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# RESEARCH MEMORANDUM

SOME EFFECTS OF STATOR CONE ANGLE AND BLADE-TIP LEAKAGE  
ON 40-PERCENT-REACTION TURBINE HAVING ROTOR-BLADE CAPS

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## NATIONAL ADVISORY COMMITTEE FOR AERONAUTICS

RESEARCH MEMORANDUMSOME EFFECTS OF STATOR CONE ANGLE AND BLADE-TIP LEAKAGE  
ON 40-PERCENT-REACTION TURBINE HAVING ROTOR-BURR CAPSBy Robert E. English, Robert J. McCready  
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## SUMMARY

For an investigation of the effects of **stator** cone angle and tip **leakage** on turbine **performance**, a **single-stage** turbine **having** 40-percent reaction **was** operated with two **stators** and two **stationary** shrouds: (1) a **stator** **having** a cone **angle** of  $70^\circ$  and a **stator-blade** height 0.92 of the **rotor-blade** height; (2) a **stator** **having** a  $0^\circ$  cone **angle** and a blade height equal to the **rotor-blade** height; (3) a labyrinth, no-leakage shroud; and (4) a cylindrical stationary **shroud** **having** a radial **clearance** 0.016 of the blade **height**. In all cases, the **rotor** blades were equipped with caps that formed a **continuous**, cylindrical rotating shroud.

The turbine was operated at an **entrance** temperature of  $660^\circ$  R with total-pressure ratios from 1.25 to 3.70 and equivalent mean blade **speeds** from 166 to 655 feet per **second**. For this **range** of conditions, the over-all **performance** of the turbine was determined.

With the  $0^\circ$ -**cone-angle** **stator**, the peak brake efficiency was approximately 0.04 higher than with the  $70^\circ$ -**cone-angle** **stator**. With the  $70^\circ$ -**cone-angle** **stator**, separation probably occurred at the tips of the rotor blades; the separation was eliminated by **changing** the **cone** angle to  $0^\circ$  and **making** the **stator-blade** height equal to the rotor-blade height. **When** the labyrinth shroud was replaced by the cylindrical shroud, the efficiency was not **changed** by a measurable amount.

## INTRODUCTION

As part of a **program** to develop techniques for designing turbines of high efficiency, an investigation is being conducted **at**

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the **NACA** Lewis laboratory to determine the performance of a **single-stage**, cold-air turbine in which the relative importance of **design** variables is evaluated by **incorporating systematic** changes in the blade design. **This** investigation should provide data for the design of turbines and **indicate where further studies may most** profitably be concentrated.

The first phase of this program **is** an investigation of the effects of **stator** cone angle and **tip leakage** on turbine performance. **In** order to observe the effect of **stator** cone angle on **over-all** turbine performance, **two stators were used: a stator** with a  $70^\circ$  **cone** angle and a stator-blade height 0.92 of the rotor-blade **height;** and **a stator with a  $0^\circ$  cone angle** and a stator-blade height equal to the **rotor-blade** height. The effect of tip leakage on turbine performance was investigated by **operating the turbine** with a labyrinth no-leakage shroud and a cylindrical shroud.

Of the possible **combinations** of these variables, three configurations of the turbine were studied: 1, the  $70^\circ$ -**cone-angle stator** and the labyrinth **shroud;** (2) the  $0^\circ$ -**cone-angle stator** and the labyrinth **shroud;** and (3) the  $0^\circ$ -**cone-angle stator** and the cylindrical shroud. For the investigation of these three configurations, the turbine **was** operated at total-pressure ratios of 1.25 to 3.70 and equivalent mean blade **speeds** of 188 to 855 feet per **second using air at a temperature** of  $660^\circ$  R.

#### SYMBOLS

The following symbols are used in this **report:**

- a velocity of **sound**, (ft/sec)
- E** turbine-shaft work, (Btu/lb)
- g** standard acceleration due to gravity, 32.17 (ft/sec<sup>2</sup>)
- h specific **enthalpy**, (Btu/lb)
- J mechanical equivalent **of** heat, 778.3 (ft-lb/Btu)
- p absolute pressure, (lb/sq ft)
- T** absolute temperature, ( $^\circ$ R)

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- $U$  mean rotor-blade speed, (ft/sec)
  - $V$  absolute velocity of gas, (ft/sec)
  - $W$  gas velocity relative to rotor, (ft/sec)
  - $w$  Weight flow of air, (lb/sec)
  - $\alpha$  angle of absolute velocity measured from direction of rotor-blade motion, (deg)
  - $\beta$  angle of relative velocity measured from direction of rotor-blade motion, (deg)
  - $\Delta$  prefix denoting change
  - $\eta$  brake efficiency
  - $\rho$  mass density, (slug/cu ft)

**Subscripts:**

- $0$  NACA sea-level air; 2116 (lb/sq ft) and 518.4' R
- $1$  stator entrance
- $2$  rotor entrance
- $3$  rotor exit
- $j$  ideal Jet
- $s$  ieentropic
- $U$  tangential component
- $X$  axial component

**Superscript:**

- ' total, or stagnation, state

**TURBINE DESIGN**

The aerodynamic design of the turbine is for the following conditions:

Entrance absolute total temperature, $T_1'$ , °R	1960
Entrance absolute total pressure, $p_1'$ , lb/sq ft	2829
	in. Hg . . . . . 40
Mean rotor-blade speed, $U$ , ft/sec	1210
Total-to-static pressure ratio, $p_1'/p_3$	4.0
Amount of reaction, $\frac{\Delta_s h_{2,3}}{(h_1' - h_3)_s}$	0.40

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Because the experimental investigation was conducted at a low turbine-entrance temperature (660° R) as contrasted with the design conditions and because the performance data are expressed in terms of the entrance and exit total states, blade speed and pressure ratio are more conveniently stated in terms of equivalent mean blade speed at NACA sea-level air conditions and total-pressure ratio. The design conditions correspond to an equivalent mean blade speed  $U\left(\frac{a_0}{a_1'}\right)$  of 645 feet per second and a total-pressure ratio  $p_1'/p_3'$  of 3.32. The velocity-vector diagram for the mean section is given in figure 1. The turbine was designed for a stator exit angle  $a_2$  of 20°, a constant along the radius. Upstream of the stator, downstream of the rotor, and in the axial clearance space between the stator and the rotor, both the static pressure and the mass flow per unit of annulus area were assumed to be constant along the radius. The expansion in the turbine was assumed to be adiabatic, and the working fluid was assumed to have a ratio of specific heats equal to 1.300 and a specific heat at constant pressure equal to 0.300.

The turbine wheel was composed of 150 blades that were 1.50 inches long; the mean diameter midway between the rotor entrance and exit was 13.25 inches and the outer wheel diameter, 14.75 inches. (See fig. 2.) The blades were die-cast aluminum, each with an integral cap at the tip, so that with the blades clamped between two disks the caps formed a continuous, rotating shroud. (See fig. 3.) The rotor-blade profile is shown in figure 4.

### Stators

A stator (fig. 5) was available from another turbine whose rotor diameters and blade height were the same as those of the experimental turbine; in addition, the trailing-edge angle of the stator blades was equal to 20°. If the assumptions used in designing the rotor blades are applied to this stator and the effects of rotor-blade overlap are neglected, the stator meets the design conditions at the entrance to the rotor. The stator had a cone angle of 70° and

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contained 28 stator blades. The atator-blade height was about  $\frac{1}{8}$  inch less than the blade height at the rotor entrance, or 0.92 of the rotor-blade height. This stator was cast in one piece.

A second stator (fig. 6), which was of welded construction, had a  $0^\circ$  cone angle, and contained 36 etator blades. The stator-blade height was made equal to the blade height at the rotor entrance. If the assumptions used in designing the rotor blades are applied to this stator, the atator meets the design conditions at the rotor entrance.

Four difference8 existed in the two stators; namely, the method of construction, the number of etator blades, the cone angle, and the stator-blade height. The internal surfaces of the  $70^\circ$ -cone-angle stator were hand-finished so that the surfaces of both stators were of approximately equal roughness. Because the solidity is high and separation is unlikely to occur in either stator, the number of etator blades should have little effect. The cone angle and the stator-blade height are the significant differences. With each stator, the minimum axial clearance between the stator blades and the rotor blades was between 0.120 and 0.140 inch.

### Shrouds

The two shrouds used in this investigation are shown in figure 7. The labyrinth, no-leakage, stationary shroud (fig. 7(a)) consists of a circular ring with two circumferential groove8 on the inner surface. The radial clearance between the labyrinth stationary shroud and the cylindrical rotating shroud formed by the blade caps is 0.025 inch. Leakage between the labyrinth shroud and the blade caps was prevented by introducing air into the space downstream of the labyrinth shroud. The amount of labyrinth-sealing air was so adjusted that the pressures in the two circumferential grooves were equalized; when the pressures were equal, it was assumed that no air leaked past the shroud.

The cylindrical stationary shroud (fig. 7(b)) is the same width as the blade caps. The radial clearance between the blade caps and the shroud was 0.025 inch, or 0.016 of the blade height.

### Configurations

The three turbine configurations investigated are shown in figure 8. Configuration 1 includes the  $70^\circ$ -cone-angle stator and

the labyrinth stationary **shroud**; configuration 2, the **0°-cone-angle stator** and the labyrinth stationary shroud; and configuration 3, the **0°-cone-angle stator** and the **cylindrical** stationary shroud.

## APPARATUS AND PROCEDURE

### Experimental Equipment

The **arrangement** of the experimental equipment is **diagrammatically** shown in figure 9. Room air enters the electrostatic precipitator where dust particles are removed and passes through a submerged flat-plate orifice, an automatically controlled **steam-supplied** air heater, a surge tank, and into a pair of ducts leading to the plenum **chamber** immediately upstream of the turbine **stator**. The air-flow path from the plenum chamber through the turbine is shown in figure 10 with the labyrinth shroud and the **70°-cone-angle stator** of **configuration 1** installed in the turbine. Air entering the plenum chamber passes through an **18-mesh**, 27-gage **screen**, turns, and flows through a straightening grid. The air then passes through the **stator** and the rotor and leaves the turbine through the annular passage between the two exhaust-guide shells coaxially mounted in the tail pipe. The space **between the** outer guide **shell and the** tail pipe is provided to carry off the sealing air supplied to the labyrinth.

Downstream of the turbine, the air flows through a **large** surge tank into a **low-pressure** exhaust system (fig. 9). The air pressure drop **across** the turbine is controlled by a butterfly valve **downstream of** the second surge tank. **A 300-horsepower water brake,** cradle-mounted for torque measurements, **was** used for power absorption.

### Instrumentation

The location of the turbine instrumentation is shown in figure 10. Entrance total pressure  $p_1'$  was indicated by a **total-pressure** tube **3/4** inch upstream of the **stator** blades; the entrance total temperature  $T_1'$  **was measured with thermocouples at four** stations within the pair of ducts leading to the plenum chamber. The exit static pressure  $p_3$  was measured by six wall taps located in the exhaust-guide shells downstream of the rotor. The exit total temperature  $T_3'$  was measured by three total-temperature **thermocouples** at the downstream end of the two exhaust-guide shells.

Water-brake torque was indicated by an **NACA balanced-diaphragm dynamometer-torque indicator (reference 1)**. Air flow  $w$  was measured by the **7-inch** submerged flat-plate orifice upstream of the air heater (fig. 9). Turbine speed was **indicated** by a calibrated electric tachometer.

The accuracy **in reading the data may be summarized as follows:**

- (1) Absolute pressure,  $\pm 0.05$  inch of mercury
- (2) **Temperatures,  $\pm 1^\circ$  R**
- (3) Orifice pressure drop,  $\pm 0.05$  inch in a range with a minimum of **15 inches**
- (4) Torque,  $\pm 0.05$  inch in a range with a minimum of 17 inches
- (5) Rotative speed,  $\pm 13$  rpm in a range with 8 minimum of 3700 rpm

An estimate of the reproducibility of the **measurements** indicates that the probable variation in the **brake efficiency for 8 single data point is** less than 0.01 for total-pressure ratios  $p_1'/p_3'$  **greater than 2.00.**

### Experimental Procedure

Data were taken at turbine speeds from 3700 to 16,800 rpm (equivalent mean blade speeds from 188 to 855 ft/sec) at total-pressure ratios  $p_1'/p_3'$  from 1.25 to 3.70. At each of ten pressure ratios, the turbine was operated at six to ten different speeds. For all runs the entrance total temperature was held between  $658^\circ$  and  $662^\circ$  R; the entrance total pressure varied with the air flow and the barometric pressure between  $26\frac{1}{2}$  and 28 inches of mercury.

### PERFORMANCE CALCULATIONS

All turbine performance data were reduced to **NACA sea-level air conditions** at the turbine entrance. The performance was determined in terms of the following variables:

- (1) Brake efficiency,  $\eta$
- (2) Total-pressure ratio,  $p_1'/p_3'$
- (3) Equivalent turbine-shaft work,  $E \left( \frac{a_0}{a_1} \right)^2$
- (4) Equivalent mean rotor-blade speed,  $U \left( \frac{a_0}{a_1} \right)$

- (5) Ratio of equivalent mean rotor-blade speed to equivalent mass flow,  $\frac{U\rho_1'}{w\rho_0}$  (This is the reciprocal of mass flow per unit blade speed.)
- (6) Ratio of mean rotor-blade speed to ideal jet speed (blade-to-jet speed ratio),  $U/V_j$

The brake efficiency of the turbine, which includes the mechanical efficiency of the turbine, is defined as

$$\eta = \frac{E}{\Delta_S h_{1,3}'}$$

The weight flow of air  $w$  was determined from the orifice measurements and the data of reference 2.

The ideal enthalpy drop  $\Delta_S h_{1,3}'$  was computed using the chart of air properties in reference 3 and the values of entrance total pressure and temperature and exit total pressure,  $p_1'$ ,  $T_1'$ , and  $p_3'$ , respectively. Because the exit total pressure  $p_3'$  was not directly measured, this value was computed. The exit static pressure  $p_3$ , the exit total temperature  $T_3'$ , the weight flow  $w$ , and the annular area between the exhaust-guide shells were used to determine the exit total pressure  $p_3'$  by adding to the measured static pressure 8 dynamic pressure computed from continuity with the assumptions that the tangential velocity is zero and the axial velocity is constant; these assumptions result in efficiency values that are always conservative.

The ideal Jet speed  $V_j$  is the Jet speed for an isentropic expansion from the entrance total state to the exit static pressure, that is,

$$V_j = \sqrt{2Jg (h_1' - h_3)_S}$$

The isentropic enthalpy drop  $(h_1' - h_3)_S$  was computed from reference 3 using the entrance total temperature and pressure and the exit static pressure,  $p_1'$ ,  $T_1'$ , and  $p_3$ , respectively.

## RESULTS AND DISCUSSION

The performance characteristics over the range of operation covered in this investigation are shown for configurations 1, 2, and 3 in figures 11, 12, and 13, respectively. For each of the

three configurations, the **curves** of **constant** equivalent blade speed  $U \left( \frac{a_0}{a_1'} \right)$  are vertical above a pressure ratio  $p_1'/p_3'$  of approximately 2.25 because the flow through the **stator** blades was **choking** and the equivalent weight flow therefore remained **constant**. The over-all **performance characteristics** are similar for the three configurations and each configuration has an efficiency **near** the maximum over the range of pressure ratio from 1.25 to 3.70.

Brake efficiency  $\eta$  plotted against the ratio of blade-to-Jet speed ratio  $U/v_j$  (fig. 14) is nearly independent of total-pressure ratio  $p_1'/p_3'$  for **configuration 2**. Because this **independence** of pressure ratio also exists for configurations 1 and 3, the **comparison** of the three configurations in figure 15 at a pressure ratio of 3.00 **is** representative of the entire range of pressure ratios. **In** addition to showing the same **changes** in brake efficiency as figures 11 to 13, figures 14 and 15 indicate that the maximum brake efficiency was obtained with a blade-to-jet speed ratio  $U/v_j$  between 0.50 and 0.55. **The** maximum brake **efficiency** was 0.795 for configuration 1 and 0.833 for **configuration 2**, a **difference** of **approximately** 0.04. With the **70°-cone-angle stator**, separation probably occurred at the tips of the rotor **blades**; the **performance** was improved by **changing** the cone angle to **0°** and making the stator-blade **height** equal to the rotor-blade height, thereby eliminating the separation (figs. 8(a) and 8(b)). **If** the efficiency were computed using the total-to-stator pressure ratio  $p_1'/p_3$  instead of the total-pressure ratio  $p_1'/p_3'$ , the relative **advantage** of configuration 2 over configuration 1 would be substantially **unchanged**.

For configuration 3 (fig. 8(c)), the **maximum brake efficiency** was 0.835, an increase of less than 0.005 over the maximum **efficiency** of configuration 2. Although any tip leakage **introduced** by **replacing** the labyrinth no-leakage shroud with the **cylindrical** shroud would cause a **reduction** in the working fluid passing through the rotor blades, the measured **efficiency** slightly increased. Within the probable reproducibility of the data, the performance of **configuration 2** is identical to the performance of configuration 3.

In order to determine the approximate magnitude of the leakage air flow between the blade caps and the cylindrical stationary shroud, the **weight** flow through 8 thin-plate Orifice of equal area was **computed** and multiplied by a factor of 0.7 (reference 4). The conditions for **maximum leakage** flow **occurred** at the highest blade speed and pressure ratio investigated; at these **conditions** the Computed leakage was about 1.7 percent of the air flow entering the turbine. Because the

leakage air flow is mainly from the boundary layer at the outer radius of the flow annulus, this air contains only a small portion of the total energy passing through the turbine.

For configuration 3 (figs. 8(c) and 13), the brake efficiency was 0.82 at the total-pressure ratio and equivalent mean blade speed for which the blades were designed.

### SUMMARY OF RESULTS

An investigation of the effects of stator cone angle and blade-tip leakage on turbine performance was conducted with a turbine having 40-percent reaction and 8 design assumption of constant static pressure over the blade height. Caps at the tips of the rotor blades formed a continuous rotating shroud. The turbine was operated with a turbine-entrance temperature of approximately  $660^{\circ}$  R at total-pressure ratios from 1.25 to 3.70 and equivalent mean rotor-blade speeds of 188 to 855 feet per second. The following results were obtained:

1. With the  $0^{\circ}$ -cone-angle stator, the peak brake efficiency was approximately 0.04 higher than with the  $70^{\circ}$ -cone-angle stator. With the  $70^{\circ}$ -cone-angle stator, separation probably occurred at the tips of the rotor blades; the performance was improved by changing the cone angle to  $0^{\circ}$  and making the stator-blade height equal to the rotor-blade height, thereby eliminating the separation.
2. Replacing the labyrinth, no-leakage shroud with a cylindrical stationary shroud that had a radial clearance of 0.016 of the blade height from the cylindrical rotating shroud and using the  $0^{\circ}$ -cone-angle stator produced no measurable change in brake efficiency.
3. The maximum brake efficiency was obtained with the blade-to-Jet speed ratio between 0.50 and 0.55.

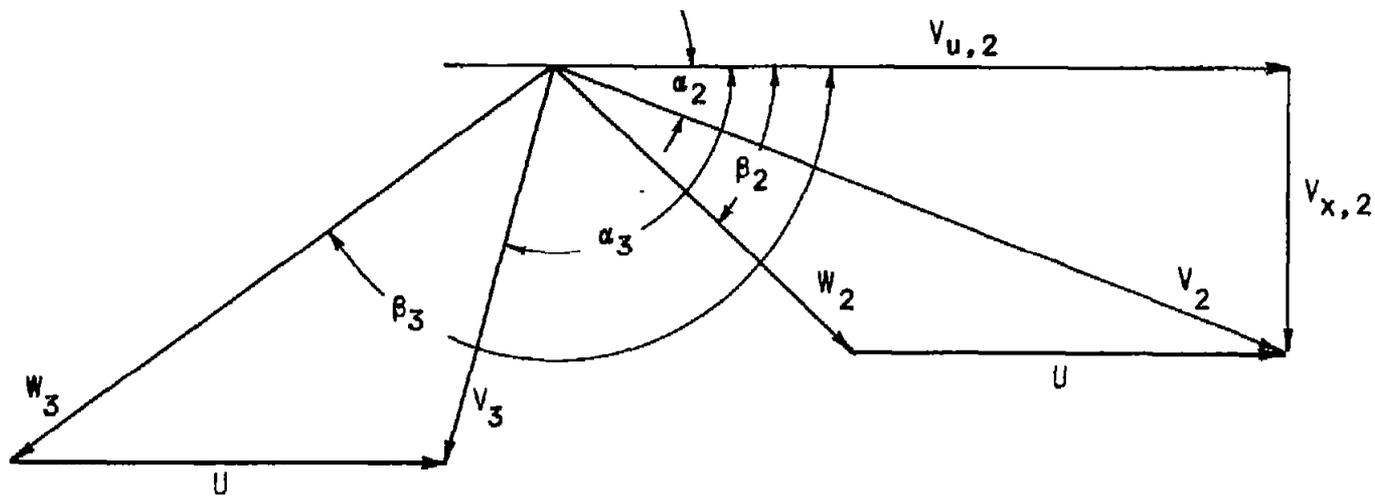
Lewis Flight Propulsion Laboratory,  
National Advisory Committee for Aeronautics,  
Cleveland, Ohio.

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Velocity (ft/sec)	Angle (deg)
U 1210	$\alpha_2$ 20.0
"2 2177	$\alpha_3$ 106.4
V <sub>3</sub> 1281	82 41.7
W <sub>2</sub> 1120	83 145.5
W <sub>3</sub> 2081	



Figure 1. - Velocity-vector diagram for mean section.

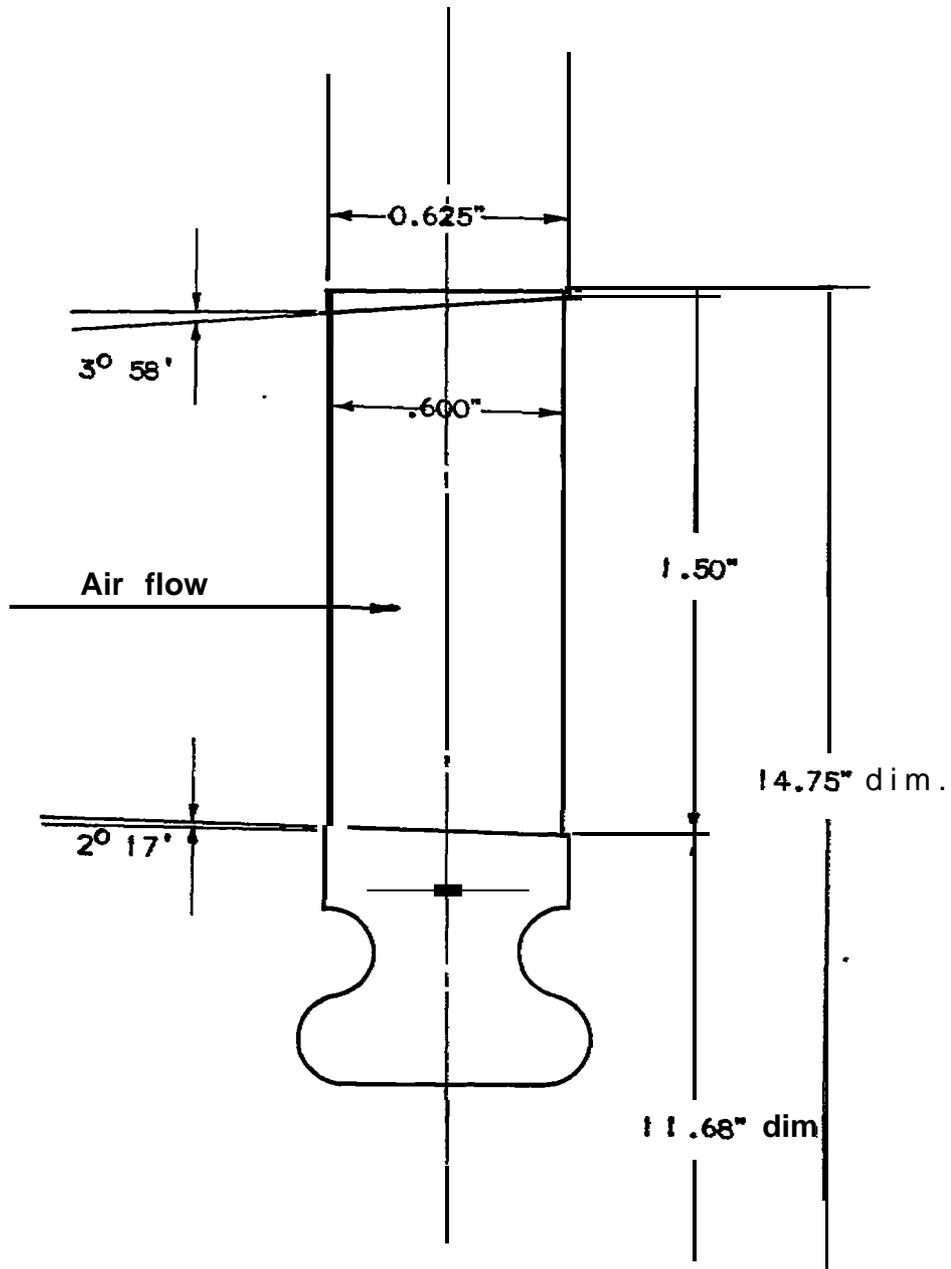


Figure 2. - Sketch of rotor blade showing turbine dimensions.



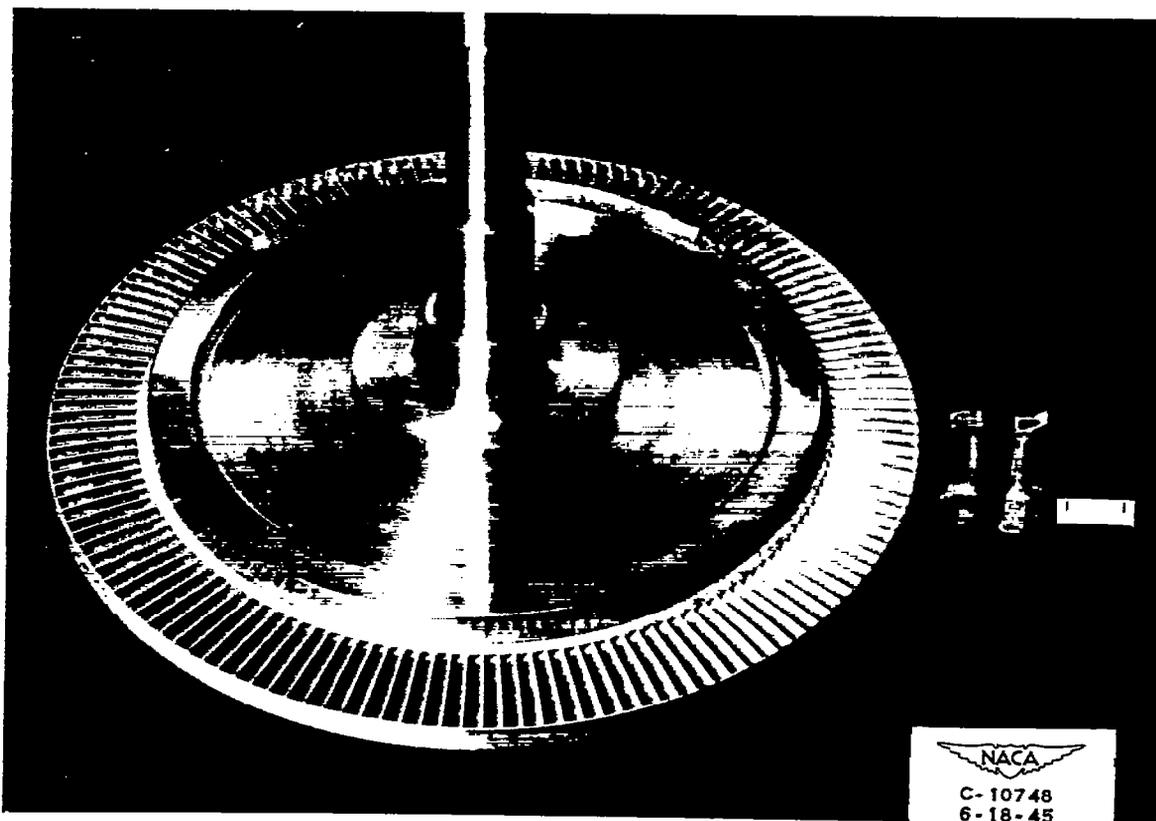


Figure 3. - Photograph of assembled rotor showing rotating cylindrical shroud formed by blade caps.



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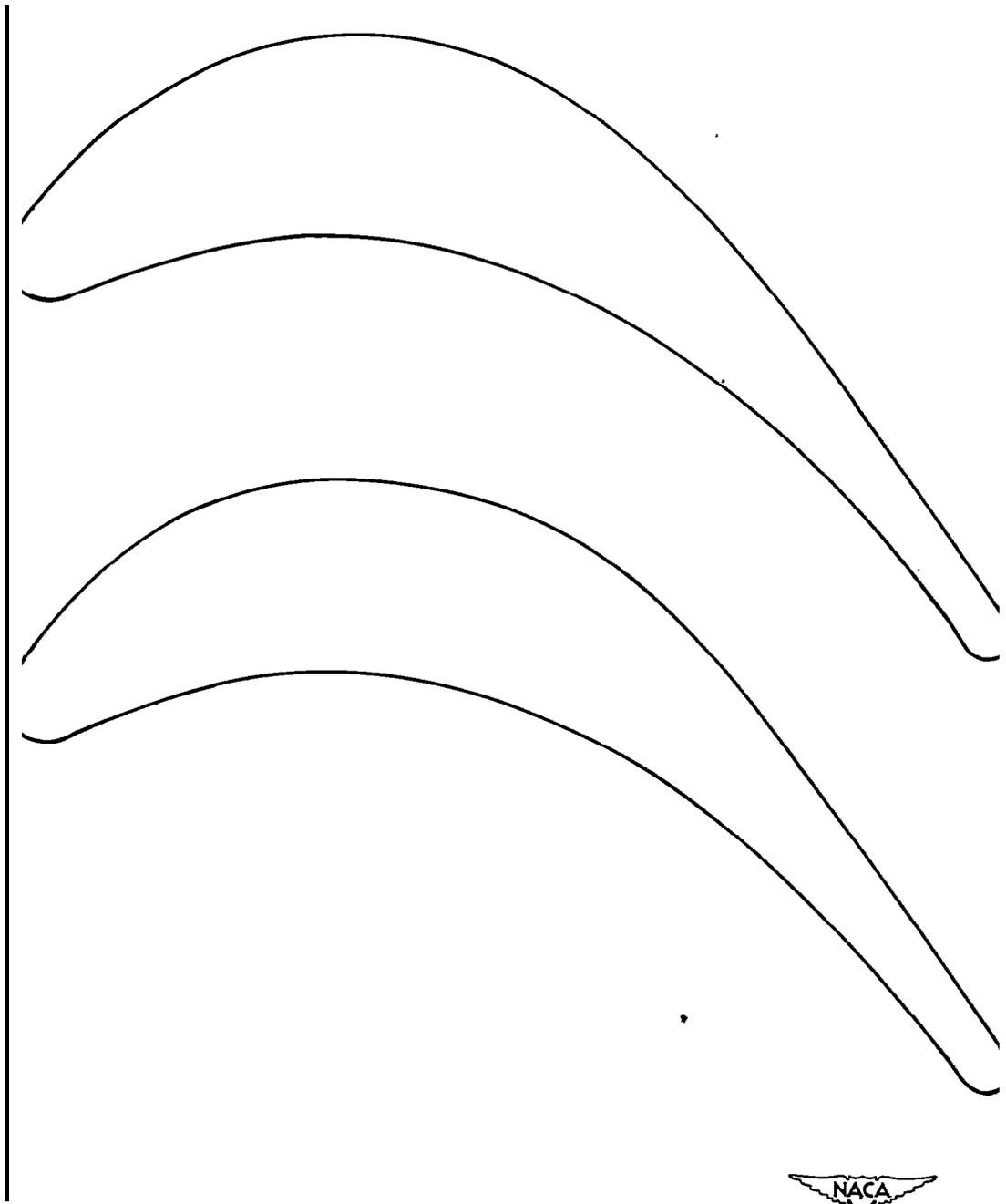
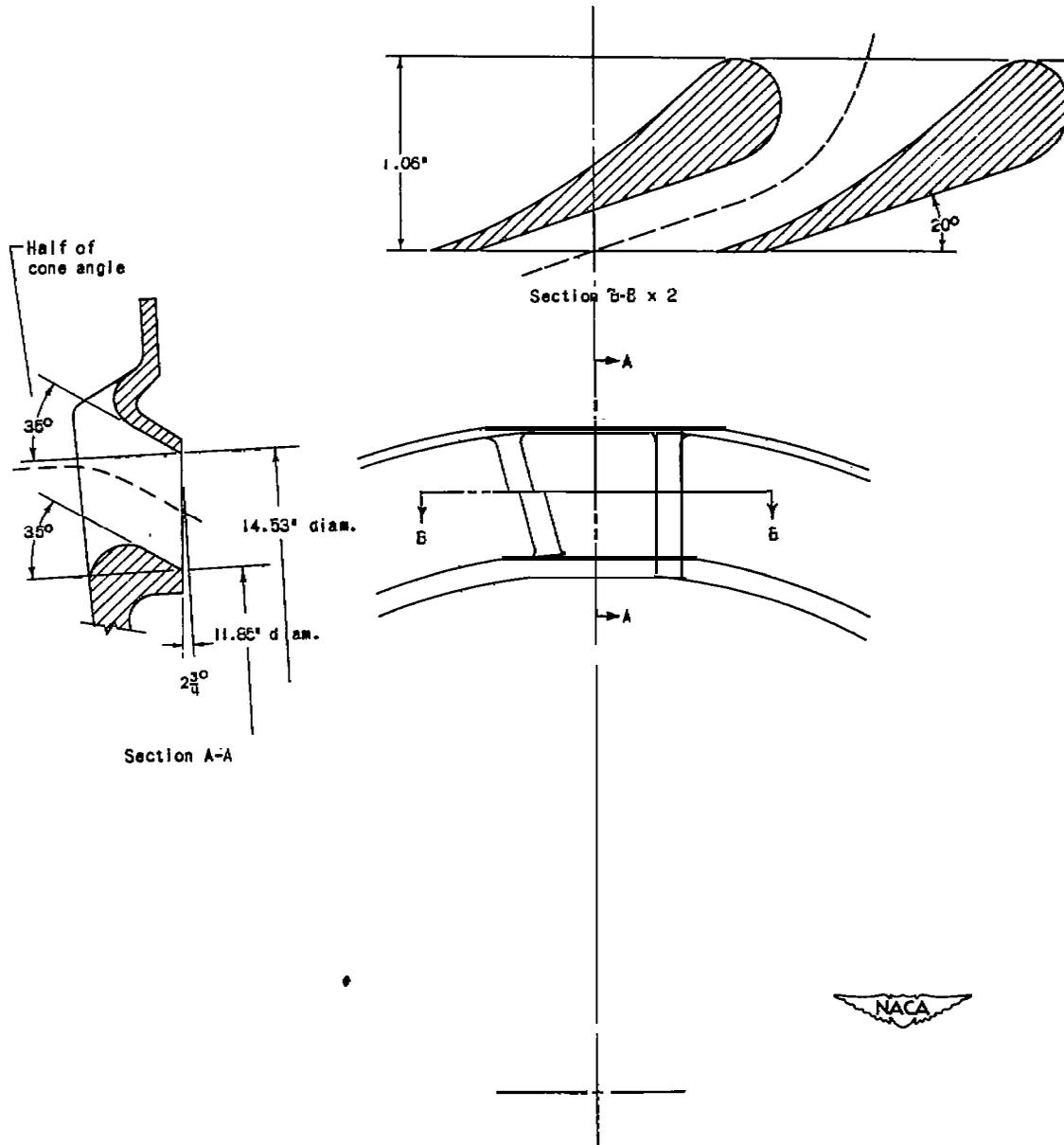


Figure 4. - Diagram showing rotor-blade profile (mean section).



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Figure 5. - 70°-cone-angle turbine stator; 28 stator blades.

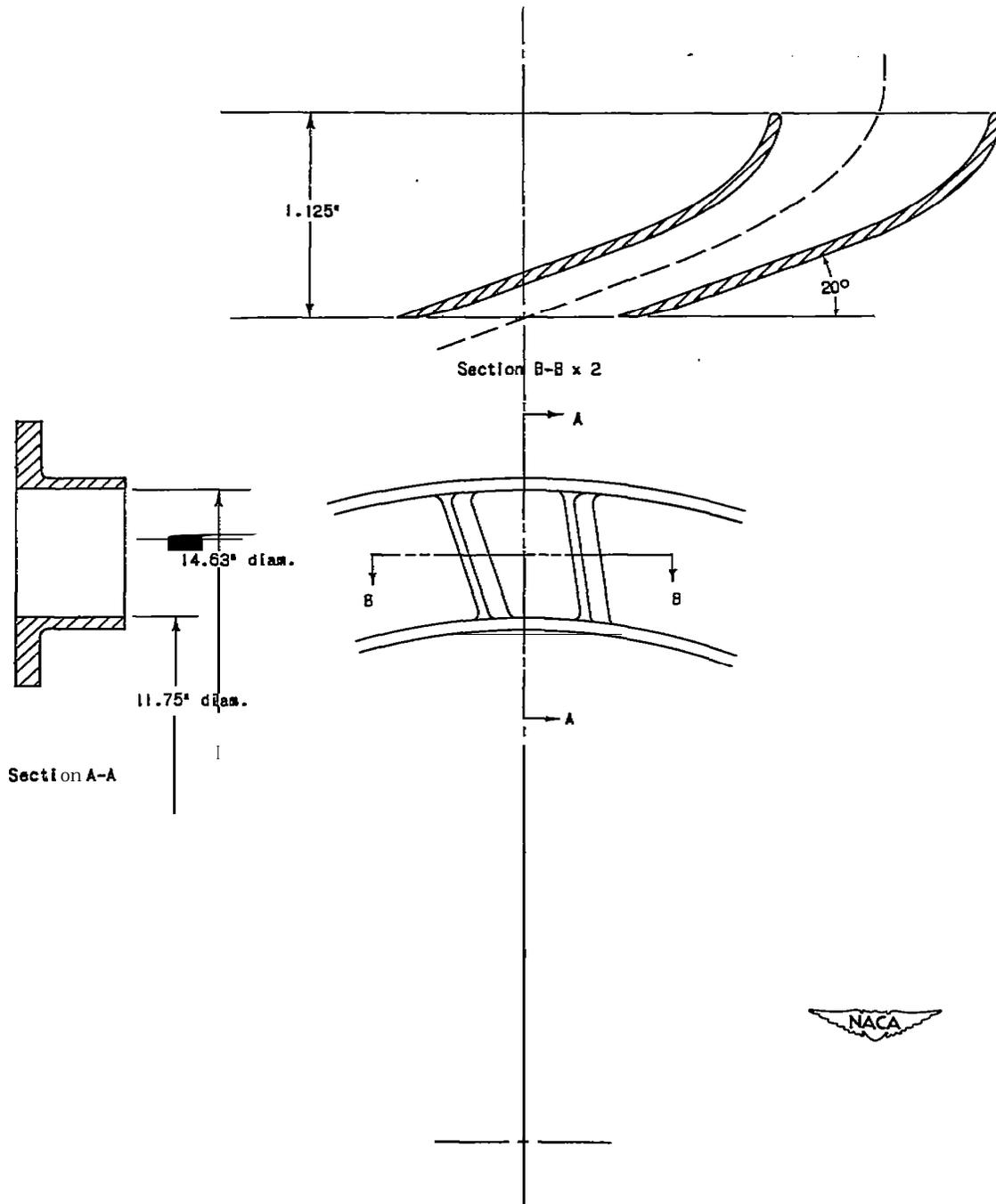


Figure 6. - 0°-cone-angle turbine stator; 36 stator blades.

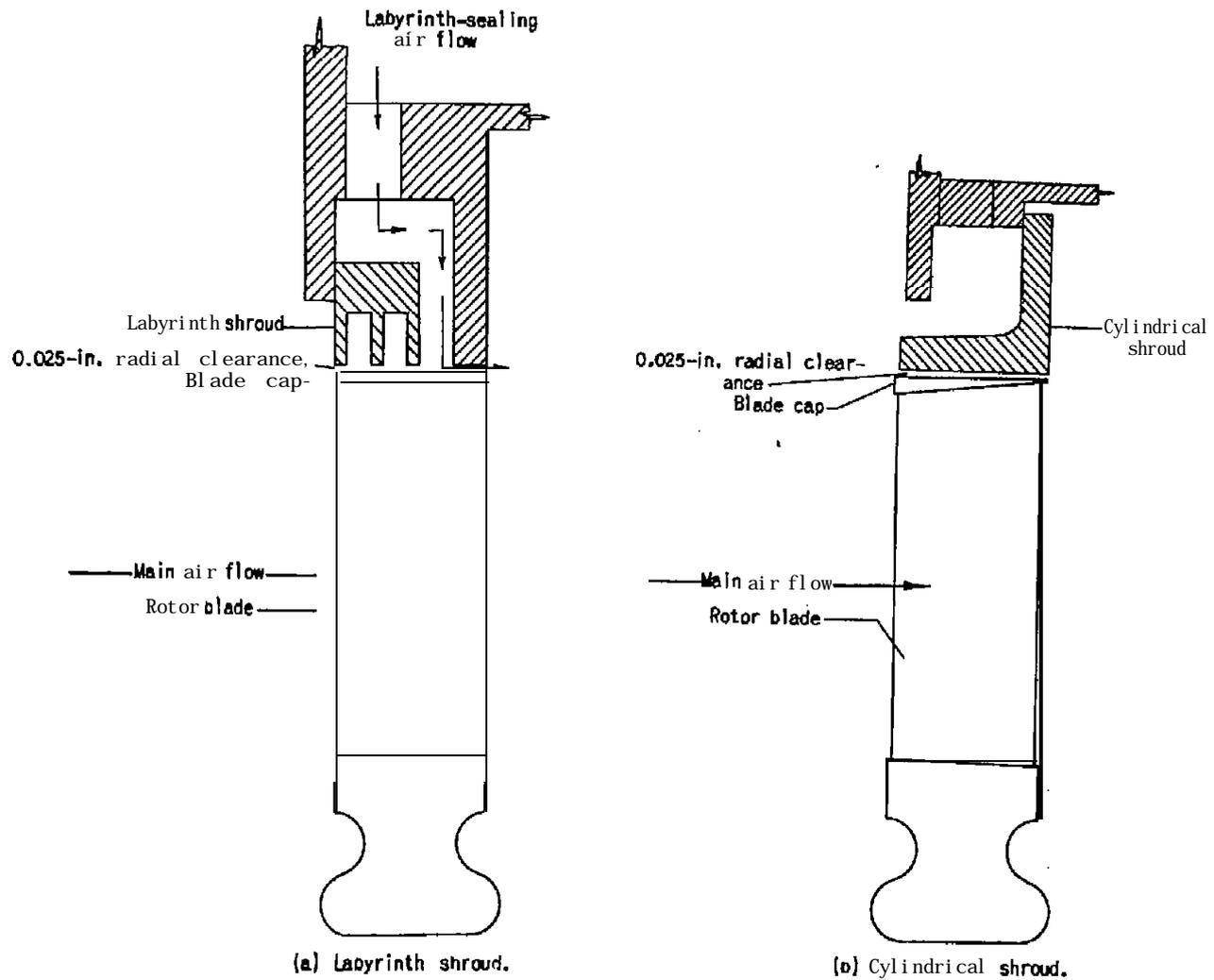
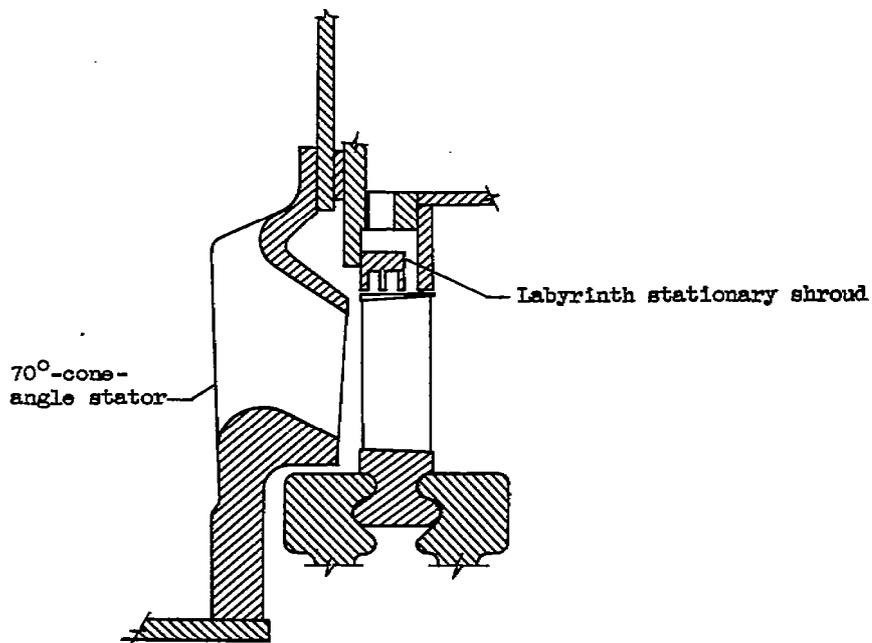
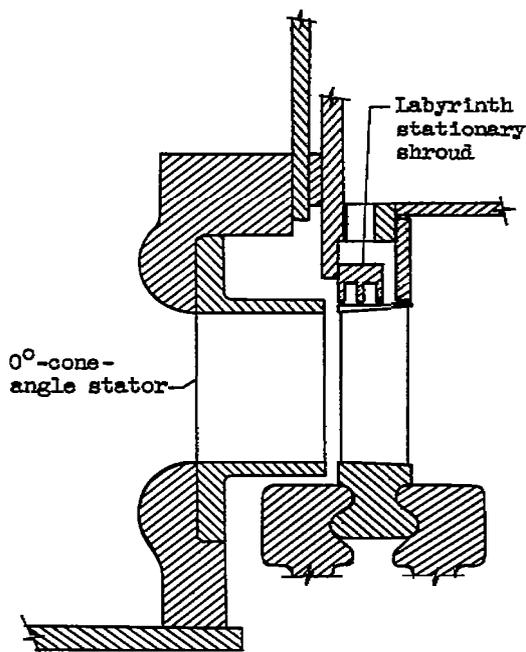


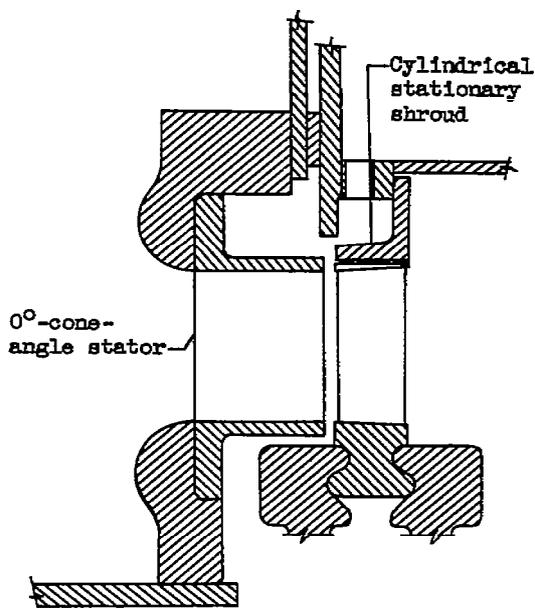
Figure 7. - Installation of stationary shrouds showing labyrinth-sealing air-flow path and radial clearance.



(a) Configuration 1.



(b) Configuration 2.



(c) Configuration 3.

Figure 8. - Three turbine configurations investigated.



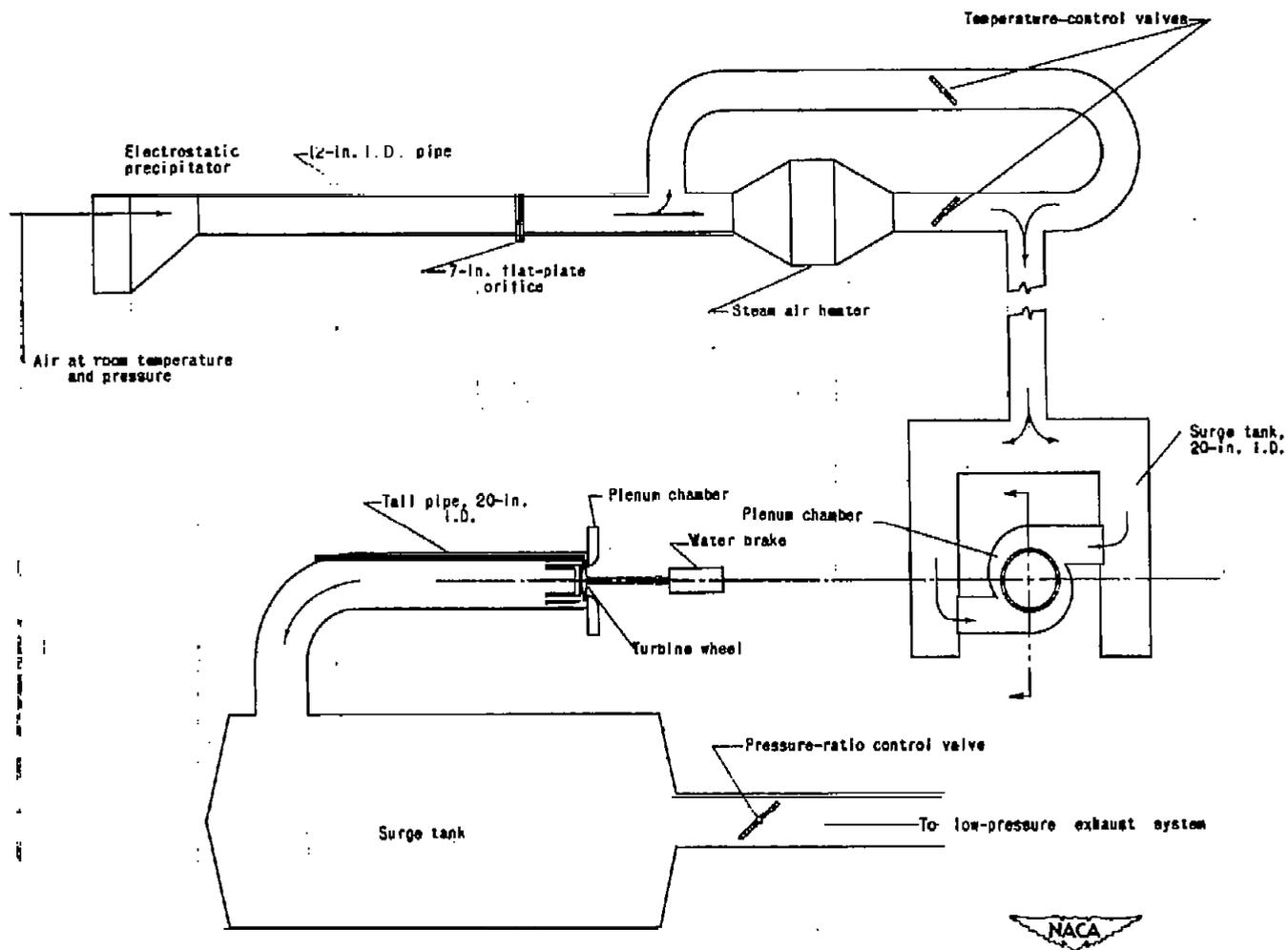


Figure 9. - Diagrammatic sketch of experimental-equipment arrangement showing pressure-ratio control valve, air-flow path, and auxiliary equipment.

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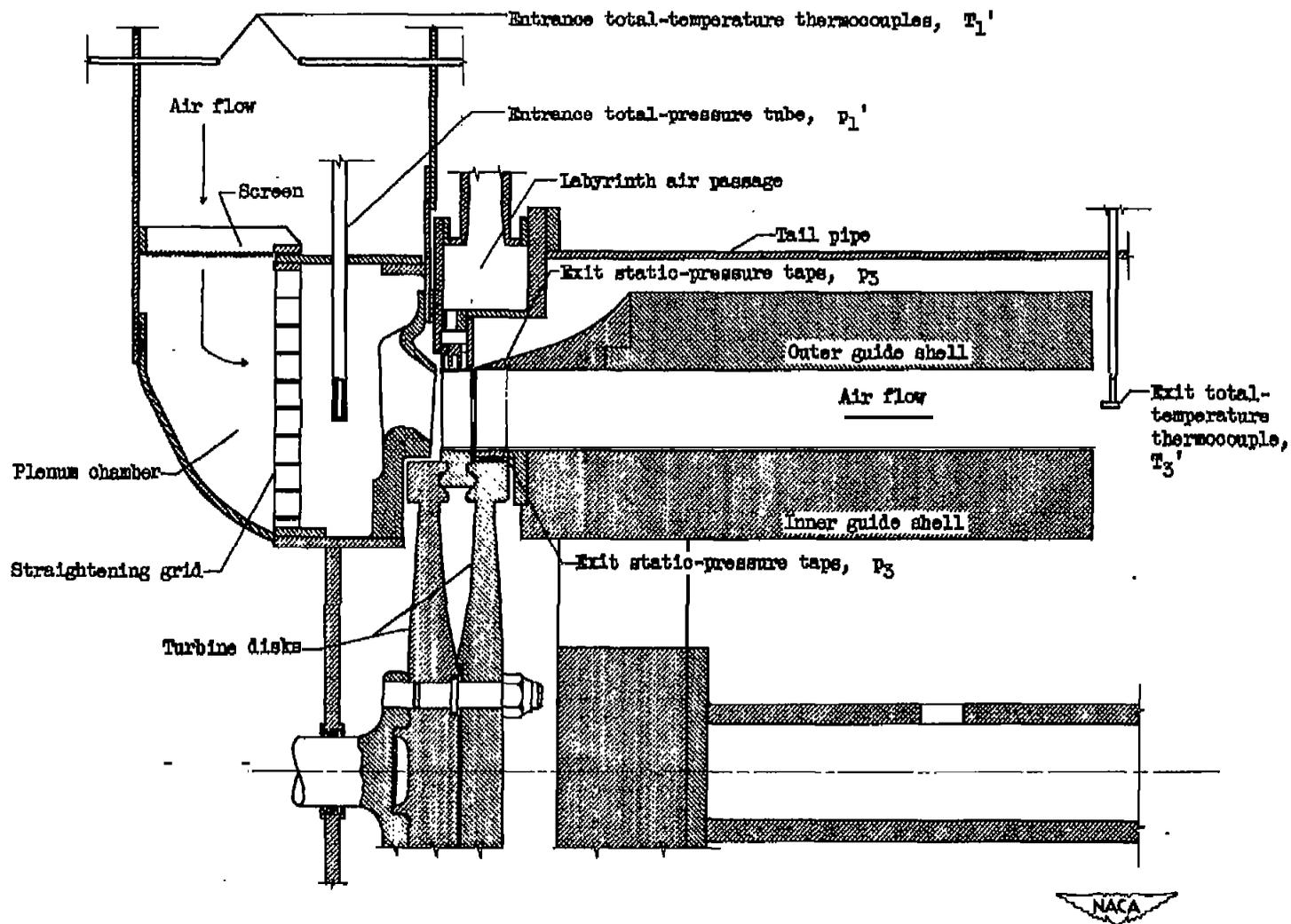


Figure 10. - Cross section through turbine center line (configuration 1) showing turbine arrangement and instrumentation.

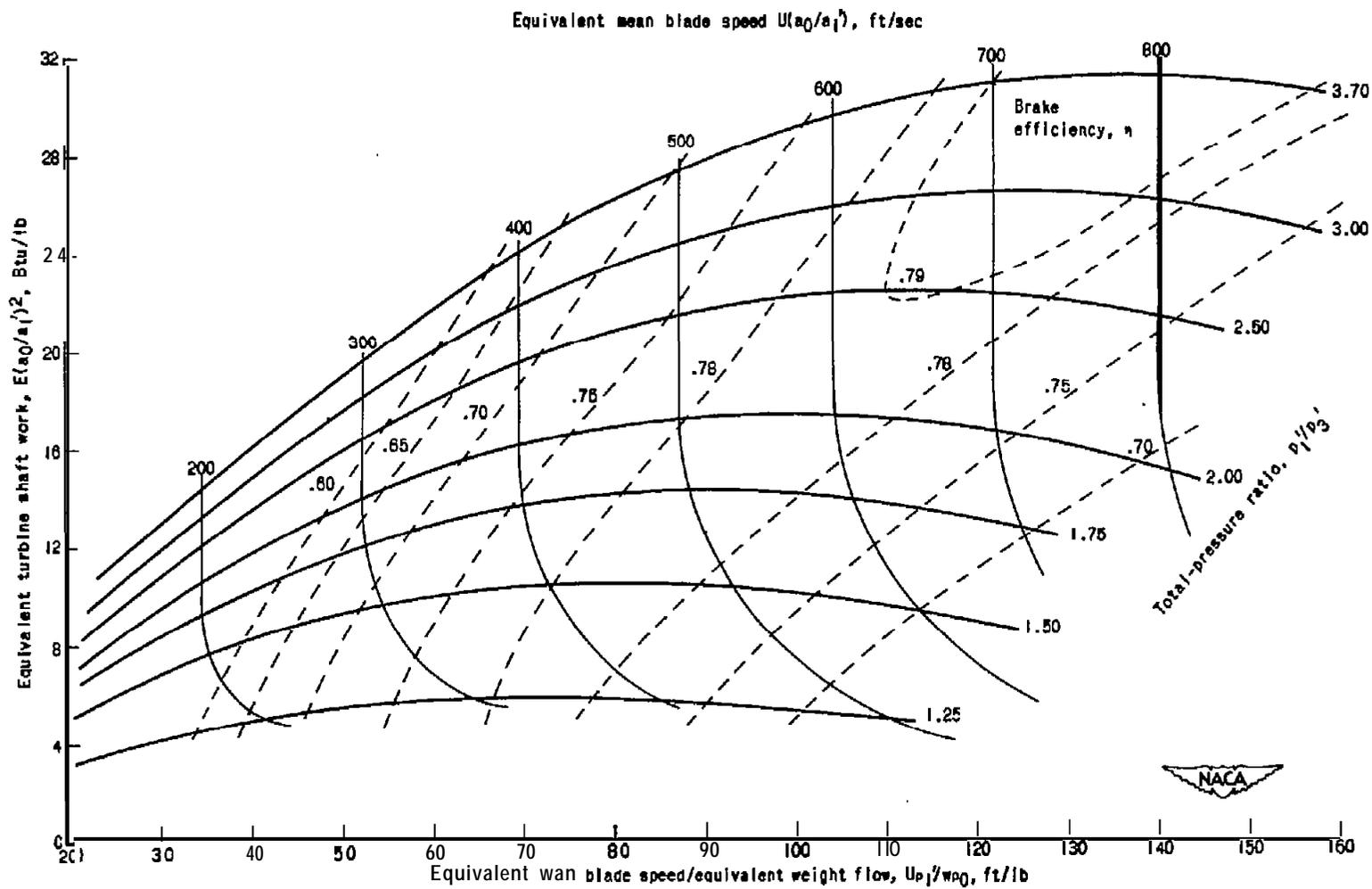


Figure 11. - Over-all performance chart with  $70^\circ$ -cone-angle stator and labyrinth shroud (configuration I).



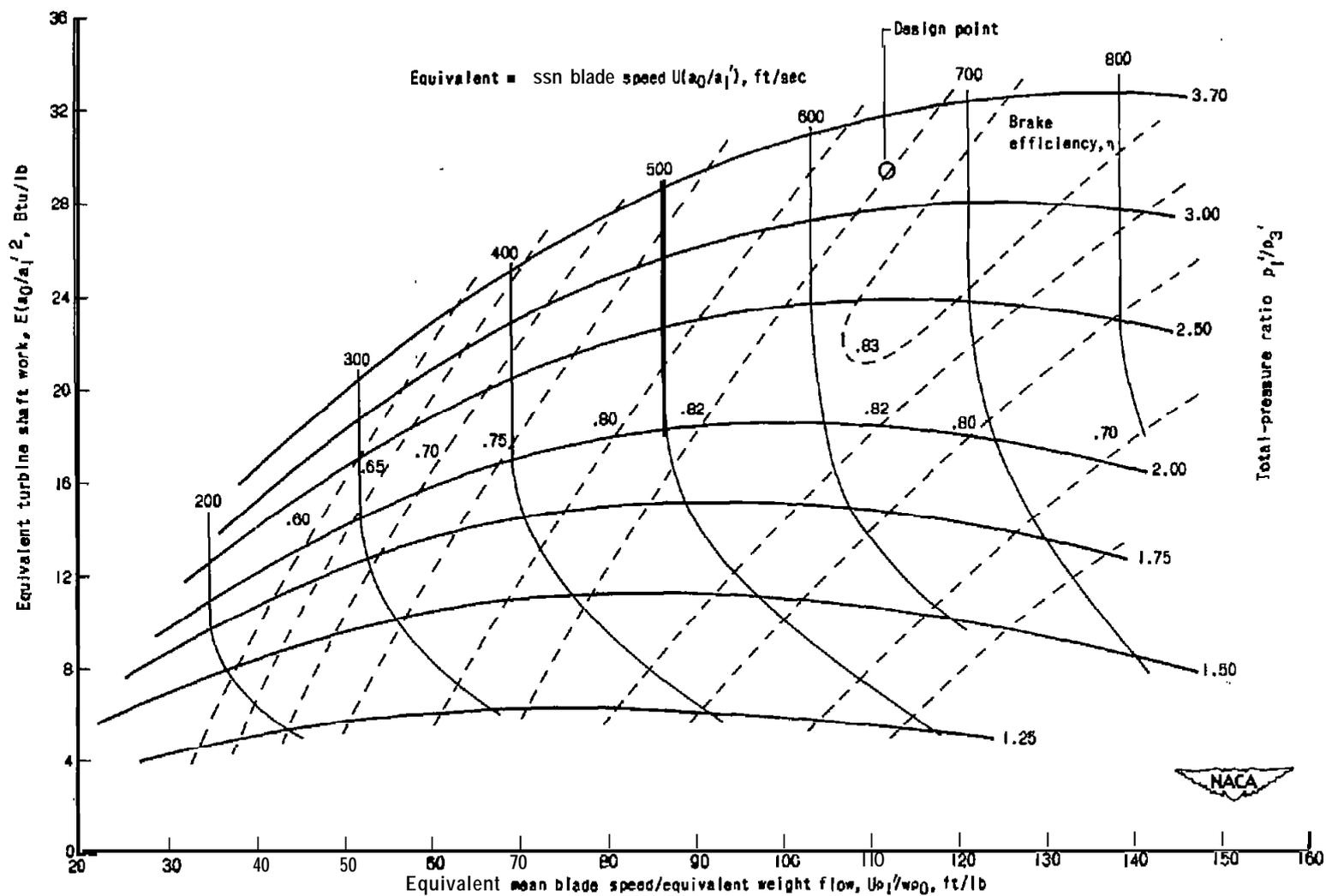


Figure 13. - Over-all performance chart with  $0^\circ$ -cone-angle stator and cylindrical shroud (configuration 3).

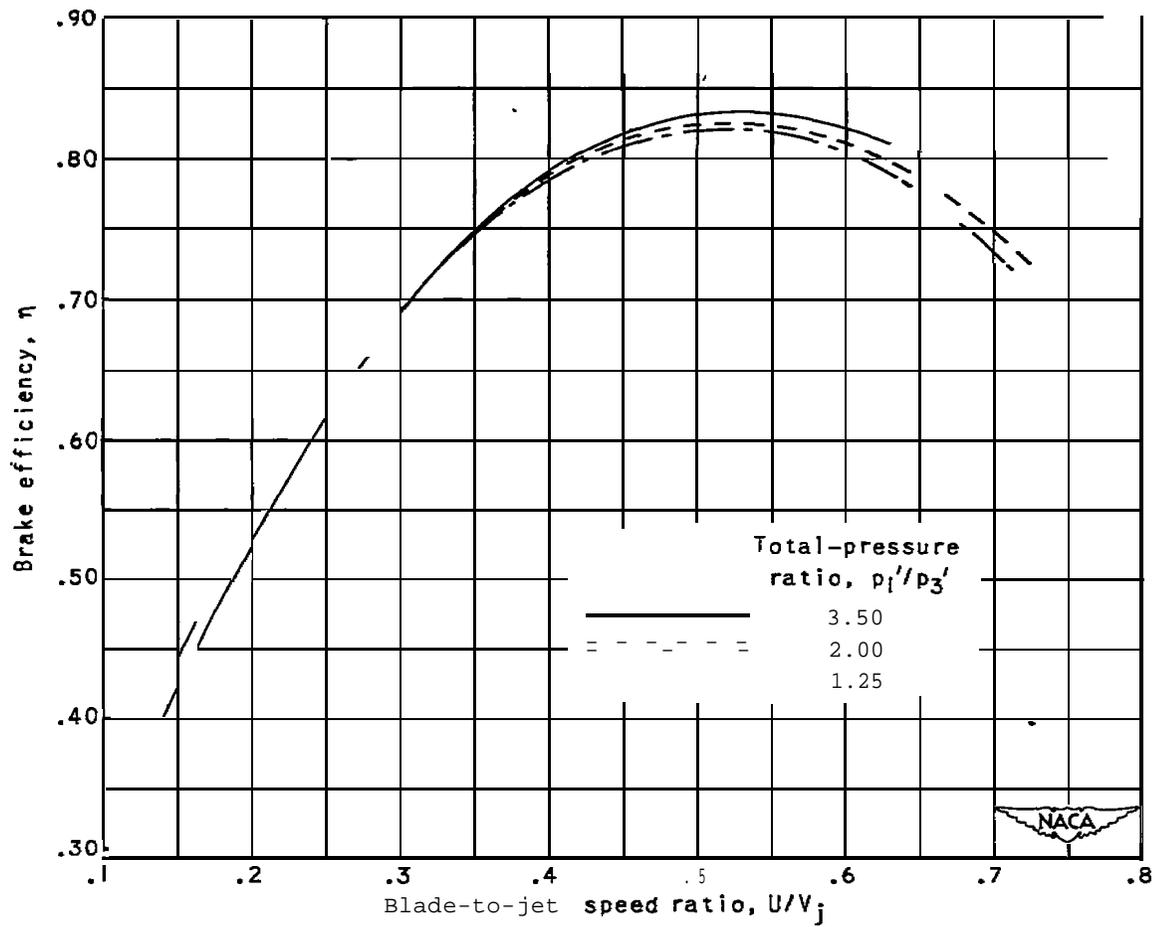


Figure 14. - Efficiency of configuration 2 at various pressure ratios.

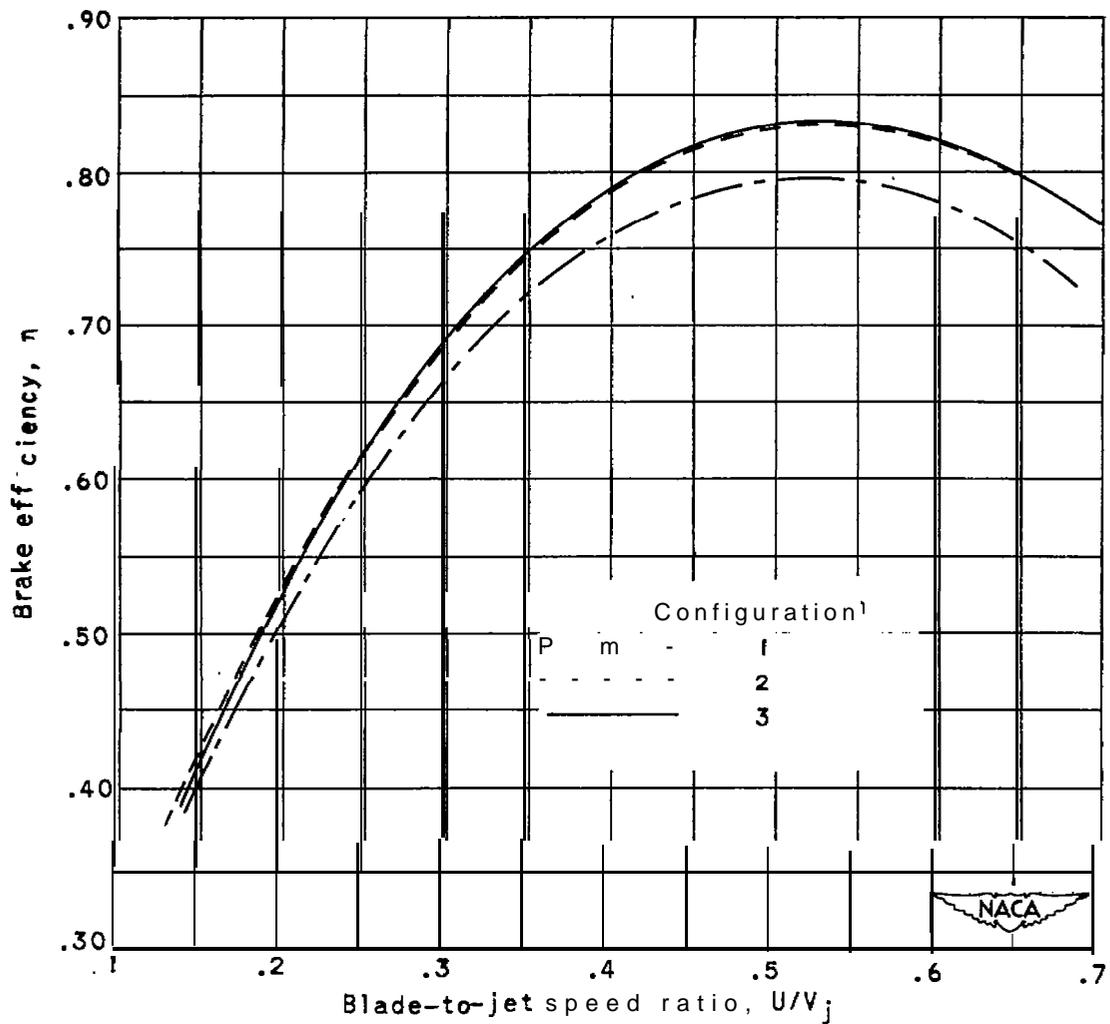


Figure 15.- Comparison of configurations 1, 2, and 3 at a total-pressure ratio of 3.00.

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