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RESEARCH MEMORANDUM

for the

Bureau of Aeronautics, Department of the Navy

PERFORMANCE OF A 13-STAGE DEVELOPEMENT COMPRESSOR FOR THE J40-WE-24 ENGINE AT EQUIVALENT SPEEDS FROM 30 TO 112 PERCENT OF DESIGN

By James E. Hatch, James G. Lucas, and Harold B. Finger

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Cleveland, Ohio

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PERFORMANCE OF A 13-STAGE DEVELOPMENT COMPRESSOR FOR THE J40-WE-24 ENGINE AT EQUIVALENT SPEEDS FROM 30 TO 112 PERCENT OF DESIGN

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SUMMARY

A 13-stage development compressor for the J40-WE-24 engine was investigated over a range of speeds from 30 to 112 percent of design speed. It was found that design equivalent flow and pressure ratio could not be attained at design speed. The design-speed peak pressure ratio of 5.58 was obtained at a flow of 145.8 pounds per second. The peak efficiency at design speed was 0.783. The maximum efficiency of 0.830 was obtained at 85 percent of design equivalent speed. The surge pressure ratio at 112 percent of design speed was 6.02 and was obtained at a flow of 156.1 pounds per second. The maximum efficiency at 112-percent speed was 0.730.

An analysis was conducted to determine the reasons for the poor performance obtained at the design and over-design speed. The results indicated that excessively high relative Mach numbers combined with negative angles of attack at the tip of the first-stage rotor limited the attainable performance of the compressor. Therefore, the failure to meet design performance specifications may be mainly attributed to the fact that the first stage was overcompromised for low-speed operation. Analytical investigation of the effect of reducing the turning angle of the inlet guide vanes indicated that even if design flow could be attained by such a modification, the efficiency of the compressor would be lower than that presented herein.

INTRODUCTION

At the request of the Bureau of Aeronautics, Department of the Navy, the performance of a J40-WE-24 13-stage development axial-flow compressor designed to produce a pressure ratio of 6.0 at an equivalent flow of 164 pounds per second and an equivalent speed of 7260 rpm was investigated at the NACA Lewis laboratory to determine design and off-design performance characteristics. The compressor configuration investigated was modified by the manufacturer to favor part-speed performance. Analysis of the design and previous experience indicated that such a modification would have a detrimental effect on performance at design and over-design speed.

In flight operation, it is required that the compressor be able to operate with satisfactory performance at equivalent speeds up to 115 percent of design. However, with the lowest available inlet-air temperature it was possible to run only 112 percent of design speed without exceeding design mechanical rpm. Tests were run over a range of flows from maximum obtainable to surge at speeds from 30 to 112 percent of design. The inlet-air pressure was adjusted so that the average Reynolds number relative to the tip of the first rotor based on the rotor blade chord was above 400,000 at all speeds. Inasmuch as inlet-air temperature has a relatively minor influence on Reynolds number, the temperature most easily obtained was used.

This report presents a discussion of the following:

1. Over-all performance of compressor

2. Factors which might contribute to poor performance obtained at speeds over 85 percent of design:

   a. Inlet duct loss

   b. Flow conditions leaving inlet guide vanes

   c. Performance of first rotor

3. Predicted effect of guide-vane resetting:

   a. Flow conditions leaving inlet guide vanes

   b. Inlet conditions relative to first rotor

APPARATUS AND INSTRUMENTATION

Compressor

Type	Axial-flow
Number of stages	13
Design equivalent weight flow, lb/sec	164
Design total-pressure ratio	6.0
Design efficiency	0.838

Design equivalent speed, rpm	7260
First rotor tip diameter, in.	35
Tip diameter, stages 5 to 13, in.	32.35
Diameter ratio at first rotor inlet	0.55
Diameter ratio at last rotor inlet	0.84

Driving Power

Drive motor	15,000 hp syncronous motor
Speed control	Variable frequency
Gearbox	2.101-ratio single-helical speed increaser
Power available at 7260 rpm	14,400 hp
Compressor power required at design point with sea-level inlet conditions	23,000 hp

Installation and Instrumentation

Air supply	Atmospheric air or refrigerated air at pressure of 50 in. Hg abs
Exhaust	Altitude exhaust at 10 in. Hg abs
Air control	Butterfly throttles in inlet and outlet ducting
Air metering	Thin-plate orifice of 18.394-in. diam. in pipe of 41.25-in. I.D.
Compressor inlet	Depression tank 6 ft in diam. and approx. 10 ft long; dual inlet ducts for compressor installed in depression tank with bell-mouth nozzle on inlet of each
Compressor-inlet pressure	Two wall static-pressure taps 180° apart and six total-pressure tubes located at area center of equal areas in depression tank; fourteen 9-tube radial total-pressure rakes in dual inlet ducts ahead of inlet guide vanes
Compressor-inlet temperature	Five bare-wire thermocouples located at area center of equal areas in depression tank

Compressor-discharge pressure	Eight wall static-pressure taps spaced around circumference; two 11-tube circumferential total-pressure rakes located at different circumferential positions and each surveyed over five equispaced radial positions 1 in. downstream of discharge guide vanes
Compressor-discharge temperature. . . .	Four rakes spaced around circumference with five spike-type thermocouples (ref. 1) equally spaced across passage and located 1 in. downstream of discharge guide vanes
Angle measurement	Claw probe after inlet and discharge guide vanes
Compressor insulation	2 in. of glass wool
Pressure measurement	Water and mercury manometers
Temperature measurement	Calibrated Brown self-balancing potentiometer
Motor-speed measurement	Electric chronometric tachometer

The precision of measurements is estimated to be within the following limits:

Temperature, ^{o}R	±1.0
Pressure, in. Hg	±0.05
Weight flow, percent	±1.50
Speed, percent	±0.30

Compressor Design

Simple radial equilibrium, corrected for the mutual interference of adjacent blade rows, was used in the calculation of the original J40-WE-24 design, referred to in the present discussion as the "equilibrium" design compressor. Compressor tests made by the Westinghouse Electric Company on an engine having a compressor similar to the equilibrium design, except for higher rotor blade-setting angles as measured from the axial direction at the hub sections, indicated severe part-speed acceleration problems. On the basis of these tests, the manufacturer decided to modify the equilibrium compressor by twisting rotor and stator blade angles in an attempt to improve the part-speed operation. The resulting compressor, whose performance is presented herein, was modified from the equilibrium compressor by twisting the first seven rotors closed and the first five stators open. This compressor is referred to as the "modified" equilibrium design.

Following is a tabulation of the amount of twist applied to the equilibrium design compressor blading by the manufacturer to produce the modified compressor reported herein:

Rotor	Blade-setting-angle change, deg (a)			Stator	Blade-setting angle change, deg (b)		
	Radial position				Radial position		
	Tip	Mean	Hub		Tip	Mean	Hub
1	+14	+7	0	1	-10	-5	0
2	12	6	0	2	-8	-4	0
3	10	5	0	3	-6	-3	0
4	8	4	0	4	-4	-2	0
5	6	3	0	5	-2	-1	0
6	4	2	0	6	0	0	0
7	2	1	0	7	0	0	0

[a]Plus sign indicates increased angle measured from axial direction.
[b]Minus sign indicates decreased angle measured from axial direction.

The change in the angles of the first seven rotor blade rows was accomplished by twisting the blades closed from the tip section, so that the rotor blades would be unloaded most near the tip for a given stator blade setting, and the stators in these stages would have been similarly unloaded. However, the change in the stator blade angles may have counteracted the unloading effect of the rotors in stages 2 through 5, because these stator angles were reduced as measured from the axial direction. Calculations indicate that the increase in angle of attack on the rotors resulting from such a stator modification is smaller than the angle-of-attack change on the stator in the outer portion of the annulus. However, such a stator change tends to increase the angle of attack on the following rotor almost equally from hub to tip. Therefore, it is apparent that the changes in the blade rows beyond the first row may have nullified each other to some extent. Thus, the major effects of the modifications would be apparent only on the first rotor row.

Whereas the reduction in the angle of attack on the first-stage rotor row, which was accomplished by the rotor blade twist, would tend to improve the part-speed performance, the design and over-design speed weight flow and performance would be adversely effected. In order to attain the design weight flow, the inlet guide vanes would have to be opened by reducing the guide-vane turning angle. The proposed inlet guide vanes for the modified equilibrium design compressor were, therefore, opened 6 percent from those for the equilibrium design. However, at the termination of the tests, it was found that the manufacturer had installed the equilibrium rather than the modified guide vanes in the present compressor. It is, therefore, to be expected that the performance of the present compressor at the design and over-design speed conditions would be worse than might have been expected had the 6-percent-open guide vanes been installed. (A discussion of the flow characteristics which might be expected in the first rotor from such a guide-vane change is presented in the RESULTS AND DISCUSSION of the present report.) The guide vanes installed in the present compressor were sheet metal, rather than the airfoil-section de-icing vanes planned for the production engine.

RESULTS AND DISCUSSION

Over-All Performance

The over-all compressor performance is presented in figure 1 as curves of total-pressure ratio and adiabatic efficiency plotted against equivalent weight flow with constant percentages of equivalent design speed as a parameter. The discharge total pressure for each data point was obtained by two methods. The "measured" total pressure is the arithmetic average of the two circumferential rake measurements at five radial positions. The "calculated" total pressure was calculated by the method recommended by the NACA Subcommittee on Compressors (ref. 2), where the average velocity head determined from the discharge static pressure, total temperature, annulus area, and the continuity equation is added to the measured wall static pressure to give a discharge total pressure. This pressure is considered to be more representative of the effective total pressure entering the diffuser, and the compressor is not credited for nonuniformity of velocity and deviations from axial discharge. Adiabatic temperature-rise efficiencies were calculated with both "measured" and "calculated" discharge pressures for the adiabatic power input and the arithmetic average of the twenty discharge temperatures for the actual power input. From figure 1, it can be seen that the largest deviations between "measured" and "calculated" values of pressure ratio and efficiency occur at high percentages of design speed and at the high-weight-flow end of the curves at low speeds. Measurement of circumferential distribution of total pressure showed much larger exit-guide-vane wakes at the high-flow ends of the curves. The deviation of discharge flow angle was small, varying from approximately 5° in the direction of rotor rotation to 10° in the opposite direction.

The curves of figure 1 as well as the following discussion are based on "calculated" values, because these are considered to represent more accurately the usable performance in the engine. At design equivalent speed, the maximum equivalent weight flow was 150.2 pounds per second, peak efficiency was 0.783, and surge total-pressure ratio was 5.58 at an equivalent weight flow of 145.8 pounds per second. At 112 percent of equivalent design speed the maximum equivalent flow was 156.9 pounds per second, peak efficiency was 0.730, and surge total-pressure ratio was 6.02 at a flow of 156.1 pounds per second. The maximum efficiency for the compressor was 0.830 and was obtained at 85 percent of equivalent design speed.

The surge point was defined as that pressure ratio and weight flow, giving stable operation, as near to audible surge as possible without danger that small transient fluctuations of flow would induce audible surge. No audible surge was encountered at 30 percent of equivalent design speed, but was encountered at all other speeds. In order to define the surge line more completely and to determine if a discontinuity in the slope of the surge line existed, surge points were run at speeds of 68, 70, and 73 percent of design. The surge limit (fig. 1) is in general smooth, except for an indication of a slight dip between 65 and 75 percent of design speed.

Flow conditions entering diffuser. - The flow conditions into the exit diffuser and combustor can be determined by consideration of the measured compressor-discharge pressures. At all speeds, the average total pressure was measured on the two circumferential rakes covering two out of 110 exit-guide-vane passages. The passages investigated were located approximately 45° apart, and the maximum difference of average total pressure obtained from the two rakes was approximately 2 percent and occurred at the high-flow points at high speed. From the total pressures obtained from one rake and discharge static pressure and total temperature, the velocity distribution radially and circumferentially across one blade passage was calculated and is presented in figures 2(a), (b), and (c) for the maximum efficiency points at 85, 100, and 112 percent of equivalent design speed, respectively. For the maximum efficiency points presented, the general shape of the velocity distributions did not change with speed. The circumferential variations were greatest at the two stations nearest the tip and were small for the other positions. It is expected that the circumferential variations might dissipate with diffusion, but the radial gradients would have to be taken into account in the design of an efficient diffuser and combustor.

Variation of flow with speed. - The variation of maximum and surge equivalent weight flow with percent of equivalent design speed is presented in figure 3. The curves are smooth except for the surge flow in the range of 65 to 69 percent of design speed. The increased rate of change of surge flow with speed in this range indicates that some inlet stage or stages became unstalled in this region. Such a result has been noted previously in other investigations (ref. 3), where unstalling of a stage or group of stages causes an abrupt rise in pressure ratio and weight flow with only a small change in speed. This characteristic will be substantiated in the following section of this report, with reference to the average performance of the first-stage rotor. No points of maximum weight flow were determined in this range of speeds, so that no discontinuity in the maximum-flow curve is indicated.

The design flow point is indicated in figure 3. The failure of this compressor to pass design flow may be attributed to the fact that the maximum flow starts leveling off below design speed. This tendency of maximum flow to level off at high speeds is common and indicates the imminence of choking conditions in an inlet stage with attendant increases in losses. The results of an analysis of the causes for this reduced flow at design and over-design speed are presented in the following section.

Factors Affecting Performance

In order to determine the location of the flow limit and to offer a possible explanation for the failure of the compressor to meet design mass flow and pressure ratio, an analytical investigation of the factors that might contribute to poor over-all performance at high rotational speed was conducted. The factors investigated were the inlet duct loss, the flow conditions at the guide-vane discharge, and the first-stage rotor performance.

Inlet duct loss. - Since the performance presented in figure 1 is based on stagnation conditions measured in the inlet depression tank upstream of the dual inlet ducts, a pressure drop in the ducts would penalize the compressor performance. The total-pressure drop through the ducts was determined by 14 radial rakes, each with 9 tubes, located ahead of the inlet guide vanes. At all speeds the variations of total pressure over the duct area were small, except for decreases indicated by the tubes in the wall boundary layer. A curve of the variation of $\Delta P/P_1$ with equivalent weight flow is presented in figure 4, where ΔP is the average drop in total pressure through the ducts and P_1 is the total pressure at the inlet to the ducts. (Symbols are defined in the appendix.) The ordinate, as plotted, represents the percentage loss in inlet total pressure. If the compressor performance were rated on conditions at the inlet to the compressor, the pressure ratio and equivalent weight flow would be increased by this percentage and the efficiency would be increased by a smaller amount. The shape of the duct pressure-loss curve is such that applying this correction to weight flow would reduce slightly the leveling-off tendency of the curves of weight flow against speed of figure 3. It can be seen that the correction on pressure ratio and equivalent weight flow at design speed is small, amounting to between 2.5 and 3 percent. Basing the design-speed performance on flow conditions without the inlet ducts (neglecting the effect of any nonuniform inlet velocity distributions caused by the dual inlet) would increase the surge pressure ratio from 5.58 to 5.73 and the surge equivalent weight flow from 145.8 to 149.7 pounds per second compared with design pressure ratio of 6.0 and design weight flow of 164 pounds per second.

Thus, inlet duct pressure loss does not seem excessive and only partly accounts for the poor design-speed performance.

Flow conditions leaving inlet guide vanes. - An imminent inlet-stage choking condition at high speed is indicated by the trend of the variation of weight flow with speed of figure 3 and the fact that design flow was not attained at design speed. If the absolute Mach number leaving the guide vanes at design speed were high, the possibility would exist that the guide-vane passages were approaching a choking condition and limiting the mass flow. In order to study this possibility, the flow conditions at the exit of the guide vanes were investigated. By using the measured air-flow angles leaving the inlet guide vanes, the continuity equation, standard sea-level stagnation pressure and temperature, and simple radial equilibrium, the flow conditions downstream of the guide vanes were determined for the equivalent flow of 149 pounds per second measured at the maximum efficiency point at design speed. Figure 5 compares measured air angles and the design air angles for the inlet guide vanes actually used in this compressor. Agreement between design and predicted angles was good except at the tip. It is improbable that the angle probe was traversing in the wake region, since it was located midway between two guide vanes. The underturning measured at the tip has been discussed in reference 4, and a method of predicting the guide-vane turning angle is presented in references 4, 5, and 6. Also presented in figure 5 is the absolute Mach number leaving the guide vanes. The low Mach numbers at the various radial stations (a maximum of 0.56 at the hub) eliminate the possibility that the guide-vane passages were choking or were instrumental in causing the leveling-off trend of flow with speed indicated by figure 3.

Performance of first rotor. - The preceding discussions indicate that the failure of the compressor to meet design-speed performance requirements could not be wholly attributed to pressure drop in the dual inlet ducts or flow conditions downstream of the inlet guide vanes. Consequently, the next objective was the investigation of flow conditions relative to the first rotor blade row, in order to determine whether this blade row acts as the performance limitation.

As the equivalent speed of an axial-flow compressor is increased, the inlet stages operate at successively lower angles of attack. If the inlet stages were designed so that optimum angle of attack occurred too far below design speed, negative angles of attack at design speed and mass flow would result, with consequent high losses and the possibility of a choking condition in the first stage. This condition would be aggravated by the existence of high Mach numbers in conjunction with the low angles of attack.

From the flow conditions leaving the inlet guide vanes, the radial variations of angle of attack and relative Mach number on the first rotor blade row were calculated for the peak-efficiency weight flow at design speed. The results of this calculation are presented in figure 6, where angle of attack and relative Mach number at the inlet of the first rotor blade row are plotted against radius ratio. Also shown are approximate cascade stall and design angles for the rotor blade row as obtained from the cascade data of reference 7. The airfoil sections used in the present compressor were correlated with the 65-series airfoils of reference 7 by assuming that blades having equal ratios of maximum camber height to blade chord would give similar turning and loss characteristics. The negative angles of attack on the tip sections of the first rotor blade row, combined with high Mach numbers, indicated the probability of high losses in the first-stage rotor. The curve of cascade design angle of attack was based on tests at low Mach numbers. However, reference 8 and unpublished information indicate that the minimum angle of attack for low loss increases as the Mach number is increased and the optimum angle of attack is increased above the design angle of attack indicated. Thus, it appears that the combination of high relative Mach number with low angle of attack at the tip region of the first rotor blade row contributes to the low efficiency, pressure ratio, and weight flow obtained at the design and over-design speeds.

This fact is further verified by consideration of the performance curve of the first rotor blade row, determined from the measured wall static pressures before and after the first-stage rotor. The performance of this rotor blade row is presented in figure 7 in terms of a static-pressure coefficient plotted against a flow coefficient for the various speeds investigated. Some scatter is indicated at the 30-percent-speed condition, where the pressure rise was extremely low, and any error is magnified by the inverse of the square of the speed. This curve indicates that at design and at 112 percent of design speed, the first-stage rotor is operating on the vertical portion of its performance curve in the turbining or pressure-drop region. Only at the surge point at design speed does a static-pressure rise exist across this rotor blade row, and even here the rise is negligible. A small drop in flow coefficient is indicated at design speed, and a greater decrease is obtained with increase in speed from design to 112 percent of design speed. These decreases are to be expected from the effect of Mach number on the low-loss range and turning angles of a blade section. The sharp drop in pressure coefficient between the surge points at 65 and 68 percent of design equivalent speed substantiated the fact that the first rotor becomes unstalled in this region, with the resulting discontinuity in the shapes of the surge curves of figures 1 and 3 as discussed under Over-All Performance.

The combination of low angle of attack and high Mach number at the tip of the first rotor blade row indicates that the failure of the compressor to meet design performance specifications may be mainly attributed to the fact that the first stage is overcompromised for low-speed operation, with resulting choking and associated high losses in the first rotor blade row at design speed and over.

Predicted Effect of Resetting Inlet Guide Vanes

In order to determine the extent of the improvement which might be expected from a change in the inlet-guide-vane angles from those used in the present compressor to those intended for the modified equilibrium compressor, an analysis was conducted to determine the flow conditions leaving the modified inlet guide vanes and entering the first rotor when the design flow of 164 pounds per second is assumed to be attained at design speed.

Predicted flow conditions leaving inlet guide vanes. - The design flow angles leaving the inlet guide vanes actually intended for use with the modified equilibrium compressor are presented in figure 8 as a function of radius ratio. Also presented are the predicted flow angles obtained by assuming that the difference between the measured and design angles for the equilibrium guide vanes (fig. 5) installed in the present compressor would be the same as the difference for the modified guide vanes. This assumption was used for expediency. The flow angles, however, could have been calculated from references 4, 5, and 6. The assumed turning angles thus determined were used to calculate the flow conditions behind the modified guide vanes for a flow of 164 pounds per second. The resulting distribution of absolute Mach number is also shown in figure 8. A maximum Mach number of 0.58 occurred at the hub. At all radii the Mach number is only slightly higher than those determined for the equilibrium guide vanes (fig. 5) for a flow of 149 pounds per second. Thus, the guide vanes would not present any flow limit at the design condition.

Predicted flow conditions entering first rotor. - From the absolute velocities after the modified inlet guide vanes, the flow conditions relative to the first-stage rotor at design speed and flow were calculated. These calculations are presented in figure 9 as curves of rotor angle of attack and relative entrance Mach number against radius ratio. Also presented are the approximate stall and design angles of attack determined from the NACA 65-series cascade data. It is apparent that at the design conditions, the first rotor would be operating well below the design angle of attack over the outer half of the annulus, with negative angles of attack in the outer quarter of the passage. The angle of attack at the tip would be 2.5° lower than the angle of attack determined for the compressor configuration reported herein (compare with fig. 6). In addition to this lower tip angle of attack, the relative Mach number at the tip section would be higher than that presented in figure 6 and would approach sonic values. Thus, it appears that even with the modified guide vanes, the first-stage rotor would appreciably limit the attainable performance of the compressor at the design conditions, so that even if design mass flow could be reached, the efficiency would be lower than that of the compressor reported herein. This again is a consequence of overcompromising the first stage for part-speed operation.

SUMMARY OF RESULTS

The following results were obtained from an investigation of the over-all performance of a 13-stage developmental version of the axial-flow compressor for the J40-WE-24 engine:

1. Design equivalent flow and pressure ratio could not be attained at design speed. Surge occurred at a pressure ratio of 5.58 and an equivalent flow of 145.8 pounds per second. Peak efficiency at design speed was 0.783.

2. The maximum efficiency of 0.830 was attained at 85 percent of design speed.

3. At 112 percent of equivalent design speed the maximum equivalent flow was 156.9 pounds per second, peak efficiency was 0.730, and surge total-pressure ratio was 6.02 at a flow of 156.1 pounds per second.

4. The pressure losses in the dual inlet duct were small at all flow conditions.

5. An analysis of the flow conditions leaving the inlet guide vanes and entering the first rotor indicated that negative angles of attack combined with high relative Mach numbers near the tip of the first rotor contributed to the poor performance obtained at the design speed. Modification of the guide vanes to reduce the turning angle would further

increase the relative Mach number and reduce the angles of attack near the tip section, if the design flow of 164 pounds per second were obtained. Thus, further reduced efficiency would be expected for such a modified configuration. In both cases, the difficulty of low angle of attack and high Mach number may be attributed to the overcompromise of the first-stage rotor to favor low-speed performance.

Lewis Flight Propulsion Laboratory
National Advisory Committee for Aeronautics
Cleveland, Ohio, April 9, 1953

APPENDIX - SYMBOLS

The following symbols are used in this report:

N	rotor speed, rpm
P	total pressure
p	static pressure
r	radius
T	total temperature
W	weight flow, lb/sec
$\frac{W\sqrt{\theta}}{\delta}$	weight flow corrected to NACA standard sea-level pressure and temperature, lb/sec
Y_s	parameter $\left(\frac{p_{or}}{p_{ir}}\right)^{\frac{\gamma-1}{\gamma}} - 1$
α	angle of attack
γ	ratio of specific heats for air
δ	ratio of inlet total pressure to NACA standard sea-level pressure
θ	ratio of inlet total temperature to NACA standard sea-level temperature

Subscripts:

ir	inlet to first rotor row
or	outlet from first rotor row
std	NACA standard sea-level conditions
t	tip radius
1	instrument station in depression tank

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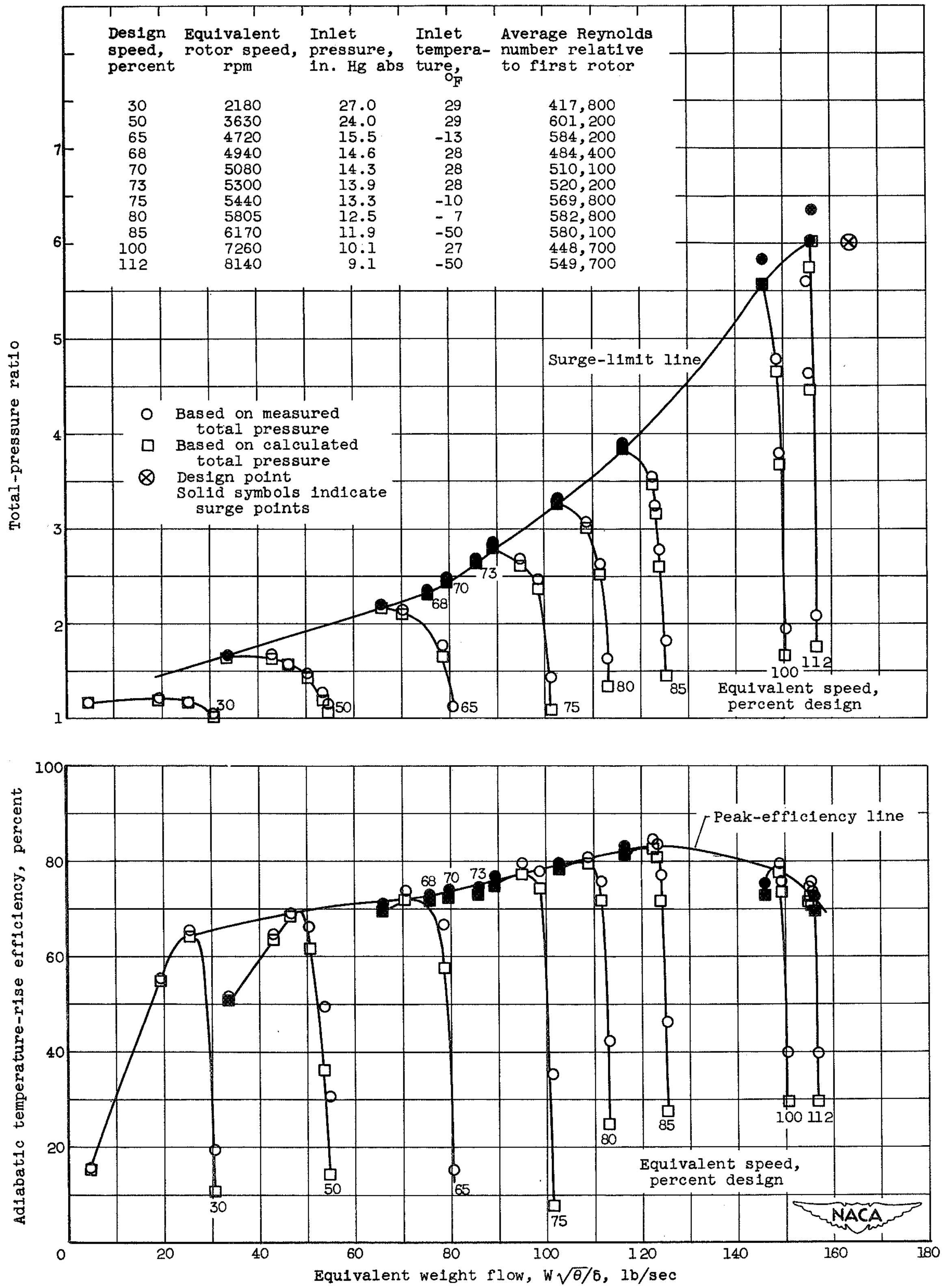
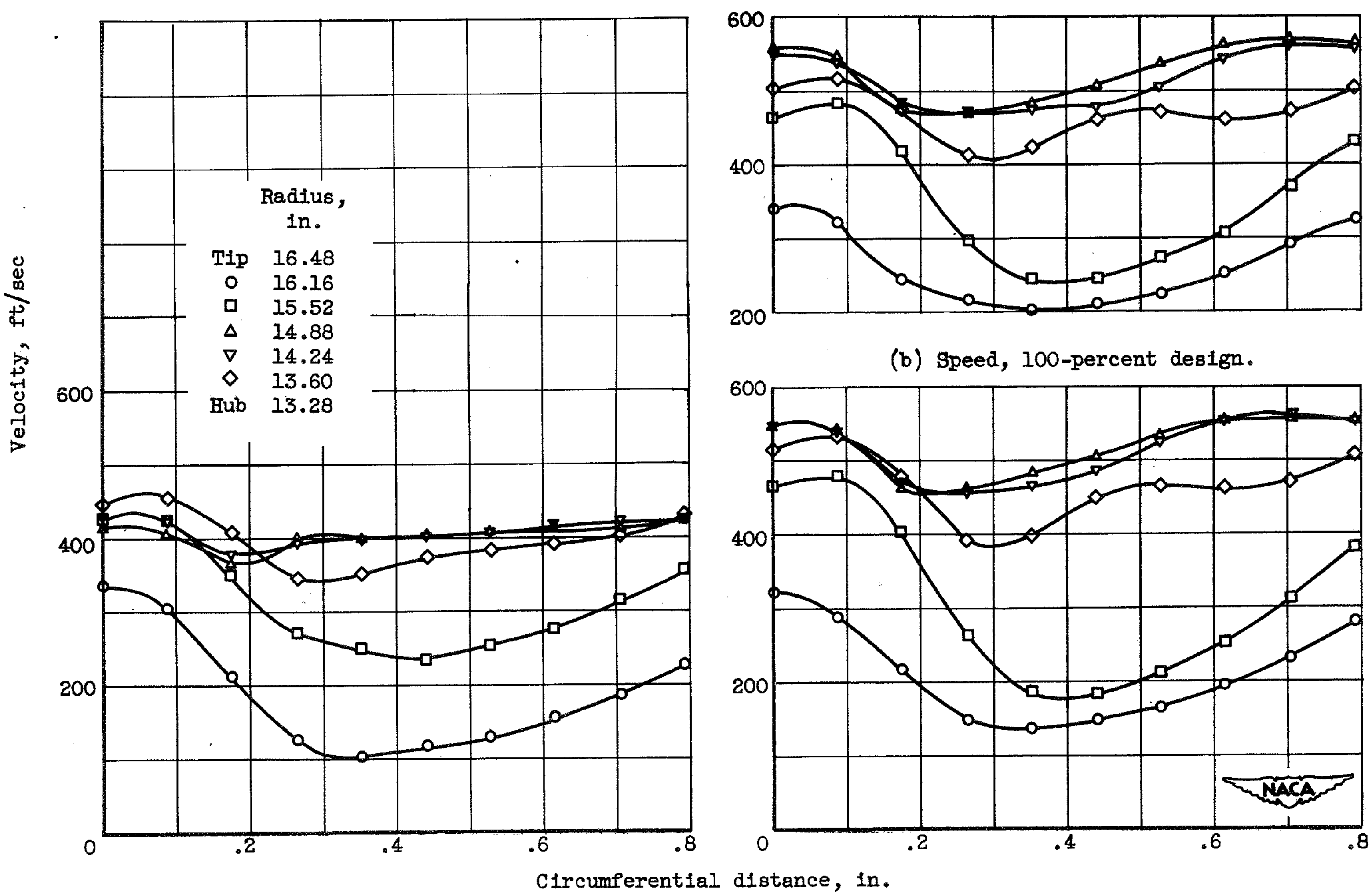


Design speed, percent	Equivalent rotor speed, rpm	Inlet pressure, in. Hg abs	Inlet temperature, °F	Average Reynolds number relative to first rotor
30	2180	27.0	29	417,800
50	3630	24.0	29	601,200
65	4720	15.5	-13	584,200
68	4940	14.6	28	484,400
70	5080	14.3	28	510,100
73	5300	13.9	28	520,200
75	5440	13.3	-10	569,800
80	5805	12.5	- 7	582,800
85	6170	11.9	-50	580,100
100	7260	10.1	27	448,700
112	8140	9.1	-50	549,700

Figure 1. - Over-all performance of J40-WE-24 modified equilibrium design compressor.



(a) Speed, 85-percent design.

(b) Speed, 100-percent design.

(c) Speed, 112-percent design.

Figure 2. - Variation of compressor-discharge velocity with radial and circumferential position for maximum efficiency points at 85, 100, and 112 percent of design speed.

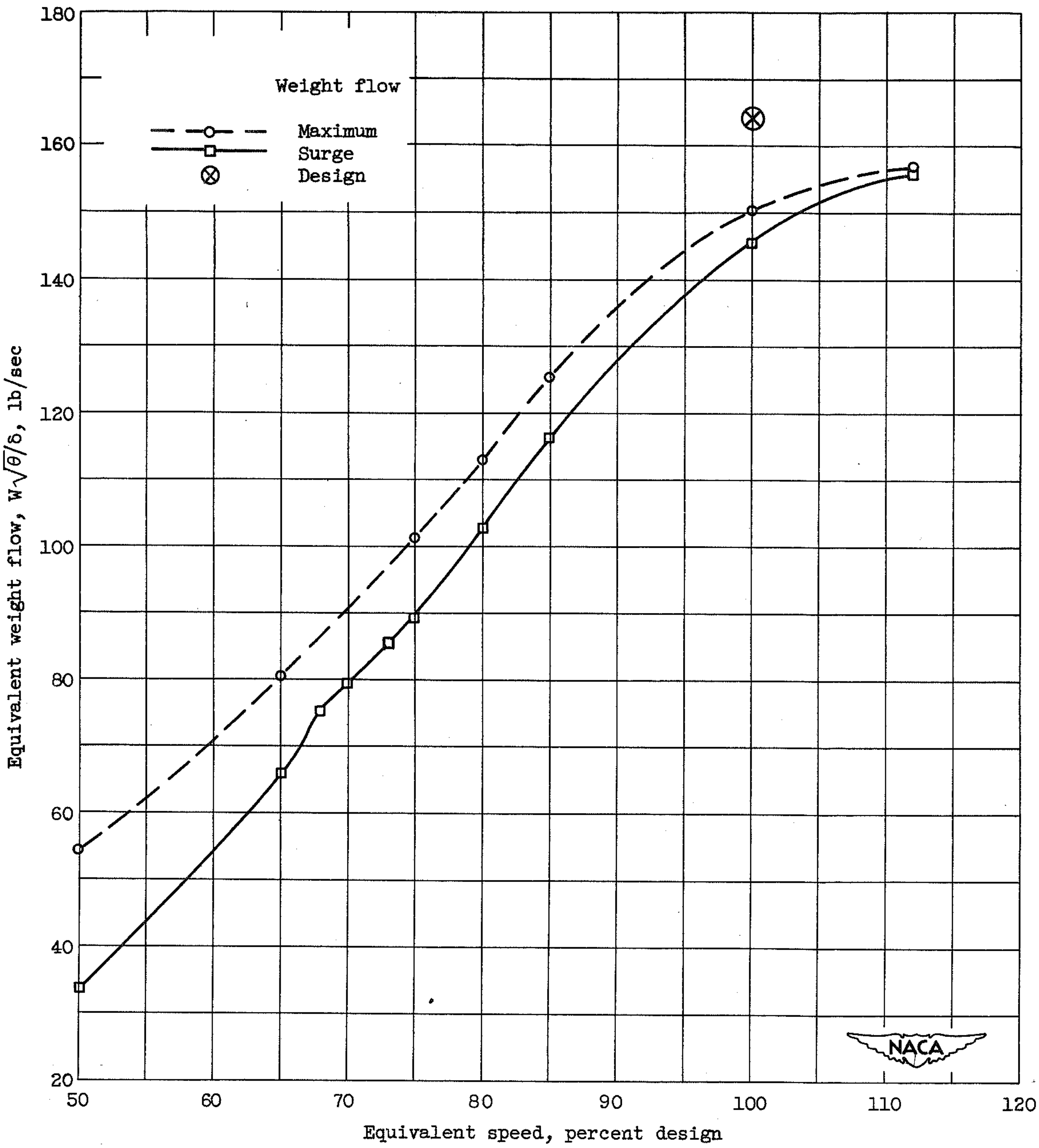


Figure 3. - Variation of maximum and surge weight flow with compressor speed.

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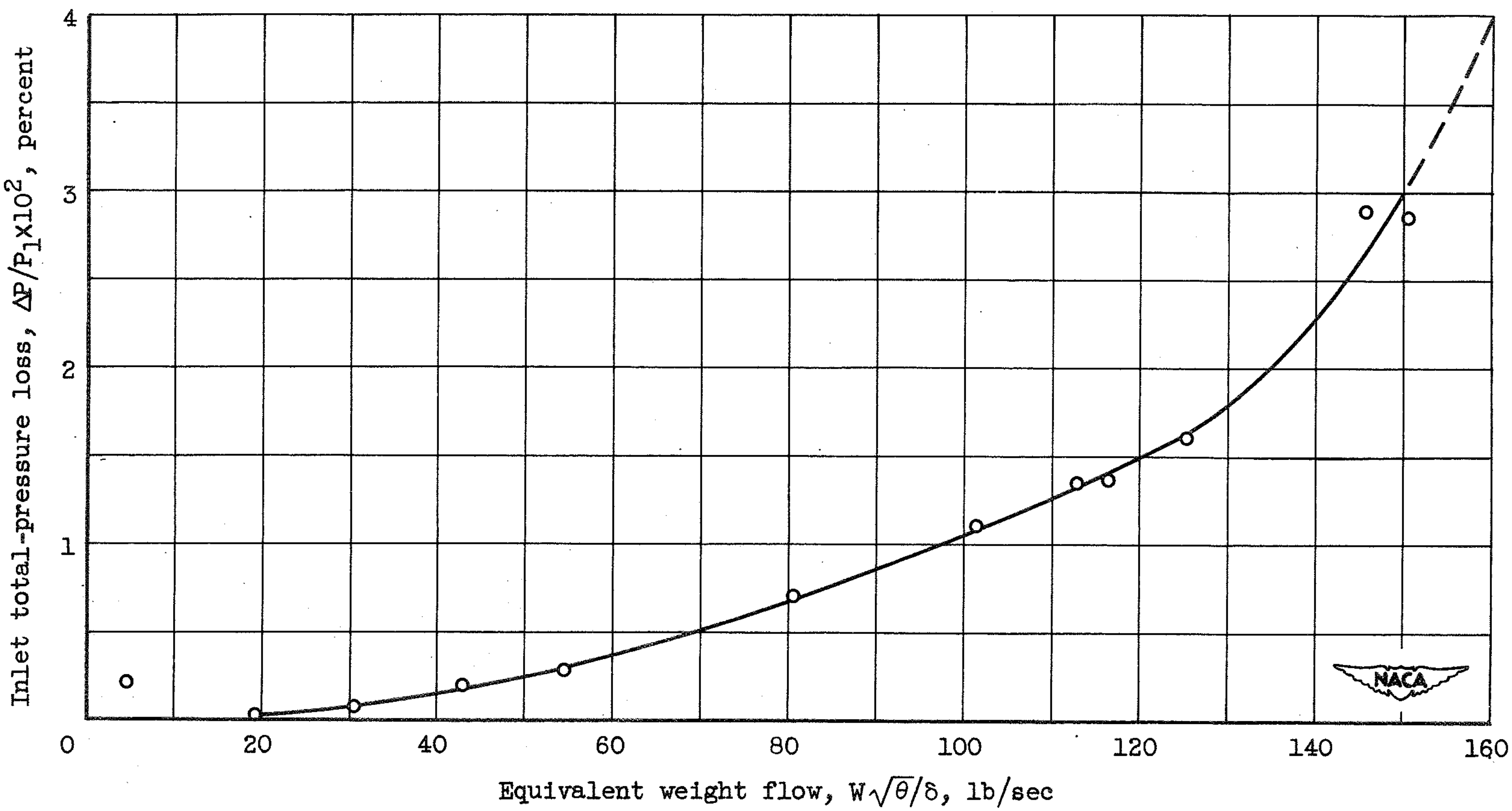

Figure 4. - Variation of pressure loss in dual inlet duct with equivalent weight flow.

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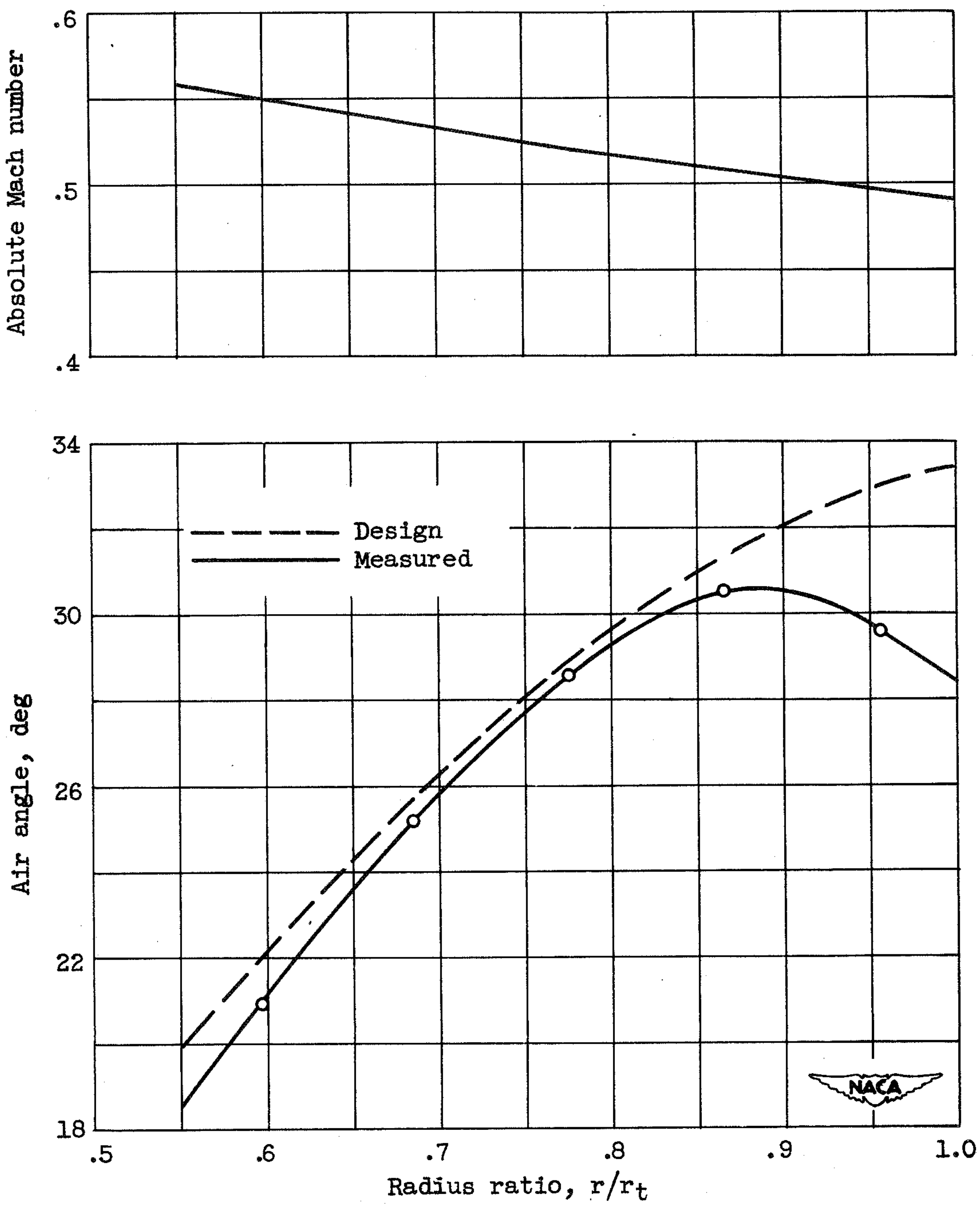

Figure 5. - Design and measured air angles and Mach numbers leaving inlet guide vanes for equilibrium design guide vanes.

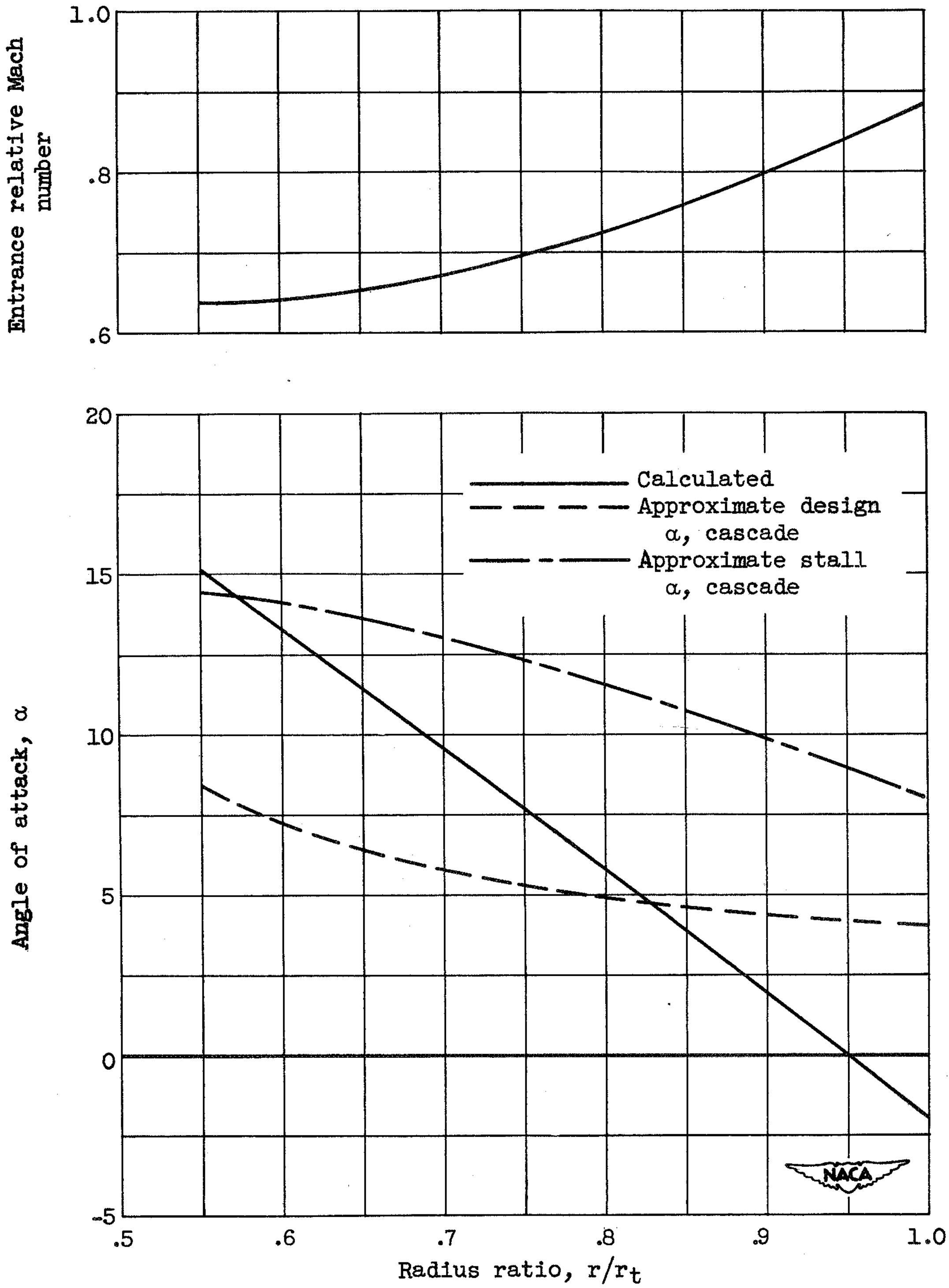

Figure 6. - Variation of first-stage-rotor flow parameters with radius for equilibrium design guide vanes at equivalent flow of 149 pounds per second and design speed.

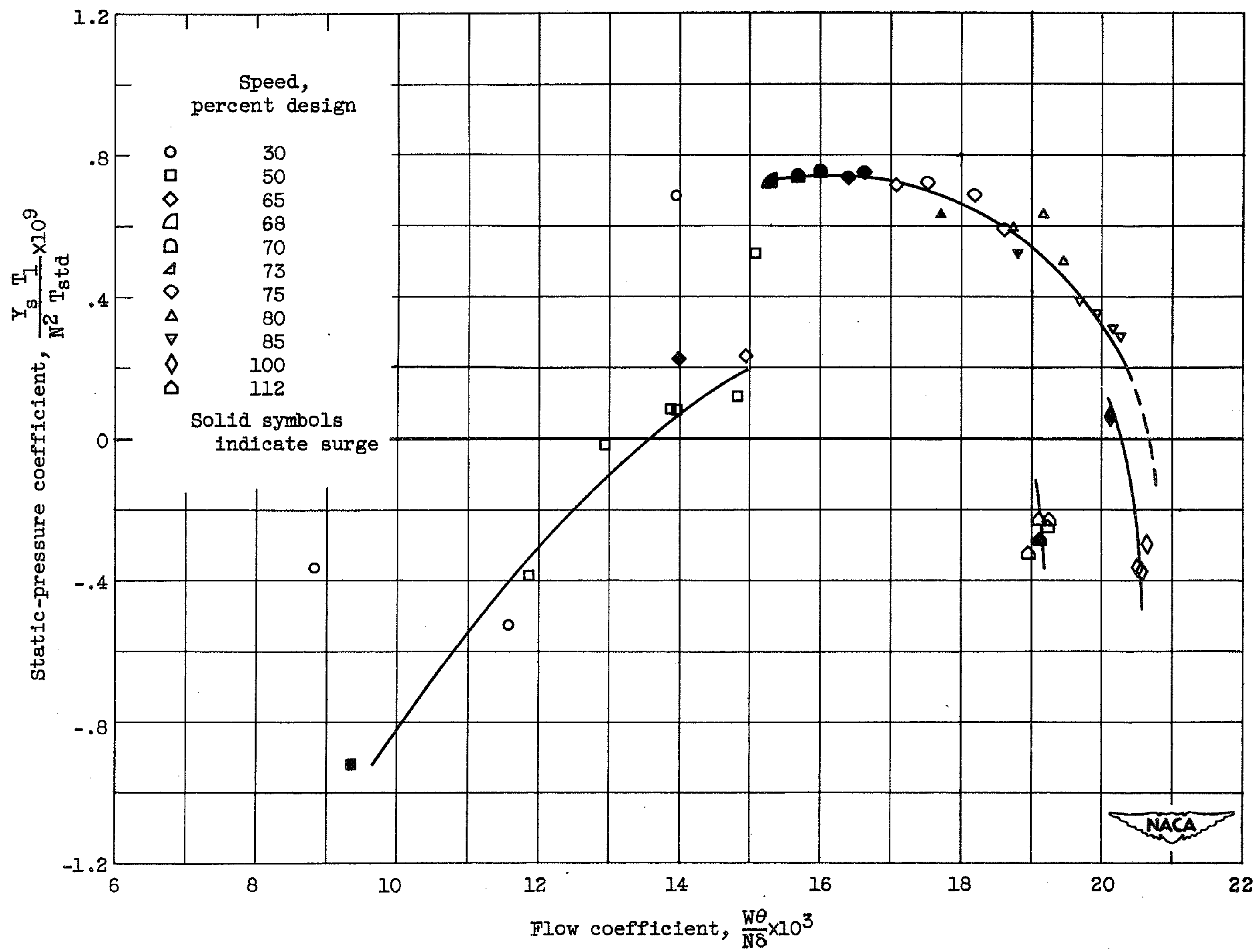

Figure 7. - Performance of first-stage rotor of J40-WE-24 13-stage compressor.

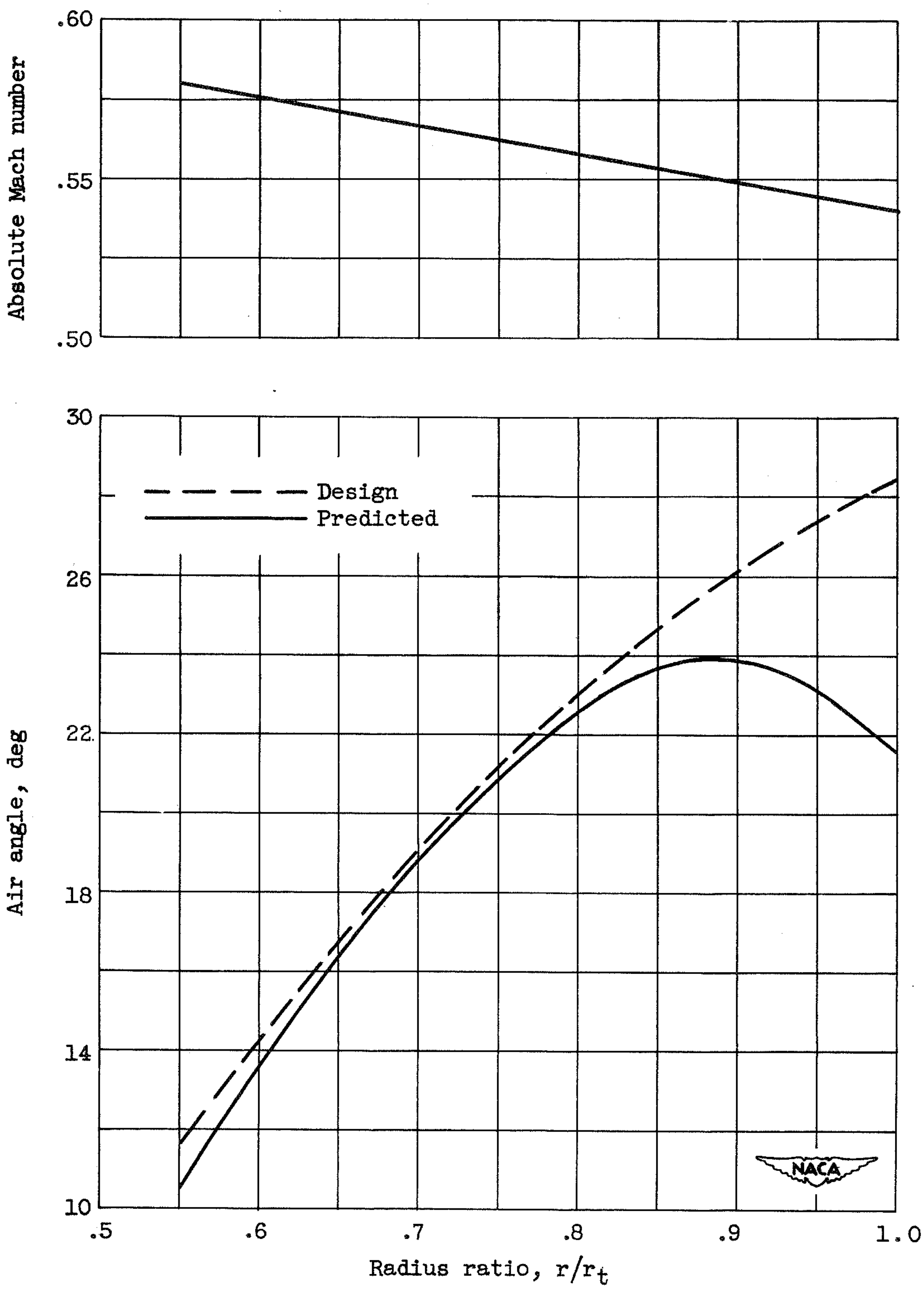

Figure 8. - Design and predicted air angles and Mach numbers leaving inlet guide vanes for modified equilibrium design guide vanes at design weight flow of 164 pounds per second.

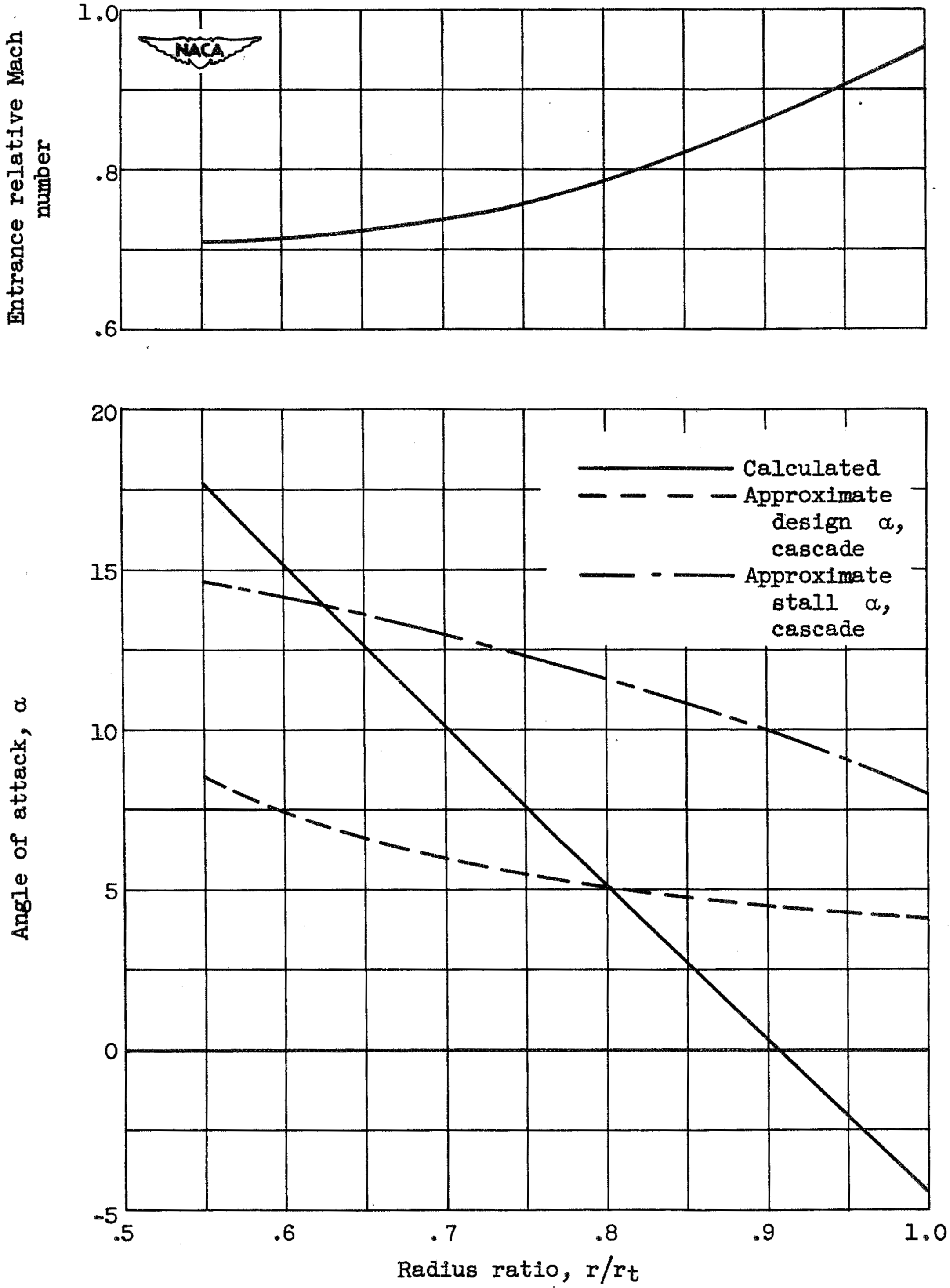

Figure 9. - Variation of first-stage-rotor flow parameters with radius for modified equilibrium design guide vanes at design equivalent flow of 164 pounds per second and design speed.

RESEARCH MEMORANDUM

for the

Bureau of Aeronautics, Department of the Navy

PERFORMANCE OF A 13-STAGE DEVELOPMENT COMPRESSOR FOR THE J40-WE-24

ENGINE AT EQUIVALENT SPEEDS FROM 30 TO 112 PERCENT OF DESIGN

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Engines, Turbojet	3.1.3
Compressors - Axial Flow	3.6.1.1

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Abstract

The performance of a 13-stage development compressor for the J40-WE-24 engine has been determined at equivalent speeds from 30 to 112 percent of design. The design total-pressure ratio of 6.0 and the design weight flow of 164 pounds per second were not attained. An analysis was conducted to determine the reasons for the poor performance at the design and over-design speed. The analysis indicated that most of the difficulty could be attributed to the fact that the first stage was overcompromised to favor part-speed performance.