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RESEARCH MEMORANDUM

EXPERIMENTAL INVESTIGATION OF COOLANT-FLOW CHARACTERISTICS OF A SINTERED POROUS TURBINE BLADE

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SUMMARY

As part of a program to study the application of transpiration cooling to gas-turbine blades, an experimental investigation was conducted on a porous stainless-steel blade to evaluate the flow characteristics and the peripheral distribution of air flowing through its walls. This evaluation was made in order to obtain sufficient data to develop the methods whereby local coolant-flow rates could be determined for the porous blade in a high-temperature, high-velocity gas stream. The local coolant-flow rates at high temperatures were required for a subsequent heat-transfer investigation.

The results of three separate experiments were combined to determine the coolant-flow characteristics of the porous turbine blade. These experiments consisted of the following: (1) the relation between pressure drop and air flow through two porous disks was checked at room temperature and at elevated temperature levels; (2) the local cooling-air-flow distribution through the walls of a porous blade was observed at room temperature with no flow past the blade for a range of imposed pressure drops; and (3) the variation in static pressure about the periphery of an impermeable blade of identical blade profile was determined in a cold-air stream over a range of Mach numbers. The results of these separate experiments were correlated and combined to determine the local flow rates through the walls of a porous blade in a static cascade. Flow distribution data are presented for the case of a 1000° F gas stream at an inlet Mach number of 0.45.

In order to check the validity of the equation used to correlate the coolant flow rates through a porous wall and the imposed pressure drops over a range of temperatures, the flow through two porous disks was observed at several temperature levels for a range of pressure drops. Results indicated that the correlation equation was sufficiently accurate for the contemplated heat-transfer work.

INTRODUCTION

Because of the limitations in the heat-transfer rates which can be obtained with the usual forced-convection methods of air-cooled gas-turbine blades, other methods of blade cooling are being investigated at the NACA Lewis laboratory. One of these methods is transpiration cooling, which consists in blowing relatively cool air through the porous walls of the blade into the gas stream. Because of the porous structure, the metal-to-coolant heat-transfer rate is high. In addition, the cooling air, in discharging into the boundary layer, keeps the mean boundary-layer temperature well below that of the hot gases flowing past the blade. The large thermal resistance of this boundary layer gives transpiration cooling its effectiveness and makes possible its use in such severe applications as rocket nozzles (reference 1). This method of cooling therefore merits consideration as a possible means of cooling gas-turbine blades. The theory and some of the aspects of the application to gas turbines are discussed in reference 2.

In such an application, two formidable problems exist with respect to cooling. The first problem is cooling an irregular shape with long leading- and trailing-edge sections in which internal coolant-supply passages cannot readily be provided. The second problem is that of obtaining adequate but not excessive cooling around the perimeter in the presence of the large pressure variations that exist around turbine blades.

In a quantitative evaluation of the heat-transfer characteristics of a transpiration-cooled blade in a hot-gas stream, a knowledge of local coolant-flow rates is as necessary as information on the wall, gas stream, and coolant temperatures. Local coolant-flow rates vary because of differences in the permeability and thickness of the blade wall and differences in local external pressures resulting from the flow of gases around the blade. The effect of the temperature level on the coolant weight flow through the blade walls may have to be considered when wide temperature variations are encountered.

The means for evaluating local cooling-air flow rates through the walls of a porous blade in a hot gas stream are presented herein. The local cooling-air-flow rates were investigated at room temperature with constant ambient pressure. The local static-pressure distribution around the periphery of an impermeable blade of identical profile was observed in a static cascade at room temperature for a series of inlet Mach numbers up to 0.46. These data were correlated and the method of correlation checked for temperature effect by observing the air-flow pressure-drop relation for two porous disks at temperatures up to 600° F. The data were then used to calculate the local cooling-air flow rates about the periphery of the porous blade in a static cascade in which hot gases with temperatures as high as 1000° F passed through the cascade at an inlet Mach number as high as 0.54. The heat-transfer aspect of these experiments is presented in reference 3. The present report, while necessarily referring to those heat-transfer experiments, is concerned with the investigations carried out and the methods used to determine the local cooling-air flow rates through the walls of the porous blade during the heat-transfer investigation.

The blades and disks, which were used in this investigation, were made at Stevens Institute of Technology under an NACA contract to develop the technique of fabricating porous stainless-steel turbine blades.

APPARATUS AND EXPERIMENTAL PROCEDURES

Obtaining measurements of cooling-air flow rates through local areas of the porous blade in a cascade through which hot gases are flowing without disturbing the coolant and gas flows and without affecting the heat-transfer processes is an imposing problem. In the present investigation no attempt was made to obtain such measurements directly; instead, a series of relatively simple tests were made and correlation plots were employed to yield the desired information. The analysis and justification for methods employed will be discussed in a subsequent section.

Three separate sets of experimental data were obtained. The pressure variation around the blade periphery was determined in a cascade investigation, which was made at room temperature using an impermeable wall blade. The blade was equipped with static-pressure orifices on the blade surface. The relations of the Mach number and total pressure at the cascade inlet to the local static pressures on the blade surface were obtained. The local coolant flow rates through the porous-wall blade were measured in tests at room temperature with no gas flow over the outside of the blade surface and then were correlated in a manner that would account for variations in porous-metal temperature. The correlation method was checked by observing the air-weight flow rates through two porous disks for a range of temperatures.

Porous Disks

Apparatus. - Inasmuch as the local coolant-flow calibration data obtained on the cold porous blade were to be used to determine local flow rates at elevated temperatures, an investigation of the effect of porous-wall temperature on air-flow rate through the wall for a given pressure drop was necessary in order to determine the possible error involved in the use of such data. In the investigation for the determination of this wall-temperature effect, two porous disks, which were made from the same powders and with the same techniques as the porous blades, and which will be described later, were used. The porosity of both disks was 20 percent. The porosity in percentage is defined as $\left(1 - \frac{\text{density of porous compact}}{\text{density of parent material}}\right) \times 100$. The diameter of both disks was 1.50 inches but one was 0.091 inch and the other 0.145 inch thick in order to check the effect of thickness on the correlation methods.

The setup used with these disks is shown in figure 1. Air at a pressure of 125 pounds per square inch gage passed successively through a pressure regulator, a filter, a rotameter, an electric furnace, an additional filter consisting of a porous blade installed in a tank to remove any scale that might have been picked up from the heater, and finally through the test section in which the porous disk was clamped. Details of the method of mounting the disks are shown in figure 1. Static-pressure taps were installed upstream and downstream of the disk and a thermocouple was spot-welded to the downstream side of each disk as shown in figure 1. This thermocouple measurement was used as the temperature of the air leaving the disk. The apparatus was insulated to minimize heat loss.

Experimental procedure. - Experiments were conducted on the two porous disks in order to determine the effect of disk temperature on air flow through the disks by measuring the air-flow rates for a series of pressure drops across the disks at several constant disk temperatures. The disks were calibrated at room temperature and also at temperatures from 315° to 490° F for the thick disk and from 460° to 600° F for the thin disk.

Porous Blade

Blade description and fabrication methods. - The porous blades used in this investigation were untwisted and untapered and had a 1.97-inch chord, a 3.5-inch span, and a nominal 0.12-inch wall thickness. On the pressure side of the blade, wall thickness was nearly constant at 0.125 inch, whereas the wall thickness on the suction surface varied from 0.090 inch near the leading and trailing edges to 0.115 inch at midchord. The external profile was the same as that of the root section of the rotor blades of a current jet engine. Several views of the blades are shown in figure 2 and their dimensions are shown in figure 3.

The blades were made of AISI type 301 stainless-steel powder. The powder was a -100+200 mesh fraction and was made up of irregular particles. The porosity of the blades was controlled by the powder particle size and the compacting pressure, which was used in forming the blades, and was influenced by the coining operation used during manufacture. No porosity producing agent was used.

The procedure that was followed in forming the blades is outlined in the following steps. The powder was first compacted with a lubricant-binder in a die, which had the contour of the blade; the die had a core rod through the center so as to produce hollow blades. A compacting pressure of 35 tons per square inch was applied in the spanwise direction. After compacting, the blade was removed from the die and was presintered in a controlled atmosphere to remove the volatile lubricant binder. This first heating was followed by a coining process; that is, the blade was placed again in the die and pressed in order to increase the density and the strength of the blade. After the coining operation, the blade was sintered in a controlled atmosphere.

The blades made by the process described were characterized by a variation in porosity along the blade span. This variation was the result of friction between the powder and the sides of the die as the blades were compacted from the blade ends. The magnitude of the porosity variation along the blade span was determined by cutting a blade into nine equal pieces and determining the density of each one of these pieces. The average porosity at the ends of the blades was approximately 12 percent and at the midspan about 30 percent. Peripheral variations existed but were not determined.

Apparatus. - Local cooling-air flows were obtained on a cold blade by the individual collection and measurement of the air that passed through a number of given areas on the blade surface for a series of pressure drops imposed across the blade wall. The apparatus that was used in the determination of these local cooling-air flow rates is shown in figure 4. Air was supplied to the setup at a pressure of 125 pounds per square inch gage. This air was then passed successively through a pressure regulator to give the desired pressure inside the blade, a filter to remove any foreign particles from the air, a rotameter to measure the total rate of air flow to the blade, and finally through the blade walls into the room in which the setup was placed. Air issuing from a selected local spot on the blade was collected by a sampling tube, which was held against the blade outer surface as illustrated in figure 4. Several sampling tubes of 1/2-inch diameter thin-walled steel tubing, which had been machined so that the ends would fit the contour of the blade at particular locations around the periphery, were used. The tubes also had the contoured end ground to a sharp edge. A static tap was installed in the wall of each sampling tube so that the pressure inside the tube could be adjusted to the same value as the ambient-air pressure around the outside of the blade. Under these conditions, the presence of a sampling tube against the wall should have a negligible effect on the flow through the wall. The air collected by each of the sampling tubes was measured with a rotameter, which was located (fig. 4) between the tube and the needle valve. The pressure and the temperature of the air inside the blade was measured with a static-pressure probe and a thermocouple, respectively, as shown in figure 4.

Experimental procedure. - The local cooling-air flow rate out of an area on the surface of a cold porous blade was obtained by placing the sampling tube against the blade wall at a particular location, equalizing the sampling-tube pressure with ambient pressure, and measuring the air-flow rates for a series of pressures inside the blade. This procedure was followed for ten locations around the blade perimeter at the midspan.

Pressure Distribution Around Impermeable Blade

Apparatus. - The apparatus used was essentially that employed in the heat-transfer experiments described in reference 3. Air drawn from a 40 pounds per square inch gage source was passed successively through an orifice, a surge tank, an inlet nozzle, a cascade wherein the air was turned through approximately $81\frac{1}{2}^{\circ}$, and was then discharged into a low-pressure exhaust system. The inlet nozzle and the approach section are shown in figure 5 and a cross section through the seven-blade cascade is presented in figure 6. Because this phase of the work was to be carried out at room temperature, it was possible to make the test section of wood and the blades of Lucite. Dimensionally, the test section and the blades were identical to those used in reference 3. The center blade of this impermeable-wall blade cascade was instrumented with 15 static-pressure taps around the blade periphery to obtain the static-pressure distribution around the blade. Six static taps were placed in a wall of the entrance duct of the wood test section approximately $1\frac{1}{2}$ blade-chord lengths upstream of the cascade. A total pressure was obtained with a probe in the upstream plenum chamber to which the test section was attached. These total and static pressures were used to obtain the Mach number at the inlet to the cascade.

Experimental procedure. - The experiments were made at room temperature and consisted in setting a series of inlet Mach numbers which ranged from 0.05 to 0.52 and noting the local pressure distribution around the blade and the upstream total pressure at each Mach number.

ANALYSIS

Correlation Equations

A method for correlating the data for the flow of a gas through a plane wall of a porous material has been developed by Green (reference 4). The method was used successfully by Green and Duwez to correlate the data for nitrogen flowing through several porous disks (reference 5). The relation used was, in the symbol notation used herein,

$$\frac{p_{a,1}^{2} - p_{a,2}^{2}}{\tau} = 2\alpha R\, T_a \mu_a\, (\rho v)_a + \frac{2\beta R T_a}{g}\, (\rho v)_a^{2} \tag{1}$$

(Symbols used in this report are defined in the appendix.) In this equation $p_{a,1}$ is the static pressure of the gas upstream of the disk and $p_{a,2}$ is the pressure downstream of the disk. The weight flow per unit area $(\rho v)_a$ is obtained by dividing the weight rate of flow of air through the disk W_a by the cross-sectional area of the disk. No procedure was determined for correlating data obtained from specimens made of different materials or by different manufacturing methods. Values of α and β must therefore be determined individually for each specimen. For the case of viscous flow through the disk, the second term on the right hand side of equation (1) is negligible, whereas for higher flow rates this term becomes appreciable. In the correlation of all flow data in the present report, the second term was always considered.

Equation (1) can be rewritten for convenience in plotting data as follows:

$$\frac{(\rho v)_a}{\mu_a} = \left(\sqrt{\left(\frac{\alpha g}{2\beta}\right)^{2} + \frac{g}{2\beta R}\, \frac{p_{a,1}^{2} - p_{a,2}^{2}}{\mu_a^{2} \tau T_a}}\right) - \frac{\alpha g}{2\beta} \tag{2}$$

Now α, β, g, and R are constants for each specimen so that equation (2) can be represented by

$$\frac{(\rho v)_a}{\mu_a} = \left(\sqrt{C_1^{2} + C_2\, \frac{p_{a,1}^{2} - p_{a,2}^{2}}{\mu_a^{2} \tau T_a}}\right) - C_1 \tag{3}$$

Thus $(\rho v)_a/\mu_a$ is a function of $(p_{a,1}^2 - p_{a,2}^2)/\mu_a^2 \tau T_a$ and the flow data can be correlated on a general basis by plotting the first factor against the second.

Porous Disk

The comparison of experimental data with equation (1), which was made in reference 5, utilized data taken at a constant temperature. Similarly, the blade coolant-flow data used in the current investigation were obtained at room temperature. Because these data are intended to be used for calculating local cooling-air flow rates through the walls of a porous blade over a range of elevated temperatures, it was thought advisable to check whether equation (1) would correlate experimental data over that temperature range. Consequently, the apparatus discussed in a previous section of this report and shown in figure 1 was devised and the relation of pressure drop and air-flow rate through two porous disks was checked at room temperature and at elevated temperatures.

A correlation of the data from the disk calibration was attempted by plotting $(\rho v)_a/\mu_{a,d}$ against $(p_{a,1}^2 - p_{a,2}^2)/\mu_{a,d}^2\tau_d T_d$. The weight flow per unit area was determined by dividing the total air weight flow through the disk by the cross-sectional area of the pipe immediately downstream of the disk. The viscosity of the air $\mu_{a,d}$ was evaluated at the temperature T_d, which was measured with the thermocouple on the downstream surface of the disk. The value of τ_d that was used was the measured thickness of the disk. The disk surface temperature was used to evaluate the air viscosity because the tests were conducted under adiabatic and therefore isothermal conditions and the temperatures of the air and the disk were the same for all air temperatures that were encountered. The flow-calibration curves that were obtained at elevated temperatures were compared with the curves obtained at room temperature in order to determine how well equation (1) would correlate air-flow data which is obtained at different temperature levels.

Porous Blade

The flow data from the cold calibration of the porous blades reported herein were correlated by plotting the term $(\rho v)_{a,x}/\mu_{a,w,x}$ against $\dfrac{p_a^2 - p_{g,x}^2}{\mu_{a,w,x}^2 T_{w,x}\tau_x}$. The weight flow per unit area $(\rho v)_{a,x}$ at each location on the blade periphery was determined by dividing the weight rate of air flow passing through the sampling tube shown in figure 4 by the cross-sectional area of the tube in contact with the blade. The viscosity $\mu_{a,w,x}$ and the temperature $T_{w,x}$ were based on the thermocouple reading of the air inside the blade as the calibration was made at room-temperature conditions. The pressure p_a was the pressure inside the blade, and the pressure $p_{g,x}$ was room pressure. The thickness of the blade wall was used for τ_x at each position considered.

Evaluation of Porous-Blade Local Cooling-Air-Flow Distribution at Elevated Temperatures in Cascade

Before the local cooling-air flows in a cascade at elevated temperatures could be determined for the heat-transfer investigation of reference 3 it was necessary to know the local pressure drops across the blade wall, the average local cooling-air temperature that exists in the blade wall, and the wall thickness. The wall thickness was measured at each of the selected locations for which the cooling-air weight flow was to be determined.

The ratio of local static to total-inlet pressure is dependent on the inlet Mach number and the gas properties as long as the total pressure and the total temperature do not change through the cascade and separation does not occur. During the heat-transfer investigation discussed in reference 3 the range of temperatures was such that gas properties were nearly constant. As a result, the relation between inlet Mach number and the ratio of local static to inlet-total pressure, which are reported herein, were considered valid and were used to evaluate local static pressures at the higher temperatures encountered in that investigation. The inlet Mach numbers for that investigation were calculated from measured values of static pressure immediately ahead of the test section and total pressure in the plenum chamber. A value of 1.4 was used for the ratio of specific heats. Some error was introduced when the Mach number-pressure ratio relation was determined on an impermeable blade and the results were applied to a porous blade. Estimates based on the restriction of gas-flow area by the addition of cooling air indicate that, barring separation, the error was under 5 percent for the worst case. A measured value of pressure in the cooling passage within the porous blade was then used to calculate the pressure drop across the blade wall.

It was shown mathematically in reference 6 that because of the large area of the wall that was in contact with the cooling air in a porous material there was practically no difference in temperature between the cooling air and the blade wall. For this reason the wall temperature was assumed equal to the air temperature at any point within the wall and measured midwall temperature was used with the local pressure drop and measured wall thickness in determining the local cooling-air flows from the calibration curves. The calibration curves relate the coolant-flow term $(\rho v)_{a,x}/\mu_{a,w,x}$ and the blade-wall pressure-drop term $\dfrac{p_a^{\,2} - p_{g,x}^{\,2}}{\mu_{a,w,x}^{\,2}\, T_{w,x}\tau_x}$. The value of the blade-wall pressure-drop term was obtained using the values of the local static pressures $p_{g,x}$, which were determined on the impermeable wall blade in the static cascade, and the measured static pressure p_a inside the blade. The measured blade midwall temperature was used to determine the viscosity of the cooling air. The wall thickness was measured at the location which was being considered. When the value of the blade-wall pressure-drop term had been determined, the corresponding value of the coolant-flow term was determined using the calibration curve for the particular location. The local value of the cooling-air weight flow $(\rho v)_{a,x}$ was then determined by multiplying the coolant-flow term by the air viscosity.

RESULTS AND DISCUSSION

Porous Disk

Results of the hot- and cold-flow tests of the porous disks are given in figure 7. The data are presented in the manner described; the factor $(\rho v)_a/\mu_{a,d}$ is plotted against $(p_{a,1}^2 - p_{a,2}^2)/\mu_{a,d}^2 T_d \tau_d$. The use of these factors did not result in an exact correlation of the hot- and cold-flow data although the differences appear to be within the accuracy of measurement and thus the use of equation (1) at elevated temperatures appears justified.

An additional point is brought out by figure 7. Although the disks were made from the same batch of powdered metal by the same techniques and had almost identical porosities, the flow characteristics were different even though they are based on unit thickness. The difference in flow may be due to surface differences resulting chiefly from the coining operation during the manufacturing process.

Porous Blade

The cooling-air-flow calibration curves obtained from the porous-blade tests at room temperature at four locations are shown in figure 8. These curves include data from two groups of observations; the second group was taken to check the reproducibility of the earlier group. The two sets of data are indistinguishable. The scatter of points on the curves is indicative of the random errors of the measurements. There was no difference in the cooling-air flow through the blade at a given pressure difference across the wall at the locations on opposite sides of the blade near the leading edge. There was a small difference for a given pressure drop between the flows for the two locations near the trailing edge.

The cooling-air flow distributions around the entire blade for two pressure differences across the blade walls obtained from the cold-flow calibration are shown in figure 9. Curves have been faired through the points that indicate flow rates measured with the sampling tubes. The flow decreased rapidly at the leading and trailing edges. The observed variations through the middle part of the blade cannot be explained on the basis of wall thickness since the walls were slightly thicker at the midchord position, the point of highest flow, than they were at the locations shown in figure 8. The variations for these sections are consequently assumed to be due to differences in permeability resulting from manufacturing methods. The rapid decrease in flow rates to negligible values at the leading and trailing edges was due to the long flow paths and the small areas of the coolant passage at these locations, which must feed air to the large metal surfaces involved. Difficulties may be expected in manufacturing porous turbine blades that are aerodynamically acceptable and yet have sufficient air flow through the leading- and trailing-edge regions. Increased permeability in these regions may partly remove this drawback.

Impermeable Blade Pressure Distribution

Typical results of the investigation to determine the pressure distribution around the impermeable blade in the cascade are shown in figure 10. The ratios of the local static pressure at four positions on the blade to the total pressure in the plenum chamber are plotted against the Mach number at the cascade inlet. The pressures at positions g and i were the same for given conditions of $M_{g,i}$ and $p'_{g,i}$. Smooth curves could be drawn through the data for each position when correlated in this manner. As brought out previously, even though these data were obtained at room temperature, they should be applicable with reasonable accuracy to the temperature range of the heat-transfer investigations.

A typical pressure distribution around the porous blade for an inlet-gas Mach number of 0.45 is shown in figure 11. An assumed pressure in the coolant passage of the blade is also shown. With this constant coolant pressure, which is typical of that available in current jet engines, the local pressure drop available for forcing air through the porous wall varies widely around the blade. The cricital nature of the cooling problem at the leading edge of the blade is emphasized in figure 11. The external pressure is greatest at the leading edge, and the available cooling pressure drop is lowest as compared with any other point on the blade. Thus less air would be expected to flow through this region as compared with the flow through other parts of the blade. The long coolant path through the leading-edge portion of the blade would tend to reduce the air flow still more for a given pressure difference as compared with the flow through the thinner walls of the blade.

Local Coolant-Flow Distribution on Porous Blade in Hot-Gas Stream

The calculated effect of variations in pressure drop across the walls of the porous blade on the local cooling-air-flow distribution around the midspan of the blade is shown in figure 12 for two ratios of mean coolant weight flow to hot-gas weight flow $(\rho v)_{a,m}/(\rho v)_{g,m}$. With constant gas-flow conditions around the blade in the cascade, the internal coolant pressure was set to yield successively the two mean ratios $(\rho v)_{a,m}/(\rho v)_{g,m}$ of 0.0144 and 0.0548. The numerator of these ratios is the mean value of the curves of figure 12 and the denominator is the gas-weight-flow rate W_g divided by the smallest cross-sectional area between the blades. The local flows were calculated using curves like those of figures 8 and 11 by methods described previously.

The difference between the shapes of the curves of figures 9 and 12 is due to the fact that the gas pressure around the blade is constant in figure 9 and varying in figure 12. As a consequence, the effects of wall permeability and wall thickness on flow distribution are shown in figure 9, whereas the effect of varying pressure drop across the walls, the condition which would actually exist in a turbine, is included in figure 12. The flow through the pressure surface varies much more with pressure drop than does the flow through the suction surface. This is to be expected as a result of the pressure-squared difference term in the correlation equation. At a flow ratio of 0.0548, the peak value of $(\rho v)_{a,x}$ across the pressure surface is approximately 0.65 pound per second per square foot as compared with 0.9 across the suction surface. At a flow ratio of 0.0144, comparable peak values of $(\rho v)_{a,x}$ are seen to be 0.025 and 0.28. Thus, for a flow ratio of 0.0548, the flow across the suction surface is only approximately 1.4 times the flow across the pressure surface, whereas, for a flow ratio of 0.0144, the flow is over 10 times as much. This is the result to be expected from variations in cooling-air pressure-drop distribution like that shown in figure 11. It is evident that the magnitude of the internal coolant pressure will have a pronounced effect on such differences. The lower the cooling-air pressure inside the blade, the more the difference between coolant flows resulting from the pressure drops across the suction-surface wall and pressure-surface wall is accentuated, whereas, with a sufficiently high internal pressure, the difference may be substantially reduced. Unfortunately, the coolant pressures attainable by bleeding from compressors of current engines are limited.

In addition to variations in pressure around the blade caused by variations of turbine operating conditions resulting from changes in atmospheric conditions (altitude effects), engine speed, tail-pipe opening, and so forth; lengths of coolant-flow paths and possible variations in permeability occurring because of manufacturing techniques will cause changes in the flow distribution around the blade. A given blade will therefore be operating under varying pressure-drop conditions. All these factors must be considered so that the blade will have an adequate supply of cooling air for all conditions at all locations.

Some method of controlling flow distributions must be embodied in porous blades if they are to be used efficiently in gas turbines; in addition, some means of governing the over-all quantity is desirable. Variations in permeability, wall thickness, or path length, will give limited distribution control. With respect to cooling alone, a separate compressor to supply cooling air would be advantageous. The separate compressor would permit operation over a wide range of conditions with nearly the optimum amount of coolant at each condition and could furnish the necessary high internal-coolant pressure to offset the differences in external static pressures about the blade and to minimize the differences in pressure drop across the walls. A separate compressor would also insure an adequate supply of blade-cooling air during the starting period of the engine when gas temperatures are higher than normal. A cooling system fed from the compressor of the engine would require an appreciable time interval to build up to capacity and would also depend on the thermal inertia of the blades during the starting period.

SUMMARY OF RESULTS

A direct measurement of the coolant-flow distribution on a porous blade in a high-temperature gas stream presents great difficulties. Therefore the results of three separate experiments were combined to determine the coolant-flow characteristics of a porous turbine blade. These experiments consisted of the following: (1) the relation of pressure drop and air flow through two porous disks was checked at room temperature and at elevated temperature levels; (2) the local cooling-air-flow distribution through the walls of a porous blade was observed at room temperature with no flow past the blade for a range of imposed pressure drops; and (3) the variation in static pressure about the periphery of an impermeable blade of identical profile was determined in a cold-air stream over a range of Mach numbers. The results of these separate tests were correlated and combined to permit the determination of local flow rates through the walls of the porous blade in a gas stream. Typical flow data for the blade in a 1000° F gas stream at an inlet Mach number of 0.45 are presented.

The agreement of air flow and pressure drop through the porous disks at room temperature and at elevated temperatures was sufficiently accurate to warrant the use of room-temperature flow data from the porous blade for heat-transfer work at higher temperatures.

The variations in local coolant-flow rates about the blade periphery are shown for two widely different total coolant-flow rates with the blade in a gas stream. It is evident that the variations in flow-path lengths and in pressure drop in a porous turbine blade will be such that some method of controlling the flow distribution will be needed to provide more uniform cooling. Additional measurements will be required if leading- and trailing-edge temperatures must be held to levels comparable with the center portion of the blade periphery.

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APPENDIX

SYMBOLS

The following symbols are used in this report:

C_1,C_2	constants (see equations (2) and (3))
g	acceleration due to gravity, $ft/(sec)^2$
M	Mach number, $(v\ \sqrt{\gamma gRT})$
p	static pressure, lb/sq ft
p'	total pressure, lb/sq ft
R	gas constant, (53.3), ft-lb/(lb)(^{o}R)
T	static temperature, ^{o}R
v	velocity, ft/sec
W	flow rate, lb/sec
z	distance from leading edge to particular location on blade, ft
α	flow coefficient in equations (1) and (2), 1/sq ft
β	flow coefficient in equations (1) and (2), 1/ft
γ	ratio of specific heats
μ	viscosity, (lb)(sec)/sq ft
ρ	density, lb/cu ft
τ	thickness of blade wall or disk, ft

Subscripts:

a	air
d	disk or downstream disk surface
g	gas outside blade
i	inlet to cascade

m mean

w midwall

x local (refers to condition at peripheral distance x from leading edge in direction of gas flow)

1 upstream

2 downstream

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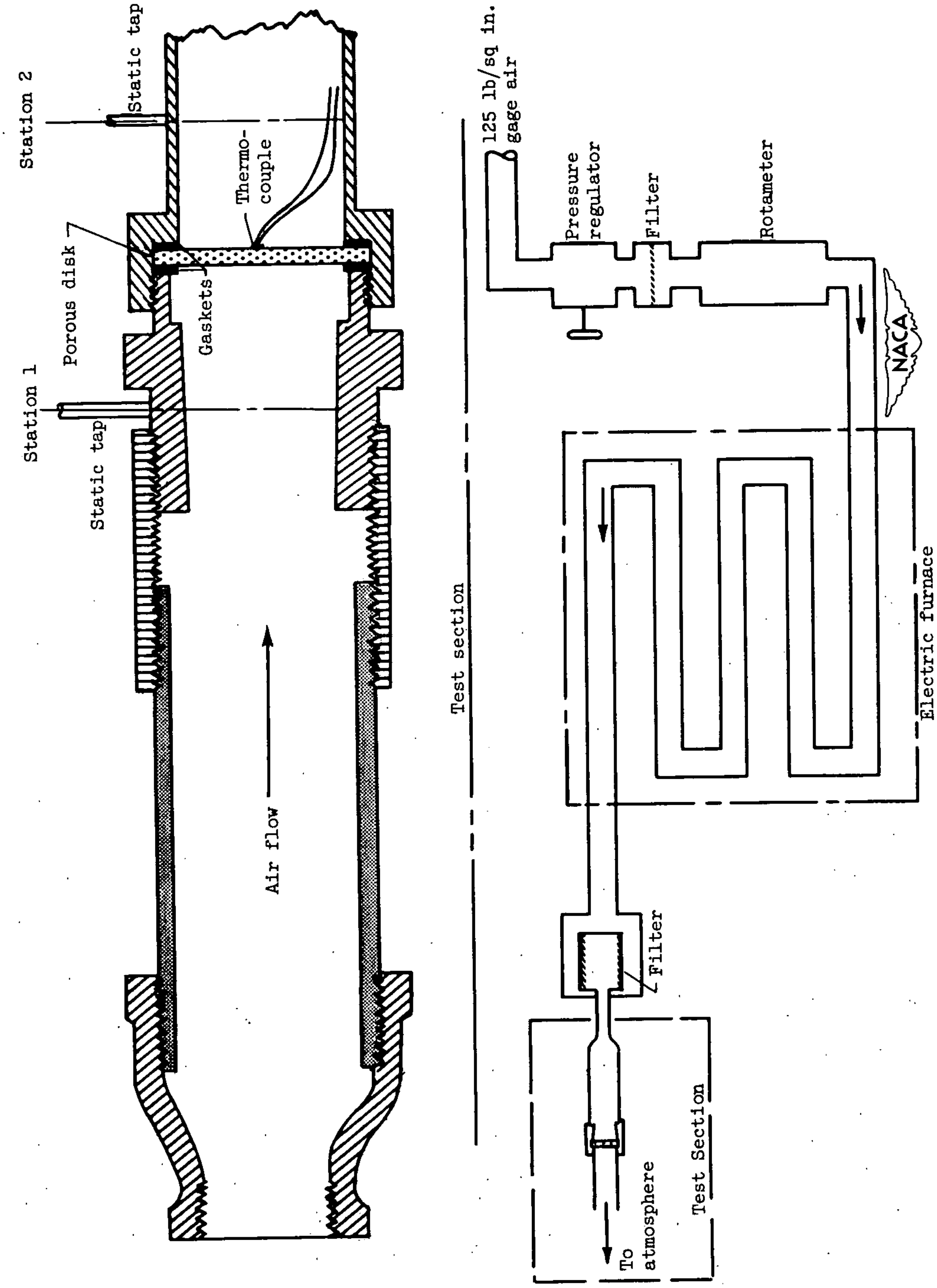


Figure 1. - Test setup for porous-disk flow experiments.

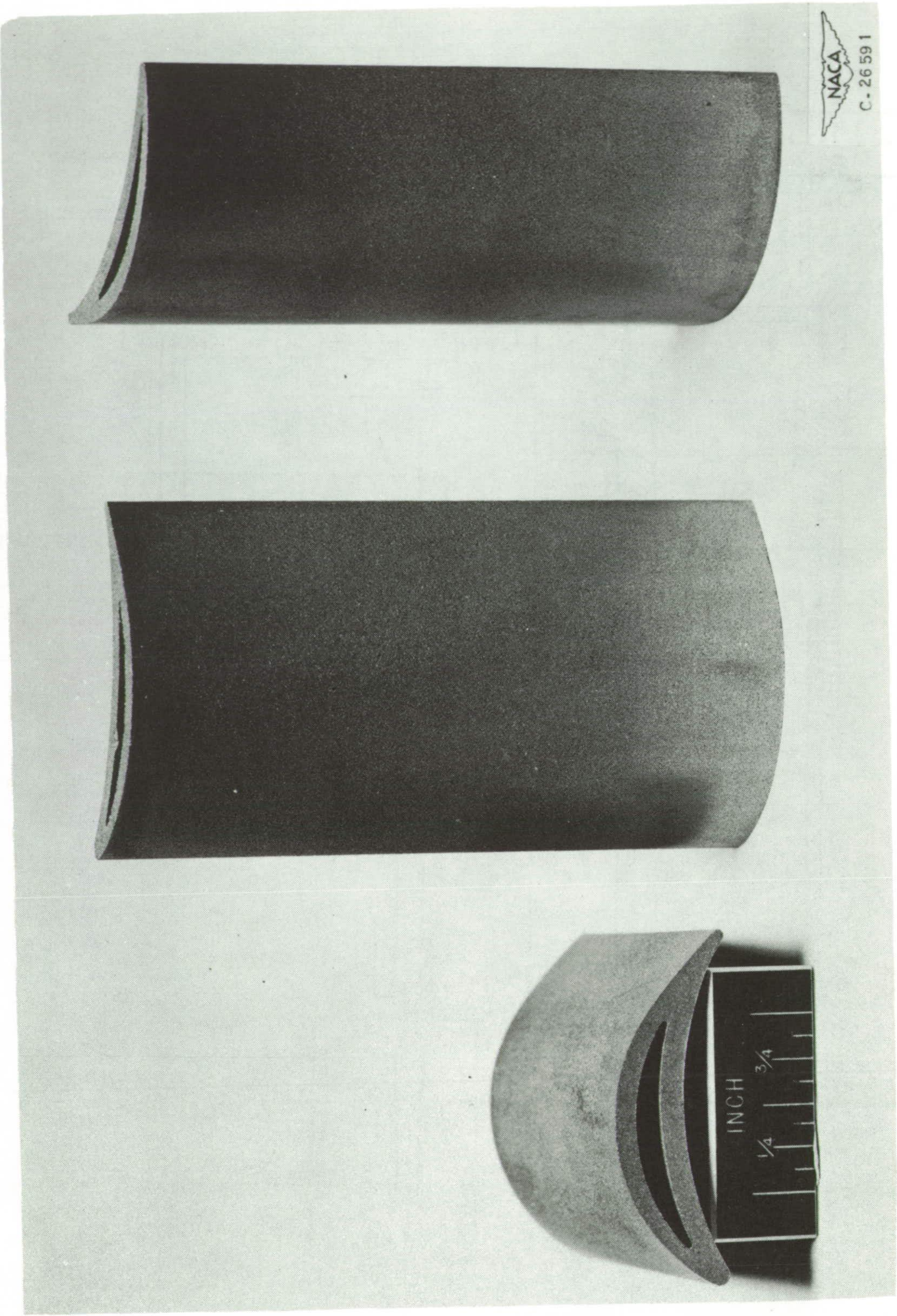
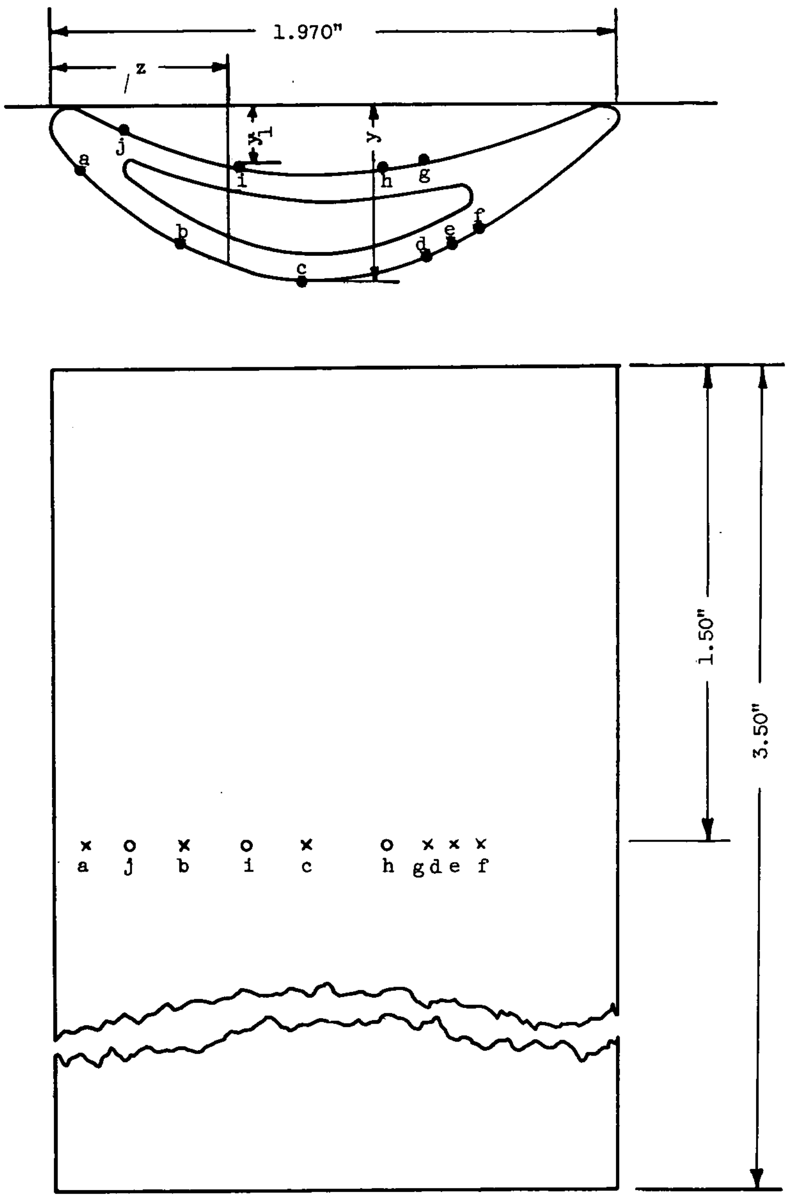


Figure 2. - Porous stainless-steel turbine blades.



Location of center line of sampling tubes

Tube	z (in.)
a	0.1
b	.464
c	.892
d	1.304
e	1.416
f	1.514
g	1.308
h	1.172
i	.654
j	.266

Blade coordinates

z	y	y_1
0	0.064	0.064
.050	.166	.002
.075	.202	.002
.100	.237	.014
.500	.548	.187
.700	.614	.235
1.000	.632	.260
1.200	.600	.252
1.500	.461	.195
1.800	.214	.082
1.900	.117	.019
1.950	.064	.002
1.970	.032	.032

Figure 3. - Blade dimensions and locations of areas where local cooling-air flow rates were measured.

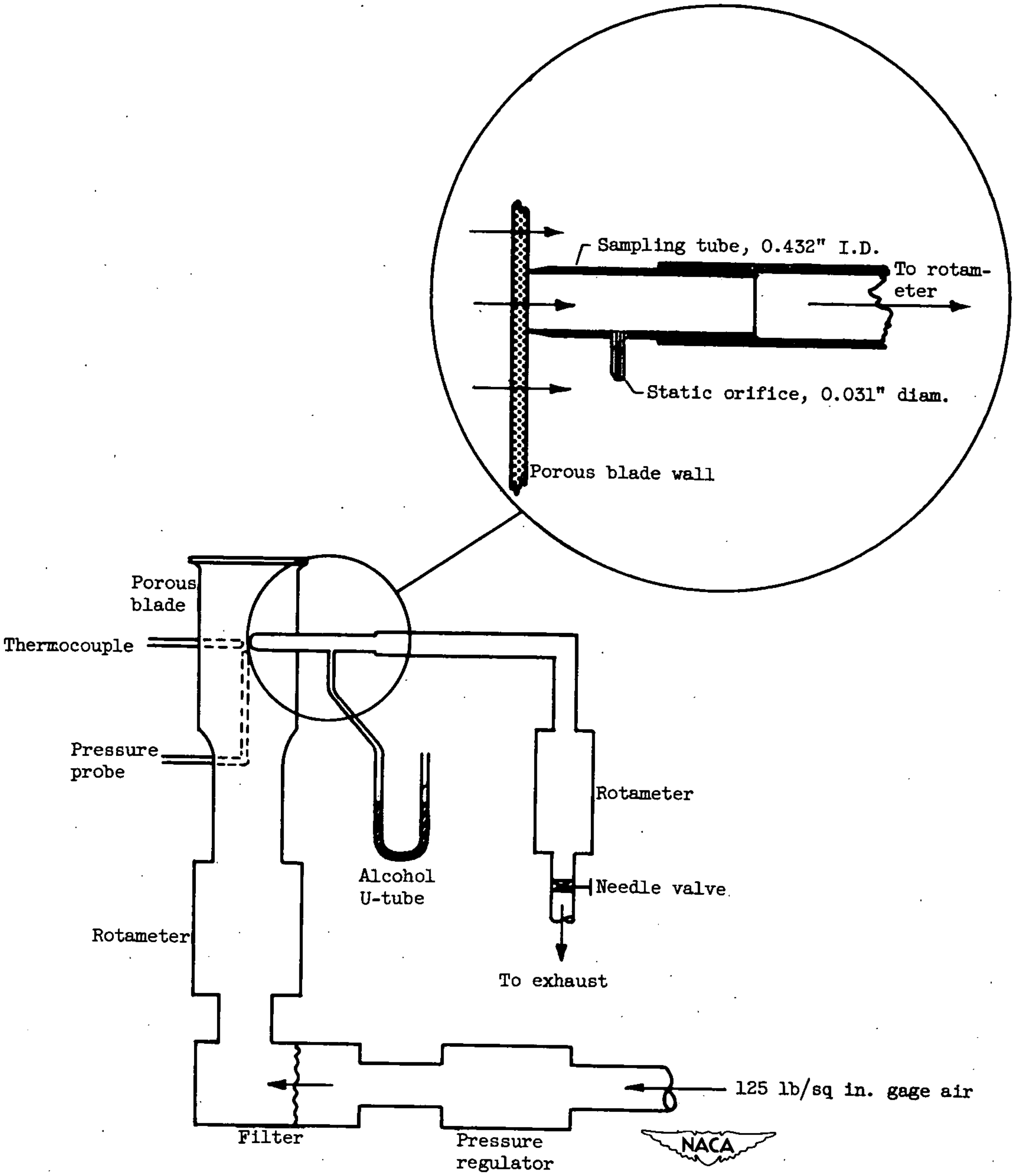


Figure 4. - Setup for measuring local cooling-air flow rates.

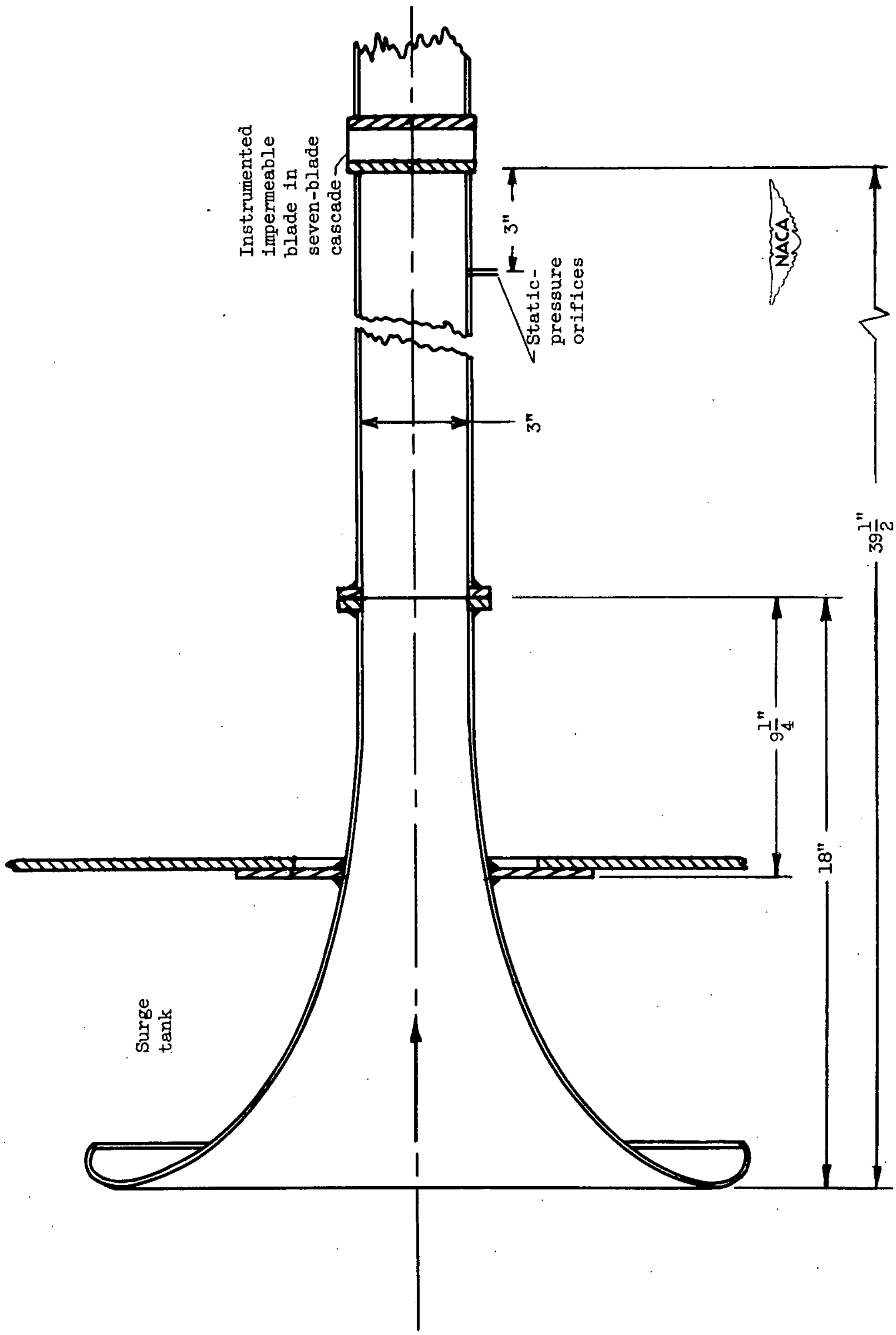


Figure 5. - Sectional view of inlet nozzle and approach section.

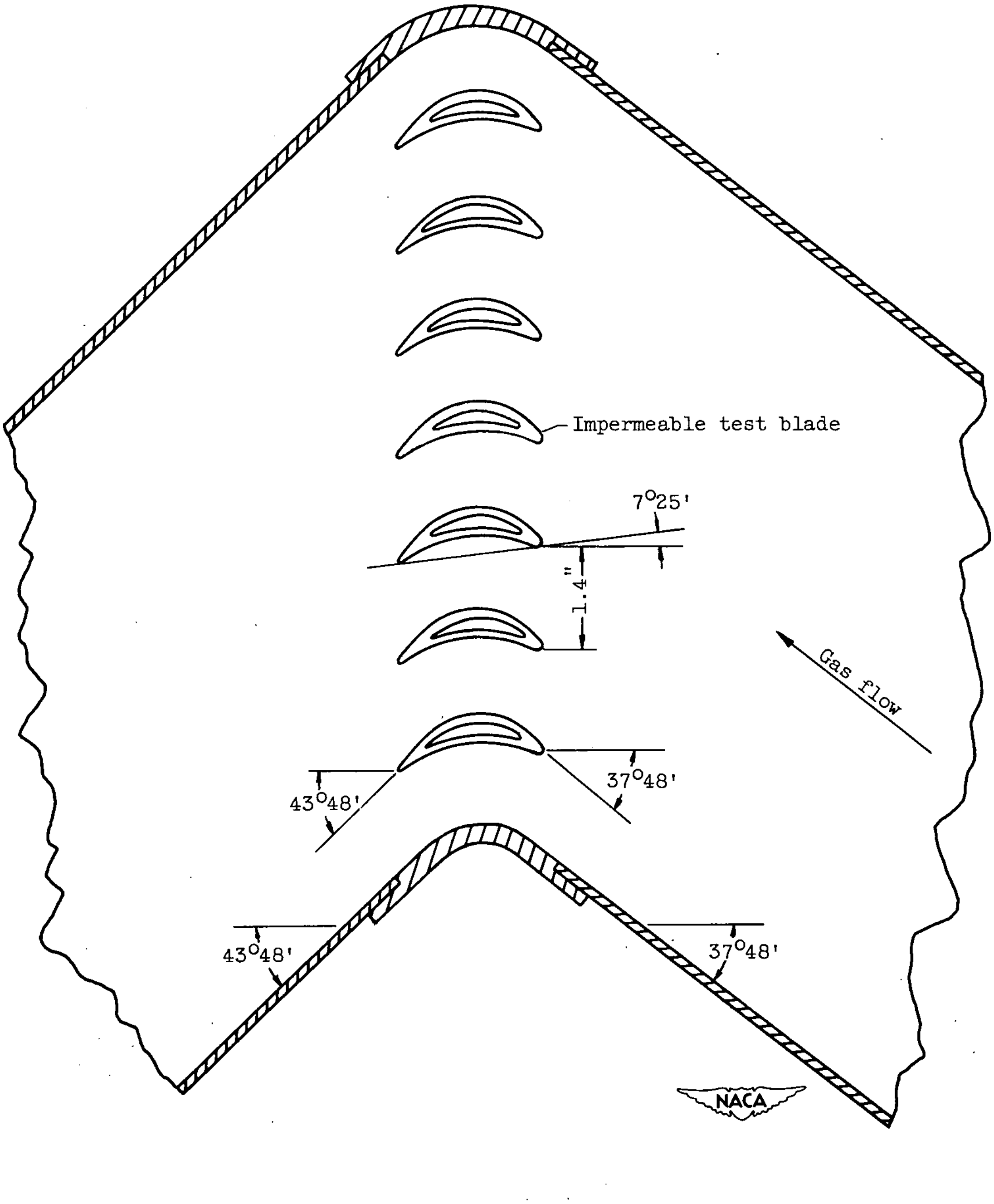


Figure 6. - Cascade geometry.

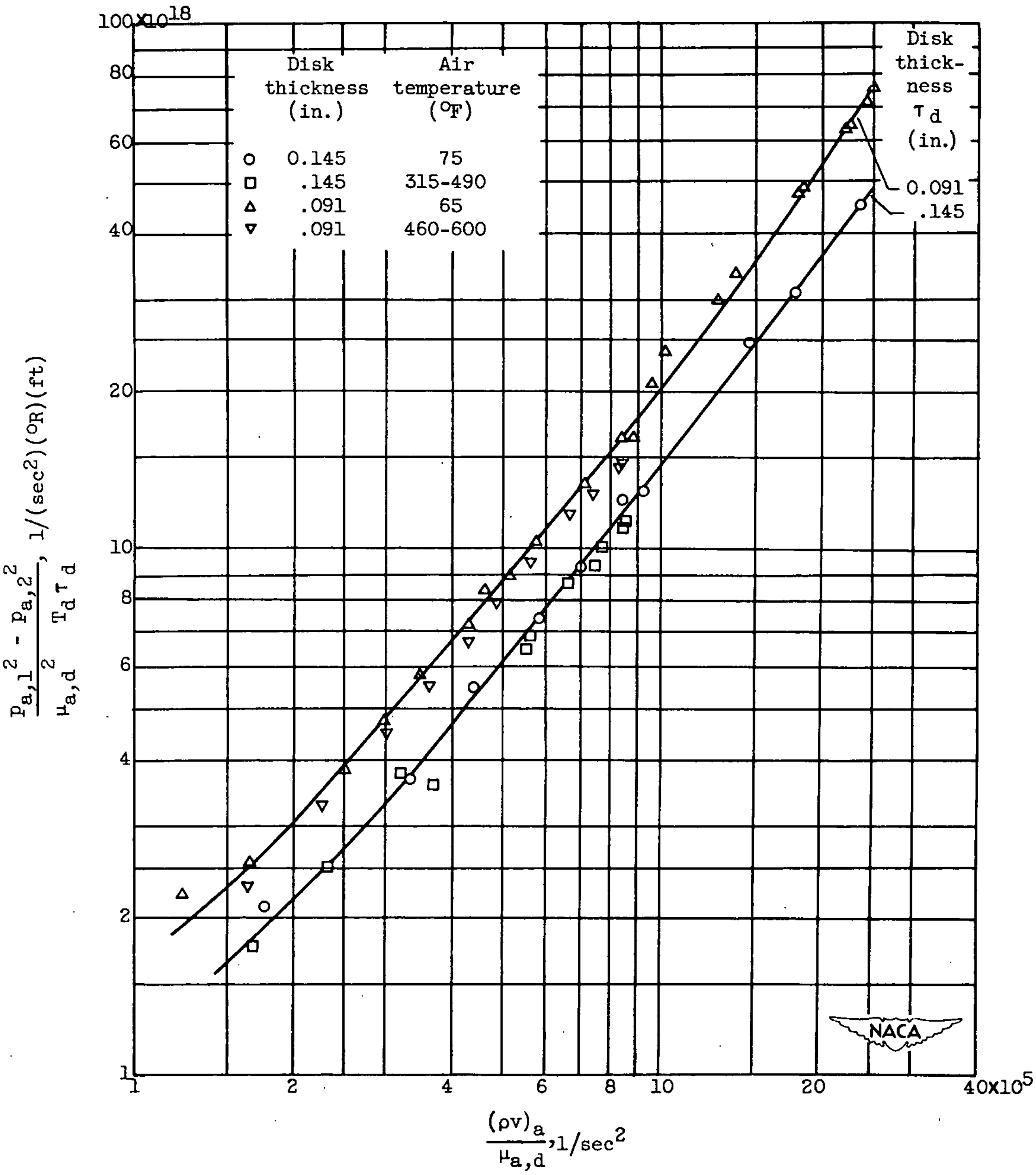


Figure 7. - Correlation of air flow through two porous disks with varying air temperatures.

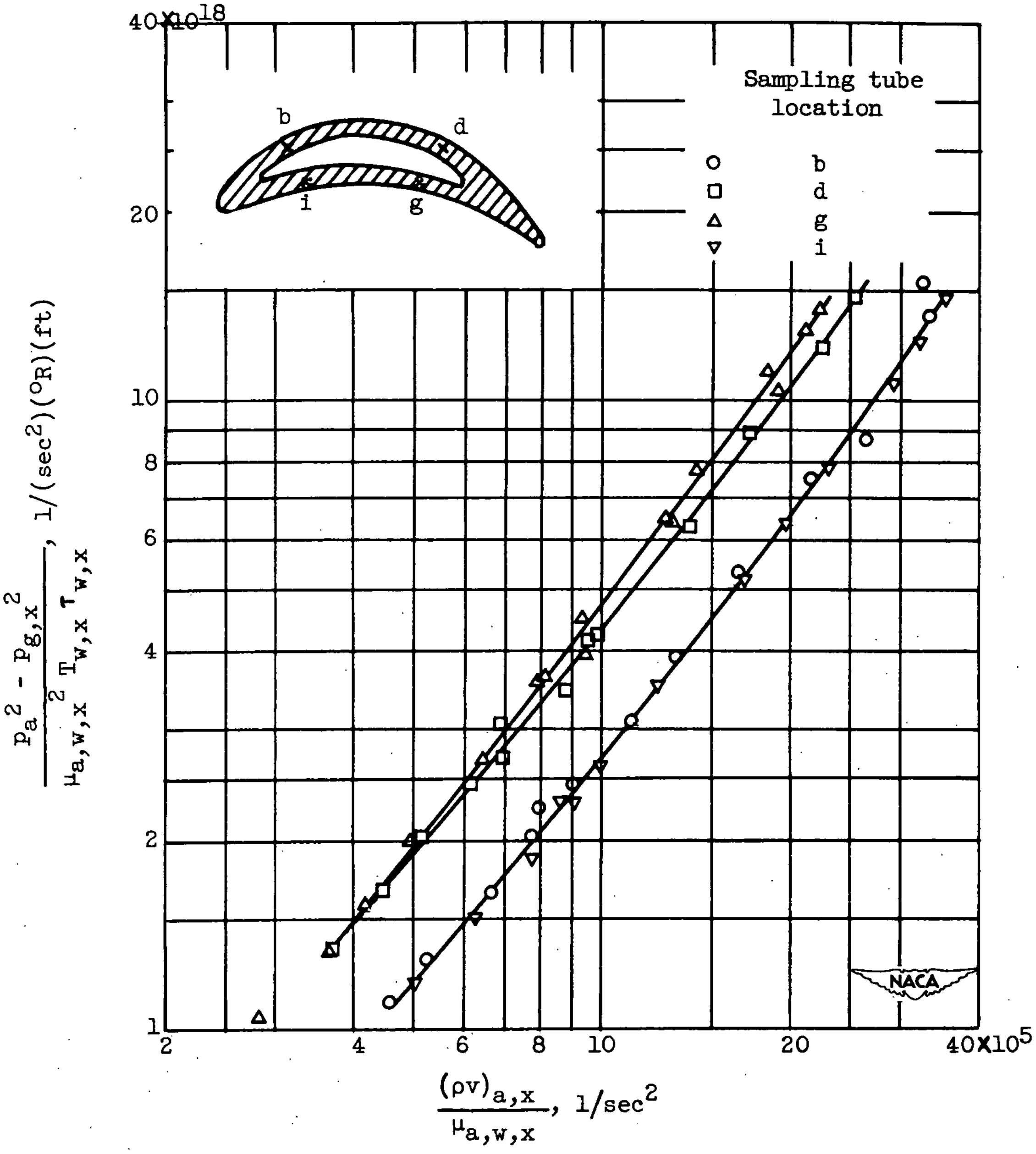


Figure 8. - Correlation of local cooling-air-flow data at four locations.

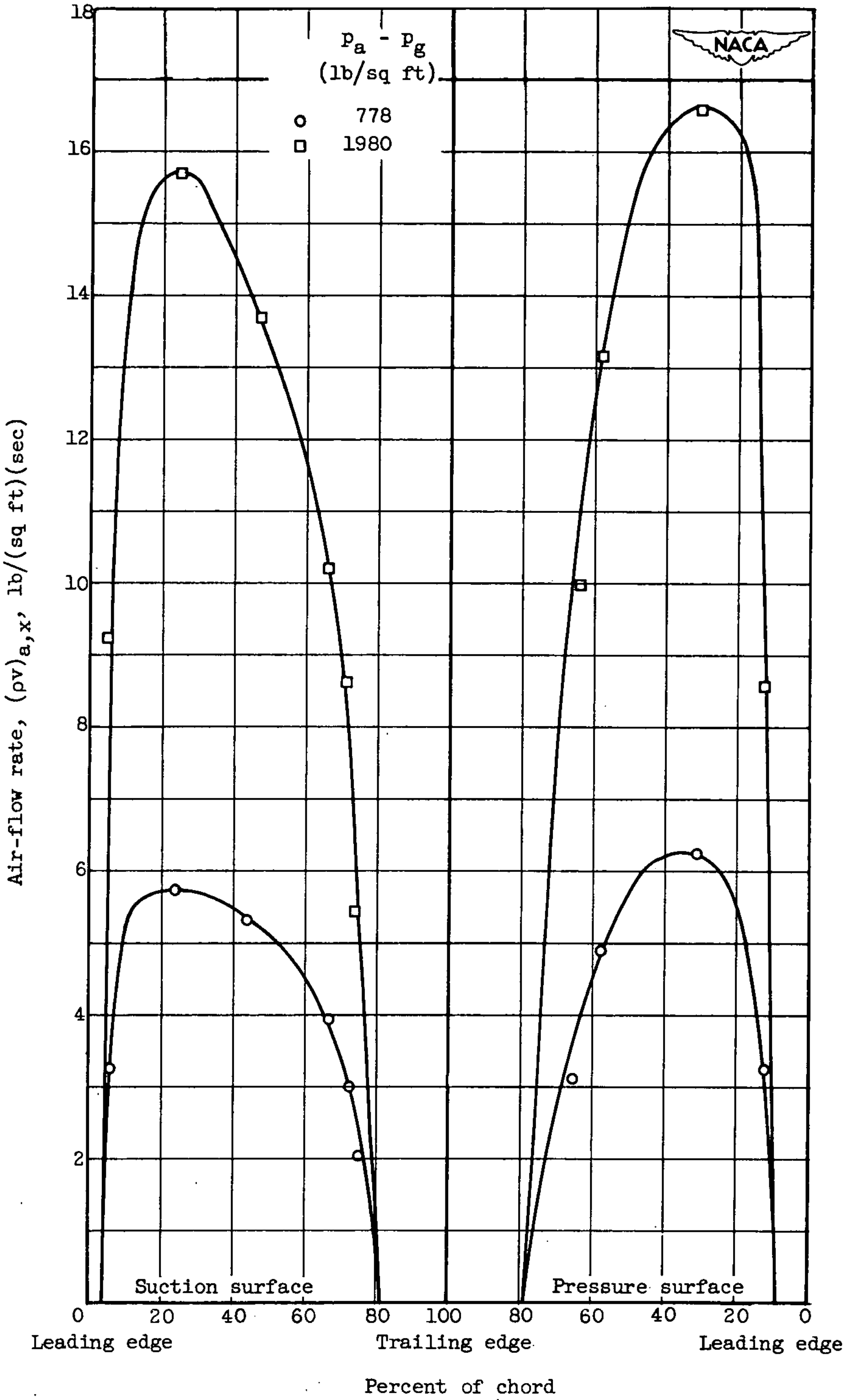


Figure 9. - Variation in coolant-flow rates about blade periphery with constant ambient pressure of 2116 pounds per square foot and coolant temperature of 80° F.

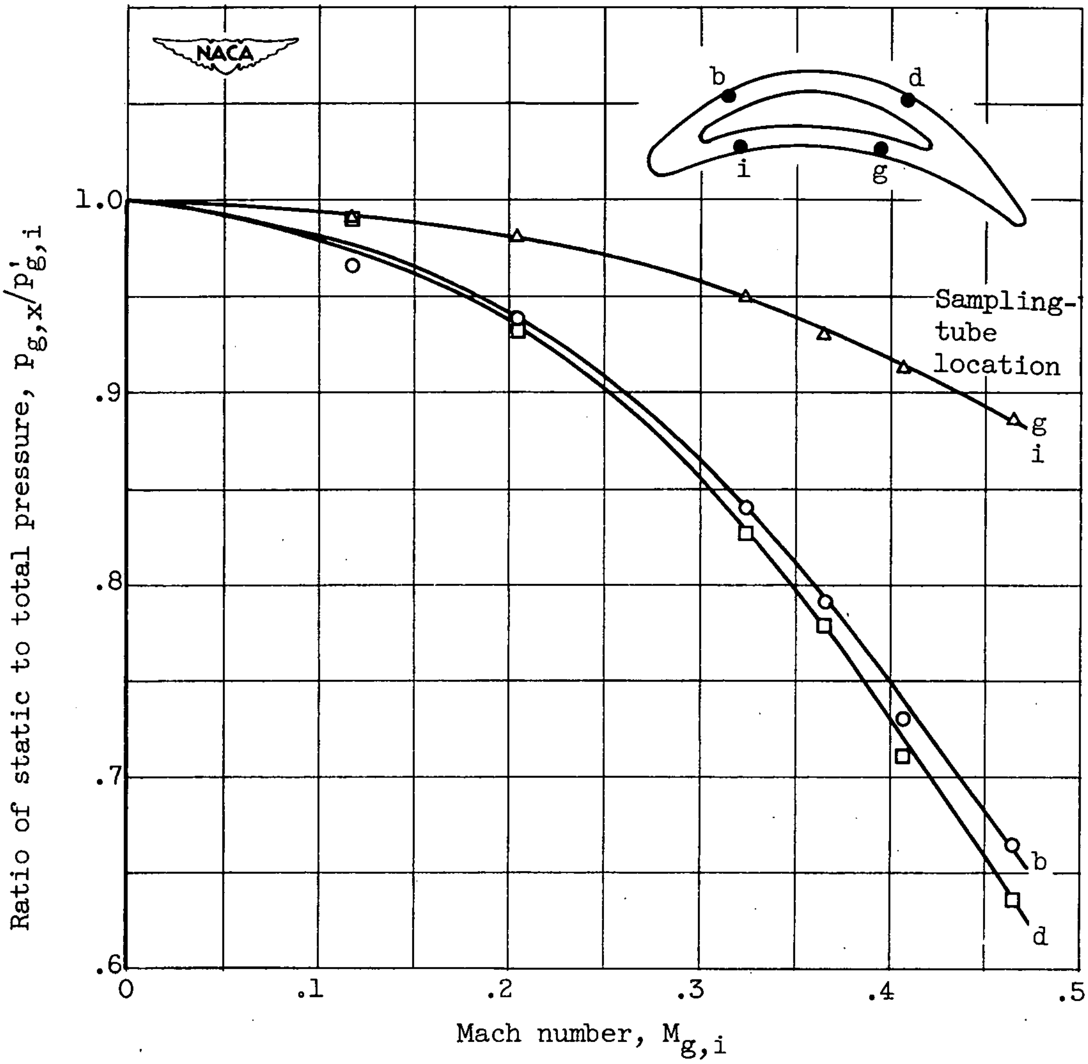


Figure 10. - Relation between inlet-gas Mach number and blade local static pressure.

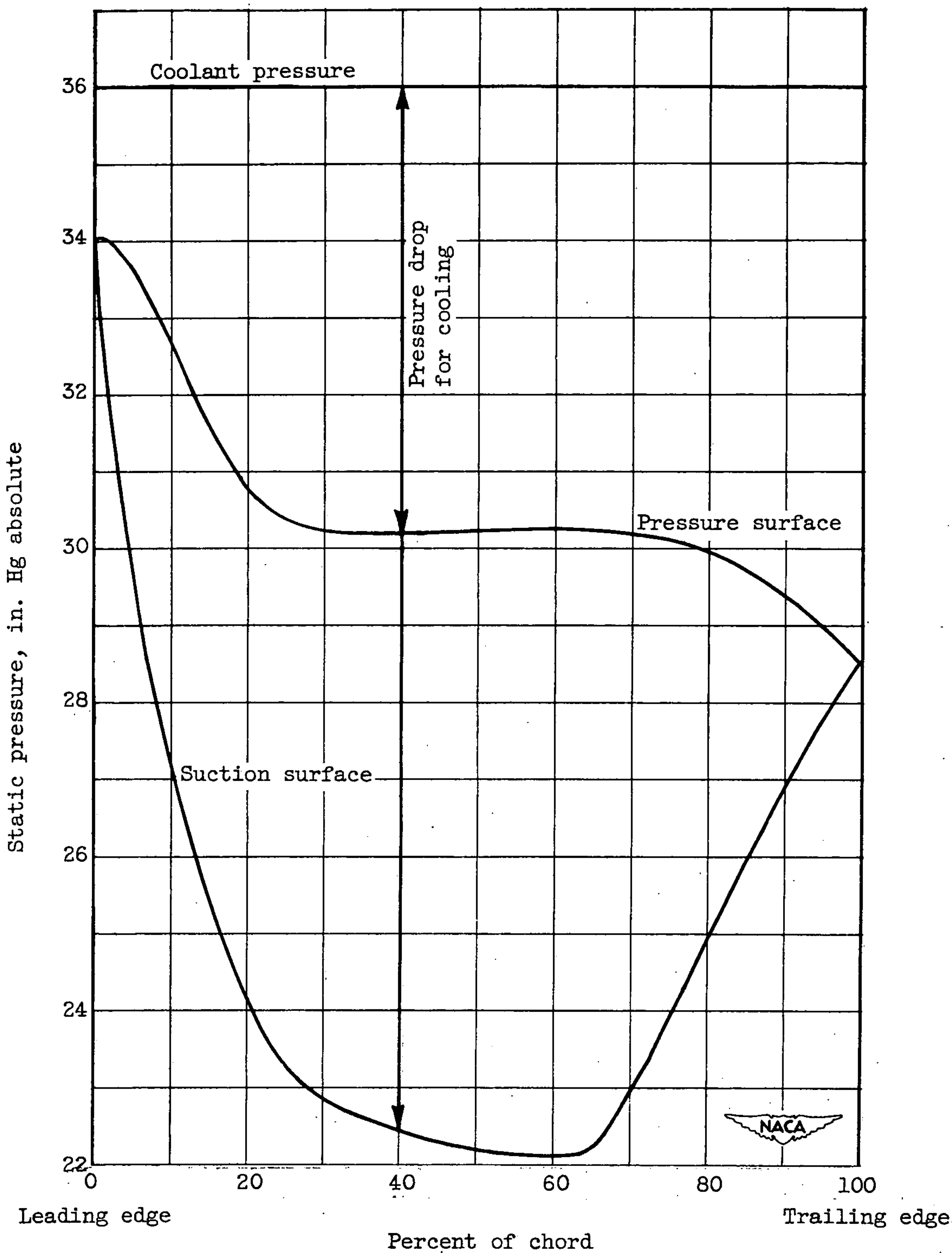


Figure 11. - Typical pressure distribution about porous blade showing variations in local pressure drop available for forcing coolant through walls. Inlet-gas Mach number, 0.45.

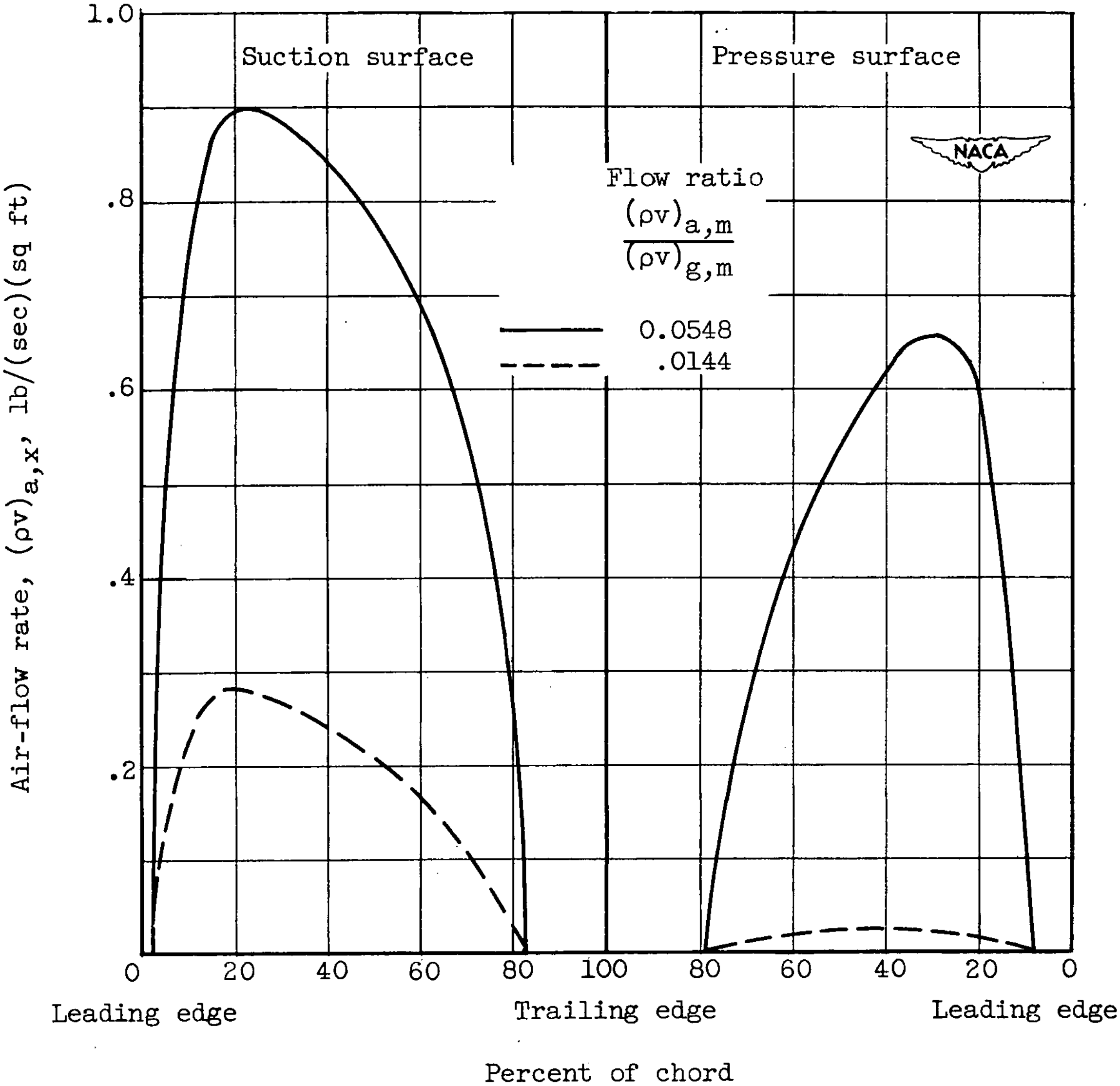


Figure 12. - Two typical coolant-flow distributions around blade perimeter in 1000° F gas stream at Mach number of 0.45. Gas-flow rate, 4 pounds per second; arithmetic mean gas flow, 24.05 pounds per second square foot.