THEORETICAL AND EXPERIMENTAL INVESTIGATION OF A HYPERSONIC MULTI-STAGE COMPRESSOR

(Title Unclassified)

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DEPARTMENT
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Part I. Aerodynamic Study of the Compressor Performance

by

Eugene Sevigny and Marian Visich, Jr.

Part II. Preliminary Analysis of the Heat Transfer to the Compressor Blades

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Robert J. Cresci, Anthony Casaccio, Samuel Lederman and Paul A. Libby

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Introduction

In reference 1 Ferri outlined the possibility of using a hypersonic multi-stage compressor to operate a hypersonic wind turnel permitting full-scale simulation at high Mach numbers (roughly 20) at reasonable altitudes (approximately 110,000 feet). In this series of reports the various aerodynamic, heat transfer and structural problems connected with such a compressor are considered. In Part I of this report an additional study of the compressor performance is reported. In Part II the heat transfer to the fourth stage blades is analyzed.

PART I. AERODYNAMIC STUDY OF THE COMPRESSOR PERFORMANCE

by

Eugene Sevigny* and Marian Visich, Jr.+

Polytechnic Institute of Brooklyn

In order to determine further the feasibility of using a supersonic compressor as a source of high stagnation temperature air for hypersonic wind tunnel testing, a detailed aerodynamic study was made of a four-stage counter-rotating supersonic compressor. The performance of the compressor was calculated assuming that the relative Mach number entering and leaving the rotor was the same and, therefore, the flow was only turned in the rotor.

Fig. 1 is a schematic installation drawing of the four-stage compressor operating from the pebble bed heater of the Polytechnic Institute of Brooklyn Aerodynamics Laboratory. The four-stage counter-rotating compressor is shown driven by two separate two-stage turbines.

The results of the aerodynamic analysis of the compressor are shown in Figs. 2 to 11. The subscript 1 corresponds to the absolute

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conditions ahead of the first stage and the subscript 8 corresponds to the absolute conditions leaving the fourth stage. The performance of the compressor was calculated assuming that the air is a calorically perfect gas.

Extensive calculations were performed for the following case:

Rotor tangential velocity u = 1300 ft/sec.

Inlet stagnation pressure P_{0_1} = 600 psia

Compressor blade angle $\alpha = 60^{\circ}$

Pressure recovery per stage 70%

The results of these calculations are cross plotted in Figs. 2 to 6. Also indicated on Figs. 3, 4 and 5 are data points for a six-stage compressor.

The effect of the pressure recovery per stage on the static pressure ratio across the compressor and the stagnation pressure of the air leaving the compressor is shown in Figs. 7 and 8, respectively.

The effect of inlet stagnation pressure on the stagnation pressure leaving the fourth stage of the compressor is shown in Fig. 9 for an inlet stagnation temperature of 3000°R.

Figs. 10 and 11 show the effect of the compressor blade angle on the final stagnation temperature and pressure for an inlet stagnation pressure of 600 psia and an inlet stagnation temperature of 3000°R.

The results of this analysis indicate that the concept of using a compressor as a source of high stagnation temperature air is feasible. Additional calculations are being performed to find the effect of lower initial inlet Mach number and more realistic pressure recovery per stage.

The preliminary design of an experimental model to measure the pressure recovery in turning passages of 120° at M=3.1 and M=6.0 has been completed. Fig. 12 shows the design of the model which will be tested at the Polytechnic Institute of Brooklyn Aerodynamics Laboratory. The sidewalls of the model will be fabricated of plexiglas to permit schlieren pictures of the flow in the passages.

PART II. PRELIMINARY ANALYSIS OF THE HEAT TRANSFER TO COMPRESSOR BLADES

by

Robert J. Cresci,* Anthony Casaccio,**

Samuel Lederman+ and Paul A. Libby++

Polytechnic Institute of Brooklyn

In order to make some estimates of the heat transfer which will be encountered by the blades of the compressor, the heating on the blades of the last stage, which will be critical, has been considered. The entrance conditions to this stage were taken to be those representative of the presently considered design. In particular, the case corresponding to inlet conditions of $M_1=5$, $P_{0_1}=600$ psia, $T_{0_1}=3000^\circ R$ and $\alpha=60^\circ$ of Part I of this report has been calculated. The significant data are as follows:

Relative Mach number entering fourth stage	$M_7=12.44$
Relative stagnation temperature of fourth stage	$T_{0_7}=16,000^\circ R$
Stagnation point pressure at leading edge of fourth stage	$P_{0_7}=77.6$ psia

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It is pointed out that the stagnation temperature is the ideal gas value.

The stagnation point heat transfer formula of reference 2 was modified for two-dimensional flow and was applied with a Newtonian velocity gradient. A preliminary analysis indicated that a nose radius on the order of 0.2" with a wall thickness of approximately 0.050" would be reasonable; the blade material was taken to be Inconel X.

The heat transfer problem involves simultaneous consideration of the aerodynamic heating, the heat conduction through the wall, and the cooling inside the blade. Aerodynamic heating occurs during only part of the revolution of the compressor whereas the internal cooling occurs throughout the cycle. The heating phase was assumed to occur in 1/600 of a second while the total revolution occurs in 1/100 of a second. These values correspond to a high energy flow passage two feet in length, to a mean diameter of approximately 3.5 feet and to a rotative speed of 6000 rpm. It is assumed that no significant low energy flow exists external to the blades during 5/6ths of the cycle. If internal cooling is sufficient, the power required from the turbines can in this way be significantly reduced.

The solution was carried out by means of the Analog Computer; initial conditions corresponding to uniform wall temperature were imposed and the temperature histories of the exposed surface and of the internal surface of the blade were examined as the number of cycles increased. It was found that with internal cooling a periodic temperature history of the exposed surface and a steady inner surface history were established after only four seconds. Since this time interval is considerably shorter than the contemplated running time of the compressor, it was concluded from this result that a heat sink solution to the compressor heating problem would not be practical even with a more conductive blade material.

A series of solutions was carried out. The internal cooling heat transfer coefficient was varied in each so that the exposed surface temperature did not exceed 2000°R. In Fig. 13 the exposed surface temperature history and the internal surface temperature history for an internal heat transfer coefficient of 0.3 BTU/ft^2 sec./$^\circ$R are shown. The stagnation temperature of the coolant was taken to be 600°R. As an independent check of the solution, the heat balance between the heat input on the exposed surface during 1/600 of a second and the heat withdrawn on the internal surface during 1/100 of a second was confirmed.

The heat transfer downstream of the leading edge of the blade has also been investigated; the same inlet conditions to the last stage as was used for the stagnation point heating were applied in this case. The boundary layer was assumed to be turbulent. A reasonable estimate was made of the flow field with a total turning angle of 120° and with a chord of 4". The calculations were carried out for both the compression and expansion surfaces. The distributions of heat transfer coefficient are shown in Fig. 14; the maximum values shown there are approximately 10% of the stagnation point heat transfer. It is pointed out that in the central region on the compression side of the blade there may exist a separated region due to shock impingement; in this region there may be increased rates of heat transfer but these would not be expected to be more than 30 or 40% of the stagnation point value.

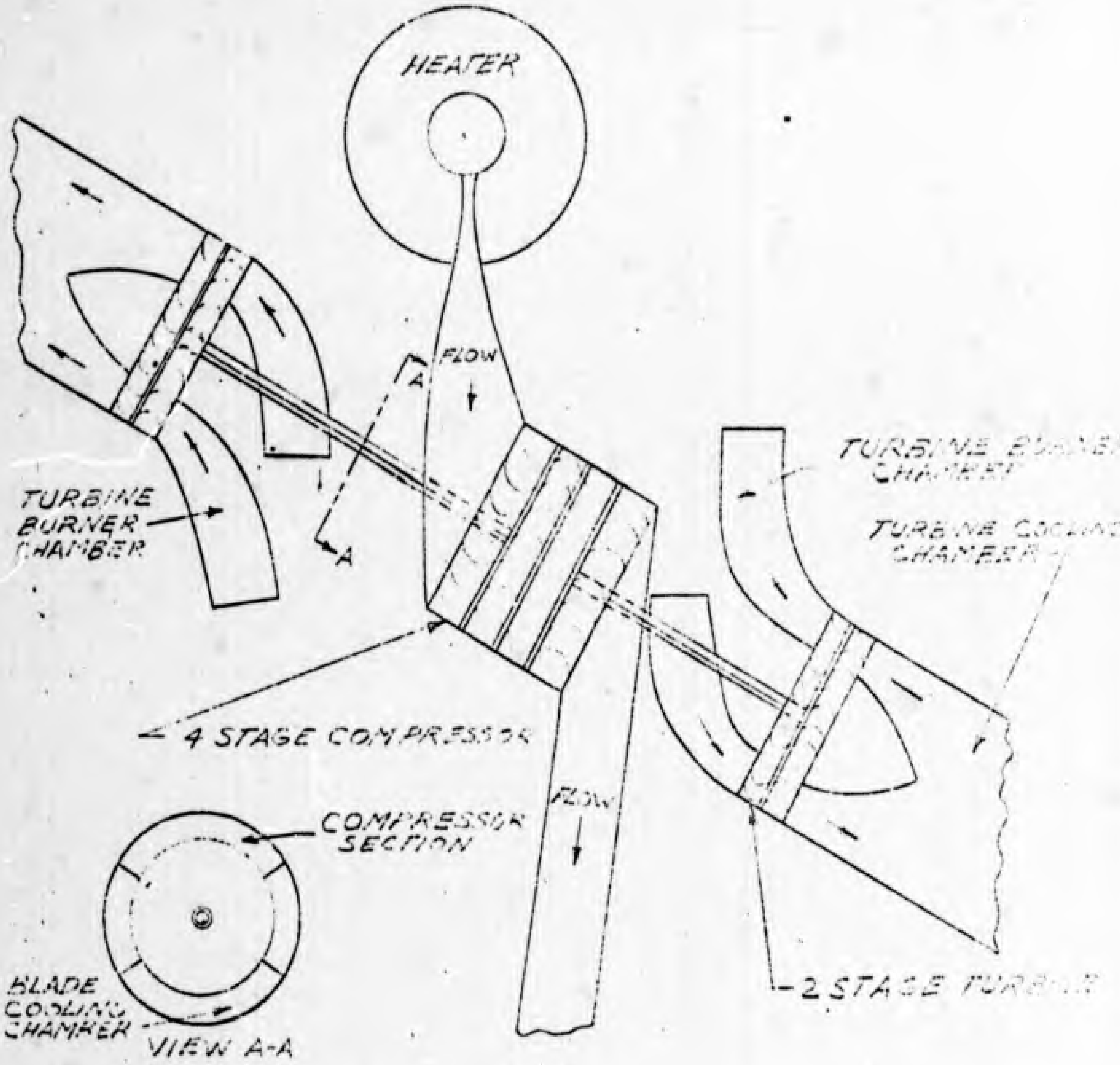
On the basis of these results it is estimated that a total heat transfer to the last stage involving 60 blades of 4" chord and 5" span will be roughly 2,000 BTU/sec. This is approximately 6% of the total energy entering the last stage of the compressor. Based on the energy of the flow leaving the last stage, the heat transfer to this stage represents roughly 4% of the total energy. A weight flow of 10 lb/sec. has been assumed.

As a next step in the analysis of the blade heating, consideration will be given to the internal cooling passages. It is expected that coolant injection will take place in the stagnation point region with the coolant flowing downstream in passages shaped so as to provide a local heat balance.

References

1. Ferri, A.: Preliminary Description of a Heating System for Hypersonic Wind Tunnels for Mach Numbers Between 15 and 30. Polytechnic Institute of Brooklyn, PIBAL Report No. 454, August 1958 (Classified Confidential).

2. Fay, J.A. and Riddell, F.R.: Theory of Stagnation Point Heat Transfer in Dissociated Air. Journal of the Aeronautical Sciences, Vol. 25, No. 2, pp. 73-85, February 1958.



INSTALLATION SCHEMATIC
4 STAGE

FIG. 1

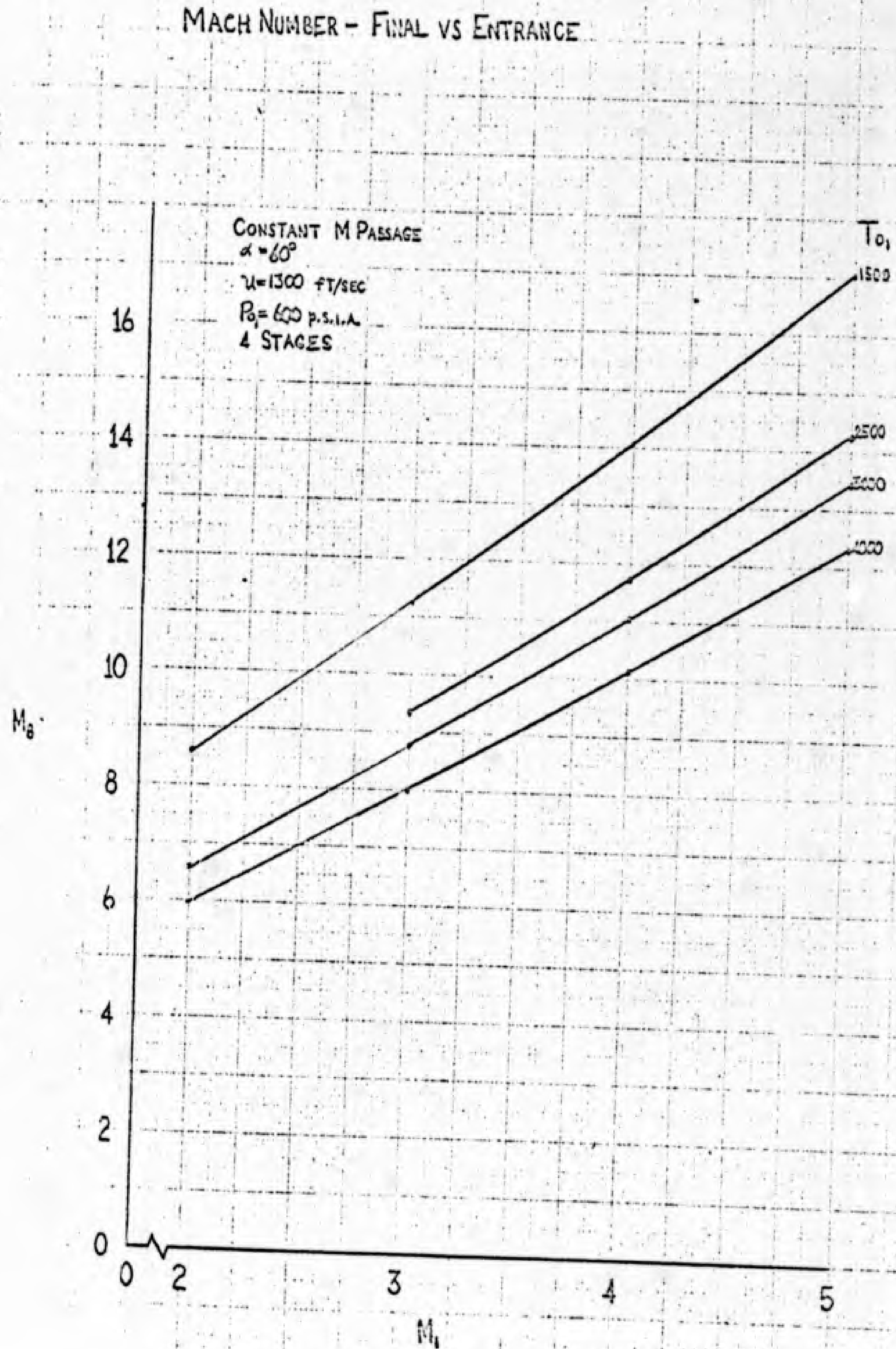

MACH NUMBER - FINAL VS ENTRANCE
CONSTANT M PASSAGE
$\alpha = 60°$
$u = 1300$ FT/SEC
$P_{0_1} = 600$ p.s.i.a.
4 STAGES
T_{0_1}
1500
2500
3000
4000
16
14
12
10
8
6
4
2
0
M_8
0
2
3
4
5
M_1

FIG. 2

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STAGNATION TEMPERATURE - FINAL VS ENTRANCE

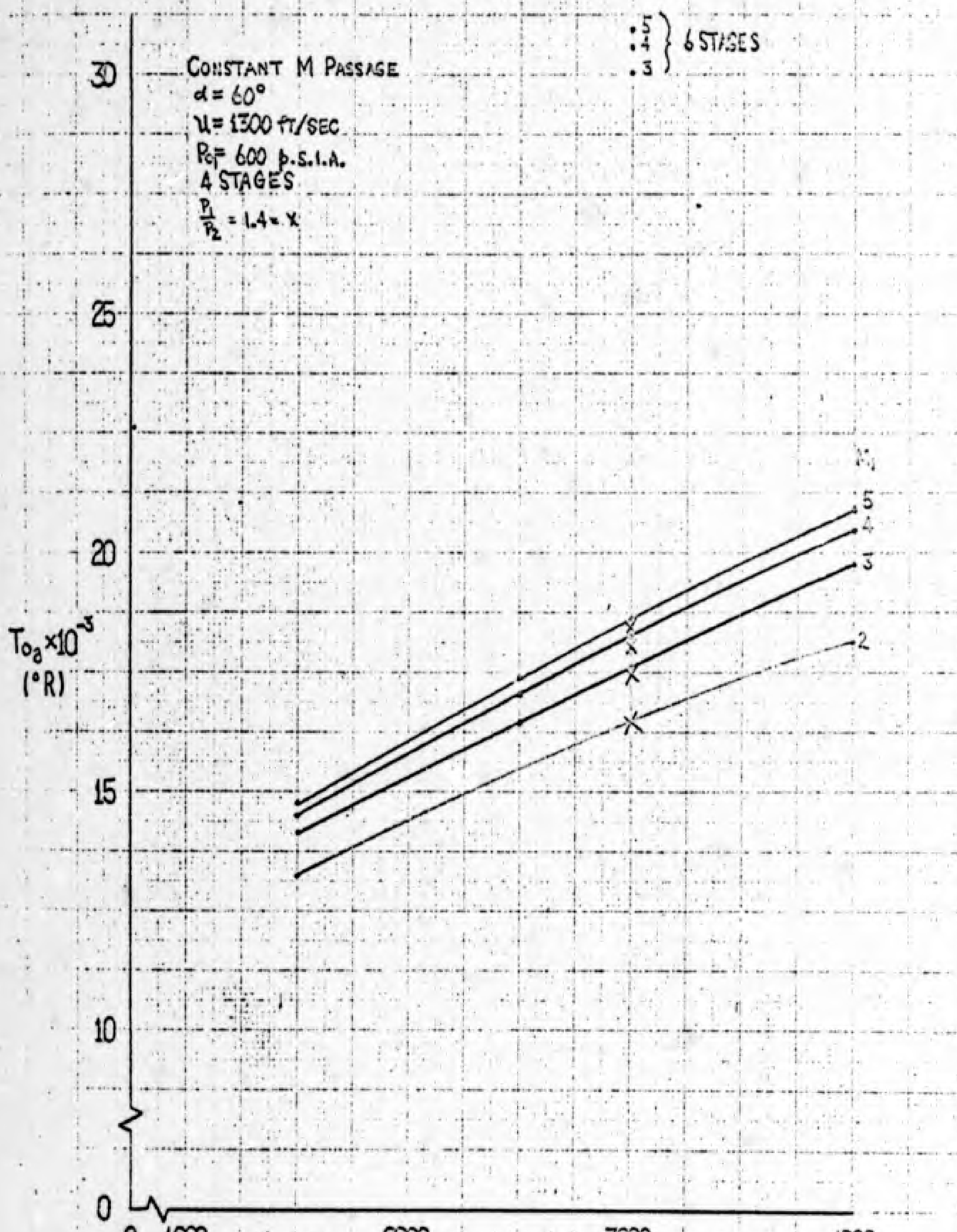

FIG. 3

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FINAL STAGNATION PRESSURE VS ENTRANCE MACH NUMBER

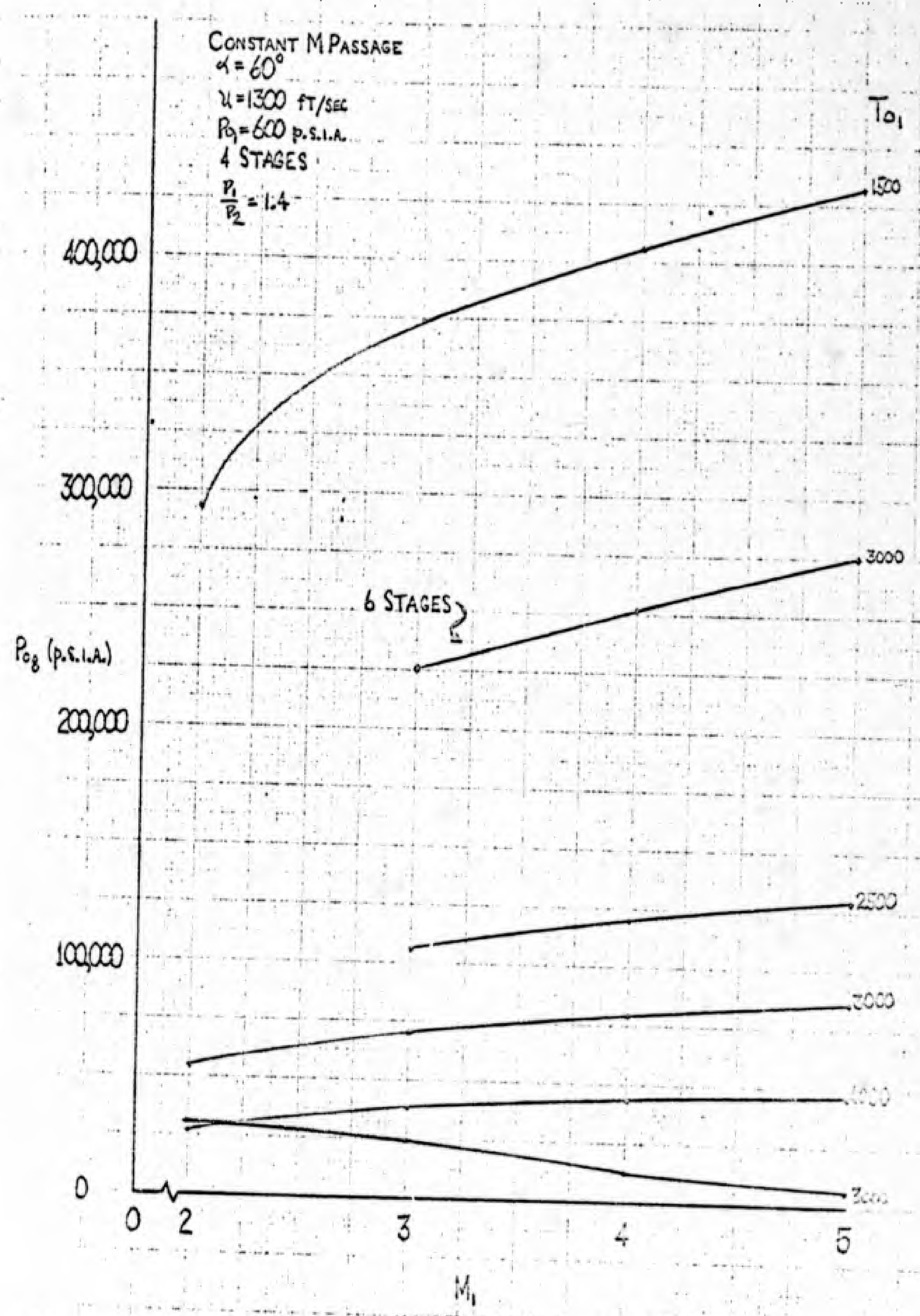


FIG. 4

[redacted]

FINAL STAGNATION PRESSURE VS ENTRANCE STAGNATION TEMPERATURE

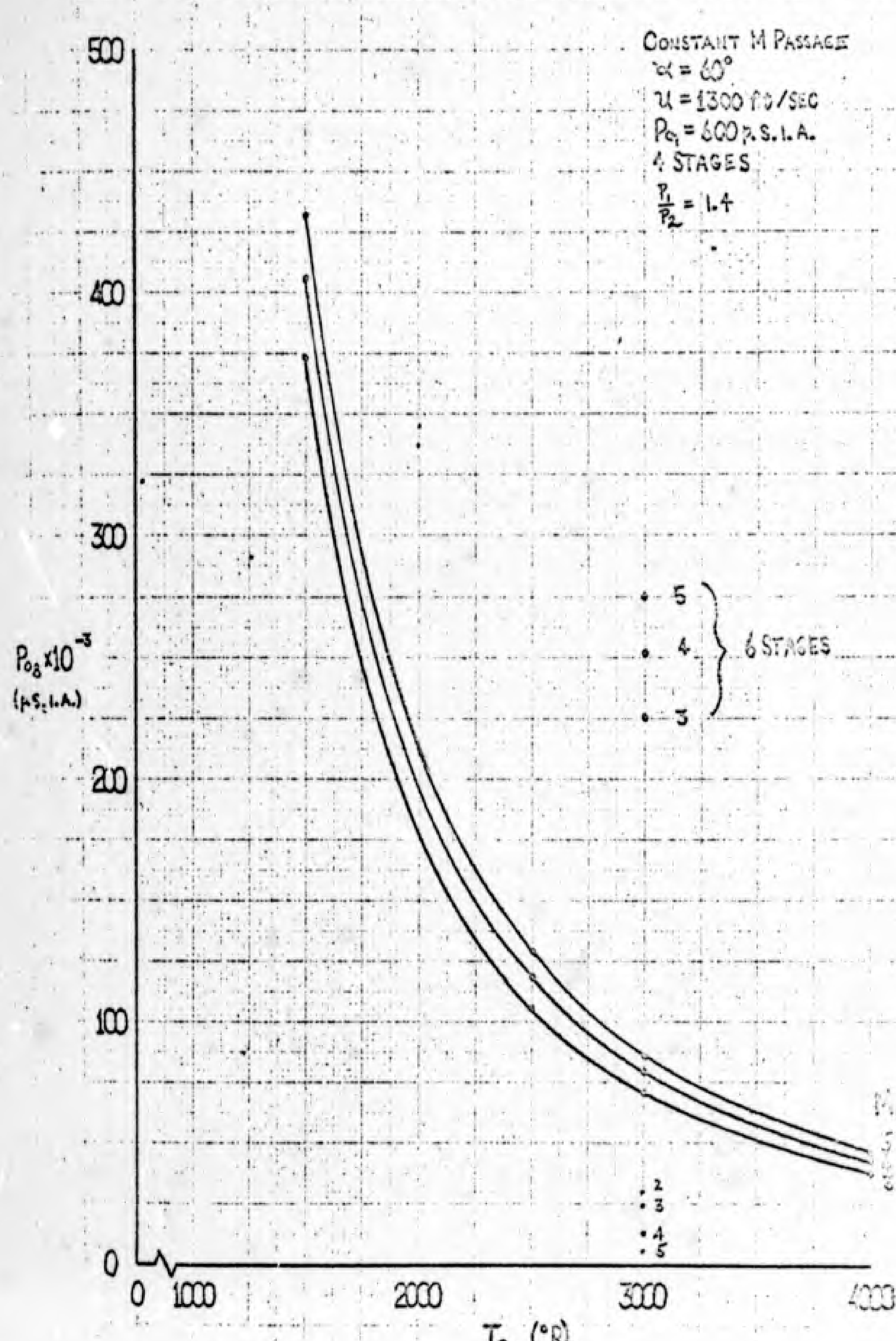
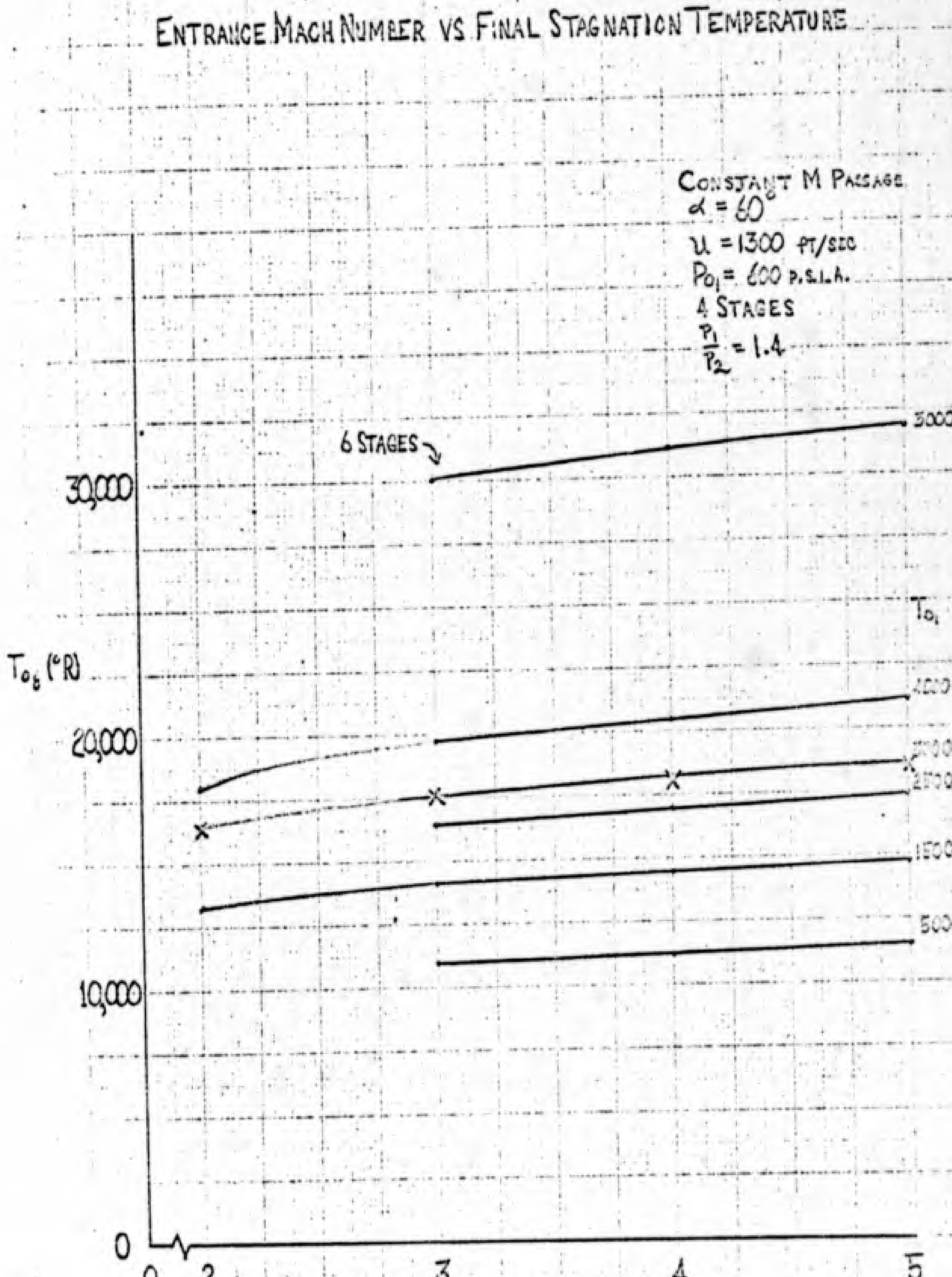


FIG. 5

[redacted]



ENTRANCE MACH NUMBER VS FINAL STAGNATION TEMPERATURE
CONSTANT M PASSAGE
$\alpha = 60°$
$u = 1300$ FT/SEC
$P_{0_1} = 600$ P.S.I.A.
4 STAGES
$\frac{P_1}{P_2} = 1.4$
6 STAGES
3000
T_{0_1}
4000
3000
2500
1500
500
30,000
20,000
10,000
0
T_{0_6} (°R)
0
2
3
4
5
M_1

FIG. 6

STATIC PRESSURE RATIO VS ENTRANCE MACH NUMBER

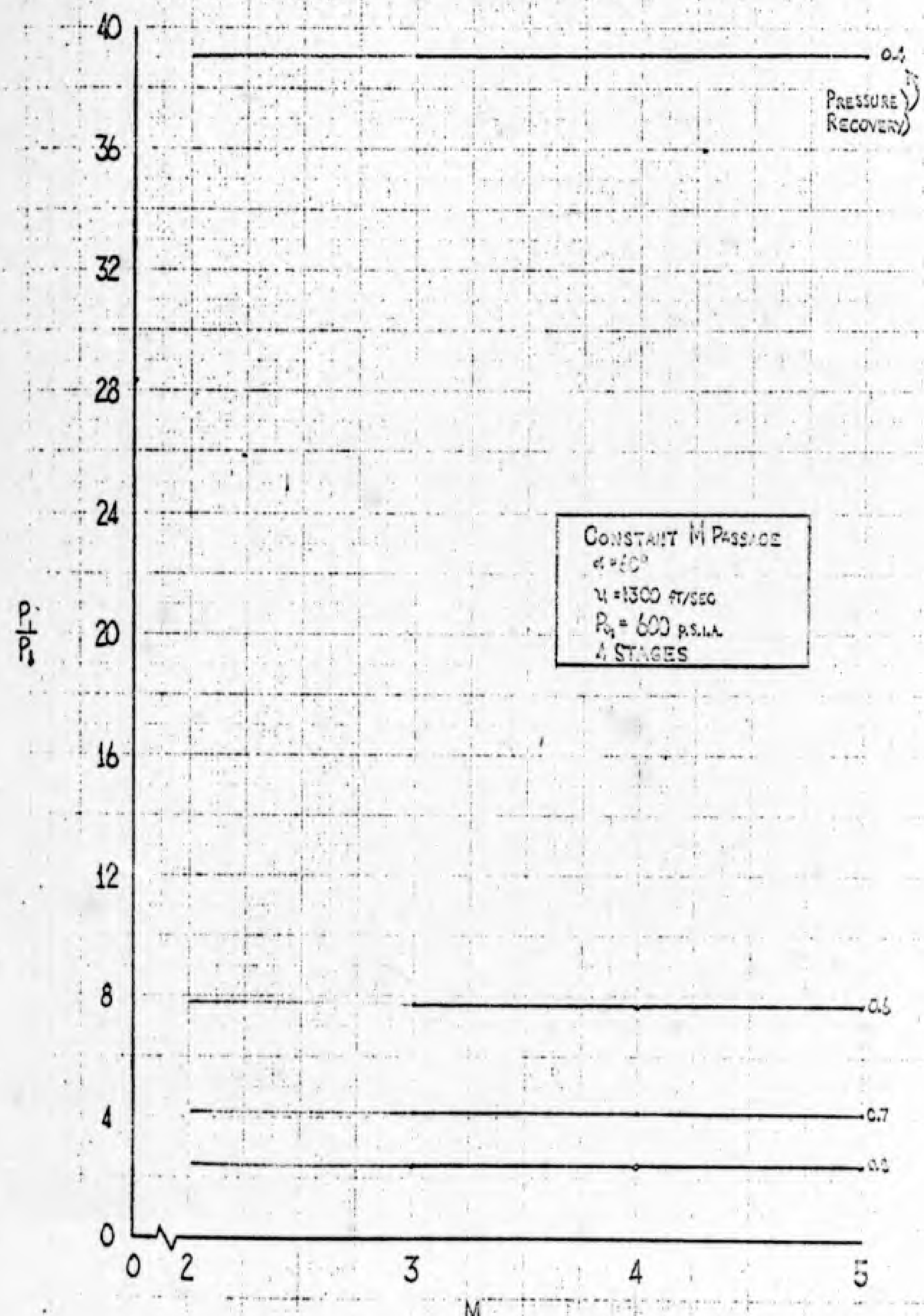


Fig. 7

FINAL STAGNATION PRESSURE VS PRESSURE RECOVERY / STAGE

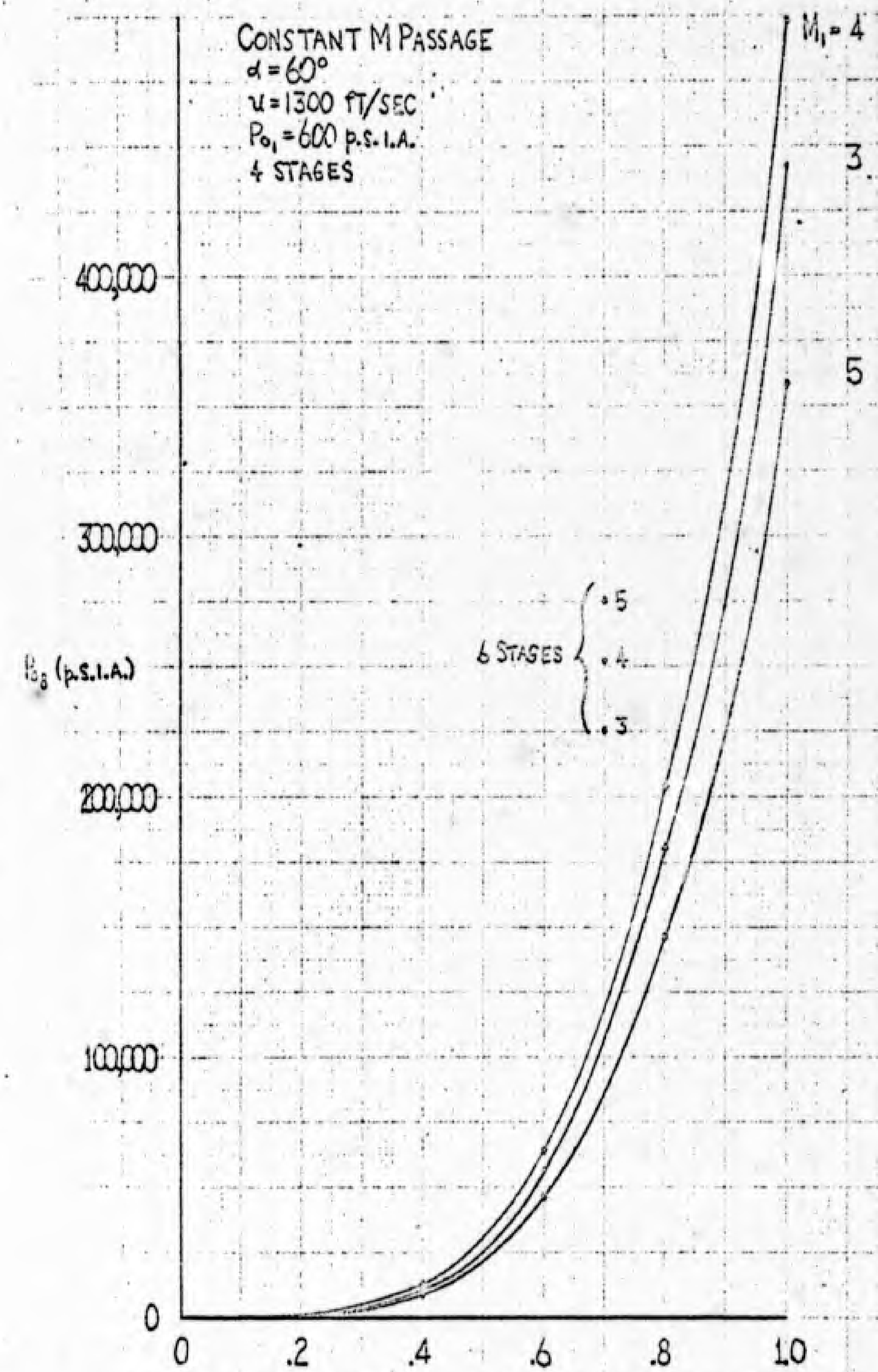


FIG. 8a

[redacted]

FINAL STAGNATION PRESSURE VS PRESSURE RECOVERY/STAGE

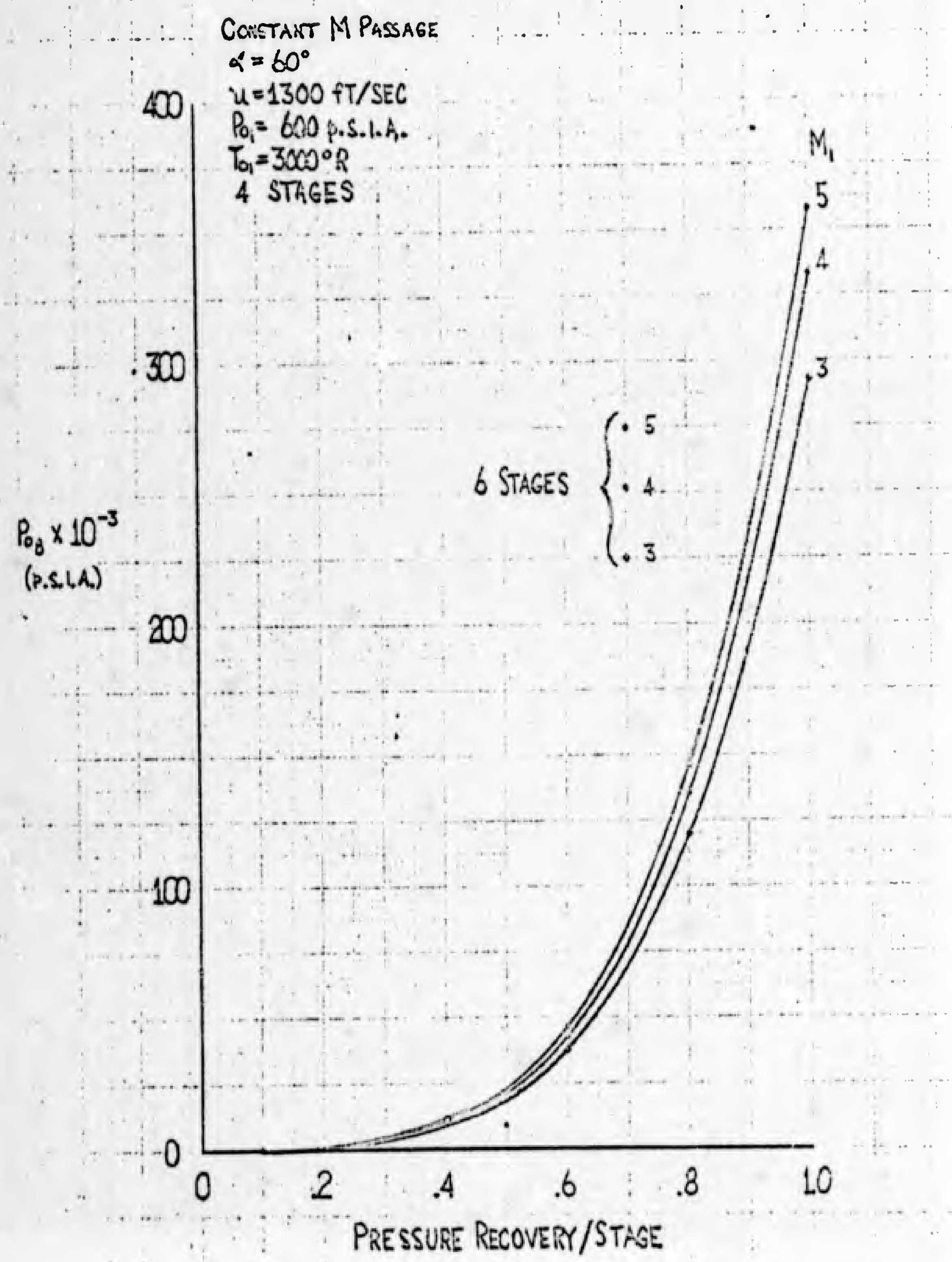
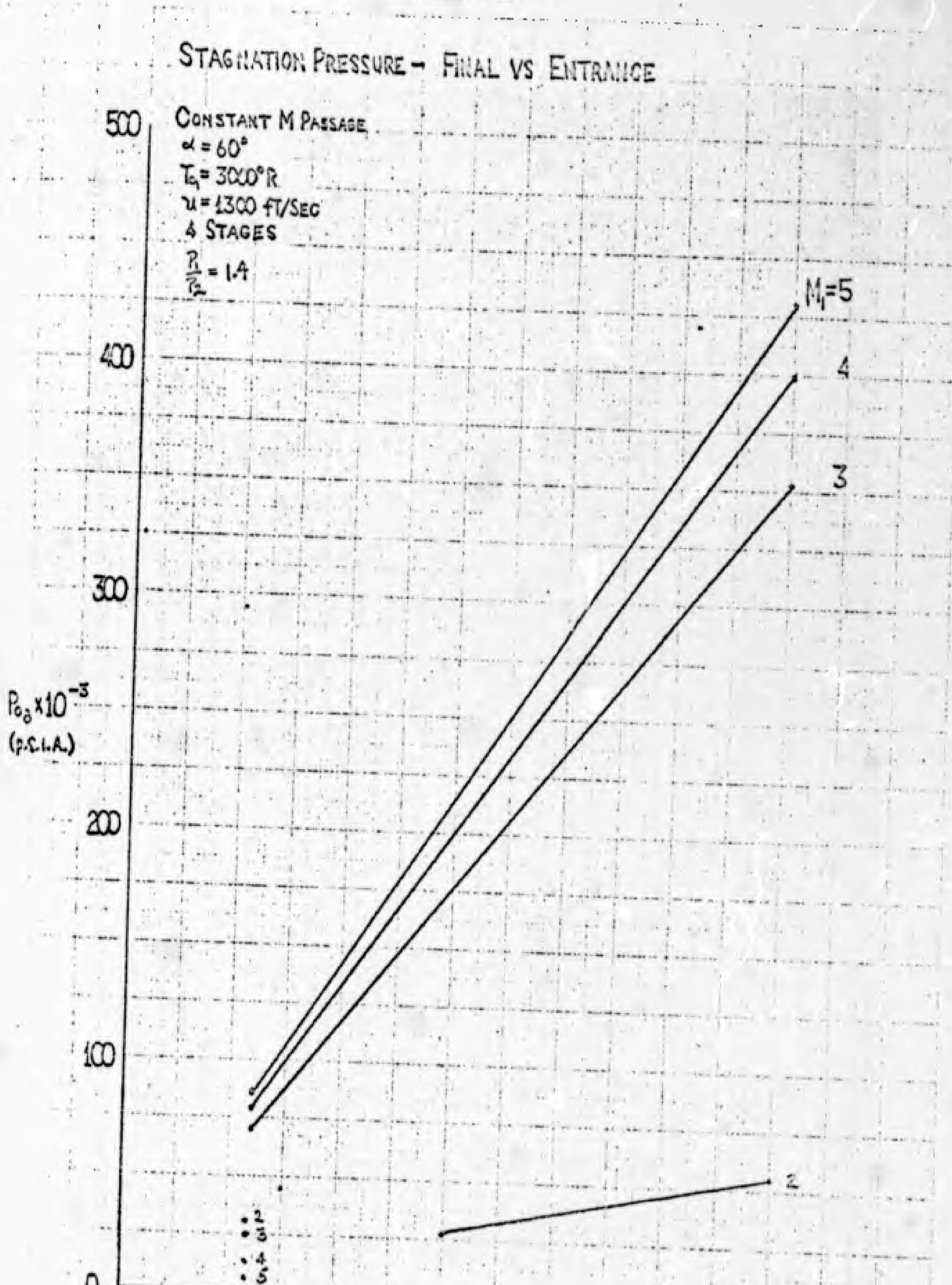


FIG. 8b

[redacted]



STAGNATION PRESSURE – FINAL VS ENTRANCE
CONSTANT M PASSAGE
α = 60°
T₀₁ = 3000°R
u = 1300 ft/sec
4 STAGES
P₁/P₂ = 1.4
500
400
300
200
100
0
$P_{0_3} \times 10^{-3}$
(p.s.i.a.)
M_1=5
4
3
2
2
3
4
5
0
500
1000
1500
2000
2500
3000
P_{0_1} (p.s.i.a.)

FIG. 9

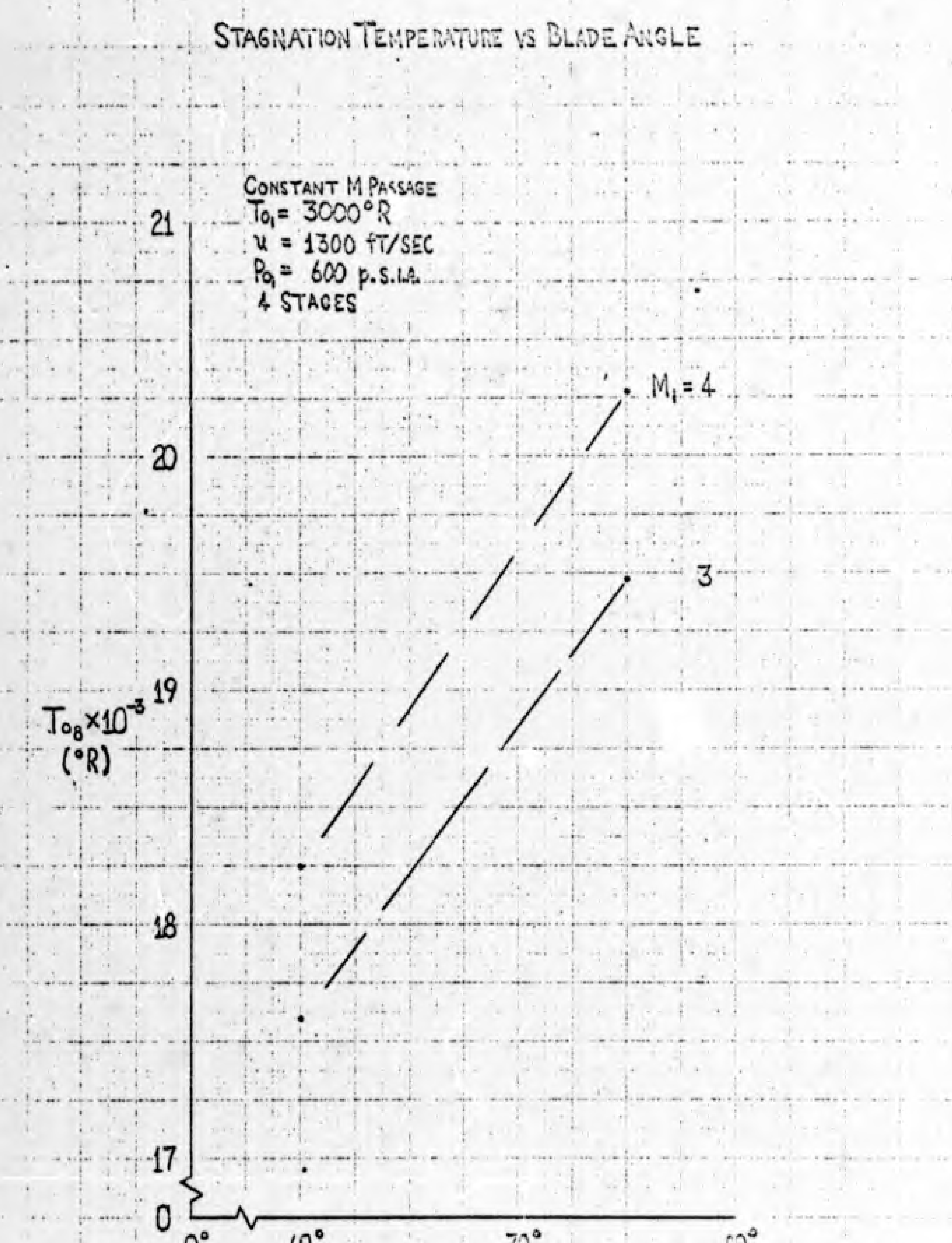

STAGNATION TEMPERATURE VS BLADE ANGLE
CONSTANT M PASSAGE
T_{O_1} = 3000°R
u = 1300 FT/SEC
P_{O_1} = 600 p.s.i.a.
4 STAGES
$M_1 = 4$
3
21
20
19
18
17
0
$T_{O_8} \times 10^{-3}$
(°R)
0°
60°
70°
80°
α

FIG. 10

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STAGNATION PRESSURE VS BLADE ANGLE

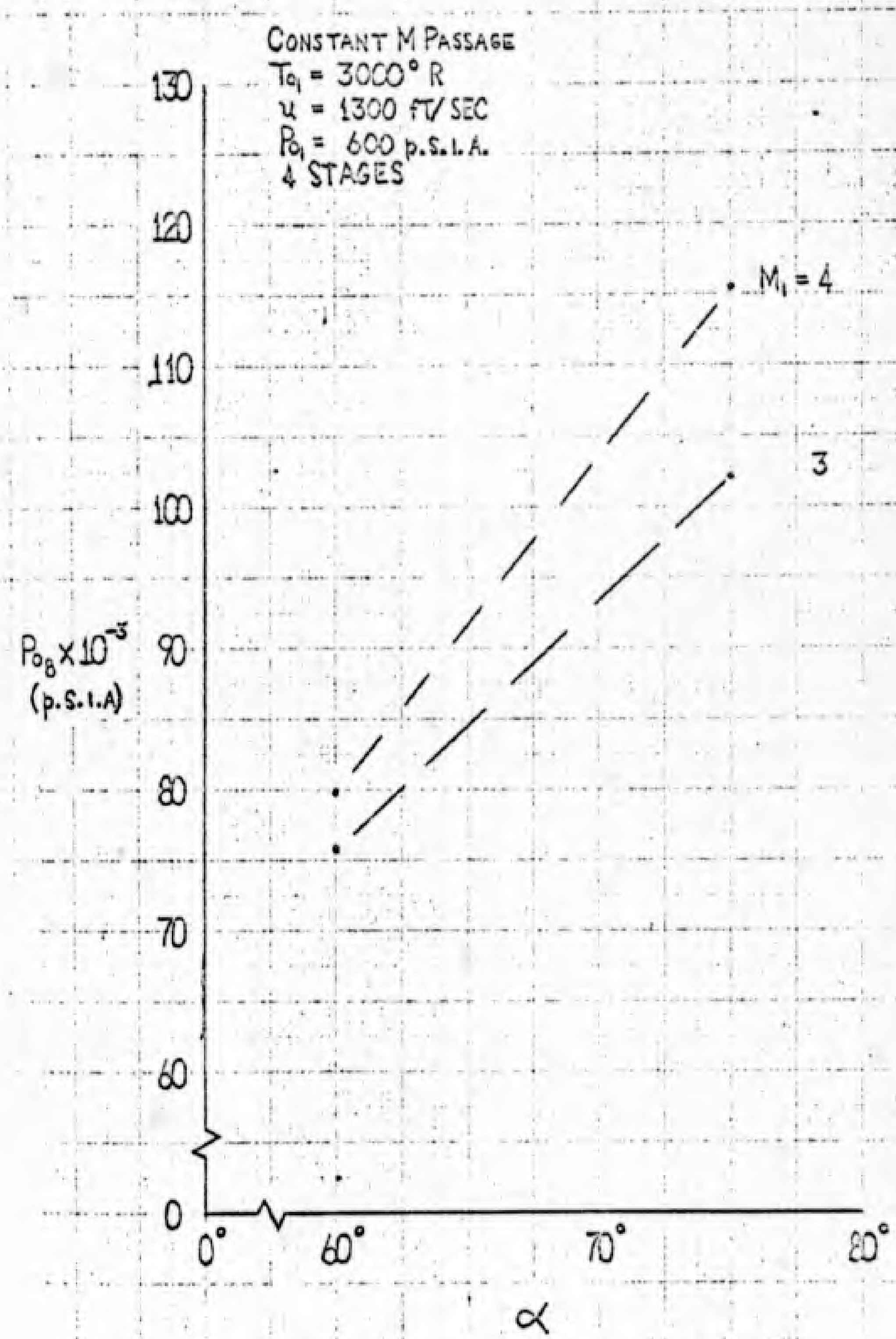

FIG. 11

L

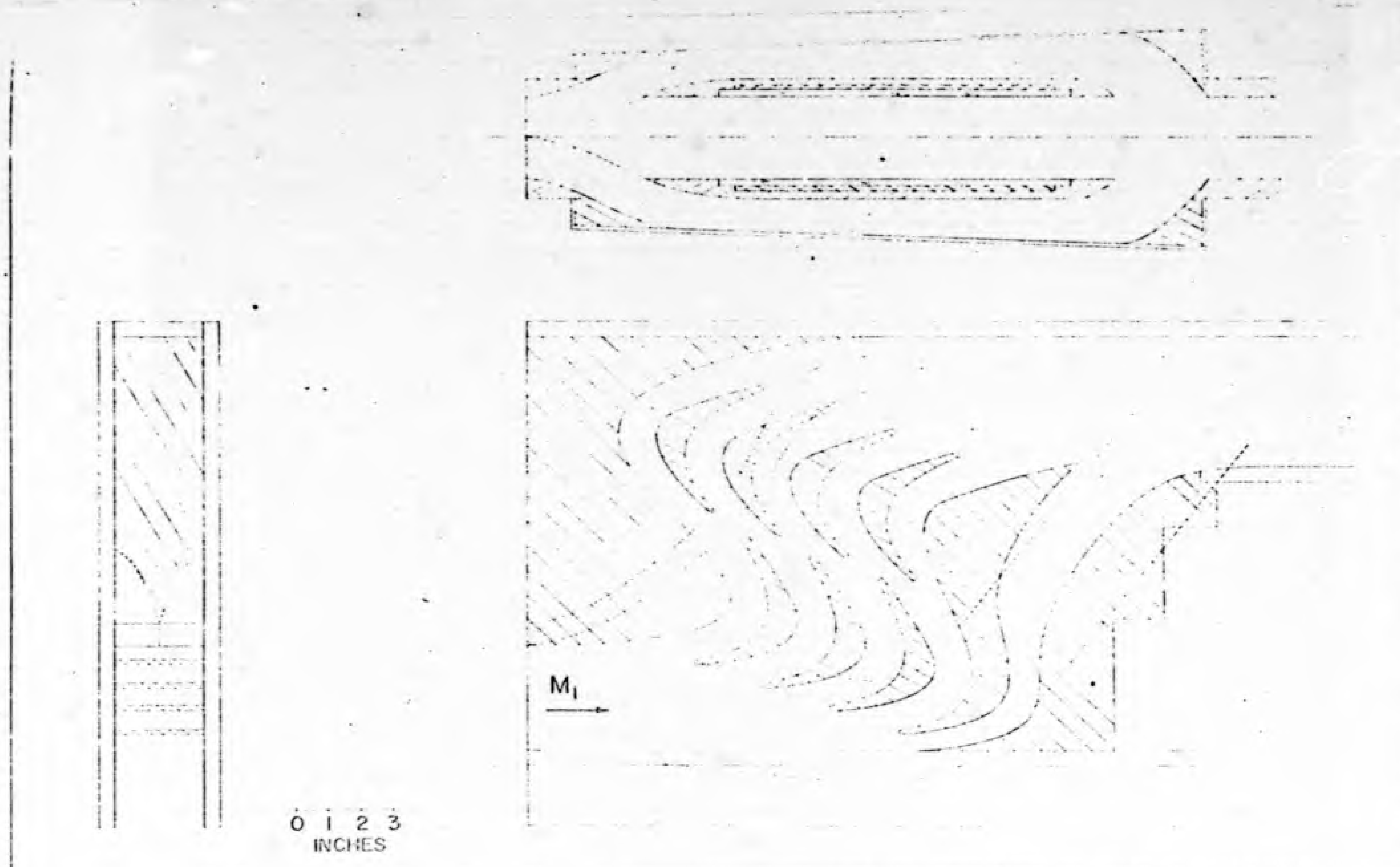

FIGURE 12 SCHEMATIC OF MODEL OF TURNING PASSAGES.

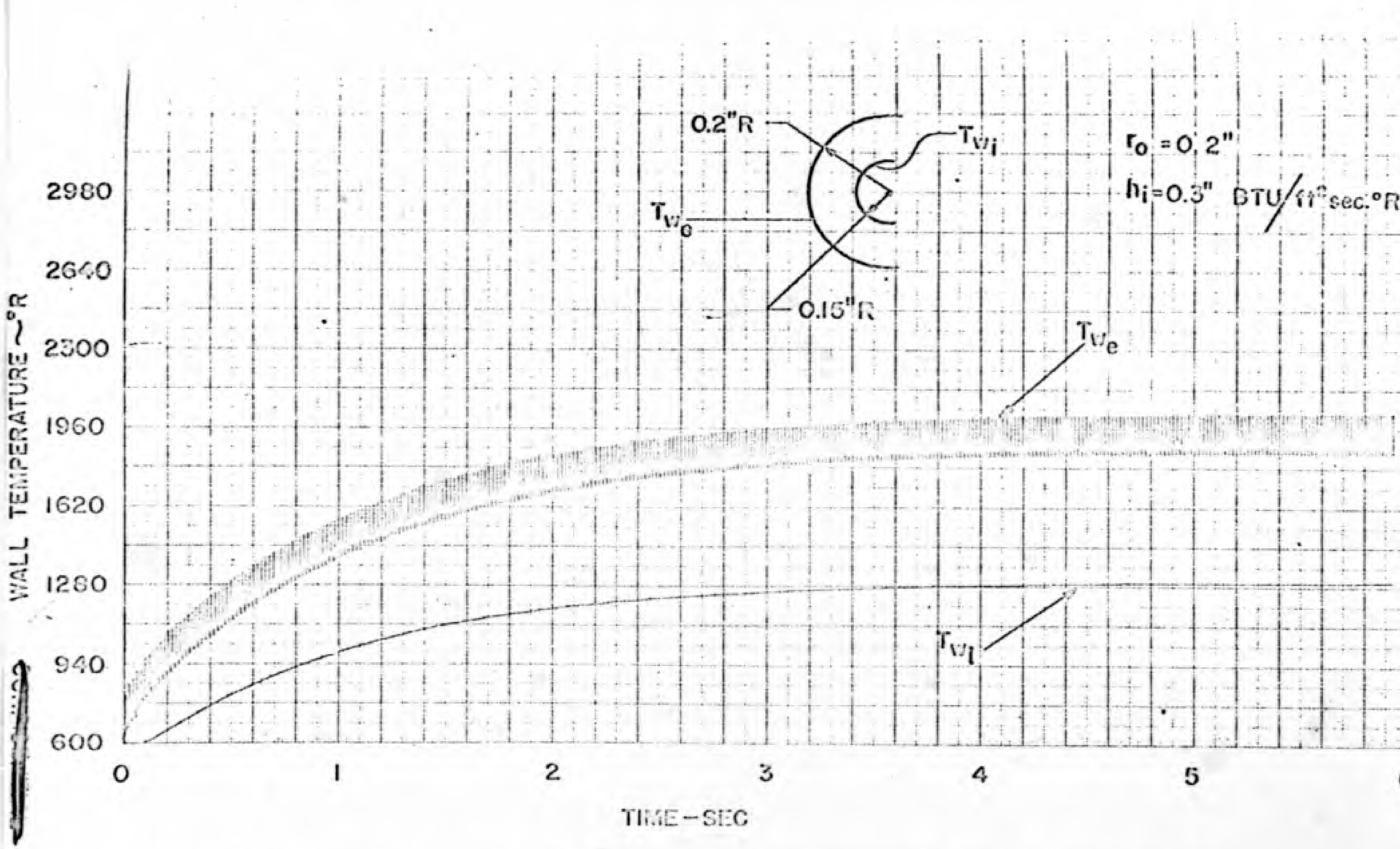

FIG.13 EXPOSED AND COOLED SURFACE TEMPERATURE HISTORIES AT THE STAGNATION POINT OF FOURTH STAGE BLADE.

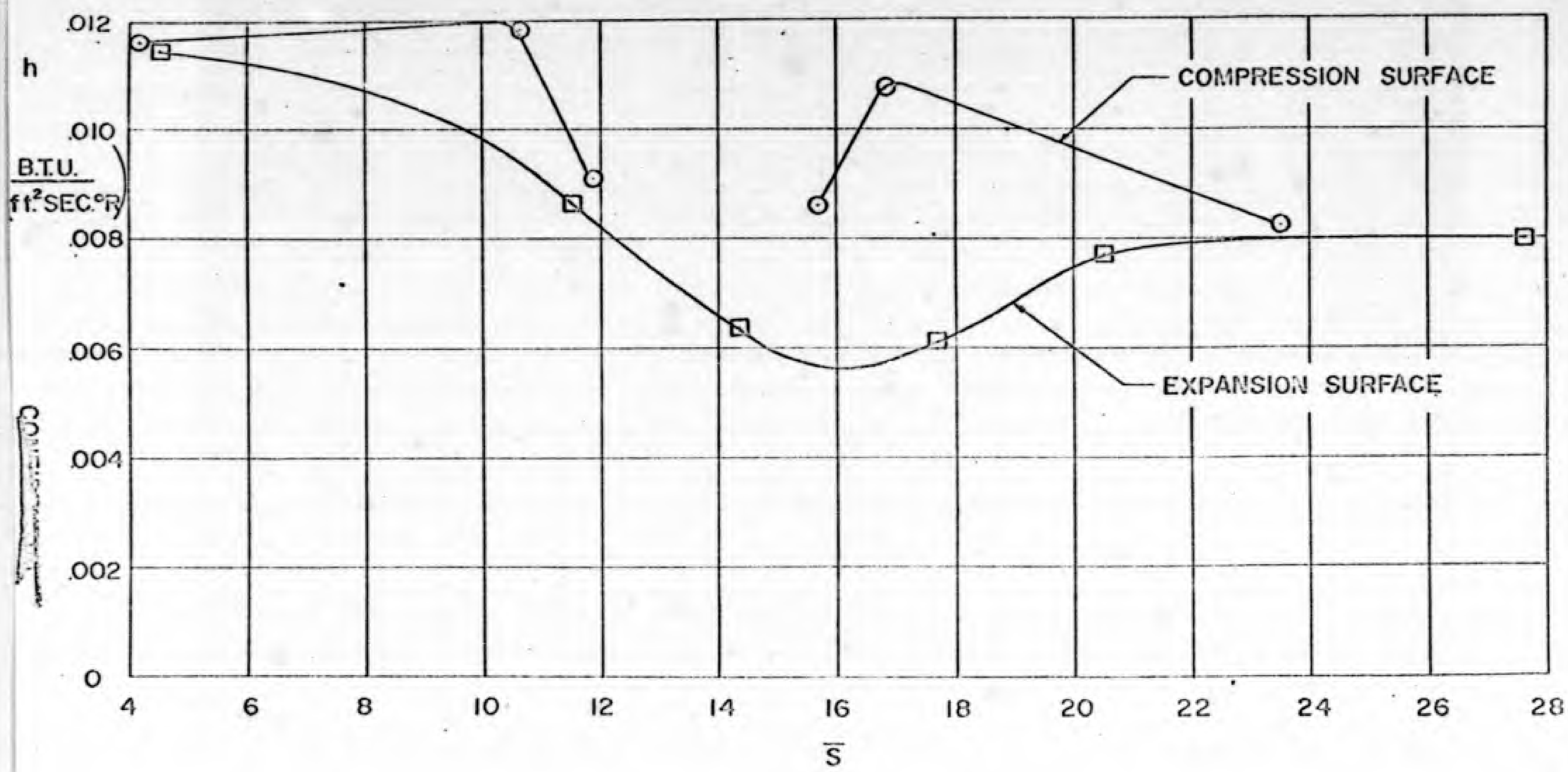

FIG. 14 TURBULENT HEAT TRANSFER ON FOURTH STAGE BLADE