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RESEARCH MEMORANDUM

EXPERIMENTAL INVESTIGATION OF AIR-COOLED TURBINE BLADES IN TURBOJET ENGINE

III - ROTOR BLADES WITH 34 STEEL TUBES IN COOLING-AIR PASSAGES

By Robert O. Hickel and Gordon T. Smith

Lewis Flight Propulsion Laboratory
Cleveland, Ohio

NATIONAL ADVISORY COMMITTEE
FOR AERONAUTICS
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SUMMARY

An investigation is being conducted to determine experimentally the effectiveness of air-cooled turbine blades in a production turbojet engine. A number of cooled-blade configurations are being investigated; the results obtained with the third configuration, namely, a blade shell with 34 steel tubes inserted to increase the internal heat-transfer surface, are presented.

The turbine of a production turbojet engine was modified to accomodate two diametrically opposite air-cooled turbine blades. The air-cooled blades were so instrumented that a radial temperature distribution near the trailing edge and a peripheral temperature distribution at a distance of about one-third of the blade span from the base could be determined. Investigations of the cooled-blade temperatures were made over a range of cooling-air flows from about 0.005 to about 0.125 pound per second per blade at engine speeds ranging from 4000 to 9000 rpm.

The results of this investigation indicated that only a limited increase, if any, in cooling effectiveness was obtained with the 34-tube blade over that of the 10-tube and the 15-fin blades previously investigated.

The pressure loss through the 34-tube blade was correlated and compared with that of the 15-fin blade. The loss through the 15-fin blade was about 40- to 70-percent less than the loss through the 34-tube blade, depending on the weight flow.

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INTRODUCTION

An investigation of various air-cooled turbine blades installed in a production turbojet engine, which had a rated engine speed of 11,500 rpm, was started at the NACA Lewis laboratory in November 1949. This investigation has the dual objective of obtaining a cooled-blade configuration that will permit engine operation at current gas temperatures when nonstrategic turbine-blade materials are used, and of increasing the engine performance through use of higher maximum-cycle temperatures still using nonstrategic materials. The status and the preliminary analytical and experimental studies of cooled turbines that led to this investigation are discussed in considerable detail in reference 1.

The first cooled-blade configurations studied in the investigation were blades with 10 tubes and with 15 fins installed in blade shells to increase the internal heat-transfer surface of the blades. The results of the studies are reported in references 1 and 2 and show, in general, that the 15-fin blade provided more effective cooling than the 10-tube blade, particularly at the midchord position. Both blades exhibited chordwise temperature distributions, which showed that the blade temperature was considerably lower at the midchord than at the leading and trailing edges. In order to increase the internal heat-transfer area of the blade (and consequently reduce the temperature), particularly near the leading and trailing edges, a blade with smaller tubes than those used in the investigation reported in reference 1 was thought to be desirable. As a consequence, a blade having 34 steel tubes was fabricated and the results of an experimental investigation of the cooling effectiveness of this blade configuration are presented herein.

Two of these cooled blades were installed diametrically opposite each other in the turbine wheel in place of conventional uncooled blades. The profile of the cooled blades at the root was the same as the root profile of the conventional solid uncooled blades used in the turbine. The cooled blades were essentially of the same cross section from root to tip but were untwisted. One uncooled blade of the same profile as the cooled blade was placed next to each of the cooled blades. These air-cooled and modified solid blades were instrumented so that blade temperatures could be obtained. In addition, modified solid blades were placed on either side of the blades mentioned to obtain two cascades of four modified blades, each of the same untwisted profile. Cooling air was supplied to each air-cooled blade through a modified tail-cone and rotor assembly. The investigation covered a range of engine speeds from 4000 to 9000 rpm and a range of coolant flows from about 0.005 to about 0.125 pound per second per blade.

SYMBOLS

The following symbols are used in this report:

n exponent

p static pressure, inches mercury absolute

p' total pressure, inches mercury absolute

R ratio of coolant flow per blade to combustion-gas flow per blade

r radius from center of rotor, feet

T static temperature, $^{\circ}F$

T' total temperature, $^{\circ}F$

w weight-flow rate, pounds per second

$\Delta p'$ cooling-air pressure loss, inches mercury

η efficiency of compression in turbine wheel

ρ density, slugs per cubic foot

φ temperature-difference ratio, $(T_{g,e}-T_B)/(T_{g,e}-T_{a,e,h})$

ω angular velocity of rotor, radians per second

Subscripts:

A combustion air

a blade-cooling air

B cooled blade

c compressor

e effective

F fuel

g combustion gas

H hub of rotor

h root of blade

i inlet

m mixture of combustion gas and scavenge, bearing, and blade-cooling air in tail pipe

S stator

T blade tip

0 NACA standard sea-level conditions

APPARATUS AND INSTRUMENTATION

Engine and Blade Modifications

A production turbojet engine was modified to accomodate and supply cooling air to two air-cooled turbine blades. The modifications were the same as those described in detail in reference 1.

The hollow- and modified-blade sections were made of cast X-40 high-temperature alloy and were of the same profile as those described in reference 1. In order to increase the internal heat-transfer surface of the hollow blade, 34 tubes were inserted in each blade. The tubes were of two different materials and sizes: 18 tubes were made of stainless steel, 0.100-inch outside diameter with a wall thickness of 0.0125 inch; and 16 tubes were made of Inconel, 0.078-inch outside diameter with a wall thickness of 0.008 inch. The tubes extended through the blades from base to tip and were brazed to each other and to the blade shell with a commercial brazing alloy. Several views of a 34-tube blade are shown in figure 1. Although it appears in figure 1(a) that cooling air could flow through some of the spaces between the tubes (as was possible in the 10-tube blade of reference 1), this is not the case because brazing material fills most of the spanwise space between tubes. Consequently, all the cooling air passes through the inside of the 34 tubes and the two triangular-shaped passages near the leading and trailing edges.

Both the hollow- and solid-modified blades were welded to conventional blade bases that were modified to provide an air-inlet slot for the cooling air. A portion of this slot, which is of the same profile as the core of the blade, is shown in figure 1(b).

Instrumentation

The engine speed was measured by a chronotachometer. Temperatures at the compressor inlet and outlet were measured by shielded iron-constantan thermocouples and the compressor-outlet pressure was determined by total-head tubes and static-pressure taps located downstream of the diffuser section. The mass flow through the engine was determined from temperature and pressure measurements made with an airfoil-type survey rake in the tail pipe about 6 feet downstream of the turbine rotor. Fuel flow was measured by two rotameters connected in series. This instrumentation was the same as that described in greater detail in reference 1.

The location of the thermocouples on the rotor blades is indicated in figure 2. The cooled blades (fig. 2(a)) were instrumented so that a radial and a peripheral temperature distribution could be determined. Thermocouples A, C, and D provided the radial temperature distribution near the trailing edge, and thermocouples G, H, I, and J, the peripheral temperature distribution at a section about one-third the spanwise distance from the blade base. Thermocouples E and K measured the temperature of the incoming cooling air at the base of each cooled blade. Thermocouples F and L were located at corresponding positions near the leading edges of the two diametrically opposed solid untwisted blades (fig. 2(b)). Thermocouple M was located near the trailing edge on one of the modified-solid blades so that when used with thermocouple F, a more representative value of the effective gas temperature might be obtained. It was found, however, that thermocouple M was influenced considerably by the coolant flow from the adjacent cooled blade and consequently was not used and is not reported upon. Additional details of the blade thermocouple instrumentation are given in reference 1.

The blade cooling-air flow was controlled and measured in the same manner as described in reference 1. The cooling-air conditions at the rotor inlet were determined by instrumentation in the tail cone as described in reference 2.

For the static calibration of the pressure loss from the rotor hub to the cooled-blade base, the static pressure at the blade base was measured by a 0.030-inch-diameter static-pressure tube, which was inserted from the tip end of one of the cooled blades through one of the cooling-air tubes. The pressure in the blade base was measured at a point about $4\frac{1}{2}$ inches from the blade tip. The static-pressure tube was differentially connected with the total-pressure tube in the cooling-air duct immediately upstream of the rotor shown in figure 3 of reference 2. The pressure differential was measured with a mercury manometer.

EXPERIMENTAL PROCEDURE

Several series of programmed runs were made during this investigation. For each series, the engine speed was held constant and the blade-cooling air was varied by means of manually controlled valves in the supply line. During each series, a certain grouping of not more than six thermocouples could be connected at one time because of the limitation of the pickup system described in reference 1.

An initial series of runs was made at an engine speed of 4000 rpm with thermocouples C, E, F, I, K, and L connected to determine if the diametrically opposite sets of blades were operating at the same temperatures. Runs were then made with thermocouple groupings A, C, D, E, and F and G, H, I, J, K, and F at engine speeds of 4000, 5400, 6000, 8000, and 9000 rpm. Tests above 9000 rpm were not made because of difficulties that were experienced with oil seals on the thermocouple-pickup shaft. Inasmuch as the 34-tube blade did not exhibit the expected cooling improvements, it was believed to be most expedient to terminate this part of the investigation and proceed with investigations of other blade configurations.

A summary of the conditions at which the runs were made is given in table I. For the various series, the engine speed was varied from 4000 to 9000 rpm, the combustion-gas flow varied from about 20.0 to about 50.8 pounds per second, and the calculated turbine-inlet total temperature varied from about 992° to about 1215° F. The cooling-air flow per blade was varied from 0.004 to about 0.129 pound per second and the cooling-air temperature at the blade root varied from 47° to 138° F.

In order that the pressure loss through the cooled blades could be evaluated, a static calibration of the cooling-air system was conducted over a range of coolant flows from about 0.01 to 0.09 pound per second per blade. The calibration determined the pressure loss from the rotor hub to the blade bases.

CALCULATION PROCEDURES

The calculation procedures for correlating cooled- and solid-blade temperatures developed in reference 1 were used in this investigation.

A method of correlating the cooling-air pressure loss that occurs when cooling air passes through the rotating coolant passages is developed in reference 2. The method involves finding the value of the pressure-loss parameter

$$\frac{\rho_{a,H}}{\rho_0}\left[p'_{a,H} - \left(p'_m - \frac{\eta\omega^2 r_T^{\,2}\rho_{a,H}}{70.7}\right)\right]$$

for various coolant flows and engine speeds and plotting this result against coolant flow. The values of $\rho_{a,H}$, $p'_{a,H}$, and p'_m are experimentally determined, whereas ω and r_T depend upon engine speed and turbine size, respectively. The efficiency of compression in the radial coolant passages is represented by η in the previously presented parameter, and a value of $\eta = 0.27$ is found in reference 2 to be the value that best correlates the data of that report. The same value of η was used in the present investigation because the rotating cooling-air passages leading to the cooled blades were the same as those of reference 2.

The cooling-air pressure loss determined from the pressure-loss parameter just described was for the entire radial coolant passage from rotor hub to blade tip. In order to determine the pressure loss through the blade only, the cooling-air pressure loss from the rotor hub to the blade base (as determined by static investigations) was subtracted from the pressure-loss parameter. The pressure loss through the blade was then

$$\Delta p'_B = \frac{\rho_{a,H}}{\rho_0}\left\{\left[p'_{a,H} - \left(p'_m - \frac{\eta\omega^2 r_T^{\,2}\rho_{a,H}}{70.7}\right)\right] - \left(p_{a,H} - p_{a,h}\right)\right\}$$

The calculation procedure for comparing the temperatures of various cooled-blade configurations for NACA standard sea-level conditions at the engine inlet is discussed in reference 2 and employed herein. Values of effective gas temperature and effective blade-cooling-air temperature for standard conditions, which were necessary for these calculations, were obtained from figure 17 of reference 2. These values can be used with very little error because the type engine employed in the present investigation was the same as that reported in reference 2.

RESULTS AND DISCUSSION

The results of an experimental investigation to determine the cooling effectiveness of an air-cooled turbine blade with 34 steel tubes are presented in the following paragraphs.

Basic Temperature Data

Comparison of blade and cooling-air temperatures. - The results of the first series of runs are given in figure 3 in order that the temperatures of the cooling air at the root of each blade and the temperatures of the cooled and uncooled blades may be compared at approximately the same conditions. A comparison of the cooling-air temperatures at the blade root of each cooled blade (thermocouples E and K) is shown in figure 3(a). The data points fall along a 45° line with very little deviation, indicating that the distribution of air to the two cooled blades was about equal. A comparison of the temperatures near the trailing edge at about 35 percent of the blade span (thermocouples C and I) of each cooled blade is presented in figure 3(b). A 45° line has been drawn for ease in comparing the data. It can be seen that thermocouple I is 20° to 25° F hotter than C. Inasmuch as the cooling-air distribution to each blade appeared to be about equal as evidenced by the temperatures in figure 3(a), it is believed that slight differences in the geometry of coolant passages, which were unavoidable in the fabrication process, may have resulted in the trailing-edge temperature of the blade with the peripheral temperature survey (thermocouples G, I, and J) being somewhat hotter than the blade with the spanwise survey at the trailing edge (A, C, and D). A comparison of the leading-edge temperatures of the two solid uncooled blades (thermocouples F and L), which are adjacent to the cooled blades, is shown in figure 3(c). The agreement is very good and because the difference between the two solid-blade temperatures appeared to be negligible, thermocouple F was arbitrarily used in all later runs.

Effect of cooling-air flow on blade, effective-gas, and cooling-air temperatures at engine speed of 9000 rpm. - The basic temperature data taken for an engine speed of 9000 rpm is shown in figure 4; the same type of data was obtained at each of the engine speeds. In general, there was appreciable cooling of the air-cooled blades, even at low coolant flows. The cooling obtained at the midchord positions was much greater than that at the leading and trailing edges. The solid-blade temperature remained essentially constant over the range of coolant flows; however, the small variations of these temperatures, particularly that of thermocouple F in figure 4(b), was attributed to changes in the ambient temperature of the test cell, which in turn influenced the temperature of the combustion gases. The cooling-air temperature at the blade root decreased as the coolant flow increased as would be expected. The behavior of the blade temperatures and the cooling-air temperature was similar to that observed in references 1 and 2.

Correlated Cooled-Blade Temperatures

Correlation curves for each cooled-blade-thermocouple location are shown in figure 5. Each set of curves is for a particular thermocouple location and includes curves for engine speeds of 4000, 6000, and 9000 rpm. The development of this correlation procedure is discussed in reference 1. These curves can be used to predict cooled-blade temperatures with considerable accuracy when values of effective combustion-gas temperature, effective blade cooling-air temperature, combustion-gas flow, coolant flow, and engine speed are known. Generally, the data for each thermocouple could be represented by a family of curves, with the temperature-difference ratio φ decreasing as the coolant flow decreases and the engine speed increases.

The dashed-line curves in figure 5 indicate the values of φ for constant ratios of coolant flow to combustion-gas flow R of 0.05 and 0.10. For constant values of R, the value of φ for thermocouples A, G, H, and J showed a consistant decrease with increase in engine speed. For thermocouples C, D, and I, however, the value of φ slightly increased as the engine speed increased from 4000 to 6000 rpm and then rapidly decreased above 6000 rpm.

Temperature-ratio data that follow trends similar to those exhibited by thermocouple C in figure 5(b) (that is, φ curves that are straight lines and parallel to one another for all engine speeds through certain coolant-flow ranges) can be further correlated in another manner, as shown in figure 6. The data values of φ for thermocouple C were divided by the measured cooling-air-flow rates raised to a power n, which represent the slopes of the curves in figure 5(b). The resulting value was plotted against the combustion-gas-flow rate for each engine speed. Two curves result, one for cooling air-flow rates below 0.02 pound of air per second and one for cooling-air-flow rates above 0.02 pound of air per second. The corresponding values of n are 0.053 and 0.173, respectively. Although some of the data points seem to be quite far from the lines faired through the data, the deviation of the line from the data would have a small effect on the blade temperature calculated. For example, at an engine speed of 5400 rpm and a coolant flow of 0.05 pound per second, an effective gas temperature of 1000° F and a cooling-air temperature of 100° F, the blade temperature computed from the curve would be only about $2\frac{1}{4}$ percent less than that computed on the basis of the data point farthest from the line.

Correlated Solid-Blade Temperatures

It is desirable that the solid-blade temperature (effective gas temperature) be calculable for any set of known engine conditions; such as engine speed, gas flow, and fuel flow. A method of calculating the solid-blade temperature was developed in reference 1 and the same method was employed to compute the solid-blade temperatures for this investigation. A plot of the calculated solid-blade temperatures against the measured solid-blade temperatures for the same set of engine conditions is shown in figure 7 for engine speeds ranging from 4000 to 9000 rpm. A 45° line on the plot represents the mean value of the data quite well. The most extreme deviation of a data point from the 45° line is about 25° F, and has a small effect, about $2\frac{1}{2}$ percent, on the calculated blade temperature when the temperature-difference ratio curves of figure 5 are employed. These correlations compare favorably with those presented in references 1 and 2 and it was concluded that the solid-blade or effective gas temperatures can be calculated by the procedure given in reference 1 with satisfactory accuracy.

Correlated Cooling-Air Pressure Loss

The upper curve of figure 8 shows the correlation of the cooling-air pressure loss through the rotating coolant passages. The pressure-loss parameter

$$\frac{\rho_{a,H}}{\rho_0}\left[p'_{a,H} - \left(p'_m - \frac{\eta\omega^2 r_T^2 \rho_{a,H}}{70.7}\right)\right]$$

is plotted against coolant flow for engine speeds of 6000, 8000, and 9000 rpm. The data correlate well; most of the data points are within 2 inches of mercury of the curve faired through the data.

The lower curve of figure 8 represents the pressure loss from the rotor hub to the blade base $(p_{a,H} - p_{a,h})$ as determined by the static calibration. The pressure loss from the blade base to the blade tip is then the difference in pressure loss between the two curves of figure 8. The pressure loss through the 34-tube blade for various coolant flows was plotted on logarithmic coordinates and is shown by the solid line in figure 9. The pressure loss through the 15-fin blade (investigated in reference 2), which was obtained by subtracting the pressure drop from rotor hub to blade base from the pressure-drop parameter of figure 10 of reference 2, is also shown in figure 9. The pressure drop from rotor hub to blade base is the same for the investigation reported in reference 2 as for the present investigation, because the radial coolant passages between the rotor hub to the blade bases were essentially the same in each investigation.

COMPARISON OF 34-TUBE, 10-TUBE, and 15-FIN BLADES

Blade Geometry

A comparison of several pertinant internal geometry factors of the 34-tube, 10-tube, and 15-fin blades is made in table II, and a schematic diagram of the three configurations is shown in figure 10. The 10-tube and 15-fin blades are discussed in detail in references 1 and 2, respectively.

The total free-flow area, the inside-surface area exposed to cooling-air, the total perimeter of cooling-air passages, and the hydraulic diameter of the 34-tube blades are considerably less than those of the 10-tube and 15-fin blades (table II). The geometric factors of the 34-tube blades were expected to be more favorable, possibly exceeding those of the 10-tube blade and approaching or exceeding those of the 15-fin blade. This improvement was prevented, however, by fabrication difficulties, which resulted in blockage by brazing material of the irregular-shaped passages formed by the outer surfaces of the tubes shown in figure 10(a). Thus in the 34-tube blade, cooling air passed only through the interiors of the 34 tubes and the two triangular-shaped passages at the leading and trailing edges. In the 10-tube blade (fig. 10(b)), the tubes were sufficiently large so that cooling air passed through the irregular-shaped passages formed by the outside walls of the tubes and the blade wall, as well as through the tubes. In the 15-fin blade, (fig. 10(c)), blockage of the spaces between the fins by brazing material was no problem. It is possible that improved fabrication techniques will permit the more efficient use of a large number of small tubes; however, it is apparent that only a limited increase in the heat-transfer effectiveness of the blade can be obtained by using increased numbers of tubes. From the geometry factors presented in table II, it appears that the 34-tube blades will not cool as well as either the 15-fin or 10-tube blades.

Chordwise Temperature Distribution

Comparisons of the chordwise temperature distribution of the 34-tube, 10-tube, and 15-fin blades are shown in figure 11 for an engine speed of 9000 rpm and coolant flows equivalent to 5 and 10 percent of the combustion-gas flow. The temperatures for thermocouples G, H, I, and J (peripheral temperatures of 34-tube blade), F (solid-blade or effective gas temperature), and K (effective cooling-air temperature at the blade root) were obtained from the curves of figure 4(b) at coolant flows of 0.047 and 0.094 pound per second, which correspond to coolant flows equivalent to 5 and 10 percent of the combustion-gas flow, respectively. The blade temperatures for the 10-tube and 15-fin blades were computed from the equation

$$T_B = T_{g,e} - \varphi(T_{g,e} - T_{a,e,h})$$

The values of φ for substitution in this equation were obtained from references 1 and 2 for coolant flows equivalent to 5 and 10 percent of the combustion-gas flow in those investigations, and the values of effective gas temperature $T_{g,e}$ and effective blade cooling-air temperature $T_{a,e,h}$ were obtained from figure 4(b) at the coolant flows mentioned. The temperature values of the 10-tube and 15-fin blades were computed in this manner for the engine conditions that existed at the time the data for the 34-tube blade were obtained; a valid comparison of the temperatures was thus obtained.

All three blade configurations have similar chordwise temperature distributions (fig. 11); the temperatures are considerably lower near the midchord than near the leading and trailing edges. Although the leading and trailing edges are hotter than the midchord positions, they are 150° F or more below the solid-blade temperatures, depending on the blade configuration and coolant-to-gas-flow ratio.

The 34-tube blade generally is hotter than the 10-tube or 15-fin blade; for example, in figure 11(a) at a coolant flow equivalent to 5 percent of the combustion-gas flow, the 34-tube blade was about 10° F higher than either the 10-tube or 15-fin blade for a point near the leading edge (thermocouple G). Near the trailing edge the 34-tube blade was about the same temperature as the 10-tube blade but 43° F higher than the 15-fin blade. At a point near the midchord on the pressure surface (thermocouple H) the 34-tube blade is 34° and 90° F hotter than the 10-tube and 15-fin blades, respectively. For the midchord point on the suction surface thermocouple J (fig. 11(b)), the 34-tube blade is 83° F cooler than the 10-tube blade and 103° F hotter than the 15-fin blade. The 34-tube blade is markedly cooler than the 10-tube blade at this position because of the rapid increase of the temperature at position J of the 10-tube blade for engine speeds between about 6000 and 9000 rpm, as observed in reference 1.

Effect of Engine Speed on Chordwise Temperature

For Standard Sea-Level Inlet Conditions

A comparison of the temperatures at thermocouple locations G, H, I, and J (chordwise temperatures at 35-percent span) for the 34-tube, 10-tube, and 15-fin blade for standard NACA sea-level inlet conditions is presented in figure 12. The temperatures of the 34-tube blade were computed by the method previously described and the temperatures for the 10-tube and 15-fin blades were obtained from reference 2.

For the leading-edge position (thermocouple G, fig. 12(a)), the 34-tube blade compared favorably with the 10-tube and 15-fin blades for a coolant flow equivalent to 5 percent of the combustion-gas flow. At a coolant-flow rate equivalent to $R = 0.10$, however, the 34-tube blade was hotter than either the 10-tube or 15-fin blade throughout most of the speed range.

At the midchord position on the pressure surface of the blade (thermocouple H, fig. 12(b)), the 34-tube blade was cooler than the 10-tube blade for engine speeds up to about 8500 rpm. The 34-tube blade was about 70° to 120° F hotter than the 15-fin blade throughout the entire speed range.

For the suction side of the blade near the midchord (thermocouple J, fig. 12(c)), the 34-tube blade is a maximum of 35° F cooler than the 10-tube blade for engine speeds up to 6000 rpm. Above 6000 rpm the 34-tube blade is considerably cooler than the 10-tube blade, being a maximum of about 130° F cooler at an engine speed of 9000 rpm and a coolant-to-gas-flow ratio of 0.10. Although data for the 34-tube blades are unavailable for engine speeds above 9000 rpm, it appears that at 10,000 rpm the 34-tube blade would be as hot or hotter than the 10-tube blade because of the rapid decrease in temperature exhibited by the 10-tube blade at engine speeds above 9000 rpm. The 34-tube blade is 50° to 75° F hotter than the 15-fin blade throughout most of the speed range.

At the trailing edge (thermocouple I, fig. 12(d)), the temperature of the 34-tube blade is about the same as the 15-fin blade for engine speeds between about 5000 to 8500 rpm, and is about 15° to 30° F cooler than the 10-tube blade in this speed range. Above 8500 rpm, the temperature of the 34-tube blade increases more rapidly than that of either the 10-tube or 15-fin blade and at 9000 rpm it is about the same temperature as the 10-tube blade and about 30° F hotter than the 15-fin blade. From the shape of the curve for the 34-tube blade, its trailing-edge temperature would be considerably above the other blades at 10,000 rpm.

These curves indicate that generally the 34-tube blade does not cool as effectively as the 15-fin blade, and provides only a slight improvement over the 10-tube blade for engine speeds below about 9000 rpm. It appears that above 9000 rpm, where the cooling performance is most vital, the cooling effectiveness of the 34-tube blade would be inferior to that of either the 10-tube or 15-fin blade.

Cooling-Air Pressure Loss for 34-Tube and 15-Fin Blades

A factor that is of importance in any air-cooled turbine installation is the pressure drop required to force the coolant through the rotor. The pressure drop through the 15-fin blade is about 70-percent less than that of the 34-tube blade at a coolant flow of 0.02 pound per second (fig. 9). At a coolant flow of 0.09 pound per second, the 15-fin blade requires about 40-percent less pressure drop. It is expected that the pressure drop through the 34-tube blade would be greater than that of the 15-fin blade because the smaller and more numerous passages increase the resistance to the coolant flow.

SUMMARY OF RESULTS

The following results were obtained in an experimental investigation to determine the effectiveness on air-cooling turbine blades with 34 steel tubes inserted for the purpose of increasing the internal heat-transfer surface:

1. The 34-tube blade exhibited the same general cooling characteristics as the 10-tube and 15-fin blades previously investigated; that is, the midchord portion of the blade cooled satisfactorily, whereas the leading and trailing edges were much hotter.

2. The cooling effectiveness of the 34-tube blade was slightly better than that of the 10-tube blade at engine speeds below about 8500 rpm for the midchord and trailing-edge positions. At an engine

speed of 9000 rpm, the 34-tube blade exhibited no advantage over the 10-tube blade except at the midchord position on the suction surface. For the entire range of engine speeds investigated, the 34-tube blade generally does not cool as well as the 15-fin blade. The trend of the data indicated that at engine speeds greater than 10,000 rpm the 34-tube blade would be considerably hotter than either the 10-tube or 15-fin blade.

3. Correlation of the cooling-air pressure drop through the cooled blades indicated that the pressure drop of the 15-fin blade was about 40- to 70-percent less than that of the 34-tube blade depending upon the coolant flow.

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

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2. Hickel, Robert O., and Ellerbrock, Herman H., Jr.: Experimental Investigation of Air-Cooled Turbine Blades in Turbojet Engine. II - Rotor Blades with 15 Fins in Cooling-Air Passages. NACA RM E50I14

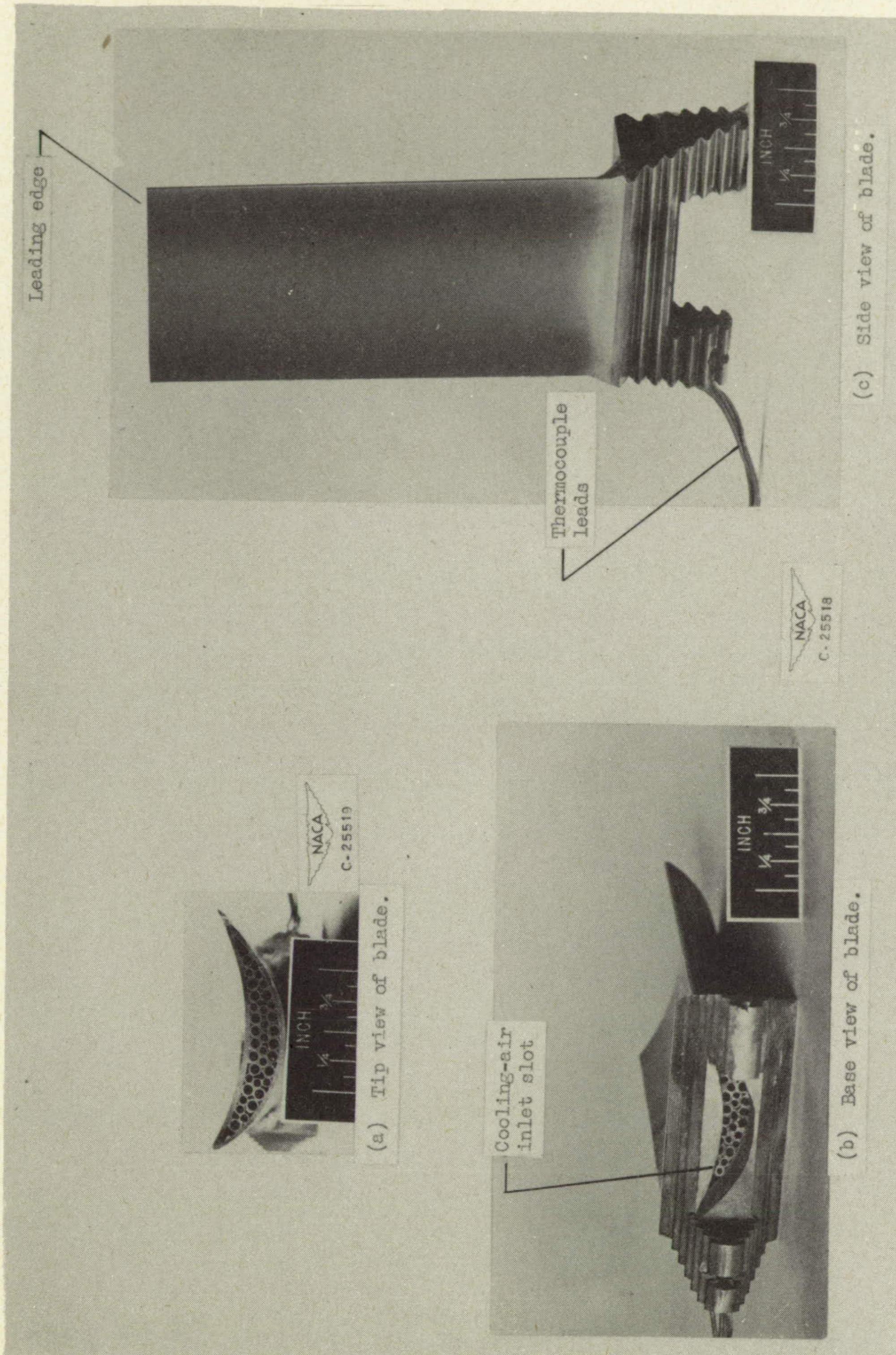
TABLE I - SUMMARY OF ENGINE OPERATING CONDITIONS

Series	Nominal engine speed (rpm)	Average compressor-inlet conditions		Thermocouple group	Combustion-gas flow, w_g (lb/sec)	Fuel flow, w_F (lb/sec)	Cooling-air flow per blade, w_a (lb/sec)	Cooling-air-inlet temperature $T'_{a,H}$ (°F)	Cooling-air temperature at blade root $T'_{a,S,h}$ (°F)	Calculated turbine-inlet total temperature $T_{g,S,i}$ (°F)
		Pressure $p'_{A,c,i}$ (in. Hg)	Temperature $T'_{A,c,i}$ (°F)							
1	4000	29.00	59 - 69	C,E,F,I,K,L	19.99 - 20.74	0.311 - 0.314	0.005 - 0.124	58 - 87	61 - 105	1056 - 1100
2	4000	28.90	52 - 63	G,H,I,J,K,F	20.11 - 20.67	0.286 - 0.317	0.004 - 0.123	58 - 82	60 - 95	1035 - 1071
3	4000	29.00	55 - 71	A,C,D,E,F	20.48 - 20.70	0.308 - 0.309	0.004 - 0.123	62 - 78	61 - 95	1036 - 1054
4	5400	29.14	55 - 56	A,C,D,E,F	28.53 - 28.71	0.402 - 0.405	0.005 - 0.122	56 - 75	70 - 101	1012 - 1034
5	5400	29.20	53 - 55	G,H,I,J,K,F	28.57 - 28.77	0.398 - 0.400	0.005 - 0.123	57 - 72	61 - 95	1015 - 1034
6	6000	29.34	38 - 40	A,C,D,E,F	32.60 - 33.70	0.435 - 0.439	0.005 - 0.124	54 - 69	62 - 93	992 - 1015
7	6000	29.30	46 - 47	G,H,I,J,K,F	32.48 - 33.13	0.439 - 0.443	0.005 - 0.124	54 - 71	60 - 95	1005 - 1031
8	8000	29.45	54 - 58	A,C,D,E,F	42.66 - 43.48	0.597 - 0.600	0.010 - 0.122	66 - 75	80 - 114	1099 - 1124
9	8000	29.56	50 - 52	G,H,I,J,K,F	42.99 - 43.61	0.600 - 0.602	0.010 - 0.124	57 - 73	72 - 106	1080 - 1099
10	9000	29.24	51 - 53	A,C,D,E,F	46.10 - 50.72	0.686 - 0.690	0.010 - 0.128	50 - 61	47 - 124	1150 - 1156
11	9000	29.52	39 - 75	G,H,I,J,K,F	49.84 - 50.78	0.688 - 0.688	0.010 - 0.129	52 - 75	74 - 138	1105 - 1215

TABLE II - COMPARISON OF GEOMETRY FACTORS FOR 34-TUBE,

10-TUBE, AND 15-FIN BLADES

Item	34-tube blade	10-tube blade	15-fin blade
Total free-flow area of cooling-air passages, square inch	0.14	0.22	0.24
Total inside-surface area exposed to cooling air, square inches	31.9	35.7	38.1
Total perimeter of cooling-air passages, inches	8.1	9.1	9.7
Hydraulic diameter, inch	0.07	0.10	0.10

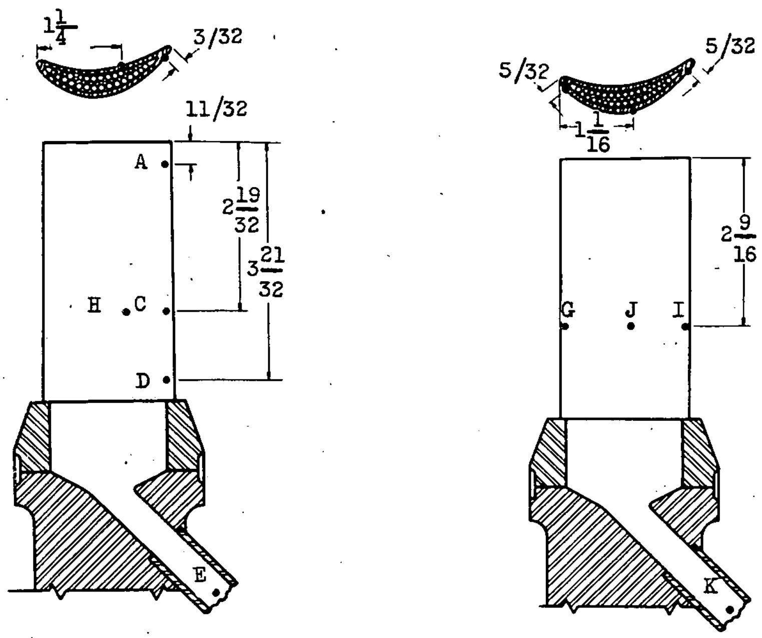


(a) Tip view of blade.

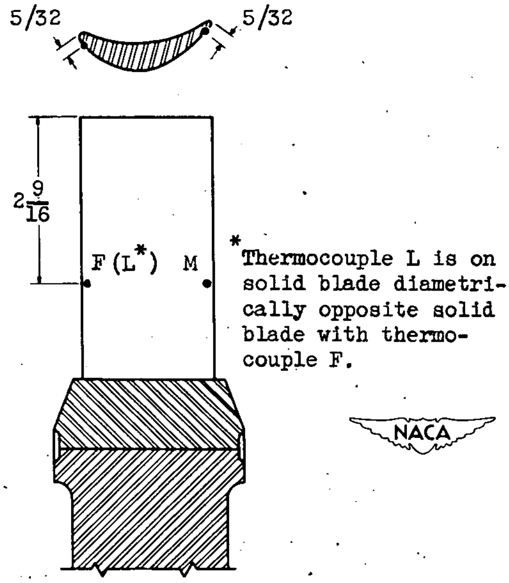
(b) Base view of blade.

(c) Side view of blade.

Figure 1. - Cooled turbine blade with 34 tubes.

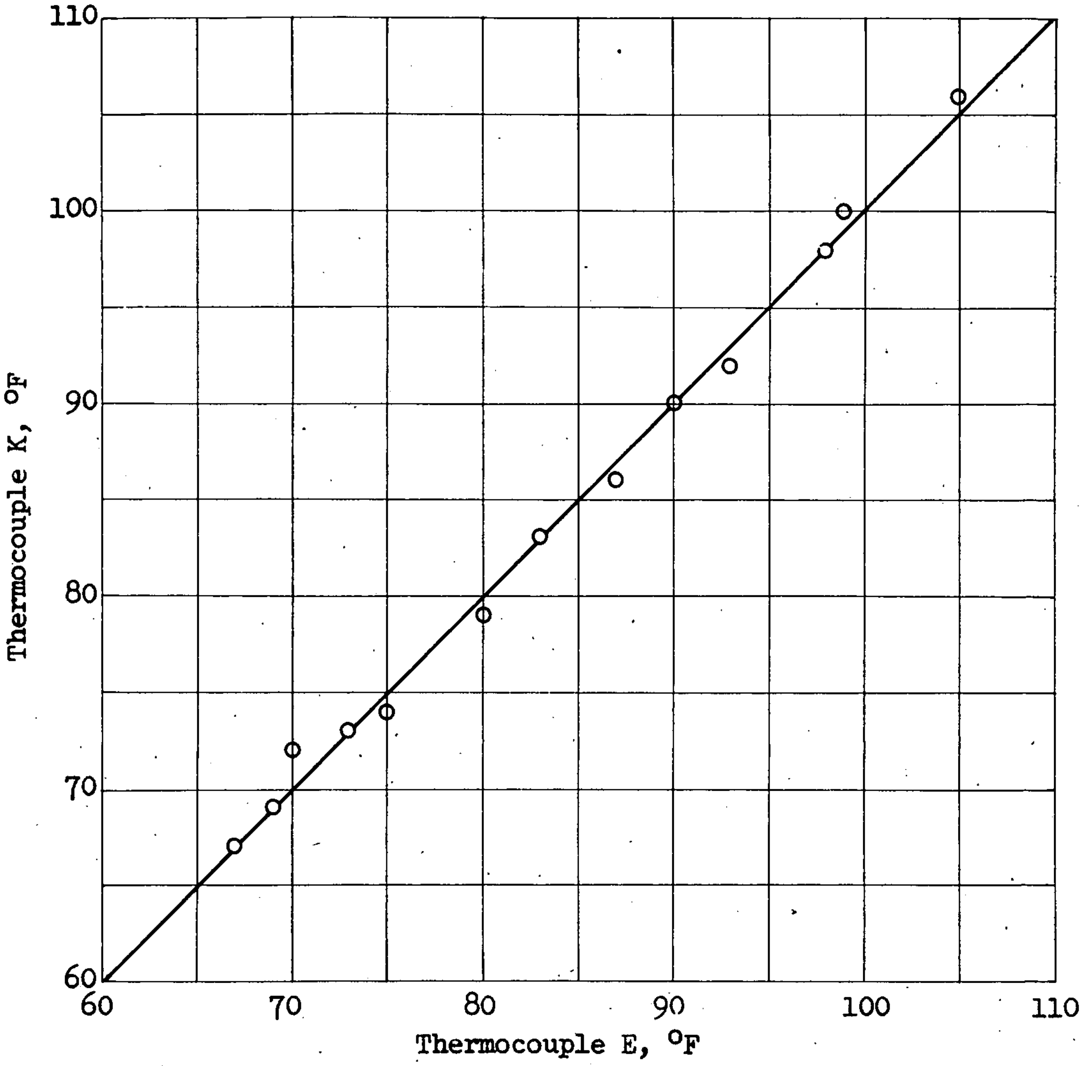


(a) Cooled blades.



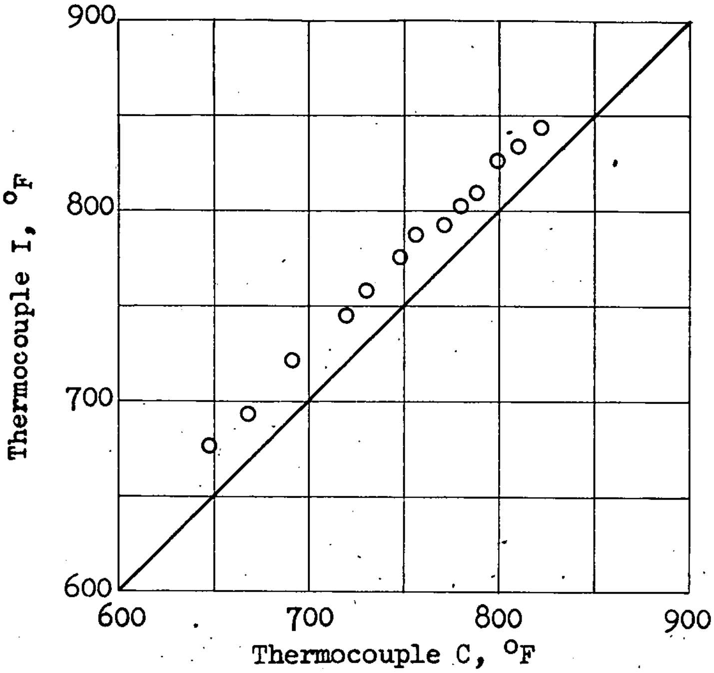
(b) Uncooled blade.

Figure 2. - Schematic diagram of thermocouple locations on cooled and uncooled blades. (All dimensions are in inches.)

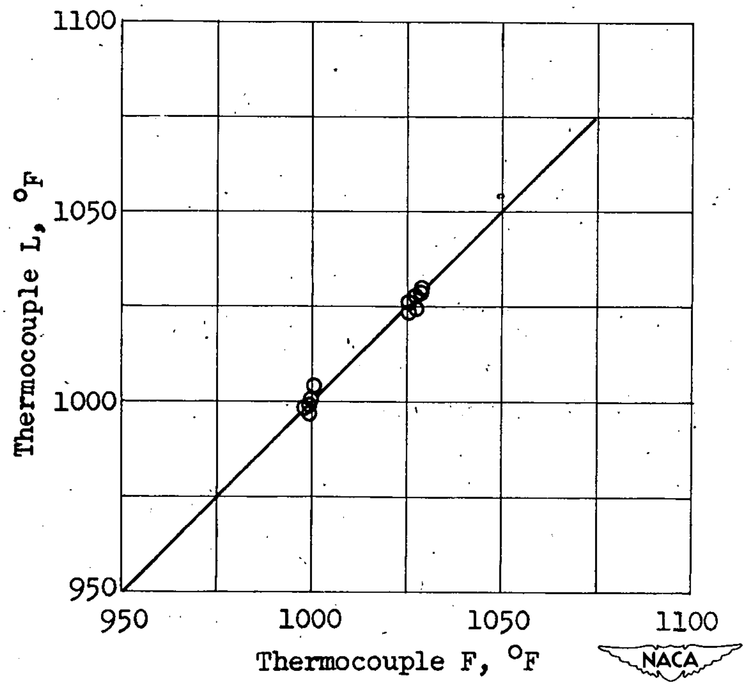


(a) Cooling-air temperature at blade root.

Figure 3. - Comparison of thermocouple readings at similar positions.

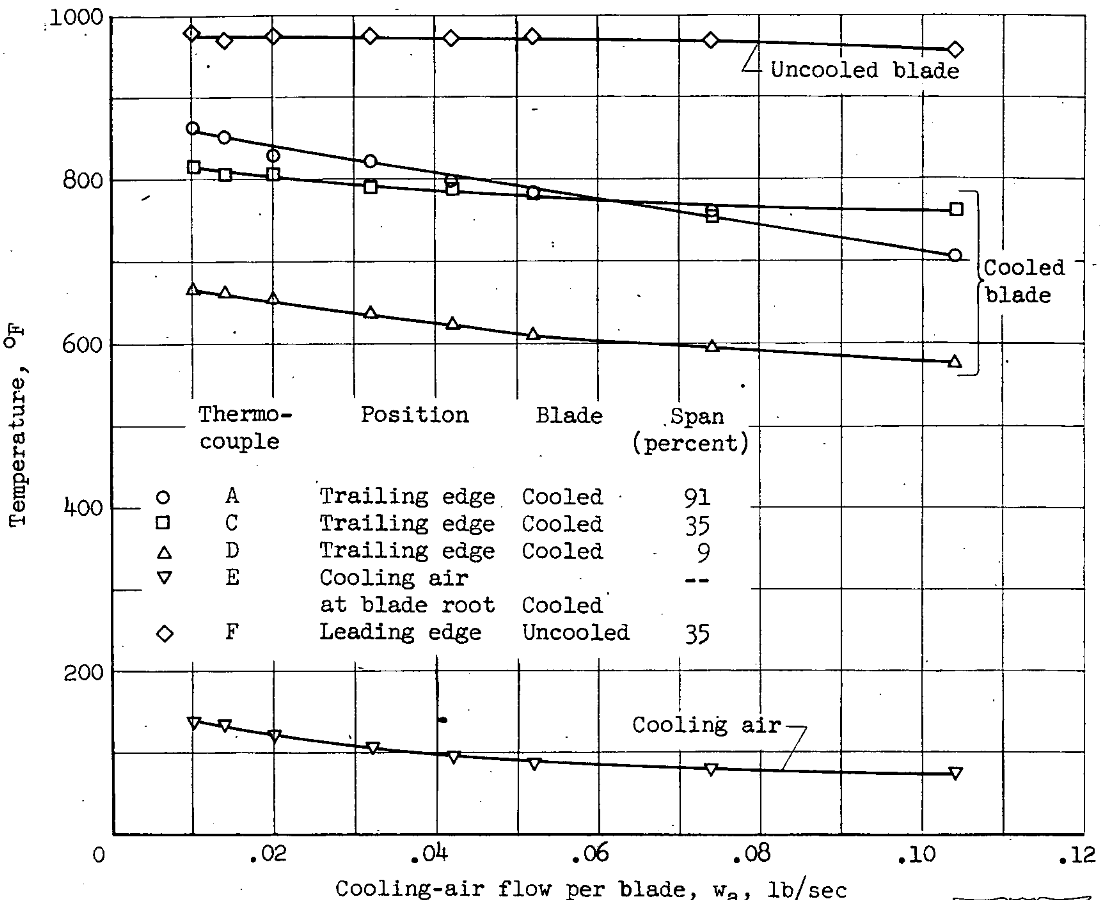


(b) Trailing-edge temperatures of cooled blade.



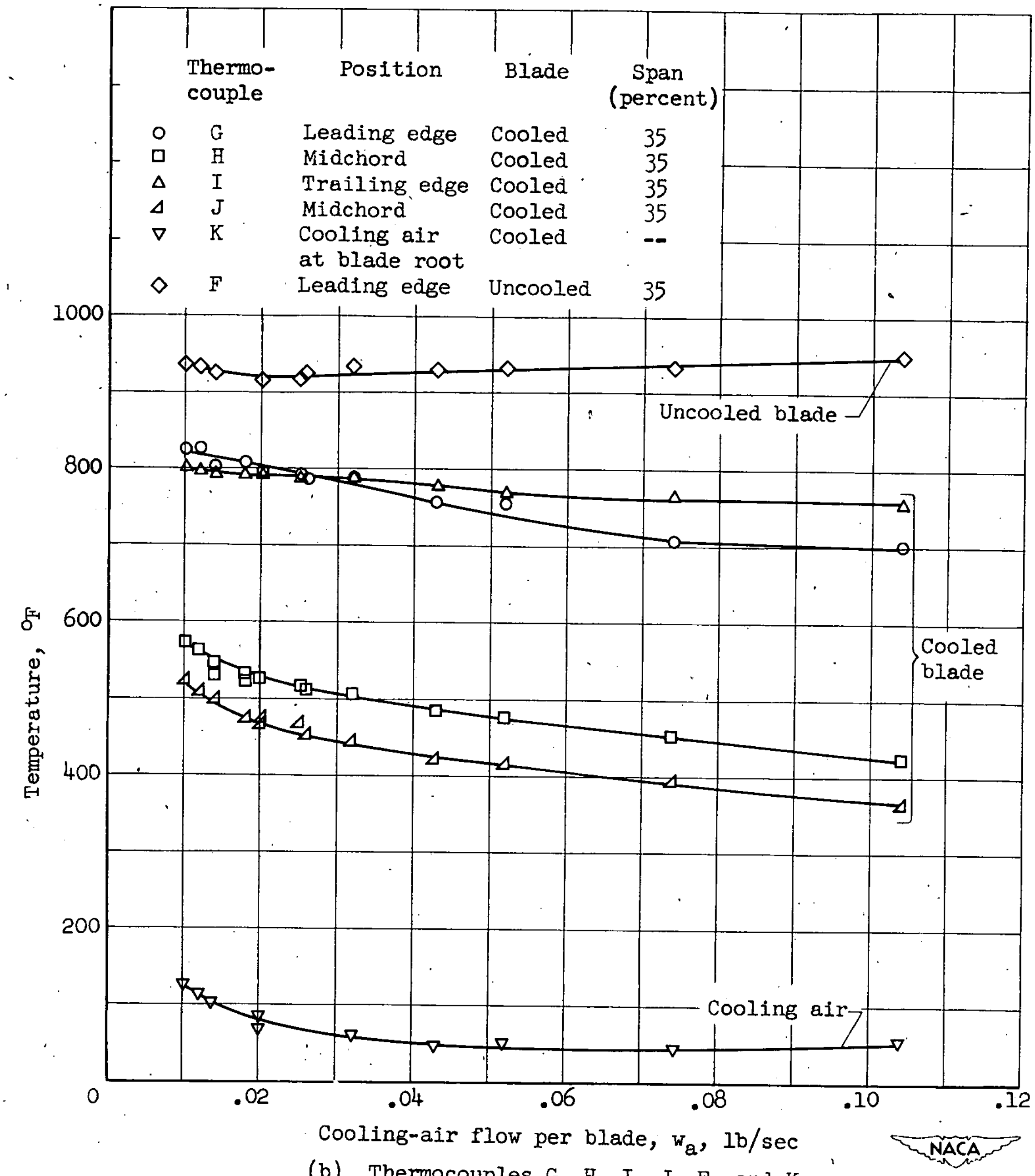
(c) Leading-edge temperatures of uncooled blade.

Figure 3. - Concluded. Comparison of thermocouple readings at similar positions.



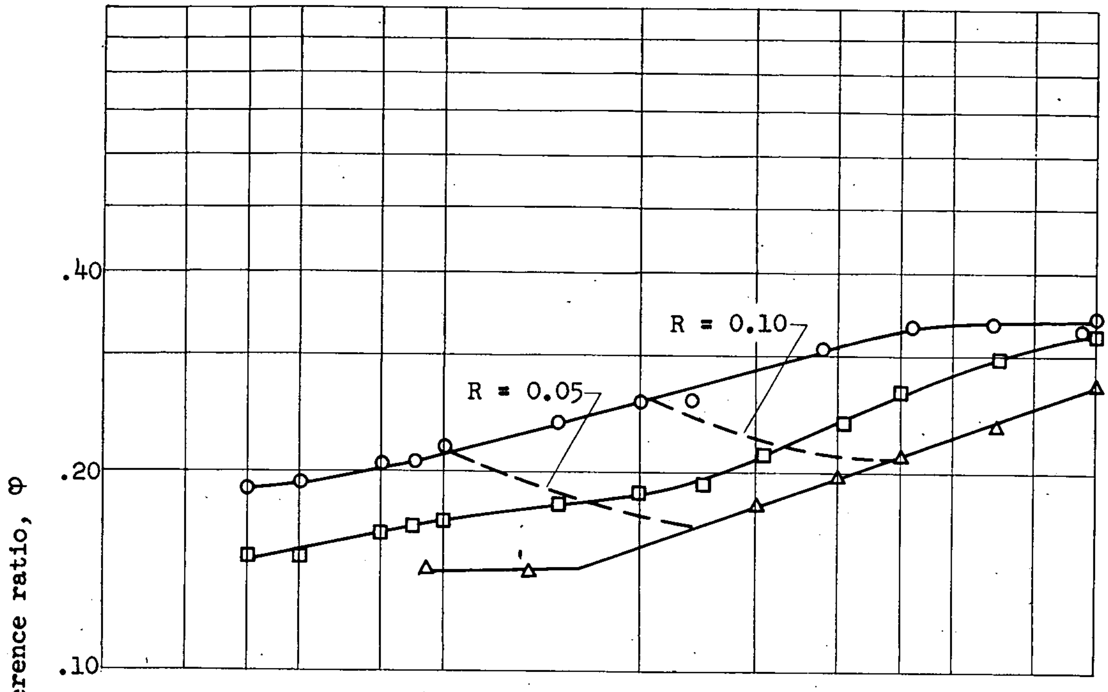
(a) Thermocouples A, C, D, E, and F.

Figure 4. - Effect of cooling-air flow on modified blades and cooling-air temperatures at engine speed of 9000 rpm.

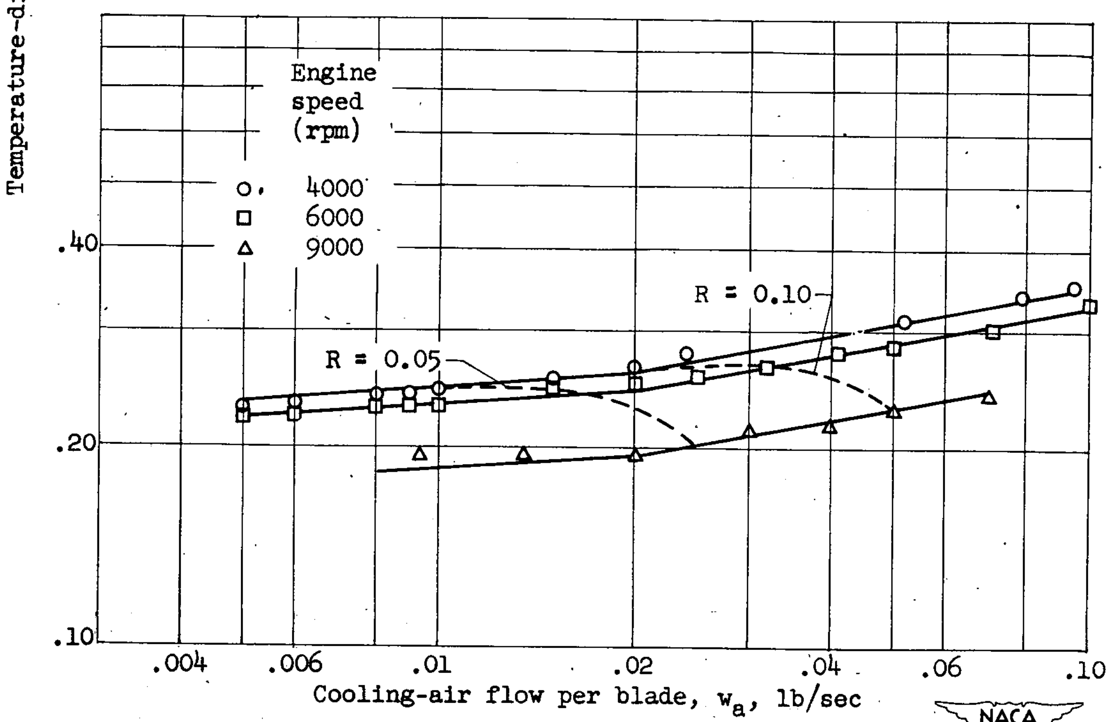


(b) Thermocouples G, H, I, J, F, and K.

Figure 4. - Concluded. Effect of cooling-air flow on modified blades and cooling-air temperatures at engine speed of 9000 rpm.

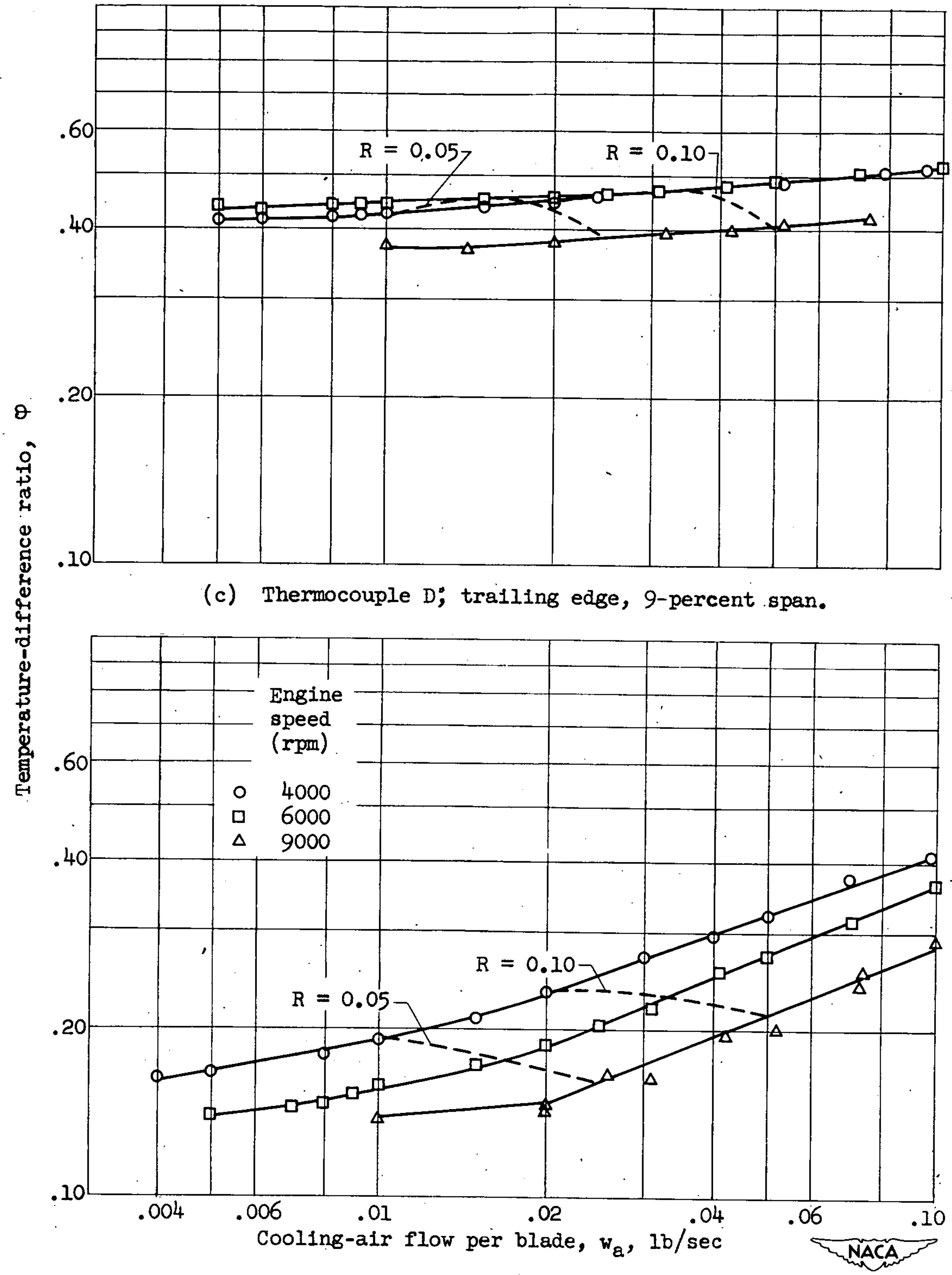


(a) Thermocouple A; trailing edge, 91-percent span.



(b) Thermocouple C; trailing edge, 35-percent span.

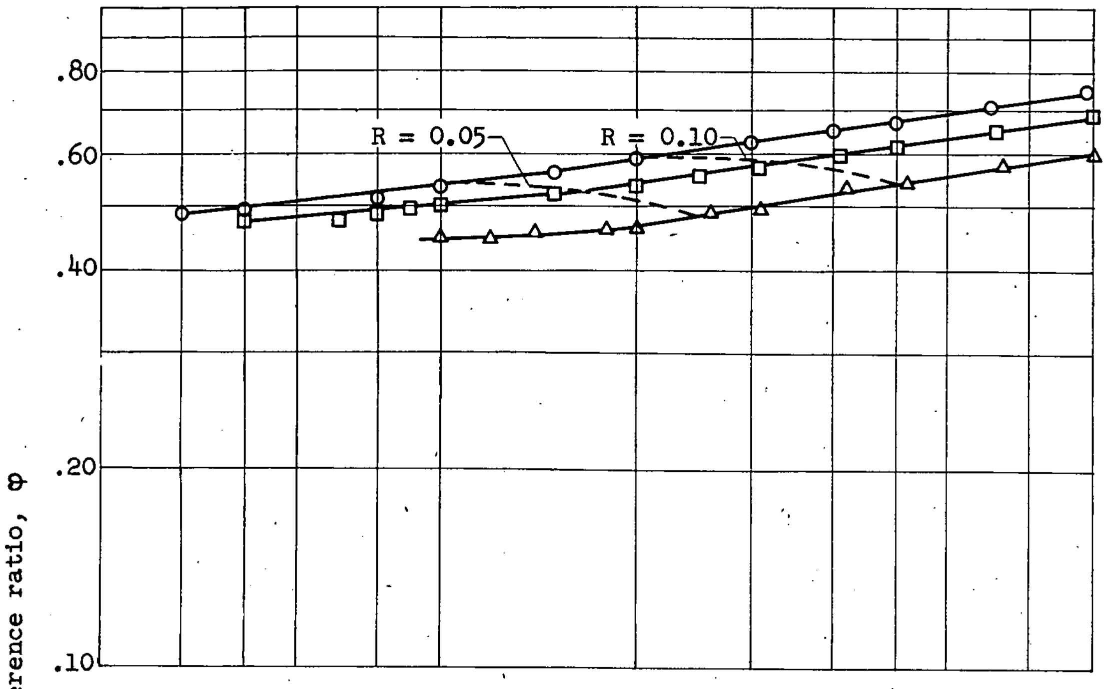
Figure 5. - Effect of cooling-air flow on temperature-difference ratio for several engine speeds.



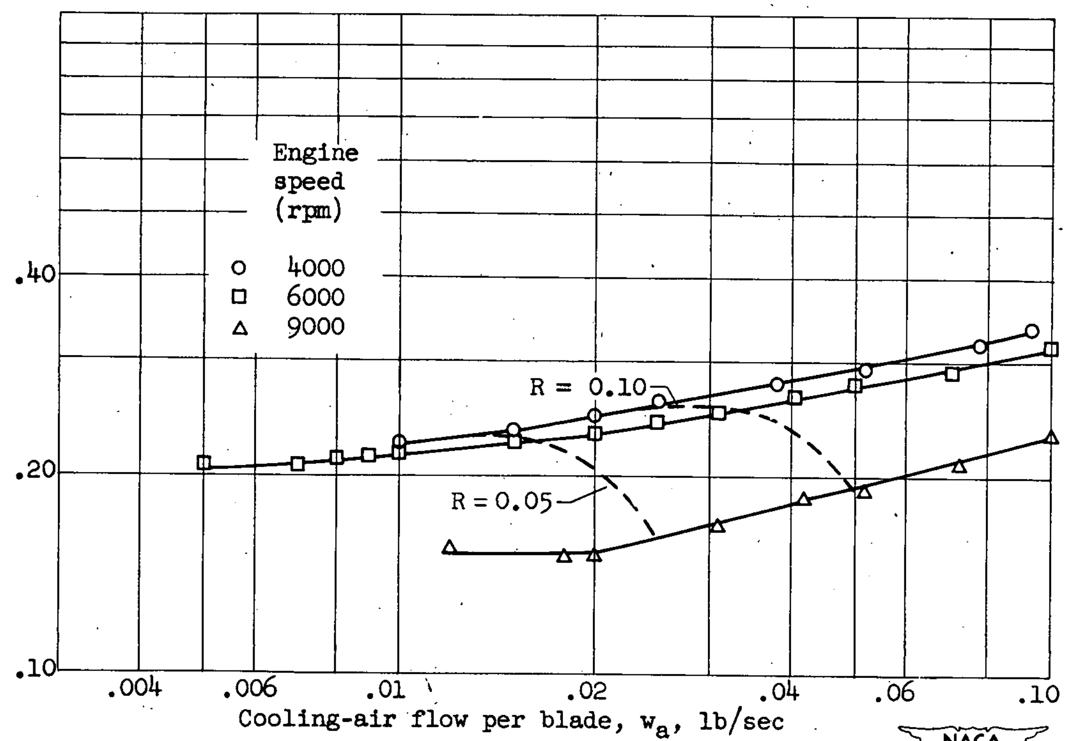
(c) Thermocouple D; trailing edge, 9-percent span.

(d) Thermocouple G; leading edge, 35-percent span.

Figure 5. - Continued. Effect of cooling-air flow on temperature-difference ratio for several engine speeds.

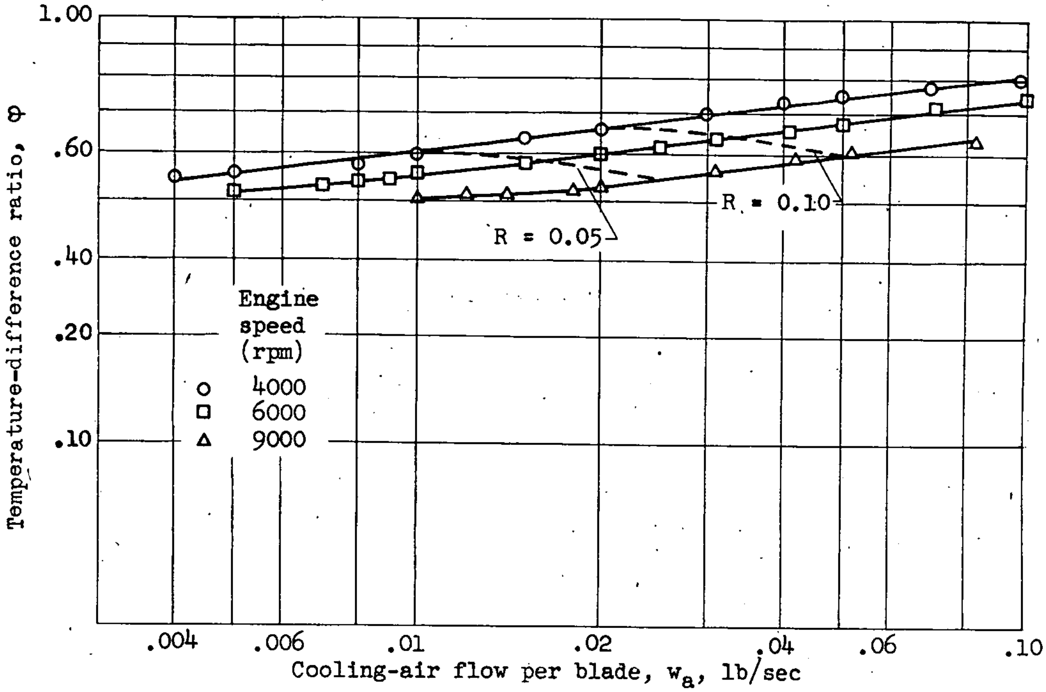


(e) Thermocouple H, midchord pressure surface, 35-percent span.



(f) Thermocouple I; trailing edge, 35-percent span.

Figure 5. - Continued. Effect of cooling-air flow on temperature-difference ratio for several engine speeds.



(g) Thermocouple J; midchord, suction surface, 35-percent span.

Figure 5. - Concluded. Effect of cooling-air flow on temperature-difference ratio for several engine speeds.

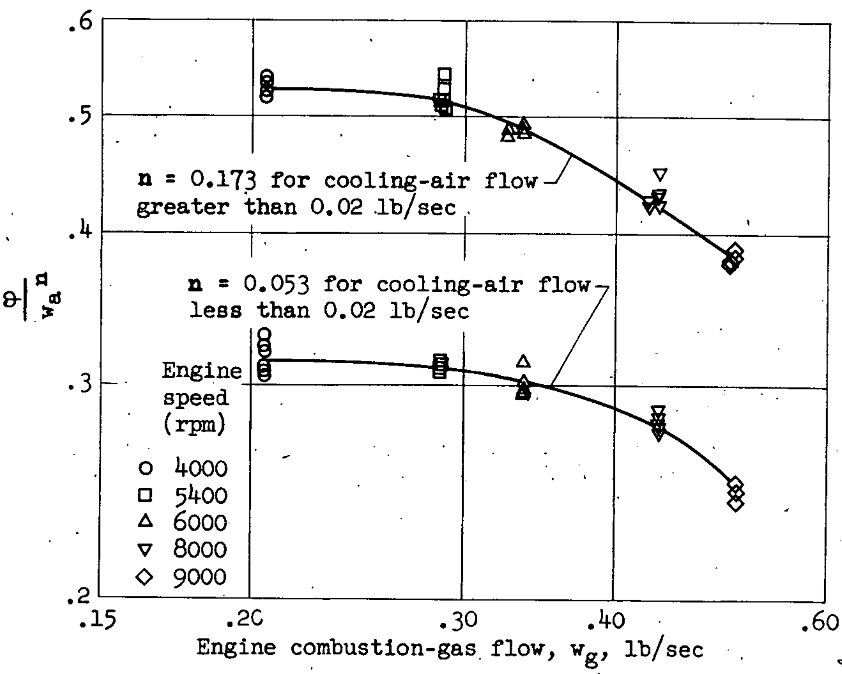


Figure 6. - Variation of $\varphi/w_a{}^n$ ratio with engine combustion-gas flow for thermocouple C; trailing edge, 35-percent span.

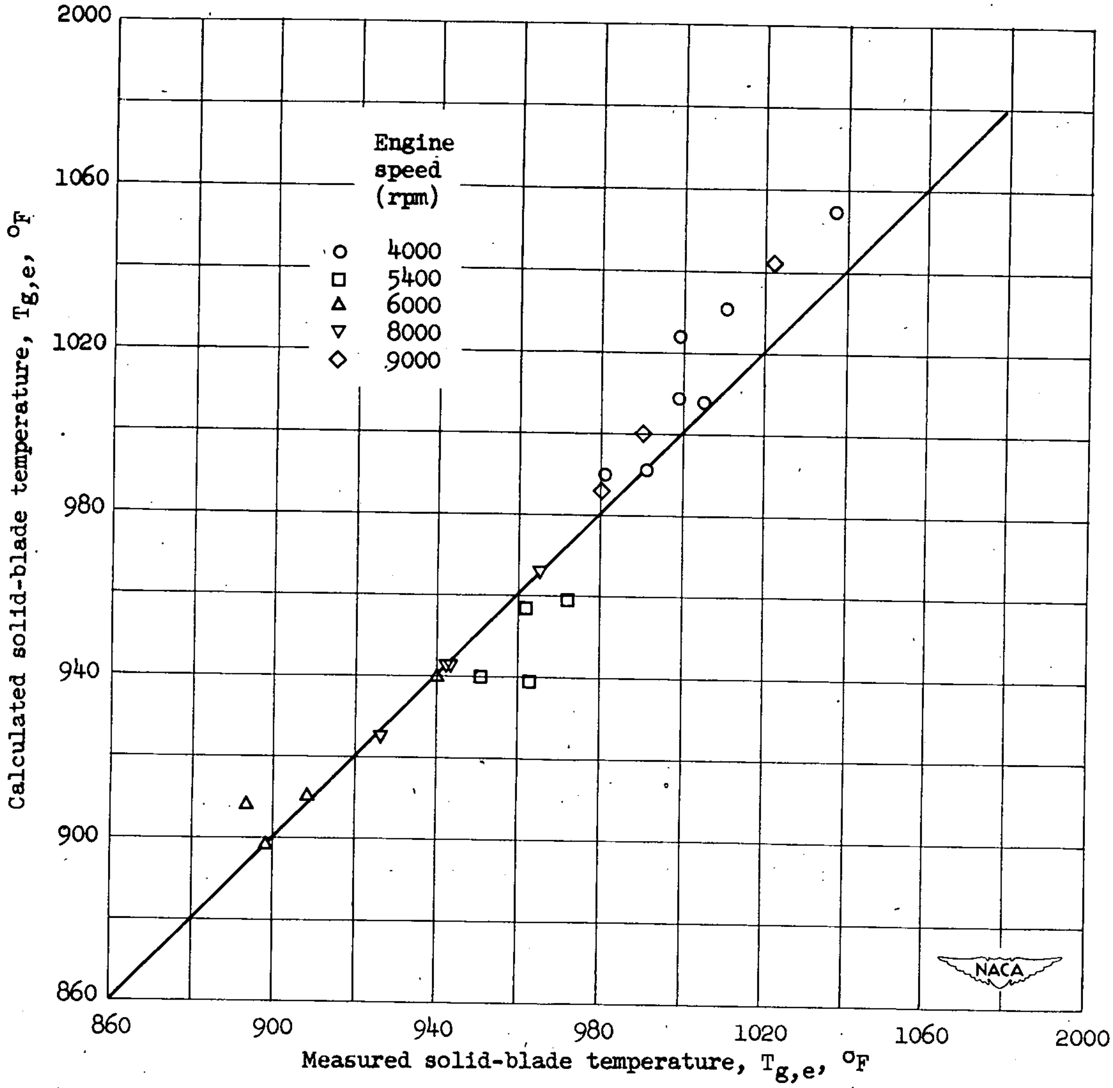


Figure 7. - Comparison of measured solid-blade temperatures with calculated solid-blade temperatures.

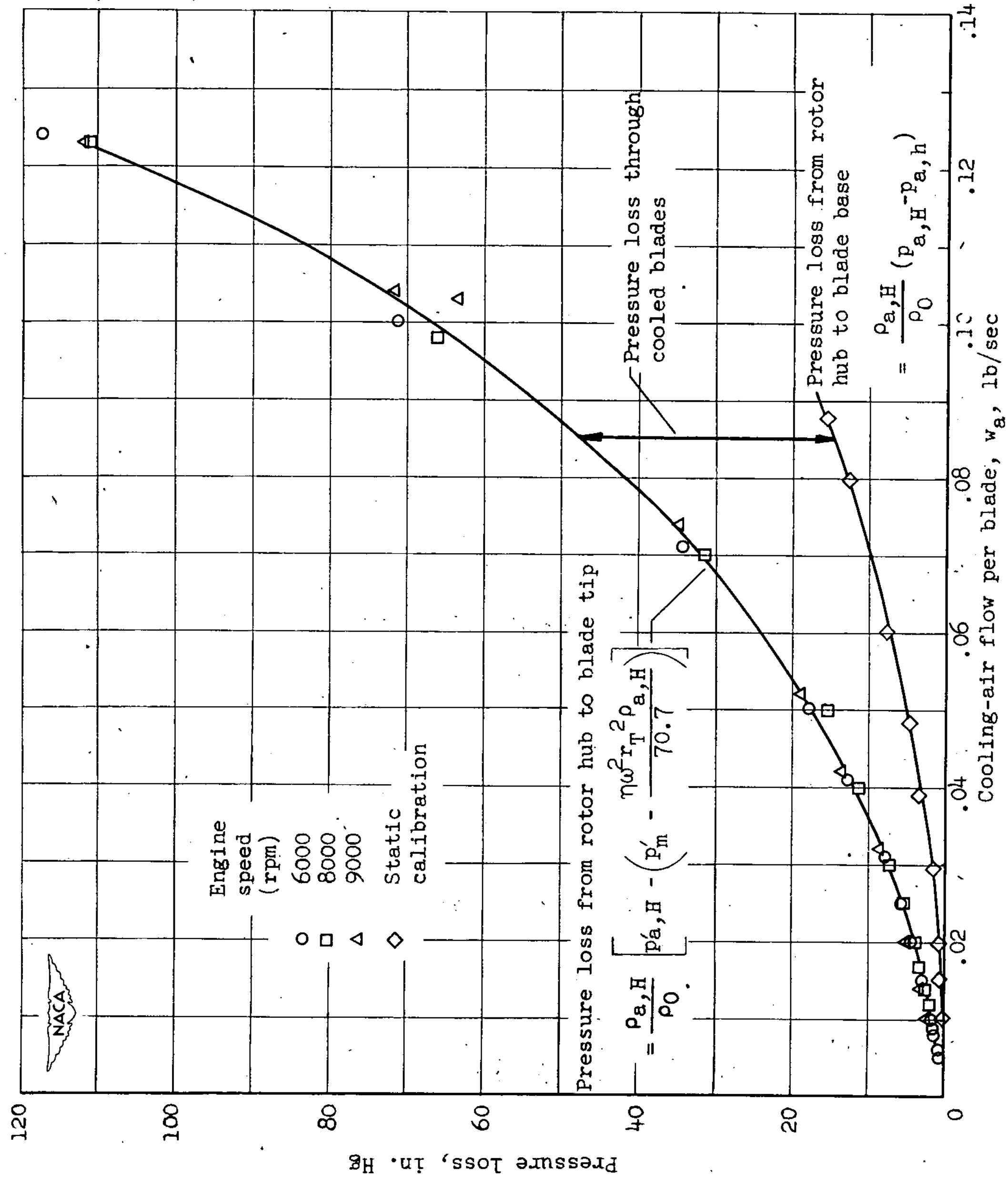


Figure 8. - Correlation of cooling-air pressure loss.

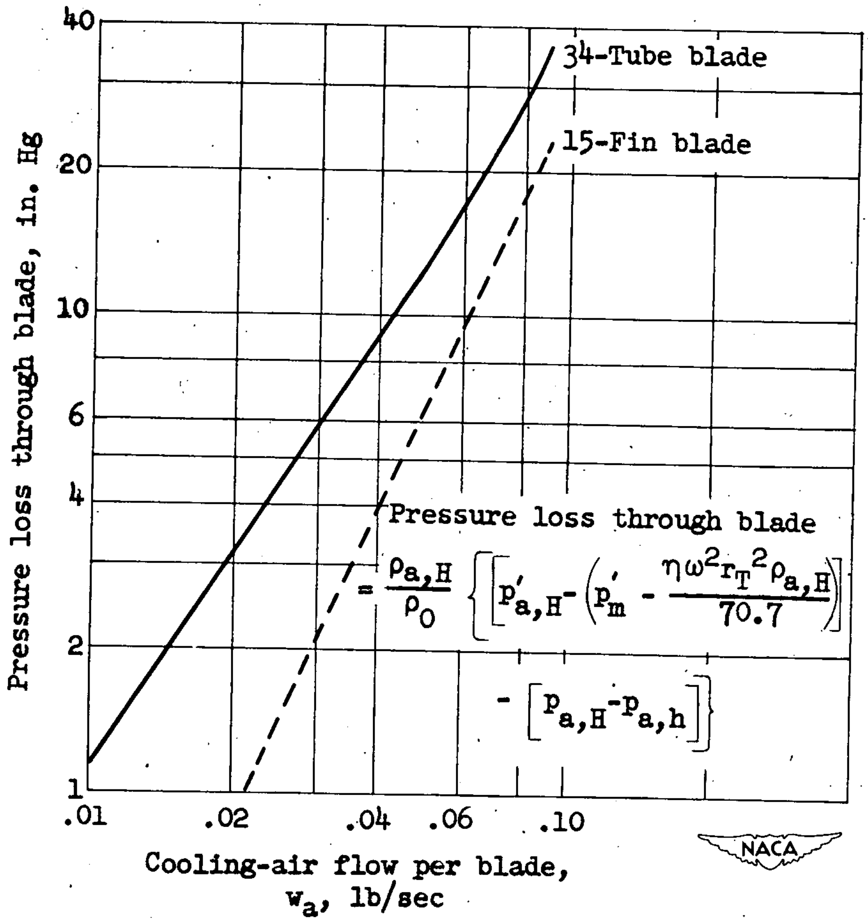
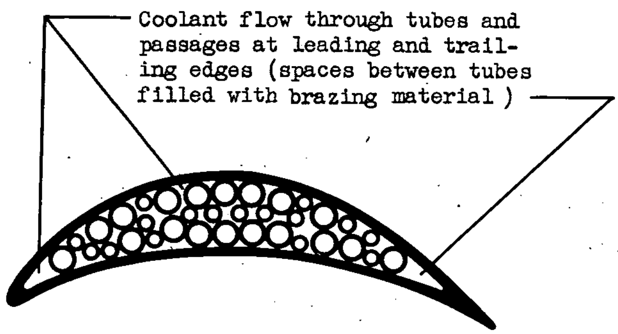
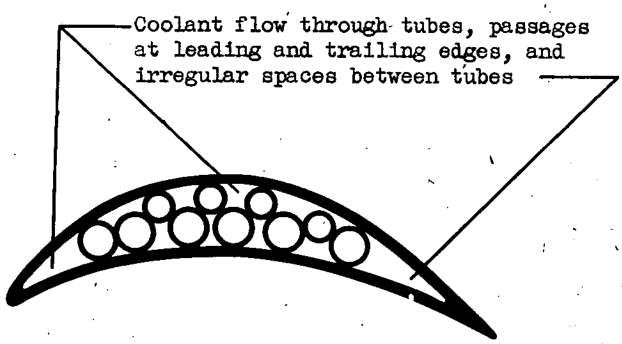


Figure 9. - Comparison of pressure loss through 34-tube and 15-fin blades.



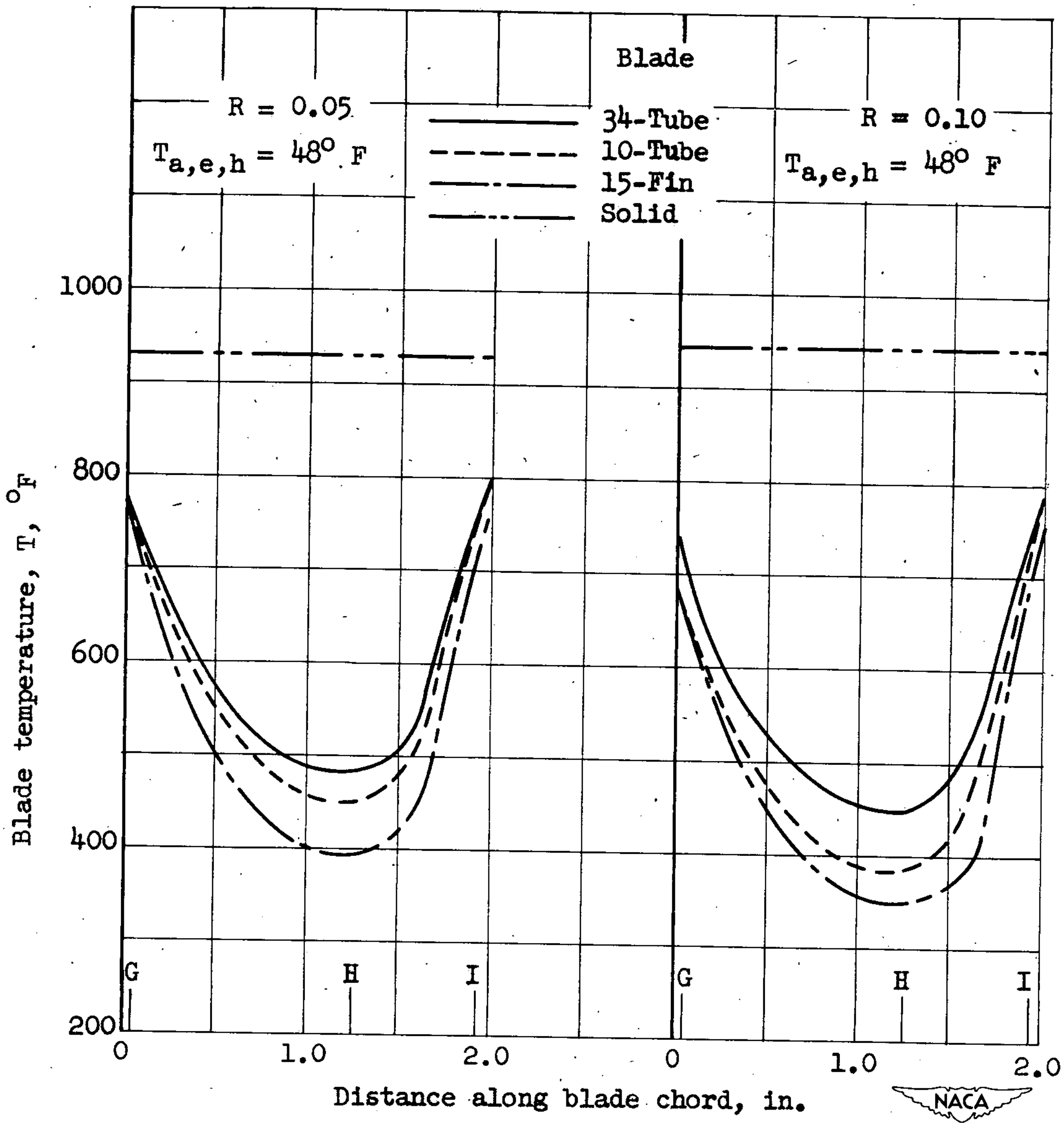
(a) 34-Tube blade.



(b) 10-Tube blade.

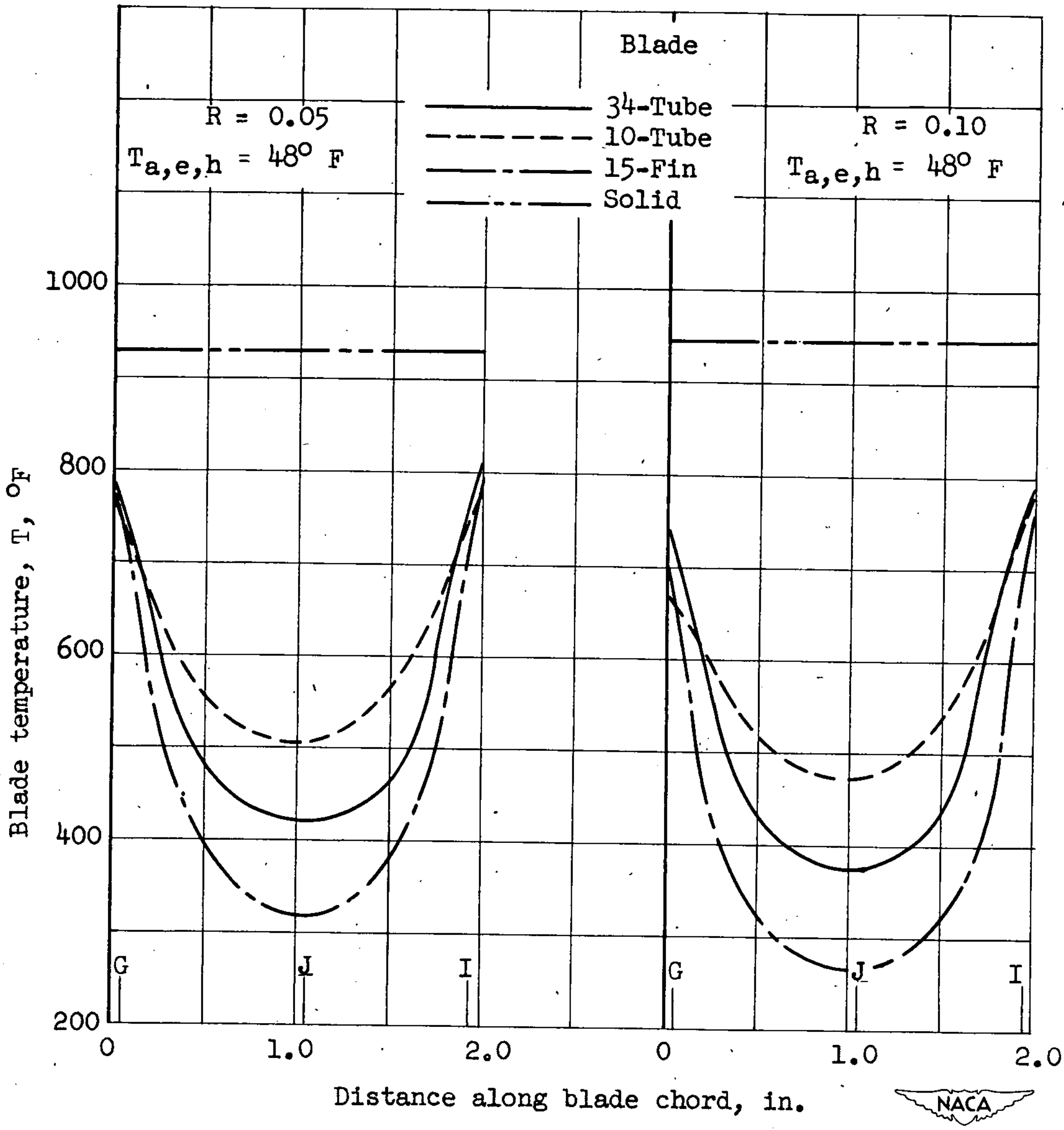
(c) 15-Fin blade.

Figure 10. - Comparison of air-cooled blades investigated.



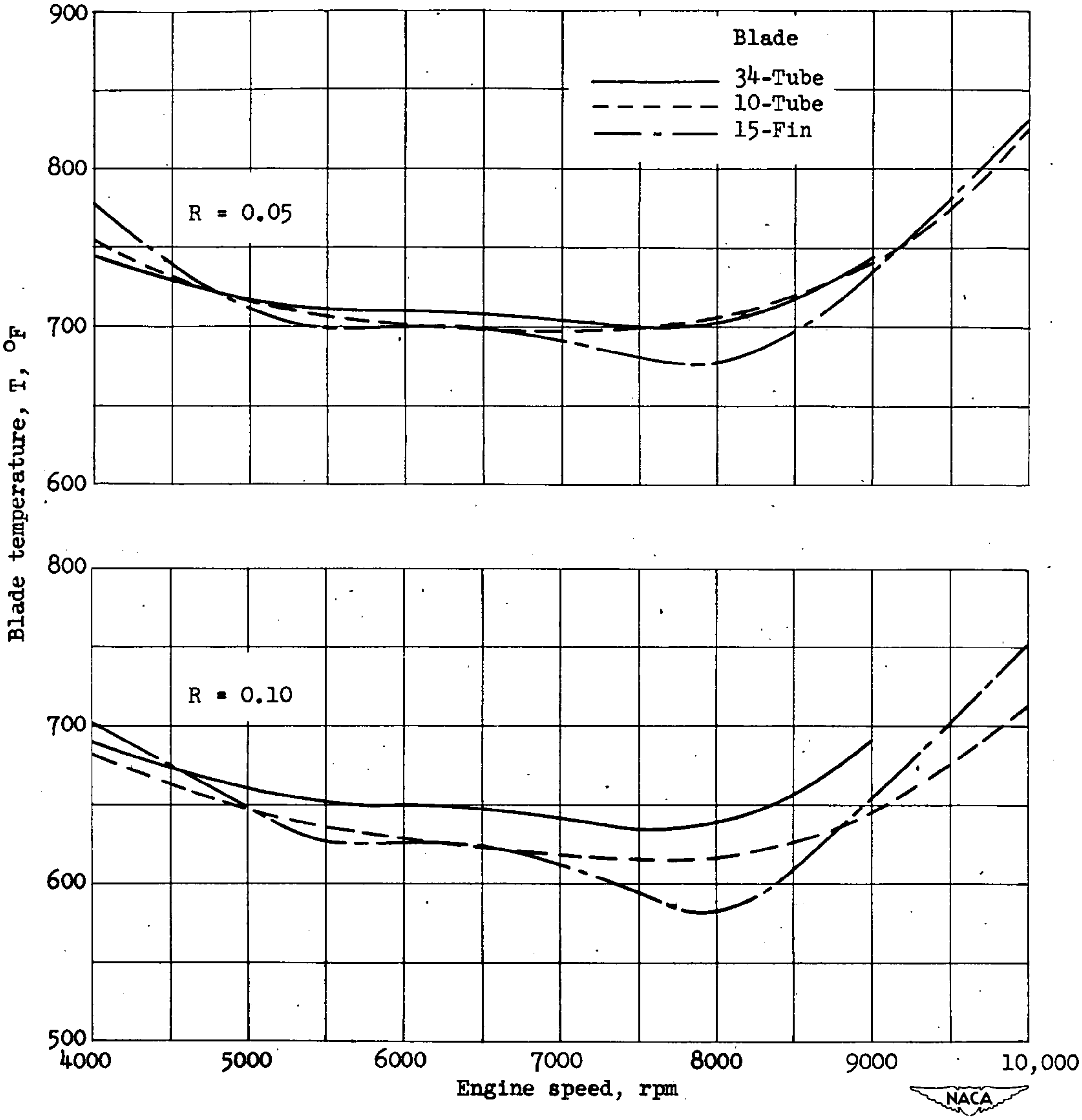
(a) Thermocouples G, H, and I; 35-percent span.

Figure 11. - Comparison of temperatures along blade chord for 34-tube, 10-tube, and 15-fin blades for several ratios of cooling-air flow to engine gas flow at an engine speed of 9000 rpm.



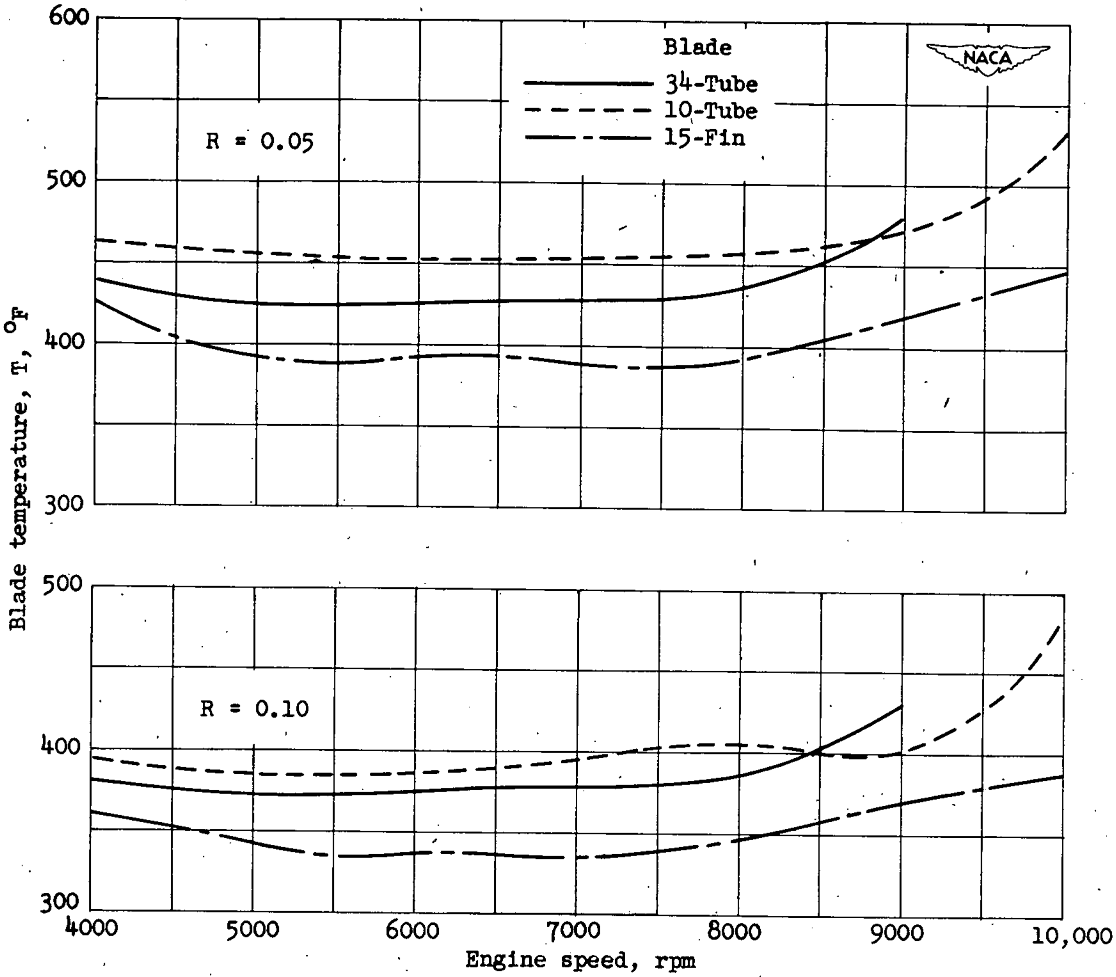
(b) Thermocouples G, J, and I; 35-percent span.

Figure 11. - Concluded. Comparison of temperatures along blade chord for 34-tube, 10-tube, and 15-fin blades for several ratios of cooling-air flow to engine gas flow at an engine speed of 9000 rpm.



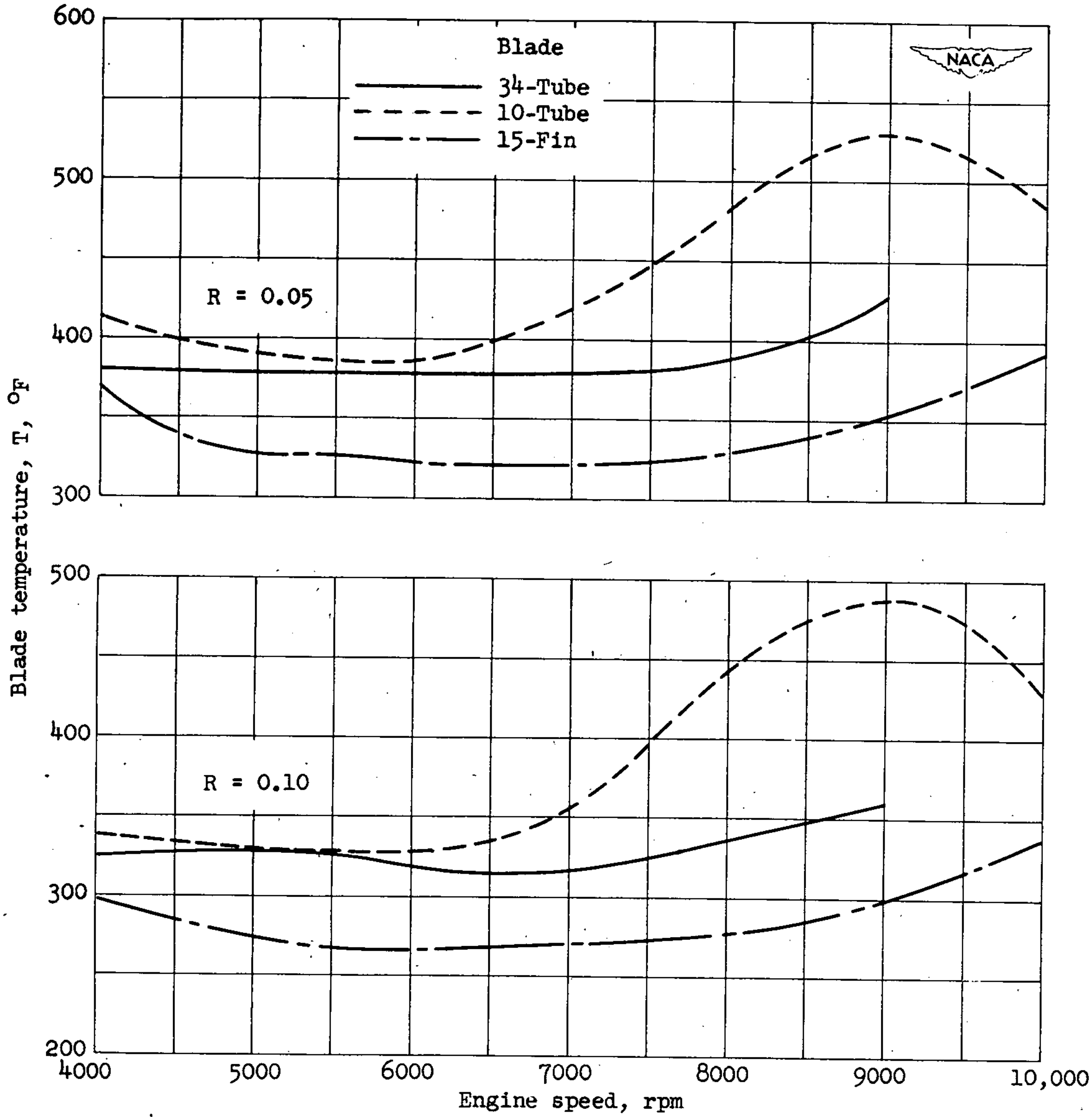
(a) Thermocouple G; leading edge, 35-percent span.

Figure 12. - Comparison of blade temperature of 34-tube, 10-tube, and 15-fin blades over a range of engine speeds for standard engine-inlet conditions.



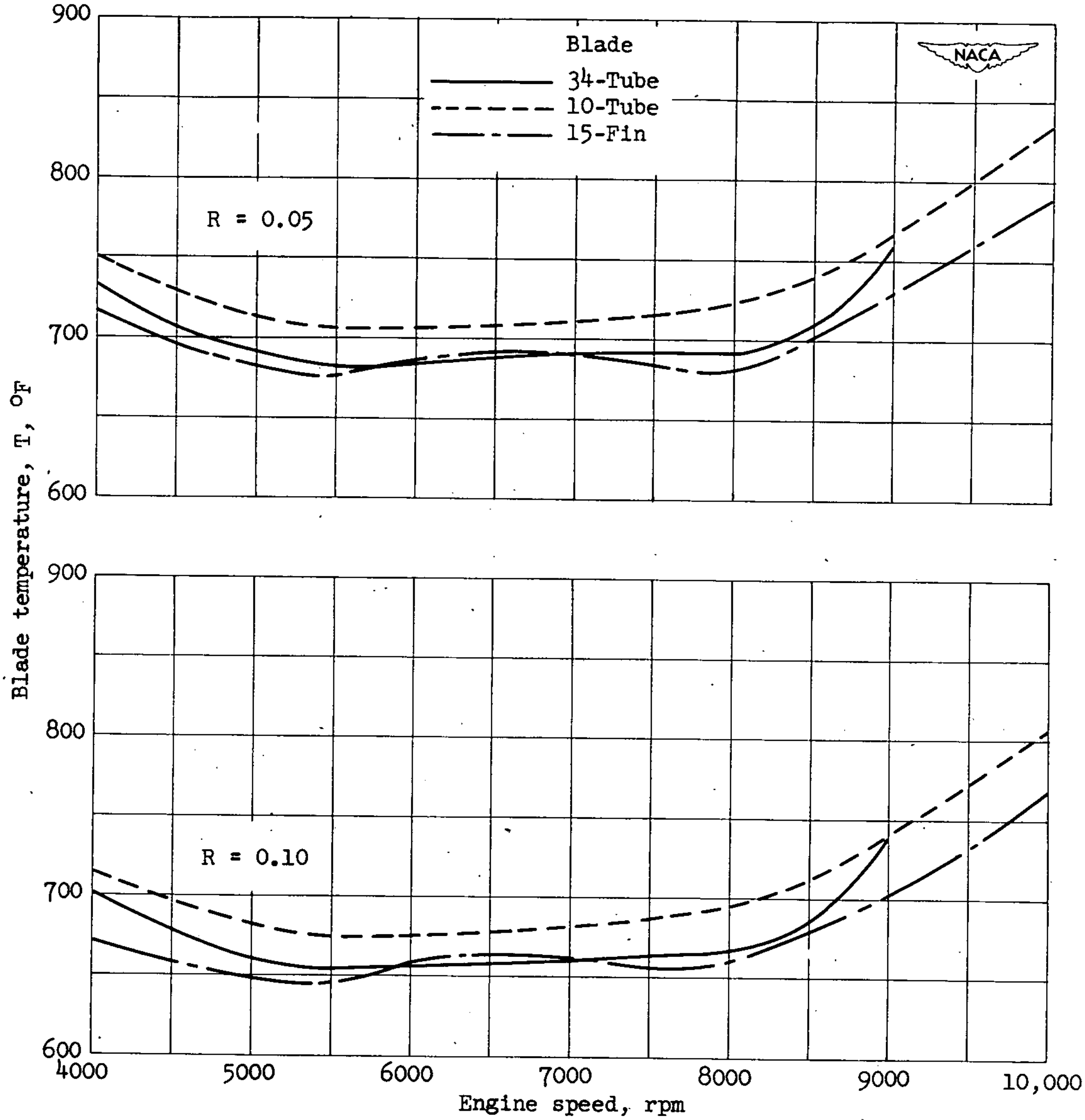
(b) Thermocouple H; midchord, pressure surface, 35-percent span.

Figure 12. - Continued. Comparison of blade temperature of 34-tube, 10-tube, and 15-fin blades over a range of engine speeds for standard engine-inlet conditions.



(c) Thermocouple J; midchord, suction surface, 35-percent span.

Figure 12. - Continued. Comparison of blade temperature of 34-tube, 10-tube, and 15-fin blades over a range of engine speeds for standard engine-inlet conditions.



(d) Thermocouple I; trailing edge, 35-percent span.

Figure 12. - Concluded. Comparison of blade temperature of 34-tube, 10-tube, and 15-fin blades over a range of engine speeds for standard engine-inlet conditions.