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TECHNICAL NOTE

No. 1138

STANDARD PROCEDURES FOR RATING AND TESTING MULTISTAGE AXIAL-FLOW COMPRESSORS

By NACA Subcommittee on Compressors

Aircraft Engine Research Laboratory
Cleveland, Ohio

Washington
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SUMMARY

In order to establish a standard procedure for rating and testing multistage axial-flow compressors, the NACA Subcommittee on Compressors appointed a panel to write such a procedure. This panel made recommendations for standardization of test setups, instrumentation, test procedure, data to be taken, and the presentation of the data. These recommendations are presented.

INTRODUCTION

In the past, axial-flow compressors have been rated and tested, insofar as practical, in accordance with the standard procedures for rating and testing centrifugal compressors. The current methods of testing centrifugal compressors are given in references 1 and 2. The procedures for centrifugal compressors, however, have often proved unsuitable for axial-flow types because of fundamental differences in design and application. The Subcommittee on Compressors therefore appointed a panel consisting of Lt. Comdr. William Bollay of the Bureau of Aeronautics, Navy Department, Mr. Arnold H. Redding of the Westinghouse Electric Corporation, Mr. E. L. Hunsaker of the General Electric Company, Mr. John Talbert of the Wright Aeronautical Corporation, and Mr. John T. Sinnette, Jr., of the NACA to formulate a standard procedure for rating and testing axial-flow compressors.

This report presents the recommendations of the panel for the standardization of the test setup, the instrumentation, the test procedure, the data to be taken, and the presentation of the data. The recommendations cover two types of setup: (1) those setups that use an external drive for the compressor and (2) those setups in which the compressor is an integral part of a gas-turbine engine, jet-propulsion engine, or composite power plant.

Derivations of important equations are presented in an appendix. Air charts obtained from reference 3, but in the notation of this report, are also presented for rapid calculation of compressor performance and thermodynamic states with greater accuracy than is possible with the equations based on normal air given in reference 1.

SYMBOLS

The following symbols are used:

A_2 cross-sectional area in plane of outlet measuring instruments, square feet

a local velocity of sound, feet per second

B_h moisture correction factor for enthalpy as defined on page 17

B_t moisture correction factor for temperature as defined on page 17

c chord of compressor blade

c_f chord of blades in final row

c_p specific heat at constant pressure, Btu per pound per $^{\circ}$F (for dry air at 59° F, 0.2400)

c_v specific heat at constant volume, Btu per pound per $^{\circ}$F (for dry air at 59° F, 0.1715)

D inside diameter of compressor casing at first row of rotor blades

D_{o1} outer diameter of annular passage at inlet measuring station (depression tank)

D_{i1} inner diameter of annular passage at inlet measuring station (depression tank)

D_{o2} outer diameter of annular passage at outlet measuring station

D_{i2} inner diameter of annular passage at outlet measuring station

g standard acceleration of gravity, 32.174 feet per second per second

H enthalpy corresponding to stagnation conditions, Btu per pound

J mechanical equivalent of heat, 778.26 foot-pounds per Btu

l	width of outlet passage after last row of compressor blades, $\frac{D_{o2} - D_{i2}}{2}$
M	Mach number, V/a
m	mass ratio of water vapor to dry air
N	rotor speed, rpm
$N/\sqrt{\theta}$	equivalent rotor speed, rpm (rotor speed corrected to NACA standard sea-level conditions)
n	number of stages
P	absolute total pressure, pounds per square foot
p	absolute static pressure, pounds per square foot
q	dynamic pressure, $\frac{1}{2} \rho V^2$, pounds per square foot
R	Reynolds number, $\rho c V/\mu$
R_d	gas constant for dry air
R_m	gas constant for moist air
r	radius, feet
S	flow area, square feet
T	total temperature, $^{\circ}R$
T_i	indicated or measured temperature, $^{\circ}R$
t	static temperature, $^{\circ}R$
U	rotor speed at tip for first stage, feet per second
U_{i1}	rotor speed at hub for first stage, feet per second
U_{o2}	rotor speed at tip for last stage, feet per second
U_{i2}	rotor speed at hub for last stage, feet per second
V	relative air velocity, feet per second
W	flow rate, pounds per second

$\frac{W\sqrt{\theta}}{\delta}$ equivalent flow rate, pounds per second (flow rate corrected to NACA standard sea-level conditions)

$Y = \frac{T}{t} - 1$

α recovery factor of thermocouple probe as defined on page 15

β angle between absolute air velocity and compressor axis

γ ratio of specific heats, c_p/c_v (for dry air at 59° F, 1.400; for other conditions, fig. 17)

Δ increment of state function for actual process

Δ increment of state function for isentropic process from inlet total pressure and temperature to outlet total pressure

δ ratio of inlet-air total pressure to NACA standard sea-level pressure (2116.2 lb/sq ft)

η adiabatic efficiency (p. 13)

η_s adiabatic shaft efficiency

η_T adiabatic temperature-rise efficiency

θ ratio of inlet-air total temperature to NACA standard sea-level temperature (518.6° R)

Π relative pressure function for dry air

$$\left(\log_e \Pi = \frac{J}{R_d}\int_{400}^{T} \frac{c_p dT}{T}\right)$$

ρ mass density, slugs per cubic foot

ψ_m mean pressure coefficient per stage

$$\left[\psi_m = \frac{J\,\Delta H}{\frac{n}{2}\left(\frac{U_{m1}^2}{2g} + \frac{U_{m2}^2}{2g}\right)}\right]$$

where

$$U_{m1} = \sqrt{1/2\left(U_{i1}^2 + U^2\right)}$$

$$U_{m2} = \sqrt{1/2\left(U_{i2}^{2} + U_{o2}^{2}\right)}$$

Subscripts:

1	inlet measuring station, station 1a
2	outlet measuring station, station 2a or 2b
a	average
c	compressor
d	dry air
i	inner diameter
o	outer diameter
m	on ψ and U indicates mean values as defined under ψ_m and U_m
m	on H, R, and T indicates values for moist air
max	maximum
min	minimum
s	on state properties indicates value for isentropic process

TEST SETUP

Setup with Compressor Externally Driven

When the compressor is externally driven by a motor or other device that is not an integral part of the power plant for which the compressor was designed, a more elaborate setup can be made in order to insure the greatest accuracy of the test results. The following setup is recommended:

Inlet. - An inlet depression tank designed to insure smooth entry of the air into the compressor should be placed immediately ahead of the compressor. Even when air is taken directly from the room, a depression tank is recommended in order to obtain the desired accuracy of inlet-temperature measurement and freedom from rotation. A suitable inlet tank is shown in figure 1(a). A tank diameter D_{o1} at least three times the compressor diameter D is recommended although a diameter as small as 1.5 times the compressor diameter is

permissible. A bellmouthed inlet should be provided to insure a smooth transition into the compressor inlet. The diameter of this bellmouthed inlet should be at least 2 compressor diameters if the tank diameter is greater than 2 compressor diameters or should extend to the wall of the tank if the tank diameter is 2 compressor diameters or less. Compressor-inlet measurements should be made approximately 1 compressor diameter ahead of the bellmouthed inlet (station 1a). The dynamic pressure and temperature should be uniform at this station within the following limits:

$$\frac{(q_{max} - q_a)}{q_a} < 0.01\left(\frac{D_{o1}}{D}\right)^2$$

$$T_{max} - T_a < 1^\circ\ F$$

At the compressor entrance (station 1b) any rotation should not cause the flow to deviate more than 3° from the axial direction as determined by a survey with a yaw tube. (Suitable yaw tubes for such a survey are described in reference 4.)

In order to secure the required degree of uniformity, one or more reinforced screens and possibly a honeycomb straightener will generally be required. The last screen or straightener should be placed approximately 2 compressor diameters ahead of the bellmouthed inlet. Screens with 50-percent openings, giving a pressure drop of about $2q$, were found to produce the maximum improvement in uniformity of flow (reference 5). Honeycomb straighteners may be required if appreciable rotation has been introduced by elbows, unsymmetrical throttles, and so forth, ahead of the tank. The straightening requirements as well as the losses can be reduced to a minimum with an axial approach to the tank and the use of a symmetrical throttle and a diffuser with a small divergence angle (7° optimum). A diffuser with fairly large divergence angles (20°) may be used without serious difficulties if a screen is placed at the downstream end to stabilize the flow (reference 6).

The preceding description assumes that the compressor is driven from the outlet end. In some cases it may be necessary or more convenient to drive the compressor from the inlet end. In this case, modification of the inlet system as shown in figure 1(b) or 1(c) is recommended. The depression tank should be large enough to enclose whatever driving equipment is required immediately ahead of the compressor and still allow the same flow area through the depression tank as in the setup where the drive is from the outlet end. The air-flow passage ahead of the compressor should be faired to insure smooth flow with a steady decrease in flow area from measuring station 1a to the compressor entrance.

Heat transfer from the drive unit, the gears, and the bearings to the inlet air should be kept below 0.2 percent of the total temperature rise of the compressor at design conditions by making appropriate insulation provisions. (See paragraph on insulation, p. 8.)

Outlet. - The compressor may be tested either with or without a diffuser. Somewhat higher accuracy can be obtained for the performance of the compressor proper when tested without a diffuser because a straight passage can be provided to obtain more accurate outlet measurements. For tests of the compressor without a diffuser, the outlet end should be set up essentially as shown in figure 2(a). The air from the compressor discharges axially into an annular collector through a uniform annular passage that has the same inner and outer diameters as the exit from the last row of blades. The length of the straight passage should be at least 2 blade heights or chords, whichever is larger. The outlet measuring station 2a for all setups is located in this passage 1 blade height or chord (whichever is larger) from the trailing edges of the last blade row. For this reason the walls of the passage must be smooth and, if possible, of constant diameter. (Inserts can be used to build up the passage walls to constant diameter if they taper.)

The discharge collector should be so designed that the static-pressure variations around the circumference at the outlet measuring station 2a are less than 5 percent of the mean dynamic pressure. If an excessive variation of static pressure due to a nonsymmetrical discharge exists, screens may be used to advantage as shown in figure 2(a). Great care must be taken that the flow resistance in these screens and in the outlet ducting does not preclude covering the desired operating range of pressure ratios and flows. The recommended collector dimensions are as follows: a minimum diameter of 2 compressor diameters and a minimum axial length of one-half compressor diameter. Other discharge arrangements such as properly designed scrolls or multiple guide vanes are permissible provided these arrangements achieve the required uniformity of static pressure at the outlet measuring station and do not appreciably restrict the flow and thereby limit the operating range of the tests.

When the compressor is furnished with a diffuser, it is generally desirable also to determine the over-all performance of the compressor and diffuser. A suitable setup for such tests is shown in figure 2(b). In general, a straight section cannot be provided after the last row of compressor blades as in the setup for test without a diffuser but outlet measurements should still be made at 1 blade height or chord, whichever is larger, from the trailing edge of the last row of blades (station 2a). Caution should be used to minimize the effect of this instrumentation on the readings taken at the following station. In

order to obtain the over-all performance of the compressor and diffuser pressure measurements should also be taken just beyond the outlet end of the diffuser (station 2b). The recommendations for the collector and outlet piping are the same as for the setup without a diffuser.

Air facilities. - Whenever possible, altitude-exhaust facilities and refrigerated inlet-air facilities should be used in order to obtain a wide range of air flows and inlet-air pressures and temperatures. In testing compressors at low pressure ratios, a blower in the air-supply system can be used to advantage to obtain a wide range of flow conditions.

Insulation. - A minimum of 2 inches of hair felt (or the equivalent) should be used to lag the compressor and any other parts where heat transfer would affect the indicated performance. If possible, the compressor and collector should be mounted separate from the gear box in order to eliminate heat transfer between these parts.

Setup with Compressor Driven by Its Own Power Plant

When the compressor is tested as an integral part of a complete power plant and thus is driven by a self-contained turbine or engine, the test setup will necessarily be different from that in which the compressor is tested as a separate unit and the range of speed and air flow will be much more limited. The inlet setup should be the same as that used for testing the compressor with an external drive. Modifications of the compressor-discharge passages to increase the accuracy of measurements will be impractical. Compressor outlet measurements should, however, be made 1 blade height or blade chord, whichever is larger, downstream of the last row of compressor blades. Care should be taken to shield properly the discharge thermocouples from radiation from burners. In order that the temperature-rise measurement may be used as a reliable indication of efficiency, the compressor should be lagged. Altitude-exhaust facilities are useful in extending the range of the tests.

INSTRUMENTATION AND MEASUREMENTS

The locations of the measuring stations are shown in figures 1 and 2. The air flow may be measured either before or after the compressor. If any possibility of appreciable air leakage exists, measurements at both the inlet and the outlet should be made as a check. Every effort, however, should be taken to eliminate leakage as it may introduce large errors in the performance results. At reduced inlet pressures these errors may be greatly aggravated. Inlet-air flow may be measured either before or after the depression tank, preferably

before. Air flow may be measured after the depression tank by calibration of the bellmouthed inlet, but sufficient sensitivity will be difficult to obtain over the entire range of the tests.

The instrumentation at the inlet station (station 1a) should consist of two total-pressure rakes located 180° apart, two thermocouple rakes 180° apart and 90° from the pressure rakes, and two wall static-pressure taps 180° apart and 45° from the temperature and pressure rakes. (See fig. 3.) Each total-pressure rake should consist of three total-pressure tubes located to read the pressure at the area center of equal areas. Each thermocouple rake should consist of three thermocouples, each located at the area center of equal areas. Because the air velocity in the depression tank is usually very low, care must be exercised to avoid heat transfer between the room and the thermocouple, especially that transfer caused by a slight leakage of room air through the probe and over the thermocouple junction.

The location of the instruments at the outlet measuring station is shown in figure 4 for the compressor without a diffuser. The required pressures are obtained with four wall static-pressure taps located 90° apart in the outer wall and four in the inner wall. These pressures should be separately read and arithmetically averaged. If the compressor is tested with a diffuser instead of a constant-area discharge section, measurements should be taken at the same axial position (station 2a) but four static-pressure probes may be used instead of wall static taps. These probes should be 90° apart and in the middle of the passage. In addition, four static-pressure probes 90° apart should be placed at the second outlet station (station 2b) in the middle of the passage to obtain over-all compressor-diffuser performance. A suitable static-pressure probe is shown in figure 5. The two openings on opposite sides of this probe are connected to separate manometer tubes and, before readings are taken, the direction of the probe is set to give equal pressures at the two openings.

The temperature at outlet station 2a should be read with six total-temperature probes spirally arranged and located 60° apart, each at the area center of equal annular areas. (See fig. 4.) The outlet thermocouples can be used to read the outlet temperature or connected in series with the inlet thermocouples to read directly the temperature rise. A suitable temperature probe is shown in figure 6. The recovery factor of this probe (based on information received from Pratt & Whitney Aircraft) is shown in figure 7 as a function of the air-stream velocity for air-stream total temperatures equal to the ambient temperature and for air-stream total temperatures approximately 100° F above the ambient temperature. As the range of pressures and temperatures in these tests was small, the separate effects

of Reynolds number, Mach number, and temperature level cannot be determined. As this probe is insensitive to yaw up to approximately ±20°, it should need to be set only once during the tests. Because of the high recovery factor and relatively low temperature error due to heat transfer of this probe, it should be satisfactory in most cases to use a recovery factor of 1. For air temperatures above 500° F or velocities below 300 feet per second, however, it may be necessary to shield the probe and take other precautions to reduce heat transfer by radiation and conduction. (See references 7 and 8.)

ACCURACY OF PERFORMANCE

Because of the large effect of compressor efficiency on over-all performance of a gas-turbine or jet engine, every effort should be made to obtain high accuracy for the performance of compressors for those applications. Accurate shaft efficiencies, which require the accurate measurement of torque input and air-weight flow, are especially valuable although difficult to obtain with existing apparatus and techniques. In order to evaluate the effect of small modifications in the compressor design, it is considered very desirable that the relative accuracy of the final efficiency be within one-fourth percent (that is, the results should be reproducible within this limit). In order to compare the merits of compressors tested on different test stands, it is considered essential that the absolute accuracy of the efficiency be within 1 percent. These accuracies require careful instrumentation and techniques of measurement. Emphasis should be given to investigations leading to the improvement of measuring apparatus and techniques, particularly with regard to the measurement of torque and air flow.

In all reports on compressor performance, careful estimates of the accuracy of the various measurements and of the over-all performance should be included in order that the reader know how reliable the test results are.

TEST METHOD

When an external drive is used, tests should be run from 10 percent of the design rotor speed $N/\sqrt{\theta}$ up to the maximum speed the mechanical design will permit in increments of 10 percent of design $N/\sqrt{\theta}$. Inlet conditions for these tests should correspond approximately to sea-level pressure and room temperature or to the highest pressure that will enable coverage of the speed range with the available power.

Additional tests are desirable at 80, 90, 100, etc., percent of design $N/\sqrt{\theta}$ up to the maximum safe speed at four additional sets of inlet conditions; that is, two additional tests at ambient inlet temperature and reduced inlet pressure, for example, one-half and one-fourth atmospheres, and two additional tests with one of the inlet pressures already used but with inlet temperatures of 0° and -70° F (or as low as practical).

Tests should be run over the whole range of pressure ratios obtainable with the equipment. Where possible, these tests should extend into the windmilling region. When test facilities do not permit operation in this region, the lowest pressure ratio should be determined with the wide-open outlet throttle. Test points should be taken in the surge range whenever it is considered safe to do so. The surge point should be indicated as accurately as possible.

When the compressor is driven by its own power plant, testing will not generally be possible over the entire range recommended for an external drive. Tests should then be run over the entire operable range of the engine.

The desired value of $N/\sqrt{\theta}$ should be maintained throughout the tests within ±0.25 percent by adjusting the speed to compensate for variations in inlet temperature. At each speed and inlet condition, at least 10 test points should be so chosen as to give approximately uniform distribution of the points along the performance curve of pressure ratio against flow but with the points somewhat closer together near peak efficiency.

After a test point has been set, test conditions should be allowed to stabilize before data are taken. Stabilized conditions have been reached when the outlet temperature ceases to change, a process that usually takes 5 to 15 minutes. Because axial-flow compressor performance is sensitive to accumulation of dirt, check tests are desirable at periodic intervals.

PRESENTATION OF DATA

The principal method of presenting compressor performance (fig. 8) should consist of curves of the over-all pressure ratio P_2/P_1 plotted against $W\sqrt{\theta}/\delta$ with percentage of design equivalent speed as a parameter and with efficiency (either temperature-rise η_T or shaft η_s) as contours. Wherever possible both temperature-rise and shaft efficiencies should be presented. The pressure ratio and efficiencies are based on outlet total pressures calculated by the method presented in

appendix A. A table on the curve sheet should list $N/\sqrt{\theta}$, percentage of design equivalent speed, and $U/\sqrt{\theta}$. Cross-section paper with 20 lines per inch is recommended for the presentation of all data. Two sheets are permissible for covering the high-speed and low-speed tests separately with a scale suitable to each speed range. A vertical line on the air-flow scale should be used to indicate the theoretical maximum air flow for the particular compressor under consideration. The theoretical maximum air flow is calculated by assuming sonic velocity at the minimum flow area ahead of the first rotor blades. The minimum flow area will usually occur immediately ahead of the first rotor blades and be given by the expression

$$S_{min} = \int_{r_i}^{r_o} 2\pi r \cos \beta dr$$

but for some compressor designs it may occur within the entrance guide vanes. If the flow angle β is known at several points between the inner and outer radius, this integral may readily be evaluated by determining the area under the curve obtained by plotting $2\pi r \cos \beta$ as the ordinate and r as the abscissa. The maximum equivalent weight flow through this area is then given by the expression

$$\left(\frac{W\sqrt{\theta}}{\delta}\right)_{max} = 49.4\ S_{min}$$

(based on dry air with $\gamma = 1.400$). For comparing different compressors, auxiliary scales showing $W\sqrt{\theta}/\delta D^2$ and the root mean pressure ratio per stage $(P_2/P_1)^{1/n}$ should be given. The surge limit should be shown wherever it is clearly defined.

The following secondary methods of presentation are also recommended:

1. Efficiency (temperature-rise and shaft) plotted against $W\sqrt{\theta}/\delta$ for the low-speed range (or the entire range) with percentage of design equivalent speed as a parameter. (See fig. 9.)

2. Efficiency (temperature-rise and shaft) plotted against pressure ratio P_2/P_1 for the high-speed range with percentage of design equivalent speed as a parameter. (See fig. 10.)

3. Pressure coefficient ψ_m plotted against $W\sqrt{\theta}/\delta$ with percent of design equivalent speed as a parameter. (See fig. 11.)

4. Work-input factor ψ_m/η (for temperature-rise and shaft efficiency) plotted against $W\sqrt{\theta}/\delta$ with percentage of design equivalent speed as a parameter. (See fig. 12.)

5. Mach number at the compressor outlet M_2 (calculated by the method given in appendix A) plotted against $W\sqrt{\theta}/\delta$ with percentage of design equivalent speed as a parameter. (See fig. 13.)

The Mach number and the Reynolds number based on the relative velocity at the entrance to the first rotor blades for design speed at maximum efficiency and the test inlet conditions should be given on the principal performance curves. (See fig. 8.) The characteristic length in the Reynolds number should be taken as the appropriate blade chord. Mach and Reynolds numbers should be given for hub, mean, and tip diameters.

Compressor efficiency is defined as the ratio of the work required for isentropic (reversible adiabatic) compression from the inlet total pressure P_1 to outlet total pressure P_2 (as determined in appendix A) to the actual work required; that is

$$\eta = \frac{\text{Isentropic work}}{\text{Actual work}}$$

The work (in Btu/lb) for isentropic compression may be found from the basic air charts (fig. 14); that is

$$\text{Isentropic work} = \Delta H$$

The actual work input (in Btu/lb) may be determined from the measured shaft horsepower and bearing-friction horsepower; thus

$$\text{Actual work} = \frac{550}{J}\,\frac{\text{Shaft horsepower - bearing-friction horsepower}}{W}$$

The actual work input may also be determined approximately from the temperature rise. If heat transfer is neglected,

$$\text{Actual work} = \Delta H = H_2 - H_1$$

The values of H_1 and H_2 can be determined directly from the air charts (fig. 14) for the temperatures T_1 and T_2. Humidity correction can be applied by the use of figure 15. These corrections should be applied when m exceeds 0.01. (Failure to correct for moisture introduces an error of about 0.3 percent for $m = 0.01$.) The methods

of calculation, both with and without humidity corrections, are illustrated by examples in appendix B. The value of m as a function of temperature for saturated air at standard sea-level pressure is given in figure 16. The actual value of m for any given inlet condition may be determined (with sufficient accuracy) by multiplying the value obtained from the curve by the relative humidity (at the compressor inlet) and dividing by the relative pressure δ.

For small temperature rise, the specific heat (and hence γ) may be considered constant and therefore the expression for the temperature-rise efficiency can be written

$$\eta_T = \frac{T_1\left[\left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} - 1\right]}{T_2 - T_1}$$

Calculation of efficiency by means of this equation is permissible for temperature rise of less than 200° F provided that a value of γ corresponding to the mean of the inlet and outlet temperatures is used. The correct value of γ to use can be determined from figure 17, which shows γ as a function of the temperature and moisture content m. (For small temperature rise, an error in γ of 0.1 percent leads to an error in efficiency of about 0.25 percent.) The use of air charts is required for large temperature rises and is preferable even for small temperature rises.

Aircraft Engine Research Laboratory,
National Advisory Committee for Aeronautics,
Cleveland, Ohio, April 23, 1946.

APPENDIX A

CALCULATION OF OUTLET TOTAL PRESSURE AND MACH NUMBER

In the past, much uncertainty has been encountered in the determination of the outlet total pressure. In addition to large radial variations, large circumferential variations in total pressure due to the wakes from the stator blades and nonuniformity of blading make the accurate determination of the average total pressure difficult. For most applications some diffusion is required. Because a uniform velocity is desirable and generally gives a greater pressure recovery than a nonuniform velocity with the same average total pressure, it was considered undesirable to credit the compressor for flow energy associated with nonuniformity of velocity. Based on these considerations the following method for calculation of the outlet total pressure P_2 from the continuity equation on the assumption of a uniform outlet velocity in the axial direction has been adopted. On this assumption, the compressor is not credited for nonuniformity of velocity and deviation from axial discharge. The known quantities are the outlet static pressure p_2, the measured outlet temperature T_{i2}, the air weight flow W, the outlet area A_2 normal to the axis, and the recovery factor α of the temperature probe.

From the definition of the recovery factor

$$\alpha = \frac{T_{i2} - t_2}{T_2 - t_2} \tag{1}$$

and the equation

$$T_2 - t_2 = \frac{\gamma - 1}{\gamma} \frac{V_2^2}{2gR_m} \tag{2}$$

$$T_{i2} - t_2 = \frac{\gamma - 1}{\gamma} \frac{V_2^2}{2gR_m} \alpha \tag{3}$$

If Y_2 is defined as

$$Y_2 = \frac{T_2}{t_2} - 1 \tag{4}$$

then equation (2) gives

$$Y_2 = \frac{\gamma - 1}{\gamma} \frac{V_2^2}{2gR_m t_2} \tag{5}$$

By the continuity equation

$$V_2 = \frac{WR_m t_2}{p_2 A_2} \tag{6}$$

the velocity can be eliminated from equations (3) and (5), which gives

$$T_{i2} - t_2 = \frac{(\gamma - 1)\ R_m \alpha W^2}{2\gamma g p_2^{\ 2}\ A_2^{\ 2}}\ t_2^{\ 2} \tag{7}$$

and

$$Y_2 = \frac{(\gamma - 1)\ R_m W^2}{2\gamma g p_2^{\ 2}\ A_2^{\ 2}}\ t_2 \tag{8}$$

If equation (7) is solved for t_2 and the result substituted in equation (8) the following equation is obtained:

$$Y_2 = \frac{\left[-1 + \sqrt{1 + \frac{2(\gamma - 1)R_m}{\gamma g} \frac{\alpha W^2 T_{i2}}{p_2^{\ 2}\ A_2^{\ 2}}}\right]}{2\alpha} \tag{9}$$

The outlet total pressure can then be obtained from the relation

$$P_2 = p_2\ (Y_2 + 1)^{\frac{\gamma}{\gamma-1}} \tag{10}$$

and the outlet Mach number from the relation

$$M_2 = \sqrt{\frac{2Y_2}{\gamma - 1}} \tag{11}$$

In order to permit a rapid determination of P_2/p_2 and M_2, these quantities, based on $R_m = 53.345$ and a mean $\gamma = 1.388$ (corresponding to dry air at 400° F), are plotted in figures 18 and 19, respectively.

APPENDIX B

AIR CHARTS AND EFFICIENCY CALCULATIONS

A description of the air charts obtained from reference 3 and examples showing the method of calculating compressor efficiencies from these charts are presented.

Description of Charts

The basic charts taken from reference 3 are for "pure dry air" defined as possessing the following mass composition:

Oxygen	23.0 percent
Nitrogen	75.6 percent
Argon	1.4 percent

In addition, correction curves are given that provide the adjustments to be applied to the air data to account for the effects of atmospheric water vapor.

The basic air charts (fig. 14) express the relation between temperature, enthalpy, and a relative pressure function Π. (See reference 9 or 10.) The air charts are based on values obtained from reference 9. The tables of reference 9 have been revised and extended (reference 10), but the differences are small and should therefore have a negligible effect on the accuracy of efficiency calculation.

The corrections to be applied to the basic air charts to account for water vapor are shown in figure 15. The correction factors are so determined that

$$\Delta H_m = \Delta H_d \ (1 + B_h)$$

and

$$\Delta T_m = \Delta T_d \ (1 - B_t)$$

Compressor-Efficiency Calculations

The procedure for the calculation of compressor efficiencies is illustrated by the following examples. The first example neglects water-vapor corrections; whereas the second example shows the method of correction for water vapor.

Example 1. - Given: $P_2/P_1 = 6.00$, $T_1 = 540^\circ$ R, $T_2 = 951.3^\circ$ R. Find the compressor efficiency neglecting water-vapor corrections as follows:

A. Isentropic compression

Find ΔH for isentropic compression of dry air as follows: From the air charts (fig. 14), for $T_1 = 540^\circ$ R, find

$$\Pi_1 = 2.862$$

and

$$H_1 = 33.62$$

For an isentropic compression

$$\Pi_2 = (P_2/P_1)\,\Pi_1 = 6 \times 2.862$$

$$= 17.172$$

From the air charts with $\Pi_2 = 17.172$ find

$$H_{2s} = 120.08$$

and thus

$$\Delta H_d = 86.46 \text{ Btu/lb}$$

B. Actual compression

From the air charts for $T_2 = 951.3^\circ$ R find

$$H_2 = 133.53$$

and hence

$$\Delta H = 133.53 - 33.62$$

$$= 99.91 \text{ Btu/lb}$$

C. Adiabatic efficiency

The adiabatic temperature-rise efficiency is

$$\eta_T = \frac{\Delta H}{\Delta H}$$

$$= \frac{86.46}{99.91} = 0.8654$$

Example 2. - Assume the same data as used in example 1 with the additional information that $m = 0.02$ and find the compressor efficiency taking account of water-vapor corrections.

A. Isentropic compression same as example 1

B. Actual compression corrected to dry air

As the air charts are for dry air, the actual temperature rise must be corrected to dry-air conditions before using the charts.

From figure 15, for $m = 0.02$ and $P_2/P_1 = 6$, find $B_t = 0.006$. Then

$$\Delta T_d = \Delta T_m/(1 - B_t)$$

$$= (951.3 - 540.0)/(1 - 0.006)$$

$$= 413.8$$

and hence

$$T_{2d} = 540 + 413.8 = 953.8^{\circ}\ R$$

From the air charts

$$H_{2d} = 134.15\ \text{Btu/lb}$$

and hence

$$\Delta H_d = 134.15 - 33.62$$

$$= 100.53\ \text{Btu/lb}$$

C. Adiabatic temperature-rise efficiency

The adiabatic temperature-rise efficiency is given by

$$\eta_T = \frac{\Delta H_m}{\Delta H_m}$$

As the enthalpy correction factor B_h is assumed to be the same for actual and isentropic compression

$$\eta_T = \frac{\Delta H_m}{\Delta H_m} = \frac{\Delta H_d\ (1 + B_h)}{\Delta H_d\ (1 + B_h)}$$

$$= \frac{\Delta H_d}{\Delta H_d} = \frac{86.46}{100.53}$$

$$= 0.8600$$

D. Adiabatic shaft efficiency

The adiabatic efficiency may be calculated from shaft power measurements. Thus

$$\Delta H_m = \frac{550}{J}\ \frac{\text{Shaft horsepower - bearing-friction horsepower}}{W}$$

The adiabatic shaft efficiency is then given by

$$\eta_s = \frac{\Delta H_m}{\Delta H_m} = \frac{\Delta H_d\ (1 + B_h)}{\Delta H_m}$$

where ΔH_d is the same as for the temperature-rise efficiency and B_h is obtained from figure 15.

REFERENCES

1. NACA Subcommittee on Supercharger Compressors: Standard Procedures for Rating and Testing Centrifugal Compressors. NACA ARR No. E5F13, 1945.

2. NACA Subcommittee on Supercharger Compressors: Standard Method of Graphical Presentation of Centrifugal Compressor Performance. NACA ARR No. E5F13a, 1945.

3. Amorosi, A.: Gas Turbine Gas Charts. Res. Memo. No. 6-44 (Nav-ship 330), Res. Branch, Bur. Ships, Navy Dept., Dec. 1944.

4. Eckert, B.: Experiences with Flow-Direction Instruments. NACA TM No. 969, 1941.

5. Collar, A. R.: The Effect of a Gauze on the Velocity Distribution in a Uniform Duct. R. & M. No. 1867, British A.R.C., 1939.

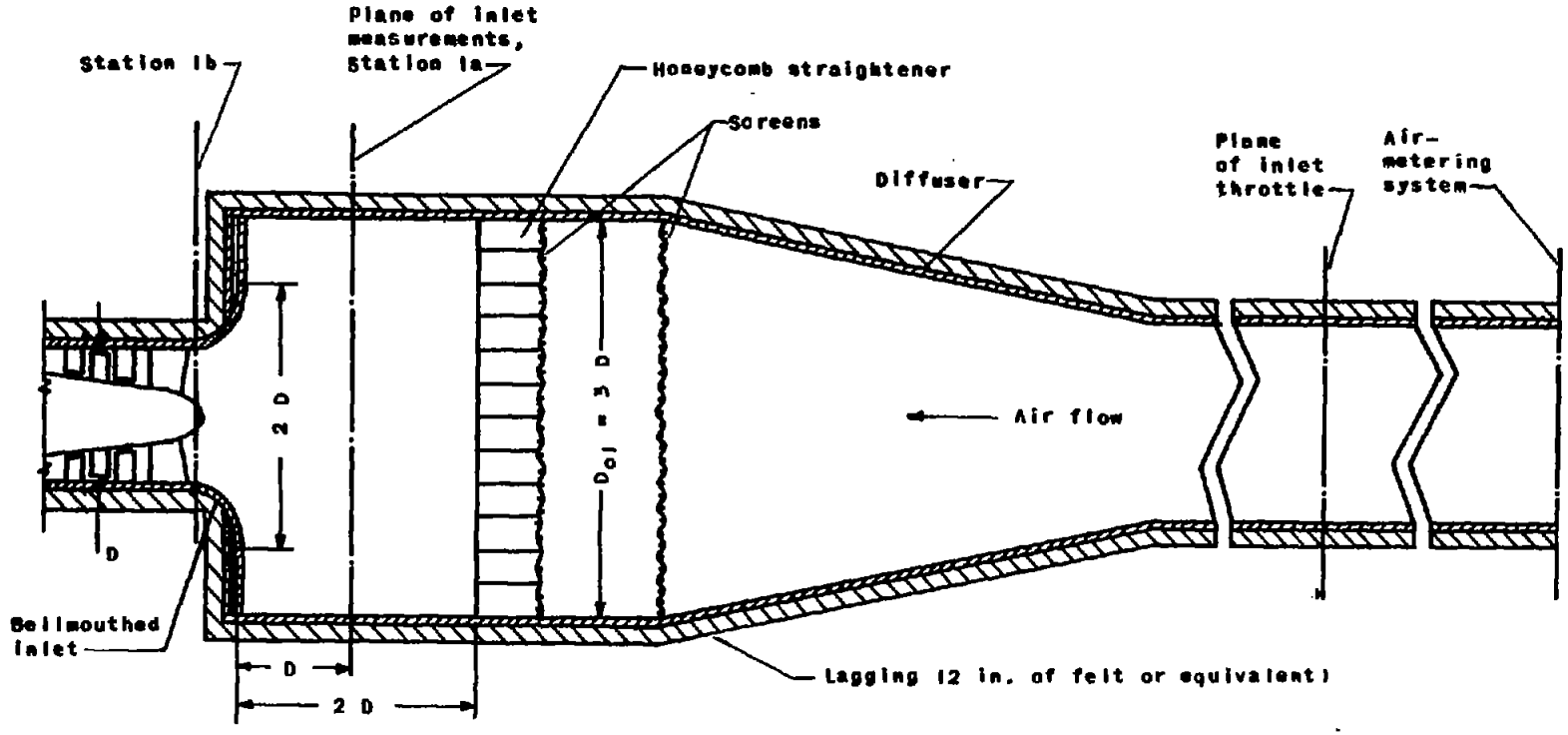
6. McLellan, Charles H., and Nichols, Mark R.: An Investigation of Diffuser-Resistance Combinations in Duct Systems. NACA ARR, Feb. 1942.

7. Hottel, H. C., and Kalitinsky, A.: Temperature Measurements in High Velocity Air Streams. Jour. Appl. Mech., vol. 12, no. 1, March 1945, pp. A-25-A-32.

8. King, W. J.: Measurement of High Temperatures in High-Velocity Gas Streams. A.S.M.E. Trans., vol. 65, no. 5, July 1943, pp. 421-428; discussion, pp. 428-431.

9. Keenan, Joseph H., and Kaye, Joseph: A Table of Thermodynamic Properties of Air. Jour. Appl. Mech., vol. 10, no. 3, Sept. 1943, pp. A-123-A-130.

10. Keenan, Joseph H., and Kaye, J.: Thermodynamic Properties of Air Including Polytropic Functions. John Wiley & Sons, Inc., 1945.



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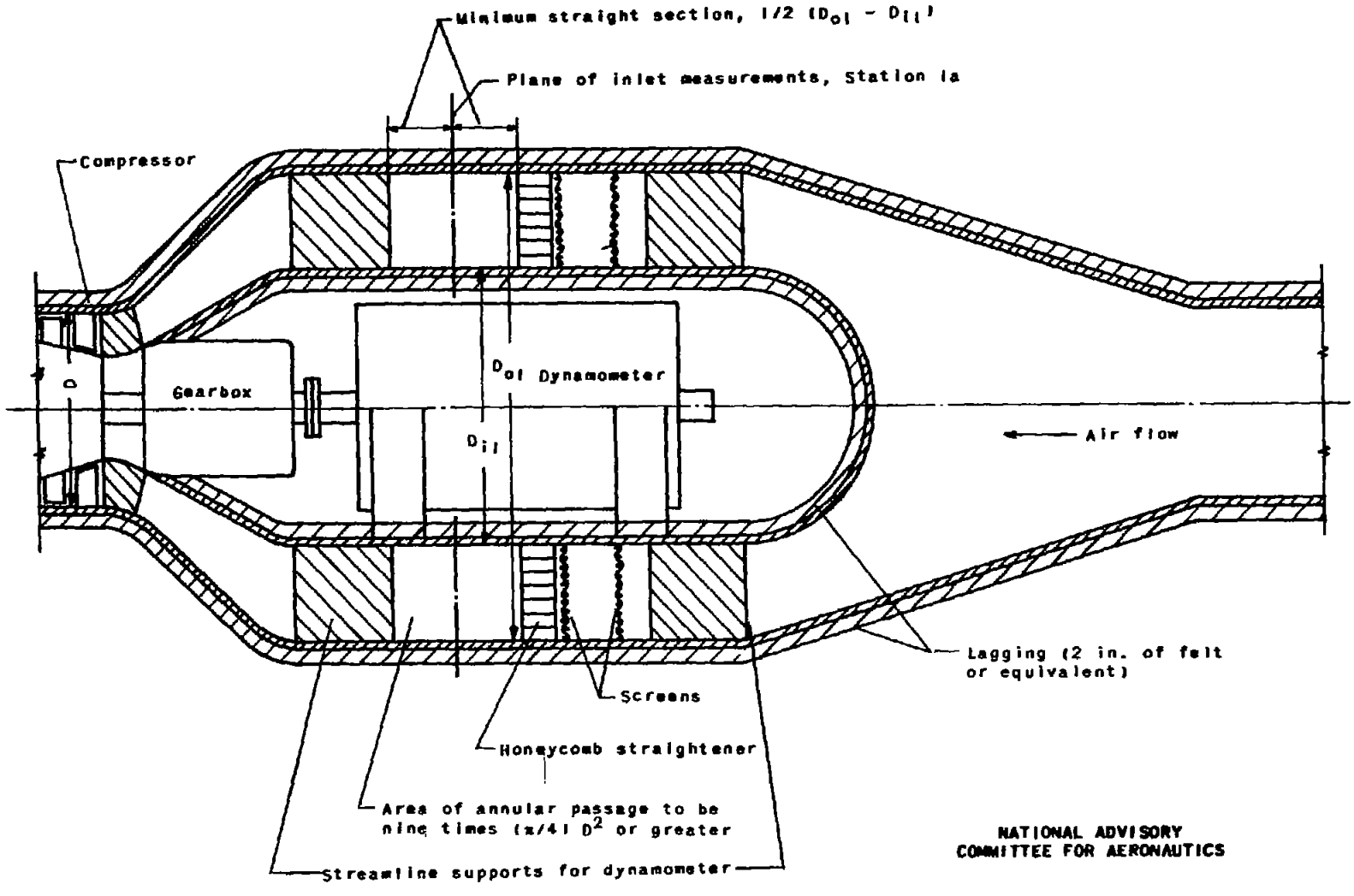
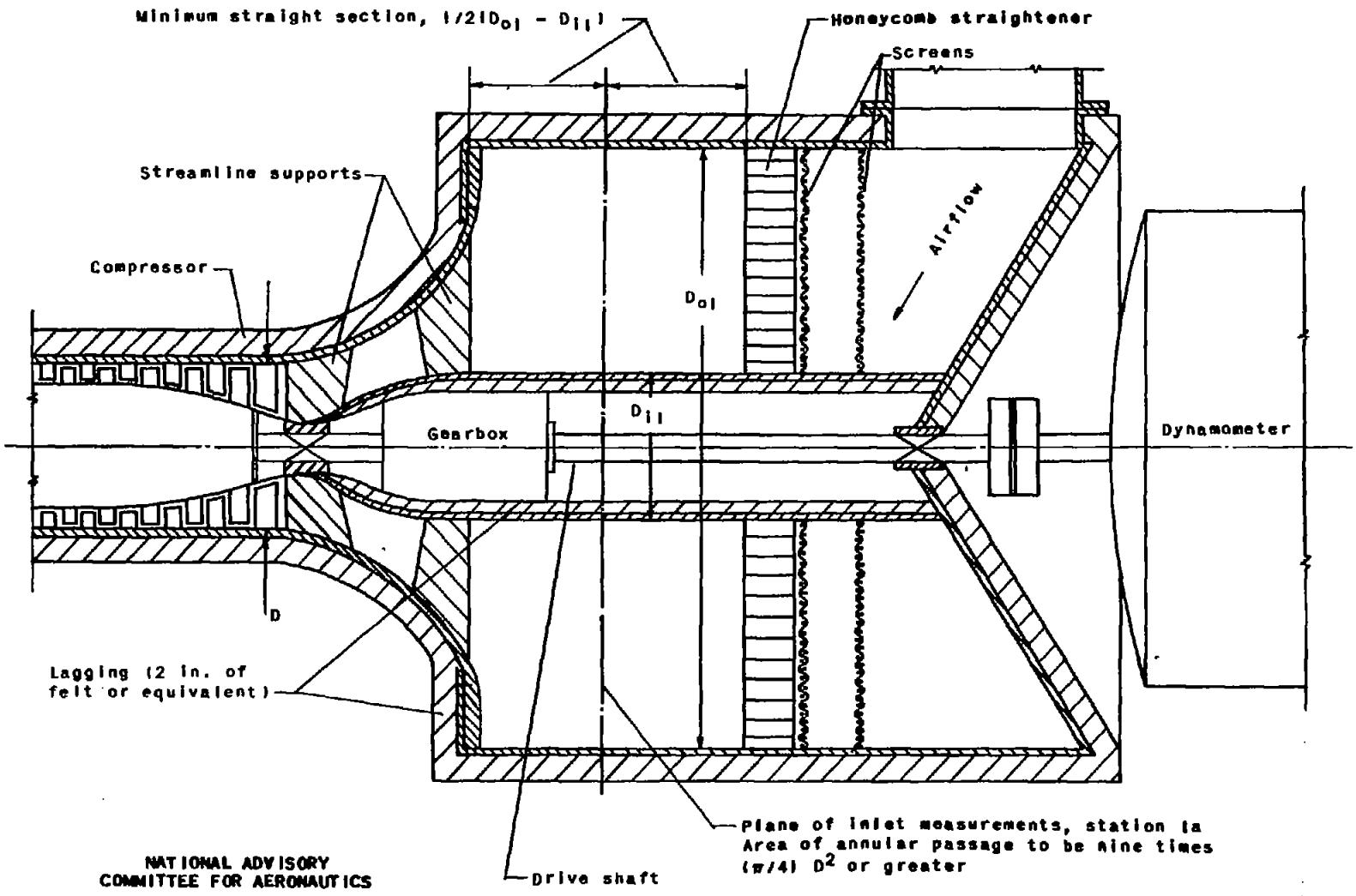
(a) For rear-end drive.

Figure 1. - Setup of inlet depression tank for axial-flow compressor.



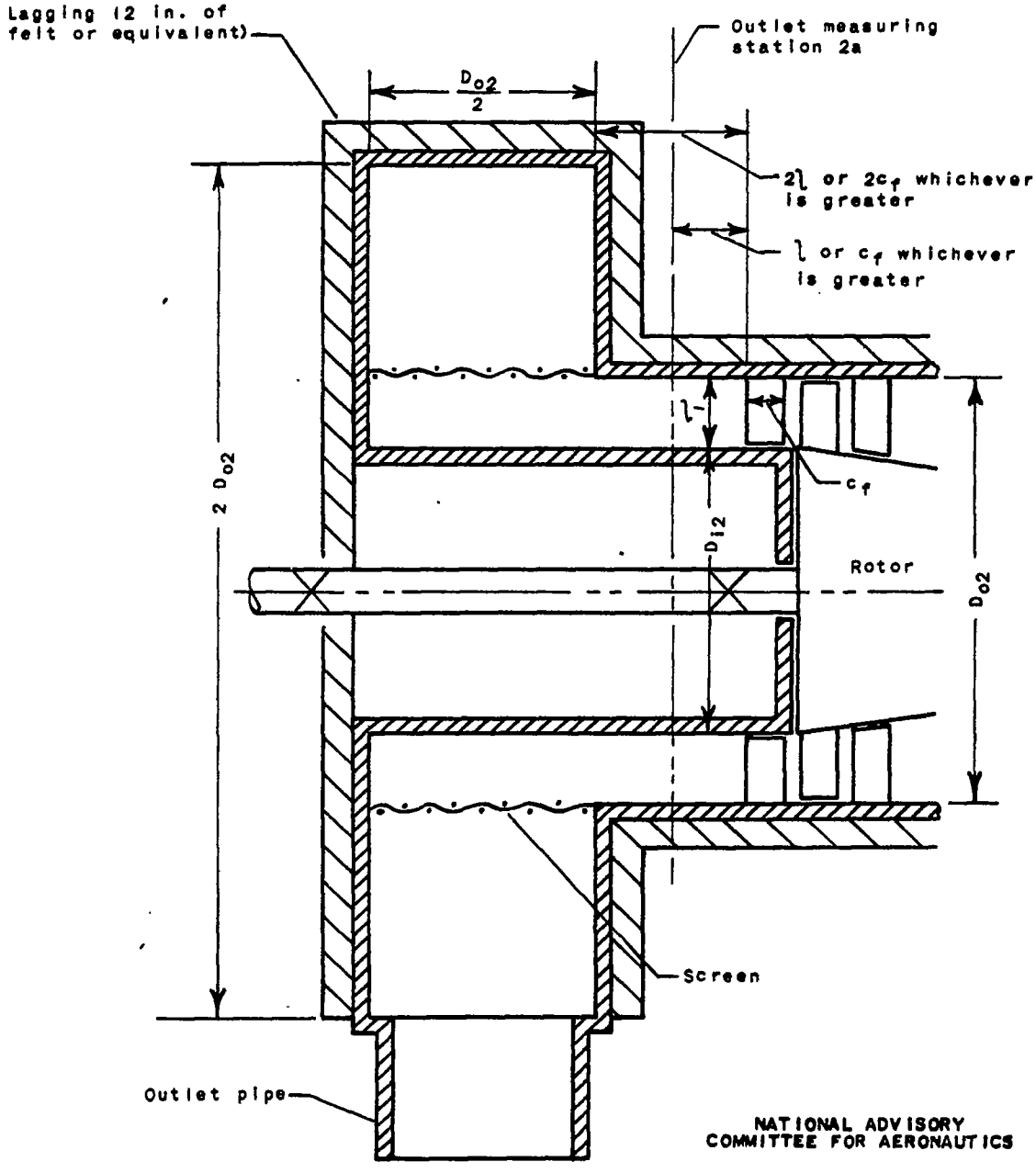
(b) For front-end drive with dynamometer inside depression tank.

Figure 1. - Continued.



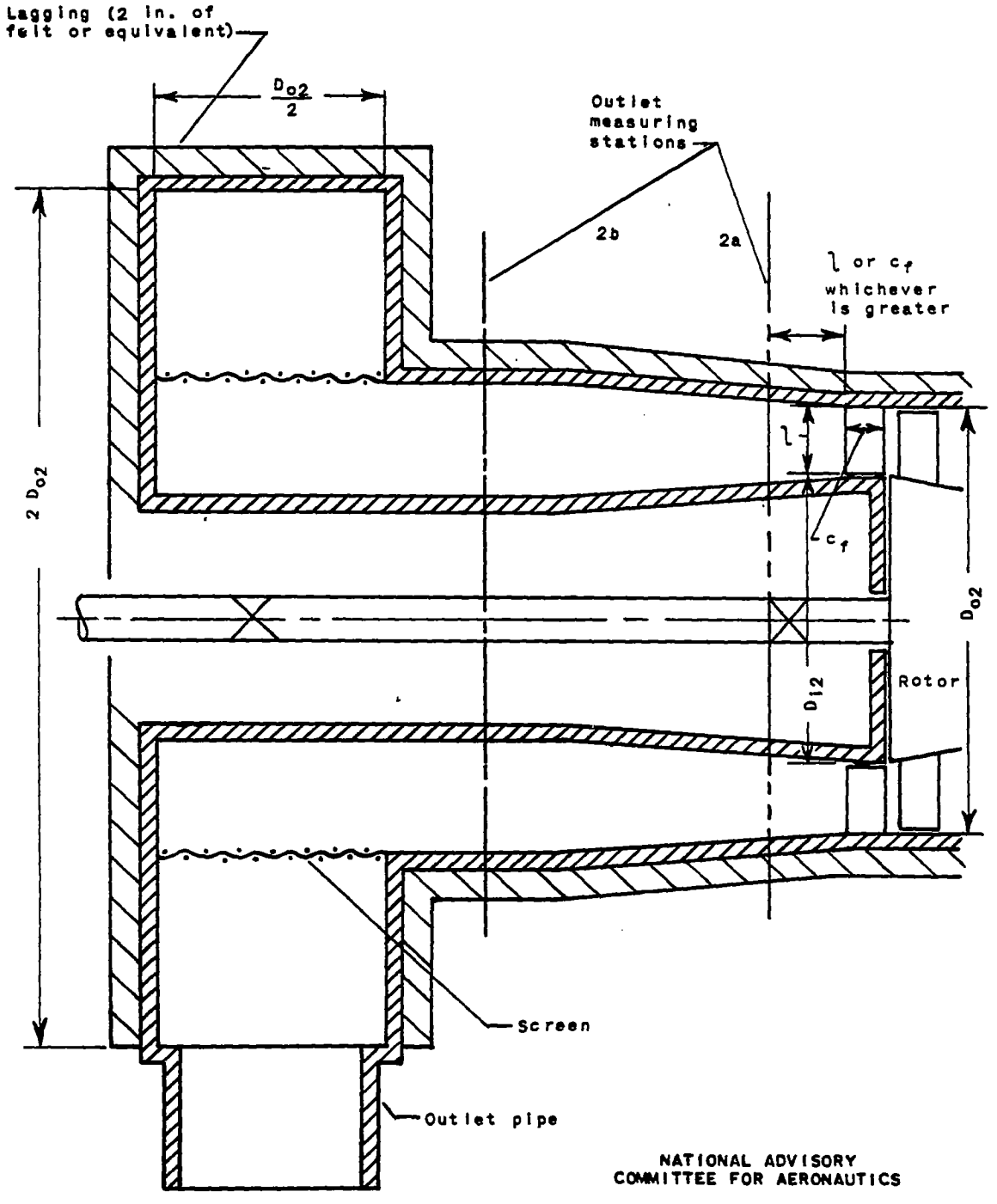
(c) For front-end drive with dynamometer outside depression tank.

Figure 1. - Concluded.



(a) Without diffuser.

Figure 2. - Outlet setup for axial-flow compressor.



(b) With diffuser.

Figure 2. - Concluded.

T Temperature probe
P Total-pressure tube
p Static-pressure tap

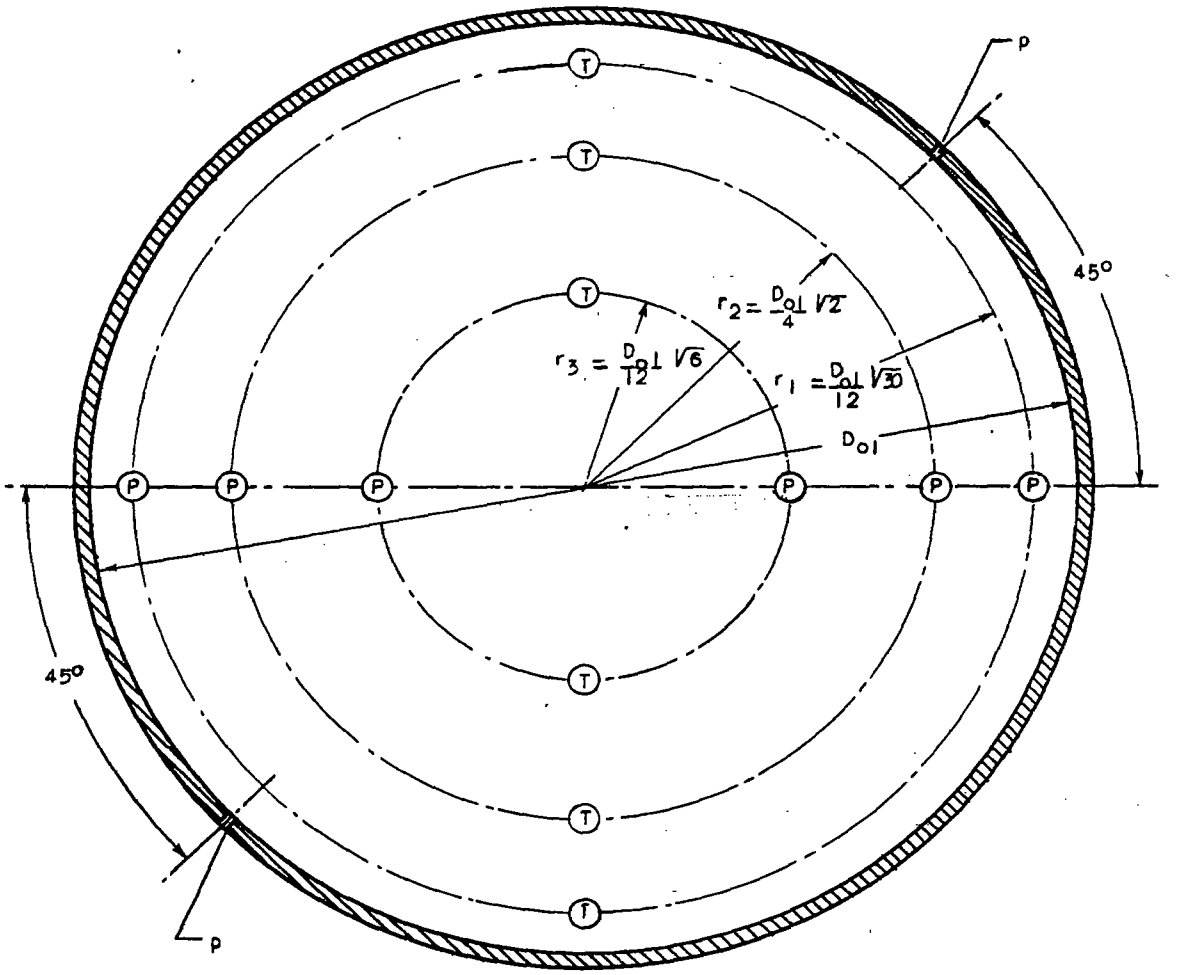


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Figure 3. - Recommended instrumentation of depression tank at inlet to axial-flow compressor (station 1a).

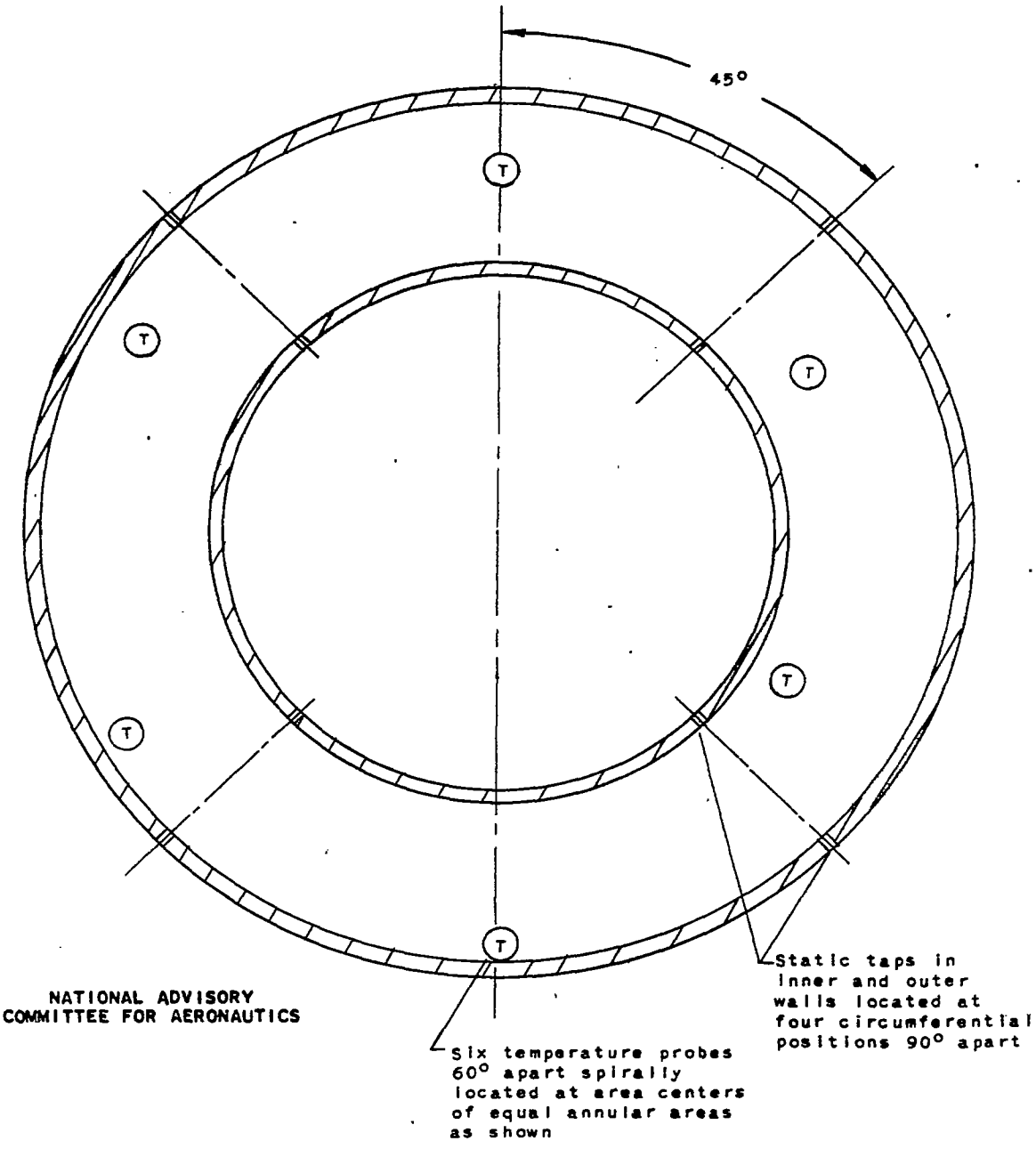

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Figure 4. - Recommended instrumentation of annular outlet passage of axial-flow compressor (station 2a).

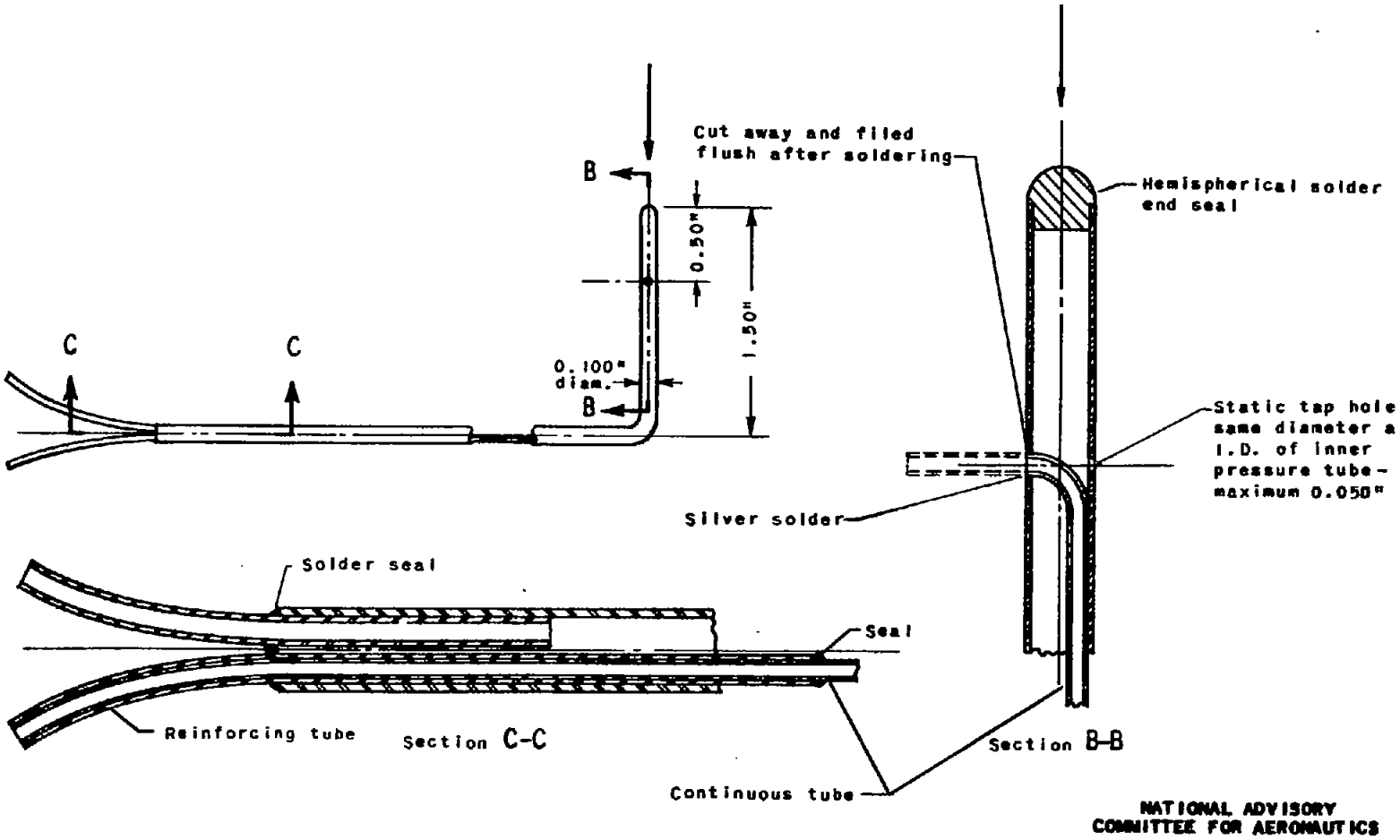

Figure 5. - A suitable static-pressure probe for compressor-outlet measurements.

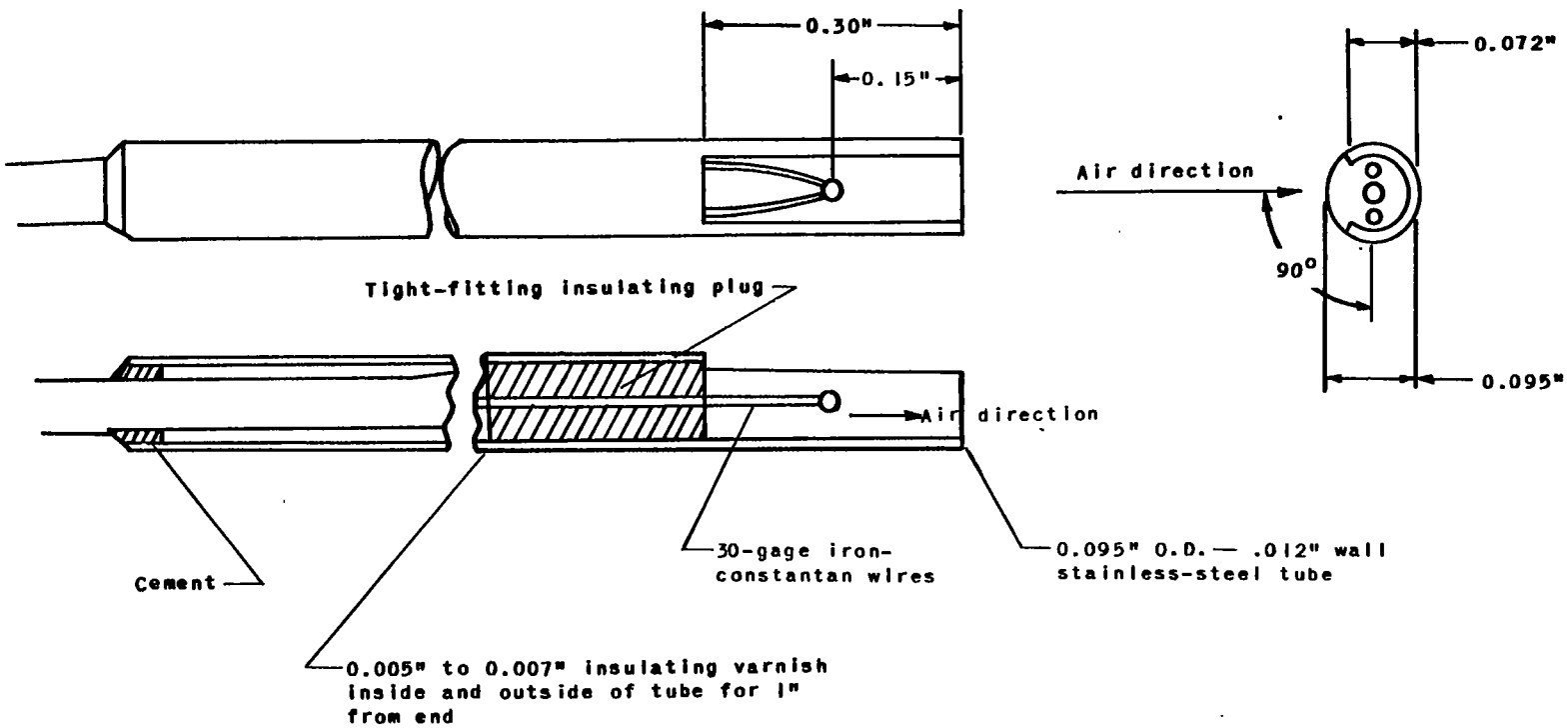

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Figure 6. - Suitable temperature probe for compressor-outlet measurements (based on information received from Pratt & Whitney Aircraft).

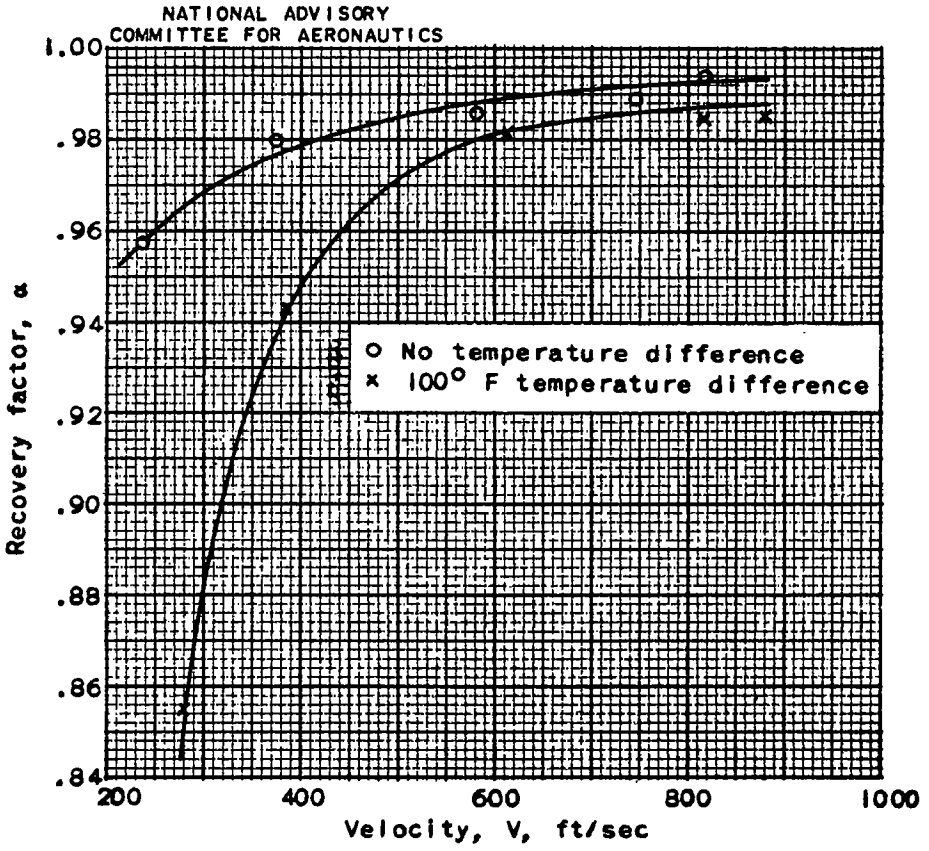
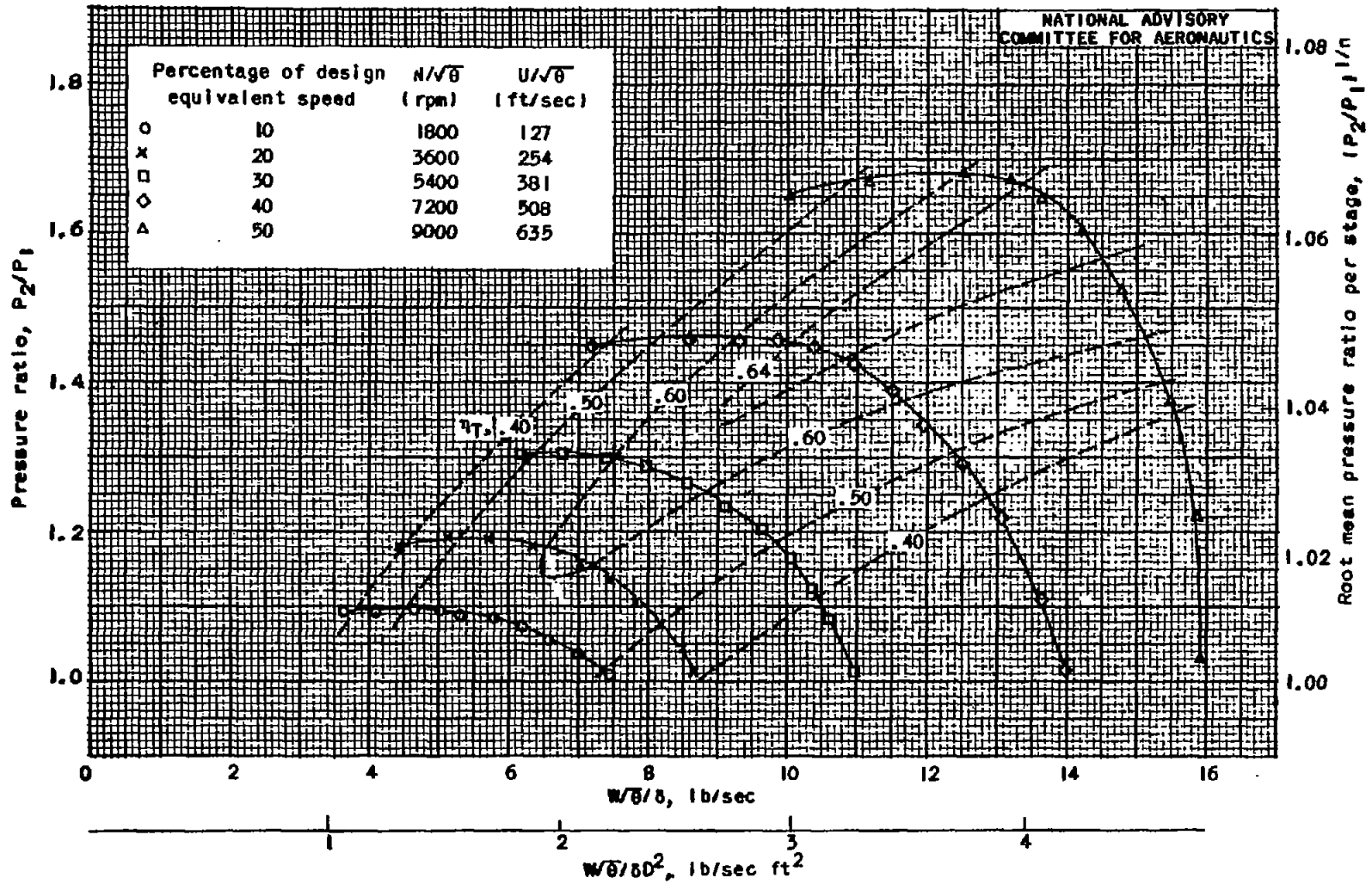
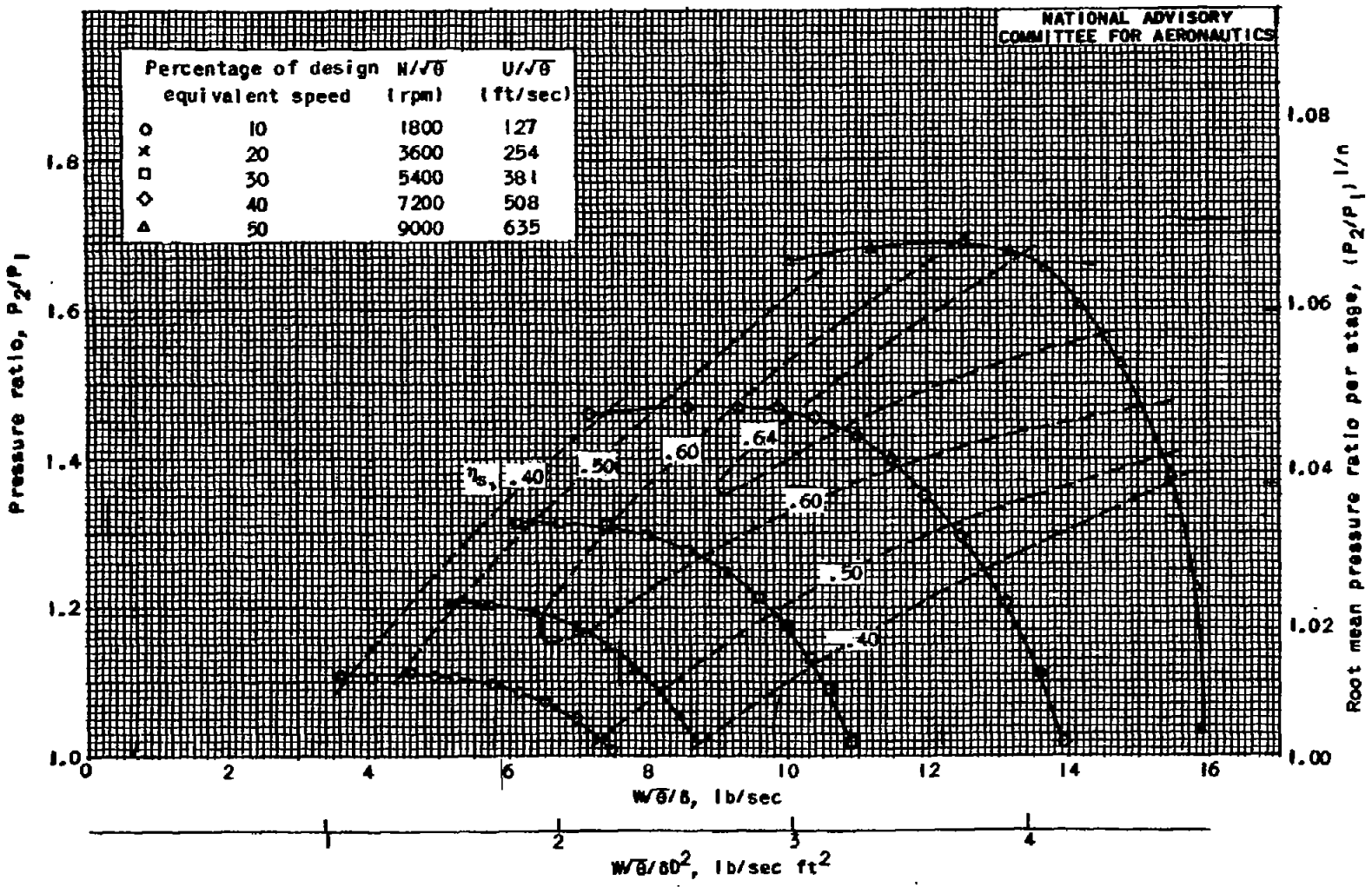


Figure 7. - Recovery factor of recommended temperature probe (fig. 6) for temperature difference of 0° and 100° F between air stream and environment (based on information received from Pratt & Whitney Aircraft).



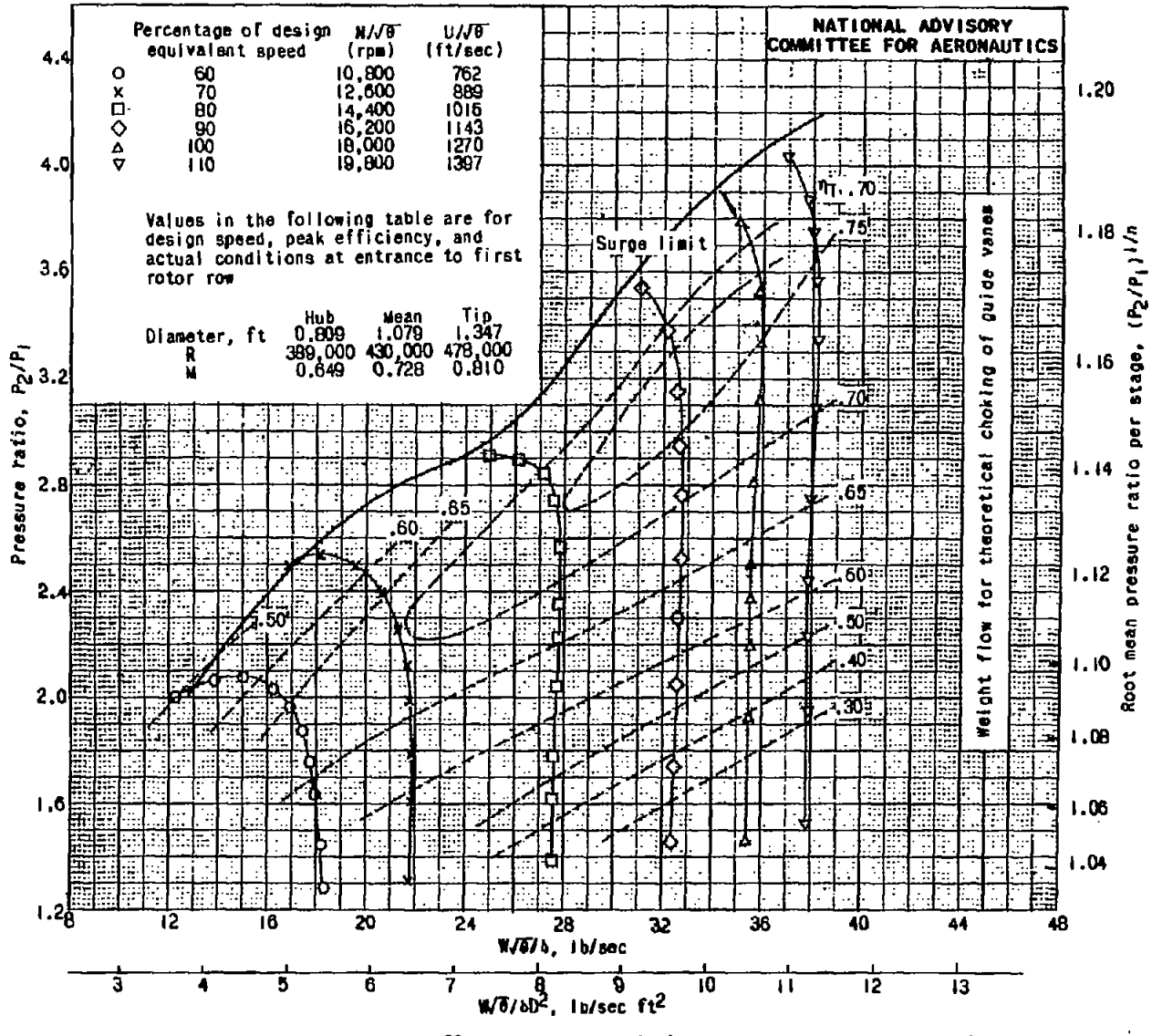
(a) With adiabatic temperature-rise efficiency contours; low percentages of design equivalent speed.

Figure 8. - Representative plot of over-all pressure ratio and root mean pressure ratio per stage against equivalent weight flow for particular inlet condition. No diffuser; inlet total pressure, 2116 pounds per square foot (29.92 in. Hg absolute); inlet total temperature, 518.6° R.



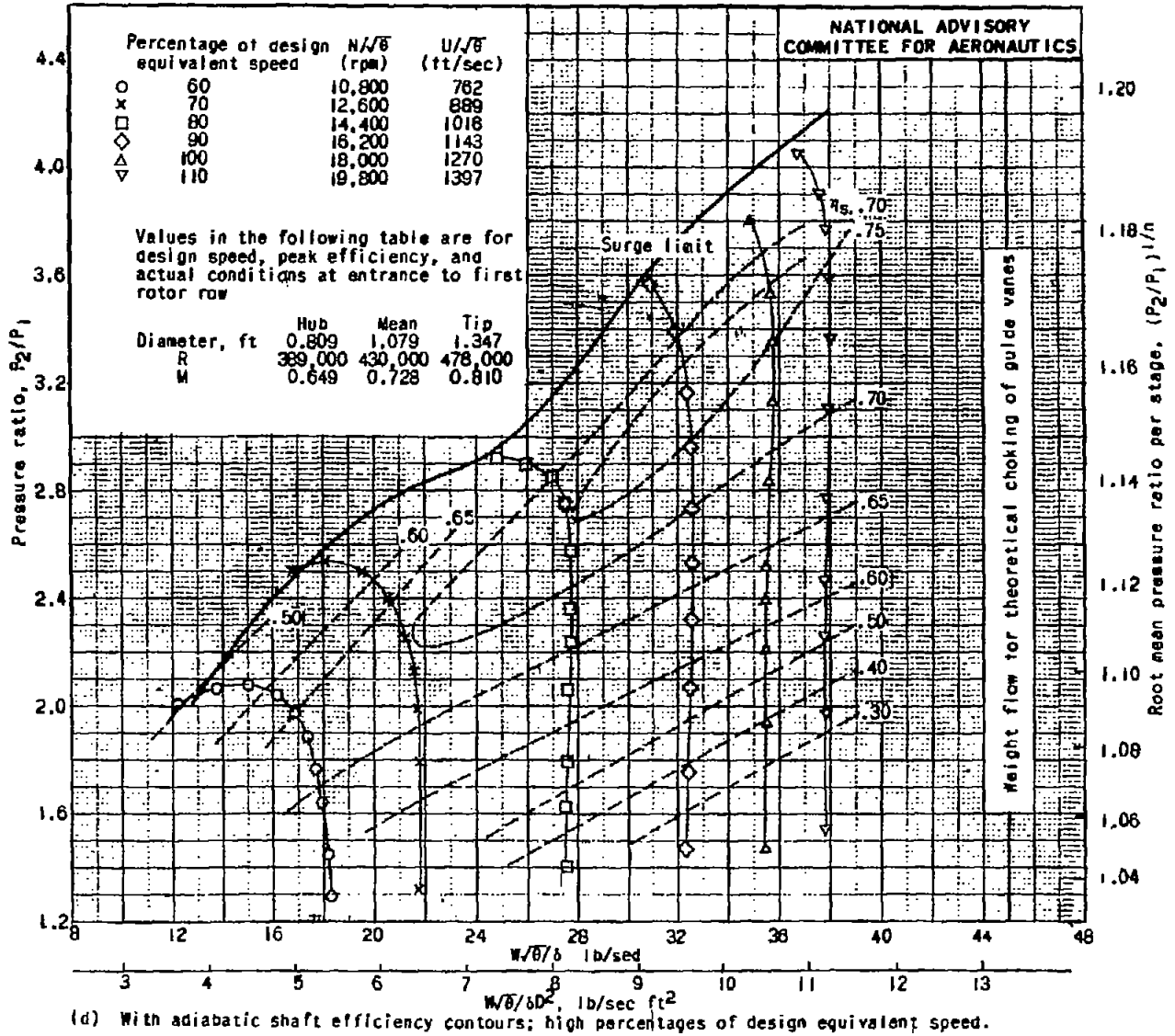
(b) With adiabatic shaft efficiency contours; low percentages of design equivalent speed.

Figure 8. - Continued.



(c) With adiabatic temperature-rise efficiency contours; high percentages of design equivalent speed.

Figure 8. - Continued.



(d) With adiabatic shaft efficiency contours; high percentages of design equivalent speed.

Figure 8. - Concluded.

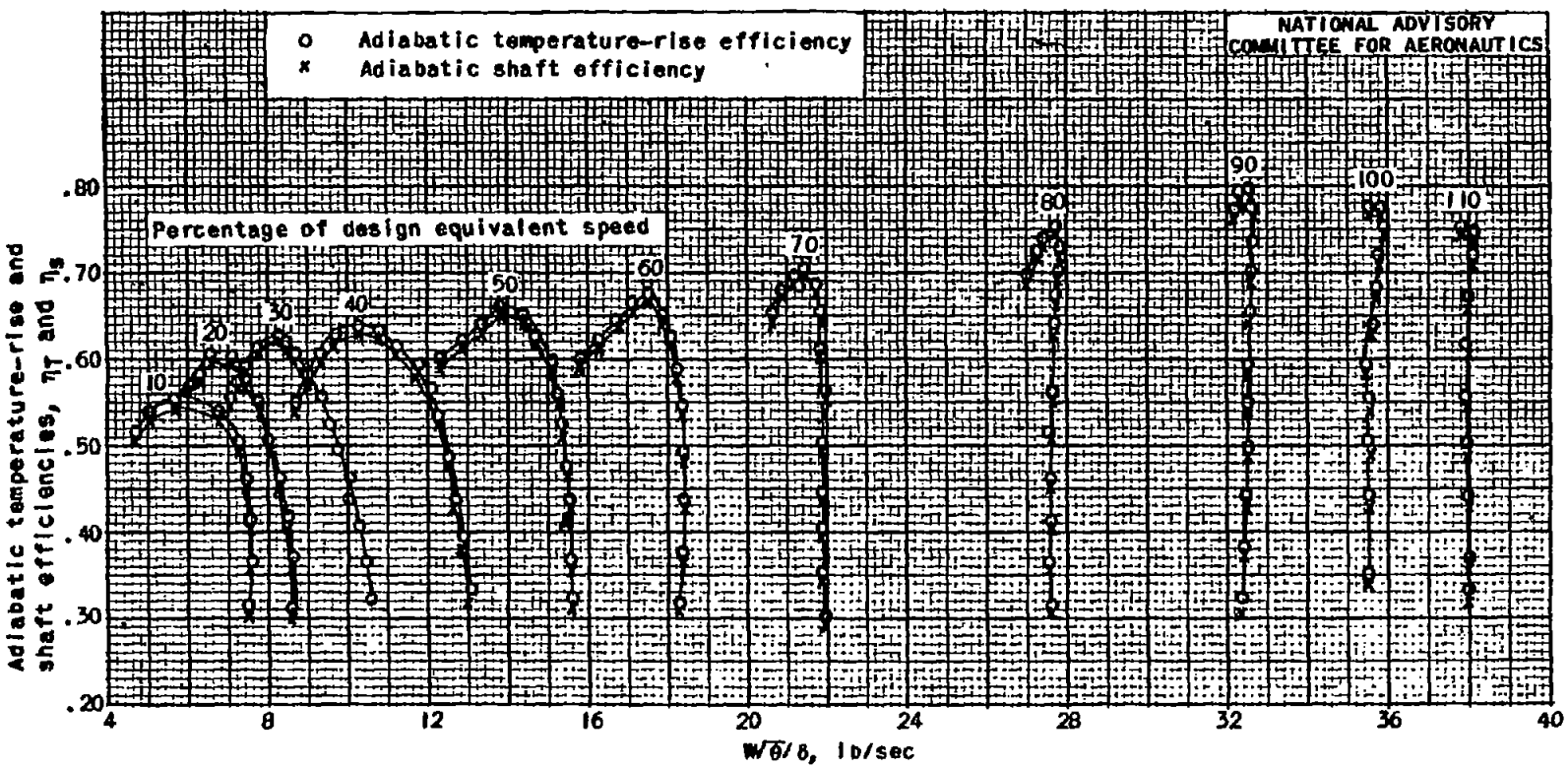


Figure 9. - Representative plot of adiabatic temperature-rise and shaft efficiencies against equivalent weight flow for particular inlet condition. No diffuser; inlet total pressure, 2116 pounds per square foot (29.92 in. Hg absolute); inlet total temperature, 518.6° R.

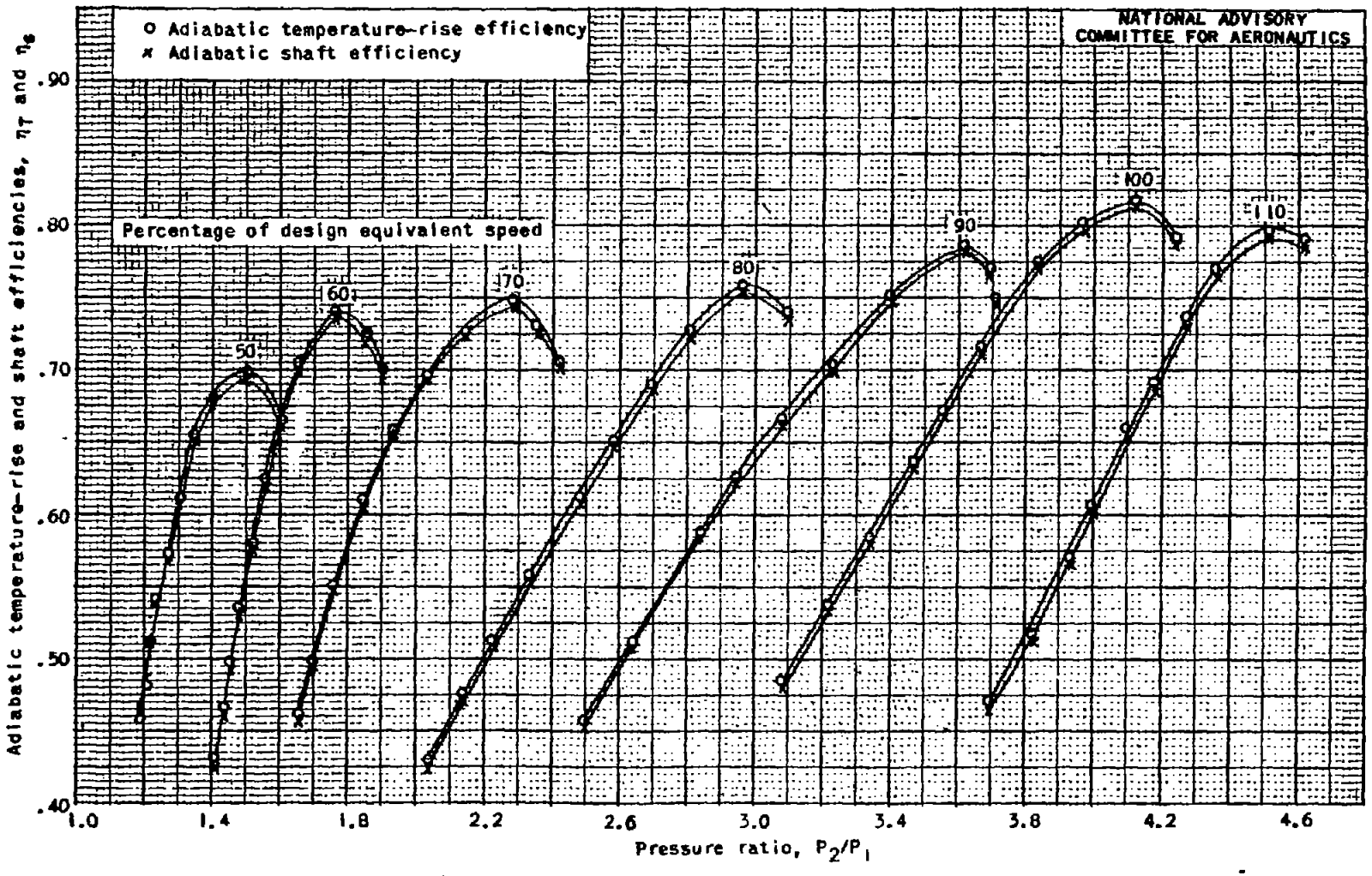


Figure 10. - Representative plot of adiabatic temperature-rise and shaft efficiencies against pressure ratio for particular inlet condition. No diffuser; inlet total pressure, 2116 pounds per square foot (29.92 in. Hg absolute); inlet total temperature, 518.6° R.

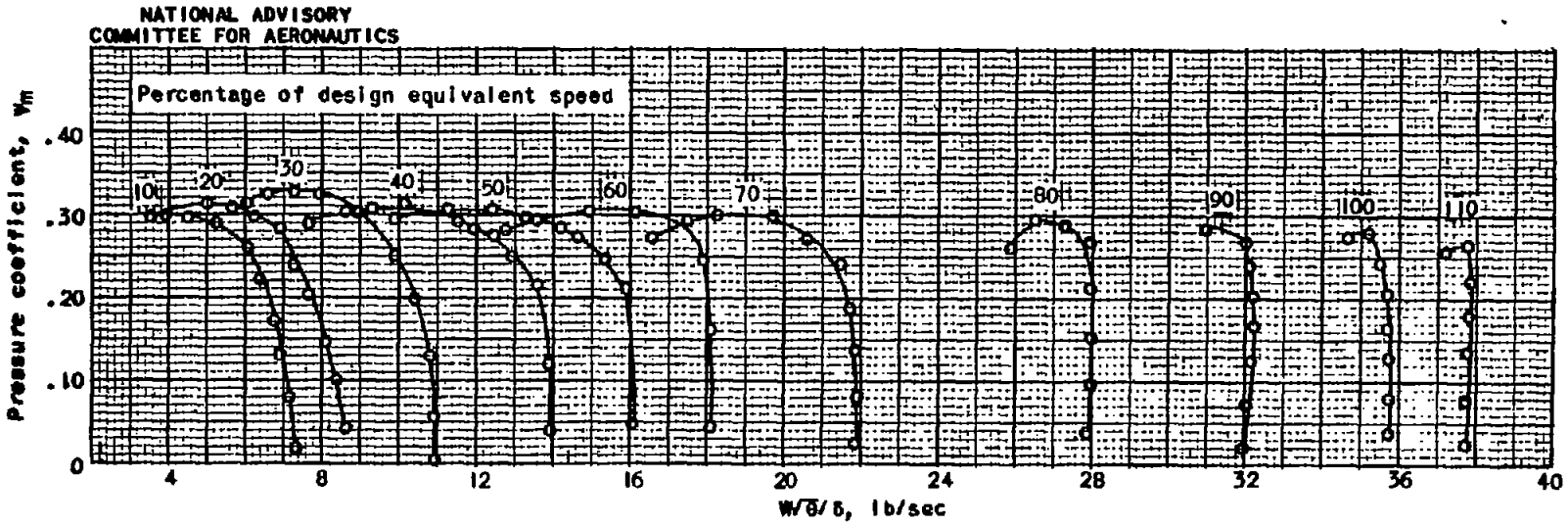
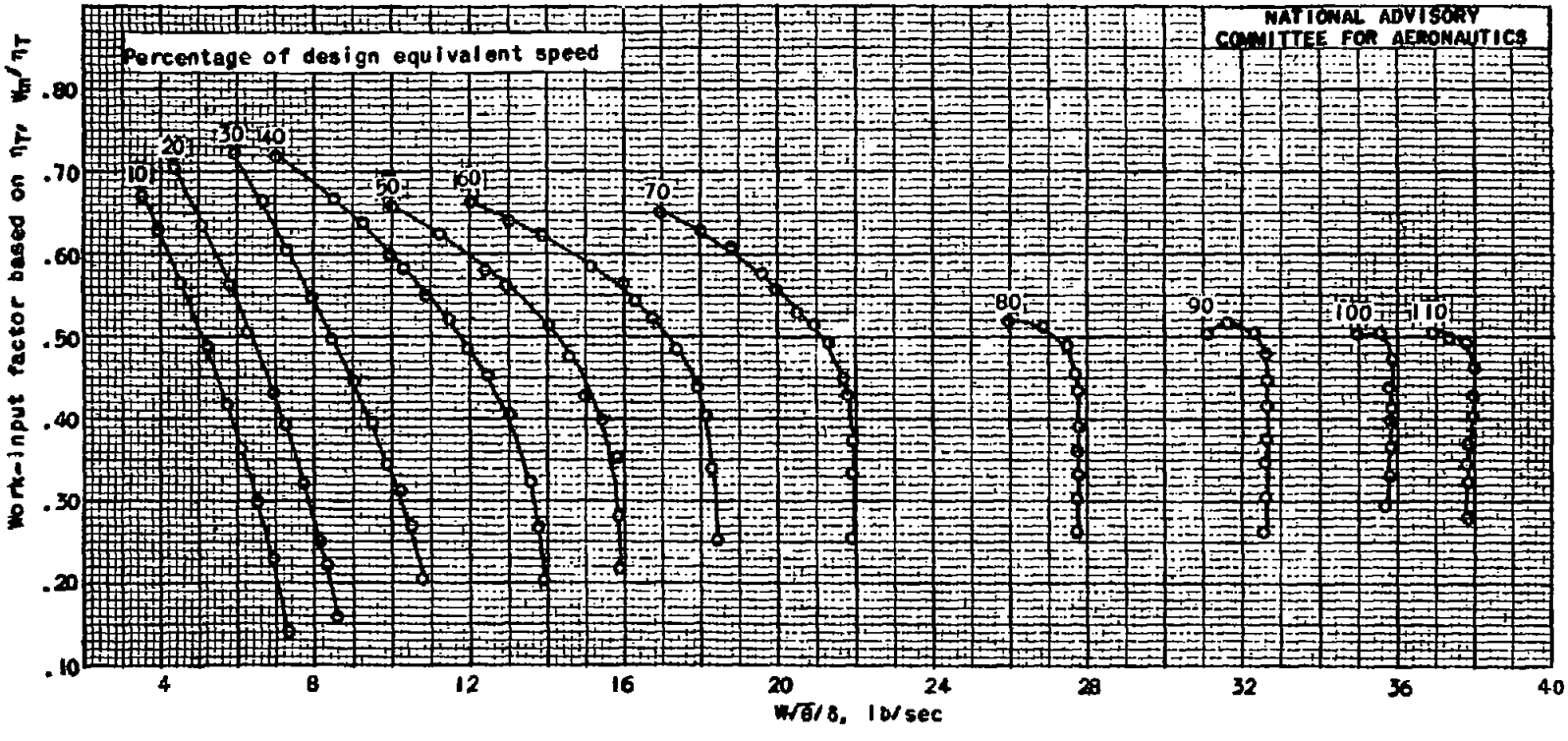
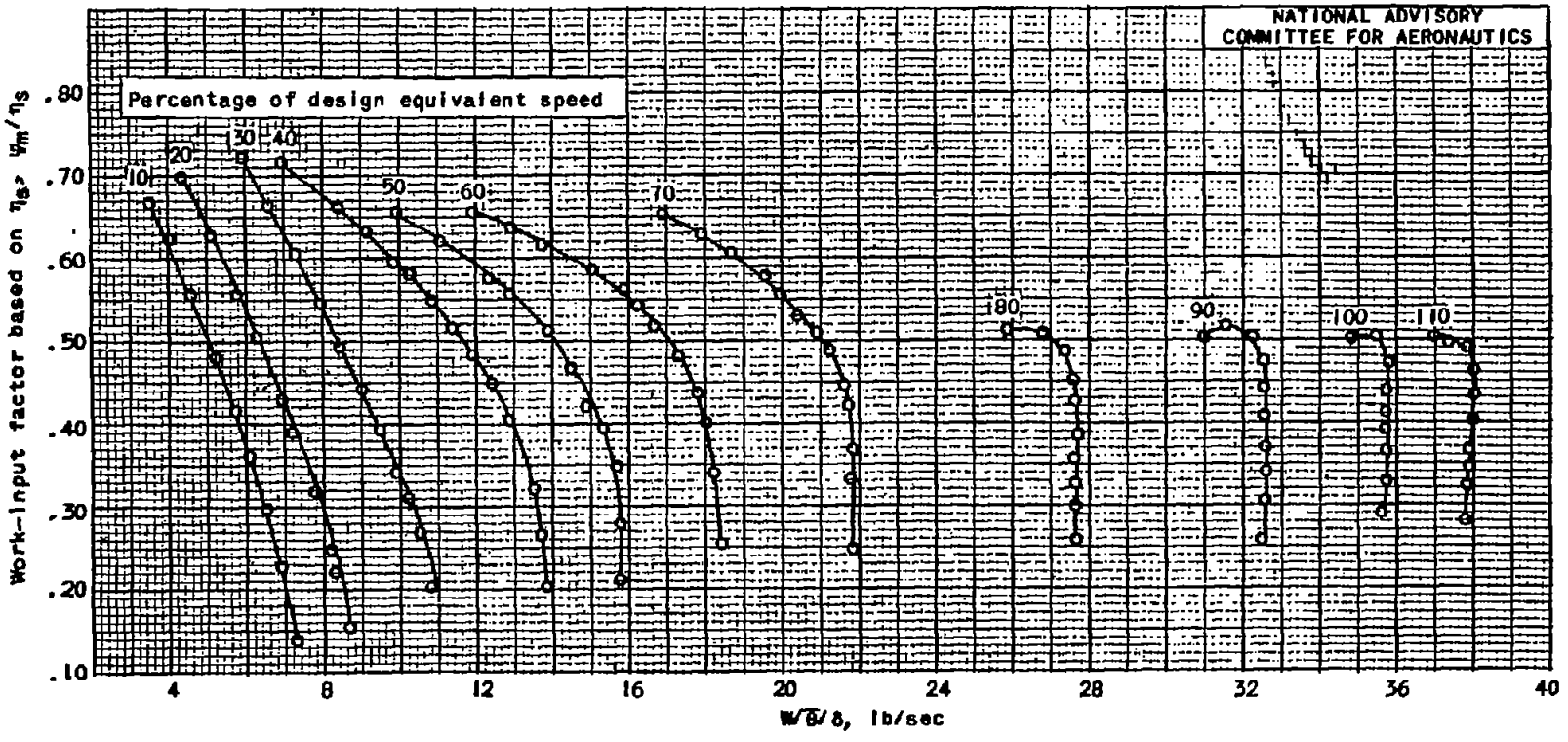


Figure 11. - Representative plot of pressure coefficient against equivalent weight flow for particular inlet condition. No diffuser; inlet total pressure, 2116 pounds per square foot (29.92 in. Hg absolute); inlet total temperature, 518.6° R.



(a) Work-input factor based on adiabatic temperature-rise efficiency, η_T.

Figure 12. - Representative plot of work-input factor plotted against equivalent weight flow. No diffuser; inlet total pressure, 2116 pounds per square foot (29.92 in. Hg absolute); inlet total temperature, 518.6° R.



(b) Work-input factor based on adiabatic shaft efficiency, η_s.

Figure 12. - Concluded.

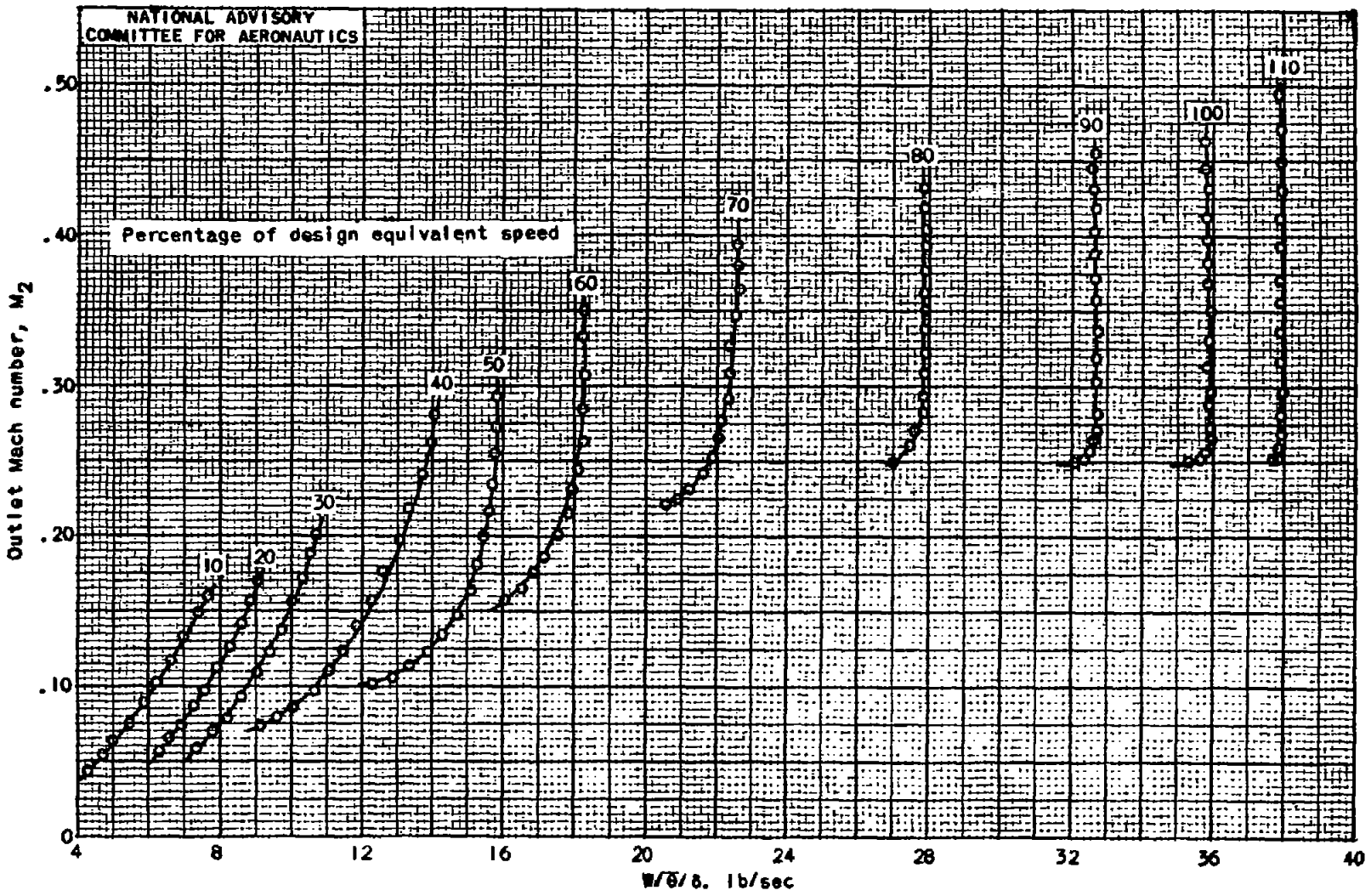


Figure 13. - Representative plot of outlet Mach number against equivalent weight flow for particular inlet condition. No diffuser; inlet total pressure, 2116 pounds per square foot (29.92 in. Hg absolute); inlet total temperature, 518.6° R.

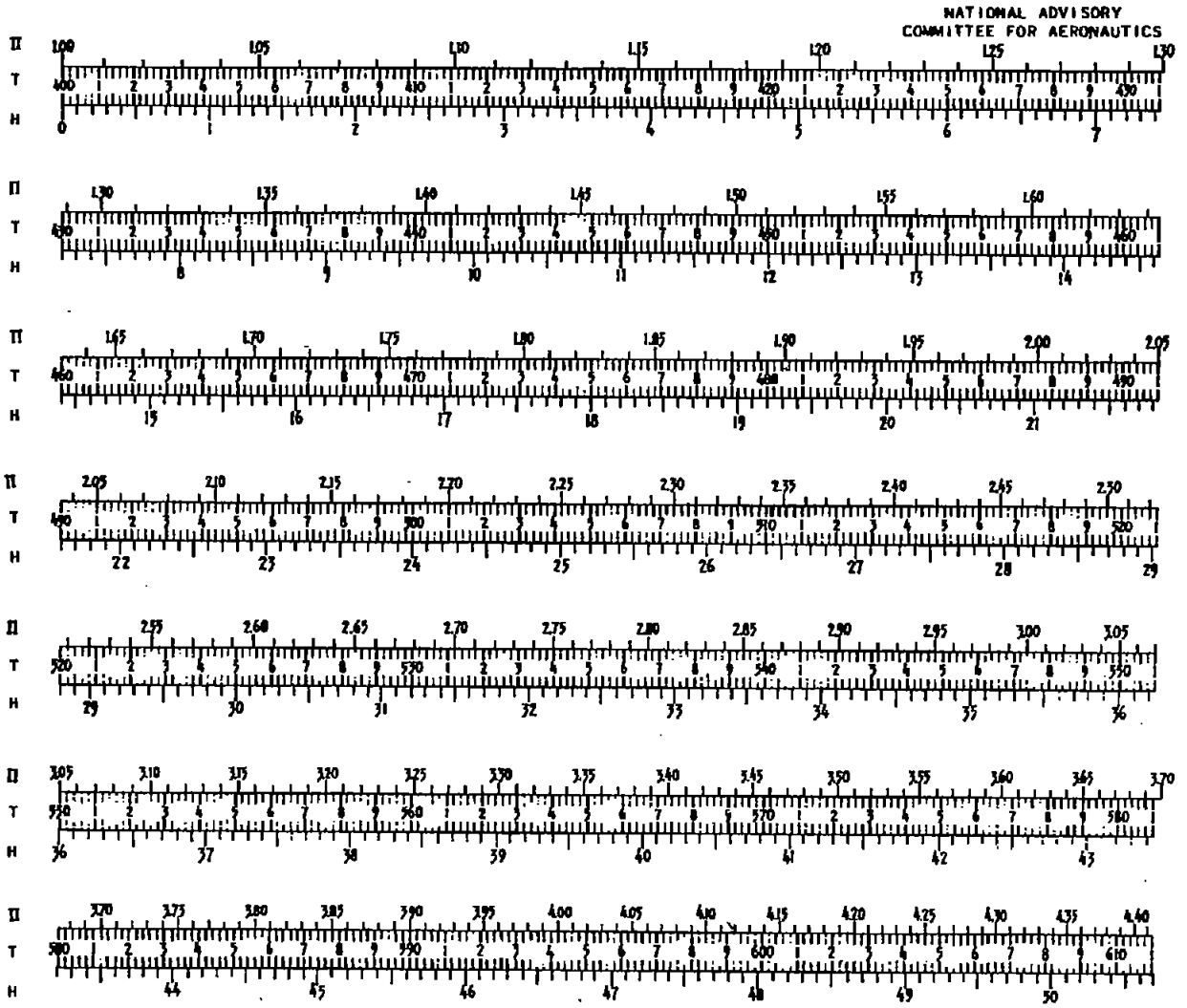


Figure 14. - Properties of air. (This chart is a direct reproduction, except for the notation, from reference 3.)

Π	1.00	to	4.41
T (degrees R)	400	to	611
H (BTU/lb.)	0.0	to	50.7

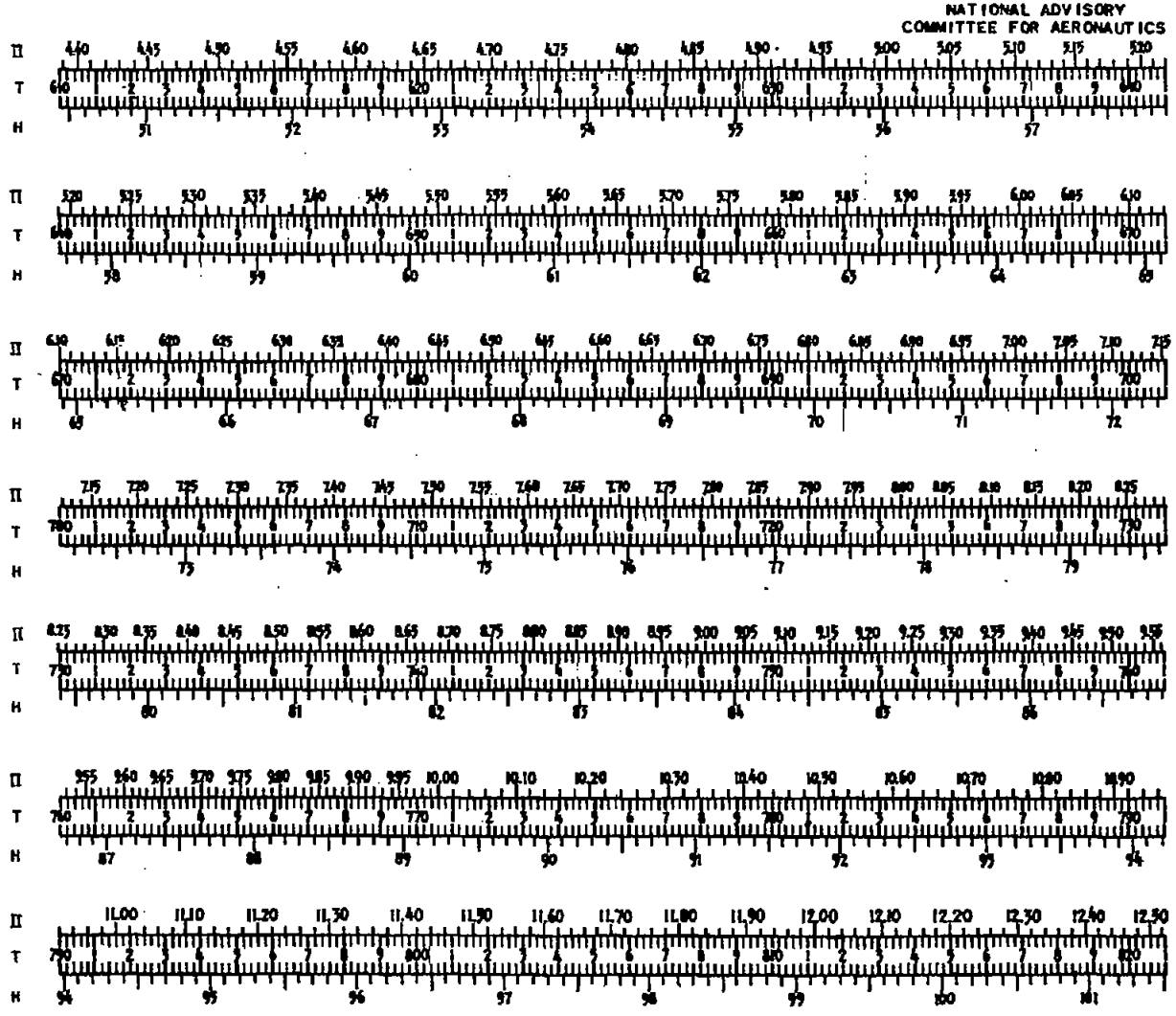


Figure 14. - Continued. (This chart is a direct reproduction, except for the notation, from reference 3.)

Π	4.39	to	12.52
T(degrees R)	610	to	821
H(BTU/lb)	50.5	to	101.5

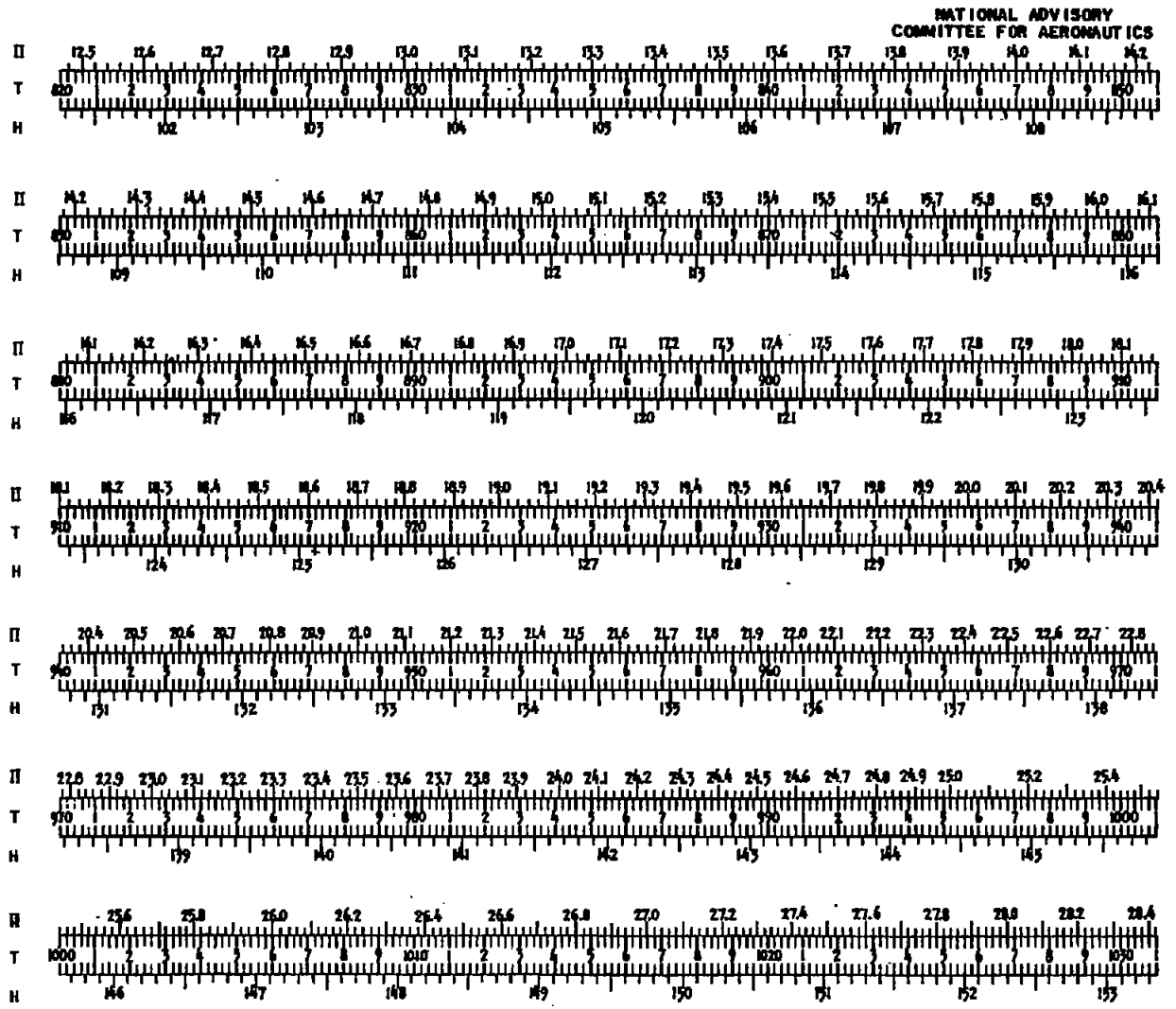


Figure 14. - Continued. (This chart is a direct reproduction, except for the notation, from reference 3.)

Π	12.48	to	28.42
T(degrees R)	820	to	1031
H(BTU/lb.)	101.3	to	153.3

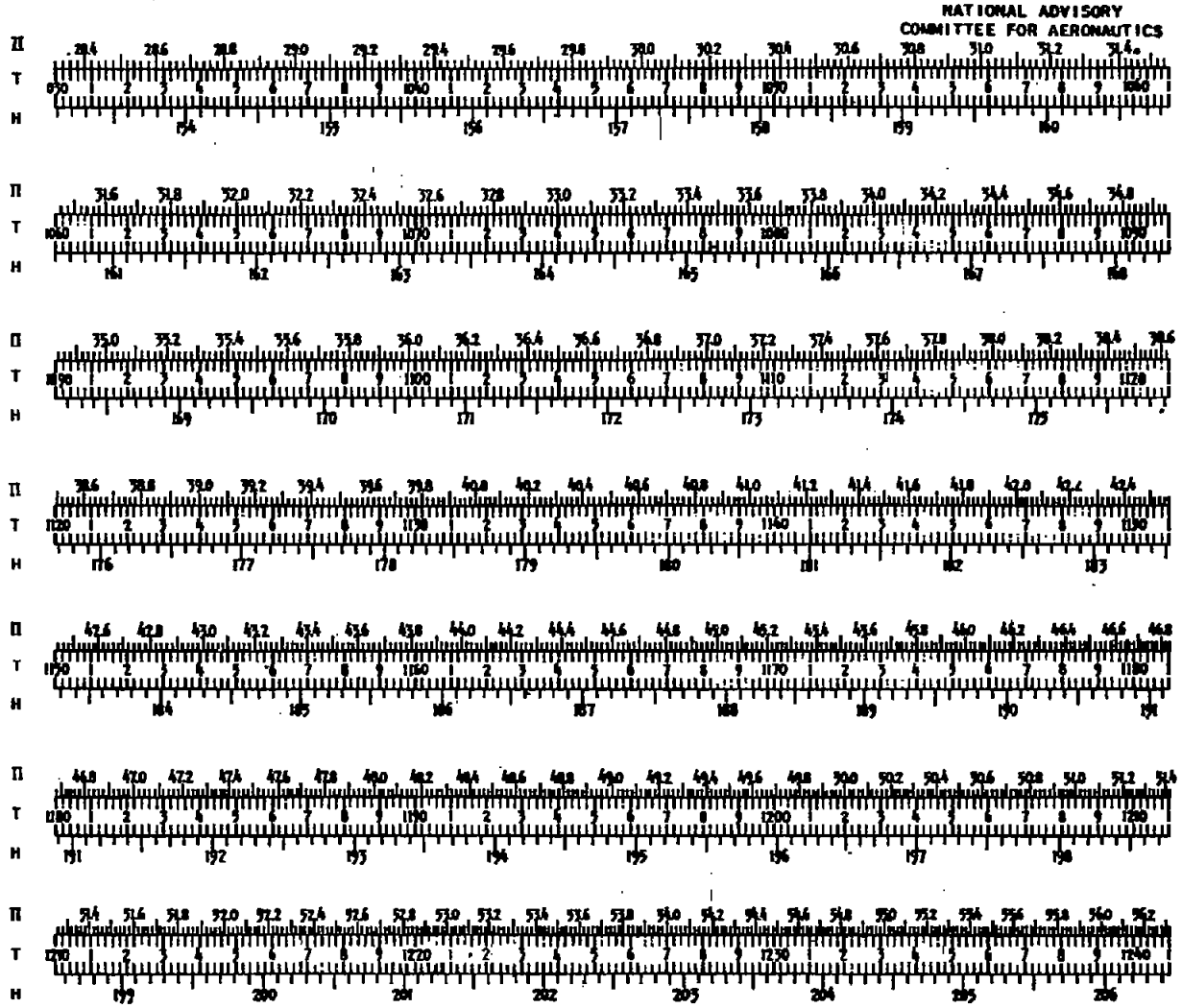


Figure 14. - Continued. (This chart is a direct reproduction, except for the notation, from reference 3.)

Π	28.32	to	56.32
T (degrees R)	1030	to	1241
H (BTU/lb.)	153.1	to	206.4

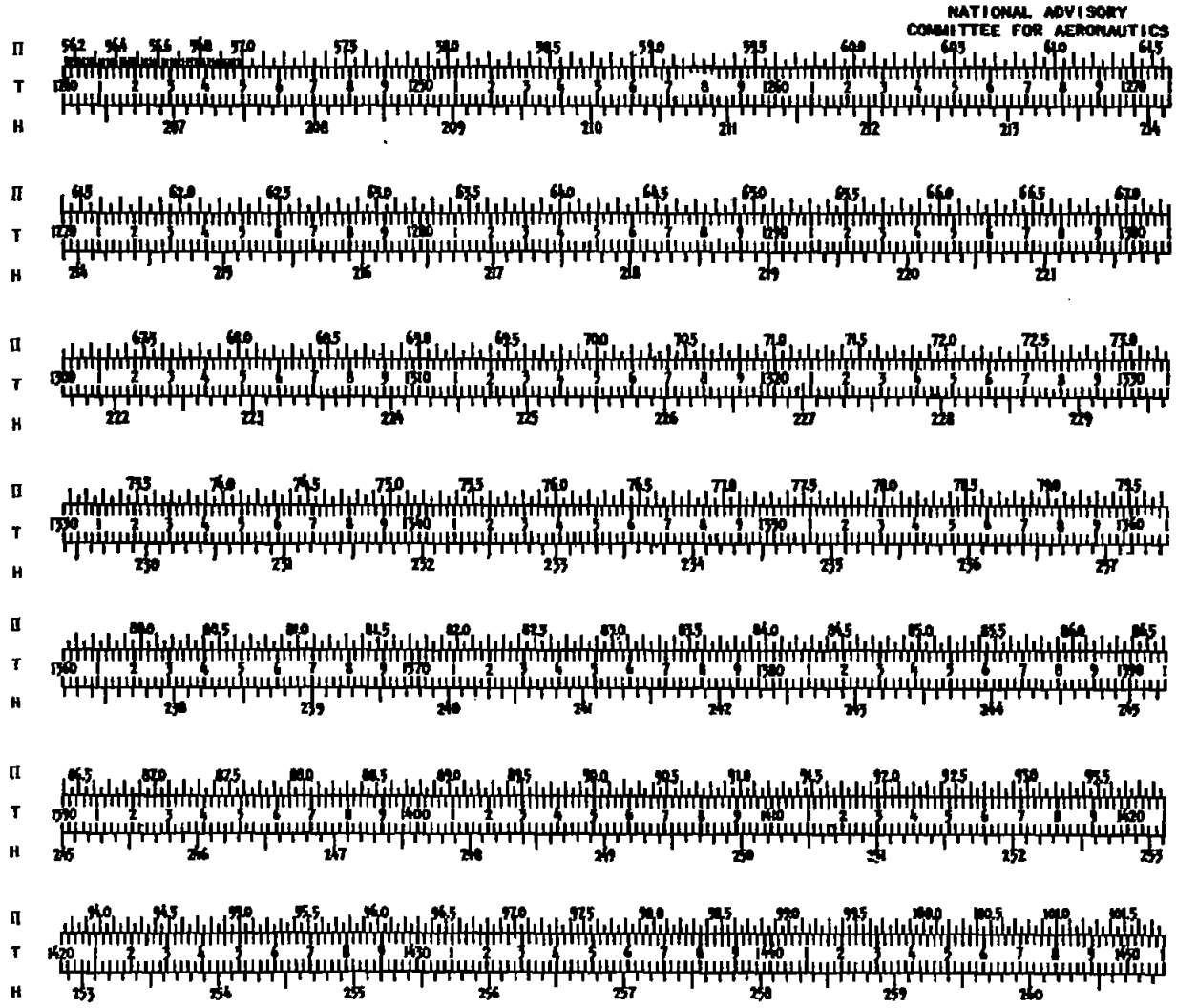


Figure 14. - Continued. (This chart is a direct reproduction, except for the notation, from reference 3.)

Π	56.16	to	101.75
T(degrees R)	1240	to	1451
H(BTU/lb)	206.2	to	260.9

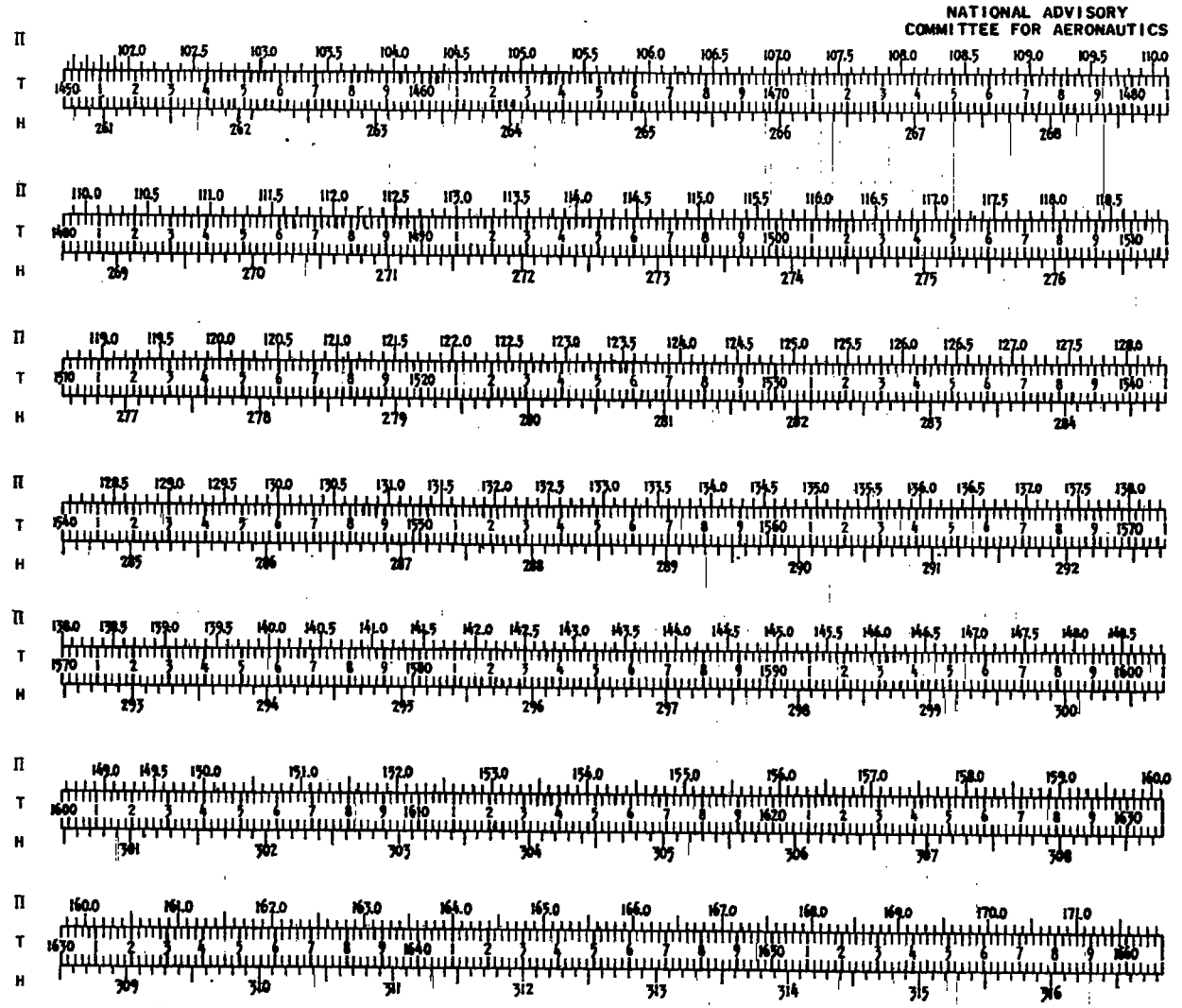


Figure 14. - Continued. (This chart is a direct reproduction, except for the notation, from reference 3.)

Π	101.55	to	171.9
T(degrees R)	1450	to	1661
H(BTU/lb.)	260.8	to	316.8

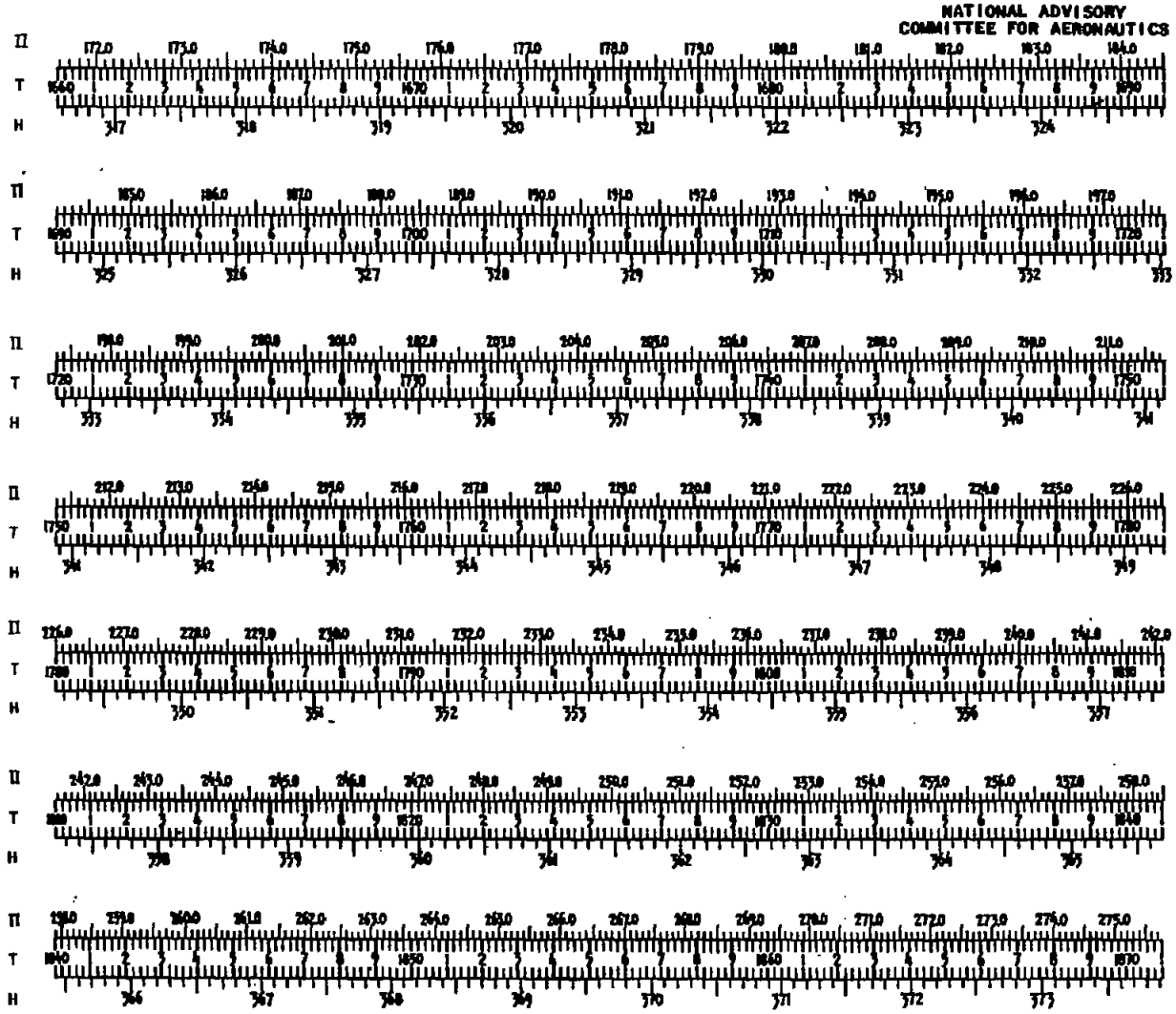


Figure 14. - Continued. (This chart is a direct reproduction, except for the notation, from reference 3.)

Π	171.6	to	275.7
T (degrees R)	1660	to	1871
H (BTU/lb.)	316.5	to	373.9

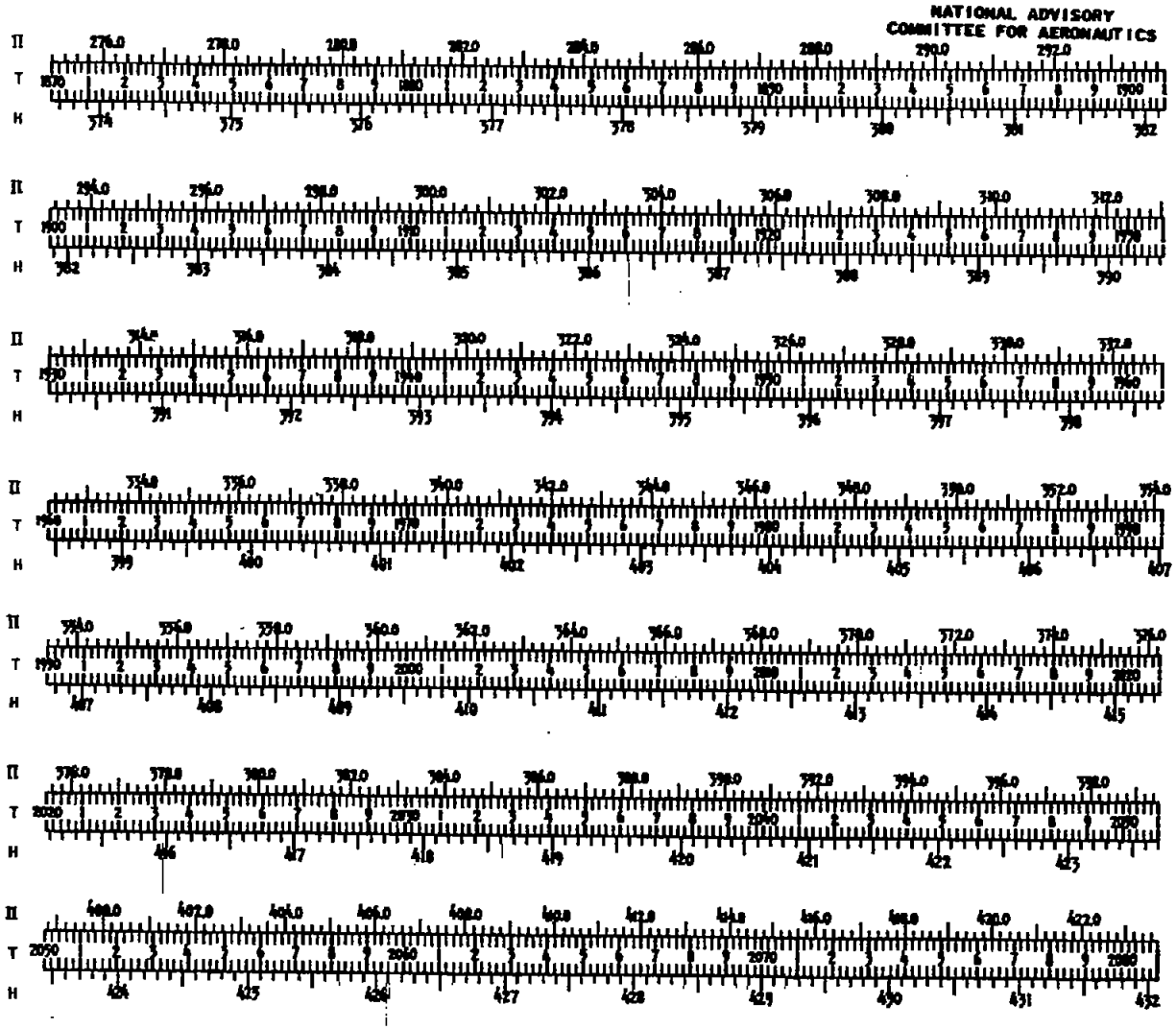


Figure 14. - Continued. (This chart is a direct reproduction, except for the notation, from reference 3.)

Π	275.2	to	423.6
T (degrees R)	1870	to	2081
H (BTU/lb)	573.7	to	432.0

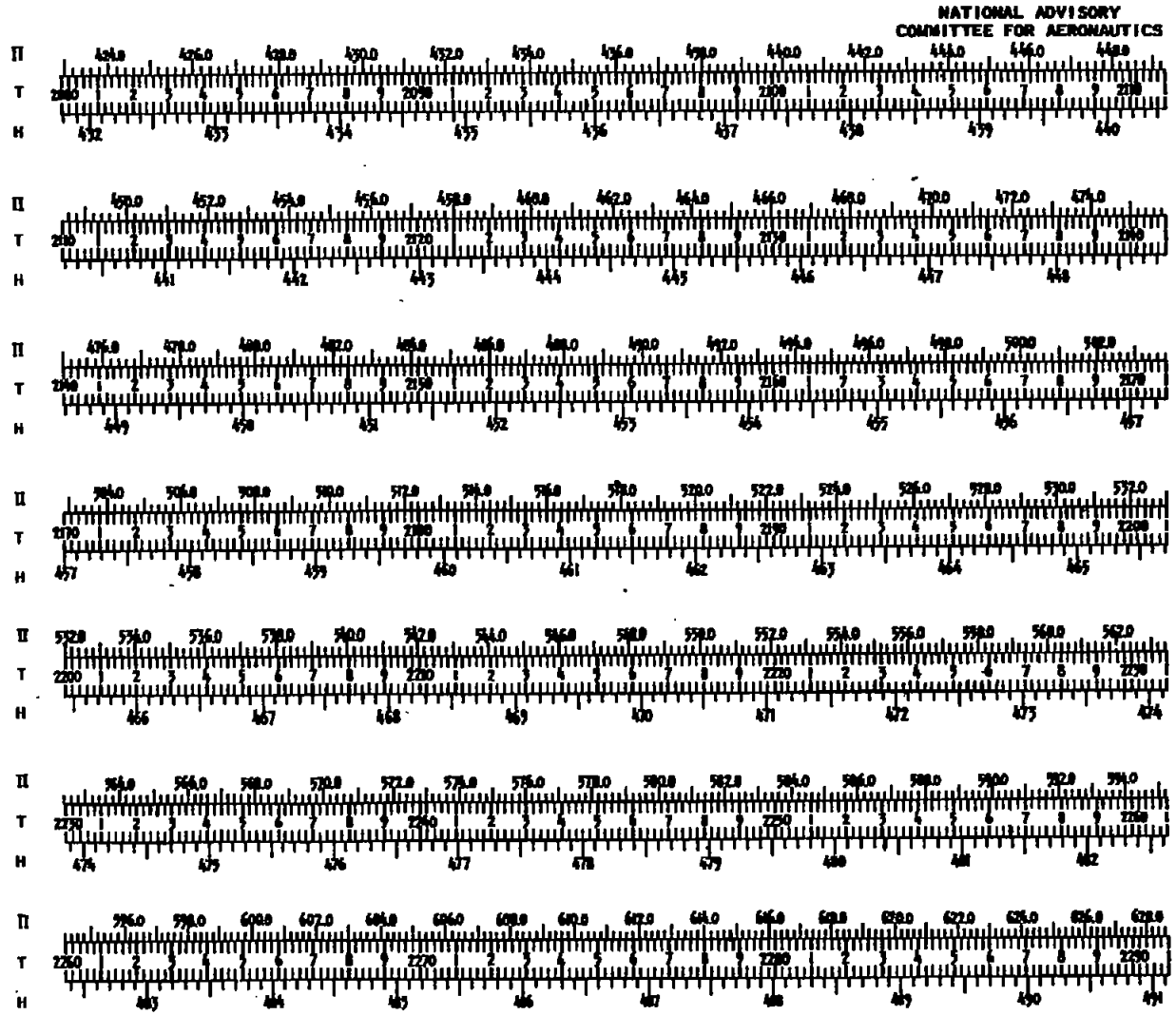


Figure 14. - Continued. (This chart is a direct reproduction, except for the notation, from reference 3.)

Π	423.0	to	628.6
T (degrees R)	2080	to	2291
H (BTU/lb)	431.8	to	491.1

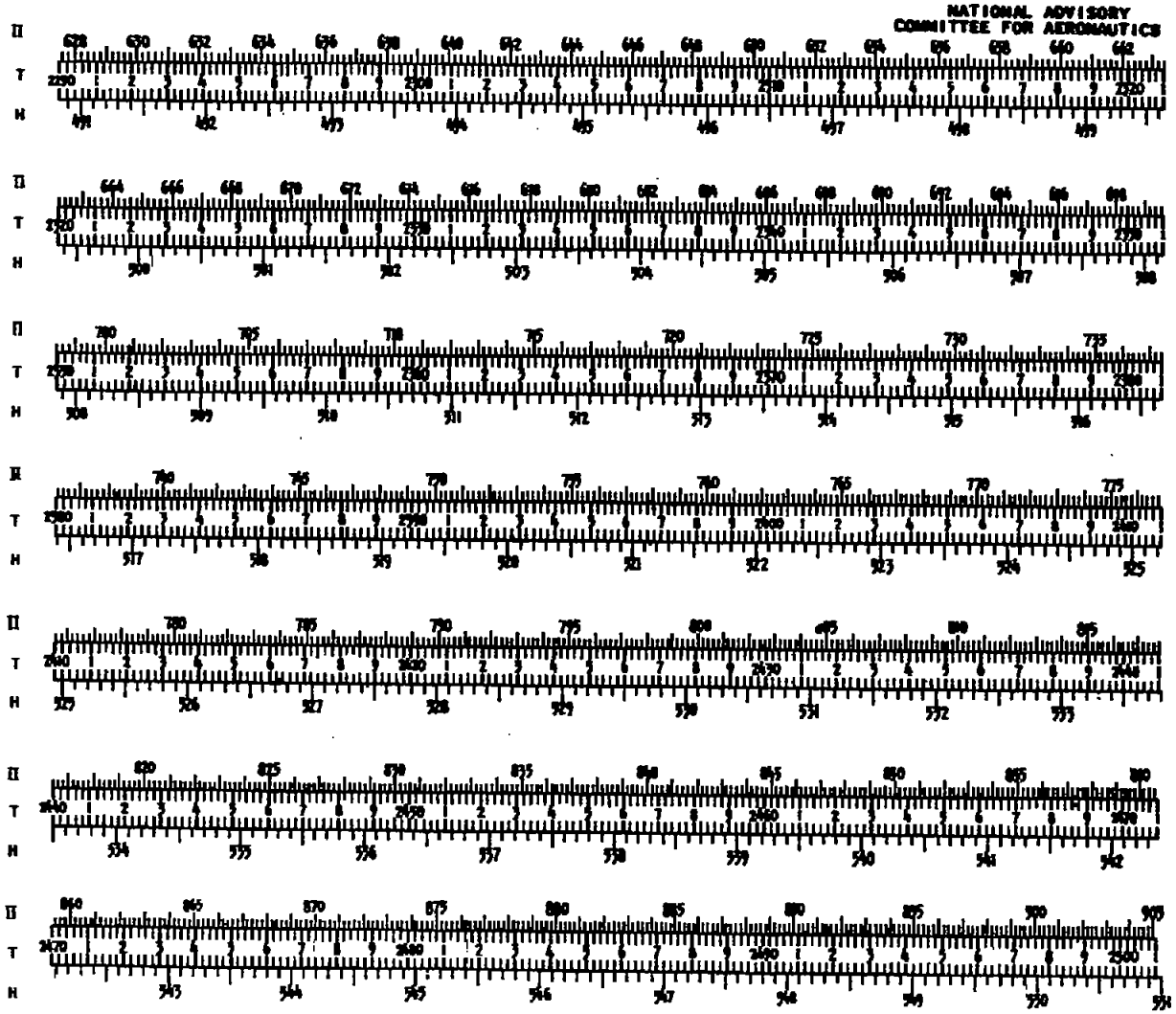


Figure 14. - Concluded. (This chart is a direct reproduction except for the notation, from reference 3.)

Π	627.6	to	905.0
T(degrees R)	2290	to	2501
H(BTU/lb.)	490.8	to	551.0

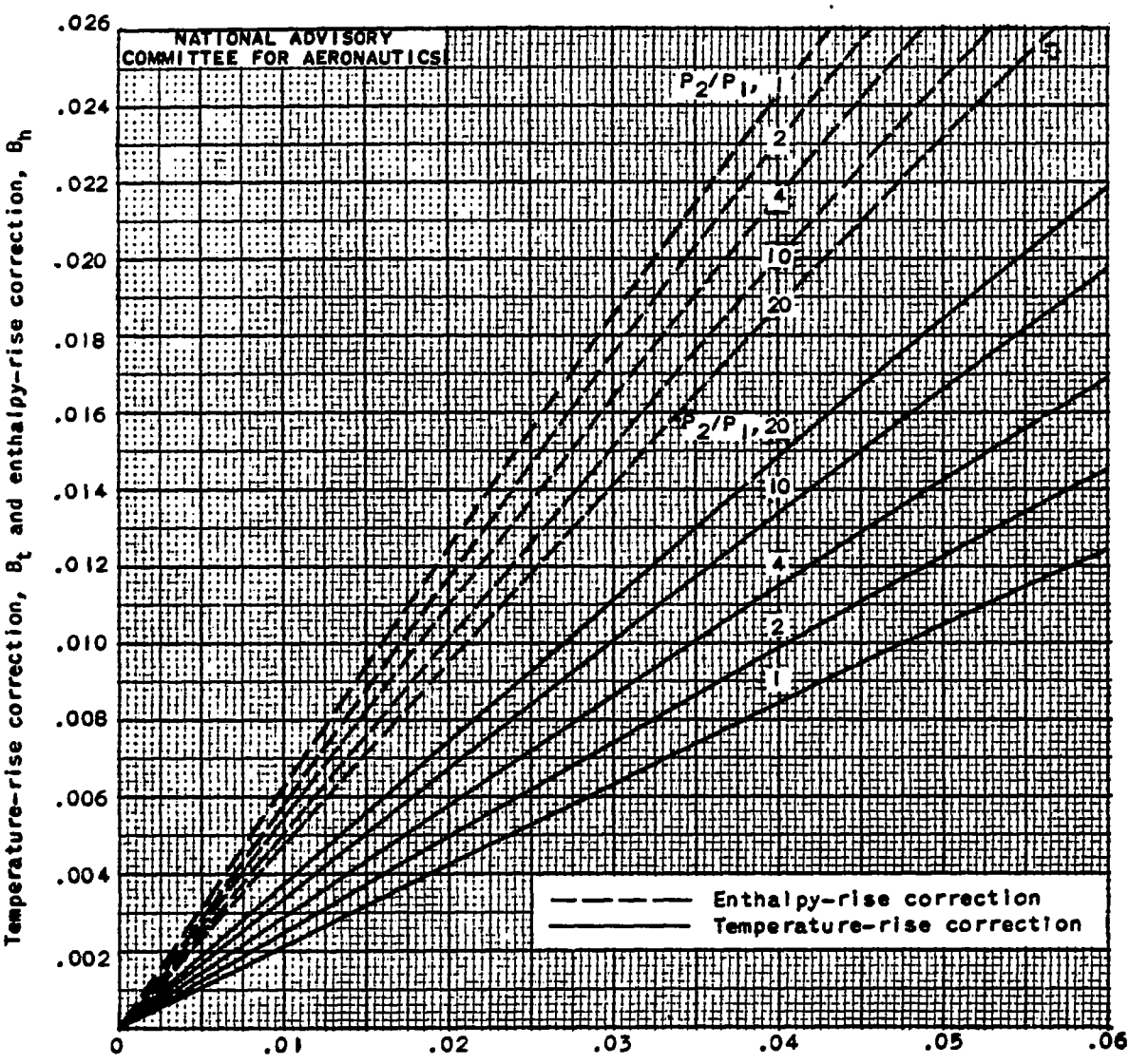


Figure 15. - Compression water-vapor correction for temperature and enthalpy rise where $\Delta T_m = \Delta T_d(1-B_t)$, $\Delta H_m = \Delta H_d(1+B_h)$. (These curves are taken from reference 3.)

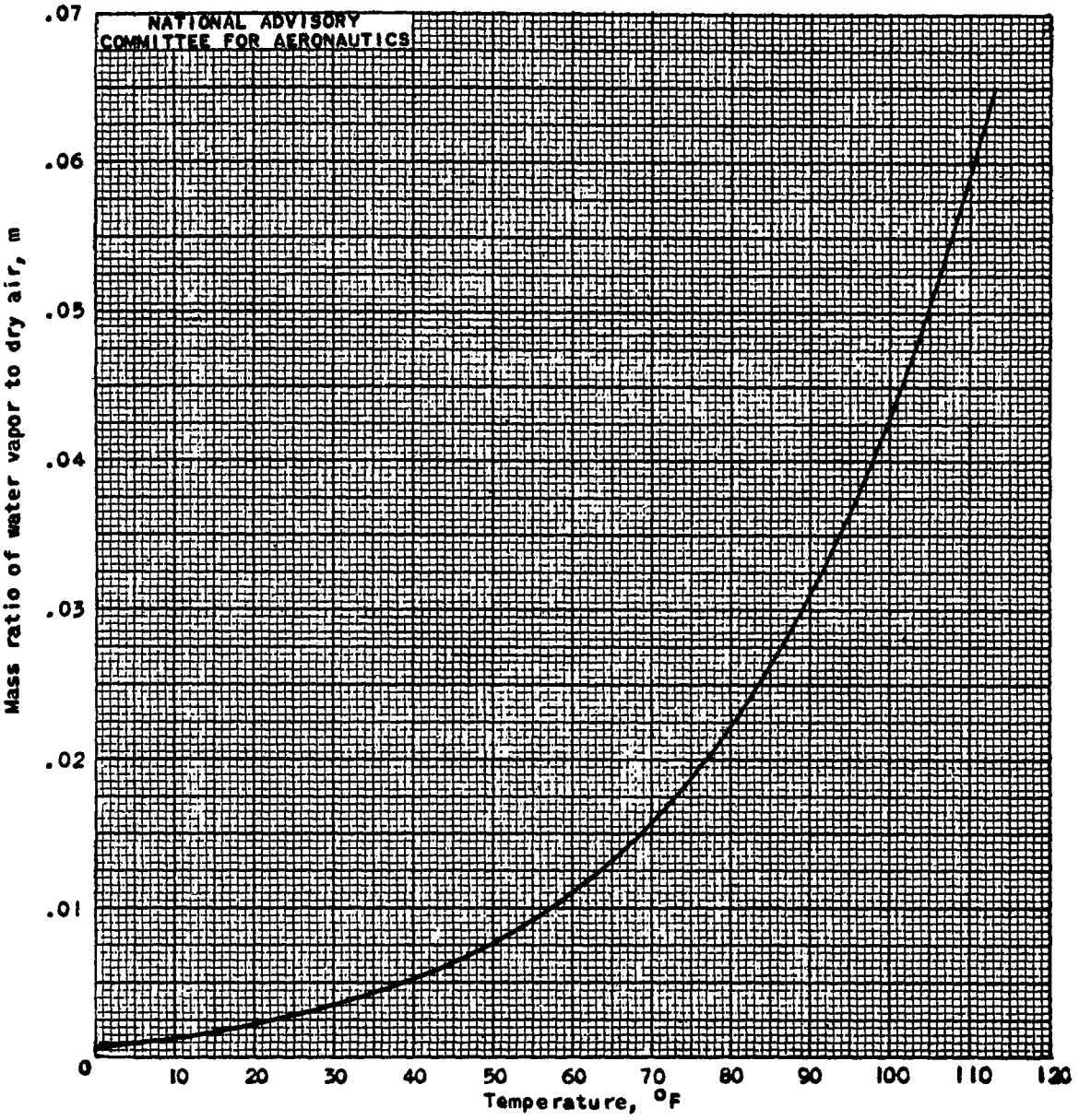


Figure 16. - Mass ratio of water vapor to dry air m for saturated air at standard sea-level pressure as function of temperature.

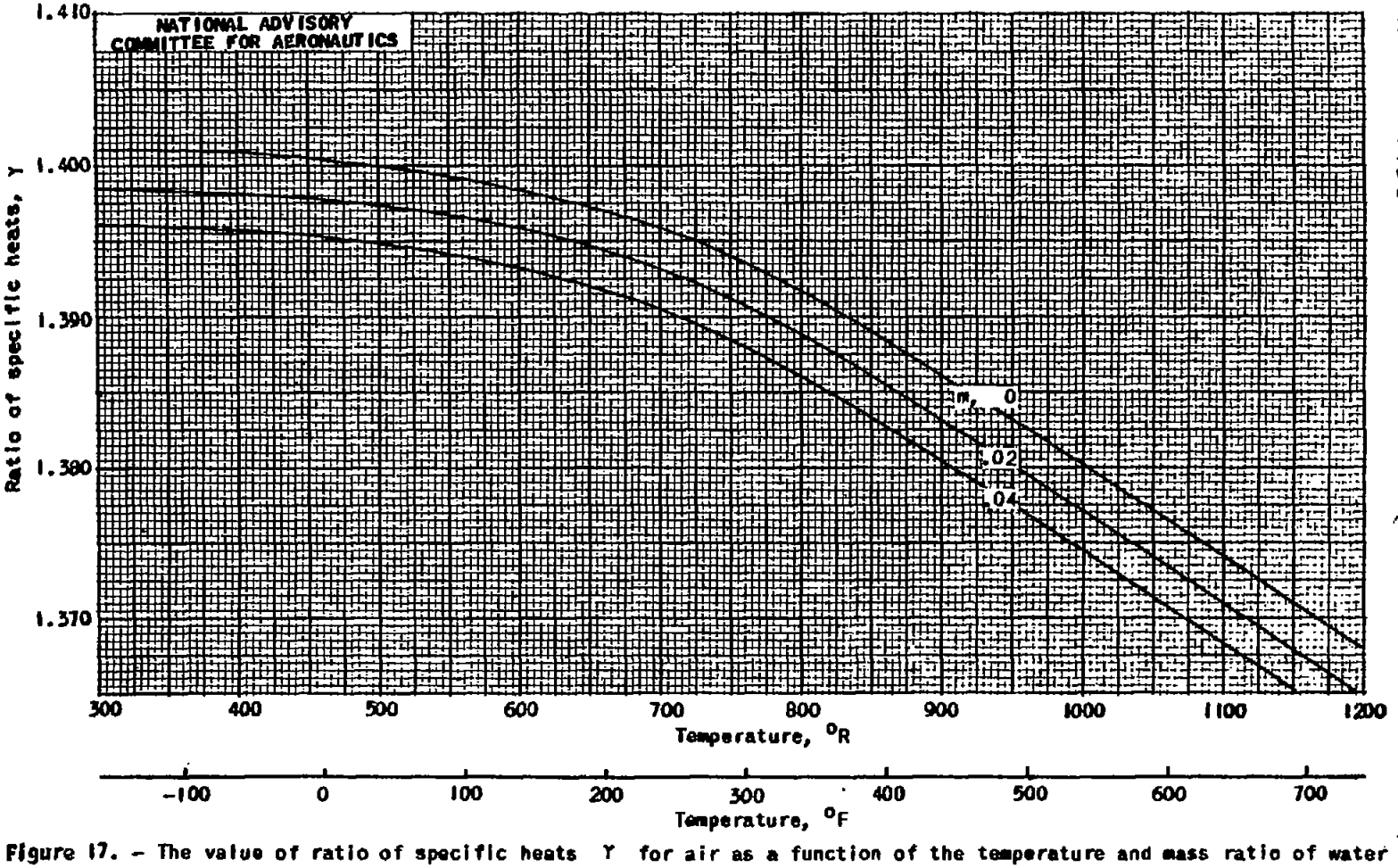


Figure 17. - The value of ratio of specific heats γ for air as a function of the temperature and mass ratio of water vapor to dry air m. (Dry-air data from reference 10).

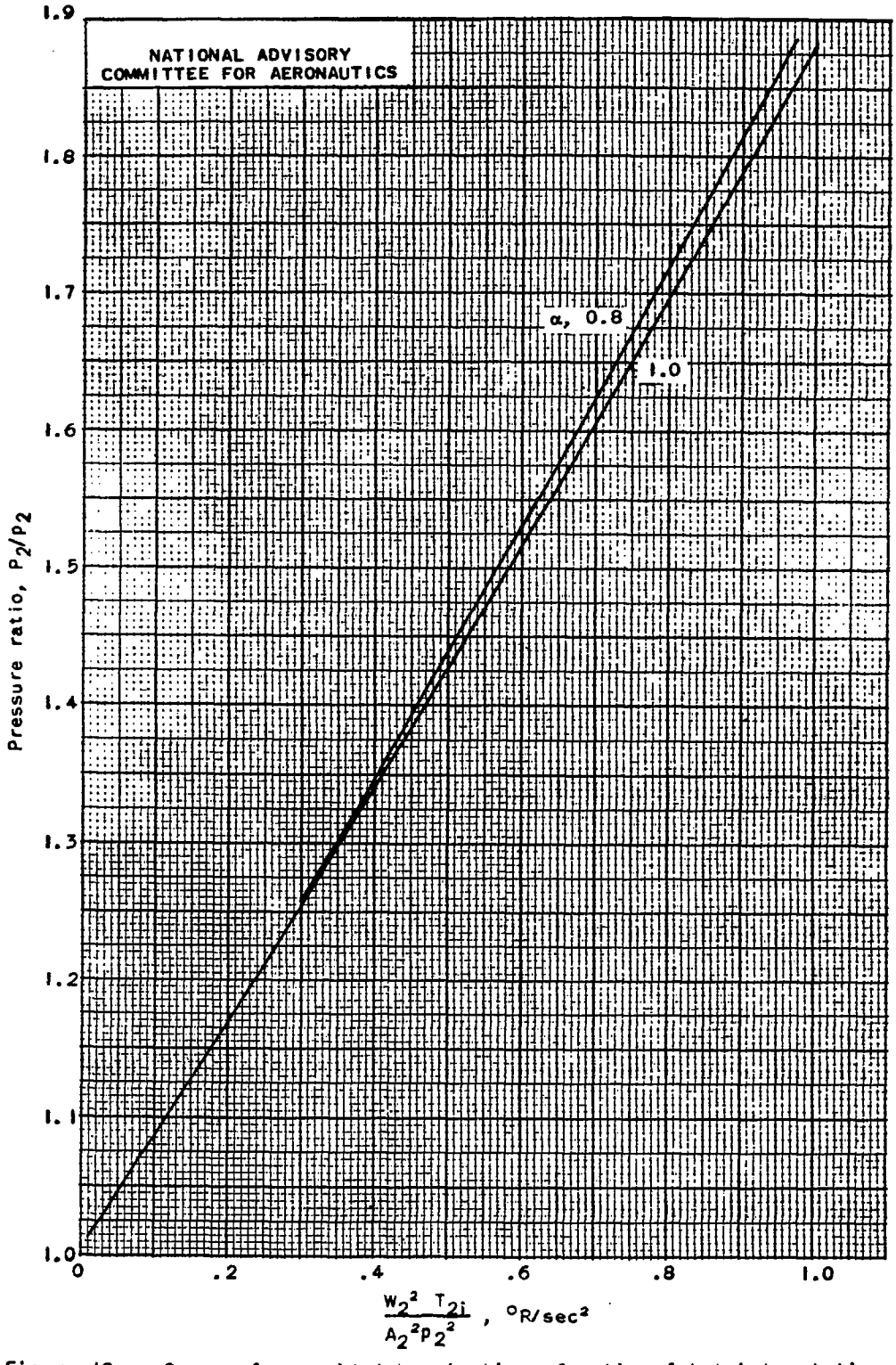


Figure 18. - Curves for rapid determination of ratio of total to static outlet pressures P_2/p_2 based on γ = 1.388 and R = 53.345.

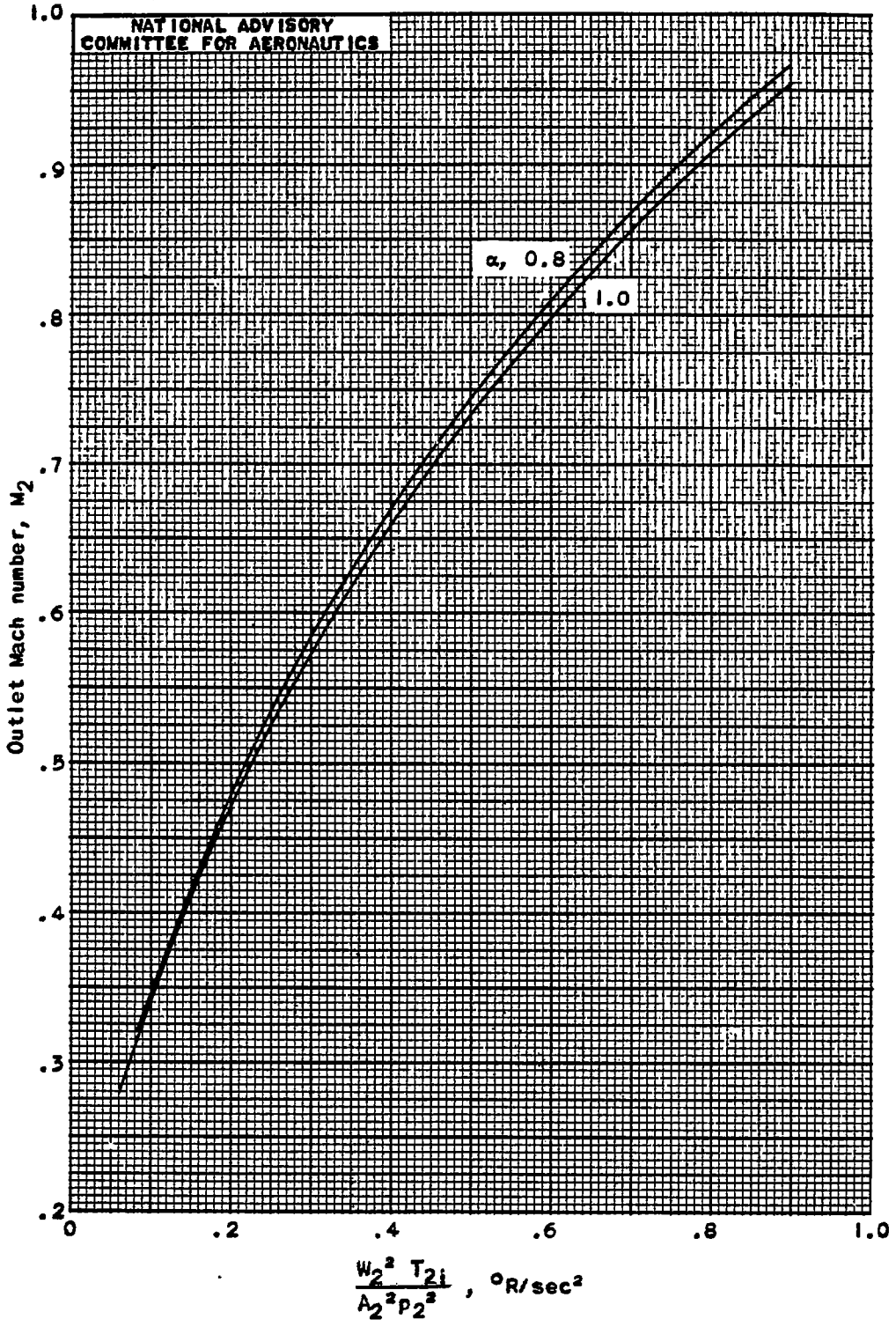


Figure 19. - Curves for rapid determination of outlet Mach number M_2 based on γ = 1.388 and R = 53.345.