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# DRIVE TURBINE SYSTEMS FOR 20-INCH TURBOFAN SIMULATORS

# I - DUCT TURBINE DESIGN

by Warren J. Whitney
Lewis Research Center
Cleveland, Ohio

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## SUMMARY

A study was made to evolve the turbine drive systems for 20-inch turbofan engine simulators. The fan designs used in the simulators included single-stage and two-stage configurations that covered a wide range of rotative speed and power requirement. The objective assumed for the study was to evolve one core turbine design that could drive all of the single-stage fans and, when operated in combination with one duct turbine design, drive all of the two-stage fans. The duct turbine power output is then needed to determine the make-up power required of the core turbine over the range of two-stage fan operating conditions.

This report is concerned with the duct turbine and includes the selection of the duct turbine velocity diagram, a description of the blade design, and a determination of its off-design performance. Adjustable stators were found to be quite advantageous to the duct turbine off-design operation. The use of adjustable stators enabled the duct turbine to accomodate fan mass flow at all operating points and caused the duct turbine power output to increase as the total power requirement increased. This in turn resulted in a core turbine make-up power requirement that was not significantly greater than that required for driving the single-stage fans.

## INTRODUCTION

A number of research projects that are currently planned at the NASA-Lewis Research Center involve the use of fan engine simulators. The rotating elements of the simulators consist of a fan and a turbine drive system on a common shaft. A source of high pressure air at moderate temperature is available at the test site for the drive turbine system. The fan designs for the simulators considered herein are all 20-inch tip diameter and include both single-stage and two-stage configurations. Their operating conditions span a wide range of tip speed, pressure ratio and power requirement. The turbine problem is to develop the necessary power over the speed range as required by the fan.

The purpose of this report is to document and describe the study to evolve the drive turbine configurations. It was desirable to meet the fan power-speed requirements with a minimum number of turbine configurations. It was also desirable to use existing turbine designs or modifications of existing designs if possible. This report discusses the overall problem and approach to its solution. Included is a description of the duct turbine design, the duct turbine off-design operation and the blading layout for the duct turbine.

## SYMBOLS

| | |
|---|---|
| $i$ | incidence angle, deg. |
| p | absolute pressure, atmospheres |
| U | blade velocity, ft/sec |
| V | absolute gas velocity, ft/sec |
| W | gas velocity relative to moving blade row, ft/sec |
| w | mass flow rate, lb/sec |
| $\alpha$ | absolute gas flow angle, measured from axial direction, deg |
| $\beta$ | gas flow angle relative to moving blade row, measured from axial direction, deg |
| $\eta$ | efficiency |
| $\rho$ | gas density, $lb/ft^3$ |
| $\phi$ | blade orientation or stagger angle, deg |

Subscripts

| | |
|---|---|
| cr | condition at Mach 1 |
| m | mean radius |
| u | tangential component |
| 0 | station at fan inlet, see figure 2 |
| 1 | station at fan outlet |
| 2 | station at duct turbine inlet |
| 3 | station at duct turbine stator outlet |
| 4 | station at duct turbine outlet |

Superscripts

| | |
|---|---|
| ' | total state |

## DRIVE SYSTEM REQUIREMENTS

The turbine drive system requirements are specified by the fan power-speed envelope as shown in figure 1. The mid-point of the speed range at each pressure ratio is denoted by a symbol for reference. All of the fans are seen to operate over a fairly narrow range of speed from ±.136 mid-range speed at a pressure ratio of 1.25 to ±.083 mid-range speed at a pressure ratio of 3. The diamond point on the figure is a two-stage fan with a 2.4 pressure ratio representative of the type of two-stage fan currently of most interest. The fan with a pressure ratio of 3 is considered to be a likely future development. The power requirements for the two-stage fans are considerably different than for the single-stage fans, figure 1. In fact the lowest requirement for a two-stage configuration is nearly twice the highest power requirement for the single-stage fans. Because of this difference and the difference in available expansion energy at the fan outlet, the turbine drive systems for the two-stage fans and single-stage fans were considered separately.

## DRIVE SYSTEM CONCEPTUAL LAYOUTS

A sketch of the drive turbine systems is shown in figure 2. The single-stage fans are driven by a multi-stage core turbine, so called because it is contained within the center body or core of the fan. The two-stage fans are driven by a duct turbine, located in the fan duct, in addition to a core turbine. An optimum turbine drive system would be one in which one core turbine design would drive all the single-stage fans; and, when used with one duct turbine design, would drive all the two-stage fans. It is necessary to evolve the duct turbine design and estimate its off-design performance to define the core turbine requirements when operated with the duct turbine to drive the two-stage fans. The duct turbine design is therefore considered first herein.

## DUCT TURBINE DESIGN

This section will include the physical limitations; design point selection and velocity diagram; off-design performance; and blade section layout.

### Physical Considerations

The size limitations imposed on the duct turbine design were:

Maximum tip 'iameter . . . . . . . . . 20 inches
Minimum hub ameter . . . . . . . . . 10 inches

These limits were adopted to be consistent with the fan geometry, figure 2. It was required to keep the turbine tip diameter within that of the fan (20 inches) and also to provide a minimum hub diameter of 10 inches to accomodate the core turbine and its supply plenum.

### Design Velocity Diagram

The operating point of the 2.4 fan shown on figure 1 was selected as the duct turbine design point. This is because there is the most immediate interest in this two-stage fan design. This point is, although, very near the center of the two-stage fan power-speed envelope.

The first step in the evolution of the duct turbine design is the selection of the turbine pressure ratio. The turbine operates between the fan outlet total pressure and atmospheric ambient pressure. The other factors that effect the turbine total pressure ratio are: (1) total pressure loss between the fan and the duct turbine, (2) turbine efficiency, and (3) the turbine outlet flow condition which specifies the outlet static-to-total pressure ratio. It was determined from the blade speed and specific work level that the turbine would be a single-stage, lightly loaded (low $\Delta V_u/U$ value) configuration with axial outlet flow. It was desirable from a facility consideration to have a moderate level of velocity ($(V/V_{cr})_4$ from .5 to .6) leaving the turbine. A 3 percent total pressure loss ($p_2' = .97p_1'$) was assumed between the fan and the turbine and an efficiency of .90 was assumed (due to the low stage loading) for the duct turbine. The turbine outlet conditions were then determined for a range of turbine pressure ratio. These results are shown in figure 3.

The two conditions that must be satisfied at the turbine outlet (fig. 3) are specific mass flow rate and static pressure. The highest pressure ratio that met both of these conditions would then be desirable since this represents the largest work extraction. It was however also thought desirable to allow a safety margin of 4 to 5 percent of excess pressure at the outlet. With this in mind a turbine pressure ratio, $p_2'/p_4'$, of 1.85 was selected. At this pressure ratio the specific mass flow rate requirement is attained at an outlet critical velocity ratio $(V/Vcr)_4$ of .554 and the corresponding outlet static pressure $p_4$ is 1.048 atmospheres.

All the quantities needed to construct the velocity diagram are known when the pressure ratio is selected. The velocity diagram is shown in figure 4. The low ratio of specific work output to blade energy level can be noted. The average (mean radius) stage loading factor, $(\Delta Vu/U)_m$ was .512 which is very low. If the turbine had extracted a specific work equal to that of the fan, the average stage loading factor would have been .892 which is still quite low. The stator blade for this turbine had a moderate velocity level, normal turning angles and presents no unusual problems. The rotor blade however is seen to operate with very high relative velocities and very low turning angles. These two features result from the high blade speed and the low stage loading factor. The power developed at the design operating condition was 2498 horsepower. This leaves a make-up power of 1852 horsepower to be supplied by the core turbine.

## Off Design Performance

In addition to the design point, the behavior of the duct turbine is of interest over the entire speed-power range of the two-stage fans. It is desirable that the duct turbine accomodates the fan mass flow and converts the available pressure drop to useful work output. The two points initially considered were the mid-range speeds of the 2.0 and 3.0 pressure ratio fans. At each point the rotative speed, fan mass flow, and turbine inlet total state conditions were imposed on the fixed geometry turbine designed for the 2.4 pressure ratio fan. At the fan pressure ratio of 3 and fan tip speed of 1625 feet per second, it was found that the turbine work output was negative for these imposed conditions. Since the actual mass flow is constant as shown on figure 1, and the turbine inlet total state is considerably increased at a fan pressure ratio of 3, the stator outlet velocity had to be decreased. This, in combination with the increased blade speed, would have caused the turbine to operate at a speed exceeding runaway speed, and the turbine work output would have been negative. When the fan outlet conditions for the pressure ratio of 2 fan were imposed on the turbine, the turbine rotor choked at about 70 percent of fan mass flow. This would require that 30 percent of the fan mass flow would have to be diverted overboard between the fan and the duct turbine and this would be intolerable for many of the research tests.

It was indicated from the foregoing that if one duct turbine design could operate successfully over the two-stage fan power-speed spectrum, it would require adjustable stator blades. This scheme was then applied to the off-design operating points. The approach was to determine the value of stator tangential velocity ratio, $(Vu/Vcr)_{3m}$ that caused the rotor to be choked at the known fan mass flow rate. This procedure was two-dimensional in that the mean radius conditions were used to represent the blade row. The resulting blade angles, $\alpha_{3m}$, are listed in Table I. Compared to the initial design angle the adjustment must reduce the outlet flow angle by 28.56° and increase it 11.56°. This adjustment to flow angle does not cause a corresponding change in relative blade entry angle as shown in figure 5. The greatest loss caused by incidence angle was less than 1 percent of the total pressure relative to the rotor blade in all cases considered herein.

The power developed by the duct turbine using adjustable stators is shown as the upper shaded areas in figure 6. The duct turbine develops an increasing amount of power with increasing fan pressure ratio. Although this is not unusual since the potential of the duct turbine has been increased, it is significant that the adjustable stator has enabled the duct turbine to assume a greater portion of the load as total power requirement increases. The make-up power required of the core turbine is fairly well evened out over the speed range as seen by the lower shaded areas of figure 6. The greatest core turbine power requirement, for the pressure ratio of 3 fan, was 2030 horsepower which is only 19 percent more than required for the highest powered single stage fan. Thus, the use of adjustable stators is seen to be an effective means of obtaining a suitable power production from the duct turbine over the range of operating conditions. The core turbine will, however, still be confronted with the problem of developing a given level of power over a wide range of speed. This problem will be considered in a subsequent study.

## Duct Turbine Blade Section Layout

The turbine blade passages were laid out for the fixed geometry turbine whose velocity diagrams are shown in figure 4. The off-design stator settings would require a device to change the stator blade orientation angle by about the amount of the required change of stator outlet flow angle. As mentioned in an earlier section, the stator presented no unusual problems with regard to turning or velocity level. This was true as well for the off-design stator settings. Thus it appeared feasible to adapt the stator blade of reference 1 to this application. The blade was first made thinner by translating the suction surface about .040" to reduce the trailing edge thickness from .070" to .030". The modified blade was then scaled to .9 of reference 1 size and the number of blades adjusted to attain the same mean radius solidity as reference 1. The stator blade outlet throat openings were determined at the hub, mean, and tip radii using the continuity and angular momentum relationships as described in reference 1. The orientation angle was then specified at the three radii to achieve these throat openings. A total of 32 blades were required for the stator assembly. The stator blade coordinates are listed in Table II.

The rotor blade was designed specifically for this application. Because of its extreme features, a similar rotor blade design could not be found which could be modified for this application. These features are; high blade speed, low stage load factor, and low radius ratio. They result in a blade design that does very little turning and operates with a lot of overall reaction. Over most of the blade span the rotor channel is required to converge to the throat opening and allow the flow to expand downstream. In addition to the aerodynamic requirements the rotor blading design is effected by mechanical considerations due to the potentially high centrifugal stress. From a mechanical standpoint it is desired to (a) incorporate blade taper; that is, a reduction in blade cross-sectional area from hub to tip and (b) overlay or stack the blade sections such that their centroids lie on a radial line in order that the centrifugal force does not induce bending stress.

The blade channels were laid out and the surface velocities were then obtained for these channels by the method of reference 2. This method requires that the axial position of the sections be specified, therefore, the axial location of the section centroids had to be determined in the design procedure. The blades were laid out to obtain as much area reduction from hub to tip as appeared feasible. The coordinates of the rotor blading are given in Table III and the blade passages and profiles are shown in figure 7. A total of 39 blades were used for the rotor.

The rotor surface velocity distributions are shown in figure 8. The mean and tip sections are seen in the figure to operate at virtually zero diffusion. This is desirable and especially important at the tip where the blade outlet critical velocity ratio is 1.27. The hub section operates at a diffusion of 14 percent, however, the velocity level is lower at this section. The resulting blade cross-sectional areas at the mean and tip sections related to that at the hub were .514 and .275 respectively. This would result in a reduction of centrifugal blade stress to about .6 that of an untapered blade. This blading design then appears to offer a satisfactory solution to the mechanical as well as the aerodynamic requirements.

## CONCLUDING REMARKS

The use of adjustable stators appears to be quite advantageous for the duct turbine. One duct turbine design with adjustable stator vanes could operate satisfactorily over the power-speed range of the two-stage fans. The duct turbine so-equipped accomodated the fan mass flow in all cases and its power production increased as the total power demand increased. This caused the power requirement of the core turbine to be fairly well evened out over the speed range.

## REFERENCES

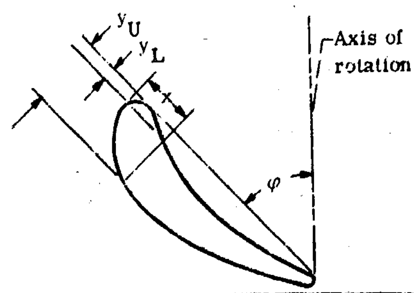
1. Whitney, Warren J.; Szanca, Edward M.; Moffitt, Thomas P.; and Monroe, Daniel E.: Cold-Air Investigation of a Turbine for High-Temperature Engine Application. I - Turbine Design and Overall Stator Performance. NASA TN D-3751, 1967.

2. Katsanis, Theodore; and Dellner, Lois T.: A Quasi-Three-Dimensional Method for Calculating Blade-Surface Velocities for an Axial Flow Turbine Blade. NASA TM X-1394.

TABLE I DUCT TURBINE STATOR OUTLET FLOW ANGLES, $\alpha_{3M}$, REQUIRED AT THE VARIOUS OPERATING POINTS

| FAN PRESSURE RATIO | LOWEST SPEED | MID RANGE SPEED | HIGHEST SPEED |
|---|---|---|---|
| 2.0 | 26.12° | 31.58° | 38.11° |
| 2.4 | 51.51° | 54.68° | 55.80° |
| 3.0 | 65.22° | 65.67° | 66.24° |

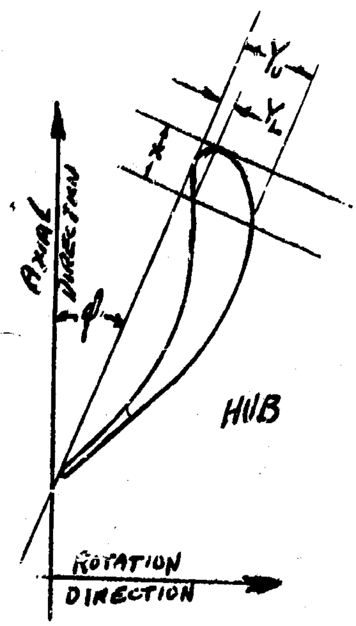
TABLE II Duct Turbine Stator Blade Coordinates

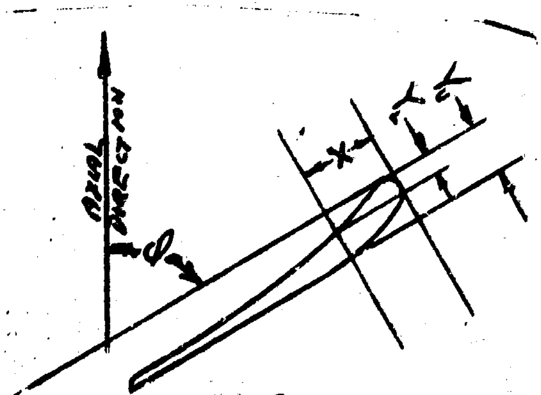



| x, in. | Hub | | Mean | | Tip | |
|---|---|---|---|---|---|---|
| | Orientation angle, $\varphi$, deg | | | | | |
| | 36° 38' | | 29° 24' | | 23° 14' | |
| | Diameter, inches | | | | | |
| | 10 | | 15 | | 20 | |
| | $y_U$, in. | $y_L$, in. | $y_U$, in. | $y_L$, in. | $y_U$, in. | $y_L$, in. |
| .000 | .117 | .117 | .117 | .117 | .117 | .117 |
| .045 | .227 | ---- | .245 | ---- | .257 | ---- |
| .090 | .292 | ---- | .319 | ---- | .344 | ---- |
| .135 | .348 | ---- | .378 | ---- | .407 | ---- |
| .180 | .392 | ---- | .423 | .019 | .454 | ---- |
| .270 | .457 | .058 | .488 | .061 | .518 | .063 |
| .360 | .498 | .096 | .529 | .099 | .558 | .102 |
| .450 | .522 | .129 | .553 | .131 | .579 | .134 |
| .540 | .534 | .157 | .563 | .157 | .586 | .161 |
| .630 | .534 | .177 | .562 | .176 | .580 | .183 |
| .720 | .526 | .193 | .552 | .189 | .567 | .195 |
| .810 | .511 | .203 | .535 | .197 | .545 | .204 |
| .900 | .490 | .207 | .512 | .201 | .522 | .208 |
| .990 | .468 | .205 | .486 | .199 | .497 | .206 |
| 1.080 | .442 | .201 | .459 | .193 | .468 | .201 |
| 1.170 | .412 | .191 | .429 | .124 | .441 | .193 |
| 1.260 | .381 | .176 | .398 | .172 | .410 | .180 |
| 1.350 | .345 | .158 | .362 | .157 | .315 | .165 |
| 1.440 | .307 | .138 | .324 | .139 | .339 | .147 |
| 1.530 | .265 | .115 | .283 | .120 | .301 | .126 |
| 1.620 | .220 | .093 | .242 | .100 | .261 | .105 |
| 1.710 | .173 | .067 | .195 | .077 | .218 | .086 |
| 1.800 | .120 | .041 | .148 | .054 | .173 | .063 |
| 1.890 | .064 | .014 | .096 | .030 | .126 | .040 |
| 1.959 | .014 | .014 | ---- | ---- | ---- | ---- |
| 1.980 | ---- | ---- | .041 | .005 | .076 | ---- |
| 2.023 | ---- | ---- | .014 | .014 | ---- | ---- |
| 2.077 | ---- | ---- | ---- | ---- | .014 | .014 |

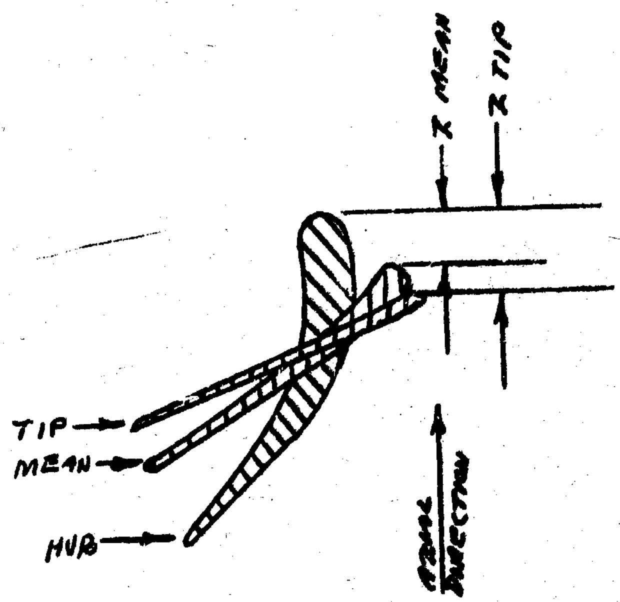
## TABLE III DUCT TURBINE ROTOR COORDINATES

| x, in. | Hub | | Mean | | Tip | |
|---|---|---|---|---|---|---|
| | Orientation angle, , deg | | | | | |
| | 22° 59' | | 59° 57' | | 68° 46' | |
| | Diameter, inches | | | | | |
| | 10 | | 15 | | 20 | |
| | $Y_U$, in. | $Y_L$, in. | $Y_U$, in. | $Y_L$, in. | $Y_U$, in. | $Y_L$, in. |
| .000 | .120 | .120 | .080 | .080 | .040 | .040 |
| .100 | ---- | ---- | .165 | ---- | .083 | .004 |
| .150 | ---- | ---- | .176 | ---- | .085 | .007 |
| .200 | .307 | .027 | ---- | ---- | .086 | .010 |
| .250 | .335 | .052 | .194 | .037 | .087 | .013 |
| .300 | .361 | .077 | .202 | .048 | | .016 |
| .350 | .383 | .101 | .208 | .058 | | .018 |
| .400 | .403 | .122 | .214 | .068 | | .021 |
| .450 | .421 | .143 | .220 | .078 | const. | .024 |
| .500 | .433 | .161 | .224 | .087 | | .026 |
| .550 | .443 | .178 | .227 | .096 | | .028 |
| .600 | .452 | .193 | .230 | .105 | | .031 |
| .650 | .456 | .206 | .232 | .113 | | .033 |
| .700 | .458 | .218 | .233 | .121 | | .035 |
| .800 | .455 | .235 | | .135 | | .039 |
| .900 | .444 | .247 | const. | .149 | | .042 |
| 1.000 | .424 | .252 | | .160 | | .045 |
| 1.100 | .396 | .249 | | .170 | | .048 |
| 1.200 | .361 | .242 | | .179 | | .051 |
| 1.300 | .320 | .226 | | .187 | | .053 |
| 1.400 | .277 | .203 | | .193 | | .054 |
| 1.500 | .230 | .175 | | .197 | | .055 |
| 1.600 | .182 | .139 | .233 | .201 | .087 | .056 |
| 1.700 | .134 | .098 | | .203 | | .057 |
| 1.793 | ----- | ----- | .218 | .218 | ---- | ---- |
| 1.794 | ----- | ----- | ----- | ----- | 072 | .072 |
| 1.800 | .087 | .054 | | | | |
| 1.900 | .040 | .006 | | | | |
| 1.933 | .015 | .015 | | | | |
| | Axial Displacement Dimension, z, inches | | | | | |
| | .000 | | .314 | | .429 | |




MEAN & TIP:
REF. LINE PARALLEL TO
AFT SUCTION SURFACE



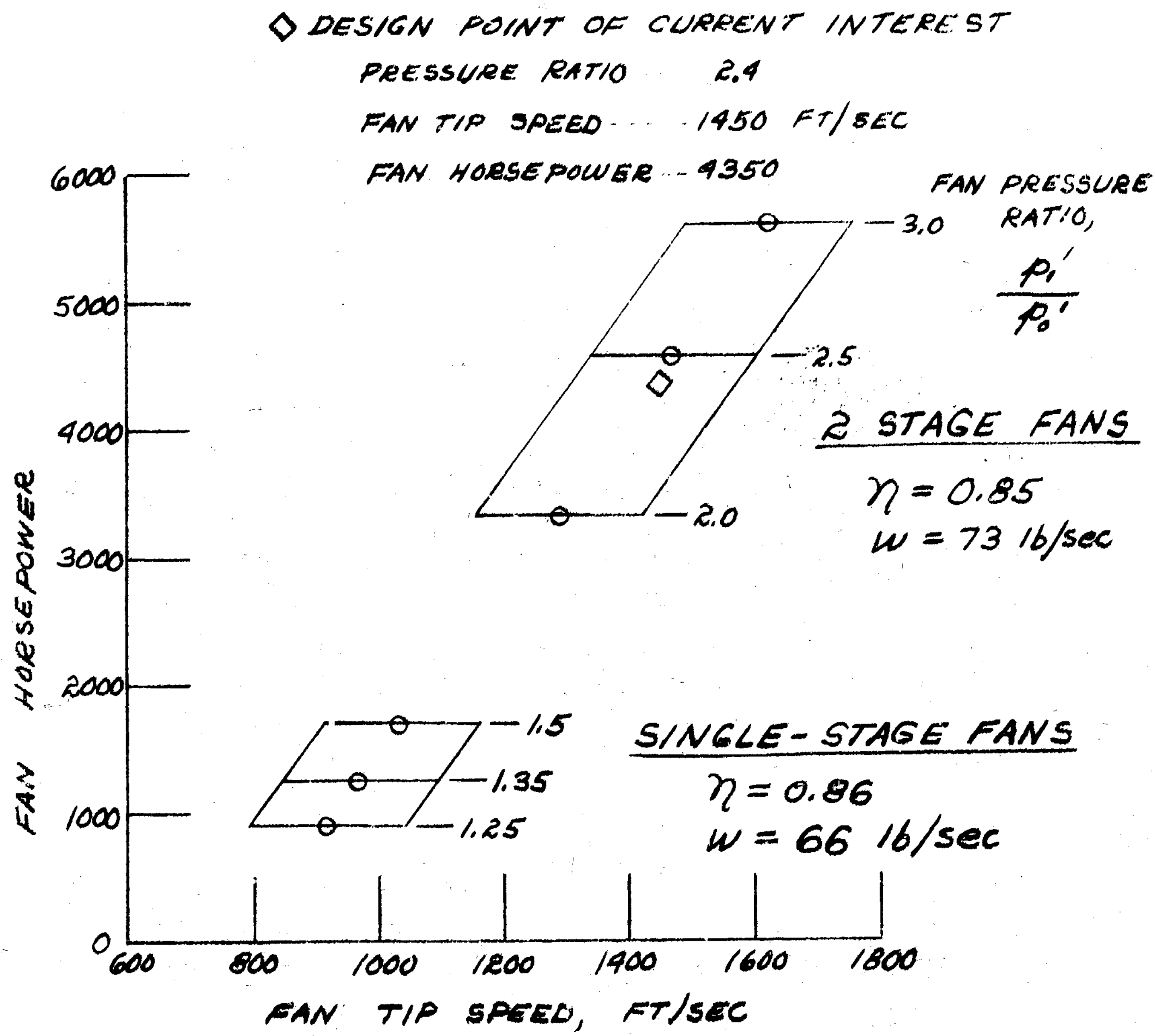



FIGURE 1.- POWER-SPEED ENVELOPES FOR SINGLE-STAGE AND 2-STAGE FANS. AMBIENT CONDITIONS — STANDARD ATMOSHERE

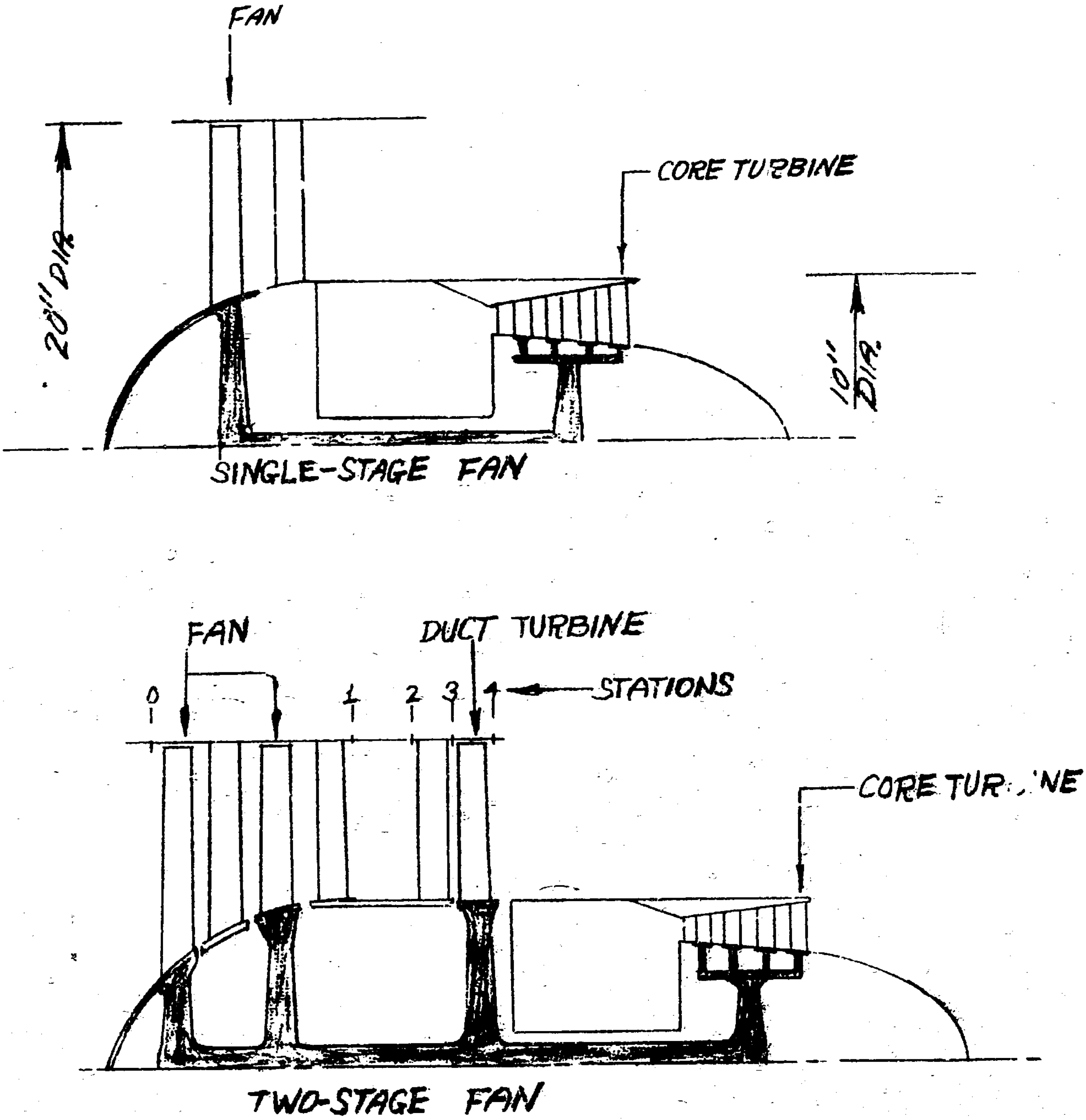
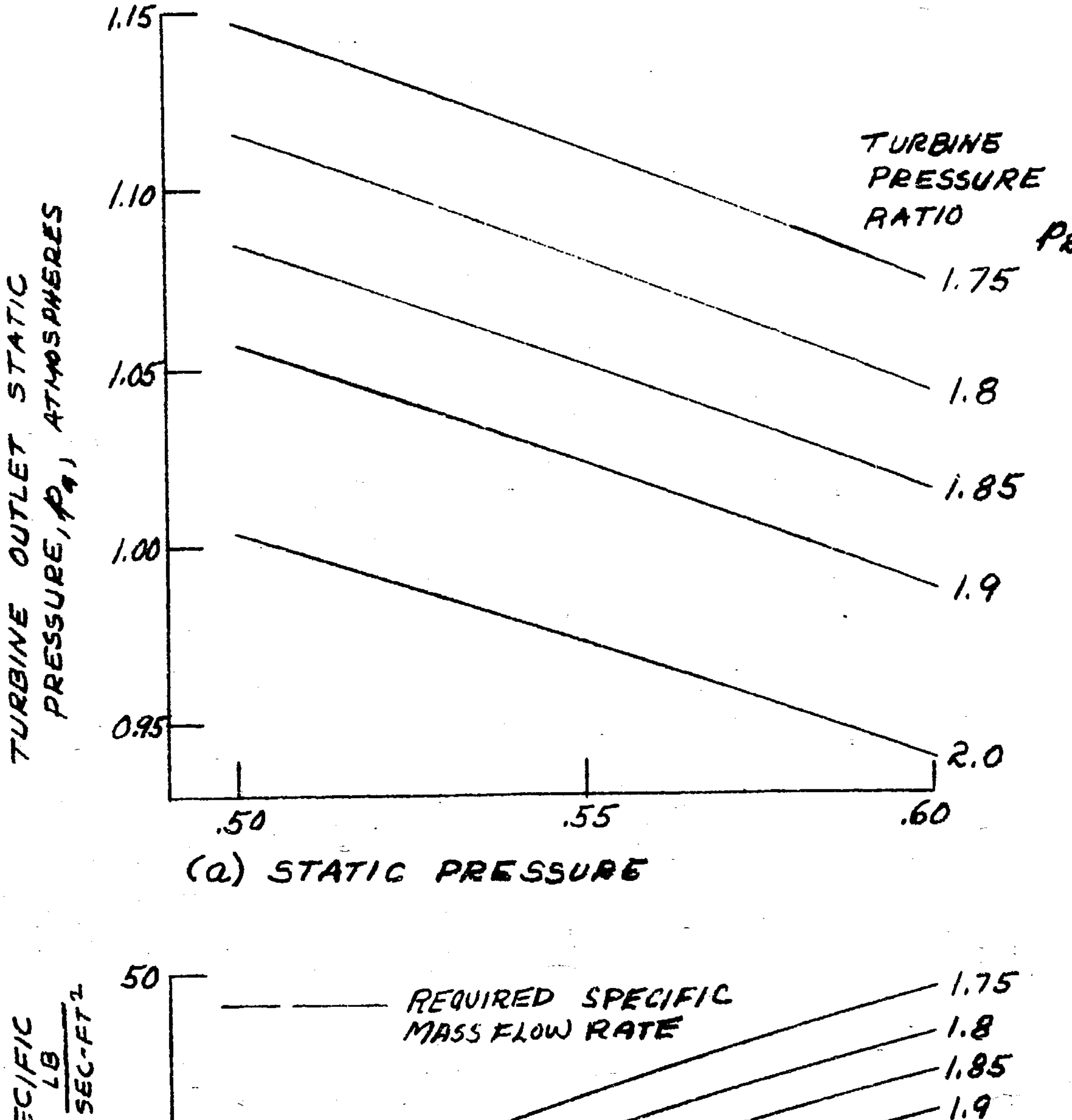
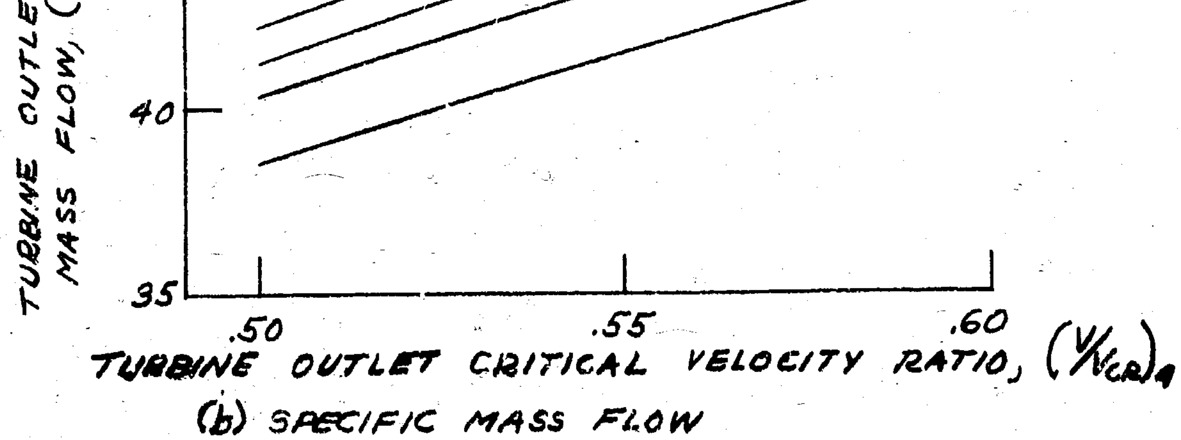



FIGURE 2 SKETCH OF TURBINE DRIVE SYSTEMS FOR 20-INCH TURBOFAN ENGINE SIMULATORS




(a) STATIC PRESSURE




(b) SPECIFIC MASS FLOW

FIGURE 3. - EFFECT OF DUCT TURBINE PRESSURE RATIO AND OUTLET VELOCITY ON OUTLET STATIC PRESSURE AND SPECIFIC FLOW RATE

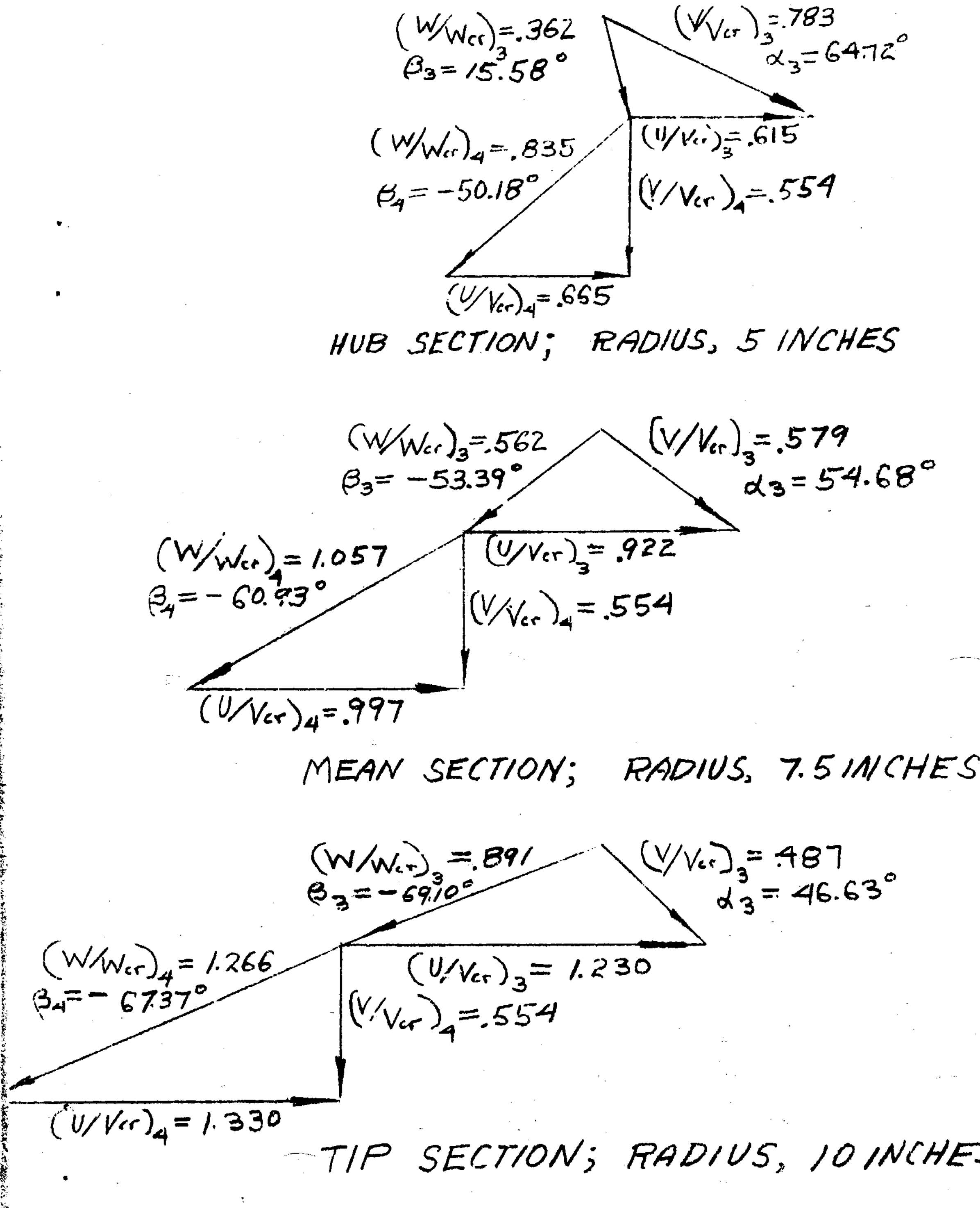
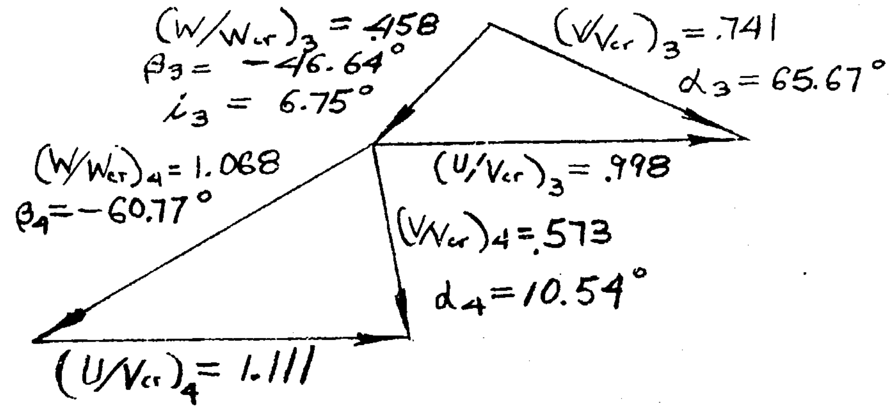
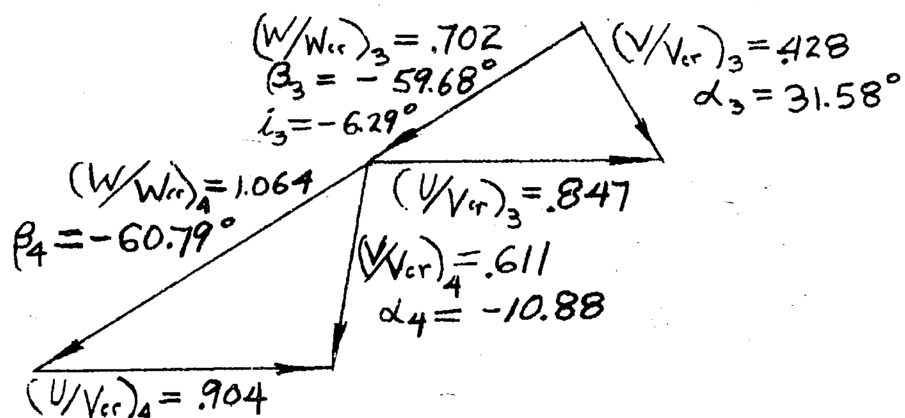



FIGURE 4 DUCT TURBINE VELOCITY DIAGRAM. BASED ON FAN TIP SPEED, 1450 FT. PER SECOND; FAN PRESSURE RATIO, 2.4; FAN POWER, 4350 HORSE POWER.




FAN PRESSURE RATIO 3
FAN TIP SPEED 1625 FT/SEC.




FAN PRESSURE RATIO 2
FAN TIP SPEED 1290 FT/SEC

FIGURE 5 MEAN RADIUS VELOCITY DIAGRAMS OBTAINED FOR THE DUCT TURBINE WITH ADJUSTABLE STATORS AT MIDRANGE SPEED FOR THE PRESSURE RATIO 3 FAN

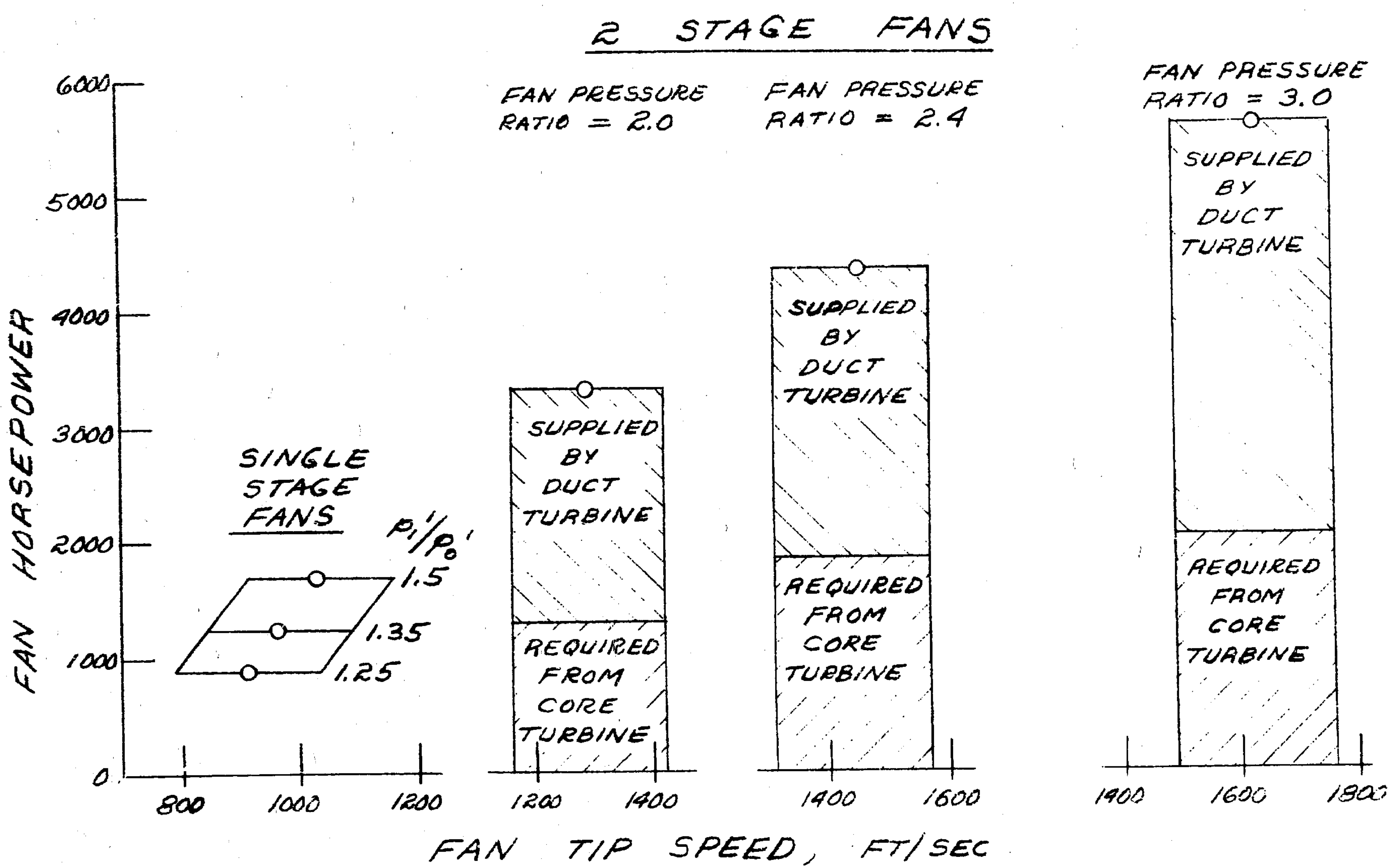



FIGURE 6. - POWER DEVELOPED BY DUCT TURBINE WITH ADJUSTABLE STATORS AND POWER REQUIRED FROM CORE TURBINE FOR THE RANGE OF FAN SPEEDS AND PRESSURE RATIOS

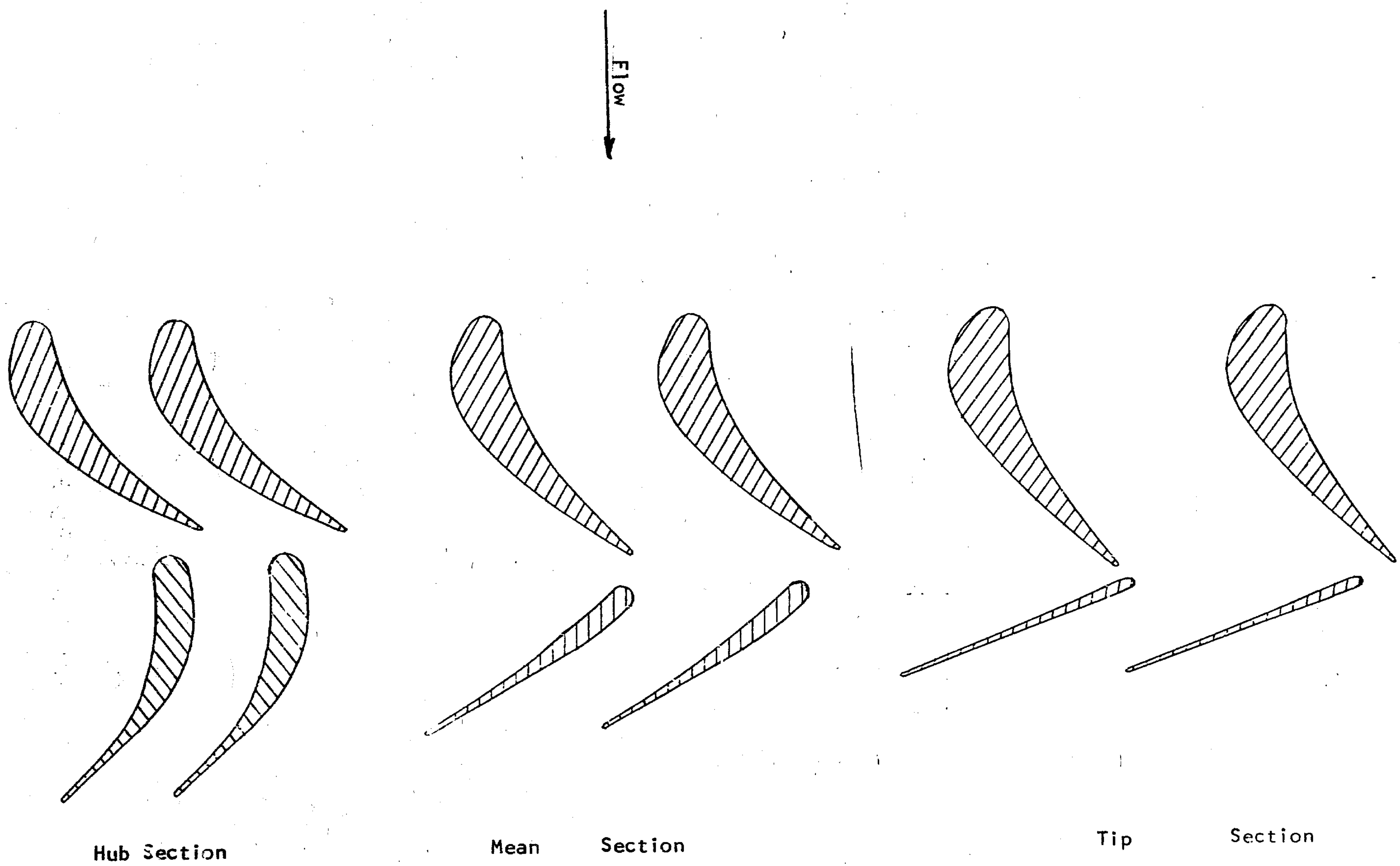



Figure 7 Sketch of Duct Turbine Stator and Rotor Blade Profiles and Flow Passages

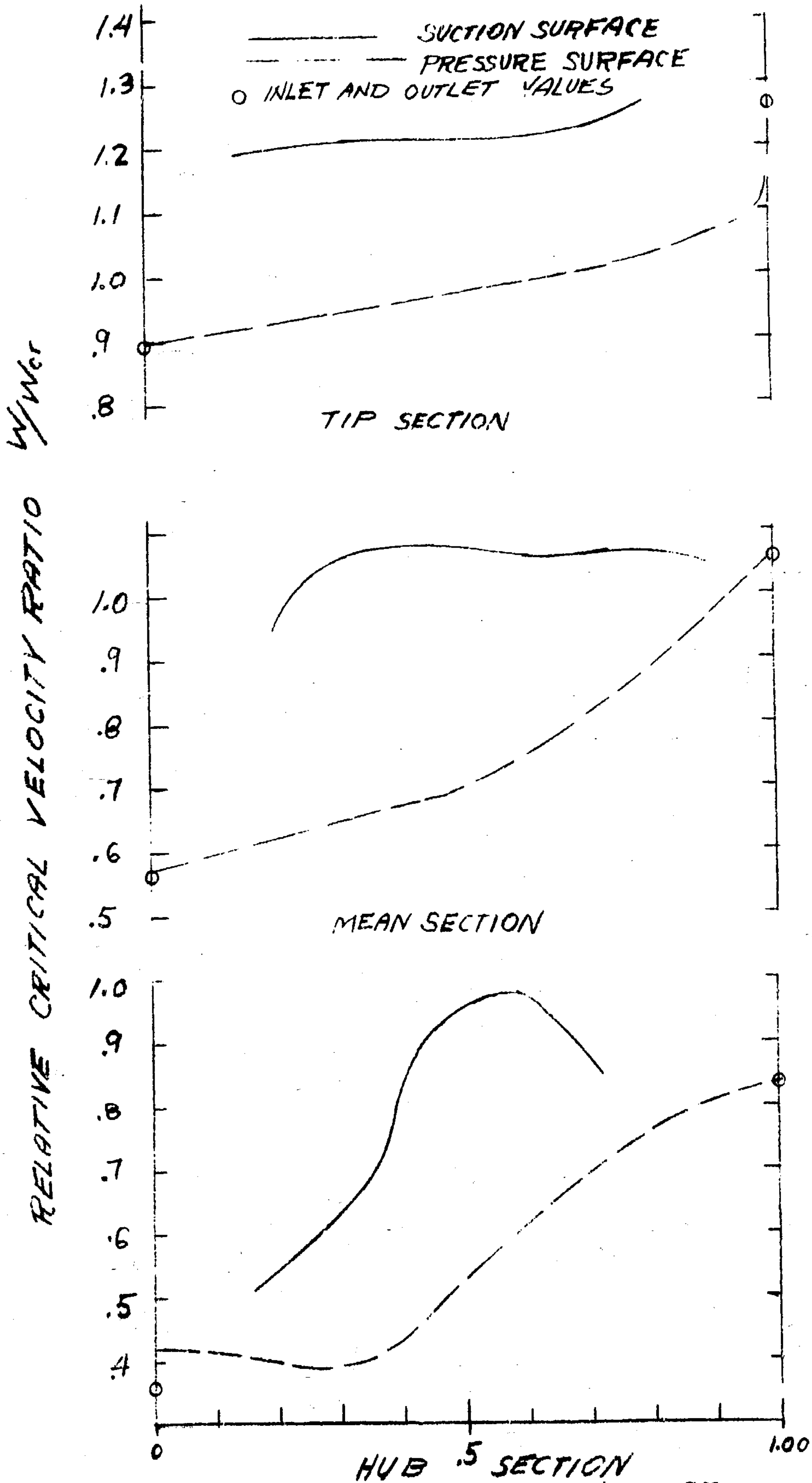



FIGURE 8 SURFACE VELOCITY DISTRIBUTIONS FOR DUCT TURBINE ROTOR BLADE