm the results of these previous tests and, if warranted, to examine the effects produced upon the pressure drop by altering the profile of the openings.

Four regulators (seris B) of circular profile were constructed so that their areas corresponded to areas in series A. The maximum area was determined by the largest diameter which could be accommodated inside the wind-tunnel. The Loss Coefficients were found in the manner previously described and then referred to the exact areas of series A by means of the formula just obtained. The results are shown in Table No.3.

TABLE No.3

Profile	Rectangular				Circular			
Regulator	A1	A3	A6	A10	B1	B3	B6	B10
Area (sq. in.)	107.4	74.6	49.9	28.4	107.5	74.6	50.3	28.1
Loss Coefficient 1	1.2	4.1	11.9	52.0	1.1	3.9	11.8	52.0
1 referred to areas in series A	1.2	4.1	11.9	52.0	1.1	3.9	12.0	51.0

The conclusions reached previously (35) that shape does assert some influence and that effects are more noticeable the bigger the opening, are corroborated by the results shown in Table No.3. The probable reason for this is that in the case of the larger regulators the loss due to eddying at the corners contributes appreciably to the total loss and is obviated by the use of circular openings.

Table No.3 does not indicate any significant difference between the Loss Coefficients for the smaller regulators of circular and rectangular profile. Notwithstanding this apparent lack of appreciable influence due to shape, however, it was decided to compare the Loss Coefficients of a range of different rectangular shapes of uniform area (series C). Since, in actual practice, only a fairly large value of r_1 would allow any wide choice in the regulator profile the openings were constructed with areas equal to the area of regulator A8. The long axis of each orifice was made horizontal. Correspondong to the dimensions of C6 a single regulator (D6) was constructed with its long axis vertical. Fig. B8 illustrates the wide difference in the shapes that were used.

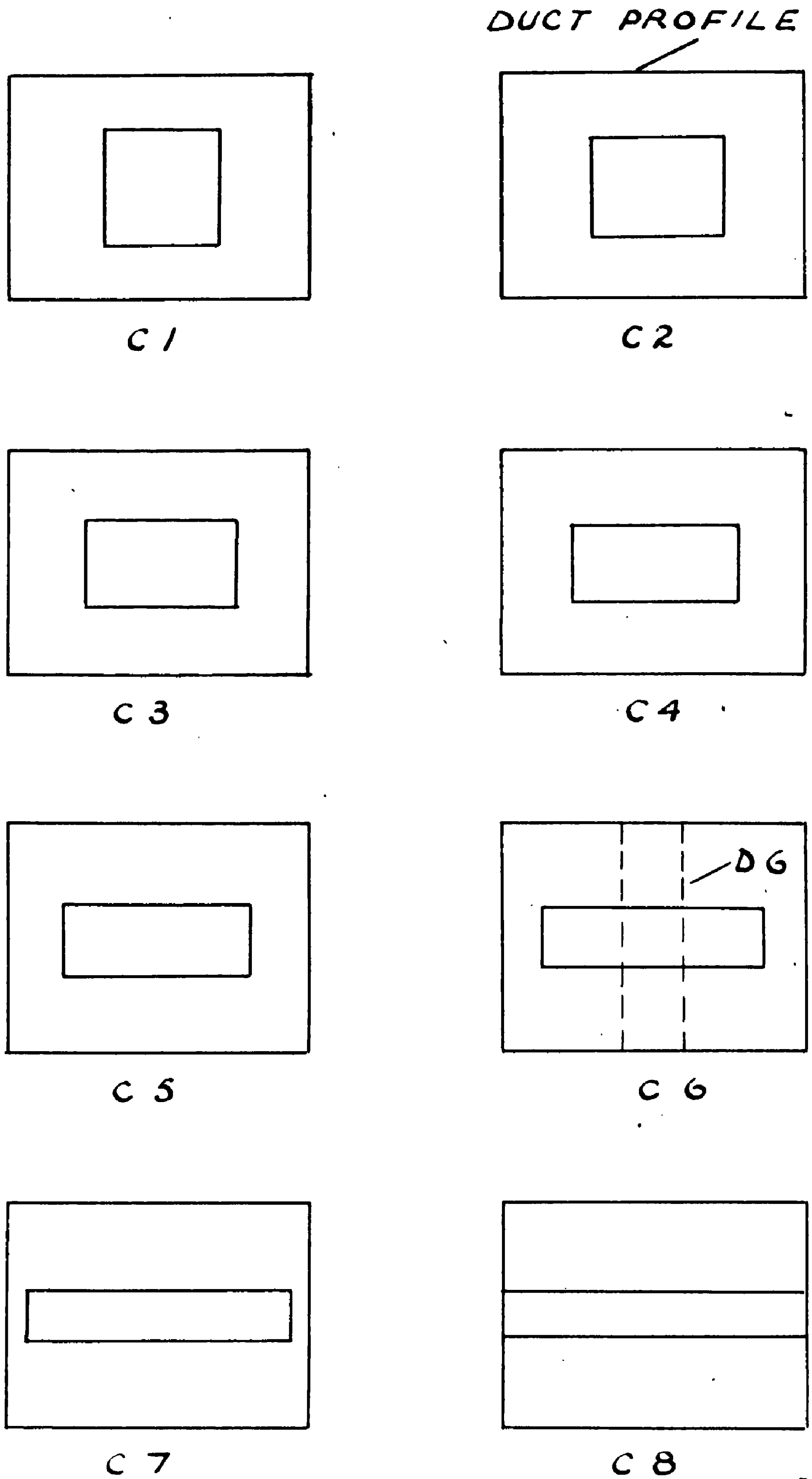


FIG. B 8.

REGULATOR SHAPES IN SERIES C.

The Loss Coefficients were obtained in the usual way and the results are recorded in Table No.4.

TABLE No.4.

Regulator	C1	C2 (A8)	C3	C4	C5	C6	C7	C8	D6
Area	38.4	38.7	38.8	38.5	38.4	38.8	38.8	38.9	38.6
Loss Coefficient l	25.9	25.1	25.2	25.6	25.3	25.0	25.2	26.0	25.2
l referred to area A8	25.4	25.1	25.4	25.2	24.8	25.2	25.4	26.4	25.0
Aerodynamic diameter	6.19	6.15	6.01	5.81	5.52	5.12	4.61	4.23	5.10
A/d^2	1.00	1.02	1.07	1.14	1.26	1.54	1.82	2.17	1.49
r_2	2.19	2.21	2.26	2.34	2.46	2.65	2.94	3.21	2.66

Discussion of Results: From Table No.4 it will be observed that:-

(a) As the factor r_2 increases there appears to be a slight tendency for the Loss Coefficient to first of all decrease and then increase. However, the accuracy of the readings (circa. 5%) does not justify any such interpretation.

(b) There is a marked increase in the value of l obtained for regulator C8. In all probability this is due to the fact that the width of the orifice was almost equal to the width of the duct and part of the opening may have been clamped between the flanges, thereby reducing the effective area.

(c) There is nearly a 50% rise in the value of r_2 and slightly more than a 100% increase in the value of A/d^2 over the range of regulators tested. The variation in the Loss Coefficient about a mean value is just over $\pm$ 1% which is much less than the indicated accuracy of the reading ($\pm$ 2.5%). Therefore it may be safely assumed that the influence of the factors A/d^2 and r_2 upon 1 is insignificant and can be ignored.

(d) Further proof of the inconsequence of shape is provided by the agreement between the results obtained for regulators C6 and D6.

3. Effect of Position

The two preceding tests have indicated that of the three dimensionless groups occuring on the left hand side of the equation, only r_1 need be considered. However, a further factor which might possibly affect the pressure drop through a regulator is its position in the airway cross-section. The object of this experiment was to examine this effect by changing the location of an orifice of constant area.

The dimensions of the regulators in series E, constructed with their long axes horizontal, agreed with those of regulator A8. The openings were located in the duct cross-section along one of the diagonals (see Fig. B9). A single regulator (F5) was constructed so that the position of the orifice was along a horizontal centre line, the same distance from the centre of the duct as E5. The Loss Coefficients were obtained in the usual way and are listed in Table No.5.

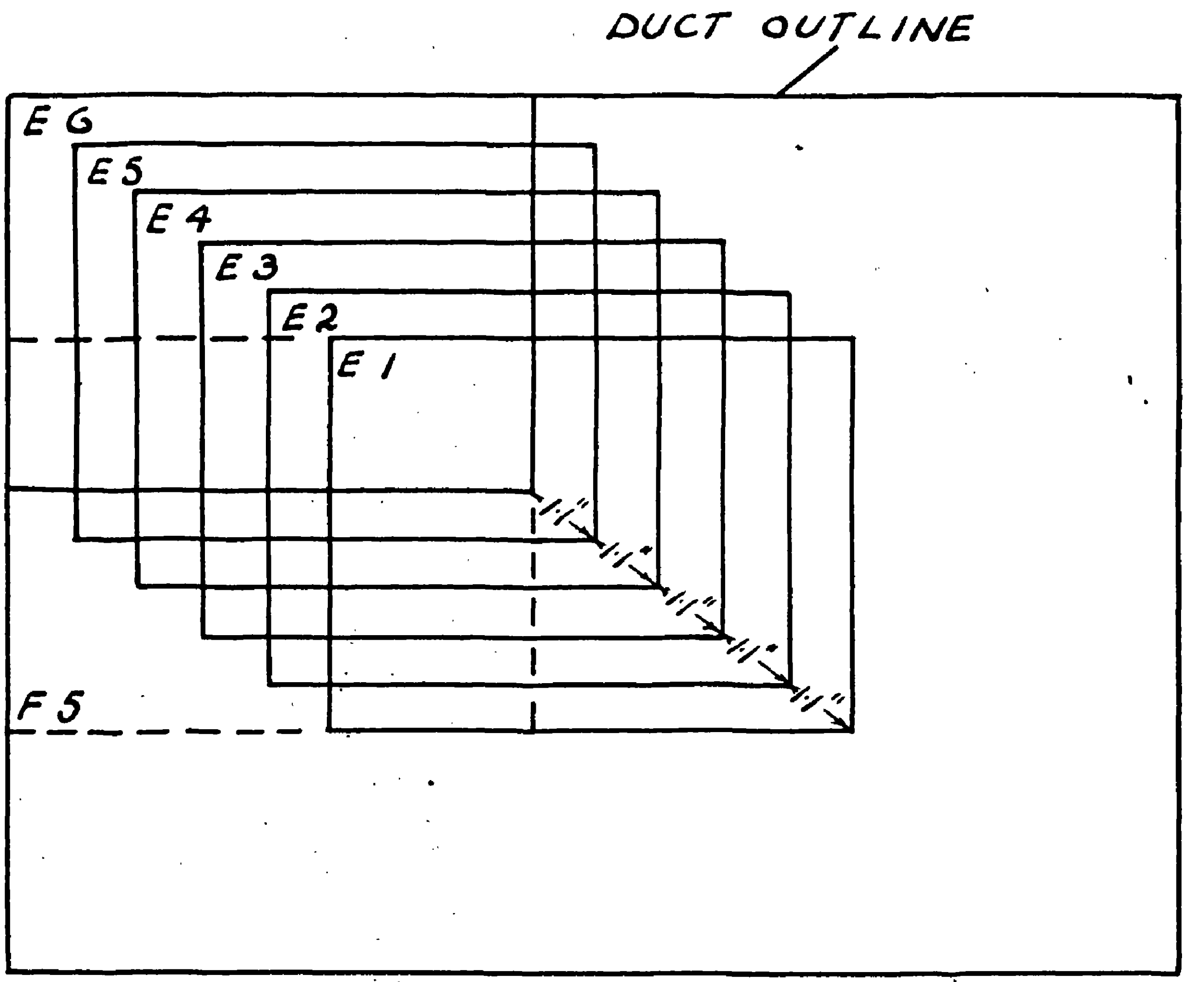


FIG. B.9.

REGULATOR POSITION SERIES E.

TABLE No.5

Regulator	E1 (A8)	E2	E3	E4	E5	E6	F5
Area (sq. in.)	38.7	38.4	38.6	38.7	38.8	38.4	38.4
Loss Coefficient 1	25.1	25.8	25.1	24.7	25.2	26.8	25.8
1 referred to area A8	25.1	25.3	24.9	24.7	25.4	26.3	25.3
Distance from centre of duct (in.)	0	1.1	2.2	3.3	4.4	5.5	4.4

Discussion of Results: From Table No.5 the following observations may be made:-

(a) There is no significant change in the values of Loss Coefficient outwith the limit of experimental accuracy(5%).

(b) Although there is a marked increase in the Loss Coefficient for regulator E6 this is almost certainly due to the fact that, since its position was chosen so that it would be located at the top left hand corner of the duct (see Fig. B9), part of the orifice was contained between the flanges whereby its area was reduced.

Construction of Nomograph

In the science of Ventilation Planning it is most desirable to be able to stipulate the cross-sectional area of a regulator which, when a certain quantity of air is passing through it, will exert a specified resistance. Therefore the empirical relationship derived between the Loss Coefficient and the ratio of the areas was rewritten in the form

$$Ar = \frac{Ad^{.242} \times Q^{.758}}{27.23 \times h^{.379}}$$

where Ar = cross-sectional area of regulator in square feet

Ad = cross-sectional area of roadway in square feet

Q = mean quantity of air at a standard density of 0.075 lb. per cubic foot passing in cubic feet per second

h = water gauge lost in regulator in inches.

Since the above formula is extremely unwieldy it was decided to construct a nomograph (see Fig. B10) in which the ranges of the different quantities were those likely to be met with in practice and whereby if Ad, Q and h were given, Ar could be readily evaluated. An example will best serve to illustrate its use.

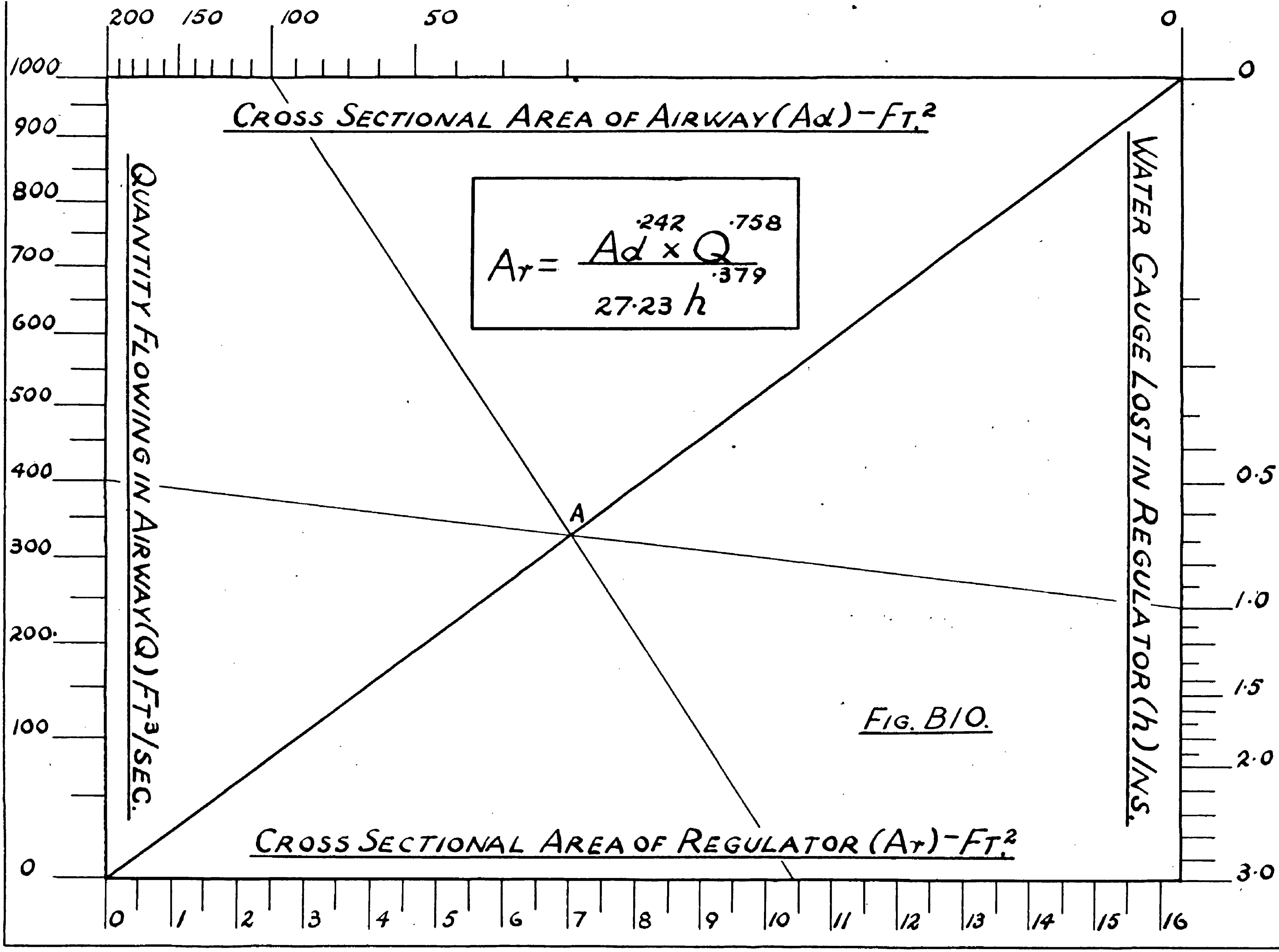
Example: A Ventilation Survey has shown 400 cubic feet per second of air at a density of 0.075 lb. per cubic foot to be flowing in a roadway of cross-sectional area 100 square feet. It is desired to increase the pressure drop in this roadway by the equivalent of 1.0 inch water gauge with the same quantity flowing. What is the cross-sectional area of the regulator required?

Instructions: With reference to Fig. B10:-

(a) Draw a straight line from the 400 mark on the Quantity scale to the 1.0 mark on the Water Gauge scale.

(b) Note where this line intersects the ungraduated diagonal line (point A).

(c) Draw a straight line from the 100 mark on the Airway Area



200 150 100 50 0
Cross Sectional Area of Airway (Ad) - Ft.²
$A_r = \dfrac{Ad^{\cdot 242} \times Q^{\cdot 758}}{27{\cdot}23\, h^{\cdot 379}}$
Quantity Flowing in Airway (Q) Ft³/sec.
1000 900 800 700 600 500 400 300 200 100 0
Water Gauge Lost in Regulator (h) Ins.
0 0·5 1·0 1·5 2·0 3·0
A
Cross Sectional Area of Regulator (Ar) - Ft.²
0 1 2 3 4 5 6 7 8 9 10 11 12 13 14 15 16

Fig. B/10.

scale through point A to meet the Regulator Area scale and read off the cross-sectional area of the regulator required, viz. 10.4 square feet.

CONCLUSIONS

From the experimental work just described and the results which it yielded the following conclusions may be drawn:-

(1) It is evident from experiment 1 that the influence of r_1 is extremely significant. Since the areas of each opening can only be reckoned correct to 1%, this influence will seriously affect the experimental accuracy of the Loss Coefficient for each regulator.

(2) For the values of r_1 likely to be met with in practice the effect of change in shape upon the loss in pressure is negligible. As a result of the limited air power available the value of r_1 in experiment 2 was restricted to 4.9. In practice higher values will usually occur but the choice of shape will not normally be quite so wide.

(3) No effect due to change of position was detected so that the influence of this factor upon the pressure drop produced by a regulator can be considered insignificant. Again it must be pointed out that in practice a relatively small opening will normally be used and this will allow a greater displacement of the regulator from the centre of the airway than was possible in experiment 3. However, as

the openings get smaller the condition will be approached where the recovery of velocity energy will be negligible and the position of the regulator, or for that matter its shape, will be unimportant.

(4) The maximum values of Reynolds Number were rather limited although there is some justification (34) in considering the stable conditions to be almost attained. Reynolds Numbers ten times as great will normally be encountered in the workings of a mine and it will be readily appreciated that this may cause a serios difference from the results obtained in these experiments.

(5) The computation of the area of a regulator from the derived empirical relationship would prove quite laborious. A simple method of evaluating this area was evolved by designing a suitable nomograph. Since the accuracy of measurement in practice is never usually greater than 5% and the quantity cannot be expected to remain absolutely constant when additional resistance is introduced, it is fully expected that the nomograph will prove satisfactory as a working guide in the estimation of regulator size.

(6) It seems fitting to include at this juncture a critical review of the manometers used to date. As will have been noted from the descriptions of the Askania Minimeter and the Casella Vernier Reading Manometer, both these instruments are extremely finely graduated. Also ample provision is made for determining the reference level really accurately by means of a pointer and its image. However, during operation, when any fluctuation of the pressure value being measured

is present it is impossible to estimate a mean position of the reference level, and consequently the practical accuracy of these instruments is very much less than that suggested by the refinement of graduation. For future experiments, therefore, it is proposed to obtain other designs of manometers as soon as supply conditions allow.

It will be quite evident already that the greatest obstacle to a more complete investigation of the factors affecting the loss of energy in regulators was the limited air power available. Nevertheless the results have more or less proved that the influence of the shape and the location of the opening are unimportant. In addition it may be assumed that the relationship between the area of the opening, the area of the roadway, the quantity of air flowing, and the water gauge lost in the regulator will be of the form obtained in these investigations. The empirical relationship established from the results of the foregoing experiments will require to be tested over a fair range of conditions to see if any adjustment is indicated. If this is the case it should be easily undertaken by a few straightforward tests on a small number of regulators.

APPENDIX B

Regulator Serial Number	Mean Dimensions a x b (in.)	Mean Area Ar (sq.in.)	Perimeter (in.)	Aerodynamic Diameter d (in.)	Area of Duct / Area of Regulator r_1	$\frac{Ar}{d^2}$
A1	8.95 x 12.00	107.4	41.9	10.26	1.77	1.02
A2	8.35 x 11.10	92.6	38.9	9.53	2.05	1.02
A3	7.50 x 9.95	74.6	34.9	8.55	2.55	1.02
A4	7.08 x 9.45	66.9	33.1	8.08	2.84	1.02
A5	6.66 x 8.25	55.0	29.8	7.38	3.46	1.01
A6	6.11 x 8.15	49.9	28.5	7.00	3.81	1.02
A7	5.76 x 7.68	44.2	26.9	6.57	4.30	1.02
A8	5.38 x 7.20	38.7	25.2	6.15	4.91	1.02
A9	5.08 x 6.76	34.4	23.7	5.81	5.52	1.02
A10	4.63 x 6.14	28.4	21.5	5.28	6.70	1.02
B1	11.70 dia.	107.5	36.8	11.70	1.77	0.785
B3	9.75 "	74.6	30.6	9.75	2.55	0.785
B6	8.00 "	50.3	25.1	8.00	3.77	0.785
B10	5.99 "	28.1	18.8	5.99	6.76	0.785
C1	6.20 x 6.20	38.4	24.8	6.19	4.95	1.00
C2(A8)	5.38 x 7.20	38.7	25.2	6.15	4.91	1.02
C3	4.76 x 8.15	38.8	25.8	6.01	4.90	1.07
C4	4.30 x 8.95	38.5	26.5	5.81	4.94	1.14
C5	3.80 x 10.10	38.4	27.8	5.52	4.95	1.26
C6	3.26 x 11.90	38.8	30.3	5.12	4.90	1.54
C7	2.76 x 14.05	38.8	33.6	4.61	4.90	1.82
C8	2.44 x 15.95	38.9	36.8	4.23	4.89	2.17
D6	3.24 x 11.90	38.6	30.3	5.10	4.92	1.49
E1(A8)	5.38 x 7.20	38.7	25.2	6.15	4.91	1.02
E2	5.36 x 7.16	38.4	25.0	6.13	4.95	1.02
E3	5.36 x 7.20	38.6	25.1	6.15	4.92	1.02
E4	5.38 x 7.18	38.7	25.1	6.16	4.91	1.02
E5	5.38 x 7.22	38.8	25.2	6.16	4.90	1.02
E6	5.36 x 7.16	38.4	25.0	6.10	4.98	1.03
F5	5.34 x 7.18	38.4	25.0	6.11	4.96	1.03

SECTION C

SHAFT PRESSURE LOSSES DUE TO CAGES

INTRODUCTION

In providing better conditions of ventilation the problem of reducing the overall resistance to air-flow of a mine must constantly receive consideration. As a result of ground movements, the maintenance of roadways of large cross-sectional area in inbye workings is usually difficult and often impossible, so that high resistances are unavoidable. Furthermore it would be impracticable to reduce the resistance of equipment installed in these roadways for the prime purpose of transporting minerals. Therefore, means of lowering the resistance to air-flow must be sought in those roadways driven in solid or settled ground or in the mine shafts themselves. The latter provide the best conditions since they are fitted with rigid structures, the position of which usually remains unaltered during the life of the mine, and very few moving components.

It has also been discovered (36) that, due chiefly to the high air velocities, the pressure loss in the shafts is in most cases greater than half the total loss for the mine. With the present-day tendency for shafts to be sunk to greater depths it has already been pointed out (37) that this proportion will appreciably increase. Hence it would appear imperative to devise ways of lowering the amount of energy removed from the air in this particular region of the mine workings.

A great deal of work has already been completed in investigating means of reducing shaft resistance by experiments on models (36, 38, 39), and also be constructing a length of underground roadway to represent a shaft layout (40). These have dealt exclusively, however, with such aspects as bunton spacing and streamlining, position of bunton in shaft cross-section, and the complete boxing-off of buntons, ladder compartments and all transverse obstructions to the air-flow. It is interesting to note that 90% of the pressure loss in a shaft is attributed to the buntons, and it is claimed that if they are boxed-off behind solid partitions the resistance is reduced by 60%. The influence of cages has been completely ignored except for mention by one investigator (41) of the "piston" effect that would result from the last named arrangement.

The purpose of this section of the research work was to investigate the factors controlling the amount by which the inclusion of a cage or cages increases the resistance of a shaft. Information on these factors would be of considerable value in the designing of cages with the object of reducing pressure losses to a minimum. The effectiveness of reducing the resistances of cages by attaching fairings was also examined.

The experiments were performed with models and, by suspending them in a horizontal wind-tunnel, corrections for altitude were obviated. The investigations were confined to a single cage rope-guide, double cage rope or rigid end-guide, and quadruple cage rigid end-guide installations. The tests were conducted over a range of Reynolds Numbers, referred to the tunnel diameter, of 300,000 - 450,000. It was considered that this range was sufficiently high to enable the findings to be applied to full-size installations geometrically similar to those examined.

All resistances were determined as a Pressure Drop Coefficient which may be defined as the loss in pressure due to any particular part of the arrangements being studied, divided by the dynamic head of the mean velocity of the air. This Coefficient was used first of all because energy losses in the air were being investigated. The drag-force of the air on the cages was not considered since any additional load on the winding engine due to this will be negligible. Secondly the Pressure Drop Coefficient can be applied directly to full-scale arrangements if geometrical similarity is maintained and change in Reynolds Number can be ignored. In addition it was found possible during the experimental work to evaluate this factor readily from a simple ratio.

The preliminary work consisted of the installation of a new wind-tunnel of adequate length and a powerful fan unit. Then the suitability of the fan for the experimental operations contemplated was established by standard test procedure. Appropriate devices were inserted in the tunnel to remove rotation in the air and induce symmetry in the velocity distribution. Finally the investigations into the factors influencing the loss in pressure caused by cages and the effect of fitting fairings were conducted.

The discussions and conclusions relevant to each test have been inserted at the end of each experiment while a general appraisal of the investigations follows the description relating to the final test. A summary of those aspects of the work which could profitably be investigated further concludes this section of the thesis.

CONSTRUCTION OF WIND TUNNEL

The wind-tunnel consisted of 6-feet lengths of steel ducting, $11\frac{1}{4}$ inches internal diameter, bolted together with rubber gaskets at the joints. The tunnel was made 90 feet long so that a large test-length could be obtained. Each 6-feet length was provided with a tapped hole, 6 inches from one end, to accommodate an airtight gland, through which a pitot tube could be inserted into the wind-tunnel. When not in use these tappings were sealed by metal plugs.

Air flow in the wind tunnel was provided by a centrifugal fan with forward facing aerofoil section blades. The fan could produce a maximum total head of 18 inches water gauge and was coupled directly to a 20 B.H.P. constant speed Squirrel-cage motor, controlled by a Star-Delta starter (see Plate IC).

A flexible canvas coupling was inserted between two lengths of ducting near the fan outlet, thus preventing vibrations being transmitted from the fan-motor unit. This damping device was deemed advisable for two reasons. Firstly, vibrations would adversely affect flow conditions generally in the wind-tunnel especially at low velocities, and reduce the effectiveness of any flow straighteners or wire screens which had been inserted. Secondly, it was proposed to insert in the line of ducting an observation length of three 4-feet Perspex sections which were relatively weak, particularly at the flanges, compared with the metal lengths. It was thought that these might be seriously affected by prolonged vibration. An additional advantage of having the coupling was the ease with which the fan-motor

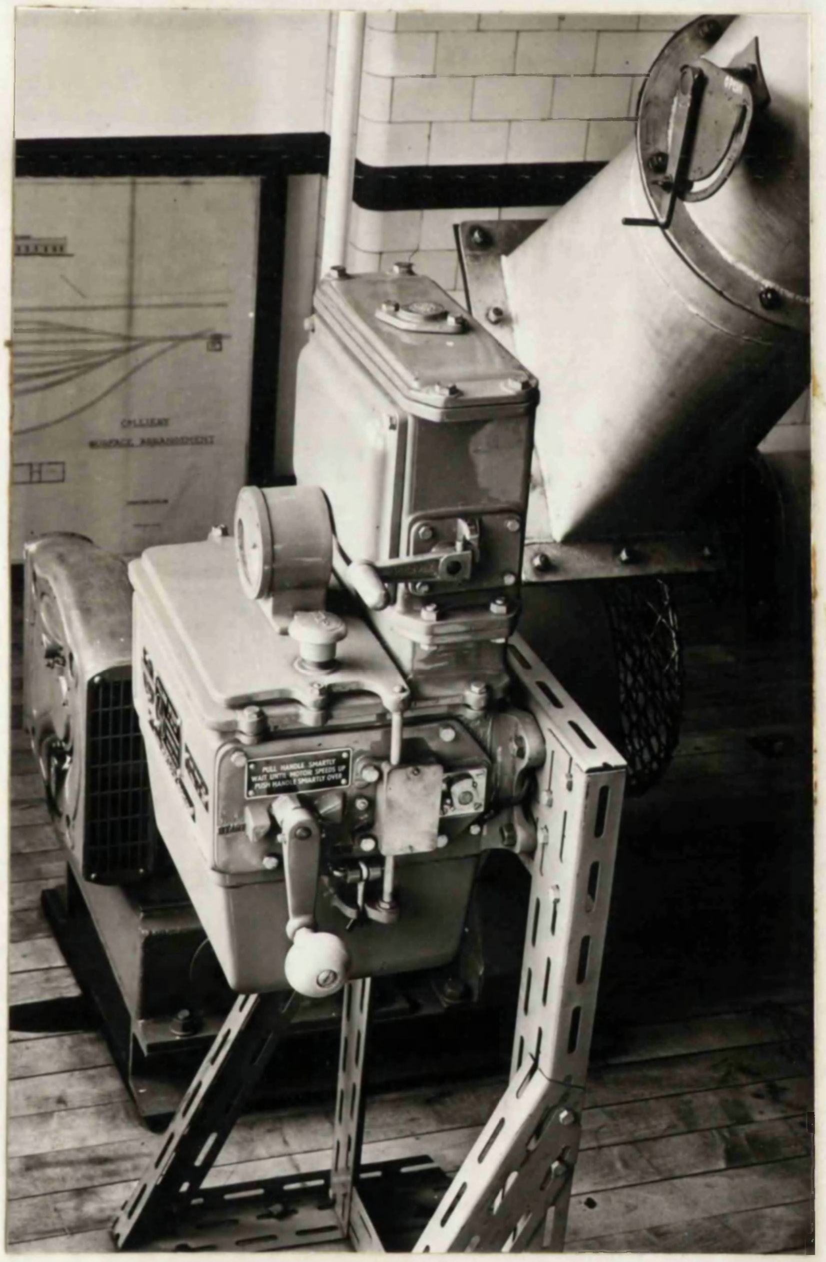


PLATE IC: Fan-Motor Unit and Star Delta Starter unit could be aligned with the ducting.

The wind-tunnel arrangements were completed by two throttling devices. The first of these was inserted near the fan evasee (see Plate IC) and consisted of a thin steel plate 11 inches diameter riveted to a steel shaft which passed through bossed sleeves to the outside of the duct. The shutter could be clamped in any position between fully closed and fully open. Since a fan of the type used overloads unless started on no-load, this contrivance was necessary to protect the driving motor and to allow the Star-Delta starter to function.

The second device was a conical shutter situated at the outlet of the wind-tunnel. It consisted of a metal cone with a threaded spindle screwed into its apex. This spindle rotated in a tapped sleeve located at the centre of the duct cross-section. The shutter allowed infinite variation from zero to maximum in the throttling of the air. It could be expected to operate without producing undue turbulence and eddying or inducing assymmetrical velocity distribution upstream.

FAN TEST

Since the fan-motor unit was a new installation it was necessary to obtain the operating characteristics of the fan, in order to find the water gauge available for experimental purposes and the velocities likely to be encountered. A complete test was carried out according to

B.S.S. 848: 1939(42) as follows:-

Power Input

The "Two-Wattmeter" method (43) was used to measure the electrical power input to the fan motor, a separate voltmeter being used to maintain a check on the line voltage.

Fan Speed

The design speed of a motor can only be obtained under certain voltage conditions and, in any case, some falling-off in speed is to be expected with increase in load. Therefore the rotational speed of the fan impeller was determined using a portable stroboscope (Philips P.R.9103 - see Plate IIC).

Static Pressure at Fan Outlet

This pressure was measured by means of a pitot-static tube connected by $\frac{1}{4}$ inch bore rubber tubing to a Casella Micromanometer.

Mean Quantity

To find the mean quantity of air flowing through the fan it was necessary to obtain the mean velocity at a section of the ducting where the area was known. This was carried out as described below.

For air flowing in a circular duct (see Fig. C1) the mean velocity Vm may be defined as follows:

$$\pi R^2 \, Vm \quad = \quad \int (2\,\pi\; r) v.\, dr \quad \ldots\ldots\ldots\ldots (1)$$

where v is the velocity existing at radius r.

If the air density remains constant

$$v \quad = \quad kh^{\frac{1}{2}}$$

where k is a constant and h is the velocity head measured on a water

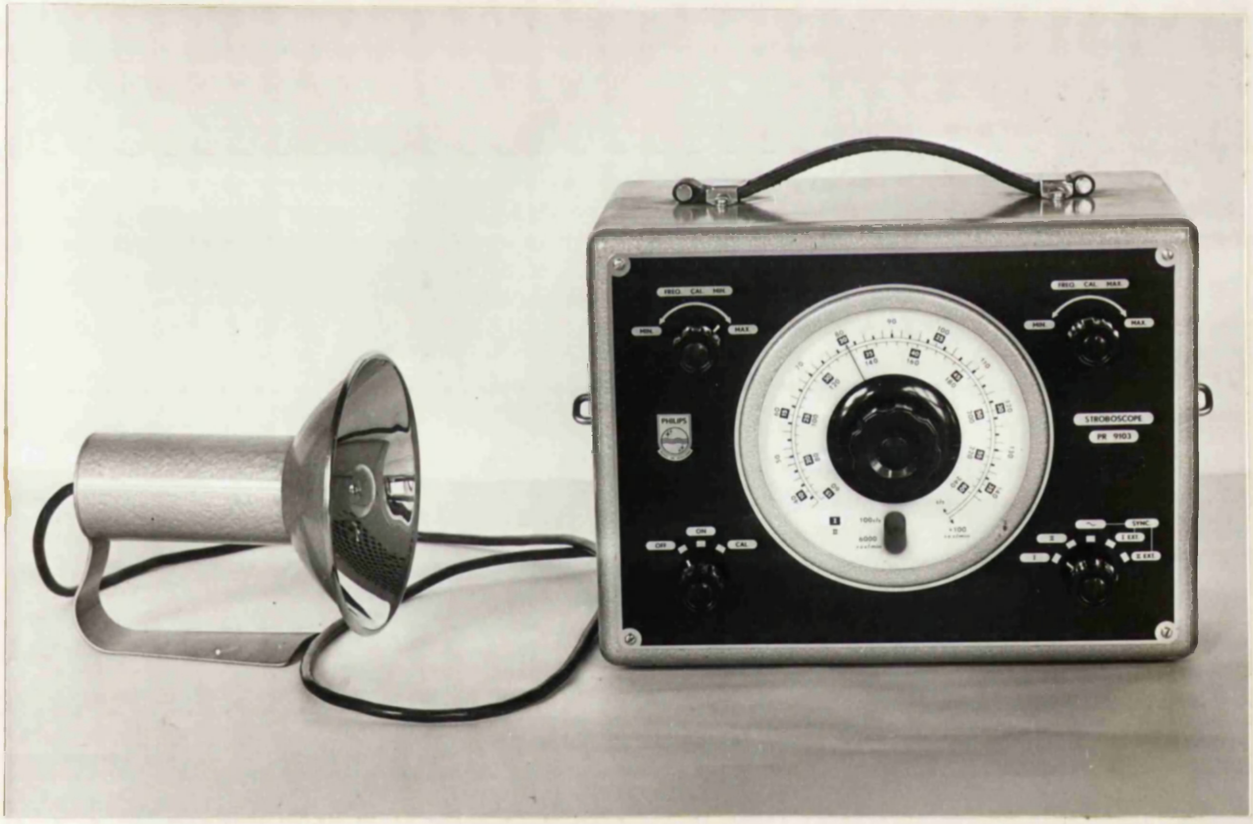


PLATE IIC: Portable Stroboscope

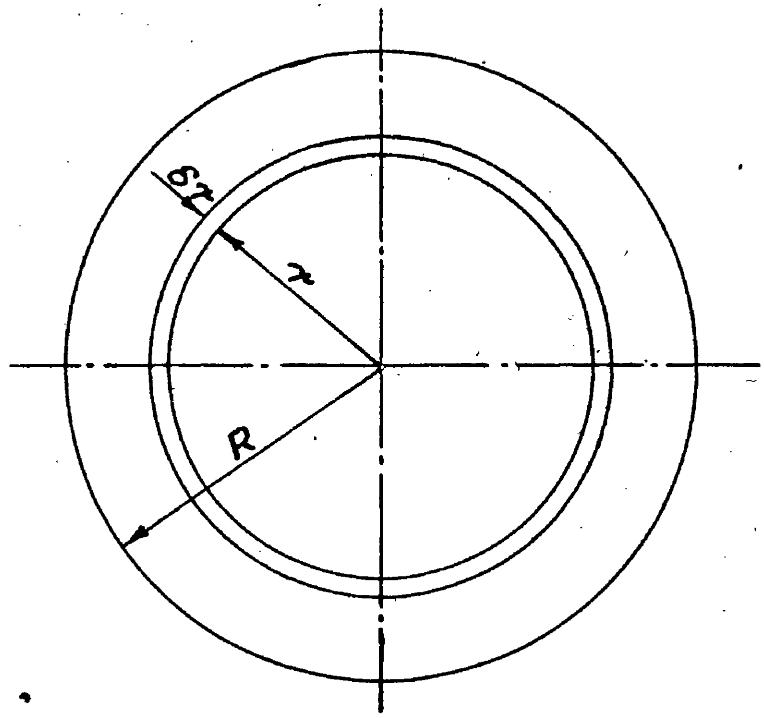


FIG. C.1

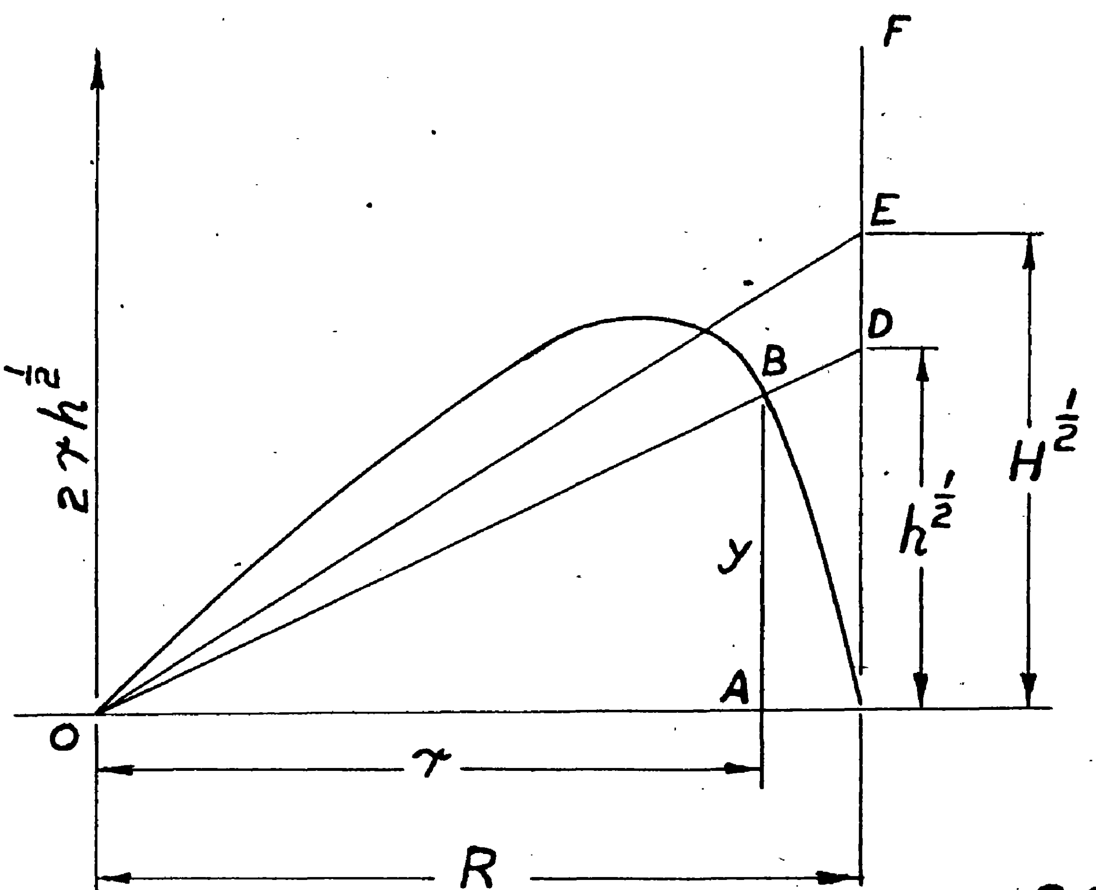


FIG. C.2

manometer.

Therefore,

$$\pi R^2 k Hm^{\frac{1}{2}} = \int k(2 \pi r)h^{\frac{1}{2}}.dr \text{ ....(2)}$$

British Standard Code B.S.1042: 1943(7) gave details of a method of evaluating $Hm^{\frac{1}{2}}$ for circular ducts but it required very accurate placing of the pitot-static tube. Instead a method recorded by Eason (44) was used (see Fig. C2).

The curve is drawn as shown so that,

$$\text{Area under curve} = \int 2 rh^{\frac{1}{2}}.dr$$

A pitot tube reading h, is taken at a distance r = OA from the centre of the duct. A perpendicular is erected at A to meet the curve at B, and the line OB extended to intercept the normal line from C at D. The square root of the velocity head existing at radius r is now represented by CD. Triangle OEC is then constructed so that it is equal in area to the area under the curve. Then CE represents the square root of the mean velocity head, in the same units as h.

The proof is as follows:-

$$R^2 Hm^{\frac{1}{2}} = \int 2 rh^{\frac{1}{2}}. dr \text{ (from equation (2))}$$

$$AB = y = (r/R)CD = rh^{\frac{1}{2}}/R$$

$$2\int y.dr = \int (2rh^{\frac{1}{2}}/R)dr = 2 \text{ area OBC} = 2 \text{ X triangle OEC}$$

But,

$$RH^{\frac{1}{2}} = 2 \text{ X triangle OEC}$$

Therefore,

$$RH^{\frac{1}{2}} = R^2 \overset{\frac{1}{2}}{Hm}/R$$

i.e.

$$H^{\frac{1}{2}} = Hm^{\frac{1}{2}}$$

Also, for constant density conditions,

$$V \text{ mean}/V \text{ centre} = (H \text{ mean})^{\frac{1}{2}}/(H \text{ centre})^{\frac{1}{2}}$$

The cone-shutter was set so that what was estimated to be half the maximum quantity of air was flowing in the duct. At a tapping 6 inches downstream from joint 13/14 (see Fig. C3) a pitot-static tube, connected by $\frac{1}{4}$ inch bore rubber tubing to an Askania Minimeter, was moved along a diameter, readings of velocity head being taken at the same radial positions (0.5 in. increments) on either side of the centre point. The procedure was repeated along a perpendicular diameter.

For each traverse the mean of the two values of the square root of the velocity head existing at the same distance from the centre, multiplied by twice this distance, was plotted to a base of radius. Applying the expression just derived the ratio of mean velocity to centre velocity was obtained. The values for the two diameters were found to agree within 5% so that the exigencies of Code 1042: 1943 were satisfied in this respect. The mean of the four values of $h^{\frac{1}{2}}$ existing at the same radial distance was determined and Graph C1 drawn.

The required ratio was evaluated as follows:-

$$\text{Area under the curve} = 22.1 \text{ square inches}$$

$$= \tfrac{1}{2}H^{\frac{1}{2}} \times R$$

$$\therefore H^{\frac{1}{2}} = 7.85 \text{ inches.}$$

At radius $r = 3.0$ inches, $h^{\frac{1}{2}} = 7.23$

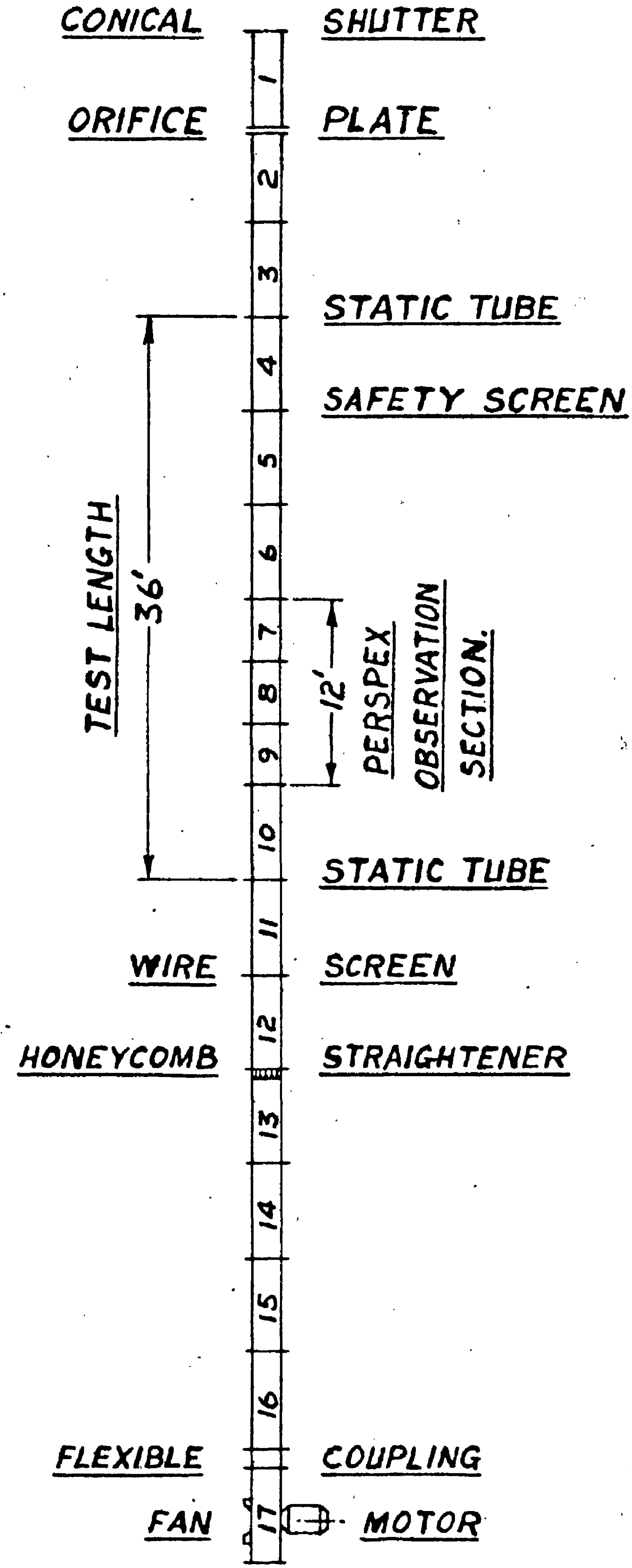
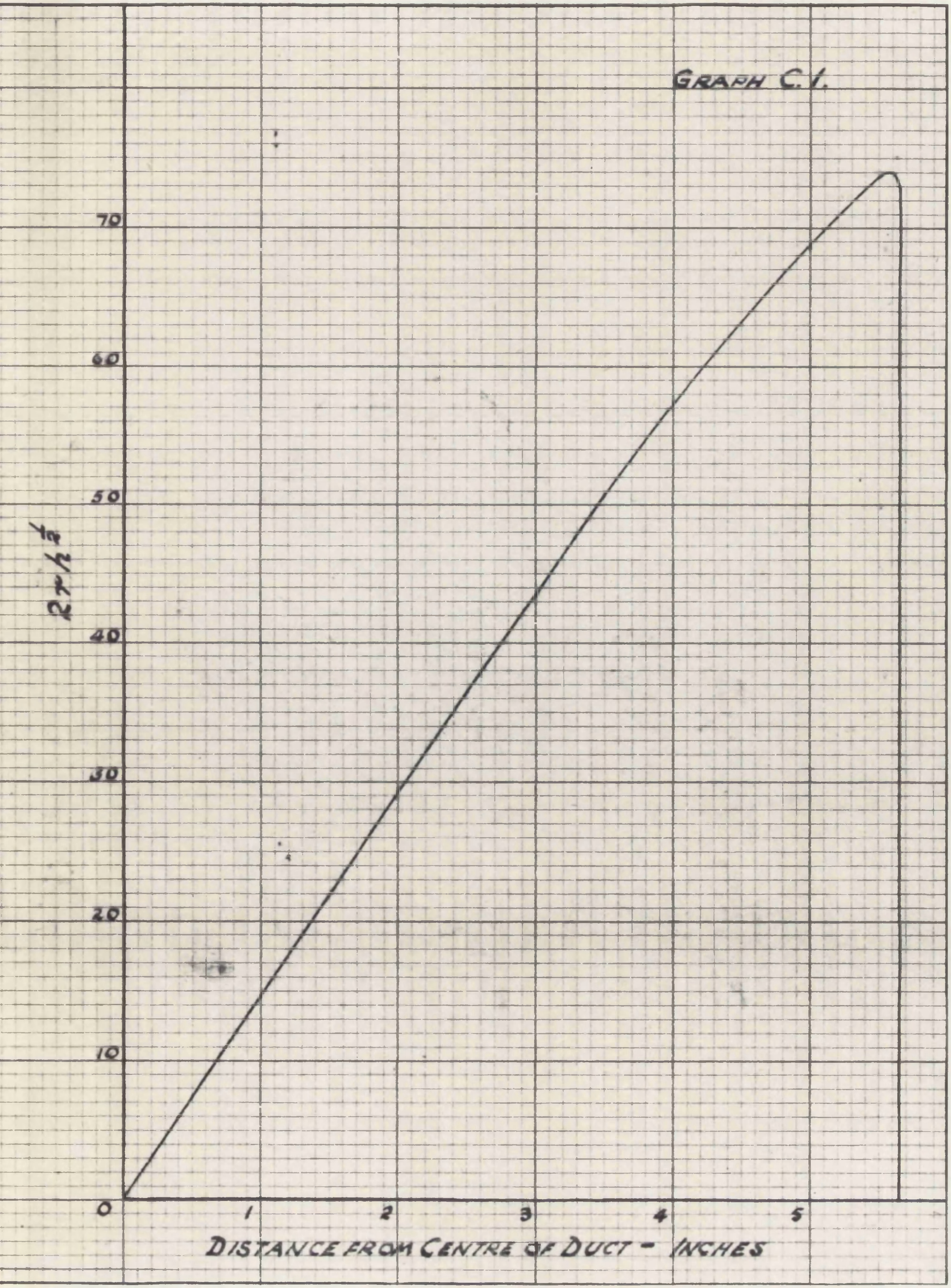


FIG. C.3

WIND TUNNEL LAYOUT.


GRAPH C.1.
70
60
50
40
30
20
10
0
1
2
3
4
5
DISTANCE FROM CENTRE OF DUCT - INCHES

$$y = 43.4/10 = 4.34 \text{ inches}$$

$$\text{Length representing } h^{\frac{1}{2}} = CD = (4.34 \times 5.625)/3.0$$

$$\text{i.e. } CD = 8.15 \text{ inches.}$$

If 8.15 inches represent a $h^{\frac{1}{2}}$ value of 7.23,

then 7.85 represents a $h^{\frac{1}{2}}$ value of $(7.23 \times 7.85)/8.15$

$$= 6.96$$

Therefore,

$$V \text{ mean}/V \text{ centre} = 6.96/7.35 = 0.95.$$

This value differs widely from what would normally be expected (45) and indicated a marked degree of asymmetry in the distribution of velocity across the duct. Since the fluctuations in velocity pressure were of such a magnitude as to render very accurate readings impossible the ratio of V mean to V centre was taken as being constant over the test range. Any additional error in the estimation of mean quantity, introduced by this assumption, was assumed to be negligible.

For each position of the cone-shutter the velocity head existing at the centre was measured and the mean velocity over the cross-section calculated. Since the area of the cross-section was known, the mean quantity flowing in the duct could be determined.

Barometric Pressure and Temperature

Readings of these two items were obtained in the same way as described in Section B (see page 33.).

Characteristic Curves

Total and Static Head-Quantity: This curve compares the total and static heads produced by a fan at constant speed and under stated air density conditions with the quantity of air passing through it. A barometric pressure corresponding to 30 inches of mercury and a temperature of 60° F were the standard conditions, giving an air density of 0.0764 lb. per cubic foot. The reference speed was taken as 2800 R.P.M., this being approximately the mean speed encountered. Total and static heads along with the quantity passing through the fan were measured and reduced to the standard conditions.

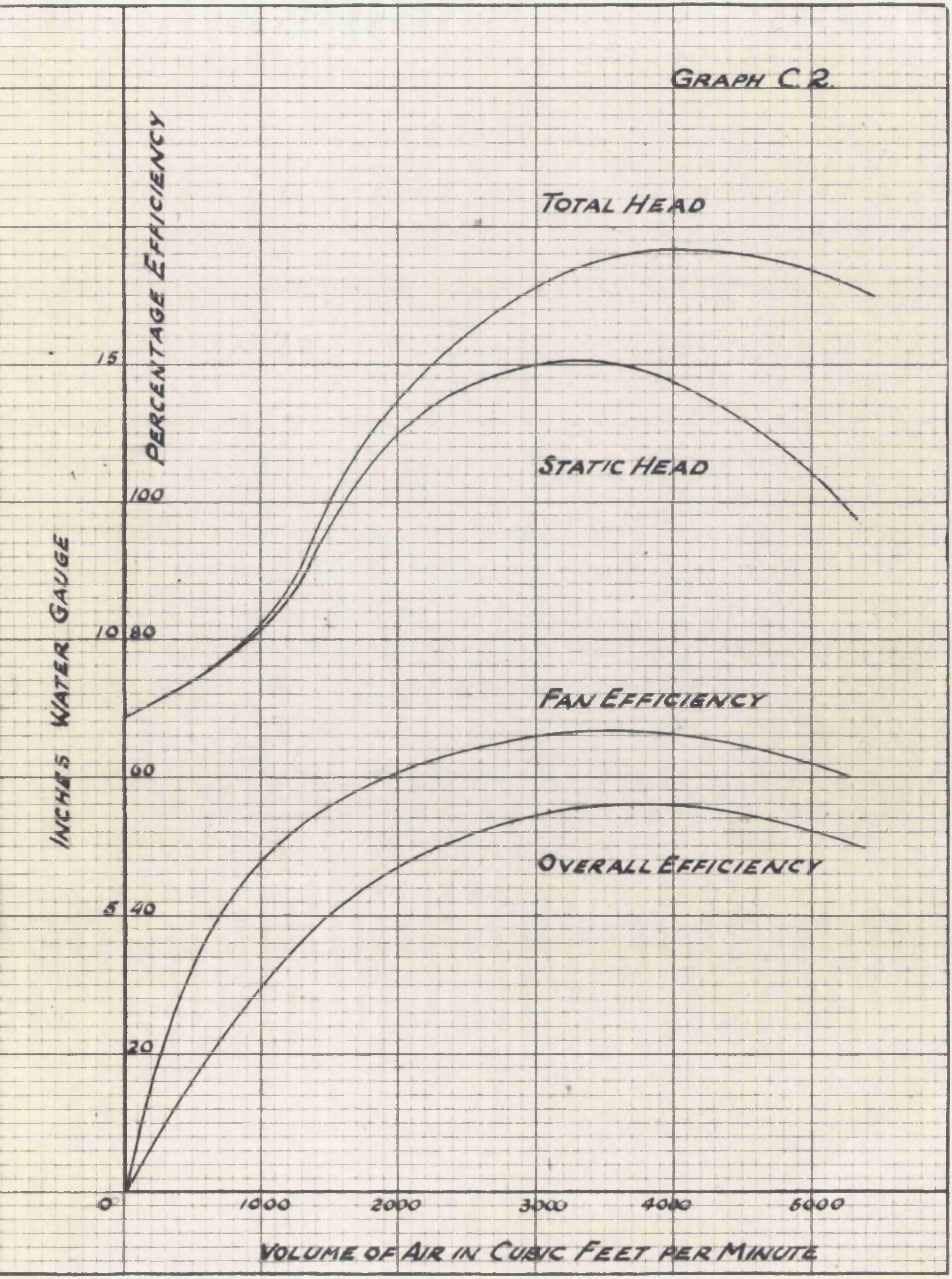
The rest of the curves were obtained in the manner described in the reference (42) and a complete set of characteristics was drawn (see Graph C2). From the graph it was noted that 14 - 15 inches of water gauge (static head) would be available for experimental purposes. The flat top region of the Static Head-Quantity curve indicated that this amount of energy would persist for quantities in the range 2,000 - 4,500 cubic feet per minute. The Fan Efficiencies obtained agreed with the maker's claims and indicated that the installation was working satisfactorily.

MODIFICATIONS TO WIND-TUNNEL

Before proceeding with the investigations the wind-tunnel arrangement was modified as shown in Fig. C3.

Observation Length

During the investigations it was proposed to use an overall test length of 36 feet and measure the pressure loss in this by means of pitot-static tubes. It was considered that due to the nature of the



GRAPH C.2.
PERCENTAGE EFFICIENCY
INCHES WATER GAUGE
TOTAL HEAD
STATIC HEAD
FAN EFFICIENCY
OVERALL EFFICIENCY
15
100
10
80
60
5
40
20
0
1000
2000
3000
4000
5000
VOLUME OF AIR IN CUBIC FEET PER MINUTE

model arrangements to be studied, their accommodation in a transparent section of wind-tunnel would be desirable. Therefore a 12-feet Perspex Observation Length was inserted into the Test Length, six feet from its upstream limit, as shown in the diagram. This meant that the pitot-static tubes would be unlikely to be affected by anything contained in the transparent part of the tunnel.

Wire Screen

A wire-mesh screen having 20 strands per inch and a porosity of 0.463 (see page 37) was used to even out the velocity distribution over the duct cross-section. From Graph B1 it was estimated that a gauze of this type would produce a loss in water gauge of about 2.0 inches when the quantity of air flowing in the duct was 4,000 cubic feet per minute. Because of the large static head developed by the fan this loss could be readily tolerated in return for the improvement in flow conditions. The screen was inserted for maximum effect on the Test Length at joint 11/12.

Honeycomb Straightener

In order to remove any rotation in the air and so increase the efficiency of the screen, a flow straightener was inserted at junction 12/13. This was made from impregnated paper honeycombing 8 inches thick, and it was estimated (23) that the pressure loss produced would not exceed one inch water gauge when the quantity flowing was 4,000 cubic feet per minute.

Standard Orifice Plate

During the experimental work connected with the fan test it was noted that fluctuations rendered accurate estimation of velocity by pitot-static tube impossible. Therefore it was decided to instal an orifice plate to measure the quantity of air flowing in the duct. The deciding factors in the choice of size of orifice were the magnitude of the differential pressure, which would control the accuracy of measurement, and the loss in water gauge produced by the orifice. A Standard Orifice Plate (see Fig. C4) was designed from details given in reference (7) which would produce a pressure loss of only 40% of the differential head.

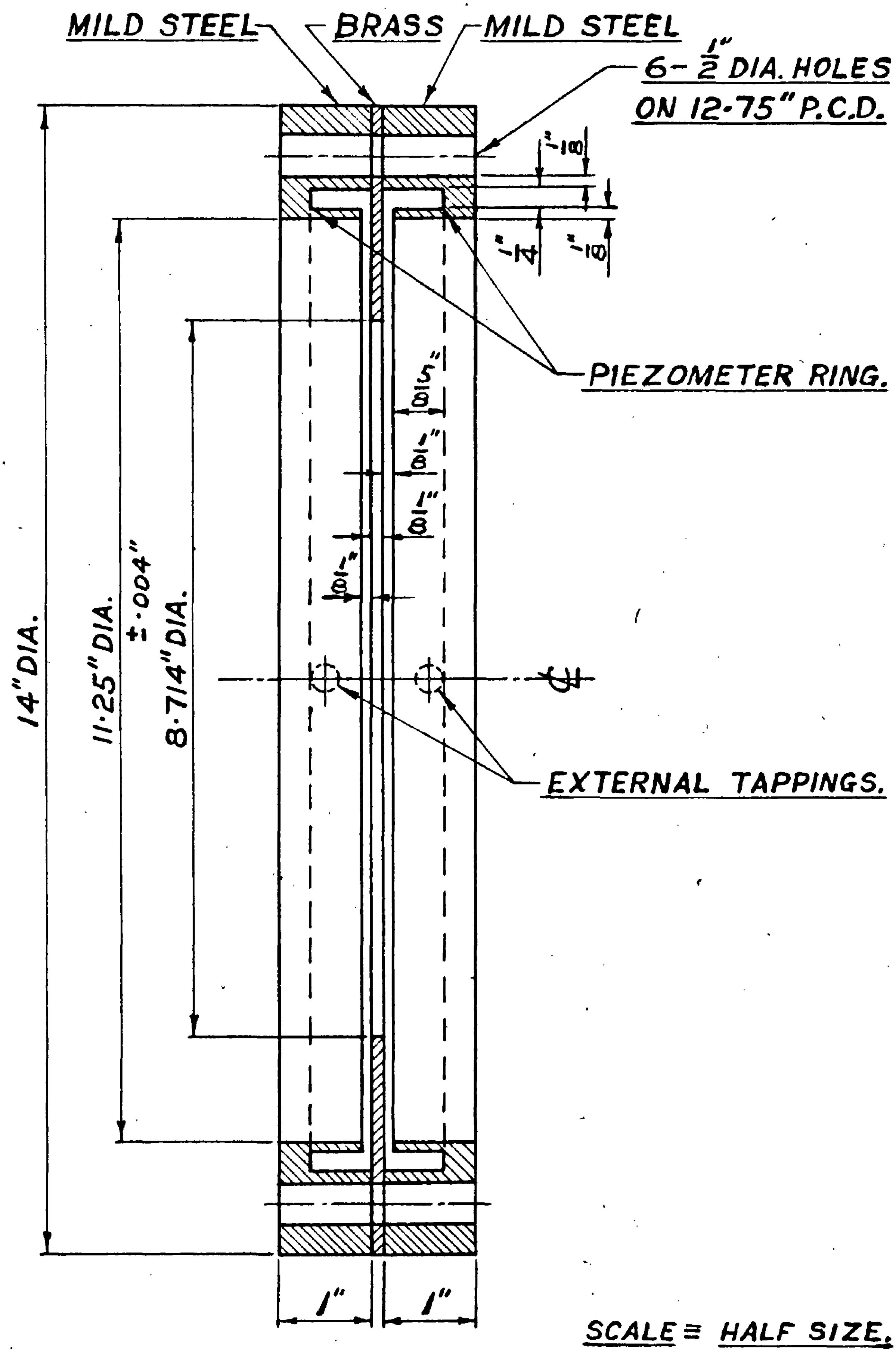
The orifice was constructed from a $\frac{1}{8}$ inch machined flat brass plate with a mild steel ring, containing an annular groove, pla ced on either side. The surfaces of contact were machined and bolt-holes provided to correspond with the flanges of the metal duct. When assembled each annular groove formed a Piezometer Ring, connected to the outside of the arrangement by a tapped hole, which averaged out the static pressure existing across each side of the plate. The orifice was located twelve feet downstream from the Test Length pitot tube so that mutual interference would be avoided. As a result of the design of the conical shutter its effect upon the orifice plate six feet upstream could be considered negligible. The quantity flowing in the duct was found from the formula:

$$Q = 0.0997\ C\ Z\ E\ d^2\ (h/w)^{\frac{1}{2}}$$

where,

Q = rate of flow in cubic feet per second

C = coefficient of discharge; the ratio



ORIFICE PLATE

FIG. C.4

of the actual discharge to the theoretical discharge

Z	=	combined multiplier; the product of three separately specified multipliers for Reynolds Number, duct diameter, and the compressibility of the air.
E	=	velocity of approach factor
	=	$1/(1 - m^2)^{\frac{1}{2}}$
m	=	area of orifice/ area of duct
	=	d^2/D^2
d	=	diameter in inches of orifice
D	=	diameter in inches of duct at high-pressure tapping
h	=	differential pressure in inches of water, measured at 60° F under air
w	=	density of the air in lb. per cubic foot, under working conditions at the position of the high-pressure tapping.

The values of C and Z were obtained from the reference. Applying the constant factors for the orifice:

$$C = 0.5925$$

$$d = 8.714 \text{ inches}$$

$$D = 11.25 \text{ inches}$$

the following formula was obtained:

$$Q = 5.61\ Z\ (h/w)^{\frac{1}{2}}$$

This equation was used throughout the experimental work to relate the differential head measured across the orifice, to the quantity of air flowing in the wind-tunnel. It was noted (46) that the value of the coefficient C could only be assumed reliable for Reynolds Numbers, referred to the duct, greater than 225,000. In effect this meant that the formula quoted above could only be applied when the differential pressure across the orifice was above 2 inches water gauge.

CHECK ON FLOW CONDITIONS

The following tests were carried out to discover whether the honeycomb flow straightener and the wire screen were having the desired effect.

Swirl

An indicator was constructed from a piece of soft wire netting. To each junction in the netting four-feet long rayon thread streamers were attached. This device was inserted in the Observation Length at junction 8/9. The disposition of the streamers at different air velocities was noted and indicated that no appreciable swirl was present.

Distribution of Velocity

Since, for constant density conditions the velocity of the air at any point in the duct is directly proportional to the square root of the corresponding height of water column, these values can be utilised to indicate the velocity distribution. A procedure similar to that described under Fan Test (see page 71.) was adopted, a special pitot-static tube arrangement being used (see Plate IIIC).

A $\frac{1}{4}$-inch outside diameter N.P.L. pitot-static tube passes through an airtight gland fitted with an external rubber sealing ring. The

PLATE IIIC: Pitot-Tube Traversing-Carriage stem of the pitot-tube is fixed to a traversing carriage which can be clamped at any position along a 0 - 13 inch scale. A vernier scale, reaching to 0.01 inch, is engraved on the carriage. The whole arrangement can be rotated through ninety degrees on either side of a reference position and the inclination determined, by means of a vernier, to the nearest 0.4 degrees. The gland fits into either of two drilled bosses welded to one of the steel sections. These bosses are accurately placed in the same plane normal to the long axis of the section and on two perpendicular diameters.

Graph C3 demonstrates the nature of the velocity distribution obtained during a horizontal (H) and a vertical (V) traverse. Although some asymmetry is indicated the displacement of the maximum velocity from the centre of the duct is not very marked. The theoretical velocity distribution (K) postulated by Karman (47) was found from the relation:

$$h^{\frac{1}{2}}/h_c^{\frac{1}{2}} = (1 - r/R)^{1/7}$$

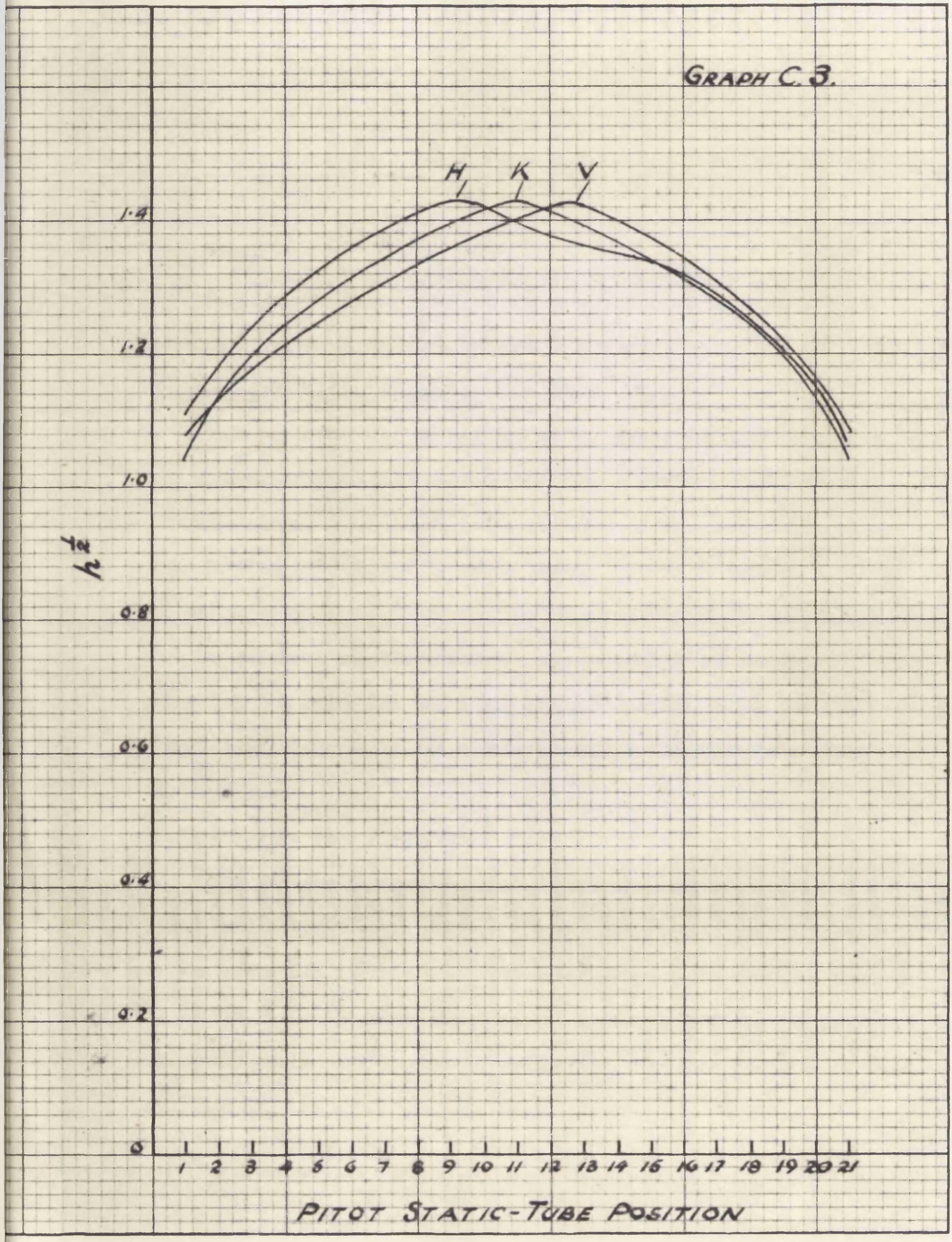
where,

h = velocity head at radius r

h_c = velocity head at the centre

R = radius of the duct.

From the profiles of all three curves it could be assumed that the distribution of velocity was very nearly normal. However, it was decided that, if possible, a check should be made at a later stage in the investigations on any effects produced by the slight asymmetry



GRAPH C. 3.
H
K
V
1·4
1·2
1·0
0·8
0·6
0·4
0·2
0
$\frac{h}{h}$
1 2 3 4 5 6 7 8 9 10 11 12 13 14 15 16 17 18 19 20 21
PITOT STATIC-TUBE POSITION

present.

MANOMETERS

Details are given below of the water manometers used in this section which have not already been described.

Chattock-Fry Tilting Manometer (see Plate IVC).

This manometer can be applied to the measurement of pressures and pressure differences corresponding to a maximum of 0.6 inches of water gauge. The least scale division of the instrument represents 0.00065 inch change in water level. Since it depends solely on the density of the fluid and the measurements of its dimensions for its calibration, the manometer can be used as an absolute standard. A hair-line was added at right angles to the one fitted inside the telescope by the manufacturers, to facillitate reading the instrument.

Inclined Gauge (see Plate VC).

This instrument was constructed from the component parts of a fixed inclination U-tube gauge. It was designed to enable pressures in the range 0.5 inch to 14.5 inch water gauge to be measured directly to within 1%. The range which can be measured, and the factor by which the scale reading must be multiplied, for each position of the inclined arm are shown in Table No.1.

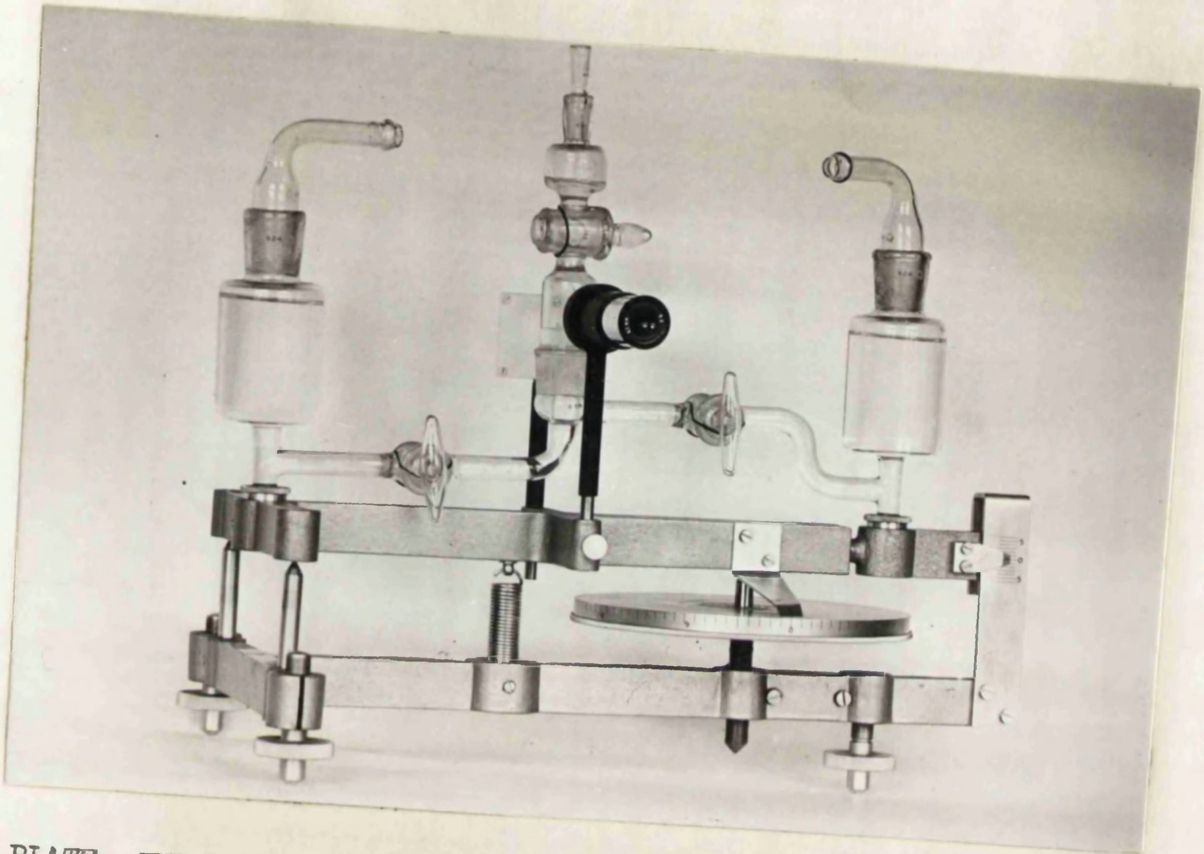


PLATE IVC: Chattock-Fry Tilting Manometer

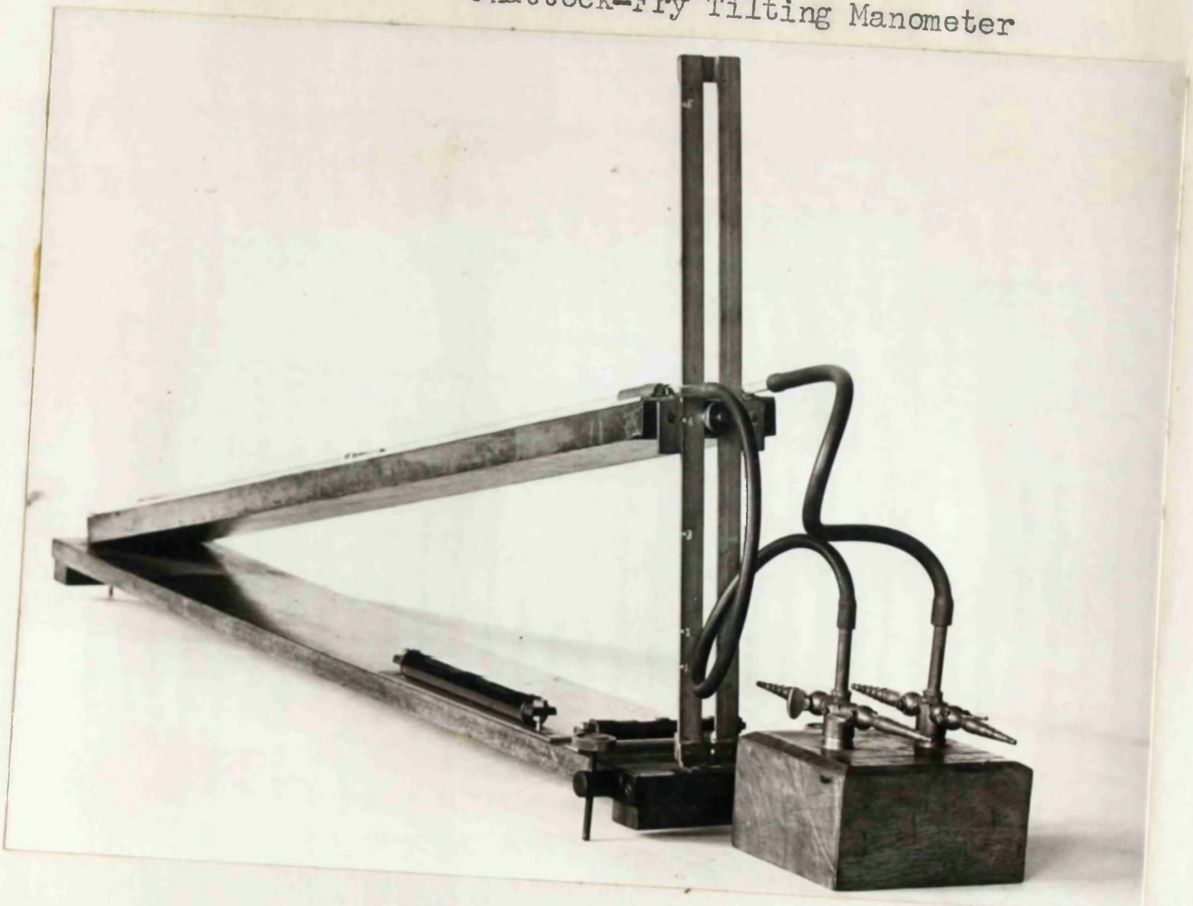


PLATE VC: Inclined Gauge

TABLE No.1

Position	Range (in. Water Gauge)	Multiplying Factor
1	0.50 - 1.65	0.050
2	0.75 - 2.50	0.075
3	1.45 - 4.75	0.144
4	2.25 - 7.35	0.223
5	4.40 - 14.50	0.438

Direct-lift Micromanometer (see Plate VIC)

This manometer was constructed at the Mining Department of the University of Nottingham. A fiducial mark is engraved on the glass tube, the inclination of which can be adjusted between 0 - 15 degrees to give varying degrees of sensitivity. The vertical scale is graduated in 0.1 inch from 0.10 inches and each division of the micrometer head represents 0.001 inch. The reliability of the instrument as an absolute standard depends on how accurately the micrometer thread is cut. If the pressure difference being measured is fluctuating, the fact that reference is made to one water level only, where the meniscus is directly visible, enables a good estimation of the mean position to be obtained.

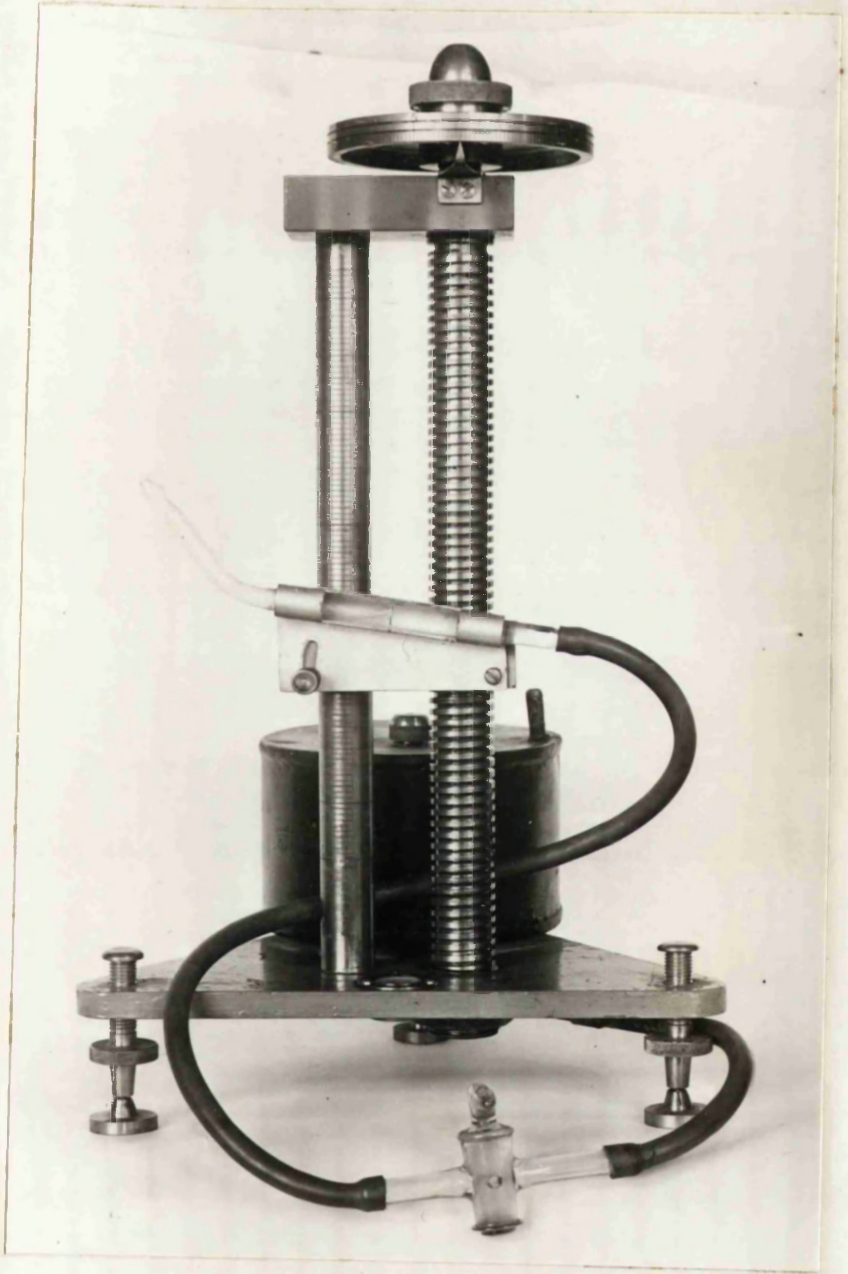


PLATE VIC: Direct-lift Micromanometer

INVESTIGATIONS

1. Relationship between k and the Number of Decks

In this experiment the connection between the Pressure Drop Coefficient (k) of a cage of given design and the number of decks it contained was examined.

Construction of Model Cages and Support Frame: The cage dimensions were defined in terms of the cage width W (see Fig. C5) so that the model experiments could be readily correlated to geometrically similar full-scale shaft arrangements. A range of values of the ratio shaft diameter to cage width, and the ratios of length of cage (L) to cage width, most likely to be met with in an actual single cage installation were used. For each length and width it was necessary to establish the size of the standard mine car or tub (48) which could be carried by each cage so that its height could be estimated. Allowances for deck thickness and overhang had also to be obtained (49).

The models were constructed using $\frac{1}{8}$ inch hardboard for the sides and plywood for the decks and cage top. The thickness of the plywood corresponded to the deck-thickness of the full-scale cage.

A frame was designed to support the models in the Perspex ducting. It consisted of two 46 inch guides of $\frac{1}{4}$ inch box-section brass channel, placed diametrically opposite in the Perspex section (see Fig. C6), with two rings of $\frac{1}{8}$ inch by $\frac{3}{4}$ inch brass strip soldered to them 44 inches apart. The support frame was anchored by two lugs which passed between the flanges through slots cut in the rubber gasket. Cross-pieces of

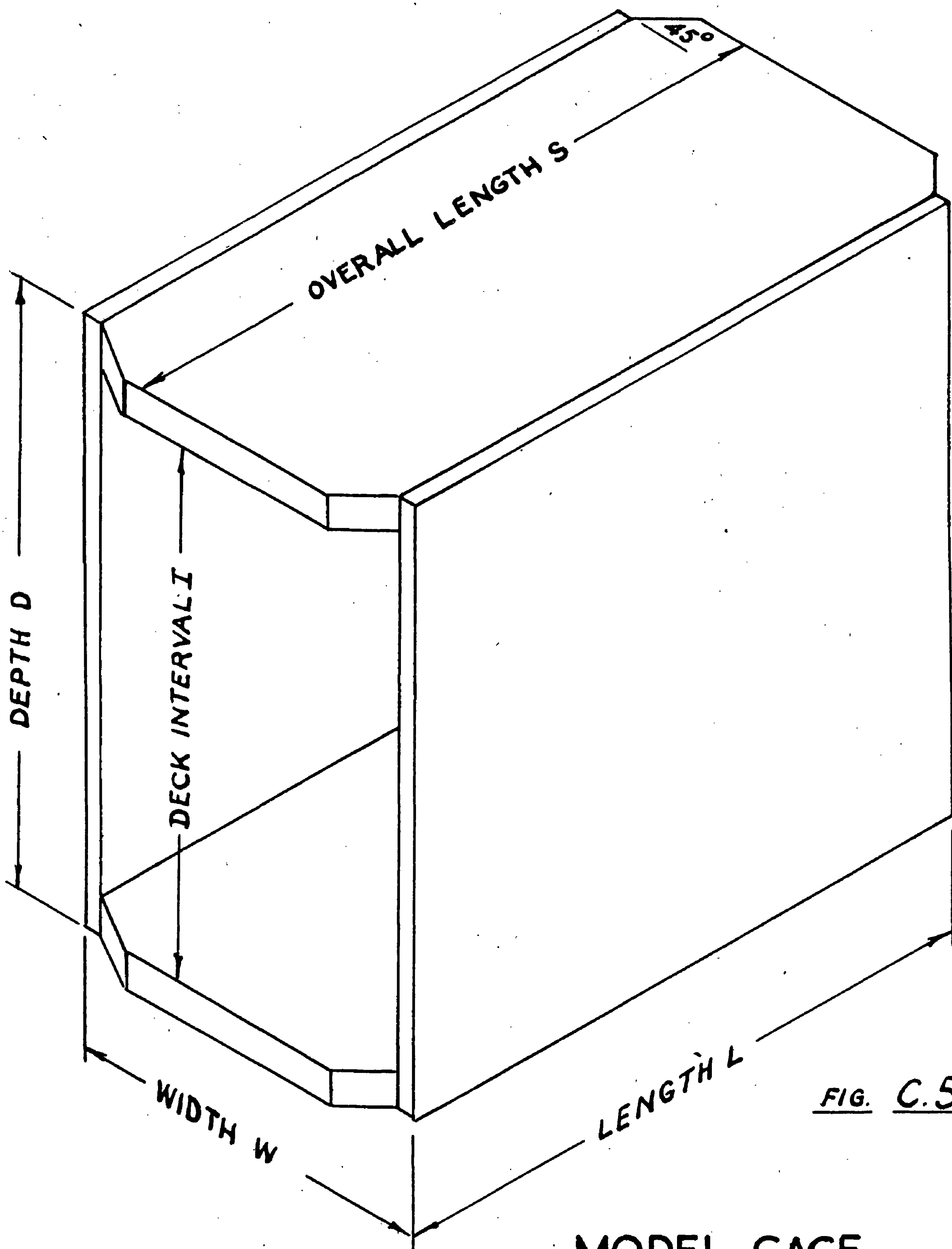
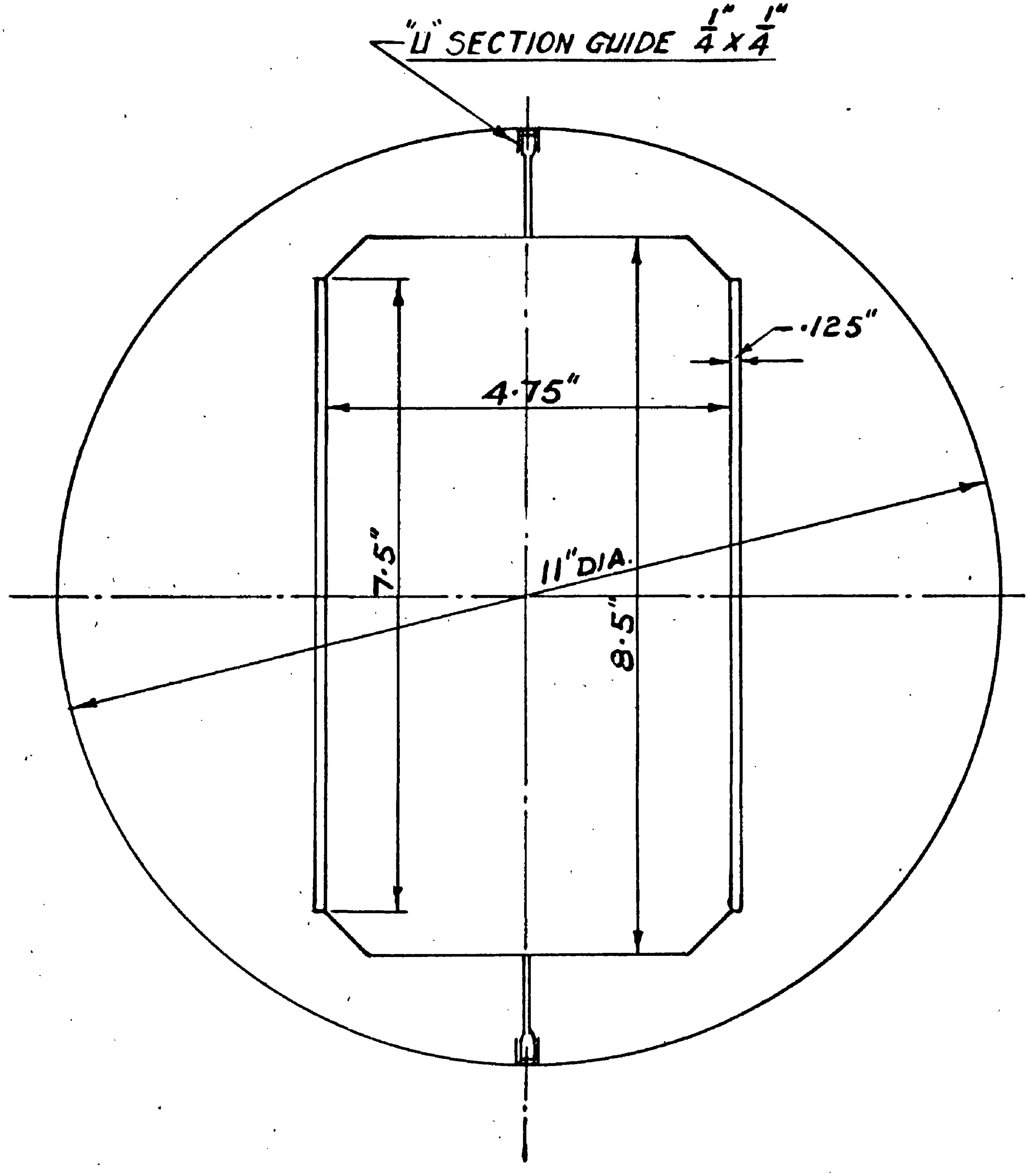


FIG. C.5.

MODEL CAGE.



SCALE ≡ HALF SIZE.

FIG. C.6

SINGLE CAGE ARRANGEMENT.

the brass box-section were bolted to the roof and floor of the cages, and located respectively inside and outside the model. The parts of these arms in the air-stream were flattened so that their effect on the resistance of the cage could be neglected.

Experimental Procedure: First of all the relation between the quantity of air flowing in the wind-tunnel, and the pressure loss produced by the Test Length with the support frame in position, had to be obtained. The air-flow was varied using the cone-throttling device so that the differential pressure produced across the orifice lay in the range corresponding to 2 - 9 inches water gauge. The orifice drop was measured by the Inclined Gauge, and the loss in pressure over the Test Length calculated from the difference in the static head existing at the start and finish. These measurement were obtained using the Chattock-Fry Tilting Manometer and the Askania Minimeter. The latter instrument was used to determine readings outside, and to check the values already obtained inside the range of the former. The experiment was repeated on different days when the density of the air had altered appreciably.

Cages of series A1 and A6 (see Appendix C) were constructed. Both series were geometrically similar and contained models of one, two, three and four deck design. The length of each model and the distance between decks was one and a half times the width. For series A1 the shaft diameter was two and a quarter times the width, while for series A6 the ratio was $3\frac{1}{3}$. The loss in pressure produced by a cage placed in the Test Length was found in the following way.

The model was anchored in the support frame; using the outlet throttle, the air quantity was varied to give readings across the orifice in the range 2 inches water gauge to maximum. The maximum value, determined by the type of cage and the number of decks, was about 6 inches for model A1/4 and 7 inches for model A6/1. For the larger set of cages the orifice tappings and the static tubes were connected to the T-piece manifold of the Inclined Gauge, and it was found possible to measure the pressure loss in the Test Length and the drop across the orifice at the same inclination. For the smaller series, however, the Askania Minimeter was used to find the drop in the Test Length in order to prevent the number of alterations to the slope of the Inclined Gauge from becoming excessive.

Before investigating the resistances of these two series of cages the ppssibility of the effect of asymmetrical velocity distribution was examined. This was done by rotating the middle section of the Perspex length successively through 60 degrees (i.e. corresponding to the pitch circle spacing of the flange holes) and obtaining the P.D.C.s for a single-deck cage of each type. The variation in k was less than 2% so the distribution of velocity was considered satisfactory.

A further preliminary in the case of the larger models was the establishment of a stable zone in the Perspex section between the support rings. Using cage A1/1 constant readings were obtained over a 30 inch length commencing 4 inches from the upstream support. All the models of series A1 were located so that the top of the cage was

6 inches from the upstream flange.

Calculation of k: The pressure loss due to the test length plus the support frame in inches water gauge was plotted against the orifice differential pressure in the same units (see Graph C4). Over the test range the relationship was discovered to be a straight line independent of changes in air density. Hence it was decided to keep the calibration curve in this form.

From the definition of Pressure Drop Coefficient (k) we get the relation

$$k = 2P/pV^2$$

where

P = pressure loss due to the cage in lb. per square foot

p = density of air in slugs per cubic foot

and V = mean velocity of approach of the air in feet per second.

Now $P = 5.2\, h_c$, $p = \frac{w}{g}$, and $V = Q/A = 5.61\, Z\, (h_o/w)^{\frac{1}{2}}/A$

where h_c = loss produced by the cage in inches water gauge

w = specific weight of the air in lb. per cubic foot

Q = quantity flowing at the test section in cubic feet per second

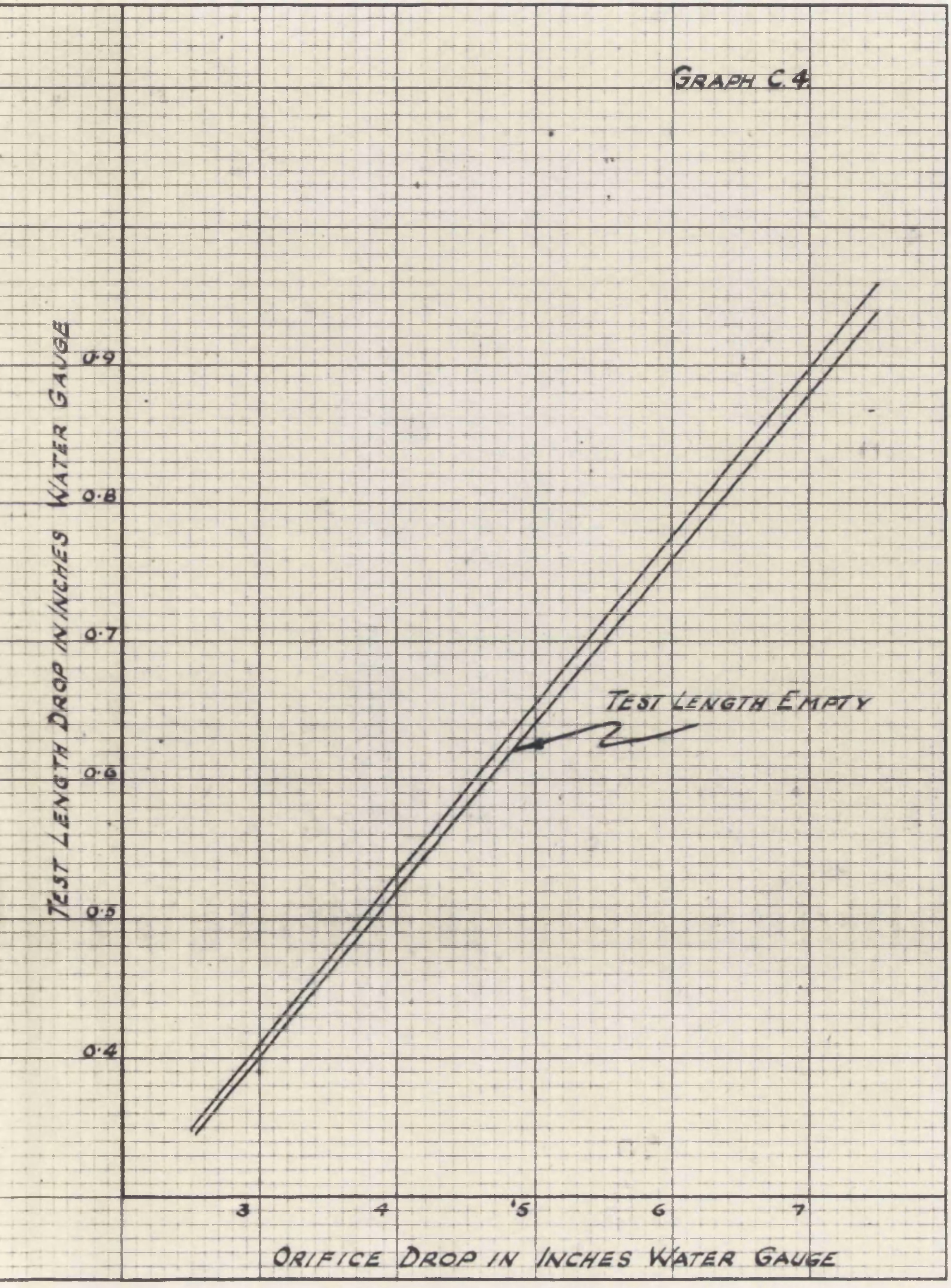
A = area of the test section in square feet

Z = multiplying factor (see page 76.)

h_o = difference in pressure across the orifice in inches water gauge.

Therefore,

$$k = (10.63\, A^2/Z^2)\,(h_c/h_o)$$


GRAPH C.4.
TEST LENGTH EMPTY
TEST LENGTH DROP IN INCHES WATER GAUGE
0·9
0·8
0·7
0·6
0·5
0·4
3
4
5
6
7
ORIFICE DROP IN INCHES WATER GAUGE

Substituting the cross-sectional area of the transparent section gives

$$k = (4.63/Z^2)\ (h_c/h_o).$$

Since h_o was directly measurable and h_c nearly so, it was evident that using the above relationship was the simplest way of deducing k. However it still remained to correlate Z with the values of h_o.

Examination of the relevant data (7) showed that Z varied approximately from 1.01 to 0.99 as h_o increased from 2 - 7 inches water gauge. Moreover the value of h_o where Z equalled unity was indeterminable. Since the total change in Z^2 was of the order of 4% it was considered that this factor could not be ignored. Consequently an attempt was made to relate Z to the pressure drop at the orifice.

The section of ducting used with the special pitot-static tube arrangement was inserted at position 3 (see Fig. C3) so that there would be minimum leakage between it and the orifice plate. For different quantities the ratio of mean velocity to centre velocity was obtained using the method described under Fan Test (see page 71). The orifice drop was measured for each rate of flow. From the pitot-static tube observations the mean quantity was derived and the substitution of this in the orifice formula enabled the factor Z to be evaluated. The method was far too inaccurate to be of any use, however, as it will be appreciated that the total change in Z is only about 2%.

Examination of the experimental results showed random fluctuation of 1 - 2% about the mean in the values of h_c/h_o for h_o in the range 3 - 7 inches water gauge (Rn 300,000 - 450,000) with no definite tendency to either increase or decrease. In the range 2 - 3 inches water gauge the results were markedly different from the rest but here there was an obvious upward trend in the value of h_c/h_o. This was consistent with the tendency for Z to increase with decrease in the quantity flowing, which would be inclined to keep k constant. For the range in which the random variation was experienced Z was assumed to be unity, and k was obtained by multiplying the arithmetic mean of the values of h_c/h_o by 4.63. The P.D.C. was calculated for each cage and so the desired relationship could be studied, but it must be remembered that there is the possibility that k will alter with large changes in Rn.

The results are shown in TableNo.2 and No.3.

TABLE No.2

Cage Serial	A1/1	A1/2	A1/3	A1/4
Number of Decks	1	2	3	4
Pressure Drop Coefficient k	2.05	2.46	2.84	3.16

TABLE No.3

Cage Serial	A6/1	A6/2	A6/3	A6/4
Number of Decks	1	2	3	4
Pressure Drop Coefficient k	0.138	0.230	0.338	.430

<u>Discussion of Results</u>: Graphs C5 and C6 illustrate the relationship between k and the number of decks which is of the form obtained by Miller and Bryan (50) for tubs standing buffered in a mine roadway. This indicates that for any particular series the overall drop in pressure is composed of a frontal and wake shock loss and a loss due to friction and eddying. The loss due to friction includes that produced by the increase in the velocity of the air flowing over the part of the internal surface of the duct affected by the cage. For each series it was found that the loss due to shock was constant while that due to friction and eddying was directly proportional to the number of decks.

The relationship may be stated algebraically as follows:-

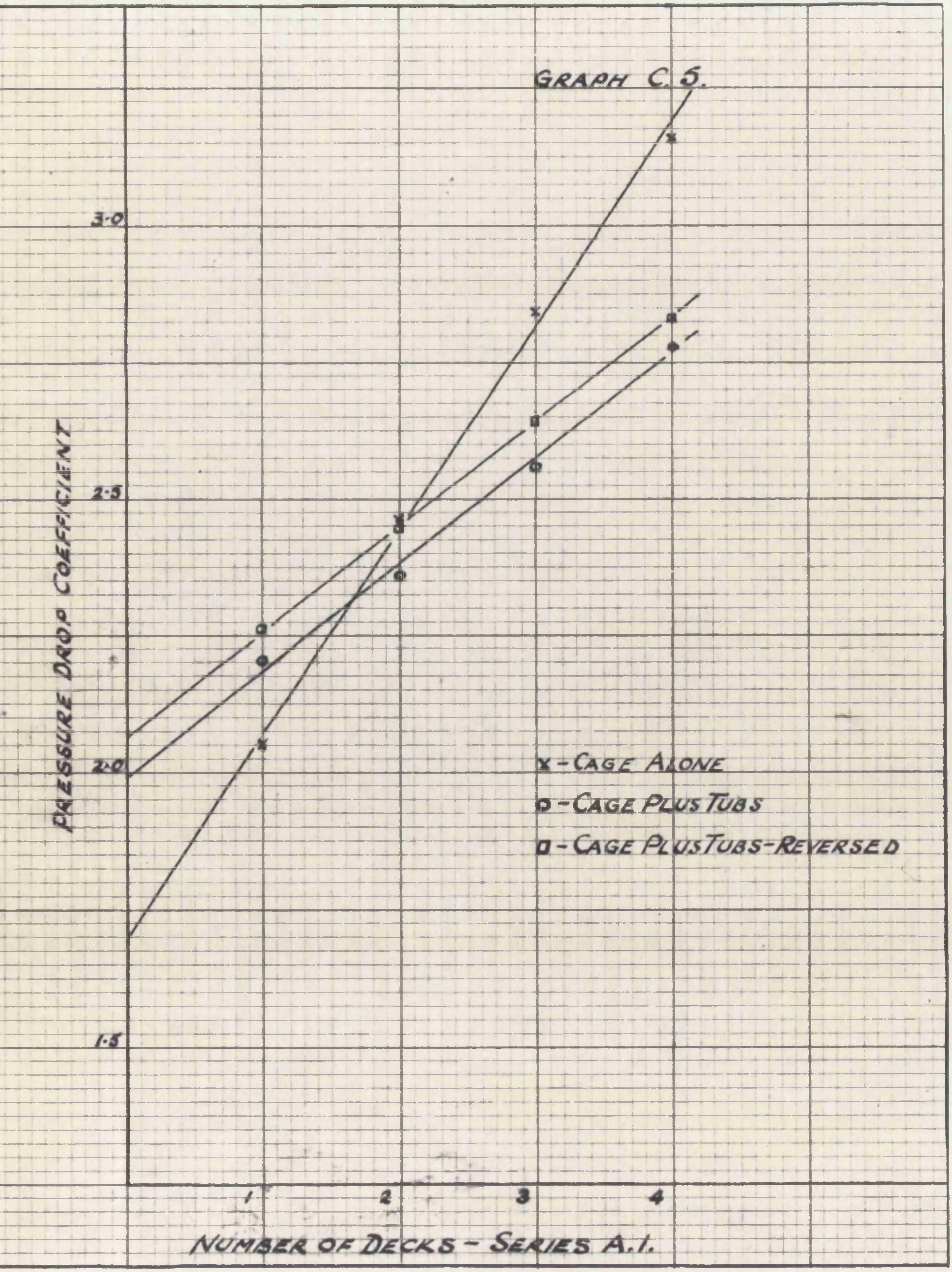
Series A1 - $k = 1.69 + 0.37\,N$

where N = number of decks

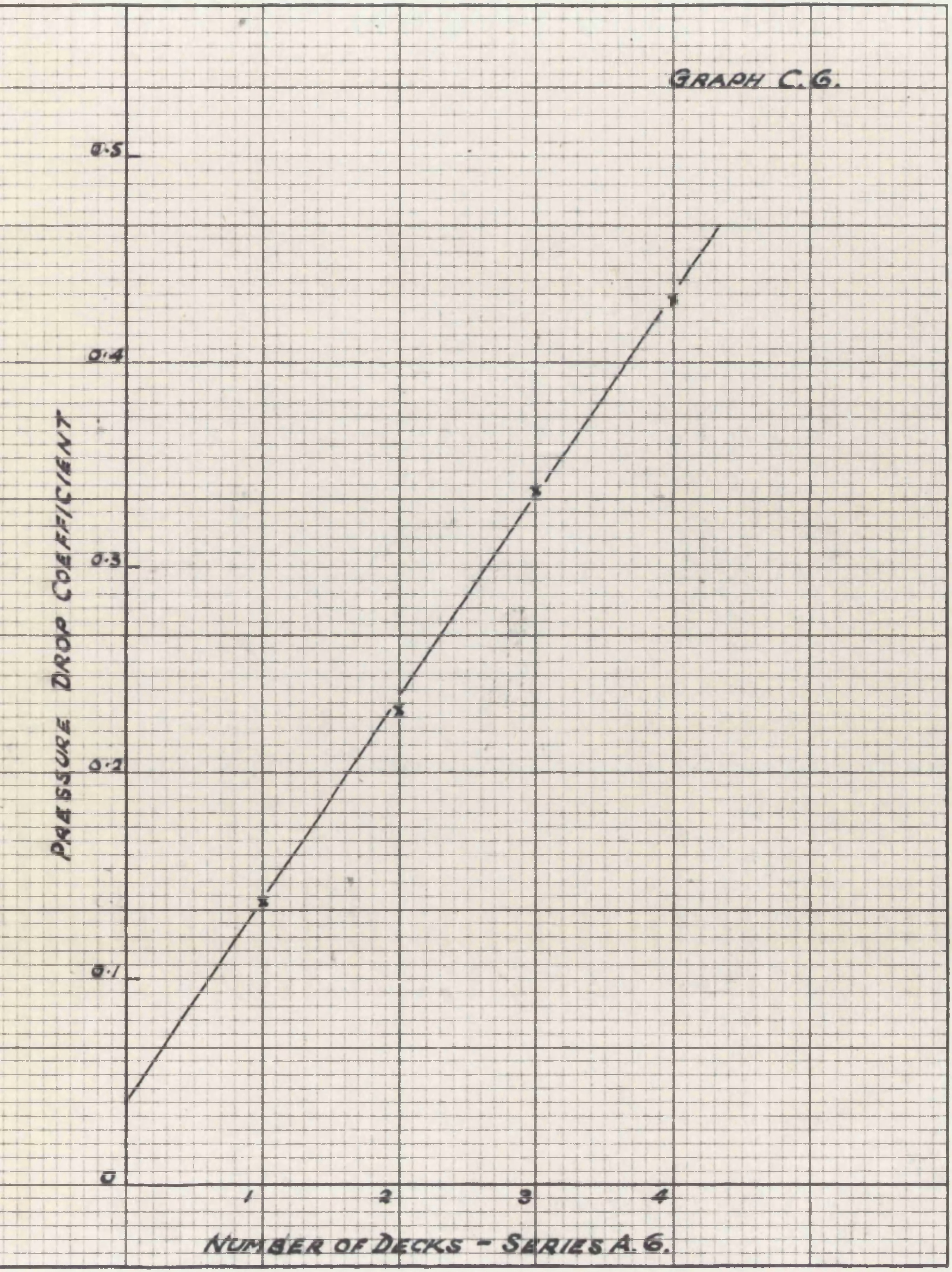
Series A6 - $k = 0.04 + 0.10\,N.$

<u>Conclusions</u>: From the results it may be assumed that for any normal design of cage the relationship between the Pressure Drop Coefficient and the number of decks will be of the form already indicated. It is reasonable to suppose that this will also hold for full-size cages. Hence, if for a particular installation, whether model or full-scale, the value of k can be obtained for two cages each having a different number of decks, the P.D.C. for any other number of decks can be calculated.

The benefit derived from the flat characteristic of the fan is demonstrated by the maximum orifice drops possible with models A1/4 and A6/1. The



GRAPH C. 5.
3·0
2·5
2·0
1·5
PRESSURE DROP COEFFICIENT
x - CAGE ALONE
o - CAGE PLUS TUBS
□ - CAGE PLUS TUBS - REVERSED
1
2
3
4
NUMBER OF DECKS - SERIES A.1.



GRAPH C.6.
PRESSURE DROP COEFFICIENT
0.5
0.4
0.3
0.2
0.1
0
1
2
3
4
NUMBER OF DECKS - SERIES A.6.

difference between the two maximum values is about 1 inch water gauge representing a change in air quantity of only 250 cubic feet per minute.

Although an extra hair-line had been added to the telescope of the Chattock-Fry Tilting Manometer it was still extremely difficult to ascertain the mean position of the top of the bubble during operation because of fluctuations. Much of the refinement of reading was lost and it would appear that only a direct reading instrument will remedy this.

2. Effect of Loading the Cage with Tubs

The open sides of the cages of series A1 were boxed-in for 2/3 of their height, to simulate the conditions produced by a cage loaded with tubs. The procedure adopted was the same as that described in section 1, and the P.D.C.s were determined for the series and also for the models reversed relative to the air-flow. Table No.4 gives a comparison of the results with the coefficients obtained minus the tubs.

TABLE No.4

Cage Serial	A1/1	A1/2	A1/3	A1/4
k for cage minus tubs	2.05	2.46	2.84	3.16
k for cage plus tubs	2.20	2.36	2.56	2.78
k for cage plus tubs - reversed	2.26	2.45	2.64	2.83

<u>Discussion of Results</u>: The results of this experiment are also shown in Graph C5 from which the following observations may be made:-

(a) The addition of sides reduces the amount of energy lost due to eddying in the space between the decks.

(b) This reduction is also obtained when the cages are reversed although it is not quite so great. In this case the position of the sides is such that they are not so effective in limiting the amount of air entering between the decks and setting up eddies.

(c) With a single-deck model the increase in rubbing friction due to the sides offsets their effect to some extent.

(d) That the presence of sides affects the constant shock losses is evident from the different intercepts on the vertical axis.

(e) For a four-deck cage an appreciable saving is produced by the sides even when the cage is reversed.

In conclusion, however, it should be pointed out that in practice the effect of tubs cannot be expected to be quite so marked as flush built-in sides.

3. <u>Effect of Increasing the Coefficient of Fill</u>

The Coefficient of Fill is defined as the ratio of the frontal cross-sectional area of the cage to the cross-sectional area of the shaft. The object of this experiment was to derive the variation in k as this ratio was increased.

Four single-deck cages geometrically similar to series A1 and A6, but intermediate in size, were constructed. The P.D.C.s were obtained using the same procedure as before with a slight exception.

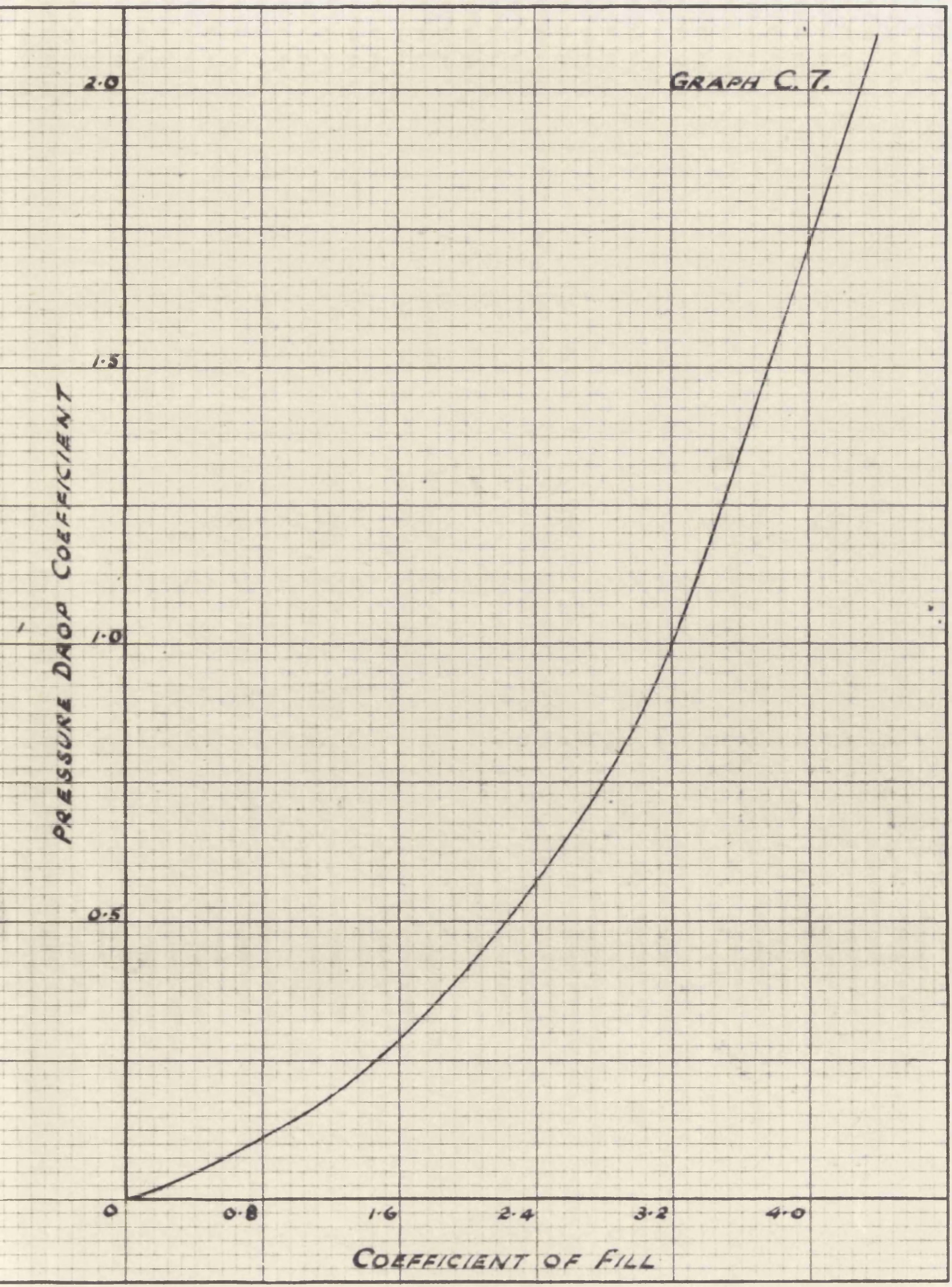
At this juncture in the experimental work the Direct-lift Micromanometer became available. This proved ideal for measuring the pressure loss over the Test Length since a wide variation in values was experienced as a result of the size range of the models. In all the tests this manometer was used to obtain the pressure drop between the static tubes while the Inclined Gauge was used to determine the orifice differential pressure. Using this arrangement enabled both readings to be taken simultaneously thus increasing the accuracy of the measurements.

The results are shown in Table No.5.

TABLE No.5

Cage Serial	A6/1	A5/1	A4/1	A3/1	A2/1	A1/1
Frontal Area of Cage (sq. in.)	8.4	14.4	22.8	29.5	37.3	41.3
Coefficient of Fill	0.088	0.152	0.240	0.310	0.393	0.435
Pressure Drop Coefficient	0.138	0.270	0.574	0.926	1.70	2.05

Discussion of Results: Graph C7 illustrates the rapid increase in k that takes place for values of Coefficient of Fill greater than 0.3. Work done on the drag of underground trains (51), however, did not show this marked rise until the corresponding figure was 0.5. The reason for this is most likely due to the fact that the trains would be considerably longer than the cages and so the pressure drop caused by


GRAPH C. 7.
PRESSURE DROP COEFFICIENT
2·0
1·5
1·0
0·5
0
0·8
1·6
2·4
3·2
4·0
COEFFICIENT OF FILL

friction would be proportionately greater. Thus the loss depending principally on the Coefficient of Fill (i.e. the frontal and wake shock loss) would not have the same influence upon the Pressure Drop Coefficient.

4. Effect of Cage Shape

From the work on Regulators it was fully expected that for a Coefficient of Fill above 0.3 the recovery of velocity energy during expansion at the rear of the cage would be very small, and hence the shape of the contracted area through which the air had to pass would be of no consequence. However, when one of the plywood decks of model A1/1 was suspended at right-angles to the air-stream it was found to have a P.D.C. of 2.75 which is far in excess of the value for the complete cage. This suggested that with the incorporation of sides some of the eddying had been eliminated and also that the energy loss due to expansion had been reduced. Therefore it was decided to investigate the effect that changing the frontal profile had on the value of k.

It seemed reasonable to assume that in addition to any effect attributable to change in shape, the P.D.C. of any single-deck model would also depend on:-

(a) the frontal area,

(b) the area affected by friction, i.e. the sides of the cage along with the surface of the shaft over which an increase in velocity due to the model took place,

(c) the distance between the roof and the floor.

It will be evident that it was impossible to maintain these three factors constant while effecting a change in the frontal profile. There was the possibility, however, that the distance between the floor and the roof of the cage would not prove too critical as a result of the shadow or "delle" effect (52). A reduction in this distance would be required when changing the profile from square to rectangular in order to keep items (a) and (b) constant.

As a result of the method and materials of construction, close control of factors (a) and (b) was only possible in the bigger models. Two cages B1/1 and F1/1 with the ratio L/W nominally equal to unity, and having Coefficients of Fill of 0.461 and 0.400 respectively, were assembled. Their P.D.C.s were obtained and again when the distance of the floor from the roof was decreased by 0.5 inch, the length of the sides remaining unaltered. When the cages were modified the value of k was reduced, but by less than 2% in both cases. The construction of the rest of the models was then undertaken, the difference of the minimum and maximum distances between decks of each series being less than 0.5 inch. The results are shown in Table No.6 and No.7.

The profile of the front of each cage is denoted by the Shape Factor which is defined as the ratio of the perimeter to the square root of the cross-sectional area. By multiplying the internal circumference of the duct by the length of the model the Duct Rubbing Surface was evaluated.

TABLE No.6

Cage Serial	B1/1	C1/1	D1/1	E1/1
Shape Factor	3.87	3.89	3.95	3.98
Length/Width	1.00	1.20	1.38	1.49
Frontal Area (sq. in.)	43.9	44.1	43.7	44.0
Area of Cage Sides (sq. in.)	105	113	118	122
Duct Rubbing Surface (sq. in.)	292	288	280	277
Total Rubbing Surface (sq. in.)	397	401	398	399
Pressure Drop Coefficient	2.66	2.67	2.66	2.53

TABLE No.7

Cage Serial	F1/1	G1/1	H1/1	I1/1
Shape Factor	3.85	3.92	3.94	3.98
Length/Width	1.00	1.19	1.34	1.46
Frontal Area (sq. in.)	38.0	37.6	37.7	37.8
Area of Cage Sides (sq. in.)	96	105	110	113
Duct Rubbing Surface (sq. in.)	290	288	283	280
Total Rubbing Surface (sq. in.)	386	393	393	393
Pressure Drop Coefficient	1.90	1.85	1.81	1.70

Discussion of Results; From the Tables it will be noted that in the case of cages B - E the reduction in the value of k is almost but not quite 5%, excluding the moderating effect of the decrease in the distance between the roof and floor. For models F - I the corresponding figure was just over 10%.

Due to the practical limitations already mentioned and also the fact that models smaller than I1/1 are not likely to be used in a single cage installation, this section of the experimental work was not extended. From the results, which are somewhat limited in scope, it may be concluded that a reduction in P.D.C. occurs as the frontal profile alters from square to rectangular. The decrease is not very considerable but is more marked the greater is the change in shape from square, and the smaller is the Coefficient of Fill.

This experiment was concluded by moving models B1/1 and F1/1 to the side of the wind-tunnel, so that in each case the clearance between the corner of the cage and the side of the duct was the minimum recommended by the National Coal Board. The reduction in k was about 2% for the larger model and about 5% for the smaller one. These results are not very significant and while a reduction was to be expected resulting from the velocity distribution, this could have been offset by the increase in the air speed in the narrow space between the models and the side of the duct. The difference in the two figures is probably due to the smaller cage being further from the centre of the duct.

5. Effect of Streamlining

Other workers (38) have shown that incorporation of streamlined section buntons caused a reduction of 55 - 60% in the resistance of a shaft. Since these figures represent a considerable decrease it was decided to find the effect on the P.D.C. of streamlining models in series A1. It was not possible, as in the case of buntons, to obtain a complete streamlined contour, so by way of a compromise the cage was inserted between a nose and a tail fairing. These were constructed to conform to the profile of a Joukowski Symmetrical Aerofoil Section. This Section is obtained from the Conformal Transformation of two-dimensional, irrotational, parallel, streamlined flow past a cylinder (53). The proof is as follows:-

(i) <u>The Joukowski Transformation</u>: The flow w_1 past a cylinder may be defined as the function of a complex variable (z). By the mathematical process of Conformal Transformation flow w_1 may be transfromed to flow w_2, which is again a function of a complex variable (t), past a closed curve. The latter flow retains the properties of irrotation and parallel streamlining.

$$\begin{aligned} w_1 &= f_1(z) \\ &= f_1(x + i\,y) \end{aligned}$$

From the theory of Conformal Transformation it is possible to obtain a complex variable t such that

$$t = f_2(z)$$

and get

$$w_2 = f_3(t)$$

$$= \quad f_3\,(u + i\,v)$$

(Note: Modulus dt/dz must have a unique value other than zero for all points in the region of the z-plane being considered.)

The Joukowski Transformation is given by the relationship

$$t \quad = \quad z + a^2/z$$

where

$$a \quad = \quad \text{a real constant.}$$

The transformation can best be illustrated graphically by considering the relationship of a point in the z-plane to a point in the t-plane (see Fig. C7).

Now

$$z \quad = \quad x + i\,y$$

$$= \quad r\,e^{i\theta}$$

and

$$t \quad = \quad z + a^2/z \text{ (the Joukowski Transformation)}$$

$$= \quad r\,e^{i\theta} + (a^2/r)e^{-i\theta}$$

$$= \quad (r + a^2/r)\cos\theta \; + \; i(r - a^2/r)\sin\theta$$

But

$$t \quad = \quad u + i\,v$$

Therefore

$$u \quad = \quad (r + a^2/r)\cos\theta$$

and

$$v \quad = \quad (r - a^2/r)\sin\theta$$

Thus the locus P' of the Joukowski Transformation Shape can be found for a given locus in the z-plane.

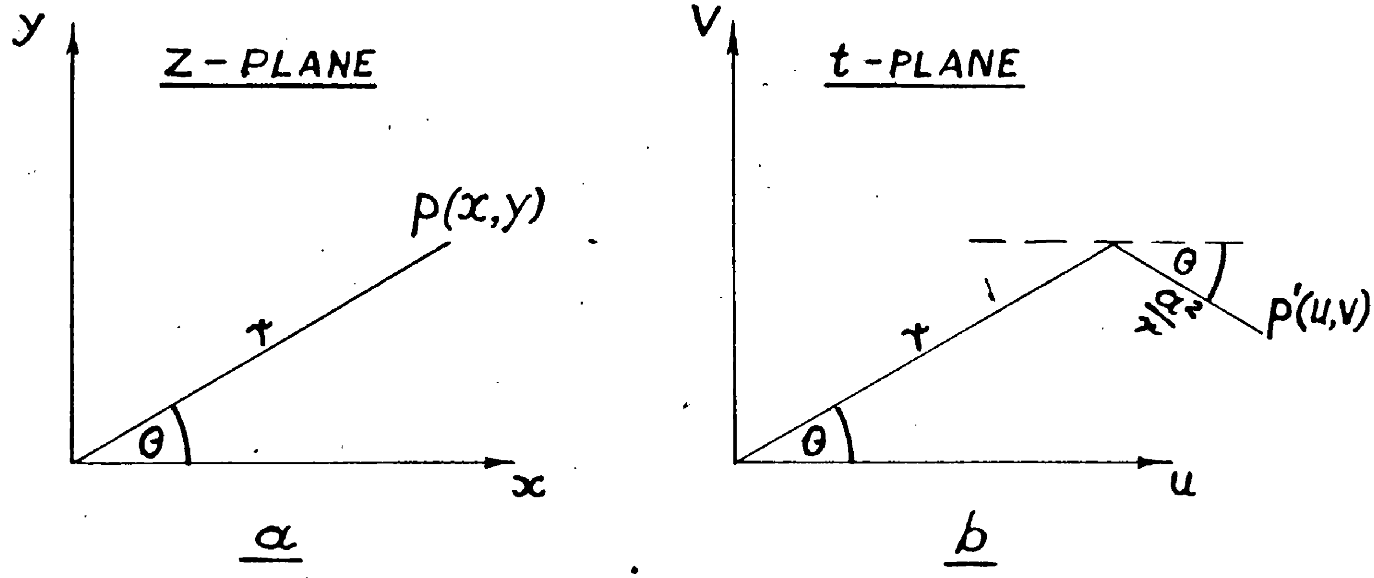

Z-PLANE
t-PLANE
y
x
P(x,y)
r
θ
a
V
u
P'(u,v)
b

FIG. C.7.

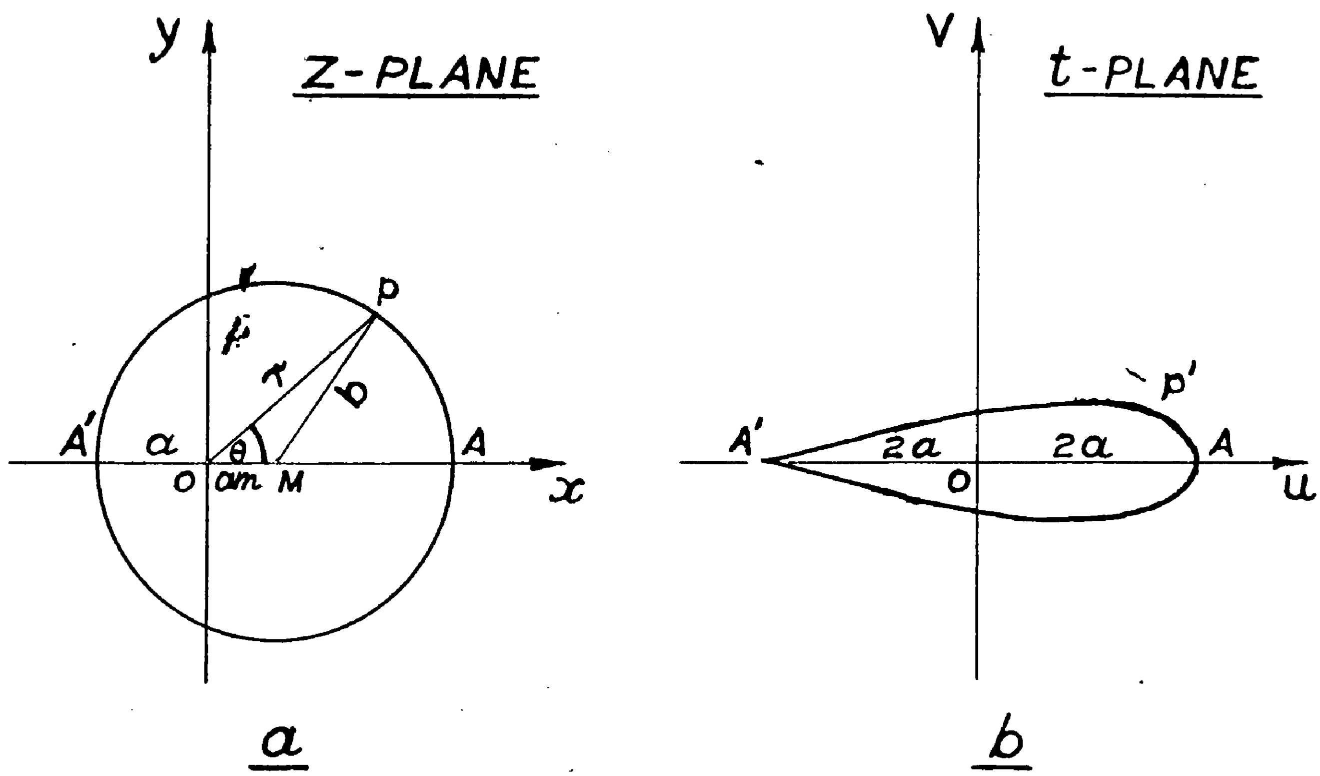

Z-PLANE
t-PLANE
y
x
P
A'
A
a
O
am
M
θ
V
u
p'
2a
2a
O
b

FIG. C.8.

(ii) <u>The Joukowski Symmetrical Aerofoil Section</u>: Consider the locus of P' (see Fig. C8) as P traces out a circle, centre M on Ox, radius b (greater than a) and passing through A'(-a, o).

Let

$$OA' = a$$

and

$$OM = ma \text{ where m is small and positive } (m^2 \text{ and higher powers will be neglected})$$

P is the point (r, θ)

Now

$$b = a + ma$$

From triangle OPM

$$b^2 = r^2 + m^2a^2 - 2\,mar\cos\theta$$

$$= a^2(1 + 2m + m^2)$$

Therefore

$$(r/a)^2 - 2m\cos\theta(r/a) - (1 + 2m) = 0.$$

Since r/a is positive the only relevant root of this quadratic equation is

$$r/a = m\cos\theta + \tfrac{1}{2}\left(4m^2\cos^2\theta + 4(1 + 2m)\right)^{\frac{1}{2}}$$

Expanding binomially and neglecting all terms containing m^2 and higher powers, gives

$$r/a = m\cos\theta + 1 + m$$

Therefore

$$r/a = 1 + m(1 + \cos\theta)$$

and

$$a/r = 1 - m(1 + \cos\theta)$$

In the t-plane

$$u = (r + a^2/r)\cos\theta$$

$$= a(r/a + a/r)\cos\theta$$

$$= 2\,a\cos\theta$$

and

$$v = (r - a^2/r)\sin\theta$$

$$= a(r/a - a/r)\sin\theta$$

$$= 2\,ma\sin\theta\,(1 + \cos\theta)$$

Thus as θ varies from $-\pi$ to π we can trace out the Joukowski Aerofoil in the t-plane. From the last two equations it will be seen that point $\pm\theta$ in the z-plane transforms to point $\pm v$ in the t-plane making the section symmetrical about the u-axis.

In addition

point A ($\theta = 0$) is $u = 2a$, $v = 0$

point A' ($\theta = \pi$) is $u = -2a$, $v = 0$.

Therefore,

Chord length $= 4a$.

The thickness at any point is

$$2v = 4\,ma\sin\theta\,(1 + \cos\theta)$$

and has its maximum value when

$$dv/d\theta = 0.$$

The maximum thickness occurs when

$$\theta = 60 \text{ degrees}$$

which gives

$$u = a.$$

Thus for this section the position of maximum thickness is a quarter of the Chord Length behind the leading edge.

At $u = a$

$$v = (3)^{3/2}\,am/2$$

$$\text{Thickness Ratio} = \frac{\text{Maximum Thickness}}{\text{Chord Length}}$$

$$= \frac{2v}{4a}$$

$$= 1.3\ m.$$

The aerofoil has a rounded leading edge and there is a cusp at the trailing edge.

The profile used during the investigations was the one having the minimum Total Drag (Skin Drag plus Form Drag) (54). The choice was purely arbitrary since it was realised that the Total Drag characteristic would be altered both by the insertion of the cage and by the proximity of the sides of the duct. For the section adopted (see Fig. C9):

$$\frac{\text{Maximum Thickness}}{\text{Chord Length}} = \frac{t\ max.}{c} = 0.2726$$

$$= 1.3\ m.$$

$$\therefore \quad m = 0.21$$

Now

$$t\ max. = \text{Cage Width } (W)$$

$$= 5.0 \text{ inches}$$

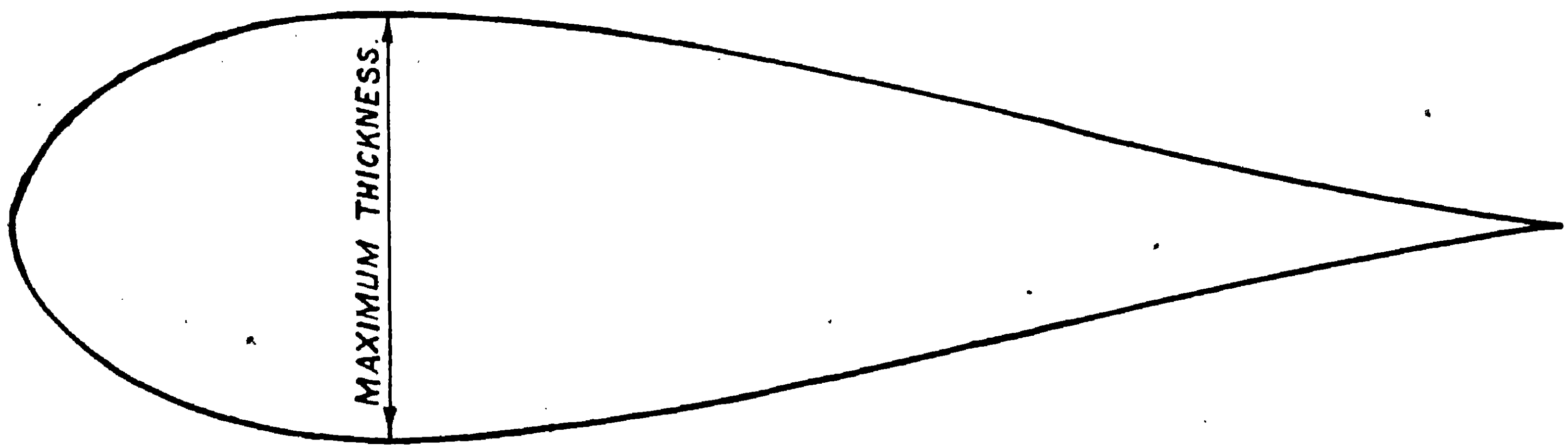


FIG. C.9.

JOUKOWSKI SYMMETRICAL AEROFOIL.

Therefore

$$c = 5.0/0.2726 = 18.32$$

$$= 4a$$

Hence

$$a = 4.58$$

$$\text{and } 2a = 9.16$$

$$\text{and } 2am = 1.92$$

Finally

$$u = 9.16 \cos \theta$$

$$v = 1.92 \sin \theta \, (1 + \cos \theta).$$

The contour was obtained by giving θ different values.

The profile was traced on to two plywood formers and these were divided along the line of maximum thickness. Using the formers as sides, nose and tail streamlined sections were then made from 20 gauge aluminium sheet of breadth equal to cage length L. Plate VIIC shows the fairings attached to model A1/1.

The nose-piece was fixed to the top of model A1/1 and the P.D.C. of the arrangement found. It had been established that, for the centre section of the Perspex observation length, the stable zone began six inches from the upstream flange and existed for thirty inches. The cage was placed in the duct with the leading edge of the nose-piece at the start of the stable length. The P.D.C. was obtained with the tail-piece added to the bottom of the model and again when the nose-section had been removed. It was found possible to place all the arrangements within the stable region with the location of the cage unaltered.

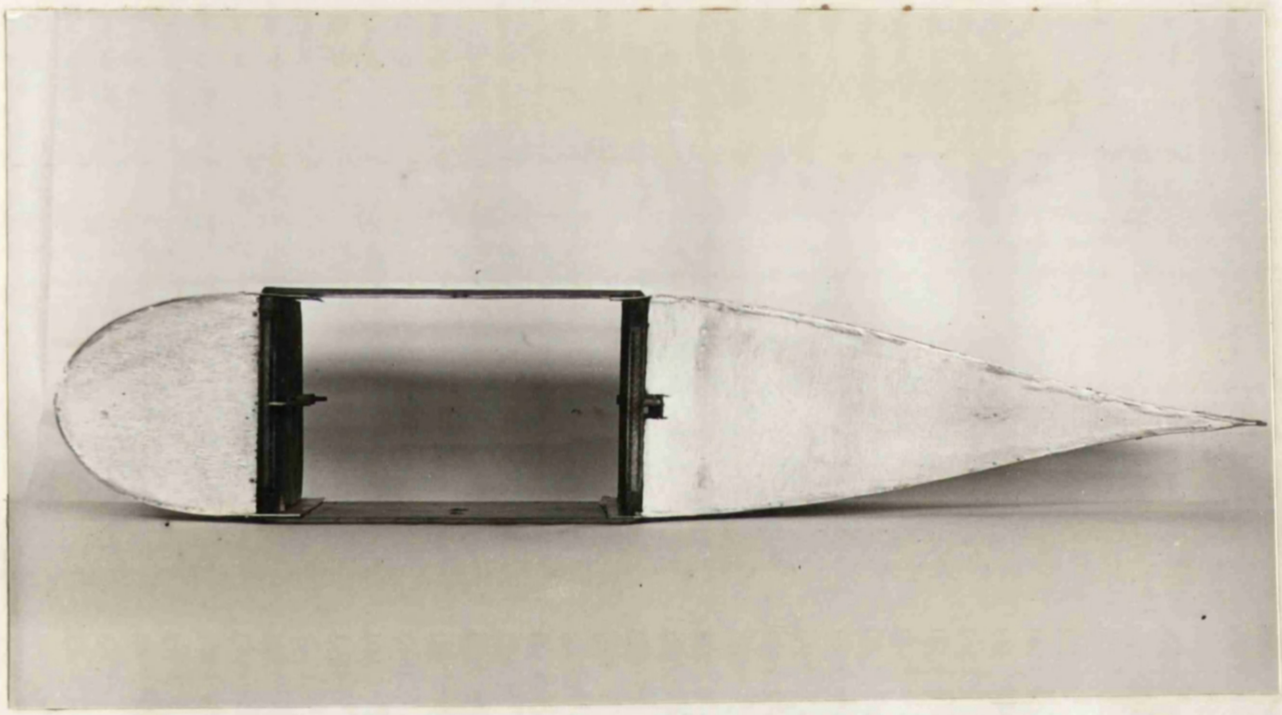


PLATE VIIC: Joukowski Sections attached to Model A1/1

These experiments were repeated with the cage reversed relative to the direction of the air-flow. This time the cusped edge of the tail-section was located six inches from the upstream flange.

The same procedure was adopted using model A1/4 but due to the length of some of the arrangements being greater than the stable zone, the cage was always located at the centre of the support frame.

The results are shown in Table No.8 and No.9, where the following symbols are used:-

SN = cage with nose-fairing attached

ST = cage with tail-fairing attached

R = arrangement reversed relative to the air-flow.

TABLE No.8

Cage Serial Number	Pressure Drop Coefficient						
	Cage Alone	SN	ST	SN + ST	SN(R)	ST(R)	SN + ST(R)
A1/1	2.05	1.53	1.56	0.95	1.64	1.62	1.32
A1/4	3.16	2.46	2.71	2.11	3.08	2.67	2.65

TABLE No.9

Cage Serial Number	Percentage Reductions in P.D.C. for Cage Alone					
	SN	ST	SN + ST	SN(R)	ST(R)	SN + ST(R)
A1/1	25.4	23.9	53.7	20.0	21.0	35.6
A1/4	22.2	14.2	33.2	2.5	15.5	16.1

It could only be expected that the fairings would reduce the loss component due to shock. Hence the percentage reductions shown in Table No.9 do not truly represent the efficiences of the streamlined sections when used with model A1/4, since in this case the shock loss is of less import.

The shock loss constituents of the arrangements used with cage A1/1 were calculated by reasoning as follows.

It was assumed that, for (say) the model plus the streamlined nose, the total pressure loss was the sum of the losses due to shock, friction and eddying produced by the cage, and friction resulting from the fairing. The friction-eddying component for the cage alone has already been found so that

$$k_1 = s + 0.37 + a$$

where s = the shock loss constituent,

a = the fairing friction constituent.

For the case in point

$$1.53 = s + 0.37 + a$$

Therefore

$$s \quad = \quad 1.16 - a.$$

If the nose-section attached to model A1/4 decreased the shock loss to the same value as for the single-deck cage and caused the same additional friction loss then the P.D.C. of the arrangement would be given by

$$\begin{aligned} k_4 &= s + 4 \times 0.37 + a \\ &= 1.16 + 1.48 \\ &= 2.64 \end{aligned}$$

This figure allows the true effect of the fairing in reducing the shock loss associated with both the single and four-deck models to be compared. Corresponding values were derived for the rest of the arrangements and these are shown in Table No.10.

TABLE No.10

Value of k for	Pressure		Drop		Coefficient	
Model A1/4	SN	ST	SN + ST	SN(R)	ST(R)	SN + ST(R)
Experimental	2.46	2.71	2.11	3.08	2.67	2.65
Calculated	2.64	2.67	2.06	2.75	2.73	2.43

Discussion of Results: From Table No.9 it will be noted that, with the exception of cage A1/1 in its normal position, the sum of the reductions produced by the nose and tail fairings independently is greater than the decrease effected when the streamlined sections are used together. Since, in the case of the single-deck model, the difference of the complete arrangement from the theoretical aerofoil profile is not so great the action of the two fairings when used simultaneously is complementary. For the other conditions, however, if it is assumed that the effect of the downstream section on the upstream section is negligible, the results suggest that the effect of the upstream section is to increase the tendency of the air to flow parallel to the sides of the cage. This induces separation at the downstream fairing thereby reducing the streamlining action.

Although in every instance the effect of the streamlining is apparently more pronounced with the single deck cage, reference to Table No.10 shows that:-

(a) the efficiency of the section used to face the oncoming air is greater where model A1/4 is concerned. This is probably due to the fact that the influence of the fairing sides is to reduce the eddying between the decks,

(b) the action of the streamlined tail in the recovery of energy is comparable for the two cages, and

(c) as might be expected, the effect of the nose section when used at the downstream end of the longer model is quite negligible.

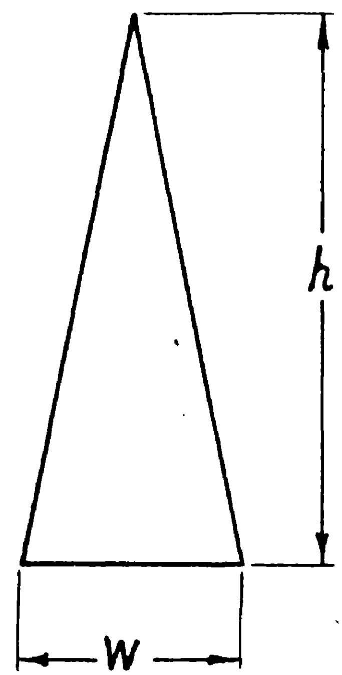
The results indicate that using aerofoil section fairings the shock losses associated with cages of either 1, 2, 3, or 4-deck design can be considerably reduced. The influence on the Pressure Drop Coefficient is of course more significant the smaller the number of decks. However, it is felt that the effect of the support rings was to diminish the efficiency of the fairings when used with model A1/4, and that a greater reduction in k than that actually obtained should have been possible.

6. Effect of Straight-sided Fairings

Although the results of Section 5 indicated that a very considerable decrease in the value of k could be obtained by employing Joukowski Aerofoil nose and tail-sections, it was realised that the manufacture of these to exact specificarion for use with full-size cages would be both complicated and costly. This experiment was carried out to determine if the reduction in P.D.C. effected by straight-sided fairings was comparable, since these could be constructed without undue difficulty.

A number of fairings to fit models in series A1 was cut from 20 gauge aluminium sheet, to form an isosceles triangle (see Fig. C10). The ratio of the altitude of the triangular profile to the width of the base (h/w) is defined as the Aspect Ratio and five fairings were constructed having values in the range 0.5 - 3.0. Detachable ends were made for the fairings from smooth-faced cardboard.

Fairings N1 - N5 were fitted in turn to the front of cage A1/1 and the P.D.C. of the arrangement with and without the paper sides obtained. As a preliminary the P.D.C. of A1/1 itself and the upstream and downstream limit of the zone of stability had been checked. The model was suspended at the downstream limit so that the effect of the support rings on the longer fairings would be minimised. The experiment was repeated with the fairings fitted to the bottom of the



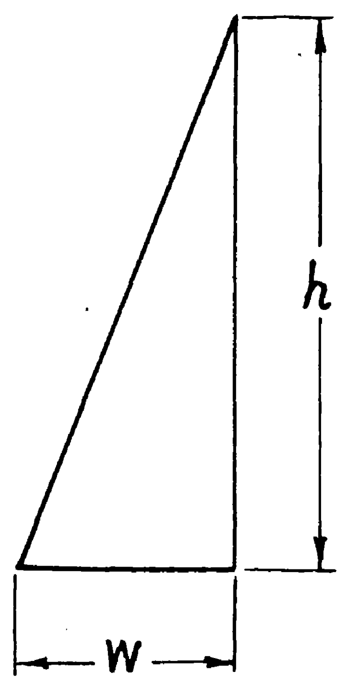


FIG. C.10.

STRAIGHT SIDED FAIRINGS.

cage which was located this time at the upstream limit of the stable region. Commencing with T5 it was observed that with T2 the reduction in the value of k was less than 5% so that there was little point in investigating the effect of T1.

The procedure was repeated using model A1/4, and the checks previously mentioned again carried out. In every instance the cage was supported in the middle of the Perspex section. It was discovered when using nose fairing N3 that the effect was less than for N2. These results were checked and found to be alright so no further nose sections were investigated. Tail fairings shorter than T3 were not examined because the maximum decrease produced by it was only 7%.

In order to get a comparison of the combined action of a nose and tail fairing, N3 and T5 were fixed to each model in turn.

The values of k for the different arrangements and the percentage decrease that they represent are listed in Table No.11.

Discussion of Results: Considering only the values relating to model A1/1 it will be seen from the Table that:-

(a) No advantage was gained by using sides along with fairings N1, N2 and T2.

(b) With N4 and N5 considerable benefit was derived from the sides while in the case of N5 the result of their omission was to practically neutralise the effect of the completed fairing.

(c) For both nose and tail fairings an increase in Aspect Ratio produced an increase in efficiency with the exception of N5. The

TABLE No.11

Fairing	Aspect Ratio	Cage Serial Number	Pressure Drop Coefficient: Cage Alone	Pressure Drop Coefficient: Cage and Fairing: Sides	Pressure Drop Coefficient: Cage and Fairing: No Sides	% Decrease: Cage and Fairing: Sides	% Decrease: Cage and Fairing: No Sides
N1	0.5	A1/1	2.13	1.82	1.78	14.6	16.6
N2	1.0			1.64	1.68	22.9	21.0
N3	1.5			1.58	1.65	25.8	22.4
N4	2.0			1.52	1.73	28.8	19.0
N5	3.0			1.52	1.95	28.8	8.3
T2	1.0			2.04	2.04	4.4	4.4
T3	1.5			1.71	1.75	19.5	17.6
T4	2.0			1.64	1.74	22.9	18.5
T5	3.0			1.59	1.79	25.4	16.1
N3 + T5				1.13	1.48	46.9	30.3
N1	0.5	A1/4	3.29	3.24	3.24	1.6	1.6
N2	1.0			2.93	2.84	10.8	13.6
N3	1.5			2.96	2.86	10.1	13.0
T3	1.5			3.06	3.21	7.0	2.5
T4	2.0			2.91	3.21	11.4	2.5
T5	3.0			2.92	3.18	11.1	3.2
N3 + T5				2.70	2.80	17.7	14.6

two most likely reasons for this were either the effect of the support ring upstream, or the additional friction loss resulting from the increase in the surface area of the fairing, or both.

The results obtained for cage A1/4 appear to have been affected to a greater extent by the support rings. The reason for the percentage decrease in the case of fairings N2 and N3 being greater without sides may be the reduction in frictional rubbing surface. However, since the figures in both columns are lower in the case of N3, it is more than likely that the real reason is that a certain amount of contraction takes place at the brass ring, and the fairings without sides afford greater room for the expansion of the air.

Comparison with the results obtained using aerofoil sections shows that, as far as nose-pieces are concerned, for model A1/1 a greater reduction in k can be obtained using straight-sided fairings. However to do this requires a fairing with an Aspect Ratio of 2 while for the streamlined nose section the figure may be taken as 0.5. For T5 and the Joukowski tail-piece the Aspect Ratios are exactly the same and the reductions in k are comparable. The combined effect is considerably less in the case of the straight-sided fairings, but this is probably due to the brass support rings.

With regard to cage A1/4 the decrease produced by T5 is less than that brought about by the aerofoil tail-section, although the difference in the two figures is not great. The maximum efficiency obtained using straight-sided nose-sections is only about half that of the streamlined nose-piece, but this is in all probability due to the much longer length of the former affecting the air-flow in the region of the support band.

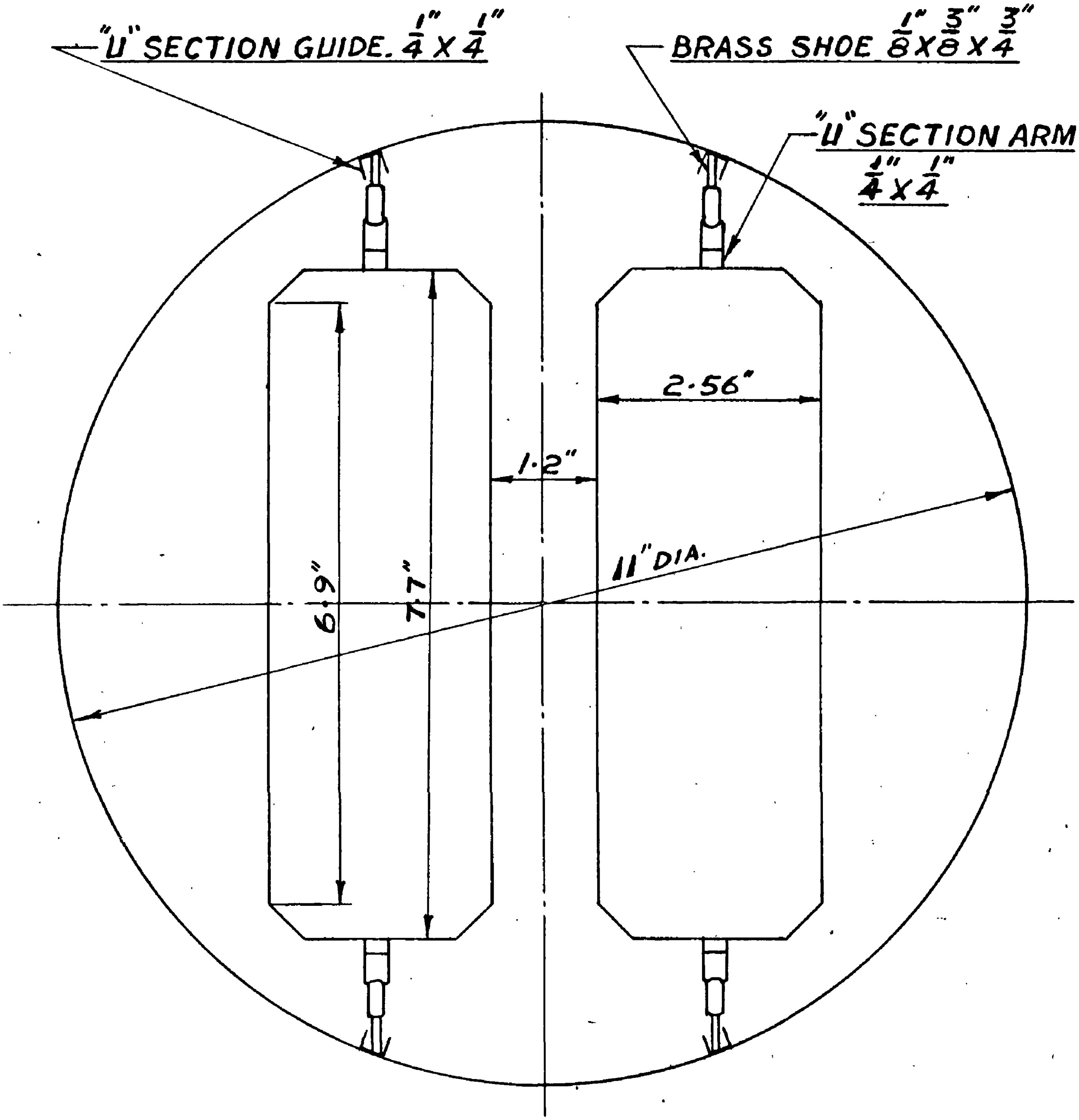
As a result of the influence of the support rings the results obtained from model A1/4 experiments are of relatively little value. However, it would appear reasonable to assume that, just as in the case of the Joukowski Sections, likewise the reduction effected by a straight-sided fairing in the shock loss associated with both a single and four-deck cage will be comparable. From the figures obtained using model A1/1, plus the fact that a straight-sided nose fairing is much more effective than its streamlined counterpart when used as a tail-section, it was decided to adopt fairings of the type used in these experiments for subsequent investigations. An additional factor in favour of this decision is the practical difficulties involved in the construction of aerofoil section fairings.

7. Two Cage Rope-Guide Installation

In this section an attempt was made to simulate a standard layout for a two-cage rope-guide installation, so that the P.D.C.s of cages suitable for use in such an arrangement could be obtained. Also, since it has been shown that the interaction of the different items within a shaft on the air-flow is of fundamental importance (39) it was necessary to examine the efficiencies of straight-sided fairings operating under the conditions met with in practice. It was also proposed to study the effect of the fairings with the cages at the point of passing. In addition, since there will probably be a tendency for two cages, each fitted with a nose and tail-section, to be drawn together when passing, as a result of the "ventüri" formed, the efficiencies of straight-sided fairings designed to counteract this suction effect were studied. Finally the zone within which the models were mutually affected was examined.

The heaviest model to be used was supported horizontally by four wire hawsers, 3/16 inch diameter, at the middle of a 12-feet span, to see if it would be ppssible to reproduce a rope-guide system. Such a span would allow a number of arrangements to be examined without the effect of support devices having to be considered. However, with 50 lb. tension on each wire there was still 0.5 inch deflection at the centre of the span. To maintain a tension of even 50 lb. inside the duct would be extremely difficult, requiring strong anchoring frames and in all probability reinforced flanges. If cheese weights were used the additional load would have to be carried by the trestles supporting the duct. Therefore the idea of supporting the cages on horizontal rope guides was abandoned as impracticable. However it was found possible to support the models on guides similar to those used in sections 1 - 6, which would only slightly interfere with the flow conditions and which would allow the cages to be located anywhere along a 12-feet length (see Fig. C11).

The positions of the guides inside the duct were carefully marked and four 12-feet lengths cut from box-section brass runner. The guides were joined in pairs by distance pieces soldered at four-feet



SCALE = HALF SIZE.

FIG. C.11

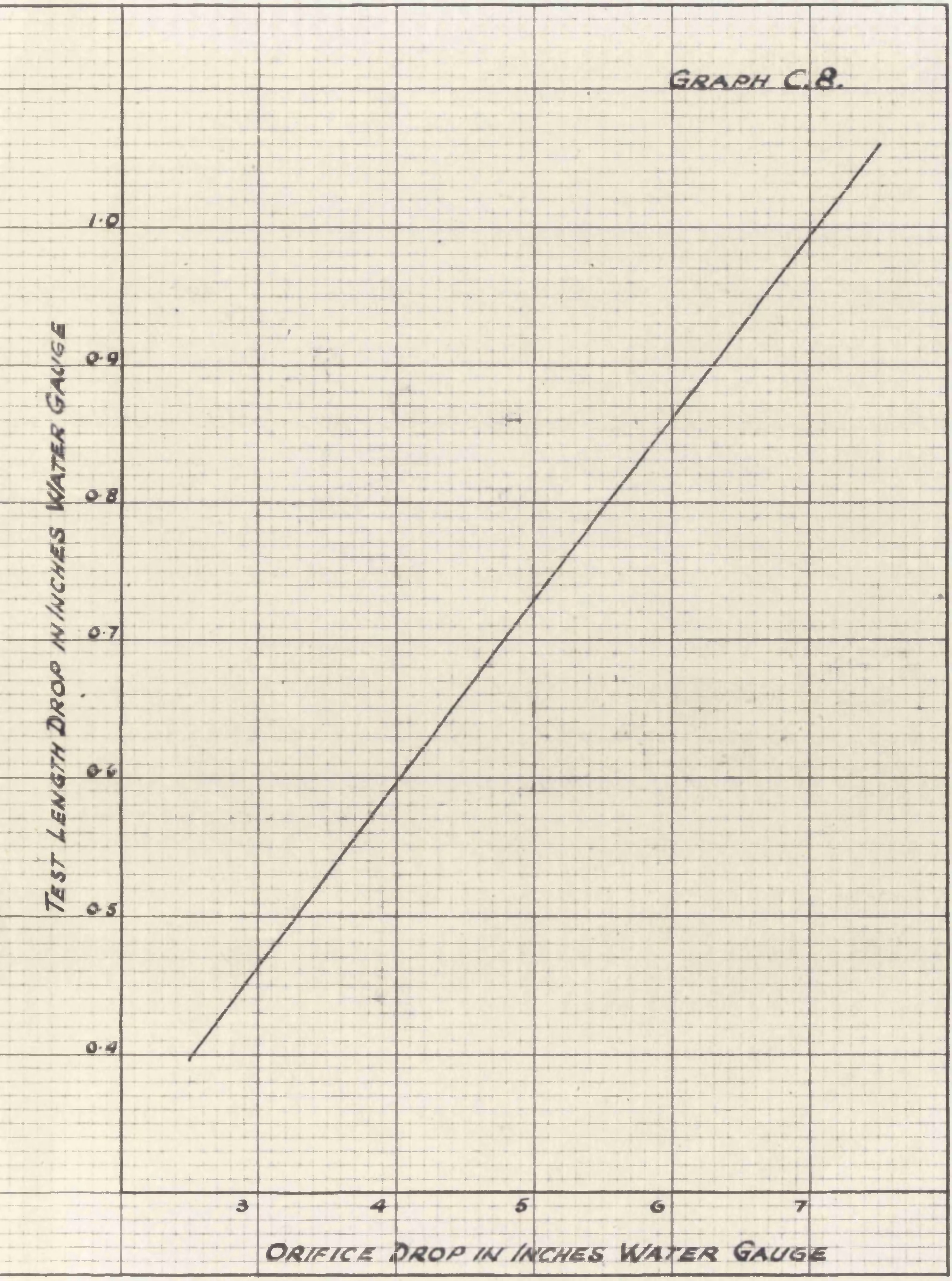
TWO CAGE ROPE GUIDE ARRANGEMENT.

intervals. These were made from $\frac{5}{8}$ inch, 25 gauge steel strip bent to the internal radius of the duct. Lugs were soldered to the runners and cemented between the rubber gaskets and the Perspex flanges, thus holding the guides securely in position. Due to small variations in the internal diameter of the Perspex ducting and slight errors in the location of the runners, supporting the cages of series L1 on rigid arms did not prove satisfactory. Spring-loaded brass guide-shoes were constructed from electric lamp-holder contacts (see Fig. C11). The models could be located at any position along the observation length by means of 3/16 inch diameter stranded wire ropes, inserted through junctions 4/5 and 11/12.

The relationship between the loss in inches water gauge produced by the Test Length plus guides, and the differential pressure across the orifice in the same units was obtained (Graph C8). The orifice drop was observed by the Inclined Gauge while the other pressure losses were measured by the Direct-lift Micromanometer.

The tests which were carried out may be summarised as follows:-

(i) Two single deck cages of series J1 and K1, and two cages of both the single and four-deck type of series L1 were constructed. The ratio of model to full-scale was taken as 1 : 15 which meant that the full-size cage represented by series L1 could accommodate a 3-feet wide by 9-feet long 2-ton mine car. In the duct cross-section the position of the cages was such that the distance between them, and the clearances between the corners and the walls, were never less than the



GRAPH C.8.
1·0
0·9
0·8
0·7
0·6
0·5
0·4
TEST LENGTH DROP IN INCHES WATER GAUGE
3
4
5
6
7
ORIFICE DROP IN INCHES WATER GAUGE

minimum values specified by the National Coal Board.

The disturbances at the inlet to the observation length caused by the guides were expected to be small and hence the final velocity profile could be assumed to occur a short distance from the start of the transparent section. To verify this cage L1/1 was moved along the entire length of the guides by means of the wire ropes and the value of k obtained at different positions. Only slight, random variations were noted, and these were probably due to small changes in the cross-sectional area of the Perspex section which were known to exist. It was decided to take the mid-point of the observation length as the test position. The velocity distribution was confirmed symmetrical by determining the P.D.C. for a single model supported on both sets of guides in turn.

For each set, one and then two models were located at the test position and the values of k obtained. The results are shown in Table No.12.

(ii) Straight-sided fairings of type A (see Fig. C10) were constructed to fit cages of series L1. The Aspect Ratios of the nose and tail-sections were two and three respectively, and permanent smooth cardboard sides were attached. A nose-piece was fitted to the two cages of type L1/1 and the value of k determined for a single model and for two of them side by side. The procedure was repeated with both the nose and tail-fairing attached and then again with only the tail-section in position. These tests were then carried out using the four-deck cages.

Triangular fairings with one side perpendicular to the base (type B) were constructed, the Aspect Ratios being the same as before. When fixed to the models these yielded an outline which could be considered as an asymmetrical aerofoil with its mean camber line facing the nearer side of the shaft. The air flowing over this section would induce a "lift-force" which would oppose any suction effect produced when the two cages were passing. The efficiency of these fairings in reducing the pressure loss caused by cages of series L1 was examined in the manner described in the previous paragraph.

Table No.13 lists the values determined from experiments (i) and (ii) and demonstrates the reductions in the value of k that were obtained for the model alone. The letters N and T indicate respectively where a nose and tail-section was affixed to the models.

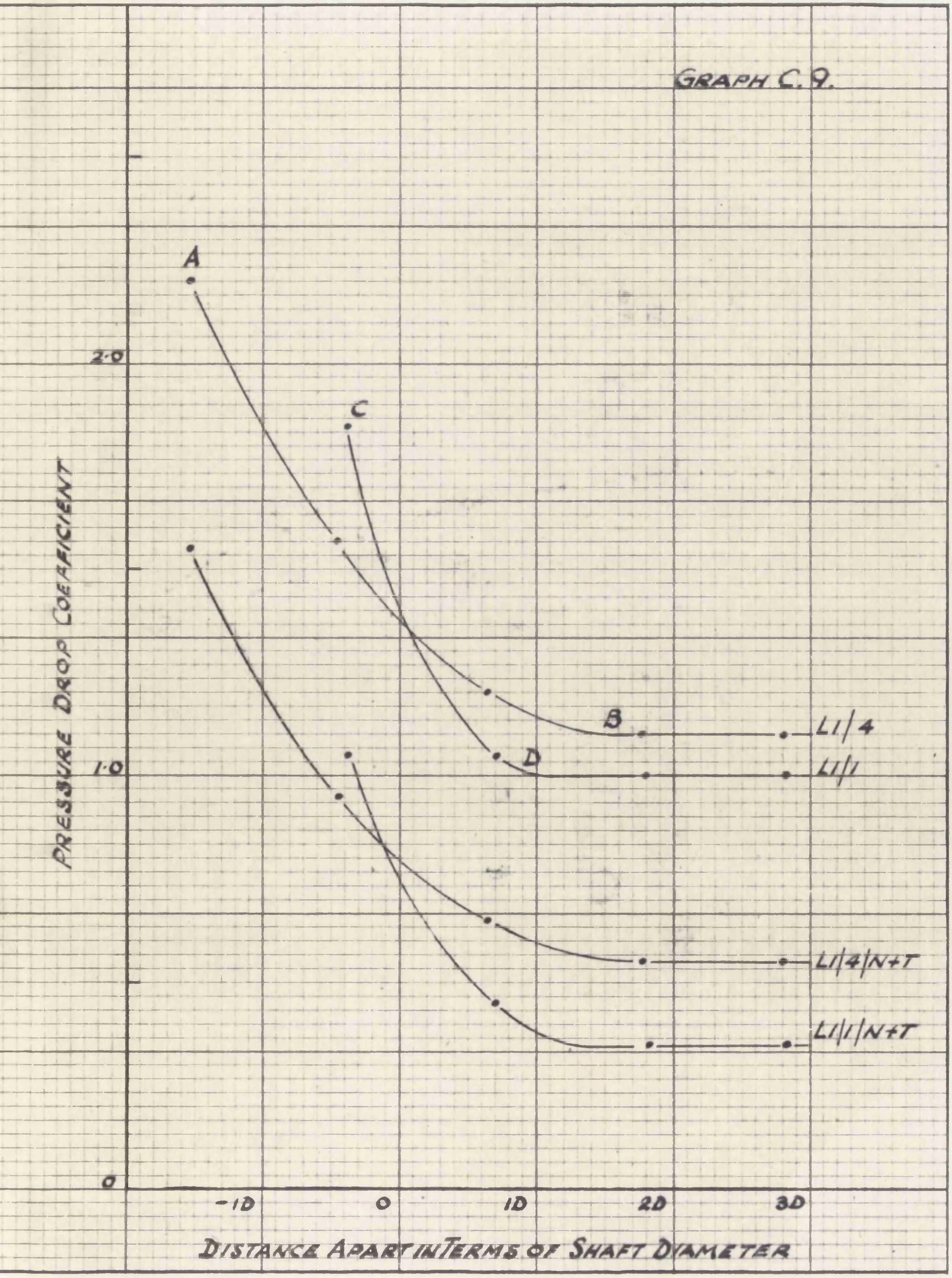
(iii) At a suitable juncture in test (i) the two single-deck cages of series L1 were moved apart and the variation in P.D.C. noted. The same procedure was adopted using the four-deck cage and again in test (ii) with models L1/1 and L1/4 fitted simultaneously with a nose and tail-fairing of type A. Graph C9 illustrates the relationship between the P.D.C. of the two cages in the wind-tunnel, and the distance between them expressed in terms of the shaft diameter D.

TABLE No.12

Cage Serial Number		J1/1	K1/1	L1/1	L1/4
Pressure Drop	Single cage	0.360	0.380	0.495	0.545
Coefficient	Cages side by side	1.42	1.64	1.85	2.18

TABLE No.13

		Single Cage							Cages side by side						
		Pressure Drop Coefficient				% Reduction			Pressure Drop Coefficient				% Reduction		
Cage Number	Type of Fairing	Cage Alone	N	T	N + T	N	T	N + T	Cages Alone	N	T	N + T	N	T	N + T
L1/1	A	0.495	0.250	0.365	0.165	49.5	26.3	66.6	1.85	1.34	1.45	1.05	27.6	21.6	43.2
	B	0.495	0.315	0.465	0.325	36.4	6.1	34.4	1.85	1.28	1.52	1.13	30.8	17.8	38.9
L1/4	A	0.545	0.340	0.475	0.270	37.6	12.8	50.5	2.18	1.89	1.86	1.57	13.3	14.7	28.0
	B	0.545	0.400	0.570	0.425	26.6	-4.6	22.0	2.18	2.05	2.00	1.84	6.0	8.3	15.6


GRAPH C. 9.
A
C
B
D
L1/4
L1/1
L1/4/N+T
L1/1/N+T
2·0
1·0
0
-1D
0
1D
2D
3D
PRESSURE DROP COEFFICIENT
DISTANCE APART IN TERMS OF SHAFT DIAMETER

Discussion of Results: Analysis of the results shown in Table No.12 revealed that for each series of models, the P.D.C. of the cages side by side was approximately four times that of the single model. This included the four-deck cage and indicated that the value of k was nearly directly proportional to the square of the Coefficient of Fill.

From Table No.13 it will be noted that:-

(a) As expected the combined effect on a single cage of a nose and tail-section of type A was much greater than had been indicated in section 6. This was almost certainly due to the elimination of interference from guide supports. The results would seem to justify the incorporation of straight-sided fairings of this type, as even with a four-deck cage the resistance produced by the model itself was halved. If the actual pit arrangements were such that tail-pieces could not be used (e.g. if loading baulks were used at the shaft bottom) then the inclusion of a nose-section would still give an appreciable reduction.

(b) With the cages side by side there is a considerable reduction in the efficiency of the fairings.

(c) Comparison of the results obtained using fairings of types A and B with single models showed that the latter type is less efficient in reducing resistance to air-flow. The efficacy of the tail-fairing by itself was very small, in all probability causing the combined effect of a nose and tail-section, in the case of model L1/1, to be less than when a nose-piece alone was used; indeed when fitted to a four-deck cage an increase in the value of k occurred.

(d) When the two single-deck models were equipped with nose-fairings of type B and suspended side by side in the air-stream a greater reduction was obtained than when type A fairings were used. This was probably due to the fact that the fairings acted like a single nose-section of type A, the two cages being considered as one. From the same reasoning may be explained why the effects of the combined nose and tail-fairings, and the tail-section by itself, were greater than for a single cage.

In the case of the four-deck model the effectiveness of the two fairings combining to form a single section was not so marked. However, comparison of the results of using tail-sections on the cages side by side and singly shows that in the former case the fairings were more effective in reducing the energy loss due to expansion at the down-stream end of the model.

From Graph C9 it may be observed that:-

(a) The distance that the four-deck cages had to be moved before the interaction between them became zero (AB) was approximately twice the distance required in the case of the single-deck cages (CD). The same ratio held when the models fitted with fairings were compared.

(b) The inclusion of the fairings only slightly increased the distance between the cages before the action of one upon the other was cancelled. The increase was greater in the case of the single-deck model which was to be expected since the combined length of the fairings represented a bigger proportion of the overall length of the arrangement.

(c) The length over which this mutual effect existed represented only a small proportion (less than 2D in the case of the four-deck cage) of a shaft of moderate depth.

Conclusions: From the results obtained during these experiments and also in section 3 and 4 it may be deduced that to minimise the resistance produced by the cages in shafts fitted with rope guides:-

(a) It is very much better to use two small cages than one large one.

(b) The ratio of cage length (L) to width (W) should be kept as high as possible.

(c) The cage should be suspended as near the side of the shaft as possible.

A considerable reduction in the proportion of shaft resistance attributable to a cage can be brought about by using fairings of type A. Due to the relatively simple layout of a shaft fitted with rope-guides and the absence of buntons, comparable decreases can be expected in actual practice even although the cages are moving.

The efficiencies of type B fairings were a good deal lower than those of type A. However, some work will have to be done on determining the true significance of the suction effect before it can be decided whether or not to extend the investigations into this aspect.

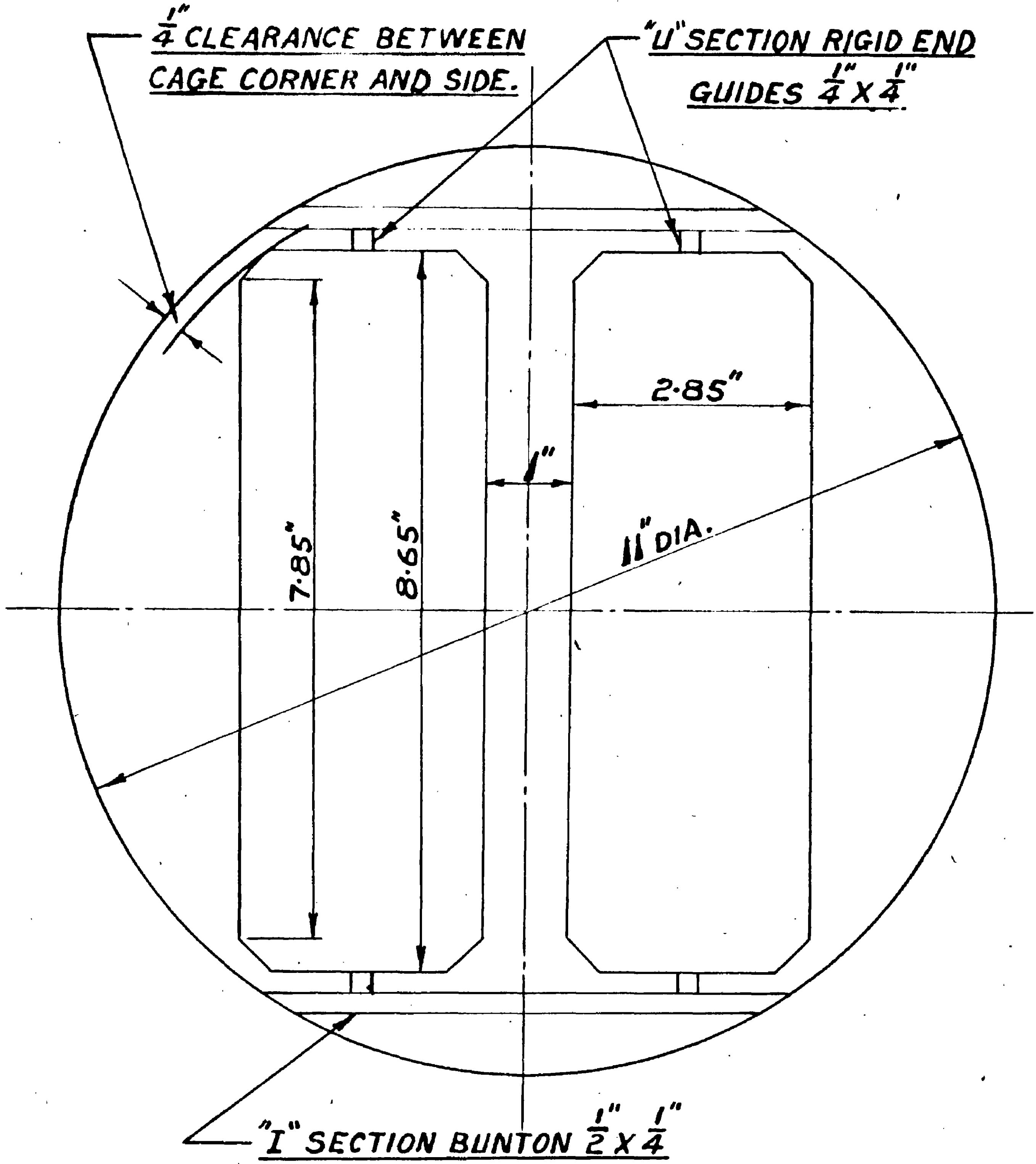
The results obtained with the models together do not indicate what conditions will be produced when two cages pass at speed. However, since the length of shaft over which the cages are mutually affected is extremely short, the increase in shaft pressure loss will be momentary. Hence the resistance in the shaft due to the cages can be considered as twice that of a single cage.

8. Two-Cage Rigid Guide Installation

This experiment was carried out to determine the P.D.C. of cages, and the effect of straight-sided fairings suitable for use in a rigid-guide arrangement. Much better simulation of the actual shaft layout (see Fig. C12), in this case vertical end-guides secured to buntons, was obtained than previously. For comparison with subsequent rigid-guide investigations the P.D.C. of the shaft lining was determined. The ratio of model to full-scale was again 1 : 15.

Only one type of cage was used since different sizes of models would have required different bunton lengths and spacing between the guides, which would have resulted in a large wastage of material. The cages to be used (series M1) were designed to accommodate a 3 feet by 10 feet standard 2-ton mine car. The clearances between cages and between corners and shaft sides were above the minimum values specified by the National Coal Board for rigid-guide installations.

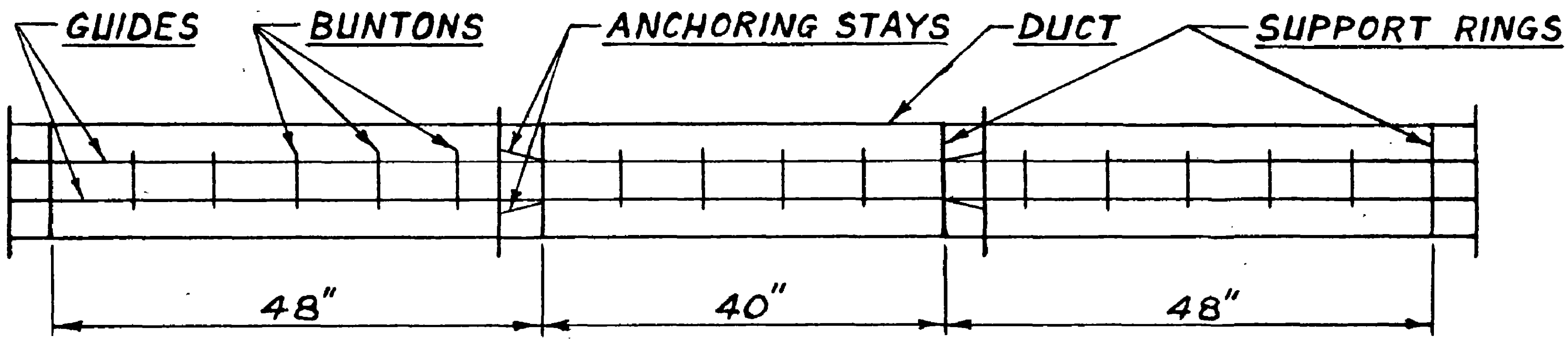
The bunton spacing (see Fig. C13) corresponded to a full-scale distance of 10 feet. The guides were made from 12 feet lengths of $\frac{1}{4}$ inch brass box channel, and were attached to the brass buntons as shown



SCALE ≡ HALF SIZE

FIG. C.12

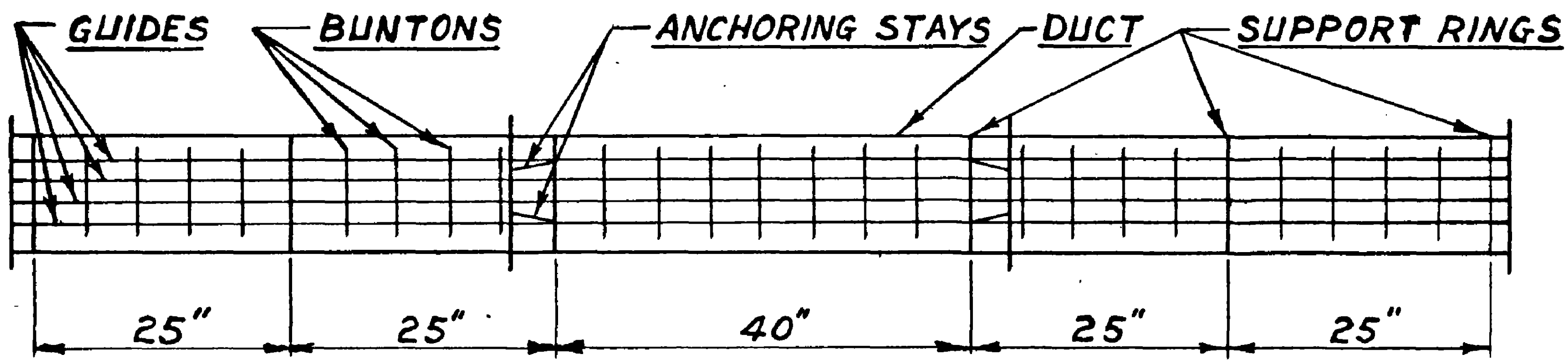
TWO CAGE RIGID GUIDE ARRANGEMENT.



OBSERVATION SECTION - 2- CAGE RIGID GUIDE ARRANGEMENT.

SCALE : 1" ≡ 20"

FIG. C.13



OBSERVATION SECTION - 4 - CAGE RIGID GUIDE ARRANGEMENT.

in Fig. C14. Push-fit side-stays supported the assembly inside the observation length. The whole guide arrangement was anchored by wire stays to the flanges of the ducting. Since the support rings were thin and spaced well apart any interference produced by them could be considered negligible.

Two each of the single and four-deck cages of series M1 were constructed. As a result of the system of guides used it was found unnecessary to fit spring-loaded guide-shoes to the cages. The type adopted were made from aluminium. Nose and tail-fairings of type A, having Aspect Ratios of two and three respectively were made from 20 gauge aluminium sheet. The wire hawsers employed in section 7 were used to move the models within the transparent length. The models were introduced to the test length by removing section 10 of the ducting.

The test length was calibrated (see Graph C10) in the same way as described in section 7.

So that the extra resistance due to the lining of buntons and guides could be ascertained, a similar calibration with no shaft fittings in the test length had been obtained first of all (see Graph C4). By subtraction the P.D.C. of the lining over the range of air-speeds used was calculated to be 0.495.

The usual preliminary precautions were taken to establish a stable length within the Perspex and to ensure that the velocity distribution was symmetrical. It was discovered that, with a single cage being moved along the guides, the values of k were slightly but noticeably

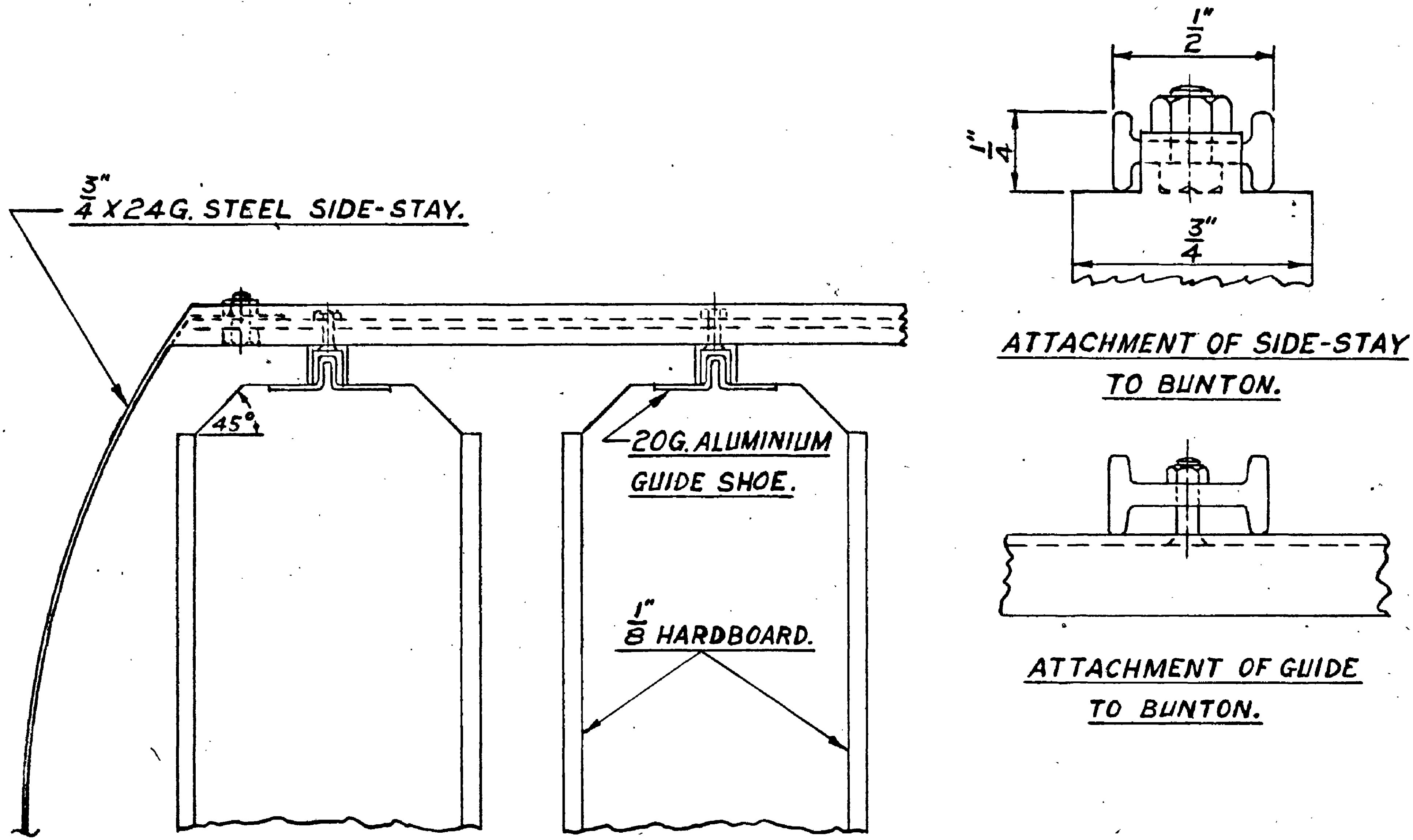
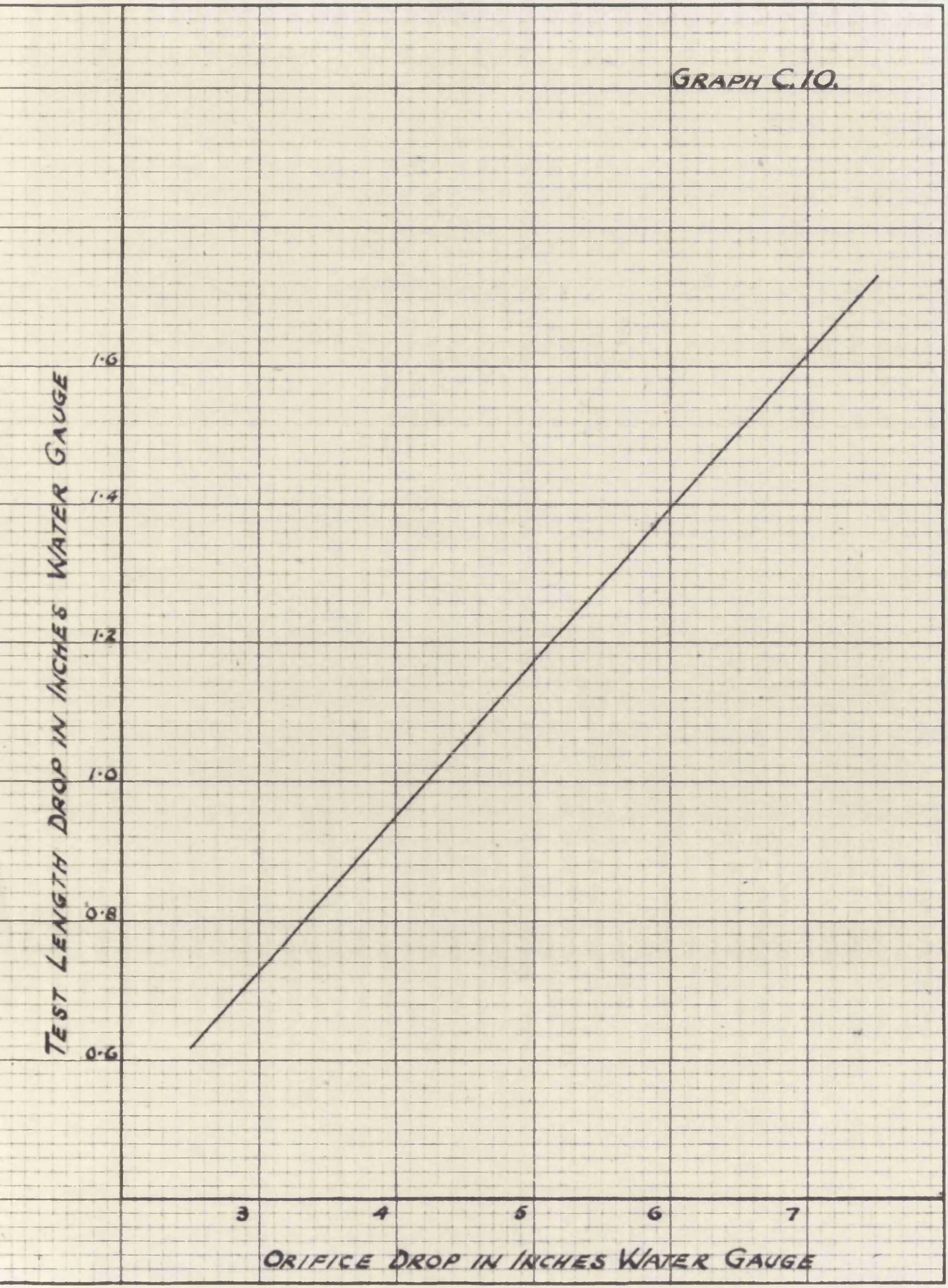


FIG. C.14

DETAILS OF SHAFT LAYOUT (FOUR CAGE ARRANGEMENT)


GRAPH C.10.
TEST LENGTH DROP IN INCHES WATER GAUGE
1·6
1·4
1·2
1·0
0·8
0·6
3
4
5
6
7
ORIFICE DROP IN INCHES WATER GAUGE

affected when the model was brought within one and two feet of the upstream and downstream ends respectively of the observation length. Therefore the mid-point of the observation length was again taken as the test position.

For cages of type M1/1 the P.D.C. was determined first of all for a single model, and then for two of them side by side. The procedure was repeated using four-deck cages. In the manner already described in section 7 the effect of fairings of type A was studied. The results are recorded in Table No.14 where the same notation as previously has been used.

Discussion of Results: From the Table it will be observed that:-

(a) The influence of the buntons is immediately noticeable when the P.D.C.s of the single and four-deck cages, individually and side by side, are compared with the values relating to the rope-guide work. The variance in the results is not accounted for by the difference in the size of the models which is not very great (see Appendix C).

(b) For M1/1 the P.D.C. of the two cages side by side is 4.9 times that of a single cage while for the four-deck model the ratio is 6 · 3. The difference between these two figures and their disagreement with the more or less uniform value of 4 obtained in section 7 is almost certainly due to the presence of the buntons.

(c) The incorporation of a nose and tail-fairing, or even just a nose-piece itself, still produced a significant decrease in resistance.

(d) Reference to Table No.13 shows that, for both the single model

TABLE No.14

	Single Cage							Cages side by side						
	Pressure Drop Coefficient				% Reduction			Pressure Drop Coefficient				% Reduction		
Cage Number	Cage Alone	N	T	N + T	N	T	N + T	Cages Alone	N	T	N + T	N	T	N + T
M1/1	0.810	0.435	0.675	0.365	46.4	16.7	55.0	3.94	2.84	3.97	2.65	27.9	-0.8	32.8
M1/4	0.940	0.595	0.900	0.545	36.7	4.3	42.0	5.89	4.86	5.65	4.68	17.5	4.1	20.6

and the cages side by side, while the effect of the nose-fairings is comparable, the efficiency of the tail-sections is much reduced when a rigid-guide arrangement is used. This suggests that the turbulence and eddying induced by the buntons decreases the effectiveness of the tail-pieces in the recovery of velocity energy during the expansion of the air at the downstream end of the model.

Conclusions: Again it would appear advantageous to utilise a nose and tail-fairing or at least a nose-fairing to reduce the resistance produced by the cages in a shaft. However, in contrast to the rope-guide experiments, the effect of the cages when moving cannot be conjectured due to the influence of the buntons. The determination of the P.D.C.s of travelling cages and the efficiency of fairings fitted to them would have to be made the subject of a special investigation. As a result of the effects contemplated when the cages were moving, it was considered pointless to try to establish a relationship between P.D.C. and the number of decks for the stationary models.

The interaction of the cages was not examined since, allowing for possible effects caused by the buntons and the small increase in the frontal area of the cage, the zone of influence was still expected to represent only a very small portion of the shaft depth.

9. Four-Cage Rigid-Guide Installation

These investigations were conducted primarily to examine the resistance of cages suitable for use in a four-cage rigid-guide installation, and the effect of straight-sided fairings when used in such an arrangement. In addition the influence of the velocity distribution was studied by producing different combinations of the cages in the wind-tunnel cross-section. Finally zones of interaction which might prove of interest in the planning of winding cycles were examined, and the P.D.C. of the shaft lining was obtained. The construction of a system of guides and buntons similar to actual conditions was again relatively simple (see Fig. C15). The ratio of model to full-scale was 1 : 24.

The cages (series N1) were designed to handle 3 feet wide by 10 feet long 2-ton standard mine cars. The clearances between the cages, and the cage corners and walls, were above the minimum specified values.

The guide arrangement was constructed in the manner described in section 8 (see Fig. C13 and C14). The bunton spacing again corresponded to a full-scale distance of 10 feet. Plate VIIIC shows the lining in position.

Four of each of the two and four-deck cages of series N1 were assembled. The double-deck model was adopted because it was thought that the pressure drop produced by a solitary single-deck cage would not be large enough to be accurately measured. The same type of nose and tail-fairings as in the previous section was employed. Plate IXC shows model N1/4 fitted with a nose and tail-section and illustrates the difference in length compared to an unmodified cage. Two additional wire hawsers were introduced so that the four models could be located at any position inside the duct.

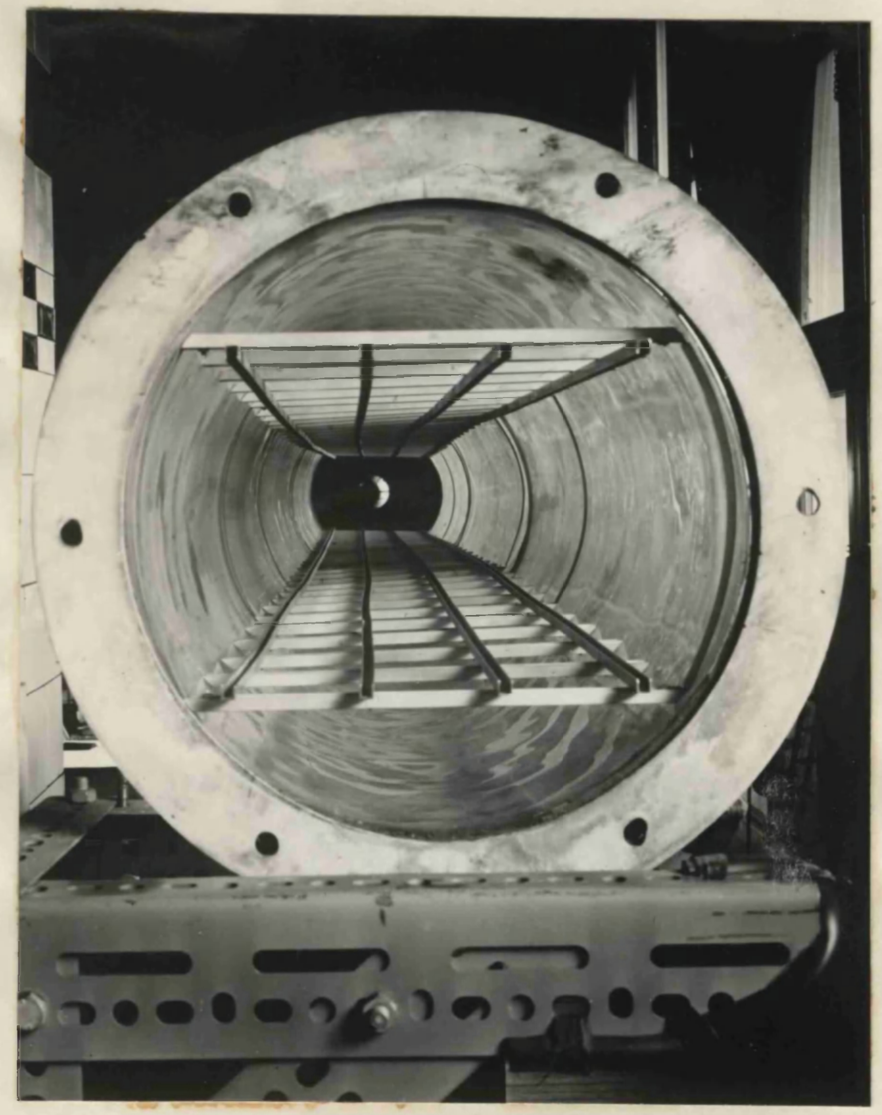


PLATE VIIIC: Four Cage Rigid-Guide Arrangement

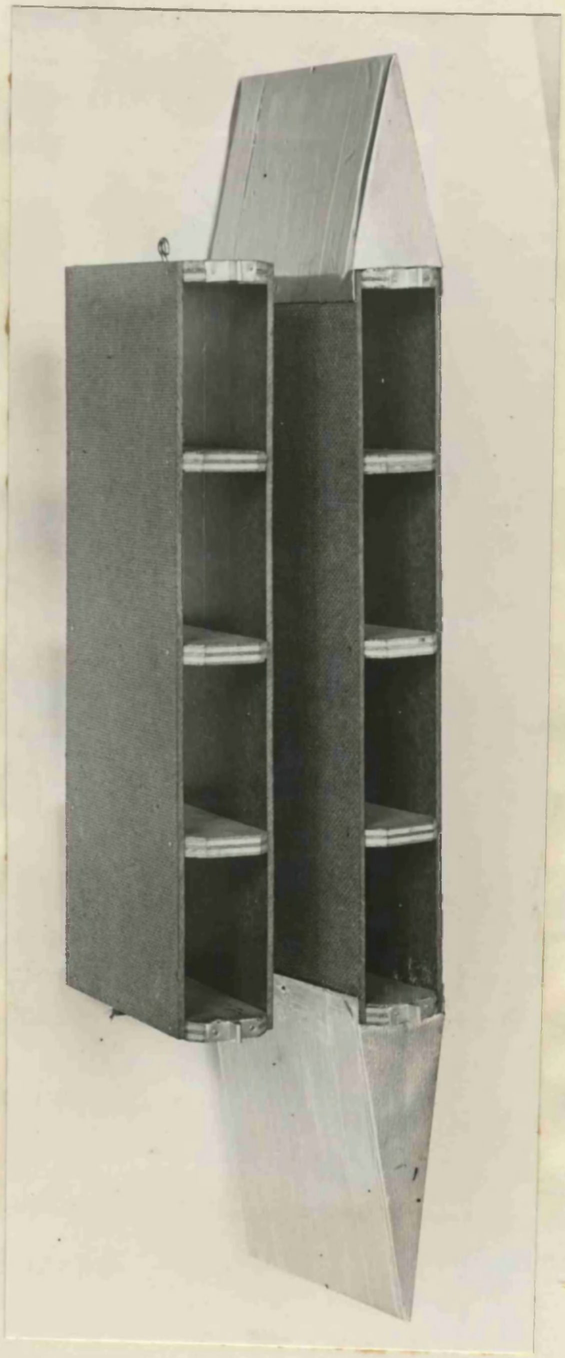


PLATE IXC: Type A Straight-sided Fairings attached to Model N1/4

The test length was calibrated (Graph C11) and the P.D.C. of the shaft lining calculated to be 1.46.

As a preliminary the zone of stability was found to correspond with that derived during the preceding rigid-guide experiments, so the test position was again located at the middle of the Perspex length. The symmetry of the air-flow was confirmed by comparing the resistance of the same cage firstly in the two outside guide positions and secondly in the two inside positions.

The Pressure Drop Coefficients were determined for the two and four-deck models, with and without fairings. Due to the small reductions in resistance effected by tail-sections alone in the preceding experiments and the unlikelihood of their being used by themselves in practice this aspect was omitted during the investigations. The cage arrangements set up at the test position are listed in Table No.15 and No.16 along with the corresponding results. The notation adopted in Table No.13 has again been used, while the symbols referring to the position of the cage in the tunnel cross-section, looking downstream, are:-

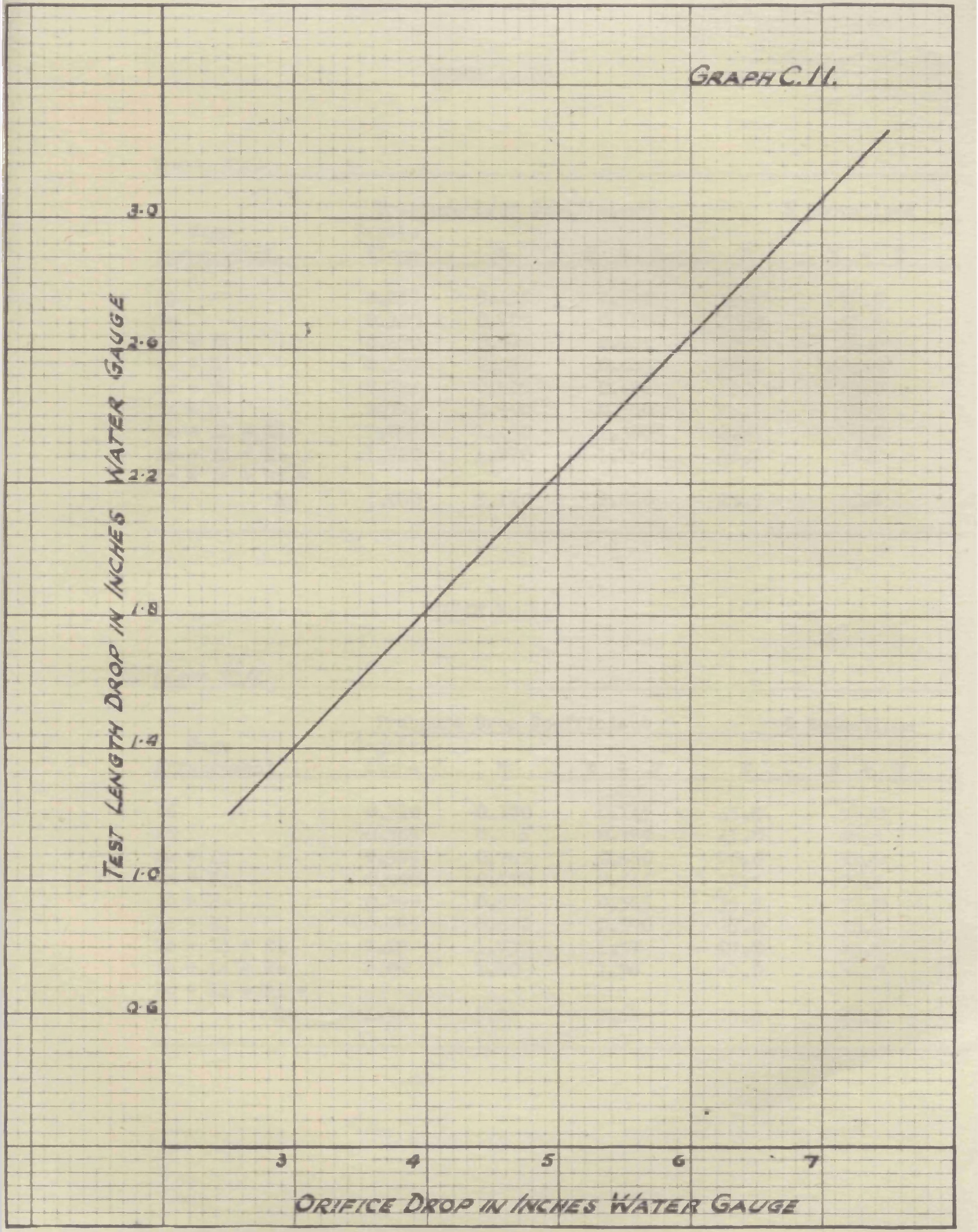
L_o - outside left-hand guide

L_i - inside left-hand guide

R_o - outside right-hand guide

R_i - inside right-hand guide.

In order to produce the longest zones of mutual influence of the cages the four-deck models equipped with both nose and tail-fairings



GRAPH C.11.
TEST LENGTH DROP IN INCHES WATER GAUGE
3·0
2·6
2·2
1·8
1·4
1·0
0·6
3
4
5
6
7
ORIFICE DROP IN INCHES WATER GAUGE

TABLE No.15

Cages N1/2

Cage Arrangement	Pressure Drop Coefficient: Cage(s) Alone	N	N + T	% Reductions: N	N + T
Li	0.255	0.155	0.110	39.0	57.0
Lo	0.330	0.205	0.180	38.0	45.5
Lo + Li	0.870	0.695	0.625	20.0	28.0
Lo + Ri	0.830	0.495	0.465	40.5	44.0
Lo + Ro	0.835	0.505	0.475	39.5	43.0
Li + Ri	0.775	0.550	0.560	29.0	28.0
Lo + Li + Ri	1.865	1.430	1.380	23.5	26.0
Lo + Li + Ro	1.780	1.270	1.150	28.5	35.5
Lo + Li + Ri + Ro	4.060	3.110	2.820	23.5	30.5

TABLE No.16

Cages N1/4

Cage Arrangement	Pressure Drop Coefficient: Cage(s) Alone	N	N + T	% Reductions: N	N + T
Li	0.310	0.170	0.145	45.0	53.0
Lo	0.365	0.215	0.195	41.0	46.5
Lo + Li	0.955	0.765	0.660	20.0	31.0
Lo + Ri	0.900	0.580	0.545	35.5	39.5
Lo + Ro	0.895	0.575	0.565	35.5	37.0
Li + Ri	0.895	0.620	0.590	27.0	30.5
Lo + Li + Ri	2.08	1.73	1.57	17.0	24.5
Lo + Li + Ro	2.02	1.60	1.39	21.0	31.0
Lo + Li + Ri + Ro	4.56	3.86	3.62	15.5	20.5

were used. The cages were moved apart and the variation in P.D.C. noted. The arrangements investigated were:-

(a) Models in guides L_o and L_i,

(b) Models in guides L_o and R_o,

(c) Models in guides L_i and R_i,

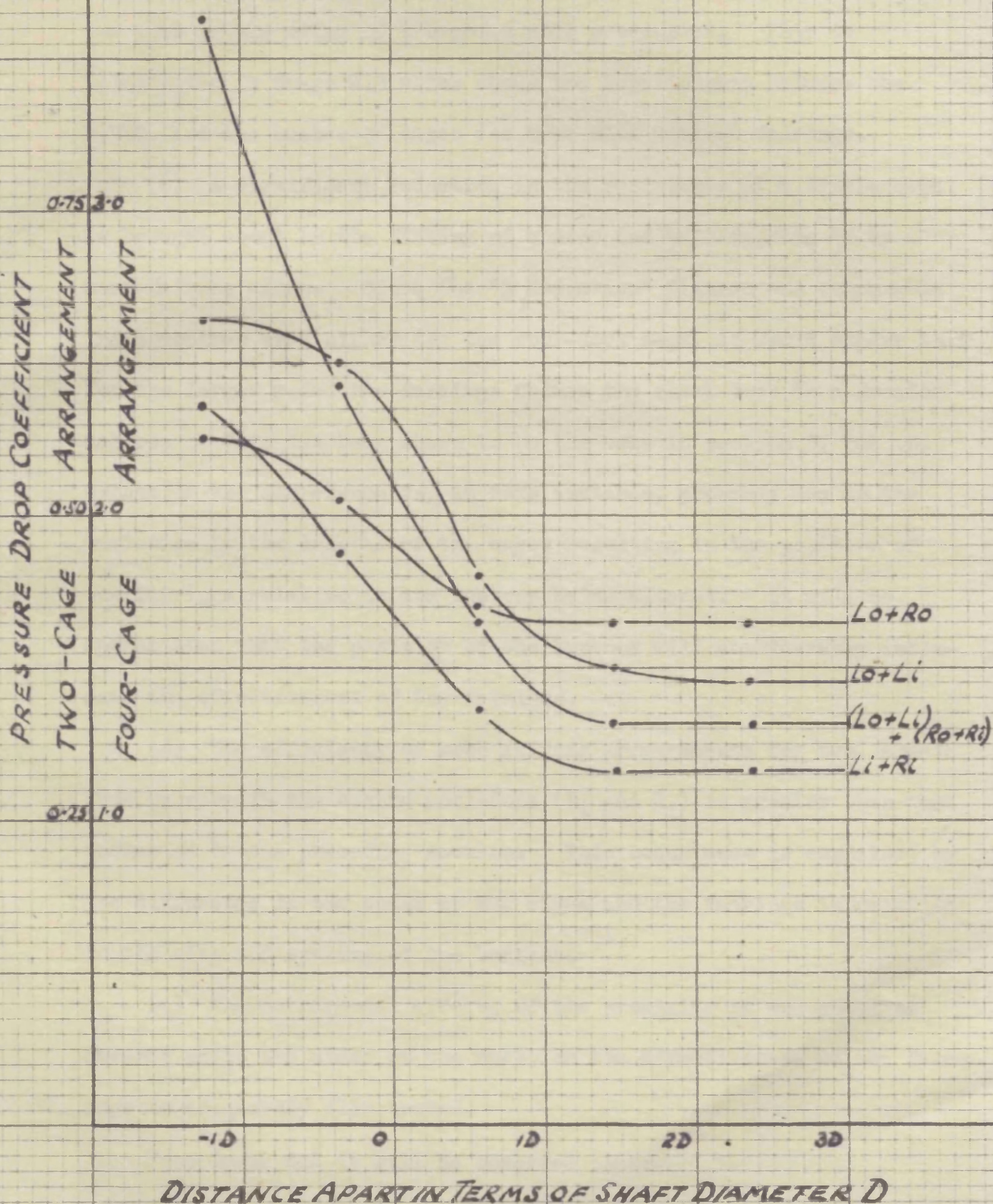
(d) Models in guides L_o and L_i maintained together and moved relative to models in guides R_i and R_o likewise maintained together.

Graph C12 illustrates the relationship between the P.D.C. and the distance between the cages, or sets of cages, expressed in terms of the shaft diameter D.

Discussion of Results: From Table No.15 and No.16 it will be noted that:-

(a) For both cage sizes the P.D.C. of comparable arrangements is greater where the outside guide positions are occupied. This contradicts what would be expected from the normal velocity profile and also the results obtained in section 1 using cages B1/1 and F1/1. The increase in resistance is probably due to the models being much closer to the walls than previously, thereby increasing the loss of energy due to friction by raising the air velocity in the space between the cage and the duct. This increase is apparently not offset to the same extent as before by moving the cage away from the high centre velocity, which is probably due to the effect of the buntons on the velocity profile (41).

GRAPH C.12.



PRESSURE DROP COEFFICIENT
TWO-CAGE ARRANGEMENT
FOUR-CAGE ARRANGEMENT
0·75 3·0
0·50 2·0
0·25 1·0
Lo+Ro
Lo+Li
(Lo+Li) + (Ro+Ri)
Li+Ri
-1D
0
1D
2D
3D
DISTANCE APART IN TERMS OF SHAFT DIAMETER D

(b) As the number of cages at the test position is increased the rise in the value of k becomes more pronounced. This is illustrated in Graph C13 which shows the relationship between the P.D.C. and the number of cages for type N1/4 without fairings.

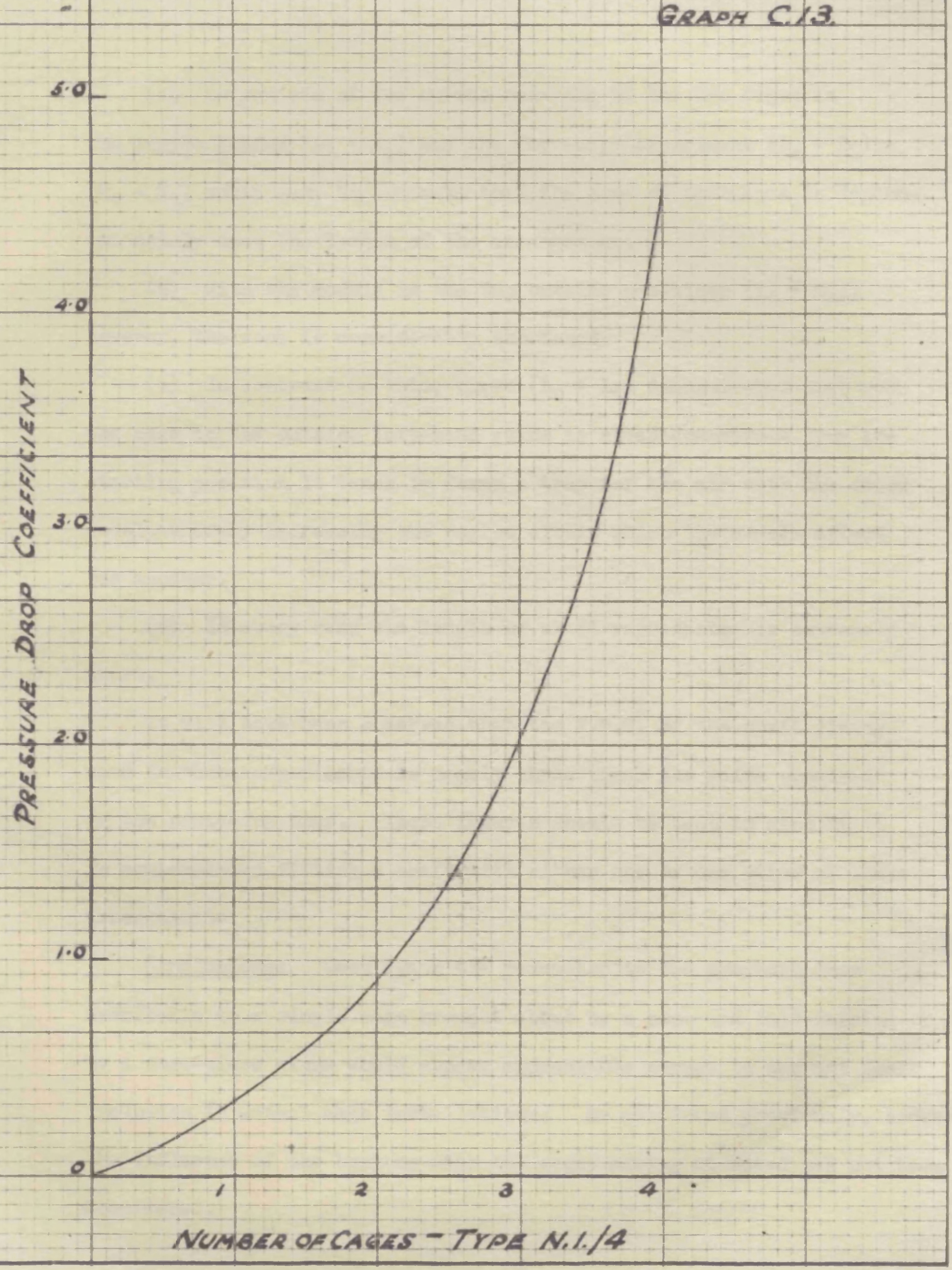
(c) A significant reduction in the resistance of a single cage is brought about by the fitting of a nose and tail-fairing or by a nose-fairing alone. Since the "% Reduction" figures are comparable and sometimes even higher for the four-deck cage, it would appear that for the larger model the fairings reduce the shock loss to a greater degree. This has been shown to be the case in section 5, but the extent of the difference is, in this instance, probably due to the variation in the lengths of the cages resulting in the position of the fairings relative to the buntons being altered. It will be appreciated that the position of the buntons will considerably influence the effectiveness of the fairings.

(d) The efficiency of both fairing arrangements for two models side by side in the centre positions (L_1 and R_1) is greater than that indicated in the preceding section. This occurrence is explained by the difference in the scale of the cages and the fairings in relation to the size and spacing of the buntons.

(e) The detrimental effect, of the proximity of one model to another or to the sides of the duct, on the effectiveness of the fairings is adequately illustrated.

From Graph C12 it will be observed that:-

GRAPH C.13.



5·0
4·0
3·0
2·0
1·0
0
PRESSURE DROP COEFFICIENT
1
2
3
4
NUMBER OF CAGES – TYPE N.1./4

(a) Comparison of the curves relating to the two cages in the inside guides (L_i + R_i) and the four-cage experiment (L_o + L_i) + (R_o + R_i) would seem to indicate that the zone of influence is dependent mainly upon the length of the arrangement.

(b) With the models in the two outside positions (L_o + R_o), however, the zone is considerably shortened.

(c) In contrast to this, curve (L_o + L_i) demonstrates that when the cage in the outside left-hand guide is moved downstream from the starting position it tends to forms a trap for the air with the other model, thereby increasing the region within which the models affect one another.

(d) In every case the length of shaft represented is extremely short.

It will have been observed that the P.D.C. of the shaft lining used in these experiments is nearly three times the value obtained in the preceding ones. These figures should be kept in mind if it is contemplated fitting a shaft with either one or the other of these arrangements.

Conclusions: Once again the reduction in the pressure loss ascribable to a single cage brought about by a nose and tail-fairing or by a nose-piece alone would appear appreciable enough to warrant their inclusion in actual cage installations. As mentioned previously, however, the influence of the buntons with the cages moving at speed has not been considered.

The remarks referring to the use of cages in rope guide arrangements (see page 114) may be applied in the case of rigid guides. That the frontal area should be kept as low as possible is again demonstrated by Graph C13. There appears to be a limit, however, to how close the cages may be brought to the shaft sides before the decrease in resistance resulting from the normal profile of the velocity is offset by the increase in rubbing friction in the narrow space so produced.

The region of the duct over which the models exerted mutual influence was in the worst case equal to twice the wind-tunnel diameter, the average value being nearer one and a half times. Therefore, in a four-cage installation, if the passing points of each pair of cages are arranged to be more than twice the shaft diameter apart, the high resistance produced by three or four cages side by side will never be experienced. If this is possible then the increase in pressure loss due to cages passing will be minimised, and will be only momentary since the cages will traverse the short length of shaft representing the zone of interaction very rapidly. Hence the resistance of the shaft due to the cages can be assumed to be the sum of the individual resistances.

GENERAL CONCLUSIONS

Conclusions from each set of experimental results have already been recorded. What follows is intended to be a general appraisal of the overall investigation.

In the main information has been obtained that will be of considerable help in planning the cage layouts to be adopted in new sinkings, and also in the reorganising of existing shaft arrangements to reduce their resistance to air-flow. With reference to the work relating to rope guide installations it is felt that, even allowing for the experimental support system employed, the full-scale effects will be comparable with those obtained during the tests. With rigid guides, however, the influence of the cages moving relative to the buntons will have to be investigated.

Due to the fact that the design of the models was extremely rudimentary (see Fig. C5) and that they were maintained stationary while the necessary measurements were taken, the Pressure Drop Coefficients obtained from the rope guide experiments cannot be applied exactly to the full-size cages. The difference is expected to be slight, however, as a result of the large proportion of the pressure loss which is controlled by the frontal area of the cage. In the case of the fairings it seems reasonable to assume that the reductions brought about will be duplicated in actual practice.

The Reynolds Numbers attained are comparable with the greatest values reached by previous workers and are very much greater than normal for this type of wind-tunnel arrangement. It can be assumed (55, 56) that resistance values derived at Reynolds Number 300,000 - 450,000 will remain unchanged at the higher values obtained in practice. The same assumption can be made regarding the efficiencies of the fairings even although radical changes in the flow conditions round streamlined objects take place at certain values of Reynolds Number (57). For a streamlined strut with its length three times its maximum thickness the critical value of Reynolds Number referred to the length of the mean chord, is in the region of 400,000 although this figure is reduced by increase in the degree of turbulence. It appears reasonable to assume that this effect, produced by breakaway of the air-stream, will be noticeable if present at all in unstreamlined profiles at lower values of Reynolds Number. For the smallest model fitted with fairings (N1/2) the Reynolds Numbers calculated on a basis of the overall length of the arrangement, which values are the minimum for these investigations, are comparable with the figures quoted above. The consistency of the experimental readings for any one test together with the general agreement of the various fairing experiments indicate the complete absence of any critical change in the flow conditions.

Apart from the rapid increase in pressure loss which takes place when the cages are side by side due to the enlarged combined frontal area, the results give no indication of any additional shock losses induced when the cages in motion pass each other.

SUGGESTIONS FOR FUTURE WORK

Although a great deal of useful information was yielded by the experiments already carried out, it is obviously necessary to extend the work to moving cages. Details of the modifications to the normal procedure that would first of all be required, and then a summary of the investigations that might be undertaken are given below.

A. Pressure Measuring Technique

The method of deriving the P.D.C. measurements of pressure loss in a suitable Test Length and the drop across a plate orifice would still appear to be applicable. However, with the cages moving, some device would be required to record the continuous change in pressure that would be experienced. Preliminary work in this field has already been undertaken in the Mining Department of The Royal Technical College, Glasgow, where a Strain Gauge Diaphragm has been developed for this purpose. This instrument consists of a thin metal diaphragm whose deflection when subjected to a differential air-pressure is measured by electrical resistance strain-gauges. By means of an oscilloscope and camera recording unit it would be possible to relate the continuous change in pressure to the travel of the cages.

B. Rope-Guide Installations

The following investigations could be made:-

(a) The determination of the P.D.C. of suitable cages and the effect of straight-sided fairings. The experiments would have to cover a range of cage travelling speeds.

(b) An examination of the possibility of suction effects being produced when cages fitted with symmetrical straight-sided fairings pass each other. Resolution of the magnitude of the side-thrusts would indicate whether deflection of the heavy cages used in practice was likely.

(c) If necessary the design of fairings to offset the suction effect mentioned above and the effectiveness of these in reducing the resistance caused by the cages. The side-thrust developed by these fairings when the cages were travelling outside the zone of mutual influence would have to be studied.

(d) An analysis of the losses produced as a result of the cages passing at speed. The effect of fairings in reducing these losses and their dependence upon the velocity of the cages might be ascertained.

C. Rigid-Guide Installation

In this case investigations similar to (a) and (d) of Section B could be carried out in order to determine the influence of the buntons.

APPENDIX C

Details of Model Dimensions (see Fig. C5)

Cage Number	Width W (in.)	Length L (in.)	Overall Length S (im.)	Depth D (in.)	Deck Inter-val (in.)	Frontal Area (sq.in.)	Shaft Diam. / Width	Length / Width
A1	4.95	7.50	8.50	8.25	7.25	41.3	2.22	1.51
A2	4.75	7.00	8.00	8.15	7.15	37.3	2.32	1.48
A3	4.25	6.30	7.05	7.15	6.40	29.5	2.59	1.48
A4	3.70	5.50	6.30	6.40	5.65	22.8	2.98	1.49
A5	2.95	4.45	4.95	5.00	4.50	14.4	3.73	1.51
A6	2.25	3.35	3.85	3.90	3.40	8.4	4.90	1.49
B1	6.25	6.25	7.25	8.50	7.50	43.9	1.76	1.00
C1	5.70	6.80	7.90	8.30	7.30	44.1	1.93	1.19
D1	5.35	7.30	8.30	8.10	7.10	43.7	2.06	1.36
E1	5.15	7.65	8.70	8.00	7.00	44.0	2.14	1.49
F1	5.75	5.70	6.75	8.40	7.40	38.0	1.9.	0.99
G1	5.30	6,30	7.25	8.35	7.35	37.6	2.08	1.19
H1	5.00	6.70	7.70	8.20	7.20	37.7	2.20	1.34
I1	4.80	7.00	8.05	8.10	7.10	37.8	2.29	1.46
J1	4.05	3.25	4.05	5.70	5.00	15.9	2.72	0.80
K1	3.45	4.35	5.15	4.85	4.15	17.2	3.19	1.26
L1	2.60	6.90	7.70	4.40	3.70	20.3	4.24	2.66
M1	2.85	7.85	8.65	5.00	4.30	24.0	3.86	2.75
N1	1.90	4.90	5.45	3.70	3.20	10.0	5.80	2.58

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