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The Effect of Lacing Wire on Axial Compressor Stage Performance at Low Speeds

By

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SUMMARY

Four stages of Free Vortex blades were tested at low speed in the 106 compressor to determine the effects of lacing wire upon stage efficiency and temperature rise.

With lacing wire of 0.14 in. diameter in the rotor and stator, and with blades of height $2\frac{1}{2}$ in., the loss in stage efficiency is about 12 per cent, but if 0.14 in. diameter tubular lacing is deformed within the blade pitch to a more streamlined section of $2\frac{1}{2}$:1 fineness ratio, the drop in stage efficiency would be about 5 per cent.

It is estimated that for the circular wire 30 per cent of the increase in loss is due to the wire drag itself and 70 per cent is due to interference by the wire in the normal blade flow.

CONTENTS

ILLUSTRATIONS

1.0 Introduction

When the problem of aerodynamic buffeting and subsequent blade failure arose in axial compressors, lacing was considered as a possible solution and an investigation to determine the aerodynamic effects of lacing wire at low speeds was initiated.

The most likely mode of blade vibration to produce failure is the fundamental in which there is a maximum amplitude at the blade tip decreasing to zero at the root, and to effectively damp this vibration the lacing wire must be near the tip. The dimensions of the lacing wire in the rotor blades are governed by the bending forces imposed on the short length of wire between blades. With increasing radius or blade speed the strength required in the lacing becomes greater and the projected area of the wire in the direction of flow is almost bound to increase. The wake produced by the wire in this direction contributes to the losses in the compressor stage.

A further consideration which can adversely affect the compressor performance is the chordwise position of the lacing wire relative to the blade. The most robust form of lacing as far as the blade is concerned is probably that in which the lacing wire passes through the thickest part of the blade section. Chordwise this position is very near the throat of the blade passage, and the wire will considerably reduce the efficiency of the diffusion process downstream of the throat. Attaching the wire to the trailing edge would probably reduce this effect to a minimum but is not a very satisfactory method mechanically.

In view of these considerations, the lacing of the rotor blades for the tests reported here was at the point of maximum thickness of the blade and two thirds of the blade height from the root. This position represents a compromise between lacing at the rotor tips for maximum damping and the other extreme of using the lacing wire as near the inner diameter as possible for minimum stresses. The stator blades were laced near the tip at the point of maximum thickness of the section.

The purpose of the tests was to find the low speed aerodynamic effects of three types of lacing wire in rotor blades and stator blades and to seek an explanation of these effects by means of detailed flow investigations between blade rows. In the analysis of the results an attempt is made to distinguish quantitatively between losses due to the lacing wire wake itself and the secondary effects of the wire on the flow over the blades.

2.0 The compressor and lacing techniques

2.1 Mechanical details

An illustration of the 106 Compressor, Ref. 1, and some details of the Free Vortex blades with which the compressor was fitted for these investigations are shown in Fig. 1. The aerodynamic design of the blades is given in Appendix 1. Four stages of blades were used at a spacing of $2/3$ chord between rows. The lengths of lacing wire were threaded through holes drilled in the blades, each length being sufficient to span ten blades. At the ends of each length the wire was either soft soldered or deformed to keep it in position, but it was not attached to each individual blade. The three forms of lacing wire used in the tests were:-

(1) Stranded cable 0.124 in. diameter made up of bundles of 20 strands of 0.009 in. wire twisted together.

(2) Smooth tube 0.143 in. diameter.

(3) The same tube pressed into approximately elliptical form between blades. Major axis 0.196 in. Minor axis 0.074 in. Subsequently called 'flattened tube'.

The cable is not a practical form of lacing but it was used in the rotor for these low speed tests because of the difficulty of lacing with stiff wire or tube without disturbing the rotor blades. The stator blades were much easier to lace because the casing is in halves, so the stators were laced successively with cable, circular tube and flattened tube. A performance investigation was made for each arrangement. The inlet guide blades were not laced for any of the tests.

All the blades were drilled to accept the lacing wire before assembly into the compressor so that they would not have to be subsequently moved. The performance of the unlaced compressor was found by testing the compressor with the lacing holes filled with wax and this performance was used as a basis for comparison.

2.2 Order of testing

The order of testing for the various arrangements of lacing was:-

(1) Four stage compressor with drilled but wax filled lacing holes.

(2) Four rotor rows laced with cable, stator rows unlaced.

(3) Four rotor rows laced with cable, four stator rows laced with cable.

(4) Four rotor rows laced with cable, four stator rows laced with circular tube.

(5) Four rotor rows laced with cable, four stator rows laced with flattened tube.

3.0 Aerodynamic properties of the lacing wire

The drag coefficient for the three forms of lacing wire shown in Fig. 2 were measured at a Reynolds number of 0.9×10^4 and are:-

	C_D
Stranded cable 0.124 in. O.D.	1.215
Smooth circular tube 0.143 O.D.	1.06
Flattened tube 0.196 in. 0.074 in. (based on minor axis)	0.47

The Reynolds number of the lacing wire in the compressor was about 1.0×10^4. Ref. 3 gives a drag coefficient (C_D) of 1.1 for a smooth cylinder at this Reynolds number and at a Reynolds number of 20×10^4 the value of C_D is 0.4. Excluding Mach number effects therefore the drag coefficient for a circular lacing wire in a compressor may be anywhere between these two values depending on the design. There is a similar variation in the drag coefficient for an elliptical tube so the results are directly applicable only for the drag coefficient values quoted.

In the compressor the velocity normal to the axis of the wire varies along the length of wire in one blade pitch. For ease of definition therefore the relevant velocity for the expression of drag coefficient and Reynolds number is taken to be the mean axial velocity corrected by a simple ratio for blade blockage in the annulus at the plane of the lacing wire.

$$\text{Velocity of approach to wire} = \frac{\bar{V}_a \times \text{Annulus Area}}{\text{Annulus Area} - \text{Blockage Area}}$$

4.0 Compressor characteristics

The overall compressor characteristics are shown in Fig. 3. They are plotted as Mean Stage Characteristics i.e. the overall pressure and temperature coefficient divided by the number of stages. The non-dimensional pressure coefficient is expressed in terms of the measured total pressure rise, the density based on the mean of total conditions at inlet and outlet, and the mean blade speed. The 'total' density used in this expression is more convenient to use than the true mean density both for computation and conversion should this necessary.

The efficiency is the isentropic efficiency derived from pressure ratio, mass flow, torque and rotational speed measurements.

The tests were carried out at a speed of 3,000 r.p.m. giving a mean blade speed of 229 ft./sec. and a Reynolds number based on this speed of 1.3×10^5.

4.1 Mean stage performance

One of the comparative tests carried out was that of rotor laced with cable and stator laced first with cable then with circular section tube. The tube had, as has been stated a somewhat larger diameter and in tunnel tests the drag of these two was about the same.

	C_D	d	$C_D \times d$
Cable	1.215	0.124	0.151
Tube	1.06	0.143	0.152

There was in fact no difference in compressor performance with these two forms of lacing. It is assumed therefore that the two are equivalent and in presenting the results they will be considered so and called circular section lacing wire as opposed to flattened tube.

The variation in mean stage performance of the compressor with circular wire and flattened tube lacing is summarised in Fig. 3. Lacing both rotor and stator with circular wire reduces the maximum stage efficiency by 12 per cent and the work done at a flow coefficient of 0.60 by about 8 per cent. Lacing the rotor only with circular wire the reduction in maximum efficiency is $7\frac{1}{2}$ per cent and of the work done 3 per cent. This reduction in efficiency and work done is prohibitive if the majority of the stages in a compressor have to be laced but if the blade rows in which failure is likely can be identified and are few in number then lacing would probably be acceptable. A reduced loss in efficiency can be obtained by lacing both rotor and stator with the more streamlined flattened tube. The test series did not include this arrangement as such because only the stator blades could be laced conveniently with the tube but an estimate of the probable performance is shown in Fig. 3. The reduction in maximum stage efficiency is about 5 per cent. This figure is still comparatively large and the explanation may be that the wire is flattened between the blades only, and retains its circular section near to the blade surface. Under these circumstances the interference by the wire on the blade boundary layer flow will be of the same degree as for the circular section wire.

4.2 Stage characteristics

The individual stage static pressure characteristics for the four stages of each lacing arrangement are shown in Figs. 4 and 5. The values for the pressure coefficient at a particular flow are about the same as for the Mean Stage characteristics Fig. 3.

The unlaced compressor has a first stage performance which is better than that of the three following stages. This effect has been noted previously in Ref. 4 and is attributed to the smallness of the annulus boundary layer at this stage. Lacing the rotor blades with circular section wire gives a set of characteristics which are almost coincident at a flow coefficient of 0.6. The lacing in the first stage rotor therefore cancels any benefit obtained from a good velocity profile but lacing all the rotor rows does not lead to any stage to stage deterioration in performance. The stage characteristics when both rotor and stator blades are laced do however indicate a steady deterioration in performance from stage one to four at low flow coefficients.

4.3 Flow investigation at the fourth stage

An investigation of the flow over one blade pitch was carried out half a chord downstream of the fourth stator for each lacing arrangement. The investigation consisted of radial traverses with yaw angle, total pressure and static instruments from outer diameter to inner diameter for a series of circumferential positions over the area of one blade pitch. By removing the stator ring and replacing it by an unbladed ring a similar traverse was obtained for the fourth stage rotor. The traverse plane was therefore one and a half chords downstream of the rotor blades and half a chord downstream of the stator blades.

The results obtained are shown in Figs. 6, 7, 8 and 9. In Figs. 6 and 7 are the fourth stage rotor gas angles and axial velocities and in Figs. 8 and 9 the fourth stage stator angles and axial velocities.

Considering first the fourth stage of the unlaced compressor as a basis for comparison, the rotor blades have gas outlet angles which correspond closely to the predicted values (see Appendix I) over most of the blade height. The gas angles at outlet from the stator blades are about one and a half degrees higher than the predicted values, the incidence in both rotor and stator being well below the stalling incidence.

Introducing lacing wire into the rotor blades at two thirds blade height from the root causes the outlet angle to increase in the rotor blade row. The increase amounts to about 7 or 8 degrees and extends over most of the outer half of the annulus.

The lacing wire also distorts the axial velocity profile so that the incidence on the following stator is above the stalling incidence. The unlaced stator is stalled therefore, over a proportion of the blade length which is almost as great as that for the laced rotor blades themselves.

Lacing the stator blades also produces a high deviation in both rotor and stator at the lacing wire radius. This radius is however near the inner diameter so the effects upon inlet and outlet angle are less extended, being more like those produced by a thick annulus boundary layer. Figs. 7 and 9 show the axial velocity profiles downstream of the fourth stage stator and rotor blade rows when all the rotor and stator blades are laced with circular wire. The profile downstream of the stator blade row shows a considerable thickening of the inner annulus boundary layer at this position but not apparently after the rotor blades. If the performance of the fourth stage is similar to the performance of the other three stages, as the stage characteristics suggest, there must be considerable radial flow between blade rows.

The circumferential mean loss coefficient distribution for a blade in the fourth stator row is shown in Fig. 9, also for comparison the two dimensional wake of the wire obtained in a wind tunnel at about the same Reynolds number. It is obvious that the losses associated with the lacing are much greater than the pure wake effects of the wire. It will be shown that the interference losses due to the effects of the wire on the flow over the blade constitute about 70 per cent of the total additional losses due to lacing in the stator blades.

5.0 Some comments on the origin of losses due to lacing wire

Considering first the circular section lacing wire. If this is placed in an annulus with some other blockage effect the pressure loss is

$$\left(\frac{\Delta\bar{\omega}}{\frac{1}{2}\rho U_m^2}\right)_{wire} = C_D.\frac{d}{h}\left(\frac{k.V_a}{U_m}\right)^2$$

where k is the blockage factor.

$$k = \frac{\text{Annulus Area}}{\text{Annulus Area - Blockage Area}}$$

In this instance the rotor blade blockage k_R is 1.16 and the stator blade blockage k_S = 1.205.

The figure for the estimated pressure loss due to drag of the circular section wire can be compared with the experimental values in Fig. 3.

Fig. 3 (V_a/U_m = 0.54)

$\Delta P/\frac{1}{2}\rho U_m^2$ Unlaced	=	0.701
$\Delta P/\frac{1}{2}\rho U_m^2$ Rotor and Stator laced with circular wire	=	0.560
Pressure loss due to lacing	=	0.141
Estimated pressure loss due to lacing wire wake in rotor and stator	=	0.051

The wire drag therefore contributes about one third of the total increase in loss. A similar estimate was made from the loss curves in Figs. 8 and 9 and the figures showed that a little more than a quarter of the total additional loss due to lacing was contributed by the wake of the lacing wire itself. Approximately therefore 30 per cent of the drop in efficiency is the drag effect of the wire and 70 per cent is interference with the flow over the blade surface.

Equivalent figures for the flattened tube are 13 per cent drag and 87 per cent interference effects. This is probably because the wire was flattened between the blades only and at the blade surface the section of the wire remained circular. The interference losses would therefore not be reduced in the same proportion as the wire drag losses.

6.0 Conclusions

With lacing wire of 0.14 in. diameter in the rotor and stator blades of a compressor stage with a blade height of 2.5 in., the maximum efficiency of the stage is reduced by 12 per cent. If the 0.14 in. diameter tube is deformed within the blade pitch to a more streamlined shape of $2\frac{1}{2}$:1 fineness ratio the drop in efficiency is estimated to be only 5 per cent which may be acceptable if only a proportion of blade rows in a multistage compressor are laced.

The test Reynolds number of the lacing wire was 1×10^4 corresponding to a C_D of 1.1 for the circular wire. At higher Reynolds numbers the C_D reaches a minimum of 0.4 and in these circumstances the effect of lacing wire on the compressor efficiency would be reduced. The same is true for the flattened wire.

In the compressor tested the lacing in a blade row produced very high gas angle deviations from the blades in that row and also from the unlaced blade row downstream. The increase in deviation caused by circular lacing in both rotor and stator blade rows caused a reduction of 8 per cent in work done at a flow coefficient of 0.6.

Of the increase in losses due to lacing with circular section wire 30 per cent is contributed by the wake from the wire and 70 per cent by the interference of the wire with the flow over the blades.

NOTATION

U	=	Blade speed
V_a	=	Axial velocity
r	=	Radius
ΔP	=	Total temperature rise
ΔT	=	Total pressure rise
ρ	=	Density
s	=	Blade pitch
c	=	Blade chord
ε	=	Gas deflection angle
ξ	=	Blade stagger angle measured from axial direction
α	=	Gas angle measured from axial direction
β	=	Blade angle measured from axial direction
Ω	=	Work done factor
$\overline{\omega}$	=	Mean total pressure loss
d	=	Diameter of lacing wire
η	=	Isentropic efficiency
h	=	Blade height

SUFFICES

0	=	After stator row of previous stage
1	=	Before rotor blade row
2	=	After rotor blade row
3	=	Before stator blade row
4	=	After stator blade row
m	=	Mean annulus diameter
M	=	Mean of compressor inlet and outlet conditions
R	=	Rotor
S	=	Stator

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2	A. D. S. Carter	The low speed performance of related aerofoils in cascade. C. P. No. 29. September, 1949.
3	S. Goldstein	Modern developments in fluid dynamics. Oxford University Press.
4	S. J. Andrews and H. Ogden	A detailed experimental comparison of axial compressor blades designed for free vortex flow and equivalent untwisted and twisted constant section blades. R. & M. 2928. July, 1953

APPENDIX I

Design of Free Vortex blades

Assumptions

$$V_a = \text{constant}$$

$$U_m/V_a = 1.5$$

$$\Delta T/\tfrac{1}{2}U_m^2 = 0.8 \text{ including work done factor}$$

$$\Omega = 0.95$$

$$\text{Reaction} = 50 \text{ per cent at } r/r_m = 0.9$$

$$\tan\alpha_3 \propto 1/r$$

$$\tan\alpha_0 \propto 1/r$$

then:-

$$\frac{\Delta T}{\tfrac{1}{2}U_m^2} = 2\frac{V_a}{U}\frac{r}{r_m}(\tan\alpha_3 - \tan\alpha_0)\Omega$$

$$= 1.27\frac{r}{r_m}(\tan\alpha_3 - \tan\alpha_0)$$

and at the radius of 50 per cent reaction that is at $r/r_m = 0.9$

$$\tan\alpha_3 - \tan\alpha_0 = \frac{U_m}{V_a}\cdot\frac{r}{r_m} = 1.35$$

The design gas angles are therefore:

r"	7.5	8.12	8.75	9.37	10.0
r/r_m	0.86	0.93	1.00	1.07	1.14
α_1	43.6	47.3	50.4	53.1	55.5
α_2	12.3	22.0	30.0	36.6	42.0
α_3	47.0	44.7	42.7	40.8	39.0
α_4	18.6	17.3	16.2	15.2	14.3

APPENDIX I (Cont'd)

The rotor has 58 blades and the stator 60 so with blades of 1.14, 1.1 and 1.06 in. chord at root boss, mean and tip diameters respectively, the blade angles are:-

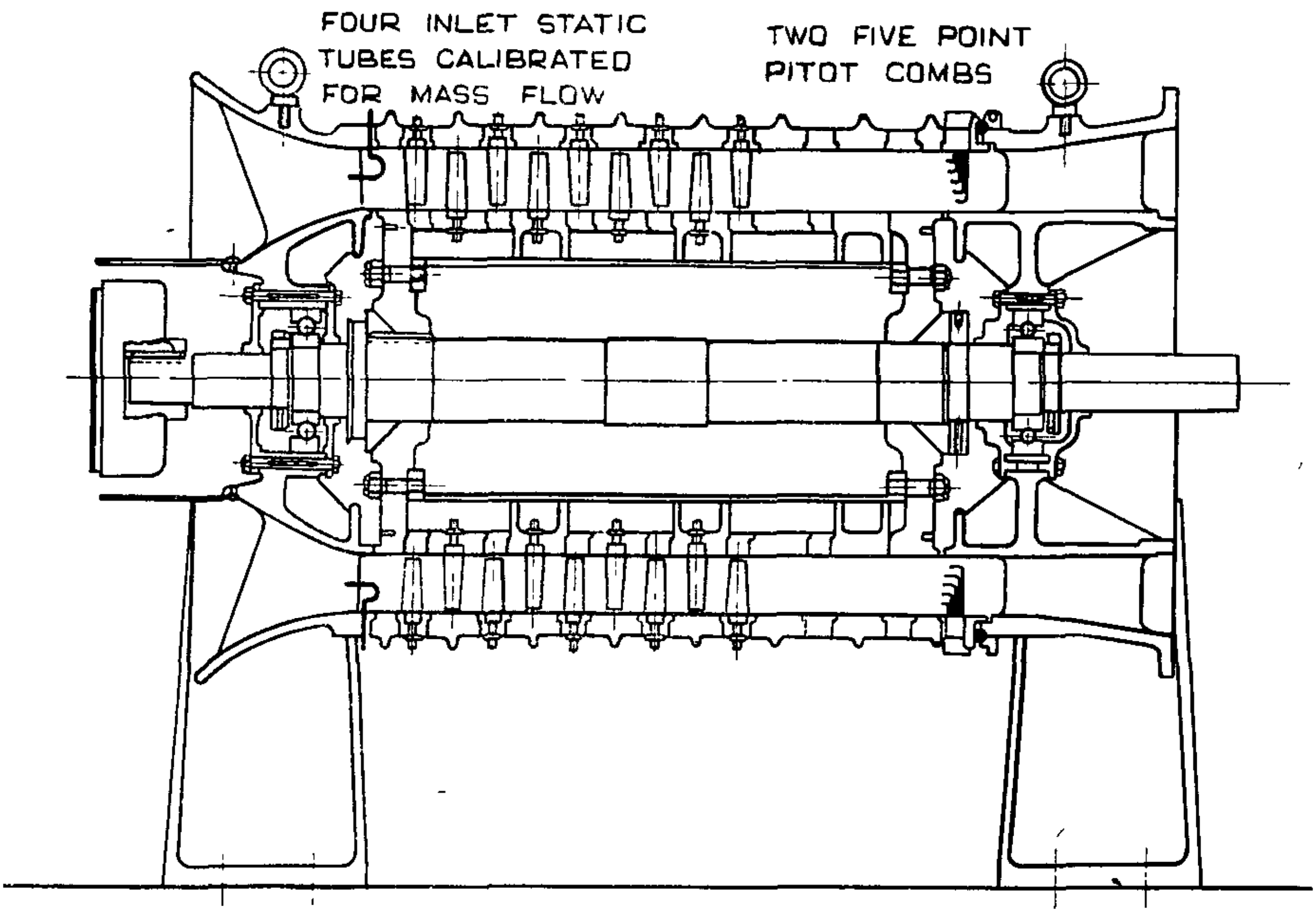
r"	7.5	8.12	8.75	9.37	10.0
β_1	43.9	47.4	49.5	51.1	52.3
β_2	2.9	13.7	23.0	31.0	37.4
θ	41.0	33.7	26.5	20.1	14.9
s/c	0.712	0.787	0.862	0.942	1.020
t/c	0.12		0.10		0.08
β_3	47.5	45.0	42.4	40.0	37.5
β_4	9.8	9.0	8.4	7.7	6.9
θ	37.7	36.0	34.0	32.3	30.6
s/c	0.740	0.788	0.833	0.876	0.918
t/c	0.10		0.11		0.12

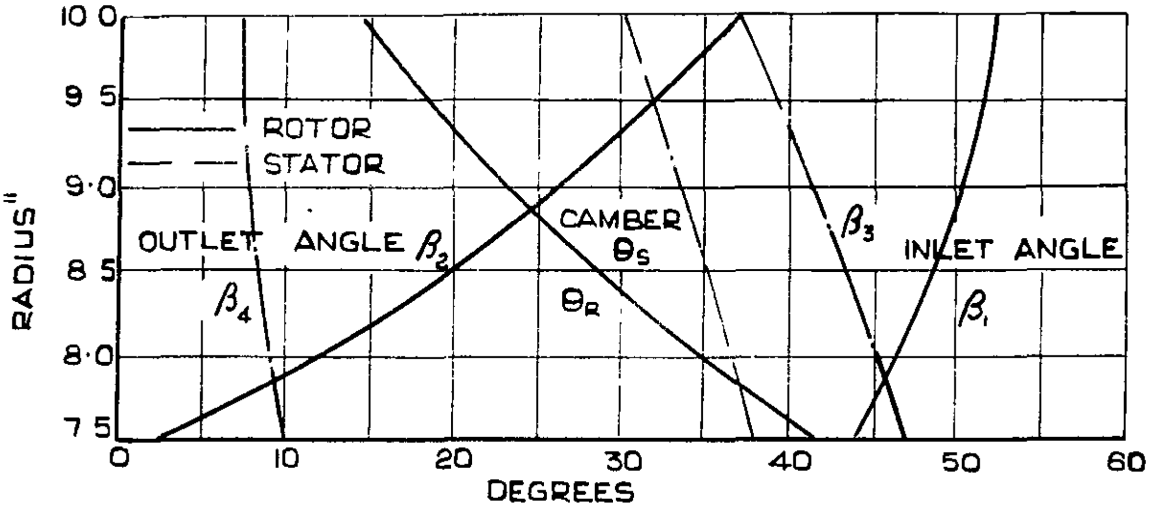
The blades have circular arc camber lines and are designed to the deviation rule $m\theta\sqrt{s/c}$ given in Reference 2.

The design details of the Inlet Guide Vanes are:-

r"	7.5	8.12	8.75	9.37	10.0
β_1	-1.1	-2.3	-3.7	-5.1	-6.4
β_2	-22.1	-20.6	-19.2	-17.8	-16.4
s/c	0.740	0.788	0.833	0.876	0.918
t/c	0.10		0.11		0.12
α_2	18.6	17.3	16.2	15.2	14.3

FIG I.





RADIUS	7·50"		8·75"		10·0"	
	R	S	R	S	R	S
CHORD	1·14	1·06	1·10	1 10	1 06	1·14
PITCH/CHORD	·711	738	862	·832	1·040	·918
THICK S/CHORD	·120	100	·100	·110	·080	·120
NUMBER BL'DS	58	60	58	60	58	60

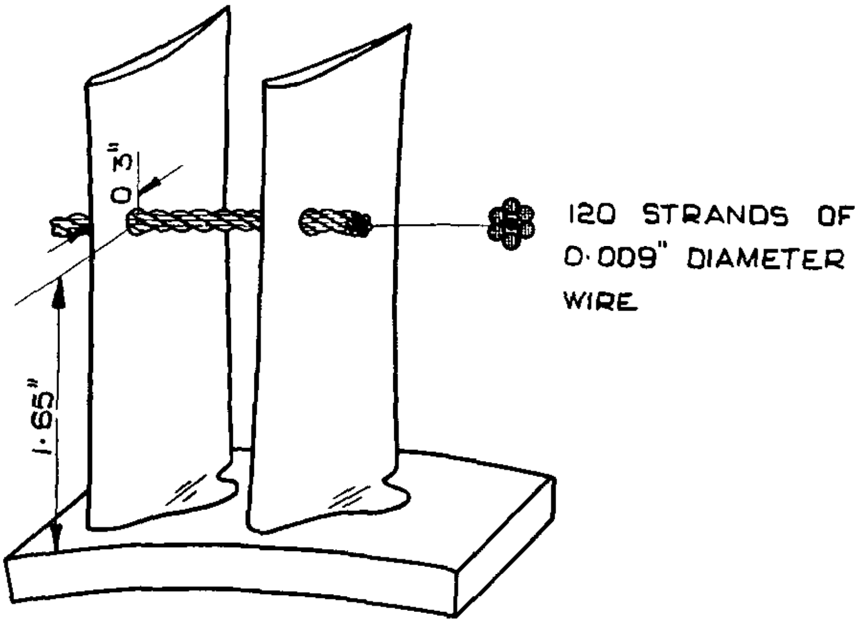
CAMBER LINE IS A CIRCULAR ARC:-

$\varsigma = \theta/2 - \beta_{1 \text{ or } 3}$

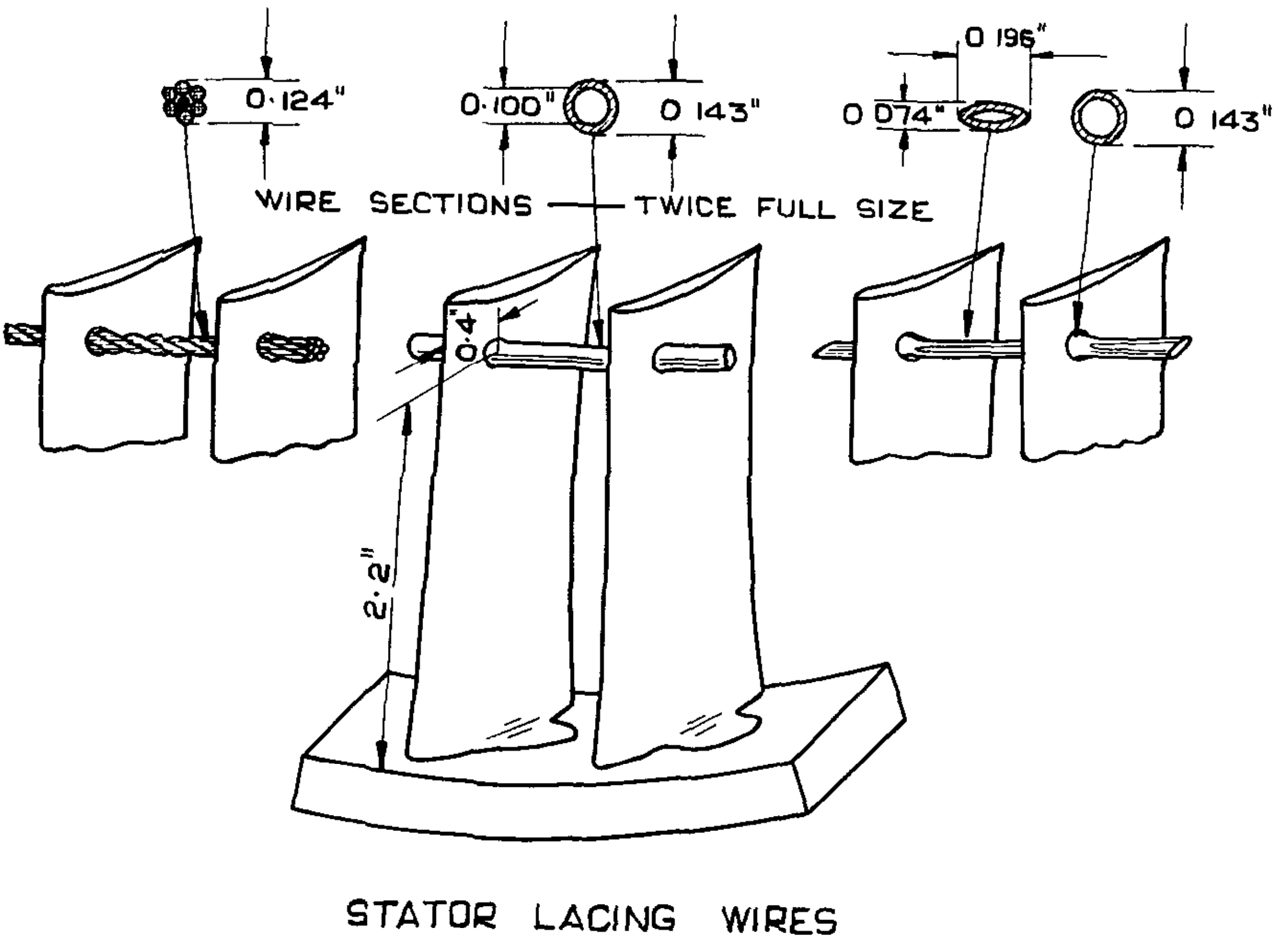
FOR DESIGN ASSUMPTIONS SEE APPENDIX I

COMPRESSOR CONSTRUCTION AND BLADE ANGLES FOR THE FREE VORTEX BLADES.

FIG. 2



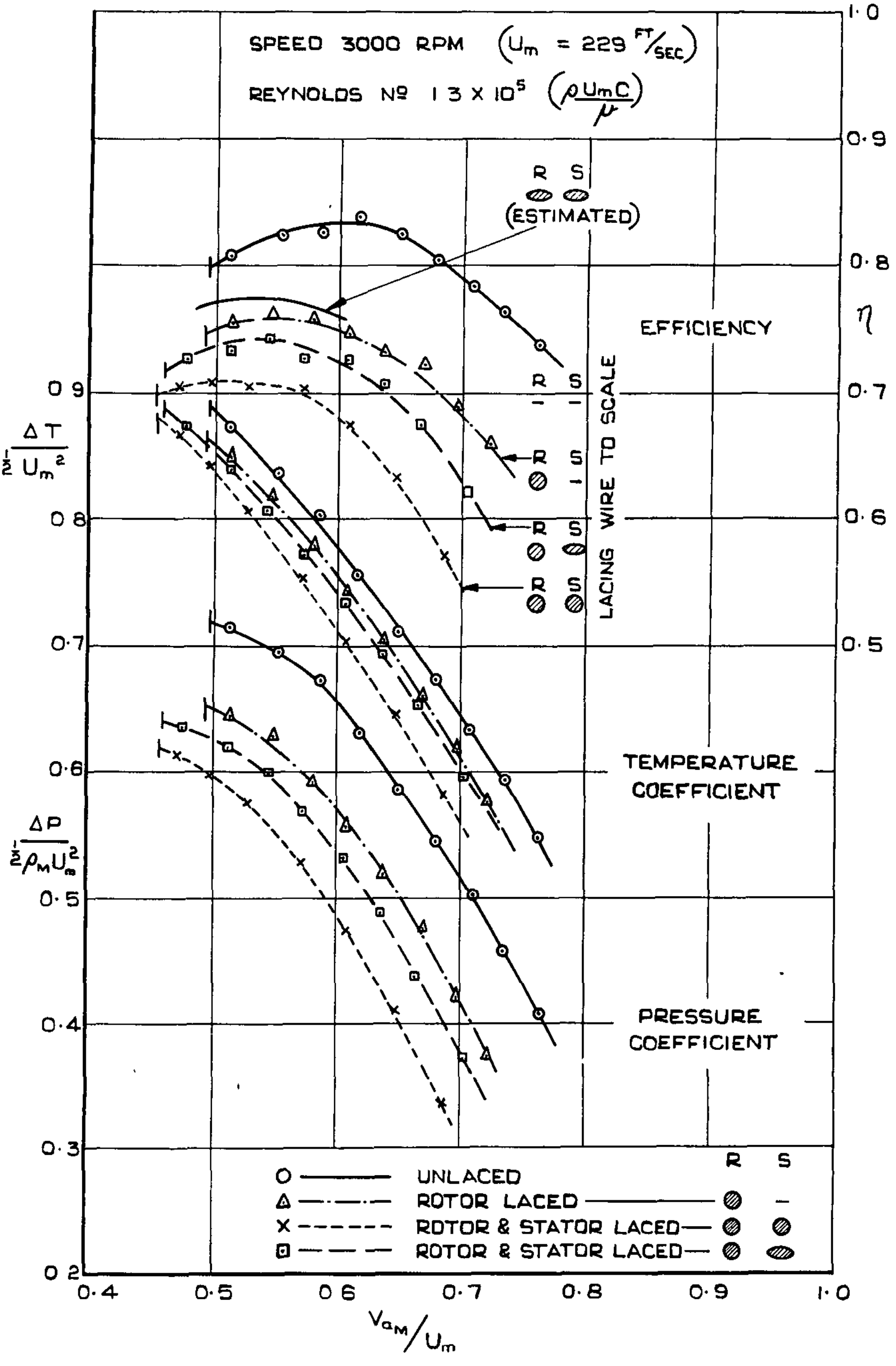
ROTOR LACING WIRE



STATOR LACING WIRES

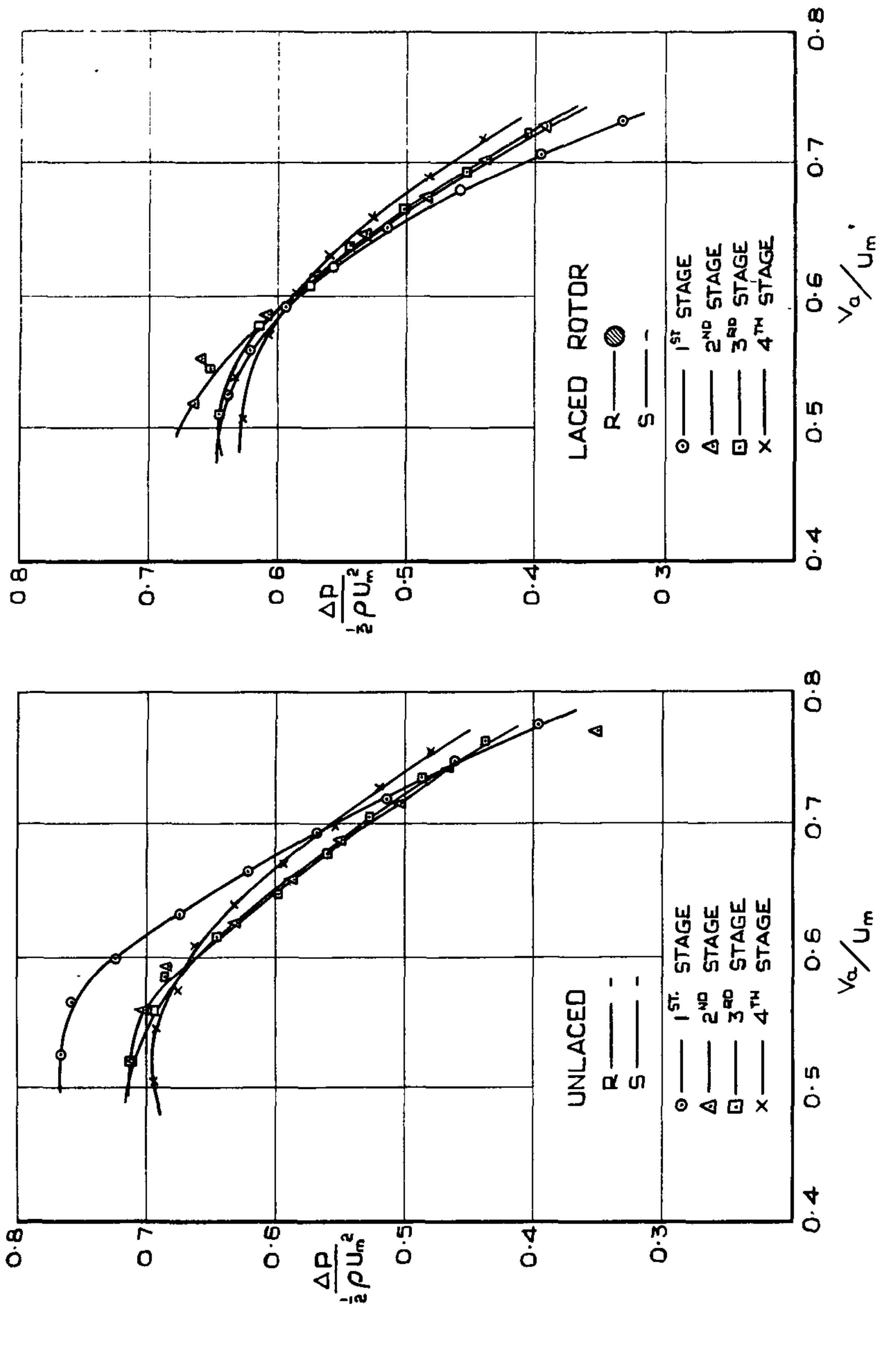
DETAILS OF THE LACING WIRES USED IN THE COMPRESSOR.

FIG. 3



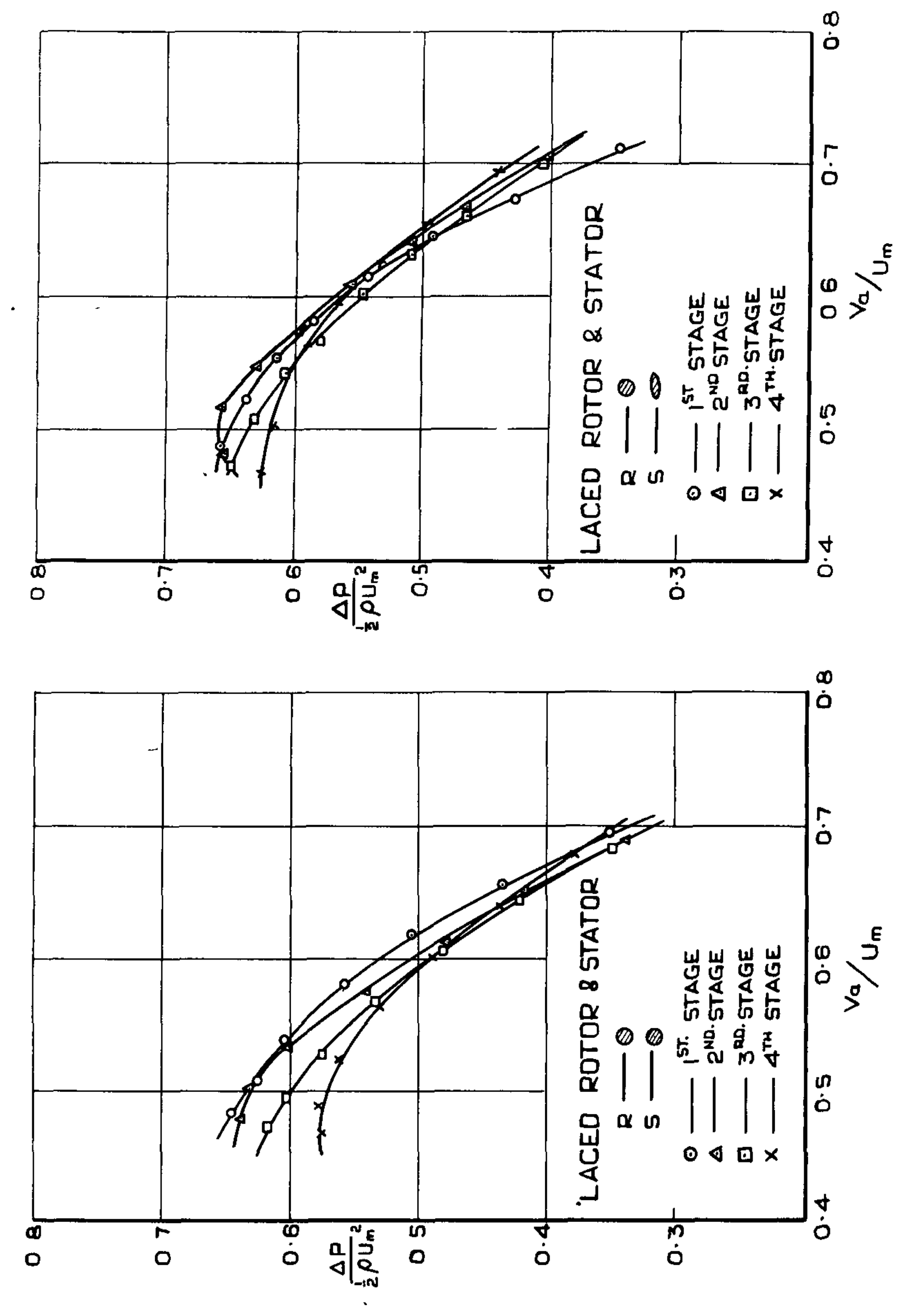
MEAN STAGE CHARACTERISTISTICS.

FIG 4



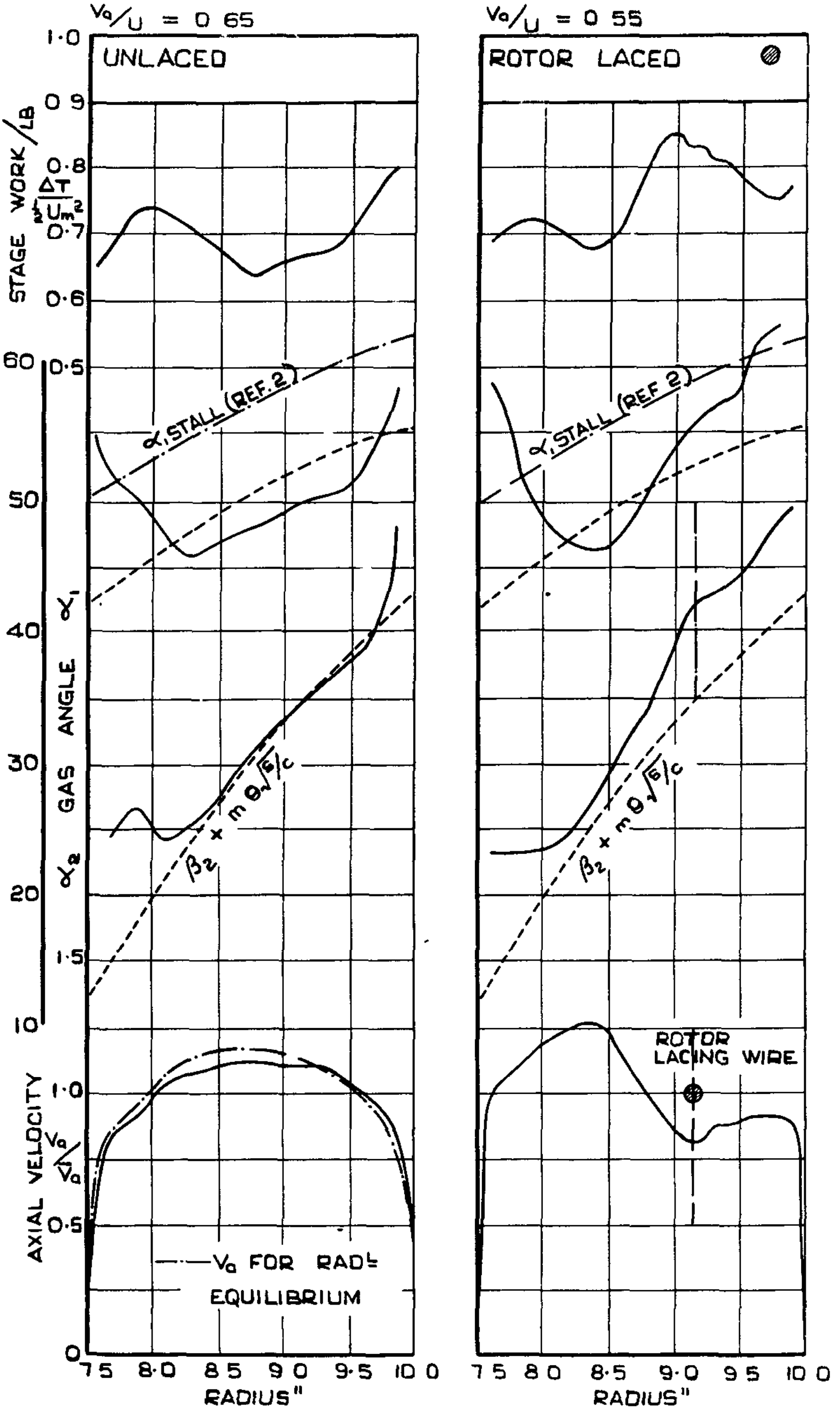
COMPRESSOR STAGE CHARACTERISTICS.

FIG. 5.

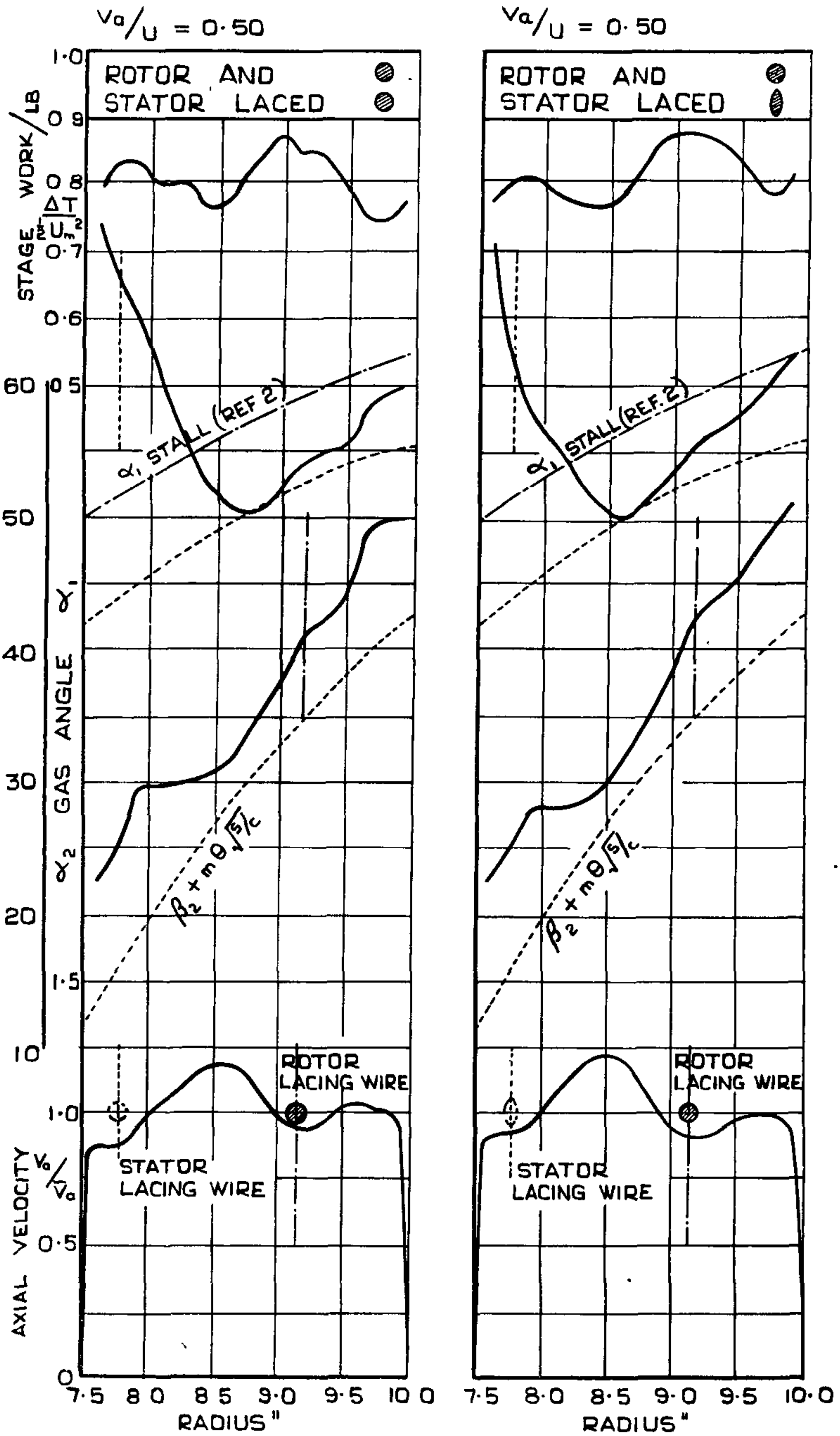


COMPRESSOR STAGE CHARACTERISTICS.

FIG. 6

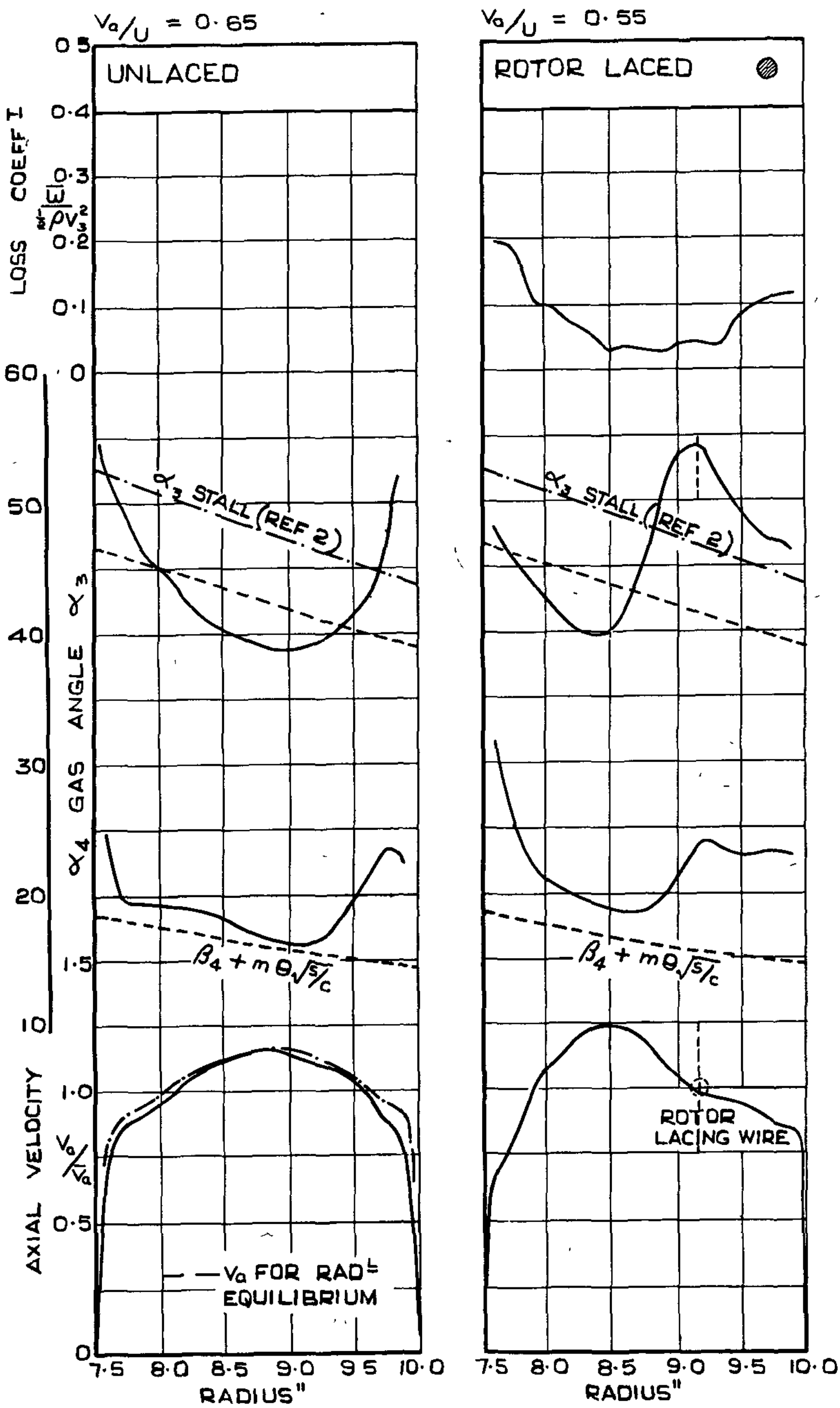


FLOW DETAILS IN FOURTH STAGE ROTOR ROW.



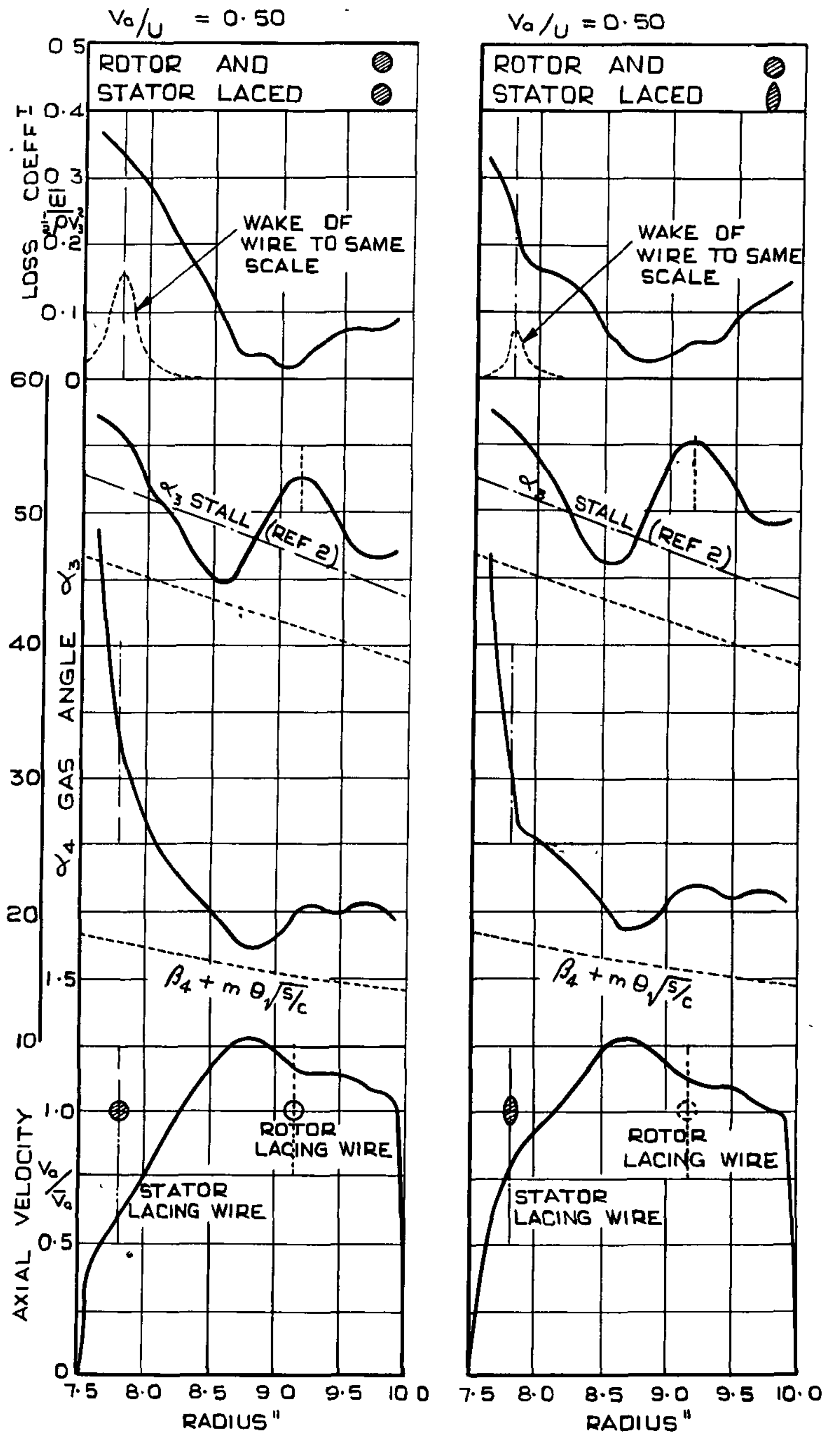
FLOW DETAILS IN FOURTH STAGE ROTOR ROW.

FIG. 8



FLOW DETAILS IN FOURTH STAGE STATOR ROW.

FIG. 9.



FLOW DETAILS IN FOURTH STAGE STATOR ROW

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