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Heat Transfer to Turbine Blades

By

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Heat Transfer to Turbine Blades.

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SUMMARY

As a preliminary to the design of cooled turbines, a knowledge of the heat transfer coefficients to turbine blades is required. The results of tests on two typical cascades, together with some other collected results, are here reported.

The two cascades investigated experimentally were of the nozzle and impulse type respectively, tested at a single incidence. The results obtained from these showed that the main factor governing the magnitude of the heat transfer coefficient is the character of the flow; whether it is laminar or turbulent. If the flow is laminar then the result agrees fairly well with that calculated by a method due to H.B. Squire and, if turbulent, the results so far known do not deviate a great deal from a common line.

An attempt was also made to decide on what temperatures and pressures to base the physical quantities, conductivity, viscosity etc., using the correlation of the experimental points as a criterion. The conclusion reached was that thermal conductivity and viscosity, based on blade temperature, and density, based on the mean of inlet and outlet pressure and the mean of blade and gas temperature, gave the best correlation of points.

LIST OF CONTENTS.

1.0. INTRODUCTION.

Much theoretical work has been done on the effect of cooling turbine blades with little experimental data as a basis for calculation. To furnish data for the blade external surface, at least in part, two cascades of typical turbine blades have been tested, by a method involving the determination of the heat transferred to a cooled turbine blade in a hot gas stream.

The first cascade had the blades set at a high stagger as in a nozzle blade row, and the second at a low stagger as in an impulse blade row, with the aim that these two extreme cases would give a rough idea of the properties of turbine blades of all degrees of reaction. The tests on the two cascades covered a range of Reynolds number, Mach number and temperature to determine the effect of each on the heat transfer coefficient, the ranges being:-

Re 0.4 - 5.0 x 10^5

Mn 0.1 - 1.0

T_g 350 - 600°K

2.0. DESCRIPTION OF APPARATUS.

The two cascades were made up of five blades each, all the blades having a common profile, but set at different staggers to give the following properties (see Fig.1).

Cascade type	Nozzle	Impulse
Inlet angle α_1	15°	45°
Outlet angle α_2	70°	45°
Pitch/chord ratio s/c	0.62	0.62
Aspect ratio	2.0	2.0
Stagger ζ	38°	10°
Chord	1.0"	1.0"

The blade form was quite conventional for a British turbine but with rather a thick trailing edge. Although this thickness was a disadvantage from an experimental investigation point of view, it may represent what will probably of necessity be adopted for cooled turbines.

Of the five blades, the middle one only was cooled. This middle blade was hollow, having a skin thickness of about 0.022" and inserted into it was a core which, being supported by small spacing pieces, left a gap of 0.015" round the periphery. The mass flow and temperature rise of water passing through the gap, when hot gas was flowing over the blade, gave the numerator of the heat transfer coefficient. Swirl chambers at inlet and outlet ensured reliable water temperature measurement.

To prevent heat leakage from the cascade walls to the blade, two "guard ring" water jackets were fixed to those walls round the blade ends. The blade cooling water passed through these before and after passing through the blade so that, whatever the inlet and outlet temperature of the water in the blade, the wall temperature was always identical. The cascade walls themselves were lagged so that no appreciable amount of heat was lost from the gas in passing through the cascade. Fig. 2 shows the experimental arrangement.

3.0. EXPERIMENTAL METHOD.

The tests were carried out on two distinct gas supplies, one high pressure and low temperature and the other high temperature and low pressure. This gave a certain amount of freedom in changing the variables, Mn, Re and temperature, independently of one another.

Gas temperatures upstream of the cascade were measured by Chromel Alumel thermocouples to an accuracy of 1 °C on a Foster potentiometer and water temperatures, measured on mercury in glass thermometers, could be estimated to 0.02 °C, an accuracy of within 2% in both cases. Gas mass flow was measured by an orifice plate downstream of the cascade, with various orifice sizes to give reasonable pressure drops. By including a suitable mixing section in the inlet duct a good velocity profile (shown in Fig. 4) was obtained, so that the inlet velocity could be calculated from a central pitot tube without any large inaccuracy due to boundary layer effect. The method of water flow measurement was to time the flow of a given quantity of water, and its circulation was forced by a pump giving 5 lbs/sq.in. pressure at the cascade.

Besides normal pressure tappings in the duct at relevant points, a small traversable pitot tube was fixed on the outlet branch to find the relation between total head loss and heat transfer.

4.0. THEORETICAL CALCULATIONS OF NUSSELT NUMBER.

Two methods of calculation (Refs.1 and 2) were used and both gave results near to that of the experimental value for the nozzle cascade, though the more reliable of the two was considered to be that by H.B. Squire (Ref.1) as it took into account the ratio of temperature and velocity boundary layer displacement thickness, which is a function of the pressure gradient and therefore most important in this case. For both methods the mainstream velocity distribution round the blade was required and this was obtained by an electrolytic tank potential flow investigation.

Knowing the velocity distribution, the Nusselt number could be calculated fairly quickly at each point, and, besides a mean overall Nusselt number, which was the value sought in the tests, a knowledge of the local Nusselt number distribution round the blade was itself valuable, and was necessary to correct the results.

4.1. Correction of results.

Due to difficulty in measuring technique at the time of construction, no provision was made for detail blade temperature distribution measurement. To ease the calculation of results, the mean water temperature was used for blade temperature in the initial plotting. Consequently, two corrections had to be applied to results to give the correct heat transfer coefficient for the external surface of the blade. The first was for the difference between mean water temperature and mean blade temperature, the former known, the latter unknown.

The second correction arose as follows. The flow of water was uniform round the periphery of the blade so that, in the region where the heat transfer was high, the water was heated to a higher temperature. Thus, in the region of high heat transfer the effective temperature difference between the gas and the blade is reduced by virtue of the increased water temperature. The method of evaluation of the correction for this effect is shown in Appendix I.

4.2. Effect of Radiation.

The correction for radiation was made assuming that the blade was a completely black body totally enclosed by surfaces at gas temperature. The maximum amount of heat radiated to the blade was about 10% of that received by convection but in most cases was only about 3%.

The assumption that the emissivity of the blade was unity is questionable but it is likely considering the blade surface condition that the emissivity was not lower than 0.9, so that the maximum error due to this correction was of the order of 1%.

5.0. RESULTS.

The test results are presented in two ways; the first, Nusselt number against Reynolds number is the more usual, and the second, $\Delta H/H$ against Reynolds number. $\Delta H/H$ can be defined as the ratio of amount of heat transferred to a blade to the amount of available heat passing through one blade pitch, and its significance lies in the analogy between it and the blade loss coefficient.

In considering the available heat or temperature difference upon which the heat transfer depends, the higher temperature was taken as the stagnation temperature of the gas (Ref.3 & 4.) defined as $T_{st} = T_g + 0.85\,\theta_v$ where θ_v is the temperature equivalent of the velocity outside the boundary layer. For these results a mean value of θ_v was taken, then:-

$$Re = \frac{\rho V c}{\mu}$$

$$\frac{\Delta H}{H} = \frac{\text{heat transferred}}{\rho_2 V_2 S \cos \alpha_2 \, K_p \, (T_{st} - T_B)}$$

$$Nu = \frac{\text{heat transferred}}{A_B \, (T_{st} - T_B)} \cdot \frac{c}{\lambda} = \frac{hc}{\lambda}$$

5.1. Interpretation of experimental data.

For each individual test the value of h for the blade could be found, but the presentation of this quantity, plotted non-dimensionally against Reynolds number, involved a choice of representative temperatures for each physical property μ, λ and ρ. The possibilities were infinite but, in this report the choices investigated are limited to four, the stagnation temperature T_{st} ($= T_g$ at low velocities), the blade temperature T_B, the mean of these two T_m, and the static temperature at outlet. The quantities μ, λ and ρ are suffixed accordingly to give, for example, μ_{st} or $_g$, μ_B, μ_m and μ_2.

The choice of μ and ρ in the evaluation of Re and λ in the evaluation of Nu was decided by two factors. (a) which was logically the more likely and (b) which gave the best correlation of test results.

Viscosity μ.

The viscosity is significant only in that it has an effect on the temperature gradient at the blade surface, because $h = \lambda\,(dT/dn)_{n=0}$ where n is normal to the surface. In the case where the flow is not near the breakaway point the maximum velocity change and therefore the maximum shear occurs at the boundary. It seems reasonable to suppose therefore that under these conditions the viscosity at the surface has its greatest affect on the temperature gradient.

The use of μ_B is common in recent German work but, in other work, both μ_g and μ_m seem to have been used, μ_g where the viscosity is assumed constant and μ_m at high temperature ratios. No definite rule is evident however.

Although strong arguments for the use of μ_B are lacking, it seems probable that μ_B is the correct one to take and it is recommended for future results.

Conductivity λ.

As quoted above h = $\lambda\ (dT/dn)_o$ and, as h is the quantity we know, it must be divided by the λ appropriate to the expression to yield the Nusselt number [= $c\ (dT/dn)_o\ (T_{st} - T_B)$] . It seems reasonable that the λ in the expression for h is λ_B and thus this is the one to use in the evaluation of h.

The choice of λ in previous work seems just as indiscriminate as for μ but, again, recent practice seems to be the use of μ_B, in most cases, though not in all.

Density ρ.

For the viscosity and conductivity there are arguments for use of blade temperature, but for the density term there are no such arguments, so the choice of relevant temperature and pressure was ruled solely by correlation of test results. In other words, the temperature and pressure used to evaluate the density were the ones which gave best distribution of test points. These proved to be the mean of inlet and outlet static pressures $\bar{p}_{1,2}$, and the mean of the blade and stagnation temperature T_m. The pressure $\bar{p}_{1,2}$ has some significance in that it takes into account the acceleration or retardation of flow in the blade passage.

Velocity V.

The velocity term in the Reynolds number was the mean outlet velocity from the cascade. If $\bar{V}_B$ the mean mainstream velocity round the blade is used, then it may produce better correlation of results when, in future widely differing profile shapes are investigated. In this case however, the values for V_B were based on potential flow theory, and were in any case, very little different from V_2, so it was not considered worth while to use the term.

To illustrate these effects a single set of experimental results was replotted in four different ways for the nozzle cascade, as given below.

Method.	1	2	3	4
μ	μ_B	μ_{st}	μ_B	μ_B
λ	λ_B	λ_{st}	λ_B	λ_B
ρ ∝	$\bar{P}_{1,2}/T_B$	$\bar{P}_{1,2}/T_{st}$	$\bar{P}_{1,2}/T_m$	$\bar{P}_{tot_2}/T_m$
V	V_2	V_2	V_2	V_2

Method 1 and 2 are self-consistent in that the same temperature is used throughout. Method 3 gives the best correlation of results with the most reasonable set of quantities in the light of the above arguments. This combination is the best at present available and should be used failing more reliable information. Method 4 is included so that the change produced by using P_{tot_2} can be illustrated. P_{tot_2} is more readily available to the designer than the static pressures so that knowledge of the Nusselt number on the basis should prove useful (Fig. 8).

Subsequently in this report the use of the phrase "plotting method 1, 2, 3 and 4" will refer to the use of one of these four combinations of physical quantities.

5.2. Discussion of Final Results.

Figs. 6 and 7, which compare the methods of calculation for one set of experimental data, contain also the theoretical curve given by the method of Ref.1. This curve gives a common basis for comparison. The most obvious factor is that, although the different methods of plotting produce a considerable change in distribution and scatter of points, the maximum ordinate displacement of the mean lines through these sets of points is about 15%. In other words, if a single curve of Nusselt number against Reynolds number were taken for design purposes, and any one of the first three sets of quantities μ, λ and ρ used to calculate h, the result would not be in error by more than 15%.

From Fig.6 it is evident that the temperature ratio effect is not eliminated by the use of the stagnation temperature but, owing to the apparent large range of Reynolds number covered, the points fall into two fairly well defined lines. Change of datum temperature to T_B however, shortens the range and bunches the points together. (Fig.6) The mean line is still almost identical with the theoretical curve and this fact is not insignificant since the theory has been proved almost correct for a flat plate and cylinder.

There are two curves plotted by method 3 (Fig. 7) one with points identified by temperature ratio and the other by Mach number. The effect of these two quantities on the results plotted in this way is either absent or small enough to be hidden by point scatter. On these two graphs the dotted line shows the curve when the correction described above has been applied. This is the mean curve which will be used to compare with other results. Fig.8 would be identical with the previous two figures but for the fact that P_{tot_2} is used in the Reynolds number instead of P_{s_m}. P_{tot_2} is more readily available during design so that knowledge of the difference produced by use of P_{tot} is useful.

The curve of $\Delta H/H$ is plotted also by method 3 (Fig.8) and has a definite temperature ratio effect, so it is evident that good correlation of Nu does not necessarily mean a good correlation of $\Delta H/H$ with the same Reynolds number. The temperature ratio effect can be removed by plotting against a Reynolds number using $\bar{\rho}_{12}$ instead of $\bar{\rho}_m$.

The results for the impulse cascade, having a more restricted temperature ratio range do not exhibit any of its effects, the Nusselt number and $\Delta H/H$ number are therefore plotted according to method 3 (Fig.9). Again no Mach number effects are evident for the cascade. It is known that a certain amount of breakaway occurred on the back of this blade. The extent of the breakaway at a low Reynolds number was 30% of the top surface and probably less at higher Reynolds numbers. The effect of this appears to be a reduction of the heat transfer to the blade, but by a small amount only, the large majority of the heat being transferred through the leading half of the blade (see Fig.5).

The graph of loss coefficient for the nozzle cascade at almost constant Mach number is shown in Fig.10. It was hoped to obtain a relation between this and the quantity $\Delta H/H$ but this proved impossible owing to the large loss contributed by the thick trailing edge, which in no way contributes to the heat transfer.

The equations of the mean curves of Nusselt number for each cascade as shown in Figs.7 and 9 are:-

<u>Nozzle.</u>

$$Nu = 0.756\,(Re)^{0.49}$$

<u>Impulse.</u>

$$Nu = 0.169\,(Re)^{0.66}$$

5.3. <u>Comparison with theoretical and other results.</u>

Fig. 11 comprises all the relevant heat transfer data available to the authors. The table at the bottom gives the origin of the results and the method of plotting.

Considering first the nozzle result and the adjacent curves, the slope of the experimental curve for the blade seems to indicate that the flow was laminar. This is to be expected in a cascade with accelerating flow of this nature. The value of the Nusselt number as it is plotted is rather higher than the theoretical prediction although similar curves for the flat plate lie closer together. This throws some doubt on the plotting method since as pointed out above, it is possible to plot the results so that the value is about equal to the theoretical one. However, the good correlation of test points gained in this case is considered to justify the method. The experimental curve for the flat plate with laminar flow does not really exist at these Reynolds numbers, which are above the critical value, but the extrapolated curve agrees very well with the theoretical curve. The fact that the critical Reynolds number of the nozzle blade is above this range can be explained in terms of the favourable pressure gradient existing in the latter.

The impulse cascade result agrees well with the other four curves of the higher group. One of these, the flat plate curve, is definitely known to be for turbulent flow conditions and as the streamlined body and cascades (1) and (3) have zero overall pressure gradient it is reasonable to suppose that all four results are for substantially turbulent flow. The highness of the curve for cascade 1 may be due to the relatively poor nose shape it has compared with the other bodies which should give rise to a much earlier transition to turbulent flow.

Cascade (2) has a considerable degree of reaction and yet gives almost as high a Nusselt number as the substantially fully turbulent cases which have no flow acceleration. This cannot be readily explained as it would be expected to be much nearer the laminar flow cases than it actually is. On the other hand the Nusselt number values for the circular cylinder seem low. It may be because the representative length to perimeter ratio is so much lower than for the cascade blades. If they were plotted on a basis of $\frac{\text{perimeter}}{\pi}$ for the representative length they would be displaced so as to bring them in the same region as the cylinder curve.

The approximate shape of these cascades is as follows:-

Cascade	1	2
Inlet angle α_1	56°	42°
Outlet angle α_2	87°	60° (approx)
Pitch/chord, ratio s/c	0.68	0.59
Aspect ratio	2.8	3.1
Chord c	4.0	2.5"
Stagger	+7°	+30°

5.4. Prediction Possibilities

From the above results it seems clear that for a laminar flow blade a fairly accurate prediction of the overall Nusselt number can be made by the theoretical method (Ref.1.) if the velocity distribution outside the boundary layer is known. The question is therefore on which blades is the flow laminar. This cannot be said with certainty as it depends on pressure gradient, temperature ratio, surface finish, to some extent the previous history of the gas, and of course the operating Reynolds number. Nozzles and high reaction blades are of course much more likely to give laminar flow than impulse blades for instance. Turbulent flow cases do not lend themselves so easily to calculation and no reliable method is as yet available.

An alternative approach for both laminar and turbulent cases is by way of Reynolds analogy. This states that if the Prandtl number = 1, then for laminar flow the shape of the velocity and temperature boundary layers is the same, giving rise to the relation:-

$$\frac{\Delta H}{H} = \frac{1}{2}\left(\frac{\overline{\omega}}{\frac{1}{2}\rho V_2^{\,2}}\right)$$

$\overline{\omega}/\frac{1}{2}\rho V_2^{\,2}$ being the mean total head loss through cascade. The analogy as put forward applies to laminar flow only of course but with suitable coefficients its application to both laminar and turbulent flow may prove useful.

6.0. CONCLUSIONS

As far as the experimental techniques is concerned the method used is satisfactory for high temperature work. To obtain a higher standard of accuracy, it would be necessary to measure surface temperature, a difficult process.

The results given by the nozzle cascade agree fairly well with the theoretical calculation, which will probably hold for say the first stage nozzles and high reaction blading at design incidence. For other cases where the mean pressure gradient is zero or negative the value of Nusselt number will probably not be far from the turbulent group of results in Fig. 11.

The suggested relevant temperature for viscosity and conductivity is T_b the blade temperature. For calculating the density, the mean of inlet and outlet static pressure divided by the mean of blade and stagnation temperature gives the best correlation of results. No temperature ratio (T_{st}/T_B) effect on Mach number effect is evident in the experimental results plotted in this way.

Acknowledgments.

The authors are indebted to Mr. T.J. Hargest at the N.G.T.E. for his work in determining velocity distributions round the blades by electrolytic tank potential flow investigation.

List of Figures.

List of References.

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Notation.

α_1	gas inlet angle.	degrees
α_2	gas outlet angle.	degrees
V	gas velocity	ft/sec.
M	mass flow	lbs/sec.
ρ	density	lbs/cu.ft.
μ	absolute viscosity	lbs/ft.sec.
C_p	specific heat of gas at constant pressure.	
c	blade chord	ft.
S	blade pitch	ft.
1	blade length	ft.
ζ	blade stagger	degrees
t_w	cooling passage width	ft.
t_b	blade skin thickness	ft.
T	temperature	°K abs.
s	blade perimeter	ft.
ΔT	temperature difference causing heat flow	°K
θ_v	temperature equivalent of velocity	°K
P_s	static pressure	lb/sq.ft.
P_{tot}	total head pressure	lb/sq.ft.
ΔH	heat transferred to blade	CHU/sec.
H	available heat passing through one blade pitch	CHU/sec.
λ	thermal conductivity	$\frac{\text{CHU}}{\text{sq.ft.degrees/ft.}}$
A	blade surface area	sq.ft.
d	representative length	ft.
h	heat transfer coefficient $\frac{\Delta H}{A.\Delta T}$	$\frac{\text{CHU}}{\text{ft}^2\text{.degrees}}$
x	distance from stagnation point round blade perimeter	ft.
y	distance along blade length	
Nu	Nusselt number $\frac{hd}{k}$	
Re	Reynolds number $\frac{\rho Vd}{\mu}$	
Pr	Prandtl number $\frac{Cp.\mu}{k}$	
Mn	Mach number V/a where a = velocity of sound	

Suffixes.

g	gas
w	water
B	blade surface
1	inlet to cascade
2	outlet from cascade
0	water inlet conditions, conditions at water inlet end of blade ($y = 0$)
st	stagnation
$\bar{z}_{1,2}$	mean of z at 1 and 2.
∞	free stream conditions
m	mean of stagnation and blade conditions

Appendix I.

In the absence of detailed blade surface temperature measurements, the values of h given by the experimental results alone were in terms of $T_{st} - T_W$ and not $T_{st} - \overline{T}_B$ which is the temperature difference on which the heat transfer to the surface depends. An estimate of $\overline{T}_B$ must therefore be made using two additional sources of information. Firstly a distribution of Nusselt number must be assumed according to Ref.1 and secondly, values of metal to cooling water Nusselt number must be taken.

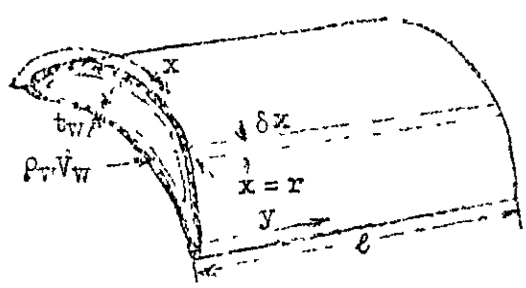
From the gas to blade Nusselt number distribution a mean value, $\overline{Nu}$, is compared with a value calculated from mean water temperature, thus arriving at two values of Nusselt starting from a single Nu distribution.

$$Nu_{true} = \overline{Nu}$$

$$Nu_{\text{as derived in experiment}} = \frac{M_W\, \Delta T_W}{A_B\,(T_{st} - \overline{T}_W)} \cdot \frac{c}{\lambda_B}$$

The method ignores the effect of chordwise and lengthwise heat flow and also the heat resistance of the metal skin thickness t_B but these effects are negligible.

Solution.



h_g = (h) from gas to blade = f (x) only

h_W = (h) from blade to water = constant

h_{tot} = (h) from gas to water = f(x) only

To satisfy continuity

$$\frac{1}{h_{tot}} = \frac{1}{h_g} + \frac{1}{h_W}$$

and the temperature rise of the water flowing in an element of blade δx at a position x = r is given by

$$\Delta T_{W_r\,1} = \frac{1}{\rho_W V_W t_W} \int_0^1 (T_{st} - T_W)\, h_{tot_r}\, dy = T_W - T_{W_0}$$

the solution of this equation is

$$\Delta T_{W_r\,1} = (T_{st} - T_{W_0})\left(1 - e^{\frac{-h_{tot_r}\,1}{\rho_W V_W t_W}}\right) \qquad (1)$$

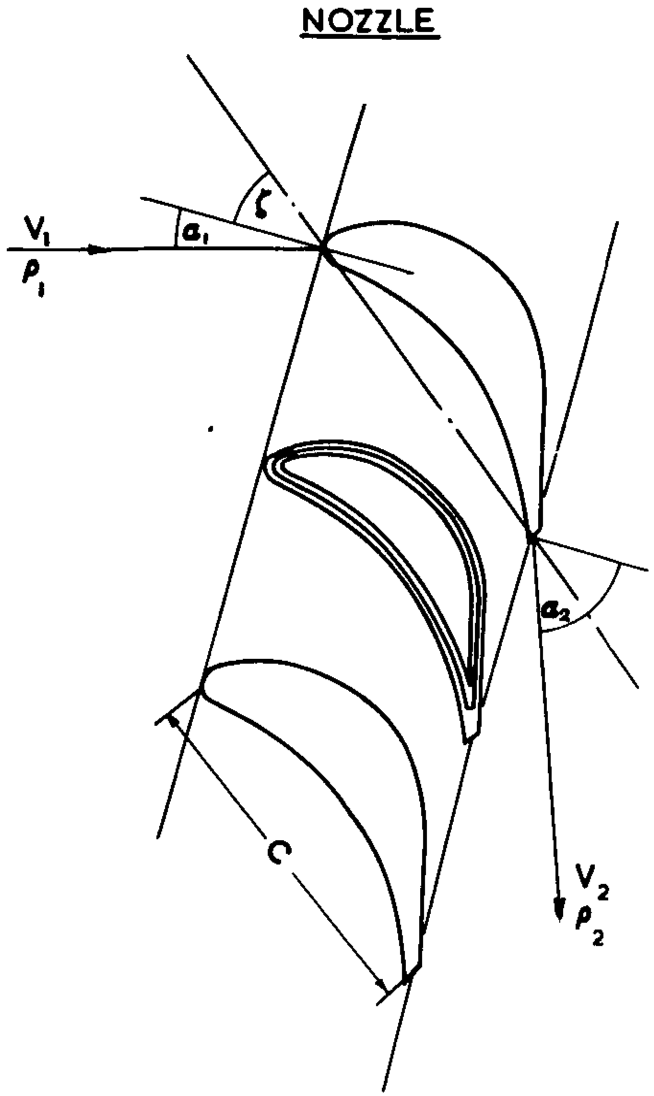
From this equation the temperature rise of the water at any value of x can be found and the mean of these over the perimeter of the blade gives $\overline{\Delta T}_W$ the quantity measured in experiment.

Then:-

$$\overline{Nu} = \frac{M_W\, \overline{\Delta T}_W}{\left(T_{st} - T_{W_0} - \frac{\overline{\Delta T}_W}{2}\right) S_B .\, 1} \cdot \frac{c}{\lambda_B}$$

This value of the Nusselt number is lower than the true value $\overline{Nu}$ by about 2.3% at the low end of the Reynolds number range and 8 - 10% at the higher end, depending to a small extent on the gas temperature. The corrections for the two experiment curves of Nu as shown in Figs. 7 and 9.

CASCADE DETAILS.



α_1	AIR INLET ANGLE
α_2	AIR OUTLET ANGLE.
ζ	STAGGER ANGLE.
V_1	AIR INLET VELOCITY
V_2	AIR OUTLET VELOCITY.
S	BLADE PITCH
C	BLADE CHORD
ρ	DENSITY

ℓ LENGTH OF BLADE = 2·0"

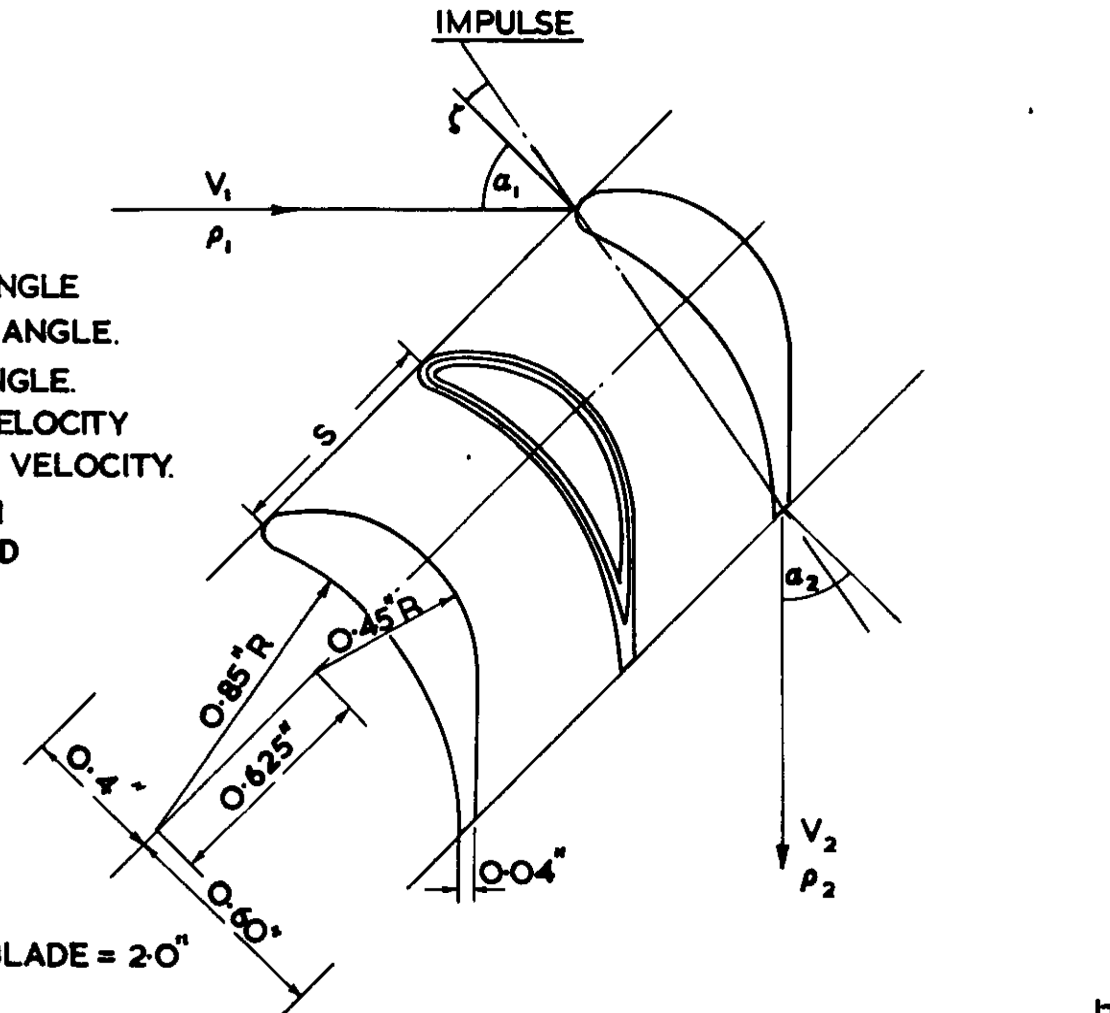


FIG. 1.

TURBINE BLADE HEAT TRANSFER INVESTIGATION APPARATUS.

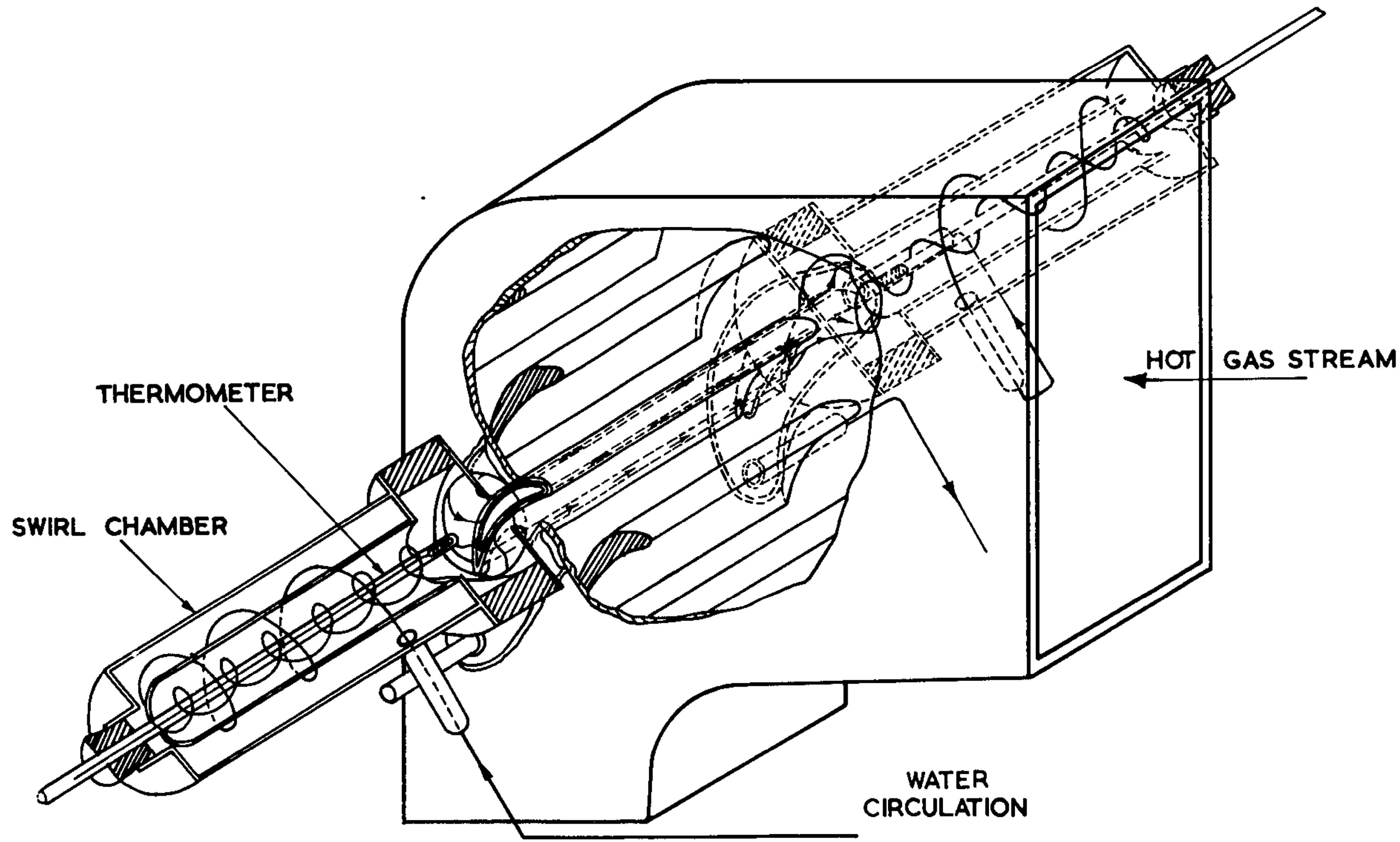
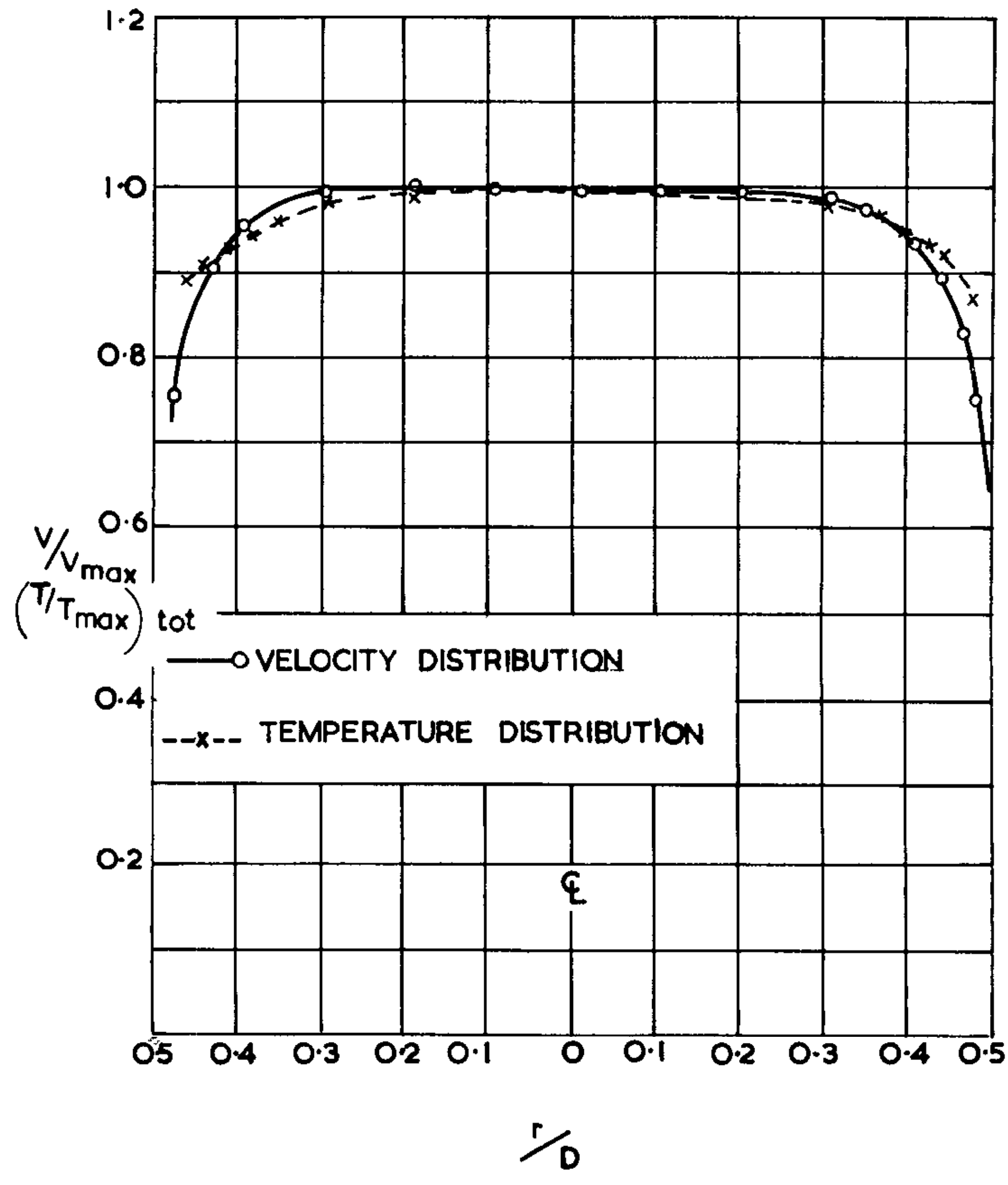


FIG. 2.

VELOCITY AND TEMPERATURE PROFILE IN
5·0" DIAM PIPE AT ENTRY TO TEST RIG



THEORETICAL NUSSELT NUMBER DISTRIBUTION FOR NOZZLE CASCADE (ASSUMING LAMINAR FLOW)

NUSSELT No. vs x/c

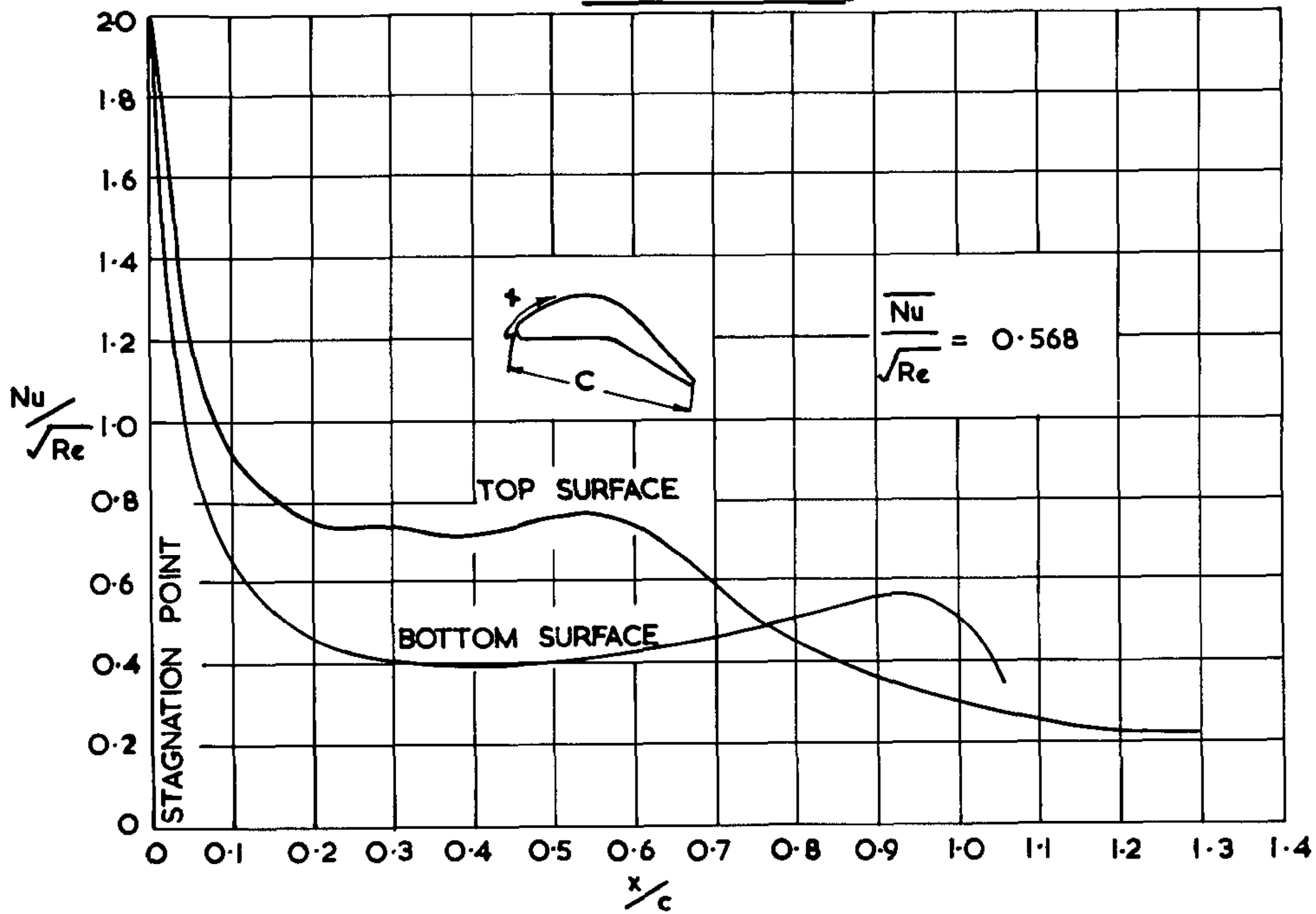


FIG 4

THEORETICAL NUSSELT NUMBER DISTRIBUTION FOR IMPULSE CASCADE (ASSUMING LAMINAR FLOW)

NUSSELT No. vs x/c

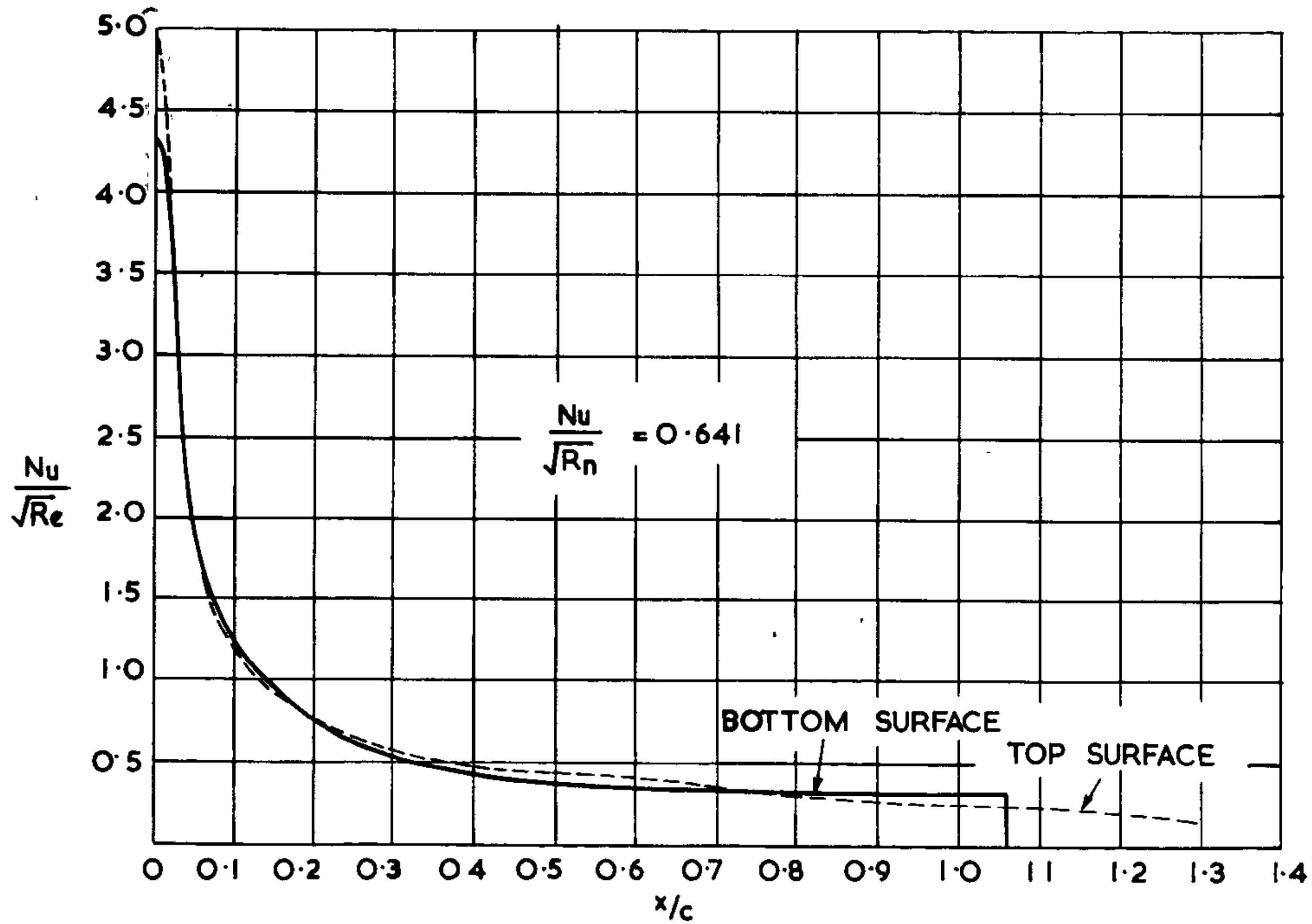
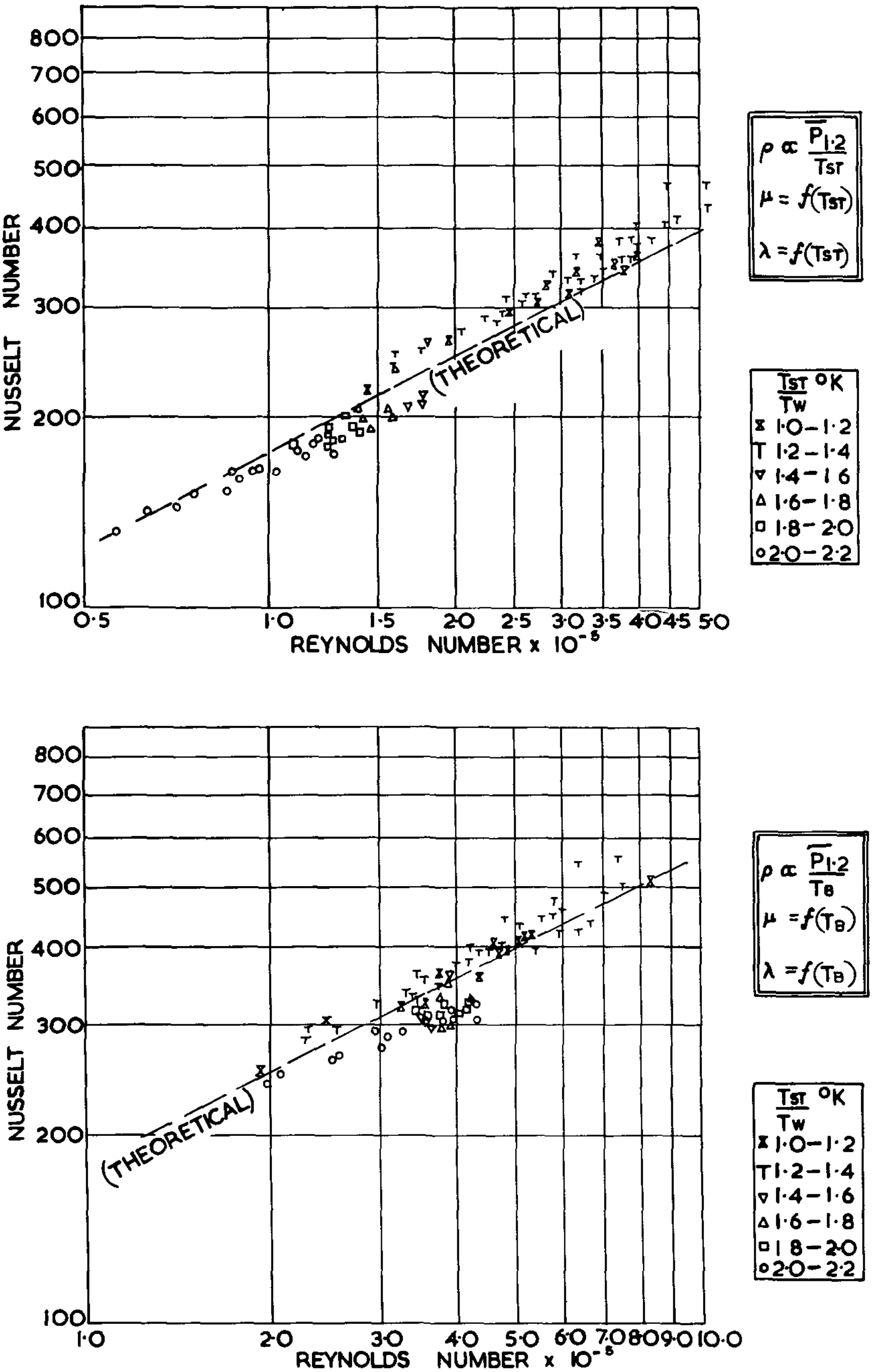


FIG. 5

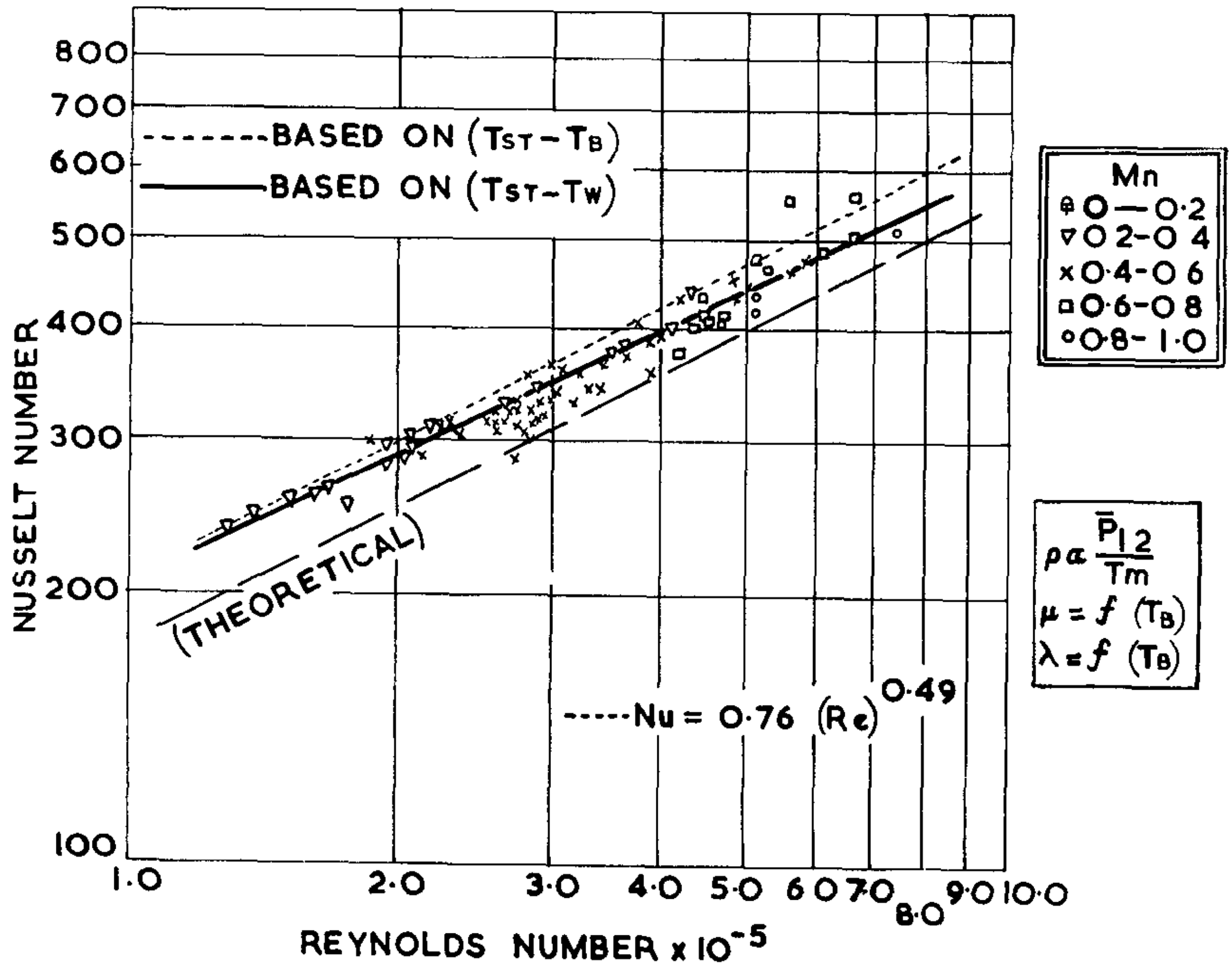
FIG. 6

NOZZLE CASCADE

EFFECTIVE TEMPERATURE COMPARISON



NOZZLE CASCADE
MACH NUMBER AND TEMPERATURE RATIO EFFECT.



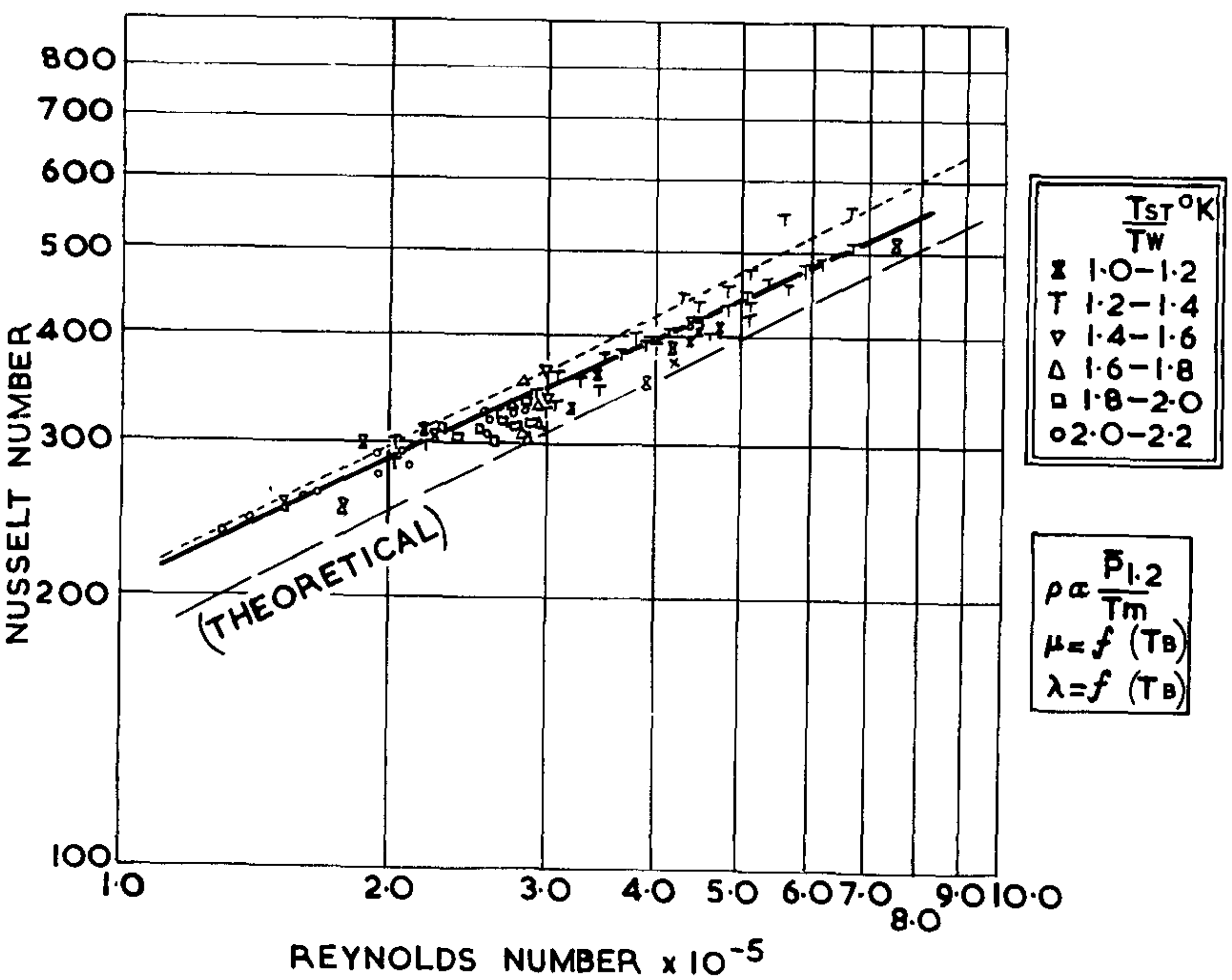


FIG. 8.

NOZZLE CASCADE

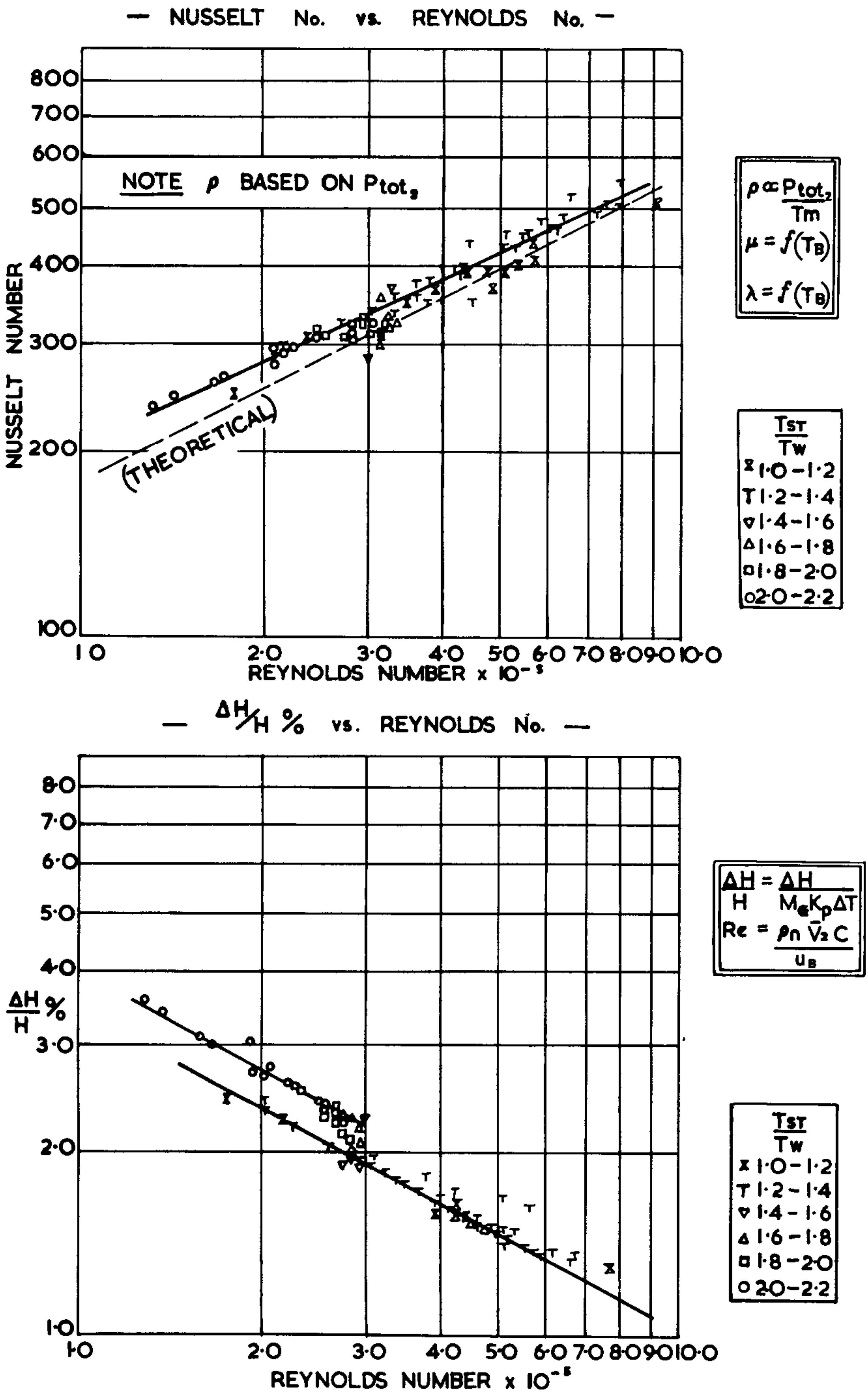
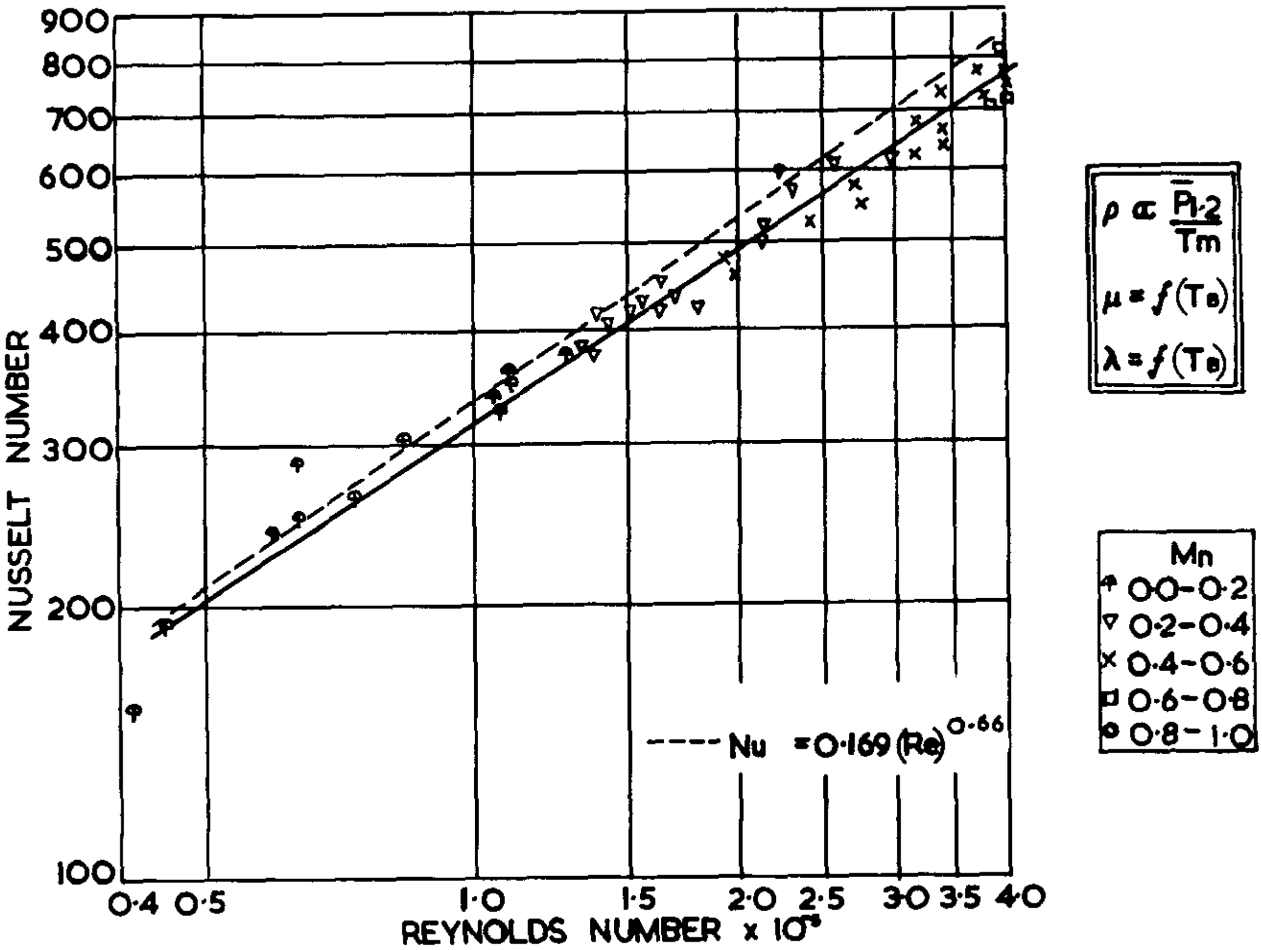


FIG. 9.

IMPULSE CASCADE

NUSSELT No. vs. REYNOLDS No.



ΔH/H % vs. REYNOLDS No.

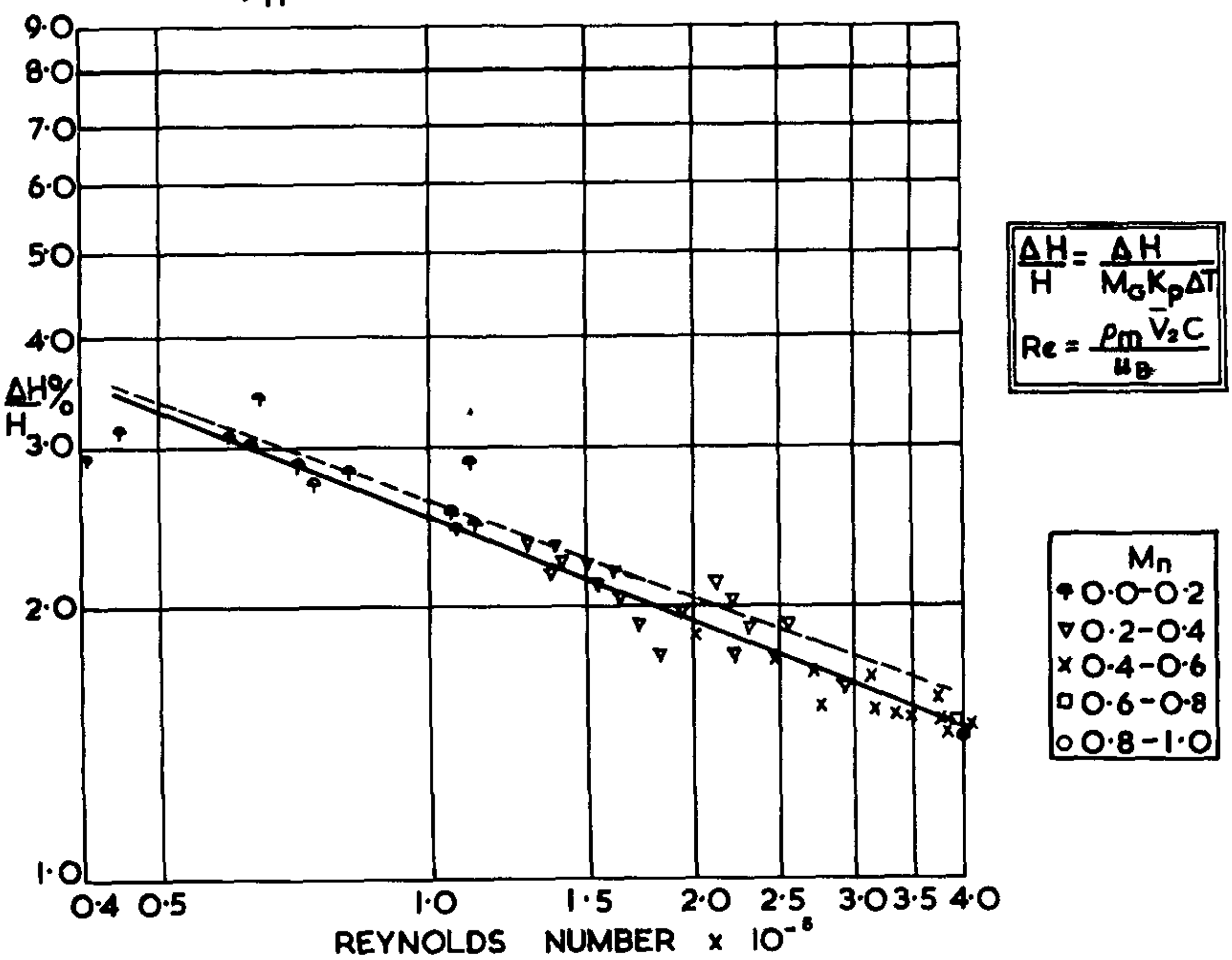
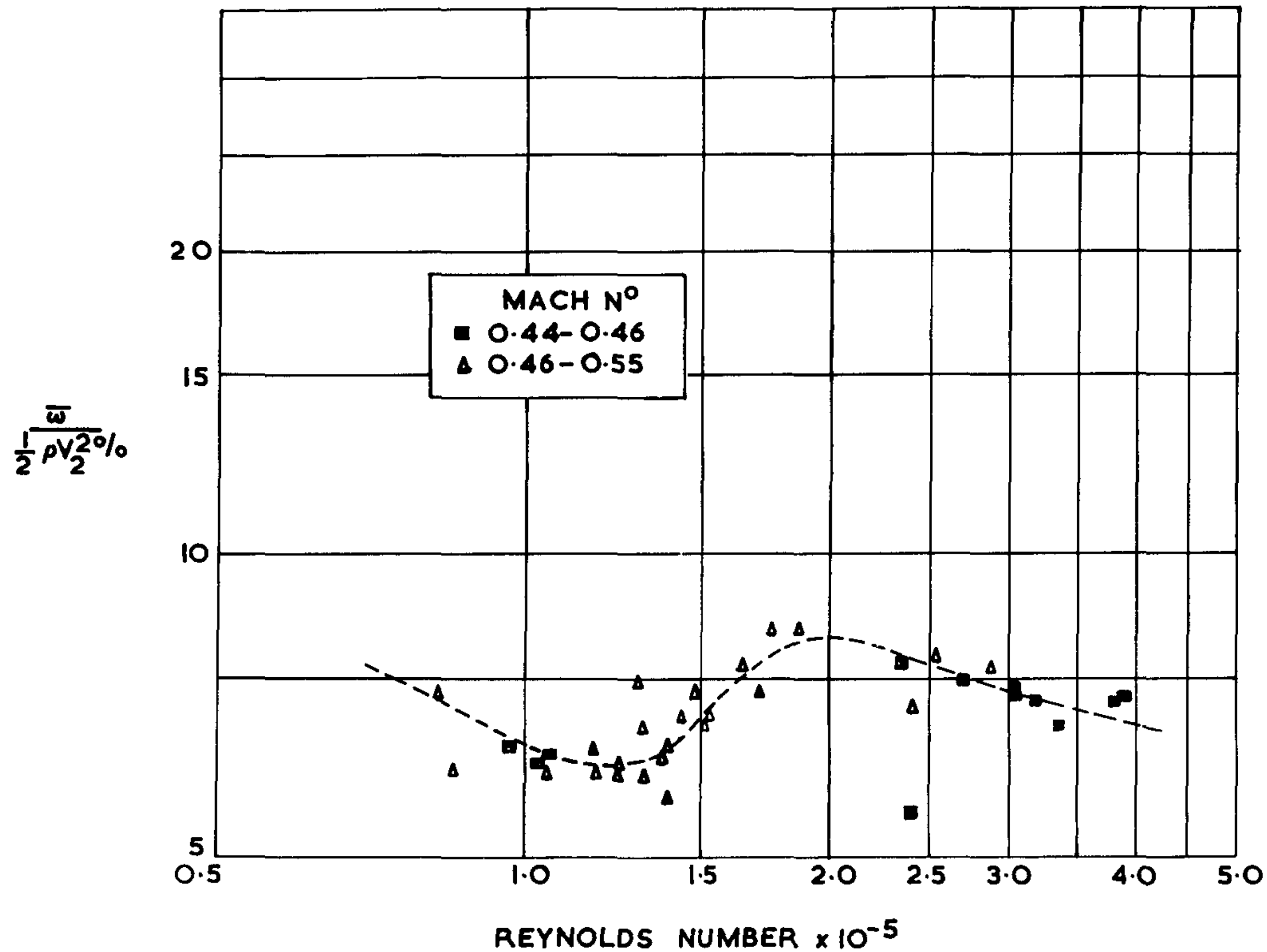


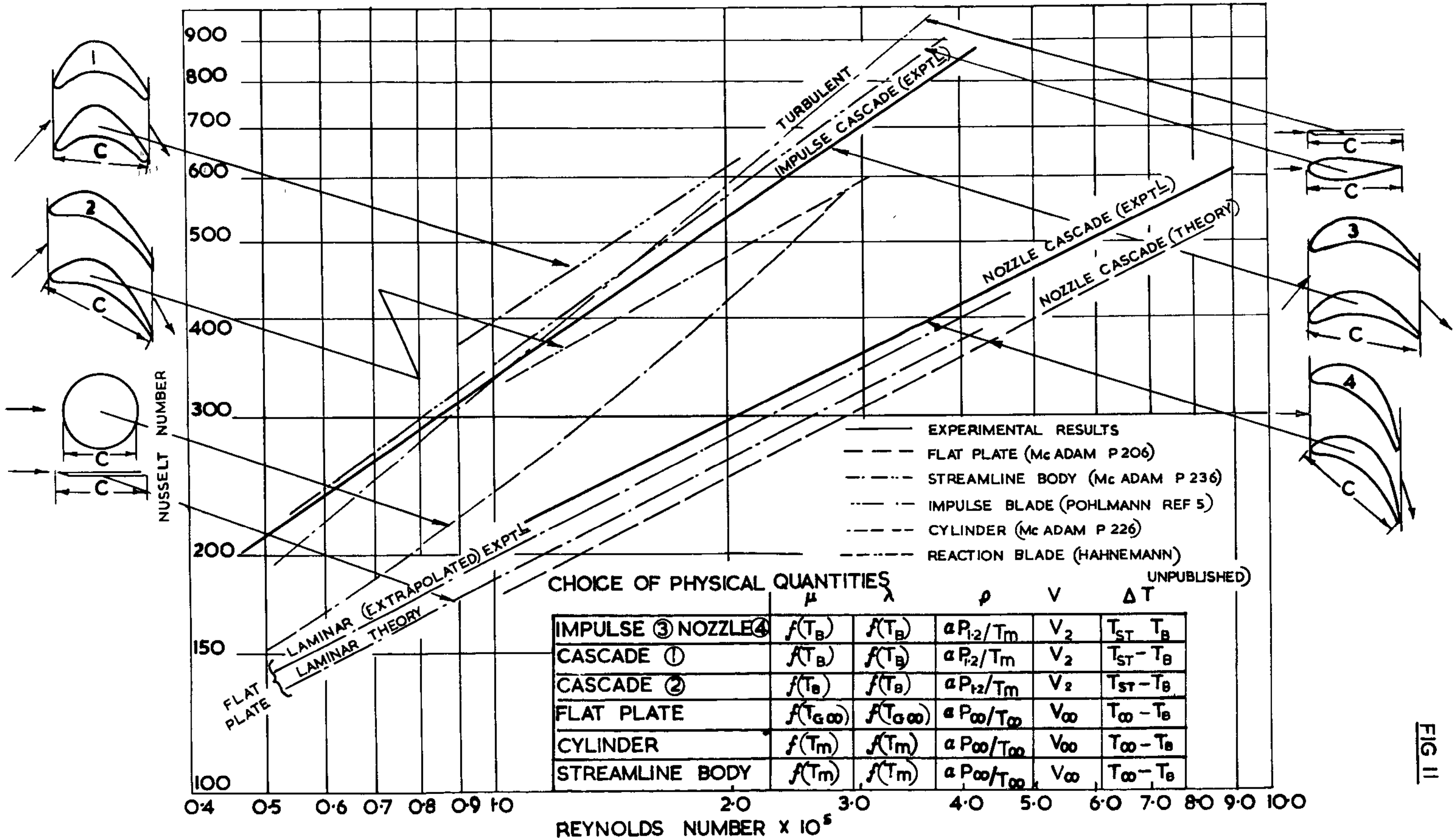
FIG. 10

NOZZLE CASCADE

BLADE PROFILE LOSS COEFFICIENT



COLLECTED HEAT TRANSFER RESULTS



CHOICE OF PHYSICAL QUANTITIES

	μ	λ	ρ	V	ΔT
IMPULSE ③ NOZZLE ④	$f(T_B)$	$f(T_B)$	$\alpha P_{1\cdot2}/T_m$	V_2	$T_{ST} - T_B$
CASCADE ①	$f(T_B)$	$f(T_B)$	$\alpha P_{1\cdot2}/T_m$	V_2	$T_{ST} - T_B$
CASCADE ②	$f(T_B)$	$f(T_B)$	$\alpha P_{1\cdot2}/T_m$	V_2	$T_{ST} - T_B$
FLAT PLATE	$f(T_{G\infty})$	$f(T_{G\infty})$	$\alpha P_\infty/T_\infty$	V_∞	$T_\infty - T_B$
CYLINDER	$f(T_m)$	$f(T_m)$	$\alpha P_\infty/T_\infty$	V_∞	$T_\infty - T_B$
STREAMLINE BODY	$f(T_m)$	$f(T_m)$	$\alpha P_\infty/T_\infty$	V_∞	$T_\infty - T_B$

FIG 11

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