EFFECT OF LENGTH

ON

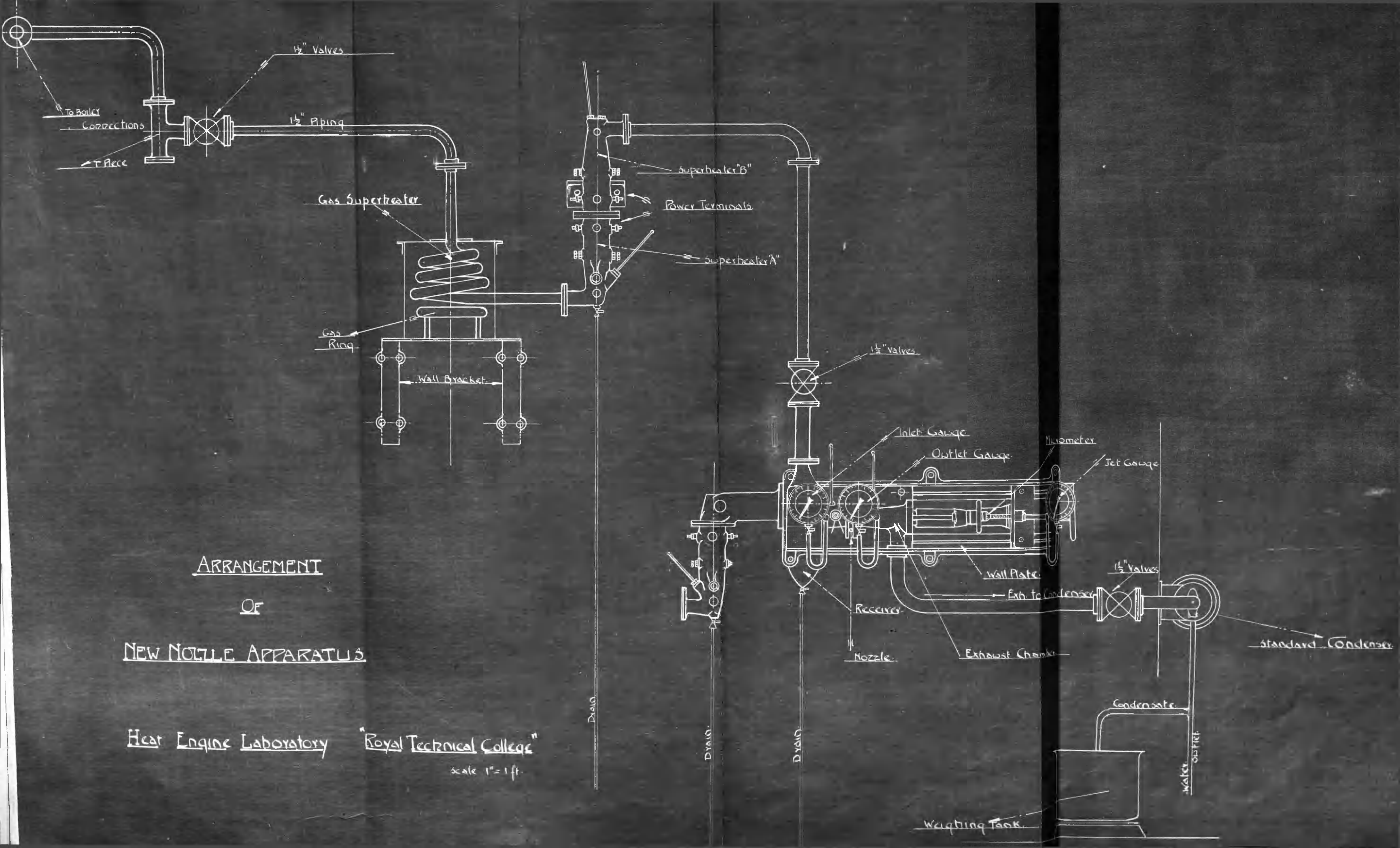
N O Z Z L E   L O S S E S

AND

THE JET ACTION OUTSIDE THE NOZZLE REGION.

BY

MOSTAFA R. YOUSSEF, B.Sc., A.R.T.C.


To Boiler Connections
1½" Valves
1½" Piping
T Piece
Gas Superheater
Superheater "B"
Power Terminals
Superheater "A"
Gas Ring
Wall Bracket
Drain
1½" Valves
Inlet Gauge
Outlet Gauge
Micrometer
Jet Gauge
Wall Plate
1½" Valves
Exh. to Condenser
Receiver
Nozzle
Exhaust Chamber
Standard Condenser
Drain
Drain
Condensate
Water Outlet
Weighing Tank
ARRANGEMENT
OF
NEW NOZZLE APPARATUS
Heat Engine Laboratory "Royal Technical College"
scale 1" = 1 ft

THE EFFECT OF LENGTH ON NOZZLE LOSSES.

INTRODUCTORY.

The subject of steam nozzles is by no means a new one, as it has been cornered by many investigators in the last few years.

A considerable amount of this has been carried out in the Heat Engine Laboratories of The Royal Technical College by Drs. Mellanby and Kerr. They adopted in their first paper [1] a method combining the measurement of flow and search tube investigations along the axis of the nozzle. They deduced a number of equations and formulae which made the analysis of nozzles possible. From their investigations it would seem that the convergent-parallel nozzle is the most suitable for detailed examination, and in a paper [2] read before the N.E.C. Inst. of Engineers and Shipbuilders, they showed that the total losses in nozzles are divisible into two parts namely friction and entry losses. It may be here mentioned that in all their experiments they used round nozzles, but Dr. D. S. Anderson has confirmed the method of their analysis by applying it to rectangular nozzles.[3] It was found necessary that the writer should carry out the present work to provide a further link in the chain of nozzle research.

The losses were investigated by a series of pressure flow experiments. The flow quantities were noted down at several periods during the experiment and a mean of these was

1. "Steam Action in Simple Nozzle Forms" B.A.Section G. August, 1920.
2. "Losses in Convergent Nozzles" N.E.C.I. of Eng. & Ship::builders, February, 1921.
3. "Losses in Nozzles of Rectangular Cross Section" Greenock Soc. Engs. & Shipbuilders: March, 1922.

FIG II

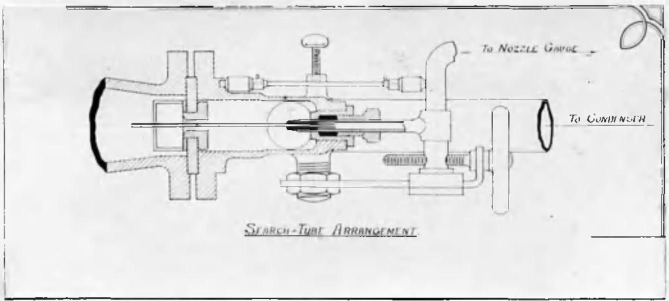
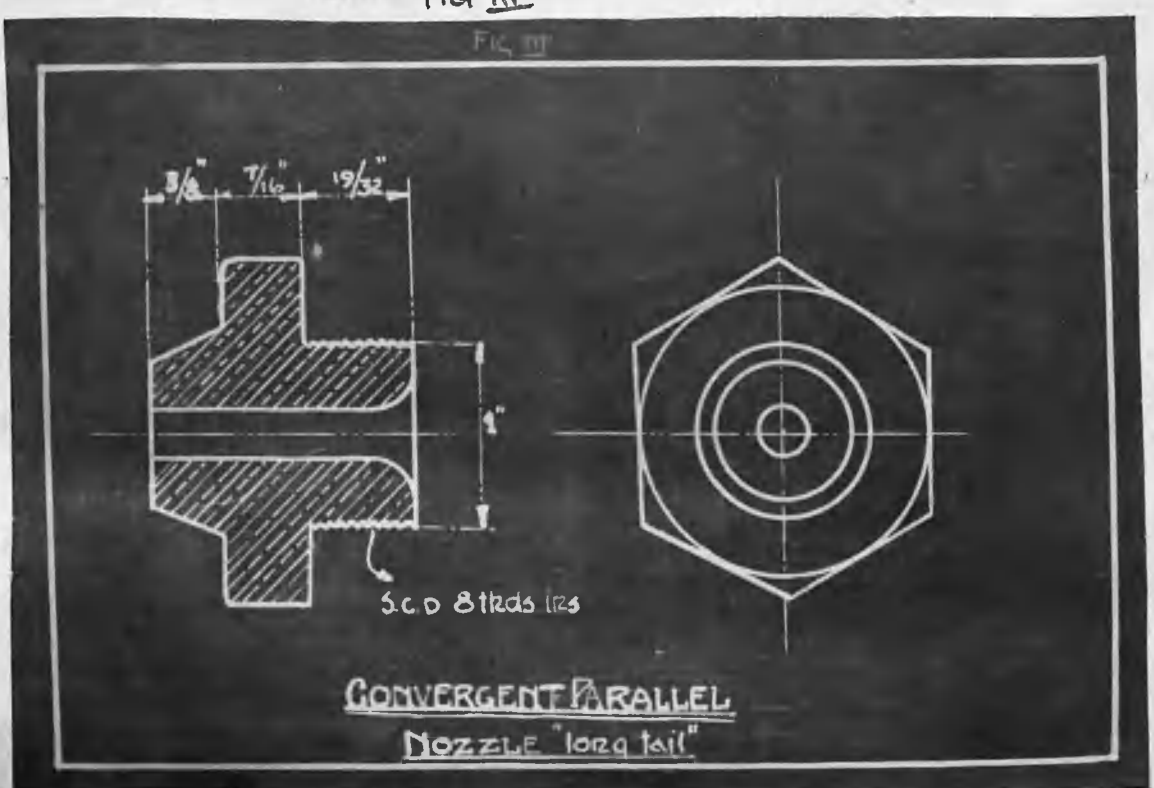


FIG III



namely 20, 40 and 50 lbs. gauge were taken. The nozzle was cut down by a $\frac{1}{4}$ of an inch for every series. Reference to these tests will be hereinafter called as Case I, Case II and Case III according to the different back pressures and Series I, II, III etc., according to the shortened length.

To avoid complications due to supersaturation in the nozzle, the steam was superheated to a fairly high degree. It will be seen from Table I that the temperatures were constant over a fairly wide range as it was practically impossible to get a finer adjustment over the gas rings heating the steam before its entry to the nozzle.

Fig III shows the nozzle before it was cut down. It is made of soft brass and is machine finished and 1.404" total length, the parallel length was 1.144". All further details after the nozzle was cut down are given in Table I. The nozzle showed in the tail portion the marks of the turning tool, and a large number of faint rings or ridges being clearly perceptible. The duration of each test was between 30 and 35 minutes. From the search tube and micrometer readings, the pressure ratio (i.e ratio of the pressure at any point to the supply pressure) was obtained. On a base of distance along the nozzle axis,pressure ratio curves were drawn, this along with the flow quantity and initial steam conditions was all that is necessary for the analysis of the nozzle.

All gauges, thermometer and weighing tanks were calibrated.

The "total length" of the nozzle is given on all the graphs.

THEORETICAL ANALYSIS.

The method of analysis used in the present work, has been adopted from Drs. Mellanby's and Kerr's investigations, due acknowledgement being here made.

This is given fully in their papers mentioned at the beginning of the present work, but it was thought advisable to give an outline of the theory involved.

The expression connecting the flow quantity, cross sectional area and initial fluid condition is

*
$$\frac{G}{A} = \frac{1}{144}\sqrt{\frac{288\,gn}{n-1}\cdot\frac{P_1}{V_1}} \cdot \frac{r\left(1-r^{\frac{n-1}{n}}\right)^{\frac{1}{2}}}{r^{\frac{n-1}{n}}}$$

which is founded on the hypothesis, that when steam is expanded adiabatically all the work done goes to create onward velocity, the adiabatic index n is fixed and unalterable.

G	=	Flow quantity	lbs/sec.
A	=	Cross Sectional area	ins^2
P_1	=	Initial pressure	lbs/ins^2 abs.
V_1	=	Initial specific volume	$ft^3/lb.$
r	=	Pressure ratio $= P/P_1$	

For superheated steam $n = 1{\cdot}3$

$\therefore$ substituting this value for n

and also substituting a for $\frac{n-1}{n}$, the above expression reduces to:

$$0{\cdot}718\left(\frac{V_1}{P_1}\right)^{\frac{1}{2}}\frac{G}{A} = \frac{r\left(1-r^a\right)^{\frac{1}{2}}}{r^a} \qquad (1)$$

The left hand side of (1) is less than the theoretical, hence it was found necessary to introduce a loss factor "K" to the right hand side of (1) so that both sides may become identical.

The form of the expression may be written as:

$$F = \left(\frac{V_1}{P_1}\right)^{\frac{1}{2}}\cdot\frac{G}{A}\times{\cdot}718 = \frac{r\left(1-K-r^a\right)^{\frac{1}{2}}}{r^a+K} \qquad (2)$$

where K is always positive, and the expression as in (2) becomes applicable to any form of actual expansion.

* Steam action in simple nozzle forms, B.A. section G. August 1920
+ Losses in convergent nozzles N.E.C.I of Eng. Shipbuilders February 1921 By Profs A.L. Mellanby D.Sc, etc, + Wm Kerr PhD. etc.

F is known as the jet function. The quantities in the expression are either got by measurement or calculation. V_1 is calculated by means of Callendar Equation for superheated steam where:

$$V_1 = 0.5948 \cdot \frac{T_1}{P_1} - 0.4213\left(\frac{671.6}{T_1}\right)^{\frac{10}{3}} + .016 \qquad (3)$$

where. T_1 is in absolute degrees.

Considering again the left hand form of F, it is seen that for fixed initial and final conditions G will be constant and the only variable will be A. Hence the values of F obtained under these conditions will vary along the nozzle length. If, however, attention is concentrated on one particu::lar section of the nozzle and the final conditions varied, V_1, A and P_1 will be constant and G the variable.

The values of the jet function will then give a flow curve to some scale for different pressure ratios of operation. The most convenient section is undoubtedly the nozzle outlet and the joining of these terminal points of F will give the flow curve. Having found F, K the loss factor could be easily spotted on the calculating chart Fig. X.

Knowing K other important expressions can be easily obtained.

Let the volume factor $= \frac{K + r^{n-1}}{r} = m$

Jet flow area varies as $\frac{1}{F}$

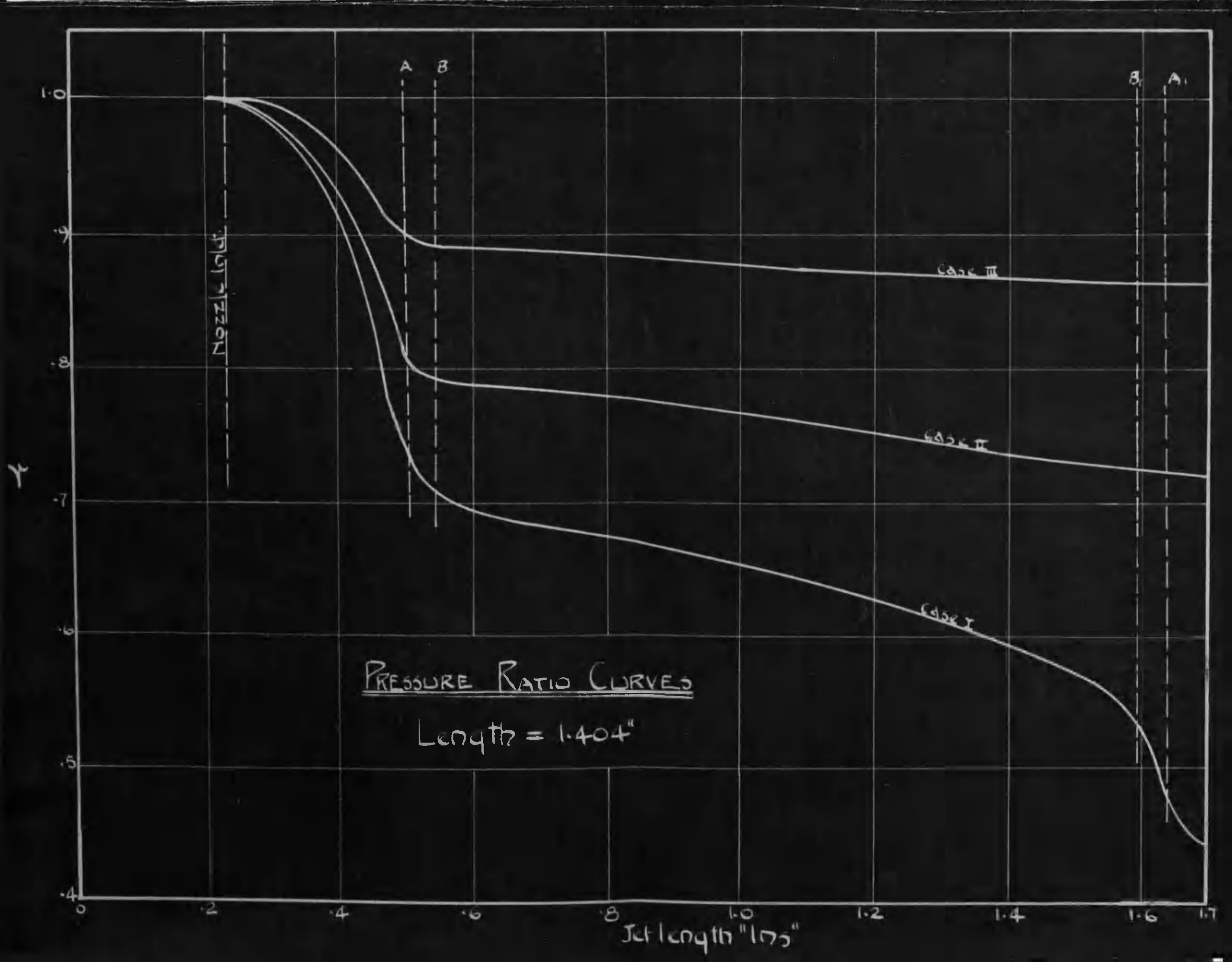
Jet energy varies as $(F_m)^2$ that is as $(1 - K - r^a)$

Flow velocity varies as (F_m) that is as $(1 - K - r^a)^{\frac{1}{2}}$

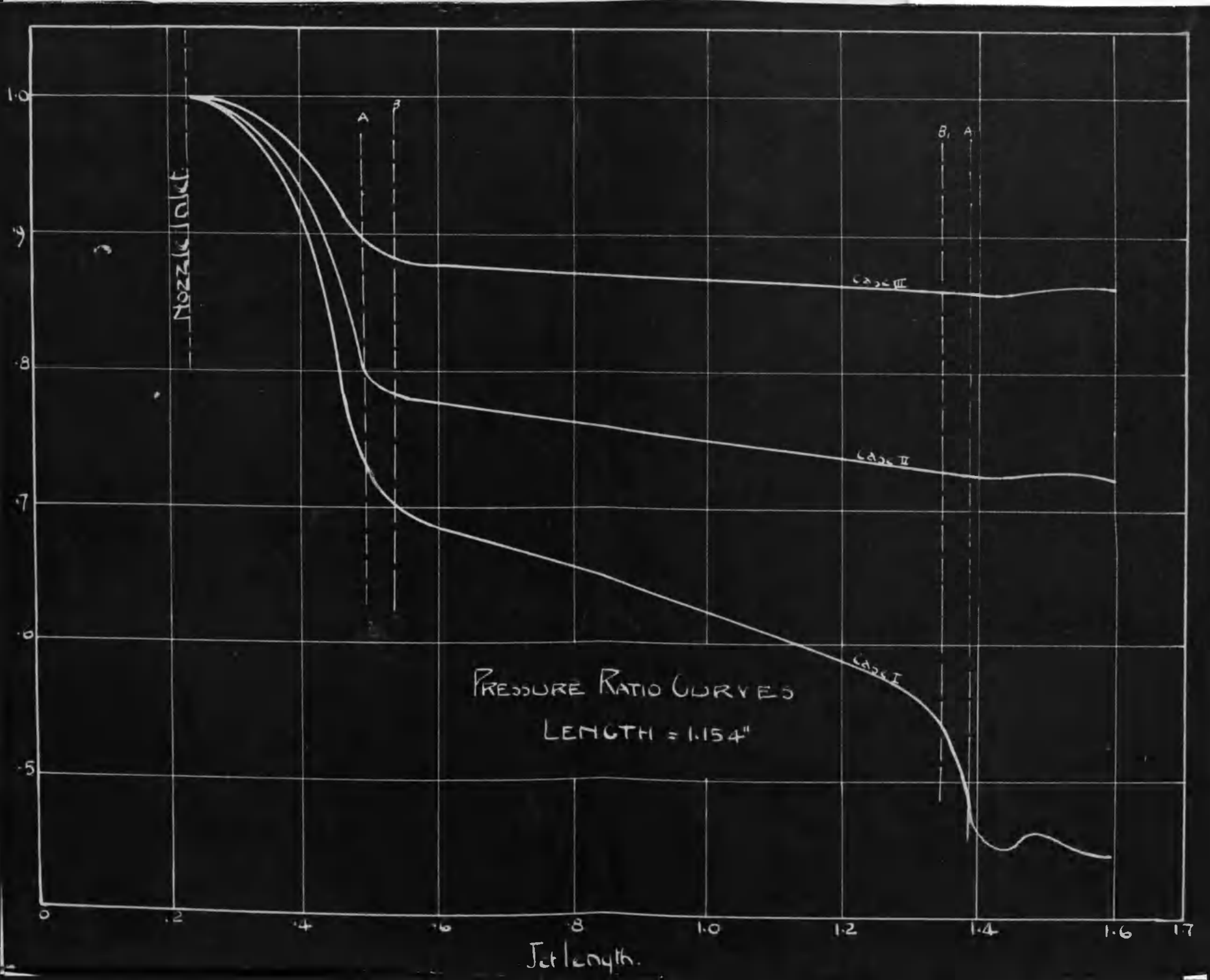
The expression for the nozzle efficiency is:

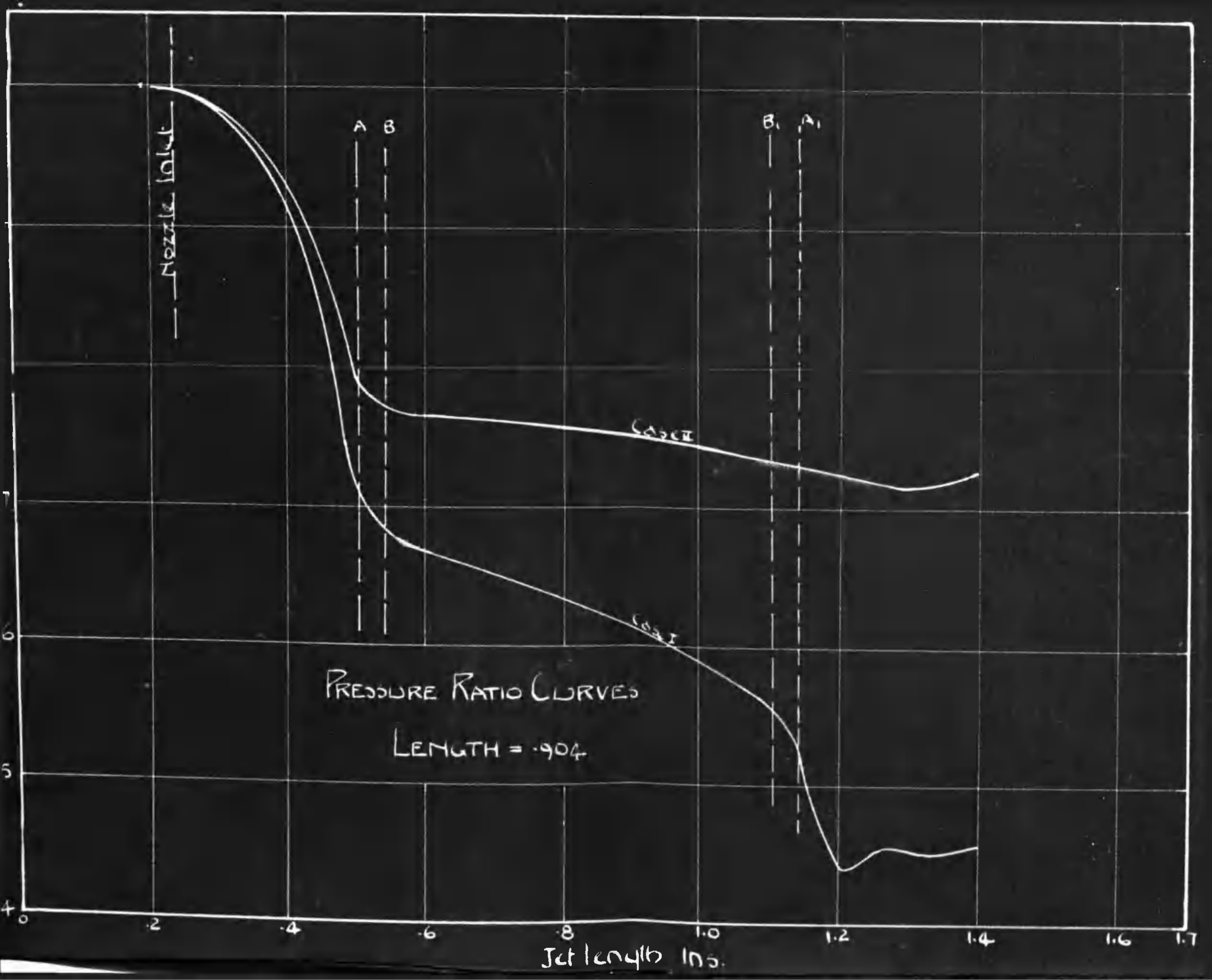
$$\eta = \frac{1 - K - r^a}{1 - r^a} \quad \text{-----------} \quad (4)$$

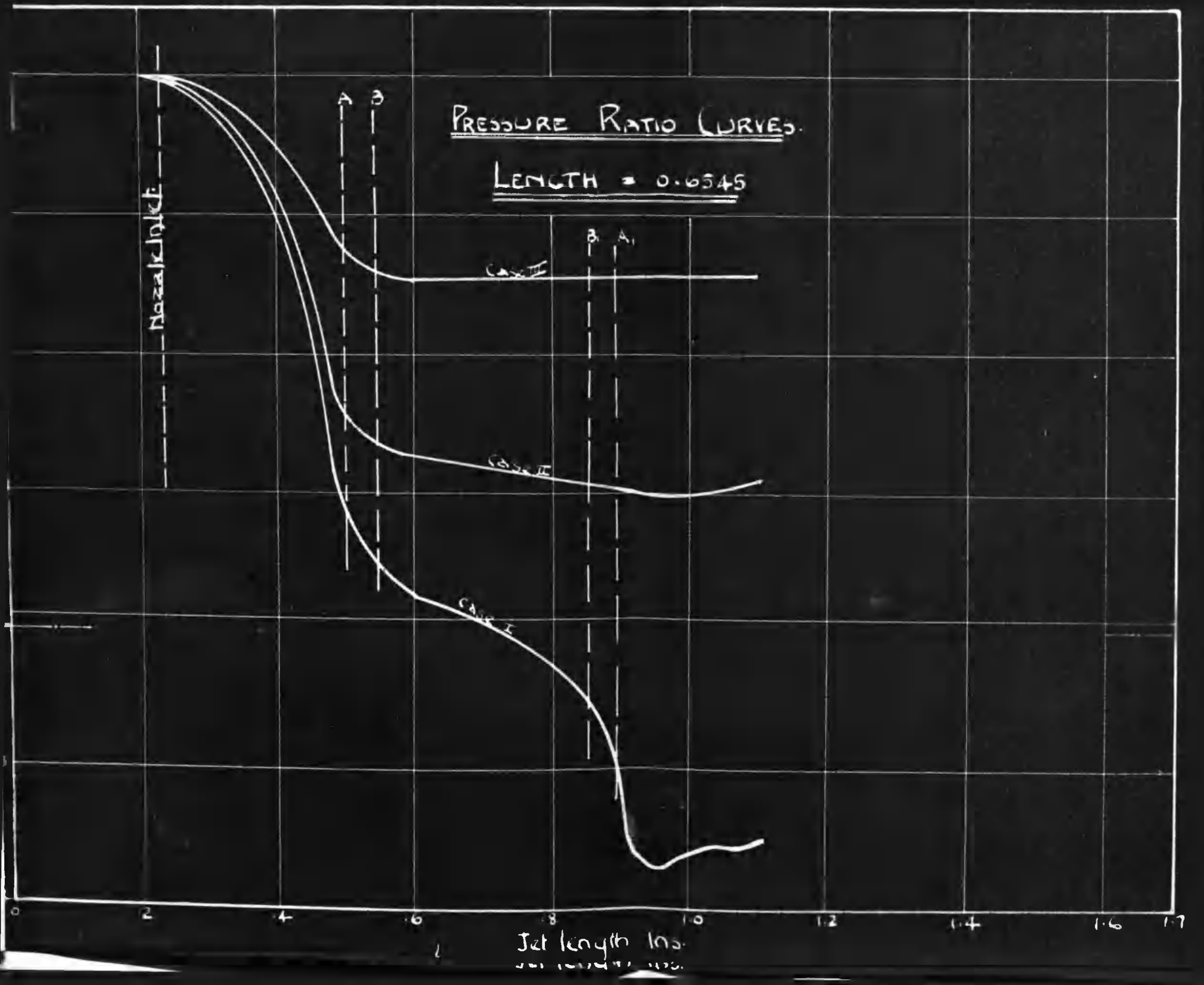
FIGS IV TO VIII



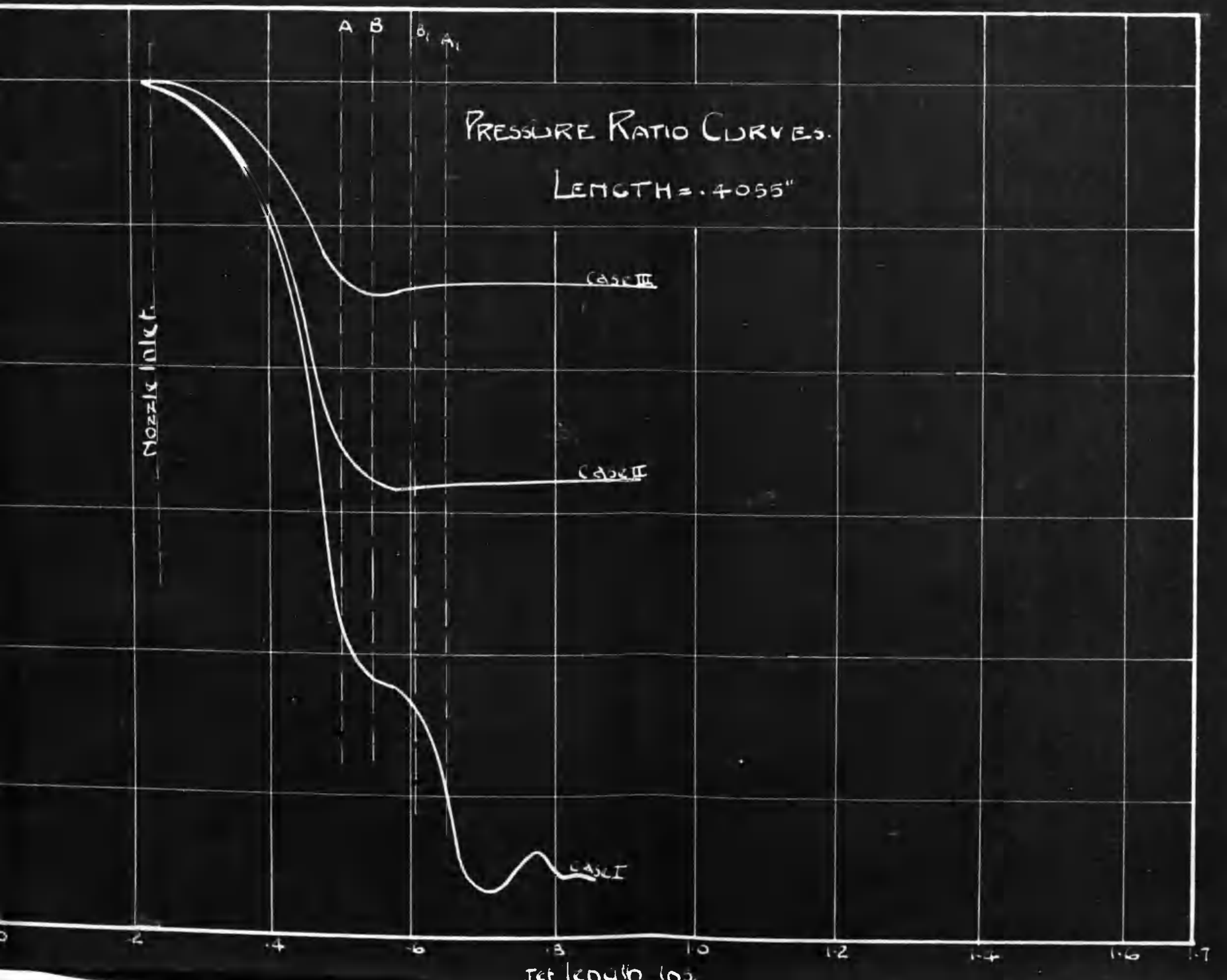
FIGS IV to VIII




Nozzle Inlet
A
B
B1
A1
Case II
Case I
PRESSURE RATIO CURVES
LENGTH = ·904
0
·2
·4
·6
·8
1·0
1·2
1·4
1·6
1·7
·4
·5
·6
·7
Jet length Ins.



PRESSURE RATIO CURVES.
LENGTH = 0.6545
Nozzle Inlet
A
B
B1
A1
Case III
Case II
Case I
0
.2
.4
.6
.8
1.0
1.2
1.4
1.6
1.7
Jet length Ins.



PRESSURE RATIO CURVES.
LENGTH = .4055"
A
B
B₁
A₁
Nozzle Inlet.
Case III
Case II
Case I
0
·2
·4
·6
·8
1·0
1·2
1·4
1·6
1·7
Jet length Ins.

The co-efficient of discharge C_d

$$\text{is } C_d = \frac{G \text{ actual}}{G \text{ theoretical}} = \frac{F \text{ actual}}{F \text{ theoretical}} = \frac{r^a . (1-K-r^a)^{\frac{1}{2}}}{(K+r^a)(1-r^a)} \quad (5)$$

The co-efficient of velocity is equal to the square root of the efficiency

$$\therefore \qquad C_v = \eta^{\frac{1}{2}} \qquad (6)$$

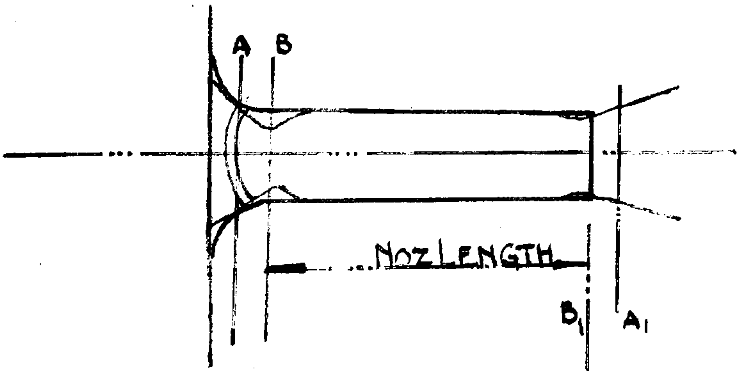
The reheating effect increasing the specific volume makes

$$C_d < C_v$$

These are the expressions derived from the expression for the jet function, their practical application will be illustrated later.

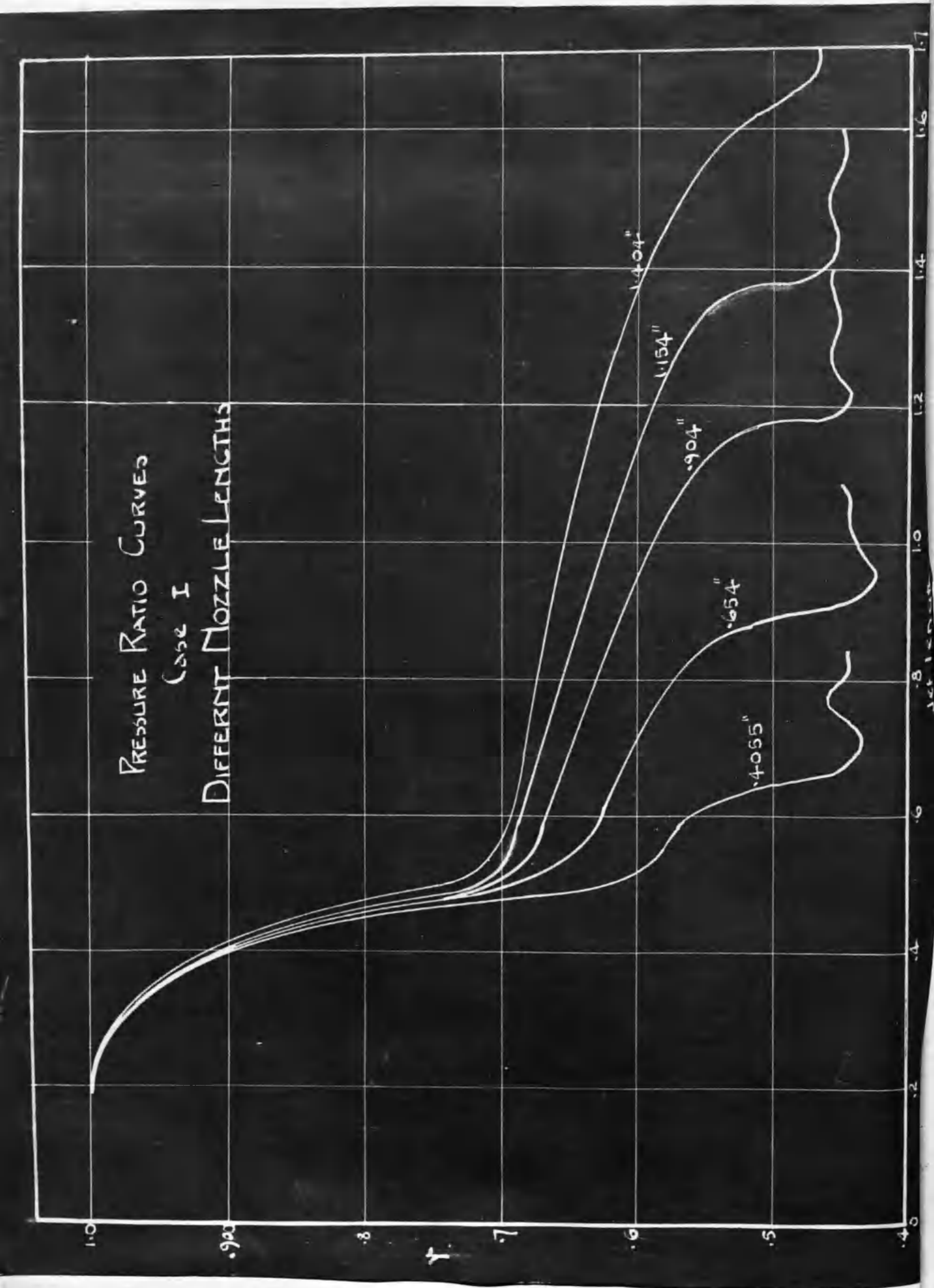
Table I gives the results of the tests and Figs. IV, V, VI, VII and VIII show the pressure ratio curves for the nozzle as it was cut down.

It will be noticed that the throat position of the nozzle lies on a steep part of the curve, this is probably due to the vena-contracta effect at entry, and owing to this effect a new throat had to be taken. This fact is better illustrated by the figure below.



The steam on leaving the nozzle follows the lines indicated at outlet as above so the outlet section had to be taken at a little distance before the outlet so as to be sure of parallel straight line motion.

FIG IX



PRESSURE RATIO CURVES
CASE I
DIFFERENT NOZZLE LENGTHS
1.404"
1.154"
.904"
.654"
.4055"

The pressure ratio curves are very smooth and well defined. The determination of F depends on A (the annular space between nozzle and search tube) and therefore it becomes an easy matter for the calculation of F from the relationship:

$$F = .718 \left(\frac{V_1}{P_1}\right)^{\frac{1}{2}} \frac{G}{A}$$

F being found the loss factor K could be easily determined from the calculating chart Fig X. The discharge coefficient is at once obtained by dividing the value of the jet function at any point on the flow curve by the theoretical value of the function corresponding to the same pressure ratio. The theore: :tical value of the jet function is obtained from the chart where $K = 0$

It was found advisable to plot all the pressure ratio curves of Case I for different lengths of the nozzle together Fig IX.

It is very interesting to notice that all the curves follow the same path while the pressure ratios at the throat drop with the shortening of the nozzle.

The next thing to be dealt with is the losses in the nozzle.

Figs. XI, XII and XIII show the loss curves, for Cases I, II and III respectively for different nozzle lengths. The curves are plotted on a base of nozzle length and indicate clearly the growth of loss as the expansion proceeds along the nozzle. The losses are rather slow at entry and they grow rapidly along the tail of the nozzle. This shows that the loss at entry is entirely different from that occurring in the tail, and also signifies the fact that there is a frictional effect dependent on the speed. The frictional loss will be dealt with at a later stage.

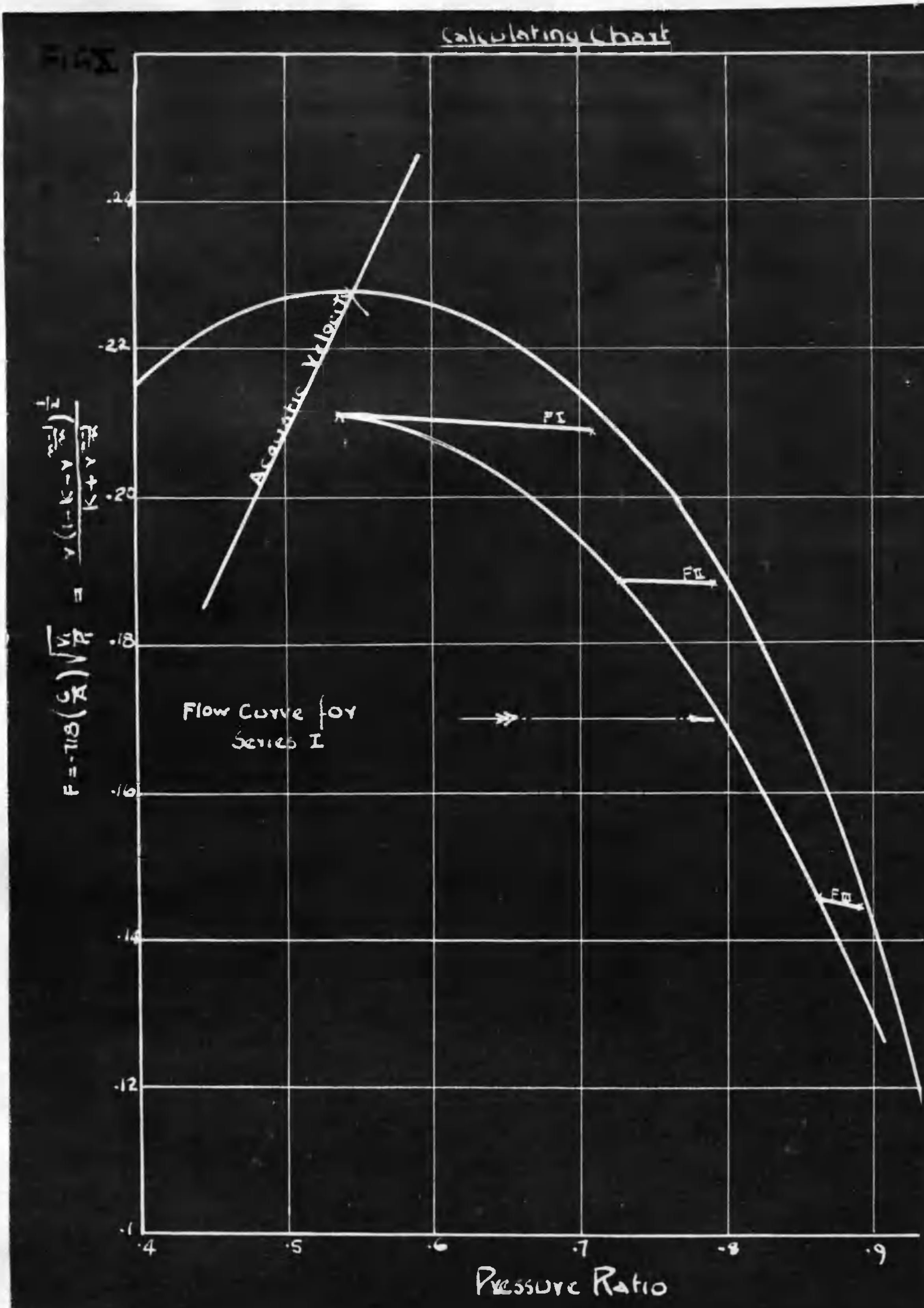
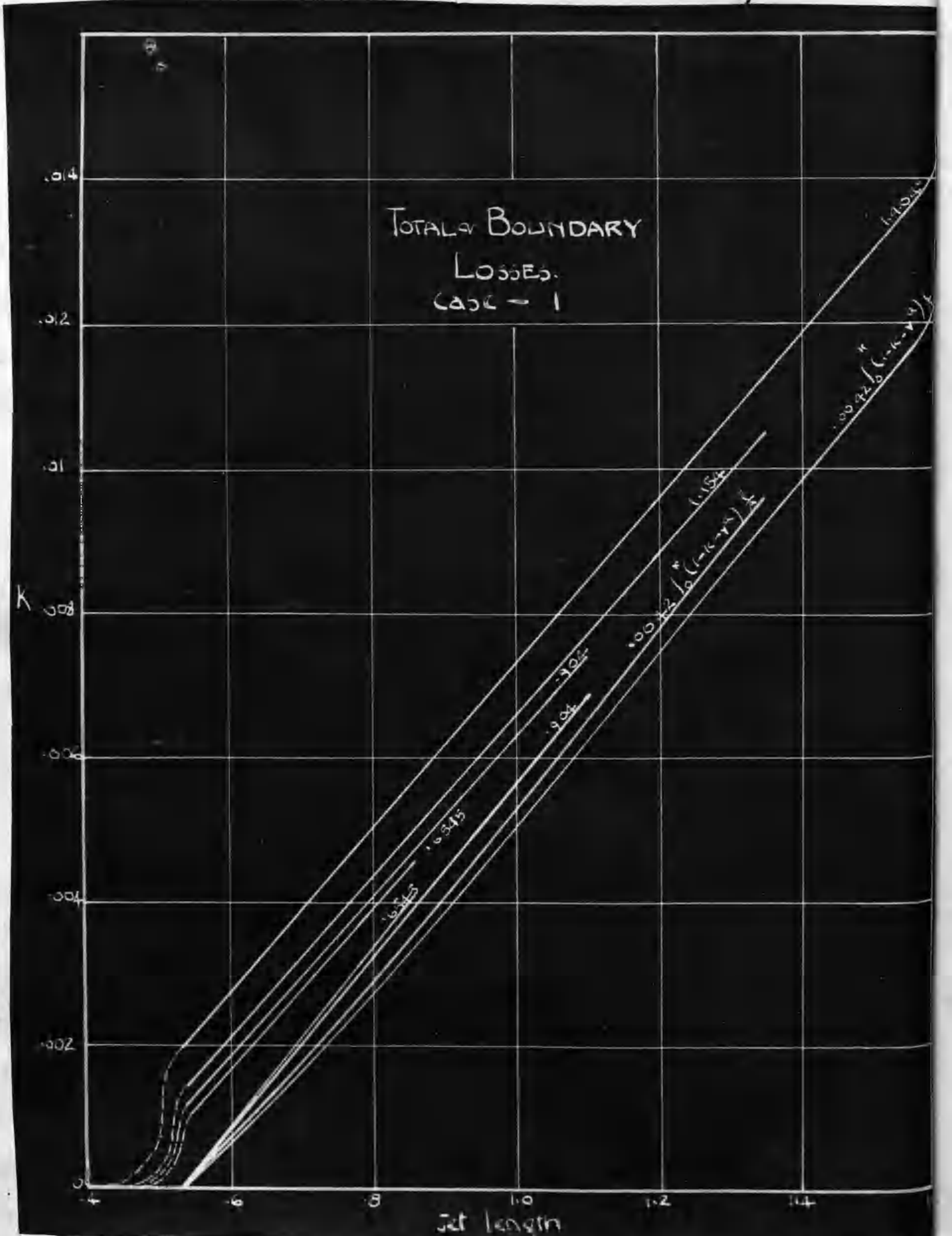


FIG. X
Calculating Chart
Acoustic Velocity
F I
F II
F III
Flow Curve for
Series I
·24
·22
·20
·18
·16
·14
·12
·1
·4
·5
·6
·7
·8
·9
Pressure Ratio



Total Boundary
Losses.
Case – 1
K
.014
.012
.01
.008
.006
.004
.002
0
.4
.6
.8
1.0
1.2
1.4
Jet length

FIG XI

Considering the loss curves for any of the three cases, at shortened lengths, one is at once struck at the diminution in loss at entry, the opposite effect would seem more reasonable, this will be dealt with at a later stage.

On the calculating chart is given a line acoustic velocity or velocity of sound in the medium at the critical con::ditions. This line marks the pressure ratios at which, with different losses, the limiting velocity of sound is reached and should definitely mark the end point of the expansion in the nozzle.

FRICTION LOSSES.

This loss can be calculated by direct methods, and the difference between this and the total losses gives the other losses.

The frictional resistance to the flow over a boundary surface depends entirely on the density, the speed of the fluid and the extent of the surface. It also depends on the nature of the surface, the influence will be on the constant co-effic::ient in the expression.

If dR = frictional resistance over length

ℓ = total perimeter (nozzle + search tube)

U = velocity of fluid.

V = specific volume of fluid

then $\frac{1}{V}$ = density of fluid

$$\therefore \quad dR = a \cdot \frac{1}{V} \cdot U^m \cdot \ell \cdot dx$$

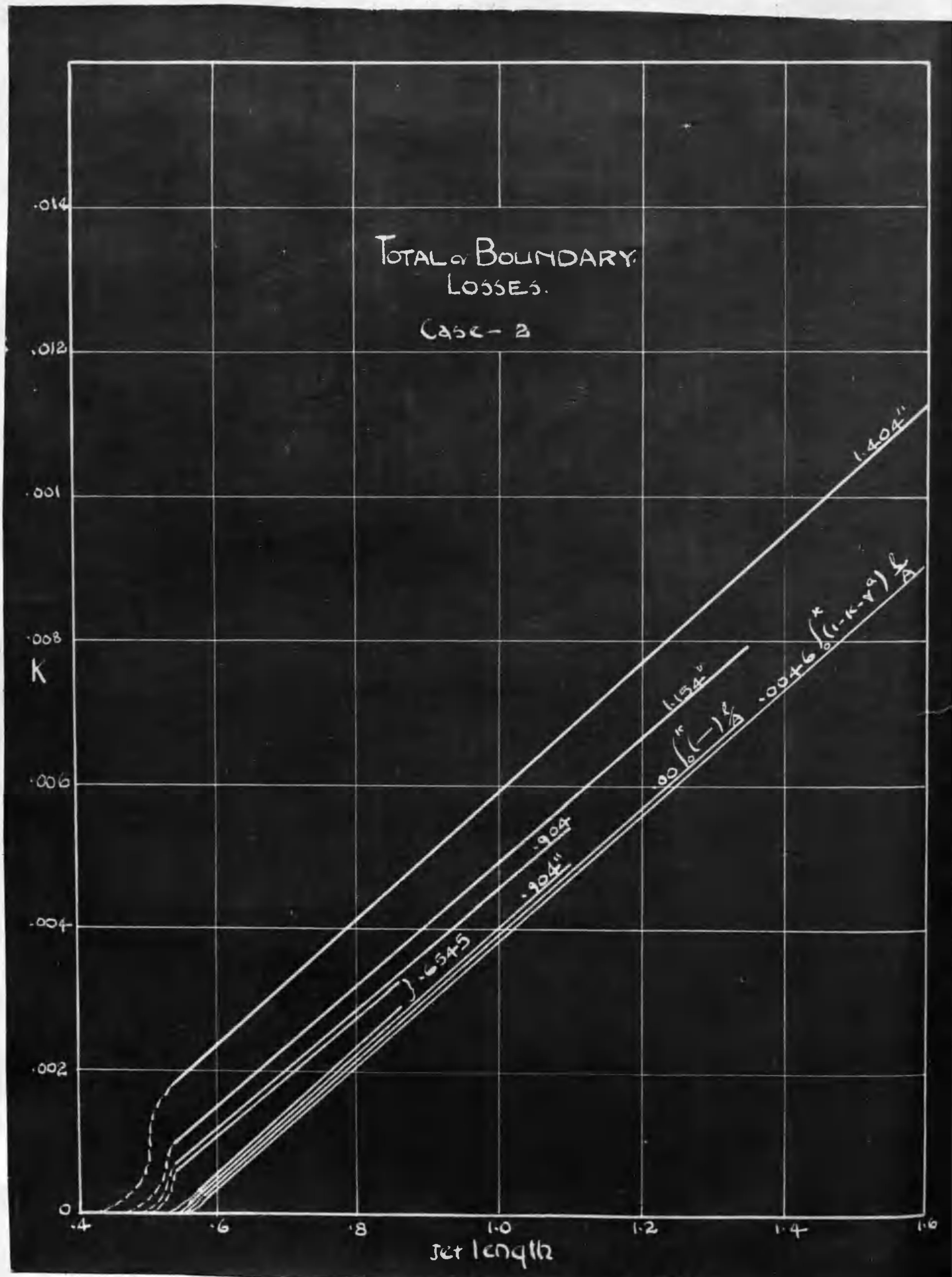
where m and a are constants

Energy loss per sec. is:-

$$dE = U\,dR$$

$$= a \cdot \frac{1}{V} \cdot U^{m+1} \cdot \ell \cdot dx \qquad (7)$$

FIG XII



TOTAL or BOUNDARY LOSSES.
Case - 2
.014
.012
.001
.008
K
.006
.004
.002
0
.4
.6
.8
1.0
1.2
1.4
1.6
Jet length
1.404"
1.154"
.904
.904"
.6545

The Equation

$$\frac{G}{A} = \frac{1}{144}\sqrt{\frac{288gn}{n-1} \cdot \frac{P_1 V_1}{V_1} \cdot \frac{1-r^a}{r^{-\frac{2}{n}}}}$$

could be rewritten thus:-

$$\frac{G}{A} = \frac{1}{144}\sqrt{\frac{288gn \cdot P_1V_1 \cdot r^2(1-K-r^a)}{n-1 \cdot V_1^2 \cdot (r^a+K)^2}} \qquad (8)$$

or:-

$$\frac{G}{A} = \text{Const.} \times \frac{\sqrt{P_1V_1 \cdot r\,(1-K-r^a)}}{\frac{V_1}{r}\,(r^a+K)}$$

From the Equation of continuity

$$\frac{G}{A} = \frac{\omega}{V} = \frac{\text{Const} \times \sqrt{E}}{V}$$

where "E" is the energy change in the ~~expression~~ expansion

$$\therefore \qquad P_1V_1(1-r^a)$$

is proportional to the energy Change with adiabatic expansion.

hence $\quad P_1V_1K$

is proportional to the energy loss on the expansion provided that: $\quad \frac{V_1K}{r}$

represents the increase in volume due to the reheating effect of the loss.

$\therefore$ Loss per lb. is given by:- $\quad b.P_1.V_1.K$

where b is constant. Hence the total loss/lb must be

$$dE = G.b.P_1.V_1\,dK.$$

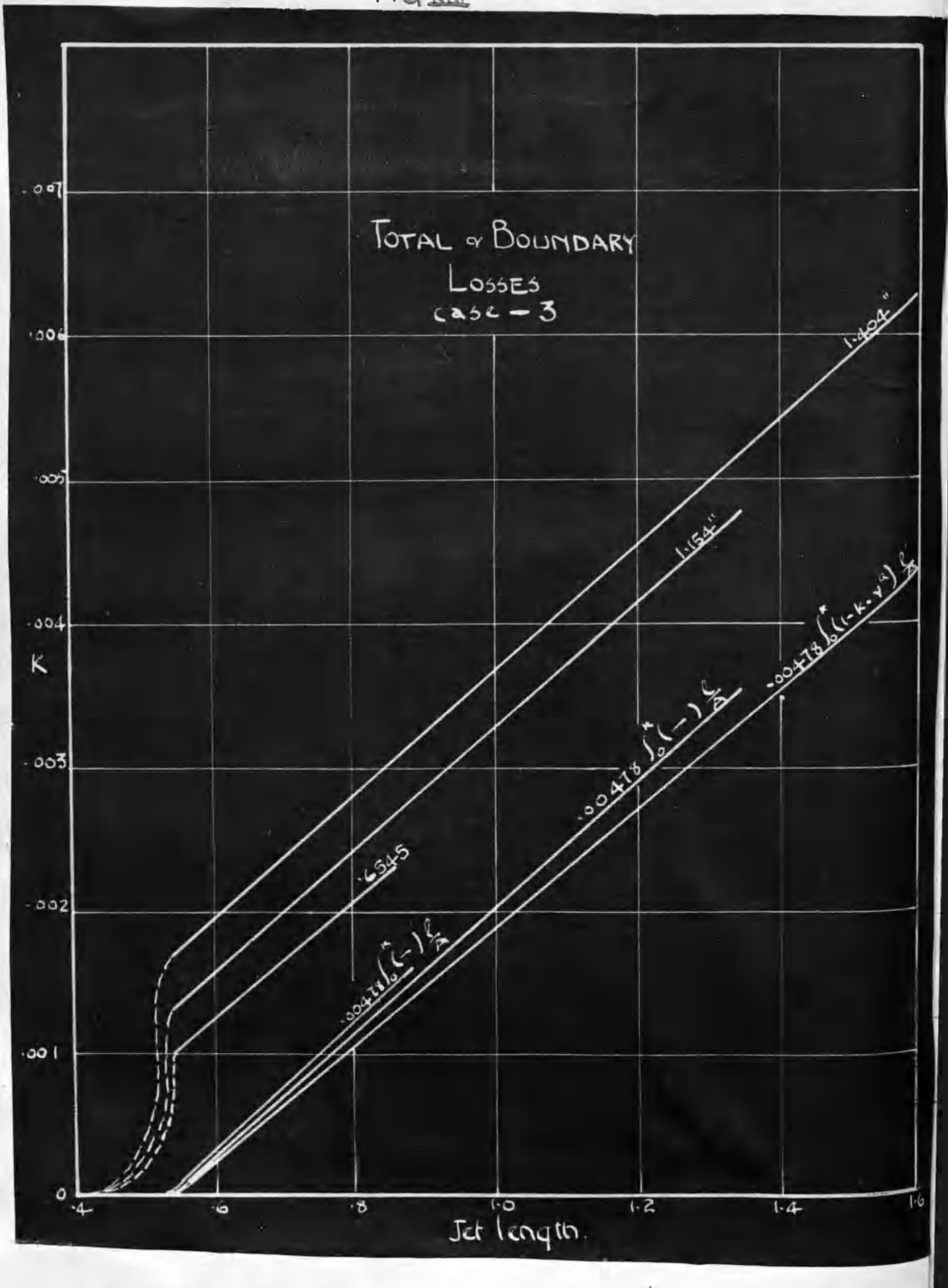
Equating this to Equation (7) gives:-

$$G.b.P_1.V_1\,dK = a.\frac{1}{V}.\omega^{m+1}.\,l\,dx.$$

Substituting the value of G from the continuity equation

$$A.b.P_1V_1\,dK = a.\omega^m.\,l.\,dx.$$

FIG XIII



TOTAL or BOUNDARY
LOSSES
case – 3
1.404"
1.154"
.6545
.00478
K
Jet length
0
.001
.002
.003
.004
.005
.006
.007
.4
.6
.8
1.0
1.2
1.4
1.6

and substituting in the continuity equation, the value of $\frac{G}{A}$ from equation (2)

$$U = (2g \cdot b \cdot P_1 V_1)^{\frac{1}{2}} (1 - K_t - r^a)^{\frac{1}{2}}$$

where K_t is the total loss at a pressure ratio r

Let K_f = Frictional loss. Hence

$$A \cdot b \cdot P_1 V_1 \, dK_f = a (2gbP_1V_1)^{\frac{m}{2}} (1 - K_t - r^a)^{\frac{m}{2}}$$

The value of "m" usually taken in practic is 2 and substituting this we should get.

$$dK_f = c(1 - K_t - r^a) \frac{l}{A} \, dx. \tag{9}$$

Where C is the frictional constant depending on the surface.

Integrating equation (9)

$$K_f = C \int_0^x (1 - K_t - r^a) \frac{l}{A} \, dx. \tag{10}$$

thus the frictional loss over an element of length δx, is proportional to the product of the energy liberated over that length.

A method for the evaluation of the constant therefore lends itself possible. All that is required is to measure the increment from x_1 to x_2 on the total loss curves and divide that by the integral of

$$(1 - K_t - r^a) \frac{l}{A} \, dx.$$

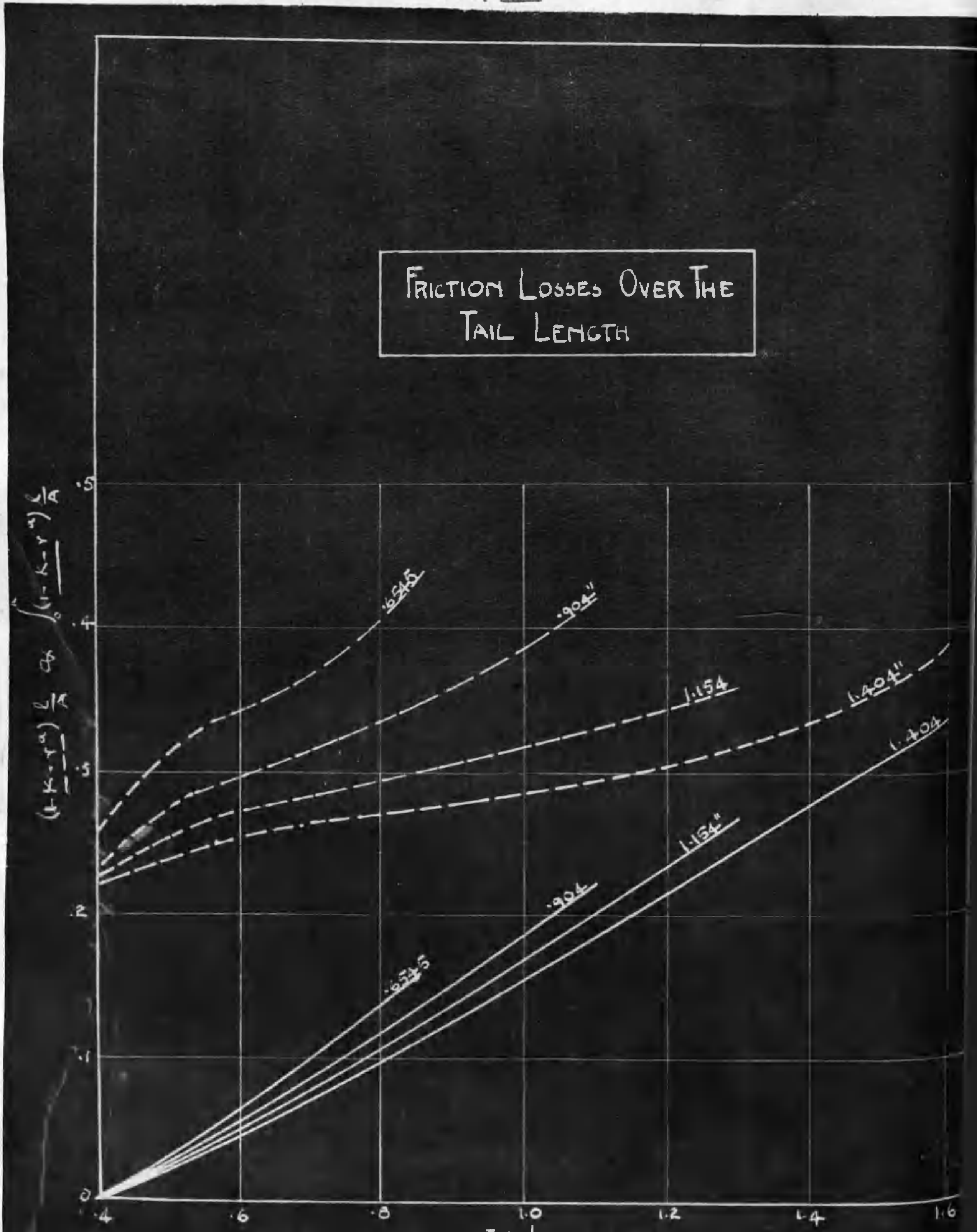
over the same length from x_1 to x_2

or

$$C = \frac{\left[K_s\right]_{x_1}^{x_2}}{\int_{x_1}^{x_2} (1 - K_t - r^a) \frac{l}{A} \, dx.} \tag{11}$$

The method for finding this constant for the nozzle is clearly illustrated in figure XIV for Case I. The integration was carried out by a graphical method.

FIG XIV



FRICTION LOSSES OVER THE TAIL LENGTH
Jet length
·4
·6
·8
1.0
1.2
1.4
1.6
0
·1
·2
·3
·4
·5
·6545
·904"
1.154
1.404"
·904
1.154"
1.404

The constants given below are those of Case I, II and III as found from the long nozzle before cutting it down.

Case	I	C	=	.0042
Case	II	C	=	.0046
Case	III	C	=	.00478

Mean Value for C = $\underline{\underline{.00452}}$

SEARCH TUBE EFFECT.

The introduction of the search tube into the nozzle causes a change in the friction.co-efficient which must be corrected.

Actually, the tube has its own co-efficient and it is the sum of the two co-efficients, (i.e. nozzle and tube) each multiplied by its corresponding surface area exposed to steam, which is equal to the coefficient as found, multiplied by the area to which it refers ~~to~~.

This may be expressed as :-

$$c_1 \cdot p_n \cdot l + c_t \cdot p_t \cdot l = c \cdot p \cdot l.$$

where c_1 = true coefficient of nozzle

c_t = true coefficient for search tube.

c = coefficient as calculated.

p_n = nozzle perimeter.

p_t = tube perimeter

p = perimeter of (nozzle + tube).

l = length of nozzle

This may be written as:-

$$c_1 p_n + c_t \cdot p_t = c p.$$

This equation has two unknowns in the quantities c_1 & c_t but c_1 & c_t could be made equal assuming that both the surfaces of search tube and nozzle are of same finish.

For the sake of easiness and also to avoid integra: :tion, the constant C could be written thus $\frac{c}{2g}$ & and the reciprocal of the hydraulic mean depth, and the velocity may be both assumed uniform, the operation of integration then becoming equivalent to multiplication by length and the expression becomes:-

$$K_f = \frac{C}{2g} . \omega^2 . \frac{p}{A} . l$$

where p = nozzle perimeter

and l = nozzle length

Total loss/sec. is then:-

$$\frac{C}{2g} . \omega^2 . \frac{p}{A} . l . G.$$

or loss/unit surface/sec

$$E = \frac{C}{2g} . \frac{\omega}{A} . G = \frac{C}{2g} . \frac{1}{V} . \omega^3 \qquad (12)$$

Now from D'Arcy's equation

$$\text{Friction head} = h = \frac{4fl}{2gd} \omega^2 \qquad (13)$$

which gives the loss in circular pipes of diameter "d" for a friction coefficient f .

Equation (13) may be rewritten

$$h = \frac{f}{2g} . \frac{p}{A} . l . \omega^2 \qquad (14)$$

which is exactly of the same form as equation (12) when changed to give the energy loss per unit surface per sec.:-

$$\text{i.e.} \quad E = \frac{f}{2g} \rho \omega^3 \qquad (15)$$

where ρ = den-sity of water

Comparing equations (12) and (15) it is obvious that C for the nozzle = f for water flow.

$\therefore$ loss of K.E over any section of the nozzle is

$$\frac{K}{(1-K-\gamma^a)} = C . \frac{p}{A} . l .$$

For design purposes it is much better to work with the true fractional loss which is:-

$$\mathcal{E} = \frac{K}{1-\gamma^a} = \frac{C . \frac{p}{A} . l}{1 + C . \frac{p}{A} . l} \qquad (16)$$

which reduces to:-

$$\xi_1 = \frac{c}{c + \frac{A}{p} \cdot l}$$

neglecting the very small loss, we could write

$$\xi = c \cdot \frac{p}{A} \cdot l$$

for the fractional loss in any part of the nozzle for which the friction factors have been assumed constant.

The values of c and f agree for similar surface conditions.

<u>BOUNDARY LOSSES</u>.

It could be easily deduced from equation (11) that:-

$$K_s = c \int_0^x \left(1 - K_t - r^{\frac{n-1}{n}}\right) \frac{l}{A}\, dx.$$

where K_s is the boundary loss, the values of c for Cases I, II and III for the long nozzle are given and a mean was also calculated, K_t and r could be easily determined from the figures of pressure-ratio curves, and total loss curves, hence K_s determined.

When it is considered that the loss factors represent the residue from the reduction data, and correct to about 1%, the agreement will be admitted to be remarkably close.

Considering the K_s curves Figs. XI, XII and XII it will be noticed that the graph of K_s is not quite a straight line in the parallel portion but has a slight concavity upwards. This is due to the increasing speed of the fluid along the tail. But by deducting this curve from the actual loss curve we are left with a loss occurring wholly in the convergent portion. This latter loss is the loss that causes the diminution of efficiency with decreasing pressure range. Prof. A. L. Mellanby and Dr. Kerr have proved that the viscosity has very little effect on the loss occurring in the convergence. The loss must then be attributed to the eddy currents in the main flow. This eddy effect will vary with different convergences.

PRESSURE RATIO CURVES.

Dealing with the full range expansion curve, Case I Series I, the pressure ratio at the outlet is approximately .535 and .708 at the throat. The theoretical value of the pressure ratio is .546 for superheated steam. With perfect expansion and no losses, the energy liberated by expansion to a pressure ratio of .546 is sufficient to generate in the fluid, a speed equal to the speed of sound in the fluid at the critical pressure conditions.

Since, however, the expansion is imperfect and losses take place, some of the energy available will be absorbed by these losses, and the fluid will not attain the speed of sound at a pressure ratio of .546. It will also be noticed that there is a big drop in the pressure ratio between the throat and outlet which shows that there is a big drop of pressure along the tail.

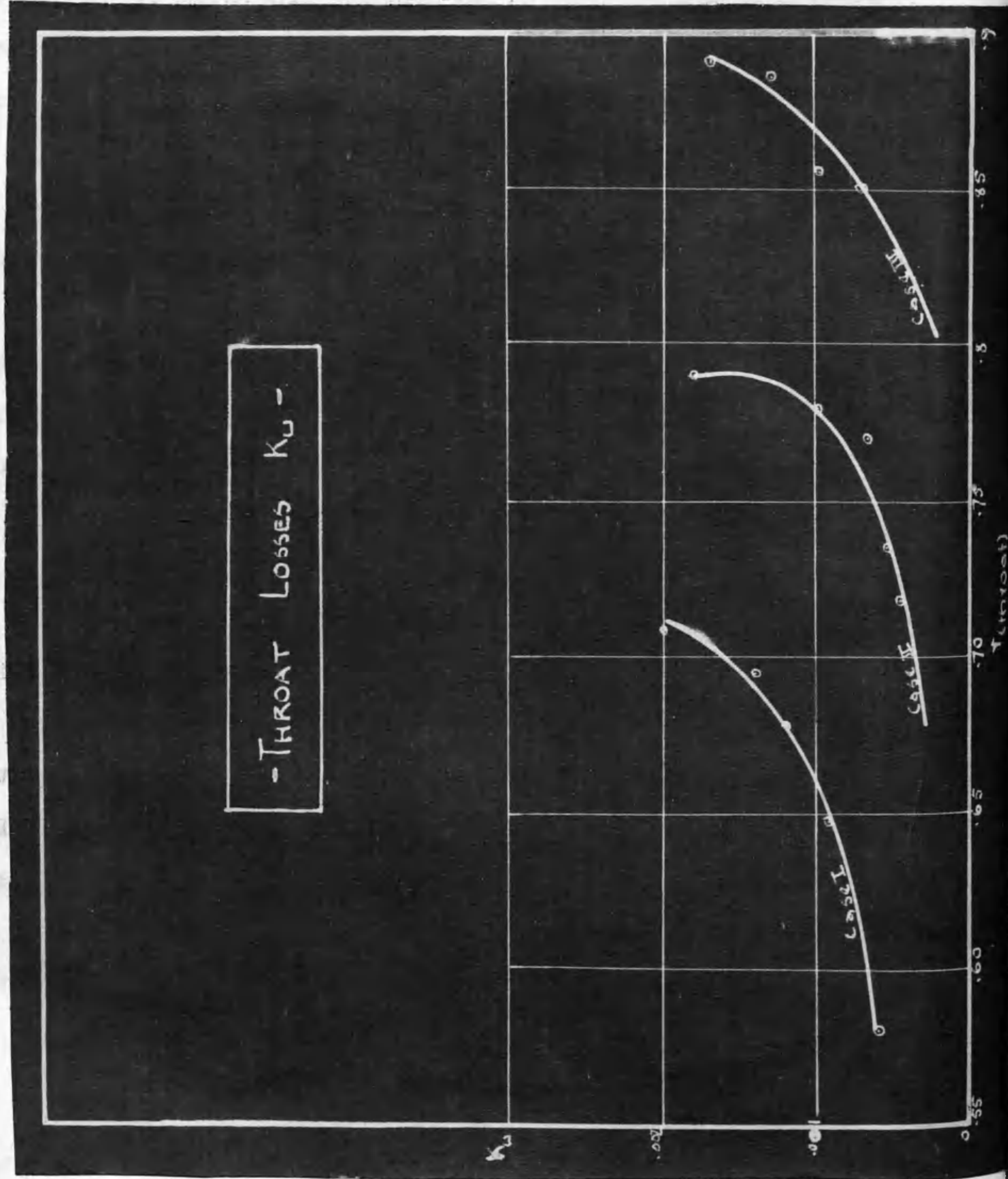
It will be of interest to note the drop of the press:ure-ratio at the throat, and the rise of pressure ratio at the outlet as the nozzle is shortened down as shown in Fig. IX. Of course, this is not only applicable to Case I, but by com:paring the pressure-ratios of Case II and Case III for the different shortened lengths, the same effect is noticeable. This effect is brought about by the fact the losses are smaller, and also the friction losses are less. This latter fact makes the flow quantity in lbs./sec. greater, for the same steam inlet conditions this is shown clearly in Table I.

LOSS CURVES.

The total loss curves for any of the three cases, and for shortened lengths of the nozzle are definitely parallel to one another. The shorter the nozzle the smaller the losses.

In Series III, IV, and V, it will be seen that if the actual values of F be plotted on the calculating chart on

FIG XV


-THROAT LOSSES Kᵤ -
Case I
Case II
Case III
.55
.60
.65
.70
.75
.8
.85
.9
.001
.002

a pressure ratio base that the points will be thrown out of the theoretical loss value $K = 0$ This is due to the steam not filling the nozzle and the throat area which was taken in the first two series becomes ineligible for these last series owing to the contraction of area at throat. The co-efficient of surface, being known for the three cases of the full length nozzles, was used for calculating the boundary losses occurring at the tail, and the total loss derived. This method also gives us the contraction of area at the throat as it will be the difference of the F calculated as above and the F given by experiment.

Fig. XV shows the variation of losses at the throat plotted to a base of pressure, this is done by the method of cross interpolation from the pressure ratio curves and the loss curves.

The curves are well defined and are of the same shape for the three cases with different lengths. This shows dis: :tinctly that the losses at the throat decrease with the pressure-ratios.

This is not what was expected, because if we consider one nozzle with the different back pressure and the same steam inlet conditions, we would at once see that the curves are of opposite characteristics, i.e. the losses increase with the pressure ratios at throat.

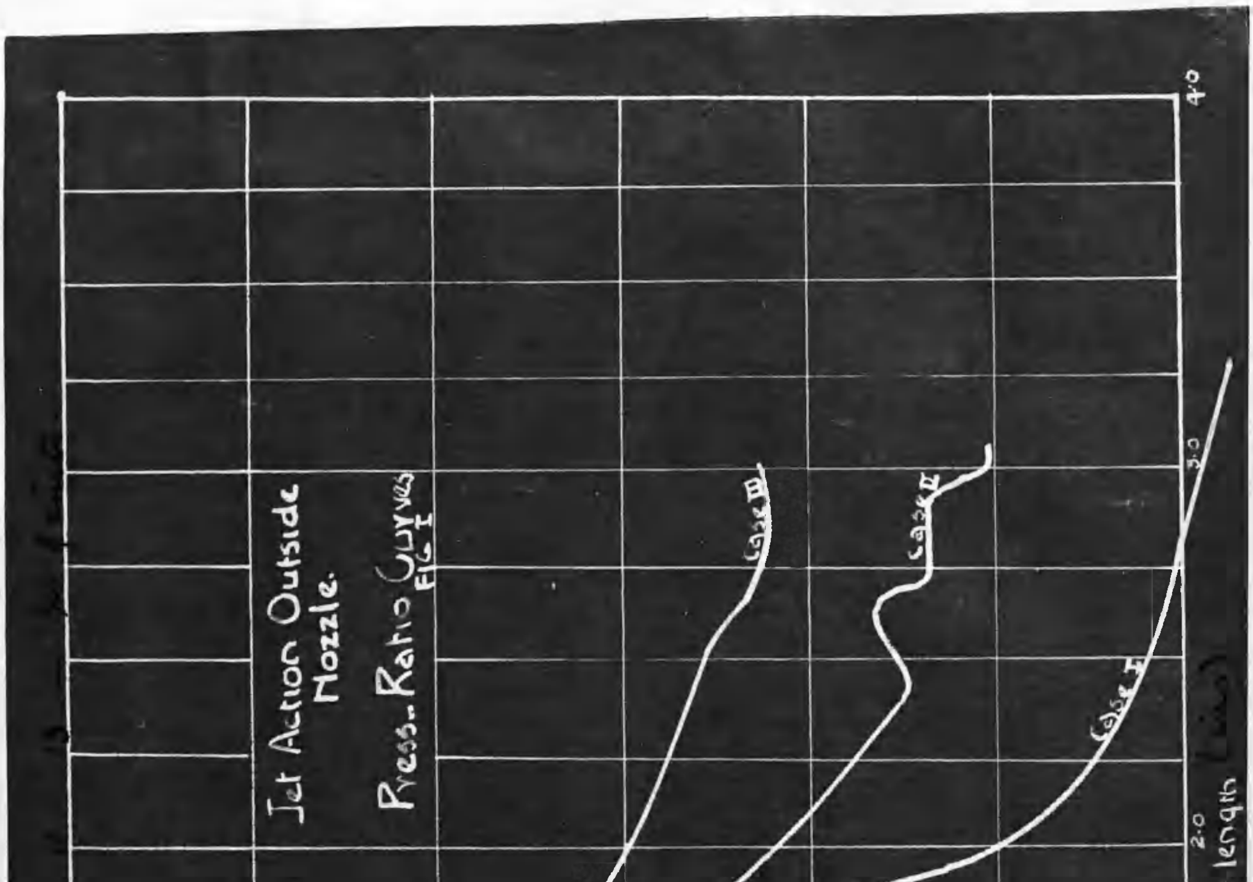
This is easily explained by the facts that the search tube is no longer fit to investigate the losses in the throat, owing to some small losses which amount up in the calculations, and that on dealing with the full lengths of the nozzle the flow area at the throat was quite stable and as the lengths were shortened down, the flow area was more and more uncertain until at shortest length the steam was not filling the nozzle at all, this effect could be very easily noticed if the values for the shortest length would be spotted on the calculating chart Fig. X, they would fall outside the $K = 0$ line. This shows clearly and definitely that other methods than the search tube should be dealt with for further work on throat losses.

S. T. diameter = .1317" throat = .0367

Area = .0135 sq. in. outlet = .0364

TABLE I.

TEST	TOTAL	PARALLEL	FLOW	STEAM			F	F	THROAT	OUTLET
No.	LENGTH OF NOZ.	LENGTH OF NOZ.	LBS/SEC.	PRESS ABS.	TEMP °F	VOL V_1	(THROAT)	(OUTLET)	DIA.	DIA.
(1)	1.404"	1.144"	.03611 .03255 .02504	76.0 76.0 76.0	408.5 412.5 407.7	6.6168 6.6494 6.6062	.2086 .1881 .1449	.2105 .1890 .1460	.2532 " "	.2525 " "
(2)	1.154"	0.894"	.03715 .0336 .0261	76.4 76.4 76.4	404.7 407.7 403.7	6.5740 6.6062 6.5514	.2130 .1930 .1502	.2141 .1948 .1515	.2532 " "	.2525 " "
(3)	0.904"	0.644"	.0380 .03412 .0266	75.95 75.95 75.89	400.7 400.7 418.2	6.5807 6.5934 6.7134	.2190 .1966 .1548	.2208 .1980 .1561	.2532 " "	.2525 " "
(4)	0.6545"	0.3945"	.0392 .0354 .0283	75.9 75.9 75.9	392.5 392.0 403.0	6.5005 6.495 6.583	.2240 .2020 .1630	.2270 .2045 .1650	.2532 " "	.251 .251 .251
(5)	0.4055"	0.1455"	.0402 .0362 .0293	75.9 75.9 75.9	390.8 390.0 384.0	6.484 6.481 6.429	.230 .207 .1680	.2308 .2079 .1685	.2532 " "	.253 .2530 .2530



In case I, where the expansion ratio through the nozzle is least, a wave formation is noticed at the nozzle outlet, this is soon damped out however and the expansion is quite regular beyond.

JET ACTION OUTSIDE THE NOZZLE REGION.

This short work was carried out on the nozzle appara: :tus described at the beginning of this paper, the object being to investigate the jet condition after leaving the nozzle.

A convergent parallel nozzle, 1" long and .25" dia. at the throat was used.

The search tube was cut at right angles to its own axis and impact readings of the pressures as the search tube was moved from the nozzle outlet taken. During these experi: :ments it was noticed that when the search tube was pushed into the nozzle the pressure ratio was the same at any length.

The inlet pressure was kept constant at 60 lbs./sq. in. G. and the back pressures were kept constant at 20, 30 and 40 lbs./sq. in. G., while the temperature was kept at 410°F. thus ensuring a high degree of superheat so as to avoid any complications due to supersaturation. The results of these tests are given in figure in a graphical form.

The flow quantities for the three tests were:-

(60/20) = .390 lbs/sec.
(60/30) = .353 lbs/sec.
(60/40) = .270 lbs/sec.

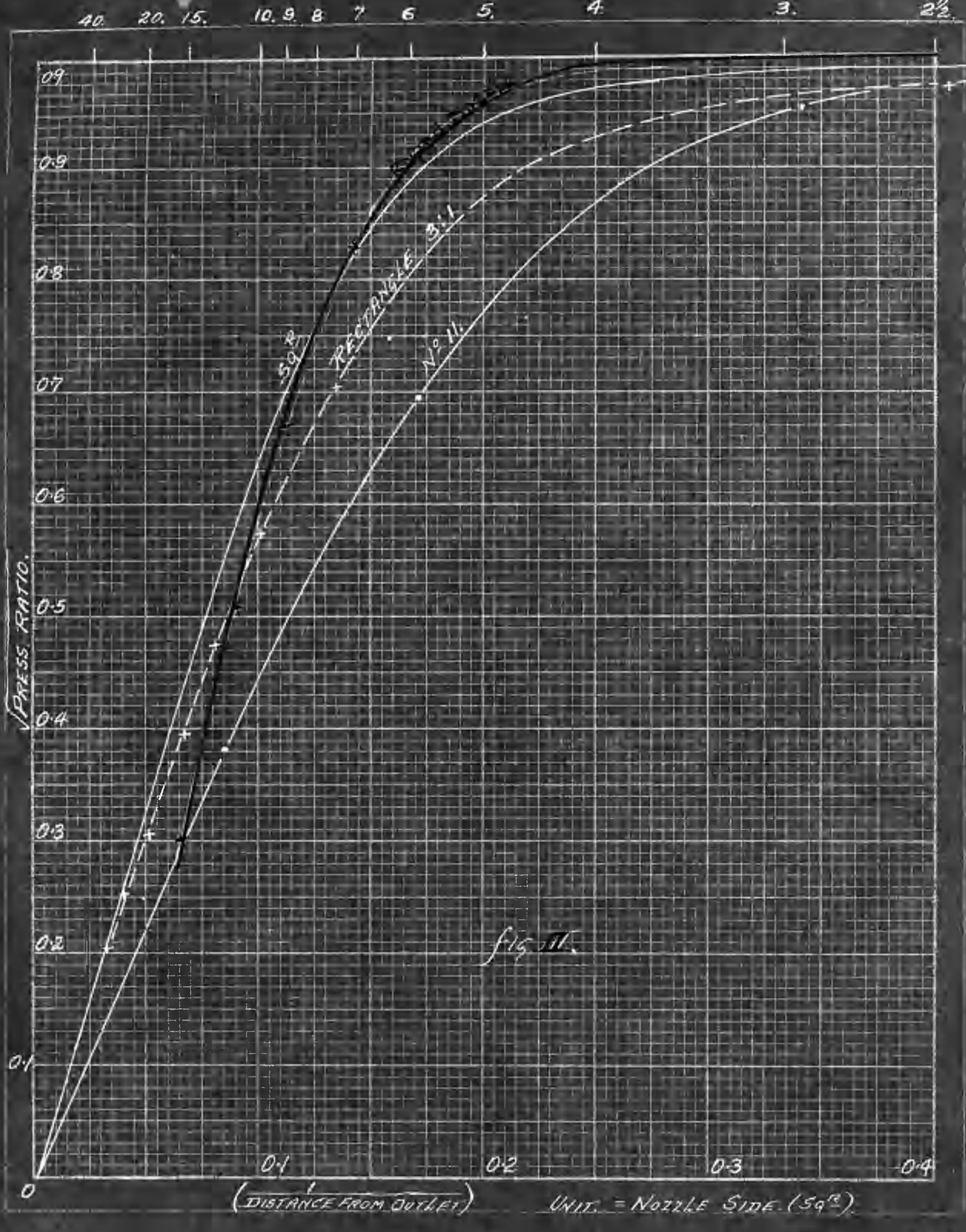
PRESSURE RATIO CURVES.

Fig I_a shows the pressure-ratio curves plotted against the jet length.

~~In all the three cases it will be seen that the jet~~ ~~constant pressure for~~ ~~follows a stream line motion for a length of nearly .4", after which the flow becomes turbulent and loses a great deal of its energy. In case (1) it will be noticed that soon after the outlet the steam was in a somewhat turbulent state, but a mean line could be drawn horizontally to represent these points.~~

Table I gives the results of the tests taking as a unit, the diameter of the nozzle.

The results are plotted on Fig. II on a base of the



40. 20. 15. 10. 9. 8 7 6 5. 4. 3. 2½.
09
0.9
0.8
0.7
0.6
0.5
0.4
0.3
0.2
0.1
0
√PRESS RATIO.
Sq. R.
RECTANGLE 3:1
No. 11.
Fig. II.
0.1 0.2 0.3 0.4
(DISTANCE FROM OUTLET)
UNIT. = NOZZLE SIDE. (Sq.)

A.12.24.

T A B L E I.

DISTANCE (Unit = $\frac{1}{4}$)		3	5	7	9	11	13	15	α
		.333	.20	.143	.111	.091	.077	.066	
55 lbs. abs.	γ (from curve)	1.00	.98	.895	.83	.80		.76	.735
	(γ - .735)	.265	.245	.16	.095	.065		.025	
	do. ÷ .265	1.0	.925	.605	.36	.245		.095	
	$\sqrt{\ }$		.96	.78	.60	.49		.31	
45 lbs. abs.	γ (from curve)	1.00	.98	.885	.78	.715		.63	.60
	(γ - .60)	.40	.38	.285	.18	.115		.03	
	do. ÷ .4	1.00	.95	.713	.45	.29		.075	
	$\sqrt{\ }$		.975	.845	.67	.54		.27	
35 lbs. abs.	γ (from curve)	1.00	.95	.86	.745	.60		.52	.47
	(γ - .47)	.53	.48	.39	.275	.13		.05	
	do. ÷ .53	1.00	.905	.735	.52	.245		.094	~~Exceeds with press. ratio~~
	$\sqrt{\ }$		.95	.86	.72	.495		.31	
	Mean		.96	.83	.67	.51		.30	

reciprocals of the jet lengths.

The reason why the square root scale of the pressure ratio was chosen, was because this approximately varies as the jet speed.

The graphs of a No. 11 turbine nozzle, a rectangular nozzle 3:1, and a square nozzle, were kindly given by Mr. J. Brown, M.B.E., due acknowledgement being here made. They were taken from his investigations on low pressure air in nozzles.

Plotting these on the same ordinates as the round nozzles, shows similar characteristics. The distinguishing feature is that the square root value becomes zero before in: :finite distance and is doubtless due to confined space in the exhaust side of the apparatus. The results are close for the following deductions.

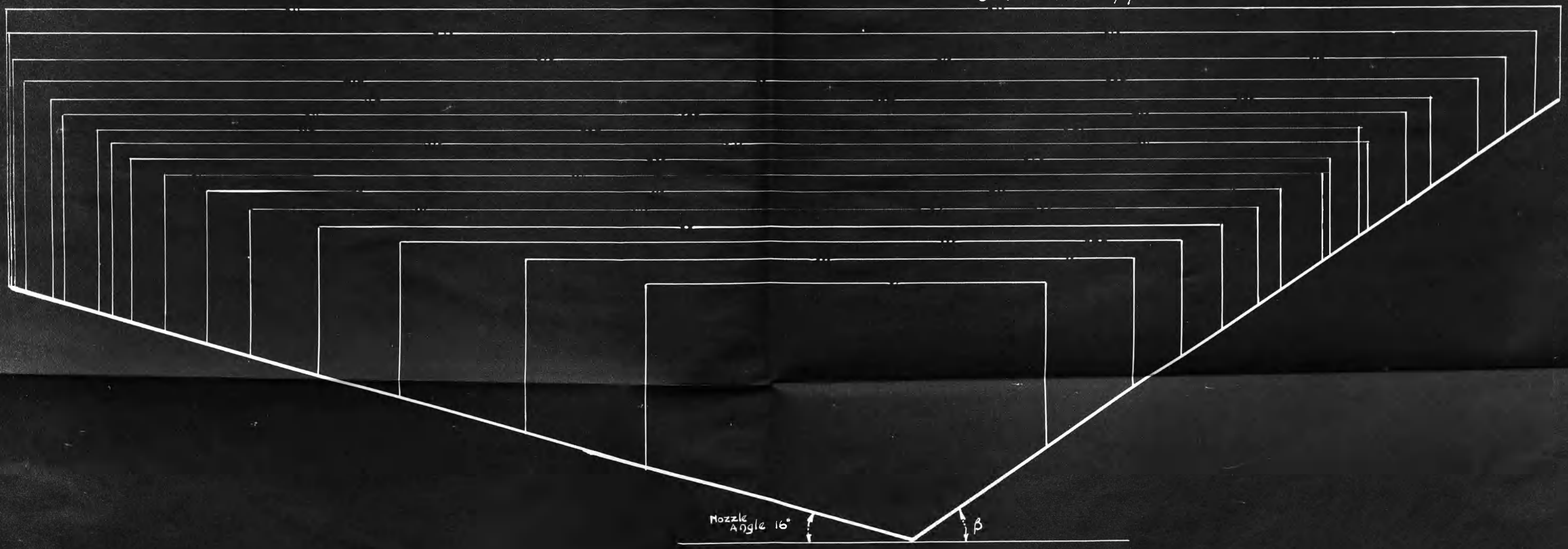
(1) Air and steam show identical characteristics.

(2) Over wide range of steam pressure there is no evidence of any secondary factors.

In conclusion, I beg to thank Professor A.L.Mellanby, D.Sc., for the interest he has taken, and the facilities granted for the progress of the work; I also beg to thank Associate-Professor Wm. Kerr, Ph.D., A.R.T.C. for the suggestion of this line of research and for the kindly help and advice he has so freely given.

Stationary Blade Tests
Blade Width 1"
Scale 1" = 200 ft/sec.

FIG. 17



INVESTIGATION ON TURBINE BLADE LOSSES.

INTRODUCTORY.

In spite of the considerable amount of research that has been done on the subject of blade losses, there is still very little known about these losses.

In every case the experimental work has been carried out with a view to determining the values of the blade velocity coefficients under different conditions.

A suitable value of this coefficient for any given case has to be obtained by analysing the performance of the machine or by the use of some empirical formula, but opinions differ as to the most suitable value of the blade coefficient for a given blading arrangement.

Rateau found from experimenting with a series of blades having angles varying from 25 to 34 that the velocity coefficient varied from .7 to .82.

Briling[1] gives the following equation as deduced from his experimental work.,

$$\psi = \psi_0 - .00432\,\theta^{\frac{4}{3}}$$

where ψ_0 = .9 to .97 when the bucket edges are rounded and from .95 to .99 when they are sharp.

θ = the total curvature of the blade as measured in degrees.

and ψ = blade velocity coefficient.

In these experiments Briling varied the blade pitch and the velocities ranged from 250 to 1,300 ft/sec, but the value

(1) "Engineering" - 25th. April, 1910.

of the pitch giving the least loss was independent of the steam speed, though the blade efficiency rose as this speed increased.

Stodola's[2] work is about the most recent work of any interest on the subject, where he gives the following equation for the blade velocity coefficient,

$$\phi = \left(\frac{P}{m\omega_1} - \cos\alpha_1\right)\frac{1}{\cos\beta_2}$$

where P = inlet pressure

m = mass flow per unit time

ω_1 = the speed of the flow

α = nozzle angle

β_2 = geometrical outlet angle of the blade.

He finds however that if air escaped in whirls through the opening then ϕ does not measure the real diminution of speed.

All the above experimenters used some special kind of apparatus consisting of groups of impulse nozzles and stationary groups of blading to represent the rotor blading in a turbine.

The question of assuming an overall blade velocity coefficient is rather unreasonable because on examining some of the velocity coefficients they were found to be so low as to suggest that the single coefficient is not a true representative of the actual occurrence, and that the blade velocity coefficient as used at present is not a true blade velocity coefficient in that it covers losses other than those occurring actually in the blade passage.

Such considerations as these led to the following experimental investigation of the blade losses at the Heat Engine Laboratories of the Royal Technical College of Glasgow.

For this purpose a special apparatus was designed

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(2) "Dampf und Gas Turbinen" pp. (145 to 160) By Prof. Stodola.

and arranged by the author, which will be referred to, hereinafter as "The Stationary Blade Apparatus", and tests were carried out by the author with different blade curvatures, and for making the investigation complete, some of the results of Mr. John G. E. Ellis, B.Sc., A.R.T.C., on "Stationary Torque Turbine Tests" were taken into account. Also some of the power tests on the Turbine were taken. These being carried out by the Post Graduate course of the Royal Technical College of whom the writer was a member in the year 1923.

Both the "Stationary Torque Turbine Tests" and the "Turbine Power Tests" were carried out on the Turbine at the Heat Engine Laboratories of the Royal Technical College, Glasgow.

These tests made possible the investigation of the blade losses both in their stationary and moving conditions.

RANGE OF EXPERIMENTAL WORK.

(a) Stationary Blade Apparatus Tests. Fig (1) gives a detailed sectional view of the "Stationary Blade Apparatus". It consists of a blade carrying lever fixed at one end by two pins that allow free motion in either the upward or downward directions, and supporting a set of stationary blades at the other end. This lever is enveloped by a cast iron casing to represent the turbine casing.

The blade carrier is so designed as to allow the blades to be moved in any direction without shifting the horizontal or zero position of the lever.

The force on these blades is measured by a spring balance outside the casing, the lever being kept always in its zero position by means of a thin steel wire attached to it and passing through a gland on top of the casing, and round a smooth running pulley on ball bearings to minimise friction which is attached along with the balance to a fixed frame. A balance weight is attached to this end of the steel wire to keep the lever in its zero position, before steam is allowed through. A nozzle box is fitted at 16^{o} to the vertical axis of the casting, thus fixing the nozzle angle. This nozzle box could slide backwards and forwards perpendicular to the nozzle angle, so as to enable the adjustment of the gap between the nozzle outlet and the blade inlet.

A Vane Indicator is fitted to the casing. This instrument consists of two aluminium vanes, one being suspended opposite the nozzle, but on the far away side of the blades. Each vane consists of a thin plate attached to a thin steel rod, which passes through the casing and at right angles to it, and is in such a position that the vane lies in the actual line of flow of the steam issuing from the blades. To the other end of the rod is attached a needle which moves over the face of a graduated brass plate in degrees, so that in whatever position the vane takes up, the actual angle can be easily read off, thereby giving the actual angle that the steam leaves the blades.

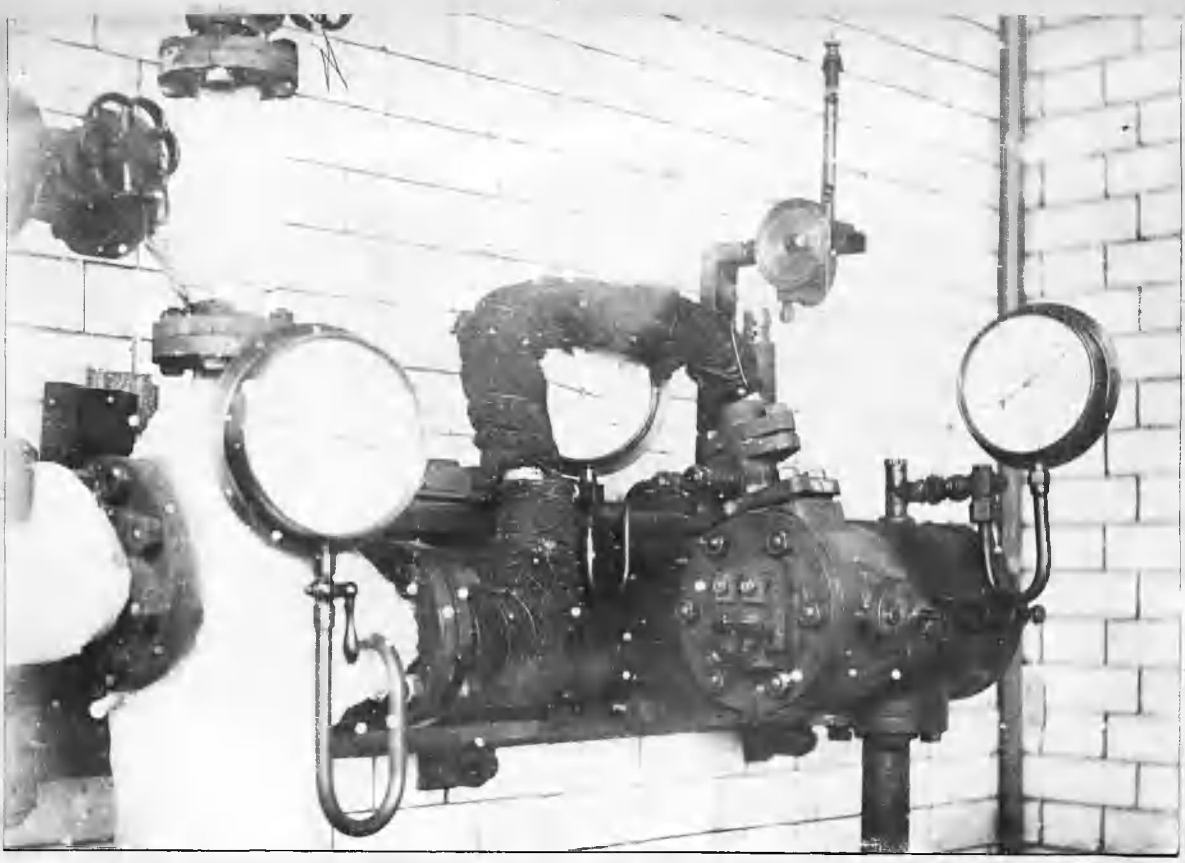
When steam is allowed through, the lever is brought back to its zero position by means of a small hand wheel attached to the spring balance and also to the fixed frame that supports the spring balance.

All balance readings are taken with the lever fixed in this zero position.

In addition, provision is made for measuring the pressure and temperature at the stop valve, nozzle box, and also in the vicinity of the exhaust pipe.

The amount of steam consumed is measured by collecting in a tank and weighing the contents at periods of

Fig II



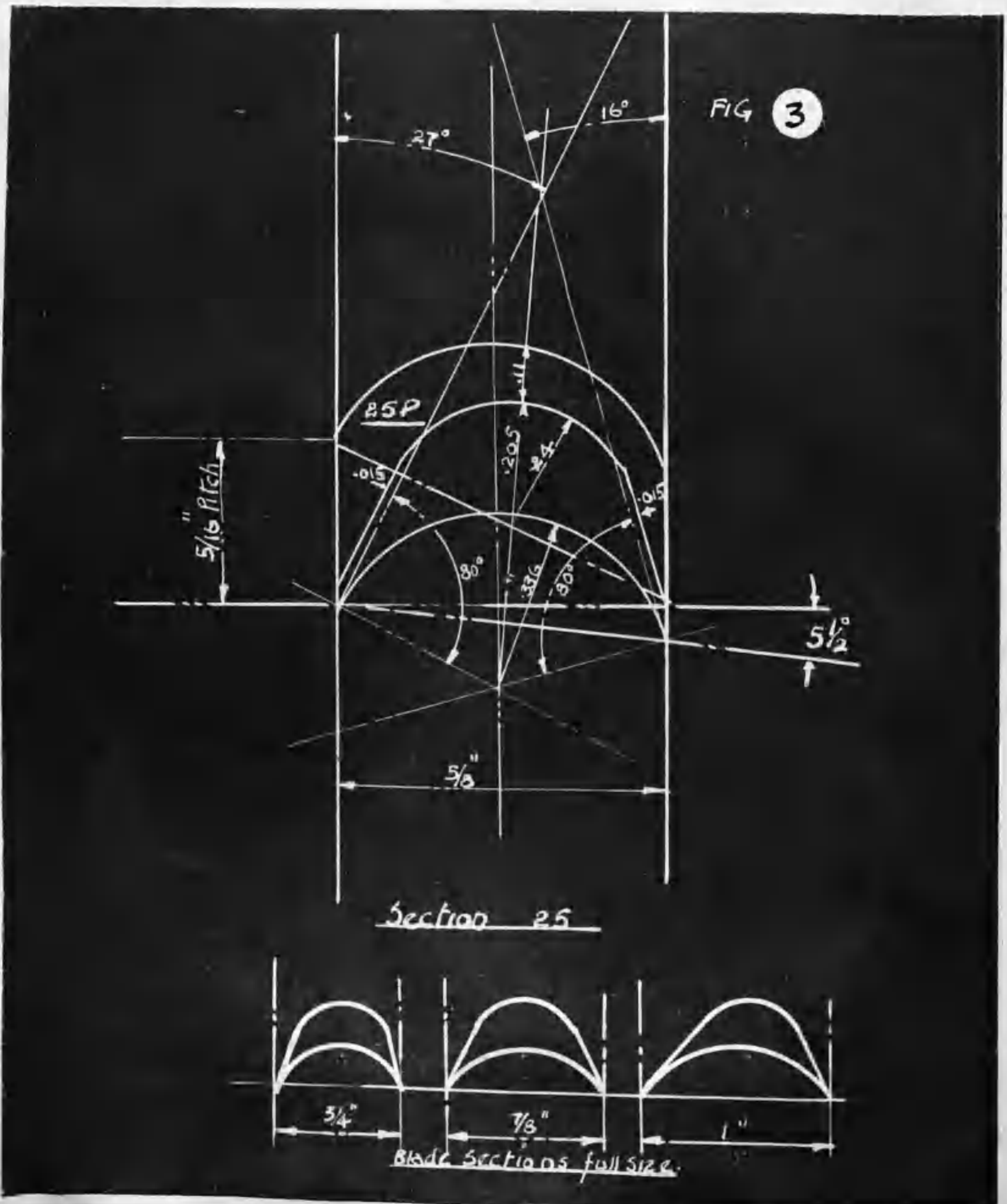


FIG 3
16°
27°
25P
.11
.205
.24
.015
.015
5/16" Pitch
80°
.336
80°
5½°
5/8"
Section 25
3/4"
7/8"
1"
Blade Sections full size.

two minutes, thus giving the consumption in lbs per second.

A Photograph of the apparatus under working conditions is shown in fig. (2).

Four sets of blades of different sizes were tested in this apparatus. Each set being mounted on a special blade carrier that could be easily fixed on the lever as shown by the drawing, every set containing seven blades and shrouded on top in the ordinary way. The blades were cut down to $1\frac{1}{4}$" in height.

The position of the blading relative to the nozzle was first fixed and the horizontal position of the lever was marked on the pulley, this being the zero mark.

The gap between the nozzle and blades was kept at 1/16" through the whole testing of the four blade sets.

Superheated steam only, was used so as to avoid any complications arising from supersaturated steam calculations.

The tests were carried out by varying the inlet pressure and keeping the exhaust pressure a constant at atmospheric, thus providing a series of pressure ratios, balance readings, and condensate for each set of blades under test.

A sectional view of the blades used is given in figure (3).

(b) Stationary Torque Turbine Tests. These experiments and also the Power experiments were carried out on "The Turbine".

The turbine was built by Parson's & Co. Ltd., and had a ratio output of 250 K. W. when running a speed of 3,000 R.P.M.

Originally this turbine was fitted with three blade wheels and each of the three wheels carried a set of blades having stationary blades between each wheel.

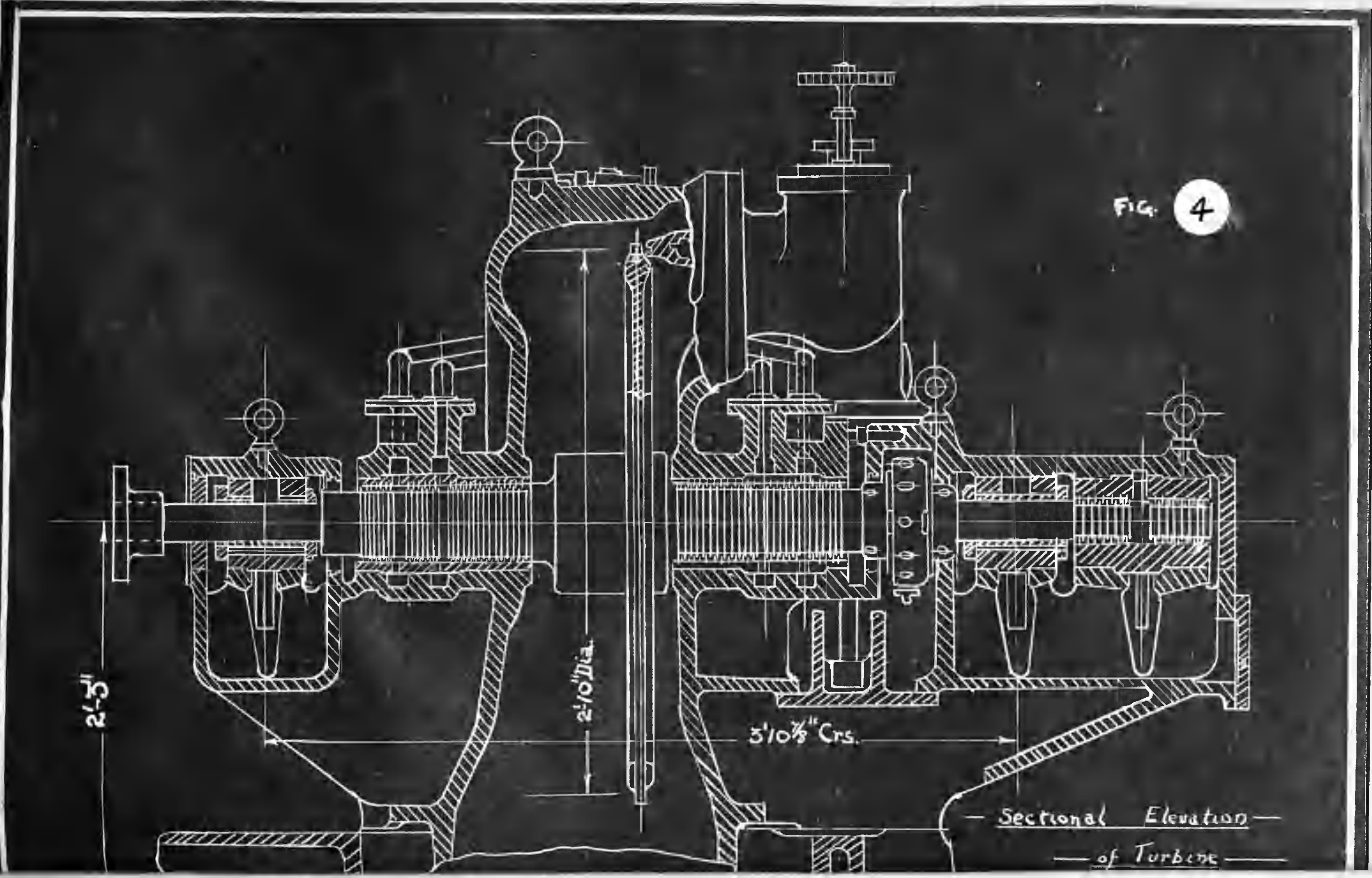


FIG. 4

Sectional Elevation of Turbine

Twelve nozzles were employed for supplying steam and these were arranged circumferentially, not equally spaced around the rotor, but grouped together in an arc at the top of the casing.

These nozzles were arranged in four groups of three nozzles each. Admission to the first of these groups was obtained by merely opening the main stop valve after which one or all the others could have steam by means of the three separate hand wheels controlling the admission valves to each of the latter three groups.

In order to facilitate experimental work the turbine had been considerably altered.

Fig. (4) gives a sectional elevation of the turbine and indicates fully the constructional arrangement and the engine details. The rotor now carries only one blade wheel, the other two having been removed. Consequently, the number of nozzles had to be diminished and so the whole arc of nozzles was removed, and for these only two were replaced, one being a convergent divergent and the other a convergent parallel, and both being of rectangular cross section. Two of the valves which previously admitted steam to the previous nozzle groups, are now utilized to supply steam to the two nozzles. On opening the main stop valve, steam is admitted to the valve chest and by one of the valves or the other, steam is allowed to pass through the corresponding nozzles. The bearings are $2\frac{1}{2}$" diameter by $6\frac{1}{2}$" long and are of the Parson's elastic sleeve type. A thrust block absorbs axial thrusts which owing to the blade angle may be of no mean order when large quantities of steam are passing. The lubrication of bearings and thrust block is obtained by oil rings. A "Run-Away" governor is connected with a valve controlling the passage between the main stop valve and the nozzle chest. This governor closes the valve when the turbine speed is about

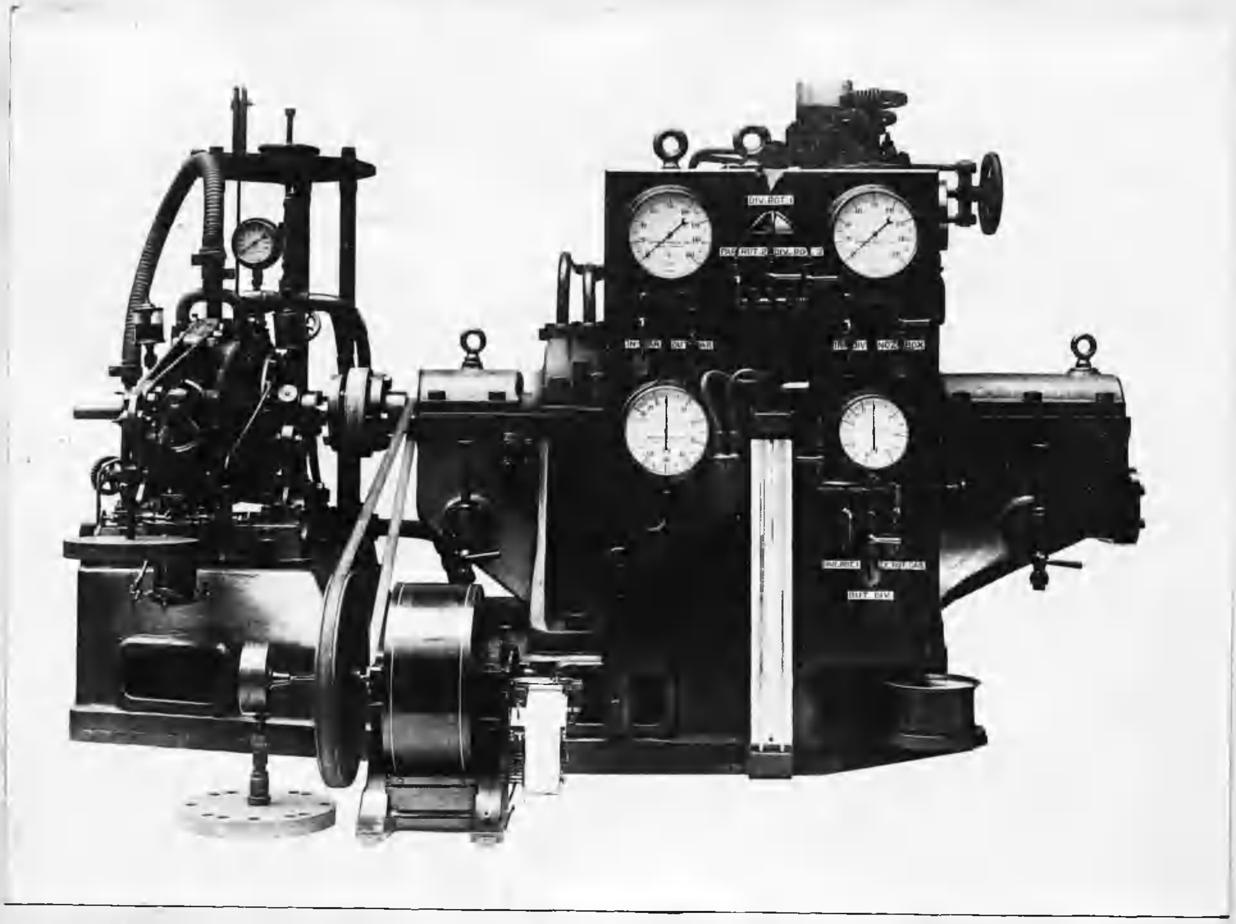


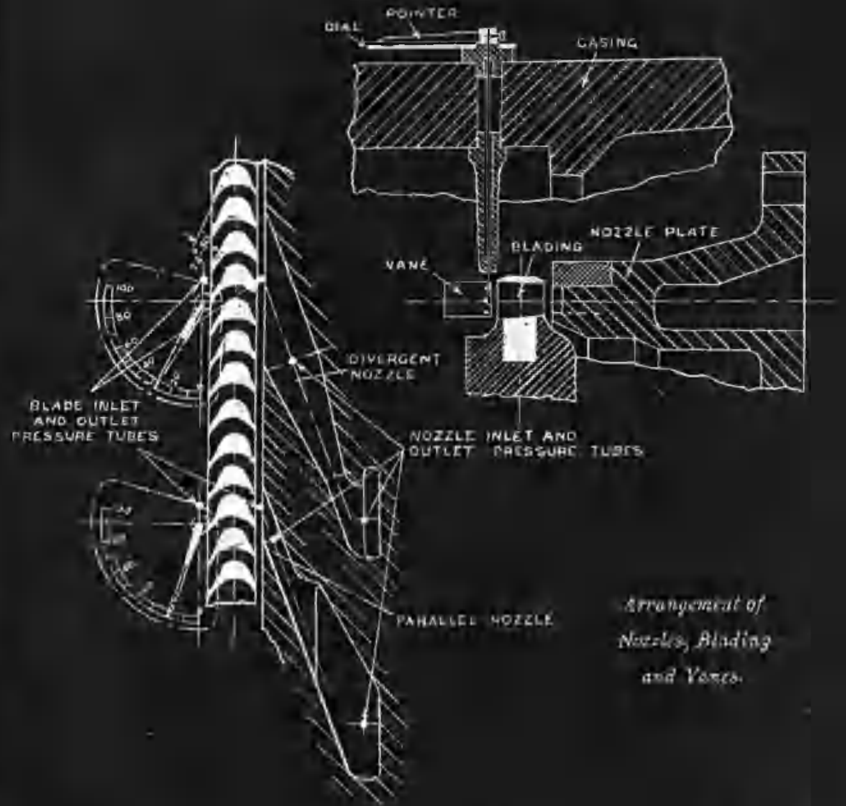
Fig (5)

15% in excess to its normal speed of 3,000 R.P.M. A vane angle indicator is fitted to the turbine. For the pressure measurements small tubes are led to various points in each nozzle, viz. nozzle inlet, nozzle outlet, rotor inlet and rotor outlet, these tubes being in communication with gauges on the instrument board. See fig (5).

For the measurement of the torque with the rotor stationary, a special dynamometer was used. This consisted of a torque bar having at one end a flange which was bolted to the rotor shaft coupling, the bolt holes in the flange being extended circumferentially to permit of adjustment of the relative angular positions of the bar and rotor. Near its outer end two spring balances are attached to the bar and to a fixed frame respectively so that one balance exercises an up:
:ward pull and the other a downward pull on the torque bar. A mark at the outer end of the bar was put so as to set up the bar in the horizontal position, this being obtained by manipulating the balance adjusting hand wheels, so as to bring the zero mark on the bar into line with a small fixed pointer on the balance frame. A large fixed pointer attached to the rotor shaft and an accompanying scale fixed to the turbine casing, was used for indicating the relative position of the rotor and nozzles for the different series of tests.

In these tests a great difficulty as to the determination of the actual position of the blading relative to the nozzles was experienced. This difficulty was totally overcome in the writer's apparatus in determining the actual position of the blading relative to the nozzles. In these "Stationary Torque Turbine Tests", the rotor position was arbitrarily chosen and marked on the scale, provided for the purpose, so that this position was the same throughout the first series, so long as the torque bar was set to the zero mark. Two rotor positions were taken in these tests.

—Turbine Power Tests.—



Arrangement of Nozzles, Blading and Vanes.

Nozzles.

	Parallel.	Divergent.
Width at Throat.	0·319"	0·319"
" " Outlet.	0·319"	0·890"
Height.	0·463"	0·463"
Length from Throat to Outlet.	0·920"	3·400"
Nozzle Angle.	20°	20°

Blading.

Height at Inlet.	0·524"
" " Outlet.	0·614"
Angle of Inlet.	27°
" " Outlet.	22°-50'
Mean Pitch.	0·539"
Thickness at Outlet.	0·015"
Width of Blade.	0·787"
" " Gap.	0·170"

Heeman and Froude Water Brake. — Radius to Load — $33\frac{3}{4}$".

Turbine Wheel — Mean Diameter. — 34".

Moment of Inertia of Wheel — 8·26 ft lbs (wt) units.

Weight of Rotor {1st Wheel Only} — 575 lbs.

Journals — $2\frac{1}{2}$" Dia. 6" Long.

— Convergent-Parallel Nozzle Efficiencies. —

r	·525	·550	·600	·650	·700	·750	·800	·850
η	·930	·926	·916	·906	·894	·880	·862	·845

Fig (6)

(c) Turbine Power Tests. A full description of the machine has already been given under the "Stationary Torque Turbine Tests". Fig (6) shows a sectional view of the vane angle indicator fitted to the turbine and also gives the details of the nozzles and blading.

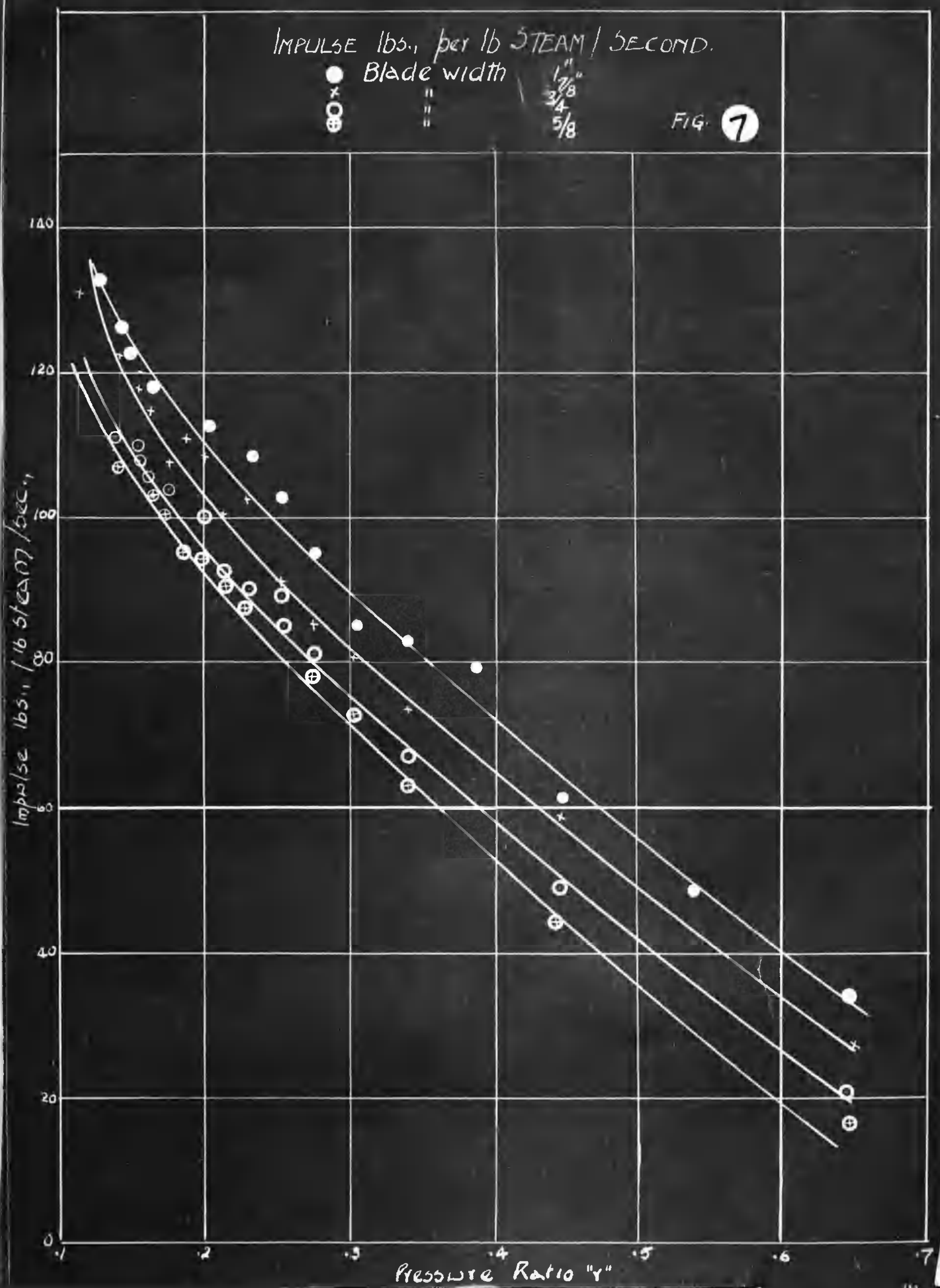
These tests were made on the turbine, the steam supply pressure ranging from 20 to 120 lbs/sq.ins., gauge and the speed from 500 to 3,200 R.P.M. The required variation in jet speed was obtained by varying both the stop valve pressure and the vacuum, thus giving the same jet speed for different steam pressures, that is to say for different mass flows.

Each test was of thirty minutes duration, the speed and torque readings being taken every five minutes and the other readings at ten minutes intervals.

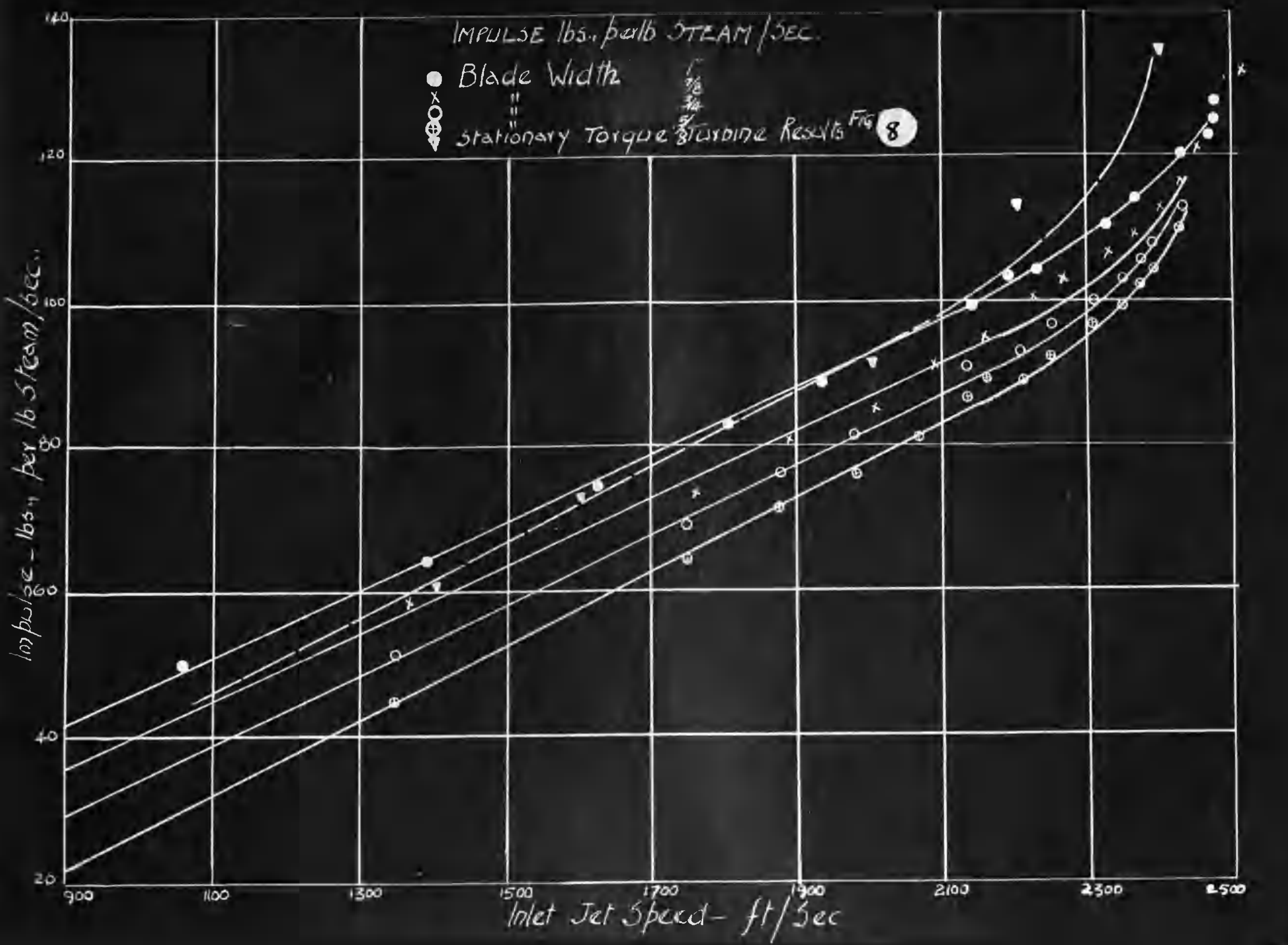
The turbine was allowed ample time in which to reach a steady temperature conditions and the machine was accordingly run slowly for at least two hours before commencing any tests.

Particular care was taken to maintain steady conditions during each test, and in this respect the speed regulation was the most troublesome factor, the dynamometer being designed for a load many times greater than any dealt with in these tests. This made it somewhat difficult to maintain a constant speed and to obtain accurate brake readings. In spite of this, however, the tests for the greater part are very satisfactory, the principal weakness being that it was not possible to cover the same speed range for each of the pressures adopted owing to the very low torque necessary to give a higher rotor speed with low steam pressure.

The steam flow was measured during these tests by collecting the condensate in a calibrated tank, the total steam



IMPULSE lbs., per lb STEAM / SECOND.
Blade width 1"
" 7/8
" 3/4
" 5/8
FIG. 7
140
120
100
80
60
40
20
0
Impulse lbs., / lb steam / sec.,
·1
·2
·3
·4
·5
·6
·7
Pressure Ratio "r"



IMPULSE lbs. per lb STEAM/SEC.
Blade Width 1"
" 7/8
" 3/4
" 5/8
Stationary Torque Turbine Results
FIG 8
Impulse - lbs. per lb Steam/sec.
Inlet Jet Speed - ft/Sec
140
120
100
80
60
40
20
900
1100
1300
1500
1700
1900
2100
2300
2500

passed during each test being measured. Detailed consideration of the flow quantities will be found in the appendix.

All the tests in this paper were made with a convergent-divergent nozzle. The flow quantities measured during the Power & Stationary tests (b) and (c) are not used as they stand in any of the subsequent calculations[x].

TEST RESULTS.

(a) Stationary Blade Apparatus. "Jet Impulses". The results for these tests are exhibited in tables $(1)_a$, $(1)_b$, $(1)_c$, and $(1)_d$. The jet speed depends very nearly on the pressure ratio alone, that is, it is independent of the supply pressure of the steam, but the mass flow through the nozzle, is directly proportional to the nozzle inlet pressure.

The steam load on the blades is expressed as "Jet Impulse" in lbs. per lb. of steam per second. This gives an excellent check on the results since the tests cover a large range both of steam pressures and of pressure ratios irrespective of the steam supply pressure.

In all these tests the vane angles measured were exactly equal to the geometrical outlet angle of the blade, the passage of steam being also clearly marked on the blade carrier and indicated the geometrical outlet angle of the blade, which angle was taken for the subsequent analysis of the results.

The Impulse on the blades is expressed as already explained in lbs per lb of steam per second, and are shown plotted against the pressure ratio "τ" in fig (7) and against the jet speed in fig (8). The dotted line in fig (8) will

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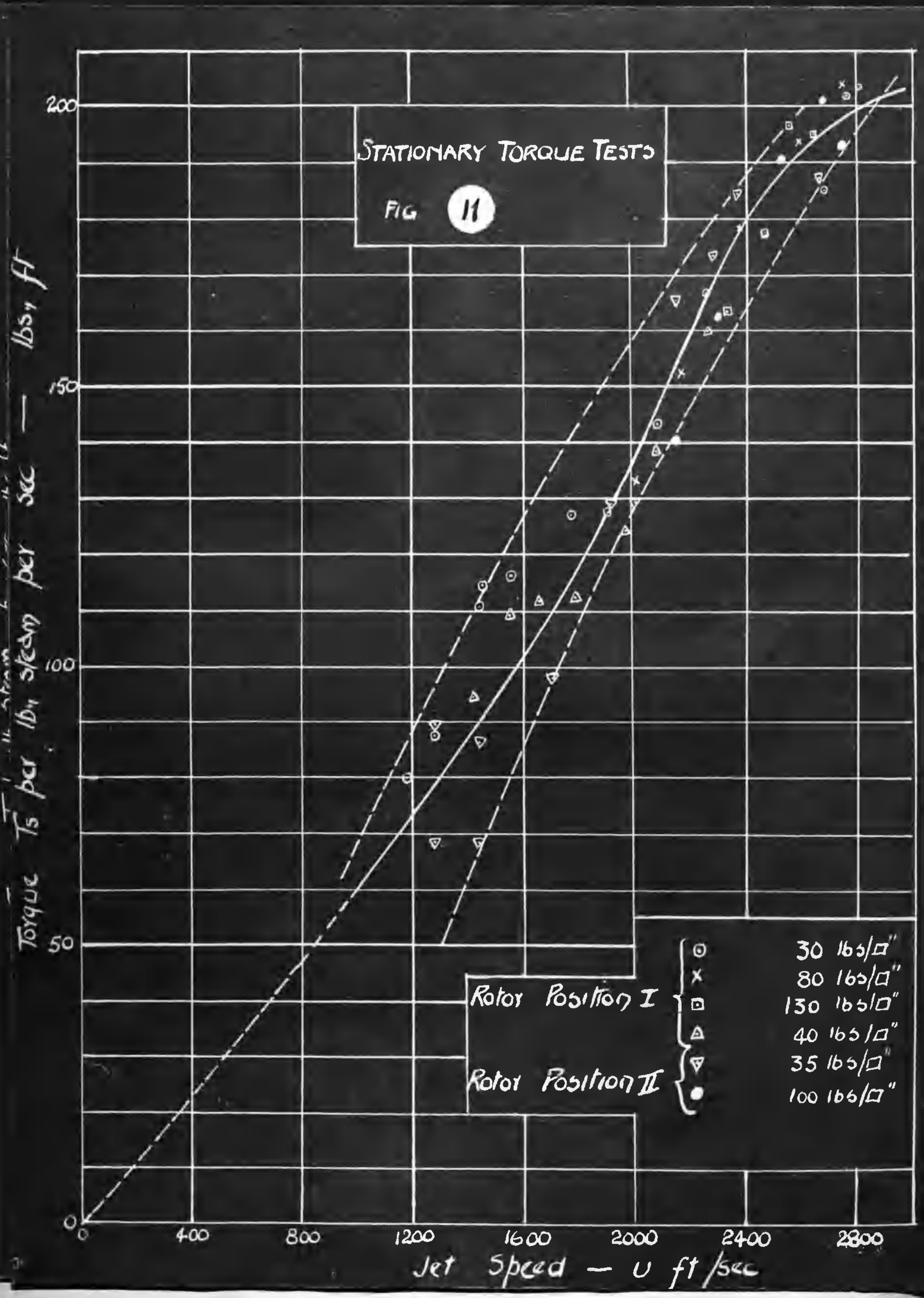
[x]See appendix.



STATIONARY TORQUE TESTS
FIG 11
Torque T_s per lb. steam per sec — lbs. ft
Jet Speed — u ft/sec
0
50
100
150
200
400
800
1200
1600
2000
2400
2800
Rotor Position I
30 lbs/□"
80 lbs/□"
130 lbs/□"
Rotor Position II
40 lbs/□"
35 lbs/□"
100 lbs/□"

be referred to later, under the "Stationary Torque Turbine Tests". The blades were so fixed in each series of tests so as to assure that all the blade passages within the jet are running full and so giving the best value of jet impulse.

The normal pressure ratio of the convergent-divergent nozzle used is about .15 and for the higher pressure ratios the jet undergoes recompression within the nozzle a condition which considerably reduces the nozzle efficiency. No doubt then, that the big drop of Jet Impulse values is due to the poor quality of jet.

The curves show a pronounced change of direction where the nozzle is working at its best, and this indicates not only is the efficiency of the nozzle a maximum for this pressure ratio, but also that the jet is of particularly good quality at this point, in that the further losses to which it is subjected beyond the nozzle outlet are less than for any other pressure ratio.

(b) Stationary Torque Turbine Tests (Jet Impulses). The results for these "Stationary Torque Turbine Tests" are shown in figs. (9) and (10). The torque values and jet speeds are taken from Mr. Ellis's curves which are reproduced in figs. (9) and (10).

No doubt that part of uncertainty which exists as to the torques at the higher pressure ratios is due to some experimental error, the dynamometer being insufficiently sensitive for the measurement of the very small balance loads, obtained in that region.

A mean curve through all the points available was drawn for the two positions of the rotor, fig. (11)

It would certainly have been desirable to investigate the definite region of the higher pressure ratios more fully and also for a variety of rotor positions. The torque values

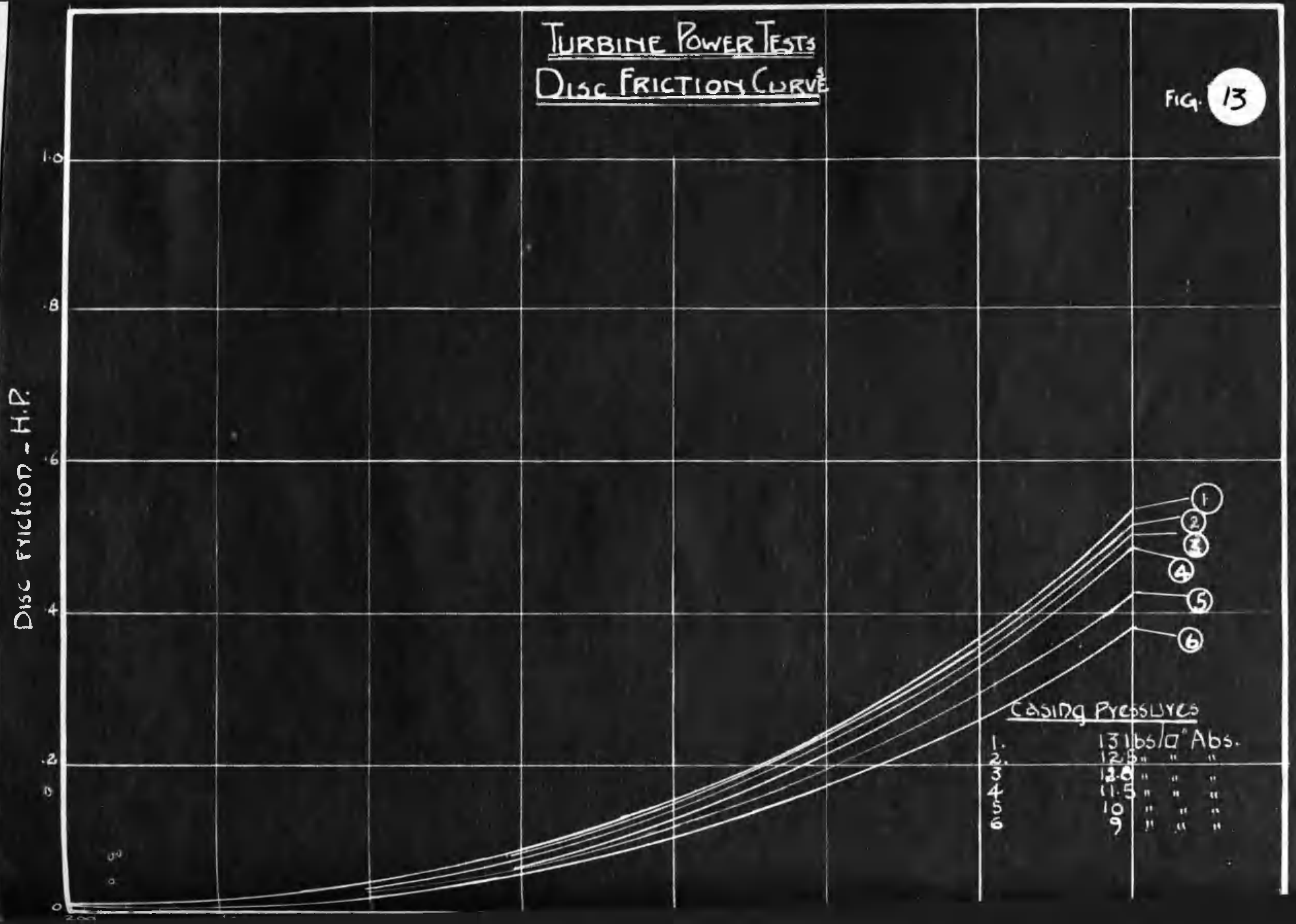
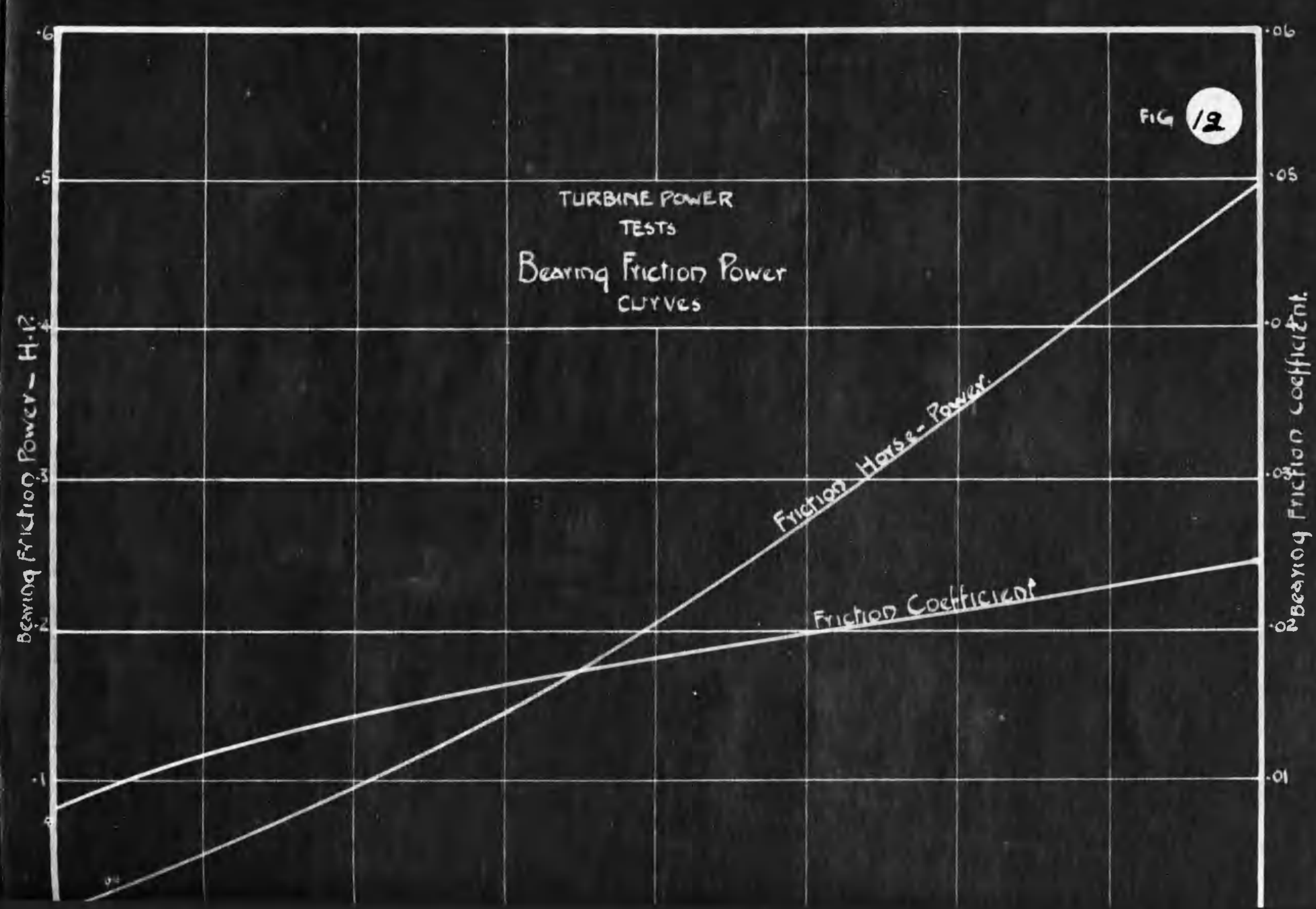

TURBINE POWER TESTS
DISC FRICTION CURVES
FIG. 13
Disc Friction - H.P.
1.0
.8
.6
.4
.2
0
1
2
3
4
5
6
Casing Pressures
1. 13 lbs/□" Abs.
2. 12.5 " " "
3 12.0 " " "
4 11.5 " " "
5 10 " " "
6 9 " " "

FIG 12



TURBINE POWER
TESTS
Bearing Friction Power
Curves
Friction Horse-Power.
Friction Coefficient
Bearing Friction Power - H.P.
Bearing Friction Coefficient.
·6
·5
·4
·3
·2
·1
·06
·05
·04
·03
·02
·01

represented by the dotted lines in fig. (11) are obtained by assuming an outlet geometrical angle of the blade and a blade velocity coefficient equal to unity.

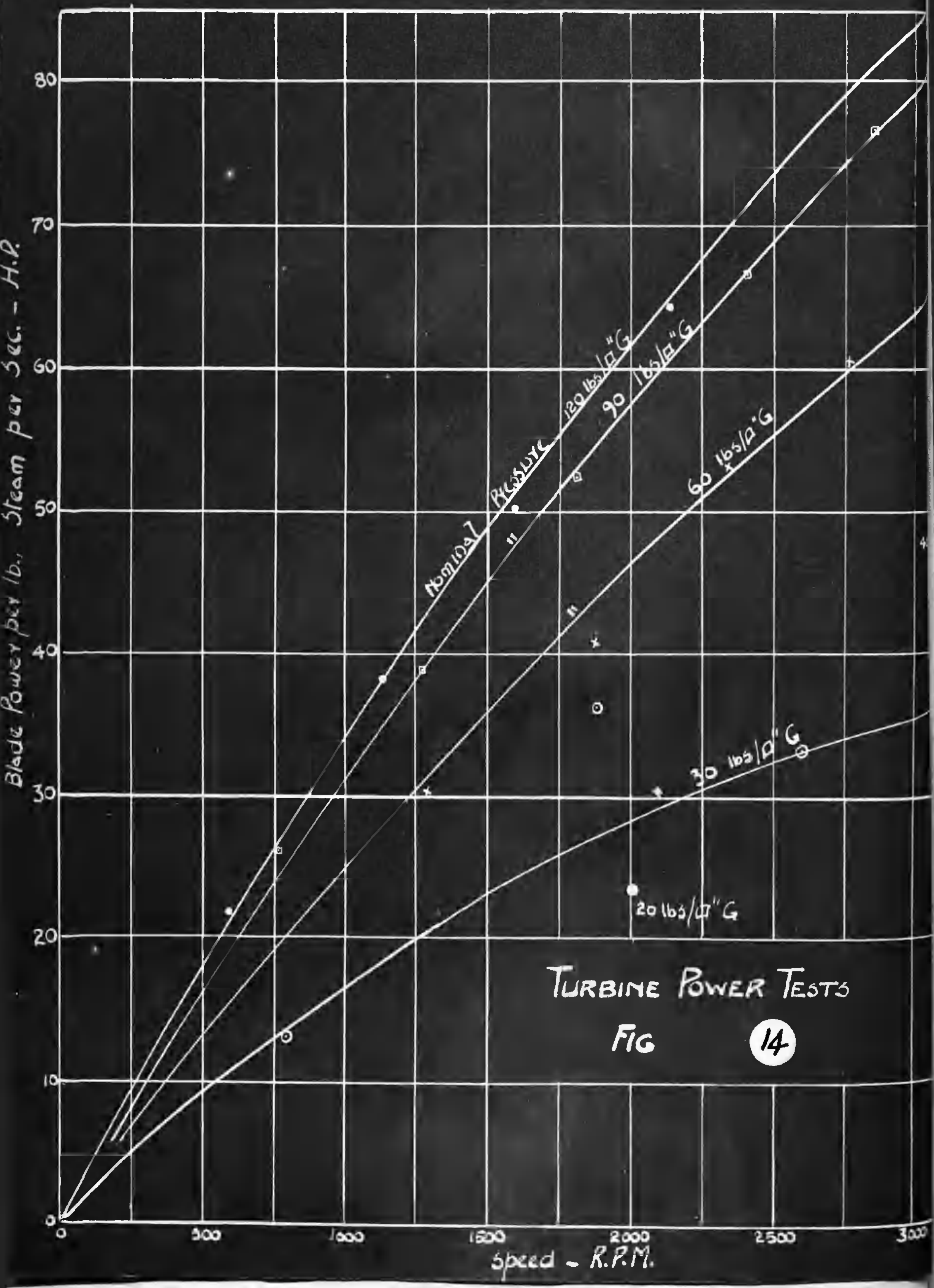
The full line curve approaches most closely to the ideal curve for a jet speed of 2,500 ft/sec corresponding to a pressure ratio of .1, which is the most effective ratio for the convergent-divergent nozzle fitted to the turbine.

We now come to the dotted line given in fig (8). The points for that line were taken from the torque curve in figure (11) and expressed in terms of impulse in lbs per lb of steam per second by simply dividing the values of the torque in ft lb. per lb steam by the radius of the rotor in feet. The width of the blades in the turbine are .787" and the blade pitch is equal to .539. This line is not in bad agreement with the writer's tests considering the different types of nozzles used in the writer's apparatus and the turbine, and also the difficulties of getting the correct torque value from the "Stationary Torque Turbine Tests" which arise from fixing the rotor in the right position.

(c) Turbine Power Tests (Jet Impulses). The test results are shown in figs. (14), (16) and (16a), and all of these tests were made with the convergent-divergent nozzle alone in operation. The results are only a few of the tests carried out, to enable of the subsequent analysis. The torque values are also seen expressed as "Jet Impulses" and plotted against Jet speeds in fig (16a).

Before calculating the torque values in these tests we have many other things to consider.

(1) Disc and Bearing Friction Power. After obtaining the brake horse power for each test, we have to determine the power actually developed in the blading. Now the only sources of loss between the blading and the brake are the frictional resistance between the turbine rotor and the steam in the



TURBINE POWER TESTS

FIG 14

casing and the friction at the rotor shaft bearings, the latter of course including the thrust bearing and the glands.

For the determination of the power lost in disc friction, the following formula was derived some years ago by Professor Wm. Kerr[1] from experimental data obtained from this same machine.

$$P_d = \left(\frac{S}{100}\right)^3 \left(\frac{d}{10}\right)^2 \frac{1}{V} \left\{ .020 \left(\frac{\sqrt{T}}{\frac{S}{100} \cdot \frac{d}{10} \cdot \frac{1}{V}} \right)^{\frac{1}{4}} + .04 \right\}$$

where:-

P_d - disc friction power .... H.P.

S = mean blade speed .... ft/sec.

d = mean diameter of blading .... inches.

V = Specific Volume of Steam in casing ...... $ft^3/lb.$

T = Temperature of the steam in casing ...... ^oF, Abs.

Inserting the numerical values of "d" we have:-

$$P_d = 11.56 \left(\frac{S}{100}\right)^3 \frac{1}{V} \left\{ .01472 \left(\frac{\sqrt{T}}{\frac{S}{100} \cdot \frac{1}{V}} \right)^{\frac{1}{4}} + .04 \right\}$$

The limiting values of T during all the power tests are 662.7^{o}F, Abs. and 672.9^{o}F, Abs., so that we may insert the mean value of T in the formula without any sensible error since this term occurs as $(T)^{1/3}$. This mean value is 667.7^{o}F, Abs., and making the substitution (1) reduces finally to :-

$$P_d = 11.56 \left(\frac{S}{100}\right)^3 \frac{1}{V} \left\{ .0332 \left(\frac{100V}{S} \right)^{\frac{1}{4}} + .04 \right\}$$

Fig. (13) gives a series of curves of P_d against rotor speeds in R.P.M. for a number of different casing pressures sufficient in cover the present tests.

For the determination of the bearing friction power

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(1) Paper by Prof. Wm. Kerr, Royal Technical College Journal 1924.

TURBINE POWER TESTS.

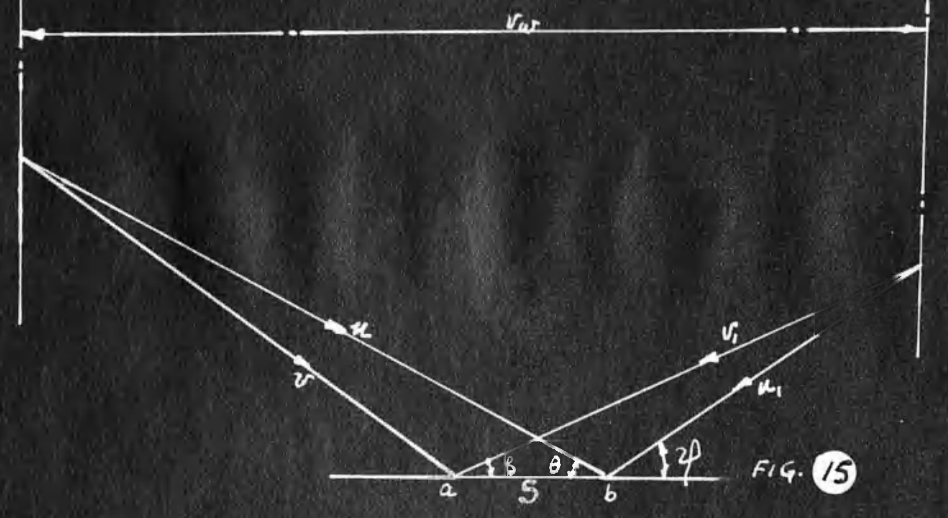


FIG. 15

we have in fig. (12) a curve of friction coefficients for various bearing surface speeds as given by Professor Wm. Kerr, and from this the curve of friction H.P. in the same figure is obtained.

<u>(2) Blade Power and Whirl Velocities</u>. To the brake power for each test we can now add, the disc and bearing friction power obtained as shown above, the sum of these three quantities being the power actually developed in the blading.

The blade power per lb of steam per second is shown plotted against rotor speed in fig. (14).

The data may now be still further reduced by calculating for each test the torque for unit steam flow and the velocity of whirl.

Considering the usual velocity diagram as shown in fig. (15).

Let :-

μ = absolute steam velocity at inlet ....ft/sec

v - relative steam velocity at inlet ....ft/sec

μ_1 = absolute steam velocity at outlet............ft/sec

v_1 = relative steam velocity at outlet............ft/sec

θ = Jet Angle.

β = geometrical angle of blade at outlet

φ = absolute exit angle of steam

S = blade speed ....ft/sec

v_w = velocity of whirl ....ft/sec

F_s = force exerted by steam on blading..lbs/lb steam/sec

T_s = Torque per lb steam/sec ....lbs. ft.

P_s = blade power per lb steam/sec .... H.P.

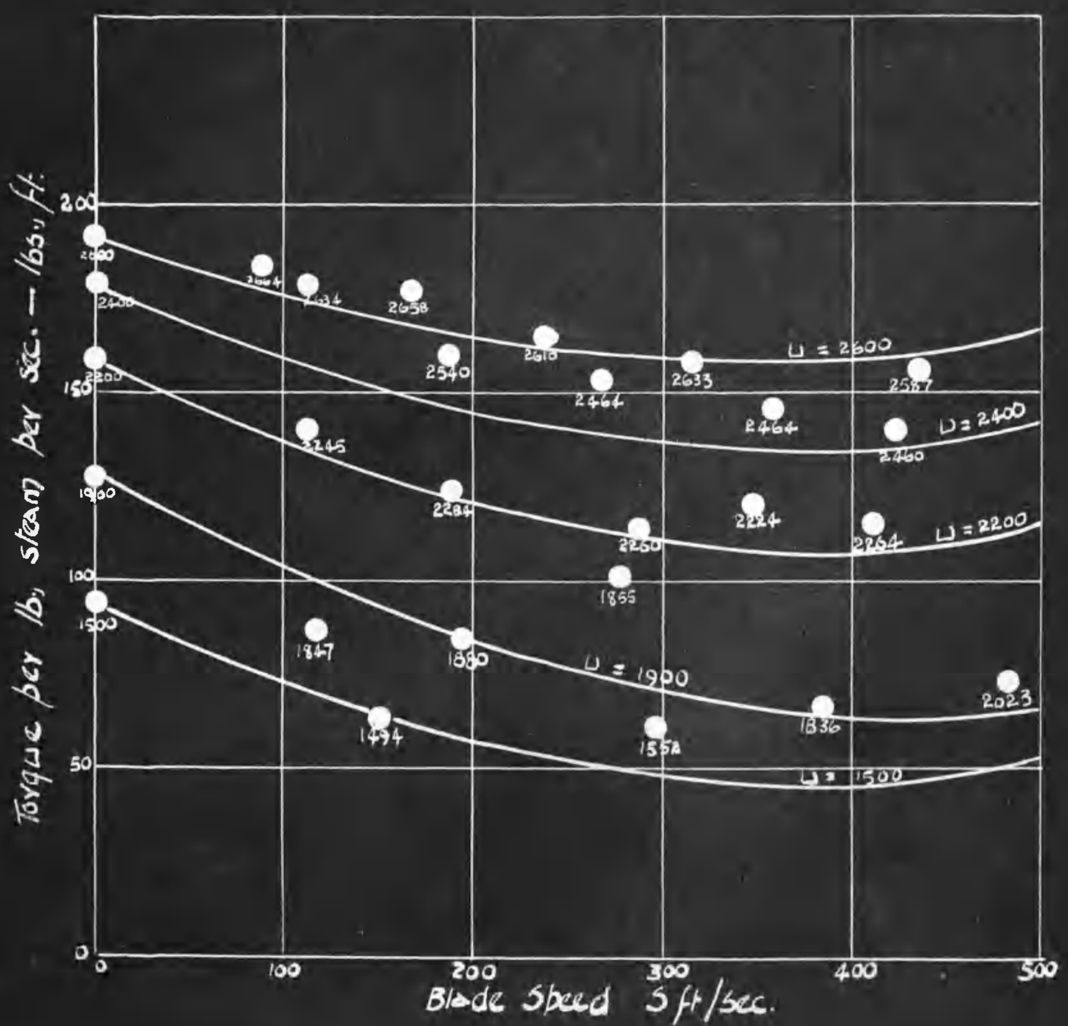
G = mass flow ....lbs/sec

g - 32.2

R = mean radius of blading .... inches

N = rotor speed ................................R.P.M.

Then we have :-



POWER AND STATIONARY.

TESTS

The Number Under Each point Indicates Jet Speed "U" ft/sec

FIG. 16

Change of velocity of steam $= u - w_1$

The component in the direction of the motion of the blading

$$= (u \cos\theta - u \cos\varphi)$$
$$= (u \cos\theta + v_1 \cos\beta - S)$$
$$= v_w$$

The change of momentum per lb of steam

$$= (u \cos\theta + v_1 \cos\beta - S) = \frac{v_w}{g} \text{ lbs wt.}$$

$$T_S = \frac{R}{12.g}(u \cos\theta + v_1 \cos\beta - S)$$
$$= v_w \frac{R}{12.g}$$
$$= \frac{v_w}{22.73}$$
$$\therefore P_S = \frac{S . v_w}{550.g}$$
$$= \frac{S}{550g}(u \cos\theta + v_1 \cos\beta - S)$$

also :-

$$T_S = \frac{33000 \, P_S}{2\pi N}$$
$$v_w = \frac{12 \, g \, T_S}{R}$$
$$= \frac{12.g \times 33000 \, P_S}{2\pi NR} = 1.192 \times 10^5 \frac{P_S}{N}$$

Fig. (16) shows T_S, the torque per lb of steam per sec, in lbs. ft. plotted against the blade speed S for both the Stationary and Power Tests on the turbine.

A series of curves are drawn in this figure, each curve being for a particular jet speed but these curves are not sufficiently defined to permit of their being used to determine torque values in preference to the plotted points. The individual test figures will thus be retained. The above curves are also shown expressed as Jet impulses and plotted against blade speeds.

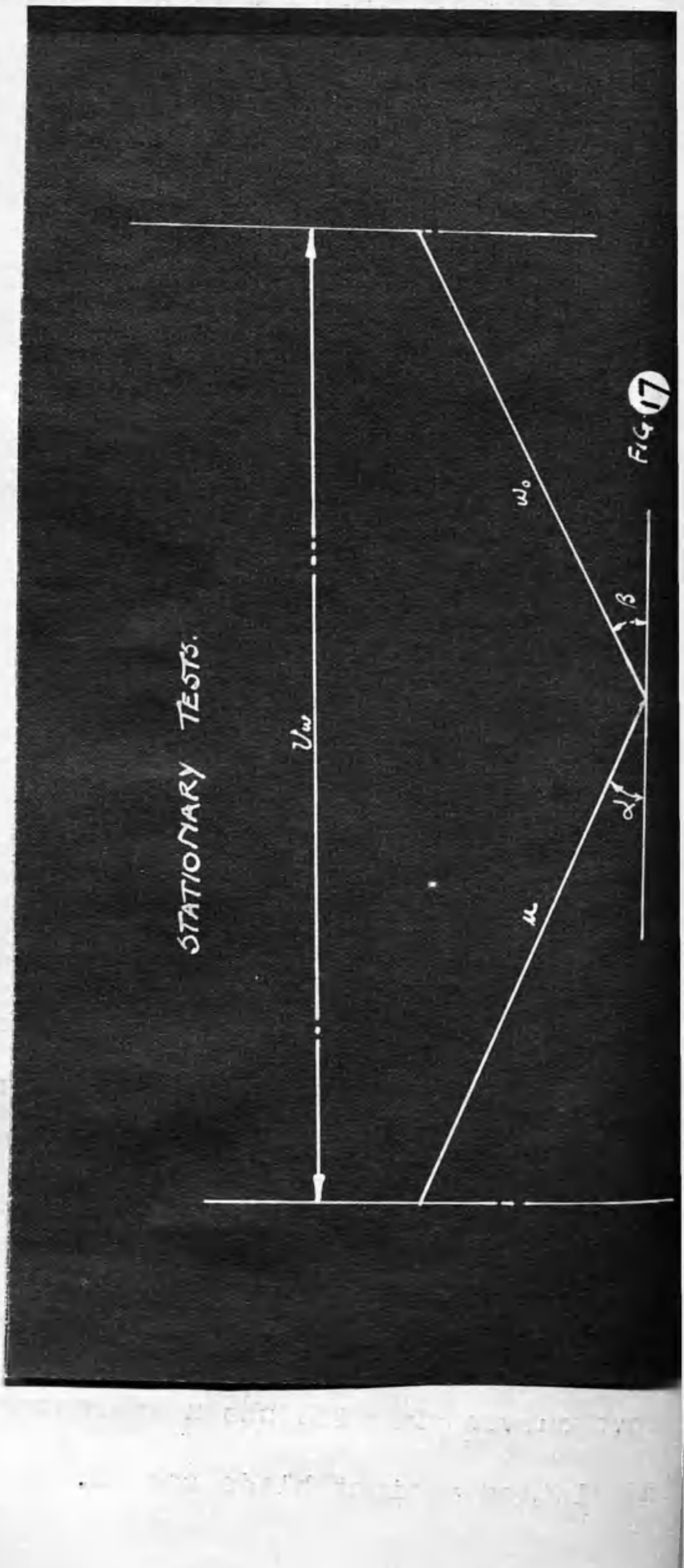


FIG. 17

BLADE VELOCITY COEFFICIENTS.

The results are now in a form to enable the examination of the blade velocity coefficients.

Fig. (18) gives the velocity diagrams of the 1" blade from the author's apparatus. Since the blades were kept stationary, then, naturally the blade speed becomes zero and the two points a and b (15)&will coincide at the point (A) Fig. (17).

μ & α fix the direction and magnitude of the jet speed μ , α being the nozzle angle.

The velocity of whirl is equal to the horizontal change of velocity.

i.e.

$$v_w = \frac{32F}{M}.$$

where

F = the pull on the blades in lbs.

M = the condensate in lbs/sec.

v_w is then set off as shown in fig. (17) and ω_o the outlet speed of the steam from the blades is measured from the fig.

The blade velocity coefficient $b = \frac{\omega_o}{\mu}$

Considering again fig. (17) we have:-

$$v_w = \mu \cos\alpha + \omega_o \cos\beta$$

But

$$\omega_o = \mu b$$

$$\therefore \quad v_w = \mu \cos\alpha + \mu b \cos\beta$$

$$\therefore \quad v_w - \mu\cos\alpha = \mu b \cos\beta$$

or

$$b = \frac{v_w - \mu\cos\alpha}{\mu\cos\beta}$$

$$\therefore \quad b = \frac{\frac{32F}{M} - \mu\cos\alpha}{\mu\cos\beta}$$

where β = the vane angle.

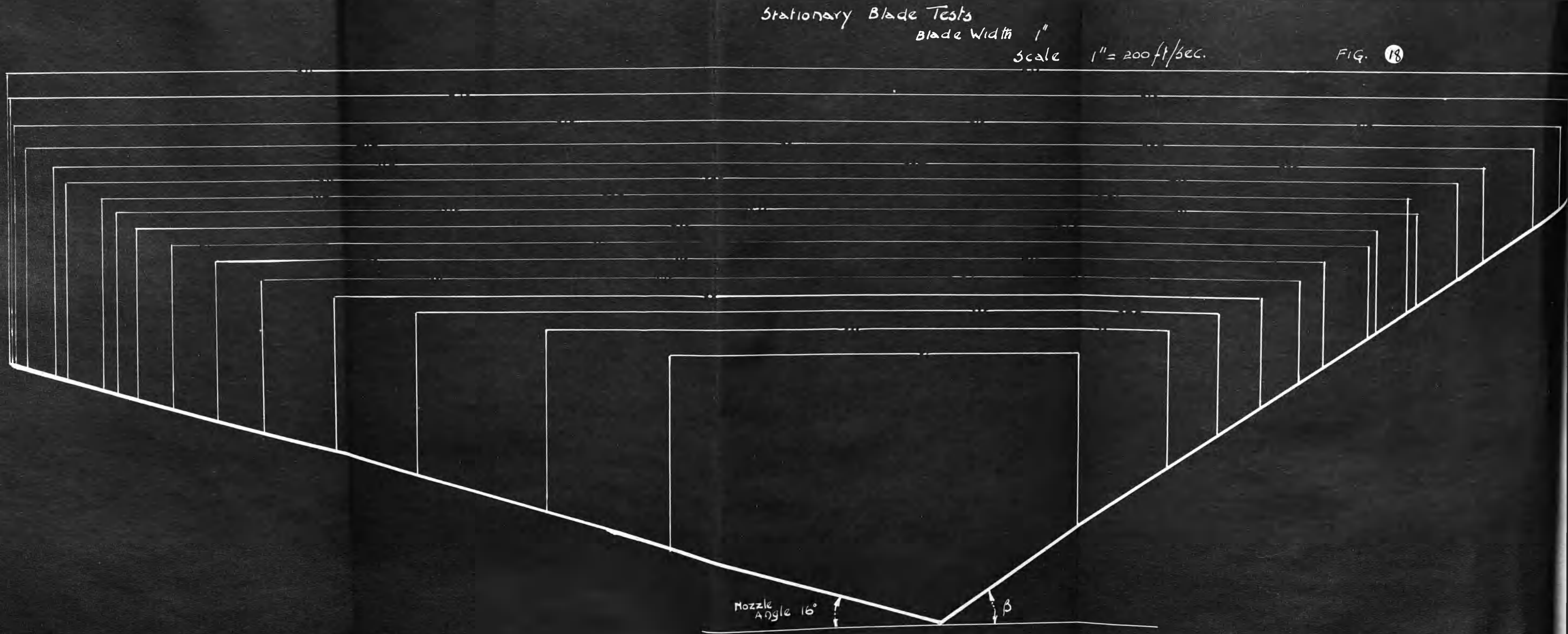

Stationary Blade Tests
Blade Width 1"
Scale 1" = 200 ft/sec.
Nozzle Angle 16°
β

FIG. 18

The above method for determining the velocity coefficients was used in all the stationary tests.

The determination of the blade velocity coefficients from the power tests were made by drawing the ordinary velocity diagram fig. (15).

The blade velocity coefficients for the "Stationary Blade Apparatus" of the author are shown plotted against a pressure ratio base in fig. (19), whereas those for the "Stationary Torque" and "Power" Turbine tests are shown in fig (20).

The coefficient "b" has a sudden drop after it reaches its maximum value at a pressure ratio of .15 in "Stationary Blade Apparatus" Tests fig (19) and after a maximum value of about .1 in all the Turbine Tests fig. (20).

These pressure ratios correspond with the best pressure ratio~~s~~ of the nozzles in operation respectively.[x]

The occurrence of the maximum value of "b" at this point confirms the impressions given by the Jet Impulse curves, namely, that in addition to having the highest efficiency, the jet also has the best form at this ratio, resulting in a reduction of the loss in the gap, and blade passage.

At the low jet speeds the curves seem to tend to the horizontal and keep a constant value which could easily be explained by the fact that the jet is so poor that it has no further effect on the blade coefficient.

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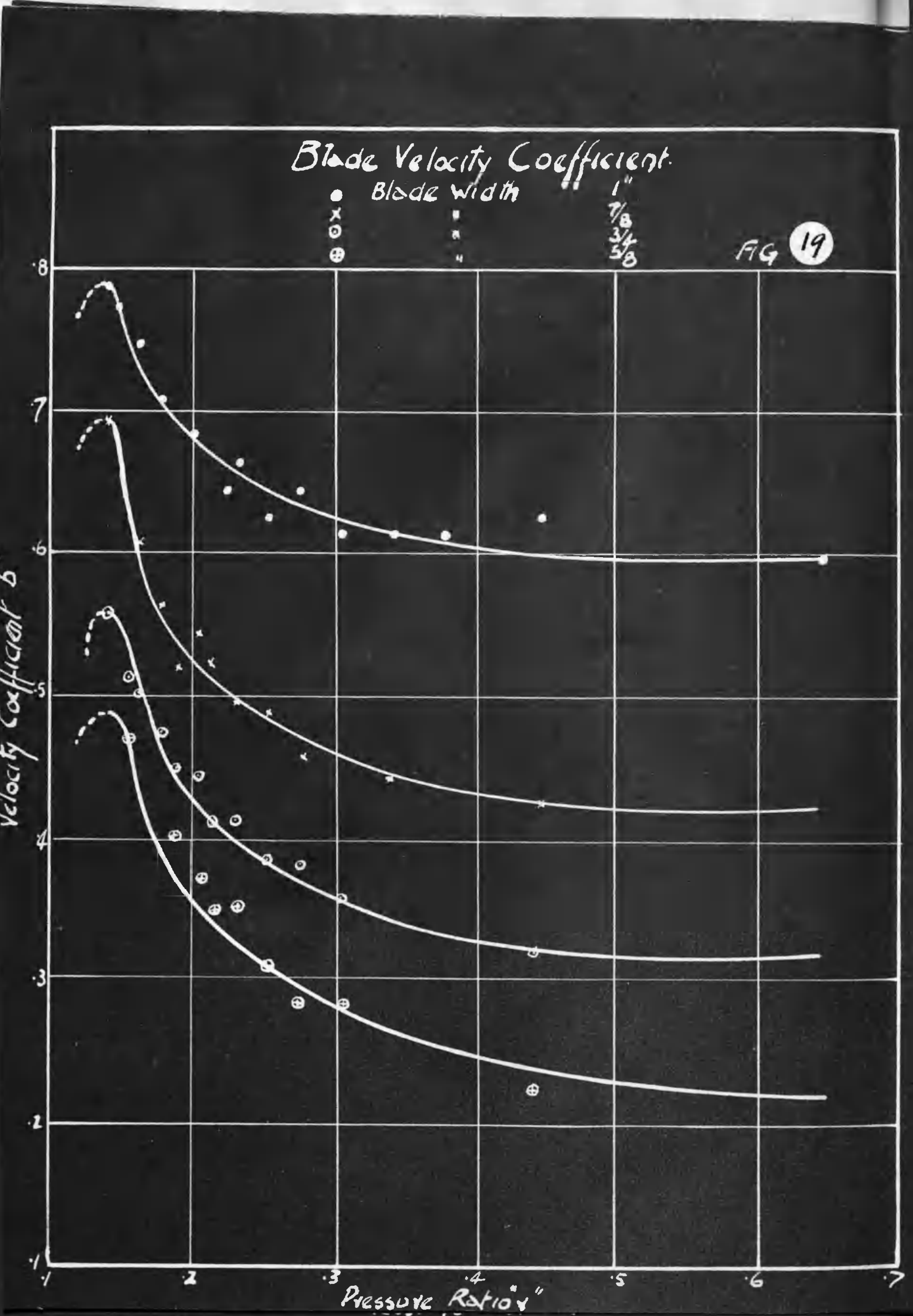
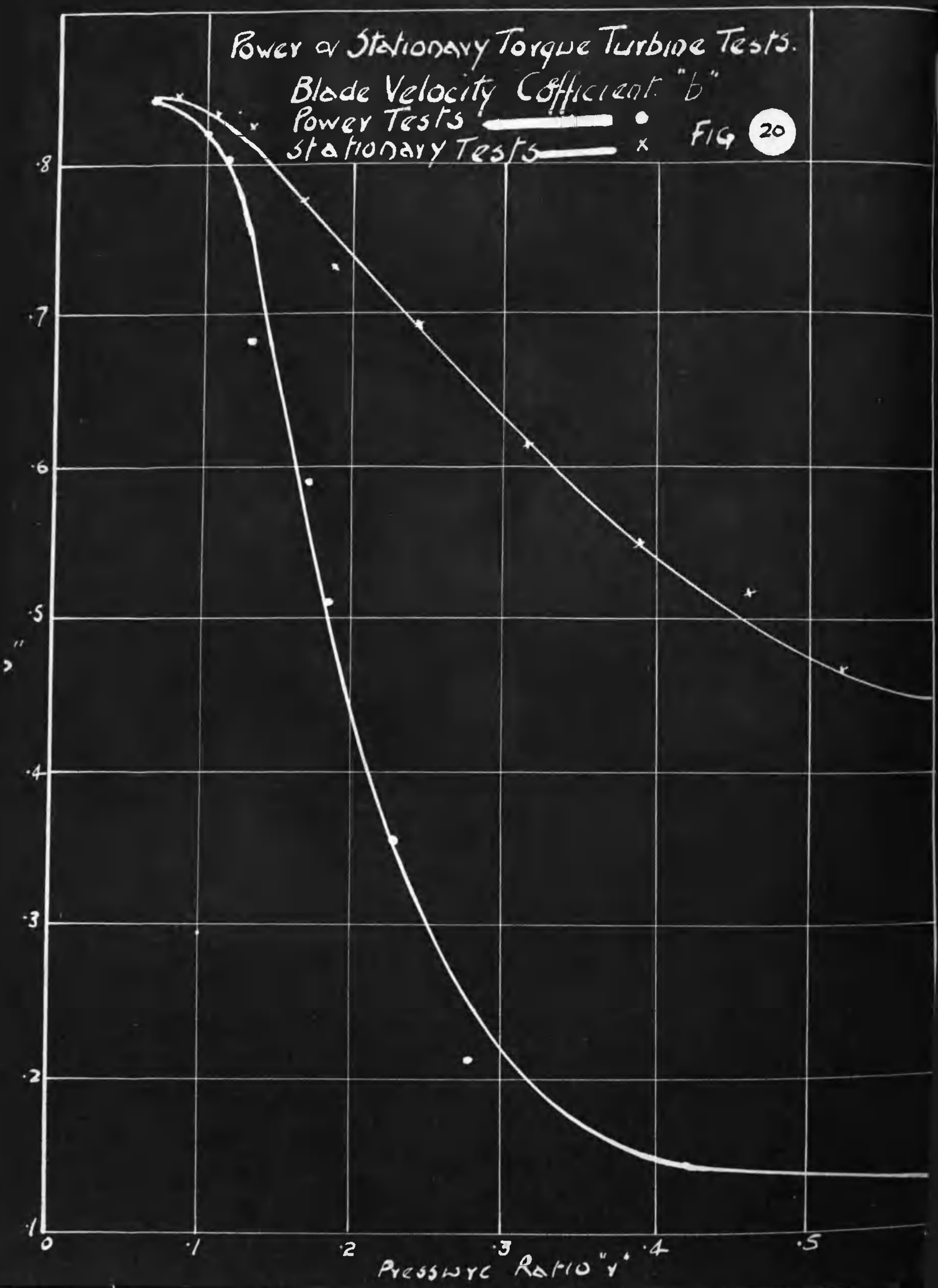
[x] See appendix.



Blade Velocity Coefficient.
• Blade Width " 1"
× " " 7/8
⊙ " " 3/4
⊕ " " 5/8
FIG 19
Velocity Coefficient "b"
·8
·7
·6
·5
·4
·3
·2
·1
·1
·2
·3
·4
·5
·6
·7
Pressure Ratio "r"


Power & Stationary Torque Turbine Tests.
Blade Velocity Coefficient "b"
Power Tests •
Stationary Tests ×
FIG 20
·8
·7
·6
·5
·4
·3
·2
·1
"b"
0
·1
·2
·3
·4
·5
Pressure Ratio "r"

LOSSES IN BLADE PASSAGE.

It has already been stated that we cannot assume a single blade velocity coefficient to cover all the energy losses in the blade passage, for the simple reason that it would have a very high value.

This being the case we are led to the conclusion that other effects are present and that these effects result in the reduction of the jet velocity or are equivalent to such a reduction.

This fact is clearly shown by the blade coefficient curves.

The separation of these effects, then become essential and is undoubtedly a decided improvement on the method which slumps together a variety of effects due to several occurrences which are distinctly different in their modes of action and in their importance.

The losses in the blade passage will thus be summed up as follows:-

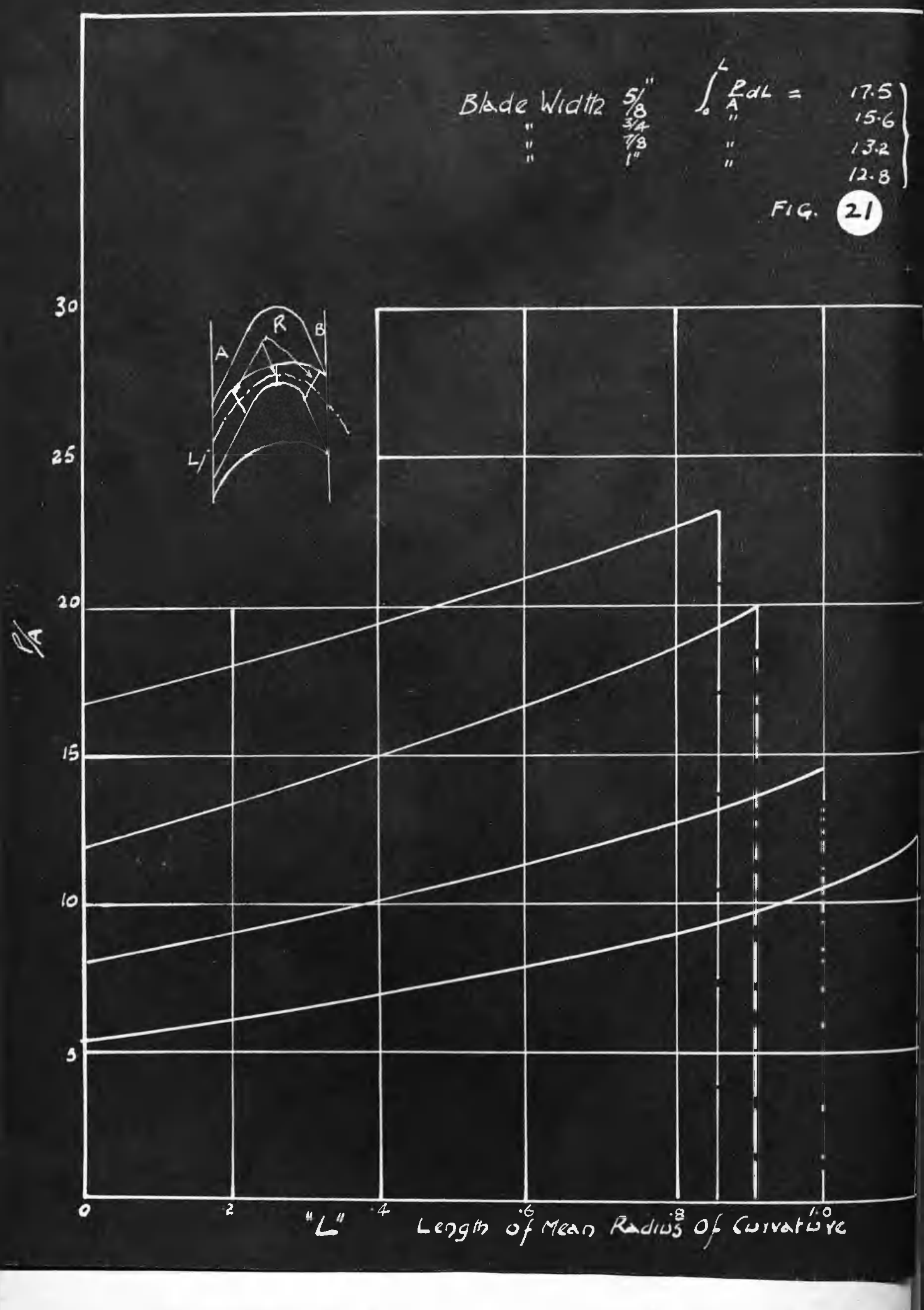
(1) Boundary Loss.
(2) Curvature Loss.
(3) Jet Quality effect.

The blade velocity coefficient curves have a maximum value at the best working pressure ratio of the nozzle which shows that at that particular point the losses are due to boundary & Curvature losses.

Let b_o = maximum blade velocity coefficient.

Then $(1 - b_o^2)$ = boundary Loss + Curvature Loss.

The drop in the curve takes place immediately after the best working pressure ratio of the nozzle is reached which shows that the loss occurring at that part of the curve where the



Blade Width 5/8" $\int_0^L \frac{R}{A} dL$ = 17.5
" 3/4 " 15.6
" 7/8 " 13.2
" 1" " 12.8
A
R
B
L
30
25
20
15
10
5
R/A
0
.2
.4
.6
.8
1.0
"L" Length of Mean Radius Of Curvature

FIG. 21

jet is very poor is due to an effect that is brought forward by the quality of the jet.

Let b = blade velocity coefficient.

Then:-

$(1 - b^2) - (1 - b_o^2)$ or the drop in the velocity coefficient must be equivalent to that effect.

(1) <u>Boundary Loss</u>. Considering the factors which determine the magnitude of the loss to which the fluid is subjected, one of the obvious factors is the frictional resistance offered by the blade surface to the passage of steam.

This boundary loss taking place in the blade passage should vary as:

$$\int_0^L \frac{p}{A}\, dL$$

where:

$\frac{A}{p}$ = average hydraulic mean depth

and is = $\dfrac{p \sin\theta\, l}{2(p + l)}$

p = Blade pitch

θ = Outlet angle of steam.

l = Blade height.

L = Total length of mean curvature of blade (See Fig. 21).

dL - A small increment in the length of mean curvature of blade.

Let C_1 = Friction coefficient.

Then:

$$\text{Boundary Loss} = C_1 \int_0^L \frac{p}{A}\, dL$$

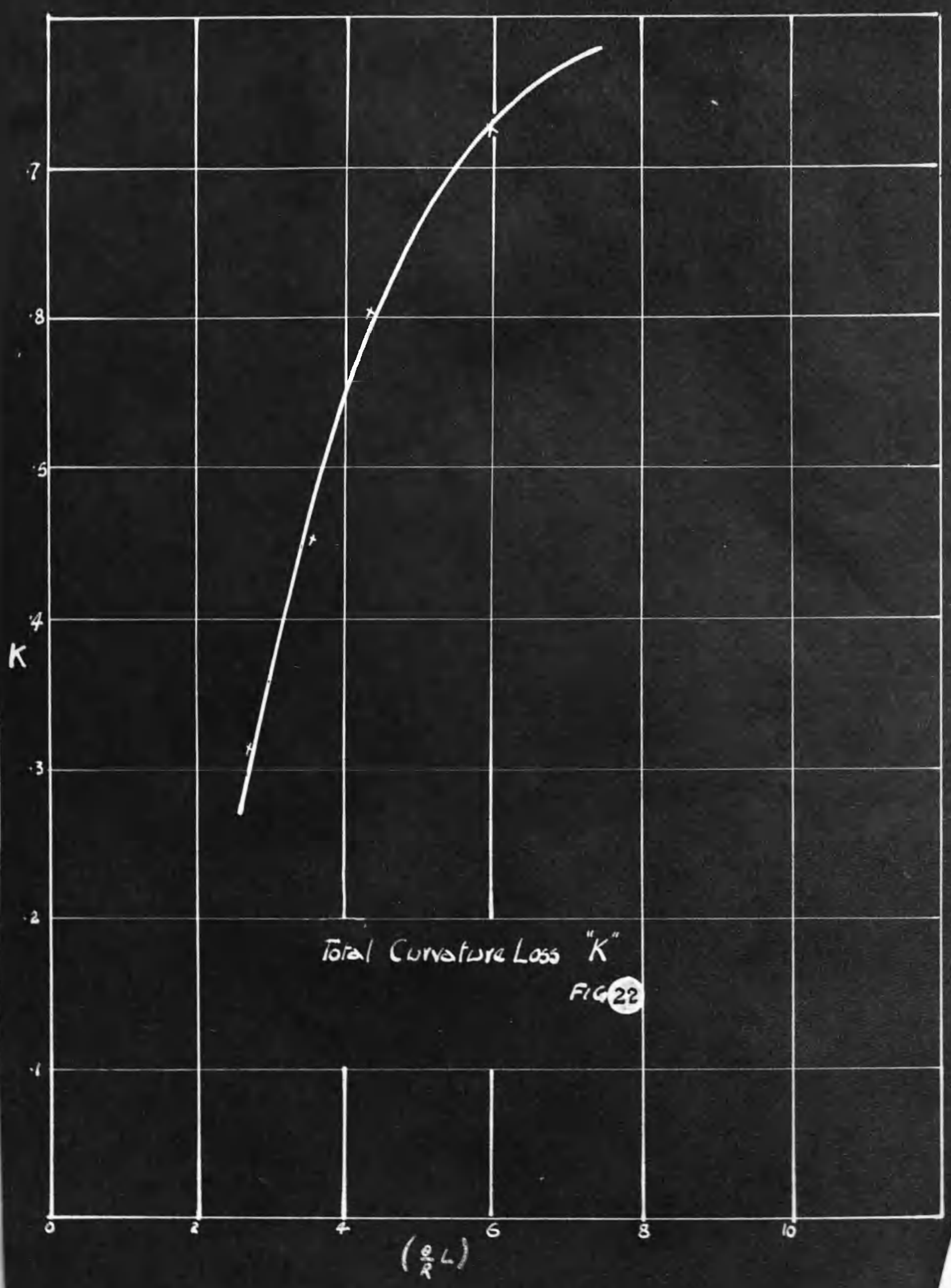

K
·7
·8
·5
·4
·3
·2
·1
0
2
4
6
8
10
Total Curvature Loss "K"
$\left(\frac{\theta}{R} L\right)$

FIG 22

C_1 [1] is taken as .005 for this kind of finish of surface.

The value of $\int_0^L \frac{p}{A}\, dL$ was found by graphical integration fig. (21) by taking sections across the blade passage and calculating the value of $\frac{p}{A}$ and then plotting it against the corresponding value of the length of mean curvature of blade " L ", the area under the curve being the value of

$$\int_0^L \frac{p}{A}\, dL .$$

(2) Curvature Loss. This loss is brought about by the continuous change of direction of the steam going through the blade passage.

It will be agreed that steam flowing through a straight pipe only entails a friction loss which varies according to the finish of the surface in contact with the steam , where as in a curved passage there is another factor which plays a big part in adding to the losses, this being the curvature loss and is due to the changing direction of the steam.

This curvature loss should vary as

$$\left(\frac{\theta}{R} L\right)$$

or to some power of

$$\left(\frac{\theta}{R} L\right)$$

It has already been stated that the loss at the maximum point of the blade velocity coefficient is due to boundary and curvature effects.

$$\therefore \quad (1 - b_o^2) = C_1 \int_0^L \frac{p}{A}\, dL \quad + \text{ Curvature loss---(1)}$$

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(1) Paper on Pipe Losses in Nozzles, by Profs., A.L. Mellanby and Wm. Kerr. Proc. I.E.C.E. & S.

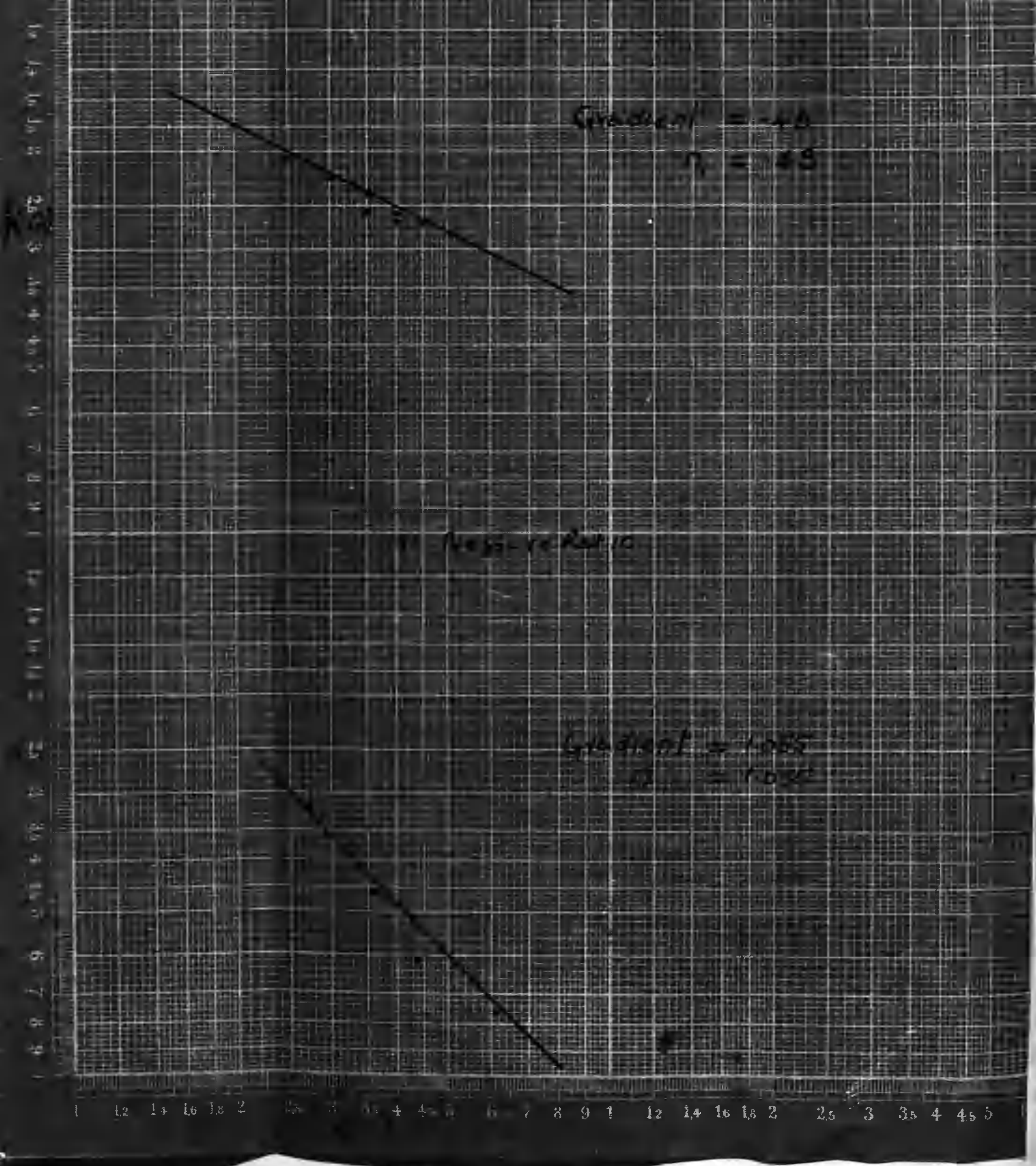


Fig (23)

Let this curvature loss be K and as we know all the other parts of expression (1) then we can find the curvature loss K for all the four sets of blades used.

Plotting this value of K against $\frac{\theta}{R}$ the nett curvature of the blade fig. (22) shows that it could be log-plotted to find the power. A log plot is shown in fig. 23 and gives a gradient of 1.055.

From the log plot fig. (23) we have

The line shown is of the order $y - mx = C$

$$\therefore \log K - n \log\left(\frac{\theta}{R}L\right) = \log C_2$$

or

$$\log \frac{K}{\left(\frac{\theta}{R}L\right)^n} = \log C_2$$

$$\therefore \frac{K}{\left(\frac{\theta}{R}L\right)^n} = C_2$$

where C_2 is a constant and equals the blade curvature coefficient.

$$\therefore K = C_2\left(\frac{\theta}{R}L\right)^n$$

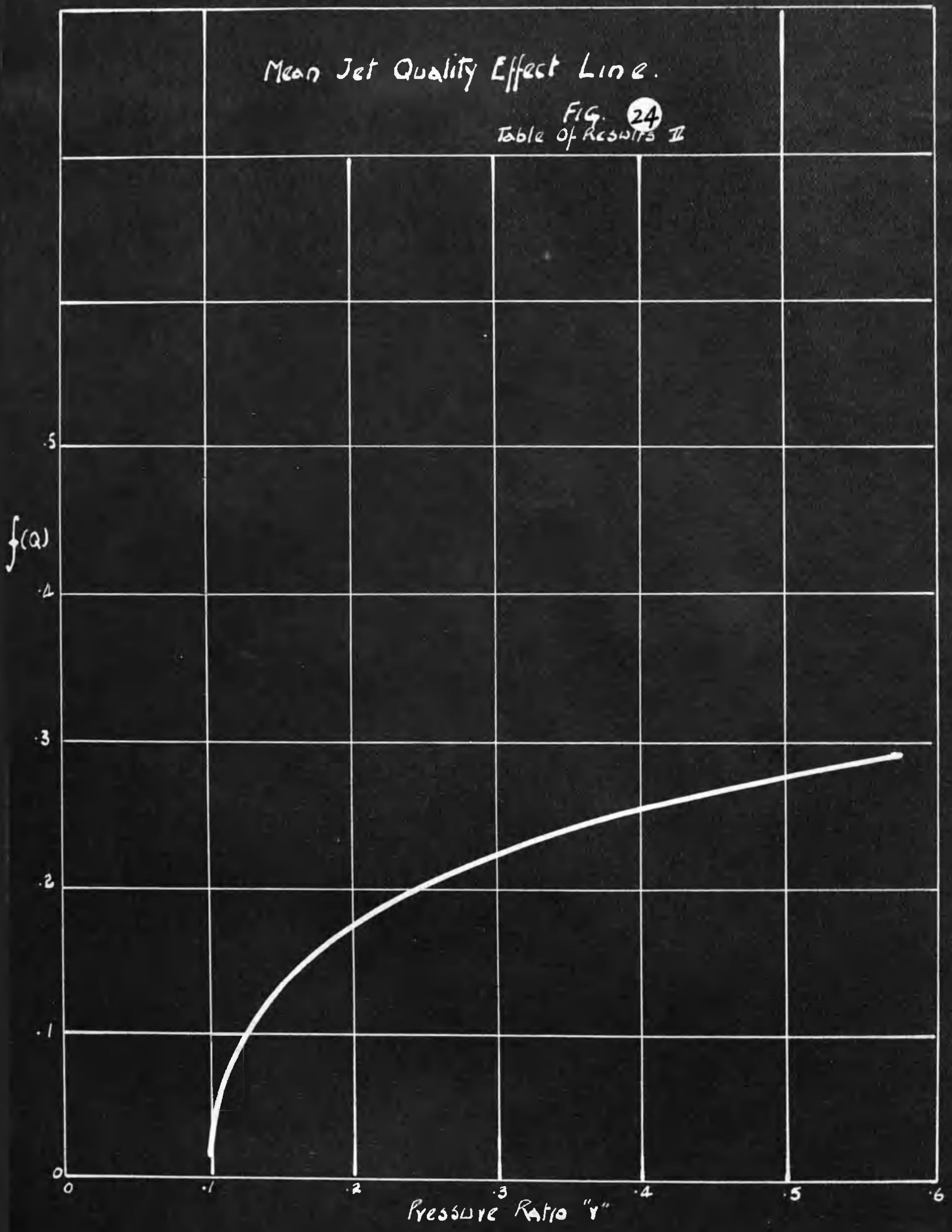
Table II gives the value of as [redacted] .1134

$$\therefore K = .1134\left(\frac{\theta}{R}L\right)^{1.055}$$

$$\therefore (1 - b_o^2) = C_1\int_0^L \frac{p}{A}\,dL + C_2\left(\frac{\theta}{R}L\right)^n$$

Substituting the values of the coefficients and the index we have:

$$(1 - b_o^2) = .005\int_0^L \frac{p}{A}\,dL + .1134\left(\frac{\theta}{R}L\right)^{1.055} \quad \text{-----(2)}$$



Mean Jet Quality Effect Line.
FIG. 24
Table Of Results II
f(Q)
·5
·4
·3
·2
·1
0
0
·1
·2
·3
·4
·5
·6
Pressure Ratio "r"

The index 1.055 is so near unity that we can write equation (2) as follows:

$$(1 - b_0^2) = .005 \int_0^L \frac{k}{A}\,dL + .1134\left(\frac{\theta}{R}L\right)$$

(3) <u>Jet Quality Effect.</u> The blade velocity coefficient as already explained is best when the pressure ratio of operation of the nozzle is at its best, which indicates that not only is the efficiency of the nozzle a maximum at this pressure ratio but also that the jet is of particularly good quality at this point.

Had the blade velocity coefficient kept steady at its maximum value b_0 then the losses would have only been due to boundary and curvature effects, but the drop in the curve shows that there is still another factor to be accounted for, which could be none other than the effects resulting from the poor quality of the jet.

The steam at the higher pressure ratios is of ~~of~~ a very poor quality as the losses resulting from recompression will increase as the jet speed falls, the recompression loss being the more severe at the higher pressure ratios.

The pressure ratio term has thus to be reduced to cover jet quality since the degree of turbulence depends upon the pressure ratio at which the nozzle is operating.

Considering all the above effects the quality effect should vary with the extent of compression in the nozzle.

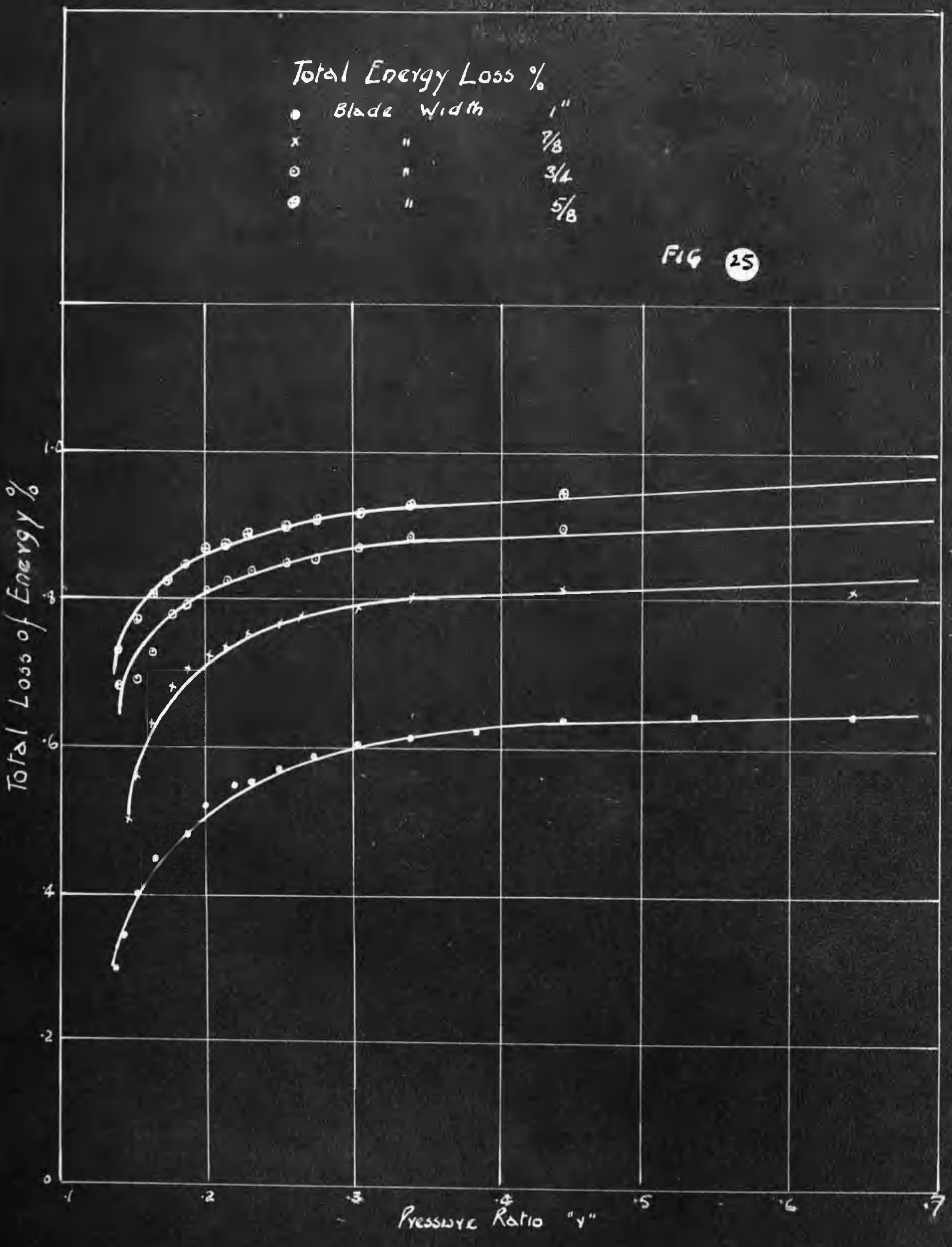
Let $Q = \dfrac{\text{Pressure Ratio}}{\text{Best Pressure Ratio.}}$

$b_0 =$ Maximum blade velocity coefficient.

$b =$ blade velocity coefficient.

The percentage energy loss at the maximum point b_0 must then be equal to

$$(1 - b_0^2)$$


Total Energy Loss %
Blade Width 1"
" 7/8
" 3/4
" 5/8
FIG 25
Total Loss of Energy %
Pressure Ratio "r"
0
·2
·4
·6
·8
1·0
·1
·2
·3
·4
·5
·6
·7

This has already been dealt with and is equal to boundary and curvature losses:

The drop is due to quality effects and must be some function of Q

$$\therefore \qquad f(Q) = (1 - b^2) - (1 - b_o^2) \text{-------} (3)$$

This value of $f(Q)$ is calculated and is shown plotted against a pressure ratio base in fig. (24). The values being shown in Table (II.)

This was again log plotted to get an expression to satisfy this curve.

This log plot is shown on the same sheet as the curvature plot Fig. (23).

The logarithmic curve shown is of the form

$$\log f(Q) - \log r^n = \log C_o$$

where C_o is a constant.

$$\therefore \qquad \frac{f(Q)}{r^{n_1}} = C_o$$

$$\therefore \qquad f(Q) = C_o r^{n_1}$$

n_1 (from the log curve) = .48

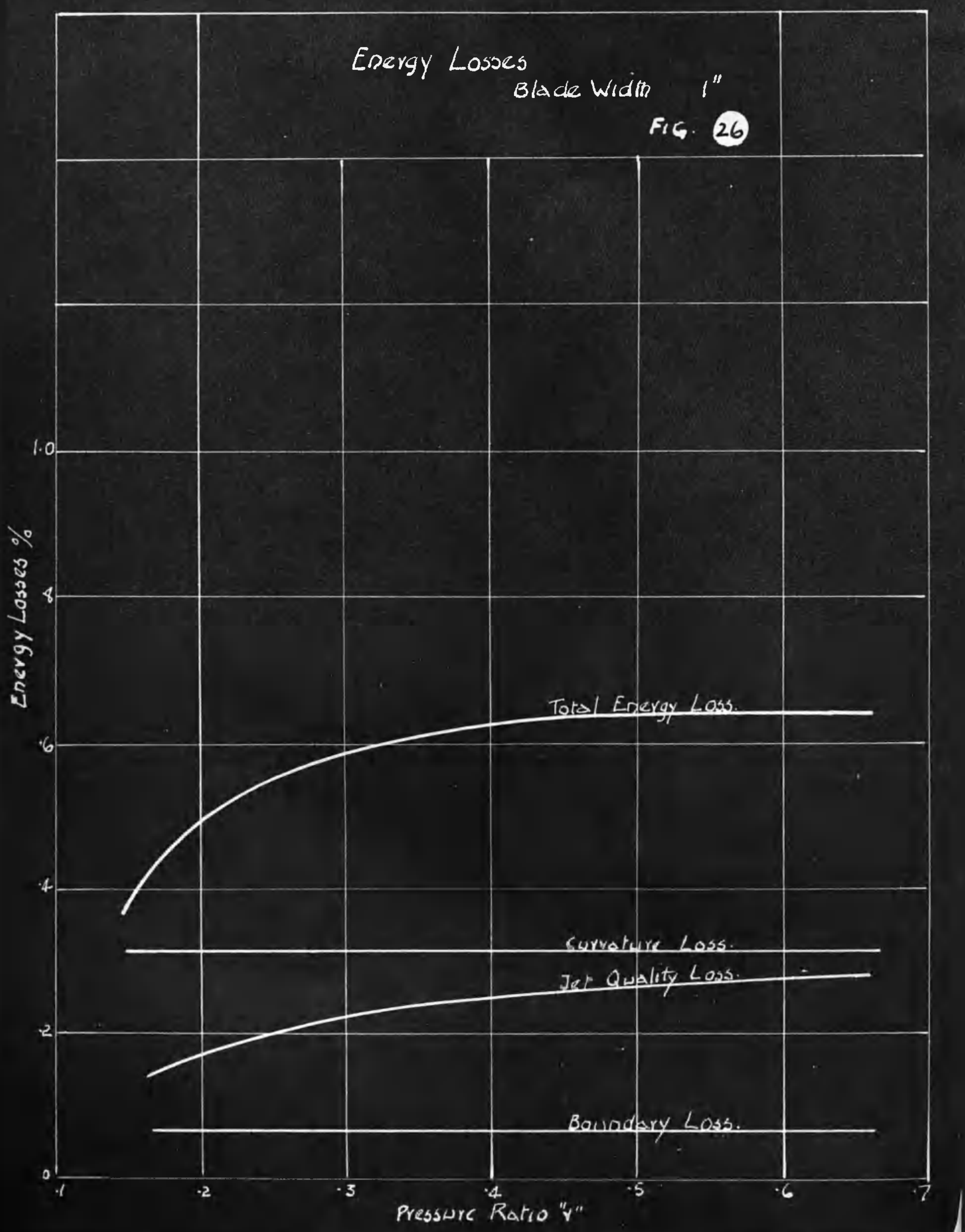
$$\therefore \qquad f(Q) = C_o r^{.48}$$

But C_o = .396

Then $f(Q)$ = $.396 r^{.48}$

From Equation (3) we have:-

$$f(Q) + (1 - b_o^2) = (1 - b^2) \text{---------} (3)_a$$


Energy Losses
Blade Width 1"
FIG. 26
Total Energy Loss.
Curvature Loss.
Jet Quality Loss.
Boundary Loss.
Energy Losses %
1.0
·8
·6
·4
·2
0
·1
·2
·3
·4
·5
·6
·7
Pressure Ratio "r"

But:-

$$(1 - b_o^2) = .005 \int_0^L \frac{p}{A} \, dL + .1134 \left(\frac{\theta}{R} L\right)^{1.055}$$

Substituting this value of $(1 - b_o^2)$ and of $f(Q)$ in equation $(3)_a$ we have:-

$$(1 - b^2) = .005 \int_0^L \frac{p}{A} \, dL + .1134 \left(\frac{\theta}{R} L\right)^{1.055} + .396 r^{.48}$$

∴ The percentage energy loss in a blade passage could be separated, in terms of a boundary loss, a curvature loss, and a jet quality effect, thus:

$$(1 - b^2) = C_1 \int_0^L \frac{p}{A} \, dL + C_2 \left(\frac{\theta}{R} L\right) + C_o \, r^{n_1} \text{------} (4)$$

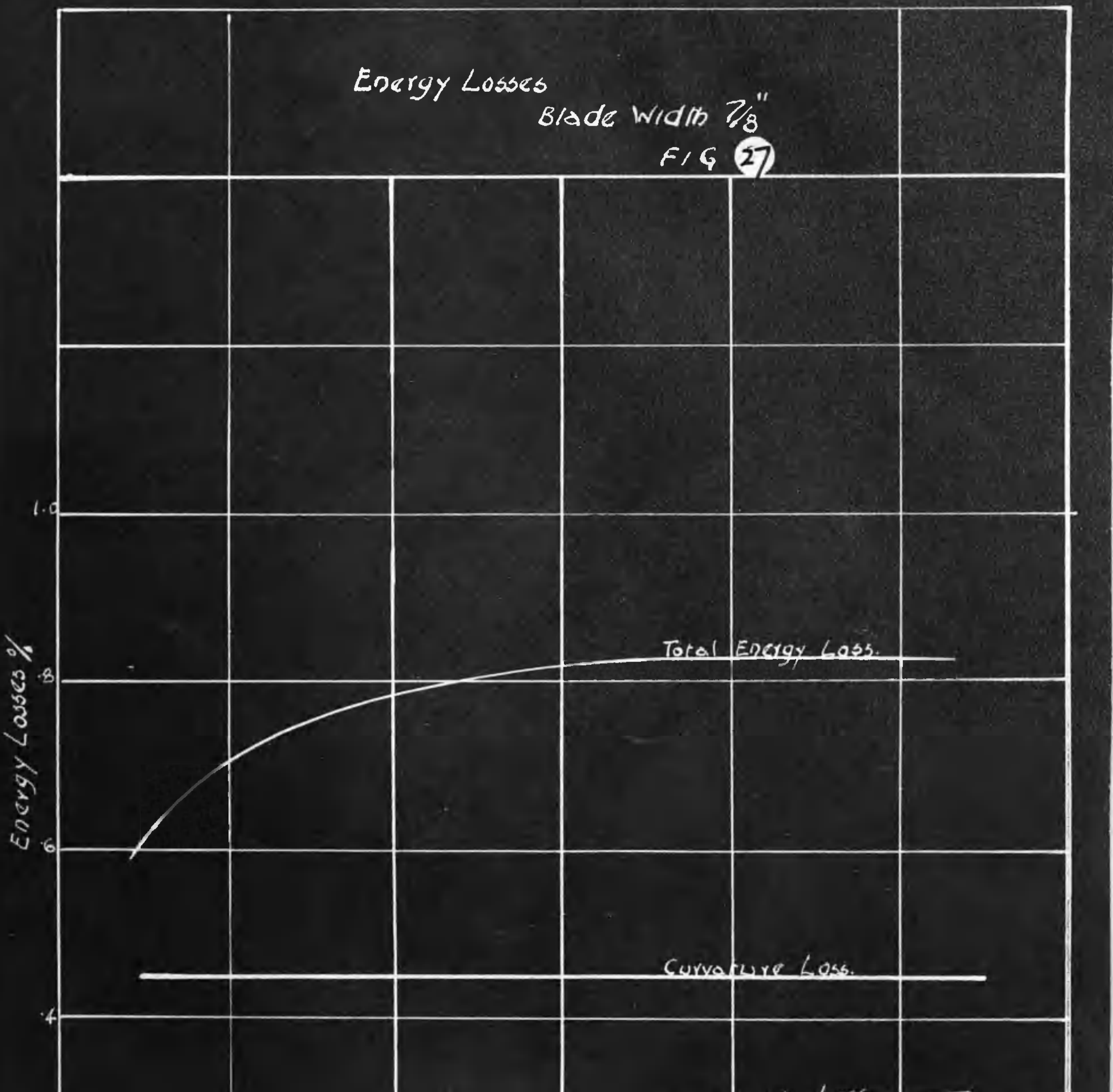
where C_1 = Friction coefficient.

C_2 = Curvature coefficient.

q C_o = A quality coefficient.

ENERGY LOSSES.

Fig. (25) gives the total Energy Loss % (which is equal to $(1 - b^2)$) against a pressure ratio base, and shows that the larger the blade the less the loss of energy %. From Figs. (26 - 29) it will be seen that the boundary loss decreases with the increase of blade width, so does the curvature loss decrease with the increase of the blade width, while the effect that is brought by the jet quality is constant as the nozzle used is the same for all the author's experiments.


Energy Losses
Blade Width 7/8"
FIG 27
Total Energy Loss.
Curvature Loss.
Jet Quality Loss.
Boundary Loss.
Energy Losses %
1.0
·8
·6
·4
·2
0
·1
·2
·3
·4
·5
·6
·7
Pressure Ratio "r"

The curvature loss plays the biggest part in the amount of energy losses in the blade passage.

The values of c_1 or c_2 were next applied to the tests on the turbine both stationary and Power and the curves are shown in fig. (30).

It will be noticed that the velocity coefficient b is much lower in the Power Tests than in the stationary tests on the Turbine fig.,(20).

This will mean, of course, that the blade is subjected to more energy losses when running than when it is stationary and seeing that the boundary and curvature losses must be the same for the same blade whether it be moving or stationary and that this extra loss could not be due to some jet quality effect, then it should be due to some steam being kept in the blade passage by the centrifugal force while the turbine is running. That quantity of steam though it may be small, will disturb the flow of the steam through the blade passage, that is in the stationary tests and hence cause this further loss.

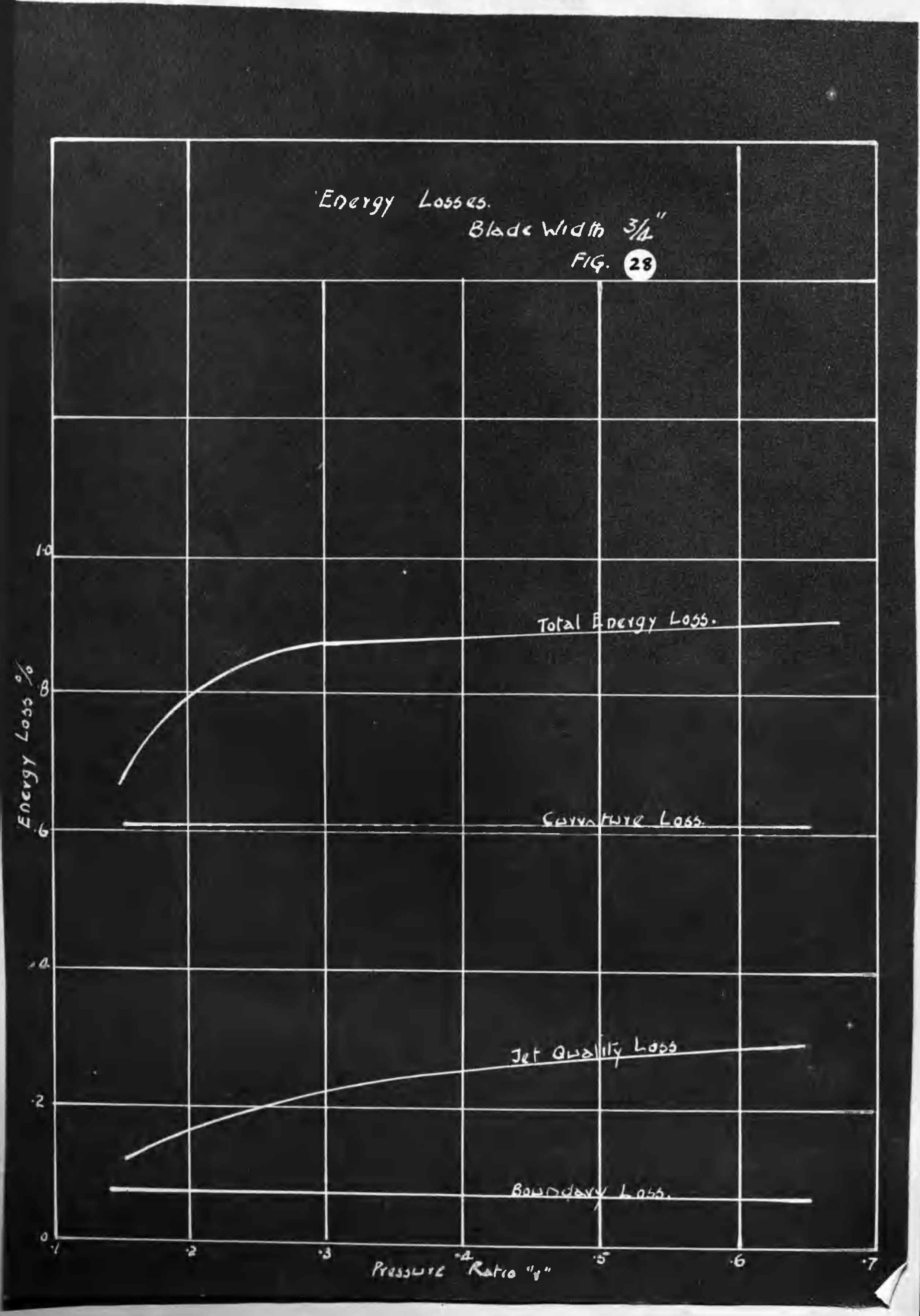
We may submit then that this use of separating the various losses in a blade passage is quite rational.

It might be well to note that the turbine from which the experimental data for the Stationary Torque and Turbine Power Tests were obtained is a somewhat abnormal machine in that the arc of admission is exceptionally short[1] and that the nozzle used (see Appendix) was of rectangular section.

It is not suggested that on this account, however, that the method of separation of the different losses is not applicable here, as there can be no doubt that these losses occur in all types of machines.

---

(1) Jet action in Turbines with short admission arcs by Prof. Wm.Kerr.

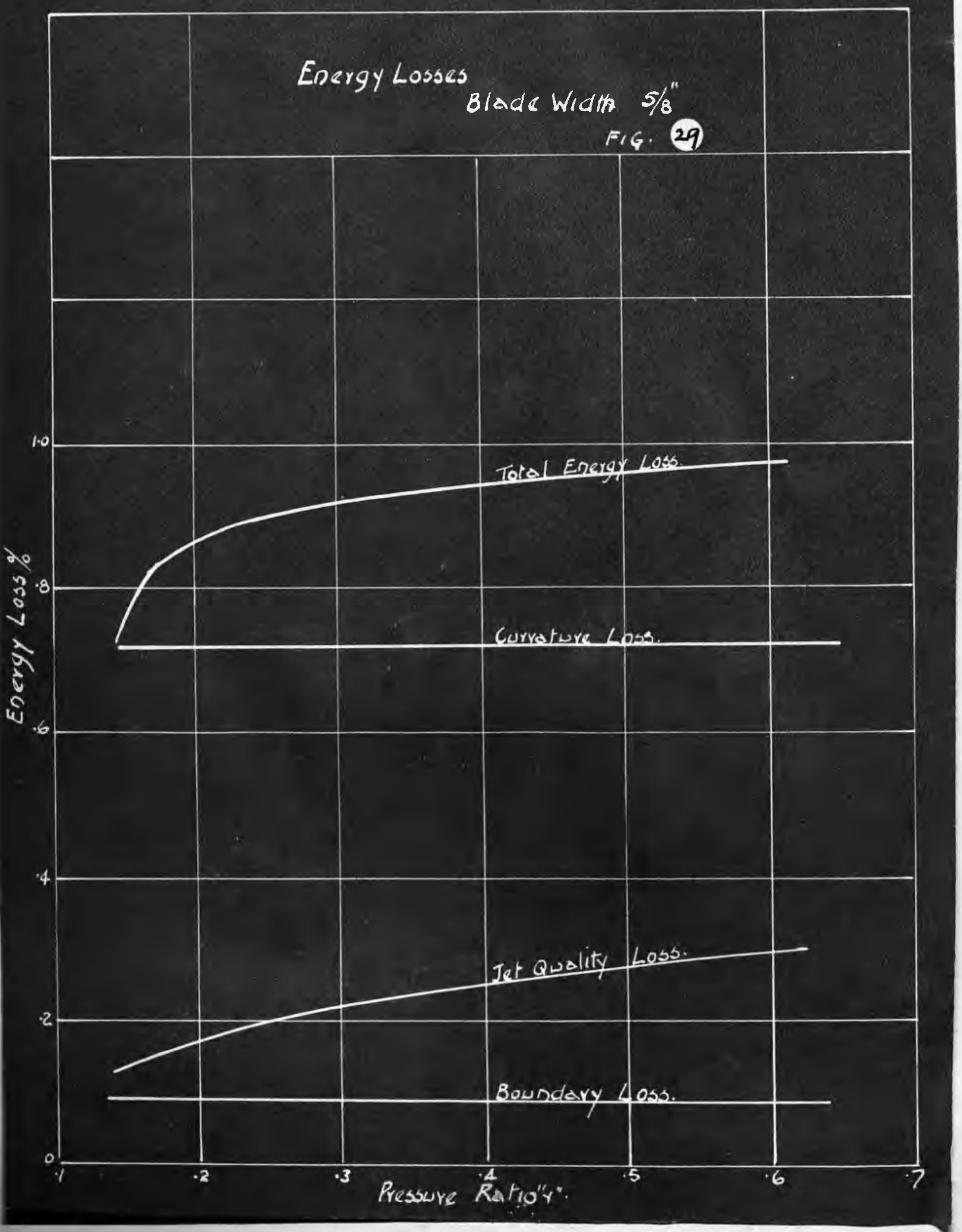


Energy Losses.
Blade Width 3/4"
FIG. 28
Total Energy Loss.
Curvature Loss.
Jet Quality Loss
Boundary Loss.
Energy Loss %
1.0
.8
.6
.4
.2
0
.1
.2
.3
.4
.5
.6
.7
Pressure Ratio "r"

STATIONARY BLADE APPARATUS.

TESTS Table 1a.

Blade Width ins	Inlet Press lbs/□" Abs.	Inlet Temp °F Abs.	Blade outlet Angle Degrees	Condensate lbs/sec.	Jet Impulse lbs	Y Press. Ratio	Nozzle velocity Coeff.	Theoretical Jet Speed ft/sec.	Actual Jet Speed ft/sec.	Corrected Jet Impulse lbs	Jet Impulse lbs/lb Steam/sec.	Corrected Jet Impulse lb/lb Steam/sec.	Horizontal Change of Speed V_w	Velocity Coefficient b	"b" Corrected	$(1 - b^2)$
	22.58	812		.0166	.57	.646	.543	1353.6	735	.57	34.3	33	1062	.6	.594	.648
	27.08	825		.018	.88	.538	.665	1596	1060	.88	49	50	1610	.679	.596	.644
	32.58	839	34.	.022	1.35	.448	.762	1835	1396	1.35	61.4	64.3	2070	.626	.6	.64
1"	37.88	846		.023	1.82	.388	.822	1980	1625	1.82	79.4	74.2	2390	.615	.614	.623
	42.88	851		.0284	2.35	.34	.861	2100	1808	2.36	83	83	2674	.615	.62	.616
	47.78	855		.0337	2.81	.305	.882	2195	1934	2.86	85	88.7	2840	.616	.628	.605
	52.68	856		.0355	3.28	.2761	.908	2271	2045	3.40	95.6	95.6	3060	.646	.64	.59
	57.68	855		.0385	3.94	.2521	.92	2326	2140	3.95	102.5	99.0	3167	.626	.65	.576
	63.08	853		.0425	4.31	.2308	.93	2360	2195	4.60	108.1	103.9	3320	.665	.665	.556
	66.0	833		.0465	4.87	.2208	.938	2380	2230	4.87	104.8	104.8	3355	.644	.67	.550
	72.61	845		.050	5.53	.2008	.94	2480	2326	5.62	112.5	111.0	3550	.684	.689	.525
	77.61	847		.0543	6.18	.1875	.948	2485	2359	6.18	114	114	3650	.71	.72	.48
	88.61	841		.0622	7.3	.1644	.95	2560	2430	7.4	118.8	120	3840	.749	.74	.451
	94.1	841		.0658	7.87	.155	.95	2600	2469	8.10	123	123	3940	.774	.774	.4
	98.61	842		.0692	8.44	.1478	.948	2610	2472	8.55	123.5	126	4030	.810	.81	.341
	103.61	841		.074	9.10	.1405	.944	2625	2479	9.35	126.2	128.0	4100	.836	.836	.299

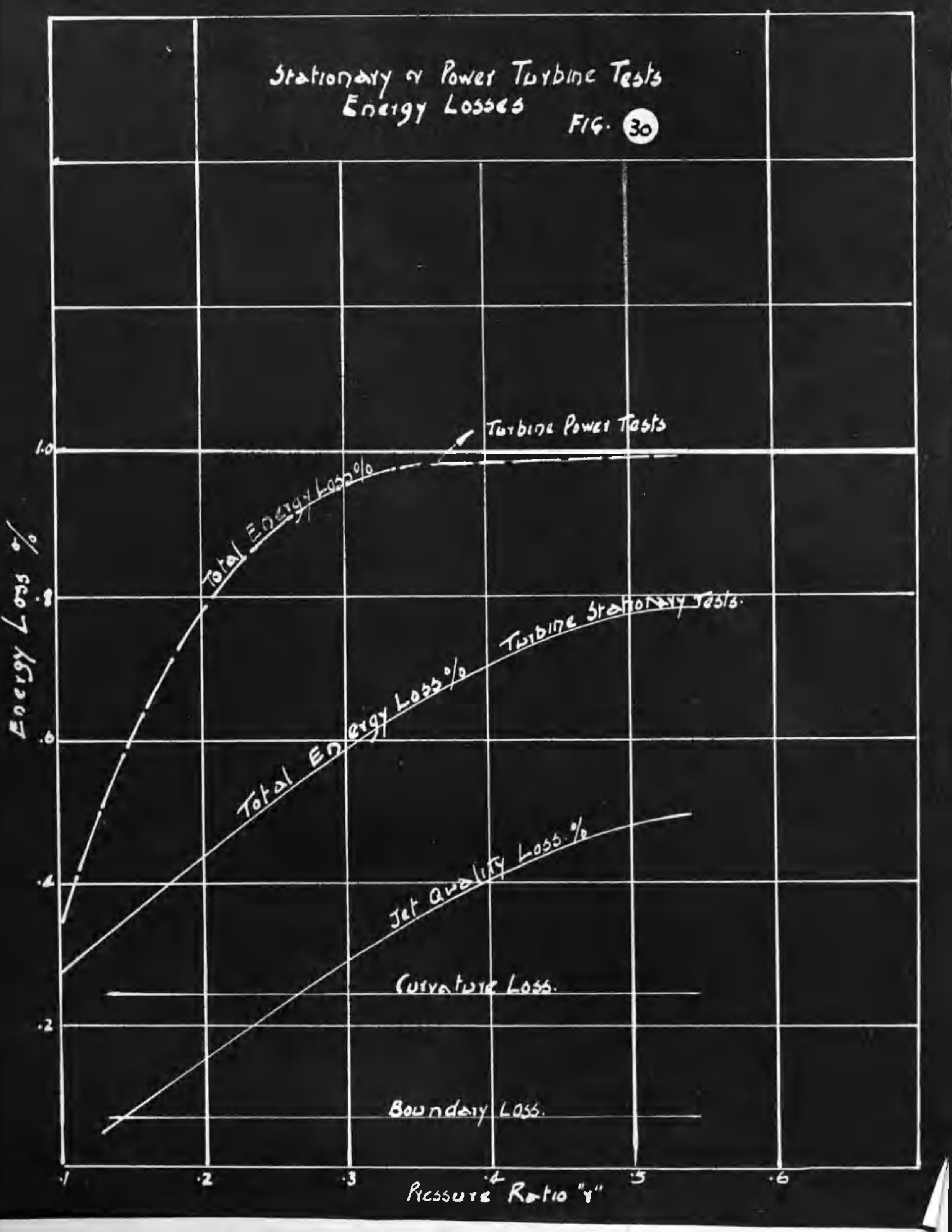


Energy Losses
Blade Width 5/8"
FIG. 29
Total Energy Loss.
Curvature Loss.
Jet Quality Loss.
Boundary Loss.
Energy Loss %
1·0
·8
·6
·4
·2
0
·1
·2
·3
·4
·5
·6
·7
Pressure Ratio "r".

STATIONARY BLADE APPARATUS.

TESTS Table 1b.

Blade Width Ins	Inlet Press lbs/□" Abs.	Inlet Tempr °F Abs	Blade outlet Angle °	Condensate lbs/sec.	Jet Impulse lbs	Press Ratio	Nozzle Velocity Coeff.	Theoretical Jet speed ft/sec.	Actual Jet speed ft/sec.	Corrected Jet Impulse lb	Jet Impulse lbs/lb steam/sec.	corrected Jet Impulse lbs/lb steam/sec	Horizontal Change of speed Uw	velocity Coefficient b	b Corrected	$(1-b^2)$
	22.58	792		.0162	..45	.646	.543	1340	727	.45	27.79	27.79	888	.3025	.425	.819
	32.58	799		.0226	1.32	.448	.762	1790	1361	1.32	58.5	58.0	1855	.426	.428	.816
7/8	42.88	803	24	.0294	2.28	.34	.861	2045	1760	2.30	73.4	75.1	2400	.443	.445	.802
	47.78	806		.0354	2.86	.305	.887	2139	1890	2.86	81.0	81.0	2590	.455	.458	.79
	52.68	806		.0387	3.18	.2761	.908	2210	2008	3.30	85.4	86.1	2760	.459	.47	.779
	57.68	811		.0416	3.83	.2521	.92	2275	2090	3.80	91.3	91.3	2920	.489	.48	.77
	63.08	819		.0422	4.22	.2308	.93	2325	2160	4.35	103	95	3040	.496	.496	.754
	67.98	823		.0482	4.78	.2145	.94	2370	2228	4.85	100.5	100	3200	.524	.516	.734
	72.58	821		.0507	5.54	.2005	.94	2410	2265	5.50	108.4	103	3295	.541	.526	.723
	77.58	826		.0534	6.10	.1875	.95	2450	2328	5.94	111.1	107	3320	.518	.54	.708
	82.08	828		.0593	6.65	.1775	.95	2490	2365	6.4	107.9	109.2	3499	.566	.565	.681
	88.58	829		.0615	7.21	.1641	.948	2530	2400	7.1	115.3	113.9	3640	.61	.61	.628
	93.98	831		.0665	7.7	.1551	.945	2559	2418	7.85	118	117.9	3770	.664	.664	.56
	103.58	822		.074	9.0	.1405	.943	2600	2448	9.10	123	122	3900	.696	.696	.514
	113.78	871		.08	10.2	.127	.94	2670	2510	10.6	132.6	132.6	4240			
	126.18	877		.09	11.6	.115	.937	2726	2553	11.8	131	135.6	4330			


Stationary & Power Turbine Tests
Energy Losses
FIG. 30
Turbine Power Tests
Total Energy Loss %
Total Energy Loss % Turbine Stationary Tests.
Jet Quality Loss. %
Curvature Loss.
Boundary Loss.
Energy Loss %
1.0
.8
.6
.4
.2
.1
.2
.3
.4
.5
.6
Pressure Ratio "r"

STATIONARY BLADE APPARATUS.

TESTS TABLE 1_c.

Blade Width ins	Inlet Press lbs/□" Abs	Inlet Temp °F Abs	Blade Outlet Angle Degrees	Condensate lbs/sec.	Jet Impulse lbs.	Pressure Ratio "r"	Nozzle Velocity Coeff.	Theoretical Jet Speed ft/sec.	Actual Jet Speed ft/sec.	Corrected Jet Impulse lbs	Jet Impulse lbs/lb steam/sec.	Corrected Jet Impulse lbs/lb steam/sec.	Horizontal change of Velocity Uw	Velocity Coefficient b	b Corrected	$(1-b^2)$
	22.58	771		.0159	.43	.646	.543	1330	723	.43	20.1	20	644			
3/4	32.58	781		.0223	1.21	.448	.762	1775	1351	1.21	48.4	51	1640	.265	.326	.894
	42.88	790	16	.0314	2.1	.34	.861	2030	1749	2.1	67	69	2220	.321	.345	.881
	47.78	793		.0334	2.6	.305	.887	2119	1880	2.6	72.8	76	2448	.361	.36	.87
	52.68	786		.0378	3.088	.2761	.908	2185	1980	3.09	81.8	81.8	2635	.383	.38	.856
	57.68	786		.0408	3.75	.2521	.92	2248	2070	3.6	88.9	85.8	2760	.387	.384	.852
	63.08	787		.0455	4.12	.2308	.93	2295	2134	4.13	90.8	90	2899	.414	.4	.84
	67.98	791		.050	4.68	.2145	.94	2350	2210	4.65	93	93.5	3010	.416	.416	.827
	72.58	793		.052	5.449	.2005	.941	2390	2250	5.2	100	96.9	3120	.445	.426	.818
	77.58	797		.0549	6.00	.1875	.950	2430	2310	5.7	95.2	100	3220	.45	.45	.797
	83.08	801		.0589	6.55	.1775	.950	2467	2348	6.15	104.1	103	3320	.471	.471	.778
	88.58	805		.0651	7.13	.1641	.948	2508	2378	6.9	106	106	3415	.502	.523	.726
	93.98	814		.0696	7.7	.1551	.945	2530	2390	7.7	110.5	108	3480	.515	.557	.69
	103.58	814		.0803	8.9	.1405	.943	2580	2430	8.9	111	113	3640	.56	.56	.686

STATIONARY BLADE APPARATUS.

TESTS TABLE 1d.

Blade width Ins	Inlet Press lbs/□" Abs.,	Inlet Temp °F Abs.	Blade outlet Angle.	Condensate lbs/sec.	Jet Impulse lbs	Y Press. Ratio,	Nozzle Velocity Coefficient	Theoretical Jet Speed ft/sec.	Actual Jet Speed ft/sec.	Corrected Jet Impulse lbs	Jet Impulse lbs/lb Steam /sec.	Corrected Jet Impulse lbs/lb Steam /sec.	Horizontal change of speed U_w	Velocity Coefficient b	b Corrected	$1 - b^2$
	22.58	771		.0159	.35	.646	.543	1330	723	.35	16.35	13	418			
	32.58	781		.0250	1.10	.448	.762	1775	1351	1.10	44.	45	1450	.124	.24	.94
5/8	42.88	789	16	.0313	2.0	.34	.861	2030	1749	2.0	63.9	64	2060	.229	.272	.926
	47.78	791		.0335	2.46	.305	.887	2119	1880	2.46	73.5	71.9	2317	.287	.287	.9175
	52.68	785		.0378	3.00	.2761	.908	2185	1980	3.0	79.5	76.0	2445	.288	.305	.907
	57.68	786		.0408	3.658	.2521	.92	2248	2070	3.5	85.8	81.0	2608	.314	.32	.898
	63.00	787		.0455	4.04	.2308	.93	2295	2134	4.02	88.5	86.5	2782	.356	.336	.887
	67.98	789		.050	4.59	.2145	.94	2350	2210	4.55	91.0	89	2861	.354	.354	.874
	72.58	792		.053	5.35	.2005	.94	2390	2250	5.03	95.0	92	2960	.373	.365	.866
	77.58	797		.0549	5.9	.1875	.95	2430	2310	5.5	100	97	3120	.407	.396	.843
	83.08	805		.0588	6.46	.1775	.95	2467	2348	6.0	101.9	99	3187	.417	.415	.828
	88.58	804		.0651	7.04	.1641	.948	2508	2378	6.8	104.2	103	3320	.422	.437	.809
	93.98	814		.0690	7.6	.1551	.945	2530	2390	7.5	108.5	105	3380	.474	.484	.765
	103.58	815		.0803	8.74	.1405	.943	2580	2430	8.6	107.0	110	3540	.518	.518	.731

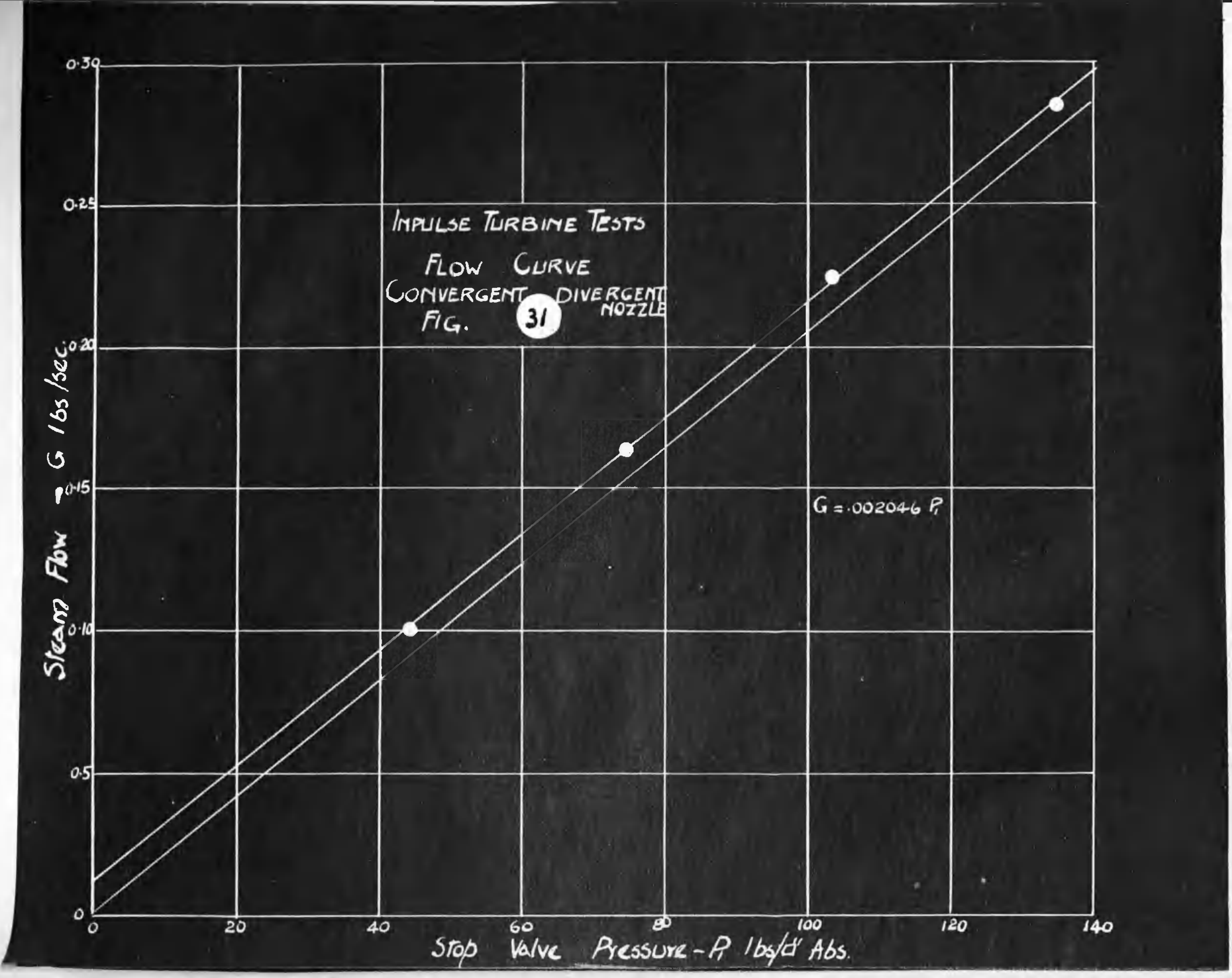
TABLE II.

From Velocity Coefficient Curves.

Blade Width Ins.	r Pressure Ratio	b_0	b	$1-b_0^2$	$1-b^2$	Friction Coefficient C_1	Curvature coefficient C_2	Quality Coefficient C_0	Nett Curvature $\frac{\theta}{R}$	Mean Curvature Length L	$\int_0^L \frac{P}{A}\,dL$	$C_1\int_0^L \frac{P}{A}\,dL$	$f(Q)$	$f(Q)$ Corrected	Ω	Ω_1	$\frac{\theta}{R}L$	$C_2\left(\frac{\theta}{R}L\right)^{.055}$
1"	.2	.79	.69	.38	.5	.005	.1134	.396	2.00	1.35	12.8	.064	.14	.17	1.055	.48	2.7	.316
	.25		.65		.56								.20	.202				
	.3		.63		.595								.235	.225				
	.35		.618		.60								.24	.24				
	.4		.607		.615								.255	.255				
	.45		.6		.635								.275	.265				
$\frac{7}{8}$	.2	.695	.525	.516	.72	.005	.1134	.396	3.04	1.2	13.2	.066	.204	.17		.48	3.65	.450
	.25		.49		.765								.249	.202				
	.3		.46		.79								.274	.225				
	.35		.445		.8								.284	.24				
	.4		.435		.81								.294	.255				
	.45		.429		.81								.294	.265				
$\frac{3}{4}$	.2	.56	.433	.686	.81	.005	.1134	.396	4.78	.91	15.6	.078	.124	.17		.48	4.35	.608
	.25		.385		.85								.164	.202				
	.3		.36		.865								.179	.225				
	.35		.342		.876								.190	.24				
	.4		.334		.88								.190	.255				
	.45		.328		.89								.204	.265				
$\frac{5}{8}$	.2	.48	.371	.770	.87	.005	.1134	.396	7.07	.86	17.5	.0875	.135	.17		.48	6.08	.720
	.25		.321		.895								.160	.202				
	.3		.29		.915								.180	.225				
	.35		.27[illegible]		.935								.200	.24				
	.4		.251		.93								.205	.255				
	.45		.24		.94								.205	.265				

APPENDIX.

ESTIMATION OF NOZZLE EFFICIENCY AND JET SPEED.



IMPULSE TURBINE TESTS
FLOW CURVE
CONVERGENT DIVERGENT NOZZLE
FIG. 31
G = ·002046 P
Steam Flow - G lbs/sec.
Stop Valve Pressure - P lbs/□" Abs.
0·30
0·25
0·20
0·15
0·10
0·5
0
0
20
40
60
80
100
120
140

STEAM FLOW.

The flow quantities as determined from the observations made in the Power Tests are shown plotted against stop valve pressures in fig. (31).

It will be seen that the plotted points lie very closely on the ~~dotted~~ Straight line but as this does not pass through the origin, these values are inadmissible. The line through the origin and parallel to the dotted line gives flow quantities about .012 lbs per sec lower than the observed flows from which it would appear that there has been a continuous leakage of steam which has passed from the steam pipe to the condenser without passing through the nozzles. As there is a half inch pipe passing from the boiler side of the stop valve to the condenser, leakage may easily have passed through the valve on this pipe. The full line in the figure may accordingly be taken as giving the quantity of steam actually passing through the nozzles for any stop valve pressure. This curve is valid only when the pressure in the casing of the Turbine is less than the critical pressure, i.e. about .6 of the stop valve pressure and this limit is not exceeded in any of the present tests.

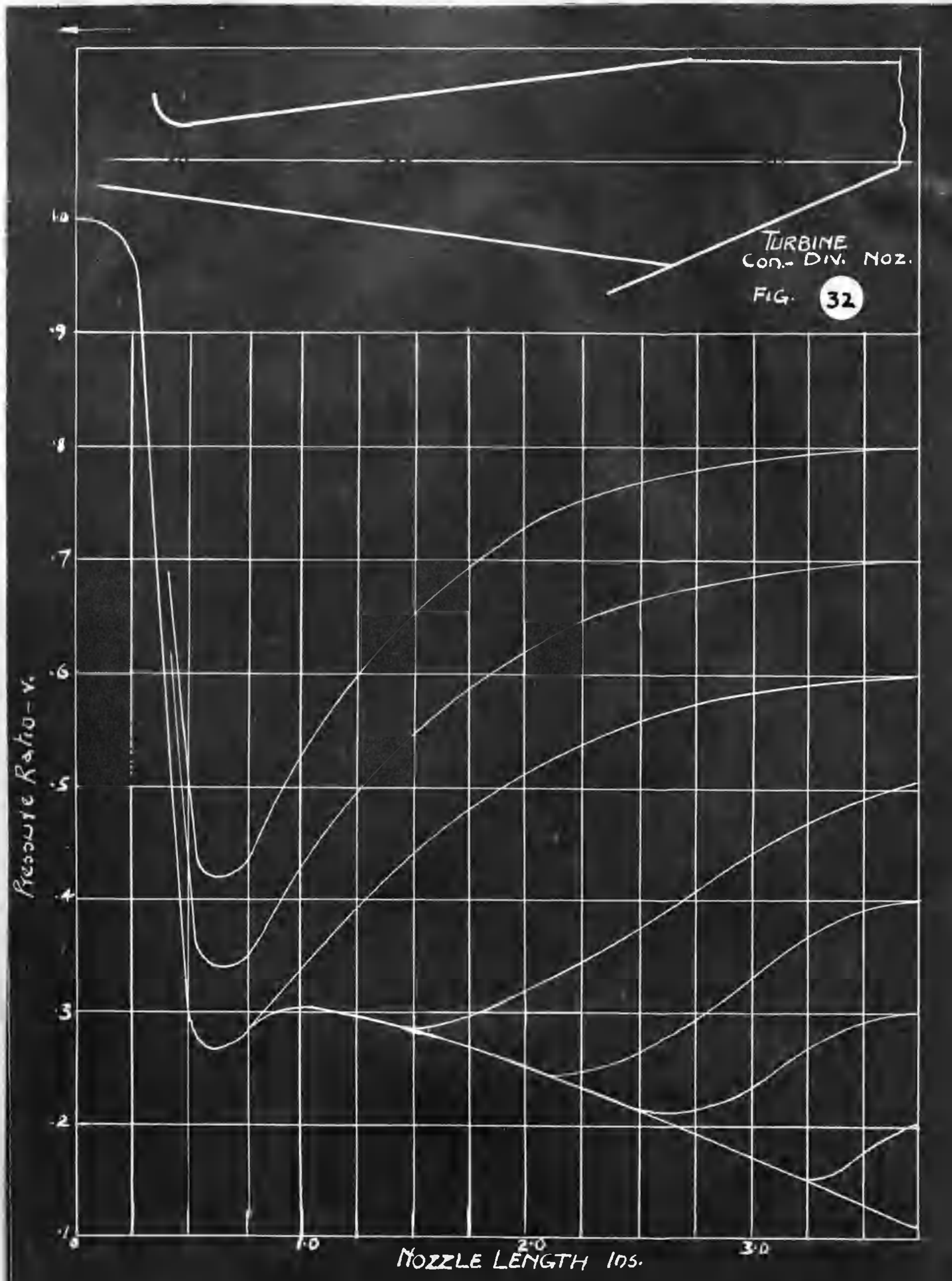
STEAM CONDITION DURING EXPANSION.

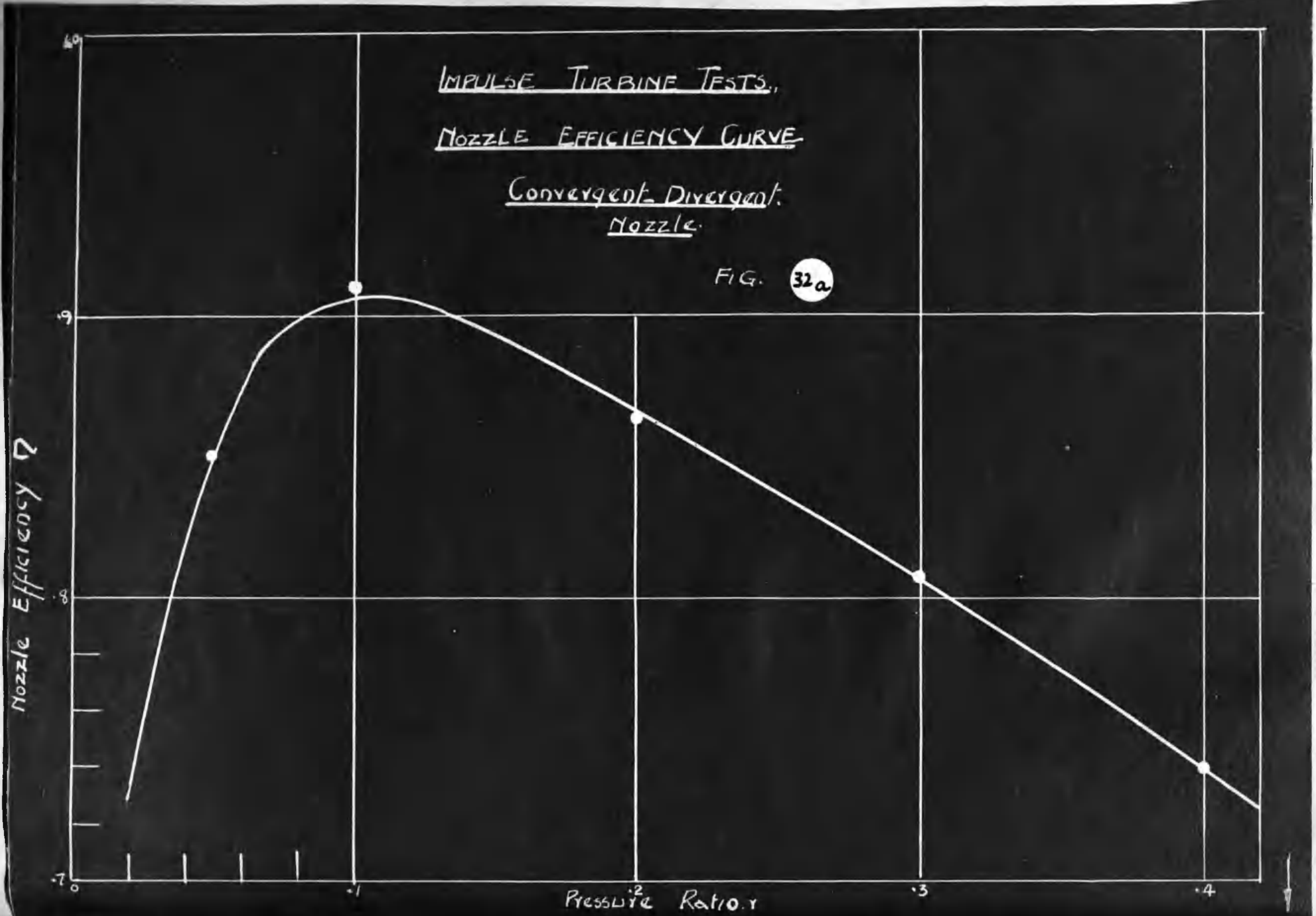
For the determination of the nozzle efficiency it is necessary to know whether or not the steam remains in the dry state during expansion, i.e. whether we are having supersaturated or wet steam.

For supersaturated expansion the adiabatic law is :-

$$P(V-.01602)^{1.3} = \text{constant}$$

and the critical ratio is .55. For the given divergent nozzle


TURBINE
CON.- DIV. NOZ.
FIG. 32
1.0
.9
.8
.7
.6
.5
.4
.3
.2
.1
0
1.0
2.0
3.0
NOZZLE LENGTH INS.
PRESSURE RATIO - r.



IMPULSE TURBINE TESTS.
NOZZLE EFFICIENCY CURVE.
Convergent Divergent.
Nozzle.
FIG. 32a
Nozzle Efficiency η
1·0
·9
·8
·7
0
·1
·2
·3
·4
Pressure Ratio r

operating with a supply pressure of 100 lbs. per sq. in. abs., the theoretical flow is .219 lbs. per sec. under superheated conditions, but with expansion in the wet field the critical ratio is .577 and the theoretical flow is .207 lbs per sec, this being only very little greater than the flow quantity, from the curve i.e. .205 lbs/sec. Since the actual expansion must be accompanied by losses which reduce the flow quantity to a value less than the theoretical, we are justified in concluding that the expansion up to the nozzle throat is within the supersaturated field. It is not certain that this steam condition will persist right through to the blade outlet but we will assume that this is so.

ESTIMATION OF EFFICIENCY.

The nozzle efficiency has not been directly determined, so has to be estimated by evaluating the losses due to :-

1. Boundary loss.
2. Convergence loss.
3. Recompression loss.

For the boundary loss we have :-

$$k_s = c_1 \int_0^l (1 - r^\alpha) \frac{p}{A} \, dx \qquad \text{..............(1)}$$

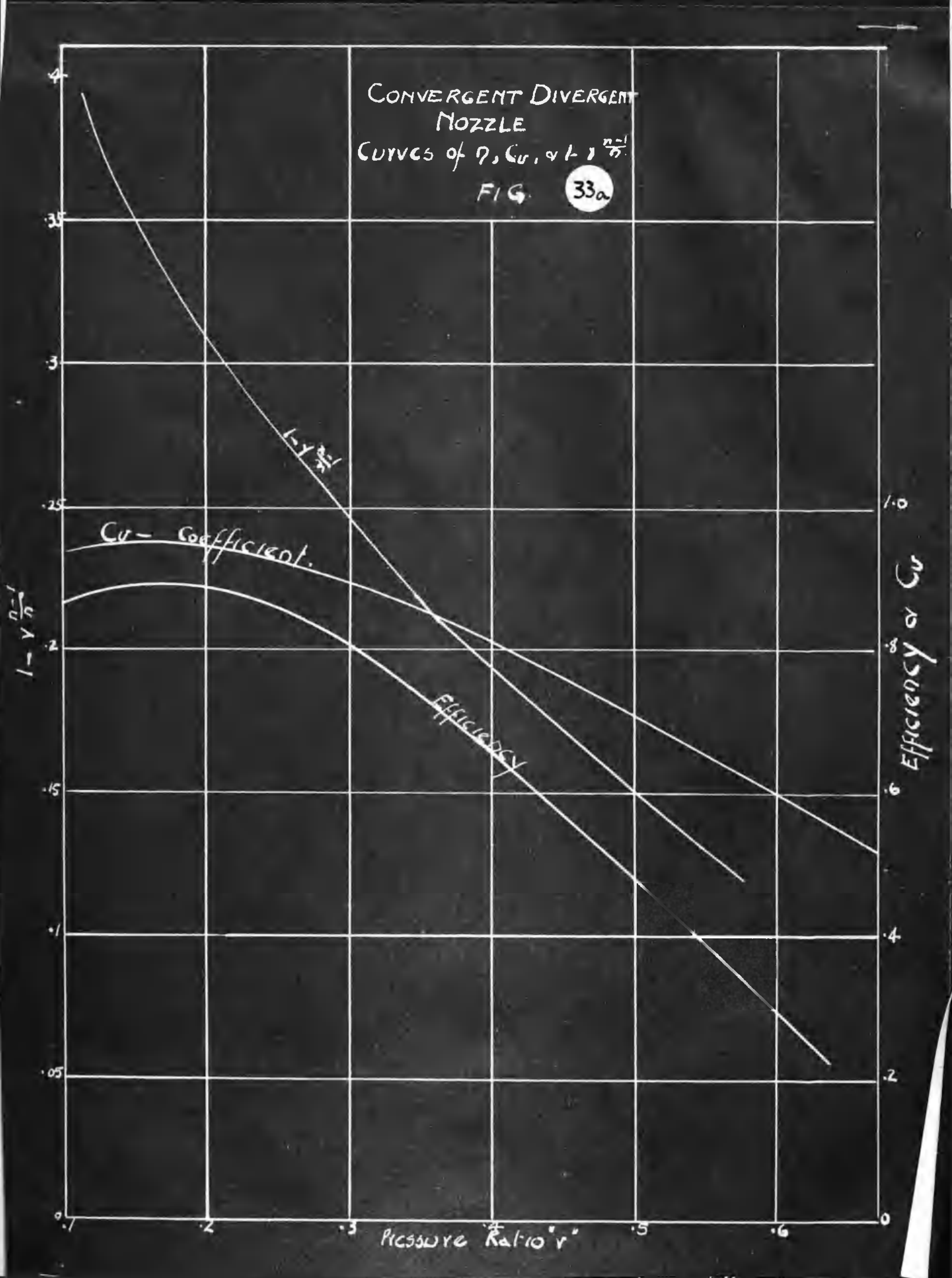
or very nearly

$$k_s = c_1 \int_0^l (1 - r^\alpha) \frac{p}{A} \, dx \qquad \text{..............(1)a}$$

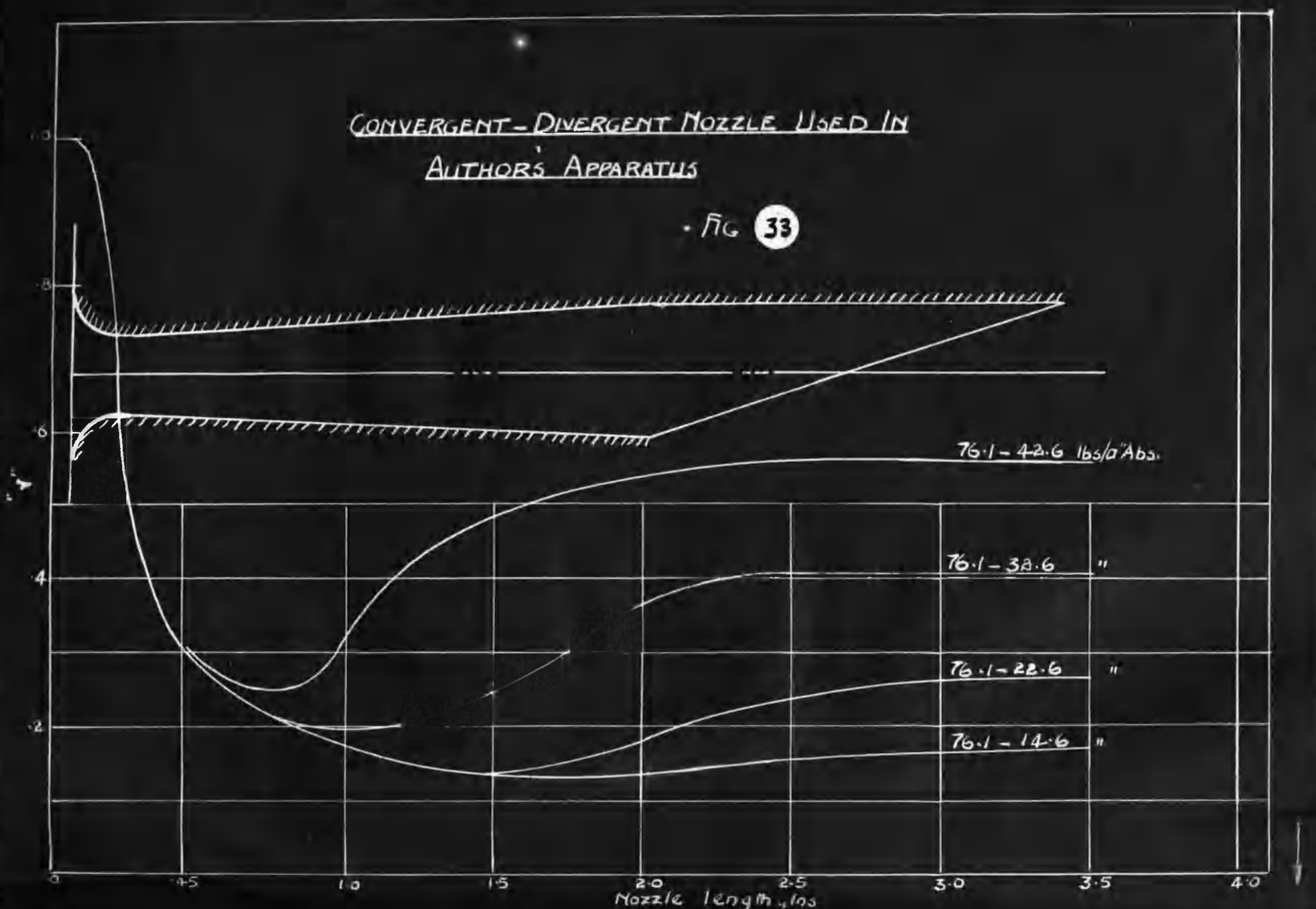
where :-

c_1 = Friction coefficient

l - distance from nozzle inlet to the point at which recompression commences.



CONVERGENT DIVERGENT NOZZLE
Curves of η, Cv, & 1-r^(n-1)/n
FIG. 33a
1-r^(n-1)/n
Cv - Coefficient.
Efficiency
·4
·35
·3
·25
·2
·15
·1
·05
0
1·0
·8
·6
·4
·2
0
·1
·2
·3
·4
·5
·6
Pressure Ratio "r"
Efficiency & Cv



CONVERGENT – DIVERGENT NOZZLE USED IN AUTHORS' APPARATUS

FIG 33

r = Pressure Ratio = $\frac{\text{Pressure in Nozzle}}{\text{Pressure at Inlet.}}$

$\alpha = \frac{(n-1)}{n}$

n = adiabatic index = 1.3

p = nozzle perimeter

A = Area of cross section of nozzle.

dx = element of length of nozzle.

The convergenceloss is given by :-

$$k_c = c_2(p_1 v_1)^{\frac{1}{2}}(1 - k_t - r_t^{\alpha})^{\frac{1}{2}} \qquad \text{.............(2)}$$

or approximately :-

$$k_c = c_2(p_1 v_1)^{\frac{1}{2}}(1 - r_t^{\alpha})^{\frac{1}{2}} \qquad \text{.............}(2)_a$$

where :-

c_2 = convergence loss coefficient

p_1 = initial pressure ....lbs. per sq.ins.abs.

v_1 = initial specific volume ....cub.ft.per lb.

r_t = pressure ratio at throat

for the loss due to recompression

$$k_r = \cdot 82\, R\,(r_2^{\alpha} - r_1^{\alpha})^2$$

where :-

$R = \frac{\text{Area at Nozzle Outlet}}{\text{Area at Nozzle Throat.}}$

r_2 = pressure ratio at end of compression

r_1 = pressure ratio at beginning of compression.

The nozzle efficiency is given by

$$\eta = \frac{1 - k_o - r^{\alpha}}{1 - r^{\alpha}} \qquad \text{.............(3)}$$

where k_o = total loss factor = $k_s + k_c + k_r$

r = pressure ratio at nozzle outlet.

For the evaluation of these expressions a series of pressure ratio curves for the nozzles are necessary. These curves along with the efficiency curves are shown in fig. (32) (32)a, and (33) (33)a, for the turbine nozzle and the Stationary Blade Apparatus' nozzle respectively. The efficiency estimated in the above manner takes account only of the losses which the fluid incurs up to the nozzle outlet, any further loss in the gap being therefore neglected, but the error on this account is not likely to be appreciable since these nozzles do not give a sudden pressure change at the outlet except for pressure ratios lower than .1 in the turbine nozzle and .15 in the Stationary Blade Apparatus nozzle.

I beg to thank Professor A. L. Mellanby, D.Sc., for his interest and kind advice in the whole course of this work and also for providing all facilities and new apparatus for the progress of the work.

I also beg to thank Professor Wm. Kerr, Ph.D., for suggesting the above line of research and for the very valuable help and advice, so readily given during the subsequent analysis of the results.

A STUDY OF HEAT TRANSFER FROM "STEAM TO WATER" AND "WATER TO WATER" IN A HIGH SPEED EXPERIMENTAL CONDENSER.

BY

MOSTAFA R. YOUSSEF, B.Sc., A.R.T.C.

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FROM

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IN

A HIGH SPEED EXPERIMENTAL CONDENSER.

INTRODUCTION.

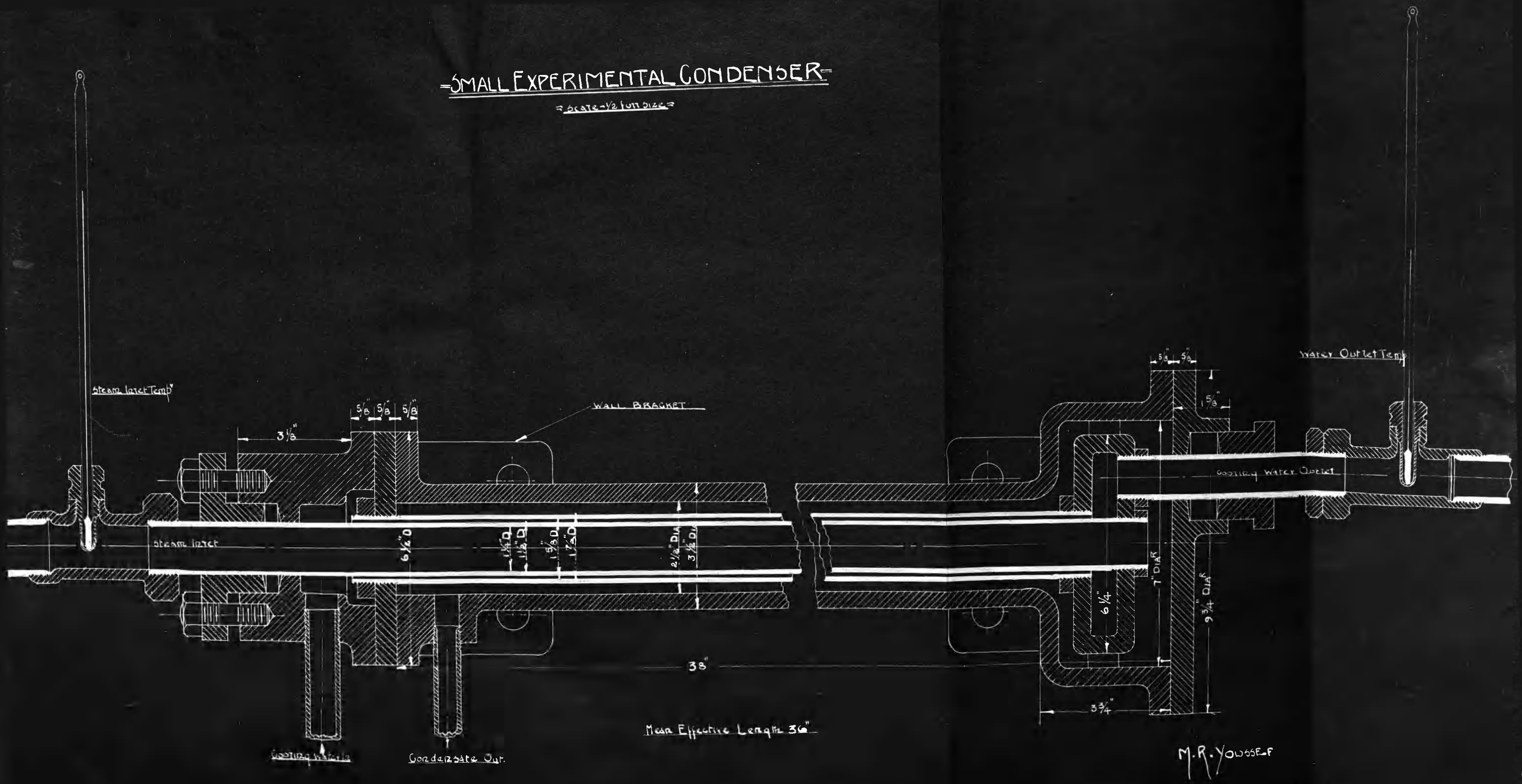
In spite of the very large amount that has been written on condensers, no complete rules have yet been given to serve as the basis of condenser design.

It is a very complicated matter, especially on the steam side of the condenser tubes, and it would not be possible from theory alone, to design a condenser.

Some very thorough experiments have been carried out a good many years ago by Dr. J.P.Joule. These experiments were carried out, under great accuracy, but unfortunately, one very important datum is lacking, which is necessary in order that the results may be analysed completely, namely the temperature of the tubes themselves.

It was in October, 1874, that Professor Osborne Reynolds brought forward his paper on "The Extent and Action of the Heating Surface of Steam Boilers". In his paper he endeavoured to deduce from theoretical considerations the laws of Heat Transmission and later verified the laws by experiment.

Many other investigations took place after that, namely by Drs. Stanton, Nicholson, etc. One of the most inter: :esting experiments was that carried out by Mr. Gordon C. Webster at the Royal Technical College and read to the Inst. of Engineers and Shipbuilders in Scotland in 1913. Great care and thoroughness were experienced and in addition the experi: :ments contain some data regarding the tube temperatures, which were determined by using thermocouples.



SMALL EXPERIMENTAL CONDENSER
Scale - 1/2 full size
Steam Inlet Temp
WALL BRACKET
Water Outlet Temp
Steam Inlet
Cooling Water Outlet
3 1/8"
5/8" 5/8" 5/8"
6 1/2" D
1 1/4" D
1 1/2" D
1 5/8" D
1 7/8" D
2 1/2" DIA
3 1/8" DIA
38"
6 1/4"
7" DIA
9 3/4" DIA
1 5/8"
3 3/4"
Mean Effective Length 36"
Cooling Water In
Condensate Out.
M.R. Youssef

With these data it was quite possible to examine the steam and the water sides separately, and finally to combine the laws so obtained so as to examine the behaviour of the condenser.

Webster deduces a certain number of laws and formulae but he does not leave the subject in a form which is most convenient to a designer.

DESCRIPTION OF CONDENSER.

Fig. 1 shows a sectional view of the condenser. Steam in "Steam to Water" or "Hot water in Water-Water" tests respec::tively flows through the middle pipe $1\frac{1}{4}$" D to the right hand end of the tube, it then changes its path backwards and comes out at the condensate outlet. Another tube of $1\frac{5}{8}$" D is circum::ferential with the first tube, both tubes are of $\frac{1}{8}$" thickness. The circulating water from the mains takes the annular space between the two tubes for its path.

The following is a diagrammatic sketch of the flows in the condenser:-

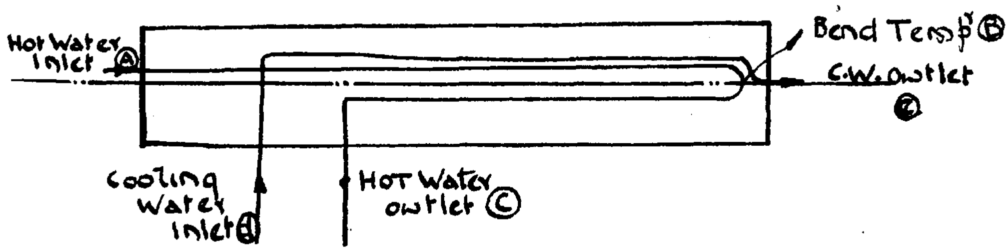


FIG II

CONDENSER DETAILS.

Length of water path = 3.10 ft.

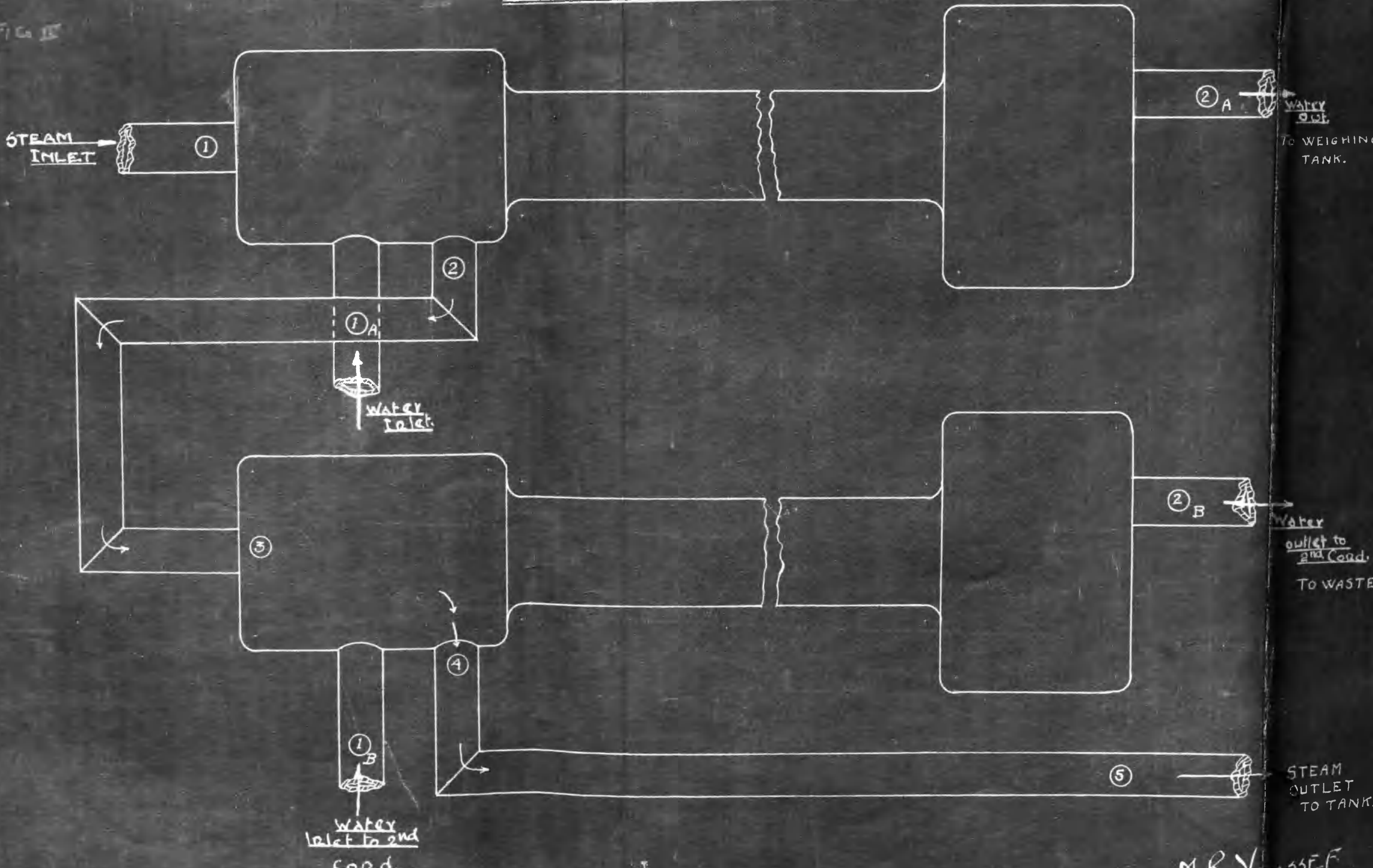
Sectional area of water flow = .0021 sq. ft.

Cooling surface of tubes in contact with steam = 2.5 sq. ft.

SMALL EXPERIMENTAL CONDENSER

ARRANGEMENT OF APPARATUS

FIG II



M.R. YOUSSEF

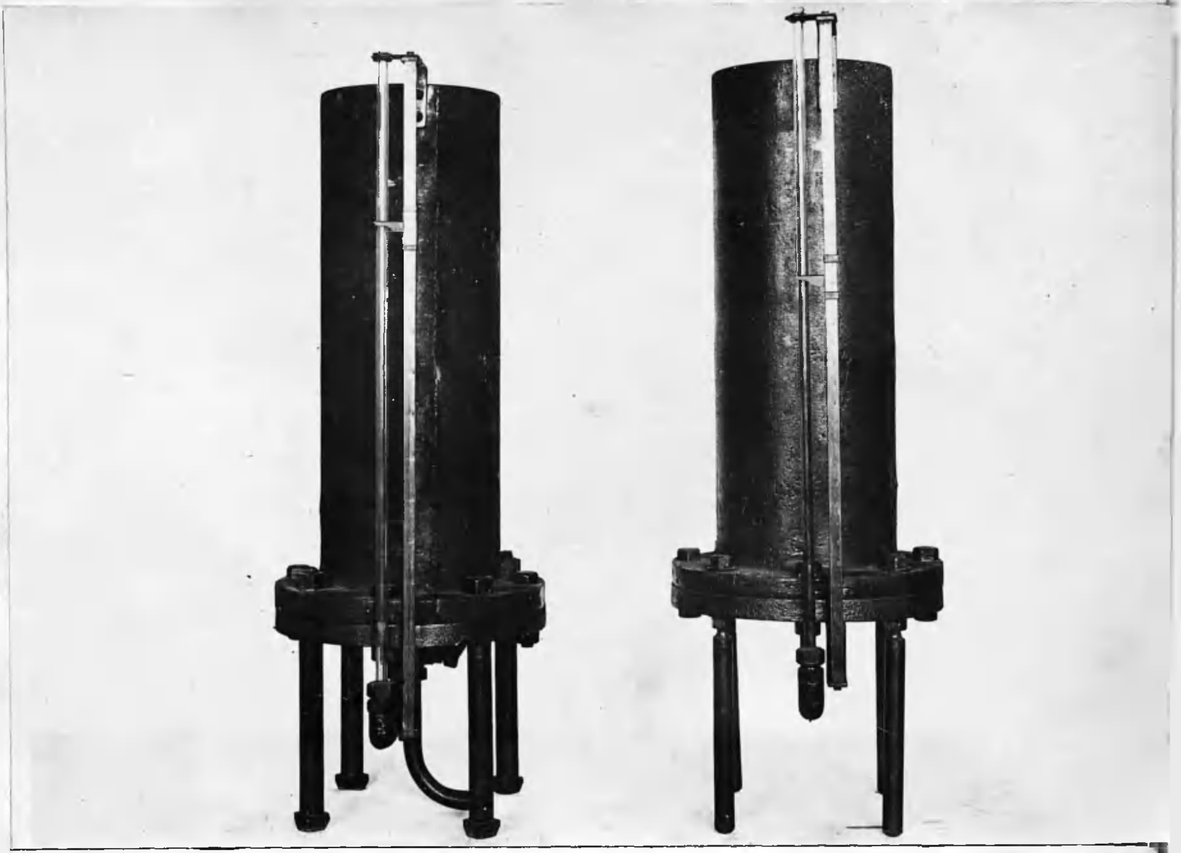
It was necessary in the case of the Water-Water tests to have a thermometer pocket at the far off flange of the condenser so as to get the temperature at the bend indicated in the sketch by (B), and so treat the condenser as two separate condensers, one as a parallel flow and the other as a counter flow condenser.

"STEAM-WATER TESTS."

Experiments on Heat Transfer were carried out by the writer at the Royal Technical College on the suggestion of Professor A. L. Mellanby, D.Sc., first on "Steam-Water" and second, on "Water-Water". The plant for the "Steam-Water" tests consisted of two condensers as in Fig. 3. The steam was allowed from the mains by means of a valve to regulate the amount of steam flow. A pressure gauge was fixed at the inlet passage of steam, and a throttle valve was adjusted at the outlet of steam from the first condenser to increase or decrease the pressure of steam in the condenser as required. The steam then followed the passage indicated by the arrows from (2) to (3) when it entered the second condenser where it left by the passage indicated by arrows from (4) to (5) to an ordinary weighing tank. The main object of the second condenser was to condense the part of steam which passed uncondensed from the first condenser. The condensers were so connected that each had a separate supply of circulating water namely, (1) A and (1) B. The water left by (2) A and (2) B after passing through the respective condensers, and the water from the first conden: :ser which was used as the experimenting one, was collected in an orifice tank, while that from the second condenser was let out to waste.

TANKS.

The orifice tanks have four holes 3", 4", 5" and 6" diameter respectively. A sliding disc with a hole cut through



is adjusted to fit closely to the bottom of the tank. The disc could be turned round so as to get the desired orifice in use. The readings given on the measuring glass, alongside Fig. 4 give the head of water, and amount of cooling water could then be easily read from a graph of heads and amount of cooling water/minute for every orifice, and thence the speed of water could be calculated by:-

$$\frac{Q}{60} = a \times v \times 62.5.$$

where $Q =$ cooling water in lbs. per minute.

$a =$ sectional area of water flow sq. ft.

$v =$ speed of water flow, ft. per second.

The ordinary weighing tanks have a shut valve at the bottom, and condensed steam is read off on the measuring glass in lbs.

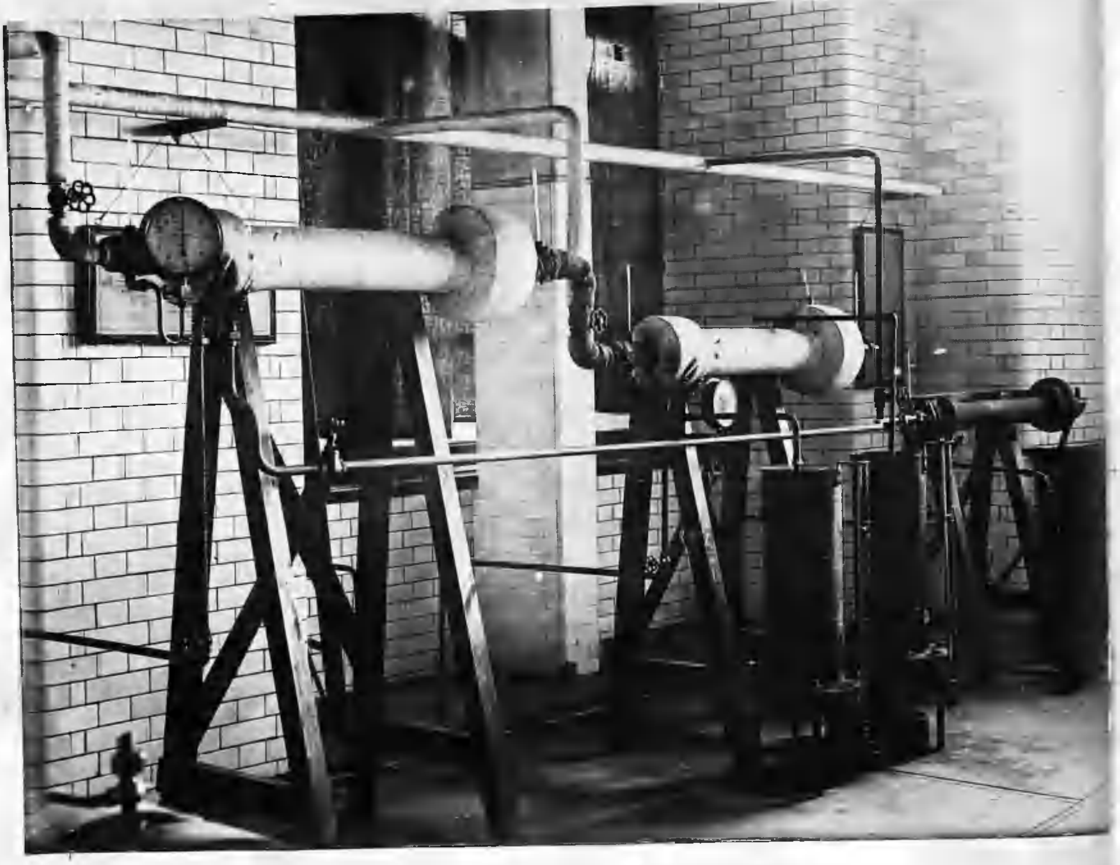
A photograph showing the bottom of the orifice tank with cover removed and also the bottom of the weighing tank is shown alongside Fig. 5.

The steam entering the first condenser was about 2 or 3% wet.

In these experiments the water speeds were varied and also the outlet water temperatures, the steam flow and pressure being kept constant throughout the tests. The pressure was only varied for every separate series of tests.

"WATER-WATER TESTS".

The plant was arranged for these tests as shown in photograph, Fig. 6. It consists of three condensers connected as shown, with three separate cooling waters. Steam is let through the first condenser by an inlet valve. The steam out::let from that condenser is led by a pipe to the third condenser the main object of which was to condense the steam passed by the first condenser. Hot water was let from the outlet of first condenser to inlet of second condenser. This hot water



was cooled down in the second condenser by a fresh supply of cooling water. Both hot and cold water were measured in orifice tanks, the speed of the former being kept constant at 4.25 ft./sec. right through, while the speed of the latter was taken at six steps of 1, 2, 3, 4, 5, and 6 feet per second. The inlet temperature of the hot water was decreased after every series of tests by opening the throttle valve fitted on the horizontal steam pipe leading to the third condenser.

All thermometer readings were corrected and also the tanks calibrated for accuracy.

The method adopted for tank calibration was to let the water flow at a certain speed into the orifice tank, and then remove the tank, and displace it by an ordinary weighing tank, and a calibration graph drawn.

THEORY OF HEAT TRANSMISSION.

The simplest form of heat transmission is that of heat transmission by conduction, in which the opposite sides of the metal of indefinitely large area are kept at two different constant temperatures, T and t. Heat then flows across the plate at a steady rate, and the amount being proportional to the temperature difference and inversely as the plate thickness, i.e.:-

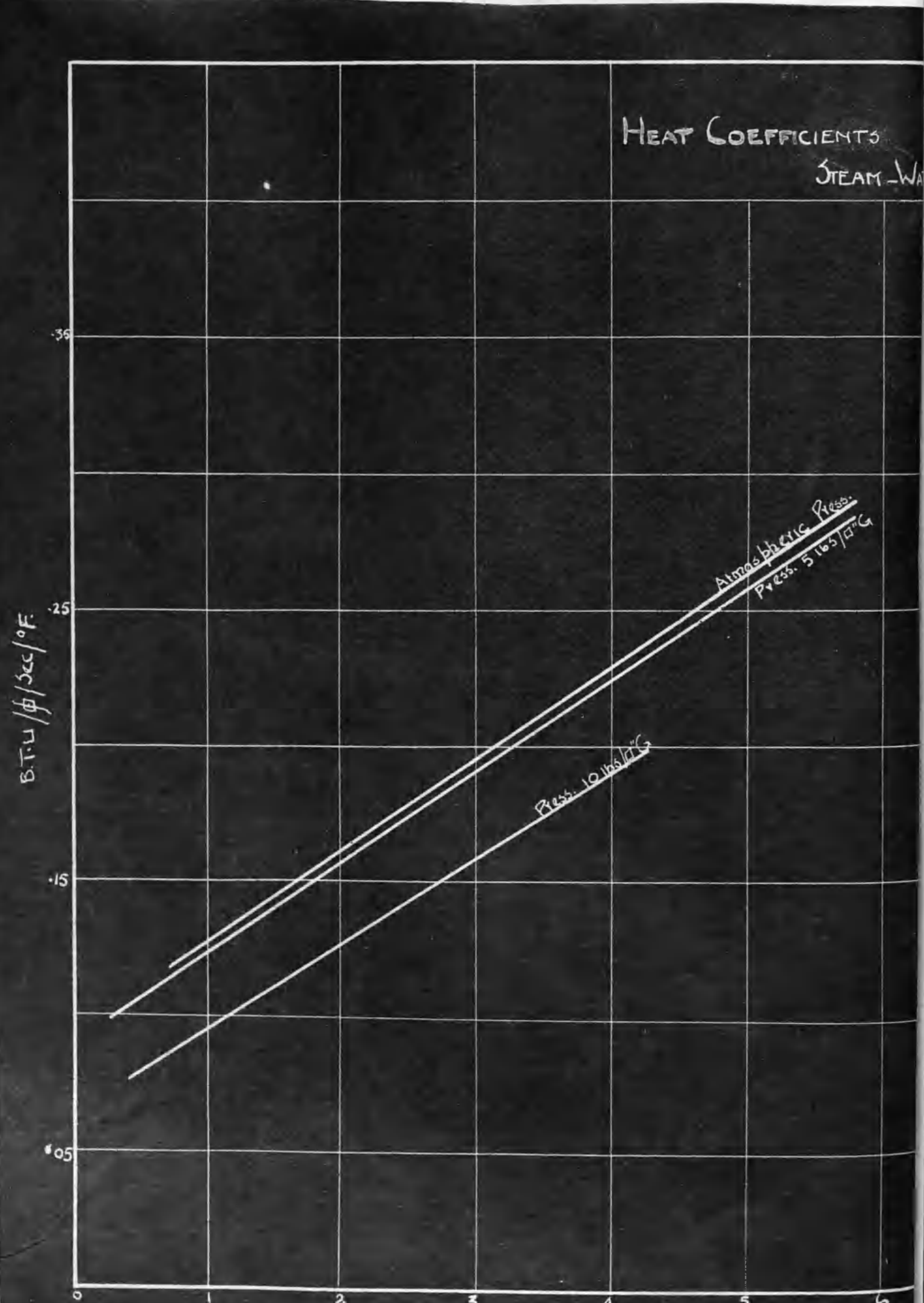
$$Q = K \frac{T-t}{d}$$

where T and t are the temperatures in degrees Fahrenheit of the two surfaces,

d is the thickness of the plate in inches and K is the thermal units passing per hour across a plate one inch thick when its sides are kept at a temperature difference of 1 degree Fahrenheit in one hour. Supposing for an example to illustrate this, we take,

$$\underline{T = 1500°F}\ ,\ \underline{t = 350°F}\ ,\ \underline{K = 450.}$$

Fi


HEAT COEFFICIENTS
STEAM_WA
Atmospheric Press.
Press. 5 lbs/□"G
Press. 10 lbs/□"G
.35
.25
.15
.05
B.T.U./□/sec/°F.
0
1
2
3
4
5
6
Cooling Water ft/sec

then we should have:-

$$Q = 450 \times \frac{1150}{5} = 1035 \times 10^3 \text{ B.T.U./sq ft/hr.}$$

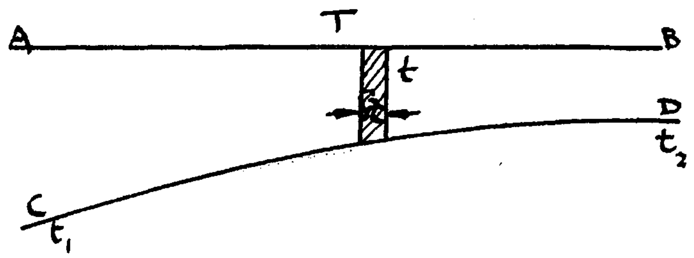
Considering the effect of high speed on conduction, we know that the more quickly gases move, or, the shorter the time they remain in contact with the plates, the greater the heat trans: :mitted.

It would indeed seem that the act of increasing the gas speed relatively to the heating surface, is equivalent to an almost proportional augmentation of the area of that surface

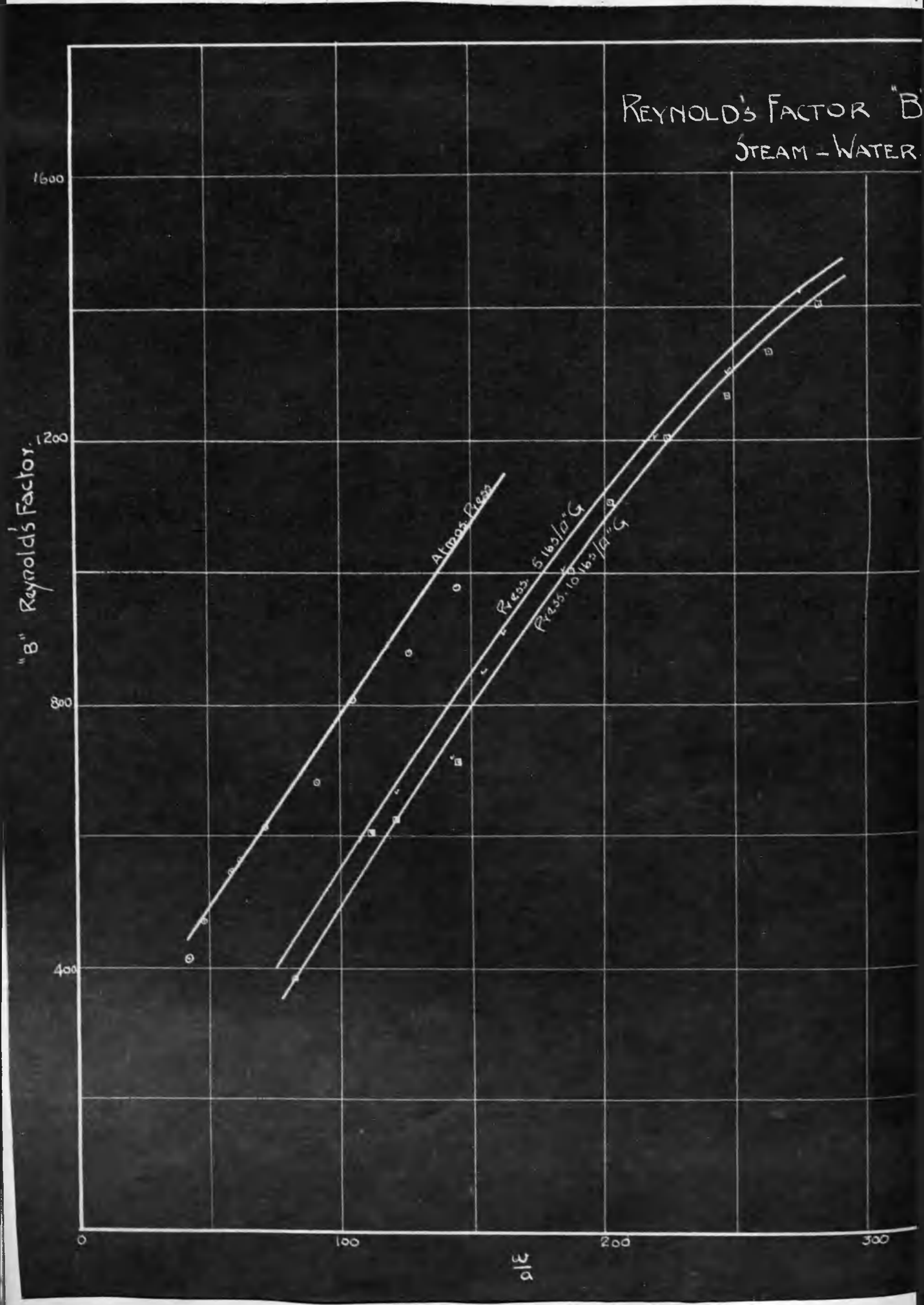
When one fluid extracts heat from another through a metal surface, it is customary to express the rate of heat transfer in B.T.U./Sec./Sq.ft/degree Fahrenheit temperature difference.

This temperature difference varies considerably and it is usually necessary to use a mean value which is never the arithmetic mean.

The question of finding this mean value will first be considered



Consider the condensation of steam in a tube or at any flat surface. With a continuous flow of steam the tempera: :ture on the steam side may be supposed constant at T and may be graphically represented by the line AB.


REYNOLD'S FACTOR "B"
STEAM - WATER.
"B" Reynold's Factor.
1600
1200
800
400
0
100
200
300
Atmos. Press
Press. 5 lbs/□" G
Press. 10 lbs/□" G

The water enters a temperature t_1 , and leaves at temperature t_2 , the form of the curve being given by CD

Let h = Rate of heat transmission B.T.U. per sec. per deg. Fah. diff.

w = Rate of water flow, lbs. per second.

dA = ~~Area of section~~ an element of surface on which the steam temperature is T and the water temp^r is t.

Then:-

$$w\,dt = +h(T-t)\,dA$$

$$\frac{dt}{T-t} = +\frac{h}{w}\,dA$$

$$\text{and} \quad \int_{t_1}^{t_2} \frac{dt}{T-t} = +\frac{h}{w}A \qquad \text{over whole area } A$$

$$\therefore \; -\log_e \frac{T-t_2}{T-t_1} = +\frac{h}{w}A$$

changing limits ie: $$+\log_e \frac{T-t_1}{T-t_2} = +\frac{hA}{w} \qquad (a)$$

Dealing with the tube as a whole,

$$w(t_2-t_1) = +hAt_m$$

where t_m = mean temperature difference. between condensing fluid and water

$$\therefore \; +\frac{hA}{w} = \frac{t_2-t_1}{t_m} \qquad (b)$$

from (a) & (b)

we have $$\log_e \frac{T-t_1}{T-t_2} = +\frac{t_2-t_1}{t_m}$$

and $$\therefore \; t_m = \frac{t_2-t_1}{\log_e \frac{T-t_1}{T-t_2}} \qquad (1)$$

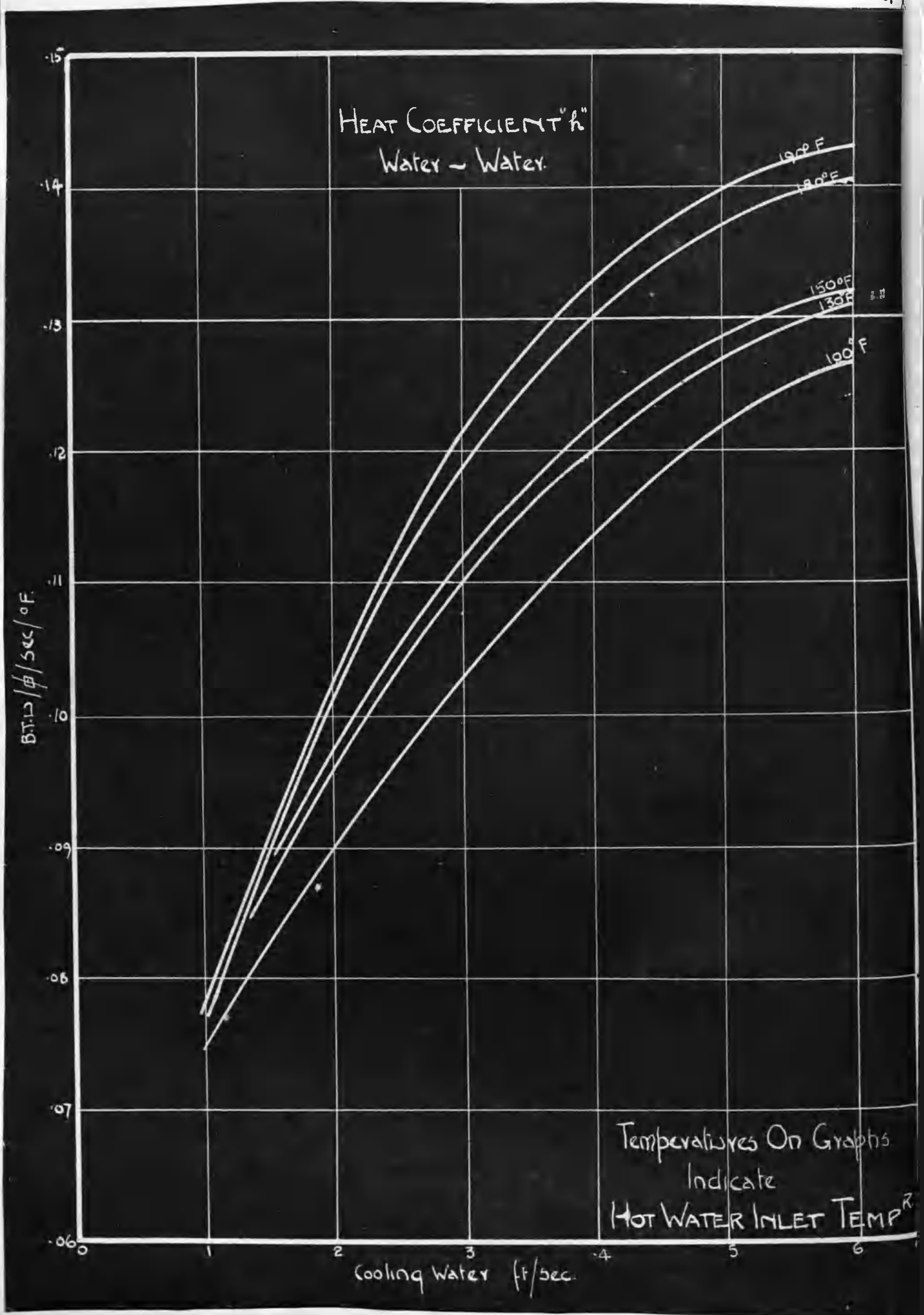
$$\cancel{\therefore \; t_m \log_e \frac{T-t_1}{T-t_2} = (t_2-t_1)}$$

Substituting in (b)

$$\therefore \; \frac{hA}{w} = \frac{t_2-t_1}{t_m}$$

$$\therefore \; h = \frac{w}{A} \times \log_e \frac{T-t_1}{T-t_2}$$

FIG. 1



Heat Coefficient "h"
Water – Water.
·15
·14
·13
·12
·11
·10
·09
·08
·07
·06
0
1
2
3
4
5
6
190°F
180°F
150°F
130°F
100°F
B.T.U/#/sec/°F.
Cooling Water ft/sec.
Temperatures On Graphs
Indicate
Hot Water Inlet Temp^R.

$$\therefore h = \frac{w}{A} \log \frac{T - t_1}{T - t_2} \qquad (2)$$

Where h = B.T.U./Sec./°F/ of difference.

A = ft^2 cooling surface.

T = steam temperature (saturated)

w = lbs. water/sec.

t_1 and t_2 = Inlet and outlet water temperatures.

When examining the manner in which water gains and steam loses heat, use is made of the Reynold's Law.

$$H = At + c\rho vt \qquad (3)$$

where H = Rate of heat transmission

A & C = Experimental constants

t' = temperature difference of fluid and metal

ρ = Fluid density.

V = Fluid velocity.

Considering first the heat flow from metal to water, then with any reasonable velocity the second term in Eqn. (3) is so large that the first term is neglected.

$$\therefore H = c\rho t' v \qquad (4)$$

~~but~~ By equation of continuity $\rho V \frac{lbs}{ft^3} \times \frac{ft}{sec} = \frac{w}{a} \frac{lbs}{sec \times ft^2}$

$$\cancel{\rho V = \frac{w}{A} \times \left[\frac{lbs}{ft^2} \times \frac{ft}{sec} \times \frac{lbs}{sec} \times \frac{1}{ft^2}\right]}$$

$$\cancel{\therefore H = \frac{Cw}{at}} \qquad \cancel{(5)}$$

~~C is a very small number and could be taken as the reciprocal of B~~.

$$\cancel{h = \frac{C}{B}}$$

again:-

If t = temperature difference between the metal and the water

and θ = metal temperature

we [redacted] have:-

$$H = \frac{w}{Ba}(\theta - t) \qquad \text{for the water side...... (6)}$$

$$\therefore (T-\theta) = (\theta - t)\frac{w}{Ba}$$

$$\text{ie} \quad T + \frac{w}{Ba}t = \theta\left(\frac{w}{Ba} + 1\right)$$

$$\therefore \theta = \frac{T + \frac{w}{Ba}t}{1 + \frac{w}{Ba}}$$

$$\text{and } \therefore (\theta - t) = \frac{T + \frac{w}{Ba}t}{1 + \frac{w}{Ba}} - t$$

$$= \frac{T + \frac{w}{Ba}t - t - \frac{w}{Ba}t}{1 + \frac{w}{Ba}}$$

$$= \frac{T - t}{1 + \frac{w}{Ba}}$$

In the case of the heat transmission on the steam side, it is quite reasonable to assume that on condensation taking place, the velocity drops to a negligible quantity so the second term of Reynold's Factor may be neglected and we may write:-

$$H = A(T-\theta) \qquad (7)$$

~~Mr. Gordon G. Webster found that varies with different pressures and temperatures.~~

Now taking the inner and outer cooling surface metal temperature as the same θ (which is only an approximation as it assumes that both surfaces are equally effective) and also that the drop through metal may be neglected, which is probably quite sound with the tubes used, we could apply the above laws.

In Equation (7) the constant A is given by Webster as nearly equal to unity, so the expression becomes $H = (T-\theta)$

Now consider any element of length δx, then the heat transmitted/sec. is:-

$(1)_a$ $$\delta Q = (T-\theta)\frac{\pi}{144}(d_1 + d_4)\,\delta x \qquad \text{(Steam Side)}$$

$(2)_a$ $$\delta Q = \frac{w}{Ba}(\theta - t)\frac{\pi}{144}(d_2 + d_3)\,\delta x \qquad \text{(Water Side)}$$

$(3)_a$ $$\delta Q = w\,\delta t \qquad \text{(Water Side)}$$

Equating $(1)_a$ and $(2)_a$

We should have:-

$$(T-\theta)\frac{\pi}{144}(d_1 + d_4)\,\delta x = \frac{w}{Ba}(\theta - t)\frac{\pi}{144}(d_2 + d_3)\,\delta x$$

But $\dfrac{d_2 + d_3}{d_1 + d_4} = 1$ for the given condenser

$$\cancel{\therefore (T-\theta) = \frac{w\theta}{Ba} - \frac{wt}{Ba}}$$

$$\cancel{\therefore (\theta - t) = \frac{T + \frac{w}{Ba} \times t - t}{1 + \frac{w}{Ba}}}$$

$$\cancel{= \frac{T - t}{1 + \frac{w}{Ba}}}$$

$\therefore$

$$\therefore \quad (\theta - t) = \frac{T - t}{1 + \frac{w}{Ba}}$$

Again equating $(2)_a$ and $(3)_a$ we should get:-

$$\frac{w}{Ba}(\theta - t)\frac{\pi}{144}(d_2 + d_3)\,\delta x = w\,\delta t$$

and by substituting the value of $(\theta - t)$ from above

$$\therefore \quad \frac{1}{Ba} \cdot \frac{T - t}{\left(1 + \frac{w}{Ba}\right)} \times \frac{\pi}{144}(d_2 + d_3)\,\delta x = \delta t$$

or

$$\frac{T - t}{\left(B + \frac{w}{a}\right)a} \times \frac{\pi}{144}(d_2 + d_3)\,\delta x = \delta t.$$

i, now $a = \frac{\pi}{4}\,\frac{(d_3^2 - d_2^2)}{144}$

$$\frac{T - t}{B + \frac{w}{a}} \times \frac{4}{(d_3 + d_2)}\,\cancel{dx}\,\delta x = \delta t$$

$\therefore$

$$\frac{dt}{T - t} = \frac{4\,dx}{\left(B + \frac{w}{a}\right)(d_3 - d_2)}$$

$$\int_{t_1}^{t_2} \frac{dt}{T-t} = \int_0^{l} \frac{4\,\delta x}{\left(B+\frac{w}{a}\right)\left(d_3-d_2\right)}$$

$$\therefore \log_e \frac{(T-t_1)}{(T-t_2)} = \frac{4l}{\left(B+\frac{w}{a}\right)\left(d_3-d_2\right)}$$

From the last expression

$$\left(B+\frac{w}{a}\right) = \frac{4l}{(d_3-d_2)} \times \frac{1}{\log_e \frac{T-t_1}{T-t_2}} \quad (8)$$

Where d_1, d_2 and l are in inches,

and w in lbs/sec.

a in ft.^{2} and hence B could be calculated.

The surface as such does not enter directly into the express::ions, the main point being that the tube length l must be sufficient to deal with the proposed water temperature rise at proposed speed,

<u>Rate of Heat Transmission/sec/sq.ft./^oF/B.T.U's</u>.

Let	the cooling water/minute	=	w
	Inlet water temperature	=	t_1
	Outlet water temperature	=	t_2
	Mean temperature difference	=	t_m
	Cooling surface in tubes in contact with steam	=	a
	Heat transmission rate	=	h

Then from above expressions, we should have:-

$$h = \frac{w(t_2-t_1)}{t_m\, a}$$

let

$$(t_2-t_1) = T_D$$

Then:-

$$h = \frac{w\,T_D}{t_m\,a} \qquad (9)$$

EXAMPLE.

Take as an example Test 1 of Series 1, (atmospheric)

Inlet temperature of cooling water	=	42°F
Outlet temperature of cooling water	=	206.5°F
Cooling water = 3.5 lbs/minute	=	.06 lbs/sec.
Mean temperature difference	=	52.5°F.
Cooling surface of tubes in contact with steam	=	2.5 sq. ft.

Applying the formula on the previous page for finding the rate of heat transmission in the usual method, we have:-

$$h = \frac{w\,T_D}{t_m \times a}$$

$$\therefore\ h = \frac{.06\,(206.5 - 42)}{52.5 \times 2.5}$$

$$= \frac{.06\ (164.5)}{52.5 \times 2.5}$$

$$= \frac{.06 \times 164.5}{52.5 \times 2.5}$$

$$= \underline{\underline{.0752}}$$ B.T.U.Sq.ft./sec./°F of difference

Again to find the value of $\frac{w}{a}$:-

Where w is the amount of cooling water per sec. and a is the sectional area of water flow in square feet:

Take w of the above example, i.e.:-

$$w = .06 \quad \text{lbs/sec.}$$

$$a = .0021 \text{ sq. ft.}$$

$$\therefore \quad \frac{w}{a} = \frac{.06}{.0021} = \underline{\underline{28.7}} \quad \text{lbs/sec./sq.ft.}$$

From Equation (8) we can get the value of Reynold's Factor B for the water side:-

The Equation is:

$$B + \frac{w}{a} = \frac{4l}{d_3 - d_2} \times \frac{1}{\log_e \frac{T - t_1}{T - t_2}}$$

or

$$B = \frac{4l}{d_3 - d_2} \times \frac{1}{\log_e \frac{T - t_1}{T - t_2}}$$

<u>EXAMPLE</u>

The tube diameters on the water side are as follows:-

$$d_2 = 1\tfrac{1}{2}'' \quad \text{dia.}$$

$$d_3 = 1\tfrac{5}{8}'' \quad \text{dia.}$$

$$l = 3.10 \quad \text{ft.}$$

then

$$\frac{4l}{d_3 - d_2} = \frac{4 \times 3.1}{1\frac{5}{8} - 1\frac{1}{2}} = \underline{\underline{1188}}$$

This value of $\frac{4l}{d_3 - d_2}$ is constant for all tests.

Again

$$T = 213.99^\circ F.$$

$$t_1 = 42.0\ ^\circ F.$$

$$t_2 = 206.5\ ^\circ F.$$

Therefore:

$$\log_e \frac{T - t_1}{T - t_2} = \log_e \frac{171.99}{7.49}$$

$$= \log_e 22.94$$

$$= \underline{\underline{3.139}}$$

$\therefore$ B (Water side) $= 1188 \times \dfrac{1}{3.139} - 28.7$

$$= 3.78 - 28.7$$

$$= \underline{\underline{349.5}}$$

For finding the water velocity the following expression is used

$$\frac{Q}{60} = a \rho v$$

where $\rho =$ density of water in lbs/ft^3

$v =$ velocity of circulating water in feet per second.

$a =$ sectional area of water flow.

$Q =$ quantity of water in lbs/min.

In our case $a =$ .0021 sq. ft.

$\rho =$ 62.5 lbs/ft^3

$\therefore$ $Q =$ 60 x .0021 x 62.5 x

$=$ 7.875

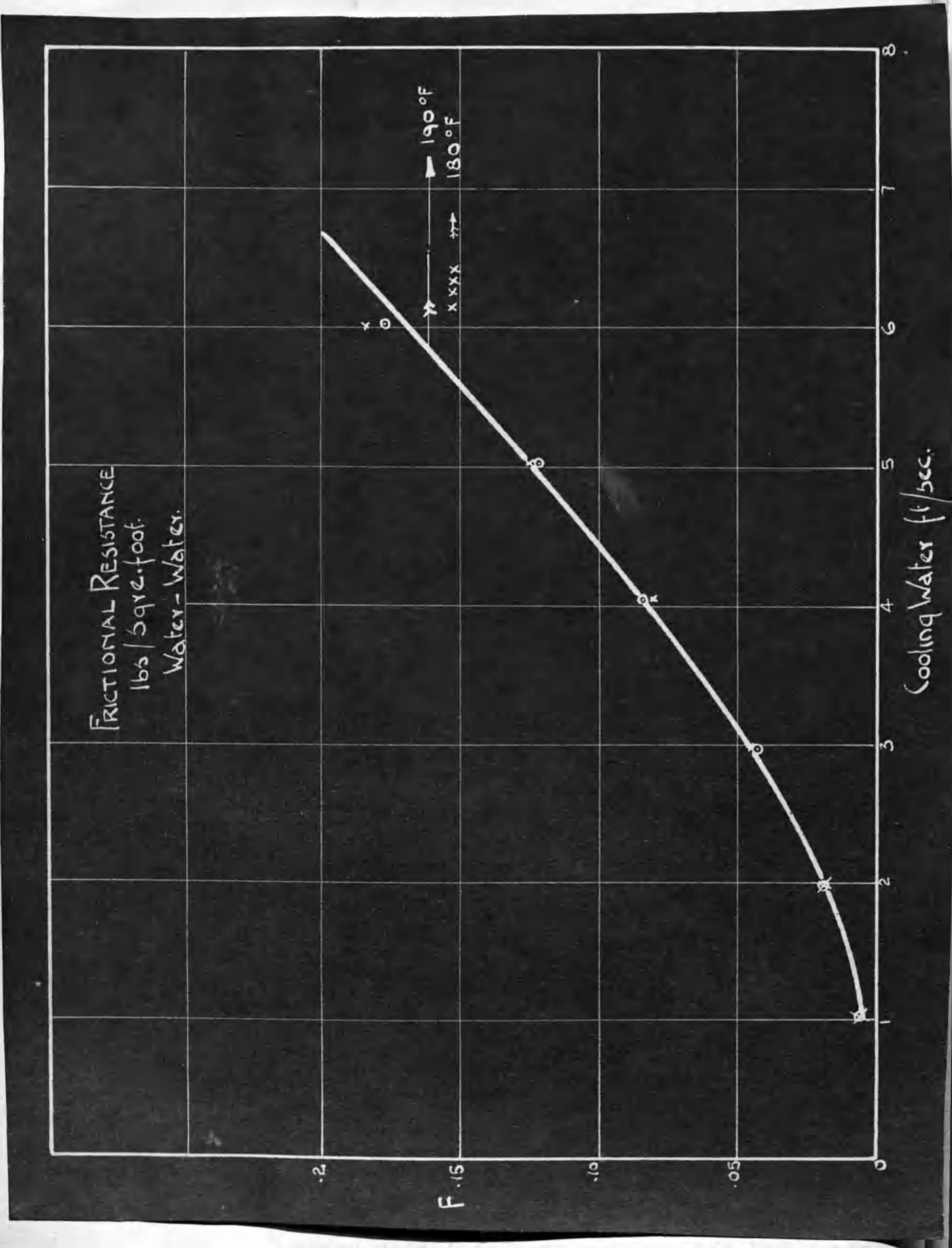
$= \underline{\underline{K v}}$

Where K is a constant.

The design of surface condensers is fundamentally based upon the rate of heat transmission between the condensing steam and the cooling or circulating water.

There are several factors affecting the resistance to heat transfer and the total resistance is the sum of the total

FIG 10



FRICTIONAL RESISTANCE
lbs / sqre. foot.
Water - Water.
F
.2
.15
.10
.05
0
1
2
3
4
5
6
7
8
Cooling Water ft/sec.
190°F
180°F

resistances taken in series. The most influential factors affecting the resistance on the steam side of the tubes are, the amount of air present with the steam, the design of the condenser with respect to the velocity of flow or the elimination of stagnant spaces, and the cleanliness of the surface of the tubes. On the water side the velocity of the water and the cleanliness of the tubes are important. The resistance of the tubes depends upon the material and the thickness.

Resistance to heat transfer between tube and water $= \dfrac{1000 \quad x \quad t_m}{\text{B.T.U./Sq.ft./hr.}}$

FRICTIONAL RESISTANCE.

Frictional resistance of pipe in lbs/sq.ft. is equal to

$$\frac{K \rho V^2}{2g}$$

$$\therefore \quad F = K\rho V^2/2g$$

Where F is the frictional resistance in pipe in lbs/sq.ft.

density of water in lbs/cubic feet

g in ft/sec^2.

V is the velocity of cooling water in ft/sec., but for brass tubes as in the condenser used for the experiments shown later:-

$$K = .005$$

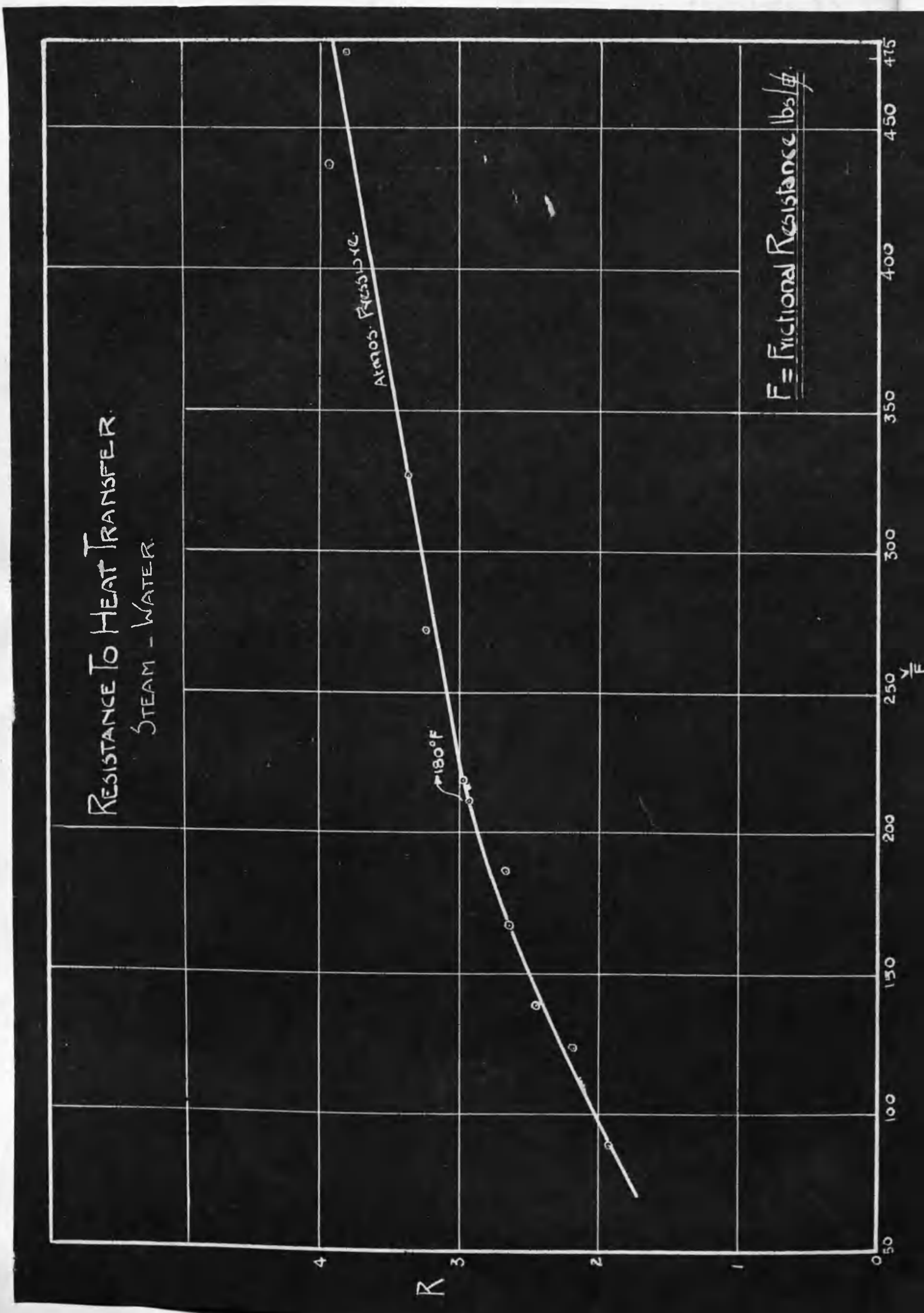
$$\therefore \quad F = \frac{.005 \rho V^2}{2g}$$

$$\therefore \quad \frac{V}{F} = \frac{2g}{.005 \rho V}$$

EXAMPLE.

Considering again the case of Test 1 of the 1st. series, to find the resistance to heat transfer between tube and water and also the value of

FIG 11


RESISTANCE TO HEAT TRANSFER
STEAM - WATER
ATMOS. PRESSURE
180°F
F = Frictional Resistance lbs/⊞
R
Y/F
0
1
2
3
4
50
100
150
200
250
300
350
400
450
475

1st.

$$R = \frac{1000 \times t_m}{B.T.U/sqft/hr}$$

$$t_m = 52.5\ ^\circ F$$

$$B.T.U/sqft/hr = 230.3 \times 60 = \underline{\underline{13818}}$$

$$\therefore\ R = \frac{1000 \times t_m}{13818} = \underline{\underline{3.8}}$$

2nd.

$$\frac{V}{F} = \frac{2g}{.005 \rho V}$$

but from tables $\rho = 62.2$ cuft/lb

$$\therefore\ \frac{V}{F} = \frac{2 \times 32.2}{.005 \times 62.2 \times .445} = \underline{\underline{476}}$$

MEAN TEMPERATURE DIFFERENCE for water on both sides of the tube.

$$t_m = \frac{\text{Initial Temp. differ. - Final Temp. Diff.}}{\log_e \dfrac{\text{Initial Temp. Diffce.}}{\text{Final Temp. Diffce.}}}$$

which is applicable to either parallel or counter flow.

REYNOLD'S FUNCTION.

Now for finding the Reynold's function for water from the "Water-Water" tests, and then applying it to the steam tests:-

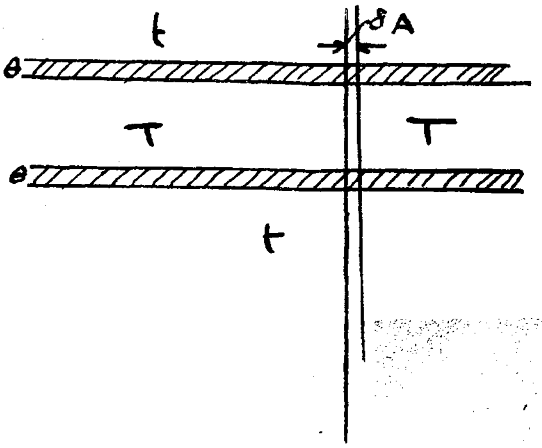
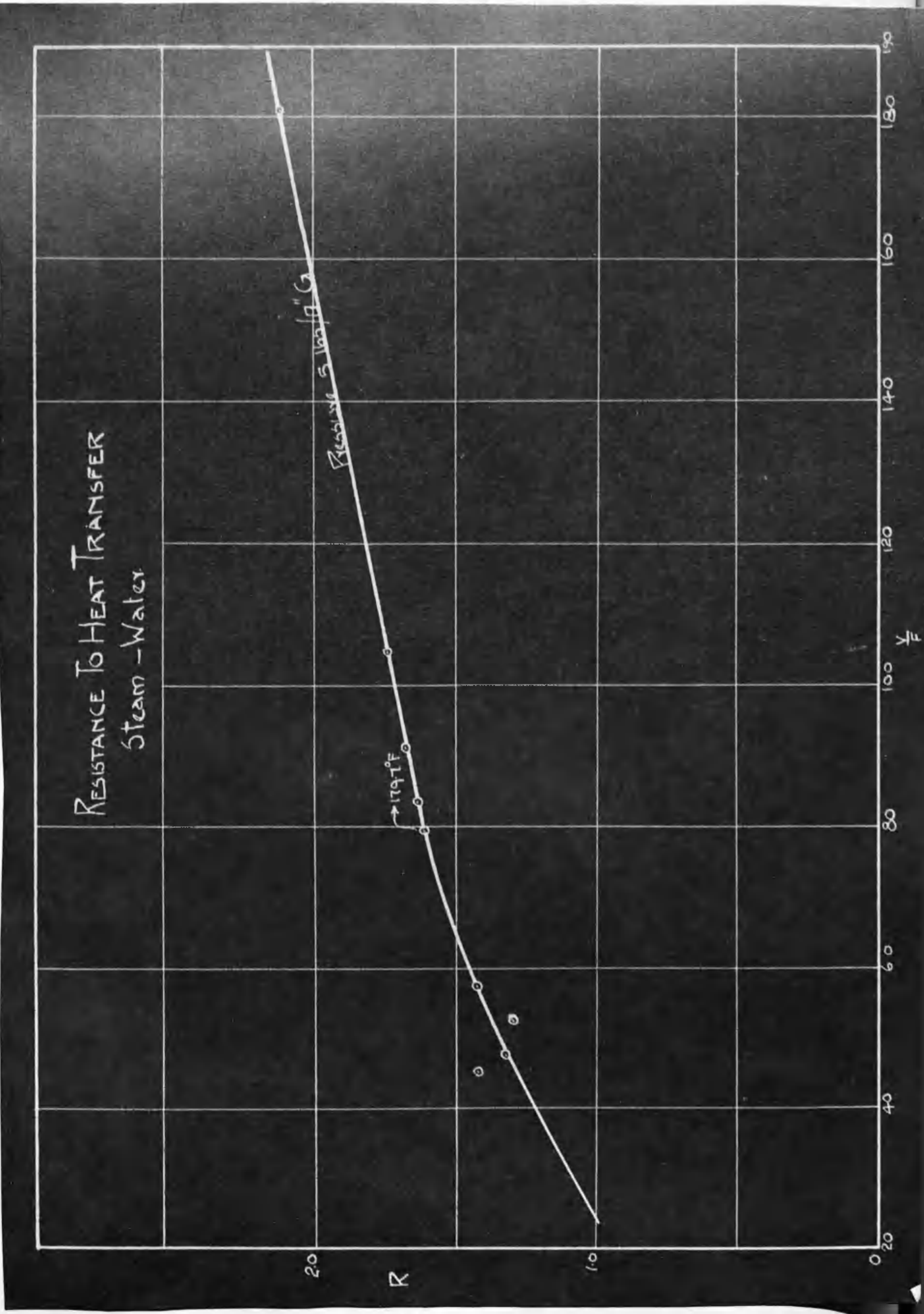


FIG 12



RESISTANCE TO HEAT TRANSFER
Steam - Water
R
2.0
1.0
0
20
40
60
80
100
120
140
160
180
190
V/F
→179.2°F

We have:-

$$f_1(T-\theta)dA_1 = f_2(\theta - t)\,dA_2$$

Where $f = \left(A + B\frac{\omega}{A}\right)$

Where f = Reynold's function for water

$$\therefore \quad \frac{T-\theta}{\theta - t} = \frac{f_2\,dA_2}{f_1\,dA_1}$$

let $\dfrac{f_2\,dA_2}{f_1\,dA_1} = \gamma$

$$\therefore \quad T-\theta = \gamma\theta - \gamma t$$

or

$$\theta = \frac{T+\gamma t}{1+\gamma}$$

But

$$f_1\left\{T - \frac{T+\gamma t}{1+\gamma}\right\}dA_1 = h(T-\cancel{\theta}t)\,dA$$

$$f_1\left\{\frac{T+\gamma T - T - \gamma t}{1+\gamma}\right\}dA_1 = h(T-t)\,dA$$

$$\therefore \quad f_1\frac{\gamma}{1+\gamma}\,dA_1 = h\,dA.$$

and hence γ found

also $\quad \gamma = \dfrac{f_2}{f_1} \times \dfrac{dA_2}{dA_1}$

but take

$$\frac{f_2}{f_1} = \frac{h_2}{h_1}$$

from curve of mean rate h, and therefore γ known, and f_1 and f_2 could be found.

Now applying the values of f_2 thus found, a graph of f_2 against speed with the mean value of f_k as a point on the graph is drawn and we are in a position to find f_1 for the steam.

$$d\theta = f_1(T-\theta)\,dA_1 = f_2(\theta - t)\,dA_2 = h(T-t)\,dA$$

$$\frac{T-\theta}{\theta - t} = \frac{f_2}{f_1}\,\frac{dA_2}{dA_1} = \gamma$$

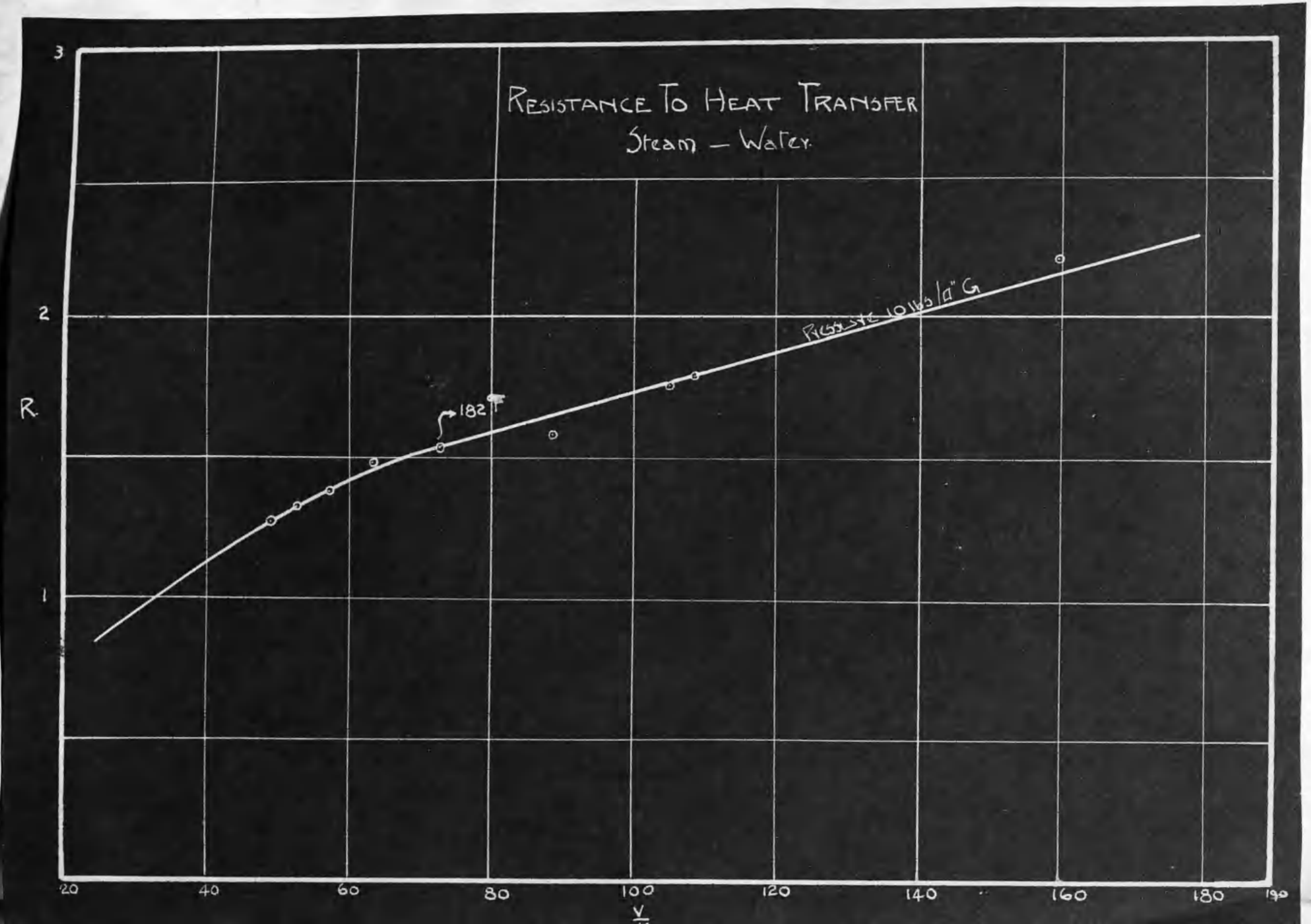

RESISTANCE TO HEAT TRANSFER
Steam — Water.
182°F
R.
3
2
1
20
40
60
80
100
120
140
160
180
190
V

FIG 1

As before

$$\theta = \frac{T + rt}{1+r}$$

$$\therefore f_2\left[\left(\frac{T+rt}{1+r}\right) - t\right] dA_2 = h(T-t)\,dA_t$$

$$f_2\left[\frac{T+rT-t-rt}{1+r}\right]dA_2 = h(T-t)\,dA_t$$

$$\frac{rf_2}{1+r}\,dA_2 = h\,dA_t$$

$$\therefore \frac{r}{1+r} = \frac{f_2}{h} \times \frac{dA_2}{dA_t} \qquad \text{hence } r \text{ found}$$

But $$f_1\,\frac{T-T+rt-rT}{1+r}\,dA_1 = h(T-t)\,dA$$

$$\therefore f_1\,\frac{T+rT-T-rt}{1+r}\,dA_1 = h(T-t)\,dA$$

$$\therefore f_1(T-t)\frac{r}{1+r} = h(T-t)\,dA$$

$$\therefore f_1\,\frac{r}{1+r}\,dA_1 = h\,dA.$$

and hence f_1 found for steam.

Apply the r found above to this last equation in which we know h for steam from the assumed mean line and we know $\frac{dA}{dA_1}$ therefore f_1 for steam could be easily determined.

METHOD FOR TUBE TEMPERATURE CALCULATION.

Divide the tube length into a number of equal divisions as shown ~~above~~ opposite page (18). T & t ~~and~~ are the two inlet temperatures.

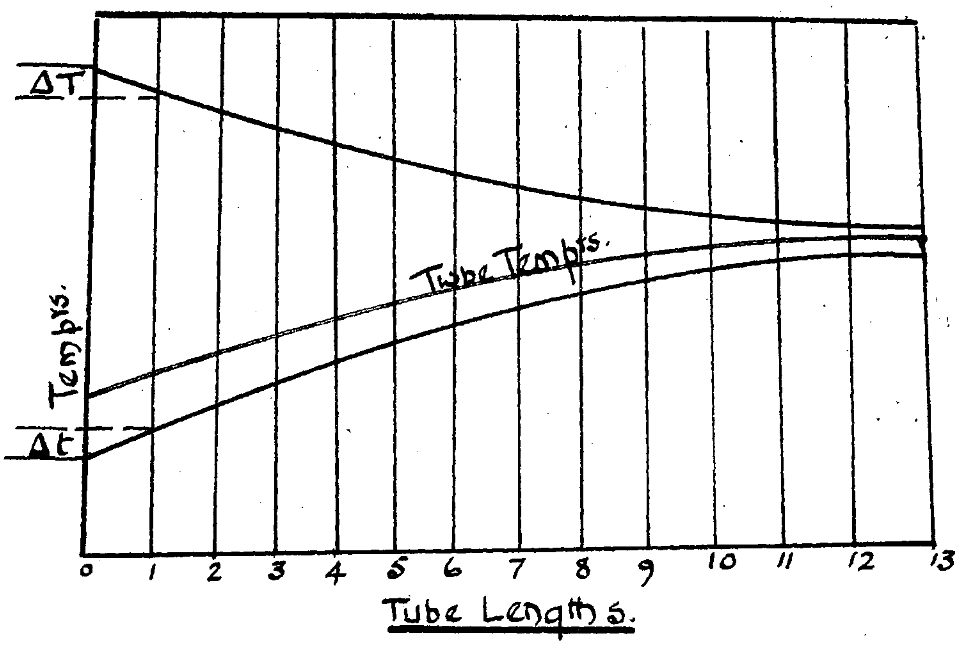
We have:-

$$h(T-t)\,dA = dQ$$

$$\therefore \frac{dQ}{dA} = h(T-t)$$

hence dQ found.

FIG.14



but $dQ = w_1 \Delta T = w_2 \Delta t$

where $w_1 =$ lbs steam/sec. and $w_2 =$ lbs water/sec.

$\therefore$ Δt and ΔT found

$\therefore$ Graph for T and t is drawn.

Now for graph of θ

We know γ from the results calculated, so by substituting the values for T and t in the formula,

$$\theta = \frac{T + \gamma t}{1 + \gamma}$$

we get θ for every division,

and then for finding the mean $\hat{\theta}$ take the area under the curve.

<u>EXAMPLE</u>.

Considering test No. 1 of the 1st. series, the mean temperatures were calculated as for two different condensers starting and finishing at the tubebend (where a thermometer was fitted to read the bend temperature). The temperatures used in these calculations were marked by A, B, C, d, and e in the tables of Results.

<u>t_m for parallel flow was</u>

$$t_m = \frac{(A - d) - (B - e)}{\log_e \dfrac{A - d}{B - e}} \qquad (1)$$

<u>t_m for counter flow</u>

$$t_m = \frac{(B - e) - (C - d)}{\log_e \dfrac{B - e}{C - d}} \qquad (2)$$

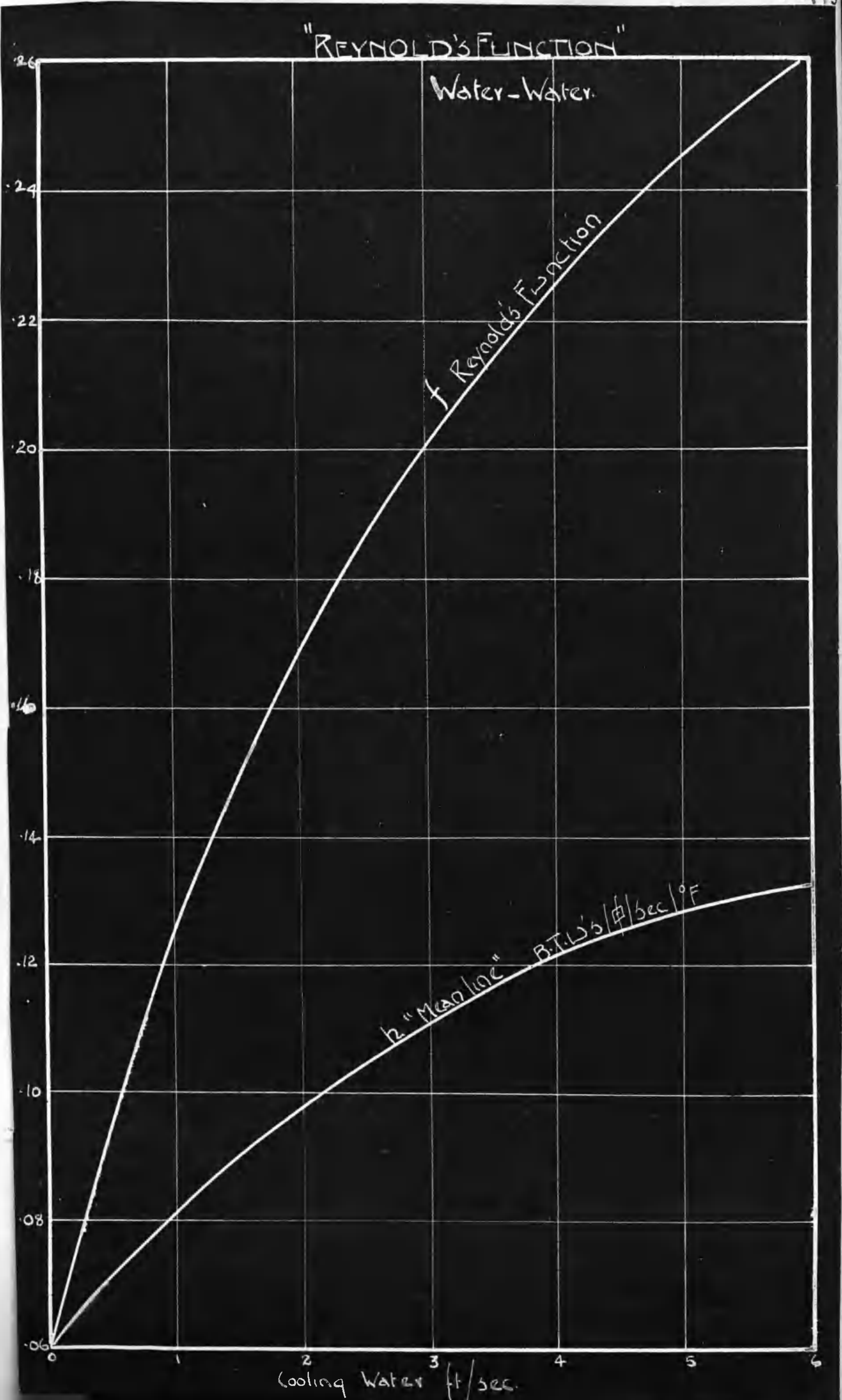
Fig. 2 gives an indication of the flows.

A = 190°F: B = 181.75°F: C = 173.5°F.
= 52.0°F: = 117.3°F.

Calculating for the parallel flow we should have

$$t_m = \frac{(190 - 52) - (181.75 - 117.3)}{\log_e \dfrac{190 - 52}{181.75 - 117.3}}$$

FIG 15


"REYNOLD'S FUNCTION"
Water-Water.
f Reynold's Function
k "Mean line" B.T.U's/#/sec/°F
·26
·24
·22
·20
·18
·16
·14
·12
·10
·08
·06
0
1
2
3
4
5
6
Cooling Water ft/sec.

$$\therefore \quad t_m = \frac{138 \ - \ 64.4}{\log_e \ \frac{138}{64.4}}$$

$$= \frac{73.6}{\text{Loge} \ \ 2.14} \quad = \quad \underline{\underline{96.5.}}$$

Whereas t_m for counter flow = 88.8. By equation (2) The heating surfaces for the parallel and counter flows respec:tively are 1.08 and 1.37 sq. ft.

So by the equation

$$h = \frac{H}{A \, t_m}$$

Where H is the total heat/sec, A the heating surface and mean temperature difference, h the rate of heat transmission could be calculated.

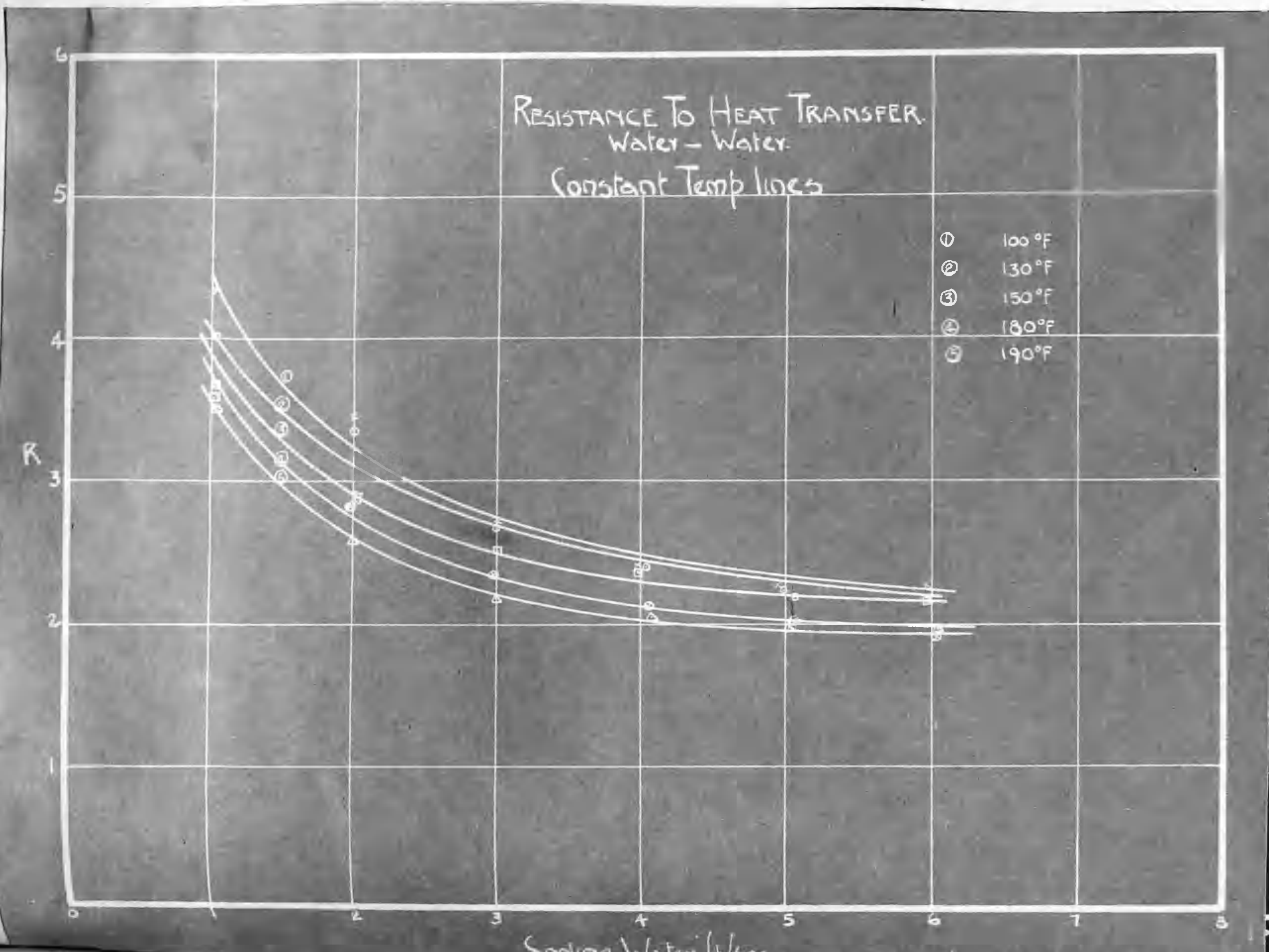
Now that we have h for both flows, a mean h was calculated and a graph was plotted against the varying speeds of the cooling water.

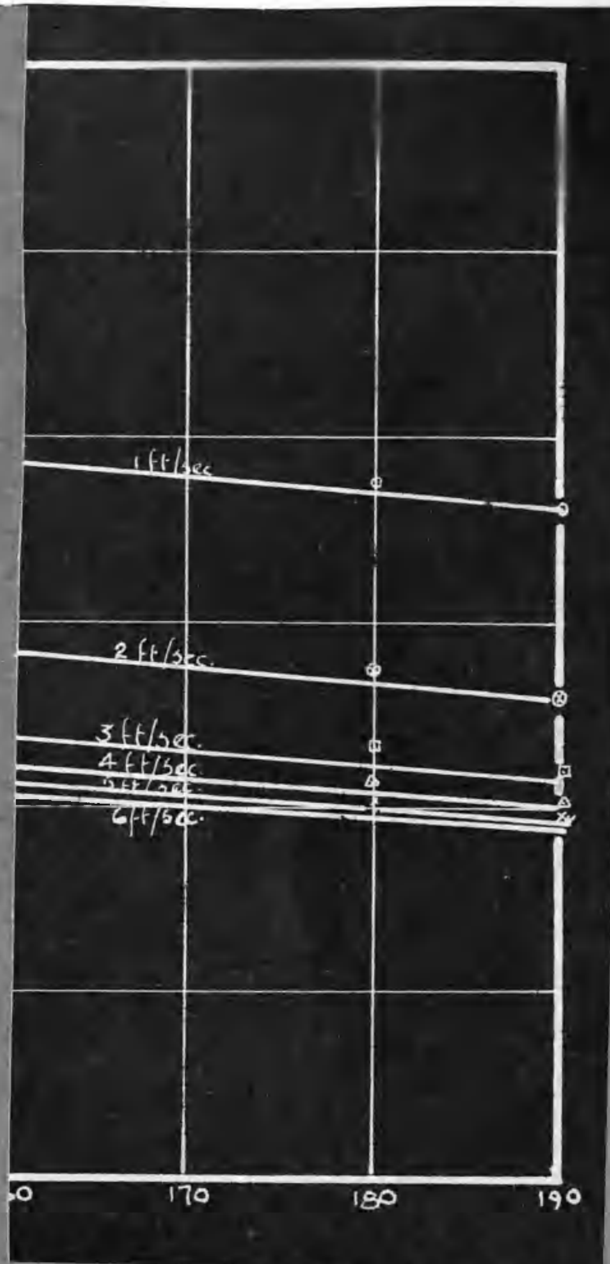
REYNOLD'S FUNCTION FOR WATER.

We mean by Reynold's Function the expression $\left(A + B\frac{w}{a}\right)$ which is denoted by f_1 and f_2 in Table 2. f_1 being for the hot water and f_2 for the cold water.

DESCRIPTION OF GRAPHS.

The curves Fig.7 (pp.6) on steam water tests show the heat transmitted in B.T.U.'s/ $\#$ /sec $^{\circ}$F. difference against a base of cooling water in ft. sec. h increases with the water velocity. It is therefore noted that the cooling water in a condenser would be sent through at a fairly high speed for the best results. Of course there is a limit, at least in practice if the cooling water has to be pumped through the condenser, as the resistance to flow increases, as the velocity to the power something like 1½ to 2 or perhaps more. A compromise has thus to be made to get the best conditions for any plant. Thus h decreases with the increase of pressure.


RESISTANCE TO HEAT TRANSFER.
Water – Water.
Constant Temp lines
100 °F
130 °F
150 °F
180 °F
190 °F
R
Cooling Water ft/sec.


1 ft/sec.
2 ft/sec.
3 ft/sec.
4 ft/sec.
6 ft/sec.
170
180
190

The curves which come next under consideration are those of the heat transmittor in B.T.U.'s/sec/Sq.ft./^{o}F in the "Water-water" tests, page8. The inlet temperature is given for each case and the lines follow a very definite path which if extended, will pass through the zero point. They show that as in the case of steam water h increases with the increase of speed. At the higher inlet temperatures the rate of heat trans: :mission is bigger than in the low inlet temperatures of the hot water. This is possibly due to some very small error in the experiment as it could be obviously seen that the whole points in the different tests should lie on a mean line.

The difference however, is very small indeed and a mean line was taken for further calculations.

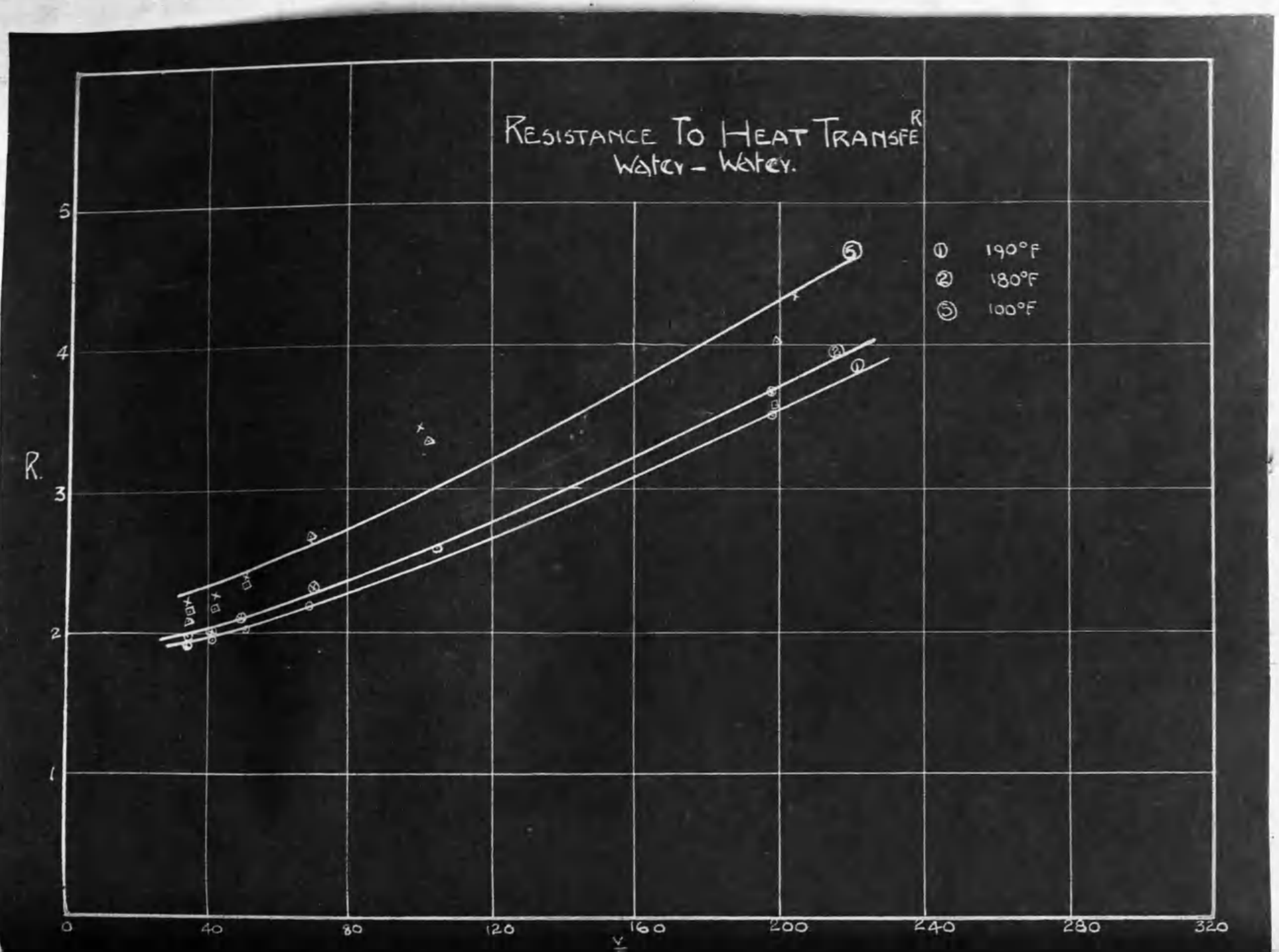
The curves shewn opposite page 7 are those of Reynold's factor, for the water side plotted against an increas: :ing $\frac{w}{a}$; Where w is the amount of cooling water in lbs/sec, and a is the sectional area of water flow in square feet.

has been calculated from the formula below

$$B + \frac{w}{a} = \frac{4l}{(d_3 - d_2)} \cdot \frac{1}{\log_e \frac{T - t_1}{T - t_2}}$$

It will be noticed that Reynold's factor from the above equation is a linear function while in the graph it follows a parabolic path, this is probably due to a power, (very small), other than unity which has been neglected in the above expression to avoid further complications. The curve shews that B increases as $\frac{w}{a}$ increases, but at the high velocity B does not increase so rapidly as at the lower velocities.

It will be interesting to see that in figure (8) of Reynold's factor that at atmospheric pressure the value of Reynold's factor is highest and that as we increase the pressure of steam, the value of B decreases. We know that the value of T in expression (7) depends on the pressure and the


RESISTANCE TO HEAT TRANSFER
Water - Water.
R.
① 190°F
② 180°F
⑤ 100°F
0
1
2
3
4
5
40
80
120
160
200
240
280
320
FIG 18

higher the pressure of steam, the higher is its saturation temperature, and the value of $\frac{4l}{d_3 - d_2}$ is a constant then the above-mentioned change depends mostly on the two other variables which are $\log \frac{T - t_1}{T - t_2}$ & $\frac{w}{a}$ the first of which changes considerably with the change of pressure.

FRICTIONAL RESISTANCE.

The frictional resistance curve page (14) between the water and the tube in the water-water tests was calculated from the formula

$$F = \frac{Kv^2}{2g}$$

where K = co-efficient of friction and taken as .005
The equation above is that of a parabola passing through the origin.

The graphs pp 15, 16 & 17 show the resistance to heat trans:fer between tube and water plotted against $\frac{V}{F}$ which shows that the resistance decreases with the increase of velocity. It will be seen that the curve suddenly changes its path at about 180°F (for the cooling water temperature). This sudden change in the rate of heat transfer is due to the pressure of air dissolved in the water which is but slowly liberated at the lower temperatures.

When the water moves forward, the temperature of the film along the walls will rise as the mean temperature rises.

The sudden change in the curve is that the liberation of air at the high temperature will be so rapid that the film in question will be broken up and the law of heat transfer suddenly changed.

It will also be noticed that on examining the set of graphs mentioned above, (and starting by series 1, which is atmospheric to series 5, which is 10 lbs/sq. in. G) that the curves become flatter than the previous ones. This is accounted for as follows:

The water enters the tubes by say an average tempera::ture of about 40°F. and leaves at about 212. In the first case the saturation temperature of steam is about 213.7°F.

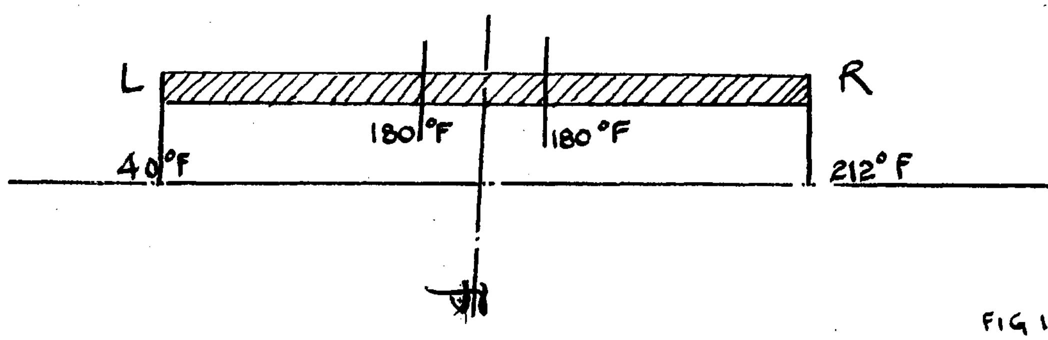


FIG 19

In this case the critical temperature will be nearer to side of outlet as we increase the pressure the saturation temperature becomes higher and the critical temperature moves towards the left.

The figures pp 20, ~~24 and 25~~ show the resistance to heat transferred plotted against speed inlet hot water tempera::ture and Coolingwater respectively.

They show very distinctly that the resistance has a sudden drop over the small speed range, and at about a speed of 3 ft./sec. the resistance is very much less and is practically constant. For this reason it would be advisable for further investigations on the subject that speed of cooling water should always be taken above 3 ft./sec.

The curves, page 21, are obtained from the above ones by means of cross-interpolation.

In conclusion I beg to thank Professor A.L.Mellanby, D.Sc., and Dr. Wm. Kerr, for the encouragement and valuable help during the course of the work.

TABLE I-

(STEAM-WATER.)

series I,Atmos.press.

Number of test.	I	2	3	4	5	6
Duration of test;mins.	I2	I2	I2	I2	I2	I2
Barometic Press INsHg.......	30.I5	30.I5	30.I5	30.I5	30.I5	30.I2
" " lbs/ins^2......	I4.80	I4.80	I4.80	I4.80	I4.80	I4.78
Steam pressure lbs/ "G......	.500	.500	.500	.430	.430	.260
" " " ab.......	I5.30	I5.30	I5.30	I5.23	I5.23	I5.04
Steam temp.Inlet °F........	2I3.95	2I3.95	2I3.69	2I3.69	2I3.69	2I3.20
" " Outlet °F.........	2I2.20	2I2.20	2I2.20	2I2.20	2I2.20	2II.00
"" " Saturated°F.......	2I3.99	2I3.99	2I3.795	2I3.795	2I3.795	2I3.I32
Coling water temp°F.inlet...	42.0	42.0	42.0	42.0	42.0	42.2
" " " outlet...	206.5	204.0	200.0	I94.00	I87.5	I85.8
" " " diff......	I64.5	I62.0	I58.0	I52.0	I45.5	I43.4
" " " mean......	52.5	56.9	64.9	70.4	77.6	78.4
Steam flow per hour.........	98.2	I03.25	I00.5	II00.25	III.25	I05.0
Coolingwater per hour lbs....	2I0.0	225.0	306.0	360.0	450.0	462.0
" " " min. lbs.....	3.50	3.75	5.I0	6.0	7.5	7.7
Heat toC.W.BT.U/min.........	576.0	603.6	805.0	9I2.50	I092.0	III3.0
" " "B.T.U/min/SQ.ft...	230.30	24I.0	322.0	364.3	436.9	446.0
Heat/sec/sq.ft/°F.B.T.U/....	.0752	.0705	.0826	.0865	.0939	.0945
Velocity of C.W.ft/sec......	.445	.476	.636	.764	.952	.976
Resistance to heat transfer.	3.800	3.93	3.36	3.22	2.980	2.920
W/A(W in lbs/sec)...........	28.77	28.9	40.5	47.6	59.5	6I.05
$\frac{4L}{d_3-d_2}$ (water side).........	II88.	II88.	II88.	II88.	II88.	II88.
Log $\frac{T - tl}{T - t_2}$ (water side)....	3.I39	2.845	2.439	2.I62	I.878	I.833
B (Reynold's Factor)	349.50	363.5	4I7.5	468 .4	535.5	548.95
Wall TEMP. ° F.						
Density of water lbs/ft^3...	62.20	62.23	62.25	62.275	62.28	62.29
V/F = 2G/.005 v	476.2	436.	426.	272?	2I8.	2I2.

TABLE .I. Contd

(Steam - water.) Series I,contd

Number of test.	7	8	9	10	11	12
Duration of test,mins	12	12	12	12	12	12
Barometric Press. Ins. Hg.	30.12	30.12	30.23	30.20	30.20	30.20
" " lbs/ins^2	14.78	14.82	14.82	14.80	14.80	14.80
Steam Pressure lbs/ " G;.	..260	.260	.260	.260	.260	.260
" " " abs.	15.04	15.08	15.08	15.08	15.06	15.06
Steam temp .Inlet°F.	213.20	213.26	213.26	213.26	213.20	213.20
" " Outlet°F.	212.0	212.0	211.0	179.0	163.0	159.0
" " Saturated°F.	213.132	213.26	213.26	213.26	213.198	213.198
Cooling water temp.°F, Inlet.	42.4	42.0	42.0	42.2	42.2	42.1
Cooling water temp Outlet.	183.0	178.7	173.0	156.5	147.0	128.0
" " " .diff.	140.6	136.7	131.0	114.3	105.0	86.0
" " " .mean.	81.4	85.4	90.4	103.9	110.5	123.0
Steam flow per hour.	105.0	105.0	103.5	99.0	104.0	107.0
Cooling water per hour.	525.0	588.0	708.0	792.0	972.0	1098.0
" " " min.	8.75	9.8	11.8	13.2	16.2	18.3
Heat to C.W.B.T.U./min.	1232.	1339.	1547.	1690.	1690.	1572.
" " " "B.T.U./min/sq. ft.	492.8	536.0	618.0	676.0	676.0	630.0
Heat/sec/sq.F/°FB.T.U.	.1005	.1045	.114	.099	.120	.0910
Velocity of C.W. ft/sec.	1.109	1.241	1.495	1.674	2.058	2.320
Resistance to Heat tranfer.	2.75	2.660	2.439	2.180	2.320	1.920
$^W/_A$ (W in lbs/sec)	69.5	77.8	93.6	104.8	128.5	145.0
$\frac{4\ L}{d_3 - d_2}$ w)(water side)	1188.	1188.	1188.	1188.	1188.	1188.
$\log \frac{T - t_1}{T - t_2}$ (water side)	1.728	1.602	1.452	1.105	.948	.698
B(Reynold's Factor)	578.0	618.2	673.9	807.0	880.	980.
Wall Temp ° F						
Density of water lbs/ft^3	62.31	62.32	62.344	62.38	62.394	62.42
$\frac{V}{F} = \frac{2\ g}{005}$	187.	166.7	138.5	123.7	100.7	89.5

T A B L E. I. Contd

(Steam-Water) Series 2. Press 5 lbs./ " Gauge.

Number of test.	I	2	3	4	5	6
Duration of Test mins	I2	I2	I2	I2	I2	I2
Barometric press. Ins Hg.	30.0I	30.0I	30.0I	30.0I	30.0I	30.0I
" " lbs/ins^2	I4.705	I4.705	I4.705	I4.705	I4.705	I4.705
Steam pressure lbs " G	5.5	5.7	5.5	5.8	5.7	5.5
" " " abs.	20.205	20.405	20.203	20.505	20.405	20.205
Steam Temp; Inlet °F	229.5	229.6	228.55	229.3	228.9	228.5
" " Outlet °F	22I.0	22I.0	22I.0	220.0	22I.0	22I.0
" " Saturated °F	228.56	228.53	22.53	229.3I	229.05	228.53
Cooling Water Temp °F Inlet.	42.0	42.0	42.0	42.0	42.0	42.0
Cooling Water Temp °F Outlet	206.8	I92.5	I86.5	I82.0	I78.5	I75.5
Cooling Water Temp.diff.	I63.8	I50.5	I44.5	I40.0	I36.5	I33.5
" " " mean.	77.8	75.4	97.4	I0I.5	I04.2	I06.5
Steam flow per hour.	267.0	270.	278.0	286.0	290.0	282.0
Cooling Water per hour.	540.0	8I6.0	939.0	I08.0	II76.	I236.
" " " min.	9.00	I3.6	I5.65	I8.00	I9.60	20.6
Heat to C.W.B.T.U./min.	I524.2	2049.0	2258.0	2520.	2675.	2749.0
" " " " B.T.U./min/sq ft.	6II.0	82I.3	925.0	I008.0	I068.	I099.
Heat/sec./sq.F/°F B.T.U.	.I3I	.I8I5.	.I57	.I655	.I705	.I720
Velocity of C.W? ft/sec.	I.I40	I.725	I.985	2.280	2.485	2.6I7
$W/_A$ (Win lbs/ sec)	7I.5	I07.9	I23.9	I42.9	I55.5	I63.2
Resistance to heat transfer	2.I22	I.529	I.750	I.680	I.629	I.6I5
$\frac{4\ L}{d_3 - d_2}$ (Water side)	II88.	II88.	II88.	II88.	II88.	II88.
Log $\frac{T - t_1}{T - t_2}$ (Water side)	2.I066	I.993	I.4839	I.3788	I.3083	I.258I
B (Reynold's Factor)	520.	488.I	678.I	7I9.6	753.5	905.
Wall Temp. °F						
Density of water lbs/ft^3	62.2I	62.275	62.30	62.3I3	62.33	62.340
$\frac{V}{F} = \frac{2\ g}{005\ \ v}$	I8I.6	I20.	I04.5	9I.0	83.5	79.45

TABLE .I. Contd.

(Steam-Water).Series.2, contd.

Number of test.	7	8	9	I0
Duration of test, mins	I2	I2	I2	I2
Barometric Press. Ins Hg.	30.II	30.II	30.II	30.II
" " lbs/ins^2	I4.75	I4.75	I4.75	I4.75
Steam pressure lbs / " G	5.7	5.40	5.5	5.5
" " " abs.	20.45	20.I2	20.25	20.25
Steam temp. Inlet °F	229.I5	228.4	228.64	228.9
" " Outlet °F	222.5	22I.5	22I.5	22I.5
" " Saturated °F	229.I7	228.39	228.65	228.65
Cooling water temp °F Inlet	4I.6	4I.6	4I.4	4I.5
" " " Outlet	I62.5	I58.8	I53.0	I45.0
" " " diff	I20.9	II7.2	III.6	I03.5
" " " mean	II6.0	II9.0	I27.8	I28.2
Steam flow per hour.	286.0	290.0	290.0	288.0
Cooling Water per hour.	I695.	I884.	2070.	2I72.
" " " min.	28.25	3I.4	34.5	36.2
Heat to C.W.B.T.U./min	34I0.	3685.	3855.	3750.
" " " " B.T.U./min/sq ft.	I363.	I465.	I540.	I507.
Heat/sec/sq ft/°F B.T.U.	.I960	.205	.225	.243
Velocity of C.W. ft/sec.	3.58	3.982	4.36	4.590
Resistance to heat Transfer.	I.4I5	I.350	I.380	I.4I7
W/A (W in lbs/sec.	224.0	249.5	274.2	287.3
$\frac{4\ L}{d_3 - d_2}$ (Water side)	II88.	II88.	II88.	II88.
Log $\frac{T - t_1}{T - t_2}$ (Water side)	I.0403	.9858	.9083	.8060
B (Reynold's Factor)	I200.	I3I0.	I420.	I600.
Wall Temp. °F				
Density of water lbs/ft^3	62.37	62.38	62.394	62.400
$\frac{V}{F} = \frac{2\ g}{005 \quad v}$	57.8	52.I	47.6	45.2

T A B L E. I .Contd

(STEAM-WATER).Series 3, IO lbs/ " G.

Number of test	I	2	3	4	5	6
Duration of test,mins	I2	I2	I2	I2	I2	I2
Barometric Press.Ins Hg.	29.54	29.54	29.54	29.54	29.54	29.566
" " lbs/ins^2	I4.50	I4.50 5	I4.50	I4.50	I4.50	I4.5I
Steam Pressure lbs/ " G	II.9	II.9	I0.7	II.08	I0.7	II.05
" " " abs.	26.4	26.4	25.2	25.58	25.2	25.56
Steam Temp. Inlet°F	243.0	243.0	240.0	24I.0	240.0	24I.0
" " Outlet °F	233.0	232.6	228.0	232.0	232.0	234.0
" " Saturated °F	243.08	243.08	240.52	24I.32	240.52	24I.27
Cooling water temp°F inlet	4I.0	4I.0	4I.0	4I.0	4I.0	4I.0
" " " "outlet	2II.0	203.7	I99.7	I90.0	I79.7	I73.2
" " " "diff.	I70.0	I62.7	I58.7	I49.9	I38.7	I32.2
" " " "mean.	92.3	99.4	I00.0	I08.2	II6.7	I2I.I
Steam flow per hour	393.4	394.0	393.5	395.0	396.0	394.0
Cooling water per hour	6I5.0	855.0	930.0	III3.0	I356.	I645.
" " " min.	I0.25	I4.25	I5.5	I8.55	22.6	25.9
Heat to C.W.B.T.U./MIN.	I743.0	23I9.0	2459.00	2782.0	3I38.	3429.
" " " " "/Min/Sq.ft	698.	927.	982.	III3.	I252.	I372.
Heat/sec/sqft/°FB.T.U.	.I263	..I565	.I635	.I7I5	.I90	.205
Vel. of C.W. ft/sec.	I.30	I.907	I.965	2.352	2.865	3.285
Resistance to Heat transfer	2.2I	I.785	I.750	I.585	I.549	I.495
W/A (Wi in lbs/sec)	8I.5	II3.2	I23.0	I47.I	I79.3	202.5
$\frac{4L}{d_3 - d_2}$ (Water side)	II88.	II88.	II88.	II88.	II88.	II88.
$\log \frac{T - t_1}{T - t_2}$ (Water side)	I.8405	I.637I	I.5872	I.38I3	I.I878	I.080I
B (Reynold's Factor)	385.5	6I2.8	626.	7II.9	822.7	898.5
Wall Temp °F						
Density of water lbs/ft^3	62.I82	62.20	62.23	62.25	62.3I2	62.33
$\frac{V}{F} = \frac{2\,g}{005 \quad v}$	I59.5	I08.9	I05.5	88.3	72.49	63.I

T A B L E . I . Contd

(STEAM - WATER) Series 3, Contd.

Number of test.	7	8	9	I0	II	I2
Duration of Test, mins.	I2	I2	I2	I2	I2	I2
Barometric Press. Ins Hg.	29.56	29.56	29.56	29.56	29.56	29.56
" " lbs/ins^2	I4.5I	I4.5I	I4.5I	I4.5I	I4.5I	I4.5I
Steam pressure lbs/ " G	I0.4	II.0	II.2	II.0	II.0	II.2
" " " abs.	24.9I	25.5I	25.7I	25.5I	25.5I	25.7I
Steam temp Inlet °F.	240.0	24I.0	24I.0	240.9	24I.0	24I.5
" " Outlet °F.	235.5	237.0	237.0	236.2	234.0	232.0
" " Saturated °F.	239.89	24I.I7	24I.59	24I.I7	24I.I7	24I.59
Cooling water temp °F. Inlet	4I.0	4I.0	4I.0	4I.0	4I.0	4I.0
" " " " Outlet	I65.7	I62.8	I59.7	I54.7	I38.9	I35.9
" " " " diff.	I24.7	I2I.8	II8.2	II3.7	97.9	94.9
" " " " mean.	I26.4	I30.I	I33.0	I35.I	I45.6	I47.8
Steam flow per hour.	393.4	393.5	394.0	394.0	395.0	394.4
Cooling water per hour.	I704.	I860.	I995.5	2I45.?	2I60.	2I75.
" " " min.	28.4	3I.0	33.25	35.75	36.0	36.25
Heat to C.W.B.T.U./min	3523.	3776.	3933.	4069.	3524.	3445.
" " " " " " " /min/sq ft.	I4I0.	I509.	I570.	I627.	I4I0.	I378.
Heat/sec/sq ft/°F BT.U.	.I859	.I935	.I970	.2000	.I6I5	.I554
Velocity of C.W. ft./Se c	3.62	3.935	4.22	4.53	4.57	4.60
Resistance to heat transfer	I.36	I.32	I.270	I.383	I.720	I.79
W/A (W in lbs/sec)	255.400	245.9	264.0	283.8	285.9	287.5
$\frac{4L}{d_3 - d_2}$ (Water side)	II88.	II88.	II88.	II88.	II88.	II88.
$\log \frac{T - t_1}{T - t_2}$ (Water side)	.9858	.936I	.8899	.84I6	.67I4	.64I8
B (Reynold's Factor)	979.6	I023.I	I073.	II27.2	I435.I	I567.5
Wall Temp. °F						
Density of water lbs/ft^3	62.35	62.369	62.372	62.385	62.409	62.4I8
$\frac{V}{F} = \frac{2g}{.005} v$	57.4	52.9	49.20	45.75	45.4	45.I

TABLE II.

SMALL HIGH SPEED CONDENSER

WATER TO WATER

Series No.	(I)					
Number of test.	I	2	3	4	5	6
Hot water circulation.						
Water, lbs/min.	33.6	33.6	33.6	33.6	33.6	33.5
Water lbs/sec.	.560	.560	.560	.560	.560	.558
Inlet temp. °F (A)	I90.	I90.	I90.	I90.	I90.	I90.
Outlet temp. °F (C)	I73.5	I66.6	I62.0	I59.5	I58.0	I56.0
Temp. difference.	I6.5	23.4	28.0	30.I	32.0	34.0
Heat B.T.U'S / sec.	9.24	I3.I0	I5.67	I6.84	I7.9	I8.99
Temp. at bend °F (B)	I8I.75	I78.3	I76.0	I74.95	I74.0	I73.0
Average hot water speed.	4.25 ft.sec.					
COLD WATER CIRCULATION						
Water lbs/min.	8.25	I5.6	23.5	32.I	39.5	47.6
Water lbs/sec.	.I375	.260	.390	.535	.658	.783
Water speed ft/sec.	I.045	I.980	2.983	4.075	5.02	6.04
Inlet temp. °F (d)	52.0	52.0	52.0	52.0	5I.0	5I.0
Outlet temp. °F (E)	II7.3	I02.0	92.0	84.0	79.0	75.0
Heat .B.T.U's/sec.	9.03	I3.0	I5.64	I7.I2	I8.4	I9.03
t m. (parallel flow)	96.5	I04.0	I08.0	II2.5	II5.2	II6.8
t m (counter flow)	90.0	94.20	95.5	99.0	I02.5	I03.I
Corrected Total heat/sec	9.03	I3.0	I5.64	I7.55	I8.5	I9.03
Heating surface (c.flow)	I.37	I.37	I.37	I.37	I.37	I.37
Heating surface (P.flow)	I.08	I.08	I.08	I.08	I.08	I.08
h (parallel flow)	.0865	.II56	.I04I	.I4I0	.I485	.I5I5
h (counter flow)	.073I	.I006	.I096	.I265	.I3I8	.I345
h (mean)	.0798	.I08I	.I268	.I337	.I402	.I425
R (mean)	3.5	2.58	2.I95	2.035	I.98	I.95
$\frac{V}{F} = \frac{2g}{005} \frac{}{v}$	I98.0	I04.5	69.5	50.8	4I.25	34.3
F	.00505	.0I86	.0428	.0803	.I2I5	.I76

TABLE. II .contd.

SMALL HIGH SPEED CONDENSER. (Water to water)

SERIES Nº.	(2)					
Number of test.	I	2	3	4	5	6
HOT WATER CIRCULATION						
water lbs/min.	33.6	33.70	33.65	33.65	33.75	33.80
water lbs /sec.	.562	.563	.562	.562	.5625	.564
Inlet temp. °F (A)	I80.	I80.	I80.	I80.	I80.	I80.
Outlet temp. °F (C)	I64.5	I58. 6	I54.0	I52.0	I50.0	I49.8
Temp.difference.	I5.5	2I.4	25.5	28.0	30.0	30.2
Heat B.T.U's/sec.	8.70	I2.0	I4.3	I5.72	I6.88	I7.0
Temp at Bend °F (B)	I72.25	I69.3	I67.25	I66.0	I65.0	I64.9
Average hot water speed			4.25			
COLD WATER CIRCULATION						
water lbs/min.	8.25	I5.25	23.4	3I.75	39.75	46.9
water lbs/sec.	.I37	.258	.39I	.527	.662	.782
water speed ft/sec.	I.045	I.96	2.97	4.03	5.04	5.96
Inlet temp. °F (D)	5I.0	5I.0	5I.0	5I.0	5I.0	5I.0
Outlet temp °F (E)	II0.5	95.3	87.0	8I.0	76.0	74.0
Heat B.T.U's/sec.	8.I6	II.42	I4.I0	I5.80	I6.52	I7.9
t m (parallel flow)	9I.25	98.7	I02.I	I05.0	I07.6	I08.5
t m (counter flow)	86.5	90.6	9I.25	92.2	95.7	95.6
Corrected total heat/sec	8.I6	II.42	I4.08	I5.8I	I7.20	I7.90
Heating surface (c.flow)	I.37	I.37	I.37	I.37	I.37	I.37
Heating surface (p.flow)	I.08	I.08	I.08	I.08	I.08	I.08
h (parallel flow)	.0828	.I09	.I278	.I392	.I420	.I50I
h (counter flow)	.0590	.0920	.II35	.I25I	.I260	.I306
h (mean)	.0759	.I09I	.I206	.I327	.I340	.I400
R(mean)	3.69	2.8I	2.32	2.I0	2.00	I.92
$\frac{V}{F} = \frac{2g}{005} \quad \overline{5v}$	I98.0	I05.6	69.7	5I.4	4I.2	34.7
F.	.00505	.0I96	.0428	.0785	.I22	.I87

T A B L E . II. contd.

SMALL HIGH SPEED CONDENSER.

WATER TO WATER

SERIES No.			(3)			
Number of Test.	I	2	3	4	5	6
HOT WATER CIRCULATION						
water lbs/min.	33.4	33.78	33.65	33.6	34.2	33.75
water lbs/sec.	.557	.5625	.562	.560	.570	.562
Inlet temp. °F (B)	I50.	I50.	I50.	I50.	I50.	I50.
Outlet temp. °F (C)	I39.0	I35.0	I3I.6	I30.0	I27.8	I27.6
Temp. difference.	II.0	I5.0	I8.4	20.0	22.2	22.9
Heat B.T.U's/sec.	6.I25	8.4I	I0.35	II.20	I2.62	I2.9
Temp. at bend °F (B)	I44.5	I42.5	I40.8	I40.0	I38.9	I39.05
Average hot water speed.			4.25			
COLD WATER CIRCULATION.						
water lbs/min	8.20	I5.75	23.6	3I.4	39.75	46.9
water lbs/sec	.I366	.262I	.4I	.524	.657	.78I
Water speed ft/sec.	I.04	2.0	3.0	3.99	5.04	5.96
Inlet temp. °F (D)	5I.0	5I.0	5I.0	5I.0	5I.0	5I.0
Outlet temp. °F (E)	98.0	82.0	77.0	73.0	70.0	68.0
Heat B.T.U's/sec	6.43	8.I4	I0.23	II.5I	I2.49	I3.29
t m (parallel flow)	69.5	72.0	80.4	82.2	85.8	84.0
t m (Counter flow)	65.2	66.2	68.2	72.8	74.3	83.7
Corrected Total heat/sec	6.43	8.I4	I0.0	II.3	I2.6I	I3.29
Heating surface (C.flow)	I.37	I.37	I.37	I.37	I.37	I.37
Heating surface (P.flow)	I.08	I.08	I.08	I.08	I.08	I.08
H. (parallel flow)	.0856	.I045	.II5	.I2I5	.I360	.I465
H. (counter flow)	.072I	.090	.I098	.II35	.I260	.II6
H. (mean)	.0788	.0972	.II24	.I205	.I3I	.I3I7
R (mean)	3.55	2.87	2.5	2.36	2.I8	2.I5
$\frac{V}{F} = \frac{2\ g}{005 \quad 5v}$	I99.0	I03.5	69.0	5I.9	4I.2	34.7

TABLE II. contd.

SMALL HIGH SPEED CONDENSER

WATER TO WATER

SERIES No			(4)			
Number of test	I	2	3	4	5	6
HOT WATER CIRCULATION.						
water lbs/min.	33.6	33.6	33.5	33.7	33.75	33.75
water lbs/sec.	.560	.560	.559	.562	.562	.563
Inlet temp. °F (A)	I30.	I30.	I30.	I30.	I30.	I30.
Outlet temp. °F (C)	I20.0	II8.5	II6.2	II4.2	II2.0	II2.5
Temp differance.	I0.0	II.5	I3.8	I5.8	I8.0	I7.5
Heat B.T.U's/sec.	5.60	6.43	7.75	8.90	I0.I0	9.85
TEmp. at bend °F (B)	I25.0	I24.25	I23.I	I22.0	I2I.0	I2I.25
Average hot water speed			4.25			
COLD WATER CIRCULATION						
water lbs/min.	8.30	I5.75	23.6	3I.70	39.2	46.9
water lbs/sec.	.I366	.262	.4I	.527	.654	.78I
water speed ft/sec.	I.04	2.0	3.0	4.02	4.98	5.96
Inlet temp. °F (D)	5I.0	5I.0	5I.0	5I.0	5I.0	5I.0
Outlet temp. °F(E)	85.0	74.0	72.0	68.0	65.0	64.0
Heat B.T.U's/sec.	4.67	6.05	8.6I	8.95	9.I5	I0.I5
t m (parallel flow)	57.3	63.6	64.0	65.8	67.I	67.5
t m (counter flow)	53.I	58.55	58.5	59.I	58.0	6I.0
Corrected Total Heat/sec	4.67	6.20	7.70	8.90	9.60	I0.I5
Heating surface (c.flow)	I.37	I.37	I.37	I.37	I.37	I.37
Heating surface (p.flow)	I.08	I.08	I.08	I.08	I.08	I.08
h (parallel flow)	.09I5	.093	.II2	.I25	.I325	.I39I
h (couter flow)	.0643	.0785	.0963	.II0	.I205	.I2I5
h (mean)	.0779	.0857	.III	.II7	.I26	.I303
R (mean)	4.0I	3.34	2.64	2.37	2.20	2.I3
$\frac{V}{F} = \frac{2g}{005} \; 5v$	I99.0	I03.5	69.0	5I.5	4I.6	34.7

T A B L E. II. contd.

SMALL HIGH SPEED CONDENSER

WATER TO WATER

SERIES No			(5)			
Number of test.	I	2	3	4	5	6
HOT WATER CIRCULATION						
water lbs/min.	33.6	33.65	33.70	33.70	33.60	33.60
water lbs/sec	.562	.5625	.563	.563	.562	.562
Inlet Temp. °F (A)	I00.	I00.	I00.	I00.	I00.	I00.
Outlet Temp. °F (C)	95.0	93.0	92.5	9I.7	90.5	90.0
Temp. difference.	5.0	7.0	7.5	8.3	9.5	I0.7
Heat B.T.U's/sec.	2.8I0	3.934	4.22	4.66	5.34	6.0I
Temp. at bend °F (B)	97.5	96.5	96.25	95.85	95.25	95.35
Average hot water speed.			4.25			
COLD WATER CIRCULATION						
water lbs/min.	8.0	I5.8	23.6	3I.70	39.0	46.6
water lbs/sec;	.I33	.263	.4I	.527	.650	.780
water speed ft/sec.	I.0I5	2.07	3.0	4.02	4.95	5.96
Inlet temp. °F (D)	5I.0	5I.0	5I.0	5I.0	5I.0	5I.0
Outlet temp. °F (E)	72.0	65.0	64.0	62.0	60.0	59.0
Heat B.T.U's/sec.	2.80	3.7	5.34	5.8	5.85	6.24
t m (parallel flow)	35.3	39.9	40.25	4I.I	42.0	42.6
t m (counter flow)	33.8	36.6	35.7	37.5	37.6	39.6
Corrected total heat/sec.	2.70	3.90	4.8I	5.60	6.02	6.25
Heating surface (c.flow)	I.37	I.37	I.37	I.37	I.37	I.37
Heating surface (p.flow)	I.08	I.08	I.08	I.08	I.08	I.08
h (parallel flow)	.07I5	.0905	.II05	.I26	.I325	.I36
h (counter flow)	.0745	.0783	.0985	.I09	.II7	.II5
h (mean)	.0730	.0844	.I05	.II8	.I25	.I26
R (mean)	4.34	3.42	2.67	2.38	2.24	2.23
$\frac{V}{F} = \frac{2\ g}{005 \qquad 5v}$	204.0	I00.0	69.0	57.5	4I.8	34.7

T A B L E . III .

WATER - WATER

Test No	Speed ft/sec	h(mean)	$\frac{dA_2}{dA_I}$	h_2 cold	h_I hot	$\frac{h_2}{h_I}=\frac{f_2}{f_I}$	$r=\frac{f_2}{f_I}(\frac{dA_2}{dA_I})$	$\frac{I+r}{r}$	$h(\frac{dA}{dA_I})$	f_I	f_2
I	I.045	.0795	I.2	.0796	.I25	.636	.7I4	2.40	.0875	.2I0	.I25
2	I.980	.097I	"	.097I	"	.776	.932	2.075	.I07	.222	.I725
3	2.985	.II35	"	.II35	"	.9075	I.09	I.9I6	.I25	.238	.2I5
4	4.075	.I238	"	.I238	"	.990	I.I89	I.840	.I3I2	.24I8	.2392
5	5.02	.I288	"	.I287	"	I.028	I.23I	I.8I0	.I420	.2570	.2635
6	6.04	.I3I5	"	.I3I5	"	I.05I	I.26I	I.795	.I450	.2605	.280

STEAM - WATER

Speed ft/sec	h(mean) steam	f_2	$\frac{f_2}{h}(\frac{dA_2}{dA})(I+r)$	r	$\frac{I+r}{r}$	f_I
I.045	.II8	.I25	I.I65	.I65	7.06	.598
I.980	.I5I	.I725	I.256	.256	4.9I	.74I
2.983	.I85I	.2I6	I.28I	.28I	4.56	.845
4.075	.224	.2392	I.I73	.I73	6.79	I.25
5.02	.258	.2635	I.I20	.I20	9.33	-----
6.04	.293	.300	I.32	.32	4.43	.94