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RESEARCH MEMORANDUM

STRUCTURAL DESIGN AND ENGINE EVALUATION OF AN AIR-COOLED TURBINE BLADE COMPOSED OF CORRUGATIONS AND A SEMISTRUT FOR OPERATION AT A TIP SPEED OF 1200 FEET PER SECOND

By Richard H. Kemp, André J. Meyer, Jr., and William C. Morgan

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NATIONAL ADVISORY COMMITTEE FOR AERONAUTICS

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STRUCTURAL DESIGN AND ENGINE EVALUATION OF AN AIR-COOLED TURBINE BLADE COMPOSED OF CORRUGATIONS AND A SEMISTRUT FOR OPERATION AT A TIP SPEED OF 1200 FEET PER SECOND

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SUMMARY

An air-cooled turbine rotor blade consisting of an internal part-span structural strut, corrugated cooling surfaces, and a thin outer shell was investigated analytically and experimentally. The effects on stresses and blade life of varying strut length, shell thickness, and corrugation dimensions are presented graphically. Blades were fabricated from two material combinations; X-40 and A-286 were used for one combination and S-816 and L-605 for the other for the strut and shells, respectively. During a 55-hour run in an engine at rated speed and temperature, cracks developed in the welded sections of the A-286 shells. The S-816 - L-605 blades were relatively undamaged after 100 hours at rated speed with cooling, 100 hours at rated speed without cooling, and after 300 cycles of stop-start operation. The cyclic test was designed to subject the cooled blades to thermal shock.

INTRODUCTION

A program to increase the structural reliability of experimental air-cooled turbine blades intended for aircraft propulsion systems was initiated at the NACA Lewis laboratory. The first report from this investigation disclosed a new blade construction (ref. 1). The engine used in the first tests imposed higher centrifugal stresses than occur in the majority of conventional engines; nevertheless, the blades endured satisfactorily. The same type of construction was used for another series of blades evaluated in another engine with operation characteristics more closely approaching present-day conventional jet engines. The tip speed was slightly less than 1200 feet per second compared with 1300 feet per second as previously reported in reference 1. The engine reported herein, however, has been known to impose more severe vibratory exciting forces on the turbine rotor blades than the engine of reference 1.

Analytical computations were made to establish the best proportions for the various blade parts. The outer-shell thickness, the corrugation amplitude and pitch, and the inner-strut length were varied in the calculations. Blades were fabricated in accordance with the analytical results and were run in an engine at maximum speed and temperature to determine their strength.

Solid uncooled blades recently have experienced leading-edge cracking attributed to thermal stresses and thermal shock (ref. 2). It has been supposed frequently that this problem would be more prevalent with cooled blades because larger temperature gradients would be present. To gain in- 4595
formation on this subject, these cooled blades were subjected to a cyclic operation known to cause cracking of conventional uncooled blades.

STRESS AND BLADE-LIFE COMPUTATIONS

The construction of the blade, the same as that reported in reference 1, is shown schematically in figure 1. An exploded view is shown in figure 2 which illustrates the relative locations of the strut, the corrugations, and the inner and outer shells. To retain a fair degree of turbine-compressor matching, the cross-sectional profile at the base of the cooled blade airfoil was made the same as that of the uncooled blade it was to replace. The tip cross-sectional area was increased slightly to permit escape of the expended cooling air. The stresses were determined for a tip diameter of 34.4 inches and a maximum speed of 7950 rpm. An airfoil length of 3.75 inches was assumed. Only the stresses due to centrifugal force were analyzed.

Stress Distribution in Blade

For the initial stress computations, sheet-metal thicknesses of 0.020, 0.005, and 0.007 inch were used for the outer shell, the corrugations, and the inner shell, respectively. The corrugation amplitude and pitch were both held to 0.050 inch, and a strut length of 1.77 inches was selected. The effect of varying some of these factors will be considered later in the report. The centrifugal stress distribution in the blade for the above conditions is shown in figure 3 with the cross-sectional areas used in the computations. The cross-sectional area at the tip end of the strut is assumed equal to the sum of the areas of the shell, the corrugations, and the strut. The area used at the base of the strut is that of the strut alone, because, at this point, the entire blade is supported only through the strut. Between the strut tip and base, the assumed effective area varies linearly.

The maximum centrifugal stress in the shell, 17,600 psi, is at the end of the strut. Immediately below this point, the strut shares the load, and the stress is reduced to 7800 psi. At the base of the strut the stress builds up to 23,400 psi, but the blade is considerably cooler here. The most critical point in the strut is slightly above rather than at the base because of the lower temperature in the base region. This will be discussed more fully in the following two sections.

Spanwise Temperature Distribution

In order to determine the critical or weakest points in the blade, the temperature distribution along the length of the blade must be known. These temperatures have not been measured accurately in this particular cooled blade design, and, therefore, the measured temperatures of conventional solid blades were used as a basis for the analysis. The most unfavorable condition for the cooled blades would be complete failure of the cooling system; under these adverse conditions, the cooled blade would soon attain about the same temperature profile as the solid blades. This profile has been measured by Morse and Johnston (unpublished NACA data) and is presented in figure 4 herein.

In order to determine the maximum blade life for varying blade proportions, the stress-rupture life for the blade materials must be used. For the low stress levels outlined in the previous section, rupture data are not available in the literature for the temperature range in question. Useful information was gained by using existing data and extrapolating as prescribed in reference 3; the results are presented in figure 5.

Stress-Rupture Life Along Blade Span

By using the stresses (fig. 3), the temperatures (fig. 4), and the material properties (fig. 5), the variations in life along the blade span, as shown in figure 6, can be determined. In the strut region, an assumed effective area is used, as mentioned previously, which takes into account the share of the load borne by the outer shell. In determining the blade life, however, the entire cross section was assumed to consist entirely of the strut material (S-816). This blade life is somewhat conservative since L-605 is stronger in stress-rupture than S-816. The two weakest points in the blade, indicated by the minimum points in the curves, are in the outer shell at the tip end of the strut and in the strut 3/4 inch above the platform. The shell in stress-rupture is much weaker than the strut, but in both places the minimum lives are exceptionally long, even in the uncooled condition. The strut could be lengthened, which would increase the shell life, reduce the strut life, and more nearly equate the two lives. For this engine application, however, it may be wise to make the strut conservative because vibratory stresses are highest at the base of the strut and rather low in the shell near the end of the strut.

The sharp rise in the temperature profile (fig. 4) near the blade base accounts for the minimum strut life occurring 3/4 inch above instead of at the base.

Effect of Strut Length and Shell Thickness on Blade Life

By assuming the condition of no cooling air at rated conventional engine conditions, the blade lives were interpolated for the calculated stresses for varying strut lengths. In making the stress calculations, 4595
the external profiles of the blade, the corrugation amplitude (0.050 in.), pitch (0.050 in.) and thickness (0.005 in.), and inner-shell thickness (0.007 in.) were all held constant. The results of these calculations are shown in figure 7 for one outer-shell thickness (0.020 in.). Additional calculations were made in which the outer-shell thickness was treated as a variable, and these results are shown in later figures.

In figure 7, the stress-rupture life is plotted as a function of the strut length. The blade-life limit, shown by the crosshatched area, is bounded by two separate curves; the solid line represents the shell life, and the broken line represents the strut life. For a blade with a strut length of 2.16 inches (the point of intersection of the two curves), the strut life is the same as the shell life (point A, fig. 7). For a blade with a strut length of 2.50 inches, the shell life is 46,000 hours, whereas the strut life is 8000 hours. The strut is therefore weaker and hence determines the blade life (point B). At a strut length of 1.55 inches the shell is weaker than the strut and gives a blade life of 4000 hours (point C). If a strut length below 1.10 inches is considered, the blade life will still be determined by the low point in the shell-life curve; for this reason, the crosshatching extends horizontally to the left. The maximum blade life possible would be 12,000 hours at the optimum strut length of 2.16 inches. For comparative purposes, the standard solid blade for this engine when operating at the same temperature as assumed for the cooled blade has a theoretical life of approximately 13,000 hours (ref. 2).

The results of the calculations in which the outer-shell thickness is assumed to be a variable are shown in figures 8(a) and (b). Four curves for different outer-shell thicknesses, ranging from 0.005 to 0.30 inch, are needed to represent the strut lives. A single curve represents the shell life, since the stress in the shell is independent of the thickness if the cross-sectional area of the shell does not vary along the length of the airfoil. Figure 8(a) shows that the optimum strut length increases as the shell thickness decreases. In addition, the maximum blade life greatly increases with decreased shell thickness. Blade life is also extremely sensitive to strut length. For example, with a 0.005-inch-thick shell, the optimum strut length is 2.56 inches; if the strut length is only 1.88 inches, the life is reduced to only one-tenth the maximum value attainable with this shell thickness. Overlength struts are not as detrimental as struts which are short.

Since all blade variations of this design are so conservative in regard to life in the engine investigated, the effect of raising the complete spanwise temperature profile 100° F was considered. The new blade-life diagram is constructed in figure 8(b). The plots appear quite similar to figure 8(a), but the ordinate values in figure 8(a) are 10 times higher than those in figure 8(b). The optimum strut length is 2.22 inches for the blade with a 0.020-inch-thick shell and results in a blade life of 1000 hours. This blade life is reduced by a factor of 12 for the 100° F higher temperature.

4595 The effect of outer-shell thickness is observed more directly in the summarizing curves in figure 9. In view of the long stress-rupture lives indicated in figures 8 and 9, it is informative to determine the stress-rupture safety factors at the critical sections in the strut and shell. This can be done by comparing the actual blade stresses with the stresses for rupture in 1000 hours for the blade materials at the presumed operating temperatures. This information is contained in table I for the material combination of S-816 (strut) and L-605 (shells and corrugations). Two sets of temperature conditions were again assumed; the normal spanwise temperature profile of the standard uncooled blade and a profile raised 100° F from the normal temperature at every span location. The data shown represent the stress conditions for the optimum strut length for each shell thickness and temperature distribution. For each of the two temperature conditions the 1000-hour rupture stress for the material at the operating temperature of the critical section is divided by the actual operating stress at the critical section to give the stress safety factor. Table I shows that appreciable increases in reliability can be theoretically achieved by reducing the outer-shell thickness. However, many other factors such as vibration susceptibility, lack of impact resistance, and welding problems along the leading and trailing edges severely limit the reduction of outer-shell thickness.

Variation of Blade Weight for Various Strut Lengths

In the preceding section it was shown that the blade design was very conservative. Other advantages might be gained by compromising the excess strength. One such advantage might be an appreciable reduction in blade weight. Computations were made of the weight of the cooled blade, and the results are shown in figure 10. The blade weight is given as a function of the strut length and the shell thickness. Relatively large changes in either the strut length or the outer-shell thickness do not appreciably affect the blade weight.

For the cooled blade with a 0.020-inch-thick shell and optimum strut length (2.16 in.), the blade weight would be 0.55 pound (fig. 10). For the blade with a 1.77-inch strut, tested in the engine, the weight was only reduced 0.03 pound per blade to 0.52 pound. Halving the outer-shell thickness to 0.010 inch and again using the optimum strut length

(2.45 in. for this case) only reduced the total blade weight 0.01 pound. For comparative purposes, the solid standard turbine blade for this particular engine has an average weight of 0.60 pound.

The advantage of weight reduction by shortening the strut is offset by increased centrifugal stresses in the shell (fig. 11) and increased shear stresses in the braze (fig. 12). Shortening the strut lowers the stresses in the strut but increases the stresses at the critical point in the shell at a much faster rate. The shear stress at the brazed joint also increases with shortening of the strut. With struts longer than $1\frac{1}{2}$ 4595
inches the shear stresses are very conservative, but for strut lengths shorter than $1\frac{1}{2}$ inches these shear stresses increase at a very rapid rate with further shortening. The ultimate shear strength of the braze at 1200° F is approximately 30,000 psi (ref. 4). The values of figure 12 are based on 100-percent effective shear area, however, and, because of machining tolerances and the complexity of braze technique, 100 percent cannot be guaranteed. From 50 to 75 percent can be relied upon for design purposes.

In consideration of the important factors, shortening of the strut to achieve the slight weight reductions does not appear to be warranted in view of the sizable increases in stresses of other blade components.

Effect of Various Corrugation Amplitudes

A series of stress computations and plots similar to figure 8 was completed for several different values of corrugation amplitude. From these graphs the maximum blade lives corresponding to the optimum strut lengths were determined for a constant corrugation pitch of 0.050 inch. The information is summarized in figure 13(a) for the temperature profile of uncooled blades and in figure 13(b) for temperatures 100° F higher. The straight horizontal parts of the curves occur where the shell life alone controls the blade life.

Reductions in corrugation amplitude increase blade life by reducing corrugation weight, increasing the strut cross-sectional area and, thus, reducing the centrifugal stresses. The effect is appreciable, as noted in the curves; but, reduced cooling efficiency, increased pressure drop, and the tendency for the smaller passages to become blocked limit the amount that the amplitude can be feasibly reduced (ref. 5).

Effect of Varying the Pitch of Corrugations

Similar computations were also made with corrugation pitch as the variable, and the amplitude held constant at 0.050 inch. Outer-shell thicknesses of 0.010 and 0.020 inch were assumed. Figure 14 shows the results for both the normal and 100° F above normal uncooled temperature profiles. The effect is not as pronounced as varying the amplitude. The larger the pitch, the longer will be the blade life. This effect is due to the decrease in the weight of the corrugations as the pitch is increased. Caution must be exercised, however, when increasing pitch, because the braze shear area carrying the outer shell is inversely proportional to the pitch. When the pitch is increased, the number of corrugation apexes, which are the points or lines of contact between the corrugations and the shell and strut, is reduced. The effect on the cooling characteristics is discussed in reference 5.

BLADE CONSTRUCTION AND MATERIALS

Blades were fabricated using two different combinations of materials. One of these combinations, L-605 (Haynes No. 25) for the shells and corrugations and S-816 for the strut, has already been discussed in the design analysis portion of this report. The selection of these materials was governed by physical strength characteristics (short and long times) and fabrication technique requirements such as formability, weldability, and brazability. The L-605 was used in the form of fully annealed rolled sheet, while the S-816 struts were fabricated by machining from standard S-816 forged turbine blades. This necessitated a reduction in strut length from the optimum value of 2.16 inches to a practical value of 1.77 inches. The blade components were assembled with a high-temperature braze composed of 20 percent chromium, 10 percent silicon, 1 percent iron, and the balance of nickel.

In addition to the previously described material combination, blades were also made using cast X-40 struts and A-286 shells and corrugations. Although the stress-rupture properties of these materials are somewhat inferior to those of the S-816 - L-605 combination in the temperature range to which the materials would be subjected, certain advantages suggested their inclusion in the program. For example, the use of X-40, a casting alloy, simplifies the fabrication procedure for producing the strut, while the use of A-286 for the shells and corrugations considerably reduces the strategic alloy content of the completed blade, an important consideration for missiles or expendable engines.

The fabrication procedure for these blades is substantially the same as that which is discussed in reference 1. Briefly, the outer shells (0.020 in. thick, fig. 2) are stretch-formed in halves and are first brazed to their respective corrugations and inner shells in dies which hold the parts in contact with each other. A soft fibrous refractory padding material is used between the dies and the sheet metal parts to assure a uniform pressure while being brazed. The corrugations were pressed from 0.005-inch-thick material with a pitch and an amplitude of 0.050 inch each. The inner shells were press-formed from 0.007-inch-thick stock.

The assembled suction and pressure halves are brazed to the strut, again by using refractory padding and clamping dies. The leading and trailing edges are shielded-arc-welded without a filler rod, and a sheet-metal cover which forms the blade platform and cooling air seal is fitted to the bottom of the outer shell. This cover is brazed to the blade in a third brazing cycle. The brazing cycle consists of heating to 1800° F and holding for 10 minutes, followed by rapid heating to 2130° F in a 1- to 4-micron mercury vacuum and holding for 15 minutes. As a final step, the tips of the blade are trimmed to length.

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ENGINE INSTALLATION

The cooled blades were operated in a turbojet engine having a turbine tip diameter of 34.4 inches and a turbine tip speed of 1195 feet per second.

Two methods of mounting the cooled blades in the rotor were employed. The first method is shown in figure 15, and a close-up view is shown in figure 16. Two blades of the X-40 strut - A-286 shell combination were operated in this manner. A serrated hollow slug occupies the space ordinarily taken by two standard blades. The slug was designed to allow the investigation of a wide variety of different root attachments. The serrated base of the cooled blade strut in turn mates with serrations cut on the inside of the slug. The cooling air is delivered to the bottom of the hollow slug and is kept from escaping by the sheet-metal rim cap brazed to the base of the airfoil and to the top and ends of the slug.

Two blades of the S-816 strut - L-605 shell combination were mounted in the rotor as shown in figure 17. In this case, the slug base was eliminated, and the serrated strut of the cooled blade was inserted directly into the rotor serration. A sheet-metal rim cap (fig. 18) was brazed to the base of the airfoil and bent over the rim of the rotor where attachment was completed by welding. The two adjacent standard solid blades were grooved slightly under their platforms so as to slide over and tightly seal the sheet-metal platform of the cooled blade. This method of mounting the cooled test blades in the rotor is the same as that described in reference 1.

Cooling air was supplied to the blades by means of tubes welded to the downstream face of the wheel as shown in figure 15. The cooling air supply tubes entered a plenum chamber at the center of the wheel; in figure 15 the cover plate of this chamber has been removed. An external source of air was connected to the plenum chamber by means of the system shown in figure 19. Air is introduced at the front of the engine into a tube passing through the center of the engine to the plenum chamber on the turbine wheel. At the position where the rotating tube extends into the stationary housing, a double labyrinth seal is provided. Leakage of blade cooling air is held to a negligible quantity by providing air from a separate source to the center of the seal at the same pressure that exists inside the housing. Static-pressure taps in the center of the seal and in the housing permit this balance to be effected. The quantity of cooling air supplied to the blades could therefore be measured accurately by means of a flat-plate orifice installed in the air line according to ASME specifications.

TEST PROCEDURE

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Engine operation of the cooled blades of the X-40 - A-286 combination was limited to a steady-state endurance test at rated engine speed and a turbine-inlet gas temperature of 1650° F. The cooling airflow to each of the cooled blades was set at 2 percent of the combustion gas mass flow handled by each of 96 blades of the engine at rated conditions.

CG-2 Engine operation of the cooled blades of the S-816 - L-605 combination was divided into three parts. The first part consisted of a steady-state endurance operation at rated speed and an inlet gas temperature of 1650° for 100 hours. Of this time, 33 hours were obtained with a cooling airflow to each cooled blade of 5 percent. The last 67 hours were obtained with a cooling airflow of 2 percent. Again, the airflow values refer to the percent of the combustion gas mass flow handled by each of 96 blades of the engine at rated conditions. The second part of the operation of these blades consisted of cyclic starting and stopping of the engine with occasional short runs up to rated speed as a rough check on the structural soundness of the blades. The cooling airflow was maintained always at 2 percent of the gas mass flow handled by each blade of the engine at rated conditions. The engine was started in the normal manner and accelerated to idle speed, which was held for 5 minutes. Fuel to the engine was shut off instantaneously, and, after rotation ceased, the engine was allowed to stand for 20 minutes with the cooling air flowing. The blades were therefore subjected to a thermal-shock cycle, probably somewhat more severe than they normally would encounter if they were supplied by compressor bleed air. At intervals of 50 cycles the blades were inspected for cracks by a penetrant-oil method.

For the third part of the operation, the S-816 - L-605 combination blades were subjected to endurance-running for a period of 100 hours at rated speed and tailpipe temperature with the cooling air to the blades completely shut off. This was done to obtain some indication of the strength of the blade when operating at actual metal temperatures similar to those that would be encountered if the blades were operated with cooling air in an engine with a much higher turbine-inlet temperature. It is recognized that the uncooled operation will not produce the identical temperature pattern that would be experienced by the cooled blade at advanced turbine-inlet temperature and, hence, would not simulate the thermal stress condition. However, it is believed that all other loading conditions are simulated to a fair degree and that a 100-hour survival would be a good indication of the structural integrity of the design.

ENGINE EVALUATION RESULTS

X-40 Strut - A-286 Shell Combination

Two blades of this type were operated in a steady-state endurance run at rated speed and tailpipe temperature with 2-percent cooling air. 4595 After 55 hours and 37 minutes of rated operation, a cooling air supply tube on the face of the rotor tore loose and terminated the test run. A photograph of the two blades after the engine operation is presented in figure 20. Although the blades were still serviceable, there was evidence of deterioration as shown in the photograph. Essentially two types of damage are evident; one is due to impact and the other is due to weld-cracking. The impact damage is prominent only on the suction surface near the leading edge. Markings in this area indicate that the foreign-particle size varied from small particles which barely nicked the surface to those large enough to indent the shell as much as 0.030 to 0.050 inch. This type of damage was not obtained when the same combination of blade design and shell thickness was operated in a different engine in the same test stand (ref. 1). A possible explanation may be the formation of hard carbon particles in the combustion chambers of the engine used herein.

The second type of damage noted was cracking in the welds of the leading and trailing edges. In the trailing edges of both blades, cracks started in the weld at the tip and progressed downward from 1/2 to 3/4 inch. On the leading edge of both blades, a crack started in the weld $2\frac{1}{4}$ inches from the tip and progressed in the chordwise direction approximately $\frac{1}{4}$ inch on the pressure surface. On the leading edge of blade B (fig. 20), a crack started in the weld at the tip and progressed downward about 1 inch. A small portion of the shell at the tip was subsequently lost as can be noted in figure 20. Also, small cracks in the rim cap are evident.

Although the struts or the main load-carrying member of these blades were probably sound and the shell cracks could have been repaired, no further running was done with these blades.

S-816 Strut - L-605 Combination

Steady-state endurance operation with cooling air. - Two blades of this type were operated, as noted previously, under two different sets of conditions. They were first subjected to steady-state endurance operation at rated speed and temperature for 100 hours. The first 33 hours were run at a coolant-flow value of 5 percent; the remaining 67 hours were run at a coolant flow of 2 percent. A photograph of one of the blades after the steady-state endurance operation is shown in figure 17. Some indenting of the shell at the leading edge on the suction surface was noted. Again, it is suspected that the foreign particles responsible were hard carbon formed in the combustion chambers. A small crack was again evident in the rim cap of each blade in the trailing-edge region. It is believed that these cracks resulted from stresses induced by the blade repositioning itself in the serration during running. Since the rim cap was welded to the rim of the wheel while stationary, stresses would be imposed when the blade shifted because of the centrifugal forces. This is a minor item associated only with the particular way in which the blade was mounted in this wheel. In a full-scale design this problem would not exist. In all other respects, the two blades were in very good condition at the end of the 100 hours.

Cyclic endurance operation. - After the 100 hours of steady-state endurance operation, the same two blades (without reworking of any kind) were subjected to a cyclic operation to obtain an indication of their resistance to thermal shock. The cycle consisted basically of starting and stopping the engine in the normal manner while maintaining, at all times, a coolant flow of 2 percent (based on the engine mass flow per blade at rated conditions). The engine was allowed to remain at rest for 20 minutes before the next start to ensure the return of the blade temperature to room value. Inspection at regular intervals consisted of visual and penetrant-oil surveys. In addition, the engine was operated briefly at rated conditions at regular intervals as a further check on the structural integrity. At the end of 300 cycles of operation the blades were still in good condition as shown in figure 21. This photograph is a combined natural-light and black-light double exposure to show the results of the final penetrant-oil inspection. Very small indications are noted along the leading edge of blade A. It is doubtful that these are actual cracks; they are probably false indications caused by irregularities in the surface produced by foreign-particle impact. The white spot at the tip of blade A near the trailing edge is definitely a false indication.

Leading-edge cracking of solid blades induced by thermal cycling has been a problem in the engine in which the cooled blades were operated. Many standard solid blades have shown evidence of cracking in the same time of operation to which the cooled blades were subjected (ref. 2). The final condition of the cooled blades after both the steady-state and cyclic operations indicates a resistance to thermal shock which is at least comparable to, if not better than, the standard solid blades.

Steady-state endurance operation without cooling air. - After the steady-state endurance operation with cooling air and the cyclic endurance operation, the same two blades were operated for 100 hours at rated speed and tailpipe temperature with the cooling air completely shut off. At the end of the 100 hours, the blades were still in good condition except for the minor indentations on the leading edge caused by foreign-particle impact as discussed previously. The excellent operating experience obtained with the two blades in all three modes of operation indicates good structural integrity of the design and suggests considerable promise for operation in an engine having advanced turbine-inlet temperatures.

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CONCLUDING REMARKS

For the type of cooled blade design described herein, the stresses and, consequently, the blade lives for the temperature profile analyzed are extremely conservative. This fact can be utilized in four ways. (1) The high safety factors existing at normal temperatures would allow a nonreplaceable type of blade attachment to be used which, if fully exploited, should produce a large savings in the weight of the turbine assembly. (2) The blade temperature can be allowed to increase commensurately with higher than normal inlet gas temperatures. (3) The amount of cooling air needed for the turbine rotor blades can be reduced for a given inlet gas temperature. (4) Less critical materials can be used for blade construction, particularly for expendable missile-type applications.

SUMMARY OF RESULTS

An air-cooled blade design having combined features of the strut and the shell-supported types was stress-analyzed and then fabricated and operated in a turbojet engine having a 34.4-inch turbine tip diameter and a turbine tip speed of 1195 feet per second.

In the stress analysis the following results were obtained for the S-816 strut - L-605 shell combination and an assumed spanwise temperature distribution:

1. The blade life is dependent on the strut length and reaches a maximum at a particular strut length for a given shell thickness.

2. The blade life can be increased markedly by decreasing the outer-shell thickness or decreasing the corrugation amplitude or both. The effect of varying the corrugation pitch was not so pronounced.

3. For the particular turbojet engine chosen, it was calculated that, for a 0.020-inch-thick shell, the optimum strut length would be 2.16 inches, which would give a maximum life of 12,000 hours at conventional

blade temperatures. If the blade temperature level was increased by 100° F, the blade configuration would optimize at a strut length of 2.22 inches with a life of 1000 hours.

Blades of two different material combinations were fabricated and operated in the turbojet with the following results:

1. Two blades having cast X-40 struts and A-286 shells were operated for approximately 55 hours at rated speed and temperature with 2-percent cooling air. At the end of that time both blades showed deterioration in the form of foreign-particle impact and cracking of the weld along the leading and trailing edges.

2. Two blades having S-816 forged struts and L-605 shells were operated for 100 hours at rated speed and temperature. Operation for 33 hours was at a cooling flow rate of 5 percent and for 67 hours at 2 percent. At the end of the 100 hours, the blades were still in good condition except for minor indenting of the suction surface near the leading edge due to foreign particle impact. The welds were in good condition.

3. The same blades which were operated for 100 hours were subjected to a cyclic endurance operation consisting of starting and stopping the engine to produce thermal-shock conditions. At the end of 300 cycles, the blades were still in good condition except for the minor impact indentations.

4. The same two blades were also subjected to 100 hours of steady-state endurance operation at rated engine speed and temperature with the cooling air shut off. After the 100 hours of operation, the blades were still in good condition with the exception of the minor indenting of the leading edge.

Lewis Flight Propulsion Laboratory
National Advisory Committee for Aeronautics
Cleveland, Ohio, August 8, 1957

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TABLE I. - COMPARISON OF ACTUAL OPERATING STRESSES IN COOLED BLADE CRITICAL SECTIONS WITH 1000-HOUR STRESS-RUPTURE DATA FOR THE BLADE MATERIAL COMBINATION S-816 (STRUT) AND L-605 (SHELLS AND CORRUGATIONS) AND FOR TWO DIFFERENT TEMPERATURE PROFILES

Critical stress location	Actual operating stress at critical section for normal temperature, psi	1000-Hour rupture stress for normal temperature, psi	Stress safety factor for normal temperature and 1000-hour operation	Actual operating stress at critical section for temperature 100° F above normal, psi	1000-Hour rupture stress for temperature 100° F above normal at critical section, psi	Stress safety factor for temperature 100° F above normal and 1000 hours
In strut for -						
0.010-In. shell	18,200	30,500	1.68	18,200	21,800	1.20
0.020-In. shell	21,600	30,500	1.41	21,800	21,800	1.00
0.030-In. shell	25,000	30,500	1.22	25,900	21,800	.84
In shell for -						
0.010-In. shell	12,400	22,600	1.82	12,400	15,700	1.27
0.020-In. shell	15,000	22,000	1.47	14,500	15,000	1.03
0.030-In. shell	19,600	23,500	1.20	18,100	16,400	.91

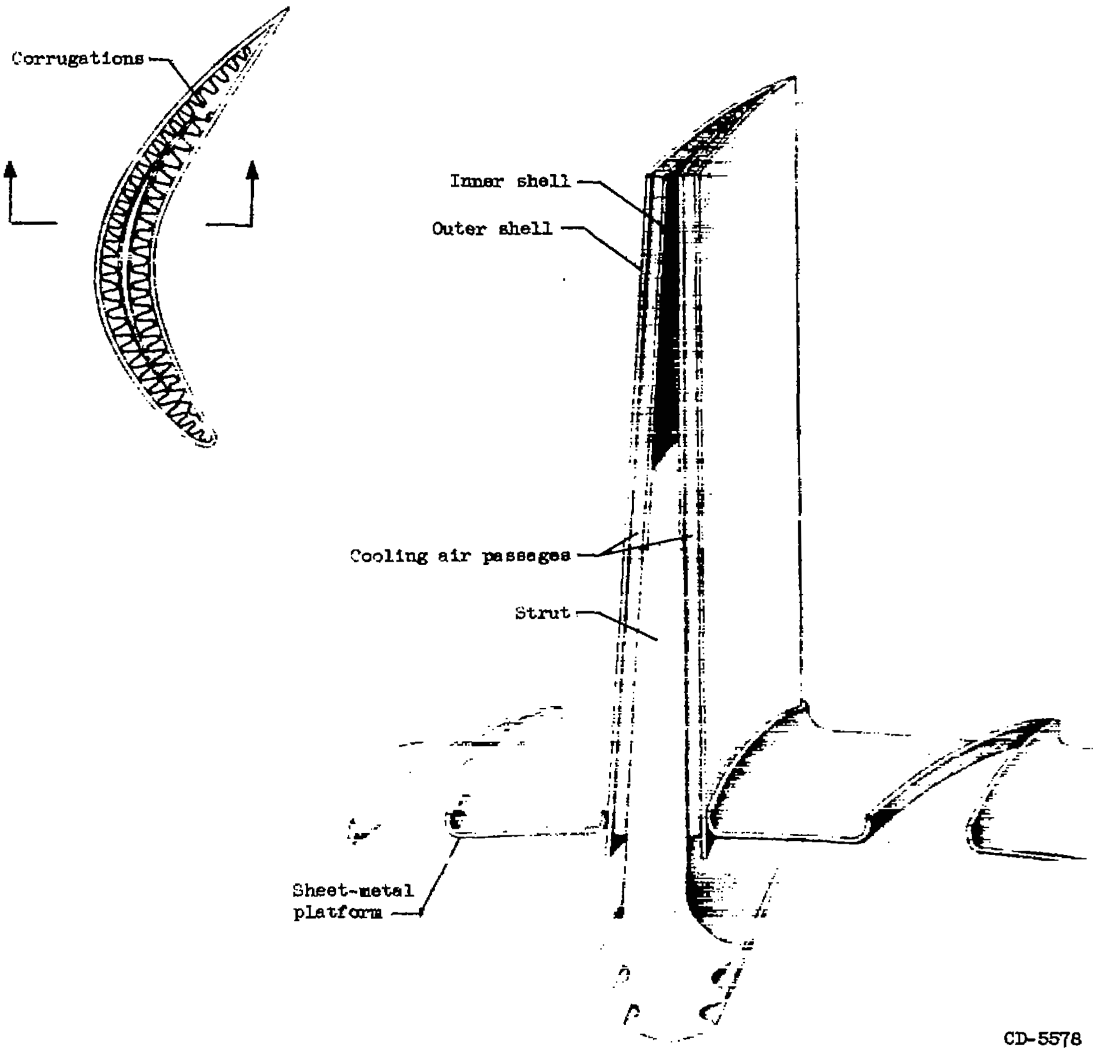

Figure 1. - Air-cooled blade construction.

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CG-3

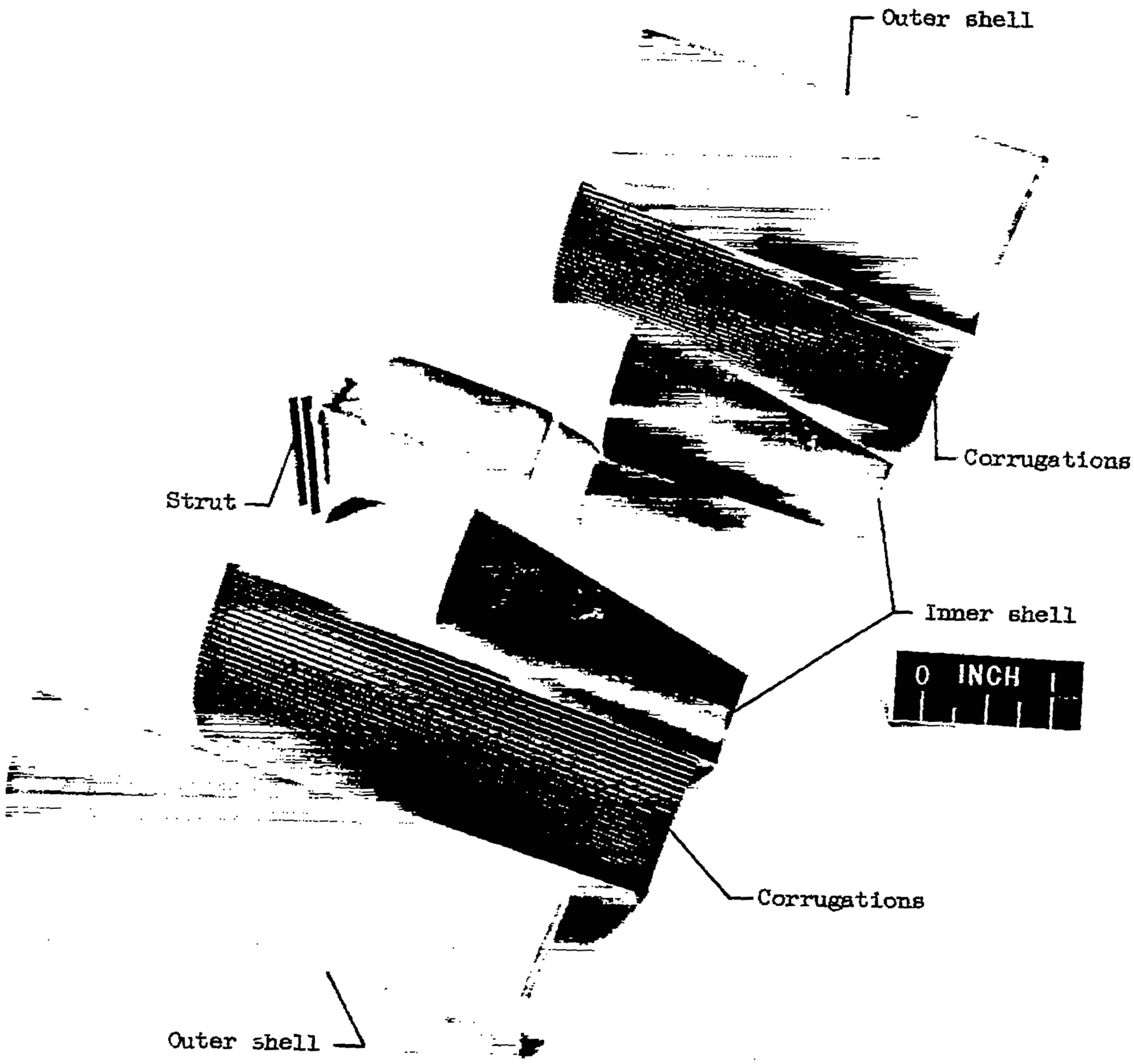

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Figure 2. - Exploded view of cooled blade components.

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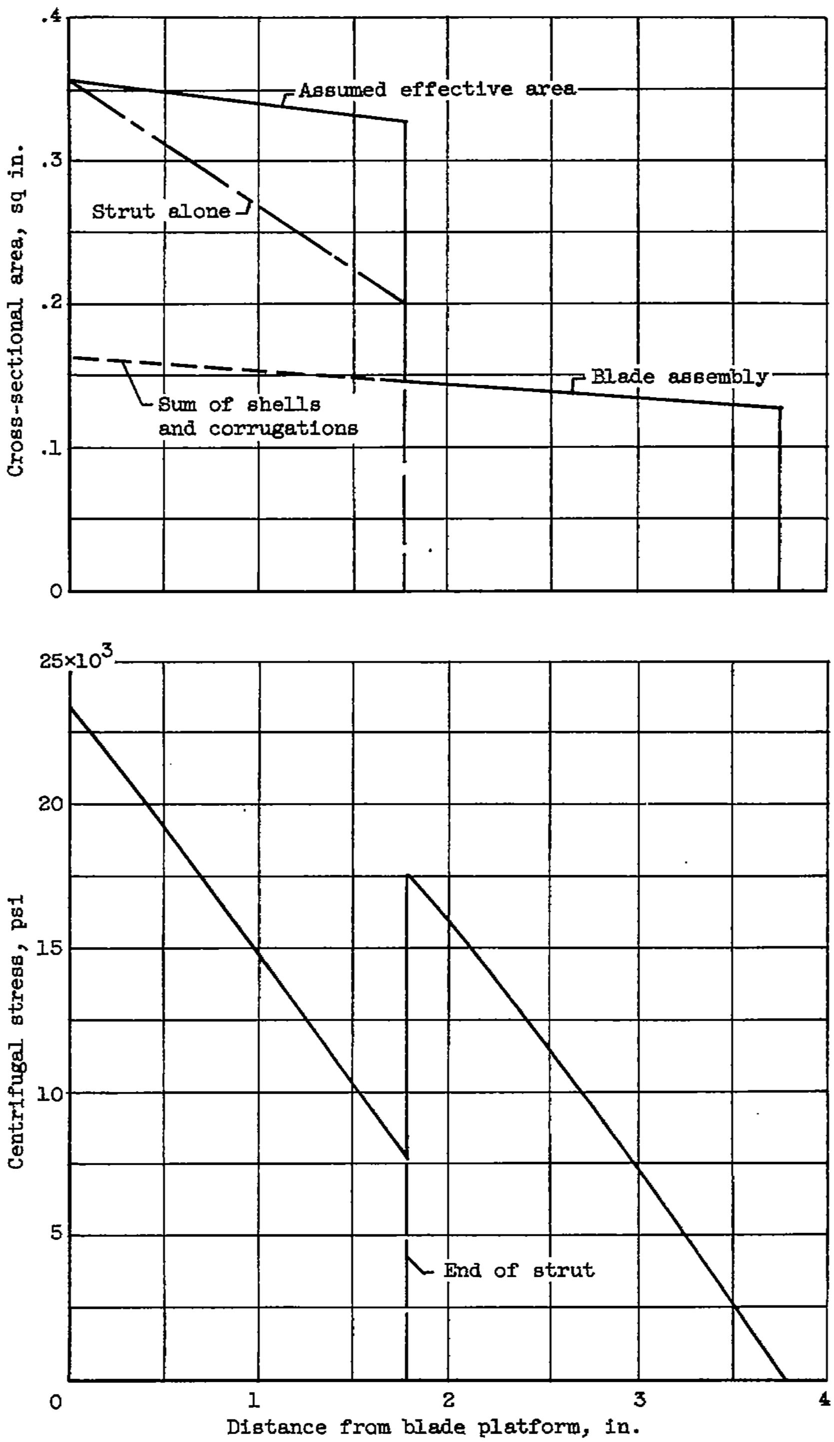

Figure 3. - Centrifugal stress and area distribution along cooled blade span.

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CG-3 back

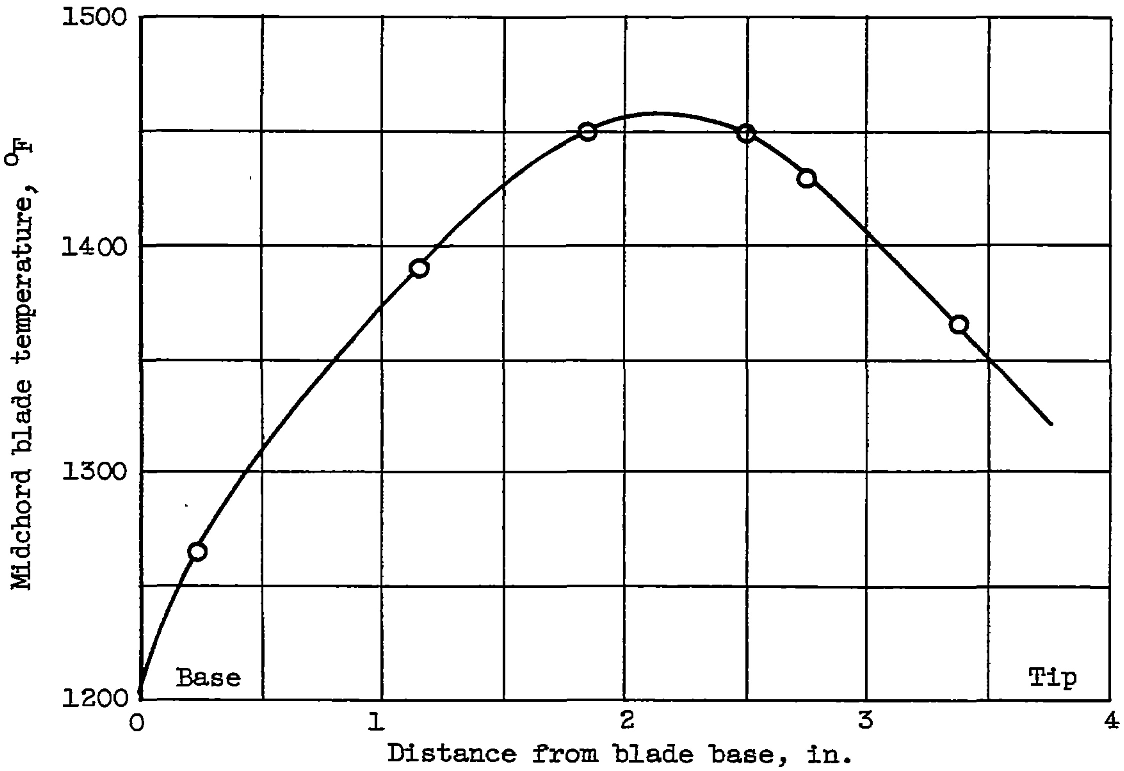

Figure 4. - Temperature distribution along span of uncooled solid blade at rated conditions of conventional engine (unpublished NACA data).

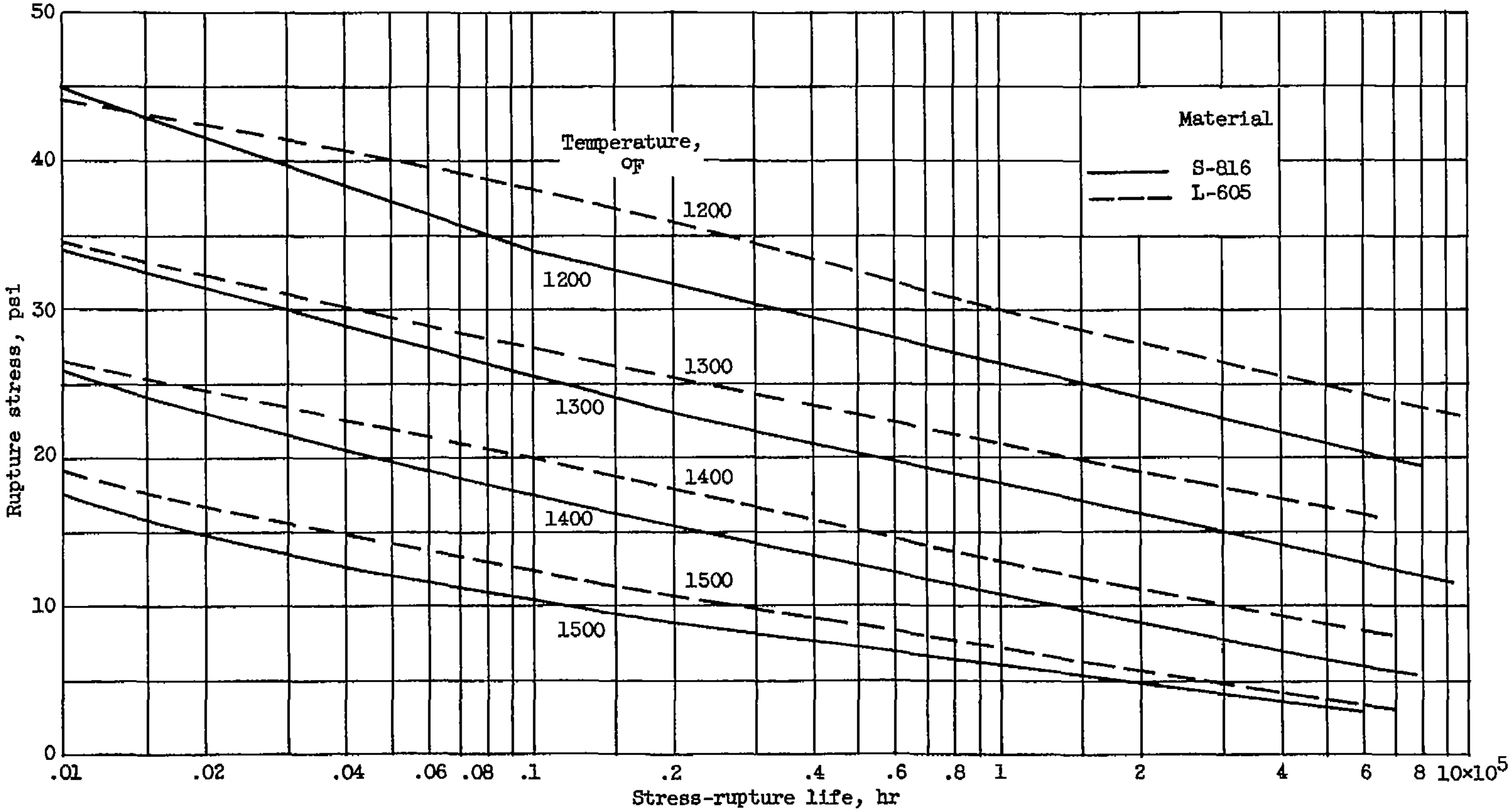

Figure 5. - Extrapolated stress-rupture properties of cooled blade materials.

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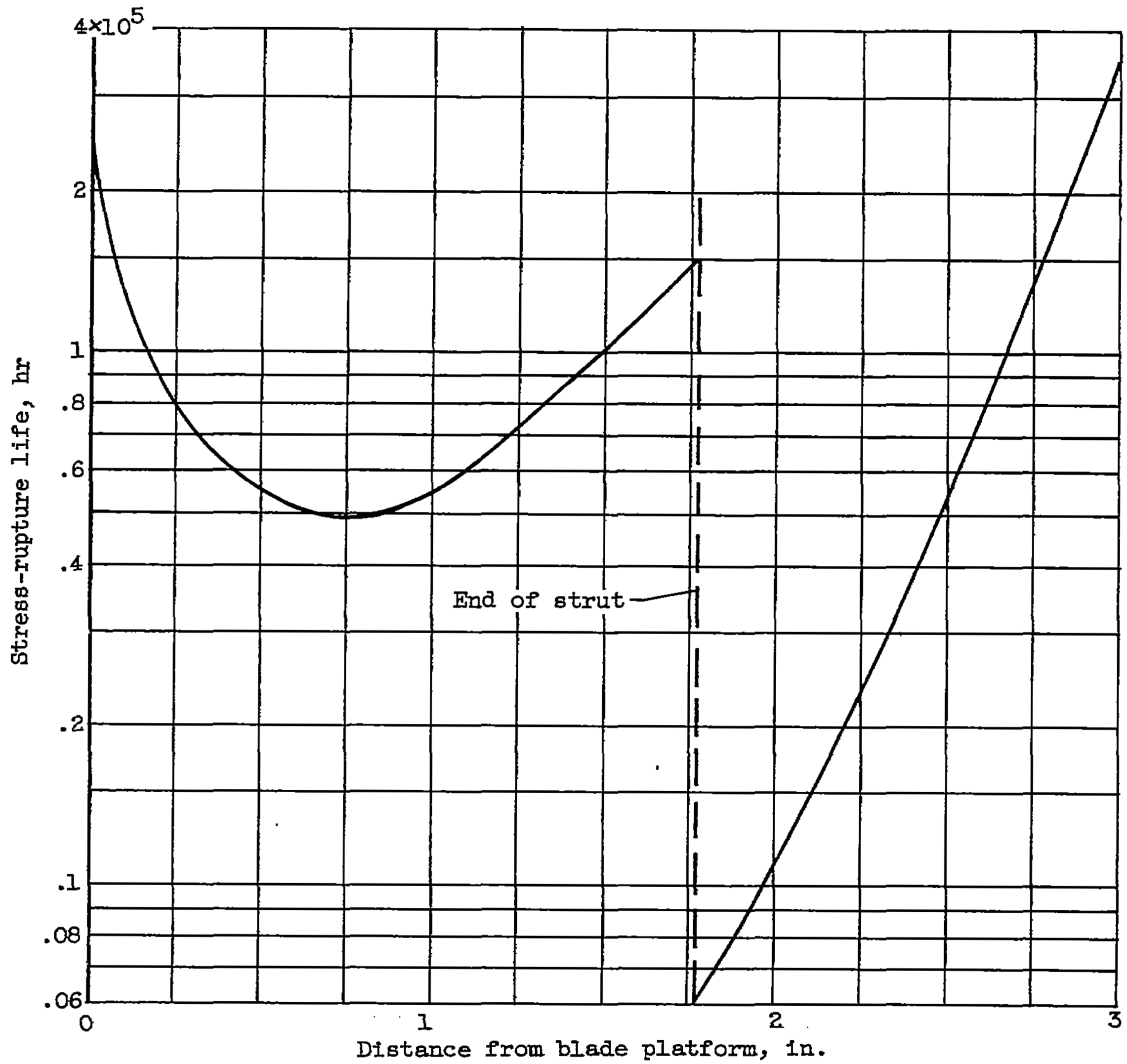

Figure 6. - Stress-rupture-life variation along span of cooled blade for assumed temperature profile. Strut length, 1.77 inches; outer-shell thickness, 0.020 inch.

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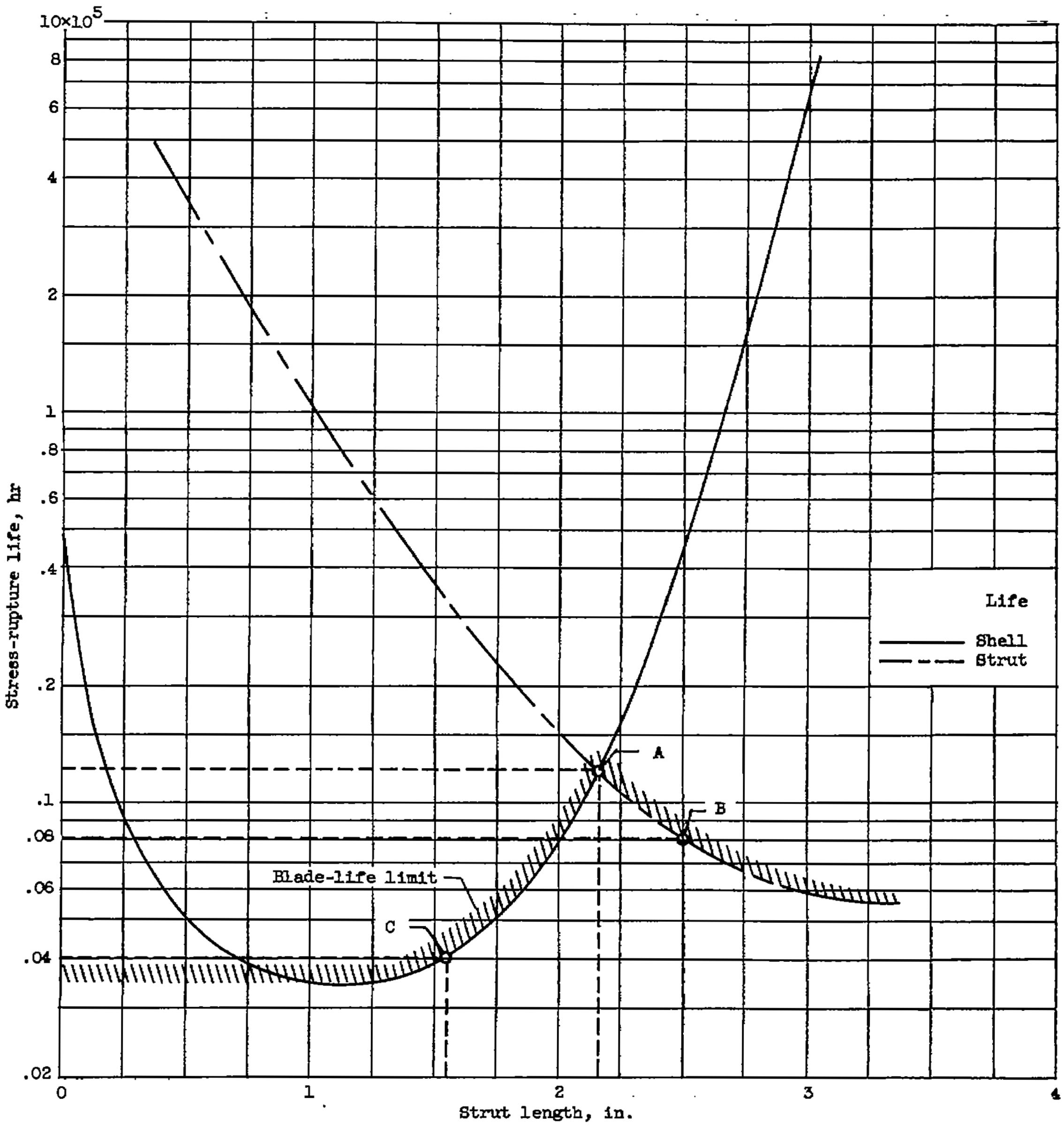

Figure 7. - Variation in blade life with strut length for 0.020-inch outer-shell thickness.

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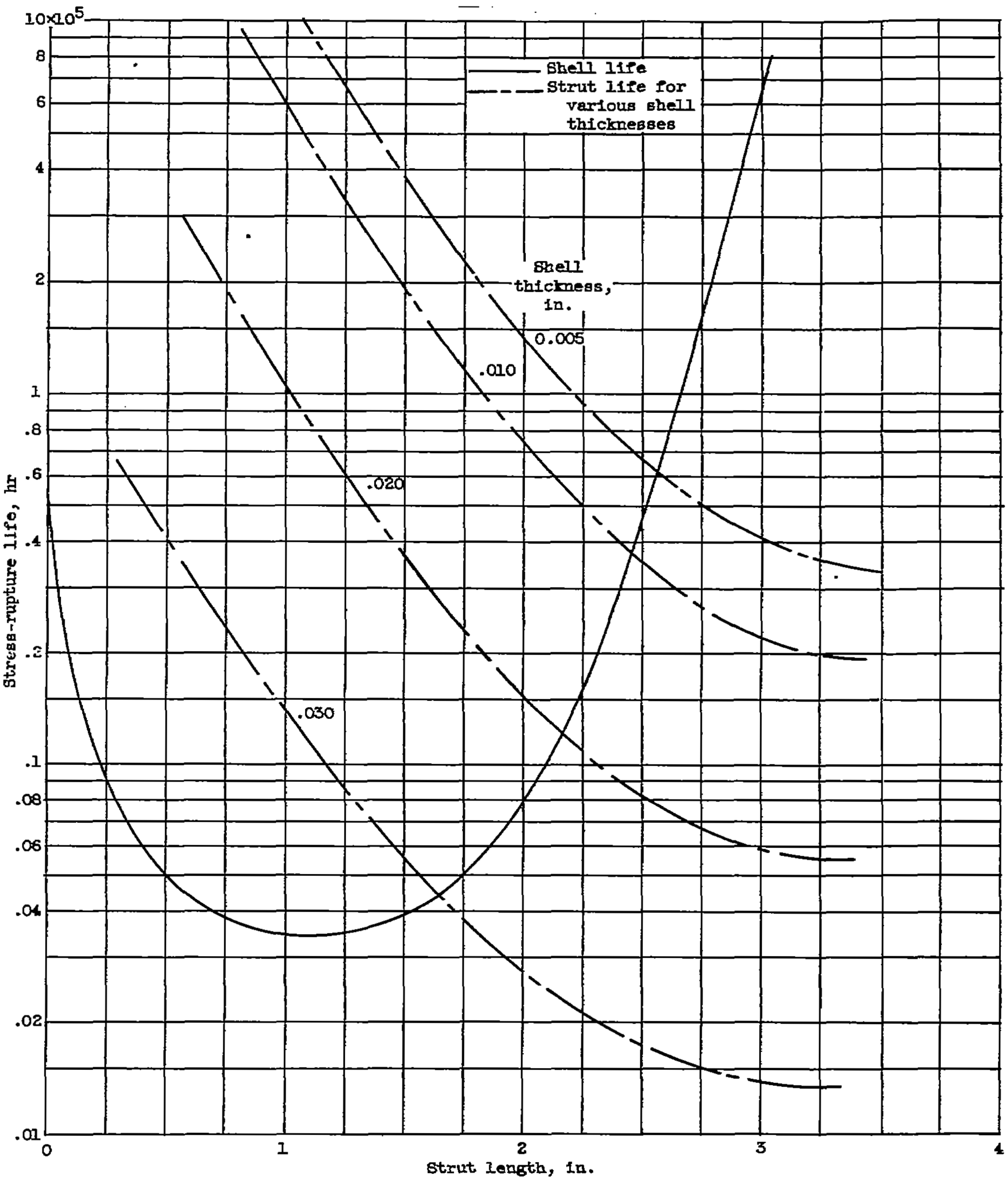
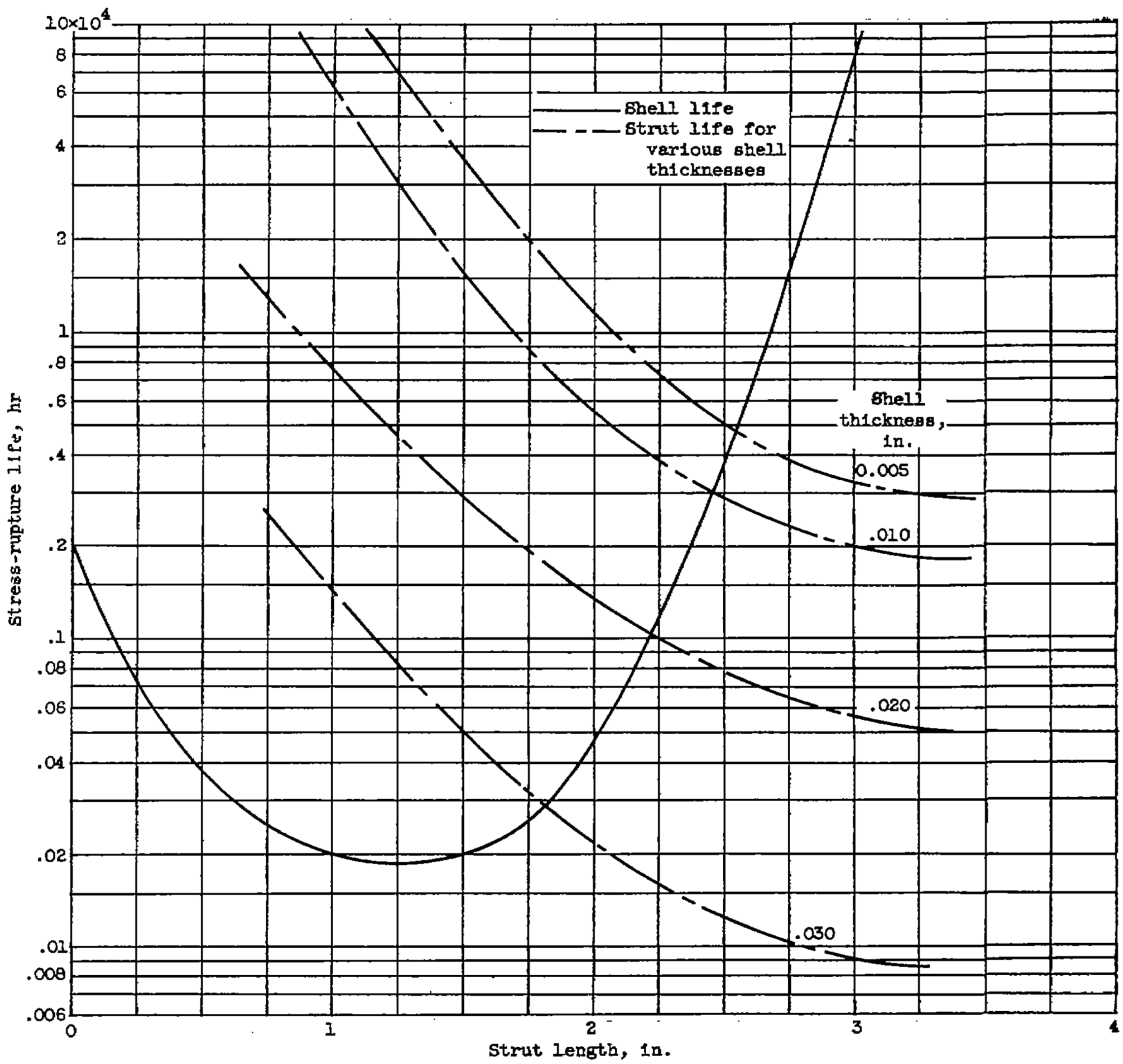


(a) Normal uncooled blade temperatures.

Figure 8. - Variation in blade life for various outer-shell thicknesses and strut lengths.



(b) Temperatures 100° F higher than normal.

Figure 8. - Concluded. Variation in blade life for various outer-shell thicknesses and strut lengths.

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CG-4

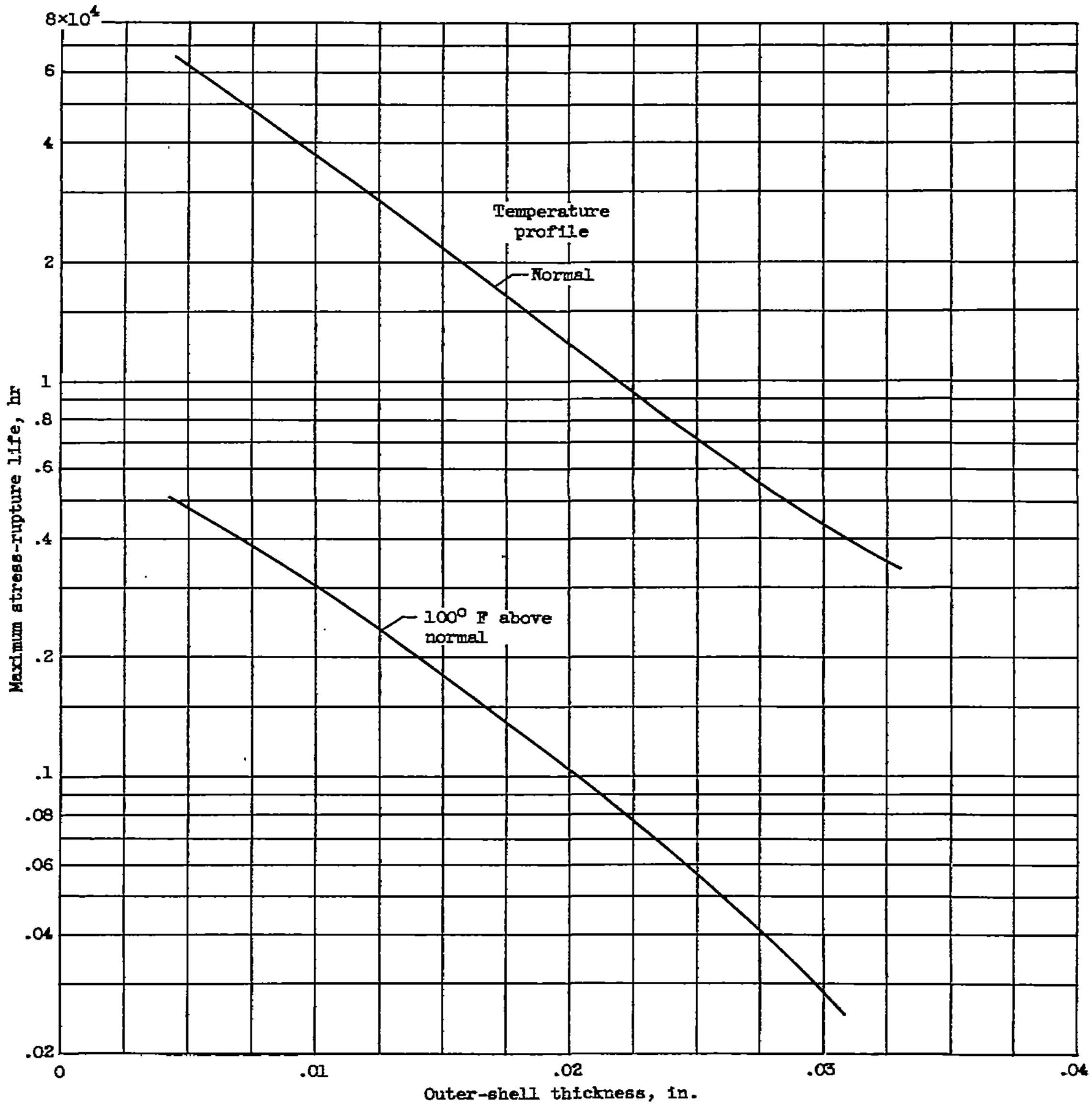

Figure 9. - Effect of outer-shell thickness on maximum blade life (optimum).

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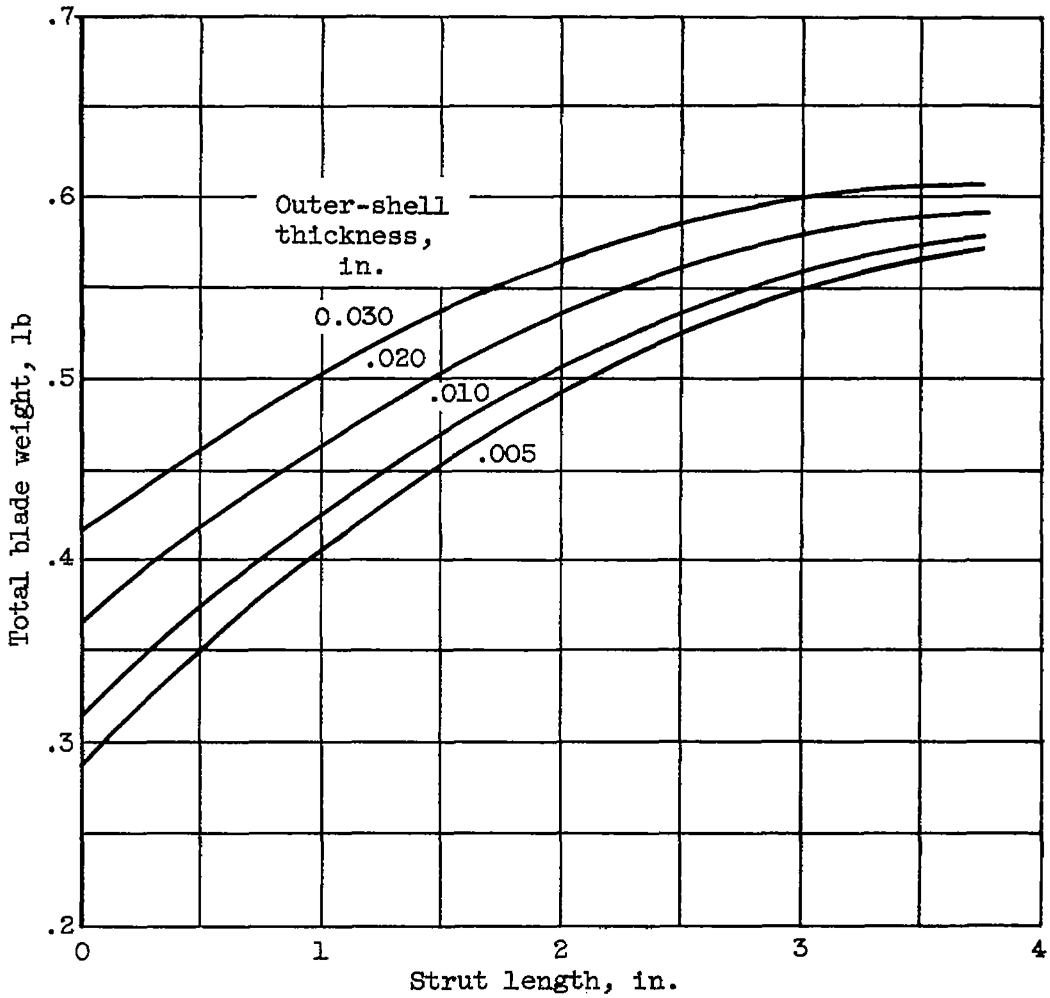

Figure 10. - Variations in total blade weight for different outer-shell thicknesses and strut lengths.

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CG-4 back

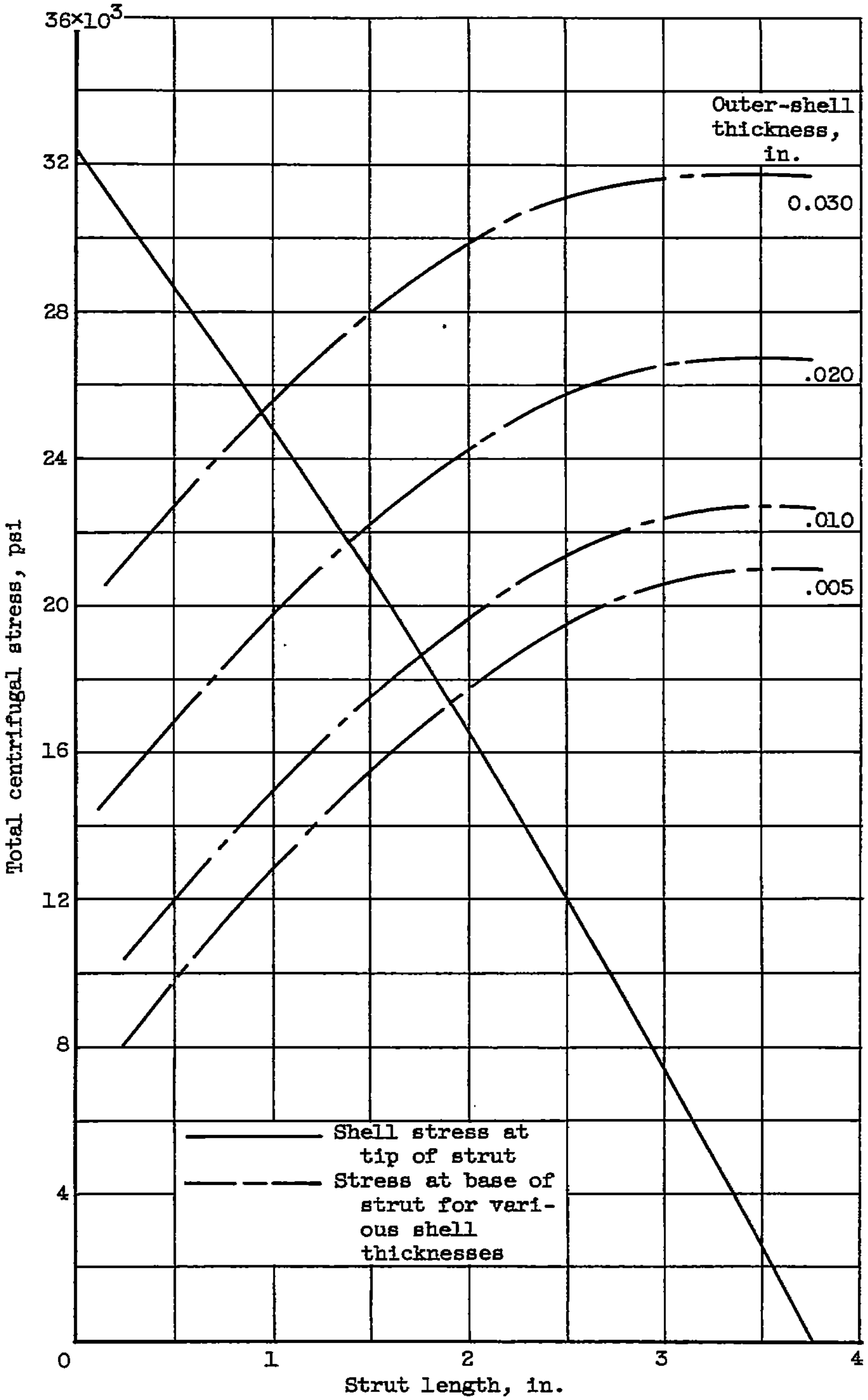

Figure 11. - Centrifugal stresses in shell and strut for various strut lengths.

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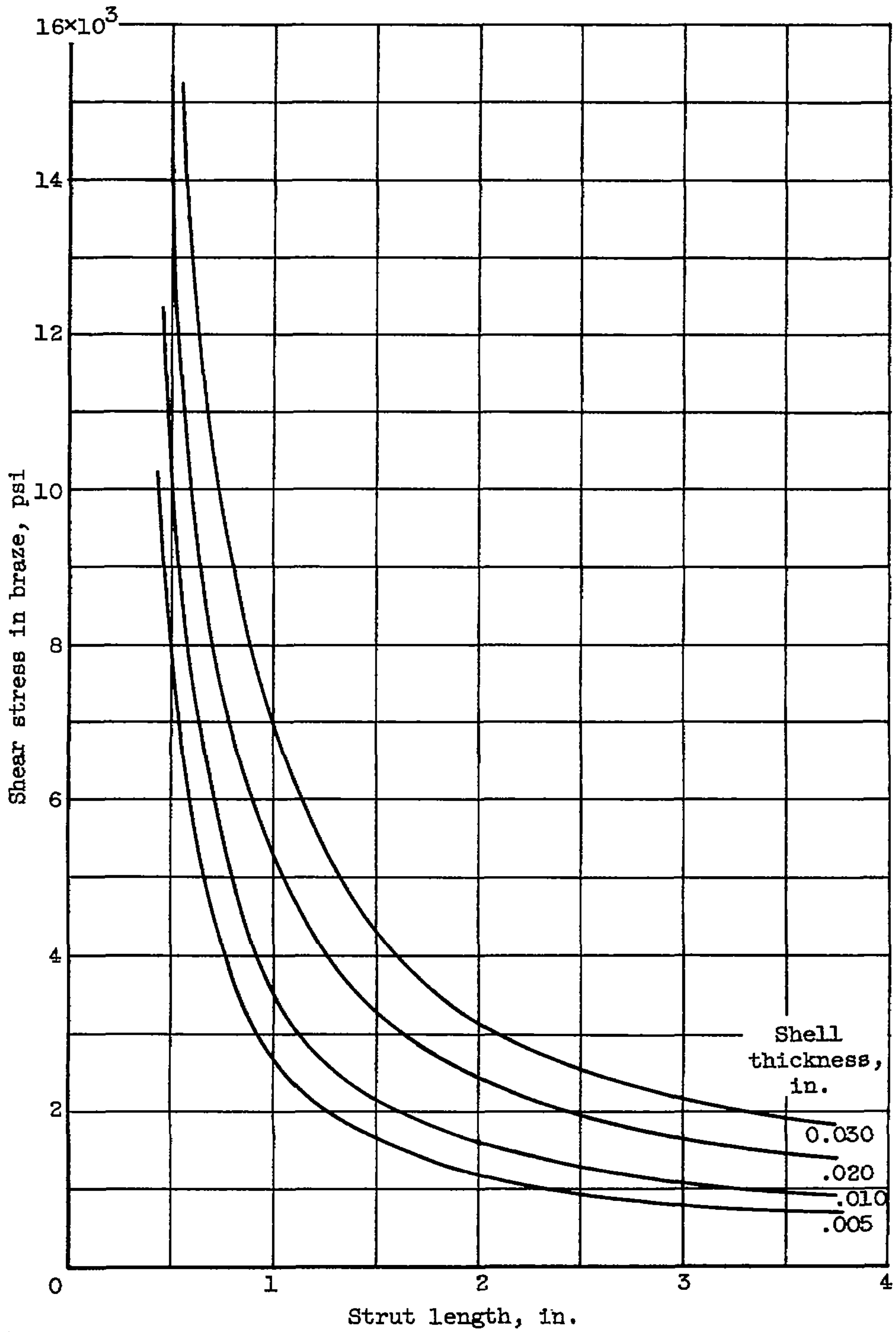

Figure 12. - Shear stress in braze as affected by strut length.

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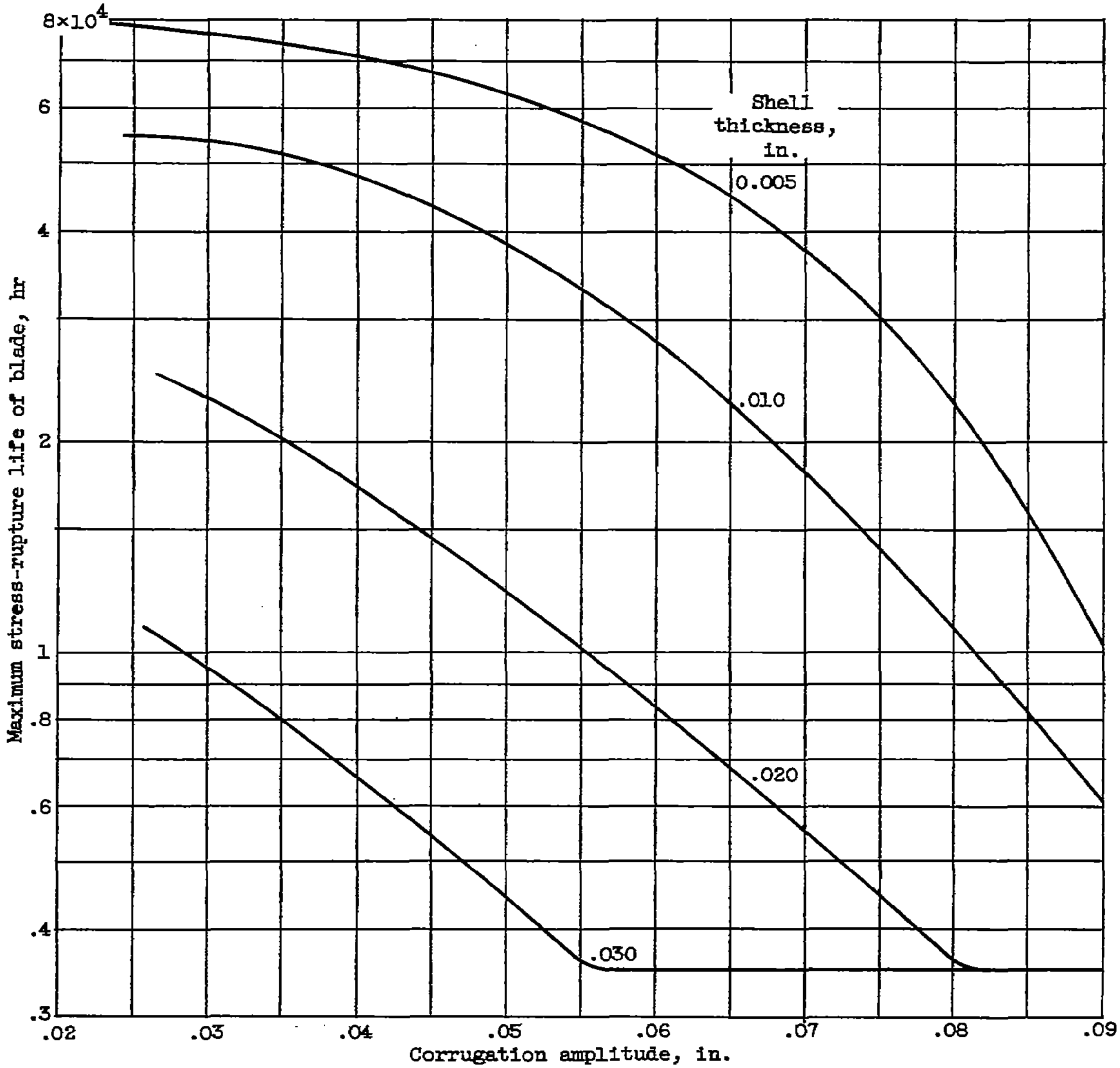

(a) Normal temperature of uncooled blade.

Figure 13. - Effect of corrugation amplitude on maximum blade stress-rupture life.

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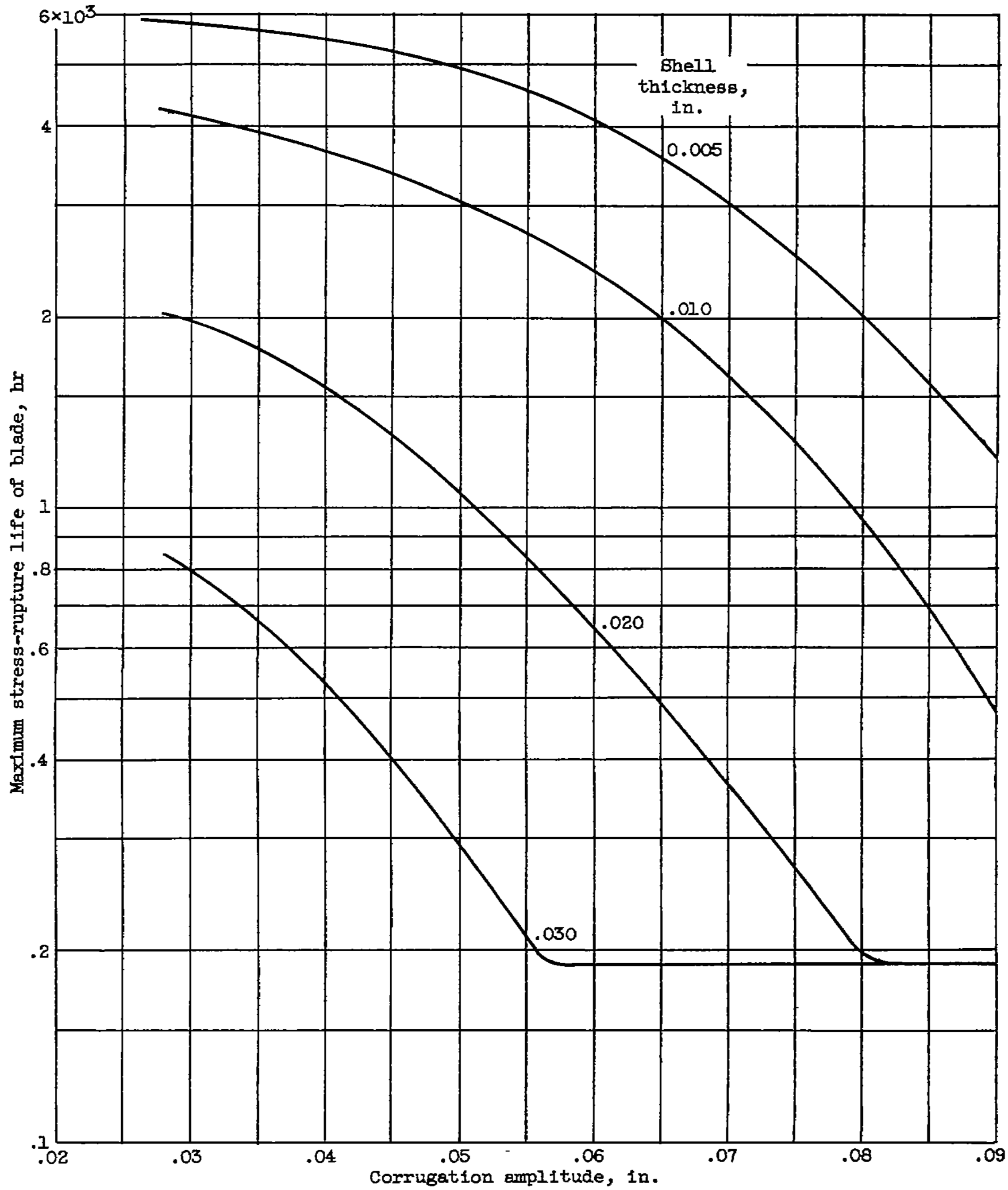

(b) Temperature 100° F higher than normal.

Figure 13. - Concluded. Effect of corrugation amplitude on maximum blade stress-rupture life.

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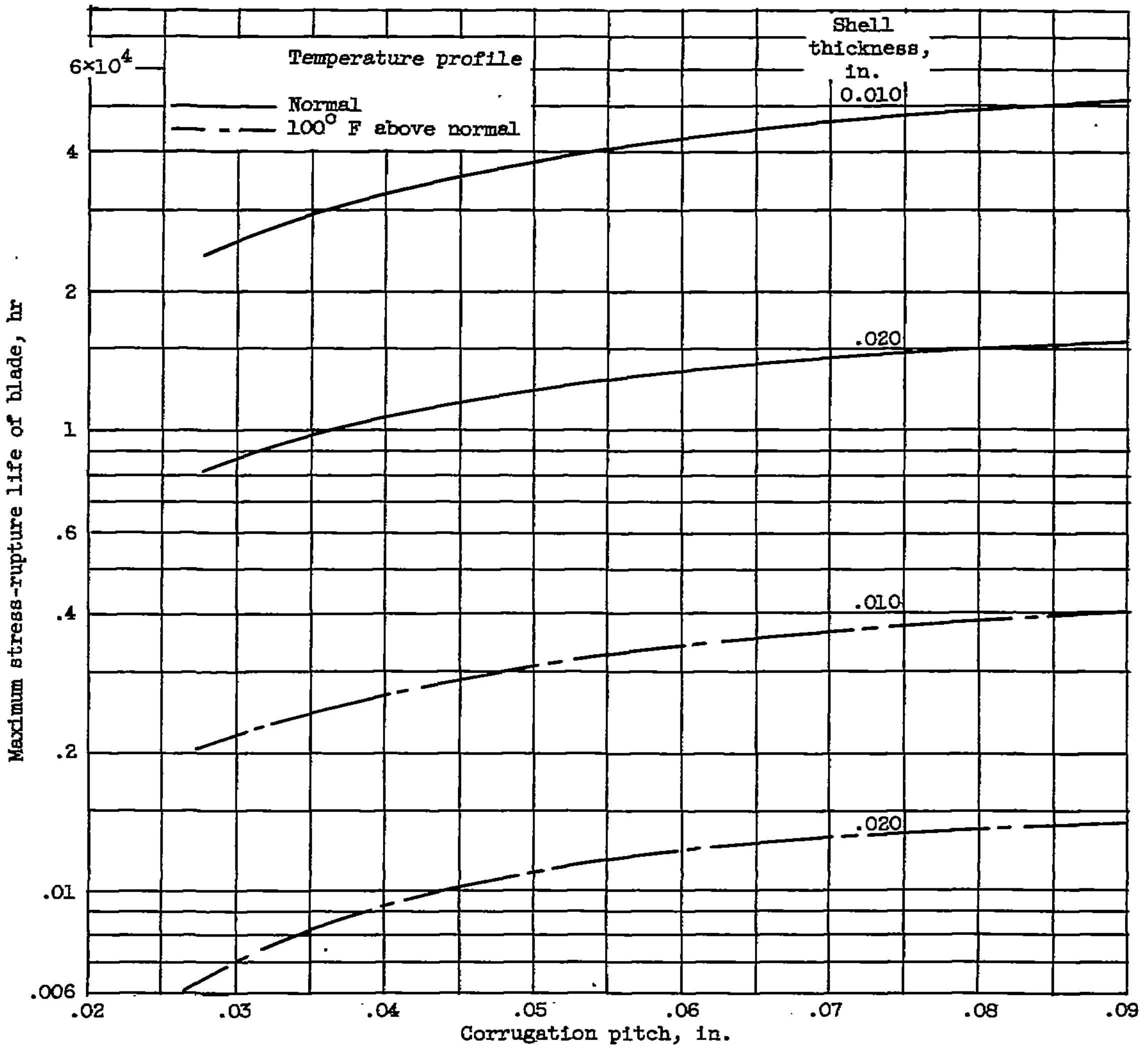

Figure 14. - Effect of corrugation pitch on maximum blade stress-rupture life.

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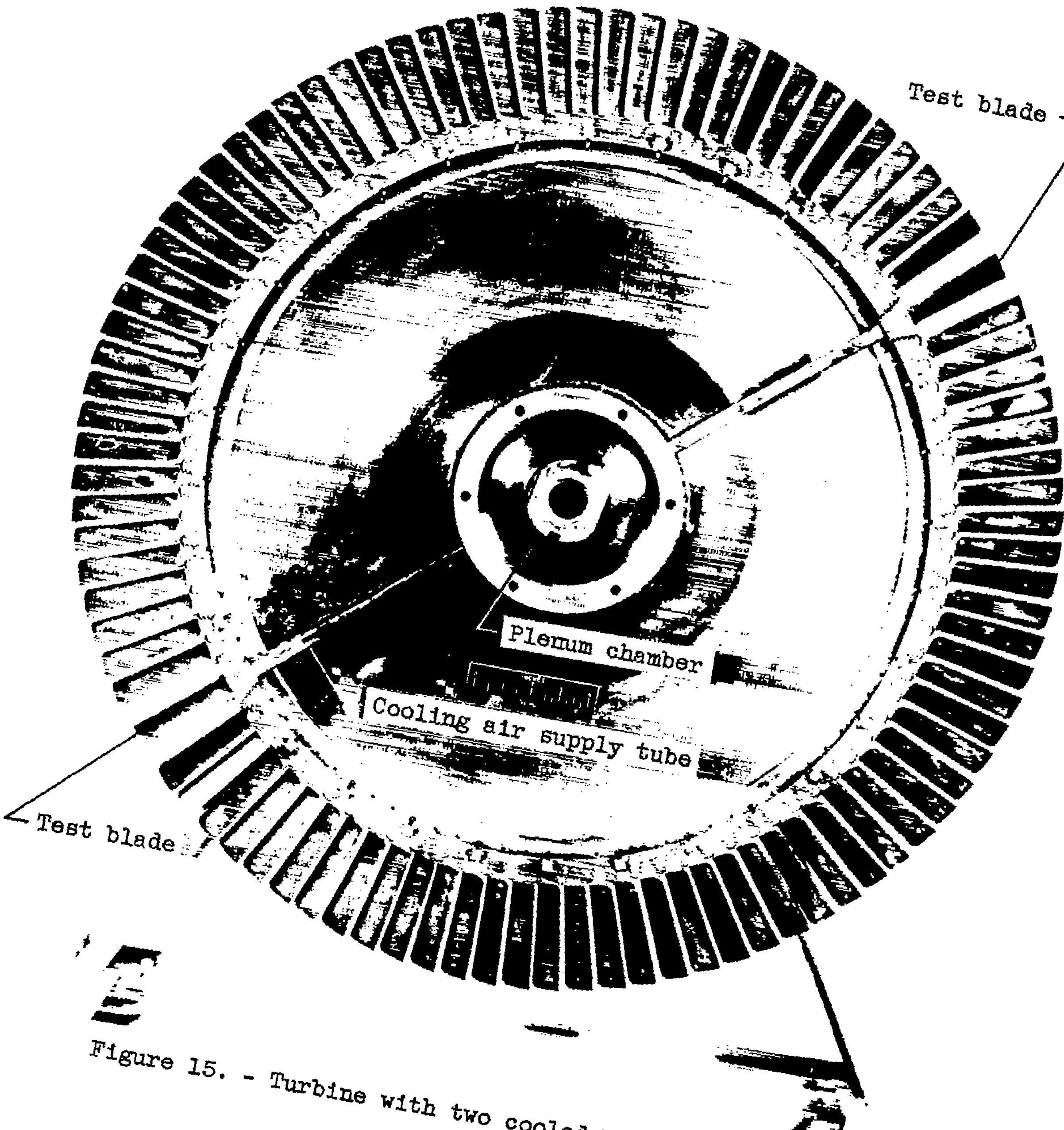

Figure 15. - Turbine with two cooled blades installed.

Figure 16. - Photograph of cooled blade before operation. Blade materials: shells, A-286; strut, X-40.

Figure 17. - S-816 - L-605 blade after 100 hours of operation with cooling air.

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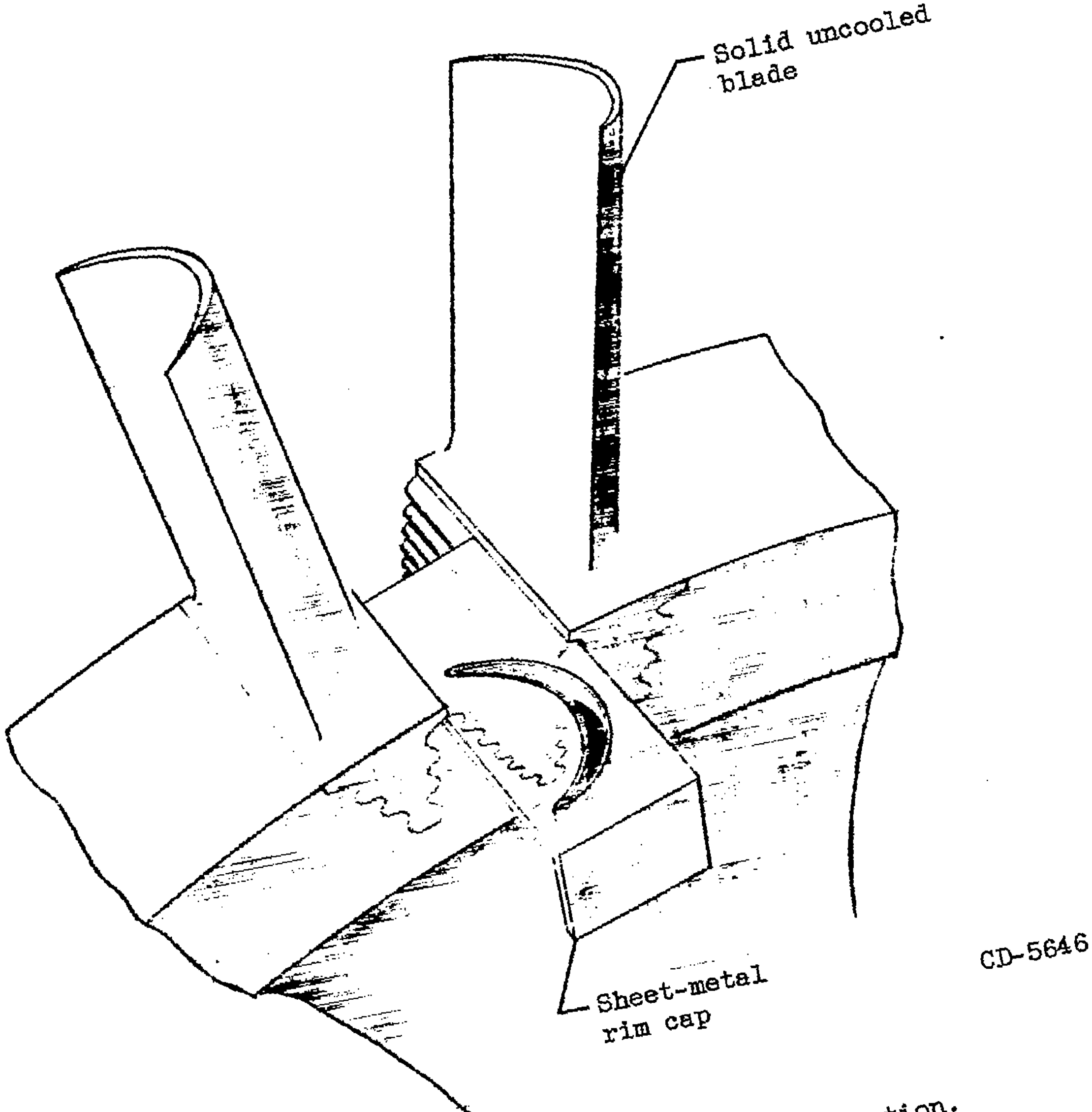

Figure 18. - Sheet-metal rim-cap construction.

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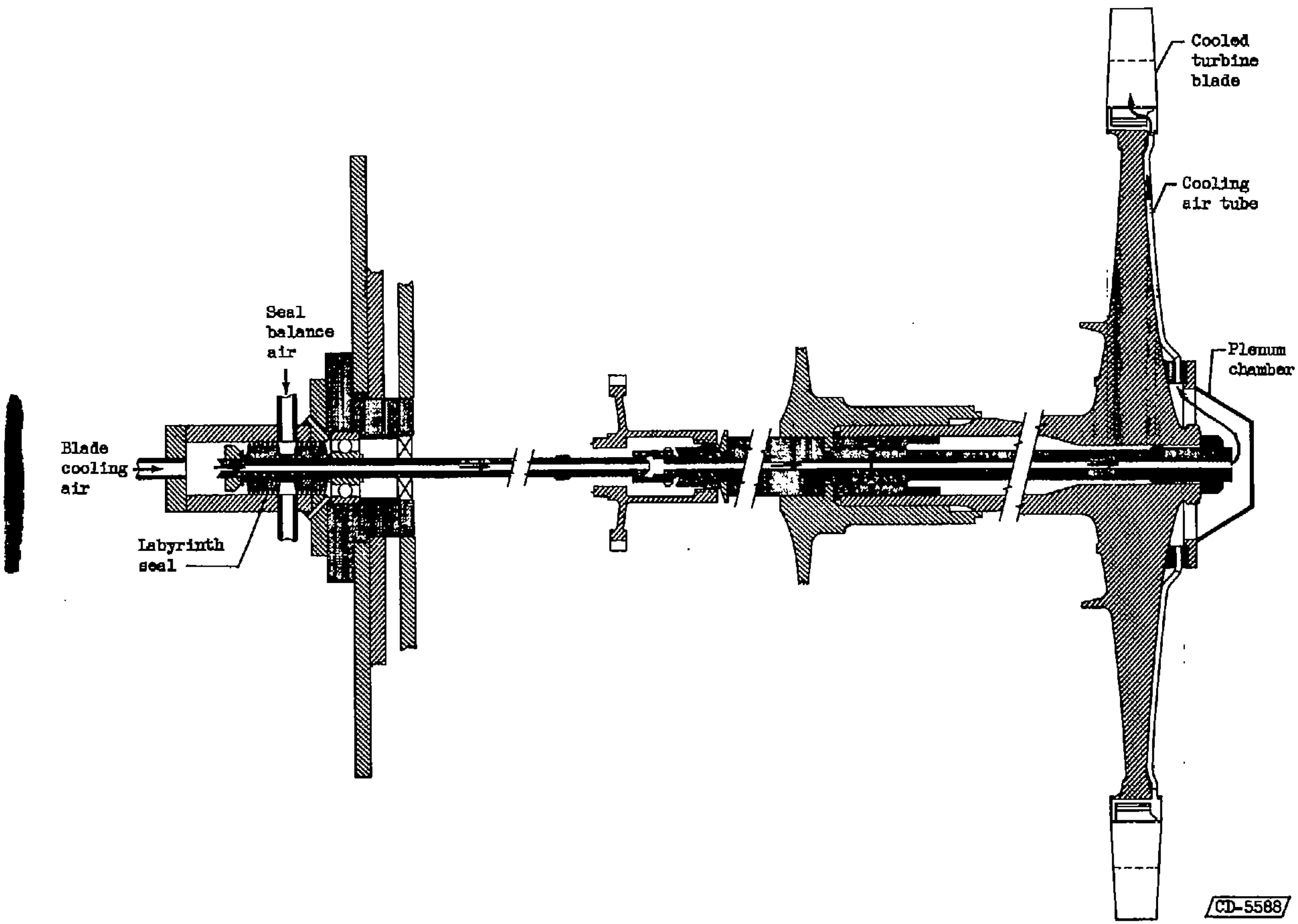

Figure 19. - Turbine cross section showing method for supplying cooling air.

Figure 16. - Photograph of cooled blade before operation. Blade materials: shells, A-286; strut, X-40.

Figure 17. - S-816 - L-605 blade after 100 hours of operation with cooling air.

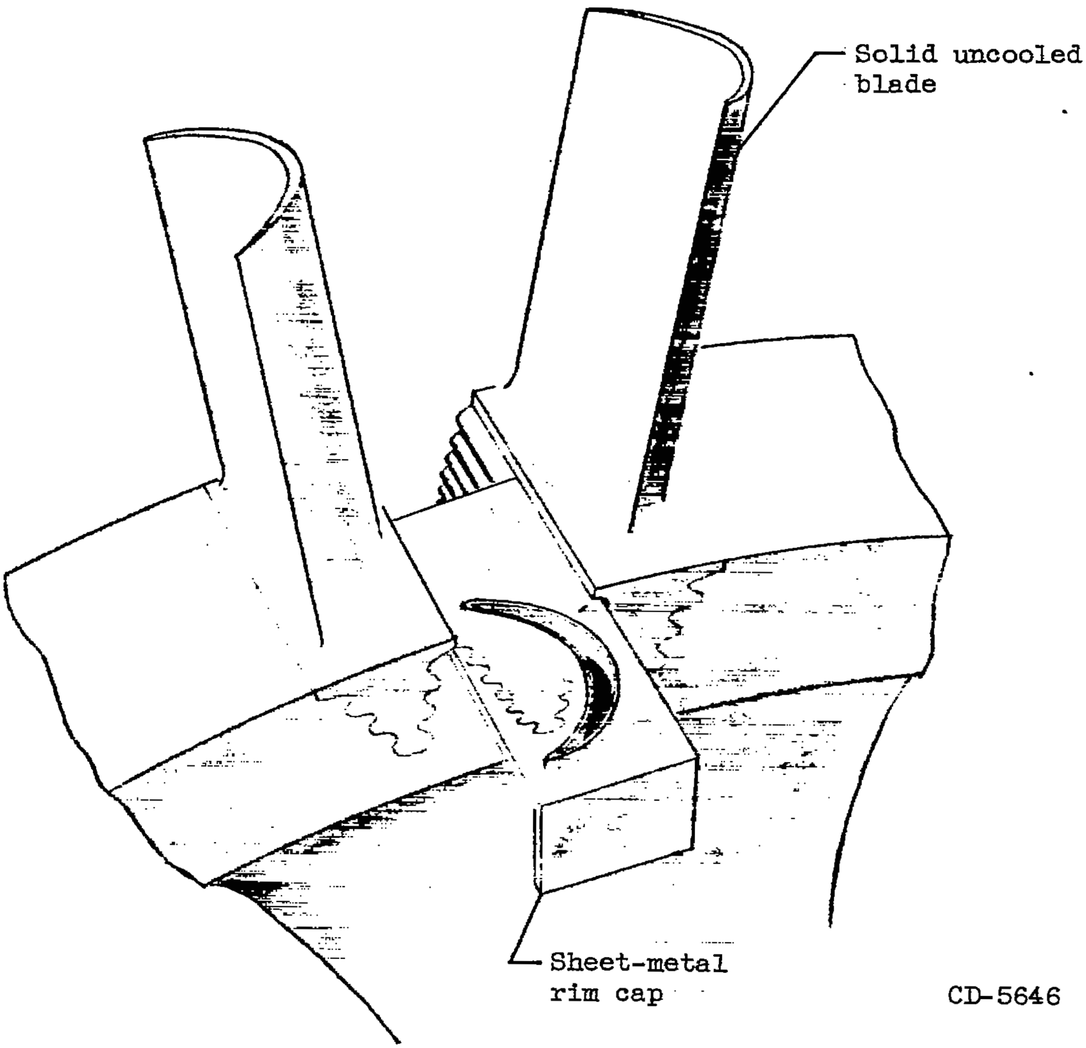

Figure 18. - Sheet-metal rim-cap construction.

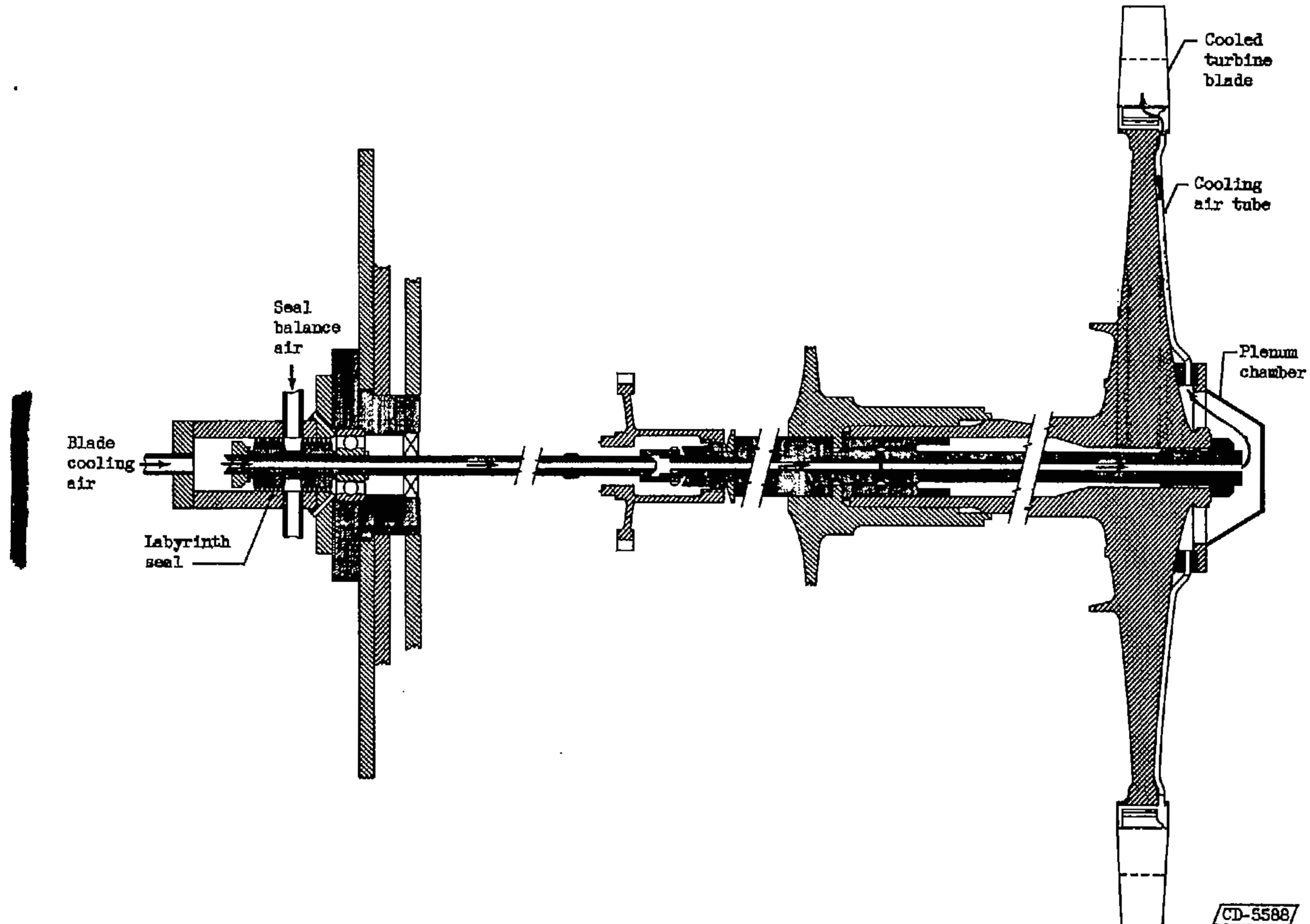

Figure 19. - Turbine cross section showing method for supplying cooling air.

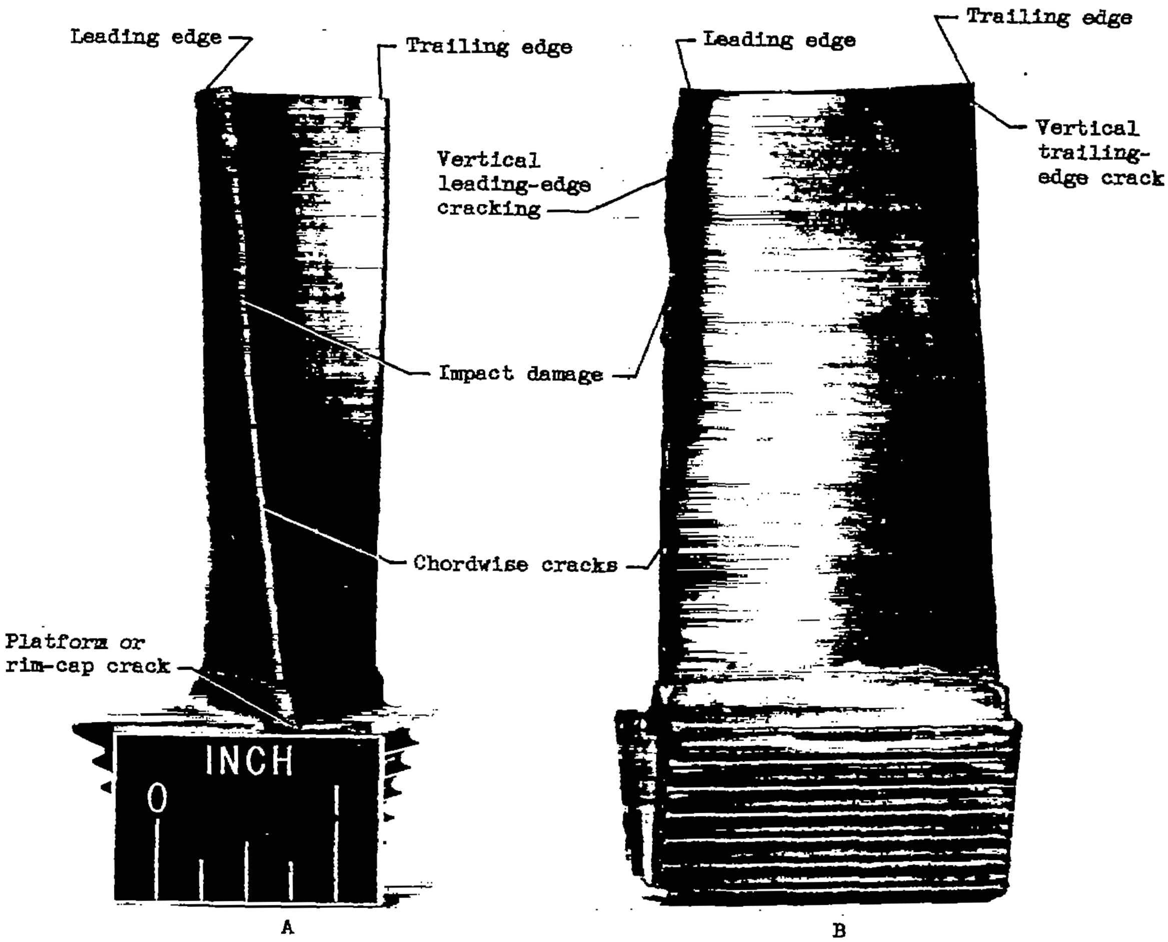

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Figure 20. - A-286 - X-40 blades after 55 hours of operation at turbine-inlet gas temperature of 1650° F.

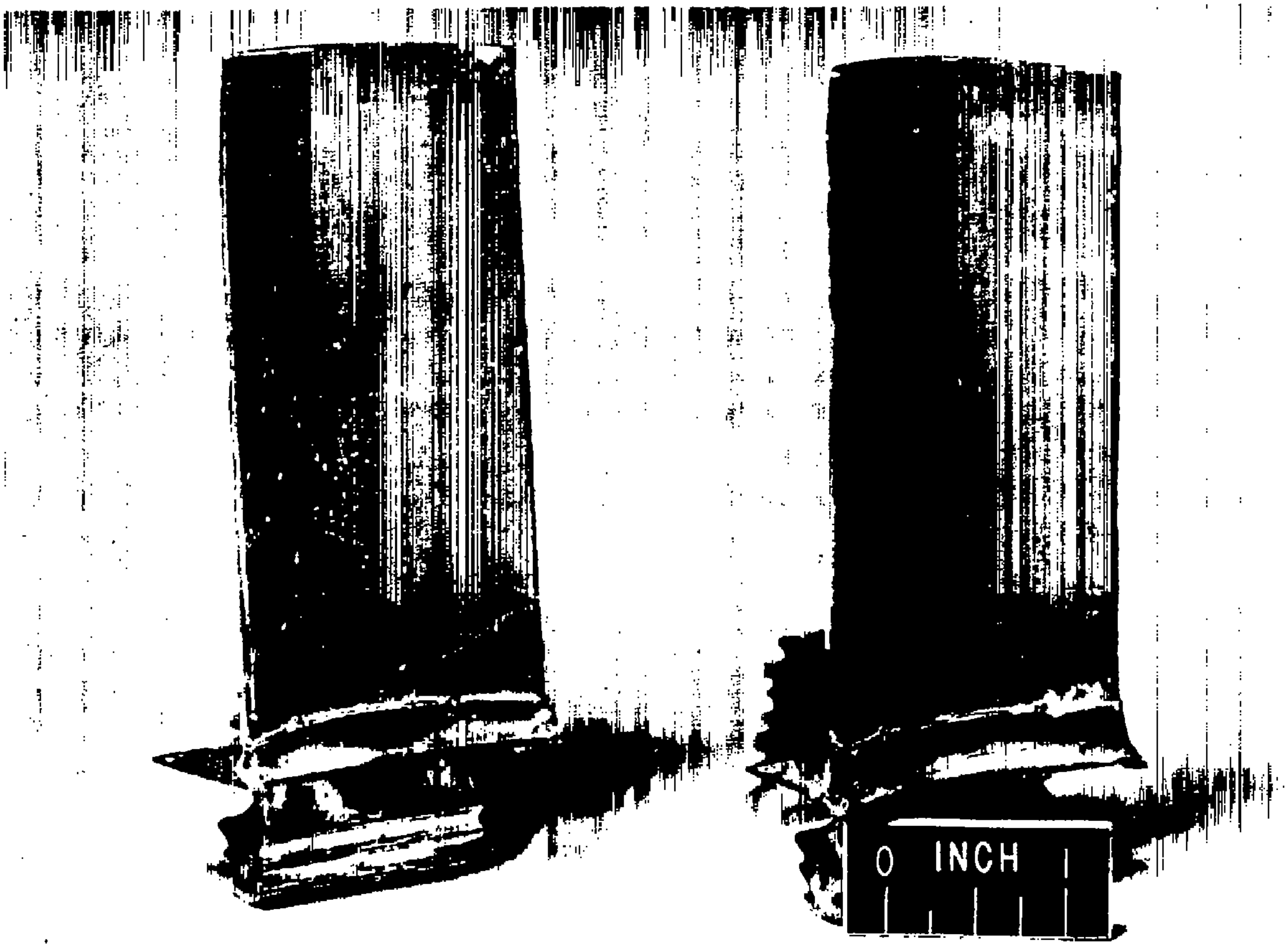

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Figure 21. - S-816 - L-605 blades after 100 hours of steady-state operation with cooling air and 300 stop-start cycles.

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