https://ntrs.nasa.gov/search.jsp?R=19930088203 2020-06-17T07:51:27+00:00Z

NACA RM E54C22

NACA

RESEARCH MEMORANDUM

ANALYSIS OF FACTORS AFFECTING SELECTION AND DESIGN OF AIR-COOLED SINGLE-STAGE TURBINES FOR TURBOJET ENGINES

I - TURBINE PERFORMANCE AND ENGINE WEIGHT-FLOW CAPACITY

By Richard J. Rossbach, Wilson B. Schramm, and James E. Hubbartt

Lewis Flight Propulsion Laboratory
Cleveland, Ohio

CLASSIFICATION CHANGED

To UNCLASSIFIED
NACA Research abstract
By authority of RN-126 Date Effective Apr. 15, 1958
Amt 5-8-58

LIBRARY COPY

MAY 27 1954

LANGLEY AERONAUTICAL LABORATORY
LIBRARY, NACA
LANGLEY FIELD, VIRGINIA

CLASSIFIED DOCUMENT

This material contains information affecting the National Defense of the United States within the meaning of the espionage laws, Title 18, U.S.C., Secs. 793 and 794, the transmission or revelation of which in any manner to an unauthorized person is prohibited by law.

NATIONAL ADVISORY COMMITTEE FOR AERONAUTICS

WASHINGTON

May 25, 1954

NACA RM E54C22

NASA Technical Library
3 1176 01435 3891

NATIONAL ADVISORY COMMITTEE FOR AERONAUTICS

RESEARCH MEMORANDUM

ANALYSIS OF FACTORS AFFECTING SELECTION AND DESIGN OF AIR-COOLED SINGLE-STAGE TURBINES FOR TURBOJET ENGINES

I - TURBINE PERFORMANCE AND ENGINE WEIGHT-FLOW CAPACITY

By Richard J. Rossbach, Wilson B. Schramm, and James E. Hubbartt

3171

CB-1

SUMMARY

An analysis was made to determine the compressor pressure ratios and specific weight flows obtainable with single-stage air-cooled turbines for both nonafterburning and afterburning turbojet engines operating at a flight Mach number of 2 in the stratosphere. Wide ranges in turbine-inlet temperature, tip speed, hub-tip radius ratio, and coolant-flow ratio were considered. In the interest of exploring outer limits as well as minimizing cooling requirements, the turbines were designed for the maximum feasible specific work through the use of high turbine aerodynamic limits and high turbine-exit whirl. Other components were correspondingly considered to operate with high aerodynamic limits comensurate with the assigned efficiencies.

The results of this analysis show that the increases in compressor pressure ratio obtainable with a single-stage air-cooled turbine are quite large, if the turbine tip speed, hub-tip radius ratio, inlet temperature, and exit whirl are increased. Increases in the compressor pressure ratio are generally reflected in increases in the gas density and, therefore, the weight-flow capacity per unit of flow area of each engine component downstream of the compressor. For the afterburning engines with the afterburner-inlet velocity employed, the engine weight-flow capacity is limited by the afterburner for essentially all conditions considered. In this case, if higher afterburner limits are permissible, large increases in engine weight flow are possible. For the nonafterburning engine, in general, either the combustor or the turbine limits the engine weight-flow capacity. At low turbine hub-tip radius ratios corresponding to low compressor pressure ratios and high turbine flow areas, the combustor limits the engine weight-flow capacity for the combustor velocities employed. For these low hub-tip radius ratios, slight increases in the engine weight flow are possible if higher combustor velocities are permissible. As the turbine hub-tip radius ratio increases, the weight-flow capacity of the combustor increases directly with the compressor pressure ratio. The changes in turbine weight-flow capacity with turbine hub-tip radius ratio depend upon the turbine blade root stresses and the turbine-inlet temperature. However, as the turbine

[redacted]

hub-tip radius ratio is increased, the weight-flow capacity of the combustor relative to that of the turbine increases so that at high turbine hub-tip radius ratios the turbine limits the nonafterburning engine weight-flow capacity. For high turbine-inlet temperatures, the turbine weight-flow capacity can be increased as the turbine hub-tip radius ratio is increased without increasing the blade root stresses, since the gas density at the turbine increases faster than the flow area decreases. For low turbine-inlet temperatures, the converse is true; and, hence, the turbine blade root stresses must be increased in order to increase the turbine weight-flow capacity. Thus, high turbine-inlet tempera- 3171 tures, which are desired for obtaining the best combination of specific thrust and specific fuel consumption at high flight speeds, are desirable for attaining high compressor pressure ratios and high weight flows per unit of frontal area with single-stage turbines.

INTRODUCTION

As contemplated flight conditions become progressively more severe, as research and development produce engine components of successively higher performance, and as new analytical techniques continue to be evolved, new turbojet-engine performance analyses are required. The objective of the present report is to obtain values of compressor pressure ratio and compressor corrected weight flow per unit of turbine frontal area (hereafter referred to as "turbine-limited specific weight flow") in a turbojet engine having a single-stage air-cooled turbine with high aerodynamic limits. As in all analytical studies of this type, the ultimate goal is the choice of basic engine design specifications for a particular mission and application that result in high thrust per unit engine frontal area and low specific fuel consumption with acceptable engine specific weight. The possibilities of improving component configurations for a particular engine application are increased as freedom in the selection of realistic engine design specifications is extended. In this respect, turbine cooling provides an important contribution to design freedom, since it permits independent choice of blade stresses and gas temperatures.

Several analyses of the performance of turbojet engines over ranges of engine design specifications have been published. References 1 to 3, which are typical of these publications, present the thrust per pound of compressor weight flow and the specific fuel consumption of a turbojet engine for ranges of flight speed, compressor pressure ratio, turbine-inlet temperature, and afterburner-outlet temperature. This type of analysis was extended in reference 4 to include the effect of altitude on thrust and specific fuel consumption and to employ the engine performance in an analysis for the range of a specific aircraft. However, in all these cases the effect of cooling of the turbine to permit the higher turbine-inlet temperatures considered was neglected. The general effect of cooling would be to distort the results toward lower values of thrust and higher values of specific fuel consumption.

A limited number of calculations to evaluate the effects of cooling have been made. References 5 to 7 show that gas-turbine performance can be greatly improved by increasing the turbine-inlet temperature and employing either air or a liquid for cooling the turbine blades. In addition, references 8 and 9 indicate that the performance of turbojet engines designed for subsonic and for supersonic flight, respectively, can be improved by increasing the turbine-inlet temperature and employing air-cooled turbine blades.

3171

The thrust per unit of frontal area and the specific fuel consumption for turbojet engines analyzed in the literature have been evaluated by independently specifying the compressor pressure ratio, the corrected engine weight flow per unit frontal area, the turbine-inlet temperature, the component efficiencies, and the coolant flow. Under these conditions, the flow Mach numbers within each component are uniquely specified for given component flow areas. These Mach numbers are, in turn, interrelated with the component efficiencies. Furthermore, the cooling requirements are interrelated with the Mach numbers and the state conditions within the turbine as well as the turbine blade stresses and geometry. Thus, the independent choice of these specifications frequently leads to designs for which any or all of the following may be true: It is impossible to pass the specified weight flow; the required Mach numbers differ from those commensurate with the specified component efficiencies; or the cooling requirements are inaccurate. Consequently, the performance results of many of the engines are not directly comparable.

CB-1 back

The freedom of choosing independent design specifications so that all engines considered in any specific analysis are directly comparable has necessarily awaited the development of adequate procedures for analyzing and limiting each of the components. A simplified procedure for evaluating the turbine specific work for a given combination of turbine design specifications, namely turbine-inlet temperature, tip speed, and hub-tip radius ratio, was published recently (ref. 10). In addition, recent component research permits good estimates of the component efficiencies and aerodynamic limits. Since the engine weight flow and the compressor pressure ratio can be related to the turbine work, the component efficiencies, and the aerodynamic limits, while the coolant-flow requirements can be evaluated from the turbine design specifications, it is possible to relate the engine performance to the turbine-inlet temperature, the tip speed, and the hub-tip radius ratio for specified component efficiencies and aerodynamic limits. All engines analyzed in this way and for which turbine blade cooling can be accomplished are directly comparable, provided that the component efficiencies and aerodynamic limits are comparable for all engines. Consequently, a series of investigations has been undertaken by the NACA to determine the performance of turbojet

engines over a wide range of turbine-inlet temperatures, tip speeds, and hub-tip radius ratios for several values of coolant flow. In the analysis of these engines, an attempt was made to obtain the highest turbine specific work that seemed feasible in the near future with a given set of turbine design specifications, in order to explore the outer limits for engines employing turbine blade cooling. This was done by choosing higher values of turbine aerodynamic limits and turbine-exit whirl than are in current use. However, a rather conservative value of turbine adiabatic efficiency was assigned. Such a procedure results in the minimum cooling load, because a specified amount of work is obtained from 3171
the turbine for the lowest values of turbine-inlet temperature and stress. The engines investigated were made comparable by utilizing the same values of the component aerodynamic limits and efficiencies commensurate with those limits for all engines. In addition, air-cooling is considered, because it is in a more advanced stage of development than is liquid cooling; and a single-stage turbine is employed in the interest of retaining mechanical simplicity and minimum blade heat-transfer surface area.

As a part of this series of investigations at the NACA Lewis laboratory, the obtainable turbine work and weight flow expressed in terms of the compressor pressure ratio and the turbine-limited specific weight flow (hereafter referred to as the turbine performance as opposed to the engine performance, which is the objective of the entire investigation) were determined over a range of turbine design specifications. In addition, the weight-flow capacity of the inlet diffuser and the compressor and the ratio of the required combustor and afterburner frontal areas to the turbine frontal area were computed, so that the compressor corrected weight flow per unit of engine frontal area could be evaluated readily. The present report presents the results of this analysis and illustrates not only the influence of turbine blade cooling on the obtainable compressor pressure ratio, turbine-limited specific weight flow, and engine specific weight flow, but also the factors affecting the selection and design of air-cooled single-stage turbines for turbojet engines. The following ranges in the turbine design specifications were considered for a flight Mach number of 2.0 in the stratosphere: turbine-inlet temperature, 2000° to 3500° R; turbine blade tip speed, 1100 to 1700 feet per second; turbine hub-tip radius ratio, 0.55 to 0.75; and coolant-flow ratio, 0 to 0.15.

ANALYTICAL PROCEDURES

The engines analyzed are considered to consist of an inlet diffuser, a compressor, a combustor, and a turbine with a zone downstream in which the cooling air is assumed to mix with the combustion gases. For the case of thrust augmentation, the engines have, in addition, a diffuser after the mixing zone downstream of the turbine and an afterburner.

Schematic diagrams of the unaugmented and the augmented engines are shown in figure 1. The symbols and definitions used throughout the report are listed in appendix A.

Turbine Specific Work

3171

The turbine-inlet temperature, turbine blade tip speed, and turbine hub-tip radius ratio were used as independent variables for specifying the engine designs. In addition, single-stage turbine designs with free-vortex velocity distributions and constant annular area were employed. Since the turbine design conditions were specified, it was necessary to start the analysis at the turbine. The simplified procedure for designing free-vortex turbines as published in reference 10 was used for evaluating the turbine specific work. As shown in reference 10, the turbine specific work is dependent upon the aerodynamic limits, namely, rotor-inlet relative Mach number and exit axial critical velocity ratio (ratio of exit axial velocity to sonic velocity at Mach number 1), turbine adiabatic efficiency, and amount of whirl or tangential velocity at turbine exit, as well as turbine-inlet temperature, tip speed, and hub-tip radius ratio.

In the present report the aerodynamic limits and the turbine-exit whirl were chosen so as to assure high turbine work. High turbine work was particularly emphasized, since, for a given compressor pressure ratio and engine weight flow, the coolant flow required can be reduced by maintaining the turbine specific work at reduced turbine tip speed. This reduction in the coolant flow is simply a reflection of the decrease in turbine blade stresses. Alternately, the compressor pressure ratio and, consequently, the engine weight flow can be increased by increasing the turbine specific work without increasing the required coolant flow by increasing the aerodynamic limits, or exit whirl, or both. In addition, the turbine size or blade stress, or both, can be reduced for a given compressor pressure ratio and engine weight flow by increasing the turbine aerodynamic limits or exit whirl, or both.

Reference 11 shows that the specific work of a single-stage turbine can be increased substantially by increasing the turbine aerodynamic limits. In addition, it is apparent from Euler's turbine equation that the turbine specific work can be directly increased by employing turbine-exit whirl, that is, by permitting the gases leaving the turbine to have a tangential component of velocity in the direction opposite that of the wheel rotation. Therefore, the present report utilized a combination of the aerodynamic limits and turbine-exit whirl so that the highest turbine specific work was obtained that seems feasible for near-future single-stage turbine designs without a significant reduction in turbine efficiency. Accordingly, the inlet relative Mach number at the turbine rotor root was limited to 0.80 and the turbine adiabatic efficiency was fixed at 0.83.

Appendix B presents the results of calculations made to estimate the permissible amount of turbine-exit whirl without a significant reduction in the turbine adiabatic efficiency. It is assumed that the pressure head associated with the exit tangential velocity component was eventually dissipated downstream of the turbine by the tail-cone struts, and deleterious effects on the downstream components were thus prevented. On the basis of the results presented in appendix B, the present report used turbine-exit whirl corresponding to a value of -0.41 for the ratio of whirl component of velocity to wheel speed at the radial position at which the product of whirl energy loss per pound and total weight flow expresses the total whirl energy loss. This turbine-exit whirl was determined as that value corresponding to the same turbine adiabatic efficiency as for no turbine-exit whirl, assuming that the blading losses were independent of turbine-exit whirl. However, as shown in appendix B, even if the blading losses are scheduled to vary with turbine-exit whirl when a loss coefficient invariant with exit whirl is used, the maximum drop in the turbine adiabatic efficiency due to this whirl is only 3.6 percentage points. Furthermore, supplementary calculations revealed that the effect of employing this exit whirl is to increase the obtainable compressor pressure ratio and turbine weight-flow capacity by about 34 and 20 percent, respectively, for a typical example. For these reasons, the use of this amount of turbine-exit whirl seemed particularly desirable, even though it is considerably in excess of values used in present design practice.

3171

The turbine-exit axial critical velocity ratio must be limited by the condition of axial choking within the blade passages at the trailing edge. For a given blade solidity and trailing-edge thickness, it is possible to relate the blade chord to the exit axial critical velocity ratio (outside the blade passage) for this choking condition. For this reason, the minimum values of permissible turbine blade chord have been computed over the range of turbine design specifications used herein for an exit axial critical velocity ratio of 0.70, a trailing-edge thickness reasonable for a cooled turbine blade, and two turbine blade tip solidities. These calculations are discussed and the results are presented in appendix B. These results show that the minimum turbine blade chords are within or below present design practice. Slight increases in the exit axial critical velocity ratio have an appreciable effect upon the maximum turbine blade chord (since as this velocity ratio approaches unity the minimum turbine blade chord approaches infinity) with no significant improvement in the turbine performance. For this reason, the exit critical velocity ratio of 0.70 was considered as the limit.

The angle through which the turbine blades turn the working fluid is a direct measure of the turbine work. It is expected that this turning angle must be limited at the hub to prevent excessive losses. Therefore, since the need for high turbine work has been particularly emphasized by using a combination of the aerodynamic limits and exit whirl corresponding to high turbine specific work, a calculation was made (see appendix B) to determine the hub values of the rotor blade turning angles over the range of turbine designs considered herein. These results as tabulated in appendix B show that only in the case of the combination of turbine-inlet temperature and tip speed of 2000° R and 1700 feet per second, respectively, do the turning angles exceed the limiting value of 120° used in reference 10.

3171

The occurrence of negative reaction over any portion of the turbine blade span is undesirable, because it may cause flow separation that would result in additional losses. Of the 80 combinations of the turbine design specifications studied herein, only two displayed this undesirable characteristic; these were for a tip speed of 1100 feet per second, a turbine-inlet temperature of 3500° R, and for hub-tip radius ratios of 0.55 and 0.60.

Coolant-Flow Ratio

In order to avoid restricting the analysis to any particular air-cooled blade configurations, a range of coolant flows from 0 to 15 percent of the compressor weight flow was considered. However, since the turbine design specifications fix the turbine blade stresses for any blade configuration as well as the aerodynamic and thermodynamic conditions around the blade, the cooling-air requirements can be uniquely evaluated for any blade configuration. The work done by the turbine upon the cooling air as it flows between the entrance of the compressor and the tips of the turbine blades was considered equivalent to the work required to compress the cooling air to the compressor-discharge stagnation temperature. Although this assumption permitted handling the cooling air as if it were compressed entirely by the compressor, in the practical application it involves compressor interstage bleed, since, in general, work is done on the cooling air as it flows within the turbine rotor.

Compressor Pressure Ratio

The compressor pressure ratio for a given adiabatic efficiency is dependent upon the compressor specific work and compressor-inlet temperature. Since the work balance between the turbine and compressor provides a relation between compressor specific work and turbine specific work, it is possible to relate the compressor pressure ratio to the turbine specific work, the coolant-flow ratio, the fuel-air ratio, and the compressor-inlet temperature. This relation was employed in conjunction with the analytical procedures and results presented in reference 12 for evaluating the compressor pressure ratio.

Turbine Specific Weight Flow

The turbine specific weight flow was determined by applying the continuity equation to the exit of the turbine and using the limiting value of the turbine-exit axial critical velocity ratio and the state conditions at the mean radial station. The state conditions were determined from the velocities at the turbine exit, the compressor, combustor, and turbine pressure ratios, and the turbine-inlet temperature. The combustor pressure ratio and the turbine specific weight flow were obtained 3171
from a graphical solution of the empirical pressure-loss characteristic of a typical low-loss combustor and the continuity equation written between the combustor-inlet and turbine-exit stations. Finally, the turbine-limited specific weight flow was computed as the turbine weight flow per unit of turbine frontal area corrected for fuel and coolant flows and for compressor-inlet stagnation conditions.

Compressor Specific-Weight-Flow Capacity

The compressor specific-weight-flow capacity was estimated for a range of compressor tip speeds for a transonic inlet stage employing solid-body-rotation guide vanes. The compressor was aerodynamically limited by the use of an inlet axial critical velocity ratio at the hub of 0.7 and a relative Mach number at the tip of 1.1. These were considered to be realistic for near-future compressors designed for high weight flow and specific work. The weight-flow capacity was computed at the station between the first guide vane and the first rotor (where the pressure is the lowest while the whirl due to the guide vane is high) with the continuity equation at the mean-radial position.

Combustor and Afterburner Performance

In order to keep the engine frontal area low and the combustor velocity (and therefore the combustor pressure drop) low, the combustor frontal area was maintained equal to the turbine frontal area, unless the required combustor reference velocity exceeded the assumed limiting value of 150 feet per second. (Reference velocity is defined as that velocity obtained from the continuity equation when the burner weight flow is divided by the inlet density and the maximum flow area.) For the high-volume flows for which the limiting velocity tended to be exceeded, the combustor frontal area was allowed to exceed the turbine frontal area, while the combustor reference velocity of 150 feet per second was maintained. For the augmented engines an afterburner reference velocity of 500 feet per second was employed, and the frontal area ratio between the turbine and the afterburner required to accommodate the weight flow was determined from the continuity relation. The afterburner reference velocity was never reduced below 500 feet per second, in order to minimize the diffusion losses at the turbine exit.

The pressure ratio of the combustor was varied with the inlet Mach number and the temperature ratio according to the experimental loss characteristics of the typical low-loss turbojet combustor studied in reference 13. The pressure losses in the afterburner were assumed to be equal to the inlet dynamic head as in reference 14.

Mixing and Diffusing at Turbine Exit

The combustion gases and the cooling air discharged from the turbine blades were assumed to mix in a constant-area section downstream of the turbine (see fig. 1). The pressure losses and the final mixture temperature were determined by using the momentum and energy equations and assuming that the combustion gases and the cooling air enter the mixing zone at the same static pressure and that no heat transfer to the cooling air occurs before mixing.

For the augmented engine, an additional pressure loss was accounted for which was due to the necessity of diffusing from the cooling-air mixing zone to the afterburner-inlet velocity. This pressure loss was computed with an adiabatic diffuser efficiency.

All assumptions involved in the analysis and the important constants are summarized in appendix C.

VARIATION OF TURBINE PERFORMANCE WITH TURBINE DESIGN VARIABLES

In the foregoing, analytical procedures are outlined which permit the determination of the turbine performance and the engine weight-flow capacity obtainable from turbojet engines with single-stage air-cooled turbines and components with high aerodynamic limits commensurate with the component efficiencies chosen. In the interest of minimizing the required coolant flow for the single-stage turbine, an attempt was made to consider turbine designs in which the highest feasible specific work could be extracted from the turbine. This was accomplished herein by employing a large tangential component of the turbine efflux velocity, which was shown to result in a net gain in the obtainable compressor pressure ratio and turbine-limited specific weight flow. The compressor pressure ratio, the turbine-limited specific weight flow, and the state conditions through the engine are presented in table I for the following values of the independent variables: turbine-inlet temperature T_4', 2000°, 2500°, 3000°, and 3500° R; blade tip speed U_t, 1100, 1300, 1500, and 1700 feet per second; hub-tip radius ratio r_h/r_t, 0.55, 0.60, 0.65, 0.70, and 0.75; coolant-flow ratio C, 0, 0.05, 0.10, and 0.15.

Uncooled-Turbine Performance

3171

Effect of turbine design specifications on turbine performance. - The effect of variations in the turbine design specifications on the performance of the uncooled turbine is presented in figure 2, where the turbine-limited specific weight flow is plotted against the compressor pressure ratio. Figure 2 illustrates that, in general, the compressor pressure ratio and the turbine-limited specific weight flow increase simultaneously as the turbine-inlet temperature and the turbine tip speed are increased. The increase in the turbine-limited specific weight flow is essentially a reflection of the increase in the compressor pressure ratio and, therefore, the gas density at the turbine exit. The simultaneous increase is particularly important, since the turbine-limited specific weight flow and compressor pressure ratio of the engine are measures of the corresponding turbojet-engine thrust and specific fuel consumption, respectively. At least within the ranges encountered with the single-stage turbine, it is generally true that increases in the engine specific weight flow and compressor pressure ratio at constant turbine-inlet temperature result in increased thrust at lower specific fuel consumption. The increases in the turbine-limited specific weight flow and compressor pressure ratio with turbine-inlet temperature are largest at high turbine tip speeds. Similarly, both of these quantities increase most with turbine tip speed at the highest turbine-inlet temperature; while, for low turbine-inlet temperature, the compressor pressure ratio increases while the turbine-limited specific weight flow first increases and then decreases with increasing tip speed. Thus, both high turbine-inlet temperatures and turbine tip speeds seem particularly advantageous. The higher tip speed is accompanied by a direct increase in the whirl component of the gases at the turbine inlet and, thus, an increase in turbine work. Similarly, a higher turbine-inlet temperature yields an increase in the turbine work in the form of an increased inlet whirl component. In addition, an increase in the turbine-inlet temperature permits this increase in turbine work to occur while the turbine pressure ratio p_7'/p_4' actually increases and therefore, in general, affects favorably the turbine weight-flow capacity. The variations in the turbine-limited specific weight flow and the compressor pressure ratio with the turbine-inlet temperature and tip speed are essentially a reflection of these characteristics.

In addition, figure 2 shows that, for a given turbine-inlet temperature and tip speed, the turbine-limited specific weight flow increases with decreasing turbine hub-tip radius ratio (increasing blade length and, thus, annular area), until a maximum weight flow is reached. Thereafter, the turbine-limited specific weight flow decreases with decreasing turbine hub-tip radius ratio. Although this peak weight flow is obtained only for the highest turbine-inlet temperature and tip speeds used herein, the trend is general and would occur for all conditions if the turbine hub-tip radius ratios were extended to lower values. The compressor pressure ratio decreases continuously with decreasing turbine hub-tip radius ratio, however, since decreasing hub wheel speed requires decreased absolute stream velocities at the turbine inlet for limited relative velocities. Thus, the gains in turbine-limited specific weight flow obtained by decreases in the turbine hub-tip radius ratio are accompanied by a decreasing compressor pressure ratio. Figure 2 illustrates that the percentage gains in the turbine-limited specific weight flow obtained by decreasing the turbine hub-tip radius ratio are largest for low turbine tip speeds and inlet temperatures. In addition, these gains are accomplished with the smallest percentage of loss in compressor pressure ratio.

3171

Increases in turbine tip speed afford additional design freedom. For instance, as shown on figure 2, at a turbine tip speed of 1100 feet per second the turbine performance is limited to a narrow range, with turbine hub-tip radius ratio being the most significant turbine-design variable; however, at a tip speed of 1700 feet per second the turbine performance can vary over a wide range, and design freedom is increased, particularly since a larger number of combinations of turbine-limited specific weight flow and pressure ratio are available. The reader should recall, however, that figure 2 depicts only the range of design freedom for the case of single-stage turbines with specified aerodynamic limits and that reducing the aerodynamic limits would shift the curves presented toward the origin, the result of which would be larger coolant-flow requirements, heavier turbines, and lower turbine performance for each set of values of the turbine design variables. The maps of the compressor pressure ratio and turbine-limited specific weight flow shown on figure 2 overlap for turbine tip speeds of 1500 and 1700 feet per second. This overlapping would occur more commonly if smaller speed increments and larger ranges of turbine-inlet temperature and hub-tip radius ratio were chosen. This overlapping illustrates that different combinations of turbine design variables may give the same turbine performance and that the evaluation of the best combination for a particular engine application depends on the combination of values of turbine design specifications that results in the least severe blade-cooling requirements for a particular cooled-blade configuration. In a similar manner, in order to determine the best combination of the compressor pressure ratio and turbine-limited specific weight flow over the full range of the turbine design specification for a particular airplane with a specified mission, it is necessary to make a complete engine analysis with a specific cooled-turbine blade configuration. Subsequently, the results would have to be applied to an aircraft-range or combat-time investigation.

CB-2 back

<u>Comparison of turbine performance results with those of production engines</u>. - In order to give the reader a basis for evaluating the turbine

designs used in the present report, a comparison is made between the turbine performance results presented herein and the turbine performance of three production engines with single-stage turbines. In order to make this comparison, the turbine performances of three analytical engines predicted upon the assumptions of this report were determined at values of turbine-inlet temperature and tip speed that resulted in the same corrected turbine-inlet temperature and corrected tip speed for both the analytical engines and the corresponding production engines. The hub-tip radius ratios for the analytical engines and the corresponding production engines were equal. The results of this comparison are presented in the following table: 3171

Turbojet engine	Corrected design specifications			Turbine design performance			
				Production engine		Analytical engine	
	Turbine-inlet temperature, °R	Turbine tip speed, ft/sec	Hub-tip radius ratio	Compressor pressure ratio	Specific weight flow, $\frac{(lb/sec)}{sq\ ft}$	Compressor pressure ratio	Specific weight flow, $\frac{(lb/sec)}{sq\ ft}$
A	2010	1358	0.680	4.25	22.8	6.165	27.82
B	2100	1168	.798	5.25	14.7	5.925	19.54
C	2060	1193	.785	5.00	16.1	5.915	20.40

The turbine performance of the analytical engines exceeds that for the corresponding production engines. This difference is essentially a result of the higher aerodynamic limits and turbine-exit whirl and, correspondingly, the higher turbine work, rather than the effect of slight differences in efficiencies. Comparison of the turbine performance on figure 2 with the design compressor pressure ratios and weight flows for the three turbojet engines indicates that, by extending the ranges of the turbine design specifications beyond present limitations, much higher turbine-limited specific weight flows and compressor pressure ratios can be obtained with a single-stage turbine.

Cooled-Turbine Performance

Since the turbine cooling air must be supplied by the compressor and the same cooling air absorbs rather than produces work in the turbine, the compressor pressure ratio and the turbine-limited specific weight flow are affected by the amount of cooling air supplied to the turbine.

Effect of coolant-flow ratio on compressor pressure ratio. - The effect of the coolant-flow ratio on the compressor pressure ratio is presented in figure 3. It is evident from the figure that the pressure ratio of the cooled engine is independent of the values of the turbine design specifications. The highest value of compressor pressure ratio shown in the figure for the cooled turbine is about 2 percent below that for no coolant flow for each percentage point of coolant flow. At a pressure ratio of 1.0, cooling has no effect, because the coolant
3171 pumping work is zero.

Effect of coolant-flow ratio on turbine performance. - The effect of coolant-flow ratio on the variation of turbine performance with turbine design specifications is presented in figure 4. If the compressor pressure ratio were invariant with coolant-flow ratio for any combination of the turbine design specifications, an increase in the coolant-flow ratio would increase the turbine-limited specific weight flow, because this weight flow is computed at the exit to the turbine annulus and the coolant bypasses the turbine annulus. However, as shown in figure 3, the coolant-flow ratio decreases the compressor pressure ratio, because the coolant requires work for its compression but supplies no work in the turbine. At the lower tip speeds, the effect of the coolant that bypasses the turbine annulus is greater than the effect of the reduced weight flow through the turbine. This weight-flow reduction is brought about by the decrease in density at the turbine exit due to the decreasing compressor pressure ratio. The result is that the turbine-limited specific weight flow increases with coolant-flow ratio at the lower tip speeds. At the higher tip speeds, the opposite set of circumstances causes the turbine-limited specific weight flow to decrease with an increase in coolant-flow ratio. As a result, an aerodynamically limited compressor must have a larger frontal area in the cooled than in the uncooled case if the turbine tip speed is 1100 feet per second; the converse is true at 1700 feet per second.

Figure 4 shows that the range of turbine performance is reduced by cooling as a result of the larger influence of cooling at the higher turbine speeds compared with the lower tip speeds. However, even for coolant-flow ratios of 0.15, the ranges in both compressor pressure ratio and turbine-limited specific weight flow are appreciably above the possible design values for uncooled turbines. With reference to the preceding table, for example, a turbine designed for an inlet temperature of 2000° R, a tip speed of 1300 feet per second, and a hub-tip radius ratio of 0.70 might be considered typical of current design practice for an uncooled turbine. For these design conditions, the turbine-limited specific weight flow is 20.4 pounds per second per square foot and the compressor pressure ratio is 3.96, as given in figure 4(a) ($C = 0$). This set of turbine performance values is designated by the circles in figure 4. For all points to the right and above these circles, the turbine-limited specific weight flow and the compressor pressure ratio are superior to the values for a typical current design. Since the engines with superior turbine performance in figure 4(a) and all the engines in figures 4(b), (c), and (d) are possible through turbine blade cooling, it is seen that this cooling, which permits the more severe turbine design conditions, is a means of realizing significant gains in single-stage turbine performance. Even with 15 percent of the compressor weight flow used for cooling, the turbine-limited specific weight flow can be increased to as much as 33 pounds per second per square foot, or the compressor pressure ratio can be increased to approximately 7.5 by increasing the turbine-inlet temperature to 3500° R (fig. 4(d)). Although these increases indicate improvements in engine thrust and specific fuel consumption, they can be accurately evaluated only by means of a complete analysis including the engine flight plan and the specific type of cooled-blade configuration.

3171

TURBINE BLADE HUB STRESSES RESULTING AT VARIOUS TURBINE PERFORMANCE LEVELS

It is impossible to evaluate completely or adequately the required coolant-flow ratio over the range of turbine design variables considered herein without making a cooling analysis of several air-cooled blade configurations. However, it is possible to gain some knowledge concerning the severity of the cooling problem over the range of the turbine design variables by studying the resulting turbine blade centrifugal stresses at the hub. Since the centrifugal stresses are a function of the blade tip speed and hub-tip radius ratio, lines of constant centrifugal stress can be superimposed upon turbine performance plots with the turbine tip speed and hub-tip radius ratio as parameters. For constant turbine-inlet temperatures, a constant-stress line can be considered as an approximation of the line of constant required coolant-flow ratio for a particular cooled-blade configuration. The reason a constant-stress line only approximates the condition of constant required coolant-flow ratio is that several of the variables affecting the required coolant-flow ratio vary along a constant-stress line. However, the effects of these variables, which include the blade inside and outside heat-transfer coefficients, the effective gas temperature, the blade pressure drop, and the compressor pressure ratio, are to some extent compensating.

The stresses corresponding to the performance levels attainable at turbine-inlet temperatures of 2000° and 3500° R are presented in figure 5 for coolant-flow ratios of 0 and 0.15. The stresses were computed on the basis of a linear metal area taper from blade hub to tip with the tip metal area being 0.4 that at the hub. This amount of taper was chosen since it can be obtained with present fabrication procedures for convection-air-cooled blade configurations.

[redacted]

A significant characteristic of the constant-stress lines is their slope. For the lower turbine-inlet temperature, the stress lines are so inclined that for constant stress the turbine-limited specific weight flow decreases with increasing compressor pressure ratio as shown in figure 5(a). However, for the high turbine-inlet temperature, the turbine-limited specific weight flow increases with compressor pressure ratio at constant stress as shown in figure 5(b). In general, for the range of turbine design variables considered herein, the engine thrust increases with the turbine-limited specific weight flow, and the specific fuel consumption decreases with increasing compressor pressure ratio. For this reason it is desirable to increase compressor weight-flow rate and pressure ratio simultaneously. At 3500° R turbine-inlet temperature, the lines of constant stress possess this desirable characteristic. If the coolant flow is assumed to be constant along a constant-stress line, it is evident from figure 5(b) that the thrust and specific fuel consumption can be improved by increasing the turbine-limited specific weight flow and compressor pressure ratios along this line without increasing the coolant required. At 2000° R turbine-inlet temperature, this characteristic of simultaneous increases in the turbine-limited specific weight flow and compressor pressure ratio is not displayed by the constant-stress lines. In fact, at constant blade stress, indications are that the obtainable thrust increases at the expense of increasing specific fuel consumption corresponding to increases in turbine-limited specific weight flow and decreases in compressor pressure ratio. Therefore, in order to obtain any knowledge concerning the best operating point along a constant-stress line, it is necessary to weigh the increase in thrust due to higher specific weight flows against the increase in specific fuel consumption. In order to increase both compressor pressure ratio and turbine-limited specific weight flow at 2000° R turbine-inlet temperature, it is necessary to increase the blade stresses. For the 2500° and 3000° R turbine-inlet temperatures (not shown) the slopes of the constant-stress lines are between those for the two extreme turbine-inlet temperatures shown.

Because the turbine blade hub stresses are dependent upon both the turbine tip speed and hub-tip radius ratio, the lines of constant stress must cross the lines of constant tip speed and the lines of constant hub-tip radius ratio as shown in figure 5. For 3500° R, the characteristic increase in the turbine-limited specific weight flow and compressor pressure ratio along the constant-stress line is accompanied by an increase in the turbine tip speed and hub-tip radius ratio. Thus, in order to exploit the gains in engine performance associated with increases in both compressor pressure ratio and turbine-limited specific weight flow along a constant-stress line, it appears that it would be desirable to design the turbine with short blades (high hub-tip radius ratio) and high tip speeds rather than to use long blades and low tip speeds. However, for 2000° R turbine-inlet temperature, the long turbine blades or low turbine hub-tip radius ratios may possibly be desirable.

At 3500° R turbine-inlet temperature, the three lowest constant-stress lines for zero coolant-flow ratio cross over each of the corresponding stress lines for 0.15 coolant-flow ratio. This cross-over indicates that, for a limited range of turbine design variables, the cooled turbines can drive compressors of higher weight flow for the same compressor pressure ratio and turbine blade hub stress than can the uncooled turbine. However, the cooled turbine cannot drive the higher-weight-flow compressor without employing a higher turbine specific work, which adversely affects the engine performance. For most of the values of the turbine design variables at 3500° R and for the complete range at 3171
2000° R, the cooled-blade hub stresses are always higher than those for the uncooled blade with the same turbine-limited specific weight flow and compressor pressure ratio.

INFLUENCE OF ENGINE COMPONENTS ON SELECTION OF TURBINE DESIGN SPECIFICATIONS

In the foregoing discussion, those factors have been emphasized that affect the selection of a set of values of turbine design variables which would result in a light-weight, air-cooled turbine with minimum blade heat-transfer area. For this purpose the turbine-limited specific weight flow, which is based upon the turbine frontal area, was employed as a criterion of merit. In the case in which the weight of the turbine is of secondary importance in comparison with the engine frontal area, the engine specific weight flow is the most important criterion of merit. The reference area in this parameter is the frontal area of the largest engine component, including the inlet diffuser, the compressor, the combustor, the turbine, and, in the case of the afterburning engine, the afterburner.

For some of the engine components, the maximum specific weight flow was computed, while, for the remainder, the ratio of the component area to the turbine frontal area required to pass the necessary weight flow was computed. The maximum specific weight flow of the compressor as a function of the compressor tip speed is presented in figure 6, in which wheel-type spanwise velocity distribution is assumed for the inlet guide vanes. A value of the inlet diffuser specific weight flow of 34.4 pounds per second per square foot was obtained by using a ram recovery of 0.85 and a flight Mach number in the stratosphere of 2.0. The primary combustor is larger than the turbine for many combinations of turbine design variables, while the afterburner is nearly always larger than the turbine.

Limitations on Turbine Design Specifications Due to Component Frontal Areas

In order to illustrate the manner in which the frontal areas of the several engine components exercise control over the choice of a desirable

3171

CB-3

set of values of turbine design variables, it was assumed to be desirable for all the engine components to have the same frontal area. From data presented in table I, however, it is possible to determine the effect of the several component frontal areas on the choice of turbine design specifications for arbitrary ratios of frontal areas of the components. In figure 7 the turbine-limited specific weight flow of the uncooled engine is plotted against the compressor pressure ratio for all values of turbine tip speed and hub-tip radius ratio and for turbine-inlet temperatures of 2000° and 3500° R. Superimposed on these plots are the boundaries beyond which the inlet diffuser (ram recovery, η_R, 0.85), the compressor, the combustor, or the afterburner must be larger than the turbine in frontal area in order that the pertinent component accommodate the turbine specific weight flow without exceeding the imposed limits. The engine specific weight flow is the same as the turbine-limited specific weight flow for all conditions for which the turbine frontal area is equal to or larger than that of any of the other components.

The regions of figure 7 in which the inlet diffuser and the compressor limit the engine specific weight flow are quite small. In fact, for the uncooled engine, only at the highest values of turbine-inlet temperature and tip speed do the inlet diffuser and the compressor have to be larger than the rest of the engine. Since, at these same conditions for the cooled engine, the turbine-limited specific weight flow is sufficiently lower than for the uncooled engine (fig. 4), the cooled engine is not limited by either the inlet diffuser or the compressor. Therefore, for the range of design variables and coolant-flow ratio considered herein, the weight-flow capacity of the engine is essentially unlimited by the inlet diffuser and the compressor.

Figure 7 indicates that the combustor limits the engine specific weight flow at low values of hub-tip radius ratio over the full range of the variables. At both values of turbine-inlet temperature and at a tip speed of 1100 feet per second, the turbine and combustor frontal areas are equal at a hub-tip radius ratio of about 0.70. If the tip speed is changed to 1700 feet per second, the hub-tip radius ratio of the boundary at which the combustor and turbine are equal in size varies from 0.59 to 0.67 as the turbine-inlet temperature varies from 2000° to 3500° R. These boundaries are located at slightly lower values of hub-tip radius ratio for cooled turbines having the same design conditions as the corresponding uncooled turbines. The reason for the shift in this boundary toward lower hub-tip radius ratios for the cooled case is that the coolant flow bypasses the combustor. With reference to the constant-stress lines in figure 5(b), the advantage of high turbine hub-tip radius ratios has been indicated for high turbine-inlet temperatures. Since, for the high values of the hub-tip radius ratio in the case of the nonafterburning engine, the turbine is the largest engine component, the turbine-limited specific weight flow is the same as engine specific weight flow. Thus, for high hub-tip radius ratios and high turbine-inlet temperatures, improvements in engine specific weight flow are a direct result of improvements in turbine-limited specific weight flow. For low turbine-inlet temperatures, however, the hub-tip radius ratio chosen for a particular design must be the result of a compromise. The inclination of the constant-stress lines for this condition in figure 5(a) is such that high turbine-limited specific weight flows are obtained at low values of hub-tip radius ratio; but figure 7(a) indicates that the combustor becomes larger than the turbine before the hub-tip radius ratio has decreased significantly. In addition, the constant-stress lines of figure 5(a) indicate that the turbine-limited specific weight flow is increased at the expense of compressor pressure ratios. Thus, at the low temperatures, little design freedom is available, and it may be necessary to choose a value of hub-tip radius ratio that requires a combustor having a frontal area considerably greater than that of the turbine. 3171

The engine specific weight flow is limited by the afterburner frontal area for nearly all conditions studied. Only for moderate and high tip speeds and for a turbine-inlet temperature of 2000° R is the turbine larger than the afterburner (fig. 7). For cooled turbines, the afterburner is larger than all components for all conditions studied, because the effect of coolant flow is to decrease the compressor pressure ratio as shown in figure 4. This decrease in pressure ratio lowers the density level in all the components, including the afterburner, and, therefore, causes an increase in the afterburner frontal area over the uncooled case. Thus, it appears that an afterburner suffers the inherent disadvantage of permitting only low engine specific weight flow. As a result, the engine thrust capacity per unit of frontal area is limited by the afterburner size. For this type of engine, the principal advantage of increasing the turbine-limited specific weight flow is to reduce both turbine weight and cooling requirements, as is extremely desirable.

Effects of Increasing Combustor and Afterburner Reference Velocity Limits

It is apparent from figure 7 that the two engine components that most seriously limit the engine specific weight flow are the combustor and the afterburner. Therefore, sample calculations were made to determine the effect on engine specific weight flow of increasing both combustor and afterburner reference velocities, if the limits on these velocities can be lifted as a result of research and development. The results for the uncooled nonafterburning and afterburning engines are presented in figures 8 and 9, respectively. For convenience in evaluating the results, the turbine-limited specific weight flow presented previously in figure 4 is shown by the dashed curves. In addition, the required combustor and afterburner reference velocities and the resulting combustor and afterburner pressure ratios corresponding to the turbine-limited specific weight flows are shown.

3171

CB-3 back

Nonafterburning engines. - Curve I for the specific weight flow in figures 8(a) and (b) was determined by dividing the turbine-limited specific weight flows of the dashed curve by the ratio of the combustor to the turbine frontal areas. The peaks on curve I of these figures correspond to the points on the boundary in figure 7 beyond which the combustor is larger than the turbine for the pertinent values of the variables. It should be noted that, although the turbine-limited specific weight flow may rise as the hub-tip radius ratio falls below that value at which the turbine and combustor have the same size, the engine specific weight flow decreases sharply, while the blade centrifugal stress increases and the compressor pressure ratio decreases. The decrease in compressor pressure ratio usually results in an increase in specific fuel consumption. The engine specific weight flow of curve I has no effect on the combustor reference velocity or pressure ratio, so that the dashed curve and curve I are identical in the lower two parts of figures 8(a) and (b).

However, for curve II, the combustor reference velocity was increased wherever necessary (middle figs. 8(a) and (b)) so as to permit a combustor frontal area equal to that of the turbine. This resulted in a decreased combustor pressure ratio as shown in the bottom sections of figure 8. It is evident from figure 8 for the case of the uncooled turbine that no increase in engine specific weight flow results from increases in the combustor reference velocity for a turbine-inlet temperature of 3500° R. At a turbine-inlet temperature of 2000° R, the increase in engine specific weight flow amounts to a maximum of 9.4 percent over the value obtainable for the case in which the combustor reference velocity is limited to 150 feet per second and the combustor and turbine have the same frontal area. The reason no significant increases in engine specific weight flow are possible when the reference velocity limit is increased is that the combustor pressure ratio falls with increased reference velocity, as is shown in figure 8.

The effect of 15-percent coolant flow on the engine specific weight flow, based on the conditions used in curve I, was computed. The engine specific weight flow increased with coolant flow, since the coolant bypassed the combustor, which limits the flow. The effect of this coolant-flow ratio for the conditions of curve II varied with tip speed. At the high tip speed the coolant reduced the engine specific weight flow, and at the low tip speeds the coolant flow raised the engine specific weight flow (for the same reasons given in the discussion of fig. 4).

Afterburning engines. - Curve I for the specific weight flow in figures 9(a) and (b) was determined by dividing the turbine-limited specific weight flows of the dashed curve by the ratio of the combustor to the turbine frontal areas or the ratio of the afterburner to the turbine frontal areas, whichever was greater. Thus, the peak on curve I for a turbine-inlet temperature of 2000° R and a tip speed of 1500 feet per second (fig. 9(b)) represents one point on the boundary in figure 7 beyond which the afterburner is larger than the turbine for the pertinent values of the variables. As in the case of the nonafterburning engine, the engine specific weight flow decreases sharply as the hub-tip radius ratio falls below that value at which the turbine and afterburner have the same size. Also, when engine frontal area is important, no advantage is gained by the choice of a value of hub-tip radius ratio less than that value for maximum engine specific weight flow, because of the deleterious effects on blade centrifugal stress, engine specific weight flow, and compressor pressure ratio. 3171

In general, significant gains in engine specific weight flow are indicated in curve II for the case when both the afterburner and the combustor reference velocities are increased beyond the limiting values. However, it should be noted that the afterburner reference velocities become extremely high. Curve II in the top sections of both figures 8 and 9 has the same value at each corresponding point, because the turbine-limited specific weight flow is determined at the turbine exit, and, therefore, the afterburner cannot control it but must accommodate it. For the case of 2000° R turbine-inlet temperature, the afterburner-inlet Mach number is so high that the exit Mach number of 1 is reached before the combustion-gas temperature has reached 3500° R. For these cases, the fact that the afterburner is thermally choked is indicated in figure 9.

Since curve II has the same values in the corresponding upper parts of figures 8 and 9, the effect of the coolant-flow ratio is the same as noted in the discussion of figure 8. With respect to curve I, however, the effect of increasing the coolant-flow ratio is to make the afterburner progressively larger than the rest of the engine, because the coolant flow reduces the compressor pressure ratio and because the afterburner must accommodate the cooling air.

CONCLUSIONS

The following conclusions are drawn from a study of nonafterburning and afterburning turbojet engines operating at a Mach number of 2 in the stratosphere and employing air-cooled single-stage turbines with high aerodynamic limits and exit whirl:

1. Values of the turbine-inlet temperature and tip speed higher than used in current design practice are particularly advantageous, since they permit appreciably higher values of compressor pressure ratio and turbine-limited specific weight flow than are currently obtainable.

2. Increasing the coolant-flow ratio reduces the obtainable compressor pressure ratio for all values of turbine design specifications by an amount depending on the uncooled pressure ratio and the coolant-flow ratio.

3. At the highest value of turbine-inlet temperature studied (3500° R), the obtainable compressor pressure ratio and turbine-limited specific weight flow can be increased simultaneously without increasing the blade centrifugal stress. However, at the lowest value of turbine-inlet temperature studied (2000° R), the obtainable compressor pressure ratio and turbine-limited specific weight flow cannot be increased simultaneously without increasing the blade centrifugal stress.

3171

4. The nonafterburning engine specific weight flow is limited by the combustor for the lower range of turbine hub-tip radius ratios studied; however, the afterburning engine specific weight flow is limited by the afterburner for essentially all turbine hub-tip radius ratios studied. Increases in the turbine-limited specific weight flow for turbine hub-tip radius ratios below the values for which either the combustor or afterburner limits the engine specific weight flow are accompanied by increases in turbine blade centrifugal stresses and decreases in engine specific weight flow and specific fuel consumption.

5. No significant increases in the nonafterburning engine specific weight flow are possible by increasing the combustor reference velocity; however, the afterburning engine specific weight flow can be increased substantially if the afterburner reference velocity is increased, except in those cases in which thermal choking occurs at the afterburner outlet.

Lewis Flight Propulsion Laboratory
National Advisory Committee for Aeronautics
Cleveland, Ohio, March 22, 1954

APPENDIX A

DEFINITIONS AND SYMBOLS

The following terms, which were employed in the text in the interest of brevity, are defined below for the convenience of the reader:

Compressor specific weight flow	Corrected weight flow per unit compressor frontal area that an aerodynamically limited compressor can pass. 3171
Engine specific weight flow	Corrected weight flow of compressor divided by area of largest engine component among the following: inlet diffuser, compressor, combustor, turbine, and afterburner (where applicable).
Inlet diffuser specific weight flow	Corrected weight flow of diffuser divided by diffuser frontal area.
Turbine design specifications	Turbine-inlet temperature, blade tip speed, and blade hub-tip radius ratio.
Turbine-limited specific weight flow	Corrected weight flow through compressor as limited by turbine, divided by turbine frontal area.
Turbine performance	Compressor pressure ratio and turbine-limited specific weight flow obtainable with single-stage aerodynamically limited turbine.
Turbine specific weight flow	Weight-flow capacity of turbine divided by turbine frontal area.

The following symbols are used in this report:

A	frontal area of any engine component, sq ft
a^*	critical velocity of sound, ft/sec
C	ratio of cooling air to compressor mass flow
f	fuel-air ratio
g	acceleration due to gravity, 32.174 ft/sec^2
J	mechanical equivalent of heat, 778.164 ft-lb/Btu

K loss factor

L loss in turbine except for whirl loss, Btu/lb

M_R Mach number relative to rotor at inlet

p' stagnation pressure, lb/sq ft

3171

r radius, ft

T' stagnation temperature, $^{\circ}R$

U blade speed, ft/sec

V velocity, ft/sec

V_u tangential component of velocity, ft/sec

V_x axial component of velocity, ft/sec

W velocity relative to blade, ft/sec

w weight flow, lb/sec

δ ratio of stagnation to standard pressure, $p'/2116$

η efficiency

η_R ram recovery

η_T' turbine adiabatic efficiency for zero exit whirl

θ ratio of stagnation to standard temperature, $T'/518.4$

σ turbine blade hub centrifugal stress, lb/sq in.

τ ratio of blade tip to root cross-sectional area

Subscripts:

AB afterburner

B combustor

C compressor

h hub or inner radius

i	radial position at which tangential velocity head times mass flow equals total exit-whirl loss
T	turbine
t	tip
1	diffuser inlet
2	compressor inlet
3	combustor inlet
4	turbine stator inlet
5	turbine rotor inlet
6	turbine exit
7	mixing-zone inlet
8	diffuser inlet
9	afterburner inlet
10	afterburner exit
∞	free stream

3171

APPENDIX B

TURBINE-EXIT WHIRL

Turbine Efficiency with Turbine-Exit Whirl

For a specific set of values for the turbine design variables, the variation in turbine-exit whirl has little effect upon the inlet Mach number vector diagrams along the blade span, because the turbines analyzed are designed for an assigned inlet relative Mach number at the hub; but the whirl does increase the blade turning angles and the exit Mach numbers, both relative and absolute. According to Ainley and Mathieson (ref. 15), when exit relative Mach numbers are increased between that exit Mach number at which sonic local velocities occur on some portion of the blade and an exit relative Mach number of 1, the profile loss may either rise or fall. Therefore, since for the present analysis turbine-exit whirl can have little effect on the Mach numbers at the turbine inlet, and since experimental results of reference 15 seem to indicate that with proper design the profile loss might be held constant with turbine-exit whirl, calculations were made to determine the turbine adiabatic efficiency over a wide range of the turbine-exit whirl parameter for blading losses invariant with exit whirl. The energy associated with the exit whirl was considered dissipated and chargeable to the adiabatic efficiency. The results of these calculations for a zero-whirl adiabatic efficiency of 0.83 and for two values of the inlet whirl parameter are presented in figure 10. As is evident from the figure, initial increments in the exit-whirl parameter result in a greater percentage increase in the turbine specific work output than in the total losses charged to the turbine, while later increments in the exit-whirl parameter have the opposite result. These circumstances are indicated by the successive rising and falling of adiabatic efficiency with increasing values of exit-whirl parameter. For a value of the exit-whirl parameter $V_{u,6,i}/U_i$ of -0.410, the adiabatic efficiency with exit whirl is equal to the value without exit whirl, regardless of the value of inlet-whirl parameter $V_{u,5,i}/U_i$. This point is designated by the break-even point in the figure.

Since reference 15 left the question open as to whether, in a specific instance, the profile loss would rise or fall with increasing values of the exit whirl, further computations were undertaken in an attempt to explore the effects of losses on efficiency. In these computations, the blading losses were expressed by $L = (K/gJ)\Sigma W^2$ as in reference 16, in which the constant of proportionality was considered invariant with exit whirl. The adiabatic efficiency was then computed for zero exit whirl with an assumed efficiency of 0.83 for a value of the exit whirl parameter $V_{u,6,i}/U_i$ of -0.41. The results are presented in the following table:

Turbine tip speed, U_t, ft/sec	Turbine-inlet temperature, T_4', $^{\circ}R$	Turbine hub-tip radius ratio, r_h/r_t	Turbine adiabatic efficiency for zero whirl, η_T'
1100	2000	0.55 .75	0.846 .849
	3500	0.55 .75	0.840 .841
1700	2000	0.55 .75	0.866 .869
	3500	0.55 .75	0.852 .854

3171

The zero-exit-whirl efficiency computed in this way increases with increased tip speed and decreased temperature, and has maximum and minimum values of 0.869 and 0.840. Thus, for the complete range of turbine design variables considered herein, the variation of the zero-exit-whirl efficiency from that for $V_{u,6,i}/U_i = -0.41$ is between 1.0 and 3.9 percentage points for a blading loss varied with the exit velocity relative to the rotor.

Experimental results presented in references 15 and 17 give the effects of exit relative Mach number, Reynolds number, and turning angle upon the profile loss coefficient. From these results it may be concluded that the profile loss coefficient increases with increased turning angle but that the coefficient decreases with increasing Reynolds number and exit relative Mach number. An increase in the exit-whirl parameter corresponds to an increase in the turning angle, because the specific work is increased. In addition, the Reynolds number and the relative Mach number increase because of the increased velocity and the increased density due to the higher specific work and compressor pressure ratio. An inspection of the data referenced indicates that, for the analysis of the present report, the increases in the Reynolds and exit relative Mach numbers influence the loss coefficient considerably more than the increased turning angle, with the result that the profile loss coefficient would decrease as the exit whirl is changed from zero to some prescribed amount. At least it seems conservative to assume a constant profile loss coefficient.

Minimum Turbine Blade Chords for Axial Choking

The turbine rotor reaches an axial choking condition when the axial Mach number immediately inside the rotor blade passage at the trailing edge is 1. The corresponding exit axial Mach number (outside the passage) is less than 1 because of the area change due to the trailing-edge blockage. The blockage is dependent upon the ratio of the blade trailing-edge thickness to the spacing, which can in turn be related to the chord, trailing-edge thickness, and solidity. Reference 18 shows that, for typical current turbine rotor blade designs operating at limiting Mach number, the exit axial critical velocity ratio is approximately 0.70. However, since the results of reference 18 are limited to turbines with small amounts of turbine-exit whirl, which influences the turbine-exit density and therefore the change in axial Mach number as the gases leave the turbine, further calculations were made to justify the use of this value of axial critical velocity ratio for the amount of whirl used herein. Since the blockage is dependent upon the chord, the minimum permissible chords for $V_{u,6,i}/U_1 = -0.410$ and the extremes of the combinations of turbine design variables were computed with an assumed exit axial critical velocity ratio of 0.7 and a trailing-edge thickness of 0.070 inch. This trailing-edge thickness is a reasonable value for cooling the trailing edge. Tip solidities of 1.1 and 1.5 were used to cover the range expected for future designs. The results of these computations are as follows:

Turbine tip speed, U_t, ft/sec	Turbine-inlet temperature, T_4', °R	Turbine hub-tip radius ratio, r_h/r_t	Minimum turbine rotor blade chord, in.	
			Solidity = 1.1	Solidity = 1.5
1100	2000	0.55	0.8956	1.221
		.75	.9381	1.279
	3500	0.55	0.8341	1.137
		.75	.8477	1.156
1700	2000	0.55	1.156	1.576
		.75	1.264	1.723
	3500	0.55	0.9670	1.319
		.75	1.017	1.386

The minimum chords are between 0.8341 and 1.264 inches for a tip solidity of 1.1 and between 1.137 and 1.723 inches for a tip solidity of 1.5. These chords are lower than most present designs for turbojet engines. In addition, it is shown in reference 19 that increasing the blade chord lengths is desirable for high turbine-inlet temperatures in order to reduce the coolant-flow requirements.

Rotor Blade Turning Angles

The rotor blade turning angles were computed at the rotor blade hub (where the greatest amount of turning occurs) for the extreme combinations of the turbine design variables to determine whether excessive turning occurs. The calculations were made from the turbine design velocity vector diagrams with a procedure analogous to that outlined in appendix E of reference 10. The results are presented in the following table:

Turbine tip speed, U_t, ft/sec	Turbine-inlet temperature, T_4', $^{\circ}R$	Turbine hub-tip radius ratio, r_h/r_t	Rotor blade turning angle at hub, deg
1100	2000	0.55	89.6
		.75	98.2
	3500	0.55	73.8
		.75	79.8
1700	2000	0.55	119.4
		.75	132.4
	3500	0.55	99.3
		.75	109.8

3171

The maximum turning angle of 132.4° occurs at the lower turbine-inlet temperature. As the temperature is increased, the turning angle is decreased.

APPENDIX C

ASSUMPTIONS AND CONSTANTS

Inasmuch as the results and procedures of reference 10 were used in the present report, the following assumptions of that reference also apply to the turbine analysis herein:

3177

(1) Simplified radial equilibrium

(2) Free-vortex blading

(3) Use of density at mean radial position for computing weight flow

(4) Stagnation temperature invariant with radial position at turbine entrance

(5) Hub, mean, and tip radii at turbine exit equal to respective values at turbine entrance

(6) Constant value of 4/3 for ratio of specific heats

In addition to the foregoing, the following assumptions were employed in this turbine analysis:

(7) Adiabatic flow through turbine rotor

(8) Work done by turbine on cooling air between compressor inlet and tips of turbine blades equivalent to work required to compress cooling air to compressor-outlet stagnation temperature

(9) Mechanical friction and work required by accessories negligible

(10) Pressure head associated with tangential component of turbine absolute efflux velocity dissipated by struts that secure tail cone, and resulting loss charged to turbine adiabatic efficiency

(11) No effect of cooling-air efflux from turbine blade tips on turbine work or adiabatic efficiency (ref. 20 and unpub. data)

In calculating the maximum specific weight mass flow of the compressor, the following assumptions were imposed:

(12) Use of simplified radial equilibrium in analysis of flow at inlet to first compressor rotor (as recommended in ref. 21 for first approximation of conditions in an axial-flow turbomachine)

(13) Computation of weight flow through compressor inlet at mean radial position

(14) Use of solid-body-rotation guide vanes in compressor-inlet analysis 3171

The following additional assumptions were imposed on the pertinent components:

(15) Constant value of 4/3 for ratio of specific heats in combustor analysis

(16) Variation of combustor pressure ratio with inlet Mach number and temperature ratio according to experimental loss characteristics of typical low-loss turbojet combustor of reference 13

(17) Entrance of combustion gases and cooling air into zone downstream of turbine in which mixing occurred at same static pressure

(18) Constant flow area in mixing zone for combustion gases and cooling air

(19) Constant-area combustion section in afterburner, as in reference 14

(20) Friction and flame-holder losses in afterburner equal to inlet dynamic head, as in reference 14

3171

The following values of constants were employed herein:

Compressor-inlet hub-tip radius ratio	0.4
Compressor adiabatic efficiency	0.83
Compressor rotor-inlet relative Mach number at blade tip	1.1
Compressor-inlet axial critical velocity ratio at hub	0.7
Ratio of combustor inner radius to turbine tip radius	0.25
Combustion efficiency	0.95
Combustor reference velocity limit, ft/sec	150
Turbine adiabatic efficiency	0.83
Turbine-inlet relative Mach number at hub	0.8
Turbine-exit axial critical velocity ratio	0.7
Afterburner combustion efficiency	0.88
Afterburner-inlet velocity, ft/sec	500
Afterburner-exit stagnation temperature, $^{\circ}R$	3500
Afterburner diffuser efficiency	0.85
Afterburner flame-holder drag coefficient	1.0

REFERENCES

1. Cleveland Laboratory Staff: Performance and Ranges of Application of Various Types of Aircraft-Propulsion System. NACA TN 1349, 1947.

2. Schey, Oscar W.: The Advantages of High Inlet Temperature for Gas Turbines and Effectivenss of Various Methods of Cooling the Blades. Paper No. 48-A-105, presented at A.S.M.E. Meeting (New York), Nov. 28-Dec. 3, 1948.

3. Ellerbrock, Herman H., Jr., and Schafer, Louis J., Jr.: Application of Blade Cooling to Gas Turbines. NACA RM E50A04, 1950.

4. Lubarsky, Bernard: Performance and Load-Range Characteristics of Turbojet Engine in Transonic Speed Range. NACA TN 2088, 1950.

5. Shreeve, Charles A., Jr., and Martin, Harold R.: A Theoretical Study of the Thermodynamic Conditions in Gas Turbines Due to Blade Cooling by the Use of Air. Final Rep. June 1, 1947-July 31, 1948, Mech. Eng. Dept., Univ. Maryland, July 7, 1948. (Office Naval Res. Proj. NR 220007, Final Rep. N7onr-397, Task Order 1.)

6. Hawthorne, W. R.: A Theoretical Study of the Thermodynamic Conditions in Gas Turbines Due to Blade Cooling by Use of Liquid. High Temperature Gas Turbines with Cooled Blades, A Preliminary Survey of Two Simple Plants. Tech. Rep. No. 6574-4, Gas Turbine Lab., M.I.T., May 1949. (ONR Contract N5ori-78, Task Order 21, NR-220-010, Proj. DIC 6574.)

7. Brown, T. W. F.: Some Factors in the Use of High Temperatures in Gas Turbines. Proc. Inst. Mech. Eng., vol. 162, no. 2, 1950, pp. 167-179; discussion, pp. 179-185.

8. Schramm, Wilson B., Nachtigall, Alfred J., and Arne, Vernon L.: Preliminary Analysis of Effects of Air Cooling Turbine Blades on Turbojet-Engine Performance. NACA RM E50E22, 1950.

9. Gabriel, David S., Krebs, Richard P., Wilcox, E. Clinton, and Koutz, Stanley L.: Analysis of the Turbojet Engine for Propulsion of Supersonic Fighter Airplanes. NACA RM E52F17, 1953. 3171

10. Cavicchi, Richard H., and English, Robert E.: A Rapid Method for Use in Design of Turbines within Specified Aerodynamic Limits. NACA TN 2905, 1953.

11. English, Robert E., and Cavicchi, Richard H.: Possible Range of Design of One-Spool Turbojet Engines within Specified Turbine-Design Limits. Paper No. 53-S-33, A.S.M.E., 1953.

12. English, Robert E., and Wachtl, William W.: Charts of Thermodynamic Properties of Air and Combustion Products from 300° to 3500° R. NACA TN 2071, 1950.

13. Zettle, Eugene V., and Cook, William P.: Performance Investigation of Can-Type Combustor. I - Instrumentation, Altitude Operational Limits, and Combustion Efficiency. NACA RM E8F17, 1948.

14. Bohanon, H. R., and Wilcox, E. C.: Theoretical Investigation of Thrust Augmentation of Turbojet Engines by Tail-Pipe Burning. NACA RM E6L02, 1947.

15. Ainley, D. G., and Mathieson, G. C. R.: An Examination of the Flow and Pressure Losses in Blade Rows of Axial Flow Turbines. Rep. No. R.86, British N.G.T.E., Mar. 1951.

16. Hawthorne, W. R., and Walker, Antonia B.: The Effect of Blade Cooling on the Stage Efficiency of a Gas Turbine. Rep. No. 6574-2, Gas Turbine Lab., M.I.T., Mar. 1949. (ONR Contract N5ori-78, Task Order 21, NR-220-010, Proj. DIC 6574.)

17. Ainley, D. G.: Performance of Axial-Flow Turbines. War Emergency Issue No. 41 pub. by Inst. Mech. Eng. (London). (Reprinted in U.S. by A.S.M.E., April 1949, pp. 230-244.)

18. English, Robert E., Silvern, David H., and Davison, Elmer H.: Investigation of Turbines Suitable for Use in a Turbojet Engine with

High Compressor Pressure Ratio and Low Compressor-Tip Speed. I - Turbine-Design Requirements for Several Engine Operating Conditions. NACA RM E52A16, 1952.

19. Schramm, Wilson B., Arne, Vernon L., and Nachtigall, Alfred J.: Analytical Comparison of Turbine-Blade Cooling Systems Designed for a Turbojet Engine Operating at Supersonic Speed and High Altitude. II - Air-Cooling Systems. NACA RM E52J30, 1953.

20. Ainley, D. G.: An Experimental Single-Stage Air-Cooled Turbine. Pt. II - Research on the Performance of a Type of Internally-Air-Cooled Turbine Blade. Aircraft Eng., vol. XXV, no. 295, Sept. 1953, pp. 269-276.

21. Wu, Chung-Hua, and Wolfenstein, Lincoln: Application of Radial-Equilibrium Condition to Axial-Flow Compressor and Turbine Design. NACA Rep. 955, 1950. (Supersedes NACA TN 1795.)

3171

CB-5

TABLE I. - TABULATION OF RESULTS

(a) Turbine blade tip speed U_t, 1100 feet per second; afterburner reference velocity V_{AB}, 500 feet per second.

Coolant-flow ratio, C	Turbine hub-tip radius ratio, r_h/r_t	Combustor fuel-air ratio, f_B	Mixture fuel-air ratio, f_8	After-burner fuel-air ratio, f_{AB}	Combustor reference velocity, V_B, ft/sec	Combustor-turbine frontal-area ratio, A_B/A_T	Afterburner-turbine-frontal area ratio, A_{AB}/A_T	Compressor-exit temperature, T_3', °R	Mixture temperature, T_8', °R	Combustor pressure ratio, p_4'/p_3'	Turbine pressure ratio, p_7'/p_4'	Mixing-zone pressure ratio, p_8'/p_7'	After-burner diffuser pressure ratio, p_9'/p_8'	After-burner pressure ratio, p_{10}'/p_9'	Compressor pressure ratio, p_3'/p_2'	Turbine-limited specific weight flow, $w_c\sqrt{\theta_2}/A_T\delta_2$, (lb/sec)/sq ft
Turbine-inlet temperature T_4', 2000° R																
0	0.55	0.017	0.017	0.053	150	1.50	1.65	929	1805	0.884	0.606	0.999	0.963	0.916	2.27	22.8
	.60	.016	.016			1.36	1.50	955	1782	.890	.569			.914	2.48	21.6
	.65	.016	.016			1.19	1.35	981	1759	.895	.534			.911	2.70	20.3
	.70	.016	.016			1.03	1.19	1009	1754	.900	.497			.906	2.95	18.8
	.75	.015	.015		130	1.00	1.01	1036	1709	.929	.462	1.00		.905	3.21	16.7
0.05	0.55	0.017	0.016	0.053	150	1.48	1.72	919	1764	0.882	0.606	0.992	0.960	0.912	2.18	23.0
	.60	.017	.016			1.33	1.57	942	1743	.887	.569	.992	.960	.909	2.37	21.6
	.65	.016	.016			1.17	1.41	967	1723	.892	.534	.992	.960	.907	2.58	20.4
	.70	.016	.015			1.01	1.23	994	1700	.897	.497	.994	.961	.904	2.82	18.4
	.75	.016	.015		126	1.00	1.05	1020	1677	.930	.462	.994	.961	.900	3.08	16.8
0.10	0.55	0.017	0.016	0.053	150	1.48	1.78	909	1723	0.880	0.606	0.984	0.956	0.907	2.11	23.4
	.60	.017	.015			1.31	1.64	931	1704	.885	.569	.985	.957	.904	2.28	22.1
	.65	.016	.015			1.18	1.47	954	1685	.890	.534	.985	.957	.901	2.47	20.6
	.70	.016	.014			1.00	1.29	979	1664	.895	.497	.986	.957	.899	2.68	18.5
	.75	.016	.014	.054	126	1.00	1.10	1004	1643	.930	.462	.987	.958	.898	2.90	16.8
0.15	0.55	0.017	0.015	0.053	150	1.44	1.88	898	1680	0.877	0.606	0.974	0.951	0.901	2.03	23.8
	.60	.017	.014	.053	150	1.29	1.72	919	1663	.882	.569	.975	.951	.898	2.18	22.3
	.65	.017		.053	150	1.14	1.54	940	1645	.887	.534	.976	.952	.896	2.38	20.7
	.70	.016		.054	147	1.00	1.35	965	1627	.896	.497	.978	.953	.893	2.56	18.8
	.75	.016		.054	124	1.00	1.16	989	1609	.930	.462	.979	.954	.890	2.77	17.0
Turbine-inlet temperature T_4', 2500° R																
0	0.55	0.026	0.026	0.053	150	1.44	1.84	940	2297	0.865	0.663	0.997	0.961	0.951	2.34	22.3
	.60	.025	.025			1.31	1.68	967	2274	.873	.631			.949	2.57	21.5
	.65	.025	.025			1.18	1.51	995	2250	.880	.598			.948	2.81	20.4
	.70	.025	.025			1.03	1.33	1022	2225	.885	.566			.947	3.07	18.9
	.75	.024	.024		135	1.00	1.14	1051	2200	.916	.535			.945	3.36	17.5
0.05	0.55	0.026	0.026	0.053	150	1.42	1.91	930	2236	0.863	0.663	0.988	0.957	0.947	2.28	22.5
	.60	.026	.024			1.29	1.74	954	2216	.870	.631	.988	.957	.946	2.47	21.7
	.65	.025	.024			1.16	1.57	981	2193	.877	.598	.988	.956	.945	2.70	20.5
	.70	.025	.024			1.01	1.38	1006	2171	.883	.566	.989	.956	.944	2.92	18.8
	.75	.024	.023		131	1.00	1.18	1033	2148	.916	.535	.990	.956	.942	3.18	17.4
0.10	0.55	0.026	0.024	0.053	150	1.40	1.99	919	2176	0.861	0.663	0.978	0.952	0.944	2.19	22.8
	.60	.026	.023		150	1.27	1.82	940	2156	.866	.631	.978	.952	.943	2.38	21.0
	.65	.025	.023		150	1.14	1.64	966	2136	.873	.598	.979	.952	.942	2.57	20.5
	.70	.025	.023		149	1.00	1.44	992	2115	.875	.566	.980	.953	.940	2.79	18.8
	.75	.025	.022		128	1.00	1.23	1016	2094	.918	.535	.980	.953	.939	3.02	17.4
0.15	0.55	0.026	0.022	0.053	150	1.38	2.09	907	2111	0.856	0.663	0.966	0.944	0.940	2.10	23.1
	.60	.026			150	1.25	1.92	930	2094	.864	.631	.966	.944	.939	2.27	22.1
	.65	.026			150	1.12	1.72	952	2076	.869	.598	.968	.945	.938	2.45	20.6
	.70	.025			147	1.00	1.51	976	2058	.881	.566	.969	.946	.937	2.65	19.0
	.75	.025	.021		127	1.00	1.29	1000	2039	.917	.535	.969	.946	.936	2.87	17.5
Turbine-inlet temperature T_4', 3000° R																
0	0.55	0.036	0.036	0.052	150	1.37	2.00	950	2791	0.847	0.704	0.994	0.960	0.967	2.44	21.5
	.60	.035	.035			1.26	1.83	975	2768	.856	.676	.995	.959	.966	2.65	20.9
	.65	.035	.035			1.14	1.65	1004	2744	.863	.648	.994	.960	.966	2.90	20.1
	.70	.034	.034			1.01	1.45	1032	2718	.870	.618	.995	.959	.965	3.17	18.7
	.75	.034	.034		132	1.00	1.24	1061	2692	.907	.590	.996	.959	.964	3.47	17.8
0.05	0.55	0.036	0.034	0.052	150	1.35	2.08	938	2712	0.843	0.704	0.985	0.955	0.965	2.33	21.8
	.60	.035	.034		150	1.24	1.90	964	2691	.852	.676	.986	.955	.964	2.54	21.0
	.65	.035	.033		150	1.12	1.71	990	2669	.859	.648	.986	.954	.964	2.78	20.1
	.70	.034	.033		149	1.00	1.50	1017	2648	.869	.618	.986	.956	.963	3.02	18.8
	.75	.034	.032	.053	130	1.00	1.28	1046	2621	.907	.590	.987	.956	.962	3.30	17.8
0.10	0.55	0.036	0.032	0.052	150	1.33	2.17	927	2631	0.840	0.704	0.974	0.948	0.963	2.26	21.9
	.60	.036	.032	.052	150	1.22	1.99	951	2612	.848	.676	.974	.949	.962	2.44	21.2
	.65	.035	.032	.052	150	1.10	1.79	975	2591	.858	.648	.974	.949	.961	2.65	20.2
	.70	.035	.031	.053	148	1.00	1.57	1000	2570	.870	.618	.976	.950	.961	2.87	18.5
	.75	.034	.031	.053	128	1.00	1.34	1026	2548	.908	.590	.976	.950	.960	3.15	17.8
0.15	0.55	0.036	0.031	0.052	150	1.31	2.29	915	2548	0.834	0.704	0.960	0.936	0.960	2.16	22.1
	.60	.036	.030	.052	150	1.20	2.09	937	2531	.844	.676	.961	.938	.959	2.35	21.5
	.65	.035		.053	150	1.08	1.88	961	2512	.851	.648	.962	.936	.959	2.52	20.9
	.70	.035		.053	144	1.00	1.66	984	2492	.870	.618	.962	.938	.958	2.75	19.0
	.75	.035		.053	126	1.00	1.41	1010	2473	.908	.590	.964	.941	.957	2.96	17.6
Turbine-inlet temperature T_4', 3500° R																
0	0.55	0.046	0.046	0.051	150	1.30	2.14	957	3285	0.827	0.736	0.993	0.959	0.976	2.50	20.5
	.60	.046	.046	.051	150	1.21	1.97	986	3262	.837	.710	.993	.958	.976	2.75	20.3
	.65	.045	.045	.051	150	1.10	1.77	1018	3236	.847	.683	.994	.958	.976	3.01	19.5
	.70	.045	.045	.052	148	1.00	1.56	1043	3210	.861	.657	.994	.956	.975	3.29	18.4
	.75	.044	.044	.052	131	1.00	1.33	1073	3185	.898	.632	.994	.959	.974	3.59	17.8
0.05	0.55	0.046	0.044	0.051	150	1.28	2.25	946	3190	0.823	0.736	0.983	0.954	0.975	2.40	20.6
	.60	.046	.044	.051	150	1.19	2.04	973	3167	.833	.710	.984	.954	.974	2.62	20.3
	.65	.045	.043	.052	150	1.08	1.84	1000	3145	.842	.683	.984	.954	.974	2.86	19.5
	.70	.045	.043	.052	146	1.00	1.62	1028	3119	.861	.657	.984	.953	.974	3.12	18.6
	.75	.044	.042	.052	128	1.00	1.38	1056	3096	.900	.632	.984	.954	.973	3.40	17.6
0.10	0.55	0.046	0.042	0.052	150	1.26	2.33	934	3091	0.818	0.736	0.972	0.948	0.973	2.30	20.7
	.60	.046	.041		150	1.17	2.15	959	3070	.828	.710	.972	.948	.973	2.50	20.3
	.65	.046	.041		150	1.06	1.91	986	3048	.838	.683	.972	.947	.972	2.75	19.7
	.70	.045	.041		143	1.00	1.69	1013	3028	.862	.657	.972	.946	.972	2.98	18.7
	.75	.045	.040		126	1.00	1.44	1036	3005	.900	.632	.973	.947	.971	3.21	17.5
0.15	0.55	0.047	0.040	0.052	150	1.24	2.46	925	2989	0.812	0.736	0.957	0.928	0.971	2.22	21.0
	.60	.046	.039		150	1.14	2.26	946	2970	.821	.710	.956	.927	.970	2.40	20.4
	.65	.046	.039		150	1.04	2.03	972	2950	.833	.683	.957	.930	.970	2.62	19.7
	.70	.045	.039		141	1.00	1.79	995	2930	.862	.657	.958	.931	.970	2.82	18.8
	.75	.045	.038		124	1.00	1.53	1020	2911	.901	.632	.959	.932	.970	3.08	17.6

[redacted]

TABLE I. - Continued. TABULATION OF RESULTS

(b) Turbine blade tip speed U_t, 1300 feet per second; afterburner reference velocity V_{AB}, 500 feet per second.

Coolant-flow ratio, C	Turbine hub-tip radius ratio, r_h/r_t	Combustor fuel-air ratio, f_B	Mixture fuel-air ratio, f_8	After-burner fuel-air ratio, f_{AB}	Combustor reference velocity, V_B, ft/sec	Combustor-turbine frontal-area ratio, A_B/A_T	Afterburner-turbine frontal-area ratio, A_{AB}/A_T	Compressor-exit temperature, T_3', °R	Mixture temperature, T_8', °R	Combustor pressure ratio, p_4'/p_3'	Turbine pressure ratio, p_5'/p_4'	Mixing-zone pressure ratio, p_8'/p_7'	After-burner diffuser pressure ratio, p_9'/p_8'	After-burner pressure ratio, p_{10}'/p_9'	Compressor pressure ratio, p_3'/p_2'	Turbine-limited specific weight flow, $w_c\sqrt{\theta_2}/A_T\delta_2$, (lb/sec)/sq ft
Turbine-inlet temperature T_4', 2000° R																
0	0.55	0.016	0.016	0.053	150	1.40	1.63	1000	1743	0.893	0.510	0.999	0.963	0.909	2.86	25.2
	.60	.015	.015	↓	150	1.24	1.48	1035	1711	.905	.465	1.00	↓	.905	3.20	23.9
	.65	.015	.015	↓	150	1.08	1.33	1069	1680	.910	.424	↓	↓	.901	3.55	22.2
	.70	.014	.014	↓	138	1.00	1.16	1106	1647	.926	.384	↓	↓	.896	3.96	20.4
	.75	.014	.014	↓	113	1.00	.99	1145	1612	.955	.345	↓	↓	.891	4.42	18.3
0.05	0.55	0.016	0.015	0.053	150	1.38	1.70	965	1708	0.896	0.510	0.993	0.961	0.904	2.74	25.2
	.60	.016	.015	↓	150	1.22	1.54	1019	1679	.902	.465	.994	↓	.901	3.05	23.9
	.65	.015	.014	↓	150	1.06	1.38	1052	1651	.908	.424	.994	↓	.897	3.37	22.1
	.70	.014	.014	↓	135	1.00	1.21	1088	1621	.929	.384	.995	↓	.892	3.74	20.3
	.75	.014	.013	.054	111	1.00	1.03	1124	1589	.955	.345	.996	.962	.887	4.16	18.1
0.10	0.55	0.016	0.015	0.053	150	1.36	1.77	971	1672	0.893	0.510	0.986	0.957	0.900	2.61	25.3
	.60	.016	.014	.053	150	1.20	1.61	1003	1645	.899	.465	.987	.958	.896	2.89	23.8
	.65	.015	.014	.054	150	1.04	1.45	1034	1620	.905	.424	.988	.958	.892	3.19	22.0
	.70	.015	.013	.054	133	1.00	1.27	1067	1593	.930	.384	.989	.958	.887	3.52	20.2
	.75	.014	.013	.054	109	1.00	1.06	1102	1564	.955	.345	.990	.959	.882	3.90	17.9
0.15	0.55	0.016	0.014	0.053	150	1.34	1.86	957	1634	0.890	0.510	0.977	0.952	0.894	2.49	25.5
	.60	.016	.014	.054	150	1.18	1.69	987	1610	.896	.465	.979	.953	.890	2.75	23.9
	.65	.016	.013	↓	150	1.02	1.52	1017	1587	.902	.424	.980	.954	.887	3.02	22.0
	.70	.015	.013	↓	131	1.00	1.33	1047	1562	.930	.384	.982	.955	.882	3.32	20.2
	.75	.015	.012	↓	107	1.00	1.13	1083	1537	.955	.345	.985	.956	.877	3.69	17.9
Turbine-inlet temperature T_4', 2500° R																
0	0.55	0.025	0.025	0.053	150	1.39	1.82	1013	2253	0.884	0.576	0.997	0.961	0.947	2.99	25.5
	.60	.024	.024	↓	150	1.26	1.66	1049	2201	.891	.537	↓	↓	.945	3.34	24.8
	.65	.024	.024	↓	150	1.11	1.49	1085	2169	.898	.499	↓	↓	.944	3.72	25.5
	.70	.023	.025	↓	145	1.00	1.30	1123	2136	.910	.461	↓	↓	.942	4.16	21.9
	.75	.022	.022	↓	122	1.00	1.11	1162	2100	.941	.424	.998	↓	.940	4.83	20.1
0.05	0.55	0.025	0.024	0.053	150	1.37	1.89	998	2178	0.880	0.576	0.989	0.957	0.944	2.85	25.5
	.60	.024	.023	↓	150	1.24	1.72	1032	2149	.888	.537	.989	.958	.943	3.17	24.6
	.65	.024	.023	↓	150	1.09	1.55	1066	2120	.895	.499	.990	↓	.941	3.52	23.5
	.70	.023	.022	↓	142	1.00	1.36	1103	2089	.911	.461	.990	↓	.939	3.92	21.6
	.75	.023	.022	↓	120	1.00	1.16	1139	2057	.941	.424	.991	↓	.937	4.35	19.9
0.10	0.55	0.025	0.023	0.053	150	1.34	1.97	983	2122	0.877	0.576	0.980	0.953	0.941	2.72	25.6
	.60	.025	.022	↓	150	1.21	1.80	1015	2095	.884	.537	.980	.953	.940	3.00	24.5
	.65	.024	.022	↓	150	1.07	1.62	1048	2069	.891	.499	.981	.954	.938	3.35	25.2
	.70	.024	.021	↓	139	1.00	1.42	1082	2041	.912	.461	.982	.954	.936	3.69	21.6
	.75	.023	.021	↓	117	1.00	1.21	1118	2011	.942	.424	.983	.954	.934	4.08	19.7
0.15	0.55	0.025	0.022	0.053	150	1.32	2.07	969	2064	0.874	0.576	0.969	0.946	0.937	2.59	25.6
	.60	.025	.021	↓	150	1.19	1.89	1000	2040	.881	.537	.970	.947	.936	2.86	24.6
	.65	.024	.021	↓	150	1.05	1.70	1030	2015	.887	.499	.971	.947	.934	3.15	25.1
	.70	.024	.020	↓	137	1.00	1.49	1063	1990	.913	.461	.972	.948	.932	3.48	21.6
	.75	.024	.020	↓	115	1.00	1.27	1095	1963	.942	.424	.974	.949	.930	3.83	19.6
Turbine-inlet temperature T_4', 3000° R																
0	0.55	0.034	0.034	0.052	150	1.36	1.98	1026	2724	0.869	0.625	0.995	0.959	0.965	3.11	25.4
	.60	.034	.034	.052	150	1.24	1.81	1062	2692	.878	.590	↓	.959	.964	3.48	24.9
	.65	.033	.033	.053	150	1.12	1.63	1099	2659	.885	.555	↓	.958	.963	3.88	24.0
	.70	.032	.032	.053	147	1.00	1.43	1137	2625	.897	.520	↓	.960	.962	4.32	22.6
	.75	.032	.032	.053	126	1.00	1.22	1178	2589	.930	.486	.996	.960	.961	4.85	21.2
0.05	0.55	0.035	0.033	0.052	150	1.34	2.06	1010	2650	0.866	0.625	0.986	0.955	0.963	2.98	25.3
	.60	.034	.032	.052	150	1.22	1.88	1045	2621	.874	.590	.987	.956	.962	3.30	24.6
	.65	.034	.032	.053	150	1.09	1.69	1080	2591	.882	.555	.986	.955	.961	3.66	23.8
	.70	.033	.031	.053	144	1.00	1.49	1116	2560	.898	.520	.987	.956	.960	4.07	22.4
	.75	.032	.031	.053	123	1.00	1.27	1155	2527	.930	.486	.988	.956	.959	4.54	20.9
0.10	0.55	0.035	0.031	0.053	150	1.31	2.15	995	2575	0.861	0.625	0.975	0.950	0.961	2.82	25.3
	.60	.034	.031	↓	150	1.20	1.97	1028	2548	.870	.590	.976	.949	.960	3.13	24.7
	.65	.034	.030	↓	150	1.07	1.77	1061	2520	.878	.555	.976	.950	.959	3.46	23.6
	.70	.033	.030	↓	141	1.00	1.55	1094	2491	.896	.520	.977	.950	.958	3.82	22.2
	.75	.033	.029	↓	121	1.00	1.32	1132	2462	.930	.486	.978	.951	.957	4.25	20.7
0.15	0.55	0.035	0.030	0.053	150	1.29	2.27	979	2497	0.857	0.625	0.962	0.936	0.958	2.66	25.5
	.60	.035	.030	↓	150	1.17	2.06	1009	2472	.866	.590	.964	.940	.957	2.95	24.5
	.65	.034	.029	↓	150	1.05	1.86	1042	2447	.873	.555	.964	.940	.956	3.27	23.5
	.70	.034	.029	↓	139	1.00	1.63	1074	2421	.899	.520	.966	.942	.955	3.60	22.2
	.75	.033	.028	↓	118	1.00	1.39	1107	2394	.931	.486	.967	.943	.954	3.97	20.5
Turbine-inlet temperature T_4', 3500° R																
0	0.55	0.045	0.045	0.052	150	1.32	2.13	1037	3217	0.854	0.664	0.993	0.958	0.975	3.22	24.9
	.60	.044	.044	↓	150	1.22	1.95	1074	3186	.864	.633	.993	.959	.975	3.58	24.7
	.65	.043	.043	↓	150	1.11	1.75	1113	3151	.874	.599	.994	.959	.974	4.03	24.2
	.70	.043	.043	↓	147	1.00	1.54	1154	3115	.887	.566	.994	.958	.974	4.52	23.2
	.75	.042	.042	↓	128	1.00	1.32	1192	3081	.922	.536	.994	.958	.973	5.04	21.9
0.05	0.55	0.045	0.043	0.052	150	1.30	2.21	1021	3126	0.848	0.664	0.984	0.954	0.974	3.06	24.8
	.60	.044	.042	↓	150	1.19	2.02	1054	3097	.859	.633	.984	↓	.973	3.39	24.4
	.65	.044	.042	↓	150	1.08	1.82	1093	3065	.869	.599	.984	↓	.973	3.80	23.9
	.70	.043	.041	↓	144	1.00	1.60	1131	3032	.888	.566	.985	↓	.972	4.25	22.9
	.75	.042	.040	↓	125	1.00	1.37	1168	3000	.922	.536	.985	↓	.971	4.72	21.6
0.10	0.55	0.046	0.041	0.052	150	1.27	2.31	1005	3032	0.844	0.664	0.972	0.947	0.972	2.91	24.7
	.60	.045	.040	↓	150	1.17	2.11	1036	3005	.854	.633	.973	.947	.971	3.21	24.3
	.65	.044	.040	↓	150	1.06	1.90	1075	2976	.864	.599	.974	.947	.971	3.58	23.6
	.70	.044	.039	↓	141	1.00	1.67	1109	2945	.868	.566	.974	.948	.970	3.99	22.7
	.75	.043	.039	↓	122	1.00	1.45	1143	2917	.923	.536	.974	.948	.970	4.40	21.2
0.15	0.55	0.046	0.039	0.052	150	1.24	2.44	988	2935	0.839	0.664	0.957	0.931	0.970	2.77	24.8
	.60	.045	.038	↓	150	1.15	2.24	1019	2911	.849	.633	.959	.932	.969	3.04	24.2
	.65	.044	.038	↓	150	1.03	2.00	1052	2884	.856	.599	.960	.934	.969	3.37	23.3
	.70	.044	.037	↓	138	1.00	1.78	1086	2856	.889	.566	.961	.934	.968	3.74	22.5
	.75	.043	.037	↓	120	1.00	1.50	1121	2830	.923	.536	.962	.936	.968	4.13	21.1

TABLE I. - Continued. TABULATION OF RESULTS

(c) Turbine blade tip speed U_t, 1500 feet per second; afterburner reference velocity V_{AB}, 500 feet per second.

Coolant-flow ratio, C	Turbine hub-tip radius ratio, r_h/r_t	Combustor fuel-air ratio, f_B	Mixture fuel-air ratio, f_6	After-burner fuel-air ratio, f_{AB}	Combustor reference velocity, V_B, ft/sec	Combustor-turbine frontal-area ratio, A_B/A_T	Afterburner-turbine frontal-area ratio, A_{AB}/A_T	Compressor-exit temperature, T_3', °R	Mixture temperature, T_6', °R	Combustor pressure ratio, p_4'/p_3'	Turbine pressure ratio, p_7'/p_4'	Mixing-zone pressure ratio, p_8'/p_7'	After-burner diffuser pressure ratio, p_9'/p_8'	After-burner pressure ratio, p_{10}'/p_9'	Compressor pressure ratio, p_3'/p_2'	Turbine-limited specific weight flow, $w_c\sqrt{\theta_2}/\delta_2 A_T$, (lb/sec)/sq ft
Turbine-inlet temperature T_4', 2000° R																
0	0.55	0.015	0.015	0.053	150	1.28	1.60	1078	1672	0.912	0.414	1.00	0.963	0.900	3.64	27.0
	.60	.014	.014	.054	150	1.11	1.46	1122	1633	.917	.368			.894	4.14	25.5
	.65	.015	.013		142	1.00	1.30	1167	1592	.931	.324			.887	4.68	23.6
	.70	.015	.013		117	1.00	1.14	1213	1549	.954	.281			.880	5.34	21.5
	.75	.012	.012		92	1.00	.96	1260	1505	.978	.243			.871	6.06	18.8
0.05	0.55	0.015	0.014	0.054	150	1.26	1.67	1060	1644	0.906	0.414	0.984	0.961	0.895	3.46	26.5
	.60	.014	.014		150	1.09	1.52	1102	1608	.914	.368	.985	.961	.890	3.90	25.2
	.65	.014	.013		139	1.00	1.36	1144	1571	.951	.324	.986	.961	.883	4.41	23.4
	.70	.013	.012		114	1.00	1.19	1189	1532	.957	.281	.987	.962	.876	5.00	21.2
	.75	.012	.012		90	1.00	1.01	1235	1492	.975	.243	.988	.962	.868	5.65	18.4
0.10	0.55	0.015	0.014	0.054	150	1.25	1.75	1042	1615	0.906	0.414	0.988	0.958	0.891	3.27	26.8
	.60	.015	.013		150	1.07	1.59	1082	1581	.912	.368	.989	.959	.886	3.68	25.1
	.65	.014	.013		136	1.00	1.42	1122	1547	.932	.324	.991	.959	.879	4.14	23.2
	.70	.013	.012		112	1.00	1.25	1163	1512	.957	.281	.993	.960	.872	4.65	20.8
	.75	.013	.012		89	1.00	1.06	1207	1476	.975	.243	.995	.961	.864	5.26	18.0
0.15	0.55	0.016	0.013	0.054	150	1.21	1.84	1023	1581	0.903	0.414	0.980	0.954	0.885	3.08	26.6
	.60	.016	.013		150	1.05	1.67	1061	1552	.909	.368	.983	.956	.880	3.48	24.8
	.65	.014	.012		134	1.00	1.50	1100	1522	.932	.324	.985	.956	.874	3.88	23.0
	.70	.014	.012		110	1.00	1.31	1140	1491	.957	.281	.988	.958	.867	4.36	20.6
	.75	.013	.011		86	1.00	1.11	1180	1458	.975	.243	.990	.959	.860	4.87	17.7
Turbine-inlet temperature T_4', 2500° R																
0	0.55	0.024	0.024	0.053	150	1.33	1.80	1094	2161	0.899	0.490	0.997	0.961	0.943	3.82	28.7
	.60	.023	.023		150	1.17	1.63	1140	2119	.906	.444	.998		.941	4.35	27.8
	.65	.022	.022		150	1.02	1.48	1186	2078	.915	.403			.938	4.96	26.3
	.70	.021	.021		131	1.00	1.28	1235	2034	.938	.362			.935	5.68	24.8
	.75	.020	.020		107	1.00	1.09	1283	1989	.961	.323			.932	6.45	22.5
0.05	0.55	0.024	0.023	0.053	150	1.30	1.87	1075	2112	0.895	0.490	0.990	0.957	0.941	3.61	28.5
	.60	.023	.022		150	1.15	1.70	1119	2074	.903	.444	.990	.956	.938	4.10	27.5
	.65	.022	.021		150	1.00	1.53	1163	2037	.908	.403	.991	.956	.936	4.65	25.9
	.70	.022	.021		128	1.00	1.34	1209	1996	.939	.362	.992	.956	.933	5.28	24.4
	.75	.021	.020		105	1.00	1.14	1255	1955	.962	.323	.992	.959	.930	5.92	22.0
0.10	0.55	0.024	0.022	0.053	150	1.28	1.95	1057	2062	0.893	0.490	0.981	0.953	0.937	3.42	28.4
	.60	.023	.021		150	1.13	1.78	1098	2027	.900	.444	.983	.954	.936	3.87	27.2
	.65	.023	.021		147	1.00	1.60	1139	1992	.909	.403	.983	.954	.933	4.34	25.8
	.70	.022	.020		126	1.00	1.40	1183	1956	.939	.362	.985	.955	.930	4.92	24.0
	.75	.021	.019	.054	98	1.00	1.19	1228	1919	.965	.323	.986	.958	.926	5.56	21.6
0.15	0.55	0.024	0.021	0.053	150	1.25	2.05	1037	2009	0.889	0.490	0.971	0.947	0.934	3.22	28.2
	.60	.024	.020		150	1.11	1.87	1078	1977	.898	.444	.973	.948	.931	3.64	27.0
	.65	.023	.020		145	1.00	1.67	1117	1947	.910	.403	.975	.949	.928	4.07	25.4
	.70	.022	.019		122	1.00	1.47	1157	1913	.940	.362	.976	.950	.926	4.56	23.5
	.75	.022	.019	.054	100	1.00	1.25	1200	1880	.962	.323	.979	.952	.923	5.15	21.1
Turbine-inlet temperature T_4', 3000° R																
0	0.55	0.033	0.033	0.053	150	1.33	1.86	1108	2650	0.887	0.548	0.995	0.958	0.963	3.98	29.4
	.60	.032	.032		150	1.20	1.79	1157	2606	.896	.504	.995		.962	4.56	[illegible]
	.65	.031	.031		150	1.06	1.61	1205	2565	.903	.464	.995		.960	5.22	[illegible]
	.70	.031	.031		138	1.00	1.41	1254	2520	.924	.424	.996		.959	5.98	[illegible]
	.75	.030	.030		115	1.00	1.20	1303	2474	.950	.387	.996		.957	6.80	26.0
0.05	0.55	0.033	0.032	0.053	150	1.30	2.04	1090	2562	0.884	0.548	0.984	0.955	0.961	3.77	[illegible]
	.60	.033	.031		150	1.18	1.87	1134	2544	.892	.504	.987	.956	.960	4.26	28.7
	.65	.032	.030		150	1.04	1.67	1181	2504	.899	.464	.988		.958	4.83	27.6
	.70	.031	.029		138	1.00	1.47	1229	2464	.924	.424	.988		.957	5.56	26.4
	.75	.030	.028		113	1.00	1.25	1276	2421	.951	.387	.989		.955	6.32	24.4
0.10	0.55	0.034	0.030	0.053	150	1.28	2.13	1070	2512	0.880	0.548	0.977	0.950	0.958	3.60	29.0
	.60	.033	.030		150	1.15	1.95	1112	2477	.888	.504	.978	.950	.957	4.02	28.5
	.65	.032	.029		149	1.02	1.74	1157	2442	.898	.464	.978	.951	.954	4.57	27.2
	.70	.031	.028		132	1.00	1.55	1202	2404	.924	.424	.979	.951	.953	5.17	25.8
	.75	.031	.028		111	1.00	1.31	1248	2365	.952	.387	.980	.952	.953	5.85	[illegible]
0.15	0.55	0.034	0.029	0.053	150	1.26	2.24	1050	2440	0.875	0.548	0.965	0.940	0.956	3.38	28.7
	.60	.033	.028		150	1.13	2.05	1091	2408	.884	.504	.966	.941	.955	3.78	28.0
	.65	.033	.028		149	1.00	1.84	1131	2375	.893	.464	.967	.942	.954	4.28	26.6
	.70	.032	.027		129	1.00	1.61	1174	2341	.928	.424	.968	.944	.952	4.78	25.4
	.75	.031	.027		106	1.00	1.37	1216	2306	.952	.387	.970	.945	.950	5.41	23.3
Turbine-inlet temperature T_4', 3500° R																
0	0.55	0.043	0.043	0.052	150	1.32	2.11	1123	3142	0.878	0.591	0.994	0.958	0.974	4.14	29.7
	.60	.042	.042		150	1.21	1.93	1171	3100	.885	.553			.973	4.75	[illegible]
	.65	.042	.042		150	1.08	1.73	1221	3056	.894	.515			.972	5.44	28.3
	.70	.041	.041		142	1.00	1.52	1272	3010	.912	.478			.972	6.26	26.3
	.75	.040	.040	.053	120	1.00	1.30	1322	2963	.941	.441			.971	7.13	24.5
0.05	0.55	0.044	0.041	0.052	150	1.29	2.19	1102	3057	0.870	0.591	0.985	0.954	0.975	3.91	[illegible]
	.60	.043	.041	.052	150	1.18	2.01	1148	3018	.881	.553	.985	.953	.972	4.45	29.3
	.65	.042	.040	.052	150	1.06	1.80	1195	2977	.890	.515	.985	.954	.971	5.08	[illegible]
	.70	.041	.039	.053	139	1.00	1.58	1245	2935	.912	.478	.986	.954	.970	5.80	27.7
	.75	.041	.039	.053	118	1.00	1.35	1292	2892	.942	.441	.986	.954	.969	6.60	26.0
0.10	0.55	0.044	0.040	0.052	150	1.26	2.29	1082	2966	0.866	0.591	0.974	0.947	0.971	3.68	29.0
	.60	.043	.038	.052	150	1.16	2.10	1126	2933	.877	.553	.974	.947	.970	4.18	28.8
	.65	.042	.038	.052	150	1.05	1.88	1169	2895	.885	.515	.975	.948	.969	4.73	28.0
	.70	.042	.038	.053	136	1.00	1.65	1218	2857	.914	.478	.976	.948	.968	5.40	27.2
	.75	.041	.037	.053	116	1.00	1.41	1263	2817	.942	.441	.977	.949	.967	6.11	25.4
0.15	0.55	0.044	0.038	0.052	150	1.25	2.42	1061	2877	0.861	0.591	0.960	0.934	0.969	3.46	28.7
	.60	.044	.037	.052	150	1.13	2.21	1102	2845	.871	.553	.961	.935	.968	3.91	28.4
	.65	.043	.037	.053	150	1.00	1.99	1143	2811	.880	.515	.962	.936	.967	4.40	27.4
	.70	.042	.036	.053	133	1.00	1.74	1189	2775	.914	.478	.963	.938	.966	5.00	26.5
	.75	.042	.035	.053	113	1.00	1.49	1234	2740	.943	.441	.965	.939	.966	5.65	24.5

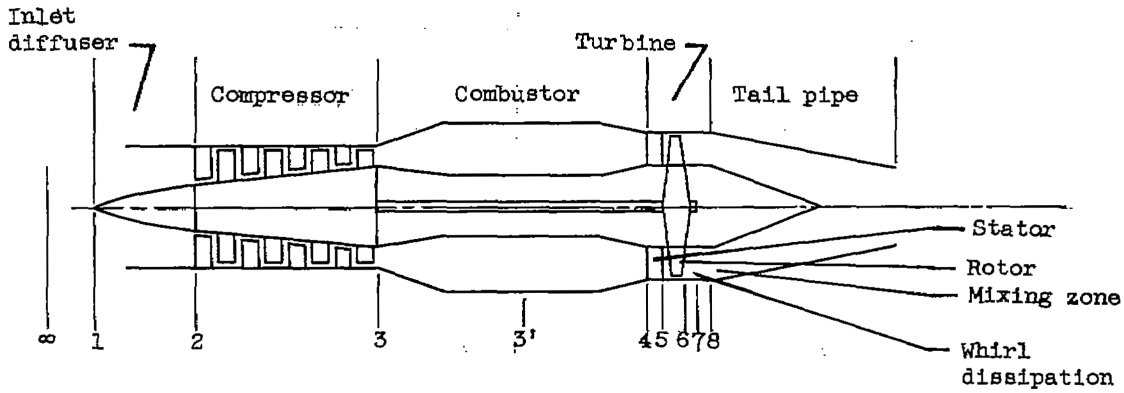
3171

TABLE I. - Concluded. TABULATION OF RESULTS

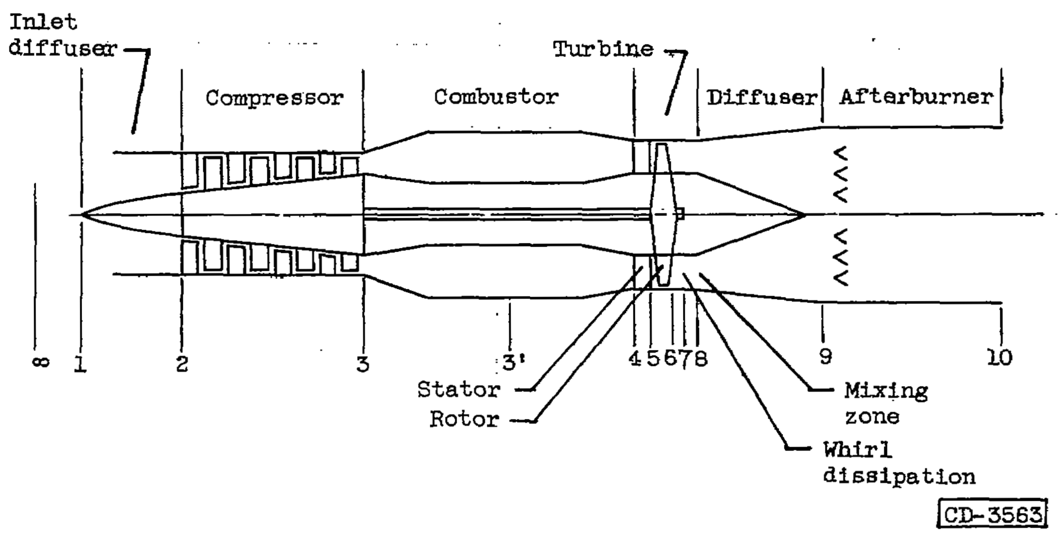
(d) Turbine blade tip speed U_t, 1700 feet per second; afterburer reference velocity V_{AB}, 500 feet per second.

Coolant-flow ratio, c	Turbine hub-tip radius ratio, r_h/r_t	Combustor fuel-air ratio, f_B	Mixture fuel-air ratio, f_8	After-burner fuel-air ratio, f_{AB}	Combustor reference velocity, V_B, ft/sec	Combustor-turbine frontal-area ratio, A_B/A_T	Afterburner-turbine frontal-area ratio, A_{AB}/A_T	Compressor-exit temperature, T_3', °R	Mixture temperature, T_8', °R	Combustor pressure ratio, p_4'/p_3'	Turbine pressure ratio, p_5'/p_4'	Mixing-zone pressure ratio, p_8'/p_7'	After-burner diffuser pressure ratio, p_9'/p_8'	After-burner pressure ratio, p_{10}'/p_9'	Compressor pressure ratio, p_3'/p_2'	Turbine-limited specific weight flow, $w_C\sqrt{\theta_2}/A_T\delta_2$, (lb/sec)/sq ft
Turbine-inlet temperature T_4', 2000° R																
0	0.55	0.014	0.014	0.054	150	1.14	1.57	1162	1597	0.922	0.329	1.00	0.963	0.888	4.62	28.3
	.60	.013	.013		144	1.00	1.42	1214	1548	.933	.280			.879	5.36	26.4
	.65	.012	.012		119	1.00	1.27	1269	1497	.956	.236			.869	6.21	24.3
	.70	.011	.011		93	1.00	1.10	1326	1443	.976	.195			.855	7.22	21.5
	.75	.010	.010		71	1.00	.94	1381	1392	.987	.161			.841	8.24	18.0
0.05	0.55	0.014	0.013	0.054	150	1.11	1.64	1139	1575	0.919	0.329	0.996	0.961	0.885	4.34	27.9
	.60	.013	.012		141	1.00	1.49	1189	1532	.934	.280	.997	.962	.877	5.00	26.0
	.65	.012	.012		116		1.33	1243	1486	.958	.236	.998	.962	.866	5.78	23.8
	.70	.011	.011		91		1.16	1297	1437	.976	.195	1.00	.962	.854	6.88	21.0
	.75	.011	.010		69		.98	1348	1390	.987	.161	1.00	.963	.840	7.82	17.5
0.10	0.55	0.014	0.013	0.054	150	1.09	1.72	1117	1551	0.917	0.329	0.991	0.959	0.880	4.08	27.5
	.60	.015	.012		136	1.00	1.56	1165	1512	.934	.280	.993	.960	.872	4.67	25.6
	.65	.013	.011		114		1.40	1215	1470	.958	.236	.995	.961	.862	5.36	23.4
	.70	.012	.011		89		1.22	1267	1426	.977	.195	.997	.962	.851	6.18	20.4
	.75	.011	.010		67		1.04	1316	1386	.987	.161	1.00	.963	.839	7.02	17.0
0.15	0.55	0.014	0.012	0.054	150	1.07	1.81	1095	1526	0.914	0.329	0.985	0.956	0.875	3.82	27.8
	.60	.014	.012		135	1.00	1.64	1142	1490	.935	.280	.988	.958	.867	4.36	25.3
	.65	.013	.011		111		1.47	1188	1453	.959	.236	.991	.958	.858	4.98	23.0
	.70	.012	.010		87		1.29	1238	1413	.976	.195	.994	.960	.847	5.70	20.0
	.75	.012	.010		66		1.09	1284	1376	.987	.161	.997	.961	.835	6.46	16.6
Turbine-inlet temperature T_4', 2500° R																
0	0.55	0.022	0.022	0.053	150	1.25	1.77	1183	2081	0.912	0.406	0.986	0.961	0.936	4.92	31.8
	.60	.021	.021	.053	150	1.07	1.61	1238	2031	.919	.359			.935	5.70	30.6
	.65	.020	.020	.053	137	1.00	1.44	1294	1978	.938	.314			.931	6.64	29.1
	.70	.019	.019	.054	113	1.00	1.26	1353	1923	.961	.272			.927	7.73	27.0
	.75	.018	.018	.054	90	1.00	1.06	1414	1867	.977	.233			.922	8.92	23.7
0.05	0.55	0.022	0.021	0.053	150	1.20	1.85	1160	2038	0.909	0.406	0.991	0.958	0.936	4.60	31.2
	.60	.022	.021	.053	150	1.05	1.68	1212	1992	.918	.358	.992	.958	.932	5.32	30.0
	.65	.021	.020	.053	134	1.00	1.50	1267	1947	.938	.314	.993	.959	.929	6.16	28.4
	.70	.020	.019	.054	110	1.00	1.31	1322	1895	.961	.272	.994	.959	.924	7.14	26.2
	.75	.019	.018	.054	88	1.00	1.11	1380	1845	.977	.233	.995	.960	.920	8.24	25.0
0.10	0.55	0.023	0.021	0.053	150	1.18	1.92	1137	1995	0.906	0.406	0.984	0.954	0.933	4.30	30.7
	.60	.022	.020	.053	150	1.03	1.76	1186	1953	.912	.359	.984	.955	.929	4.98	29.3
	.65	.021	.019	.054	131	1.00	1.57	1239	1910	.938	.314	.986	.956	.926	5.72	27.6
	.70	.020	.018	.054	108	1.00	1.37	1292	1865	.961	.272	.988	.956	.922	6.60	25.6
	.75	.020	.018	.054	85	1.00	1.17	1345	1818	.978	.233	.990	.957	.917	7.57	22.4
0.15	0.55	0.023	0.020	0.053	150	1.15	2.02	1113	1948	0.902	0.406	0.974	0.949	0.929	4.03	30.3
	.60	.022	.019	.053	150	1.00	1.84	1160	1911	.908	.359	.976	.950	.926	4.62	28.8
	.65	.022	.018	.053	128		1.65	1210	1882	.939	.314	.981	.953	.923	5.29	27.3
	.70	.021	.018	.054	105		1.44	1261	1831	.962	.272	.981	.952	.918	6.07	24.9
	.75	.020	.017	.054	83		1.23	1312	1789	.977	.233	.984	.954	.914	6.94	21.7
Turbine-inlet temperature T_4', 3000° R																
0	0.55	0.031	0.031	0.053	150	1.27	1.94	1202	2567	0.903	0.466	0.996	0.959	0.960	5.18	33.9
	.60	.030	.030		150	1.13	1.77	1258	2516	.910	.421	.995	.960	.958	6.04	33.4
	.65	.030	.030		148	1.00	1.58	1315	2463	.919	.378	.996	.959	.957	7.01	32.2
	.70	.029	.028		125	1.00	1.38	1376	2407	.947	.337	.997	.958	.955	8.16	30.7
	.75	.028	.028	.054	102	1.00	1.18	1439	2349	.967	.298	.997	.959	.952	9.42	27.9
0.05	0.55	0.032	0.030	0.053	150	1.26	2.02	1178	2507	0.899	0.466	0.988	0.956	0.958	4.85	33.3
	.60	.031	.029		150	1.11	1.84	1232	2461	.907	.421	.989	.956	.957	5.62	32.6
	.65	.030	.029		145	1.00	1.65	1287	2412	.920	.378	.989	.956	.955	6.50	31.4
	.70	.029	.028		123	1.00	1.45	1343	2361	.947	.337	.990	.956	.953	7.54	29.9
	.75	.028	.027		100	1.00	1.23	1403	2306	.968	.298	.990	.957	.951	8.70	27.1
0.10	0.55	0.032	0.029	0.053	150	1.22	2.11	1155	2443	0.895	0.466	0.978	0.950	0.956	4.53	32.8
	.60	.031	.028		150	1.08	1.92	1206	2400	.903	.421	.980	.951	.954	5.22	31.9
	.65	.030	.027		141	1.00	1.72	1258	2356	.920	.378	.980	.952	.953	6.02	30.8
	.70	.030	.027		120	1.00	1.51	1312	2309	.947	.337	.982	.952	.951	6.95	29.1
	.75	.029	.026	.054	98	1.00	1.28	1368	2261	.968	.298	.984	.953	.948	8.01	26.3
0.15	0.55	0.033	0.028	0.053	150	1.19	2.22	1128	2377	0.891	0.466	0.967	0.942	0.954	4.22	32.0
	.60	.032	.027		150	1.05	2.02	1178	2336	.899	.421	.969	.944	.952	4.85	31.2
	.65	.031	.026		136	1.00	1.81	1228	2298	.922	.378	.970	.944	.950	5.56	30.1
	.70	.030	.026		117	1.00	1.59	1280	2256	.948	.337	.972	.946	.948	6.40	28.4
	.75	.029	.025	.054	95	1.00	1.35	1333	2211	.968	.298	.974	.946	.946	7.34	25.6
Turbine-inlet temperature T_4', 3500° R																
0	0.55	0.042	0.042	0.052	150	1.30	2.09	1218	3059	0.894	0.518	0.994	0.956	0.973	5.40	35.2
	.60	.041	.041	.053	150	1.17	1.91	1276	3005	.903	.474	.994	.959	.971	6.33	35.4
	.65	.040	.040	.053	150	1.03	1.71	1335	2950	.910	.432	.995	.956	.970	7.38	34.6
	.70	.039	.039	.053	134	1.00	1.50	1397	2894	.934	.392	.994	.958	.969	8.56	33.4
	.75	.038	.038	.054	112	1.00	1.28	1462	2835	.958	.354	.994	.958	.968	9.88	30.9
0.05	0.55	0.042	0.040	0.052	150	1.27	2.18	1192	2981	0.889	0.518	0.985	0.954	0.971	5.04	34.4
	.60	.041	.039	.053	150	1.14	1.98	1249	2930	.898	.474	.986	.954	.970	5.88	34.4
	.65	.040	.038	.053	150	1.01	1.76	1306	2880	.906	.432	.986	.954	.969	6.85	33.7
	.70	.039	.037	.053	130	1.00	1.56	1364	2827	.935	.392	.986	.954	.968	7.92	32.6
	.75	.039	.037	.054	109	1.00	1.33	1424	2773	.958	.354	.987	.955	.966	9.12	30.1
0.10	0.55	0.043	0.038	0.052	150	1.23	2.27	1167	2898	0.884	0.518	0.975	0.948	0.969	4.70	33.7
	.60	.042	.038	.053	150	1.11	2.07	1222	2853	.894	.474	.978	.948	.968	5.47	33.6
	.65	.041	.037		147	1.00	1.86	1276	2806	.906	.432	.977	.948	.967	6.32	32.8
	.70	.040	.036		127	1.00	1.63	1330	2759	.935	.392	.977	.949	.966	7.30	31.6
	.75	.039	.035		106	1.00	1.39	1388	2709	.959	.354	.979	.950	.965	8.40	29.2
0.15	0.55	0.043	0.037	0.053	150	1.20	2.40	1142	2813	0.880	0.518	0.962	0.936	0.967	4.37	32.9
	.60	.042	.036		150	1.08	2.19	1194	2772	.890	.474	.963	.937	.966	5.06	32.8
	.65	.041	.035		144	1.00	1.95	1247	2730	.906	.432	.965	.939	.966	5.84	32.0
	.70	.040	.034		124	1.00	1.72	1298	2686	.936	.392	.966	.940	.964	6.70	30.8
	.75	.040	.034		103	1.00	1.46	1352	2642	.959	.354	.968	.942	.962	7.70	28.4

3171



(a) Nonafterburning engine.



(b) Afterburning engine.

Figure 1. - Schematic diagram of two turbojet engines.

3171

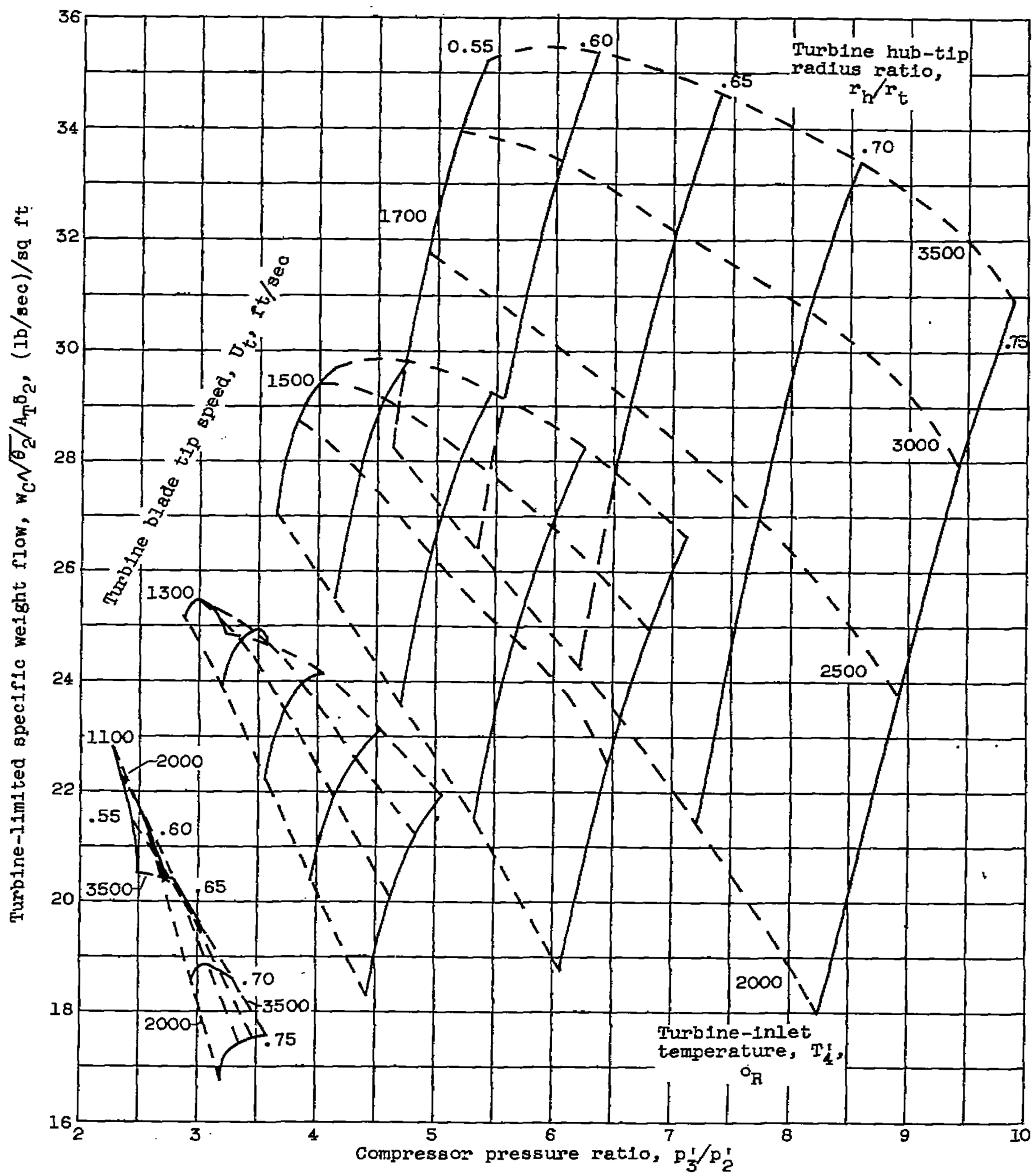


Figure 2. - Effect of turbine design variables on turbine performance. Coolant-flow ratio, 0; turbine-exit whirl parameter $V_{u,6,i}/U_i$, -0.410; turbine adiabatic efficiency, 0.83.

3171

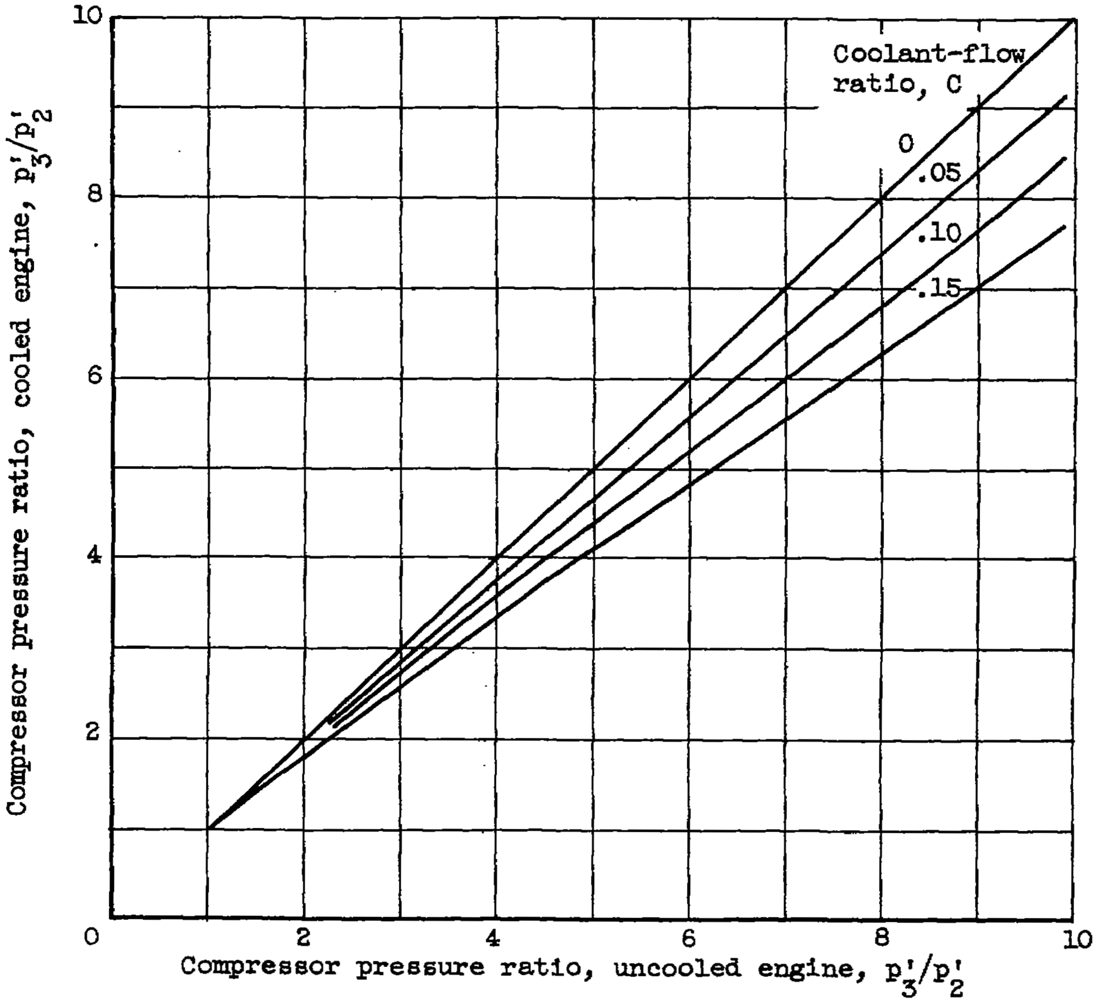
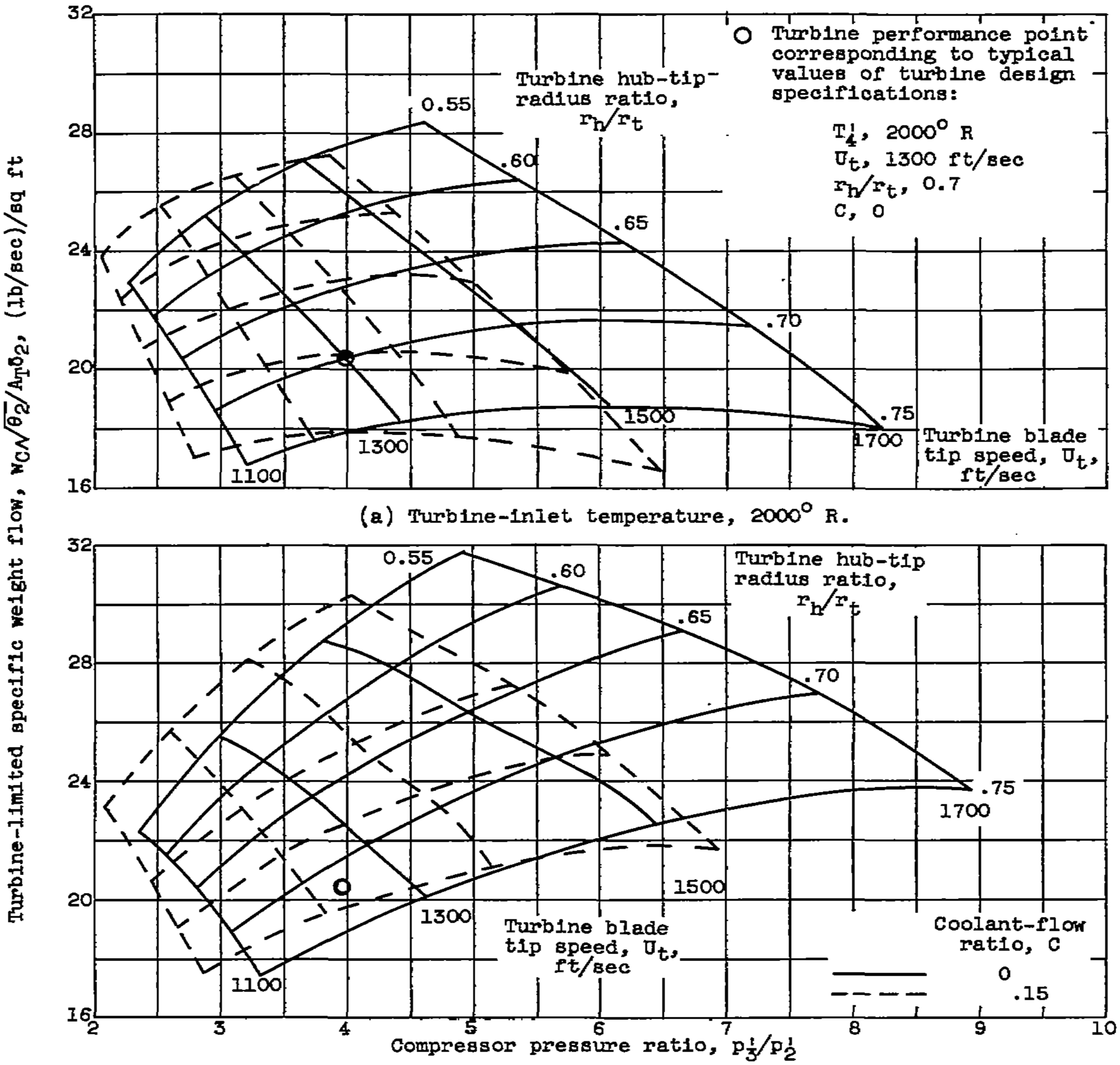


Figure 3. - Effect of coolant-flow ratio on compressor pressure ratio.



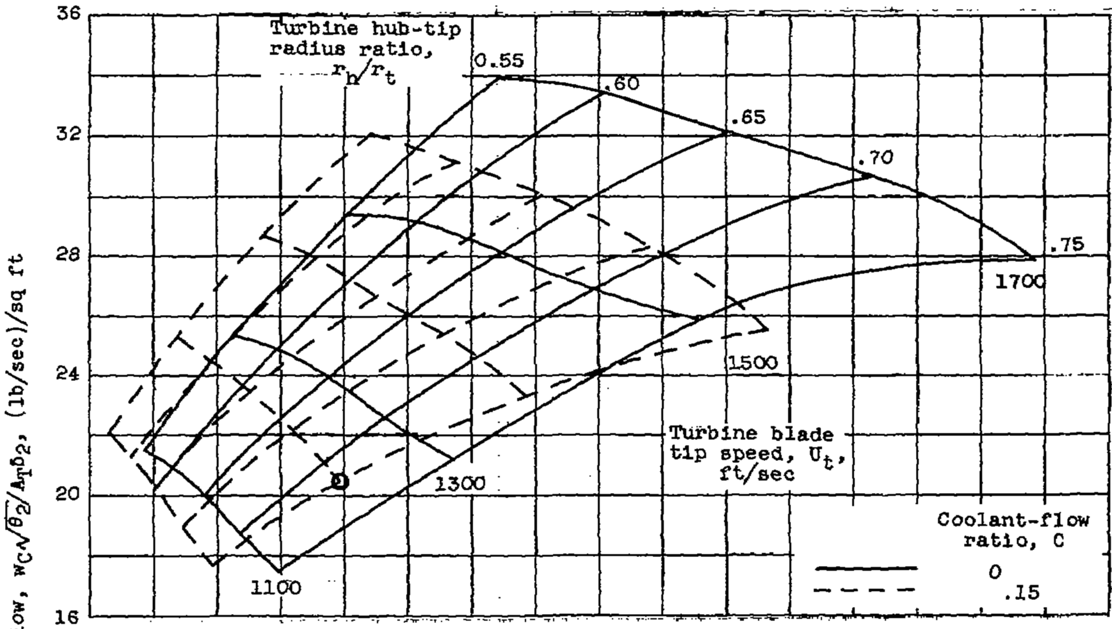
(a) Turbine-inlet temperature, 2000° R.

(b) Turbine-inlet temperature, 2500° R.

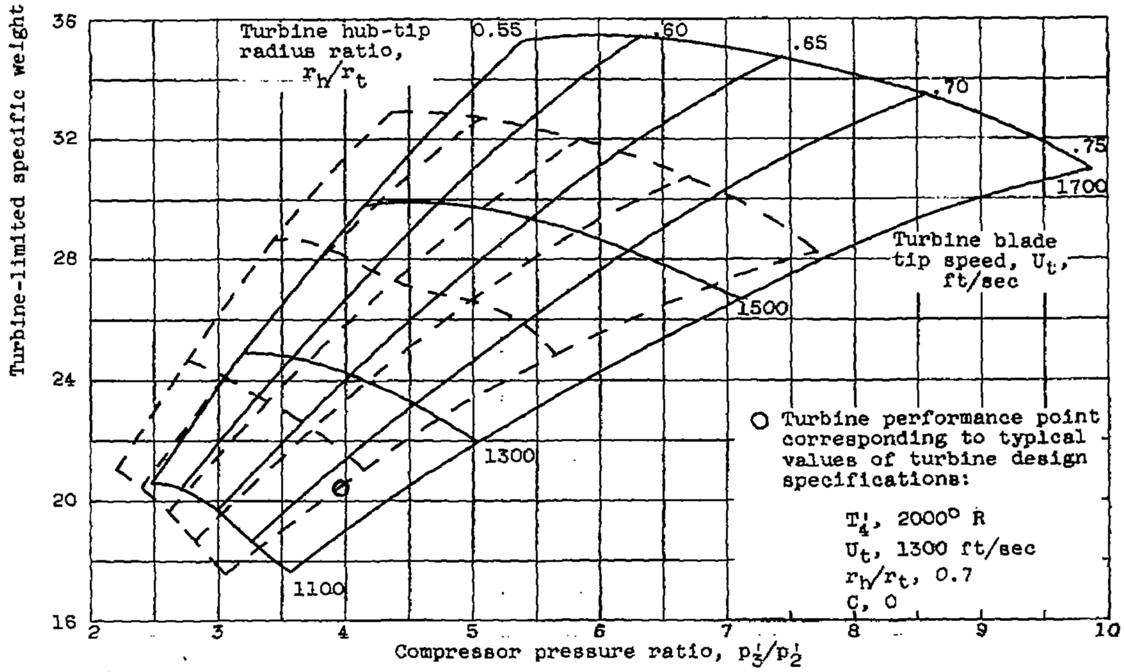
Figure 4. - Effect of coolant-flow ratio on turbine performance for range of turbine design variables. Turbine-exit whirl parameter $V_{u,6,i}/U_i$, -0.410; turbine adiabatic efficiency, 0.83.

[redacted]

3171



(c) Turbine-inlet temperature, 3000° R.



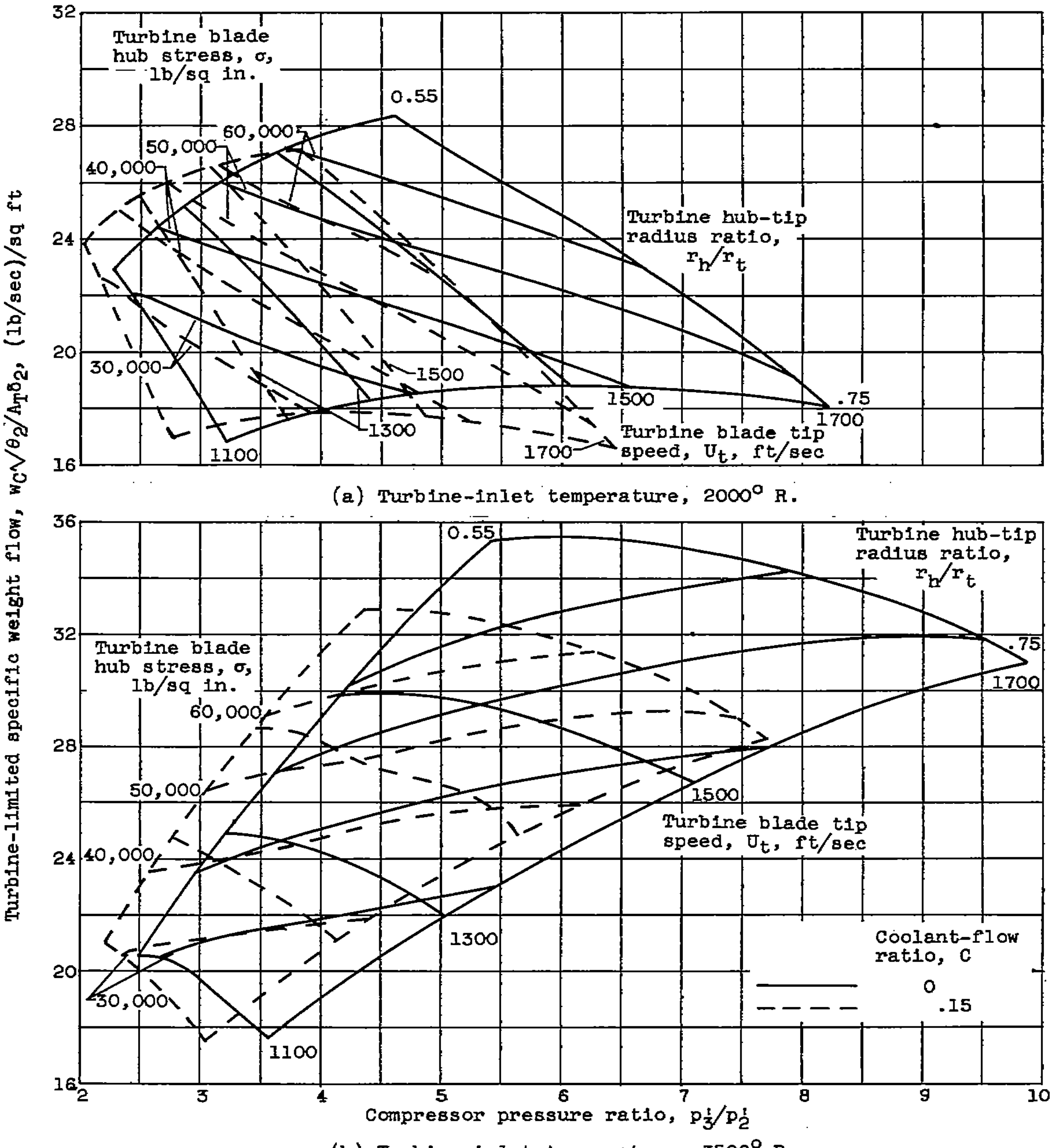
(d) Turbine-inlet temperature, 3500° R.

Figure 4. - Concluded. Effect of coolant-flow ratio on turbine performance for range of turbine design variables. Turbine-exit whirl parameter $V_{u,6,i}/U_i$, -0.410; turbine adiabatic efficiency, 0.83.

[redacted]

3171

CB-6 back



(a) Turbine-inlet temperature, 2000° R.

(b) Turbine-inlet temperature, 3500° R.

Figure 5. - Comparison of stresses resulting at various turbine performance levels for two coolant-flow ratios. Turbine-exit whirl parameter $V_{u,6,i}/U_i$, -0.410; turbine blade taper ratio, 0.40; turbine adiabatic efficiency, 0.83.

3171

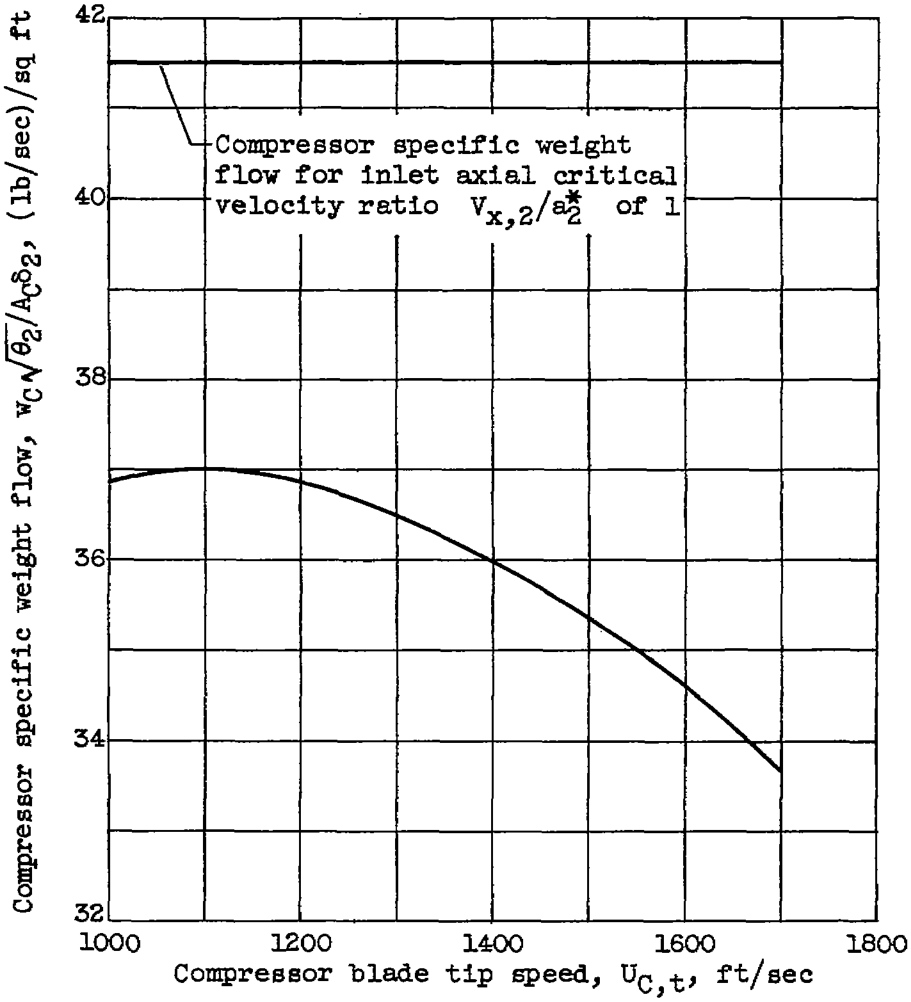
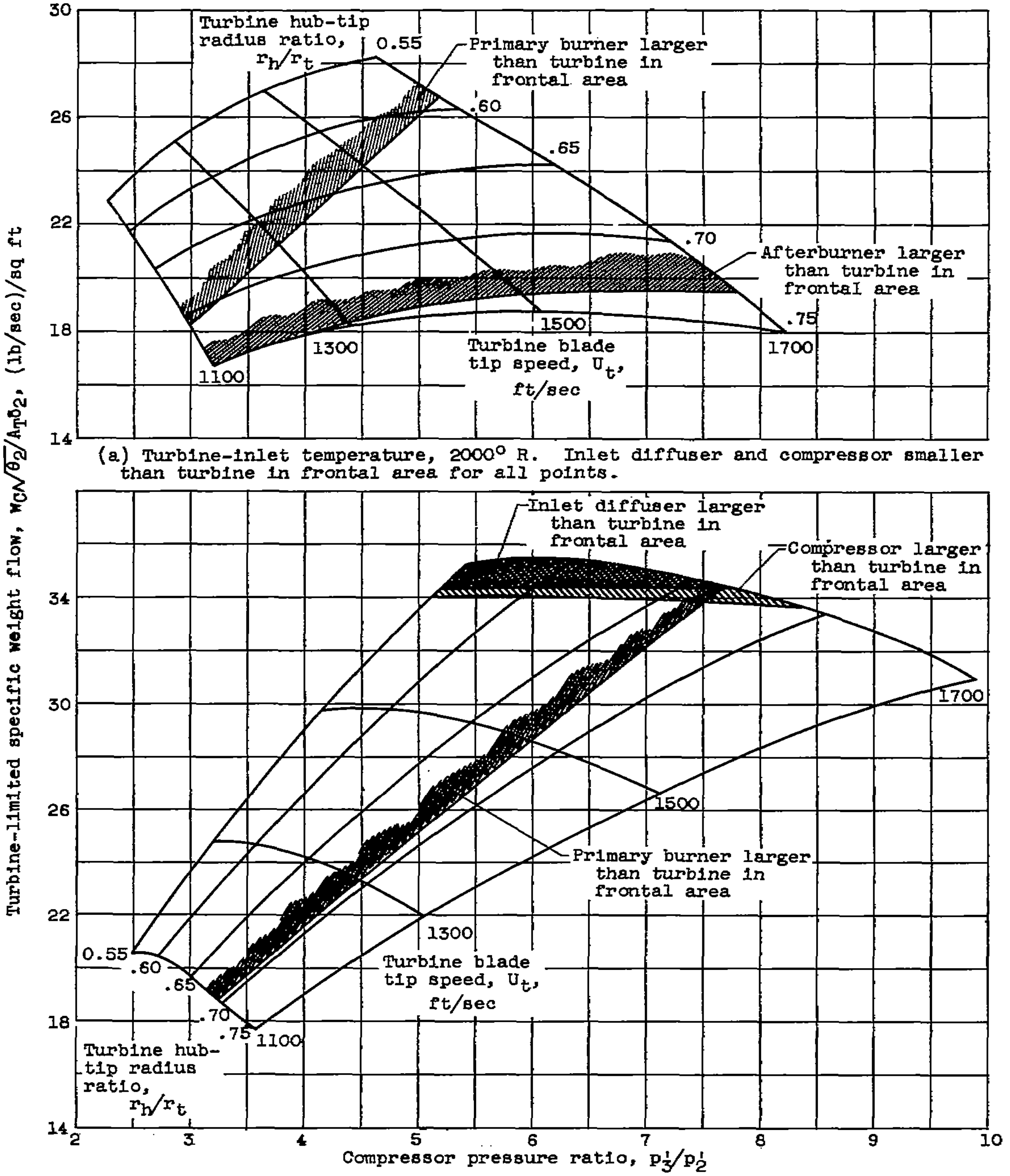


Figure 6. - Effect of compressor blade tip speed on compressor specific weight flow for aerodynamically limited compressor with solid-body-rotation velocity distribution. Inlet axial critical velocity ratio at hub $V_{x,2,h}/a_2^*$, 0.7; inlet relative tip Mach number $M_{R,2,t}$, 1.1; compressor-inlet hub-tip radius ratio $(r_h/r_t)_C$, 0.4.

3171

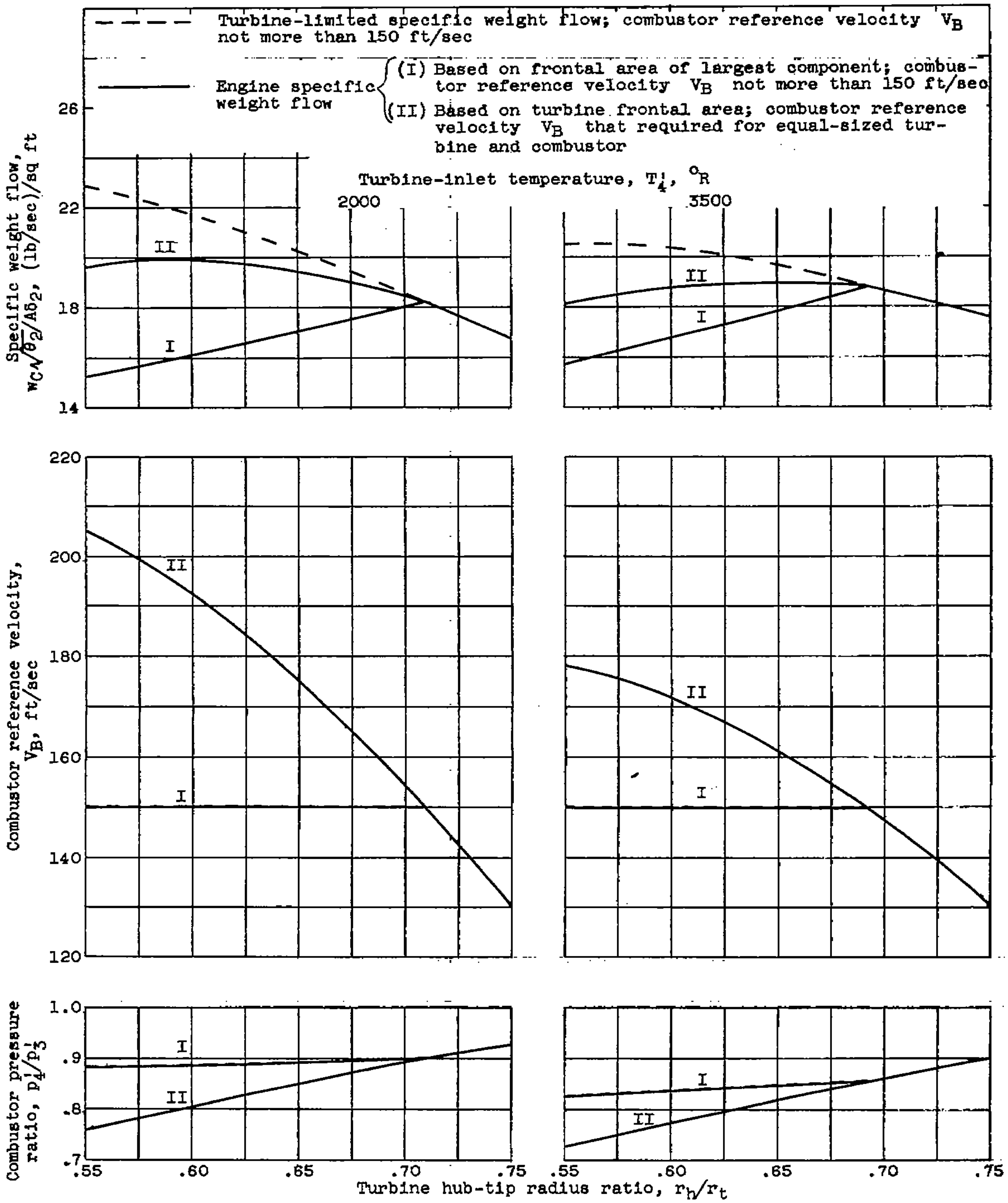


(a) Turbine-inlet temperature, 2000° R. Inlet diffuser and compressor smaller than turbine in frontal area for all points.

(b) Turbine-inlet temperature, 3500° R. Afterburner larger than turbine in frontal area for all points.

Figure 7. - Limitations on turbine design specifications due to relative engine-component frontal areas. Coolant-flow ratio, 0; turbine-exit whirl parameter $V_{u,6,i}/U_i$, -0.410; turbine adiabatic efficiency, 0.83.

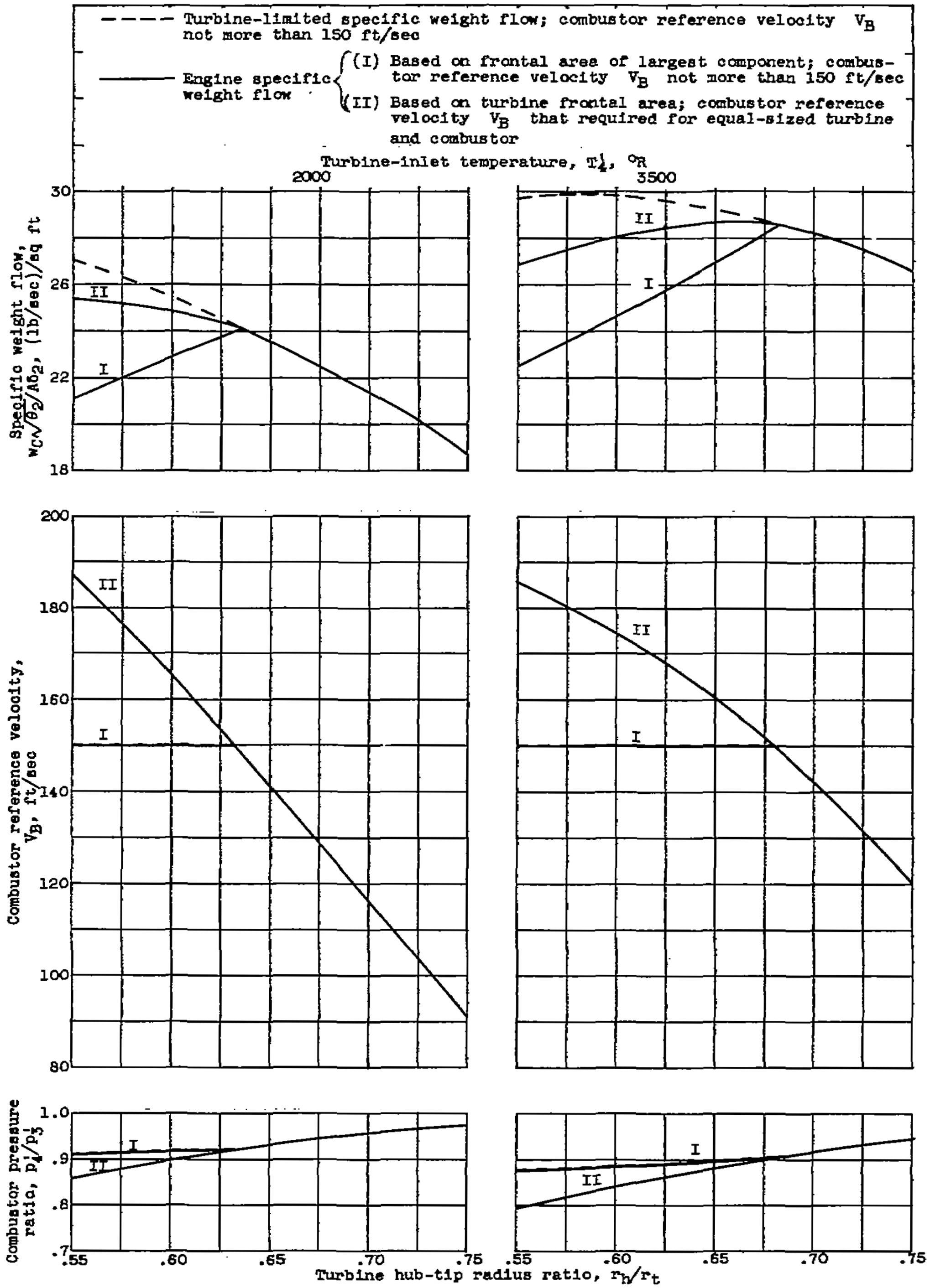
3171



(a) Turbine blade tip speed, 1100 feet per second.

Figure 8. - Effects of increasing reference velocity limit on engine specific weight flow and combustor pressure ratio for nonafterburning engine. Coolant-flow ratio, 0; turbine-exit whirl parameter $V_{u,6,1}/U_1$, -0.410; turbine adiabatic efficiency, 0.83; combustor inner radius to turbine tip radius ratio $(r_h/r_t)_B$, 0.25.

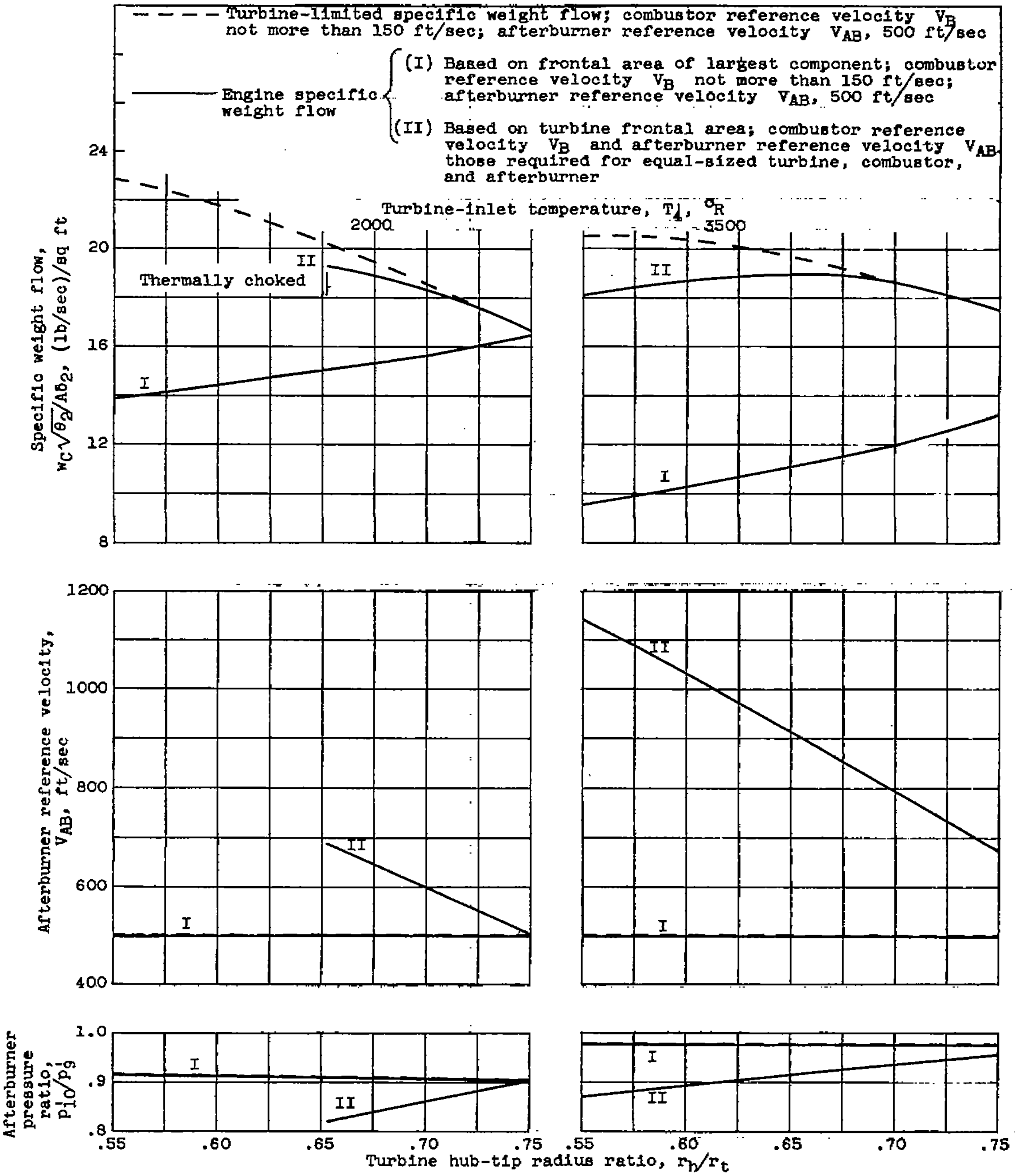
3171



(b) Turbine blade tip speed, 1500 feet per second.

Figure 8. - Concluded. Effects of increasing combustor reference velocity limit on engine specific weight flow and combustor pressure ratio for nonafterburning engine. Coolant-flow ratio, 0; turbine-exit whirl parameter $V_{u,6,i}/U_i$, -0.410; turbine adiabatic efficiency, 0.83; combustor inner radius to turbine tip radius ratio $(r_h/r_t)_B$, 0.25.

3171

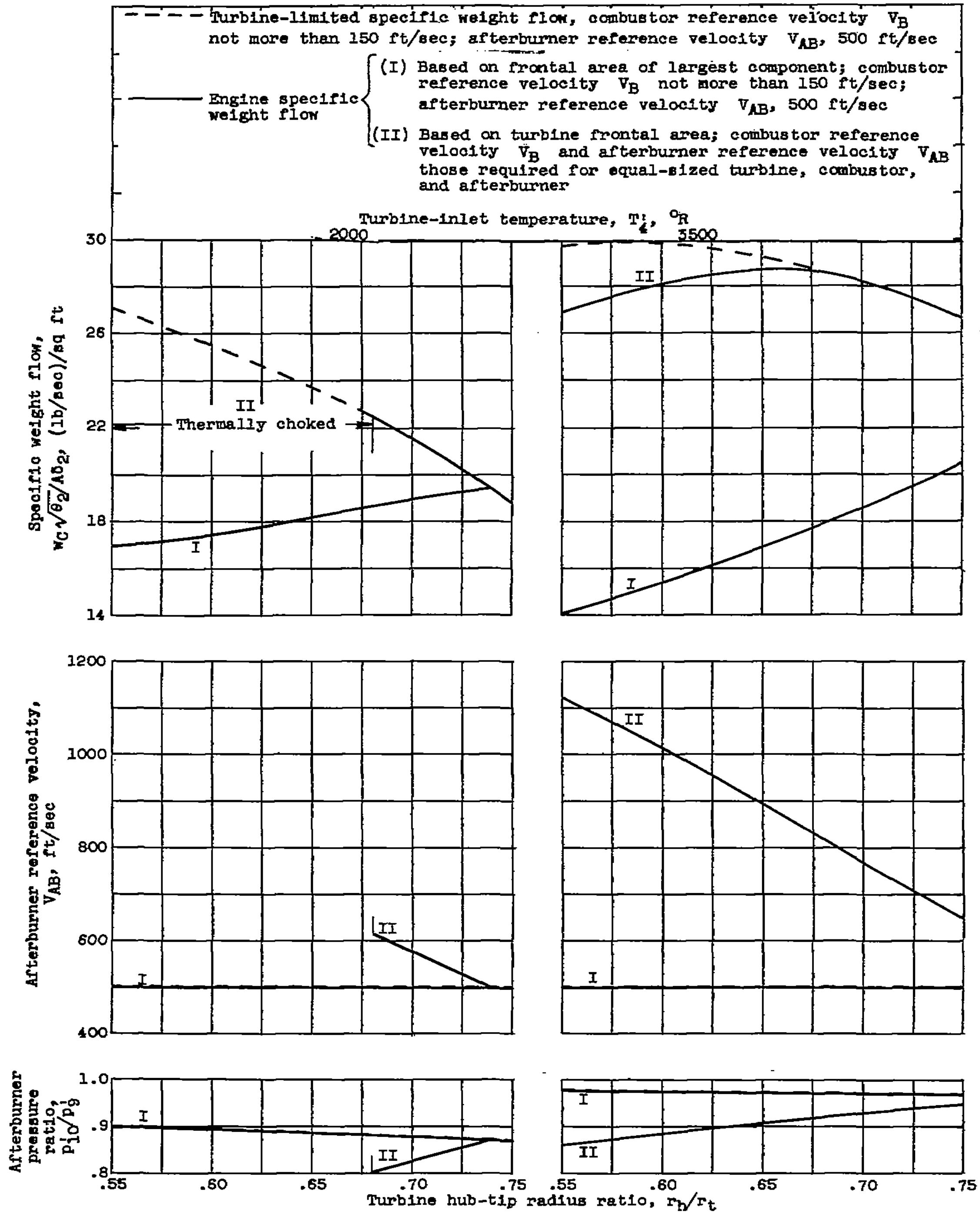


(a) Turbine blade tip speed, 1100 feet per second.

Figure 9. - Effects of increasing afterburner reference velocity limit on engine specific weight flow and afterburner pressure ratio for afterburning engine. Coolant-flow ratio, 0; turbine-exit whirl parameter $V_{u,6,i}/U_i$, -0.410; turbine adiabatic efficiency, 0.83; afterburner-exit temperature, 3500° R.

[redacted]

3171



(b) Turbine blade tip speed, 1500 feet per second.

Figure 9. - Concluded. Effects of increasing afterburner reference velocity limit on engine specific weight flow and afterburner pressure ratio for afterburning engine. Coolant-flow ratio, 0; turbine-exit whirl parameter $V_{u,6,i}/U_i$, -0.410; turbine adiabatic efficiency, 0.83; afterburner-exit temperature, 3500° R.

[redacted]

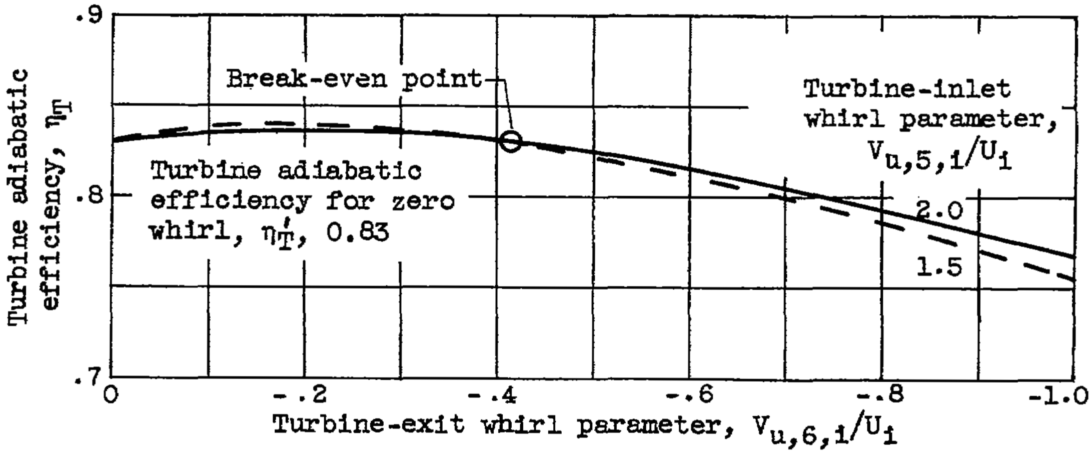


Figure 10. - Variation of turbine adiabatic efficiency with turbine-exit whirl parameter. Turbine adiabatic efficiency for zero exit whirl, 0.83.

NACA-Langley - 5-25-54 - 350

NASA Technical Library
3 1176 01435 3891