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TURBINE FOR ORDNANCE TURBOJET ENGINE

II - Cold-Air Performance With Opened Stator

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16. Abstract

A single-stage axial-flow turbine was investigated to determine the effect of increased stator throat area on the performance level of a turbojet engine turbine. The stator blades were bent to increase the throat area in order to move the compressor operating point in the engine farther away from surge. Results are compared with those obtained with the as-cast stator setting.

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II - COLD-AIR PERFORMANCE WITH OPENED STATOR

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SUMMARY

An experimental investigation of a single-stage axial-flow turbine was made to determine the effect of increased stator throat area on the performance level of a turbine for a turbojet engine. The stator blades were bent to increase the throat area in order to move the compressor operating point in the engine further away from surge.

The results of the investigation indicated that an 11.4-percent increase in stator throat area resulted in an 11.2-percent increase in mass flow. The equivalent mass flow for the opened stator was 2.55 kilograms per second (5.63 lb/sec) at equivalent design speed and pressure ratio. This is about 1.6 percent higher than the design equivalent value.

The total efficiency was slightly less than 0.92 at equivalent design speed and pressure ratio for the opened stator. This is about one-half point higher than the value of efficiency obtained with the as cast stator. The improvement in turbine efficiency was attributed to reduced incidence losses and reduced rotor tip clearance losses as a result of reduced rotor reaction when the stator throat area was increased.

Results of a rotor-exit radial survey indicated a nearly uniform total pressure from hub to tip. There was considerable exit flow angle variation from hub to tip. This variation in exit flow angle with radius ratio would result in a corresponding 5.6 percent variation in local specific work.

INTRODUCTION

One of the major problems in the use of gas turbine engines for civilian single- and twin-engine light planes is the high cost of the engine. Studies (refs. 1 and 2) have been made at the Lewis Research Center for achieving lower cost gas turbine engines for this application. To achieve lower cost, a moderate turbine inlet temperature was selected for reliability as well as its use of low-cost materials for the turbine. Studies have been made for both fan-jet and turbojet engines designed for 4448 newtons (1000 lb) of sea-level static thrust and a flight Mach number of 0.65 at an altitude of 7620 meters (25 000 ft). The turbine for the fan-jet engine was designed and fabricated for a cold-air performance evaluation. The results of the performance evaluation are given in references 3 and 4.

The turbojet engine is of interest for general aviation as well as such military applications as drones, missiles, and various remotely piloted aircraft. A model of the turbine for the turbojet engine was built and tested in a cold air facility. The results of a cold-air investigation of the turbine are reported in reference 5.

The results indicated that the total efficiency was about three points higher than design and the equivalent mass flow was about 9.5 percent lower than design value. The deficiency in mass flow was due to the stator throat area being 9.4 percent smaller than that required for design mass flow. The smaller throat area resulted from manufacturing errors and from the use of cold air in this investigation.

During engine development programs stator blades are sometimes bent to change stator throat area in order to obtain design mass flow. Also engine tests on an experimental model may indicate that the compressor is operating very near surge when the engine operates near design speed and design turbine-inlet temperature. Therefore, to move the compressor operating point farther from surge, the turbine stator blades would be bent open to give an increase in stator throat area.

To determine the effect on performance of bending the blades as well as changing the the stator throat area, the stator used in the cold performance investigation of reference 5 was also modified to give a 11.4-percent larger throat area than the as-cast stator. This represents the maximum value that the stator blade throat could be increased by bending the blades.

A cold-air investigation was then made to determine the turbine performance. Turbine inlet conditions were approximately constant at a temperature of 300 K (540° R) and a pressure of 10.13 newtons per square centimeter (14.7 psia). The turbine was operated over a range of speeds from 0 to 105 percent of design and over a range of total-pressure ratio from 1.22 to 2.60.

This report presents the turbine performance and compares the performance with that obtained with the as-cast stator throat area. Test results are presented in terms of mass flow, torque, efficiency, and rotor-exit flow angle. Included are results of a radial survey of rotor-exit total pressure and flow angle obtained at equivalent design speed and near design total pressure ratio.

SYMBOLS

A area, cm^2; $in.^2$

g dimensional constant, SI = 1.0; 32.174 ft/sec^2

Δh specific work, J/g; Btu/lb

J mechanical equivalent of heat, SI = 1.0; 778.2 ft/lb/Btu

N turbine speed, rpm

p pressure, N/cm^2 abs; psia

R gas constant, J/(kg)(K); (ft-lb)/(lb)(^{O}R)

r radius, m; ft

T absolute temperature, K; ^{O}R

U blade velocity, m/sec; ft/sec

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

W relative gas velocity, m/sec; ft/sec

w mass flow, kg/sec; lb/sec

α absolute gas flow angle measured from axial direction, deg (positive values in direction of rotation)

γ ratio of specific heats

δ ratio inlet total pressure to U.S. standard sea level pressure, p_1'/p^*

ϵ function of γ used in relating parameters to those using air inlet conditions at U.S. standard sea level conditions, $(0.740/\gamma)\left[(\gamma + 1)/2\right]^{\gamma/(\gamma-1)}$

η_s static efficiency (based on inlet-total to exit-static pressure ratio)

η_t total efficiency (based on inlet-total to exit-total pressure ratio)

θ_{cr} squared ratio of critical velocity at turbine inlet to critical velocity at U.S. standard sea level air, $(V_{cr}/V_{cr}^*)^2$

ω turbine speed, rad/sec

Subscripts:

cr condition corresponding to Mach 1

eq equivalent

m mean radius

u tangential component

1 station at turbine inlet (see fig. 3)

2 station at stator exit (see fig. 3)

3 station at rotor exit (see fig. 3)

Superscripts:

' absolute total state

* U.S. standard sea-level conditions (temperature, 288.15 K (518.67° R); pressure, 10.13 N/cm^2 (14.70 psia))

TURBINE DESCRIPTION

The component test hardware and engine hardware have some physical differences at their respective operating temperatures. The turbine blading and passages are slightly larger in the engine that in the component rig due to thermal expansion. In reference 5 calculations indicated that thermal expansion would increase the flow area by 2.5 percent. Since the component tests were made with air near ambinet temperature, the equivalent mass flow in the component rig would be expected to be about 97.5 percent of the equivalent mass flow determined from hot operation conditions. The turbine total efficiency was expected to be higher in the component rig than in the engine because, in the component rig, the rotor tip clearance is smaller, the turbine inlet temperature is uniform, and the hub seal leakage is zero. Reference 5 stated that these differences could result in the component test efficiency about 3.5 percentage points higher than the assumed design value (0.88) or that obtained in the engine. The tip clearance for the component tests was 0.028 centimeter (0.011 in.) compared with 0.079 centimeter (0.031 in.) assumed for hot operation in the engine.

A meridional view of the ordnance turbine flowpath is shown in figure 2 (dimensions given are for hot operation). The stator and rotor blading information is given in table II. Stator and rotor profiles and flow passages are shown in figure 3. Both the stator and rotor have changing cross sections and twist from hub to tip. The blade surface velocities at the mean section as calcualted from the computer program of reference 7 are shown in figure 4 for the stator and rotor. These velocities were computed for the engine design conditions. The figure shows that there was no large diffusion predicted for the stator or rotor at the mean diameter.

The stator blades were bent to increase the throat area by 11.4 percent. This represents the maximum value that the stator throat could be opened by bending the blades. Bends in the blades were made as close to the inner and other walls as possible. As a result, a very small portion of the blade remained at the original setting at the inner and outer walls while the rest of the blade was at the new setting. Figure 5, a downstream view of the stator assembly, shows an abrupt change in stator angle, particularly near the outer wall. This abrupt change might have an adverse effect on the rotor efficiency. A photograph of the turbine rotor is shown in figure 6. The blade twist from hub to tip is quite apparent in this figure.

APPARATUS

The apparatus, which is the same as that used in the investigation of reference 5, consisted of the turbine, described in the preceding section, a cradled dynamometer to absorb and measure the power output of the turbine while controlling the speed, and an inlet and exhaust piping system with flow control valves. The arrangement of the apparatus is shown schematically in figure 7(a). High-pressure dry air was supplied from the laboratory air system. It passed through a filter, a mass flow measuring station, consisting of a calibrated flat-plate orifice, a remotely controlled pressure-regulating valve, an inlet plenum, and the turbine. After flowing through the turbine, the air was exhausted through a system of piping and a remotely operated valve into the laboratory low-pressure exhaust system.

A 223.7-kilowatt (300-hp) cradled dynamometer was used to absorb the turbine power, control speed, and measure torque (fig. 7(a)). The dynamometer was coupled to the turbine shafting through a gearbox, which provided relative rotative speeds between dynamometer and turbine of 1.00 to 8.25. The dynamometer and gearbox was cradled on hydrostatic oil-trunion bearings. Figure 7(b) shows the turbine installed in the test facility.

INSTRUMENTATION

The instrumentation stations are shown in figure 8. Overall performance was determined by measurements taken at stations 1 and 3. Instrumentation at the turbine inlet (station 1) measured static pressure and total temperature. Static pressures were obtained from six taps with three on the inner wall and three on the outer wall. The outer and inner taps were located opposite each other and were spaced 120° about the circumference. The temperature was measured with three thermocouple rakes, each containing three thermocouples at the area center radii of three equal annular areas.

The instrumentation at station 2, between the stator and the rotor, consisted of six static-pressure taps. These taps were spaced 120° apart with three on the outer wall and three on the inner wall as described for station 1.

At station 3, approximately one axial chord length downstream of the rotor, the instrumentation measured static pressure, total pressure, total temperature, and flow angle. The static pressure was measured with six wall taps equally spaced circumferentially (three each on the inner and outer walls). A self-alining probe was used for the measurement of total pressure, total temperature, and flow angle. There were also five total-temperature rakes, each containing three thermocouples, at a downstream station located about 45.7 centimeters (18.0 in.) from the rotor exit. These rakes were used as a check on turbine efficiency as calculated from torque, mass flow, and speed measurements.

A torque arm attached to the dynamometer stator transmitted the dynamometer torque to a commercial strain-gage load cell. The rotational speed was detected by a magnetic pickup and a shaft-mounted gear. The static pressures were measured by commercial strain-gage absolute transducers. A 200-channel data acquisition system was used to measure and record all the electrical signals.

PROCEDURE

Data were obtained at nominal inlet total flow conditions of 300 K (540° R) and 10.13 newtons per square centimeters (14.7 psia). The turbine was operated over a range of speeds from 0 to 105 percent of design and over a range of total pressure ratio from 1.22 to 2.60.

The turbine was rated on the basis of both total and static efficiency. The total pressures used in determining these efficiencies were calculated from mass flow, static pressure, total temperature, and flow angle from the following equation:

$$p' = p\left\{\frac{1}{2} + \frac{1}{2}\left[1 + \frac{2(\gamma - 1)}{\gamma}\frac{R}{g}\left(\frac{W\sqrt{T'}}{pA\cos\alpha}\right)^2\right]^{1/2}\right\}^{\gamma/(\gamma-1)}$$

In the calculation of turbine-inlet total pressure, the flow angle was assumed to be zero.

RESULTS AND DISCUSSION

The results of this investigation are given in two sections. The first section covers performance obtained with the increased stator throat area. The second section compares these performance results with those obtained with the original or as cast stator.

Performance With Increased Stator Throat Area

Mass flow. - Equivalent mass flow $\epsilon w \sqrt{\theta_{cr}}/\delta$ is shown in figure 9 as a function of total-to-total pressure ratio. An equivalent mass flow of 2.55 kilograms per second (5.63 lb/sec) was obtained at equivalent design speed and at the design total-to-total pressure ratio of 1.909. This value of mass flow is about 1.6 percent larger than design value. At this design pressure ratio and speed, the flow chokes. Because there is no observable variation of choking mass flow with speed, the stator is choked and therefore controls the flow through the turbine.

Equivalent torque. - The variation of equivalent torque $\epsilon \Gamma/\delta$ with total pressure for lines of constant speed are shown in figure 10. At an equivalent design speed and pressure ratio of 1.909, the equivalent torque was about 61.3 newton meters (45.2 ft-lb). This value is 7.2 percent higher than the design value. Since the mass flow was only 1.6 percent higher than design value, the 7.2 percent higher torque value would indicate that the turbine is performing better than the 0.88 total efficiency value assumed in the design.

The torque curves continually increase with increasing pressure ratio for all speeds investigated. All data shown are for pressure ratios below limiting loading, that is, the point above which an increase in pressure ratio results in no increase in torque output.

Overall performance. - The overall performance map is shown in figure 11 in terms of specific work output $\Delta h/\theta_{cr}$ and a mass flow speed parameter $\epsilon w \omega/\delta$ for the various equivalent speeds investigated. Lines of constant pressure ratio and efficiency contours are shown. Figure 11(a) shows performance in terms of total conditions. Turbine efficiencies of 0.70 to 0.92 were obtained for the range of pressure ratios and speeds investigated. At equivalent design speed and total-to-total pressure ratio of 1.909, the total efficiency was about 0.92. This is four points higher than the design value of 0.88. As was pointed out in the TURBINE DESCRIPTION section, differences between an engine installation and the component installation could result in about 3.5 points higher efficiency in the component setup. Therefore, the turbine is performing as would be expected. The vertical lines on the speed curves indicate choked flow. This is consistent with the results shown in figure 9.

Figure 11(b) shows the performance map based on turbine inlet-total to exit-static conditions. Static efficiency varied from about 0.40 to about 0.78. At the design operating point the static efficiency was 0.78. The difference between total and static efficiencies at the design operating point indicates that there was 14 percent of ideal enthalpy in exit kinetic energy.

Rotor exit flow angle. - Figure 12 shows the variation of rotor flow angle with pressure ratio for lines of constant speed. These flow angles were obtained at the mean radius. The figure shows that the rotor exit flow angle was about -1.3° from axial at design pressure ratio and speed. Design value for the full size turbine was -3.8°. It must be remembered that the subject turbine was designed to operate at a slightly different pressure ratio and speed. The figure also shows that for a constant pressure ratio the flow angles decrease and become negative as rotative speed is decreased. For the higher speed constant speed lines the flow angle decreases as pressure ratio is increased and then tends to level off.

Radial survey of rotor exit total pressure and angle. - Figure 13 shows the results of a radial rotor exit survey obtained at a design equivalent speed and a total pressure ratio of 1.93. Difficulty in setting the design total pressure ratio of 1.909 resulted in the survey being obtained at this near design pressure ratio. However, this difference was small and was not expected to affect the results.

Figure 13(a) shows the radial variation of total-pressure ratio with radius ratio. The main point to draw from this curve is that the total pressure was nearly uniform from hub to tip. Figure 13(b) shows the radial variation of absolute rotor exit flow angle with radius ratio. The figure shows that there was considerable variation in rotor exit flow angle from hub to tip. Negative angles indicate a positive contribution to the specific work. This variation in flow angle with radius ratio resulted in a corresponding 5.6 percent variation in local specific work.

Static-pressure distribution. - Figure 14 shows the variation of stator-exit tip static pressure with turbine inlet total to exit static pressure ratio for lines of constant speed. Stator-exit static pressures at the hub are not presented because they were considered unreliable since the static taps were located approximately 0.025 to 0.032 centimeter (0.010 to 0.013 in.) upstream of the plane tangent to the stator trailing edge. This was a result of the axial length of the hub wall being smaller than the design value. In addition, casting roughness prevented a smooth surface on which to locate the hub static taps. The curve indicates that, for a given turbine pressure ratio, rotor tip reaction decreases with decreasing rotative speed. The shape of the curve for a given speed line is what is to be expected. At the high pressure ratios the tip static pressure is constant, indicating that the rotor is choked in the tip region. After the rotor unchokes, the stator tip pressure ratio increases because of the decreasing mass flow.

Figure 15 shows the variation of turbine total pressure ratio with turbine inlet-total to exit-static pressure ratio for lines of constant speed. This figure was included only as a working or reference curve for the reader.

Performance Comparison With Original Stator Setting

A comparison of mass flow obtained at equivalent design total pressure ratio and speed indicated that an 11.4 percent increase in stator throat area resulted in an 11.2 percent increase in mass flow. This would therefore indicate that there was no significant change in loss between the turbines with the as-cast and the opened stator.

Figure 16 is the performance map of the turbine, based on total-to-total pressure ratio, as presented in reference 5. Comparison of total efficiency at design point operation (fig. 11) shows that there was about a one point improvement in efficiency when the stator throat area was increased by bending the stator blades. This improvement in performance may be attributed to reduced rotor incidence losses and a reduction in rotor reaction when the stator throat area was increased. Rotor tip clearance losses, for a given tip clearance, would be expected to decrease with a decrease in rotor reaction. These results, therefore, indicate that this method of increasing the stator throat area by bending the blades did not increase turbine losses.

SUMMARY OF RESULTS

An experimental investigation was made to determine the effect of bending stator blades as well as changing the stator throat area on performance of a turbojet engine turbine. Results of the cold-air investigation may be summarized as follows:

1. An equivalent mass flow of 2.55 kilograms per second (5.63 lb/sec) was obtained at equivalent design speed and total pressure ratio for the opened stator. This is about 1.6 percent higher than design value.

2. An 11.4-percent increase in stator throat area resulted in an 11.2-percent increase in mass flow. Thus, no significant change in loss occurred for the turbines with the as-cast and the opened stator.

3. The total efficiency was nearly 0.92 at equivalent design speed and pressure ratio. This is about one point higher than the value obtained with the original stator.

4. The improvement in turbine performance when the stator blades were bent open may be attributed to reduced incidence losses and reduced rotor tip clearance losses as a result of reduced rotor reaction.

5. Results of a rotor exit survey of total pressure and flow angle taken at equivalent design speed and at a pressure ratio of 1.93 indicated nearly uniform total pressure from hub to tip. There was considerable variation in rotor exit flow angle from hub to

tip. This variation in flow angle with radius ratio resulted in a corresponding 5.6-percent variation in local specific work.

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National Aeronautics and Space Administration,
and
U.S. Army Air Mobility R&D Laboratory,
Cleveland, Ohio, January 11, 1974,
501-24.

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7. Katsanis, Theodore: Fortran Program for Calculating Transonic Velocities on a Blade-to-Blade Stream Surface of a Turbomachine. NASA TN D-5427, 1969.

TABLE I. - TURBINE DESIGN CONDITIONS

Parameter	Hot conditions	Equivalent conditions
Inlet total temperature, T_1', K; $^{\circ}R$	1089; 1960	288.2; 518.7
Inlet total pressure, p_1', N/cm^2; psia	26.3; 38.2	10.1; 14.7
Mass flow, w, kg/sec; lb/sec	3.28; 7.23	2.51; 5.53
Rotative speed, N, rad/sec; rpm	3633.8; 34 700	1892.8; 18 075
Specific work, Δh, J/g; Btu/lb	159.3; 68.4	43.2; 18.6
Rotor tip diameter, cm; in.	24.70; 9.72	24.70; 9.72
Energy function, $\Delta h/T'$, J/kg K; Btu/(lb)($^{\circ}R$)	0.1463; 0.0349	0.1500; 0.0358
Speed function, $U_m/\sqrt{T_1'}$, $m/sec\sqrt{K}$; $ft/sec\sqrt{^{\circ}R}$	11.26; 27.53	11.41; 27.89
Work factor, $\Delta V_u/U$	1.153	1.153
Total efficiency, η_t	0.88	0.88
Total to total pressure ratio, P_1'/P_3'	1.862	1.909

TABLE II. - TURBINE PHYSICAL PARAMETERS

Blading		Radius		Axial chord		Solidity	Throat		Number of blades
		cm	in.	cm	in.		cm	in.	
Stator	Hub	8.36	3.24	1.615	0.636	1.076	0.510	0.201	35
	Mean	10.35	4.08	1.900	.748	1.022	.750	.295	
	Tip	12.35	6.24	2.185	.860	.985	1.053	.415	
Rotor inlet	Hub	8.31	3.27	2.090	0.823	2.268	0.535	0.211	57
	Mean	10.35	4.08	1.757	.692	1.540	.650	.256	
	Tip	12.35	6.24	1.425	.561	1.047	.720	.283	

	Hub	Mean	Tip
Radius, cm (in.), at -			
Station 1	11.18 (4.40)	13.84 (5.45)	16.51 (6.50)
Station 2	11.18 (4.40)	13.84 (5.45)	16.51 (6.50)
Station 3	10.72 (4.22)	13.61 (5.36)	16.53 (6.51)
α_2, deg	70.0	66.0	61.4
β_2, deg	52.4	23.2	-15.7
α_3, deg	-5.8	-3.8	-3.1
β_3, deg	-53.3	-54.6	-58.0

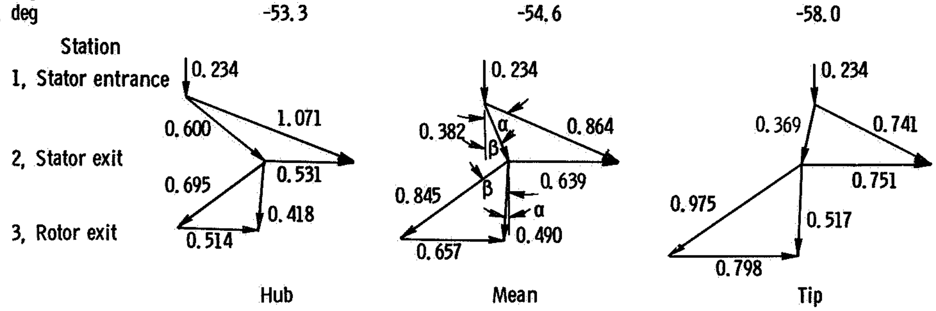


Figure 1. - Design velocity diagrams (given in terms of Mach number).

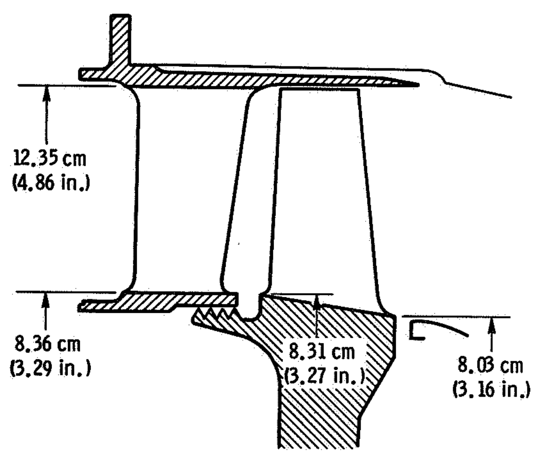


Figure 2. - Turbine meridional flowpath. Radii shown are for hot-engine conditions.

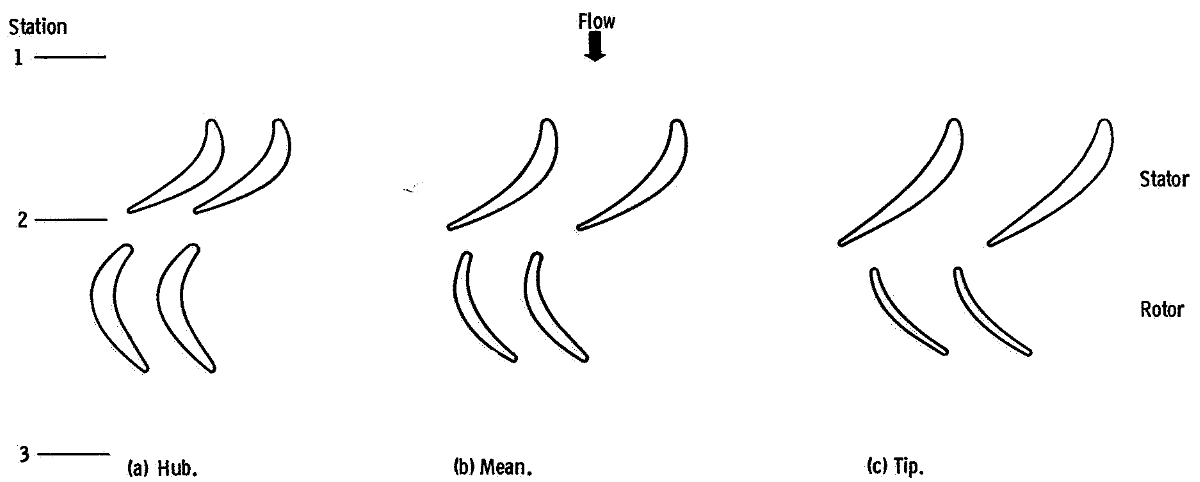


Figure 3. - Stator- and rotor-blade passages and profiles.

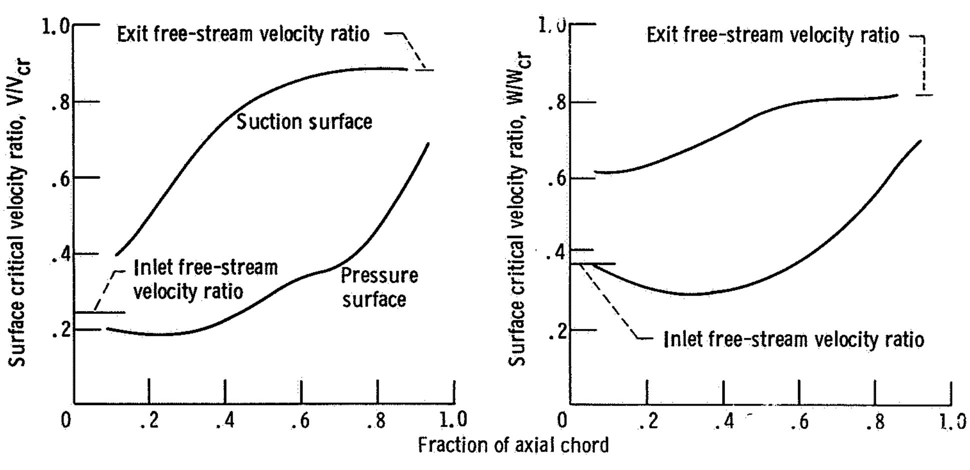


Figure 4. - Blade surface velocity distribution at mean diameter.

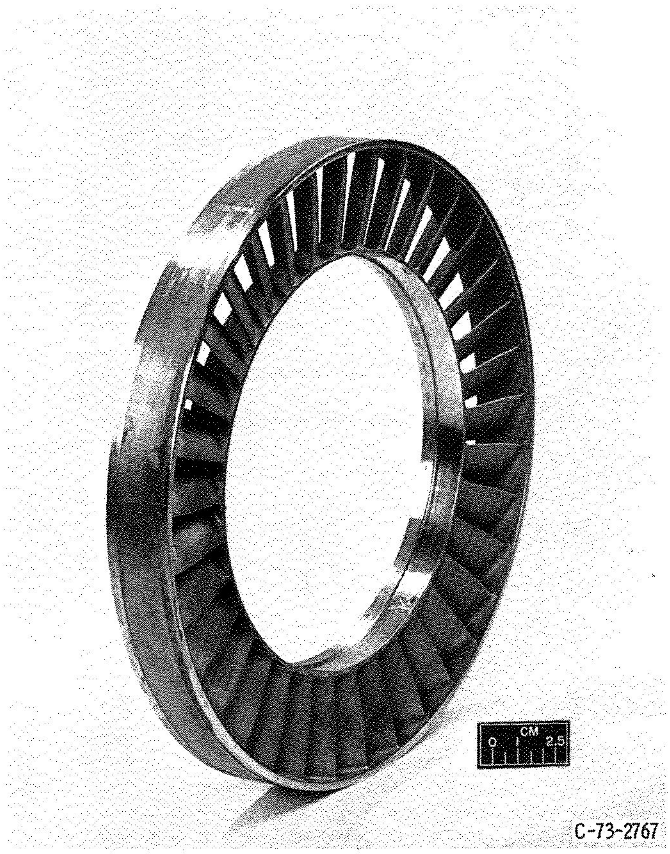


Figure 5. - Downstream view of opened stator.

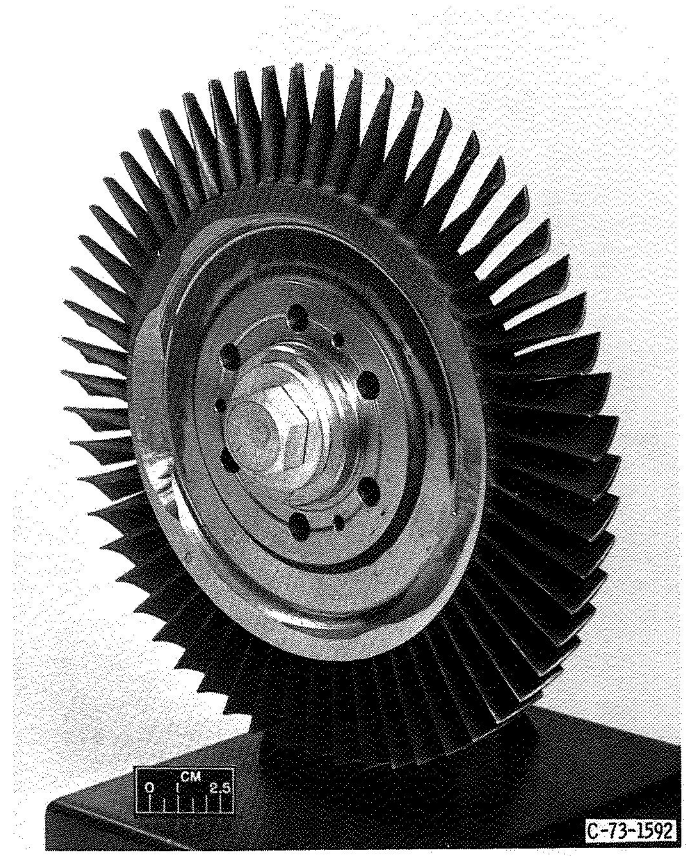
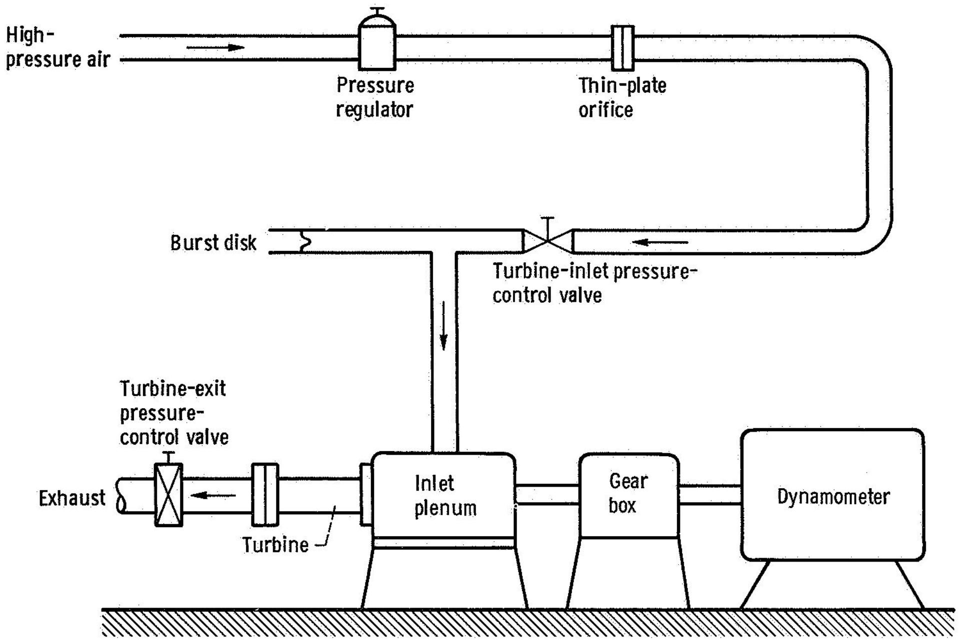
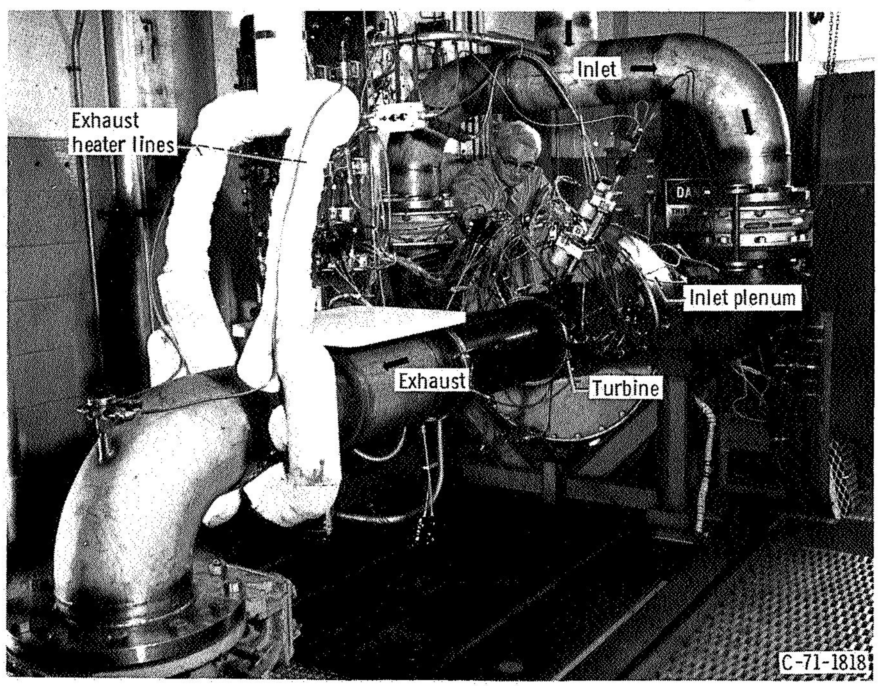


Figure 6. - Turbine rotor.



(a) Arrangement of piping and test equipment.



(b) Turbine test apparatus.

Figure 7. - Experiment equipment.

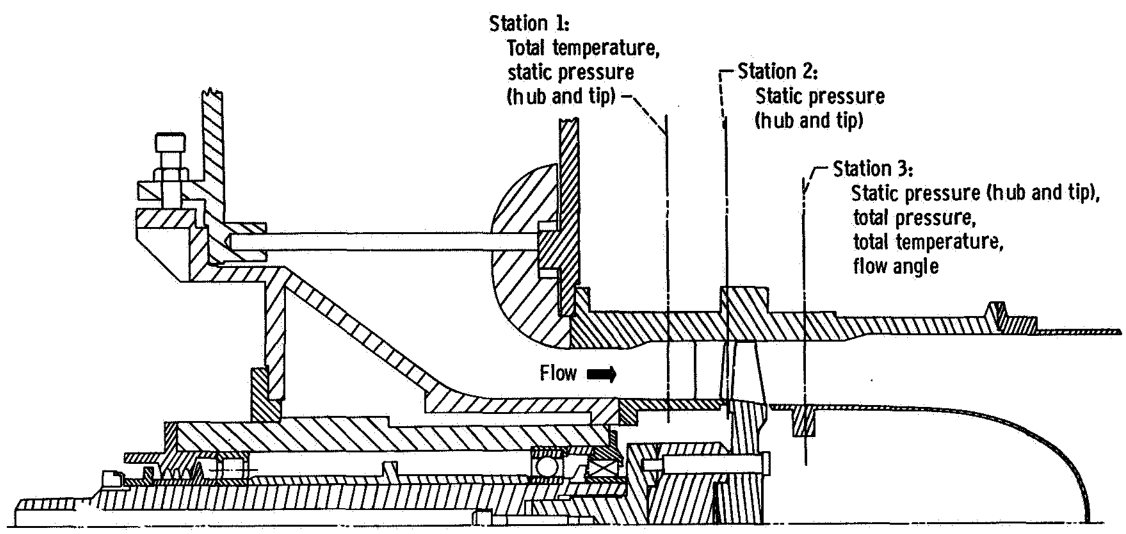


Figure 8. - Schematic of turbine for component tests.

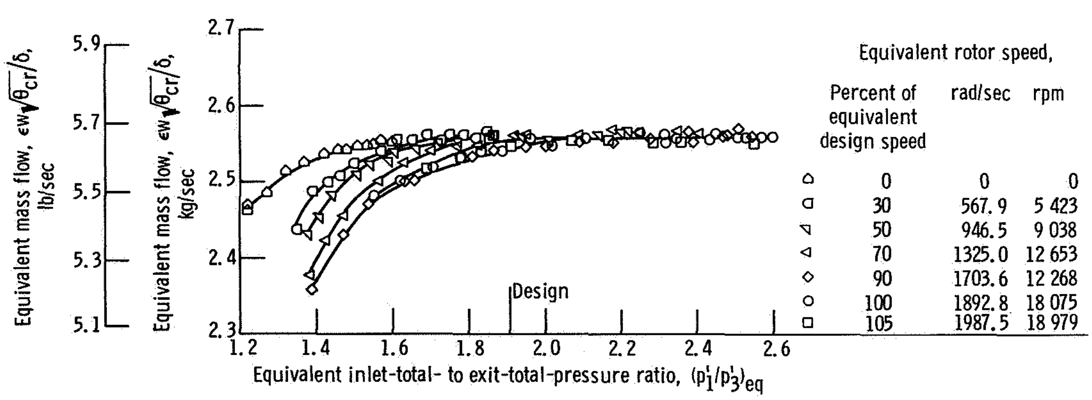


Figure 9. - Variation at mass flow with pressure ratio and speed.

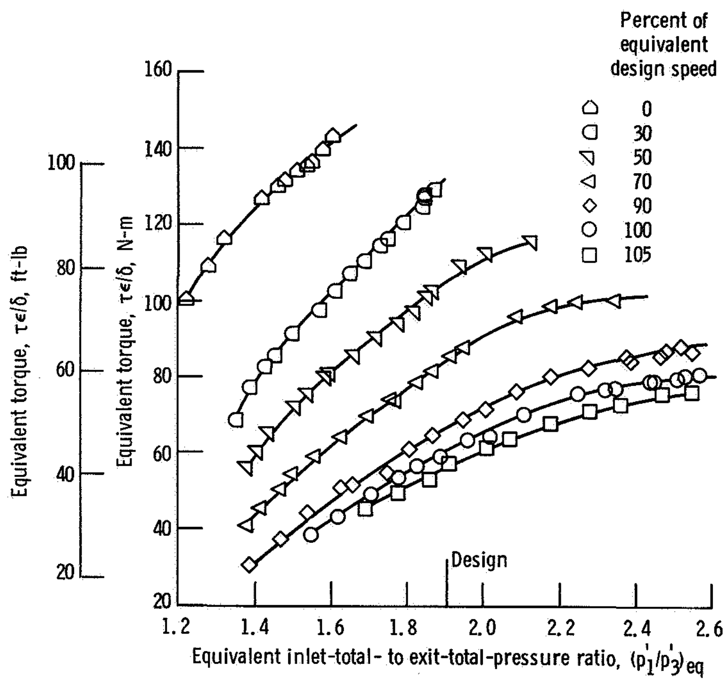


Figure 10. - Variation of torque with pressure ratio and speed.

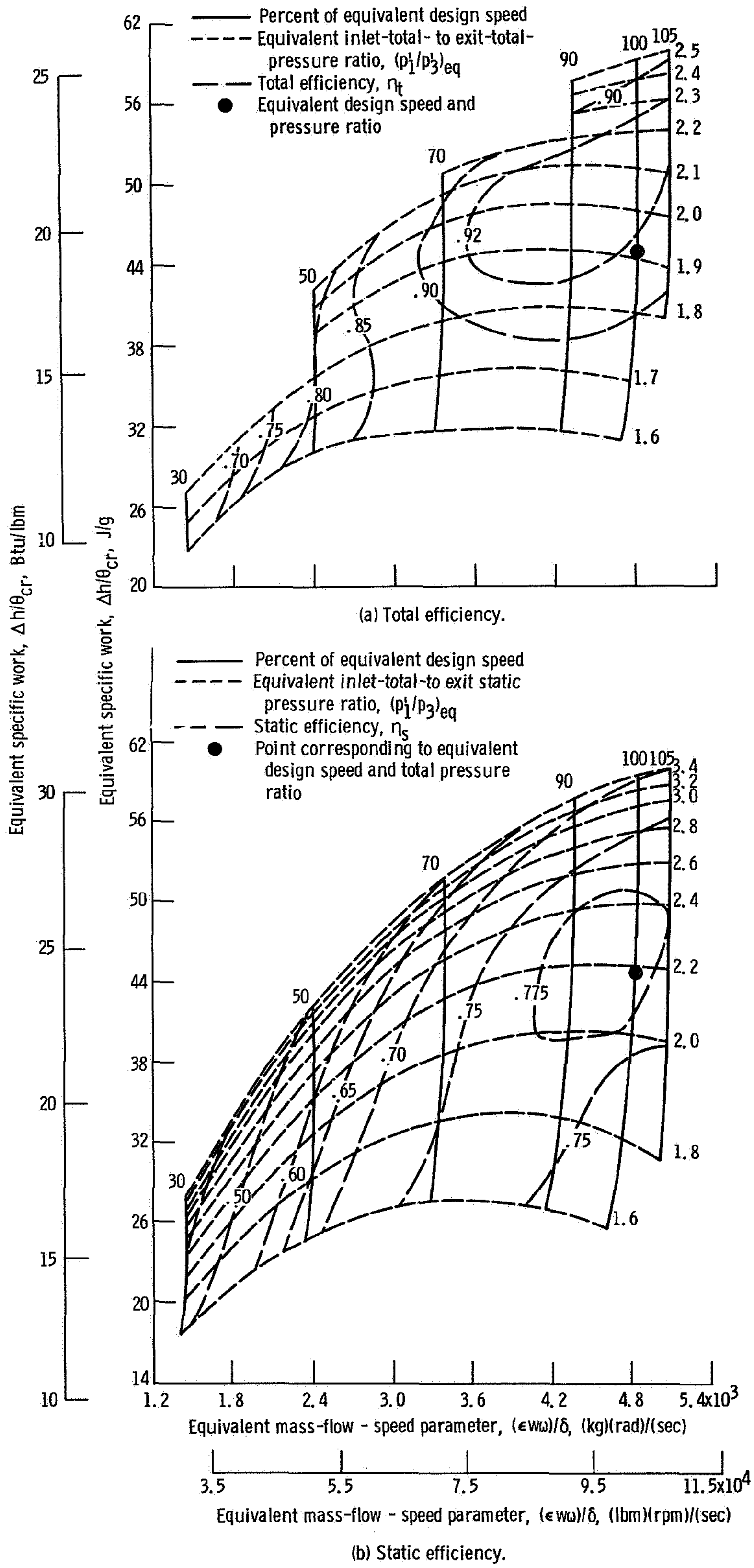


Figure 11. - Overall turbine performance map for opened stator configuration.

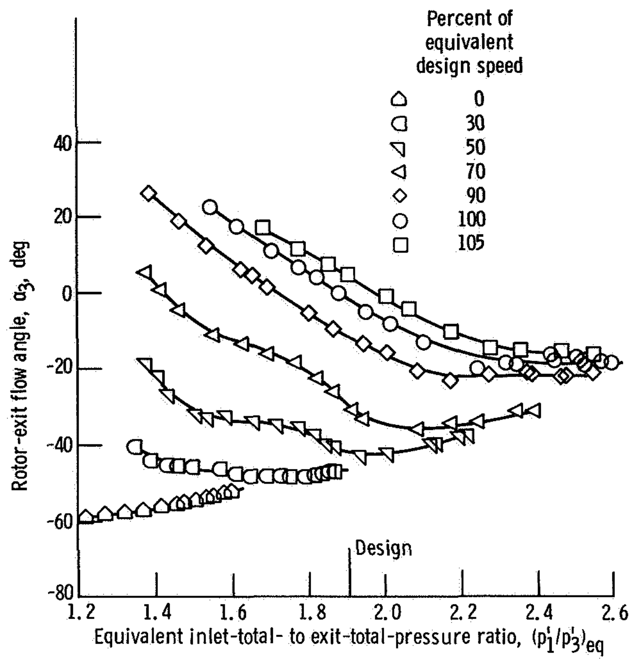


Figure 12. - Variation of rotor-exit flow angle (at mean radius) with pressure ratio and speed.

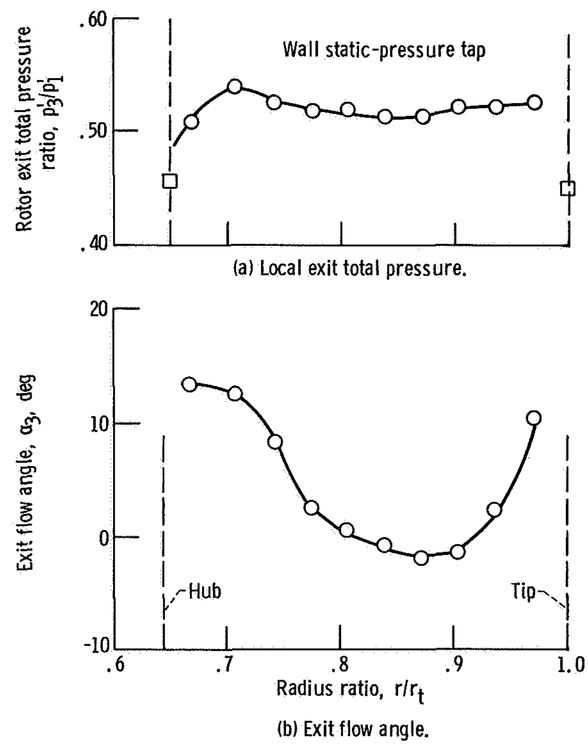


Figure 13. - Variation of rotor-exit total pressure and flow angle with radius ratio at equivalent design speed and design inlet-total- to exit-static-pressure ratio.

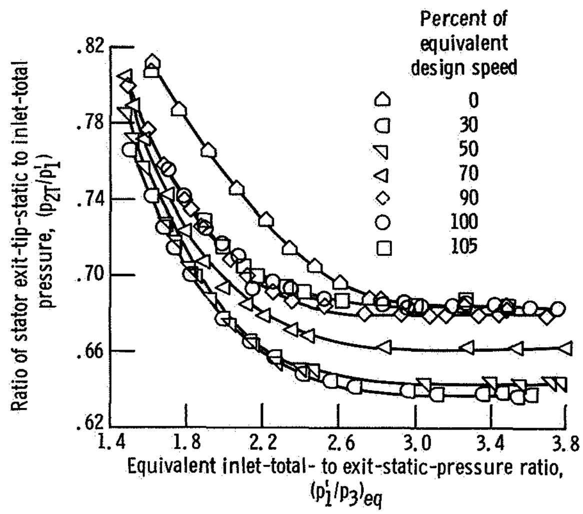


Figure 14. - Variation of stator exit-tip-static pressure with pressure ratio and speed.

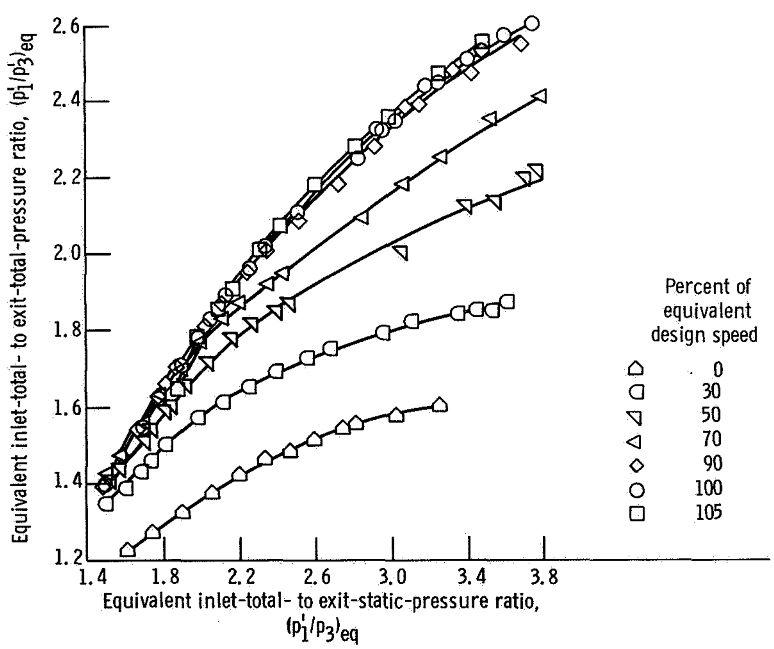


Figure 15. - Variation of total-pressure ratio with static-pressure ratio and speed.

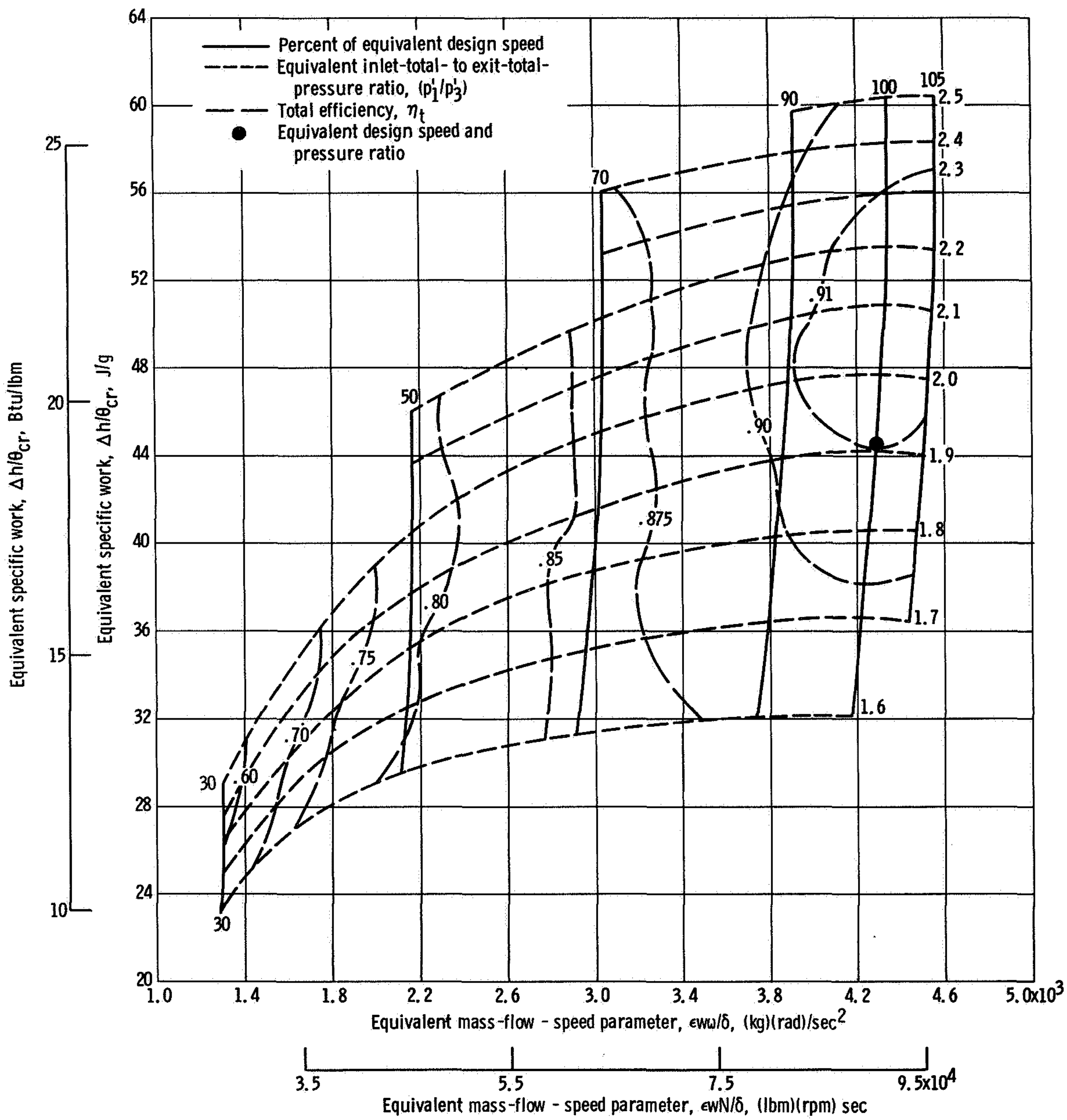


Figure 16. - Overall turbine performance map with as-cast stator.

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