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RESEARCH MEMORANDUM

ANALYTICAL INVESTIGATION OF EFFECT OF WATER-COOLED TURBINE BLADES ON PERFORMANCE OF TURBINE-PROPELLER POWER PLANTS

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ANALYTICAL INVESTIGATION OF EFFECT OF WATER-COOLED TURBINE BLADES ON PERFORMANCE OF TURBINE-PROPELLER POWER PLANTS

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SUMMARY

The results of previous NACA investigations on the heat-transfer processes of liquid-cooled turbine blades are applied to an analysis of the performance of a series of six turbine-propeller power plants, each having a multistage turbine equipped with water-cooled blades, and designed for maximum turbine-inlet temperatures of 2000°, 2500°, 3000°, 3500°, 4000°, and 4500° R, respectively. Of greatest interest is the case where continuous operation at turbine-inlet temperatures of 3500° to 4500° R is attained through liquid cooling of the turbine blades by circulating water through cooling passages within the blades.

Methods are developed for computing the power losses incurred by the cooling process, namely: the direct heat loss to the cooling fluid, the pumping power required to circulate the cooling water through the turbine blading and plumbing system, and that part of the turbine-coolant heat-exchanger aerodynamic drag loss caused by friction- and momentum-pressure loss due to heating of the cooling air flowing through the turbine-coolant heat-exchanger core. The relation between optimum-power pressure ratio and engine temperature ratio derived in this report is used to establish the proper pressure ratio for each turbine-inlet gas temperature.

The results of the analysis showed that, with pressure ratios appropriate for the turbine-inlet temperature, the high-turbine-inlet-temperature operation possible through liquid cooling gave equivalent shaft specific power outputs that increased from 164.6 to 775 horsepower per pound per second of compressor air flow as the turbine-inlet temperature was increased from 2000° to 4500° R. Over the same temperature interval, the equivalent shaft thermal efficiency increased

from 43 to 52 percent and the equivalent shaft specific fuel consumption decreased from 0.314 to 0.264 pound per brake horsepower-hour. The specific weight, based on the sea-level take-off horsepower, decreased from 0.8084 to 0.5386 pound per brake horsepower over the same turbine-inlet-temperature range. Although the specific output increased almost linearly with the turbine-inlet temperature, there was little gain in efficiency or specific fuel consumption above a turbine-inlet temperature of 3500° R. Increase in speed or altitude slightly improved efficiency, specific fuel consumption, and specific output, but the specific weight increased steadily with altitude and almost doubled its sea-level take-off value at 35,000 feet.

From the results of the analysis it is concluded that:

1. The improvements in gas-turbine performance gained by operating at the high turbine-inlet temperatures permissible with liquid cooling are not vitiated by the losses incurred by the cooling process.

2. Full utilization of the high operational turbine-inlet temperatures made possible by liquid cooling demands an increase in compressor and turbine pressure ratios from the present range of 4 to 8 to a range from 30 to 40.

3. The required turbine-coolant heat-exchanger capacity and size per unit horsepower for the liquid-cooled turbine engine is approximately 20 percent of that of a liquid-cooled reciprocating engine of the same output.

4. Successful application of liquid cooling to the blading of gas turbines will permit high-temperature turbine blades, wheels, and other parts exposed to the combustion process and products to be constructed entirely from alloy steels such as SAE nickel, chromium, molybdenum, chrome-nickel and chrome-molybdenum series.

INTRODUCTION

The contemporary combustion-gas-turbine power plant operates on the Brayton cycle; slight departures from the ideal efficiency of the cycle are caused by imperfections in the thermodynamic processes. At a given pressure ratio, the specific power output is directly proportional to the turbine-inlet temperature; the ideal cycle efficiency, however, is a unique function of the compressor pressure ratio (reference 1), which must be increased with the turbine-inlet temperature to gain the maximum increase in power output and efficiency made possible by operation at elevated temperatures. Accompanying the increases in specific power output and efficiency resulting from operation at elevated temperatures, and their properly associated pressure ratios, are reductions in the specific fuel consumption, specific weight, and the thrust specific frontal area of the engine.

At the present stage of gas-turbine development, the turbine-inlet gas temperature is limited to approximately 2000° R by the necessity of keeping the components of the engine exposed to the combustion process and products at temperatures low enough to insure mechanical reliability. Although the allowable operating temperature may be increased by air cooling, reference 2 indicates that continuous operation at 2700° R is apparently the ultimate that can be obtained with air cooling.

For several years a theoretical study of the liquid cooling of turbine blades has been in process at the NACA Cleveland laboratory as a means of circumventing the materials limitation and thereby providing practical and continuous operation at the highest constant-pressure combustion temperatures possible with commercial liquid fuels. The analytical investigations of references 3 and 4 show that very effective cooling of turbine blades can be obtained by circulating water through cooling passages of circular cross section within the blade. Such cooling will permit continuous operation at the gas temperatures (3600° to 4000° F) obtainable with stoichiometric mixtures, with the simultaneous assurance that average blade-metal temperatures will be kept within safe limits for alloy steels such as SAE nickel, chromium, molybdenum, chrome-nickel, chrome-molybdenum series, and possibly certain high-conductivity materials such as aluminum.

Although the application of liquid cooling to gas turbines offers the possibility of improved performance, it incurs additional losses that must be sustained by the power plant. Consequently, any study of the high-temperature performance of the gas-turbine power plant, which assumes such operation to be obtained through liquid cooling, must include a study of the following losses:

(1) The direct heat loss to the cooling fluid

(2) The pumping power required to circulate the coolant

(3) The aerodynamic drag of the turbine-coolant heat-exchanger core

Methods for computing these losses are developed herein and are used to evaluate the performance of six multistage liquid-cooled turbine-propeller power plants designed for operation at combustion-gas temperatures of 2000°, 2500°, 3000°, 3500°, 4000°, and 4500° R, respectively. For each engine, the maximum-power pressure ratio for operation at a flight speed of 600 miles per hour and an altitude of 35,000 feet (fig. 1) is selected and is computed from the relation between compressor pressure ratio and engine temperature ratio developed in appendix A. This range of pressure ratios extends from 11 at a turbine-inlet temperature of 2000° R to 42 at a temperature of 4500° R. In order to simplify comparison and facilitate the calculation of performance at flight conditions other than those for which the compressor pressure ratio is selected, it is assumed that the compressor element in all the engines is designed for the same rated air flow and operated at constant nondimensional speed and constant nondimensional weight-flow parameters. These assumptions insure operation at constant efficiency and constant pressure ratio for each engine over the entire range of flight conditions covered, and a sensibly constant physical diameter for all of them. By combining the equation of continuity with the expression found herein that defines the relation between compressor pressure ratio and engine temperature ratio, it is easily shown that the required cross-sectional flow area of the turbine decreases slowly as the turbine-inlet temperature is increased. Physically, for optimum power, the ratio of the design turbine-inlet pressures increases faster than the ratio of the design inlet temperatures and the increase in specific volume of the combustion gas associated with the increase in turbine-inlet temperature is more than compensated for by the effect of increasing turbine-inlet pressure in decreasing the specific volume. For the variation of compressor pressure ratio with turbine-inlet temperature used in this report, the assumption that the engine diameter does not change with the turbine-inlet temperature therefore leads to a conservative value of the weight estimate. In order to facilitate further the preparation of a weight estimate, the analysis is based on a representative axial-flow jet-propulsion engine having a sea-level static air flow of 55 pounds per second. Where necessary, the data on the engine are supplemented by data taken from the catalogs of manufacturers of aircraft power-plant components and accessories. In order to insure uniformity in comparison of the results, the fundamental equations for polytropic processes are used in computing compressor and turbine work because available special gas-turbine cycle-analysis methods do not cover the range of cycle temperatures and pressure ratios used in this report.

It is assumed that the combustion chamber is lined with ceramic materials that will require no cooling, and there is ample reason to assume that ancillary heat losses can be reduced to a negligible minimum. The analysis therefore assumes that the use of cooling extends only to the turbine blades. The large pressure ratios associated with the combustion temperatures insure that in all cases most of the thermal energy is extracted in the turbines and the exhaust-gas temperature is not high enough to warrant cooling of the tail pipe.

For each engine, calculations are made for five flight speeds increasing in 100-mile-per-hour increments from 200 to 600 miles per hour at each of five altitudes in a range extending from sea level to 35,000 feet.

The results of the calculations are presented as graphs of brake specific fuel consumption, net equivalent shaft horsepower, engine shaft thermal efficiency, specific weight, and the variation of the specific power losses. In every instance the specific performance variables are based on the power that the engine delivers to its propulsion elements because it is desired to evaluate only the effect of cooling on the brake power output of the engine. The specific-power-loss parameters, however, are based on the fuel horsepower because they are functions of the fuel horsepower and rotative speed of the engine rather than of its net output.

Finally the work of the report is applied exclusively to consideration of the turbine-propeller power plant because previous performance analyses of the various gas-turbine propulsion systems combined with recent improvements in propeller design indicate that it is this combination which, with the high pressure ratios required for high-temperature operation, will give the best over-all performance in the speed ranges considered.

SYMBOLS

The following symbols and abbreviations are used in the report:

C dimensionless coefficient in heat-transfer equation

c_p specific heat at constant pressure, Btu per pound per $^{\circ}F$

D characteristic linear dimension of engine, feet

d internal diameter of engine shell casting, inches

f specific fuel consumption, pounds per brake horsepower-hour

G engine air mass velocity, pounds per second per square foot of cross-sectional area of flow

g acceleration due to gravity, 32.2 feet per second per second

H enthalpy, Btu per pound

h heat-transfer coefficient, Btu per hour per square foot per °F

J specific power, horsepower per pound of fluid

K exhaust-nozzle coefficient

k thermal conductivity of combustion gas, Btu per hour per foot per °F

L length of engine shell, inches

M weight flow of fluid, pounds per second

N rotational speed, rpm

n polytropic exponent

P total pressure, pounds per square foot

p static pressure, pounds per square foot

Q turbine heat-rejection rate to coolant, Btu per hour

R gas constant, foot-pounds per pound per °F

r radial length from rotation axis to extremity of cooling passage, feet

S heat-transfer area per turbine stage, square feet

T total temperature, °R

T_{corr} corrected total temperature, °R

t static temperature, °R

V velocity, feet per second

W installed weight of power plant, pounds

w weight of engine component, pounds

X installed weight of engine, pounds

x linear distance, feet

Z number of turbine stages

α constant factor in exponent

γ ratio of specific heats

Δ increment

δ thickness of equivalent steel engine shell, inches

η efficiency

μ viscosity of combustion gas, pounds per second per square foot

ρ density, slugs per cubic foot for fluids or pounds per cubic foot for metals

σ altitude density ratio, (p/p,)

ψ perimeter, feet

Ω angular velocity, radians per second

f/a fuel-air ratio, pounds of fuel per pound of air

hp power, horsepower

Subscripts:

A altitude

8 air

b blade

C coolant passage

c compressor

corr corrected

e engine shell

F	friction due to resistance to fluid flow in blade cooling passage
f	fuel
g	gas
h	turbine-coolant heat exchanger
i	metal-to-coolant inside blade cooling passage
j	jet
L	liquid cooled
l	coolant
m	mechanical
o	gas-to-blade on outside of blade
P	propeller
p	coolant pumping or coolant pump
r	propeller reduction gear
S	shaft (used with η to denote shaft thermal efficiency)
SL	sea level
s	steel
t	turbine
u	uncooled
x	point on turbine expansion-path length
y	base values
z	general values
0	free stream
1	compressor inlet or front face of turbine-coolant heat-exchanger core

2 compressor outlet

2, h downstream or outlet face of turbine-coolant heat-exchanger core

3 turbine Inlet

3, h turbine-coolant heat-exchanger duct outlet

4 turbine outlet

5 exhaust-nozzle outlet

6 point on rotation axis of turbine

7 extremity of cooling passage

net used with hp to denote net equivalent shaft horsepower of power plant

The location at' the stations is shown in figures 2 and 5.

ANALYSIS

The analysis of the performance of a water-cooled turbine-propeller engine assumes an engine having essentially the components and arrangement shown in figure 2. Except for the provision for handling the cooling fluid, the arrangement follows conventional practice; the temperature-entropy and the pressure-volume diagrams for the cycle are shown in figure 3. Cooling water enters the turbine wheel from the turbine shaft and is circulated through U-shaped passages in each blade of the turbine. The water is returned through the turbine shaft to a ducted turbine-coolant heat exchanger, which is exposed in the air stream, and after cooling is recirculated through the system.

The performance analysis will be developed in the following order:

(1) Derivation of the equations for over-all power-plant performance

(2) Formulation of the equations for power losses due to liquid cooling

(3) Calculation of the installed. weight of the power plant

(4) Discussion of the cycle temperatures and pressures

Over-all Power-Plant Performance

Net propulsive horsepower of power plant. - The net equivalent shaft horsepower delivered by the liquid-cooled turbine-propeller power plant to its propulsion elements is defined as the difference between the power developed in the turbine and the jet, and the sum of the power absorbed by the compressor, lost to the cooling fluid, used to circulate the cooling fluid through the turbine blading and plumbing system, and that lost to the aerodynamic drag of the turbine-coolant heat-exchanger core. Auxiliary drag increments induced by the turbine-coolant heat-exchanger installation are neglected. If the net equivalent shaft horsepower is hp_{net}

$$hp_{net} = hp_t - hp_c - hp_l - hp_p + hp_j - hp_h \tag{1}$$

The sum of the first four terms on the right-hand side of equation (1) represents the shaft horsepower delivered by the engine to the propeller. The two terms hp_l and hp_p are the power lost to the cooling fluid by heat conduction through the turbine blades and the power expended in the coolant circulating process, respectively. The term hp_h represents that part of the aerodynamic drag of the turbine-coolant heat-exchanger installation due to friction and momentum pressure losses arising from heating of the cooling air in the heat-exchanger core. In general, the calculations show that a net thrust is obtained from the turbine-coolant heat-exchanger core in the speed ranges of primary interest, but because the drag effect of the heat-exchanger duct is neglected, the thrust or drag effect of the turbine-coolant heat-exchanger core is treated in the section of the report dealing with losses.

Calculation of the power developed in the turbine presupposes a polytropic expansion process using a polytropic exponent calculated from an average value of the ratio of specific heats of the combustion gas on the basis of an assumed adiabatic stage efficiency. The value of the specific-heat ratio is arbitrarily selected so that the combination of values for the specific-heat ratio and the polytropic exponent give turbine-work values that are in close agreement with those obtained from constant-volume alinement-chart calculations. By trial-and-error methods the necessary value of γ_t was found to be 1.22. On this basis the power developed in the turbine is

$$hp_t = \frac{\eta_{m,t}\gamma_t R_t T_3}{(\gamma_t - 1)} \left[1 - \left(\frac{P_4}{P_3}\right)^{\frac{n_t-1}{n_t}}\right] \frac{M_a}{550}\left(1 + \frac{f}{a}\right) \tag{2}$$

The characteristic gas constant R_t in equation (2) is evaluated at the arithmetic mean of the turbine-inlet and ideal turbine-exhaust gas total temperatures by the methods and data of reference 5,

It is admitted that the methods used to calculate the turbine power 'do not precisely represent the complicated thermodynamic process that occurs in the liquid-cooled turbine. The methods used in this report are sufficiently accurate, how-ever, for engineering calculations, because their end result agrees with that of an accurate stage-by-stage calculation within 5 percent.

Similar considerations govern the calculation of the power absorbed by the compressor except that the polytropic exponent is calculated from average values of the ratio of specific heats for the compression process from the data of *reference 6. The* compressor power is

$$hp_c = \frac{\gamma_c R_c T_1}{\eta_{m,c}(\gamma_c - 1)} \left[\left(\frac{p_2}{p_1}\right)^{\frac{n_c - 1}{n_c}} - 1 \right] \frac{M_a}{550} \qquad (3)$$

After the gases leave the last stage of the turbine, they are expanded through the exhaust nozzle with an exit velocity

$$V_j = K \sqrt{2g \frac{\gamma_t R_t T_{4,corr}}{(\gamma_t - 1)} \left[1 - \left(\frac{p_0}{p_4}\right)^{\frac{\gamma_t - 1}{\gamma_t}} \right]} \qquad (4)$$

The ratio of specific heats for the turbine expansion process may be used here because the exhaust-nozzle pressure ratios used in this report are so low that calculations show that no error results. The thrust horsepower of the jet is

$$hp_j = \frac{V_0 M_a}{17{,}700 \eta_p} \left[\left(1 + \frac{f}{a}\right) V_j - V_0 \right] \qquad (5)$$

By the postulation that the compressor is run at a constant nondimensional speed, the compressor-air weight flow M_a is a function of the state of the air at the compressor inlet and will vary with the flight speed and altitude of *the* airplane. If N is the design speed at which the compressor delivers its rated sea-level performance, reference 7 shows that if for all other inlet conditions it is run at a constant nondimensional speed

$$ND\sqrt{\frac{\rho_{a,1}}{P_1}}$$

its nondimensional weight flow

$$\frac{M_a}{gD^2\sqrt{P_1\rho_{a,1}}}$$

is constant *for* all inlet oonditiona. Evaluation of the nondimensional weight-flow parameter for the design conditions at which the compressor delivers its design flow permits calculation of the weight flow at other compressor-inlet conditiona.

Calculation of the power outputs and power losses is baaed on data for a representative jet-propulsion engine, supplemented by information gathered from the catalogs of the manufacturers of air craft power-plant components and accessories. In this instance, specific relations are probably of greater significance than absolute magnitudes, and the results are presented in the form of the following specific performance parameters.

Specific power output. - The specific parer output is defined as

$$\frac{hp_{net}}{M_a}$$

Specific fuel consumption. - The specific fuel consumption f is the same as that used in rating reciprocating engines, namely pounds of *fuel* per brake horsepower-hour.

Specific weight. - The specific weight is defined as

$$\frac{W}{hp_{net}}$$

Power-plant brake thermal efficiency. - The brake thermal efficiency of the power plant is defined as

$$\frac{hp_{net}}{hp_f}$$

Specific heat loss. - The specific heat lose is defined 88

$$\frac{hp_l}{hp_f}$$

Specific pumping power. -The specific pumping power is defined as

$$\frac{hp_p}{hp_f}$$

Power Lasses

Direct heat loss to coolant. - For&turbine blade with cooling passages extending almost the entire length of the blade, reference 3 shows that the difference between the combustion-gas temperature and the average blade temperature is approximately

$$T_g - T_b = C\,(T_g - T_l) \tag{6}$$

Here T_g, T_b, T_l are the values of the free-stream total temperature of the combustion gas, the average blade-metal temperature, and the average temperature of the coolant flowing in the blade passage, respectively. The factor C is a nondimensional constant whose value depends on the liquid-to-metal and the gas-to-metal heat-transfer coefficients, and the right-section perimeters of the coolant passage and turbine blade.

Equation (6) gives the temperature difference between gas stream and turbine blade at any point along the expansion path through a multistage turbine, but as the gas expands through the turbine both temperature and pressure decrease. Accurate determination of the heat loss therefore requires knowledge of the variation of gas temperature with turbine expansion-path length. A preoiee calculation of the heat lose requires a step-by-step rotor-to-et&or calculation through the turbine because of the discontinuous variation in temperature difference occasioned by the abrupt changes in relative gas velocity between turbine blades and nozzle blades. Although the process is not difficult, It is a tedious one; the method presented herein permits a rapid computation of the heat loss, yet gives results that vary no more than 5 percent from those of the more preoiee method.

Each stage of the turbine consists of a row of rotor blades and a row of stator blades. The cross sections of stator and rotor blades are usually similar and for analytical purposes they will be assumed to be the same; whereupon it follows that the length of the expansion path per stage is approximately equal to the perimeter of one blade.

Unpublished data indicate that turbine-stage pressure ratios of 5 are possible without serious losses in efficiency, a value that will not be exceeded in the work of this report. The over-all pressure ratio of the turbine is P_3/P_4 and therefore the minimum number of stages required is

$$Z = \frac{\log_{10} \frac{P_3}{P_4}}{\log_{10} 5} \qquad (7)$$

In general, Z as determined by equation (7) will be an irrational number; moreover, considerations of design and performance will exert additional influence on the final choice of the number of stages and the division of power among them. In approximating the heat loss, however, two stages are assumed for the over-all pressure-ratio range $0 < P_3/P_4 < 16$, three stages are assumed in the range $16 < P_3/P_4 < 50$, and the turbine-stage pressure ratio is taken as $\sqrt[Z]{P_3/P_4}$.

At any point x along the expansion path through the turbine, the temperature of the combustion gas is

$$T_g = T_3 \left(\frac{P_x}{P_3}\right)^{\frac{n_t - 1}{n_t}} \qquad (8)$$

Both the over-all pressure ratio and the stage pressure ratio for the turbine are known; hence, by taking values of Px at the junction of adjacent stages, a graph of gas temperature plotted against turbine expansion-path length may be constructed. The approximate curve thus obtained, together with the actual curve for a six-stage turbine, are shown in figure 4. The approximating curve is exponential in form and can be described by an empirical equation of the form

$$T_g = T_3 e^{\alpha x} \tag{9}$$

Knowledge of the pressure and temperature at any point on the curve makes possible the evaluation of α. Substitution of the equivalent of T_g in equation (6) gives for the temperature difference at any point through the turbine

$$T_g - T_b = C\left(T_3 e^{\alpha x} - T_2\right) \tag{10}$$

At any point along the turbine, the heat loss to the coolant is

$$\frac{dQ}{dx} = Ch_o \frac{S_o}{\Psi_b}\left(T_3 e^{\alpha x} - T_l\right) \tag{11}$$

where S_o/Ψ_b is the total heat-transfer area per unit length of turbine expansion-path length, and h_o is the gas-to-blade heat-transfer coefficient, From reference 3 it can be shown that

$$C = \frac{1}{1 + \frac{h_o \Psi_b}{h_i \Psi_C}} \tag{12}$$

where ho and h_1 are the gas-to-blade and metal-to-coolant heat-transfer coefficients, respectively, whioh are computed from the methods and data of reference 8, and Ψ C and Ψ_b are the right-section perimeters of the cooling passage and of the turbine blade, respectively. For a fixed gas flow and turbine-inlet gas temperature, the right-hand member of equation (12) changes only slightly and C maybe evaluated at the turbine inlet and considered constant for a given set of operating conditions.

The basic value of the heat-transfer coefficient h_o is obtained from data given in reference 4. In reference 8 (p. 221, fig. 111), from which the data for the calculation of h_o are taken, it is shown that in the range of gas-flow Reynolds numbers used in turbine cooling, the relation between h_o, μ, G, and k is *accurately given* by

$$h_o \propto k \left(\frac{G}{\mu}\right)^{0.8} \tag{13}$$

Knowledge of h_o at one set of turbine-inlet and weight-flow conditions permits its computation at any other set of conditions by ratio and proportion. For a given engine-air weight flow and turbine-inlet temperature, h_o also varies only slightly through the turbine and neglecting variation along the flow path in h_o and C, the heat loss from the turbine to its coolant is given by the integral of equation (11) over the length of the turbine expansion path and is

$$hp_l = \frac{778Q}{1.98 \times 10^6} = \frac{778}{1.98 \times 10^6} Ch_o \frac{S_o}{\psi_b} \int_0^{Z\psi_b} \left(T_3 e^{\alpha x} - T_l\right) dx$$

$$= \frac{778}{1.98 \times 10^6} Ch_o \frac{S_o}{\psi_b} \left[\frac{T_3}{\alpha}\left(e^{\alpha Z\psi_b} - 1\right) - T_l Z\psi_b\right] \tag{14}$$

<u>Coolant-pumping power loss.</u> - Experiments by Föttinger (reference 9) on the resistance to fluid flow in U-shaped channels rotating at high angular velocities show that in many instances the resistance to suoh flow may be several hundred times that computed by conventional fluid-flow formulas and methods. No rigorous treatment of this phenomenon has yet been developed and the power required to pump the coolant through the rotating blade passage will be approximated 'from energy considerations by treating the blade passage as a hydraulic pump with an efficiency of η_p.

A sketch of the cooling system is shown in figure 5. The coolant enters the blade at its base in the rim of the wheel, travels around the passage in the direction indicated by the arrows, and returns to the wheel. The coolant enters the wheel at the axis of rotation and is therefore, in effect, pumped from the center of the wheel (station 6) to the extremity of the blade passage. At the axis of the wheel the translational velocity of the fluid in the passage due to rotation of the wheel V_6 is equal to zero. At the outer extremity of the cooling passage, this velocity is V_7, which is equal to Ωr_7 (where Ω is the angular velocity of rotation of the wheel and r_7 the radial length from the axis of the wheel to the extremity of the blade passage), If M_l is the rate of flow of the coolant through the wheel, the power required to pump the coolant from the axis of rotation to the extremity of the passage is

$$J_{6-7} = \frac{M_l}{550g}\left(\Omega r_7 V_7 - \Omega r_6 V_6\right) \tag{15}$$

If there were no losses, the system would be a conservative one and the power that would be returned to the wheel by the fluid in flowing from the extremity of the passage to the rotation axis would be given by equation (15). Losses, however, render the system imperfeot and, if the pumping process is assumed to be carried out with an efficiency of η_p, the power available at the extremity of the coolant passage is

$$\eta_p J_{6-7} = \eta_p \frac{M_l}{550g}\left(\Omega r_7 V_7 - \Omega r_6 V_6\right) \tag{16}$$

The power returned to the wheel in the return-flow process is

$$J_{7-6} = \eta_p^2 J_{6-7} = \eta_p^2 \frac{M_l}{550g}\left(\Omega r_7 V_7 - \Omega r_6 V_6\right) \tag{17}$$

and the net pumping power is

$$J_{6-7} - J_{7-6} = \frac{Id_l}{550g}\left(\Omega r_7 V_7 - \Omega r_6 V_6\right)\left(1 - \eta_p^2\right) \qquad (18)$$

To this power must be added hp_F, the power required to overcome the fluid frictional resistance in the coolant passages of the stator blades. Methods for the calculation of this fluid-friction power loss can be found in reference 10. The pumping power required to circulate the coolant through the heat-exchanger core is computed from information from the Harrison Radiator Corporation, and the pumping power expended in the heat-exchanger plumbing is arbitrarily taken as one-half the heat-exchanger pumping power. As computed, these losses are negligible, and assessing their sum at 1 percent of the turbine pumping power is more than sufficient. The total pumping power is then

$$hp_p = 1.01Z\left(J_{6-7} - J_{7-6} + hp_F\right) \qquad (19)$$

In estimating the coolant pumping power for the turbine, consideration has been limited to the blade cooling passage proper. In practice, the rotor will probably be hollow with the cooling water occupying the chamber thus formed and passing from it to the blade passage. Any friction loss in such a chamber would be that between contiguous elements of fluid and would be very small; this loss is therefore neglected.

Turbine-coolant heat-exchanger-core drag loss. - Turbine-coolant heat-exchanger-core drag estimates are based on the premise that the heat exchanger is well ducted and equipped with a variable-area exit that maintains a constant coolant-exit temperature by varying the weight flow of cooling air through the heat exchanger. This type of heat-exchanger installation gives the minimum pressure drop across the heat-exchanger core and therefore the least core drag or maximum core thrust under all operational conditions. A pressurized system is assumed, calculations are based on coolant entering the heat exchanger at a temperature of 250° F, and a heat-exchanger-core area sufficient to provide adequate cooling for sea-level operation at a flight speed of 200 miles per hour is selected.

A sketch of the heat-exchanger element of the cooling system is shown in figure 5. Air at station 0 in the free stream of the atmosphere is diffused in the intake duct as static pressure at the face of the heat-exchanger core; perfect recovery of the dynamic pressure of the free stream is assumed. It can be assumed with negligible error that the heat-rejection rate of the coolant to the turbine-coolant heat exchanger equals that of the engine to the coolant and the required turbine-coolant heat-exchanger air flow and associated heat-exchanger-m pressure drop may be read off performance charts of Beat-exchanger blook tests (information from Harrison Radiator Corp.).

The process in the exit section of the turbine-coolant heat-exchanger duct is treated in the same way as the process in the engine exhaust nozzle and use of equations (4) and (5), with appropriate values for γ and T (reference 6), and f/a set equal to 0, gives the exit velocity of the cooling air from the turbine-coolant heat exchanger and Its thrust or drag horsepower.

Engine-Weight Analysis

In order to obtain a substantially correct value of the installed weight of the liquid-cooled turbine-propeller engine, the weight estimate is based on a representative production axial-flow jet-propulsion engine and a oomposite selection of physical dimensions and air-capacity data on several similar engines that have attained operational success. The size and capacity of starters, pumps, and auxiliaries in this type of power plant are primarily functions of engine size rather than engine output and any increase in the diameter of the liquid-cooled engine over that of the uncooled engine is assumed to be negligible. Consequently, the principal contributions to increased weight in the cooled engine will arise from the following causes:

(a) Increase in thickness of engine shell to withstand increased pressures because the optimum-power pressure ratios used in the liquid-cooled engine are higher than current values

(b) Addition of turbine-coolant heat-exchanger plumbing and coolant

(c) Addition of coolant pump

(d) Increase in number of compressor and turbine stages to accommodate increased pressure ratios

(e) Addition of propellers because representative engines furnishing data for the analysis are jet-propulsion engines

(f) Addition of propeller reduction gear

(g) Increase in compressor weight because rear half of compressor must be made of steel to withstand high air temperatures associated with optimum-power pressure ratios

<u>Increase in engine-shell weight.</u> - The increased weight of the engine shell will be taken as equivalent to the weight of a cast steel shell whose internal diameter is that of the representative machine, and of a thickness 8 such that the difference in maximum pressure between the liquid-cooled engine and the uncooled engine will develop a hoop stress of 40,000 pounds per square inch in the material of the steel shell. If d denotes the internal diameter of the aluminum shell of the representative engine, and 8 the thickness of the stell shell in inches 971

$$\delta = \frac{\left(P_{2,L} - P_{2,u}\right)}{144 \times 80,000} d$$

The length L of the equivalent steel shell is assumed to be equal to the length of the aluminum shell for *a* representative jet engine and its weight is

$$w_e = \frac{\pi \rho_s L}{4 \times 1728} \; 28 \; (2d + 28) \qquad (20)$$

<u>Weight of turbine-coolant heat *exchanger and* coolant.</u> - The weight of the turbine-coolant heat exchanger and its attendent coolant is assessed at 20.9 pounds per cubic foot, which is the weight of a suitable production-prototype aluminum heat-exchanger design. The value allowed for the plumbing and its attendant coolant is one-quarter of the turbine-coolant heat exchanger and its attendant coolant weight.

<u>Increasein weight of turbine.</u> - It is assumed that the turbine stages added to accommodate the increased pressure ratio are the same as those of the representative turbine used as a basis for the analysis. The weight of a complete turbine stage consisting of rotor disk, blades, and nozzle diaphragm is taken as 204 pounds.

<u>Weight of coolant pump.</u> - The weight of a commercial gear-type water pump having the required capacity varies from 15 to 30 pounds and is arbitrarily assessed at 30 pounds.

<u>Weight of compressor.</u> - The increased compressor pressure ratio requires an increase in the number of compressor stages and consequently au increase in the weight of the compressor. Component-weight data on several successful representative jet engines having aluminum-alloy compressors indicates that the ratio of compressor weight to turbine weight ranges from 1.6 to 2.4; an average value of 2 is

chosen for this analysis. In the cases under consideration, the temperature of the air in the latter stages of compression is above the level where aluminum alloys are suitable; the rear half of the compressor will therefore be assumed to be made of steel and, consequently, the initial compressor-turbine weight ratio must be multiplied by one-half the ratio of the densities of steel and aluminum to get a corrected value of 2.8.

Weight of reduction gear. - Propeller reduction-gear units having efficiencies of 95 percent at a reductionratio of 10:1 can be currently built at a specific weight of 0.05 pound per transmitted horsepower (reference 11). This value is used in forming the weight estimate.

Weight of propeller. - A survey of the available data on uncooled turbine-propeller engines shows that the ratio of propeller weight to shaft horsepower varies from 0.437 to 0.188. A value of 0.302 will be used in forming the weight estimates.

Cycle Temperatures and Pressures

The combustion-gas turbine operates on the Brayton cycle. A sketch of the engine with the stations numbered at which the important events of the cyole occur is shown in figure 2, and in figure 3 are shown the pressure-volume and temperature-entropy diagrams for one cycle of operation. Compression begins in the atmosphere at some point 0 and it is assumed that the sir obeys the Bernoulli equation for compressible flow of gases. It can be assumed with but negligible error that full recovery of the free-stream dynamic pressure as static pressure is effected in the intake ducts of the ccmpregsor and heat exchanger (reference 12) and the pressure at the face of the heat-exchanger core and compressor intake is represented as follows:

$$p_1 = p_0\left(1 + \frac{\gamma_0 - 1}{2}\,\frac{V_0^2}{\gamma_0 g R_0 T_0}\right)^{\frac{\gamma_0}{\gamma_0 - 1}} \qquad (21)$$

The temperature of the diffused air is

$$T_1 = t_0 + \frac{(\gamma_0 - 1)V_0^2}{2g\gamma_0 R\, t_0} \tag{22}$$

The design compressor pressure ratio for optimum power is

$$\frac{P_2}{P_1} = 0.949 \left(\eta_{m,t}\eta_{m,c} \frac{T_3}{T_1}\right)^{\frac{n_c}{2(n_c - 1)}} \tag{23}$$

which is derived in appendix A, and the temperature of the air at the compressor outlet is

$$T_2 = T_1 \left(\frac{P_2}{P_1}\right)^{\frac{n_c - 1}{n_c}} \tag{24}$$

The fuel-air ratio f/a for the specified turbine-inlet temperature is obtained from the charts of reference 13. Perfect combustion may be assumed because of the high precombustion temperature and pressure of the air, but a 2-percent pressure loss is assumed to occur between compressor outlet and turbine inlet; hence

$$P_3 = 0.98\ P_2 \tag{25}$$

The turbine-outlet pressure is held to a constant value of 0.90 P_1. This value is less than ambient pressure at some of the lower flight speeds and in these cases the turbine is allowed to exhaust to ambient pressure with negligible exit velocity and jet thrust. When knowledge of its value is required, the exit temperature of the combustion gases from the turbine is obtained by computing its enthalpy from the relation

$$H_4 = H_1 + \Delta H_c + \Delta H_f - \Delta H_t \tag{26}$$

where ΔH_c and ΔH_f are the enthalpy increases in the compression ad combustion processes, respectively (references 6 and 13) and

ΔH_t is the enthalpy drop in the expansion process through the turbine. The temperature is then obtained from the combustion-air temperature-enthalpy charts of reference 14.

The application of the foregoing methods is illustrated by a sample calculation in appendix B.

RESULTS OF CALCULATIONS

Over-all Power-Plant Performance

Specific power output. - The variation of the specific power output of the water-cooled turbine-propeller power plant with turbine-inlet temperature is shown in figure 6. It is immediately apparent that the high specific power output made possible by high-temperature operation is not rendered unavailable by the losses incurred in the cooling process. At the design speed and altitude (600 mph and 35,000 ft, respectively), the specific power output increases almost linearly *f'rm184.6* horsepower per pound of compressor air flow per second at a turbine-inlet gas temperature of 2000° R to 506.0 horsepower per pound of compressor air flow per second at a turbine-inlet temperature of 3500° R. At a given turbine-inlet gas temperature, an increase in efficiency and specific power output accompanies an increase in either the flight speed or the altitude. This effect is caused by an increase in the engine air flow as the speed increases and the decrease in compressor work associated with reduction in ambient temperature as the altitude is increased.

In the interests of generality, the results have been given on the basis of equivalent shaft horsepower per pound of compressor air flow, but the calculations show that a turbine-propeller engine, 21 inches in diameter, designed *for a sea-level* static air flow of 55 pounds per second, could develop 24,850 net equivalent shaft horsepower for take-off without appreciable increase in diameter if operated at 8 turbine-inlet temperature of 3500° R and a compressor pressure ratio of 27.64.

With a given *compressor* gas-turbine power plant, the reduction in air flow with altitude causes a decrease in the power available as the flight altitude is increased. The calculated variation of power with altitude for the water-cooled turbine-propeller power plant is shown in figure 7. It is seen that approximately 36 percent of the rated sea-level power is available at the design altitude.

<u>Over-all efficiency</u>. - The ideal efficiency of the compressor gas-turbine cycle is an exponential function of only the compressor pressure ratio (reference 1). The optimum-power pressure ratios associated with the turbine-inlet gas temperatures used in this analysis are in the range where the efficiency rises very slowly with increasing pressure ratio, and it is primarily for this reason that the efficiency curves have the sensibly constant character exhibited in figure 8. The curves show that with appropriate pressure ratios, turbine-inlet gas temperatures of 3000° to 3500° R will give over-all equivalent shaft efficiencies of 46 to 48 percent. The discontinuity in the curves at a temperature of 2500° is caused by the increase in the cooling-power loss, which is a result of the change from a two-stage to a three-stage turbine at that point.

The efficiency curves of figure 8 show that the low-temperature engines attain efficiencies that approach those obtained in the high-temperature engines. This fact is primarily due to the increase in the ratio of the cooling-power loss to the turbine power as the turbine-inlet temperature is increased, and the increase in heat loss caused by the change from a two-stage turbine in the low-temperature engines to a three-stage turbine in the high-temperature engines. A contributing cause lies in the fact that the use of constant polytropic exponents in the evaluation of the compressor and turbine work confers higher over-all component efficiencies on the low-pressure-ratio engines than those that result in the higher-pressure-ratio engines. The pressure ratio is increased with the turbine-inlet temperature; hence the engine efficiency increases as the pressure ratio is decreased with an effect on the over-all efficiency that is more favorable than might be expected.

In making the performance calculations, it has been assumed that in each engine there is no change in the turbine or compressor pressure ratio over the entire range of operating conditions for which calculations are made. At a given turbine-inlet temperature, the turbine power per unit of compressor air flow is therefore sensibly constant but the decrease in ambient temperature with altitude results in a decrease in compressor work and outlet air temperature and an increase in both the combustion temperature rise to the given turbine-inlet temperature and the fuel-air ratio necessary to attain it. In addition, at a given speed, the specific heat loss to the cooling fluid increases with altitude. These last effects balance the decrease in compressor work with altitude so that at a constant flight speed there is little gain in thermal efficiency with altitude in the water-cooled turbine-propeller engine.

At a given altitude, the ram temperature of the air entering the compressor increases with the flight speed; hence the compressor work and outlet air temperature increase with the flight speed but the combustion temperature rise and fuel-air ratio for a given turbine-inlet temperature decrease as does the specific heat loss to the turbine coolant. The last three effects outweigh the increase in compressor work and at a given altitude there is an increase in thermal efficiency with flight speed.

It should be realized, however, that the theoretical gains in efficiency accompaning an increase in pressure ratio cannot be realized without simultaneously increasing the turbine-inlet temperature. In general, the polytropic exponent for the compression process is greater than that for the expansion process; therefore at a constant turbine-inlet temperature, the compressor work will increase faster than the turbine work as the oompressor pressure ratio is increased. The net specific power output will be diminished and because there will be no appreciable change in the fuel horsepower the efficiency will be lowered. Conversely, an increase in the turbine-inlet temperature without a corresponding increase in the pressure ratio will yield an increase in the specific power output, but only at the cost of a lowered. efficiency because the exhaust gases must be rejected to the air stream at a higher temperature.

It should therefore be emphasized that the compressor pressure ratio must be increased with the turbine-inlet temperature to gain the maximum benefits made possible by liquid cooling of the turbine.

Brake specific fuel consumption. - The preceding discussion of efficiency applies directly to the brake specific fuel consumption, because one is but the reciprocal of the other multiplied by a constant conversion factor. This fact is illustrated. in the curves of brake specific fuel consumption in figure 9. At appropriate pressure ratios, the high-temperature operation made possible by liquid cooling will give brake specific fuel consumptions of 0.283 to 0.260 pound per brake horsepower-hour at the turbine-inlet temperatures capable of attainment. Again the discontinuity in the curves at a turbine-inlet temperature of 2500° R is causedbythe changefroma two-stage to a three-stage turbine at this turbine-inlet gas temperature.

Specific weight. - The specific-weight characteristics of the water-cooled turbine-propeller power plant are shown in figure 10. At a flight speed of 200 miles per hour at sea level, the calculations indicate an installed specific weight of 0.596 pound per equivalent shaft horse-power for a power plant designed to operate at a turbine-inlet temperature of 3500° R.

At a given turbine-inlet gas temperature and compressor pressure ratio, the Rower developed by the gas turbine is directly proportional to the amount of air handled in unit time, which is in turn a function of the flight speed and altitude. The over-all effect is a decrease in air weight flow with altitude, a consequent reduction in power, and an increase in specific weight. At the design speed of 600 miles per hour at an altitude of 35,000 feet, the sea-level specific-weight value of 0.596 pound per brake horsepower for the engine designed to operate at a turbine-inlet temperature of 3500° R has almost doubled its sea-level take-off value. At a given altitude, a marked decrease in specific weight attends an increase in airspeed because of the increase in air weight flow through the engine with flight speed.

Power Losses

Specific heat loss to turbine coolant. - The largest single power loss introduced by the cooling process in the liquid-cooled turbine is the direct heat loss to the turbine coolant. The character of this loss is shown in figure 11. The calculations show that the direct heat loss to the cooling fluid of the engine with a turbine-inlet temperature of 3500° R amounts to 8.6 percent of the fuel horsepower when operated at a flight speed of 600 miles per hour at an altitude of 35,000 feet.

Although all the heat lost to the cooling fluid is considered unusable because no effort is made to recover it, actually most of this heat would be lost if no cooling were employed because the gases leaving the tail pipe would be at a higher temperature and their thermal potential would be beyond recovery. When considered in this light, the actual power lost to the cooling fluid is $\left(\eta_{S,u} - \eta_{S,L}\right)hp_l$. For the particular case under consideration, the amount that can actually be said to be lost as a result of the cooling process is 28.5 percent of the calculated loss or 2.5 percent of the fuel horsepower. All the power losses are expressed in terms of the fuel horsepower, but in terms of the brake horsepower the values are approximately double those shown on the power-loss curves.

At a given turbine-inlet gas temperature, the heat loss to the coolant varies with the heat-transfer coefficient, which in turn, varies with the 0.8 power of the engine air flow. Under the same conditions, the engine power and the fuel-consumption rate are in direct proportion to the engine air flow; hence, the specific-heat loss decreases slightly as the engine air flow and power are increased. As previously explained, the discontinuity in the curves at a turbine-inlet temperature of 2500° R is caused by the change from a two-stage to a three-stage turbine at this turbine-inlet temperature.

The coolant circulation rate (6.42 lb/sec) used in calculating the heat loss gives a maximum average blade-metal temperature of approximately 860° F at a turbine-inlet gas temperature of 2000° R (reference 4). This blade temperature increases to 1085° R as the turbine-inlet temperature is increased to 3500° R. Unfortunately, the characteristics of the available liquid coolants make it difficult to decrease the heat loss by operating the turbine blades at the highest metal temperatures permissible, that is, in the neighborhood of 2000° R. At the elevated coolant temperatures required by high average blade temperatures, organic cooling agents such as ethylene glycol decompose and yield solid products that would plug the cooling passages and block cooling to the blades. With water as the coolant, the plumbing and heat-exchanging equipment necessary for high-pressure application would result in a large and bulky arrangement of considerable weight and doubtful efficiency. Inasmuch as operating the blades at the maximum allowable metal temperature can increase the net output by only 4 percent, the safety margin afforded by the lower blade-metal temperature outweighs the slight loss in available power.

Several practical advantages are to be gained by operating at the low average blade temperatures possible with a coolant flow in excess of that required for the maximum average blade temperature permissible. The requirements for good aerodynamic efficiency of the turbine blades dictates small-diameter cooling passages within them; hence, in practice, correct distribution of coolant flow might prove difficult to maintain in a finely designed system. A coolant flow in excess of the minimum requirement insures that each of the blades on the turbine will always receive at least the minimum requirement for adequate cooling. The low average blade temperature obtained insures that the extremities of the turbine blade will not reach temperatures high enough to cause damage. All the critical components of high-temperature gas-turbine power plants may be constructed from the cheap and more common alloy steels that are easily forged and machined. The amount of cooling used in this analysis insures an average blade temperature in the range where blade strength is unaffected by the temperature of the turbine-blade metal.

The direct heat loss to the turbine coolant, though of considerable absolute magnitude, is a small percentage of the fuel horsepower and net horsepower of the power plant. There is, however, one possibility of reducing this loss. The stator blades in turbines are not subjected to centrifugal stresses and could be made of ceramic materiels, which would require no cooling. Because cooling of the stator blades has been taken into account instead of assuming the use of ceramic stators, the losses presented are higher than those that might be realized in practice.

Coolant pumping power. - At a given coolant flow through each turbine stage, the coolant pumping power is almost proportional to the square of the rotative speed of the turbine and is directly proportional to the number of stages in the turbine. The pumping power, for a given coolant flow is therefore independent of the power output; hence, the percentage of the fuel horsepower expended in the coolant pumping process decreases as the power output increases. This trend is shown in the increase of the specific pumping power with altitude in figure 12. Over the entire range of flight conditions for all the engines, the pumping power varies from a minimum of 0.26 to 2.25 percent of the fuel horsepower. Again the discontinuity in all the curves at a temperature of 2500° R is caused by the change from a two-stage to a three-stage turbine at that point.

Turbine-coolant heat exchanger. - The specific turbine-coolant heat-exchanger requirement is shown in figure 13. Because the ratio of the power output to the cross-sectional air-flow area is very high for the liquid-cooled, high-temperature, gas-turbine engine, for a given engine the frontal area of the turbine-coolant heat-exchanger core will be larger than the frontal area of the engine proper. When compared on the basis of square feet of frontal area per net horsepower, or pounds of heat-exchanger weight per net horsepower, however, the turbine-coolant heat-exchanger requirement of the liquid-cooled turbine is approximately 20 percent of that of contemporary liquid-cooled reciprocating engines of the same power output. Calculations consistently show that the heat-exchanger core gives positive thrust rather than drag when operated at or near design flight speed, and with careful ducting, the heat exchanger will recover a small part of the power lost to the cooling fluid.

GENERAL DISCUSSION

The results of the calculations indicate that with all the losses incurred by liquid cooling deducted, a water-cooled turbine turbine-propeller power plant designed for operation at a turbine-inlet temperature of 3500° R, with a compressor pressure ratio of 27.6, and a sea-level static air flow of 55 pounds per second, would give a specific power output of 506.0 horsepower per pound of compressor air flow per second when operated at a flight speed of 600 miles per hour at an altitude of 35,000 feet. The equivalent shaft thermal efficiency and specific fuel consumption would be 48.2 percent and 0.283 pound per equivalent shaft horsepmr-hour, respectively. The specific heat loss to the cooling fluid would be 0.0855 horsepower per fuel horsepower, the specific coolant pumping power would be 0.00989 horsepower per fuel horsepower, and the installed specific weight, based on the take-off horsepower (flight speed, 200 mph) at sea level, would be 0.5964 pound per horsepower. The weight of heat exchanger, plumbing, attendant coolant, propellers, and their associated reduction gearing is included in the specific-weight estimate.

The improvement in performance indicated by these calculations is the result only of the use of high turbine-inlet temperature operation made possible through liquid cooling, in conjunction with appropriate pressure ratios. Its realization is dependent only upon successful solution of the problems incident to circulating the coolant through the turbine blading and plumbing system, and the development of compressors combining high pressure ratios with good efficiencies. The performance ie calculated for current turbine materials and is capable of attainment with most of the cheaper and more common alloy steels that are easily forged and machined by common shop methods and production techniques.

These predicted values of the performance of liquid-cooled turbine-propeller power plants are conservative. 5 estimate of the heat loss to the cooling fluid is particularly conservative because to facilitate its calculation the total temperature of the gas stream rather than the effective gas temperature is used in the calculation of the heat-transfer temperature difference thereby overestimating the heat loss. The insulating effect of boundary layer on the turbine blades is also neglected in the heat-loss calculation. The boundary-layer phenomenon is of considerable importance in reducing both the heat loss and the average blade temperature by shielding the turbine blades from direct contact with the hot gas of the free stream. 5 weight estimate is also believed to be conservative because in selecting the principal power-plant auxiliaries, those whose weights fell in the heavier range of their respective classes were generally taken. In all cases the performance and physical characteristics of the paver-plant auxiliaries are of components already developed end currently in production.

SUMMARY OF RESULTS

From an analytical investigation of the performance of six turbine-propeller power plants having turbines equipped with water-cooled blades and designed to operate at turbine-inlet temperatures of 2000°, 2500°, 3000°, 3500°, 4000°, and 4500° R, respectively, the following results were obtained :

1. At the design speed and altitude of 600 miles per hour and 35,000 feet, respectively, the net specific equivalent shaft horsepower increased almost linearly with the turbine-inlet temperature from 184.6 to 775 horsepower per pound of compressor air flow per second as the turbine-inlet temperature was increased from 2000° to 4500° R. 971

2. At the same conditions and over the same turbine-inlet-temperature range, the over-all efficiency, baaed on the net equivalent shaft horsepower, increased from 43 to 52 percent as the specific fuel consumption decreased from 0.314 to 0.264 pound per equivalent shaft horsepower-hour. Although the net specific equivalent shaft horsepower increased almost linearly with the turbine-inlet temperature, there was little gain in efficiency or decrease in specific fuel consumption at any operating condition above a turbine-inlet temperature of 3500° R.

3. The installed specific weight, based on the sea-level take-off horsepower decreased from 0.8084 pound per equivalent shaft horsepower for the engine designed to operate at a turbine-inlet temperature of 2000° R to 0.5386 pound per equivalent shaft horsepower for the engine designed to operate at 4500° R. For each engine, the specific weight increased steadily with altitude and almost doubled its sea-level value at 35,000 feet.

4. The heat loss to the cooling fluid was the largest single loss incurred in the cooling process. In a given engine this loss varied slightly with operating conditions. Over the entire range of operating conditions for which performance calculations were made, the power lost to the cooling fluid in cooling the turbine blades varied from a minimum of 5.8 to a maximum of 6.5 percent of the fuel horsepower in the engine designed for a turbine-inlet temperature of 2000° R. In the engine designed for a turbine-inlet temperature of 4500° R, this variation was from a minimum of 6.8 to a maximum of 8.8 percent of the fuel horsepower. The increase in the specific heat lose in these two cases was due entirely to the increase in heat-transfer area resulting from the difference in the number of turbine stages (two in the engine designed for a turbine-inlet temperature of 2000° R and three in the engine designed to operate at a turbine-inlet temperature of 4500° R). With a fixed number of turbine stages, the specific heat loss decreased with the turbine-inlet temperature.

5. The turbine-coolant heat-exchanger-core frontal-area requirement of the water-cooled turbine-propeller engine varied from 0.440 square foot per 1000 take-off equivalent shaft horsepower for the 2000° R engine to 0.3880 square foot per 1000 take-off equivalent shaft horsepower for the 4500° R engine.

6. The pumping power required to circulate the coolant was negligible and it never exceeded 2.25 percent of the fuel horsepower in any of the engines.

7. For the particular case of the engine designed with a compressor pressure ratio of 27.6 to operate at a turbine-inlet temperature of 3500° R, the following performance, which takes full consideration of all the losses incurred by the cooling process, was obtained at a flight speed of 600 miles per hour at an altitude of 35,000 feet.

(a) Net specific output, 506.0 equivalent shaft horsepower per pound of compressor air flow per second

(b) Brake thermal efficiency, 48.2 percent

(c) Brake specific fuel consumption, 0.283 pound per equivalent shaft horsepower-hour

(d) Installed specific weight at sea level including heat exchangers and propeller, 0.5964 pound per horsepower

(e) Specific heat loss to turbine coolant in cooling turbine blades, 0.0855 horsepower per fuel horsepower

(f) Specific coolant pumping power, 0.00989 horsepower per fuel horsepower

CONCLUSIONS

The following general conclusions may be drawn from the study of the effect of water-cooled turbine blades on the performance of turbine-propeller power plants;

1. The improvement in gas-turbine performance obtained by operating at the high turbine-met temperatures possible through liquid cooling is not vitiated by the losses incurred in the cooling process if the proper pressure ratios associated with those high turbine-inlet temperatures are used.

2. Full utilization of the high operational turbine-inlet temperatures made possible by liquid cooling demands an increase in compressor and turbine pressure ratios from the present values of 4 to 8 to a range from 30 to 40.

3. The required turbine-coolant heat-exchanger capacity and size for the liquid-cooled turbine engine is approximately 20 percent of that of a liquid-cooled reciprocating engine of equal rated power.

4. Successful application of liquid cooling to the blading of gas turbines will permit high-temperature turbine blades, wheels, and other parts exposed to the combustion process and products to be constructed entirely from common alloy steels such as SAE nickel, chromium, molybdenum, chrome-nickel, and chrome-molybdenum series.

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APPENDIX A

COMPRESSOR PRESSURE RATIO FOR OPTIMUM POWER

If it is assumed that there is no pressure loss between the compressor outlet and the turbine inlet and that the adiabatic exponents for the compression and expansion processes are the same, it is possible to derive a simple relation between the compressor pressure ratio for best power and the ratio of maximum cycle temperature to compressor-inlet temperature.

The net power of the compressor and turbine combination per pound of air is

$$J_t - J_c = \frac{\eta_{m,t}\gamma R_t T_3 M_a}{(\gamma-1)550g}\left[1-\left(\frac{P_4}{P_3}\right)^{\frac{\gamma-1}{\gamma}}\right] - \frac{\gamma R_c T_1 M_a}{\eta_{m,c}(\gamma-1)550g}\left[\left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} - 1\right]$$

If it is assumed that

$$P_2 = P_3$$

and

$$R_t = R_c = R$$

it follows that

$$J_t - J_c = \frac{\gamma R M_a}{(\gamma-1)550g}\left\{\eta_{m,t}T_3\left[1-\left(\frac{P_4}{P_2}\right)^{\frac{\gamma-1}{\gamma}}\right] - \frac{T_1}{\eta_{m,c}}\left[\left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} - 1\right]\right\}$$

$$= \frac{\gamma R T_1 M_a}{(\gamma-1)\eta_{m,c}550g}\left\{\eta_{m,t}\eta_{m,c}\frac{T_3}{T_1}\left[1-\left(\frac{P_4}{P_2}\right)^{\frac{\gamma-1}{\gamma}}\right] - \left[\left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} - 1\right]\right\}$$

$$= \frac{\gamma R T_1 M_a}{(\gamma-1)\eta_{m,c}550g}\left\{\eta_{m,t}\eta_{m,c}\,\frac{T_3}{T_1}\left[1-\left(\frac{P_4}{P_1}\right)^{\frac{\gamma-1}{\gamma}}\left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}}\right]-\left[\left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}}-1\right]\right\}$$

When the right-hand member is differentiated with respect to P_2/P_1 and the derivative is set equal to zero, there is obtained

$$\frac{\partial(J_t-J_c)}{\partial(P_2/P_1)}=\frac{\gamma R T_1 M_a}{\eta_{m,c}(\gamma-1)550g}\left\{\eta_{m,t}\eta_{m,c}\,\frac{T_3}{T_1}\left[0-\frac{1-\gamma}{\gamma}\left(\frac{P_2}{P_1}\right)^{\frac{1-2\gamma}{\gamma}}\left(\frac{P_4}{P_1}\right)^{\frac{\gamma-1}{\gamma}}\right]-\left[\frac{\gamma-1}{\gamma}\left(\frac{P_2}{P_1}\right)^{-\frac{1}{\gamma}}\right]\right\}=0$$

Solving this equation for P_2/P_1 yields

$$\frac{P_2}{P_1}=\sqrt{\frac{P_4}{P_1}}\left(\eta_{m,t}\eta_{m,c}\,\frac{T_3}{T_1}\right)^{\frac{\gamma}{2(\gamma-1)}}$$

For the turbine-propeller engine, P_4/P_1 is usually slightly less than unity and for all computational purposes in this report it is arbitrarily kept equal to 0.9. In addition, the polytropic exponent n_c will be substituted for γ. With these values, the compressor pressure ratio for best power becomes

$$\frac{P_2}{P_1}=0.949\left(\eta_{m,t}\eta_{m,c}\,\frac{T_3}{T_1}\right)^{\frac{n_c}{2(n_c-1)}}$$

APPENDIX B

SAMPLE COMPUTATION

The use of the methods developed in the analysis section is illustrated in a sample computation for the 3500° R engine operating at a flight speed of 600 miles per hour at an altitude of 35,000 feet.

Design Operating Conditions and Data

In order to insure a common basis for comparing the results, the following quantities are kept constant over the entire scope of the analysis :

Ratio of specific heats for compression, γ_c	1.376
Ratio of specific heats for expansion, γ_t	1.220
Polytropic exponent of compression, n_c	1.430
Polytropic exponent of expansion, n_t	1.200
Mechanical efficiency of compressor, $\eta_{m,c}$	0.985
Mechanical efficiency of turbine, $\eta_{m,t}$	0.965
Exhaust-nozzle coefficient, K	0.970
Air weight flow at sea-level static pressure M_a, lb/sec	55
Turbine coolant pumping efficiency, η_p	0.900
Cooling water flow per stage, M_l, lb/sec	12.640
Turbine-coolant, heat-exchanger and compressor-inlet pressure-recovery efficiency	Perfect
Heating value of fuel, Btu/lb	18,700
Right-section perimeter of turbine blade, ψ_b, ft	0.2514
Propeller efficiency used to calculate equivalent shaft horsepower of turbine-coolant heat-exchanger and jet thrust, η_p	0.85

The following variables are dependent on the flight conditions and therefore apply only to the example given:

Altitude, ft	35,000
Flight speed V_0, mph	600
Ambient temperature, T_0, °R	394.35
Ambient pressure, p_0, lb/sq ft	498

Work of Compressor and Turbine Combination

At station 1 (fig.2) in the diffuser, the ram pressure is given by equation (21) and with the value of γ of 1.401 taken from reference 5

$$P_1 = p_0 \left(1 + \frac{\gamma_0 - 1}{2} \frac{V_0^2}{\gamma_0 g R_0 T_0}\right)^{\frac{\gamma_0}{\gamma_0 - 1}}$$

$$= 498 \left[1 + \frac{0.401 \times (880)^2}{2 \times 32.2 \times 1.401 \times 53.33 \times 394.36}\right]^{\frac{1.401}{0.401}}$$

= 850 pounds per square foot

The temperature of the diffused air (equation (22)) is

$$T_1 = t_0 + \frac{(\gamma_0 - 1)\, V_0^2}{2 g \gamma_0 R_0}$$

$$= 394.35 + \frac{(0.401)(880)^2}{2 \times 32.2 \times 53.33 \times 1.401} = 458.6^{\circ}\ R$$

The compressor pressure ratio is computed from the relation between compressor pressure ratio and engine temperature ratio derived in appendix A and is

$$\frac{P_2}{P_1} = 0.949 \left(\eta_{m,c}\, \eta_{m,t} \frac{T_3}{T_1}\right)^{\frac{n_c}{2(n_c - 1)}}$$

$$= 0.949 \left[0.985 \times 0.985 \times \frac{3500}{458.6}\right]^{\frac{1.430}{2(0.430)}}$$

$$= 27.64$$

The engine from which the basic data for the report are taken has a sea-level static air flow of 55 pounds per second. With a given engine, there is no change in its characteristic physical dimensions: The gas constant R_c is a constant al80 and the non-dimensional weight-flow parameter $M_a/gD^2\sqrt{P_1\rho_{s,1}}$ reduces to $M_a\sqrt{T_1}/P_1$. The corrected air Plow is obtained by equating the non-dimensional weight-flow parameter at sea-level static performance conditions to its value at the inlet conditions at altitude. Therefore

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$$\left(\frac{M_a\sqrt{T_1}}{P_1}\right)_A = \left(\frac{M_a\sqrt{T_1}}{P_1}\right)_{SL}$$

$$M_{a,A} = \left(\frac{M_a\sqrt{T_1}}{P_1}\right)_{SL}\left(\frac{P_1}{\sqrt{T_1}}\right)_A$$

$$= 0.5915\frac{850}{\sqrt{458.6}}$$

$$= 23.48 \text{ pounds per second}$$

By equation (3) the power absorbed in the compressor is

$$hp_c = \frac{\gamma_c R_c T_1}{\eta_{m,c}(\gamma_c - 1)}\left[\left(\frac{P_2}{P_1}\right)^{\frac{n_c-1}{n_c}} - 1\right]\frac{M_a}{550}$$

$$= \frac{1.376 \times 52.3594 \times 458.6}{0.985 \times 0.376}\left(27.64^{\frac{0.43}{1.43}} - 1\right)\frac{23.48}{550}$$

$$= 6525 \text{ horsepower}$$

Equation (24) gives for the compressor-outlet temperature

$$T_2 = T_1 \left(\frac{P_2}{P_1}\right)^{\frac{n_c-1}{n_c}} = 458.6 \times 27.64^{\frac{0.43}{1.43}}$$

$$= 1245^{\circ}\ R$$

The temperature rise from compressor outlet to turbine inlet is 2255° F and the fuel-air ratio required to attain this temperature *is* 0.03945 (reference 13).

From equation (25) the turbine-inlet pressure is

$$P_3 = 0.98\ P_2 = 0.98 \times 27.64 \times P_1$$

$$= 27.09\ P_1$$

and

$$P_4 = 0.9\ P_1$$

By use of equation (2) the power developed in the turbine is found to be

$$hp_t = \frac{\eta_{m,t}\gamma_t R_t T_3}{(\gamma_t - 1)} \left[1 - \left(\frac{P_4}{P_3}\right)^{\frac{n_t-1}{n_t}}\right] \frac{M_a}{550} \left(1 + \frac{f}{a}\right)$$

$$= \frac{0.985 \times 1.22 \times 53.56 \times 3500}{0.22} \left[1 - \left(\frac{0.9\ P_1}{27.09 P_1}\right)^{\frac{0.2}{1.2}}\right] \frac{23.48}{550} \times 1.03945$$

$$= 19,650 \text{ horsepower.}$$

Power Losses

Heat loss to cooling fluid. - From reference 4 the combustion-gas-to-blade heat-transfer coefficient h_o is 217 Btu per hour per square foot per °F. This *value* is for the sea-level static air weight flow of 55 pounds per second. The corrected heat-transfer coefficient (equation (13)) is

$$h_{o,z} = \frac{\left(\frac{M}{\mu}\right)_z^{0.8} k_z}{\left(\frac{W}{\mu}\right)_y^{0.8} k_y} h_o y$$

$$= \frac{\left(\frac{23.48}{0.1002}\right)^{0.8} \times 0.0433}{\left(\frac{55}{0.1401}\right)^{0.8} \times 0.0578} 217$$

$= 104$ Btu per hour per square foot per $^{\circ}F$

The nondimensional cooling coefficient, given by equation (12) is

$$C = \frac{1}{1 + \frac{h_o \psi_b}{h_i \psi_c}}$$

$$= \frac{1}{1 + \frac{104 \times 0.2514}{0.1310 \times 2370}} = 0.9215$$

The heat-transfer area per unit length of the turbine expansion path S_o/ψ_b is 37.25 square feet per foot and the average temperature of the coolant liquid flowing in the blade cooling passage T_l is 200° F. There are three stages in the turbine and the right-section perimeter of a turbine blade is 0.2514 foot. Application of equations (8) and (9) at the turbine inlet and the junction of the first and second stages yields the following equation and value of a:

$$\alpha\psi_b \log e = \frac{n_t-1}{n_t}\frac{1}{Z}\log 10 \frac{0.98\ P_2}{0.9\ P_1}$$

$$\alpha = -\frac{n_t-1}{n_t Z \psi_b \log e}\log\frac{0.98\ P_2}{0.9\ P_1}$$

$$= -\frac{0.2}{1.2 \times 3 \times 0.2514 \times 0.434294}\log\frac{0.98 \times 27.64}{0.90}$$

$$= -0.75169$$

When the preceding values are substituted. into equation (14), the power loss to the cooling fluid is

$$hp_l = \frac{778}{1.98 \times 10^6} Ch_o \frac{S_o}{\psi_b}\left[\frac{T_3}{\alpha}\left(e^{\alpha Z\psi_b} - 1\right) - T_l Z\psi_b\right]$$

$$= \frac{778 \times 0.9215 \times 104 \times 37.25}{1.98 \times 10^6}\left[\frac{3500}{0.75169} - \left(e^{-0.75169 \times 3 \times 0.2514} - 1\right)\right.$$

$$\left. -(659.6 \times 3 \times 0.2514)\right]$$

$$= 2126.61 \text{ horsepower}$$

The temperature drop due to the direct heat loss is

$$\Delta T_l = \frac{550}{778}\frac{hp_l}{c_{p,3}M_a(1 + f/a)}$$

$$= \frac{550 \times 2126.61}{778 \times 0.28715 \times 23.48 \times 1.03945} = 211.80\ ^{\circ}F$$

From equation (26) and reference 6, the uncorrected enthalpy is

$$H_4 = H_1 + \Delta H_o + \Delta H_f - \Delta H_t = 14.044 + 207.62 + 570 - 592$$

$$= 199.664 \text{ Btu per pound}$$

When the uncorrected turbine-outlet temperature T_4 of 1991.80° R (*reference* 14) is decreased by the temperature drop caused by cooling, the actual turbine-outlet temperature is found to be 1780° R.

From equation (4), the exit velocity of the jet from the exhaust nozzle is

$$V_j = K\sqrt{2g\,\frac{\gamma_t R_t T_{4,corr}}{(\gamma_t-1)}\left[1-\left(\frac{p_0}{P_4}\right)^{\frac{\gamma_t-1}{\gamma_t}}\right]}$$

$$= 0.97\sqrt{2 \times 32.2\,\frac{1.2 \times 53.56 \times 1780}{0.2}\left[1-\left(\frac{498}{765}\right)^{\frac{0.2}{1.2}}\right]}$$

$$= 1545 \text{ feet per second}$$

And by equation (5) the thrust horsepower of the exhaust jet is

$$hp_j = \frac{V_0 M_a}{17{,}700\eta_p}\left[\left(1+\frac{f}{a}\right)V_j - V_0\right]$$

$$= \frac{880 \times 23.48}{(17{,}700)(0.85)}\left(1.03495 \times 1545 - 880\right) = 996 \text{ horsepower}$$

<u>Coolant pumping power.</u> - The turbine in this instance has three stages; thus with the aid of equations (18) and (19) and reference 10 the coolant pumping power is

$$hpp = 1.012\left[\frac{M_l}{550g}\left(\Omega r_7 V_7 - \Omega r_6 V_6\right)\left(1 - \eta_p^{2}\right) + hp_F\right]$$

$$= 1.01 \times 3\left\{\frac{6.42}{550 \times 32.2}\left[\left(\frac{200\pi \times 21}{12}\right)^2 - 0\right]\left[1 - (0.90)^2\right] + 0.43651\right.$$

$$= 242.29 \text{ horsepower}$$

Aerodynamic drag of turbine-coolant heat-exchanger core. - From information supplied by the Harrison Radiator Corporation, a heat exchanger of sufficient size to provide adequate cooling at 200 miles per hour at sea level wae selected. For the case under consideration, the following values occur:

Heat-dissipation rate, Btu/min	170,413.61
Temperature of coolant entering heat exchanger $^{\circ}F$	250
Initial turbine-coolant heat-exchanger temperature difference, $^{\circ}F$	183.43

When an air flow of 500 pounds per minute and a coolant flow of 130 gallons per minute are used, the heat-dissipation rate is 6600 Btu per 100° F initial temperature difference per square foot of core frontal area, and the required turbine-coolant heat-exchanger-core frontal area is 13.59 square feet.

At a flight speed of 600 miles per hour at an altitude of 35,000 feet, the heat-dissipation rate of the engine is 90,203.25 Btu per minute, and the initial turbine-coolant heat-exchanger temperature difference is 208.6 °F. Dividing the engine heat-dissipation rate by the product of the core frontal area and 1/100 of the initial turbine-coolant heat-exchanger temperature difference gives a chart value heat-transfer rate of 3182.31 Btu per minute per square foot (test section) of core frontal area per 100° initial temperature difference. From the information obtained from the Harrison Radiator Corporation, the required air flow is 187.5 pounds per minute. The chart value of the required cooling-air pressure drop is 2.58 inches of water, which must be divided by the altitude density ratio σ to get a corrected value of

$$\Delta P_h = \frac{2.85}{\sigma} = \frac{2.85}{0.3098}$$

$$= 9.2 \text{ inches of water}$$

which corresponds to 47.83 pounds per square foot.

From equation (21), the pressure at the front face of the turbine-coolant heat-exchanger core is found to be 850 pounds per square foot. At the outlet face of the core, the pressure is

$$P_{2,h} = P_1 - \Delta p_h = 850 - 47.83$$

$$= 802.17 \text{ pounds per square foot}$$

The exit temperature of the cooling air from the turbine-coolant heat-exchanger core is

$$T_{2,h} = T_1 + \frac{Q}{3600\ c_{p,2,h} M_{a,h}} = 458.6 + \frac{90203.25}{13.588 \times 0.2401 \times 187.5}$$

$$= 605.25\ ^{\circ}R$$

From equation (4), with appropriate value6 of n_0 and R_0 from reference 6, the exit velocity of the cooling air from the turbine-coolant heat-exchanger-duct nozzle is

$$V_{3,h} = K \sqrt{2g \frac{\gamma_0 R_0 T_{2,h}}{\gamma_0 - 1} \left(1 - \frac{p_0}{P_{2,h}}\right)^{\frac{\gamma_0 - 1}{\gamma_0}}}$$

$$= 0.97 \sqrt{2 \times 32.2\ \frac{1.401 \times 53.33 \times 605.25}{0.401} \left[1 - \left(\frac{498}{802.17}\right)^{\frac{0.401}{1.401}}\right]}$$

$$= 933.5 \text{ feet per second}$$

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Equation (5) with f/a set equal to 0 and $M_{a,h}$ substituted for M_a permits the calculation of the thrust horsepower of the turbine-coolant heat-exchanger core. With appropriate substitutions for the case under consideration, the duct thrust horsepower is

$$hp_h = \frac{V_0\left(V_{3,h} - V_0\right)}{17,700\ \eta_p} M_{a,h} = \frac{880(933.5 - 880)\ 187.5 \times 13.588}{17,700 \times 0.85 \times 60}$$

$$= 128.8 \text{ horsepower}$$

Engine-Weight Analysis

Engine shell-casting weight. - The thickness of the equivalent steel shell is

$$\delta = \frac{\left(P_{2,L} - P_{2,u}\right) d}{144 \times 80,000} = \frac{(27.64 - 4.5)\ 3170 \times 21}{144 \times 80,000}$$

$$= 0.1337 \text{ inch}$$

and by use of equation (20), the weight of the equivalent steel shell is found to be

$$w_e = \frac{\pi\rho_s L}{4 \times 1728}\ 28\ (2d + 2\delta) = \frac{\pi \times 498 \times 80}{4 \times 1728}\ 2 \times 0.1337 \times 42.267$$

$$= 204.64 \text{ pounds}$$

Weight of turbine. - The weight of a complete turbine stage Including rotor dish, blades, and nozzle diaphragm is assessed at 204 pounds. There are three such stages and the weight of the turbine is 612 pounds.

Weight of compressor. - From the section of the report entitled "ANALYSIS", the weight of the compressor is

$$w_c = 2.8 \times 612 = 1713.60 \text{ pounds}$$

Coolant pump weight. - This weight is assessed at 30 pounds.

Engine-coolant heat-exchanger weight. - The weight of a suitable tube and fin aluminum radiator and its attendant coolant is 20.9 pounds per cubic foot. An allowance of 25 percent of the heat-exchanger and its coolant weight is made for the plumbing, and the weight of the turbine-coolant heat-exchanger installation is

$$w_h = 1.25 \text{ (turbine-coolant heat-exchanger and coolant weight)}$$

$$= 1.25 \times 20.9 \times 13.568 = 354.99 \text{ pounds}$$

Weight of propeller-reduction gears. - By the derivation given in "ANALYSIS" this weight is

$$w_r = 0.05 \times 29799.4 = 1489.97 \text{ pounds}$$

Weight of propellers. - By the relations developed in "ANALYSIS", the weight of the propellers is estimated at

$$w_p = 0.0302 \times 29{,}799.4 = 8999.42$$

The weights of the principal engine components are tabulated as follows:

Component	Weight (lb)
Engine shell casting	204.64
Turbine-coolant heat exchanger, plumbing and coolant	354.99
Turbine (three stages)	612.00
Compressor	1713.60
coolant pump	30.00
Propeller reduction gear unit	1489.97
Propellers	8999.42
Basic engine weight	1174.00
Cross weight	14,578.62

Specific Performance Values

Specific power output. - From equation (1) the net propulsive horsepower of the power plant is

$$hp_{net} = hp_t - hp_c - hp_l - hp_p + hp_j - hp_R$$

$$= 19,650 - 6525 - 2126.61 - 242.29 + 996.0 + 128.8$$

$$= 11,880.9 \text{ horsepower}$$

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The specific power output is

$$\frac{hp}{M_a} = \frac{11,880.9}{23.48}$$

= 506.0 horsepower per pound of compressor air flow per second

Specific weight. - The specific weight is

$$\frac{W}{hp_{net}} = \frac{14578.6}{11,880.9}$$

= 1.227 pounds per net equivalent shaft horsepower

Specific fuel consumption. - The specific fuel consumption is

$$f = \frac{f/a \; M_a \; 3600}{hp_{net}} = \frac{0.03945 \times 23.48 \times 3600}{11,880.9}$$

= 0.2807 pound per brake horsepower-hour

Power-plant brake thermal efficiency. - The brake thermal efficiency of the power plant is

$$\eta_S = \frac{hp_{net}}{hp_f} = \frac{hp_{net}}{f/a \; (M_a) \frac{(60 \times 14700 \times 778)}{33000}}$$

$$= \frac{33,000 \times 11,880.9}{0.03945 \times 23.48 \times 60 \times 18,700 \times 778}$$

$$= 0.485 = 48.5 \text{ percent}$$

Specific Power Losses

Specific heat loss. - The specific heat loss is

$$\frac{hp_l}{hp_f} = \frac{2126.61}{\frac{0.03945 \times 23.48 \times 60 \times 18{,}700 \times 9778}{33{,}000}}$$

$$= 0.08679$$

= 8.679 percent of the fuel horsepower

Specific pumping power. - The specific pumping power is

$$\frac{hp_p}{hp_f} = \frac{242.29}{\frac{0.03945 \times 23.48 \times 60 \times 18{,}700 \times 778}{33{,}000}}$$

= 0.009889 = 0.9889 percent of the fuel horsepower

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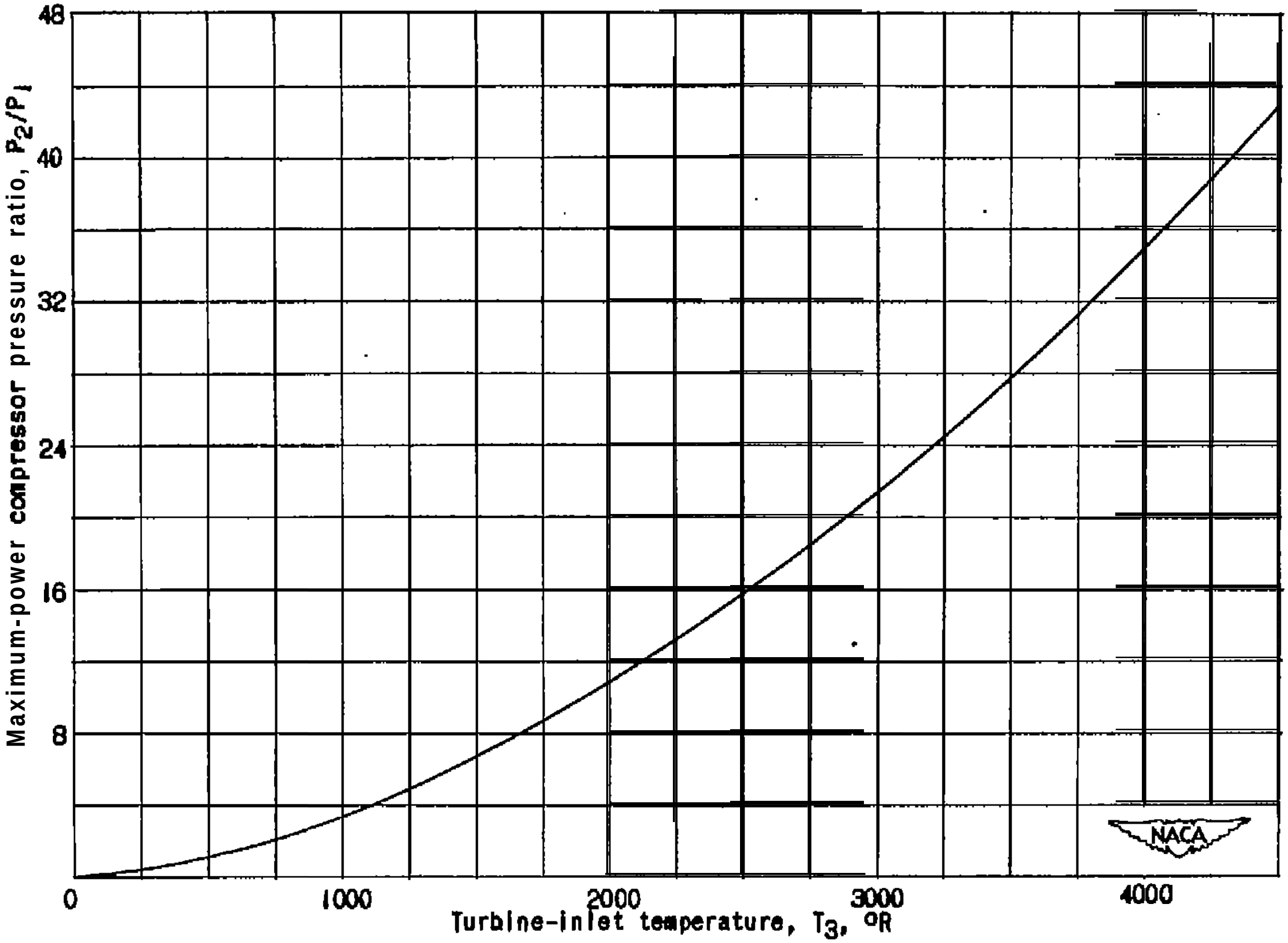

Figure 1. - Variation of maximum-power compressor pressure ratio with turbine-inlet temperature for multistage-turbine turbine-propeller engine with water-cooled turbine blades. Flight speed, 600 miles per hour; altitude, 35,000 feet.

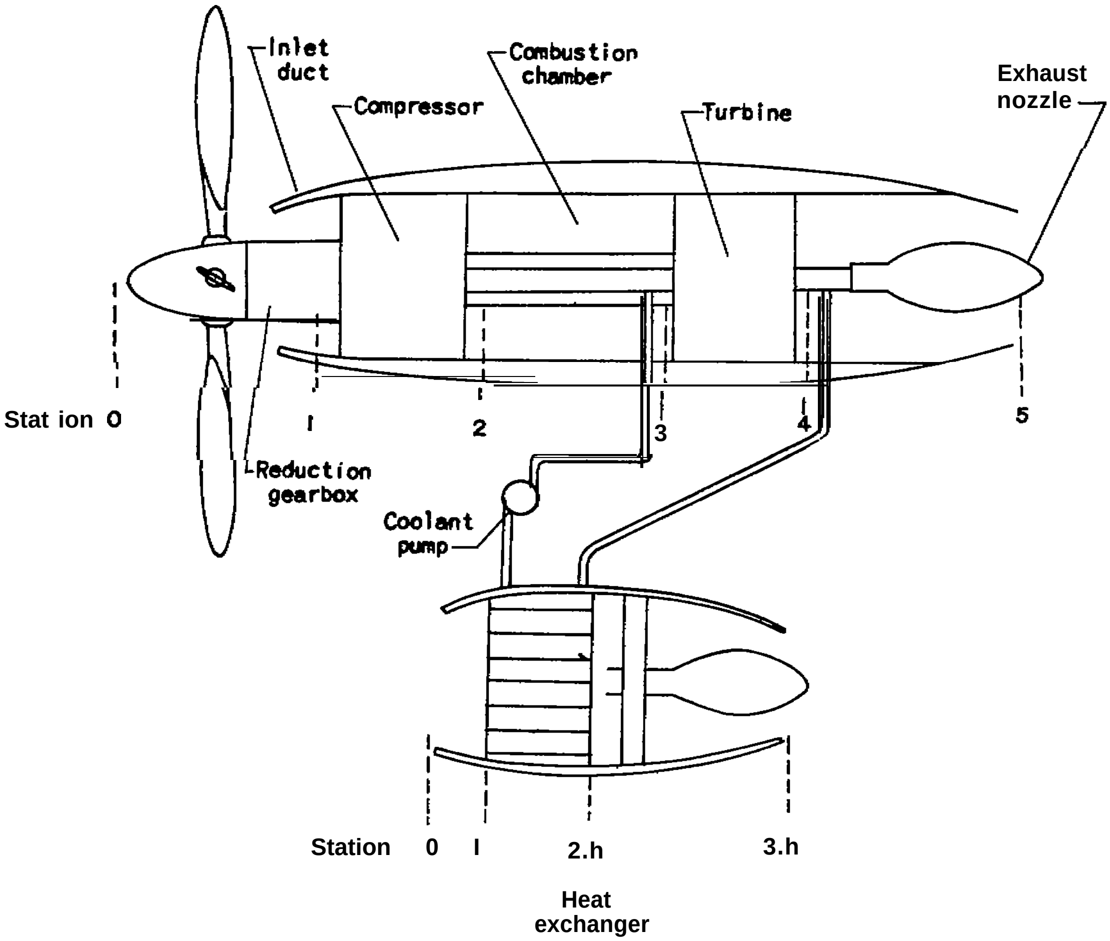

Figure 2. - Schematic diagram of engine and heat exchanger for multistage-turbine turbine-propeller engine with water-cooled turbine blades.

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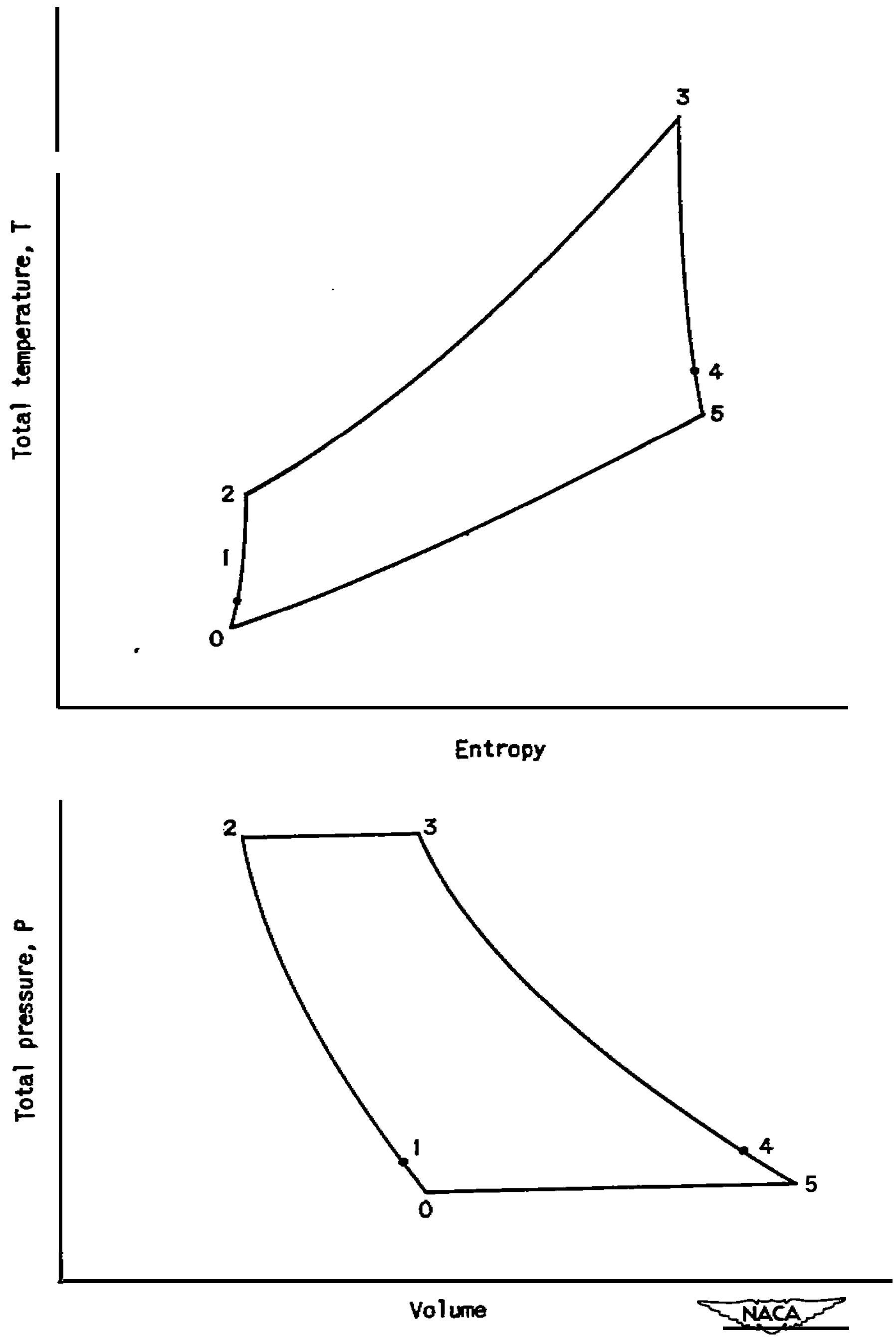

Figure 3. - Chamcterist ic temperature-entropy and pressure-volume diagrams for thermodynamic cycle of multistage-turbine turbine-propeller engine with water-cooled turbine blades. (Numbers refer to stations.)

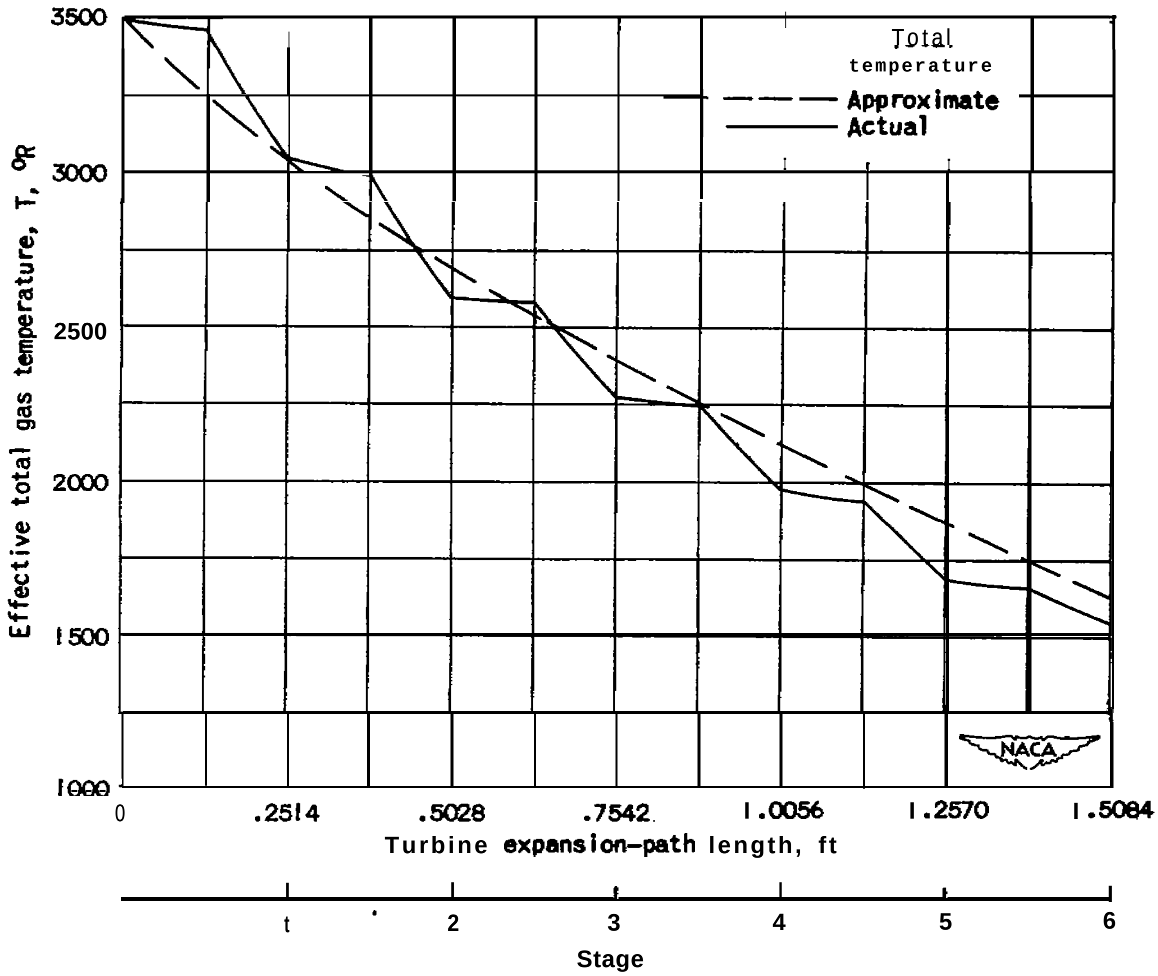

Figure 4. - Comparison of experimental and ideal relations between calculated total gas temperature and turbine expansion-path length for multistage-turbine turbine-propeller engine with water-cooled turbine blades.

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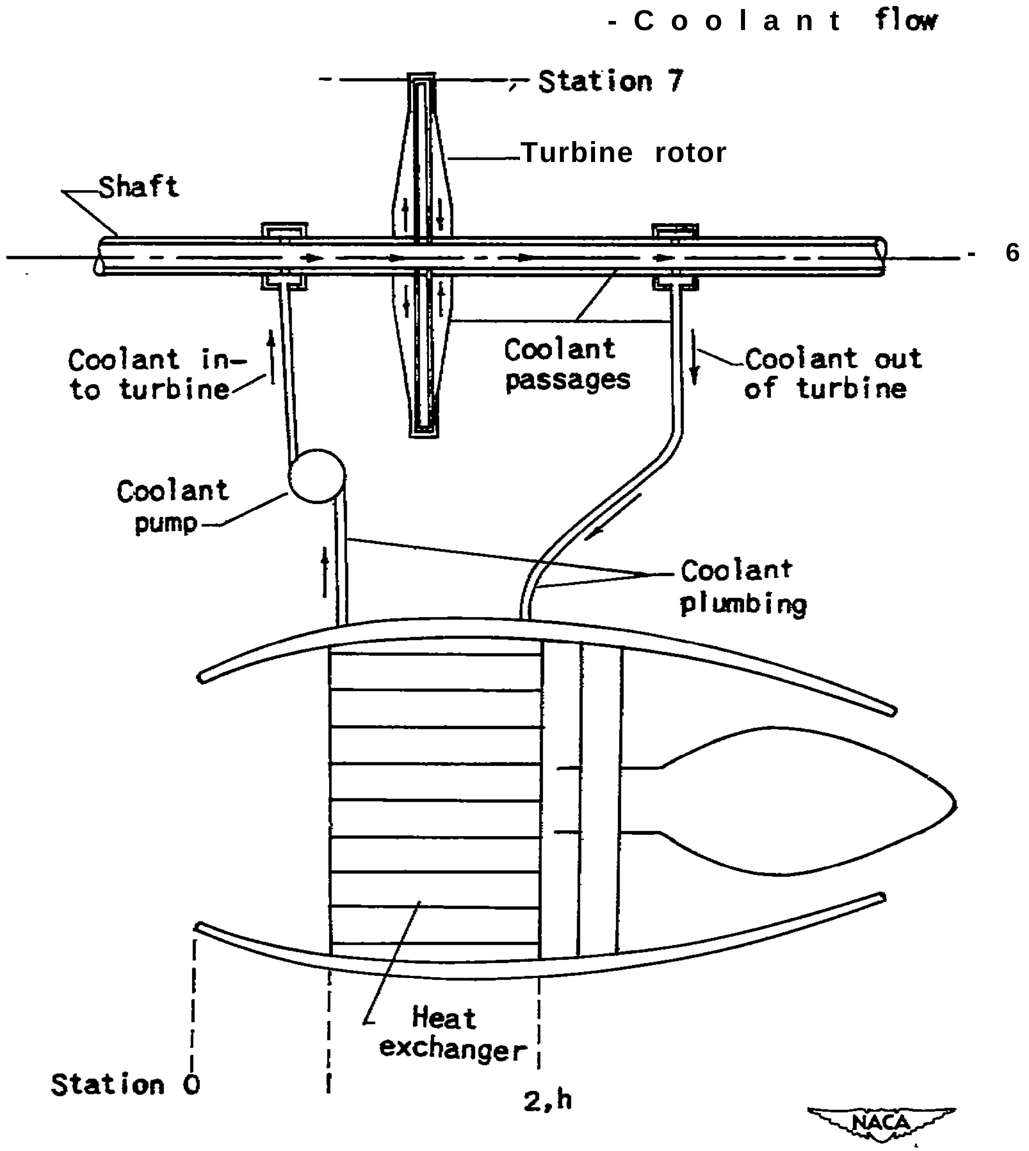

Figure 5. - Sketch of coolant circulation system for multistage-turbine turbine-propeller engine with water-cooled turbine blades.

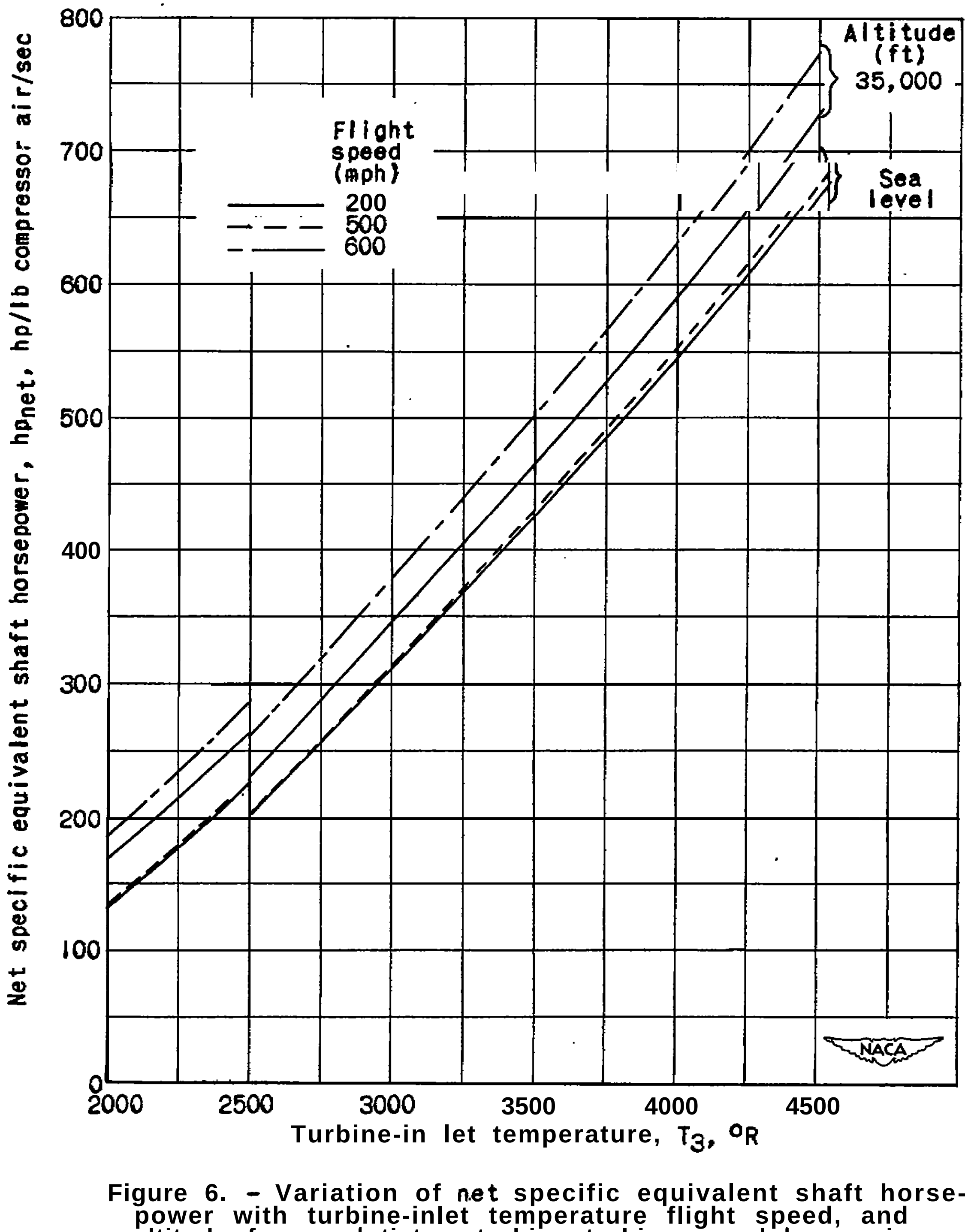

Figure 6. - Variation of net specific equivalent shaft horsepower with turbine-inlet temperature flight speed, and altitude for multistage-turbine turbine-propeller engine with water-cooled turbine blades.

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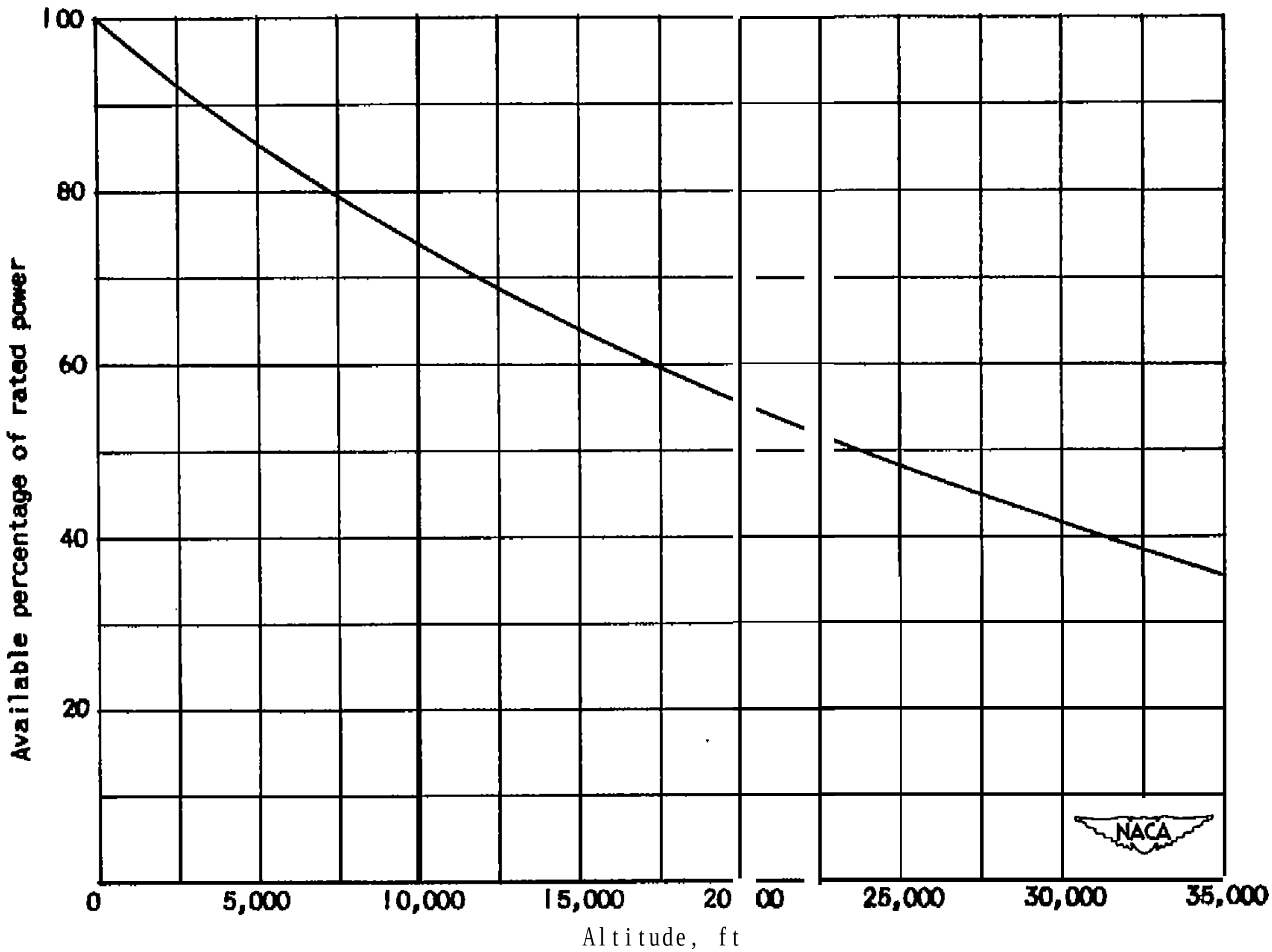


Figure 7. - Variation of percentage of sea-level power with altitude for multistage-turbine turbine-propeller engine with rater-cooled turbine biades.

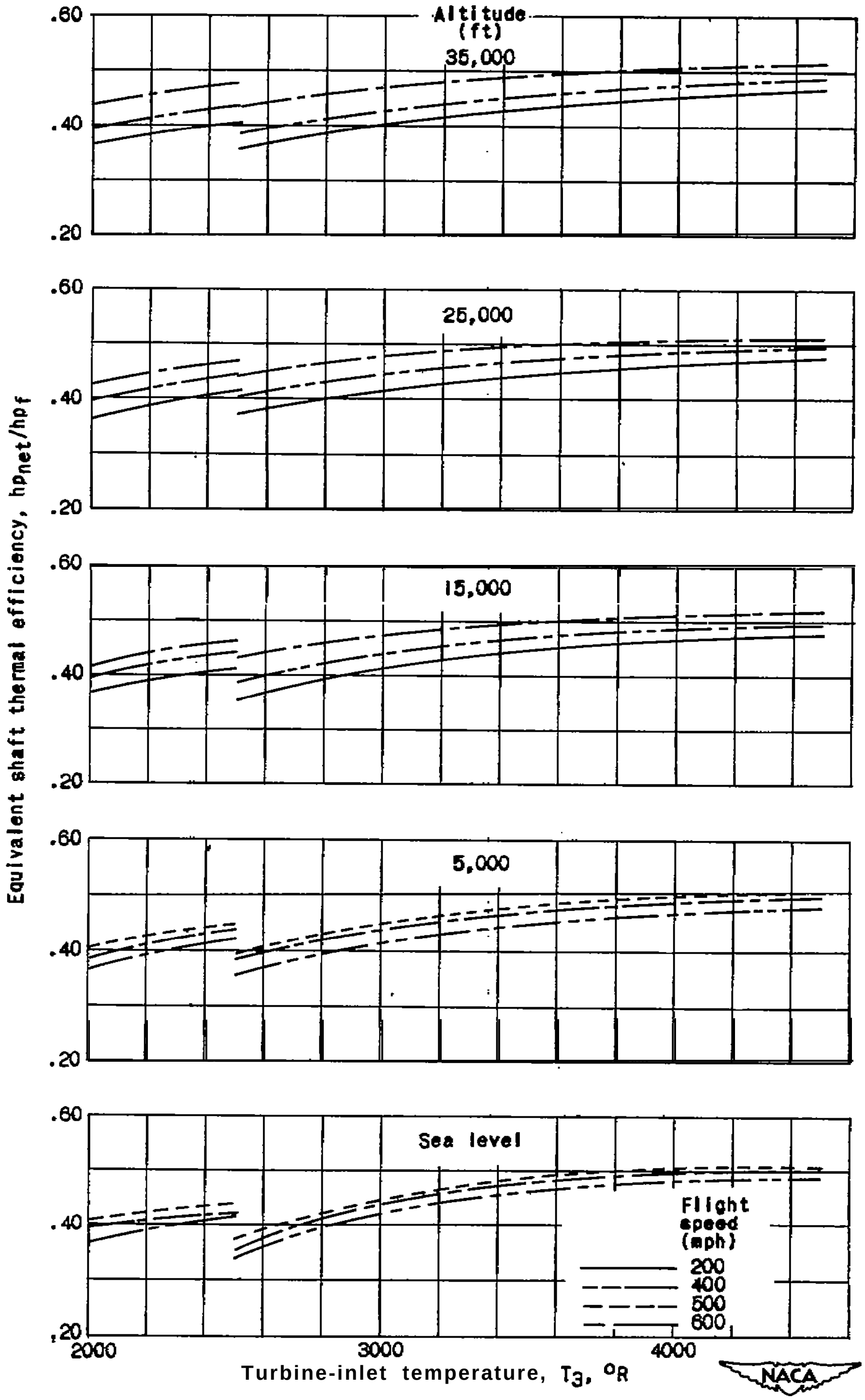


Figure 8. - Variation of equivalent shaft thermal efficiency with turbine-inlet temperature, flight speed, and altitude for multi-stage-turbine turbine-propeller engine with water-cooled turbine blades.

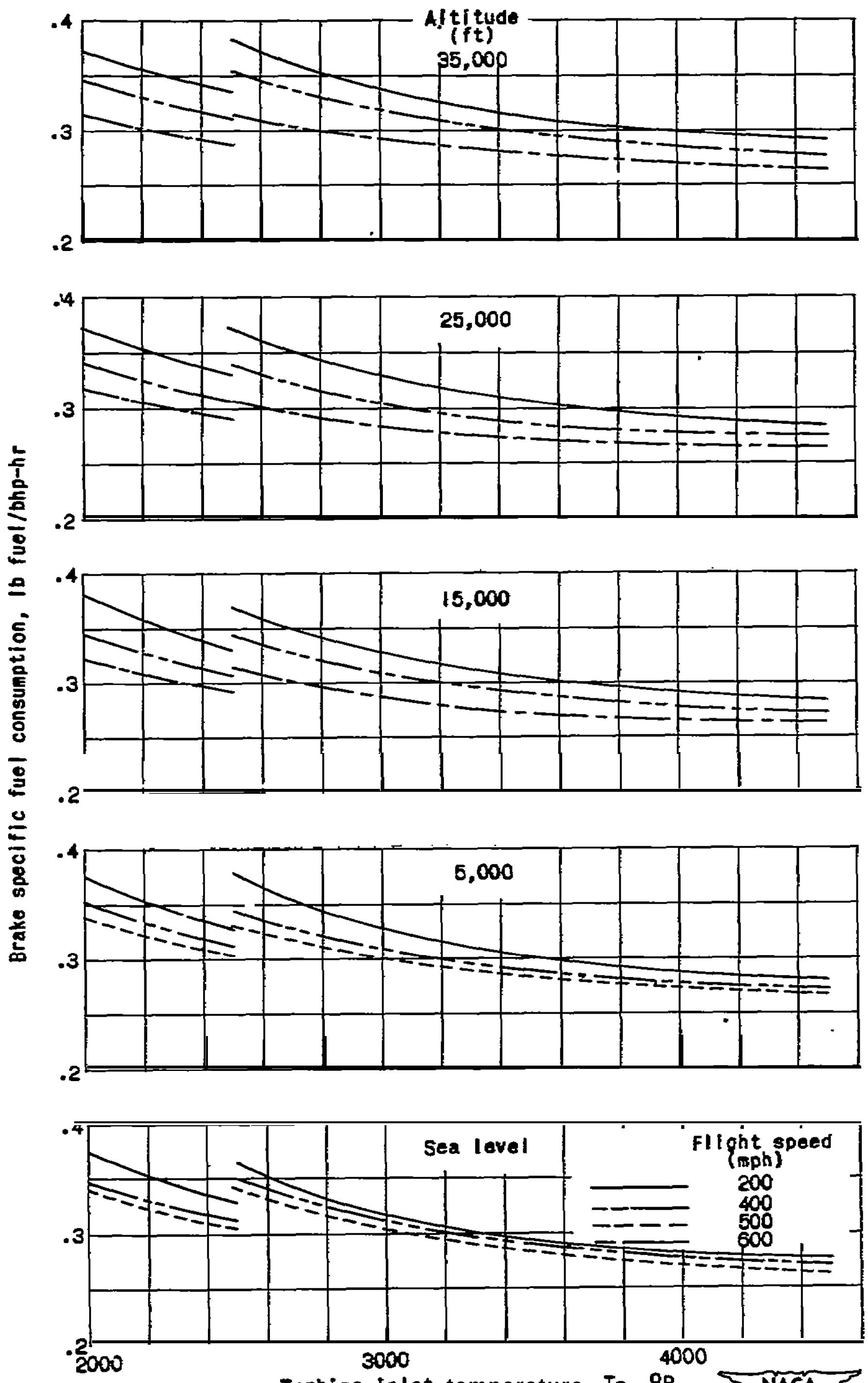
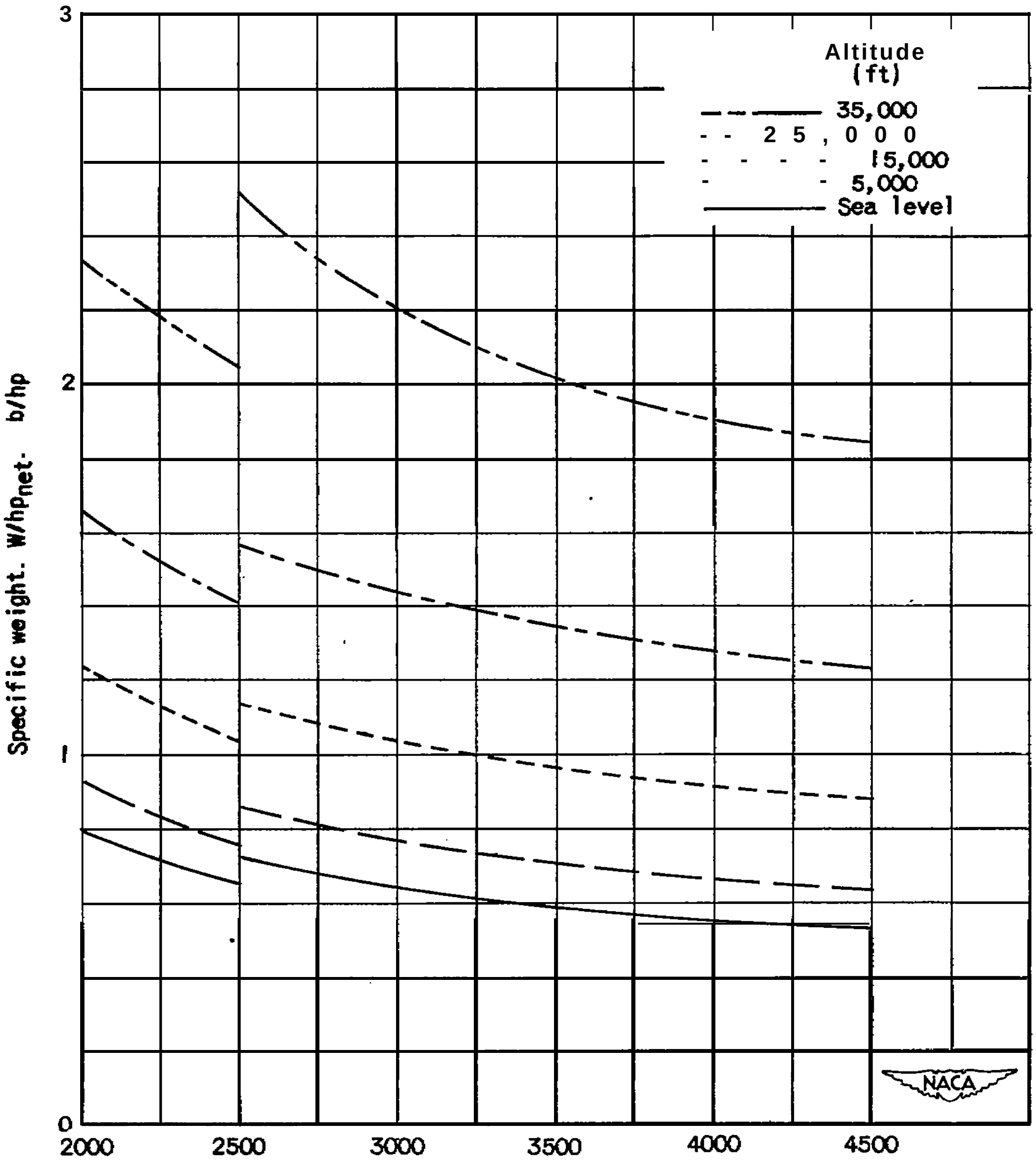


Figure 9. - Variation of brake specific fuel consumption with turbine-inlet temperature, flight speed, and altitude for multistage-turbine turbine-propeller engine with rater-cooled turbtne blades.



(a) Flight speed, 200 miles per hour.

Figure 10. – Variation of specific weight with turbine-inlet temperature, flight speed, and altitude for multistage-turbine turbine-propeller engine with water-cooled turbine blades.

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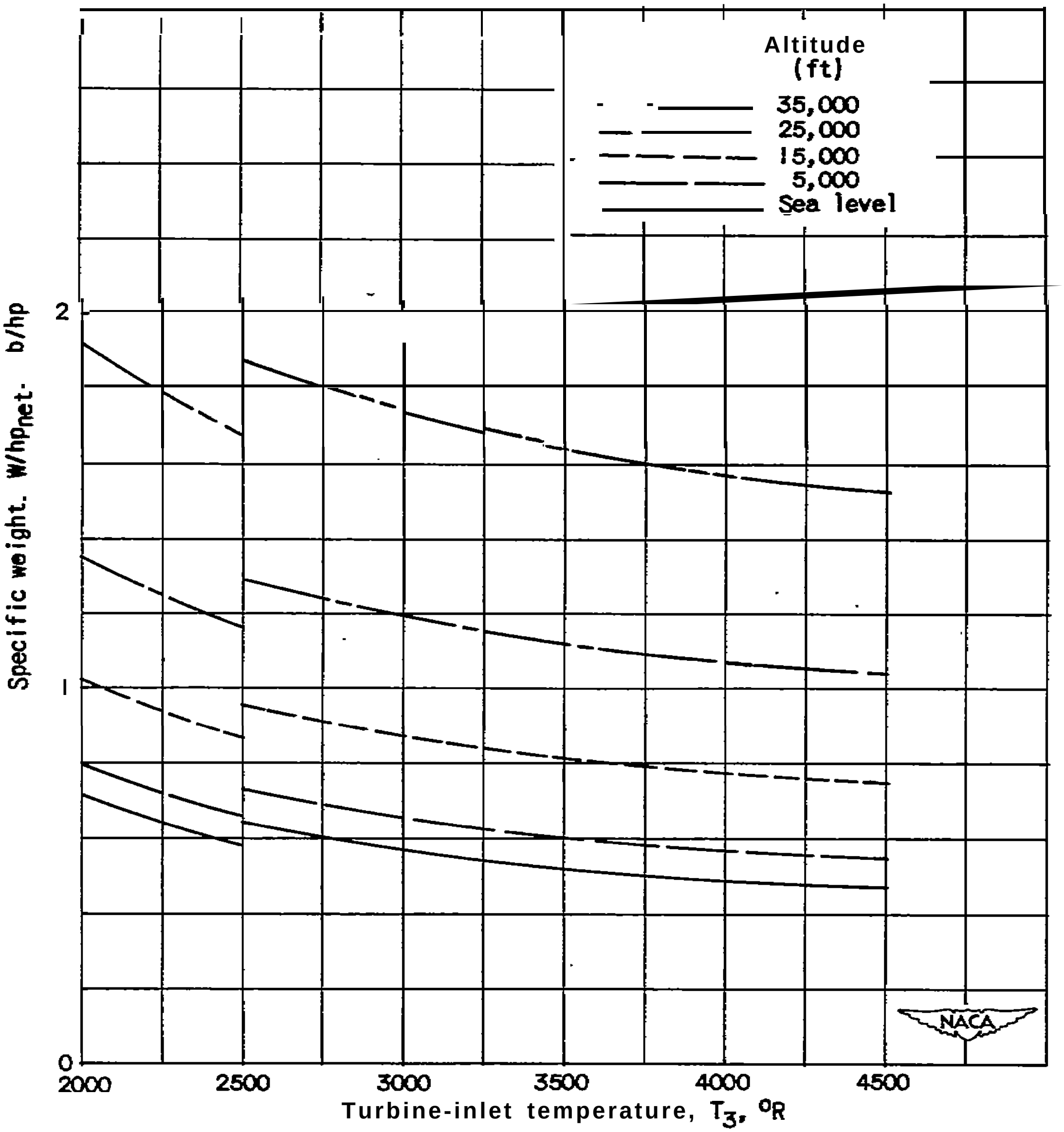
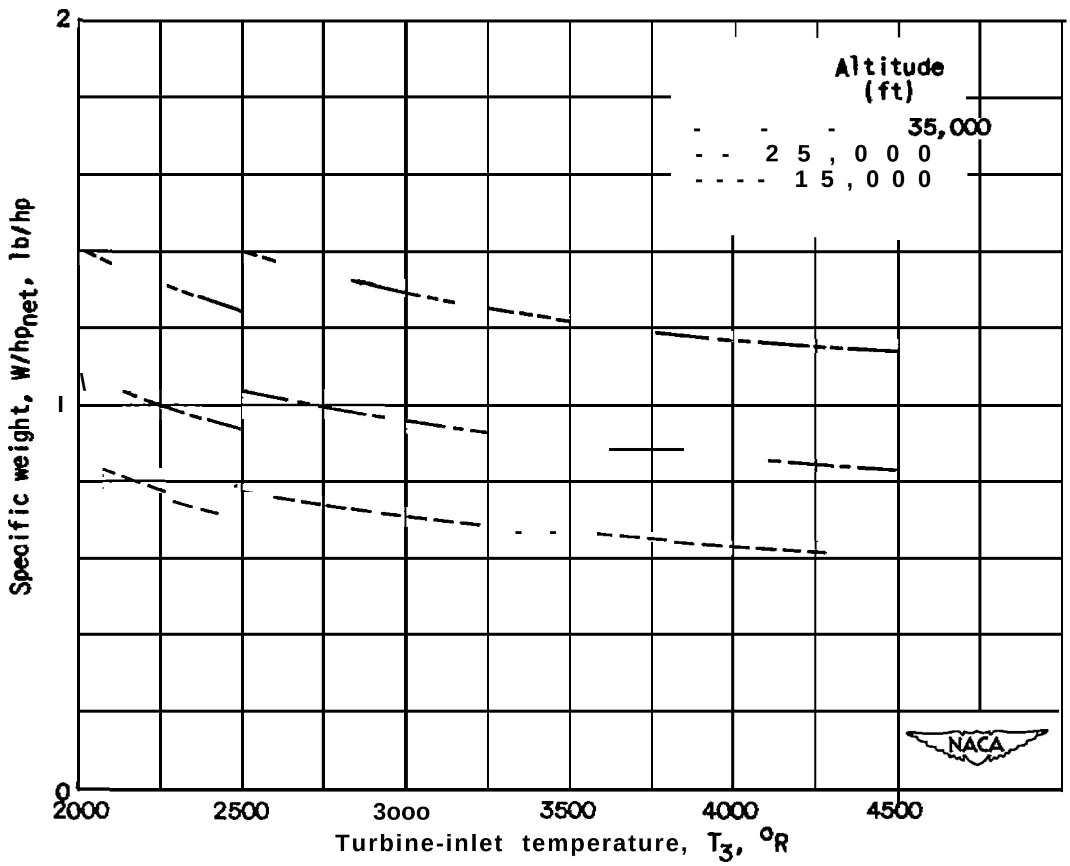


(b) Flight speed, 400 miles per hour.

Figure 10. – Continued. Variation of specific weight with turbine-inlet temperature, flight speed, and altitude for multistage-turbine turbine-propeller engine with water-cooled turbine blades.



(c) Flight speed, 600 miles per hour.

Figure 10. - Concluded. Variation of specific weight with turbine-inlet temperature, flight speed, and altitude for multistage-turbine turbine-propeller engine with water-cooled turbine blades.

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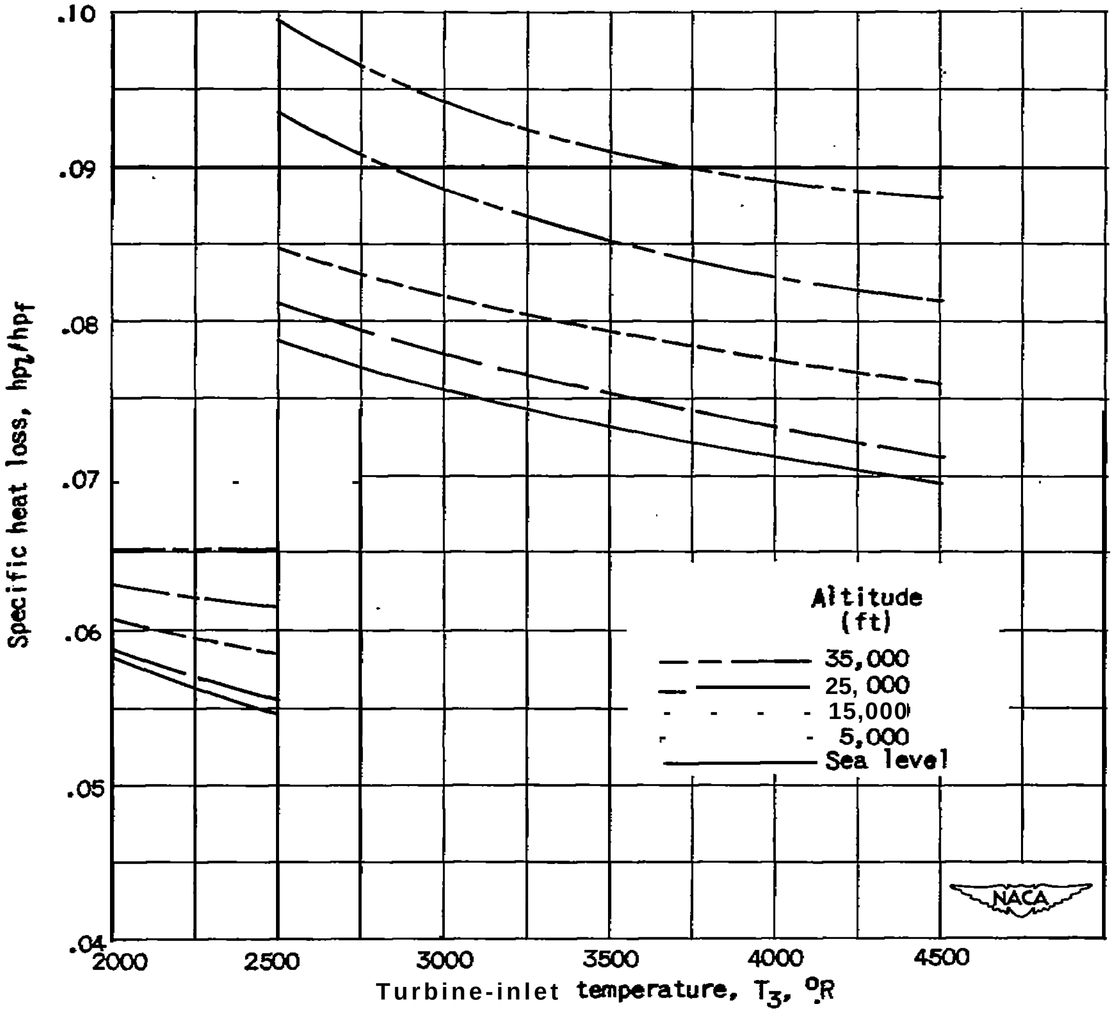
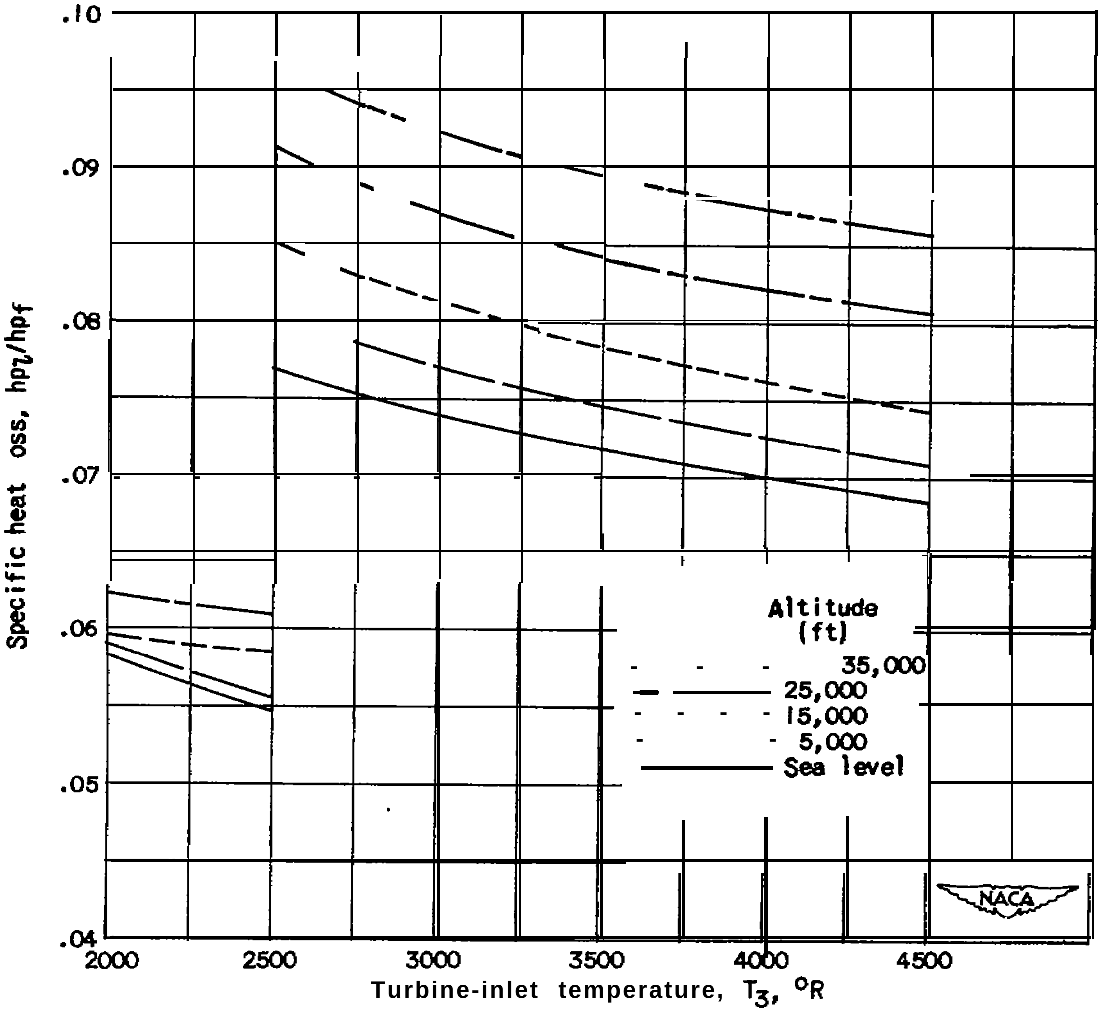


(a) Flight speed, 200 miles per hour.

Figure 11. – Variation of specific heat loss with turbine-inlet temperature, flight speed, and altitude for multistage-turbine turbine-propeller engine with water-cooled turbine blades.



(b) Flight speed, 400 miles per hour.

Figure 11. – Continued. Variation of specific heat loss with turbine-inlet temperature, flight speed, and altitude for multistage-turbine turbine-propeller engine with water-cooled turbine blades.

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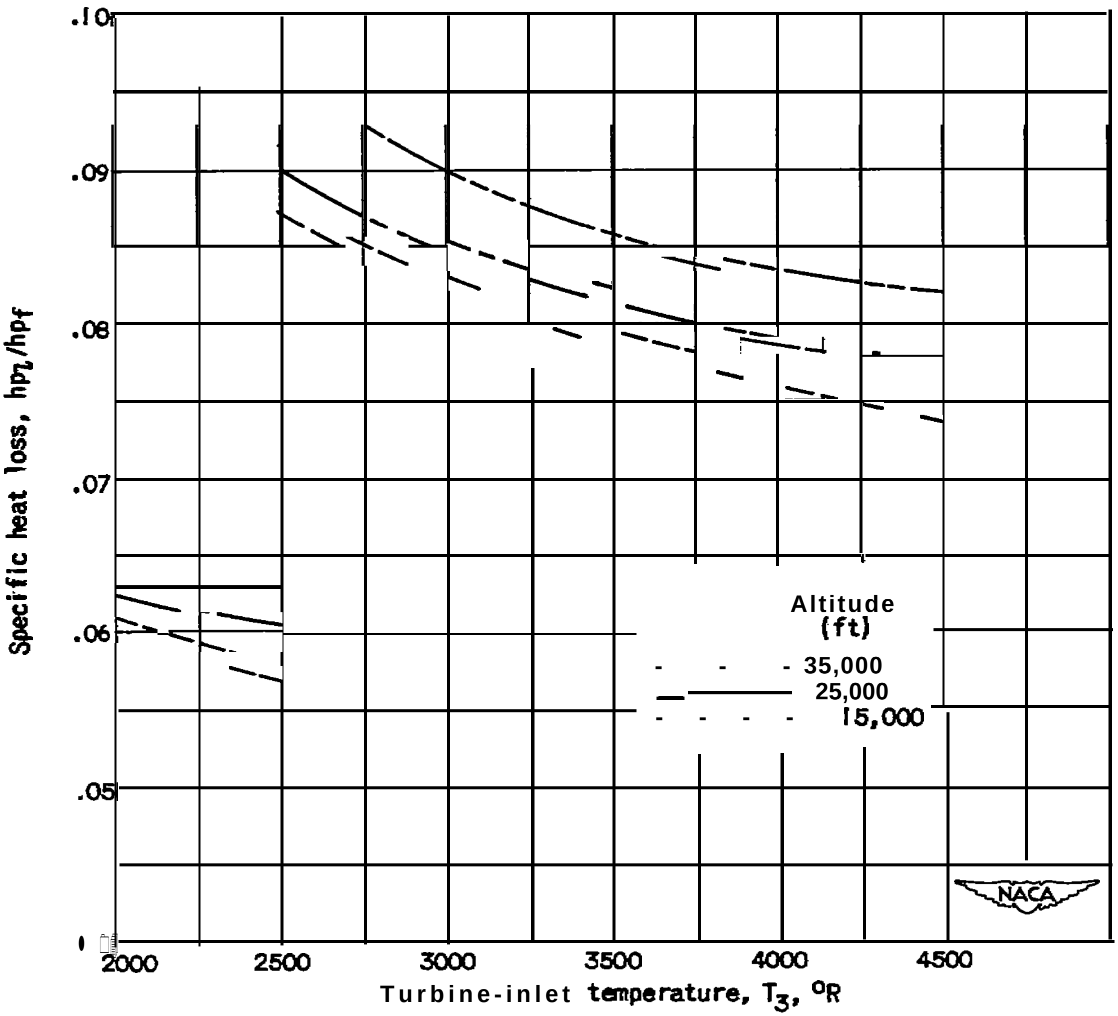
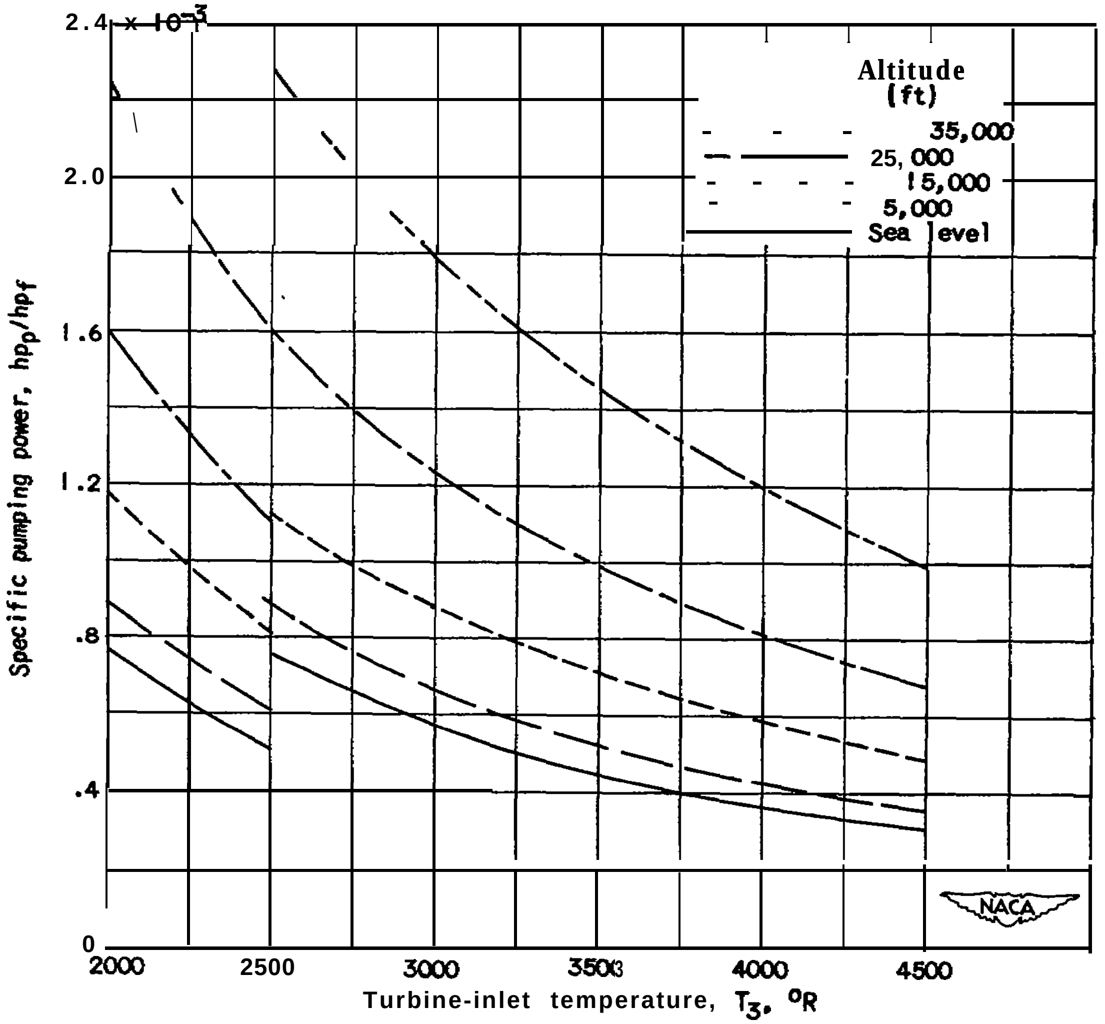


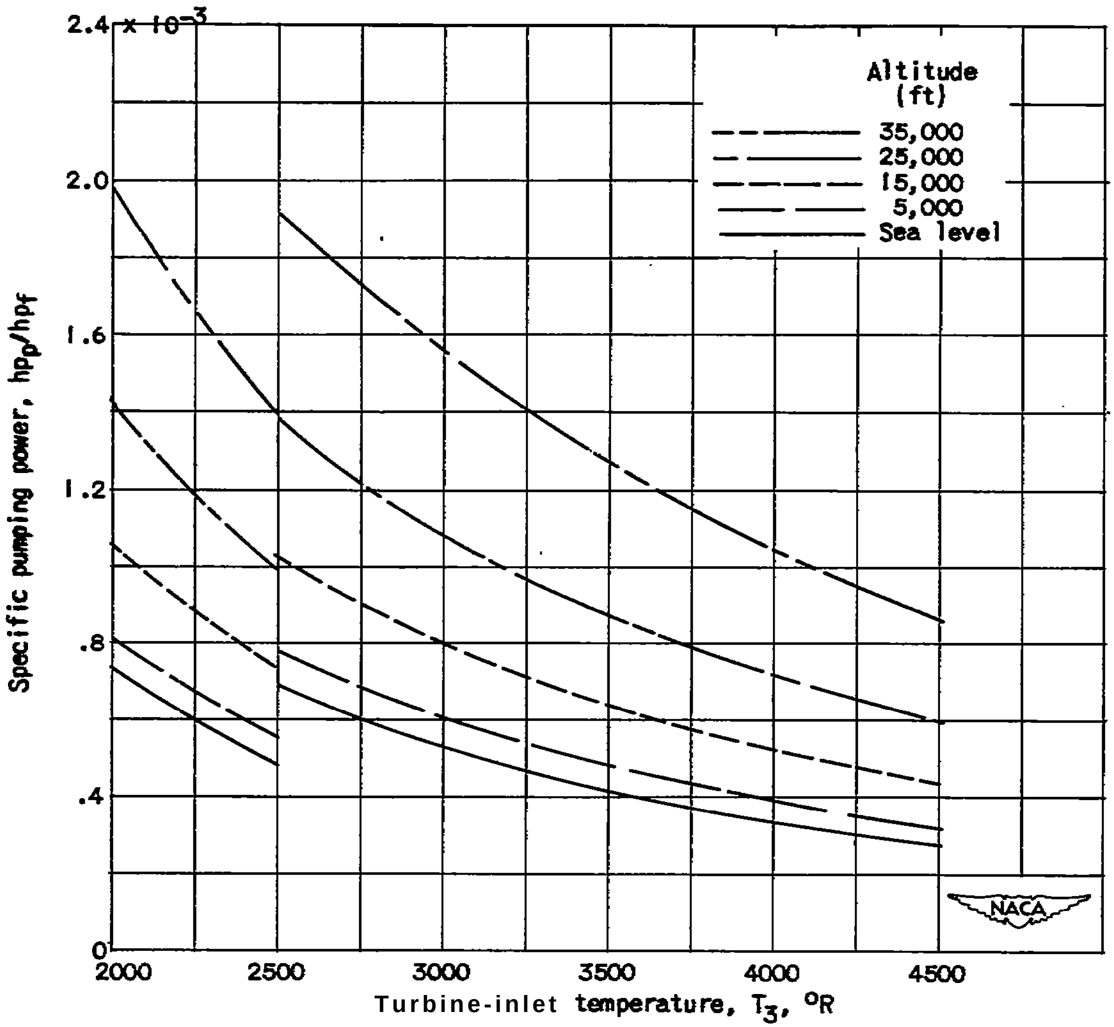
(c) Flight speed, 600 miles per hour.

Figure 11. - Concluded. Variation of specific heat loss with turbine-inlet temperature, flight speed, and altitude for multistage-turbine turbine-propeller engine with water-cooled turbine blades.



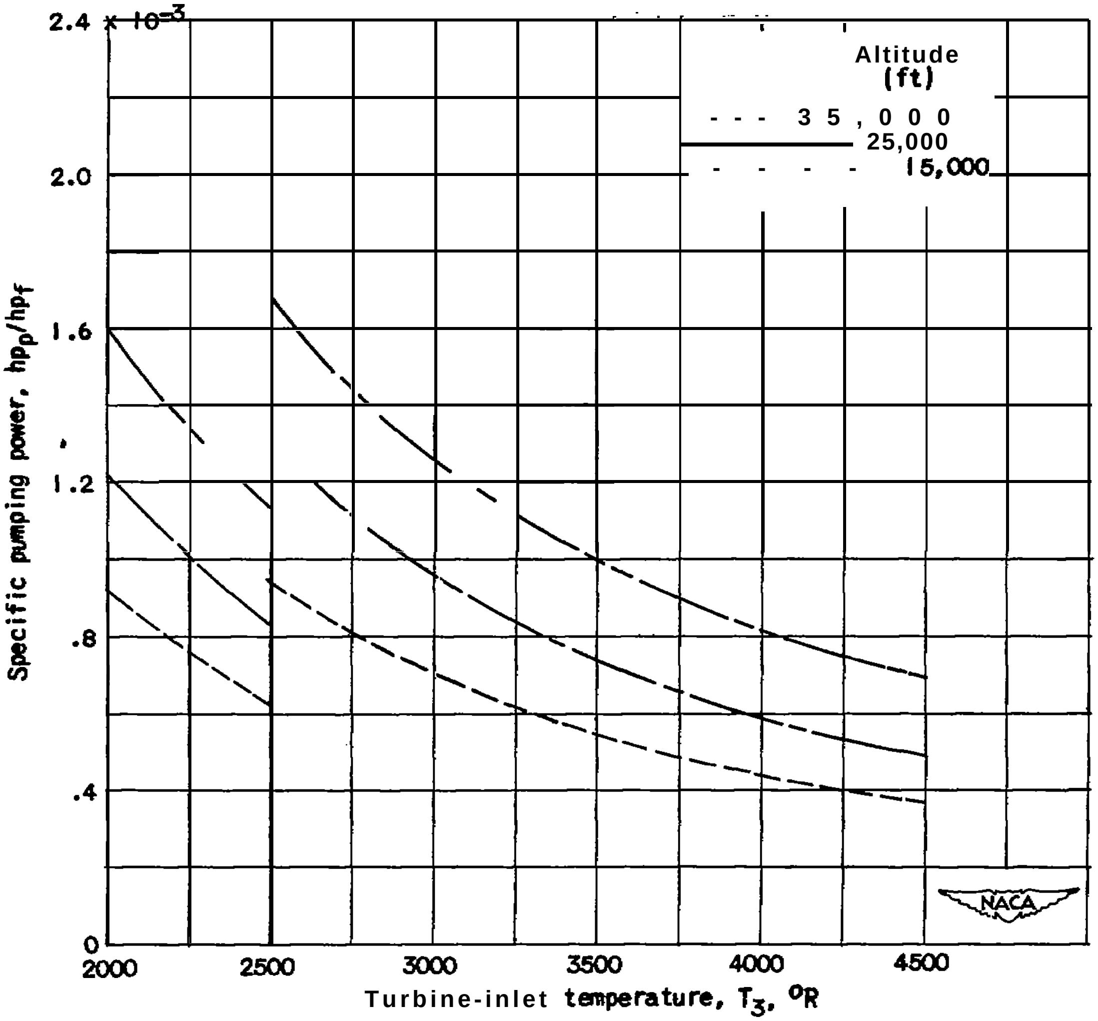
(a) Flight speed, 200 miles per hour.

Figure 12. – Variation of specific coolant pumping power with turbine-inlet temperature, flight speed, and altitude for multistage-turbine turbine-propeller engine with water-cooled turbine blades.



(b) Flight speed, 400 miles per hour.

Figure 12. - Continued. Variation of specific coolant pumping power with turbine-inlet temperature, flight speed, and altitude for multi-stage-turbine turbine-propeller engine with water-cooled turbine blades.



(c) Flight speed, 600 miles per hour.

Figure 12. - Concluded. Variation of specific coolant pumping power with turbine-inlet temperature, flight speed, and altitude for multistage-turbine turbine-propeller engine with water-cooled turbine blades.

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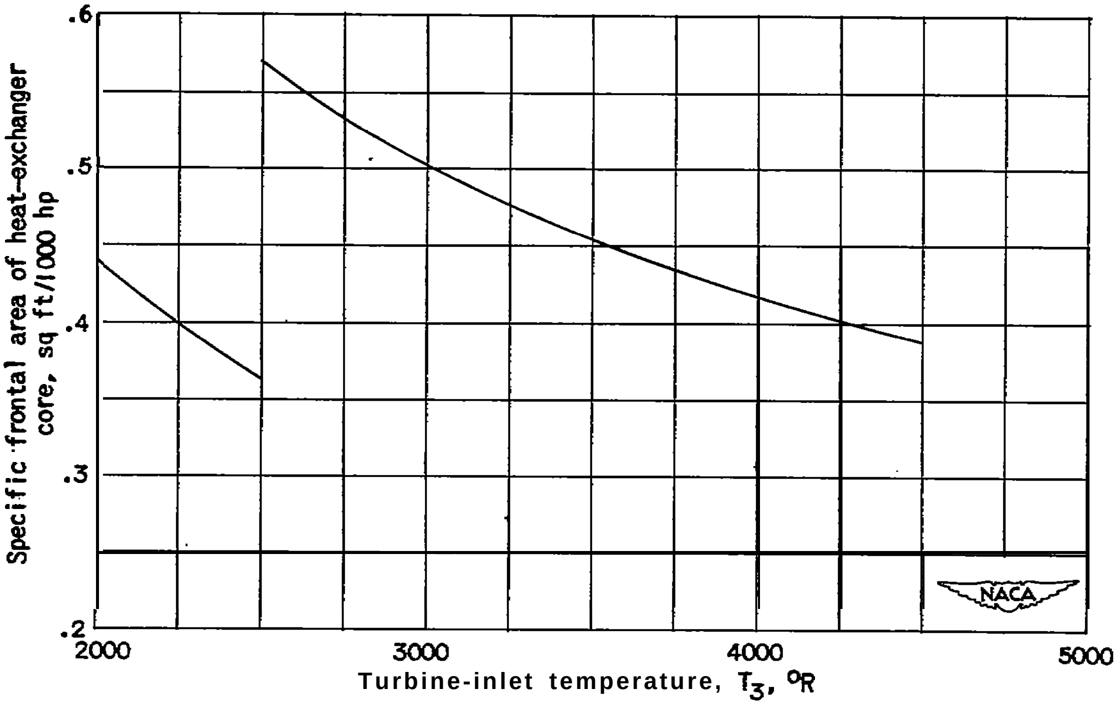

Figure 13. – Variation of specific frontal-area requirement far heat-exchanger core in multistage-turbine turbine-propeller engine with water-cooled turbine blades.