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TECHNICAL NOTE 2068

THEORETICAL EFFECT OF INLET HUB-TIP-RADIUS RATIO AND DESIGN SPECIFIC MASS FLOW ON DESIGN PERFORMANCE OF AXIAL-FLOW COMPRESSORS

By Chung-Hua Wu, John T. Sinnette, Jr.
and Robert E. Forrette

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THEORETICAL EFFECT OF INLET HUB-TIP-RADIUS RATIO AND DESIGN SPECIFIC MASS FLOW ON DESIGN PERFORMANCE OF AXIAL-FLOW COMPRESSORS

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SUMMARY

An analysis was made of the effect at design condition of inlet hub-tip-radius ratio and mass flow per unit frontal area on the allowable rotor-tip speed, pressure ratio, and power input per unit frontal area of multistage axial-flow compressors employing approximately symmetrical velocity diagrams at all radii, constant total enthalpy along the radius, a maximum Mach number of 0.8 for all stages, and a prescribed empirical turning limitation. The analysis covers inlet hub-tip-radius ratios of 0.6, 0.5, and 0.4, the practical range of design specific mass flows for each, and the number of stages sufficient to give an exit hub-tip ratio of approximately 0.9.

The analysis indicated that for any given inlet hub-tip-radius ratio the design rotor speed and the pressure ratio per stage decrease with increasing design mass flow per unit frontal area. The maximum pressure ratio and rotor speed, determined by the minimum allowable value of the axial velocity behind the first-stage rotor, decrease with decreasing inlet hub-tip-radius ratio. The power input per stage per unit frontal area was fairly constant over the practical range of design mass flow per unit frontal area and varied only slightly with inlet hub-tip-radius ratio.

When limited by an exit hub-tip-radius ratio of 0.9 and a constant maximum Mach number of 0.8 for all stages, the over-all pressure ratio, power input per unit frontal area, and mass flow per unit frontal area increased with decreasing inlet hub-tip-radius ratio over most of the range. For seven stages and the same limitations, the maximum values of specific mass flow, total-pressure ratio, and specific power input were reached at an inlet hub-tip ratio of about 0.4 and were 31.3 pounds per second per square foot, 5.9, and 4180 horsepower per square foot, respectively.

INTRODUCTION

Little information is available in current literature on the variation of the design (maximum allowable) rotor-tip speed, pressure ratio, and power input per unit frontal area of multistage axial-flow compressors with the design mass flow per unit frontal area and the inlet hub-tip radius ratio. A theoretical investigation was conducted at the NACA Lewis laboratory in order to obtain these relations for compressors designed with: (1) constant work input to the rotor along the radius; (2) velocity distributions giving approximately symmetrical velocity diagrams at all radii; (3) a rotor speed that gives a maximum Mach number of 0.8 in the first stage; (4) a variable hub radius to give a maximum inlet Mach number of 0.8 for all succeeding stages; and (5) a turning limitation based on Howell's work (reference 1) with a blade solidity of 1.5 at the hub for all stages. The compressibility effect was considered, but the effect of the curvature of streamlines, caused by radial flow, was neglected in the computation of radial variation of the state of the air. Calculations were made for inlet hub-tip-radius ratios of 0.6, 0.5, and 0.4 over the practical range of design mass flow for each ratio. The number of stages used for each design was limited to that giving an exit hub-tip-radius ratio of approximately 0.9.

All values given in this report are design values based on the chosen maximum allowable Mach number and turning of the air. The values of rotor-tip speed, inlet-air velocity, mass flow, and power input are equivalent values; that is, values corrected to NACA standard sea-level inlet conditions. Equivalent mass flow and equivalent power input per unit frontal area are hereinafter referred to as "specific mass flow" and "specific power input," respectively. The hub-tip-radius ratio is hereinafter referred to as the "hub-tip ratio."

Although the examples considered are for specific Mach number and turning limitations, the trend of the design performance with varying inlet hub-tip ratio and specific mass flow should be the same for any similar specified limitation on Mach number and turning.

SYMBOLS

The following symbols are used in this report:

A_a annulus area

A_f frontal area

a velocity of sound

H total enthalpy

h static enthalpy

M relative Mach number at inlet to blade

P total pressure

p static pressure

$\bar{p}$ average static pressure

r radial coordinate

U rotor speed

V absolute velocity of air

W velocity of air relative to rotor blades

w mass flow per second

z axial coordinate

δ ratio of inlet-air total pressure to NACA standard sea-level pressure

θ ratio of inlet-air total temperature to NACA standard sea-level temperature

σ blade element solidity

ω angular velocity of rotor

Subscripts: (See fig. 1 for station notation and coordinate system.)

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1 station upstream of first rotor

2 station upstream of first stator and so forth

e exit station downstream of last stator

h hub

i inlet station upstream of inlet guide vane

r radial component

t tip

z axial component

θ tangential component

DESIGN CRITERIONS

The following design criterions are adopted for this analysis:

Velocity diagram. - The velocity diagram chosen is one that gives a constant work input to the rotor along the radius and an approximately symmetrical velocity diagram at all radii. The deviation of the velocity diagram from exact symmetry is due to the change in axial velocity across the blade and to the radial displacement of streamlines (reference 2). Previous calculations (references 2 and 3) show that this type of velocity diagram gives significantly higher values of specific mass flow, pressure rise, and design rotor-tip speed than does the usual vortex type for the same design limitations.

Maximum Mach number. - The maximum Mach number of the air flow relative to the blade was chosen to be 0.8 for all stages. This value gives a high pressure rise through all stages and, according to unpublished data obtained at the NACA Langley and Lewis laboratories, should not result in an appreciable loss in efficiency.

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Maximum turning. - Information regarding the amount of turning the air can efficiently undergo for various design conditions is relatively unavailable. The formula suggested by Howell in reference 1 is on the conservative side and is convenient to use. Because the turning is largest at the hub for the present type of design, and the axial velocity at the hub at the entrance to the rotor is always lower than that at the entrance to the stator, it was assumed that, in general, the critical condition exists at the hub of the rotor, and the turning limitation was applied there. Because of the change in axial velocity across the blade, Howell's formula was modified to the following form (which reduces to his formula when the axial velocity V_z is constant):

$$V_{\theta,2} - V_{\theta,1} = \frac{1.55}{1 + \frac{1.5}{\sigma}} V_{z,1} \tag{1}$$

Solidity. - The solidity at the hub for all stages was chosen to be 1.5. The efficiency tends to drop if the solidity is too high, but unpublished data obtained at the NACA Langley laboratory indicate that high efficiency can be obtained with a solidity of 1.5.

Tip radius. - The tip radius was held constant for all stages in order to keep the over-all diameter of the compressor the same as that of the inlet stage.

Hub radius. - The hub radius was kept constant across the inlet guide vanes and first rotor row, and at subsequent stations was increased along the axis in such a manner as to give a maximum Mach number of 0.8 for all stages. The percentage rise in hub radius was the same across a stator row and the following rotor row.

Polytropic efficiency. - A polytropic efficiency of 0.9 was assumed at the mean radius.

Mass-flow range. - For convenience, the parameter used to vary the design mass flow was taken as the ratio $V_{z,h}/U_t$ at the inlet to the first rotor. For each hub-tip ratio, this velocity ratio was varied in increments of 0.1; the minimum value was chosen so as to avoid excessively low axial velocities at the tip following the first rotor row, and the maximum value was extended to 1.0, at which point the pressure ratio had dropped excessively and the use of a lower hub-tip ratio would be preferable if a higher specific mass flow were required.

Number of stages. - The number of stages used for each case was limited to that giving the exit hub-tip ratio of the last rotor closest to 0.9. It was then specified that the flow be turned in the axial direction through one or two stators, 1241 depending on whether the turning in a single row was below or above the turning limitation.

METHOD OF CALCULATION

For this investigation, it was desirable to cover a wide range of design specific mass flows and inlet hub-tip ratios. Only the over-all performance parameters such as design rotor-tip speed, pressure ratio, and power input were considered; the details of the flow in the compressor required for accurate blade design were not considered. A method of computation called the simplified radial-equilibrium calculation (reference 2) was therefore used. The effect of streamline curvature caused by the radial motion of the air was neglected in the computation of the radial variation of the state of the air. In determining the tangential component of air velocity downstream of a rotor for constant work input along the radius, the air particle was assumed to take the same relative position as upstream of the rotor.

The calculations then involved a station-to-station computation, starting from the station upstream of the first rotor, with all quantities expressed in dimensionless ratios. For each inlet hub-tip ratio, calculations were made for three to five selected values of the velocity ratio $V_{z,1,h}/U_t$ ranging from 0.6 to 1.0. In order to avoid a very low axial velocity entering at the tip of the stator, the minimum allowable velocity ratio for this type of design is approximately 0.6.

For each inlet hub-tip ratio and velocity ratio $V_{z,1,h}/U_t$ used, the radial distributions of $V_{z,1}/U_t$ and $V_{\theta,1}/U_t$ were first determined by equations (D13) and (E6), respectively, of reference 2. In this type of design, with the simplified method of computation, the maximum Mach number in the stage occurs at the hub at the entrance to the stator blade; that is, at station 2. A tentative relative Mach number therefore had to be assumed entering at the hub of the preceding rotor blade; that is, at station 1. The distribution of tangential velocity at station 2 was then determined by equation (E6) of reference 2, the total enthalpy at station 2 by equation (18a) of reference 2, the axial velocity at the hub at station 2 by the specified value of $M_{2,h}$

equal to 0.8, and the distribution of axial velocity at station 2 by equation (E8) of reference 2. The total mass flow at station 2 was then integrated and compared with that at station 1. This computation was repeated with various Mach numbers at station 1 until the integrated mass flows at stations 1 and 2 agreed within 1 percent. With the correct value of the Mach number at station 1 thus determined, the total enthalpy at the inlet, the design rotor-tip speed, and the inlet mass flow were obtained in terms of standard inlet temperature and pressure.

The computations for the succeeding stages were similar to that for the first stage except that the variable in the computation was the increase in hub radius across the blade rather than the maximum Mach number relative to the rotor blade. For simplicity, it was assumed that the percentage increase in hub radius was the same for the stator- and the following rotor-blade rows. The correct value of this increase in hub radius was the one that gave the maximum Mach number of 0.8 within 1 percent at the entrance to the stator-blade row. Because of the hub rise, a change in tangential velocity smaller than that given by equation (1) was assigned at the hub of the rotor blades. The value of the change in tangential velocity was obtained by the condition that the change in total enthalpy was the same as in the case of no hub rise; that is,

$$\frac{r_{4,h} V_{\theta,4,h} - r_{3,h} V_{\theta,3,h}}{r_{3,h} V_{z,3,h}} = \frac{1.55}{1 + \frac{1.5}{\sigma}}$$

or

$$\frac{r_{4,h}}{r_{3,h}} V_{\theta,4,h} - V_{\theta,3,h} = \frac{1.55}{1 + \frac{1.5}{\sigma}} V_{z,3,h} \tag{2}$$

This computation was repeated in succeeding stages until the final hub-tip ratio was about 0.9, which is considered to be the maximum practical value.

Calculations were also made for the inlet and exit guide vanes, using a constant hub diameter through each.

The details of the method of calculating the flow and the derivation of the equations involved are given in reference 2.

RESULTS OF CALCULATIONS AND DISCUSSION

Results of the calculations are presented in terms of equivalent values; that is, the values correspond to NACA standard sea-level inlet conditions. The calculations cover a range of mass flows for inlet hub-tip ratios of 0.6, 0.5, and 0.4. The ratio of 0.4 is lower than the value generally used, but a decrease in hub-tip ratio from 0.5 to 0.4 results in a gain of about 12 percent in specific mass flow (mass flow per unit frontal area, fig. 2) corresponding to the increase in annulus area. The mass flow per unit annulus area for the range considered (fig. 2) varies from 69 percent to 87 percent of the theoretical maximum value of 49.4 pounds per second per square foot.

The variation of rotor-tip speed with specific mass flow for the three inlet hub-tip ratios is shown in figure 3. For each inlet hub-tip ratio, the design speed decreases with increasing specific mass flow. The decrease occurs because, at higher mass flows, the ratio of air velocity to rotor-tip speed is higher and therefore, for the same Mach number limitation, the rotor-tip speed has to be lower. The lower rotor-tip speed will, of course, give a lower pressure rise in all stages. Figure 3 also shows that at a given rotor speed, the specific mass flow increases with decreasing inlet hub-tip ratio. The cross line at the top of the curves indicates the lower limit of design specific mass flow below which an excessively low value of the axial velocity at the tip behind the first rotor would occur. Because of this limit, the maximum value of design rotor-tip speed decreases from 1040 to 960 feet per second when the inlet hub-tip ratio decreases from 0.6 to 0.4.

The variation of hub-tip ratios at various stations with the specific mass flow for the three inlet hub-tip ratios is shown in figure 4. In general, for a given inlet hub-tip ratio, the required increase in hub radius per stage is less for higher mass flow; hence, additional stages may be added as the mass flow is increased before the assigned limiting value of the exit hub-tip ratio of 0.9 is reached. Also, the lower the inlet hub-tip ratio, the greater is the number of stages permitted.

The ratios of total and static pressures to the inlet total pressure, averaged over the annulus area after the successive blade rows, are given in figures 5 and 6, respectively. The curves of each figure are very similar in shape, but the value for the static-pressure ratio is lower than that for the total-pressure ratio at the same station. For any given number of stages in which the limiting hub-tip ratio is not reached, the highest pressure ratio is obtained with the highest inlet hub-tip ratio and the lowest specific mass flow. The increase in pressure ratio with a decrease in specific mass flow is due to the increase in the allowable rotor-tip speed U_t with the decrease in specific mass flow. The increase in the maximum pressure ratio with increasing inlet hub-tip ratio is due to the increase in rotor-hub speed U_h. The increase in rotor-hub speed results from an increase in the maximum allowable rotor-tip speed (fig. 3) and from the higher value of U_h/U_t due to the higher hub-tip ratio.

The maximum over-all pressure ratio, as limited by the exit hub-tip ratio of 0.9, is shown by the crossed lines on the curves of figure 5. For a given inlet hub-tip ratio, the maximum allowable over-all pressure ratio can usually be increased by increasing the number of stages, provided the specific mass flow is also increased by a suitable amount to prevent exceeding the allowable exit hub-tip ratio. When the inlet hub-tip ratio varies, the relation between the maximum over-all pressure ratio and the specific mass flow follows the same trend as when the inlet hub-tip ratio is constant, but the practical range of specific mass flow is extended.

The design criterions chosen give high over-all pressure ratios with a small number of stages, which not only reduces the axial length, the weight, and the cost of the machine, but also shortens the time required in design, development, and manufacture.

The variation of specific power input to the rotors with specific mass flow and inlet hub-tip ratio is shown in figure 7. At a given inlet hub-tip ratio, the decreasing total-pressure rise with increasing specific mass flow (fig. 5) gives a maximum practical value of specific power input to the whole compressor a little higher than that at the lowest allowable specific mass flow (fig. 3). For a given number of stages, the specific power input is about the same for the three different inlet hub-tip ratios - a result of the smaller total-pressure rise connected with higher specific mass flow.

A comparison of the characteristics of seven compressors with an exit hub-tip ratio of 0.9, obtained from figures 2 to 7, is presented in the following table:

Compressor	Inlet hub-tip ratio	Specific mass flow (lb/(sec)(sq ft))	Rotor-tip speed (ft/sec)	Specific power input (hp/sq ft)	Number of stages	Over-all total-pressure ratio	Average total-pressure ratio per stage
1	0.6	23.2	970	2460	5	4.44	1.35
2	.6	26.2	800	3015	6	4.87	1.30
3	.6	28.1	655	3300	7	5.02	1.25
4	.5	27.3	932	3290	6	5.24	1.32
5	.5	30.5	784	3940	7	5.67	1.28
6	.4	31.3	886	4180	7	5.92	1.29
7	.4	33.9	768	4710	8	6.31	1.26

For a given number of stages, the rotor-tip speed, the specific mass flow, the specific power input, and the over-all pressure ratio increase with decreasing inlet hub-tip ratio over most of the range (fig. 8). For seven stages, the specific mass flow, the specific power input, and the over-all pressure ratio seem to have reached their maximum values at an inlet hub-tip ratio of about 0.4. For eight stages, the optimum condition probably occurs at a still lower hub-tip ratio. For a given inlet hub-tip ratio, the rotor-tip speed decreases with an increasing number of stages; whereas the specific mass flow, the over-all pressure ratio, and the specific power input increase.

In addition to determining the trend of the design performance of multistage axial-flow compressors, the results obtained may also be used to give preliminary design information. For a given pressure ratio, several possible designs may be obtained from figures 5 and 8, each involving a set of inlet and exit hub-tip ratios, design rotor-tip speed, design mass flow per unit frontal area, and power input per unit frontal area. The choice will depend upon whether the emphasis is on the smallest frontal area or on the shortest axial length of the compressor. If a higher over-all pressure ratio than those considered is desired in a single unit, the maximum Mach number may be reduced in the later stages to prevent excessive hub-tip ratios, thereby using more stages with a resultant reduction in pressure ratio per stage for these stages.

In order to illustrate the flow characteristics of the type of design employed in this investigation, the hub shape of a typical multistage compressor with corresponding velocity diagrams at the hub, the mean radius, and the tip, and the variations of velocities, Mach numbers, static pressures, and static enthalpies

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throughout the compressor, as determined by the simplified method, are shown in figures 9, 10, and 11 and briefly discussed in the appendix. For an actual detailed design, it is recommended that more accurate methods of flow calculation be used.

SUMMARY OF RESULTS

From an analysis based on a design employing symmetrical velocity diagrams, constant total enthalpy along the radius, and a set of chosen design values, the following results were obtained regarding the design performance of axial-flow multistage compressors:

1. For a given inlet hub-tip ratio, the design rotor-tip speed decreased rather rapidly with increasing design mass flow per unit frontal area. The maximum value of design rotor-tip speed slightly decreased with decreasing inlet hub-tip ratio. At a given rotor-tip speed, the mass flow per unit frontal area was greater at the lower inlet hub-tip ratio.

2. At a given inlet hub-tip ratio, the pressure ratio per stage decreased with increasing design mass flow per unit frontal area. The maximum value of pressure ratio per stage decreased with decreasing inlet hub-tip ratio.

3. The power input per stage per unit frontal area was fairly constant over most of the range of mass flow considered and dropped off at the maximum mass flow. The maximum value per stage for different inlet hub-tip ratios was nearly the same.

4. With the limitations of an exit hub-tip ratio of 0.9 and a constant maximum Mach number of 0.8 for all stages:

(a) The number of stages that can be used increased with increasing design mass flow per unit frontal area and decreasing inlet hub-tip ratio.

(b) For a given number of stages, the design mass flow per unit frontal area, the rotor-tip speed, the over-all pressure ratio, and the power input per unit frontal area increased with decreasing inlet hub-tip ratio over most of the range. For seven stages, maximum mass flow, power input, and pressure ratio were reached at a hub-tip ratio of about 0.4 and were 31.3 pounds per second per square foot, 4180 horsepower per square foot, and 5.9, respectively.

(c) For a given inlet hub-tip ratio, as the mass flow per unit frontal area increased, the rotor-tip speed decreased but the over-all pressure ratio increased because the number of stages was increased. Inasmuch as the over-all pressure ratio and the mass flow were both increased, the power input per unit frontal area was also increased.

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All these results were obtained on the basis of maximum power input per stage corresponding to a constant maximum Mach number of 0.8 for all stages. Higher over-all pressure ratios may be obtained by reducing the Mach number in the later stages, thereby using more stages with a resultant reduction in pressure ratio per stage for these stages.

Lewis Flight Propulsion Laboratory,
National Advisory Committee for Aeronautics,
Cleveland, Ohio, August 17, 1949.

APPENDIX

FLOW CONDITIONS THROUGH TYPICAL COMPRESSOR DESIGN WITH INLET HUB-TIP RATIO OF 0.5 AND MASS FLOW OF 30.1 POUNDS PER SECOND PER SQUARE FOOT OF FRONTAL AREA

The hub shape of a typical multistage compressor with inlet hub-tip ratio of 0.5 and specific mass flow of 30.1 pounds per second per square foot is shown in figure 9. The figure was drawn for a constant Reynolds number, based on the axial length of the blade, of 8×10^5 at sea level or 2×10^5 at an altitude of 35,000 feet. In the drawing, the tip radius is taken as 1 foot. The dashed line gives the hub shape directly obtained from the calculation and shows sharp breaks in the slope. A suggested fairing for practical design is shown by the solid curve. For an actual design, of course, it would be desirable to repeat the flow calculations with the modified contour.

The velocity diagrams at the hub, the mean radius, and the tip of the typical multistage compressor are shown in figure 10. They are not exactly symmetrical in the usual sense because of the change in axial velocity and the radial displacements of the streamlines across the blade. The design criterion adopted for maximum turning at the hub of the rotor gives reasonable flow conditions at all stations except at the tip of the first rotor, where a large reduction in relative velocity occurs because of the large reduction in axial velocity. The relative inlet Mach number, however, is low at the tip and the more accurate calculation, which includes the effect of the curvature caused by the radial motion, tends to reduce this change in axial velocity (reference 2) and consequently improves the condition at that point. For a low inlet hub-tip ratio and for a low mass flow, it is recommended that the flow over the tip of the first rotor be taken into account more carefully by reducing the turning over the first rotor or by adding more work toward the tip than at the hub, thereby increasing the axial velocity at the exit of the rotor tip.

In the design criterions chosen, the maximum Mach number occurring at the entrance to the station at the hub is limited to 0.8 for all stages, and a constant percentage increase in hub radius is assumed for a stator and the following rotor. This criterion gives the relative Mach number at the inlet to the blade (fig. 11(a)). The maximum relative Mach number at the inlet to the rotors could be increased to 0.8 if a greater hub increase were taken through the stator row, which is not done in the present calculations because of the additional time required and because the additional sharp breaks in the hub shape would be undesirable.

The velocities relative to the stators are shown in figure 11(b); the velocities relative to the rotor, in figure 11(c); the axial component of the velocities, in figure 11(d); and the absolute tangential component of velocities, in figure 11(e).

For both absolute and relative velocities, the variation with radius (fig. 11(b) and 11(c)) is larger at stations following the rotor than at stations following the stator. This variation is the result of large variation in axial velocity at stations following the rotor (fig. 11(d)), which is characteristic of this type of design. A more accurate calculation that includes the effect of streamline curvature caused by the radial motion would decrease this variation in stations following the rotor and increase the variation in stations following the stator (reference 2).

The tangential component of absolute velocities (fig. 11(e)) generally increases with radius, and the sum of the velocities entering and leaving a rotor approximately equals the rotor speed. The deviation is caused by the radial displacement of the streamlines.

The absolute velocity entering the stators is increased in successive stages in order to maintain the constant Mach number at the hub with the increasing air temperature in successive stages. This increase is achieved primarily through the increase in axial components. The variation of absolute velocity entering successive rotors is determined by the specified taper of the hub across the stages and increases in successive stages except from station 1 to 3, because there is no taper across the first rotor. This progressively higher axial velocity entering the rotors gives progressively larger changes in tangential velocities across successive rotors, as calculated from equation (1).

The static-pressure ratio across successive stages along the assumed streamlines is plotted against the entering radius in figure 11(f). The pressure ratios are very nearly constant with respect to the radius.

The increase in static enthalpy throughout the compressor is shown in figure 11(g); the variation with radius is small. The unit used here for static enthalpy is square feet per second per second; in order to express this unit in Btu per pound, the value should be divided by 778 x 32.2.

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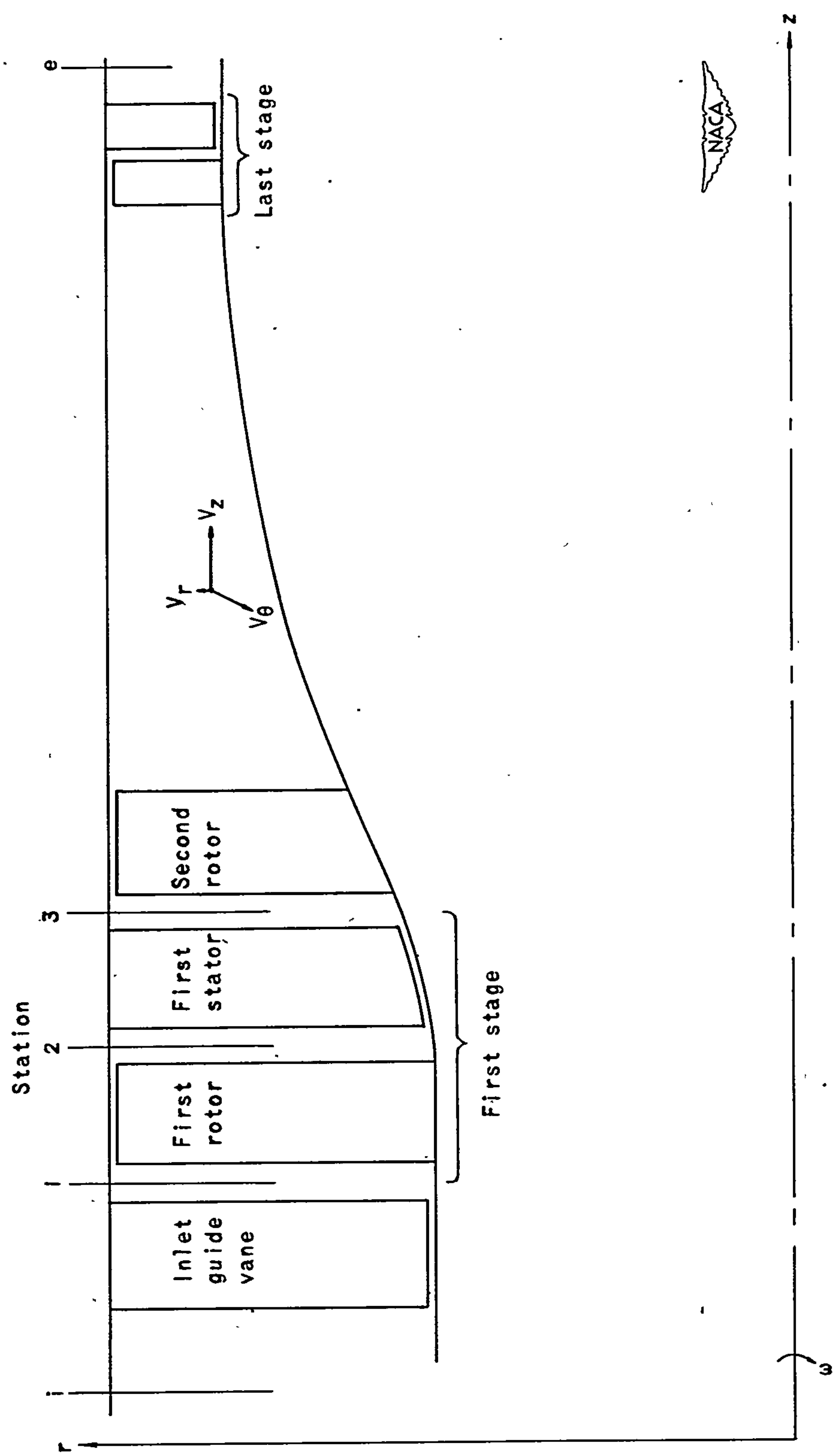

Figure 1. - Coordinate system and station notation.

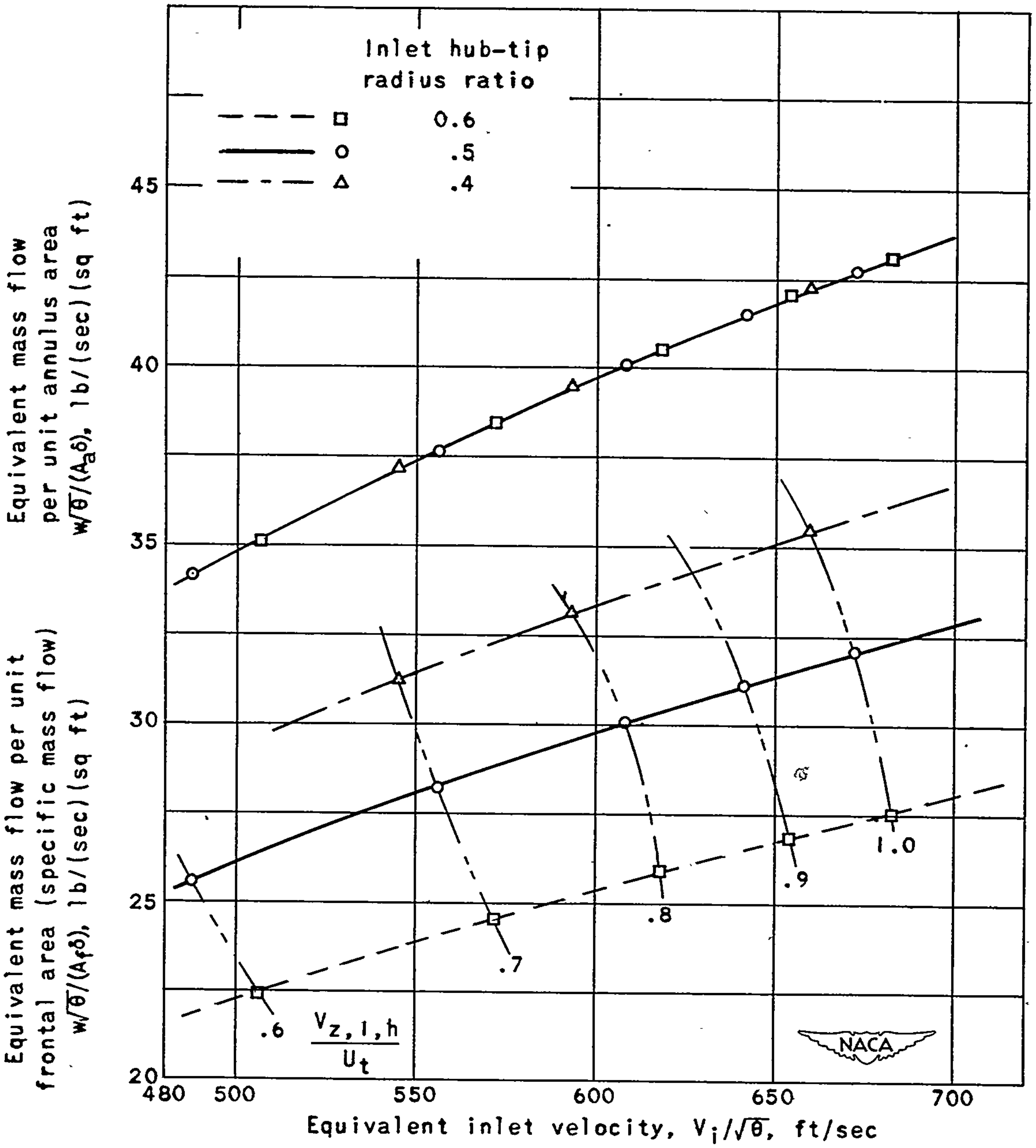

Figure 2. - Variation of equivalent specific mass flows with inlet hub-tip radius ratio and equivalent inlet velocity.

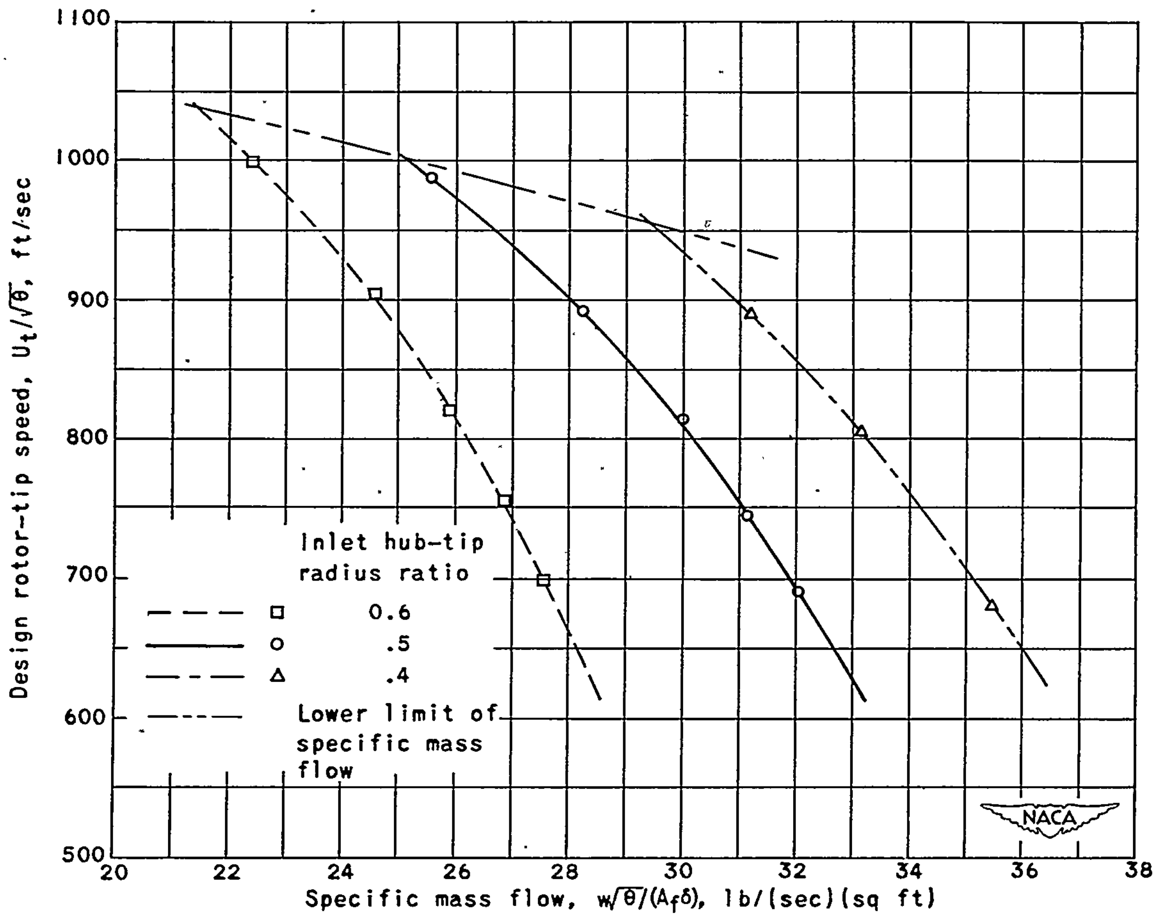

Figure 3. - Variation of design rotor-tip speed with specific mass flow.

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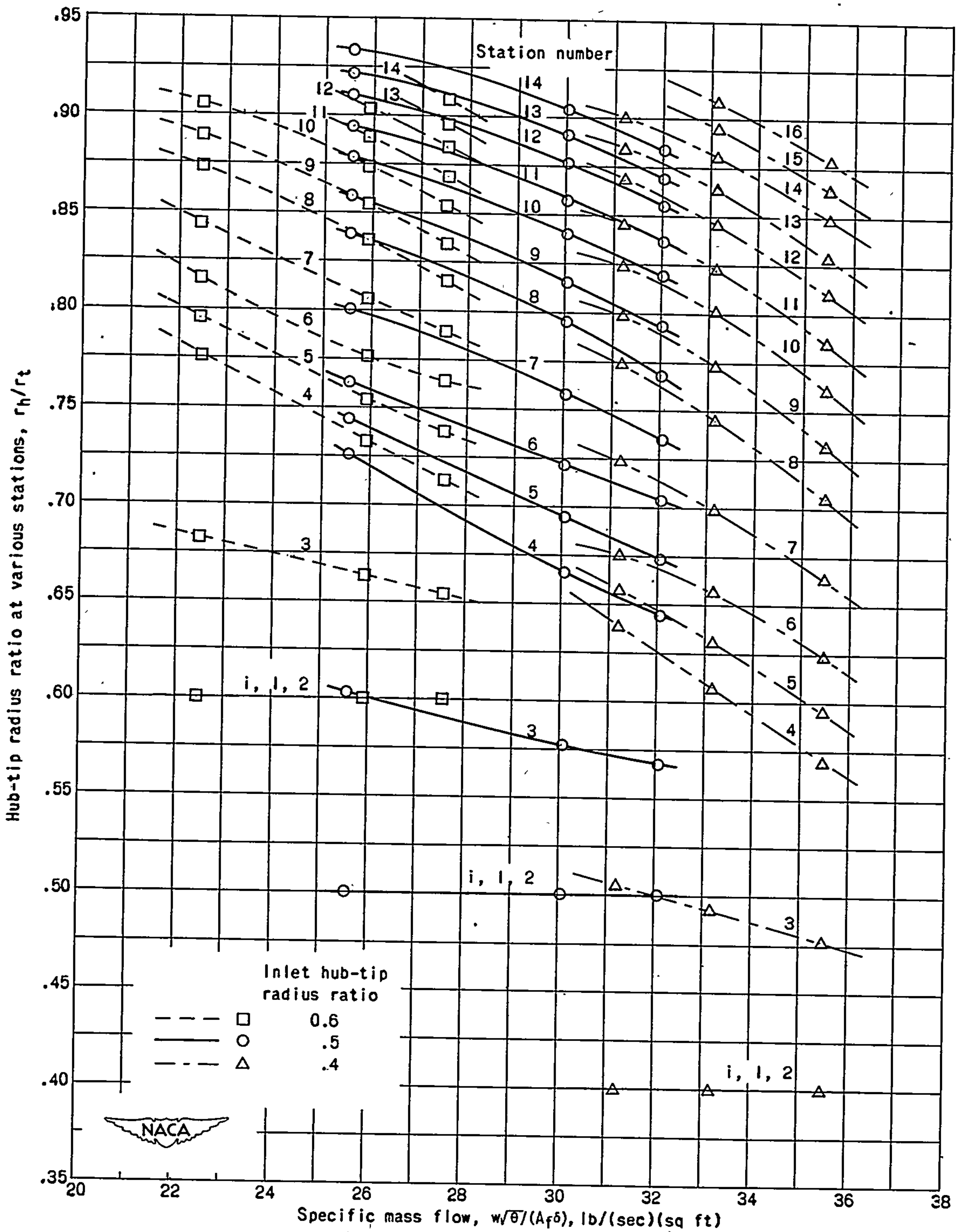

Figure 4. - Hub-tip radius ratios at various stations for various inlet hub-tip ratios and specific mass flows.

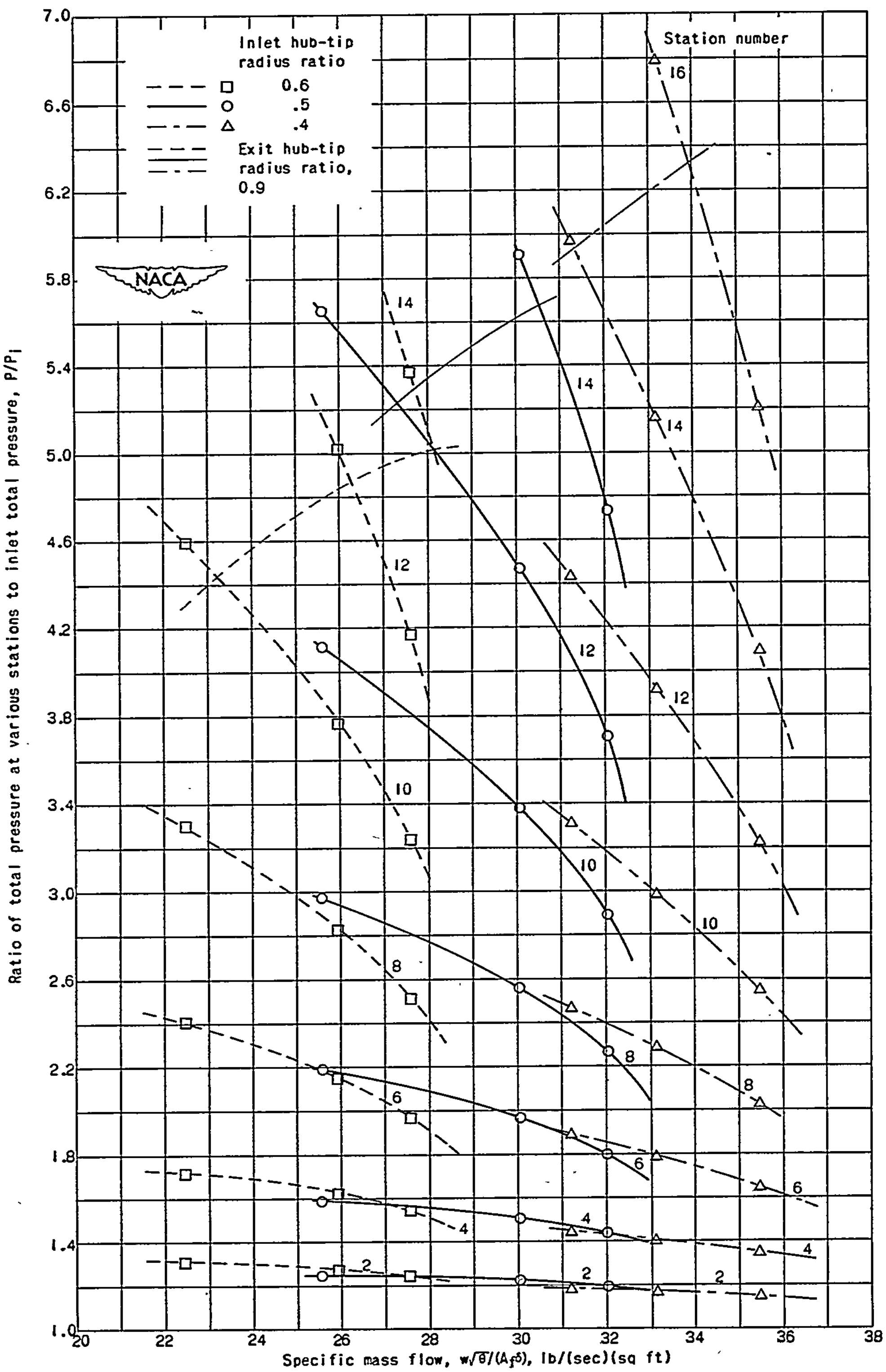

Figure 5. - Variation of total-pressure ratio with inlet hub-tip radius ratio and specific mass flow.

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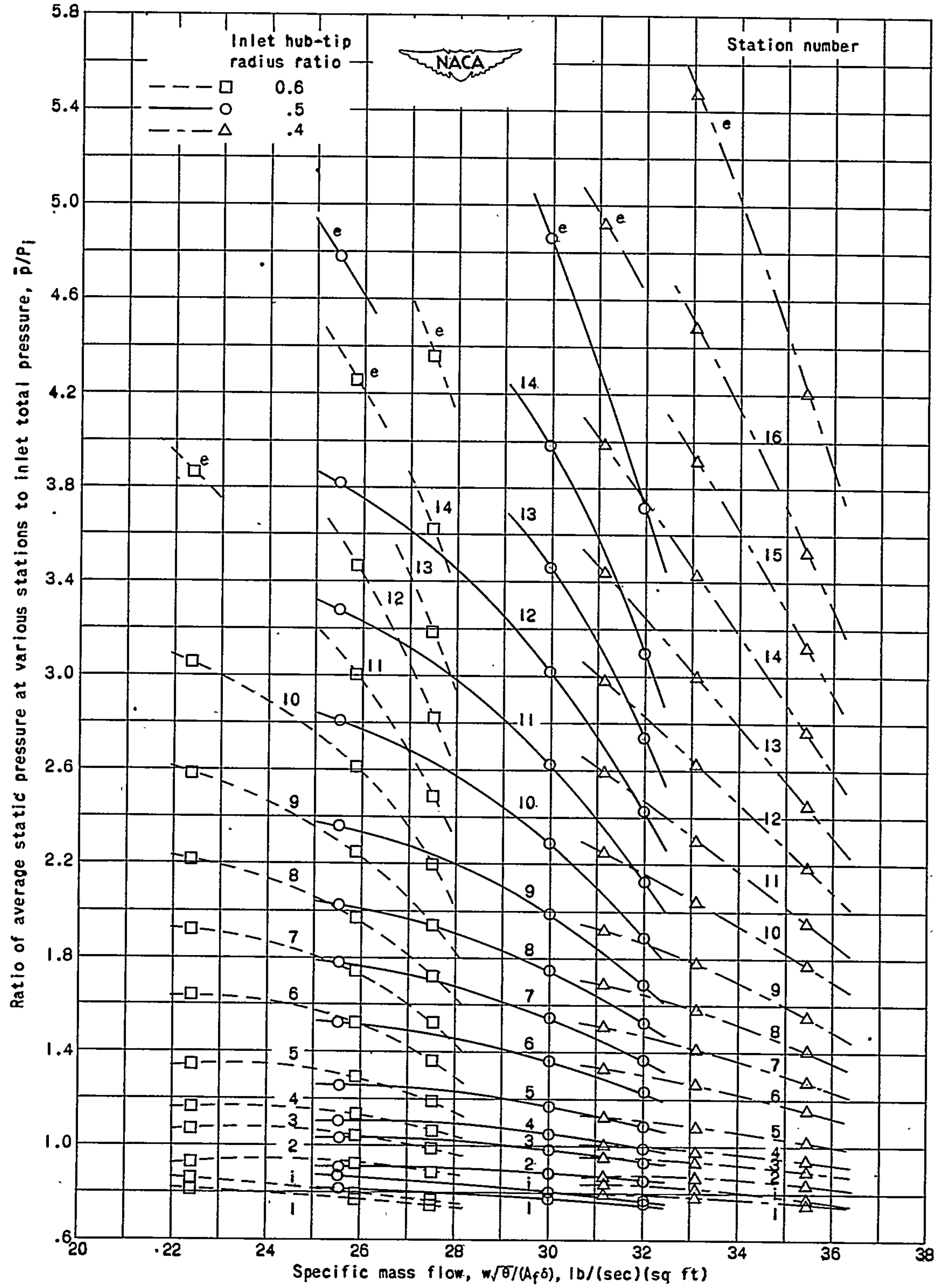

Figure 6. - Variation of average static-pressure ratio with inlet hub-tip radius ratio and specific mass flow.

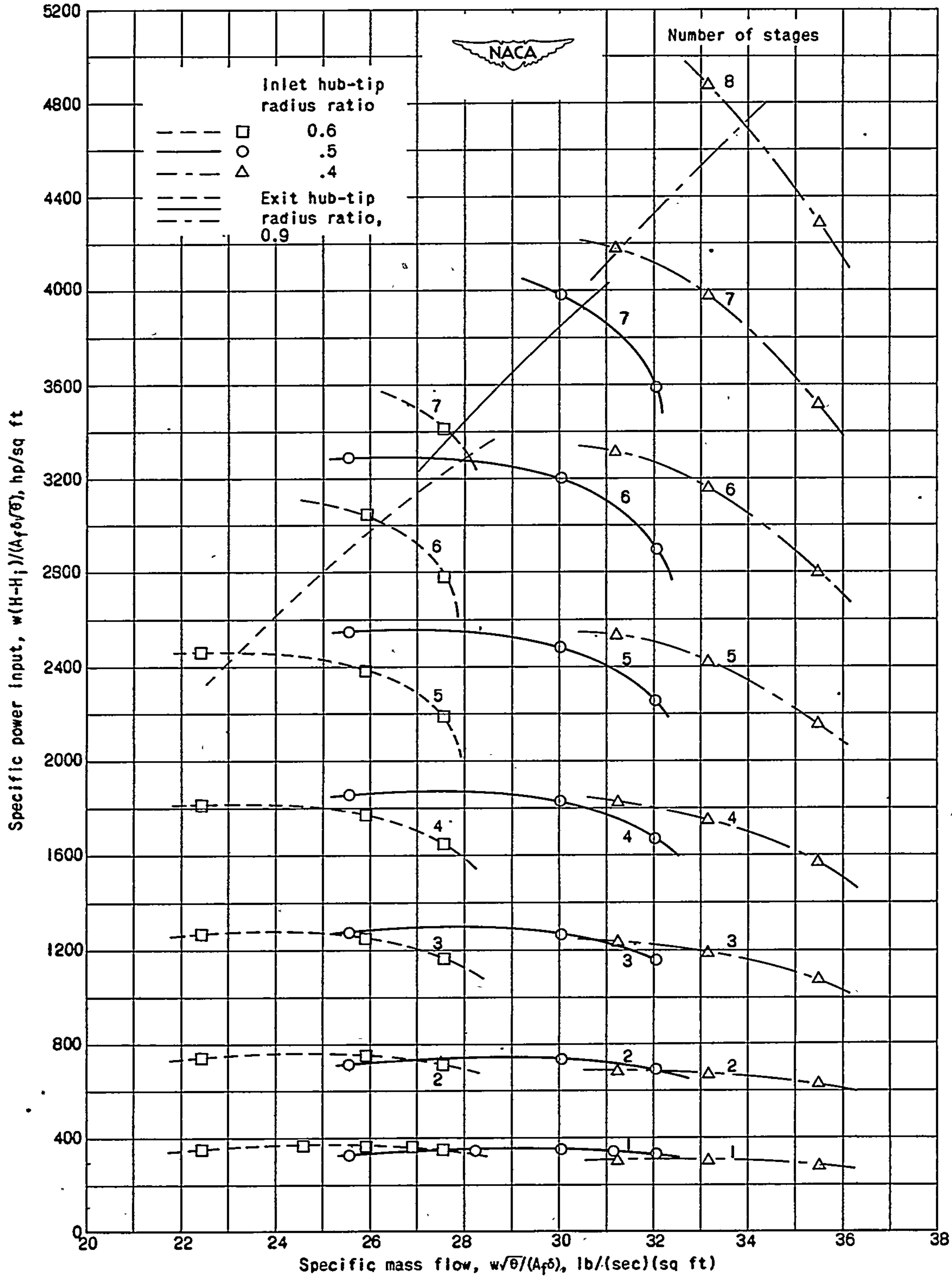

Figure 7. - Specific power input to compressor for various number of stages, specific mass flows, and inlet hub-tip radius ratios.

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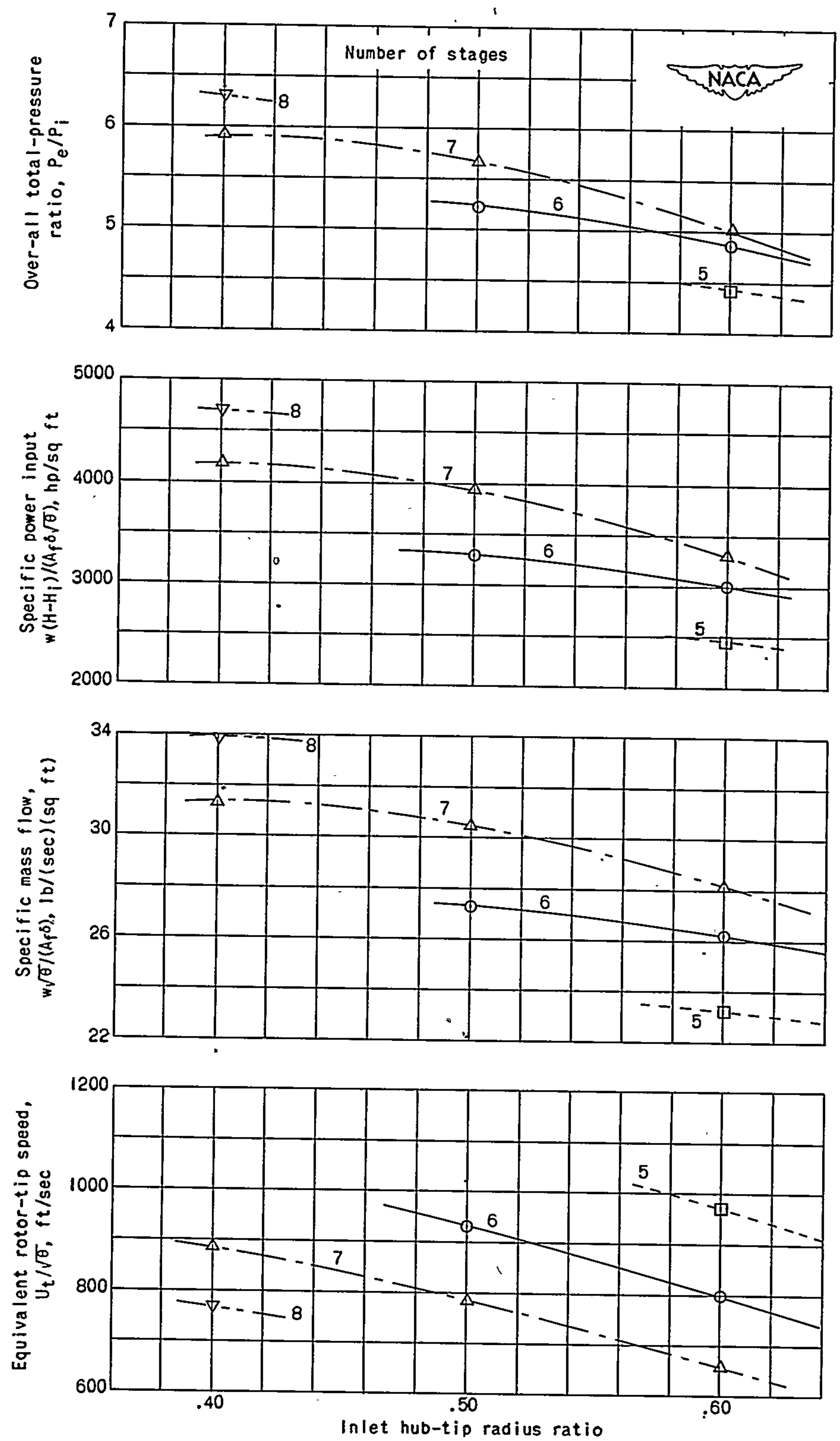

Figure 8. - Variation of design performance of compressors with inlet hub-tip ratio for constant exit hub-tip radius ratio of 0.9.

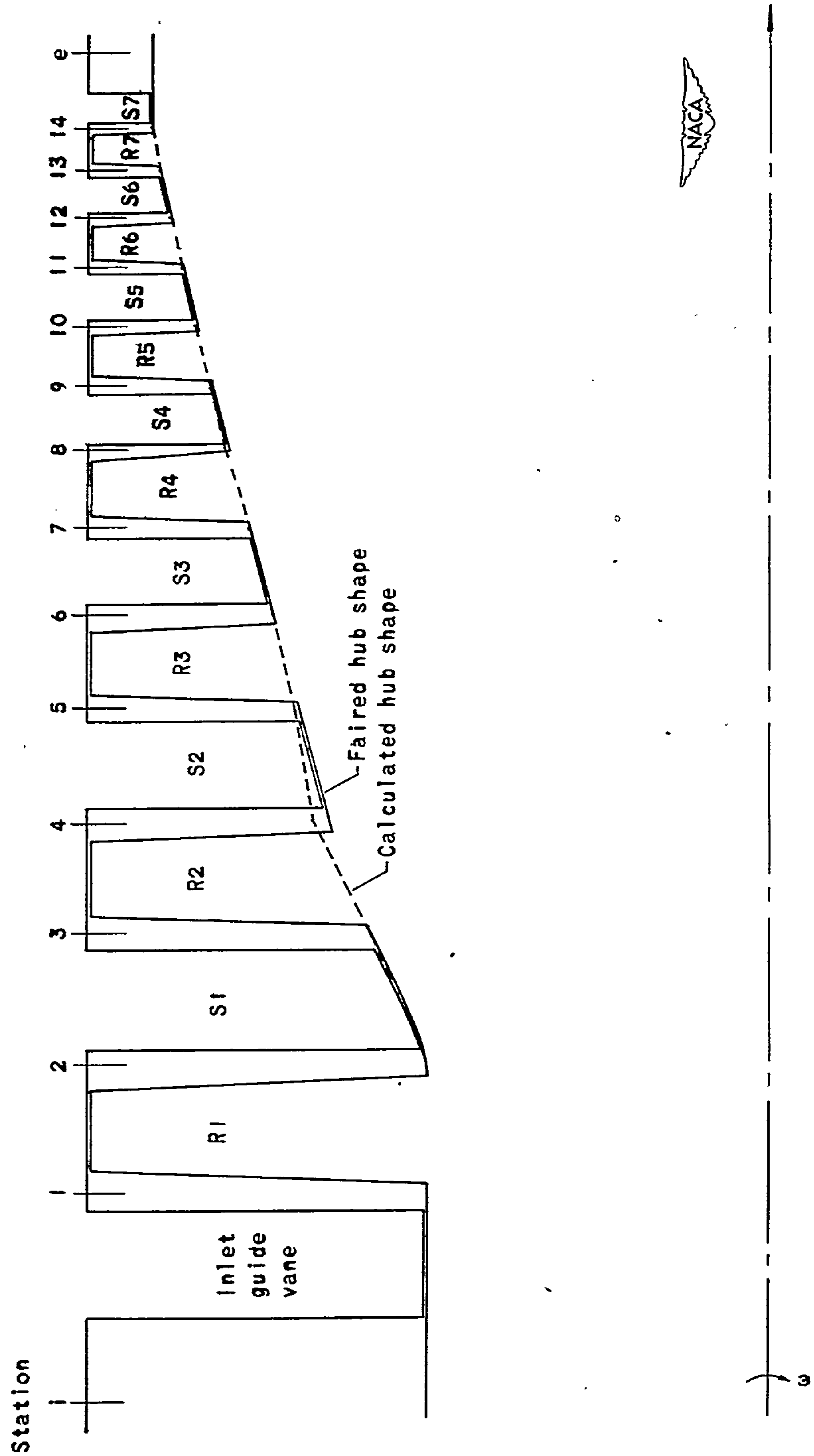

Figure 9. - Axial section of typical multistage compressor with inlet hub-tip radius ratio of 0.5 and design specific mass flow of 30.1 pounds per square foot.

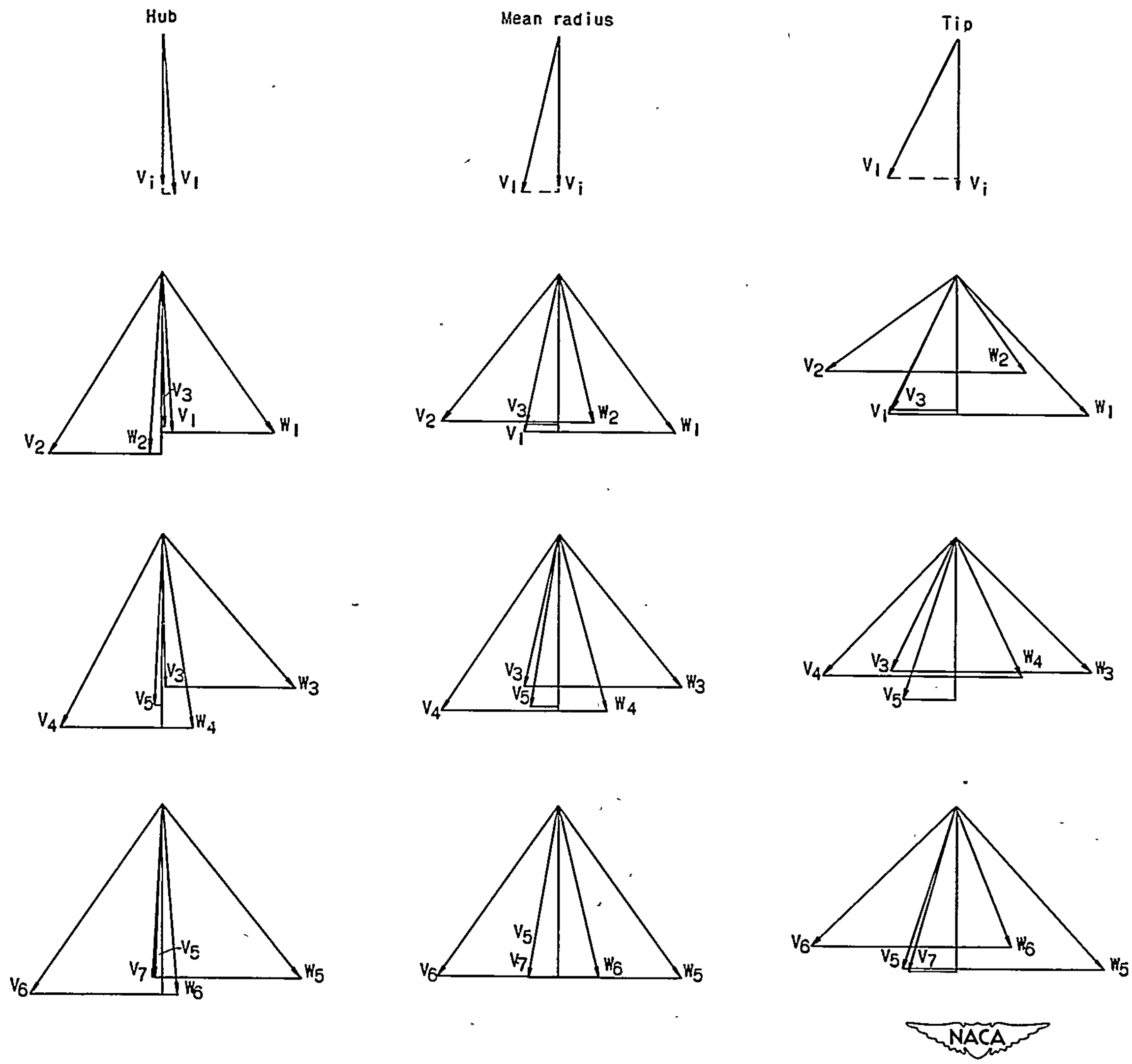

Figure 10. - Velocity diagrams of typical multistage compressor with inlet hub-tip radius ratio of 0.5 and specific mass flow of 30.1 pounds per second per square foot.

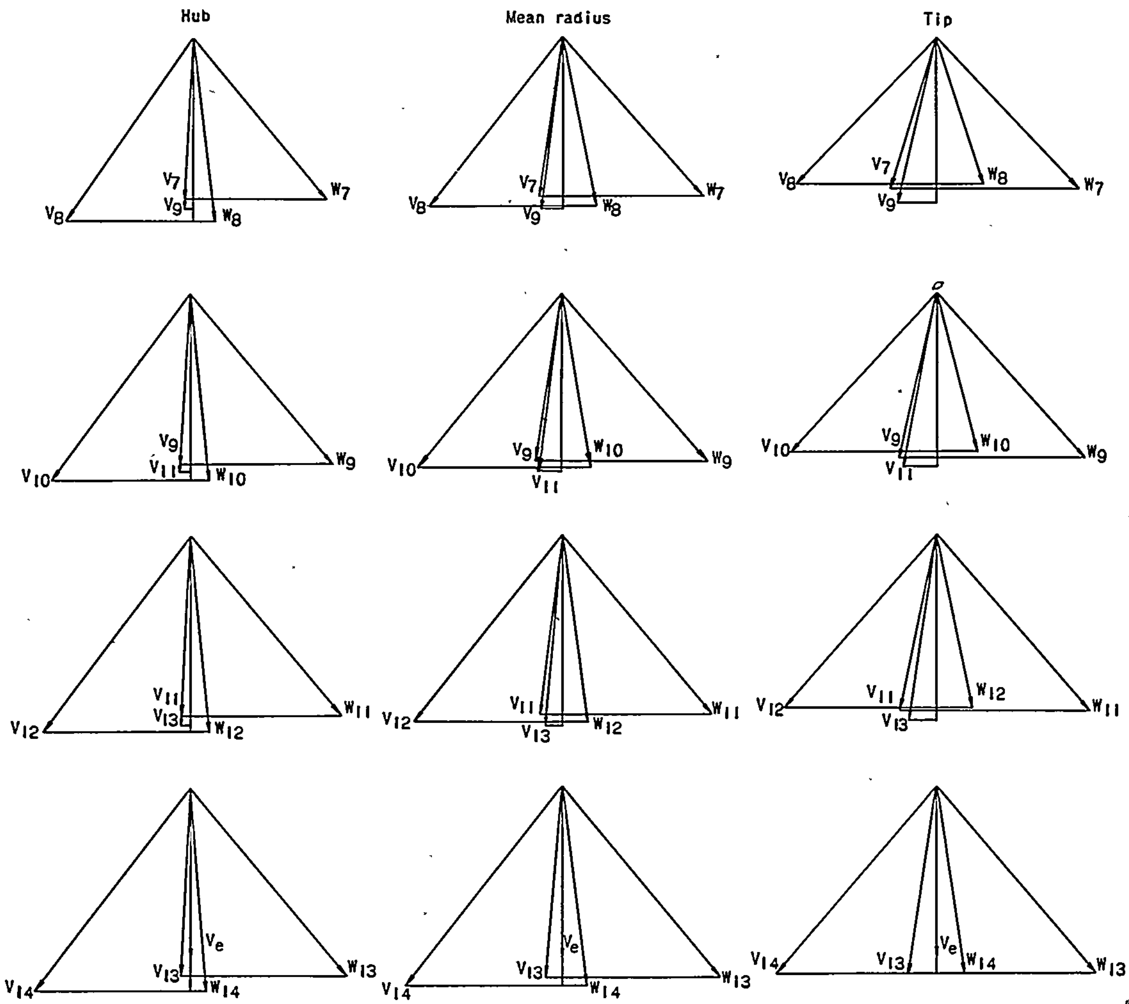

Figure 10. - Concluded. Velocity diagrams of typical multistage compressor with inlet hub-tip radius ratio of 0.5 and specific mass flow of 30.1 pounds per second per square foot.

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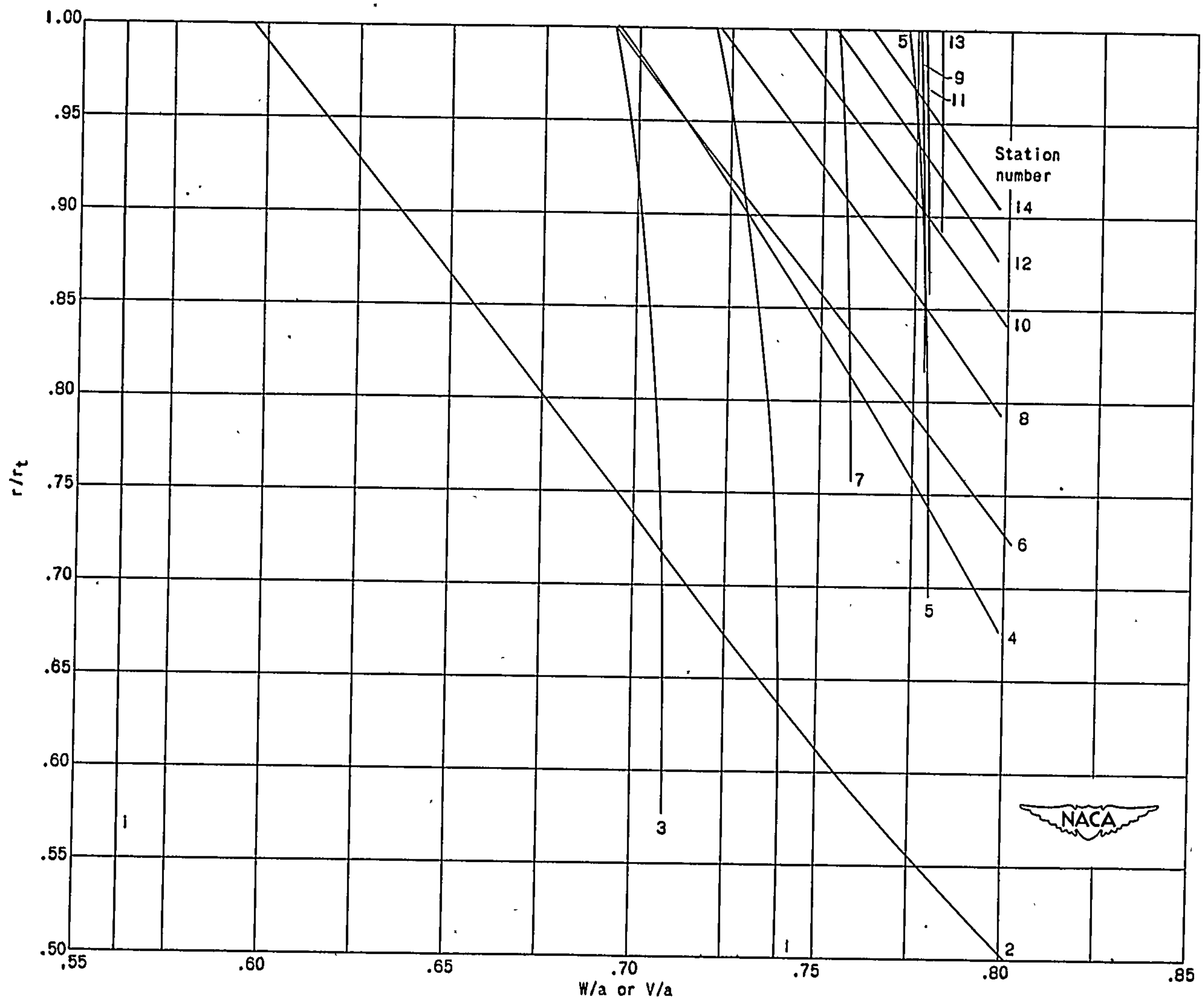
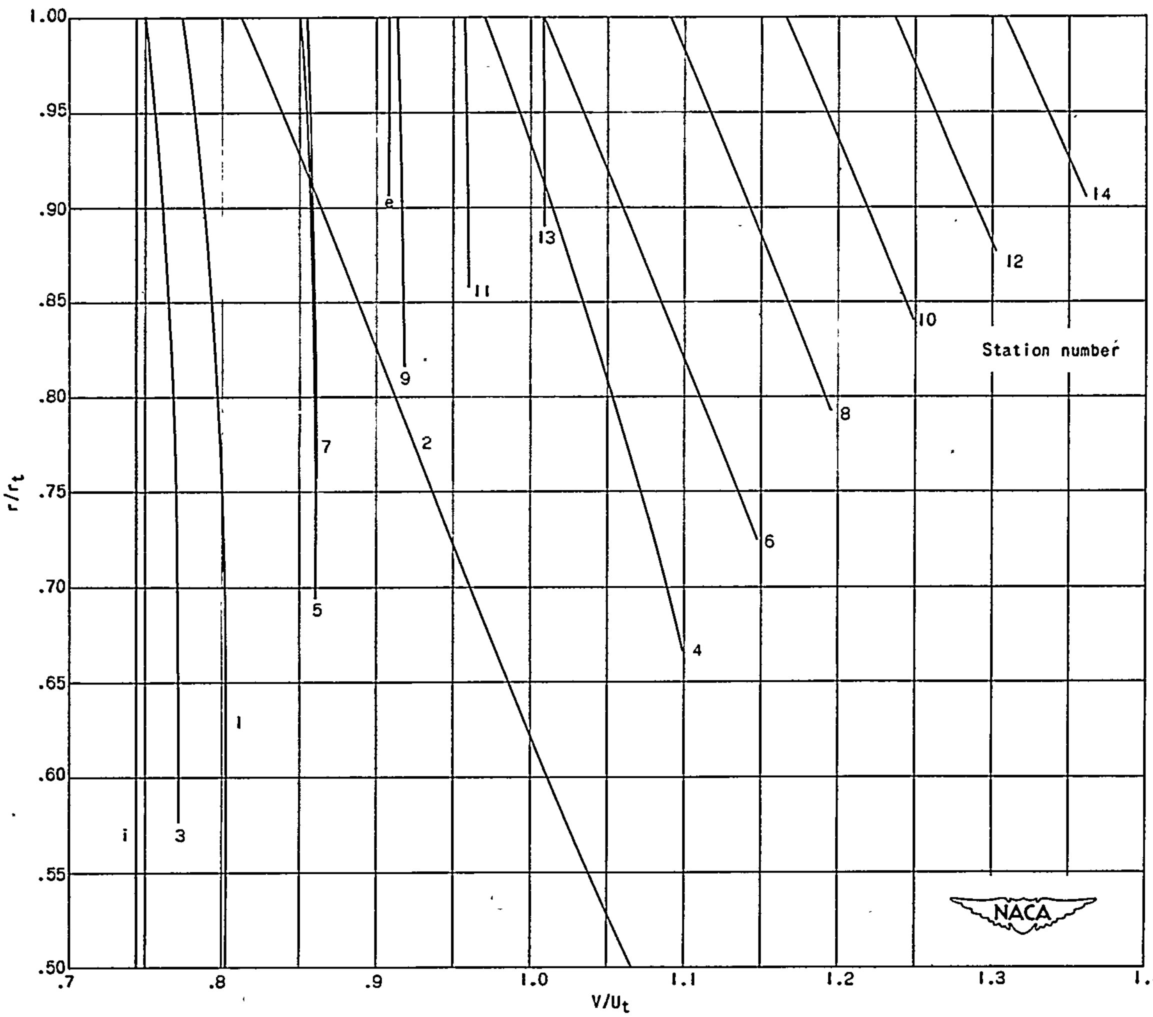


(a) Relative inlet Mach number.

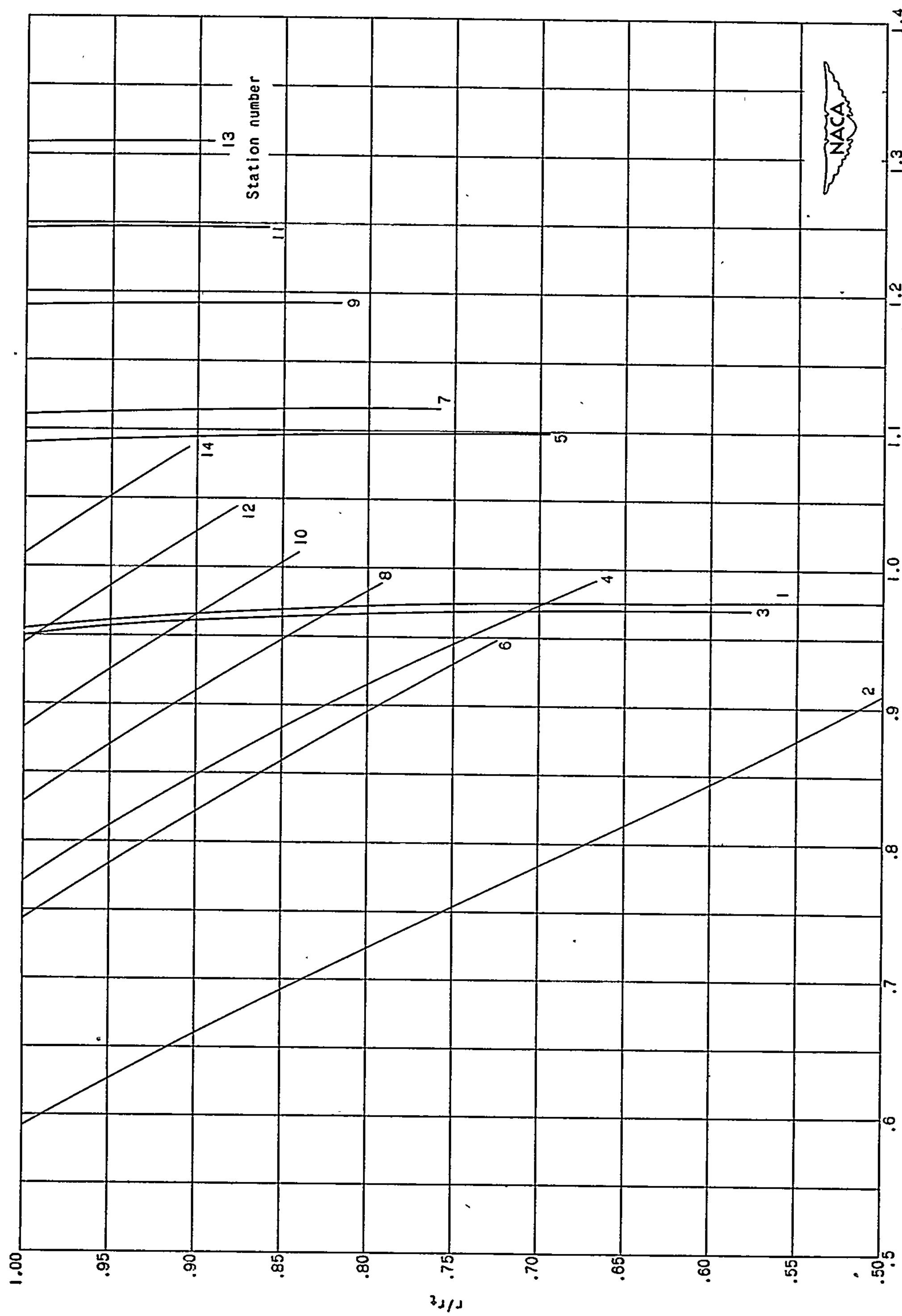
Figure 11. - Flow conditions through typical multistage compressor with inlet hub-tip radius ratio of 0.5 and specific mass flow of 30.1 pounds per second per square foot.



(b) Velocities relative to stator.

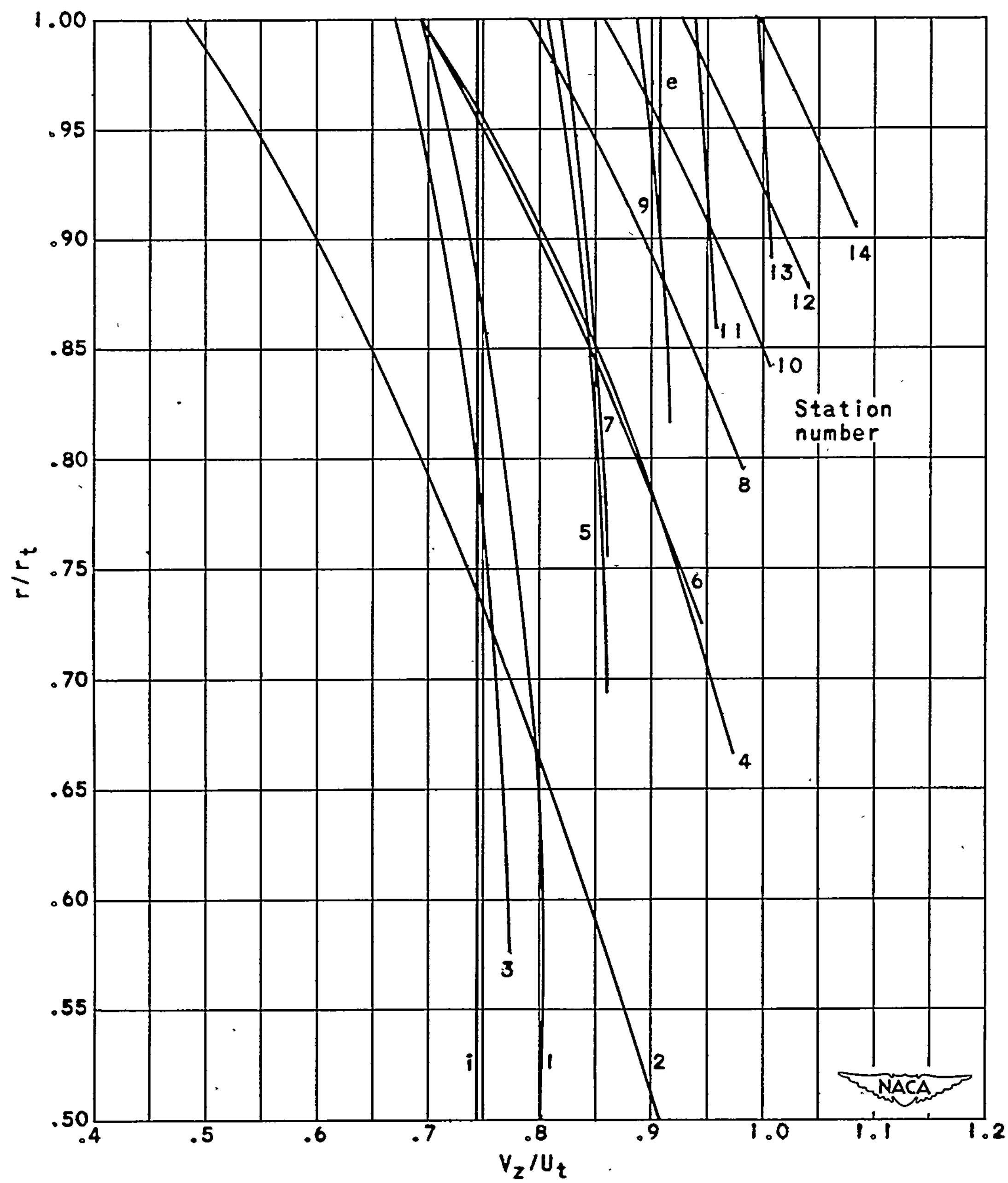
Figure 11. - Continued. Flow conditions through typical multistage compressor with inlet hub-tip radius ratio of 0.5 and specific mass flow of 30.1 pounds per second per square foot.

1241



(c) Velocities relative to rotor.

Figure 11. - Continued. Flow conditions through typical multistage compressor with inlet hub-tip radius ratio of 0.5 and specific mass flow of 30.1 pounds per second per square foot.



(d) Axial velocities.

Figure 11. - Continued. Flow conditions through typical multistage compressor with inlet hub-tip radius ratio of 0.5 and specific mass flow of 30.1 pounds per second per square foot.

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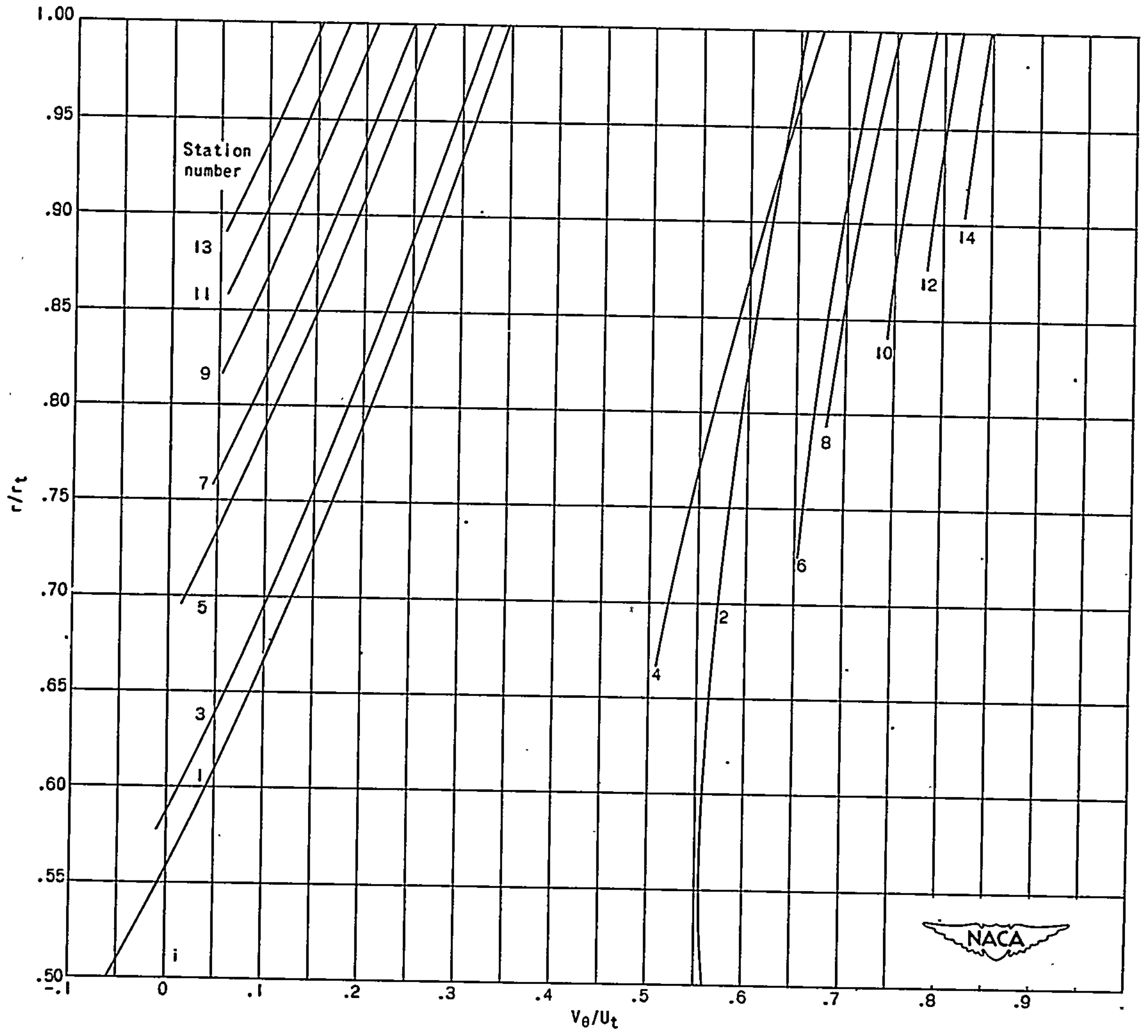
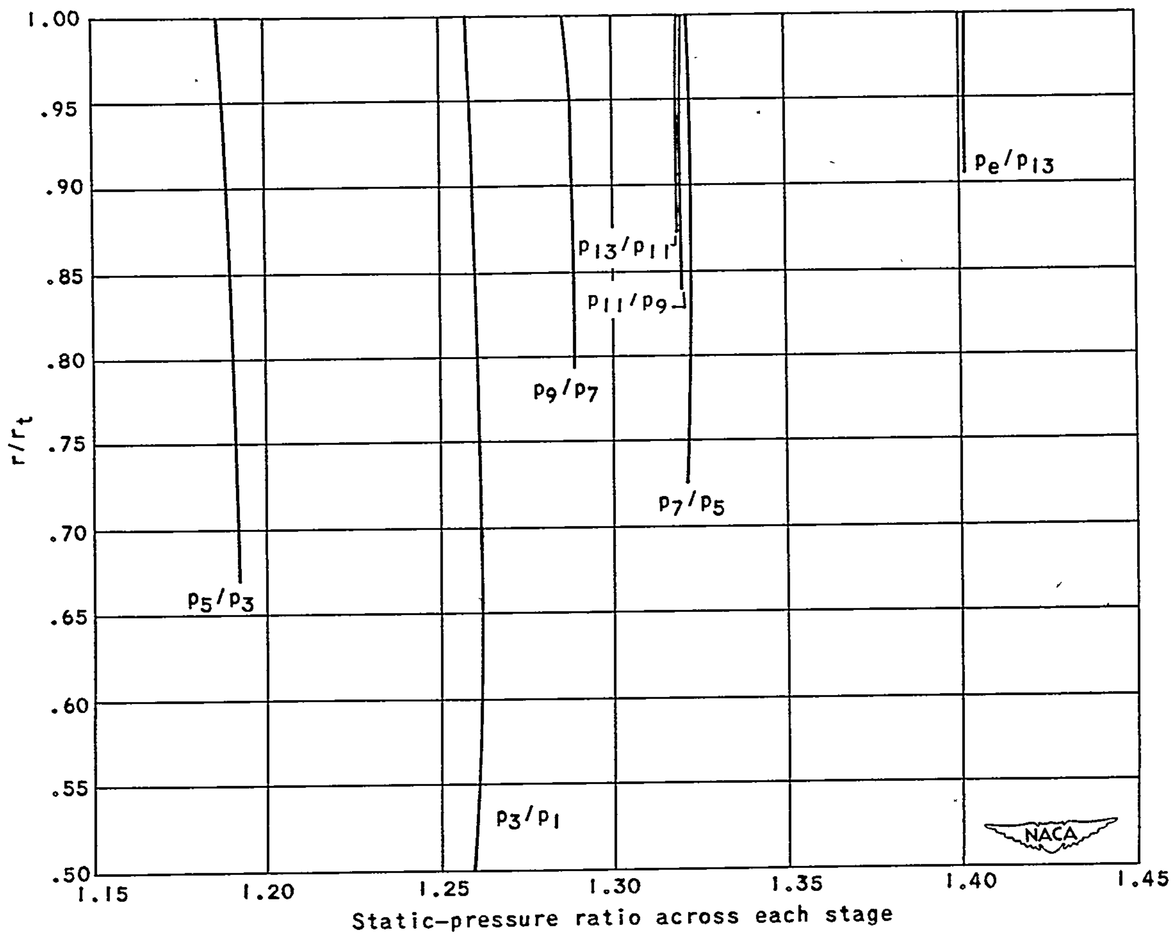


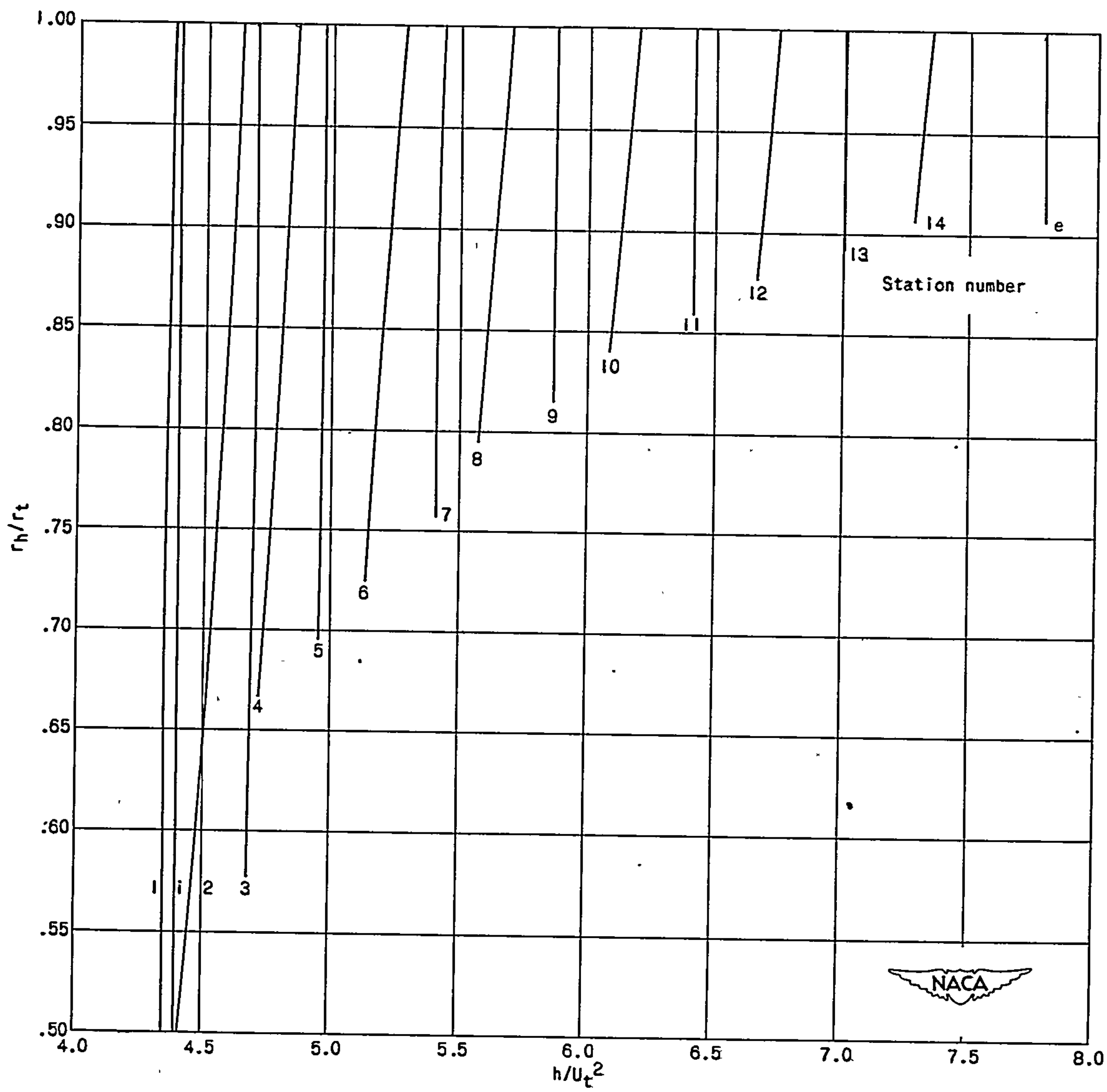
(e) Tangential velocities.

Figure 11. - Continued. Flow conditions through typical multistage compressor with inlet hub-tip radius ratio of 0.5 and specific mass flow of 30.1 pounds per second per square foot.



(f) Static-pressure ratios.

Figure 11. - Continued. Flow conditions through typical multistage compressor with inlet hub-tip radius ratio of 0.5 and specific mass flow of 30.1 pounds per second per square foot.



(g) Static enthalpy.

Figure 11. - Concluded. Flow conditions through typical multistage compressor with inlet hub-tip radius ratio of 0.5 and specific mass flow of 30.1 pounds per second per square foot.

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