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The Mixing between Hot and Cold Airstreams in a Centrifugal Field

By

B.S. Stratford, Z.M. Jawor and Miss Maureen Smith

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The mixing between hot and cold airstreams in a centrifugal field

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B. S. Stratford, Z. M. Jawor and Miss Maureen M. Smith

SUMMARY

The mixing rate attainable between hot and cold airstreams influences the length and weight of aircraft gas turbine engines. Previous work has suggested that an improvement might be obtained by causing the mixing to take place in the centrifugal field generated by a curved flow path.

Theory indicates that when the stream of higher total pressure is at the inner radius of the turn the mixing is augmented by the eddies of higher total pressure being centrifuged differentially through the eddies of lower total pressure. When the initial total pressures of the two streams are equal it seems that a similar effect can occur provided the higher density fluid is at the inner radius. During the mixing process the higher density eddies could temporarily increase in total pressure relative to the lower density eddies if momentum were transferred before heat. The resultant increase in mixing rate is proportional to the time elapsing between the sharing of momentum and the sharing of heat.

A limited experimental investigation indicates increases in the local mixing rate of up to 100 per cent, when the total pressures of the two streams are equal.

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1.0 Introduction

One of the factors determining the length and weight of a gas turbine engine is the mixing rate that can be achieved in the dilution zone, where the cold air reduces the temperature of the products of combustion to a value acceptable to the turbine. Burning in the primary zone takes place at stoichiometric mixture strength, so that the temperature of the hot air is three to four times as high as that of the relatively cold dilution air coming direct from the compressor. Suggestions have been made that the resulting density difference could be utilized in a high intensity mixing arrangement: the air streams would be arranged to flow in a curved path such that the high density cold air tended to be centrifuged through the low density hot air. References 1 to 3 indicate that there does appear to be an increase in mixing rate between hot and cold streams in a centrifugal field. A theory is suggested in Reference 4 for the mixing between two cold airstreams in a centrifugal field, experimental results being given for the increase in mixing rate for a cold jet in still air. In the present paper the same type of theory as in Reference 4 is applied to the mixing of the hot and cold streams but a new empirical factor is found to be required. An experiment appears to confirm that there is an increase in mixing rate and provides a tentative value for the empirical constant.

2.0 Theory

Two effects of density may be distinguished. The first is dependent upon there being a difference of total pressure between the two streams as in Reference 4 and may be analysed by modifying the analysis of Reference 4. The second effect of density can be present when the two streams are at equal total pressures; its treatment appears to require an added concept in the basic mixing length theory.

In both analyses the flow is treated as incompressible. Also, the analyses neglect the slight loss of total pressure which results from the mixing and which, together with the wake from the partition upstream, would cause a dip in the centre of the total pressure profile. The results, except for the final substitution of temperature, should apply to gases of different densities at the same temperature, as well as to the hot and cold streams so far discussed.

2.1 The mixing rate increase which is dependent upon there being a difference of total pressures between the two streams

The derivation of Reference 4 for two cold streams is readily generalized to include streams of different densities. The result is given by Equations (9) and (10) of the present paper. The derivation will not be given in full as the main interest in the present paper concerns streams with equal total pressures. The main changes from the previous derivation are as follows.

Equation (4) of Reference 4 becomes

$$Y - Y_e = \tfrac{1}{2}d^2\kappa(1 - \rho u^2/(\rho u^2)_e) \qquad \ldots.(1)$$

Equation (5):-

$$(\rho u^2)_e = \rho u^2 + \delta(\rho u^2) \qquad \ldots.(2)$$

and Equation (6)

$$Y - Y_e = \tfrac{1}{2} d^2 \kappa \delta(\rho u^2)/\rho u^2 \qquad \ldots.(3)$$

leading to

$$\overline{L} = 1 \Big/ \left\{1 - (\tfrac{1}{2} d^2 \kappa / \rho u^2)(-\partial P/\partial y)\right\} \qquad \ldots.(4)$$

The distance d is

$$d = 2u/|\partial u/\partial y| \qquad \ldots.(5)$$

so that

$$\overline{L} = \left\{1 - (-2\kappa \partial P/\partial y)/\rho(\partial u/\partial y)^2\right\}^{-1} \qquad \ldots.(6)$$

Put

$$\left[-\kappa \frac{\partial P}{\partial y} \Big/ \rho\left(\frac{\partial u}{\partial y}\right)^2\right]_{mean} = \frac{1}{r} \frac{P_1 - P_2}{\frac{2}{3}b} \Big/ \frac{\rho_1 + \rho_2}{2} \left(\frac{U_2 - U_1}{\frac{2}{3}b}\right)^2$$

$$= \frac{4}{3} \frac{b}{r} \frac{P_1 - P_2}{(\rho_1 + \rho_2)(U_2 - U_1)^2} \qquad \ldots.(7)$$

and then Equation (6) becomes

$$\overline{L} = \left\{1 - \frac{8}{3} \frac{b}{r} \frac{P_1 - P_2}{(\rho_1 + \rho_2)(U_2 - U_1)^2}\right\}^{-1} \qquad \ldots.(8)$$

or

$$\overline{L} = \left\{1 - \frac{2}{3} \frac{b}{r} \frac{\Delta P}{\frac{1}{2}\rho_m (\Delta U)^2}\right\}^{-1} \qquad \ldots.(9)$$

where

$$\tfrac{1}{2}\rho_m (\Delta U)^2 = \tfrac{1}{2}\left(\frac{\rho_1 + \rho_2}{2}\right)(U_2 - U_1)^2 \qquad \ldots.(10)$$

Thus the increase in mixing rate associated with the difference of total pressures is approximately proportional to $\Delta P/\frac{1}{2}\rho_m(\Delta U)^2$. In some circumstances in an engine it would seem possible for the ratio $\Delta P/\frac{1}{2}\rho_m(\Delta U)^2$ to be as great as unity, so that the effect considered would then be as great as for a cold jet in still air at the same value of b/r.

2.2 The increase for streams of equal total pressures

The initial velocities U_h and U_c at the common surface of two streams of equal total pressures are given by

$$p + \tfrac{1}{2}\rho_h U_h^2 = p + \tfrac{1}{2}\rho_c U_c^2 = P \qquad \ldots.(11)$$

so that

$$\rho_h U_h^2 = \rho_c U_c^2 \qquad \ldots.(12)$$

The radius of curvature, κ, of any flow path has the same form as for the mixing of cold streams[4], i.e.,

$$\kappa = (\partial p/\partial y)/\rho u^2 \qquad \ldots.(13)$$

A difficulty immediately arises. The centrifugal action upon the mixing of the cold air streams was found to depend upon the inner stream having a greater total pressure than the outer, so that when an eddy from the inner stream moved into the outer its value for ρu^2 was greater than that of the surrounding fluid. Hence, from Equation (13), its flow path curvature, κ, was less, and it became differentially centrifuged through the surrounding fluid. In the present flow, however, this effect cannot occur - at least not as just described. The total pressures of the two streams are assumed equal and hence ρu^2 for an eddy of the cold stream that has moved into the hot would be the same as that of the surrounding hot stream.

The hypothesis now put forward is that, although an eddy retains its properties in the conventionally accepted manner until reaching the end of its mean free path and, on being brought to relative rest at the end of its mean free path, it shares its momentum with the surrounding fluid, an eddy could retain its heat content beyond the time at which it is brought to rest and that it only shares this heat content when, in effect, it loses its identity at the beginning of the next movement.

The theoretical model implied in the preceding hypothesis seems not unreasonable. The eddy could be brought to relative rest either by gross form drag or by impact with another eddy, and these would be fairly rapid processes. Heat transfer, however, it would seem, could occur only by conduction and radiation, by smaller scale eddy motion, or by one eddy becoming bound and interwoven by a swirling action into a new eddy movement. The conduction, radiation, and small scale eddy motion would seem rather slow mechanisms for transferring heat, as compared with the rapidity with which an impact could transfer momentum. On the other hand the swirling and interweaving action which would accompany the formation of a new eddy would, on a coarse scale, very rapidly transfer heat. Thus it would seem that an eddy would tend to retain most of its temperature differential after it has been brought to rest, but that parts of it would effectively become common with the surrounding fluid, so that the temperature would become common, as soon as those parts became absorbed into a new eddy.

With the preceding model there is one period of the motion - between the end of one transverse movement and the beginning of the next - when an eddy would have transferred its momentum but not its heat content; it would then have a total pressure different from that of the surrounding fluid. In consequence during this period it would become differentially centrifuged through the surrounding fluid, thus increasing the total transverse movement and the mixing rate.

Suppose that the "waiting time" between the end of one eddy movement and the beginning of another is a proportion ϕ of the time taken for the movement, ϕ being a quantity to be supplied empirically. Suppose also that there is no momentum exchange during this waiting time other than that which occurs immediately at the end of the main movement, and that therefore the eddy retains a constant total pressure until the beginning of the next movement. With these assumptions, and following broadly similar arguments to those used for the mixing of cold streams, the density differential for an outward moving eddy at the end of the main transverse movement is

$$\delta\rho = L(-\partial\rho/\partial y) \qquad \text{....(14)}$$

so that the ratio of the curvature of the eddy flow to that of the surrounding flow, immediately momentum has been transferred and velocities thereby equalized, is, from Equation (13)

$$\kappa_e/\kappa = \rho u^2/(\rho u^2)_e = \rho/\rho_e = \rho/(\rho + \delta\rho) \qquad \text{....(15)}$$

Since the total pressure of the eddy remains constant during the waiting time and the total pressure of the surrounding flow is constant everywhere, the curvature ratio κ_e/κ will, to a first order, be constant throughout the waiting time, and the value calculated from Equations (14) and (15) for the beginning of this time can be used throughout. Now the longitudinal distance travelled during the waiting time is, on average ϕd, where d is the distance travelled during the main movement. Hence, by integration, or, more crudely, as in Figure 2, the additional transverse movement produced by centrifugal action during the waiting time is

$$Y - Y_e = \tfrac{1}{2}\phi^2 d^2 \kappa(1 - \kappa_e/\kappa) \qquad \text{....(16)}$$

Equations (14), (15) and (16) give, to a first order,

$$Y - Y_e = \tfrac{1}{2}\phi^2 d^2 \kappa L(-\partial\rho/\partial y)/\rho \qquad \text{....(17)}$$

On the assumptions made in Reference 4 d is found to be

$$d = 2u/|\partial u/\partial y| \qquad \ldots.(18)$$

Consequently the ratio, $\overline{L}$, of the total transverse movement in curved flow to that in straight flow, is

$$\overline{L} = \left\{L + (Y - Y_e)\right\} \Big/ L = 1 + 2\phi\, \kappa u^2(-\partial\rho/\partial y/\rho(\partial u/\partial y)^2 \qquad \ldots.(19)$$

Equation (19) has been obtained for an outward moving eddy but the result for $\overline{L}$ will be found to hold for an inward moving eddy. As in Reference 4 a mean value across the mixing region is required for the various quantities in Equation (19). Let the mixing region be of total width b and mean radius of curvature r, and suppose as in Reference 4 that the 'tails' of the profile may be neglected. Let T represent temperature so that, from Equation (12), $u^2 \propto \rho^{-1} \propto T$ at any point on the mean profile (the static pressure is here taken as constant and equal to that on the dividing streamline). It would then seem that reasonable mean values are given by

$$\left.\begin{aligned} -\partial\rho/\partial y &= (\rho_c - \rho_h)/\tfrac{2}{3}b \\ \partial u/\partial y &= (U_h - U_c)/\tfrac{2}{3}b \\ T &= \tfrac{1}{2}(T_h + T_c) \\ u^2 &= \tfrac{1}{2}(U_h{}^2 + U_c{}^2) \\ 1/\rho &= \tfrac{1}{2}(1/\rho_h + 1/\rho_c) \\ \kappa &= 1/r \end{aligned}\right\} \ldots.(20)$$

Substitution gives

$$\left.\begin{aligned} \overline{L} &= 1 + f_0\phi^2 b/r \\ \text{where} \quad f_0 &= (t+1)^2\ (t^{\frac{1}{2}} + 1)/3t(t^{\frac{1}{2}} - 1) \\ \text{and} \quad t &= T_h/T_c \end{aligned}\right\} \ldots.(21)$$

the function f_0 being plotted in Figure 3.

Typical values for the temperature ratio between the primary and secondary flows in an engine would lie between $2\frac{1}{2}$ and 4, in which region f_0 is almost constant at a value close to $6\frac{1}{2}$. Now when the mixing is nearly complete the temperature ratio would be much lower, say 1.2 to 1.5. In the latter region f_0 is much larger, so that the centrifugal field might be particularly useful in accelerating the later stages of the mixing, which ordinarily would occur slowly and occupy a correspondingly large amount of space.

Although the transverse movement now occurs in two stages, detailed analysis shows that the mixing rates and, in particular, the rate of heat transfer, are still proportional to the square of the total transverse movement. The increase in mixing rate resulting from the centrifugal field can therefore be readily deduced from Equation (21) once the quantity ϕ has been determined empirically.

There would seem the possibility that the increased mixing rate and turbulence level of the curved flow could persist to some extent downstream, just as suggested for the cold curved flows discussed in References 4 and 5.

2.3 The combined increase in mixing rate

In a flow in which the hot and cold streams are at different total pressures the effect considered in Section 2.2 would be additional to that considered in Section 2.1. Two minor changes occur in the derivation of Section 2.2 itself. The function f_0 of Equation (21) becomes generalized to f, where

$$f = \frac{t^2 - 1}{3t} \cdot \frac{U_h^2 + U_c^2}{(U_h - U_c)^2} \qquad \ldots.(22)$$

as the ratio $(U_h^2 + U_c^2)/(U_h - U_c)^2$, derived from Equations (19) and (20), now depends upon the difference of total pressures as well as upon the temperature ratio. Also, the length L of Equation (14) becomes factored by $\bar{L}$ of Equation (8). The final result is

$$\bar{L} = \left\{1 - \frac{8}{3}\frac{b}{r}\frac{P_c - P_h}{(\rho_c + \rho_h)(U_h - U_c)^2}\right\}^{-1} \left\{1 + f\phi^2 b/r\right\} \qquad \ldots.(23)$$

where the mixing rate in the curved flow is proportional to $\bar{L}^2$.

The value of f increases when the total pressure of the cold stream is increased above that of the hot - as $(U_h - U_c)$ then decreases. Hence an increase in the difference of pressures $(P_c - P_h)$ causes a two-fold increase in the increase of mixing rate.

3.0 An experiment

3.1 Apparatus and preliminary results

Figure 4 shows the general layout of the apparatus for supplying a single hot stream and a single cold stream to a test mixing section. Uniform total pressures were obtained by gauzes and honeycombs and, as indicated in the figure, by two two-dimensional contractions of 5/1 area ratio. In all the tests the total pressures of the two streams were equal and the respective temperatures were 650°C and 30°C. Care was taken to obtain a consistent and reasonably uniform temperature in the hot stream. The construction largely consisted of "Nimonic 75" material.

The cross-section of the flow in the test section featured an aspect ratio of 8 - based on a span of 24 in. and a total stream depth of 3 in. - in order to prevent secondary flow from the end walls affecting the flow at mid-span[4]. The test sections are shown in Figure 5. Temperature traverses were obtained using the sonic suction thermocouple[6] and traversing gear of Figures 6 to 9; the measurements from the traversing thermocouple were recorded in millivolts on a Negretti Zambra "quick reading potentiometer", while the fixed thermocouples recorded direct on Cambridge indicators.

The design values for the depths of the hot and cold streams were 2 in. and 1 in. respectively. However the dividing wall between the contractions was left unrestrained and, because of the temperature difference across it, bowed considerably. The resulting streams at mid-span measured 1.6 in. deep for the hot and 1.4 in. for the cold. The streams also appeared to have a slight inclination, the temperature traverses indicating that the flow at mid-span was not entering the working section parallel to the walls. By superposition of the profiles as in Figure 10a the lateral misalignment could be approximately determined for each profile. The plot of the misalignment in Figure 11 indicates that in most of the test section the mean mixing line was inclined at about 1.4°. The mode of bowing of the dividing wall presumably caused the inclination; in a future test it would be advisable to stiffen the wall, perhaps by stays at the quarter span positions. The analysis of Figure 10a would slightly exaggerate the misalignment as the mass flow per unit depth is greater for the cold stream than for the hot.

Figure 12 shows the distributions of static pressure. The rise in static pressure on the inner radius wall of the curved flow at, say, the 60° traverse station is 0.8 in. of water, on an inlet dynamic head for that wall of 7.7 in. of water. (The 60° traverse station is at x = 7.76 in. along the mean mixing line and at just over 6 in. along the inner radius wall.) It has been assumed in the analysis that such a small amount of diffusion would not affect the mixing rate in the mainstream. In any subsequent experiment, however, it would be desirable to obtain the same streamwise pressure distribution in the curved flow as in the straight.

3.2 Main results and analysis

The main results from the test are represented by the typical traverses for total temperature in Figure 13 and the plots of the width of the mixing regions in Figure 14.

For the analysis it is necessary to define a rate of mixing and to compare corresponding values for the two flows. The most obvious definition for a mixing rate is db/dx, where b is the total width of the temperature profile at any streamwise station. The 'tails' to the profiles, however, prevent the exact width, b, from being readily decided for any given traverse, so that the differential db/dx would be subject to considerable error. The main analysis was therefore based on an area, or heat transfer, method for estimating the degree of mixing at any station. A vertical line was drawn on each traverse profile so that the shaded areas as shown in Figure 10b were equal. A linear dimension was then obtained for the average width of each area by dividing by the vertical height of that area. The two dimensions thus obtained for each profile were summed and termed β. It will be found that β is closely related to the total heat that has been transferred from the hot to the cold stream, and a further property demonstrating the usefulness of β may be shown as follows. A common method of avoiding the problem of determining accurately the full width b of a profile is to use a width say $b_{0.95}$, where the suffix 0.95 indicates that the width is measured to the points where the temperature difference from a mean has become 0.95 of its maximum value. In a similar manner a generalized width b_θ may be defined. The value of β will be seen to be

$$\beta = \int_0^1 (b_{1,\theta} + b_{2,\theta})\, d\theta = \int_0^1 b_\theta \, d\theta$$

where b_θ is a generalised form of $b_{0.95}$ and where $b_{1,\theta}$ and $b_{2,\theta}$ are the contributions to b_θ from the two parts of the profile. Thus β is the integral of all possible b_θ, it utilises information from all parts of each experimental curve instead of from just one pair of points, and should be accurately determinable for any given experimental profile.

Values of β are plotted for the two flows in Figures 15 and 16, the distance x in each flow being measured along the mean mixing line. The mixing rate $d\beta/dx$ appears to be up to about 100 per cent or more greater for the curved flow than for the straight flow, for values of b/r in the curved flow of up to 0.25. Moreover the increase in mixing rate appears to increase with increase in x, and therefore with increase in b/r, as suggested by the theory - although local differentiations of the experimental curves should be treated with caution. (Strictly the comparison of $d\beta/dx$ between the two flows should be made at equal values of b, if it is assumed that the fall-off in $d\beta/dx$ with increase of x in the straight flow is due to a damping effect from the walls.) The corresponding values for the parameter ϕ, deduced by substitution of the experimental results into Equation (21), are shown in Figure 17, and appear to be in the region of 50 or 60 per cent. Such a value for ϕ seems intuitively reasonable. The results basing the analysis on db/dx are rather similar.

The present experimental investigation is limited in extent and further work would be desirable. The experimental techniques might, perhaps, be simplified by using unheated gases of different densities.

4.0 Conclusions

Theory indicates two ways in which a centrifugal field can increase the rate of mixing between two streams when their temperatures are different. When the total pressures of the two streams differ the eddies from the stream of higher total pressure may be centrifuged through the stream of lower total pressure, the temperature difference merely influencing the quantitative results. When the total pressures are equal, it is suggested that the effect occurs in two stages. In the first stage eddies from the two streams mix and exchange momentum, but retain their temperature and identity. Consequently for the second stage, in the waiting period prior to the formation of the next eddy, the total pressures have become unequal and the differential centrifugal action can occur. The time before the loss of identity in the next eddy formation thus becomes a new unknown parameter, to be determined empirically.

In a limited experimental investigation it appeared that the local mixing rate could be roughly doubled by a centrifugal field when the two streams had the same total pressure. The value derived for the waiting time was about equal to 50 or 60 per cent of the time spent in the main eddy motion. Further experiments would be desirable, perhaps using unheated gases of different densities.

ACKNOWLEDGEMENT

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List of symbols

b	width of the mixing region (see Figure 1)
d	distance travelled by the eddy along the jet, during its transverse movement
f	defined by Equation (22)
f_o	value of f when the total pressures of the two streams are equal; f_o is defined by Equation (21)
L	transverse movement of an eddy in straight flow
$\bar{L}$	ratio of the total transverse movement of an eddy in curved flow (relative to the rest of the fluid) to that in straight flow
p	static pressure at the central plane, or mean flow path, of the mixing region
P	total pressure
r	radius of curvature of the central plane, or mean flow path, in the mixing region (see Figure 1)
t	T_h/T_c
T	temperature
u	local velocity
U	fluid velocity either before mixing, or at the edge of the mixing region. In the latter instance U is an equivalent velocity given by $P = p + \frac{1}{2}\rho U^2$, where only P and U are necessarily taken at the same transverse position.
x	distance along model mean mixing line
y	distance measured outwards across the flow
Y	displacement inwards from reference tangent line as a result of the curvature
ρ_m, ΔP, ΔU	defined by Equations (8), (9) and (10)
β	$\int_0^1 b_\theta d\theta$
$\delta\rho$	density difference between the eddy and the surrounding fluid at the end of the transverse movement
$\delta(\rho u^2)$	value of $(\rho u^2)_e - \rho u^2$ at the middle of the transverse movement
θ	$(T - T_m)/(T_h - T_m)$ when $T > T_m$, and $(T_m - T)/(T_m - T_c)$ when $T < T_m$, where T_m is defined by Figure 10b

κ	curvature of the flow path
ρ	local fluid density
ϕ	ratio of the "waiting time" between movements to the time of a movement

<u>Suffices</u>

e	refers to an eddy
c, or 1	refers to the trough (or peak) values in the cold stream, i.e., the stream at the inner radius
h, or 2	refers to the peak (or trough) values in the hot stream, i.e., the stream at the outer radius
θ	refers to the pair of positions on any profile where the non-dimensional temperature difference from the mean is θ

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FIG. I

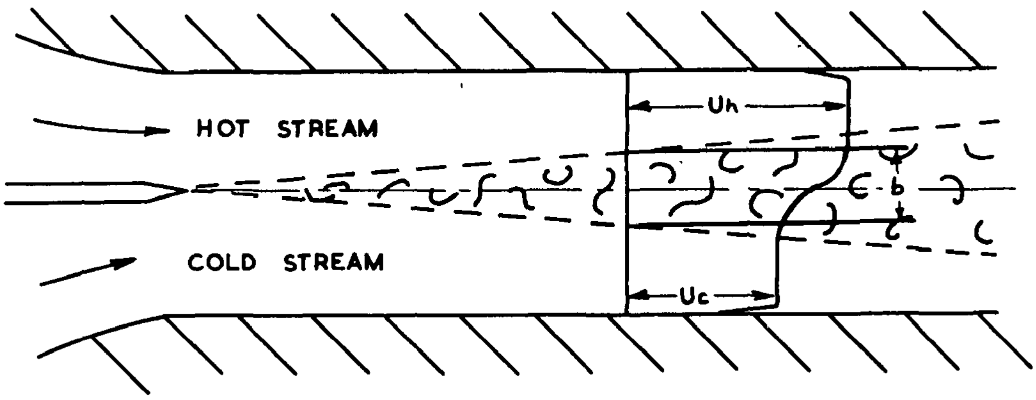


FIG. 1a. STRAIGHT FLOW – NO CENTRIFUGAL FIELD.

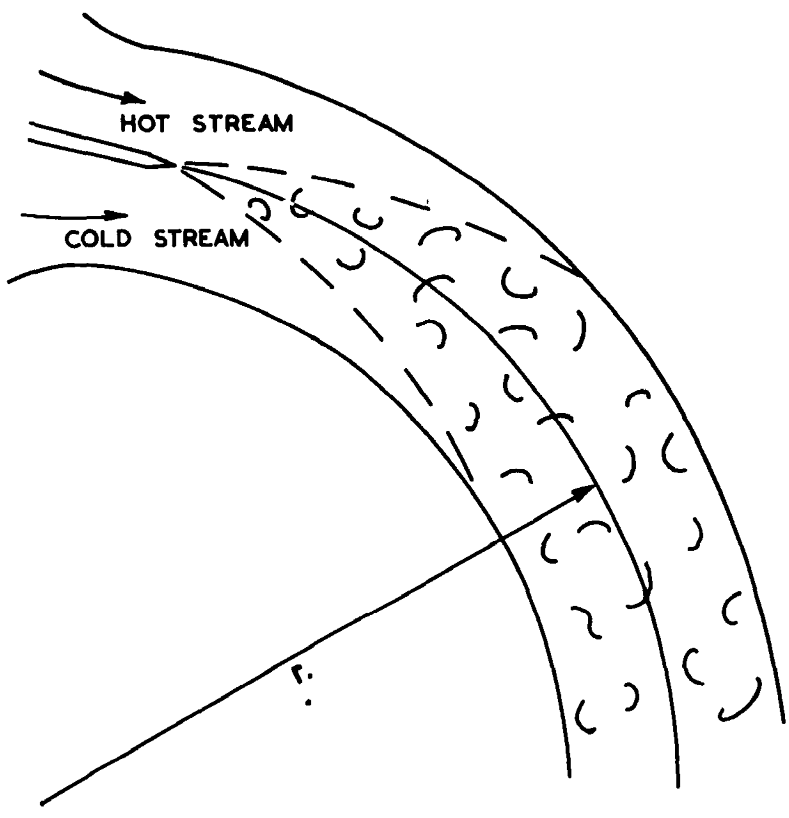


FIG. 1b. CURVED FLOW WITH CENTRIFUGAL FIELD.

THE MIXING OF SINGLE HOT AND COLD STREAMS.

FIG. 2

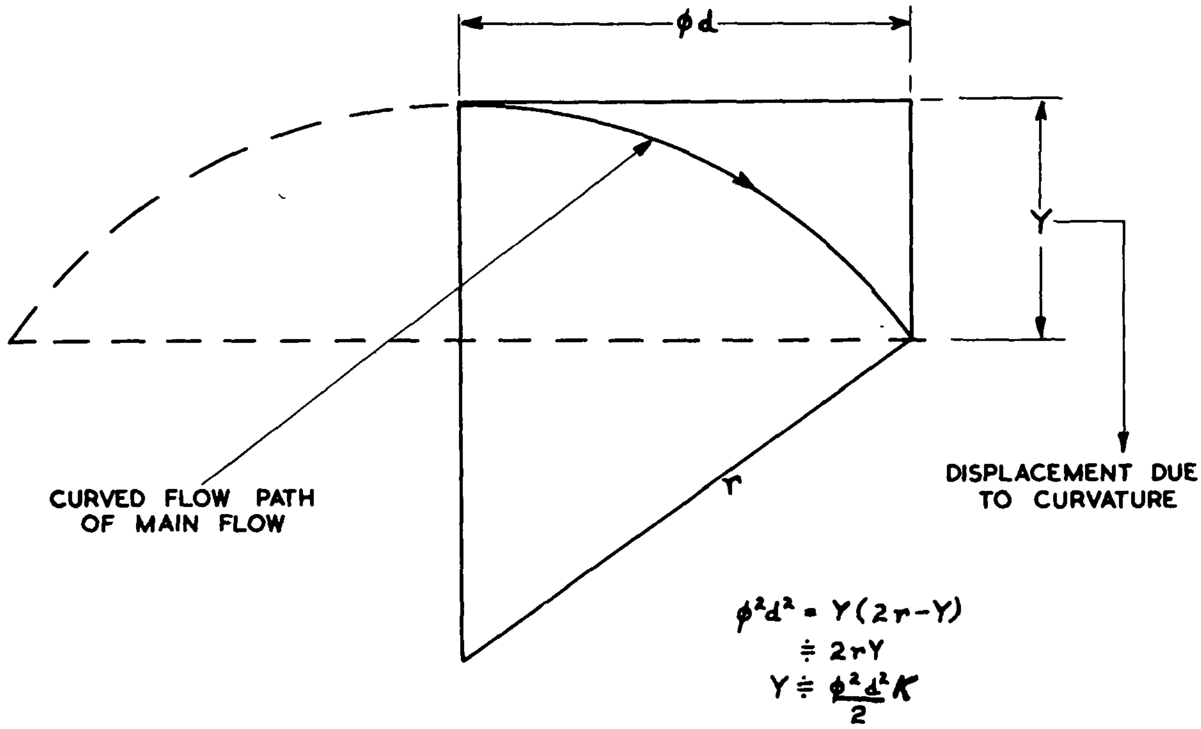


FIG. 2a. DISPLACEMENT FROM A REFERENCE TANGENT LINE DUE TO CURVATURE.

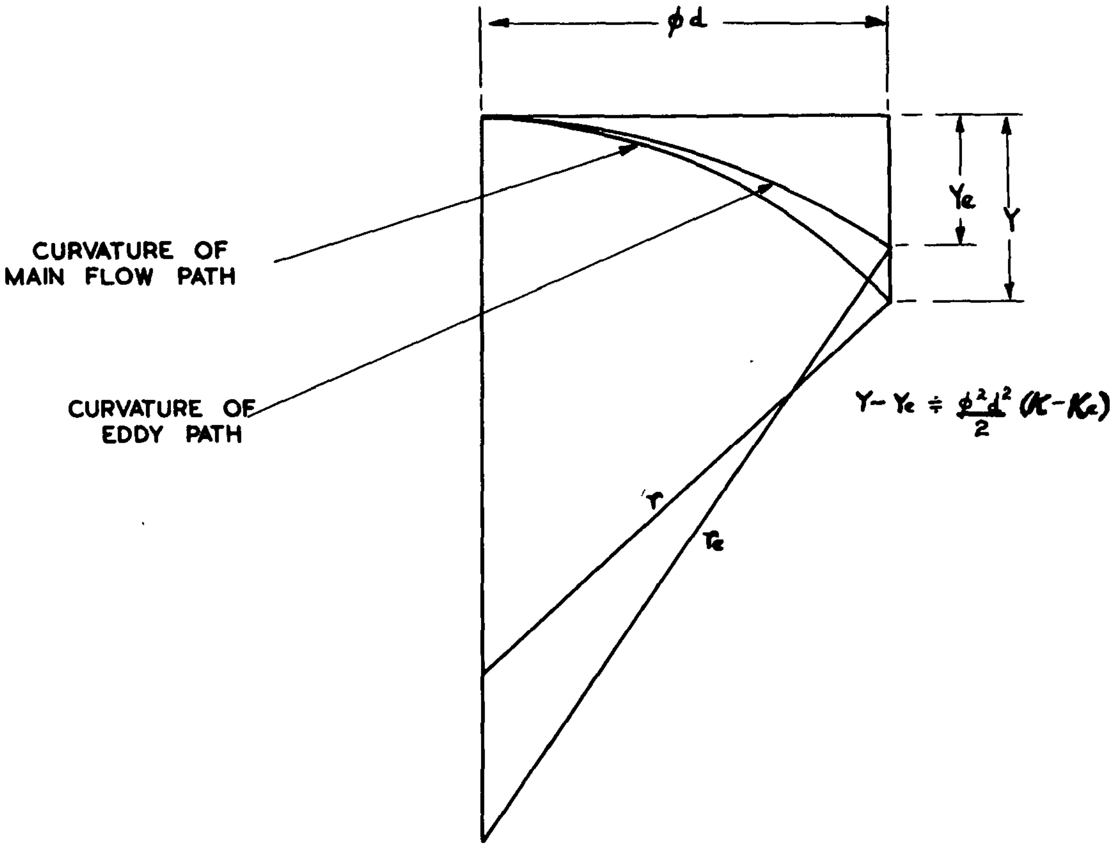
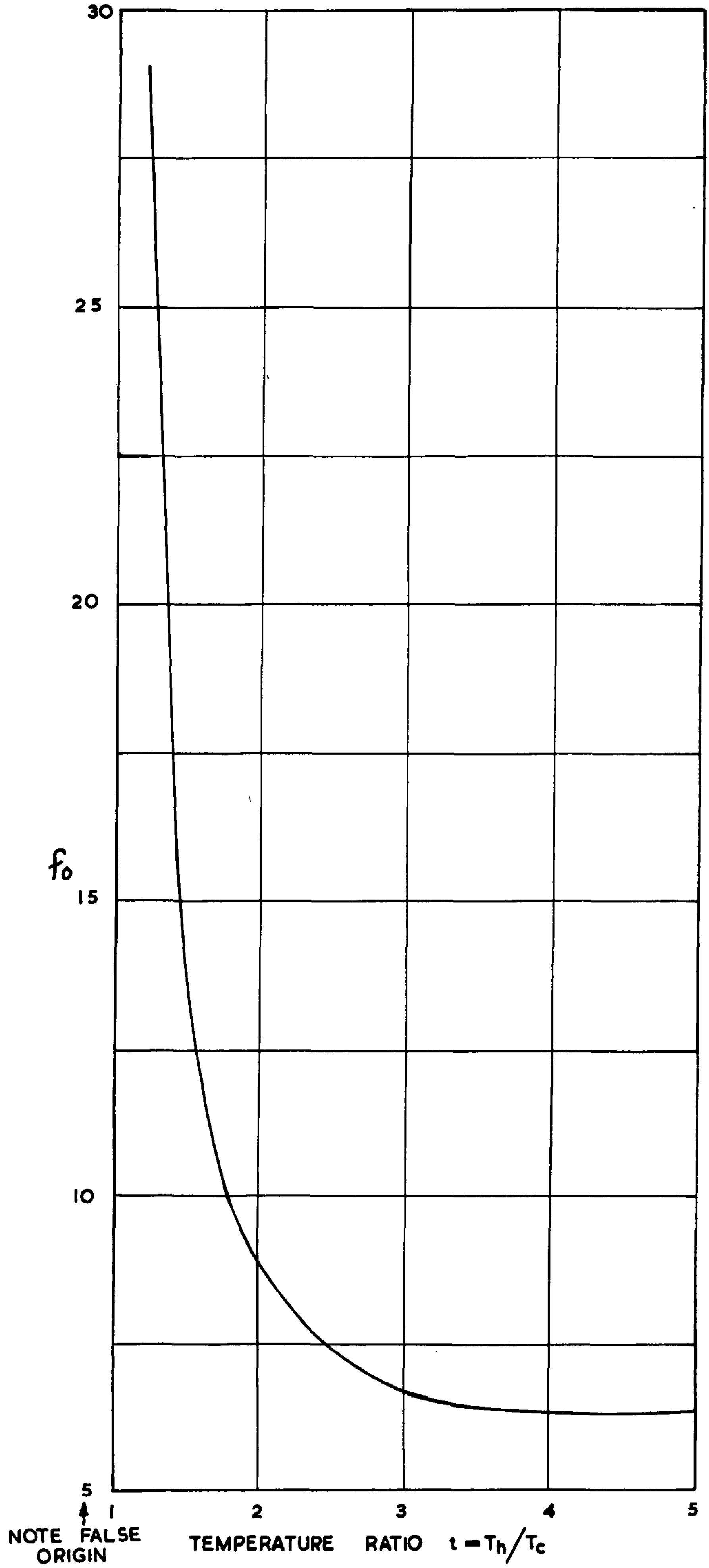


FIG. 2b. DIFFERENTIAL DISPLACEMENT RESULTING FROM DIFFERENTIAL CURVATURE OF PATHS.

THE DIFFERENTIAL CENTRIFUGAL MOVEMENT.

FIG. 3



VALUES OF THE FUNCTION f_0.

DIAGRAM OF THE TEST RIG.

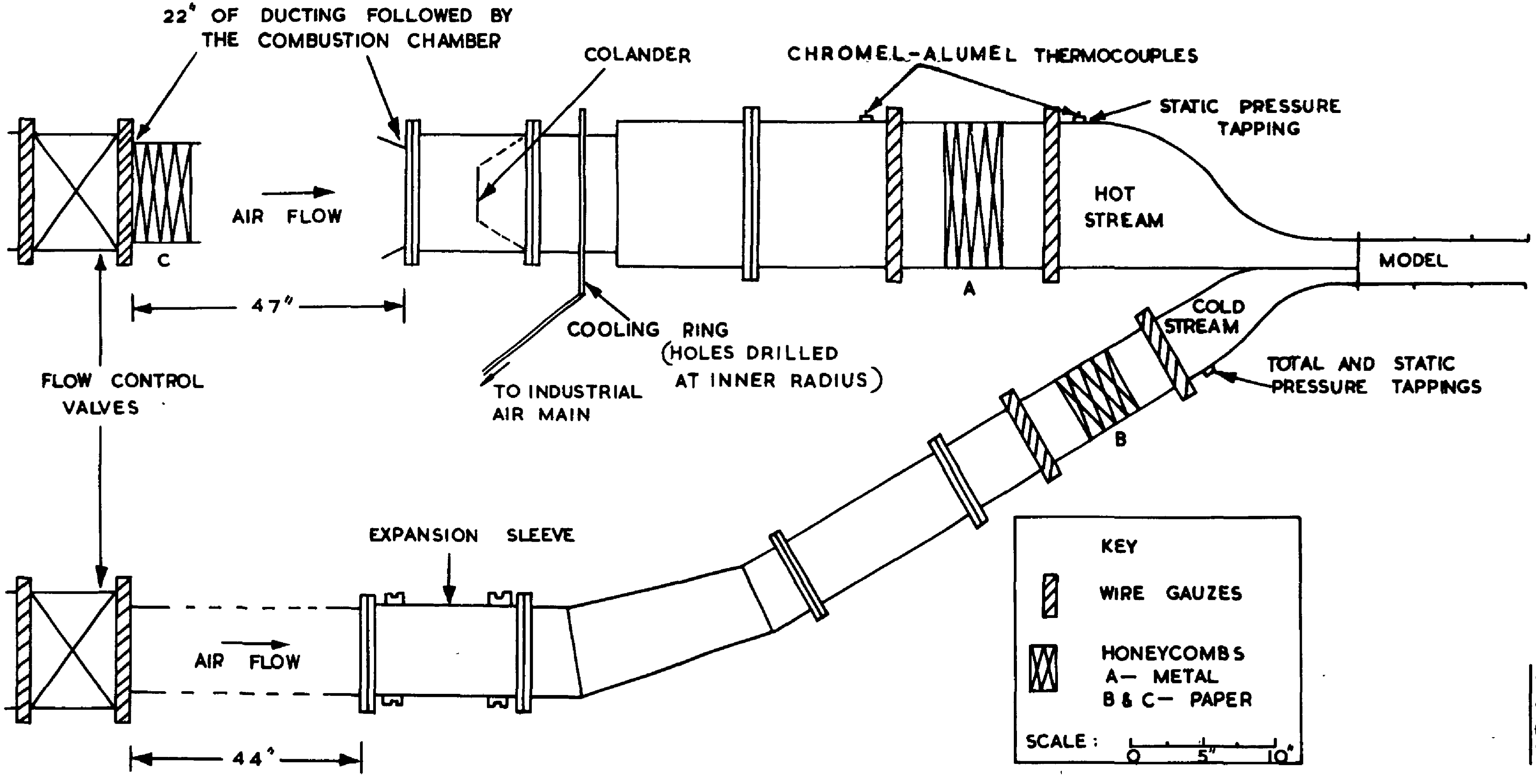
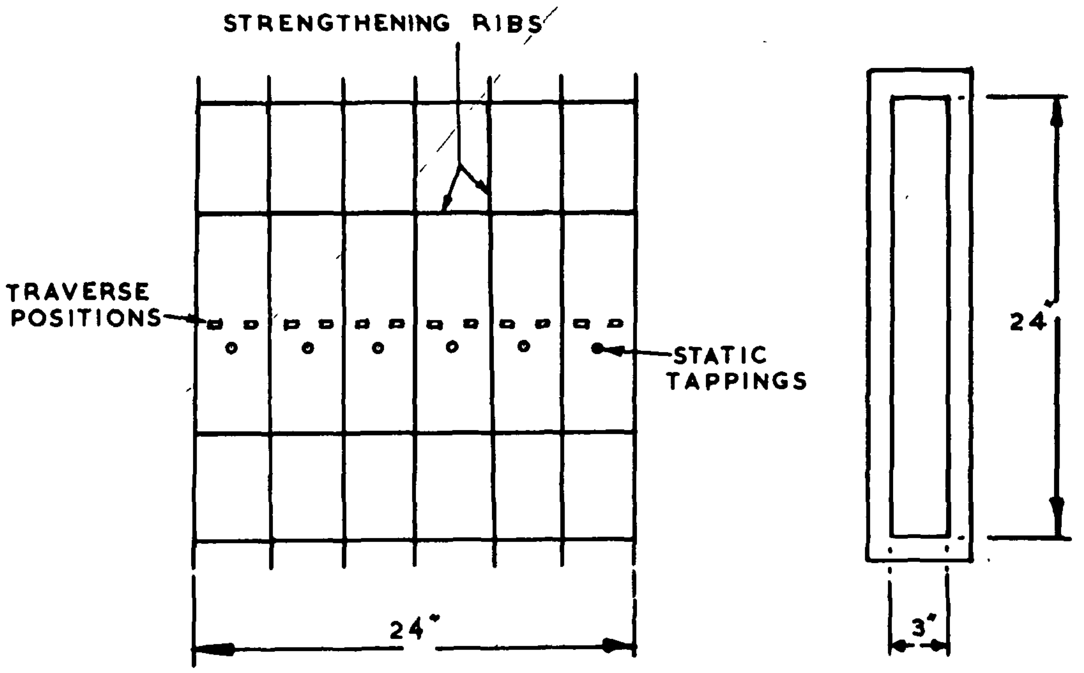
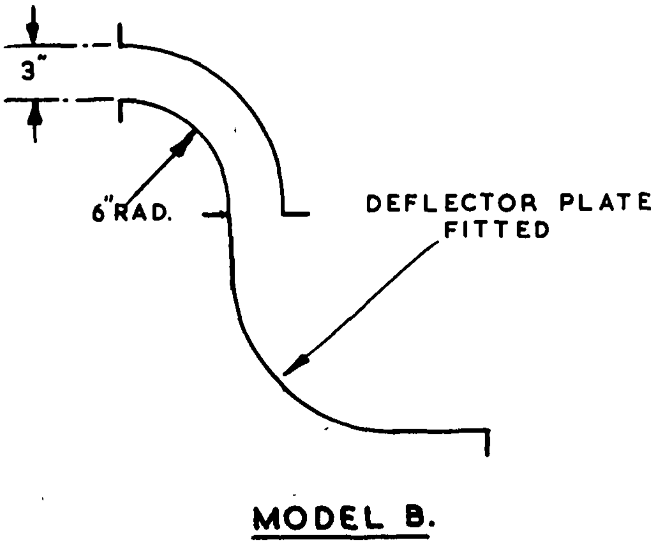


FIG.4

FIG.5



MODEL A.

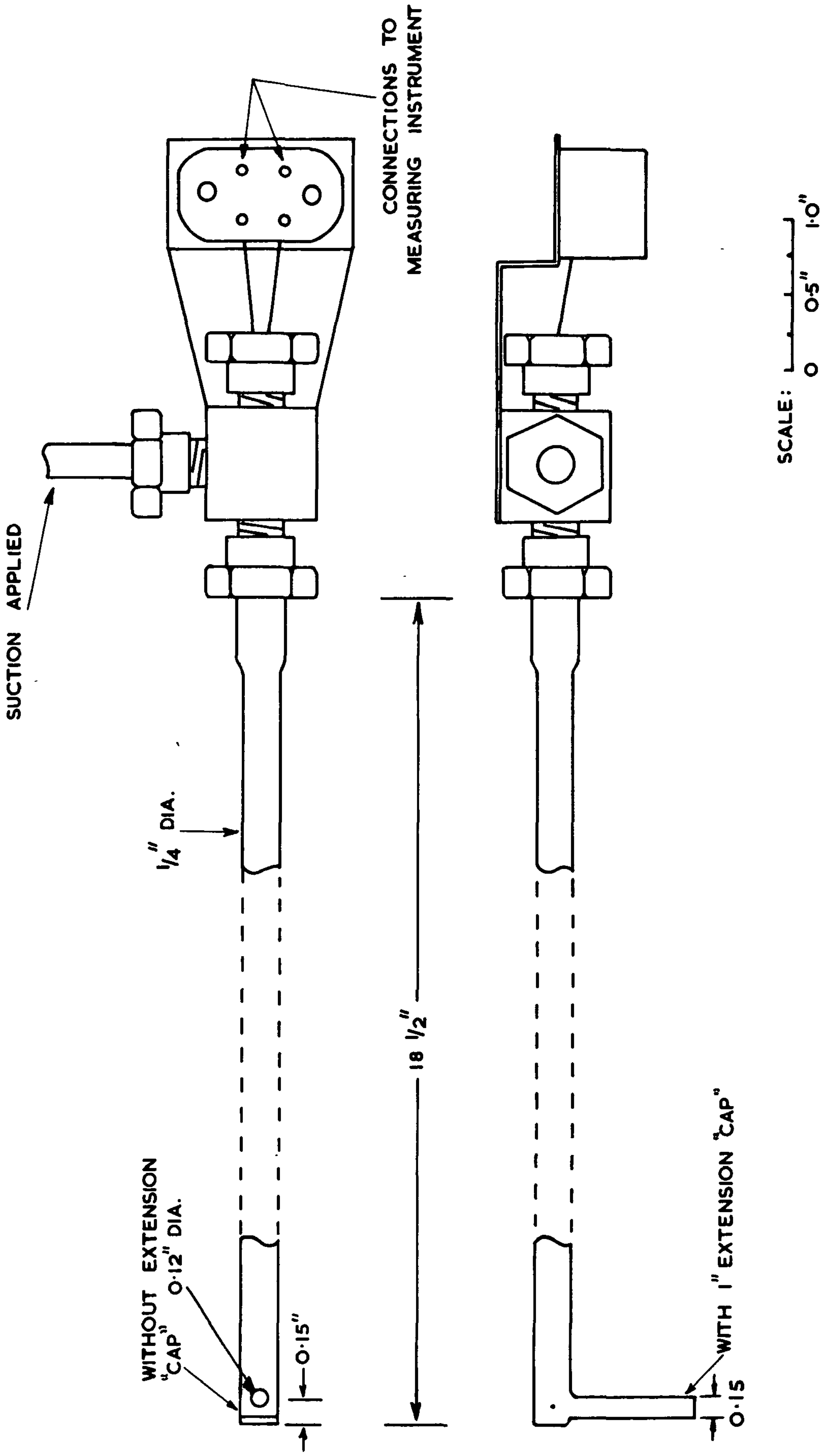


MODEL B.

SCALE: 0 5" 10"

THE TEST MODELS.

FIG. 6



THE SONIC SUCTION THERMOCOUPLE
(CHROMEL ALUMEL)

LINE DIAGRAM OF TRAVERSING HEAD.

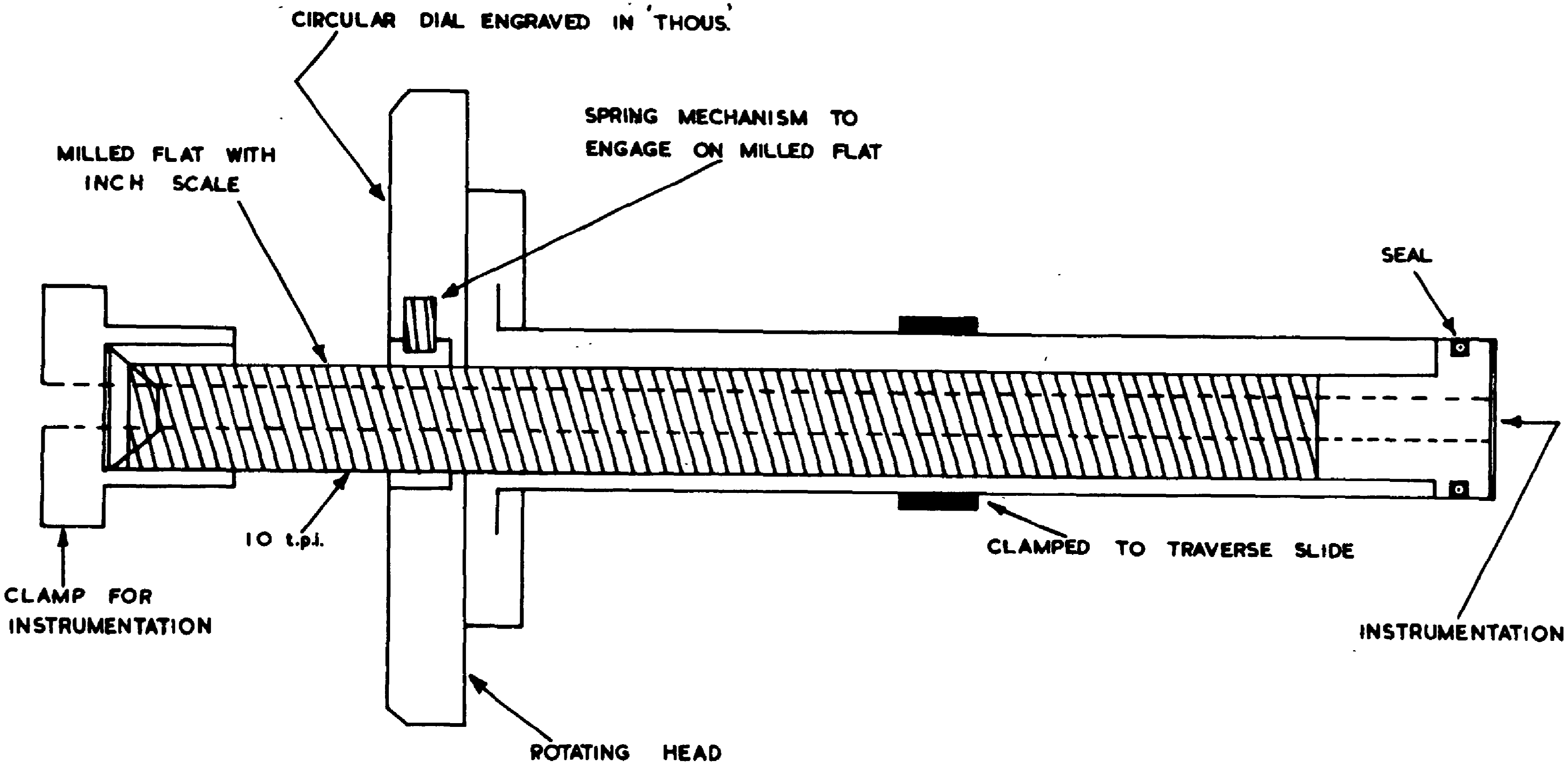


FIG.7

FIG. 8

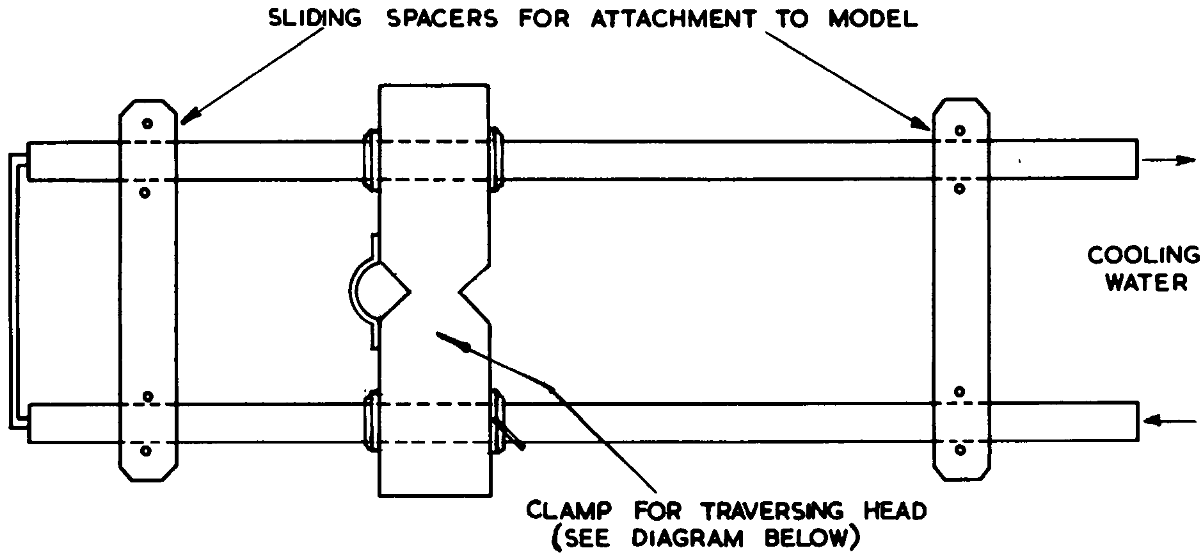


FIG. 8a. THE SLIDING GEAR.

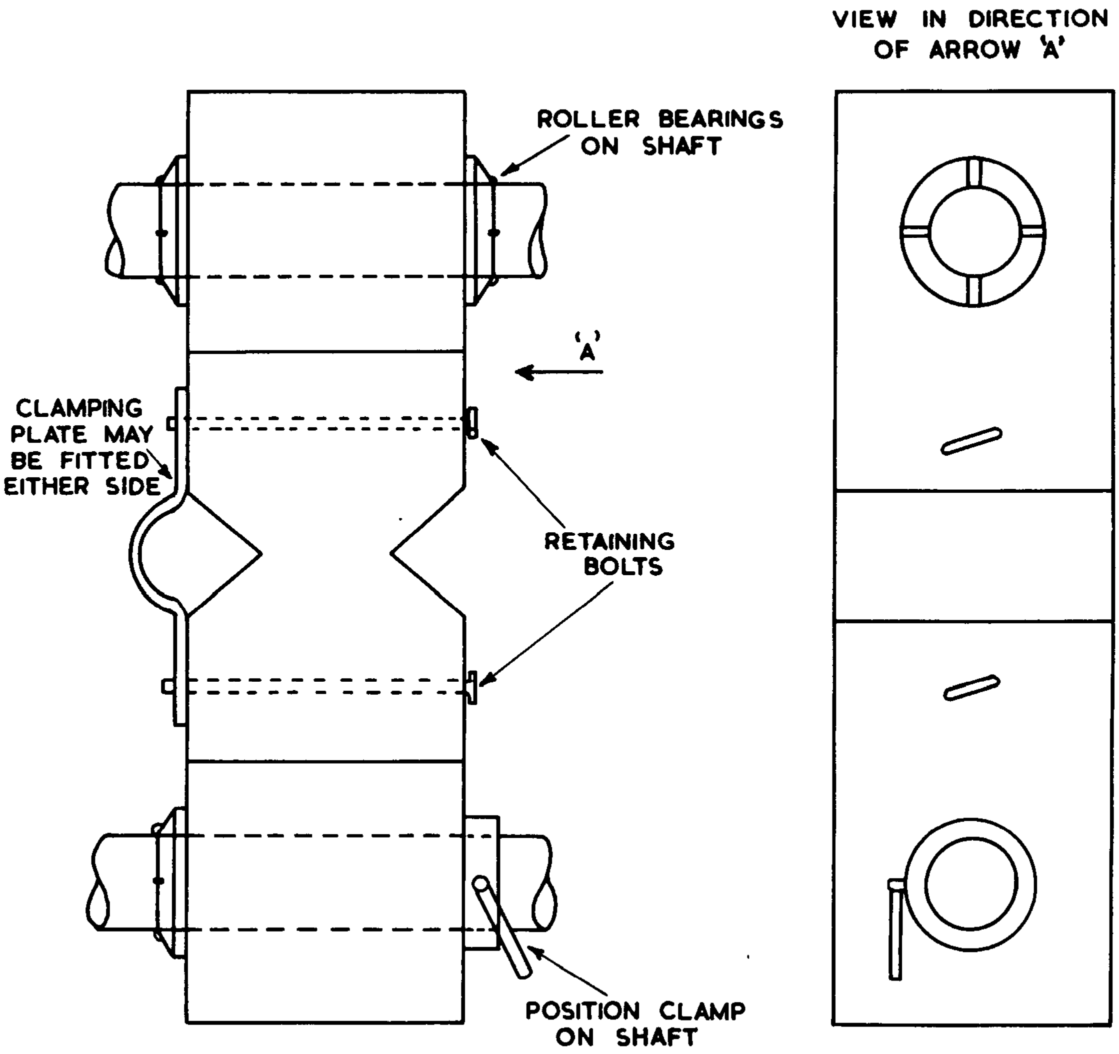
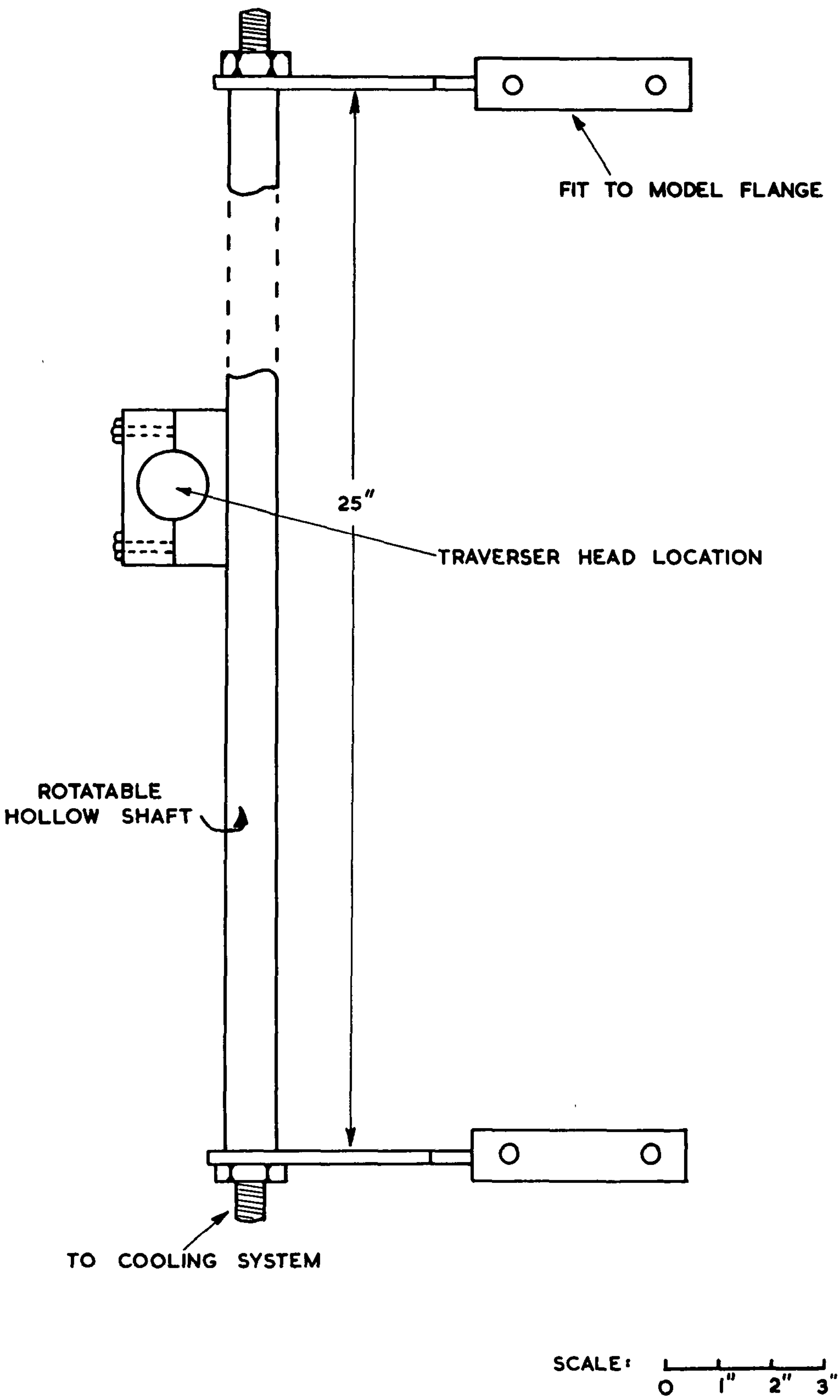


FIG. 8b. CLAMP FOR THE TRAVERSING HEAD.

THE TRAVERSER FOR THE STRAIGHT MODEL.

FIG. 9



TRAVERSER FOR THE CURVED MODEL.

FIG. 10

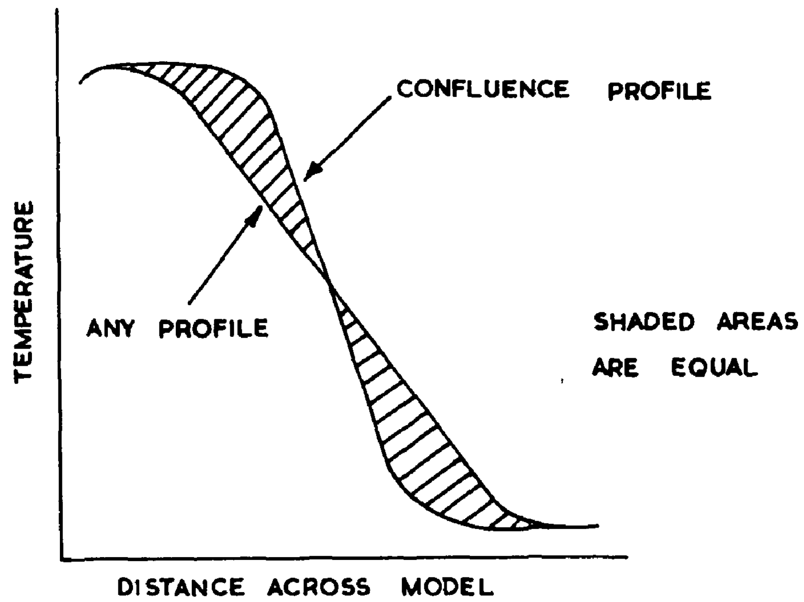


FIG. 10a. DETERMINATION OF ANGLE BETWEEN STREAMS.

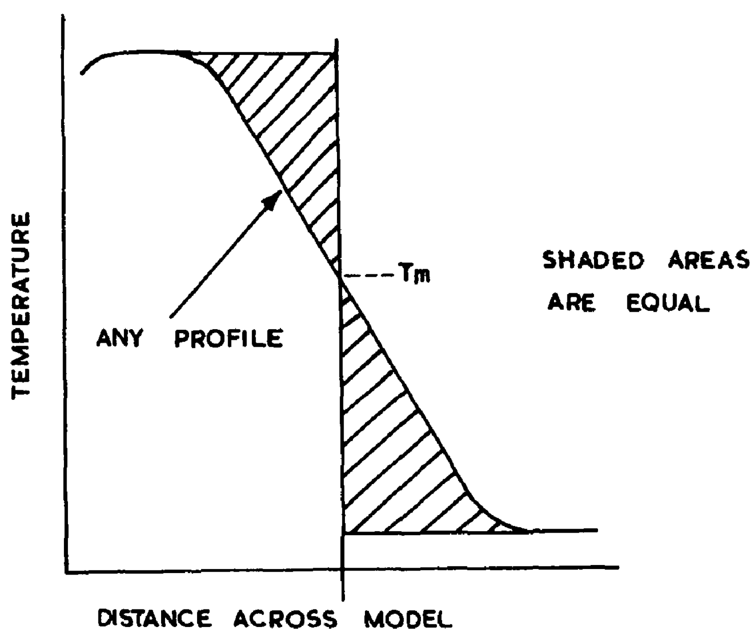
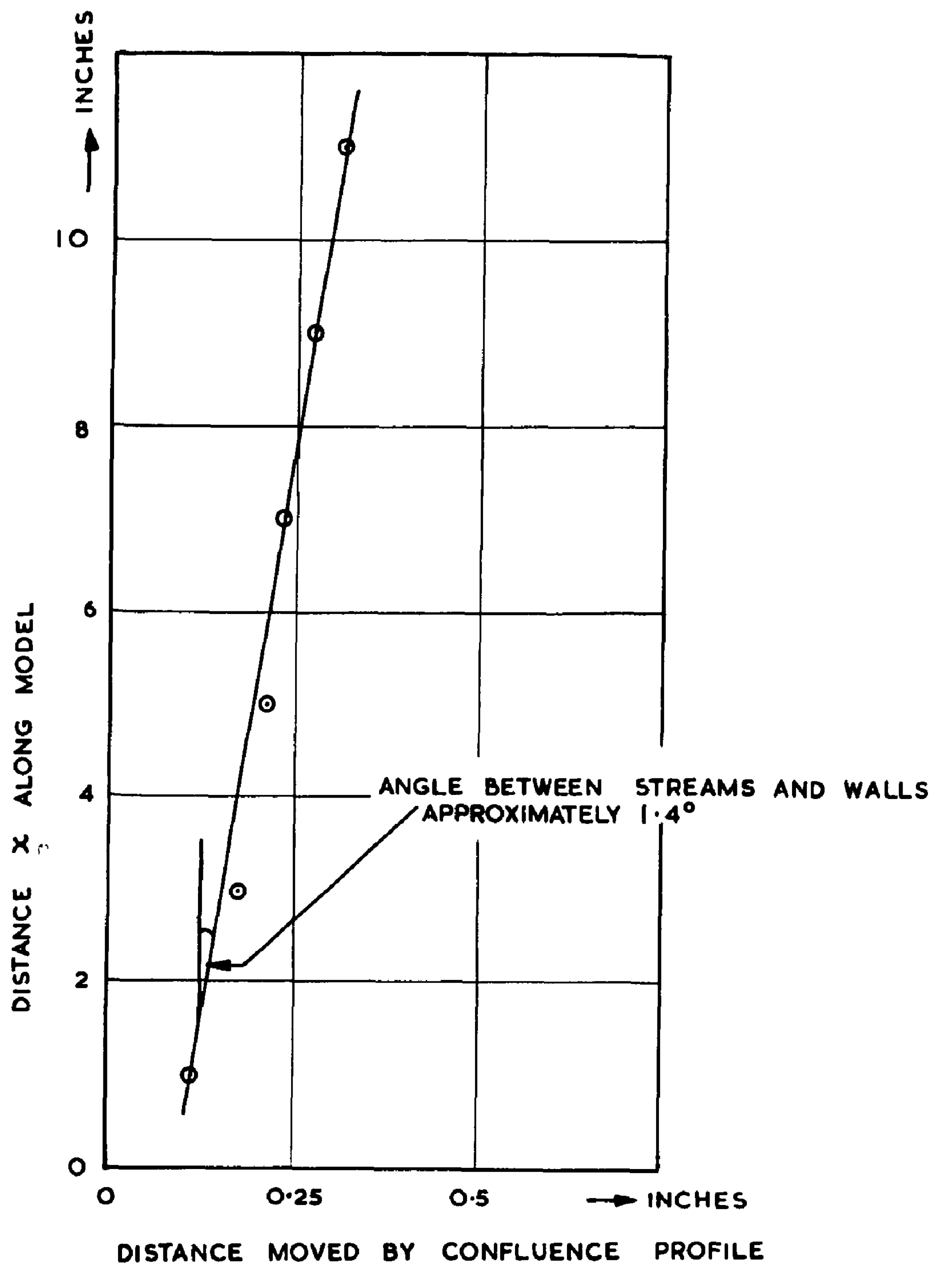


FIG. 10b. DETERMINATION OF β.

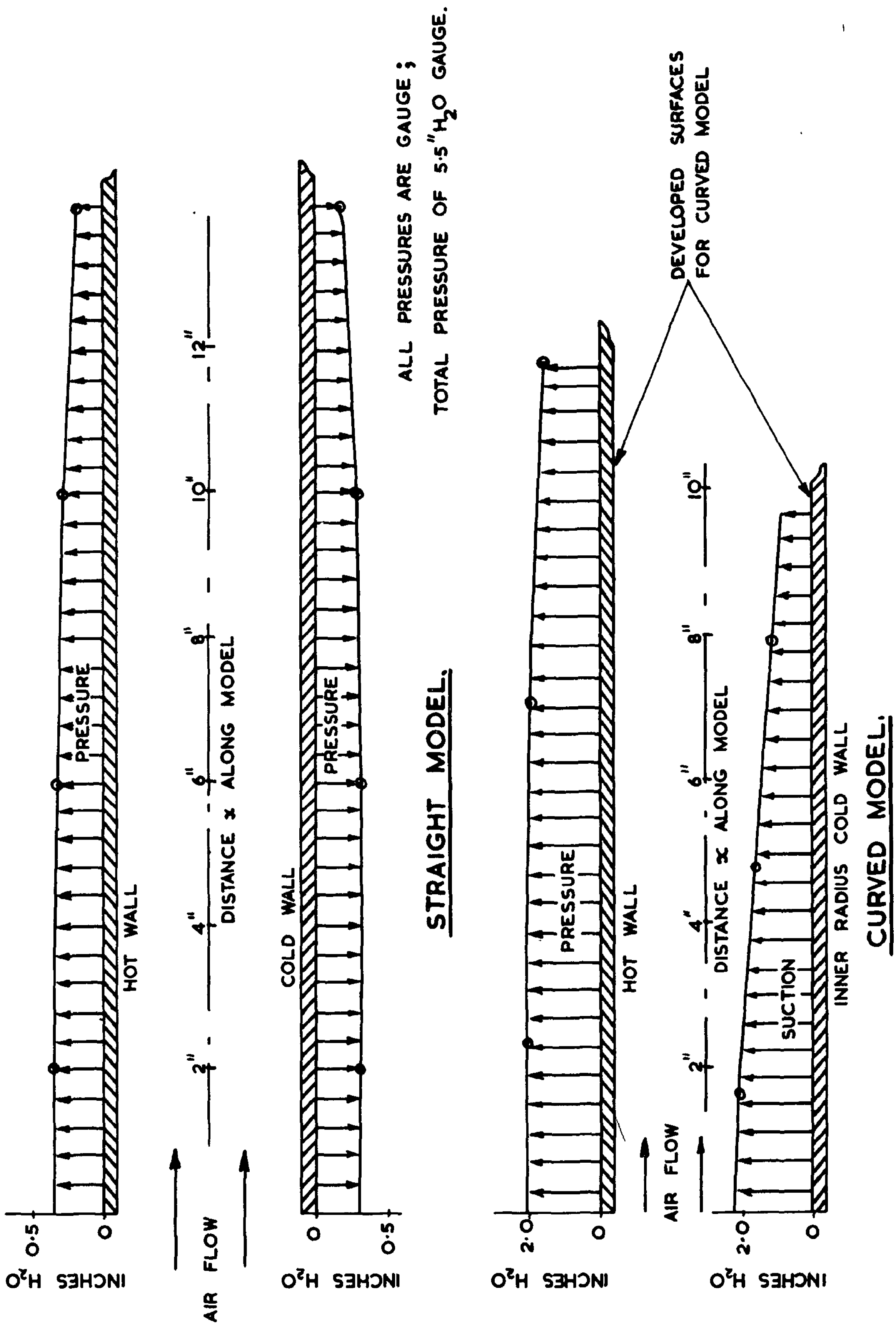
METHODS USED IN ANALYSIS OF THE EXPERIMENTAL RESULTS.

FIG. 11



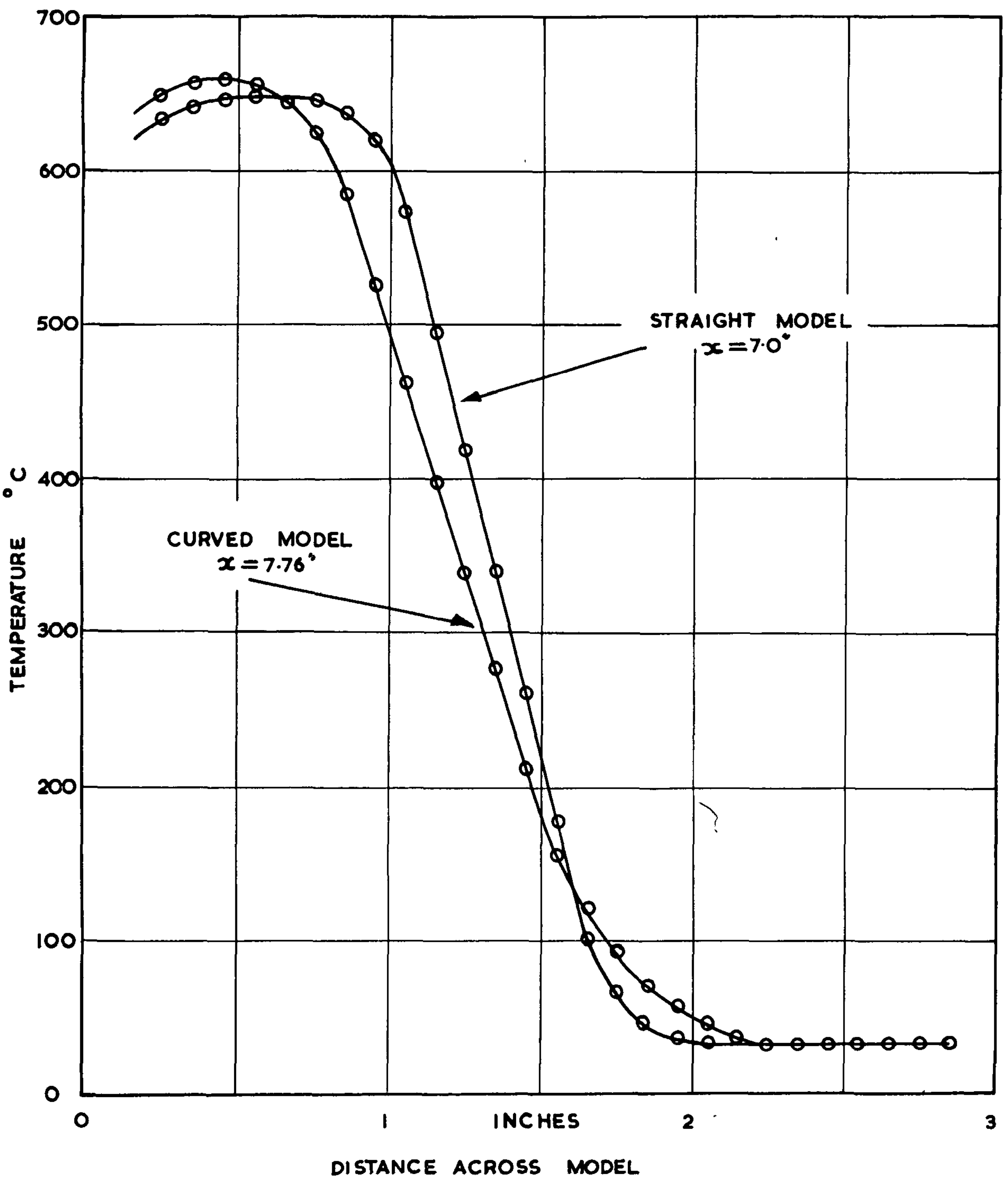
INCLINATION OF THE TWO STREAMS.

FIG.12



STATIC PRESSURE FOR STRAIGHT AND CURVED MODELS.

FIG. 13



TYPICAL TEMPERATURE PROFILES FOR STRAIGHT AND CURVED MODELS.

FIG. 14

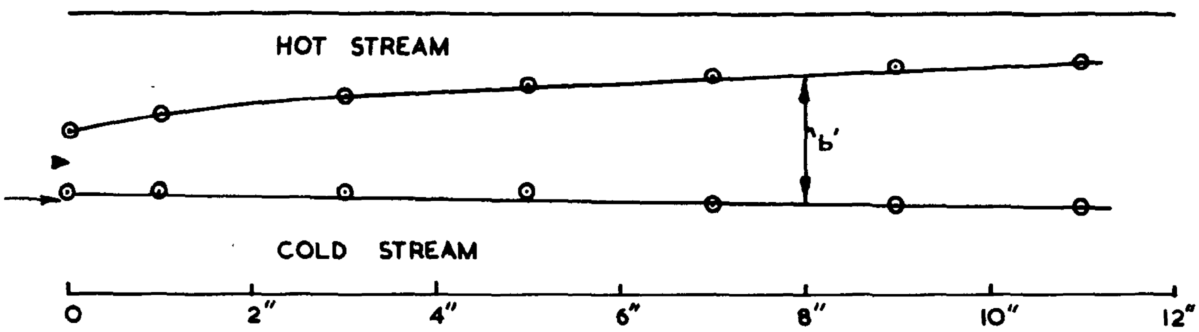


FIG. 14 a. THE STRAIGHT MODEL.

THE ARROWS INDICATE THE POSITION OF THE 'SPLITTER PLATE' PRIOR TO TEST.

THE TRIANGLES INDICATE THE ESTIMATED POSITION OF THE 'SPLITTER PLATE' DURING TESTS.

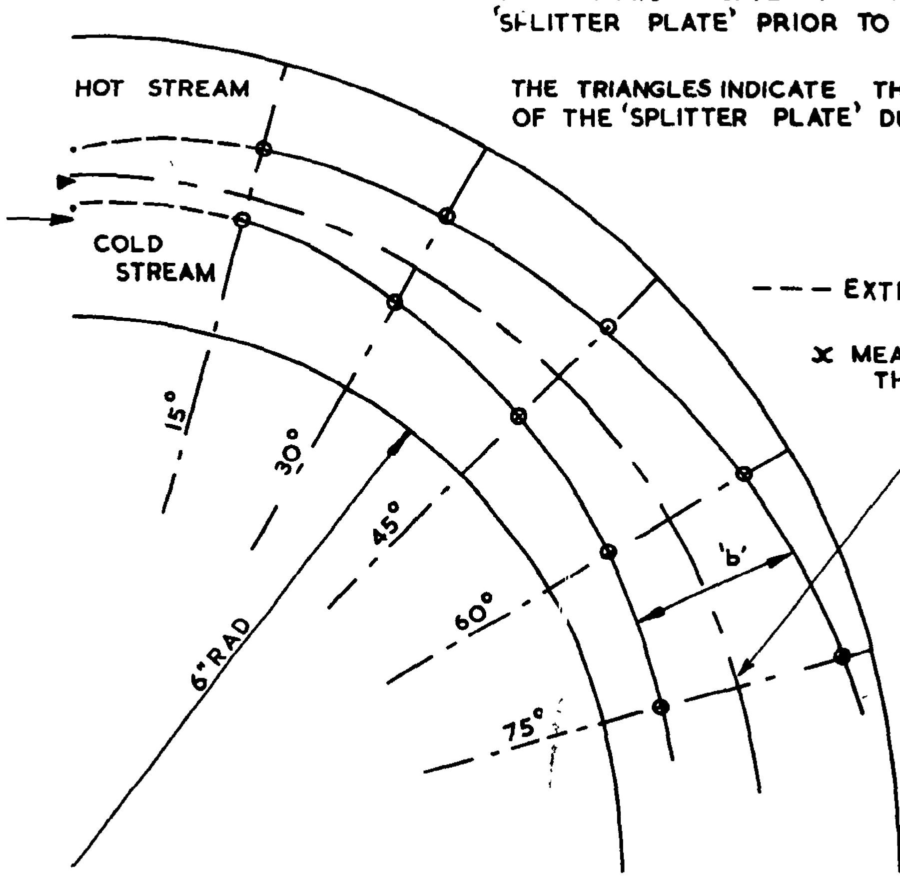
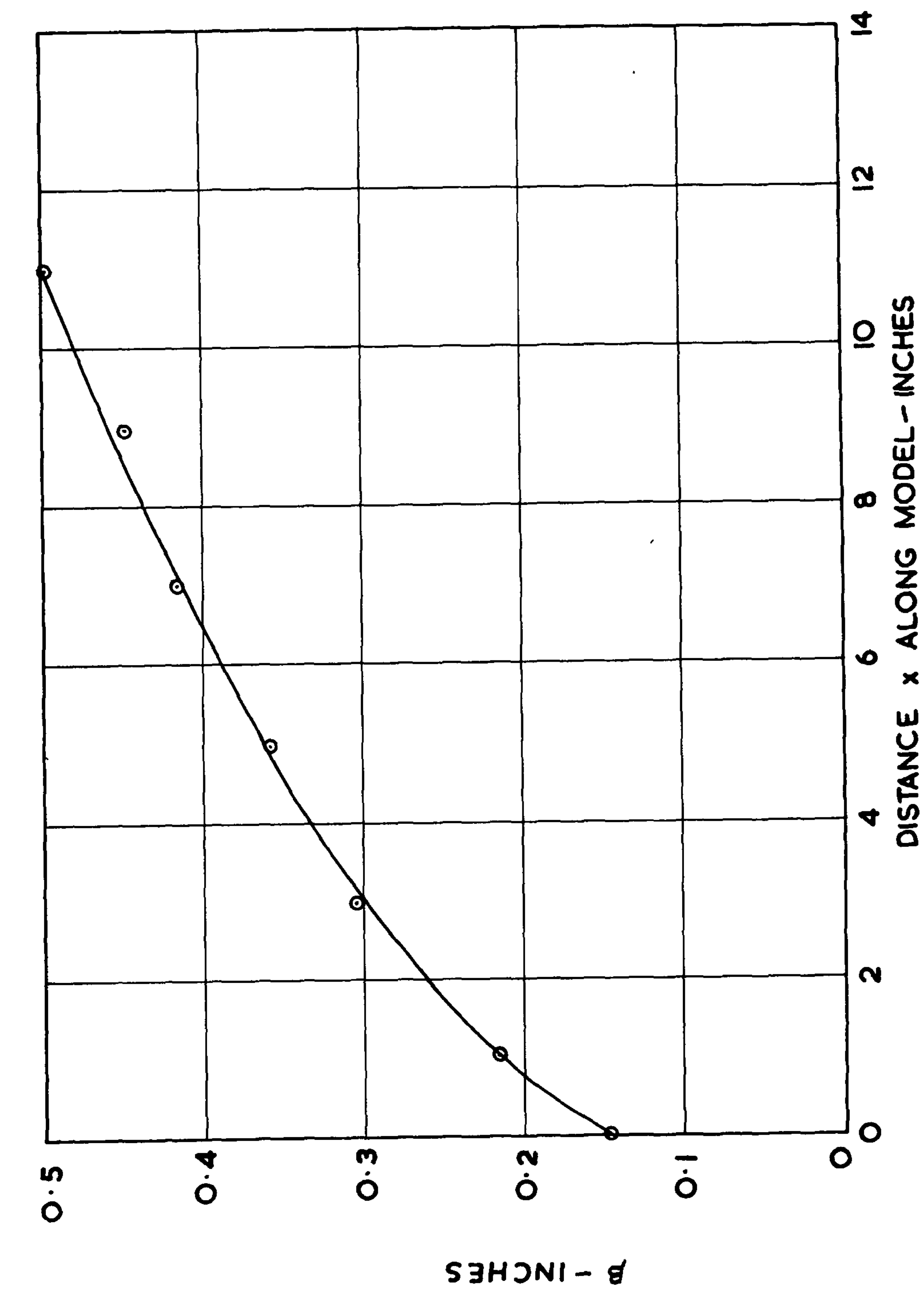


FIG. 14 b. THE CURVED MODEL.

FIGS. 14 a & 14 b DRAWN TO SAME SCALE.

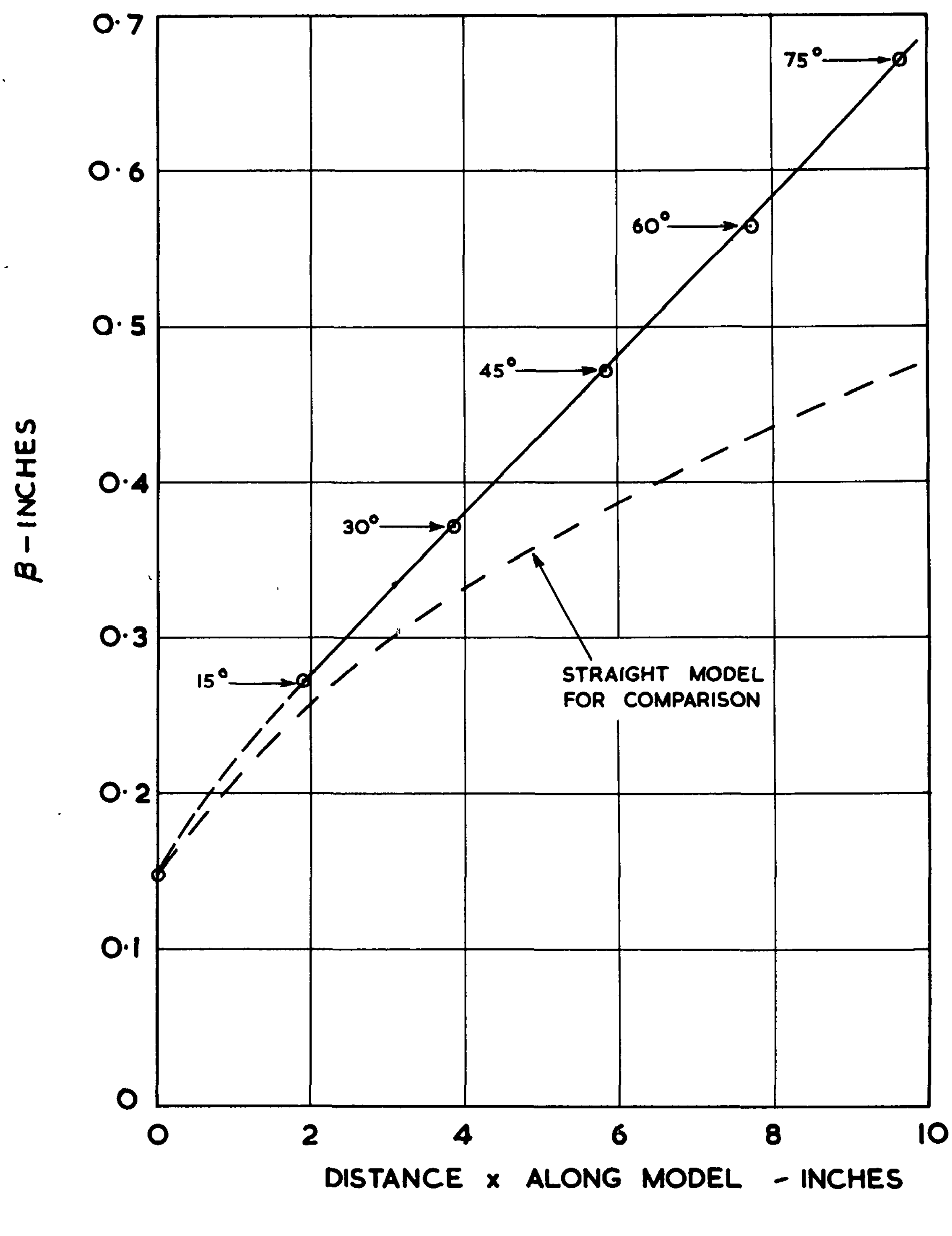
DETERMINATION OF MIXING WIDTH FROM TEMPERATURE PROFILES.

FIG. 15



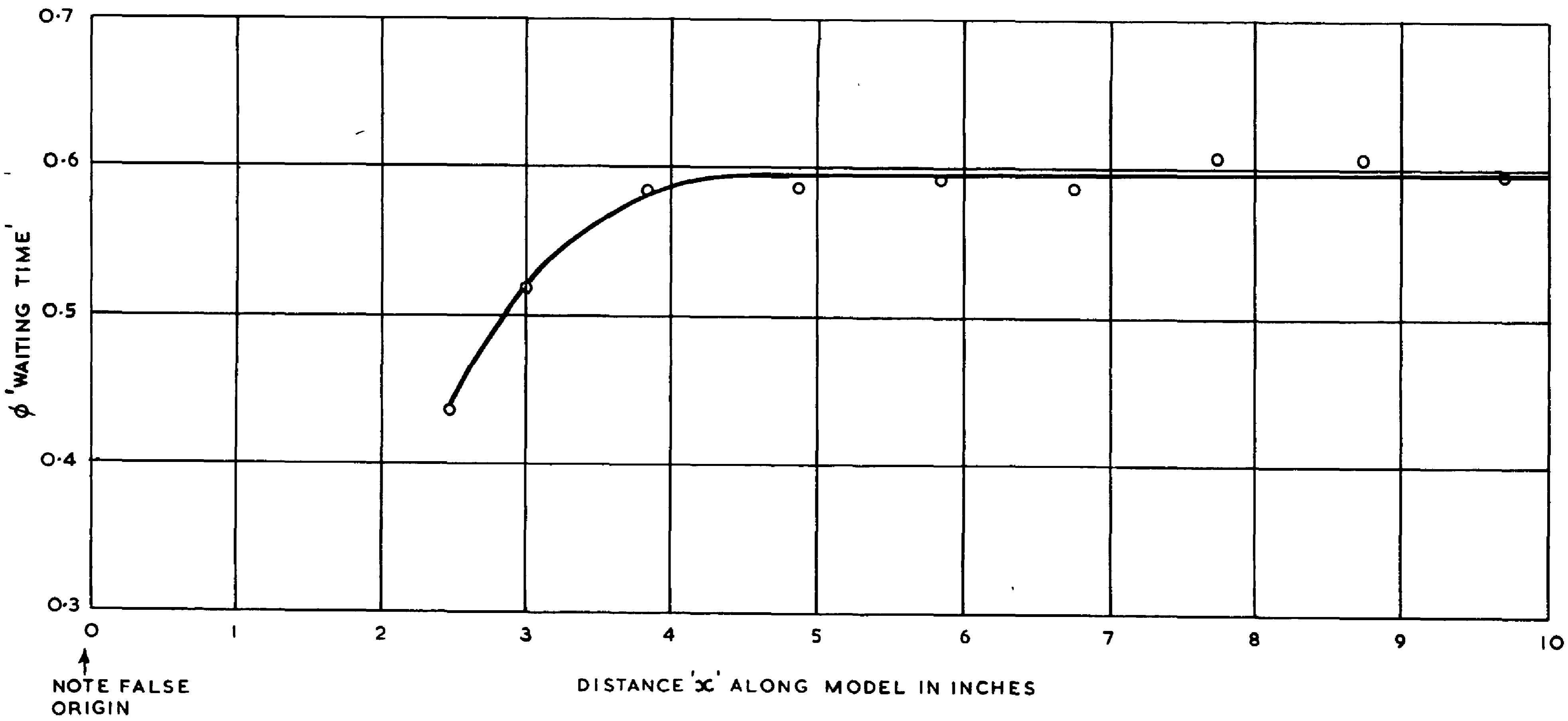
MIXING WIDTH β BASED ON HEAT TRANSFER FOR THE STRAIGHT MODEL

FIG. 16



MIXING WIDTH β BASED ON HEAT TRANSFER FOR THE CURVED MODEL

$\phi = \dfrac{\text{TIME BETWEEN MOVEMENT}}{\text{TIME FOR MOVEMENT}}$



EXPERIMENTAL ESTIMATE OF THE WAITING TIME ϕ

FIG.17

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The mixing rate attainable between hot and cold airstreams influences the length and weight of aircraft gas turbine engines. Previous work has suggested that an improvement might be obtained by causing the mixing to take place in the centrifugal field generated by a curved flow path.

Theory indicates that when the stream of higher total pressure is at the inner radius of the turn the mixing is augmented by the eddies of higher total pressure being centrifuged differentially through the eddies of lower total pressure. When the initial total pressures of the two

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P.T.O.

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A limited experimental investigation indicates increases in the local mixing rate of up to 100 per cent, when the total pressures of the two streams are equal.

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