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RESEARCH MEMORANDUM

BLADE-TO-COOLANT HEAT-TRANSFER RESULTS AND OPERATING DATA FROM A NATURAL-CONVECTION WATER-COOLED SINGLE-STAGE TURBINE

By Anthony J. Diaguila and John C. Freche

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NATIONAL ADVISORY COMMITTEE
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BLADE-TO-COOLANT HEAT-TRANSFER RESULTS AND OPERATING DATA FROM A NATURAL-CONVECTION WATER-COOLED SINGLE-STAGE TURBINE

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SUMMARY

Blade-to-coolant heat-transfer data and operating data were obtained with a natural-convection water-cooled turbine over a range of turbine speeds and inlet-gas temperatures. The blade-to-coolant convection coefficients were correlated by the general relation for natural-convection heat transfer that expresses Nusselt number as a function of Grashof and Prandtl numbers. Blade-to-coolant heat-transfer data were obtained over a range of the product of Grashof and Prandtl numbers from 3.00×10^{12} to 1.20×10^{14}. Scatter of the heat-transfer data was less than that obtained in an earlier investigation with the same turbine. The data are displaced from a theoretical equation for natural-convection heat transfer in the turbulent region and from natural-convection data obtained with vertical cylinders and plates. The displacement indicates possible disruption of natural-convection circulation within the blade coolant passages. A constant rate of heat transfer exists for this turbine over the major portion of the Grashof - Prandtl number range investigated. Coolant flow rate has no significant influence on blade-to-coolant heat transfer for this turbine over the range of operating conditions considered.

Comparison of the cooling effectiveness of the blade leading-edge, midchord, and trailing-edge sections by means of a nondimensional temperature-ratio parameter indicates the blade cooling effectiveness is greatest at the midchord and least at the trailing edge.

INTRODUCTION

In order to design cooled turbine blades, blade temperatures must be calculated for the desired design operating conditions. Although blade temperatures can be calculated from theoretically derived equations, it is necessary to know gas-to-blade and blade-to-coolant heat-transfer coefficients. Operating data for similarly cooled blades are also desirable for design purposes. An investigation was therefore conducted to determine blade-to-coolant heat-transfer and operating data for a natural-convection water-cooled single-stage turbine. The results of this investigation are described herein.

A general theoretical relation for natural-convection heat transfer along flat plates is presented in reference 1. In this relation, surface-to-coolant natural-convection heat transfer is expressed by dimensionless parameters. The relation expresses Nusselt number as a function of the Grashof and Prandtl numbers. Experimental investigations of natural-convection flow on vertical cylindrical surfaces and along plane vertical surfaces have been made by other investigators and are summarized in references 1 and 2. In these investigations, correlation of the heat-transfer data was achieved by the general relation for natural-convection heat transfer.

The application of natural-convection flow to the cooling of gas-turbine blades has also been studied analytically. A theoretical analysis (reference 3) shows that in the design of natural-convection water-cooled turbine blades the minimum diameter for a passage of a given length is important with respect to coolant circulation within the passage. The analysis (reference 3) indicates that coolant circulation within the passage may be restricted by the heated boundary layer which builds up around the cylindrical surface of the passage.

Application of natural-convection cooling to an experimental single-stage turbine has also been investigated previously (reference 4). A limited range of operating conditions was covered in that investigation and the blade-to-coolant heat-transfer data obtained were correlated by the general relation for natural-convection heat transfer.

The natural-convection turbine described in reference 4 was redesigned to eliminate rotor coolant leakage and was reinstrumented. The investigation was then continued at the NACA Lewis laboratory to determine (a) blade-to-coolant heat-transfer data over an extended range of Grashof numbers, (b) if the coolant circulation restrictions indicated by theoretical analysis in reference 3 were apparent in this turbine, and (c) turbine operating data.

Blade-to-coolant heat-transfer data were obtained over a range of turbine speeds from 2546 to 9107 rpm and over a range of inlet-gas temperatures from 523° to 836° F. Coolant flow rates of approximately 1.5 to 5.6 gallons per minute were employed. Blade temperatures, coolant temperatures in the blade coolant passages, and cooling effectiveness of various sections of the blade are also presented.

SYMBOLS

The following symbols are used in this report:

c_p specific heat of coolant at constant pressure, (Btu/(lb)($^{\circ}$F))

Gr Grashof number, $\dfrac{l^3\rho^2\beta\omega^2 r(T_{B,av} - T_{l,1})}{\mu^2}$

H heat-transfer coefficient, (Btu/($^{\circ}$F)(sq ft)(sec))

k thermal conductivity of coolant, (Btu/($^{\circ}$F)(ft)(sec))

l characteristic length of coolant passages, (ft)

Nu Nusselt number, $\dfrac{H_l l}{k}$

Pr Prandtl number of coolant, $\dfrac{c_p\mu}{k}$

Q heat flow, (Btu/sec)

r radius to centroid of water column, (ft)

S surface area, (sq ft)

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

w weight flow, (lb/sec)

β coefficient of thermal expansion of coolant, (1/$^{\circ}$F)

μ absolute viscosity of coolant, (lb/(ft)(sec))

ρ density of coolant, (lb/cu ft)

φ cooling effectiveness parameter $\dfrac{T_{g,e} - T_B}{T_{g,e} - T_{l,t}}$

ω angular velocity, (radians/sec)

Subscripts:

av average

B blade

e effective

g combustion gas

i inside

l liquid

o outside

R rim

r root

T total

t tip

1 entering rotor

2 leaving rotor

APPARATUS

Turbine. - The natural-convection water-cooled single-stage turbine operated for this investigation is fully described in reference 4. The test facility consists of a turbine coupled to a dynamometer through a gearbox. The turbine, the turbine installation, and the combustion-gas system are also described in reference 4. A cross section of the turbine rotor and the rotor coolant supply assembly are shown in figure 1. The rotor was fabricated from stainless steel (AISI type 403). The turbine has 31 reaction blades; these blades are of constant cross section with a chord of 1.74 inches, span of 2.44 inches, and no twist. Five blind radial passages extending to within 0.060 inch from the blade tip and ranging in diameter from 0.060 to 0.125 inch are drilled in each blade as indicated in figure 1. The rotor tip diameter is 13.88 inches. The coolant flow path through the rotor is shown in figure 1, and the coolant flow path through the entire turbine is shown in figure 2. Figure 2 indicates that the nozzle blades were provided with coolant passages and a coolant supply system separate from that of the rotor; however, in this investigation the nozzle blades were not cooled. Cracks which could not be repaired and which permitted coolant leakage had developed in the welded nozzle ring construction as a result of previous operation.

Instrumentation. - Coolant inlet and outlet temperatures were obtained with stationary thermocouples as shown in figure 1. Combustion-gas conditions were measured at the inlet-gas measuring station shown in figure 2. Rotor-blade temperatures were obtained with chromel-alumel thermocouples in conjunction with a rotating thermocouple pickup consisting of a slip-ring and brush system. Locations of the thermocouples on the rotor blades and within the rotor-blade coolant passages are shown in figure 3. In order to obtain coolant temperature measurements in the blade coolant passages, the rotor thermocouple instrumentation was altered from that described in reference 4. A thermocouple was inserted at the tip section of the central coolant passage of one blade; a second thermocouple was inserted at the midspan section of the central coolant passage of another blade; and a third thermocouple was inserted at the root section of the central coolant passage of a third blade. The thermocouples, each of which was enclosed in a 0.060-inch tube, were supported along the center line of the passage by a perforated spacer, which contacted the passage wall. The thermocouple leads were brought out of the coolant passage at the blade root. This method of measuring coolant-passage water temperatures caused a flow restriction within the passage; however, an indication of the coolant temperature at several pertinent points of blade-to-coolant heat transfer was nevertheless provided. Inlet-gas temperatures were measured by four shielded chromel-alumel thermocouples spaced in equal radial increments along the blade span 90° apart in a plane 1/4 inch upstream of the nozzle blades. Total and static pressures were measured at the same locations in the same plane as the temperatures. Turbine speed was measured by an electric tachometer and checked with a chronometric tachometer. Rotameters were used to measure fuel and coolant flow rates.

METHOD OF CALCULATION

Evaluation of blade-to-coolant heat-transfer coefficients. - For this turbine, the total heat absorbed by the coolant consisted of the heat transferred to the coolant by convection in the blade coolant passages and the heat transferred to the coolant from the peripheral disk rim area exposed to the hot gases. Heat transferred by the turbine shaft was considered negligible. The total heat transferred to the coolant was determined from the coolant weight flow and coolant temperature rise by the expression

$$Q_T = w_l c_p (T_{l,2} - T_{l,1}) \tag{1}$$

In order to calculate the heat flow to the coolant within the turbine blade, it was first necessary to obtain the heat transferred from the hot gas to the turbine blade. For the determination of heat transfer from the gas to the turbine blade, the required average gas-to-blade heat-transfer coefficient was obtained from the equation

$$H_o = \frac{Q_T}{S_{B,o}(T_{g,e} - T_{B,av}) + S_{R,o}(T_{g,e} - T_{B,r})} \tag{2}$$

It should be noted that the coefficient obtained by equation (2) is an average for the blade and the disk rim between the blades. Because of the uncertainty of the heat flow path and the limited rotor instrumentation, an average coefficient was used. A separate evaluation of the amount of heat absorbed by the coolant through the blade and through the disk rim requires more elaborate instrumentation than was available. The outside blade surface area $S_{B,o}$ included the area of the blade tip. The effective gas temperature $T_{g,e}$ was obtained from equation (3) of reference 4. The integrated blade temperature $T_{B,av}$ was determined from plots of chordwise and spanwise blade temperature against the blade periphery and blade span, respectively. The temperature of the rim section was considered to be the same as the blade root temperature $T_{B,r}$.

By use of the average gas-to-blade heat-transfer coefficient, heat flow from the gas to the blade was obtained from the equation

$$Q_B = H_o S_{B,o}(T_{g,e} - T_{B,av}) \tag{3}$$

Average blade-to-coolant convection coefficients were then calculated by the equation

$$H_i = \frac{Q_B}{S_{B,i}(T_{B,av} - T_{l,t})} \tag{4}$$

In equation (4) the area term $S_{B,i}$ consists of the coolant-passage surface area in the blade. The coolant temperature $T_{l,t}$ was measured at the tip of the coolant passage. Consideration of the natural-convection flow characteristics within the passage indicates the reason for using the coolant temperature at the passage tip. The flow within the coolant passage probably consists of a cooled core of fluid flowing radially outward and a heated column of fluid flowing radially inward along the passage walls. Since flow reversal probably occurs at the tip of the passage, the tip thermocouple provides a more representative average of the coolant temperature affecting heat transfer.

Correlation of blade-to-coolant heat-transfer coefficients. - Blade-to-coolant convection heat-transfer coefficients obtained over a range of Grashof numbers were correlated on the basis of an expression for natural-convection heat transfer

$$Nu = f(GrPr) \tag{5}$$

Fluid properties were based on a film temperature which was an average of the integrated average blade temperature $T_{B,av}$ and the temperature of the coolant at the tip of the coolant passage $T_{l,t}$. Use of a film temperature is a general method of correlating natural-convection data as cited in reference 1. The acceleration term $\omega^2 r$ in the Grashof number replaced the gravity term since the centrifugal forces acting on the fluid far exceed the force due to gravity. The radius r was considered to be the radius to the centroid of a column of water with a length equivalent to the average coolant-passage length. The average coolant-passage length was used as the characteristic dimension l in the Nusselt and Grashof numbers in accordance with natural-convection theory (reference 5).

Determination of blade-temperature effectiveness parameters. - The calculation procedure for correlating cooled blade temperatures developed in reference 6 was used in this investigation except for determination of coolant flow rate. Because of the coolant reservoir at the base of the blades and natural-convection circulation employed in this turbine, some of the coolant flow through the turbine rotor can bypass the blades. Because the coolant flow rate through the blade coolant passage could not be measured in this installation, total flow through the rotor was used in determining the blade cooling effectiveness. The blade temperature data were correlated by the expression

$$\varphi = f(w_l, w_g) \tag{6}$$

where

$$\varphi = \frac{T_{g,e} - T_B}{T_{g,e} - T_{l,t}}$$

Separate temperature ratios or cooling effectiveness parameters were determined for the blade midchord, leading-edge, and trailing-edge sections. Midchord blade temperatures were obtained from an arithmetic average of the temperatures on the blade surface near the midchord (thermocouples 2, 3, 5, 6, 8, 9, 11, 12, 14, 15, 17, and 18 of fig. 3). The leading- and trailing-edge temperatures were determined from an arithmetic average of the three leading-edge thermocouples (1, 7, and 13 of fig. 3) and three trailing-edge thermocouples (4, 10, and 16 of fig. 3), respectively. The coolant temperature $T_{l,t}$ was measured by thermocouple 21 (fig. 3). The effective gas temperature was determined from equation (3) of reference 4. The weight flow of the combustion gas w_g was determined by the summation of turbine air flow and fuel flow. Coolant flow w_l was considered to be the total coolant flow rate through the turbine rotor.

PROCEDURE

The turbine was operated to obtain blade-to-coolant heat-transfer coefficients and operating data over a range of coolant flows, speeds, and inlet-gas temperatures; however, the range of turbine operation is not indicative of turbine design conditions. Turbine speed range was determined by the maximum permissible surface speed of the large diameter carbon seal at the rotor coolant inlet and by the minimum (base) speed of the dynamometer. Maximum inlet-gas temperature was determined by structural limitations of welded sections of the nozzle blade assembly. Minimum gas temperature was fixed by the burner design. The range of turbine operating conditions is given in table I.

In order to obtain blade-to-coolant heat-transfer data, it was necessary to vary the Grashof number. The acceleration term $\omega^2 r$ in the Grashof number was varied by alterations in the turbine speed. The difference between the average blade and entering coolant temperature $T_{B,av} - T_{l,1}$ was varied by altering the turbine-inlet-gas temperature. These terms are the major factors affecting the magnitude of the Grashof number. At each Grashof number the coolant flow was independently varied to determine if coolant flow rate had a measurable effect on blade-to-coolant heat transfer.

As shown in table I, turbine operating data were also obtained for very low coolant flow rates and for slightly higher gas temperatures than were included in the portion of the investigation dealing with blade-to-coolant heat transfer. Partial steam formation in the rotor coolant discharge at flows less than 1.50 gallons per minute prevented accurate evaluation of the heat flow to the coolant. Consequently, heat-transfer coefficients could not be calculated accurately for these runs. Mechanical limitations of the test installation prevented operation of the turbine at coolant flows higher than 5.61 gallons per minute.

RESULTS AND DISCUSSION

Blade-to-coolant heat-transfer results. - Turbine blade-to-coolant convection data for various coolant flows correlated by the general relation for natural-convection heat transfer are shown in figure 4(a). Over the entire range of the product of Grashof and Prandtl numbers GrPr investigated, from 3.00×10^{12} to 1.20×10^{14}, the heat-transfer rate increases slightly as indicated by a Nusselt number increase from 280 to 500. The data indicate a trend of slightly increasing heat transfer up to a GrPr value of 10^{13}. It appears that a constant rate of heat transfer exists for this turbine at GrPr above this value. Although the turbine data were obtained at several coolant flows, separate curves for individual coolant flows are not discernable. Also, no significant increase in Nusselt number is apparent for increasing values of coolant flow, which indicates that coolant flow rate does not affect blade-to-coolant heat transfer for this turbine over the range of operating conditions investigated. As stated previously, coolant flow runs less than 1.50 gallons per minute could not be included in these data because partial steam formation within the water discharge prevented accurate determination of the heat transferred to the coolant.

Blade-to-coolant heat-transfer results obtained in an earlier investigation with this turbine (reference 4) are shown in figure 4(b). An appreciable reduction in scatter can be observed in the results of the present investigation when compared with the data of reference 4. The reduction in scatter is attributed to the elimination of rotor coolant leakage, additional instrumentation which permitted determination of coolant temperature within the blade-coolant passages, and refinements in the calculation procedure for determining the heat flow to the coolant.

The effect of very low coolant flows upon blade-to-coolant heat transfer is shown in figure 5, in which it is further emphasized that coolant flow rate does not significantly affect blade-to-coolant heat transfer for this turbine over the normal coolant flow range. A plot of integrated blade temperatures and coolant temperatures measured within the blade passage at the blade tip against coolant flow rate for nearly constant turbine speed and gas conditions is presented in figure 5. Throughout the entire coolant flow range investigated, from 0.24 to 5.61 gallons per minute, a temperature difference of approximately 115° F between blade and coolant temperature exists, which indicates a constant rate of heat flow from blade to coolant. From a consideration of basic heat-transfer equations, it is apparent that for constant gas-to-blade heat-flow conditions the rate of heat flow from blade to coolant is dependent upon the temperature difference between blade and coolant.

The turbine data are compared in figure 6 with a theoretical equation for natural-convection heat transfer in the turbulent range (reference 3) and with data obtained by other investigators (references 1 and 2) for natural-convection heat transfer outside vertical cylinders and high plates in the turbulent region. Vertical-cylinder and high-plate data were used for this comparison because of a lack of published data for natural-convection heat transfer inside vertical cylinders. The turbine data indicate a scatter of similar magnitude to that of the stationary investigations. The GrPr range investigated with this turbine extends beyond that covered in the stationary investigations. As indicated previously, various factors inherent in the test facility limited turbine operation to a certain range of conditions. Consequently, accumulation of data over a comparable GrPr range could not be achieved. The turbine data are displaced below the theoretical equation for natural-convection heat transfer, as well as below the experimental results of investigations with vertical cylinders and plates. The turbine data do not follow the trend of the theoretical relation and appear to display a constant heat-transfer rate for values of GrPr between 10^{13} and 1.20×10^{14}. This lower constant rate of blade-to-coolant heat transfer indicates a possible disruption of natural-convection circulation within the blade coolant passages and thus corroborates the analysis of reference 3, in which an analysis was made for a coolant-passage length-to-diameter ratio of 25, such as exists for this turbine. The analysis indicated the possibility of a build-up of the heated boundary layer adjacent to the coolant-passage wall and a resulting coolant flow restriction.

Operating results. - Values of one minus a cooling effectiveness parameter $(1 - \varphi)$ for various coolant-to-gas flow ratios for this turbine are presented in figure 7. This method of plotting deviates from that of reference 6, in which the parameter φ is plotted directly against the coolant-to-gas flow ratio. A more favorable comparison is effected since displacement between the individual curves is more pronounced and the relative cooling effectiveness of the various blade portions is readily apparent when plotted in this manner. The section of the blade with the lowest values of $1 - \varphi$ represents the most effectively cooled portion. A comparison of the effectiveness parameters for various sections of the blade indicates the cooling effectiveness to be least at the trailing edge, slightly greater at the leading edge, and greatest at the midchord, a distribution similar to that obtained with air-cooled blades (reference 7). The cooling effectiveness parameter φ for the natural-convection water-cooled turbine blades is higher than for air-cooled turbine blades as reported in reference 7. However, a direct quantitative comparison of these air-cooled-blade data with the water-cooled-blade data obtained from this turbine cannot be made for several reasons. For example, much higher gas flow per unit area at the turbine inlet (three to seven times greater) passed through the air-cooled turbines investigated. Also, the total flow through the rotor was used in determining the blade cooling effectiveness.

Typical chordwise temperature distributions at the tip and the root section of the turbine blade are presented in figure 8. For relatively constant gas flow, an average trailing-edge temperature of 515° F and an average leading-edge temperature of 470° F at the blade tip occur for coolant flows of 3.10 and 5.61 gallons per minute. At the root section the blade temperature distribution is approximately 150° F lower than around the blade tip except for the trailing-edge temperature. A marked decrease of about 350° F in the trailing-edge temperature occurs at the blade root. This decrease in temperature may result from increased cooling by conduction near the trailing edge at the blade root. As indicated in a cross section of the turbine rotor in figure 1, the rim thickness at the trailing edge is considerably less than at the leading edge. Consequently, cooling by conduction through the rim at the trailing edge should be more effective. The general decrease in temperature at the blade root may be attributed to rim cooling because of a full flowing coolant reservoir at the blade base. Displacement between the two chordwise temperature-distribution curves around the blade root for two coolant flows is probably the result of more pronounced rim cooling as the flow rate through the rotor is increased. It should be noted that, although the leading- and trailing-edge temperatures at the blade-tip section considerably exceed midchord temperatures, they are well below the temperature level of approximately 1000° F permitted by the strength properties of the turbine material.

Coolant temperature levels at three radial locations in the blade coolant passage are shown in figure 9 for constant gas mass flow and for relatively constant turbine-speed conditions. For increasing gas temperatures and decreasing coolant flow rates, the temperature level of the coolant in the passages increases. At coolant flows of 3.10 gallons per minute or greater, the coolant temperature increases uniformly from root to tip. For coolant flows of 1.50 gallons per minute or less, a marked increase in coolant temperature from midchord to tip can be observed. Displacement between temperature curves for different coolant flows results from the rotor design characteristics together with the method of coolant circulation employed and changes in the gas temperature. As the amount of coolant entering the rotor decreases, the temperature level within the rotor coolant reservoir increases; the temperature of the coolant entering the blade coolant passages is thus increased. It appears from consideration of figure 9 that coolant flow rate affects the temperature level of the coolant more than does gas temperature. This influence is evidenced by the large increase in coolant temperature that results from increasing the coolant flow from 1.5 to 3.1 gallons per minute while the gas temperature is held at approximately 809° F.

SUMMARY OF RESULTS

The following results were obtained from an investigation with a natural-convection water-cooled single-stage turbine:

1. Blade-to-coolant heat-transfer data were correlated by the general expression for natural-convection heat transfer, which expresses Nusselt number as a function of Grashof and Prandtl numbers. Scatter of the data was reduced from that obtained in an earlier investigation with this turbine.

2. Displacement of the turbine data from a theoretical equation for natural-convection heat transfer in the turbulent region and from vertical cylinder and plate data indicated the possibility of disruption of natural-convection circulation within the blade coolant passages.

3. A constant rate of heat transfer existed for this turbine over the major portion of the range of the product of Grashof and Prandtl numbers investigated.

4. Coolant flow rate had no significant effect on blade-to-coolant heat transfer for this turbine over the range of operating conditions considered.

5. Cooling effectiveness of the turbine blade was greatest near the blade midchord and least at the trailing edge.

Lewis Flight Propulsion Laboratory
National Advisory Committee for Aeronautics
Cleveland, Ohio, September 5, 1951

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TABLE I - RANGE OF TURBINE OPERATING CONDITIONS FOR INVESTIGATION

TO OBTAIN BLADE-TO-COOLANT HEAT-TRANSFER AND OPERATING DATA

	Coolant flow (gal/min)	Indicated turbine speed (rpm)	Effective gas temperature (^{o}F)	Temperature difference: integrated blade-inlet coolant (^{o}F)
Blade-to-coolant heat-transfer and operating data	1.50	2546-3721	542-548	155-169
	2.19	2546-8912	515-810	145-261
	3.10	2546-8912	520-812	127-244
	4.24	3819-9010	505-816	157-233
	5.61	4603-9107	788-806	200-213
Operating data	0.24	5631-9010	864-883	307-318
	.46	5631-8912	863-879	312-316
	.91	3917-8912	848-885	298-322
	1.50	4603-9107	773-810	254-266

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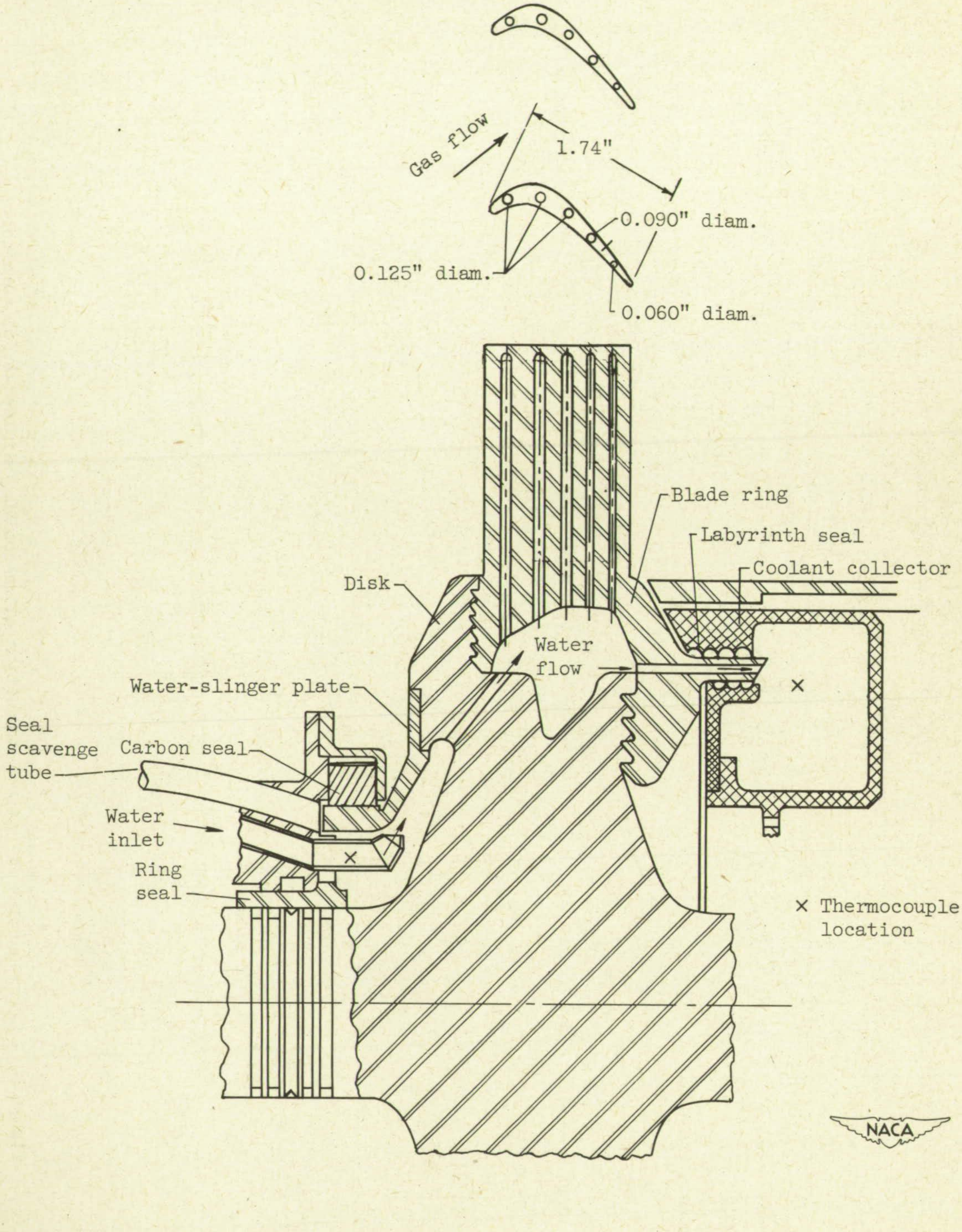


Figure 1. - Cross section of rotor and rotor-coolant system of natural-convection water-cooled stainless-steel turbine.

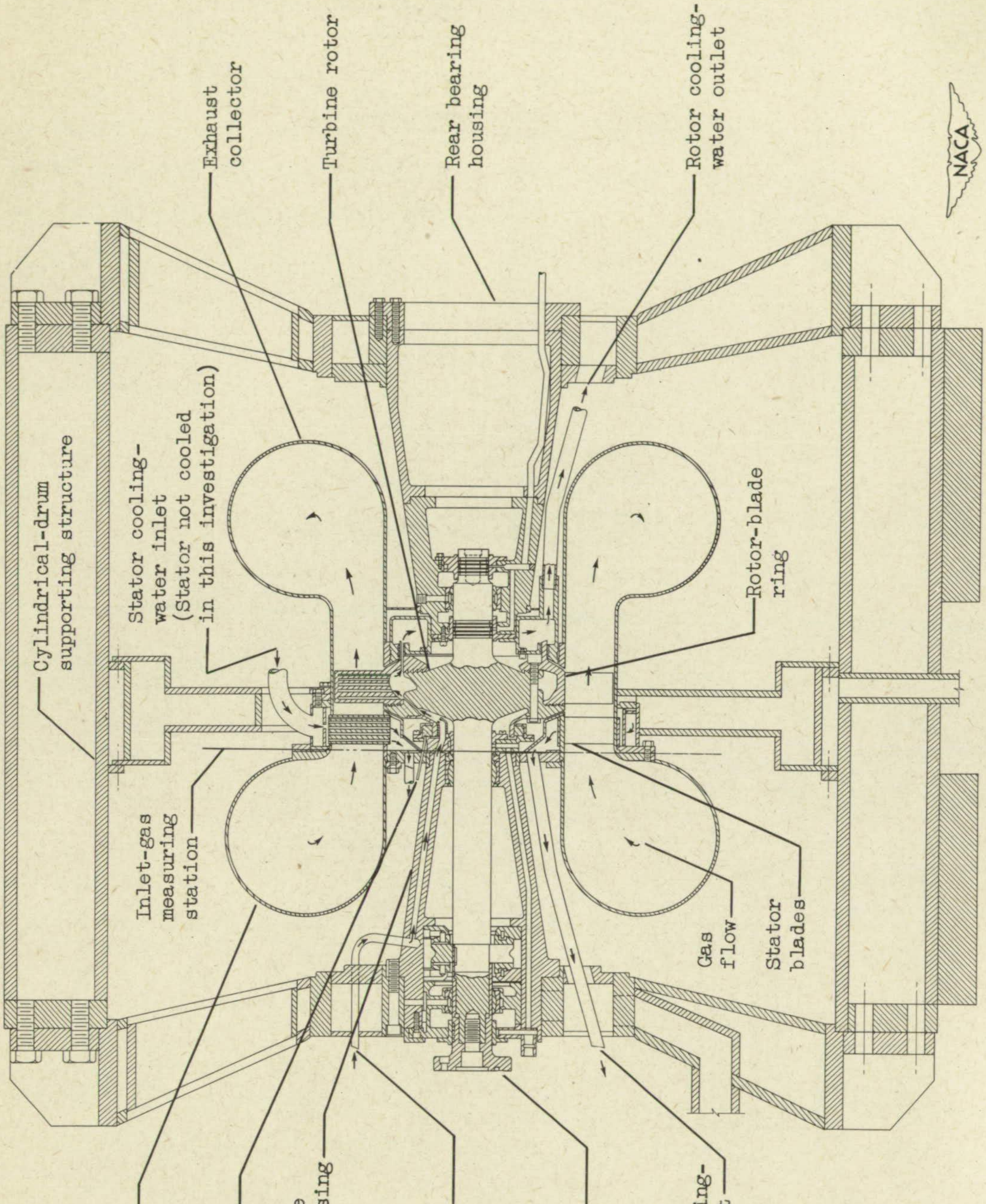


Figure 2. - Cross-section of natural-convection water-cooled stainless-steel turbine.

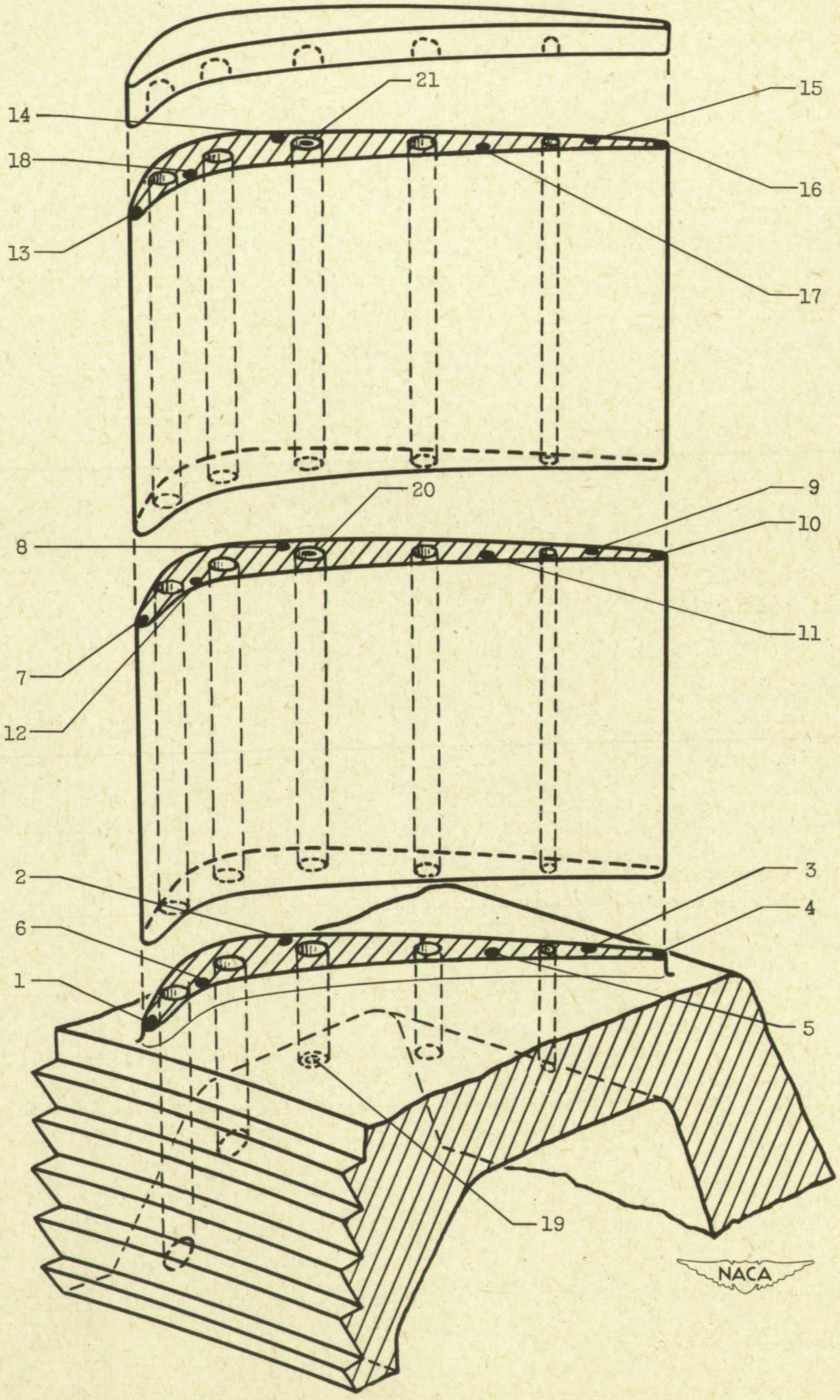
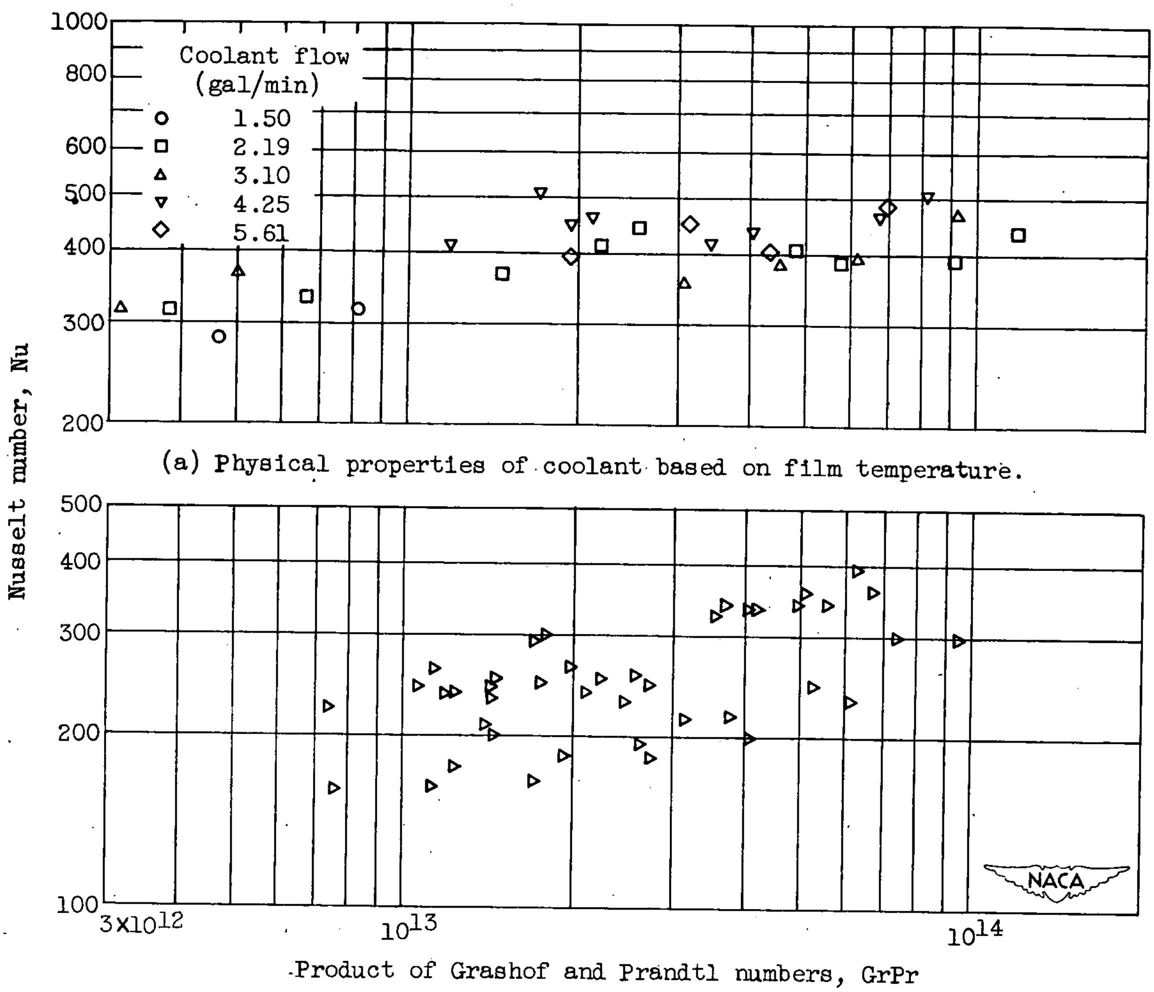


Figure 3. - Diagrammatic representation of blade thermocouple locations on natural-convection water-cooled turbine.



(b) Physical properties of coolant based on bulk temperature. Coolant flow range, 2 to 8 gallons per minute. (Data from reference 4.)

Figure 4. - Correlation of blade-to-coolant heat-transfer data for natural-convection water-cooled turbine.

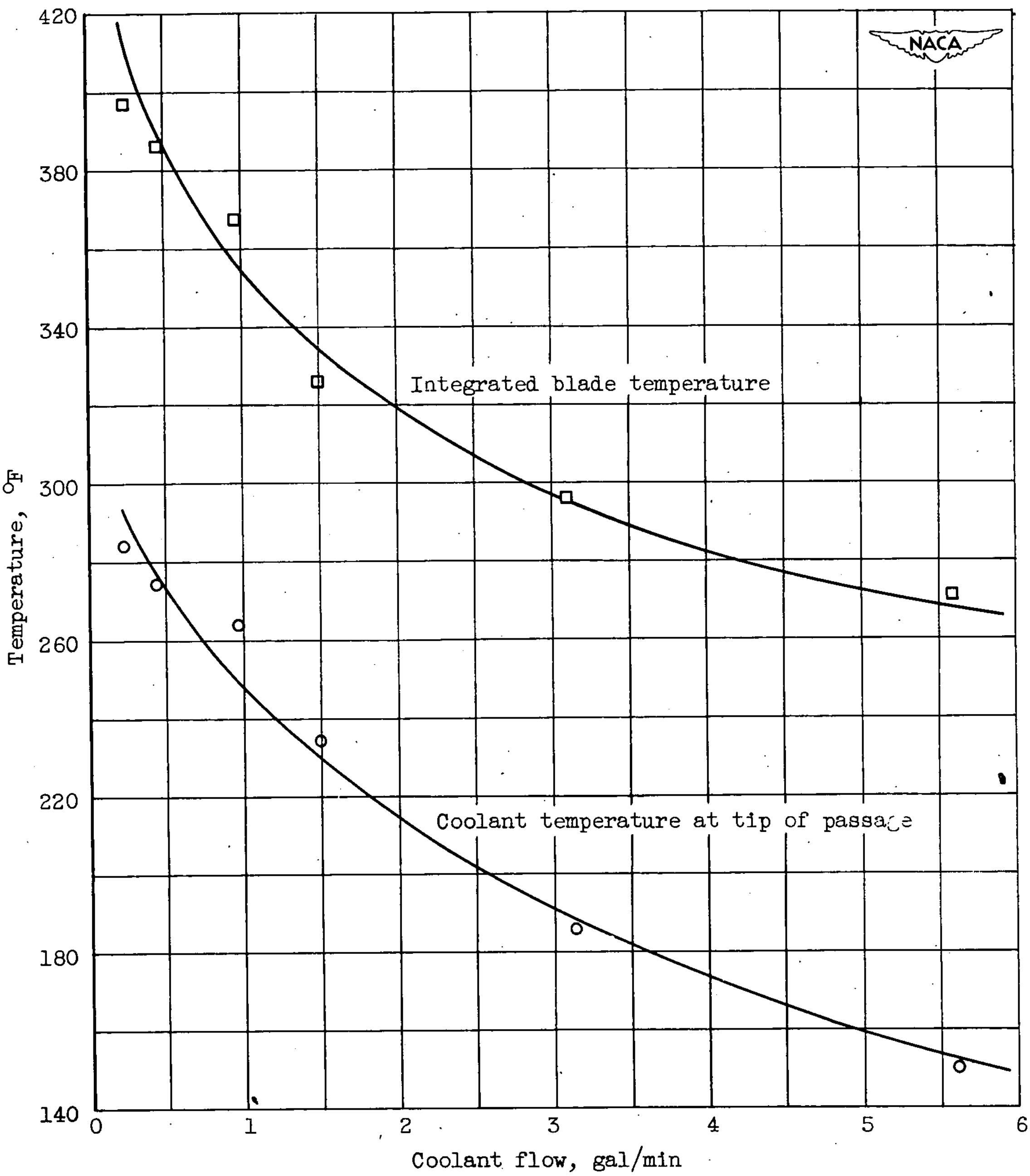


Figure 5. - Integrated blade temperature and coolant temperature at blade passage tip over the total range of coolant flows. Turbine speed, 8912 to 9107 rpm; effective gas temperatures, 773° to 883° F; gas flow, 2.5 pounds per second.

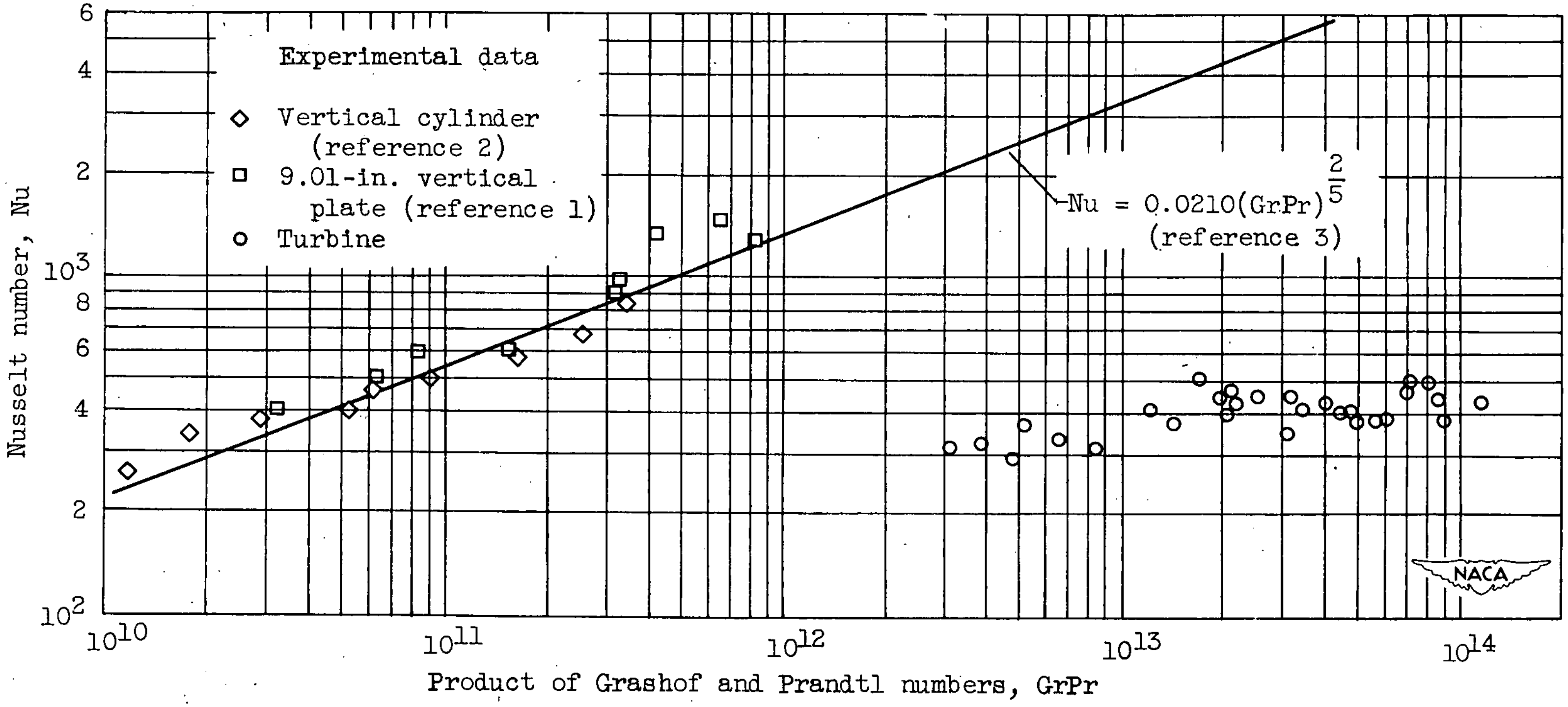


Figure 6. - Comparison of turbine data with NACA theoretical equation and with data for natural-convection heat transfer on vertical plates and cylinders in turbulent region. Physical properties of coolant evaluated at film temperature.

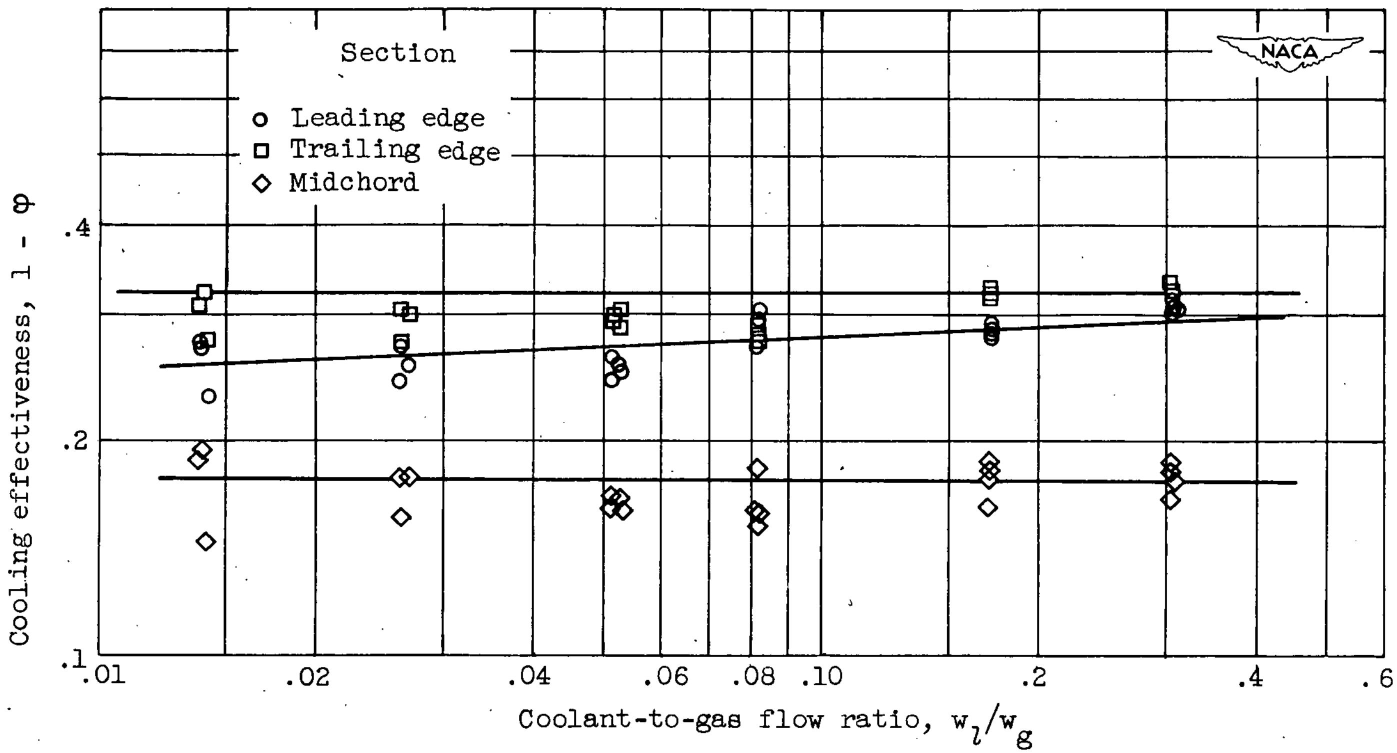


Figure 7. - Cooling effectiveness for three sections of natural-convection water-cooled turbine blades.

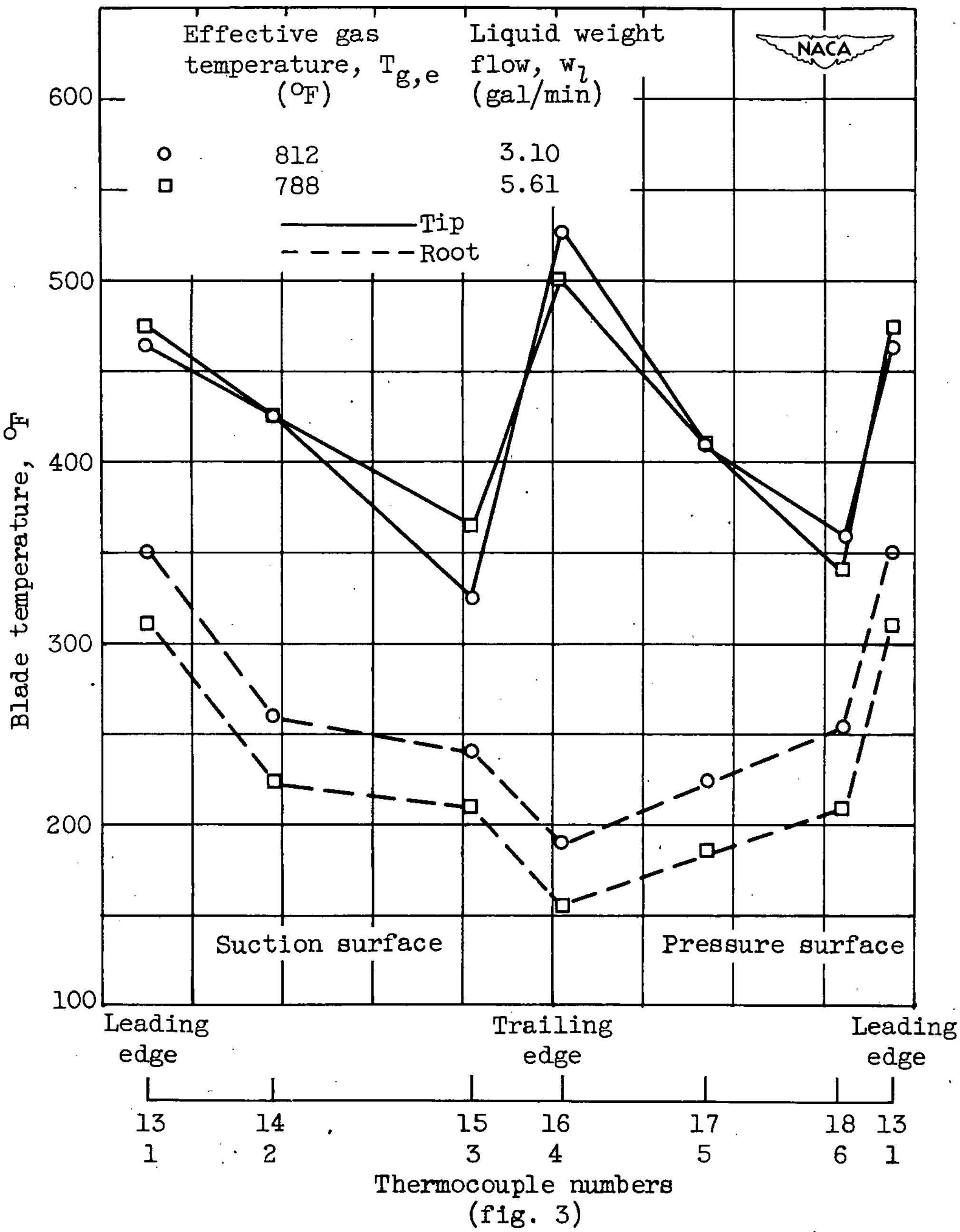


Figure 8. - Typical chordwise temperature distributions around tip and root sections of natural-convection water-cooled turbine blade. Turbine speed, 8912 to 9107 rpm; gas flow, 2.5 pounds per second.

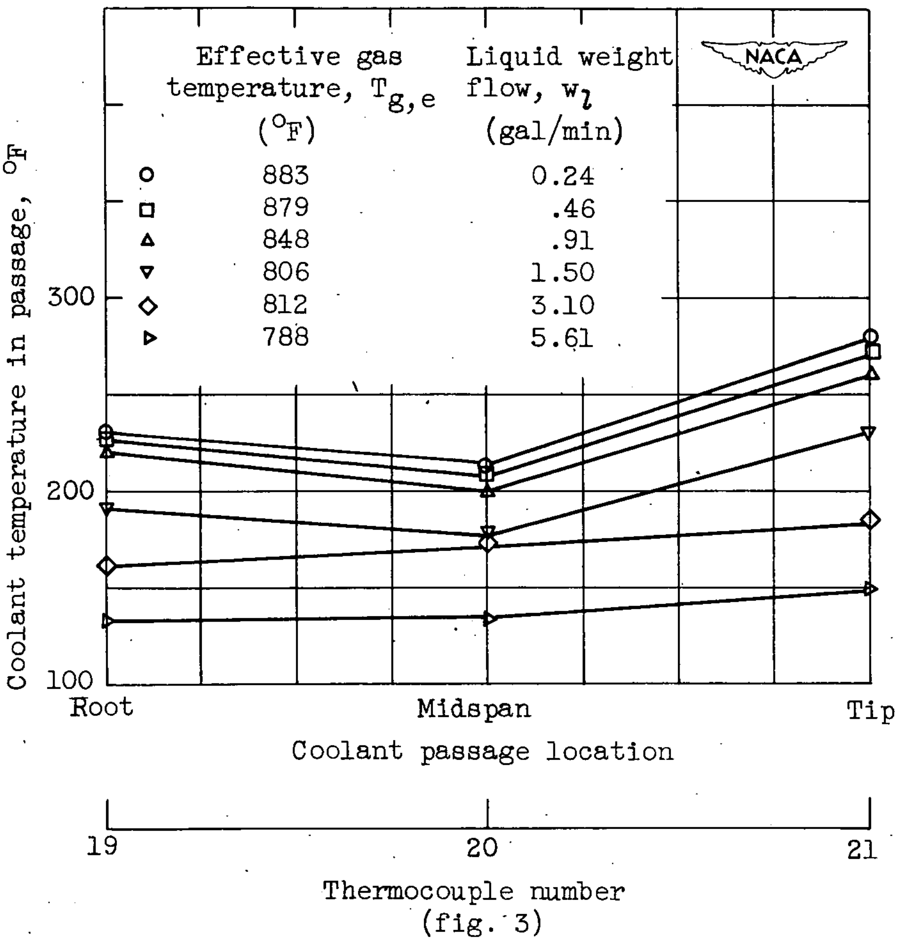


Figure 9. - Radial coolant temperature distribution in blade passage. Turbine speed, 8912 to 9107 rpm; gas flow, 2.5 pounds per second.