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Some Surge Investigations on a Low Speed Compressor

By

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An experimental investigation of the effects of stage matching on compressor surge

- by -

R. C. Turner and R. A. Burrows

SUMMARY

This paper summarises some exploratory tests on a low speed low pressure ratio multi-stage axial compressor, in which mismatching was simulated by appropriate staggering of the stages, the last stage alone being unchanged. Overall and stage characteristics and surge flow coefficients were determined for two degrees of mismatching and for the fully matched condition. In addition, velocity and yaw traverses were made at the first and last stages, and flow fluctuation measurements were also made at selected positions.

In the two mismatched builds, the surge flow coefficients were found to be identical, and were higher than the value for the matched build. This corresponds with the phenomenon sometimes noted on high pressure ratio compressors, in which surge at speeds near the matched condition may tend to occur at a constant value of the flow coefficient at the last stage, and at lower speeds at a constant higher value.

The tests appear to suggest that the velocity profile at the last stage may be one significant factor in the determination of the surge point of a compressor.

It is also suggested that flow visualisation techniques could provide useful detailed qualitative information on the internal flow conditions of a compressor before and after the surge point.

Previously issued as N.G.T.E. Memorandum No.M.337 - A.R.C.21,952.

CONTENTS

APPENDICES

ILLUSTRATIONS

1.0 Introduction

It has been observed in some compressors of conventional design that, just before surge occurs, the last stage is operating at one or other of two positions on its pressure rise coefficient versus flow coefficient characteristic. The position for speeds above those corresponding to the "kink" in the surge line, i.e. when the compressor is at or near its matched condition, is at a comparatively low flow coefficient, at or near the peak pressure rise; while for lower speeds, it is at a higher value, i.e. lower down the stage characteristic.

In some compressors the effect is more marked than in others, but it has appeared sufficiently established to justify further investigation. The present tests were therefore carried out on the N.G.T.E. "106" low speed compressor. In this compressor, mismatching corresponding to that at low speeds in a high pressure ratio compressor can be to some extent simulated by suitable variation of the stagger of the stages. The same flow coefficient is imposed on all stages since the annulus is of constant area and the density change through the compressor is negligible. With the stagger varying from stage to stage, each stage thus operates at a different point on its characteristic relative to some significant point such as the theoretical stall point, the maximum efficiency point, or the peak pressure rise point. This broadly corresponds to an off-design condition in a conventional high pressure ratio compressor (i.e. a compressor with a pressure ratio in excess say of 2.0), where although the stages may be similar aerodynamically, a different flow coefficient is imposed on each.

In the present tests, the last stage was maintained at one stagger setting whatever the degree of mismatching. The degree of mismatching was varied by means of stagger changes of the previous stages. Flow conditions at or near surge were investigated for three degrees of mismatching.

2.0 Apparatus and measurements

2.1 The compressor

The "106" compressor is described in Reference 1. It is a low speed multi-stage machine 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 running speed of 1500 rev/min the mean diameter blade speed is 114.5 ft/sec and the blade Reynolds number based on this speed is 0.65×10^5. The blade chord is approximately 1.1 in. The six-stage assembly of the compressor as used in the present tests is shown in Figure 1.

In the present investigation, the blades were of medium stagger free vortex design, with 50 per cent reaction at the mean diameter. Details of the blades as designed are given in Appendix I. Three builds of the compressor were tested, the variation of stagger from the design value being given in degrees in the following table, the positive sign indicating a numerical increase:-

Blade row	Build 1	Build 2	Build 3
G	- 4	+ 6	+16
1R	- 4	+ 6	+16
1S	0	+ 8	+16
2R	0	+ 8	+16
2S	+ 4	+10	+16
3R	+ 4	+10	+16
3S	+ 8	+12	+16
4R	+ 8	+12	+16
4S	+12	+14	+16
5R	+12	+14	+16
5S	+16	+16	+16
6R	+16	+16	+16
6S	+16	+16	+16

The last stage rotor and stator and the fifth-stage stator blades were thus maintained at the same setting in all three builds. Build 1 corresponded to the mismatching in a high pressure ratio compressor at a sub-design speed, and Build 3 to the completely matched condition, Build 2 being intermediate between the two. In all builds, the stagger settings were generally considerably higher than the design values, and hence it was to be expected that the general level of efficiency would be low.

2.2 Measurements

Entry and delivery total pressures and interstage static pressures, driving torque, mass flow and speed were measured as steady values.

Pressure fluctuations were indicated by means of capacity type pick-ups connected to pitot tubes inserted in the compressor where required. The oscilloscope traces, obtained via a frequency modulated circuit, were recorded on moving film. Calibration was made under steady conditions only. A check calculation was made, however, to ensure that Helmholtz type resonance did not occur in the pitot tube and connection to the pick-up at the frequencies encountered in the test. The pitot tubes were situated about one chord downstream of the adjacent stator blade row, and care was taken to avoid blade wakes. They could be traversed across the annulus.

3.0 Test technique and presentation of results

For each build, the overall and stage characteristics were determined up to the surge point. They are presented in Figures 2 and 3, 8 and 9, and 12 and 13 for Builds 1, 2 and 3 respectively. Figure 14 shows the characteristics of Stage 6 for all three builds. Details of the method of calculation of the parameters are given in Appendix II. The "minor surge" noted on Figure 2, was manifested as a rapid audible pulsation of approximately 25 cycles/sec, similar to that previously referred to as the "rapid pulse surge" in References 2 and 3, which describe tests of two highly mismatched stages. The surge proper was accompanied by the sudden onset of slow audible pulsations of about 10 cycles/sec, with sharp discontinuities in the performance characteristics.

Axial velocity and yaw traverses after Stages 1 and 6 for Build 1 are presented in Figures 4 and 5, Figure 4 referring to conditions just prior to the minor surge point and Figure 5 to conditions just prior to the surge point. Figure 10 shows axial velocity and yaw traverses after Stages 1 and 6 for Build 2 just prior to the surge point; there was no minor surge in either of Builds 2 or 3. Figure 15 shows axial velocity and yaw traverses after Stage 6, just prior to the surge point of Build 3, no traverse having been made after Stage 1.

Figures 6 and 7 summarise the flow fluctuation measurements in Build 1. Figure 6 refers to conditions before the surge point, the compressor being then in the condition of minor surge. Figure 7 refers to conditions with the compressor just surged. Figure 11 summarises the flow fluctuation measurements before and after the surge of Build 2. Figure 16 shows the corresponding curves for Build 3, but is less complete, as measurements were made after the first, fifth and sixth stages only. It should be pointed out that the amplitudes of the fluctuations presented in Figures 6, 7, 11 and 16 are peak to peak values scaled off the oscilloscope records. As even the ordered fluctuations were in general very irregular both with regard to duration and amplitude the assessment of the amplitudes depended very much on the judgement of the observer. These figures should, therefore, be taken as indicative of trends only.

Other measurements not mentioned above were also made as the occasion demanded; these are referred to as required in the discussion.

4.0 Discussion

4.1 Significance of tests

It is of course impossible to reproduce full scale compressor mismatching exactly in a constant annulus constant density machine such as the "106". The "106" however provides much better conditions for detailed flow investigations, and appeared to be the obvious choice for initial experiments. It is of course not even certain that surge in the "106" is the same phenomenon as that in a full scale compressor, but it seems reasonable to make this provisional assumption.

Mismatching could have been accomplished with a series of similar stages by tapering the annulus, but this would have involved lengthy and expensive modifications to the blades, and rotor drum. The main assumption underlying the "mismatching by restaggering" which was employed in the present tests is that the initiation of surge primarily depends on the position of the working point of each stage relative to its stall or peak pressure rise point. In a full scale compressor, the stages often have similar characteristics; in the mismatched version of the "106", the stages have different characteristics. A full scale compressor has a tapered annulus; the "106" does not. These differences may be important, but the authors feel that it is reasonable to assume provisionally that they will not change the primary mechanism of surge initiation.

Another criticism which might be made of the tests is that restaggering of blades originally designed for free vortex flow will result in stages with a typical radial variation of angle. Here again, the authors feel that the implications with regard to surge initiation will probably be secondary only. Many tests have been carried out on this and other compressors using blades with various kinds of twist (e.g. free vortex, constant section untwisted). The characteristics differ in detail, but not in general shape, and there is no definite evidence known to the authors of a fundamental change in surge behaviour which could be ascribed to change of blade twist.

4.2 General

The results show several interesting features:-

(a) The surge flow coefficient was practically the same for Build 1 as for Build 2, but was appreciably lower for Build 3. In Figure 17, the mean diameter rotor incidence angles through the compressor are shown for each build surge, relative to the blade stalling incidence angles. The incidence angles were calculated assuming a radially uniform axial velocity distribution and the "nominal" air outlet angles from the previous blade rows (see Reference 4). The stalling incidences were calculated from Reference 5. Figure 17 illustrates the relative degree of mismatching in the three builds and the fact that in Builds 1 and 2 at surge the first stages were more stalled than the final stages. As the last stage was identical in all three builds, the incidence on it is an indication of the surge flow coefficient. The existence of two distinct surge flow coefficients at the last stage, depending on the degree of mismatching, is in agreement with the behaviour sometimes observed in full scale compressors and already noted in Section 1.0. This result suggests (but does not prove) that the surge behaviour of this compressor is similar to that of a high pressure ratio compressor.

(b) Surge in all three builds was caused by a single rotating stall cell or region of disturbed flow extending right through the compressor and occupying the full annulus height; the speed of rotation was approximately 0.35 times the rotor speed. It was accompanied by major discontinuities in the compressor characteristics.

(c) The "minor" surge, present only in Build 1 (which had the greatest degree of mismatching) was caused by three rather ill-defined rotating stall cells, situated towards the outer diameter of the first stages and rotating at 0.35 times the rotor speed. These were attenuated and gradually lost their separate identities through the stages until Stage 5 was reached, after which the flow was steady; irregular fluctuations were apparent after Stage 6, however (Figure 6). After the first stage, the axial velocity as indicated by conventional pitot and static tubes was reversed towards the outer diameter (Figure 5); the presence of the stall cells may have affected the measurements to some extent. There was no discontinuity in the overall characteristics, but the efficiency reached its peak value when the minor surge commenced, with a slight change of curvature in the temperature rise coefficient curve. The stage characteristics (Figure 3) show that the first three stages were operating on the low flow side of their peak pressure rises. Stall cells are of course commonly found in the first stages of high pressure ratio compressors operating at sub-design speeds near the surge line. When they do not give rise to discontinuities in the stage characteristics they are usually described as "progressive", as distinct from "abrupt" and generally extend across part of the annulus only, as in the present case - see Reference 6.

(d) In Build 2 (which had a lesser degree of mismatching), just before surge, no stall cells were apparent, but there were appreciable flow fluctuations, the largest being at the outer diameter after the first stages. They were attenuated through the later stages. Inspection of the yaw traverses after the first stage (Figure 10) suggested moderate stalling towards the outer diameter. The first stage alone passed its peak pressure rise before surge.

(e) In Build 3 (in which the stages were fully matched), just before surge, no stall cells were apparent, but there were large random pressure pulses towards the inner diameter after the last stage stator blades.

(f) Change of stagger of the last stator blade row by $\pm 10^{\circ}$ in any of the three builds did not affect the surge flow coefficient.

(g) In Builds 1 and 2, the axial velocity and yaw traverses after the last stage were similar in general shape just prior to surge, the peak velocity being towards the inner diameter; the profiles after the first stage were, however, considerably different, with reversed mean flow in Build 1 at the outer diameter. In Build 3, the velocity profile after the last stage at surge was peaked towards the outer diameter, with a well marked region of low flow at the inner diameter.

4.3 Some related work on surge

Reference 7 gives a useful account of surging in high pressure ratio multi-stage compressors The type of surge usually met with is described as surge due to "abrupt stall", consisting of a cyclical entry into an abrupt stall followed by recovery, with corresponding changes in the net flow. By "abrupt stall" is meant a major discontinuous change in the compressor characteristics, accompanied by a rotating stall cell extending right through the compressor. This obviously corresponds to the "surge" of the present tests, in which no cyclic variation in the net flow was however noted. The existence and characteristics of the cyclic variation would be expected to depend largely on the associated ducting.

The factor which appears common to the "surge" of the full scale compressor connected to ducting and to the "surge" of the present tests, however defined, is the sudden change of the flow pattern, with the inception of the rotating cell extending through the compressor. The instrumentation in the present tests was not adaptable to detailed investigations of the flow patterns during surge, and so it is not possible to say definitely how the cell differed if at all from the kind which is usually localised in the first stages of a mismatched compressor - as in the "minor surge" of Build 1 - and which has been given extensive theoretical treatment on a two-dimensional basis, as for instance summarised in Reference 6.

There is, however, evidence that the flow during surge of a full scale multi-stage compressor embodies a significant degree of recirculation or reverse flow. Unpublished test results at N.G.T.E. have shown that the temperature of the middle stages of a compressor during surge may rise to a value which is explicable only on the assumption of recirculation, and which may result in destruction of the blades.

It is therefore possible that surge as normally understood is caused by the inception of a rotating regime of reverse or recirculating flow. The instrumentation of the present tests was not suitable for distinguishing reverse flow, and this may well have applied to the tests described in Reference 6. That there was flow reversal at surge was suggested by the behaviour of wool tufts held in front of the inlet. Reference 8 does in fact state that reverse flows are often detectable in low speed fans when surged, and possibly also in high pressure ratio compressors. Reference 8 also gives a qualitative treatment of surge on the basis of one half of the compressor blowing back through the other half.

Reference 9 suggests that instability may be initiated in a stage by reverse flow occurring at the inner diameter, consequent upon deterioration of the velocity profile, the deterioration occurring independently of blade stalling, which, however, in a practical case would accompany it. In the present tests, the velocity profile at the outlet of Build 3 was very low towards the inner diameter just prior to surge, and some such criterion might therefore apply with this build. An examination of the outlet velocity profiles at surge of a number of full scale compressors might be rewarding in this respect.

In Builds 1 and 2 however, the outlet profiles showed no evidence of deterioration, although there was an average reverse flow at the outer diameter of the first stage during the minor surge. Further theoretical treatment would obviously be facilitated by a more detailed qualitative knowledge of the flow just before and just after surge; this might well be obtained by some form of flow visualisation technique used for instance on a low speed low pressure ratio compressor assembled as in the present tests to simulate the various degrees of mismatching encountered in a full scale compressor.

5.0 Conclusions

Six stages of blading have been tested in a low speed low pressure ratio compressor in the matched condition and with two degrees of mismatching. The mismatching was obtained by the progressive re-staggering of the stages through the compressor, the last stage being at the same stagger setting in all three builds. The flow conditions have been examined in detail at flows near surge, both velocity distributions and pressure fluctuations being investigated.

The surge flow was found to be identical for the two mismatched conditions and was higher than that for the matched condition. This result agrees with experience on some full scale compressors where the surge at speeds in the mismatched condition below the surge line kink tends to occur at one flow coefficient at the last stage, while that at higher speeds (i.e. nearer the matched condition) occurs at a lower value. It also suggests that the surge behaviour of the present tests satisfactorily simulates that to be found in some high pressure ratio compressors.

Before the surge in the two mismatched builds, the initial stages showed evidence of stalling at the outer diameter, in one case with definite rotating cells and reverse mean flow, the flow improving progressively through the compressor. Just prior to surge, the velocity profile after the last stage was very similar in the two builds, the flow being peaked towards the inner diameter.

In the matched build, there was no evidence of stalling the first stage prior to surge. The outlet axial velocity was peaked towards the outer diameter, with a region of very low velocity towards the inner diameter, where large random pressure fluctuations were observed.

It is suggested that normal compressor surge could be associated with rotating regions of reverse or recirculating flow, and that one of the conditions determining its inception may be the velocity profile at the last stage. The visualisation of the flow just before and just after surge in a multi-stage compressor together with an examination of the outlet velocity profiles at surge of typical high pressure ratio compressors might therefore prove to be useful experimental techniques.

NOTATION

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

Suffices

m	at mean diameter
s	static or stall
1	at entry to rotor blade row
2	at exit from rotor blade row
3	at entry to stator blade row
4	at exit from stator blade row

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No	Author(s)	Title, etc.
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6	Members of the Compressor and Turbine Research Division, Lewis Flight Propulsion Laboratory	Aerodynamic design of axial-flow compressors. Vol. III, Chapter 11. N.A.C.A. RM E56B03b, NACA/TIL/5509, August, 1956.
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8	A. G. Smith P. J. Fletcher	Observations on the surging of various low speed fans and compressors. Unpublished M.O.A. Report A.R.C. 17,139. July, 1954.
9	W. Merchant	An approximate investigation of the off-design performance of a turbo-compressor stage. Proc.I.Mech.E., Vol. 161, W.E.P. No. 53, 1949.

APPENDIX I

Blade design details

The blades were of free vortex design (design flow coefficient = 0.667, 50 per cent reaction at mean diameter). The section was C.4 on circular arc camber lines. The blade height was 2.5 in. and the mean radius 8.75 in. The rotor has 58 blades and the stator 60. Further details are given below. The angles are in degrees and the chords in inches.

		Rotor	
r/r_m	0.874	1.0	1.14
β_1	39.2	46.5	50.6
β_2	-3.5	15.6	32.0
θ	42.7	30.9	18.6
c	1.14	1.10	1.06
s/c	0.726	0.862	1.020
t/c	0.12	0.10	0.08
α_2	5.5	23.4	37.8
		Stator	
r/r_m	0.860	1.0	1.125
β_3	52.0	46.1	42.2
β_4	18.6	15.9	14.0
θ	33.4	30.2	28.2
c	1.06	1.10	1.14
s/c	0.744	0.833	0.905
t/c	0.10	0.11	0.12
α_4	26.7	23.4	21.1
		Inlet guides	
r/r_m	0.86	1.0	1.125
β_3	0	0	0
β_4	-31.6	-28.2	-25.9
θ	31.6	28.2	25.9
c	1.06	1.10	1.14
s/c	0.744	0.833	0.905
t/c	0.10	0.11	0.12
α_4	26.7	23.4	21.1

For the tests described in this Memorandum, the blade staggers were varied in the manner indicated in the text.

APPENDIX II

Note on the calculation of the parameters

Density ρ

The density was taken as that of the atmosphere at the time of the tests.

Mean flow coefficient V_a/U

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

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

The inlet total pressure was taken as atmospheric. The outlet total pressure was taken as the arithmetic mean value derived from the readings of two five-point pitot combs two chords downstream of the last stator blades. The points of the comb were equi-spaced across the annulus, and corresponding points of the two combs were led to common manometers. The overall pressure rise ΔP was taken as the difference between the inlet and outlet total pressures divided by the number of stages.

Stage pressure rise coefficient $\Delta P_s/\frac{1}{2}\rho U^2$

The stage pressure rise was taken as the difference between the measured static pressures before and after the stage, these pressures being the mean values derived by connecting four static tappings at each position.

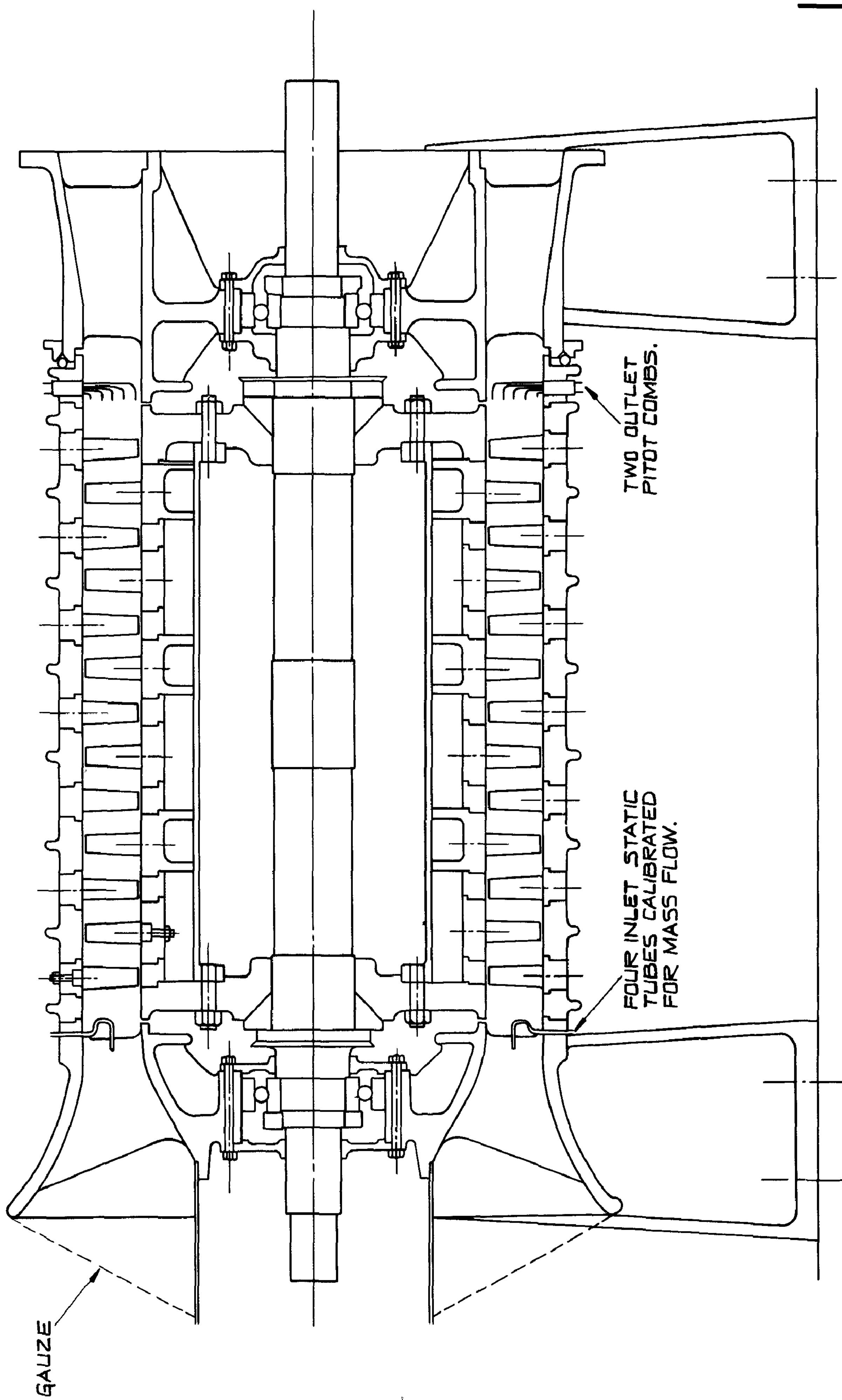
Temperature rise coefficient $\Delta T/\frac{1}{2}U^2$

The value plotted is $2gJK_p\delta T/U^2$ where δT was the overall total temperature rise of the air in $^{\circ}C$, divided by the number of stages, the other quantities being in consistent units. δT was calculated from the input torque, mass flow, and speed. A correction was made for bearing friction.

Isentropic efficiency

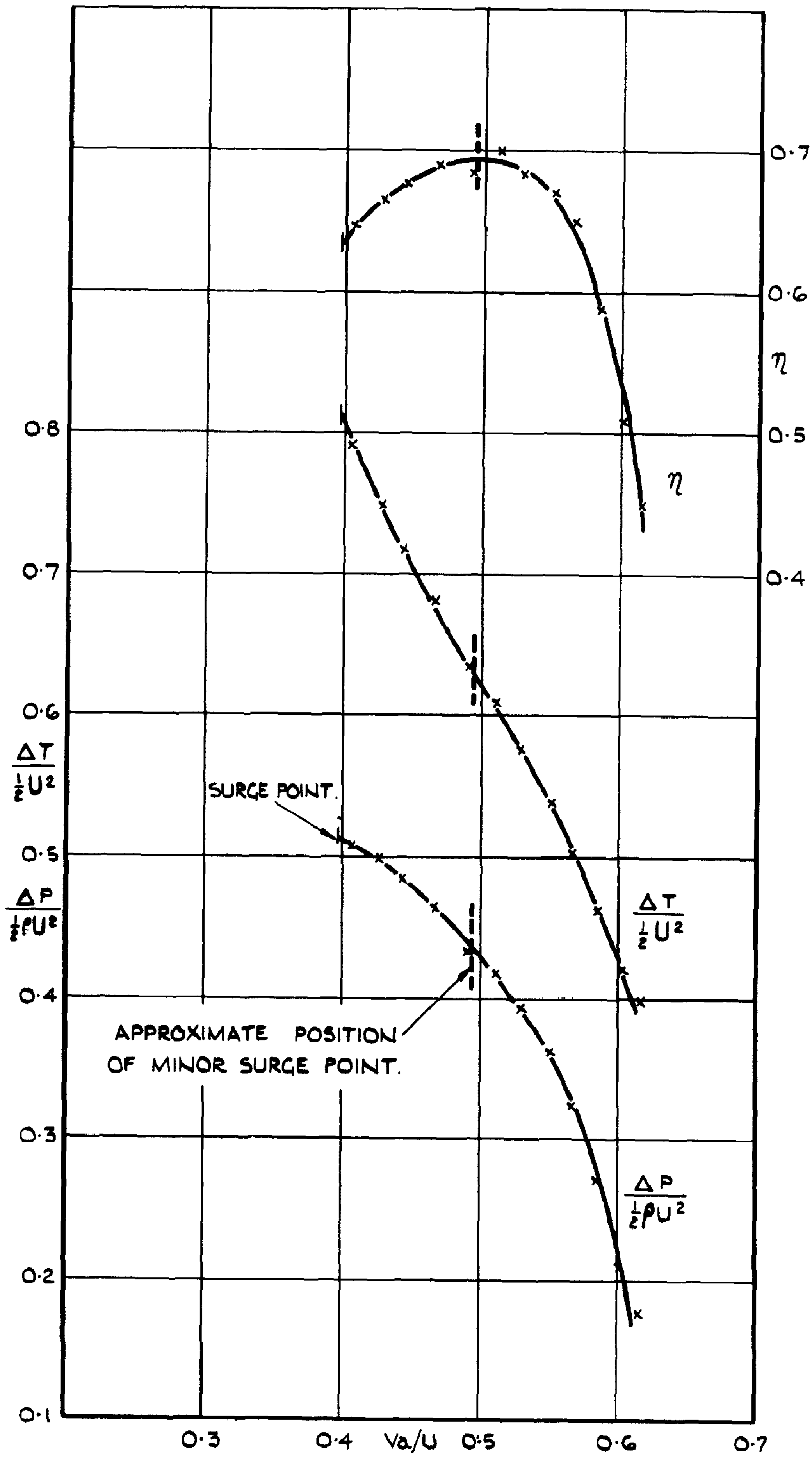
The isentropic efficiency was based on the overall total pressure rise and the overall total temperature rise.

FIG. I



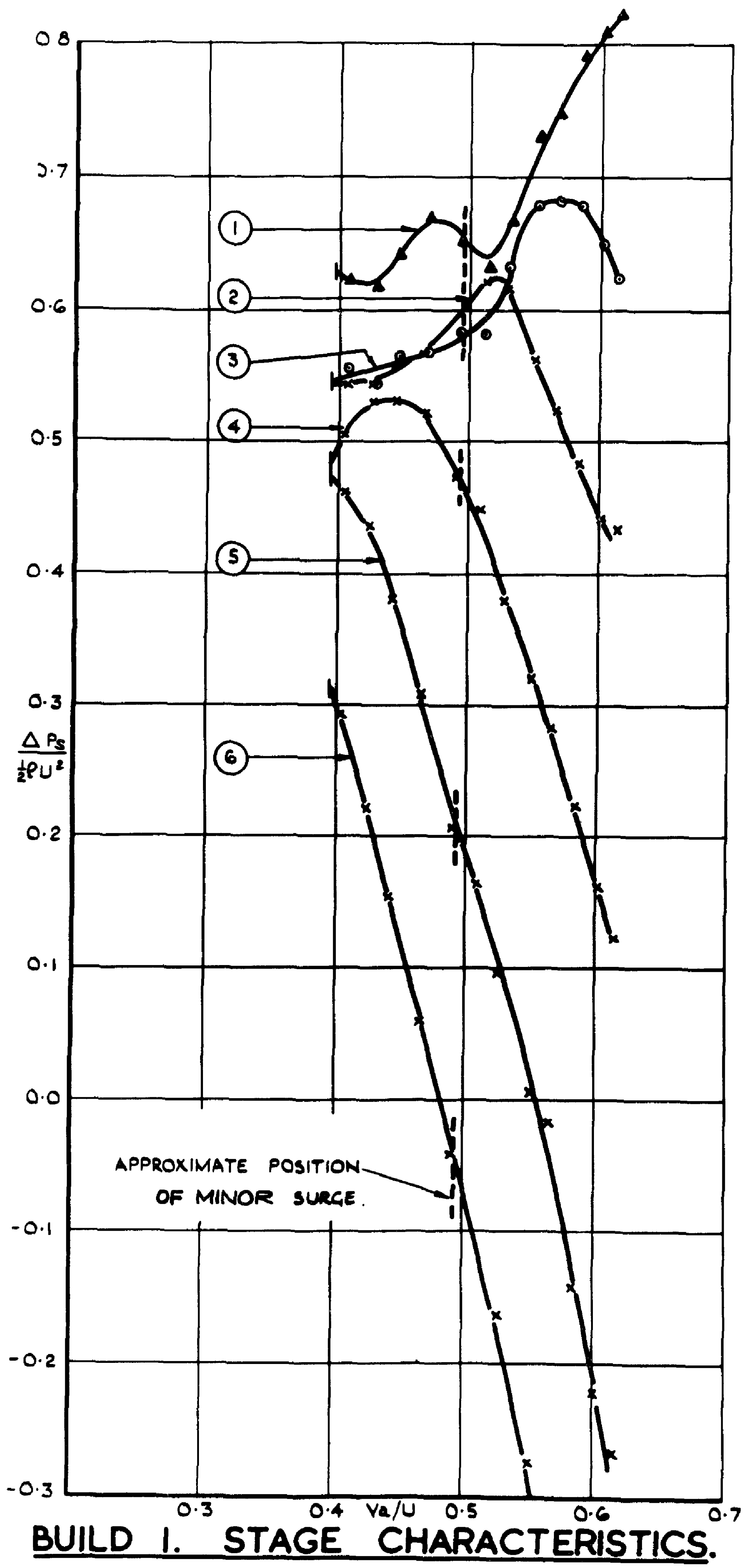
ASSEMBLY OF COMPRESSOR.

FIG. 2.



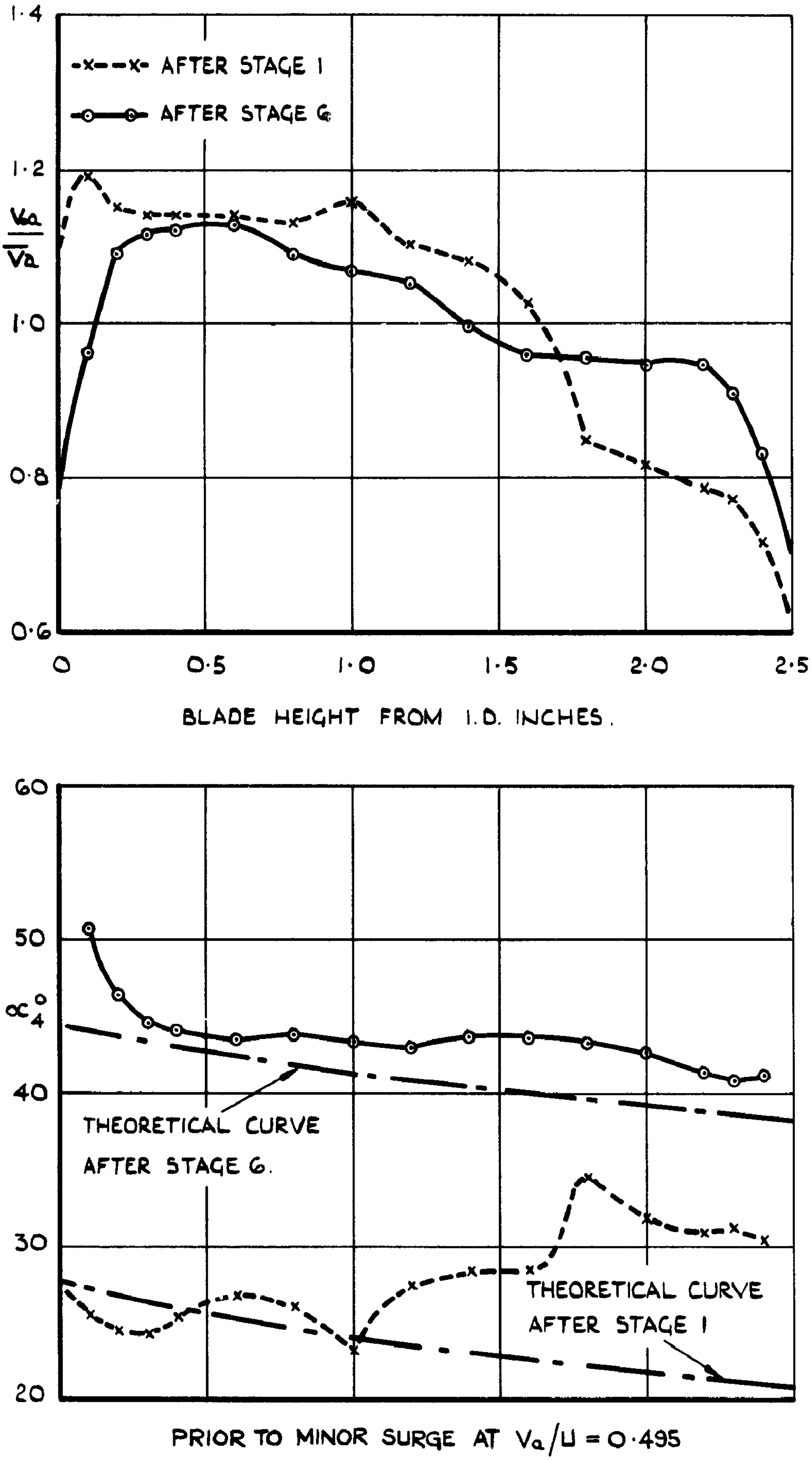
BUILD I. OVERALL CHARACTERISTICS.

FIG.3

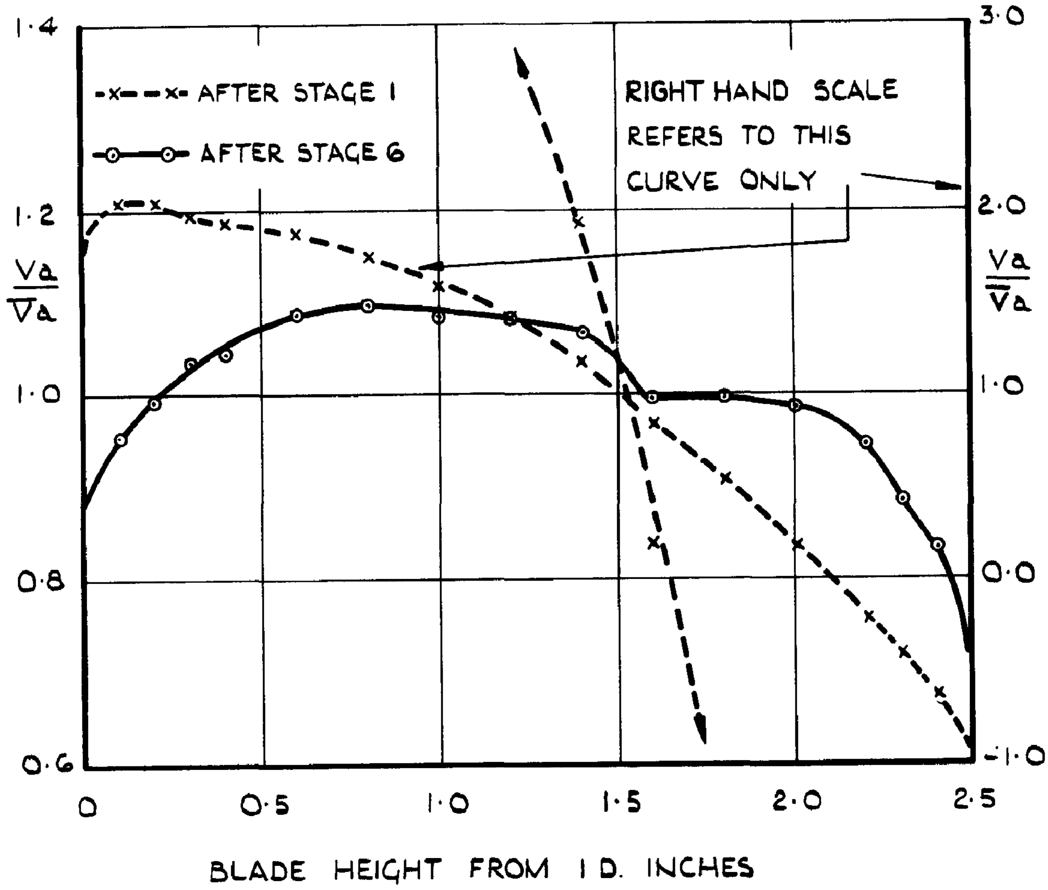


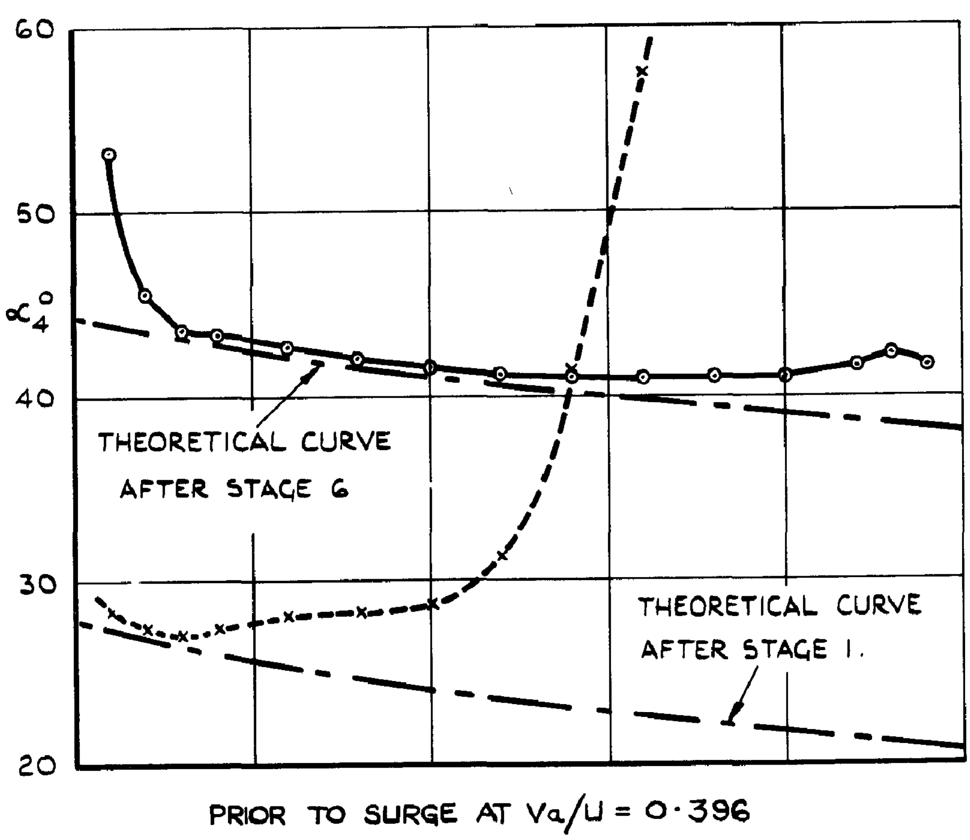
BUILD I. STAGE CHARACTERISTICS.

FIG. 4.



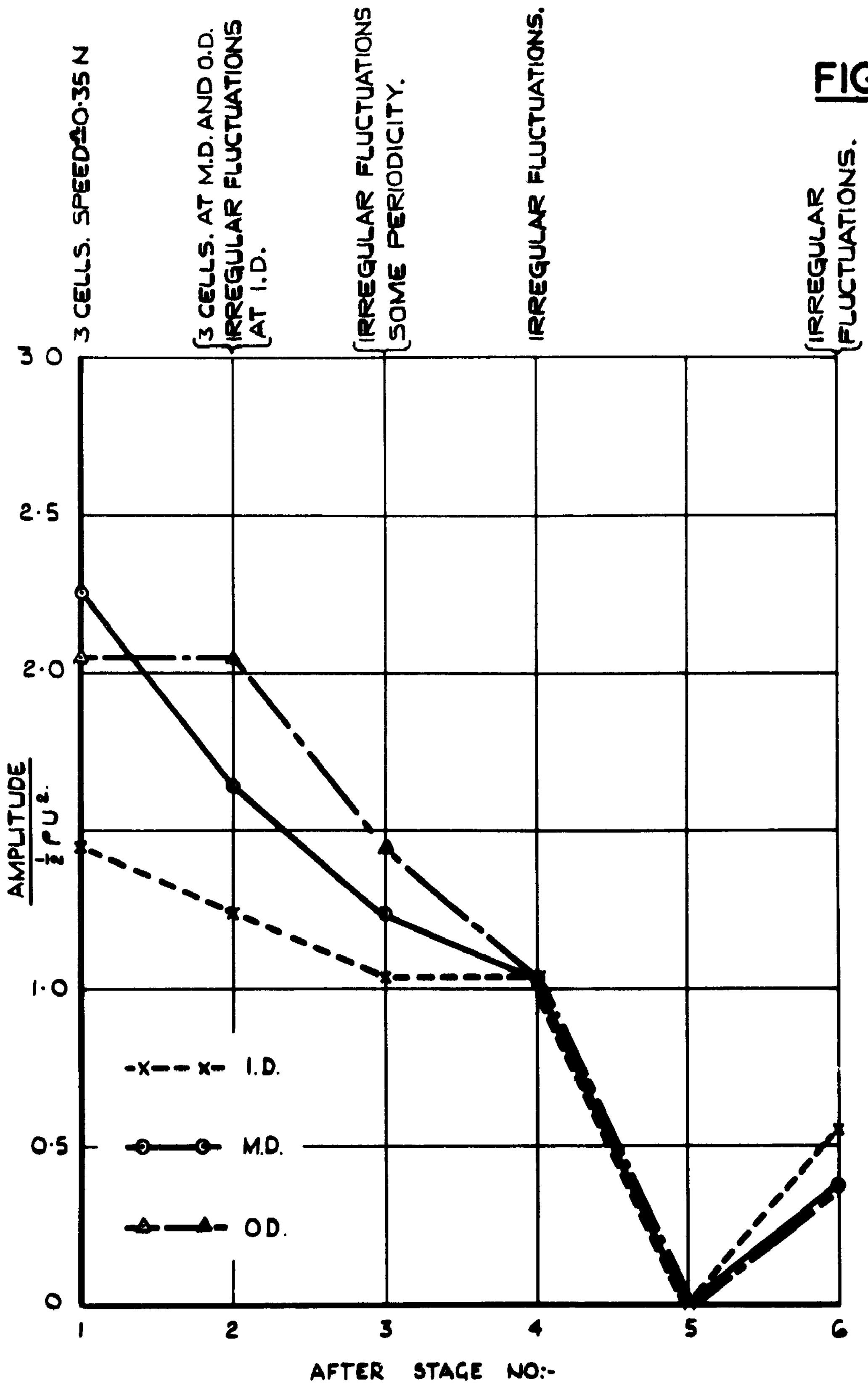
BUILD I. AXIAL VELOCITY AND YAW TRAVERSES AFTER STAGES 1 AND 6 PRIOR TO MINOR SURGE.





BUILD I. AXIAL VELOCITY AND YAW TRAVERSES AFTER STAGES I AND 6 PRIOR TO SURGE.

FIG. 6

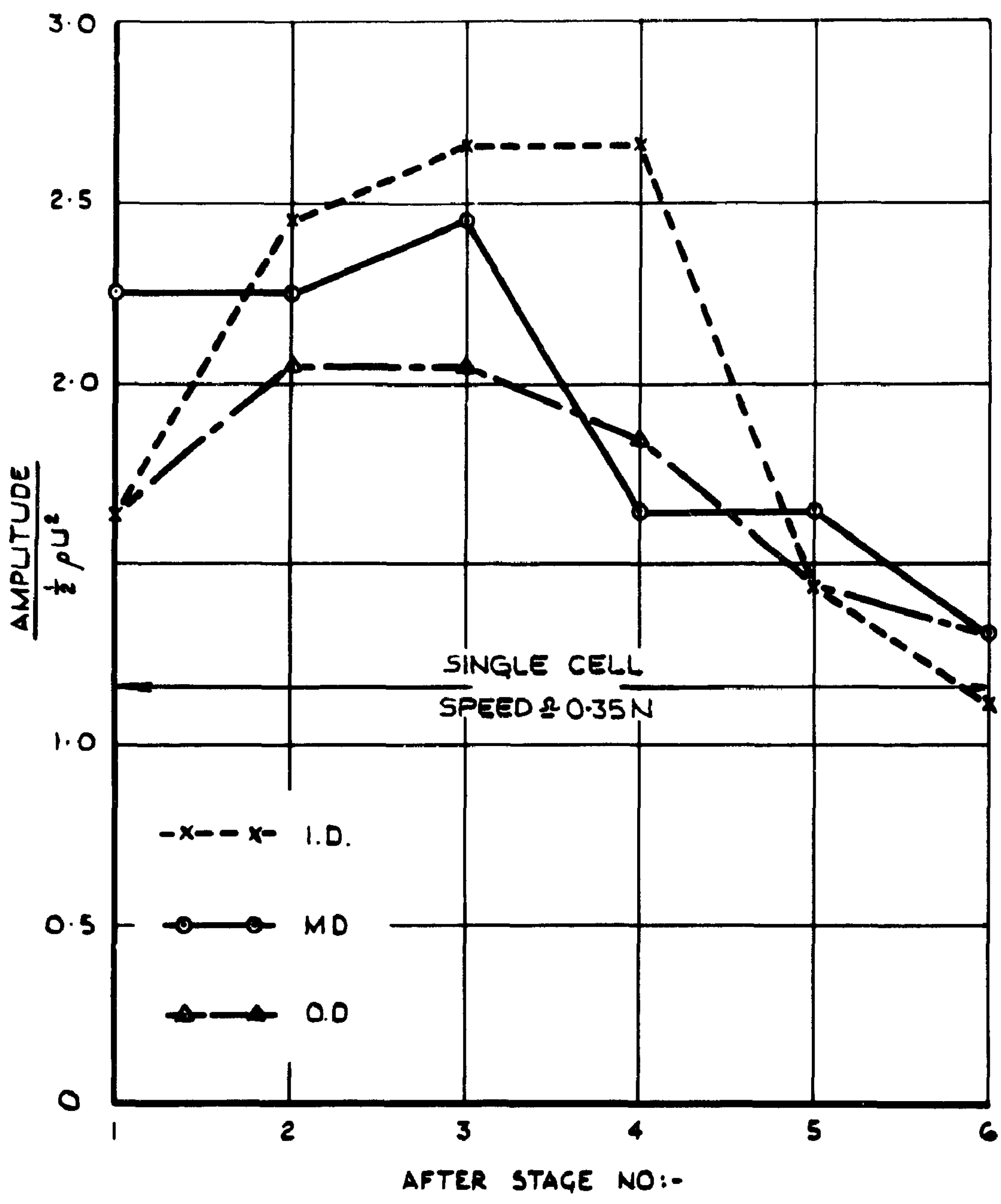


PRIOR TO SURGE AT $V_a/U = 0{\cdot}396$

NOTE COMPRESSOR IN "MINOR SURGE" CONDITION.

BUILD I. AMPLITUDES OF PRESSURE FLUCTUATIONS THROUGH STAGES PRIOR TO SURGE.

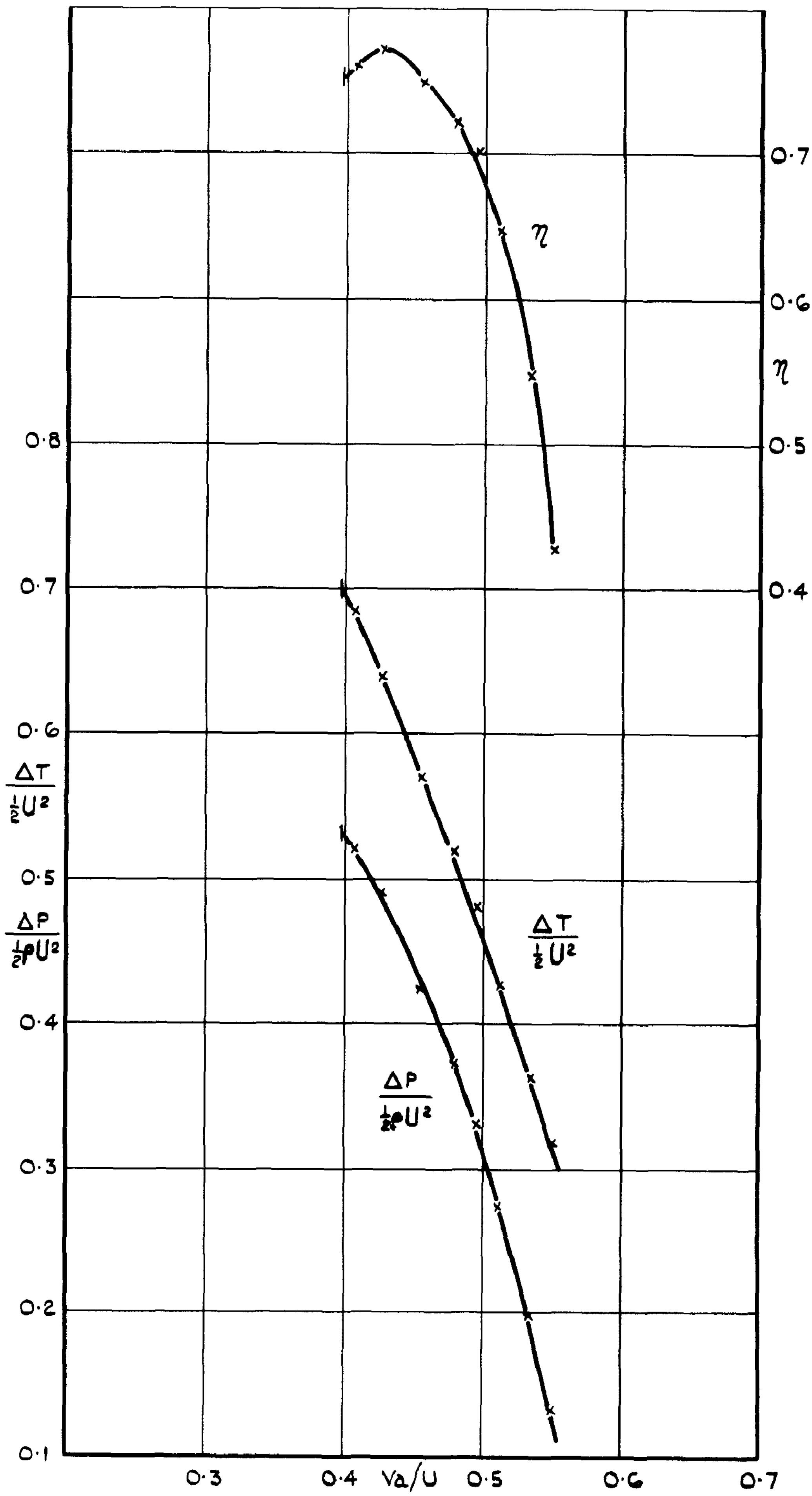
FIG. 7



SURGE $Va/U = 0{\cdot}396$

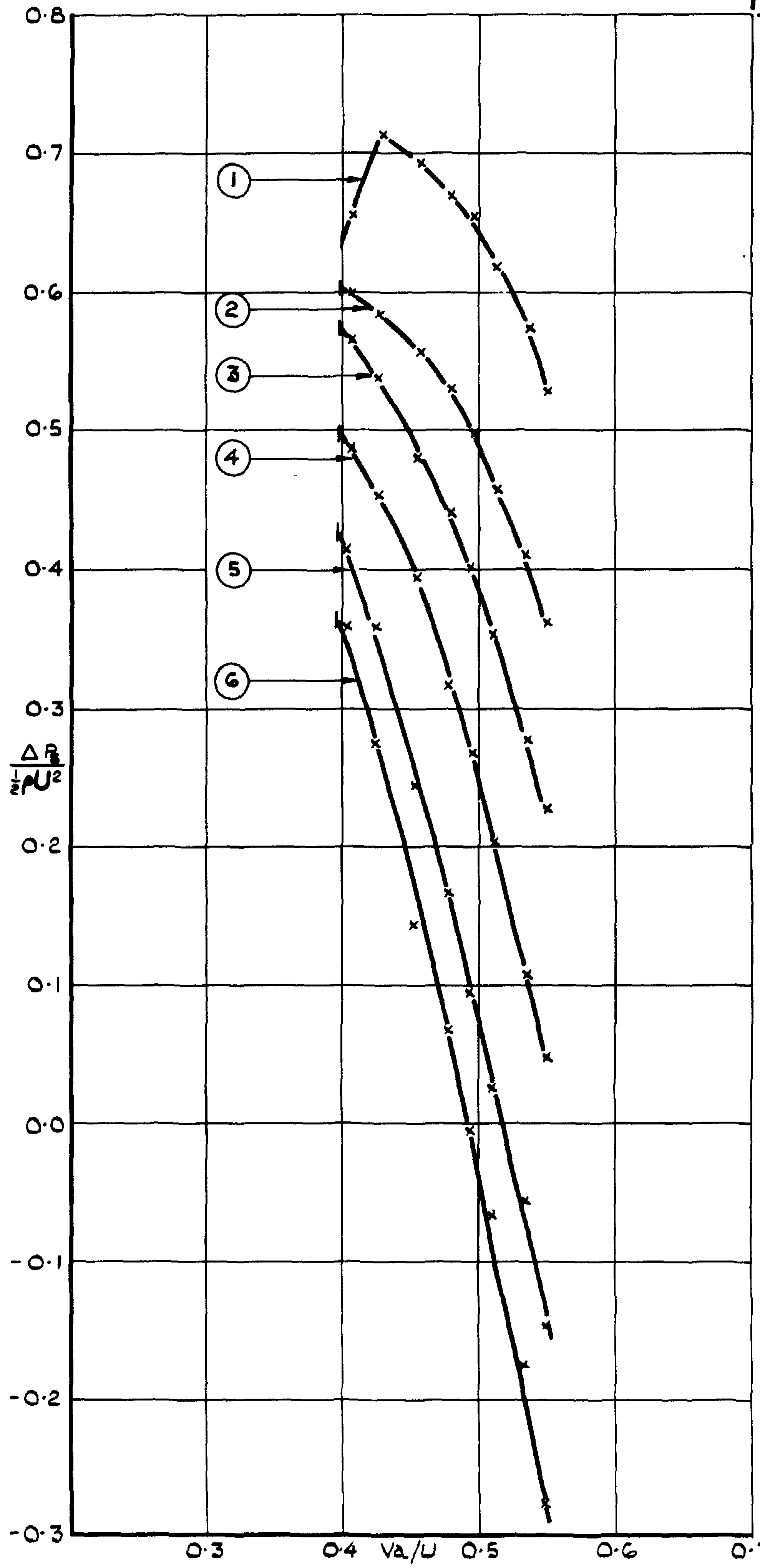
BUILD I. AMPLITUDES OF PRESSURE FLUCTUATIONS THROUGH STAGES DURING SURGE.

FIG. 8.



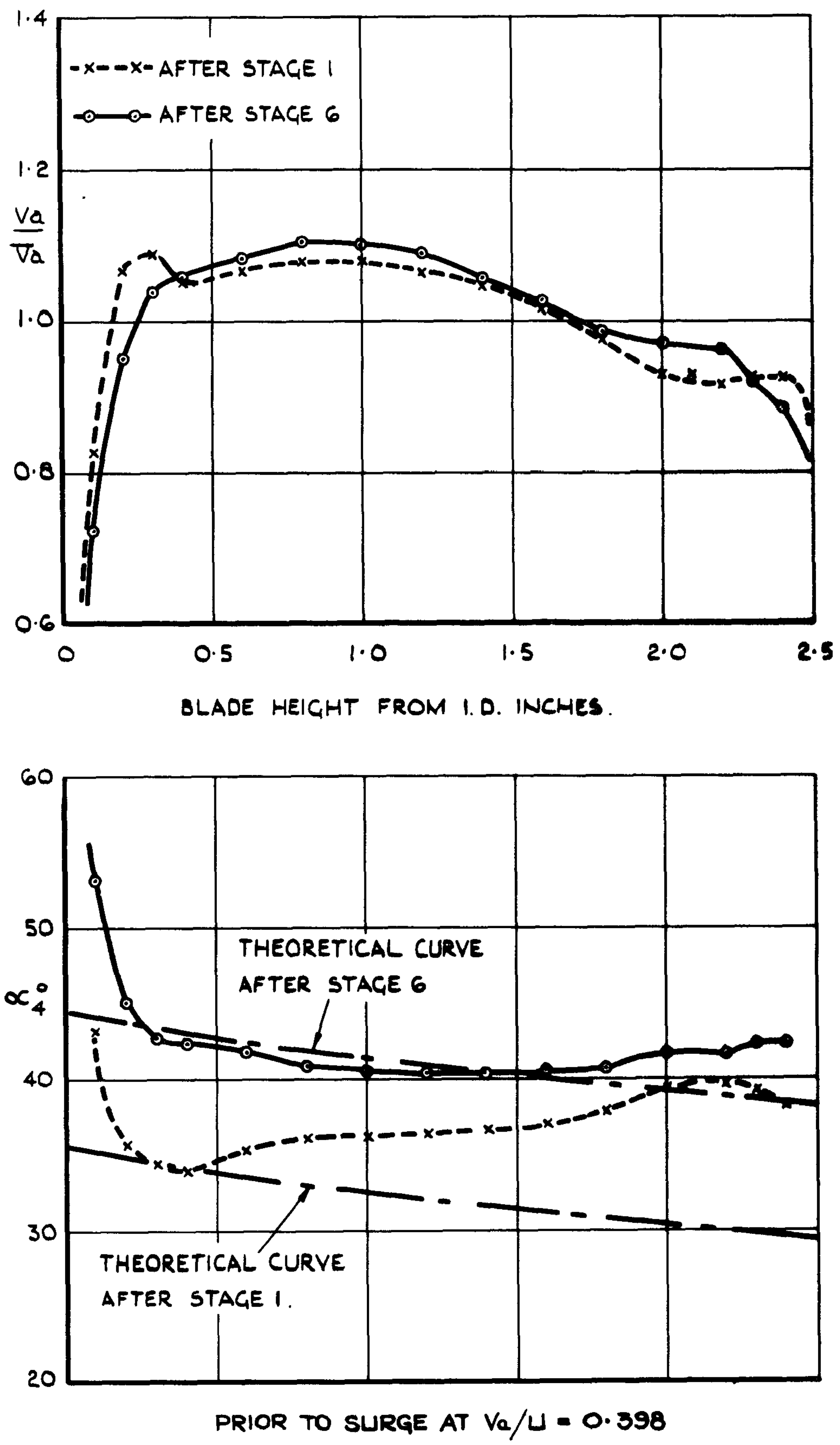
BUILD 2. OVERALL CHARACTERISTICS.

FIG. 9.



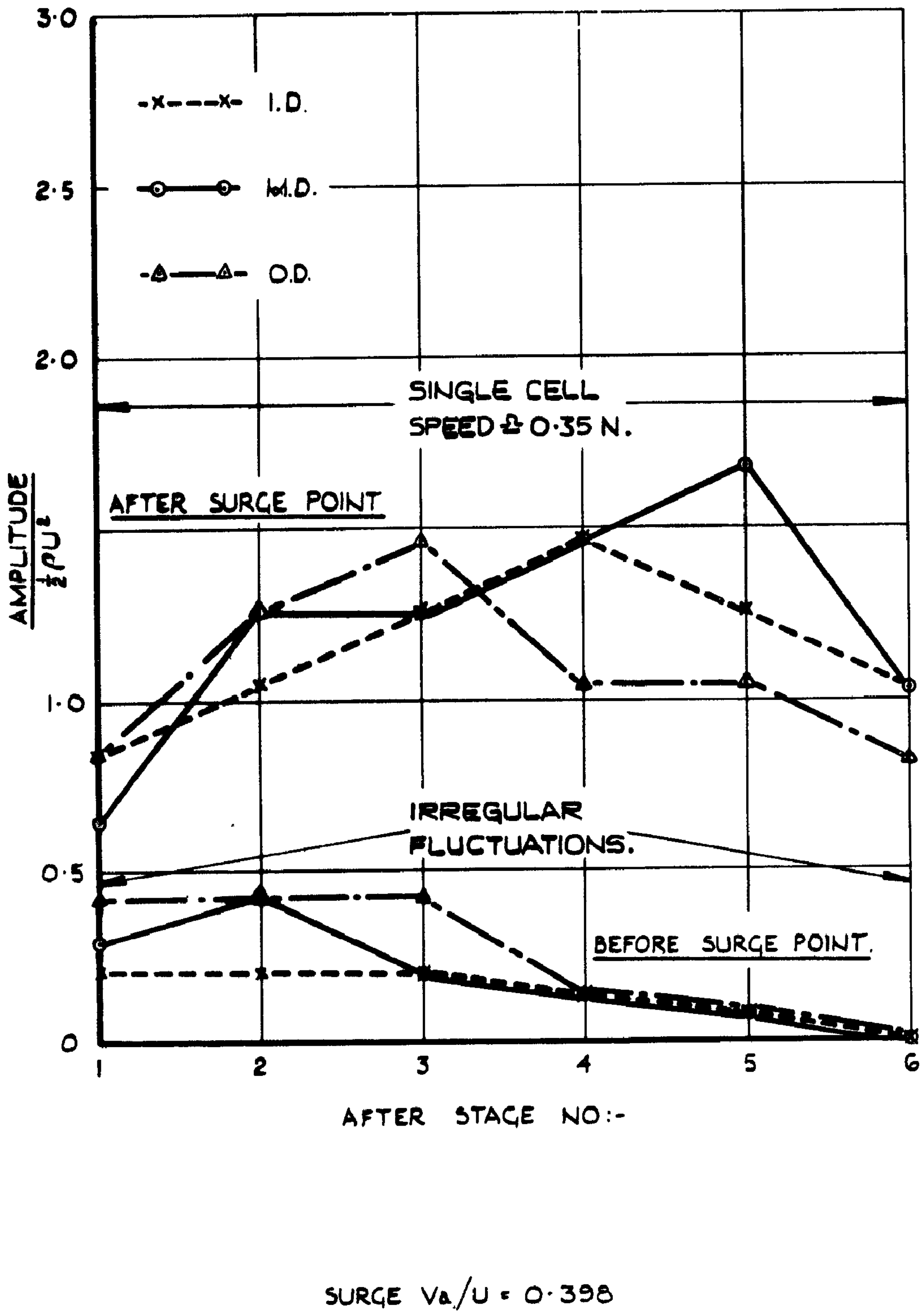
BUILD 2. STAGE CHARACTERISTICS.

FIG. 10.



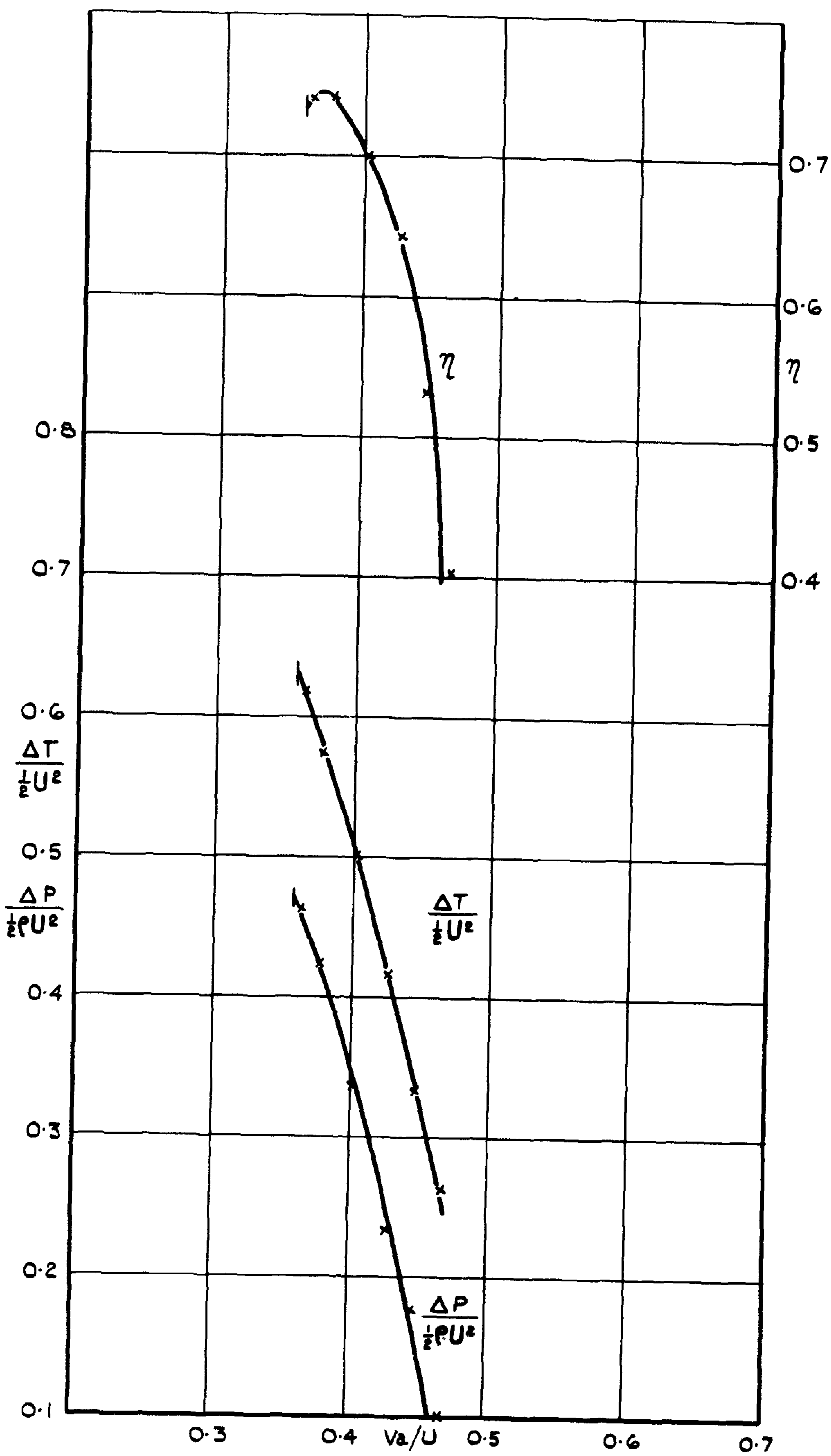
BUILD 2. AXIAL VELOCITY AND YAW TRAVERSES AFTER STAGES 1 AND 6 PRIOR TO SURGE.

FIG. 11.



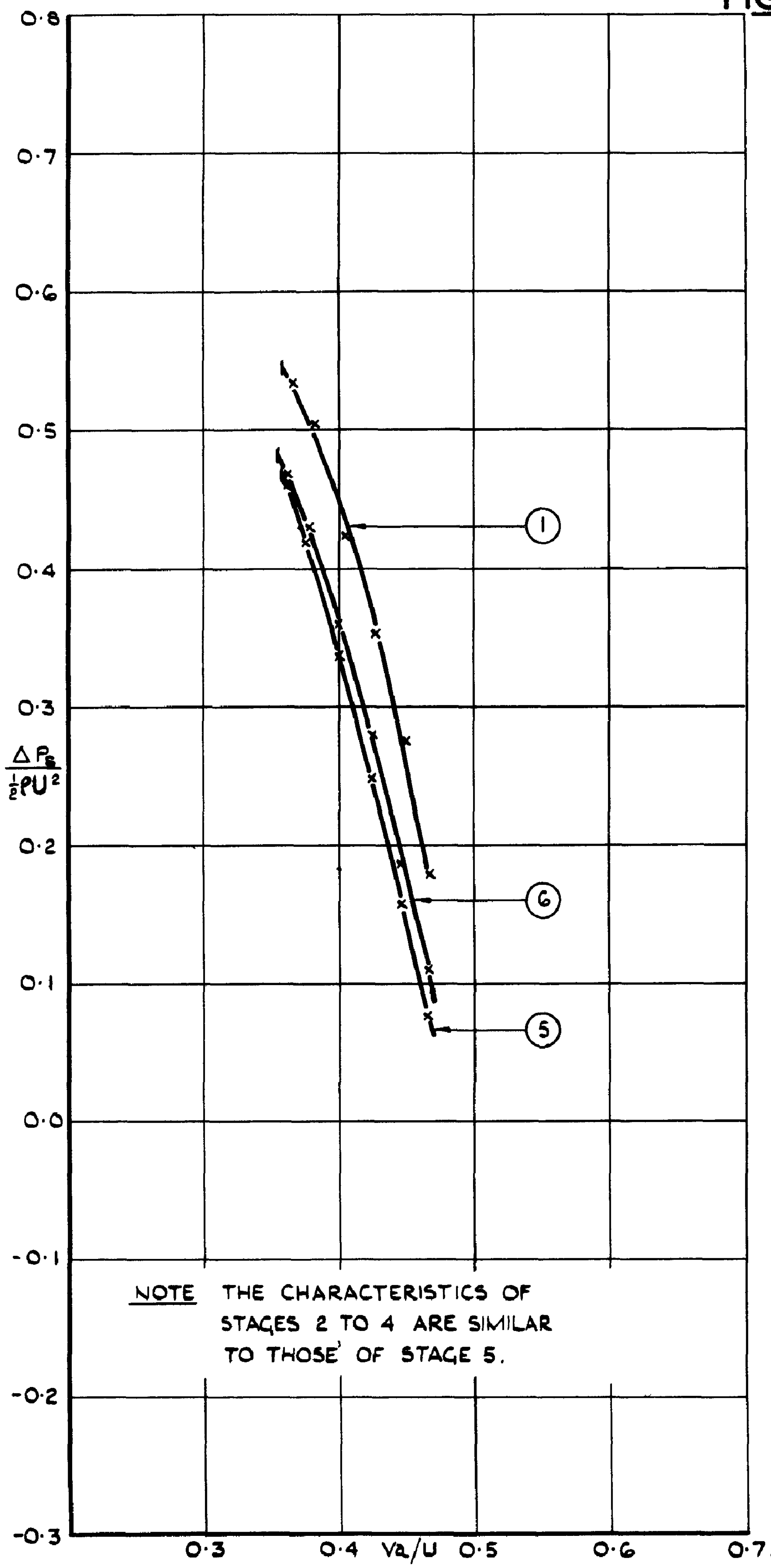
BUILD 2. AMPLITUDES OF PRESSURE FLUCTUATIONS THROUGH STAGES BEFORE AND AFTER SURGE POINT.

FIG. 12.



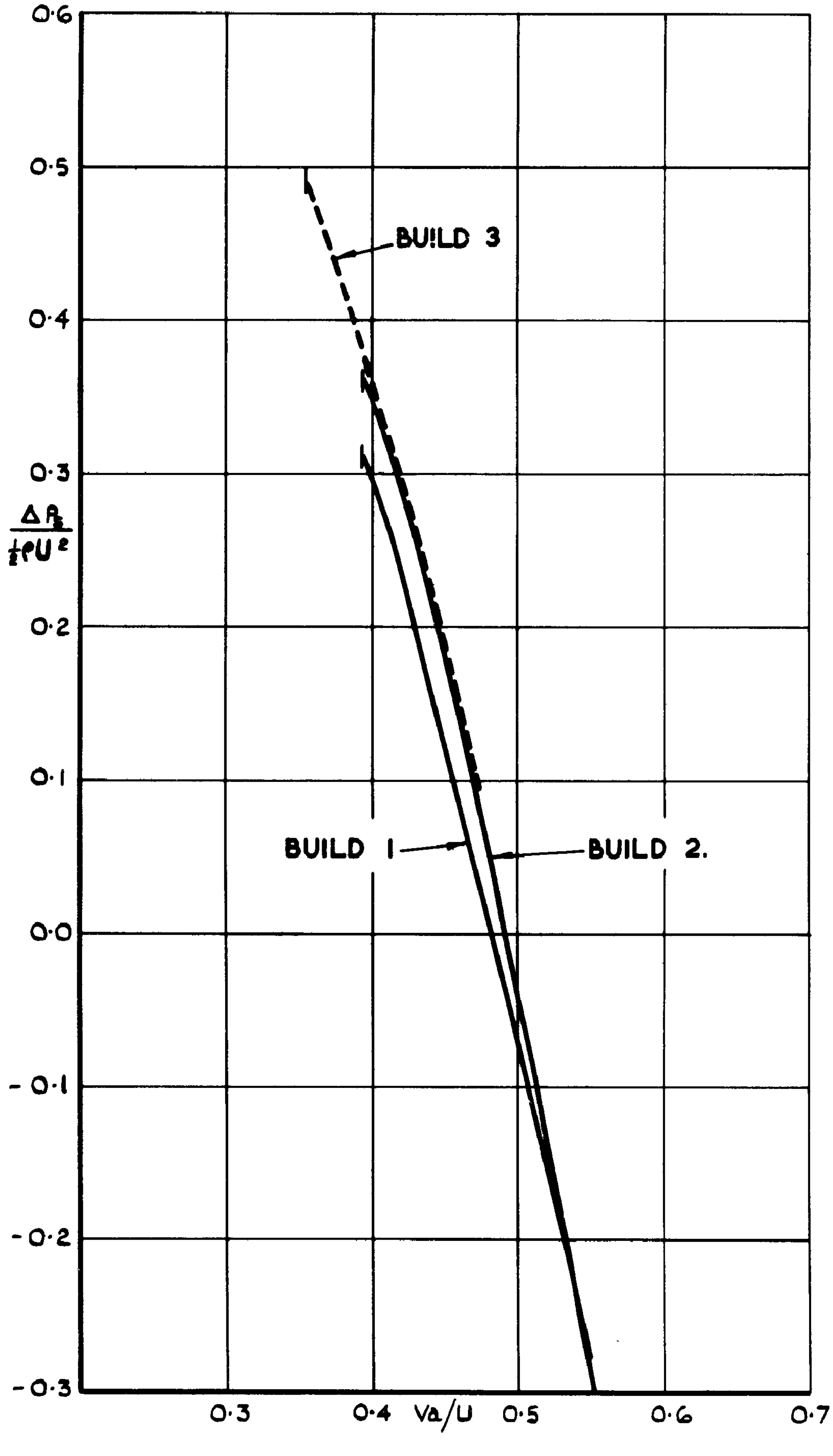
BUILD 3. OVERALL CHARACTERISTICS.

FIG. 13.



BUILD 3. STAGE CHARACTERISTICS.

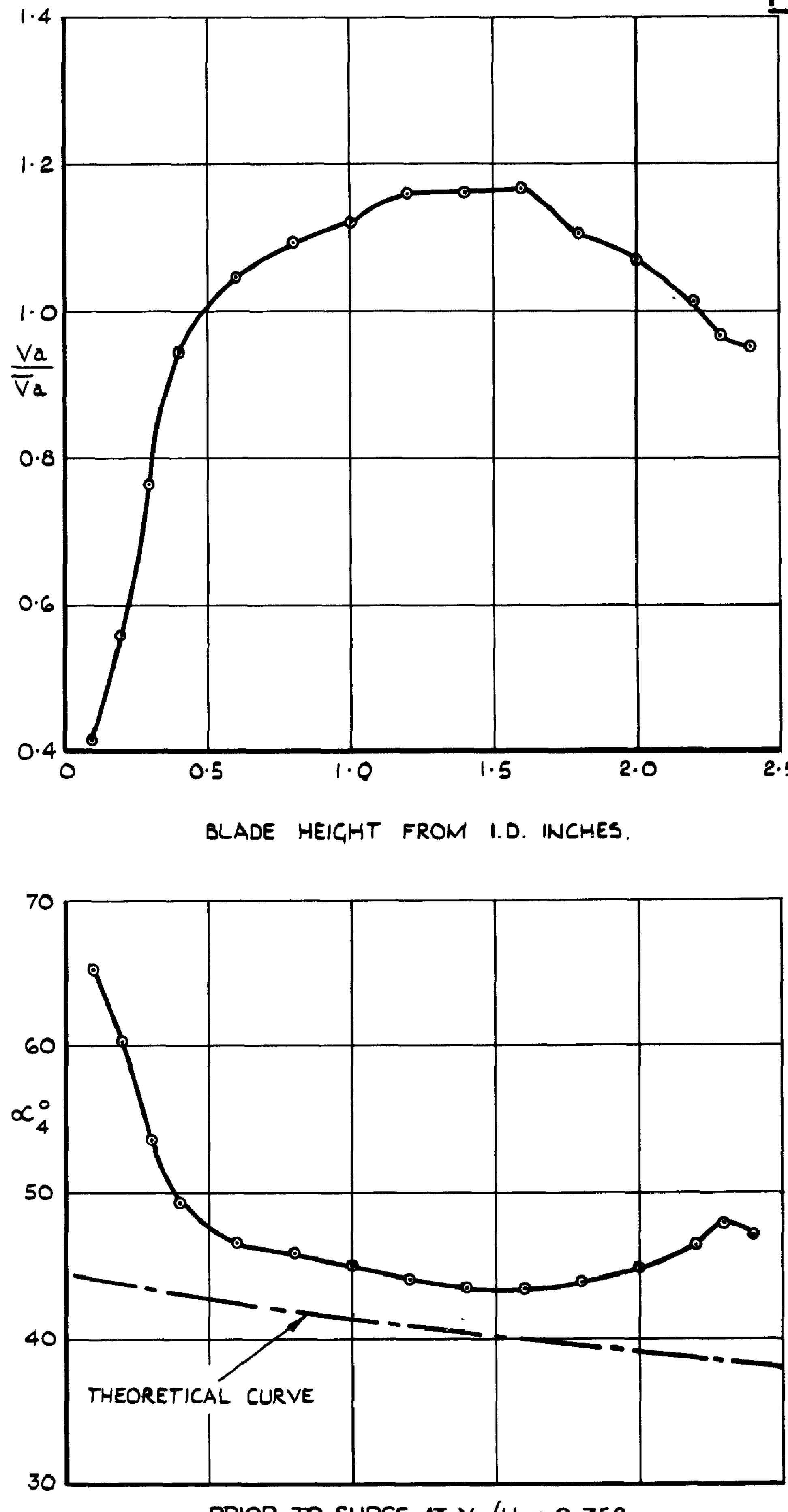
FIG. 14.



ALL BUILDS.

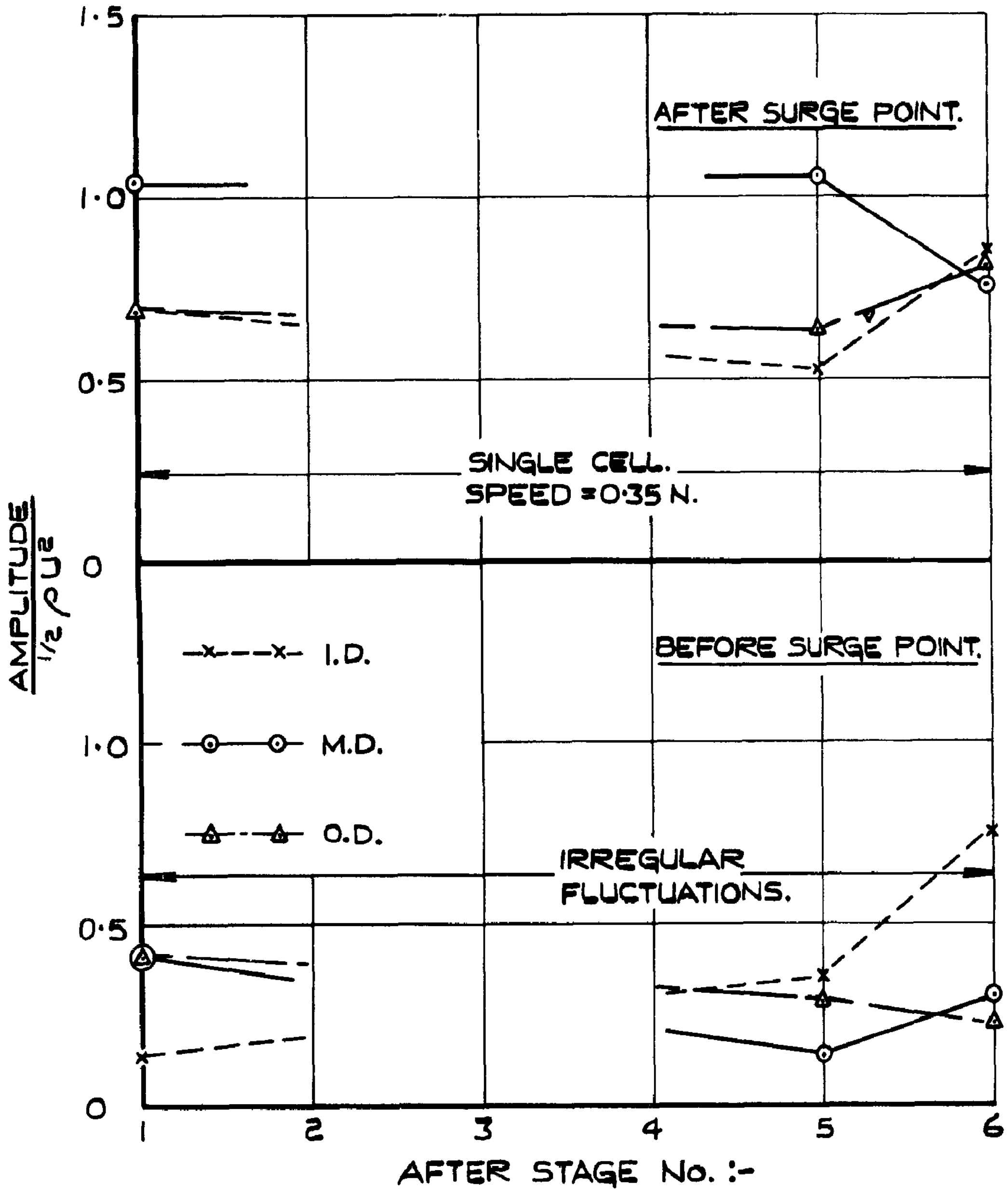
CHARACTERISTICS OF STAGE 6.

FIG. 15.



BUILD 3. AXIAL VELOCITY AND YAW TRAVERSES AFTER STAGE 6 PRIOR TO SURGE.

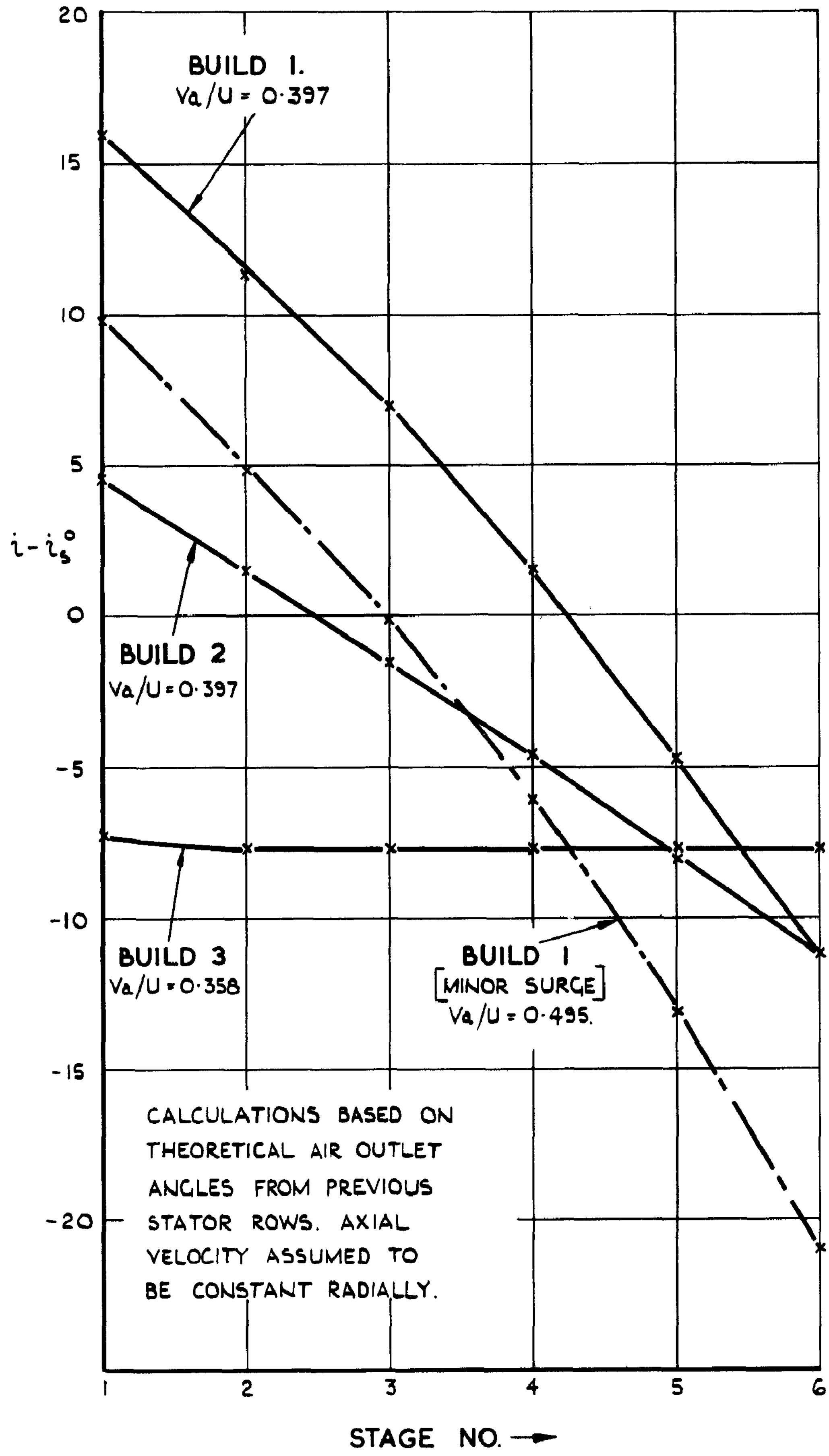
FIG. 16



SURGE $V_a/U = 0{\cdot}358$.

BUILD 3. AMPLITUDES OF PRESSURE FLUCTUATIONS THROUGH STAGES BEFORE AND AFTER SURGE POINT.

FIG. 17.



ALL BUILDS. MEAN DIAMETER ROTOR INCIDENCES AT SURGE RELATIVE TO THEORETICAL STALLING INCIDENCES.

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