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AD481394

C.W.R. REPORT NO. 300-38

A0-481-394

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C.W.R. REPORT NO. 300-38

## TABLE OF CONTENTS

## Foreword

Under BuAer contract No(as) 56-495-C, the Research Division, Curtiss-Wright Corporation conducted analytical, design, and development work on a transpiration air cooled turbine rotor blade concept. The following work statement is quoted, in part, from the Contract under the heading of Articles or Services:

> "Item I. Conduct a design and test study for the purpose of determining the feasibility of increasing the permissible turbine inlet temperature of a J-65-W-6 turbo-jet engine by the use of transpiration air cooled turbine rotor blades."

(Under this Item, reference is made to the Statement of Work as put forth in C.W.R. letter dated 10/5/55 which proposed a program to derive data between 1560°F and 1800°F turbine inlet temperature operation.)

This report is submitted in fulfillment of the requirement to summarize the work conducted on Item I, above, which culminated in the design, fabrication, and full-scale engine testing of a transpiration air cooled rotor stage in a J-65 engine.

## Object

To establish the feasibility of increasing the operating temperature of aircraft turbine engines by the use of a transpiration air cooled turbine rotor assembly.

## Summary

The design, fabrication, and full-scale engine test program to establish the feasibility of increasing the turbine inlet temperature of jet engines, through the use of a transpiration air-cooled rotor stage, was successfully completed. Testing was performed on a modified J65 engine using a variable area exhaust nozzle to match the compressor to the turbine for the elevated temperature operation. The maximum condition tested was at an average turbine inlet temperature of 1845°F with an increase in cooling air flow over the standard uncooled (1640°F) configuration of approximately 2%. While this percentage was more than required by the cooled blade assembly, it was the minimum flow from a controllability standpoint. The test data clearly indicates this percentage of coolant would be sufficient for turbine inlet temperatures up to approximately 2100°F. While the program required investigation only between 1560°F and 1800°F turbine inlet temperature, 2 hours and 20 minutes of testing was conducted between 1800°F and 1845°F to demonstrate the structural durability of the design, to collect additional data relevant to blade metal temperatures, and to determine engine performance characteristics. The 1845°F turbine inlet temperature was not exceeded because it was felt this was the approximate maximum temperature to which the other components in the test vehicle could be expected to perform without failure.

The engine performance data derived from this program indicates that by the increase in turbine inlet temperature afforded through the use of transpiration air-cooled blades substantial increases in engine thrust are obtainable, and that any degradation in turbine performance is negligible. Temperature measurements taken at the 1845°F operating points indicated the following: (1) skin temperature of the blade trailing edge did not exceed 1184°F; and (2) the temperature of the structural strut, taken at the root area, did not exceed 740°F. Inspection of the condition of the blades after the engine test clearly indicated the feasibility of the concept was successfully established. Except for some additional development required in one or two minor problem areas, achievement of ultra-high turbine inlet temperatures through the use of transpiration air cooled blades appears entirely within the realm of possibility.

Prior to initiation of the test phase of the program, a review was made of the porous materials available, or under development, which indicated that the inherent weakness of the porous material would necessitate a unique blade design to withstand the stringent temperature and stress requirements of a turbine rotor blade. Preliminary design studies showed that a strut-supported blade would be the most practical. The design finally selected was a brazed assembly in which a porous airfoil skin forms the main gas passage, a sheet metal shelf separates the cooling flow from the main gas stream as well as metering it to the blade, and an internal strut which forms the main structural element, carries the airfoil and shelf, as well as provides the root attachment. The material used for the porous airfoil was a constant permeability woven wire cloth which was

sintered and cold worked to give the required strength and porosity characteristics. It was not considered practical, for this feasibility program, to resort to a variable permeability airfoil material. However, by judicious design in metering of the coolant flow to the blade passages, and also in the design of the passages per se, some degree of effective variable permeability was achieved. The blade was made mechanically and aerodynamically interchangeable with the first stage turbine rotor blade of the J65 test vehicle. Coolant air to the rotor blade assembly was provided by internally routing a portion of the compressor discharge flow through a series of orifices, a rotating seal, and then through a simple impeller attached to the rotor disc which fed the air radially to the blades.

Fabrication of the transpiration air cooled turbine blades required extensive development in the following areas: (1) control of porosity and dimensional characteristics of the porous tube from which the airfoil was made; (2) stretch-forming of the porous metal tube into an airfoil shape; (3) development of a brazing technique to bond the porous airfoil to the strut; and (4) development of a satisfactory method of inspecting the brazed joint. It is considered significant that solutions to the above problem areas were satisfactorily developed within the scope of this feasibility program. The process was sufficiently developed so that blade assemblies were made from combinations of the following materials: Airfoil - N-155 porous material and H.S.-25 porous material; Struts - H.S.-31 and Inconel 713; Shelves - H.S.-25 sheet alloy. In addition, the blade fabrication process appears practical and adaptable to low cost mass production techniques without the utilization of specialized tooling. Inasmuch as the program was to establish the feasibility of the concept, only two sets (220 blades total) of transpiration cooled rotor blades were fabricated. In the first set, the blades consisted of H.S.-31 struts, H.S.-25 shelves, and airfoil skins made either from N-155 or H.S.-25 porous material. The second set consisted of struts made from Inconel 713C alloy with the airfoil skin and shelf material the same as in the first set.

## Conclusions.

1. The feasibility of achieving higher turbine inlet temperatures and increasing engine performance by the use of transpiration air cooled blades has been established.
2. Compromises in the basic aerodynamic or mechanical design of the turbine are not necessary for the type of transpiration cooled blades developed herein.
3. This transpiration air cooled turbine rotor blade can be fabricated by low cost, mass production techniques.
4. Significant advances have been made in brazing of the porous material, but the brazing cycle is critical and requires additional research to broaden its time-temperature relationship.

## Recommendations

1. Conduct further research and development of the unique transpiration air cooled turbine blade concepts developed herein to extend the operating turbine inlet temperature of turbine engines to 2500°F, and above.

2. Conduct analytical studies to determine the optimum utilization of gas turbine power plants incorporating the increase in operating gas temperatures afforded by the use of transpiration cooled turbine concept.

C.W.R. REPORT NO. 300-38

# I. DESIGN

Reported by: N. Lauziere
N. Lauziere

## A. Transpiration Cooled Turbine Rotor Blade

1. Heat Transfer Design Procedure
2. Heat Transfer Final Design
3. Structural Design

## B. Engine Studies

1. Combustion Chamber
2. Turbine Blades
3. Rotor Assembly
4. Variable Area Exhaust Nozzle
5. Air Filter Study

# I. Design

## A. Transpiration Cooled Turbine Rotor Blade

### 1. Heat Transfer Design Procedure

The work on the transpirational cooled turbine rotor blade program began with the heat transfer analysis which was required in order to determine the permeability and coolant flow requirements for operation up to 2000°F turbine inlet temperatures. Basically, the problem here was to calculate the airfoil skin temperature distribution of the rotor blade with a permeability coefficient-to-thickness ratio $(K/t)$ distribution specified. In addition, the metering orifices at the base of each cooling air passage, required to obtain the cooling airflow distribution to the skin, was determined. The design procedure used for this analysis is developed in Reference 1.

Previous theoretical analysis and experiemental investigations indicate that transpiration cooling of gas turbine blades is very effective. The local cooling airflow theoretically required for a constant blade wall temperature is described below.

To obtain an ideal cooling airflow distribution, it is necessary to fabricate a blade shell of variable spanwise and chordwise permeability. With present state of development of permeable materials, this is practically impossible. Moreover, even if obtainable, the required spanwise variation in permeability would give a blade which would be undesirable from stress considerations. The problem then becomes one of determining the blade surface temperature distributions for a particular air pressure at the base of the blade, and for a shell permeability coefficient-to-thickness ratio $(K/t)$ which is specified throughout the blade, to obtain a reasonable blade surface temperature distribution. Orifices may be installed at the base of the blade at the entrance to each cooling-air passage to vary the cooling-air pressure of each passage. The method of calculation which was utilized in this design is based on the following assumptions:

1. The film coefficient of heat transfer from gas to blade is constant throughout the blade surface.
2. The effective gas temperature is constant, and the gas temperature distribution is known.
3. The cooling passage is radial, and only centrifugal pressure effects are considered in the balance of forces acting on the control area in the calculation of the pressure distribution of the cooling air in a passage.

4. The overall temperature of the cooling air inside the passage is assumed, and the spanwise variation of cooling-air temperature is linear from root to tip.

The method requires a knowledge of the following:

1. Complete geometry of the blade, both skin and cooling passages.
2. Gas velocity and gas pressure distribution around the blade surface.
3. Variation of permeability coefficient-to-thickness ratio $(K/t)$ around the blade skin.
4. Pressure and temperature of the cooling air.
5. The angular velocity of the turbine blades.

The method of calculation is as follows:

The internal pressure distribution of the cooling air in a passage is calculated from the following equation:

$$\ln\left(\frac{P_i}{P_{i,r}}\right) = \frac{\omega^2 T_{c,r}}{8 R a^2}\left(\frac{a\, r_r}{T_{c,r}} - 1\right)\ln\left(1 + \frac{a}{T_{c,r}} X\right) + \frac{\omega^2}{8 R a} X \qquad (1)$$

Eq. 17, Ref. 6

This is an exact solution of the differential equation of flow through a rotating passage with ejection through a porous wall, using assumptions 3 and 4 above. The error resulting from these assumptions is small at low coolant flows. The pressure, $P_{i,r}$, is chosen so that it is less than $P_a$ but greater than $P_{e,r}$ and such that it remains greater than $P_e$ throughout the passage. This insures coolant flow through the openings of the porous blade wall throughout the entire blade. Various values of $P_{i,r}$ may be chosen until this condition is satisfied, and also, the optimum distribution of $T_w$ can be obtained.

The blade skin temperature at any spanwise position, $X$, is now calculated by a trial and error process which at the same time gives the local cooling air flow, $\rho v$. This procedure is as follows:

## Laminar Gas Flow Region

A value of $T_w$ is assumed. Based on this value $C_k$ is calculated from the known $(K/t)$ at the particular value of $X$, and the equation giving $C_k$ is a function of $T_w$ and $K/t$. For a wire mesh blade shell the following equation is used:

$$C_k = 3.050\left[(K/t)\frac{1}{\mu T_w}\right]^{5/8} \quad (2)$$

Ref. 6

For the value of $C_k$, $\rho V$ may be calculated from:

$$\rho V = C_k \left(P_i^2 - P_e^2\right)^n \quad (3)$$

where $n = 5/8$

The coolant velocity, $V$ is now calculated from:

$$V = \frac{\rho V}{\rho} = (\rho V)\left(\frac{R T_w}{P_e}\right) \quad (4)$$

Using this $V$, $-f_w$ the coolant flow parameter is calculated from:

$$-f_w = \frac{2}{E_u + 1} \frac{V}{W} \sqrt{R_e} \quad (5)$$

where: $$E_u = \frac{y}{W} \frac{dW}{dy} \quad (6)$$

The Euler number, $E_u$, is determined from the known gas velocity distribution and blade geometry. The properties for determining the Reynolds number, $Re$, are evaluated at the assumed value of $T_w$. For this value of $-f_w$, and the ratio of $T_g/T_w$, a value of $(1-\phi)$ is obtained from Fig. 5, Ref. 6. This is graphical solution of the flow equation, the heat transfer coefficient equation, and the heat balance equation (Ref. 4). The parameter $(1-\phi)$ is defined as:

$$(1-\phi) = \frac{T_w - T_c}{T_g - T_c} \quad (7)$$

$T_w$ can now be calculated. This must agree with the assumed value or the calculation must be repeated until agreement is reached. For the leading edge passage, the only modification is that $E_u = 1$ and $-f_w$ is calculated from:

$$-f_w = 0.525 \frac{V}{U} \sqrt{Re_D} \quad (8)$$

Here the characteristic length of the Reynolds number is the leading edge diameter.

## Turbulent Flow Region

A value of $T_w$ is assumed as before in the laminar region, and $C_k$ is calculated in the same manner. Now $\rho v$ is calculated from equation (3). The parameter $(1-\phi)$ is then calculated from:

$$1-\phi = \frac{\frac{2.11}{Re^{0.1}}}{\epsilon^{\left[\frac{71.3(\rho v)\,P_r^{2/3}}{Re^{0.9}}\,\frac{y}{\mu}\right]} + \frac{2.11}{Re^{0.1}} - 1} \qquad (9)$$

Ref. 6

The assumed value of $T_w$ is checked as before using equation (7).

The points of transition from laminar to turbulent flow around the blade surfaces are taken as those points where the pressure gradients vanish; in most cases, these points are points of minimum pressure (Refs. 4 and 7). With the use of this criterion, and the known gas pressure distribution, it can be determined which cooling passages lie in the laminar gas-flow region and which passages lie in the turbulent gas-flow region.

For each spanwise position, $X$, in a particular passage, a value of the local cooling air flow, $\rho v$, is obtained for every value of $T_w$ that is checked out by trial and error. With the spanwise distribution of $\rho v$ for each passage known, the cooling-air flow to the passage, $W_r$, is found by intergrating the following equation between $X = 0$ and $X = L$:

$$W_r = W_t + \int_0^L b(\rho v)\,dx \qquad (10)$$

For a capped blade $W_t = 0$
The total cooling air required is equal to the sum of all the $W_r$ terms.

## Orifice Calculation

Each passage is equipped with an orifice so that the cooling-air pressure available, $P_a$, is attenuated to the desired value of $P_{i,r}$. The diameter of each orifice is calculated from the following equations:

For subcritical pressure drops, $\left(\frac{P_{i,r}}{P_a} > 0.528\right)$

$$A_n = \sqrt{\frac{R}{2g}}\,\frac{W_r}{B}\,\sqrt{\frac{T_{c,r}}{P_{i,r}\,(P_a - P_{i,r})}} \qquad (11)$$

For supercritical pressure drops, $\left(\frac{P_{1,r}}{P_2} \leq 0.528\right)$

$$A_n = 2.067\sqrt{\frac{R}{2g}}\;\frac{W_r}{B}\;\frac{\sqrt{T_{c,r}}}{P_2} \qquad (12)$$

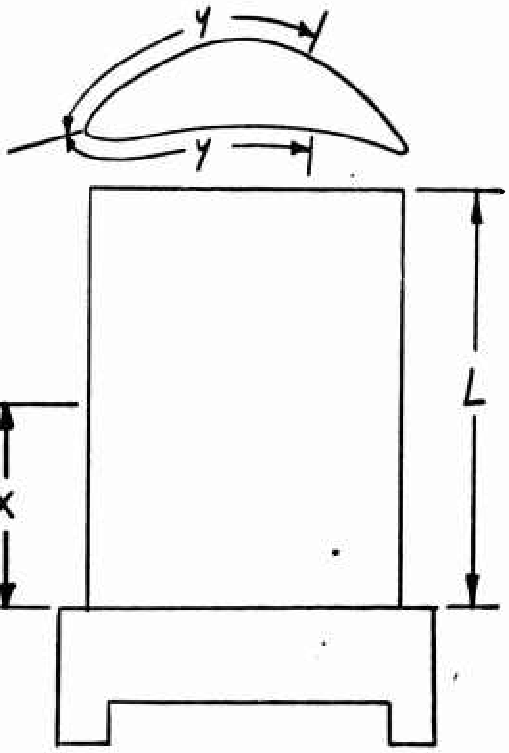
The orifice coefficient, $B$, is obtained from Figure 7, p. 47, Ref. 8.

There are much more cumbersome equations available in place of equation (1) which would eliminate assumption 3. However, its solution, combined with the trial and error procedure of finding $T_W$, results in a tedious calculation that is hardly justifiable, especially since there is no way of eliminating assumptions 1 and 4. As previously mentioned equation (1) gives only a small error at a low coolant flow.

The rate of heat addition to the coolant on its way through the flow passages is usually not accurately known, and therefore, has to be estimated. In a typical design a linear increase of approximately 200°F in cooling air temperature through the blade span is assumed arbitrarily, (Ref. 7).

A list of symbols is given in Appendix I, and references are listed in Appendix II.

The following sketch indicates the coordinate system used in the transpiration cooled blade design procedure.



## 2. Heat Transfer Final Design

Design calculations were made at 2000°F turbine inlet temperature for zero flight speed condition, and for 60,000 ft. altitude, 600 knots speed, on a transpiration air cooled strut supported rotor blade design made of a constant skin permeability. These calculations were made to determine the optimum skin permeability, blade base orifice areas, and to provide an estimate of the amount of bleed air required to maintain the skin and strut within the design temperature limits.

The calculations were made according to the method described in Part 1 of this section of the report.

Turbine airfoil external gas relative velocity distribution and static pressure distribution for sea level, zero flight speed conditions, shown in Figure 1, were calculated using the standard, but approximate methods. Velocity distribution at flight conditions remains the same as long as inlet gas total temperature and engine speed are unchanged; the pressure distribution retains the same shape but is proportionally lowered according to the change in inlet gas total pressure.

Figure 2 shows the general layout of the cooling air passages. Passages 1 and 2 were assumed to be in the laminar gas flow region, and passages 3, 4, 5, and 6 in the turbulent gas flow region. Table II gives the blade section geometry.

A pressure drop of about 5% was assumed through the combustion chamber for all conditions. The pressure drop through the stator was neglected for it was felt that this would tend to compensate for that of the cooling air from the last compressor stage to the base of the blade. Actual data at sea level static condition gave an available cooling air pressure of 192 in. Hg. Abs. at 540°F. The pressure at the exit of the combustion chamber is 182 in. Hg. Abs. At 600 knots and 60,000 ft. altitude condition, the available cooling air was calculated to have a pressure of 30.5 in. Hg. Abs. and a temperature of 456°F. Assuming a pressure drop of 5%, the combustion chamber exit pressure, at this condition, is 29.0 in. Hg. Abs.

The coolant temperature rise in the blade passages was assumed to be 250°F and 100°F for the central and end passages, respectively. These assumptions are based on results obtained on previous calculations. The coolant temperature rise is greater in the central passages because of the smaller mass rate of cooling air in comparison to the end passages. Detail calculations indicate that for 2000°F turbine inlet temperature, the optimum skin permeability for the sea level condition is $K/t = 8 \times 10^{-11}$ ft. with 3.6% cooling air flow. For the altitude condition, the optimum permeability is $K/t = 5 \times 10^{-10}$ ft. with a cooling air flow rate of 4.8%. The apparent incompatibility between the optimum permeability requirements for the sea level and altitude design condition is caused by great differences in the ratio of engine mass flow and pressures between these conditions. The engine mass flow at the sea level condition is approximately 125 lbs/sec. If a blade skin were made with a permeability of $K/t = 8 \times 10^{-1}$ ft, (optimum for sea level) at the altitude condition the ratio of pressures available to flow the coolant through the skin are reduced, and hence there would be insufficient amount of cooling air to cool the skin, this would result in excessive skin temperatures.

When the permeability factor of $K/t = 5 \times 10^{-10}$ ft. (optimum for altitude) is used at the sea level condition, the wall temperature dropped considerably and the calculations showed the coolant flow increased to approximately 10%. If, however, the available cooling air pressure at static conditions is reduced, the rate of effusing air can be reduced, and still give an acceptable temperature distribution. The permeability calculated for flight conditions could then be used at static conditions. All that is required to accomplish this is to accurately control the cooling air pressure. The small pressure difference required to give an acceptable air flow is such that it approaches the same magnitude as the accuracy to which they can be controlled or predicted. The calculations show, however, that a value of cooling air pressure at static conditions can also satisfactorily operate with the permeability calculated at flight conditions. The new cooling air pressure arrived at was 86.1 psia. On this basis new orifice diameters were calculated, and Figure 5 shows the wall temperature distribution for these conditions. The air bleed comes out to be 5.9%.

The new calculated orifice diameters cause the air bleed temperature to rise somewhat at flight conditions, and the skin temperature to drop somewhat. Figure 6 gives the new skin temperature distribution for the altitude condition with the new orifice diameters. In this case the altitude air bleed came out to be 5.9%, the same as the sea level condition. However, it is only coincidental in that the air bleed for static and flight conditions are the same for the final design values used.

From the information compiled on Figures 5 and 6, it can be seen that the skin temperature is nowhere above 1650°F, and greater than 1600°F at only two points. These local "hot spots" should not be too detrimental, as these temperature gradients will tend to be smaller since conduction in the permeable skin materials, not accounted for in the analysis, would even out the gradients.

It also can be seen from Figures 5 and 6 that a decrease in permeability from root to tip (constant chordwise) would result in excessive temperatures along passages 3 and 4. The chordwise temperature distributions appear too erratic to derive any benefit from varying the permeability in the spanwise direction alone. A variable permeability both chordwise and spanwise would be desired, but this is highly impractical from fabrication considerations as explained before.

Temperature gradients around the skin, due to non-transpiration cooled portions (brazed areas) are very small and can be neglected. The magnitude, calculated by approximate methods, is around 10°F between any transpiration cooled zone, and the brazed zone where braze prevents any transpiration cooling.

Because of two opposing effects, the higher values of permeability are required for higher altitude flight conditions and this is independent of flight speed. The higher altitude, on the one hand, lowers the pressure level which diminishes the driving force for the effusing air. The speed increases the pressure, because of the ram effect, but at the same time, this increases the cooling air temperature. This would tend to nullify the effect of speed completely and hence lead to the approximate conclusion that the permeability required increases with altitude and is almost independent of speed. Therefore, for any one condition of operation, there is some value of available cooling air pressure which will give an acceptable coolant flow and skin temperature distribution for the permeability which is optimum at the highest altitude. Some device which will automatically fix the available cooling air pressure at the desired value, at any elevation and speed , could be designed. Such a device might be an automatically operated needle valve or variable orifice in the coolant supply line.

The orifice sizes tabulated in Table I show the values of areas from the calculated orifice diameters. However, as the orifice calculated diameters are a little large for the blade passage geometry, equivalent orifices of non-circular cross-section have been designed, these are also tabulated in Table I. These orifice areas are a little larger than the calculated values due to the fact that the orifices are more of a slot than a circle, dictated by blade geometry, which does not have as great an effective orifice area. If these orifice areas should prove out to be too large during engine testing, restriction to the cooling air flow can be added at some downstream point in the cooling air ducting system.

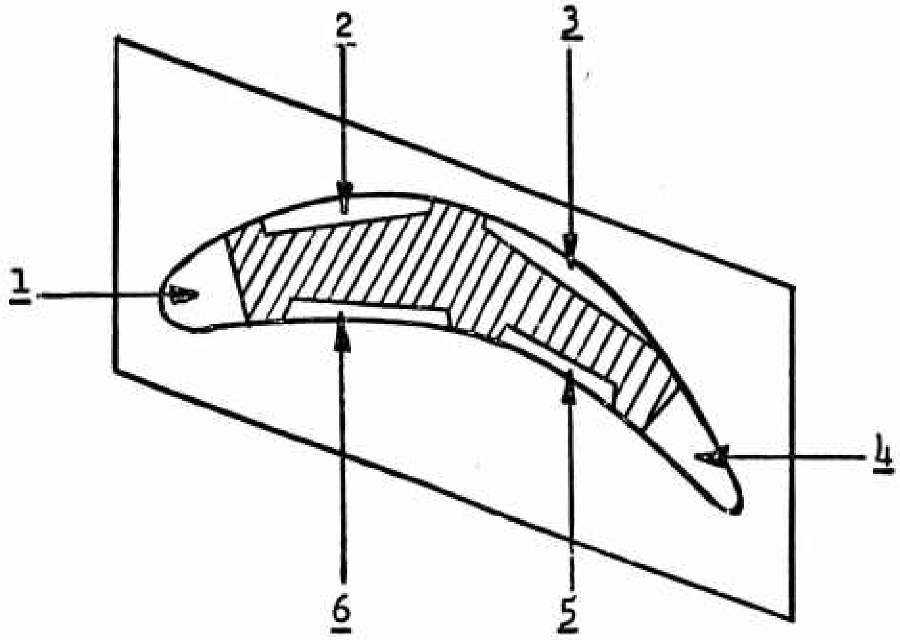
The corresponding Poroloy designation for a permeability equivalent to $K/t = 8 \times 10^{-11}$ ft. is an air flow of .0057 lb/in$^2$-sec. at $P^2/T$ of 20,000 lb$^2$/in$^5$. For a $K/t$ of $5 \times 10^{-10}$ ft. is an air flow of .018 lb./in.$^2$-sec. at a $P^2/T$ of 20,000 lb$^2$/in$^5$ and a thickness of 0.022 in.

For the test engine, the 5.9% air bleed for the final design is somewhat high, and it is believed that this can be reduced when more positive information is available from actual engine tests. A method to reduce the coolant flow, once experimental data has been obtained to verify calculated values, is by reducing to cooling air temperature before it enters the transpiration cooled blade. If necessary, this could be accomplished by the use of a heat exhanger between the last compressor stage and the inlet to the base of the turbine blade.

NACA has conducted investigations for the design of heat exchangers to reduce the temperature of the cooling air for turbine blades, using ram-air as a cooling medium. A reduction in temperature as high as 400°F was obtained with a heat exchanger weighing around 60 lbs. (Reference 10). The disadvantage of this type of exchanger is that at static conditions the cooling effect will be quite small.

It is felt, however, that there is merit to both of these approaches, and some combination of the two will probably be required to give the optimum configuration.

## BLADE COOLING AIR PASSAGE ORIFICES



| Passage | Calculated Orifice -in.$^2$ | Designed Orifice -in.$^2$ |
|---|---|---|
| 1 | None | .02380 |
| 2 | .00705 | .00904 |
| 3 | .00783 | .01087 |
| 4 | .01152 | .01626 |
| 5 | .00396 | .00568 |
| 6 | .00749 | .01082 |

Table I

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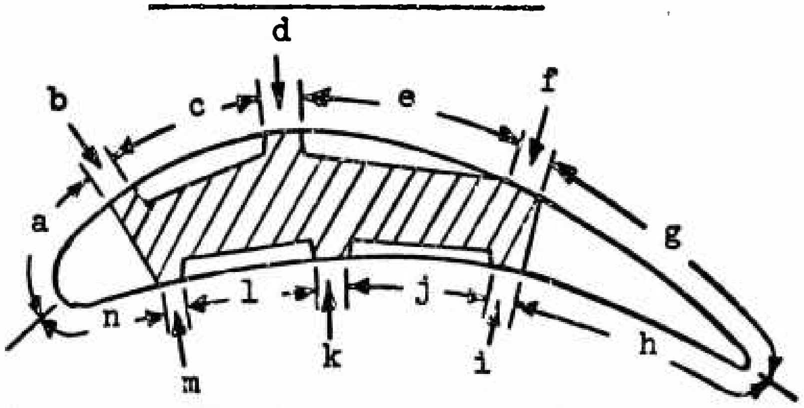
## BLADE SECTION GEOMETRY



| Dimension | Root Section (R=11.805 in.) | Mid Section (R=13.780 in.) | Tip Section (R=15.125 in.) |
|---|---|---|---|
| a | .2510 | .1830 | .1736 |
| b | .1000 | .0856 | .0720 |
| c | .2850 | .2840 | .2956 |
| d | .0700 | .0700 | .0760 |
| e | .3636 | .3850 | .3400 |
| f | .1000 | .0790 | .1430 |
| g | .3300 | .2830 | .2160 |
| h | .3762 | .2780 | .2496 |
| i | .0690 | .0680 | .0760 |
| j | .1500 | .1582 | .1650 |
| k | .0680 | .0625 | .0760 |
| l | .2116 | .2184 | .2350 |
| m | .0700 | .0700 | .0720 |
| n | .2228 | .2590 | .2664 |

Root Section (Total Perimeters - Suction 1.4996 in. - Pressure 1.1676 in.)

| Passage | Outer Perimeter (in.) | Inner Perimeter (in.) | Passage Area (in.$^2$) | y (in.) |
|---|---|---|---|---|
| 1 | .4738 | .640 | .02399 | 0 |
| 2 | .2850 | .544 | .00905 | .4935 |
| 3 | .3636 | .747 | .01039 | .8878 |
| 4 | .7026 | .652 | .01553 | 1.4996 |
| 5 | .1500 | .342 | .00520 | .6474 |
| 6 | .2116 | .492 | .00859 | .3986 |
| Mid Section (Total Perimeters - Suction 1.3696 in. - Pressure 1.1141 in.) | | | | |
| 1 | .4420 | .5930 | .01644 | 0 |
| 2 | .2840 | .5480 | .00882 | .4106 |
| 3 | .3850 | .7566 | .00770 | .8151 |
| 4 | .5610 | .5560 | .00977 | 1.3696 |
| 5 | .1582 | .3390 | .00334 | .6890 |
| 6 | .2184 | .4892 | .00700 | .4382 |
| Tip Section (Total Perimeters - Suction 1.3162 in. - Pressure 1.1460 in.) | | | | |
| 1 | .4400 | .5480 | .01030 | 0 |
| 2 | .2956 | .5736 | .00859 | .3934 |
| 3 | .3400 | .7900 | .00638 | .7872 |
| 4 | .4656 | .4450 | .00601 | 1.3162 |
| 5 | .1650 | .3470 | .00522 | .7319 |
| 6 | .2350 | .5120 | .00839 | .4559 |

Table II

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## CHORDWISE VARIATIONS OF RELATIVE GAS VELOCITY AND GAS PRESSURE

### Static, Sea Level Conditions

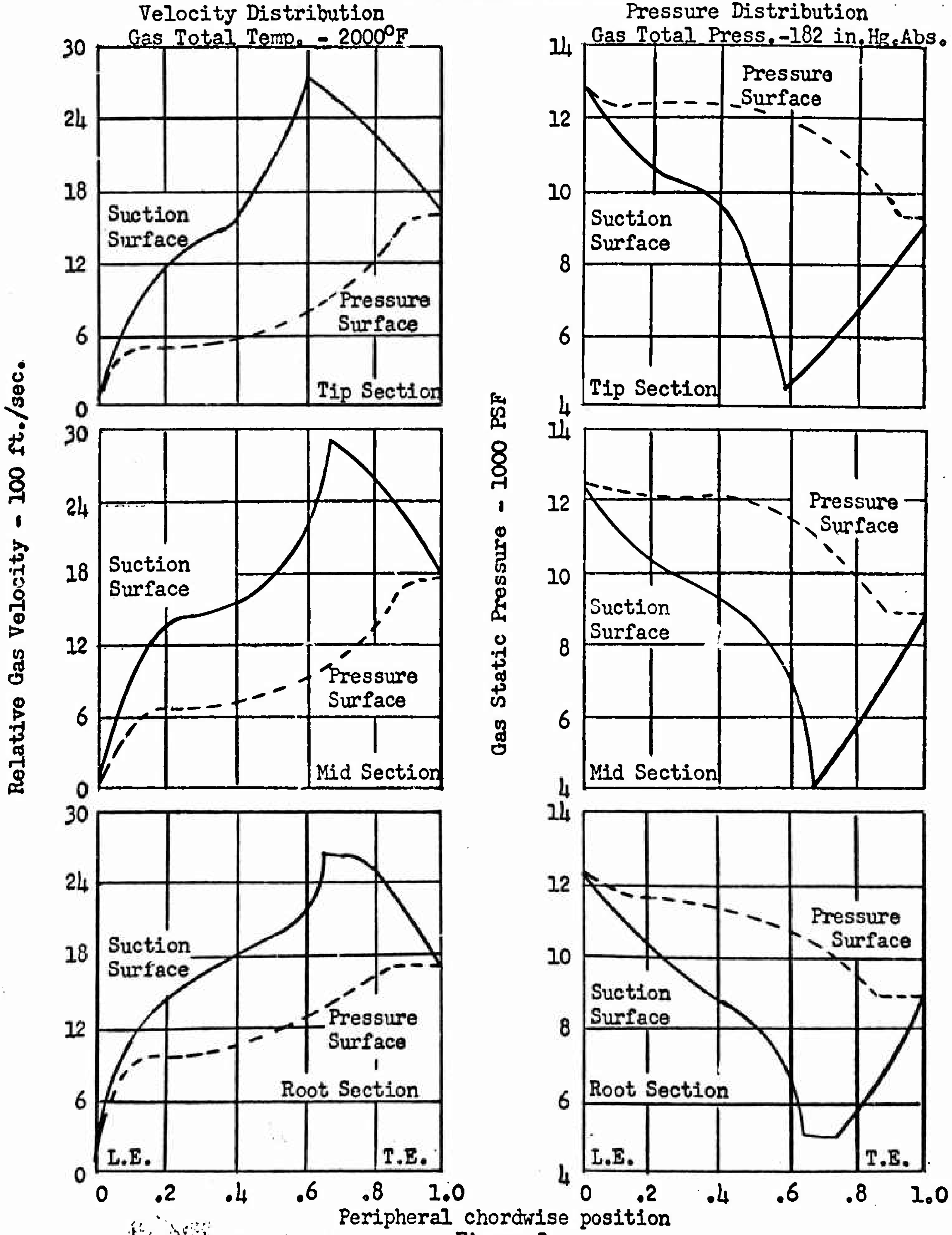
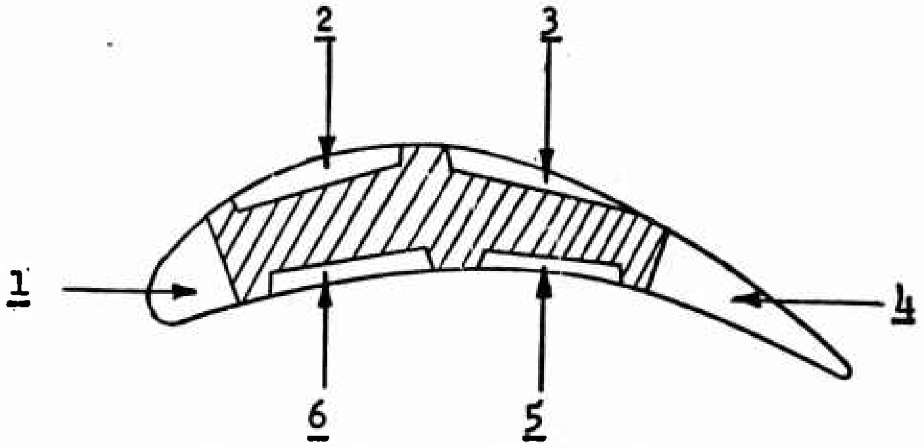
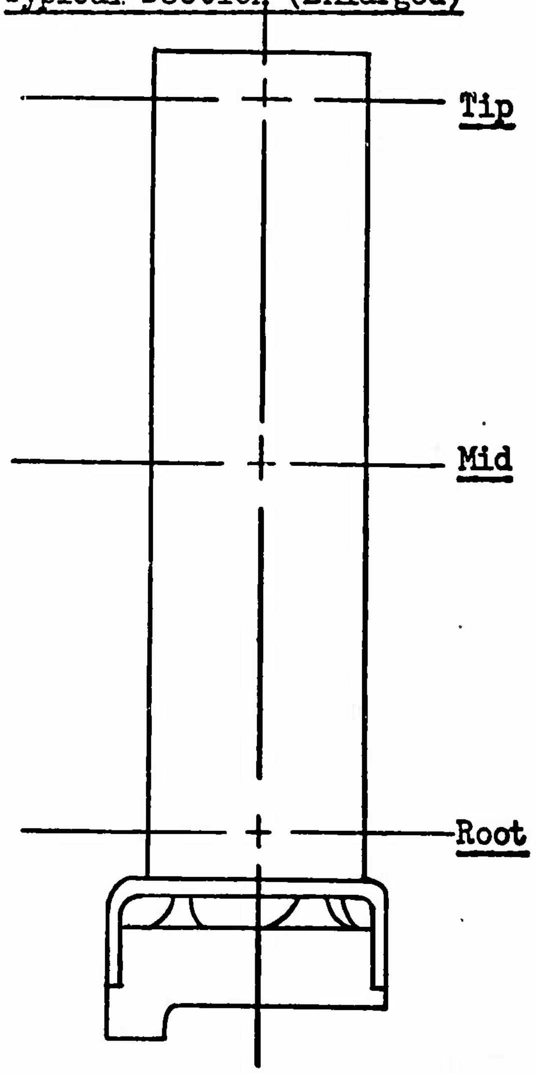


Figure 1

## J65 TRANSPIRATION COOLED
## TURBINE ROTOR BLADE



Typical Section (Enlarged)



Blade Sections

Figure 2

C.W.R. REPORT NO. 300-38

## CHORDWISE WALL TEMPERATURE DISTRIBUTION

Static, Sea Level - 3.6% Air Bleed - K/t=8x10$^{-11}$ ft.

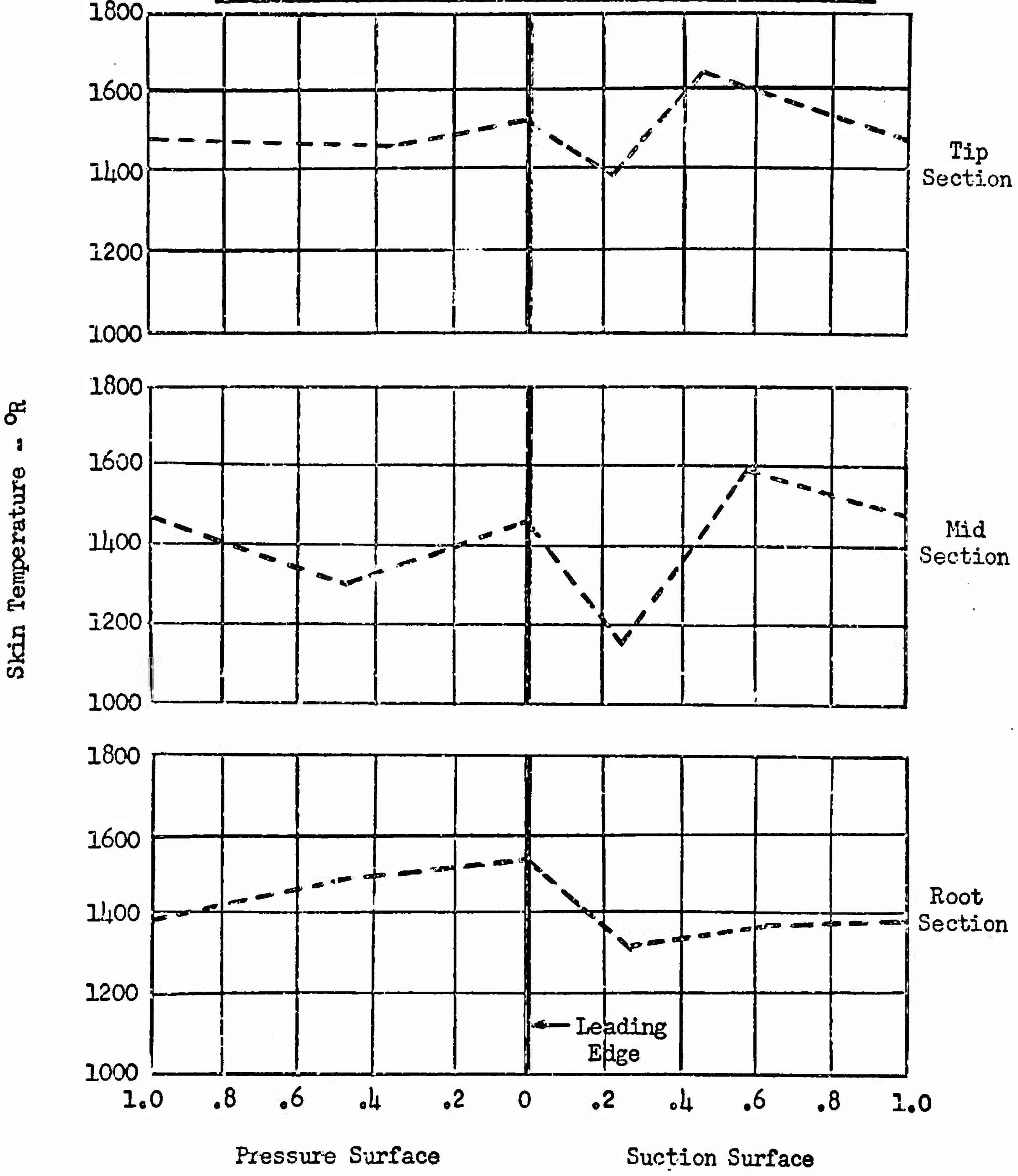


Figure 3

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CHORDWISE WALL TEMPERATURE DISTRIBUTION

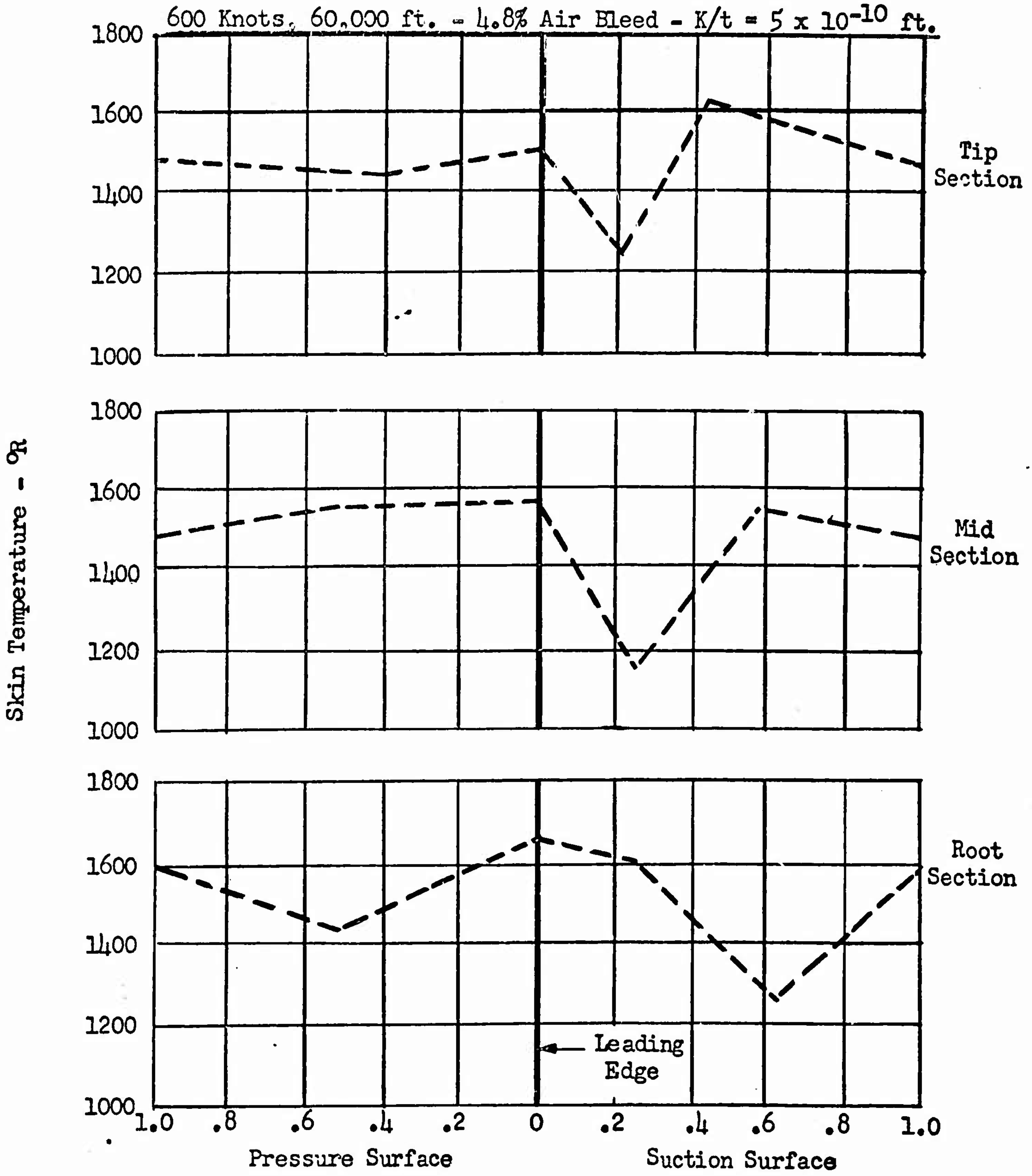
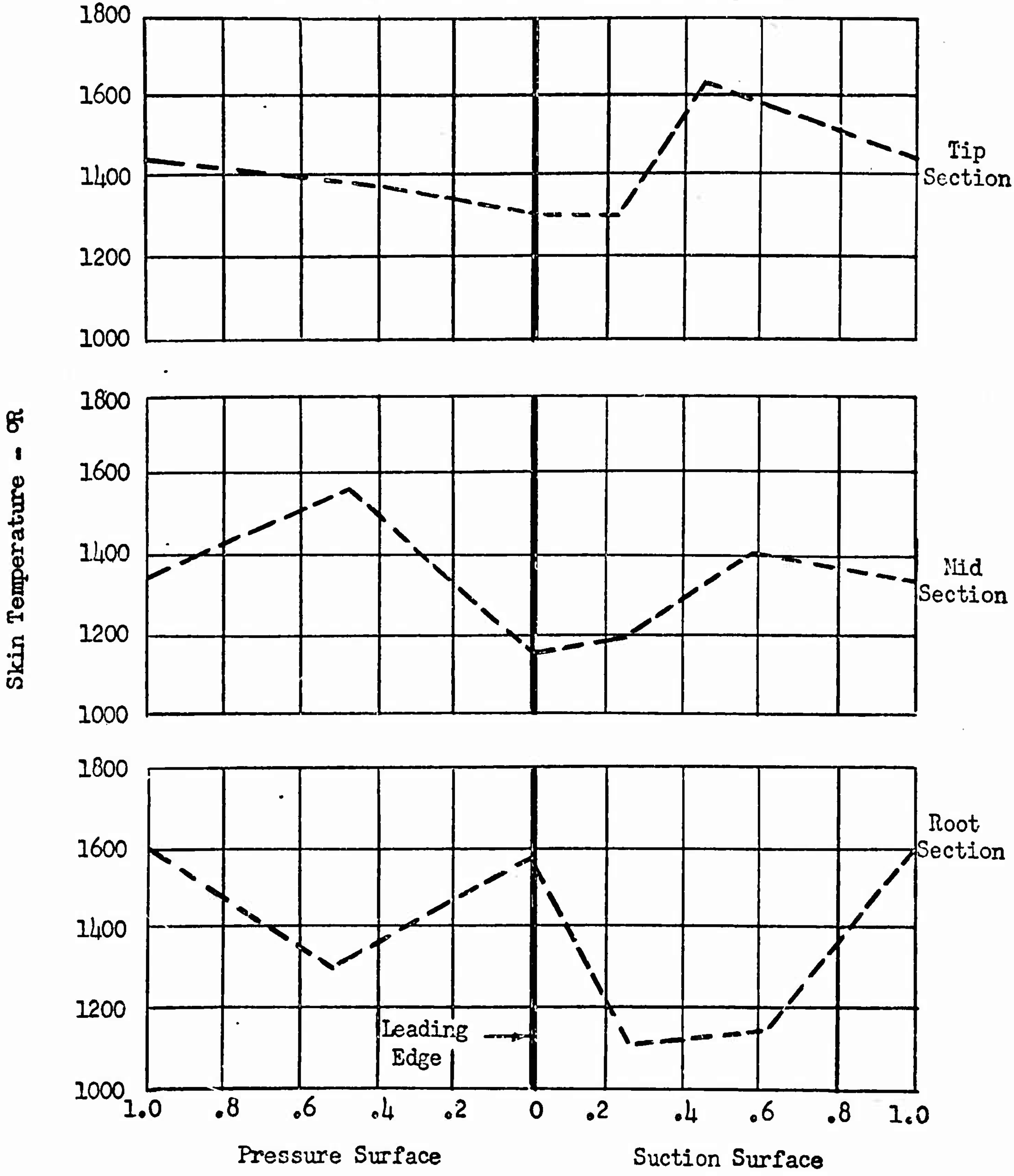


Figure 4

CONFIDENTIAL

CHORDWISE WALL TEMPERATURE DISTRIBUTION

Static, Sea Level - 5.9% Air Bleed - K/t=5x10$^{-10}$ ft.
(Cooling Air Pressure Reduced to 86 psia)



Peripheral Chordwise Position

Figure 5

C.W.R. REPORT NO. 300-38

## CHORDWISE WALL TEMPERATURE DISTRIBUTION

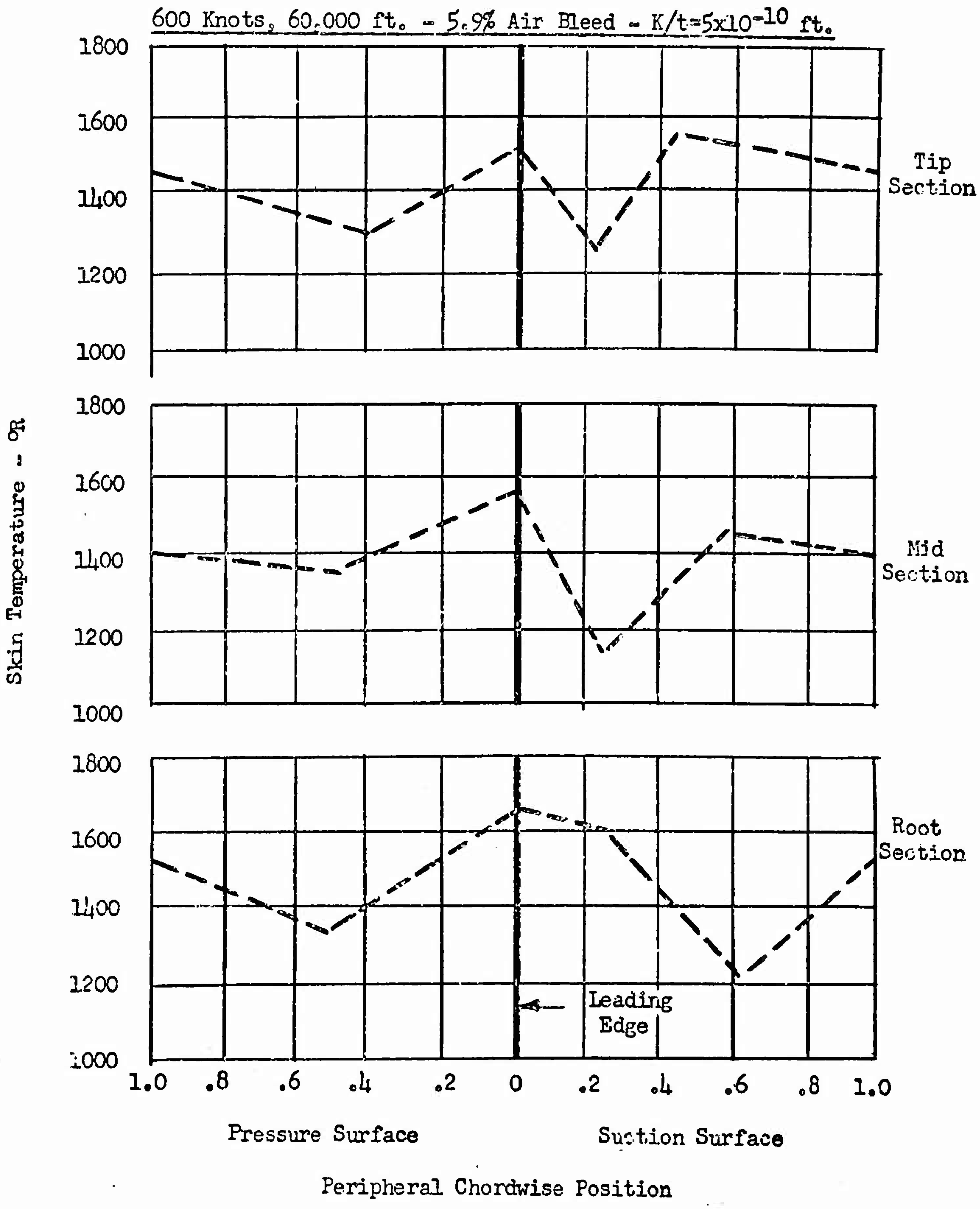
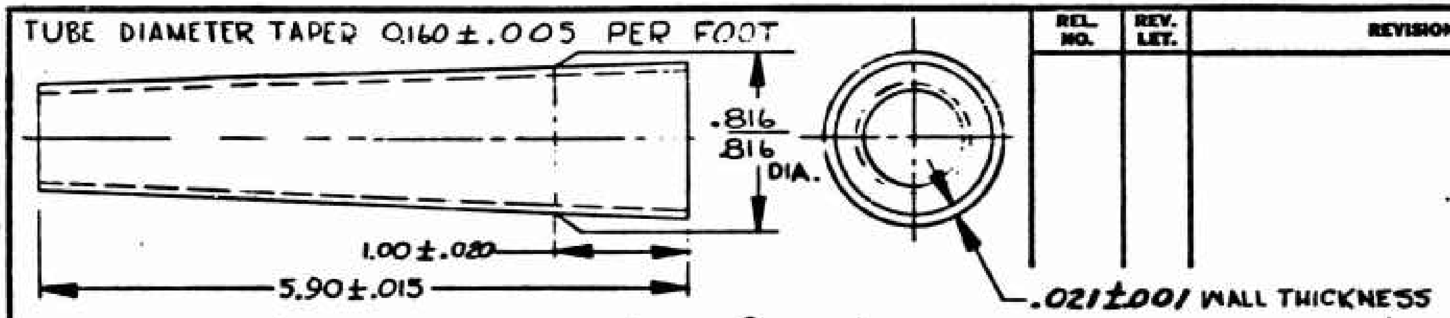


Figure 6

C.W.R. REPORT NO. 300-38



| REL. NO. | REV. LET. | REVISION | DATE | CKR. | APP'D. |
|---|---|---|---|---|---|
| | | | | | |

| TUBE NO. | PERMEABILITY COEFFICIENT K SQ. IN. | $\frac{\Delta P^2}{L}$ $\frac{LB.^2}{IN.^5}$ | STANDARD AIR FLOW G $\frac{LB.}{SQ.IN.-SEC.}$ | STANDARD AIR FLOW G $\frac{CU. FT.}{HR. SQ.IN.}$ | TO MAKE AIRFOIL NO. |
|---|---|---|---|---|---|
| R-530465 | 1.215 x $10^{-10}$ | 20,000 | 0.01703 | 802 | R-530464 |

## NOTES

1) STANDARD AIR TAKEN AT 14.7 P.S.I.A. AND 59°F.
2) ONLY THE CENTER SECTION, 3.90 INCHES OF THE ENTIRE TUBE IS TO BE FINALLY USED.
3) AIR EFFUSION TOLERANCE:
   A) LOCAL VARIATION WITHIN EACH INDIVIDUAL TUBE ± 4% IS DESIRED ±8% IS ACCEPTABLE
   B) OVERALL EFFUSION MAY VARY $^{+0}_{-30}$ % FROM NOMINAL
4) AIR EFFUSION MEASUREMENT AT CIRCUMFERENTIAL SECTIONS
   A) TUBE MIDSPAN
   B) 0.95 INCHES TO EITHER SIDE OF MIDSPAN
5) PORES OF TUBES MUST BE CLEAR OF FOREIGN BODIES

| | | | | | |
|---|---|---|---|---|---|
| L. Cohen | | | R.W.S. | L.E.F. | |
| 2-14-57 | | | 2/11/57 | 2/14/57 | |
| DRFTM. | CKR. | CHIEF DRFTM. | DESIGN ENGR. | MODEL ENGR. | SPECIFICATIONS |
| | | | | | HAYNES ALLOY No 25 |
| MET. | PROCESS ENGR. | ENGR. | ENGR. | ENGR. | MAT'L SPECS. |

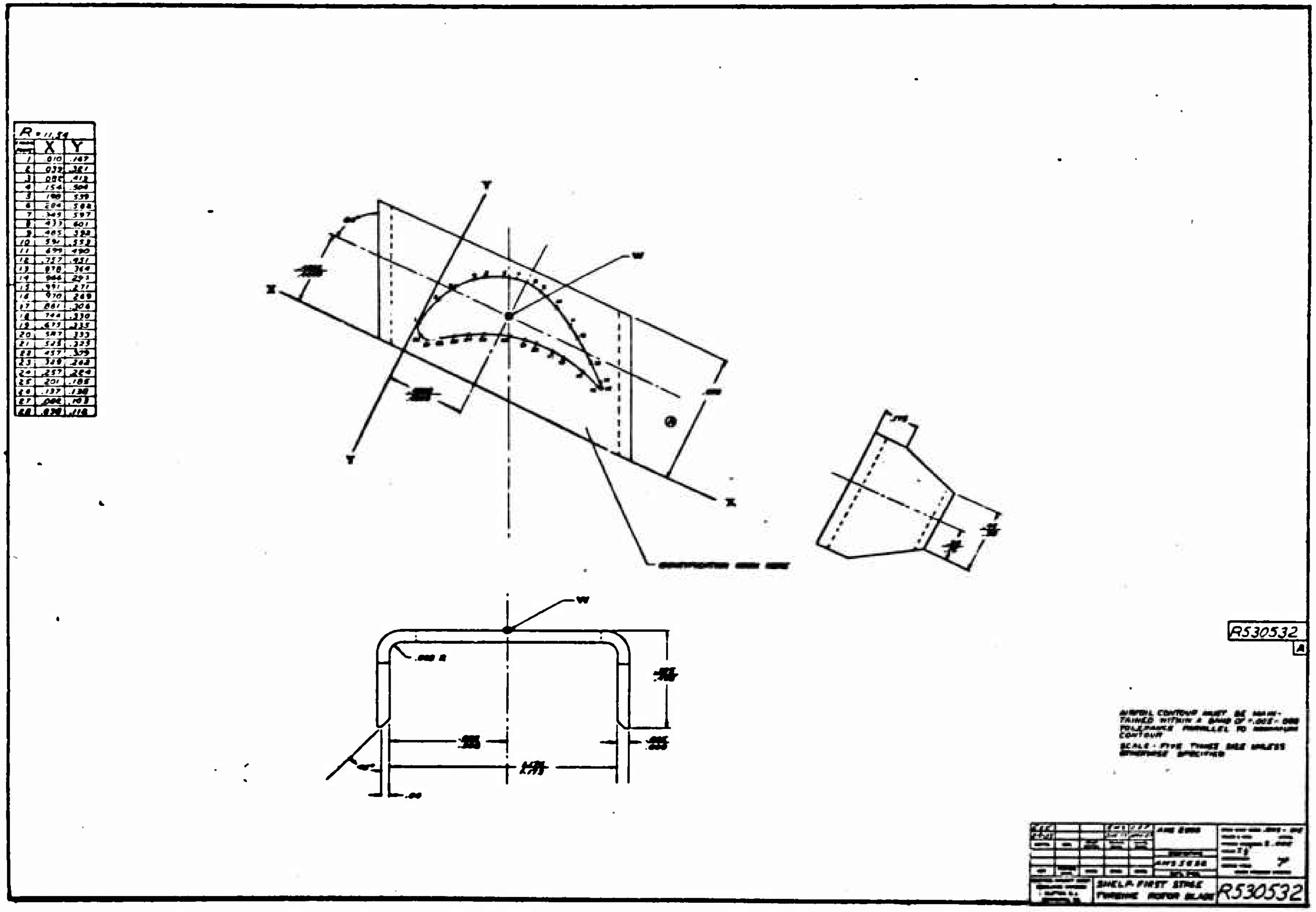
BREAK SHARP EDGES
FILLETS & RADII
FINISHED DIMENSIONS
ANGLES
CONCENTRICITY F.I.R.
SURFACE FINISH √
APPROX.
UNLESS OTHERWISE SPECIFIED

CURTISS-WRIGHT CORP.
RESEARCH DIVISION
CLIFTON, N. J.
QUEHANNA, PA.

WIREWOUND POROUS TUBE FOR AIR COOLED TURBINE ROTOR BLADE AIRFOILS

R-530465

Figure 9



R = 11.54

| | X | Y |
|---|---|---|
| 1 | .010 | .167 |
| 2 | .032 | .361 |
| 3 | .085 | .413 |
| 4 | .154 | .504 |
| 5 | .190 | .559 |
| 6 | .204 | .588 |
| 7 | .349 | .597 |
| 8 | .433 | .601 |
| 9 | .485 | .582 |
| 10 | .591 | .553 |
| 11 | .699 | .490 |
| 12 | .757 | .451 |
| 13 | .878 | .364 |
| 14 | .946 | .293 |
| 15 | .991 | .271 |
| 16 | .970 | .249 |
| 17 | .861 | .306 |
| 18 | .744 | .330 |
| 19 | .675 | .335 |
| 20 | .587 | .333 |
| 21 | .528 | .323 |
| 22 | .457 | .379 |
| 23 | .328 | .262 |
| 24 | .257 | .224 |
| 25 | .201 | .185 |
| 26 | .137 | .138 |
| 27 | .082 | .103 |
| 28 | .038 | .118 |

Figure 10

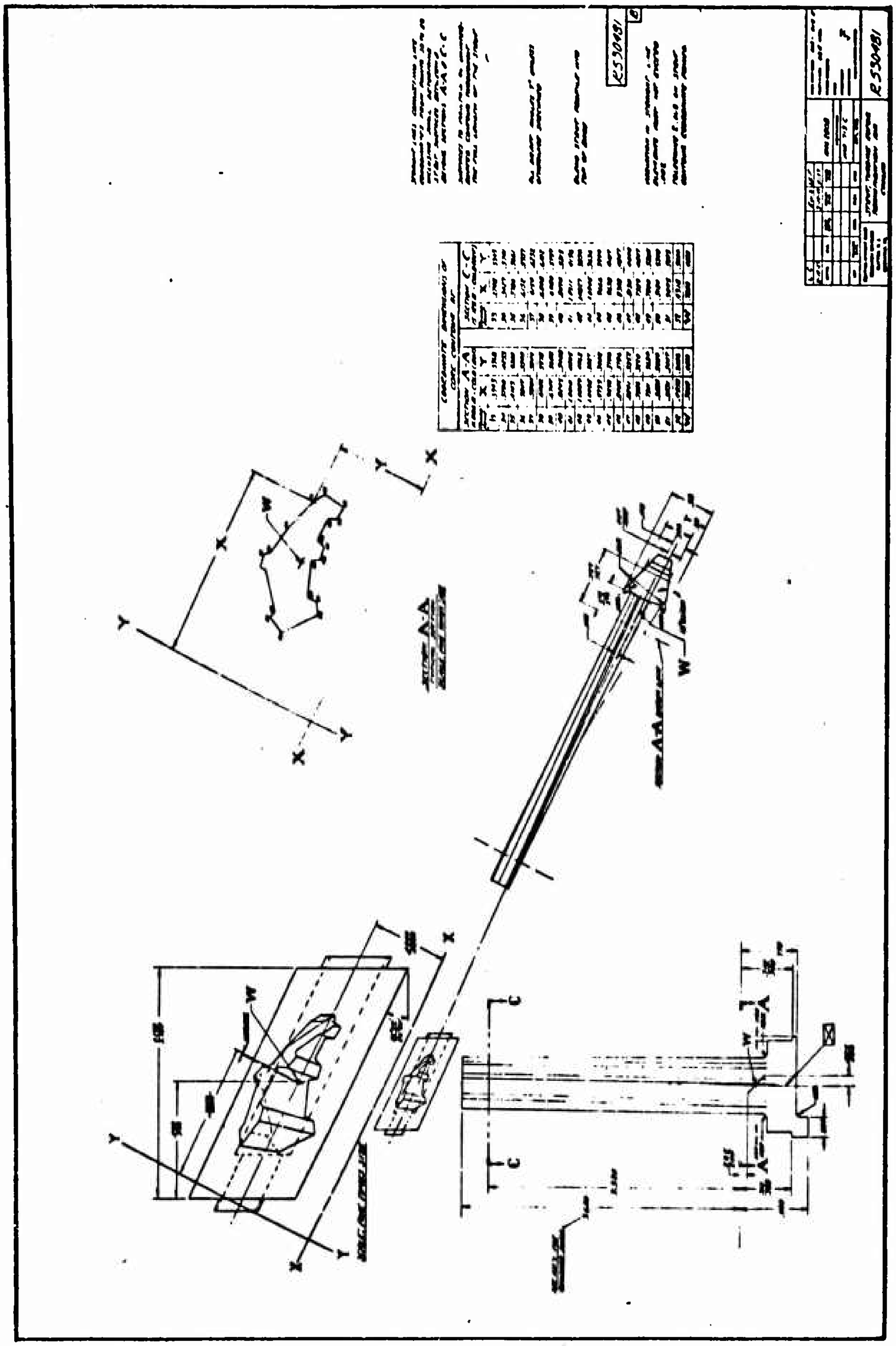


Figure 11

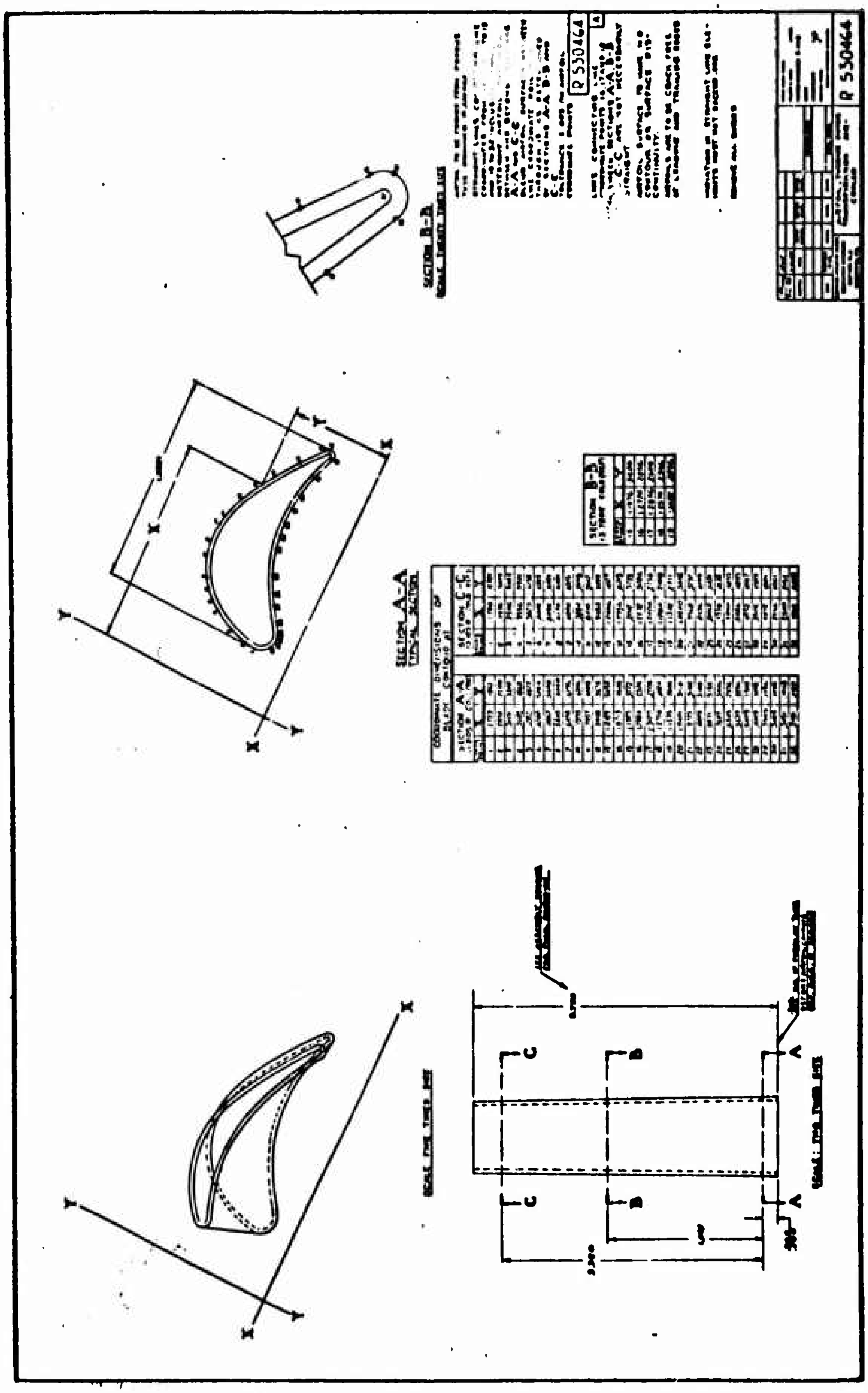


Figure 12

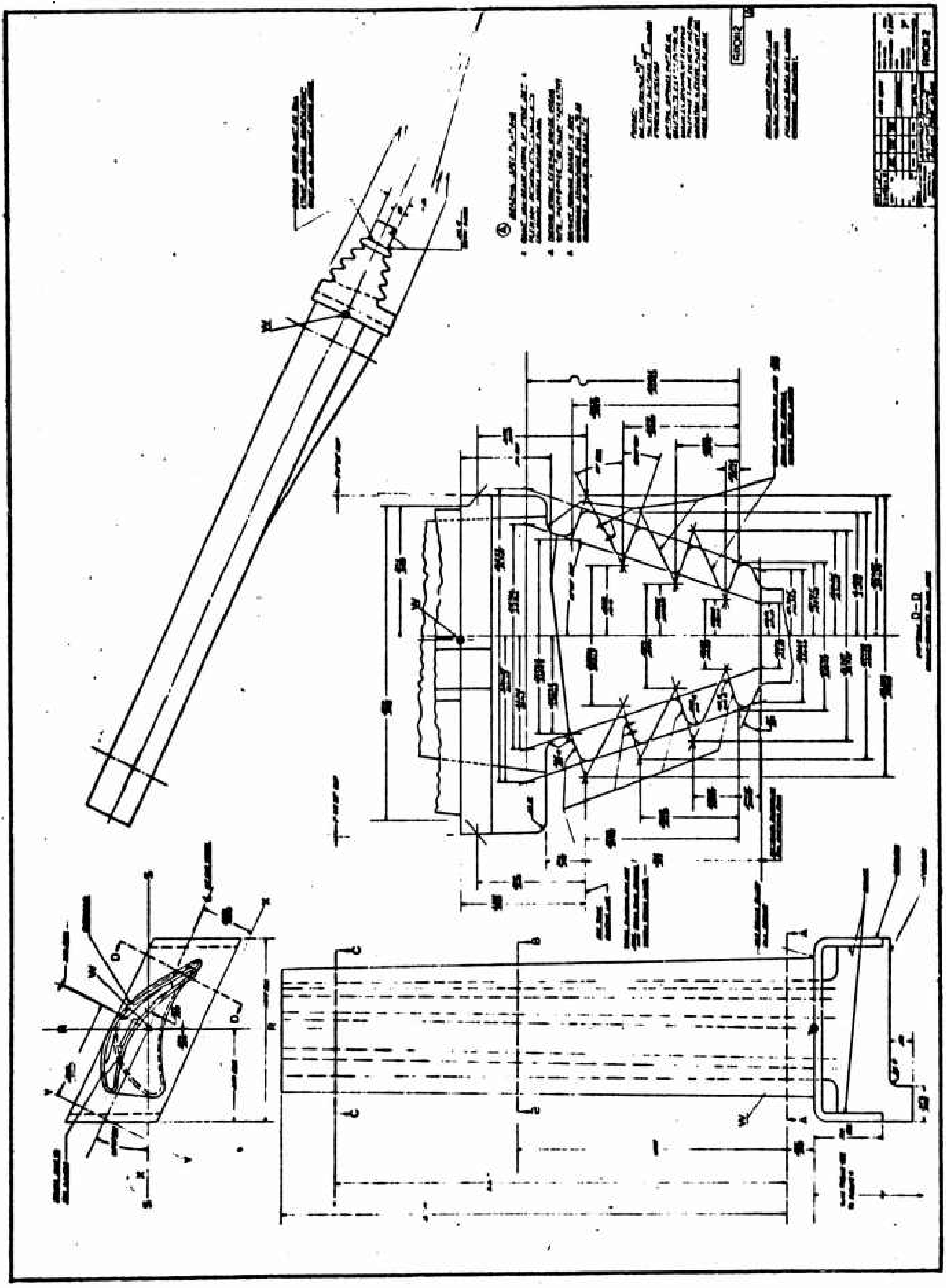


Figure 13

## B. Engine Studies

### 1. Combustion Chamber

In the studies made of the standard J65 engine to determine what areas would have to be modified to operate at a turbine inlet temperature of 1800°F, indicated that the materials of the combustion chamber liners and the flames tube must be modified. Investigation of other materials that would be suitable led to the conclusion that Haynes-Stellite 25 or N-155 alloys were quite satisfactory. As later production models of the J65 engine incorporated combustion chamber made of Haynes-Stellite 25, it was decided to utilize production facilities and use a liner made of Haynes-Stellite 25.

To avoid possible burn-out of the flame tubes the use of newly developed flame tubes was included for the transpiration-cooled blade tests at elevated temperatures. These tubes are fabricated of N-155 alloy which is aluminum dipped and capable of withstanding rather high temperatures.

### 2. Turbine Blades

Original stress data on the J65 first and second turbine stator indicated that they would not be strong enough with the standard materials used. In order for the first stage stator blades to withstand the stress loads at 1800°F turbine inlet temperature, it was found necessary to use Haynes-Stellite 31 material which has sufficient strength at the temperature to withstand the stress loads. As an added protection, these blades were aluminum dipped to improve thermal shock properties. as indicated in recent development tests.

The design of a convection cooled first-stage turbine stator blade was undertaken to provide a stand-by for running at 1800°F if the previously modified first-stage turbine stator blade gave any indication of being too close to critical loading during testing. For this cooled stator blade design, the strut-supported airfoil principle was utilized. In the standard first-stage stator blade design the mounting was a simple cantilever, pinned and bolted at the outside diameter of the turbine annulus. For the convection cooled blade design, the theory of a simply supported beam is utilized, thus, greatly reducing the stresses. Details of the blade consist of only two parts to make up the blade assembly. These parts consisted of an airfoil formed from thin-walled seamless tubing, and a cast strut incorporating the passages for the cooling air. Assembly of these two parts is accomplished by high temperature furnace brazing.

Design stresses for this blade at sea level, zero speed, and standard conditions did not exceed 20,000 psi which is conservative for operations at 1800°F turbine inlet temperatures.

There was no change in the airfoil contour as the original blade design was a constant section airfoil with no twist. This type of blade is very readily adaptable to convection or transpiration cooled blade designs and does not present any design or fabrication problems as aerodynamic characteristics are not changed and all tooling is conventional.

In the case of the second stage turbine stator blades, analysis indicated that a simple modification to the existing blades would be satisfactory for 1800°F turbine inlet operation. This modification consisted of converting these blades from single simply supported cantilever beams to double simply supported cantilever beams by welding the shrouded ends of two blades together at the inner diameter of the turbine annulus. This forms a box-type structure which greatly increases the section modulus, thereby, drastically reducing the stress on the individual blades. With this design it was not necessary to change the blade material.

Studies conducted on the second-stage turbine rotor blades disclosed that the material these blades were made of could not withstand the elevated operating temperatures, and that the fir tree attachment was critical. Investigations also revealed that Inco 700 and 713C materials are satisfactory, with Inco 713C allowing a greater margin of safety. However, Inco 700 second-stage turbine blades with redesigned fir tree attachments were in production for later model J65 engines. As studies of the redesigned fir tree were found to satisfy the operating conditions of transpirational cooled blade program, it was decided to use the Inco 700 blades which could be readily procured.

## 3. Rotor Assembly

Determination of turbine rotor disc temperatures for operation at the elevated temperatures required for transpirational cooled blade testing disclosed that the allowable temperatures would be exceeded. Several solutions existed to overcome this situation such as a redesigned disc, material change, increased cooling, etc. It was felt that the simplest solution would be to use the increased cooling approach as it appeared to be the simplest and most economical.

The increase in cooling air flows to the discs was accomplised by plugging the aft end of the turbine rotor shaft to permit all the thirteenth stage compressor bleed air to go between the first and second stage turbine rotor disc faces. Normally this cooling air is metered between the disc faces, and the rear face of the second stage turbine rotor disc. Cooling of the second stage disc rear face was accomplished by modifying the exhaust cone to duct cooling air to the outer periphery of the disc, and discharging perpendicularly for more effective cooling. Figure 14 is a schematic of the cooling-air ducting in the J65 test vehicle.

As discussed previously the fir tree attachment of the second stage turbine rotor blades became a critical area at the elevated operating temperatures. To remedy this a turbine rotor disc assembly from a later model J65 engine was utilized which incorporated a redesigned fir tree that correlated with Inco 700 turbine blades.

A design change in the turbine rotor disc assembly was made to instrument the transpiration cooled first stage turbine rotor blades for temperature measurement. This involved putting holes through the two discs so that the leads could be carried out for connection to slip rings. The stresses in the discs, by putting these holes in, only slightly changed the overall stress. Basically the result is that a local higher stress occurs but is not detrimental. Another part of the design change was to modify the forward face of the first stage rotor disc to adapt the impeller plate required to duct the cooling air up the face of the disc and direct the cooling air radially to the base of the transpirational cooled blades. This again involved drilling holes in the first stage disc which created local stresses, which did not significently change the overall disc stress. The impeller plate itself is self-supporting with the drilled and tapped holes acting as positioners only.

The design of the impeller plate has been based on data obtained by NACA and the Research Division on tests conducted on narrow rotating passages. It has been found that a tangential component is imposed on the cooling air in a narrow rotating passage. To utilize the centrifugal pumping effect to a greater extent incorporation of radial vanes on the impeller plate is considered necessary.

The outer diameter of the first stage rotor disc also had to be modified for adaption of the cooled transpiration turbine rotor blade. As discussed in the blade design, the bottom edge of the blade shelf was made to be parallel to disc periphery to prevent leakage of blade cooling air. In order to obtain a close fit between the blade shelf and the disc it was necessary to reduce the existing disc rim diameter to eliminate the angular clearance slope above the fir tree, required in the broaching operation, so that a relatively flat surface existed for the lips of the blade shelf to fit on. As the contact area between the lip of the blade shelf and the disc rim is so small the curvature of the disc rim can be considered flat (Shelf lip is less than 0.10 inch long each side of fir tree).

## 4. Variable Area Exhaust Nozzle

Analysis of the engine to determine the influence of elevated turbine inlet temperature on the various operating parameters revealed that a fixed area exhaust nozzle to operate at the elevated temperatures would cause the compressor to surge in the lower speed range. In order to avoid surge of the compressor it was found necessary to start the engine with a larger area exhaust nozzle and run the engine speed up to rated speed, or the test point to be run, and then start closing down on the exhaust nozzle area. To accomplish this the use of a variable area nozzle is necessary. The standard variable nozzles available would not cover the range of areas for both the large and small area requirements. A variable area nozzle was redesigned to meet the requirements of transpiration-cooled turbine blade testing.

## 5. Air Filter Study

An anticipated problem in cooling turbine rotor blade airfoils by means of a porous skin is that of clogging the pores in the airfoil. The dust or dirt which could cause this would be picked up by the main air-stream, and bled off from the compressor into the ducting leading to the porous blades. To prevent this, a type of low pressure drop air filter would be required. Several types of filters exist, but the most promising appears to be an electrostatic type from a low pressure drop point of view. However, overcoming the problems attendant to the high voltage necessary would probably require extensive research and development work.

Drawings were prepared on maximum space envelopes and other requirement specifications. These drawings were presented to several filter designing and manufacturing sources. Little positive response was received from the majority of vendors; however, some of these concerns who were familiar with this type of work (basically atomic energy filtering problems) were willing to present development programs and fabricate prototype units.

This search of the filter industry soon indicated the practical preferences of a non-electrical type of filter. The electrical power required to operate an electro-static filter was greater than that available from present-day aircraft jet engines. Not a single vendor could offer a unit suitable for this project. Out of several hundred common filter vendors contacted, thirty presented an active interest. Most manufacturers were specializing in large industrial installations and were either not prepared, or interested, in small high performance units for aircraft application. In general, either a mat-type or cyclone separator type of filter action was submitted. Most of the installations submitted were external to the jet engine. It is desirable to have an internal unit since this does not increase the engine frontal area. Of the three which could achieve this, one has been selected. Proposal No. 1, drawing R-15099, is a design for a compact internal installation, shown in Figure 15. The unit is located in the empty volume between the engine shafting and the rear main bearing support assembly. This has in no way altered the external engine geometry. Air is bled off at four holes in the inner ring of the center main bearing support, but between the supporting struts. A design flow of 2.74 #/sec at 100 psig and 700°F is manifolded to a toroidal filter, housing 15, Farr Co. Rotonamic inertial-type filters. The Rotonamic nominal design condition is 20 CFM per tube. This is mainly to minimize the abrasive action of the separated particles on the cyclone stator blades, tubes, etc., and have a low pressure drop across the filter. However, it can perform satisfactorily at much greater flows which increase the intensity of the cyclone action. Hence, the required air flow of 40 CFM per tube.

Filter operation is minutely affected by the relative pressure conditions of the air being filtered.

As the air enters the filter unit, it is given a spinning motion and this swirl converts almost all of the axial velocity into rotation. The developed centrifugal forces on the dust particles throw them outward to a liner which guides the dust to the dust bleed-off lines. Ten per cent of the filter flow is required as scavenge air to carry the dust overboard. A study is planned to find a suitable means of conserving this lost air for some useful purpose in the engine. Filtering efficiency for 5 micron and greater dust particles is more than 97 per cent at maximum flow.

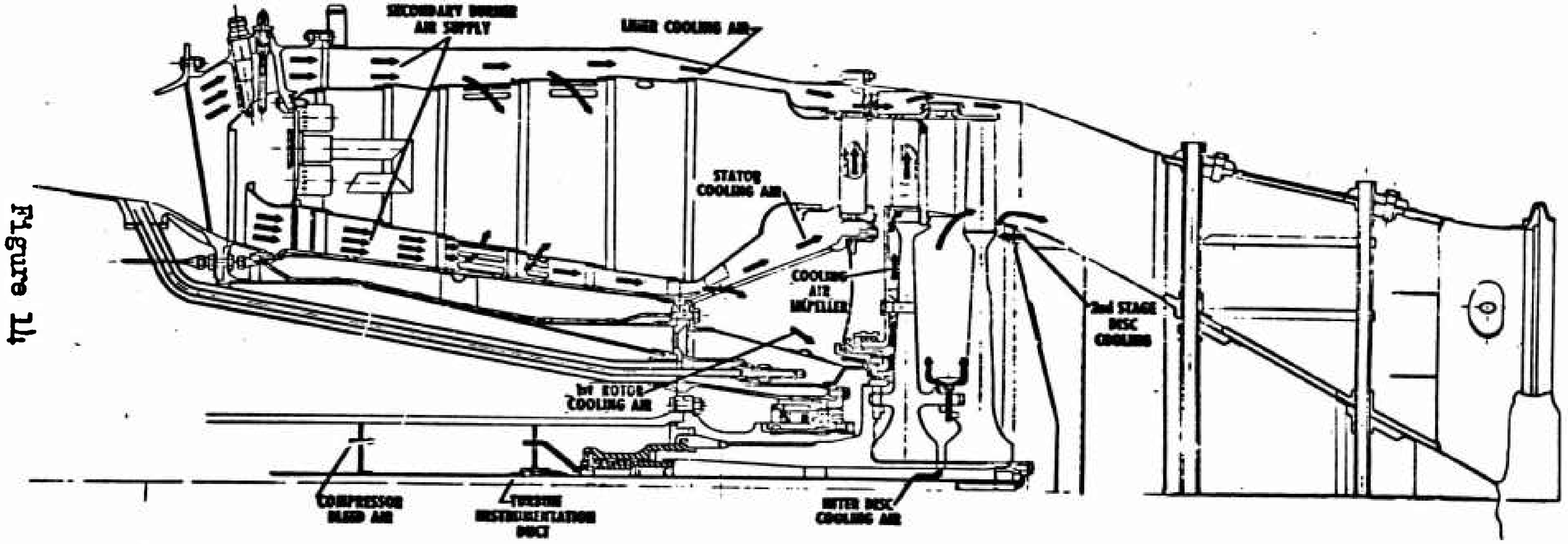
Pressure drop across this filter has been stated as 61 inches water gage (2.2 psig) and future design improvements will decrease this value. The successful operation of the rotors is unaffected by altitude or gravitational forces imposed while the aircraft engine is in maneuvering flight.

The rotors are constructed of 16 gage (.062) type 330 stainless steel and are not adversely affected by 600°F thermal shock. In fact, the design and material construction are expected to withstand temperatures of almost 1000°F, which is more than ample for the present engine. A single Rotonamic tube assembly is approximately 14 ounces. For simplicity, the whole installation is seam-welded, as suggested by the vendor. A brazing and welding option is given to allow a wider range of applicable fabrication techniques. As for durability, these filters have operated over 20,000 hours on air systems for abrasive particles such as sand and grindings without measurable erosion to the spinners, tubes, etc. At engine tear down, any inspection of the unit should include a check on dust build-up in the tubes or bleed-off line. If this happens, the unit can be cleaned by compressed air blown through it.

Operating at design conditions, 2.45 #/sec of clean air is put into the filter exit collecting ring. Four exhaust pipes direct the air to the aft annular plenum chamber. This plenum is connected by slots to an existing void ahead of the turbine rotor heat shield. Since the cooling air is expected to flow through the rotor blade pores at all times, the overall total pressure drop is less than the 4 psia drop across the combustion chamber plus turbine stator seal to the disc impeller and up into the cooling passages of the turbine rotor blades. This impeller also has given some evidence of slightly raising the total pressure of the cooling air by centrifugal compressor action.

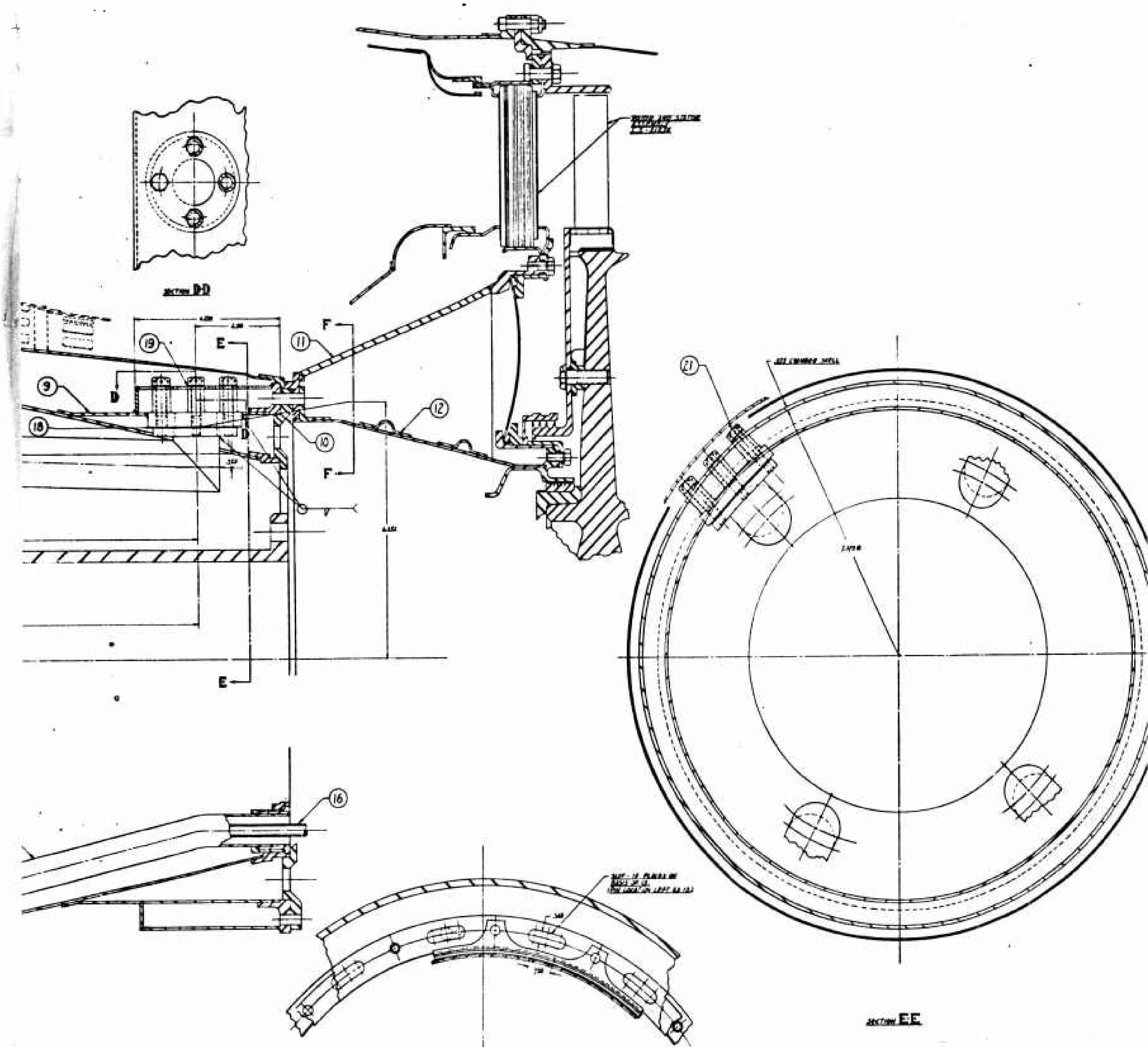
Not knowing whether any serious clogging of the pores would occur in the porous airfoils, a decision was reached to stop at this point until results from actual engine tests indicated a need for such a filtering system.

CURTISS WRIGHT RESEARCH DIVISION



J65 (SLAVE) ENGINE CONFIGURATION WITH AIR COOLED FIRST STATOR AND ROTOR STAGE

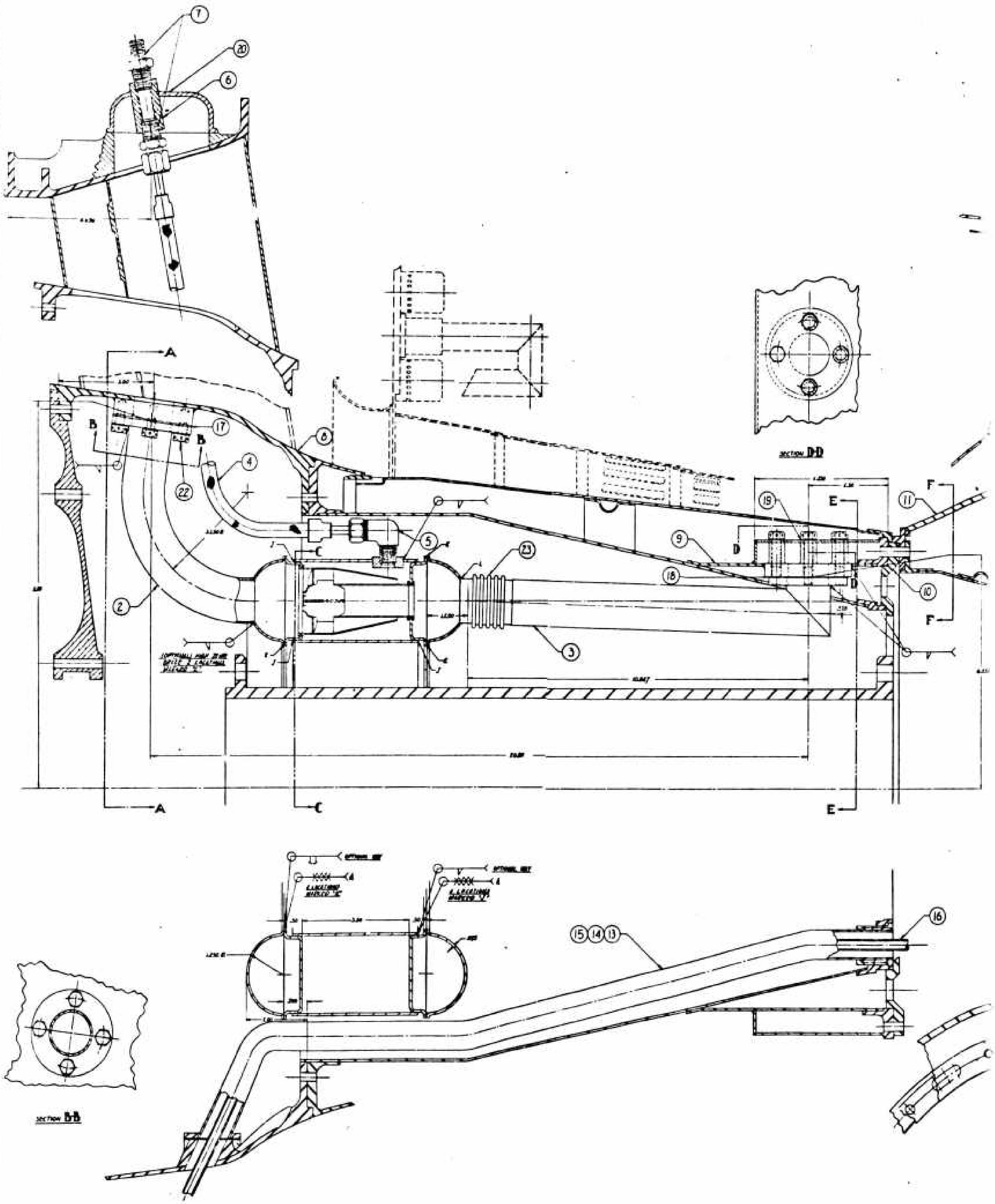
Figure 14



| ITEM NO. | PART NO. | REFERENCE | DESCRIPTION | SHT. NO. | QUAN. | MATERIAL | SPEC. |
|---|---|---|---|---|---|---|---|
| 23 | | | FLEXON BELLOWS-A-10006-OR EQUIV. | | 4 | STAINLESS STEEL | |
| 22 | | | AN-[illegible]-16-.250-28-UNF-3A-HEX HEAD | | 16 | STAINLESS STEEL | |
| 21 | | | AN-[illegible]-.250-28-UNF-3A-HEX HEAD | | 16 | STAINLESS STEEL | |
| 20 | | 220041 (DWG.) | MANIFOLD-CENTER MAIN BEARING | | 2 | | |
| 19 | | | GASKET-.500 O.D. [illegible] THICKNESS | | 16 | [illegible] | |
| 18 | | | GASKET-.250 O.D. [illegible] THICKNESS | | 4 | [illegible] | |
| 17 | | | GASKET-.[illegible]50 O.D. [illegible] THICKNESS | | 4 | [illegible] | |
| 16 | | 220397 (DWG.) | OIL LINE | | 1 | | |
| 15 | | 603033 DWG. | TUBE ASSY-[illegible] MAIN BEARING | | 1 | | |
| 14 | | 603031 (DWG.) | TUBE ASSY-CENTER MAIN BEARING | | 2 | | |
| 13 | | 603036 DWG. | TUBE ASSY-REAR MAIN BEARING | | 1 | | |
| 12 | | 603075 DWG. | DUCT ASSY-REAR MAIN BEARING | | 1 | | |
| 11 | | 603325 DWG. | SUPPORT ASSY [illegible] | | 1 | | |
| 10 | | 603033 DWG. | SUPPORT ASSY-REAR MAIN BEARING | | 1 | | |
| 9 | | 603029 DWG. | CAP ASSY-REAR MAIN BEARING | | 1 | | |
| 8 | | 506722 (DWG.) | HOUSING COMPLETE ASSY | | 1 | | |
| 7 | | | PARKER-MALE CONNECTOR [illegible] | | 4 | STAINLESS STEEL | |
| 6 | | | COUPLING | | 2 | STAINLESS STEEL | |
| 5 | | | PARKER-MALE ELBOW-[illegible] | | 2 | STAINLESS STEEL | |
| 4 | | | STAINLESS TUBE [illegible] | | 2 | STAINLESS STEEL | |
| 3 | | | EXIT TUBE AND FLANGE [illegible] | | 4 | STAINLESS STEEL | |
| 2 | | | INLET TUBE AND FLANGE [illegible] | | 4 | STAINLESS STEEL | |
| 1 | | | FILTER ROTAIRE [illegible] FARR CO. | | 13 | STAINLESS STEEL | |

PARTS LIST

Figure 15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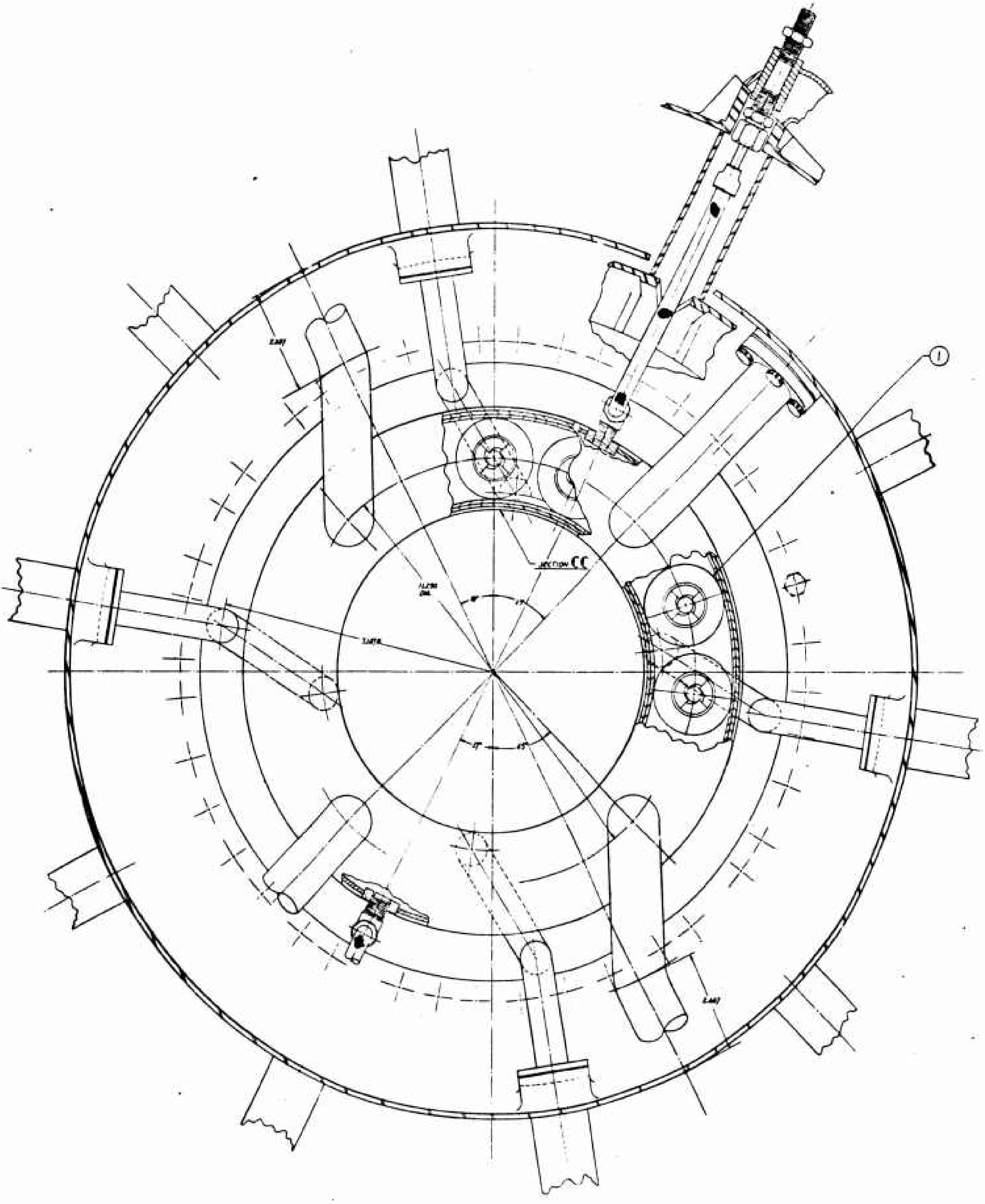
⑯
SECTION B-B



SECTION A-A

C.W.R. REPORT NO. 300-38

## II. FABRICATION

Reported by: N. Lauziere
N. Lauziere

### A. Transpiration Cooled Turbine Rotor Blade

1. Porous Airfoil
2. Cast Struts
3. Blade Shelf
4. Blade Assembly

### B. Engine Adapting Parts

1. Combustion Chamber
2. Turbine Blades
3. Cooling Air Ducting
4. Instrumentation
5. Variable Area Nozzle
6. Miscellaneous Fabrication

## II. Fabrication

### A. Transpiration Cooled Turbine Rotor Blade

#### 1. Porous Airfoil

Prior to completing the design studies of heat transfer and structure, procurement of various porous materials was initiated. Upon receipt of these materials tests were conducted to determine permeability variations, strength, and formability. Results of these tests indicated that permeabilities varied greatly and in most cases were above ± 30% of the specified permeability with the exception of Poroloy, which is a wire wound tube. The permeabilities checked on this material was usually within ± 20% depending on the angle of wind. Greater deviations were found to exist in the low porosity ranges or when a greater angle of wind was used. The greater angle of wind is when the angle of wire wind is approaching the longitudinal axis of the tube. This condition results in a higher strength of the tube in the spanwise direction, but with resultant poor permeability control.

Another factor that presented a problem in the fabrication of the porous airfoils is that all porous materials were in sheet form with the exception of Poroloy. This meant that a seam would exist, resulting in an area on the blade airfoil of zero porosity. This could be overcome by having this seam aligned with one of the lands on the strut where attachment by brazing or welding takes place; however, this would be very difficult to do and would involve special tooling. In addition, it was not considered a very promising production approach. The joining seam could also be placed at the trailing edge of the airfoil and not greatly effect blade cooling as the air being discharged forward could give sufficient film cooling for limited temperature operation. To fully utilize the benefits of transpiration cooling it was decided to make the airfoils from Poroloy tubes having a fully porous surface on the airfoil with the exception of the areas where the skin is bonded to the supporting struts.

Poroloy tubes of three different permeabilities were procured in order to obtain a closer check of porosity deviations, and also to initiate preliminary forming operations.

Porosity tests on these three blade permeabilities again indicated that permeabilities varied within ± 20% with a wire wind of 60° from the axis of the tube.

The material used in these tubes was N-155 as previous experience had indicated no inherent problems in the forming of blade airfoils from the tube. However, difficulties were encountered not only in the forming of the airfoils but also in the sintering of the tubes after winding. Investigations, conducted to determine the cause of these difficulties disclosed the fact that in the annealing between stages of making the wire tube hydrogen was utilized to eliminate oxidation of the material. The use of this hydrogen created embrittlement of the wires, and in the forming of the airfoil, with small radii at the trailing edge, cracking of the Poroloy occurred. Problems occuring in the sintering process are believed to be partially attributed to this embrittlement, but were believed to be chiefly due lack of proper control during the sintering phase. Correction of these difficulties was achieved by going to an argon atmosphere when annealing between steps of making the tubes, and carefully governing the sintering phase.

Another wire material was utilized in making of tubes with somewhat better properties than the N-155. This material was Haynes- Stellite 25 with a slightly better oxidation resistance at higher temperature and roughly the same strength characteristics. There appeared to be no difficulties encountered in making tubes or forming the airfoils following the same process developed for the N-155 material. Some difficulty was encountered in procuring the wire in reasonable quantities. Three sources were found for obtaining wire drawn down to .003 inches diameter. The problem was in drawing the wire to this diameter with a specified tensile strength, and in minimum quantities of one-pound rolls. The difficulties were apparently from the lack of quality in the wire before it is drawn to these small sizes. The various initial wire sizes were .250, .095, and .063 inches in diameter. Previously .250 wire diameter was used in drawing down to .003 inches diameter, but after a few passes and annealing, cracking occurred in the furnace. In order to obtain the proper wire size, the specifications were revised to allow quarter-pound rolls to be supplied. Even at that, a high rejection rate existed in drawing of the wire to the .003 inch size.

It was found upon receipt of Poroloy tubes that dimensional tolerances were not per blueprint specifications. Checking into this matter disclosed that to obtain the specified permeabilities, pressure rolling of the wire wound sintered tubes took place, and the resulting expansion in the material varied. The drawings called for the inside diameter, outside diameter, and wall thickness of the tube to be held to very close tolerances. Due to the varying expansion resulting from rolling to obtain specified permeabilities, only one close dimension could be held. The manner in which this problem was overcome was to slightly taper the tubes, .016 inches per foot, holding the wall thickness to the specified tolerances. In this manner a section of the tube is gaged to the required diameters, and the required length cut out.

The method of forming the airfoil is by filling the porous tube with beeswax, and placing it between dies having the prescribed airfoil contour. These dies are then forced together under pressure to a predetermined point. The result is basically the same as would be obtained in a stretch forming operation of non-porous metals. There is very little spring back of airfoil contour. In this case the amount of spring back encountered was very small and at this time, it was not considered worthwhile to modify the dies. Figure 16 shows the steps in making the airfoil from the porous tube. Detail (1) is the porous tube into which a cast beeswax cone (2) is inserted. The next step is after the pressing operation, and the fourth and final step is after the formed tube has been placed in a locating fixture and trimmed to the proper length. Figures 17 through 20 show the dies in the forming operation of the airfoils.

One set of airfoils was made from N-155 porous tubes, and another from Haynes-Stellite 25 porous tubes to determine their operational characteristics. These airfoils were made to conform with the redesigned first stage turbine rotor blades.

## 2. Cast Struts

Upon completion of the geometric design of the turbine rotor blade for transpiration cooling the material selected for the strut was Haynes-Stellite 31, a casting alloy. Heat transfer analysis indicated that this material was satisfactory for operation at 1800°F turbine inlet temperatures with a very large margin of safety. However, due to the request that higher temperatures be attained during engine tests, if at all possible during this program, selection of one of the recently developed high temperature casting alloys was also made. The material selected was Inco 713C. On this basis, two sets of struts were procured: one set of Haynes-Stellite 31, and the other of Inco 713C. The reason for ordering the Haynes-Stellite material was that very little was known about the characteristics of Inco 713C alloy in brazing and welding.

These struts were made by the conventional investment casting methods, Figure 21 shows one of the production cast struts. Upon receiving these cast struts deviations were noted from dimensional requirements. These deviations were basically spanwise bowing of the strut section. This bowing is probably due the slenderness of the strut, and when the wax pattern is transferred, and packed with the investment, some distortion occurs. As this bowing was not consistent, the only correction that could be exercised was in taking greater precaution in handling. Even with this precaution some bowing still existed; however, the deviation in most cases was minor and it was felt that the struts were acceptable.

## 3. Blade Shelf

Fabrication of the rotor blade shelf was performed by standard sheet metal practices. This involved stamping out blanks, and then bending the leading and trailing edge flaps over a die. The final operation consisted of punching out the contoured hole which allows the strut to pass through, and provides the orifice passages at the root of the blade airfoil. The material used in these shelves is Haynes-Stellite 25 sheet one-sixteenth of an inch thick.

There were no difficulties encountered in the fabrication of these parts.

## 4. Blade Assembly

The transpiration cooled turbine rotor blade consists of three details made up of a porous airfoil, cast strut, and a sheet metal shelf described in detail above and shown in Figures 22 and 23. Attachment of these details to make a blade assembly is accomplished by brazing. Development of the technique and other studies conducted leading to the method adapted to braze the blade assemblies are described in detail in the brazing development section of this report.

The next operation after brazing of the blade assembly is the machining of the fir tree on the blade butt. In this particular design, grinding of the fir tree on both sides of the blade at one time is not possible due to the shelf configuration. The shelf design is such that the lower edges of the shelf lip are parallel to a tangent line on the disc periphery. This does not allow any clearance angle for the fir tree grinding wheel unless the blade is tipped sufficiently.

To grind the fir tree one side of the blade at a time, the blade assemblies are first located, by compartor charts, to reference surfaces and set into matrix boxes, also located to the same reference surfaces. When the blade assembly and the matrix box are located relative to each other, the void left in the matrix box is filled with "Cerrotru", a low melting metal. Figures 24 and 25 show the brazed blade assembly in the matrix box before and after the fir tree has been machined. From this point on, the blades are set up in fixtures to give the proper machining positions. To grind the fir tree use was made of a "Hoglund" dresser tool which reproduces the proper fir tree on the grinding wheel from a template. The reason for this grinding method is that greater accuracy can be obtained due to the grinding angle necessary to obtain the grinding wheel clearance and the ability to maintain close tolerances without continuous reforming of the grinding wheel. The machine set-up for grinding the fir tree is shown on Figure 26. As only one set of blades is fabricated at one time, the fir tree of each blade is ground to fit the disc in which it will be used. Normally blade fir trees are ground in production quantities and then assorted in various "N" sizes to fit deviations on the broached discs.

After the fir trees are ground to size the blades are assembled into the rotor and tip ground to obtain the proper clearance with respect to the rotor shroud. The leading edge butt faces of the blades roots are also ground in the rotor assembly to obtain a flat contact surface with the impeller plate to keep cooling air leakage at a minimum. Figures 27 and 28 show a completed rotor assembly with and without the cooling air impeller plate.

## B. Engine Adapting Parts

### 1. Combustion Chamber

New combustion chamber liners were procured utilizing Haynes-Stellite 25 material in order to withstand the higher combustion chamber operating temperatures. These were fabricated in the same manner as production parts. The flame tubes to be used in these tests were fabricated of N-155 material, and were aluminum dipped.

### 2. Turbine Blades

To run at the elevated temperatures without resorting to expensive reblading of other stages, the first and second stage stator blades and the second stage rotor blade had to be modified to withstand the stresses at 1800°F.

Analysis of the first stator stage indicated that using the same design blade made of Haynes-Stellite 31 material and aluminum dipped would be required to withstand turbine inlet temperature of 1800°F with a satisfactory margin of safety. One set of these blades was fabricated by standard production methods.

In order to have a standby for engine operation above the normal 1650°F temperature, a set of convection cooled stator blades was procured. The design of these blades is covered in the design section of this report. The strut, as shown in Figure 29, for this blade was cast of Haynes-Stellite 31 material using the conventional investment casting process with no difficulties. The airfoils were formed from tubes made of Inconel X. The forming process used is the same as the transpiration cooled airfoils; that is, by filling the tubes with beeswax and pressing them between dies with the proper airfoil shape. These blades have a constant airfoil section, and straight line elements with no twist, making it much easier to hold the required tolerances. Some spring back was noted, but was very easily corrected due the fact that the blades were straight. Figure 30 shows a formed airfoil for the convection cooled stator blade.

The strut and airfoil were bonded together with "Nicrobraze 120" brazing alloy in furnaces with a very low dew point. Brazing of these blades presented no real problems, except that the fit between the airfoil and the strut was not perfect and the resulting braze area was approximately 70 to 80 per cent. Figure 31 shows the brazed cooled stator blade assembly. On this type of blade, where the stresses are quite low, 20 to 30 per cent brazed area on the lands is considered more than sufficient.

The second stage stator blades were modified by welding the cantilevered ends of the blades in pairs, as discussed in the design analysis, to reinforce the blades making them capable of withstanding the stress loads at the elevated operating temperatures.

Modification of the second stage turbine rotor blades involved the procuring of another rotor assembly that is normally used in advanced models of the J65 engine which incorporates a new fir tree design of greater strength. With this modified second stage rotor disc, it was possible to procure second stage turbine rotor blades made of Inco 700 material which is capable of operating at the elevated temperature operation of 1800°F turbine inlet quite satisfactorily.

## 3. Cooling Air Ducting

To provide cooling air to the first stage transpiration cooled turbine rotor blades, and also cool the adjacent parts in the turbine and exhaust sections, required the fabrication of some new parts and modification of several standard engine parts.

The ducting of the cooling air to the rotor blades involved reworking of the inner stator blade support to permit a sufficient amount of secondary combustion chamber air for the blades. This involved enlarging the metering holes that previously supplied cooling air to the face of the first stage turbine rotor disc. Revision of the heat shield was also necessary in order to give sufficient clearance to the impeller plate which ducts the air on the face of the disc up to the blades. Relocation of the heat shield involved reworking of the stator and bearing support members, along with the fabrication of new adapting rings for retaining the heat shield and ducting the cooling air to the face of the rotor disc. Carrying of the cooling air up the face of the disc was accomplished by utilizing an impeller plate, as previously mentioned. This plate incorporated a labyrinth seal at its inner diameter, as shown on Figure 32, to create an air passage with the normal engine labyrinth seal at the rear main bearing. The impeller plate utilized radial vanes on the face toward the disc to permit radial discharge of the cooling air to the root of the cooled blade. The vanes are shown on Figures 33 and 34. This was done to reduce any tangential components of the air stream created in the narrow passage between the plate and the disc, utilizing the full effect of the pumping action.

To increase the cooling air flow between the first and second stage discs the metering hole at the end of the turbine shaft was plugged allowing all the thirteenth stage bleed air to be discharged between the two discs. To cool the rear face of the second stage rotor disc the inner exhaust duct cone was modified so that cooling air was ducted through the supporting strut rods. From there cooling air is piped to a hollow reinforcing section at the inner diameter of the cone section directly behind the rim of the second stage turbine rotor disc. Holes were then drilled through the heat shield directly into this hollow chamber. The discharge of air from these holes is perpendicular to the rear face of the second stage disc rim.

## 4. Instrumentation

To obtain temperature measurements of the transpiration cooled turbine rotor blades, reworking of the blade strut was performed to incorporate thermocouples in the base of the blades. This involved drilling a hole in the strut from the base, to insert a thermocouple into the strut for a depth of approximately one inch. This locates the thermocouple about one quarter of an inch above the blade shelf. Drilling of the hole was accomplished by the "Zelox" method as conventional drills would not cut the Haynes-Stellite 31 or Inco 713C material.

To route the thermocouple leads, the turbine rotor disc assembly was modified so that tubes could be inserted to span the gap between the two discs, so that the leads could be carried down the rear face of the second stage disc. In this way the leads could be carried through the engine shafting to slip rings mounted on the front main bearing support housing. Figure 35 shows the installation of the instrumented blades in the first stage of the cooled turbine rotor.

In order to route the instrumentation through the engine shafting, a series of tube assemblies was fabricated to support the leads in the center of the shafting so that lead wire damage would not occur on sharp shaft shoulders from centrifugal force effects and also make it possible for easier repair and maintenance of the wiring. This involved complete disassembly of the engine, and reworking of some engine parts for installation of this wire ducting.

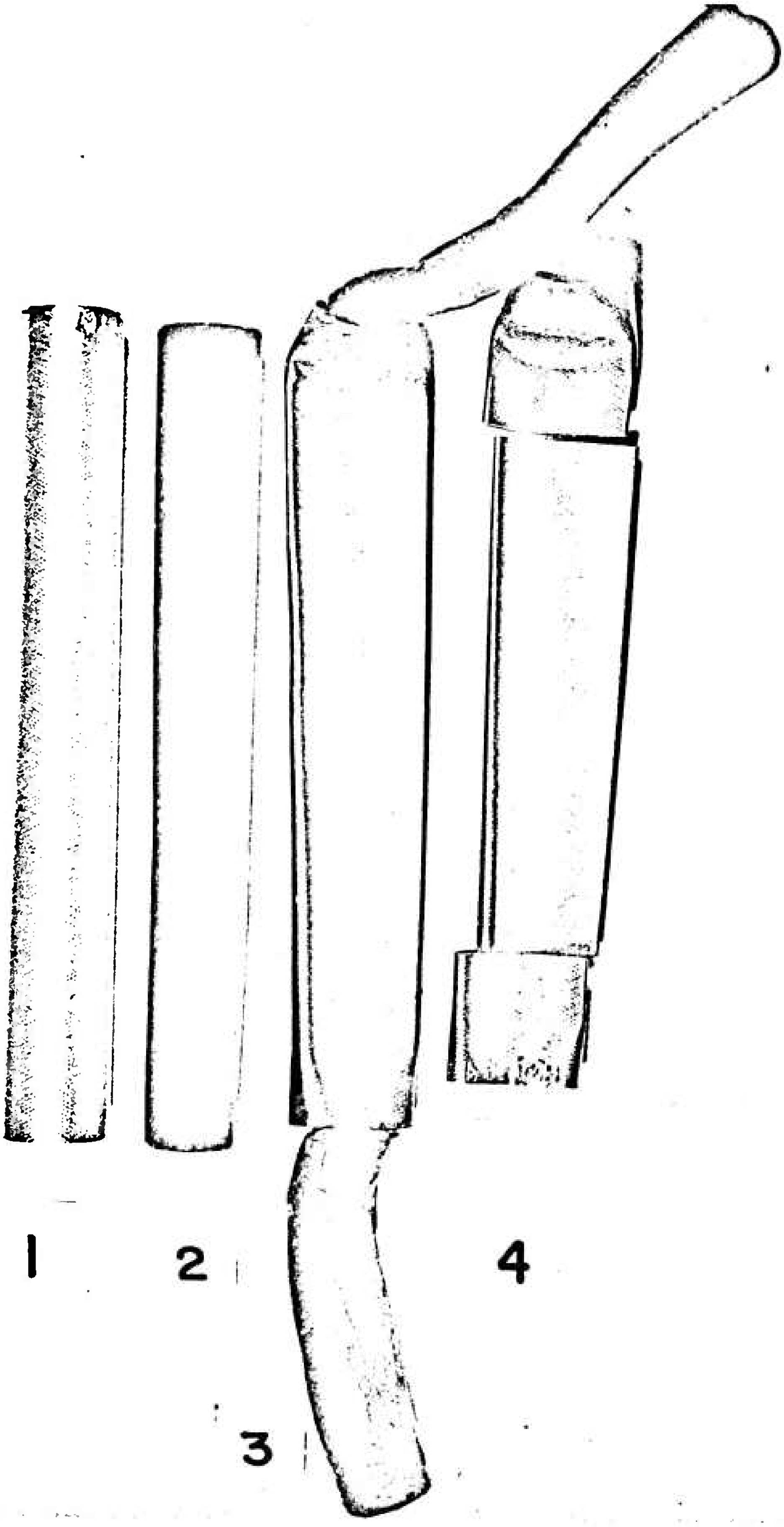
Fabrication and reworked parts to adapt slip rings and other instrumentation were done by methods normally followed for installations of this type.

## 5. Variable Area Nozzle

Operating at elevated temperatures of the J65 test engine required the use of a variable area exhaust nozzle, as discussed in the design analysis section. Modification of a standard variable area nozzle was necessary for operating at 1800°F turbine inlet temperature or any higher temperature. This modification consisted of changing the expanding and contracting ring so that a larger nozzle area could be obtained on the engine in order to avoid compressor surge at the lower speed points. This nozzle also had to contract more than the standard nozzle so that the higher turbine inlet temperature could be attained.

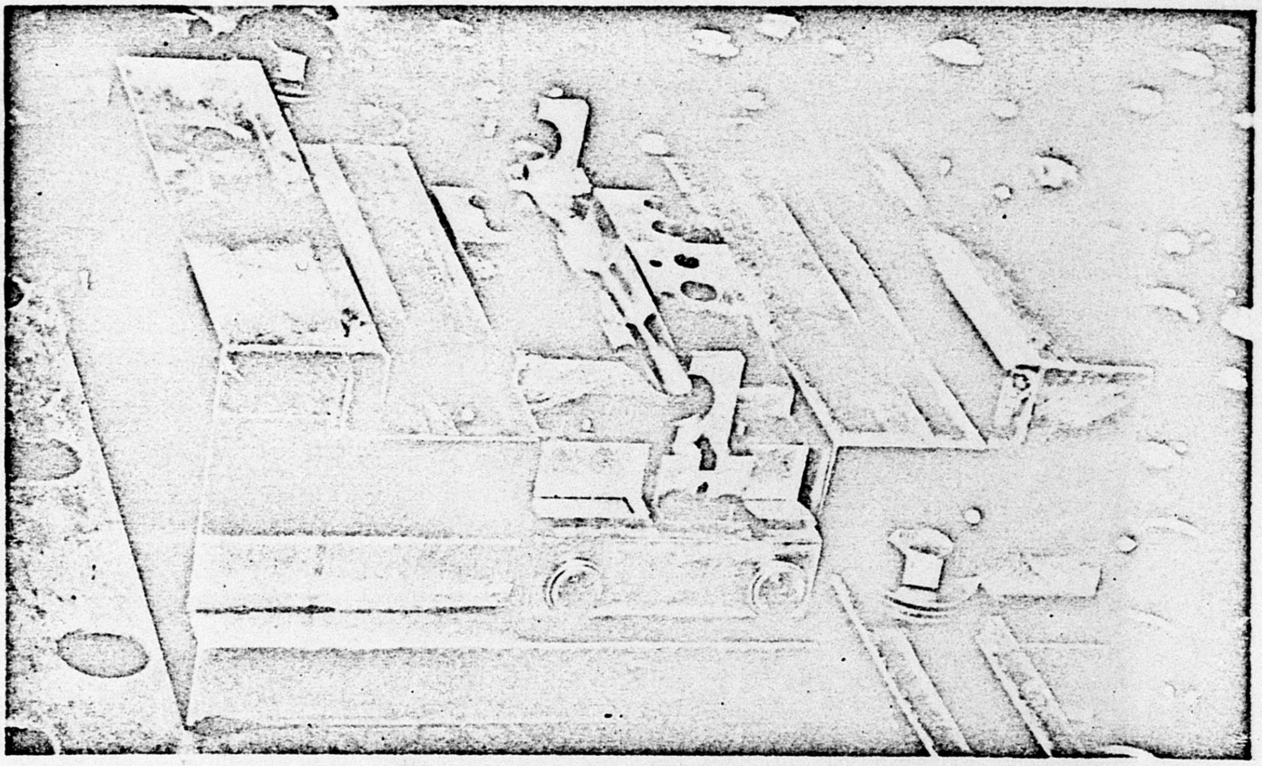
## 6. Miscellaneous Fabrication

Adapting parts for the convection cooled first stage turbine stator blades were fabricated. This basically consisted of adapting rings and shrouds. With this design of cooled stator blades, the shrouds were punched with holes conforming to the airfoil shape, and the brazed assemblies inserted therein to form a cooled stator assembly.

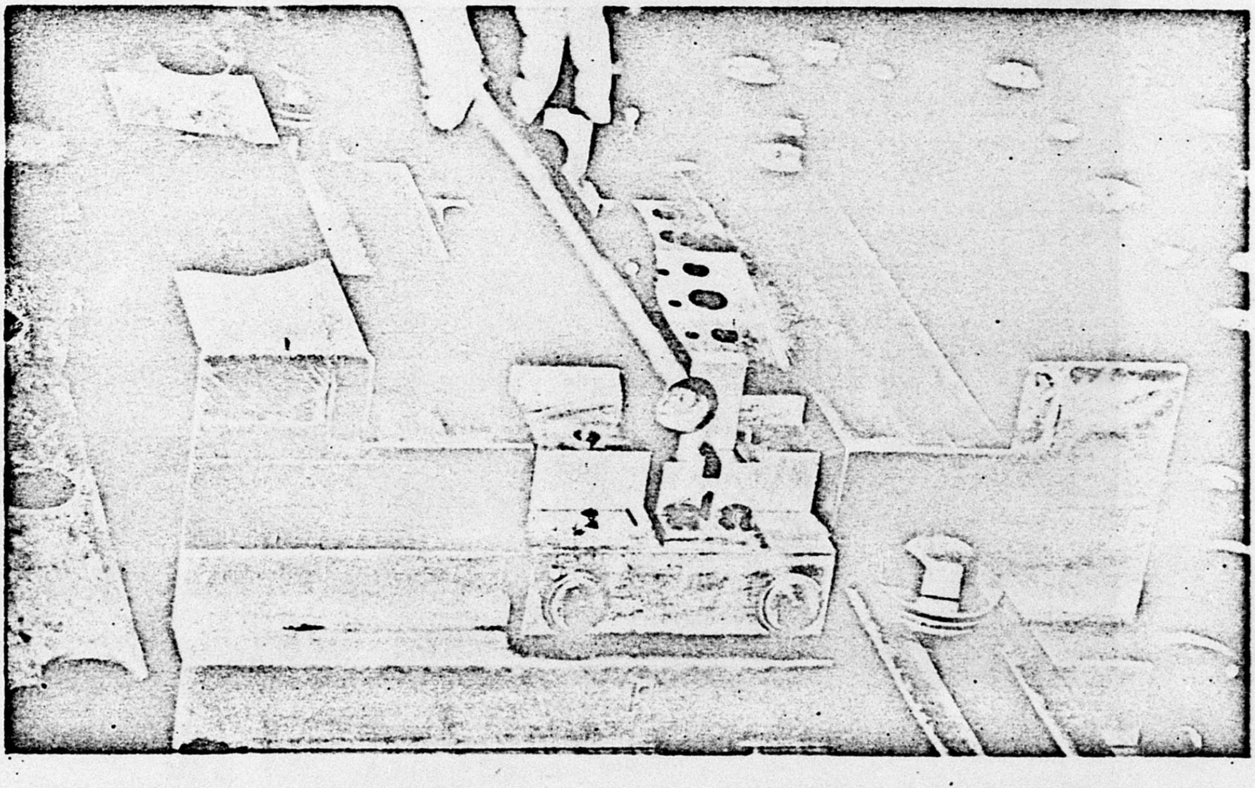


Airfoil Forming

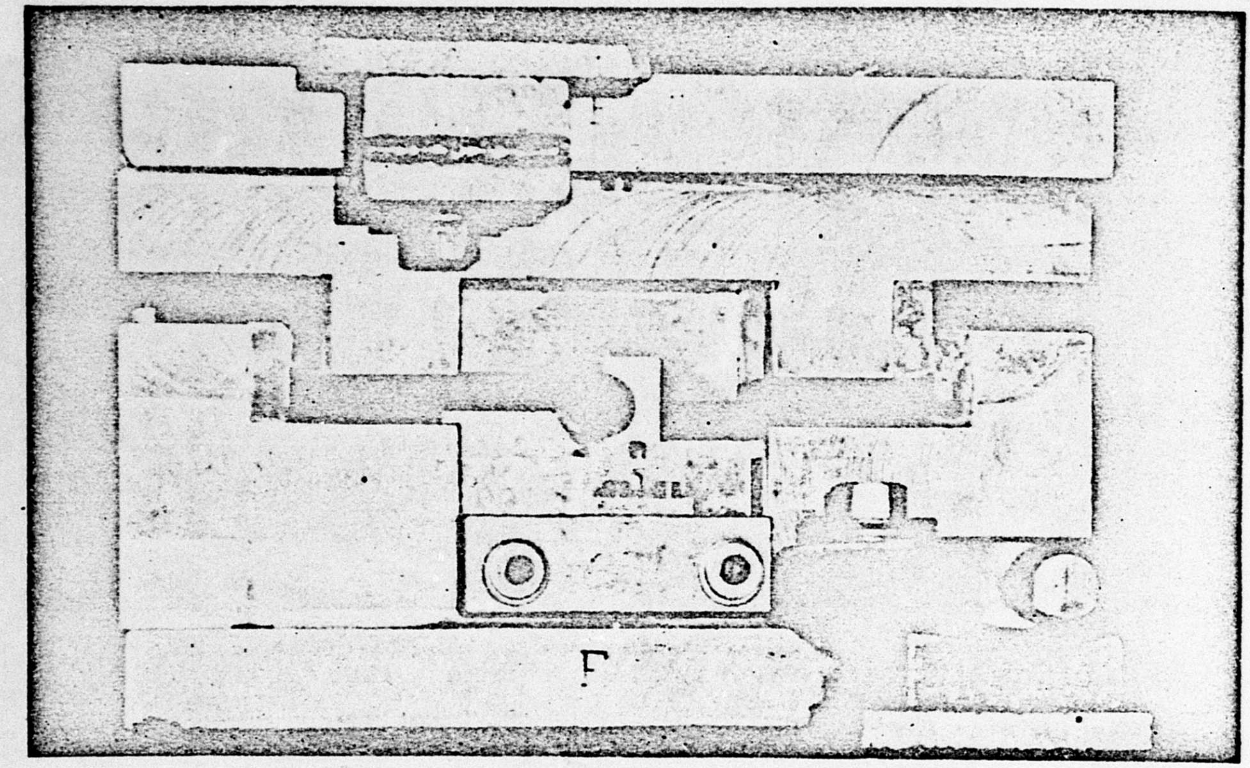
Figure 16



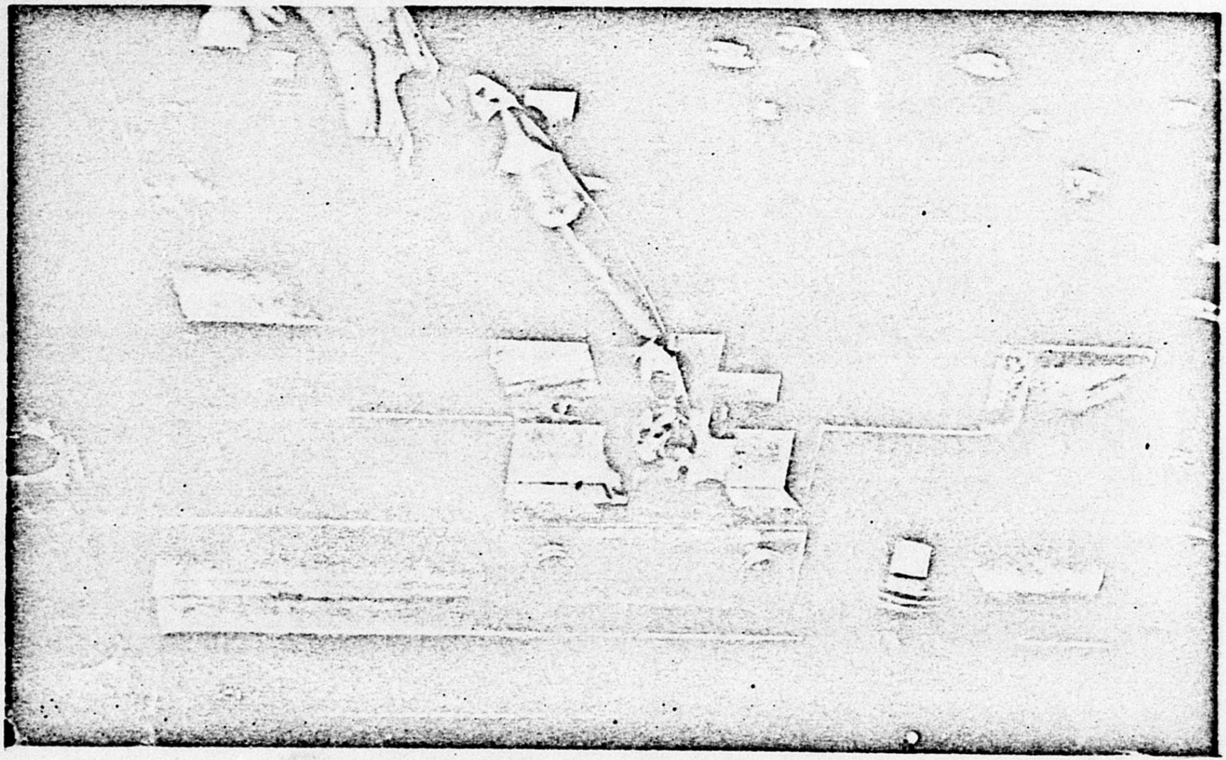
Lower Airfoil Die Figure 17



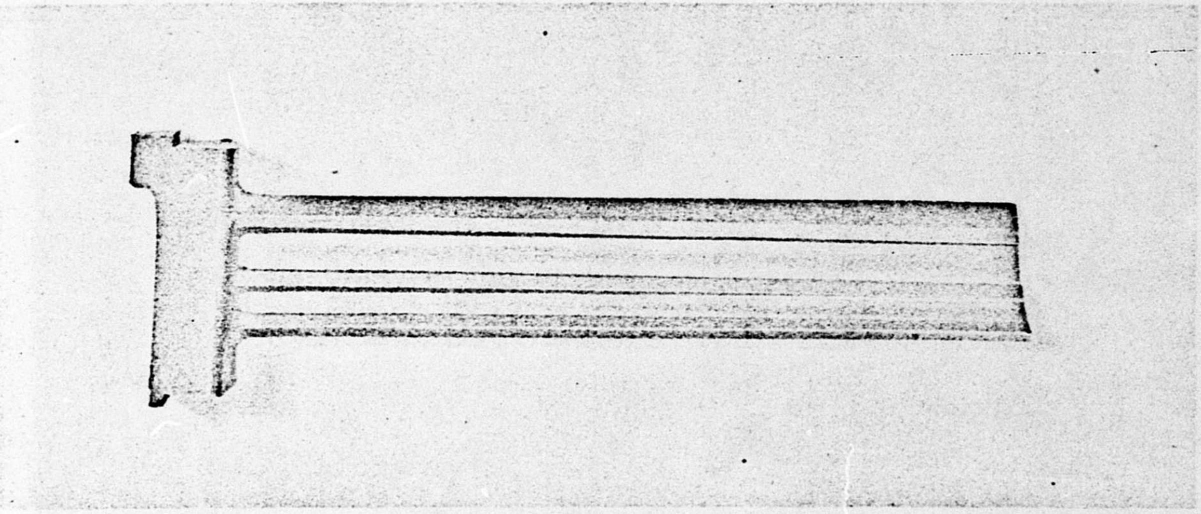
Insertion Of Tube In Lower Die Figure 18



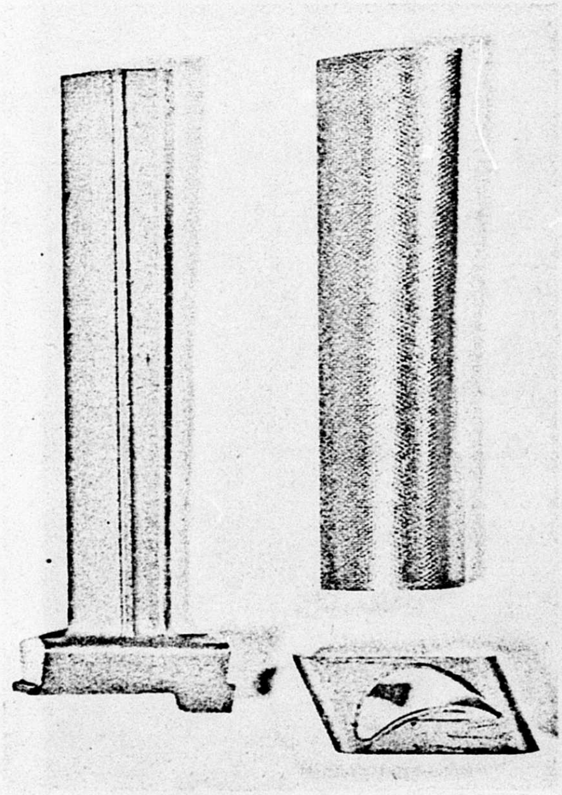
Airfoil Dies Prior To Pressing Figure 19



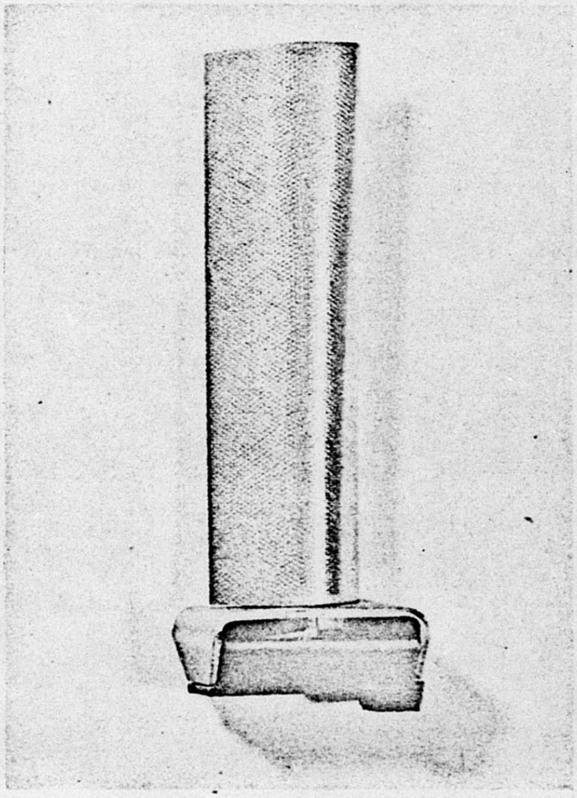
Formed Tube In Die Figure 20



**Cast Strut** **Figure 21**

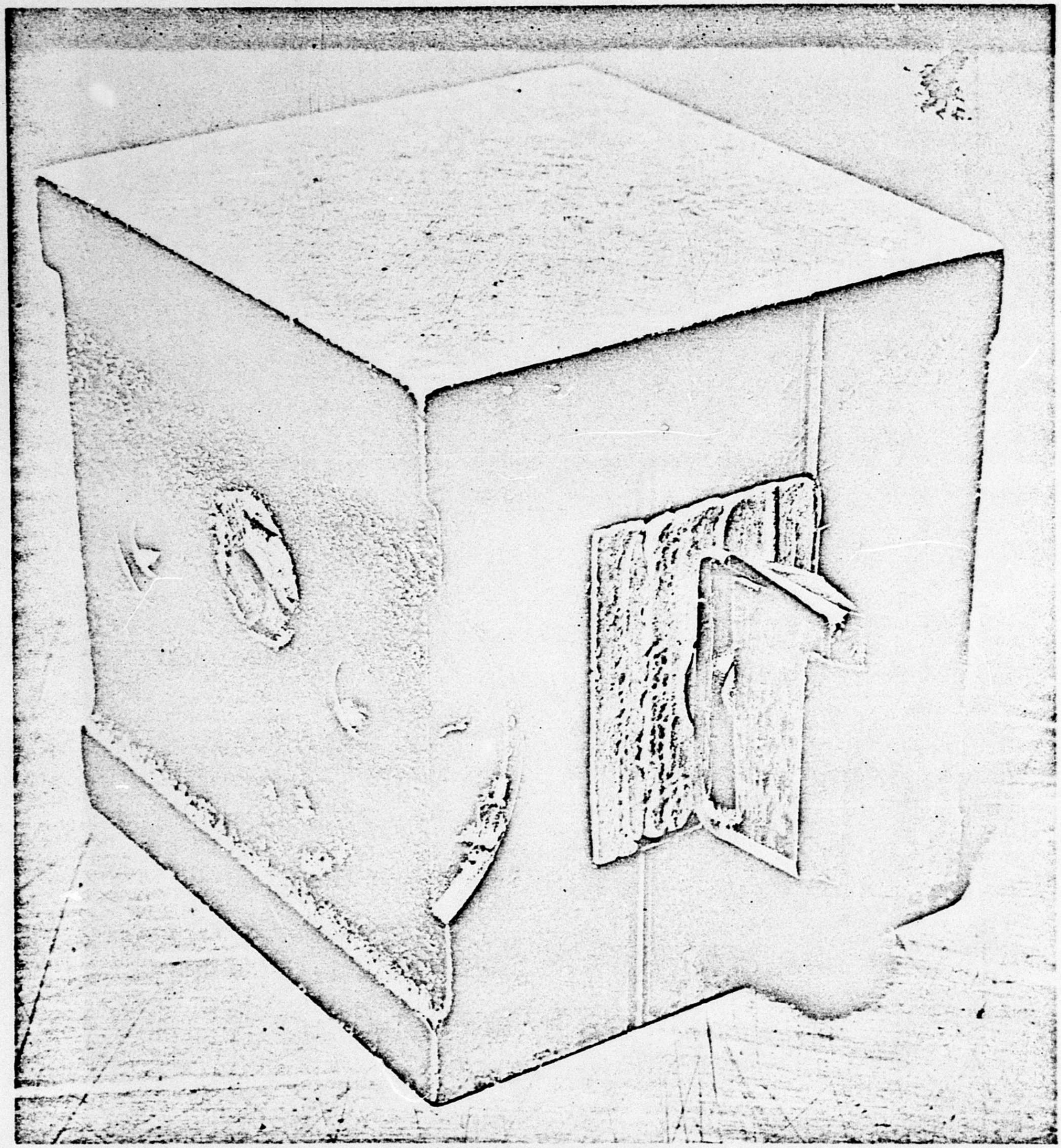


**Blade Details** **Fig. 22**



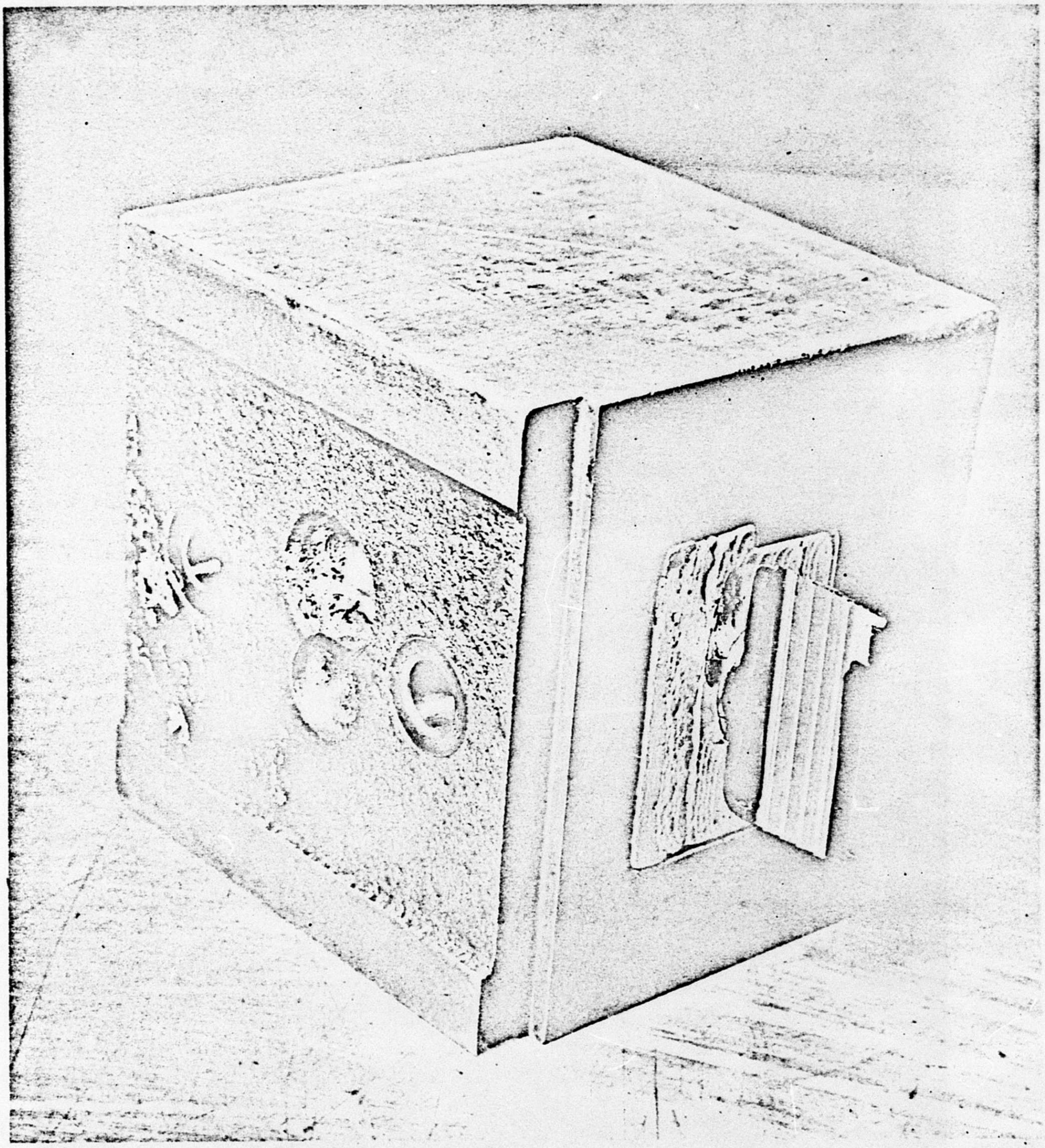
**Blade Assembled** **Fig. 23**

[redacted]



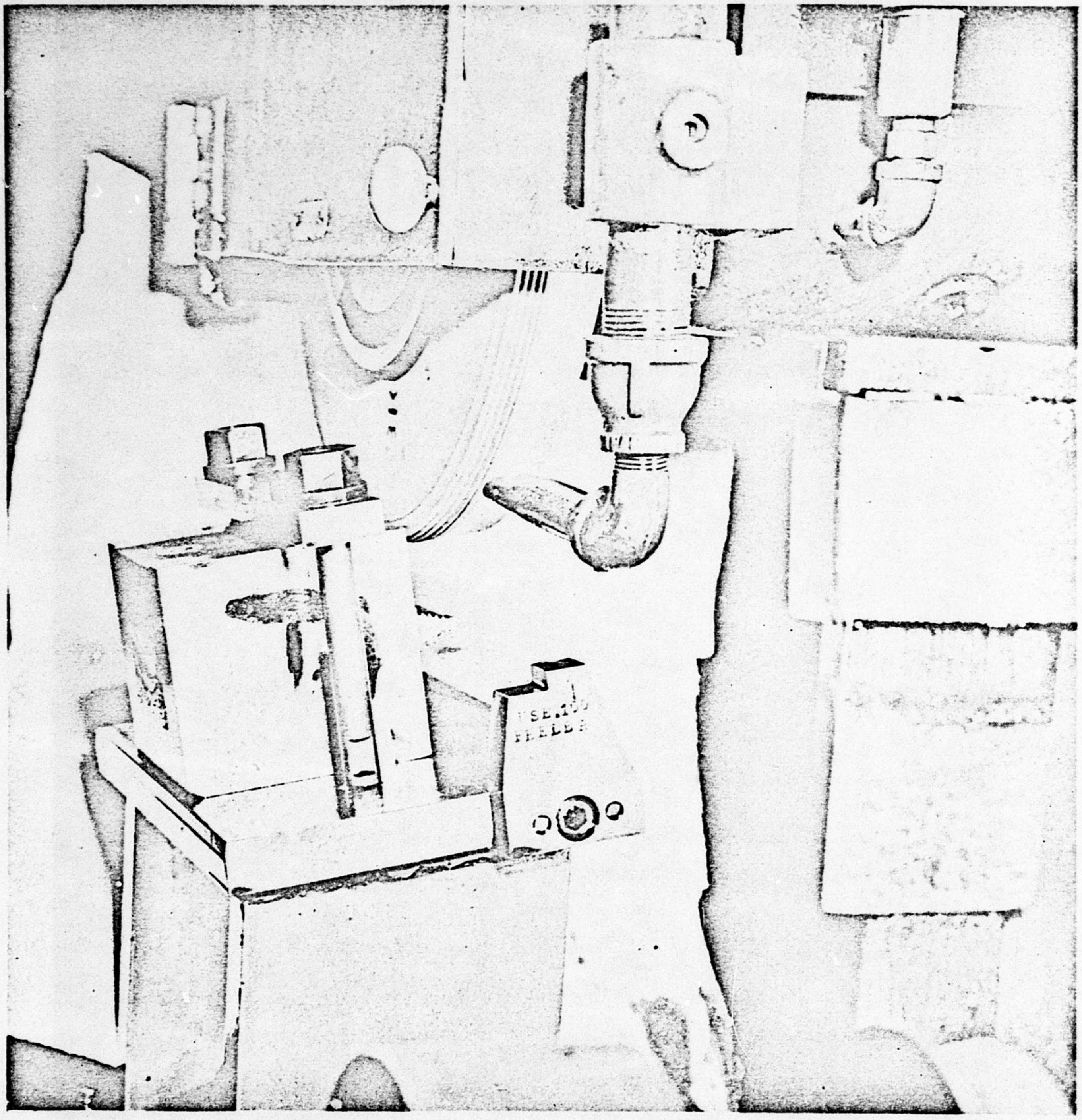
Blade Assembly In Matrix Box

Figure 24



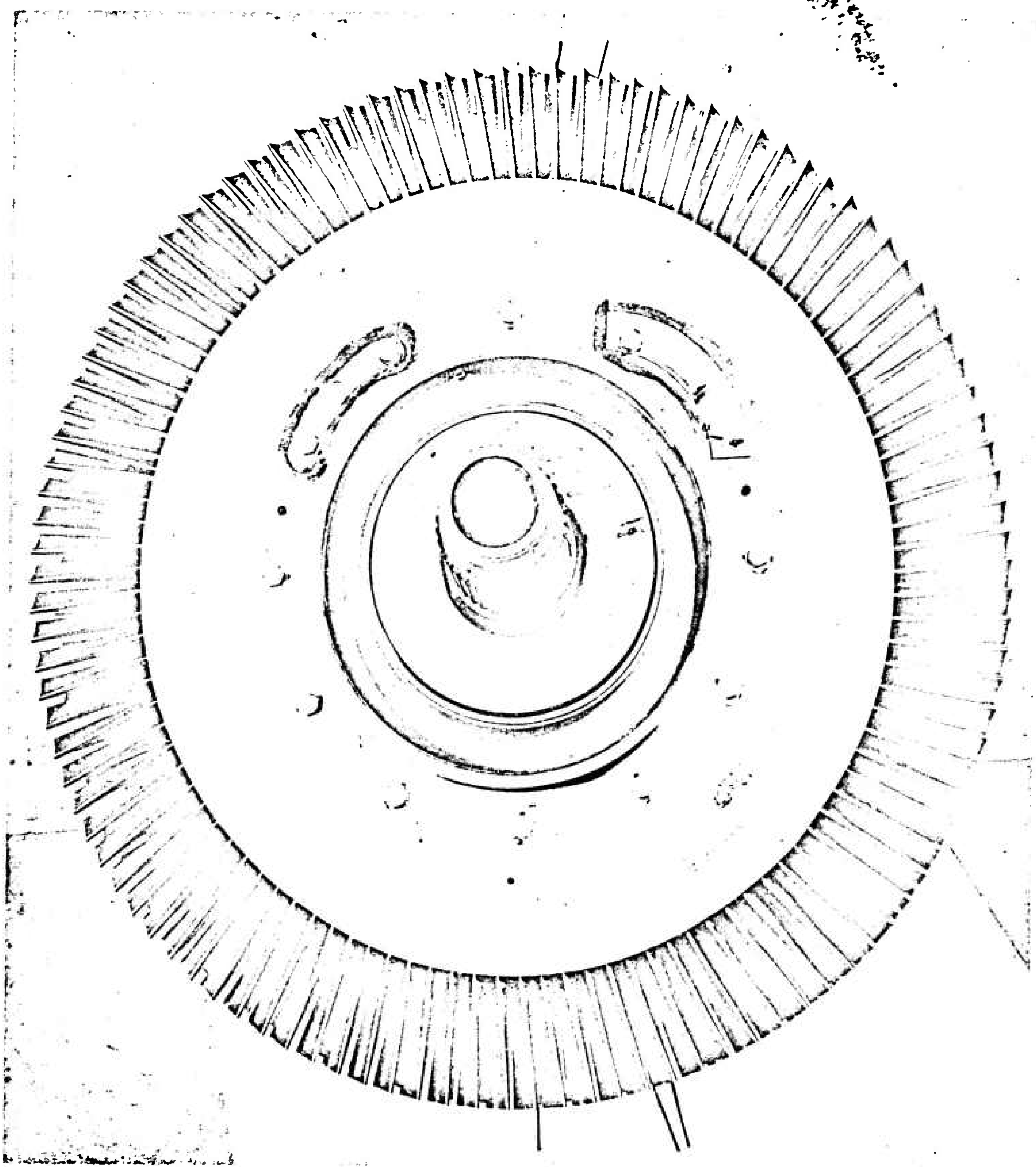
Fir Tree Machined On Blade In Matrix Box

Figure 25



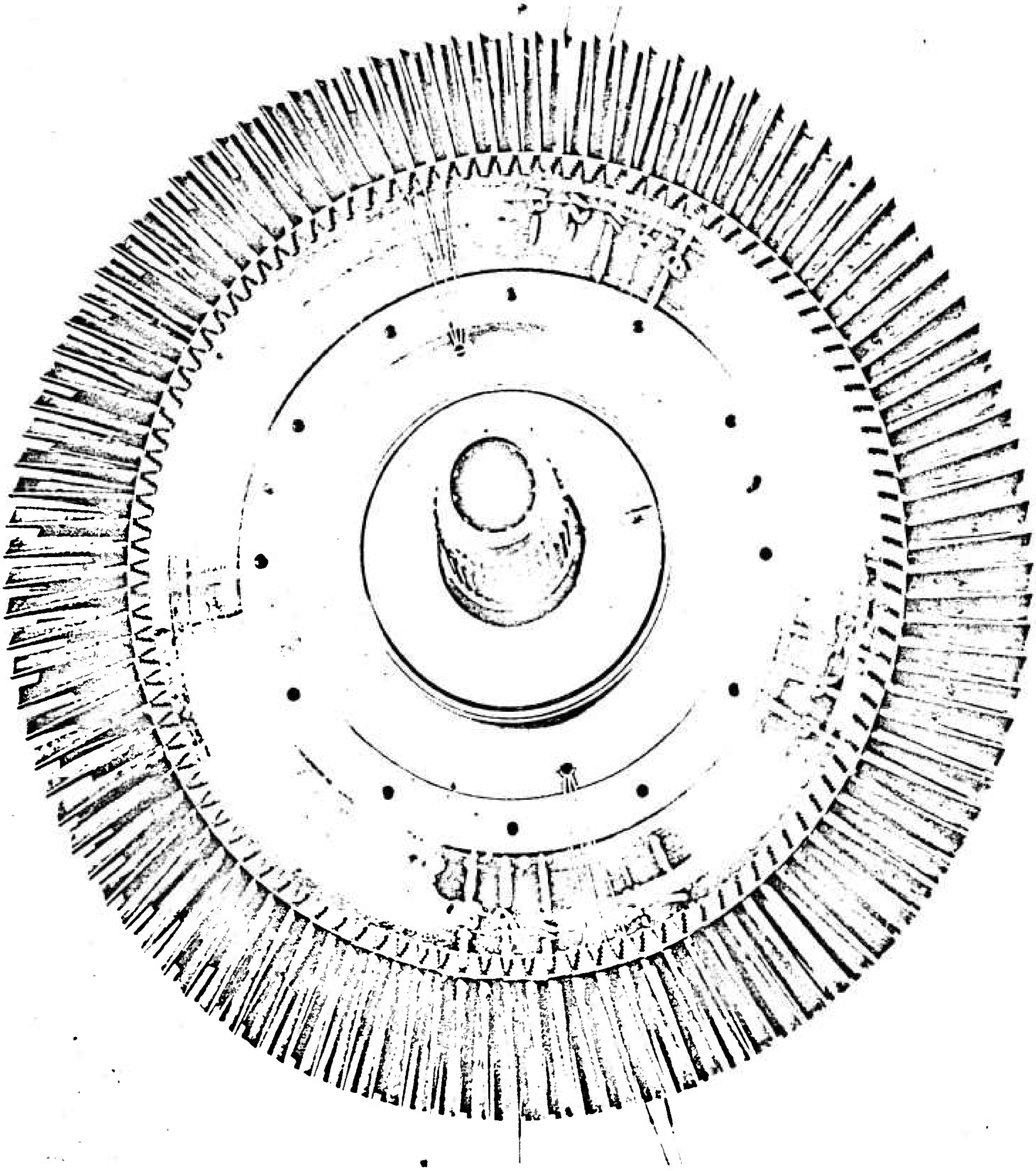
Machine Set-Up For Grinding Fir Tree

Figure 26



Cooled Rotor Assembly With Impeller Plate Figure 27

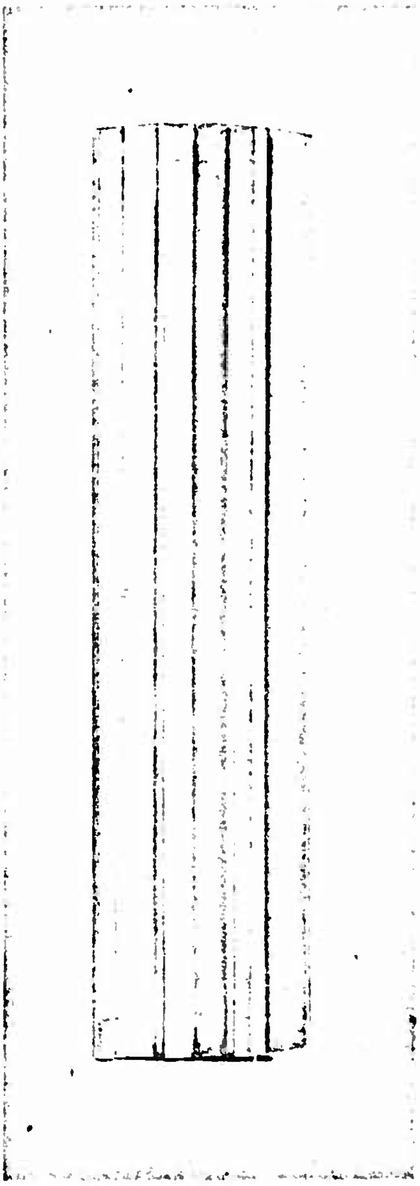
[redacted]



Cooled Rotor Assembly Without Impeller Plate Figure 28

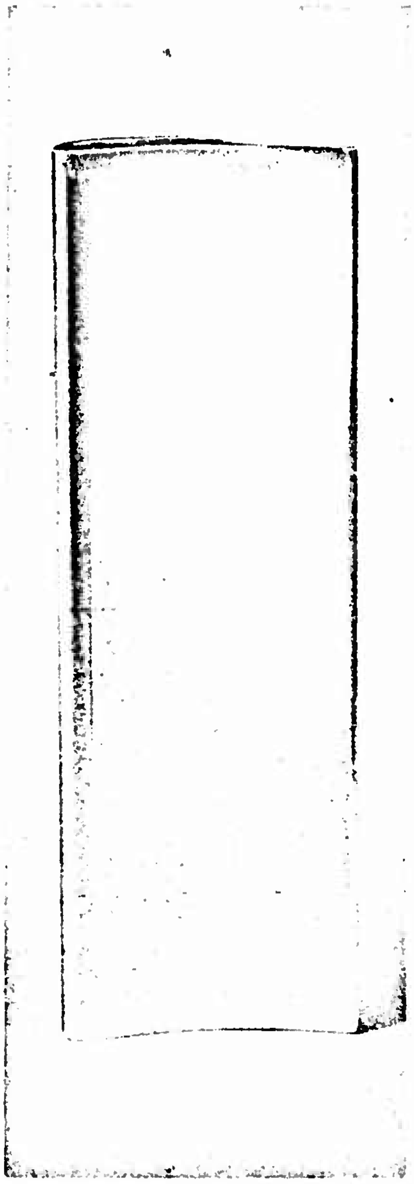
[redacted]

[redacted]



Cast Stator Blade Strut

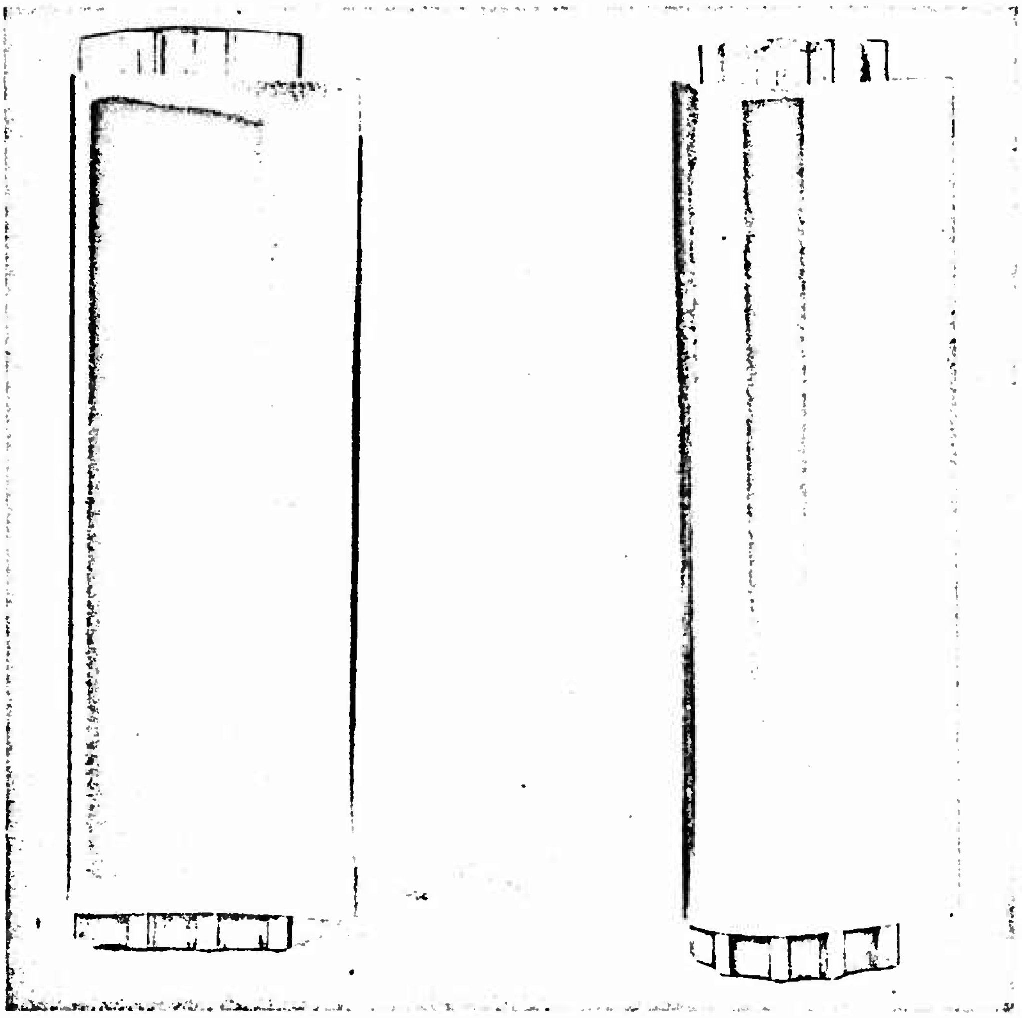
Figure 29



Cooled Stator Blade Airfoil

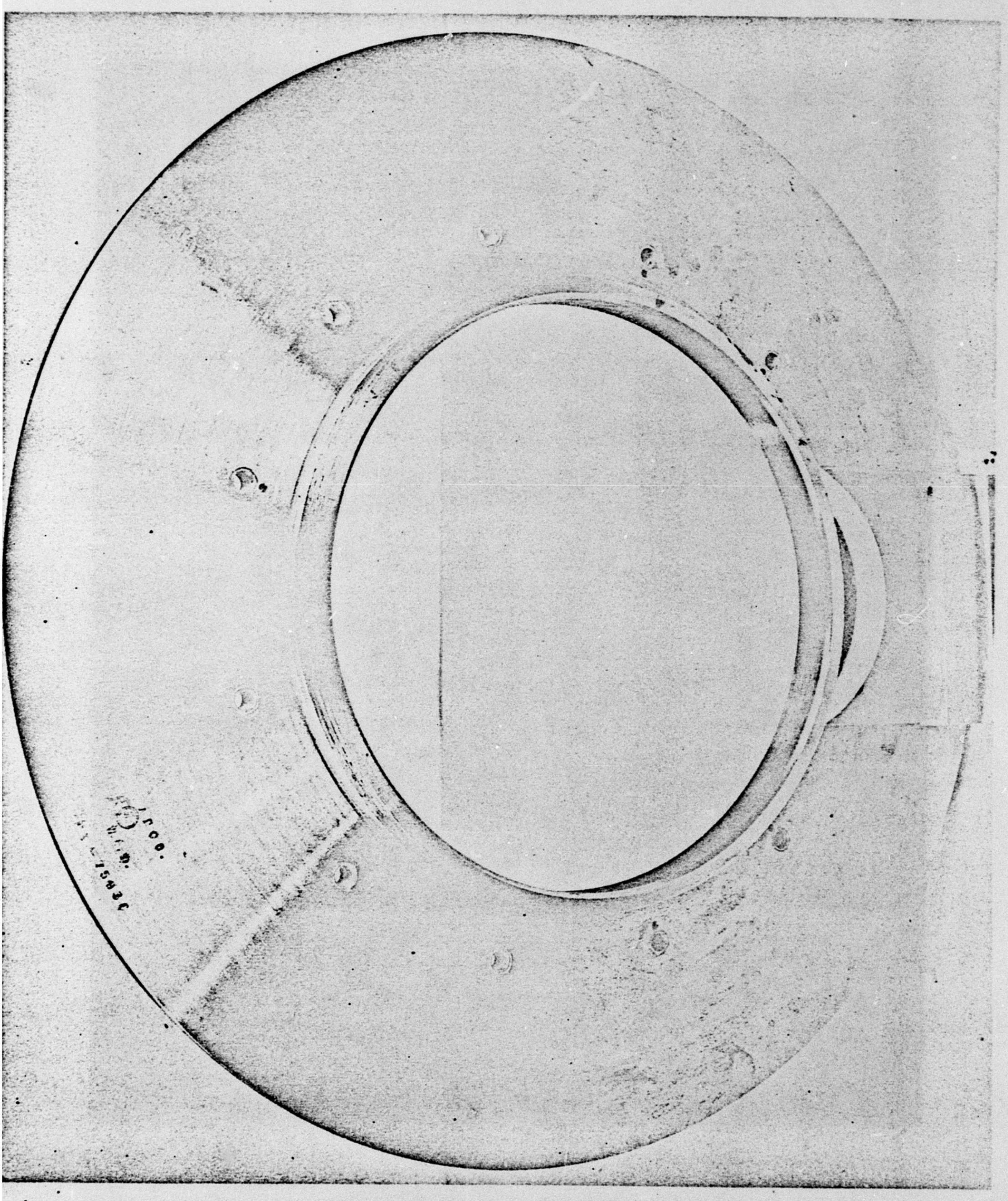
Figure 30

[redacted]

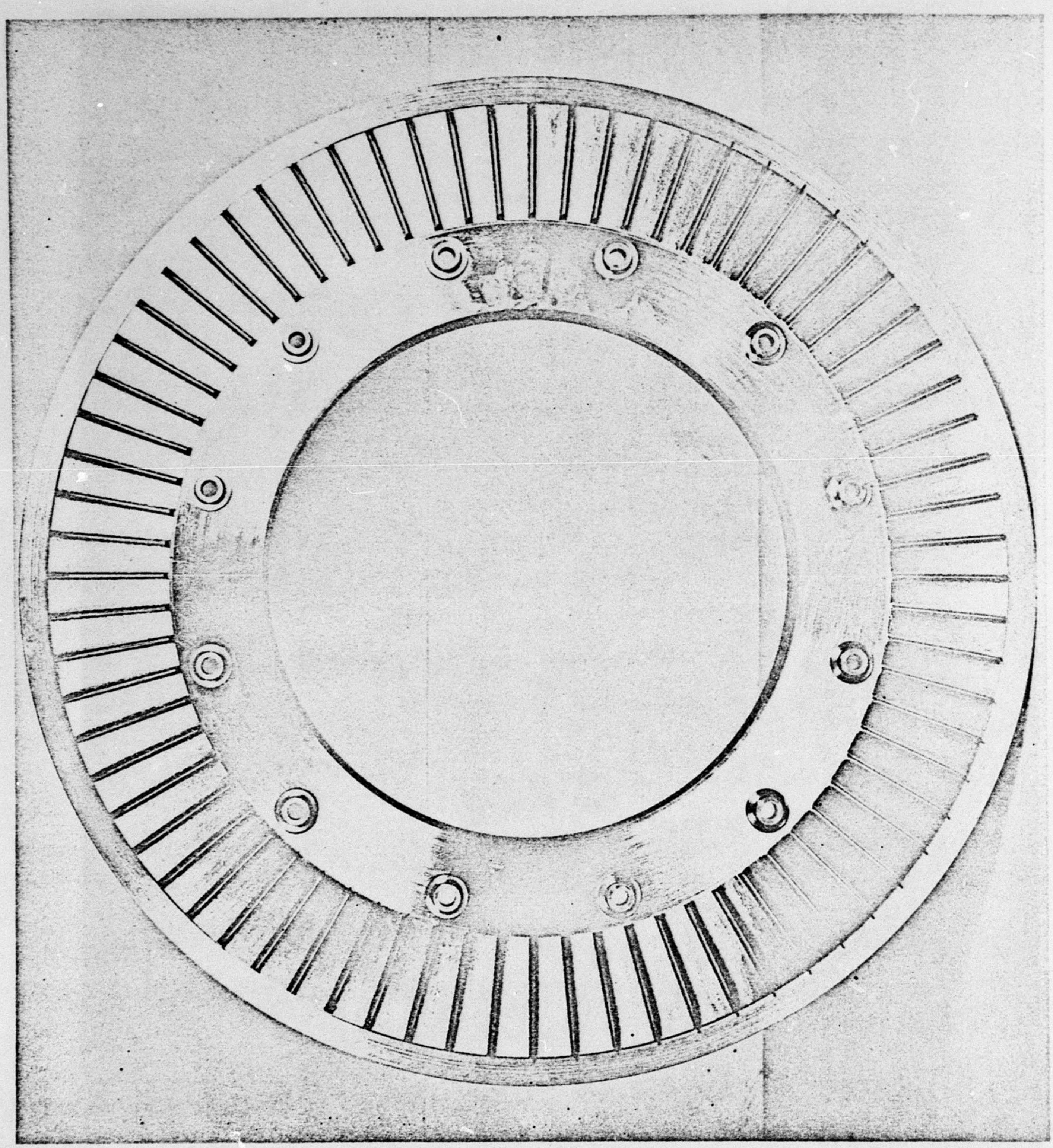


Brazed Stator Blade Assembly

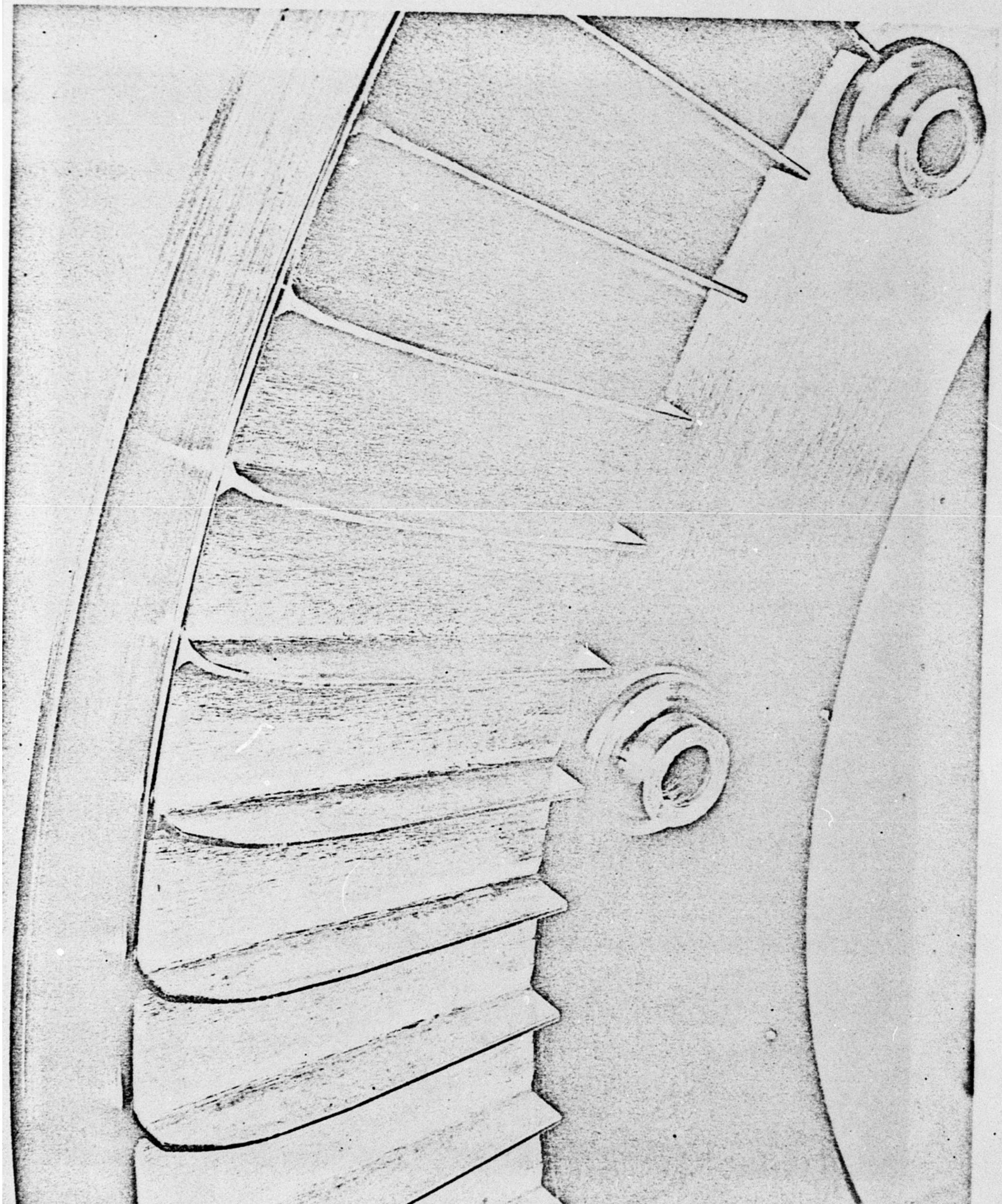
Figure 31



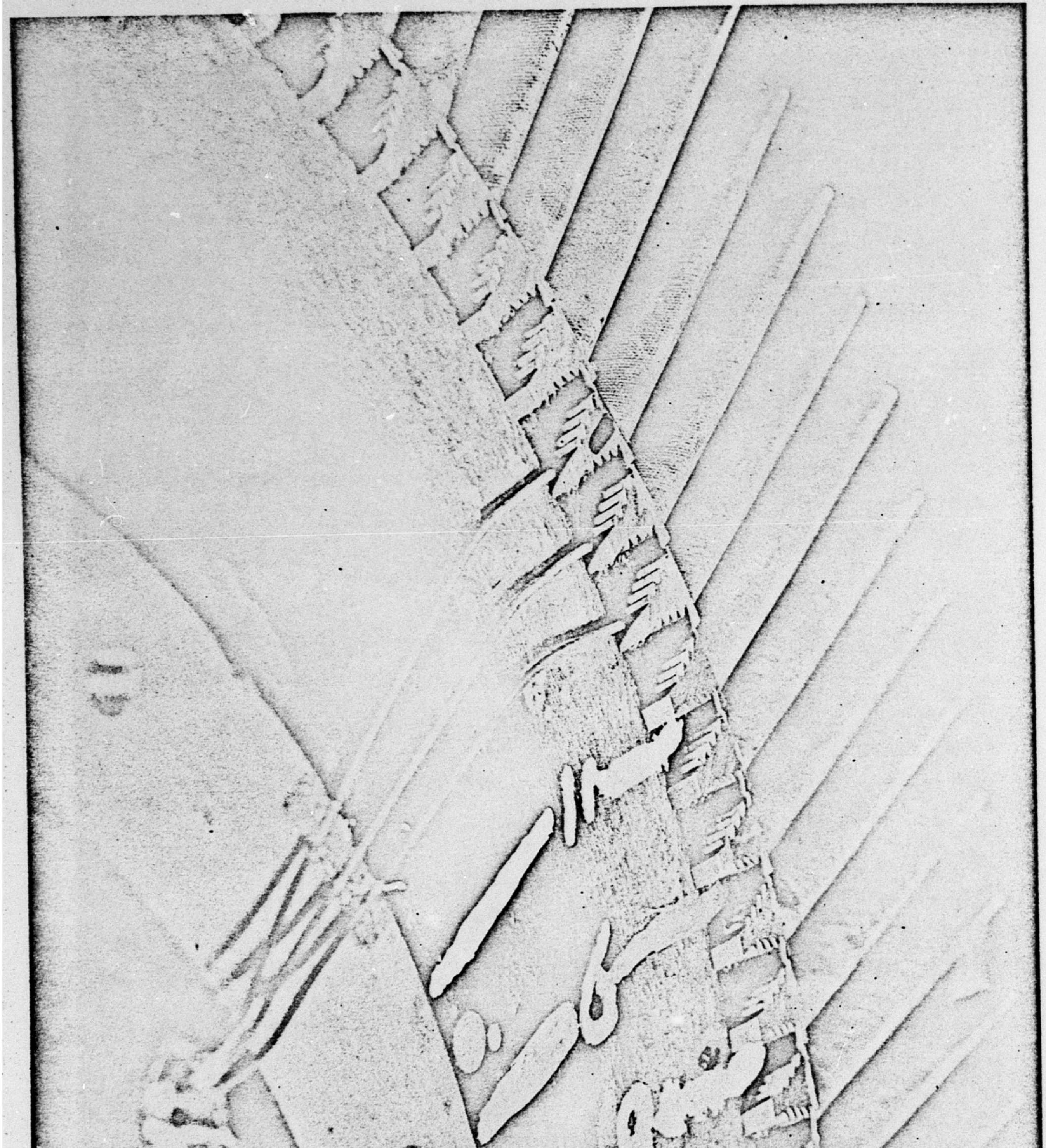
Impeller Plate (Labyrinth Seal) **Figure 32**



Impeller Plate (Radial Vanes) Figure 33



Impeller Plate. (Radial Vanes Close-Up) Figure 34



Cooled Turbine Rotor Blade Instrumentation

Figure 35

[redacted]

C.W.R. REPORT NO. 300-38

## III. ENGINE TESTING

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[redacted]

## III. Engine Testing

### A. General

Testing of the transpirational air cooled turbine assembly, shown in Figure 27, consisted of the program listed on Table III. This testing was directed toward demonstrating the feasibility of transpiration air cooled turbine rotor blades at elevated temperature operation, and collecting data revelant to their performance in a full-scale engine.

Several on-the-stand inspections were made during testing, as noted on the test program in Table III, so that skin temperature readings could be recorded from the temperature indicating paints (Thermocolor), and also to detect any incipient failures. This way any change in condition developed on the blades would probably be detected in time and noted so that any relation to an engine operation point that may have given rise to any blade change might be determined.

### B. Airfoil Temperatures

The method used to determine the transpiration turbine blade airfoil temperature was by using temperature indicating paints (Thermocolor) on the trailing edges of the porous airfoil. The installation of any other type of temperature indicating device would have resulted in local clogging of the porous airfoil and indicated a false temperature reading, which would be much higher than the actual temperature. By painting only a narrow band along the trailing edge where film cooling would occur from the air transpiring forward on the porous blade airfoil, a fairly accurate airfoil temperature can be obtained even though the porous material is clogged to some extent by the paint.

The temperatures recorded from the various changes in paint colors at the blade inspection periods during testing have indicated that the maximum airfoil temperature at the trailing edge was 1184°F. This temperature was indicated after the 1845°F (average) turbine inlet temperature operation was completed. As there was hardly any noticeable area change in color of the paint indicating that this temperature had just been reached. In Figure 36 is plotted the observed skin temperatures indicated by the temperature indicating paints.

### C. Strut Temperatures

Data obtained from the thermocouples installed in the transpiration cooled turbine blade struts are also plotted in Figure 36. Reliable readings from these thermocouples are only available for the initial phases of the cooled blade testing. Temperature readings observed at the higher temperature points were very erratic and were much lower than those recorded at lower turbine inlet gas temperatures, indicating malfunctioning of the thermocouples.

In order to estimate the strut temperature at the root of the blade for a 1845°F turbine inlet condition the data taken at the lower points was correlated with the theoretical design. From this analysis an estimated strut temperature of 740°F was obtained. Another temperature reading available that indicated the above temperature on the root of the blade strut to be approximately correct is that the temperature indicating paints on the butt of the blade showed that the lower half of the fir tree attachment did not exceed 824°F.

## D. Blade Cooling Air

The cooling air flows to the transpiration cooled turbine blades measured during testing up to 1845°F turbine inlet temperature did not exceed 3.25% of the engine flow, as shown on Figure 37.

Initially the cooling air flow through the blades was calculated to be in the vicinity of 5 to 6% metered by the orifices at the root of the cooling air passages at the base of the blade. In this case the actual metering occurred on stator support orifices that had been designed to pass a 3 to 4% cooling air flow if the originally designed higher cooling air flow was not necessary. As it turned out, the 3.25% cooling air flow was sufficient with results exceeding the anticipated cooling effects calculated. Cooling air and other relative pressures recorded during these tests have been plotted in Figure 38.

## E. Engine Performance

As the basic intent of these tests was to prove the feasibility of transpiration cooled turbine blades at high temperature operation, extensive instrumentation to measure engine performance was not incorporated into the engine test vehicle. However, sufficient instrumentation was employed to give general operating characteristics to compare with data obtained on a basic calibration of the standard engine. Figure 39 shows the plotted data from both the standard engine calibration and the transpiration cooled turbine engine tests.

One of the things that became very obvious in the initial testing of the transpiration cooled turbine blade was that the exhaust gas temperature observed was approximately 200°F higher than the standard engine for the same PRM's. Investigations and analysis of data, from a previous convection cooled turbine test, with approximately the same amount of bleed used as in the transpiration cooled test, showed an increase in turbine exhaust temperature of approximately 70°F when compared with a standard uncooled engine configuration. Analysis of the performance data from the transpiration cooled engine configuration showed that there was a loss in mass flow somewhere after the compressor discharge. Calculations indicate that this loss amounted from 9.0 to 9.5 lbs/sec at the higher RPM's. Heavy "breathing" noted at the center main bearing support indicated the main compressor labyrinth seal was leaking. This was later confirmed upon the disassembly of the engine after the test. Therefore, comparison of the performance characteristics of the standard uncooled engine with the transpiration cooled configuration, as shown in Figure 39, is not truly indicative of actual cooled engine performance. A valid comparison is shown in Figure 40, which compares the performance of the transpiration air cooled blade configuration and the uncooled blade configuration with the same leakage conditions. Comparison of this test data indicates that the transpiration air cooled configuration ran approximately 50°F higher in exhaust gas temperatures. Consideration must be given to the fact that in the cooled configuration the turbine rotor tip clearances were much higher (.110") to allow for higher operating temperatures so that any thermal distortion of the turbine casings, at these elevated temperature conditions, would not affect the mechanical performance of the cooled rotor stage. Even though it is conservatively assumed that the 50°F increase in exhaust temperature (see Figure 40) is directly attributed to the loss in turbine efficiency caused by the effusion of cooling air into the gas passage, it is clear that the increases in operating gas temperature afforded through the use of transpiration cooled blades will result in substantial increases in thrust, and that any reduction in turbine performance will be negligible.

## F. Transpiration Cooled Turbine Performance

As mentioned in the design section of this report, the airfoils of the transpiration cooled turbine blade were modified to incorporate the porous airfoils formed from porous tubing without requiring a seam on some part of the airfoil, so that a completely porous airfoil surface could be tested. The modification of reducing the chord length at the airfoil tip, changes the vector diagram of velocities on the blade, and redistributes the radial pressures along the blade, thus reducing the efficiency of this stage slightly. As mentioned above, another factor that tended to reduce the overall turbine efficiency was that higher than normal rotor blade tip clearances (.110") were used.

## G. Blade and Engine Test Discussion

Two complete sets of rotor assemblies with transpiration air cooled blades were fabricated. Since each assembly consisted of 110 blades, 220 blades, plus a few spares, were fabricated for the test program. In order to derive the maximum amount of data on materials of construction, four basic types of blades were fabricated to make up the two rotor assemblies:

| | |
|---|---|
| Type I | H.S.-31 Struts, H.S.-25 Shelves, H.S.-25 Airfoils |
| Type II | H.S.-31 Struts, H.S.-25 Shelves, N-155 Airfoils |
| Type III | Inconel 713C Struts, H.S.-25 Shelves, H.S.-25 Airfoils |
| Type IV | Inconel 713C Struts, H.S.-25 Shelves, N-155 Airfoils |

The first rotor was assembled with Type I and Type II blades. In the engine test of the initial assembly, it was noticed that engine light-up was difficult to achieve, and excessive torching of hot gases in the tail pipe persisted for very long periods on the order of 120 seconds. After reaching 57% of rated RPM, an inspection of the cooled rotor blades was made. This inspection revealed the airfoil skins of the cooled rotor stage were damaged. This condition was not unexpected, since approximately twelve hot starts were made which resulted in extremely high turbine temperatures. Measurements made at the outer turbine support indicated at this point, gas temperatures in excess of 2300°F were experienced. The visual inspection of the blades showed three or four of the airfoil skins separated from their struts. In addition, the skins of remaining blades were blackened and appeared brittle in the tip region. None of the struts were damaged. As can be seen by the photos shown in Figures 41 and 42, the secondary damage to the cooled rotor stage was rather extensive. Total engine time accumulated on this rotor before damage was approximately 1 hour and 40 minutes. It is considered significant that no other damage to the engine resulted from this data. It is logical to conclude that loss in turbine performance, as shown by the 50°F rise in exhaust gas temperature, is negligible from this failure.

In order to correct the engine starting difficulties and the excessive torching condition, the damaged rotor was removed and tests were conducted on the engine using solid uncooled rotor blades. Two items became immediately apparent: the main fuel control of the engine was malfunctioning because of a faulty fuel pressure spring, and there was a loss in turbine air flow through a leak in the labyrinth seal. After correcting the fuel control problem and calibrating the engine again to account for the loss of turbine airfoil, testing of the second cooled rotor assembly was initiated.

The second cooled rotor was assembled with Type III and Type IV transpiration cooled blades.

Inspection of the cooled blade assembly was made after the idle point, 57% RPM, and 63% RPM. The condition of the blades after the idle and 57% RPM test point revealed the condition of the blades was excellent, and there was no noticeable discoloration of the porous material. After the 63% RPM test point, it was found that two porous airfoils had separated from the blade strut and caused some secondary damage to the other cooled blades. This secondary damage occurred on the trailing edge tips of the cooled blades. The other areas of the blade airfoil were in excellent condition. Figures 43 and 44 show the transpiration cooled turbine rotor assembly after this test and the condition of the blades. Several of these blades were still in good condition, and several that had received secondary damage at the tip trailing edge were repaired by cutting back the damaged area and blending. Although these repaired blades would affect engine performance the feasibility of transpiration cooled blades at elevated temperatures could still be evaluated. Replacement of the few damaged blades was made and testing continued.

The next series of tests were uneventful and a stepped program up to 1816°F inlet temperature was completed. After having run approximately twenty minutes at this temperature a slight increase in vibration amplitude of three to four mils on the turbine flange was noted, indicating unbalance developing in the turbine section. Inspection of rotor revealed that one of the porous airfoils had separated from a blade strut. In this case secondary damage was quite restricted as can be seen in Figure 45. Excellent data was obtained from the cooled blades after this test in regard to blade temperatures presented earlier.

Replacement of the failed blade was made, and testing continued in which the turbine inlet temperature was increased to 1845°F.

Shortly after taking a reading, when two hours of endurance had been completed, turbine flange vibration amplitudes increased 2 to 3 mils. A test stand inspection of the blades again revealed that another blade airfoil had separated from the blade strut. Figures 46 and 47 show the rotor assembly and the strut from which the porous airfoil came off along with the resulting secondary damage. The remaining transpiration cooled turbine blades were in very good condition. Figures 47a and 47b show blades that were tested in the engine through the entire high temperature operation.

Inspection of the completely disassembled engine after the final test on these transpiration cooled turbine blades revealed that the other critical components in the combustion chamber, turbine section, and exhaust ductwork were in good condition after the high temperature testing. Some buckling occurred in the exhaust ducts but was not of any great consequence. The combustion chamber liners were in excellent condition after operating at an average inlet temperature of 1845°F, along with known uneven temperature distribution detected by thermocouples in the exhaust duct and jet pipe. When the average exhaust gas temperature of 1402°F was recorded, indicating an average turbine inlet temperature of 1845°F, the peak exhaust gas temperatures from individual thermocouples recorded were as high as 1575°F. This meant that turbine inlet gas temperature peaks were above 2000°F in some areas. The cause for this may be in the fuel distributors which were operating over their normal range. The only change noted in the combustion chamber after 18 hours and 43 minutes of testing, of which approximately two and one-half hours was at 1800°F to 1845°F, there was more than the normal amount of carbon deposit on the combustion chamber flame tubes as shown on Figures 48 and 49.

Another effect of the high temperature turbine inlet operation was noted on the first stage turbine stator blades. The aluminum coating on the cast Haynes-Stellite 31 blades had started to peel in some areas, but this did not seem to adversely affect the base material.

Inspection of the labyrinth seal, upon disassembly of the engine, showed that this part was not properly seated in the last major disassembly and that a large passage was created allowing a high leakage shortly after the discharge of the compressor. Calculations made to determine the leakage indicated that it was possible for a flow of approximately 10 lbs/sec to pass through. This ties in quite closely with the calculations made of the loss in turbine mass flow of 9.0 to 9.5 lbs/sec.

## H. Description of Engine

The engine used in this test was a J65-W4B reworked as follows:

1. Installation of transpiration air cooled blades in final stage rotor.
2. Holes for cooling air to the first stage transpiration cooled rotor blades in the inner stator blade support.
3. Plugged metering air hole at the aft end of turbine shaft.
4. Provision for cooling air at periphery of rear face of second stage disc by modifying exhaust duct cone for an external air supply.
5. First stage standard stator blades replaced with Haynes-Stellite 31 stator blades aluminum dipped.
6. Second stage stator blades welded together in pairs at the inner periphery to form continuous frame box structures, with increased clearances between pairs of stators.
7. Substitution of second stage turbine rotor blades of Inco 700 with strengthened fir tree attachment.
8. Substitution of second stage turbine rotor disc with new fir tree design.
9. Substitution of Haynes-Stellite 25 combustion chamber with aluminum dipped N-155 flame tubes.
10. Variable area exhaust nozzle.
11. Increased turbine rotor blade tip clearance.

## I. Description of Instrumentation

The following instrumentation was installed for measurement of blade temperatures and basic engine performance:

1. "Ceramo" thermocouples inserted in cooled blade strut to one-quarter inch above blade shelf.
2. Temperature indicating paints (Thermocolor) applied to housings and liners in combustion chamber, turbine rotor discs, turbine blades, and exhaust ducting.
3. Four thermocouples at the exit of the combustion chamber on the inner liner.
4. Four thermocouples in the exhaust cone.
5. Four thermocouples in the jet pipe.
6. Total pressure probe in the exhaust cone.
7. Static pressure tap in the exhaust cone.
8. Three thermocouples at the inner stator support cooling air orifices.
9. Six static pressure taps, three on each side, of the inner stator support cooling air orifices.

10. Three thermocouples on the metering bellmouth screen.
11. One thermocouple at the compressor exit.
12. Total pressure probe at the compressor exit.
13. Six static pressure taps in the calibrated bellmouth.
14. Vibration pickup; two each at the front main bearing support, turbine flange, and one at the center main bearing support.
15. Slip ring assembly and thermocouples lead ducting through the engine shafting.
16. Standard pressure and temperature instrumentation for fuel flow, oil, and bearings.
17. Cooling air to the slip-ring assembly.
18. Plenum chamber pressure tap and temperature.
19. Standard engine equipment for measurement of speed, thrust, and controls.

## J. Test Installation

The engine was mounted on the standard outside test stand at Quehanna, Pennsylvania. Actual mounting consisted of attaching the engine to the flat bed of a trailer then securing the trailer to a floating bed which contained the thrust load measuring equipment.

Cooling air to the rear face of the second stage disc was supplied by a 600 CFM Schramm portable compressor.

A mechanically driven variable area nozzle was used to obtain the desired value of exhaust gas temperature by "sizing" the nozzle. The engine was started and accelerated to the desired speed before closing the nozzle to increase the turbine inlet temperature. The nozzle was also opened before decelerating from the speed point.

## Table III

## Transpiration Cooled Turbine Blade Engine Test Program

| Type Test | Conditions | Remarks |
|---|---|---|
| 1. Calibration | Standard | |
| 2. Transpiration Cooled Turbine | a. Starting | High exhaust gas temperatures encountered resulted in several attempted starts with excessive flaming at exhaust. |
| | b. 4000 RPM | Hot start |
| | c. 5000 RPM | |
| | d. 6000 RPM | |
| | e. 6500 RPM | Vibration erratic. Inspected blades on stand. Removed engine from stand for closer inspection of damaged blades. |
| 3. Starting Tests | a. Starting | Hot starts initially encountered. Corrected by modifying fuel control to manual at start, and using larger exhaust nozzle. |
| | b. 7000 RPM | Found exhaust gas temperatures higher than expected for 3% (approx.) cooling airbleed. |
| | c. 7500 RPM | |
| | d. 7880 RPM | Maximum allowable temperature of 1200°F of exhaust gas with standard nozzle. |
| 4. Transpiration Cooled Turbine | a. 4000 RPM | Normal cool start. Inspected blades on stand. Excellent condition. |
| | b. 6000 RPM | |
| | c. 6500 RPM | |
| | d. 7000 RPM | Inspected blades on stand. Excellent condition. |
| | e. 7500 RPM | Inspected blades on stand. Found failure of some cooled rotor blade airfoils. Removed rotor from engine to replace and repair blades. |

## Table III (Cont'd.)

## Transpiration Cooled Turbine Blade Engine Test Program

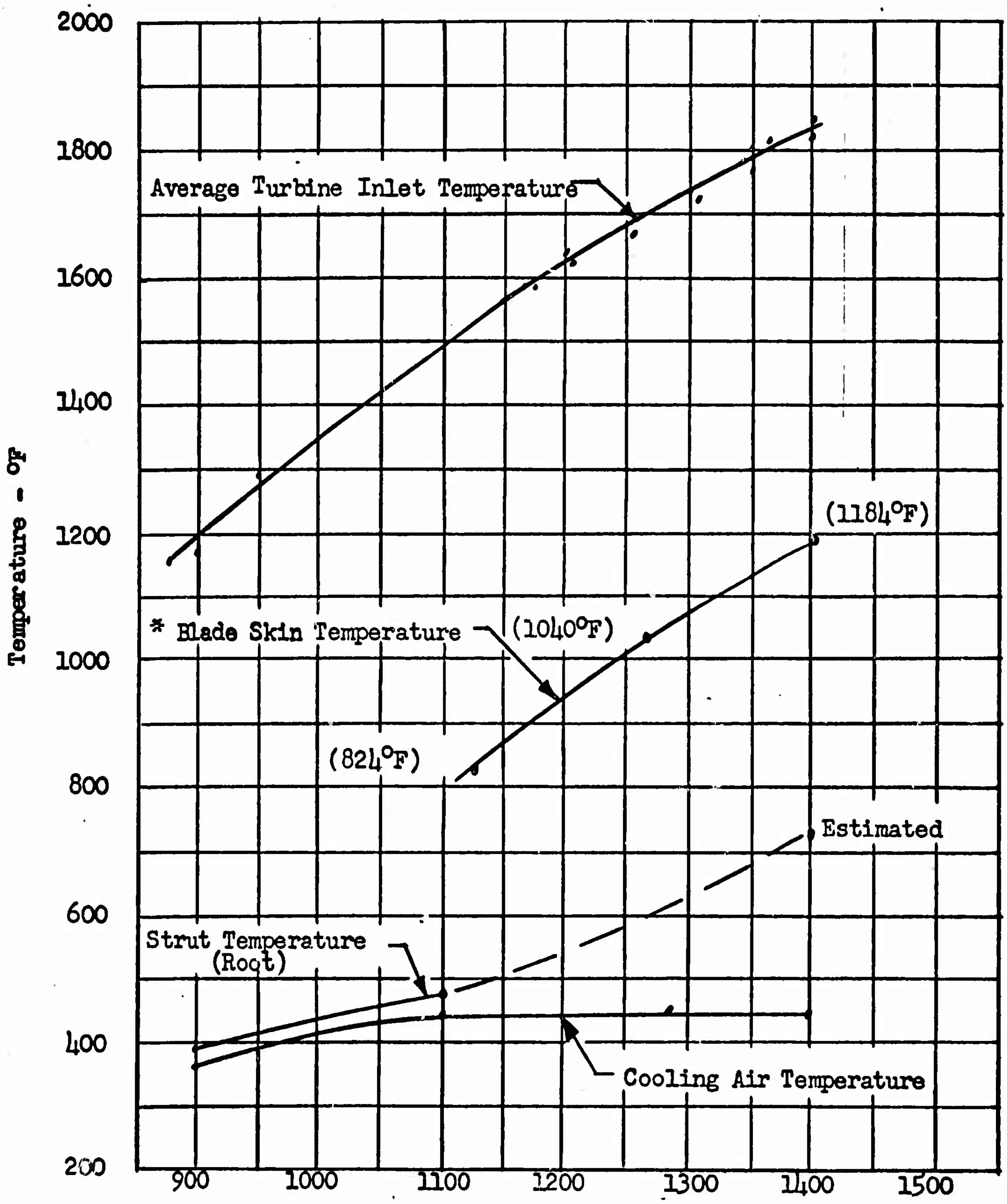
| Type Test | Conditions | Remarks |
|---|---|---|
| 5. Transpiration Cooled Turbine | a. 4000 RPM | Normal cool start. |
| | b. 7800 RPM | Set limit of 1200°F E.G.T. for RPM setting. Inspected blades on stand. In good condition. |
| | c. 7800 RPM | Closed variable area nozzle to get 1250°F E.G.T. |
| | d. 7800 RPM | Closed V.A.N. to get 1300°F E.G.T. |
| | e. 7800 RPM | Closed V.A.N. to get 1350°F E.G.T. |
| | f. 7800 RPM | Closed V.A.N. to get 1400°F E.G.T. Slight increase (3 to 4 mils) of vibration noted after 20 minutes of running at this point. Stopped engine. Normal rundown. Inspected blades on stand. Found primary failure of one airfoil skin. Removed rotor from engine to replace blade and repair secondary damage to other blades. |
| 6. Transpiration Cooled Turbine | a. 4000 RPM | Slightly hot start. |
| | b. 7800 RPM | Turbine inlet temperature - 1625°F. |
| | c. 7800 RPM | Close V.A.N. to get T.I.T.- 1725°F. |
| | d. 7800 RPM | Close V.A.N. to get T.I.T.- 1800°F. Ran for 2 hours at this point. Shortly after taking reading after 2 hours of endurance a slight increase in vibration (2-3 mils) at turbine was noted. Shut down and inspected blades on stand. Found one cooled airfoil had separated from strut. End of test. |

Total testing time: 18 hours and 43 minutes

C.W.R. REPORT NO. 300-38

TRANSPIRATION COOLED TURBINE BLADES

Gas and Blade Temperatures



Average Turbine Exit Temperature - °F

Figure 36

TRANSPIRATION COOLED TURBINE BLADES

Air Flows

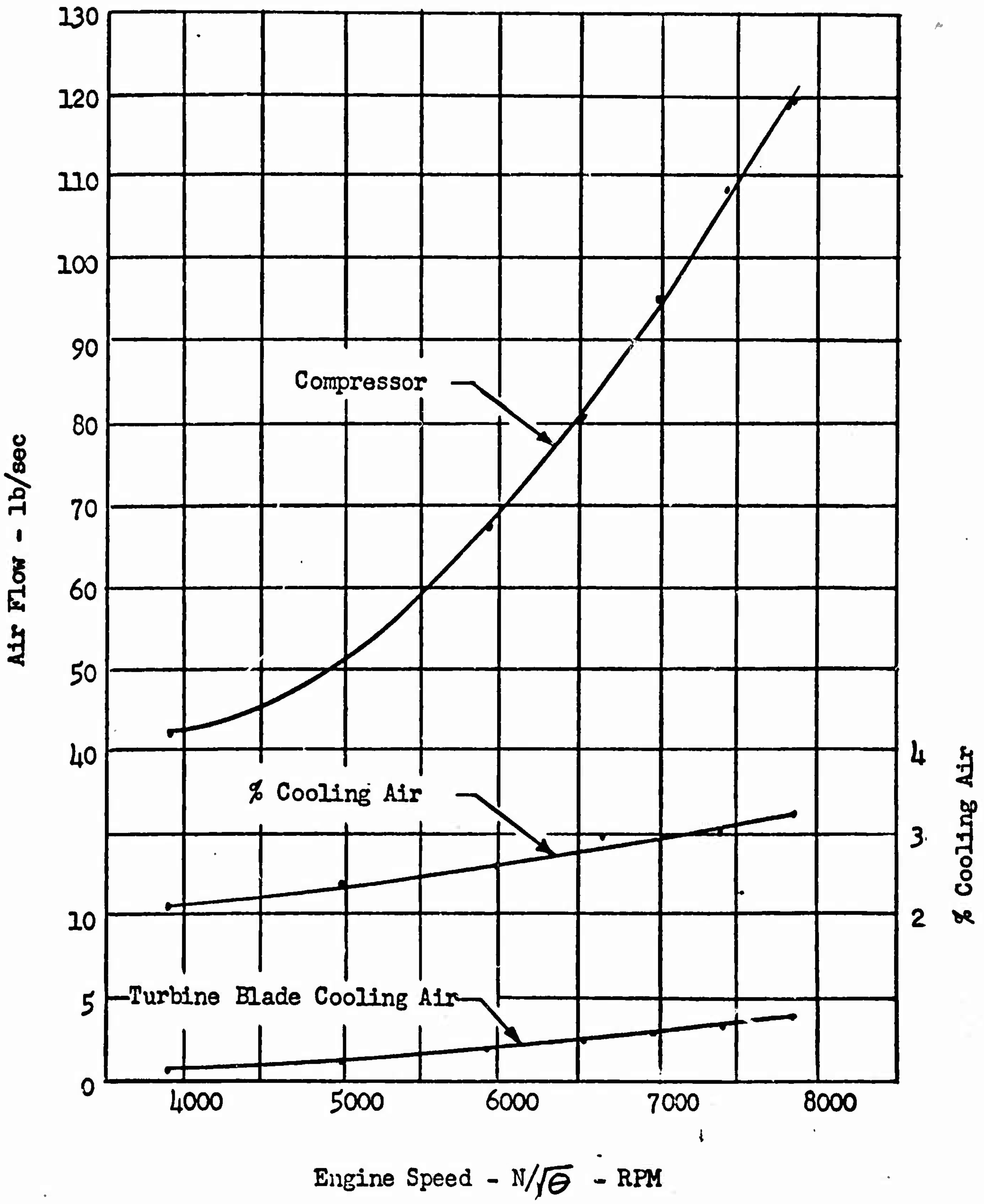
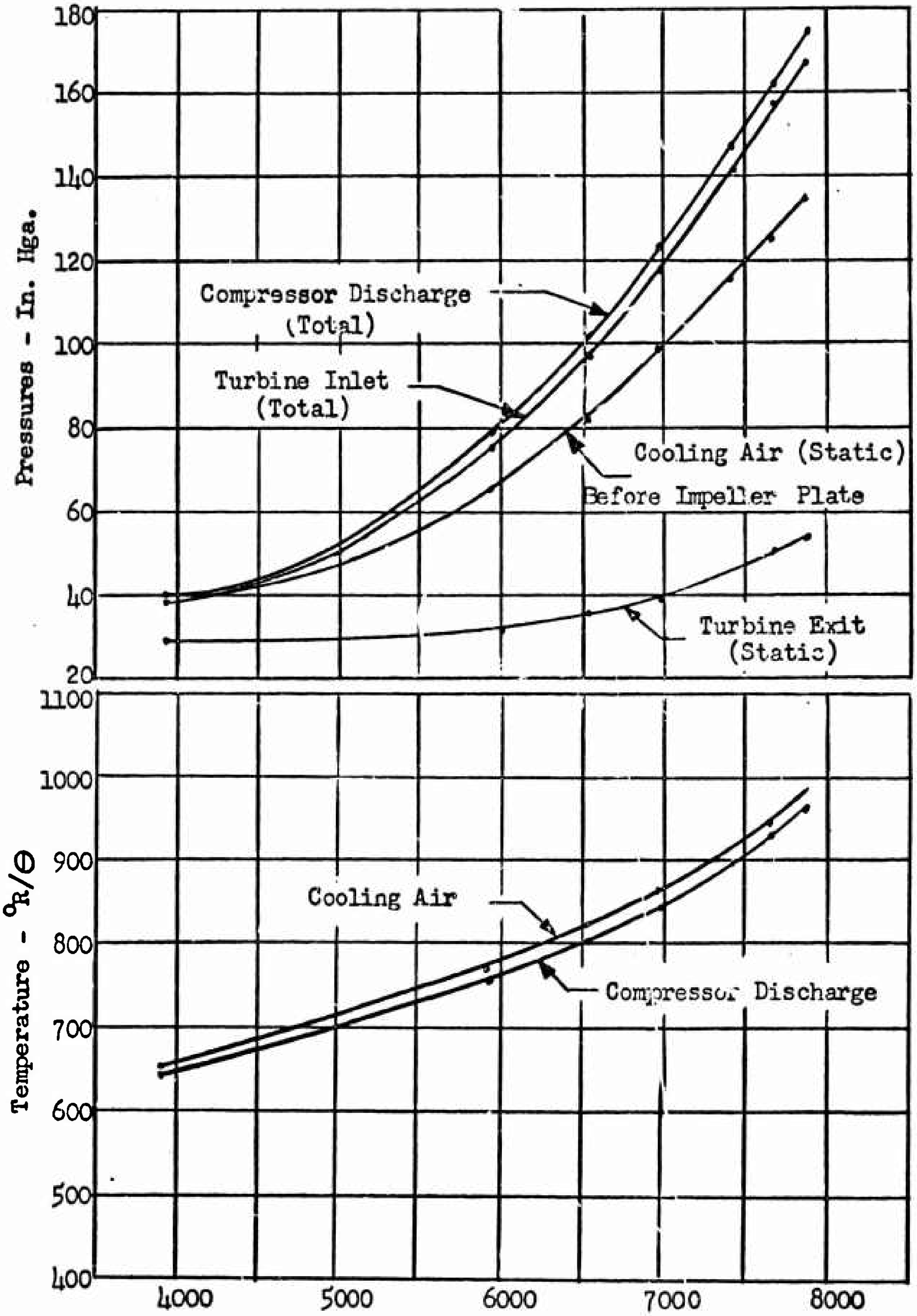


Figure 37

Pressures and Temperatures



Engine Speed - $N/\sqrt{\Theta}$ - RPM

Figure 38

TRANSPIRATION COOLED TURBINE BLADES

Engine Performance

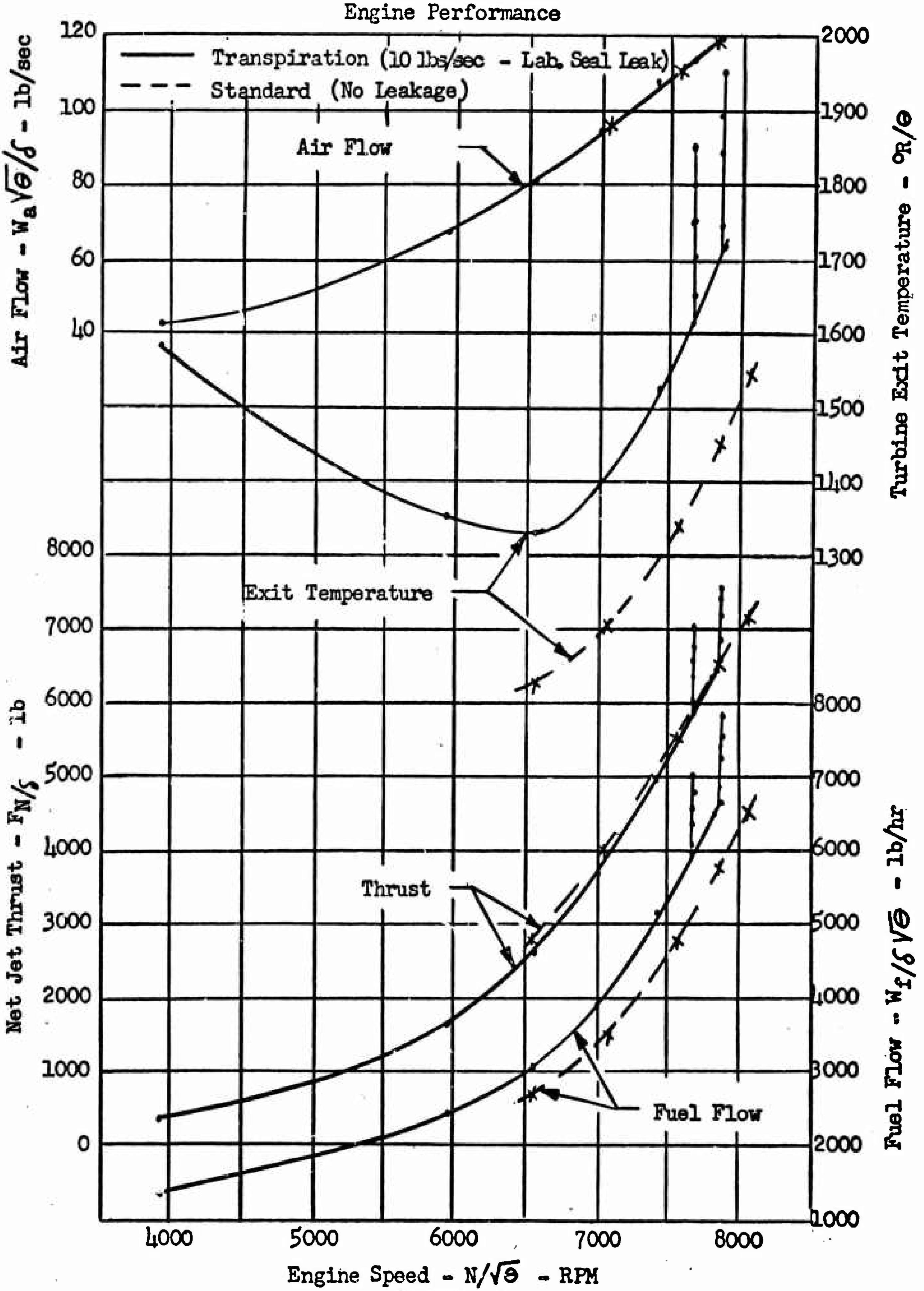


Figure 39

## TRANSPIRATION COOLED TURBINE BLADES

### Thrust vs Exhaust Gas Temperature *

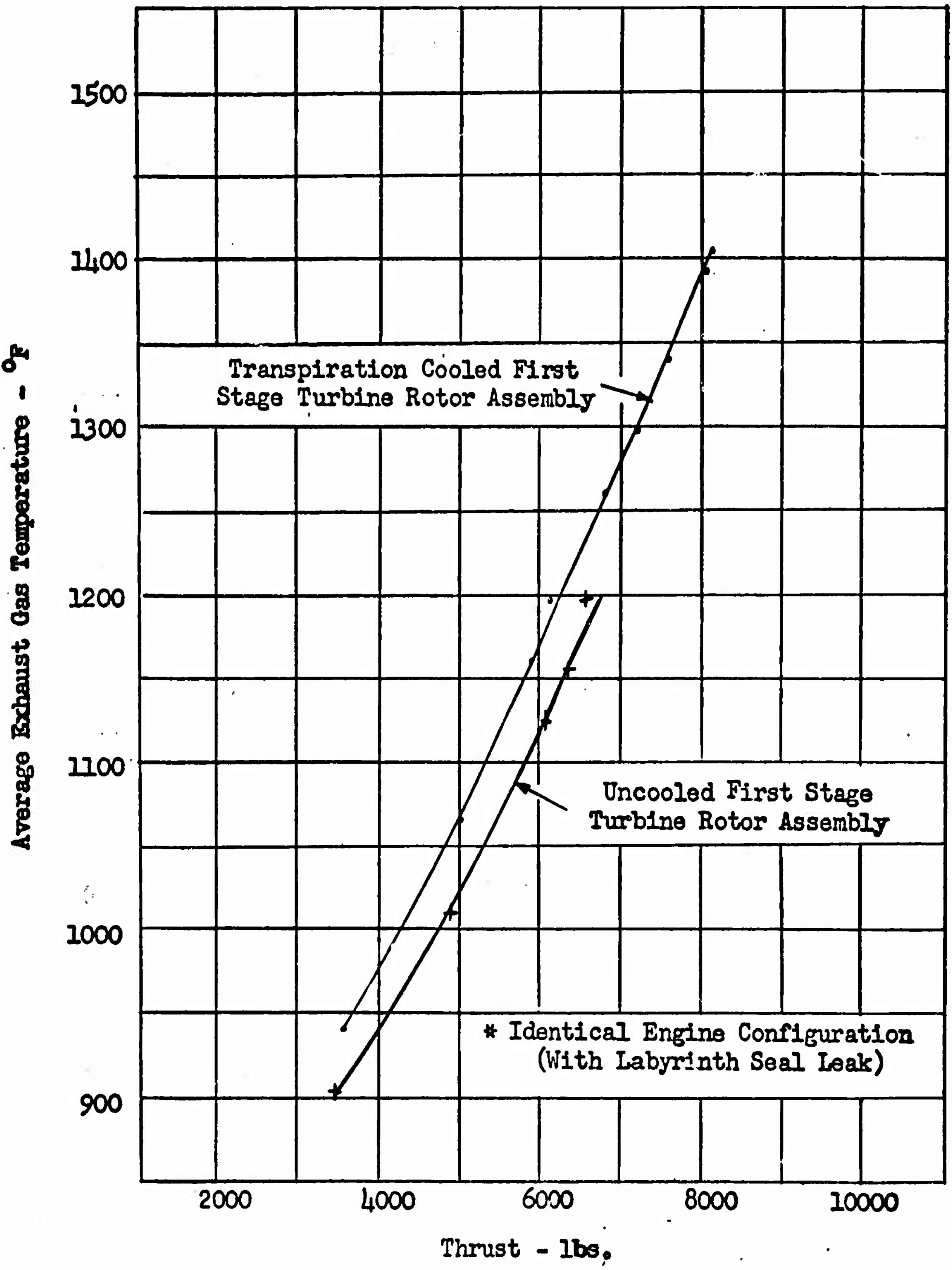
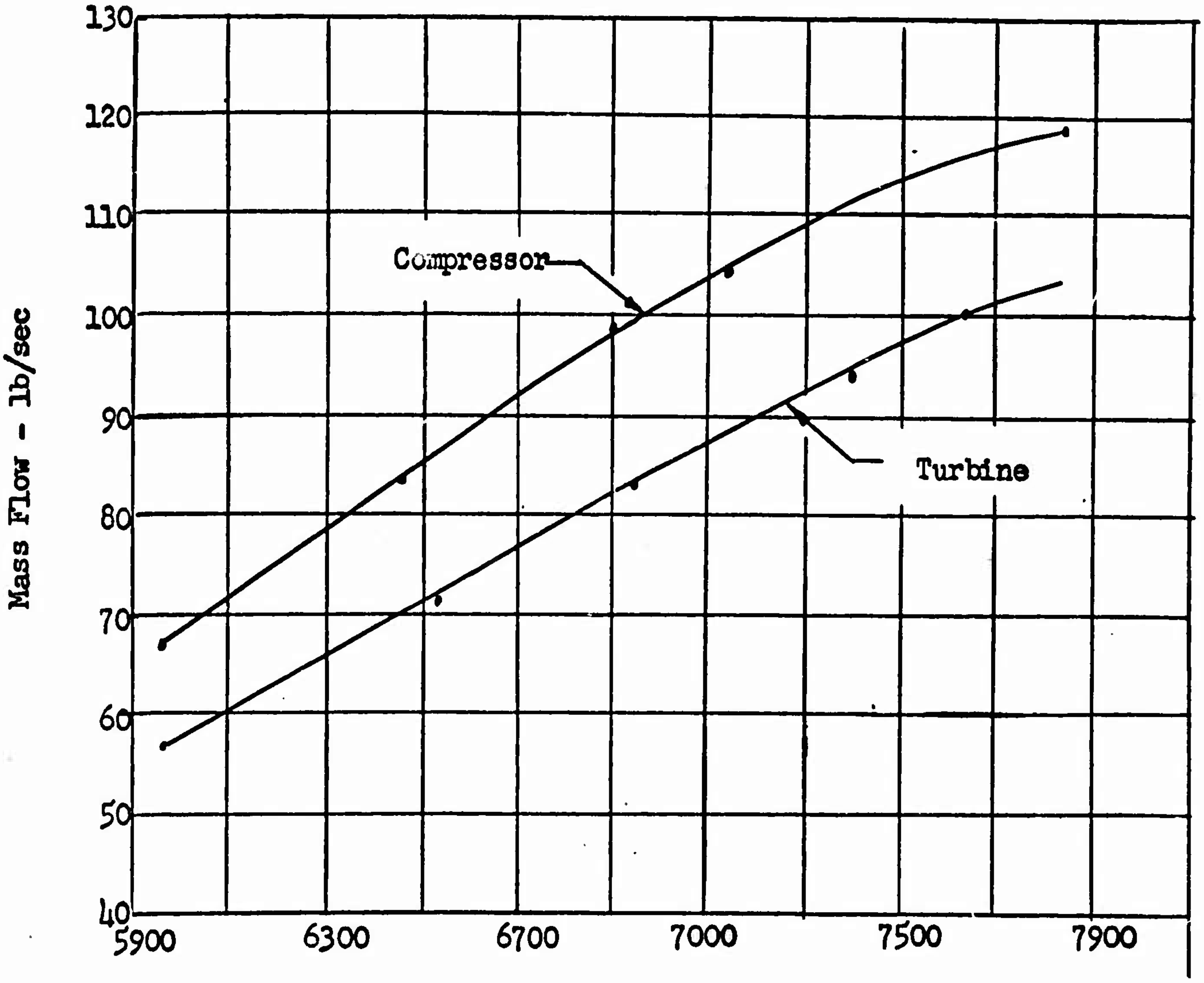


Figure 40

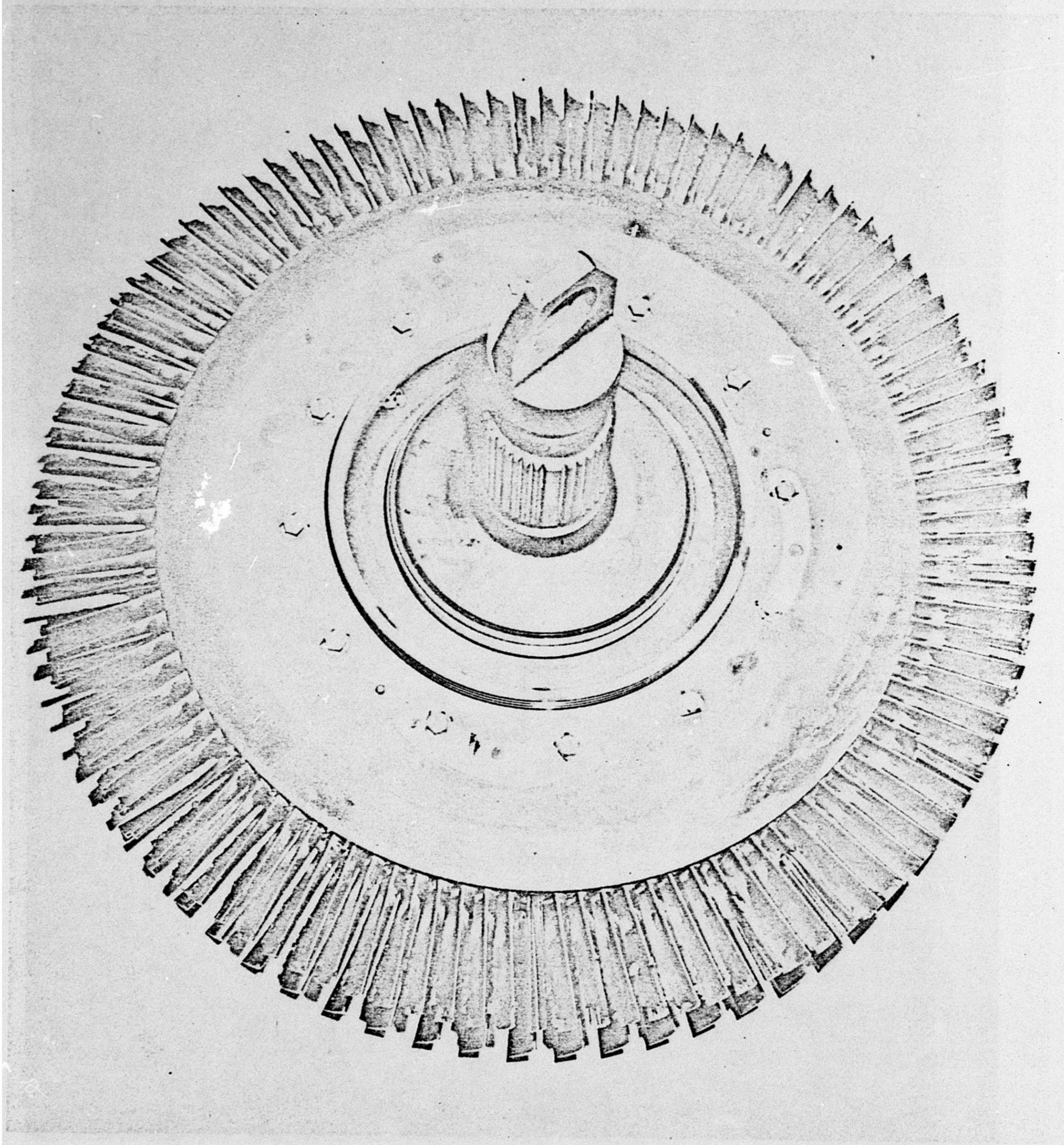
C.W.R. REPORT NO. 300-38

TRANSPIRATION COOLED TURBINE BLADES



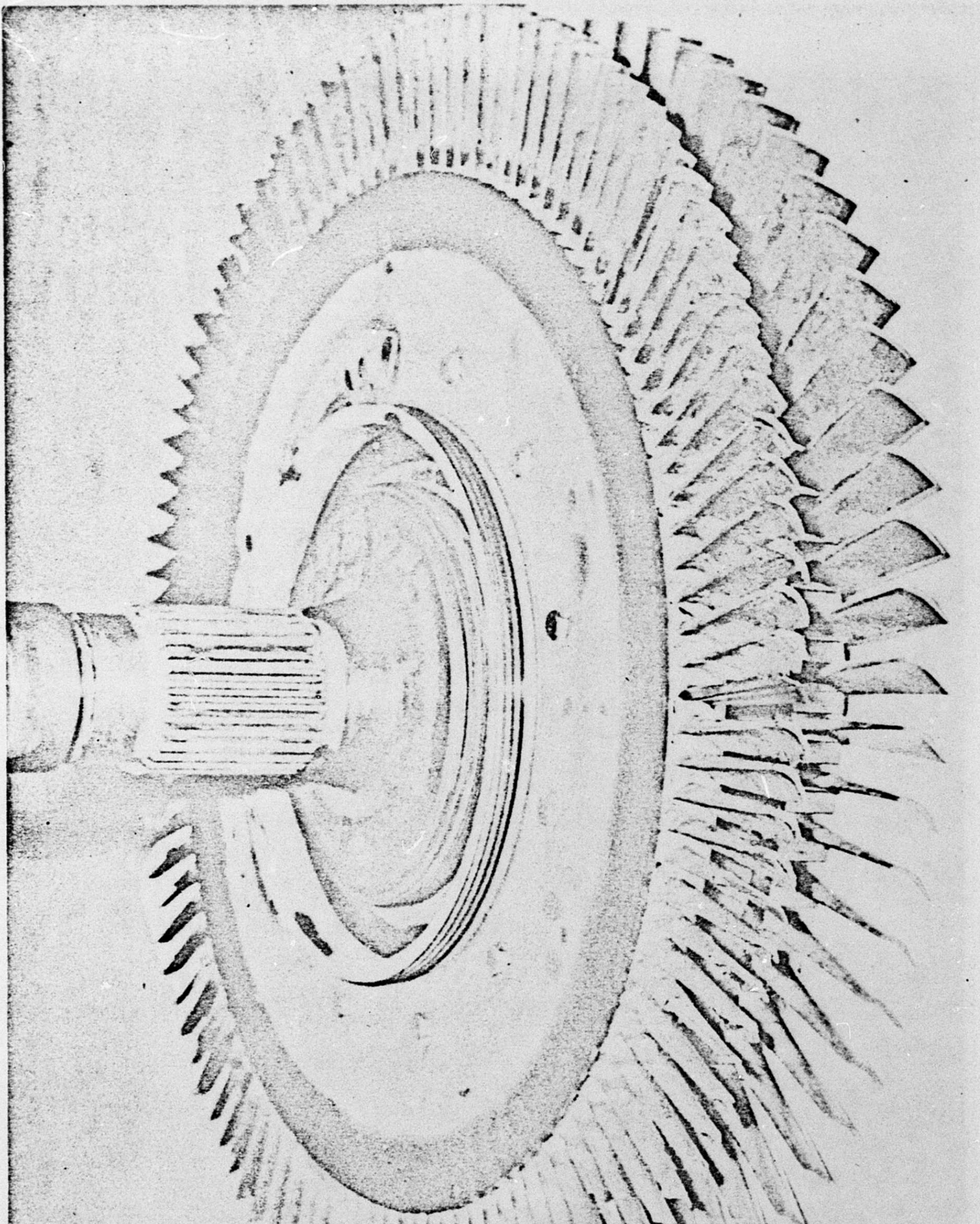
Engine Speed - N/√θ - RPM

Figure 40a



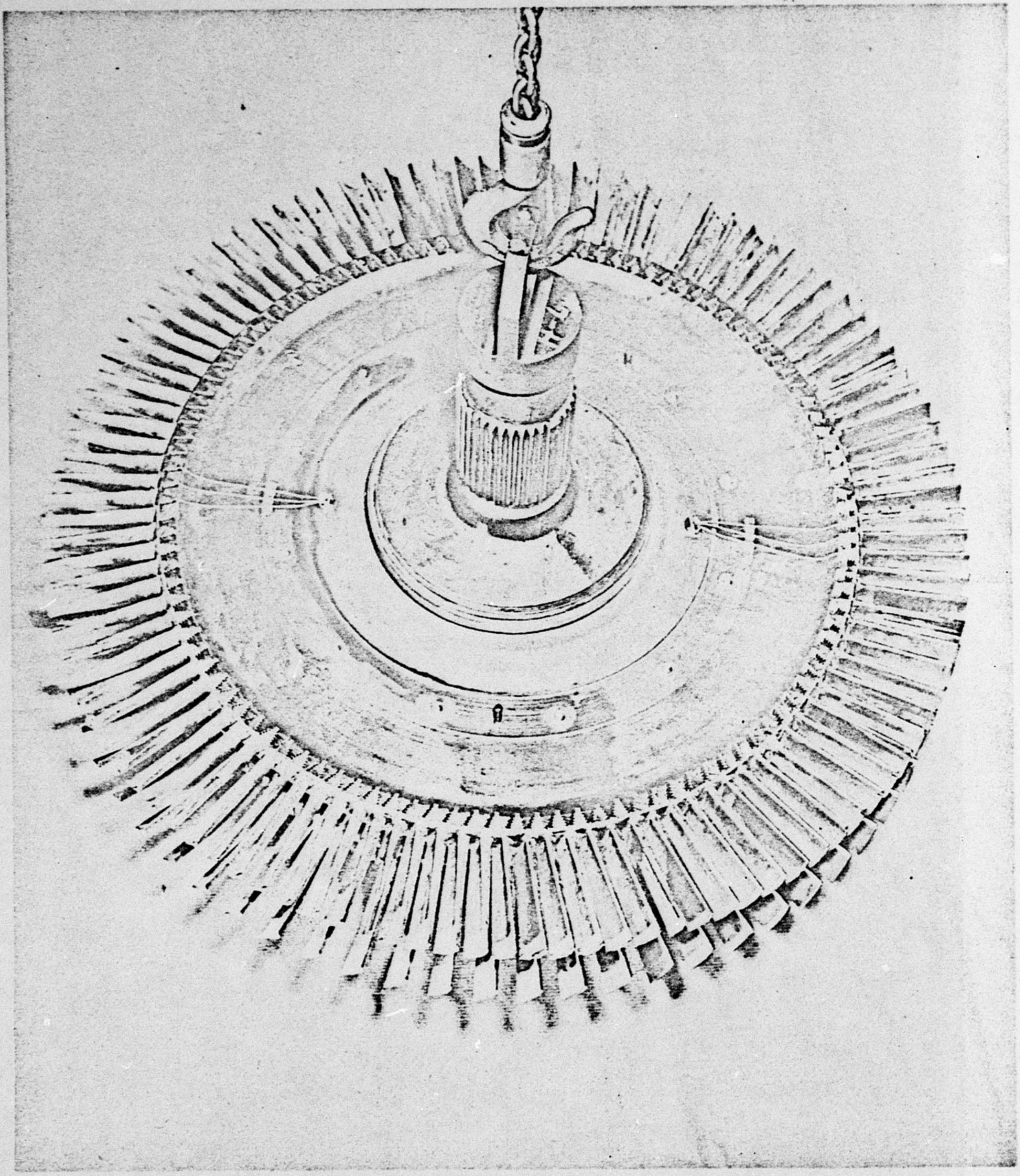
Overall View Of The Initially Tested Transpiration
Cooled Turbine Rotor On Which Torching Occurred

Figure 41



Close-Up Side View Of The Initially Tested Cooled Turbine Rotor
Showing Primary Airfoil Failures And Secondary Damage

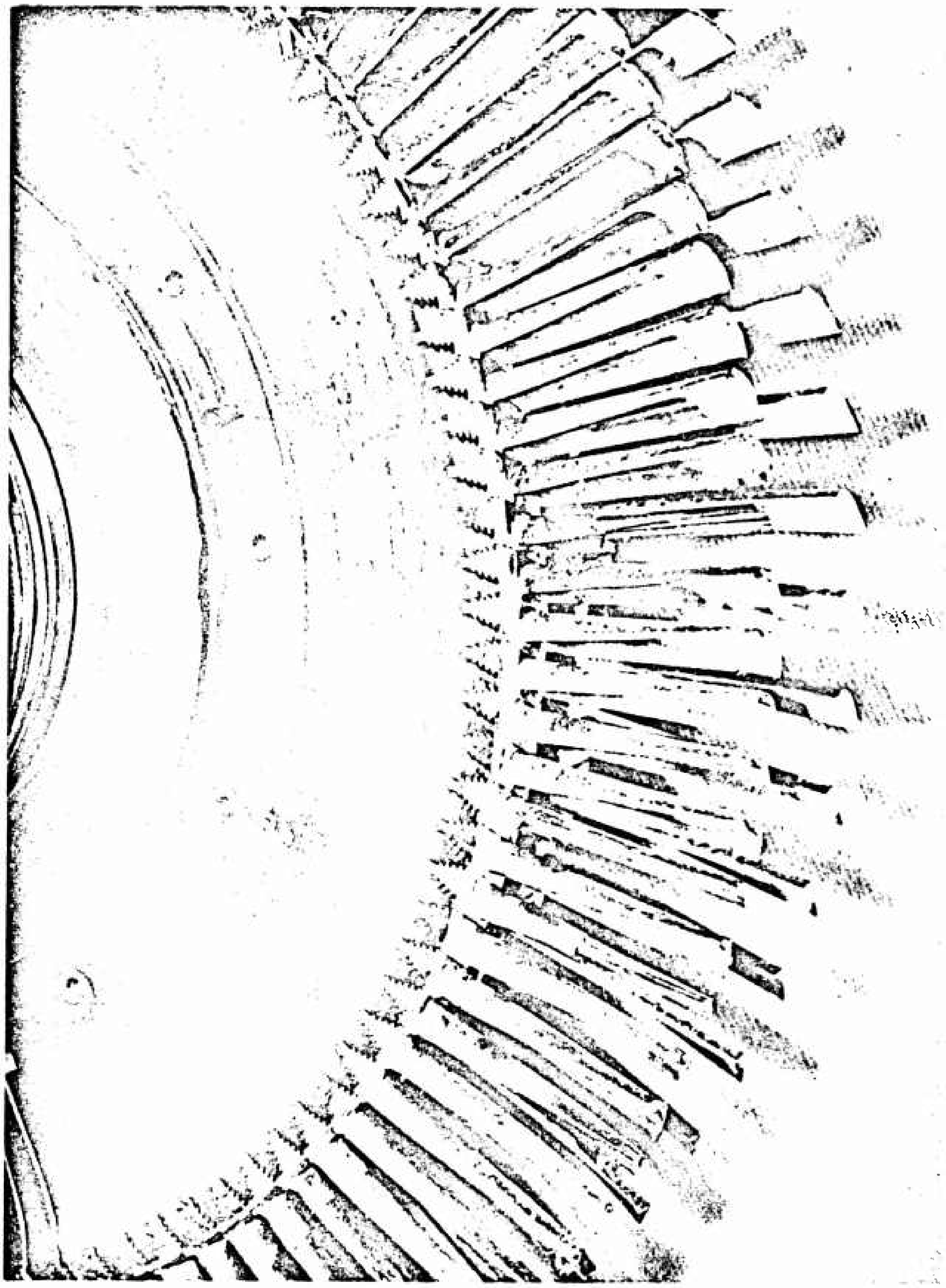
Figure 42



Overall View Of The Cooled Turbine Rotor After The
Second Test With Primary Failure Of Two Airfoils

Figure 43

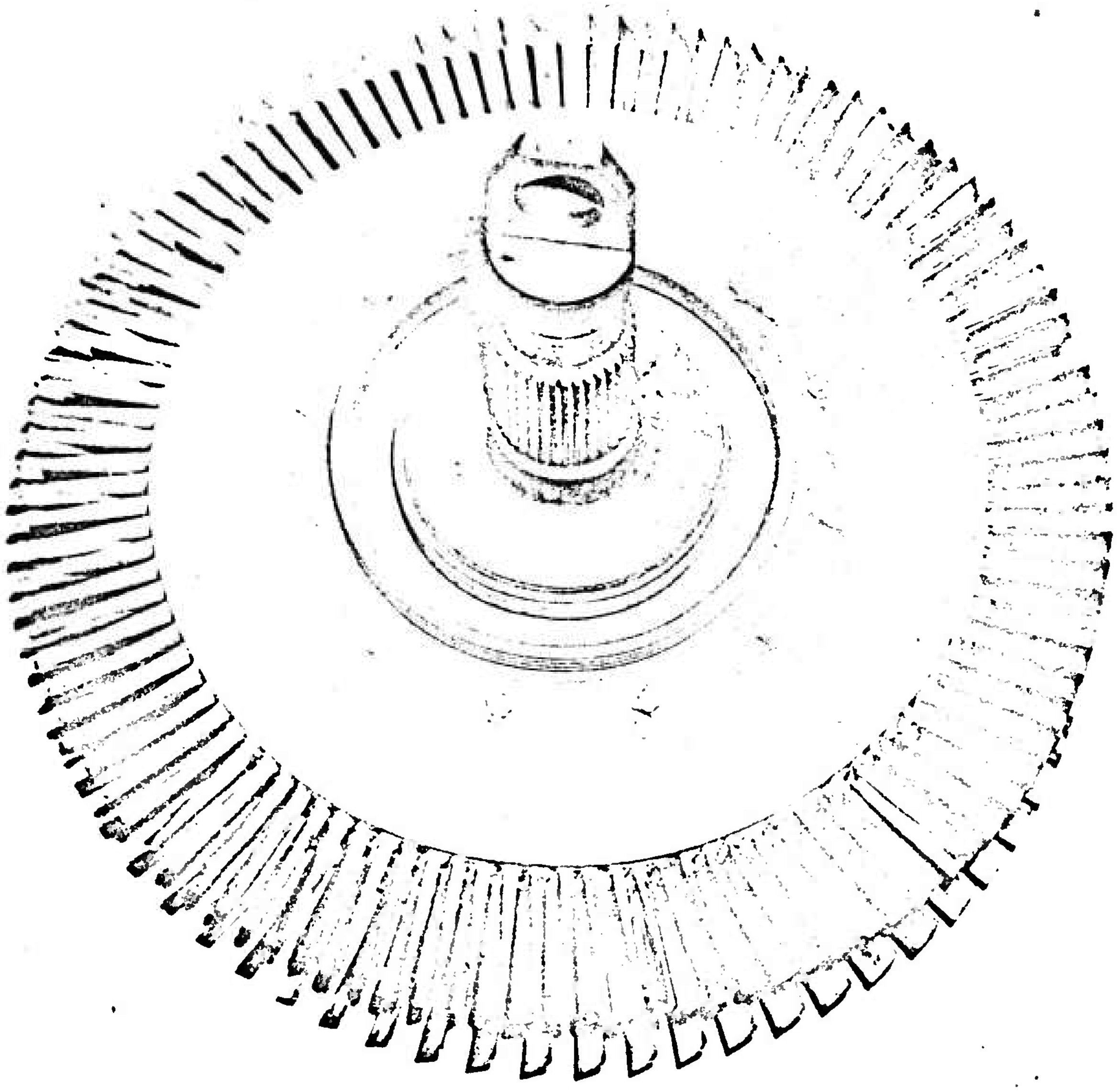
[redacted]



Close-Up Of The Cooled Turbine Rotor After The Second Test Showing The Secondary Damage Resulting From The Two Primary Airfoil Failures

Figure 44

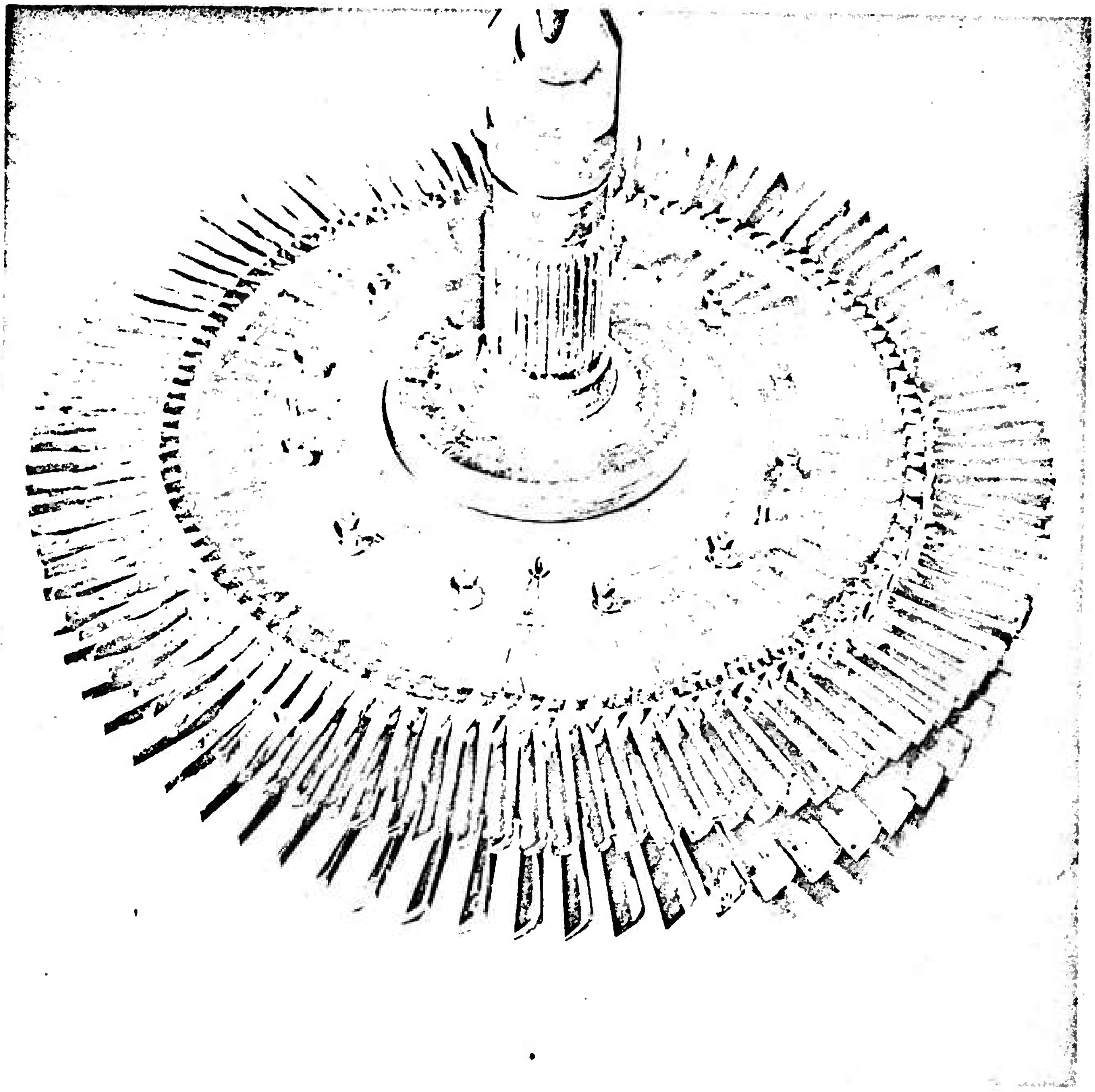
[redacted]



Overall View Of The Cooled Turbine Rotor After Twenty Minutes
Of Running At 1800°F (+) Showing The Secondary Damage
Resulting From The Primary Failure Of One Airfoil

Figure 45

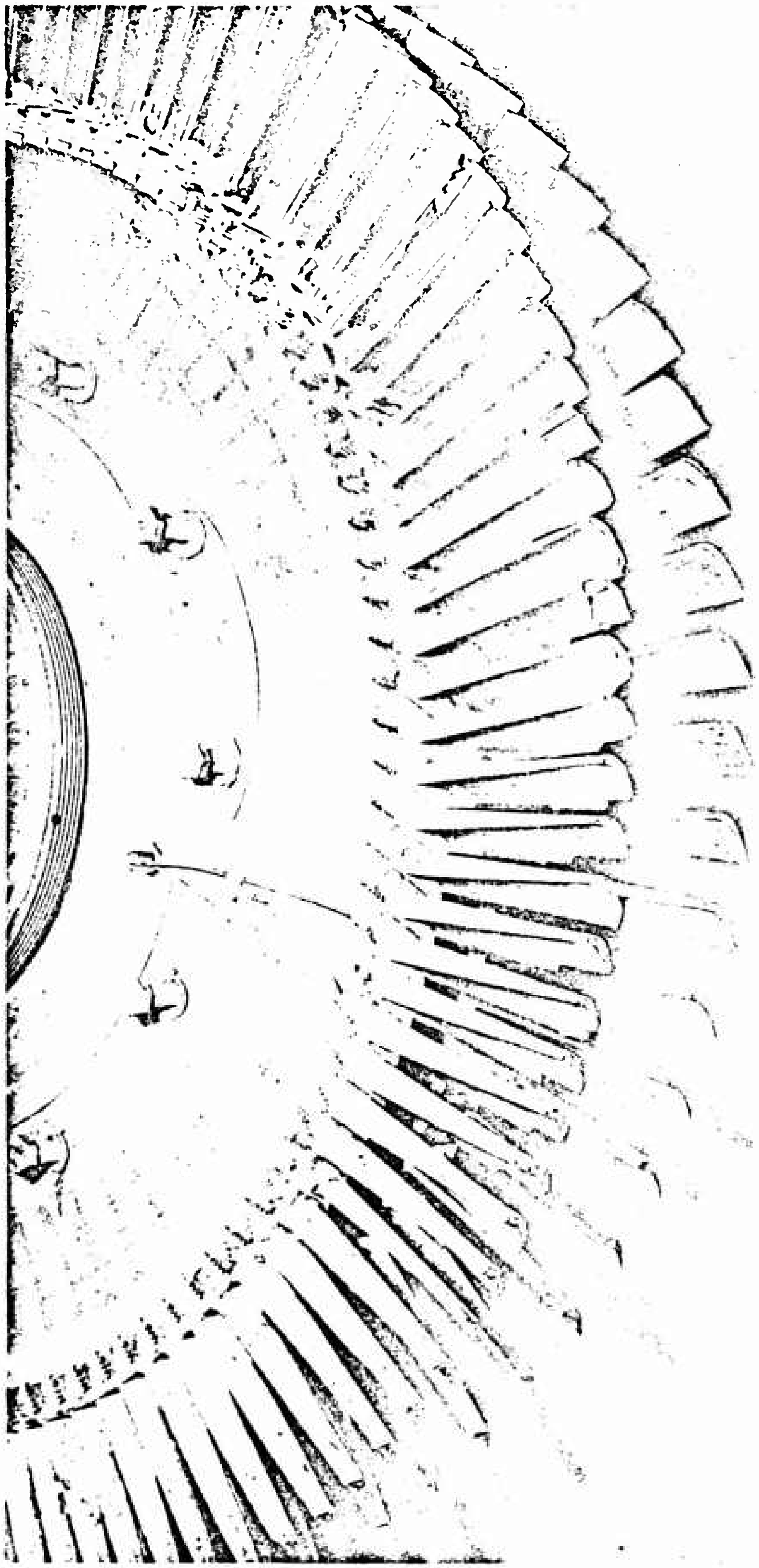
[redacted]



**Overall View Of The Cooled Turbine Rotor After Two Hours Of Running At 1800°F (+)** **Figure 46**

[redacted]

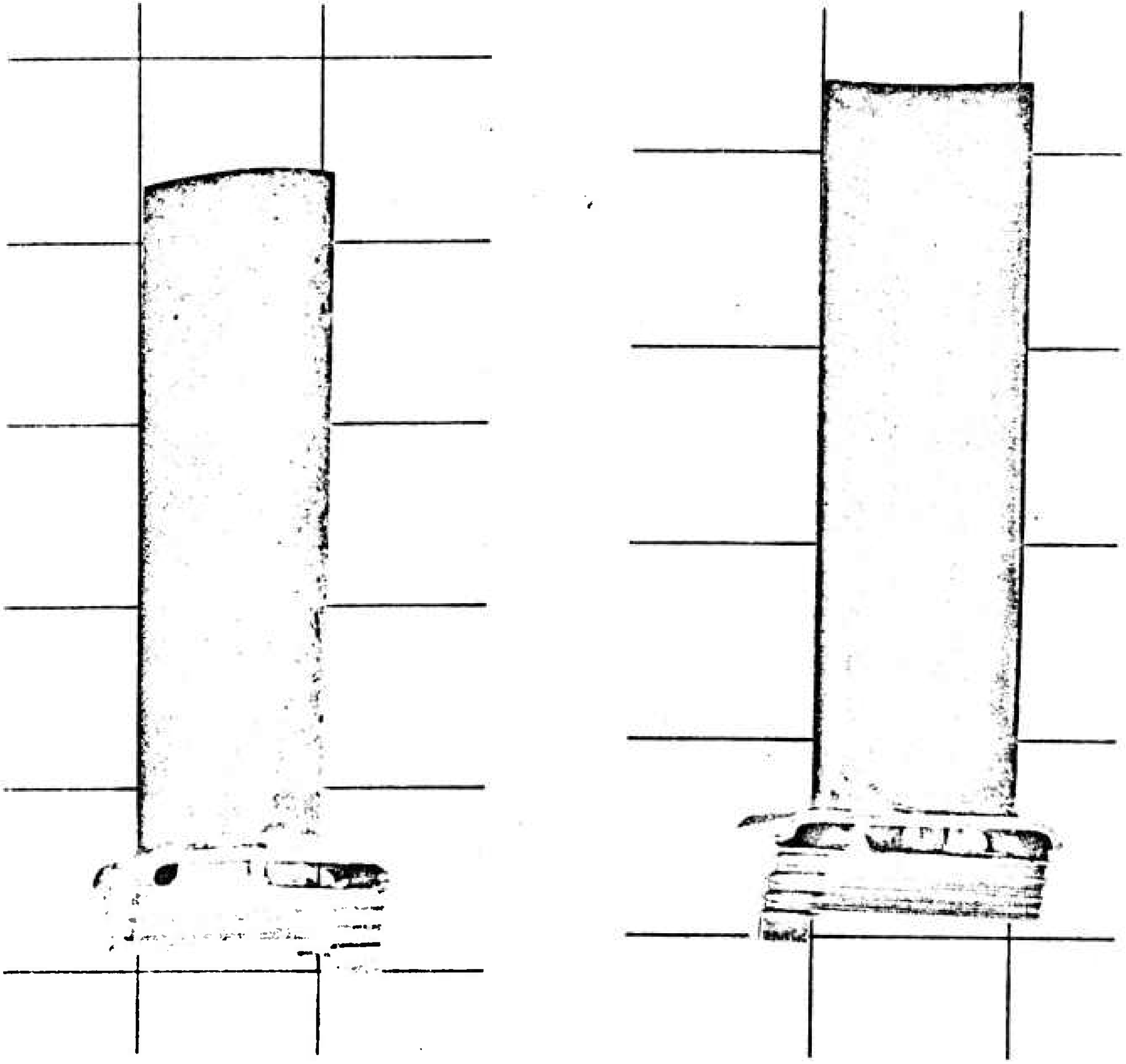
[redacted]



Close-Up Of The Cooled Turbine Rotor After Two Hours Of Running At 1800°F (±) Showing The Secondary Damage Resulting From Primary Failure Of One Airfoil

Figure 47

[redacted]

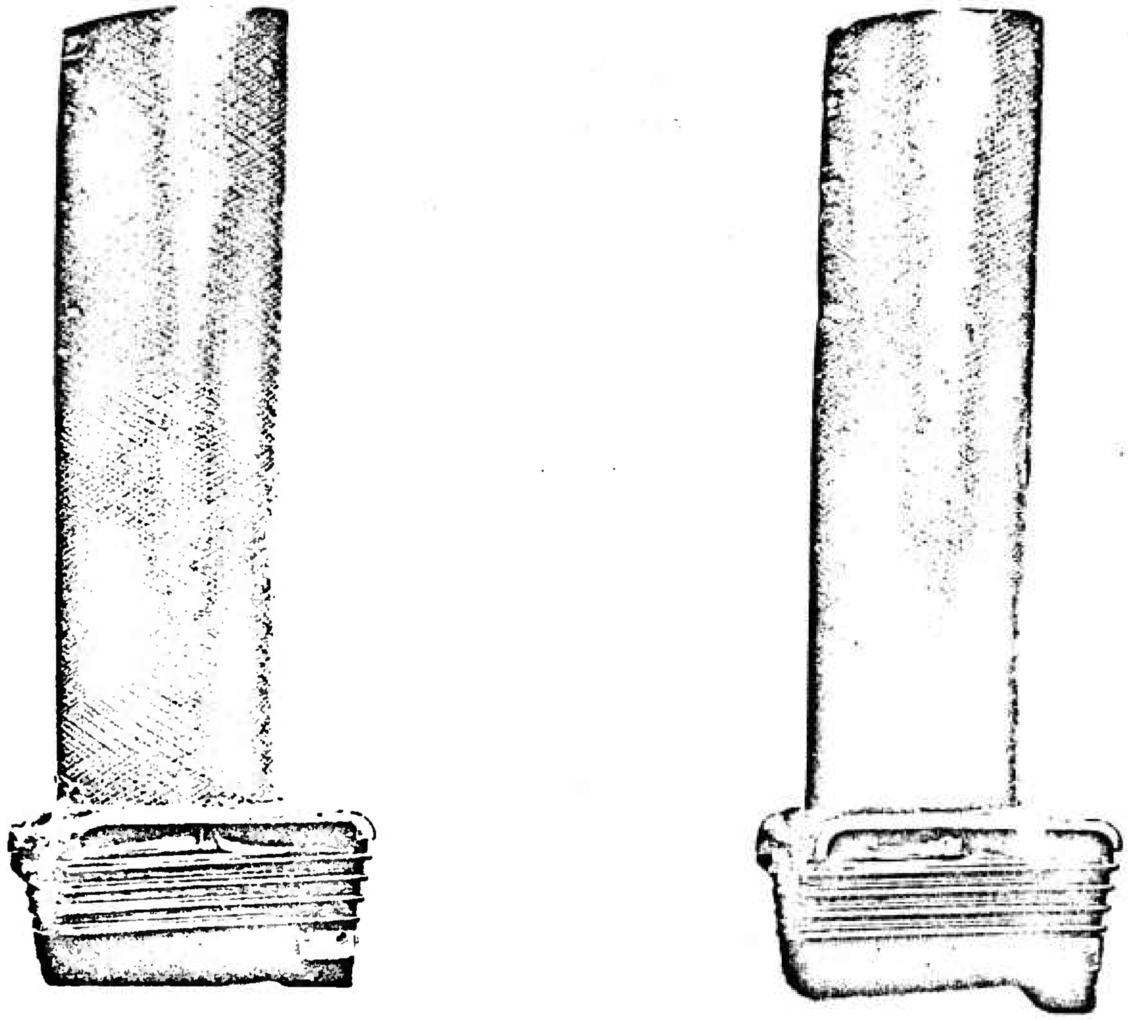


Convex

Concave

Figure 47a

Photograph of a turbine blade which has been engine tested



Blade No. 75

Blade No. 85

Figure 47b

Photograph of engine run turbine blades Nos. 75 and 85. The blades are still in good condition in spite of secondary damage on the trailing edge.

Flame Tubes - Carbon Build-Up - 3 O'clock Position Figure 48

Flame Tubes - Carbon Build-Up - 11 O'clock Position Figure 49