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TECHNICAL MEMORANDUM

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PERFORMANCE EVALUATION OF A TWO-STAGE TURBINE DESIGNED FOR A RATIO OF BLADE SPEED TO JET SPEED OF 0.146

By Milton G. Kofskey

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[redacted]

NATIONAL AERONAUTICS AND SPACE ADMINISTRATION

WASHINGTON January 1960

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SUMMARY

The design and experimental investigation of a two-stage turbine with low ratio of blade speed to jet speed, suitable for bleed-type high-energy liquid-rocket applications, are presented. First-stage and two-stage performance are described for operation with the first-stage rotor shrouded and unshrouded. With the exception of the final test, the equivalent weight flow was 15.3 percent greater than design value.

Operation of the first stage with the rotor shrouded gave an efficiency of 0.422 and an equivalent specific work of 13.9 Btu per pound at design ratio of total to static pressure and speed. Design equivalent specific work (16.7 Btu/lb) was obtained at a total- to static-pressure ratio of 5.25 and design speed. Removal of the rotor shroud and reduction of the blade height for rotor tip clearance resulted in an efficiency of 0.397 and equivalent specific work of 12.9 Btu per pound at design pressure ratio and speed. The decrease of 2.5 points in efficiency results from blade tip unloading and the reduction in blade height for tip clearance.

Performance of the two-stage turbine indicated an efficiency of 0.457 and equivalent specific work of 25.8 Btu per pound at design speed and pressure ratio (8.34) for the shrouded-first-stage-rotor configuration. Design equivalent specific work, 33.4 Btu per pound, was not obtained. Removal of the rotor shroud and a reduction in rotor blade height resulted in an efficiency of 0.443 and equivalent specific work of 25.1 Btu per pound at design speed and pressure ratio. When the equivalent weight flow was reduced to within 2.1 percent of the design value by reducing the first-stator throats, the two-stage-turbine efficiency was 0.454, and equivalent specific work was 25.6 Btu per pound for the unshrouded case at design pressure ratio and speed.

Comparison of experimental and theoretical efficiencies based on an experimentally obtained loss coefficient and design velocity diagrams indicated that the experimental efficiencies were 3.8 to 6.3 points lower for all configurations investigated.

*Title, Unclassified. [redacted]

INTRODUCTION

One of the problems associated with the design of pump-drive turbines suitable for bleed-type liquid-rocket applications is the requirement of low flow rate and high work per pound of working fluid. As a result of rotor stress limitations and the desire for minimum flow rate with high specific work, the turbine will operate at a speed-work parameter considerably lower than those used in low-energy-propellant rocket applications. The speed-work parameter λ is defined as the ratio of the square of the mean-section blade speed to the specific work output at design operation $U^2/gJ\Delta h_a$. E-443

Results of an analytical investigation of the effect of speed-work parameter on turbine efficiency indicated that as the speed-work parameter is reduced there is a corresponding decrease in turbine efficiency (refs. 1 to 3). Good correlation of experimental results with theoretical predictions in the high speed-work range has been obtained.

Since there is little published information on the performance of multistage turbines designed in the low range of speed-work parameter, a two-stage axial-flow turbine having a speed-work parameter of 0.072 was designed and experimentally investigated. As a result of the two-stage design efficiency of 0.59 and the speed-work parameter of 0.072, the ratio of blade to jet speed was 0.146. Because of the small blade height and the desire to minimize tip unloading, the first-stage rotor was designed with a blade tip shroud. Cold-air performance results for first-stage and for two-stage operation, with and without first-stage-rotor shrouding, are presented in terms of (1) efficiency based on total-to static-pressure ratio and (2) equivalent specific work output, as functions of pressure ratio, over a range of speed. The performance obtained with a shrouded and an unshrouded first-stage rotor is compared.

The comparison of experimental and theoretical efficiencies made herein should facilitate the selection of design efficiency for turbines operating in the low range of speed-work parameter.

SYMBOLS

c_p specific heat at constant pressure, Btu/(lb)(^{O}R)

D_p pressure-surface diffusion parameter, $1 - (V'_{S,min}/V'_i)$

D_s suction-surface diffusion parameter, $1 - (V'_o/V'_{S,max})$

D_{tot} sum of suction- and pressure-surface diffusion parameters, $D_p + D_s$

g	gravitational constant, 32.17 ft/sec^2
Δh	specific work output, Btu/lb
J	mechanical equivalent of heat, 778.2 ft-lb/Btu
P	total pressure, lb/sq ft
p	static pressure, lb/sq ft
r	radius, ft
T	total temperature, ^{O}R
U	mean-section blade speed, ft/sec
V	gas velocity, ft/sec
V_j	ideal gas velocity corresponding to total- to static-pressure ratio across turbine, $\sqrt{2gJc_pT_0\left[1-\left(\frac{p_4}{P_0}\right)^{(\gamma-1)/\gamma}\right]}$, ft/sec
w	weight-flow rate, lb/sec
γ	ratio of specific heats
δ	ratio of inlet total pressure to NASA standard sea-level pressure of 2116 lb/sq ft
η_{ad}	adiabatic efficiency, based on total- to static-pressure ratio across turbine
θ_{cr}	squared ratio of critical velocity at turbine inlet to critical velocity at NASA standard sea-level temperature of 518.7^{O} R
λ	speed-work parameter, $U^2/gJ\Delta h_a$
σ	solidity, ratio of blade chord to spacing

Subscripts:

a	absolute total state
cr	conditions at Mach number of 1.00
i	blade inlet
max	maximum

[redacted]

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min minimum

o blade outlet

p pressure surface

S blade surface

s suction surface

t tip

z axial component

θ tangential component

0 upstream of turbine

1 between first-stage stator and rotor

2 downstream of first-stage rotor

3 between second-stage stator and rotor

4 downstream of turbine

Superscript:

' relative

TURBINE DESIGN

The design of the two-stage turbine was critical in two respects: The speed-work parameter λ is low, which indicates high change in whirl relative to blade speed; and the hub-tip radius ratio is high, 0.95 and 0.91 in the first and second stages, respectively. These factors result in small blade heights, so that blade tip clearances can become an appreciable percentage of the blade heights. The design parameters listed herein are representative of high-pressure pump-drive turbines for high-energy liquid-rocket applications. The values of blade speed and specific work correspond roughly to those of a hydrogen-reactor rocket with a chamber pressure of 1000 pounds per square inch absolute and a bleed rate of 0.035, or of a chemical hydrogen-oxygen rocket with a chamber pressure of 1000 pounds per square inch absolute and a bleed rate of 0.020.

[redacted]

Design Requirements

The design requirements for the two-stage turbine with 10.2-inch mean blade diameter are as follows:

Overall equivalent specific work, $\Delta h_a/\theta_{cr}$, Btu/lb	33.4
Equivalent weight flow, $w\sqrt{\theta_{cr}}/\delta$, lb/sec	0.484
Equivalent mean blade speed, $U/\sqrt{\theta_{cr}}$, ft/sec	246
Speed-work parameter, λ	0.072

Turbine losses, and therefore turbine efficiency, were computed from boundary-layer characteristics and experimental boundary-layer parameters using the method described in references 4 and 5. These calculations indicated a turbine efficiency of 0.59 based on a ratio of inlet total to exit static pressure of 8.34. The ratio of blade to jet speed corresponding to this pressure ratio is 0.146. The first-stage design efficiency of 0.51 is based on a ratio of inlet total to exit static pressure of 2.94. With this pressure ratio, the first-stage ratio of blade to jet speed is 0.191. Since the turbine was designed with an equal work split, the equivalent specific work for the first stage is 16.7 Btu per pound.

Velocity Diagrams

The design velocity diagrams were calculated at the free-stream stations (fig. 1) for the mean blade radius to meet the design work requirements and are based on the following assumptions:

(1) Two-dimensional flow

(2) Rotor inlet and outlet relative whirl components of equal magnitude

(3) Equal work split

(4) Exit hub-tip radius ratio of 0.91

Free-stream velocity diagrams are shown in figure 2. From this figure, it can be seen that design free-stream turning through both stators is comparatively high, being 80.0° and 145.0° for the first and second stages, respectively. Free-stream design turning through the rotors was also high, being 152.0° and 141.3° in the first and second stages, respectively. Figure 2 also shows that the turbine was designed with approximately 62° exit whirl.

[redacted]

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Stator Design

Stator pressure losses were determined from boundary-layer characteristics as developed in references 4 and 5. The blade profiles laid out for the mean blade radius were analyzed for continuity and surface velocities by using the method of reference 6, with the exception that radial variations were ignored because of the high hub-tip radius ratio.

Since the outlet velocity of the first stator was slightly supersonic ($(V/V_{cr})_1 = 1.095$), slight divergence downstream of the throat, located inside the blade passage, was used. The stator design resulted in a throat dimension of 0.049 inch, 110 blades, and a blade height of 0.286 inch. The stator had a solidity σ of 1.8.

The outlet velocity from the second-stage stator was also slightly supersonic ($(V/V_{cr})_3 = 1.195$), and slight divergence in passage area downstream of the throat, located inside the passage, was used. In order to obtain design axial velocity in the second stage, the annulus area was increased. This was accomplished by increasing the blade height in equal amounts at the hub and tip to maintain the 10.2-inch mean blade diameter. The increase in blade height from the first to the second stage was accomplished within the second-stage stator. The stator design resulted in 90 blades with a blade height of 0.484 inch at the exit. The stator had a solidity of 2.0.

Blade surface and midchannel velocity distributions are presented in figures 3(a) and (b). The total diffusion D_{tot} for each stator was low, 0 and 0.18 for the first and second stages, respectively. Stator blade coordinates for both stages are given in table I.

Rotor Design

Rotor blade profiles determined at the mean radius were designed by the same method used in the stator design. Blade surface and midchannel velocity distributions are presented in figures 3(c) and (d). The total diffusion D_{tot} was low, 0.19 and 0.28 for the first- and second-stage rotors, respectively. The design resulted in 110 blades for each stage, with solidities of 2.0 and 2.1 for the first and second stages, respectively. Coordinates for both rotor blade profiles are given in table II.

[redacted]

APPARATUS

The experimental investigation of the turbine was conducted in the same turbine test facility used in reference 7. The apparatus consisted of the turbine configuration, suitable housing to give uniform turbine-inlet flow conditions, and a cradled dynamometer to absorb turbine power output. A diagrammatic sketch of the turbine test section is shown in figure 4(a). Figures 4(b) to (e) show the four turbine configurations investigated. A photograph of the two-stage rotor is shown in figure 5, and a cutaway sketch of the two-stage-turbine assembly is shown in figure 6. The turbine was driven with dry pressurized air.

Stator blades ground from SAE 4340 steel bar stock were located in slotted rings. The rotor blades were also ground from SAE 4340 steel bar stock and were attached to the rotor disk with an H-type base as shown in figures 4(b) to (e). The first-stage-rotor blades were shrouded with a shrink-fit steel band.

First-stage and two-stage performance tests were made with a shrouded and an unshrouded first-stage rotor (figs. 4(b) to (e)). When the shroud band was removed, the blade tips were ground to obtain a tip clearance of 0.014 inch, and an insert was placed in the casing over the rotor blade to give a continuous outer wall. The tip clearance for the second-stage rotor was 0.018 inch, and an axial clearance of 0.020 inch was used between the stators and rotors.

A labyrinth seal was used over the first-stage shroud to minimize air leakage over the rotor shroud. In addition, a labyrinth-shaft seal was used between the first and second stages to minimize leakage between stages.

INSTRUMENTATION

The actual specific work output was computed from weight flow, torque, and speed measurements. The weight flow was measured with a calibrated ASME flat-plate orifice. The turbine output torque was measured with a commercial self-balancing torque cell and a mercury manometer. Turbine rotative speed was measured with an electronic events-per-unit-time meter.

Turbine-inlet measurements were taken in the annulus upstream of the stator inlet (station 0, fig. 4(a)). Two thermocouple - total-pressure rakes were used for measurement of inlet total pressure and temperature.

Turbine-outlet static pressures were measured in the annulus downstream of the rotor outlet (station 4, fig. 4(a)) from three static-pressure taps spaced 120° apart on each of the inner and outer walls.

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EXPERIMENTAL PROCEDURE

The experimental investigation was conducted with a turbine-inlet temperature of 250° F. Because of bearing thrust limitations, the inlet pressure was limited to a maximum of 57 inches of mercury absolute for two-stage operation. However, with first-stage operation, the rotor thrust was well below the bearing thrust limitations; therefore, the inlet pressure was increased to 101 inches of mercury absolute to increase the turbine power output and thereby improve the accuracy of power measurement.

Performance data were obtained at turbine speeds of 70, 80, 90, 100, and 110 percent of design; and the exit static pressure was varied to give ratios of inlet total to exit static pressure from 2 to approximately 16.

EXPERIMENTAL RESULTS

First-Stage Performance

The performance of the first stage of the two-stage turbine is presented in figure 7. In this figure, equivalent specific work output $\Delta h_a/\theta_{cr}$ and adiabatic efficiency η_{ad} (based on total- to static-pressure ratio) are plotted against ratio of inlet total to exit static pressure for turbine speeds of 70 to 110 percent of design speed.

Figure 7(a) presents the performance with the rotor shrouded. At design pressure ratio (2.94) and design speed, the efficiency was 0.422, and the equivalent specific work was 13.9 Btu per pound. Design equivalent work (16.7 Btu/lb) was not obtained at design pressure ratio and speed, because the losses were higher than assumed in the design. Design equivalent work was obtained at a pressure ratio of 5.25 and an efficiency of 0.350. The significance of the second-stage-choke line will be discussed in a later portion of the report.

The choking equivalent weight flow was 0.558 pound per second, which is 15.3 percent higher than the design value. The stator throats were measured and found to be approximately 8.7 percent larger than the design value, so that 6.6-percent excess flow could not be attributed to excess area. To account for the excess weight flow, the maximum possible equivalent weight flow was determined using measured stator throat areas and assuming a flow coefficient of 1.000. The difference between the calculated theoretical maximum weight flow (0.577 lb/sec) and the experimentally measured weight flow (0.558 lb/sec) indicates that the flow coefficient was approximately 0.967. In the design of the stator, the flow coefficient was estimated to be 0.899 because high blockage was expected owing to the large ratio of wetted perimeter to flow area.

Figure 7(b) presents the overall performance with the rotor shroud removed and the blade height reduced to provide a tip clearance of 0.014 inch. At design pressure ratio and speed, the efficiency was 0.397, and the equivalent specific work was 12.9 Btu per pound. Design specific work could not be obtained at design speed because of increased losses.

Comparison of the performance obtained with the shrouded and unshrouded rotors (figs. 7(a) and (b)) at design pressure ratio and speed shows decreases of 7 percent in equivalent specific work and 2.5 points in efficiency when the rotor shroud was removed and the blade height was reduced to provide for tip clearance. This decrease results from blade tip unloading and from reduced effective blade height.

Two-Stage Performance

Two-stage performance is presented in figure 8, where adiabatic efficiency and equivalent specific work are plotted against total- to static-pressure ratio. With the first-stage rotor shrouded, figure 8(a) shows that the efficiency was 0.457 at design pressure ratio (8.34) and speed. The corresponding equivalent specific work output was 25.8 Btu per pound. Design equivalent specific work of 33.4 Btu per pound could not be obtained, since the second stage choked before design specific work of the first stage could be obtained and also because the losses were higher than assumed in the design.

The second-stage choke line shown in figure 7(a) (first-stage performance) indicates total- to static-pressure ratio when the second stage choked. Equivalent specific work to the right of the line is not obtainable when operated with the second stage. At design speed, the maximum equivalent specific work of the first stage was 13.5 Btu per pound, or 81 percent of design value.

When the first-stage-rotor shroud was removed, figure 8(b) shows that the efficiency was 0.443 at design pressure ratio and speed. The equivalent specific work output was 25.1 Btu per pound. The second-stage-choke line on the equivalent-specific-work curves of figure 7(b) (first-stage performance, unshrouded rotor) shows the maximum total- to static-pressure ratio and work obtainable when the second stage choked. Figure 7(b) shows that, at design speed, first-stage equivalent specific work is limited to a maximum of 12.6 Btu per pound, 75 percent of design value, when operated with the second stage.

Comparison of two-stage performance at design speed and pressure ratio for the shrouded and unshrouded first-stage rotor (figs. 8(a) and (b)) shows decreases of 3 percent in equivalent specific work and 1.4 points in efficiency when the rotor tip shroud is removed. As mentioned previously, the decrease results from blade tip unloading due to tip leakage and from reduced effective blade height.

Comparison of the specific-work split for the shrouded-first-stage-rotor configurations (first stage and two-stage at design speed and total- to static-pressure ratios of 2.75 and 8.34, respectively, figs. 7(a) and 8(a)) shows that the specific-work output of the first stage (13.5) was approximately 9 percent higher than that of the second stage (12.3). This may be attributed to the fact that the second-stage rotor was not shrouded, and hence tip leakage and reduced effective blade height (due to tip clearance) would be reflected in a reduction in specific-work output.

Comparison of the first-stage specific work (12.6) with the two-stage total specific work (25.1) for the unshrouded case (figs. 7(b) and 8(b)) at design speed and two-stage and corresponding first-stage pressure ratios (8.34 and 2.77) indicates that the design condition of equal work split was obtained experimentally. The specific work of the first and second stages was 12.6 and 12.5 Btu per pound, respectively.

In order to determine the effect of excess weight flow on turbine performance, the first-stage-stator throats were reduced to 0.047 inch, which is 4.1 percent smaller than the design value of 0.049 inch. With the reduced stator throats, the equivalent weight flow was 0.494 pound per second, which was 2.1 percent higher than the design value of 0.484 pound per second. At design pressure ratio and speed (fig. 9), the two-stage efficiency was 0.454, and the equivalent specific work output was 25.6 Btu per pound with the first rotor unshrouded. Reduction of weight flow by closing the stator throats therefore resulted in increases of 2 percent in equivalent specific work and 1.1 points in efficiency.

ANALYSIS OF RESULTS

As was mentioned in the section on turbine design, turbine losses and therefore turbine efficiency were computed with boundary-layer characteristics and experimental boundary-layer parameters as described in references 4 and 5. A later report (ref. 3) presents an improved method of estimating design efficiency using the velocity diagrams. Briefly, the level of efficiency obtained with this improved method depends on a loss coefficient that must be obtained experimentally. Using the base loss coefficient of the example turbine of reference 3 and the modified method developed in reference 8, which includes the effect of Reynolds number based on blade height and effective channel velocity, the variation of theoretical efficiency with ratio of blade to jet speed U/V_j was obtained.

The experimental and the theoretical efficiencies are plotted against ratio of blade to jet speed U/V_j in figure 10 for the first-stage performance with the shrouded- and unshrouded-rotor configurations. The blade- to jet-speed ratio was used as a method of generalizing the data

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for correlating theoretical and experimental efficiency with pressure ratio for lines of constant turbine speed. Figure 10(a)(shrouded rotor) indicates that, at design pressure ratio and speed, good agreement is obtained, in that the theoretical efficiency is 0.460, and the experimental efficiency 0.422, a difference of 3.8 points. It will also be noted that the trend of efficiency variation over the range of blade- to jet-speed ratio investigated agrees closely with that of the theoretical efficiency.

When the rotor shroud is removed and the blade height reduced, figure 10(b) shows that the difference in theoretical and experimental efficiencies increases to 6.3 points. As was stated previously, the decrease in experimental efficiency results from blade tip unloading due to tip leakage and from the reduced effective blade height due to required tip clearance.

Theoretical and experimental efficiencies are plotted against blade- to jet-speed ratio for the three two-stage configurations investigated in figure 11. Fair correlation between experimental and theoretical efficiencies is obtained over the range of blade- to jet-speed ratio investigated for the shrouded configuration (fig. 11(a)). At design pressure ratio and speed, the difference between experimental and theoretical efficiencies is 4.2 points. Operation without the first-stage-rotor shroud (fig. 11(b)) resulted in 5.6 points difference between theoretical and experimental efficiencies at design pressure ratio and speed. Figure 11(c) shows that, with the first-stage-rotor shroud removed and the equivalent weight flow reduced to within 2.1 percent of design value, there is a 4.5-point difference between the theoretical and experimental efficiencies at design pressure ratio and speed.

Part of the differences in efficiencies is a result of the mechanical losses, since the theoretical efficiency is based on the aerodynamic performance of the blading, and the experimental efficiency includes the mechanical losses such as bearing and windage losses and also losses due to leakage across the second-stage-stator labyrinth seal during two-stage operation. It will be noted that, for all configurations investigated (first- and two-stage), the trend of experimental efficiency variation over the range of blade- to jet-speed ratio investigated agrees closely with that of the theoretical efficiency.

Results of the experimental investigation for design speed and pressure ratio are presented in table III.

SUMMARY OF RESULTS

Results of the investigation of a two-stage turbine with 10.2-inch mean blade diameter, tested with excess weight flow with the exception of the last test, give basic performance information that should be useful in the design of future turbines operating in the low range of speed-work ratio.

The following results were obtained for the first-stage performance:

1. At design total- to static-pressure ratio (2.94) and speed, the efficiency was 0.422, and the specific equivalent work was 13.9 Btu per pound for the shrouded-rotor configuration. Design equivalent work (16.7 Btu/lb) was obtained at a total- to static-pressure ratio of 5.25 and design speed.

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2. Removal of the rotor shroud and reduction of the rotor blade height resulted in an efficiency of 0.397 and an equivalent specific work of 12.9 Btu per pound. Design equivalent specific work was not obtained because of the increased losses.

3. Comparison of experimental with theoretical efficiencies based on an experimentally determined loss coefficient indicated a lower efficiency than used in the design. At design pressure ratio and speed, the agreement was within 3.8 to 6.3 points in efficiency.

4. Results based on weight-flow calculations using measured stator throat areas indicated that the flow coefficient was 0.967 in spite of the large wetted area.

The following results were obtained for two-stage performance:

1. At design total- to static-pressure ratio (8.34) and speed, the efficiency was 0.457, and the equivalent specific work was 25.8 Btu per pound for the shrouded-first-stage-rotor configuration. Design equivalent specific work (33.4 Btu/lb) was not obtained, because the experimental losses were higher than the losses assumed in the design and the second stage choked before design specific work of the first stage could be obtained.

2. Removal of the rotor shroud and reduction of the rotor blade height for tip clearance resulted in an efficiency of 0.443 and an equivalent specific work of 25.1 Btu per pound at design speed and total- to static-pressure ratio. This corresponds to a drop of 3 percent in equivalent specific work and 1.4 points in efficiency compared with the shrouded configuration.

3. Reduction of the equivalent weight flow to within 2.1 percent of design value resulted in an efficiency of 0.454 and equivalent specific work of 25.6 Btu per pound at design total- to static-pressure ratio and speed for the unshrouded-first-stage-rotor configuration. This represents an increase of 2 percent in equivalent specific work and 1.1 points in efficiency compared with the higher-weight-flow case.

4. Comparison of theoretical and experimental efficiencies indicated that the experimental efficiencies were 4.2 to 5.6 points lower than the theoretical efficiencies for the three configurations investigated.

5. At design speed and pressure ratio, the first and second stages produced equal work when operated with the first-stage rotor unshrouded. For the shrouded-first-stage-rotor configuration, specific work of the first stage was 9 percent higher than that of the second stage.

Lewis Research Center
National Aeronautics and Space Administration
Cleveland, Ohio, September 23, 1959

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[redacted]

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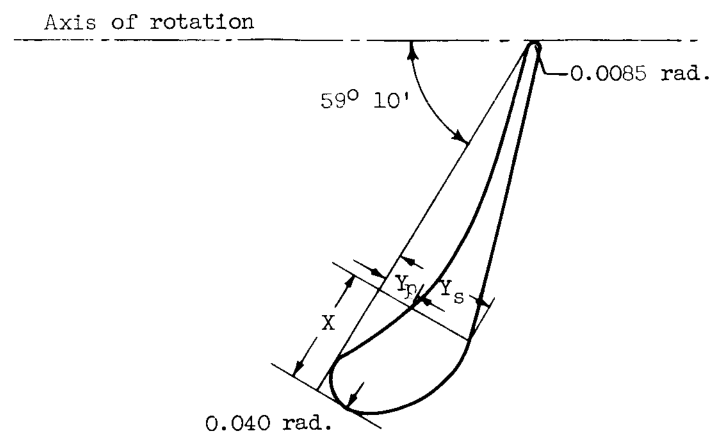
TABLE I. - STATOR BLADE COORDINATES FOR HUB, MEAN, AND TIP SECTIONS

(a) First stage



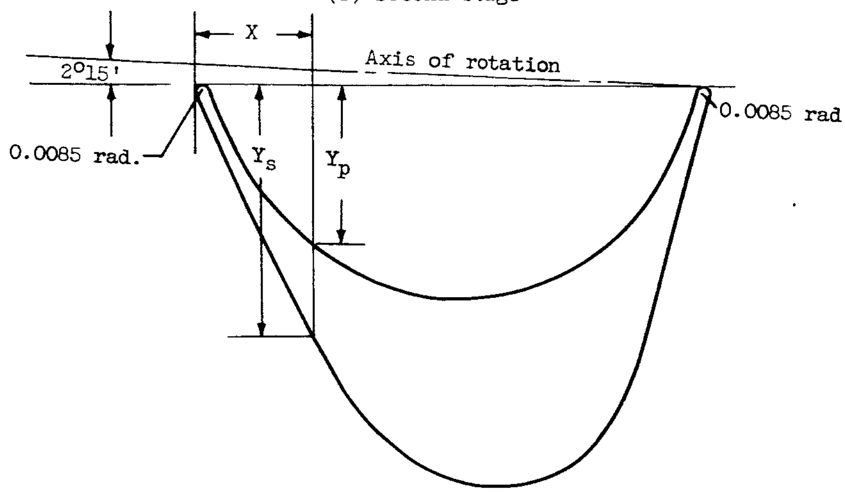
r/r_t		
Hub	Mean	Tip
0.945	0.972	1.000
X, in.	Y_s, in.	Y_p, in.
0.000	0.0400	0.0400
.0250	.0912	.0030
.0400	.1052	.0000
.0500	.1125	.0015
.0750	.1241	.0136
.1000	.1295	.0258
.1250	.1305	.0363
.1500	.1276	.0444
.1750	.1224	.0503
.2000	.1153	.0541
.2250	.1075	.0556
.2500		.0553
.2750		.0536
.3000	Straight	.0507
.3250	line	.0466
.3500		.0418
.3750		.0363
.4000		.0303
.4250		.0240
.4500		.0174
.4750		.0106
.5000		.0038
.5147	.0170	.0000
.5232	.0085	.0085

[redacted]

TABLE I. - Concluded. STATOR BLADE COORDINATES

FOR HUB, MEAN, AND TIP SECTIONS

(b) Second stage



X, in.	Y_s, in.	Y_p, in.
0.0000	0.0085	0.0085
.0085		.0000
.0250	Straight	.0237
.0500	line	.0777
.0750		.1206
.1000	.2229	.1552
.1250	.2741	.1835
.1500	.3242	.2073
.1750	.3705	.2273
.2000	.4110	.2444
.2250	.4446	.2585
.2500	.4717	.2697
.2750	.4939	.2777
.3000	.5110	.2832
.3250	.5242	.2861
.3500	.5330	.2871
.3750	.5391	.2860
.4000	.5420	.2834
.4250	.5417	.2791
.4500	.5380	.2725
.4750	.5306	.2636
.5000	.5186	.2522
.5250	.5006	.2384
.5500	.4753	.2211
.5750	.4375	.2000
.6000	.3718	.1733
.6250	.2841	.1390
.6500	Straight	.0920
.6750	line	.0307
.6917		.0000
.7002	.0085	.0085

[redacted]

[redacted]

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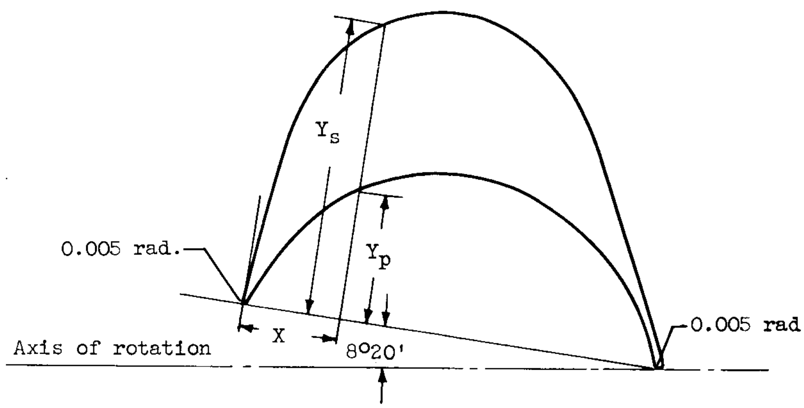
TABLE II. - ROTOR BLADE COORDINATES FOR HUB, MEAN, AND TIP SECTIONS

(a) First stage



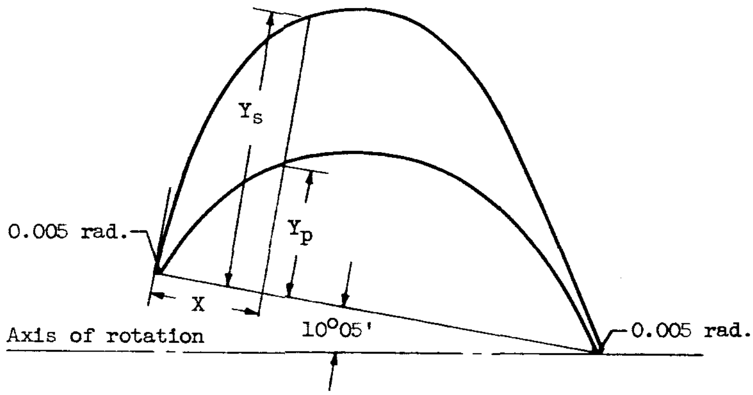
r/r_t		
Hub	Mean	Tip
0.945	0.972	1.000
X, in.	Y_s, in.	Y_p, in.
0.0000	0.0050	0.0050
.0100	.0665	.0000
.0250	.2444	.0465
.0500	.3180	.0944
.0750	.3577	.1276
.1000	.3844	.1531
.1250	.4037	.1727
.1500	.4176	.1882
.1750	.4276	.2003
.2000	.4338	.2095
.2250	.4368	.2158
.2500	.4362	.2194
.2750	.4321	.2209
.3000	.4246	.2196
.3250	.4137	.2161
.3500	.3990	.2105
.3750	.3796	.2022
.4000	.3530	.1910
.4250	.3162	.1769
.4500	.2685	.1596
.4750	Straight	.1378
.5000	line	.1114
.5250		.0782
.5500		.0357
.5715		.0000
.5765	.0050	.0050

[redacted]

TABLE II. - Concluded. ROTOR BLADE COORDINATES FOR HUB,

MEAN, AND TIP SECTIONS

(b) Second stage



r/r_t		
Hub	Mean	Tip
0.908	0.955	1.000
X, in.	Y_s, in.	Y_p, in.
0.0000	0.0050	0.0050
.0100	Straight line	.0000
.0193	.1436	----
.0250	.1774	.0406
.0500	.2650	.0846
.0750	.3130	.1166
.1000	.3453	.1418
.1250	.3666	.1616
.1500	.3819	.1775
.1750	.3923	.1901
.2000	.3990	.1997
.2250	.4022	.2067
.2500	.4020	.2115
.2750	.3981	.2138
.3000	.3914	.2141
.3250	.3809	.2122
.3500	.3666	.2077
.3750	.3481	.2012
.4000	.3248	.1922
.4250	.2969	.1811
.4500	.2622	.1668
.4750	.2243	.1506
.5000	.1860	.1303
.5250	Straight line	.1066
.5500		.0790
.5750		.0481
.6000		.0141
.6128		.0000
.6178	.0050	.0050

[redacted]

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TABLE III. - DESIGN REQUIREMENTS AND EXPERIMENTAL RESULTS FOR TWO-STAGE

TURBINE AT DESIGN SPEED AND PRESSURE RATIO

[Design equivalent weight flow, 0.484 lb/sec.]

Configuration	Experimental equivalent weight flow, lb/sec	Design total- to static-pressure ratio	Design equivalent specific work, Btu/lb	Experimental equivalent specific work, Btu/lb	Design efficiency	Experimental efficiency	Theoretical efficiency
Single-stage, shrouded rotor	0.558	2.94	16.7	13.9	0.51	0.422	0.460
Single-stage, rotor shroud removed	.558	2.94	16.7	12.9	.51	.397	.460
Two-stage, first-stage rotor shrouded	.550	8.34	33.4	25.8	.59	.457	.499
Two-stage, first-stage-rotor shroud removed	.549	8.34	33.4	25.1	.59	.443	.499
Two-stage, first-stage-rotor shroud removed and first-stage-stator throat reduced	.494	8.34	33.4	25.6	.59	.454	.499

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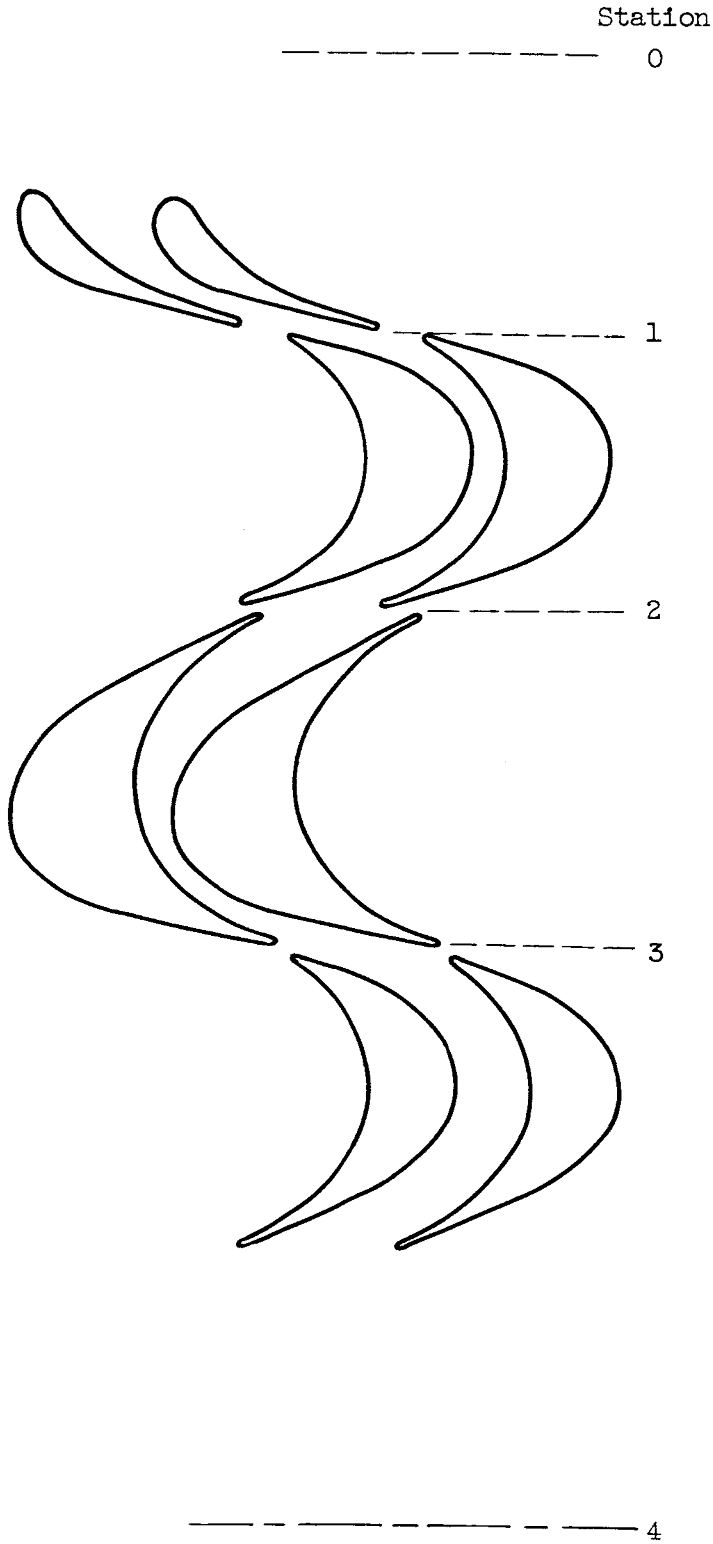


Figure 1. - Stator and rotor blade profiles.

[redacted]

[redacted]

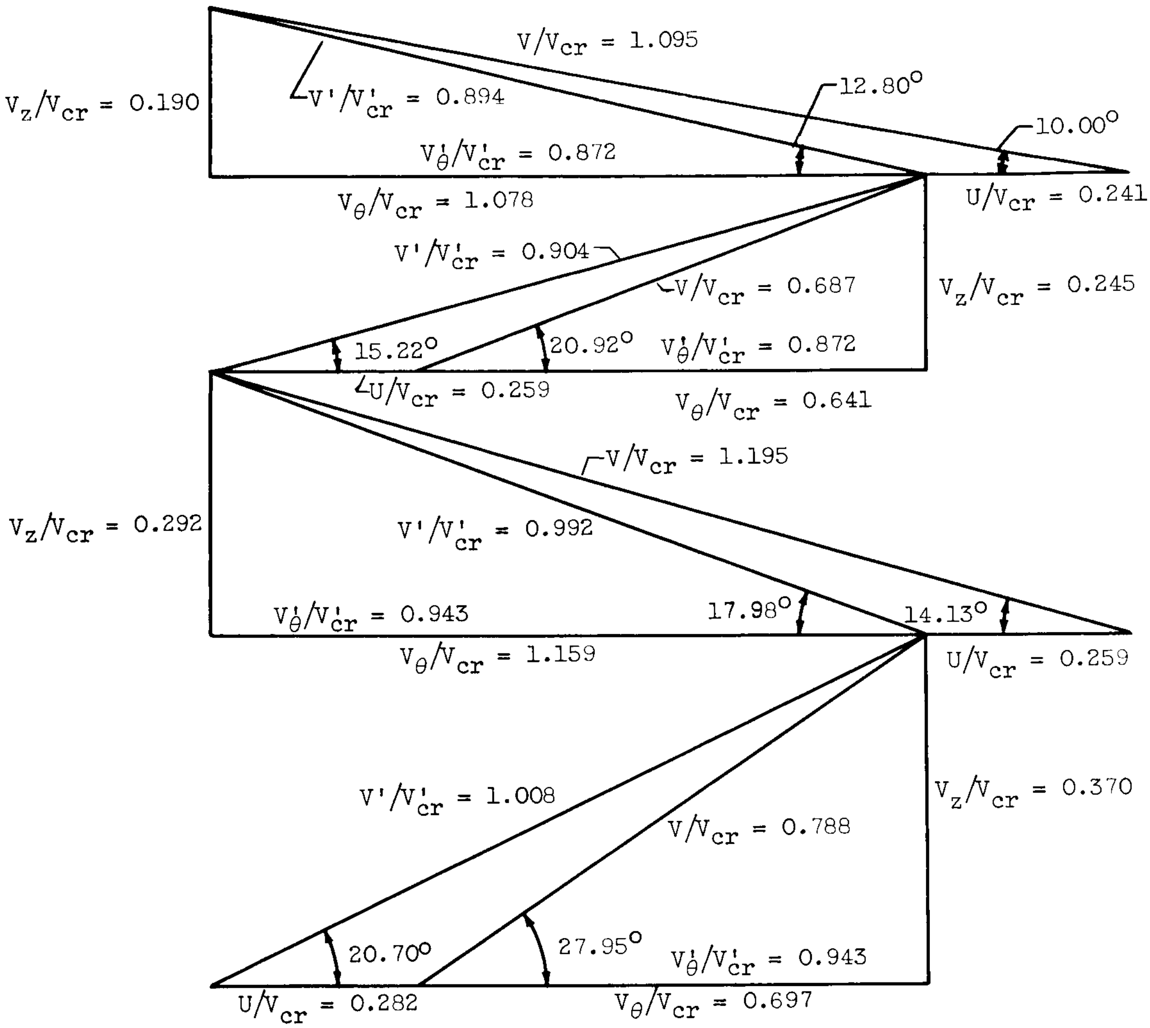
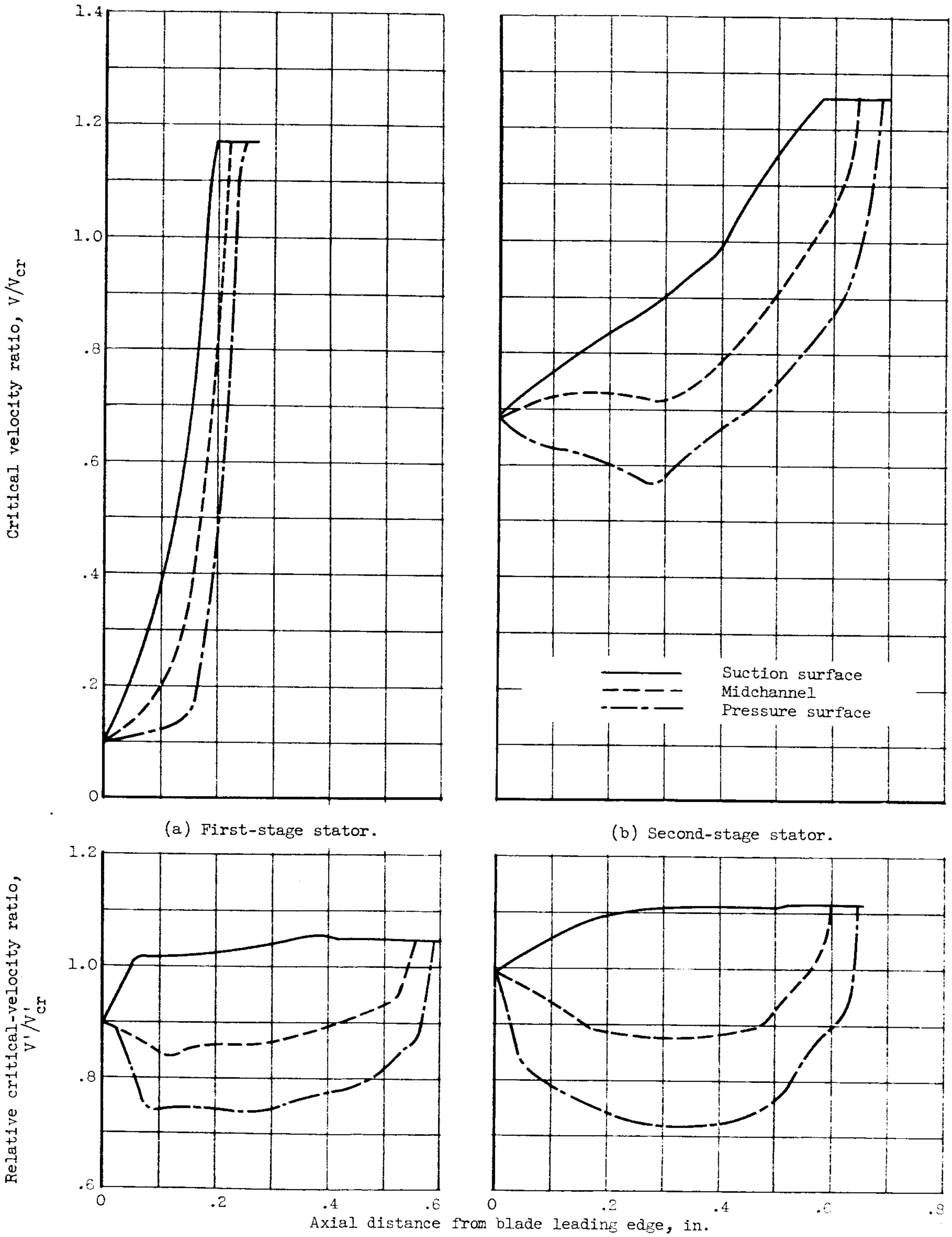


Figure 2. - Design free-stream velocity diagrams for two-stage turbine with 10.2-inch mean diameter.

[redacted]

E-443



(a) First-stage stator.

(b) Second-stage stator.

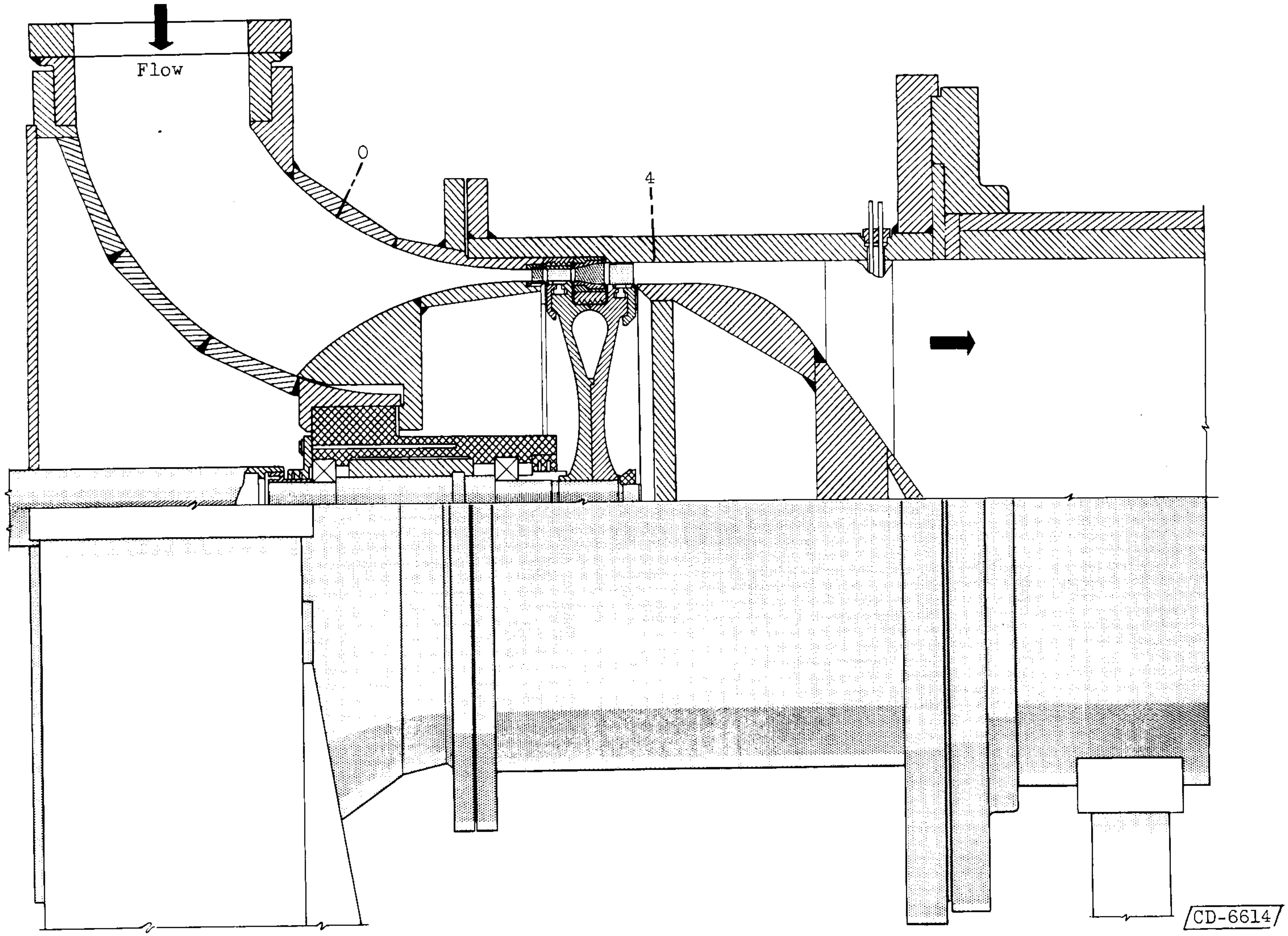
(c) First-stage rotor.

(d) Second-stage rotor.

Figure 3. - Design midchannel and surface velocity distributions.

[redacted]

[redacted]



(a) Overall view.

Figure 4. - Diagrammatic sketch of turbine test section.

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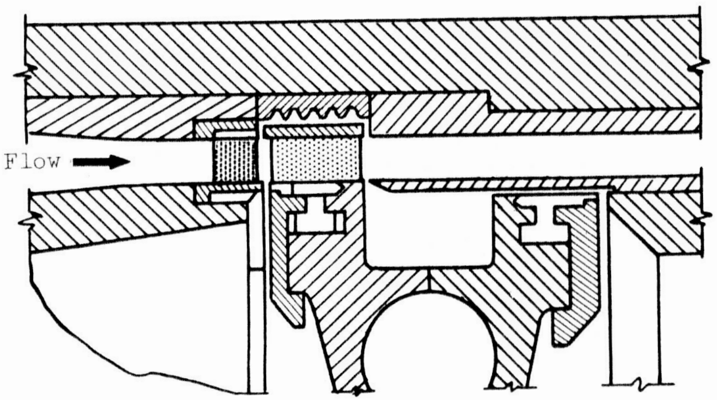
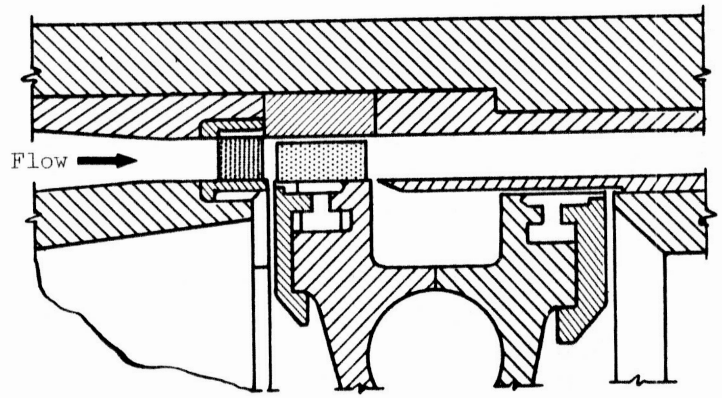
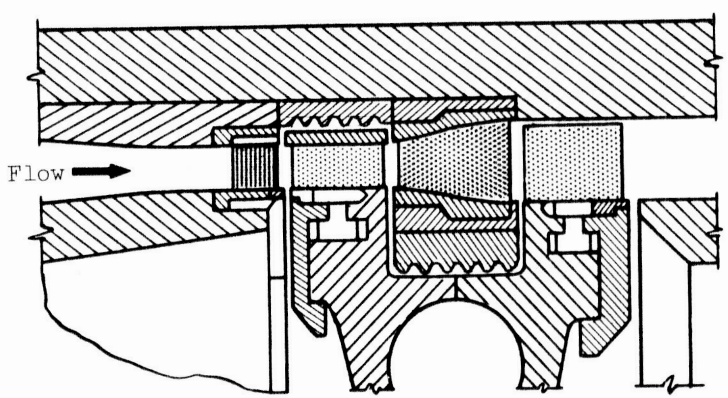


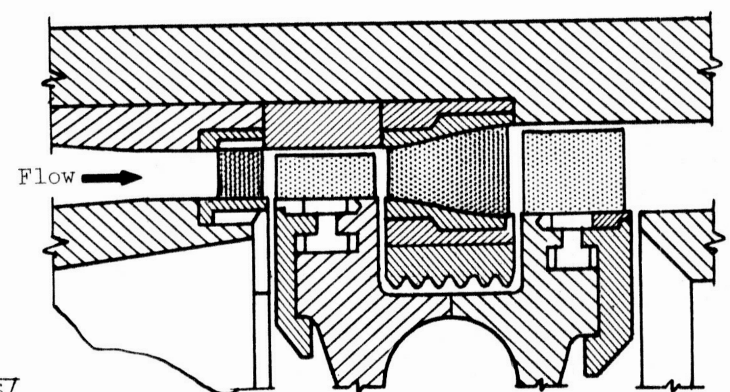
(b) First stage, shrouded rotor.



(c) First stage, rotor shroud removed.



(d) Two stages, first-stage rotor shrouded.



(e) Two stages, first-stage-rotor shroud removed.

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Figure 4. - Concluded. Diagrammatic sketch of turbine test section.

[redacted]

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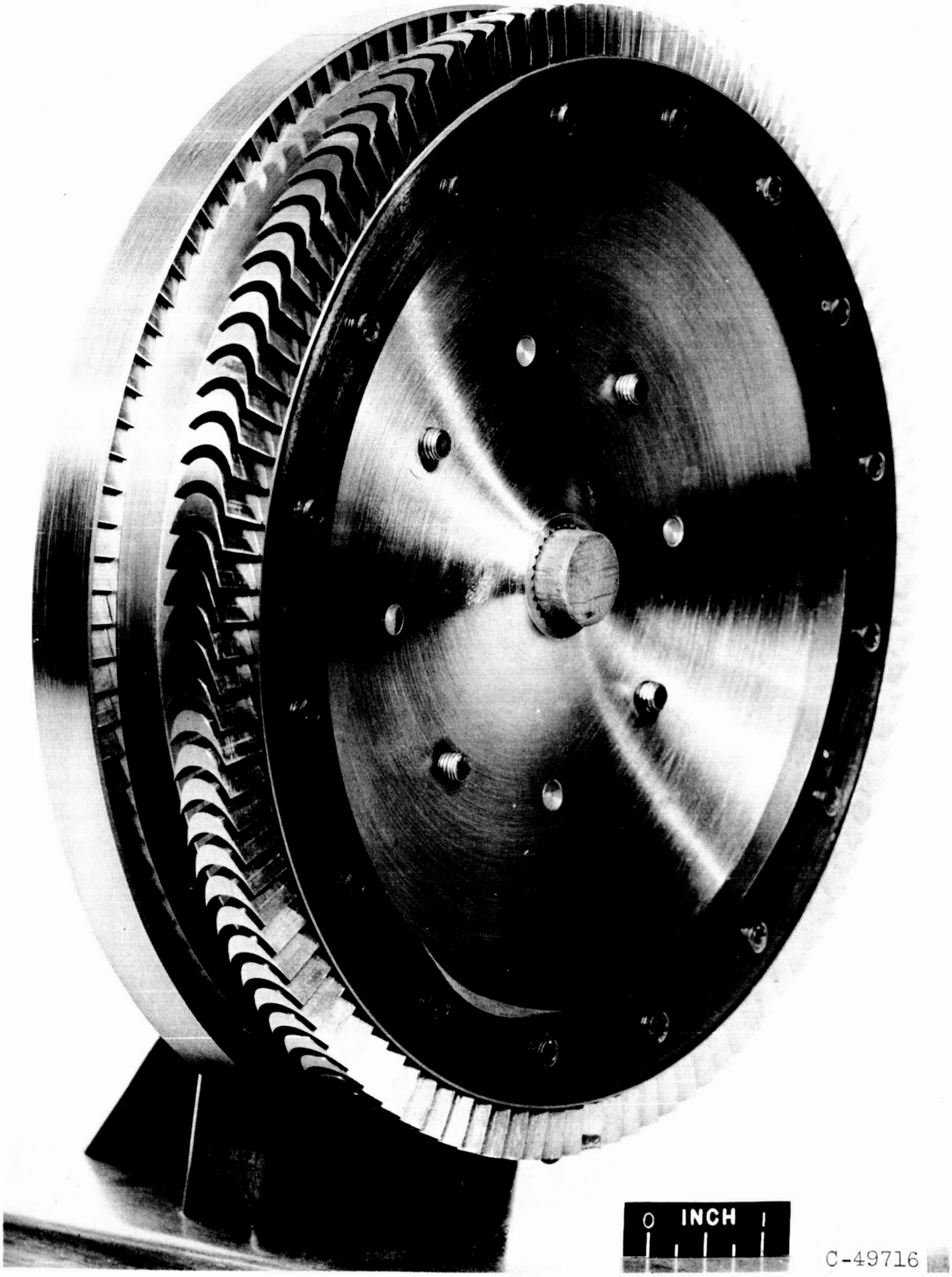

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Figure 5. - Two-stage-turbine rotor with first stage shrouded.

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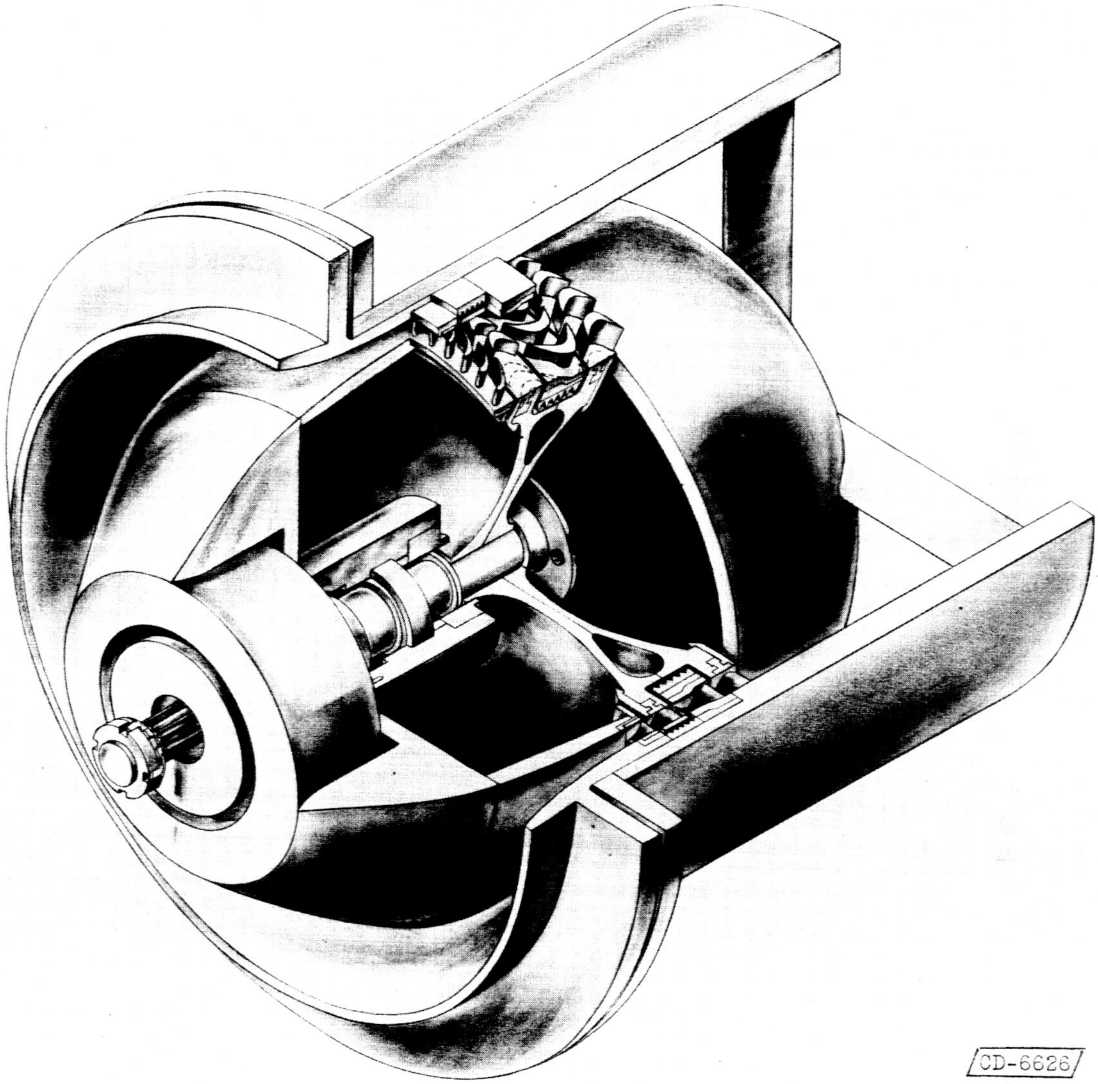

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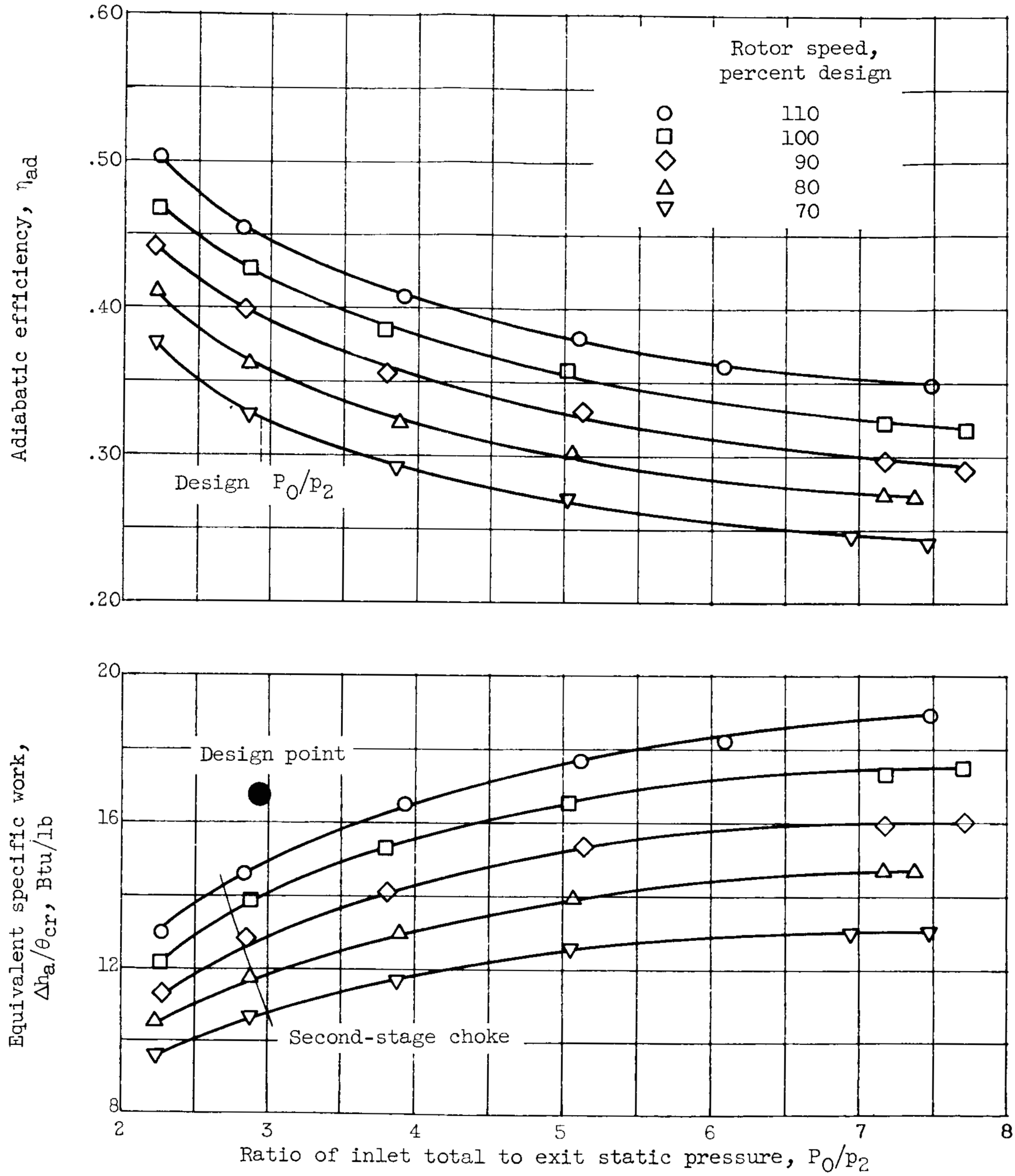
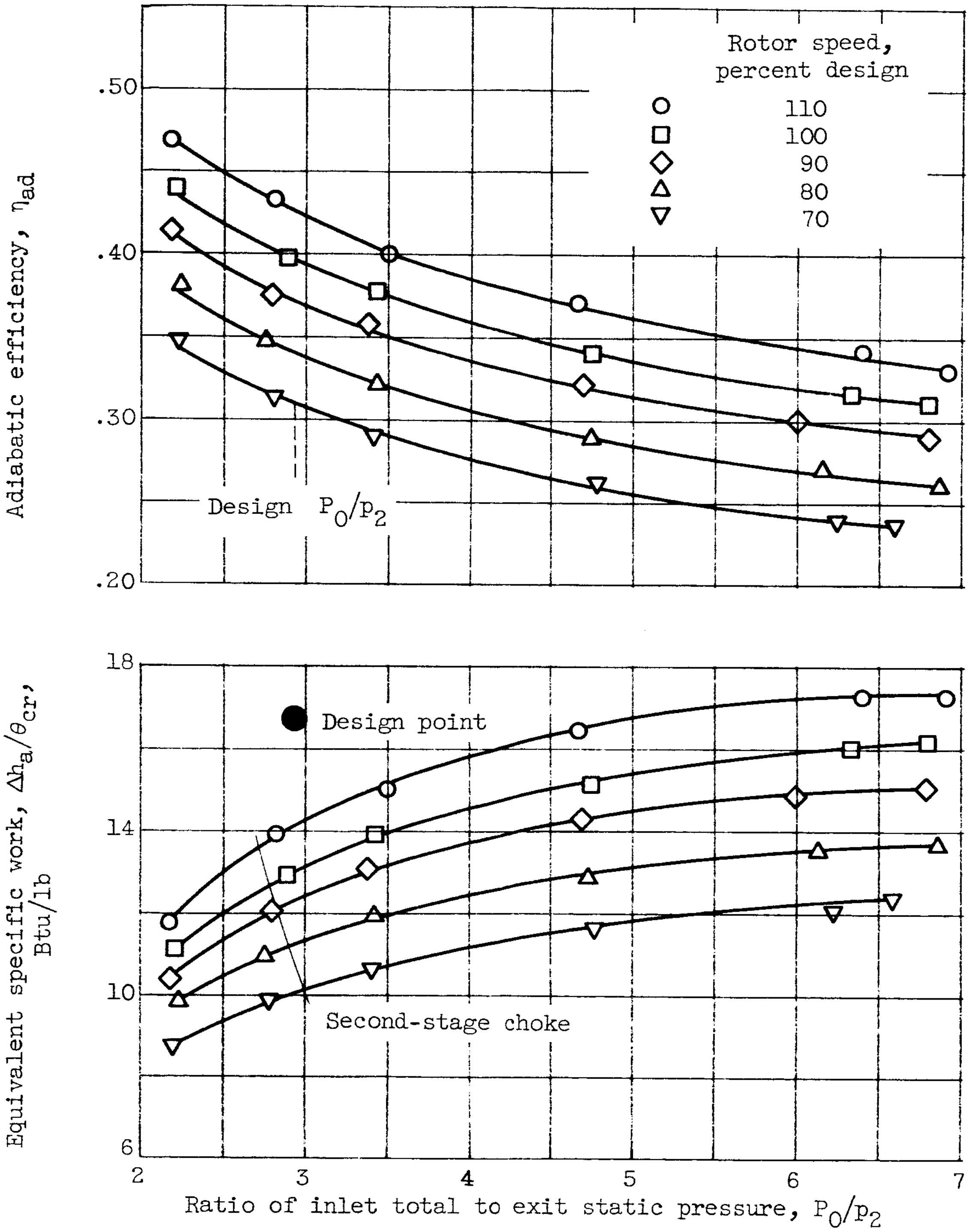
Figure 6. - Cutaway view of two-stage turbine.



(a) Shrouded rotor.

Figure 7. - Experimental first-stage performance of two-stage turbine with 10.2-inch mean diameter.



(b) Rotor shroud removed.

Figure 7. - Concluded. Experimental first-stage performance of two-stage turbine with 10.2-inch mean diameter.

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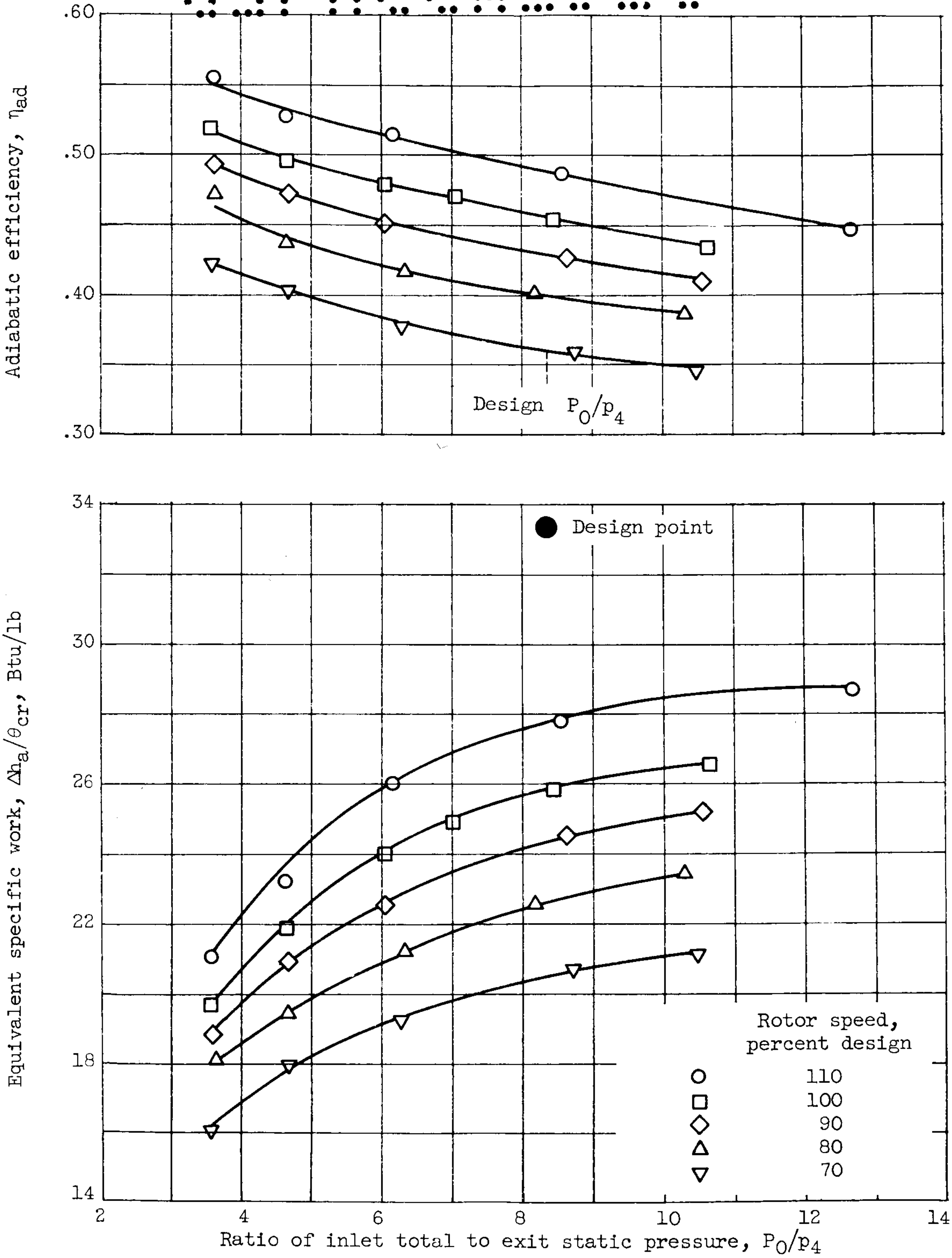
[redacted]



(a) Shrouded first-stage rotor.

Figure 8. - Experimental two-stage performance of two-stage turbine with 10.2-inch mean diameter.

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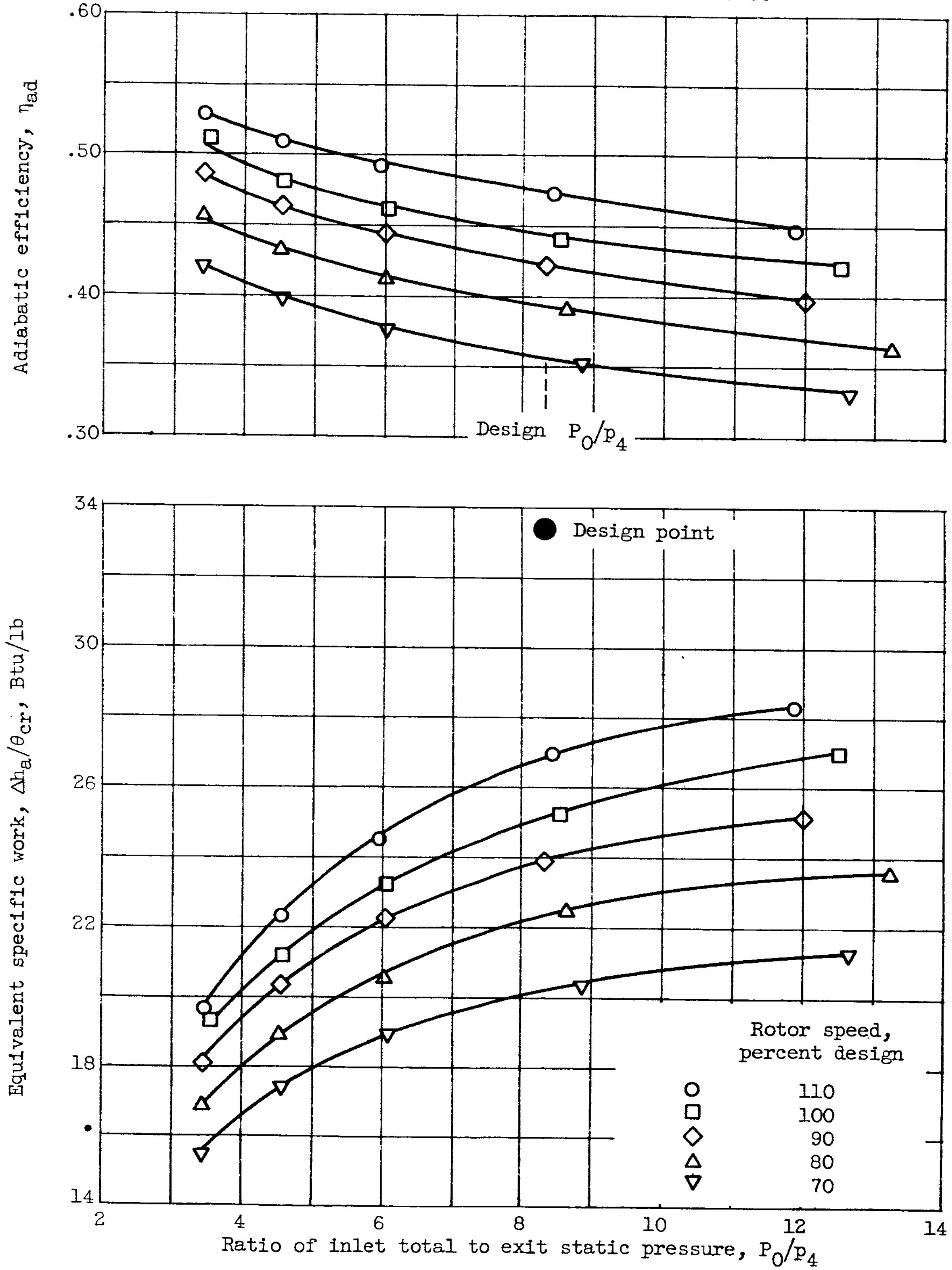

(b) Unshrouded first-stage rotor.

Figure 8. - Concluded. Experimental two-stage performance of two-stage turbine with 10.2-inch mean diameter.

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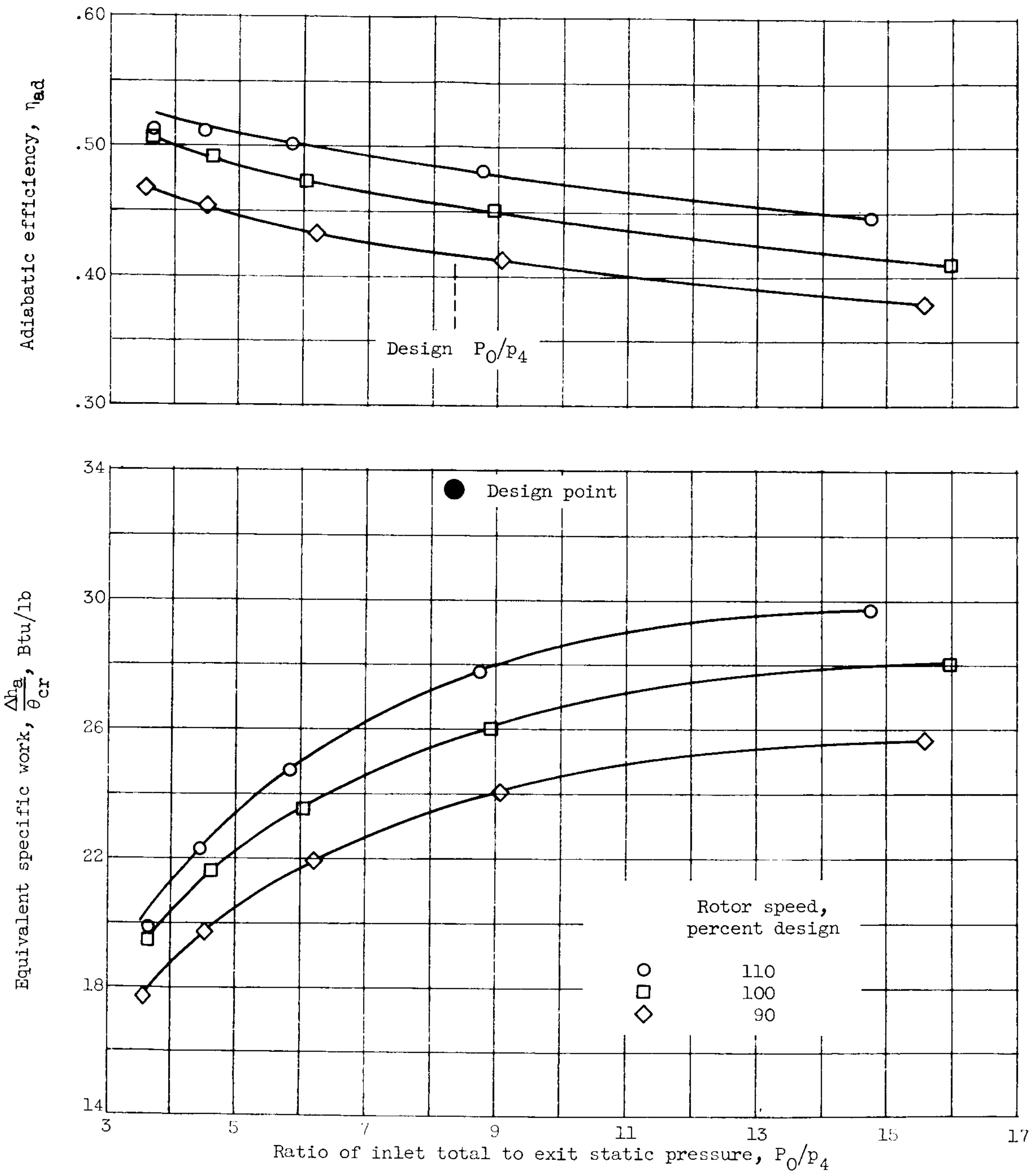

Figure 9. - Experimental two-stage performance of two-stage turbine with 10.2-inch mean diameter (reduced weight flow, unshrouded first-stage rotor).

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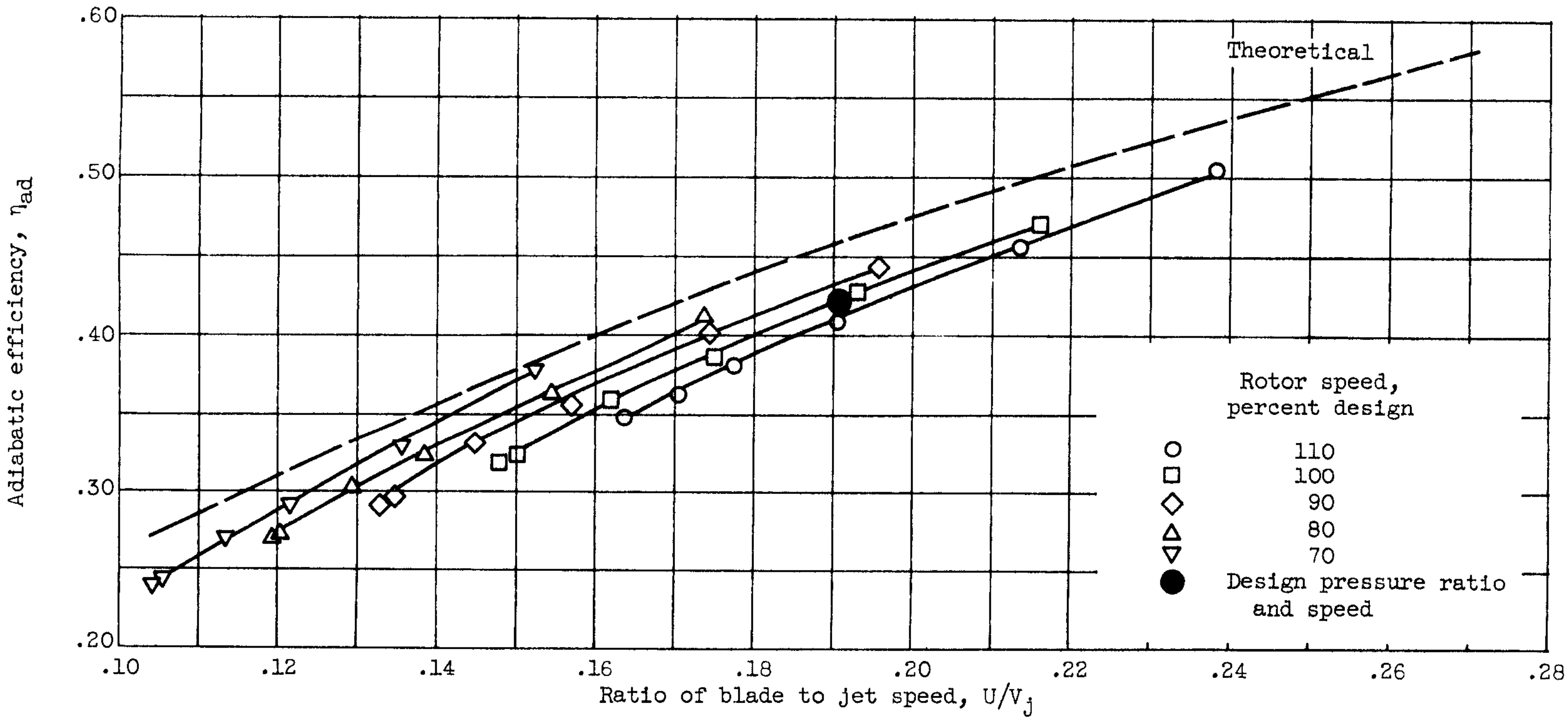
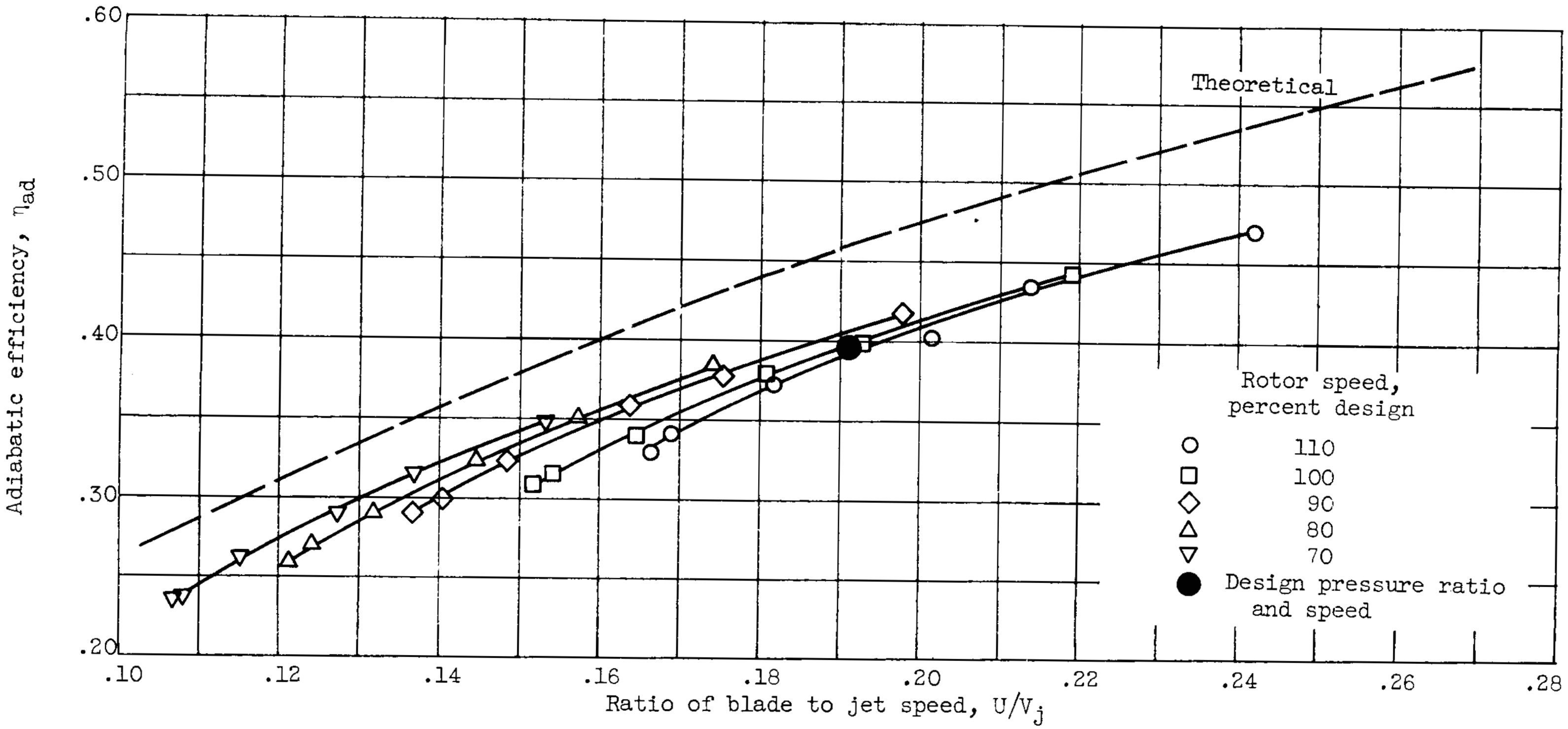


(a) First-stage rotor shrouded.

Figure 10. - Variation of theoretical and experimental efficiencies with blade- to jet-speed ratio for first-stage operation.

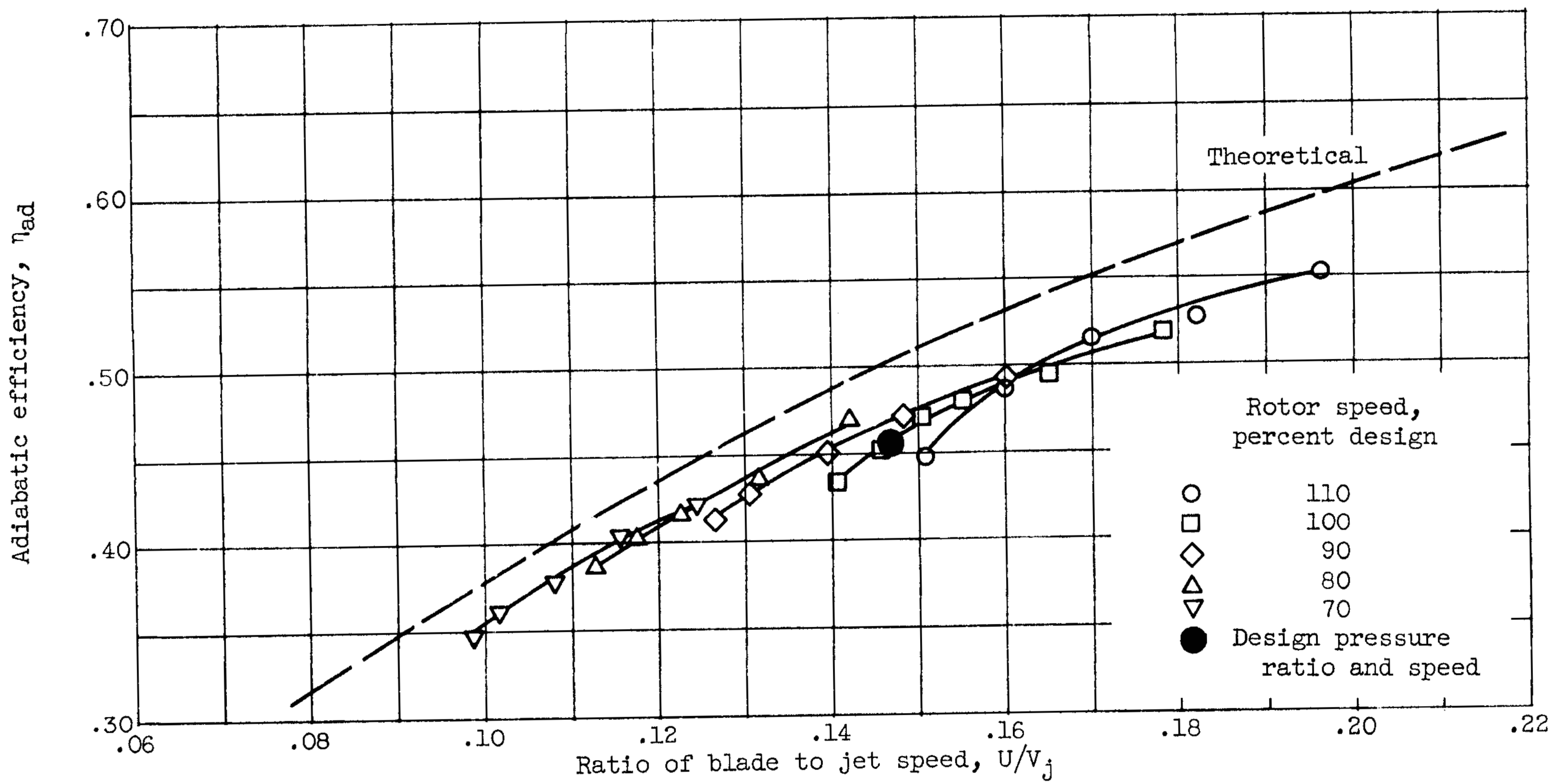
[redacted]



(b) First-stage rotor unshrouded.

Figure 10. - Concluded. Variation of theoretical and experimental efficiencies with blade- to jet-speed ratio for first-stage operation.

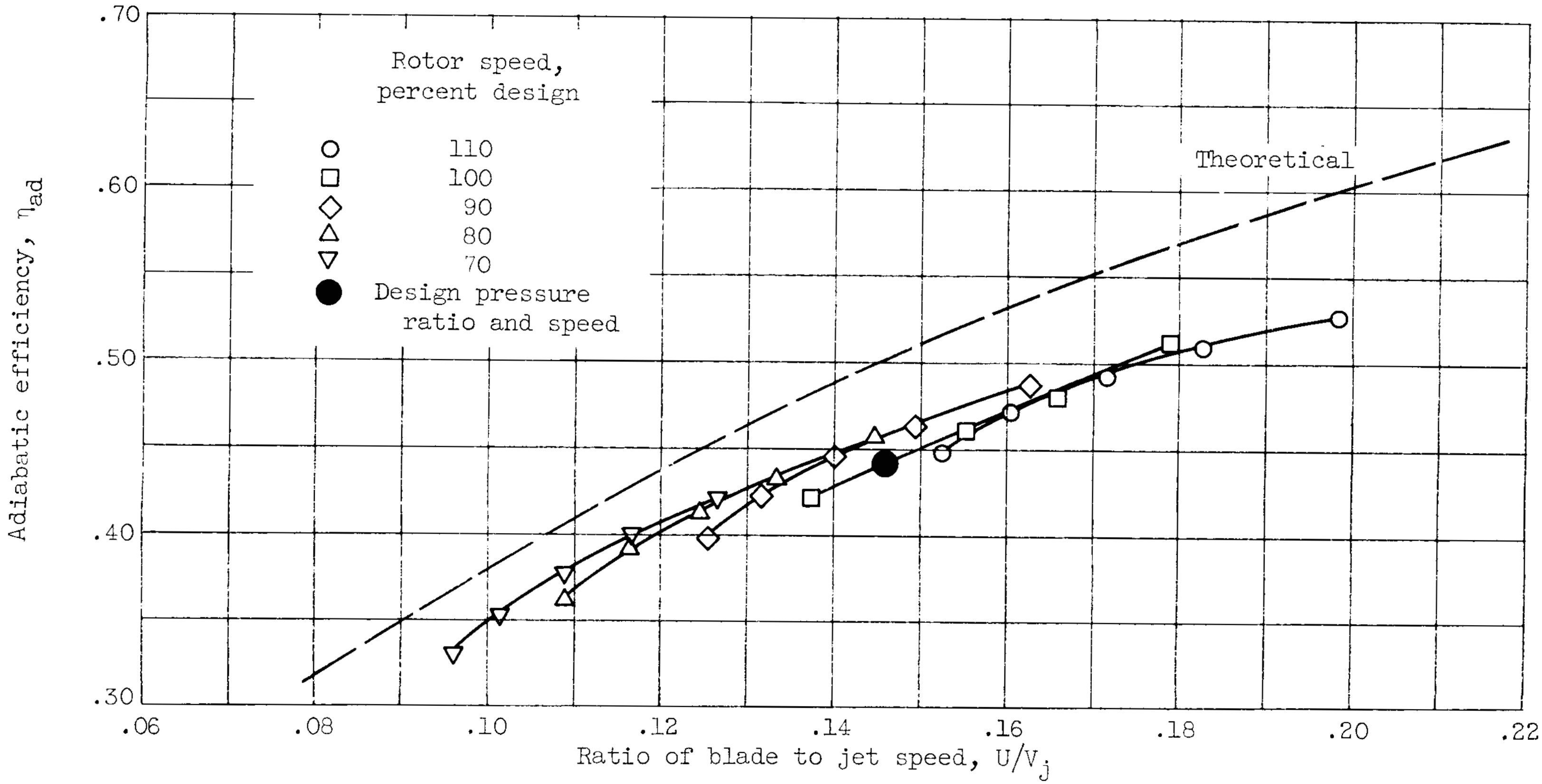
[redacted]



(a) Shrouded first-stage rotor.

Figure 11. - Variation of theoretical and experimental efficiencies with blade- to jet-speed ratio for two-stage operation.

[redacted]



(b) Unshrouded first-stage rotor.

Figure 11. - Continued. Variation of theoretical and experimental efficiencies with blade- to jet-speed ratio for two-stage operation.

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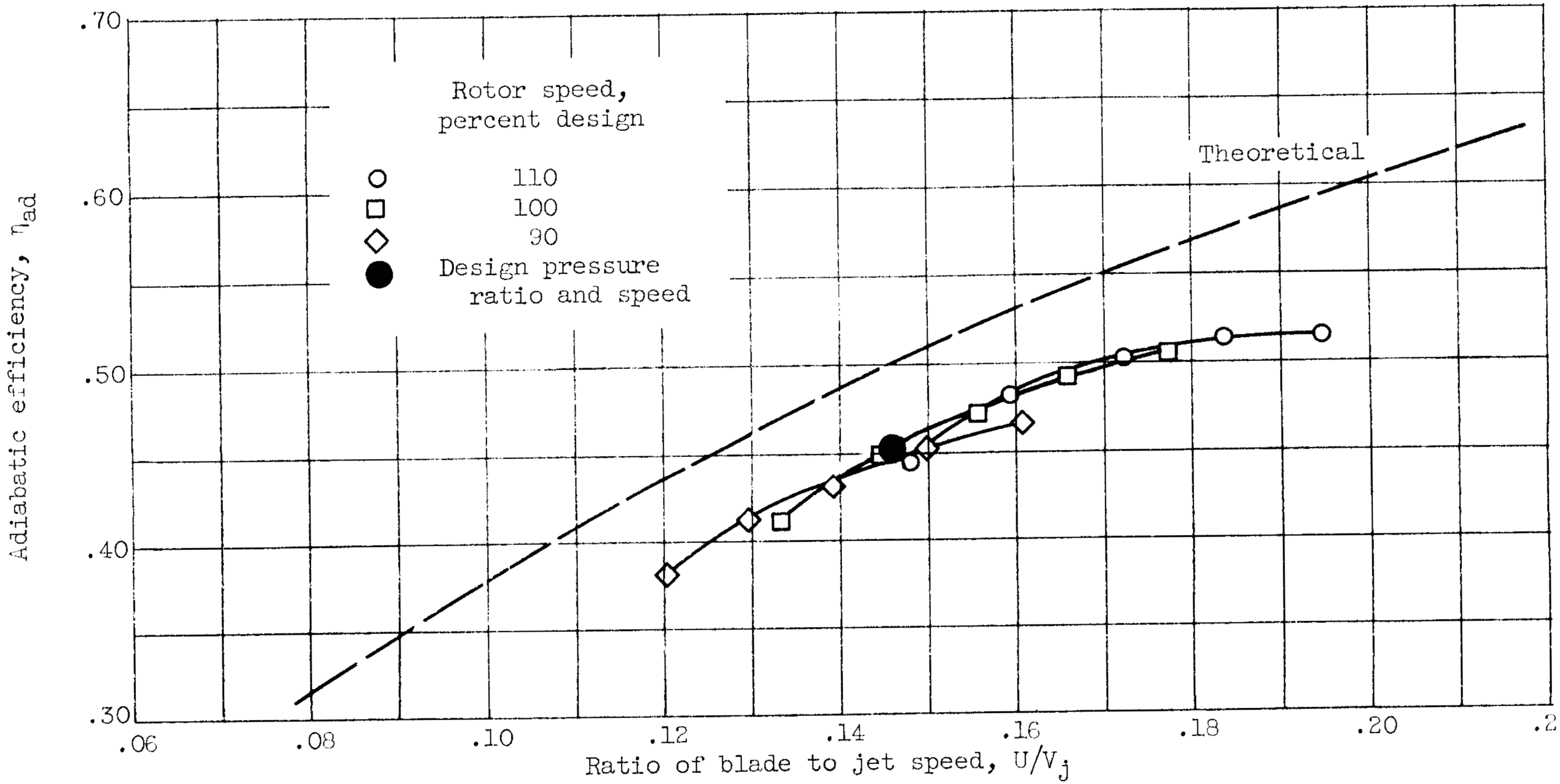

(c) Unshrouded first-stage rotor with reduced equivalent weight flow.

Figure 11. - Concluded. Variation of theoretical and experimental efficiencies with blade- to jet-speed ratio for two-stage operation.

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