MEMO 10-5-58E

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STRUCTURAL DESIGN AND PRELIMINARY EVALUATION OF A LIGHTWEIGHT, BRAZED, AIR-COOLED TURBINE ROTOR ASSEMBLY

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NATIONAL AERONAUTICS AND SPACE ADMINISTRATION
WASHINGTON
December 1958

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SUMMARY

A lightweight turbine rotor assembly was devised, and components were evaluated in a full-scale jet engine. Thin sheet-metal airfoils were brazed to radial fingers that were an integral part of a number of thin disks composing the turbine rotor. Passages were provided between the disks and in the blades for air cooling. The computed weight of the assembly was 50 percent less than that of a similar turbine of normal construction used in a conventional turbojet engine.

Two configurations of sheet-metal test blades simulating the manner of attachment were fabricated and tested in a turbojet engine at rated speed and temperature. After $8\frac{1}{2}$ hours of operation pieces broke loose from the tip sections of the better blades. Severe cracking produced by vibration was determined as the cause of failure. Several methods of overcoming the vibration problem are suggested.

INTRODUCTION

The turbojet engine is being used for the main propulsive power for several missiles such as the Snark, the Matador, and the Regulus and is being considered for some advanced design missiles with higher flight speeds. Specific engine weight for this application, especially for long range at high supersonic speeds, becomes extremely important. The turbine rotor assembly (the blades, disks, and shafting) constitutes about one-quarter of the total jet-engine weight. Reduced turbine weight would permit additional weight reductions in the bearings, bearing supports, and total engine framework because both the static and gyroscopic forces due to maneuvering would be decreased. Therefore, any large weight savings in the turbine would have a pronounced effect on over-all engine weight.

Furthermore, the turbine assembly constitutes the largest concentration of the important alloying metals such as nickel, chromium, cobalt, molybdenum, and columbium in the conventional engine. For expendable missiles, it would be desirable to keep the amount of these strategic materials used to a minimum. For these reasons the NACA investigated the feasibility of building up a turbine rotor assembly by brazing together parts stamped from thin sheet materials. In the investigation reported in reference 1 hollow sheet-metal uncooled blades were evaluated, while in the investigation reported herein a different design concept was used, which also provides for a lightweight rotor assembly. The design criteria for missiles are relatively short lives but extreme reliability and the ability to be readily produced in large quantities at rather low costs.

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In order to study the problems with lightweight sheet-metal construction, several designs were devised and stress analyzed. Turbine blades of the most promising design were fabricated so as to duplicate the fully bladed wheel type of attachment. These blades were tested in a full-scale engine at rated speed and temperature. To diminish the need for critical alloying materials, air cooling of both the rotor disks and the blades was incorporated.

DESIGN PRINCIPLES

The primary load governing the weight of the rotating turbine assembly and indirectly the weight of the engine framework supporting the turbine is the centrifugal force on the blade airfoils. The sole purpose of the blade airfoil is to turn the hot gases and thus absorb power to turn the compressor. A hollow thin-walled sheet-metal structure properly reinforced can accomplish this goal as well as a heavier solid blade. The reduction in wall thickness is limited by the required resistance to impact, erosion, vibration, and normal gas bending loads acting on the blade. Experience with hollow blades (ref. 1) and air-cooled turbine blade designs (ref. 2) has shown the minimum practical wall thickness to be between 0.030 and 0.020 inch.

As an example of the weight savings possible by sheet-metal construction, a widely produced solid blade airfoil (without the root fastening section) weighs 0.43 pound and produces a centrifugal load of 11,450 pounds at rated speed. A constant-wall-thickness airfoil 0.030 inch thick having nearly the same external contour would weigh only 0.18 pound and would impose a centrifugal load of 4167 pounds, a 58-percent reduction. Weight and load vary directly as the thickness; thus a 0.020-inch-thick airfoil would weigh 0.12 pound and would impose a load of 2778 pounds, or 72 percent less than that for the solid blade. The greatly reduced blade loading allows substantial decreases in rotor weight.

As much as 50 percent of the total blade weight of the conventional solid bucket is in the root attachment section (which permits replacement of defective blades). The rotor rim and the remainder of the rotor, which supports the blades and rim, are heavy as a result of the replaceable-blade feature. For short-lived applications and where minimum weight is a very important item, the replaceable-blade feature can be sacrificed to gain a sizeable weight saving.

Thin airfoil shells need some internal stiffeners to suppress vibrations. If the shells are brazed to the stiffeners and the stiffeners are made an integral part of the rotor disks, a very lightweight assembly is possible. Relatively thin parallel-sided disks, which could be punched from plate material, would suffice for the rotor. Radial fingers protruding from the disk periphery could serve to retain and reinforce the blades. This proposed type of construction is illustrated in figure 1.

Continuing with the previous comparison, the turbine rotor alone that is normally used with the solid blades weighs 205 pounds. The four thin disks with protruding fingers and hub reinforcing rings (fig. 1) would weigh 114 pounds, or 44 percent less. Thus with 0.020-inch-thick shells and the four thin disks a weight saving of 134 pounds, or over 50 percent, could be effected, not considering the shafting, bearings, and engine frame. The shafting would also be less for the sheet-metal assembly because shafts are primarily designed on the basis of gyroscopic bending forces, which would be proportionally reduced for the lighter turbine assembly.

In the illustrated design, cooling air enters the turbine at the hub between adjacent pairs of disks. Inducing vanes can be added between disks to improve distribution, to aid the air in attaining the required rotational speed, to help pump the air through the blades, and to space adjacent disks properly. Cooling air was not incorporated to achieve higher gas temperatures but merely to permit the use of less strategic blade and disk materials and to permit slightly higher stresses in the blades and disks. The use of cooling is not necessarily required for the type of lightweight construction discussed herein.

OPERATING STRESSES

In order to determine some of the structural problems associated with lightweight sheet-metal turbine construction, a number of turbine configurations were devised and stress analyzed. To begin with logical design parameters and to obtain an inexpensive test vehicle, the geometric values and operating conditions for a widely used jet engine were selected. This unit had a turbine tip speed close to 1200 feet per second, a tip diameter of 34.3 inches, and a blade height of 3.75 inches. Heat-resistant materials having an average density of 0.31 pound per cubic inch were assumed for all parts.

Tensile Stresses in the Blade

Using the above constants, the average centrifugal stresses (centrifugal load divided by total cross-sectional area) were computed along the span of a constant-thickness shell and are presented in figure 2. These stresses are independent of shell thickness as long as the thickness does not vary along the span. For comparison, also included in the figure are the tensile stresses for the conventional solid blade. The stress for the untapered shell is about 50 percent higher than for the conventional blade; but for short lives these stresses are still very conservative, especially since air cooling can be readily introduced. Tapering the sheet metal reduces the stresses somewhat, as is illustrated by the curve for a 2-to-1 taper ratio. Further reductions in stress are possible with greater taper ratios.

Because the reinforcing fingers (fig. 1) are appreciably tapered and are integrally brazed to the shell, they lower the stresses considerably by increasing the blade cross-sectional area. The actual decrease in stress depends on the shell thickness chosen. A curve showing the spanwise stress distribution for a 0.020-inch-thick shell with four fingers is also shown in figure 2. The stresses for a thicker shell would not be reduced as much. The discontinuity in the curve is caused by the termination (at 1.85 in. from the base) of two of the four fingers. The stress could be further reduced by also tapering the fingers from the blade base to the tip in the chordwise direction. The case shown is for a constant chordal or disk thickness of 1/8 inch for each finger. By using tapered shell material in conjunction with highly tapered fingers, the tensile stresses in a sheet-metal blade could be reduced below those for the conventional solid blades.

At the base of the airfoil, the fingers alone are assumed to carry the entire blade load. For the example illustrated (0.020-in.-thick shell) the stress in the fingers is 33,650 pounds per square inch, which is acceptable considering that all the cooling air supplied to the blades passes this point. However, the stresses with a 0.030-inch constant-thickness shell would be excessive (50,500 psi).

Shear Stresses in the Braze

The average shear stresses in the braze were computed for the four 1/8-inch-thick fingers illustrated in figure 1. Neglecting the slots for distributing air into the hollow airfoil, the average shear stresses are 815 and 1225 pounds per square inch for shell thicknesses of 0.020 and 0.030 inch, respectively. If even half the contact area in shear were removed for cooling air slots, these stresses, which would then be doubled, would still be extremely conservative. Tapering the fingers in the chordwise direction until the tips were pointed, thus again doubling the shear stresses, would still not produce excessive stresses.

At the operating temperatures of the blade, a safe shear stress for heat-resistant nickel-chromium brazes would be about 20,000 pounds per square inch, provided that exact clearances were maintained between parts during brazing. However, because of the complexity of the brazing technique and the difficulty of maintaining clearances during the process, lower design stresses should be used.

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Stresses in the Parallel-Sided Rotor Disks

The elastic stress distribution was calculated using the procedure outlined in reference 3 for several disk configurations assuming a relatively severe gradient between hub and rim temperatures of 300° and 1200° F, respectively, and a temperature variation between these points in accordance with the third power of the radius. About 300° to 400° F is the maximum allowable hub temperature range to give satisfactory bearing operation and would be produced by cooling the hub region if the blade cooling air were omitted. Bleeding cooling air for the blades from the high-pressure end of the compressor and introducing it at the center of the disks would also produce a hub temperature in the range 300° to 400° F. In all cases the radial stress at the rim was assumed to be 11,500 pounds per square inch, which corresponds to the centrifugal load of 96 blades with 0.020-inch-thick shells and four fingers per blade supported by four disks 1/8 inch thick. The combined thermal and centrifugal elastic stresses are shown in figure 3. The stresses in disks without central holes (fig. 3(a)) are higher than used in conventional turbine design. However, because plate material is more homogeneous than large forgings, if a high-quality disk material is used, further disk reinforcement may not be necessary, and thus the disk weight and complexity can be kept to a minimum. The high compressive stresses in the rim which result from the thermal gradient are not of major concern. If the compressive yield point is exceeded at the first time of operation, the material will flow plastically, and from then on the stresses will never exceed the yield point. Similar high compressive stresses exist in conventional turbines, as shown in figure 4. In this figure are plotted the calculated stresses for the equilibrium temperature gradient (fig. 4(a)) and the worst transient temperature gradient (fig. 4(b)), both of which are plotted in figure 5. At equilibrium the temperature difference between the hub and the rim was 630° F, and under transient conditions the maximum difference was close to 900° F, the difference assumed for stress computations of the parallel-sided disks. Excessive cyclic operation (not likely for missile application) may prove troublesome with the thin disks because of cracking or warping, as it has with some conventional turbine rotors.

The tangential stress for the disks with central holes for introducing blade cooling air is high at the hub (fig. 3(b)). This stress is very localized and will probably also be relieved the first time rated speed is reached. Because this tangential stress is tensile and of much higher magnitude than the compressive rim thermal stress, it might cause cracking in a relatively few cycles. Its seriousness will have to be evaluated experimentally.

There are simple methods of overcoming the cracking danger. One is illustrated by the stresses plotted in figure 3(c). Hub rings equal in thickness to the main disks were assumed to be brazed to the main disks. The outer peripheries of the rings were tapered from the 3-inch diameter to a knife edge at the 4-inch diameter, as shown in the sketch in the figure. The resultant tangential stress was lowered from 203,000 to 94,000 pounds per square inch, about the same stress as in the conventional disk. Another cure may be effected by avoiding the central hole and introducing the cooling air nearer to the rim, for example, at the 11.5-inch radius where the disk stresses are lower (fig. 3(a)). Schemes for scooping in the cooling air at this location are sketched in figure 6. These are only a few suggestions, none of which have been tried. Introducing air at this location would require considerable development to obtain good air distribution with a reasonable pressure drop and to obtain an adequate seal between the stationary air supply manifold and the rotating openings in the disks. There are probably better methods than those illustrated to introduce cooling air without using central holes. 5088

Passing cooling air either through a central hole or nearer the rim and then radially outward between the disks will probably produce a temperature gradient differing from the assumed gradient. Conditions may be better or worse than those computed. Temperatures would have to be determined analytically or experimentally before stresses with coolant flow could be accurately predicted.

FABRICATION OF TEST BLADES

The cost and time required for constructing a fully bladed turbine rotor for strength evaluation alone is prohibitive. However, a few special test blades that simulate the type of construction shown in figure 1 can be made easily and tested in a standard engine. A blade design was evolved that would fit into a standard turbine rotor and which duplicated the blade attachment to radial fingers. Figure 7 shows an exploded view of the test blade base and the radial fingers to which the shell was attached as well as the passages for coolant flow. This construction results in a root attachment that is considerably wider than normal; consequently, two standard blades had to be replaced by each special test blade.

The finger pieces and spacers were made from A-286 material (low-strategic-alloy content) and had to be nickel plated to produce a satisfactory braze. These pieces were brazed together first to form the base slug shown in figure 8. Then 0.020-inch-thick L-605 material was stretch-formed to the airfoil contour and was also brazed in position. Two 0.020-inch-thick sheet-metal pieces simulating the rim cap of a full wheel were brazed to the top surface of the root slug. The braze material was 20 percent chromium, 10 percent silicon, 1 percent iron, and the balance nickel. The brazing cycle consisted of heating to 1800° F, holding for 10 minutes, rapid heating to 2130° F in a vacuum (1 to 4 microns of mercury), and holding for 15 minutes. The next operation was welding the two shell halves together at the leading and trailing edges with a shielded arc but without filler rod. Finally, the blades were trimmed to length, and serrations were ground into the base slug.

ENGINE INSTALLATION AND TEST PROCEDURE

Two test blades were installed opposite one another in a turbine disk that had tubes for cooling air welded to its rear face. The mechanism for transferring cooling air to the rotor is sketched in figure 9. Laboratory service air was supplied to the test blades at a rate equivalent to the amount that each blade would receive if a total of 2 percent of the total compressor flow were extracted for distribution to the complete set of blades. The test conditions for blade evaluation with the engine installed in a sea-level type test stand were rated speed (1200 ft/sec tip speed) and temperature (1260° F exhaust). These operating conditions were maintained until failure was noted.

RESULTS OF ENGINE TEST ON FIRST PAIR OF BLADES

The total operating time in the engine before failure of the first two test blades was 6 hours and 41 minutes, but only 2 hours and 33 minutes were at rated speed and temperature. Considerable difficulty was experienced with the bearing of the cooling air transfer mechanism, and this accounted for the excessive running done at less than rated speed. Photographs of the failed blades are given in figure 10. Examination of the failed pieces of the blade shown in figure 10(a) revealed that a very poor braze between the shell and the radial fingers was responsible for the failure. The two fingers near the trailing edge were attached to the shell by braze on only about 25 percent of the surface provided, probably because of excessive clearance between the original parts. Failure of this small shear area shifted the entire blade load to the two leading-edge fingers still securely brazed to the shell. Calculations indicated that this action caused the tensile stress in these two fingers to increase to about 60,000 pounds per square inch and in addition imposed a severe bending load because of the unsymmetrical support. The two leading-edge fingers then failed in tension, and as the shell moved radially outward and through the thin blowout shroud band installed on the engine, probably the longer trailing-edge finger was broken off 3/4 inch above the base. The shell then slipped free of the shorter trailing-edge finger.

The other test blade failed in fatigue in the tip region as shown in figure 10(b), but as a result of vibration. A crack probably originating adjacent to the full-length finger proceeded radially inward until the inadequately supported piece tore loose.

DESIGN AND FABRICATION OF AN IMPROVED TEST BLADE

The mechanism of failure in the preceding test indicated the need for a design change that would remove some of the shell load from the fingers. Therefore, the shell length was increased 1/2 inch radially inward to provide direct attachment to the rim (fig. 11). In a full wheel it will be necessary to modify the disks by cutting 1/2-inch-deep slots into the rim of all disks on both sides and immediately adjacent to each finger. This alteration unfortunately requires more machining on the disks, but it simplifies the brazing procedure by reducing the amount of clamping needed and by automatically providing the proper clearance for brazing between the shell and the fingers.

The change improves the blade strength by increasing the quantity of braze area in shear; but more important, it transfers the shell load from the fingers only to the whole disk rim. The total weight increase due to the change is only 0.016 pound per blade for the 0.020-inch-thick shell. Also shown in the sketch of the hub section of this figure are hub reinforcing rings with interlocking spiral threads to give more intimate connection between the disk and the ring. Hub reinforcements are being studied under separate programs.

In order to simulate the full wheel blade attachment in a two-blade test, the base slug was machined out of two solid pieces (fig. 12). The fingers also were machined from a single piece of stock for ease of positioning during brazing. The shells, base blocks, and fingers were brazed in a single heat. The shell edges were welded as the first test blades were, and then the tips were cut to length to free the long fingers from each other. The test blades were finished by adding the rim cap and grinding serrations into the root.

RESULTS OF ENGINE TEST ON SECOND PAIR OF BLADES

The blades were installed in the same turbine rotor, engine, and test cell used for the first test. Because much less difficulty was encountered with the bearing of the cooling air transfer mechanism, the rated time was a higher percentage of the total engine operating time. After 5 hours and

40 minutes at rated speed and temperature the blades were examined, and no damage visible to the naked eye could be detected. Running continued until a total time of 9 hours and 13 minutes, including 8 hours and 29 minutes at rated speed, was accumulated. At this time several blade tip fragments tore through the thin shroud band. The condition of the blades is shown in figure 13. Again vibration caused cracking in the tip region of the hollow blades. The mode of vibration was probably the one known as the "breathing" mode, where opposite faces of the airfoil move in opposite directions.

DISCUSSION

The engine experience with the built-up turbine design presented herein shows that a substantial weight saving can be achieved by utilizing brazed sheet-metal structure. Although the tests conducted were quite limited, for missile application the present state of development is nearly adequate. In the 8 hours of successful operation even a slow missile could cover about 5000 miles. In the 8-hour test the engine was operated continuously at rated conditions, which are normally limited to a maximum time of 15 minutes. In missile flight the engine would probably be throttled slightly and would more closely approach normal cruise conditions. The failure with the second pair of test blades was due to vibration, probably resulting from the mode of testing.

Extensive past experience at the NACA with unevenly spaced experimental turbine blades (one blade replacing two standard blades) has conclusively demonstrated that this procedure greatly accelerates fatigue failures. It must be resorted to, however, for testing special root attachments because of the limited tangential spacing distance in the standard wheel. The effects of uneven spacing are mentioned in reference 4. In unpublished NACA tests, unevenly spaced 0.030-inch-thick blade shells with no internal reinforcement and no cooling air failed because of vibration within 30 minutes, while the same type blades survived 30 hours without failure when evenly spaced.

With additional refinements it should be possible to obtain a more reliable lightweight air-cooled blade design for manned aircraft or effect further reduction in weight by using thinner or tapered shell material. To improve the vibration resistance of sheet-metal blades, several logical approaches should be investigated. First, the tip region of the blade should be stiffened to preclude "breathing" vibrations. Methods of tip stiffening are (1) applying a tip cap over the blade (with circular holes for the escape of cooling air), (2) extending the short fingers closer to the blade tip, (3) increasing the shell thickness, and (4) increasing the number of fingers by using more but thinner disks. Another possibility is to explore other parent and braze materials which when combined have better fatigue properties. Also, dampers might be added inside the blade shell.

Cooling air was provided for in this hollow blade construction because it was so easy to incorporate. The amount introduced was intended only to improve reliability and to reduce the need for strategic alloying materials, not to permit higher gas temperatures. However, it may be possible to design for higher turbine-inlet gas temperatures by improving the heat-transfer properties of the shell by providing fins or corrugations and increasing the coolant flow rate. Corrugations and internal fins would also serve to stiffen the blade and improve the vibration characteristics.

The proposed construction may also be applicable to the power turbines used to drive the pumps for liquid-propellant rockets, particularly when the units become larger.

SUMMARY OF RESULTS

In a program to investigate the problems associated with the design and fabrication of lightweight, brazed, sheet-metal turbine assemblies, the following results were obtained:

1. A design study using thin sheet metal for the blade airfoils and relatively thin parallel-sided disks for the rotor showed that a 50-percent weight saving could be effected over a specific widely used conventional turbine assembly.

2. The first pair of test blades with sheet-metal airfoils brazed to radial fingers by a construction that simulated the lightweight rotor attachment failed in $2\frac{1}{2}$ hours at rated conditions in a full-scale engine. The principal failure was in the brazed joints.

3. The second pair of test blades, which were redesigned to increase the brazing area, endured $8\frac{1}{2}$ hours at rated speed and temperature before failure.

4. Vibration was determined as the mechanism of failure of the second test blades and was probably excessive because of the uneven blade spacing that was required to test blades attached to the rotor by unconventional means.

Lewis Research Center
National Aeronautics and Space Administration
Cleveland, Ohio, July 15, 1958.

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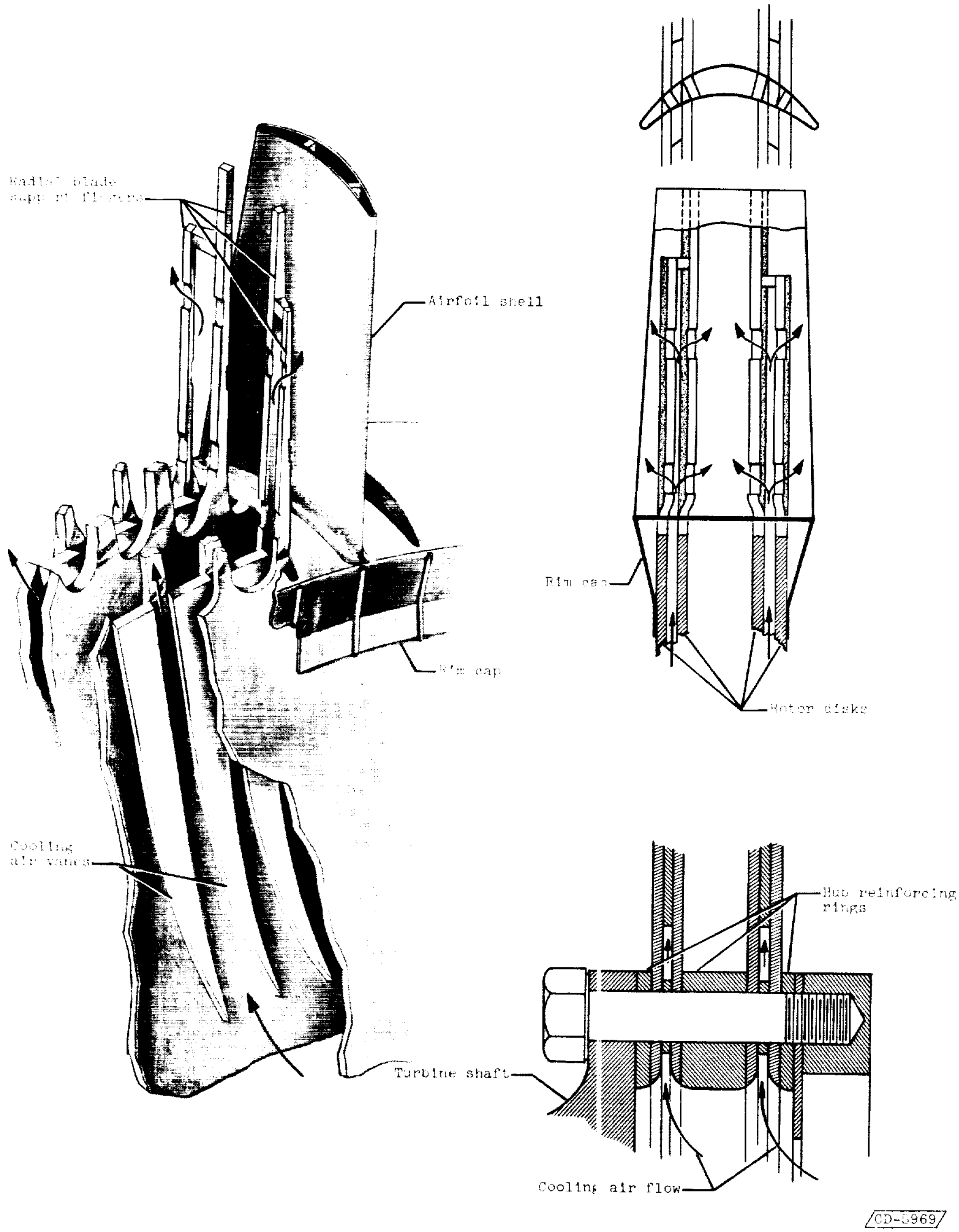


Figure 1. - Schematic drawing of proposed lightwe ght turbine assembly.

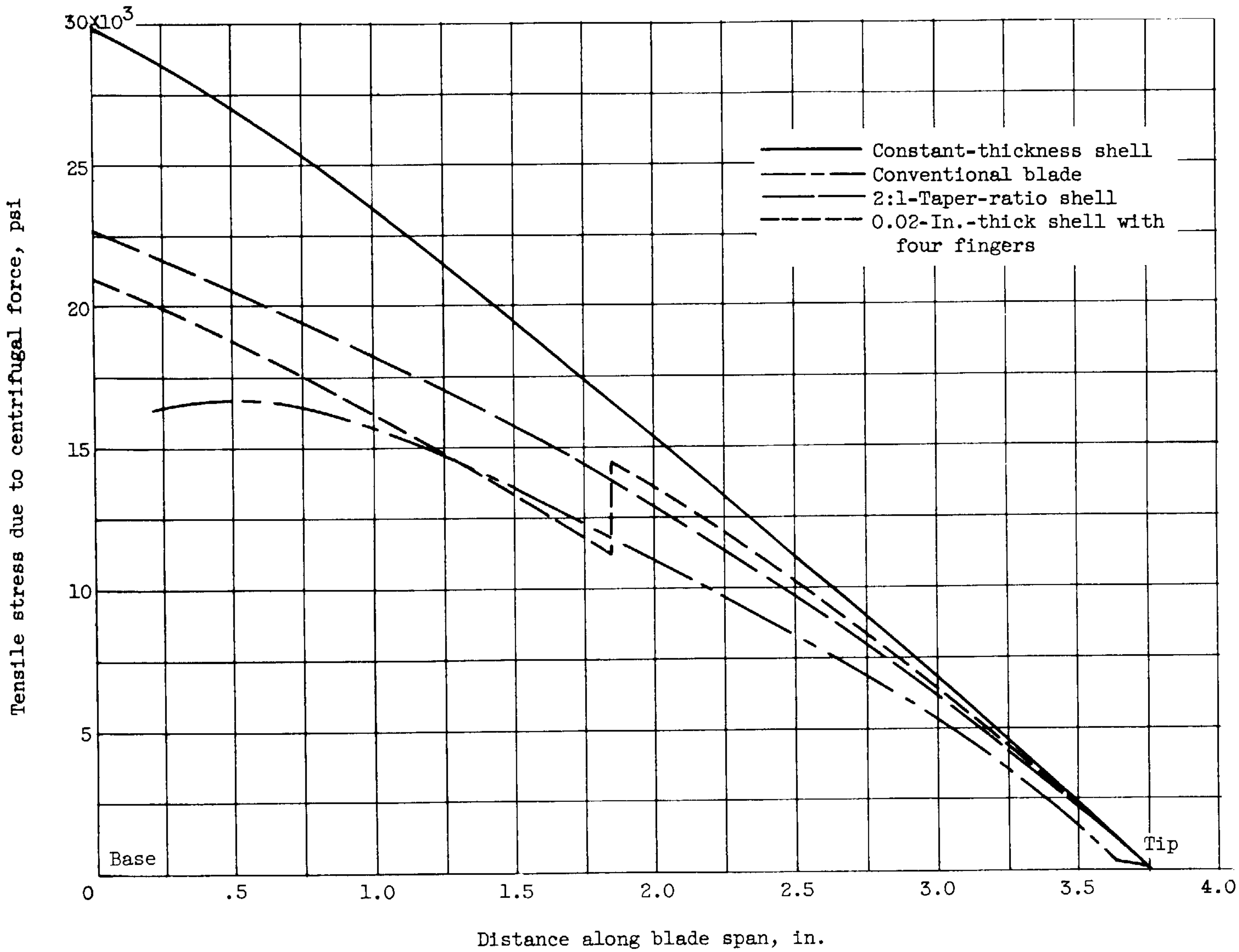
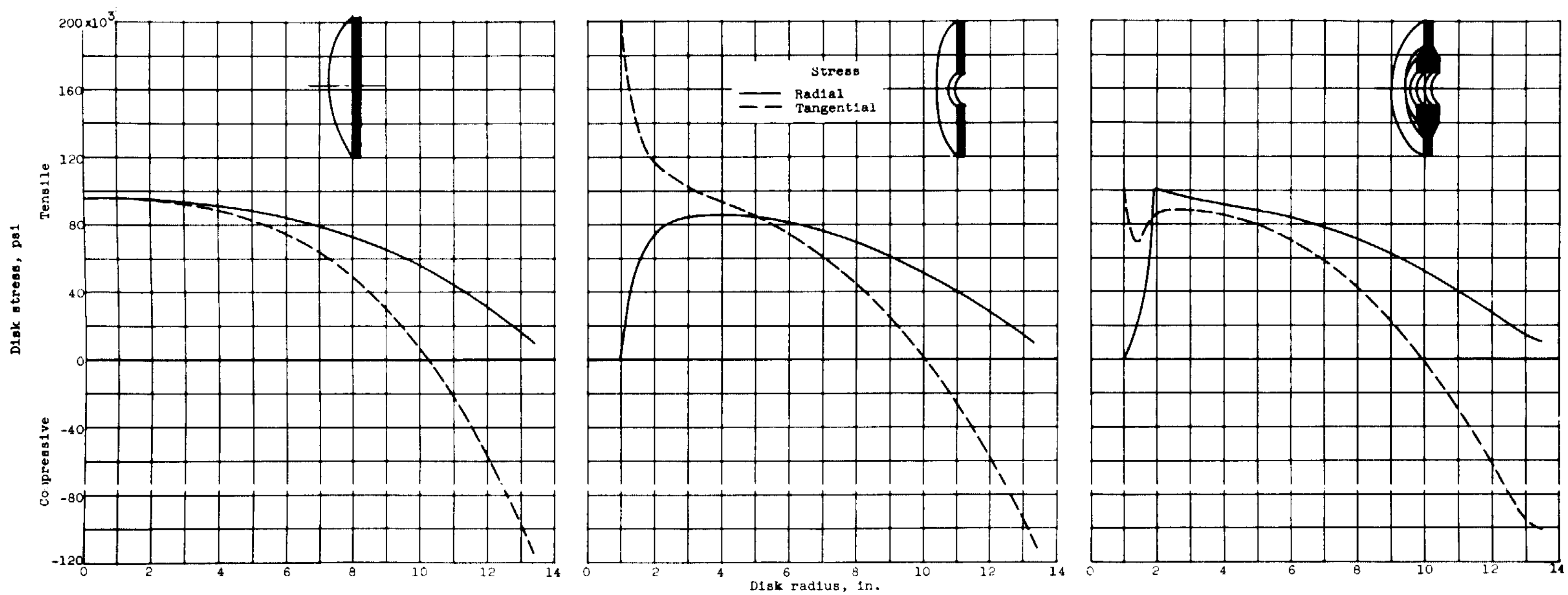
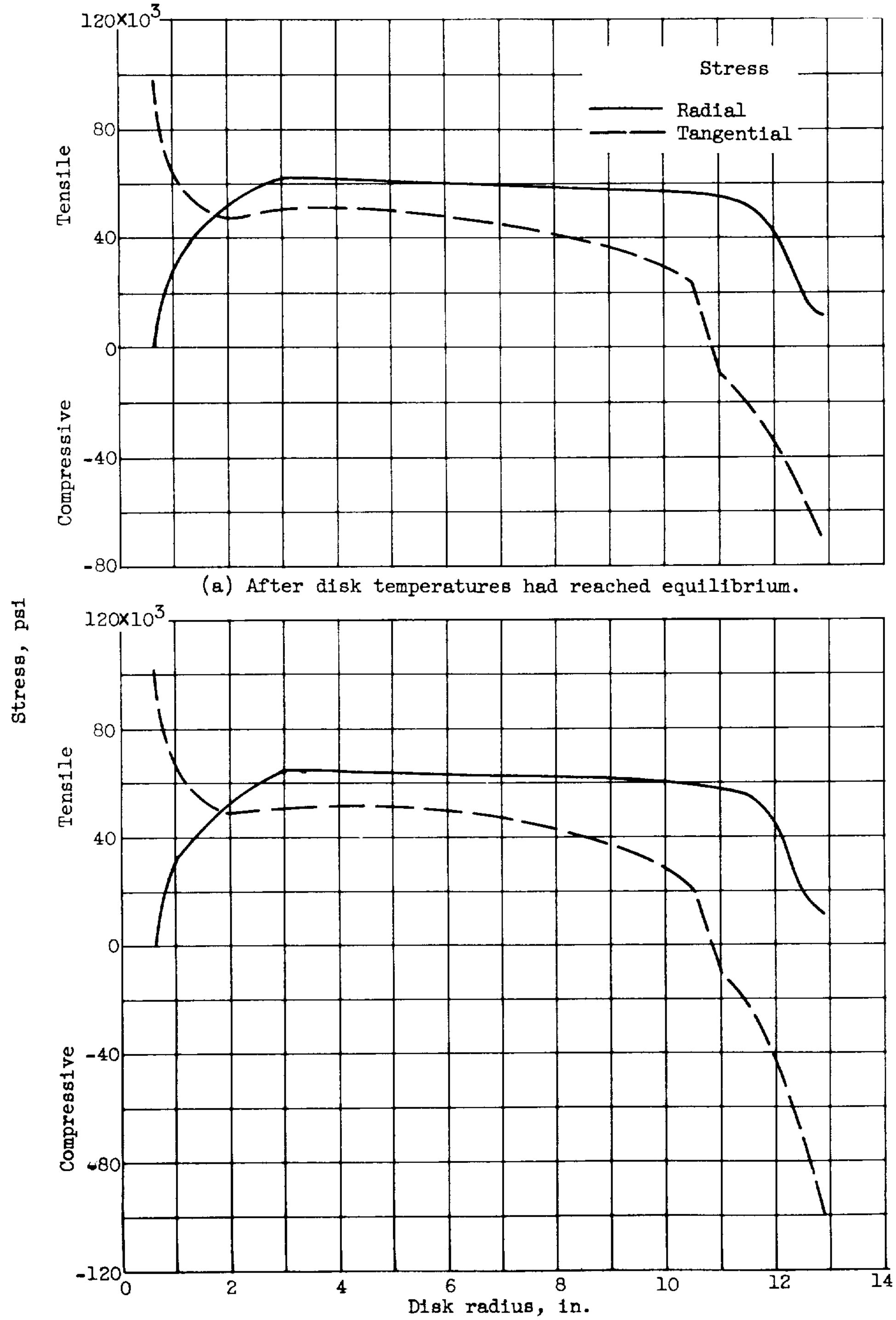


Figure 2. - Centrifugal stresses along blade span.



(a) Disk without central hole.　(b) Disk with 2-inch-diameter central hole.　(c) Disk with reinforced central hole.

Figure 3. - Combined thermal and centrifugal elastic stresses in parallel-sided disks with hub-to-rim temperature gradient of 300° to 1200° F.



(b) Five minutes after ignition when hub-to-rim temperature gradient is maximum.

Figure 4. - Combined thermal and centrifugal stresses in turbine disk at rated speed and rated tailpipe temperatures.

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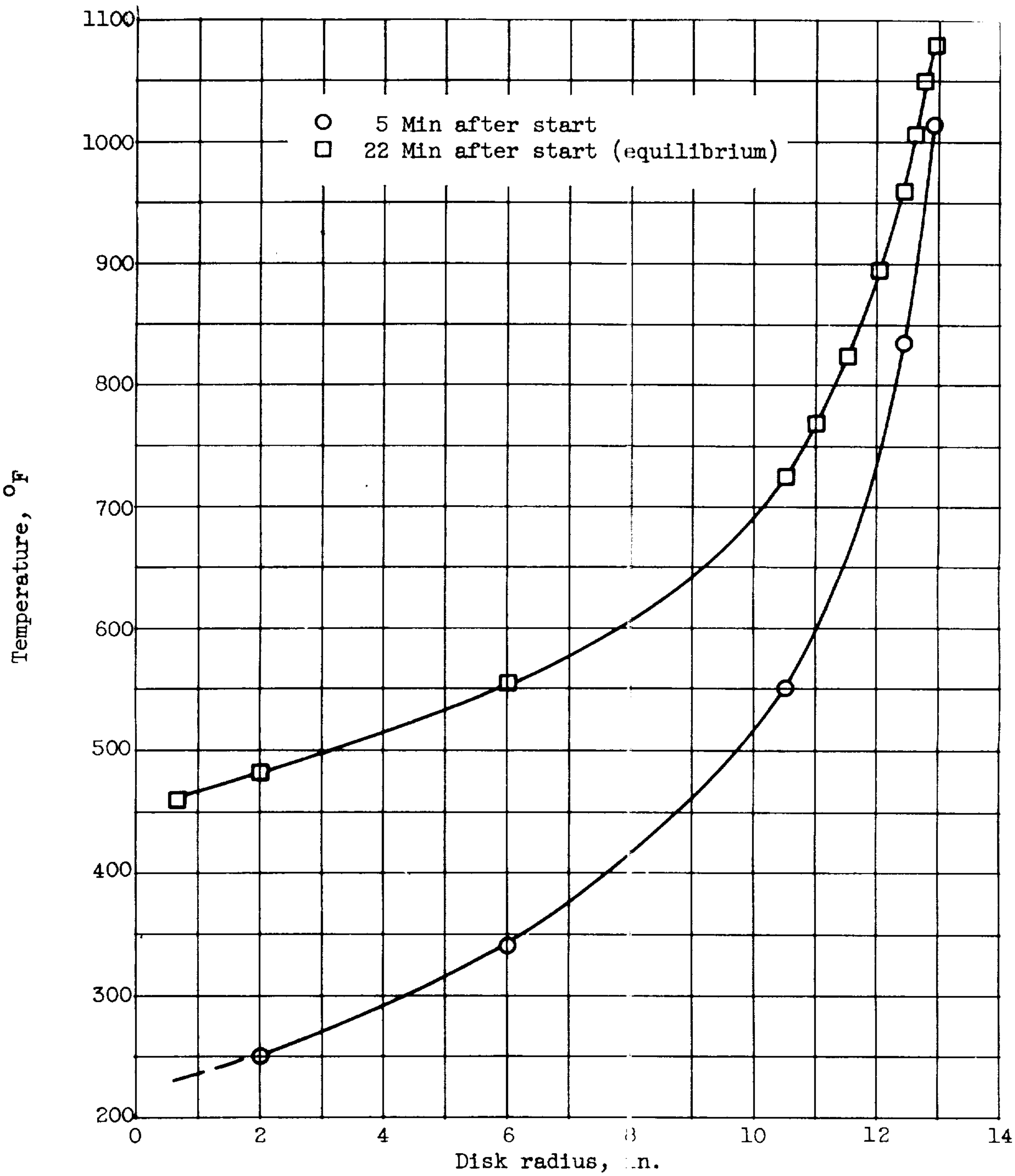

Figure 5. - Radial temperature distribution for equilibrium and transient conditions.

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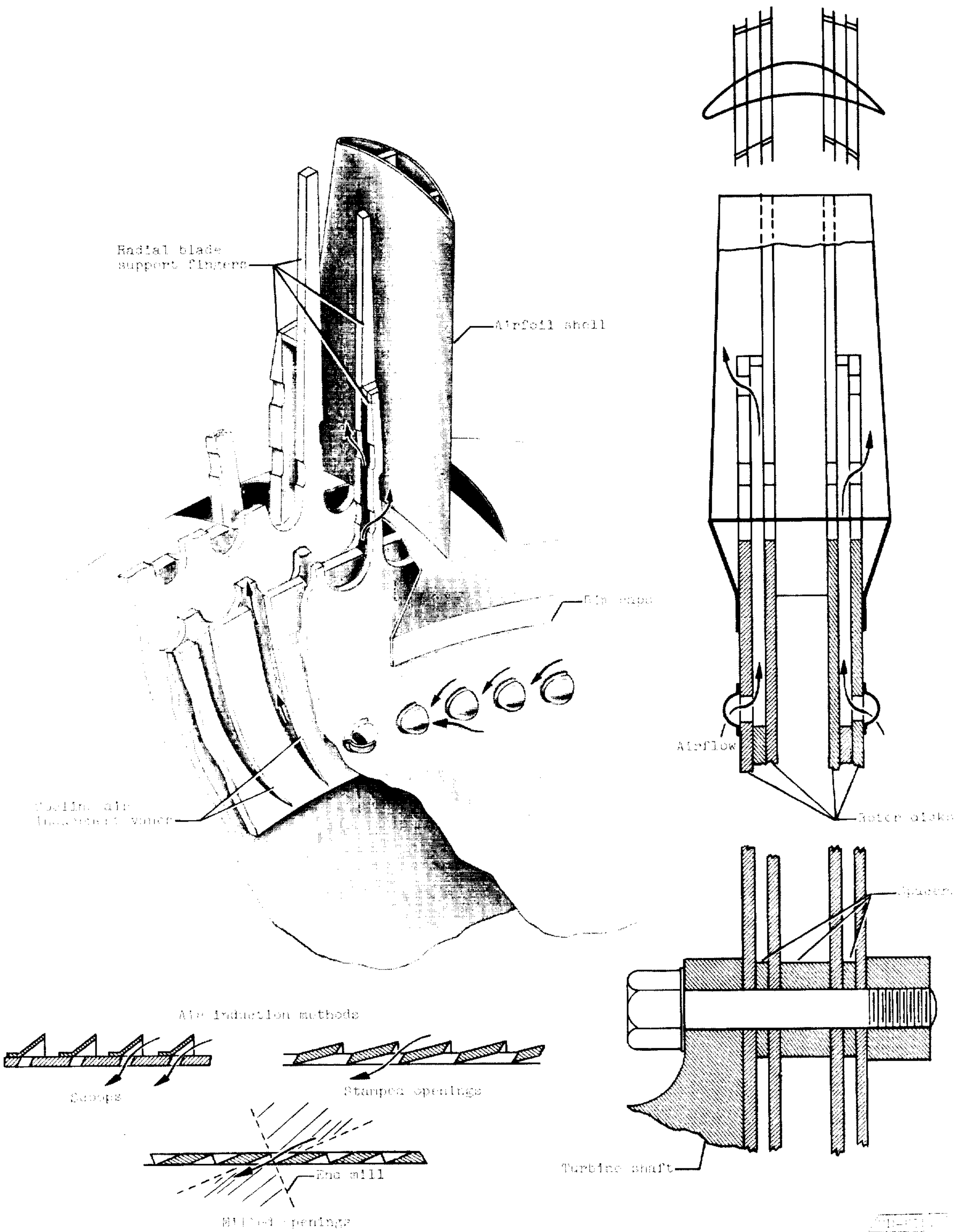

Figure 6. - Methods of scooping in cooling air without use of central hole.

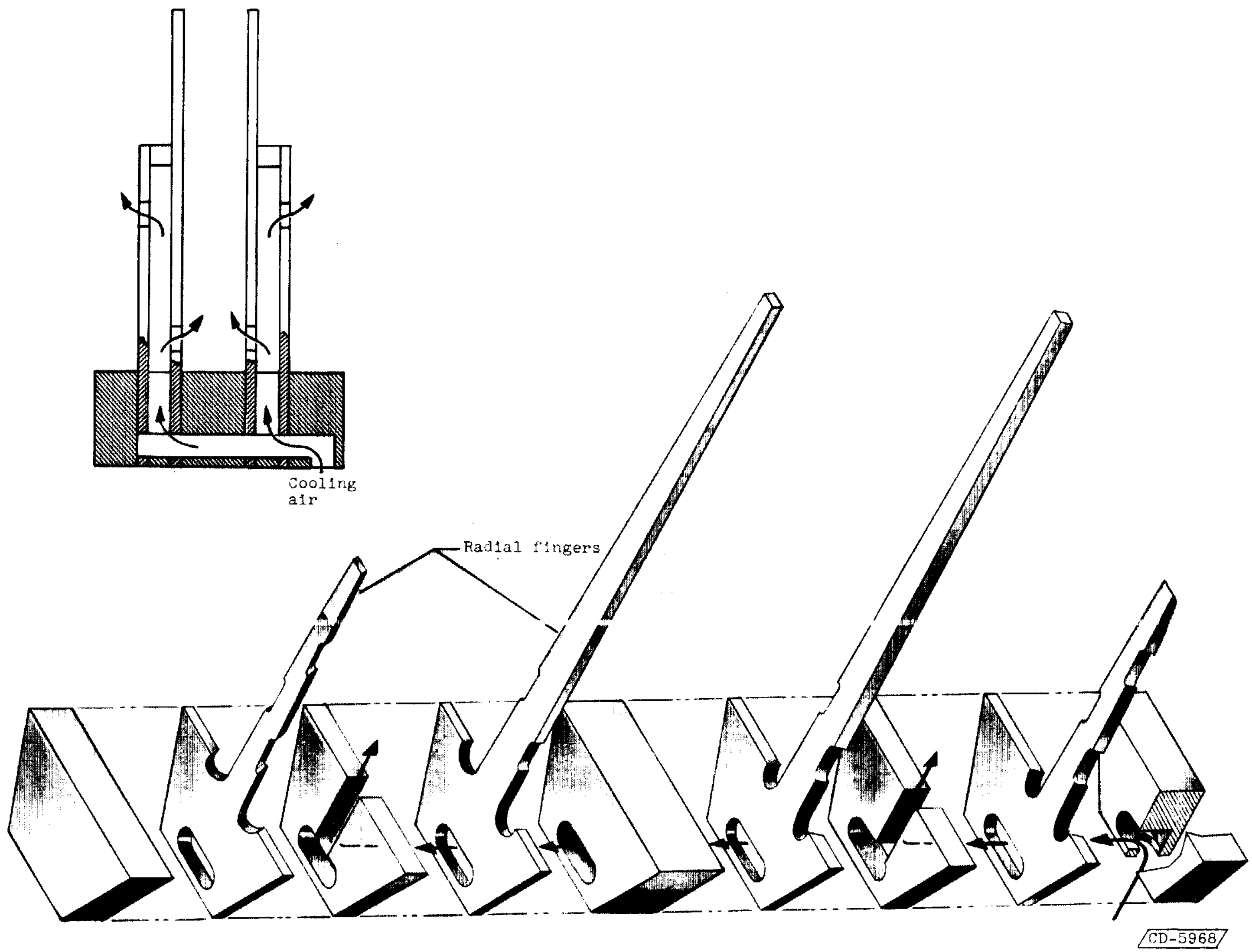

Figure 7. - Exploded view of first test blade base and radial finger construction.

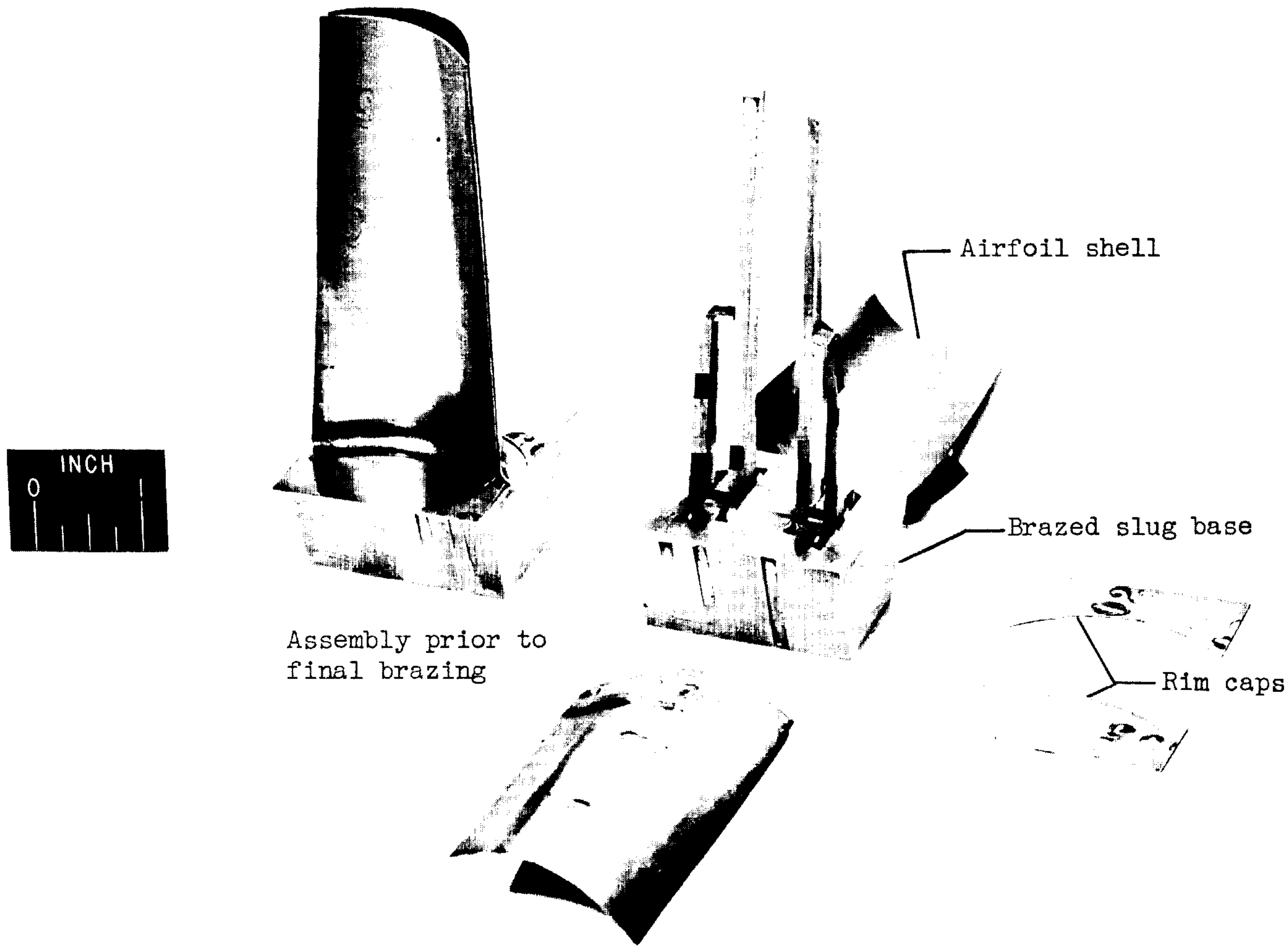

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Figure 8. - Slug base and fingers brazed into integral unit. Airfoil shells with rim cap piece.

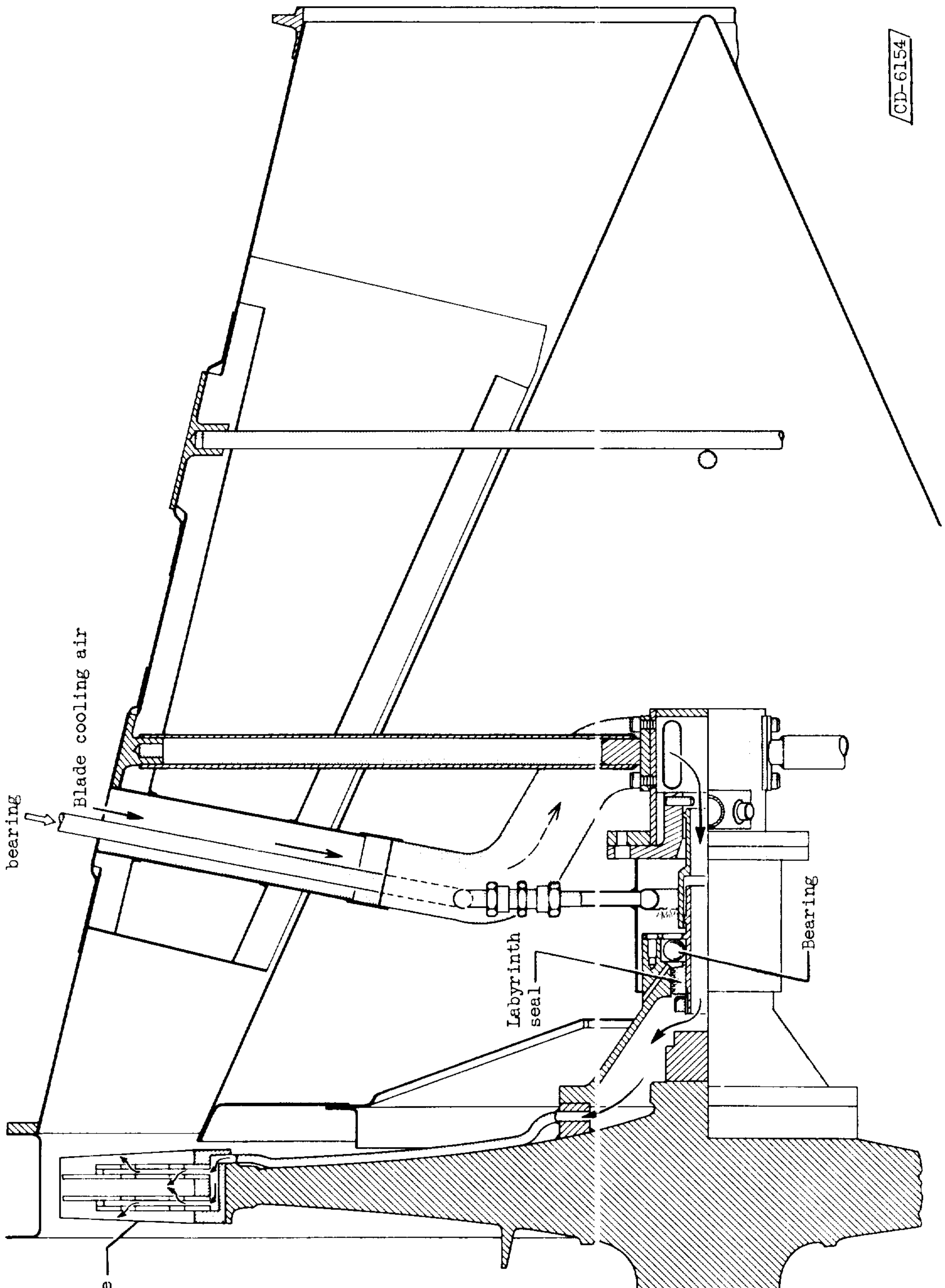

Figure 9. - Cooling air transfer mechanism.

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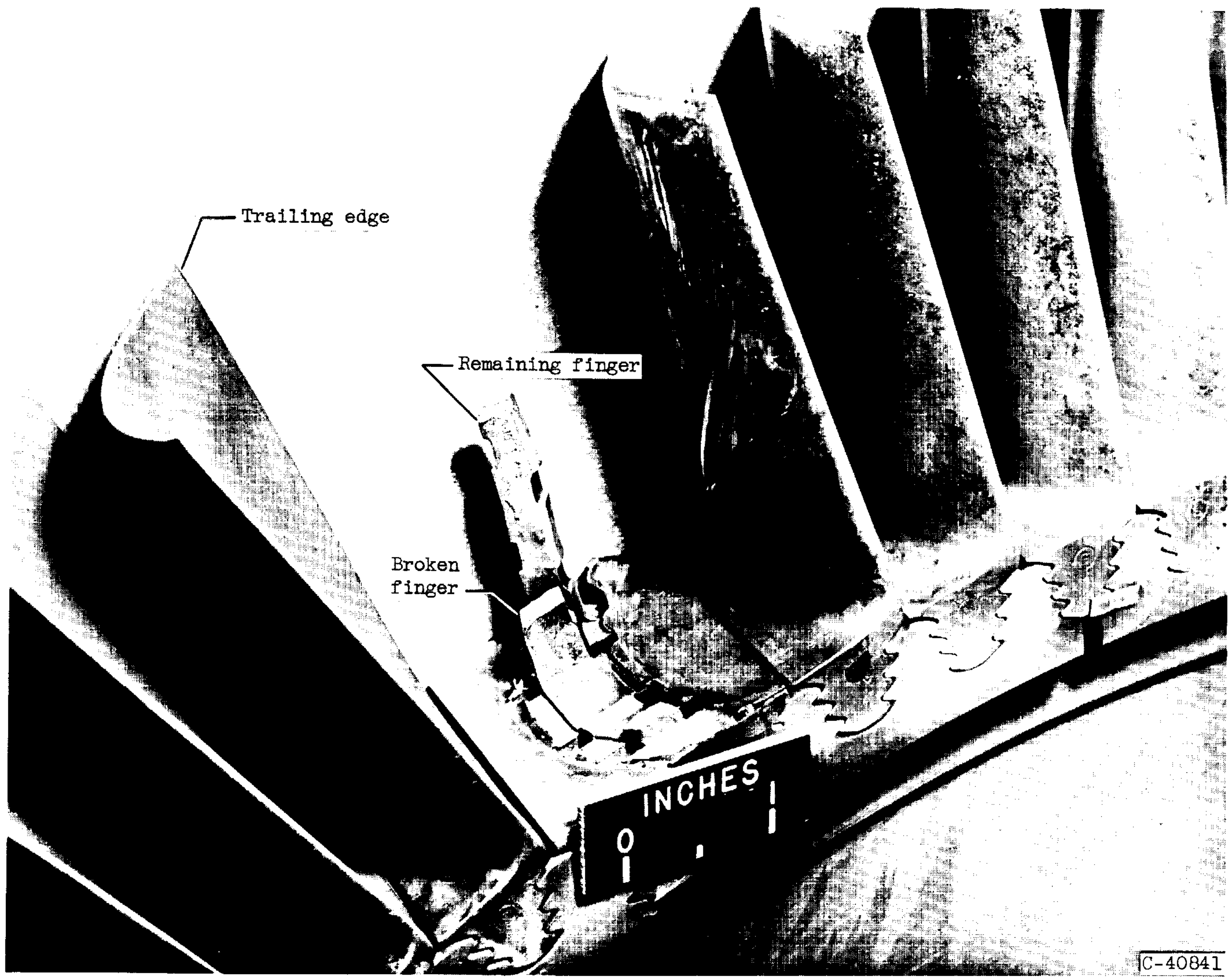
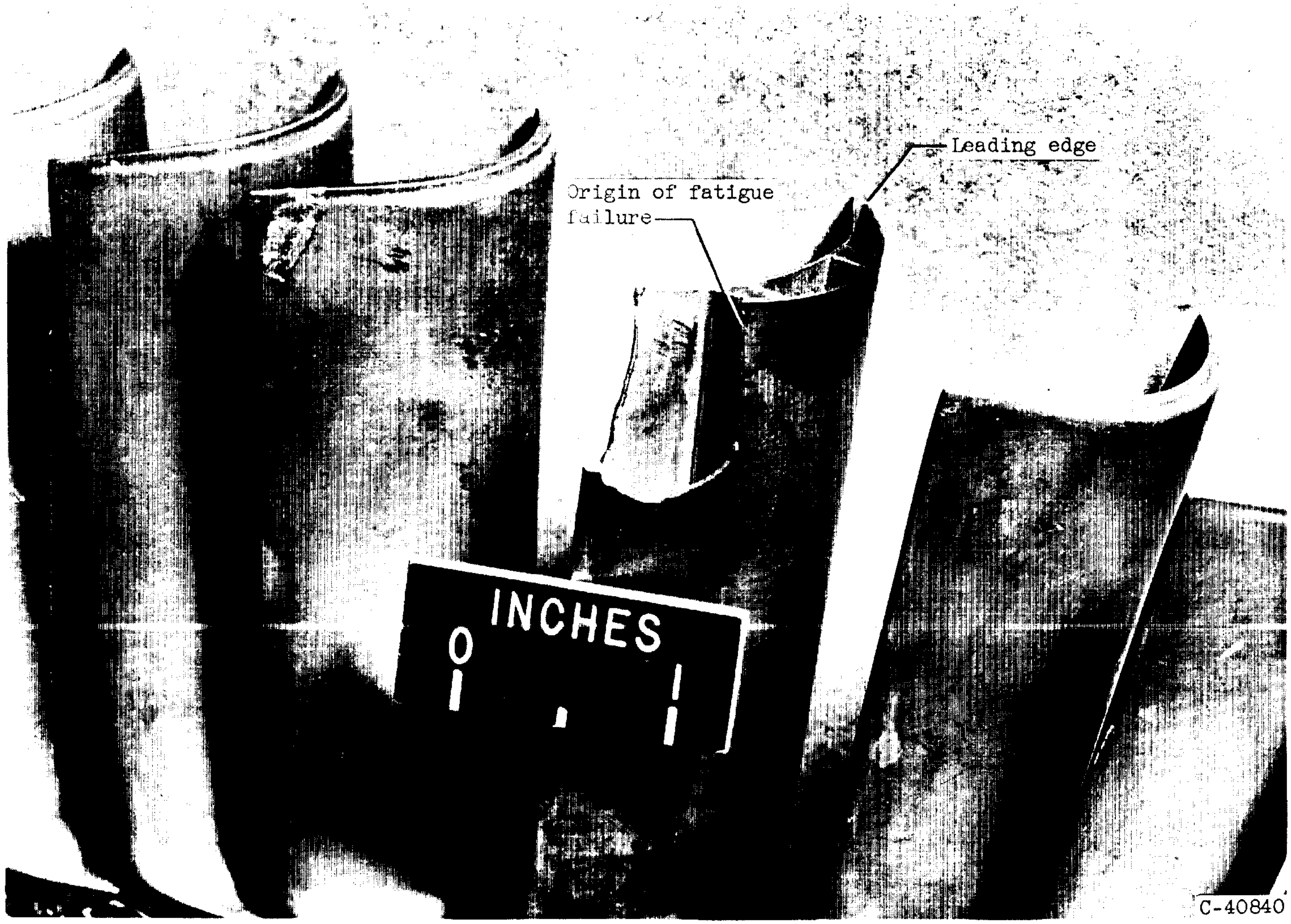


(a) Shear failure.

Figure 10. - Blade failures after $2\frac{1}{2}$ hours at rated speed.



(b) Vibration failure.

Figure 10. - Concluded. Blade failures after $2\frac{1}{2}$ hours at rated speed.

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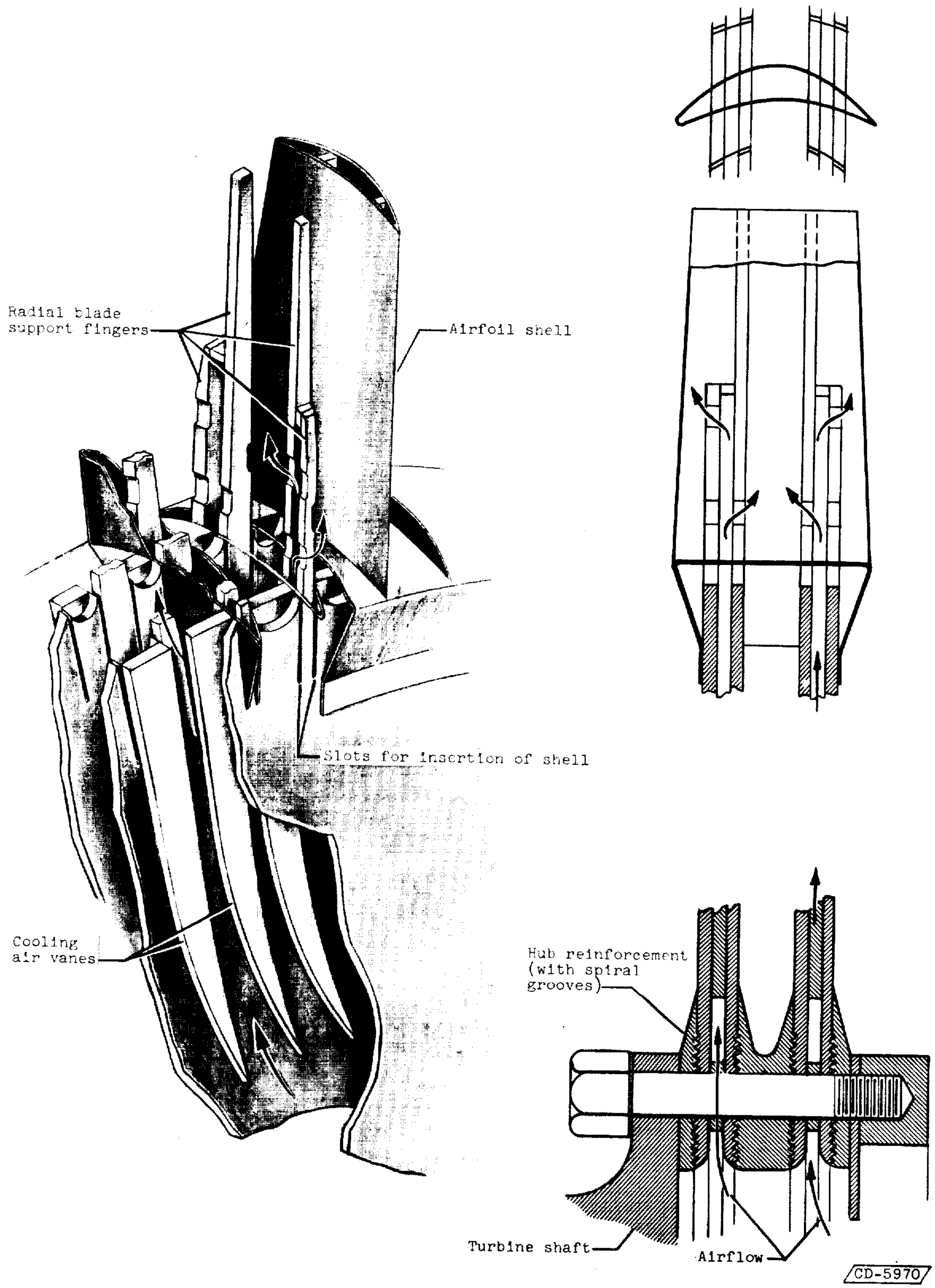

Figure 11. - Blade modification to avoid type failure encountered with first test blades.

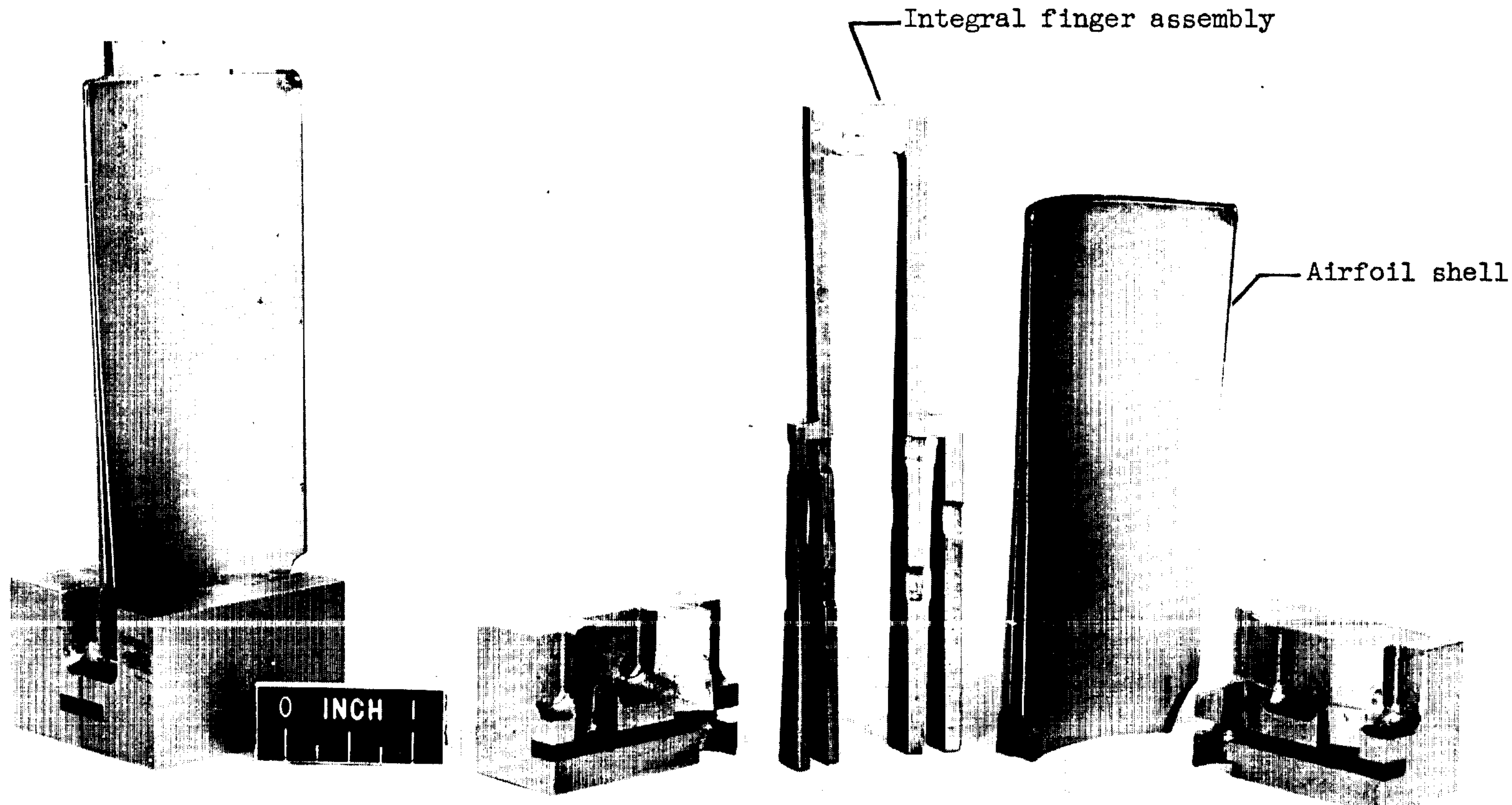

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Figure 12. - Modified test blade construction.

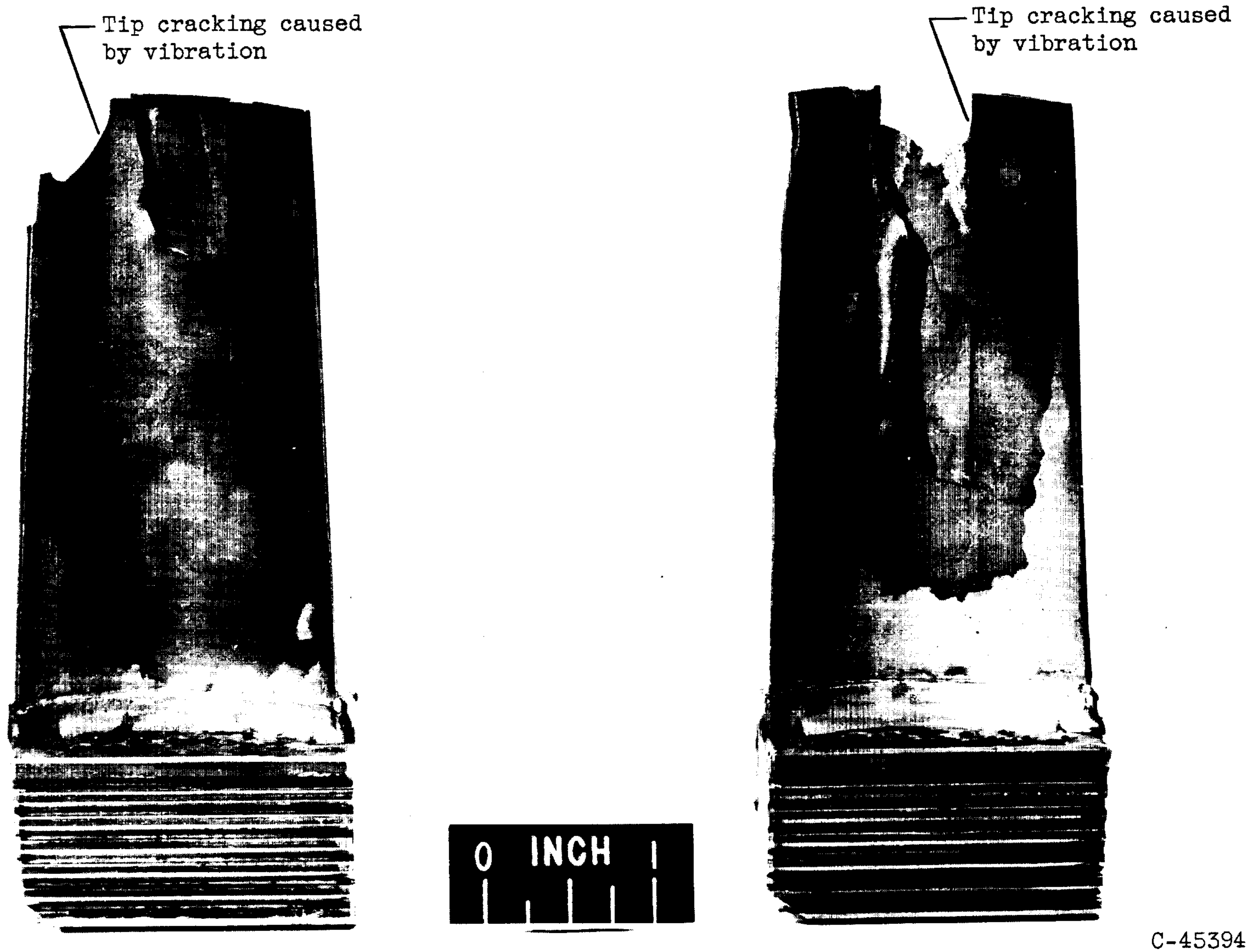

Figure 13. - Vibration failures of modified test blades.