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An Experimental Investigation of the Blade Vibratory Stresses in a Single-Stage Compressor

By

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An experimental investigation of the blade vibratory - stresses in a single stage compressor

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SUMMARY

The compressor blade flutter investigation has so far been concentrated primarily on static cascades. As a link with full scale compressor results a series of tests has been undertaken on the untwisted stator of a single stage compressor. The tests were carried out in a variable density return circuit rig. The results of the investigation can be summarised briefly:

(1) Stalled flutter, which appears to be identical with that experienced in cascades, can occur in a compressor.

(2) Stalled flutter occurred at all incidences above stall, for sufficiently high air velocities. In a few instances some evidence of a rotating stall cell was also observed.

(3) The critical flutter velocity is close to, but slightly lower than that for a similar blade in cascade.

(4) There is no simple and direct relationship between flutter alternating stress and density.

(5) The maximum flutter alternating stress seems roughly independent of density, provided the air velocity is high enough, over the range of practical interest.

(6) The critical flutter incidence is independent of air density i.e. of Reynolds number, over the range covered.

(7) The critical flutter velocity is a function of air density. Its variation is less than would be expected on theoretical grounds. The experimental variation is given in the text of the Report.

(8) Buffeting stresses, i.e. miscellaneous alternating stresses under normal operating conditions, are proportional to the steady gas bending stress.

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ILLUSTRATIONS

ILLUSTRATIONS (Cont'd)

1.0 Introduction

The compressor blade flutter investigation at N.G.T.E. has been concentrated primarily on static cascades (see References 1, 2 and 3, for example). One compressor investigation has been reported (Reference 4) but the results are not conclusive, and different interpretations have been put upon them (cf. References 4 and 5). On one side it can be argued that, in an annulus, as soon as the blades become stalled the air flow will break down into discrete stall bands. Stall bands have been known to rotate, usually at some 30 to 60 per cent of the compressor speed in typical British designs. An individual blade will thus experience stalled and unstalled flow alternately, and since in typical operating conditions the unstalled portion of the flow is greater than the stalled, aerodynamic damping would predominate over excitation. In Reference 4 it is further suggested that the whole process may be completely random. On the basis of non-stationary stall bands, stalled flutter as experienced in cascades would have no significance in a compressor. The other view point is that the stall bands are not nearly as discrete as might at first appear (unless perhaps the compressor is surged), and that stalling flutter can occur as in a cascade. Tests were therefore carried out to see how far the cascade results are applicable to compressors. To avoid the restrictions arising from stage matching such as would occur in a multi-stage compressor, the tests were carried out on a single stage high speed fan.

A further unknown is the effect of air density on the flutter and/or buffeting characteristics. It seemed reasonable to expect that the larger exciting aerodynamic forces associated with higher air densities would give rise to higher vibratory stresses. However, experience arising from testing compressors as component items under throttled inlet conditions would suggest that density has less effect on flutter than would be expected from simple theoretical considerations. As density becomes an important parameter for high speed flight at low altitudes, the tests were carried out in a variable density rig so that the effect of density on the blade vibration could also be investigated.

2.0 Test equipment and technique

2.1 Description of test rig

The tests were carried out in the N.G.T.E. No. 114 Variable Density Return Circuit Rig. A full description of this rig is outside the scope of this Report, but a functional layout is reproduced in Figure 1. It will be seen that it has a return circuit comprised of the working section, a load throttle, a circuit loss make up fan, a cooler, and a venturi meter. The driving turbine stator assembly, together with all compressor bearing housings, are swung so that the torque input to the model compressor can be measured. The rig absolute pressure can be varied from 4 atmospheres to 1/6 atmosphere by an auxilliary piston compressor. The air is pre-dried with activated alumina when pumping to high pressure. The maximum model compressor speed is limited to 20,000 r.p.m. with the existing rotor.

The blade alternating stresses were measured by means of strain gauges. In order to avoid complications arising from the use of slip rings it was decided to concentrate the initial set of tests on a stator row. The rotor blades were made very stiff so that no flutter could be expected from them even under the arduous conditions at which they would have to operate during the tests.

2.2 Details of test blades

In order to facilitate comparison with the cascade results it was decided to use a constant section stator having a profile and stagger identical with one already tested in the cascade tunnel. The section chosen was 10C4/40C50, in standard notation (Reference 6), and this was set at a stagger of -24.5^{o}. At the time of the design it was thought that a section near the tip of the blade would be a better reference for flutter work than the mean diameter section. Accordingly the number of blades was chosen to give a pitch/chord ratio of unity at 80 per cent of the blade height from the root (outside diameter). This was the pitch/chord ratio used in the cascade tests. The blade height was fixed by the existing compressor annulus dimensions. It was 1.82 in. at inlet at 1.78 in. at outlet from the stator. A chord length of 0.60 in. was adopted giving an aspect ratio of 3.0. Both cascade and compressor blades were cast in H.R. Crown Max. The test compressor blades were thus an identical 0.8 scale version of the cascade blades. Direct comparison with the results of Reference 2 is thus possible.

The cascade aerodynamic tests (Reference 7) at a pitch/chord ratio of unity had shown that the cascade stalled at an inlet angle of approximately 48^{o} at a Mach number of 0.5, decreasing to about 45^{o} at a Mach number of 0.6. At the slightly higher mean diameter pitch/chord ratio of these stators it was anticipated that the stalling angle would be slightly lower than this. It is clearly impossible to maintain a constant incidence along the whole length of a compressor stator, as in a cascade test, for all operating conditions. As we were primarily interested in stalling phenomena it was arranged that the rotor should deliver a constant inlet angle to the stator at its low speed stalling point, i.e. at 48^{o}. The corresponding mean flow coefficient for the rotor is 0.622. At this flow the rotor is operating, theoretically, at "nominal" incidence. Previous tests with this rotor as part of a 50 per cent reaction stage had shown the surge flow coefficient to be 0.58. It was therefore hoped that any stall and surge phenomena would be initiated by the test stator blades, though at lower flows stalling of the rotor might influence the results.

The test stator blades were cast with integral roots, which were made as large as the stator casing would allow. The blade roots were bolted firmly to a ring to minimise root damping. Details of the arrangement are shown in Figure 2. The blades themselves were first checked for soundness, then selected and positioned from a frequency test. The frequencies of the blades ranged from 662 to 747 c/s, and included two sets of six tuned blades for comparison purposes. The mechanical damping of the blades was very low. Tests on some of the blades in still air at normal atmospheric pressure gave a logarithmic decrement of 0.007 (measured in situ).

2.3 Strain gauge equipment

The strain gauge amplifier channels were made up of a number of standard 14 in. panel mounted units supplied by Messrs. Southern Instruments Ltd. Two amplifier channels were used, each comprising a strain gauge panel (selection, D.C. polarising source and calibration), an R.C. preamplifier panel and a D.C. driver amplifier panel, giving an overall voltage gain of about 4×10^{5}. The stabilised power supply is common to both channels. A three-channel cathode-ray tube unit employing high performance monitor-type tubes (VCRX 214) was constructed for the display of the strain gauge signals and for recording by a 35 mm. continuous moving film camera. A film speed of about 40 in. per second was used for these tests. A two-channel marker amplifier was also constructed to apply "time" and "event" signals simultaneously to the third cathode-ray tube. In this case the "event" was rotor speed. Photographs of the strain gauge recording equipment and some typical test records are shown in Figures 3(b) and 4(b) and 4(c). To facilitate visual monitoring of the strain gauge signals, between actual exposures, a time-base panel is included in the amplifier unit

A total of 20 blades were strain gauged. All gauges were of the nitro-cellulose bonded type and were affixed with "Durofix" cement. 18 gauges were Tinsley type 6H, which have S.E. wire (similar to Eureka) of 0.001 in. dia., giving a resistance of 50 ohms. The other two gauges were Rotol type DC½/1/100, using nickel copper wire of 0.001 in. dia. giving a resistance of 100 ohms. 15 blades were gauged in the flexural position (13 Tinsley and 2 Rotol gauges) as shown in Figure 2, while 5 blades had gauges at 45° to the flexural position. This latter position was thought to be the most suitable for detecting any possible torsional or complex blade vibration modes, having regard to the size of the gauge relative to the blade.

The lead-out wires (4/0.006 in. wire, cotton covered and lacquered) were taken through holes drilled in the blade roots and rings, as shown in Figure 2, and hence through the casing to an external terminal board. From this, screened cables led to the remotely situated amplifying and recording equipment. The strain gauge selection panels provide for the connection of 6 gauges each; thus the immediate selection of one out of six gauges in each channel was possible.

2.4 Test procedure

The following procedure was generally adopted for test purposes. The compressor was run up to the desired speed at maximum flow. The flow was then gradually reduced, and simultaneous records of aerodynamic performance and blade alternating stress taken at suitable intervals up to and through stall and surge and to as small a flow as practicably possible. The procedure was repeated at speed intervals of 2,500 r.p.m. from 7,500 r.p.m., to 20,000 r.p.m., the limit of the rig. Complete tests were carried out at nominal rig pressures of ¼ atmosphere (part repeated), 1 atmosphere, and 2 atmospheres (up to 17,500 r.p.m.) total absolute pressure. All records were taken photographically The duration of a flutter record was about 1 to 1½ seconds. In order to extend the blade and gauge life as much as possible, little "settling time" was allowed for some tests, particularly at high stress levels. Some slight error may have crept into the aerodynamic results from this source. It would be small.

The aerodynamic records gave mass flow, stage temperature rise, and the static pressure at six points equally spaced around each of the inside and outside annulus casings, both before and after the test stator. The static pressure at the outside was also measured before the rotor and before the inlet guide vanes.

The tests were finally terminated by failure of the strain gauges and a suspected compressor shaft thrust bearing failure.

3.0 Test results

The results obtained can be presented in several different ways. The most direct is given in Figures 5, 6 and 7, where the measured alternating stress has been plotted against the flow coefficient at entry to the rotor for ¼ atmosphere, 1 atmosphere and 2 atmospheres nominal rig density respectively. The stage pressure rise coefficient has also been plotted on each of these Figures so that the corresponding aerodynamic performance is known. The results for one blade only have been quoted, although measurements were always taken simultaneously on two blades. Stresses recorded on both blades were, generally speaking, the same, although in one or two instances discrepancies were obtained. These discrepancies are probably due to the somewhat short duration of the records combined with the random amplitude variations. It is interesting to note, too, that surveys of the 12 gauges connected to the measuring equipment always gave stress readings of the same order. The 6 static pressure tappings around the inside and outside annulus walls also gave uniform readings, suggesting that the flow conditions and the resulting blade behaviour were uniform around the circumference.

A more comprehensive presentation of the results can be obtained by plotting alternating stress contours with air inlet angle and inlet Mach number as reference axes as in Figures 8, 9 and 10. This follows the method of presentation usually adopted in the case of cascade results. The air inlet angle to the stator was obtained by yawmeter measurements after the rotor in a conventional stage. Previous test results had shown that density and speed (i.e. Reynolds number) had no effect on the outlet angle from this rotor, over the range to be encountered in these tests. The angle quoted is that at the mean diameter. The inlet angle at other diameters will differ slightly from this, being closest to uniform at the stall point. The variation will be such that well above stall the inlet angle at the tip of the stator will be less than at the mean diameter, while well below stall the angle at the tip will be higher than at the mean diameter. In all calculations a constant axial velocity was assumed across the annulus height at all axial positions.

4.0 Discussion of results

4.1 Comparison with cascade results

There seems little doubt that the vibration observed on these stators is stalling flutter, identical with that observed on a simple cascade. The contours derived from the tests show a striking resemblance with those obtained from the cascade tests. Above the stalling incidence vibration is widespread in both cases. Likewise when stalled there seems to be a critical Mach number or velocity above which the alternating stresses rise rapidly. The nature of the vibration is also the same in both cases. Each blade tested vibrated at its own natural frequency. The vibration was of that haphazard, sporadic nature which is so characteristic of stalling flutter in cascade. This can easily be seen from Figure 4 which compares records from the cascade and compressor test. Additionally it can be seen from this Figure that the vibration had, in general, no discernable amplitude modulation which could be traced to rotating stall cells. In a few cases however it did seem possible to trace a modulation which could be attributed to a stall cell rotating at about 35 per cent compressor speed (see Figure 4(c) for example), but such records were rare and the stall cell rotation did not persist. If the whole vibration were due to some exciting force such as a stall cell, one would expect that compressor speed would be the significant parameter The results do not show this to be so.

The results obtained were perfectly repeatable. Two separate runs over the stalled regions were carried out at $\frac{1}{4}$ atmosphere nominal rig pressure. The results are in agreement. In every case when the flow coefficient was less than about 95 per cent of that for peak pressure rise some flutter was observed. The magnitude of the flutter depended, of course, on compressor speed and rig density.

On the basis of this evidence one must conclude, therefore, that stalling flutter, as experienced in a cascade, can exist in a compressor. It should be pointed out, however, that these tests were carried out well away from resonance with any stall cells. If a single stall cell rotates at 35 per cent compressor speed resonance would be obtained with these blades at 120,000 rev/min which is well outside the test running range. If the frequency of the stators had been such as to bring resonance within the running range, stall cell phenomena would probably have played a greater part in the results. That problem is outside the scope of this investigation.

Assuming, then, that we are dealing with the same physical problem in compressor and cascade, it is necessary to examine the quantitative relationship between the two results. In one sense the results are in complete agreement: when the incidence is high so that the blades are stalled, flutter occurs provided the Mach number is high enough. In so far as the cascade results can be used to predict stall aerodynamically, they can also be used to predict the stalling flutter incidence limit of a compressor blade.

The critical flutter velocity or Mach number (i.e. that at which flutter commences) is lower for the compressor stators than for the cascade. To a certain extent it is to be expected that rotor wakes and the generally less steady flow conditions in a compressor will decrease the aerodynamic damping. This will be most prominent when the total damping approaches zero, i.e. near the critical flutter velocity. It had been hoped to deduce the extent to which a stalled and unstalled rotor blade affected the aerodynamic damping. In Figure 12 the pressure rises across the rotor, the stator, and the complete stage have been plotted against flow coefficient for a number of representative conditions It will be seen that, contrary to design expectations there is no flow range over which the stator is stalled and the rotor remains unstalled. A detailed examination of the stall and surge phenomena involved was outside the scope of the present investigation, and information on this particular aspect of flutter was not obtained.

It is interesting to note that the concept of a limiting flutter velocity and incidence (as used in References 3 and 5) is more nearly approached by the compressor results than by the cascade results. The cascade results indicate that the flutter boundary follows a "stalling incidence - Mach number" boundary. In the compressor stalling incidence seems more independent of Mach number, both aerodynamically and as regards stalling flutter.

It would be unwise to make any deductions from these limited test results, but the agreement is considered so close that it should be possible to correlate compressor and cascade results, using perhaps only one empirical coefficient, when more results are available.

4.2 Effect of density on flutter

As might be expected, increasing the air density increased the general level of flutter stress. In Figure 13 the alternating stress, referred to atmospheric conditions as datum have been plotted against relative air density for a number of operating conditions. It is obvious that no simple direct relationship between stress and density exists.

Looking at the results in more detail it appears that the maximum stress is roughly independent of air density over the practical range. For the blades under test it was limited to about ±30 tons/in². This is not altogether surprising. Non-linearity in the mechanical and/or aerodynamic damping forces is to be expected at high amplitudes and has obviously limited the amplitude and stress in this case. The maximum stress of ±30 tons/in² was not in fact reached at $\frac{1}{4}$ atmosphere relative density. However, the stresses were still increasing when the maximum speed was reached, and had higher speeds been possible the authors believe higher stresses would have been recorded. Maximum stresses are of little practical interest, however, and even at the lowest density the maximum alternating stress level was unacceptable for prolonged operation.

If one regards the high stress flutter zone as bounded by a limiting incidence and a limiting velocity then it is possible to make some generalisation. One can say that the limiting incidence is independent of density. It would, of course, be dependent on Reynolds number rather than density itself. Thus these tests only confirm previous ones that stalling is to a first approximation unaffected by Reynolds number over the range covered. The main effect of density is therefore a change in the critical flutter velocity, i.e. the air velocity at which high stress flutter commences when the blades are stalled. The measured alternating stresses have been plotted against velocity, in Figures 14, 15 and 16, for various inlet angles above stall. It will be seen that the critical flutter velocity is well defined in most cases, and is a function of air density. In Reference 3 it has been shown that the critical flutter velocity and the density are related by the following expression

$$V_f = K_f \; f \; t \; \delta \; {}^{\sigma}/_{\rho} \qquad \ldots \quad \ldots \quad \ldots \quad \ldots \qquad (1)$$

$$\text{or} \quad V_f \propto {}^{1}/_{\rho} \qquad \ldots \quad \ldots \quad \ldots \quad \ldots \quad \ldots \quad \ldots \qquad (2)$$

where V_f = critical flutter velocity

f = frequency of blade

t = blade maximum thickness

δ = logarithmic decrement in vacuo

σ = blade material density

ρ = air density

and K_f = a constant depending on the aerodynamic design of the blade

These results would show, however, that changes in air density have much less effect than the theoretical treatment would suggest. It is possible that the aerodynamic coefficient, K_f, is a function of Reynolds number, but more likely the theoretical treatment of Reference 3 is over simplified.

It has been argued that the frequency parameter is a suitable parameter controlling the critical flutter velocity. These results do not support this view. For it to be an adequate criterion the critical flutter velocity would have to be independent of density.

Of the two hypotheses it would obviously be safer to assume that the critical flutter velocity is inversely proportional to density, at least for densities above atmospheric. It is doubtful if compressors are designed for less than atmospheric inlet air densities. and these special cases would have to be treated on their own merits.

4.3 Effect of density on buffeting characteristics

As an incidental the results obtained during these tests also provide information on the effect of density on the unstalled buffeting effects such as occur under normal operating conditions. The buffeting stresses are due to many adventitious causes, as for example impulses from rotor wakes, pressure fields from other blade rows, somewhat unsteady or non-uniform flow and so forth. These stresses would probably be lower in a single stage than in a multi-stage compressor, but the effects of density should be comparable.

The buffeting stresses measured in these tests have been plotted against velocity and against density in Figures 16 and 17 for two constant unstalled incidences. The scatter in the results is probably due to using amplifier gain settings appropriate to the stalling flutter encountered at that speed and density. The results are quite interesting. It is seen from the Figures that the alternating stresses are substantially proportional to the square of the inlet velocity and directly proportional to air density. This is not surprising in view of the small amplitudes. The buffeting stresses are thus proportional to the steady gas bending stress, and a constant factor of safety on this quantity would provide constant safeguard against failure from this source. An adequate factor of safety would have to be obtained by experience gained from multi-stage compressors.

5.0 Conclusions

An experimental study of the factors governing blade vibration in a single stage axial compressor has been completed. The results of the investigation can be summarised:

(1) Stalled flutter, which appears to be identical with that experienced in cascades, can occur in a compressor.

(2) Stalled flutter occurred at all incidences above stall, for sufficiently high air velocities. In a few instances some evidence of a rotating stall cell was also observed.

(3) The critical flutter velocity is close to, but slightly lower than that for a similar blade in cascade.

(4) There is no simple and direct relationship between flutter alternating stress and density.

(5) The maximum flutter alternating stress seems roughly independent of density, provided the air velocity is high enough, over the range of practical interest.

(6) The critical flutter incidence is independent of air density, i.e. of Reynolds number, over the range covered.

(7) The critical flutter velocity is a function of air density. Its variation is less than would be expected on theoretical grounds. The experimental variation is given in the text of the Report.

(8) Buffeting stresses i.e. miscellaneous alternating stresses under normal operating conditions, are proportional to the steady gas bending stress.

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6	S. Gray	"Fluid dynamic notation in current use at N.G.T.E. A.R.C. current paper No. 97. July, 1950.
7	A. D. S. Carter	"Some tests on compressor cascades of related aerofoils having different positions of maximum camber." R. & M. 2694. December, 1948.

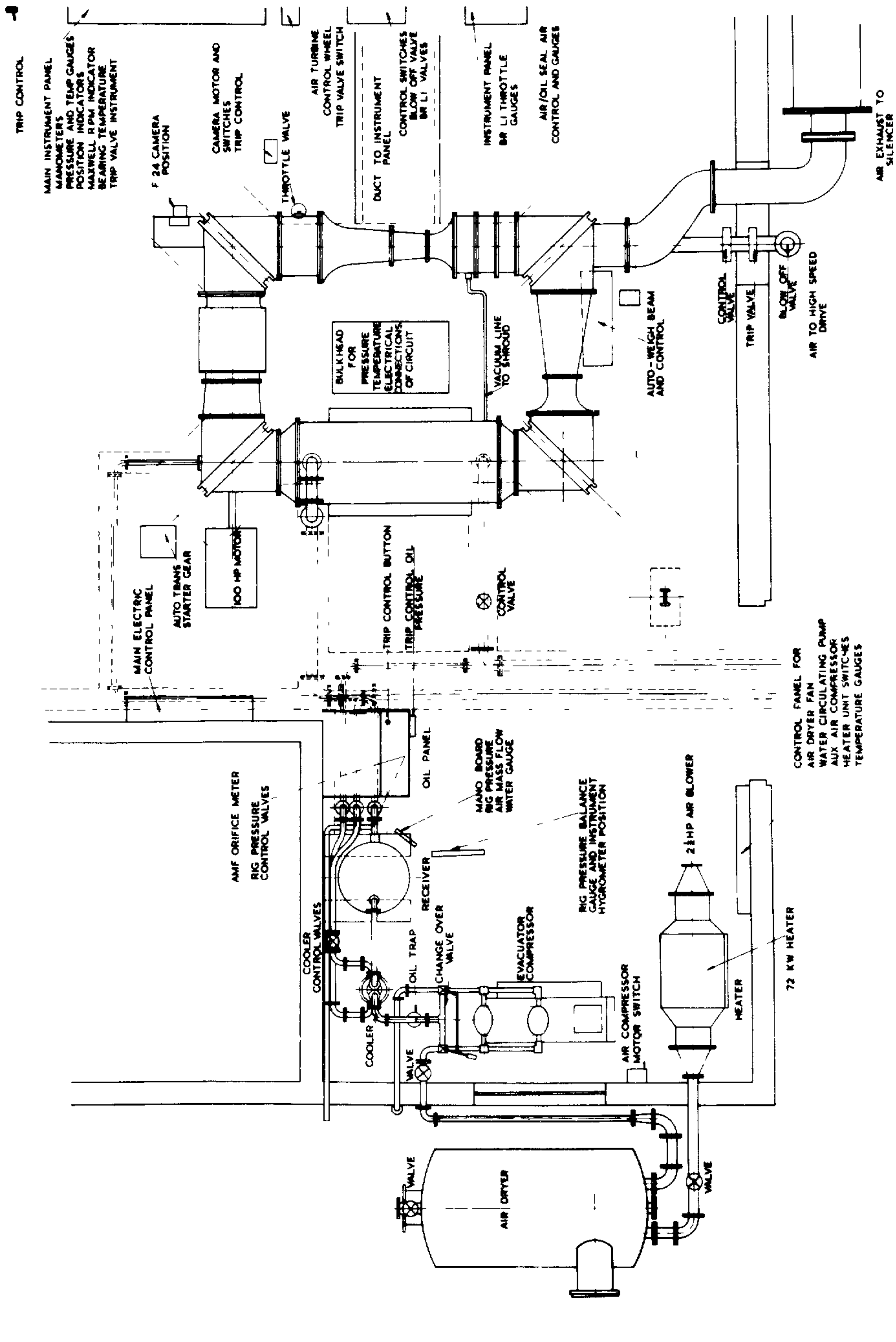
APPENDIX I

Details of Mean Static Density before Test Stator Row

Compressor Speed	Nominal Rig Pressure (Atmospheres)		
r.p.m.	1/4	1	2
20,000	0.0171	0.0682	
17,500	0.0174	0.0700	0.1450
15,000	0.0176	0.0706	0.1469
12,500	0.0186	0.0706	0.1469
10,000	0.0186	0.0722	0.1482
7,500		0.0725	0.1492

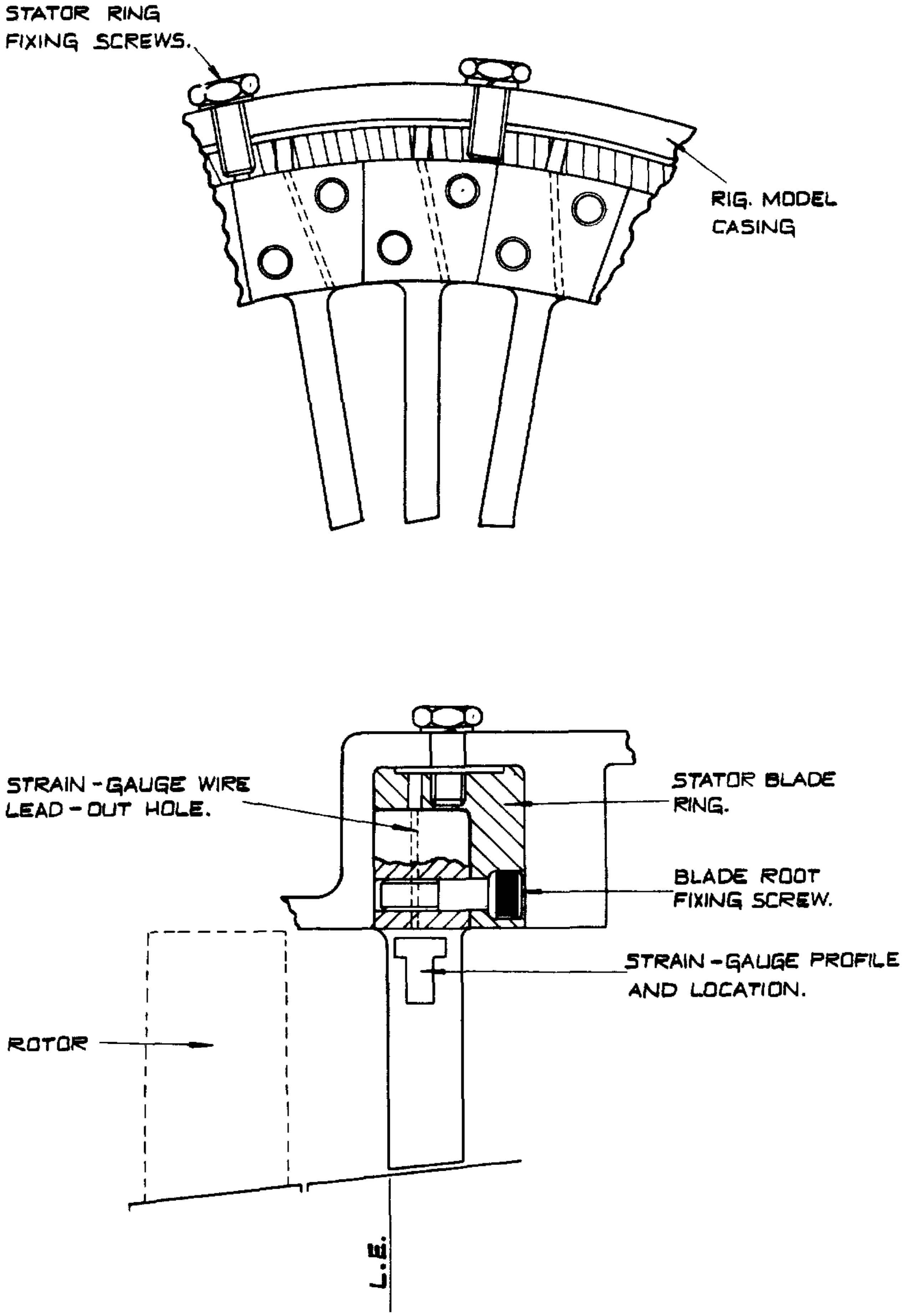
Density quoted in lb/ft^3

FIG. 1



FUNCTIONAL PLAN OF
114 VARIABLE DENSITY TUNNEL.

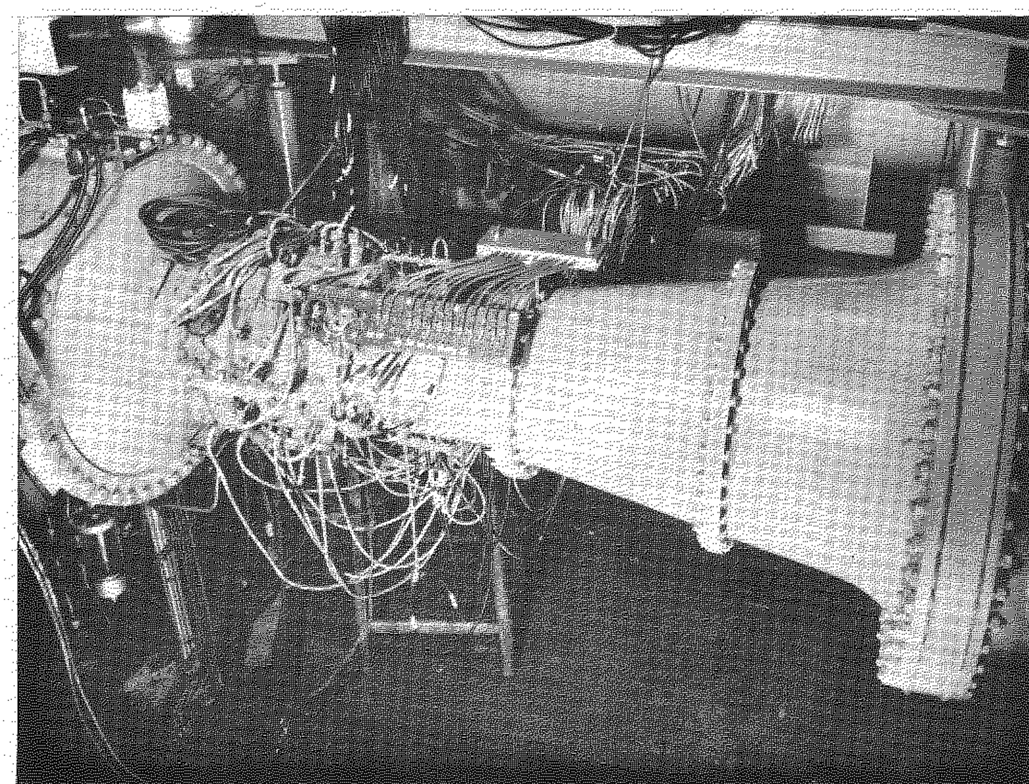
FIG. 2



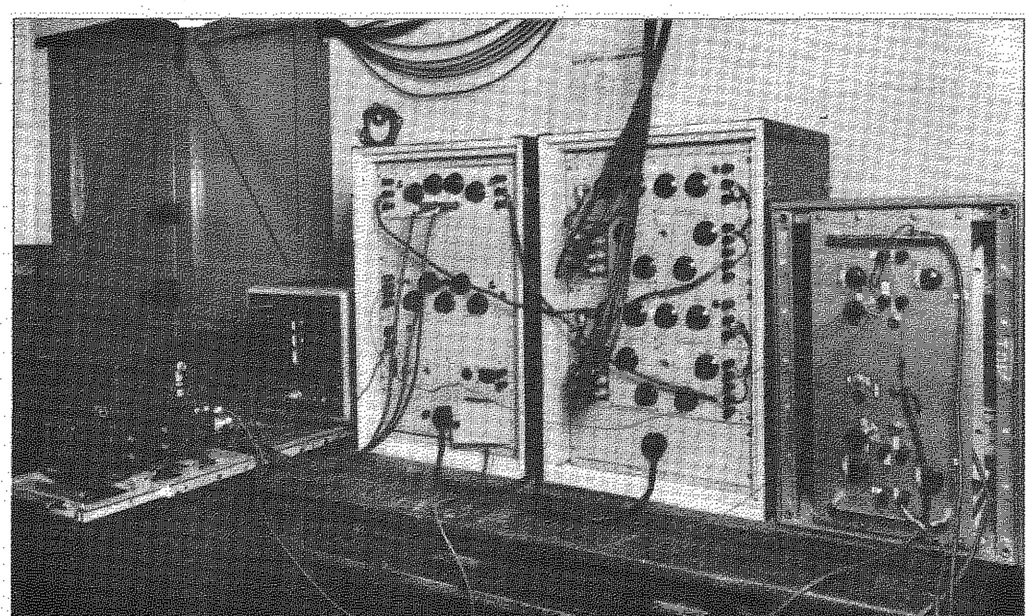
DETAILS OF TEST BLADE MOUNTING.

(SCALE: FULL SIZE.)

FIG. 3

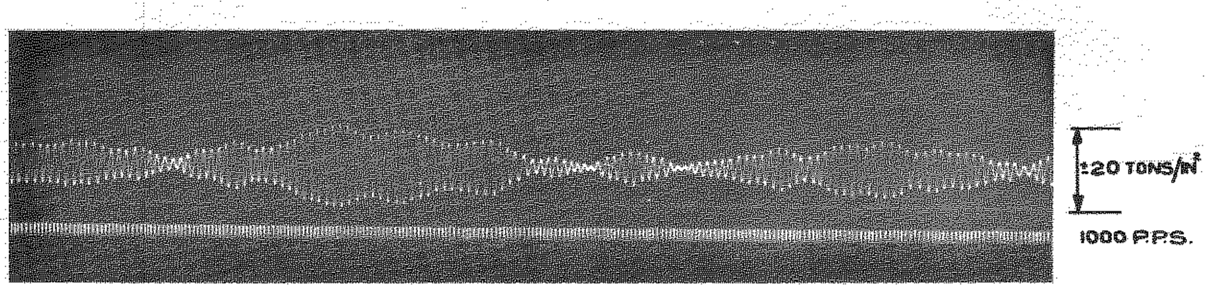


(a) VIEW OF WORKING SECTION OF RIG



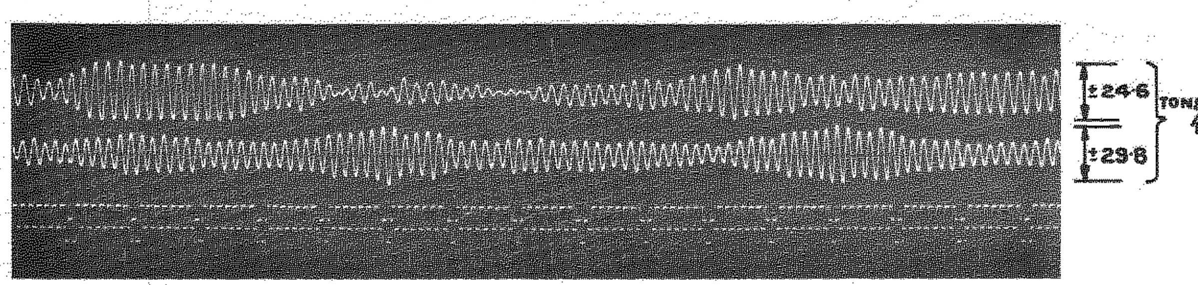
(b) AMPLIFYING AND RECORDING EQUIPMENT

FIG 4

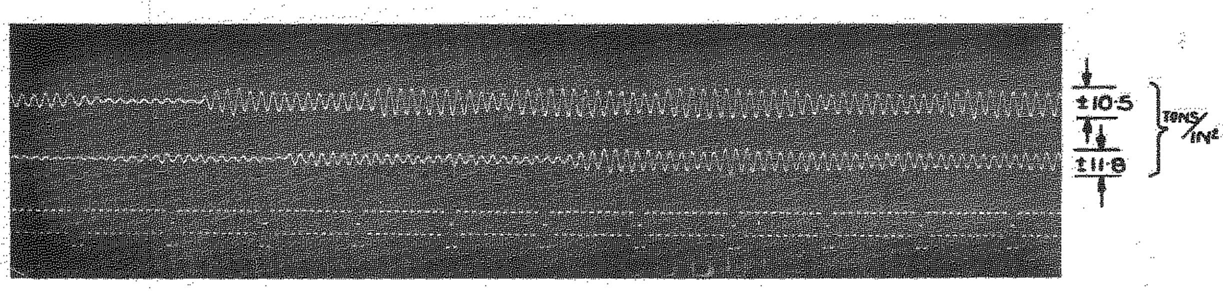


(a) CASCADE TEST, (OPTICAL DETECTION OF TIP MOVEMENT)

$M_n = 0{\cdot}5,\ \alpha_1 = 56{\cdot}3^\circ$



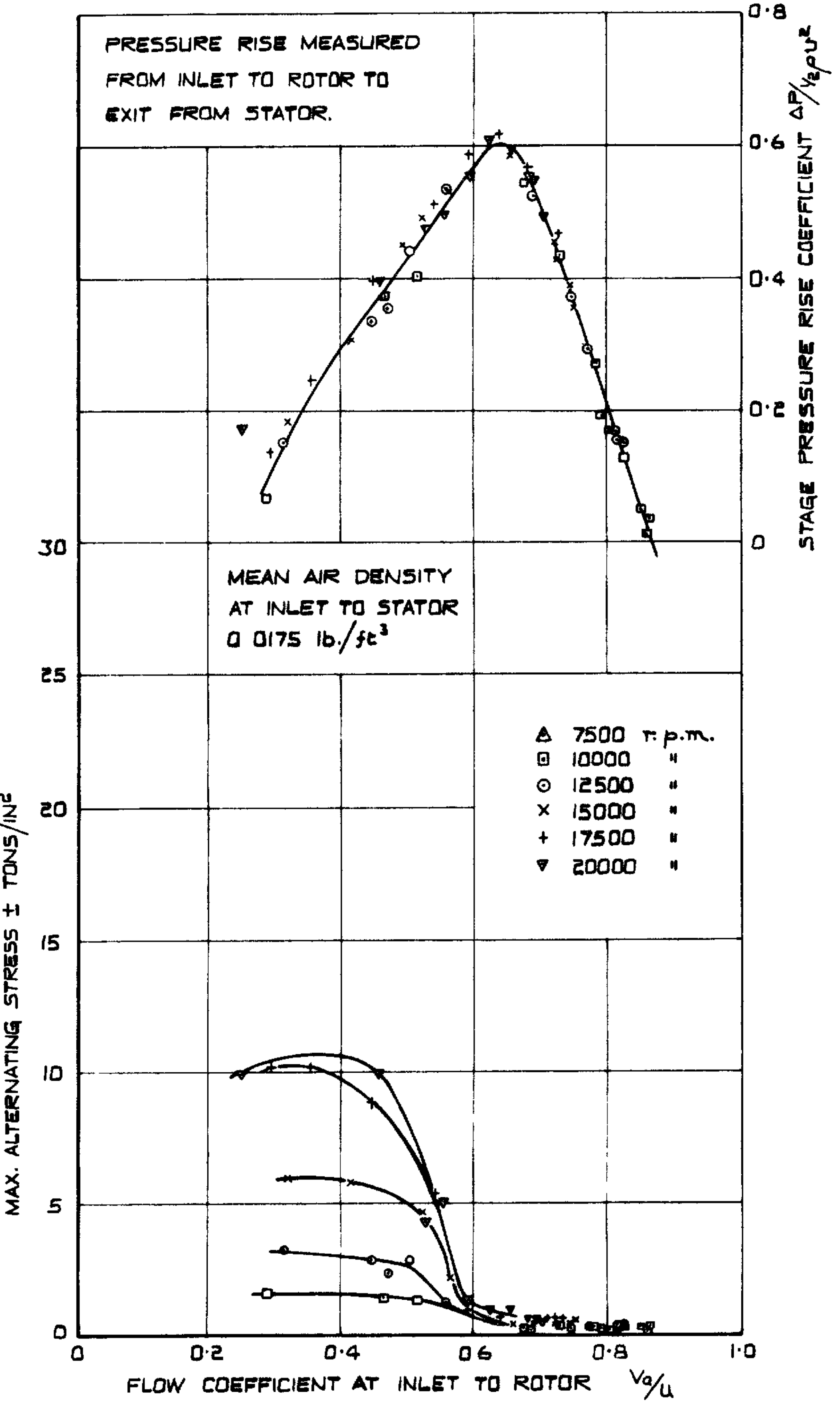
(b) 15,000 R.P.M., $V_a/u = {\cdot}26$, NOM. RIG PRESSURE = 2 ATMOS.



(c) 10,000 R.P.M., $V_a/u = {\cdot}44$, NOM. RIG PRESSURE = 2 ATMOS. SHOWING ROTATING STALL AT APPROX. 35% ROTOR SPEED.

(NOTE:- IN RECORDS (b) & (c) TIMING SIGNAL IS 1,000 PULSES PER SEC. WITH 1 PULSE PER 2 ROTOR REVS. SUPERIMPOSED.)

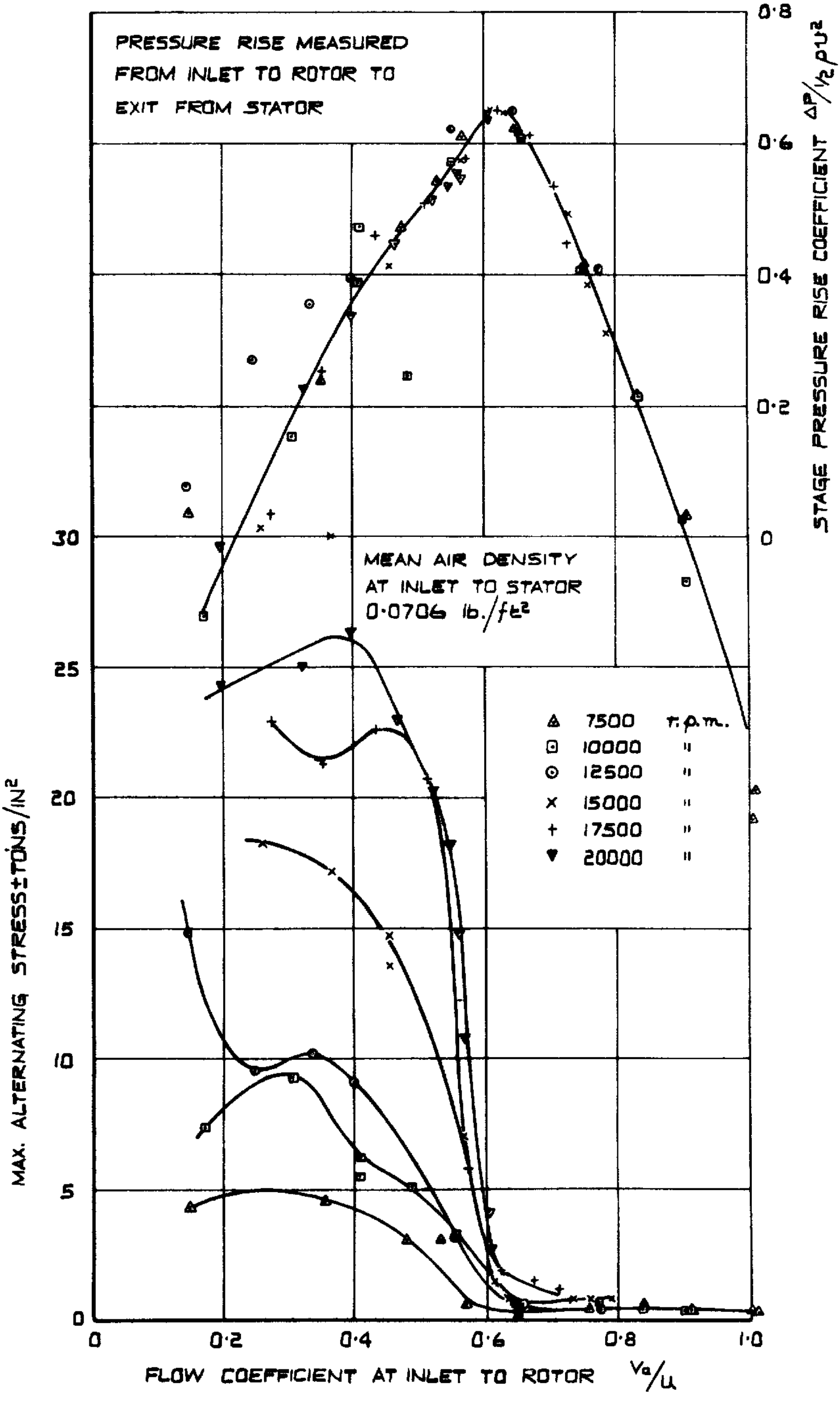
FIG.5



STAGE CHARACTERISTICS AND STRESSES

(1/4 ATMOSPHERE NOMINAL RIG PRESSURE.)

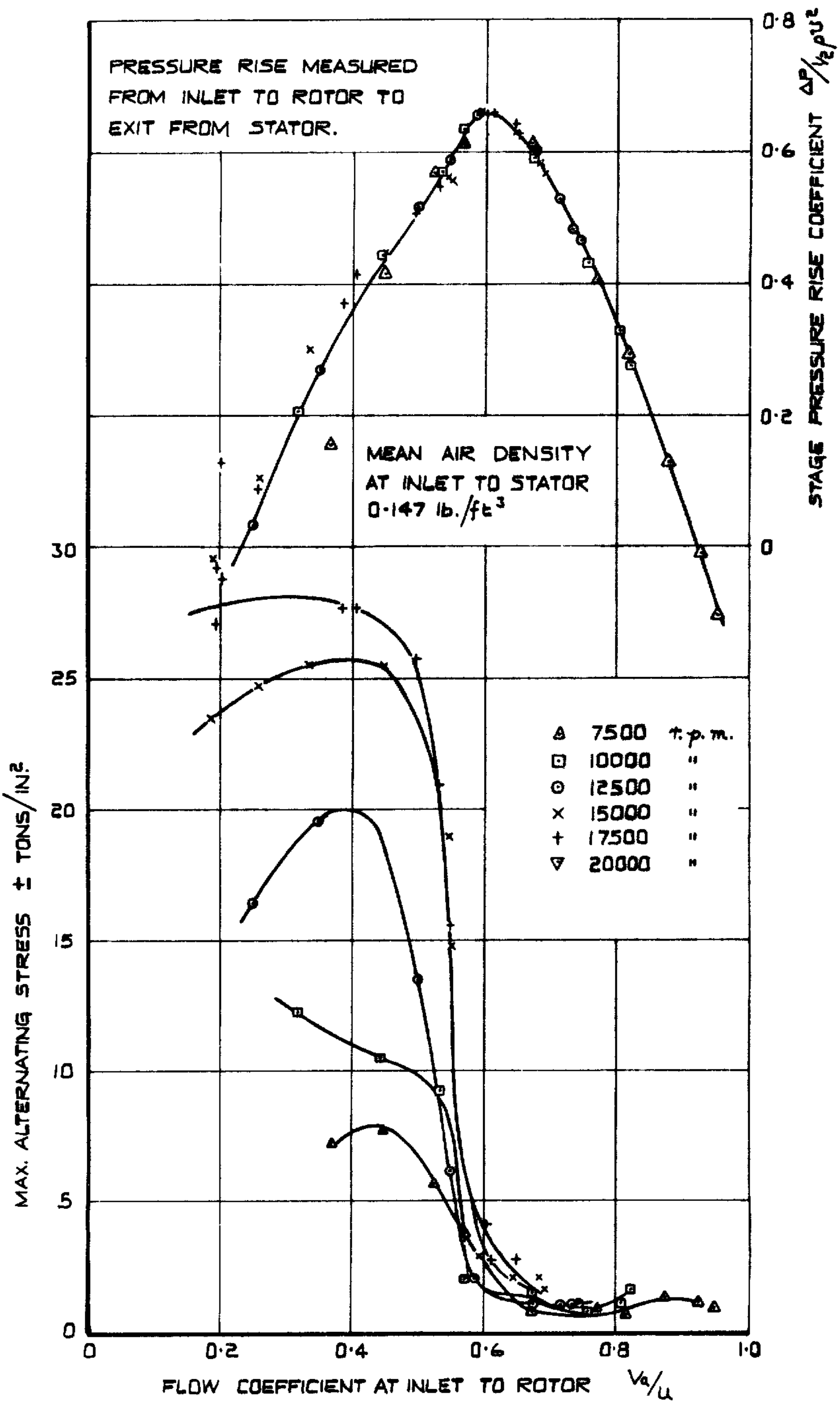
FIG. 6



STAGE CHARACTERISTICS AND STRESSES

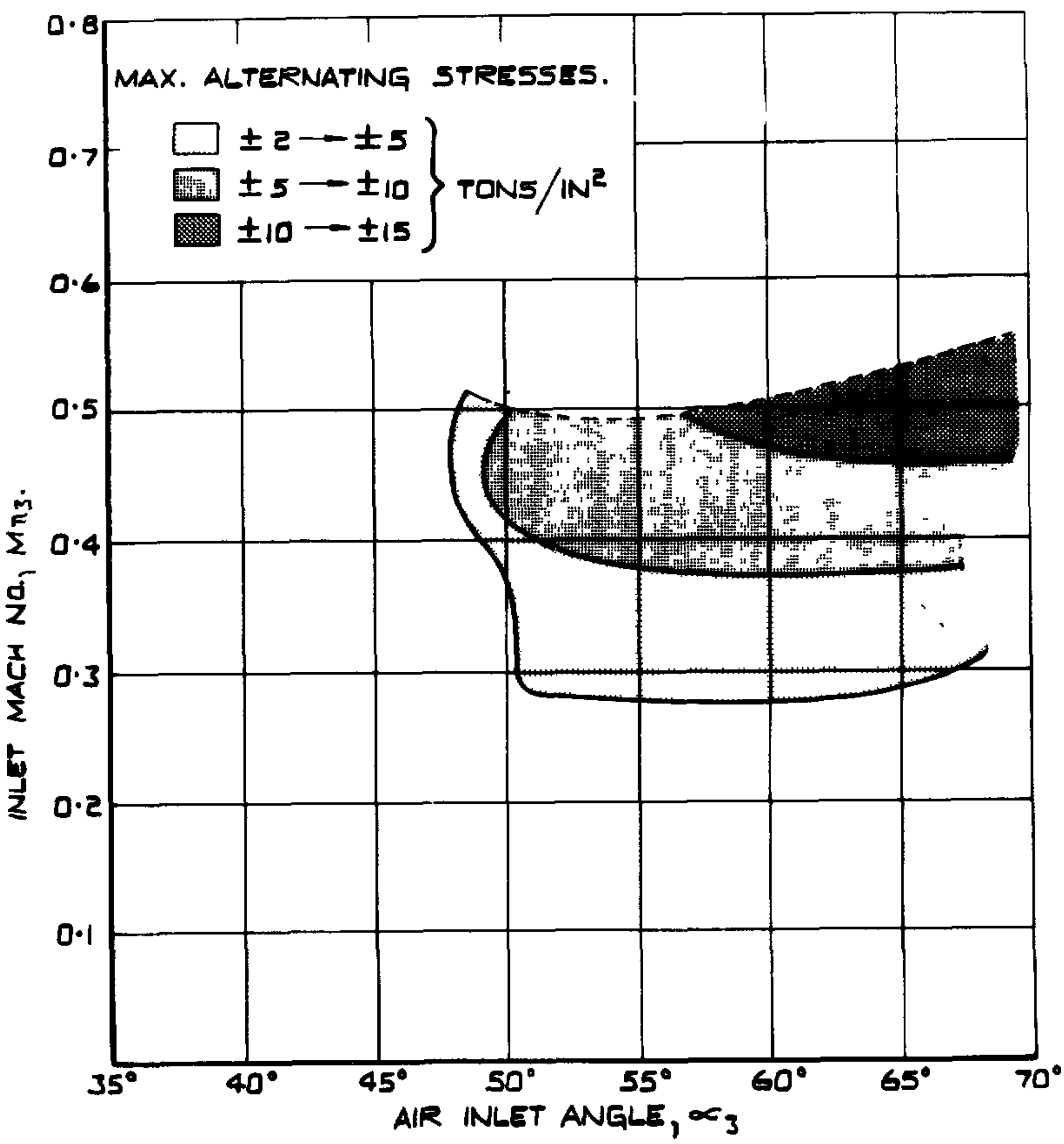
(1 ATMOSPHERE NOMINAL RIG PRESSURE.)

FIG. 7



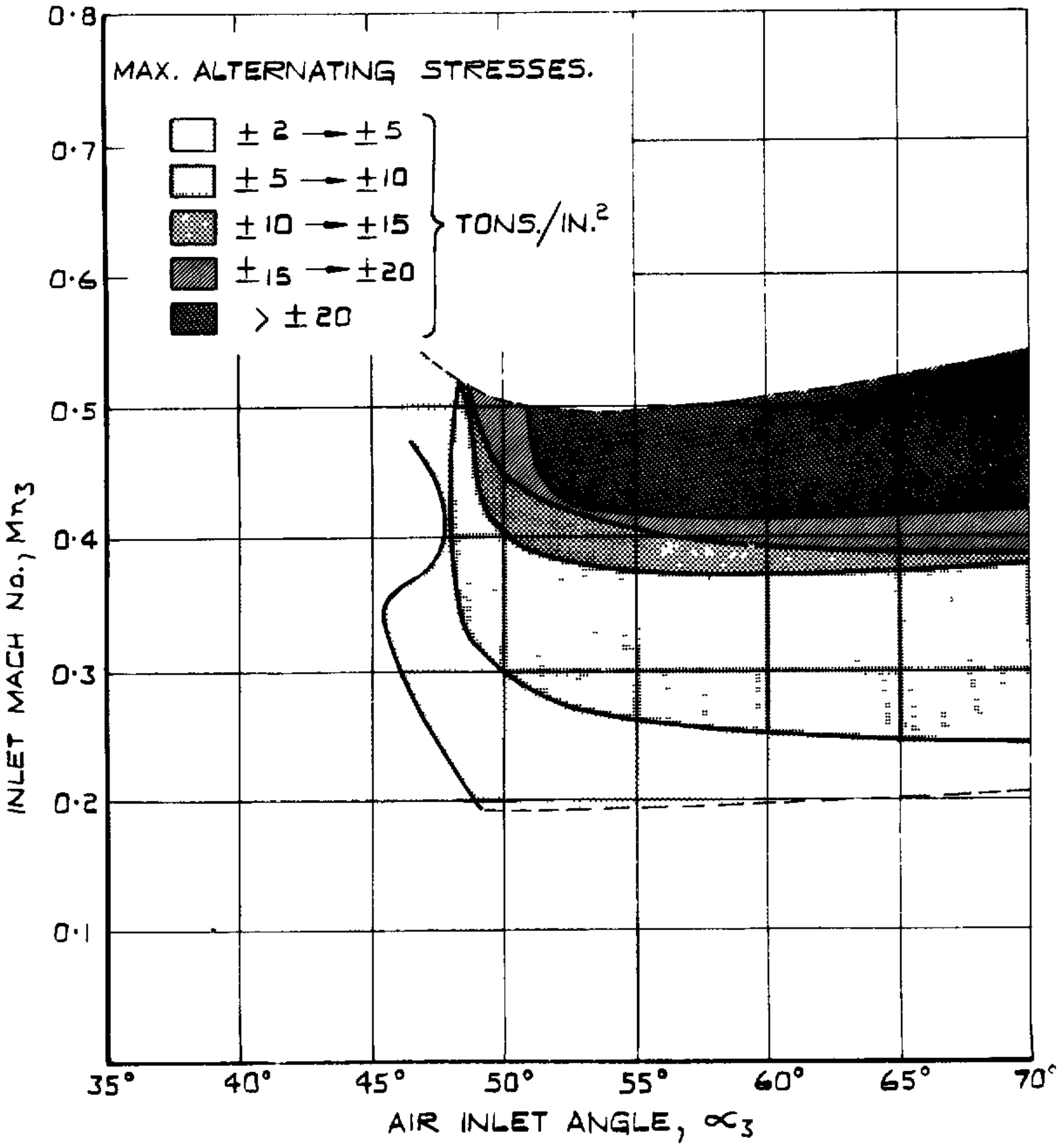
STAGE CHARACTERISTICS AND STRESSES

(2 ATMOSPHERES NOMINAL RIG PRESSURE.)



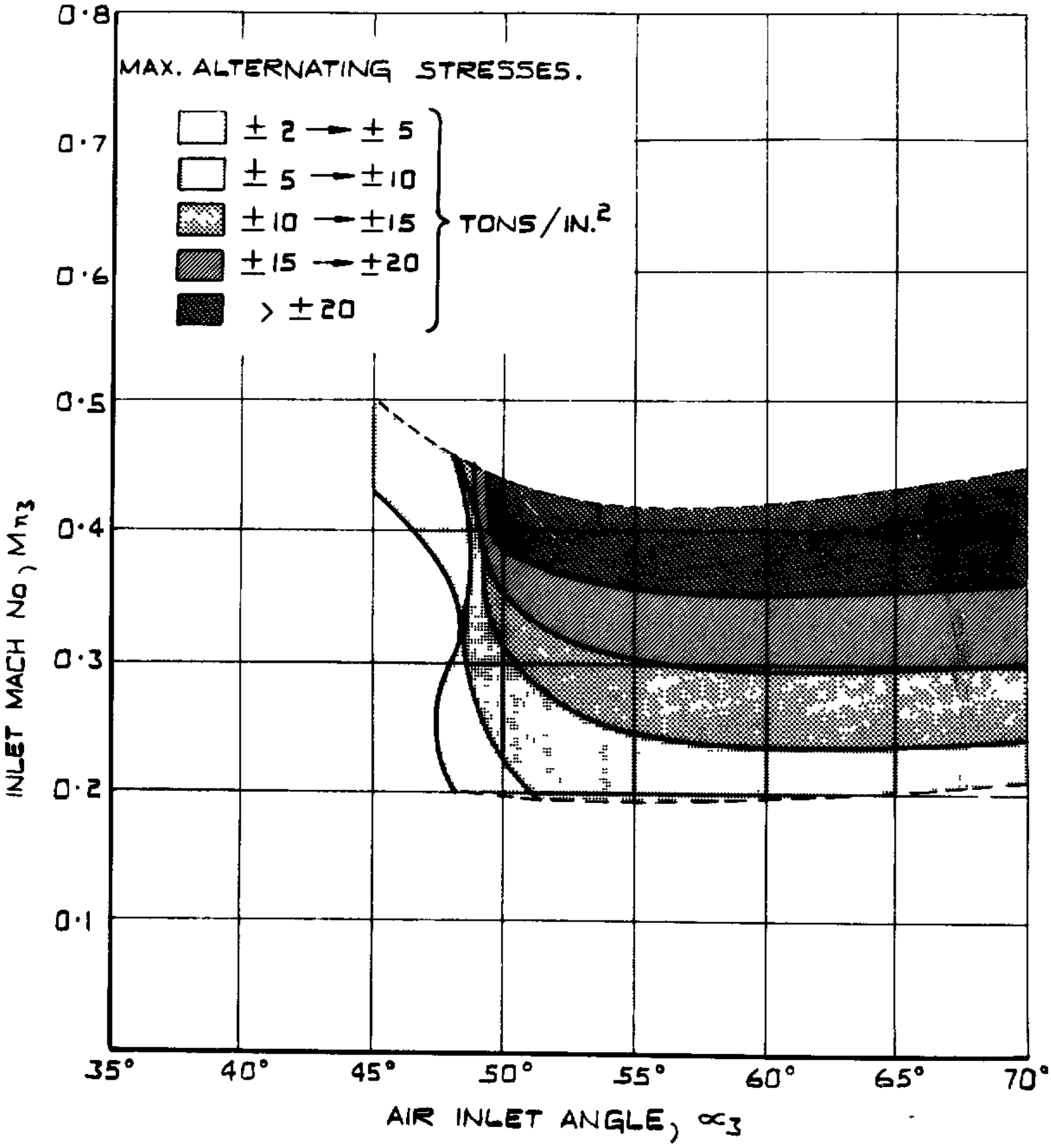
STRESS CONTOURS WITH STATOR INLET AIR PARAMETERS, (¼ ATMOS.)

FIG. 9

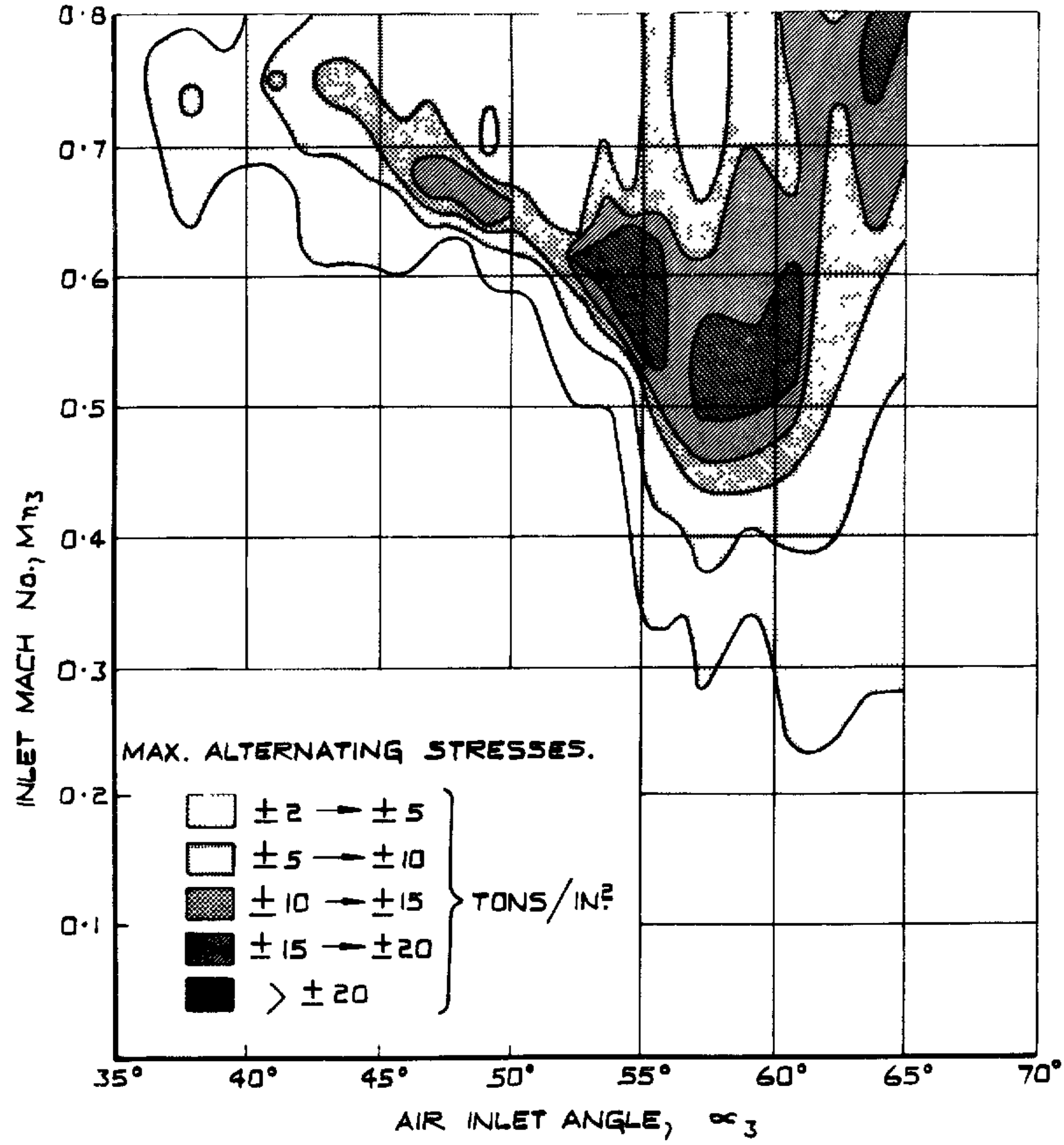


STRESS CONTOURS WITH STATOR INLET AIR PARAMETERS, (1 ATMOS)

FIG. 10



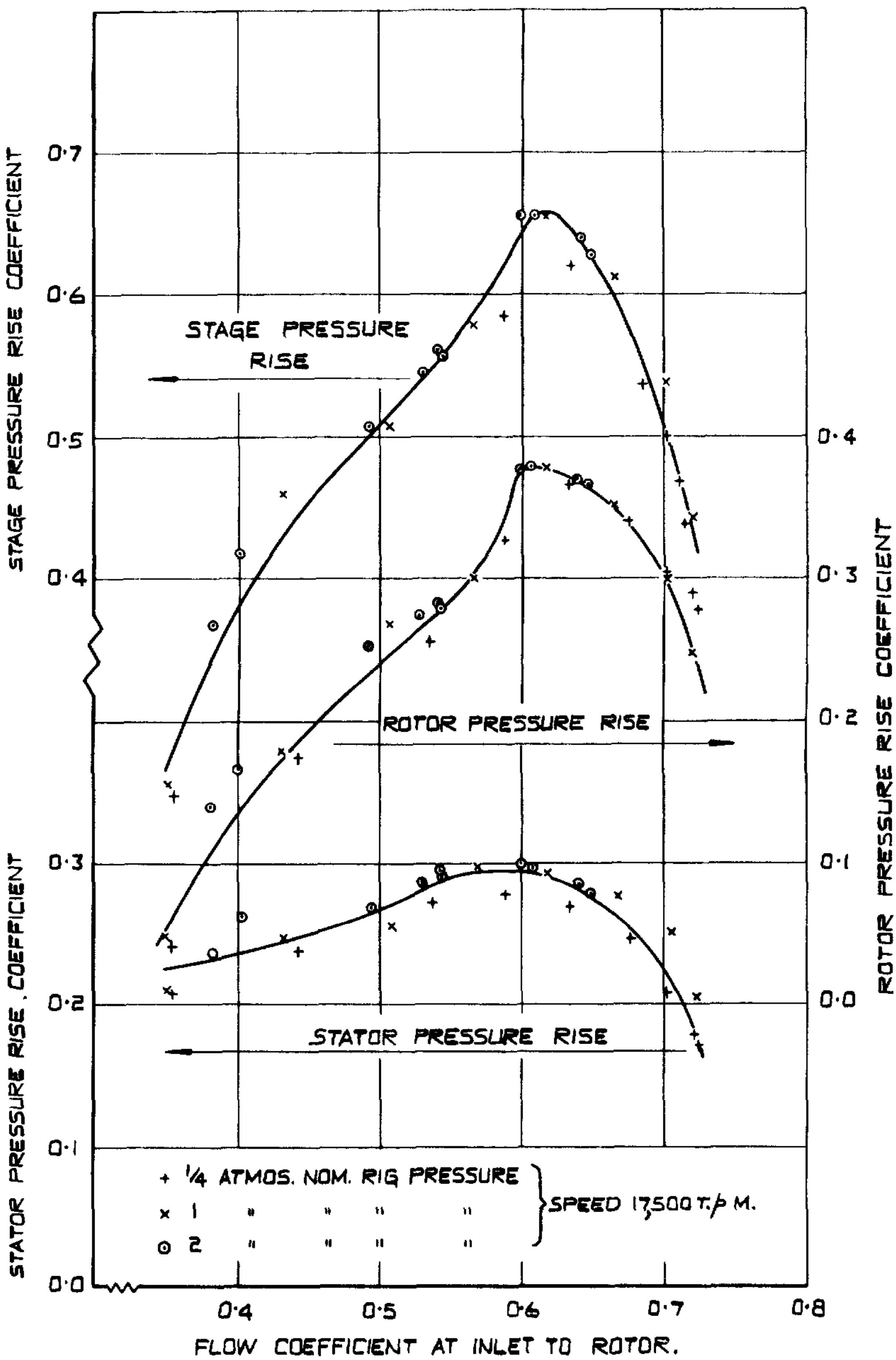
STRESS CONTOURS WITH STATOR INLET AIR PARAMETERS, (2 ATMOS.)



STRESS CONTOURS FROM CASCADE TESTS

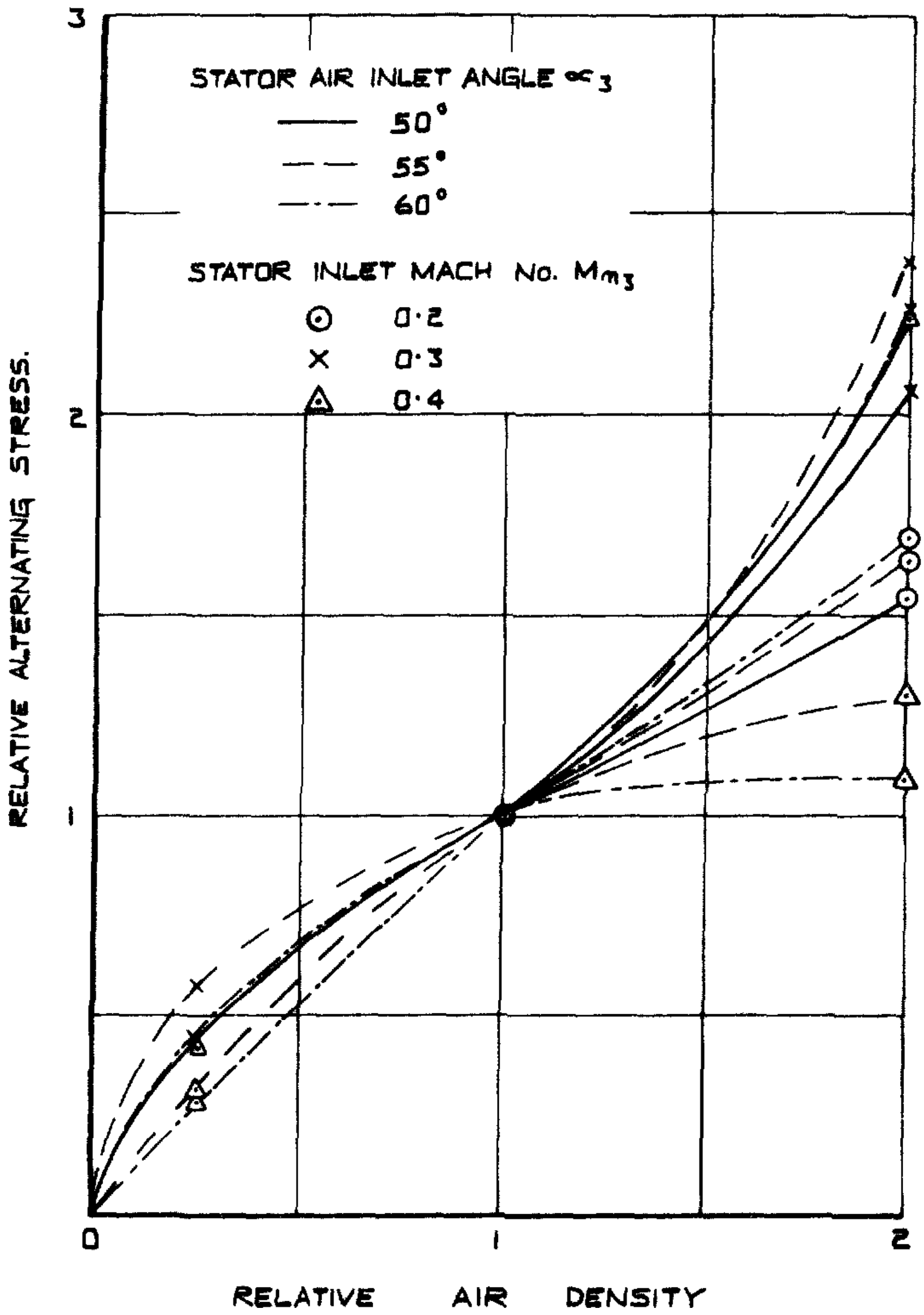
(1 ATMOS.)

FIG. 12



COMPARISON OF PRESSURE RISE COEFFICIENTS

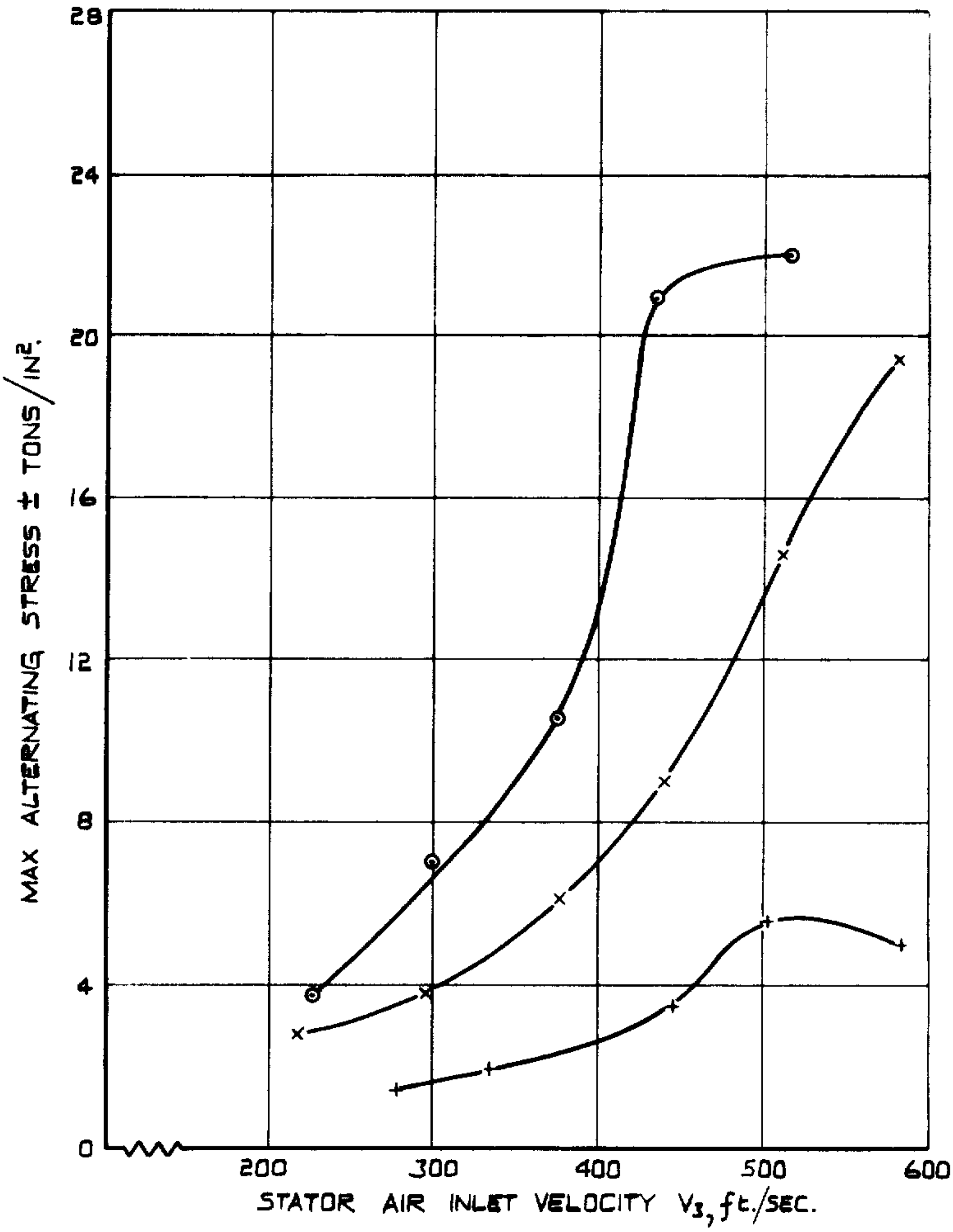
FIG. 13



POINT (1, 1) RELATES TO CONDITIONS AT NOMINAL RIG PRESSURE = 1 ATMOSPHERE.

SELECTED STRESS-DENSITY CURVES.

FIG. 14

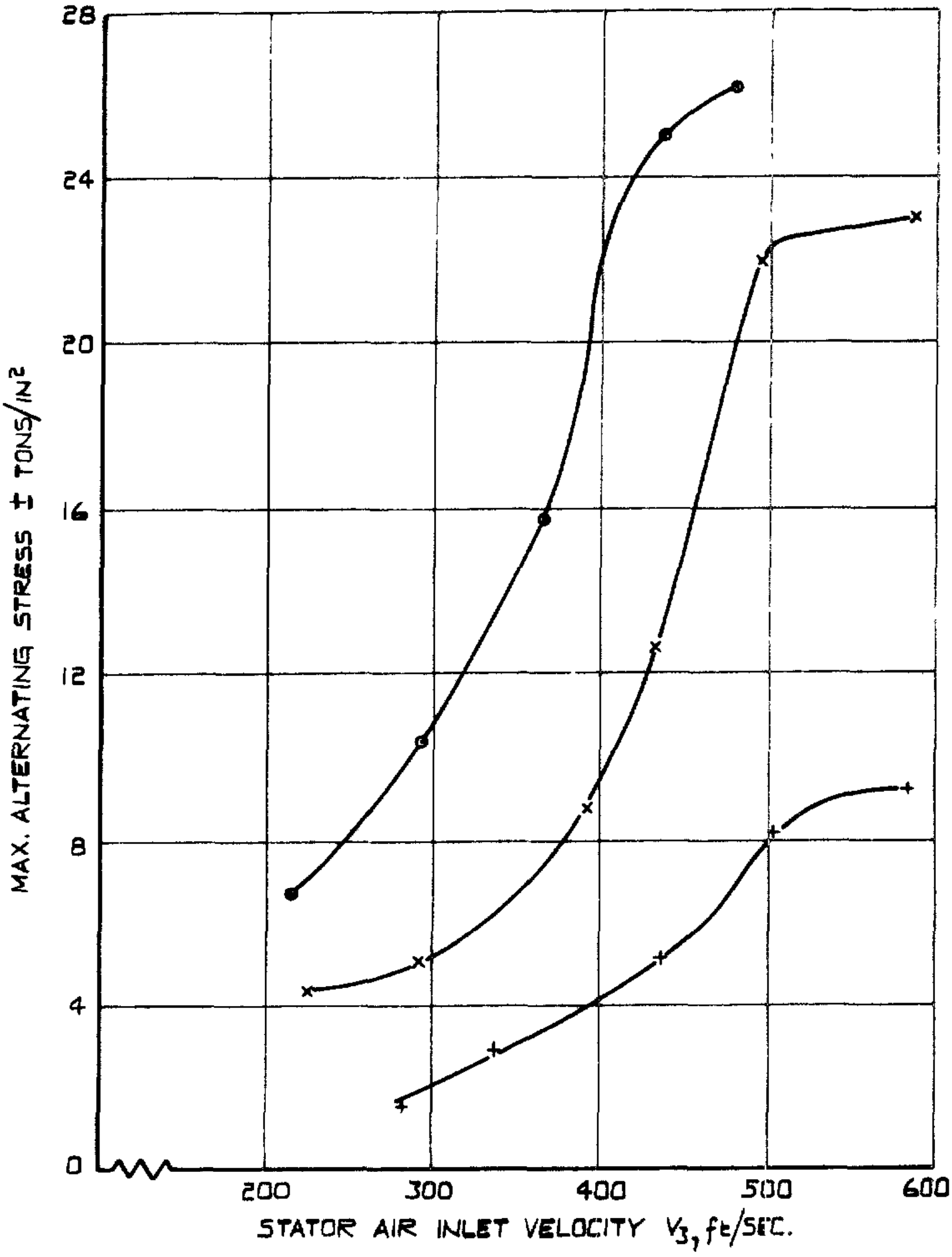


	NOMINAL RIG PRESSURE (ATMOSPHERES)	MEAN AIR DENSITY AT STATOR INLET. (lb/ft³)
+	1/4	0·0175
×	1	0·0706
⊙	2	0·147

STRESS - VELOCITY CURVES

(STATOR INLET ANGLE α_3 = 50°)

FIG. 15

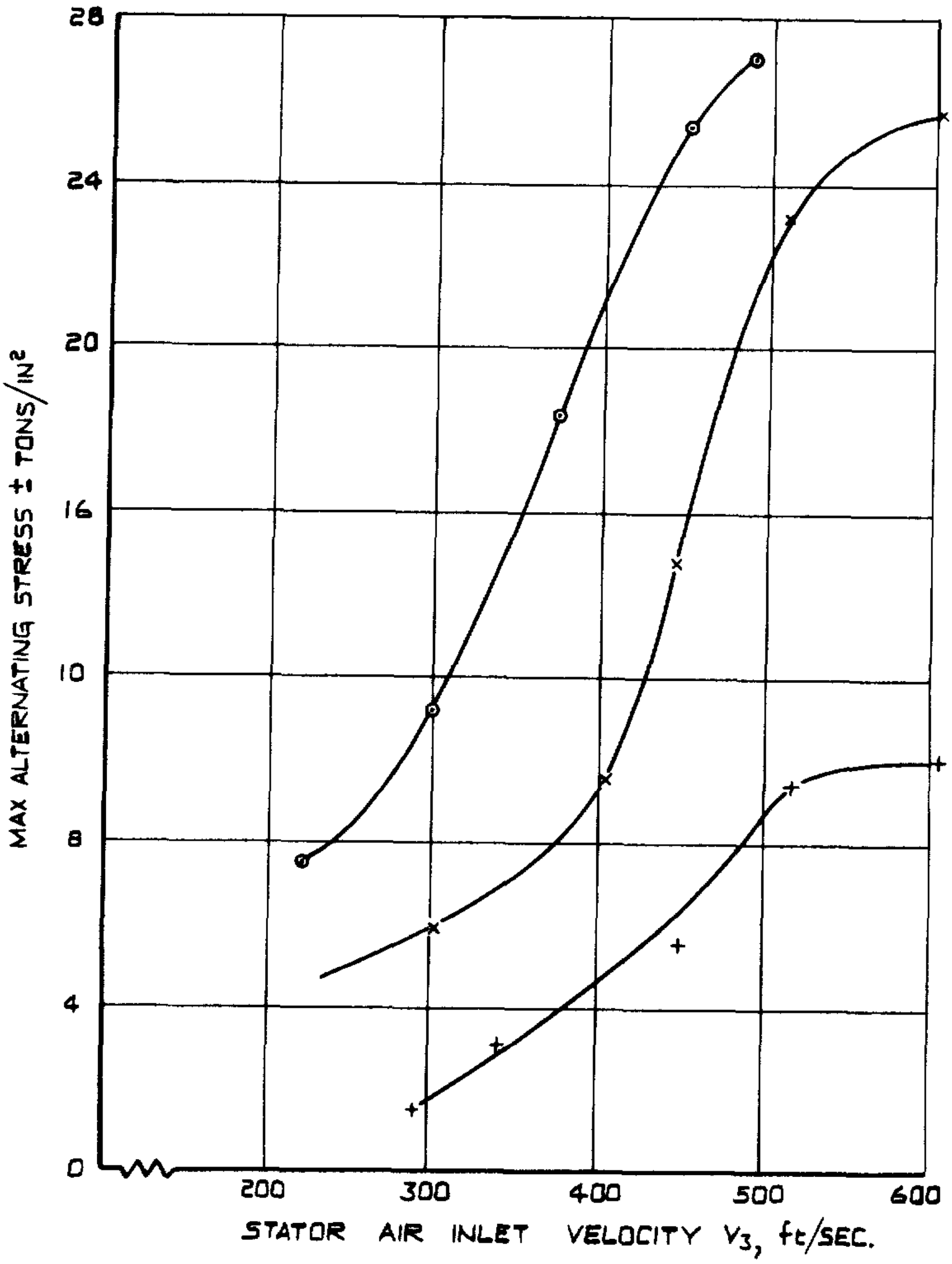


	NOMINAL RIG PRESSURE (ATMOSPHERES)	MEAN AIR DENSITY AT STATOR INLET (lb/ft³)
+	1/4	0·0175
×	1	0·0705
⊙	2	0·147

STRESS – VELOCITY CURVES

(STATOR INLET ANGLE $\alpha_3 = 55^\circ$)

FIG. 16

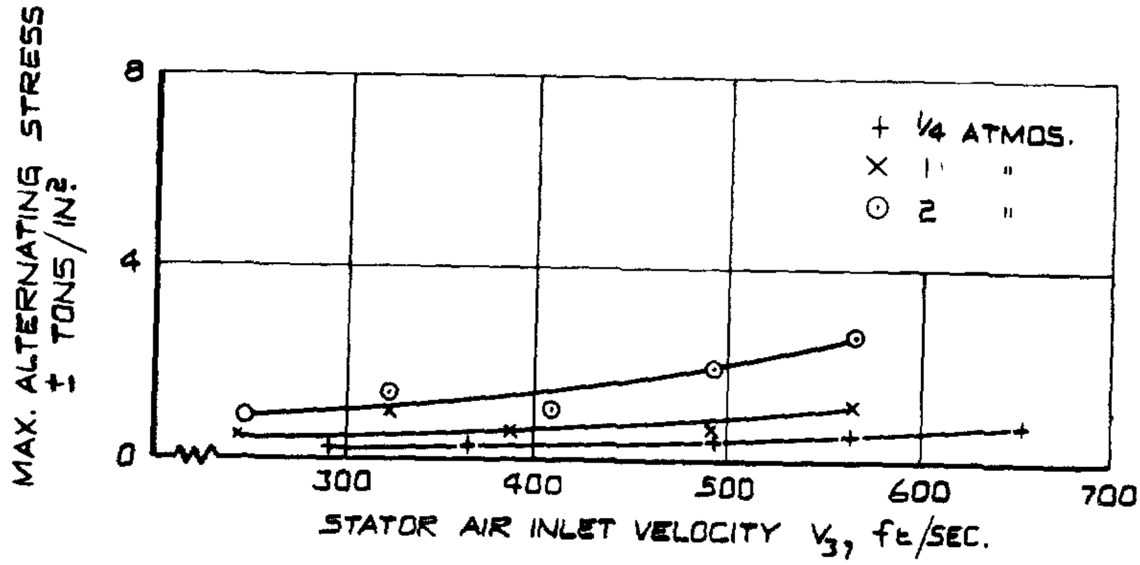


	NOMINAL RIG PRESSURE (ATMOSPHERES)	MEAN AIR DENSITY AT STATOR INLET (lb/ft^3)
+	1/4	0·0175
×	1	0·0706
⊙	2	0·147

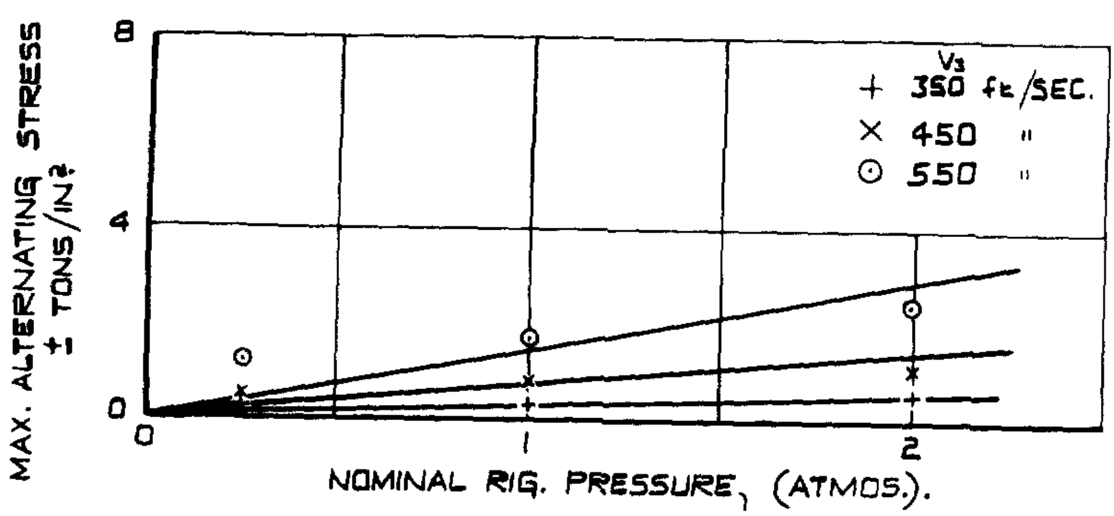
STRESS - VELOCITY CURVES

(STATOR INLET ANGLE $\alpha_3 = 60°$)

FIG. 17



(a) STRESS v VELOCITY.



(b) STRESS v DENSITY

BUFFETING STRESSES.

(STATOR INLET ANGLE $\alpha_3 = 45°$)

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