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The Low Speed Performance of Low Stagger Compressor Blading at Three Pitch/Chord Ratios

by

R.C. Turner and R.A. Burrows

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SUMMARY

Two stages of low speed, low stagger blading were tested at mean diameter pitch/chord ratios of 0.49, 0.66 and 0.94. Contrary to the indications of some earlier published work, there was no significant fall-off in efficiency at the lowest pitching, and the trend of the performance was in line with that predicted by simple theory.

These results have a considerable bearing on the compression of light gases, where low pitch/chord ratios are needed to produce high stage temperature rises, in order to keep the number of stages reasonably small.

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1.0 Introduction

In compressors intended for use with light gases (e.g. hydrogen or helium), it is desirable to design for a high stage work so that the number of stages may be kept to a reasonably low figure. The high specific heats of these gases (e.g. 1.24 helium) result in low stage temperature rises for given work inputs, stage work being a function of stage velocities and geometry, both of which are usually confined within fairly narrow limits. Since the overall temperature rises required are of the same order as those for air compressors, a large number of stages may be necessary. This could result in high costs and also in mechanical difficulties, particularly those associated with low whirling speeds.

There are several methods of increasing the stage work; they are reviewed in Reference 1. One is to design for a high axial velocity. Since the peripheral blade speed is limited by stress considerations, this approach results in "high V_a/U" or "low stagger" blading, if blading of approximately 50 per cent reaction is considered. The magnitude of the axial velocity may be limited by the permissible entry and exit losses and also by Mach number effects on the blades. The latter are less likely to be encountered with light gases however, as the sonic velocities are high compared with that of air (3240 ft/sec for helium, 1100 ft/sec for air at 288°C).

Another method is to use lower pitch/chord ratios than are currently employed in the blading of conventional compressors. Reduction of pitch/chord ratio enables a higher deflection and therefore stage work to be obtained - see for instance Reference 2. Here again, Mach number and choking difficulties which may arise with air at small pitch/chord ratios are less likely to be encountered with the light gases. There is, however, little published information on the performance of compressors designed with low pitch/chord ratios, and some tests reported in Reference 3 have suggested that at least with medium stagger blading, stage performance falls off sharply at the lower values (e.g. about 0.5) in a manner which is not predicted by simple theoretical methods[2].

The purpose of the present tests was to compare the performance of low stagger blading at low and conventional pitch/chord ratios and to find if any performance change could be predicted by conventional methods. In particular, it was desired to find whether there was any large drop in performance at the lowest pitching such as that described in Reference 3. The tests were carried out in air at low Mach numbers on two stages of blading, no facility being available at the time for tests on a larger number of stages.

2.0 Apparatus

2.1 The compressor

The 108 compressor is a low speed machine of constant annulus dimensions. It is described in detail in Reference 4 and a layout is shown in Figure 1, the associated ducting being shown in Figure 2. The outside diameter is 40 in. and the inside/outside diameter ratio is 0.7. For the present tests, two stages of blading were used. The maximum running speed is 900 rev/min, this giving a Reynolds number of 2.1×10^5, based on the blade chord and the mean diameter blade peripheral speed.

2.2 The blading

The blading was that described in Reference 4. It was originally designed for free vortex conditions with 50 per cent reaction at the inside diameter and a mean diameter flow coefficient of 0.8. The blade chord was 3 in., giving an aspect ratio of 2.0. The blade profiles were C4 on a parabolic (P40) camber line. Full details are given in Appendix I. For the present tests, the roots of both rotor and stator blades were modified and spacers were provided so that tests could be carried out with 72, 54 and 38 blades/row, the stagger angles being unchanged. The inlet guide blades were kept at the original figure of 48 blades/row. The pitch/chord ratios for the rotor and stator blades at the mean and inside and outside gauge diameters are given below. The general appearance of the blade passages for the 72 and 38 blade builds is shown in Figure 3. In all the tests, the numbers of rotor and stator blades were equal.

Position	i.d.	m.d.	o.d.
Diameter (in.)	29	34	39
s/c (72 blades/row)	0.42	0.49	0.57
s/c (54 blades/row)	0.56	0.66	0.76
s/c (38 blades/row)	0.80	0.94	1.07

The blading is fairly described as of the "low stagger" variety, although the effect of the 50 per cent reaction condition at the inside diameter and the free vortex twist is to put the rotor mean diameter section, when considered individually, into the "medium stagger" category (air outlet angle in the 20° to 35° range). For the 54 blades/row build, the theoretical mean diameter air outlet angle is 24.6° for the rotor and 1.5° for the stator, the mean of these two values being 13.1°.

2.3 Instrumentation and measurements

Mass flow was measured by means of a venturi in the delivery ducting (Figure 2), which had previously been calibrated by detailed traversing. Torque was measured via the reaction on the stator of the driving motor, which was swung in trunnions. The bearing friction torque was estimated by measuring the torque at a series of low speeds, at which the aerodynamic torque was negligible, and extrapolating to zero speed. Speed was measured by means of a tachronometer, a device which times a known number of revolutions. Static pressures were measured at nine points round the circumference of the casing after the inlet guide blades and were used, with the measured mass flow, in the estimation of the inlet total pressure. The total pressure after the blading was estimated from the readings of four nine-point pitot combs equi-spaced round the annulus, two chords downstream of the second-stage stator blades. Pressures were indicated on Betz manometers and sloping water manometers.

3.0 Test procedure

For each blade pitching, the mean characteristics of the two stages were determined at speeds of 900 and 500 rev/min. In addition, measurements were also made at 250 rev/min for the 72 and 38 blades/row builds. Two separate tests were run at each speed to provide an indication of the likely scatter of the test points.

The mean stage characteristics for the three pitchings at 900 rev/min are presented in Figures 4 to 6, and those at 500 rev/min in Figures 7 to 9. Figures 10 and 11 summarise the results for the two speeds.

4.0 Discussion

4.1 Limitations of tests

The tests should be regarded as comparative only, the absolute values of the overall parameters being of somewhat less significance. There are three reasons for this:-

(i) In the torque estimation, it was not possible to allow exactly for the bearing friction torque. An allowance was made as described in Section 2.3, but the friction torque of ball bearings is known to vary with speed and axial loading, although not in an easily determined manner. Any errors would, however, be practically the same for corresponding tests.

(ii) Tests on one or two stages are always open to doubts because small irregularities in the flow pattern can easily give rise to pressure variations which are significantly large proportions of the necessarily small overall pressure rise.

(iii) Even in the absence of the previous two considerations, it is well known that one or two stages when operating in isolation may give a significantly different performance from that when they are part of a series of stages as in a multi-stage compressor.

Allied to (iii) is a possibility which could invalidate even the comparative significance of the tests. It is conceivable that velocity profile deterioration though a number of stages could result in much greater deterioration of performance in the later stages in (say) the close pitched than in the wider pitched blading. The effect would not be observed in tests on a small number of stages where the velocity has not sufficiently deteriorated.

These limitations being borne in mind, the authors feel that the conclusions of the present tests are significant and encouraging.

4.2 Mean stage performance

The scatter of the test points is seen to be reasonably small in Figures 4 to 9. The surge flow coefficients shown are the mean values for the two tests, the difference between the two being in general under 1 per cent. The mean curves indicated in Figures 10 and 11 can therefore be taken as giving a fair representation of the performance at 900 and 500 rev/min respectively. Figure 12 shows the theoretical low speed characteristics at the three pitchings. These are derived from the data of References 2, 5 and 6. Figure 13 presents three leading parameters as functions of mean diameter pitch/chord ratio for the 900 rev/min, 500 rev/min, and theoretical performances. These parameters are:-

(a) The peak efficiency.

(b) The flow coefficient at the peak pressure rise coefficient.

(c) The temperature rise coefficient at a flow coefficient of 1.15 times the value given by (b).

The peak pressure rise coefficient represents the limit of stable operation. For the 72 and 54 blade pitchings, it occurs at the surge point; for the 38 blade pitching, surge occurs at a lower flow coefficient, but it is felt that operation between the peak pressure rise and surge would be ill-advised in a practical case; there was in fact evidence of minor instability during the tests. For the theoretica performance, it is common practice to take the surge as occurring at the peak pressure rise coefficient in the absence of contrary information The temperature rise coefficient at a flow coefficient 1.15 times th value at the peak pressure rise can be regarded as an acceptable working value.

Comparison of Figures 10, 12 and 13 shows a good agreement of trend though not of detail between the test and theoretical performance at 900 rev/min. In particular, neither the test nor the theoretical results show a large fall-off in peak efficiency at the lowest pitching.

The theoretical characteristics were calculated for a work done factor of 1.0. Comparison with the test results apparently shows the test value to be even higher. This phenomenon has been noted before in this compressor, and is discussed in Reference 4. It has no importance in the present tests.

Comparison of Figures 11, 12 and 13 shows the same general trends at 500 rev/min, although the peak efficiency is lower and the surge flow is lowest at the intermediate pitching. The former effect is probably due partly to the low Reynolds number and partly to the deficiencies of the torque estimation (Section 4.1). The low surge flow at the intermediate pitching may be due at least in part to some flow phenomenon connected with the low Reynolds number.

The tests at 250 rev/min, carried out at the two extreme pitchings, showed a very large scatter of the experimental points, presumably due to the decreased relative accuracy of the measurements and also to the difficulty of holding the speed constant. This made firm deductions difficult, but there was no indication that the efficiency at the low pitching was significantly inferior. The results of these tests are not included in the figures.

4.3 Comparison with Reference 3

Reference 3 describes tests on three stages of constant section untwisted blading in a water compressor. Rotor and stator blades were identical in geometry and were set to give 50 per cent reaction. The

(a) The significant blade Reynolds numbers is that based on blade pitch rather than on blade chord. The values for the tests at 0.5 and 1.0 pitch/chord ratios were then 0.33×10^5 and 0.67×10^5 respectively, the value based on blade chord being 0.67×10^5 in both cases. It is suggested in the Reference that a critical value might lie between the two former values. The corresponding values for the present tests are given in the following table; they are based on the mean diameter blade pitch and the estimated velocity relative to the rotor blade at mean diameter at a flow coefficient of 0.8.

Reynolds number $\times 10^{-5}$			
Number of blades/row	72	54	38
Speed = 900 rev/min	1.283	1.712	2.435
Speed = 500 rev/min	0.577	0.769	1.092
Speed = 250 rev/min	0.288	(0.385)	0.547

The corresponding figures for the stator blades are less than the above approximately by a factor of 0.8, the relative air inlet velocity being smaller. The present tests thus lend no support for the existence of a critical Reynolds number based on blade pitch in the region $0.3 - 0.7 \times 10^5$. No measurements of the turbulence levels were made in either the present tests or those of Reference 3, but it is not thought that any difference would be large enough to explain the apparent lack of agreement.

(b) Reference 3 suggests that reduction of pitch/chord ratio has an additional effect not allowed for in simple theoretical concepts of blade performance. It could be connected with the poorer geometry of the blade passages (regarded as diffusers); this applies increasingly as the stagger is increased; the significant difference between the present and the Reference 3 tests is the higher mean stagger of the latter.

Another factor, briefly mentioned in Section 4.1, concerns the deterioration of velocity profile through a number of stages. This deterioration is known to cause a tendency to local stalling of the inside diameter of a series of constant section untwisted stages[7] and it is possible that in the Reference 3 tests the closer pitching precipitated local stalling of the last stage, with consequent loss of efficiency; this is of course conjecture, and would need to be substantiated by a detailed investigation. It is interesting to note that in some unpublished tests at N.G.T.E. on a three-stage air compressor, consisting of variable stagger constant section untwisted blades, abnormally low efficiencies were recorded at the higher staggers. No firm conclusions could be drawn, however, since there were other factors present which obscured the significance of the results.

5.0 Conclusions

Two stages of low stagger free vortex blading have been tested at mean diameter pitch/chord ratios of 0.49, 0.66 and 0.94, in a low speed compressor of 0.70 diameter ratio. The object of the tests was to provide information for use in the design of compressors of high stage work, such as are required for instance in the compression of light gases.

The performance was found to be broadly in line with that predicted by simple theory. In particular, there was no indication of a sharp drop in efficiency at the lowest pitch/chord ratio, as had been suggested by some earlier work. The reasons for the apparent disagreement with the earlier work are discussed, and attention is drawn to the limitations of the present tests.

NOTATION

J	mechanical equivalent of heat
K_p	specific heat at constant pressure
U	blade peripheral velocity at mean diameter
V_a	mean air axial velocity
ΔP	mean stage total pressure rise
ΔT	mean stage total temperature rise
c	blade chord
g	acceleration due to gravity
s	blade pitching
ρ	air density
η	isentropic efficiency
β	blade angle
θ	blade camber angle
γ	blade stagger angle
Ω	work done factor

Suffices

$_0$	inlet guide blade outlet
$_1$	rotor blade inlet
$_2$	rotor blade outlet
$_3$	stator blade inlet
$_4$	stator blade outlet

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APPENDIX I

Blade design details

The blades were of free vortex design (design flow coefficient = 0.800) with details as below. The blade chord was 3.0 in. throughout and the thickness/chord ratio was 0.10.

Inlet guide blades

dia. (in.)	29.0	34.0	39.0
θ (degrees)	4.5	4.0	3.2
γ	3.1	2.8	2.2
β_3	0	0	0
β_0	4.5	4.0	3.2

Rotor blades

dia.	29.0	34.0	39.0
θ	42.0	30.0	21.3
γ	15.5	29.0	39.6
β_1	44.2	50.0	54.7
β_2	2.2	20.0	33.4

Stator blades

dia.	29.0	34.0	39.0
θ	44.1	41.0	39.0
γ	11.7	9.5	6.9
β_3	41.9	37.6	33.7
β_4	-2.2	-3.4	-5.3

The compressor outside diameter was 40 in. and the inside diameter was 28 in. All blade sections were C4 on parabolic arc (P40) camber lines.

APPENDIX II

Note on the calculation of the parameters

Density ρ

The density was taken as the mean of the calculated inlet and outlet densities.

Mean flow coefficient V_a/U

The axial velocity V_a was based on the density, mass flow and annulus area. The blade velocity U was calculated at the mean diameter.

Mean stage pressure rise coefficient, $\Delta P/\frac{1}{2}\rho U^2$

The inlet total pressure was taken as the mean value based on the mass flow and static pressure after the inlet guide blades; the outlet pressure was the arithmetic mean value as read on four nine-point pitot combs, situated two chords downstream of the last stator blade row, this being not significantly different from a true area mean value. The mean stage pressure rise ΔP was taken as half the difference between the inlet and outlet pressures.

Mean stage temperature rise coefficient, $\Delta T/\frac{1}{2}U^2$

The value plotted is $2gJK_p\delta T/U^2$, where δT was half the overall total temperature rise of the air in °C, the other quantities being in consistent units. δT was calculated from the input torque, speed and mass flow. A correction for bearing friction was made to the input torque as described in Section 2.3.

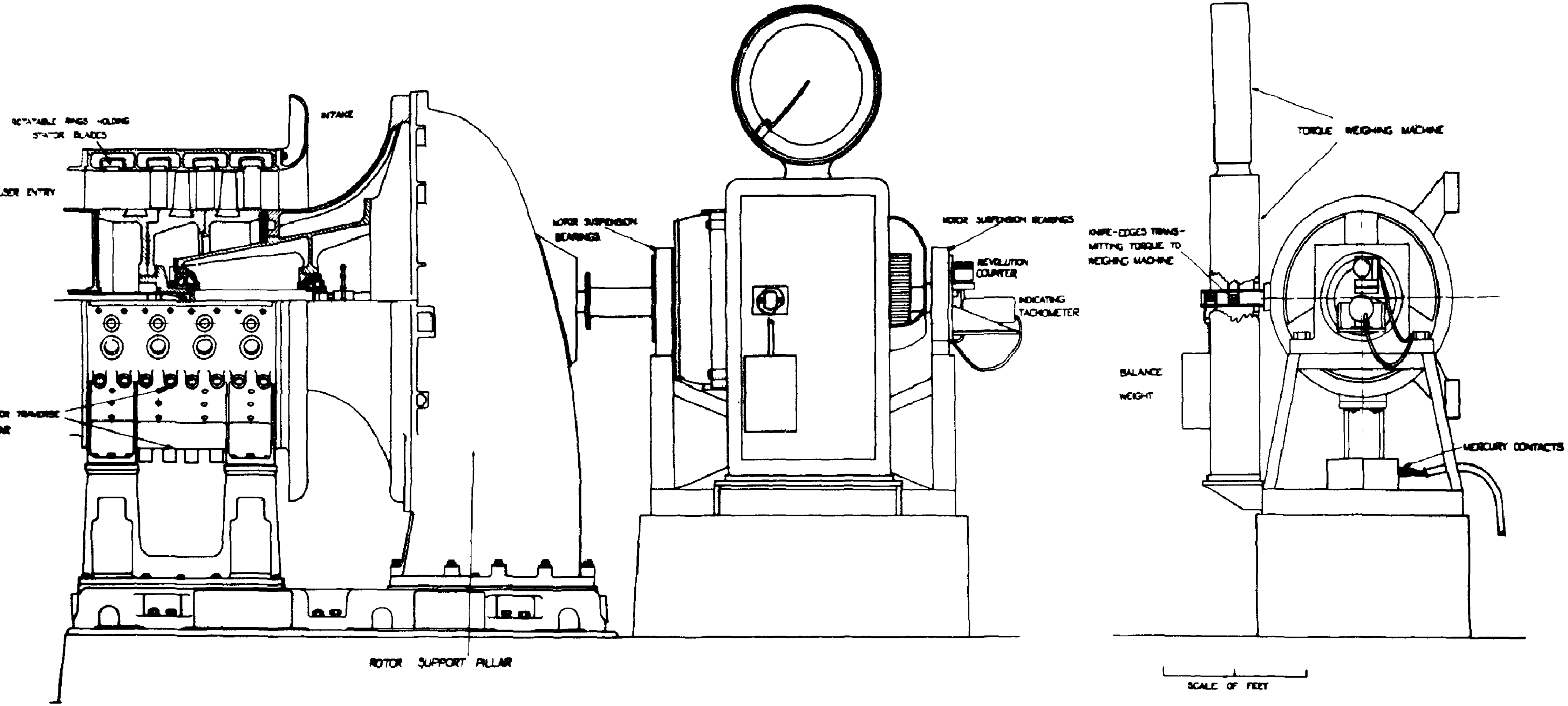
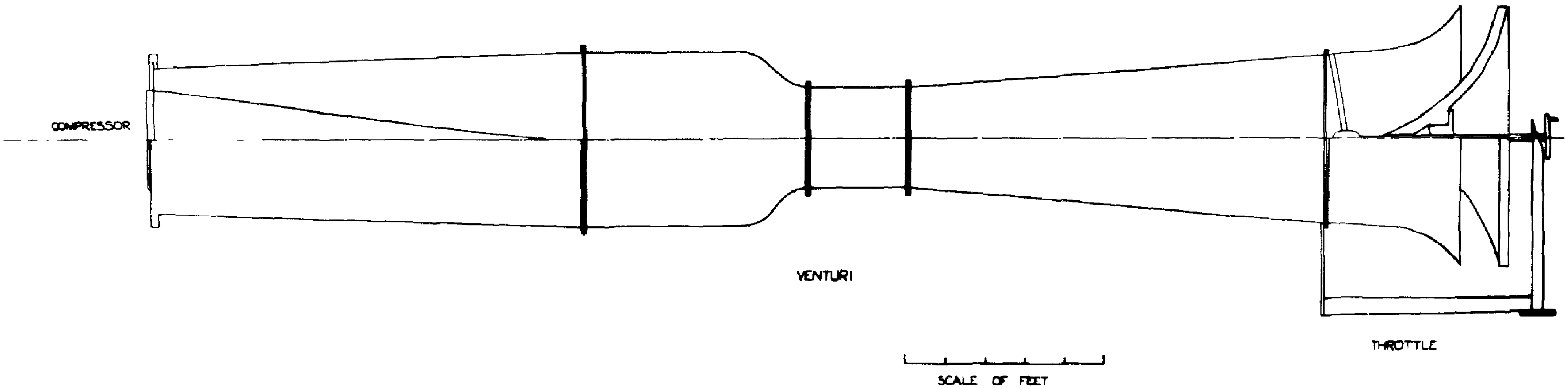


FIG. 1

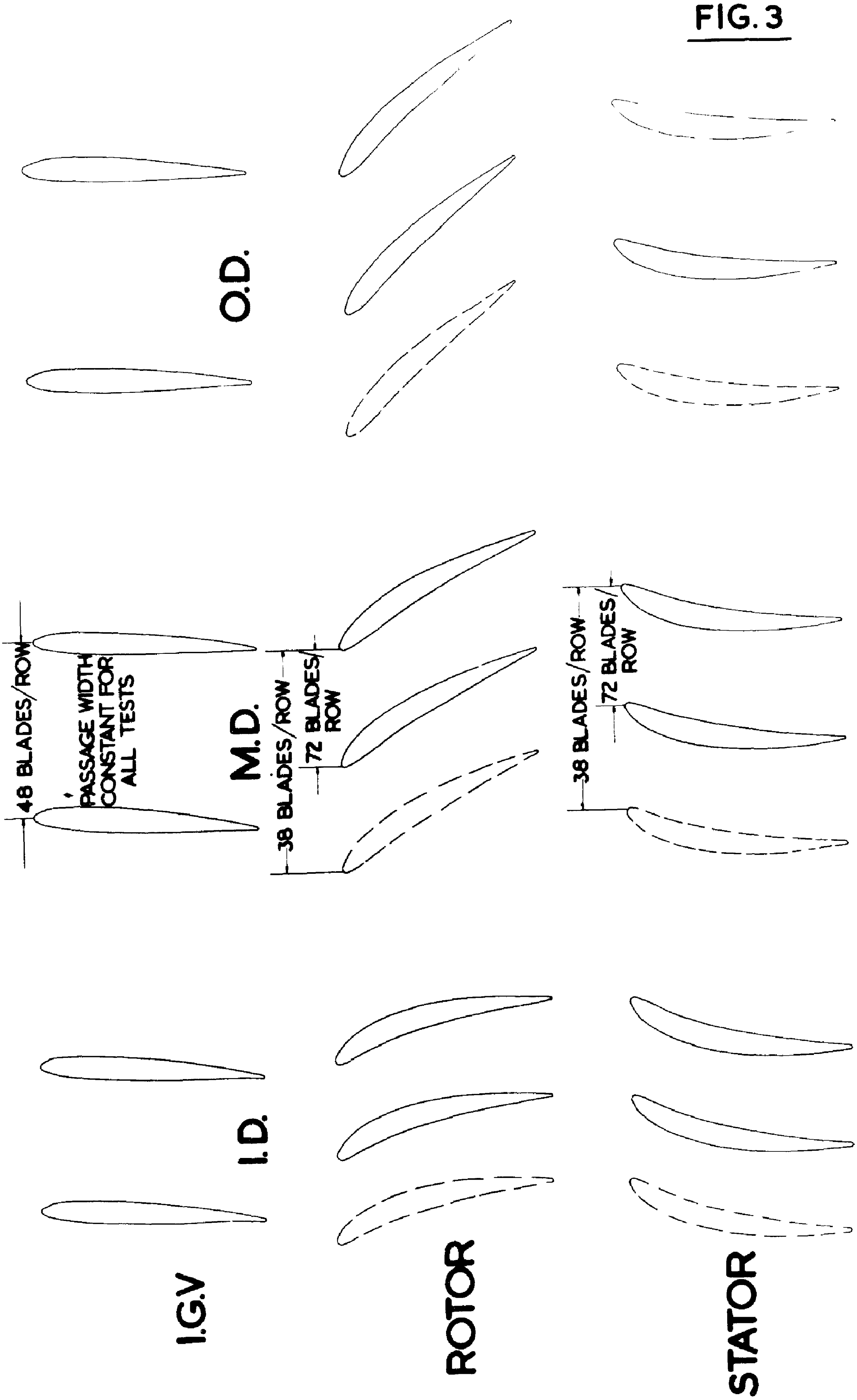
LAYOUT OF COMPRESSOR



OUTLET DUCTING IN ARRANGEMENT

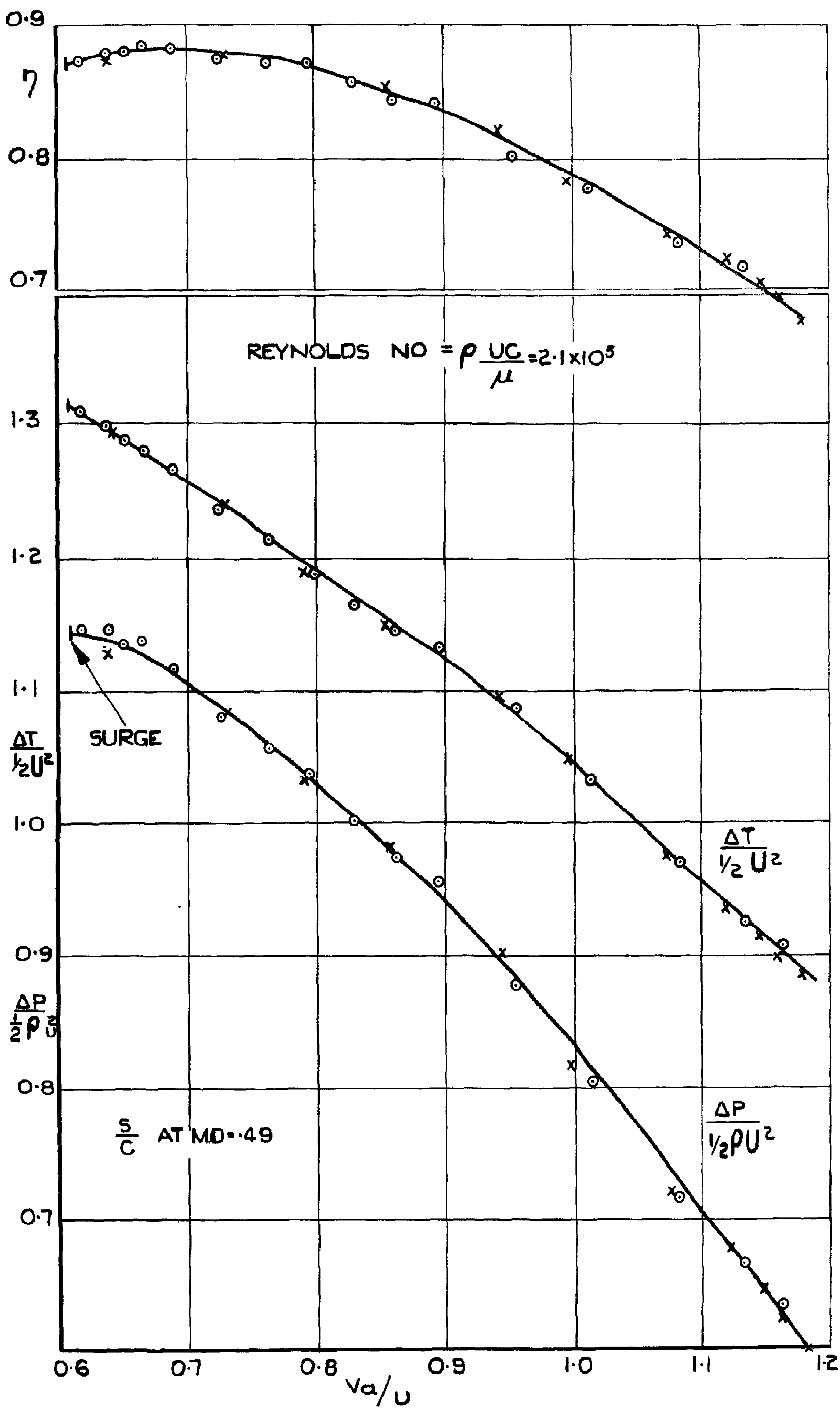
FIG. 2.

FIG. 3



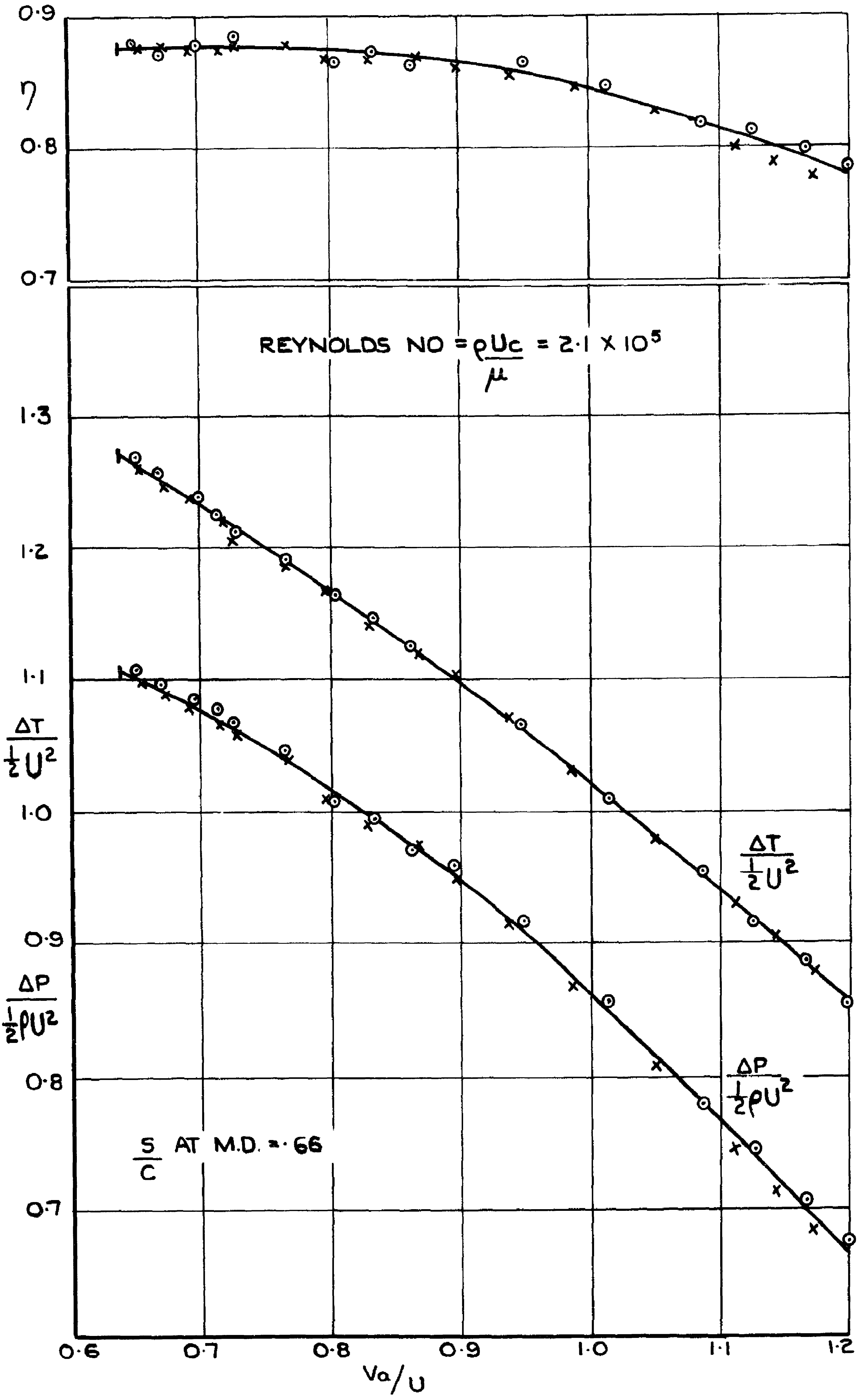
BLADE PASSAGES FOR 72 AND 38 BLADES/ROW AT THREE BLADE HEIGHTS.

FIG. 4



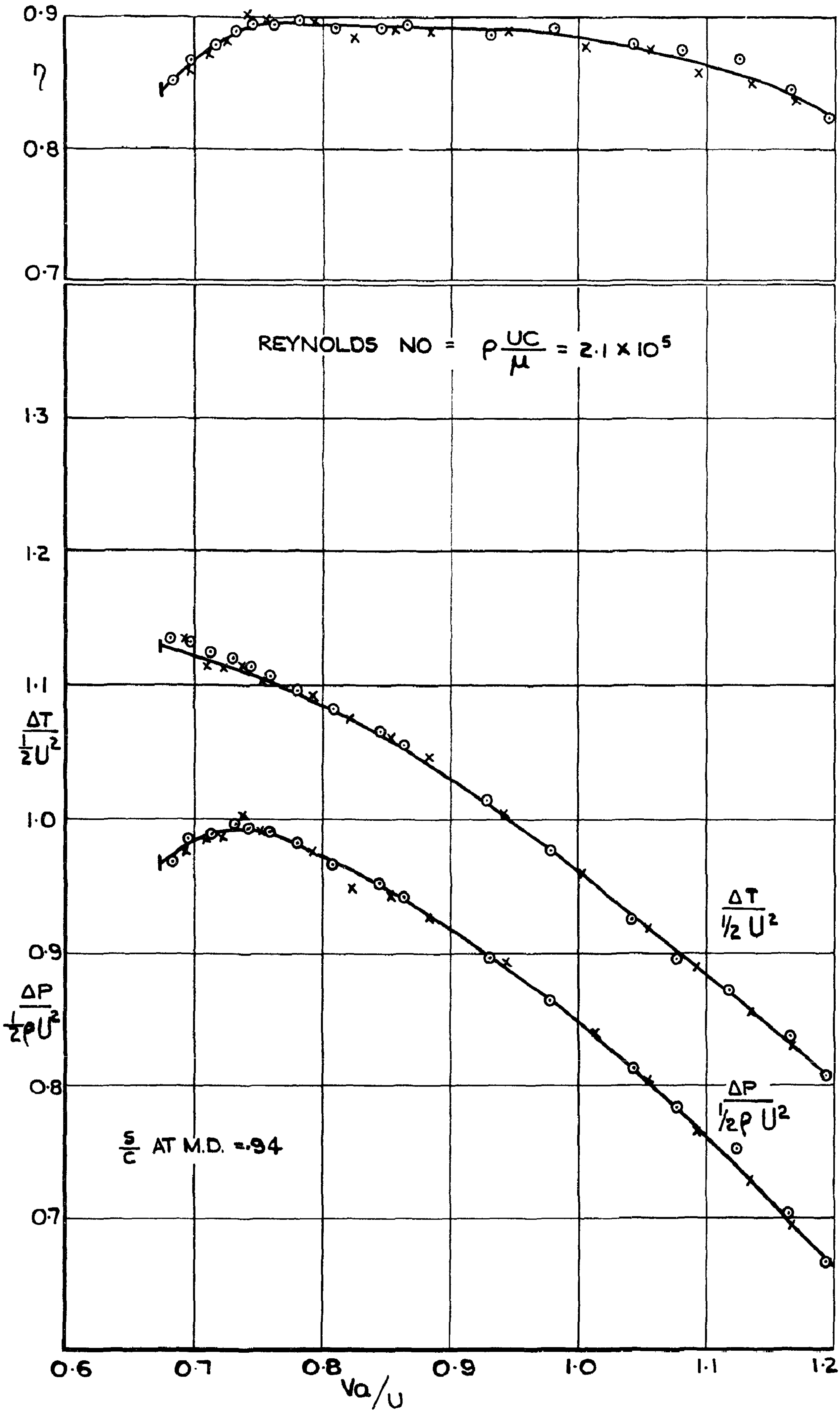
MEAN STAGE CHARACTERISTICS
900 REV./ MIN. 72 BLADES / ROW.

FIG. 5



MEAN STAGE CHARACTERISTICS
900 REV/ MIN. 54 BLADES/ROW.

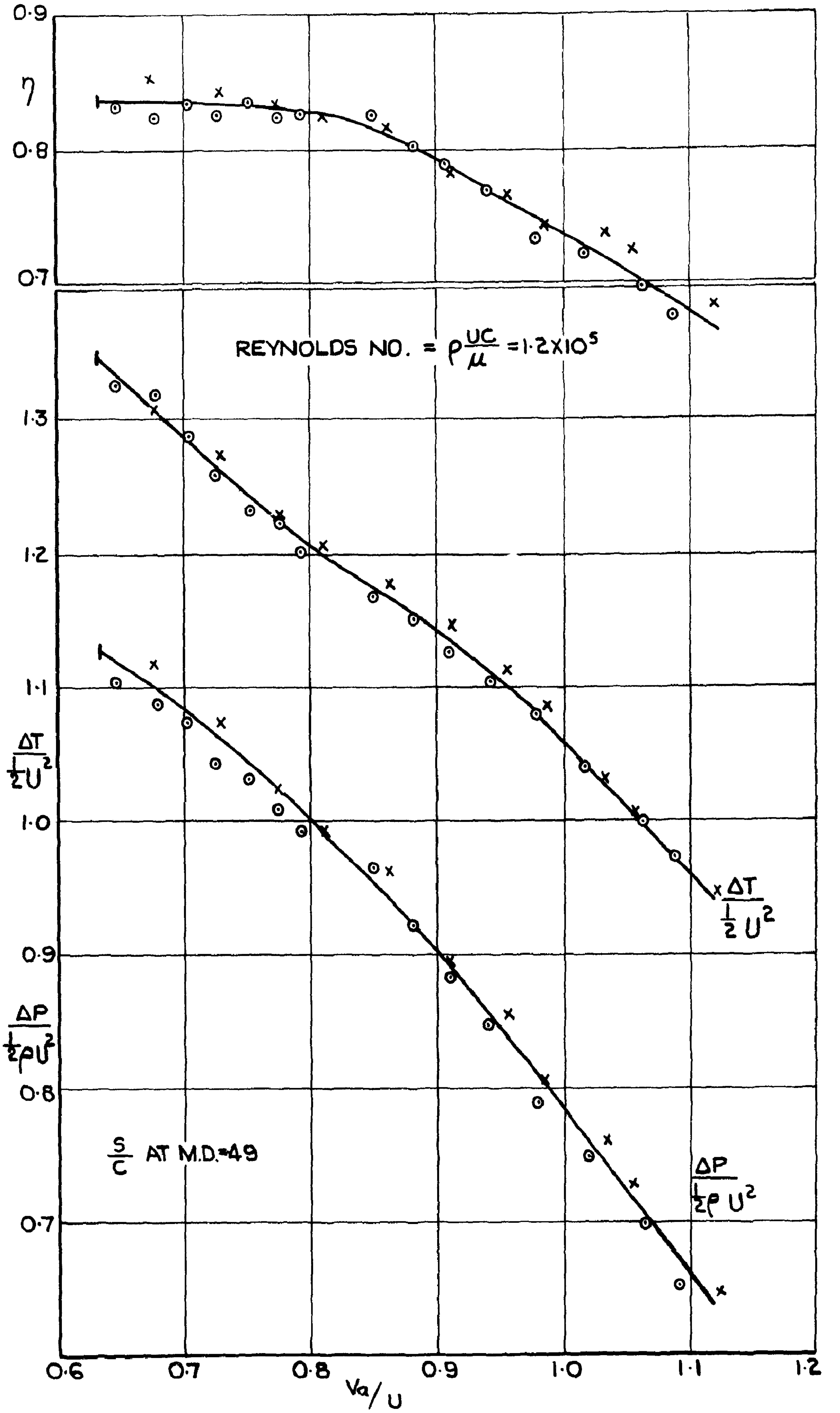
FIG.6



MEAN STAGE CHARACTERISTICS

900 REV/MIN 38 BLADES/ROW

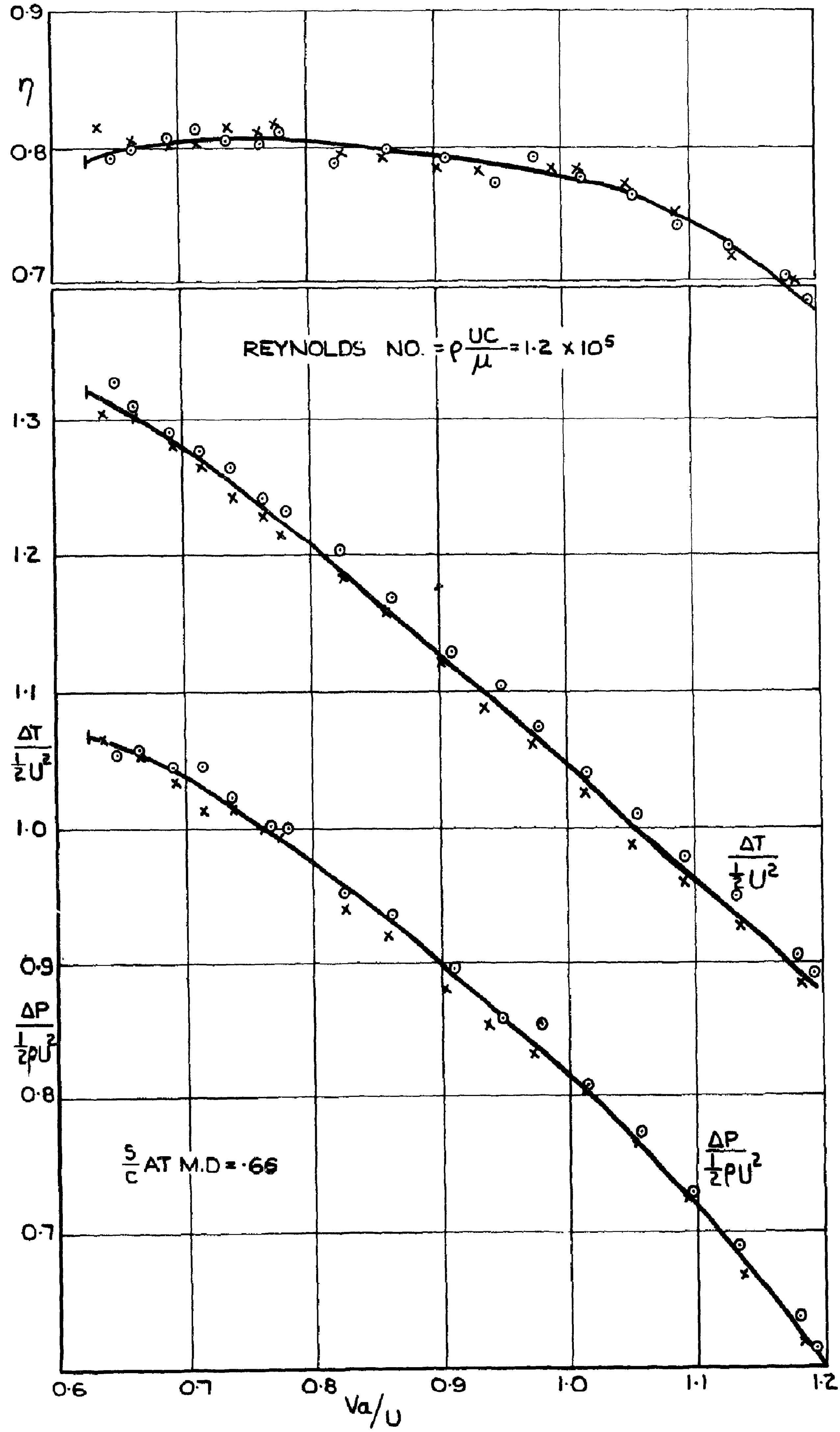
FIG.7



MEAN STAGE CHARACTERISTICS

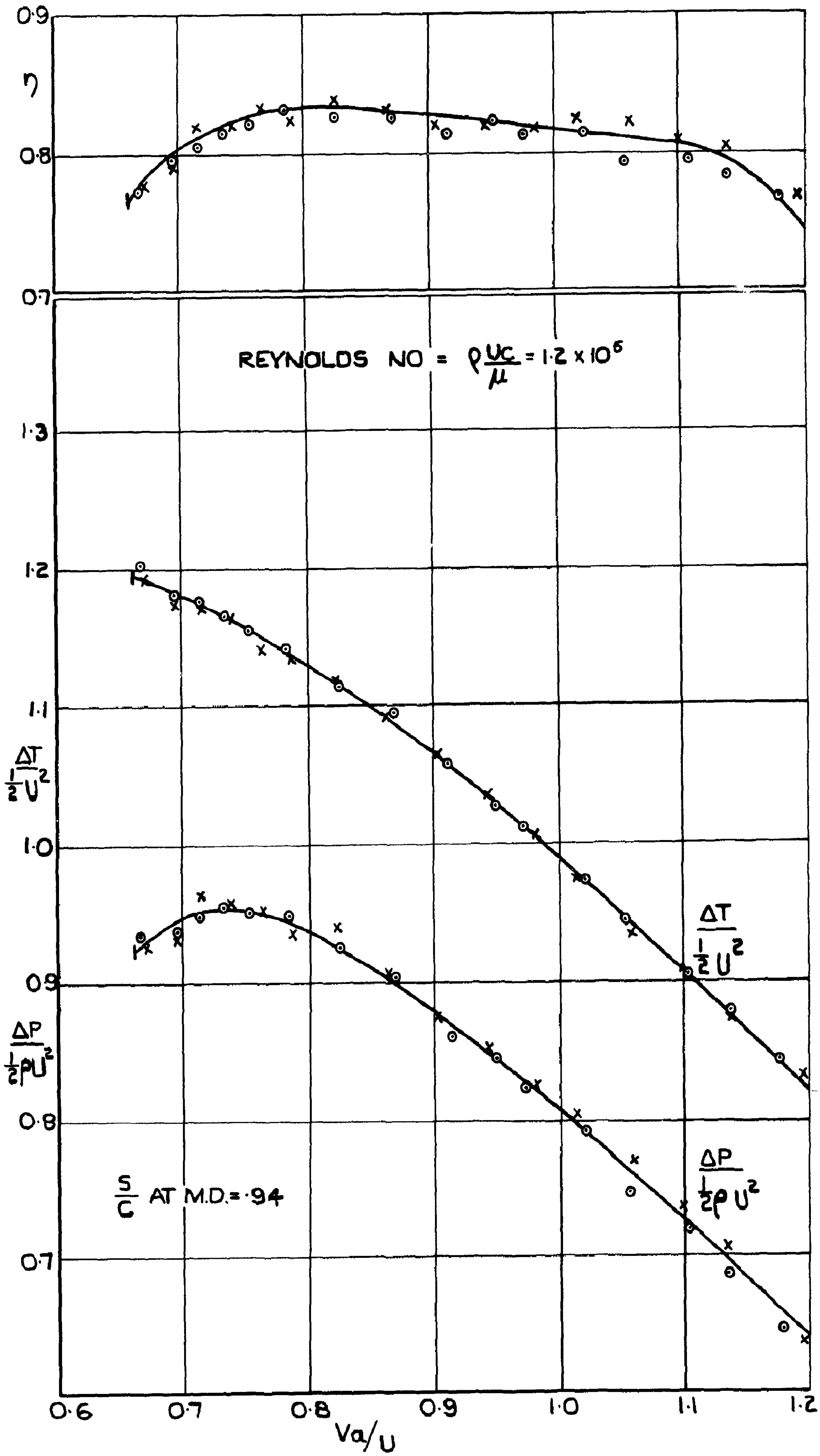
500 REV/MIN 72 BLADES/ROW

FIG.8



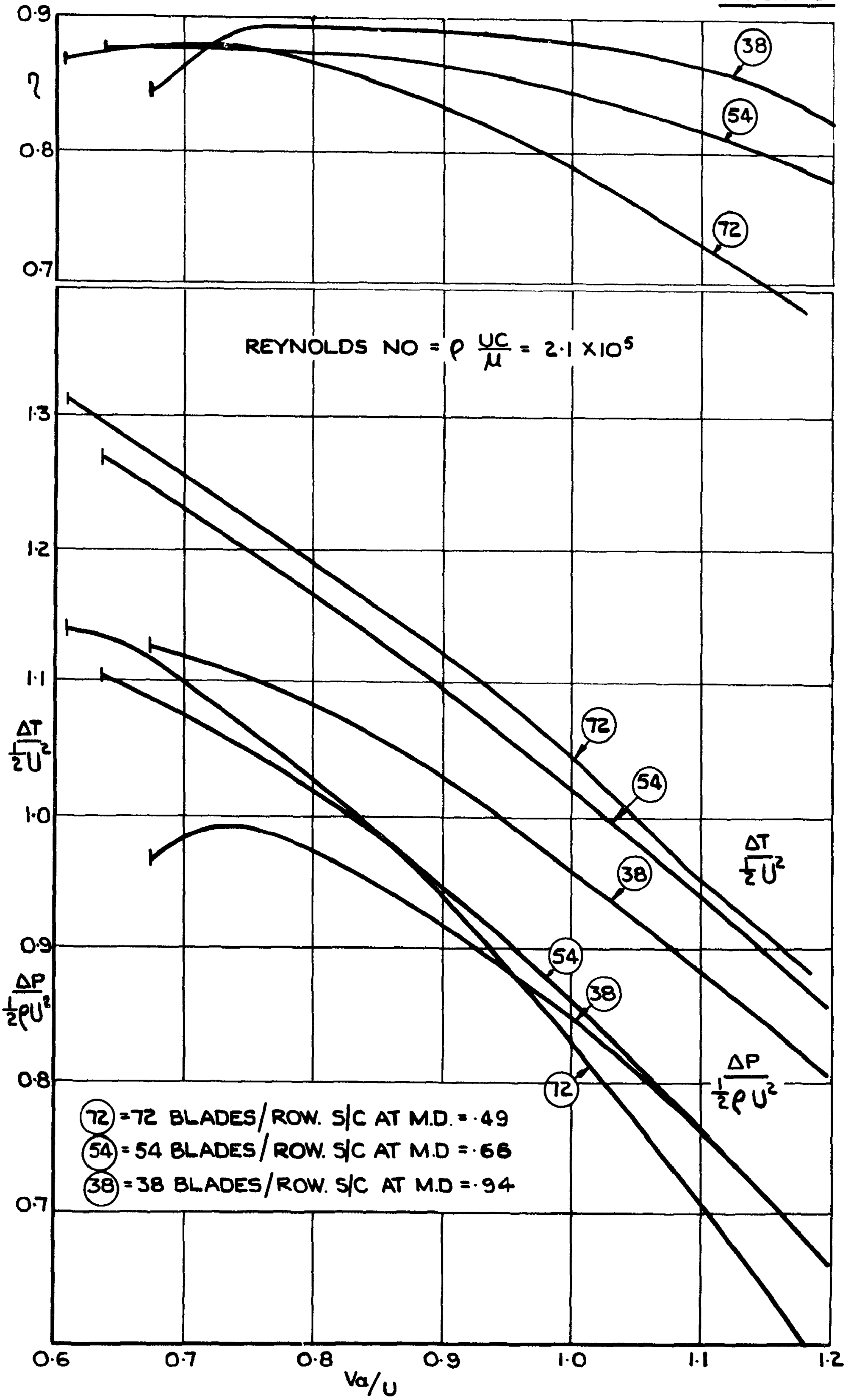
MEAN STAGE CHARACTERISTICS

FIG. 9



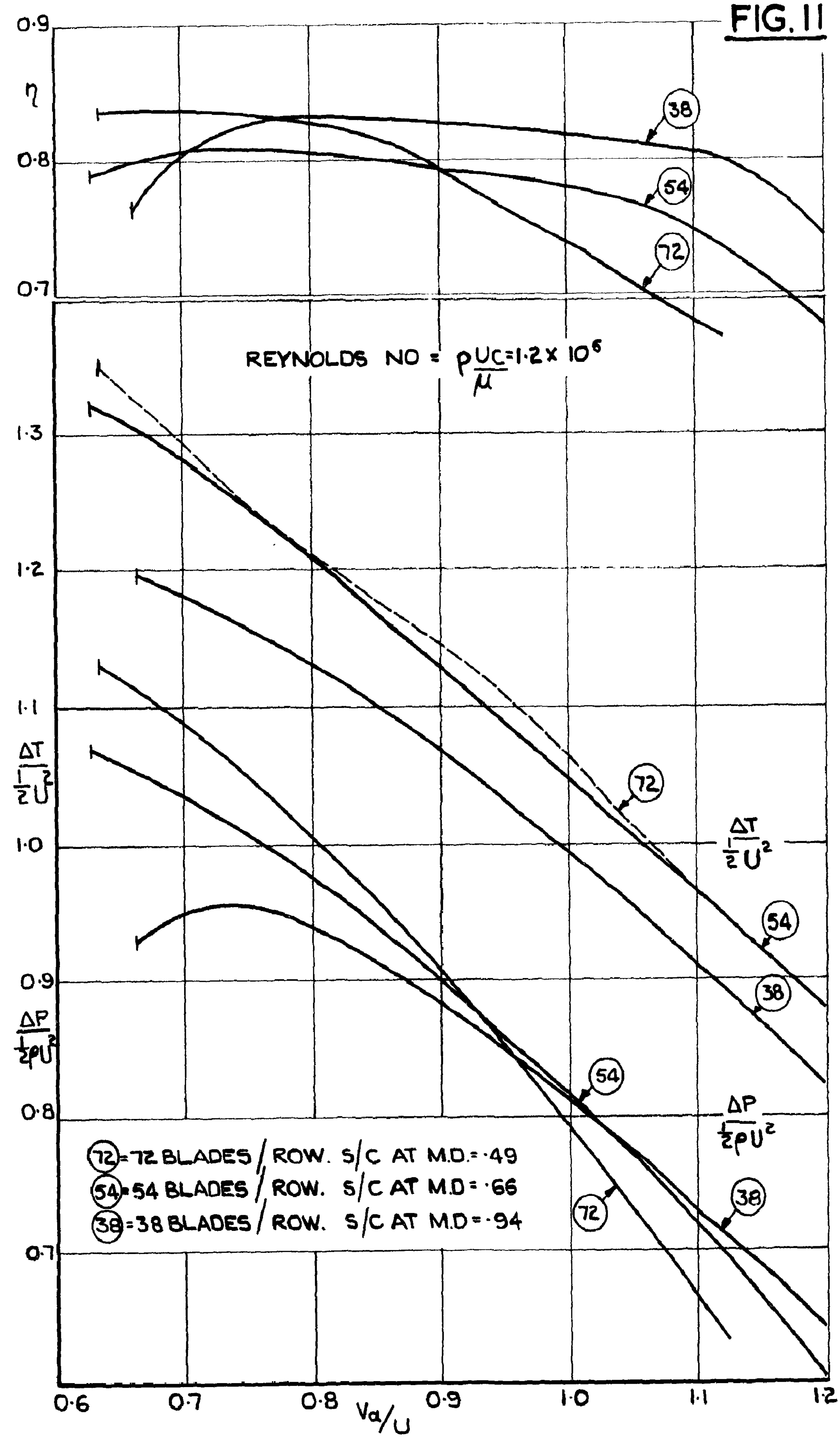
η
REYNOLDS NO = $\rho \frac{Uc}{\mu} = 1.2 \times 10^5$
$\frac{\Delta T}{\frac{1}{2}U^2}$
$\frac{\Delta P}{\frac{1}{2}\rho U^2}$
$\frac{S}{C}$ AT M.D. = ·94
V_a/U
0·9
0·8
0·7
1·3
1·2
1·1
1·0
0·9
0·8
0·7
0·6
0·7
0·8
0·9
1·0
1·1
1·2

FIG. 10



COMPARISON OF MEAN STAGE CHARACTERISTICS AT THREE BLADE PITCHINGS. 900 REV/MIN.

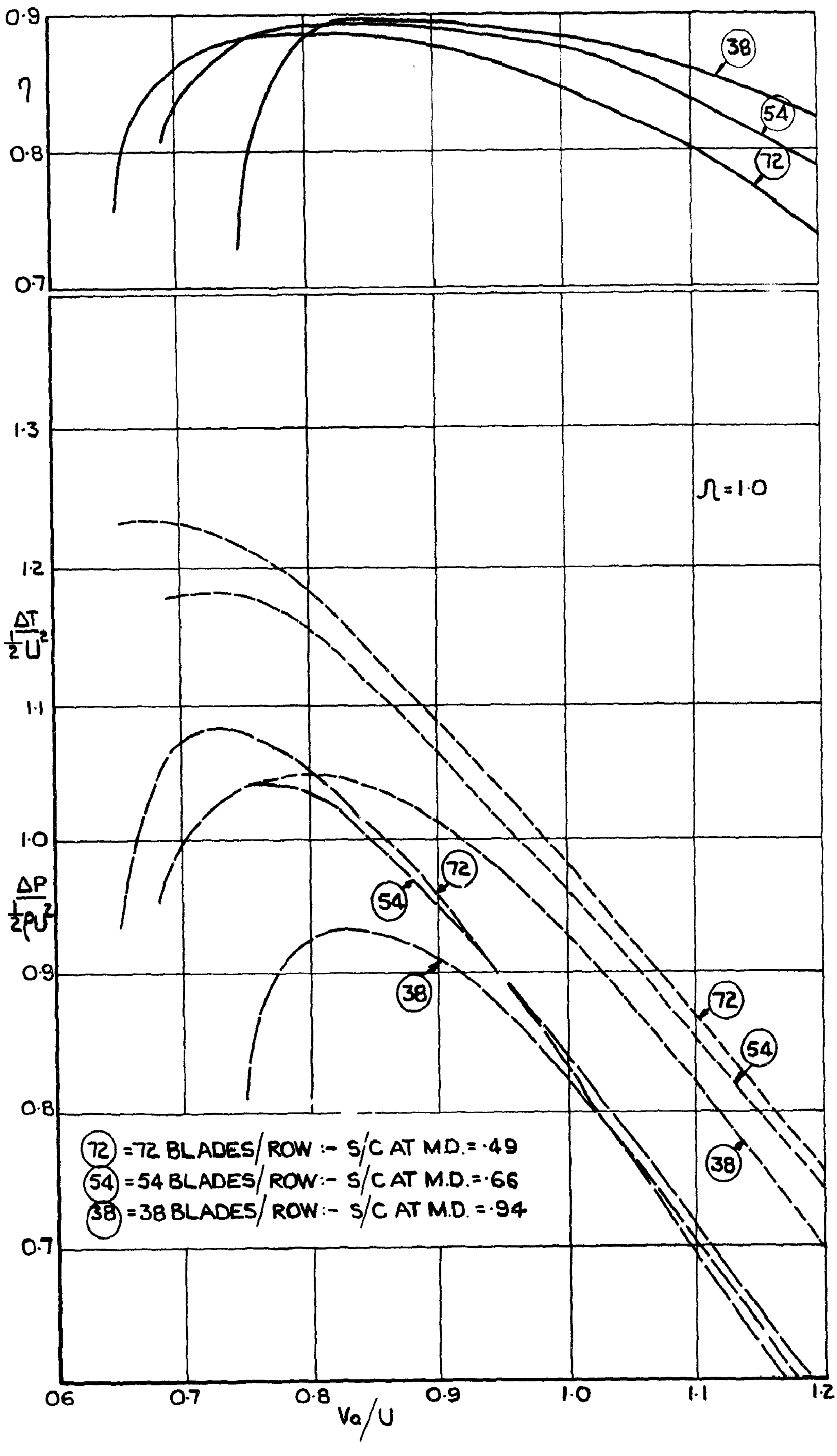
FIG. 11



COMPARISON OF MEAN STAGE

CHARACTERISTICS AT THREE

FIG.12



THEORETICAL CHARACTERISTICS AT ALL BLADE PITCHINGS

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