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The Design and Testing of an Axial-Flow Compressor having Different Blade Profiles in Each Stage

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Summary.—The report opens with a brief survey of various blade profiles and their two-dimensional cascade performance characteristics. The requirements of the different stages of an axial compressor are then discussed. This is followed by the design details of a suitable test compressor embodying in each stage the blade profile most suited to the duty that stage has to perform.

The test results for this compressor are given and discussed at some length. It is concluded that a compressor with suitable blade profiles in each stage has considerable advantages over one in which the same profile is used throughout. These advantages are :

(*a*) The compressor will have a good surge line
(*b*) Its part-load efficiency will be high
(*c*) It will have a wide characteristic at constant speed
(*d*) It may have a high efficiency at high flow
(*e*) It will have a higher peak efficiency than some designs
(*f*) It should be more reliable.

A possible disadvantage is that it may have a lower surge pressure ratio at the design speed than a compressor with high load blade sections throughout.

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1. *Introduction.*—It is customary in the design of axial compressors to select a blade profile and to use that profile for every stage of the compressor. It is known that different profiles have different characteristics. It is also known that the duty any individual stage of a multi-stage compressor has to carry out depends on the position of that stage in the compressor. In Ref. 1 the logical suggestion was made that improved performance could be obtained by using in each stage of the compressor the blade profile most suited to the duty of that stage. Ref. 1 went on to describe how that could best be done practically.

The main deterrent to the suggestion is the difficulty of matching the stages. Unknown small errors in the design of each stage may build up so that the compressor is not matched at any speed, whereas if the same profile is used throughout the errors will tend to cancel, and the compressor will be matched, even though it may not be at the designed speed. The latter fault is very easily corrected.

This report contains the design and the test results for a compressor having different blade profiles in each stage, substantially in accordance with the recommendations of Ref. 1. It starts with a brief recapitulation of the blade profiles and their two-dimensional characteristics.

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* N.G.T.E. Report R.217, received 10th March, 1958.

2. *Blade Section Performance.*—2.1. *General.*—Considerable theoretical and experimental work has been carried out at the National Gas Turbine Establishment on the various blade profiles that could be used in axial compressors. The work has been reported in Refs. 2, 3, 4, 5, 6, 7 8, 9, 10, 11 and 12.

In general it was believed that a blade profile could be adequately defined by :

(*a*) The blade maximum thickness

(*b*) The position of the blade maximum thickness

(*c*) The camber-line

(*d*) The position of the blade maximum camber

(*e*) The thickness of the leading and trailing edges.

These five quantities do not completely define the profile, of course, but given these quantities the possible alternatives available result in profiles that lay, to all intents and purposes, within the variations one would expect from manufacturing techniques.

Of the quantities quoted above the blade maximum thickness is not available for the control of the section aerodynamic characteristics, since it is closely associated with the mechanical strength of the blade. With the exception that the trailing-edge thickness must be chosen to meet certain production limitations, the remaining quantities can be chosen purely on aerodynamic grounds since they do not affect the mechanical criteria (section area, modulus, etc.). However, brief tests on leading-edge and trailing-edge thickness suggest these have only a minor effect on the aerodynamic performance provided the trailing-edge thickness is not allowed to become excessive. Sharper leading-edges seemed beneficial at high speeds, but the gain is small.

At the National Gas Turbine Establishment it has been customary to adhere to a camber-line in the form of a parabolic arc, except that, when the position of maximum camber is in the mid-chord position, the camber-line so closely approximates to a circular arc that the latter form is often adopted. Other camber-lines are obviously possible, for example, a semi-sine wave has often been suggested. However, it is believed that such variations would have a second-order effect, and for all the work on blade profiles a parabolic-arc camber-line has been adopted. The position of maximum camber then alone defines the camber-line.

Thus it will be seen that of the five quantities defining the profile we need only consider the position of the maximum camber and the position of the maximum thickness as major variables controlling the aerodynamic performance. These are discussed in detail in the next two Sub-sections, but before doing so it may well be helpful to recapitulate the various blade profiles in use at N.T.G.E., since the numbering system has chronological but no technical significance :

C.1 (Refs. 2, 13 and 14.) This is the original blade profile used in axial compressors. It is a general-purpose aerofoil (it is substantially RAF 27), having its position of maximum thickness 30 per cent chord from the leading edge.

C.2 (Refs. 2 and 13.) An asymmetric aerofoil not now used.

C.3 (Ref. 2.) A high-speed section having its position of maximum thickness 50 per cent chord from the leading edge.

C.4 (Refs. 2, 13 and 14.) A modified version of C.1 having a slightly thicker trailing edge to facilitate manufacture. It is the most used profile.

C.5 (Ref. 2.) A further modification of C.1 having an even thicker trailing edge. It is used on small-chord blades.

T.6 (Ref. 15.) A turbine aerofoil.

C.7 (Ref. 16.) A semi-high-speed section having its position of maximum thickness 40 per cent chord from the leading edge.

2.2. *Position of Maximum Camber*.—Theoretical work on the effect of the position of maximum camber is given in Refs. 3 and 6, and test results are given in Refs. 2, 4, 5 and 10. This work can be summarised as follows :

The high curvature near the leading edge of blades having a forward position of maximum camber results in a high suction peak on the convex surface of the blade in that region. This has two consequences. The low-speed stall of these profiles is sudden since separation tends to originate in the rapid diffusion region behind the suction peak, although at incidences below the stall the rapid diffusion is advantageous in that it takes place where the boundary layer is thinnest. Secondly, unity Mach number is reached locally on the blade surface at much lower inlet Mach numbers. Sections having a forward position of maximum camber will therefore have a lower drag critical Mach number than those with the maximum camber in the more rearward position. On the other hand cascades of blades having their position of maximum camber in the forward position have larger throat areas as shown in Fig. 1, and are therefore able to pass a larger mass flow before choking. At low incidences their performance is generally better since the high suction peak has disappeared from the convex surface, and the peak on the concave surface is only formed at very low incidences.

It follows that by locating the position of maximum camber towards the rear one can ensure that the stall is gradual, *i.e.*, that the low-speed stall is well separated from the incidence for maximum efficiency, and that the drag critical Mach number is high. This is accompanied by a lower choking mass flow and a somewhat worse low-incidence performance. Figs. 2 and 3 show two typical and practical blade profiles having positions of maximum camber 40 per cent and 50 per cent of the chord from the leading edge. The calculated pressure distributions and the cascade performance are given to illustrate the above remarks.

2.3. *Position of Maximum Thickness*.—Theoretical work on the effect of the position of maximum thickness is given in Ref. 6 and test results are given in Refs. 7, 8, 9, 10, 11 and 12. Very broadly, the effect of locating the position of maximum thickness towards the rear is very much the same as locating the maximum camber there. It results in a more rounded pressure distribution on the convex surface of the blade and consequently a higher drag critical Mach number. The loss of throat area resulting from a rearward movement of the position of maximum thickness is not so great as that which accompanies similar changes in the position of maximum camber. Since the two variables have similar effects most of the work has been carried out with the position of maximum camber at 50 per cent chord from the leading edge. This is about as far back as can be tolerated on account of throat area. The test results show that the increase in drag critical Mach number is accompanied by a reduction in the incidence range particularly for the maximum thickness greater than 50 per cent chord from the leading edge. In some of the tests quoted above, negative stall may have been due to laminar separation from the convex surface, so the significance of this feature should not be over-emphasized. Additionally, as the position of maximum thickness is moved back, the incidence for maximum drag critical Mach number tends to increase and approach the stalling incidence. In Figs. 3, 4 and 5 some typical results have been given to illustrate these points.

Limited tests have been carried out on the position of maximum thickness with the position of maximum camber slightly forward of the mid-chord point (Ref. 10). These suggest that moving the maximum thickness back from the leading edge is as powerful as moving the maximum camber. Since it entails a smaller loss in throat area it is the preferable practical method of achieving a good high-speed blade shape.

3. *Choice of Blade Profile for Stage Duty*.—3.1. *General*.—It is assumed that a primary design objective is a matched compressor, *i.e.*, that all stages operate at their peak efficiency point simultaneously at some mass flow and speed. This usually coincides with or is just below the design speed. At speeds away from the matching speeds the operating conditions of each stage will vary systematically through the compressor with the middle stages operating more or less

at their peak efficiency point, while the first and last stages will be furthest away from it. The principles governing the choice of blade profile can therefore be fully covered by concentrating on the first and last stages.

3.2. *Blade Profiles for the First Stage.*—Due to the low temperature at inlet the first stage has to operate at high inlet Mach numbers at the design point. This is particularly true of the tip section. At part load this stage has to operate at incidences above design. As is well known the objectionable kink in the surge line at part load is due to the stalling of the first stage or stages. It is essential that this stalling should be delayed to as low a speed as possible if a good surge line shape is to be achieved. One of the major requirements is, therefore, that the low-speed stalling incidence should be acceptably higher than the incidence at which the maximum critical Mach number and the maximum lift/drag ratio occur. This may not be essential for the first stage if adjustable guide vanes can be used, but would most certainly apply to the second stage.

From the remarks made in the previous Section it will be appreciated that a section with its position of maximum camber towards the rear will be most suitable for this duty. In order to obtain better high-speed performance at the design point, it would also be desirable to keep the position of maximum thickness towards the rear, the limit being determined by the necessity to maintain an adequate working incidence range.

3.3. *Blade Profiles for the Last Stage.*—Due to the temperature rise through the compressor the last stage does not have to operate at very high Mach numbers at the design point. At part load the last stage tends to operate at incidences below the design value and, due to reduced temperature rise, at high Mach numbers. It is, of course, choking of the last stage which limits the flow capacity of an axial compressor at part load. The requirements for this stage demand a profile giving as large a throat area as possible in order to obtain a wide characteristic. Obviously a blade section with its position of maximum camber in the forward part of the blade fulfils this demand. Since the design Mach number is moderate it will also be satisfactory to use a profile in which the position of maximum thickness is kept forward. This will assist in maintaining a high throat area and a wide operating range. It is also desirable to use a blade profile giving a high throat area in the high pressure stages for another reason. Due to the well known deterioration of the velocity profile through the compressor, the Mach number at the mid-blade height will be higher than design, while the incidence will be lower. In order to permit this profile to develop without choking the middle of the blade, the throat area should be some 20 per cent to 30 per cent greater than the calculated minimum. Otherwise even the centre of the blades in the high-pressure stages will be operating at low efficiency. Furthermore, attempts to control the velocity profile have always resulted in some loss of efficiency even at low speeds. This aspect is discussed further in Section 6 in connection with the observed test results.

4. *Design of Test Compressor.*—As a suitable vehicle for testing the ideas advanced in the previous Sections, use was made of the N.G.T.E. High-Duty Compressor, Reference number 109. A full description of this compressor and its first set of blading, together with the test results obtained from that blading, is given in Ref. 17. As originally bladed, the blade sections had a parabolic-arc camber-line with the position of maximum camber at 40 per cent chord from the leading edge throughout. The position of maximum thickness remained 30 per cent chord from the leading edge for all blades. It will be appreciated from the foregoing that this blading is suitable for the last stage, and so it was decided to retain this stage in the new design. Actually, for practical reasons, the last two stages of the original blading were retained. The profiles for the first four stages were then modified in accordance with the ideas advanced. Partly to reduce the manufacturing demands, and partly to facilitate comparison, the same number of blades and the same blade chord (and thus pitch/chord ratio) was used for the revised blading as for the original design. For the same reason the design speed was retained at 9,500 r.p.m. and the design mass flow of 74 lb/sec and pressure ratio of $4 \cdot 5/1$ chosen to agree with the test values obtained with the original blading.

A circular-arc camber-line, *i.e.*, position of maximum camber 50 per cent chord from the leading edge, was the obvious choice for the first stage. It was decided to use a C.7 section, position of maximum thickness 40 per cent chord of the leading edge, on this camber-line. The C.3 section having its position of maximum thickness further towards the rear was not used because it was thought that its incidence range was too narrow and this would adversely affect the surge line.

Having selected what was thought to be the best profile, it seemed only reasonable, at the time of the design, that every endeavour should be made to ensure that it operated at its optimum incidence. Consequently this stage was designed on the assumption that radial equilibrium would be satisfied after each blade row. The form of vortex flow adopted assumed that the whirl velocity at inlet to the stator was radially constant. This form was adopted since, if radial equilibrium is neglected (*i.e.*, the axial velocity is assumed radially constant), it then reduces to the well known ' constant α_3 ' twist used in the design of the original blading. Associated with this method of calculating the flow angles, the blade angles were chosen to give ' nominal ' conditions at all radii, using the data given in Ref. 18 or the derived charts in Refs. 19 and 20. The departure from a constant axial velocity design may now be regretted. It is believed to have no significant effect, but introduces an additional variable when comparing the test results from the two sets of blading.

The aim in the design of the intermediate stages was to provide a smooth and a gradual transition from the blading adopted for the first stage to that adopted for the last. In this way it was hoped to preserve the matching of the compressor. Thus in subsequent stages both the position of maximum camber and the position of maximum thickness were brought forward in small steps. In order to provide adequate throat area the position of maximum camber was gradually brought forward in second and third stages. The position of maximum thickness was retained at 40 per cent chord from the leading edge to give good high-speed performance and only brought forward in the very last stages where large throat area and good negative-incidence performance are paramount. On the same basis of ' gradual change ' stage two was designed half-way between radial equilibrium and a radially constant axial velocity. Subsequent stages were designed for a radially constant axial velocity and thus had the ' constant α_3 ' blade twist adopted in the original design. The mean-diameter blade profiles have been drawn out and fully designated in Fig. 6, which also summarizes the design philosophy. Fig. 7 shows the stage-by-stage variation of throat area for this design and the original. Every endeavour was made to keep this as smooth as possible, but it will be noted that the throat area of the first-stage rotor is smaller than might have been ideally desirable. This is due to using the C.7 section on a circular-arc camber-line. It could not have been corrected without using excessively thin blades.

5. *Testing of Compressor*.—5.1. *Test Technique*.—For the purpose of these tests the compressor was driven from the outlet end. This enabled air to be fed to the compressor *via* an unrestricted inlet duct as shown. The inlet flow distribution was thus much better than was obtained in the cell tests quoted in Ref. 17.

The mass flow was measured by an I.S.A. nozzle situated in the inlet duct. To cater for velocity profiles outlet total-pressure measurements were taken by means of four combs of five pitots, equally spaced around the annulus. The tubes were equi-spaced radially, and an arithmetic mean was taken as the total pressure at the comb. The inlet total pressure was taken equal to the mean static pressure in the inlet duct. Wall statics were used to measure static pressures. In addition to the overall measurements, the static pressure at the casing was measured between each stage to obtain some indication of the stage operating conditions. Two points located diametrically opposite each other were used for this purpose. The arithmetic mean of these was taken as the static pressure between the rows.

The temperature rise across the blading was measured directly. Resistance elements were used at inlet and six thermo-couples equally spaced around and across the annulus at outlet. In addition, check thermo-couples were placed in the inlet duct and the outlet duct for the later tests.

The instrumentation is fully illustrated in Fig. 8, showing location of instruments, etc. It is considered that the absolute accuracy of the major performance parameters are :

Mass flow	$\pm$ 2 per cent
Efficiency	$\pm$ 2 per cent
Pressure	$\pm$ 1 per cent.

Relative accuracy is probably about half the above values.

In order to keep within the available power it was necessary to resort to inlet throttling at the higher speeds. It is not thought that this would have any significant effect on the performance, though the efficiency may perhaps be some 1 per cent below the value that would be obtained with standard inlet conditions, at the design speed. The Table below gives the mean inlet pressure and temperature for each speed, and indicates the degree of throttling employed.

Speed (r.p.m.)	Inlet total pressure (lb/sq in.)	Inlet total temperature (deg K)
4,000	14·5	288
5,000	14·4	288
6,000	14·4	291
6,500	14·1	299
7,000	13·6	295
7,500	11·1	294
8,000	10·3	295
8,500	8·6	293
9,000	8·2	296
9,500	7·3	295
10,000	6·6	295
10,500	5·3	295

The Reynolds number at entry to the first-stage rotor at the design point is thus 3×10^5. This is well above the critical value.

5.2. *Test Results.*—The overall characteristics obtained from two series of tests are given in Fig. 9. The first series of tests gave characteristics up to design speed ; the second series includes overspeed tests and also forms a check. For the second series two blades were removed from stage one. This was necessary because the fir-tree broachings in the discs had previously crumbled and were unsafe for overspeed running.

It will be seen that over the duplicated range, agreement between the two series of tests is good except at 4,000 and 5,000 r.p.m. The latter curves have, in fact, been checked several times and it would appear that the second series is the true result. The two tests of the first series were conducted at sub-zero inlet temperatures and it is believed, but cannot be substantiated, that icing of the inlet thermometers gave spurious readings. In Fig. 10 the test results obtained on the new blading have been compared with those obtained on the original uniform blading with both a ducted inlet and cell test technique.

The individual stage characteristics have been calculated from the inter-stage static pressures and plotted in the usual non-dimensional manner in Figs. 11 and 12. The corresponding curves for the original blading are given in Figs. 13 and 14. The points exhibit the usual scatter associated with this work, but mean curves have been drawn through the points. The curves for each stage of the two compressors have been reproduced together for comparison purposes in Fig. 15.

6. *Discussion of Test Results.*—The most significant feature of the test results is the disappearance of the pronounced kink which existed in the surge line of the original compressor. Considerable improvement has been achieved in the speed range 7,500 to 9,000 r.p.m. as can be

seen from Fig. 10. The efficiency is also much higher in the mid-speed range. Both these features are, of course, attributable to a lower stalling speed (*i.e.*, high stalling incidence) of the first stage or stages. Reference to the comparative stage characteristics plotted in Fig. 15 confirms that the flow coefficient for peak pressure rise has been considerably reduced for each of the first two stages. This was the design objective. However, it may be queried how much of this is due to the change of blade section, and how much due to change of blade twist. Clearly such factors cannot be separated other than by elaborate test, for not only are the fluid velocity triangles resulting from radial equilibrium or otherwise in question ; additionally, the choice of design incidence for a given velocity triangle could materially influence the stage performance. Fortunately, the hub/tip ratio for this stage is not low (it is $0 \cdot 6$ at entry to the rotor) and in such circumstances it is the authors' experience that minor changes in blade twist have little effect on the stalling flow coefficient (this may not be true of low hub/tip ratio stages). It is therefore believed that the delayed stalling of the revised design is due to the change in blade profile.

It was also hoped that the change in profile would result in a higher efficiency at overspeed (*i.e.*, at 10,000 r.p.m. and 10,500 r.p.m.) by virtue of the higher drag critical Mach number. Direct comparison with the original blading is dogged by discrepancies between the ducted inlet and cell test techniques, but generally speaking, this design objective was not fulfilled, though a peak efficiency of 86 per cent was recorded at 10,000 r.p.m. (mean stage temperature rise of 33 deg C). This is an acceptable working efficiency, but it is on the falling part of the efficiency envelope. Furthermore, the mass flow achieved at overspeed has been limited by the small throat area of the circular-arc cambered first stage (*see* Fig. 7), choking of this stage being the controlling factor at overspeed. The slightly higher general efficiency of the revised design could be attributed to experimental error, but the much flatter efficiency characteristics must be due to the circular-arc cambered blades in the low pressure stages. The efficiency of a reasonable working point at 9,000 r.p.m., which is the highest speed at which direct comparison is possible, would be several points higher with the new blading. Unpublished model tests of the original first stage having parabolic ($a/c = 40$ per cent) arc camber-lines showed that the efficiency started falling, particularly at high flow coefficient, just below an equivalent speed of 9,000 r.p.m. ; this is presumably due to shock stalling. There is no evidence of any narrowing of the range in this blading at that speed. Otherwise there is little to indicate that very much has been achieved at high speed by the redesign.

The above comparison of the variable-profile and constant-profile blading used in this compressor is perhaps of secondary interest. The superiority, for general purposes, of the circular-arc camber-line has long been accepted. Nearly all axial compressors nowadays have circular-arc cambered blades in all stages, and the practical question is whether any advantage can be gained by using parabolic camber-lines with the position of maximum camber 40 per cent chord from the leading edge in the high pressure stages.

In order to illustrate this aspect, a design exercise was carried out in which the last stator row was supposedly replaced by one having circular-arc cambered blades. The assumptions made were that the fluid outlet angle from the row had to be identical with the original values and that the blade inlet angle remained unchanged, *i.e.*, for a given air inlet angle the incidence was the same (an exercise in which the incidence was increased by two degrees was also carried out but the conclusions reached were substantially the same). The throat areas for the circular-arc cambered blades were found to be only 88 per cent of those possible using parabolic camber-lines. It follows that a compressor having such blades would choke at a lower flow, and in Fig. 16 the characteristics that would be obtained have been compared with the test results. It illustrates the severe curtailment of the characteristics which would occur if circular-arc cambered blading had been used throughout. In particular the appearance of the familiar ' short characteristic ' (*i.e.*, one of very limited flow range) will be observed at 7,000 r.p.m. which is the kink speed for this compressor. Such characteristics often appear in the kink region. It is also significant that the peak efficiency points have been lost at some speeds due to the choking, so that the effective maximum efficiency is reduced. It should, however, be noted that the superior high-incidence performance of the circular-arc cambered blades could result in a higher surge pressure ratio at the higher speeds. Since the low-speed performance would be unchanged, a kink would thus be reintroduced into the surge line. This is also illustrated in Fig. 16 where the revised surge line has been estimated by assuming that, at the matching speed, the circular-arc blades could operate at 2·5 deg higher incidence on the last stage before surge occurred. This may not be fully realised in practice. Even so, one would expect that a compressor with circular-arc blading throughout would achieve a higher design-speed surge pressure ratio but at the expense of peak efficiency, loss of working range, and a poor surge line. But this is not the whole story. Fig. 17 shows the radial velocity distribution (expressed as Mach-number distribution) that was measured at outlet from the compressor at 9,500 r.p.m. and 8,000 r.p.m. Curves are given for a point near surge and for an operating point on each characteristic. Also plotted on these Figures are the outlet Mach-number distributions which would be obtained if each blade-section throat were choked and the flow was confined to concentric cylinders. In calculating these curves no loss was assumed to occur between the throat and the downstream measuring section. This is not true and the calculated values will thus be somewhat lower than the actual values. In spite of this they illustrate a significant feature. If circular-arc cambered blades had been used, the centre section of the blade would probably have been choked even at the surge flow. Some loss of efficiency could be expected on this score. Additionally, a redistribution of the flow would be necessary within the compressor, and such forced redistribution is usually accompanied by additional losses, even at low speeds (*see* Ref. 21, for example). The loss of efficiency from this source cannot be calculated and so efficiency curves for the fully circular-arc compressor have not been included in Fig. 16. By contrast, the larger throat areas of the parabolic-arc ($a/c = 40$ per cent) cambered blades allow the natural velocity profile to develop without any choking and loss of efficiency. This profile may not appeal to the combustion-chamber designer, but is most probably desirable for high compressor efficiency.

One noteworthy feature of the test results which was not expected is the high efficiency recorded at the high-flow end of the constant-speed characteristics. For example, at choke at 8,000 r.p.m. the efficiency of the variable-profile blading is some 15 per cent higher than that recorded on the constant-profile blading. Choking phenomena, as discussed above, cannot account for this because both compressors have the same outlet stages. It can be argued that in the mid-speed range the earlier stages will not be stalled, and therefore present the latter stages with a much more uniform velocity profile. Although some improvement can be expected on this score, it is difficult to believe it would give rise to the increase in efficiency actually observed. No evidence can be obtained from the stage characteristics, since interpretation of detail effects is obscured by the scatter of the results. This feature must therefore remain partially unexplained.

Clearly, the main advantages of the type of blading discussed in this report are associated with part-load operation, though the high efficiency near choke could be a valuable asset in certain applications. Ease of starting is therefore the salient gain for orthodox gas turbines, but in any application which demands operation over a wide speed range the good surge-line shape and high efficiency at mid and low speeds are obviously of paramount importance. The speed range may be mechanical, as in some naval or industrial applications, or aerodynamic as in the case of aero-engines for high-speed flight. The flow per unit of frontal area of the test compressor is too low for the aero-engine application, but contemporary values could easily be achieved using a transonic ' O stage '. It is interesting to note that such a stage, having blades of circular-arc camber-line and position of maximum thickness 50 per cent chord from the leading edge, falls into the general design pattern and philosophy of the test compressor. The design technique for such blades is now available, though adjustable inlet guide vanes will probably be necessary to maintain the incidence of these blades within its narrow limitations.

Finally, some reference must be made to reliability. The whole object of the design has been to provide blades which operate at high efficiency over as much of the operating range as possible, with a view to improving the aerodynamic performance. Inasmuch as stalling, shock stalling and choking are delayed, one may expect that flutter and stall cell excitation of the blades will be confined to a minimum. These are the causes of most blade failures. So although no experimental evidence is available, other than the fact that the compressor survived 55 hours of test running without a blade failure, it seems reasonable to believe that a compressor having properly chosen blade profiles in each stage will be more reliable than one having the same profile in all stages.

7. *Conclusions.*—As a result of the work carried out it may be concluded that a compressor with appropriately chosen blade profiles in each stage offers some or all of the following advantages :

(*a*) It will have a good surge line

(*b*) Its part-load efficiency will be high

(*c*) It will have a wide characteristic at constant speed

(*d*) It may have a high efficiency at high flow

(*e*) It will have a higher peak efficiency than some designs

(*f*) It should be more reliable.

A possible disadvantage is that it may have a lower surge pressure ratio at design speed than a compressor with ' high load ' blade sections throughout.

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18	A. D. S. Carter	The low-speed performance of related aerofoils in cascade. C.P. 29. September, 1949.
19	A. D. S. Carter and A. F. Hounsell	General performance data for aerofoils having C1, C2 or C4 base profiles on circular-arc camber-lines. (Unpublished M.o.S. Report.)
20	R. A. Jeffs, A. F. Hounsell, and R. G. Adams	Further performance data for aerofoils having C1, C2 or C4 base profiles on circular-arc camber-lines. (Unpublished M.o.S. Report.)
21	S. J. Andrews, R. A. Jeffs, and E. L. Hartley	Tests concerning novel designs of blades for axial compressors. R. & M. 2929. October, 1951.

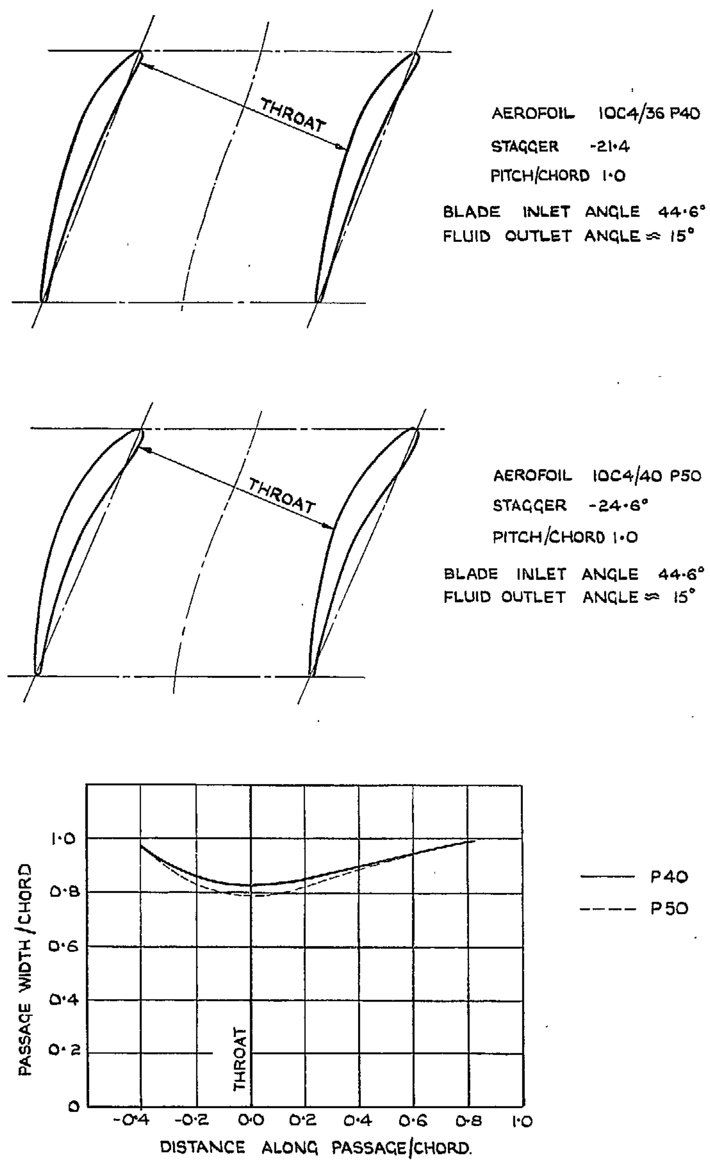

FIG. 1. Throat widths for P.40 and P.50 camber lines.

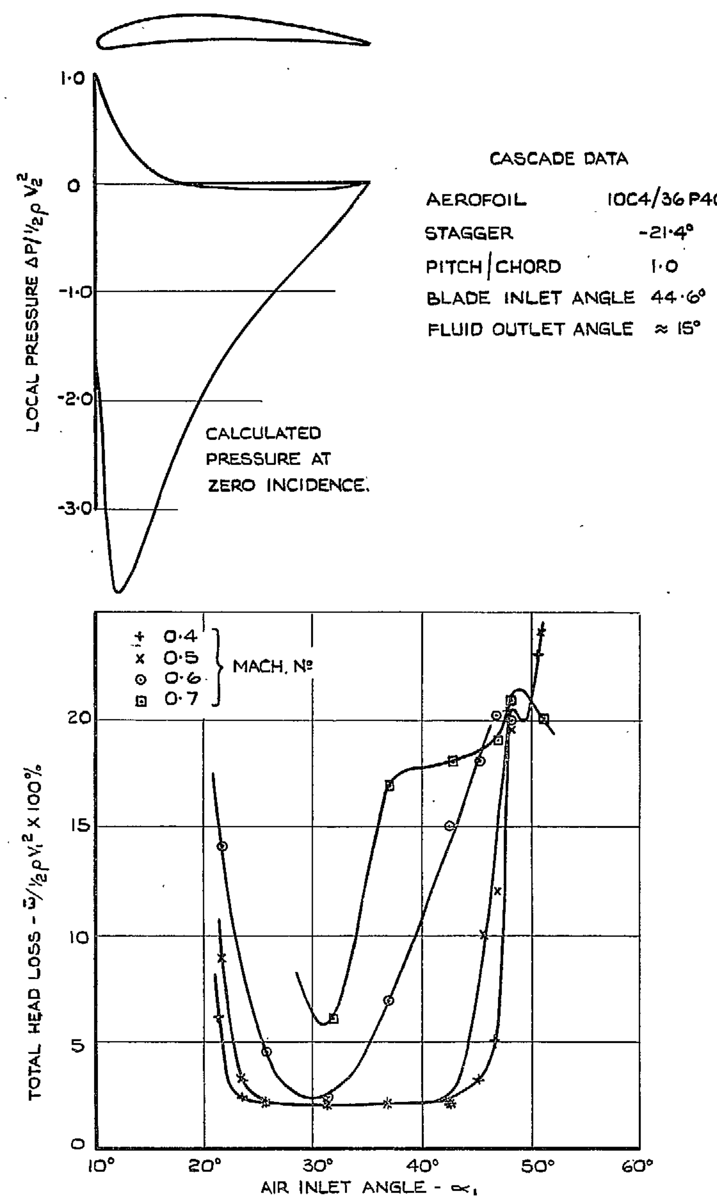

FIG. 2. Cascade performance. C.4 base profile. P.40 camber.

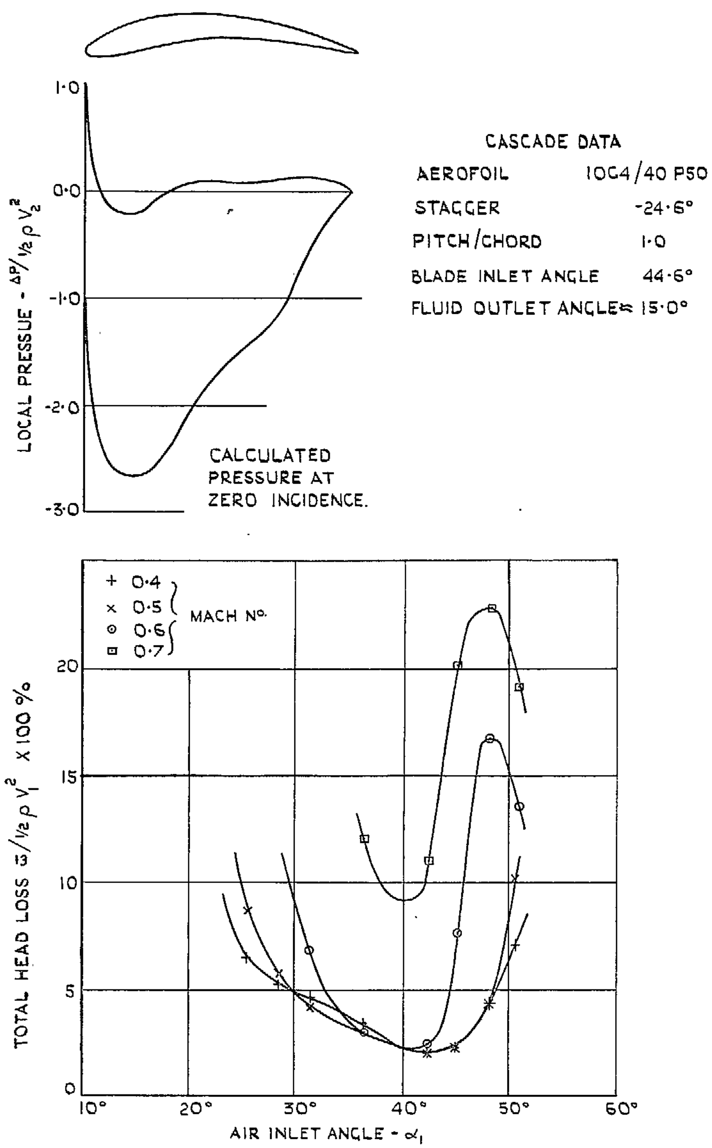

Fig. 3. Cascade performance. C.4 base profile.
P.50 camber.

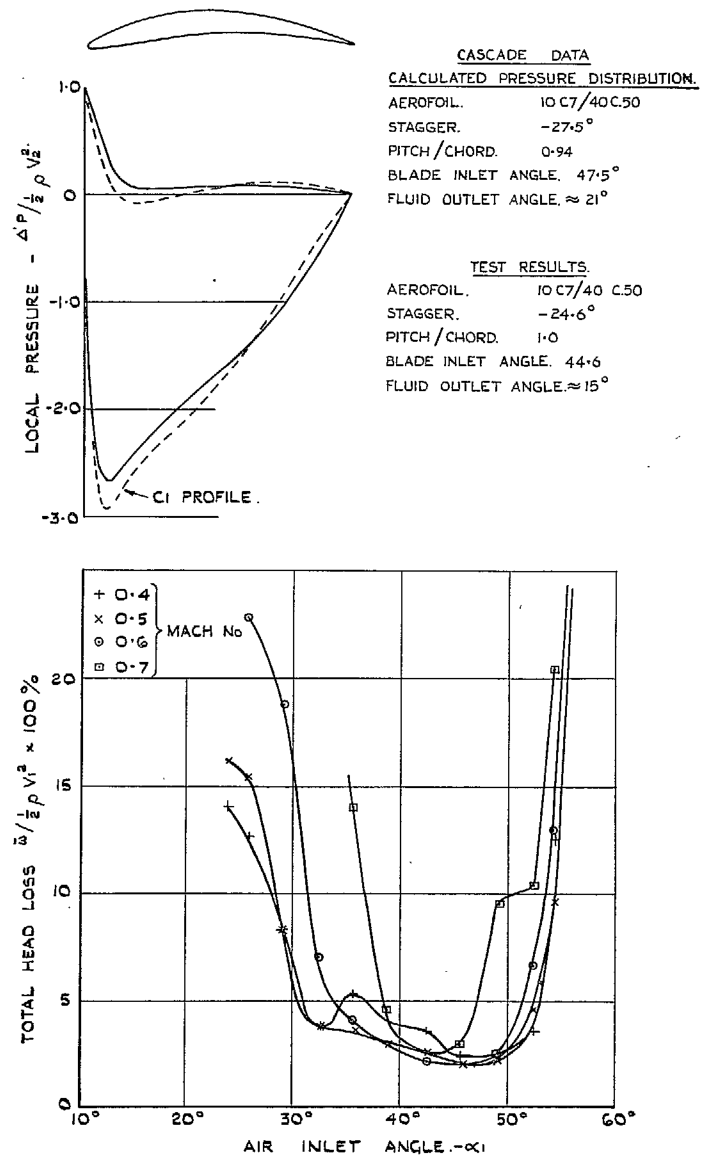

Fig. 4. Cascade performance. C.7 base profile.
C.50 camber.

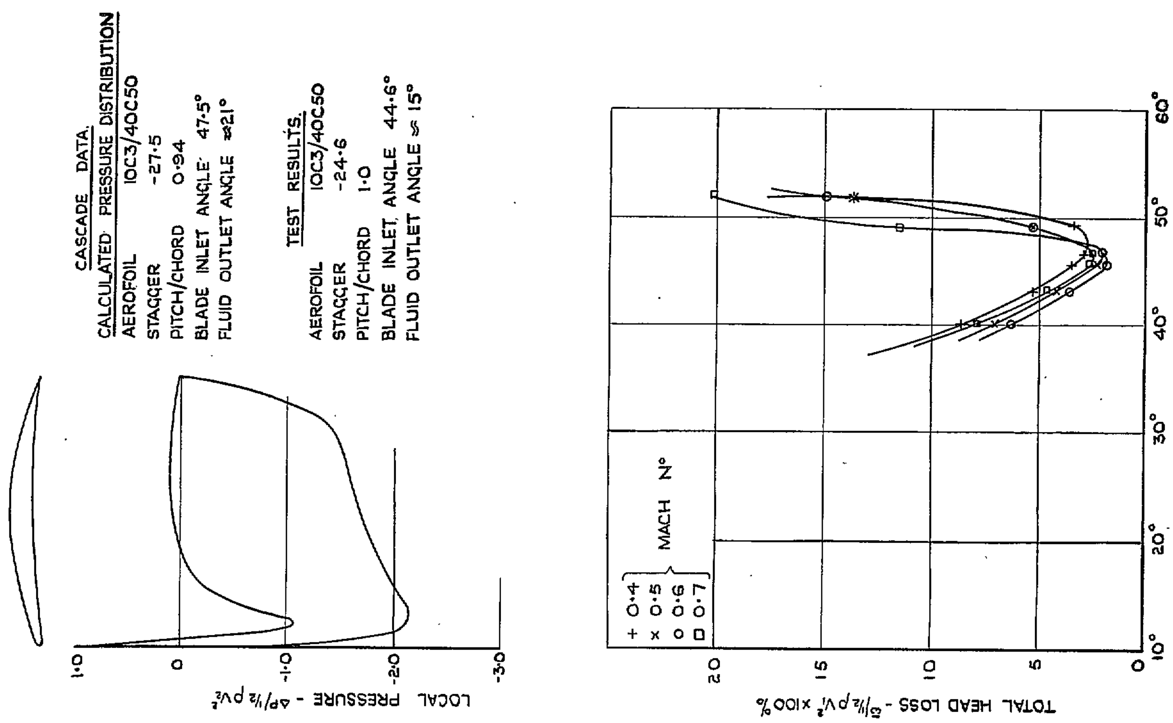

FIG. 5. Cascade performance. C.3 base profile. C.50 camber.

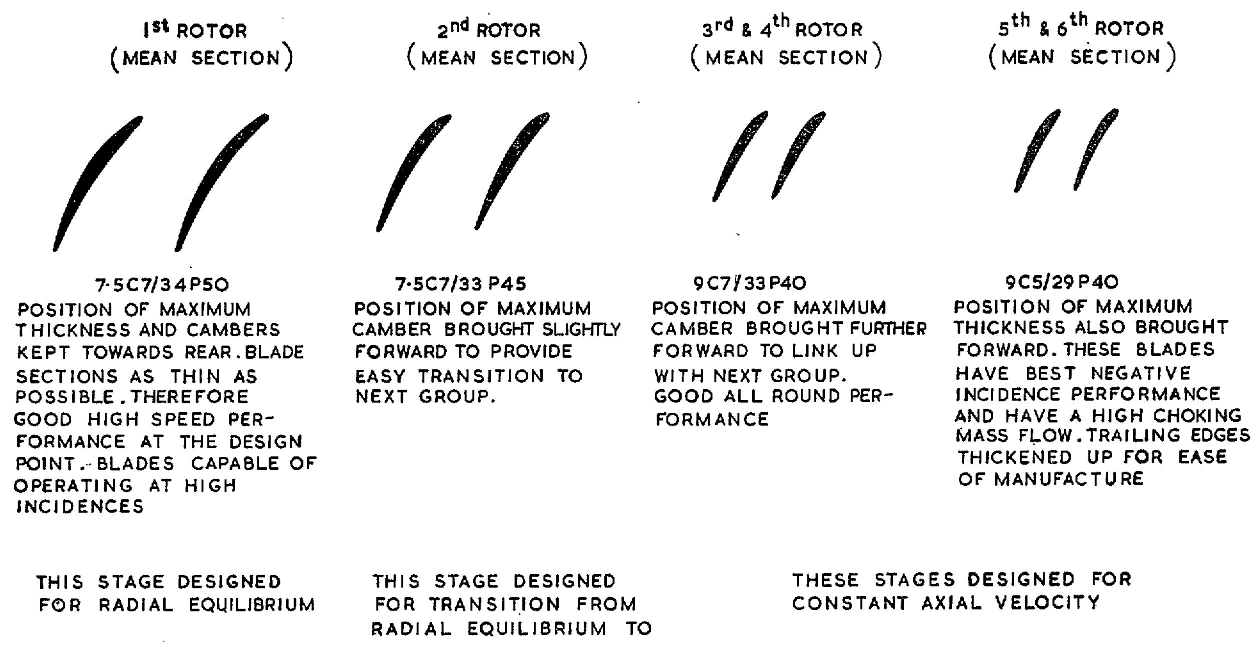

FIG. 6. Compressor blade mean-diameter profiles and notes.

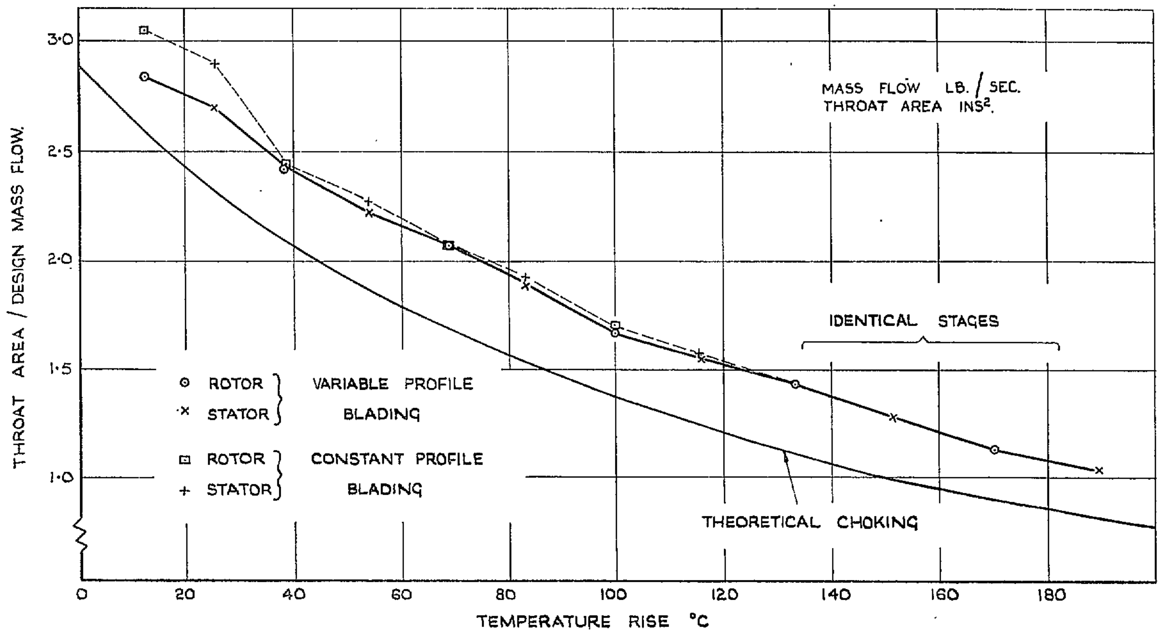

FIG. 7. Comparison of compressor blade throat areas.

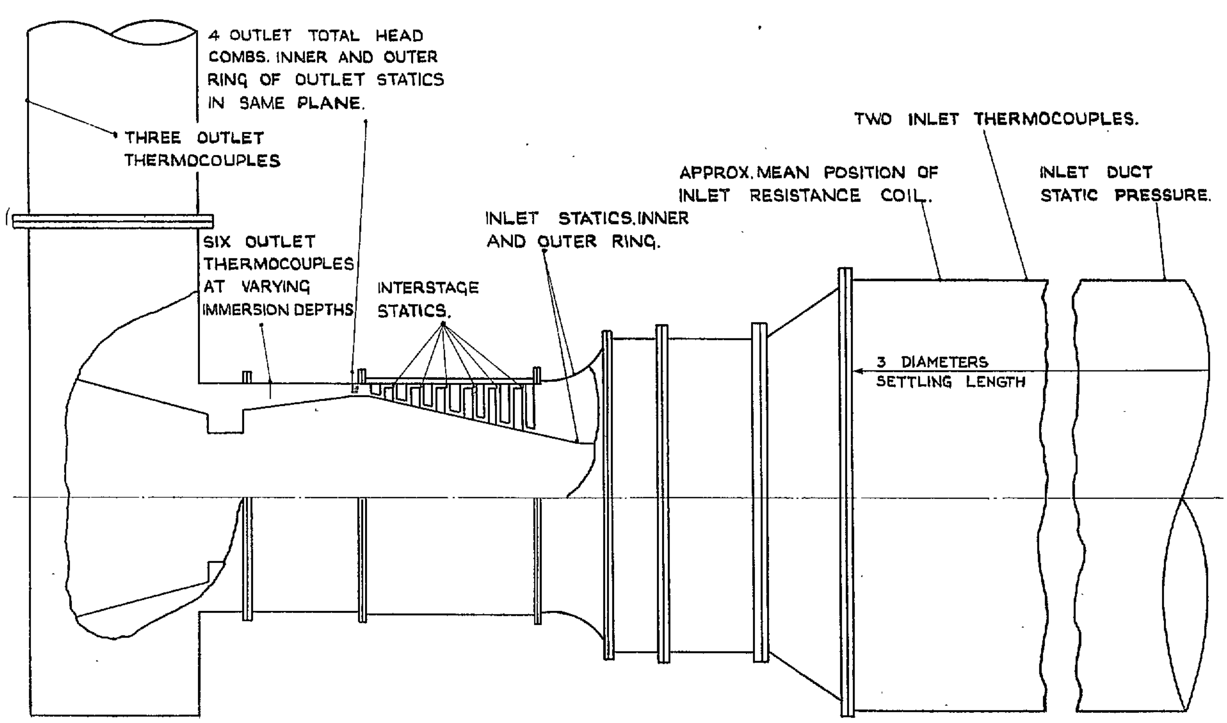

FIG. 8. Instrumentation.

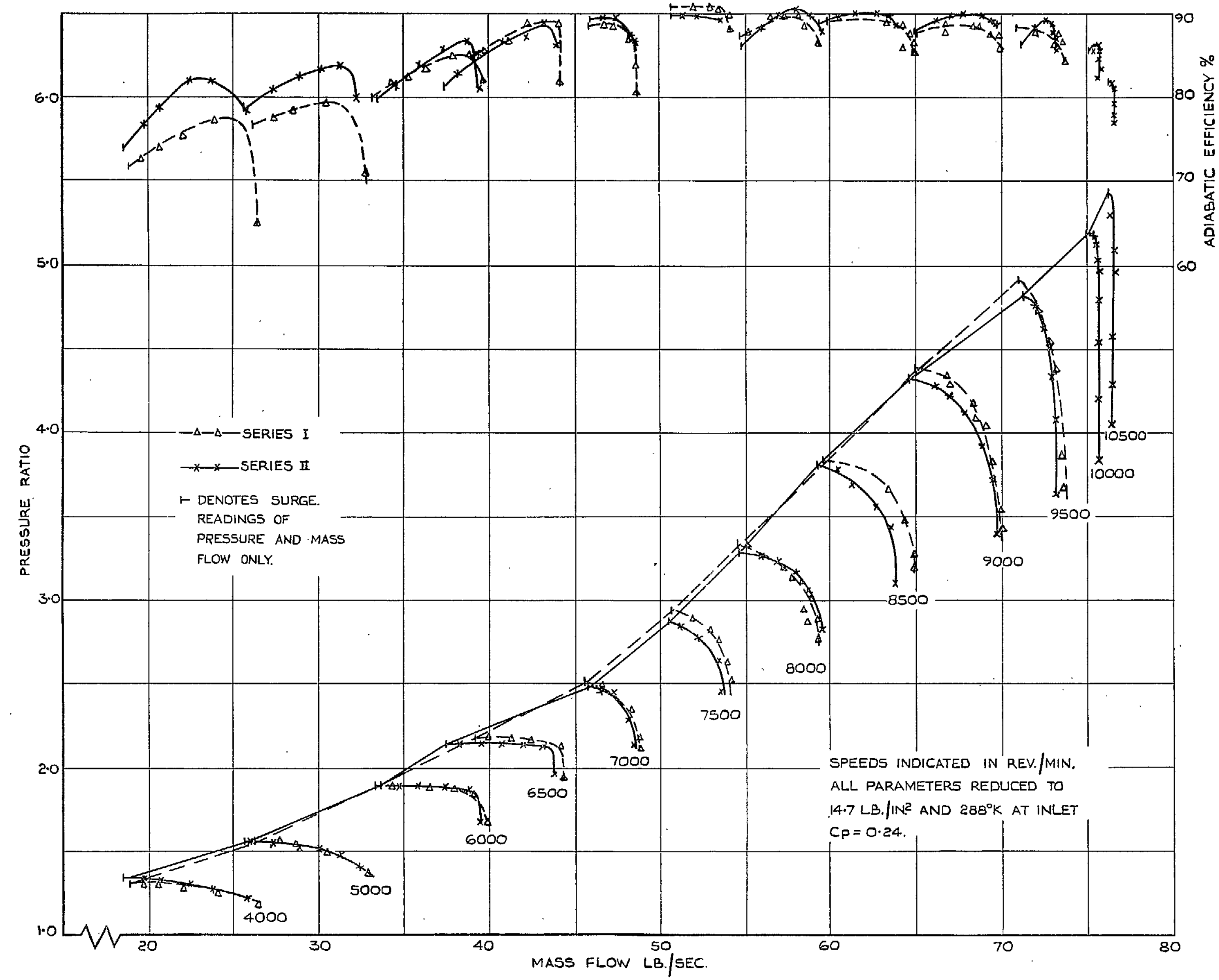

FIG. 9. Overall characteristics.—Variable-profile blading.

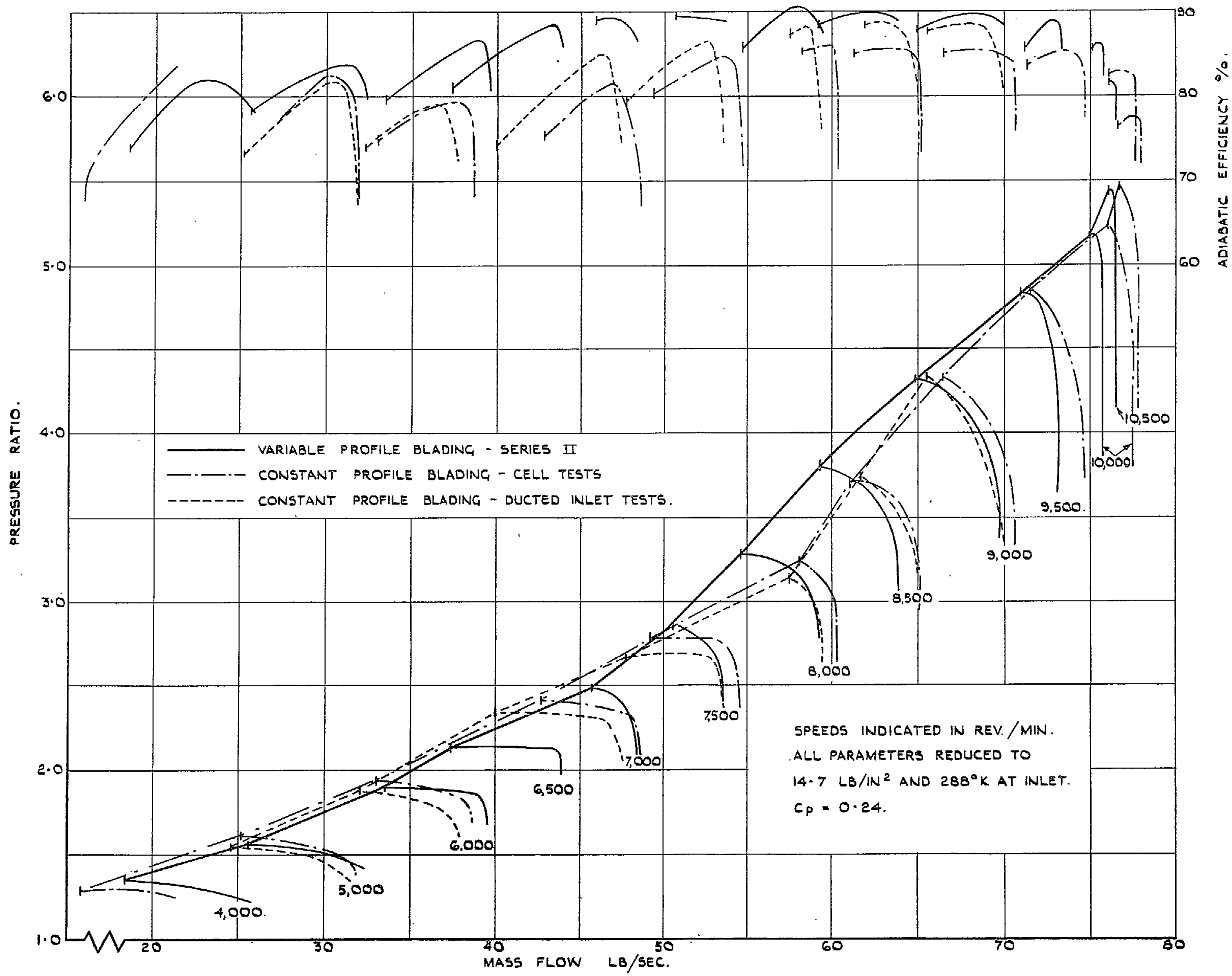

FIG. 10. Comparison of overall characteristics.

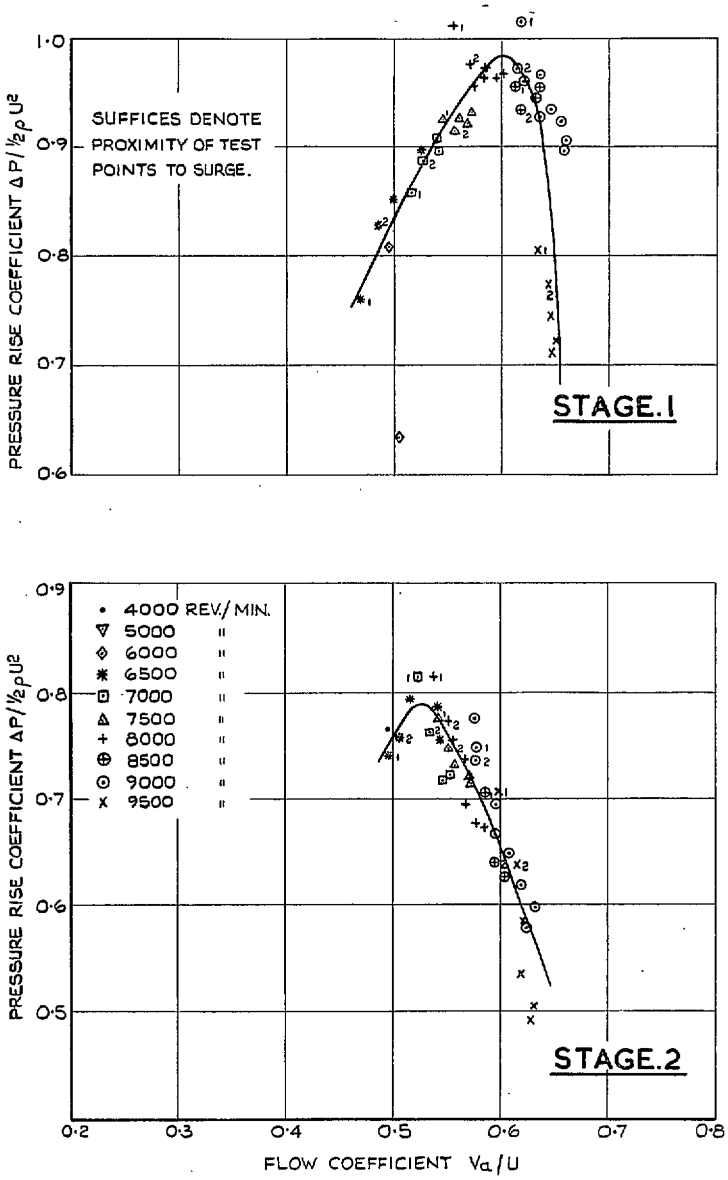

FIG. 11. Variable-profile blading. Stage Characteristics. —Stages 1 and 2.

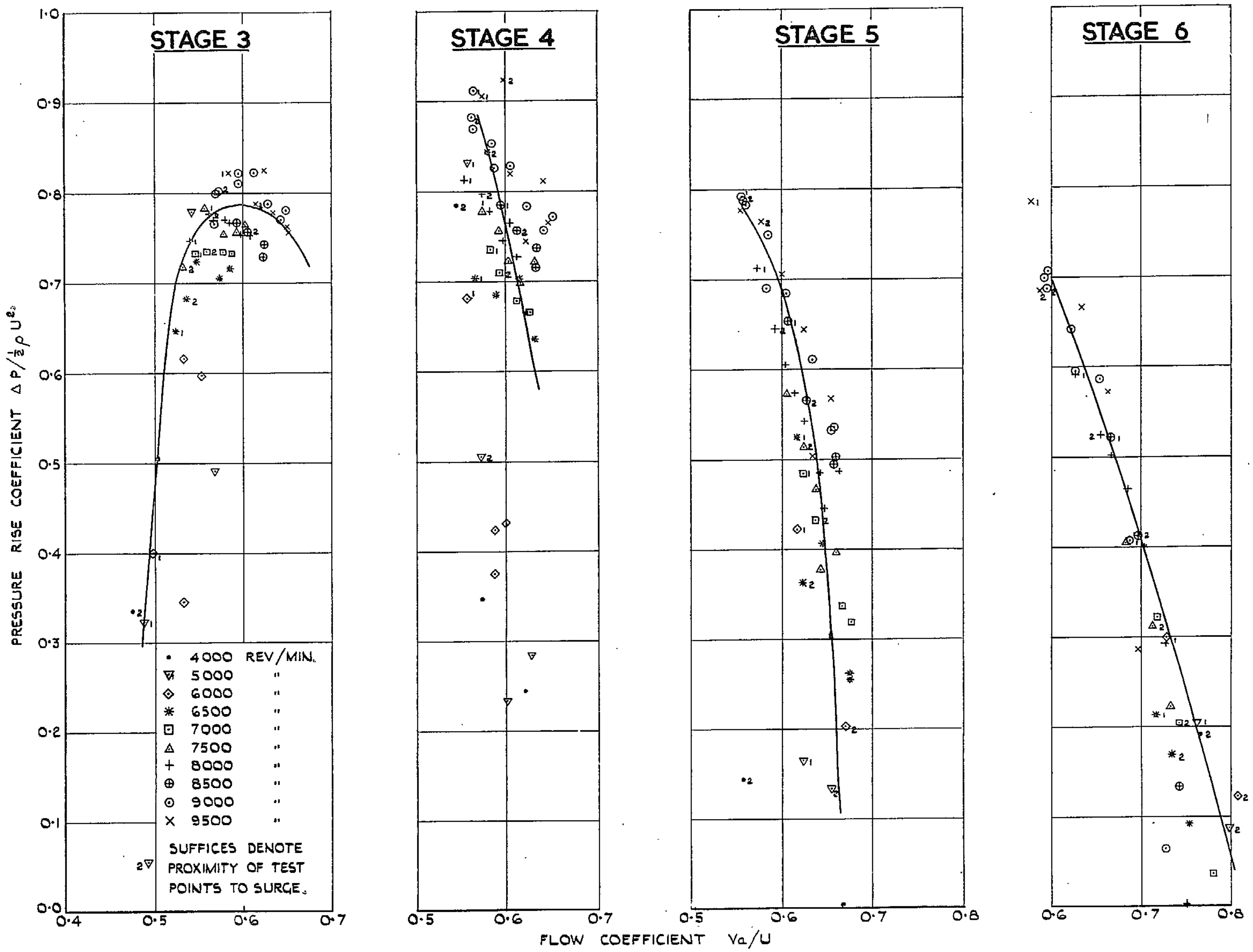

FIG. 12. Variable-profile blading. Stage characteristics.—Stages 3 to 6.

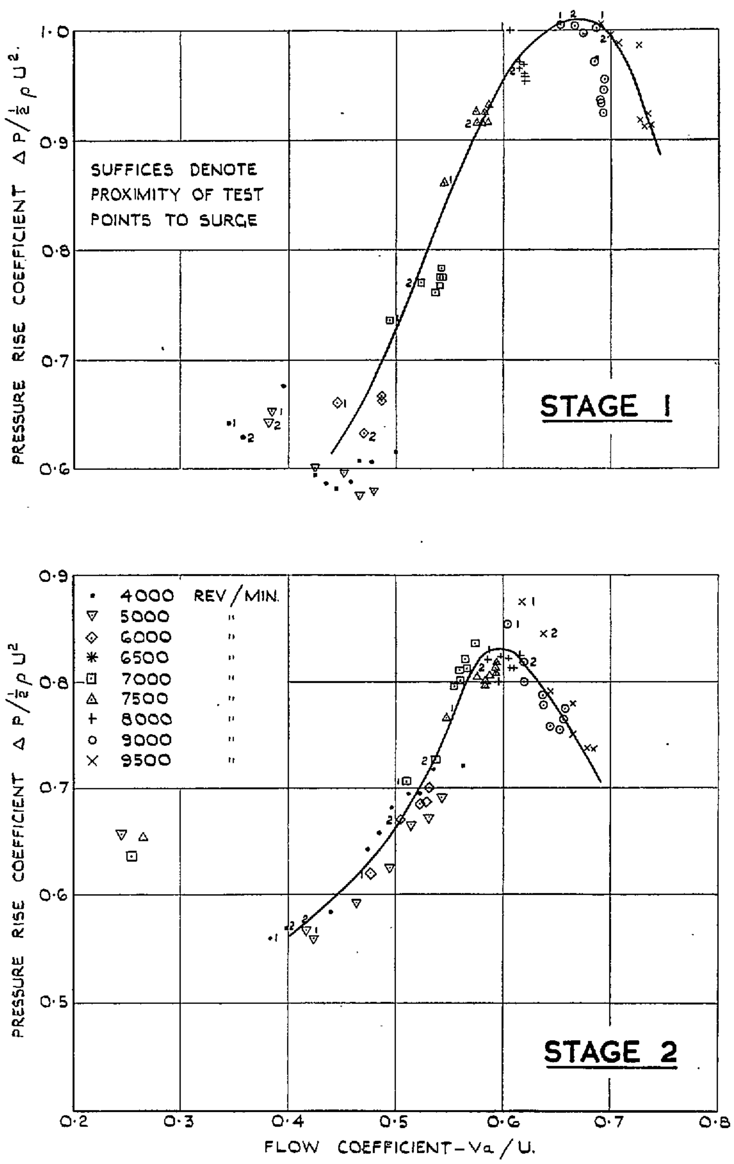

FIG. 13. Constant-profile blading. Stage characteristics. —Stages 1 and 2.

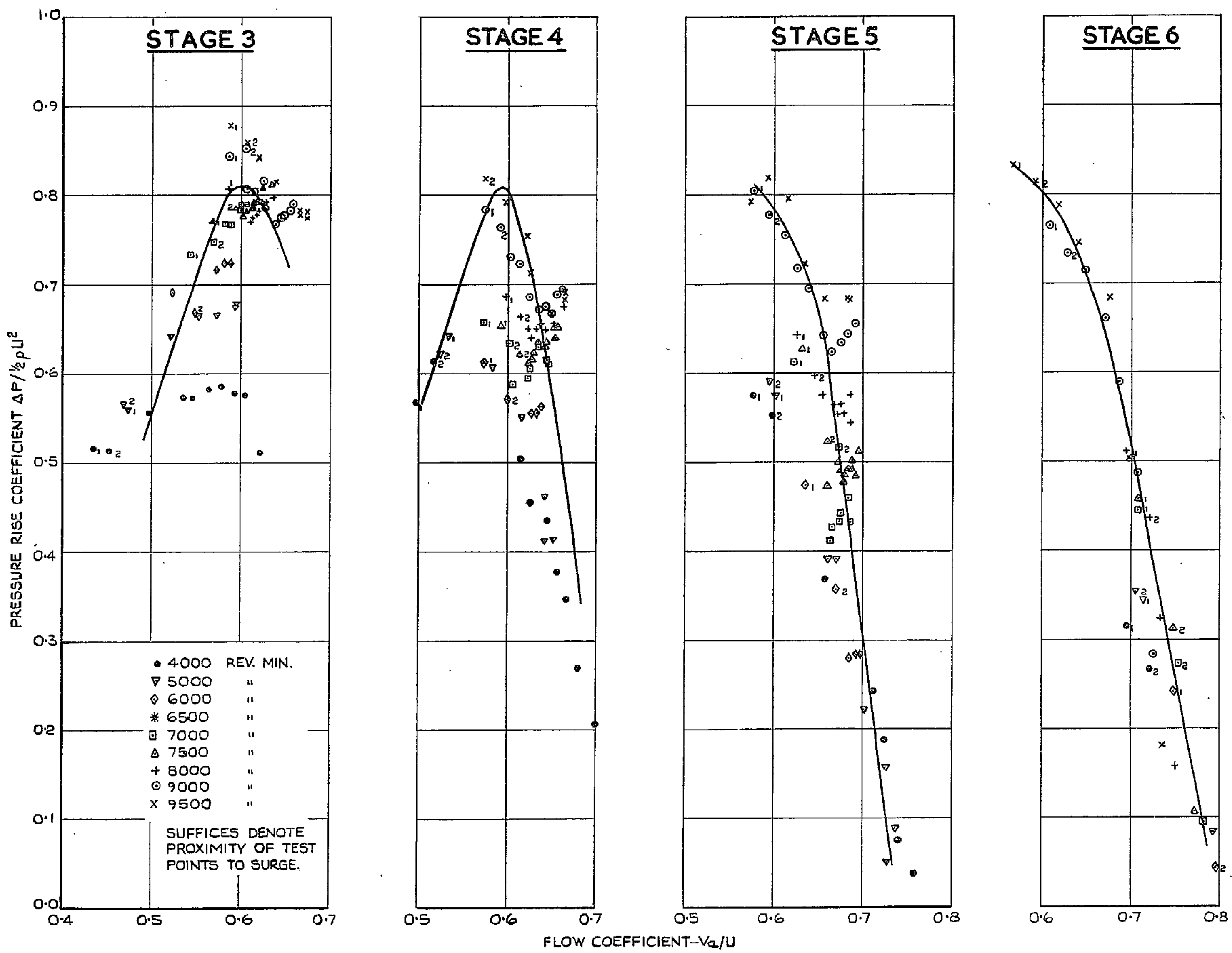

FIG. 14. Constant-profile blading. Stage characteristics.—Stages 3 to 6.

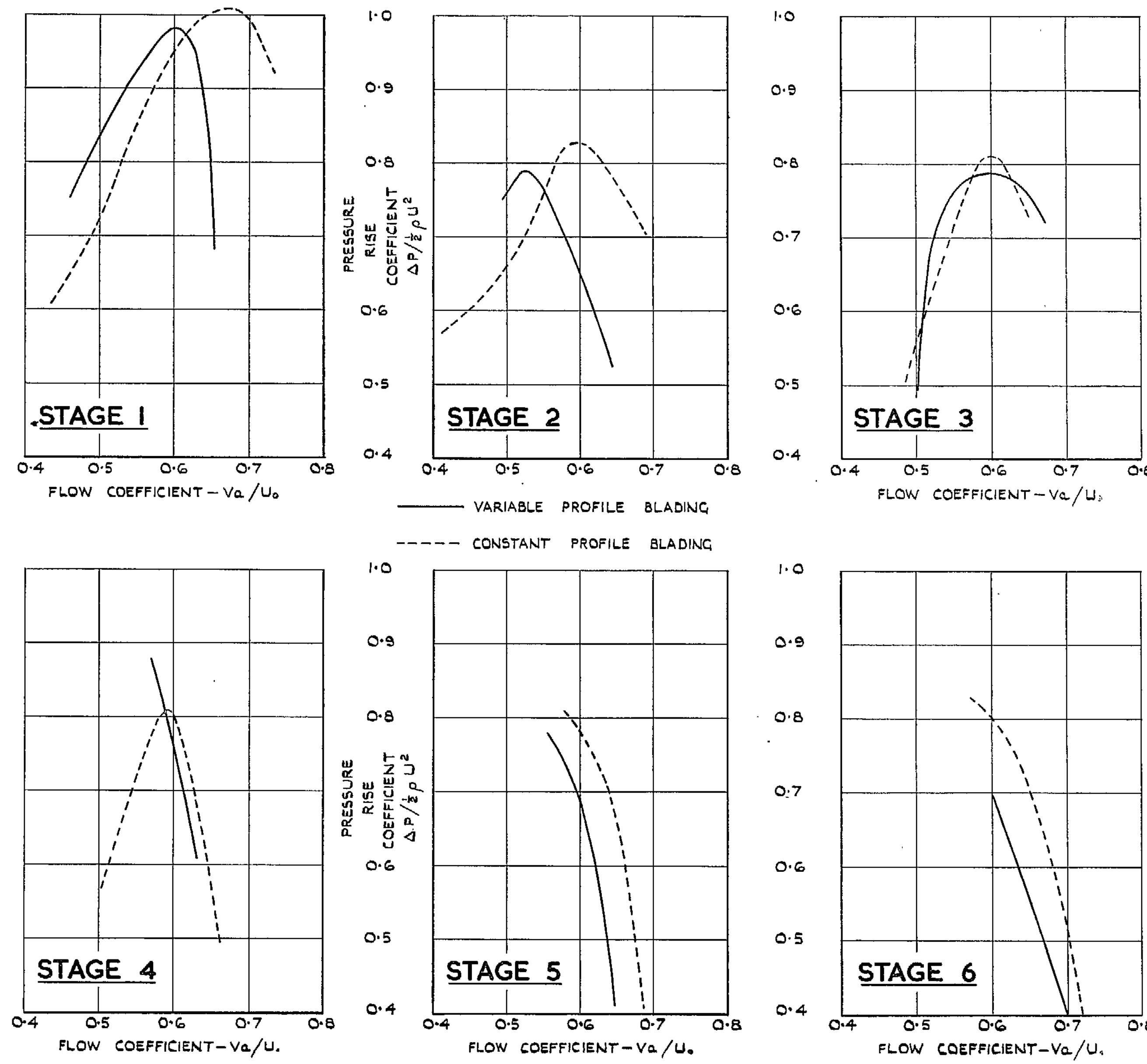

FIG. 15. Comparison of stage characteristics.

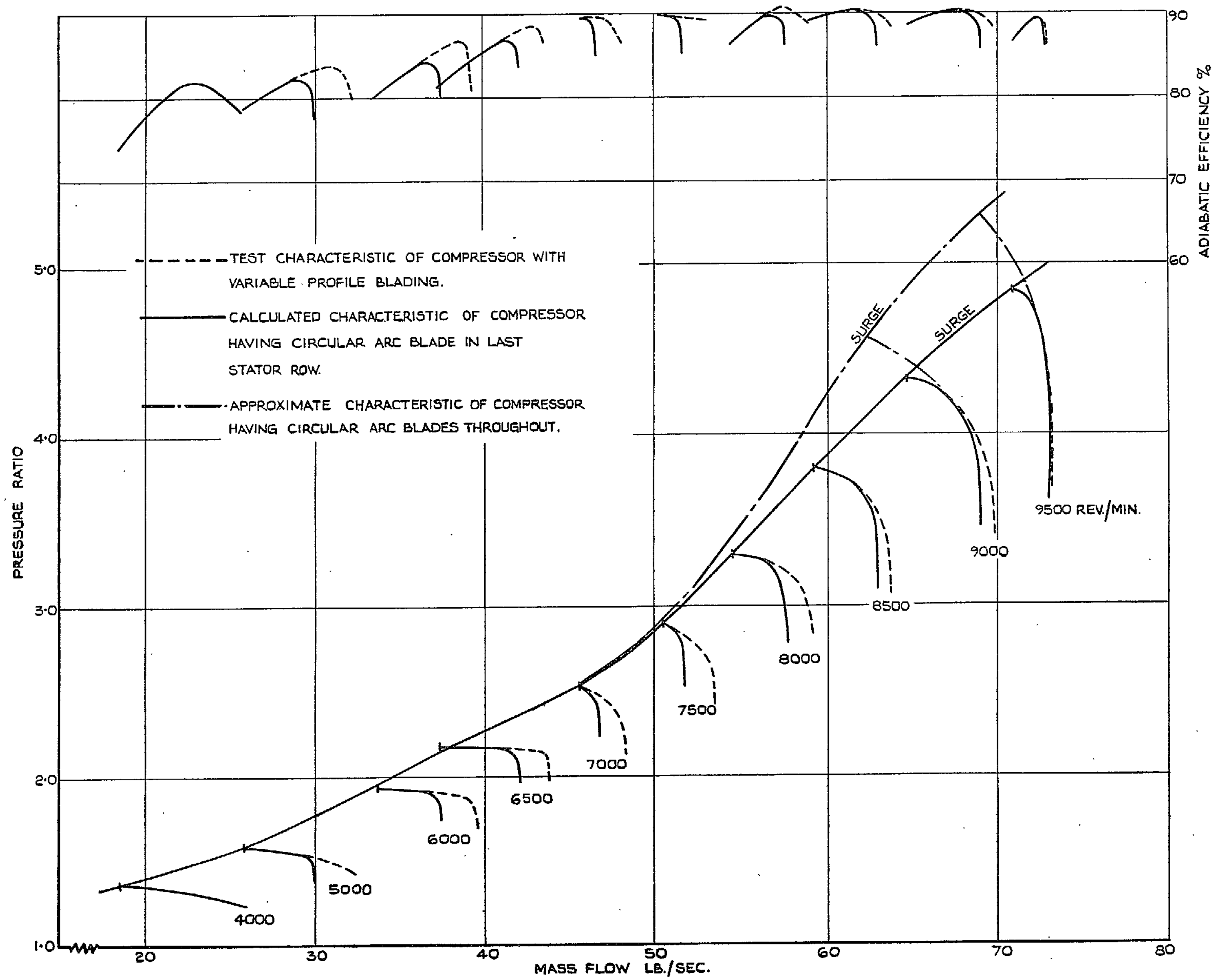

FIG. 16. Calculated characteristics with hypothetical circular-arc blading.

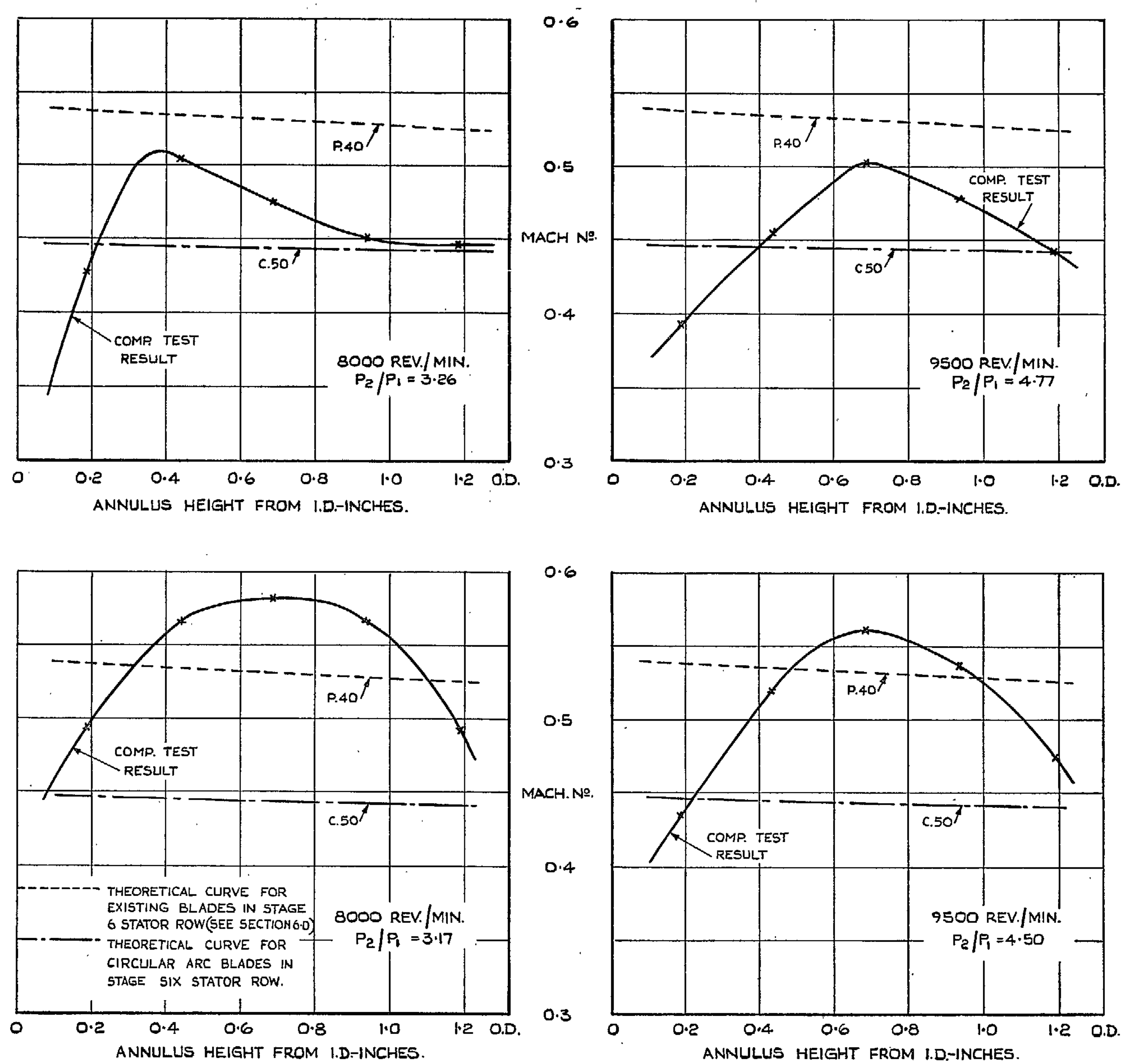

FIG. 17. Mach-number profiles at outlet.

(80622) Wt. 61/991 K.7 12/60 Hw.

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