C.P. No. 306
(17,933)
A.R.C. Technical Report

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MINISTRY OF SUPPLY

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CURRENT PAPERS

Tests on Rough Surfaced Compressor Blading

By

R. C. Turner and Hazel P. Hughes

LONDON : HER MAJESTY'S STATIONERY OFFICE

1956

PRICE 5s 6d NET

U D.C. No. 621.438.031.3-226.2

C.P. No. 306

Memorandum No. M. 241

March, 1955.

NATIONAL GAS TURBINE ESTABLISHMENT

Tests on rough surfaced compressor blading

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R. C. Turner

and

Hazel P. Hughes

SUMMARY

Six stages of medium stagger free vortex blading, with rough surface finish, were tested in the N.G.T.E. 106 compressor. Overall characteristics were determined at various speeds and compared with those of similar blading with the conventional smooth finish.

It was established that appreciable losses in temperature rise coefficient and efficiency were exhibited, and hoped-for gains in performance at low Reynolds numbers were not realised. There was no change in the general shape of the characteristics at any speed.

The condition of the boundary layer on a blade of the fifth stage stator row was investigated by means of pitot traverse and sensitive film techniques. No appreciable difference was found in this respect between the two sets of blades.

CONTENTS

APPENDICES

ILLUSTRATIONS

1.0 Introduction

The 106 axial flow compressor (Reference 1) is a low speed multi-stage experimental machine intended for the comparison of the aerodynamic performance of various designs of blades. Blade manufacture, for this compressor as for others, tends to be a costly process, and it is obvious that useful economies could be made if blades cast by the lost wax process were to be used without subsequent surface finishing. It was realised that the imperfection of finish would result in a change of performance, but it was hoped that this would not be sufficient to invalidate comparative tests (i.e. between different blade designs with similar surface finish) such as are normally performed on this compressor. The present tests were intended to investigate the change in performance, due to the surface imperfections, of blades of conventional design.

An important secondary aspect of the tests related to the effect of surface roughness and excrescences at low Reynolds numbers. It is well-known (see for instance Reference 2) that under certain conditions of low Reynolds number, the profile drag of a body in smooth fluid flow can be reduced by the artificial induction of turbulence in the boundary layer; the explanation being that separation of the otherwise laminar boundary layer is prevented or delayed, with a resultant reduction of form drag, this reduction being greater than the concomitant increase of skin drag. Two methods of inducing turbulence in the boundary layer are the use of a trip wire (of dimensions comparable to the boundary layer thickness) set on the surface in a suitable position perpendicular to the direction of flow; and the roughening of the surface. The latter method has been the subject of a patent (Reference 3).

The present blades could be regarded as using both of these methods. In the manufacturing process, the wax model blades were cast in three-part dies; the die joints left flashes in the wax, extending the whole length of the model blades on either side of but near the leading edge, and along the trailing edge. These flashes were reproduced on the actual blades. It was thought that the flash near the leading edge on the convex surface might act as a turbulence inducing wire, and that the surface roughness of the blades might have the same effect.

It would have been possible to have used two-part dies, with the resulting flashes on the leading edge and on the trailing edge, but it was thought that this would have resulted in poorer performance at off-design incidences.

If the supposedly beneficial effects of the convex surface flash and roughness were in fact realised, the resultant effect at low Reynolds numbers on the overall compressor performance would probably be an increased efficiency, at any rate at low flow coefficients, and possibly a decrease in the surge flow coefficient resulting from an increased blade stalling incidence.

For convenience, the rough-surfaced and conventional blades will be described as "rough" and "smooth" respectively

2.0 Description of apparatus

2.1 The compressor

The 106 compressor is described in Reference 1. It is a low speed multi-stage compressor of constant annulus dimensions and a diameter ratio of 0.75. The blade height is 2.5 in. and the mean diameter is 17.5 in. At the normal running speed of 3,000 r.p.m., the mean blade speed is 229 ft./sec., and the blade Reynolds number based on this speed is 1.3×10^5. A general arrangement is shown in Figure 1. In these tests, six stages were used with an axial spacing between the blade rows of about 0.9 in., the blade chord being approximately 1.1 in. A new feature was incorporated in stage five; a segment, holding two adjacent blades of the stator ring, was made so that it could be removed and replaced during running; this enabled the two blades to be treated with a flow-sensitive film and replaced for a short time, the object being to obtain an indication of the state of the blade boundary layer. The segment is shown in Figure 2. In practice, two interchangeable segments were used, holding rough and smooth blades respectively.

2.2 The blades

2.2.1 Design and general finish

The blades were of medium stagger free vortex design and C4 section. Design details are given in Appendix 1. The smooth blades were of aluminium alloy cast in metal dies, with subsequent hand finishing and anodising. The rough blades were cast by the lost wax process in H.R. Crown Max, a smooth rotor blade and a smooth stator blade being used as masters.

The die flashes on the rough blades were rather variable, both on any particular blade, and from blade to blade. Approximate dimensions are indicated in Figure 3. The two long nose flashes extended nearly the whole length of the blade, on either side of the leading edge and about 0.1 in. from it; near the tip they turned through 90° and merged in a short fin-like flash of irregular shape situated on the leading edge. Typical long flashes were 0.001 to 0.002 in. high and 0.010 in. wide; although on some blades it is probable that these dimensions were exceeded, while on others the flash was barely distinguishable. These dimensions were found by cutting sections of a typical blade and optically projecting them. On about 5 per cent of the rough blades the leading edge itself was very rough and irregular. The trailing edge flash was a thin jagged fin; on some blades it was negligibly small.

The surface finish was also a somewhat variable quantity on the rough blades. Typical graphs derived from "Talysurf" records are shown on Figure 4 for both rough and smooth blades.

The radius of the fillet where the blade proper joins the root was about 0.2 in. as compared with the nominal value of 0.125 in. for the smooth blades.

Photographs of rough and smooth stator blades are shown in Figure 5. The smooth blade has a mottled appearance which reduces the contrast in the photographs.

2.2.2 Setting and accuracy

Both rough and smooth blades were set at the required stagger angle by means of a straight edge set against the concave side of the blade at mean diameter. The same setting angles were used for both rough and smooth blades, no allowance being made for the effect of the nose flash on the concave side. A flash which was 0.002 in. proud of the blade surface would result in an angular error of about 0.1° (i.e. the blade outlet angle would be decreased by this amount). The accuracy of setting was about ±0.1°.

Sections of six rotor and six stator blades, rough and smooth, were projected optically to 20 times full size. The blade forms of all the blades appeared to be satisfactory. The following are the arithmetic means of measurements of camber and thickness derived from the projections for the mean diameter position.

Blades	Rotor		Stator	
	Camber	t/c	Camber	t/c
Smooth	26.7°	0.103	32.1	0.110
Rough	27.5°	0.107	32.4	0.113
Design	26.5°	0.100	34.0	0.110

These figures indicate that for the same setting angle, the rough blades had outlet angles about 0.5° and 0.3° smaller than those of the smooth blades for the rotor and stator respectively, allowing for a 0.002 in. flash on the rough blades. It is assumed that the sample blades are a fair representation of the remainder.

2.2.3 Tip clearances

Although the tip clearances on the 106 compressor are checked during assembly, it is probable that some relative movement between the stator rings and the rotor occurs during the final locking of the stator rings in position. Thus precise running values are not known, and may vary circumferentially. The assembly values for the cast blades were about 0.025 in. for both rotor and stator and 0.030 in. for the smooth blades. It is not thought that this difference is important.

2.3 Instrumentation

2.3.1 Total pressure rise

Two five-point pitot combs spaced 180° apart, were installed on the outlet annulus as indicated in Figure 1. Each pitot tube of the comb faced the same way; the corresponding tubes in the two combs were connected together and to fluid manometer columns; the combs were yawed to give maximum readings (which occurred simultaneously on all the pitot tubes) at a flow coefficient of about 0.67. Each pitot tube sampled an equal increment of radius, and the arithmetic mean of the five readings was taken as the delivery pressure. The inlet total pressure was taken as the prevailing atmospheric pressure. The loss across the intake gauze and in the inlet annulus was therefore neglected; but these losses were small and in any case they would not invalidate comparative tests.

2.3.2 Mass flow

Four static tubes placed in the inlet annulus as indicated in Figure 1 were commoned and connected to a Betz manometer. The readings were calibrated for mass flow against a standard orifice in the outlet ducting.

2.3.3 Speed

Speed was measured for all the tests by means of a Hasler hand tachometer.

2.3.4 Inlet temperature

This was measured by means of a mercury-in-glass thermometer at the compressor inlet.

2.3.5 Torque

The driving motor was supported in trunnions at each end, and the torque reaction on the casing was measured by means of an Avery balance. The balance had a variable linkage which was adjusted to give zero deflection of the motor casing at each torque reading; error due to out-of-balance of the casing and stiffness in the electrical connections were thus avoided. Each trunnion was supported in contra-rotating outer races, the object being to minimise hysteresis.

The zero reading of the balance was taken as that obtained by the extrapolation to zero speed of a series of torque readings at very low speeds (where the aerodynamic torque is negligible). This makes an allowance for compressor bearing friction. Unfortunately, the friction torque of ball bearings is known to rise with speed, but not in an easily determined manner. Thus although comparisons of parameters depending on torque (between different types of blades) at the same speed are valid, those at different speeds (between the same or different types of blades) are not. The important parameters thus affected are temperature rise coefficient and efficiency.

Windage torque is also not allowed for, but it is probable that it amounts to a fairly constant proportion of the blade aerodynamic torque over the whole speed range.

2.3.6 Measurements at very low speeds

For tests at speeds of 60 to 200 r.p.m., a micro-manometer had to be used for the measurement of the overall pressure rise and inlet static depression. The micro-manometer was very slow in reaching its equilibrium position, and to speed up the observations it was necessary to replace the two pitot combs by two single pitot tubes of 0.25 in. internal diameter. No attempt was made to correct for the effect of low Reynolds number on either the outlet pitot readings or the inlet static mass flow calibration.

It was not possible to measure the torque at these low speeds. The measuring gear was not sufficiently sensitive, and the aerodynamic torque was small compared to the bearing friction. Thus neither temperature rise coefficients nor efficiencies could be calculated.

2.3.7 Blade surface pitot traverses

A device was constructed whereby a fine pitot tube could be traversed chordwise across the convex surface of a stator blade at mean diameter, thereby sampling a part of the boundary layer. Hypodermic tubing of 1.0 mm outside diameter was used for these tests. The stator blade chosen was one of those in the removable segment position of stage 5 (para. 2.1 and Figure 2). The traverse could thus be performed on rough and smooth blades successively, under the same running conditions, by replacing one of the two available segments by the other, fitted with the alternative blades. The pressure was measured differentially against the outlet static pressure of the blade row at outside diameter, on a sensitive manometer. Alternatively, a stethoscope could be connected to the pitot tube, in order to investigate changes in noise across the blade surface.

3.0 Test technique

3.1 Overall measurements

Pressure rise coefficient, temperature rise coefficient, and total head efficiency, were determined for successively decreasing values of flow coefficient. The method of calculation is indicated in Appendix 2. This was done for speeds of 3,000, 2,250, 1,500, 1,250, 1,000 and 750 r.p.m. for both rough and smooth blades. Three test runs were made at each speed.

At very low speeds (60 to 200 r.p.m.) it was possible only to determine the pressure rise coefficient and flow coefficient (see paragraph 2.3.6)

3.2 Blade boundary layer investigations

These were all performed on the stage 5 removable stator blades (Figure 2 and paragraphs 2.1 and 2.3.7). The object was to attempt to determine, on the convex surfaces of both rough and smooth blades, the extent of laminar, turbulent and/or separated flow.

3.2.1 Pitot traverses

These were performed at 3,000, 1,500, 750, and 400 r.p.m. at a flow coefficient of 0.66, on both rough and smooth blades. The reading (paragraph 2.3.7) was rendered non-dimensional by dividing it by the inlet velocity head of the compressor.

It was possible to detect, by means of a stethoscope connected to the pitot tube, positions of marked increase in noise, at 750 and 1,500 r.p.m. At 400 r.p.m. there was no noticeable change, and at 3,000 r.p.m. the general noise level made a detection impossible.

3.2.2 Sensitive film methods

Both liquid and solid films were tried for this purpose, the method being to coat the blades with the film, place them in the compressor for a definite time, and then to remove them for examination. The degree of removal of the film should indicate the kind of flow in the boundary layer. (References 4, 5, 6, 7 and 8).

The solid films tried (soot and 1 · 4 diethoxybenzene) gave indistinct results, especially at low speeds. It would appear that the air speed was too low for the satisfactory use of either of these methods in this compressor, and it was extremely difficult to obtain films of even thickness.

Various liquid films were tried. In general, the more volatile liquids could be satisfactorily applied as thin films (by smearing with the finger), but evaporated too rapidly. The less volatile liquids tended to form into globules on the blade surface. The most satisfactory liquid from the point of view of giving a definite indication, was kerosine, either distilled into various fractions or as supplied for use as engine fuel. The required time of immersion in the compressor depended on the particular fraction used and on the compressor speed; there appeared to be a critical time for each fraction and speed, after which the pattern of removal remained comparatively stable; but often the evidence was somewhat conflicting on this point.

In general, the upstream part of the convex surface dried first, with sometimes a strip near the trailing edge. On the concave side the result was more variable, but this investigation was directed chiefly towards the convex surface. The smooth blades after removal from the compressor could be dusted with french chalk to emphasise the wet regions for photographic purposes. The rough blades were best examined visually, as the french chalk tended to adhere to both wet and dry areas of the blade surfaces.

4.0 Test results and discussion

4.1 Overall characteristics

These are plotted for speeds between 3,000 and 750 r.p.m. for both rough and smooth blades in Figures 6 to 11, and the curves are replotted with speed as a secondary parameter in Figures 12 to 14.

A serious difficulty encountered in these tests was the apparent change of performance between successive test runs at the same speed. The obvious possible causes - faulty test technique or apparatus and fouling of the blades - were fully investigated without result. The changes were not due to observers' errors, which probably would have resulted in a random scatter of points rather than the displacement of the characteristics as a whole. An attempt was made, without success, to correlate changes in performance with changes in atmospheric humidity.

It seems possible that the same compressor may have several slightly different stable regimes of flow for the same externally imposed conditions. The difficulty of exactly repeating test results in full scale compressors is well-known. In order to obtain truly representative results, it would probably be necessary to perform a very large number of test runs and deal with them on a statistical basis. The amount of testing involved would not be justified in general by the value of the results obtained. In the present series of tests, a compromise was made in that three runs were performed at each speed. The test points for separate runs are shown by separate symbols in Figures 6 to 11. Curves which appear to be reasonable means have been drawn through the points. It should be realised that if further test runs had been made, at any speed, the position of these curves might have been altered somewhat; it is highly unlikely, however, that the general conclusions to be derived from these tests would be affected. Comparisons between the rough and smooth blades are based on these curves.

The pressure coefficients derived from the low speed tests (50 to 100 r.p.m.) are shown on Figure 15. Only one or two test runs were considered justified at each of the low speeds.

In Figure 16, the overall parameters (derived from the mean curves) are plotted against speed at the design flow coefficient of 0.667. Included on this figure for completeness are the results of the low speed tests, extrapolated where necessary. The drop in the efficiencies and the rise in the temperature rise coefficients for both rough and smooth blades are partly due to the inclusion of bearing friction in the torque measurement. The rough blades show drops in efficiency of 4 per cent (of the smooth blade value) at 3,000 r.p.m. to 6 per cent at 750 r.p.m.; in temperature rise coefficient of 7 per cent to 1 per cent; in pressure rise coefficient of 12 per cent to 6 per cent respectively; and a constant drop in surge flow coefficient of 2 per cent from 3,000 r.p.m. to 750 r.p.m. Comparisons at very low speeds are perhaps best made by direct reference to Figure 15, where the rough and smooth blade characteristics show broadly similar variations with speed; the rough blades give smaller values of pressure rise coefficient at flow coefficients below about 0.55, and larger values at flow coefficients above 0.55.

It is interesting to note that the effect of the roughness and flashes on the performance appears to be similar to that of blade fouling. A test on the rough blades at 3,000 r.p.m. was made inadvertently when the blades were considerably fouled after extensive running. The characteristics are indicated in Figure 6, and show losses in pressure and temperature rise coefficients and in efficiency, together with a diminution of the surge flow coefficient.

The rough blades are thus inferior to the smooth blades over most of the speed range. The general shape of the characteristics, however, is unchanged, so that it would be reasonable to use rough blades for comparative tests of differing aerodynamic designs. It would, of course, be necessary to ensure that the different batches of blades were similar as regards the surface finish and flashes, and where the expected difference in performance was likely to be small, the added expense of the smooth blades might be justified by the elimination of doubts on this point. Considering the case of a service compressor, the consequences of the inferior performance of rough blades would have to be weighed against their lower production costs.

4.2 Blade boundary layers

4.2.1 Surface traverses

The results of surface pitot traverses at 3,000, 1,500, 750 and 400 r.p.m. for a flow coefficient of 0.66, are shown in Figure 17. All were taken with the same pitot tube. This is important, since the reading at any position is very dependent on the shape and size of the pitot orifice.

Notable features of the curves are:-

(a) The comparatively rapid change in the shape of the curves below 1,500 r.p.m. It is interesting to note that the value of overall pressure rise coefficient (Figure 16) shows a tendency to drop more rapidly with decrease in speed below about 1,500 r.p.m. ; a similar tendency is apparent with the efficiency, but this is probably partly due to the effect of bearing friction on the torque reading.

(b) A general similarity of the corresponding curves for the rough and smooth blades.

(c) A marked change of slope from approximately zero near the leading edge, as the trailing edge is approached; the position of this change moving towards the leading edge as the speed is reduced, and apparently being related to the increase in stethoscope noise at the speeds where this was investigated. At 750 and 400 r.p.m. the slope again approaches zero as the trailing edge is approached.

This method of traversing has been used on a turbine cascade (Reference 7) for the purpose of detecting the transition from laminar to turbulent flow. It is assumed that the boundary layer velocity gradient (measured in a direction perpendicular to the surface) is greater in the turbulent region after transition than in the laminar region before transition, at any rate near to the surface. The pressure picked up by the pitot tube (if this is of suitably small dimensions) will thus show a sudden rise as the tube is traversed from the laminar to the turbulent region.

Considered on this basis, the curves gave no indication of transition on the forward part of the surface. At 750 and 400 r.p.m. there is a change of slope towards the trailing edge, but it seems unlikely the transition would occur in this region; probably separation has occurred here to such an extent that the pitot tube is immersed in a region of zero flow and is picking up the prevailing static pressure.

4.2.2 Liquid films

A generalised pattern of wet and dry areas on the convex surface of a blade is shown on Figure 13. It must be emphasised that this figure is somewhat idealised, the actual patterns obtained being rather variable, and dependent upon the time of immersion, the degree of wetting of the surface prior to immersion, and the liquid used. Photographs of typical patterns (made visible by the application of french chalk) are shown on Figure 19. As far as could be judged there was no significant difference between the patterns obtained on the smooth and the rough blades under similar operating conditions.

According to Reference 4, the drying time is considerably greater for laminar than for turbulent boundary layers; the dry regions should thus indicate turbulent flow (in the boundary layer) and the wet areas laminar flow if the immersion time is suitably chosen. However, the scale of the experiments of Reference 4 was much larger than in the present tests (e.g. aeroplane wings of 48 in. chord were used with the same order of air velocity as in the present tests) although the liquid films were probably of the same order of thickness. If a liquid film of geometrically similar thickness could have been applied to the present blades, it probably would have evaporated in a prohibitively short time. Thus a different interpretation from that of Reference 4 would probably apply to the present tests.

It seems likely that besides evaporation, which will be greater in the more turbulent regions, two other effects will modify the surface pattern; the static pressure field over the blade surface and the traction force of the air on the liquid film. Both of these will result in a movement of the film. The static pressure field will tend to move the liquid to the suction peak of the aerofoil; the air friction effect, which is probably greater in a turbulent than in a laminar region, will tend to displace it towards the rear of the blade, unless a region of reverse flow occurs. Visual examination of the liquid film immediately after immersion in the compressor suggested that the liquid had been forced on to the area "B" in Figure 18. The very thin film remaining on the rest of the blade would be quickly removed by evaporation in either laminar or turbulent flow; this would explain the dry areas.

There appears to be a rough correspondence between the regions of greatest slope in the curves of Figure 17 and the wet regions "B" of Figure 18. A possible hypothesis would be that the wet area corresponds to a region of separation, followed by re-attachment. However, no definite statement can be made without more detailed investigation of the boundary layer; this would involve traversing perpendicular to the surface at several positions, and also the measurement of the static pressure distribution along the blade. The blade chords of the compressor are somewhat small for this purpose.

As far as the comparison between the rough and the smooth blades is concerned, it can be concluded that neither the present series of traverses nor the sensitive film tests showed any significant difference between the boundary layers of the two sets of blades.

4.3 General remarks

It appears that the only change of performance at any Reynolds number above about 0.3×10^5 caused by the roughness and flashes is a general fall in the primary parameters at all flow coefficients; the evidence from the rather limited tests, at lower Reynolds numbers is considered to be inconclusive. In any case, the presence of imperfections on the concave surfaces might be expected to cause a general drop in efficiency irrespective of Reynolds number; the arguments concerning the positions of transition and separation apply only to the convex surface (paragraph 1.0). The drop in temperature rise coefficient of the rough blades, as compared with the smooth blades, does not appear to be due to imperfections of blade form or to inaccuracies in the setting of the blades; the mean air angle at outlet from any row is probably slightly increased by the surface imperfections. This could best be checked in a cascade tunnel. About 1° rise in air outlet angle for both rotor and stator blades would be needed to account for the difference in temperature rise coefficient. The blade outlet angles are in fact appreciably lower for the cast blades (paragraph 2.2.2)

Reference 9 contains an account of some tests performed on a three stage high speed compressor. The rotor blades were smooth for all the tests, but the stator blades used were successively smooth, rough sand cast but smooth for the leading one-third of the chord, and rough sand cast all over. The results appear to be at least qualitatively in agreement with those of the present tests. The rough surface finish gave lower pressure rises and efficiencies and also lower temperature rises (as calculated from the published data). The reduction in peak efficiency was

5 to 8 per cent. depending on the speed as between the smooth and the totally rough blades. The rough blades also showed a reduction in surge flow.

It is probable that the flow in a compressor is in general highly turbulent, largely because of the effect of the wakes from one blade row moving relatively to the next row. Hence, any device designed to improve the performance by causing transition from laminar to turbulent flow in the boundary layers would be ineffective; and surface roughness and other excrescences on both the concave and convex surfaces would be expected merely to result in a drop of performance due to the increased thickness of the (turbulent) boundary layers. More detailed investigation of the flow on the blade surfaces would be needed in order to establish the exact processes occurring and to throw light on the possible effect of increased roughness and changes in the size and position of the flashes.

5.0 Conclusions

(1) The rough blades with die flashes show an inferior performance to the smooth blades at all speeds above and including 750 r.p.m. . The tests at lower speeds were too limited for general conclusions to be drawn. At 3,000 r.p.m. (Reynolds number = 1.3×10^5) there is a loss in efficiency of 4 per cent of the smooth blade value; and 6 per cent at 750 r.p.m. (Reynolds number = 0.33×10^5), these figures referring to the design flow coefficient. Corresponding losses in the temperature rise coefficient are 7 per cent and 1 per cent respectively. The tests indicate that the rough blade performance is probably still inferior down to a Reynolds number of 0.24×10^4. The surge flow coefficient of the rough blades is slightly smaller than that of the smooth blades. These figures are based on mean curves drawn from the points of three test runs at each speed; at some speeds considerable experimental scatter of test points was encountered.

(2) For simple comparisons of different aerodynamic designs of blades the rough blades appear to be as suitable as the smooth blades, there being no radical alteration in the shape of the overall characteristics. It would, however, be essential to ensure that no significant variation in surface finish occurred between the various batches of blades.

(3) Blade surface traverses and sensitive film tests suggest that the boundary layer conditions on rough and smooth blades are very similar. There is no indication that transition from laminar to turbulent flow in the boundary layer of the convex surface is caused by the roughness or die flashes. It seems probable that there is no significant region of laminar flow on the blades over the range of the tests.

NOTATION

U	=	blade speed at mean diameter
V_a	=	axial velocity
ΔT	=	total temperature rise per stage
ΔP	=	total pressure rise per stage
ρ	=	density
η	=	isentropic efficiency
μ	=	air viscosity
c	=	blade chord
s	=	blade pitch
t	=	blade maximum thickness
r	=	radius
α	=	air angle measured from the axial direction
β	=	blade angle measured from the axial direction
θ	=	blade camber

Suffices

$_1$	before rotor blade row
$_2$	after rotor blade row
$_3$	before stator blade row
$_4$	after stator blade row

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APPENDIX 1

Blade design details

The smooth blades were of free vortex design (design flow coefficient = 0.667) with details as below. The section was C4, on circular arc camber lines.

The cast blades were produced by the lost wax process using selected smooth blades as masters. Smooth inlet guide blades were used for both the rough blade and the smooth blade tests.

The rotor has 58 blades and the stator 60, so that with blades of 1.14, 1.10 and 1.06 in. chord at root mean and tip diameters respectively, the details are as follows :-

r in.	7.5	8.12	8.75	9.37	10.0
β_1	43.9	47.4	49.5	51.1	52.3
β_2	2.9	13.7	23.0	31.0	37.4
θ	41.0	33.7	26.5	20.1	14.9
s/c	0.712	0.787	0.862	0.942	1.020
t/c	0.12		0.10		0.08
α_2	12.3	22.0	30.0	36.6	42.0
β_3	47.5	45.0	42.4	40.0	37.5
β_4	9.8	9.0	8.4	7.7	6.9
θ	37.7	36.0	34.0	32.3	30.6
s/c	0.740	0.788	0.833	0.876	0.918
t/c	0.10		0.11		0.12
α_4	18.6	17.3	16.2	15.2	14.3

The design details of the inlet guide blades are as follows:-

r in.	7.5	8.12	8.75	9.37	10.0
β_3	-1.1	-2.3	-3.7	-5.1	-6.4
β_4	-22.1	-20.6	-19.2	-17.8	-16.4
θ	21.0	18.3	15.5	12.7	10.0
s/c	0.740	0.788	0.833	0.876	0.918
t/c	0.10		0.11		0.12
α_4	18.6	17.3	16.2	15.2	14.3

APPENDIX 2

Note on the calculation and significance of the overall parameters

General significance of parameters

At the highest running speed (3,000 r.p.m.) the maximum increase in air density across the six stages in these tests was only about 8 per cent; thus the six stages were effectively matched at all flow coefficients. The performance can thus be regarded as that of a single stage without compressibility effects but with two major advantages over conventional single stage tests:-

(1) Readings are more easily made because of their larger magnitude.

(2) The effect of velocity profile changes and the effect of the stages on each other are taken into account.

Density ρ

The overall mean density of the air is taken as the mean of the atmospheric density and an arbitrary value of the density at compressor outlet based on the measured total pressure and the corresponding calculated isentropic temperature rise. This value is used in the calculation of the overall pressure rise and overall flow coefficients.

Flow coefficient V_a/U

The axial velocity V_a is based on the overall mean density. The peripheral blade speed U is taken at the mean diameter for this and for the other non-dimensional coefficients.

Pressure rise coefficient $\Delta P/\frac{1}{2}\rho U^2$

The density ρ is the overall mean value defined above; the pressure rise ΔP is the measured total pressure rise, i.e. the excess above atmospheric pressure of the arithmetic mean readings of the two five-point pitot combs in the outlet annulus, divided by the number of stages

Temperature rise coefficient $\Delta T/\frac{1}{2}U^2$

The value plotted is $2gJK_p\delta T/U^2$ where

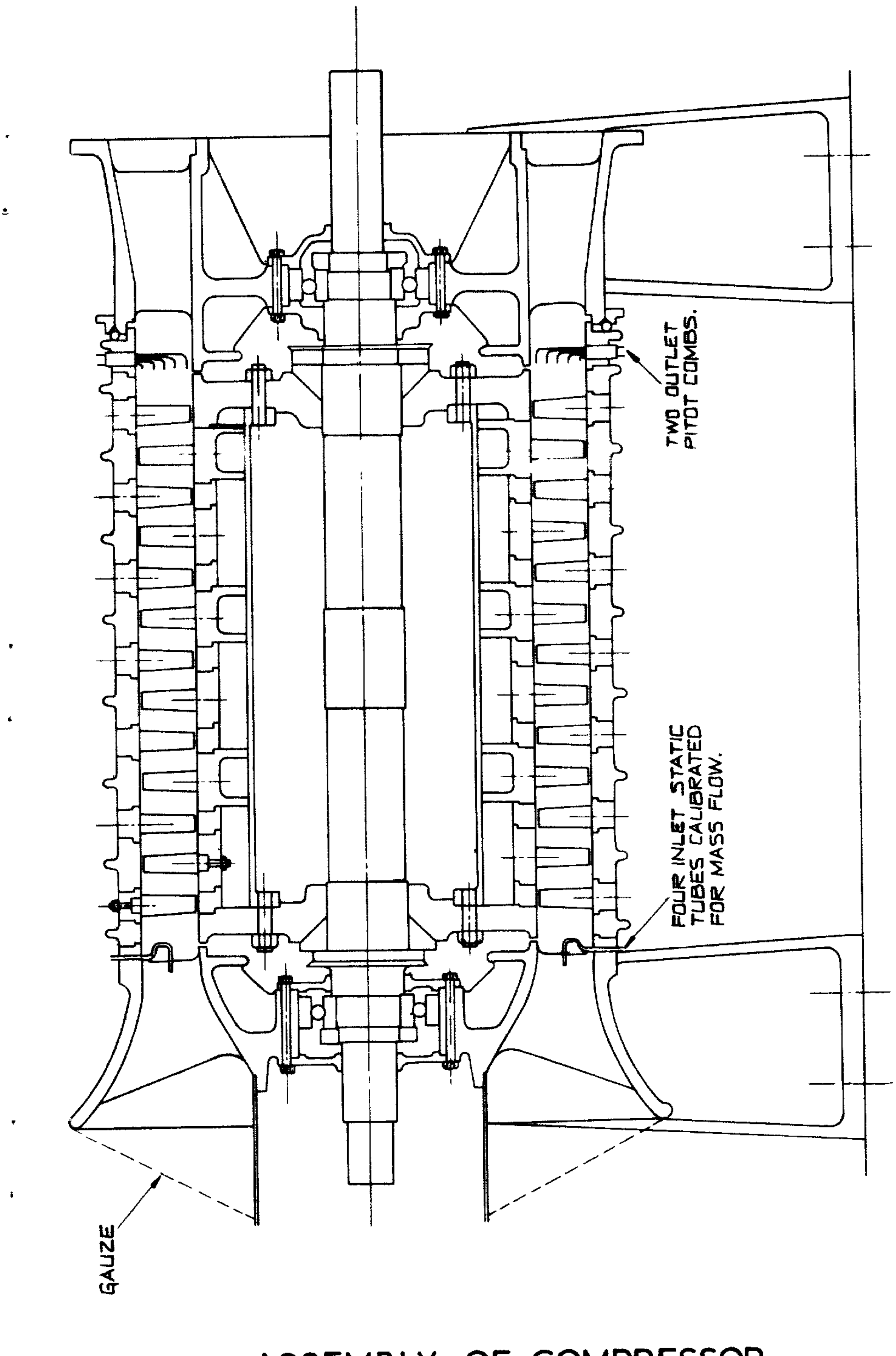
g	=	gravitational acceleration
J	=	mechanical equivalent of heat (in Centigrade units)
K_p	=	specific heat of air at constant pressure
δT	=	overall total temperature rise of the air in degrees Centigrade, divided by the number of stages

δT is calculated from the measured torque input and speed and mass flow. A correction is made for the bearing friction torque which is estimated by the extrapolation to zero speed of the torque values measured at a series of very low speeds.

Isentropic efficiency

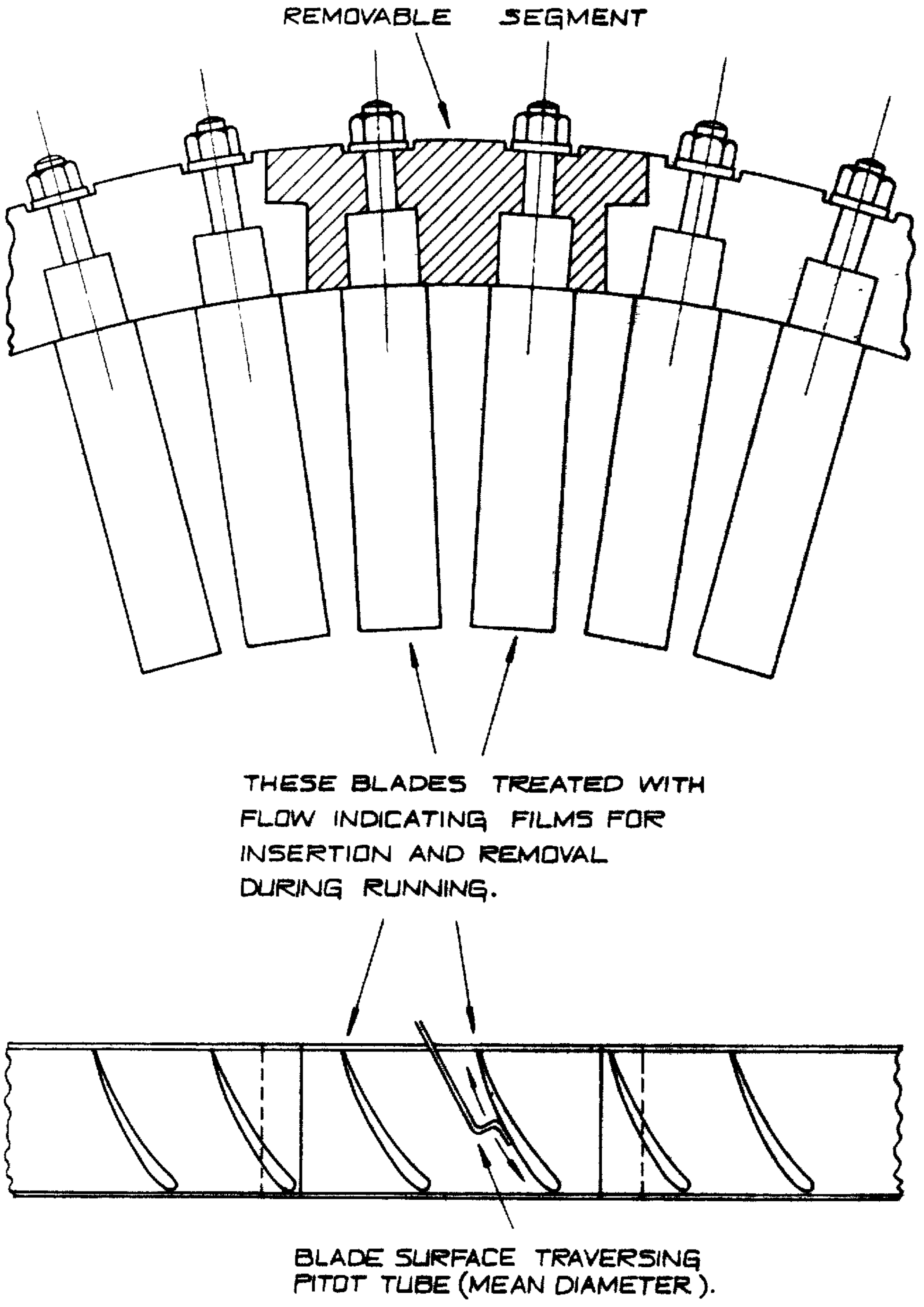
This is calculated from the measured total pressure rise and the calculated total temperature rise.

FIG. 1



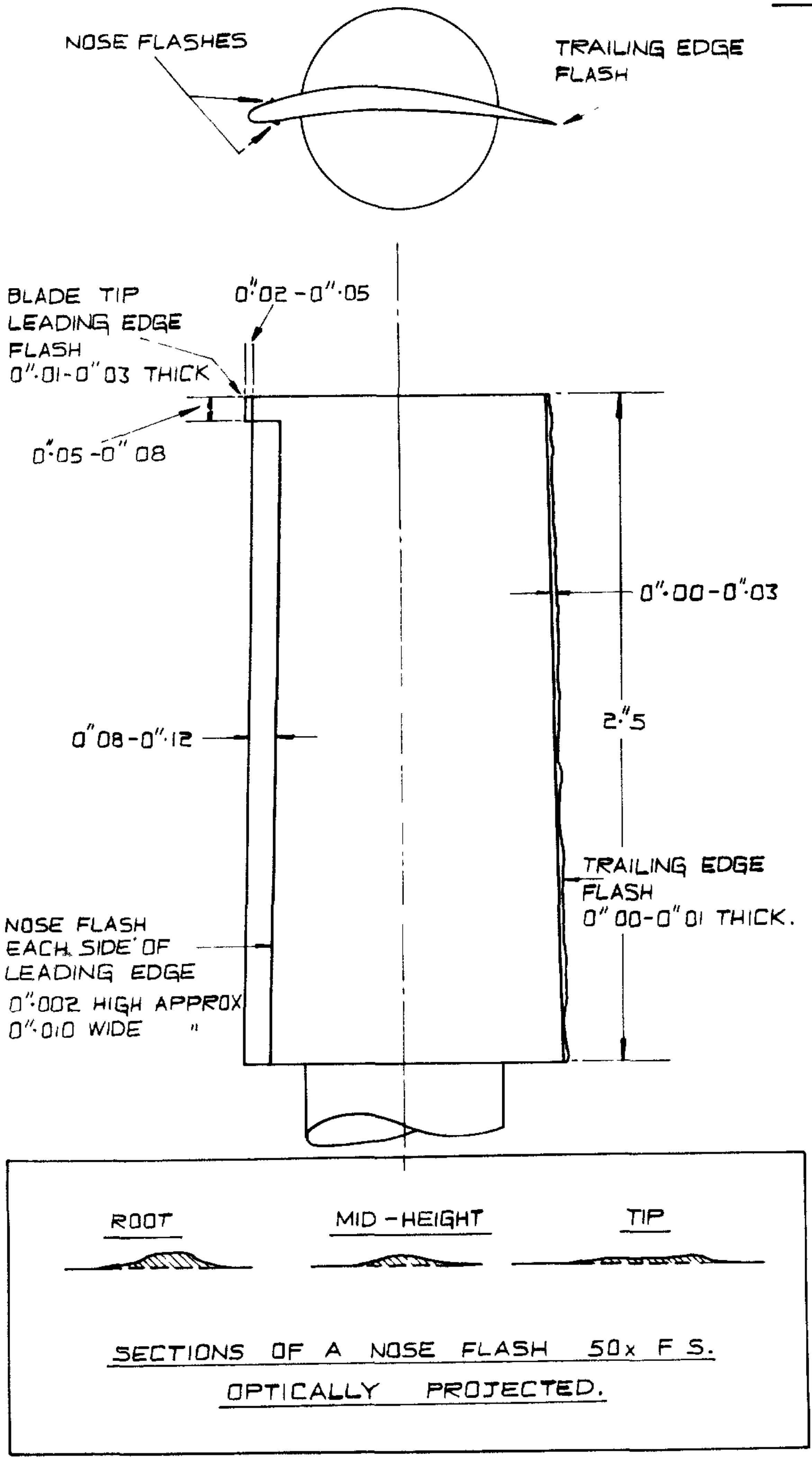
ASSEMBLY OF COMPRESSOR.

FIG. 2



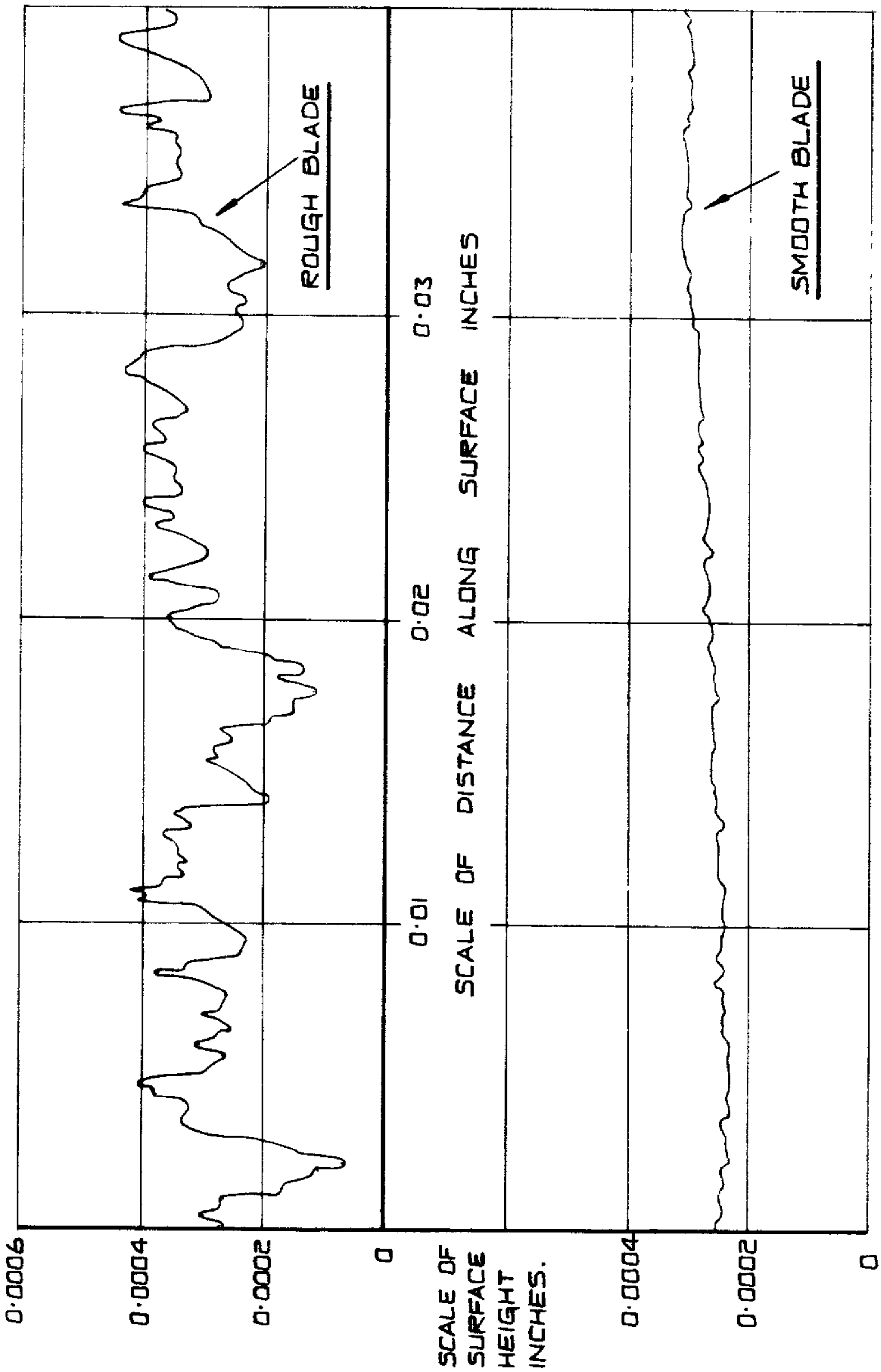
BLADE BOUNDARY LAYER INDICATION
STAGE FIVE STATOR BLADES.

FIG. 3



DIE FLASHES ON ROUGH BLADE.

FIG. 4



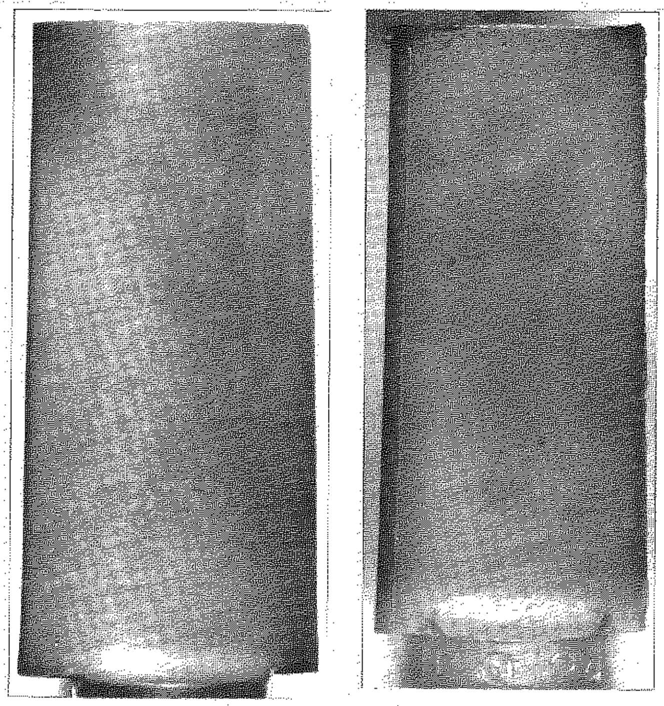
CURVES DERIVED FROM "TALYSURF" RECORDS.

TYPICAL BLADE SURFACE FINISHES.

FIG. 5

SMOOTH ROUGH

LEADING EDGE

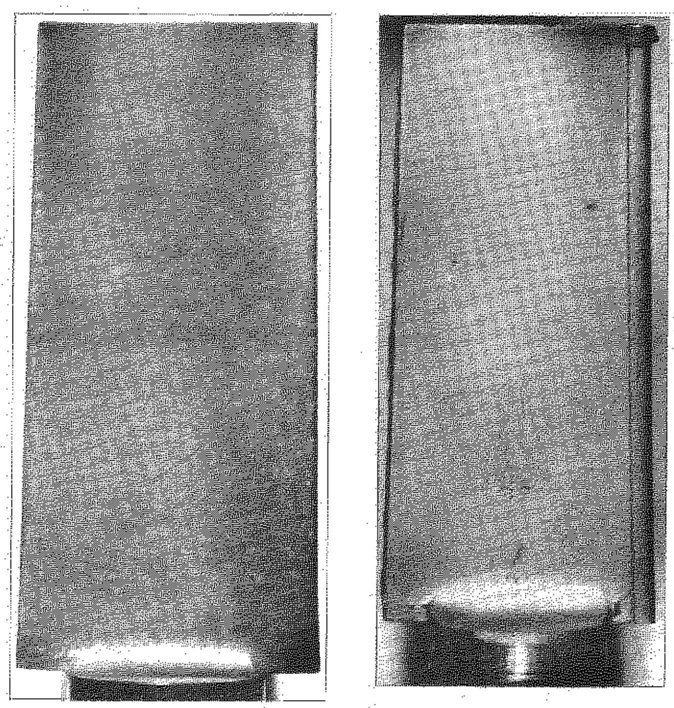


TRAILING EDGE

CONVEX SURFACE

SMOOTH ROUGH

TRAILING EDGE

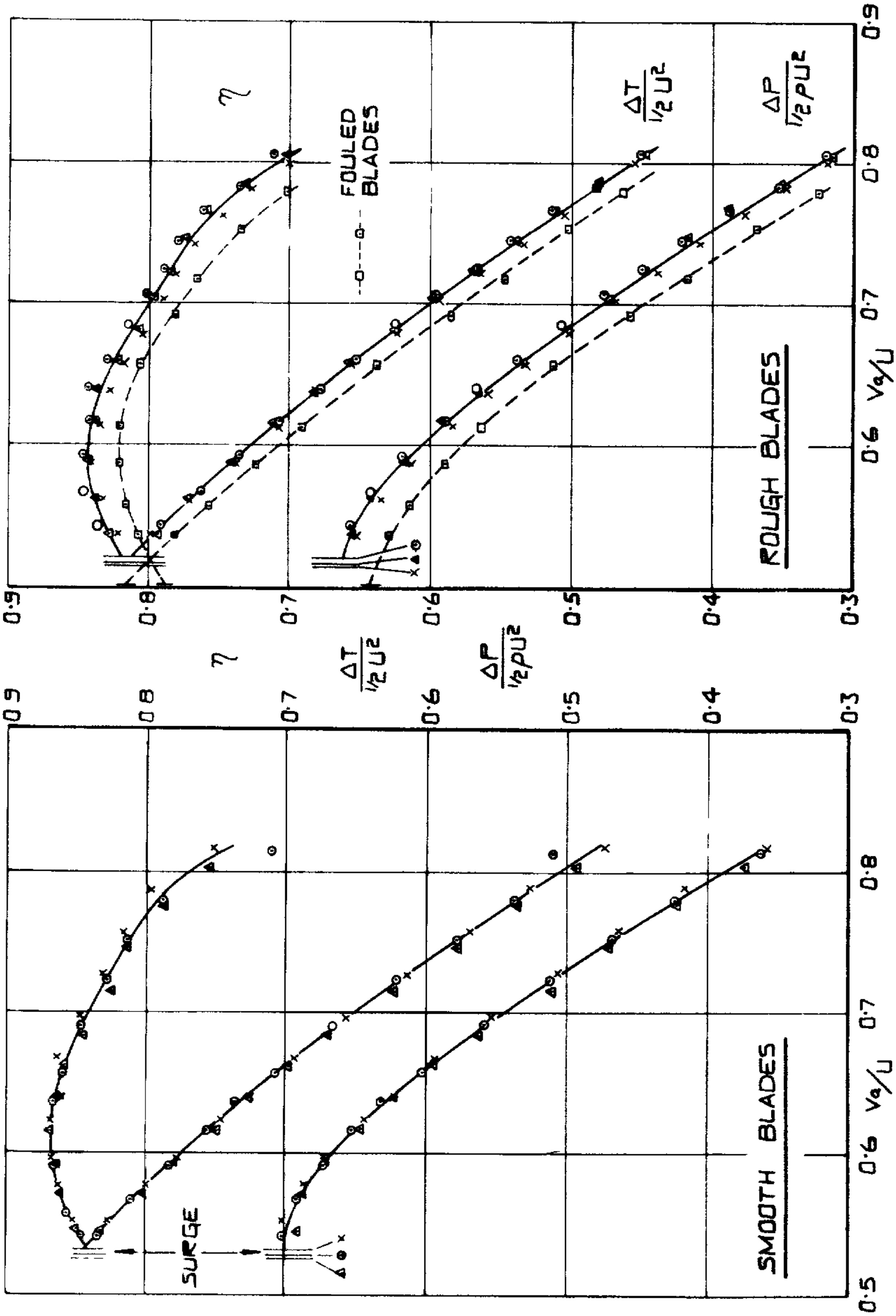


LEADING EDGE

CONCAVE SURFACE

GENERAL APPEARANCE OF STATOR BLADES.

FIG. 6

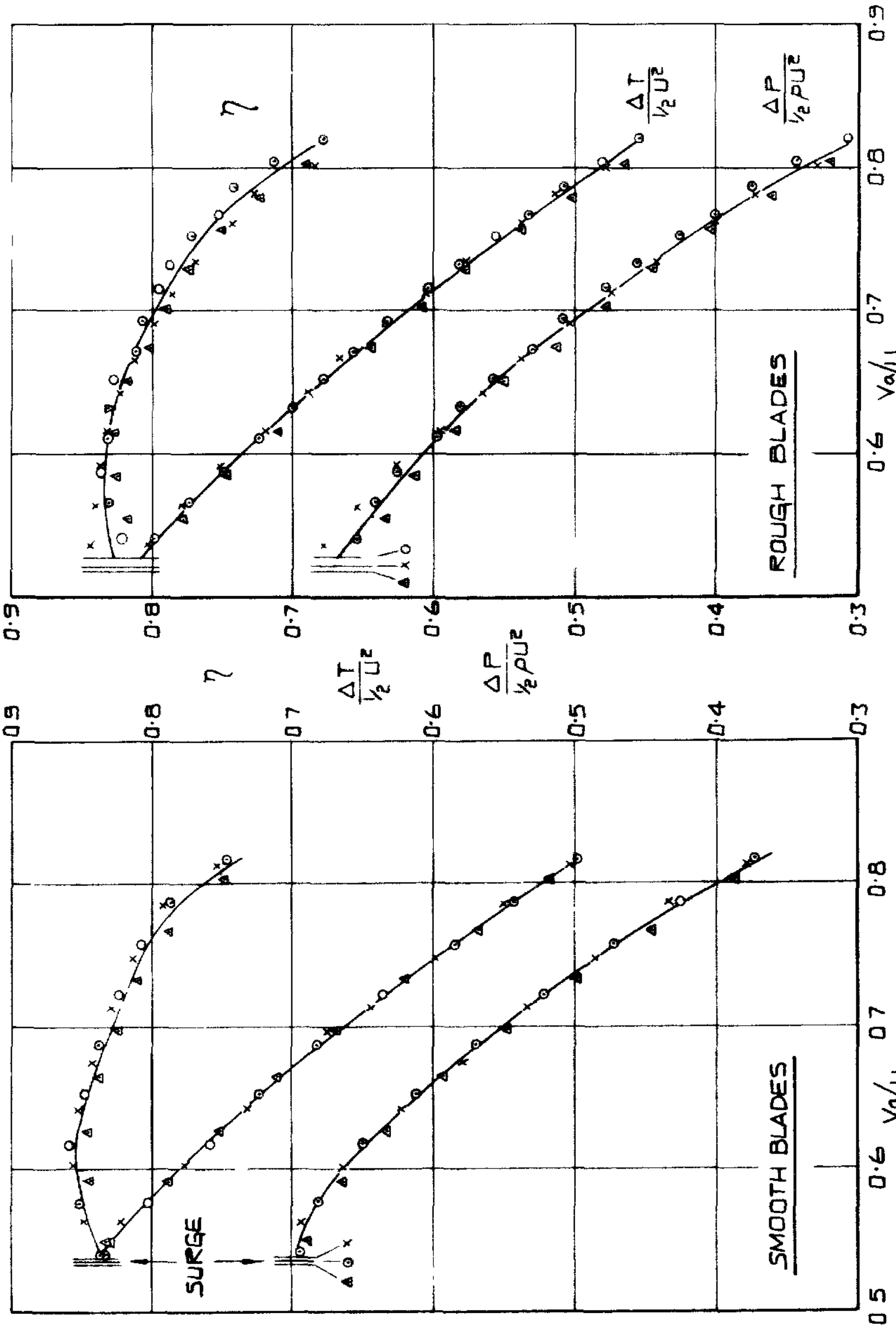


TEST RUNS DISTINGUISHED BY SYMBOLS

REYNOLDS NUMBER $= \frac{\rho U C}{\mu} = 1 \cdot 3 \times 10^5$ AT 3000 R.P.M.

OVERALL CHARACTERISTICS–3000 R.P.M.

FIG. 7

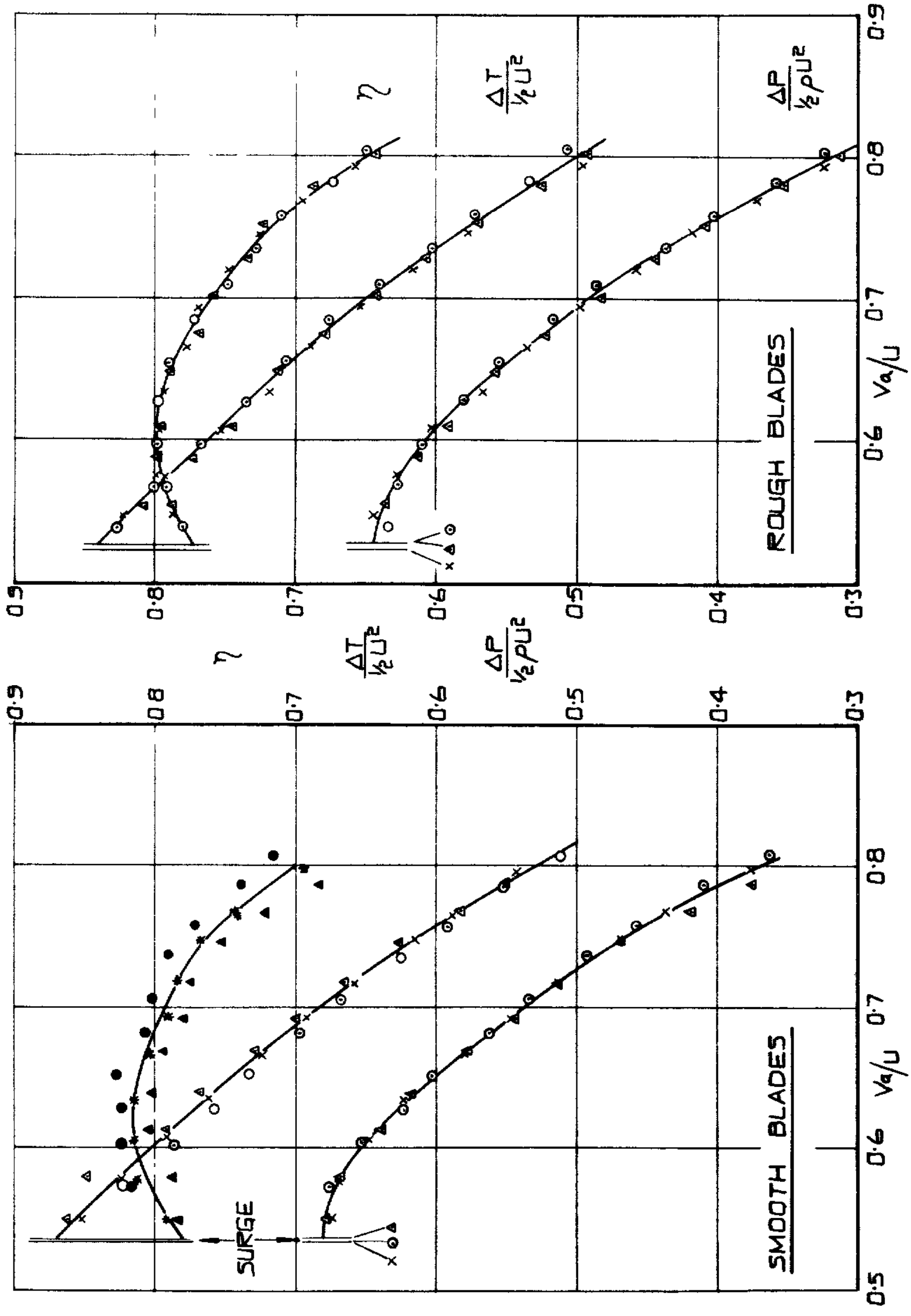


TEST RUNS DISTINGUISHED BY SYMBOLS.

REYNOLDS NUMBER $= \frac{\rho U c}{\mu} = 0{\cdot}98 \times 10^5$.

OVERALL CHARACTERISTICS-2250 R.P.M.

FIG. 8

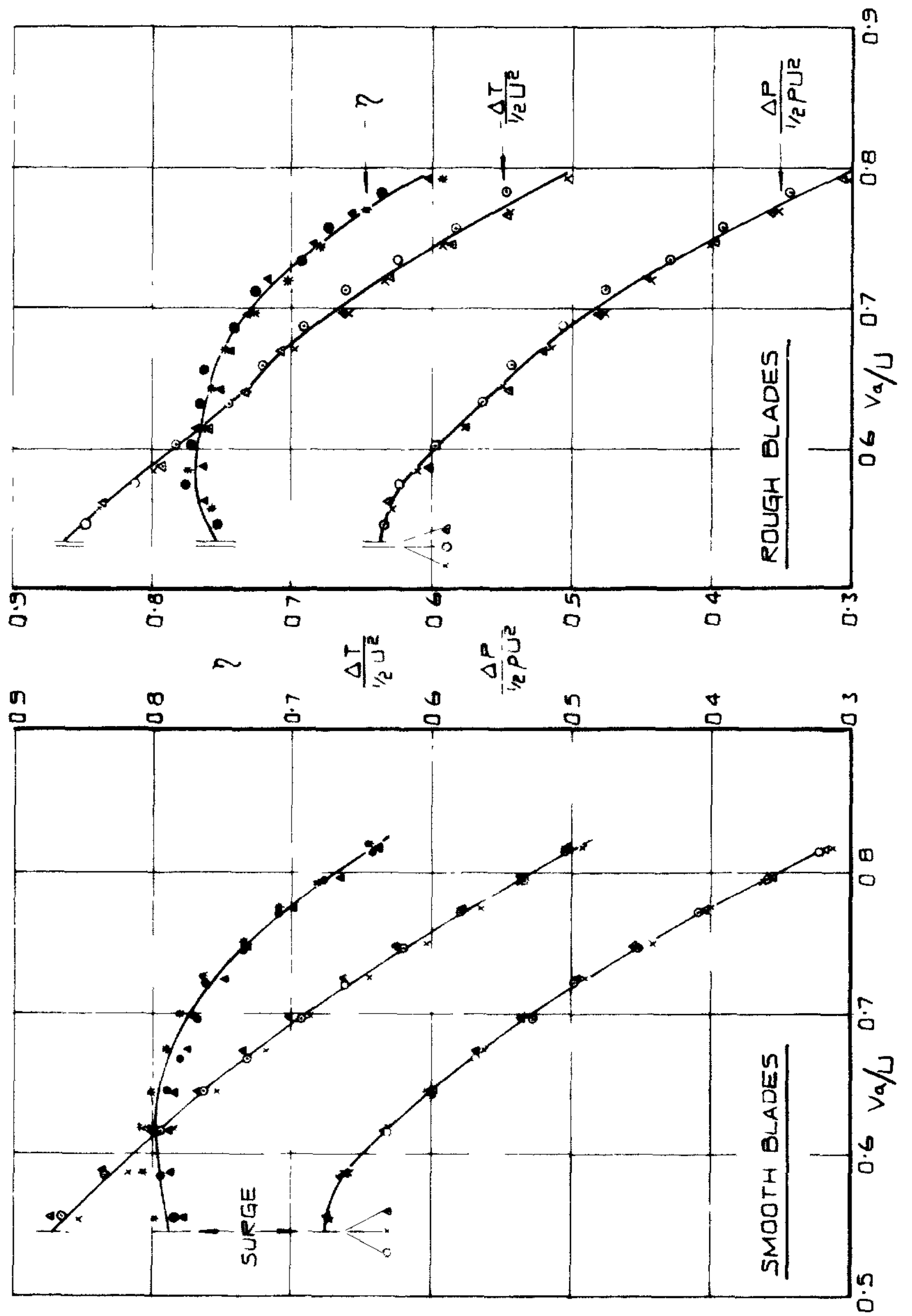


TEST RUNS DISTINGUISHED BY SYMBOLS

REYNOLDS NUMBER = $\frac{\rho U c}{\mu}$ = $0{\cdot}65 \times 10^5$

OVERALL CHARACTERISTICS-1500 R.P.M.

FIG. 9

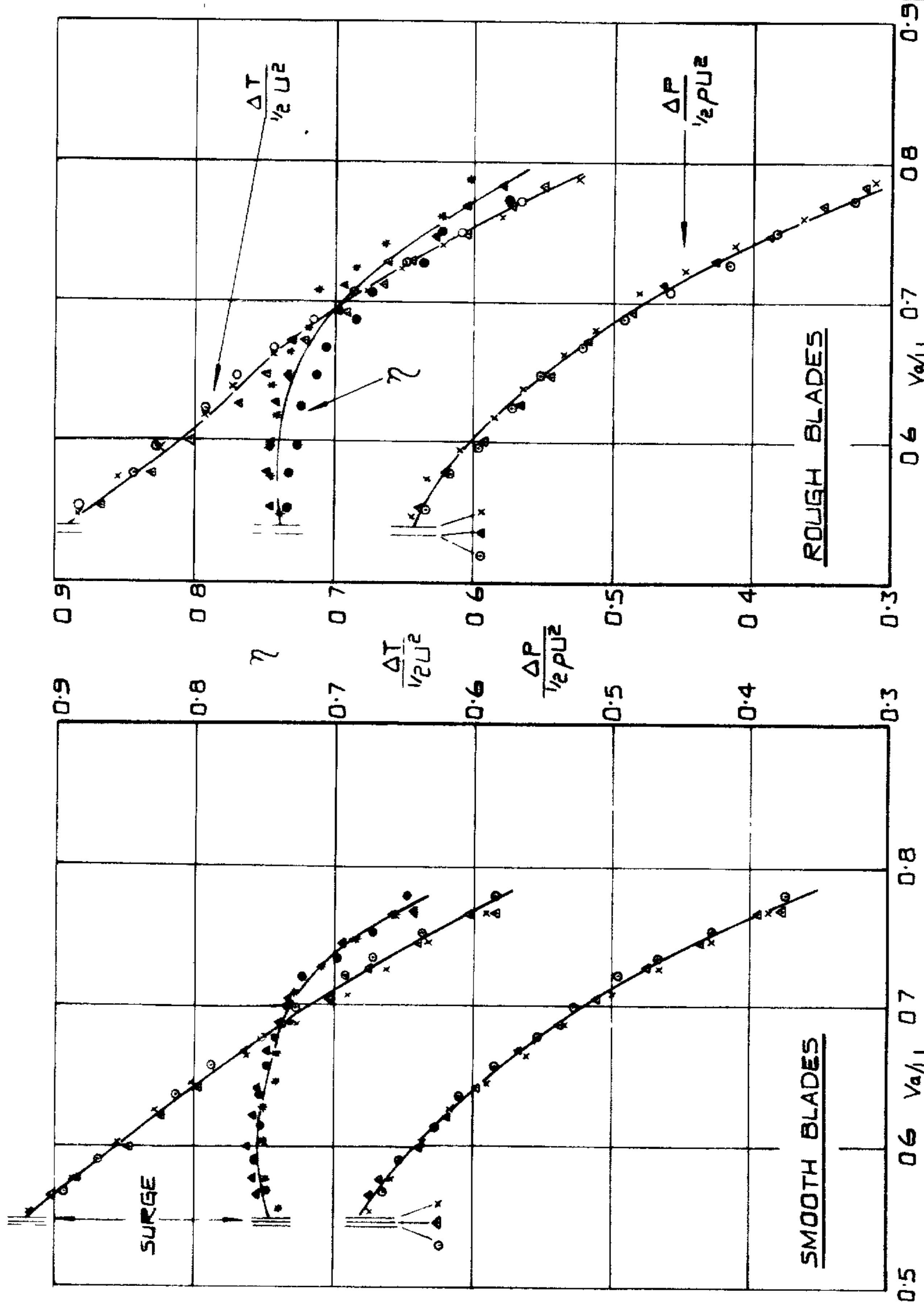


TEST RUNS DISTINGUISHED BY SYMBOLS

REYNOLDS NUMBER $= \frac{\rho U C}{\mu} = 0.54 \times 10^5$

OVERALL CHARACTERISTICS-1250 R.P.M.

FIG.10

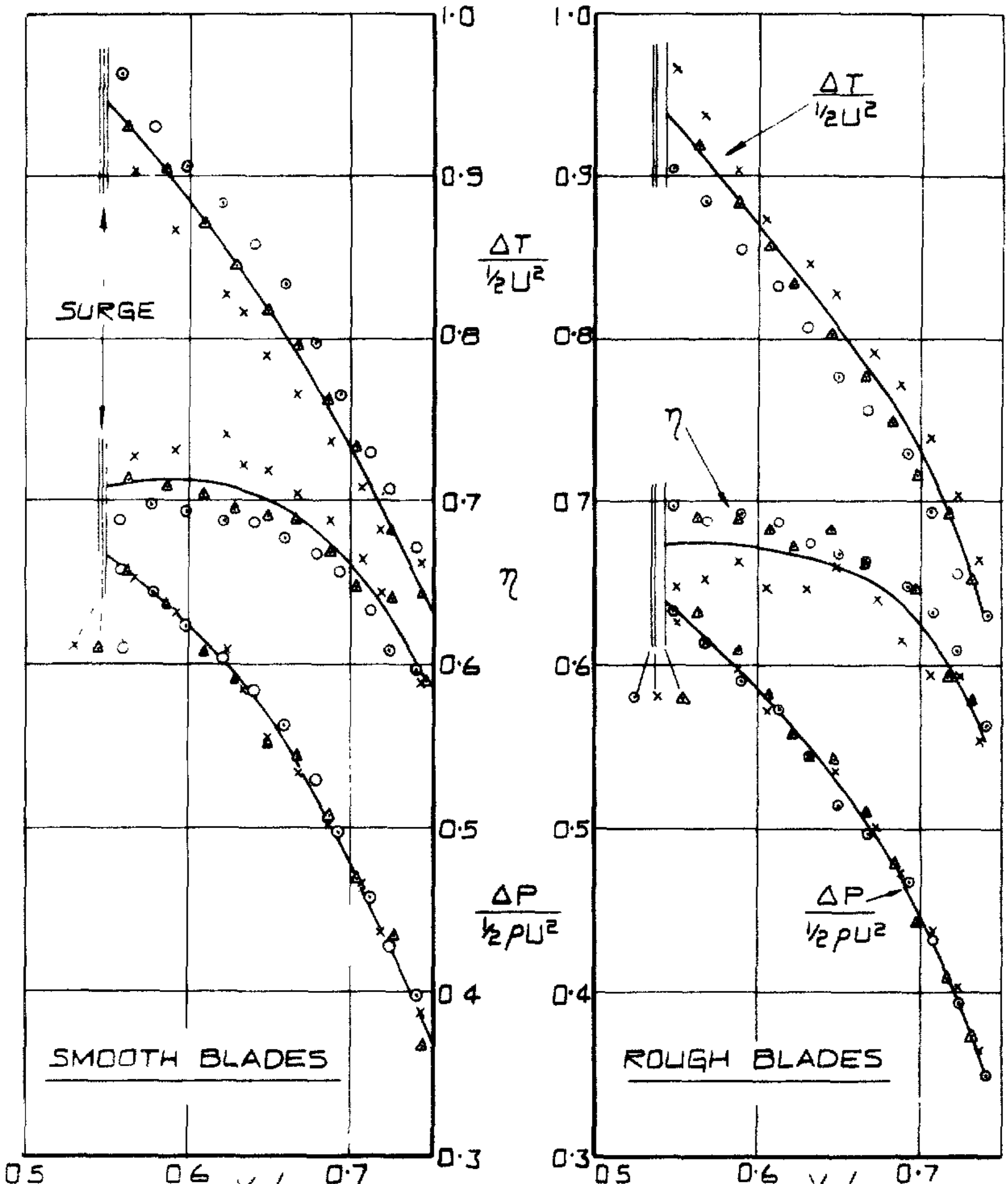


TEST RUNS DISTINGUISHED BY SYMBOLS

REYNOLDS NUMBER $= \frac{\rho U C}{\mu} = 0{\cdot}43 \times 10^5$.

OVERALL CHARACTERISTICS-1000 R.P.M.

FIG. 11

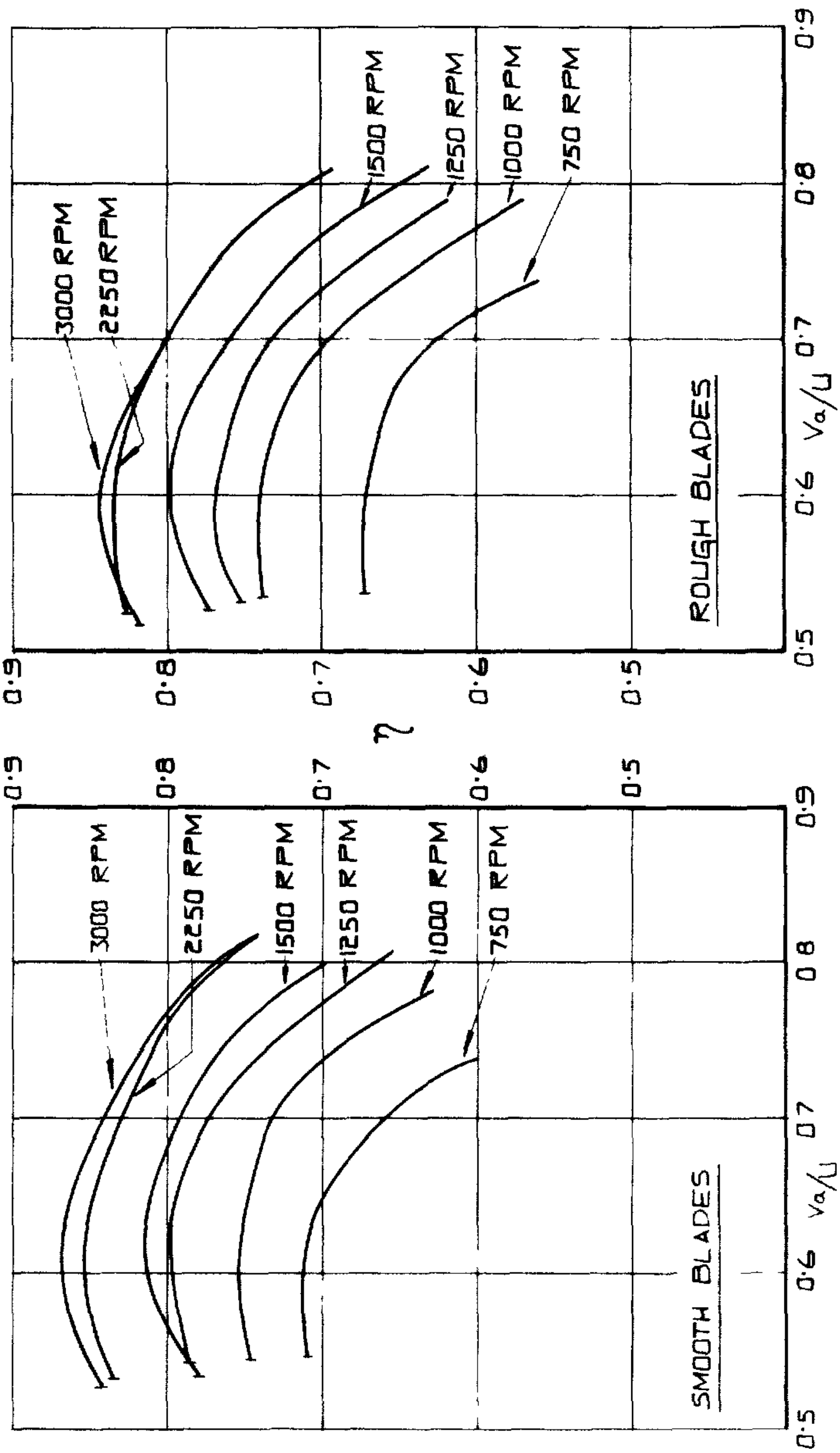


TEST RUNS DISTINGUISHED BY SYMBOLS

REYNOLDS NUMBER $= \frac{\rho U C}{\mu} = 0{\cdot}33 \times 10^5$

OVERALL CHARACTERISTICS–750 R.P.M.

FIG. 12

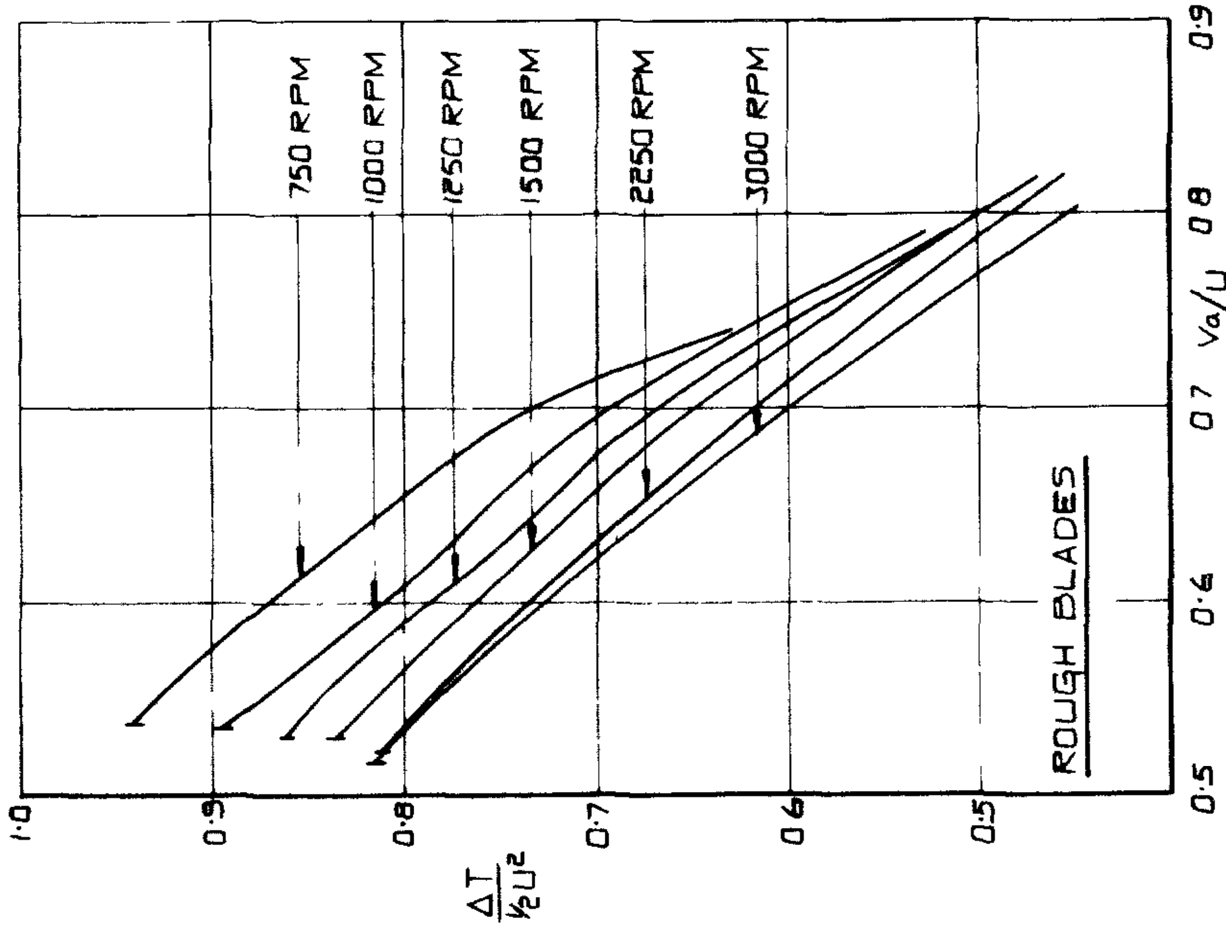


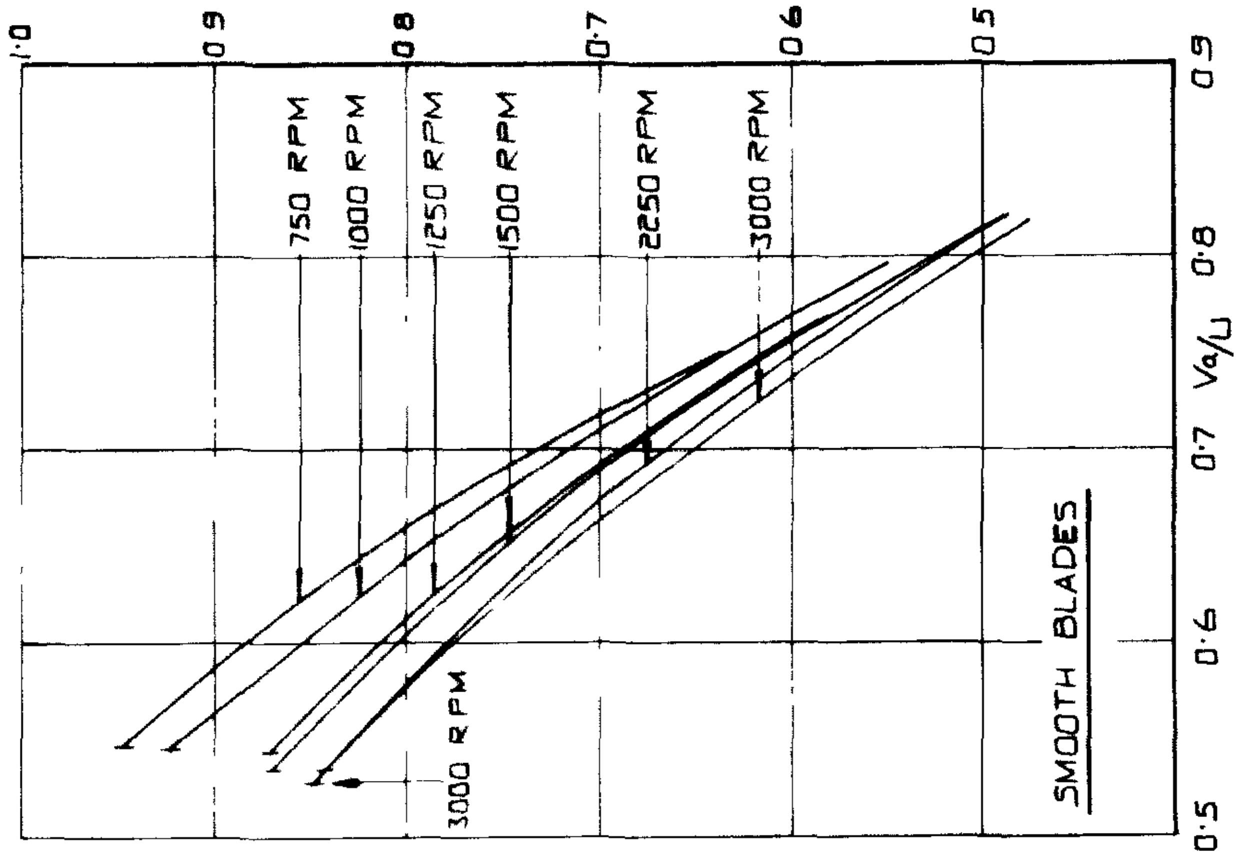
NOTE ERROR IN EFFICIENCY DUE TO BEARING FRICTION IS AN UNKNOWN FUNCTION OF SPEED.

REYNOLDS NUMBER $= \frac{\rho UC}{\mu} = 1{\cdot}3 \times 10^5$ AT 3000 R.P.M.

EFFICIENCIES AT 750–3000 R.P.M.

FIG. 13



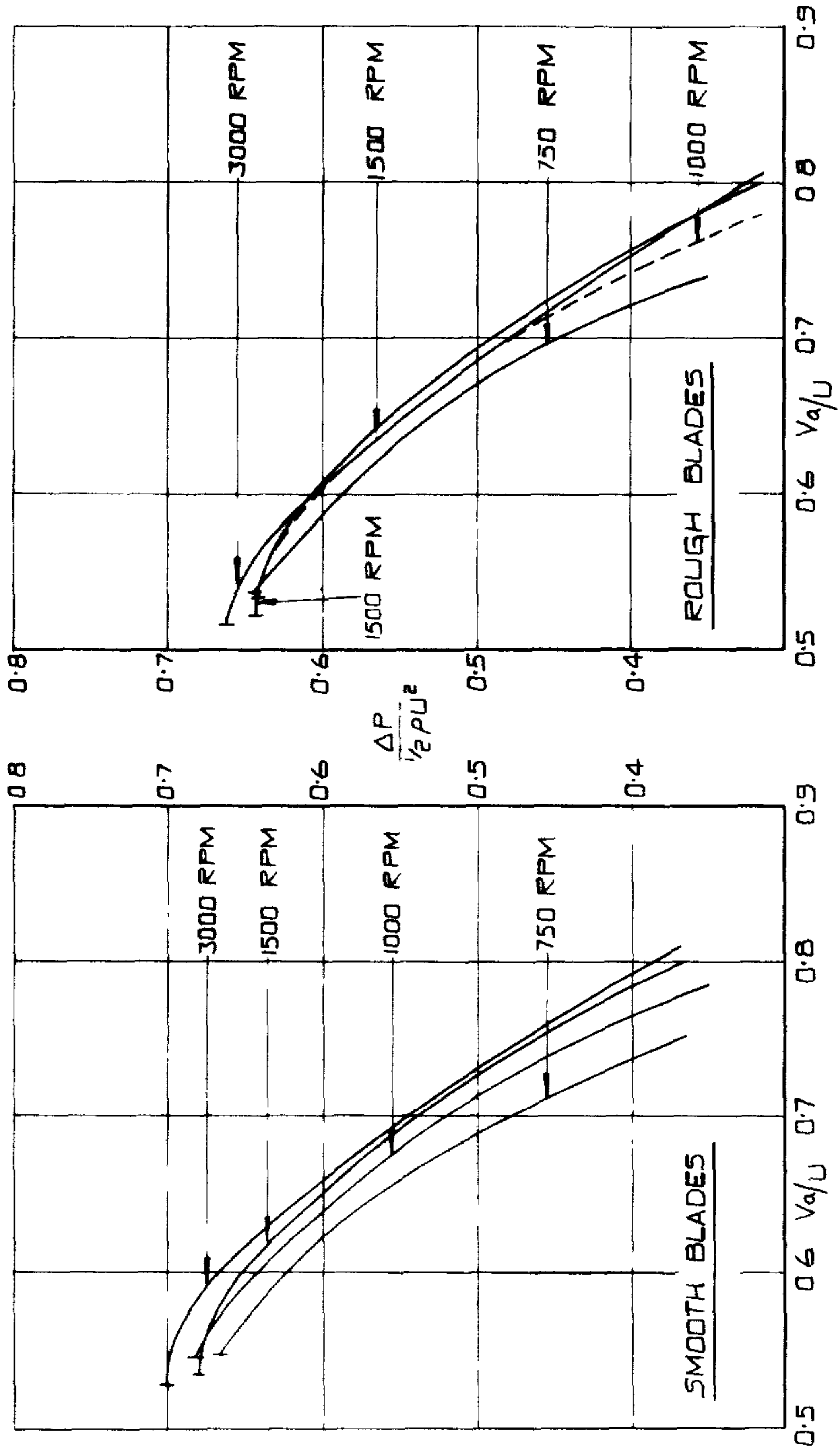


NOTE ERROR IN TEMPERATURE RISE COEFFICIENT DUE TO BEARING FRICTION IS AN UNKNOWN FUNCTION OF SPEED.

REYNOLDS NUMBER $= \frac{\rho U C}{\mu} = 1{\cdot}3 \times 10^5$ AT 3000 R.P.M.

TEMPERATURE RISE COEFFICIENTS AT 750-3000 R.P.M.

FIG. 14



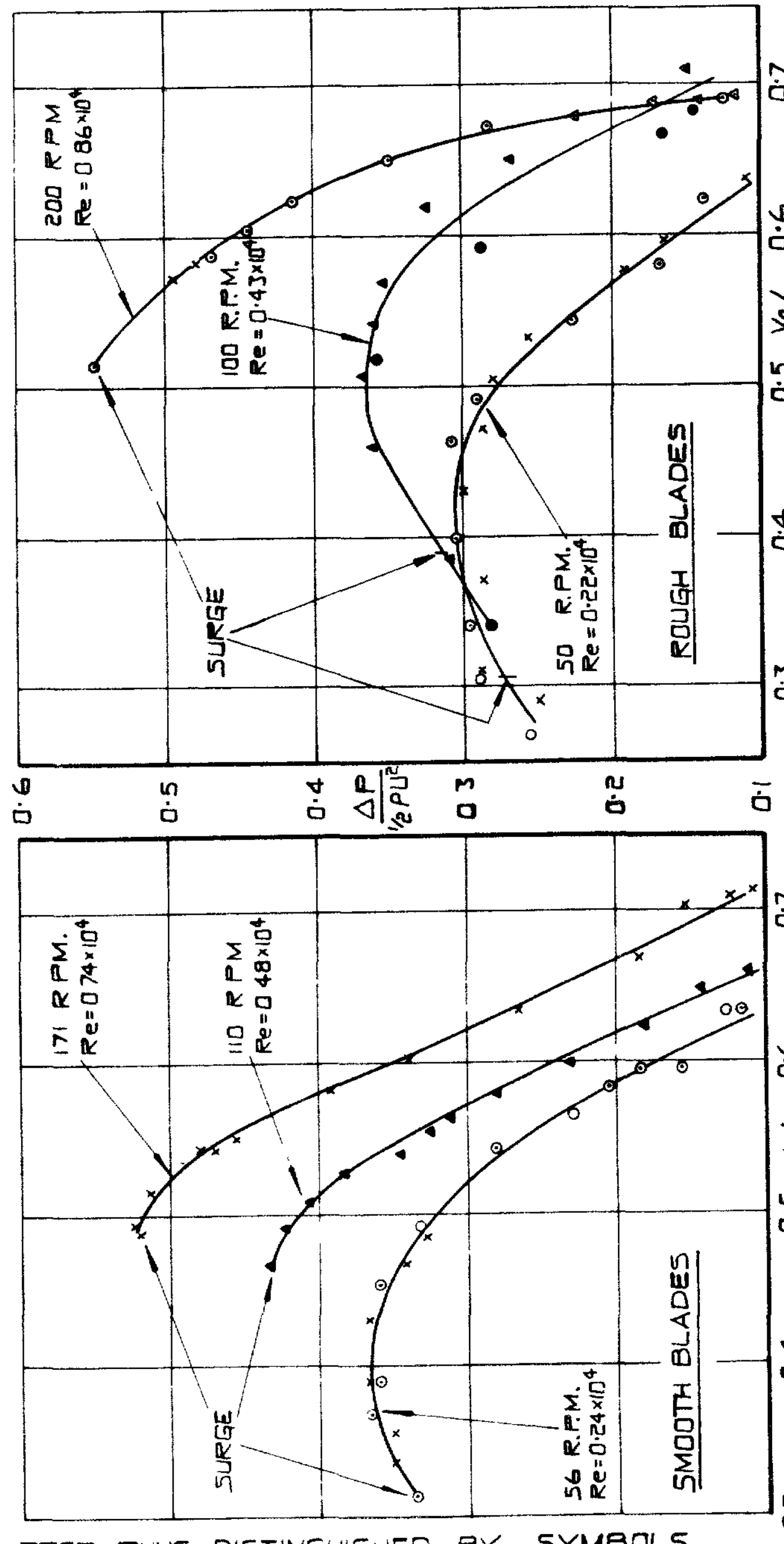
REYNOLDS NUMBER $= \rho \frac{Uc}{\mu} = 1{\cdot}3 \times 10^5$ AT 3000 R.P.M.

PRESSURE RISE COEFFICIENTS AT

750–3000 R.P.M.

FIG. 15

NOTE 1) INLET STATIC TUBE MASS FLOW CALIBRATION NOT CORRECTED FOR REYNOLDS NUMBER
2) NORMAL OUTLET PITOT COMBS REPLACED BY SINGLE POINT 1/4" DIAMETER PITOT TUBES.

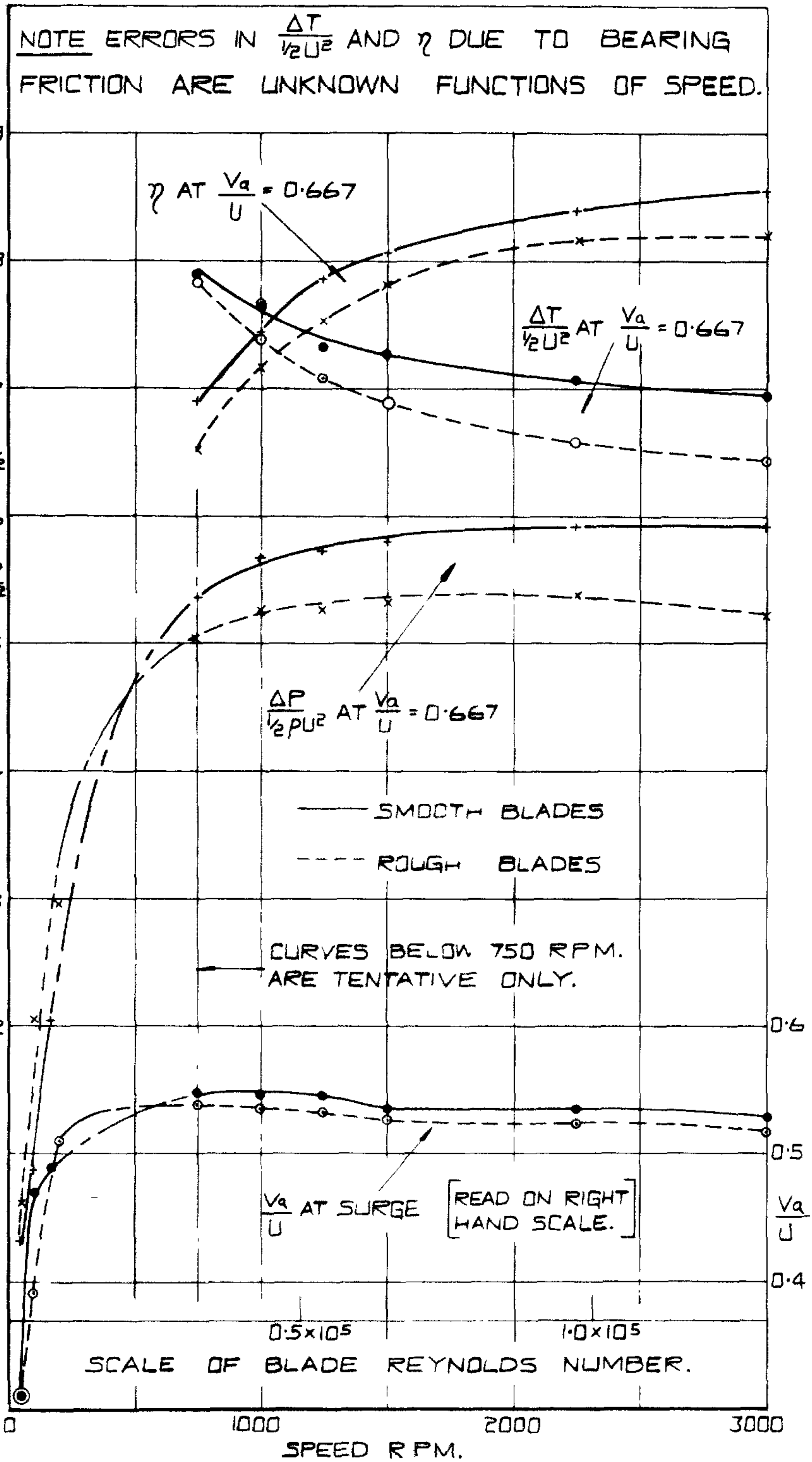


TEST RUNS DISTINGUISHED BY SYMBOLS

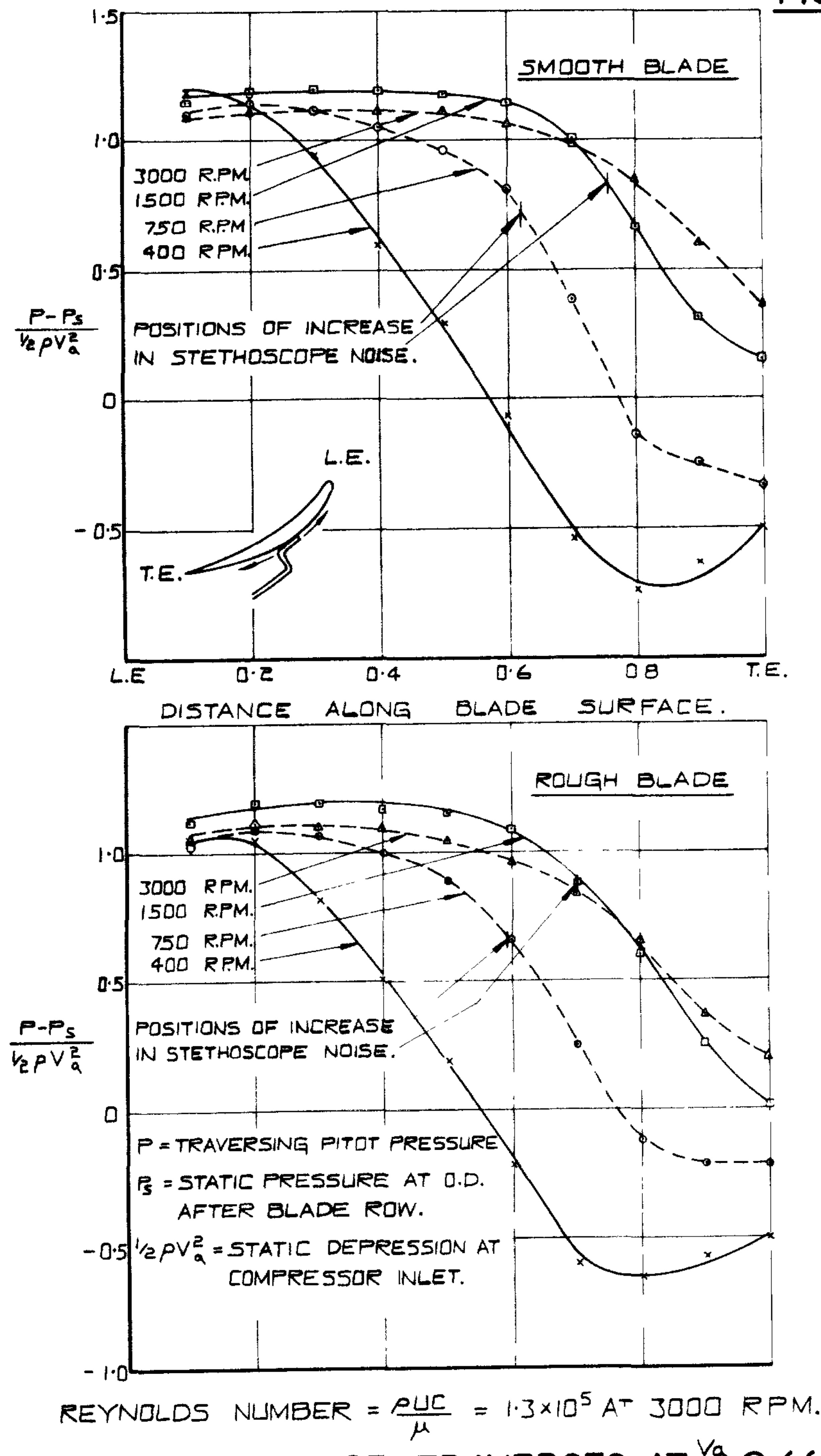
REYNOLDS NUMBER $= \frac{\rho UC}{\mu} = 1{\cdot}3 \times 10^5$ AT 3000 R.P.M.

PRESSURE RISE COEFFICIENTS AT 50–200 R.P.M.

FIG. 16



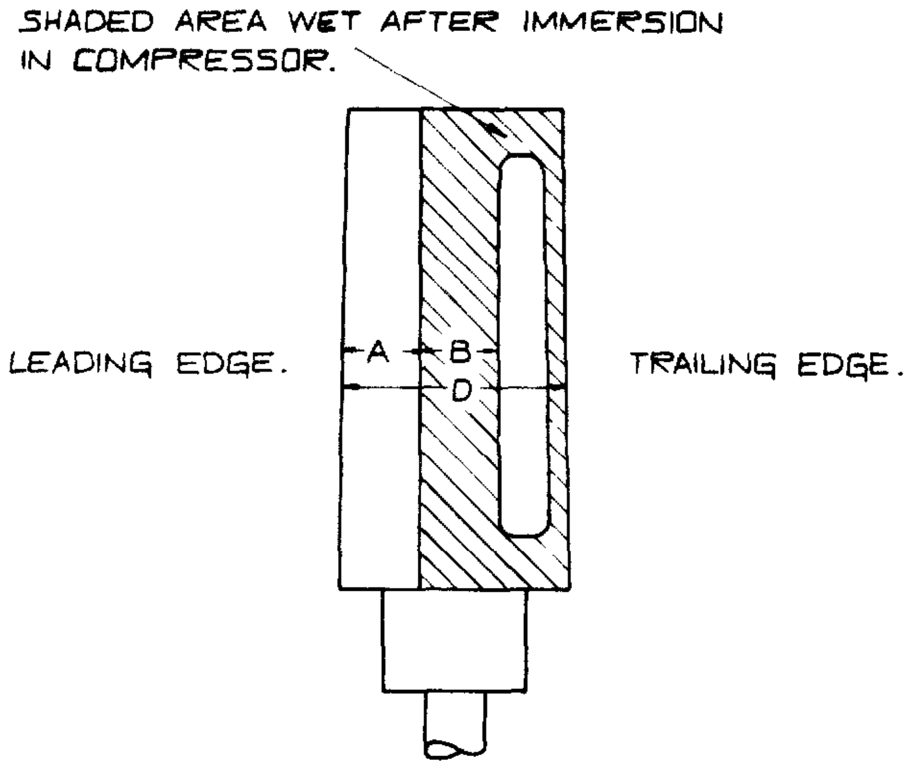
PERFORMANCE VARIATION WITH SPEED.



REYNOLDS NUMBER $= \frac{\rho U C}{\mu} = 1\cdot3 \times 10^5$ AT 3000 RPM.

CONVEX SURFACE TRAVERSES AT $\frac{V_a}{U} = 0\cdot66$ ON STAGE FIVE STATOR BLADE.

FIG. 18



SPEED R.P.M	$\frac{A}{D}$	$\frac{B}{D}$	FILM	IMMERSION TIME, MINS
200	0·09	0 55	220-240°C PARAFFIN	1-2
400	0·23	0 27	"	UNDER 1
750	0 58	0 24	"	"
1500	0 64	0 36	240-260°C PARAFFIN	1-2

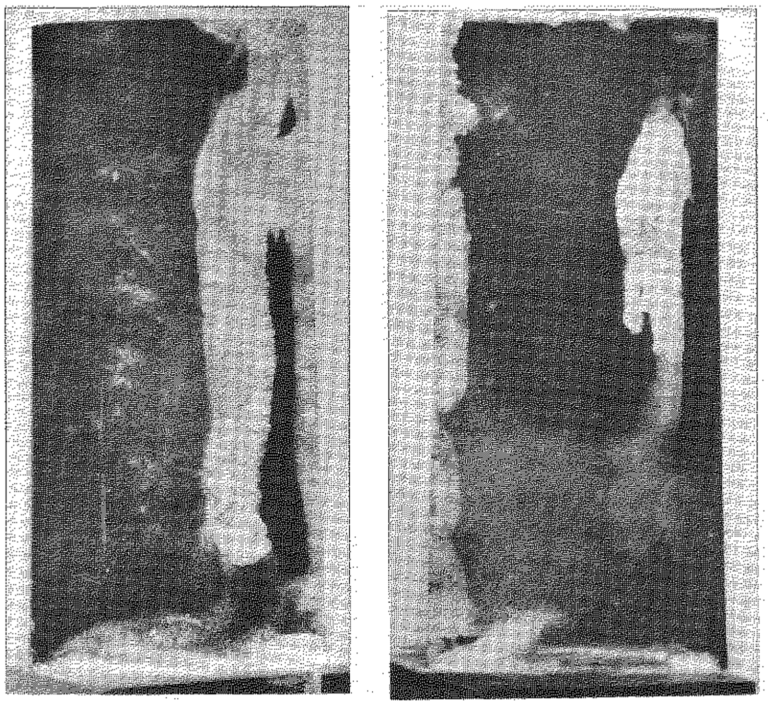
REYNOLDS NUMBER = $\frac{\rho U C}{\mu}$ = $0 65 \times 10^5$ AT 1500 RPM.

V_a/U = 0·66 AT ALL SPEEDS.

SIMPLIFIED LIQUID FILM DISTRIBUTION ON THE CONVEX SURFACE OF A STATOR BLADE IN STAGE FIVE.

FIG. 19

L.E. T.E. T.E. L.E.



CONVEX SURFACE

CONCAVE SURFACE

$V_a/U = 0{\cdot}66$ AT 1500 R.P.M.

REYNOLDS NUMBER = $\rho \frac{UC}{\mu} = 0{\cdot}65 \times 10^5$

LIQUID WAS 240–260°C FRACTION PARAFFIN

IMMERSION TIME IN COMPRESSOR = 2 MINS.

BLADE DUSTED WITH FRENCH CHALK FOR PHOTOGRAPHING.

TYPICAL LIQUID FILM DISTRIBUTION ON A STATOR BLADE OF STAGE FIVE.

Printed and published by
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S.O. Code No. 23-9010-6

C.P. No. 306