JET LOSSES IN A PARTIAL ADMISSION TURBINE

by

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ABSTRACT

In partial admission turbines there are severe losses that do not show up in their full admission counterparts. These are due to windage, pumping and mixing, and filling and emptying. This report is the result of work done in an attempt to formulate these losses. It was found that for a large aspect ratio turbine (2.66), the loss can be predicted using a formula worked out by Stenning (1), if a suitable windage loss can be approximated.

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Title: Assistant Professor of Mechanical Eng.

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1. INTRODUCTION

There are three major sources responsible for the low efficiency observed in partial admission turbines. These are the windage losses, the filling and emptying losses, and the pumping and mixing losses. The only attempt to formulate these has been made by Stenning (1), and the results are an analytical expression which has been checked out in the low aspect ratio regime. However, the windage loss was not readly observed due to the fact that the low aspect ratio turbine induces very little windage flow. Consequently, there was no term in the formulation to compensate for this loss.

Now with the results of Wennerstrom (2), who did a study of windage losses on a normal aspect ratio turbine (2.66), this formula can be checked out in this regime using the same turbine and investigating the flow in the nozzle area. The losses that are of primary concern in this region are due to the pumping and mixing and the filling and emptying. The filling and emptying losses are the work needed to move the stagnant air being brought into nozzle arc by the turbine wheel from the arc of non-admission. The loss due to pumping and mixing is the work needed to bring the air into the jet times the mixing coefficient (1).

To thoroughly investigate the losses, the velocity profiles in the nozzle area were mapped using a stagnation pressure probe. Having the velocity triangles resolved for the turbine, the net output work can be calculated, knowing the windage loss. The efficiency obtained by dividing the net output work by the kinetic energy of the nozzle jet should then correlate with value formulated by Stenning (1).

2. TEST APPARATUS

2.1. Test Turbine

The turbine wheel, shaft, and bearing carriage used for the test were from a General Electric turbo-supercharger, Model 7s-B31-A1. The wheel has 144 impulse-type, axial-flow blades approximately 1.25 inches long. The outer diameter of the turbine wheel, including a shrouding ring on the blades, is 12.25 inches.

The turbine nozzle is composed of a round steel tube of 0.930 inches inside diameter, that has been formed into a rectangular cross section 0.375 inches wide and 1.0625 inches high, giving an outlet area of 0.400 square inches (fig. 6.4). The nozzle is placed at an angle of 60 degrees measured from the axial direction. Air approaches the nozzle tube from a 2.375 inch inside diameter pipe. The converging section is made from a turned oak plug. The angle of convergence is 30 degrees and the surface exposed to the flow is lacquered and polished.

The turbine is encases in a cylindrical steel shroud which is free to rotate on a backing ring made of 0.75 inch hard maple. The backing ring has an axial clearance with the turbine wheel of about 0.030 inches, and is bolted rigidly to the supporting framework. The shroud can be rotated to any position and is then pinned to the backing ring with three fine steel pins to prevent movement. The radial clearance between the turbine and the shroud varies from 0.010 to 0.030 inches around the periphery, as the shroud is constructed of welded steel stock and is not precision bored. An inner shroud of diameter equal to the blade root diameter is mounted concentrically to provide a cylindrical annulus for the exit flow. This annulus has an axial length of 4.75 inches. The inner shroud is plugged to prevent flow from entering at the center and leaking into the exit annulus due to wheel windage. Three large rivets at the hub of the wheel induce a large flow without this plug.

For the 3540 rpm runs, the turbine was connected to a 1/4 horsepower, 1800 rpm, synchronous motor, which operated as a dynamometer. The pulley system used gave a ratio of turbine to motor speed of 2.038. To obtain a lower turbine speed such that a lower nozzle speed to wheel speed ratio could be obtained, a 3/4 hr D.C. motor was installed and acted as a generator when driven by the turbine. The load was varied by using a resistor board composed of three lightbulbs and thereby acted as a speed control. In the latter set-up the speed ratio of turbine to motor speed was approximately 1.08.

2.2 Instrumentation

The objective of this test was to map the velocities in the nozzle area. The probe used for this was a three-hole, cylinderical, yaw probe. This type of probe was chosen as the best of available probes suited for traversing the entire annulus radial width and rotating through 360 degrees at all points. The probe was positioned with its axis corresponding to a radial line from, and perpendicular to, the turbine axis. The probe centerline was located 1.125 inches downstream from the blade exit. The traversing device holding the probe was adjustable radially (along the probe axis), and angularly (about the probe axis). For radial adjustments, a millimeter scale was used, and for angular adjustments, a 360 degree protractor. The traversing device was mounted on the turbine shroud, which could be rotated completely around the turbine axis.

The yaw pressure taps were connected to opposite sides of a U-tube with one of its legs inclined about 60 degrees to amplify readings. A fixed reference line was located on the inclined leg of the U-tube and was placed tangent to the fluid meniscus at static equilibrium conditions. It is necessary to align the pointer of the traversing device with the stagnation tap of the probe by balancing the yaw tap pressures in a stream of known direction. The stagnation pressure tap was connected to a vertical water manometer. Pressures measured were the differences between stagnation pressure and atmospheric pressure.

Two of the manometer-bank tubes were converted to a mercury U-tube and used to measure the pressure drop across the junction of the large diameter pipe and the nozzle tube. The nozzle tube side of the U-tube also had a petcock that could be opened to the atmosphere, allowing a measurement of the total upstream static pressure. A glass tube thermometer was placed in the line ahead of the converging section. Line air velocity was low enough (35 fps) to consider the stagnation and static temperatures essentially equal. Knowing the static pressures of the pipe and the nozzle tube along with the upstream temperature, the mass flow can be calculated assuming no losses using Bernoulli's equation.

Turbine speed was measured with the use of a Strobotac. Details of the instrumentation can be seen in figures 6.1, 6.2, and 6.3.

3. OPERATIONAL PROCEDURE

3.1. Probe Calibration

To align the pointer of the traversing device with the stagnation pressure taps, the probe must be calibrated in a stream of know direction. This was accomplished by installing the probe in the outlet of a small, low velocity wind tunnel and determining the direction of the flow. The flow direction was determined by installing the probe first from the top of the outlet and then rotating it 180 degrees and protruding it from the bottom. In this manner, two reading are gotten assuming the pointer to be randomly located with respect to the probe axis, but firmly affixed. To determine the actual angle of flow the angle between the two readings is split. Then, having the flow angle determined, the pointer is set pointing in the direction of the flow and the probe body is "zeroed in" using an inclined U-tube to null the yaw pressure taps.

The scale, measuring position along the axis of the probe, was adjusted in such a way that a reading of 11.35 centimeters corresponded to the probe pressure taps being tangent to the inner surface of the outer shroud. The outer surface of the inner shroud then corresponded to a reading of 8.15 centimeters with the probe taps tangent to this surface. The annulus width was 3.2 centimeters at all points of the annulus.

To locate the probe around the annulus with respect to the nozzle, the backing ring was scribed off in degrees.

The centerline of the nozzle at its exit corresponds to ϕ = 0 degrees, and ϕ increases in the clockwise direction as the turbine is viewed from the exhaust side. (see figure 6.4) The turbine rotates counterclockwise when viewed in this manner.

3.2. High Speed Ratio Runs ($n = .347$)

To achieve a speed ratio of .347, the nozzle exit velocity was adjusted to 490 fps, which corresponded to a turbine rpm of 3540 giving a mean blade speed of 170 fps. The pressure drop across the converging section at these conditions was approximately 1.39 inches of mercury and the total upstream pressure was approximately 17.6 psia at 72 degrees F. After suitable warm-up time, readings were taken at one degree intervals of rotation about the axis of the turbine at traverse positions of the annulus corresponding to readings of 8.95, 9.75, and 10.55 centimeters on the scale. These readings are 1/4, 1/2, and 3/4 of the blade height respectively. For convenience, they will be referred to as the root position, center position and the tip position of the blade respectively in the remainder of this report.

3.3. Low Speed Ratio Runs ($n = .210$)

In this set of runs the nozzle exit velocity was 434 fps, which corresponded to a turbine rpm of 1900 giving a mean blade speed of 91 fps. The pressure drop across the converging section was 1.05 inches of mercury and the total upstream pressure was 16.77 psia at 72 degrees F. In this series of runs data was taken at one degree intervals of rotation and at radial positions corresponding to root, center, and tip position of the blades as defined in (3.2).

4. RESULTS

The data for the two series of runs is broken down and plotted in figures 6.5 through 6.10. Figures 6.5 and 6.6 show the spread of the jet as it comes out the blade exit. There is a marked speed effect along the blade height that is illustrated by the "lead" in the direction of rotation of the tip profile compared to the root profile. In the high speed ratio runs, the blade speeds from root to tip are 161, 170, and 179 fps, where root refers to 1/4 blade height and so forth.

Looking at the tangential velocity profiles (figures 6.7 and 6.8) there is a complete change in direction for the runs at high speed compared to the low speed runs. This is what would be expected when the velocity of the nozzle flow is kept almost constant and the turbine is doubled by decreasing the load. The output work of the turbine, neglecting the windage loss, is just the change in the tangential momentum of the jet per unit time multiplied by the turbine speed. Therefore, by picking mean values of exit tangential velocity off of figures 6.7 and 6.8 and subtracting these values from their respective inlet components, the work done per pound of mass of jet flow can be calculated (figure 6.11). This times the mass flow of the jet gives the work per unit time done by the jet. In the case of the high speed runs, this work is 234 foot-lbs. per second and for the low speed ratio runs, 116 foot-lb. per second is the output of the jet.

From these values, the windage work must be subtracted to obtain the output of the turbine. From Wennerstrom (2), the corrected windage loss for the high speed runs is

$$Work_{corr} = \left(\frac{3540}{3530}\right)^3 (70.0) = 70.6 \text{ ft.-lb./sec.}$$

and for the low speed runs is

$$Work_{corr} = \left(\frac{1900}{3530}\right)^3 (70.0) = 10.85 \text{ ft.-lb./sec.}$$

Subtracting these values from the jet work, the turbine output work for the high speed runs is 153.4 ft.-lb./sec. The low speed work is 105.2 ft.-lb. sec.

To find the efficiency of the turbine in these two speed ranges the output work must be divided by the input energy or the ideal kinetic energy of the jet (figure 6.11.). The values for the efficiency calculated in this manner are .327 for the high speed runs and .344 for the low speed runs.

However when the efficiency is calculated using Stenning's formula (1), the values are considerably higher, .491 for the high speed runs and .385 for the low speed runs. One of the probable reasons for this discrepancy is that Stenning's formula was derived using data obtained from low aspect ratio turbines, in which the windage work was neglible. This can be checked out by neglecting the windage work as predicated by Wennerstrom (2) and calculating the efficiencies. In this case the values are .498 for the high speed runs and .380 for the low speed runs. These values show excellent correlation, which checks out Stenning's formula for pumping and mixing losses and filling and emptying losses.

One source of error in using a yaw probe is that in a pressure gradient, the probe must point into the pressure gradient to balance the yaw taps, which give a slight angle deviation. This was checked using Lima's formula (3),

$$\Delta\alpha = -25\,\frac{s}{2}\left(\frac{\frac{\partial p_o}{\partial y}}{p_o - p}\right)$$

s = distance between yaw taps (inches)

$\frac{\partial p_o}{\partial y}$ = pressure gradient (inches water/inch)

and found to have a value of 4 degrees at the edges of the jet. Also, this error was in an opposite direction at one edge as compared with the other. In the central section of the jet the error was only about one degree. Consequently, this was neglected in subsequent runs.

5. CONCLUSIONS

The following conclusions are drawn from this study:

(1) For the turbine and under the conditions which it was run, Stenning's formula for predicting the losses due to pumping and mixing and due to filling and emptying has excellent correlation. (See figure 6.11 and 6.12)

(2) To be able to predict overall efficiency, a windage loss term must be added. This would have the form of some coefficient times the speed ratio cubed. This coefficient would be dependent upon the geometry of the turbine.

(3) Obviously, this relationship has to be checked out on many other types of partial admission turbines, using various speed ratios before it can be relied upon.

6. APPENDIX

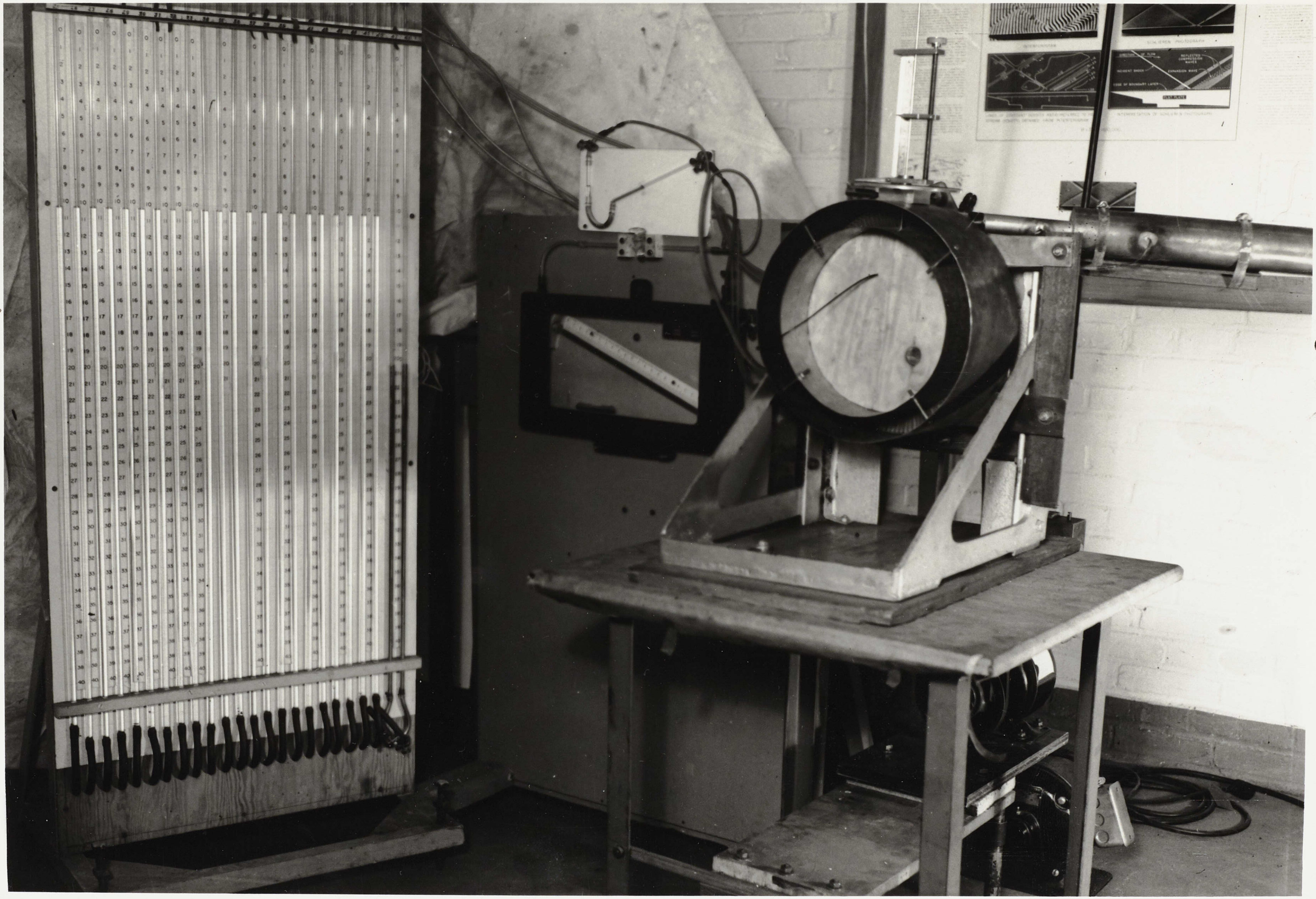


figure 6.1.

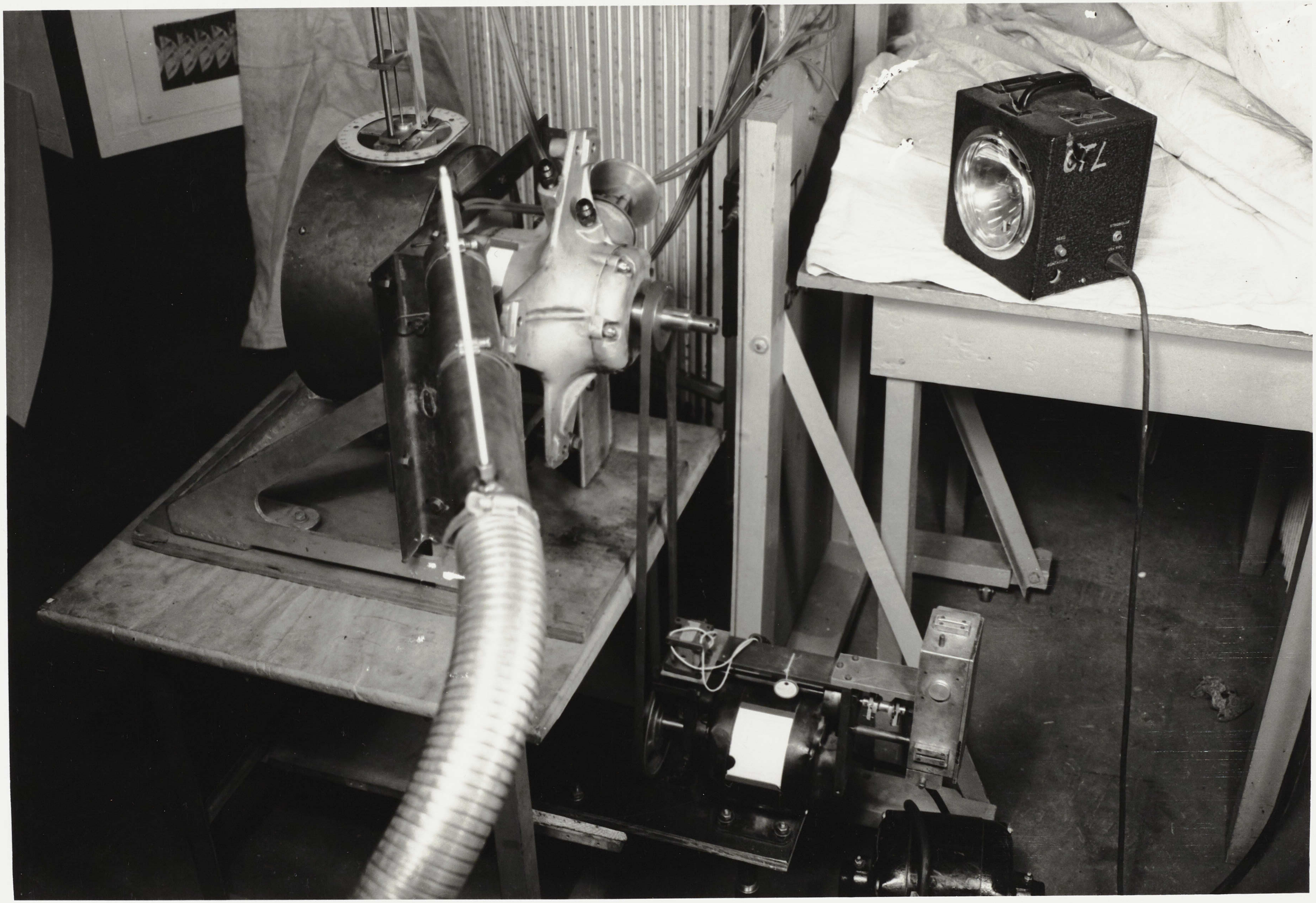


figure 6.2.

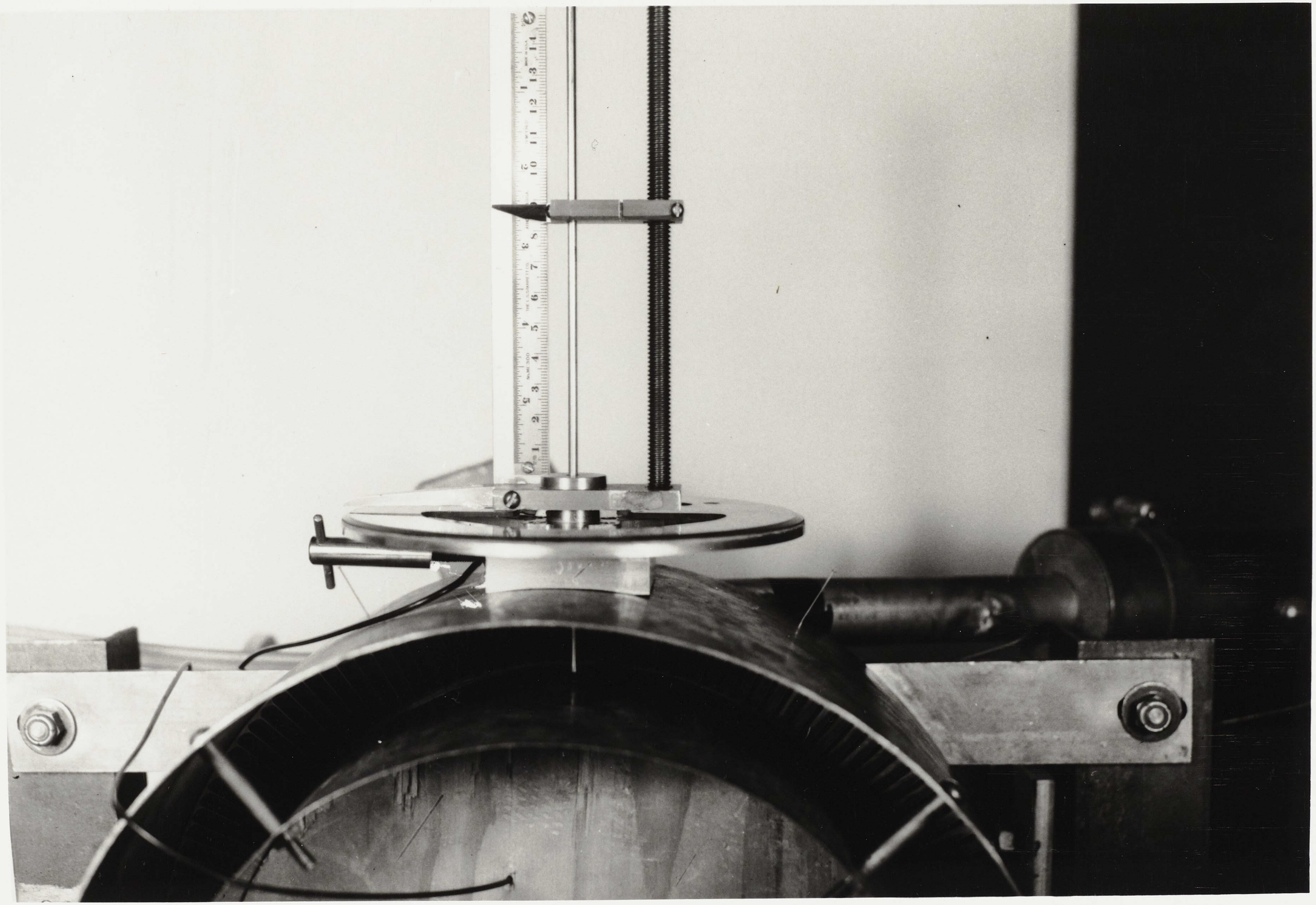


figure 6.3.

figure 6.4.

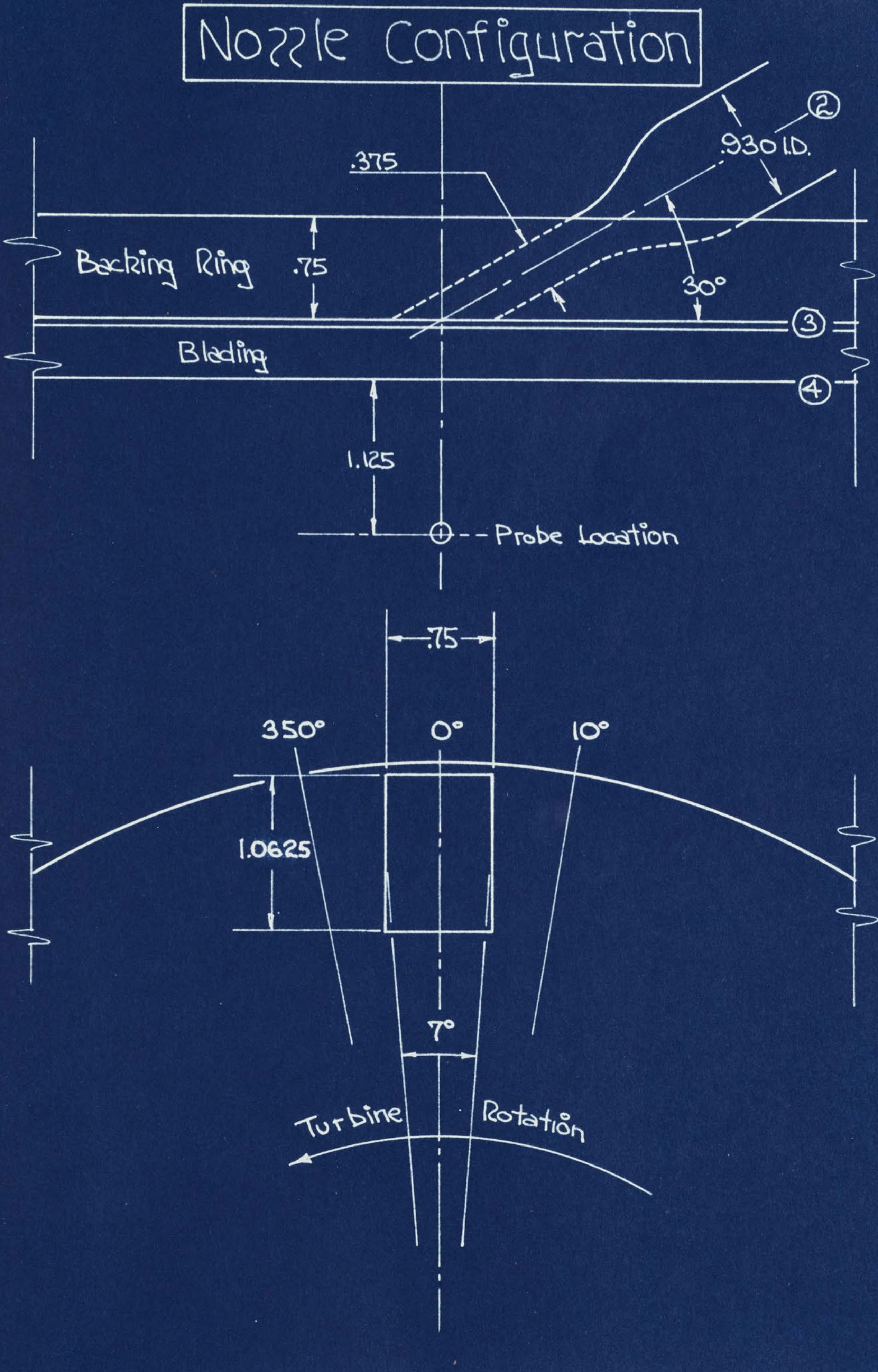

Nozzle Configuration
.375
.930 I.D.
2
Backing Ring
.75
30°
3
Blading
4
1.125
Probe Location
.75
350°
0°
10°
1.0625
7°
Turbine Rotation

figure 6.5.

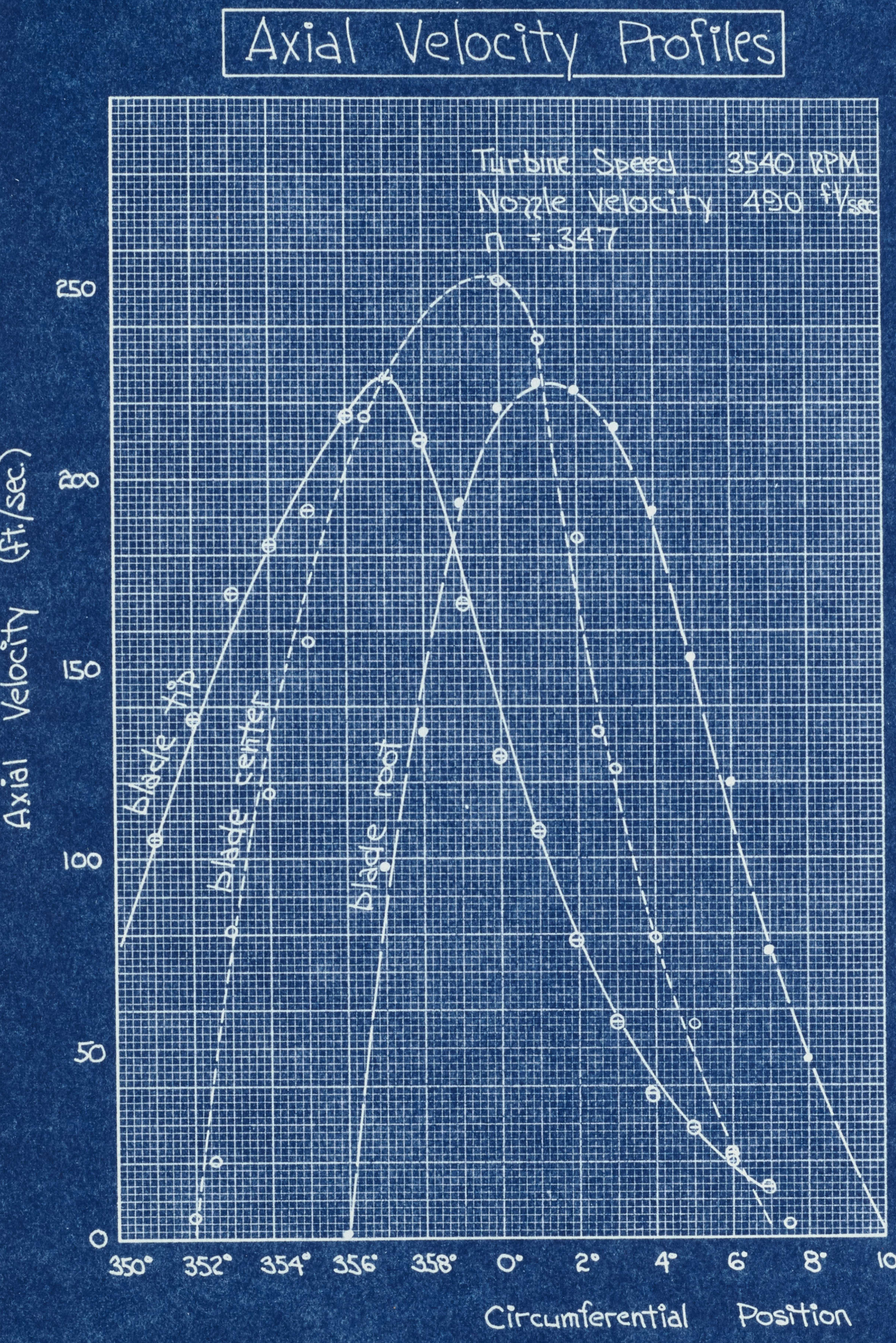

Axial Velocity Profiles
Turbine Speed 3540 RPM
Nozzle Velocity 490 ft/sec
n = .347
blade tip
blade center
blade root
Axial Velocity (ft./sec.)
0
50
100
150
200
250
350°
352°
354°
356°
358°
0°
2°
4°
6°
8°
10°
Circumferential Position

figure 6.6.

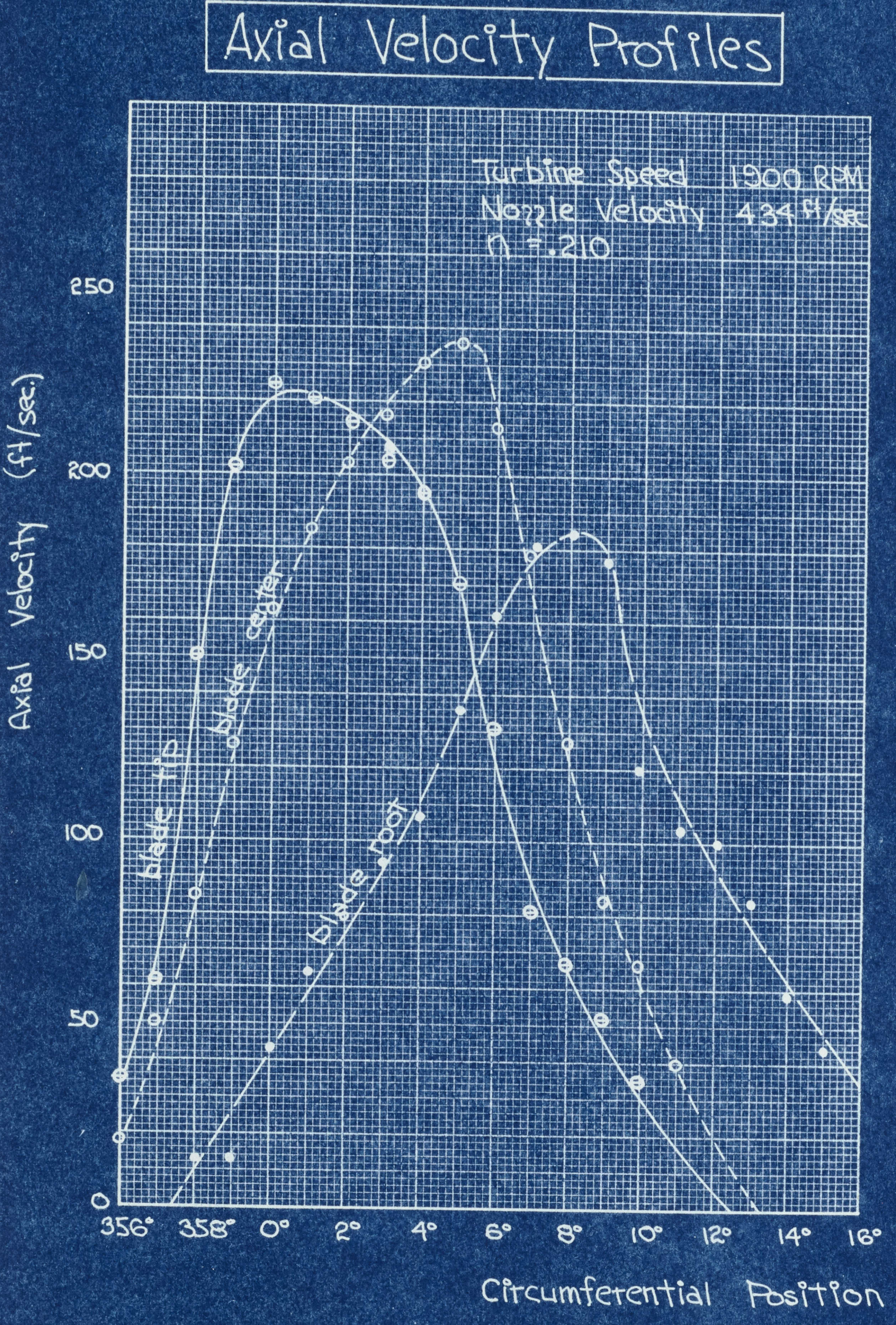
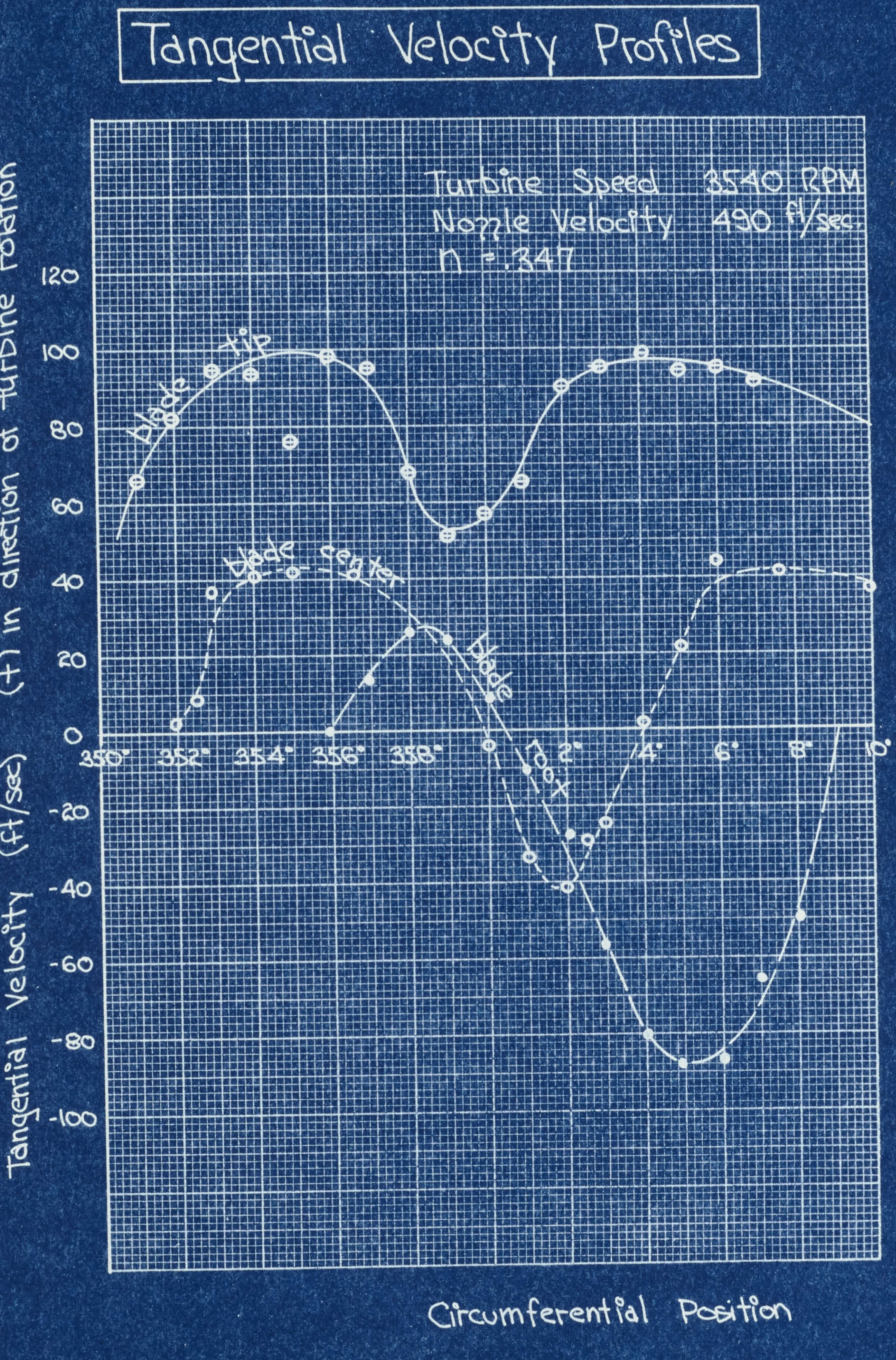

Axial Velocity Profiles
Turbine Speed 1900 RPM
Nozzle Velocity 434 ft/sec
η = .210
Axial Velocity (ft/sec.)
250
200
150
100
50
0
356°
358°
0°
2°
4°
6°
8°
10°
12°
14°
16°
Circumferential Position
blade tip
blade center
blade root

figure 6.7.



Tangential Velocity Profiles
Turbine Speed 3540 RPM
Nozzle Velocity 490 ft/sec.
n = .347
blade tip
blade center
blade root
120
100
80
60
40
20
0
-20
-40
-60
-80
-100
350°
352°
354°
356°
358°
2°
4°
6°
8°
10°
Tangential Velocity (ft/sec) (+) in direction of turbine rotation
Circumferential Position

figure 6.8.

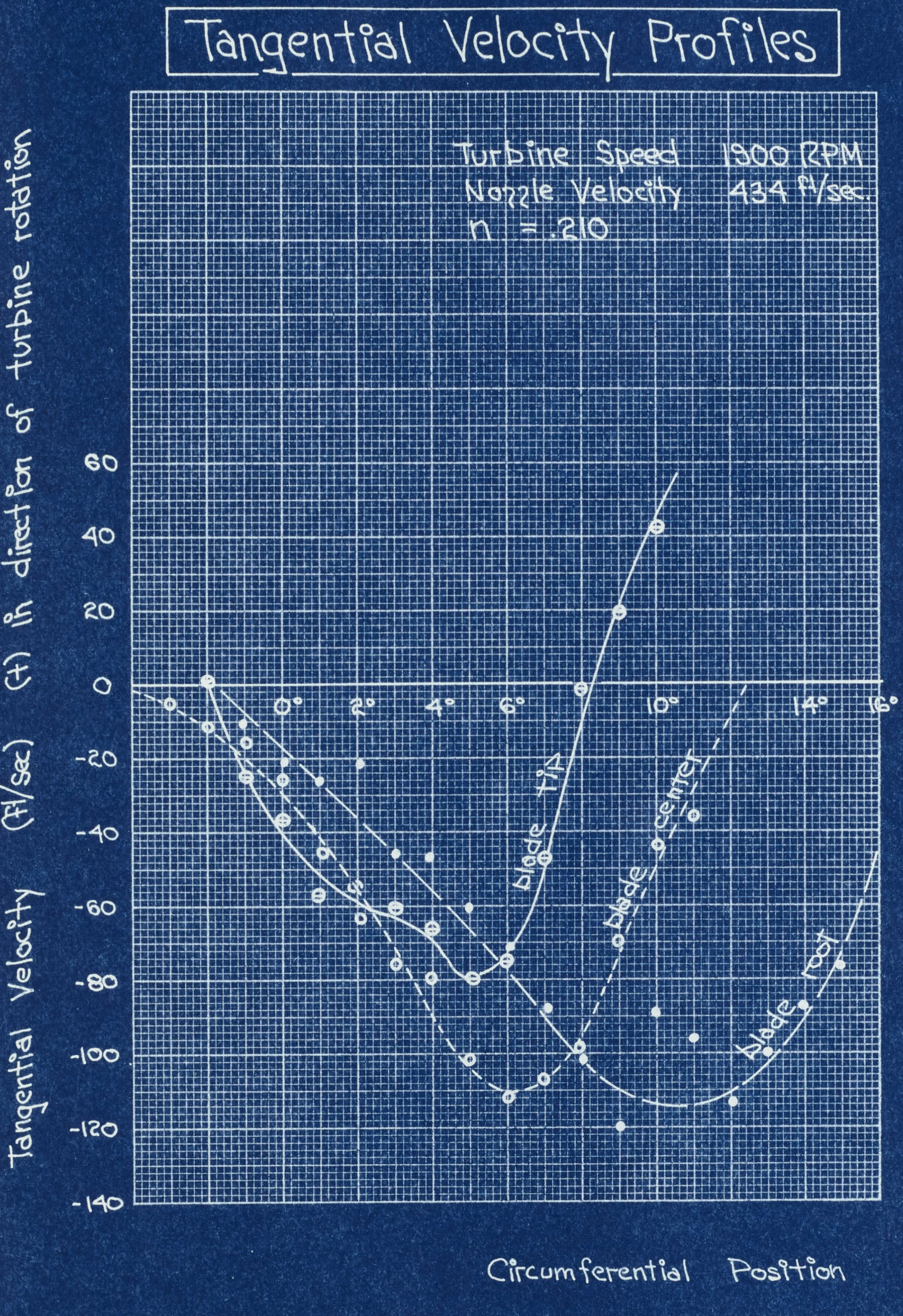

Tangential Velocity Profiles
Turbine Speed 1900 RPM
Nozzle Velocity 434 ft/sec.
n = .210
blade tip
blade center
blade root
Tangential Velocity (ft/sec) (+) in direction of turbine rotation
60
40
20
0
-20
-40
-60
-80
-100
-120
-140
0°
2°
4°
6°
10°
14°
16°
Circumferential Position

figure 6.9.

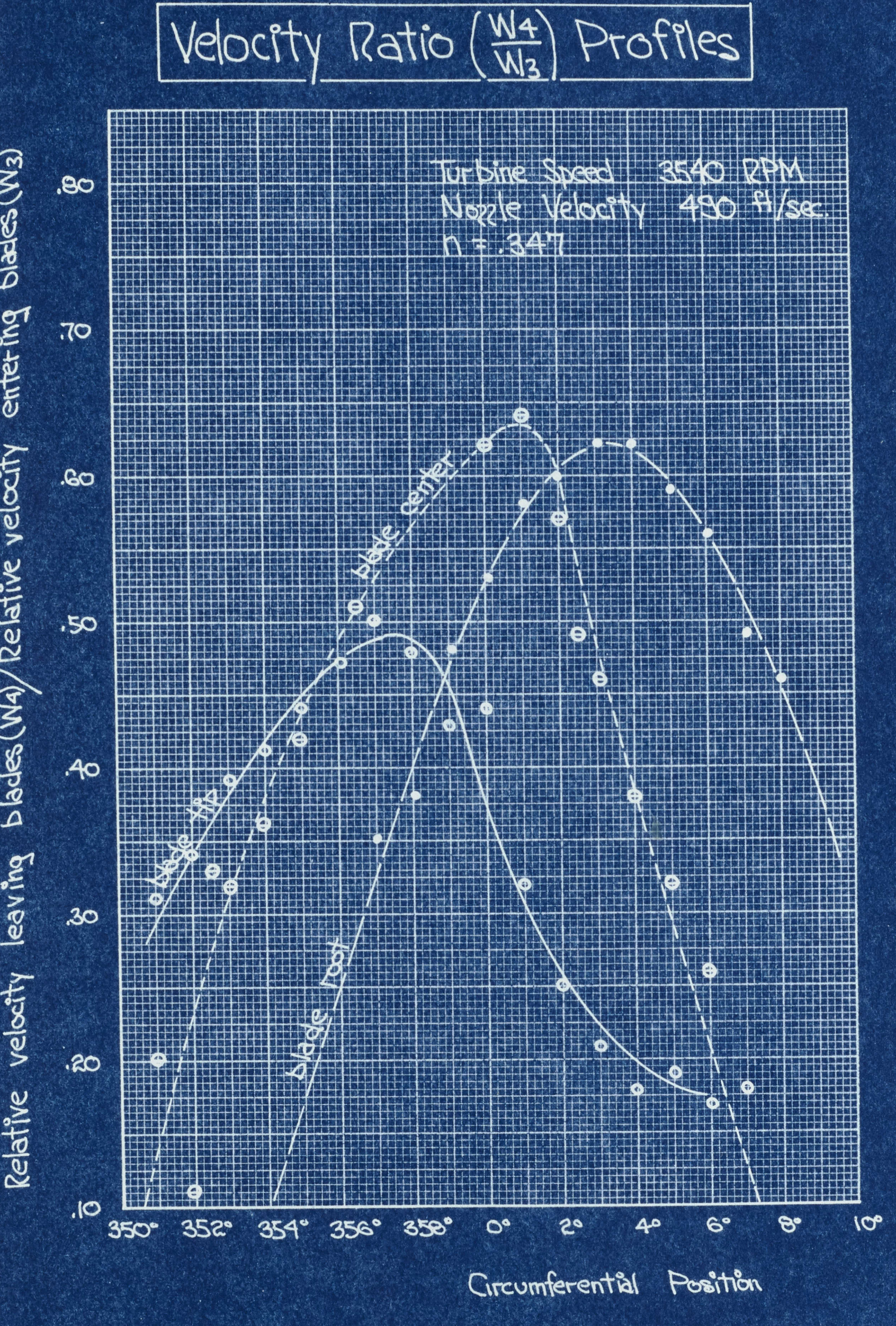
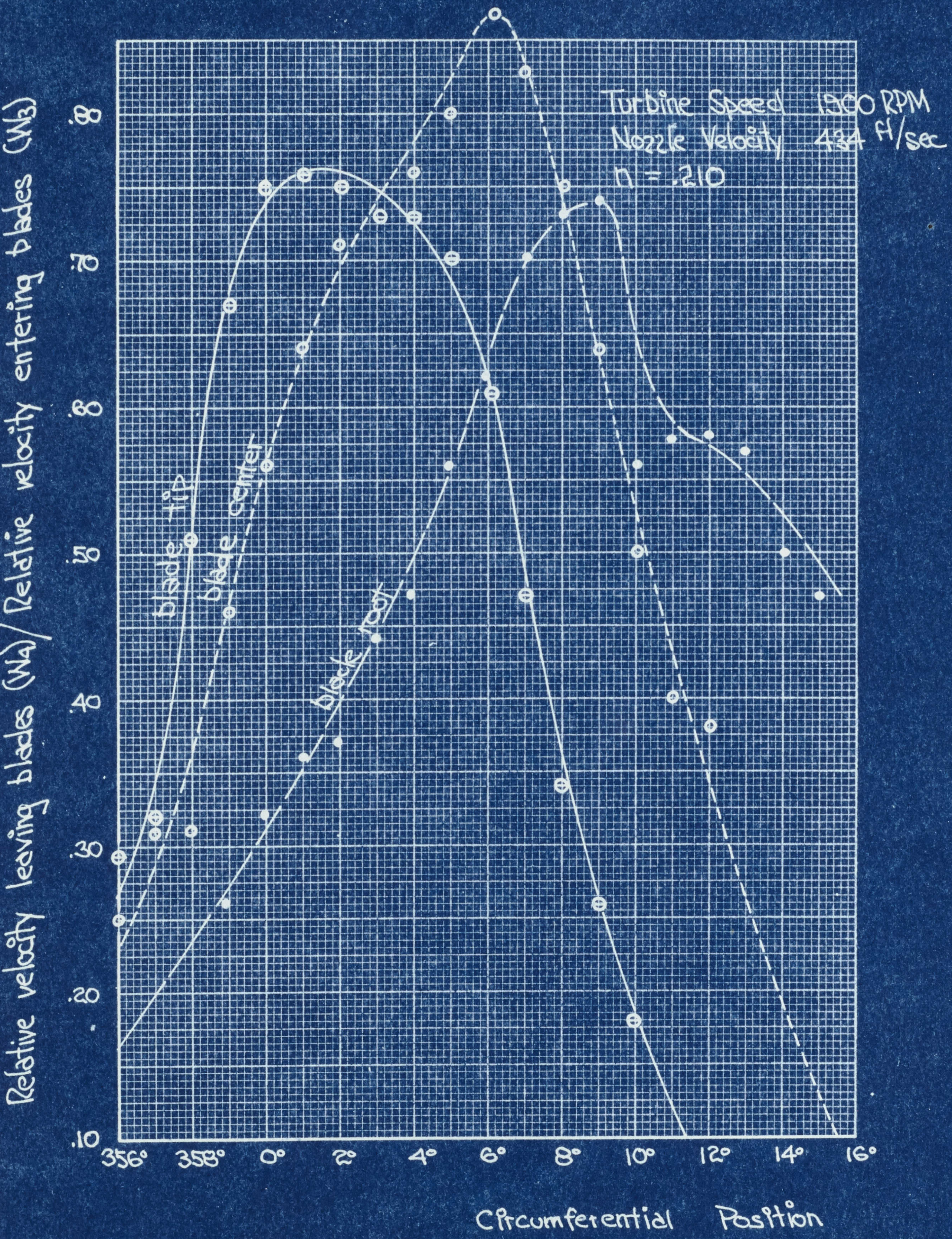

Velocity Ratio ($\frac{W_4}{W_3}$) Profiles
Turbine Speed 3540 RPM
Nozzle Velocity 490 ft/sec.
n = .347
blade tip
blade center
blade root
Relative velocity leaving blades (W4)/Relative velocity entering blades (W3)
.10
.20
.30
.40
.50
.60
.70
.80
350°
352°
354°
356°
358°
0°
2°
4°
6°
8°
10°
Circumferential Position

figure 6.10.

Velocity Ratio $\left(\frac{W_4}{W_3}\right)$ Profiles



6.11. Calculated efficiency for high speed run

turbine speed = 3540 rpm
blade speed = 170 fps
nozzle velocity = 490
tangential noz. vel. = 490 x cos 30° = 424 fps
mean tangential exit velocity (fig 6.7.) = +32 fps

$$Z = \frac{\text{blade speed}}{g_0}\left[V_3 \cos 30° - V_4 \cos \Theta\right]$$

$$= \frac{170}{32.2}\left[424 - 32\right] = 2070 \ \frac{\text{ft-lb}}{\text{lbm}}$$

$$\dot{m} = V_1 A_1 \rho_1 = \frac{41(4.43).09}{144} = .113 \ \frac{\text{lbm}}{\text{sec}}$$

Output = $Z \times \dot{m}$ - windage loss

From Wennerstrom (2)

$$\text{Windage loss Work} = \left(\frac{3540}{3530}\right)^3 70 = 70.6 \ \frac{\text{ft-lb}}{\text{sec}}$$

Therefore; Output = (2070)(.113) - 70.6

$$= 234 - 70.6 = 153.4 \ \frac{\text{ft-lb}}{\text{sec}}$$

$$\eta = \frac{\text{output}}{\text{K.E. of jet}} = \frac{Z \times \dot{m} - \text{wind loss}}{\dot{m} c^2 / 2 g_0 \eta_{noz}}$$

$$\eta = \frac{153.4}{.113(490)^2 / 64.4(.90)} = .327$$

$$\eta \text{ (neglecting windage loss)} = \frac{234}{153.4}(.327) = .498$$

6.12. Efficiency as predicted by Stenning (2)

$$\eta_{tp} = \frac{1 + K\left(1 - \frac{s}{a}\right)}{1+K}\eta_t - \frac{1.4\, n^3 c\, p}{a \sin\alpha}\eta_n$$

K = ratio of velocity leaving wheel to velocity entering wheel for full admission $\approx .86$
s = blade spacing $\sim .020$ ft.
a = nozzle arc width $\sim .0625$ ft.
η_t = turbine efficiency at full admission $\approx .65$ for $n = .347$
n = velocity ratio (wheel speed / nozzle velocity)
c = wheel disk width $\sim .035$ ft.
p = leakage factor ~ 1
η_n = nozzle efficiency $\sim .90$
α = nozzle angle $\sim 30°$

$$\eta_{tp} = \frac{1 + .86\left(1 - \frac{.020}{.0625}\right)}{1 + .86}(.65) - \frac{1.4(.347)^3(.035)\,1(.90)}{.0625 \sin 30°}$$

$$\eta_{tp} = .491$$

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(3) Lima, H., Yaw Probes in a Total Pressure Gradient. Thesis (S.M.), Mass. Inst. of Tech., Dept of Mech. Engr., January, 1958.