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RESEARCH MEMORANDUM

EXPERIMENTAL INVESTIGATION OF SEVERAL WATER-INJECTION CONFIGURATIONS FOR TURBINE-BLADE SPRAY COOLING IN A TURBOJET ENGINE

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NATIONAL ADVISORY COMMITTEE FOR AERONAUTICS

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EXPERIMENTAL INVESTIGATION OF SEVERAL WATER-INJECTION CONFIGURATIONS FOR TURBINE-BLADE SPRAY COOLING IN A TURBOJET ENGINE

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SUMMARY

As part of an investigation of water spray cooling of turbine rotor blades with a representative centrifugal-flow turbojet engine, water injection from several stationary configurations was investigated in order to determine the most favorable configuration on an over-all basis.

Three general types of injection configuration were investigated at rated engine speed (11,500 rpm) and at turbine inlet-gas temperatures ranging from 1515° to 1683° F. The most favorable configuration with reference to general applicability, which includes factors such as cooling efficiency, configuration life, ease of fabrication, installation, and maintenance, as well as installation safety, consisted of four injection orifices located in the inner ring of the stator diaphragm near the stator blade trailing edge. Effective coolant dispersion over the rotor blade surface was achieved with this injection configuration by use of different orifice diameters. Cooling at the blade root was effected by water sprays from two 0.078-inch-diameter orifices, while the blade surface above the root to the midspan was cooled by sprays from two larger orifices, 0.135- or 0.150-inch diameter, depending upon the maximum flow rate desired. A chordwise temperature difference of 100° F between the leading and trailing edges at the blade midspan and an average blade midspan temperature of 292° F were obtained with this configuration at rated engine speed (11,500 rpm), approximately 1568° F turbine inlet-gas temperature, and a coolant-to-gas flow ratio of 0.0278 (7600 lb water/hr). Moving the axial location of this type injection configuration progressively upstream of the rotor blades resulted in less effective cooling.

The maximum chordwise temperature difference between leading and trailing edges always occurred near the lightly stressed blade tip. Attempts to cool the tip section by use of additional orifices located in the outer ring of the stator diaphragm were ineffective.

Rotor blades of S-816 material apparently afforded greater resistance to failure caused by the introduction of large chordwise temperature differences than blades of Stellite 21 material.

INTRODUCTION

External water spray cooling of gas-turbine blades is being investigated at the NACA Lewis laboratory as a means of permitting short periods of operation at increased thrust. The phase of the investigation reported herein is designed to determine the most favorable stationary water-injection configuration on an over-all basis of general applicability and to indicate possible limitations on the coolant-to-gas flow ratio requirements with this configuration at overspeed, overtemperature conditions.

In the spray cooling application, water spray impinges on the rotor blade surfaces and boiling occurs upon contact, thereby utilizing the latent heat of vaporization of the water to dissipate heat from the blades. For simplicity, water sprays were injected into the gas stream upstream of the rotor. Spray cooling is limited in aircraft turbine engine installations to short-time application since the water is not recovered and the supply available is fixed by space and weight considerations. Nevertheless, even short-time application is desirable for take-off, climb, or combat if large thrust increases can be realized by use of spray cooling.

The analysis of reference 1 indicates that large potential gains in thrust may be realized by overspeed, overtemperature operation. For example, at the sea-level static condition a 40-percent increase above rated thrust may be realized with a representative centrifugal-flow engine by operation at 2000° F inlet-gas temperature and 10-percent overspeed. This calculation assumes that the turbine blades are adequately spray-cooled to withstand overspeed, overtemperature operation at these conditions and that other engine components can withstand such operation as well. In view of the favorable analytical results, an experimental investigation of spray cooling was also initiated in reference 1 with a modified centrifugal-flow engine having blades made of Stellite 21 material. Three types of water-injection configuration were investigated over a range of engine speeds up to rated. All provided large chordwise temperature differences across the blade at the midspan position, which is detrimental from stress and consequently blade life considerations. The most favorable configuration investigated in reference 1 consisted of two spray orifices located in the inner ring of the stator diaphragm between adjacent stator blades approximately midway across the ring axially and approximately 180° apart. With this injection configuration, a temperature difference of 600° F between the blade midspan leading and trailing edges was achieved at rated engine speed and a coolant-to-gas flow ratio of 0.0089. Increasing the coolant-flow rate with this configuration resulted merely in lowering the temperature level without decreasing the blade chordwise temperature differences.

As a continuation of the work described in reference 1, an investigation of additional water-injection configurations was conducted at the NACA Lewis laboratory with the same centrifugal-flow engine. The investigation utilized rotor blades made of S-816 alloy in order to obtain experience with another blade material. The configurations investigated consisted of variations of the general type (inner ring of stator diaphragm) established as being the most favorable in reference 1, a spray bar, and a combination of inner-ring stator diaphragm orifices and orifices located in either the stator diaphragm outer ring or the tail-cone outer shell. The spray bar was located in the gas-flow passage between adjacent stator blades in a manner similar to that employed by other investigators (ref. 2, fig. 3).

Configurations of the first general type were investigated at various stations located progressively upstream of the turbine rotor. These variations in location were considered because both the water entrance angle relative to the rotor blades and the water evaporation rate could thus be altered. For example, the analysis of reference 1 indicates that high water velocities are required to achieve the proper water entrance angle to the rotor blades. Higher velocities can be achieved by permitting the gas stream to accelerate the water spray particles for a greater period of time. Thus, the farther upstream the injection station is located, the higher the resulting spray velocity at the rotor blade entrance. Evaporative losses encountered by the spray prior to striking the rotor blade surface are also a function of the injection station distance from the rotor blade. Consequently, several locations were considered.

All configurations were investigated at rated engine speed (11,500 rpm) and over a range of turbine inlet-gas temperatures from 1515° to 1683° F. The configuration proving to be most effective at this speed was checked at 4000 and 8000 rpm (turbine inlet-gas temperatures of 973° and 1001° F) as well. The range of coolant-to-gas flow ratio covered was from 0.0046 to 0.0279, equivalent to coolant flows of 360 and 7600 pounds per hour, respectively.

APPARATUS

A centrifugal-flow turbojet engine was modified to permit a continuation of the spray cooling investigation. The test installation is identical to that described in detail in reference 1 except for the following alterations.

Engine Modifications

Water-injection configurations. - The axial locations of the water-injection configurations in relation to various parts of the engine are shown in figure 1. Specific locations and types of configuration investigated are illustrated in figure 2. One general type of injection configuration consisted of orifices located in the inner ring of the stator diaphragm and the inner shell of the ring and tube assembly. As shown in figure 2, these orifices are located to direct the water spray into the gas-flow passage between adjacent stator blades at several axial stations upstream of the stator blade leading edge, at the blade leading edge, at the blade midchord, and at the blade trailing edge. Two 0.135-inch-diameter orifices, 180° apart, at each axial station comprised one injection configuration. Also, at the stator blade trailing-edge station, two 0.135-inch-diameter orifices and two 0.078-inch-diameter orifices alternately spaced 90° apart comprised one injection configuration. During the final phase of the investigation, the 0.135-inch-diameter orifices at the stator blade trailing edge were increased to 0.150-inch diameter. As a means of simplification, an abbreviated nomenclature will be used for this general type of injection configuration. A tabular presentation of the nomenclature employed to designate the various types of injection configuration investigated is presented in table I. The general location of the orifices will be designated by ID, meaning inside diameter, and the specific axial location by number, beginning with 1 at the position upstream of the stator blade, 2 at the stator blade leading edge, and so forth. Finally, the orifice size will be designated by L', L, or S, where L' corresponds to a 0.150-inch diameter, L corresponds to a 0.135-inch diameter, and S corresponds to a 0.078-inch diameter. Thus, the abbreviated nomenclature for an injection configuration in the inner ring of the stator diaphragm at the blade trailing edge consisting of two 0.135-inch- and two 0.078-inch-diameter orifices is ID-4(L) and (S).

Another general type of injection configuration investigated consisted of two spray bars inserted into the gas-flow passage between adjacent stator blades at the blade midchord position 180° apart. The spray bars were constructed of 3/8-inch-outside-diameter Inconel tubing. Four 0.067-inch-diameter orifices directed downstream were provided in each spray bar which extended from the outer ring of the stator diaphragm to approximately 1/4 inch from the inner ring. Orifices in the spray bar were spaced 3/4 inch apart with the one nearest the inner ring 1/2 inch from the end of the tube. Since the location of this configuration was not varied, an abbreviated nomenclature was not designated for this general type.

The third general type of injection configuration investigated consisted of a combination of orifices provided in the inner ring of the stator diaphragm and in either the outer ring of the stator diaphragm or the outer shell of the tail cone (fig. 2). The outer ring orifices were all 0.078 inch in diameter and were located 180° apart at two axial stations, near the stator blade midchord and downstream of the stator blade trailing edge immediately opposite the rotor blade tips. The abbreviated nomenclature for outer ring orifices is similar to that described for the inner ring orifices with the letters OD representing outside diameter and a number designating the axial station.

General installation details. - A complete set of standard rotor blades made of S-816 material was provided. Cooling air was introduced through the tail cone in the manner described in reference 1. However, it was directed only against the hub on the rear face of the turbine rotor in order to maintain the thermocouple junctions located there below the melting point of the insulation employed. City water was again utilized as the cooling spray medium. The water supply system was identical to that described in reference 1.

Instrumentation

General engine instrumentation and that required for measuring the cooling air, which was used in this investigation solely as a means of cooling the thermocouple junctions on the rotor hub, are described in detail in reference 3.

Water coolant measurements. - The water-flow rates were measured by calibrated rotameters and the injection pressures were indicated by Bourdon gages. Water-supply temperature was measured by a thermocouple in the manifold which supplied the water lines leading to individual injection stations on the engine. The water-injection pressure for all orifices was measured in the manifold.

Rotor blade instrumentation. - A total of 22 thermocouples was distributed in 17 positions on eight turbine rotor blades. The thermocouple positions are shown in figure 3. A slip-ring thermocouple pickup similar to that reported in reference 3 was used to achieve transition from the rotating thermocouples to the stationary measuring potentiometer.

CALCULATIONS AND PROCEDURE

Spray cooling efficiency calculations. - The spray cooling process efficiency was derived in reference 1 and is expressed by the following equation:

$$\sigma = 1.024\times10^{-6}\left(\frac{T_{g,e}^{0.75}}{T_{B,av}^{0.425}}\right)\left(\frac{T_{g,e} - T_{B,av}}{\frac{w_W}{w_g^{0.75}}}\right) \qquad (1)$$

(All symbols are defined in the appendix, and the term $\left(\frac{w_w}{w_g^{0.75}}\right)$ will hereinafter be designated the coolant-flow parameter.) Equation (1) was evaluated for each water-injection configuration at every condition of gas temperature and coolant-flow investigated. The average blade temperature $T_{B,av}$ was taken as the arithmetic average of the integrated averages of the root and midspan blade temperatures. This represents a departure from the procedure of reference 1 in which the integrated average around the blade midspan was considered to be the average blade temperature. As a consequence of this change, neither direct quantitative comparisons with the efficiencies shown in reference 1 nor comparisons between coolant-flow requirements calculated using the differently evaluated spray cooling process efficiencies should be made. However, the coolant flows and resulting temperature distributions may be compared directly with those obtained in reference 1. The reason for including the blade root in evaluating $T_{B,av}$ is that marked differences occur in the temperature of the highly stressed blade root with the various injection configurations evaluated in the present investigation. In the initial investigation described in reference 1, these temperature differences were not apparent because of instrumentation difficulties. The reason for not including the blade tip temperature was twofold: first, since the region near the blade tip is not highly stressed, there appears to be less need for cooling this region; therefore, the tip temperature should not influence an evaluation of the cooling efficiency. Also, thermocouples could not readily be maintained at the blade tip so that a complete temperature distribution in this region was not available for many runs. The effective gas temperature $T_{g,e}$ (uncooled blade temperature) was considered to be the integrated average blade temperature around the blade midspan for a condition of zero coolant flow.

Equation (1) was also applied to estimate the coolant-flow rate required at engine operating conditions above rated with the most favorable water-injection configuration. This calculation was made for an operating condition of 2000° F inlet-gas temperature and 10 percent overspeed, which probably represents the maximum overspeed, overtemperature conditions obtainable without major modifications of this engine. The calculation procedure was similar to that described in reference 1. The allowable blade temperature was substituted for $T_{B,av}$ and an effective gas temperature $T_{g,e}$ equivalent to an inlet-gas temperature of 2000° F was substituted in equation (1) to provide the required coolant-to-gas flow ratio. The value of allowable blade temperature specified must be consistent with material strength properties as well as satisfactory with respect to cooling. Since the term $T_{B,av}$ in equation (1) represents an average blade temperature and experimental evidence indicates that chordwise temperature differences exist with spray cooling, a low value of $T_{B,av}$ should be specified in order to compensate for these temperature differences. However, yield strength or stress-to-rupture properties which are customarily used in determining the allowable blade temperature employed in cooled blade design must also be considered in specifying $T_{B,av}$. For example, with S-816 material, which is employed in the current investigation, the yield strength remains practically constant between 1200° F and room temperature. Therefore, the selection of an average blade temperature less than 1200° F for substitution into equation (1) when S-816 material is considered would provide no increase in margin between yield strength and blade centrifugal stress. The average blade temperature of 1200° F does not compensate for large chordwise temperature differences; consequently, these calculations indicate a theoretical minimum coolant flow. For other materials, such as Stellite 21, it is advantageous to specify a lower average blade temperature since the material properties show a marked increase in strength as the material temperature approaches room temperature. Thus, a greater margin of safety between the strength of the material and the blade centrifugal stress is achieved, which tends to compensate for any chordwise temperature differences that may exist and results in a more realistic calculated value of coolant flow. Effective gas temperature $T_{g,e}$ was determined in the manner described in reference 1 from the assumed value of inlet-gas temperature. Because no data are available for the spray cooling process efficiency at engine overspeed, the value of spray cooling efficiency used was chosen from those obtained during engine operation at rated speed with the water-injection configuration under consideration.

Engine operation. - The complete range of engine operating conditions covered in the investigation of various water-injection configurations is presented in table II. All configurations were investigated at rated speed (11,500 rpm) and over a range of turbine inlet-gas temperatures from 1515° to 1683° F. The range of effective gas temperatures was from 1362° to 1521° F, as indicated in table II. The configuration consisting of stator diaphragm inner-ring orifices (ID-4(L) and (S)) was also investigated at two lower engine speeds (4000 and 8000 rpm). For each configuration, the water-flow rate was varied by manually adjusting the water supply pressure to provide a range of coolant-to-gas flow ratios as indicated in table II. In each case, a complete set of data was also obtained at zero water flow. As in the investigation described in reference 1, water sprays were turned on simultaneously with the engine starts to minimize thermal shock conditions in the blades, and the engine was shut down for blade inspection after a range of water-flow rates from maximum to zero flow had been investigated with a particular configuration. The adjustable exhaust nozzle was maintained at a fully open position during engine operation at all speeds except for one run at rated speed with the injection configuration consisting of stator diaphragm inner-ring orifices (ID-4(L) and (S)). The exhaust-nozzle position was partially closed at this condition in order to increase the turbine inlet-gas temperature.

RESULTS AND DISCUSSION

In this section are presented blade temperature distributions obtained with the configurations investigated, the cooling efficiencies of these configurations, the rotor blade failures encountered, the rotor blade spray patterns, and a comparison of the injection configurations on the basis of general applicability.

Blade Temperature Distributions

Blade temperature data were obtained for every water-injection configuration investigated and at each coolant flow considered. Because of the resulting large number of temperature distributions (both spanwise and chordwise) and because the maximum coolant flow provides the best cooling at the critical blade midspan position, only those distributions obtained at maximum or near maximum coolant flow with each configuration are presented. The uncooled temperature distribution is also shown in each case to provide a basis for comparison. Thus the data presentation is considerably simplified and comparison between injection configurations is facilitated. Temperature distributions obtained at similar coolant-to-gas flow ratios and rated engine speed with most of the configurations investigated are presented first. Next, temperature distributions obtained at several engine speeds with the most favorable configuration with respect to cooling are presented. Finally, the chordwise temperature distribution achieved with the most favorable configuration at rated speed with the maximum coolant flow permitted by the installation is presented. Temperature distributions obtained with configurations consisting of a combination of inside diameter and outside diameter orifices are not shown because of the extremely unsatisfactory results obtained.

<u>Spanwise temperature distributions</u>. - In figure 4(a) are presented the spanwise temperature distributions obtained at rated speed with a spray bar and with inside diameter injection configurations located at each axial station under consideration. Temperatures obtained on the blade suction surface at the midchord position are plotted against three spanwise stations - root, midspan, and tip. The suction surface midchord position was chosen for these plots to permit direct comparison with the data presented in reference 1. The coolant-to-gas flow ratio for each configuration considered in the figure varies slightly. This is primarily a result of engine mass-flow variations caused by day-to-day weather changes, since the coolant-flow rate

settings were generally the same for each configuration investigated. Figure 4(a) indicates that only one configuration (ID-4(L) and (S)) gives a lower temperature (200° F) at the highly stressed blade root than at the other spanwise stations. This lower temperature apparently is due to the two smaller diameter orifices which provide additional cooling at the blade root. Configuration ID-4(L), which was identical with configuration ID-4(L) and (S) except for the smaller orifices, had a blade root temperature much higher than the midspan temperature for a similar coolant-to-gas flow ratio. As the axial location of the injection configuration was moved upstream, the blade root temperature steadily increased until it equaled the uncooled blade root temperature (1280° F) with configuration ID-1(L), and a trend of decreasing blade tip temperature up to station 2 was observed. The more effective cooling at the blade tip may be due to the radial displacement of liquid particles in the centrifugal force field. Increased evaporation of the liquid particles, which probably occurs as the injection station is moved upstream of the rotor, would seem to limit this trend. In view of the generally less favorable cooling encountered as the injection station was located farther upstream of the rotor, it appears that any advantages accruing from acceleration of the water spray for a greater period of time by the gas stream are subordinate to the evaporative losses encountered. Further inspection of the figure indicates the existence of a reversal in temperature gradient along the blade span for all configurations except ID-4(L) and (S), ID-2(L), and ID-1(L). Such a reversal in gradient may be detrimental with regard to stress. Thus, on the basis of the foregoing, configuration ID-4(L) and (S) appears to afford more effective coolant dispersion over the blade surface than the other inside diameter configurations or the spray bar, for it provides more adequate cooling at the highly stressed blade root and midspan sections and does not produce a reversal in spanwise blade temperature gradient.

CI-2

Comparison of the results obtained with configuration ID-3(L) with the data of reference 1 (fig. 11(a)), for a similar configuration, indicates that much higher blade root temperatures occurred even though the coolant-to-gas flow ratio was twice as great as that shown in reference 1. This is not a contradiction of earlier results; it merely indicates that as a result of the increased injection pressures which necessarily accompany high coolant-flow rates for this configuration, the spray jet penetrates the gas stream farther, thereby tending to bypass the blade root and cool it less effectively. Operation with an identical configuration in the present investigation at a similarly low coolant-to-gas flow ratio indicated an identically low blade root temperature.

Chordwise temperature distributions. - The injection configurations considered in the discussion of figure 4(a) are compared on the basis of chordwise temperature distributions obtained at rated engine speed in figure 4(b). Chordwise temperature distributions around the blade root and midspan for the spray cooled condition are shown, and the midspan chordwise temperature distribution is also presented for the uncooled condition. Blade tip chordwise temperature distributions are not presented because none of the configurations considered cooled the blade tip effectively. Large temperature differences between the leading and trailing edges or between the midchord and trailing edge usually occurred at the blade tip. Because this section of the blade is not highly stressed and no experimental blade failures have occurred at the tip, such temperature differences are not considered serious. The number of thermocouple locations for which blade temperatures are plotted in figure 4(b) does not always coincide with the total number of blade thermocouple locations indicated in figure 3 because of thermocouple failures during engine operation.

A general trend of increasing temperature around the blade root as the inside-diameter-type injection configuration with two orifices is located increasingly farther upstream of the rotor is apparent from figure 4(b). Also apparent is the higher temperature level around the blade root in comparison with the blade midspan. Most of the configurations provided large chordwise temperature differences between the leading and trailing edges, either at the blade root or midspan. Such temperature differences are undesirable because of the additional stresses introduced in the blades. For example, the blade may be considered as a beam fixed at one end and free to expand in a spanwise direction. At any spanwise station a constant chordwise temperature distribution will result in equal expansion over the entire blade chord. Existence of a chordwise temperature difference, however, will tend to make the portion of the blade chord having a higher temperature expand more than the cooler portion. Since the cooler portion of the blade restrains the higher-temperature portion, the former section is in tension and the latter in compression so that a shear stress is imposed. When this condition occurs within the normally highly stressed regions of the blade (from blade root to blade midspan), the added stress may cause blade failure or seriously affect blade life. Thus, in addition to reducing the blade temperature level at the blade root and midspan sections, effective spray cooling must also minimize chordwise temperature differences at these sections. Of the configurations compared in figure 4(b), ID-4(L) and (S) provided the smallest chordwise temperature difference (approximately 100° F between the blade leading and trailing edges) as well as the lowest level of blade root temperatures (approximately 200° F). The low temperature level is undesirable in that it represents a degree of overcooling. Effective reduction of large chordwise temperature gradients, however, apparently requires such a degree of cooling with the configurations considered. Configuration ID-4(L) and (S) also provided a lower chordwise temperature difference at the midspan and a lower temperature level than any other configuration except ID-4(L) and ID-3(L). The midspan chordwise temperature distribution with configuration ID-4(L) and (S) was improved by operating at higher coolant flows, as will be shown. On the other hand, operation at higher coolant-to-gas flow ratios did not appreciably lower the relatively high root chordwise temperature differences of configurations ID-3(L) and ID-4(L) for the reasons pointed out previously. On this basis, it again seems that configuration ID-4(L) and (S) is the most favorable with respect to cooling.

Temperature distributions with most favorable configuration type at several speeds. - The spanwise temperature distributions for the most favorable configuration, type ID-4(L) and (S), at several turbine speeds are shown in figure 5(a). The results are shown for lower speeds merely to provide an indication of the effectiveness which may be realized with spray cooling at lower gas temperatures and blade tip speeds. In general, a similar trend in the spanwise temperature distribution (low root and midspan and relatively high tip temperature) was obtained at all engine speeds. A negligible reversal in gradient occurred at the lower speeds investigated. The maximum temperature difference between the blade root and blade tip occurred at rated speed and was about 500° F as compared with about 100° F for the uncooled condition. Maximum root-to-tip temperature differences for the spray cooled case were approximately 250° and 200° F at 8000 and 4000 rpm, respectively, which is almost identical to the root-to-tip temperature differences encountered for the uncooled condition at these speeds.

A comparison of the chordwise temperature distributions for configuration ID-4(L) and (S) at several engine speeds is shown in figure 5(b). No appreciable gradients occurred at either the blade root or midspan at 4000 and 8000 rpm. Of these two speeds the maximum chordwise temperature difference (between the leading and trailing edges) occurred at the root at 4000 rpm and was 200° F. At rated speed, the maximum temperature difference was 470° F and occurred between the trailing edge and the midchord on the pressure surface at the midspan position.

In order to determine whether the midspan chordwise temperature difference obtained at rated speed with this type of configuration could be further reduced, the coolant flow was increased by increasing the 0.135-inch-diameter orifices to 0.150-inch diameter. The midspan chordwise temperature distribution obtained with the resulting configuration, ID-4(L') and (S), at rated speed and a turbine inlet-gas temperature of 1568° F with a coolant-to-gas flow ratio of 0.0278 is presented in figure 6. For ease of comparison, the temperature distribution obtained with configuration ID-4(L) and (S) at a coolant-to-gas flow ratio of 0.0235 is also shown. The temperature difference between trailing edge and midchord was greatly reduced from that occurring at the lower coolant-to-gas flow ratio. A maximum temperature of 400° F was obtained at the blade midspan on the trailing edge. This represents a reduction in temperature of 930° F from the uncooled condition. The maximum chordwise temperature difference occurred between the trailing edge and midchord on the blade suction surface and was 180° F. The

CI-2 back

temperature difference between the leading and trailing edges was 100° F and the average midspan blade temperature was 292° F. A chordwise temperature distribution around the blade root is not shown because of loss of most of the root thermocouples during previous operation. From the few temperature readings obtained at the blade root during operation with configuration ID-4(L') and (S), no apparent change occurred in the value of the root temperature level from that demonstrated previously in figure 5(b) for configuration ID-4(L) and (S). It appears that with regard to minimizing chordwise temperature differences at rated engine speed, an inner-diameter-type injection configuration with two small and two large orifices located near the stator blade trailing edge is the most satisfactory.

Cooling Efficiencies of Various Configurations

Another means of determining the cooling effectiveness of various water-injection configurations is by a comparison of the spray cooling process efficiency obtained with each configuration. Such a comparison is made in the following section, and the efficiency of the most favorable configuration is applied to determine the amount of water required for engine operation at one condition of overtemperature and overspeed.

Comparison of injection configuration cooling efficiencies. - The efficiency of the spray cooling process σ, plotted against the coolant-flow parameter $\left(\frac{w_w}{w_g^{0.75}}\right)$ for various inside-diameter injection configurations investigated at rated engine speed, is presented in figure 7. No efficiencies are presented for the spray bar and outside-diameter orifice configurations because of the limited number of flow rates investigated and the unsatisfactory cooling obtained with these configurations. Several facts are to be noted from these curves. A trend of decreasing efficiency level is apparent as the injection configurations were located successively farther upstream of the rotor. The lowest efficiencies range from 3.5 to 17 percent and were obtained with configuration ID-1(L), which was located upstream of the stator leading edge. It should be noted that this trend of decreasing cooling efficiency with configurations located successively farther upstream of the rotor indicates greater evaporative losses and is also reflected by increasingly unfavorable blade temperature distributions discussed previously. Only configuration ID-1(L) exhibits a curve with a clearly defined maximum efficiency point. The maximum efficiency occurs at lower values of the coolant-flow parameter with the other configurations. Because of the possibility of introducing thermal stresses in the blades, it was not considered desirable to operate at lower values of coolant flow in order to establish the peak efficiency point for all configurations. At rated speed, the efficiency curves of all configurations except ID-4(L) and (S) present steep negative slopes with minimum efficiency values occurring in each case at the highest values of the coolant-flow parameter. It has already been shown that favorable blade temperature distributions were not obtained at rated engine conditions with these injection configurations at the maximum coolant flows considered in this investigation. In order to obtain more desirable blade temperature distributions with these configurations, operation at even higher coolant flows would be required, and extrapolation of efficiency curves with the sharply negative slopes indicates that this could result in extremely inefficient operation. Configuration ID-4(L) and (S), however, presents an efficiency curve with a relatively flat slope and a higher efficiency (27.5 percent) at the maximum value of coolant flow parameter (0.067 equivalent to a 0.0235 coolant-to-gas flow ratio) considered in this investigation than that of the other configurations. Also, extrapolation of the relatively flat efficiency curve indicates that further coolant flow increases, such as might be required for over-temperature, overspeed operation, would not appreciably decrease the cooling efficiency with configuration ID-4(L) and (S). Therefore, for efficient coolant utilization at coolant-flow rates likely to be encountered, it appears that configuration ID-4(L) and (S) is the most favorable of those investigated. Because of the limited number of runs made with configuration ID-4(L') and (S), an efficiency curve for this configuration is not shown. At rated speed and a coolant-to-gas flow ratio of 0.0278, which provided the favorable midspan chordwise temperature distribution shown in figure 6, the cooling efficiency with configuration ID-4(L') and (S) was 24 percent.

The efficiencies obtained with configuration ID-4(L) and (S) at several speeds are presented in figure 8. The rapidly decreasing slope common to the efficiency curves of the other configurations at rated engine conditions is apparent with this configuration at 4000 and 8000 rpm. Minimum efficiencies of 40 and 34 percent occurred at 4000 and 8000 rpm, respectively, and at coolant-flow parameter values of 0.027 and 0.0336, respectively. These coolant-flow parameter values are equivalent to coolant-to-gas flow ratios of 0.0126 and 0.0129. Satisfactory blade temperature distributions (a low-temperature level and no chordwise gradients at either blade root or midspan) were obtained at these points; consequently, the engine was not operated at higher coolant-flow values to extend these curves. These data indicate that a slightly higher coolant utilization effectiveness is realized at lower engine speeds than at rated speed with this configuration. Efficiency curves with approximately horizontal slopes result at rated engine speed for effective gas temperatures of 1399° and 1521° F and the curves extend to coolant-flow parameters of 0.067 and 0.0663, respectively, equivalent to coolant-to-gas flow ratios of 0.0235 and 0.0232, respectively. Although a slightly higher efficiency level occurs at the higher gas temperature, a trend of increasing efficiency level with increasing gas temperature should not be interpreted from the limited data available.

Application to operating conditions above rated. - The spray cooling efficiencies obtained at rated speed with two configurations, ID-4(L) and (S) and ID-3(L), were applied in calculations designed to obtain an indication of the coolant requirements to operate at 2000° F inlet-gas temperature and 10 percent overspeed with the engine under consideration while employing blades of S-816 material. These configurations were chosen because ID-3(L) provided the most effective cooling in the investigation described in reference 1, while ID-4(L) and (S) is the most effective of those described in this report. The lowest values of efficiency obtained experimentally with these configurations (27.5 percent for ID-4(L) and (S) and 22.5 percent for ID-3(L)) were employed to incorporate an additional safety factor. The calculations indicated that coolant-to-gas flow ratios of 0.0090 (equivalent to a coolant-flow rate of 5.80 gal/min) and 0.0109 (7.02 gal/min) are required with configurations ID-4(L) and (S) and ID-3(L), respectively.

These values are lower than the coolant-to-gas flow ratios shown herein to be necessary to provide satisfactory blade temperature distributions at standard engine operating conditions. The apparent discrepancy is clarified, however, if it is recalled that these calculations were made for an average blade temperature ($T_{B,av}$ in eq. (1)) of 1200° F. Blade temperature distribution plots in this report indicate that when chordwise blade temperature differences are satisfactorily reduced, a much lower blade temperature level than 1200° F results. Consequently, if consideration is given to minimizing thermal gradients and attendant thermal stresses, these calculated coolant-flow values provide an unrealistic picture of the coolant requirements at overspeed, overtemperature conditions. The calculations do indicate, however, what may be termed a theoretical minimum coolant-flow requirement for overspeed, overtemperature operation with S-816 material blades and serve to illustrate the effect of differences in spray cooling efficiency in terms of coolant-flow rate. Since the average blade temperature $T_{B,av}$ which will occur when undesirable chordwise temperature differences are eliminated cannot be specified accurately for the conditions under consideration, the actual coolant-flow requirements can be determined only experimentally. The foregoing discussion again serves to illustrate the fact that a certain amount of overcooling is involved in spray cooling with the configurations under consideration because of the lower temperature level which is a result of the reduction of large blade temperature gradients. It should be emphasized that these calculations assume all other engine components capable of withstanding operation at overspeed, overtemperature conditions.

Rotor Blade Failures

The effect of spray cooling upon blade life is a factor to be considered in view of (1) possible thermal shocks imposed, and (2) thermal stresses introduced in the blades when large chordwise blade temperature differences occur. It is not the purpose of the present investigation to evaluate either of these problems. In fact, in order to minimize the thermal shock problem, which should be investigated separately prior to any final application of spray cooling, water was injected simultaneously with the engine starts. However, some information can be obtained from the current investigation as to life characteristics of S-816 blades when subjected to large chordwise temperature differences. Three blades developed cracks. One failure occurred on a thermocoupled blade. In a fourth failure, the upper half of the blade was lost, probably as the result of continued propagation of a crack such as that shown in figure 9, which presents a typical blade failure. In all failures, the crack extended forward from the trailing edge at approximately the midspan, which is generally considered a critically stressed blade section for this turbine. The extent of the cracks (penetration from the trailing edge toward the midchord) varies from a minimum of 1/4 to 1 inch. The first blade failure occurred after 15 hours of spray cooling operation. The difference in total blade life between the first and the final failed blade was approximately 2 hours. Three of the four failures occurred during operation with injection configurations which provided the most severe chordwise temperature differences encountered, these configurations being the spray bar and the combined outer- and inner-diameter injection orifices. A limited comparison can be made with the Stellite 21 blades used in the spray cooling investigation described in reference 1. The bulk of the engine operation in the investigation of reference 1 was conducted at lower speeds and at stress levels approximately 1/8 and 1/2 the blade stress encountered at rated speed. Practically all engine operation in the current investigation was conducted at rated engine speed. On this basis, it appears that S-816 blades are more resistant to failure induced by the added stresses introduced through large chordwise temperature differences. It should be noted again that if these temperature differences can be eliminated by use of suitable injection configurations, blade life probably will be increased. In that case, the magnitude of possible thermal shock to the blades must still be evaluated to determine the full effect of spray cooling on blade life.

Rotor Blade Spray Patterns

Typical patterns resulting from the deposition of minerals from the city water used in the spray cooling process onto the blade suction and pressure surfaces after operation with configuration ID-4(L) and (S) are shown in figures 10(a) and 10(b), respectively. As indicated in reference 1, the boundary between the wetted and unwetted portions of the blade surface is probably defined by the heavy white lines. The orderly pattern of the lines of deposition which extend progressively farther back from the leading edge on the suction surface and farther upward in a spanwise direction on the pressure surface evidently corresponds to increasing values of coolant flow. This difference between patterns on the suction and pressure surfaces, the former presenting a spanwise or vertical interface between the wetted and unwetted sections and the latter presenting a horizontal interface, would seem to indicate a major difference in the spray dispersion over the two blade surfaces.

On the basis of these patterns the flow mechanism can readily be visualized for the suction surface; however, this is more difficult insofar as the pressure surface is concerned. The analysis of reference 1 indicates that at rated engine speed, a water velocity relative to the rotor of approximately 2000 feet per second is required in order for the water spray to strike the rotor blades at the aerodynamic gas entrance angle. In the case of configuration ID-4(L) and (S), with which the patterns of figure 10 were obtained, it is reasonable to conclude that the spray could not have been accelerated to this velocity in the short distance between the injection orifices and the rotor. Consequently, the spray entrance angle relative to the plane of rotation is less than the aerodynamic entrance angle and the water strikes primarily the suction surface. The pattern indicates that the water spray is slung radially along the suction surface and that with increasing flow rates, the degree of surface coverage from leading to trailing edges is increased. Several possible flow mechanisms can be postulated to account for cooling on the pressure surface. One obvious assumption is that cooling on this surface may be achieved primarily by conduction rather than by spray cooling. The validity of this assumption was checked in an approximate manner. The heat input to the blade was calculated using a gas-to-blade convection coefficient of approximately 200 Btu per hour per square foot per $^{\circ}F$ (the same coefficient used in reference 4 in calculating blade temperatures) and neglecting any possible temperature drop occurring through such a water film as may exist along the blade surface. Then by means of a simple heat balance the temperature differences through the blade were calculated for runs made at several coolant flows at various blade locations and were compared with measured temperature differences. The results of these calculations were inconclusive, indicating that conduction alone could account for the temperature differences encountered only in a few instances. Neither could a trend be established as to specific locations on the blade where conduction could account for the temperature differences encountered.

Visual flow studies of secondary flow conducted by the NACA indicate the formation of passage vortices (refs. 5 and 6). The presence of such vortices could possibly account for the manner of spray dispersion encountered on the blade pressure surface and the consequent pattern of mineral deposition. Further spray cooling investigations, perhaps with provision for visual flow determination, are required to obtain an accurate picture of the spray cooling flow mechanism.

The difference between the spray pattern shown in figure 10(b) and that shown on the blade with the eroded tip which will be discussed later (fig. 11) is primarily due to the different methods of injection employed and the fact that only two coolant-flow rates were used in achieving the pattern in figure 11. A marked similarity exists, nevertheless, with the pronounced areas of deposition extending across the blade (fig. 11) representing the boundaries between wetted and unwetted sections as established by the two coolant flows considered. The area of deposition in the upper right section of the blade probably is the result of spray injected from the outside-diameter orifices in the tail-cone outer shell.

Over-All Comparison of Injection Configurations

In order to establish clearly the advantages and disadvantages of the various configurations investigated, an over-all comparison is presented. The comparison is made on the basis of general applicability, which includes such factors as configuration life, ease of installation and maintenance, and safety of installation, as well as cooling effectiveness which has been discussed in detail previously.

One of the major considerations in evaluating an injection configuration is cooling effectiveness. It has already been shown that a configuration of the inside-diameter type is the most favorable from this consideration. In finally evaluating this configuration as well as the other configurations investigated, however, other factors must be considered. Logically, the first of these is ease of fabrication and installation. Difficult fabrication and assembly procedures detract from the desirability of any configuration regardless of its cooling effectiveness. In this regard, none of the inside- or outside-diameter injection configurations presents any problems. Actually, this type of configuration is perhaps as simply installed as any that can be devised, consisting essentially of easily drilled orifices of the desired diameter. The method of introducing the coolant supply through tubes welded to the stator diaphragm or the ring and tube assembly is also readily accomplished. A spray bar of the type considered herein is also easily fabricated.

A second consideration in evaluating the general applicability of an injection configuration is the life expectancy of the installation and the ease with which it may be maintained in working order. Again the various inside- and outside-diameter injection configurations seem to afford the least difficulty. Except for the possibility of clogging extremely small diameter orifices, a condition not encountered during the present investigation and one that can readily be remedied by a simple reaming process during normal engine inspections, these configurations can be operated indefinitely without adjustment. This is in marked contrast to the welding cracks encountered around such configurations as the stator trailing-edge orifices described in reference 1

CI-3

and the spray bar used in the present investigation. Figure 12 indicates the condition of an Inconel spray bar after approximately 1 hour of rated speed operation both with and without spray cooling. It is apparent that the combination of gas forces and gas temperature has bent the bar after a relatively short period of operation and that prolonged operation would probably have resulted in failure of the spray bar. Although it may be possible to provide sufficient reinforcement to prevent such bending or even to provide a means of retracting the bar from the gas stream when not in use, additional undesirable complexities would be introduced. Thus, besides its relatively low cooling effectiveness, the spray bar has other major disadvantages.

Finally, consideration must be given to possible detrimental effects imposed upon the engine by the injection configuration. The detrimental effect of thermal shock upon the rotor blades exists regardless of configuration type. That it varies appreciably with the configuration employed is doubtful. Similar thermal shock and temperature gradient effects can conceivably present problems in regard to the stator blades when injection configurations in the region of the stator blades are employed. At overtemperature conditions, the hazards of improper gas-stream temperature distributions may be intensified and stator-blade cooling may be necessary for operation at these conditions. Another possibility is erosion of engine parts due to the water stream. An example of rotor blade erosion is shown in figure 11. The blade tip has been eroded after a brief period of engine operation with an injection configuration which included orifices in the tail-cone outer shell. The erosion is apparently the result of direct impingement of the water spray upon the rotor blade tips. Operation under these conditions is, of course, intolerable and would rule out this type of injection configuration regardless of its cooling effectiveness (in this case, poor). The final consideration of possible detrimental effects upon the engine also becomes a major factor if a spray bar or any injection configuration involving a protuberance into the gas stream upstream of the rotor is employed. The danger of failure and damage to rotor blades is then a constant possibility.

The preceding discussion indicates that the inside-diameter type configuration, consisting of two relatively large (0.135- or 0.150-in.-diam.) and two relatively small (0.078-in. diam.) orifices located near the stator blade trailing edge, is the most favorable on an over-all as well as a cooling basis. Furthermore, many of the considerations which determine the general applicability of a configuration, such as installation and maintenance ease, actually increase the desirability of this configuration in comparison with others considered in this investigation as well as in the investigation described in reference 1.

SUMMARY OF RESULTS

The following results were obtained from an investigation of water-spray cooling with several stationary configurations in order to determine the most favorable one for application of water spray cooling to turbine blades in a turbojet engine:

1. In evaluating water-injection configurations for spray cooling from an over-all consideration of cooling effectiveness, ease of fabrication, installation, and maintenance, configuration durability, and safety of the configuration installation insofar as it affects other parts of the turbine, one general type was found to be the most favorable for application to a typical centrifugal-flow turbojet engine. This type configuration was located in the inner ring of the stator diaphragm near the stator blade trailing edge and consisted of two relatively large (0.135- or 0.150-in. diam.) and two small (0.078-in. diam.) orifices.

2. The most effective configuration with regard to cooling provided adequate spray coverage at the rotor blade root by means of the smaller orifices and at the midspan by means of the larger orifices. At rated speed (11,500 rpm) and 1568° F turbine inlet-gas temperature, use of this configuration resulted in a midspan chordwise temperature difference between leading and trailing edges of 100° F and an average midspan blade temperature of 292° F with a coolant-to-gas flow ratio of 0.0278 (7600 lb/hr) and a spray cooling process efficiency of 24 percent.

3. Location of inside-diameter injection orifices close to the downstream edge of the stator diaphragm provided more effective rotor blade cooling than locations farther upstream.

4. Injection configurations such as a spray bar or orifices located in the tail-cone outer shell were ineffective with regard to cooling and presented hazards to safe turbine operation.

5. Rotor blades of S-816 material appear to afford greater resistance to failure caused by the introduction of large chordwise temperature differences than blades of Stellite 21 material.

Lewis Flight Propulsion Laboratory
National Advisory Committee for Aeronautics
Cleveland, Ohio, August 14, 1953

CI-3 back

APPENDIX - SYMBOLS

The following symbols are used in this report:

ID	inside diameter
OD	outside diameter
L	large orifice diameter, 0.135 in.
L'	large orifice diameter, 0.150 in.
S	small orifice diameter, 0.078 in.
T	temperature, $^{\circ}R$
w	weight flow, lb/sec
σ	efficiency of spray-cooling process
1,2,3, 4,5	axial stations of injection configurations

Subscripts:

av	average
B	blade
e	effective
g	gas
w	water

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TABLE I. - WATER-INJECTION CONFIGURATION NOMENCLATURE

General orifice type	ID	Inside diameter
	OD	Outside diameter
Orifice axial location	1	Upstream of stator blade leading edge in ring and tube assembly
	2	At stator blade leading edge in stator diaphragm
	3	At stator blade midchord in stator diaphragm
	4	At stator blade trailing edge in stator diaphragm
	5	Downstream of stator blade trailing edge in tail cone
Orifice diameter	L'	Large diameter, 0.150 in.
	L	Large diameter, 0.135 in.
	S	Small diameter, 0.078 in.
Sample configuration designation	ID-4(L) and (S)	Stator diaphragm inner diameter orifices at trailing edge of stator blade. Orifice sizes, 0.135 and 0.078 in. diam.

TABLE II. - RANGE OF OPERATING CONDITIONS

Configuration[1]	Speed, rpm	Effective gas temperature, °F	Range of coolant-to-gas flow ratio
ID-4(L)	11,500	1362	0.00653 to 0.02286
ID-3(L)	11,500	1372	0.00657 to 0.02316
ID-2(L)	11,500	1394	0.00620 to 0.02220
ID-1(L)	11,500	1419	0.0065 to 0.0205
ID-4(L) and (S)	11,500	1399	0.0059 to 0.0279
		1521	0.0114 to .0232
		1413	.0220 to .0231
	4,000	969	.0046 to .0126
	8,000	943	.0046 to .0129
ID-4(L) and (S), ID-1, -2, and -3	11,500	1398	0.0220 to 0.0222
Spray bar	11,500	1437	0.0093 to 0.0205
ID-4(L) and (S) and OD-5(S)	11,500	1405	0.0167 to 0.0221
ID-4(L') and (S)	11,500	1368	0.0139 to 0.0278

[1]Each configuration consisted of two orifices except configurations ID-4(L') and (S), and ID-4(L) and (S). These each had 4 orifices.

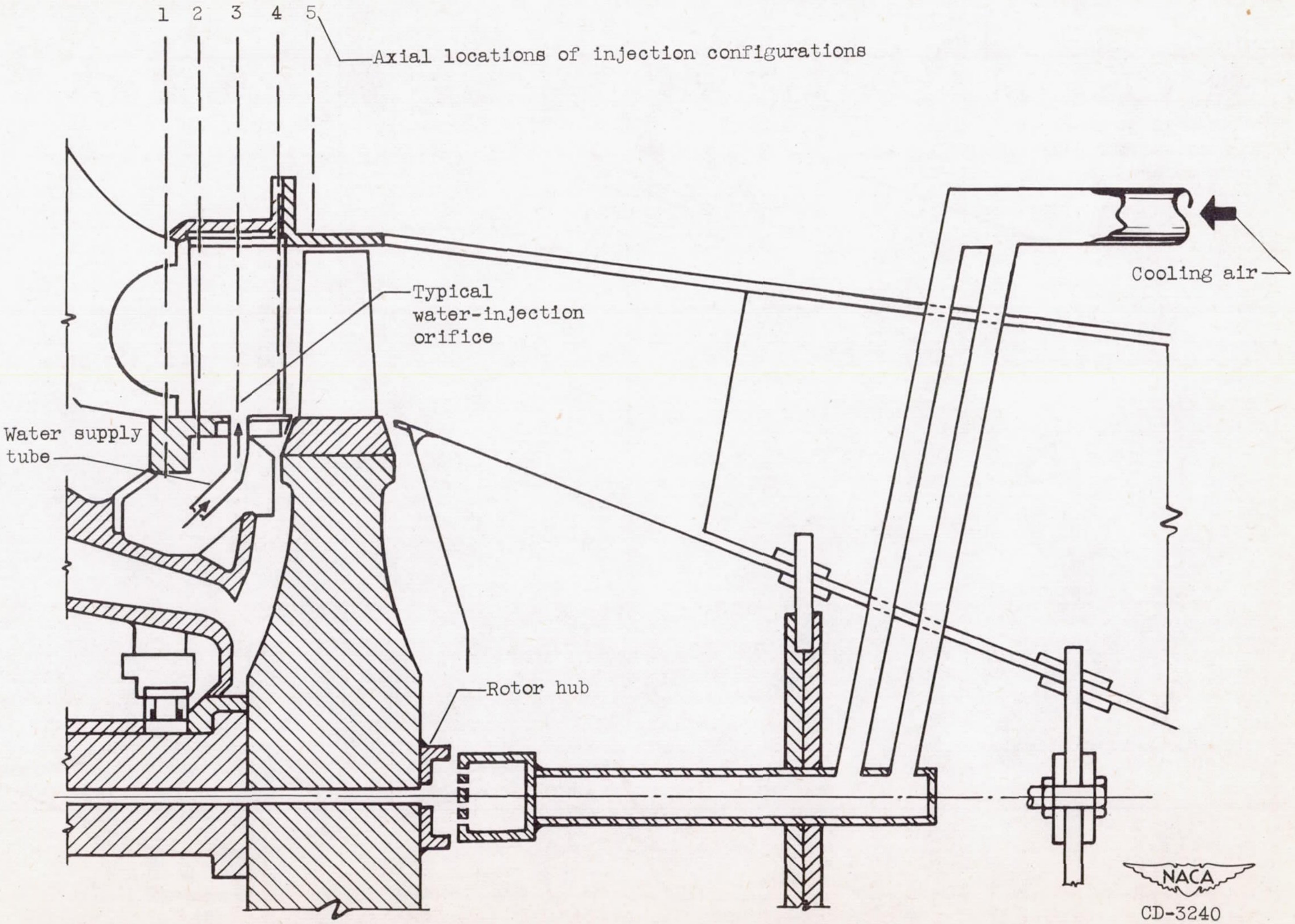


Figure 1. - Section of engine showing typical water-injection configuration and axial locations of various water-injection configurations investigated.

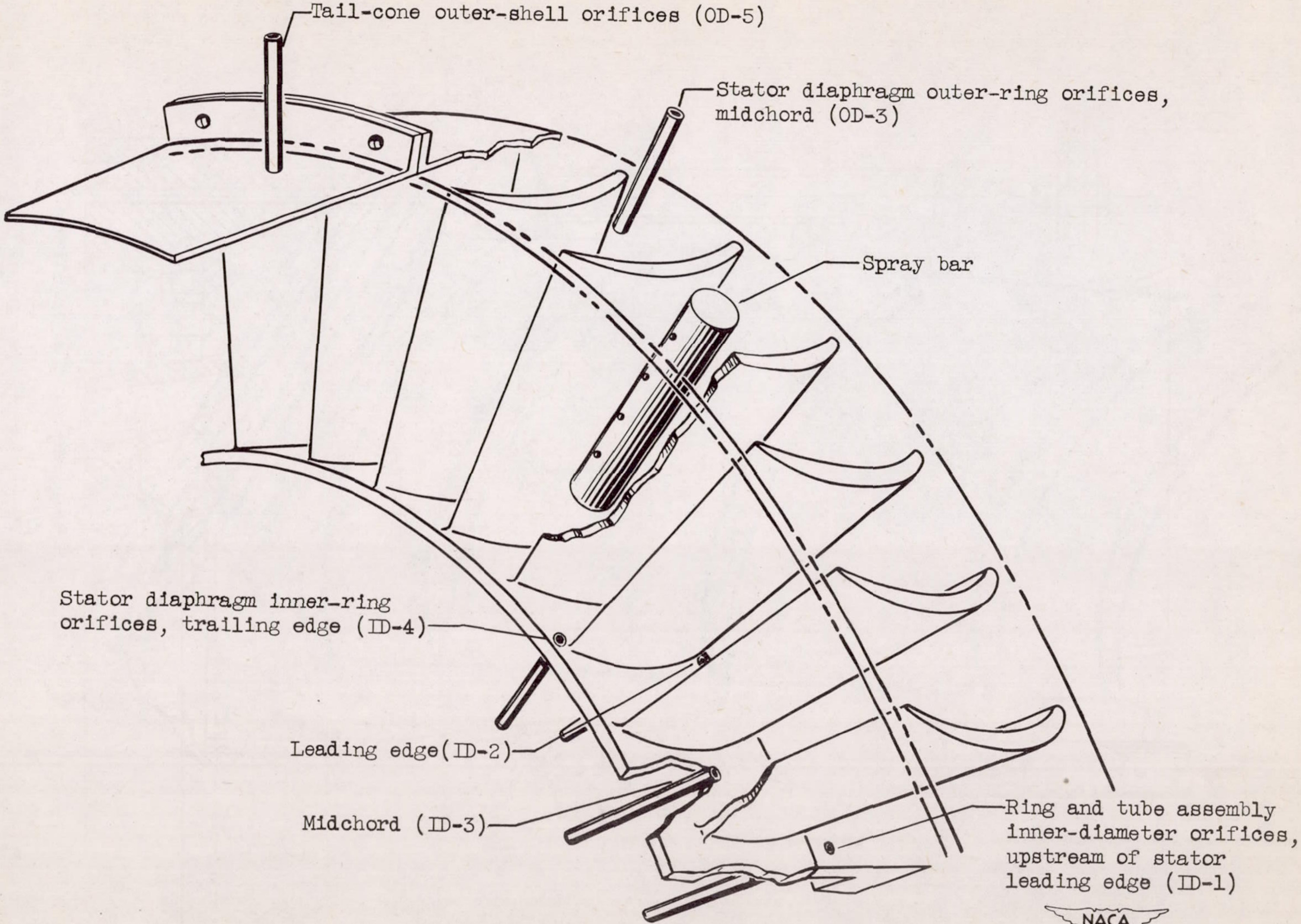


Figure 2. - Types of water-injection configuration investigated.

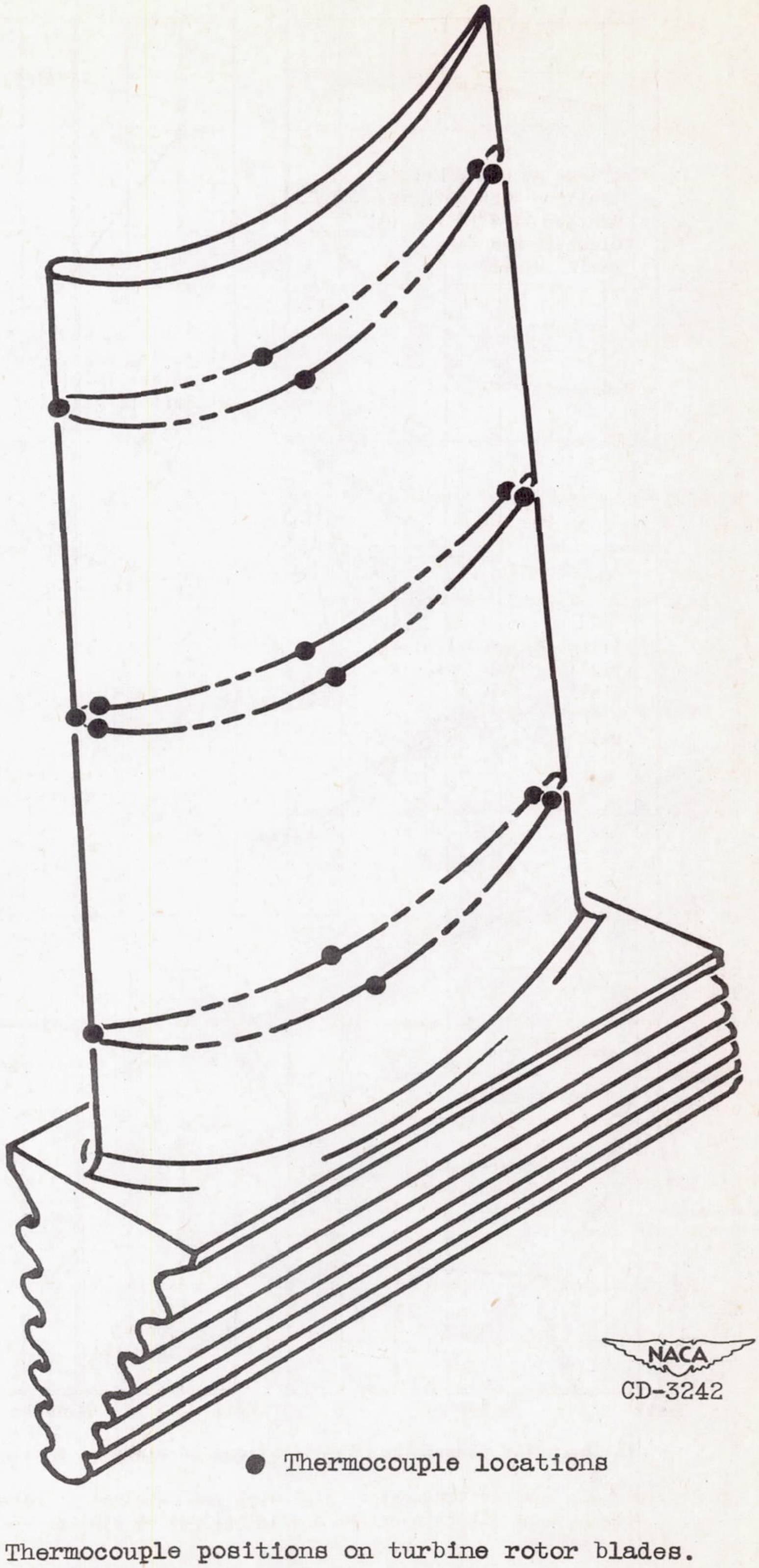
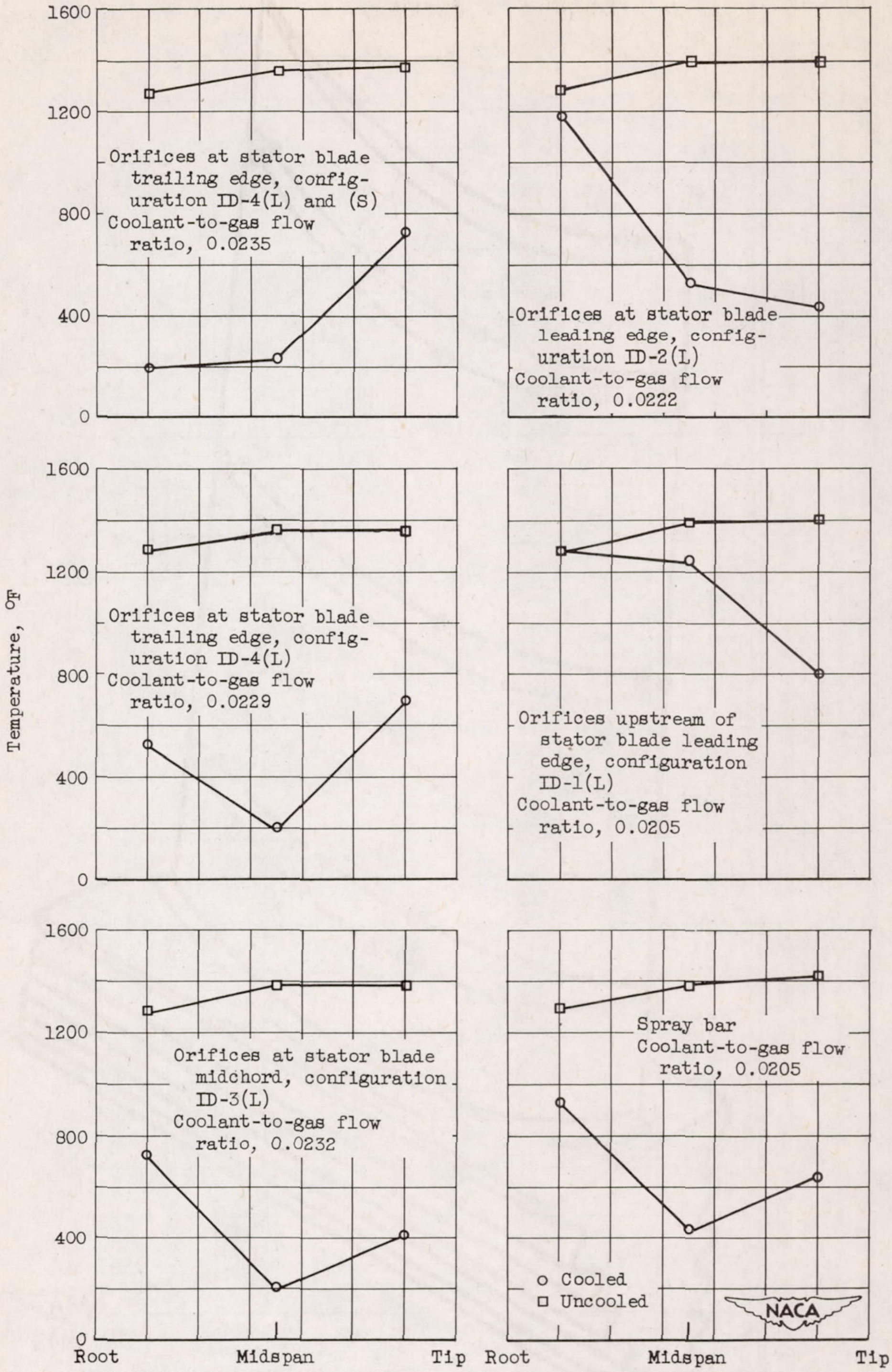


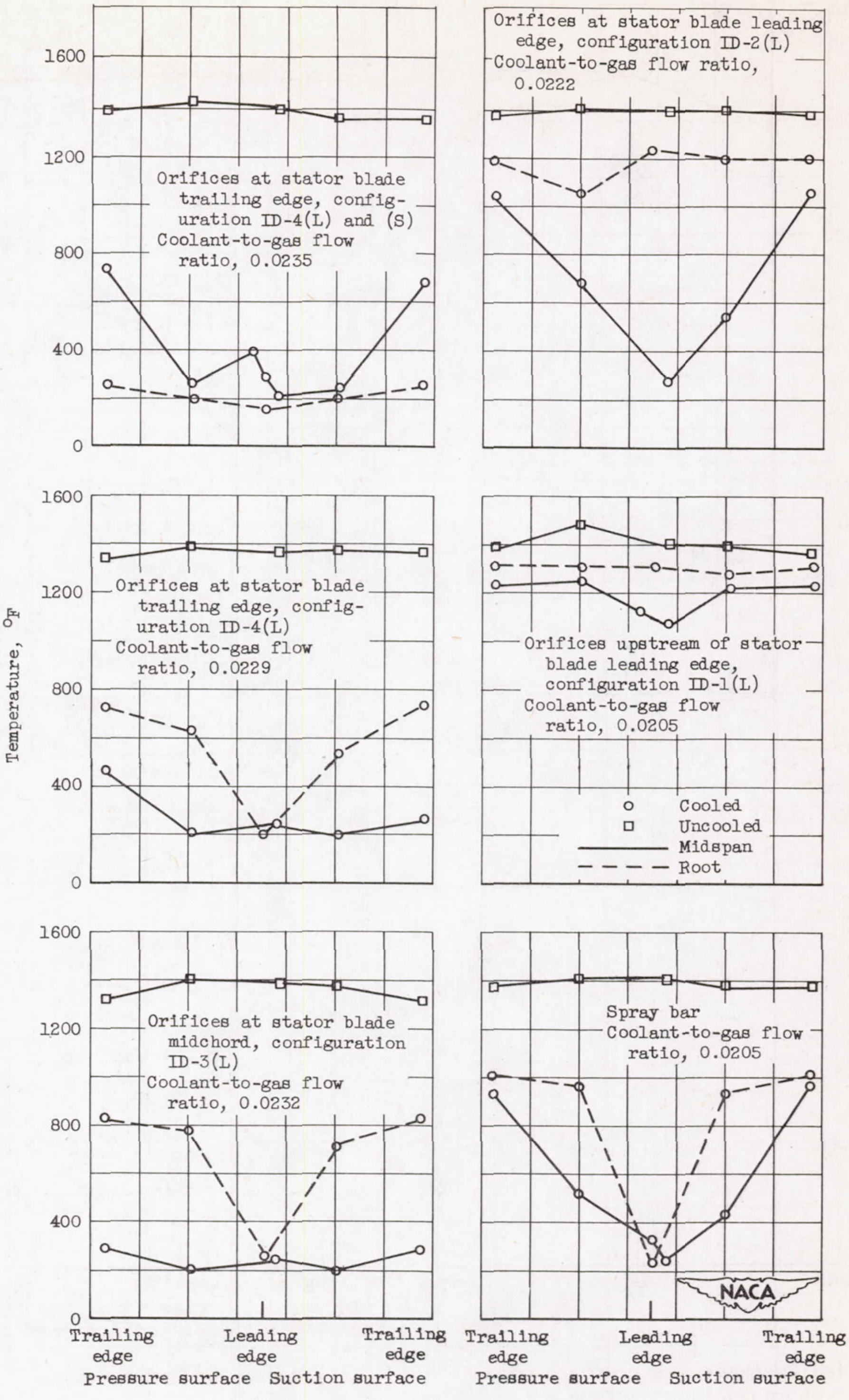
Figure 3. - Thermocouple positions on turbine rotor blades.

CI-4 back



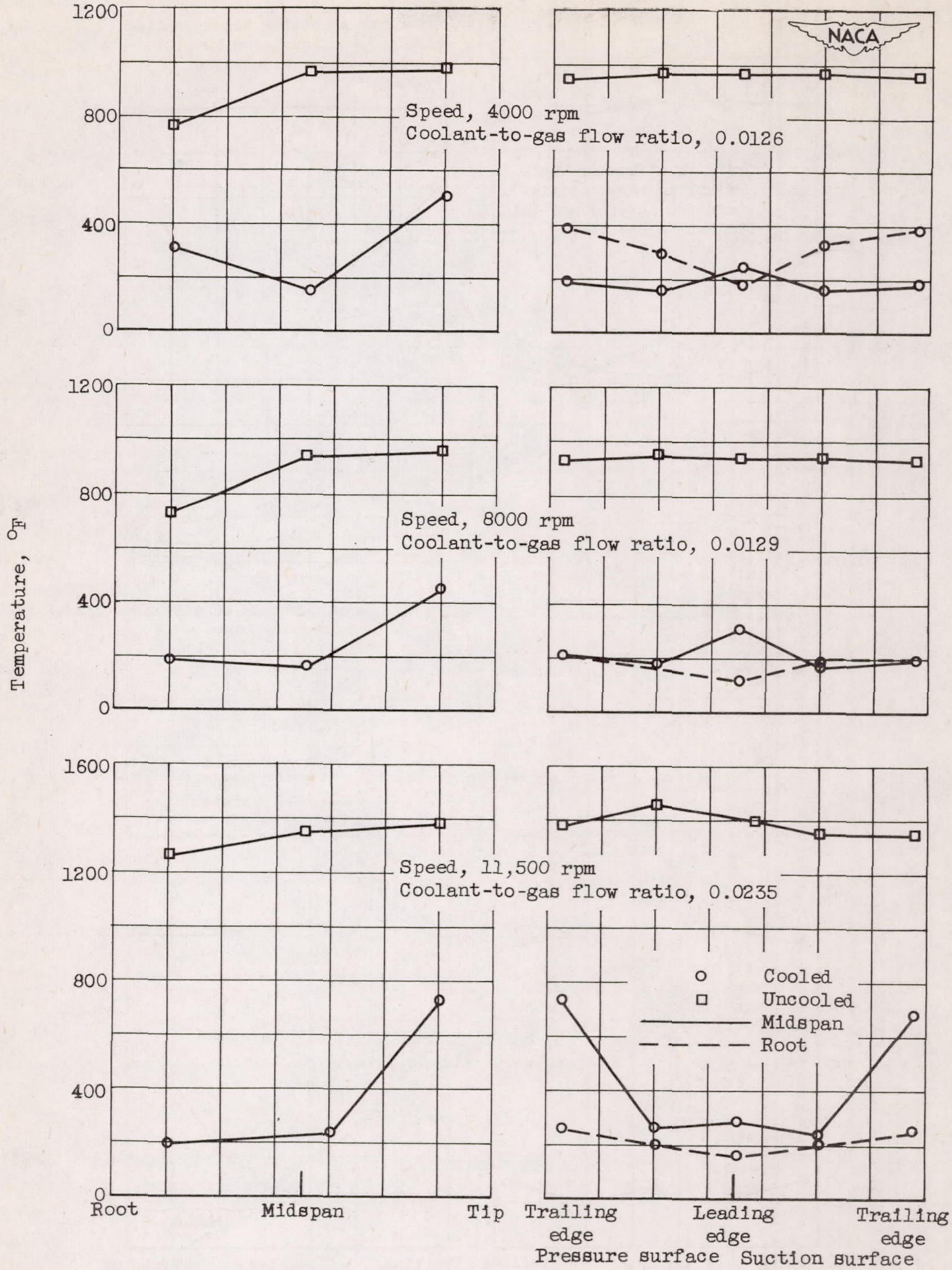
(a) Spanwise temperature distributions on midchord suction surface.

Figure 4. - Blade temperature distributions obtained at rated engine speed with various water-injection configurations at similar coolant-to-gas flow ratios.



(b) Chordwise temperature distributions.

Figure 4. - Concluded. Blade temperature distributions obtained at rated engine speed with various water-injection configurations at similar coolant-to-gas flow ratios.



(a) Spanwise temperatures on midchord suction surface.

(b) Chordwise temperatures.

Figure 5. - Blade temperature distributions obtained at various engine speeds with one injection configuration. Configuration type - stator diaphragm inner-ring orifices near trailing edge, ID-4(L) and (S).

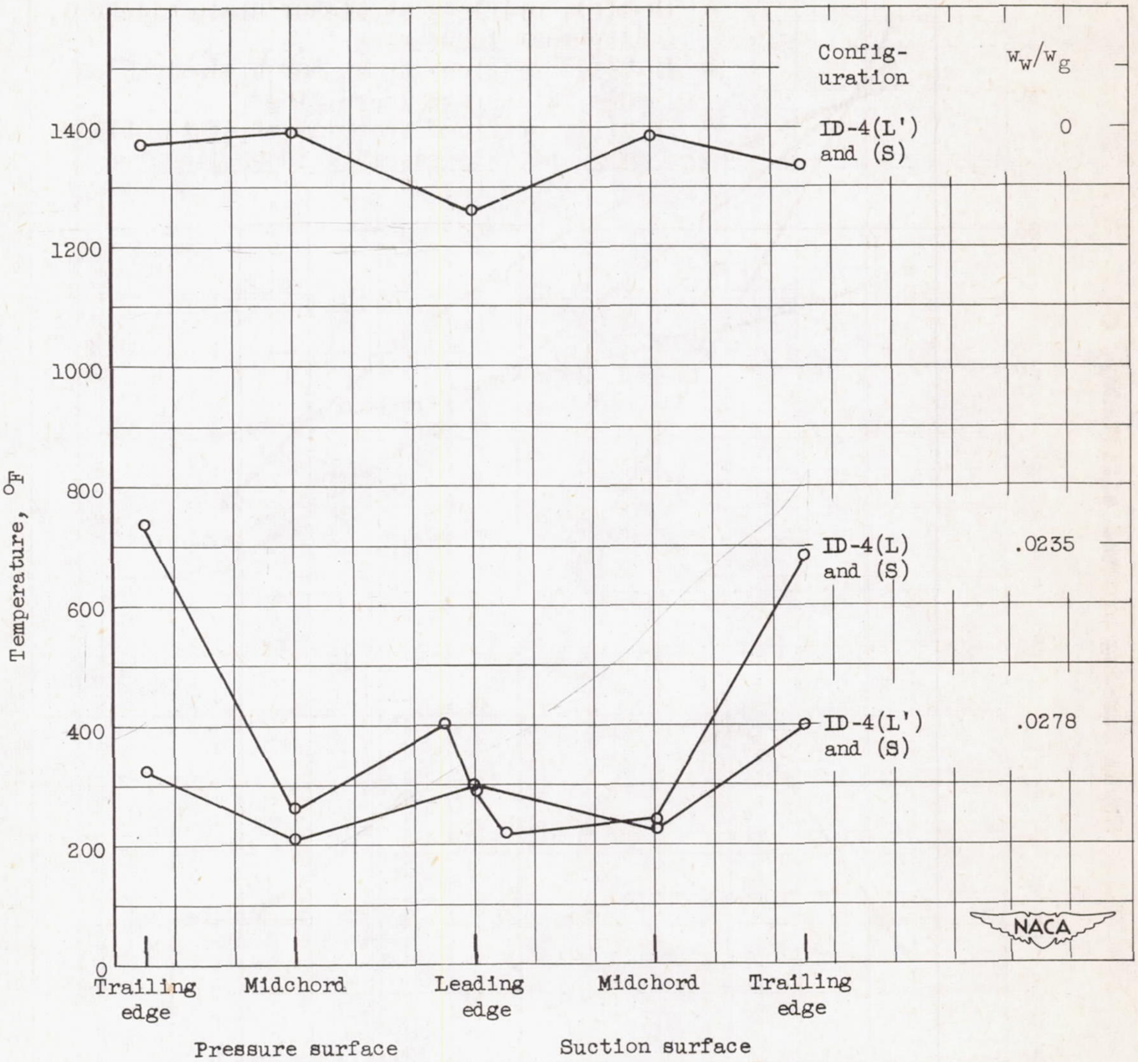


Figure 6. - Midspan chordwise blade temperature distributions obtained with configurations ID-4(L') and (S) and ID-4(L) and (S) at rated engine speed.

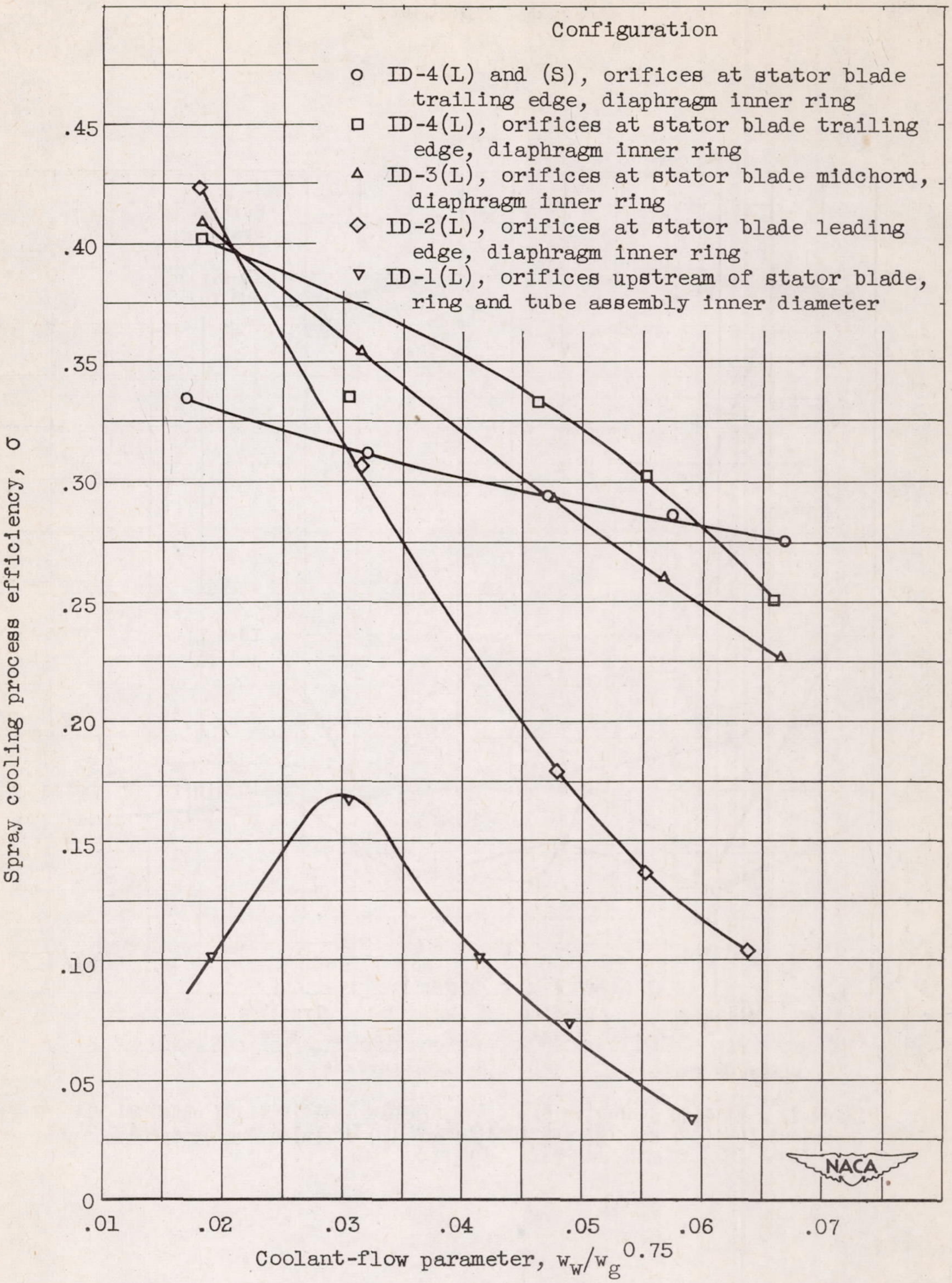


Figure 7. - Efficiency of spray cooling process plotted against coolant-flow parameter for various injection configurations investigated at rated engine speed.

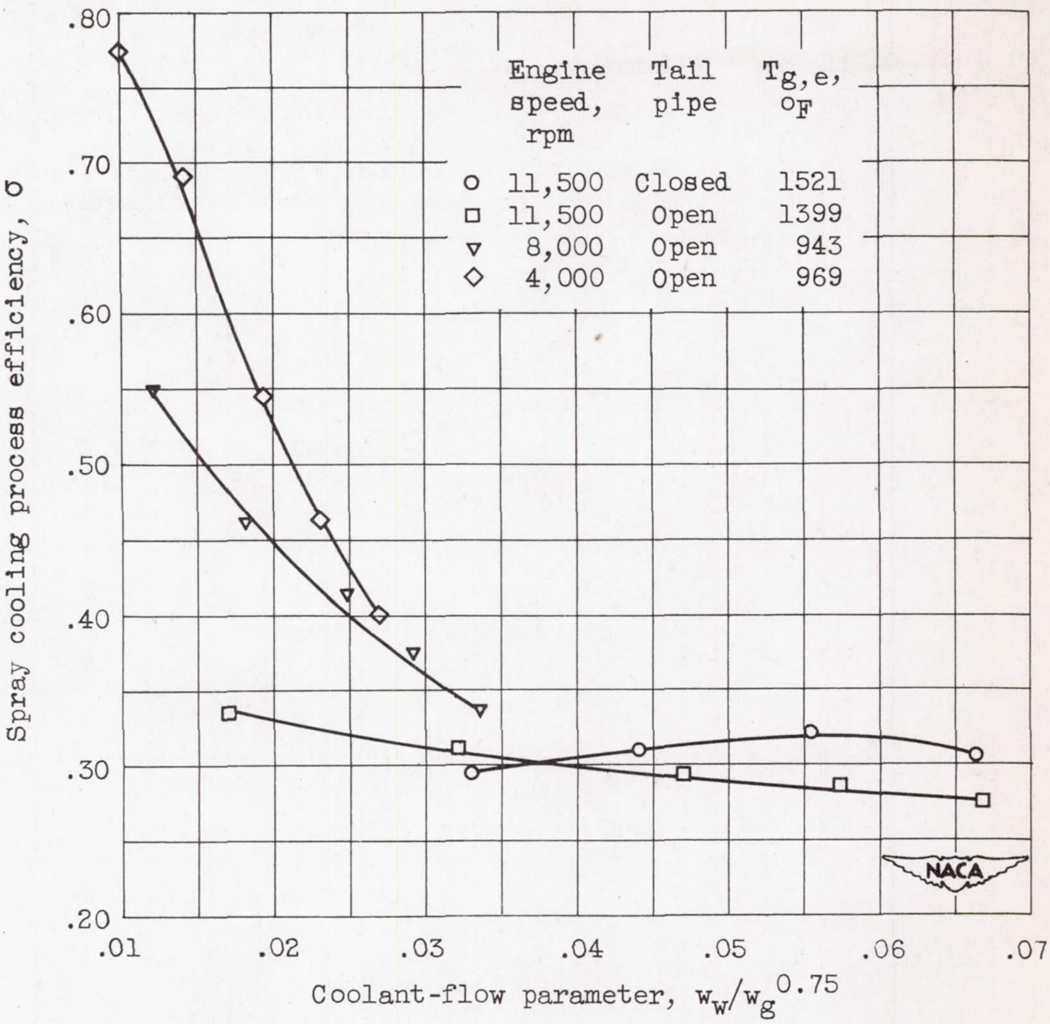
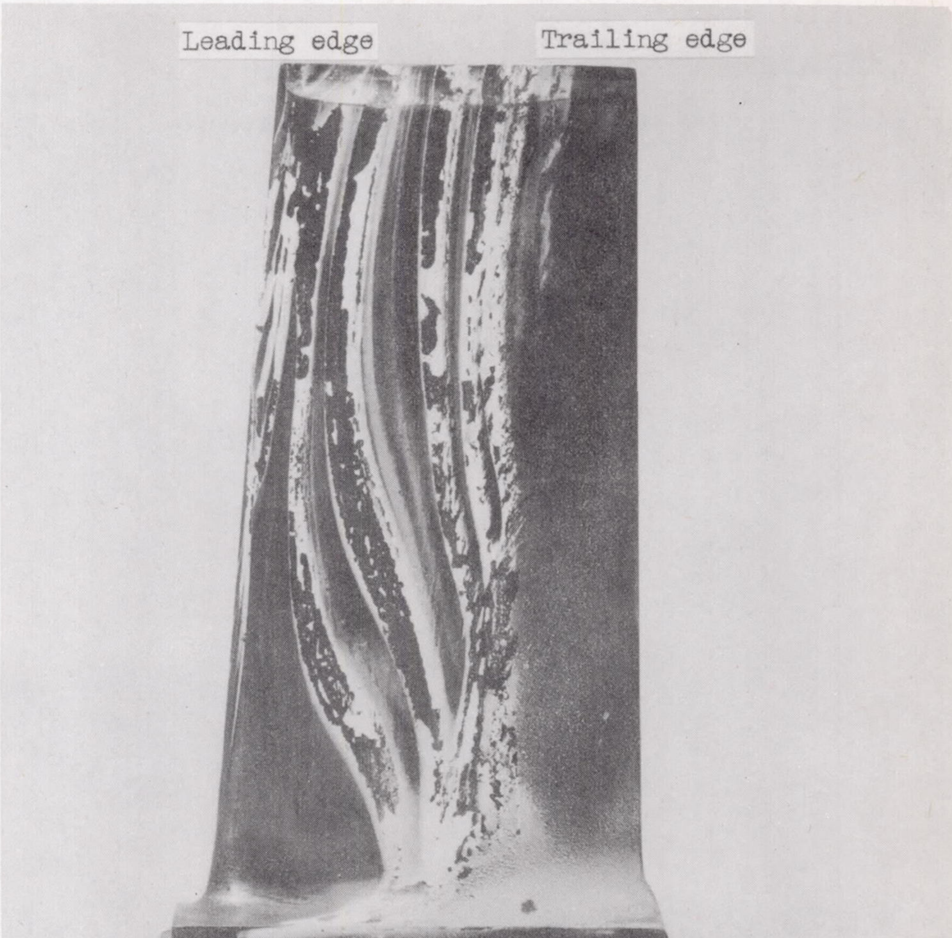


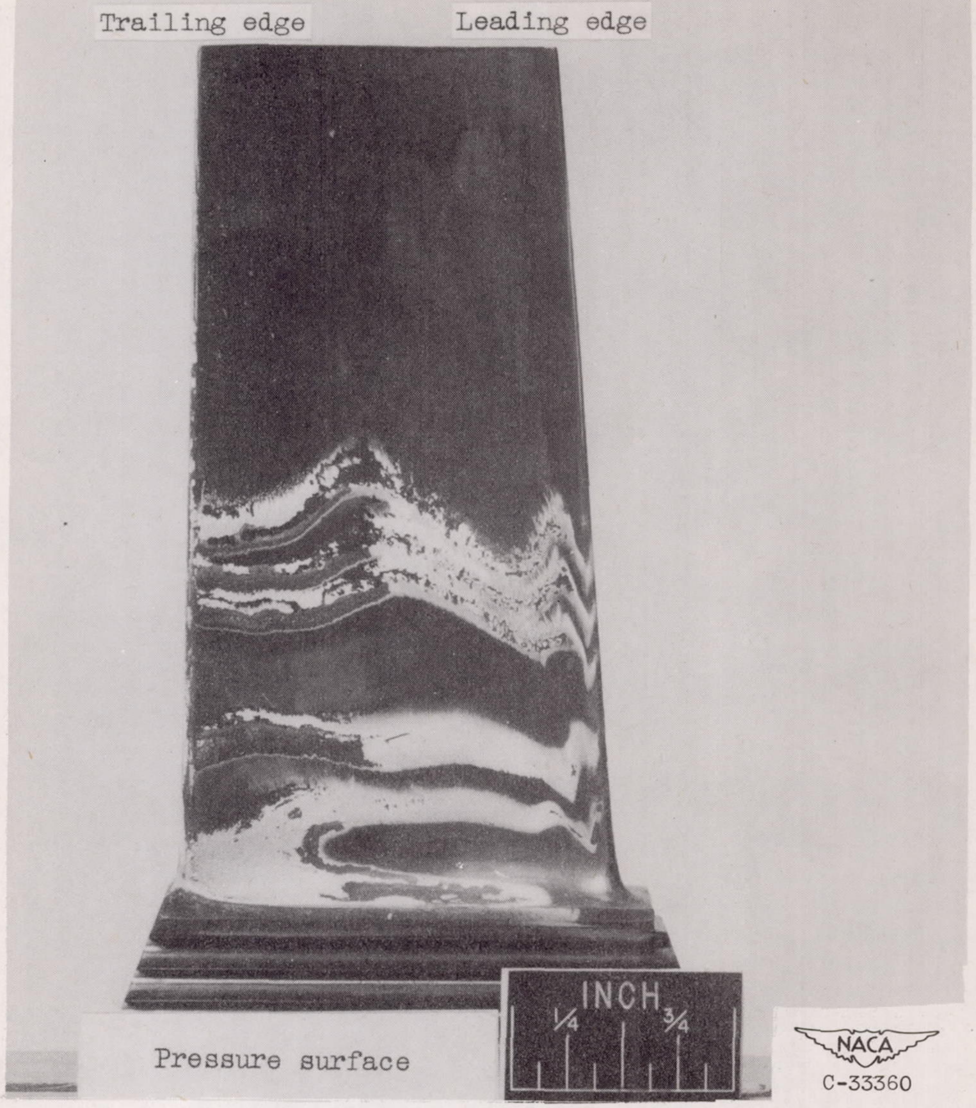
Figure 8. - Efficiency of spray cooling process plotted against coolant-flow parameter for one injection configuration investigated at several engine speeds. Configuration type - stator diaphragm inner-ring orifices near trailing edge, ID-4(L) and (S).

Figure 9. - Rotor blade after spray cooling operation showing trailing edge crack typical of blade failures encountered when large chordwise temperature differences were measured at midspan.



(a) Suction surface.

Figure 10. - Deposits on blade surfaces after engine operation at rated speed. Configuration type - stator diaphragm inner-ring orifices near trailing edge, ID-4(L) and (S).



(b) Pressure surface.

Figure 10. - Concluded. Deposits on blade surfaces after engine operation at rated speed. Configuration type - stator diaphragm inner-ring orifices near trailing edge, ID-4(L) and (S).

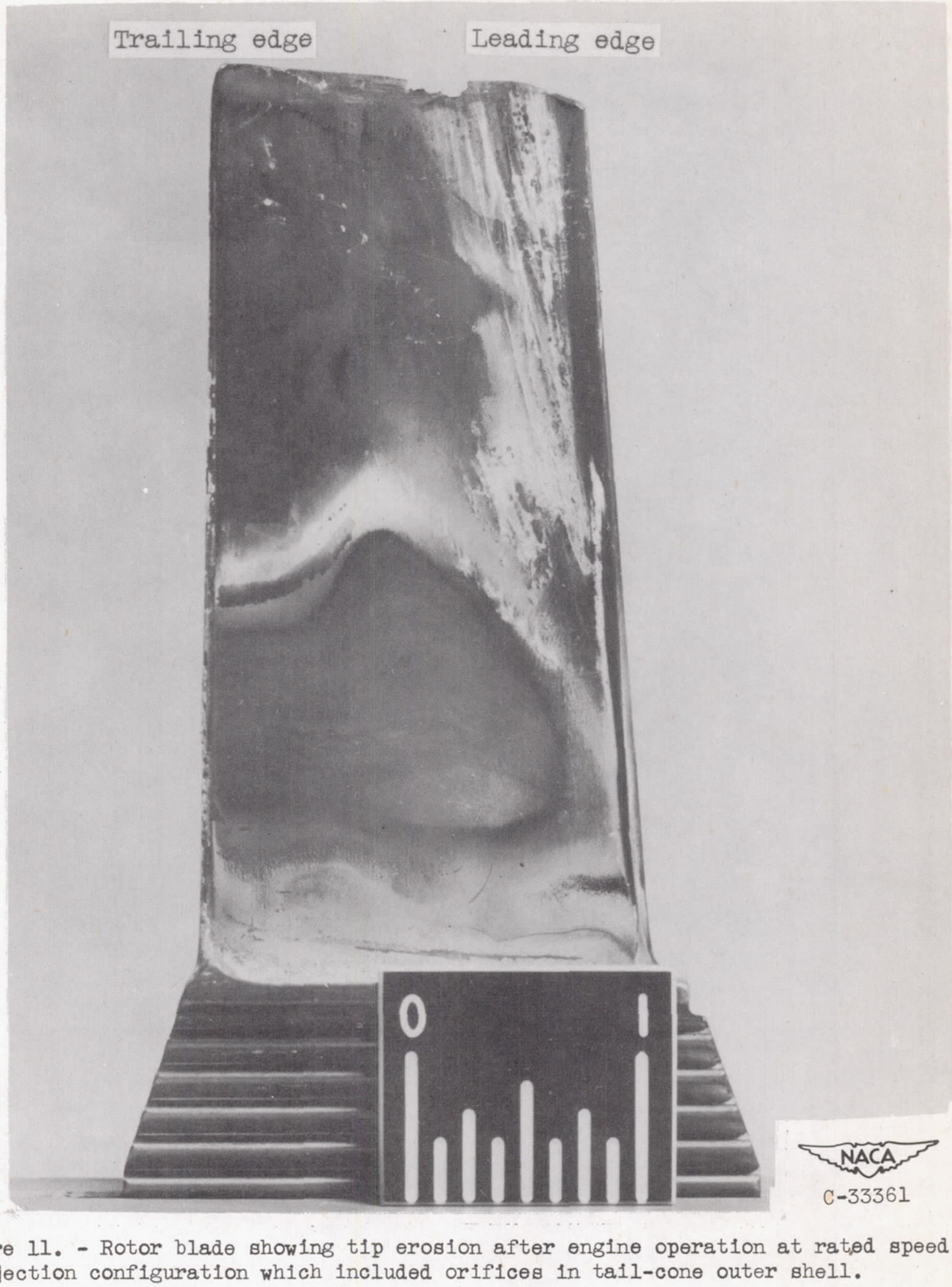


Figure 11. - Rotor blade showing tip erosion after engine operation at rated speed with injection configuration which included orifices in tail-cone outer shell.

Figure 12. - Inconel spray bar after approximately 1 hour of engine operation at rated speed with and without spray cooling.