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EXPERIMENTAL INVESTIGATION OF AIR-COOLED TURBINE BLADES IN TURBOJET ENGINE

V - ROTOR BLADES WITH SPLIT TRAILING EDGES

By Gordon T. Smith and Robert O. Hickel

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Cleveland, Ohio

NATIONAL ADVISORY COMMITTEE
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SUMMARY

An air-cooled blade configuration that has special provisions for cooling the trailing edge by passing cooling air through a series of radial slots was investigated under conditions of actual engine operation. The results of this investigation are presented and compared with the results of other blade configurations previously investigated.

The cooling effectiveness was determined over a range of engine speeds from 4000 to 10,000 rpm; the cooling-air flow per blade was varied from about 0.01 to about 0.10 pound per second. The results indicated that the trailing edge of the blade was effectively cooled. For example, at an engine speed of 10,000 rpm, an effective gas temperature of 1145° F, a cooling-air temperature of 138° F, and a coolant-to-combustion-gas-flow ratio of about 0.063, the trailing-edge temperature at the midspan position was 755° F, which was only about 30° F hotter than the best blade configuration thus far investigated. The cooling-air pressure loss, based on coolant flow rate, was found to be about the same as that for the best blades previously investigated.

INTRODUCTION

A general program of investigation of air-cooled turbine-blade configurations under conditions of actual engine operation is being conducted at the NACA Lewis laboratory in order to obtain a cooled blade that will permit engine operation at current turbine-inlet temperatures or higher while using blades fabricated from nonstrategic materials. A summary of the preliminary analytical and experimental research that preceded this program is presented in reference 1.

The results obtained with one of the blade configurations investigated are presented herein; these results are compared with results obtained with the two most effective configurations previously investigated. The first three blade configurations, investigated under conditions of actual engine operation, consisted of untwisted blade shells with a variety of tubes and fins placed in the hollow blade to improve the cooling effectiveness by increasing the heat-transfer surface area exposed to the cooling air. The results of these investigations are presented in references 1 to 3. In general, the results show that the blades were effectively cooled at the midchord positions but that the leading and trailing edges were considerably hotter (as much as 500° F) causing large chordwise temperature gradients. A series of cooled-blade modifications was subsequently devised for the purpose of reducing these chordwise temperature gradients by special methods of cooling the leading- and trailing-edge areas. The results of an investigation of six special configurations intended to reduce these chordwise temperature gradients are presented in reference 4, where they are designated as blades 4, 5, 6, 7, 8, and 9. The results obtained with the seventh configuration, which has a special method for cooling the trailing edge are presented herein; this configuration is hereinafter designated blade 10. Blade 10 was specifically investigated to determine the cooling effectiveness of a split trailing-edge configuration and no special attempt was made to provide leading-edge cooling modifications. The results are compared with the results of blades 8 and 9 (the two coolest blades of reference 4).

The cooled-blade temperature distribution and the cooling-air pressure losses of blade 10 were investigated over a range of constant engine speeds from 4000 to 10,000 rpm and over a range of cooling-air flow per blade from about 0.01 to about 0.10 pound per second. The turbine-inlet temperature varied from about 960° F at 4000 rpm to about 1170° F at 10,000 rpm.

SYMBOLS

The following symbols are used in this report:

N engine speed, (rpm)

p static pressure, (in. Hg abs.)

p' total pressure, (in. Hg abs.)

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

r radius, (ft)

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

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

w weight flow rate, (lb/sec)

η efficiency of cooling-air compression in turbine wheel

ρ density, (slugs/cu ft)

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

ω angular velocity of rotor, (radians/sec)

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

T blade tip

0 NACA standard sea-level conditions

APPARATUS AND INSTRUMENTATION

Engine

The modified turbojet engine and the instrumentation used for this investigation are described in detail in references 1 and 2 with the exception that the two cooled blades installed diametrically opposite one another in the turbine were of a different configuration.

Cooled Blades

The cooled-blade configuration investigated is shown in figure 1. The blade shell has an outside geometry identical with that of blades 4, 5, 7, 8, and 9 of reference 4 except that a radial slot was ground through the trailing edge of the blade. This slot was 0.010 inch wide and extended from a point about 1/4 inch from the blade base to a point about 1/4 inch from the tip. In order to reduce vibratory stresses in the trailing-edge section of the blade and also to maintain the slot width, four struts were spot-welded in the radial slot. These struts were placed about 5/8 inch apart and had a radial length of 1/8 inch. Four 0.156-inch outside diameter, three 0.125-inch outside diameter, and one 0.190-inch outside diameter steel tubes were brazed in the shell interior. (See fig. 1.) The passage formed by the exterior of the rearmost tube and the interior of the blade shell near the trailing edge was capped at the tip, thus forcing the cooling air to flow axially through the five trailing-edge slots and thereby to cool the trailing-edge section of the blade by forced convection. The blade shell was welded to a conventional blade base that had been modified to permit the introduction of the blade coolant, as described in reference 1.

The location of the blade-thermocouple instrumentation is schematically shown in figure 2. The spanwise cooled-blade temperatures near the trailing edge were obtained from thermocouples A, B, C, and D located on one of the two cooled blades. The chordwise temperature distribution at about the one-third span position was obtained from thermocouple C and from thermocouples G, H, and J, which were located on the cooled blade that was diametrically opposite the blade with the spanwise instrumentation. The blade-metal temperature midway between two trailing-edge struts, which formed a trailing-edge coolant passage on the cooled blades, was obtained with thermocouples B and I. The effective gas temperature (uncooled-blade temperature) was obtained from thermocouples F and L, which were located near the leading edge at about the one-third span position on the uncooled blades that were adjacent to each of the cooled blades. Thermocouples E and K provided a measurement of the cooling-air temperature at the base of the cooled blades.

Measurements of the total pressure in the cooling-air passage at the rotor hub and in the combustion gas stream downstream of the turbine were used to evaluate the pressure loss through the coolant system from the rotor hub to the blade tip.

PROCEDURE

Experimental Procedure

The cooling effectiveness of the blade 10 configuration was evaluated over a range of constant engine speeds from 4000 to 10,000 rpm in increments of 1000 rpm. The combustion-gas flow varied from about 19 pounds per second at an engine speed of 4000 rpm to about 60 pounds per second at 10,000 rpm. The cooling-air flow per blade was varied from about 0.01 to 0.10 pound per second at each engine speed. Two sets of runs were conducted at each engine speed with a separate grouping of six rotating thermocouples connected for each run. Only six thermocouples were used because of limitations in the rotating thermocouple pickup, which are discussed in reference 1.

Calculation Procedure

The general calculation procedures for this series of cooled turbine-blade investigations are discussed in reference 1. The calculation procedure required for adjusting cooled-blade temperatures to engine conditions other than those at which they were measured is contained in reference 2. The method for correlating the cooling-air pressure loss through the turbine from the rotor hub to the blade root is developed in reference 2, and the method of evaluating the cooling-air pressure drop through only the blade is discussed in reference 3.

RESULTS AND DISCUSSION

The temperatures of blade 10 were correlated by use of a temperature-difference ratio $(T_{g,e}-T_B)/(T_{g,e}-T_{a,e,h})$, which is designated φ or cooling effectiveness. The effective gas temperature $T_{g,e}$ was measured by thermocouples F and L, the cooling-air inlet temperatures $T_{a,e,h}$ were measured by thermocouples E and K, and the cooled blade temperatures T_B were measured by the previously described instrumentation on the cooled blades. The development of φ is discussed in detail in reference 1. Values of φ and a summary of several pertinent engine operating conditions for low, medium, and high engine speeds of 4000, 7000, and 10,000 rpm, respectively, are presented in table I. The values of φ in table I may be plotted for each engine speed and range of coolant flow for extrapolation or interpolation purposes, as is shown in references 1 to 3. Comparisons are presented for the chordwise and spanwise temperature distributions of blade 10 with blades 8 and 9. Blade 10 is compared to blades 8 and 9 because these blades had the most successfully cooled trailing-edge sections thus far investigated. The data from which the temperatures of blades 8 and 9 were calculated is contained in reference 4.

Chordwise Blade-Temperature Comparisons

A comparison of the chordwise temperature distribution of blade 10 with that of blades 8 and 9 at an engine speed of 10,000 rpm, a coolant-to-combustion-gas-flow ratio of 0.063, a cooling-air temperature at the blade root of 138° F, and an effective gas temperature of about 1145° F is presented in figure 3. A schematic diagram of blades 8, 9, and 10 is shown in figure 4 so that the pertinent design features of each blade can be compared. The trailing-edge temperature of blade 10 was about 755° F or 35° F cooler than blade 9 and 30° F hotter than blade 8 (fig. 3). The midchord suction surface of all three blades was well cooled; the maximum difference in temperature between blades at this point was about 50° F. The midchord pressure surface of blade 10 was considerably hotter than blades 8 and 9; however, this excessive temperature was caused by a fabrication flaw that resulted in an internal tube, which was located at the position of thermocouple H, not being brazed to the blade shell. Because of this flaw, the measured temperature of the midchord pressure surface of blade 10 was not regarded as a true representative temperature for this configuration and consequently the curve indicating the pressure surface profile was omitted. The measured temperature values are indicated, however.

The difference between the midchord temperature on the suction surface and the temperature at the trailing edge for blade 10 was about 200° F. This gradient was comparable to the temperature gradients for blades 8 and 9, which were the lowest yet achieved with the blade configurations so far investigated. Because no special attempt was made to cool the leading edge of blade 10 the temperature gradient between the midchord and the leading edge was relatively high, about 400° F.

A composite blade configuration incorporating the split trailing-edge feature of blade 10 and the radial leading-edge slots and reverse-flow features of blade 8 would appear to promise a well-cooled blade cross section with low temperature gradients. Such a blade would probably have vibratory-stress characteristics superior to blade 8 because of its thicker cross section at the trailing edge and more numerous struts in the trailing-edge slot. The use of a larger number of trailing-edge struts is permissible in the split trailing-edge type because the thermal gradients about the struts are not nearly as severe as the gradients about similar struts on the film-cooled configuration. The low gradients about the struts in the split trailing edge are the result of effective cooling of the struts by forced convection, whereas on the film-cooled configuration the induced turbulence and interruption of the film coverage downstream of the struts results in considerably higher gradients.

Spanwise Blade-Temperature Comparisons

The trailing-edge spanwise temperature profiles for blades 8, 9, and 10 for an engine speed of 10,000 rpm and a coolant-to-combustion-gas-flow ratio of 0.063 are presented in figure 5. The conditions for the calculations were the same as those in figure 3. Blades that were cooled by coolant flowing radially outward through the internal coolant passages exhibit, in general, temperature profiles similar to those of blades 9 and 10 (fig. 5), that is, the blade temperature generally tends to increase from root to tip. Blades cooled by coolant flowing radially outward through coolant passages within the midchord section and then reversing near the tip and flowing radially inward near the leading and trailing edges exhibit a spanwise profile at the leading and trailing edges similar to that of blade 8; that is, the blade temperature decreases from root to tip. The curves for blade 10 and blade 8 are drawn as smooth lines. Actually, a series of local hot areas should be indicated for each spanwise position of a strut. The thermocouple instrumentation used in these investigations was not extensive enough to evaluate the temperature increase at each strut location; however, trends of the spanwise temperature profiles were established by the data available. An allowable temperature curve (based on the simple centrifugal stresses and 1000-hr stress-to-rupture data) for a cooled blade made of a typical nonstrategic material is also shown in order to illustrate what is believed to be a nearly optimum spanwise temperature profile. Because the allowable-temperature curve is relatively flat for about the first 40 percent of the spanwise distance for a typical nonstrategic material, it is apparent that the actual spanwise temperature gradient in the cooled blades should be small over the first 40 percent of the blade span. The spanwise temperature distribution exhibited by blade 8 is particularly undesirable because the hottest spanwise temperature is at or near the blade root where the stresses are the highest. It would be possible to adjust the spanwise temperature patterns of blades 8 and 10 by varying the size of the trailing edge slots along the blade span so that the radial temperature gradients approach the optimum, as indicated by the allowable-temperature curve. Adjustment of the spanwise temperature distribution in this manner cannot readily be accomplished unless the coolant is discharged entirely or partly along the blade span.

Cooling-Air Pressure Loss

The pressure loss from the rotor hub to the blade tip was calculated from data measurements at engine speeds of 5000, 7000, 9000, and 10,000 rpm by the method presented in reference 2, which results in a correlation of the pressure loss from rotor hub to blade tip for all engine speeds. Data for engine speeds of 4000, 6000, and 8000 rpm are not presented because mechanical difficulties at these speeds resulted in cooling-air pressure data that were unreliable. The results of the pressure-loss calculations are presented as the upper curve of figure 6, which shows the variation of correlated cooling-air pressure loss with coolant flow for four engine speeds. Excellent correlation for the range of engine speeds investigated was obtained as evidenced by the small deviation of the data points.

The pressure loss through only the blade, from root to tip, is represented by the lower curve of figure 6. This curve was obtained by the method presented in reference 3. The magnitude of the pressure drop for a given coolant flow is about the same for blade 10 as that for the best blades (with respect to pressure-drop) thus far investigated. (See references 2 and 4.)

In addition to the comparison of cooling-air pressure losses for different blades over a range of coolant flows, a comparison of the pressure drops required for a range of cooling effectiveness φ is also significant. The pressure losses for blades 8, 9, and 10 over a range of trailing-edge cooling effectiveness are compared in figure 7. Blades 8 and 9 were selected for this comparison because they were superior as regards trailing-edge cooling effectiveness. This comparison would probably be changed if the split-trailing-edge feature of blade 10 were used in conjunction with a special modification for cooling the leading edge, but it is apparent from figure 7 that blade 10 affords the best combination of trailing-edge cooling and coolant pressure loss of the three configurations considered.

SUMMARY OF RESULTS

The results of an experimental investigation in a modified commercial turbojet engine of an air-cooled turbine-blade configuration having a split trailing edge were as follows:

1. The split trailing-edge configuration provided effective cooling of the trailing edge of the blade. It compared favorably with the best previous methods investigated for special cooling of the trailing-edge portion of the blade. For example, at an engine speed of 10,000 rpm, an effective gas temperature of 1145° F, a cooling-air temperature of 138° F, and a coolant-to-combustion-gas-flow-ratio of about 0.063, the trailing-edge temperature at the midspan position was 755° F, which was only about 30° F hotter than the best blade configuration thus far investigated.

2. The cooling-air pressure loss based on coolant flow rate was found to be about the same as the best blades previously investigated.

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REFERENCES

1. Ellerbrock, Herman H., Jr., and Stepka, Francis S.: Experimental Investigation of Air-Cooled Turbine Blades in Turbojet Engine. I - Rotor Blades with 10 Tubes in Cooling-Air Passages. NACA RM E50I04, 1950.

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, 1950.

3. Hickel, Robert O., and Smith, Gordon T.: Experimental Investigation of Air-Cooled Turbine Blades in Turbojet Engine. III - Rotor Blades with 34 Steel Tubes in Cooling-Air Passages. NACA RM E50J06, 1950.

4. Ellerbrock, Herman H. Jr., Zalabak, Charles F., and Smith, Gordon T.: Experimental Investigation of Air-Cooled Turbine Blades in Turbojet Engine. IV - Effects of Special Leading- and Trailing-Edge Modifications on Blade Temperatures. NACA RM E51A19, 1951.

TABLE I - SUMMARY OF ENGINE OPERATING CONDITIONS AND TEMPERATURE-DIFFERENCE RATIOS

FOR BLADE 10 AT ENGINE SPEEDS OF 4000, 7000, AND 10,000 rpm

Series	Engine speed N (rpm)	Fuel flow w_F (lb/hr)	Average conditions at compressor inlet: Total pressure $P'_{A,c,i}$ (in. Hg abs.)	Average conditions at compressor inlet: Total temperature $T'_{A,c,i}$ (°F)	Cooling-air flow per blade w_a (lb/sec)	Coolant-to-gas flow ratio R	Temperature-difference ratio φ, Thermocouple G	H	I	J	Engine speed N (rpm)	Fuel flow w_F (lb/hr)	Average conditions at compressor inlet: Total pressure $P'_{A,c,i}$ (in. Hg abs.)	Average conditions at compressor inlet: Total temperature $T'_{A,c,i}$ (°F)	Cooling-air flow per blade w_a (lb/sec)	Coolant-to-gas flow ratio R	Temperature difference ratio φ, Thermocouple A	B	C	D
1	3,997	1120	29.26	76	0.0889	0.245	0.439	0.620	0.559	0.727	4,010	1127	29.30	69	0.0891	0.235	0.425	0.500	0.497	0.627
	4,002				.0696	.192	.401	.584	.507	.694	4,010				.0699	.184	.406	.462	.452	.588
	4,005				.0478	.132	.355	.536	.466	.641	3,997				.0493	.130	.369	.410	.392	.538
	4,005				.0399	.110	.328	.511	.427	.610	4,005				.0402	.106	.338	.377	.364	.510
	4,004				.0299	.082	.281	.467	.388	.572	4,003				.0302	.080	.293	.349	.327	.480
	4,003				.0243	.067	.262	.453	.373	.544	4,003				.0294	.078	.278	.321	.296	.457
	4,004				.0196	.054	.235	.423	----	.513	4,005				.0196	.052	.256	.304	.272	.442
	4,004				.0148	.041	.212	.403	----	.486	4,005				.0148	.039	.237	.281	.245	.418
	3,997				.0100	.028	.196	.384	----	.454	4,005				.0101	.027	.221	.250	.213	.406
2	7,000	1842	29.17	78	0.0889	0.125	0.342	0.489	0.463	0.618	7,000	1846	29.24	83	0.0885	0.124	0.424	0.437	0.437	0.525
	6,995				.0698	.098	.308	.460	.422	.584	7,001				.0716	.101	.368	.397	.389	.486
	7,003				.0492	.069	.255	.417	.387	.535	7,022				.0490	.069	.313	.351	.336	.443
	6,993				.0401	.057	.230	.395	.359	.510	7,042				.0399	.056	.292	.327	.313	.420
	5,998				.0301	.042	.203	.372	.333	.476	6,997				.0300	.042	.270	.302	.280	.392
	7,000				.0244	.034	.183	.355	.307	.457	7,000				.0242	.034	.258	.281	.264	.380
	6,990				.0196	.028	.175	.345	.293	.436	7,008				.0194	.027	.247	.267	.247	.368
	6,999				.0149	.021	.153	.318	.275	.405	6,988				.0147	.021	.230	.244	.214	.347
	7,000				.0101	.014	.138	.288	.243	.357	7,000				.0100	.014	.213	.207	.174	.324
3	9,990	3150	29.07	85	0.0985	0.088	0.405	0.434	0.457	0.643	10,000	3140	29.20	85	0.0982	0.088	0.429	0.381	0.411	0.498
	10,010				.0891	.080	.390	.421	.435	.630	10,000				.0891	.080	.423	.365	.393	.487
	10,015				.0699	.063	.357	.394	.401	.596	10,001				.0699	.063	.392	.337	.363	.451
	9,980				.0494	.044	.318	.358	.372	.546	9,995				.0493	.044	.365	.297	.326	.409
	9,995				.0403	.036	.298	.340	.350	.512	9,985				.0399	.036	.331	.269	.294	.387
	9,992				.0300	.027	.287	.328	----	.475	9,990				.0300	.027	.315	.251	.276	.362
	9,992				.0250	.022	.270	.317	----	.454	10,000				.0252	.023	.288	.232	.259	.346

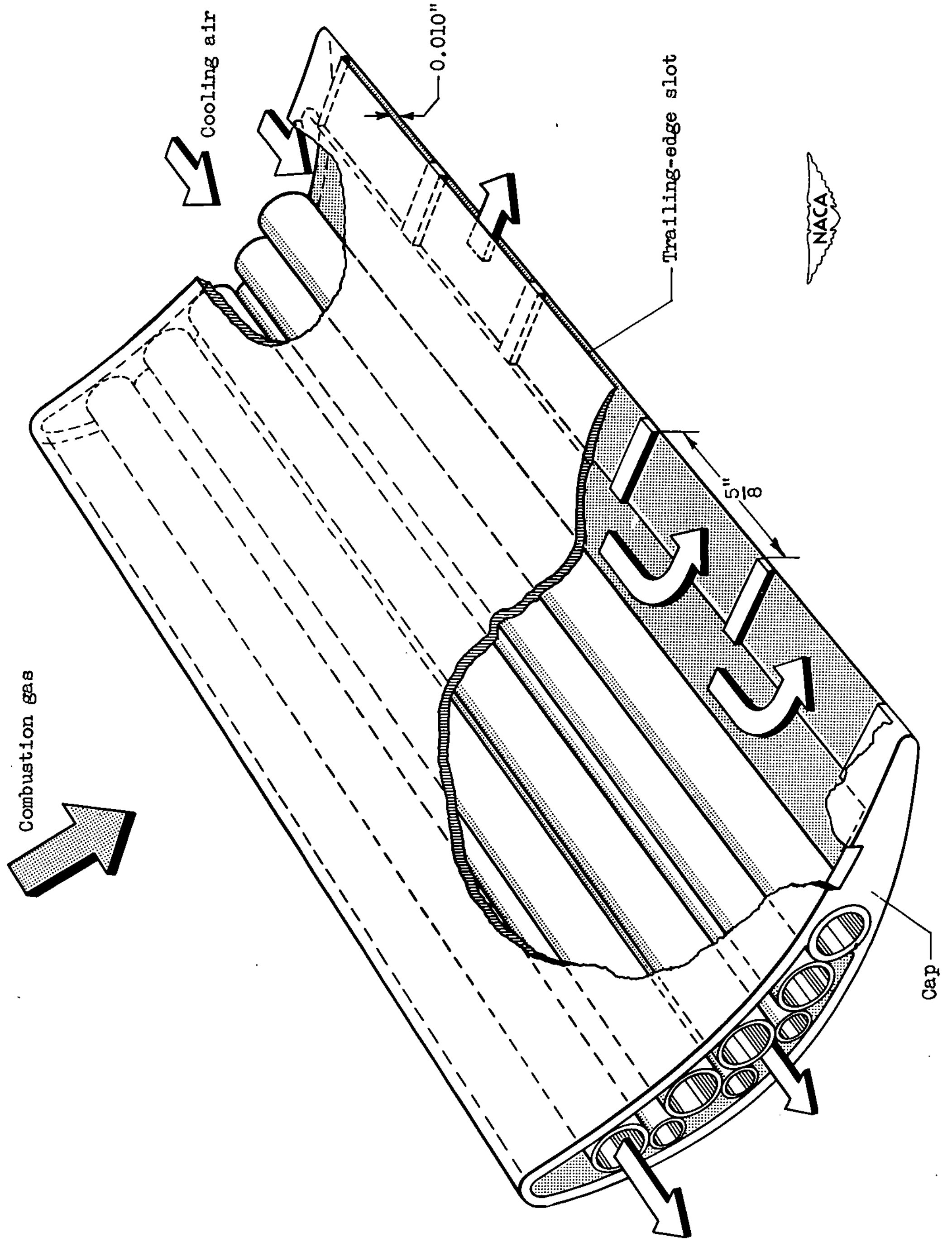

Figure 1. - Internal coolant-passage configuration of cooled blade 10.

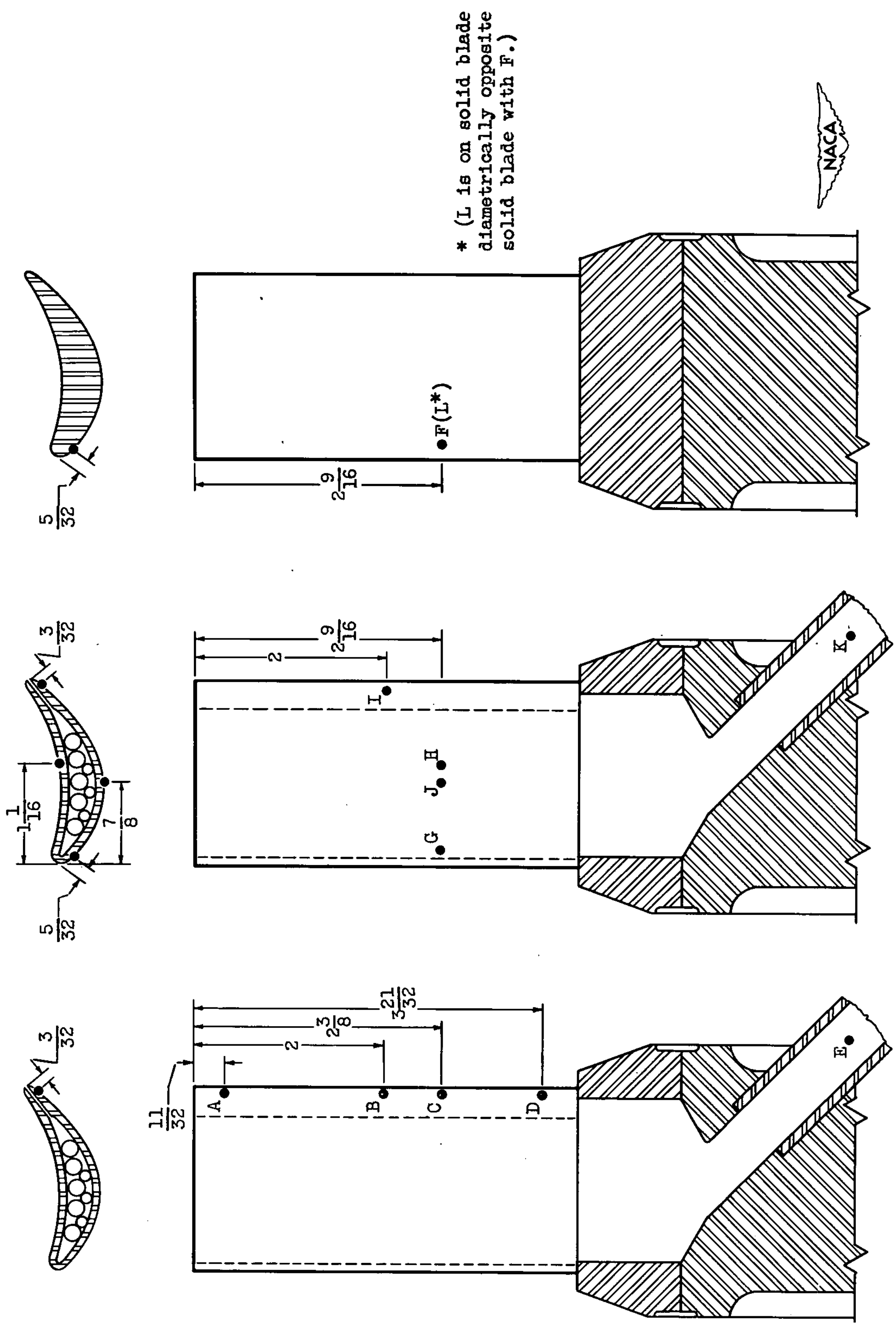

Figure 2. - Schematic diagram of thermocouple locations on cooled and uncooled blades. (All dimensions are in inches.)

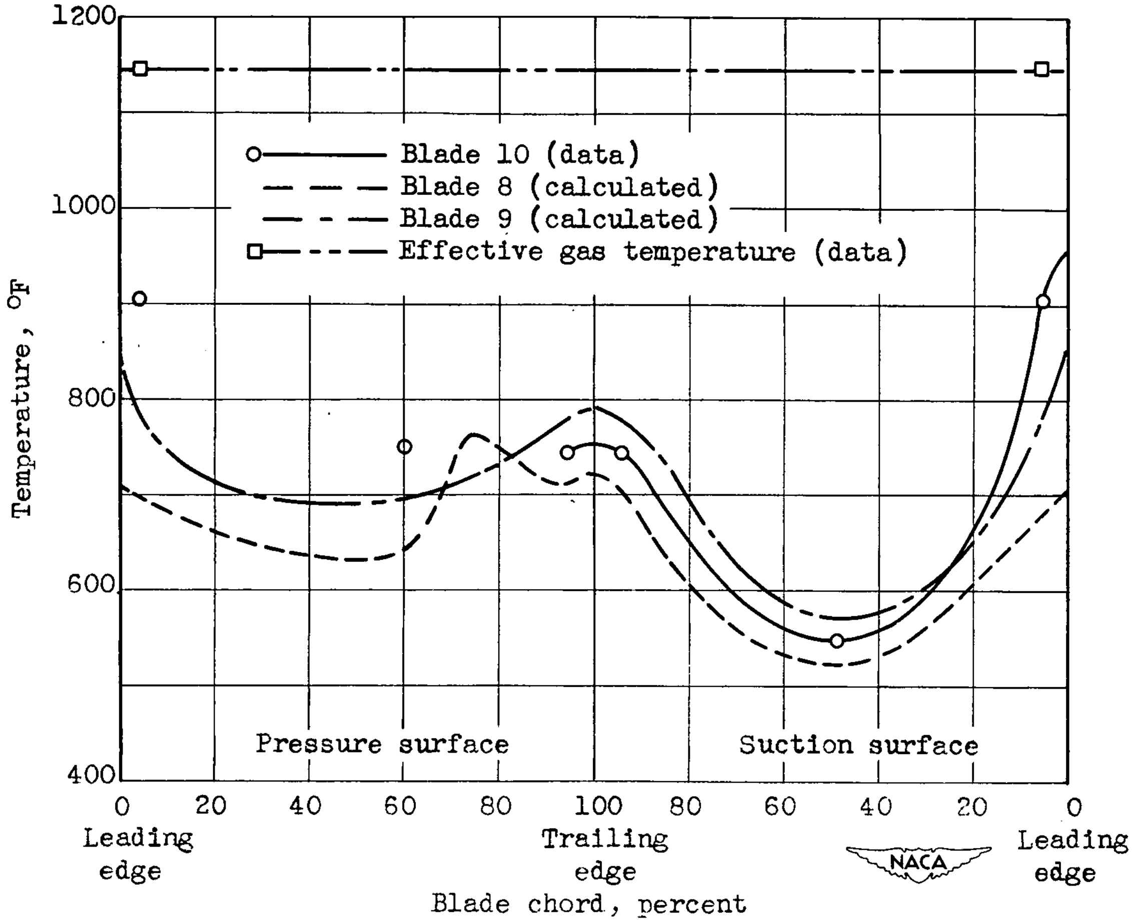

Figure 3. - Comparison of chordwise temperature distribution for blades 8, 9, and 10. Engine speed, 10,000 rpm; coolant-to-combustion-gas-flow ratio, 0.063; cooling-air temperature, at blade root, 138° F.

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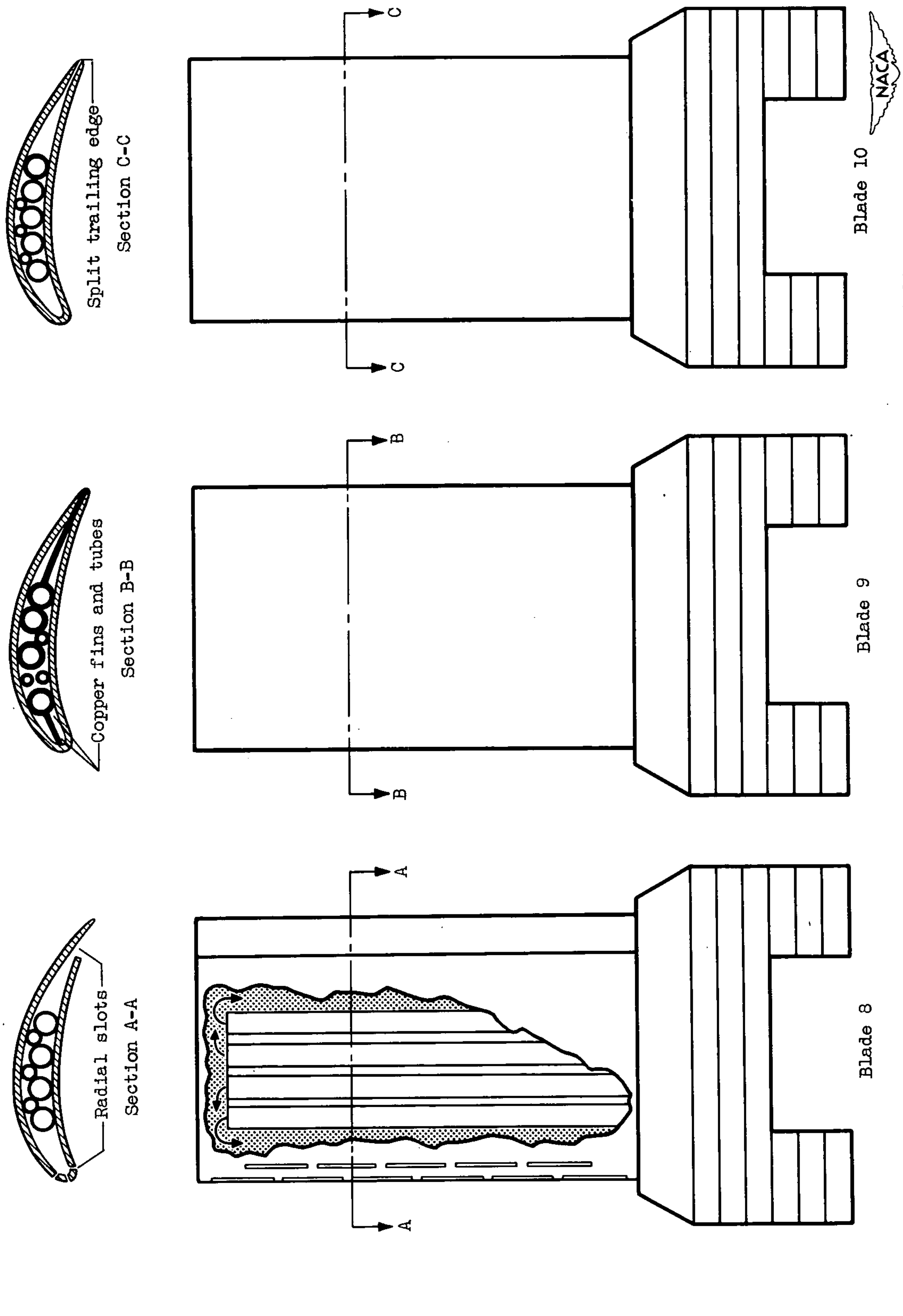

Figure 4. - Comparison of coolant passages of blades 8, 9 and 10.

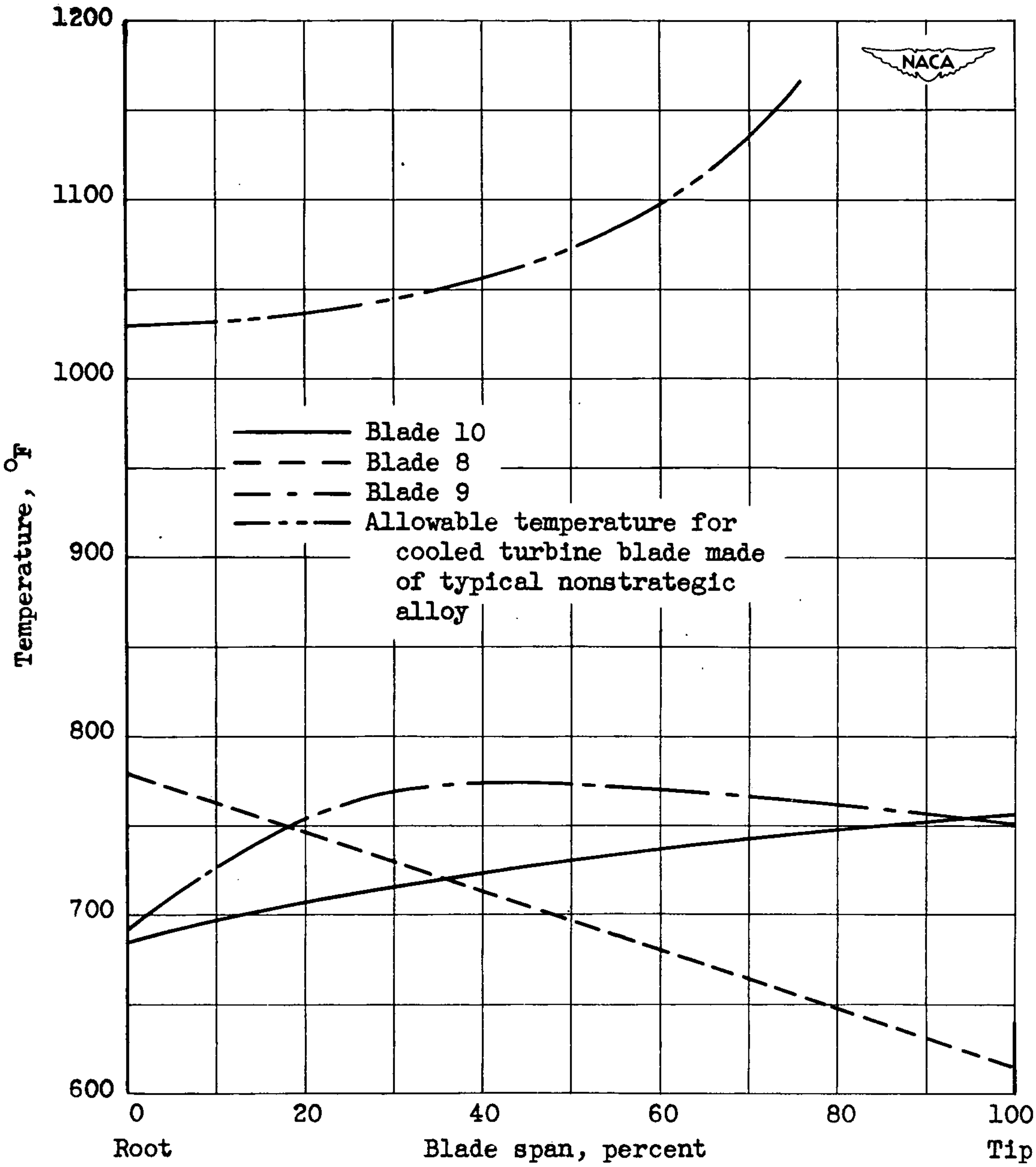

Figure 5. - Comparison of spanwise temperature distribution at trailing edge for blades 8, 9, and 10. Engine speed, 10,000 rpm; coolant-to-combustion-gas-flow ratio, 0.063; cooling-air temperature at blade root, 138° F.

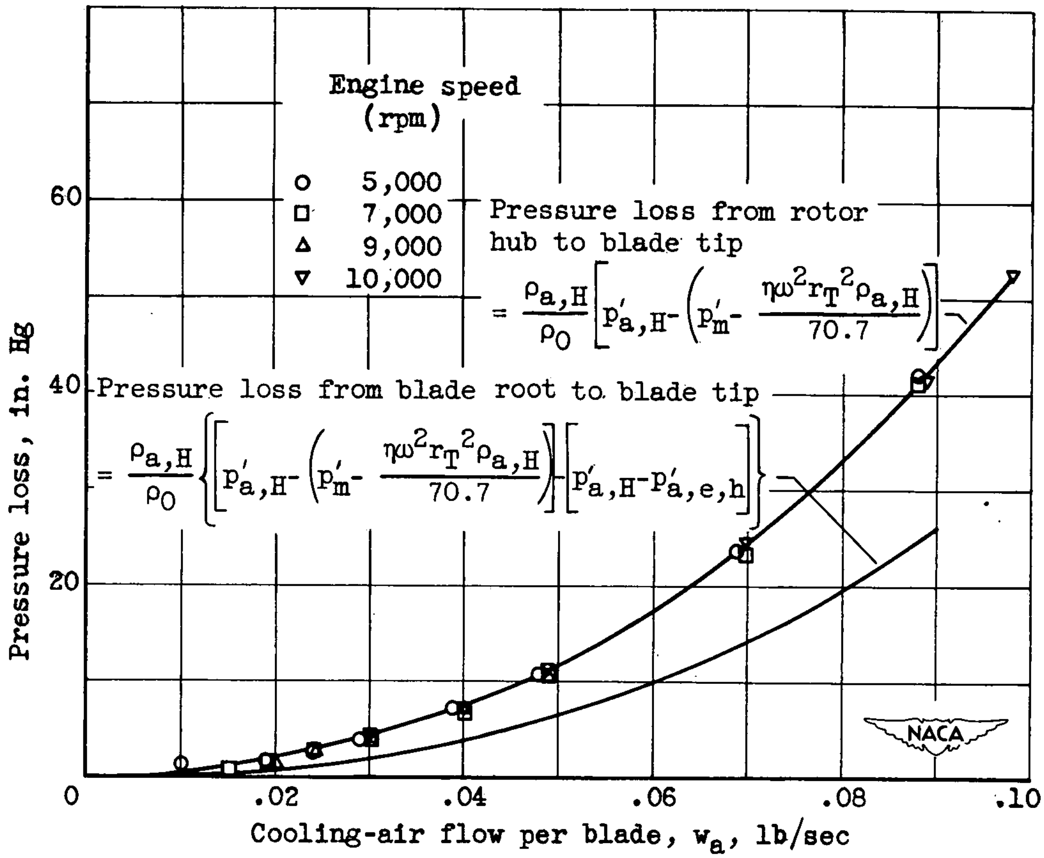

Figure 6. - Correlation of pressure loss from rotor hub to blade tip and from blade root to blade tip.

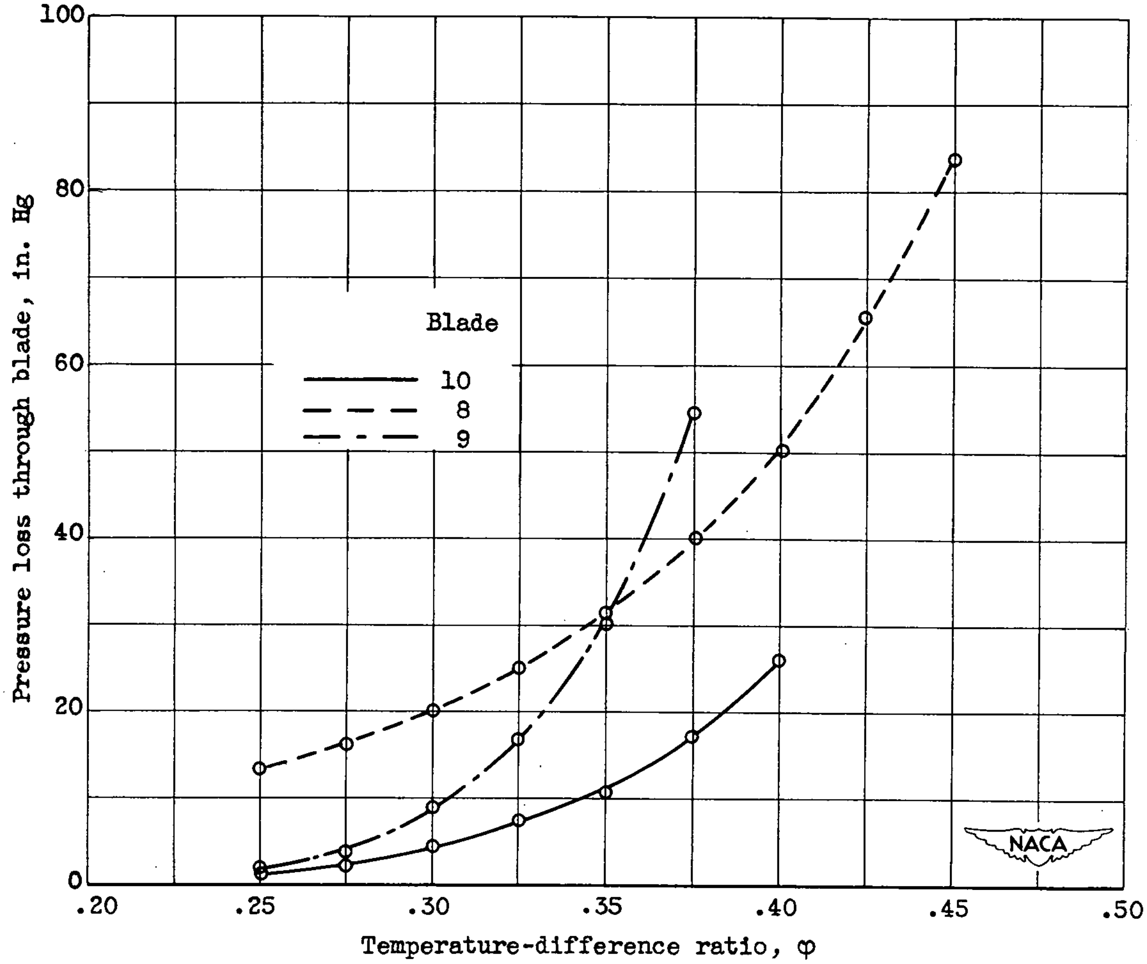

Figure 7. - Comparison of pressure loss required by blades 8, 9, and 10 for range of values of cooling effectiveness. Thermocouple, C; engine speed, 10,000 rpm.