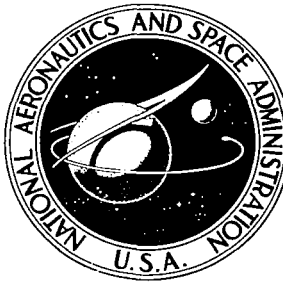


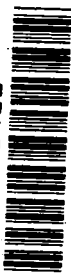
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EXPERIMENTAL INVESTIGATION OF ADVANCED CONCEPTS TO INCREASE TURBINE BLADE LOADING

IV. Performance Evaluation of Plain Rotor Blade With Plow Type Vortex Generators

by H. G. Lueders

Prepared by

GENERAL MOTORS

Indianapolis, Ind.

for Lewis Research Center



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Prepared under Contract No. NAS 3-7902 by
ALLISON DIVISION
GENERAL MOTORS
Indianapolis, Ind.

for Lewis Research Center

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FOREWORD

The research described herein, which was conducted by the Allison Division of General Motors, was performed under NASA Contract NAS3-7902. The work was done under the project management of Mr. Edward L. Warren, Airbreathing Engines Division, NASA-Lewis Research Center, with Mr. Richard J. Roelke, Fluid System Components Division, NASA-Lewis Research Center, as research consultant. The report was originally issued as Allison Division, General Motors EDR 4909, Volume IV, October 1968.

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EXPERIMENTAL INVESTIGATION OF ADVANCED CONCEPTS TO INCREASE TURBINE BLADE LOADING

IV. PERFORMANCE EVALUATION OF PLAIN ROTOR BLADE WITH PLOW TYPE VORTEX GENERATORS

by H. G. Lueders

Allison Division, General Motors

SUMMARY

The overall performance of a single-stage turbine designed with a rotor blade suction surface diffusion factor of 0.3 and negative hub reaction was investigated over a range of equivalent speeds and expansion ratios. Plow type vortex generators were attached to the forward portion of the blade suction surface to prevent flow separation. Rotor exit surveys were taken of total pressure, flow angle, temperature, and hot-wire data. A number of rotor blades had static pressure orifices located at the mean-line section to record blade surface static pressures during rotation. The results of this investigation are compared with the performance of the same blade without the vortex generators attached.

The overall performance characteristics of the turbine with vortex generators were very similar to the turbine without vortex generators (plain blade turbine). The only noticeable differences occurred at the higher pressure ratios. In this operating region, the vortex generator turbine developed slightly more work and higher efficiency than the plain blade turbine. These differences, however, were minor. The total efficiencies at design equivalent speed and expansion ratio were 88.6 and 88.4 percent for the vortex generator and plain blade turbine, respectively.

The turbine loss patterns, as determined from exit surveys, were very similar for the two tests. The higher losses measured in the hub region of the plain blade turbine were not significantly affected by the addition of vortex generators.

INTRODUCTION

The analysis and optimization of propulsion systems have always involved a balance or trade between (1) turbine efficiency and (2) turbine size and weight reduction. Generally, the reduction of turbine diameter, solidity, and/or stage reaction results in a smaller and/or lighter turbine but at some sacrifice in efficiency because of losses associated with increased blade loading. If the size and weight reduction can be accomplished without a loss in efficiency, considerable gains are available to the overall propulsion system.

NASA has initiated a program to investigate concepts to increase turbine blade loading without the associated losses currently encountered. The first phase of this test program consisted of the testing of a plain rotor blade which had high suction surface diffusion. The performance of this blade, which formed the program base line, is presented in reference 1. For comparison, the following three different blading concepts are being evaluated:

- Vortex generators, often called boundary layer trip devices
- Tandem airfoils
- Jet flap

The analyses and design of these concepts are presented in reference 2. The tangential jet blade described in reference 2 is not scheduled for testing at this time because of vibrational problems. The performance of the tandem blade is presented in reference 3. The performance of the plain blade with the triangular plow vortex generators attached is discussed herein. The plain blade with the vortex generators attached is shown in Figure 1. The plain blade with corotating vortex generators described in reference 2 is not scheduled for testing because of the results presented herein and the test results of a similar configuration conducted under NASA Contract NAS3-9404.

The blades used for the tests presented herein are the same blades used in reference 1 except for triangular plow vortex generators attached to the blade suction surface at an axial distance of 0.64 inch from the leading edge. The principle of the triangular plow vortex generator is to shed a pair of counterrotating vortices which induce high energy air from the free stream into the low energy boundary layer to produce a reenergization of this viscous layer of fluid. This reenergization results in a thinning of the boundary layer to retard or prevent flow separation.

This report presents the test results of a single-stage turbine incorporating a plain blade with triangular plow vortex generators. A comparison with the plain blade results of reference 1 is included.

Performance data were taken from 70 to 110 percent of design equivalent speed in increments of 10 percent over a range of expansion ratios from 1.4 to a level near limiting loading.

Rotor exit surveys were conducted at the design equivalent speed and expansion ratio. Circumferential traverses with a combination total pressure, temperature, and yaw angle probe were made at constant radii to map the flow characteristics at the rotor trailing edge. A hot-wire anemometer survey was also made at the rotor trailing edge to provide additional insight to the rotor exit flow characteristics. Measurements were taken of the blade surface static pressure at the mean line for design equivalent speed at expansion ratios below design value, at design, and above design. All testing was conducted while operating the test rig with inlet conditions of approximately 2.9 atmospheres absolute pressure and 660°R temperature.

SYMBOLS

E	specific work output, Btu/lb
$\dot{m}$	mass flow rate, lb/sec
N	rotational speed, rpm
P	pressure, psia
T	temperature, °R
u_r	blade tangential velocity, ft/sec
V	absolute gas velocity, ft/sec
W	relative gas velocity, ft/sec
Γ	torque, ft-lb
δ_o	ratio of inlet air total pressure to standard sea level conditions
ϵ	function of γ defined as $\frac{\gamma^*}{\gamma} \frac{\left(\frac{\gamma+1}{2}\right)^{\gamma/(\gamma-1)}}{\left(\frac{\gamma^*+1}{2}\right)^{\gamma^*/(\gamma^*-1)}}$
η_T	adiabatic efficiency defined as the ratio of turbine work based on torque, primary weight flow, and speed measurements to ideal work based on inlet total temperature, and inlet and outlet total pressure both defined as sum of static pressure plus pressure corresponding to gas velocity
η_t	adiabatic efficiency defined as the ratio of turbine work based upon measured inlet and exit total temperature to ideal work based on measured inlet total temperature and pressure and measured exit total pressure
Θ_{cr}	squared ratio of critical velocity at turbine inlet temperature to critical velocity at standard sea level temperature
ν	ratio of blade speed to isentropic gas velocity based on inlet total temperature and pressure and exit static pressure $(u_r)m/V'$

Subscripts

0	station at stator inlet (all stations shown in Figure 4)
3	station at free-stream conditions between stator and rotor
5	station at outlet of rotor just downstream of trailing edge
6	station downstream from turbine
m	mean radius
rel	relative condition
st	static condition
T	stagnation or total conditions
t	tip radius

Superscripts

'	ideal or isentropic
*	standard conditions

APPARATUS AND INSTRUMENTATION

The analyses and design of the turbine test rig and the blading are discussed in detail in reference 2. This reference includes the tabulated coordinates for the stator and rotor blades. The turbine has a constant hub diameter of 21.0 inches and a constant tip diameter of 30.0 inches. The unit has 40 stator blades and 76 rotor blades. The overall design point equivalent characteristics are:

- Equivalent specific work output, $\frac{E}{\theta_{cr}}$, Btu/lb 20.0
- Equivalent weight flow, $\frac{\dot{m}\sqrt{\theta_{cr}}}{\delta_o}$, lb/sec 45.51
- Equivalent blade tip speed, $\frac{(ur)_t}{\sqrt{\theta_{cr}}}$, ft/sec 610.0

The design velocity diagrams are shown in Figure 2 and the section profiles are shown in Figure 3. The dimensions of the vortex generators and their location on the blades are shown in Figure 4. A photograph of the plain blade with the triangular plow vortex generators is shown in Figure 1. The radial clearance between the blade tips on the turbine casing is approximately 0.030 inch.

The apparatus used in this investigation consisted of a single-stage cold air turbine test rig, suitable housings to provide uniform inlet flow conditions, and a dynamometer to absorb and measure the turbine power output. Airflow is supplied to the test rig by a separate air compressor facility. The air is supplied at approximately 3 atmospheres pressure and a temperature of approximately 700°R. The facility air can be heated or cooled by heat exchangers. The facility air enters the plenum chamber through two diametrically opposed pipes. The air passes through the turbine blading and discharges into the facility exhaust system. The inlet pressure is controlled by the separate air compressor supply and/or by a throttle valve in the inlet supply line. The turbine expansion ratio is controlled by a throttle valve in the exhaust system duct. The turbine exhaust may either be discharged to the atmosphere or directed to an evacuator facility to provide below ambient exhaust conditions.

The turbine test rig instrumentation is described in detail in reference 2. The airflow is measured using a Bailey adjustable orifice calibrated with an ASME flow nozzle. The turbine power output is absorbed by two Dynamatic dry-gap eddy current brakes. The torque of each dynamometer is measured separately by a dual output strain gage load cell connected in tension to the dynamometer torque arm.

Measurements of total temperature and total pressure were made at stations 0 and 6 (Figure 5). Turbine inlet temperature was measured with 20 iron-constantan thermocouples arranged five to a rake. The sensing elements were located on centers of equal annular areas, and the rakes were spaced 90 degrees apart. Four Kiel type total pressure probes, also located at the inlet, were used to establish the desired inlet total pressure.

The turbine exit measuring station (station 6, Figure 5) was instrumented with five combination total pressure, total temperature, self-aligning flow angle probes. The sensing elements of the five combination probes were located at the center of five equal annular areas.

Four static pressure taps on both the inner and outer walls were located around the annulus at stations 0, 3, 5, and 6. Stator outlet (station 3) static pressure taps were centrally located on the projected stator flow passage immediately downstream of the stator blade trailing edge.

Two surveys were made approximately 1/8-inch downstream of the rotor blade trailing edge (station 5, Figure 5). Total pressure, total temperature, and flow angle were measured at five radii from hub to tip for a circumferential arc of 22 degrees. The measurements were taken concurrently with a single combination probe. The second survey was made with a hot-film anemometer probe consisting of a radially mounted 0.002-inch diameter quartz rod with a very thin platinum plating over the quartz. The maximum frequency response of the sensing element is approximately 40,000 cycles per second. This probe was installed in the same mounting pad as the pressure, temperature, flow angle probe and was positioned circumferentially to avoid the stator wakes determined from the total pressure survey presented in reference 1.

The blade surface static pressure measurements were obtained by means of a Scanni-Valve pressure switch located on the axis of rotation of the rear turbine rotor shaft. The pressure switch outlet line is directed through a rotating-to-stationary seal after which the signal is measured by means of a transducer. The pressure switch is indexed electrically through a slip ring assembly.

CALCULATION PROCEDURE

OVERALL TURBINE PERFORMANCE

The turbine performance was rated on the basis of two expansion ratios defined as (1) the ratio of the inlet total pressure to rotor discharge static pressure and (2) the ratio of inlet total pressure to rotor discharge total pressure. The inlet total pressure at station 0 was calculated from continuity using the average of the 20 measured total temperatures, the average of the hub and tip static pressures, the mass flow rate, and the inlet annulus area. The flow was assumed to be axial. The exit total pressure at station 6 was also calculated from continuity using the mass flow rate, the annulus area, the average of the hub and tip static pressures, the average flow angle, and the total temperature. The total temperature was calculated from the enthalpy drop which in turn was based on the measured airflow, torque, and speed.

The efficiencies were calculated as a ratio of the actual enthalpy drop as obtained from torque, mass flow rate, and rotor speed measurements to the ideal enthalpy drop as obtained from the inlet total temperature and the associated calculated expansion ratio.

ROTOR EXIT SURVEY

The performance of the turbine as described by a rotor exit survey at the design point condition is based on measured expansion ratio, inlet temperature, and exit temperature. The measured expansion ratio is based on the average total pressure indicated by the four inlet Kiel probes and the exit total pressure measured by the survey probe. The inlet total temperature is the average temperature of the 20 inlet thermocouples; the exit total temperature is measured by the thermocouple on the survey probe. These thermocouples were corrected for Mach number based on a linear variation of hub and tip static pressure and the measured total pressure. The isentropic work of the turbine is based on the measured inlet temperature and measured total pressure ratio. The actual work is the difference of the enthalpies associated with the measured inlet and exit temperatures. The efficiency at each station in the survey is the ratio of the actual work to the isentropic work.

The measured absolute flow angle and an assumed linear variation of static pressure from hub to tip were used to determine the velocity diagrams at the rotor exit. The stator exit whirl was calculated as a function of circumferential position at each radial depth from the work based on measured temperatures, the blade speed, and the rotor exit whirl. These stator exit whirls and rotor exit relative angles were then used to determine the rotor relative total pressure loss from the hot-wire survey.

HOT-WIRE SURVEY

The procedure followed for the reduction of the basic hot-wire electronic signals is described in reference 4. This data reduction procedure produces the variation of absolute gas velocity and total temperature with blade passage circumferential position at a given radial-circumferential position. The reduction of these variables to a loss of rotor relative total pressure is described in reference 5. The rotor exit relative discharge angle and the stator exit whirl velocity were determined from the rotor exit survey calculation rather than using the design value. The stator exit absolute total pressure was based on the measurement of the stator loss survey (reference 1).

EXPERIMENTAL RESULTS

TURBINE OVERALL PERFORMANCE

The overall performance of the turbine with triangular plow vortex generators on the plain blade is shown in Figure 6 as a composite map. This map presents the equivalent shaft work, E/ϕ_{cr} , as a function of the equivalent flow-speed parameter, $\dot{m} \sqrt{\phi_{cr}} \epsilon / \delta_o$, for lines of constant total-to-total expansion ratio, P_{T0}/P_{T6} , and equivalent rotor speed, $N/\sqrt{\phi_{cr}}$. Contours of constant total efficiency, η_T , are also included. The design equivalent work and speed are indicated by point A on the map and the design equivalent work and flow-speed parameter are shown by point B. Comparison of the flow-speed parameter of these two points indicates the turbine flow capacity was approximately 5 percent greater than design. This is comparable to the flow capacity of the turbine with the plain blade installed. The excess flow in both tests was caused by an increase in stator area that resulted from a welding operation of the stator (reference 1).

The variation of equivalent torque, $\Gamma \epsilon / \delta_o$, and equivalent weight flow, $\dot{m} \sqrt{\phi_{cr}} \epsilon / \delta_o$, with overall total-to-total expansion ratio and equivalent speed, are shown in Figures 7 and 8, respectively. The torque characteristic of Figure 7 indicates that limiting loading had not yet been achieved at the high expansion ratio condition. The flow characteristics of Figure 8 indicate that at design equivalent speed and pressure ratio the measured weight flow was 47.7 lb/sec which is 4.8 percent higher than the design value of 45.5 lb/sec. The turbine choked at an equivalent flow of 47.85 lb/sec and an expansion ratio of 2.4. There is only a minor influence of equivalent speed on the flow capacity of the turbine.

A plot of total-to-total efficiency is presented in Figure 9 as a function of blade jet-speed ratio for lines of equivalent speed. At the design blade jet-speed ratio of 0.438, the efficiency is 88.6 percent compared with the design value of 84 percent. The design value of efficiency is premised on the design analysis indicating separation from the entire span of the rotor suction surface. Test results, discussed herein, show that separation probably occurred only in the hub region, causing the rotor to operate at a higher level of reaction than design. This produces a higher than design rotor static pressure ratio which reduces the potential for separation and, therefore, improves the efficiency from the design value.

An insight into the degree of reaction achieved is shown in Figure 10 showing the hub and tip static pressure variation as a function of overall total-to-total expansion ratio at the design speed condition. At the turbine design expansion ratio of 2.1, the rotor hub and tip exit static pressure was 0.407 compared with the design value of 0.41 and the stator exit hub static pressure

was 0.393 compared with the design value of 0.363. This indicates that the rotor hub was near impulse conditions; it was designed for negative hub reaction. This condition is caused by the increase in stator throat area. Another contributing factor, however, is the reduced rotor hub effective flow area which is discussed in the following subsection.

The blade surface static pressure measurements taken at the rotor blade mean line for design equivalent speed and expansion ratios of 1.86, 2.09 (design), and 2.33 are shown in Figure 11. The data between 1.86 and 2.09 expansion ratios show that an increasing expansion ratio caused a significant reduction in the blade surface static pressure. However, increasing the expansion ratio from 2.09 to 2.33 resulted in only those pressures on the suction surface downstream of the 1.3-inch axial location to be significantly reduced in magnitude. This indicates that rotor choking was probably occurring in this region. The average static pressure variation of Figure 12 also verifies that the rotor was unchoked at an expansion ratio of 1.86, was very close to choking at 2.1, and was definitely choked at 2.3. This probable area of choking is upstream of the trailing edge flow orthogonal.

TURBINE LOCAL EFFICIENCY SURVEY

Circumferential traverses of a combination total pressure, temperature, and yaw angle probe were made at constant radii to map the flow characteristics at the rotor trailing edge. These surveys yield the circumferential variations of temperature ratio, $(T_{T0} - T_{T5})/T_{T0}$, total pressure ratio, P_{T0}/P_{T5} , blade exit absolute flow angle, and local efficiency. Typical examples of these survey traces are shown in Figures 13 through 16. From these surveys, a contour map of turbine efficiency was constructed based on the locally measured total-to-total expansion ratio. This contour map is shown in Figure 17. The efficiency contours shown are caused by the flow losses incurred in both the stator and rotor. The stator wakes and the cores of high efficiency between these wakes in the upper two-thirds of the annulus can readily be seen. The large areas of low efficiency at the hub, however, indicate additional significant rotor losses in this region. Additional information about the nature of these losses is presented in the following subsection.

ROTOR EXIT HOT-WIRE SURVEY

A hot-wire anemometer survey was conducted at the rotor trailing edge. The turbine exit total pressure survey was used to define the radial-circumferential path of the hot-wire probe to avoid the stator wakes. The results of this survey are presented as contours of rotor relative total pressure ratio, P_{T5rel}/P_{T3rel} , in Figure 18. The rotor trailing edge positions, as indicated by an electronic measuring device, are shown together with the pressure and suction sides of the blade wakes. The blade wakes, relatively efficient flow between these wakes, and secondary flow cores near the blade tips are discernible in Figure 18. In the region of the blade hub, areas of both high and

low relative total pressure exist, although the low pressure areas are dominant. The areas of high pressure are adjacent to the pressure surface, while the areas of low pressure appear to emanate from the suction surface indicating probable flow separation in this region. The hub region of the blade is most susceptible to separation. Separation in this region reduces the effective flow area at the hub and would change the turbine reaction characteristics in a manner consistent with that shown in Figure 10.

The design analysis predicted that flow separation would not be encountered over the entire blade surface using the triangular plow vortex generator. An examination of the local efficiency survey and the hot-wire survey indicates that separation at the rotor hub probably did occur. This forced a flow shift radially outward which improved the general flow conditions on the outer portion of the rotor blade.

COMPARISON OF PERFORMANCE OF PLAIN BLADE WITH AND WITHOUT PLOW TYPE VORTEX GENERATORS

A comparison of the overall performance maps of the plain blade with and without plow type vortex generators is shown in Figures 6 and 19. The overall characteristics of the two maps are very similar. The prime difference in performance occurs at expansion rates of approximately 1.9 and above. The turbine with vortex generators exhibits slightly better performance in this area at all speeds investigated. At an expansion ratio of 2.3 and 70 percent design speed, the turbine with vortex generators produced an equivalent work of 21.1 Btu/lb compared to 20.9 Btu/lb for the turbine without vortex generators. At 110 percent design speed and the same expansion ratio, the equivalent work was 23.35 Btu/lb for the turbine with vortex generators compared to 23.2 Btu/lb for the turbine without vortex generators. An examination of Figure 9, showing the characteristics of total-to-total efficiency with equivalent speed and blade jet-speed ratio, indicates a peak efficiency of 89.6 percent for the turbine with vortex generators. This compares to a peak efficiency in excess of 90 percent (Figure 20) for the turbine without vortex generators.

The equivalent flow characteristics of the plain blade with and without vortex generators are also similar as shown in Figures 8 and 21. Only minor differences exist which are within the accuracy of allowable measurement. The tendency of a similar characteristic is also shown by a comparison of the variation of torque with expansion ratio and equivalent speed. The torque characteristics for the two turbines are shown in Figure 22. The data symbols have been removed for clarity. The solid lines are for the turbine without vortex generators and the dashed lines are for the turbine with vortex generators. The primary difference between the two turbines occurs at the higher expansion ratios. The turbine with the vortex generators had slightly higher torque at a given expansion ratio than the turbine without vortex generators.

A small difference between the two turbines is shown in the rotor exit static pressure variation with overall total expansion ratio. This distribution at design equivalent speed is shown in Figure 23 for the turbine with and without vortex generators. Both units exhibit nearly the same characteristic to an expansion ratio of 2.3. Beyond this level of expansion ratio, the turbine with vortex generators exhibits a larger decrease in rotor exit static pressure than the blade without vortex generators. This indicates that in this range of expansion ratios, the turbine with vortex generators has the highest exit absolute Mach number. This characteristic, in conjunction with the previously mentioned higher torque, equal flow characteristic, and equal stator exit (station 3) hub and tip static pressures, combines to form overall averaged exit flow conditions which indicate improved turning characteristics for

the turbine with vortex generators in the region of design speed and higher than design expansion ratio. This characteristic is also substantiated by a comparison of the average rotor exit absolute flow angle of the two units shown in Figure 24 as a function of the total-to-total expansion ratio and equivalent speed. The solid line in this figure represents the turbine with vortex generators and the dashed line is the turbine without vortex generators. The data symbols have been omitted for clarity. At 100 percent speed and expansion ratios greater than 2.0, there is a pronounced increase in swirl angle with increasing expansion ratio for the turbine with vortex generators. This characteristic is in contradiction to the data shown for the rotor exit survey at station 5 shown in Figure 15 and in Figure 21 of reference 1. Comparison of these figures shows that the plain blade has 5 degrees more turning capacity than the unit with vortex generators at design speed and expansion ratio. This difference is partly caused by the differences in radial depths of the two surveys. The average exit absolute flow angle obtained by graphical integration of the rotor exit survey traces is shown in Figure 25 for both turbines as a function of radius. These data show the plain rotor blade as having about 2.5 degrees more turning in all areas except the tip region. No apparent cause is known for the discrepancy of angle measurements between stations 5 and 6.

Similarity of the characteristics of both turbines is also evident in the efficiency surveys shown in Figure 17 for the turbine with vortex generators and in Figure 26 for the turbine without vortex generators. Both units exhibit peak efficiencies in excess of 94 percent. The relative shape and magnitude of the efficiency contours for both units are similar and no significant improvement in the hub region efficiency can be attributed to the vortex generators.

SUMMARY OF RESULTS

The overall performance of a single-stage turbine designed with a rotor blade suction surface diffusion parameter of 0.3 and a negative hub reaction was investigated over a range of equivalent speeds and expansion ratios. The rotor blade incorporated triangular plow vortex generators attached to the forward portion of the blade suction surface to prevent flow separation. The performance results of the turbine are compared to the same set of blading without the vortex generators attached. The following results were obtained.

- The general overall performance characteristics were very similar for both turbines. The primary difference occurred at the higher expansion ratios where the turbine with vortex generators developed slightly more work and, therefore, slightly higher efficiency.
- The total efficiency of the turbine with the plow type vortex generators attached was calculated to be 88.6 percent compared to 88.4 percent for the same blade without the vortex generators. These conditions are compared at the design equivalent speed and blade-jet speed ratio.
- The static pressure distribution through both turbines was very similar to an expansion ratio of 2.3. At higher expansion ratios, the turbine with vortex generators exhibited progressively lower rotor exit static pressure than the same blading without vortex generators. This is the result of the turbine with vortex generators having slightly better rotor turning characteristics in the operating region of high rotor exit relative Mach numbers.
- A survey taken at the rotor trailing edge showed the general magnitude and shape of the efficiency contours to be essentially the same for the blading with and without plow type vortex generators. No noticeable improvement in efficiency at the rotor hub was observed for the turbine with vortex generators.

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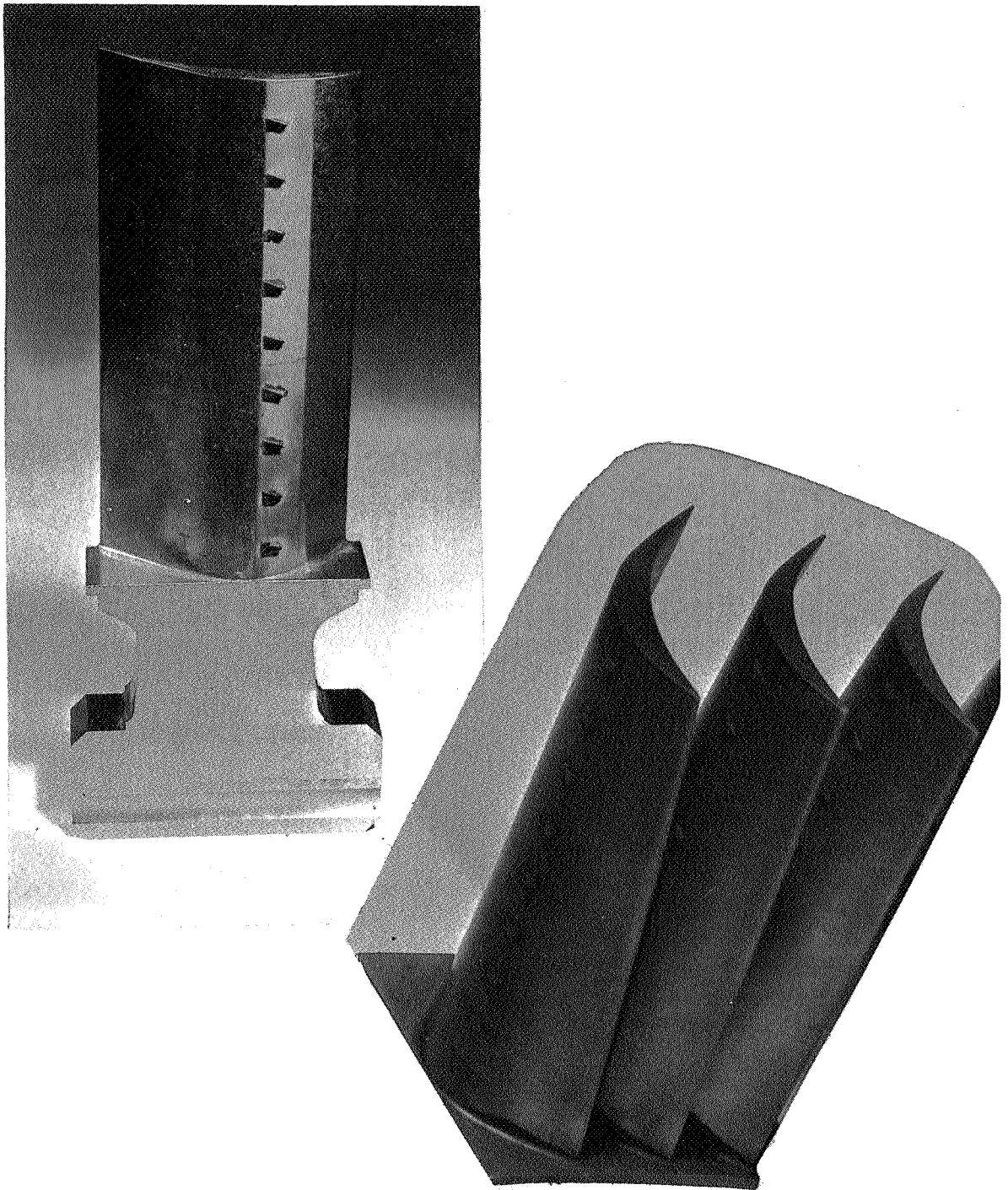
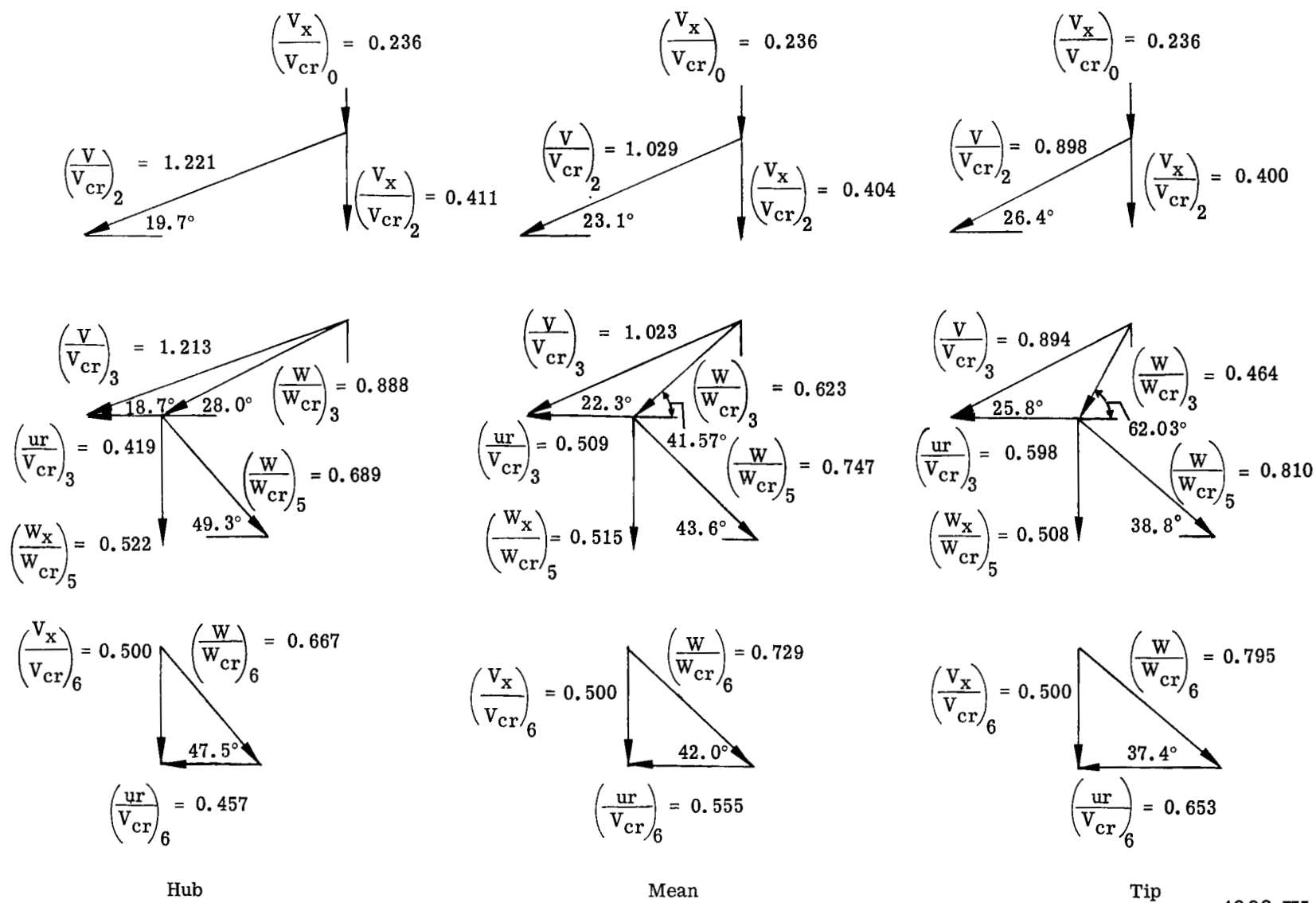
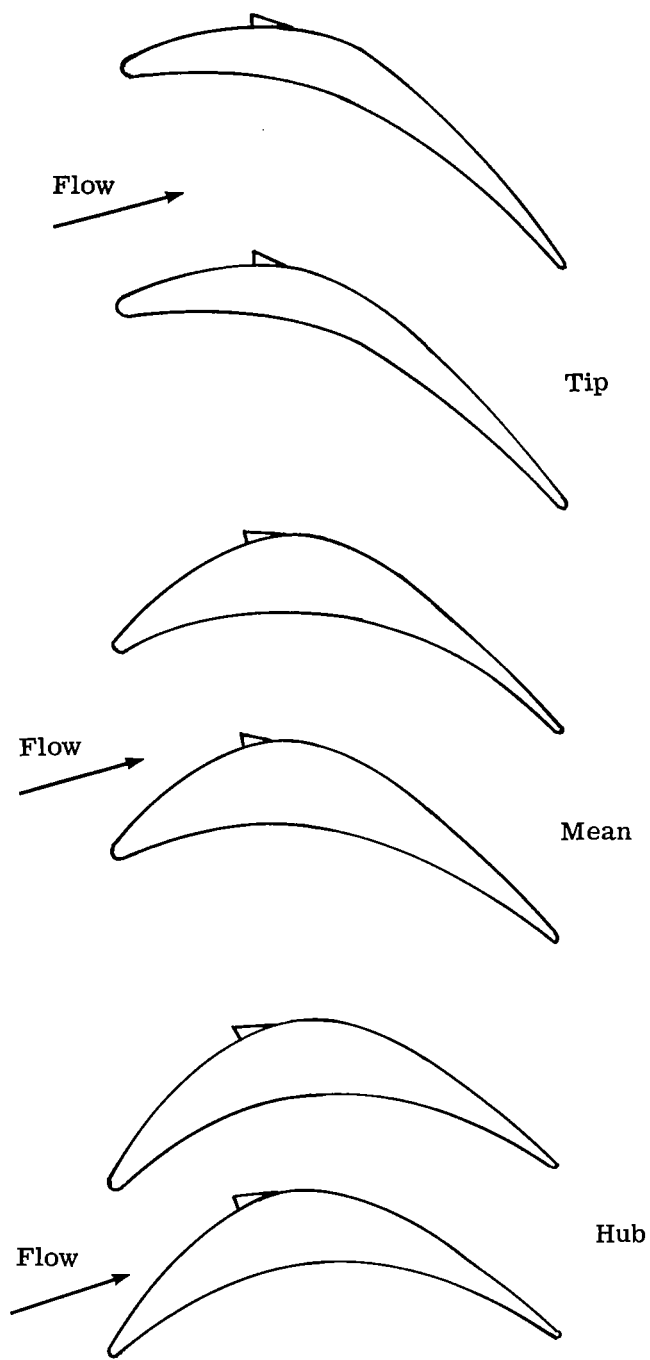


Figure 1. Plain blade with triangular plow vortex generators.



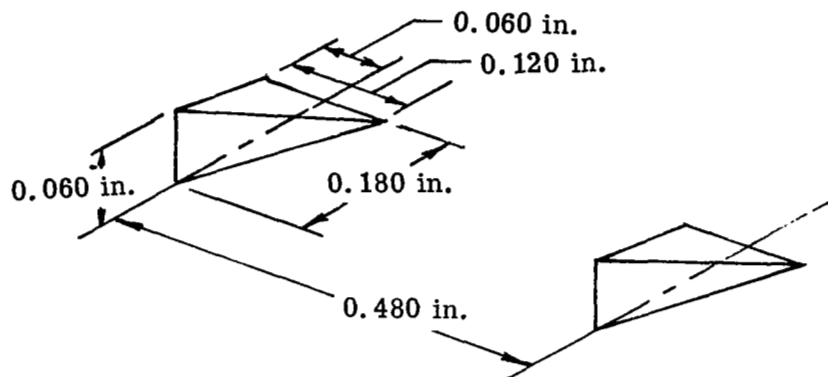
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Figure 2. Design velocity triangles.

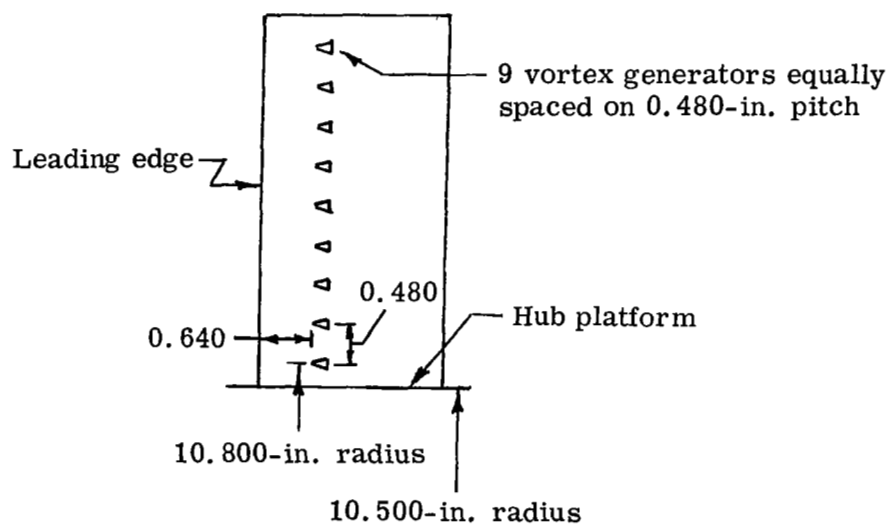


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Figure 3. Rotor-blade profiles and channels.



a. Triangular plow dimensions



b. Location of vortex generators on blade

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Figure 4. Geometric details of the vortex generators.

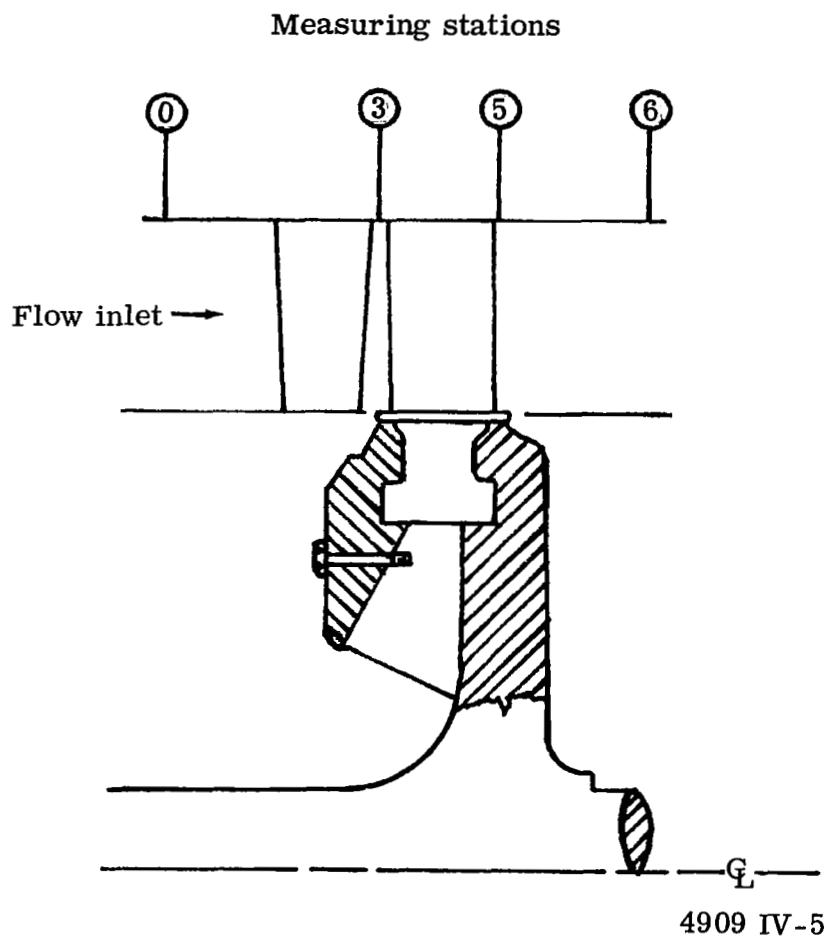


Figure 5. Schematic of flow path and station nomenclature.

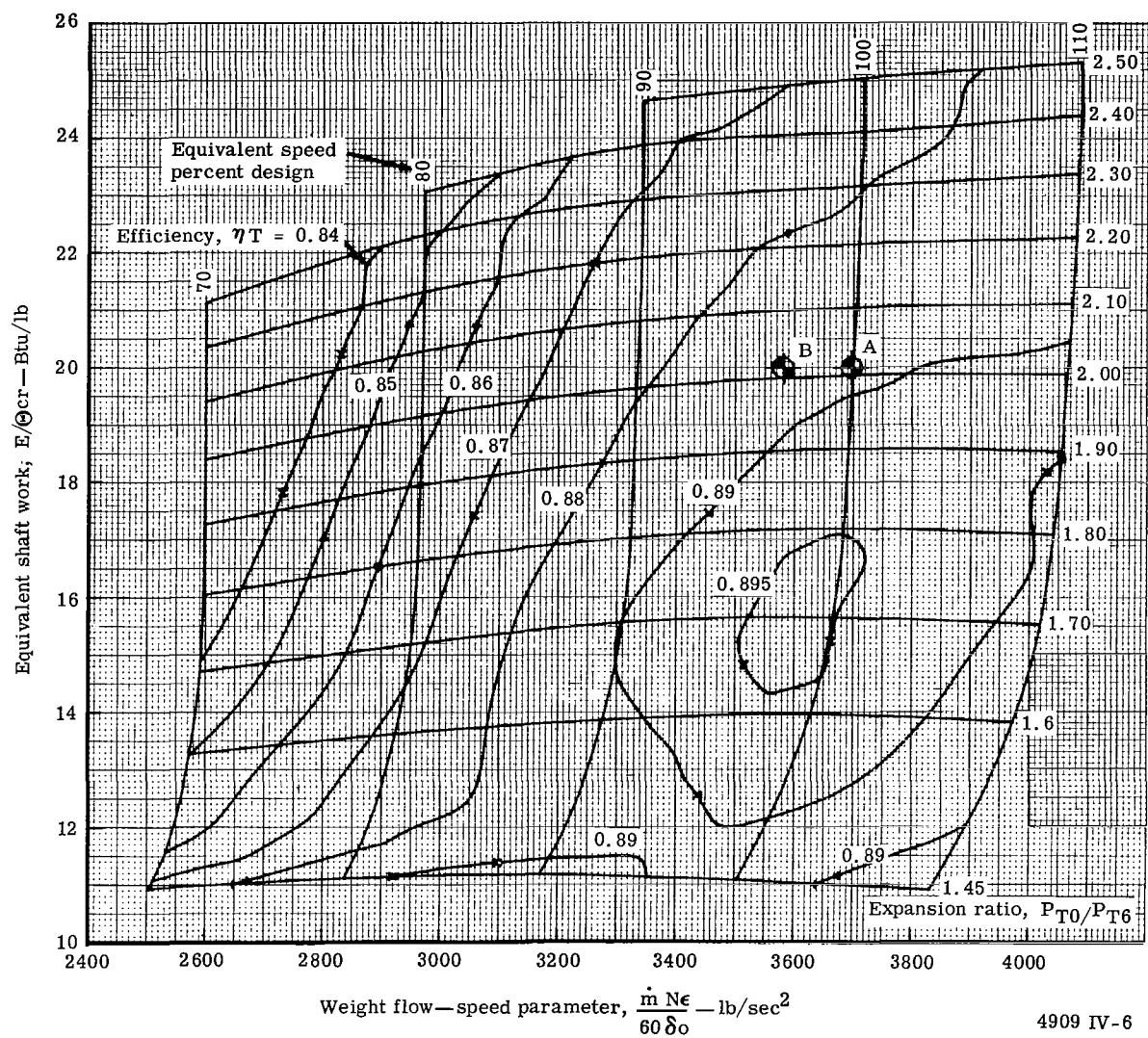


Figure 6. Overall performance of turbine with plain rotor blade having triangular plow vortex generators attached.

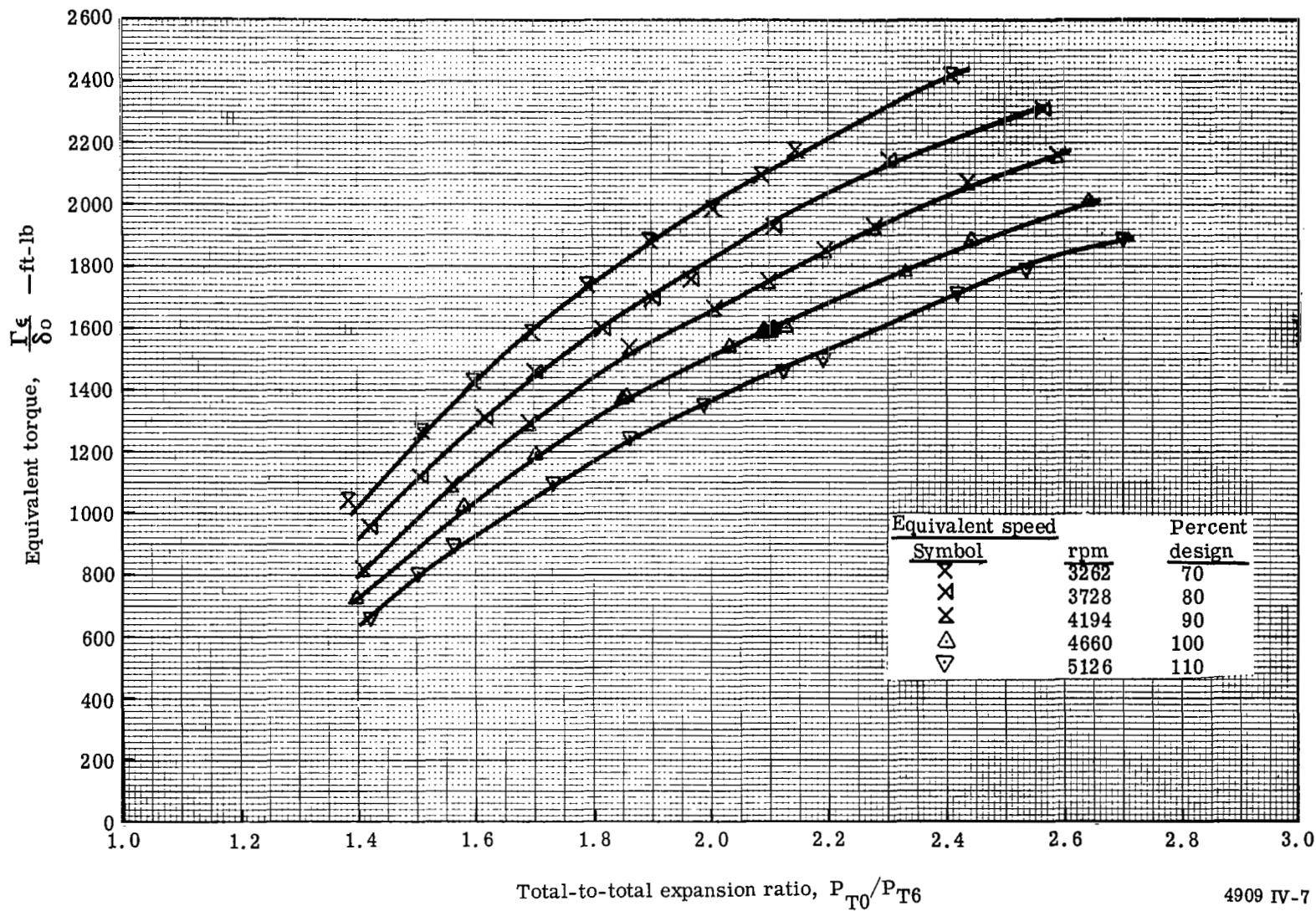
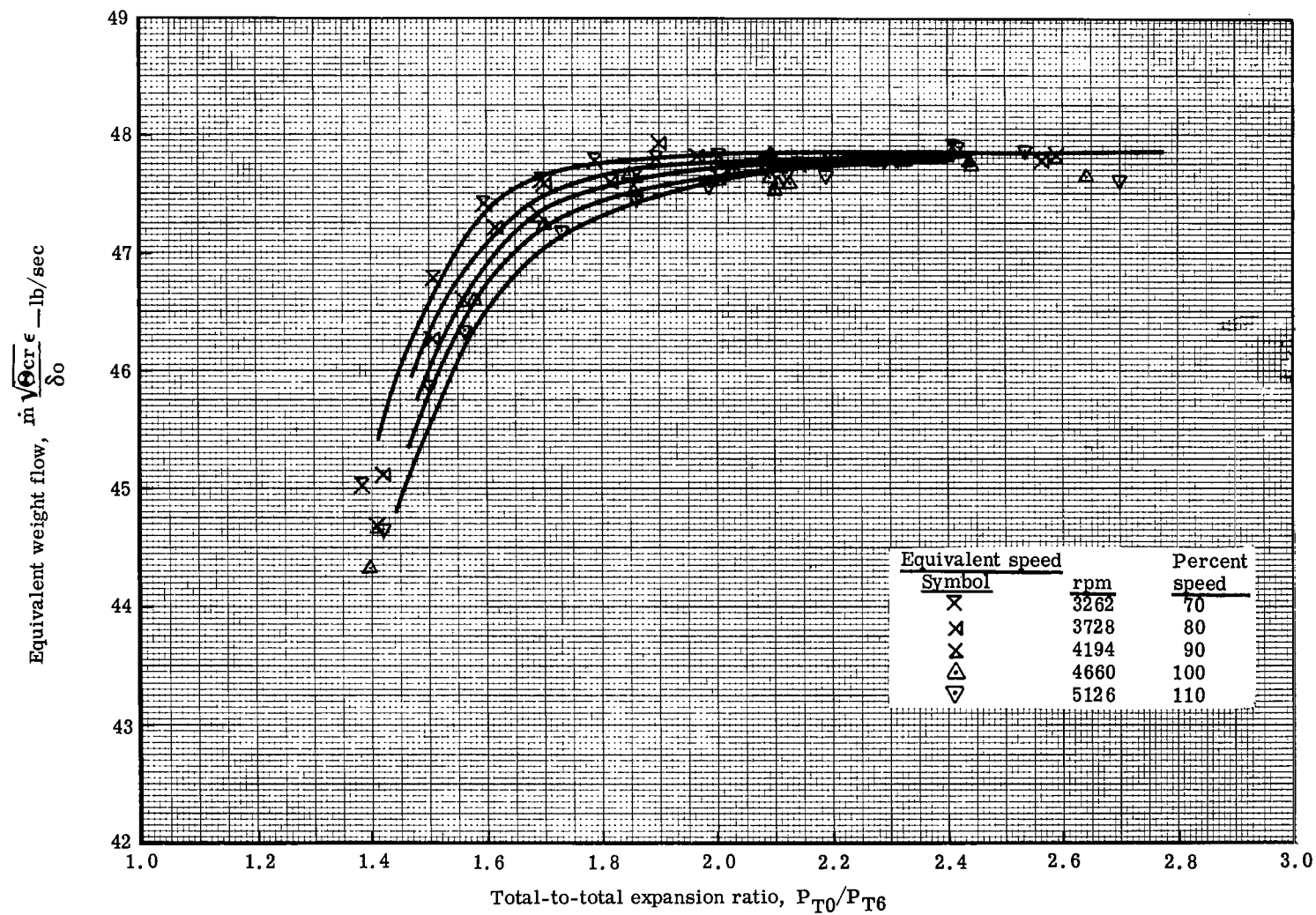


Figure 7. Variation of equivalent torque with expansion ratio for lines of constant equivalent speed.

Turbine rotor consists of plain blade with triangular plow vortex generators.



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Figure 8. Variation of equivalent weight flow with expansion ratio for line of constant equivalent speed.

Turbine rotor consists of plain blade with triangular plow vortex generators.

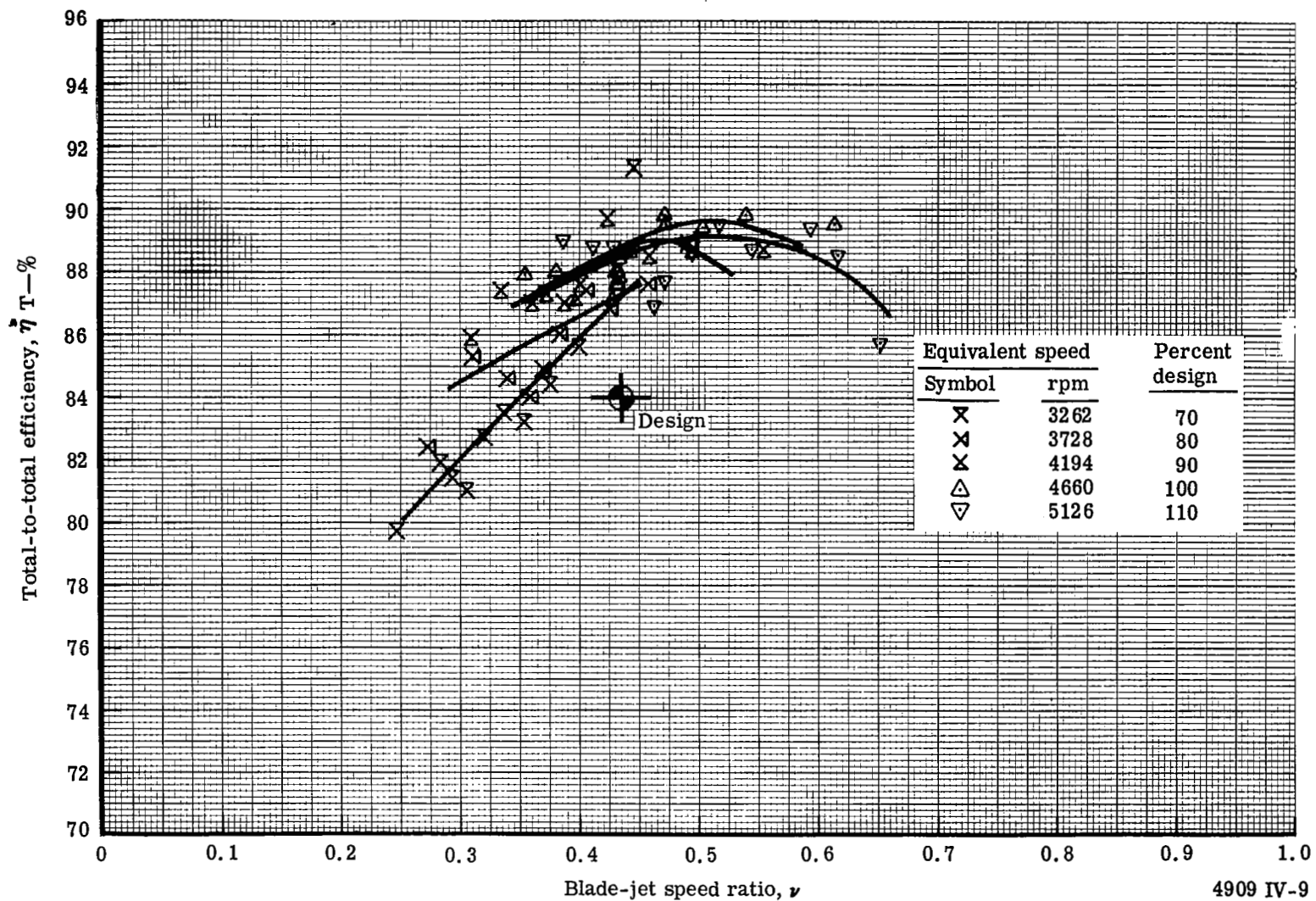


Figure 9. Variation of efficiency with blade-jet speed ratio for lines of constant equivalent speed. Turbine rotor consists of plain blade with triangular plow vortex generators.

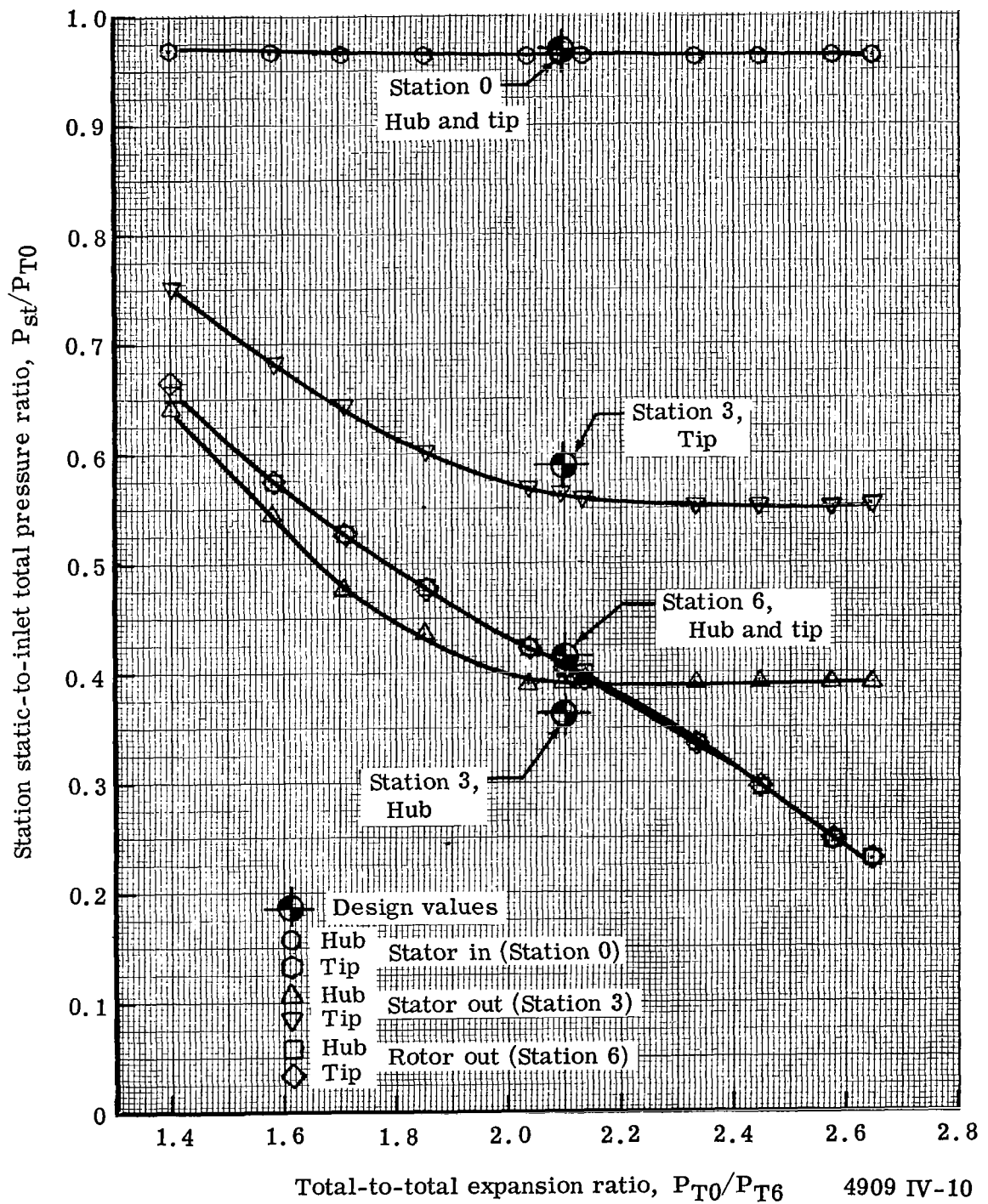
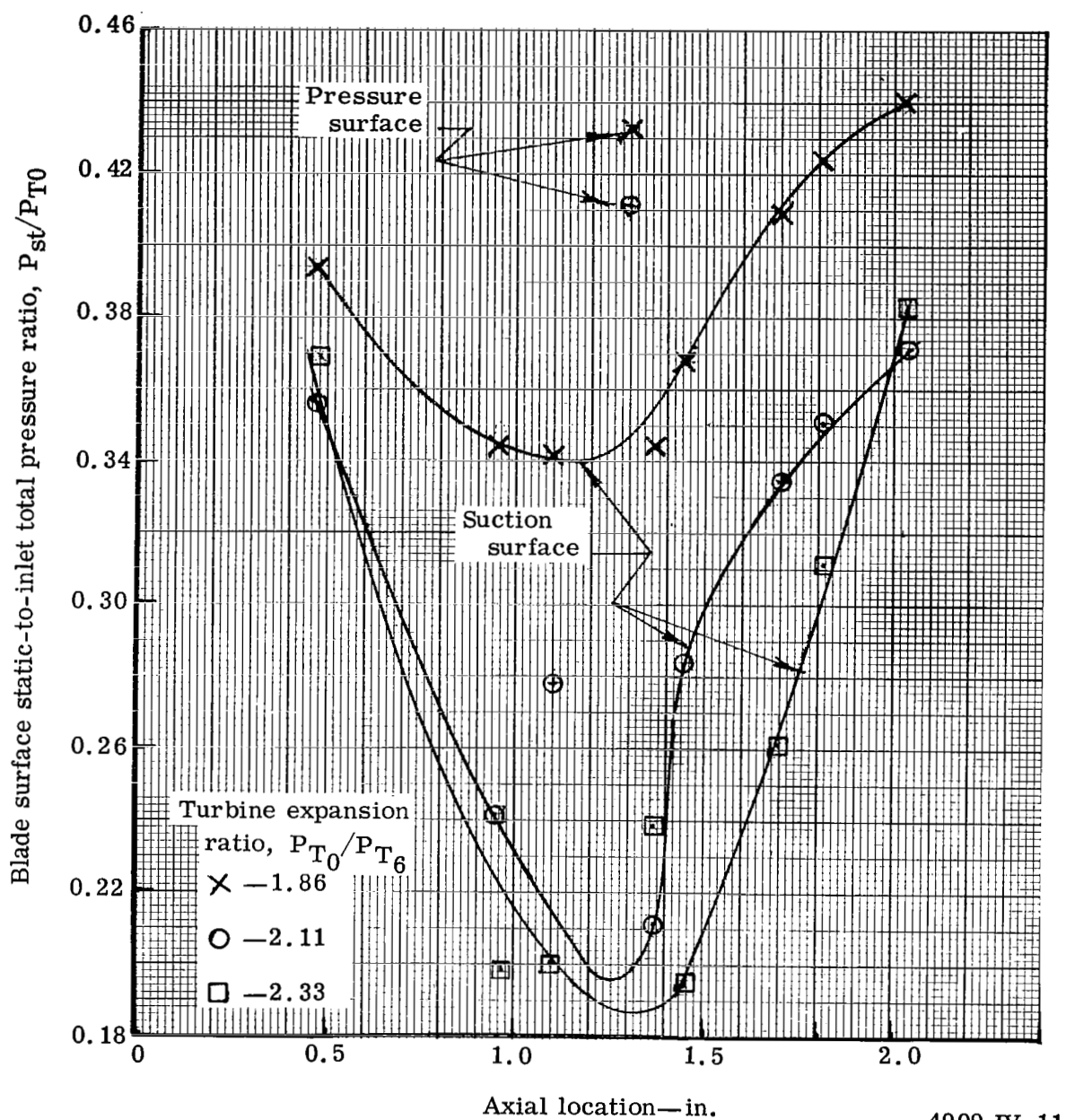


Figure 10. Variation of station hub and tip static pressure with turbine expansion ratio at design equivalent speed for plain blade with plow type vortex generators.



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Figure 11. Variation of measured blade surface static pressure at design equivalent speed for plain blade with plow type vortex generators.

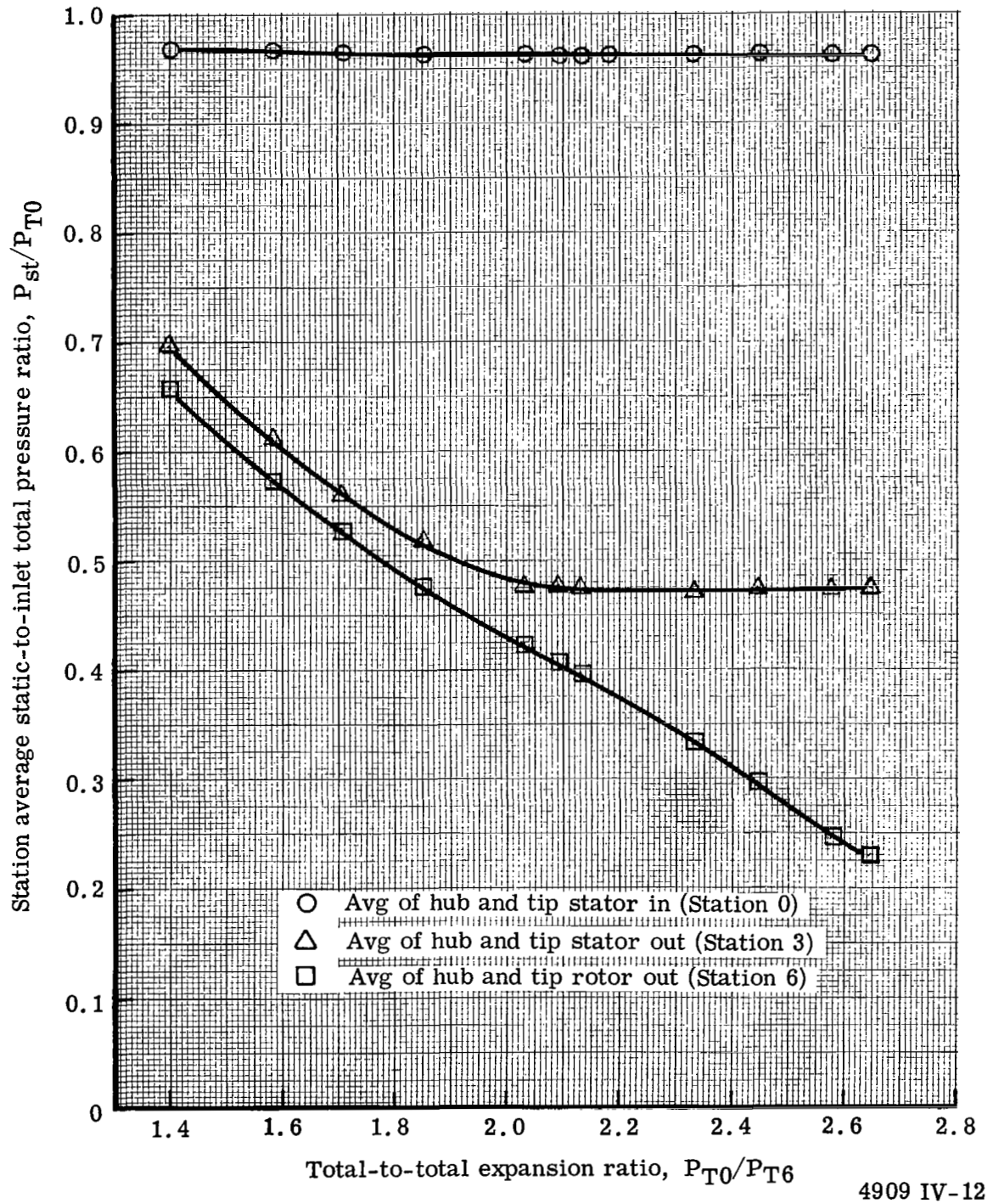


Figure 12. Variation in station average static pressure with overall turbine expansion ratio at design equivalent speed for plain blade with plow type vortex generators.

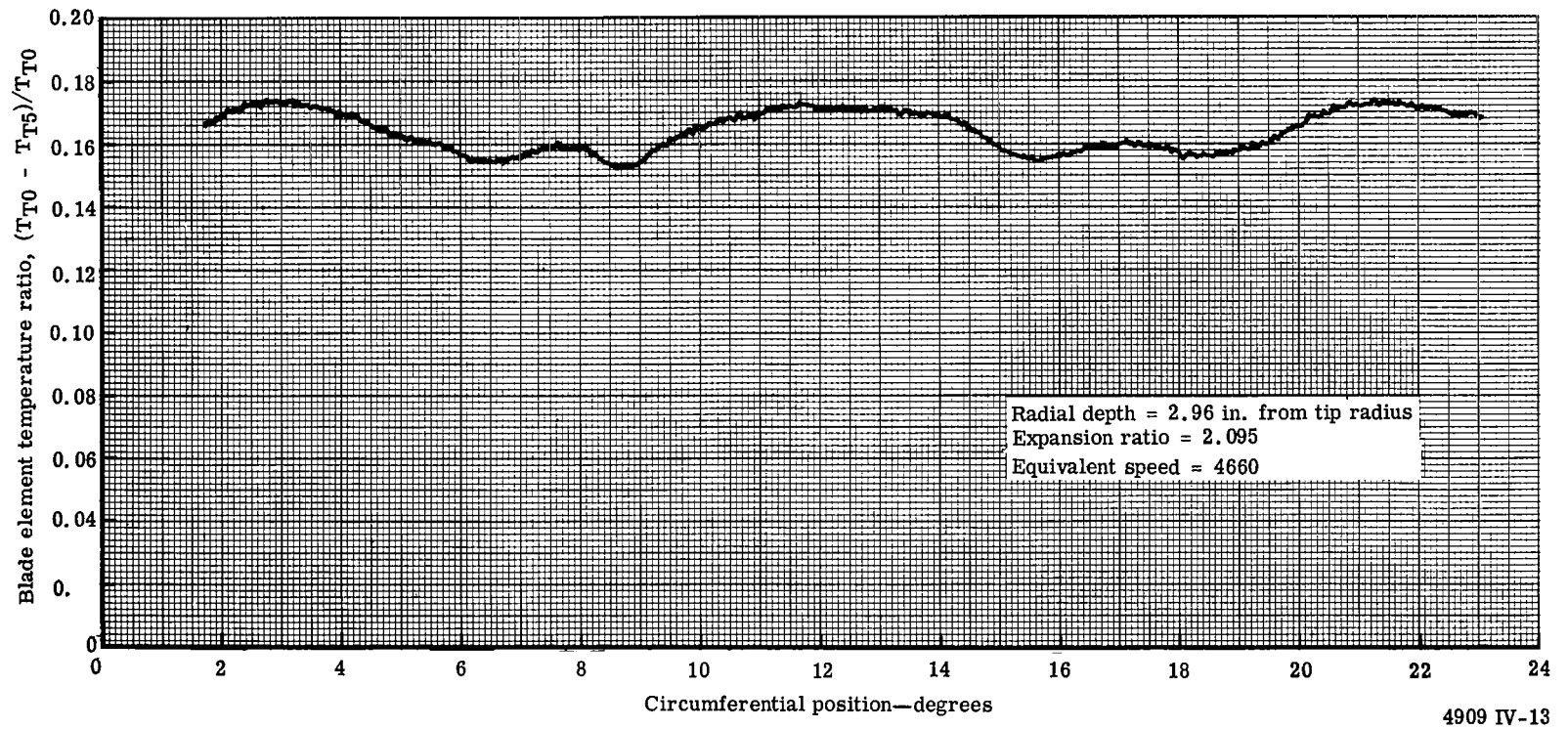
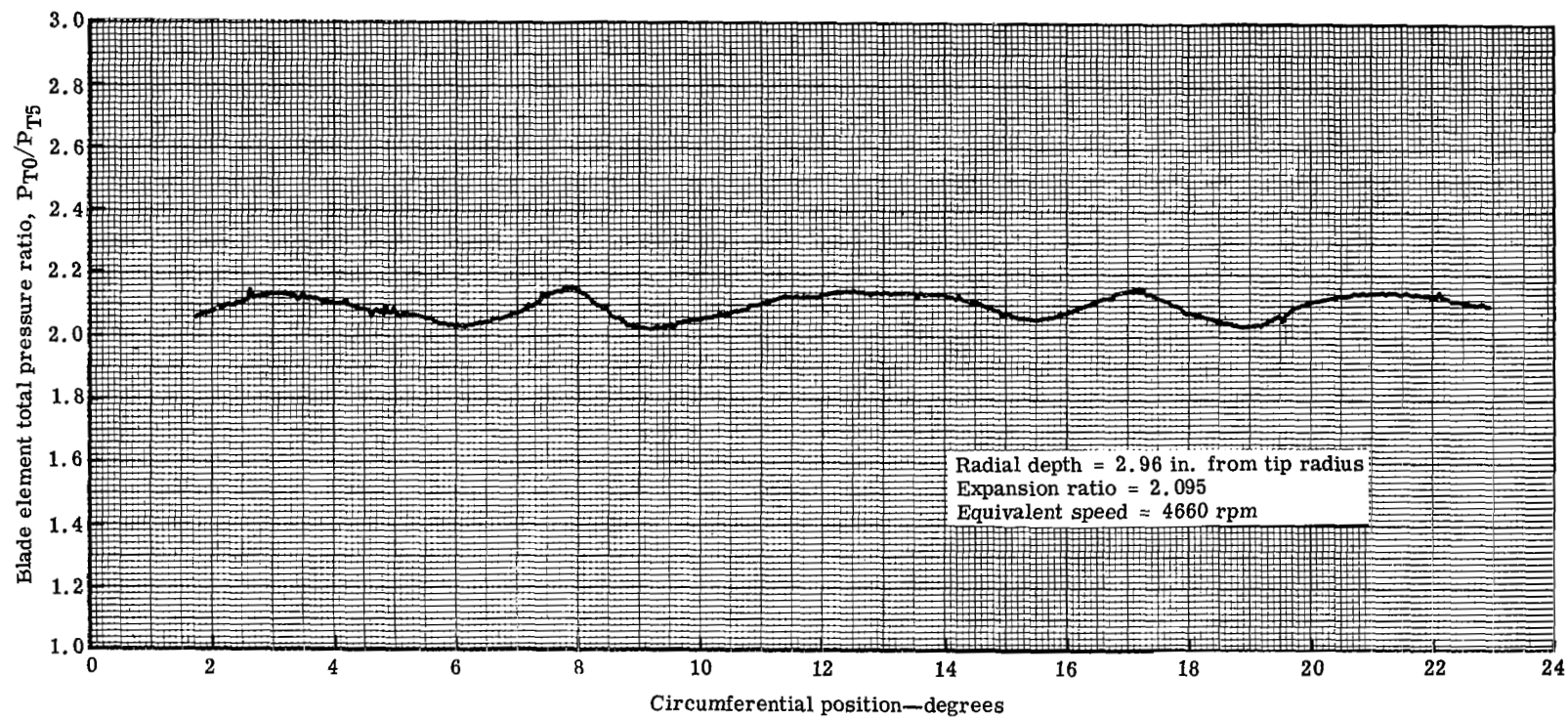
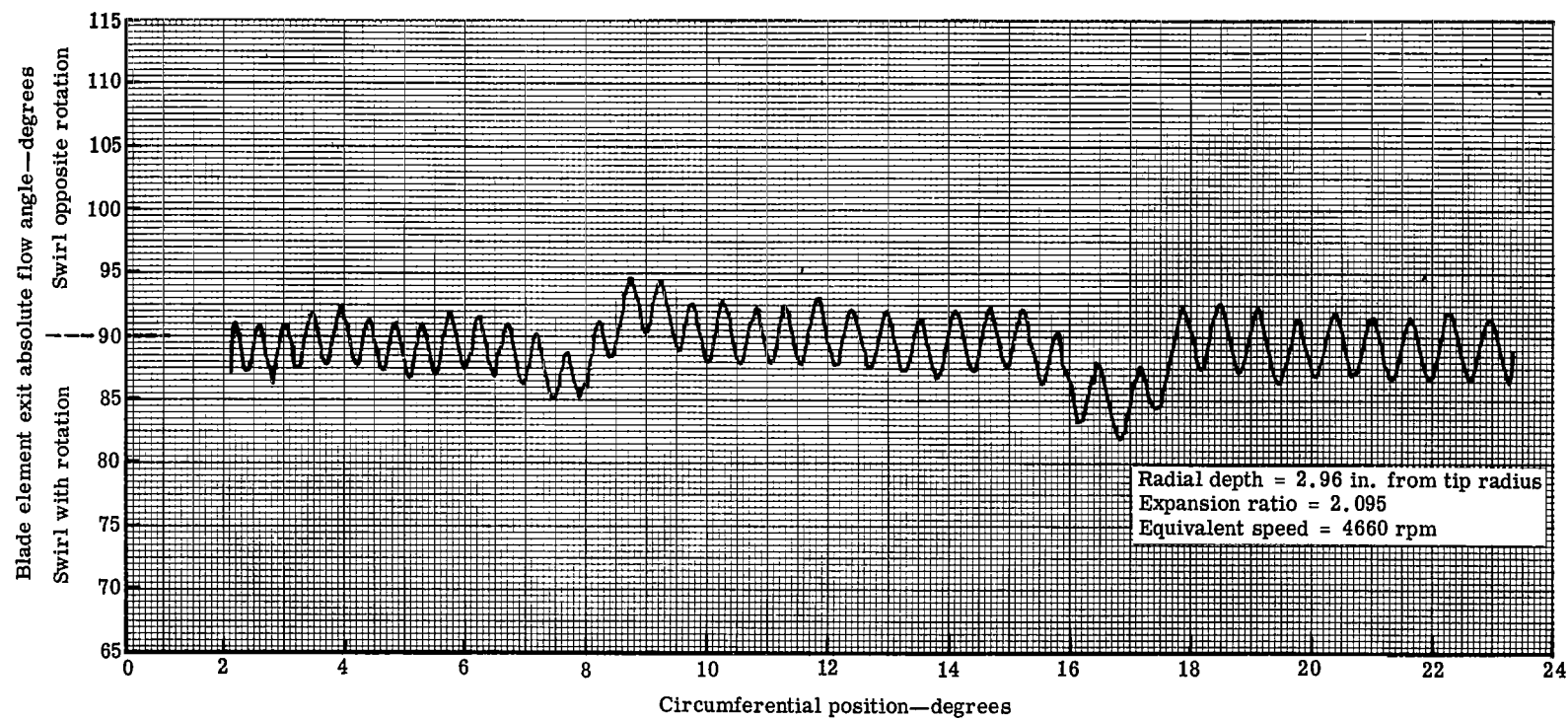


Figure 13. Circumferential variation of blade element temperature ratio recorded during rotor exit survey. Turbine rotor consists of plain blade with triangular plow vortex generators.



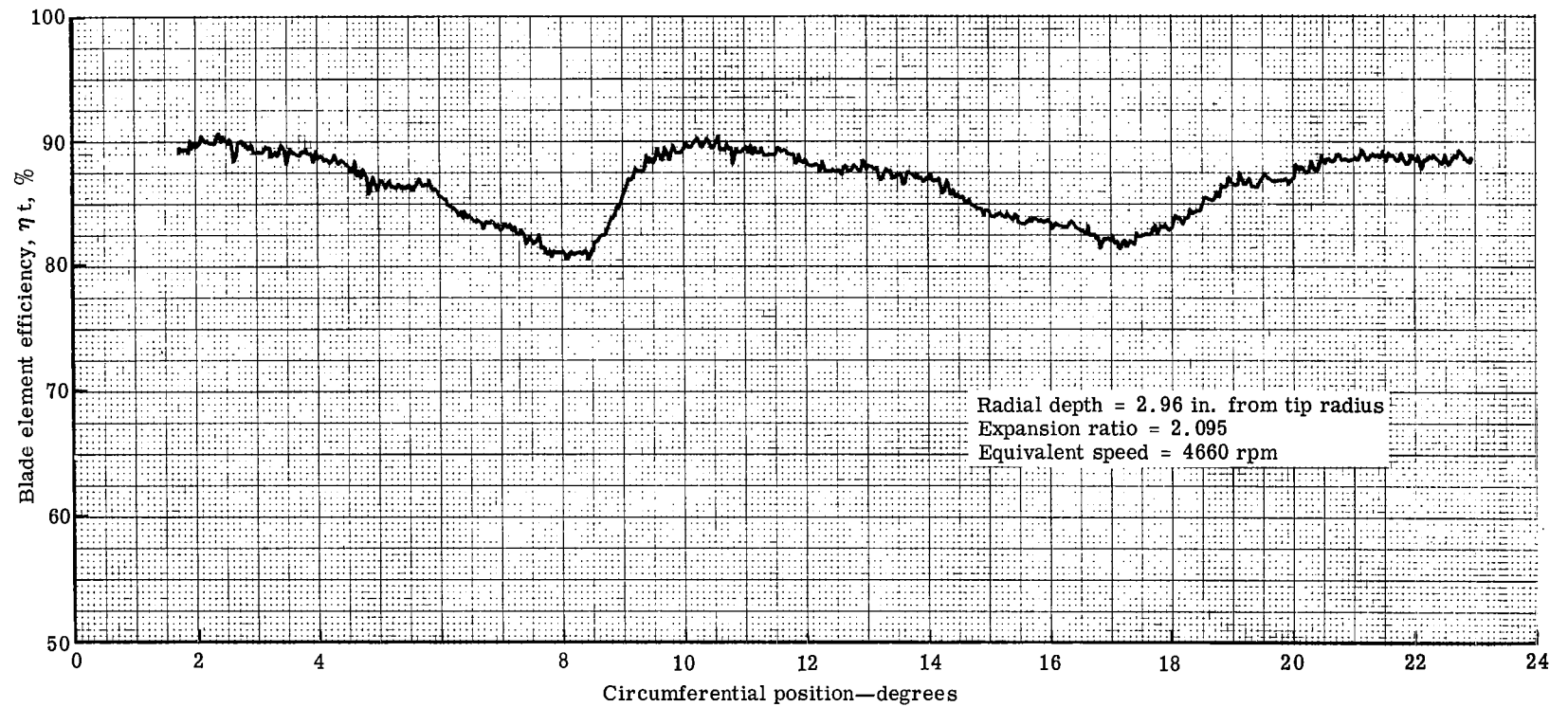
4909 IV-14

Figure 14. Circumferential variation of blade element total pressure ratio recorded during rotor exit survey. Turbine rotor consists of plain blade with triangular plow vortex generators.



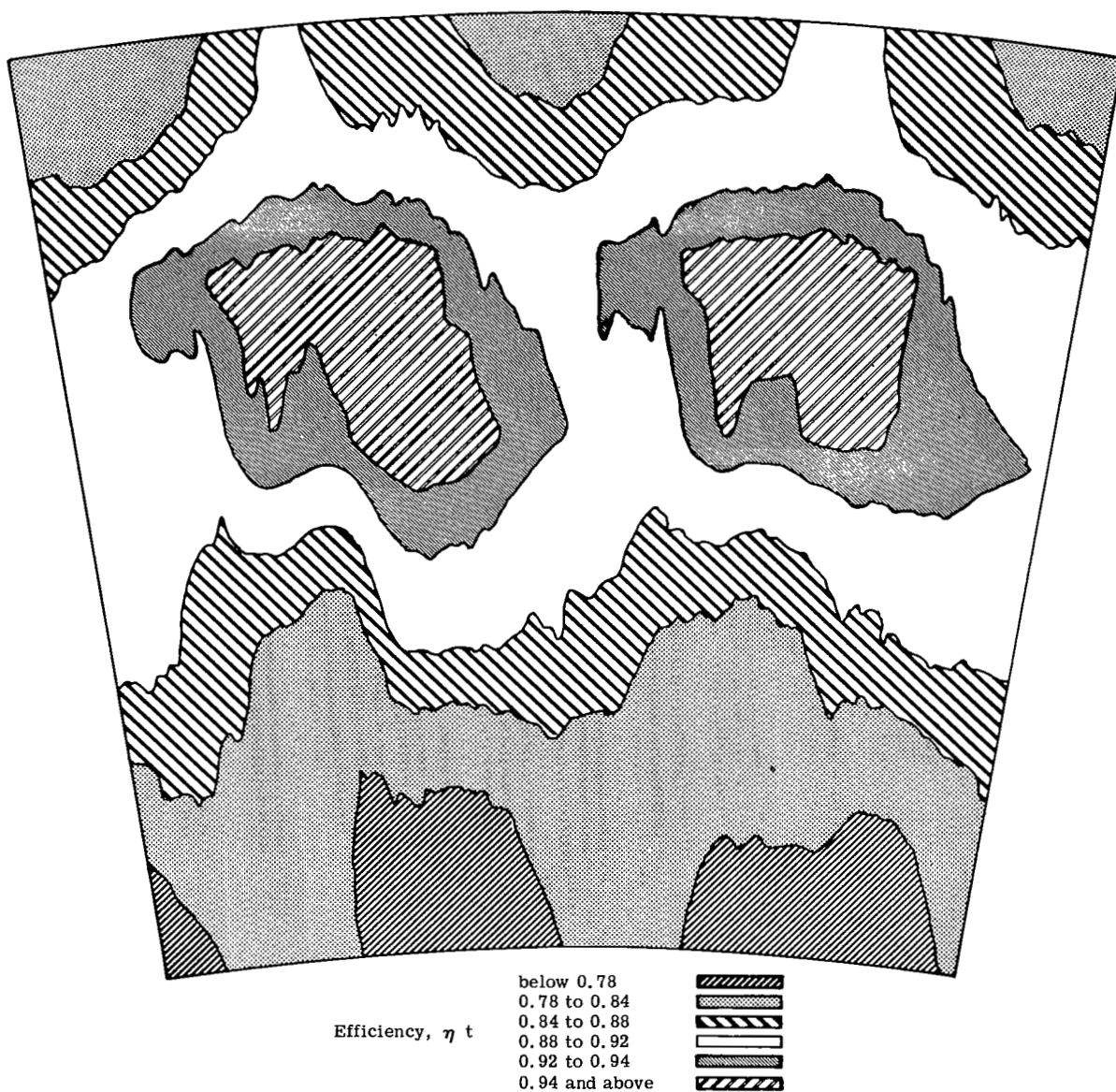
4909-IV-15

Figure 15. Circumferential variation of blade element exit absolute flow angle recorded during rotor exit survey. Turbine rotor consists of plain blade with triangular plow vortex generators.



4909-IV-16

Figure 16. Circumferential variation of blade element total efficiency calculated from rotor exit survey data. Turbine rotor consists of plain blade with triangular plow vortex generators.



Viewed looking upstream, $P_{T0}/P_{T6} = 2.095$, $N/\sqrt{\Theta_{cr}} = 4660$ rpm

4909-IV-17

Figure 17. Turbine stage total efficiency contours. Turbine rotor consists of plain blade with triangular plow vortex generators.

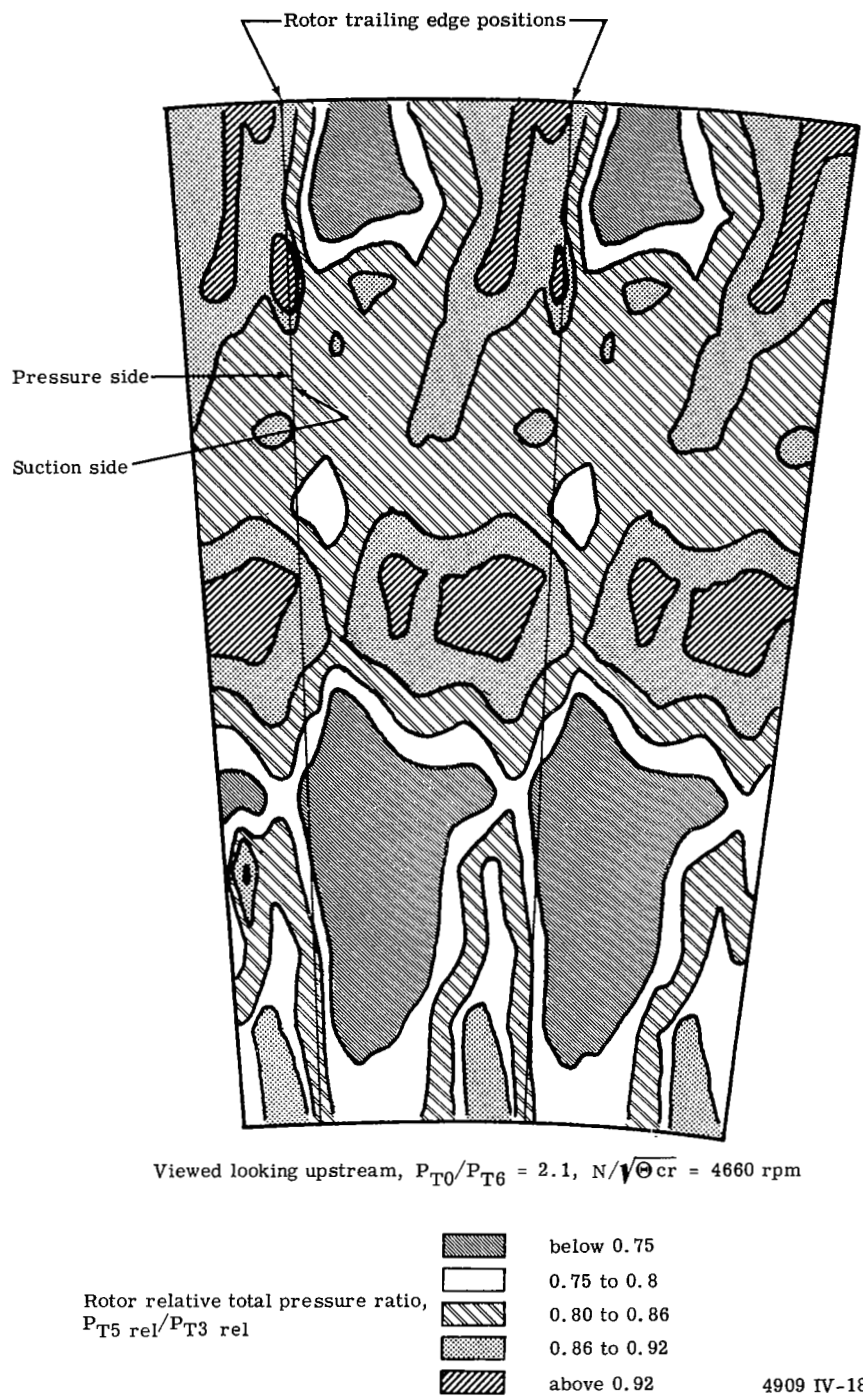


Figure 18. Rotor relative total pressure ratio contours as determined from hot-wire survey for plain blade with plow type vortex generators.

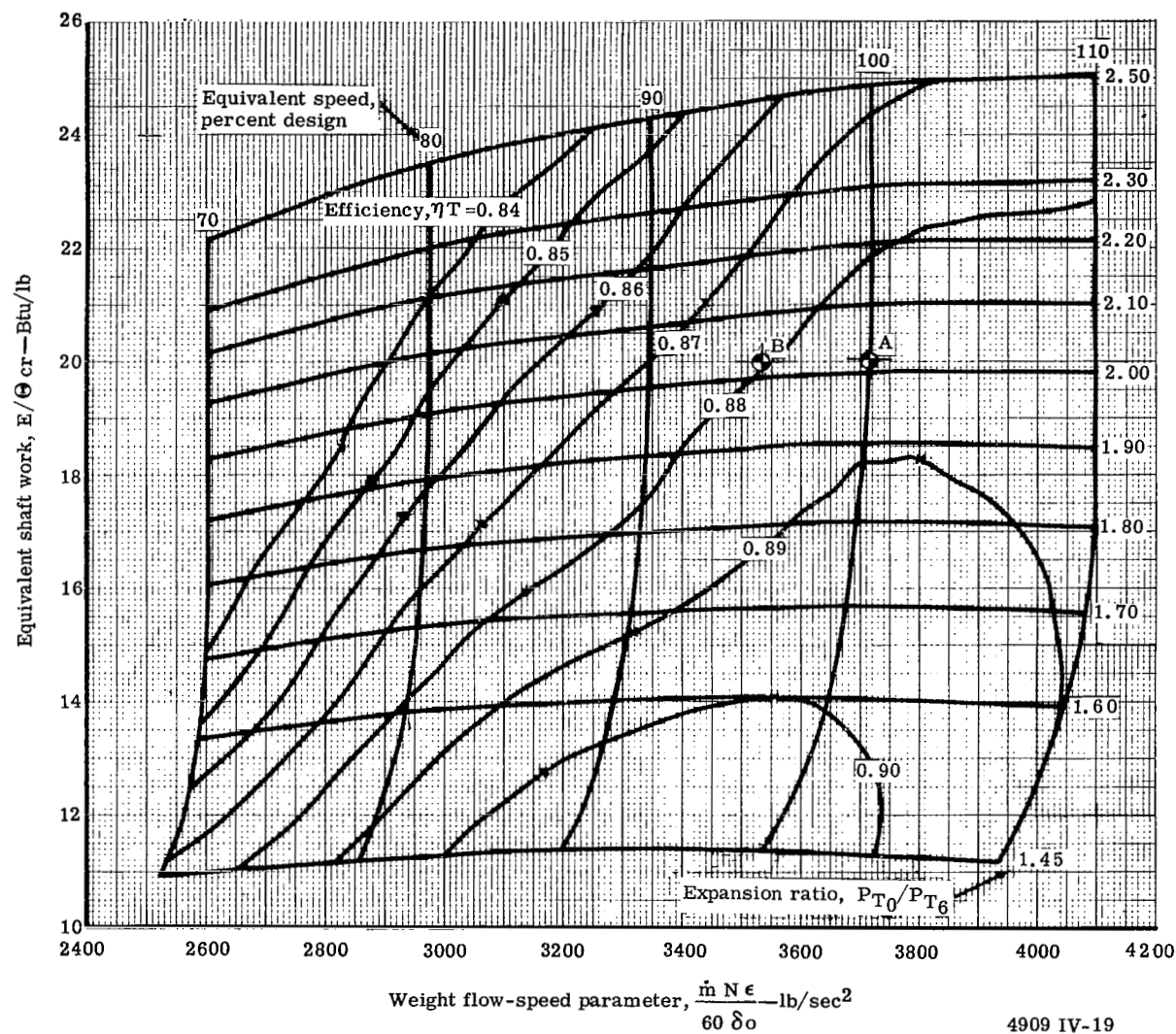


Figure 19. Overall performance of plain rotor blade turbine.

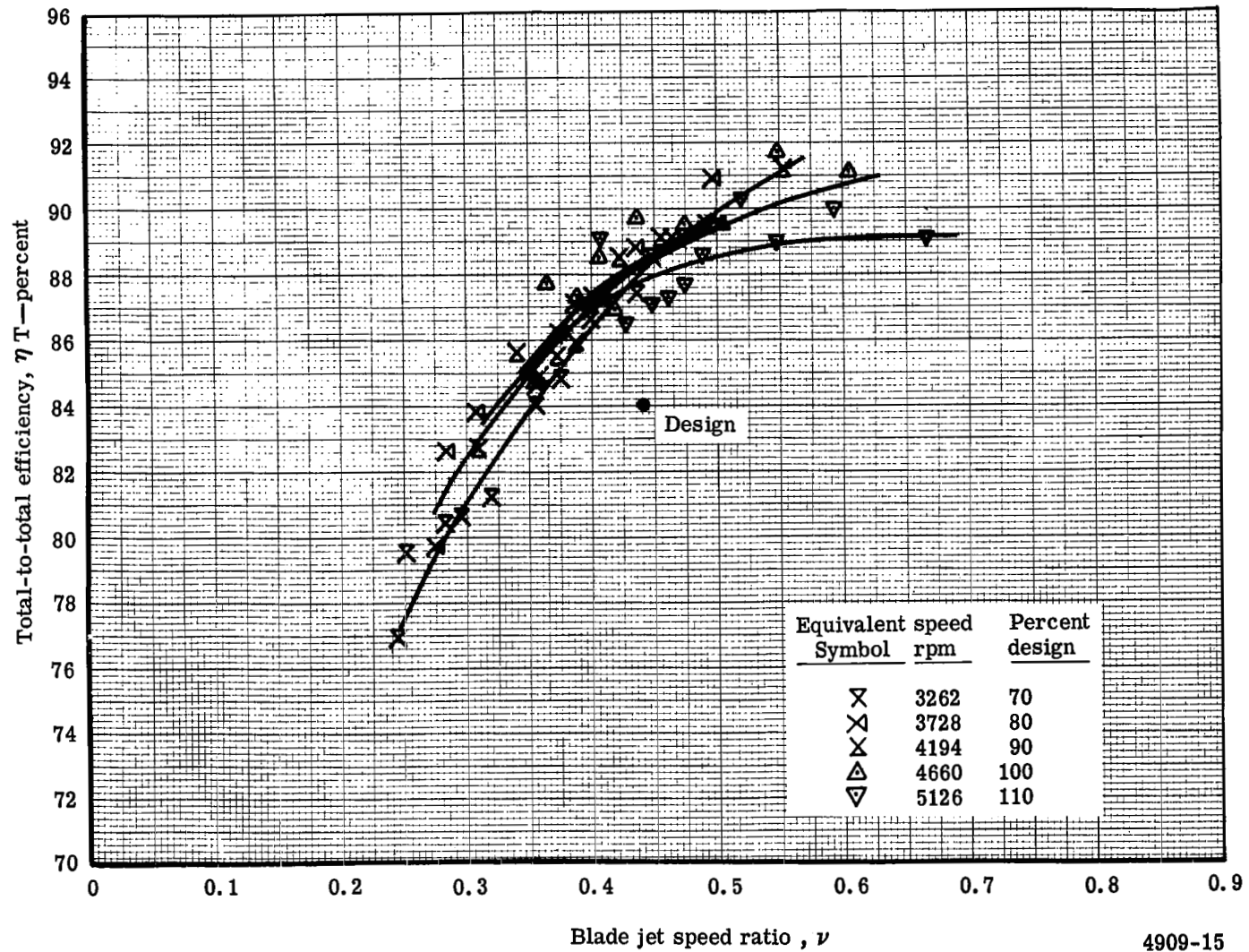
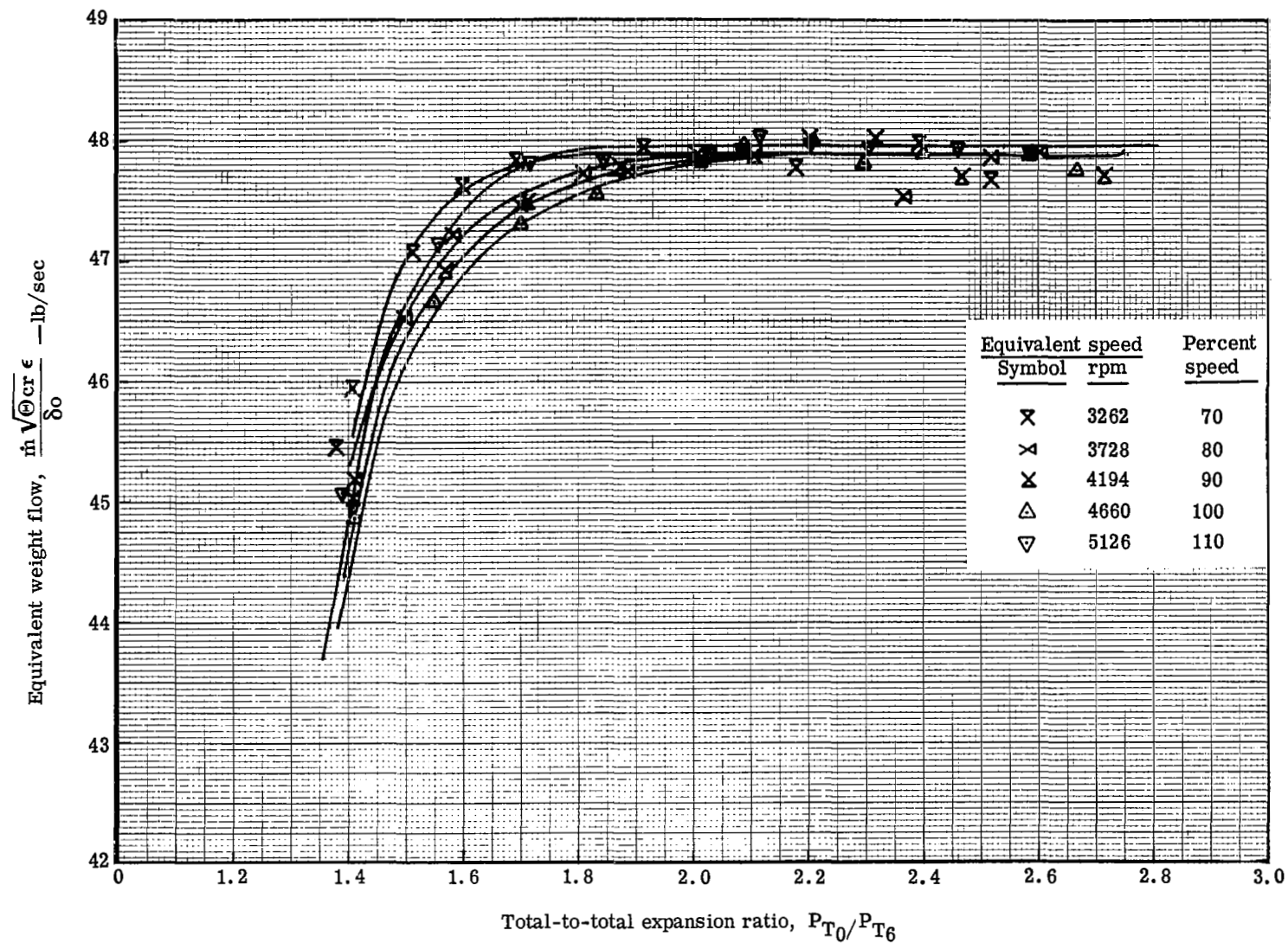
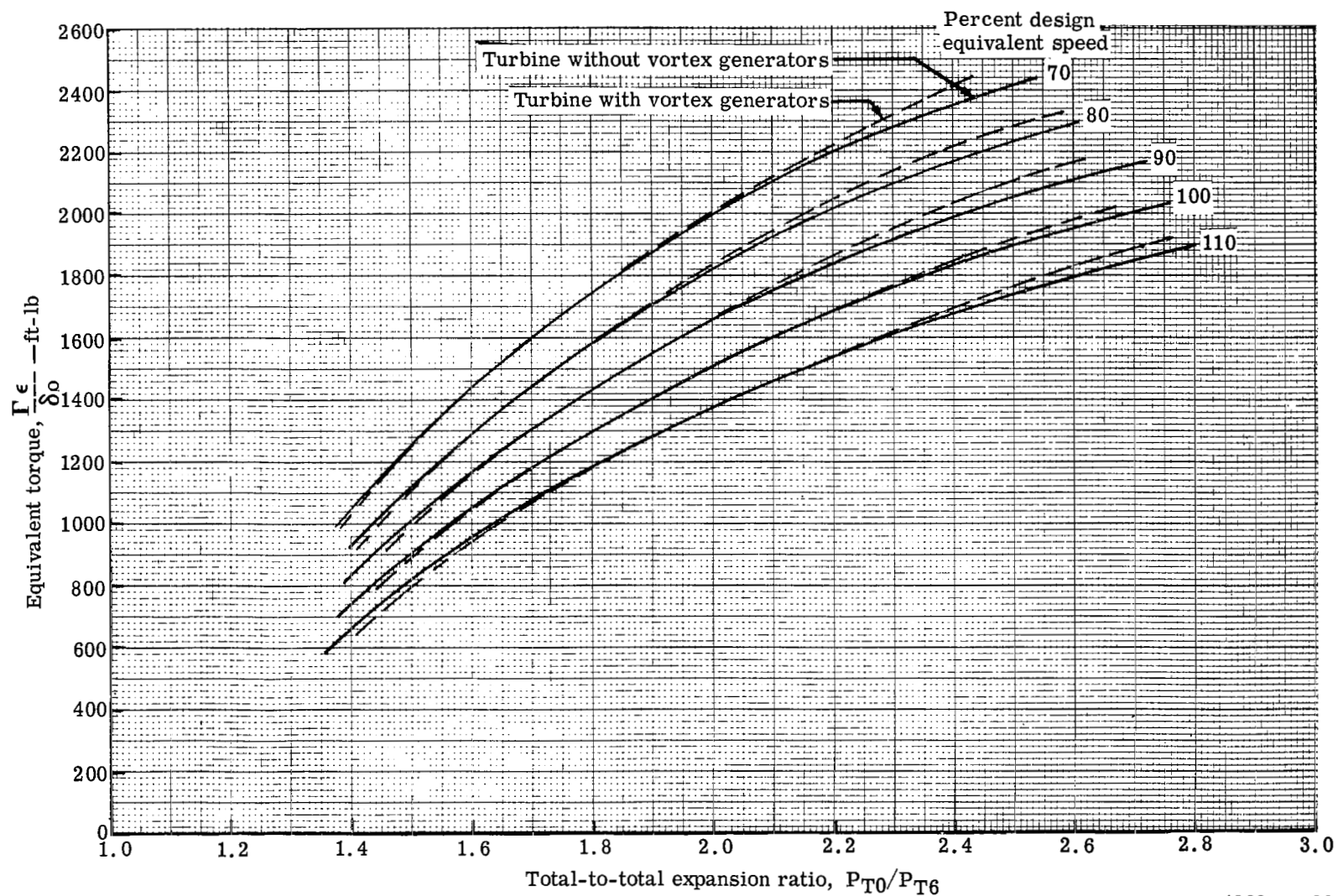


Figure 20. Variation of efficiency with blade jet-speed ratio for lines of constant equivalent speed for turbine without vortex generators.



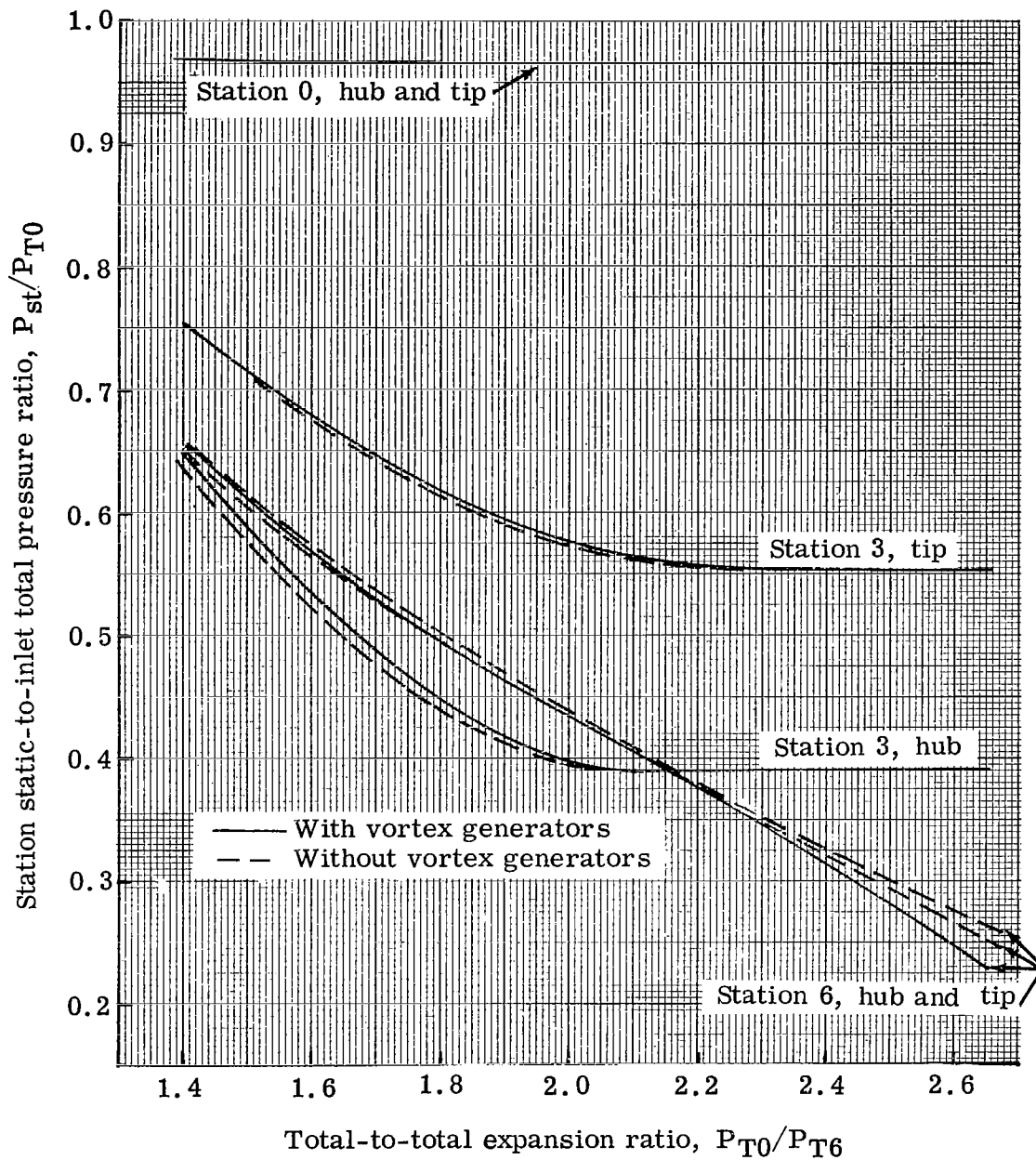
4909 IV-21

Figure 21. Variation of equivalent weight flow with expansion ratio for lines of constant equivalent speed for the turbine without vortex generators.



4909-IV-22

Figure 22. Variation of equivalent torque with expansion ratio and equivalent speed with and without vortex generators.



4909-IV-23

Figure 23. Variation of station hub and tip static pressure ratio with expansion ratio at design equivalent speed for turbine with and without plow type vortex generators.

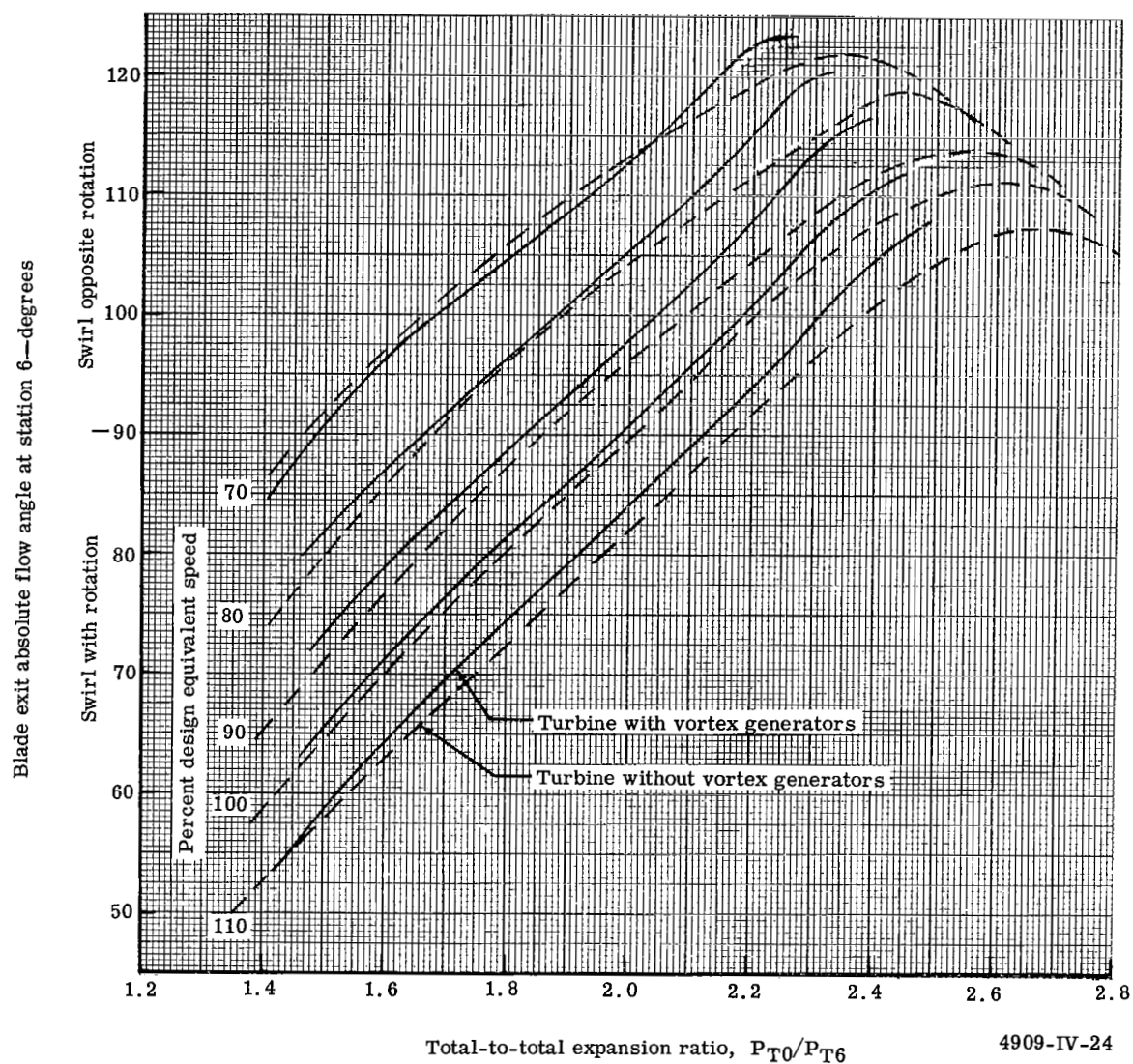
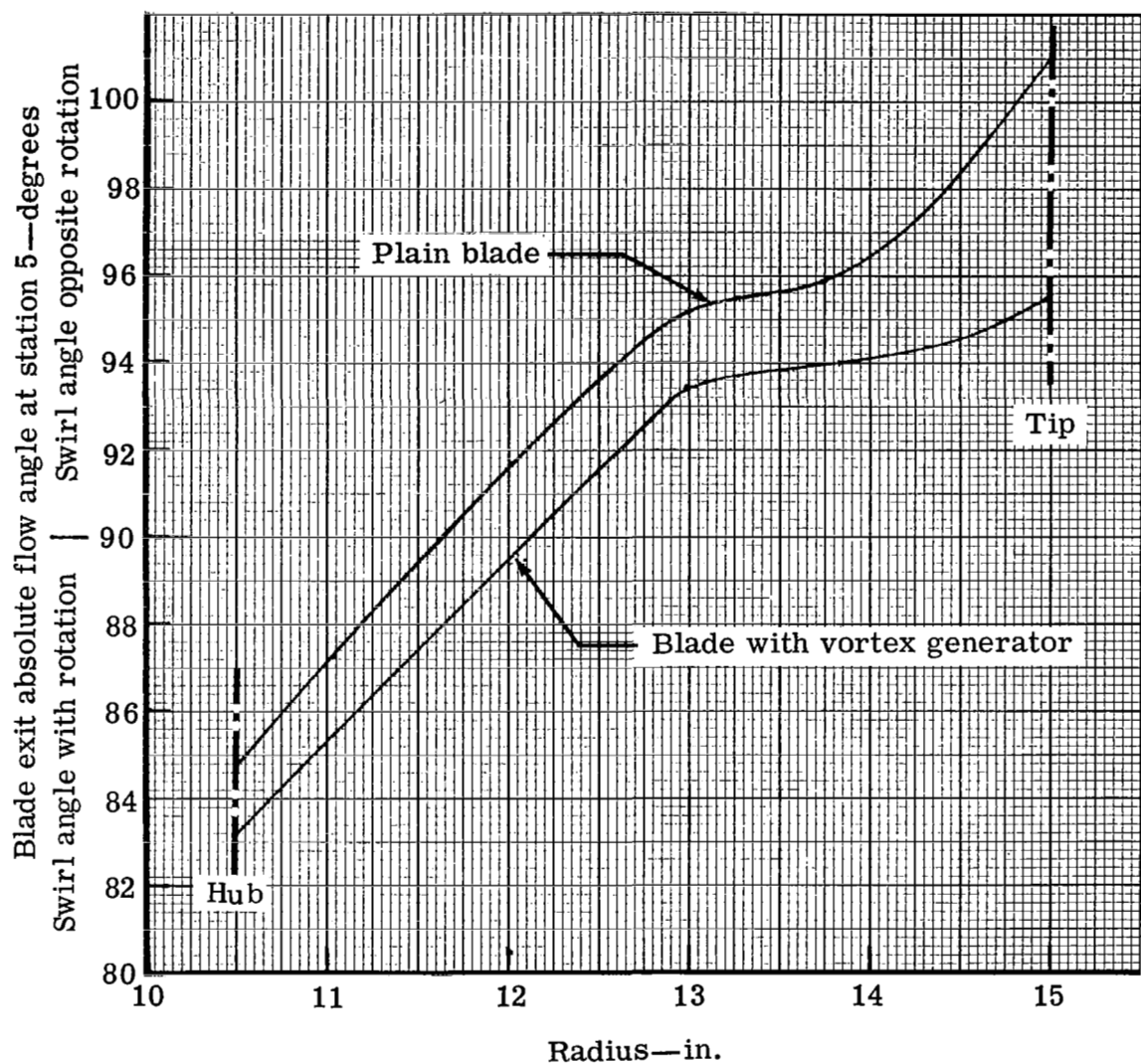
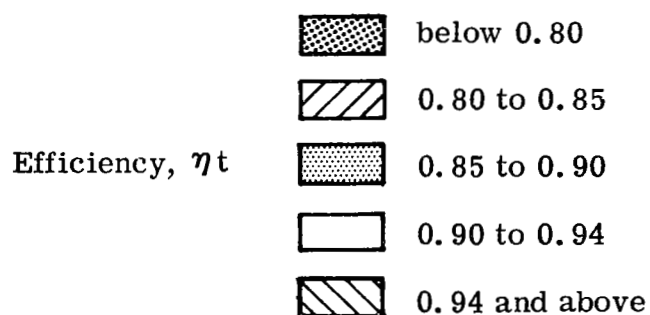


Figure 24. Variation of rotor exit absolute flow angle with expansion ratio and equivalent speed for turbine with and without vortex generators.



4909-IV-25

Figure 25. Average blade exit flow angle at station 5 for turbine with and without vortex generators when operating at design equivalent speed and expansion ratio.



Viewed looking upstream, $\frac{P_{T0}}{P_{T6}} = 2.10$, $\frac{N}{\sqrt{\Theta_{cr}}} = 4660 \text{ rpm}$

4909 III-19

Figure 26. Turbine stage total efficiency contours for turbine without vortex generators.