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RESEARCH MEMORANDUM

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EXPERIMENTAL INVESTIGATION OF AN AXIAL-FLOW COMPRESSOR INLET STAGE OPERATING AT TRANSONIC RELATIVE INLET MACH NUMBERS

I - OVER-ALL PERFORMANCE OF STAGE WITH TRANSONIC ROTOR AND SUBSONIC STATORS UP TO ROTOR RELATIVE INLET MACH NUMBER OF 1.1

By Seymour Lieblein, George W. Lewis, Jr., and Donald M. Sandercock

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[redacted]

NATIONAL ADVISORY COMMITTEE
FOR AERONAUTICS
WASHINGTON
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RESEARCH MEMORANDUM

EXPERIMENTAL INVESTIGATION OF AN AXIAL-FLOW COMPRESSOR INLET STAGE OPERATING AT TRANSONIC RELATIVE INLET MACH NUMBERS

I - OVER-ALL PERFORMANCE OF STAGE WITH TRANSONIC ROTOR AND SUBSONIC STATORS UP TO ROTOR RELATIVE INLET MACH NUMBER OF 1.1

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SUMMARY

An axial-flow compressor inlet stage having a maximum rotor relative inlet Mach number of 1.1 was designed, constructed, and investigated. The rotor was designed for a high axial inlet velocity, no inlet guide vanes, and a tip speed of 1000 feet per second. The hub-tip ratio at the rotor inlet was 0.525. The stators were designed for current limiting values of stator inlet Mach number. The general results of the investigation indicated that axial-flow compressor inlet stages can be designed to operate efficiently at rotor relative inlet Mach numbers up to 1.1 at the tip.

At design corrected tip speed of 1000 feet per second, a peak stage adiabatic efficiency of 0.90 was obtained at a corrected weight flow of 44.5 pounds per second and a pressure ratio of 1.47. The maximum pressure ratio was 1.48 at an efficiency of 0.89. A maximum weight flow of 48.0 pounds per second was attained, which corresponded to a weight flow per unit frontal area of 29.2 pounds per second per square foot and a maximum rotor relative inlet Mach number of 1.1. At tip speeds lower than design, maximum efficiency at each speed remained very close to 0.90. Stage weight flow range was generally equivalent to the range of current inlet stages. Stator outlet conditions and general stage performance appeared satisfactory for purposes of multistaging with current stage designs.

INTRODUCTION

The requirements of efficient aircraft propulsion indicate the desirability of operating lightweight, compact turbojet engines that are highly efficient over wide ranges of operation. For current turbojet engines incorporating the multistage axial-flow compressor, reductions in compressor size and weight can be effected by increasing stage pressure ratio (reducing the number of stages) and increasing mass flow per

unit frontal area. A reduction in the number of stages for a given over-all pressure ratio may also be considered as a means of improving compressor range by reducing the stage matching difficulties generally encountered with large numbers of stages.

In general, increases in compressor unit mass flow can be obtained by increasing the axial Mach number at the inlet to the first rotor row. Increases in stage pressure ratio can be achieved from increases in wheel speed and from increases in rotor relative inlet Mach number, which are obtained from increases in both axial Mach number and wheel speed. In current compressor design practice, increases in both mass flow and pressure ratio without sacrifice of efficiency are restricted by the maximum allowable Mach number relative to the rotor blades of the stage. For current inlet Mach number limits of approximately 0.75, the relations among mass flow, stage pressure ratio, and wheel speed are very closely defined and, in general, an increase in any one factor can be accomplished only at the expense of another. Increased relative inlet Mach numbers can permit the selection of wheel speed levels in accordance, primarily, with compressor and turbine stress limitations and allow greater freedom in velocity diagram design. The successful extension of allowable relative inlet Mach numbers beyond present limiting values is therefore necessary if substantial improvements in engine performance are to be realized with the multistage axial-flow compressor. 2487

Results of recent investigations of supersonic axial-flow rotors (references 1, 2, and 3) indicated acceptable performance when these units were operated in the region of relative inlet Mach number from approximately 0.75 to 1.2 (transonic range). In addition, a general conviction existed that by correct reduction of blade leading-edge and maximum thickness, a continuous spectrum of performance could be obtained between present subsonic and supersonic units. In view of these considerations, an exploratory axial-flow compressor inlet stage designed specifically to operate in the transonic range of relative inlet Mach number at moderate wheel speeds was designed and constructed at the NACA Lewis laboratory for use in an investigation of the general level of performance and problems associated with compressor operation in this range.

The transonic inlet stage contained no inlet guide vanes and was designed to produce a pressure ratio of 1.35 at a corrected tip speed of 1000 feet per second, a hub-to-tip ratio of 0.525, and relative rotor inlet Mach numbers ranging from 0.75 at the hub to 1.13 at the tip. The stators were designed for conventional values of stator inlet Mach number.

The compressor stage was installed in a variable-component test rig. As a first phase of the investigation, the over-all performance of an inlet stage with 21 transonic rotor blades and 18 subsonic stator blades was determined at several tip speeds from 43 to 100 percent of design speed and is presented herein.

APPARATUS

Compressor Design

The transonic compressor rotor used in the investigation is shown in figure 1. The rotor consisted of 21 blades and had a tip diameter of 17.36 inches, an inlet hub-tip radius ratio of 0.525, an outlet hub-tip ratio of 0.60, and an axial depth of about 3.0 inches. Rotor chord length was 3.25 inches at the tip and 3.0 inches at the hub to give solidities of 1.25 and 2.05, respectively. Stator chord lengths were 3.2 inches at the tip and 2.65 inches at the hub, giving solidities of 1.05 and 1.42, respectively, for 18 blades. Stator hub radius was constant at 5.2 inches. Blade leading- and trailing-edge radii were about 0.015 to 0.020 inch at all sections for both rotor and stator.

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Design corrected tip speed of 1000 feet per second was sufficient to produce moderate transonic relative inlet Mach numbers up to about 1.13 without inlet guide vanes at an average inlet axial Mach number of 0.60. Design corrected weight flow (with inlet wall boundary-layer allowance of 2 percent) was 48.6 pounds per second, or 29.6 pounds per second per square foot of frontal area.

The over-all stage velocity diagram was determined to produce an average pressure ratio of 1.35, an adiabatic efficiency of 0.85, and stator inlet Mach numbers consistent with current practice (approximately 0.75). The rotor-outlet annulus area was determined to produce a slight increase in average axial velocity across the row at design conditions. In order to obtain these conditions, it was necessary to design for a radially increasing total-pressure variation (1.41 at the tip and 1.29 at the hub). A combined wake and wall boundary-layer area reduction of 8 percent was assumed across the rotor. The design velocity diagrams for rotor and stator are shown in figure 2. (The symbols used herein are defined in the appendix.) The radial variation of inlet axial velocity was based on results of stationary mock-up tests of the inlet annulus. Stator turning angles were designed to result in maximum Mach numbers of approximately unity relative to a succeeding rotor row.

In the absence of a practical reliable method of blade design in transonic flow, rotor blade profiles at hub, mean, and tip radii were determined by means of the method of moment of momentum (similar to methods in reference 4). Although the method is known to be approximate and nonrigorous, it was believed to be applicable as a preliminary design tool because of its simplicity and the relatively high rotor solidities employed. The method was developed from the equality of the total tangential torque exerted by the air on an element of a rotor blade and the change in moment of momentum across the element, or in simplified form for no inlet whirl,

$$\int_0^{c_z} r\,\Delta p\,h\,dc_z = m\,r\overline{V}_\theta \tag{1}$$

where Δp is the static pressure difference across the blade upper and lower surfaces (linear variation assumed) and $\overline{V}_\theta$ is average absolute tangential velocity at z. Because of the rotor hub taper, the blade sections were designed along conical surfaces with an assumed linear variation of stream tube height h across the blade row. Inlet and outlet values of h were determined from conditions of the design velocity diagram (simple radial equilibrium at rotor outlet).

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The variation of average air angle along the chord (air mean line) of a blade element was determined by equation (1) from the prescription of an elliptical chordwise pressure loading and a selected variation of average velocity along the blade chord from inlet to outlet values. An elliptical pressure loading was found satisfactory for the avoidance of sharp peaks in suction-surface Mach number and for ease of calculation. For the subsonic inlet Mach numbers, the average relative velocity variation was prescribed to avoid the attainment of the sonic value along the flow path and was obtained by adjustment of the approximate blade thickness and solidity of the element consistent with requirement of strength and pressure rise. At the tip, all waves were assumed to fall outside of or near the leading edge and a continuous decrease in velocity across the blade was prescribed. Considerations of assumed blade boundary-layer growth (about 0.08 in. at trailing edge) and outlet air deviation angle (about 4°) were then used to establish the blade mean camber line and the exact blade thickness variation. The rotor blades were set for average angle of incidence of about 3°. (Incidence angle is defined as angle between inlet air and tangent to blade mean line at leading edge.) Maximum ratio of blade thickness to blade chord was about 0.06 at the tip and 0.085 at the hub. The blade shape at the tip was very similar to a symmetrical circular-arc profile.

Because of the possibility of obtaining stator inlet Mach numbers greater than the design values, symmetrical circular-arc camber line profiles were used at all radii for the stator. Results of reference 5 indicate very low losses for this type of blade at inlet Mach numbers of 0.8 in the good incidence range. Maximum stator thickness ratio was 0.06 at the hub and 0.08 at the tip. Stator incidence and camber angles were determined from an analysis of the data of reference 5. Blade setting angles were determined from initial runs of the rotor alone. Rotor and stator blade profiles at the hub and the tip are shown in figure 3.

Compressor Installation

The transonic compressor installation was obtained from modification of the 16-inch variable-component supersonic compressor test rig described in reference 2. Principal modifications were the enlarging of the outer casing diameter to 17.36 inches (to provide a smaller inlet

hub-tip ratio) and the addition of piping to provide an open system with atmospheric air inlet and suction exhaust. The inlet system consisted of a 36-inch-dimeter orifice tank with a 26-inch-diameter thin-plate open-end orifice, a filter tank, and an inlet depression tank (4 ft in diameter by 6 ft in length). Inlet total pressure and air flow were controlled by motor-driven butterfly valves in the inlet and outlet ducting, respectively. The compressor rotor was driven by a 3000-horsepower variable-frequency motor through a speed increaser. A sketch of the transonic compressor test rig is shown in figure 4.

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Instrumentation

Air flow through the compressor was determined from the pressure drop across the thin-plate orifice and the orifice ambient temperature, which was measured by two iron-constantan thermocouples. Additional checks of air flow were made by drawing the compressor air through the refrigerated-air inlet system which contained a calibrated adjustable orifice, and by integrating measured velocities and densities in the straight section of the inlet annulus upstream of the rotor at station 2.

Data for integrated weight flows were obtained by two methods: (1) radial surveys of static and total pressure with a combination-type probe (similar to one shown in reference 6), and (2) a low-calibration-error static-pressure rake containing five tubes equally spaced across the passage (fig. 5(a)). Each tube of the static-pressure rake had four manifolded holes at 90°. The deviation of indicated velocity head from correct calibration was about 1 percent for angles of pitch and yaw up to $\pm 5^{\circ}$. Weight flow was obtained from the rake readings, the inlet-tank total pressure, and a boundary-layer blockage factor determined from the total-pressure surveys. Measurements of weight flow obtained from all methods agreed within $1\frac{1}{2}$ percent.

Depression tank temperature measurements were made with six iron-constantan thermocouples equally spaced around the circumference of the tank at varying radii. Stagnation pressure was obtained from four equally spaced pressure taps; tank velocities were sufficiently small to neglect velocity head.

Rotor inlet conditions were determined from the inlet-tank total pressure and 8 static-pressure wall taps at hub and tip located an average of 3/4 inch upstream of the rotor blade leading edge (station 3, fig. 4). The radial variation of static pressure was faired according to the static-pressure variation measured by the rake at station 2. Preliminary surveys of inlet flow angles as measured by the combination probe at station 2 were within approximately 1° of the axial direction.

Compressor outlet conditions were determined at station 6 (3 in. behind the stator-row trailing edge) from 24 individual shielded total-pressure probes and 24 iron-constantan thermocouples (fig. 5(b) and (c)). The outlet instrumentation was intended to provide data for the evaluation of stage efficiencies closely representative of the true mass-averaged adiabatic efficiency of the stage. The total-pressure probes were located radially at the centers of six equal areas and were spaced around the circumference to duplicate four equally spaced positions across the stator blade passage. The thermocouples were located at corresponding radial and circumferential positions by means of four rakes. The total-temperature and total-pressure probes were insensitive to angle over the range of stator exit angle encountered. 2487

In addition, radial surveys of discharge angle and total pressure were made at mid-passage between blades about 1/2 inch behind the stator (station 5) with a claw-type probe (fig. 5(d)). Static-pressure wall taps were located at each measuring station.

All instruments were calibrated for Mach number and angle in an open-jet calibration tunnel, and all pressures were indicated in inches of tetrabromoethane. Compressor speed was controlled electronically within $\pm$ 10 rpm and was frequently checked with a chronometric tachometer.

PROCEDURE

Complete-stage runs were conducted for five corrected tip speeds ranging from 430 to 1000 feet per second. Inlet tank pressure was maintained constant at approximately 20 inches of mercury absolute, while inlet tank temperatures varied from about 75° to 87° F. At 800, 900, and 1000 feet per second, the flow was varied from maximum to that resulting in blade stall as indicated by fluctuations of the outlet measurements. For the lower tip speeds, stall and choked flow were not observable, and weight flow was varied until efficiency decreased markedly at both ends of the curve. Stator exit surveys of angle and total pressure were taken at selected points at each tip speed.

Adiabatic temperature-rise efficiency was calculated from arithmetic averages of the outlet total-temperature and total-pressure readings as

$$\eta_T = \frac{T_1\left[\left(\frac{P_6}{P_1}\right)^{0.283} - 1\right]}{T_6 - T_1} \tag{2}$$

The efficiency values given by equation (2) and the outlet instrumentation are, strictly speaking, area-averaged efficiencies. However, in view of the circumferential (four positions across passage) and radial (two positions near end walls) locations of the probes and the probable sizable dissipation of the stator wakes at the measuring station, the calculated efficiencies are believed to be reliable measures of the mass-averaged efficiencies of the stage.

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The radial variation of Mach number leaving the stators was determined from probe total-pressure surveys and a radial variation of static pressure calculated by a method based on wall pressure-tap readings and the measured angle variation (assuming simple radial equilibrium).

RESULTS AND DISCUSSION

Over-all performance. - The over-all performance of the transonic inlet stage as measured 3 inches downstream of the stator trailing edge is shown in figure 6 as plots of adiabatic efficiency and total-pressure ratio. At design speed, a peak efficiency of 0.90 was obtained at a pressure ratio of 1.47 and a corrected weight flow of 44.5 pounds per second. Peak efficiency remained at approximately 0.90 for all tip speeds investigated. At 80 percent speed and above, efficiency as well as pressure ratio curves are fairly flat, with efficiencies above 0.80 obtained over practically the entire weight flow range.

Maximum average pressure ratio at design speed was 1.48 at an efficiency of 0.89. The greater pressure ratio obtained in the tests compared with the design value is attributed primarily to observed efficiencies and rotor angles of incidence greater than the design values. It may also be that actual rotor blade outlet deviation angles and area blockage are somewhat less than the design values, which would tend to produce greater turning and diffusion. The radial variation of total-pressure ratio as measured by both the discharge shielded probes and the stator outlet survey probe at peak efficiency at design speed is shown in figure 7. A substantially greater radial variation of pressure ratio was obtained compared with design values.

Design weight flow of 48.6 pounds per second was not attained because actual optimum incidence angles and radial variations of incidence angle were greater than the values assumed for the design. Maximum weight flow at design speed was 48.0 pounds per second and represents a weight flow per unit frontal area of 29.2 pounds per second. Operating range at high tip speeds is slightly smaller than the ranges obtained with subsonic inlet stages (references 7 and 8, for example). This trend is to be expected in view of the higher inlet Mach number levels. Operating range at the lower tip speeds, however, is generally equivalent to current inlet stage variations. The application of transonic inlet

stages to multistage compressors may therefore not be expected to introduce matching difficulties because of the range of the inlet stage.

Rotor inlet conditions. - Relative inlet Mach number and relative air inlet angle at the inlet to the rotor (station 3) at design speed are shown in figure 8. Maximum rotor relative inlet Mach numbers were somewhat lower than design values and attained values of about 1.1 and 1.06 at the tip at maximum weight flow and peak efficiency, respectively. Only a very small part of the passage in the tip region was operating at Mach numbers greater than unity. The angle of incidence at peak efficiency (difference between observed air angle and blade mean-line angle, fig. 8(b)) was considerably greater than the design value (difference between design air angle and blade mean line), which indicated observed optimum angles of incidence of from about 4° to 9° for this design. The slightly larger inlet angle variation from hub to tip was due to measured radial variations of absolute inlet velocity greater than the values used for the design.

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Stator exit conditions. - The radial variations of air angle and Mach number leaving the stators (station 5) are illustrated in figure 9. Although the stator outlet radial survey was conducted at one circumferential position approximately midway between blades, the survey readings are believed to be representative of average values across the blade spacing. Stator outlet Mach numbers less than 0.65 were obtained over the entire useful range of operation of the stage. The radial variation of stator outlet angle at a given speed shows little change in angle across the blade height, except for the wall boundary-layer regions where a sharp rise in angle is observed. The variation in outlet angle over the speed range was about 3°. For the measured outlet angles and Mach numbers, all Mach numbers relative to a succeeding rotor row would fall within current limits. Stator outlet conditions are thus seen to be favorable for the matching of succeeding stages of current design.

SUMMARY OF RESULTS

The following results were obtained from an experimental investigation of an axial-flow compressor inlet stage having a maximum relative inlet Mach number of 1.1 at 1000 feet per second corrected tip speed and a 0.525 inlet hub-tip ratio:

(1) Peak efficiency at design speed was 0.90 at a corrected weight flow of 44.5 pounds per second and a pressure ratio of 1.47. Peak efficiency remained at approximately 0.90 for all tip speeds.

(2) Maximum pressure ratio at design speed was 1.48 at an efficiency of 0.89 and maximum weight flow was 48.0 pounds per second corresponding to a weight flow per unit frontal area of 29.2 pounds per second. Maximum relative inlet Mach number was 1.1.

(3) The weight flow range was generally equivalent to the range of current inlet stages.

(4) Stator outlet conditions and general stage performance appeared satisfactory for purposes of multistaging with current stage designs.

CONCLUSION

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From the results of this investigation, it is concluded that axial-flow compressor inlet stages can be designed to operate efficiently at rotor relative inlet Mach numbers up to 1.1 at the tip.

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APPENDIX - SYMBOLS

The following symbols are used in this report:

c blade chord, ft

h incremental stream tube height, ft

M absolute Mach number

M' relative Mach number

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m mass flow, slugs/sec

P total pressure, lb/sq ft

p static pressure, lb/sq ft

r radius, ft

T total temperature, $^{\circ}R$

U wheel speed, ft/sec

V absolute velocity, ft/sec

W weight flow, lb/sec

β air angle (with axial), deg

δ ratio of inlet total pressure to standard sea-level pressure

η_T adiabatic temperature-rise efficiency

θ ratio of inlet total temperature to standard sea-level temperature

Subscripts:

1 compressor inlet (depression tank)

2 weight flow measuring station

3 rotor inlet

4 rotor outlet

5 stator outlet

6 compressor outlet (discharge measuring station)

t compressor tip

z axial direction

θ tangential direction

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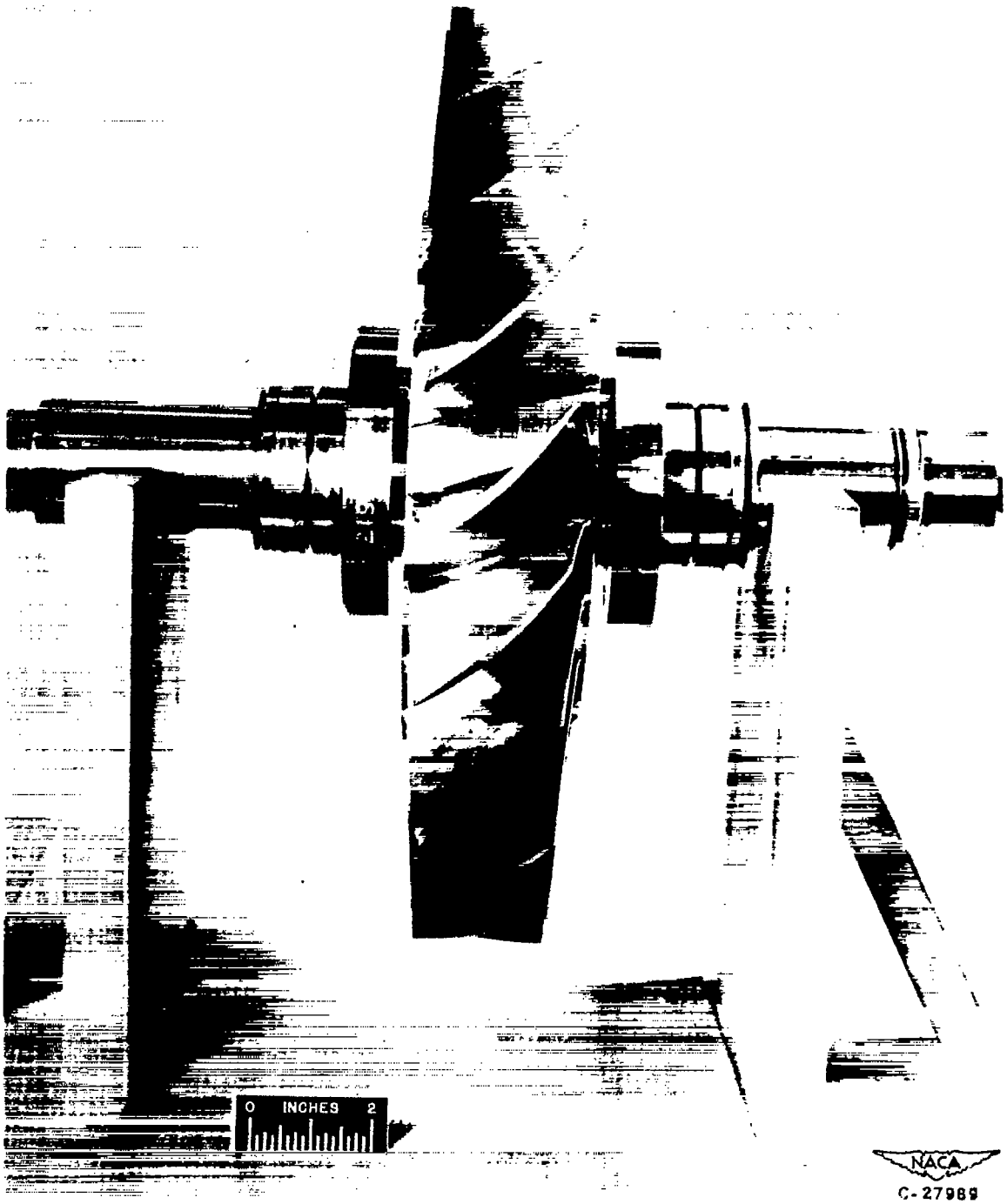

Figure 1. - Transonic compressor rotor.

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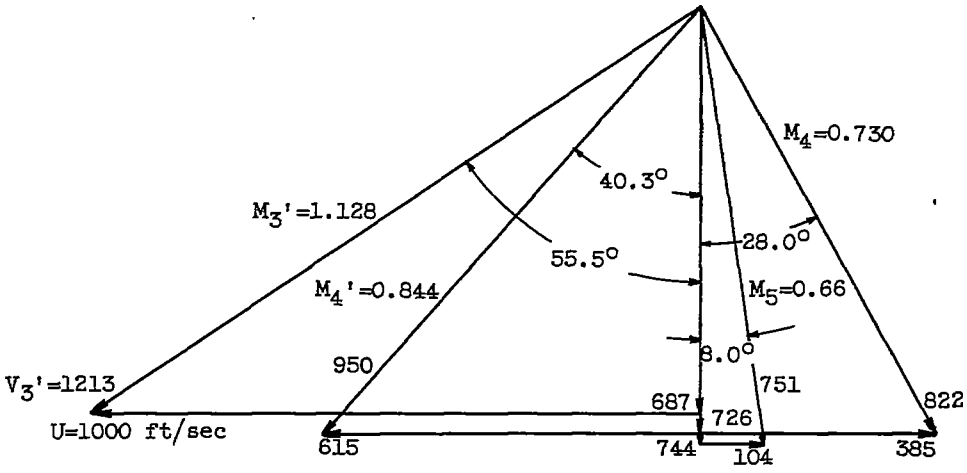
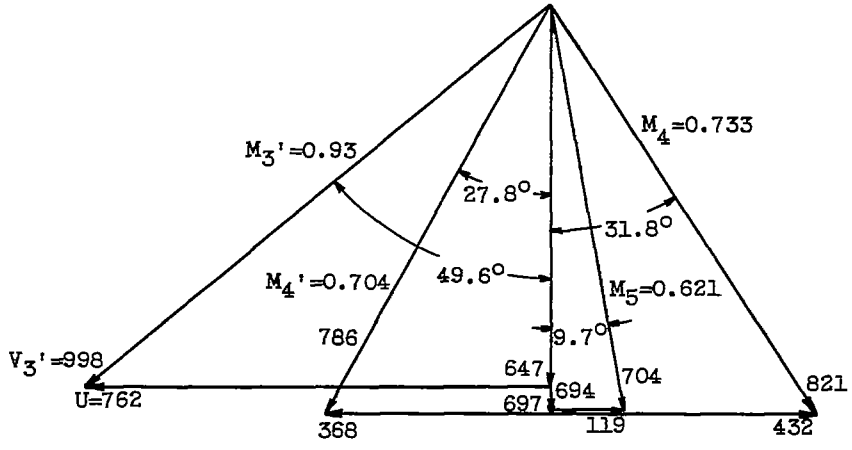
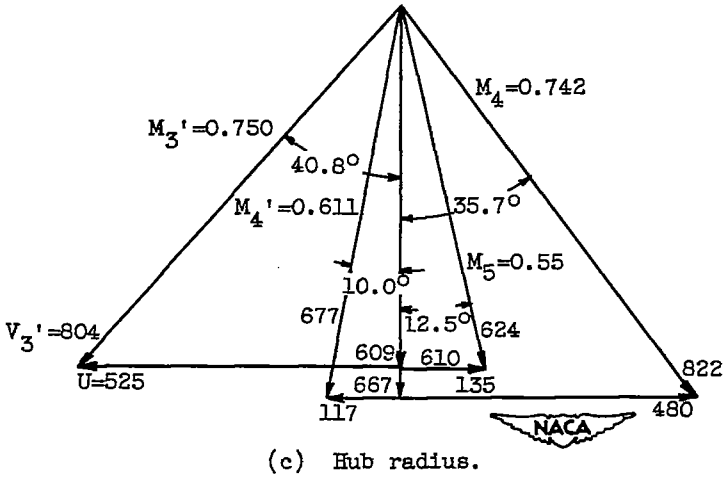


(a) Tip radius.



(b) Mean radius.



(c) Hub radius.

Figure 2. - Design velocity diagram for transonic inlet stage.

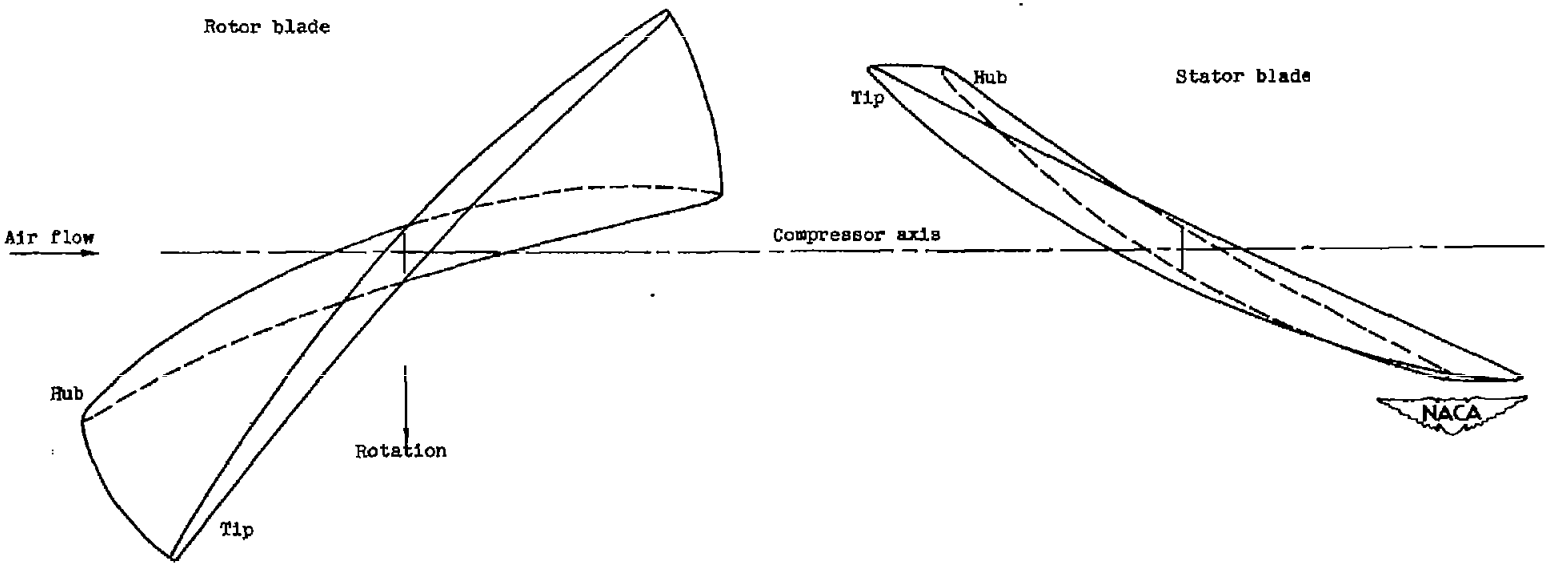


Figure 3. - Rotor and stator blade profiles at hub and tip.

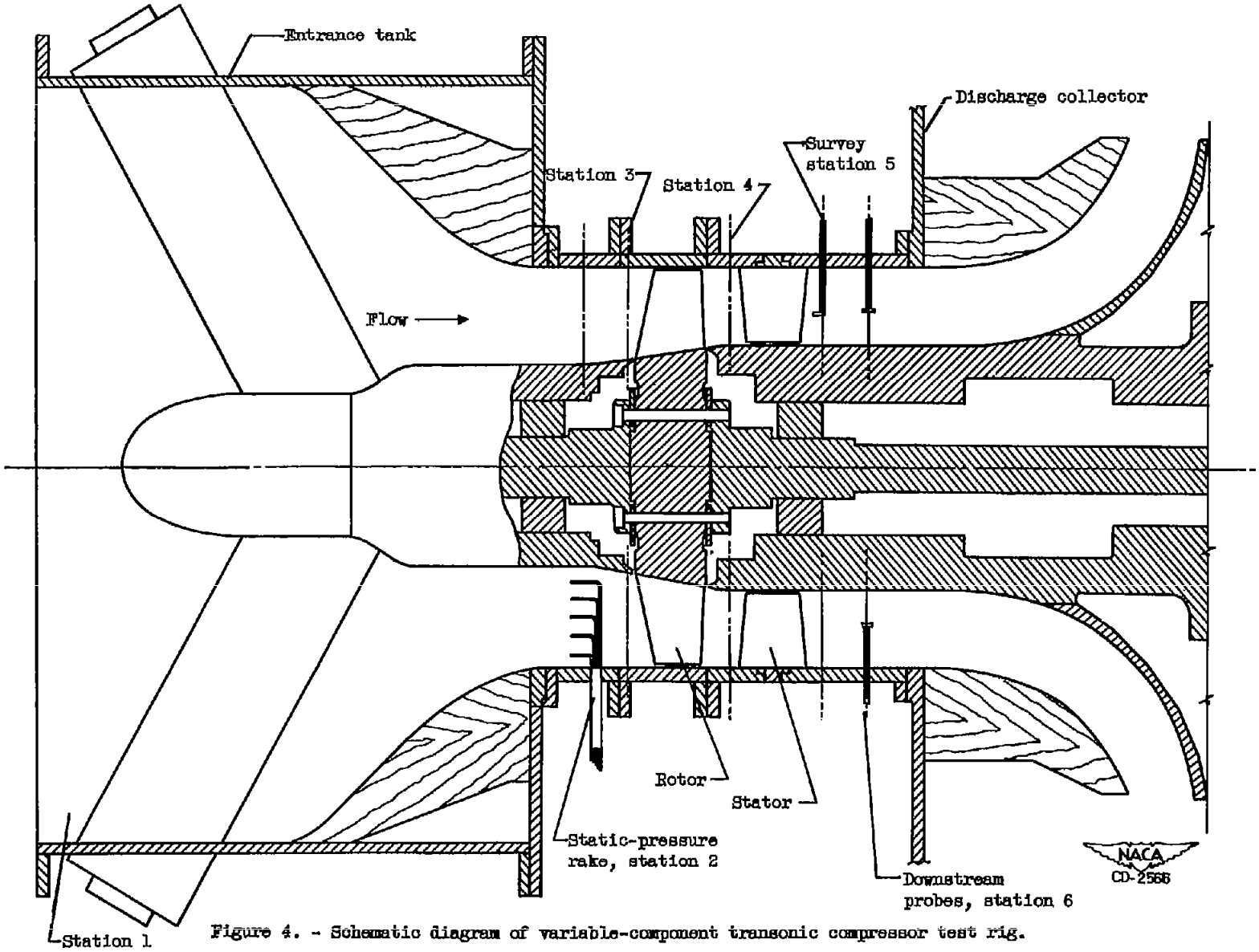


Figure 4. - Schematic diagram of variable-component transonic compressor test rig.

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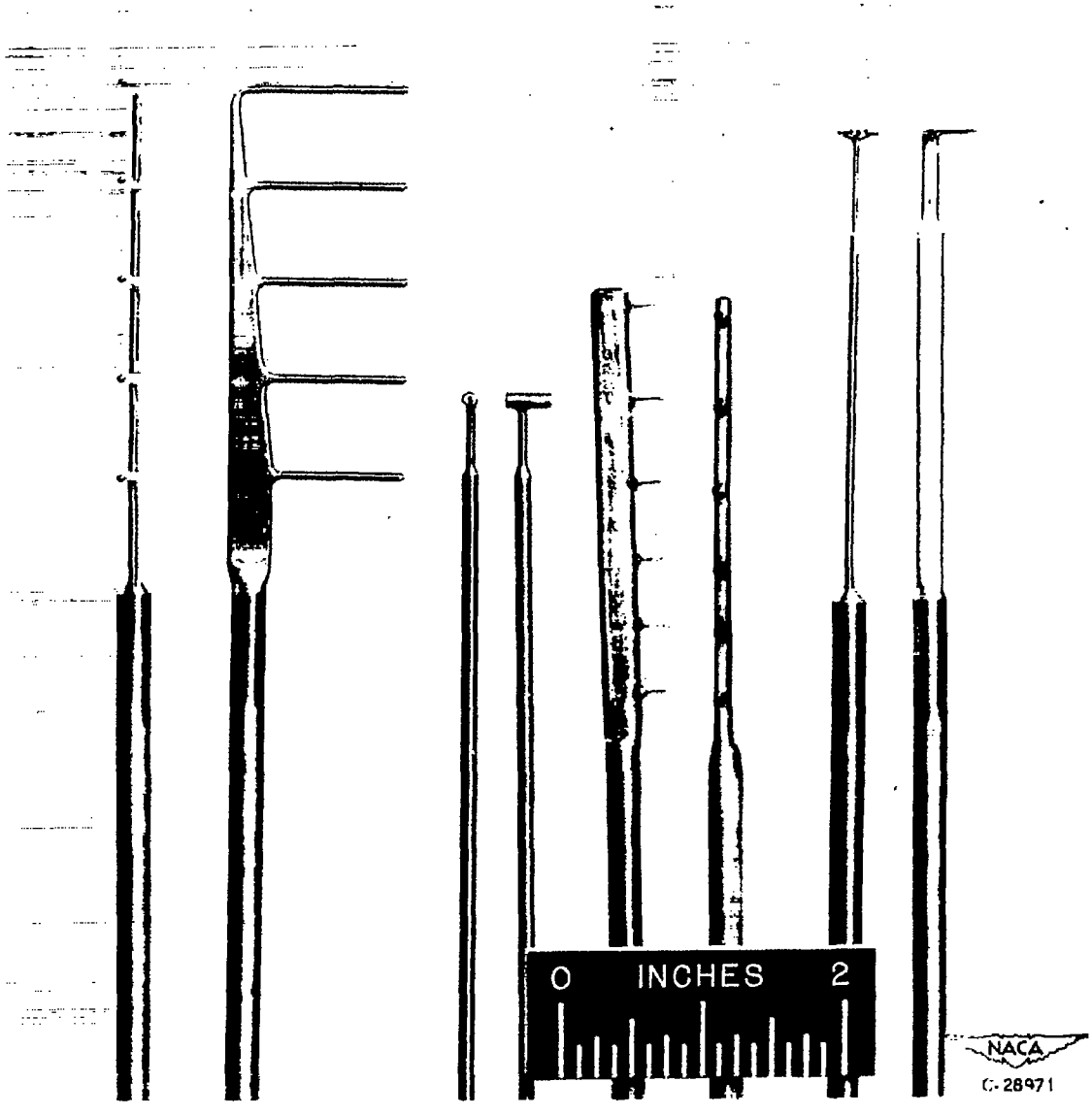

(a) Static-pressure rake. (b) Shielded probes. (c) Temperature rakes. (d) Claw total-pressure probe.

Figure 5. - Instrumentation used for investigation of transonic inlet stage.

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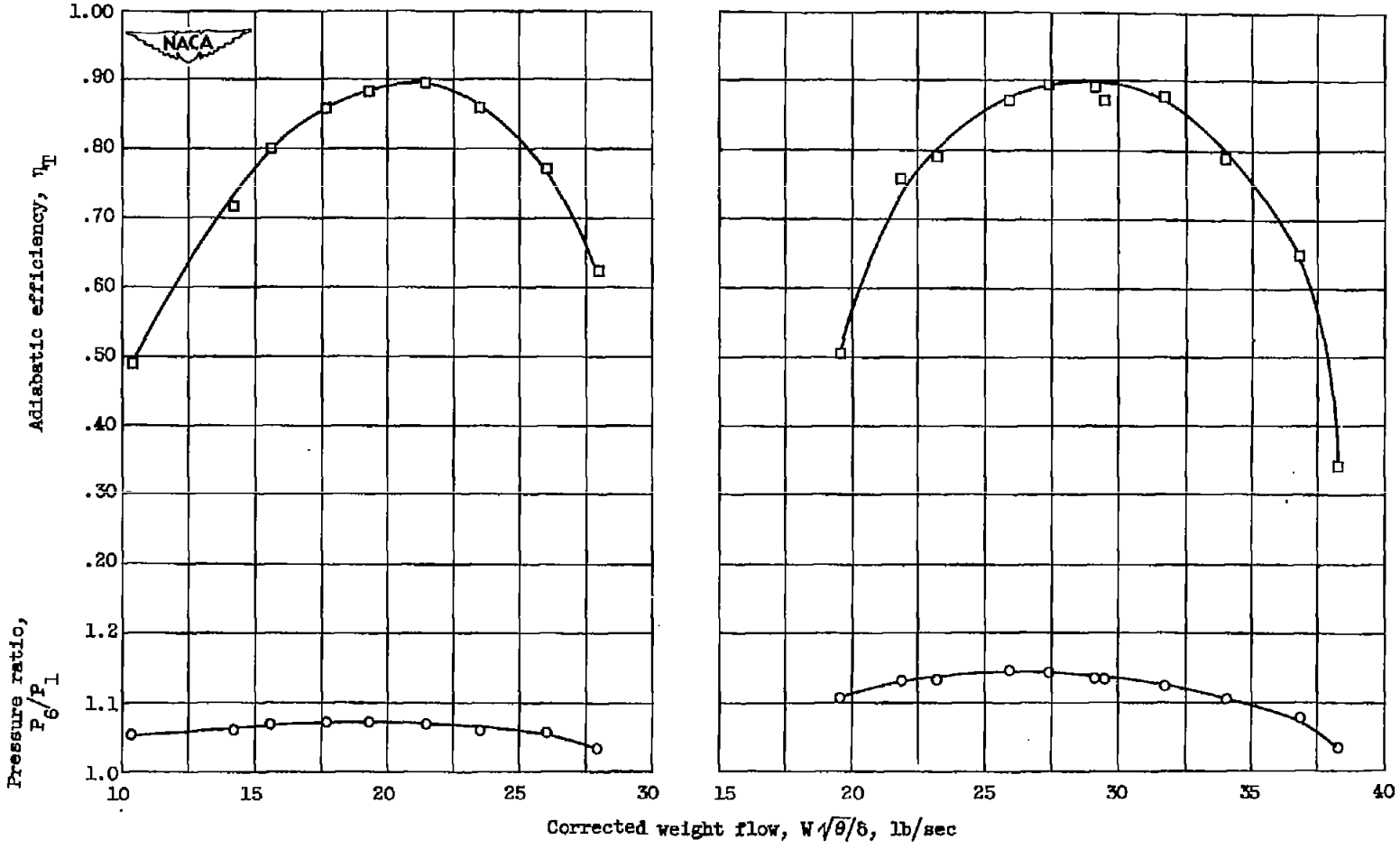

(a) Corrected tip speed, 430 feet per second.

(b) Corrected tip speed, 600 feet per second.

Figure 6. - Over-all performance of transonic inlet stage.

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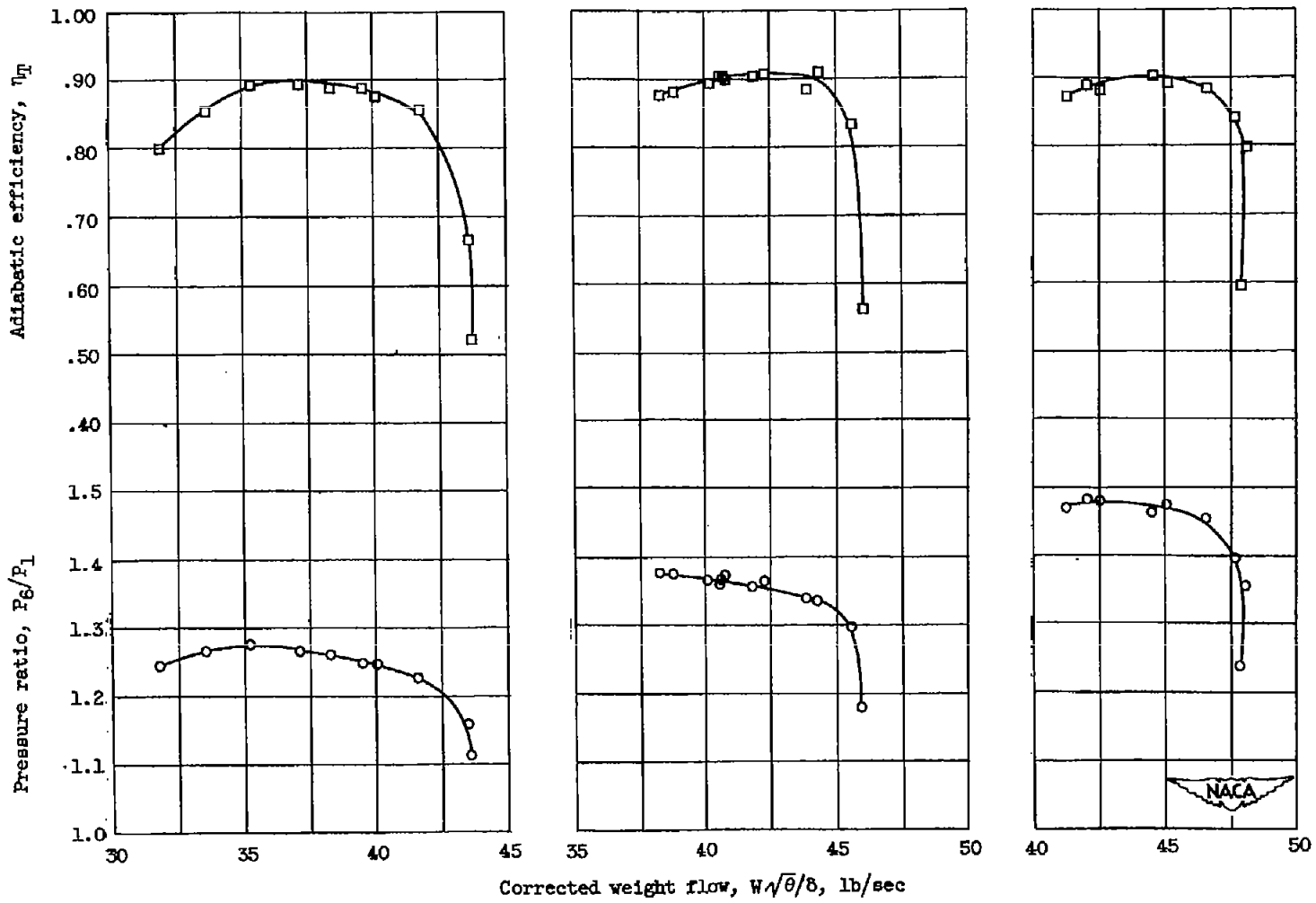

(c) Corrected tip speed, 800 feet per second.

(d) Corrected tip speed, 900 feet per second.

(e) Corrected tip speed, 1000 feet per second.

Figure 6. - Concluded. Over-all performance of transonic inlet stage.

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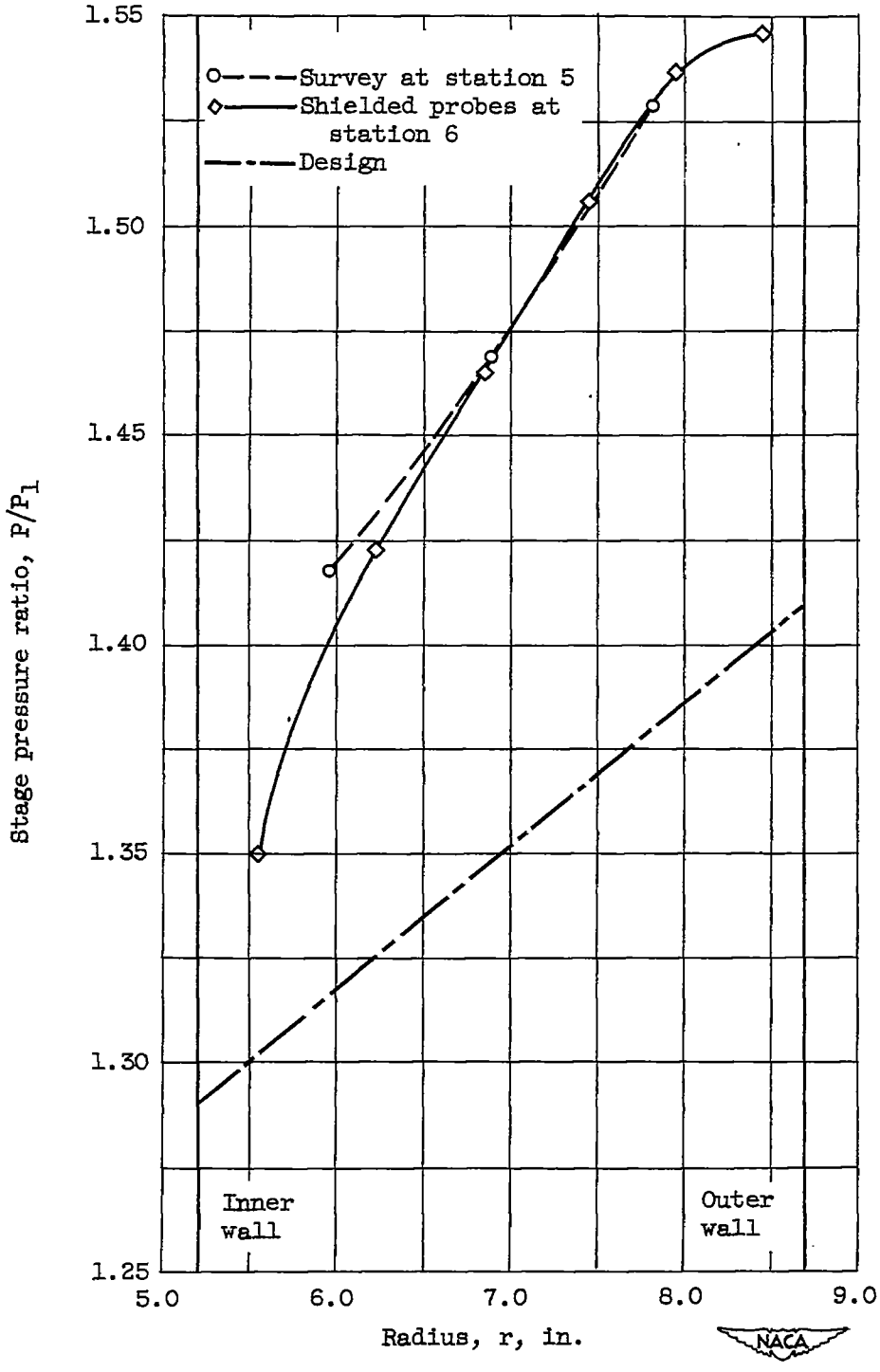
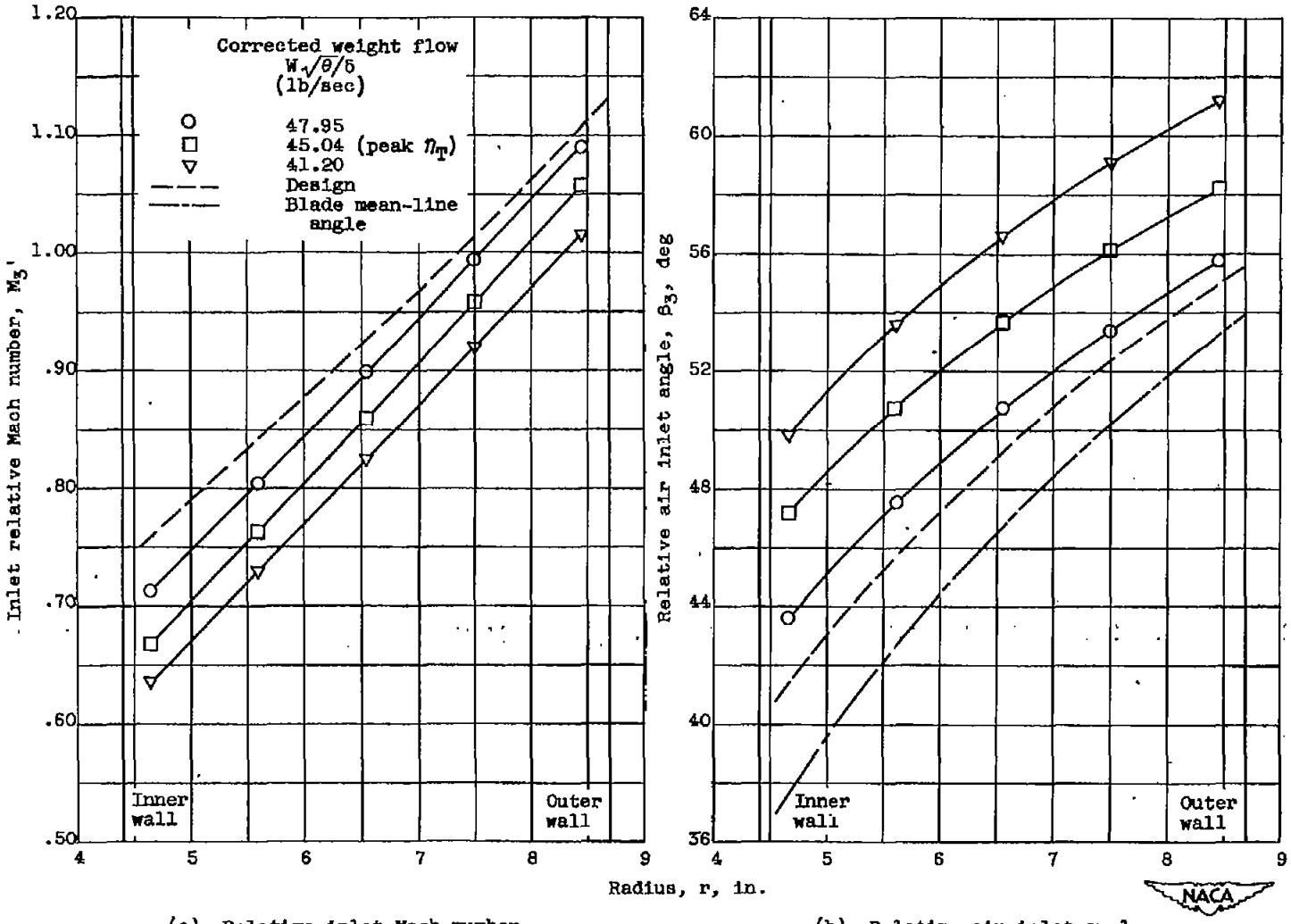


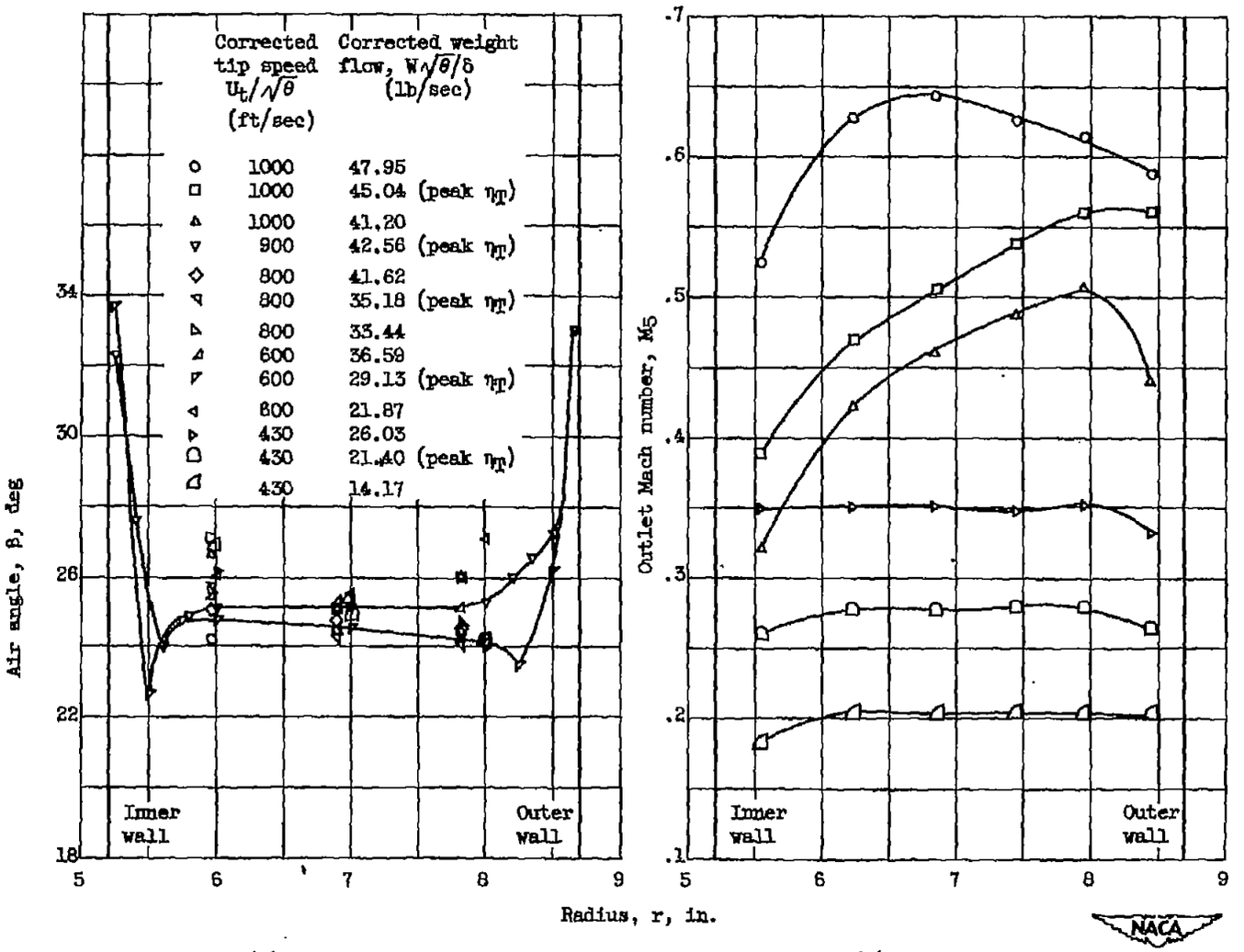
Figure 7. - Radial variation of total-pressure ratio at stage outlet. Peak efficiency at design corrected tip speed of 1000 feet per second.



(a) Relative inlet Mach number.

(b) Relative air inlet angle.

Figure 8. - Radial variation of relative Mach number and relative air angle at rotor inlet (station 3). Design corrected tip speed of 1000 feet per second.



(a) Air angle.

(b) Mach number.

Figure 9. - Radial variation of air angle and Mach number at stator outlet.

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