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COLD-AIR INVESTIGATION OF A TURBINE FOR HIGH-TEMPERATURE-ENGINE APPLICATION

V - Two-Stage Turbine Performance as Affected by Variable Stator Area

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16. Abstract

The stator areas of the design two-stage turbine were both decreased and increased by nominally 30 percent, and the performances of the two turbines are compared with that of the design stator area turbine. Turbine efficiency decreased with stator area changes. Closing the stator area resulted in the more severe efficiency loss. The decrease in efficiency for both turbines is attributable to rotor incidence, off-design blade-surface velocities, and adverse reaction changes across the blade rows.

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COLD-AIR INVESTIGATION OF A TURBINE FOR HIGH-TEMPERATURE-ENGINE APPLICATION

V - TWO-STAGE TURBINE PERFORMANCE AS AFFECTED BY VARIABLE STATOR AREA

by Frank P. Behning, Harold J. Schum, and Edward M. Szanca

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SUMMARY

The changes in turbine performance resulting from adjusting stator flow area of an experimental 0.762-meter (30-in.) two-stage turbine has been investigated. The performance of the turbine having design area stators has been previously reported. This report presents the performance of the same turbine when equipped with stators having areas both decreased and increased nominally by 30 percent of design. The area changes were effected by reorienting the design blades to that stagger angle required to provide the desired flow area. The same two-stage rotor was used for all three experimental programs.

Comparing the results of all three turbine builds indicated that changing the stator area from that of design decreased the turbine efficiency. Closing the stator area resulted in the more severe efficiency loss. The losses were attributed to rotor incidence, off-design blade-surface velocities, and adverse reaction changes across the blade rows.

INTRODUCTION

Engines for such advanced applications as the supersonic transport and multimission military aircraft must operate over a wide range of flight Mach number and altitude. To accommodate these varying conditions and to maintain the best engine performance, some degree of flexibility is desired within the engine. One way of providing this flexibility is to incorporate variable stators within the turbine component. This results in control of the turbine flow capacity, which in turn can be used to adjust to a more favorable engine operating point.

The potential benefits of using variable stator turbines must not be nullified by an associated reduction in turbine efficiency. Thus, it is essential not only to understand the effect of variable turbine geometry on efficiency, but also to have sufficient knowledge of the factors affecting this efficiency so that such performance penalties can be minimized.

In view of these considerations an experimental study of the performance of a two-stage turbine with variable stator flow area in each stage was conducted at Lewis. The turbine tested is representative of a two-stage turbine suitable for a high-temperature turbojet engine in a supersonic aircraft. The design and test results of the study of the first stage of this turbine, when tested as a separate component, are presented in references 1 to 9 and are summarized in reference 10. It was determined that both opening and closing the stator area by 30 percent resulted in lower turbine efficiencies than that obtained using the design stator setting. The peak efficiency at design speed decreased from 0.923 at design stator setting area to 0.91 and 0.87 for the opened and closed stator-area settings, respectively. The much larger penalty for the closed setting resulted from overexpansion of the flow within the stator with resultant high velocities entering the rotor, flow deceleration (static pressure rise) across the rotor hub, and probable flow separation from the rotor hub surface.

The purpose of this report is to present results of the tests of a two-stage turbine equipped with variable-stator areas. Whereas the first stage of the turbine had constant hub and tip diameters of 55.9 and 76.2 centimeters (22 and 30 in.), the flow area increased through the second stage such that the exit hub and tip diameters were 51.2 and 80.9 centimeters (20.15 and 31.85 in.). The design of the second stage of the turbine and the performance of the as-designed two-stage turbine is presented in reference 11. This turbine yielded an efficiency of 0.932 at equivalent design speed and work output.

For the experimental investigation reported herein, new stator blade rows were fabricated using the design turbine profile. But they were reoriented by stagger-angle changes to provide flow areas nominally 30-percent less than or greater than the design stator flow areas. Both first-stage and second-stage stator areas were varied. In fact, the same first-stage stators as those used in the single-stage investigations were used for the subject two-stage tests. Both the closed- and the opened-stator area turbines were tested using the same two-stage rotor assembly as that used for the design area turbine investigation (ref. 11).

Tests were made over a range of speed and pressure ratio with turbine inlet pressure and temperature maintained at 10.159 newtons per square centimeter absolute (14.735 lb/in.^{2}) and 378 K (680° R). Results are presented in terms of equivalent mass flow, equivalent torque, equivalent specific work output, turbine efficiency, and outlet flow angle as functions of equivalent speed and pressure ratio. Also presented are the end-wall static-pressure distributions through the turbines at the equivalent design speed.

The results of the variable-stator-area turbines are compared with those obtained from the design turbine investigation reported in reference 11.

SYMBOLS

h specific enthalpy, J/kg; Btu/lb

N rotational speed, rad/sec; rpm

p absolute pressure, N/m^2; lb/ft^2

R gas constant for mixture of air and combustion products used, 288 J/(kg)(K); 53.527 (ft-lb)/(lb)(^{o}R)

T temperature, K; ^{o}R

U blade velocity, m/sec; ft/sec

V absolute gas velocity, m/sec; ft/sec

W gas velocity relative to rotor blade, m/sec; ft/sec

w mass-flow rate (sum of airflow and fuel flow), kg/sec; lb/sec

α absolute gas flow angle measured from axial direction, deg

$\bar{\alpha}$ average absolute gas flow angle at turbine outlet measured as average deviation from axial direction

β relative gas flow angle measured from axial direction

γ ratio of specific heats, 1.398, for mixture of air and combustion products used

δ ratio of inlet pressure to U.S. standard sea-level pressure

ϵ function of γ, $\frac{0.73959}{\gamma}\left[\left(\frac{\gamma+1}{2}\right)^{\gamma/(\gamma-1)}\right]$

η efficiency based on total pressure ratio

θ_{cr} squared ratio of critical velocity at turbine inlet to critical velocity of U.S. standard sea-level air

τ torque, N-m; ft-lb

Subscripts:

cr condition at Mach 1

h turbine blade hub section

m turbine blade mean section

t turbine blade tip section

u tangential component

x axial component

0 station at turbine inlet, see fig. 6

1 station at first-stage stator outlet

2 station at first-stage rotor outlet

3 station at second-stage stator outlet

4 station at second-stage rotor outlet

Superscript:

' total state

APPARATUS

Turbine Description

The design of the first stage of the research turbine is described in reference 1; the second-stage design is presented in reference 11. The turbine has an outer diameter of 76.2 centimeters (30 in.) and an inner diameter of 55.9 centimeters (22 in.) through the first stage. The flow area increases through the second stage. At the outlet the inner and outer diameters are 51.2 and 80.9 centimeters (20.15 and 31.85 in.), respectively. This flow area increase is depicted in the flow-path projection shown in figure 1. The blading had a constant mean diameter of 66.04 centimeters (26.0 in.). Also shown in figure 1 is the number of blades per blade row.

The turbine was designed for the following equivalent conditions, based on U.S. standard sea-level pressure and temperature:

Specific work output, $\Delta h/\theta_{cr}$, J/kg (Btu/lb) 76 818 (33.0)
Mass flow, $\epsilon w \sqrt{\theta_{cr}}/\delta$, kg/sec (lb/sec) 18.10 (39.9)
Torque, $\epsilon\tau/\delta$, N-m (ft-lb) 3010 (2220)
Rotational speed, $N/\sqrt{\theta_{cr}}$, rpm 4407

The resultant turbine design velocity diagram and blade-surface velocity distributions corresponding to these conditions are reproduced from reference 11 as figures 2 and 3 herein. A schematic of the blading and the flow path through the blading for the design turbine are shown in figure 4. The blade profiles and the solidity for each blade row can be noted for the blade hub, mean, and tip sections. The first stage of the turbine was designed for 51.5 percent of the total turbine work, and the second stage for 48.5 percent. Test results of reference 11 showed the corresponding experimentally obtained stage work distribution was 50.5 and 49.5 percent. A photograph of the two-stage turbine rotor assembly is shown in figure 5. This turbine is hereinafter referred to as the design turbine.

For the subject variable-stator-area-turbine test program, the first-stage stator blades were reoriented to that stagger angle required to effect a change in the mean-section channel outlet orthogonal of ±30 percent from that of design. The resultant stagger-angle changes for the first-stage stator blading are noted in figure 6(a) (a reproduction from ref. 9). This reference reports the results of tests made on the first stage of this turbine with the same stator stagger-angle (or flow area, as represented by the orthogonal length) changes.

The second-stage stator blade area was also varied for the tests reported herein. For the turbine with the 70-percent area first-stage stators, the second-stage stator area was similarly reduced. This turbine configuration is hereinafter referred to as the closed turbine.

It was reported in reference 6 that increasing the first-stage stator area to 130 percent of its design area resulted in an increase in the choking equivalent mass flow at the design speed of only 16.4 percent. And the flow limitation was caused by rotor-blade-exit choke. Accordingly, for the subject opened two-stage turbine configuration, the same first-stage stator was used, but the second-stage stator area was increased by only 20 percent (nominal). This was slightly more than that required to pass the expected mass flow. It was felt that the second-stage stator losses would be lessened somewhat with the stator flow area matching the expected flow. The second-stage area changes were also effected by varying the blade-profile stagger angle. These changes are shown in figure 6(b). The final mean-section stagger angle for both stator blade rows and the three different turbine configurations are shown in the following table:

Turbine configuration	First stage stator, deg	Second stage stator, deg
Design	41.03	37.12
Closed	48.82	47.90
Opened	32.59	31.72

Figure 6 shows the stator blading angular reorientation was effected by pivoting all blades about the trailing-edge radius. An exception was made for the closed-second-stage blading configuration. Here, the blading was moved axially forward somewhat in order to provide a more even distribution of axial clearance between the adjacent rotor blades. All stators were constructed with zero end-wall clearance.

Test Equipment

The test facility for the variable stator area turbines was identical to that used for the performance evaluation of the reference design turbine (ref. 11). A photograph of the experimental facility is presented in figure 7. A sketch of the turbine test section is shown in figure 8.

Airflow to the turbine was supplied by the laboratory 37.7-newton-per-square-centimeter (40-psig) combustion air system. The air was metered and throttled to provide the desired turbine-inlet pressure. A portion of the metered high-pressure air was heated by a modified commercial jet-engine burner and permitted to reenter the primary air supply. By regulating the amount of bypassed air and the fuel flow (natural gas) to the burner the air temperature to the turbine could be maintained at the desired value of 378 K (680° R). This heated air was directed into two pipes, and entered a plenum chamber from opposite sides. This plenum, along with a screen at the inlet of the converging passage upstream of the turbine blading (see fig. 8), minimized pressure distortion at the turbine inlet measuring station. After passing through the turbine, the spent air was discharged into the laboratory altitude exhaust system. Pressure ratio across the turbine was varied by butterfly throttle valves located in the exhaust duct. Two 3730-watt (5000-hp) eddy-current, wet-gap dynamometers, each cradled on hydrostatic bearings, were used to absorb turbine power output, control the turbine speed, and measure torque output.

INSTRUMENTATION

The instrumentation required to determine the overall performance for the opened- and closed-stator turbine configurations was the same as that for the design turbine (ref. 11). Airflow was metered using a calibrated Dall tube, which is a special type of venturi meter. Fuel flow to the air heater was measured using a flat-plate orifice. Turbine primary airflow was corrected for this fuel addition.

Rotative speed was measured with an electronic counter in conjunction with a magnetic pickup and a toothed sprocket secured to the turbine shaft. A torque arm attached to the coupled stators of the two dynamometers transmitted the turbine torque output to a strain-gage load-cell transducer with a digital voltmeter readout.

Turbine research measurements were made before and after each blade row. The axial locations of these measuring stations are shown in figure 8. Pertinent measurements at each of these measuring stations are given in figure 9. The static pressure taps behind each row of stator blades were adjusted for the changes in stagger angle such that the taps were always located midway between adjacent blades and on the mean projected through-flow path. Research pressure and temperature data were digitized and recorded on paper tape from their respective transducers. At each data-set point, five readings of each transducer were recorded and were subsequently averaged before computing turbine performance parameters.

PROCEDURE

Both turbine builds with the modified stator areas were tested with the same turbine air-inlet conditions as was the design turbine (ref. 11), that is, a turbine inlet pressure of 10.159 newtons per square meter absolute (14.735 lb/in.^{2}) and a temperature of 378 K (680° R). All three turbine configurations were operated over a range of equivalent speed $N/\sqrt{\theta_{cr}}$ from 40 to about 110 percent of the design equivalent speed.

Inlet total pressure p_0' was calculated from measured values of static pressure, mass-flow rate, and inlet-total temperature. The outlet total pressure p_4' was similarly calculated except that allowance is made for the flow being at some angle other than axial. Appropriate equations for these calculations are defined and discussed in reference 11.

The equivalent work $\Delta h/\theta_{cr}$ and the efficiency η as used herein are based on measured torque values. Calculated values of equivalent specific mass flow $\epsilon w\sqrt{\theta_{cr}}/\delta$ were first plotted as functions of total-pressure ratio p_0'/p_4'. Curves were drawn through these data points for each test speed. Then, for even increments of pressure ratio the actual- and ideal-work outputs were computed, and turbine performance maps were constructed. These data are presented herein for the closed and the opened turbine configurations. Also included for comparison are pertinent corresponding curves from reference 11 for the standard (design) turbine tests.

RESULTS AND DISCUSSION

This section of the report will be presented in three parts. First, the overall experimental performance for the closed (70 percent) stator area turbine configuration will be presented and discussed. Then the results for the opened (130 percent) stator area turbine will be presented. Finally, these performance results will be compared with those previously reported for the turbine having the design stator areas (ref. 11), with particular emphasis on data obtained at the equivalent design speed. Specific comparisons for all three turbine configurations are made at the pressure ratio (3.215) where equivalent specific design work output (76 818 J/kg; 33.0 Btu/lb) was obtained for the design turbine (ref. 11). For the convenience of the reader, the performance map for this reference turbine is reproduced and presented herein as figure 10.

Overall Performance of Closed Turbine

Basic turbine data are shown in figures 11 and 12 with equivalent mass flow $\epsilon w \sqrt{\theta_{cr}}/\delta$ and equivalent torque $\epsilon \tau/\delta$ shown as functions of total-pressure ratio p_0'/p_4' and equivalent speed $N/\sqrt{\theta_{cr}}$. Figure 11 shows that speed has only a small effect on the equivalent mass flow over the range of pressure ratios investigated. In fact, at pressure ratios greater than about 3.4, data for all speeds fall on a single line with a maximum random scatter less than about 1/4 percent. These data indicate that the first-stage stator controls the turbine flow, as would be expected since the area was reduced to nominally 70 percent of design. Actually, the data show that a maximum flow of 13.32 kilograms per second (29.36 lb/sec) was attained. This value corresponds to 0.723 of the maximum mass flow obtained for the design turbine at equivalent design speed.

Figure 12 shows that equivalent torque increases with pressure ratio for all speeds investigated. Hence, limiting loading of the blading was not attained in these tests. It will also be noted that equivalent design torque (3010 N-m; 2220 ft-lb) was not obtained at equivalent speeds above 70 percent of design. At equivalent design speed a maximum torque of 2488 newton-meters (1835 ft-lb) was obtained at the maximum test pressure ratio of 4.58. To a large extent, this limitation results from the aforementioned flow deficiency.

Figure 12 shows that the 110 percent of equivalent speed line ended at a pressure ratio of 2.96. This was necessitated because of excessive rig vibration. At the higher pressure ratio range the overspeed was therefore limited to 105 percent of the design equivalent speed.

Data from figures 11 and 12 were used to evolve the performance map shown in figure 13. Thereon, equivalent specific work output $\Delta h/\theta_{cr}$ is shown as a function of equivalent mass flow-speed parameter $\epsilon wN/\delta$. Lines of constant total-pressure ratio and efficiency η are superimposed. A peak efficiency of over 0.88 was obtained in the high-speed - low-pressure-ratio regime. At equivalent design speed and a pressure ratio of 3.215, corresponding to the value where equivalent design work was obtained for the design turbine (ref. 11), the efficiency was 0.858. This point is denoted by a symbol on the map. It relates to an equivalent specific work output of about 70 530 joules per kilogram (30.3 Btu/lb) and corresponds to about 0.92 of design. It should be noted that at a pressure ratio of 3.215 the equivalent weight flow (fig. 11) is very nearly choked. This can also be seen in figure 13, wherein the 100-percent speed line is not quite vertical.

Each data point in figure 14 is an arithmetic average $\overline{\alpha}_4$ of five actuator readouts of turbine outlet flow angle. These data are shown as a function of total-pressure ratio for the various speeds. Negative angle values correspond to outlet flows opposite to rotor rotation, and this swirl would represent a positive contribution to turbine work output. Figure 14 shows a definite speed effect. At any given pressure ratio, lowering the speed results in outlet flow angles becoming more negative. At the previously mentioned total pressure ratio of 3.215 and at the design equivalent speed, the outlet flow angle was 23°.

The preceding discussion was limited to salient findings for the closed stator-area turbine. No comprehensive comparisons with the design turbine (ref. 11) was attempted. The ensuing section discusses the opened stator area turbine, after which the performance of the three turbine configurations will be compared.

Overall Performance of Opened Turbine

Data for the opened stator-area turbine configuration are presented in figures 15 and 16 in terms of equivalent mass flow and torque, respectively, as functions of pressure ratio and equivalent speed. The performance map is shown on figure 17.

Figure 15 shows equivalent mass flow to be significantly affected by speed. Also, in the higher pressure-ratio range and for all speeds tested, choking flow can be noted. That these choking values of flow decrease with increasing speed is indicative that the first-stage stator was unchoked. Some downstream blade row was limiting the flow. It will be noted that the maximum flow obtained at the equivalent design speed was 22.04 kilograms per second (48.60 lb/sec). This corresponds to 1.197 that of the choked flow for the design turbine (18.42 kg/sec or 40.60 lb/sec; ref. 11) at the same equivalent speed. This choking value is slightly higher than the value as found in the opened, single-stage tests of reference 6. The difference probably results from the effect of adding the second stage.

The same general comments pertain to the equivalent torque curve (fig. 16) as expressed for the closed turbine configuration (fig. 12). The overall level of torque, however, is higher. Also, at the higher speeds, the torque curves appear to flatten out; that is, large increases in pressure ratio result in small increases in equivalent torque. This indicates that the turbine is approaching limiting-blade loading.

The performance map (fig. 17) shows a peak efficiency contour of 0.90, occurring at an equivalent speed of 90 percent and at low pressure ratios. The symbol on the equivalent design speed curve at a pressure ratio of 3.215, again, corresponds to the turbine operating point where equivalent design work was attained with the design turbine (ref. 11). At this operating point the subject opened turbine produced 73 210 joules per kilogram (31.45 Btu/lb) or about 0.95 of the design work. The attendant efficiency at this point (fig. 17) was 0.889. Further, in this higher pressure ratio range, small changes in equivalent specific work output result with increases in pressure ratio. In fact, equivalent design specific work output was not attained at the equivalent design rotor speed. A maximum of 0.98 of design work is indicated. These pressure ratio-work characteristics, coupled with the observable rapid deterioration of turbine efficiency, indicate that the turbine is near limiting loading, as educed from the torque curve (fig. 16) discussion.

Turbine outlet flow angle for the opened turbine configuration is shown in figure 18. At comparable pressure ratios, these data show outlet angles considerably lower (more negative) than those noted for the closed turbine (fig. 14). This was to be expected from velocity diagram considerations. At the equivalent design speed and the previously stated pressure ratio of 3.215, the outlet flow angle was -20°.

COMPARATIVE STUDY OF TURBINE PERFORMANCE

Turbine performance maps for the design-, the closed-, and the opened-turbine configurations have been presented in figures 10, 13, and 17, respectively. It is of interest to compare the overall efficiency levels for the three turbines. The design turbine yielded the highest efficiency. This was expected in view of the significant changes in stator area (±30 percent of design) incorporated in the closed- and the opened-turbine configurations. Further, the area of highest efficiency on the map (fig. 10) occurs near the design operating point. Both the closed- and the open-turbine performance maps (figs. 13 and 17) show that the peak efficiencies were decreased, and occur at relatively low pressure ratios.

Engine cycle analysis based on a specific mode of engine operation for a given mission dictates the turbine operating point, and it will change with different stator areas. There are any number of different modes of engine operation at which turbine performance can be compared. With the three performance maps presented, and within the scope of stator area changes tested, turbine performance can be determined at any speed and pressure ratio. As for the first-stage variable stator-area turbine studies (summarized in ref. 10), the authors chose to compare turbine performance at the equivalent design speed. At this speed the specific operating point chosen for comparison is herein designated as that pressure ratio (3.215) at which equivalent specific design work was obtained for the design turbine (ref. 11). This is in contrast to the comparison point used in reference 10 wherein results were compared at the point at which each turbine yielded the design specific work output of the design single-stage turbine.

The ensuing discussion will first compare some measured test variables at the equivalent design speed and over the range of pressure ratio investigated for the three turbines. Corresponding turbine efficiencies will similarly be compared. Emphasis will be placed on the performance obtained at the aforementioned reference pressure ratio of 3.215. Possible reasons for the efficiency variations will be discussed, primarily in terms of reaction differences across the blading for the three turbines.

Flow, torque, and exit angle. - The variation of equivalent mass flow at the equivalent design speed for the three variable stator area turbines is presented in figure 19 over the ranges of pressure ratio investigated. Data for the design turbine are reproduced from reference 11. The trends of the three curves are obviously similar. Indicated on the abscissa is the pressure ratio (3.215) corresponding to the point of comparison. At this point all three turbines were operating at or close to their respective choking flow values.

The closed turbine (fig. 19) shows a maximum (choking) flow value of 13.32 kilograms per second (29.36 lb/sec), or 0.723 of the maximum flow obtained for the design turbine (18.42 kg/sec; 40.60 lb/sec). The choking flow for the opened turbine was 22.04 kilograms per second (48.60 lb/sec), which is about 19.7 percent greater than the choking flow for the design turbine, even though the first-stator area was increased nominally by 30 percent. This results from the fact that the first-stage stator did not control the flow (see fig. 15). Further, when the same opened first-stage stator was tested with the first-stage rotor as a separate (single-stage) component, the choking flow was increased only 16.4 percent (ref. 6). This increase indicated that the first stage operated differently when the second stage was added. The change to a 19.7-percent flow increase probably results from differences in reaction (pressure drop) across the first stage as affected by the addition of the second stage. The same reasoning may apply for the closed turbine. Here, the stator areas were decreased nominally by 30 percent of the design area, yet 0.723 of the choked flow for the design turbine was observed. The reaction characteristics of the three turbines at equivalent design speed and over a range of pressure ratio will be discussed later.

The equivalent torque curves for the three turbines at equivalent design speed are presented in figure 20 as functions of turbine pressure ratio. The curves show that at any given pressure ratio, increasing the stator area increases the torque output. This results from the aforementioned concomitant flow increase (fig. 19). Comparing the equivalent torque values at the pressure ratio of 3.215 shows the closed-turbine produced 2028 newton-meters (1496 ft-lb), the design turbine 3052 newton-meters (2251 ft-lb), and the open turbine 3493 newton-meters (2576 ft-lb). In other words, closing the stator area resulted in the turbine obtaining a torque only 66.5 percent of the design turbine torque. Opening the stator area increased the torque by only 14.4 percent.

It is also of interest to note (fig. 20) that a maximum torque value of 2488 newton-meters (1835 ft-lb) was obtained for the closed turbine configuration at the maximum pressure ratio tested. This corresponds to 0.826 of equivalent design torque for the design turbine.

Figure 21 compares the average turbine outlet flow angle $\overline{\alpha}_4$ for the three different two-stage turbines over the ranges of pressure ratio tested at the equivalent design speed. Note that the trends of the three curves are similar. A negative angle means the flow direction is opposite rotor rotation, and this contributes to turbine work output. Figure 21 shows that for any given pressure ratio, opening the stator area resulted in outlet angles tending toward the negative (overturning). The converse occurred when the stator area was closed (underturning).

At a pressure ratio of 3.215 and for the design turbine, the exit flow angle was -5° (ref. 11). The design exit flow angle was -10°. Also, at this pressure ratio, the exit flow angle for the closed turbine was 23°, considerably underturned. The exit flow angle for the opened turbine was -20° (overturned).

Efficiency. - A comparison of the efficiency variation with pressure ratio at the equivalent design speed for the three turbines is shown in figure 22. As expected, the design turbine exhibited the highest efficiency over the entire range of pressure ratio tested. The peak efficiency for the design turbine was about 0.934 (ref. 11). This compares with peak values of 0.897 for the opened turbine and 0.887 for the closed turbine.

Also shown in figure 22 is the 3.215 pressure ratio for design two-stage specific work output. At this point the design turbine efficiency was 0.932 (ref. 11), the closed turbine efficiency was 0.858, and the open turbine efficiency was 0.889.

In general, the efficiency characteristics for the two-stage turbines (fig. 22) were similar to those exhibited for the three single-stage variable stator area turbines. These data are presented in figure 23 (reproduced from refs. 9 and 10). A brief comparison of the two efficiency curves (figs. 22 and 23) shows that the design turbines produced the highest efficiencies and the closed turbines produced the lowest efficiencies. Further, adding the second stage to the open turbine lowered its peak efficiency value. The range of peak efficiency for the two-stage, opened turbine, however, was considerably broadened.

The vertical line in figure 23 at a single-stage turbine pressure ratio of 1.751 corresponds to that pressure ratio at which the design single-stage turbine produced the design specific work output (39 570 J/kg; 17.0 Btu/lb). Using this pressure ratio as a basis for comparison (instead of the design stage work used in refs. 9 and 10) the reported efficiency for the design turbine was 0.923, for the open turbine 0.903, and for the closed turbine 0.853. The peak- and comparison-point efficiencies for both the single- and two-stage turbines and for the three configurations are summarized in the following table:

Turbine	Peak efficiency		Comparison point efficiency	
	One stage (ref. 10)	Two stages	One stage (a)	Two stages (b)
Design	0.923	0.934	0.923	.932
Closed	.869	.887	.853	.858
Open	.909	.897	.903	.889

[a]Pressure ratio, 1.751.

[b]Pressure ratio, 3.215.

Static-pressure distribution. - Changes in stator throat areas have been shown to have significant effects on measured turbine parameters. Also, either opening or closing the stator areas resulted in decreased turbine efficiency. This was expected inasmuch as the turbine blade rows were designed to accept the flow at its optimum flow angle. The degree to which the stator flow angle was varied for the tests reported herein would result in rotor-blade incidence losses. Further, the resultant flow area changes would cause off-design blade surface velocities because of changes in reaction across the blade rows for the three different turbine configurations.

Accordingly, the hub and tip static pressure variations through the design-, the closed-, and the open-turbine configurations are presented in figure 24. The abscissas indicate the location of the hub- and tip-static pressure measurements (see figs. 8 and 9). The ordinates correspond to the numerically averaged measured static pressures at the different measuring stations, normalized by dividing by the calculated inlet total pressure p_0'. Data for different total pressure ratios for each two-stage turbine configuration are presented. The data for the design turbine (fig. 24(a)) were obtained during the investigation of reference 11, but were not reported. They are presented herein for purposes of comparison.

The design turbine configuration data (fig. 24(a)) show conservative reaction through all four blade rows. The accumulation of data points at measuring station 2 (behind the first-stage rotor) for pressure ratios greater than 3.354 indicates choking of the second-stage stator. Subsequent choking of the second-stage rotor occurred at pressure ratios of about 3.788 and above. Both choked conditions, however, occurred at pressure ratios above 3.215, where design equivalent work was obtained (ref. 11).

Comparing figures 24(b) with (a) shows the significant effect on the reaction characteristics when the first-stator flow area was decreased to nominally 70 percent that of design. It is immediately apparent that at the hub, the air was greatly over expanded when passing through the stators (fig. 24(b)), with negative reaction (static-pressure rise) across the rotors. The concomitant effects of rotor blade incidence, the increased level of velocities through the stators, and the observed rotor negative reaction all tend toward flow separation from the blading and could explain the decrease in efficiency noted in figure 22. Figure 24(b) shows the second-stage stator of the closed turbine to be choked at pressure ratios of about 3.887 and above.

Comparable data for the open turbine (fig. 24(c)) show that the reaction across the first stage stator decreased as compared with the design turbine (fig. 24(a)). Conversely, however, the first-stage-rotor reaction was considerably increased. The second-stage shows similar reaction characteristics. The second-stage stator choked at both the hub and tip at pressure ratios above 2.887, followed by second-stage rotor blade choking at pressure ratios above 3.011.

The experimental data shown in figure 24 for the three turbines were cross-plotted so that the hub- and tip-static pressure variation at equivalent design speed and a total-pressure ratio of 3.215 could be evolved. These data are shown in figure 25. The data at the hub compares the aforementioned significant static-pressure (reaction) changes with turbine stator area changes. At the tip the reaction changes are not as severe, particularly through the second stage of the turbine.

SUMMARY OF RESULTS

A 0.762-meter (30-in.) two-stage turbine, designed to exemplify the aerodynamic problems associated with turbines for high-temperature-engine application, was experimentally investigated to determine the effect of variable stator area on turbine performance. This report presents results of two turbines with their respective stator areas either decreased or increased from design and compares their results with those previously obtained for the design stator area two-stage turbine. Pertinent results are as follows:

1. Decreasing the first-stage stator area by 30 percent from design resulted in the first-stage stator blade row choking at an equivalent mass flow 27.7 percent less than the choking flow for the design turbine at equivalent design speed.

2. Increasing the first-stage stator area by 30 percent resulted in an increase in the choking equivalent mass flow at the design equivalent speed of 19.7 percent. The flow was limited by choking conditions in the second-stage blading at high turbine pressure ratios.

3. Either increasing or decreasing the design stator area reduced turbine efficiency when compared with the design turbine at comparable operating conditions. Closing stator area resulted in the more severe efficiency loss. Specifically, the peak efficiencies found for the design-, the closed-, and the opened-turbine configurations were 0.934, 0.887, and 0.897, respectively. The corresponding match-point efficiencies (at a pressure ratio of 3.215) were 0.932, 0.858, and 0.889.

4. The losses in turbine efficiency effected by changes in stator area are attributable to rotor incidence, off-design blade-surface velocities, and adverse reaction changes

across the blade rows. The higher losses for the closed turbine probably resulted from the observed negative reaction at the rotor hubs, with possible local blade-flow separation.

Lewis Research Center,
National Aeronautics and Space Administration,
Cleveland, Ohio, November 1, 1973.
501-24.

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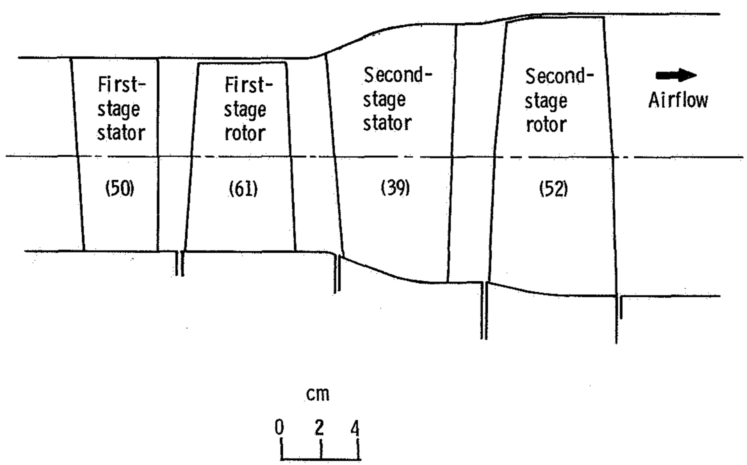


Figure 1. - Projection of flow path in radial-axial plane for two-stage cold-air turbine.

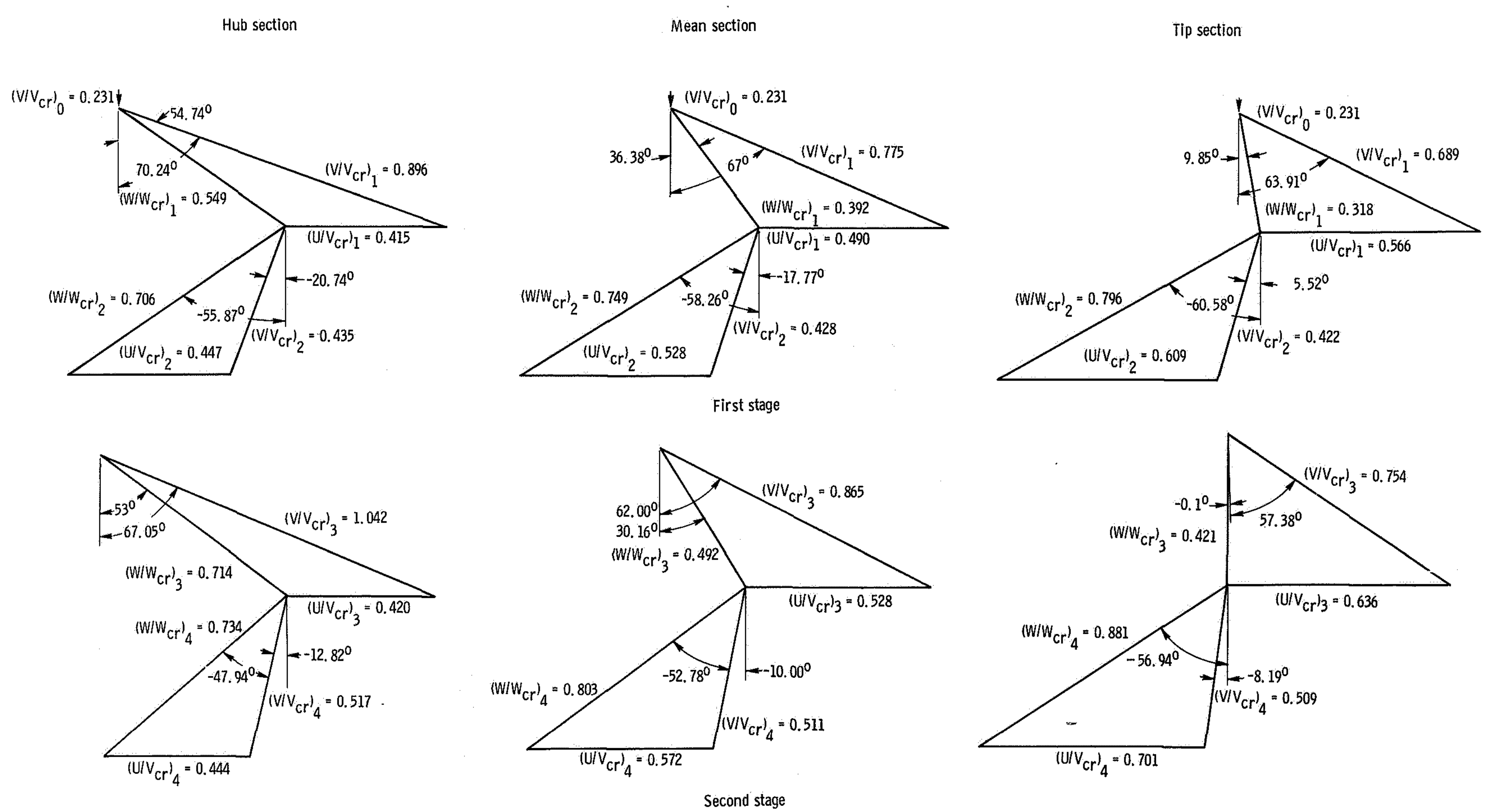
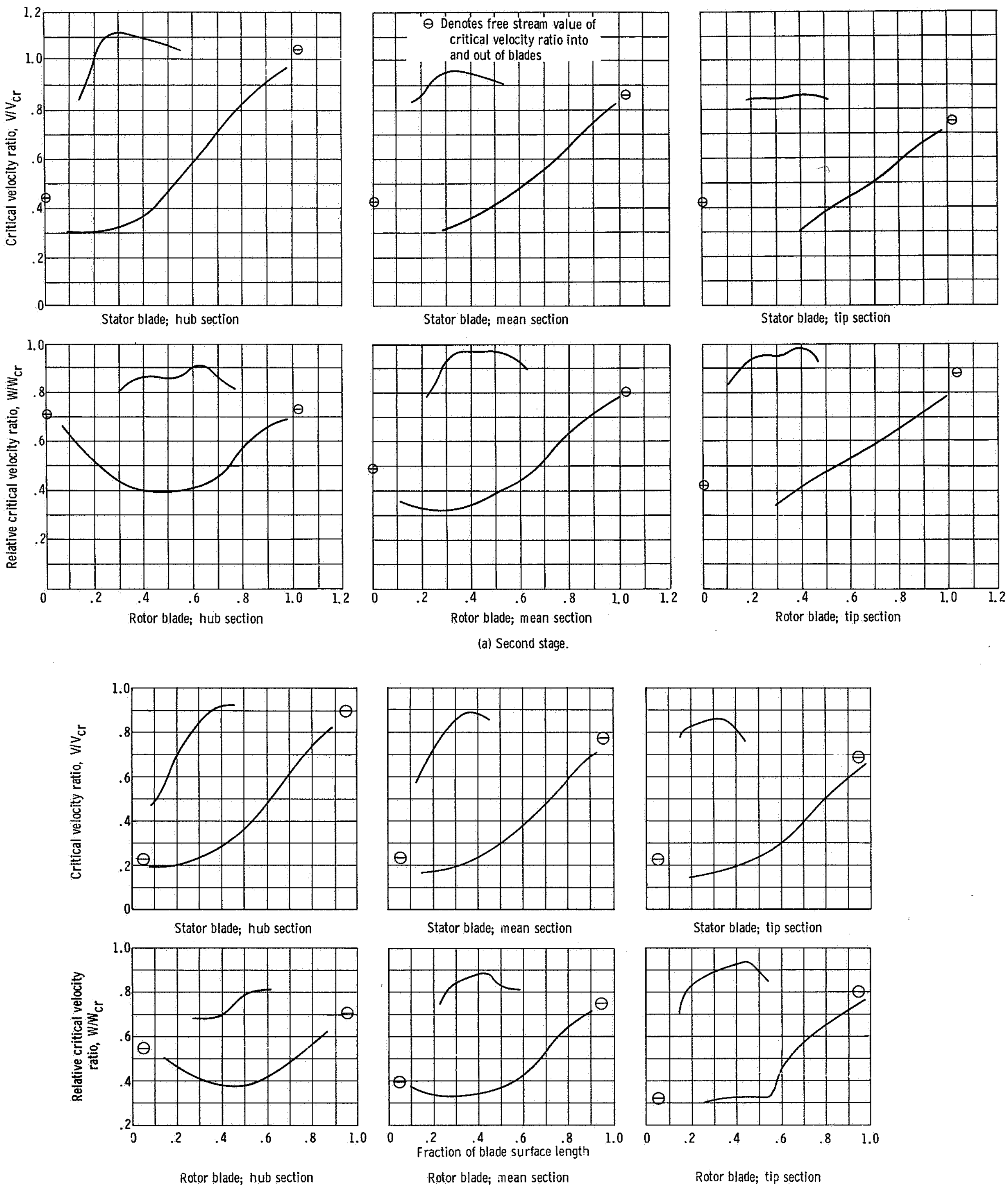


Figure 2. - Turbine design velocity diagram (from ref. 11).



(a) Second stage.

(b) First stage.

Figure 3. - Design blade-surface velocity distributions (from ref. 11).

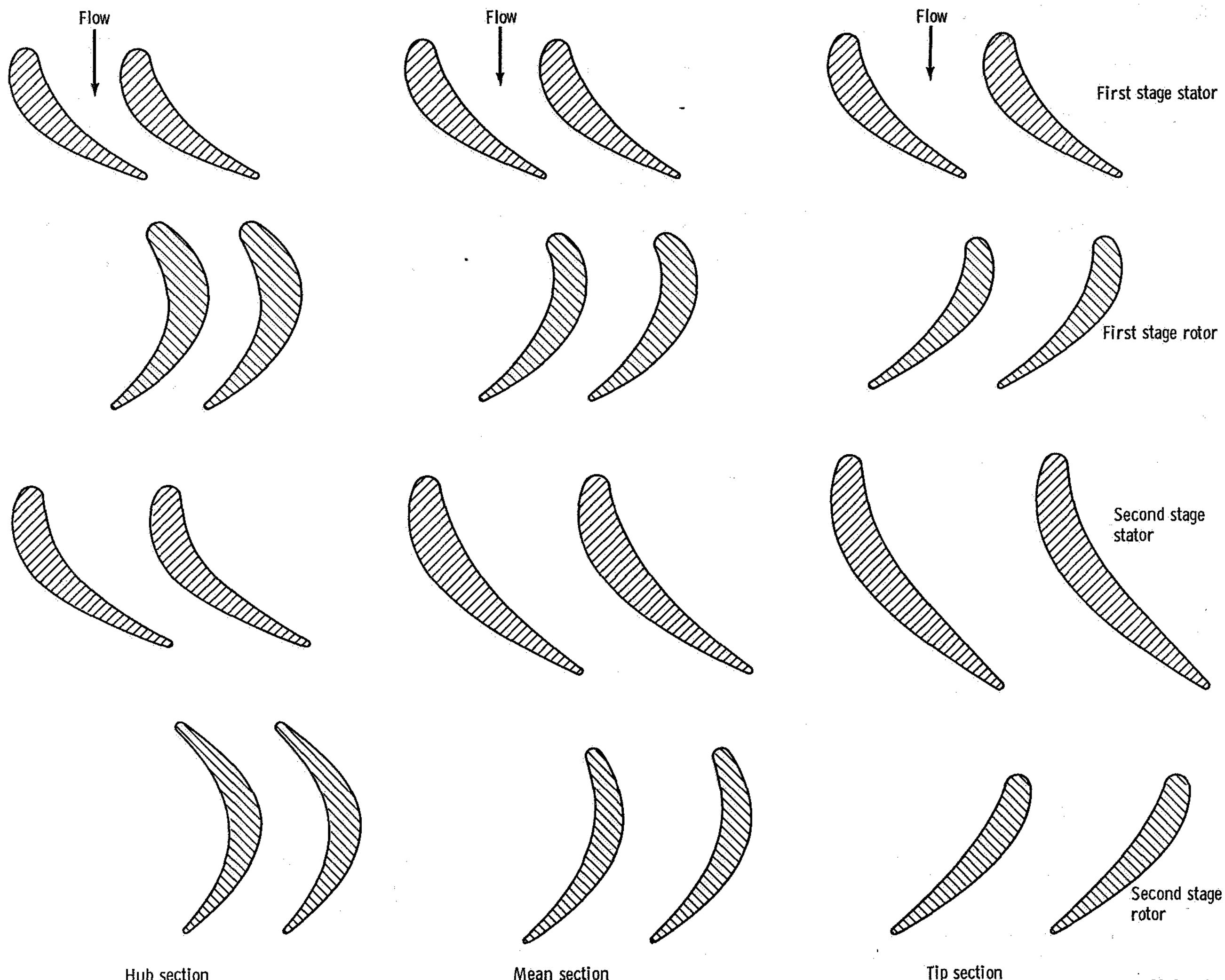


Figure 4. - Schematic of design turbine blading profiles and flow passages.

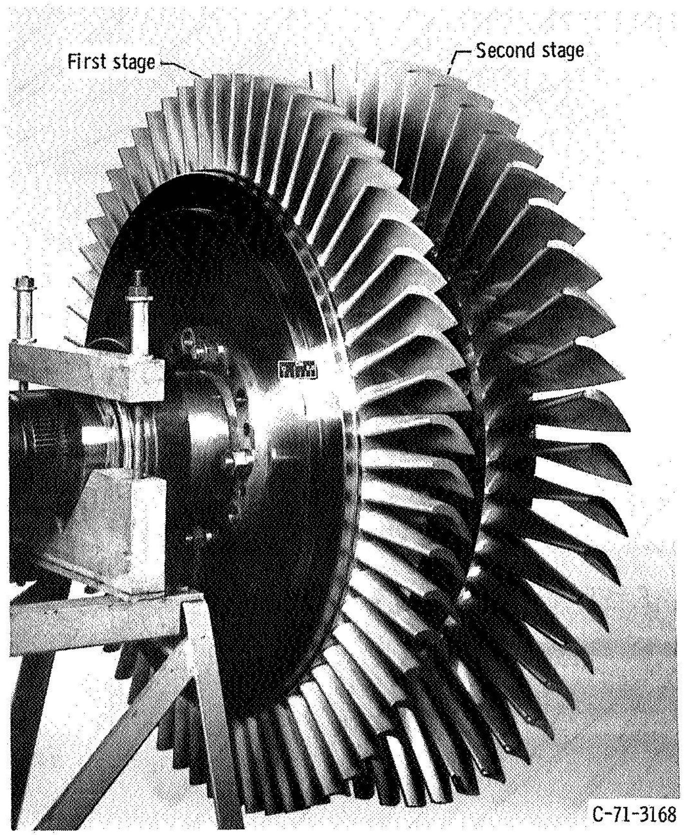
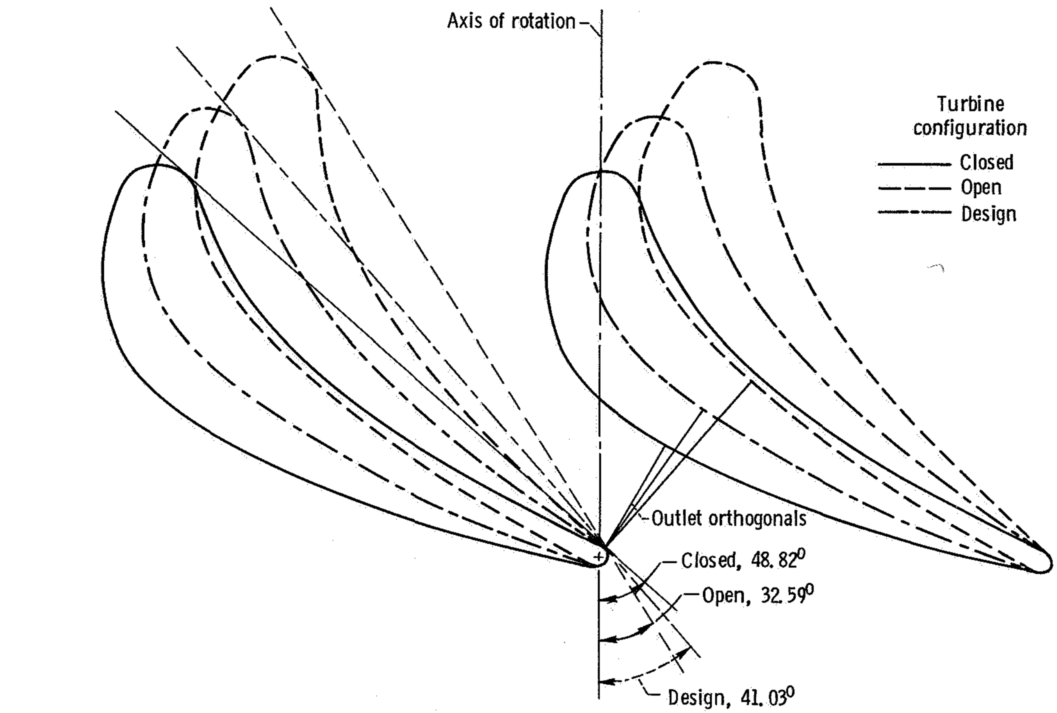
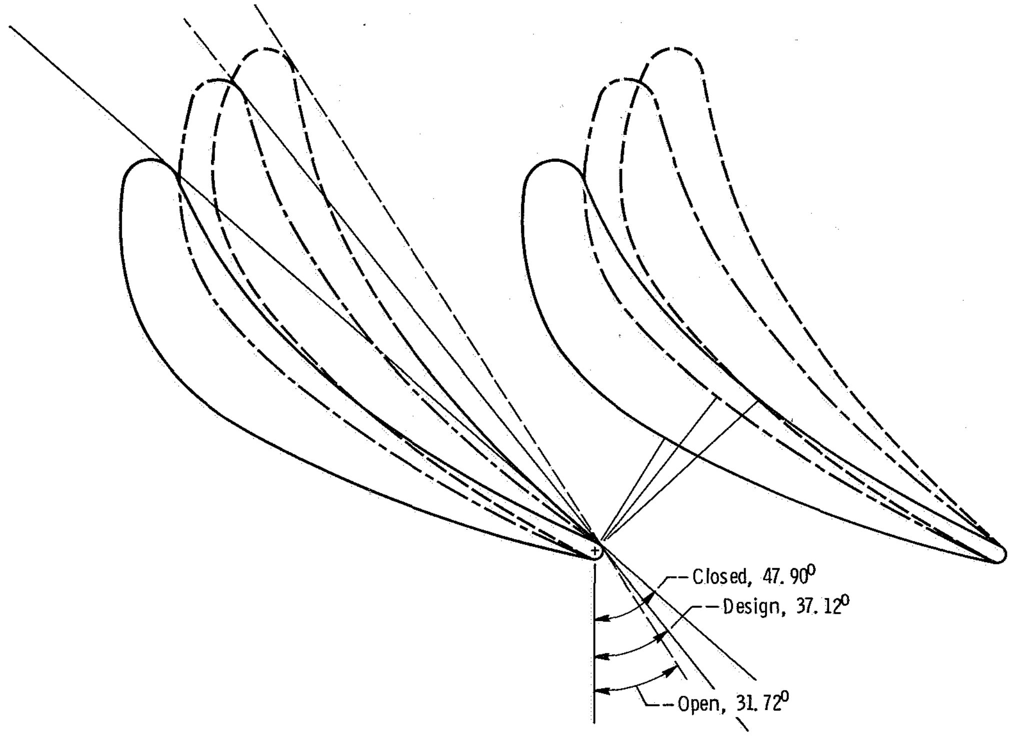


Figure 5. - Two-stage rotor assembly.



(a) First-stage stator (from ref. 9).



(b) Second-stage stator.

Figure 6. - Stator-blade mean-section orientation for three turbine configurations.

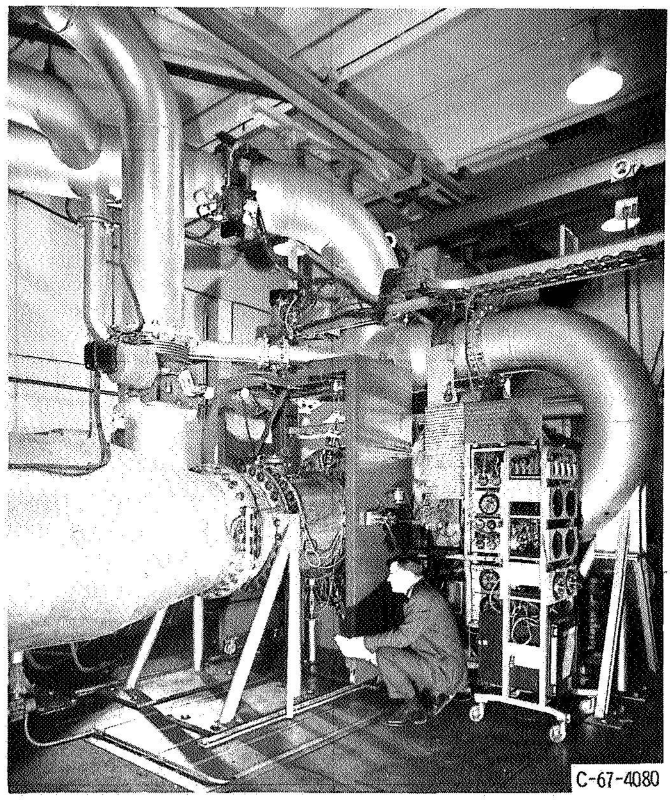


Figure 7. - Test facility.

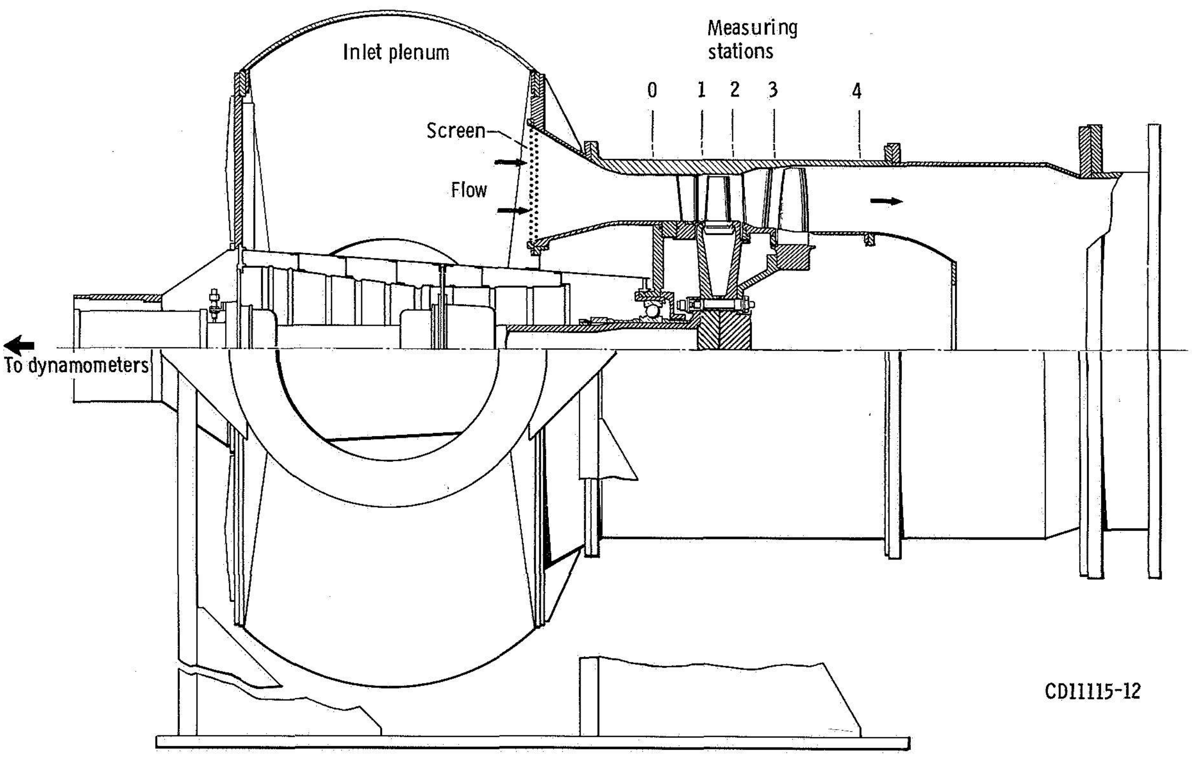


Figure 8. - Turbine test section and axial locations of measuring stations.

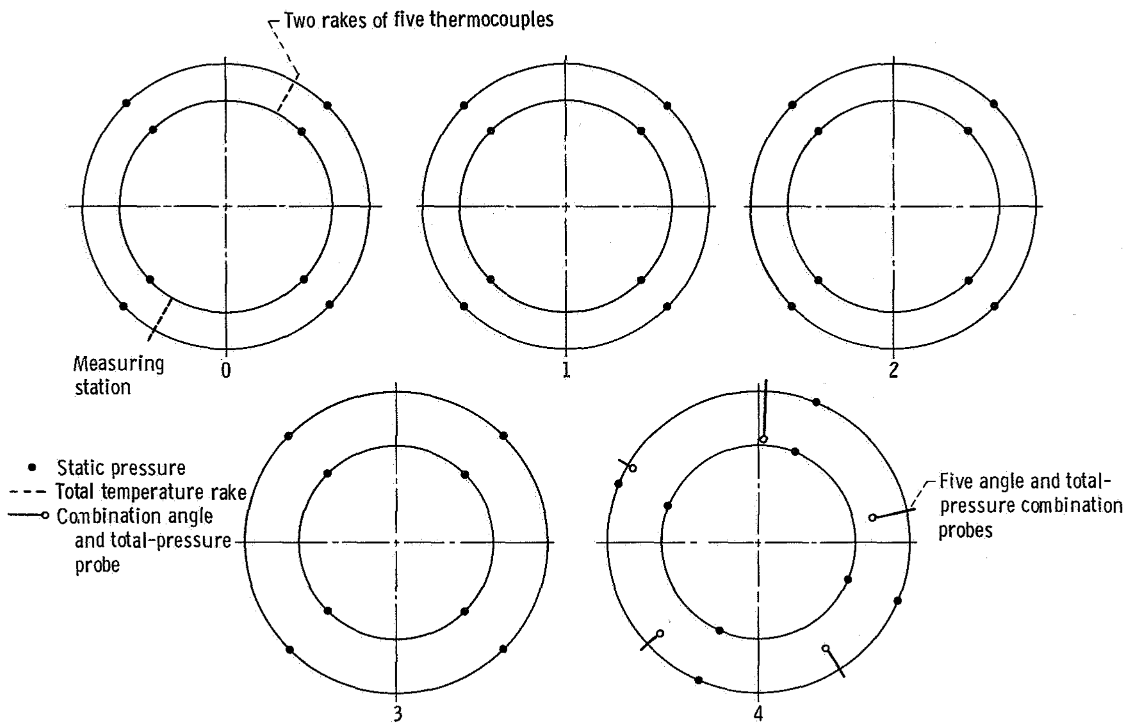


Figure 9. - Turbine instrumentation (looking upstream).

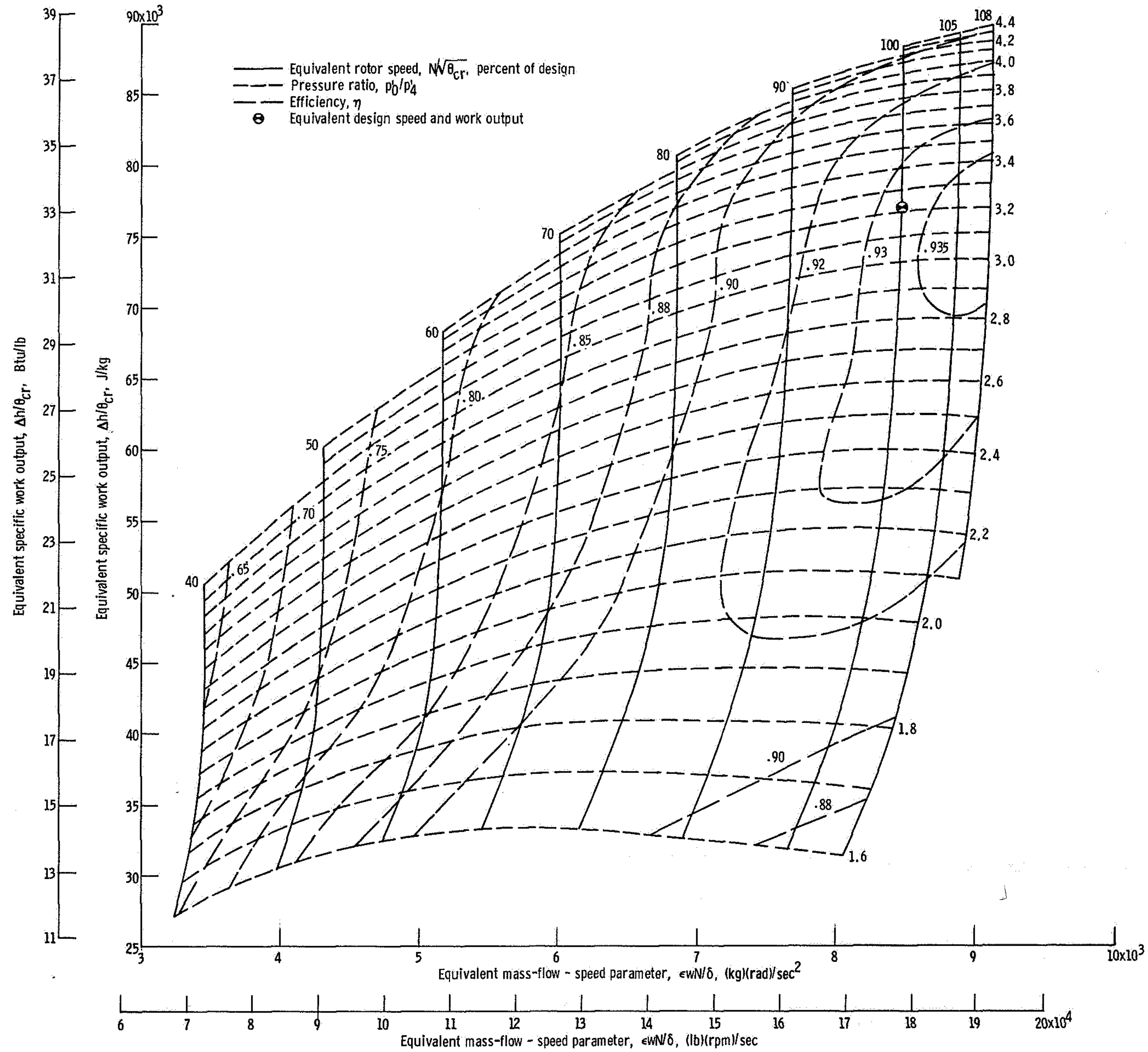


Figure 10. - Performance map for design turbine (from ref. 11).

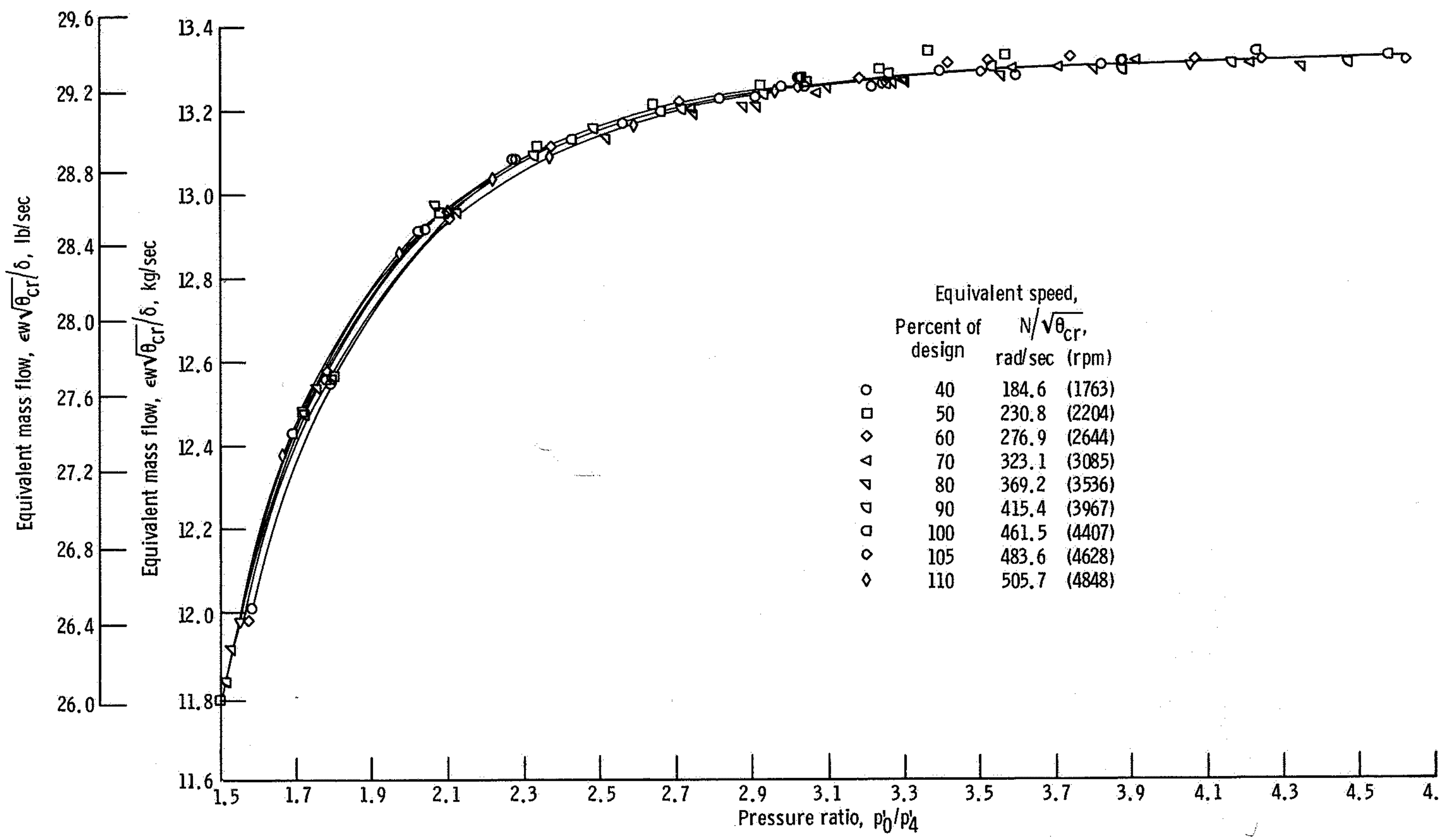


Figure 11. - Variation of equivalent mass flow with pressure ratio and equivalent speed for closed stator-area turbine.

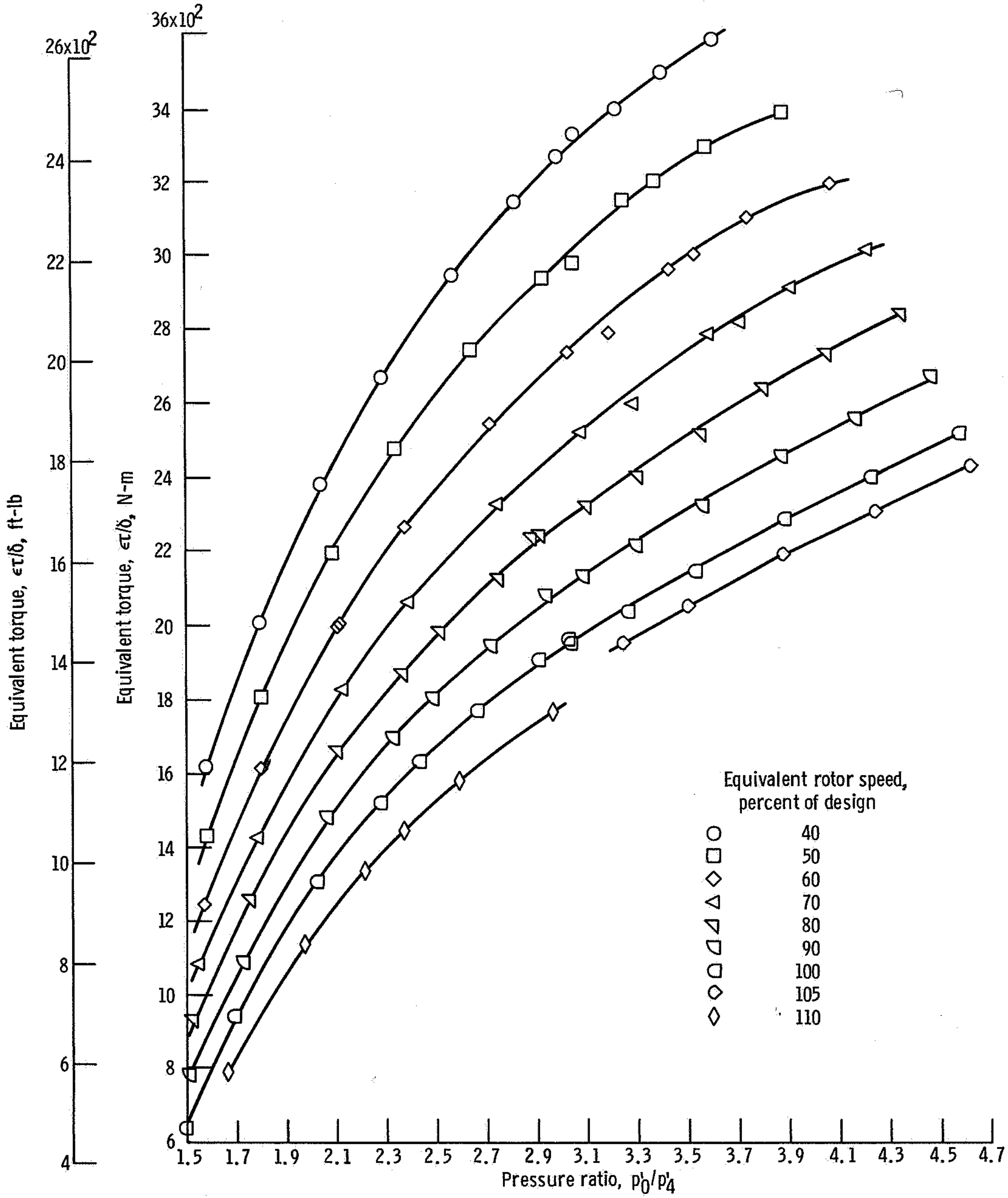


Figure 12. - Variation of equivalent torque with pressure ratio and equivalent speed for closed stator-area turbine.

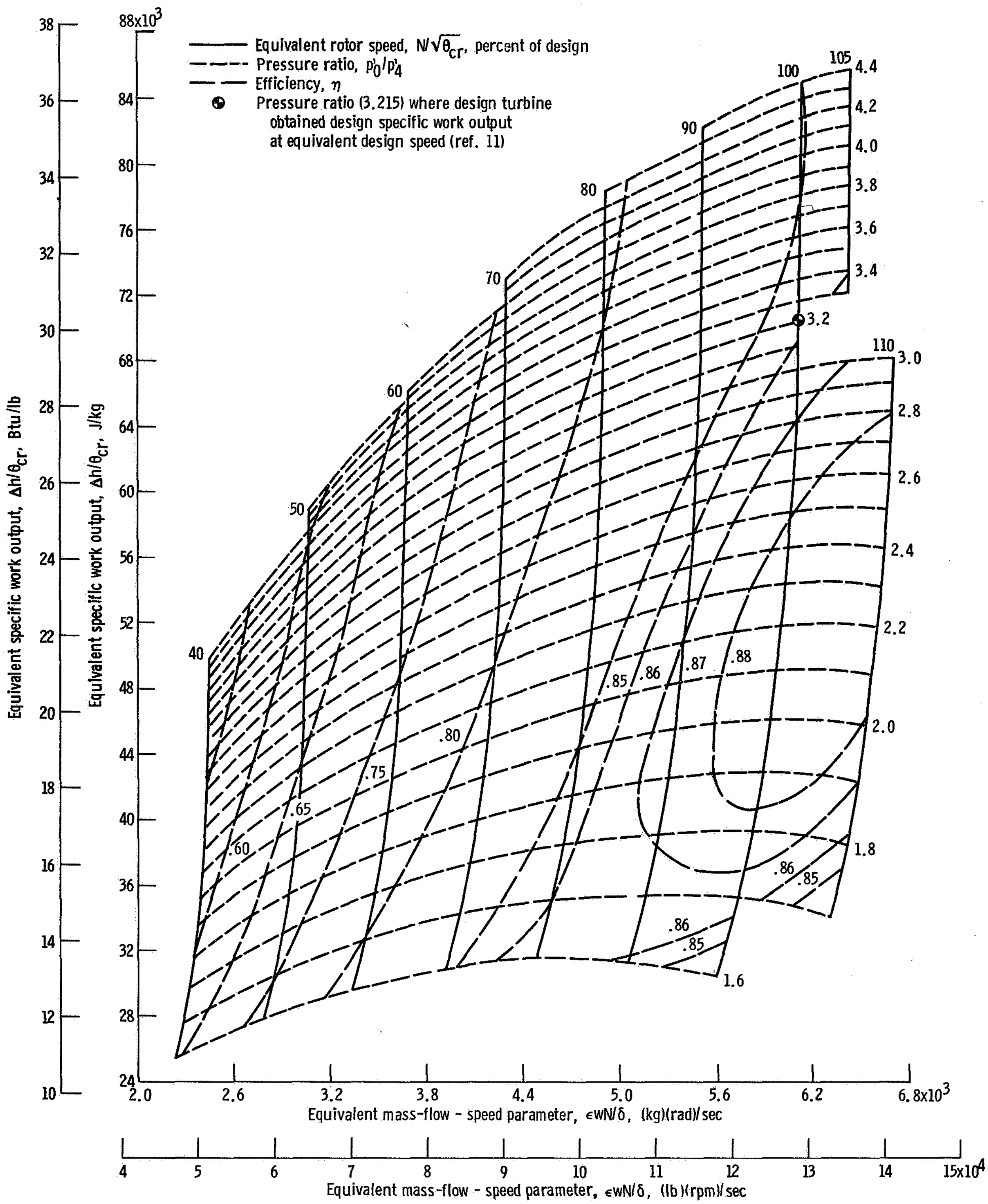


Figure 13. - Performance map for closed stator-area turbine.

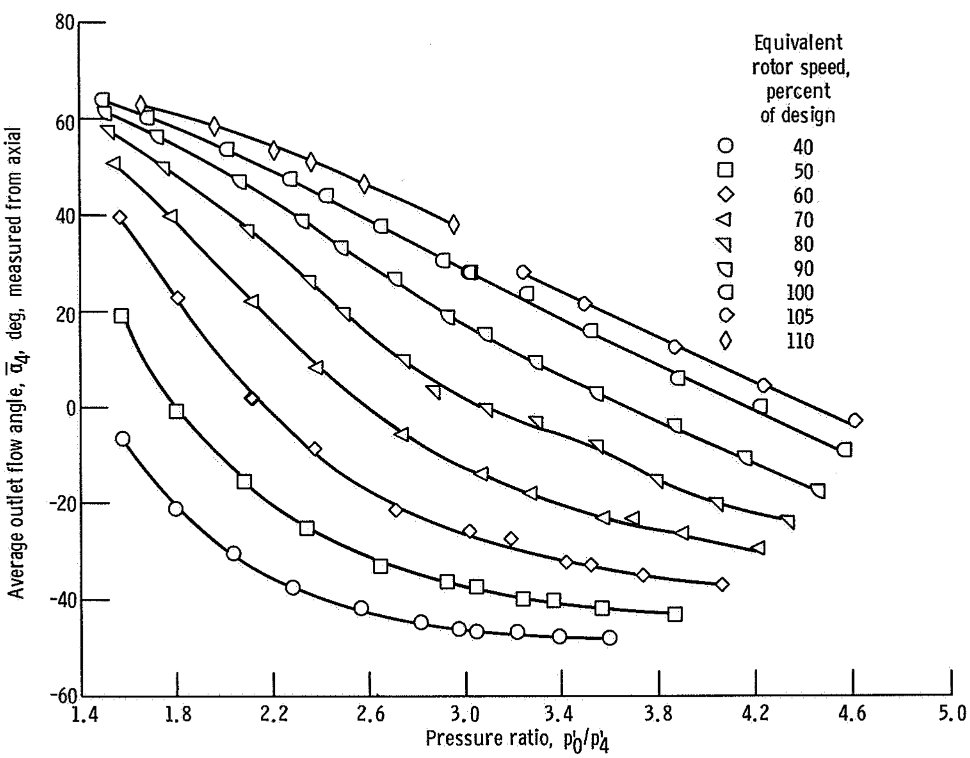


Figure 14. - Variation of average outlet flow angle with pressure ratio and equivalent speed for closed stator-area turbine.

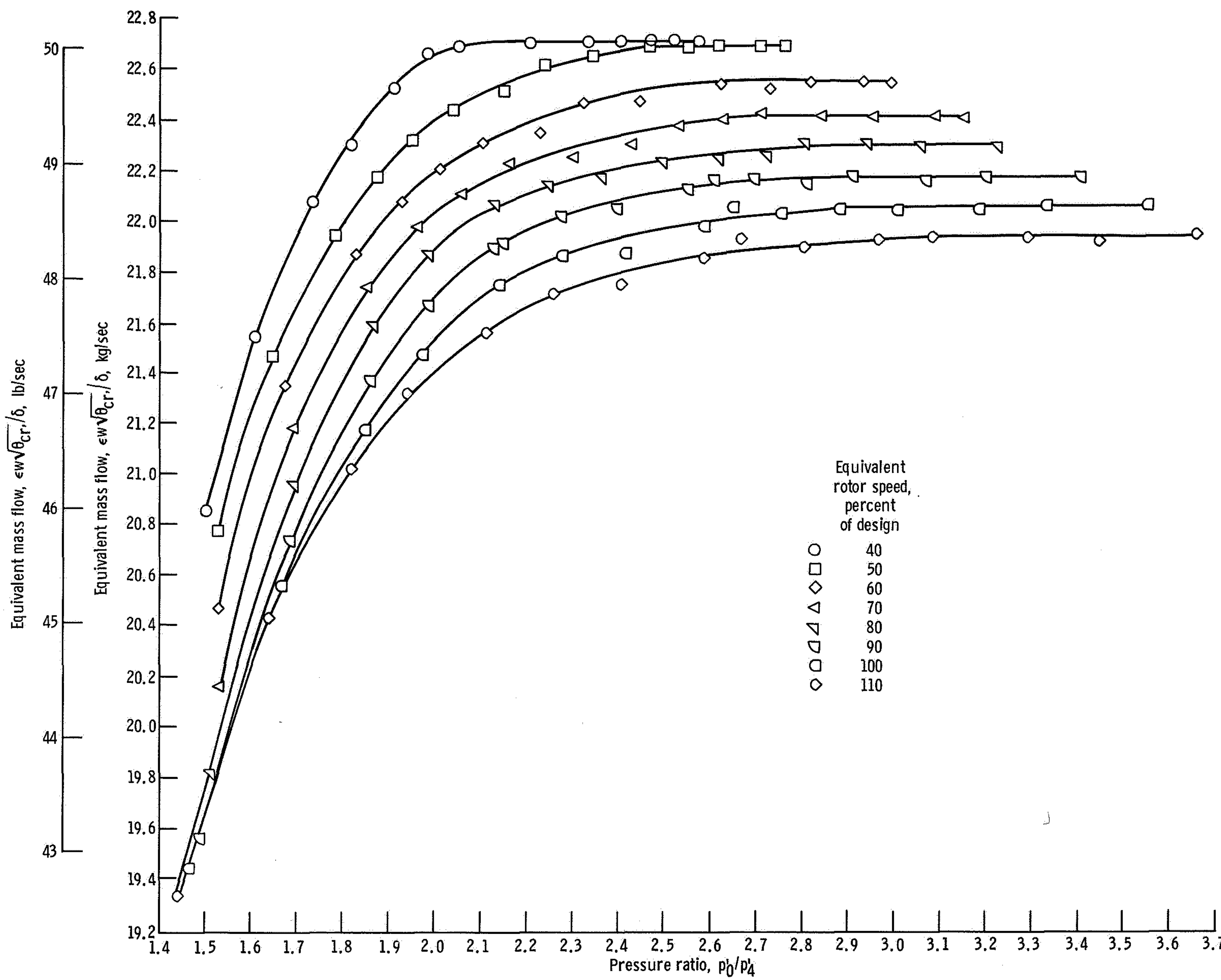


Figure 15. - Variation of equivalent mass flow with pressure ratio and equivalent speed for open stator-area turbine.

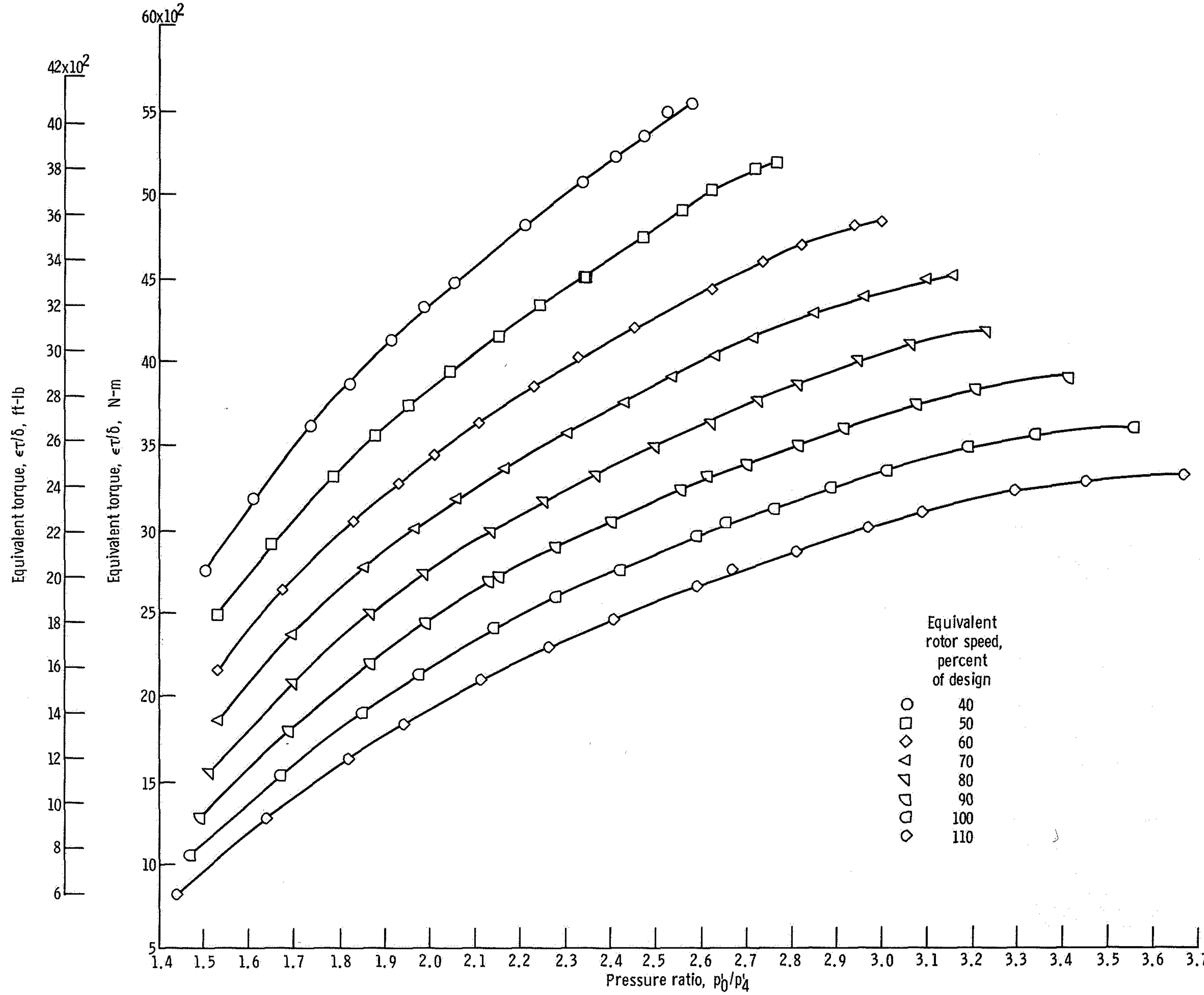


Figure 16. - Variation of equivalent torque with pressure ratio and equivalent speed for open stator-area turbine.

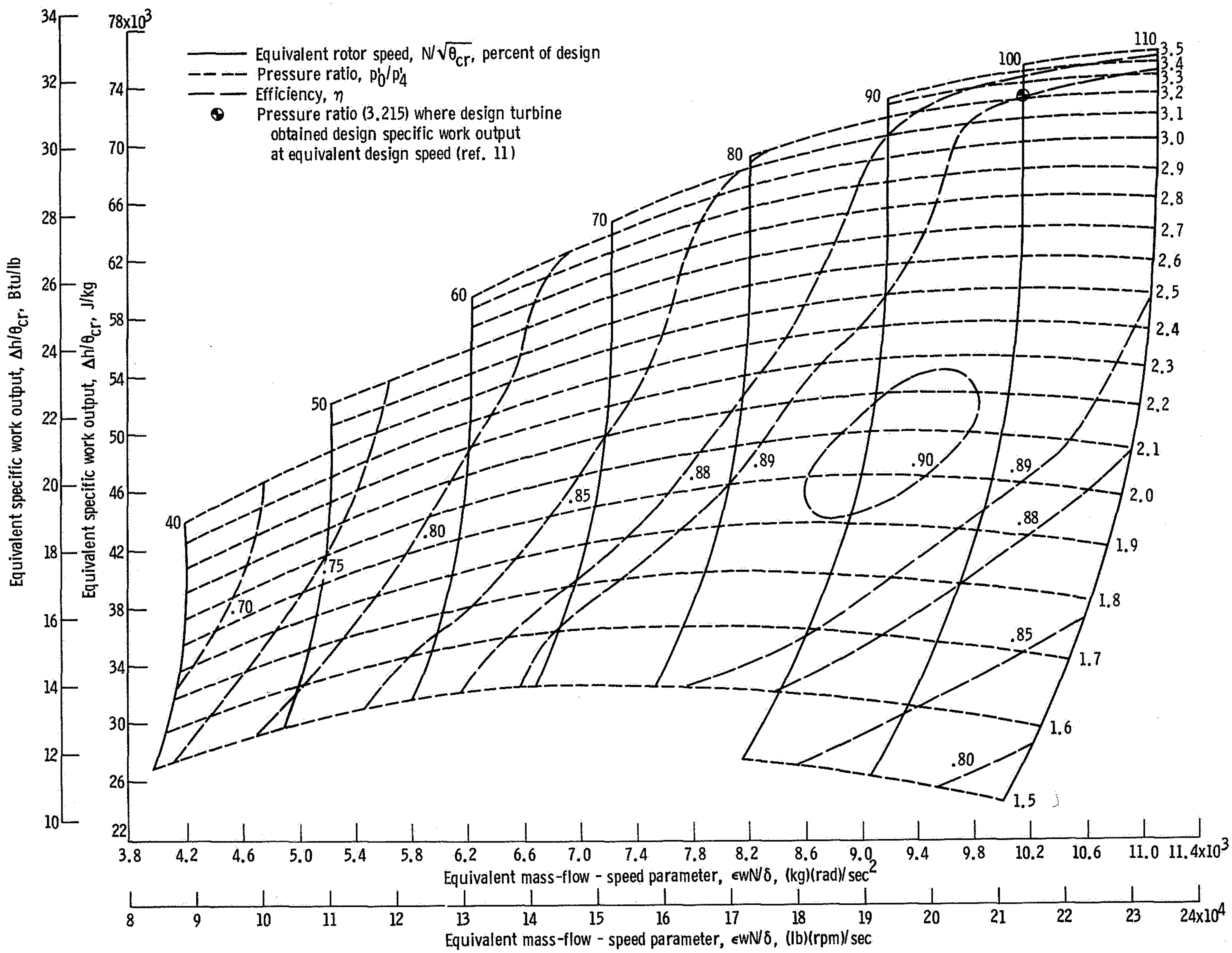


Figure 17. - Performance map for open stator-area turbine.

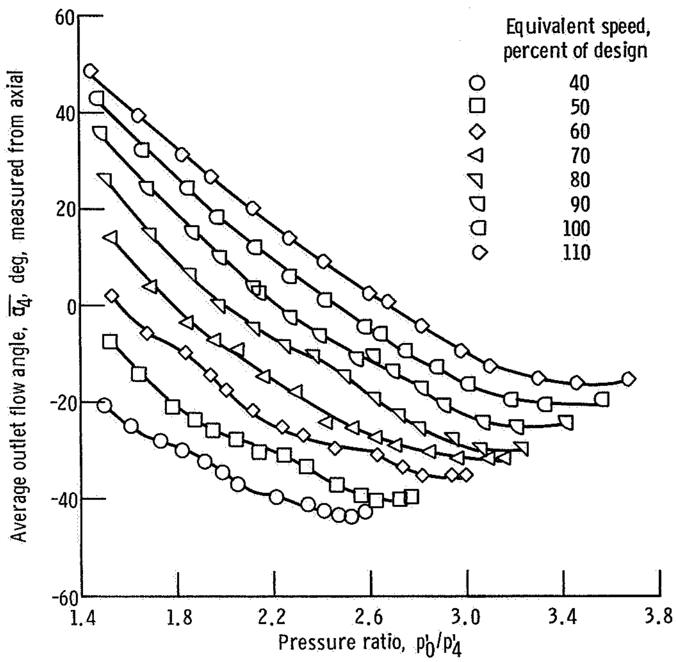


Figure 18. - Variation of average outlet flow angle with pressure ratio and equivalent speed for open stator-area turbine.

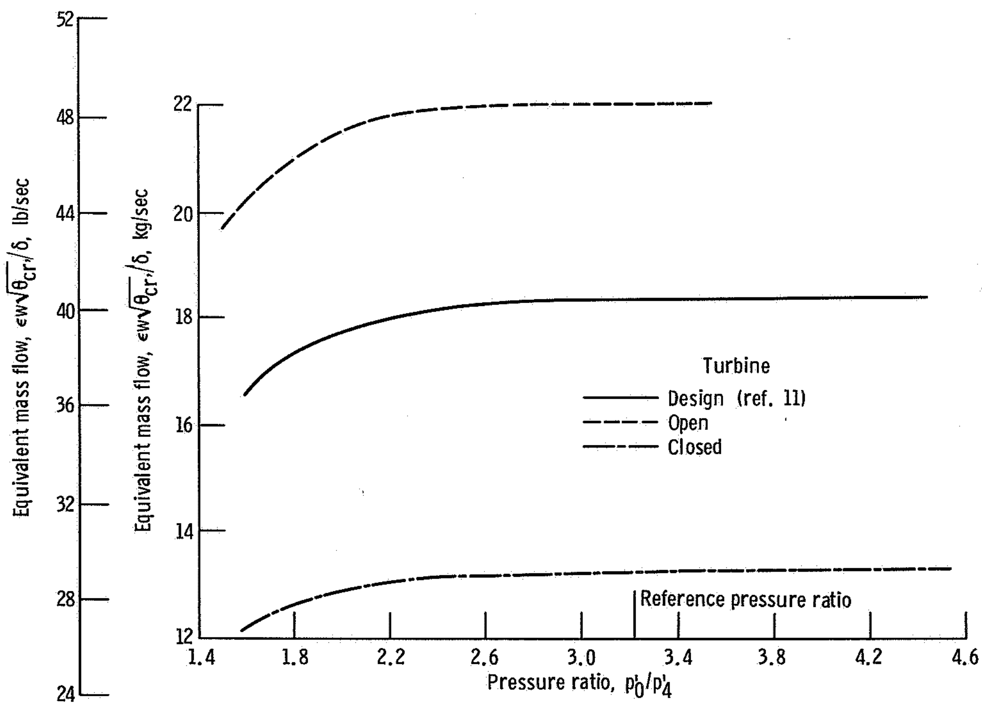


Figure 19. - Variation of equivalent mass flow with pressure ratio at equivalent design speed for three turbine configurations.

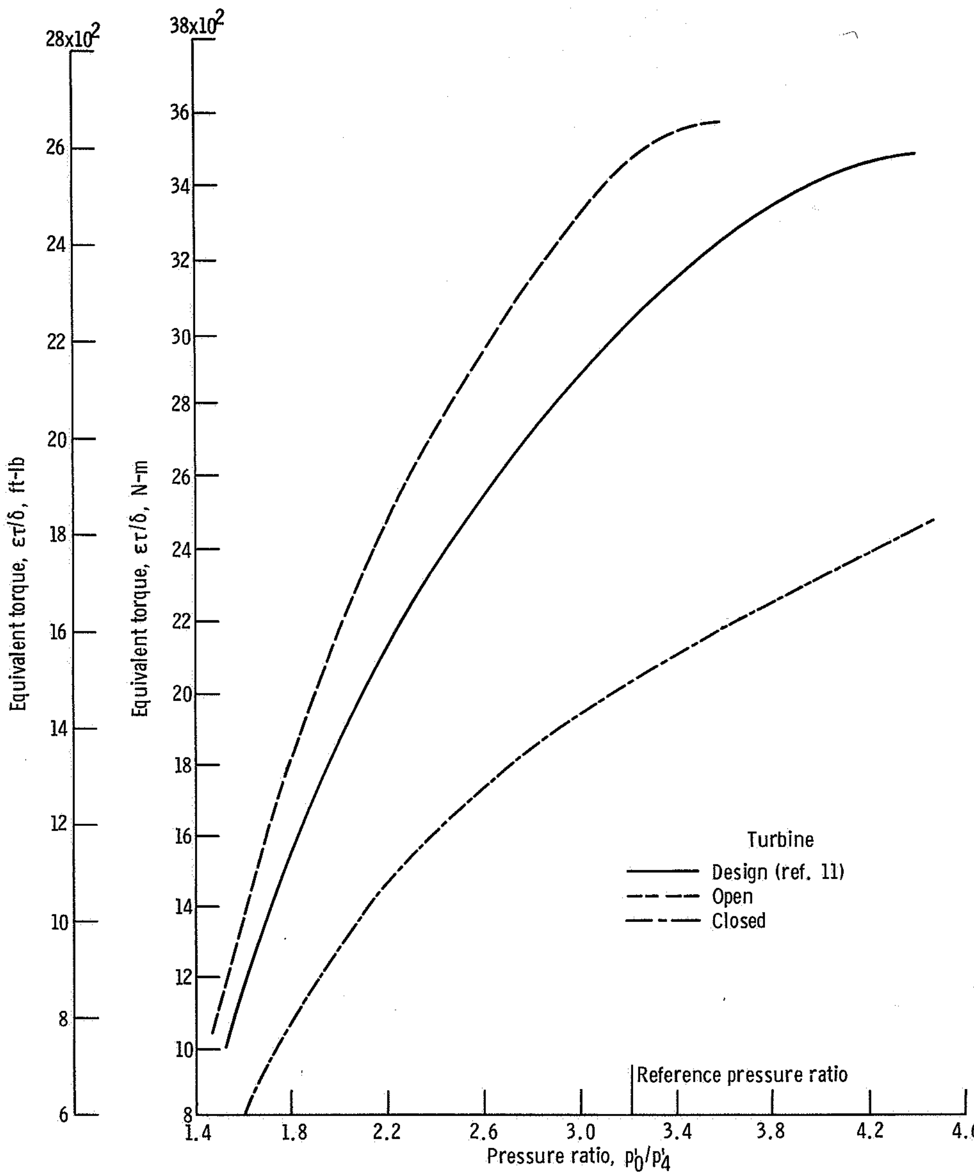


Figure 20. - Variation of equivalent torque with pressure ratio at equivalent design speed for three turbine configurations.

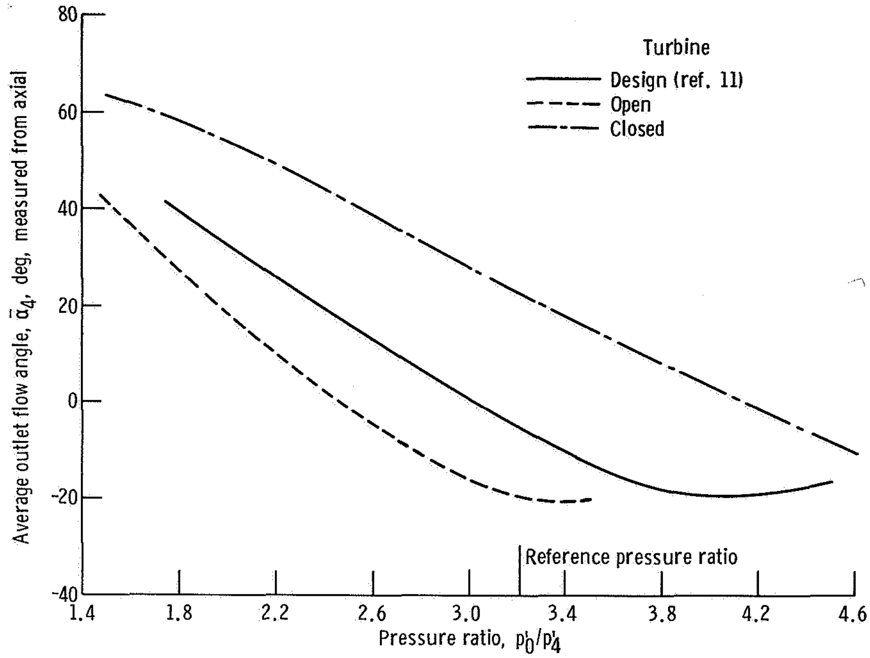


Figure 21. - Variation of average outlet flow angle with pressure ratio at design speed for three turbines.

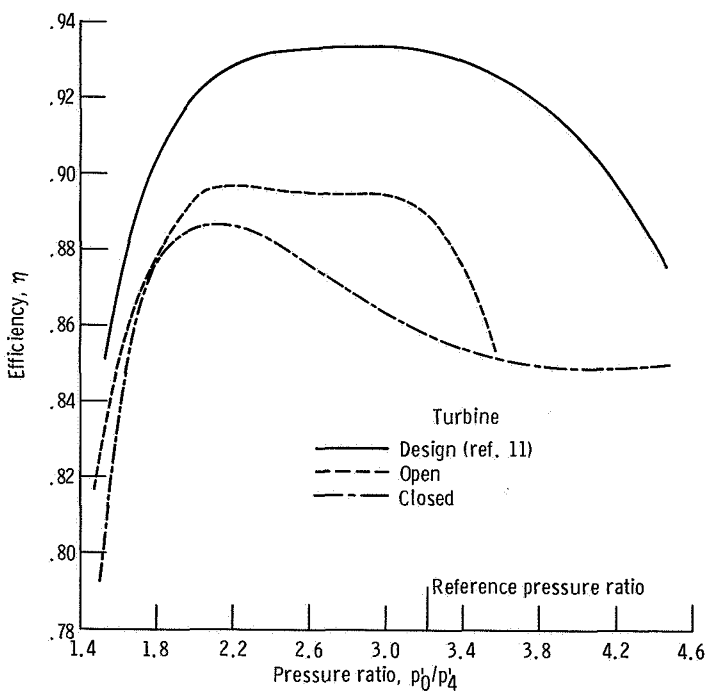


Figure 22. - Variation of efficiency with pressure ratio at equivalent design speed for three turbine configurations.

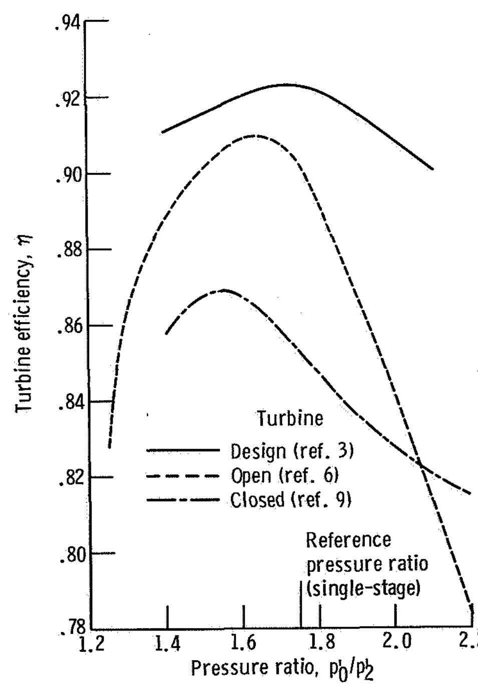
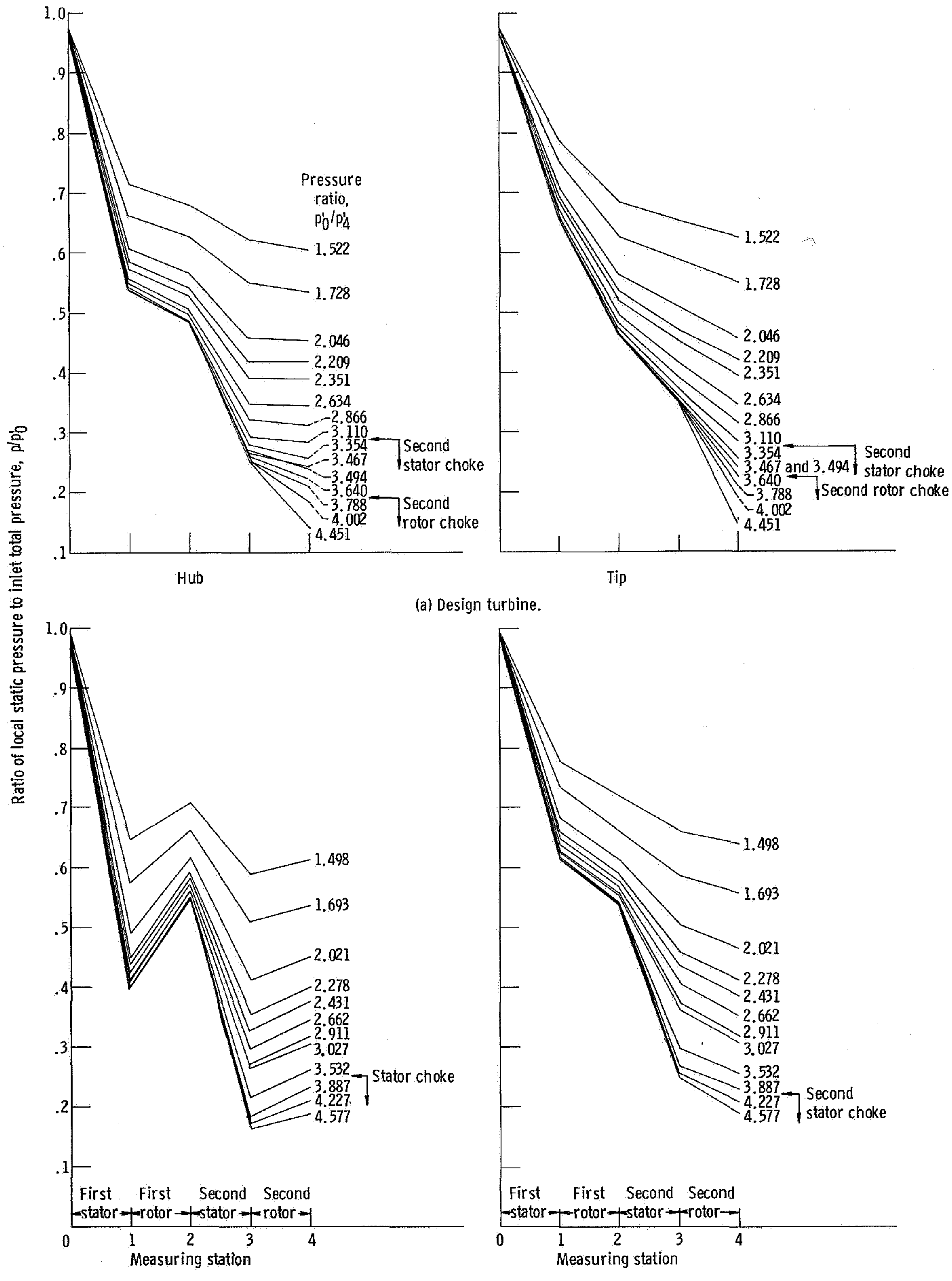


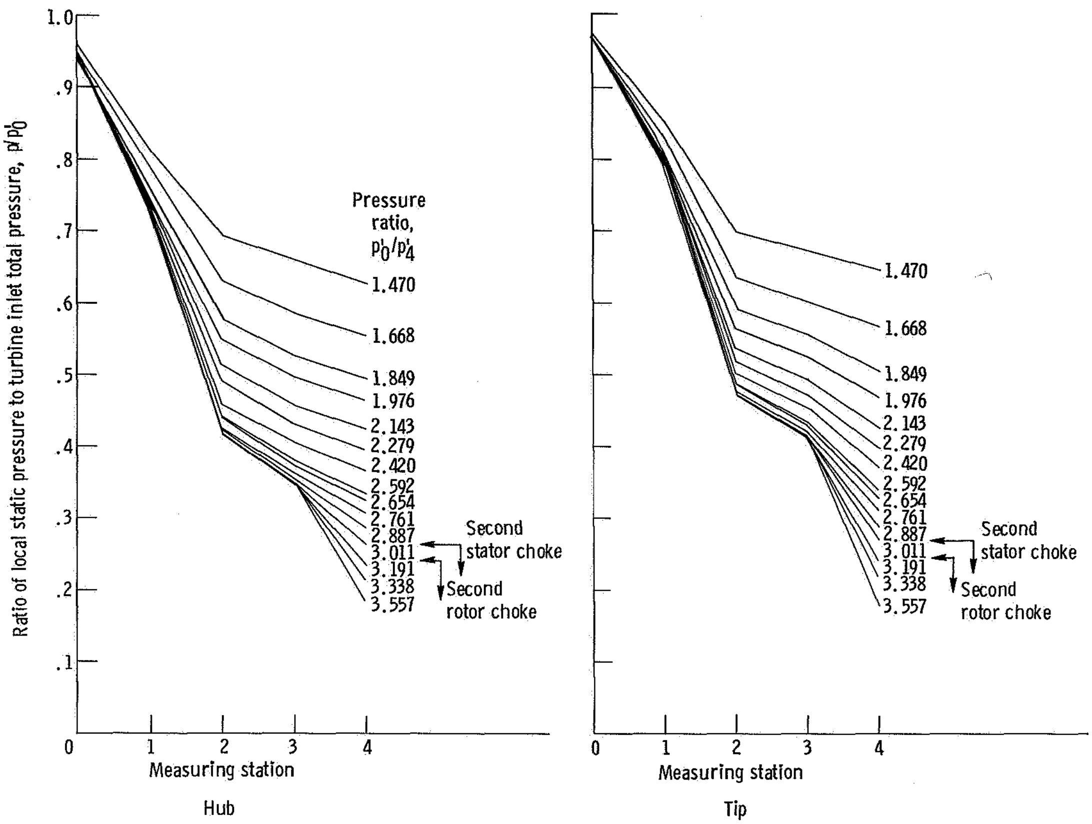
Figure 23. - Variation of single-stage turbine efficiency with pressure ratio for three turbines at equivalent design speed.



(a) Design turbine.

(b) Closed turbine.

Figure 24. - Variation of static pressure at different measuring stations and pressure ratios for equivalent design speed.



(c) Open turbine.

Figure 24. - Concluded.

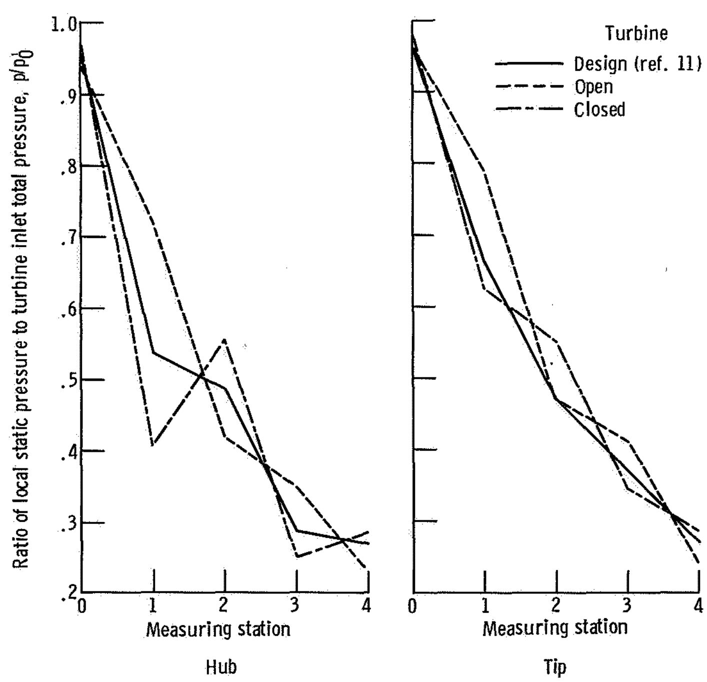


Figure 25. - Comparison of static pressure variation through turbines at equivalent design speed and at pressure ratio of 3.215.

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