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FINAL REPORT

ANALYSIS OF GEOMETRY AND DESIGN POINT PERFORMANCE OF AXIAL FLOW TURBINES

PART III - DESIGN ANALYSIS OF SELECTED EXAMPLES

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SUMMARY

This report is the final one in a series of three. Based on a stream-filament analysis procedure and a correlation of total-pressure-loss coefficients developed in the Part I report (NREC Report No. 1125-1), a computer program for the analysis of the geometry and design-point performance of axial flow turbines was prepared. The computer program is presented in the Part II report (NREC Report No. 1125-2). This Part III report is concerned with the application of the computer program to the analysis of turbine design requirements.

The report presents the results of a general investigation of the effects of changes in the radial variation of design specifications such as the stator exit tangential velocity and the work output. These results are intended to provide guidance to future users of the program who are presented with considerably more freedom in turbine design using a stream-filament approach to the design problem. The report also presents the results of a specific investigation of the geometry and design-point performance of four turbines which satisfy a selected design requirement. The four turbines cover a range of tip diameters from a value consistent with a conservative design to a value which is 75 per cent of that selected for the conservative stage. The performance predictions show a 5.6 per cent drop in total-to-total efficiency and a 9.2 per cent drop in total-to-static efficiency for the increase in stage loading which accompanies the diameter reduction.

INTRODUCTION

This is the third part of a three-part report concerning the development of a computer program for the design-point analysis of axial flow turbines. The Part I report, Reference 1, presents the development of the analysis method and a loss coefficient correlation. The Part II report, Reference 2, describes the computer program. This final report of the series considers the application of the program to particular turbine design examples.

The computer program is based on a stream-filament approach to turbine design which includes consideration of the meridional components of streamline slope and curvature. Solutions of the flow field are obtained where the flow can be considered axisymmetric; that is, at the turbine inlet, all interblade row design stations, and the final stage exit. For an analysis of the design-point geometry and performance of a turbine, the design requirements are conventionally expressed by the inlet weight flow, the inlet total temperature and total pressure, the power output of the individual spools in a multispool unit, and the rotative speed of the spools. In addition to the standard design analysis variables of annulus geometry, number of stages, and power output split between stages, the program enables the turbine designer to consider as analysis variables the radial variations of the inlet conditions, streamline curvature and slope, element loss coefficient or efficiency, stator exit whirl velocity or absolute flow angle, and power output. The program also incorporates a total-pressure-loss coefficient correlation so that it is possible to make comparison of alternative designs using parameters which are fully consistent with the assumed correlation of total-pressure loss for the individual elements of the blading.

The contents of the report are in two major categories: one, a general investigation of the effects on predicted geometry and performance of the parameters which may be selected by the designer; and two, the analysis of a series of four turbines to satisfy the same design-point requirements but differing in outside diameter. The computer program permits the specification of radial variations of such design variables as the stator exit flow angle or whirl velocity and the power output of individual stream filaments. However, the requirements that the solution of the flow field must satisfy radial equilibrium and that the geometry of the blading should be mechanically acceptable place boundaries on the analysis variables. Thus, the principal objective of the work in the first category was to qualitatively investigate the effects of the analysis variables and hence to provide guidance to future users of the program. For the investigation, three sets of turbine design requirements are used; these correspond to a single-stage turbine having a design-point pressure ratio of approximately 2:1, a multistage turbine having a two-stage high pressure (hp) and a five-stage low pressure (lp) spool, and a small single-stage turbine of high pressure ratio. The first two requirements were supplied by NASA as selected applications for the program; the third design requirement is typical of a fuel-pump application with supersonic flow at stator and a near-impulse rotor and is, in fact, similar to the first stage of the turbine of Reference 3. The stage total-to-static pressure ratio of this stage is in excess of 4:1.

For the analysis of a particular design requirement, the single stage supplied by NASA is investigated at four values of outside diameter. Starting with a conservative design for which the outside diameter is 36 inches (91.4 cms), the outside diameter is decreased in three steps to a value of 27 inches (68.6 cms). The hub diameter of the stage is varied to maintain a constant annulus area, and hence, the series of turbines cover a wide range of rotor hub section loading when the stage work is related to the blade speed at the hub. The results of the analysis provide a means of assessing the trade-off between a reduction in turbine size and the reduction in efficiency which accompanies the resultant increase in stage loading.

Report Arrangement

The report is divided into three main sections. The first section presents the results of the general investigation of the effects of specified analysis variables. The principal variables considered are the radial distribution of whirl velocities or absolute flow angles at stator exits, the radial distribution of the power output, and interfilament mixing. The second section presents the results of the design analysis of the four versions of the NASA-selected single-stage application. This section details the predicted performance of the stages, their geometries, and their computed velocity diagrams. An appendix giving the complete computer printouts of the selected designs forms the third section of the report.

THE EFFECT OF CHANGES IN ANALYSIS VARIABLES

Introduction

Once the conventional turbine design restrictions of a free-vortex distribution of tangential velocities and a radially constant work extraction have been removed, a turbine designer is given considerably more design freedom. For example, the computer program permits arbitrary specification of the radial variations of stator exit tangential velocity (or absolute flow angle) and the work output.

However, it does not necessarily follow that aerodynamically and mechanically acceptable designs will be produced by the program unless the input specifications are selected with reasonable care. For example, a solution of the flow field with positive values of the through-flow velocity at all radial stations will only be possible for a relatively narrow range of tangential velocity distributions when the absolute flow angle is high as is the case at most stator exit design stations. Similarly, the amount of radial variation in the work output which will be possible in a stage is limited; large variations in the streamline total temperature drop will not be possible in some designs because it will not be possible to obtain a physically acceptable solution of the meridional velocity distribution. The first part of this section is devoted to these two aspects of turbine design. The principal intent is to provide some guidance to future users of the program by illustrating the effect of changes in the specifications of stator exit conditions and the power output function for sample design requirements.

Also considered in this section are the effects of three other analysis variables: the specification of the loss correlation used in the analysis, the departure from the stream-filament flow as simulated by interfilament mixing, and the curvature of the annulus walls.

Stator Exit Flow Parameters

The computer program provides the option to specify either stator exit tangential velocities or absolute flow angles as a function of radius. Use of the program to date has shown that the range of distribution of tangential velocities for which a solution of the flow field is possible is relatively small. This point is illustrated by data obtained from investigations of two versions of the single-stage turbine, the highly loaded stage with an outside diameter of 27 inches (68.6 cms) and a hub/tip ratio of 0.608, and the conservative design with an outside diameter of 36 inches (91.4 cms) and a hub/tip diameter ratio of 0.803. (The specified design requirements of these turbines are presented in the following chapter.)

For the highly loaded stage, three absolute stator exit angle distributions were specified, and these are shown in Figure 1 as a function of radius. The angle distribution which varies by approximately 10 degrees across the annulus closely approximates a "free-vortex" variation for which the tangent of the angle varies inversely with radius; the second distribution reduces the radial variation to approximately 5 degrees, and the third is radially constant. The computed tangential and meridional velocity distributions are also shown in this figure. The velocities have been normalized in each case by dividing by the appropriate value of velocity at the mean streamline. It can be seen that the meridional velocity distribution for the first specification is approximately constant with radius. The constant angle design, however, produces a significant variation of meridional velocity with the hub velocity approximately 50 per cent greater than that at the tip. The change in tangential velocity distributions, however, is not large. The ratios of the tangential velocity at the casing radius to that at the hub are 0.631 and 0.668 for the constant angle and free-vortex angle design, respectively. These numbers can be compared with a value of 0.608 which would be obtained from a conventional free-vortex design in which a radially constant axial velocity would have been assumed. Figure 1 also shows the shift of the mean streamline position towards the hub which accompanies the change from a free-vortex angle distribution to a constant flow angle design; the radii at which the normalized tangential velocities are unity correspond to the mean streamline.

Although all three angle distributions were selected to have the same angle at the arithmetic mean radius, the three designs will have differing effective flow areas at stator exit. Hence, the stage reaction changes and, accompanying the change, the streamline values of the blade row velocity ratios also change. In Figure 2 the radial variations of the stator velocity ratio, V_{∞}/V_1 , are shown for the three angle distributions. Although the mean level of velocity ratio has changed, the slopes of the curves are approximately equal. It would seem likely that if the angle distributions had been selected to yield constant values of effective flow areas for the three distributions that the radial variations of blade element velocity ratios would have been almost identical. Also shown on Figure 2 are the radial variations of stator blade row efficiency (defined as $1-e$ where e is the kinetic-energy-loss coefficient). It will be seen that, for the total-pressure-loss correlation used, the effect of the angle changes has been to increase the efficiency of the hub section and decrease it at the tip by changing from a free-vortex to a constant flow angle distribution.

The stage efficiencies predicted for the three stators in stage designs which used identical specifications for the distribution of power output function showed a decrease in total-to-total efficiency from 89.08 to 88.25 per cent for the change from a free-vortex to a constant flow angle stator. The total-to-static efficiency predictions, however, yielded an opposite effect with an increase from 68.65 to 69.68 per cent. Later in the report, where the predicted performances of various stages are discussed in greater detail, it is shown that these changes are the result of changes in the stage reaction rather than the result of a redistribution of the stator exit absolute flow angles. When angle distributions are selected to maintain a constant value of effective flow area, the change in stage efficiency is negligibly small.

A similar investigation was carried out using the large diameter turbine for the same design requirements. The angle and velocity distributions for the higher hub/tip ratio turbine are shown in Figure 3. The characteristics exhibited are similar to those of Figure 1. The changes again occur in the meridional velocity distribution rather than in the tangential velocity distribution.

From Figures 1 and 3 it can be concluded that the stator exit meridional velocity distributions are very sensitive to changes in tangential velocity distributions and that there exists only a narrow band of the latter over which it will be possible to obtain a satisfactory design solution. Therefore, it is recommended that the option to specify flow angles, rather than tangential velocities, at the stator exit design station should be used. Experience with the program has shown that tangential velocity distributions which depart significantly from a free-vortex distribution will not yield a solution. Large radial gradients of meridional velocity are required to sustain radial equilibrium with some tangential velocity distributions and unless the mean level of through-flow velocity is sufficiently high, the meridional velocity will approach a zero value at some point within the annulus.

The sensitivity of the solution increases as the mean flow angle increases. Large gradients of static pressure are required to maintain radial equilibrium at stator exits, but the static pressures are principally dependent on the tangential velocities. Hence, relatively large changes in meridional velocity may be required to achieve the required static pressure distribution when the tangential velocity distribution is arbitrarily specified.

Stage Power Output Function

The computer program permits the specification of a radial distribution of a power output function as an analysis variable. The power output function is, by definition, the fraction of the total stage power output produced by the flow passing between any selected streamline and the hub. Thus, a designer is able to redistribute the power output between the filaments of the flow. However, experience with the program has shown that acceptable design solutions can only be obtained for a relatively narrow band of power output functions. The power output function is specified versus streamline number so that the power output of individual equal-flow stream filaments can be controlled independently of the streamline locations. For the investigation of the effects of changing the power output distribution, parabolic distributions of the power output were used. These distributions produce a linear variation of streamline total temperature drop for the streamlines used in the analysis; since streamlines are located to define equal flows, the total temperature drop is not necessarily a linear function of radius. The departure of the power output function from a constant work output design is indicated by the index used to generate the parabolic power output function. An index of 1.0 corresponds to a radially constant total temperature, and the factors 0.85, 0.87, and 0.89 produce hub streamline temperature drops which are 91.9, 93.2, and 94.3 per cent of the mean value, respectively, with corresponding increases in the tip streamline total temperature drop.

In Figures 4 and 5 the results obtained for three distributions of power output are shown. The turbine is the small-diameter single stage and a constant angle stator is used for all three designs. The actual variations of streamline total temperature drop are shown in Figure 4 for the selected power output distribution, together with the normalized meridional velocity distribution at the stage exit and the variation of local values of rotor blade relative velocity ratio, V_1'/V_2'. It will be seen that as the local value of total temperature drop increases, the computed meridional velocity decreases. For this particular turbine design a solution could not be obtained with a uniform distribution of power output. The value of meridional velocity decreased rapidly to zero at a point near the hub section. The design specifications near the hub in that particular case implied a loading which exceeded a limiting loading value. By progressively decreasing the power output requirement of the inner part of the annulus flow, the meridional velocity at the hub is increased. However, since the tip section loading is increased, the meridional velocity at the tip decreases. It will be seen that the changes in power output distribution have produced relatively little change in the local velocity ratios for the rotor.

While it is convenient to generate power output functions if a parabolic variation is assumed, it does not necessarily follow that such distributions would be selected for final designs. From Figure 4 it can be inferred that the power output should be redistributed so that the reduced power output of the hub section should be compensated by an increase in the central section of the design. It would appear that increasing the tip section total temperature drop by an amount equal to the reduction at the hub could in many designs merely transfer the limiting loading problem from hub to the tip. The selection of a final design will, of course, be made after consideration of both the predicted performance of the stage and the geometry of the blading. In Figure 5 the computed inlet and exit rotor relative flow angles are shown for the three power output distributions. Reducing the power output of the hub section has reduced the hub section deflection by approximately 5 degrees and increased the tip deflection by 3.7 degrees. With the loss correlation used in the analysis, the changes in reaction and deflection produced by the change in power output function result in very little change in the mass averaged value of stage efficiency.

The turbine discussed above is a relatively low hub/tip ratio stage. However, limiting loading problems can be expected in any highly loaded stage. This point is illustrated by Sample Case IV given in Reference 2. This sample design, which includes the full computer program output for three power output distributions, is for a design requirement similar to the fuel-pump turbine of Reference 3. As was illustrated in the sample output, a solution could not be obtained with a uniform distribution of power output even though the stage hub/tip ratio was 0.88. The normalized meridional velocity distributions for the two sets of analysis variables which produced solutions are shown in Figure 6. This figure also shows the radial variation of total temperature drop implicit in the power output function distributions. Changing the power function to reduce the hub total temperature drop by approximately 1 per cent has produced a significant change in the predicted meridional velocity distribution; the higher value can be concluded to be extremely close to a limiting loading. The data of Figures 4 and 6 both show that only limited variations of work output with radius will produce mechanically acceptable blade geometries. Hence, it is important that future users of the program appreciate the fact that it is possible to exceed a limiting loading condition at any point in the annulus. At the limiting loading condition it becomes impossible to obtain a value of static pressure which will satisfy radial equilibrium. To assist in the selection of a suitable power output distribution, the computer program provides output at the lowest value of mass flow for which a solution could be obtained, thus providing a basis from which to modify the specification of the analysis variables including the power output function.

The Total-Pressure-Loss Coefficient Correlation

The program provides various optional specifications of the performance of the elements of the blading. The correlation developed in Reference 1 has been made an integral part of the program. However, the coefficients of the correlation have been made part of the input specification. Hence, these coefficients together with the additional loss factor can be regarded as analysis variables. It is believed that the coefficients recommended in Reference 1 produce realistic total-pressure-loss coefficients for the type of turbine on which the correlation was based. Nevertheless, it is almost inevitable that the correlation will be revised at a later date and that the additional loss factor will have to be used in some case to produce realistic analyses for particular turbines.

To illustrate this point, a large additional loss factor was used in the analysis of the fuel-pump turbine (Sample Case IV of Ref 2). The turbine is a high pressure ratio single stage in which the stator exit Mach number is supersonic and the rotor is of near-impulse design. With the stator exit flow angle specified, analyses of the stator exit flow field were carried out using the internal correlation of loss with and without an additional loss factor. (The program output for the latter case is given in Ref 2). The total-pressure-loss coefficients for the two analyses were virtually independent of radius and can be considered constant at 0.125 and 0.368. The computed absolute and relative Mach numbers at the stator exit plane are shown in Figure 7, together with the rotor inlet relative flow angles. For this particular design the change in the computed rotor inlet flow angle is not large, principally because of the type of velocity triangle. However, the most notable change in the design analysis is in the stage reaction; for the same power output the analysis with the low stator loss coefficient produces high stator exit Mach numbers and low rotor relative exit Mach numbers. Since the test data of Reference 3 show static pressures consistent with the higher loss coefficient, it must be concluded that an additional loss factor should be used for the analysis of interblade row conditions. However, this does not necessarily mean that the total-pressure-loss coefficient correlation used for the stator row is, in general, completely invalid for supersonic stator designs. It is quite possible that the additional loss occurs as a result of the shock system ahead of the rotor. When the program is used for turbine designs in which the predicted rotor relative Mach number is supersonic over the entire span of the rotor, it is recommended that the assumptions concerning the losses should be reviewed. The internal correlation of total-pressure-loss coefficient and the recommended coefficients of that correlation are unlikely to be valid when the mean level of rotor inlet relative Mach number is supersonic.

The dependence of the total-pressure-loss coefficient on the row reaction and the deflection incorporated in the loss correlation is such that the limiting loading condition is likely to occur first at a rotor hub section. As illustrated by Figure 6, a small change in the loading of hub section can produce significant changes in the meridional velocity gradient when the section is close to its limiting loading. The actual value of limiting loading will depend on the loss correlation being used in the analysis; the higher the total pressure loss in any given section of the blading, the more difficult it becomes to achieve a value of static pressure which will satisfy radial equilibrium.

Interfilament Mixing

The computer program is based on a stream-filament analysis of the flow through a turbine. Hence, if an analysis of a multistage unit is performed with a radially uniform distribution of power output for each stage, a total-pressure profile will develop as a result of lower efficiency of the hub sections compared with the mean or tip values. Undoubtedly, the flow within a turbine is considerably more complex than that assumed for the analysis procedure. In order to provide a means of simulating the redistribution of the flow within blade rows, the analysis procedure incorporates a simple interfilament mixing model (Ref 1). In the regions of high loss, low-momentum flows will occur and low-momentum fluid will migrate under the influence of the mainstream static pressure field and/or the centrifugal force field of the rotor. Thus, it is to be expected that total-pressure profiles predicted on a stream-filament basis will be less uniform than those obtained from test data. The simple mixing model used in the analysis can be used to reduce or eliminate total-pressure and total-temperature profiles. It should be pointed out that at the present time the selection of the mixing factors will have to be based on the judgment of the program user. However, the effects of mixing can be assessed analytically using the program.

One of the design examples supplied by NASA was a twin-spool turbine with two hp spool stages and five lp spool stages. Sample Case II of Reference 2 is based on the design requirements of the hp spool and the first three stages of the lp spool. This turbine was analyzed using the mixing parameter to produce results which are consistent with the assumption that the value of stage inlet total pressure used in the calculation of stator exit total pressure is radially constant. Hence, the total-pressure profile was not allowed to develop in the manner it would have done in a purely stream-filament analysis. An analysis with identically the same specifications as Sample Case II, with the exception of the mixing in the stator rows, failed to produce a solution at the exit of the second rotor of the lp spool. The results of the two analyses for the stator row which preceded the design station where the limiting loading condition was encountered are shown in Figures 8 and 9. Figure 8 shows the absolute flow angles at the stator inlet and stator exit and the following rotor relative inlet flow angle; Figure 9 shows the corresponding absolute and relative total pressures. With interfilament mixing assumed in the stator rows, the computed rotor relative inlet flow angle and relative total-pressure distributions are such that it is possible to obtain a solution for the next design station. With the purely stream-filament analysis, the meridional velocity near the hub of the stator exit plane falls significantly producing an increase of approximately 8 degrees in the absolute flow angle. This change is due to the predicted change in the stator exit absolute pressures. As a result, the rotor relative inlet flow angle is increased by approximately 12 degrees and the rotor relative total pressure is decreased by 1.2 psi (8270 N/m^2) at the hub. These changes are sufficient to make it impossible for the rotor to extract the implicitly specified amount of work at the hub section and simultaneously satisfy the requirement for radial equilibrium.

Both analyses used radially uniform power output specifications. Hence, without interfilament mixing within the stator rows it would have been necessary to reduce the power output along the hub stream filament by an amount which would have raised the inlet total pressure at the inlet to the second stage of the lp spool by approximately 1.2 psi (8270 N/m^2) in order to obtain a solution at the exit of that stage.

Annulus Geometry

The annulus geometry is an important design variable. Obviously, the mean diameter of a stage and the annulus areas have always been used as variables in the design of turbines and the program will, of course, provide a means of comparing alternative designs. However, the effects of the meridional slope and curvature of the flow are included in the flow field solution which has been programmed. Hence, slope and curvature of the annulus walls can be considered as analysis variables. The effects of these variables are qualitatively predictable from the radial equilibrium equation used in the analysis. The precise effect of a change in the slope or curvature of the annulus wall, however, cannot readily be obtained unless the computer program is used. At stator exit design stations, where the tangential velocity rather than meridional velocity dominates the radial equilibrium solution, it is quite possible that effects of changes in annulus slope and curvature will be small. At stage exit design stations, the effects could be considerably more pronounced; at stage exits the meridional velocity rather than tangential velocity will dominate the radial equilibrium equation. It is possible, therefore, that curvature of the annulus could be used to offset the deterioration of meridional velocity distribution that can occur in the presence of a total-pressure profile.

In the single-stage analyses presented later, a constant annulus area design is considered. Hence, since the analysis assumes that the meridional components of slope and curvature within the flow field are dependent on the annulus geometry, the effects of changes in annulus slopes and curvatures are not considered. However, in the multistage analysis of Sample Case II (Ref 2) the dimensions of the annulus were selected to produce a change of annulus flare at the inlet to the lp spool. The analysis of the multistage unit was repeated with the slope and curvatures of the annulus walls set equal to zero. The effect of the change on the meridional velocity distribution at the inlet to the lp spool is shown in Figure 10. For this design both hub and casing radii increase through the lp spool and the slopes and curvatures are positive across the entire annulus. As was to be expected, the meridional velocity is increased at the casing and reduced at the hub.

PERFORMANCE PREDICTIONS FOR A SINGLE STAGE AT FOUR TIP DIAMETERS

Introduction

The design requirements of the single stage, used as an example of the application of the computer program, were specified by NASA and are as follows:

Inlet Total Temperature	518.7 deg R (288.2 deg K)
Inlet Total Pressure	14.696 psia (10.133 N/cm^2)
Inlet Flow Angle	0 deg (--)
Mass Flow	45.51 lbm per sec (20.643 kg/sec)
Power Output	1287.5 hp (960.089 kw)
Rotational Speed	4660 rpm (--)
Specific Heat at Constant Pressure	0.24 Btu per lbm deg R (1004 J/kg deg K)
Specific Gas Constant	53.35 ft lbf per lbm deg R (287.0 J/kg deg K)

For the largest diameter turbine the casing and hub radii were specified to be constant at 18 and 14.465 inches (45.72 and 36.741 cms), respectively. In addition, the conservative design was specified to have a 50 per cent mean radius reaction corresponding to a tangential velocity of 758.7 ft per sec (231.5 m/sec) at the stator exit.

The geometry and performance of four stages differing in tip diameter, but having the same annulus areas were to be predicted. For the smallest diameter turbine in the series a value of tip radius of 13.5 inches (34.29 cms) was selected, this value being 75 per cent of the conservative design value. The dimension of the turbines, hereafter referred to as Turbines A, B, C, and D are tabulated below.

Turbine	A	B	C	D
Casing Radius (ins)	18.0	16.5	15.0	13.5
(cms)	45.72	41.91	38.10	34.39
Hub Radius (ins)	14.465	12.549	10.499	8.215
(cms)	36.741	31.874	26.667	20.866

The stage hub-to-tip radius ratios vary from 0.803 to 0.608 and the hub speed of Turbine D is approximately 57 per cent of that for Turbine A.

The validity of the comparison of the predicted performances is, of course, dependent on the loss correlation assumed. For the analysis, the performance of the individual elements of the blading is internally computed from a correlation of total-pressure-loss coefficients. No interfilament mixing was specified.

The Total-Pressure-Loss Assumptions

The program input includes the nine coefficients of the loss correlation developed in Reference 1. For this particular analysis the total-pressure-loss coefficient of an element of the blading is defined as follows:

$$Y = \frac{|\tan\beta_{in} - \tan\beta_{ex}|}{(0.6 + 0.8\cos\beta_{ex})}\left\{0.03 + 0.157255\left(\frac{V_{in}}{V_{ex}}\right)^{2.6}\right\} \quad \text{if} \quad \frac{V_{in}}{V_{ex}} \leq 0.6$$

$$Y = \frac{|\tan\beta_{in} - \tan\beta_{ex}|}{(0.6 + 0.8\cos\beta_{ex})}\left\{0.055 + 0.15\left(\frac{V_{in}}{V_{ex}} - 0.6\right)\right\} \quad \text{if} \quad \frac{V_{in}}{V_{ex}} > 0.6$$

where suffices in and ex denote inlet and exit conditions relative to a stator or a rotor section. In addition, the total-pressure-loss coefficient was limited to a value of 1.0. Throughout the analysis no additional loss factors were specified. Hence, no attempt is made to account for tip clearance losses.

Design Optimizations

At the relatively high hub/tip ratio of Turbine A, no great variation of stage efficiency with the radial variation of the analysis variables is to be expected. Since the annulus dimensions were preselected together with a mean-line stage reaction, the only remaining analysis variables are the radial distribution of stator exit angle (or tangential velocity) and the power output function. As previously indicated, the variations of tangential velocity and power output are somewhat limited. The two sets of analysis variables of Sample Case I of Reference 2 showed a less than 0.1 per cent difference in efficiency between a design with a free-vortex distribution of stator exit flow angles and a uniform distribution of power output and a design with radially constant stator exit flow angle and a distributed power output. Therefore, the major investigation of the effect on over-all stage performance of alternative sets of analysis variables was carried out using the annulus geometry of Turbine D.

Variations in the power output function were systematically generated using parabolic distributions of the power output function with respect to the nondimensional mass flow function. The use of the parabolic function has the property that a linear variation of streamline total temperature drop is obtained. In addition, values of the power output function are readily computed for selected ratios of hub and casing total temperature drops. The expression used for the nondimensional power function ϕ is as follows:

$$\phi = \frac{2f}{1+f}\,\omega(\tau) + \frac{1-f}{1+f}\left\{\omega(\tau)\right\}^2$$

where $\omega(r) = \frac{j-1}{n-1}$ and $j = 1$ at the hub and $j = n$ at the casing. The power coefficient, f, is equal to the ratio of total temperature drop at the hub to that at the casing; the ratio of hub-to-mean streamline total temperature drop is $\frac{2f}{1+f}$. The coefficient f is used later to identify the power output specification.

The other variables considered in the optimization of Turbine D were the mean radius value of the stator exit flow angle and the radial variation of this angle. These two analysis variables also control the mean streamline value of stage reaction.

Variation of Efficiency With Power Output Distribution

Three alternative distributions of power output were investigated for a stage with a radially constant stator exit absolute flow angle of 67 degrees. These same distributions had coefficients of 0.85, 0.87, and 0.89 in a parabolic power output function. This series of analyses has been discussed earlier in connection with the possible range of the power output function and the principal features of the stage aerodynamics have been illustrated in Figures 4 and 5. The design with the lowest value of total temperature drop at the hub (f = 0.85) had the highest total-to-total efficiency, but the highest total-to-static efficiency was obtained with the intermediate value of f. The actual efficiencies predicted are presented in the following table.

Design Number	1	2	3
Power Coefficient	0.85	0.87	0.89
Total-to-Total Efficiency (per cent)	88.34	88.25	88.16
Total-to-Static Efficiency (per cent)	69.60	69.68	69.63

The actual variations of efficiency are relatively small. While some further improvements might be expected from the specification of power output distributions other than the parabolic type used, it is unlikely that the highest predicted efficiency will be significantly greater than any of those above.

Variation of Efficiency With Stator Angle Distribution

With a constant power output distribution (using $\mathcal{f}$ = 0.87), alternative stage designs were analyzed for three radial variations of the absolute flow angle at stator exit. Results from this investigation have been presented earlier in the discussion of the analysis variables as Figures 1 and 2. It was pointed out that the major effect on the velocity diagrams of the changes in the angle distribution was to change the meridional velocity distribution; the tangential velocities were not significantly altered by the angle change. The predicted efficiencies of the three stages are tabulated below.

Design Number	4	5	2
Hub-to-Tip Stator Angle Variation (deg)	72.1 - 62.1	69.6 - 64.6	67
Total-to-Total Efficiency (per cent)	89.08	88.74	88.25
Total-to-Static Efficiency (per cent)	68.65	69.14	69.68
Mean Streamline Stage Reaction	0.669	0.615	0.536

From these results it would at first appear that there is a significant improvement in total-to-static efficiency by specifying a constant angle stator design rather than a "free-vortex". Conversely, the free-vortex design yields a higher total-to-total efficiency. However, the change in efficiency levels is due to a change in the mean stage reaction rather than due to the variation in stator angle distribution as such. The effective flow area at stator exit is changed by the variation in stator exit flow angles; the constant angle design has a smaller effective flow area than the free-vortex design and, hence, its mean stage reaction is lower. To provide a valid comparison of the effect of angle distribution, the radially constant angle has to be decreased to increase the effective stator exit flow area and hence stage reaction. By decreasing the level of constant angle to 66.8 degrees, a stage with a mean streamline stage reaction of 0.621 is produced. The predicted efficiencies of this new stage are 88.82 and 69.76 for total-to-total and total-to-static, respectively.

The radial variations of the row velocity ratio and the local total-to-total isentropic efficiency of Design 5 above and the constant section design of nearly comparable mean streamline stage reaction (Design 7) are compared in Figure 11. It will be seen that the radial variations of local blade velocity ratios are similar. Although the constant section stator design now has a higher total-to-total efficiency, it is extremely likely that the difference in efficiency level would have been even less significant if the constant angle had been selected to yield exactly the same level of mean streamline stage reaction.

Although the investigation has covered a relatively small range, it must be concluded that the stream-filament analysis will not predict significant variations in stage efficiencies for changes in stator exit angle distribution when these distributions are selected to produce a constant value of mean streamline stage reaction.

The Variation of Efficiency With Stage Reaction

To investigate the effect of changes in the stage reaction, a series of four stage designs were investigated. These analyses used constant stator exit angle specifications and a constant power output distribution (f = 0.87). The principal results of this investigation are tabulated below.

Design Number	8	9	2	7
Stator Exit Flow Angle (deg)	67.16	67.08	67.00	66.8
Total-to-Total Efficiency (per cent)	87.48	87.945	88.25	88.82
Total-to-Static Efficiency (per cent)	70.77	70.13	69.68	68.76
Mean Streamline Stage Reaction	0.411	0.488	0.536	0.621

Stage reaction is conventionally defined as the ratio of static temperature drop across the rotor to the stage total temperature drop; the mean streamline value is obtained from the temperatures corresponding to the mass flow mean streamline. The results given in the above table are also shown in Figure 12. The opposing tendencies are not entirely unexpected and which value of stage reaction would be considered optimum will depend on the particular application of the turbine stage.

The predicted effect of a change in reaction is, of course, dependent on the loss correlation used in the analysis. The major factor in the change in the stage performance is the change in the relative velocity ratio for the rotor blade elements; Figure 13 compares rotor velocity ratios of the two extremes of stage reaction (Designs 7 and 8). This figure also compares the computed total-pressure-loss coefficients and the rotor relative exit Mach numbers of the two designs. The lower reaction design has a 33 per cent high rotor hub total-pressure loss coefficient, but the effect on mass flow weighted stage total-to-total efficiency is offset to some extent by the lower rotor relative exit Mach number; the change in total-to-total efficiency is only 1.34 percentage points.

Variation of Efficiency With Stage Loading

From the design optimization investigation it can be concluded that changes in the power output distribution and in the stator exit flow angle distribution do not significantly affect the performance of the resulting design. However, the ranges of these design parameters for which mechanically acceptable blading geometries are computed are relatively small. The effect of a change in mean stage reaction on the predicted stage efficiency of Turbine D is more significant. Nevertheless, the range of efficiencies predicted for a range of mean stage reaction of the small diameter turbine is less significant than the reduction in efficiency level resulting from an increase in mean stage loading which accompanies the reduction of the tip diameter from the value used in Turbine A. Hence, the over-all effect on performance of a reduction in tip diameter can be reasonably illustrated by any systematic variation of the other analysis variables. For the four designs finally selected, the complete computer outputs are given in Appendix I.

For the four turbines, a constant stator flow angle is specified for each. The actual angles specified are 66.7, 66.9, 67.1, and 67.16 degrees for Turbines A, B, C, and D, respectively. These specifications vary the mean streamline reaction from 0.5 for Turbine A to 0.411 for Turbine D. The particular variation finally selected was judged to be a suitable compromise between the levels of total-to-total and total-to-static efficiencies for the small diameter turbine.

The specifications for the power output distribution all use the parabolic variation of the nondimensional power function with the nondimensional mass flow function. The actual distributions are identified by the values of f which are 0.95, 0.94, 0.93, and 0.87 for Turbines A, B, C, and D, respectively. The mean streamline total temperature drop for each turbine is constant and the hub streamline temperature drops are 97.5, 97, 96.5, and 93 per cent of the mean streamline value for Turbines A, B, C, and D, respectively. The reduction of hub temperature drop with decreasing hub speed and hub section reaction was found to be necessary in order to avoid a limiting loading condition.

In addition to the full computer output of Appendix I, which can be used for detailed comparisons of the designs, the effects of the tip radius change are summarized in Figures 14, 15, 16, 17, and 18. In Figure 14 the total-to-total and total-to-static efficiency variations with tip radius are shown. As stated earlier, by varying the mean streamline stage reactions of Turbines B, C, and D, the rate of deterioration of total-to-total efficiency could have been lessened at the expense of the deterioration in total-to-static efficiency. For the selected designs, the reductions in efficiencies corresponding to a 25 per cent reduction in tip radius are 5.6 and 9.2 percentage point for the total-to-total and total-to-static, respectively.

In Figure 15, the turbines are compared in terms of schematic side view. Also noted on this figure are the specifications for the analysis variables and a performance summary of each design. The mean level of loading for the stages are indicated by the mean blade-to-jet speed ratio.

This ratio, ν , is defined as follows:

$$\nu = \frac{\bar{u}}{\sqrt{2 g_0 J C_p \Delta T_{os}}}$$

where $\bar{u}$ is the mean blade speed and ΔT_{os} is the isentropic total temperature drop. The value of ν for the four designs are 0.592, 0.526, 0.456, and 0.378.

Velocity diagrams for hub, mean, and casing radii are given in separate figures. Figure 16 compares the four sets of hub section diagrams; Figure 17, the mean sections; and Figure 18 presents the casing section diagrams. In the case of the mean section diagrams, it should be noted that the radial location of the mean streamline rather than the arithmetic mean radius is used. Hence, the stator exit and stage exit diagram for the mean section have differing section radii and blade speeds.

Concluding Remarks

The predicted variation of total-to-total efficiency with increasing stage loading can be considered satisfactory. The predicted values are, of course, directly dependent on the loss correlation assumed in the analysis. The correlation used was based on the correlation of achievable stage efficiency presented as Figure 4 of Reference 1. This data used for the loss coefficient correlation were derived from a mean-line analysis and the correlation has been applied in the computer program to individual elements, without any distinction between stator and rotor or hub and casing sections of the blading. A review of the predicted variations of streamline efficiencies suggests that the assessment of element performance has produced realistic values. It is interesting to compare the predicted efficiencies with the "achievable" efficiency obtained from

Figure 4 of Reference 1. The values of mean stage loading factor and stage flow factor, used in the simple correlation of achievable turbine efficiency together with a comparison of the predicted efficiency and "achievable" efficiency are given in the following table.

	Loading Factor $\psi = \frac{g_0 J c_p \Delta T_0}{U^2}$	Flow Factor $\phi = \frac{V_x}{U}$	"Achievable" Efficiency	Predicted Efficiency
Turbine A	1.15	0.57	0.942	0.931
Turbine B	1.43	0.66	0.937	0.920
Turbine C	1.87	0.77	0.921	0.902
Turbine D	2.57	0.97	0.881	0.875

The "achievable" efficiency does not make any allowance for the hub/tip ratio of the stage. Hence, until such time as experimental data are available from stages designed using the design analysis program, the loss correlation recommended and used in the current analysis can be considered satisfactory.

During the investigation of the effect on predicted performance of changes in the analysis variables, no significant change in efficiency level was predicted if the mean stage reaction was maintained constant. It is true that the detailed investigation was concentrated mainly on one stage and that only relatively simple radial variations of the stator exit conditions and stage power output were considered. Nevertheless, it is quite probable that a purely stream-filament analysis, limited to the solution of the flow field at interblade row stations and with total pressure losses assessed for individual streamlines, will not predict the actual performance difference of designs having the same annulus geometry. The actual performance of a blade row is undoubtedly affected by the overall design of the row. While relatively small differences in performances are likely to be predicted for alternative blade twists using the stream-filament approach, the actual difference in performance of the turbine stages could be significant. However, a design analysis program based on the stream-filament approach could be an important tool for use in a fully integrated turbine research program.

CONCLUSIONS

1. The prediction of the design-point blade row geometry and stage performance of four single stages having the same over-all design requirements has shown a 5.6 per cent drop in total-to-total efficiency and a 9.2 per cent drop in the total-to-static efficiency for a 25 per cent reduction in tip diameter from a conservative design value. These changes were for a particular variation of stage reaction; higher values of total-to-total efficiency for the most highly loaded stage could have been achieved at the expense of the total-to-static efficiency and vice versa.
2. In an investigation of the effects of changes in the analysis variables, it was established that the radial variation of tangential velocities at stator exits are limited by the radial equilibrium requirement to distributions which are close to "free-vortex" distributions. Small changes in whirl distribution specifications produce relatively large changes in the meridional velocity and absolute flow angle distributions.
3. Using a realistic correlation of the total-pressure-loss coefficients for individual blade elements, it has been shown that the specification of a uniform distribution of power output frequently produces a limiting loading condition at rotor hub sections. It is then necessary to reduce the work output of these sections in order to obtain a solution of the stage exit flow field which will simultaneously satisfy the loss specification and the radial equilibrium requirement. The redistribution of the power output, however, requires careful consideration in the case of highly loaded stages to avoid the transfer of the limiting

loading condition to other sections of the blading.

4. While the correlation of total-pressure-loss coefficient used in the present analysis appears to give realistic solutions, it will be necessary to review the correlation using experimental data from stages designed using the stream-filament approach. The extent to which the performance of a stage is affected by interfilament mixing and the over-all design of the blading (as opposed to the design of individual blade elements) should also be investigated analytically and experimentally.

REFERENCES

1. Analysis of Geometry and Design Point Performance of Axial Flow Turbines, Part I - Development of the Analysis Method and the Loss Coefficient Correlation (NREC Report No. 1125-1), Northern Research and Engineering Corporation, Cambridge, Massachusetts, September 14, 1967.

2. Analysis of Geometry and Design Point Performance of Axial Flow Turbines, Part II - Computer Program (NREC Report No. 1125-2), Northern Research and Engineering Corporation, Cambridge, Massachusetts, January 31, 1968.

3. Stabe, Roy G., et al, Cold-Air Performance Evaluation of a Scale-Model Fuel Pump Turbine for the M-1 Hydrogen-Oxygen Rocket Engine (NASA TN D-3819), National Aeronautics and Space Administration, Washington, D. C., February, 1967.

FIGURES

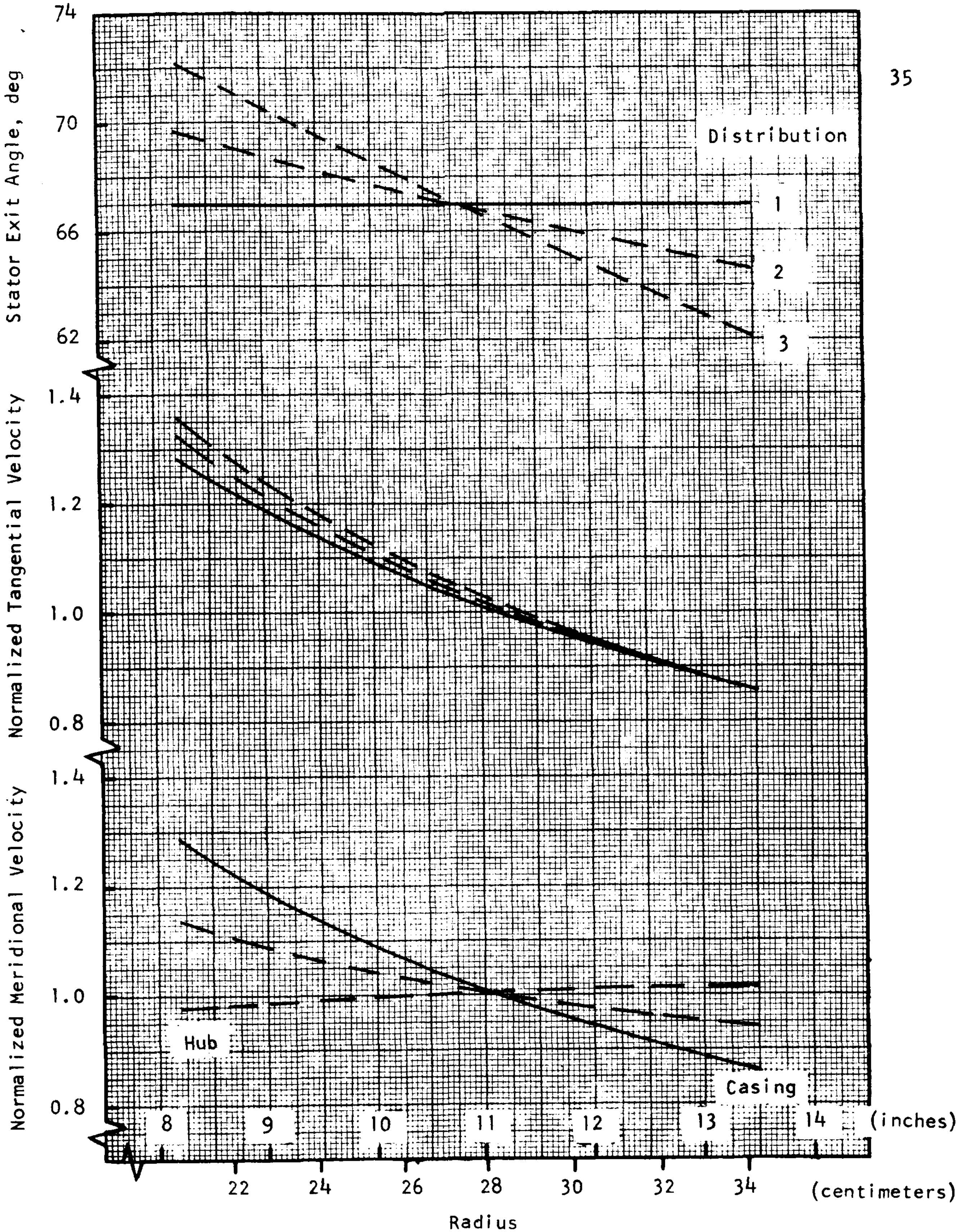


FIGURE 1 - TANGENTIAL AND MERIDIONAL VELOCITY DISTRIBUTIONS AT STATOR EXIT FOR THREE STATOR EXIT ANGLE DISTRIBUTIONS (HIGHLY LOADED STAGE - TURBINE D)

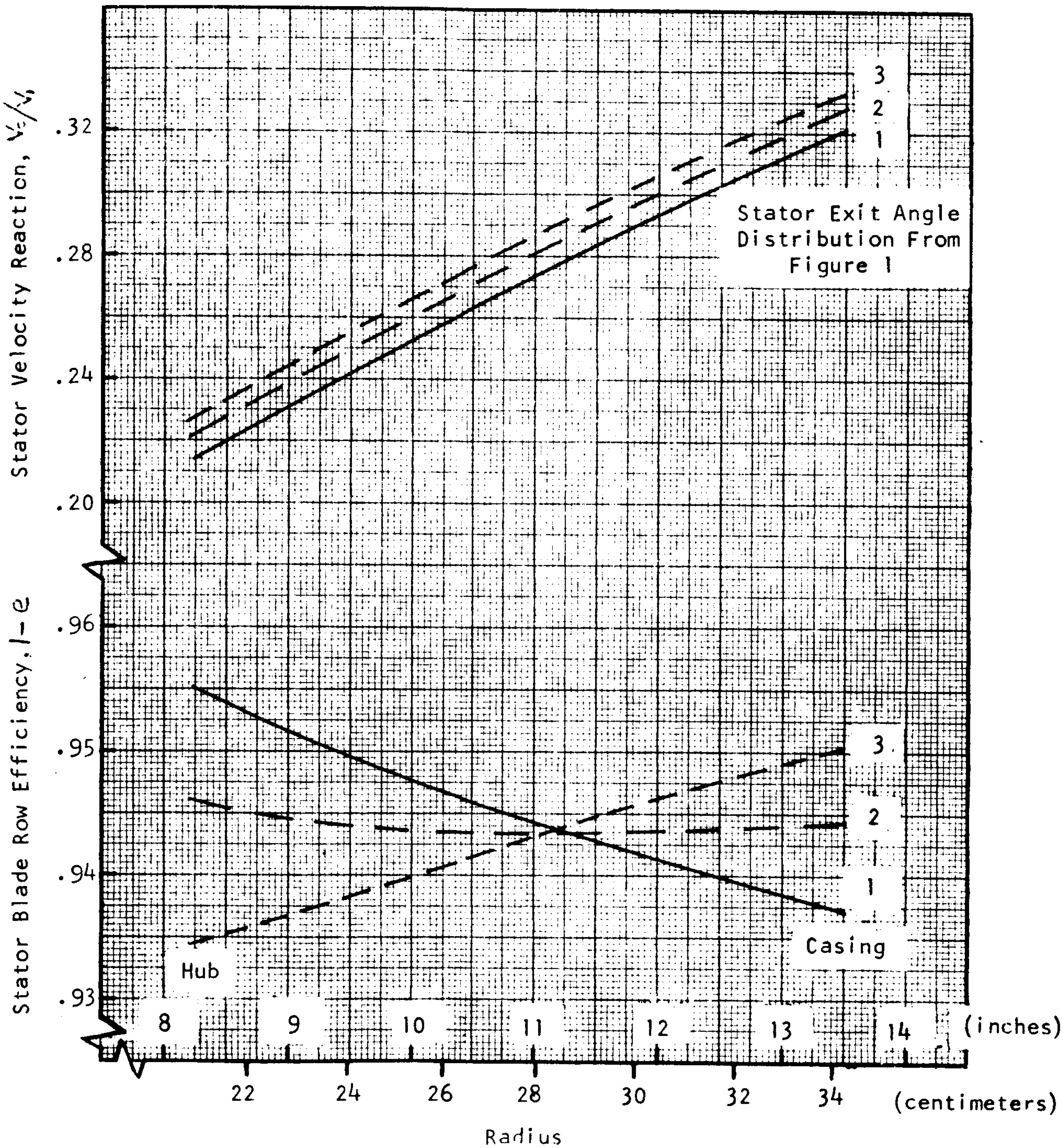


FIGURE 2 - STATOR VELOCITY RATIO AND BLADE ROW EFFICIENCY VARIATION FOR THREE STATOR EXIT ANGLE DISTRIBUTIONS (TURBINE D)

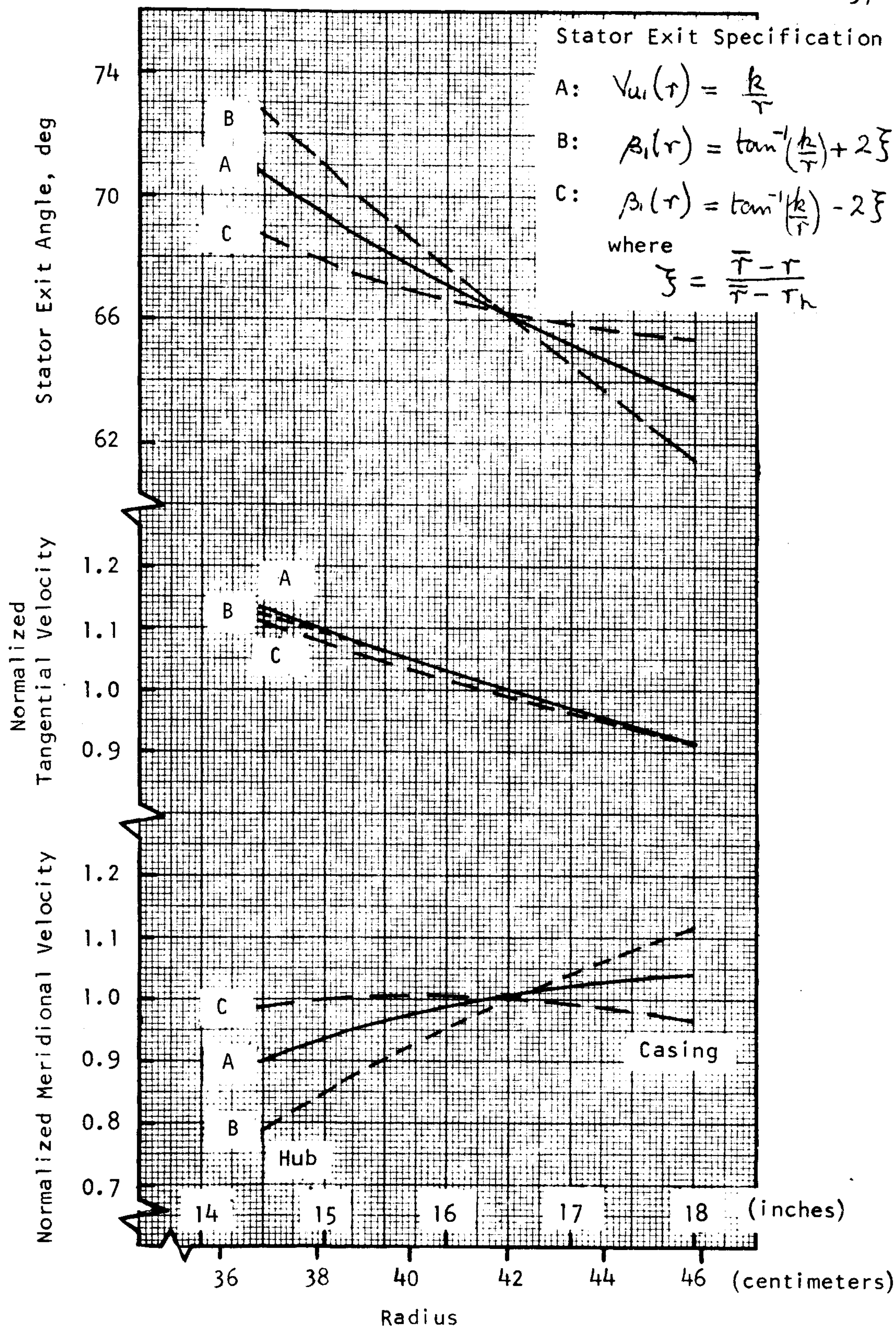


FIGURE 3 - TANGENTIAL AND MERIDIONAL VELOCITY DISTRIBUTIONS AT STATOR EXIT FOR THREE SPECIFICATIONS (CONSERVATIVE STAGE - TURBINE A)

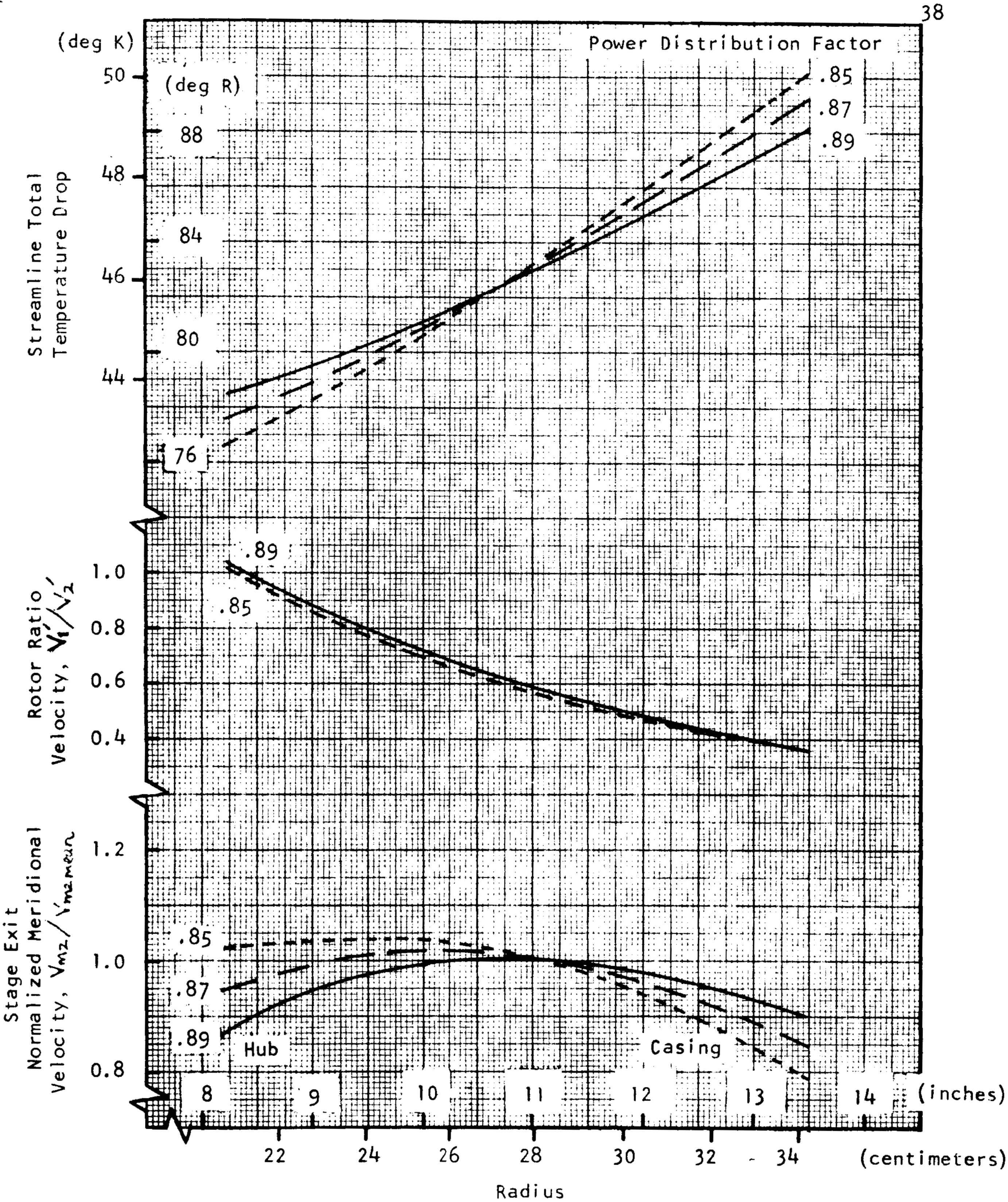


FIGURE 4 - THE EFFECT OF THE POWER OUTPUT DISTRIBUTION ON THE MERIDIONAL VELOCITY AT STAGE EXIT, STREAMLINE TOTAL TEMPERATURE DROP, AND ROTOR REACTION (TURBINE D)

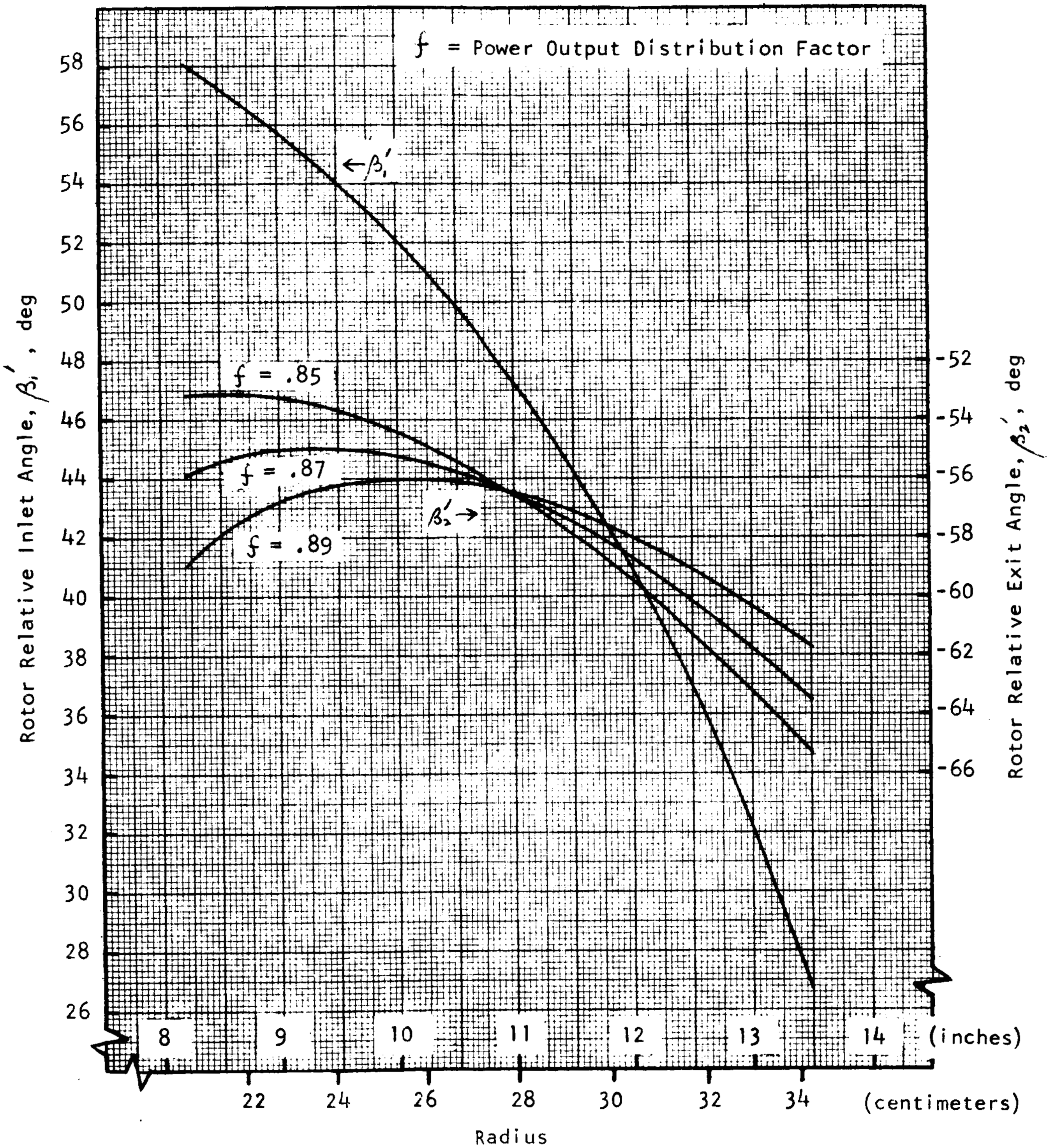


FIGURE 5 - ROTOR RELATIVE FLOW ANGLES FOR THREE POWER OUTPUT DISTRIBUTIONS (TURBINE D)

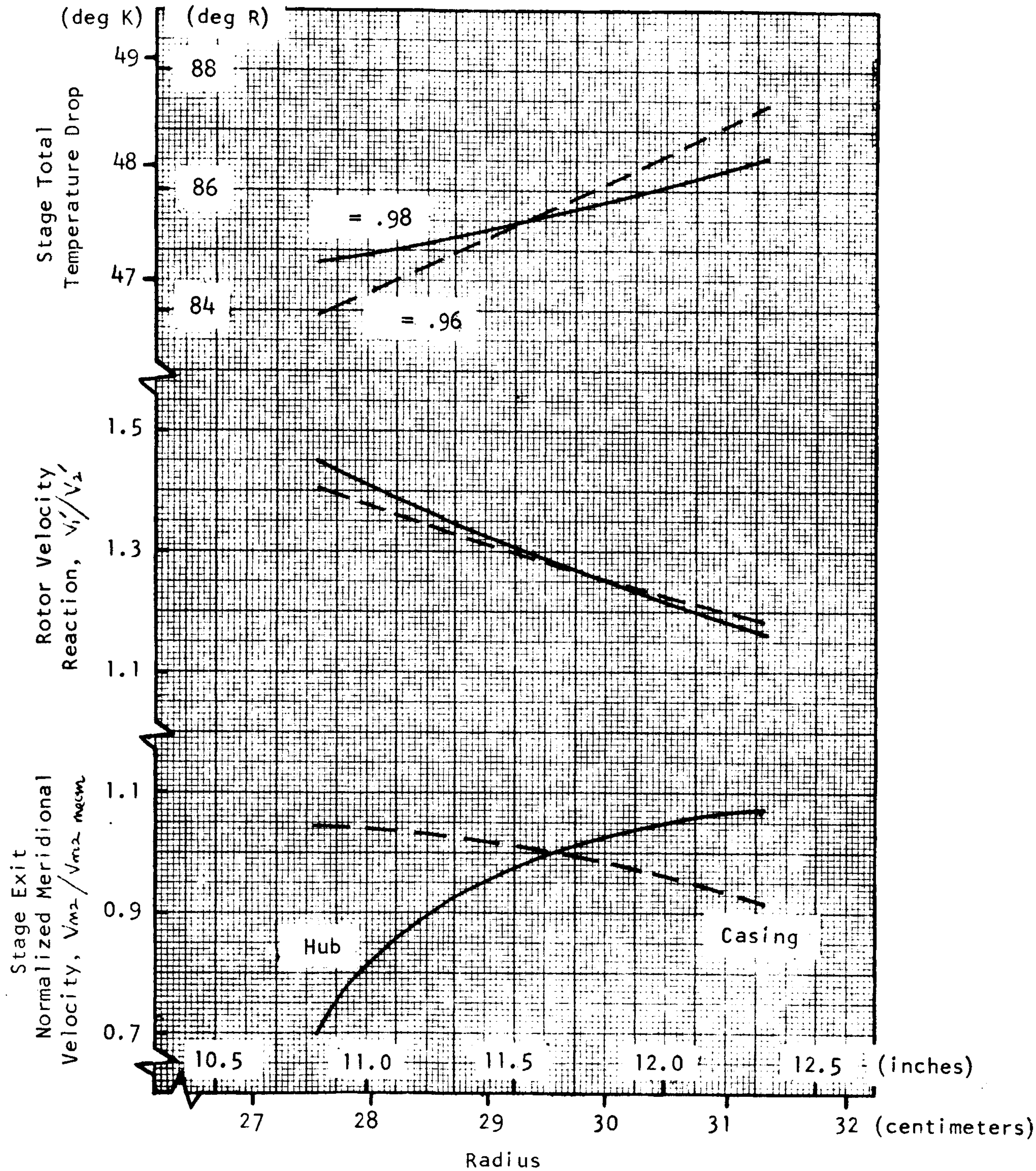


FIGURE 6 - THE EFFECT OF POWER OUTPUT DISTRIBUTION ON THE STAGE EXIT MERIDIONAL VELOCITY AND ROTOR REACTION DISTRIBUTIONS (FUEL-PUMP TURBINE)

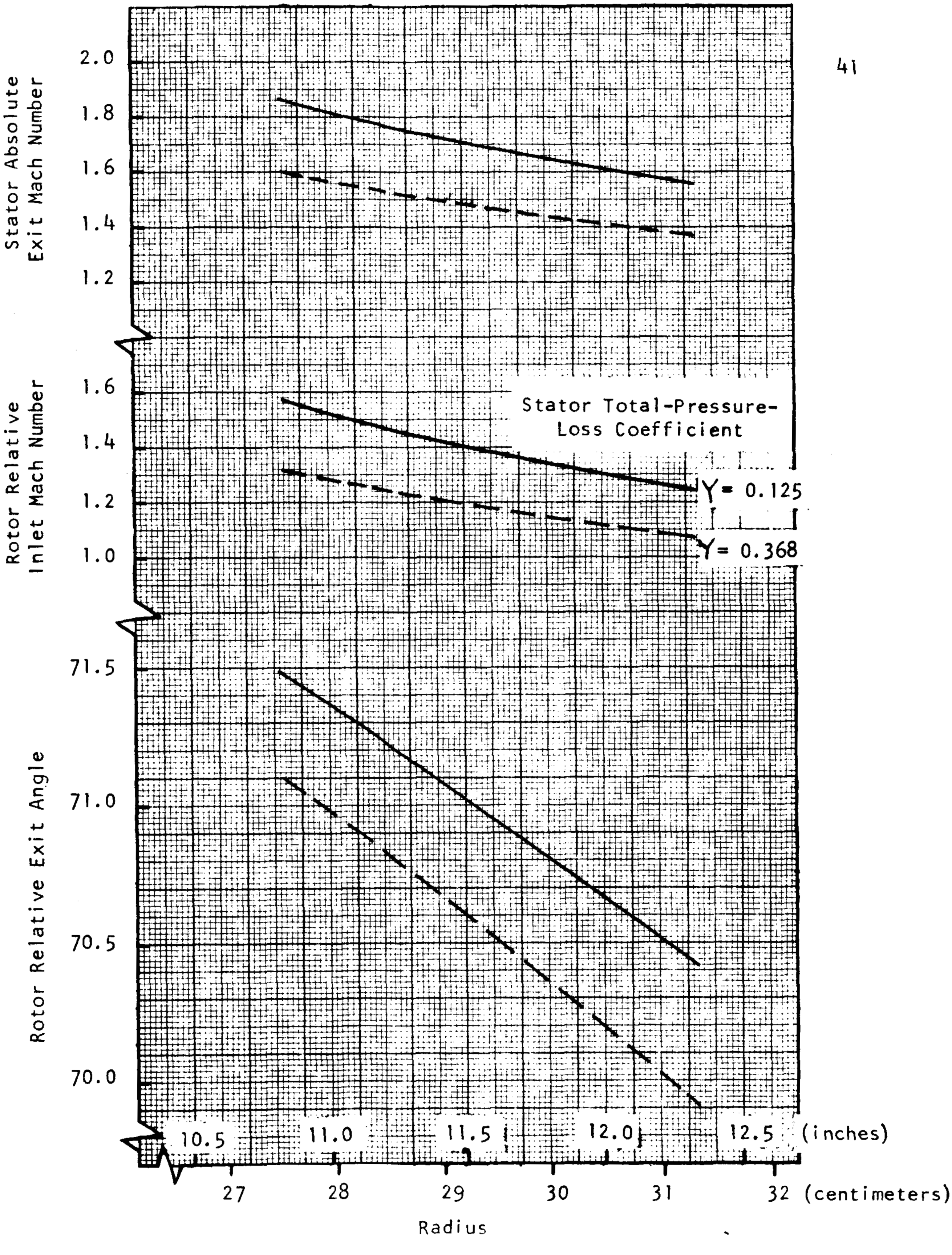


FIGURE 7 - THE EFFECT OF THE ASSUMED STATOR LOSS ON THE FLOW PARAMETERS AT EXIT FROM A SUPERSONIC STATOR (FUEL-PUMP TURBINE)

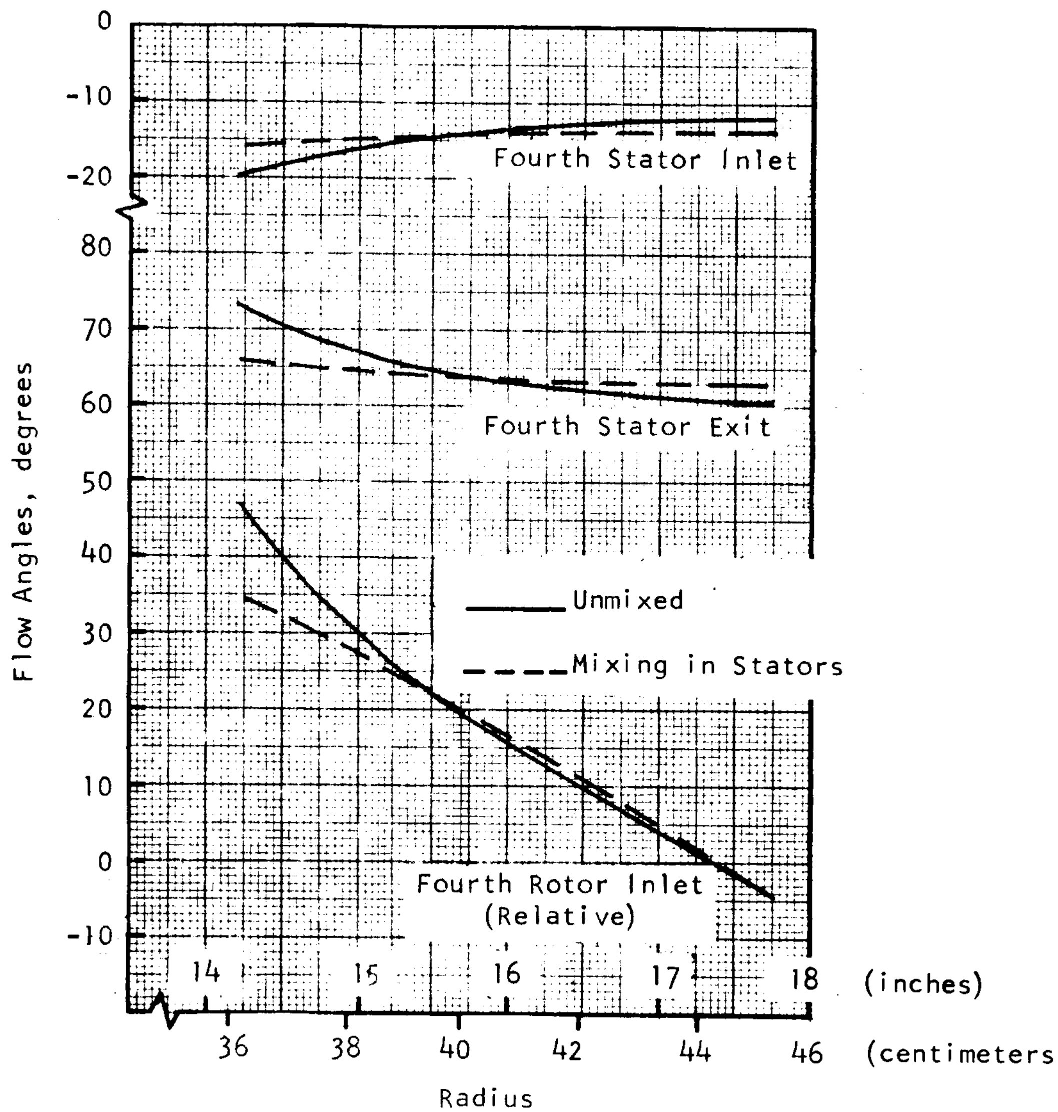


FIGURE 8 - A COMPARISON OF THE FLOW ANGLES AT THE THIRD STATOR OF A MULTISTAGE MACHINE WITH AND WITHOUT INTERFILAMENT MIXING

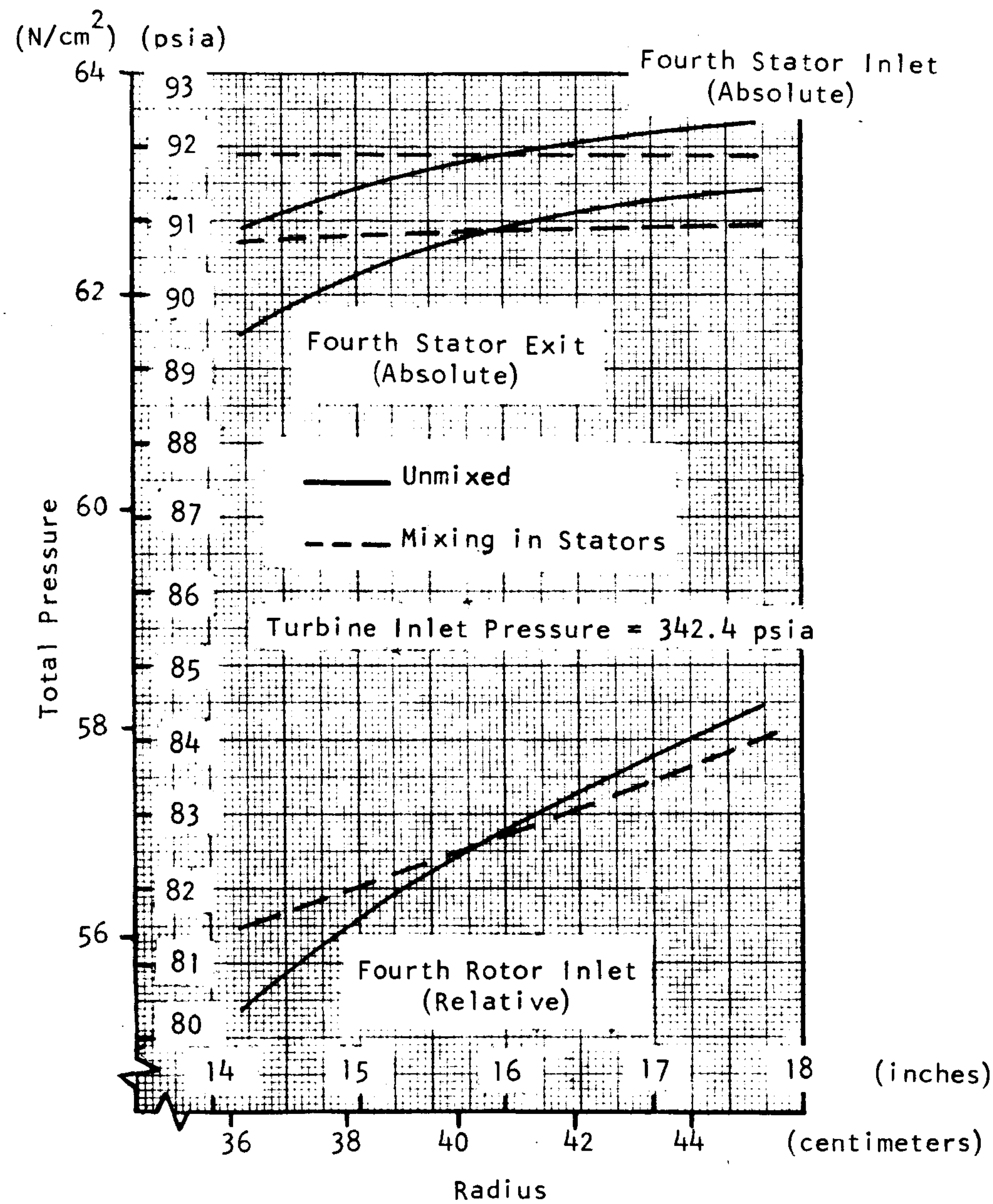


FIGURE 9 - A COMPARISON OF THE PRESSURES AT THE THIRD STATOR OF A MULTISTAGE MACHINE WITH AND WITHOUT INTERFILAMENT MIXING

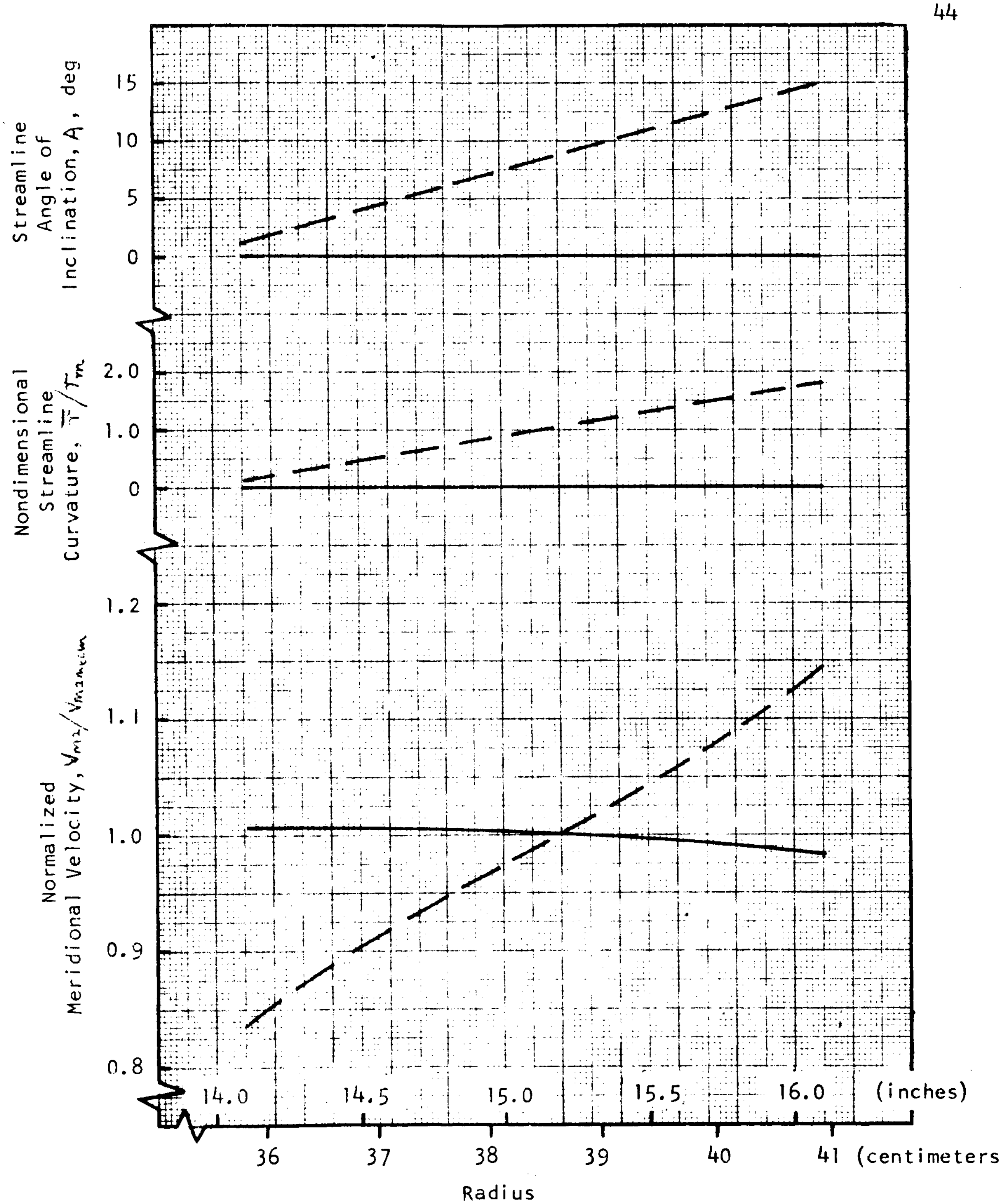


FIGURE 10 - THE EFFECT OF THE MERIDIONAL COMPONENTS OF STREAMLINE SLOPE AND CURVATURE ON THE MERIDIONAL VELOCITY DISTRIBUTION AT A STAGE EXIT

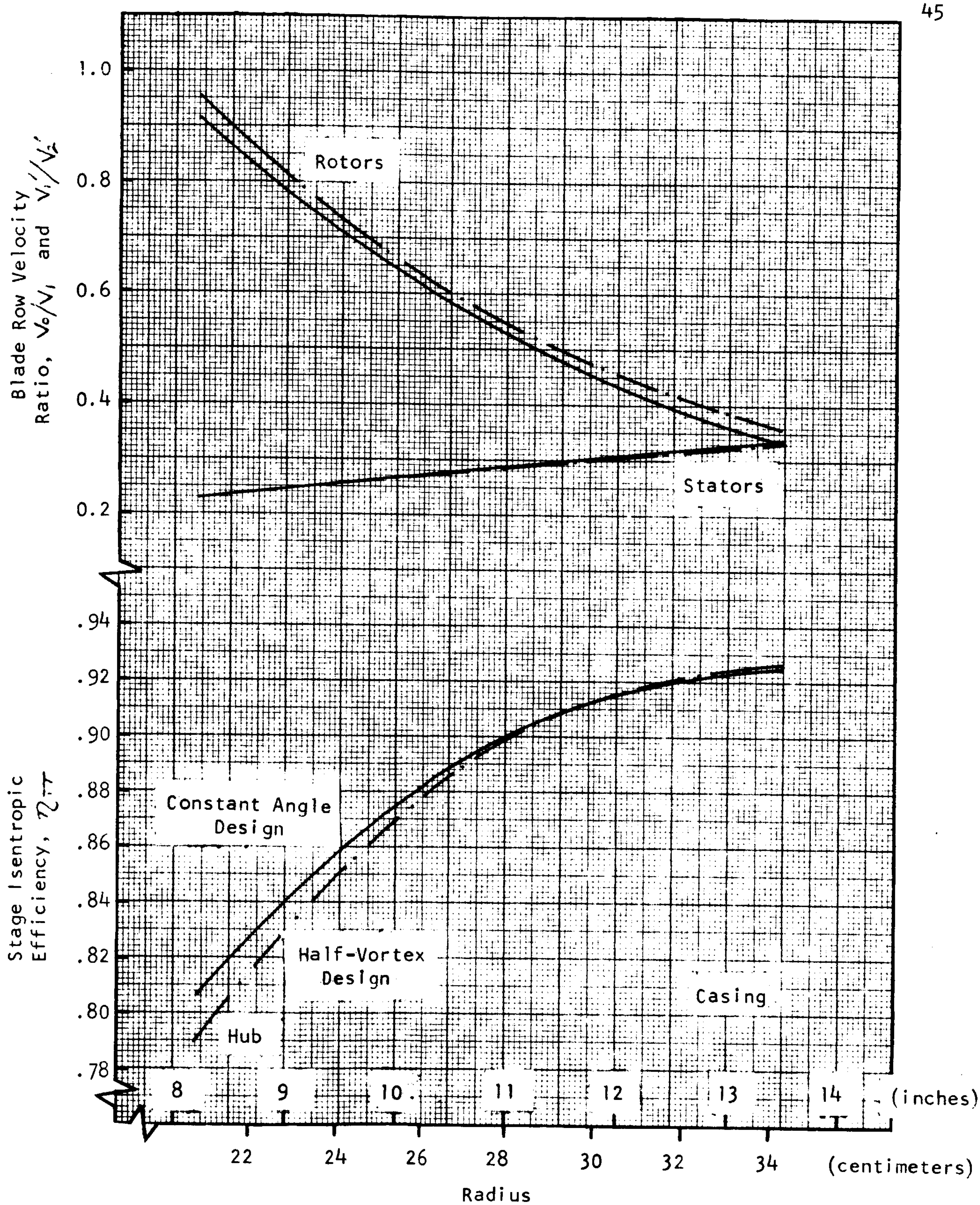


FIGURE 11 - THE VARIATIONS OF LOCAL TOTAL-TO-TOTAL EFFICIENCY AND ROW VELOCITY RATIOS FOR ALTERNATIVE STATOR SPECIFICATION (TURBINE D)

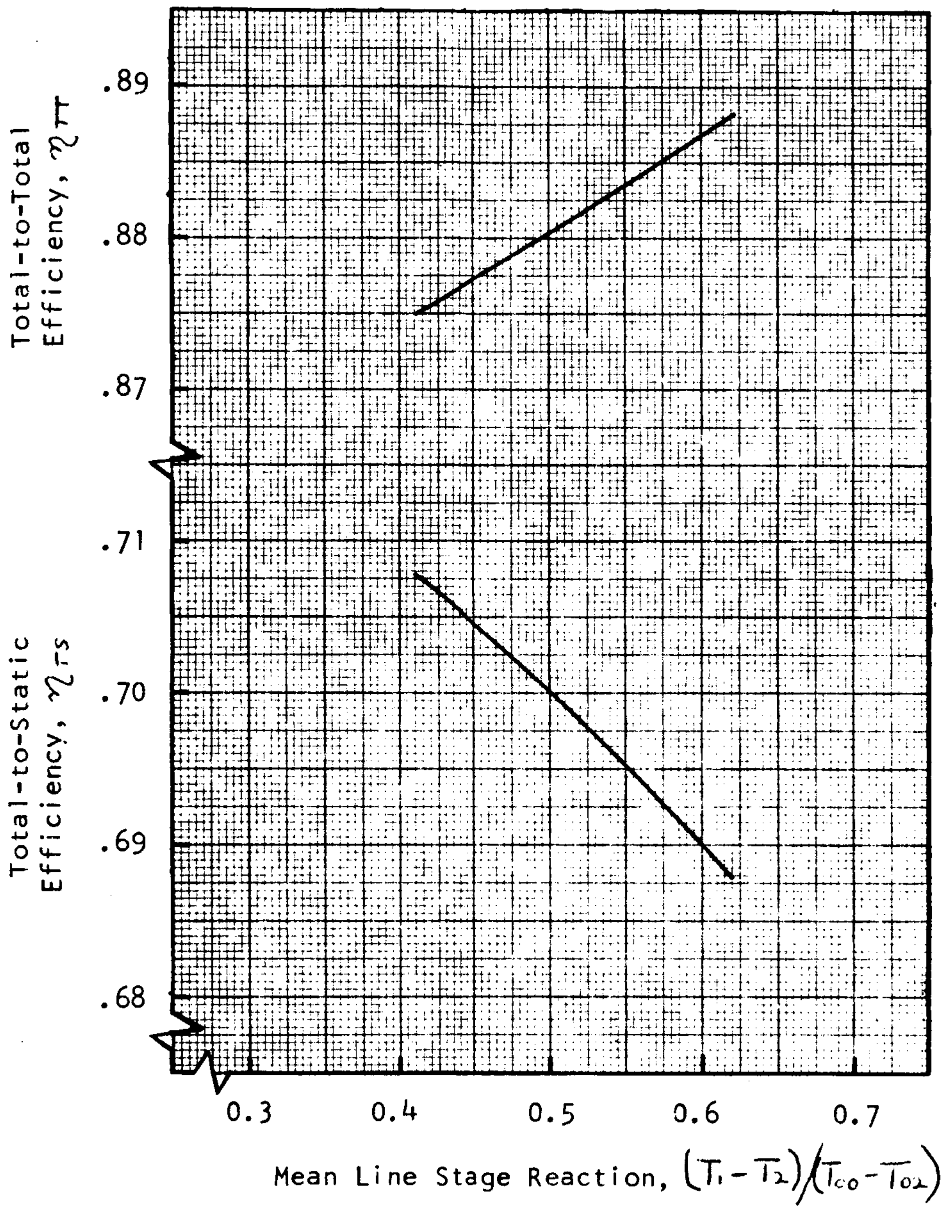


FIGURE 12 - VARIATIONS OF TOTAL-TO-TOTAL AND TOTAL-TO-STATIC EFFICIENCIES WITH MEAN STREAMLINE STAGE REACTION (TURBINE D)

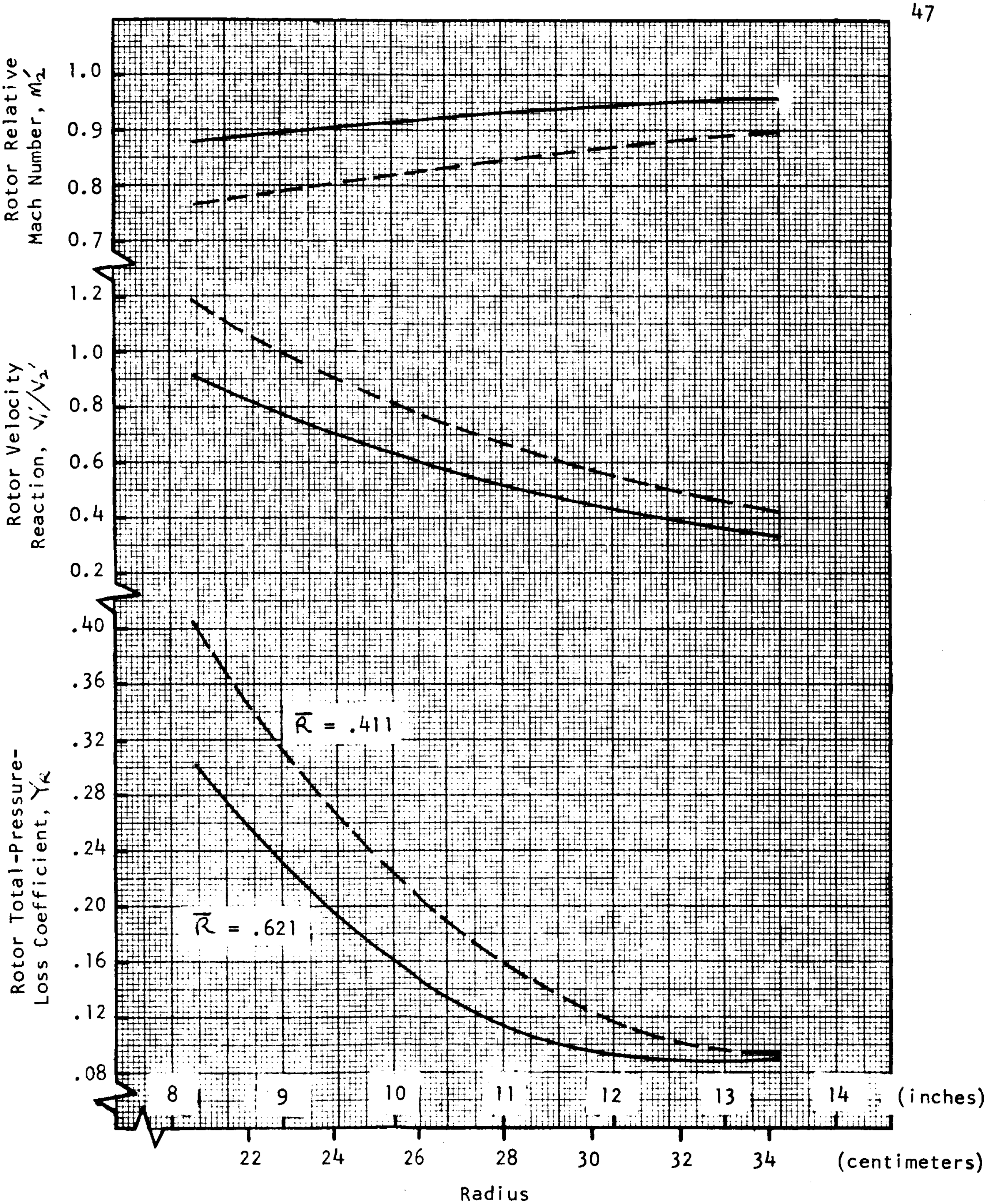


FIGURE 13 - ROTOR EXIT MACH NUMBER, VELOCITY RATIO, AND TOTAL-PRESSURE-LOSS COEFFICIENT VARIATION FOR TWO LEVELS OF MEAN LINE STAGE REACTION (TURBINE D)

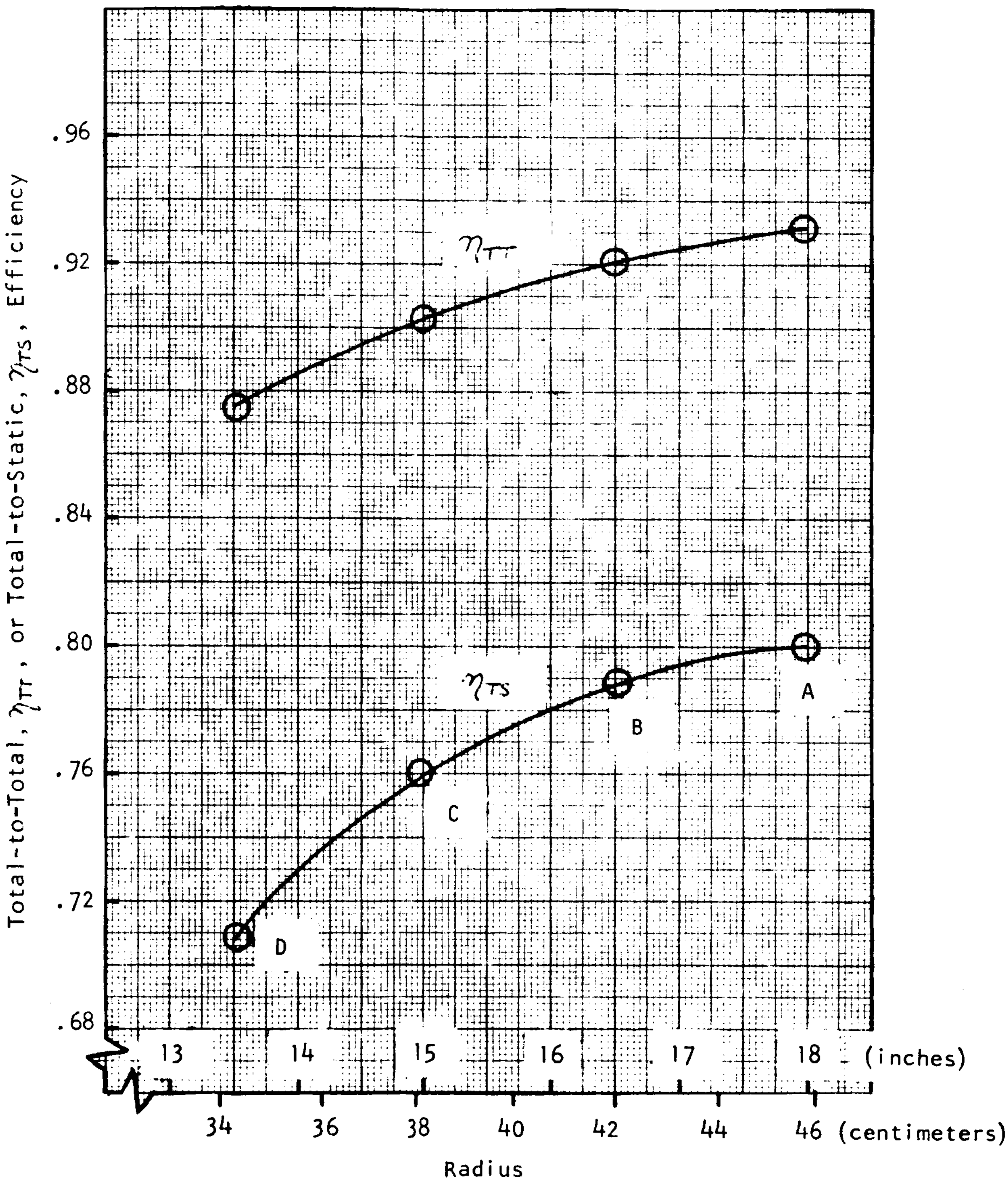
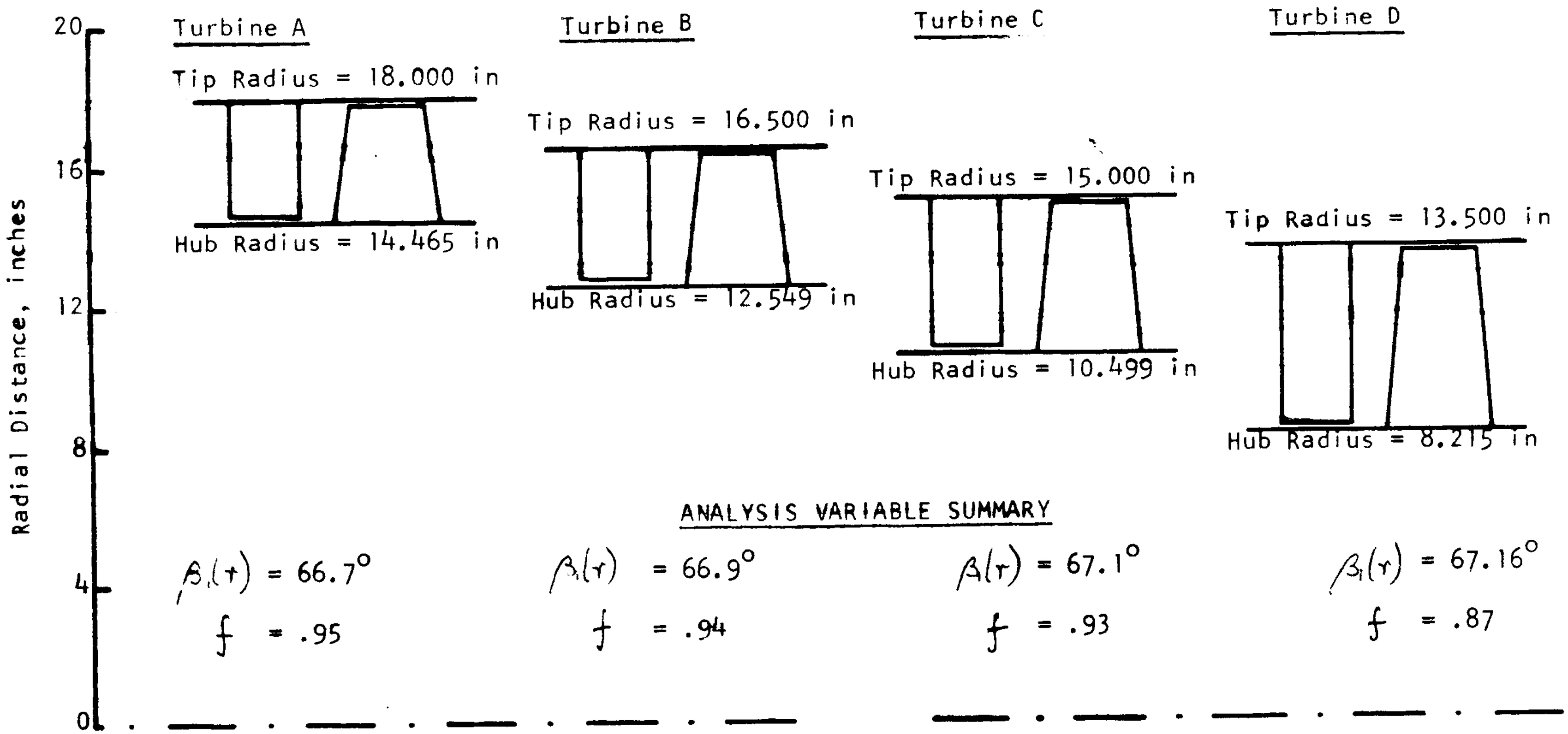


FIGURE 14 - TOTAL-TO-TOTAL AND TOTAL-TO-STATIC EFFICIENCY VARIATIONS WITH STAGE TIP RADIUS (TURBINES A, B, C, AND D)



ANALYSIS VARIABLE SUMMARY

	Turbine A	Turbine B	Turbine C	Turbine D
$\beta_1(r)$	= 66.7°	= 66.9°	= 67.1°	= 67.16°
f	= .95	= .94	= .93	= .87

PERFORMANCE SUMMARY

Total-to-Total Efficiency	= .931	.920	.902	.875
Total-to-Static Efficiency	= .800	.787	.759	.708
Total Pressure Ratio	= 1.940	1.957	1.986	2.034
Total-to-Static Pressure Ratio	= 2.192	2.223	2.299	2.462
Blade-to-Jet Speed Ratio	= .592	.526	.456	.378

FIGURE 15 - SUMMARY OF ANNULUS GEOMETRIES, SPECIFICATIONS, AND PERFORMANCE DATA FOR THE FOUR TURBINES

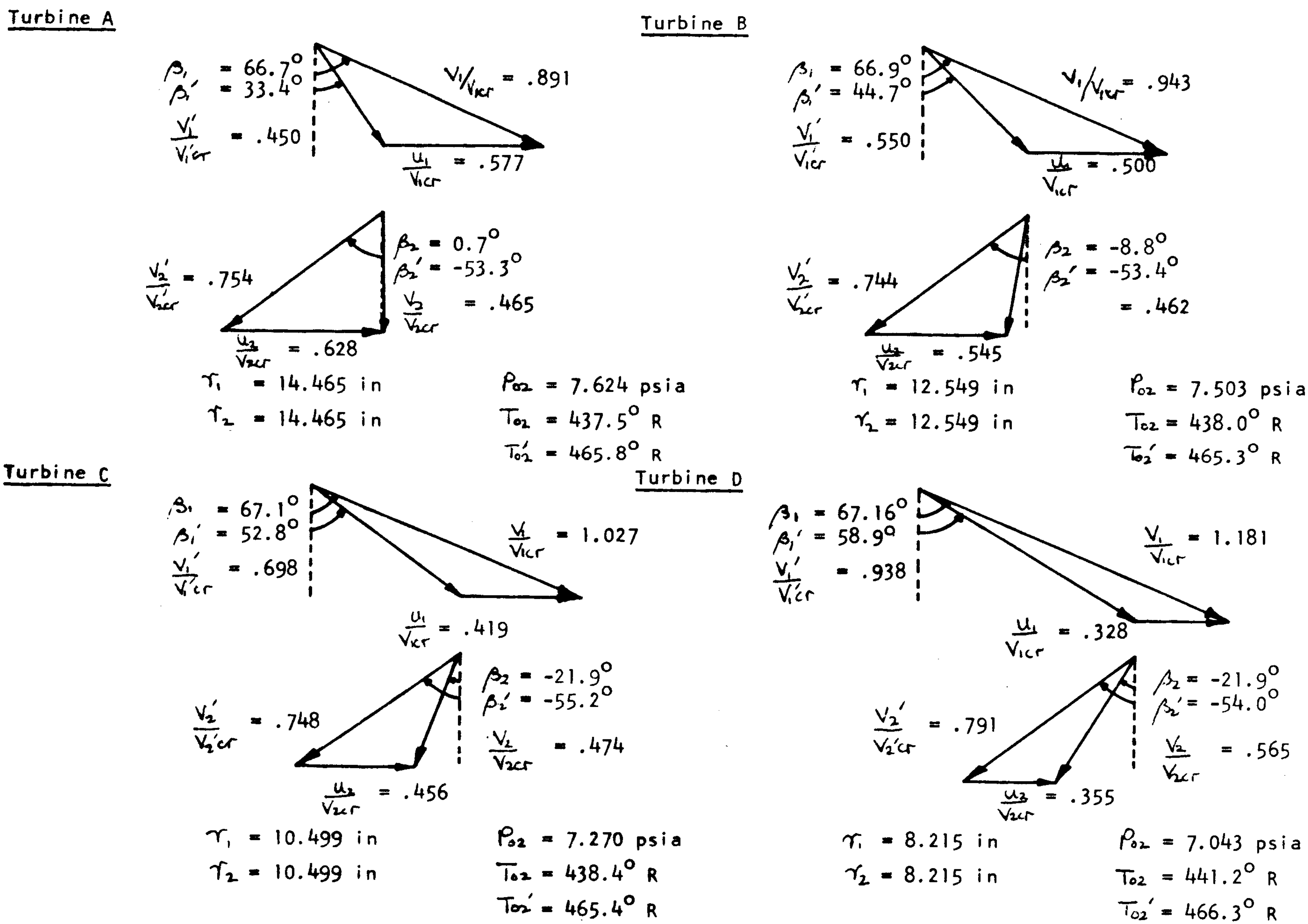


FIGURE 16 - COMPARISON OF THE HUB SECTION VELOCITY DIAGRAMS

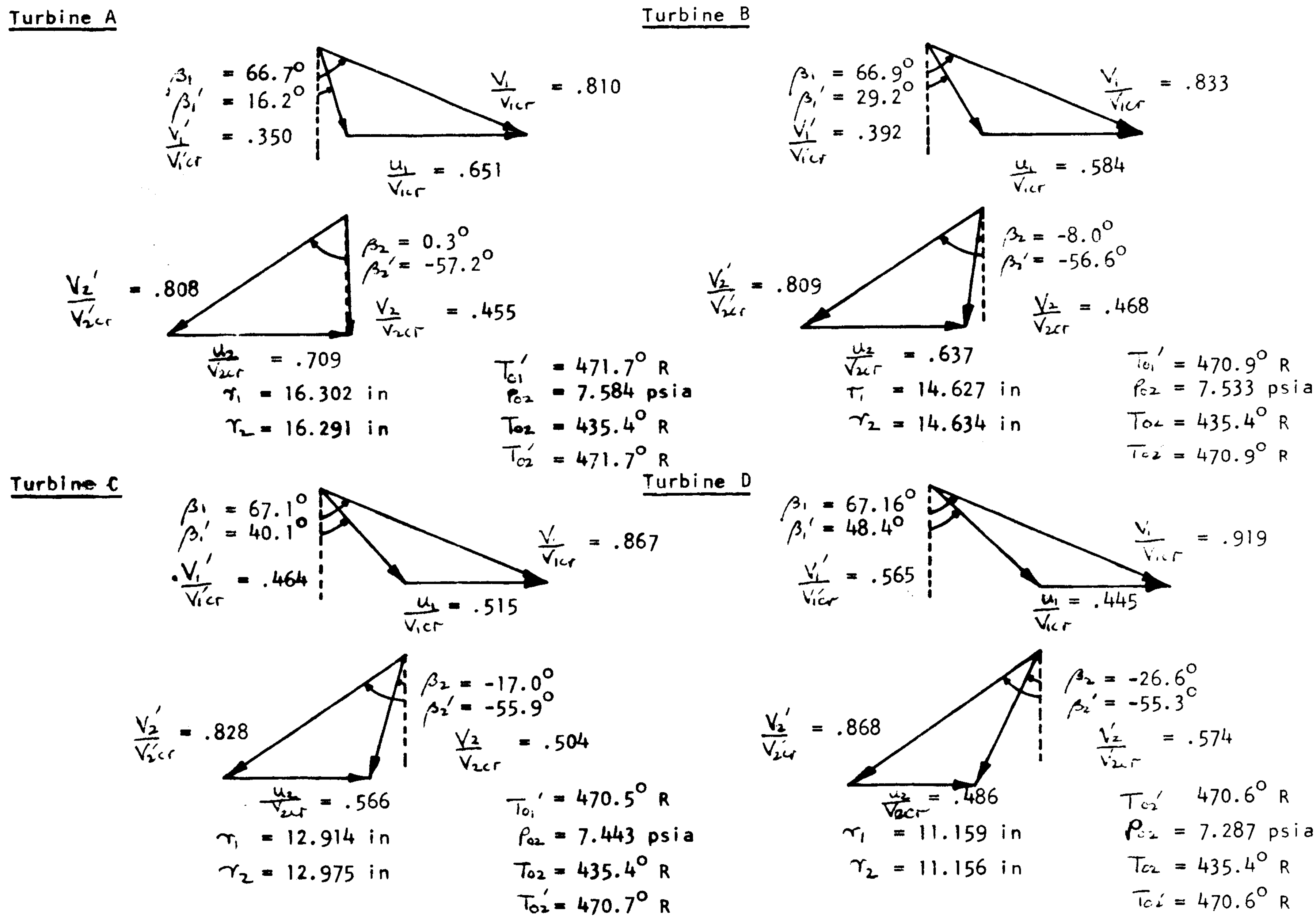


FIGURE 17 - COMPARISON OF THE MEAN SECTION VELOCITY DIAGRAMS

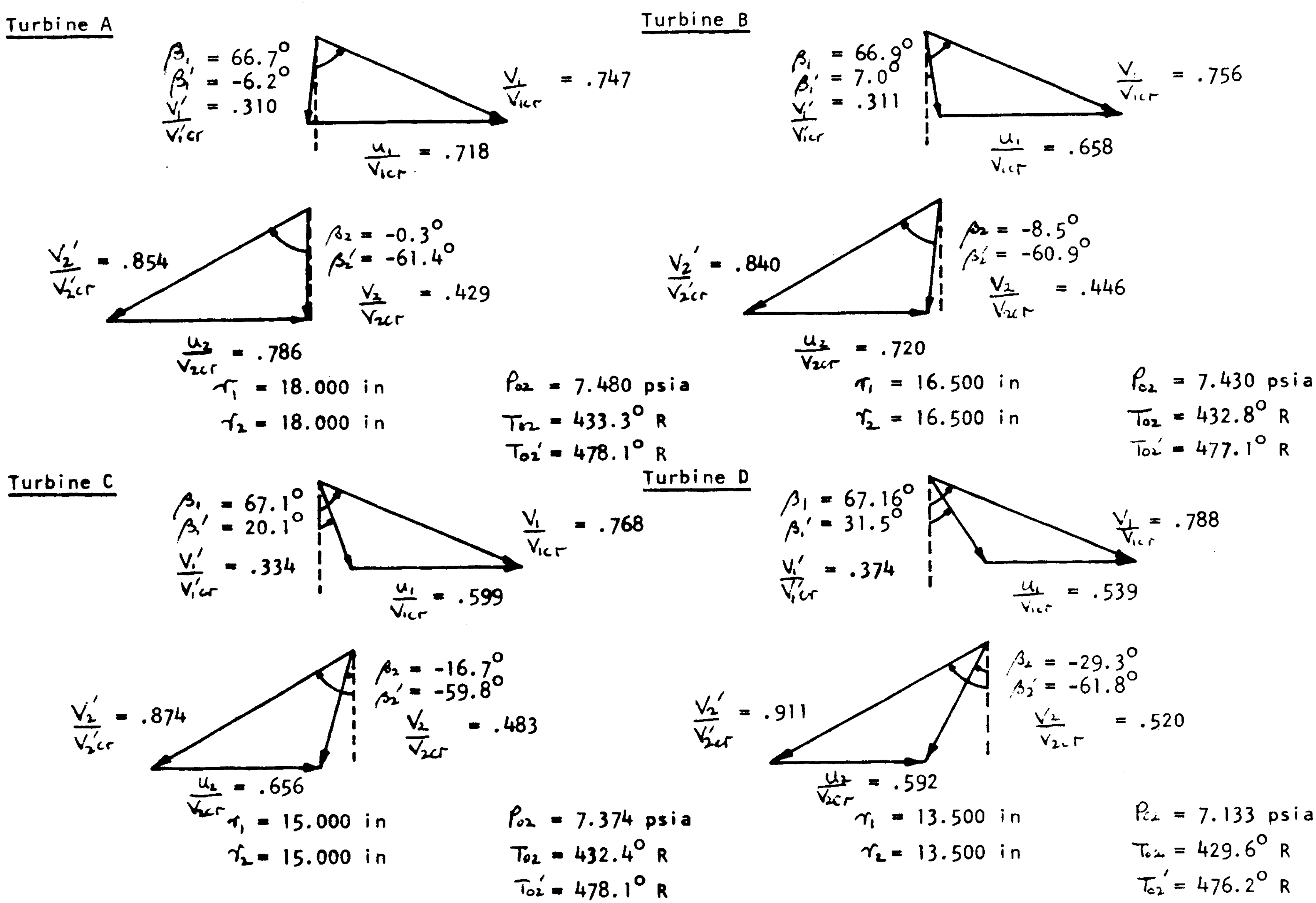


FIGURE 18 - COMPARISON OF THE CASING SECTION VELOCITY DIAGRAMS

NOMENCLATURE

Symbols	Description	Units
A	Angle of streamline slope in the meridional plane	deg
C_p	Specific heat at constant pressure	Btu/lbm deg R (J/kg deg K)
e	Kinetic-energy-loss coefficient ($= 1 - V_{ar}^2/V_{ans}^2$)	--
f	Index for parabolic power output distribution	--
g_o	Constant in Newton's law	lbm/lbf ft/sec^2
J	Mechanical equivalent of heat	ft lbf/Btu
j	Index on streamlines	--
n	Number of streamlines	--
P_o	Total pressure	psi (N/cm^2)
p	Nondimensional power function $= \frac{\omega_T J C_p}{550\, HP_T} \int_0^{\omega(r)} \Delta T_c \, d\omega(r)$	--
$\bar{R}$	Stage reaction $= \frac{T_1 - T_2}{T_{o0} - T_{o2}}$	--
r	Radius	in or ft (cms)
$\frac{1}{r_m}$	Meridional component of streamline curvature	ft^{-1} (m^{-1})
T_o	Total temperature	deg R (deg K)
u	Blade speed	ft/sec (m/sec)
V	Velocity	ft/sec (m/sec)
$\omega(r)$	Nondimensional mass flow function $= \frac{2\pi}{\omega_T R} \int_0^r \frac{P_o}{T_o} \left[1 - \frac{V^2}{2 g_o J C_p T_o} \right]^{\frac{1}{\gamma - 1}} V_m \cos A\, r\, dr$	--
β	Flow angle	deg

Symbols	Description	Units
η_{TT}	Total-to-total isentropic efficiency	--
η_{TS}	Total-to-static stage isentropic efficiency	--
ν	Blade-to-jet speed ratio = $\frac{\bar{u}}{\sqrt{2 j c_p \Delta T_{os}}}$	--

Subscripts	Description
c	Casing
cr	Condition at Mach 1
ex	Exit
h	Hub
in	Inlet
m	Meridional
s	Isentropic
u	Tangential
x	Axial
o	Stage inlet
1	Stator exit/rotor inlet
2	Stage exit

Superscripts	Description
$'$	Relative to rotor
$\bar{\ }$	Mean or mass flow weighted value

APPENDIX I

COMPUTER OUTPUT FOR TURBINE DESIGNS A, B, C, AND D

The following pages contain the full computer output for the four single-stage designs. The design requirements and loss coefficient correlation are held constant for the four designs as the tip diameter is decreased by 25 per cent from a value consistent with a conservative design.

** PROGRAM TD - AERODYNAMIC CALCULATIONS FOR THE DESIGN OF AXIAL TURBINES **

NASA SINGLE STAGE TURBINE

*** GENERAL INPUT DATA ***

NUMBER OF SPOOLS = 1
NUMBER OF SETS OF ANALYSIS VARIABLES = 1
NUMBER OF STREAMLINES = 9

GAS CONSTANT = 53.35000 LBF FT/LBM DEG R
INLET MASS FLOW = 45.51000 LBM/SEC

* TABULAR INLET SPECIFICATIONS *

RADIAL COORDINATE (IN)	TOTAL TEMPERATURE (DEG R)	TOTAL PRESSURE (PSI)	ABSOLUTE FLOW ANGLE (DEG)
16.0000	518.70	14.6960	0.

*** SPOOL INPUT DATA ***

** DESIGN REQUIREMENTS **

ROTATIVE SPEED = 4660.0 RPM
POWER OUTPUT = 1287.50 HP

** ANALYSIS VARIABLES **

NUMBER OF STAGES = 1

* POWER-OUTPUT SPLIT *

STAGE NUMBER	FRACTION OF SPOOL POWER OUTPUT
1	1.00000

* SPECIFIC-HEAT SPECIFICATION *

DESIGN STATION NUMBER	SPECIFIC HEAT (BTU/LBM DEG R)
1	0.24000
2	0.24000
3	0.24000

* ANNULUS SPECIFICATION *

STATION NUMBER	AXIAL POSITION (IN)	HUB RADIUS (IN)	CASING RADIUS (IN)
1	0.	14.4650	18.0000
2	1.0000	14.4650	18.0000
3	2.0000	14.4650	18.0000
4	3.0000	14.4650	18.0000
5	4.0000	14.4650	18.0000

• BLADE-ROW EXIT CONDITIONS •

STATOR 1

RADIAL POSITION (IN)	WHIRL ANGLE (DEG)
14.4650	66.700
15.3487	66.700
16.2325	66.700
17.1163	66.700
18.0000	66.700

ROTOR 1

STREAMLINE NUMBER	NONDIMENSIONAL POWER OUTPUT FUNCTION
1	0.
2	0.12220
3	0.24519
4	0.36859
5	0.49359
6	0.61899
7	0.74519
8	0.87220
9	1.00000

• BASIC INTERNAL LOSS CORRELATION •

```
          TAN(INLET ANGLE) + TAN(EXIT ANGLE)              (0.02999999 + 0.15725499 * (V RATIO)** 3.60)  IF (V RATIO) .LT. 0.60000000 *
Y = ------------------------------------------ *TIMES*
    0.60000000 * 0.80000000 * COS(EXIT ANGLE)             (0.05500000 + 0.15000000 *((V RATIO)-0.600))  IF (V RATIO) .GT. 0.60000000
```

THE PRESSURE-LOSS COEFFICIENT COMPUTED IN THIS MANNER MAY NOT EXCEED A LIMIT OF 1.00000000

*** OUTPUT OF SPOOL DESIGN ANALYSIS ***

** STATOR INLET 1 **

STREAMLINE NUMBER	RADIAL POSITION (IN)	MASS-FLOW FUNCTION (LBM/SEC)	MERIDIONAL VELOCITY (FPS)	AXIAL VELOCITY (FPS)	WHIRL VELOCITY (FPS)	ABSOLUTE VELOCITY (FPS)	ABSOLUTE MACH NUMBER	ABSOLUTE TOTAL PRESSURE (PSI)	ABSOLUTE TOTAL TEMPERATURE (DEG R)	ABSOLUTE FLOW ANGLE (DEG)
1	14.4650	0.	243.432	243.432	0.	243.432	0.21909	14.6960	518.70	0.
2	14.9526	5.68875	243.432	243.432	0.	243.432	0.21909	14.6960	518.70	0.
3	15.4249	11.37750	243.432	243.432	0.	243.432	0.21909	14.6960	518.70	0.
4	15.8831	17.06625	243.432	243.432	0.	243.432	0.21909	14.6960	518.70	0.
5	16.3284	22.75500	243.432	243.432	0.	243.432	0.21909	14.6960	518.70	0.
6	16.7620	28.44375	243.432	243.432	0.	243.432	0.21909	14.6960	518.70	0.
7	17.1846	34.13250	243.432	243.432	0.	243.432	0.21909	14.6960	518.70	0.
8	17.5970	39.82125	243.432	243.432	0.	243.432	0.21909	14.6960	518.70	0.
9	18.0000	45.51000	243.432	243.432	0.	243.432	0.21909	14.6960	518.70	0.

STREAMLINE NUMBER	STATIC PRESSURE (PSI)	STATIC TEMPERATURE (DEG R)	STREAMLINE SLOPE ANGLE (DEG)	STREAMLINE CURVATURE (PER IN)
1	14.2127	513.77	0.	0.
2	14.2127	513.77	0.	0.
3	14.2127	513.77	0.	0.
4	14.2127	513.77	0.	0.
5	14.2127	513.77	0.	0.
6	14.2127	513.77	0.	0.
7	14.2127	513.77	0.	0.
8	14.2127	513.77	0.	0.
9	14.2127	513.77	0.	0.

** STATOR EXIT - ROTOR INLET 1 **

STREAMLINE NUMBER	RADIAL POSITION (IN)	MASS-FLOW FUNCTION (LBM/SEC)	MERIDIONAL VELOCITY (FPS)	AXIAL VELOCITY (FPS)	WHIRL VELOCITY (FPS)	ABSOLUTE VELOCITY (FPS)	ABSOLUTE MACH NUMBER	ABSOLUTE TOTAL PRESSURE (PSI)	ABSOLUTE TOTAL TEMPERATURE (DEG R)	ABSOLUTE FLOW ANGLE (DEG)
1	14.4650	0.	359.289	359.289	834.260	908.338	0.87348	14.2523	518.70	66.700
2	14.9410	5.68853	350.004	350.004	812.699	884.863	0.84761	14.2689	518.70	66.700
3	15.4050	11.37708	341.460	341.460	792.861	863.264	0.82407	14.2841	518.70	66.700
4	15.8584	17.06564	333.556	333.556	774.508	843.281	0.80251	14.2979	518.70	66.700
5	16.3023	22.75422	326.208	326.208	757.447	824.705	0.78264	14.3107	518.70	66.700
6	16.7377	28.44282	319.349	319.349	741.520	807.364	0.76424	14.3225	518.70	66.700
7	17.1653	34.13143	312.922	312.922	726.597	791.116	0.74712	14.3334	518.70	66.700
8	17.5859	39.82005	306.881	306.881	712.569	775.841	0.73114	14.3435	518.70	66.700
9	18.0000	45.50869	301.184	301.184	699.341	761.439	0.71616	14.3529	518.70	66.700

STREAMLINE NUMBER	STATIC PRESSURE (PSI)	STATIC TEMPERATURE (DEG R)	STREAMLINE SLOPE ANGLE (DEG)	STREAMLINE CURVATURE (PER IN)	BLADE VELOCITY (FPS)	RELATIVE VELOCITY (FPS)	RELATIVE MACH NUMBER	RELATIVE TOTAL PRESSURE (PSI)	RELATIVE TOTAL TEMPERATURE (DEG R)	RELATIVE FLOW ANGLE (DEG)
1	8.6702	450.04	0.	0.	588.236	435.450	0.41874	9.7818	465.82	34.401
2	8.9191	453.55	0.	0.	607.591	405.674	0.38860	9.8980	467.24	30.371
3	9.1470	456.69	0.	0.	626.461	379.848	0.36260	10.0169	468.69	25.981
4	9.3566	459.53	0.	0.	644.899	357.852	0.34055	10.1384	470.18	21.235
5	9.5502	462.10	0.	0.	662.951	339.619	0.32230	10.2628	471.70	16.155
6	9.7297	464.46	0.	0.	680.657	325.097	0.30773	10.3900	473.25	10.790
7	9.8966	466.67	0.	0.	698.048	314.222	0.29675	10.5201	474.84	5.213
8	10.0524	468.61	0.	0.	715.151	306.891	0.28921	10.6533	476.45	-0.482
9	10.1981	470.45	0.	0.	731.991	302.948	0.28493	10.7894	478.09	-6.187

** STAGE EXIT 1 **

STREAMLINE NUMBER	RADIAL POSITION (IN)	MASS-FLOW FUNCTION (LBM/SEC)	MERIDIONAL VELOCITY (FPS)	AXIAL VELOCITY (FPS)	WHIRL VELOCITY (FPS)	ABSOLUTE VELOCITY (FPS)	ABSOLUTE MACH NUMBER	ABSOLUTE TOTAL PRESSURE (PSI)	ABSOLUTE TOTAL TEMPERATURE (DEG R)	ABSOLUTE FLOW ANGLE (DEG)
1	14.4650	0.	435.073	435.073	5.032	435.102	0.43220	7.6243	437.52	0.663
2	14.9396	5.68869	435.111	435.111	4.605	435.136	0.43251	7.6257	436.99	0.606
3	15.4002	11.37738	433.083	433.083	3.986	433.102	0.43069	7.6176	436.45	0.527
4	15.8498	17.06609	429.609	429.609	3.211	429.621	0.42737	7.6030	435.92	0.428
5	16.2908	22.75482	425.072	425.072	2.313	425.079	0.42296	7.5837	435.39	0.312
6	16.7250	28.44361	419.710	419.710	1.314	419.712	0.41770	7.5610	434.85	0.179
7	17.1538	34.13243	413.666	413.666	0.234	413.666	0.41173	7.5358	434.32	0.032
8	17.5784	39.82128	407.028	407.028	-0.911	407.029	0.40517	7.5085	433.78	-0.128
9	18.0000	45.51007	399.835	399.835	-2.109	399.841	0.39804	7.4795	433.25	-0.302

STREAMLINE NUMBER	STATIC PRESSURE (PSI)	STATIC TEMPERATURE (DEG R)	STREAMLINE SLOPE ANGLE (DEG)	STREAMLINE CURVATURE (PER IN)	BLADE VELOCITY (FPS)	RELATIVE VELOCITY (FPS)	RELATIVE MACH NUMBER	RELATIVE TOTAL PRESSURE (PSI)	RELATIVE TOTAL TEMPERATURE (DEG R)	RELATIVE FLOW ANGLE (DEG)
1	6.7058	421.77	0.	0.	588.236	727.609	0.72276	9.4949	465.82	-53.277
2	6.7059	421.23	0.	0.	607.537	743.538	0.73905	9.6390	467.24	-54.184
3	6.7058	420.84	0.	0.	626.266	758.151	0.75392	9.7748	468.67	-55.164
4	6.7058	420.56	0.	0.	644.551	771.933	0.76789	9.9059	470.14	-56.183
5	6.7058	420.35	0.	0.	662.486	785.185	0.78127	10.0350	471.65	-57.223
6	6.7058	420.19	0.	0.	680.143	798.101	0.79427	10.1638	473.20	-58.272
7	6.7058	420.08	0.	0.	697.580	810.809	0.80702	10.2934	474.78	-59.324
8	6.7058	420.00	0.	0.	714.848	823.397	0.81963	10.4249	476.41	-60.375
9	6.7059	419.95	0.	0.	731.991	835.925	0.83215	10.5587	478.09	-61.425

** STAGE 1 PERFORMANCE **

STREAMLINE NUMBER	STATOR REACTION	ROTOR REACTION	STATOR PRESSURE LOSS COEFFICIENT	ROTOR PRESSURE LOSS COEFFICIENT	STATOR BLADE ROW EFFICIENCY	ROTOR BLADE ROW EFFICIENCY	ROTOR ISENTROPIC EFFICIENCY	STAGE ISENTROPIC EFFICIENCY
1	0.26800	0.59847	0.07949	0.10286	0.94593	0.92501	0.95635	0.91555
2	0.27511	0.54560	0.07984	0.08815	0.94482	0.93547	0.96127	0.92181
3	0.28199	0.50102	0.08019	0.07840	0.94378	0.94259	0.96457	0.92648
4	0.28867	0.46358	0.08056	0.07178	0.94279	0.94754	0.96677	0.93002
5	0.29517	0.43253	0.08094	0.06724	0.94185	0.95104	0.96821	0.93275
6	0.30151	0.40734	0.08133	0.06412	0.94096	0.95354	0.96911	0.93489
7	0.30771	0.38754	0.08173	0.06201	0.94010	0.95532	0.96961	0.93657
8	0.31376	0.37271	0.08215	0.06066	0.93926	0.95658	0.96979	0.93788
9	0.31970	0.36241	0.08258	0.05990	0.93845	0.95742	0.96973	0.93887

* MASS-AVERAGED QUANTITIES *

STATOR BLADE-ROW EFFICIENCY = 0.94197

ROTOR BLADE-ROW EFFICIENCY = 0.94791

STAGE WORK = 19.596 BTU PER LBM
STAGE TOTAL EFFICIENCY = 0.93102
STAGE STATIC EFFICIENCY = 0.79996
STAGE BLADE- TO JET-SPEED RATIO = 0.59158

*** SPOOL PERFORMANCE SUMMARY (MASS-AVERAGED QUANTITIES) ***

SPOOL WORK = 19.596 BTU PER LBM
SPOOL POWER = 1287.50 HP
SPOOL TOTAL- TO TOTAL-PRESSURE RATIO = 1.54047
SPOOL TOTAL- TO STATIC-PRESSURE RATIO = 2.19152
SPOOL TOTAL EFFICIENCY = 0.93102
SPOOL STATIC EFFICIENCY = 0.79996
SPOOL BLADE- TO JET-SPEED RATIO = 0.59158

** PROGRAM TD - AERODYNAMIC CALCULATIONS FOR THE DESIGN OF AXIAL TURBINES **

HIGHLY LOADED SINGLE STAGE DESIGNS - HUB/TIP RATIO = .61,.70,.76 RESP.

*** GENERAL INPUT DATA ***

NUMBER OF SPOOLS = 1
NUMBER OF SETS OF ANALYSIS VARIABLES = 3
NUMBER OF STREAMLINES = 9

GAS CONSTANT = 53.35000 LBF FT/LBM DEG R
INLET MASS FLOW = 45.51000 LBM/SEC

* TABULAR INLET SPECIFICATIONS *

RADIAL COORDINATE (IN)	TOTAL TEMPERATURE (DEG R)	TOTAL PRESSURE (PSI)	ABSOLUTE FLOW ANGLE (DEG)
10.0000	518.70	14.6960	0.

*** SPOOL INPUT DATA ***

** DESIGN REQUIREMENTS **

ROTATIVE SPEED = 4660.0 RPM
POWER OUTPUT = 1287.50 HP

** SET 3 OF ANALYSIS VARIABLES **

NUMBER OF STAGES = 1

* POWER-OUTPUT SPLIT *

STAGE NUMBER	FRACTION OF SPOOL POWER OUTPUT
1	1.CC000.

* SPECIFIC-HEAT SPECIFICATION *

DESIGN STATION NUMBER	SPECIFIC HEAT (BTU/LBM DEG R)
1	0.2400C
2	C.24000
3	0.24000

* ANNULUS SPECIFICATION *

STATION NUMBER	AXIAL POSITION (IN)	HUB RADIUS (IN)	CASING RADIUS (IN)
1	0.	12.549C	16.5000
2	1.0000	12.5490	16.5000
3	2.0000	12.5490	16.5000
4	3.0000	12.549C	16.5000
5	4.0000	12.549C	16.5000

* BLADE-ROW EXIT CONDITIONS *

STATOR 1	RADIAL POSITION (IN)	WHIRL ANGLE (DEG)
	13.0000	66.900

ROTOR 1	STREAMLINE NUMBER	NONDIMENSIONAL POWER OUTPUT FUNCTION
	1	0.
	2	0.12162
	3	0.24420
	4	0.36775
	5	0.49227
	6	0.61775
	7	0.74420
	8	0.87162
	9	1.00000

* BASIC INTERNAL LOSS CORRELATION *

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        TAN(INLET ANGLE) + TAN(EXIT ANGLE)                  (0.02999999 + 0.15725499 * (V RATIO)** 3.60)  IF (V RATIO) .LT. 0.60000000
Y = ------------------------------------------ *TIMES*
     0.60000000 + 0.80000000 * COS(EXIT ANGLE)              (0.05500000 + 0.15000000 *((V RATIO)-0.600))  IF (V RATIO) .GT. 0.60000000 *
```

THE PRESSURE-LOSS COEFFICIENT COMPUTED IN THIS MANNER MAY NOT EXCEED A LIMIT OF 1.00000000

*** OUTPUT OF SPOOL DESIGN ANALYSIS (SET 3 OF ANALYSIS VARIABLES) ***

** STATOR INLET 1 **

STREAMLINE NUMBER	RADIAL POSITION (IN)	MASS-FLOW FUNCTION (LBM/SEC)	MERIDIONAL VELOCITY (FPS)	AXIAL VELOCITY (FPS)	WHIRL VELOCITY (FPS)	ABSOLUTE VELOCITY (FPS)	ABSOLUTE MACH NUMBER	ABSOLUTE TOTAL PRESSURE (PSI)	ABSOLUTE TOTAL TEMPERATURE (DEG R)	ABSOLUTE FLOW ANGLE (DEG)
1	12.5490	0.	243.412	243.412	0.	243.412	0.21907	14.6960	518.70	0.
2	13.1082	5.68875	243.412	243.412	0.	243.412	0.21907	14.6960	518.70	0.
3	13.6444	11.37750	243.412	243.412	0.	243.412	0.21907	14.6960	518.70	0.
4	14.1604	17.06625	243.412	243.412	0.	243.412	0.21907	14.6960	518.70	0.
5	14.6582	22.75500	243.412	243.412	0.	243.412	0.21907	14.6960	518.70	0.
6	15.1397	28.44375	243.412	243.412	0.	243.412	0.21907	14.6960	518.70	0.
7	15.6063	34.13250	243.412	243.412	0.	243.412	0.21907	14.6960	518.70	0.
8	16.0594	39.82125	243.412	243.412	0.	243.412	0.21907	14.6960	518.70	0.
9	16.5000	45.51000	243.412	243.412	0.	243.412	0.21907	14.6960	518.70	0.

STREAMLINE NUMBER	STATIC PRESSURE (PSI)	STATIC TEMPERATURE (DEG R)	STREAMLINE SLOPE ANGLE (DEG)	STREAMLINE CURVATURE (PER IN)
1	14.2128	513.77	0.	0.
2	14.2128	513.77	0.	0.
3	14.2128	513.77	0.	0.
4	14.2128	513.77	0.	0.
5	14.2128	513.77	0.	0.
6	14.2128	513.77	0.	0.
7	14.2128	513.77	0.	0.
8	14.2128	513.77	0.	0.
9	14.2128	513.77	0.	0.

** STATOR EXIT - ROTOR INLET 1 **

STREAMLINE NUMBER	RADIAL POSITION (IN)	MASS-FLOW FUNCTION (LBM/SEC)	MERIDIONAL VELOCITY (FPS)	AXIAL VELOCITY (FPS)	WHIRL VELOCITY (FPS)	ABSOLUTE VELOCITY (FPS)	ABSOLUTE MACH NUMBER	ABSOLUTE TOTAL PRESSURE (PSI)	ABSOLUTE TOTAL TEMPERATURE (DEG R)	ABSOLUTE FLOW ANGLE (DEG)
1	12.5490	0.	377.093	377.093	884.089	961.146	0.93281	14.2086	518.70	66.900
2	13.0948	5.68851	364.273	364.273	854.029	928.468	0.89590	14.2324	518.70	66.900
3	13.6212	11.37706	352.807	352.807	827.144	899.244	0.86342	14.2534	518.70	66.900
4	14.1312	17.06563	342.451	342.451	802.865	872.849	0.83449	14.2723	518.70	66.900
5	14.6271	22.75422	333.021	333.021	780.757	848.814	0.80846	14.2892	518.70	66.900
6	15.1106	28.44284	324.375	324.375	760.487	826.776	0.78485	14.3046	518.70	66.900
7	15.5831	34.13148	316.401	316.401	741.791	806.451	0.76327	14.3185	518.70	66.900
8	16.0463	39.82014	309.009	309.009	724.461	787.610	0.74344	14.3313	518.70	66.900
9	16.5000	45.50882	302.126	302.126	708.324	770.067	0.72512	14.3430	518.70	66.900

STREAMLINE NUMBER	STATIC PRESSURE (PSI)	STATIC TEMPERATURE (DEG R)	STREAMLINE SLOPE ANGLE (DEG)	STREAMLINE CURVATURE (PER IN)	BLADE VELOCITY (FPS)	RELATIVE VELOCITY (FPS)	RELATIVE MACH NUMBER	RELATIVE TOTAL PRESSURE (PSI)	RELATIVE TOTAL TEMPERATURE (DEG R)	RELATIVE FLOW ANGLE (DEG)
1	8.1038	441.83	0.	0.	510.320	530.941	0.51529	9.7126	465.29	44.746
2	8.4527	446.97	0.	0.	532.514	485.864	0.46882	9.8261	466.61	41.432
3	8.7635	451.41	0.	0.	553.922	446.232	0.42846	9.9422	467.98	37.755
4	9.0428	455.30	0.	0.	574.662	411.521	0.39344	10.0611	469.40	33.679
5	9.2955	458.75	0.	0.	594.827	381.409	0.36328	10.1829	470.85	29.175
6	9.5255	461.82	0.	0.	614.489	355.717	0.33768	10.3077	472.35	24.232
7	9.7360	464.58	0.	0.	633.707	334.353	0.31645	10.4357	473.88	18.861
8	9.9294	467.08	0.	0.	652.527	317.271	0.29948	10.5668	475.46	13.104
9	10.1079	469.35	0.	0.	670.992	304.424	0.28666	10.7013	477.07	7.044

** STAGE EXIT 1 **

STREAMLINE NUMBER	RADIAL POSITION (IN)	MASS-FLOW FUNCTION (LBM/SEC)	MERIDIONAL VELOCITY (FPS)	AXIAL VELOCITY (FPS)	WHIRL VELOCITY (FPS)	ABSOLUTE VELOCITY (FPS)	ABSOLUTE MACH NUMBER	ABSOLUTE TOTAL PRESSURE (PSI)	ABSOLUTE TOTAL TEMPERATURE (DEG R)	ABSOLUTE FLOW ANGLE (DEG)
1	12.5490	0.	427.979	427.979	-66.567	433.125	0.42994	7.5027	437.96	-8.841
2	13.1074	5.68878	432.928	432.928	-64.211	437.663	0.43495	7.5258	437.32	-8.437
3	13.6369	11.37755	435.971	435.971	-62.579	440.440	0.43815	7.5415	436.67	-8.168
4	14.1439	17.06628	435.948	435.948	-61.496	440.264	0.43830	7.5436	436.03	-8.029
5	14.6343	22.75500	433.142	433.142	-60.833	437.393	0.43567	7.5333	435.39	-7.995
6	15.1123	28.44372	428.457	428.457	-60.496	432.707	0.43116	7.5147	434.74	-8.037
7	15.5811	34.13248	422.399	422.399	-60.417	426.698	0.42528	7.4903	434.10	-8.140
8	16.0430	39.82125	415.262	415.262	-60.545	419.652	0.41833	7.4618	433.45	-8.295
9	16.5000	45.51000	407.206	407.206	-60.838	411.726	0.41048	7.4301	432.81	-8.497

STREAMLINE NUMBER	STATIC PRESSURE (PSI)	STATIC TEMPERATURE (DEG R)	STREAMLINE SLOPE ANGLE (DEG)	STREAMLINE CURVATURE (PER IN)	BLADE VELOCITY (FPS)	RELATIVE VELOCITY (FPS)	RELATIVE MACH NUMBER	RELATIVE TOTAL PRESSURE (PSI)	RELATIVE TOTAL TEMPERATURE (DEG R)	RELATIVE FLOW ANGLE (DEG)
1	6.6076	422.35	0.	0.	510.320	718.306	0.71303	9.2732	465.29	-53.429
2	6.6086	421.38	0.	0.	533.026	737.644	0.73307	9.4465	466.66	-54.062
3	6.6099	420.53	0.	0.	554.559	755.600	0.75167	9.6143	468.04	-54.761
4	6.6112	419.90	0.	0.	575.178	771.625	0.76818	9.7688	469.44	-55.600
5	6.6123	419.47	0.	0.	595.120	786.057	0.78296	9.9115	470.88	-56.562
6	6.6134	419.16	0.	0.	614.560	799.547	0.79668	10.0478	472.36	-57.597
7	6.6144	418.95	0.	0.	633.624	812.474	0.80977	10.1811	473.88	-58.675
8	6.6154	418.80	0.	0.	652.409	825.073	0.82247	10.3138	475.44	-59.781
9	6.6163	418.70	0.	0.	670.992	837.491	0.83495	10.4475	477.07	-60.907

** STAGE 1 PERFORMANCE **

STREAMLINE NUMBER	STATOR REACTION	ROTOR REACTION	STATOR PRESSURE LOSS COEFFICIENT	ROTOR PRESSURE LOSS COEFFICIENT	STATOR BLADE ROW EFFICIENCY	ROTOR BLADE ROW EFFICIENCY	ROTOR ISENTROPIC EFFICIENCY	STAGE ISENTROPIC EFFICIENCY
1	0.25325	0.73916	0.07984	0.16484	0.94777	0.88555	0.93347	0.89079
2	0.26217	0.65867	0.08022	0.13494	0.94625	0.90493	0.94281	0.90164
3	0.27069	0.59057	0.08062	0.11060	0.94487	0.92123	0.95109	0.91135
4	0.27887	0.53332	0.08103	0.09373	0.94360	0.93288	0.95714	0.91886
5	0.28677	0.48522	0.08146	0.08292	0.94241	0.94055	0.96109	0.92430
6	0.29441	0.44490	0.08191	0.07585	0.94129	0.94572	0.96365	0.92833
7	0.30183	0.41152	0.08237	0.07118	0.94023	0.94924	0.96528	0.93136
8	0.30905	0.38454	0.08285	0.06814	0.93921	0.95165	0.96624	0.93365
9	0.31609	0.36349	0.08335	0.06626	0.93824	0.95326	0.96671	0.93539

* MASS-AVERAGED QUANTITIES *

STATOR BLADE-ROW EFFICIENCY = 0.94261

ROTOR BLADE-ROW EFFICIENCY = 0.93320

STAGE WORK = 19.996 BTU PER LBM
STAGE TOTAL EFFICIENCY = 0.92037
STAGE STATIC EFFICIENCY = 0.78738
STAGE BLADE- TO JET-SPEED RATIO = 0.52634

*** SPOOL PERFORMANCE SUMMARY (MASS-AVERAGED QUANTITIES) ***

SPOOL WORK = 19.996 BTU PER LBM
SPOOL POWER = 1287.50 HP
SPOOL TOTAL- TO TOTAL-PRESSURE RATIO = 1.95694
SPOOL TOTAL- TO STATIC-PRESSURE RATIO = 2.22258
SPOOL TOTAL EFFICIENCY = 0.92037
SPOOL STATIC EFFICIENCY = 0.78738
SPOOL BLADE- TO JET-SPEED RATIO = 0.52634

** PROGRAM TD - AERODYNAMIC CALCULATIONS FOR THE DESIGN OF AXIAL TURBINES **

HIGHLY LOADED SINGLE STAGE DESIGNS - HUB/TIP RATIO = .61,.70,.76 RESP.

*** GENERAL INPUT DATA ***

NUMBER OF SPOOLS = 1
NUMBER OF SETS OF ANALYSIS VARIABLES = 3
NUMBER OF STREAMLINES = 9

GAS CONSTANT = 53.35000 LBF FT/LBM DEG R
INLET MASS FLOW = 45.51000 LBM/SEC

* TABULAR INLET SPECIFICATIONS *

RADIAL COORDINATE (IN)	TOTAL TEMPERATURE (DEG R)	TOTAL PRESSURE (PSI)	ABSOLUTE FLOW ANGLE (DEG)
10.0000	518.70	14.6960	0.

*** SPOOL INPUT DATA ***

** DESIGN REQUIREMENTS **

ROTATIVE SPEED = 4660.0 RPM
POWER OUTPUT = 1287.50 HP

** SET 2 OF ANALYSIS VARIABLES **

NUMBER OF STAGES = 1

* POWER-OUTPUT SPLIT *

STAGE NUMBER	FRACTION OF SPOOL POWER OUTPUT
1	1.00000

* SPECIFIC-HEAT SPECIFICATION *

DESIGN STATION NUMBER	SPECIFIC HEAT (BTU/LBM DEG R)
1	0.24000
2	0.24000
3	0.24000

* ANNULUS SPECIFICATION *

STATION NUMBER	AXIAL POSITION (IN)	HUB RADIUS (IN)	CASING RADIUS (IN)
1	0.	10.4990	15.0000
2	1.0000	10.4990	15.0000
3	2.0000	10.4990	15.0000
4	3.0000	10.4990	15.0000
5	4.0000	10.4990	15.0000

* BLADE-ROW EXIT CONDITIONS *

STATOR 1	RADIAL POSITION (IN)	WHIRL ANGLE (DEG)
	13.0000	67.100

ROTOR 1	STREAMLINE NUMBER	NONDIMENSIONAL POWER OUTPUT FUNCTION
	1	0.
	2	C.12103
	3	0.24320
	4	0.36650
	5	0.49093
	6	C.61650
	7	0.74320
	8	0.87103
	9	1.C0000

* BASIC INTERNAL LOSS CORRELATION *

```
         TAN(INLET ANGLE) + TAN(EXIT ANGLE)                (0.02999999 + 0.15725499 * (V RATIO)** 3.60)  IF (V RATIO) .LT. 0.60000000 *
Y = ------------------------------------------ *TIMES*
     0.60000000 * 0.80000000 * COS(EXIT ANGLE)             (0.05500000 * 0.15000000 *((V RATIO)-0.600))  IF (V RATIO) .GT. 0.60000000
```

THE PRESSURE-LOSS COEFFICIENT COMPUTED IN THIS MANNER MAY NOT EXCEED A LIMIT OF 1.00000000

*** OUTPUT OF SPOOL DESIGN ANALYSIS (SET 2 OF ANALYSIS VARIABLES) ***

** STATOR INLET 1 **

STREAMLINE NUMBER	RADIAL POSITION (IN)	MASS-FLOW FUNCTION (LBM/SEC)	MERIDIONAL VELOCITY (FPS)	AXIAL VELOCITY (FPS)	WHIRL VELOCITY (FPS)	ABSOLUTE VELOCITY (FPS)	ABSOLUTE MACH NUMBER	ABSOLUTE TOTAL PRESSURE (PSI)	ABSOLUTE TOTAL TEMPERATURE (DEG R)	ABSOLUTE FLOW ANGLE (DEG)
1	10.4990	0.	243.416	243.416	0.	243.416	0.21908	14.6960	518.70	0.
2	11.1613	5.68875	243.416	243.416	0.	243.416	0.21908	14.6960	518.70	0.
3	11.7865	11.37750	243.416	243.416	0.	243.416	0.21908	14.6960	518.70	0.
4	12.3802	17.06625	243.416	243.416	0.	243.416	0.21908	14.6960	518.70	0.
5	12.9466	22.75500	243.416	243.416	0.	243.416	0.21908	14.6960	518.70	0.
6	13.4893	28.44375	243.416	243.416	0.	243.416	0.21908	14.6960	518.70	0.
7	14.0113	34.13250	243.416	243.416	0.	243.416	0.21908	14.6960	518.70	0.
8	14.5139	39.82125	243.416	243.416	0.	243.416	0.21908	14.6960	518.70	0.
9	15.0000	45.51000	243.416	243.416	0.	243.416	0.21908	14.6960	518.70	0.

STREAMLINE NUMBER	STATIC PRESSURE (PSI)	STATIC TEMPERATURE (DEG R)	STREAMLINE SLOPE ANGLE (DEG)	STREAMLINE CURVATURE (PER IN)
1	14.2128	513.77	0.	0.
2	14.2128	513.77	0.	0.
3	14.2128	513.77	0.	0.
4	14.2128	513.77	0.	0.
5	14.2128	513.77	0.	0.
6	14.2128	513.77	0.	0.
7	14.2128	513.77	0.	0.
8	14.2128	513.77	0.	0.
9	14.2128	513.77	0.	0.

** STATOR EXIT - ROTOR INLET 1 **

STREAMLINE NUMBER	RADIAL POSITION (IN)	MASS-FLOW FUNCTION (LBM/SEC)	MERIDIONAL VELOCITY (FPS)	AXIAL VELOCITY (FPS)	WHIRL VELOCITY (FPS)	ABSOLUTE VELOCITY (FPS)	ABSOLUTE MACH NUMBER	ABSOLUTE TOTAL PRESSURE (PSI)	ABSOLUTE TOTAL TEMPERATURE (DEG R)	ABSOLUTE FLOW ANGLE (DEG)
1	10.4990	0.	407.347	407.347	964.326	1046.832	1.03285	14.1392	518.70	67.100
2	11.1497	5.68834	387.806	387.806	918.066	996.614	0.97362	14.1764	518.70	67.100
3	11.7643	11.37676	371.199	371.199	878.752	953.936	0.92461	14.2080	518.70	67.100
4	12.3505	17.06523	356.790	356.790	844.640	916.906	0.88300	14.2352	518.70	67.100
5	12.9136	22.75374	344.089	344.089	814.573	884.266	0.84696	14.2589	518.70	67.100
6	13.4576	28.44229	332.754	332.754	787.739	855.136	0.81528	14.2797	518.70	67.100
7	13.9852	34.13088	322.536	322.536	763.549	828.877	0.78709	14.2983	518.70	67.100
8	14.4987	39.81950	313.248	313.248	741.561	805.007	0.76175	14.3149	518.70	67.100
9	15.0000	45.50814	304.745	304.745	721.433	783.158	0.73878	14.3298	518.70	67.100

STREAMLINE NUMBER	STATIC PRESSURE (PSI)	STATIC TEMPERATURE (DEG R)	STREAMLINE SLOPE ANGLE (DEG)	STREAMLINE CURVATURE (PER IN)	BLADE VELOCITY (FPS)	RELATIVE VELOCITY (FPS)	RELATIVE MACH NUMBER	RELATIVE TOTAL PRESSURE (PSI)	RELATIVE TOTAL TEMPERATURE (DEG R)	RELATIVE FLOW ANGLE (DEG)
1	7.1859	427.51	0.	0.	426.954	674.315	0.66531	9.6697	465.35	52.837
2	7.7213	436.05	0.	0.	453.416	605.222	0.59126	9.7816	466.53	50.151
3	8.1775	442.98	0.	0.	478.408	545.952	0.52917	9.8956	467.78	47.163
4	8.5725	448.74	0.	0.	502.247	494.502	0.47621	10.0121	469.09	43.820
5	8.9189	453.63	0.	0.	525.147	449.627	0.43066	10.1314	470.46	40.068
6	9.2260	457.85	0.	0.	547.267	410.551	0.39141	10.2538	471.88	35.855
7	9.5005	461.53	0.	0.	568.724	376.811	0.35781	10.3795	473.35	31.134
8	9.7477	464.78	0.	0.	589.608	348.158	0.32945	10.5086	474.86	25.877
9	9.9718	467.66	0.	0.	609.992	324.482	0.30610	10.6412	476.42	20.087

** STAGE EXIT 1 **

STREAMLINE NUMBER	RADIAL POSITION (IN)	MASS-FLOW FUNCTION (LBM/SEC)	MERIDIONAL VELOCITY (FPS)	AXIAL VELOCITY (FPS)	WHIRL VELOCITY (FPS)	ABSOLUTE VELOCITY (FPS)	ABSOLUTE MACH NUMBER	ABSOLUTE TOTAL PRESSURE (PSI)	ABSOLUTE TOTAL TEMPERATURE (DEG R)	ABSOLUTE FLOW ANGLE (DEG)
1	10.4990	0.	412.181	412.181	-165.681	444.233	0.44116	7.2697	438.41	-21.898
2	11.1986	5.68874	429.328	429.328	-155.324	456.561	0.45431	7.3390	437.65	-19.889
3	11.8333	11.37748	440.286	440.286	-147.829	464.441	0.46290	7.3883	436.90	-18.560
4	12.4215	17.06622	447.097	447.097	-142.262	469.184	0.46825	7.4231	436.14	-17.651
5	12.9750	22.75496	450.365	450.365	-138.063	471.052	0.47062	7.4432	435.39	-17.043
6	13.5032	28.44371	448.987	448.987	-134.859	468.803	0.46870	7.4424	434.63	-16.718
7	14.0138	34.13246	444.584	444.584	-132.393	463.878	0.46399	7.4276	433.87	-16.583
8	14.5115	39.82122	438.136	438.136	-130.491	457.155	0.45739	7.4037	433.12	-16.585
9	15.0000	45.50997	430.148	430.148	-129.029	449.084	0.44940	7.3735	432.36	-16.697

STREAMLINE NUMBER	STATIC PRESSURE (PSI)	STATIC TEMPERATURE (DEG R)	STREAMLINE SLOPE ANGLE (DEG)	STREAMLINE CURVATURE (PER IN)	BLADE VELOCITY (FPS)	RELATIVE VELOCITY (FPS)	RELATIVE MACH NUMBER	RELATIVE TOTAL PRESSURE (PSI)	RELATIVE TOTAL TEMPERATURE (DEG R)	RELATIVE FLOW ANGLE (DEG)
1	6.3603	421.99	0.	0.	426.954	721.879	0.71688	8.9574	465.35	-55.181
2	6.3702	420.31	0.	0.	455.402	746.531	0.74284	9.1891	466.68	-54.894
3	6.3792	418.95	0.	0.	481.214	767.820	0.76527	9.3997	468.00	-55.011
4	6.3878	417.82	0.	0.	505.134	786.777	0.78521	9.5960	469.33	-55.371
5	6.3956	416.92	0.	0.	527.642	803.736	0.80301	9.7780	470.68	-55.921
6	6.4026	416.34	0.	0.	549.124	818.182	0.81801	9.9371	472.05	-56.718
7	6.4088	415.97	0.	0.	569.886	831.174	0.83137	10.0827	473.46	-57.664
8	6.4142	415.73	0.	0.	590.127	843.358	0.84380	10.2211	474.91	-58.700
9	6.4193	415.58	0.	0.	609.992	855.091	0.85569	10.3564	476.42	-59.798

** STAGE 1 PERFORMANCE **

STREAMLINE NUMBER	STATOR REACTION	ROTOR REACTION	STATOR PRESSURE LOSS COEFFICIENT	ROTOR PRESSURE LOSS COEFFICIENT	STATOR BLADE ROW EFFICIENCY	ROTOR BLADE ROW EFFICIENCY	ROTOR ISENTROPIC EFFICIENCY	STAGE ISENTROPIC EFFICIENCY
1	0.23253	0.93411	0.08007	0.27423	0.95108	0.82618	0.89442	0.84987
2	0.24424	0.81071	0.08049	0.21411	0.94879	0.85969	0.91138	0.86845
3	0.25517	0.71104	0.08092	0.16969	0.94682	0.88609	0.92559	0.88426
4	0.26548	0.62852	0.08138	0.13533	0.94508	0.90753	0.93777	0.89797
5	0.27527	0.55942	0.08186	0.10936	0.94350	0.92435	0.94774	0.90946
6	0.28465	0.50178	0.08237	0.09325	0.94206	0.93515	0.95423	0.91757
7	0.29367	0.45335	0.08289	0.08310	0.94071	0.94216	0.95841	0.92337
8	0.30238	0.41282	0.08344	0.07656	0.93945	0.94680	0.96109	0.92762
9	0.31081	0.37947	0.08402	0.07235	0.93826	0.94991	0.96278	0.93079

* MASS-AVERAGED QUANTITIES *

STATOR BLADE-ROW EFFICIENCY = 0.94389

ROTOR BLADE-ROW EFFICIENCY = 0.91123

STAGE WORK = 19.996 BTU PER LBM
STAGE TOTAL EFFICIENCY = 0.90224
STAGE STATIC EFFICIENCY = 0.75907
STAGE BLADE- TO JET-SPEED RATIO = 0.45605

*** SPOOL PERFORMANCE SUMMARY (MASS-AVERAGED QUANTITIES) ***

SPOOL WORK = 19.996 BTU PER LBM
SPOOL POWER = 1287.50 HP
SPOOL TOTAL- TO TOTAL-PRESSURE RATIO = 1.98632
SPOOL TOTAL- TO STATIC-PRESSURE RATIO = 2.29858
SPOOL TOTAL EFFICIENCY = 0.90224
SPOOL STATIC EFFICIENCY = 0.75907
SPOOL BLADE- TO JET-SPEED RATIO = 0.45605

** PRCGRAM TD - AERCCYNAMIC CALCULATIONS FOR THE CESIGN OF AXIAL TURBINES **

ALTERNATIVE ANGLE DISTRIBUTIONS FCR STAGE OF HIGHEST LOADING

*** GENERAL INPUT CATA ***

NUMBER OF SPOCLS = 1
NUMBER OF SETS OF ANALYSIS VARIABLES = 2
NUMBER OF STREAMLINES = 9

GAS CONSTANT = 53.35COC LBF FT/LBM DEG R
INLET MASS FLOW = 45.51C00 LBM/SEC

* TABULAR INLET SPECIFICATICNS *

RADIAL COORDINATE (IN)	TOTAL TEMPERATURE (DEG R)	TOTAL PRESSURE (PSI)	ABSCLUTE FLOW ANGLE (DEG)
10.0000	518.70	14.656C	C.

*** SPCOL INPUT DATA ***

** DESICN REQUIREMENTS **

ROTATIVE SPEED = 4660.0 RPM
POWER OUTPLT = 1287.50 HP

** SET 1 OF ANALYSIS VARIABLES **

NUMBER CF STAGES = 1

* POWER-CUTPUT SPLIT *

STAGE NUMBER	FRACTICN OF SPOOL PCWER OLTPLT
1	1.CCCCC

* SPECIFIC-HEAT SPECIFICATICN *

CESIGN STATION NUMBER	SPECIFIC HEAT (BTU/LBM DEG R)
1	C.24CCC
2	C.2400C
3	0.24C0C

* ANNULUS SPECIFICATION *

STATICN NUMBER	AXIAL POSITION (IN)	HUB RADIUS (IN)	CASING RADIUS (IN)
1	0.	8.215C	13.5CCO
2	1.0000	8.215C	13.5000
3	2.0000	8.215C	13.5CCC
4	3.0000	8.215C	13.5C00
5	4.00C0	8.2150	13.5000

• BLADE-RCW EXIT CCNDITIONS •

STATOR 1	RADIAL POSITION (IN)	WHIRL ANGLE (DEG)
	13.0000	67.16C

ROTOR 1	STREAMLINE NUMBER	NONDIMENSIONAL POWER OUTPUT FUNCTION
	1	C.
	2	0.1174C
	3	C.23657
	4	C.35871
	5	C.48262
	6	C.60871
	7	C.73657
	8	C.86740
	9	1.CCOCC

• BASIC INTERNAL LOSS CORRELATION •

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       TAN(INLET ANGLE) + TAN(EXIT ANGLE)                 (0.02999959 + 0.15725495 * (V RATIO)** 3.60)   IF (V RATIO) .LT. 0.60000000
Y = ------------------------------------------ *TIMES*
    0.60000000 + 0.80000000 * COS(EXIT ANGLE)             (0.05500000 + 0.15CCCCCC *((V RATIO)-0.600))   IF (V RATIO) .GT. 0.60000000
```

THE PRESSURE-LOSS CCEFFICIENT COMPUTED IN THIS MANNER MAY NOT EXCEED A LIMIT CF 1.0000000C

*** CUTPUT OF SPCCL CESIGN ANALYSIS (SET 1 CF ANALYSIS VARIABLES) ***

** STATOR INLET 1 **

STREAMLINE NUMBER	RADIAL POSITION (IN)	MASS-FLCW FUNCTICN (LBM/SEC)	MERIDIONAL VELOCITY (FPS)	AXIAL VELOCITY (FPS)	WHIRL VELOCITY (FPS)	ABSOLUTE VELOCITY (FPS)	ABSOLUTE MACH NUMBER	ABSOLUTE TOTAL PRESSURE (PSI)	ABSOLUTE TOTAL TEMPERATURE (DEG R)	ABSOLUTE FLOW ANGLE (DEG)
1	8.2150	0.	243.432	243.432	0.	243.432	0.21909	14.6960	518.70	C.
2	9.0461	5.68875	243.432	243.432	C.	243.432	C.21909	14.6960	518.70	0.
3	9.8070	11.37750	243.432	243.432	C.	243.432	C.21909	14.6960	518.70	0.
4	10.5130	17.06625	243.432	243.432	C.	243.432	C.21909	14.6960	518.70	0.
5	11.1744	22.75500	243.432	243.432	0.	243.432	0.21909	14.6960	518.70	0.
6	11.7989	28.44375	243.432	243.432	C.	243.432	C.21909	14.6960	518.70	0.
7	12.3919	34.13250	243.432	243.432	0.	243.432	C.21909	14.6960	518.70	0.
8	12.9578	39.82125	243.432	243.432	C.	243.432	C.21909	14.6960	518.70	0.
9	13.5000	45.51000	243.432	243.432	0.	243.432	0.21909	14.6960	518.70	0.

STREAMLINE NUMBER	STATIC PRESSURE (PSI)	STATIC TEMPERATURE (DEG R)	STREAMLINE SLOPE ANGLE (DEG)	STREAMLINE CURVATURE (PER IN)
1	14.2127	513.77	C.	C.
2	14.2127	513.77	0.	0.
3	14.2127	513.77	0.	0.
4	14.2127	513.77	0.	C.
5	14.2127	513.77	0.	C.
6	14.2127	513.77	0.	0.
7	14.2127	513.77	0.	0.
8	14.2127	513.77	0.	0.
9	14.2127	513.77	C.	C.

** STATOR EXIT - ROTOR INLET 1 **

STREAMLINE NUMBER	RADIAL POSITION (IN)	MASS-FLOW FUNCTION (LBM/SEC)	MERIDIONAL VELOCITY (FPS)	AXIAL VELOCITY (FPS)	WHIRL VELOCITY (FPS)	ABSOLUTE VELOCITY (FPS)	ABSOLUTE MACH NUMBER	ABSOLUTE TOTAL PRESSURE (PSI)	ABSOLUTE TOTAL TEMPERATURE (DEG R)	ABSOLUTE FLOW ANGLE (DEG)
1	8.2150	0.	467.445	467.445	1109.837	1204.260	1.23140	14.0222	518.70	67.160
2	9.0576	5.68807	431.323	431.323	1024.075	1111.202	1.11148	14.0896	518.70	67.160
3	9.8114	11.37625	403.947	403.947	959.076	1040.673	1.02549	14.1418	518.70	67.160
4	10.5064	17.06451	381.961	381.961	906.875	984.031	0.95905	14.1839	518.70	67.160
5	11.1588	22.75284	363.642	363.642	863.383	936.838	0.90530	14.2188	518.70	67.160
6	11.7787	28.44121	347.984	347.984	826.206	896.499	0.86040	14.2484	518.70	67.160
7	12.3726	34.12961	334.345	334.345	793.822	861.360	0.82201	14.2738	518.70	67.160
8	12.9453	39.81805	322.289	322.289	765.199	830.301	0.78861	14.2958	518.70	67.160
9	13.5000	45.50652	311.508	311.508	739.603	802.527	0.75913	14.3152	518.70	67.160

STREAMLINE NUMBER	STATIC PRESSURE (PSI)	STATIC TEMPERATURE (DEG R)	STREAMLINE SLOPE ANGLE (DEG)	STREAMLINE CURVATURE (PER IN)	BLADE VELOCITY (FPS)	RELATIVE VELOCITY (FPS)	RELATIVE MACH NUMBER	RELATIVE TOTAL PRESSURE (PSI)	RELATIVE TOTAL TEMPERATURE (DEG R)	RELATIVE FLOW ANGLE (DEG)
1	5.5490	398.02	0.	0.	334.072	905.713	0.92612	9.6572	466.28	58.929
2	6.5055	415.95	0.	0.	368.338	784.876	0.78508	9.7716	467.21	56.664
3	7.2504	428.58	0.	0.	398.991	690.557	0.68048	9.8851	468.26	54.200
4	7.8548	438.12	0.	0.	427.253	613.133	0.59757	9.9996	469.41	51.467
5	8.3590	445.67	0.	0.	453.785	547.727	0.52929	10.1161	470.63	48.401
6	8.7883	451.82	0.	0.	478.994	491.578	0.47178	10.2354	471.93	44.936
7	9.1596	456.96	0.	0.	503.147	443.033	0.42279	10.3578	473.29	41.003
8	9.4847	461.33	0.	0.	526.434	401.097	0.38096	10.4836	474.72	36.533
9	9.7723	465.11	0.	0.	548.993	365.198	0.34545	10.6132	476.21	31.462

** STAGE EXIT 1 **

STREAMLINE NUMBER	RADIAL POSITION (IN)	MASS-FLOW FUNCTION (LBM/SEC)	MERIDIONAL VELOCITY (FPS)	AXIAL VELOCITY (FPS)	WHIRL VELOCITY (FPS)	ABSOLUTE VELOCITY (FPS)	ABSOLUTE MACH NUMBER	ABSOLUTE TOTAL PRESSURE (PSI)	ABSOLUTE TOTAL TEMPERATURE (DEG R)	ABSOLUTE FLOW ANGLE (DEG)
1	8.2150	0.	448.788	448.788	-284.519	531.377	0.53041	7.0432	441.18	-32.374
2	9.0778	5.68878	472.361	472.361	-263.602	540.935	0.54145	7.1705	439.73	-29.164
3	9.8306	11.37755	481.525	481.525	-251.529	543.261	0.54486	7.2385	438.28	-27.581
4	10.5162	17.06630	482.780	482.780	-244.246	541.048	0.54346	7.2734	436.83	-26.836
5	11.1562	22.75504	479.380	479.380	-239.879	536.047	0.53909	7.2867	435.39	-26.583
6	11.7638	28.44378	472.427	472.427	-237.404	528.723	0.53224	7.2828	433.94	-26.680
7	12.3503	34.13253	459.733	459.733	-236.162	516.843	0.52055	7.2515	432.49	-27.189
8	12.9262	39.82128	442.268	442.268	-235.692	501.150	0.50482	7.1995	431.04	-28.054
9	13.5000	45.51004	420.691	420.691	-235.675	482.207	0.48567	7.1330	429.59	-29.258

STREAMLINE NUMBER	STATIC PRESSURE (PSI)	STATIC TEMPERATURE (DEG R)	STREAMLINE SLOPE ANGLE (DEG)	STREAMLINE CURVATURE (PER IN)	BLADE VELOCITY (FPS)	RELATIVE VELOCITY (FPS)	RELATIVE MACH NUMBER	RELATIVE TOTAL PRESSURE (PSI)	RELATIVE TOTAL TEMPERATURE (DEG R)	RELATIVE FLOW ANGLE (DEG)
1	5.8152	417.68	0.	0.	334.072	764.242	0.76285	8.5489	466.28	-54.039
2	5.8742	415.38	0.	0.	369.159	789.627	0.79037	8.8692	467.26	-53.258
3	5.9154	413.72	0.	0.	399.774	809.976	0.81236	9.1289	468.31	-53.524
4	5.9499	412.47	0.	0.	427.653	827.260	0.83105	9.3577	469.43	-54.302
5	5.9794	411.47	0.	0.	453.678	843.105	0.84790	9.5688	470.62	-55.348
6	6.0053	410.68	0.	0.	478.388	857.640	0.86335	9.7666	471.88	-56.575
7	6.0285	410.26	0.	0.	502.241	869.824	0.87606	9.9370	473.22	-58.093
8	6.0498	410.14	0.	0.	525.659	880.487	0.88693	10.0885	474.65	-59.848
9	6.0703	410.24	0.	0.	548.993	890.328	0.89673	10.2302	476.21	-61.803

** STAGE 1 PERFORMANCE **

STREAMLINE NUMBER	STATCR REACTION	RCTCR REACTICN	STATCR PRESSURE LOSS COEFFICIENT	ROTCR PRESSURF LOSS COEFFICIENT	STATCR BLACE ROW EFFICIENCY	ROTCR BLADE RCW EFFICIENCY	RCTCR ISENTRCPIC EFFICIENCY	STACE ISENTRCPIC EFFICIENCY
1	0.20214	1.18511	C.07952	C.40543	C.95792	0.77273	0.83699	0.78867
2	0.21907	C.99398	0.C7996	0.30253	C.95380	C.82007	0.86760	0.82142
3	C.23392	0.85256	0.C8C42	0.23654	0.95C63	C.85384	C.89039	0.84654
4	0.24738	C.74107	C.08091	C.189C0	C.948C3	C.87998	0.90868	C.867C9
5	0.25984	0.64965	0.08143	C.15233	C.94583	C.90126	0.92396	0.88451
6	0.27154	0.57318	0.08198	0.12365	C.94390	C.91863	0.9367C	C.89926
7	0.28261	0.50934	0.08256	0.10616	0.94217	C.92962	0.94493	0.90959
8	0.29319	C.45554	0.08318	C.C9653	C.94C58	C.9359C	0.94969	C.91644
9	0.30333	C.41018	0.08382	C.C9209	C.93911	0.93899	0.95205	C.92C80

* MASS-AVERAGED QUANTITIES *

STATCR BLACE-ROW EFFICIENCY = 0.94668

ROTOR BLACE-ROW EFFICIENCY = C.88650

STAGE WORK = 19.996 BTL PER LBP
STAGE TOTAL EFFICIENCY = 0.87477
STAGE STATIC EFFICIENCY = 0.70772
STAGE BLADE- TO JET-SPEED RATIO = 0.37815

*** SPCCL PERFORMANCE SUMMARY (MASS-AVERAGED QUANTITIES) ***

SPCCL WORK = 19.996 BTL PER LBP
SPOOL POWER = 1287.50 HP
SPOOL TCTAL- TO TCTAL-PRESSURE RATIO = 2.03436
SPOOL TCTAL- TO STATIC-PRESSURE RATIO = 2.4624C
SPOOL TOTAL EFFICIENCY = C.87477
SPCCL STATIC EFFICIENCY = C.70772
SPCOL BLADE- TO JET-SPEEC RATIO = 0.37815