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FINAL ANALYSIS & DESIGN OF A THERMAL PROTECTION SYSTEM FOR 8-FOOT HTST COMBUSTOR

TASK II

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FOREWORD

This report was prepared by the Power Systems Division of Curtiss-Wright, Wood-Ridge, New Jersey, to describe the Task II work conducted under Contract NAS 1-11714 Modification 2 for the final analysis and design of a thermal protection system for the high pressure combustor of the NASA Langley Research Center's 8-foot High Temperature Structures Tunnel. The program guidance provided by Messrs. J. Karns, E. Bruce, and Dr. M. Anderson of the NASA Langley Research Center is gratefully acknowledged.

Major technical contributions toward the program were provided by Messrs. R. Cole, A. De Feo and M. Kulina of Curtiss-Wright, and their efforts are gratefully acknowledged.

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FINAL ANALYSIS & DESIGN OF A THERMAL PROTECTION SYSTEM FOR 8-FOOT HTST COMBUSTOR

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SUMMARY

The cylindrical shell combustor with T-bar supports in the 8-foot HTST at the NASA-Langley Research Center encountered vibratory fatigue cracking over a period of 50-250 tunnel tests within a limited range of the required operating envelope. During the Task I program, a preliminary design study provided several suitable thermal protection system designs for the combustor, one of which was a two-pass regenerative type air-cooled omega-shaped segment liner.

During the Task II program, a final design layout of the omega segment liner was prepared and analyzed for steady-state and transient conditions. The design of a support system for the fuel spray bar assembly was also included. Detail drawings suitable for fabrication purposes were also prepared.

Liner design problems defined during the Task I preliminary study included

(a) the ingress of gas into the attachment bulb section of the omega segment,

(b) the large thermal gradient along the leg of the omega bulb attachment section and

(c) the local peak metal temperature at the radius between the liner ID and the leg of the bulb attachment.

These were resolved during the final design task.

Analyses of the final design of the omega segment liner indicated that all design goals were met and the design provided the capability of operating over the required test envelope with a life expectancy substantially above the goal of 1500 cycles.

The cost for fabrication of the liner assembly was estimated. The method of liner installation, inspection requirements, repair procedures and instrumentation required for monitoring liner performance and safety during tunnel runs were established. A visual-aid model of a 60° sector describing the liner design was fabricated.

INTRODUCTION

The Langley Research Center's 8-foot High Temperature Structures Tunnel (HTST) is a facility used for testing large scale structural models and thermal protection systems applicable to hypersonic vehicles. The general arrangement of the HTST facility is presented in Appendix Figure A-1. The facility is a large hypersonic blow-down tunnel which uses methane-air products of combustion as a test medium.

The energy level of the test gas is provided by burning methane gas in air in a high pressure combustor. The resulting combustion gases are expended through an axisymmetric nozzle with an exit diameter of 8-feet into an open jet test section. Downstream of the test section the flow passes through a supersonic diffuser and is pumped by a single stage air ejector through a subsonic diffuser to the atmosphere.

The facility was designed to operate within a wide operating envelope with gas temperatures between 2500° and 4000°R and pressures up to 4000 psia. A description of the combustor operating procedure is presented in Reference 1.

The general arrangement of this high pressure combustor system is presented in Appendix Figure A-2. The outer housing or barrel pressure vessel is protected from the combustion gases by a double-pass regenerative-type thermal protection system. High pressure air supplied from a bank of 6000 psi bottles adjacent to the facility enters the outer housing through two inlet flanges located at the upstream station. The air flows downstream in the annular passage created by the outer housing and the cylindrical combustor liner support. The air returns upstream in the annular passage created by the cylindrical combustor liner support and the combustor liner. The air enters the combustor liner ID and flows downstream for the combustion process.

Fuel is introduced under separate control through two large axial tubes supplying a network of concentric fuel distributor rings in a plane about 11.33 feet from the upstream closure end of the outer housing.

A separate system of air film cooling and convective water cooling is used to cool the nozzle throat and the exhaust nozzle and is also shown in Figure A-2.

For the T-bar liner design arrangement, described in Appendix A, the service life of the combustor liner has been less than 250 tunnel runs. Axial cracking has occurred in the inner liner about 1 to 5 feet downstream of the fuel distributor rings. Where the crack progressed through the entire wall thickness, a burnout of the liner has occurred. The liner failures have required one of the following actions:

(a) crack repair by welding,

(b) removal of the burnout sections and replacement by welding of an insert patch, or

(c) replacement of the entire liner.

Furthermore, the liner failures and its design limitations have prevented operation of the HTST facility over the full operating envelope and especially near the 4000°R gas condition.

As a result of this problem, NASA formulated a multi-phase program directed toward the redesign of the thermal protection system which would provide a life expectancy of at least 1500 tunnel runs and expand the operating envelope capability. Task I program incorporated the analytical studies and preliminary designs of alternate thermal protection systems to the extent that design feasibility was established, fabrication and installation requirements were evaluated and preliminary manufacturing costs were estimated.

A program was also conducted which analyzed the failed combustor liner hardware of the T-bar design and determined that the mechanism of failure was vibratory fatigue. A vibration damper system using wave springs located axially between the liner T-bars and the liner support was designed as an intermediate solution only to extend the fatigue life of the current liner design but within an operating envelope limited by

(a) 1800°F metal temperature at gas conditions to 4000°R in the low pressure region, and

(b) buckling instability at the cold flow high pressure conditions.

Several new liner designs evaluated for feasibility to operate over the complete operating envelope included:

(a) Regenerative air-cooled ring-stiffened liner to resist a buckling failure mode at cold flow high pressure conditions

(b) Regenerative air-cooled omega-shaped segment liner similar to turbojet afterburner designs

(c) Water-cooled nickel shell liner with circumferential water grooves

(d) Water-cooled nickel tube liner

Based on these studies, the two-pass regenerative type air-cooled omega-shaped segment liner and the water-cooled nickel shell liner were considered most suitable for the 8-foot HTST facility and capable of operating over the required operating envelope. The air-cooled omega liner, which was estimated to cost less than half of the water-cooled liner plus auxiliary equipment, was selected for the Task II program directed at final analysis and design.

This report presents the results of the Task II program. The program's major objectives are summarized as follows:

(a) Prepare a final layout design of a double-pass regenerative air-cooled combustor liner

(b) Prepare detail drawings and bill of materials necessary for fabrication purposes

(c) Perform thermal and stress analyses of the design for tunnel transient and steady-state operating conditions and determine cyclic life expectancy

(d) Establish methods for handling, installation, preventative maintenance, inspection and repair

(e) Establish instrumentation for monitoring liner performance and safety and recommend operating envelope limits

(f) Prepare cost estimate for fabricating the liner

(g) Fabricate a visual-aid model of a representative section of the liner design

DEFINITION OF SYMBOLS

A	area, sq. in.
D_h	hydraulic diameter, ft
E	modulus of elasticity, psi
Gr	Grashof Number
h	heat transfer coefficient, Btu/hr-sq ft-°F
k	thermal conductivity, Btu/hr-ft-°F
p	pressure, psia
Pr	Prandtl Number
Re	Reynolds Number
S	stress, psi
T	temperature, °F or °R
y	distance from neutral axis, in
Δ	difference in or change in
α	coefficient of thermal expansion, in/in-°F

Subscripts:

a	air-side
c	convective

Superscript:

1	Free-convection

FINAL DESIGN

Design Requirements

The final design of a combustor liner for the HTST facility was directed toward satisfying the following key requiredments:

(a) Gas temperature = 4000°R maximum
2500°R minimum

(b) Gas pressure = 4000 psia maximum
600 psia minimum

(c) Service life = 1500 tunnel tests

(d) No erosion which creates foreign object damage to the test model

(e) Air pressure (cold) = 2000 psia maximum

The physical envelope limitations were as follows:

(a) Maximum OD = 42 inches

(b) Minimum ID = 36 inches

(c) Maximum length = 23 feet

Recognizing that the failures of the T-bar combustor have occurred within a restricted operating envelope of the facility, it was considered necessary to evaluate the new design for other potential failure modes which may occur at the higher operating gas conditions. Consequently, the following design criteria were considered:

(a) Minimum low cycle fatigue life = 1500 cycles as predicted by Manson's method

(b) Maximum liner temperature = 1850°F

(c) Critical buckling pressure = 2.5 x actual pressure load for hot flow operation

(d) Critical buckling pressure = 1.5 x actual pressure load for cold flow operation

(e) Minimum vibratory fatigue life = 1500 tunnel tests (approximately 75 hours)

(f) Pressure stress ≤ 80% of yield strength

The maximum liner temperature limit was based on the oxidation characteristics that severely reduce material properties, and the limited properties data available in the high temperature range.

The low cycle fatigue criterion was necessary since a significant increase in thermal gradient across the liner cross-section and wall thickness at the high gas temperature conditions was expected.

The critical buckling criteria were established as a result of the large pressure load imposed on the liner resulting from the high gas pressure differential between the cooling air passage on the outer surface of the liner and the main gas chamber inside the liner. The higher factor for hot flow conditions was selected for conservatism at elevated metal temperatures.

The thermal analysis performed during Task I also indicated several potential problem areas:

(a) The ingress of hot gas into the attachment bulb section of the omega segment

(b) The large thermal gradient along the leg of the bulb attachment section of the liner segment

(c) The local peak temperature at the radius between the liner ID and the leg of the bulb attachment

These problems appeared to have the following practical solutions which were studied during the final design and discussed in the following section:

(a) Provide a tubular seal spline in the attachment bulb section to limit ingress of hot gas flow

(b) Extend the length of the segment retaining clamp to increase the coverage or insulation of the segment leg

(c) Increase the radius between the liner ID and the leg of the bulb attachment to increase the ratio of air-side to gas-side local surface area or revise the cooling air passage geometry to increase the cooling flow velocity and heat transfer coefficients.

Liner General Arrangement

A type of liner design which is used for high temperature combustors and for turbojet afterburners consists of an assembly of axially parallel segments formed to an omega-shaped cross-section such that the liner supports are integral with the inner wall. Attachment of the supports to the housing is by an axial channel-shaped clamp. Advantages of this design with regard to heat transfer are that thinner material can be used without structural problems and the coolant stream is not seriously obstructed by liner supports or brackets since there is no requirement to provide clearance for large radial growth.

Figure 1 presents the final design layout (LS 34799 Sheet 1) of the omega-shaped segment liner basic configuration using a two-pass regenerative air cooling arrangement.

The omega shape for the liner segments was selected since it has been successful in afterburner liners in aircraft turbojet engines and is known to successfully accommodate circumferential thermal growth. A cold gap of about 0.60 inch between segments is provided for circumferential expansion. The gap at operating conditions is less than 0.018 inches.

The liner consists of 36 omega-shaped segments which are formed from 14 feet long and 0.125 inch thick sheet stock of Hastelloy X material. The omega segments are longitudinally welded to a tubular seal spline located between segments. The 36 splines are formed from Hastelloy X tubes. The omega segments are clamped to the liner support by 36 channel-shaped rails. The rails are bolted to the liner support at 4 inch intervals along the length of the liner. Self-locking two lug floating anchor nuts are attached to the inner side of the rails to receive the bolts. The liner support is a 0.75 inch thick AISI 304L cylinder about 23 feet long which is bolted to an internal flange on the existing pressure vessel or barrel housing at the upstream end.

The omega segments are capped by an end seal plate located at the downstream end. The end plate is contoured to receive the 36 omega-shaped segments and is welded completely around the sgement periphery to form a leak-proof configuration. Sandwiched between the end plate and the aft-face of the liner support is an 0.008 inch Raybestos Manhattan A-56 gasket capable of operating up to 900°F. The end plate, gasket, and liner support are retained by a ring which is bolted to the aft-face of the liner support with 36 bolts.

The liner support at the downstream end incorporates a groove containing a Viton-A O-ring seal which engages with an L-shaped ring extension welded to the forward face of the nozzle approach section. This seal prevents leakage of cooling air from the passage between the liner support and the barrel housing. The L-shaped ring incorporates a generous chamfer and a cylindrical pilot for receiving the aft-end of the liner support.

The air flow path for the omega segment liner arrangement is similar to the original design. Air enters the annular passage formed between the barrel housing and the liner support near the upstream end closure. Air flows downstream and enters the annular passage formed between the liner support and the liner through radial holes in the liner support at a location several inches upstream of the nozzle approach section. The air then turns 180° and flows upstream, discharging into the inlet area to the combustor at approximately 3 feet upstream of the fuel spray bars.

The combustor liner assembly is bolted at the downstream flange and along the length of the liner support ID.

The liner and liner support assembly is attached to the front flange of the barrel housing and centered by the seal ring at the front face of the nozzle approach section. An axial gap of 0.450 inches is also provided between the liner support aft-face and the nozzle section front face to accommodate rearward thermal expansion of the liner support. The thermal expansion of the liner is forward for approximately 1.2 inches and the liner support is rearward for 0.17 inches.

The barrel housing is reworked to remove the channels on the ID of the housing which were used for supporting the previous liner T-bars.

Gas thermocouple self-aligning bosses are incorporated in four of the omega segments near the downstream end.

The fuel spray bar and supply tube assembly is mounted as in the original design to the housing upstream end closure; however, a four-spoke support has been attached to the two fuel supply tubes to change the natural frequency of the system and to prevent pounding of the omega segments by the spray bar radial struts. The four spokes engage into longitudinal rails which are bolted to the ID of the liner support and extend about 40 inches upstream.

The various elements of the omega segment liner design are described in the following paragraphs.

Mechanical Design

<u>Liner Support</u>.- The liner support is a 0.75 inch thick cylindrical section, approximately 23 feet long with a flange at the upstream end. The support is made of AISI 304L material, which has extremely good corrosion resistance.

The liner support is bolted to the existing ring flange (ALCO Drawing No. 47241-1-14, Part No. 14-9) which is an integral part of the high pressure combustor barrel. To facilitate assembly of the liner support in the barrel, the existing flange is reworked by providing an entrance and exit chamfer to protect the O-ring seal located on the outside diameter of the liner support at the downstream end and four slots to provide clearance for four installation skids. These skids are located 4.20 inches upstream of the aft-face on the OD and are 1.0 inch thick with a 4.0 inch radius. The skids are positioned on the vertical and horizontal centerline of the liner support.

The outside diameter of the liner support is machined at the downstream end for the 0.275 inch diameter Viton-A O-ring. At the downstream end of the cylindrical liner support, on a 40.06-inch bolt circle, are 36 1/4-inch 24 UNF tapped holes which are used to attach the liner retainer ring.

Located 3.70 inches from the downstream face of the liner are 36 equally spaced air supply holes, 2.0 inches in diameter, drilled at an angle of 32° so that the cooling air is directed toward the end plate of the liner assembly. Four of these holes located 15° on either side of the vertical centerline and 15° below the horizontal centerline are modified by blending with the clearance holes for the water-cooled gas thermocouple probes.

The liner support at 36 equally spaced locations around the circumference has provisions every four inches in a longitudinal direction for mounting the retaining rail assembly which positions the liner assembly. A total of 44 holes in each of the 36 rows is required. The liner support has provisions for mounting four equally spaced channels, each 42 inches long, at a distance of 4 feet from the liner support mounting flange to the upstream edge of the rails. The rails are dowelled and attached to the inside diameter of the liner support by ten 3/8-24 UNF bolts per rail. These rails radially locate the fuel spray bar assembly concentric with the liner assembly.

A transient and steady-state thermal analysis of the liner support was conducted to establish axial growth. After 120 seconds at 600 psia and 4000°R gas conditions,the liner support has a downstream growth of 0.170 inches. (For unlimited steady-state operation at gas conditions of 600 psia and 4000°R, the liner support would move downstream for a length of 0.403 inches). The liner support is designed to accommodate an axial growth of 0.450 inches and the thermocouple probe boss is designed to accommodate a growth of .400 $\pm$ .09 inches. The axial clearance between the end face of the liner support and the forward face of the exhaust nozzle approach section which is required to accommodate liner support thermal growth is established with an AISI 304 stainless steel shim sandwiched between the liner support to barrel housing mounting flanges.

Liner Assembly - The liner assembly consists of 36 omega segments, 36 oval tube splines, 36 spline caps, a one-piece omega segment end plate, four gas thermocouple probe bosses, and 36 shear pin blocks.

The liner assembly is a 360° welded structure consisting of 36 omega segments fillet welded for its entire length to 36 tube splines; the fillet welds are located on the outside diameter of the assembly, and there are two fillet welds per omega segment for a total of 72 longitudinal welds.

The downstream end of each tube spline is capped with a formed cup which is welded to the spline end face.

The downstream end of the 36 omega segments is capped with a full round annular end seal plate. The inside diameter of the plate is contoured to fit inside the omega segments and over the outside diameter of the tube splines. This plate is continuously welded to the omega segments and tube splines with a 100 percent penetration weld plus a fillet weld. The downstream face of the liner assembly is then machined within 0.06 inch of the rear surface of the end plate to minimize the uncooled projection of the omega segments.

Located just upstream of the omega segment end plate, in the plane of the centerline of each oval tube spline at the OD of the liner are 36 Hastelloy X T-shaped blocks 4.43 inches long by 1.06 inches wide by 0.375 inches overall height with 0.125 inch thick flange and 0.875 inch width body. A nut plate containing two 7/16-20 threads is plug welded to the 0.875 inch width body. These blocks are full penetration welded longitudinally to the omega segment outer flange and fillet welded circumferentially to the tube spline. The block and liner support are reamed to accept two (7/16 inch - 20 thread) shear pin-bolts with 0.5625 inch diameter shear body. One row of 36 pin-bolts anchor the liner longitudinally so that the thermal growth of the liner assembly in relation to the liner support occurs upstream of the shear bolt plane in order to minimize loads on the omega segment end plate weld joint. The second (upstream) row of 36 pin bolts provide a restoring moment to resist the thermal moment developed in the unsupported length of the liner beyond the end of the retaining rail.

Hastelloy X was selected for this liner application because it has excellent resistance to oxidizing and reducing atmospheres. Hastelloy X is a nickel-base alloy with a low strategic alloy content. It possesses exceptional strength and oxidation resistance up to 2200°F.

The alloy has excellent forming and welding characteristics. The alloy can be forged and, because of its good ductility, can be cold-worked. It can be welded by both manual and automatic welding methods including metallic arc, inert-gas shielded arc, submerged-melt, and ISGMA methods. For welding of the liner assembly the TIG process is specified with MIG method listed as an alternate. For heavy sections, the MIG process can be used more expeditiously. Welding Hastelloy X to components made of stainless steel is performed using 308L weld wire.

The surfaces in contact with the rail and liner support are spray coated with a solid film lubricant MIL-L-23398.

Rail Assembly. - The rail assembly consists of a U-shape rail, an end cap, 0.25 inch self-locking floating two-lug anchor nuts and an aerodynamic shield to cover the nut assemblies. The rail assembly is formed from 0.188 inch Inconel X (AMS 5542) sheet.

The rail is contoured to contact the omega segment below the two rolled edges. Located at 4-inch intervals is a 0.250-28 UNJF self-locking floating nut assembly which is projection welded to the rail. The total number of nut assemblies per rail is 44. The downstream face of the rail is cut at a 32° angle sloping downstream in order to match the cooling air supply holes in the liner support and provide directionality for the cooling air to the liner-to-end plate weld joint at the ID. The rail end face is also capped to provide a smooth surface for the cooling air. The rail is capped along its length with an arch-shaped aerodynamic shield to minimize the pressure drop in the cooling air flow passage and still be flexible to accommodate thermal deformations. The shield which is formed of 0.0312 inch Inconel X sheet is tack-welded to the rail along both edges. Both the rail and shield are slotted at 4-inch intervals to minimize thermal loads due to the radial temperature distribution in the rail cross-section.

The bolt hole clearance and the compatible thermal growths between the liner support and the rail insure that shear loads are not imposed on the bolts that attach the rail assembly. Upstream of the shear bolt plane the rail assembly and the omega segment liner has a 0.010 inch cold radial clearance.

The surfaces in contact with the liner and the ID of the liner support are coated with solid film lubricant MIL-L-23398.

Rework of Water Cooled Exhaust Nozzle. - The water cooled exhaust nozzle approach section (ALCO Drawing No. 47241-1-18, Part No. 18-1) is reworked by welding an L-shape liner support seal ring. This ring is machined from carbon steel (Type 1015-1025) and welded to the outside diameter of the upstream end of the exhaust nozzle approach section with a full penetration weld.

A 120° segment docking cone is provided at the lower half of the reworked water-cooled exhaust nozzle section. The cone segment is bolted to the liner support seal ring and contains three dowel pins for accurately locating the segment since the cone is bolted after the reworked nozzle approach section is installed in the barrel assembly. During installation of the liner and support assembly, the docking cone will receive the aft-section of the liner support and guide the assembly over the cone angle so that the O-ring in the liner support OD smoothly engages the seal ring.

Fuel Distribution Support System. - The fuel distributor spray bar support system consists of a four-spoke support machined from AISI 304 which is clamped to the two existing 4.5 inch OD fuel supply pipes. The spoked support consists of a center section which integrally incorporates two vertical spokes, a center hole to accommodate the combustor ignition pilot tube and the two inner halves of the horizontal spokes for the fuel supply pipes. Two individual struts machined from AISI 304 bolt to the inner half sections to form the remainder of the horizontal support spokes for the fuel spray bar system. The 0.001 to 0.003 inch interference or clamping fit with the supply tubes is obtained with shims between the inner and outer halves of the horizontal spokes.

Four rectangular rails, 2.80 inches wide and spaced 90° apart, are bolted and dowelled to the inside surface of the liner support. The rails which are machined from AISI 304 bar stock are grooved to receive the four square tangs on the ends of the spoked support. The thermal expansion takes place longitudinally along the rails. The 42-inch length of the rails was established so that the spokes engage the rails before the fuel spray bar enters the inside diameter of the liner assembly. This design arrangement permits installation of the fuel spray bar assembly after the liner and support assembly is installed in the barrel housing. The initial installation of this assembly is described on Drawing No. ES160912.

For this design, the bumper pad at the tip of each of the six radial struts supporting the fuel spray rings are machined to a diameter 0.060 inch larger than the outside diameter of the largest fuel spray ring but with clearance which is ample to assure that neither the fuel spray rings nor the pads contact the inside diameter of the liner since the deflection of the spray ring is 0.0198 inches.

Gas Thermocouple Installation Study. - A design study was conducted to accommodate the water-cooled gas thermocouple probes used in the present T-bar liner design. The existing thermocouple bosses in the barrel housing near the exhaust nozzle approach section are angularly located at 15°, 60°, 105°, 255°, 300° and 345°.

One approach for accommodating all six thermocouple probes involved the use of a water-cooled instrumentation cylinder attached to the nozzle approach section. This design was considered the most desirable installation arrangement since it eliminated the problem of accommodating the large difference in thermal expansion between the liner and the liner support. However, this separate double-walled cylindrical section would add a substantial cost element.

A design layout (LS 34799, Sheet 2) of a 2.625-inch axial length water-cooled section which is bolted to the upstream face of the existing nozzle approach section is presented in Figure 2. The construction and materials are identical to the present nozzle approach section. The inlet and outlet water passages are interconnected with the existing nozzle approach section and thus would require no new water feed lines.

The instrumentation cylinder consists of a liner made of Nickel A and a liner support or jacket made of SAE 1010-1025 material. The liner has circumferential ribs on the OD which form the water cooling channels between the liner and jacket. The liner overall wall thickness is 0.205 inches and the liner thickness at the full radius root of the ribs is 0.070 inches. The width of the coolant channel is 0.200 inches and the width of the rib is 0.120 inches.

The liner and jacket are circumferentially welded together at both ends. The liner and jacket assembly is attached by 0.75-10 UNC bolts to the nozzle approach section forward face through 24 existing tapped holes. The assembly is also centered on the nozzle approach section by hollow dowels at two locations using the existing 1.0 inch diameter counterbore of the tapped holes.

To accommodate six existing thermocouple probe locations, the liner and jacket incorporate a 1.25-inch diameter flanged boss made of Nickel A radially inserted into the assembly and welded to the ID of the liner and the OD of the jacket. The boss has a 0.600-inch diameter radial hole for insertion of the probe. To limit leakage around the probe, a Viton-A O-ring seal may be added to the design by providing a groove at the ID of the boss.

The liner of the instrumentation section also incorporates an L-shaped seal ring similar to the one shown attached to the front face of the nozzle approach section in the omega segment liner design (Figure 1). This ring engages the O-ring seal located near the aft-end of the liner support main-section.

The instrumentation section is supplied with cooling water through a transfer hole located at the top vertical centerline location of the jacket and matching the water inlet slot in the nozzle approach section. The cooling water in the instrumentation section returns to the nozzle approach section water outlet slot through a hole at the bottom vertical centerline location. The only rework to the nozzle approach section required for accommodating the instrumentation section is the drilling of the two water transfer holes in the front face. The 0.5 inch transfer holes are sealed by an O-ring in the mating face of the instrumentation section.

The advantages of this water-cooled instrumentation section is that all six existing thermocouple probes can be accommodated and that the thermocouple probes need not be removed when the combustor liner and support assembly is removed from the barrel housing.

If the instrumentation section is utilized, a design change is required to (a) decrease the length of the new liner and support assembly by a length equal to the instrumentation section, and (b) delete the thermocouple bosses provided in the liner assembly shown in Figure 1.

The following design arrangements for thermocouple probe installation without the addition of a water-cooled section were also considered:

1. Provide a 24-omega segment design which is compatible with the 15° angular spacing of the thermocouple bosses at 15° multiples. Six probes are accommodated.

2. Provide a 34-omega segment design with 20 segments based on an included angle of 10° with four segments straddling the top vertical centerline and 16 straddling the bottom vertical centerline; the remaining 14 segments are based on an included angle of 11.43° with seven on each side of the vertical centerline. Six probes are accommodated.

3. Provide a 36-omega segment design and circumferentially locate the center of four segments to align with four of the six probes. The probes at the 60° and 300° angular location would be omitted.

The 24-omega segment design includes a segment width of 4.71 inches as compared with 3.14 inches for the 36-segment design. To evaluate the efficacy of the 24-segment design, a preliminary stress analysis of the pressure loading was performed. The stress in the liner segment due to the pressure differential across the liner increases inversely with the number of segments. For the same pressure load, the maximum bending stress is a function of span length squared and span length in a circumferential direction is a direct function of the number of segments. Therefore, the pressure stress in the liner is increased by a factor of 2.25 when the number of segments is reduced from 36 to 24.

Preliminary calculations indicated that the maximum pressure drop across the omega segment liner for the 4000°R and 4000 psia gas condition is about 31 psi. For the maximum flow condition in the facility envelope, a peak pressure drop of 4.8 times the hot condition or 149 psi may occur. For the 24-segment configuration using 0.125 inch liner thickness, the maximum pressure stress would exceed 100,000 psi. Increasing the number of segments to 36 would reduce this stress to less than 50,000 psi.

A preliminary thermal analysis of the 24-omega segment design also indicated problems in that higher metal temperatures would occur in the wider segment. At the 4000°R and 600 psia gas condition, the temperature would increase approximately 37°F at the center of the omega segment and 32°F at the critical radius area where the highest temperature of the segment occurs. Consequently, the temperature level in the radius would approach 1900°F.

Based on this preliminary review, the 24-segment design was discarded.

The second method using 34 segments of two different widths (3.14 and 3.59 inches) provides all the advantages of the first method by locating each thermocouple at the center of each omega segment and in addition uses a reduced width segment. However, a disadvantage of this method is in the additional tooling cost involved in producing two different width segments.

The third method using 36 omega segments and accommodating only four of the six thermocouple probes was selected as a practical compromise.

The thermocouple probes to be utilized are symmetrically located about the vertical centerline at 15°, 105°, 255° and 345°.

The four thermocouple probe bosses as shown in Figure 1 are welded to the liner at the centerline of the omega segment with a full penetration weld. The probe boss has an oval shape with an opening to accommodate the water-cooled probe of 1.17 inches length, 0.770 inches wide and a 0.385 inch radius at the ends. The top face is rectangular in shape with two U-shaped guides, an oval groove for a spring loaded Stellite 25 face seal and a riveted shoulder pin stop. The rectangular seal plate has a 0.604 inch diameter hole with provisions for a Viton-A O-ring seal with a 0.551 inch ID and 0.07 inch cross-section. The seal plate also contains a partial conical ramp on the longitudinal centerline at the downstream section of the sealing hole and a slotted groove on the upstream end for engaging the shoulder pin stop. The seal plate is channeled in the U-shape guides of the probe boss. The spring loaded Stellite 25 seal contacts the bottom surface of the seal plate with a unit pressure of 27 psi. Regardless of the location of the seal plate, the probe can be radially inserted since the conical ramp cam will self-align the hole in the seal plate when installing the probe.

The material for the probe seal plate is aged AMS 5662 (bar). The shouldered pin on the thermocouple boss is made of annealed Inconel 600. The probe seal semi-circular wave spring material is Inconel 600 which is cold worked to attain the following mechanical properties: Ultimate tensile strength of 140,000 to 170,000 psi, 0.2% yield strength of 110,000 to 135,000 psi, and Rockwell C30-40.

<u>Summary of Weight</u>. - The following summarizes the weights of each of the major assemblies in the omega segment liner design:

Liner support (1 unit)	7519 lb.
Liner assembly (1 unit)	2510 lb.
Rail assembly (36 units)	834 lb.
L-shape seal ring (1 unit)	9 lb.
Fuel spray bar centering rail (4 units)	132 lb.
Total	11,004 lb.

Fuel spray bar spoked support assembly (1 unit) 64 lb.

Thermal Analysis

Method. - The thermal analysis of the omega segment liner performed during Task I revealed two problem areas to be solved in the final design. First, the ingress of hot gas into the bulb section between adjacent omega segments tended to bring the outer liner wall temperature to an undesirable level and therefore, a means was incorporated to prevent the ingress of hot gas. Second, the estimated maximum temperature of 1895°F in the omega segment for the gas conditions of 600 psia and 4000°R was considered excessive and changes were made to reduce the metal temperature to the 1800°F level.

The first problem was resolved by sealing the bulb section. Since the second problem is caused by the radius of curvature of the bend in the omega segment, this suggested that changing the radius may minimize the problem. However, studies made early in Task II showed limited improvement at gas conditions of 600 psia and 4000°R when the radius of curvature of the bend was increased from 0.375 to 0.50 inch. It was concluded that a more effective method of reducing the maximum liner temperature was to increase the air velocity in the omega segment cooling passage and hence, increase the air-side heat transfer coefficient.

The final design arrangement of the omega segment liner which emerged from the above thermal problems and other mechanical considerations is shown in Figure 1. An oval tube spline was inserted in the bulb section of the omega segment to reduce ingress of hot gas. Bolts and nut plates in the omega segment cooling passage were covered by a partition to provide a smooth cooling passage and to increase cooling air velocity.

The heat transfer analysis of the final design arrangements of the omega segment was based on a two-dimensional model shown in Figure 3. The model neglects heat flow in the longitudinal direction. This assumption is justified along the heated section except in the vicinity of the fuel spray bars where a rapid change in gas temperature occurs. NASA test data presented in Reference 2 indicate that the maximum liner wall temperature occurs at a distance of 1.5 ft. downstream of the fuel spray bars and the wall temperature decreases gradually beyond that station.

Barring unexpected non-uniformity in flow and thermal conditions in the circumferential direction, the temperature field may be assumed identical for each of the 36 omega segments and symmetrical about the axis of geometrical symmetry. It thus suffices to consider only one half the temperature field of an omega segment. Unlike the thermal model adopted in Task I, the new model now includes the 0.75 inch thick liner support and accounts for all the heat flow between the hot combustor gas and the cold air on the liner support.

The gas-side heat transfer coefficients evaluated during Task I on the basis of test data at various operating conditions were used in the final analysis. Determination of heat transfer coefficients for the omega segment cooling passage and the liner support was based on the well accepted empirical equation (Reference 3)

$$\frac{h_{ca_{D_h}}}{k} = 0.023\,(Re_{D_h})^{0.8}\,(Pr)^{0.4}$$

using a hydraulic diameter appropriate to the passage of interest. Preliminary estimation of Grashof Number predicted that free convection will occur in cavities 26, 27, 28 and 29 (Figure 3). The following correlations taken from Reference 4 were used to evaluate the heat transfer coefficient for free convection:

$$\frac{h^1 D_h}{k} = 0.21 \, (Gr_{D_h} Pr)^{1/4} \qquad \text{for} \qquad 10^4 < Gr_{D_h} < 3.2 \times 10^5$$

$$\frac{h^1 D_h}{k} = 0.075 \, (Gr_{D_h} Pr)^{1/3} \qquad \text{for} \qquad 3.2 \times 10^5 < Gr_{D_h} < 10^7$$

Air temperatures in these cavities were obtained by imposing the condition that at steady-state the net heat flow to each cavity is zero. Data presented in Reference 5 on thermal contact resistance were applied to interfaces between contiguous parts.

Omega Segment. - The temperature distribution in the omega segment and the liner support at a location 1.5 feet downstream of the fuel spray bars was analyzed for two extreme operating conditions, namely, 4000°R gas temperature at pressures of 600 and 4000 psia. This particular axial location was chosen for analysis because it is the location of maximum temperature measured on the T-bar liner. The pressure levels of 600 and 4000 psia represent the lower and upper limits, respectively, of the operating range at the maximum gas temperature of 4000°R. The low air flow rate at 600 psia is expected to result in a maximum liner temperature level while the high heat flow rate at 4000 psia will give a maximum temperature gradient across the liner thickness.

The temperature distribution obtained for the 600 psia and 4000°R gas conditions after 120 seconds of operation is shown in Figure 4. The maximum local liner wall temperature of 1844°F is slightly higher than the 1800°F level desired. However, stress analysis indicated that this temperature is acceptable. The temperatures of the omega segment are fairly steady after 40 seconds of operation. The liner support temperatures are still rising even after 120 seconds. Since the operating envelope for the combustor shows an operating time capability for the facility of 90 seconds at this condition, the liner support is not expected to exceed the temperatures shown in Figure 4.

The liner wall temperature distribution for the 4000 psia and 4000°R gas conditions after 12 seconds of operation is shown in Figure 5. For this case the liner temperatures are essentially steady within 8 seconds of operation. As anticipated, the temperature differential across the liner thickness is much greater, although the temperature level is much lower, than at the 600 psia condition. The operating envelope for the combustor shows an operating time capability of about 12 seconds for the gas conditions of 4000 psia and 4000°R.

To indicate the longitudinal variation of liner temperature, the temperature distributions at the downstream end where the liner cooling air passage begins are presented in Figures 6 and 7 for the two critical operating conditions, namely, 4000°R gas temperature at pressures of 600 psia and 4000 psia. Due to the low air temperature at this location, the omega segment temperature is about 200°F lower and the liner support is about 50° lower than the corresponding temperatures at the hot-end for the 600 psia gas condition. At the 4000 psia condition, there is no change in liner support temperatures but the temperature level of the omega liner at this location is about 90°F lower and the temperature differential across the liner thickness is about 20°F greater than at the hot-end.

Further downstream, at a location between where the rail assembly ends and the omega segment end plate begins, the part of the omega segment which was covered by the rail is now directly exposed to the cooling air. Therefore, the temperature distributions in the omega segment are expected to be slightly different from those shown in Figures 6 and 7. Figures 8 and 9 present the temperature distributions at this downstream section, corresponding to the gas conditions of Figures 6 and 7, respectively.

The axial temperature distribution of the liner in the vicinity of the fuel spray bar is of special concern because the large temperature gradient brought about by the sudden change in gas temperature may cause excessive thermal stress in this region. Preliminary study indicated that the liner temperature gradient is extremely sensitive to the axial variation of gas temperature. A gas temperature variation of the form shown in Figure 10 was assumed, resulting in a mean metal temperature distribution also shown in this figure.

That the assumed gas temperature distribution is a realistic one was confirmed by a comparison shown in Figure 11 of the predicted metal temperature distribution with that measured by NASA. The forms are similar but the levels are different since the measured data were for the gas conditions of 603 psia and 3300°R while the prediction was made for the 600 psia and 4000°R conditions. The estimated axial temperature distribution for gas conditions of 600 psia and 4000°R is shown in Figure 12 at a time of 120 seconds after light-off.

Figure 10 indicates that the maximum axial temperature gradient of the liner occurs at a position 2.5 inches downstream of the fuel spray bar. At this location the gas temperature is 1700°F and the mean mid-section temperature of the liner is 900°F. Using this mean temperature at the mid-section, the temperature distribution throughout the cross-section of the omega segment was estimated and is presented in Figure 13.

Transient temperature history at four selected locations of the omega segment liner are presented in Figure 14 for the gas conditions of 600 psia and 4000°R. Location "a" represents the hottest point of the omega section and location "b" is a point on the cold-side opposite location "a". Location "d" is at the edge of the omega segment where the temperature is the lowest, and location "c" is midway between locations "a" and "d". At any instant, the temperature difference between locations "a" and "b" may be considered the maximum temperature gradient across the thickness of the omega segment, while the temperature difference between "a" and "d" represents the maximum radial gradient of the liner.

The temperature history of the same four locations for 4000 psia and 4000°R gas conditions is shown in Figure 15 for a period of 12 seconds.

End Plate. - The radial temperature distribution of the end wall which caps the omega segment at the downstream end was analyzed using the thermal model shown in Figure 16. This figure shows the cross-section of the end wall forms a 0.28 inch gap with the water-cooled nozzle approach section front face. Free convection is expected to previal in the gap with the gas temperature estimated to be 3200°R. This estimate is based on NASA data (at an operating condition in the range of 3460°R) which indicated that gas temperature measurements at about 0.5 inches from the wall of a T-bar liner at the downstream location was about 80% of the combustor average gas temperature.

Two areas of the end plate required special attention. First, the extension or lip of the omega segment which projects beyond the outer surface of the end plate to facilitate welding was minimized to avoid overheating. Second, an arrangement was provided to insure adequate cooling in the corner where the end plate and the omega segments join.

Based on a segment lip overhang of 0.0625 inches in length and assuming that the heat transfer coefficient in the corner has the same value as calculated for the omega segment passage in the absence of the rail assembly, the radial temperature distribution presented in Figure 16 is predicted for the gas conditions of 600 psia and 4000°R. Because of the effect of redius of curvature, the maximum temperature in the bend of the omega segment is expected to be approximately 144°F higher than that shown in the figure.

The temperature distribution along the edge of the omega-shape end plate was estimated as an input for the stress analysis and is shown in Figures 17 and 18 for the 600 psia and 4000°R gas conditions. The figures present both the gas-side and air-side metal temperatures of the end plate.

Pressure Drop Analysis

The pressure drop in the liner and support cooling air passages was estimated for the gas conditions of 4000 psia and 4000°R. This condition has the highest pressure drop for the maximum operating temperature and is, therefore, the critical design point for evaluating pressure loading. The low air velocity (23 ft/sec) produces a frictional pressure drop along the liner support OD of only 2 psi. The flow is predicted to suffer an additional loss of 6 psi when it is required to make a 180° turn through the air supply holes at the downstream end of the liner support. The frictional loss in the liner cooling passage is 22 psi. The dump loss from the liner cooling passage into the entrance section of the combustor accounts for an 8 psi pressure drop. As compared to the above pressure losses, the loss in the combustor is negligible. Thus, the pressure differential across the liner varies from 8 psi at the upstream end to 30 psi at the downstream end.

The corresponding pressure drop for the gas conditions of 600 psia and 4000°R may be obtained by multiplying the above values by 0.15.

Pressure drop is a function of the air velocity head in the system and therefore, increases as the square of the airflow but decreases with increasing air density. Based on the curve of Combustion Air Flow versus Total Pressure Operating Envelope supplied in Reference 1, the maximum steady state operating pressure drop occurs at gas condition of 4000 psia and 2500°R and is 1.7 times as large as the values given above.

The maximum pressure drop occurs in the tunnel starting and stopping zone of the operating envelope at gas conditions of 2000 psia and 450°R and is 4.8 times the value at the 4000 psia and 4000°R condition.

Stress Analysis

The basic philosophy that guided the stress analysis effort was to perform separate analyses on each component which would isolate the effects of each specific type of loading. In this way analyses were conducted while the mechanical design was in progress and modifications to the design were accomplished that would lead directly to satisfactory stress or strain values. In performing these analyses, selective conservative assumptions were used to describe component geometry, boundary conditions and load distribution. The principle of superposition was used.

A summary of the significant stresses on each of the components that make up the thermal protection system for the high pressure combustor is presented in the following table.

TABLE 1. - LINER STRESS ANALYSIS SUMMARY

Gas Condition	Item	Location (Refer to Fig. 29)	Temp, °F	Effective Stress, psi	Allowable Stress (80% Y.S.), psi	Strain Range 10^{-6} in/in	Life Cycles	Factor of Safety
1	Liner	14*	1844	--	--	-8052	1600	-
		17*	226	--	--	7474	4300	-
		14**	1763	--	--	-7689	2050	-
	Spline	31*	323	--	--	7068	> 5000	-
		32*	1597	--	--	-4996	9000	-
	Rail	1*	202	65,100	89,600	-	-	1.38
	End Plate	21	1676	--	--	-4056	>10,000	-
	Pin-Bolt	-	400	97,300	100,000	--	--	-
2	Liner	14*	1834	--	--	-9544	1150	-
		17*	135	--	--	7313	4400	-
3	Rail	1**	70	67,268	118,500	--	--	1.76
	Rail Bolt	-	-	62,270	125,000	--	--	2.01

* 1.5 Feet downstream of fuel spray bar

** Near nozzle approach section beyond rail

Condition 1 - Steady-state at 600 psia/4000°R after 120 seconds

Condition 2 - Transient condition at 600 psia/4000°R after 30 seconds

Condition 3 - Maximum transient pressure condition (ΔP 144 psi)

The longitudinal stress level in the omega segment liner due to the radial temperature distribution results in a finite life. The point of maximum compressive stress occurs in the area of highest metal temperature. A service life of 1600 cycles is calculated for the most severe operating condition of 600 psia and 4000°R. Operating at any other gas condition results in a significant increase in life cycles. The maximum tensile strains occur in the "cold" regions of the liner and have over twice the life cycles of the hot regions. The stresses in the tube spline are also due primarily to temperature gradients. These conditions are much lower which results in a significantly higher life cycles.

The maximum stress in the rail assembly is 65,100 psi due to the combined effect of pressure load, thermal expansion and thermal gradients. This stress value is less than 75% of the minimum allowable yield stress.

The highest stressed bolt is the pin-bolt (near the nozzle approach end), holding the omega segment liner to the liner support. Loads on the bolt are due to the thermal moment tending to curl the end of the segment away from the liner support and the axial load due to thermal expansion of the omega segment overcoming the friction load between the omega segment and the rail. The effective stress in the bolt is 97,300 psi which is within the minimum allowable stress of 100,000 psi. The analysis is considered conservative since it is based on a coefficient of friction of 0.5; however, the contacting surfaces will be coated with a dry-film lubricant so that the expected coefficient of friction will be about 0.2.

The following paragraphs describe the analysis of major stresses associated with each of the components in the liner design.

Liner Radial Gradient.- The portion of the omega segment liner that forms the wall of the combustion chamber operates at a much higher temperature than that portion adjacent to the liner support. Except for a short axial distance from either end, the liner will be constrained radially forcing all elements to elongate the same amount. The radial gradient will cause a longitudinal stress in the liner, compression on the hot edge and tension on the cold edge. The magnitude of the stress will be directly proportional to the difference between the local temperature and the average temperature. In addition to the radial gradient across the section, the radial gradient across the thickness of the liner must be taken into consideration. This is particularly significant where the metal temperature is a maximum.

The two operating conditions that will cause the largest radial gradients are 600 psia and 4000 psia at 4000°R gas temperature are shown in the following table. The 600 psia condition results in the larger cross-section thermal gradient and the higher metal temperature, while the 4000 psia condition has the larger thickness thermal gradient. The maximum strain occurs at the 600 psia condition, the point of maximum combined thermal gradient.

TABLE 2. - LINER RADIAL TEMPERATURE GRADIENT SUMMARY

Gas Condition	600 psia/4000°R		4000 psia/4000°R	
Liner Location	Upstream	Downstream	Upstream	Downstream
Gradient	ΔT, °F	ΔT, °F	ΔT, °F	ΔT, °F
Cross-Section	1402	1215	972	874
Thickness	176	196	467	486
Total	1578	1411	1439	1360

The thermal analysis shows that the maximum liner temperature occurs at 1.5 feet downstream of the fuel spray bars. The metal temperature gradually falls off in the axial direction toward the nozzle approach section. The liner support temperature also falls off, but to a lesser extent, resulting in a smaller cross-section gradient at the downstream end of the liner. This reduction is sufficient to offset a slight increase in thickness gradient. Thus, the largest radial thermal gradients exist at the liner axial location where the metal temperature is the highest.

Figure 19 presents the maximum longitudinal strain (due to the thermal gradients) as a function of the developed surface of the omega segment. Since the equivalent elastic stress is above the yield strength, the liner will have a finite life.

Manson's equation for low cycle fatigue (Reference 7) was used to calculate the expected life cycles for the Hastelloy X material as a function of temperature. The data presented in Figure 20 represents twenty percent of the upper bound data. The life cycles established from this data is considered conservative since (a) the maximum strain encountered is compressive and (b) the temperature is taken as the maximum metal temperature (although Reference 8 investigator uses the mean temperature).

For those areas of the segment liner and the tube spline where tensile strains exist, the low cycle fatigue life is based on ten percent of the upper bound data. Therefore, life cycles will be one-half those presented in Figure 20.

With a maximum strain range of 8052×10^{-6} in/in and a maximum metal temperature of 1844°F, the expected life of the liner was estimated to be over 1600 cycles. Since this is by far the most severe operating condition, the expected service life for a normal distribution of test conditions would substantially exceed the 1500 cycle design goal.

The following table shows the effect of several assumptions for low cycle fatigue analysis on the estimate of service life cycles:

TABLE 3. - LINER SERVICE LIFE CYCLE SUMMARY

	600 psia/4000°R		Test Pressure Distribution*/4000°R	
% of Upper Bound	10	20	10	20
Cycles Based on Mean Temperature	3300	6600	21,500	43,000
Cycles Based on Peak Temperature	800	1600	7,500	15,000

* Assumes 70% testing at 600 psia and 30% at 4000 psia is representative of a wide range of test conditions (Reference Figure 28).

Liner Axial Gradient.- The air cooling technique that effectively controls the metal temperature of the liner in the combustion zone causes a sudden change in metal temperature near the fuel spray bar. Figure 10 shows the axial thermal gradient that exists in that area at the gas condition of 600 psia and 4000°R.

Using a simplified but conservative approach the maximum thermal stress will exist at 2.5 inches downstream of the fuel spray bar. The axial gradient, equivalent to 140°F/inch, will result in a longitudinal stress of 15,000 psi in the center-section of the omega segment and 2000 psi in the hot radius area. The hoop stress at this point is negligible.

Combining this longitudinal stress (or equivalent strain) with the longitudinal strain due to the radial gradient at this location results in a maximum strain of 3208×10^{-6} in/in (compression). This strain is much lower than the strain due just to the peak radial gradient at 1.5 ft. downstream of the spray bar.

Liner Pressure Load.- The maximum pressure across the omega segment liner occurs at the downstream end of the liner at the steady-state operating condition of 4000 psia and 2500°R. The pressure drop is equal to 51 psi. Figure 21 shows the bending stress distribution around the liner for this condition. The maximum bending stress will be 11,676 psi which is well below the yield strength for annealed Hastelloy X (AMS 5536).

The pressure stresses for the two critical conditions were analyzed at the highest pressure condition (maximum stress) and the highest metal temperature condition (minimum strength). The following table presents these limiting cases.

TABLE 4. - LINER PRESSURE STRESS SUMMARY

Steady-State Condition	ΔP, psi	Metal Temperature, °F	Location (Refer to Figure 21)	Stress, psi	Yield Strength, psi
2500 psia/4000°R	51	500 1000	A B	11,676 9,379	37,000 33,000
4000 psia/4000°R	30	173 1178 1406	A B C	6,868 5,517 3,231	42,000 30,000 26,500

A transient condition during start-up may result in a peak pressure drop of 144 psi. This condition would produce a maximum bending stress of 34,647 psi. Since the metal temperature at this condition is about 100°F as compared to a yield strength of 44,000 psi for Hastelloy X, this transiend peak pressure would not cause any permanent deformation in the liner.

Spline Circumferential Growth.- The arch section of the tube spline that faces the combustion zone has an average temperature of 1225°F and expands circumferentially more than that portion of the spline adjacent to the cold liner support. The resulting stress induced by the thermal expansion in the spline is in the form of bending in the arch-section. The maximum stress is compressive and occurs on the inside fiber of the center-section and has a value of 16,932 psi. The maximum tensile stress in the spline is 15,174 psi. These stresses are well below the Hastelloy X allowable yield stress of 25,400 psi at 1450°F metal temperature.

Spline Radial Gradient. - As in the omega segment, the tube spline has a thermal gradient in the radial direction. The temperature gradient across the 0.0625 inch thickness of the spline does not exceed 10°F and so the cross-section gradient as shown in Figure 4 constitutes the major thermal loading.

The conditions influencing the thermal stress on the spline are similar to those for the omega segment liner. The maximum strain occurs at the section of the combustor, 1.5 ft. downstream of the fuel spray bar. The operating condition of 600 psia and 4000°R produces the longitudinal strains shown on Figure 19. The maximum strain values are 7068 x 10^{-6} in/in (tension) and 4996 x 10^{-6} in/in (compression). Since the equivalent elastic stress for the peak strains exceeds the yield strength of the material, the spline has a finite service life. For the most severe operating condition (600 psia and 4000°R), a low cycle fatigue life of 5,000 cycles is expected for the spline.

Rail Radial Gradient. - The temperature of the rail assembly retaining the liner varies radially in a manner similar to the omega segment liner and tube spline. The radial cross-section gradient causes a longitudinal thermal stress in the rail. The maximum stress occurs at the 600 psia and 4000°R gas condition at a location 1.5 feet downstream of the fuel spray bar. Figure 4 shows the temperature gradient that exists after 120 seconds of operation. The resulting thermal stress presented in Figure 22 shows a satisfactory margin at all locations when compared with an allowable stress (equal to 80% of the yield strength) of Inconel X material. The radial gradient across the 0.1875 inch thickness at the bolt holes is 18°F which results in less than 3000 psi additional thermal stress.

These thermal stresses are conservative since they are based on a rail design that is not slotted in the radial direction. Slots provided at four inch intervals reduce the thermal stresses.

Rail Pressure Load and Circumferential Growth. - The principal transverse stress in the rail is due to the pressure load on the liner components and the radial expansion of the omega segments.

The major stress contribution for operation at 600 psia and 4000°R results from the thermal expansion of the various components and is 15,907 psi. The stress due to the pressure load of 30 psi is 10,231 psi. Since the final design of the omega segment liner provides for a radial clearance between the liner and the liner support, this thermal stress will be significantly reduced.

As the gas temperature is reduced the stress due to thermal expansion is also reduced but the pressure drop increases causing pressure stresses to increase. The pressure stress at 2500°R operation is 17,393 psi and adding the thermal stress results in a total bending stress of 24,200 psi (compression) in the area of the bolt holes.

Since the bending stress and the longitudinal stress due to thermal gradients are perpendicular, they are combined to calculate the maximum effective stress. The Von Mises criterion is used for combining principal stresses. An effective stress of 65,100 psi is calculated which is well below 80% of Inconel X yield strength (89,600 psi) at 350°F.

The peak pressure during start-up may result in a transient stress of 67,268 psi (compression on the inside and tension on the outside) which is well below the material's allowable yield stress.

End Plate. - The ends of each omega segment is welded to a continuous ring seal plate at the downstream end of the combustor. Loading on the plate occurs due to the following:

1. Pressure
2. Temperature gradients in the radial, circumferential and axial directions
3. Compatibility between the plate and liner caused by differential thermal gradients and expansions

In analyzing the liner end plate, it was assumed that the plate is relatively rigid compared to the omega segment liner when acting as a membrane. In axial bending the plate is relatively flexible compared to the liner. The effect of each load is analyzed separately and the principle of superposition is considered valid. The omega segment liner is modeled as segments of cylinders and plates. The mathematical model for the interaction between the seal plate and liner is a plate joining a cylinder or a plate joining a plate. The stresses caused by the radial gradient are calculated at points along the center axis of the omega segment and are assumed to be a long distance from the radial edges of the end plate. The edges are assumed fixed to the liner and it is assumed that this condition will sustain at the ends the condition of stress from along the center axis.

The radial gradient causes stresses in the circumferential direction as well as in the direction perpendicular to the surface of the plate. The latter stresses are small because the plate is only 0.125 inches thick. Figure 23 presents the transverse strain in the plate due to the radial temperature distribution. Assuming the temperature variation in the circumferential direction is linear and relatively small, the transverse stresses do not vary along the circumference but are a function of the radius.

The transverse gradient is small and the resulting stresses are considered negligible. The thermal stresses due to the axial gradient across the plate thickness are also presented in Figure 23. The maximum gradient in the plate occurs at the junction with the liner and is taken into account in the compatibility between the liner and the end plate.

The compatibility between the end plate and the omega segment liner is established along the line joining the mid-surface of the end plate with the mid-surface of the cylinder (Location 5 on the Figure 23) is a representative point on this line. Calculations are based on the average temperature of the end plate and cylinder and the average gradient.

The maximum strain occurs in the hot corner-section of the end plate (Point 5) where the compressive strain is 4056×10^{-6} in/in at a metal temperature of 1676°F. The low cycle fatigue data for Hastelloy X predicts a service life of over 10,000 cycles.

The longitudinal strain in the liner near the end plate is presented in Figure 24. Since the omega segment liner is not fully restrained near the end, the thermal strains as shown in Figure 19, 1.5 feet downstream of the fuel spray bars, are reduced. The compatibility strains due to the interaction of the liner and end plate are in addition. The maximum strain value calculated provides an estimated service life of over 1500 cycles.

<u>Liner Pin - Bolt</u>. - The temperature variation through the cross-section of the omega segment liner creates a displacement thermal moment. This moment is equal to $E \alpha \int_A TydA$. For the gas condition of 600 psia and 4000°R, this moment is equal to 38,200 inch-lb, based on elastic material properties (and assuming the maximum gradient exists throughout the length of the liner). Since the hot surface of the liner yields this moment is effectively reduced to 16,267 inch-lb. At locations remote from the ends of the liner, the moment is balanced; but at the ends the moment tends to curl the liner away from the support. A liner bolting arrangement is used near the end plate to provide an equal and opposite moment which eliminates excessive loads on the rail assembly. Figure 1 shows this pin-bolt arrangement. The tensile load on the bolts which offset the thermal moment is 8134 lb. This results in a tensile stress of 32,730 psi. At a liner location upstream of the fuel spray bars, the

displacement thermal moment is small because the temperature distribution across the liner is nearly uniform.

Since the average temperature along the length of the omega segment liner is higher than that of the rail, the liner expands axially more than the rail. The downstream pin-bolt arrangement forces the liner to expand in the upstream direction. In order to move, the liner must overcome the friction force between it and the rail. A radial clearance between the liner and liner support is provided to eliminate significant friction force in that area. The axial friction load between the liner and the rail resulting from thermal expansion and pressure loads is 9865 lb for each omega segment. This load is based on a coefficient of friction equal to 0.5. This is a conservative estimace since the coating on the liner and rail contacting surfaces reduces the coefficient of friction to 0.2. The maximum shear stress in the pin-bolt is 66,855 psi which is within the bolt's minimum allowable shear stress of 71,250 psi. The effective or combined stress is 97,300 psi which is within the minimum allowable yield stress of 100,000 psi for the AMS 6322 (Rc 32-36) bolt material.

The bearing stress in the liner bolt plate is 52,600 psi which is within the material's allowable bearing stress of 61,000 psi.

Rail Bolts. - The bolts that attach the liner retaining rail are installed with a pre-load of 800 lb per bolt. Based on a coefficient of friction of 0.5, the 44 bolts will offer a resistance to liner drag of 17,600 lb. This is about twice the 9865 pound drag load anticipated.

The bolt has a small radial clearance in the liner support and a large clearance in the rail. The bending stress in the bolt due to the axial friction load is 24,461 psi and the shear stress is 16,730 psi. The combined stress is 32,960 psi, well below the allowable stress of 82,400 psi.

Fuel Distributor Support System. - The resulting stresses in the fuel supply pipe adjacent to the mounting flange at the closure are 550 psi in bending and a negligible shear stress (7.6 psi). The resulting stresses in the fuel pipe upstream of the fuel distributor support plane are 1,101 psi in bending and a negligible shear stress (9.4 psi).

Transient Operation. - Two types of transient loads occur during a normal operation of the combustor. One transient load occurs prior to light-off and results in a sharp increase in the pressure drop across the liner. This peak pressure drop, based on NASA supplied test data, may be as high as 4.8 times the steady-state pressure drop condition calculated for 4000 psia and 4000°R operation. Therefore, the maximum transiend pressure drop for the liner would be 144 psi. The following table shows the maximum stress for this condition for each of the major liner components.

TABLE 5. PRESSURE STRESS MARGIN OF SAFETY

	Stress, psi	Location (Refer to Figure 29)	Margin of Safety	
			Transient	Steady-State
Omega Liner	34,647	12	1.3	2.8
Tube Spline	17,608	31	2.6	4.7
End Plate	27,137	22	1.7	2.1
Rail	67,268	Bolt Hole	1.8	4.1
Rail Bolt	62,270	Bolt (Bending)	2.0	4.6
Liner Pin-Bolt	4,405	Bolt (Shear)	10	>20

The margin of safety is defined as the allowable stress (taken as 80% of the minimum yield strength) divided by the calculated stress. The table shows that the margins of safety are increased for the steady-state operating condition. The steady-state condition analyzed was 4000 psia and 2500°R, which produces the highest pressure drop. The margin of safety is directly influenced by the metal temperature. Although the stress at location 16 (Figure 29) is lower than at location 12, the higher temperature at location 16 results in a steady-state margin of safety of 1.6.

The second transient load that occurs during normal combustor operation is the thermal transient. Since the most severe thermal gradients occur on the liner at the 600 psia and 4000°R steady-state operating condition, a detailed transient study at this operation was performed. Figure 14 shows the liner metal temperatures as a function of operating time. The liner surface exposed to the hot gas reaches its maximum temperature rapidly while the areas near the liner support are still increasing after 120 seconds (which exceeds the maximum possible operating time at this gas condition). Figure 25 presents the cross-section temperature gradient for the liner as a function of time. The peak temperature gradient occurs at 30 seconds. Figure 26 shows the metal temperature distribution after 30 seconds at 1.5 feet downstream of the fuel spray bars. The thermal analysis was based on a step-change in gas temperature from ambient to 4000°R. This conservative assumption results in calculation of temperature gradient that is higher than actually expected.

The temperature distribution presented in Figure 26 was used to calculate the longitudinal strains presented in Figure 27. The maximum strain of 9500 X 10^{-6} in/in at 1834°F results in a predicted life of 1130 cycles for operation entirely at the most severe gas condition of 600 psia and 4000°R. Figure 28 shows the effect upon life cycles of operation at conditions other than the most severe case. If operation at 4000 psia and 4000°R gas condition is assumed to represent the average effect of all other operating conditions, then the predicted life increases significantly as the percentage of tunnel runs at the most severe case is decreased. For example, if 80% of the tunnel runs occur at the most severe condition, the predicted life of the liner increases from 1130 to over 3500 cycles.

COST ESTIMATE

An estimate of approximate costs for fabricating the omega segment liner assembly is listed as follows:

1.	Raw materials	$ 28,500
2.	Liner new parts	42,950
3.	Assembly & welding associated with (2)	20,000
4.	Tooling	29,850
5.	Manufacturing engrg., quality control, etc.	11,500
	Total	$132,800

This cost is above the estimate prepared about one year ago, based on the preliminary design of Task I, as a result of the following factors: (a) an 8-10% labor rate annual increase due to inflation, CPI adjustment or union contract agreements, (b) an inflationary 50% or more increase in the cost of raw materials due to supply shortages, (c) the additional design features such as the accommodations for the gas thermocouple probes, the tube splines to limit hot gas ingress into the omega bulb attachment, etc. and (d) the additional design complexity to satisfy the design criteria as determined by the final stress and thermal analyses.

NASA-Langley indicated that as an intermediate solution, the current 0.375 inch thick liner support may be replaced with an 0.75 inch thick unit. The replacement of the liner support (which was not contemplated during Task I) with one of increased wall thickness is estimated to cost approximately $60,000.

Table 6 presents the estimated cost of one complete set of detail parts and the welding and assembly of the omega liner configuration. The manufacturing engineering, quality control and inspection costs stated above may be pro-rated, for the items listed for the inner liner in the table.

TABLE 6. - ESTIMATED COST OF OMEGA LINER DETAILS

Part No.*	Material Cost	Fabrication Cost	Tooling Cost
Inner Liner			
ES160874	$ 15	$ 100	$ --
ES160902	10	150	--
ES160901	15	100	--
ES160890	450	1,400	--
ES160881	6,800	2,750	900
ES160875	800	1,000	1,600
ES160876	300	600	750
ES160911	50	2,000	--
ES160910	450	1,100	--
ES160900	50	1,800	--

TABLE 6. - ESTIMATED COST OF OMEGA LINER DETAILS (Continued)

Part No.*	Material Cost	Fabrication Cost	Tooling Cost
Inner Liner (continued)			
ES160887	$ 350	$ 500	$ --
ES160884	200	1,000	--
ES160883	600	1,000	850
ES160877	3,400	1,200	2,200
ES160878	200	400	450
ES160880	800	1,800	--
ES160879	13,000	1,850	6,000
ES160905	80	100	--
ES160907	250	600	--
ES160882	--	6,300	1,000
ES160908	--	21,000	15,500
ES160885	450	1,700	--
ES160909	200	1,100	600
ES160891	--	300	--
ES160949	10	350	--
ES160945	20	100	--
ES160904	--	300	--
Misc. Nuts, Bolts	--	4,400	--
Total	$28,500	$62,950	$29,850
Liner Support			
ES160906	$30,000	$25,000	$ 2,000
ES160912	--	3,000	--
Total	$30,000	$28,000	$ 2,000

*Reference LS 34799, Sheet 1

The table does not include minor costs involved in reworks to existing hardware and instrumentation which may be accomplished at the NASA facility. These reworks include the following:

1. Remove and replace studs from front ID flange of the barrel housing and chamfer flange

2. Remove T-bar channel supports from the barrel housing

3. Machine upstream face of the exhaust nozzle approach section, weld O-ring seal liner to this face, finish-machine seal liner.

Cycle times for fabrication of the liner assembly are estimated as follows:

1.	Raw materials	6–9 months
2.	Tooling	(6 months)*
3.	Fabrication	5 months
4.	Assembly and welding	2 months

*Concurrent with cycle time for raw materials

OPERATIONAL CONSIDERATIONS

Installation Methods

In preparation for installing the omega segment liner assembly into the liner support, the 36 rail assemblies are installed in position with only a few threads of engagement of the bolts which attach the rails to the inside diameter of the liner support. The gasket is positioned on the aft-face of the liner support. The liner assembly weighing 3353 lb. is then inserted into the downstream end of the liner support.

An L-shape ring which pilots on the O-ring groove land is bolted to the liner using 36 flat countersink (82°) slotted screws (0.250-28 UNF) with 80-85 in-lb torque. The L-shape ring is then peened into the slot of the screws to lock the screws in position. The 72 shear bolts (0.5625 inch shank with 0.4375-20 UNF are installed in the liner support with 550-585 in-lb torque. The bolts attaching the rails are then torqued to clamp the liner assembly into position on the ID of the liner support with 115-125 in-lb torque.

At the upstream end of the liner support the four rectangular rails which engage the fuel supply pipe spoked support are positioned with dowels and bolted to the inside diameter of the liner support with 225-250 in-lb torque.

This liner and support assembly weighing 11,004 lb is then inserted into the high pressure barrel utilizing a new handling fixture adapter (LS 34805) in conjunction with the existing installation fixture used for the end closure and fuel distributor assembly.

The fixture adapter is bolted to the rear face of the liner support assembly at the top and bottom quadrants of the flange that bolts to the barrel ID flange. Clearance holes are located in the sector flange of the fixture adapter to clear the studs in the barrel ID flange which is used for bolting the liner and support assembly into the barrel and axially positioning the liner assembly. The upstream face of this fixture adapter is in the plane locating the upstream face of the closure. The existing handling fixture is bolted to the new fixture adapter with the same 1.5 inch diameter bolts presently used. To facilitate assembly in the high pressure barrel the liner support has four 1.0 inch skids with a 4.0 inch radius welded to the OD at the vertical and horizontal centerline. These skids prevent the O-ring from contacting the ID of the high pressure barrel during insertion of the liner assembly. With the liner and support assembly in position, the assembly can be bolted to the high pressure combustor barrel on either side of the sectored fixture adapter. The remainder of the nuts are installed on the studs for attaching the liner support to the barrel after removing the handling fixture.

The gas thermocouple probes are installed after bolting the liner support to the barrel.

The fuel spray bar assembly and end closure assembly are installed in the high pressure barrel housing with the existing handling fixture.

If inspection of the liner assembly is required, removal of the fuel spray bar assembly, the end closure and the liner and support assembly is accomplished in a reverse procedure as described above.

The installation procedure for the liner and liner support assembly is unaffected by the use of the water cooled instrumentation section.

Since the present monorail hoist lifting capacity is rated at 5 tons, there is a need to provide one of the following so that the weight of the fixture adapter (1316 lb) and the existing fixture (750 lb) can be lifted with the liner assembly.

1. Replace the monorail hoist with an increased capacity unit

2. Add another hoist to the monorail to supplement the existing unit

3. Use a mobile A-frame support hoist to supplement the existing hoist

Inspection & Repair Procedures

Inspection Procedures. - Although analysis of the life expectancy of the liner details indicate that the design goal of 1500 tunnel tests will be satisfied, it is good practice to periodically perform borescope inspections of the liner assembly and periodic visual inspections of the liner details. The areas listed below may be visually inspected and/or dye-penetrant checked at a convenient time during the initial several hundred tunnel tests. The items asterisked are critical areas which may be periodically inspected by borescope methods. The other itmes listed are not critical and are only inspected infrequently as a preventative measure.

1. Omega segment

* (a) ID for cracks

* (b) radius near ID at the bulb section for cracks and excessive metal temperature

(c) fillet radius weld attachment to the seal spline for cracks

(d) ID in area adjacent to the six bumper tangs on the fuel spray bar assembly for pounding

* (e) weld attachment to the end plate for cracks

(f) weld attachment to the four thermocouple probe bosses for cracks

(g) weld attachment to the 36 shear pin bosses for cracks and shear pin bolts for excessive bearing or shear loads

2. Tube seal spline at the surface facing the liner centerline for cracks

3. Liner Support

(a) O-ring and gasket (downstream end) for hardness and pinch

(b) Attaching flange to barrel for nut looseness (stud stretch)

(c) downstream face for interference with nozzle approach section

4. Liner Rail Assembly

(a) weld attachment to shield and end cap for cracks

(b) attaching bolts and clearance holes for excessive bearing or shear loads and bolt elongation

5. Fuel spray bar spoked support radial lugs and rails for pounding or excessive bearing loads

6. Thermocouple probes for chafing or excessive bearings loads where the probe immerses through the liner section

Based on the results of these inspections, rework and critical areas requiring future inspections on a periodic basis may be established.

Repair Procedures. - Several recommended methods for repair of damage to the liner are suitable and selection of a method will depend upon the extent of damage. The repair methods are categorized as follows:

1. weld-repair cracks in segment

2. weld patch in segment

3. replace full length segment

The liner assembly is made of Hastelloy X material which can be weld-repaired with tungsten inert gas (TIG) weld using Hastelloy X wire without further processing. The repair methods are described as follows:

1. If the liner develops a crack, the crack is routed out so that virgin metal is exposed. Subsequently, the omega segment that requires repair is gas-backed by introducing a small tube to within a few inches of the area to be welded. The tube is connected to an inert gas bottle. The crack is TIG welded with Hastelloy X wire.

2. If the liner develops a local burned-out area, this area is cut-out and a 0.125 inch thick sheet of Hastelloy X is inserted into the cut-out in the omega segment. The patch is tack-welded into position. With inert gas backing as described above, the patch is TIG welded using Hastelloy X wire.

3. For more extensive cracking or burn-out damage of the liner, the full length omega segment may be replaced with a new segment. This repair is accomplished by removing the liner from the liner support in reverse of the installation procedure previously described in this report. The liner assembly is supported on a fixture used for the longitudinal welds during the original manufacture of the unit. The fillet weld between the damaged omega segment and the seal spline is machined-off. The weld between the damaged omega segment and the end plate is also removed. A replacement omega segment is welded to the seal spline and end plate in the manner established for the initial manufacture of the liner assembly.

Although repair is not expected to be required, materials selected for all details of the design except seals are weldable which offers an obvious method of repair.

Operating Envelope Limits

The thermal protection system has been designed to meet the specified operating envelope presented in Reference 1 for a minimum 1500 cycles at any firing condition, except that the minimum operating pressure at 4000°R temperature is 600 psia. Because of the reduced effectiveness of regenerative cooling at lower pressures, operation at 4000°R and 400 psia will cause excessive liner temperatures and reduced cyclic life expectancy. Consequently, the shaded area of the operating envelope presented in Figure 30 in the pressure range below 600 psia at the 4000°R gas condition should be avoided.

Because of the improved ratio of radiant to convective heat transfer at high pressures, the effectiveness of regenerative cooling is increased and the resulting gas-side wall temperatures are lower. For this reason, the thermal stress levels in the omega segment are not as close to material strength limits at the high pressure region of the steady state operating range, even though the total heat flux and corresponding local temperature gradients through the metal thickness are about 2.5 times higher at 4000 psia than at 600 psia. The life expectancy in the high pressure region is, therefore, substantially greater than the 1500 cycle design goal.

Since the tunnel tests are predominantly in an operating region of moderate gas pressures and temperatures, it is expected that a normal distribution of test conditions would yield an estimated low cycle fatigue life substantially in excess of the 1500 cycle goal. Therefore, some operation in the shaded region may be tolerated within the 1500 cycle goal when combined with the moderate loading of other tunnel test conditions.

Pressure loading of the liner results from the pressure drop through the cooling passages. Since this is proportional to the velocity head in the cooling system, it increases directly as the square of the airflow but inversely as the average density of coolant. The highest pressure drop occurs in the tunnel starting and stopping zone at the maximum airflow condition of 1200 lb/sec at 2000 psia chamber pressure. This condition causes nearly five times the pressure loading of the 4000 psia and 4000°R steady state operating condition; but it is an extreme transient, not likely to be encountered when using the normal starting and stopping routines.

The maximum recorded tunnel test transients as supplied by NASA have been less than half of this pressure loading condition. Although, the liner is designed to withstand this higher loading without permanent deformation, it is recommended that such operation be restricted to accidental or emergency conditions.

There is a possibility that an abnormal shut-down of the tunnel without a cool-down period may cause the loss of the 0.450 inch axial clearance between the nozzle approach section front face and the downstream face of the liner support. This clearance is set by a shim at the upstream ID flange of the pressure vessel, which is used for attachment of the liner support. A gap is maintained during all operating conditions. However, if the air supply for the normal cool-down period (as described in Reference 1) is instantaneously shut off after the most severe operating condition of 600 psia and 4000°R, the resulting heat soak condition will cause an increase in the temperature of the liner support to a maximum of 309°F. This temperature causes the liner support to expand 0.620 inches which would produce an interference condition with the nozzle approach section. If this occurs, it is expected that the liner support studs at the upstream end will stretch. After experiencing this unusual condition, inspection of the studs or a check of the torque on the attaching nuts is recommended to determine the necessity for stud replacement.

Several assumptions make this failure mode a very remote possibility. The cursory analysis assumes operation at the most severe gas condition. At this time the air supply is shut-off. It is assumed that shut-off is instantaneous without cooling for any period of time except for residual low pressure air in the 100 ft. length of 18 inch pipe between the combustor and the air regulator valve. Since the heat soak requires several minutes for the liner support to reach 244°F (the zero clearance condition), several potential safeguards may be arranged as follows:

1. Procedure for manually re-opening of the main air valve

2. By-pass loop to provide cooling air if the main valve cannot be opened

3. Auxiliary cooling air supply system.

Since the potential hazard is limited to the studs, the cost of items 2 or 3 may not be warranted.

Instrumentation

During the initial test period, selected temperature and pressure information should be measured to assure that manufacture and assembly of the thermal protection system has been performed correctly and to verify those assumptions used in the thermal analysis which were based on testing of the present liner design, particularly in areas where stress levels are expected to be high. The areas of primary interest are as follows:

1. Maximum temperature of the liner inner wall at known steady-state operating conditions

2. Temperature gradient through the omega segment radially between the ID exposed to hot gas the the OD adjacent to the liner support

3. Axial temperature gradient along the hot-side of the omega segment from about 1.5 feet upstream of the fuel spray bar to about 3 feet downstream of the spray bar during steady-state operation

4. Axial temperature distribution in the liner support during steady-state operation

5. Air temperatures and pressure drops through the cooling system passages

6. Liner pressure differentials for structural loading during normal start-up and shut-down transients

7. Airflow, fuel flow, chamber pressure, chamber temperature and air inlet temperature history during start-up, steady-state and shut-down conditions as currently recorded on all tunnel tests

The recommended instrumentation plan for the omega liner is summarized in Figure 31.

Data supplied by NASA for previous liner configurations indicate that the temperature of the liner is slightly higher in the bottom half of the combustor chamber than at the top (Reference 10). Since it is not possible at this point to identify critical areas more closely than the above general observation, as a practical compromise it is recommended that two omega sections in the bottom half of the combustion chamber, one in line with one of the six fuel spray bar radial supports and one between supports, be instrumented to measure metal temperatures as follows:

1. Along the centerline of the omega segments on the coolant side of the inner wall at six inch intervals from six inches upstream to 24 inches downstream of the fuel spray bar location plus one at 18 inches upstream and one at 36 inches downstream

2. Along the outer leg of the same omega segments, on either side, where the leg is welded to the tube seal spline and adjacent to the outer support liner at 18-inch intervals from 18 inches upstream of the spray bar location to 36 inches downstream

3. Along the outside surface of the liner support at the same circumference locations as the instrumented omega segments at five foot intervals beginning one foot from the attaching bolt flange

4. Along the inside surface of the liner support at locations corresponding to the outside locations but only in the area not covered by the liner.

In addition to the metal temperatures, selected sections should also be instrumented to measure the following:

1. Cooling air temperature at

   (a) inlet to the barrel housing

   (b) downstream end (exit) of the liner support

   (c) upstream end (exit) of the liner

2. Cooling air pressure as a differential of combustor pressure at

   (a) inlet to the barrel housing

   (b) downstream end (exit) of the liner support

   (c) downstream end (entrance) of the liner

   (d) upstream end (exit) of the liner

These pressures shall be recorded during several transients, i.e., start-up and shut-down conditions, in order to verify the design pressure load multiplier used in the stress analysis.

Previous liner configurations have suffered fatigue failures because of their response to the severe vibrational environment. To verify that the new thermal protection system will not experience similar difficulty, strain gages shall be attached to twelve alternate omega segments at a position about six inches upstream of the fuel spray bars, and at six locations on the liner support corresponding to every other instrumented omega segment at a position about six inches upstream of the liner.

Vibration recordings shall be made during start-up, steady-state and shut-down conditions. These recordings shall be analyzed prior to extensive tunnel operation to identify problem areas, if any, and to permit changes in procedure, assembly or design that assures that operation under high vibratory response conditions does not occur.

Spare Parts

A recommended list of spare detail parts of the omega segment liner configuration is presented below.

TABLE 7. SPARE PARTS LIST

Item	Part No.	Units Per Assy.	Unit Spares	Configuration
Omega Segment	ES160879*	36	7-9	Basic & Alternate
Tube Spline	ES160877*	36	7-9	Basic & Alternate
Retaining Rail Assy	ES160882*	36	4-6	Basic & Alternate
Stud - Liner Support Attaching	14-12	40	60	Basic & Alternate
Nut - Liner Support Attaching	14-13	40	80	Basic & Alternate
Nuts, Bolts, Washers, Dowels	Various	Various	15%	Basic & Alternate
O-Ring - Liner Support	ES160886	1	6	Basic & Alternate
Pin - Bolt	ES160949	72	12	Basic & Alternate
Spacer - Spray Bar Support	ES160874	2	2	Basic & Alternate
Gasket	ES160945	1	4	Basic & Alternate
Spring - Probe Seal	ES160901	8	16	Basic
Seal - Probe	ES160902	4	2	Basic
O-Ring - H_2O Transfer	LS 34799-203	2	6	Alternate

* Part No. has suffix - H when applicable to the alternate configuration (water cooled instrumentation section design).

Procurement of these detail parts is recommended during the outset of fabrication of the liner assembly since the added cost of machining set-up time, raw material trim and the normal learning curve associated with a later purchase of limited quantities may be avoided. Also, the costs of nuts and bolts are significantly affected by the quantity ordered.

The determination of spare parts and quantities was based on assumptions that (a) catastrophic damage is not reasonably expected, (b) liner damage, if any, would be limited to cracks in the omega segments, (c) most cracks are repairable by welding or patching, (d) during an infrequent major disassembly and reassembly of the liner, a minimal number of expendable items such as nuts, bolts, washers, o-rings, may require replacement.

CONCLUDING REMARKS

Based on the final design and analyses of the omega segment liner and thermal protection system for the 8-foot HTST combustor pressure vessel, the following conclusions are presented:

(a) The new liner design is expected to provide a substantial improvement in the cyclic life expectancy over the present liner design.

(b) The new liner design is operable over the full operating envelope of the tunnel test facility.

REFERENCES

1. Statement of Work-Design of Thermal Protection System for 8-Foot HTST Combustor, NASA - Langley Research Center, 1-13-2681 Exhibit A, March 24, 1972

2. Moskowitz, S., "Design of Thermal Protection Systems for 8-Foot HTST Combustor," NASA CR-112319

3. Knudsen, J. G., and Katz, D. L., "Fluid Dynamics and Heat Transfer", Chapter 14, McGraw-Hill, 1958

4. McAdams, W. H., "Heat Transmission", Chapter 7, McGraw-Hill, 1954

5. Rohsenow, W. M., and Hartnett, J. P., "Handbook of Heat Transfer," Chapter 3, McGraw-Hill, 1973

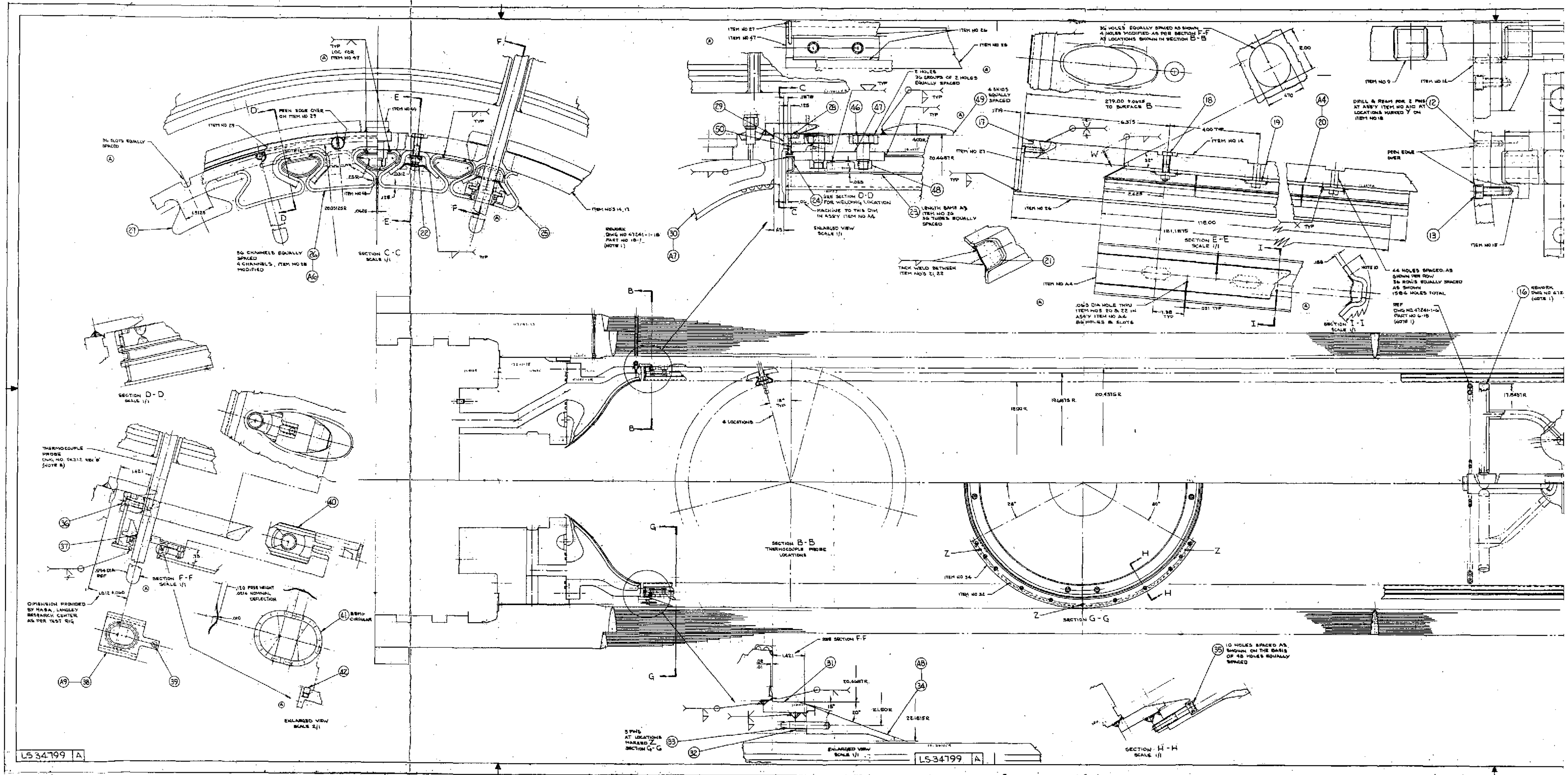
6. "Design Report on the Combustor Unit for the NASA High Temperature Structural Dynamics Facility", ALCO Products Inc., 1961

7. Halford, G. R., and Manson, S., "Application of a Method of Estimating High Temperature Low Cycle Fatigue Behavior of Materials", Transactions of the ASME, Vol. 61, 1968

8. Taira, S., "Thermal Fatigue and Its Relation to Creep Rupture and Mechanical Fatigue", Kyoto University, 1963

9. Curtiss-Wright Materials Properties Handbook

10. NASA drawing LB-HMG031170 and ten accompanying curves supplied during Task I



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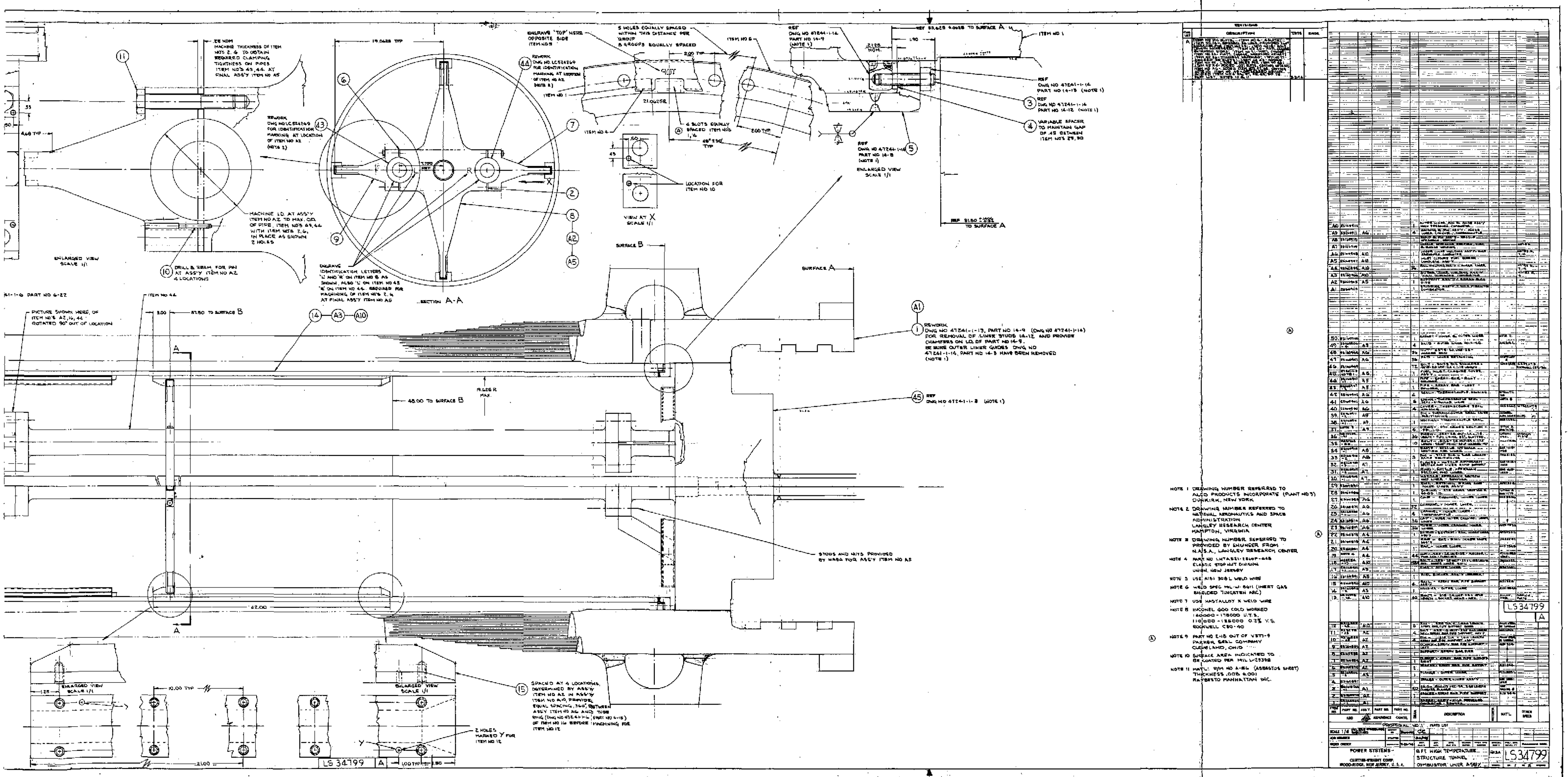

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Figure 1. Omega Segment Liner Design Layout.

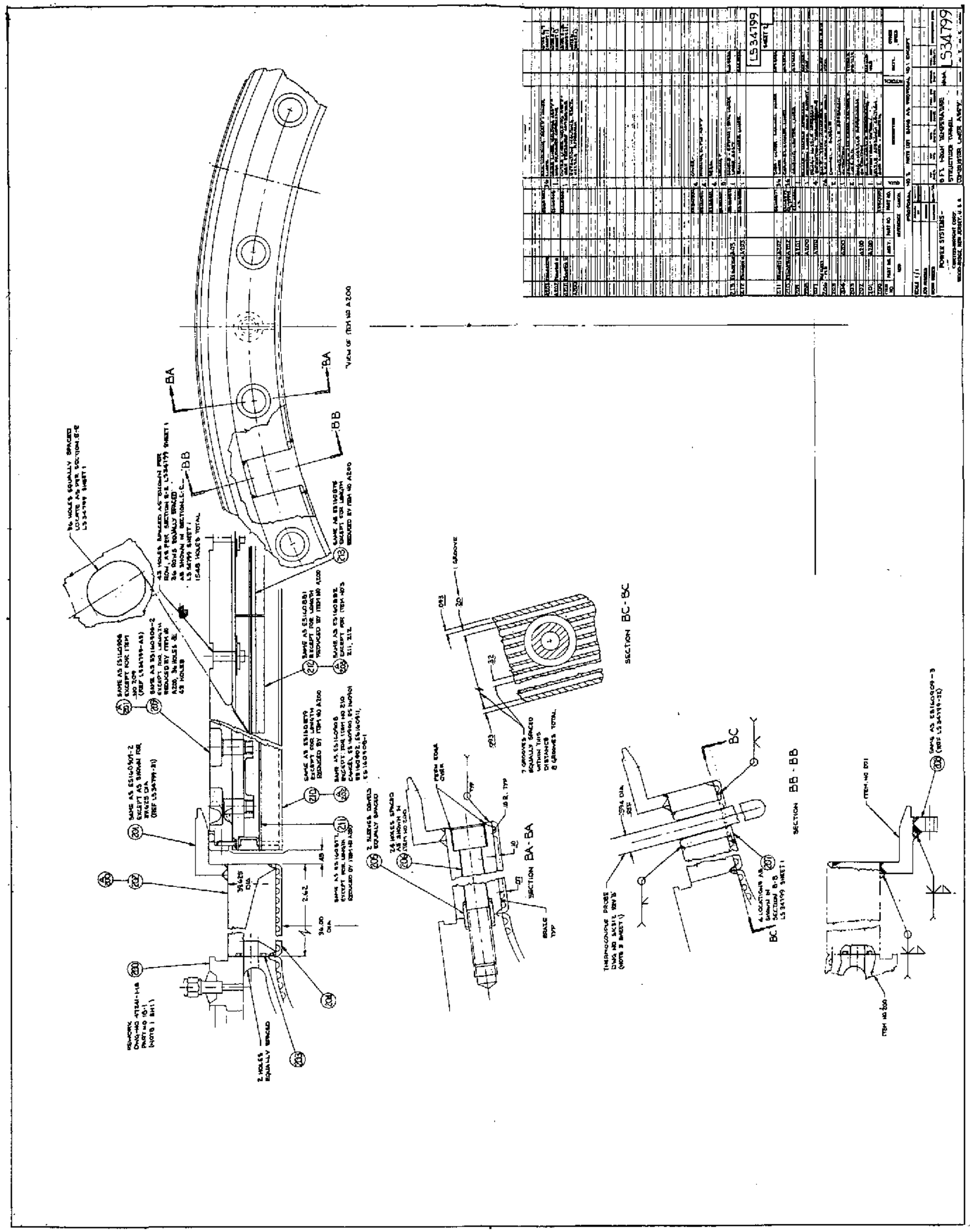

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Figure 2. Water Cooled Instrumentation Section Design Layout.

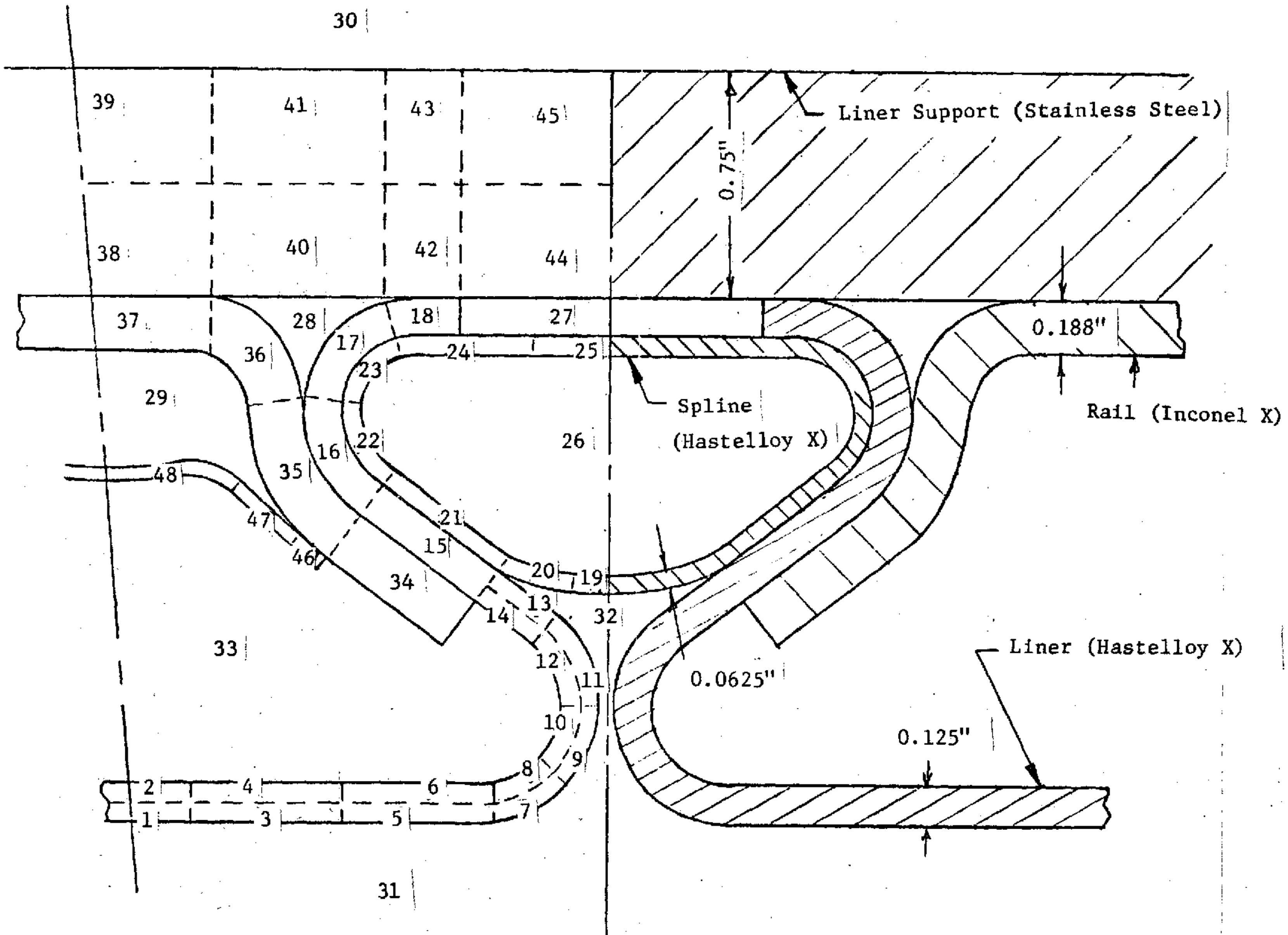


Figure 3. Liner Two-Dimensional Thermal Model.

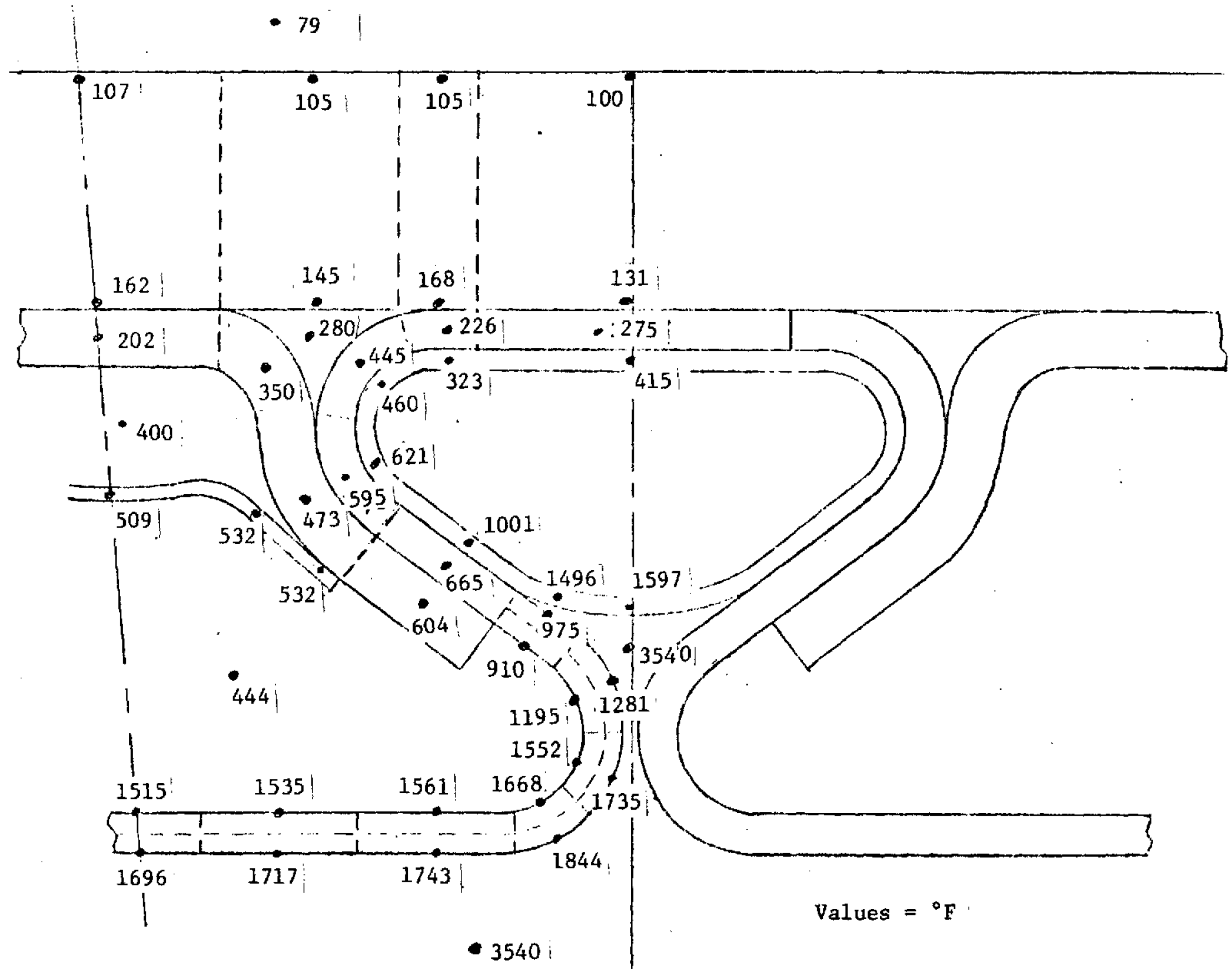


Figure 4. Temperature Distribution at 600 psia after 120 Seconds at 1.5 Ft. Downstream of Spray Bar.

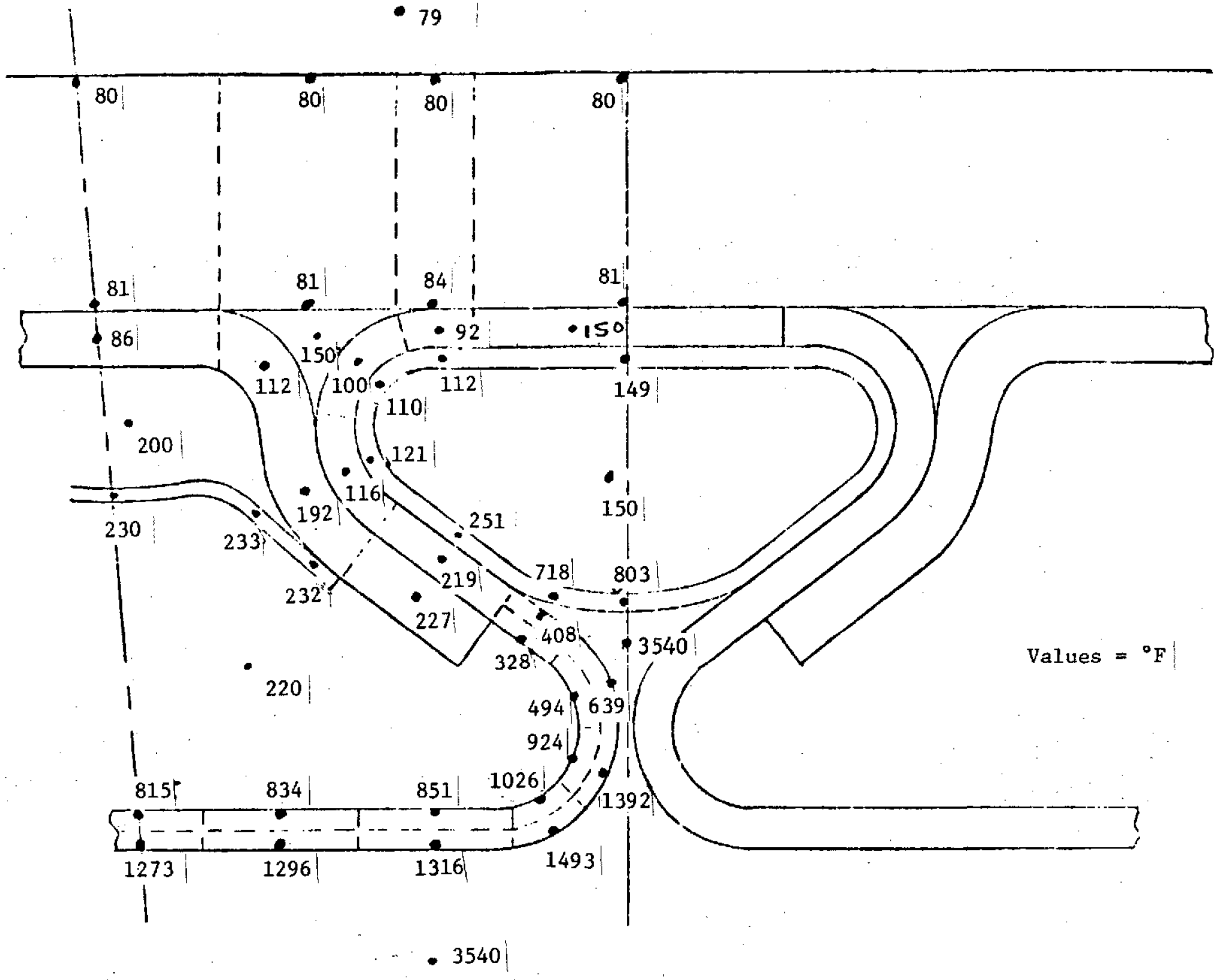


Figure 5. Temperature Distribution at 4000 psia After 12 Seconds at 1.5 Ft. Downstream of Spray Bar.

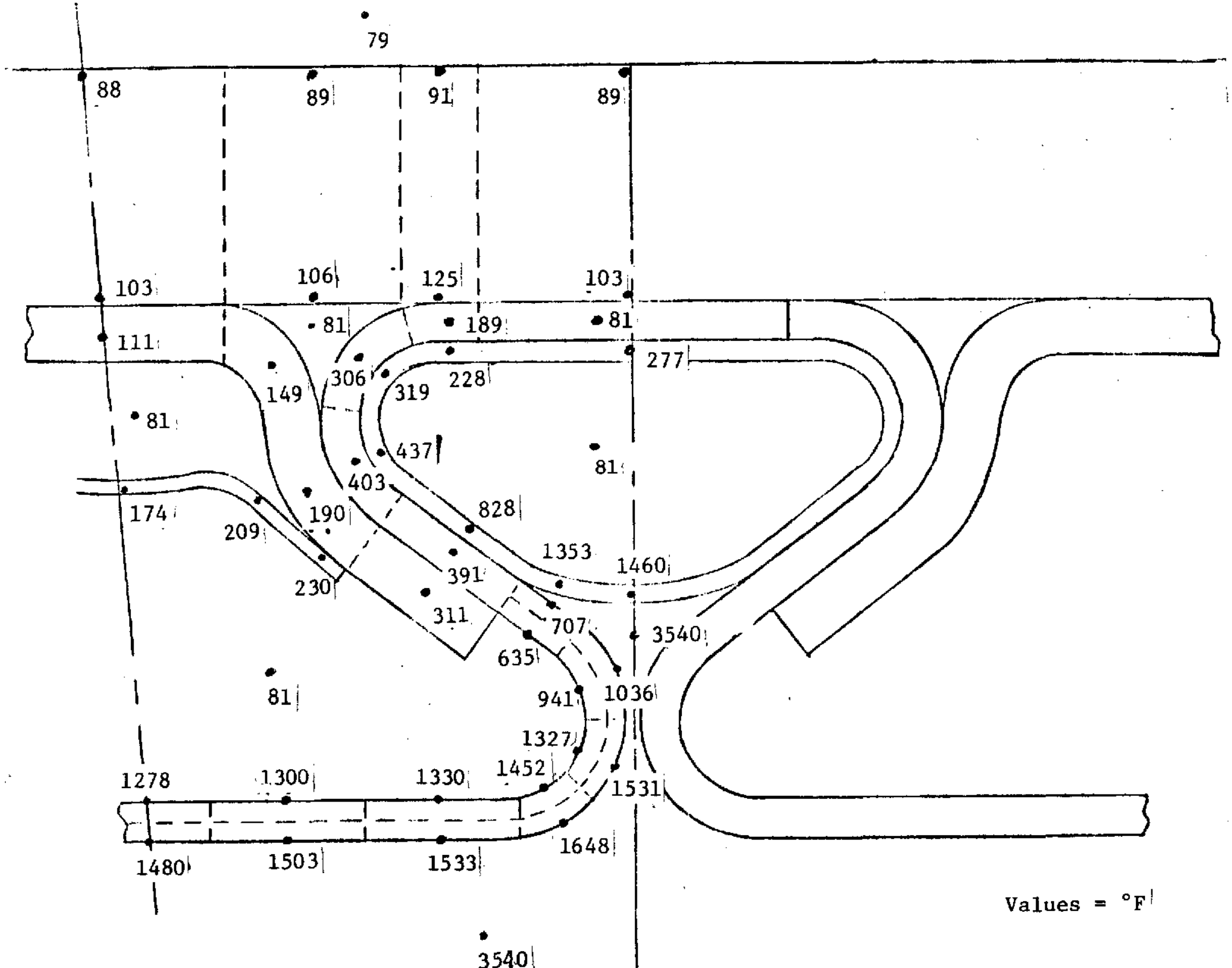


Figure 6. Temperature Distribution at 600 psia after 120 Seconds at 5 Inches Upstream of Nozzle Section.

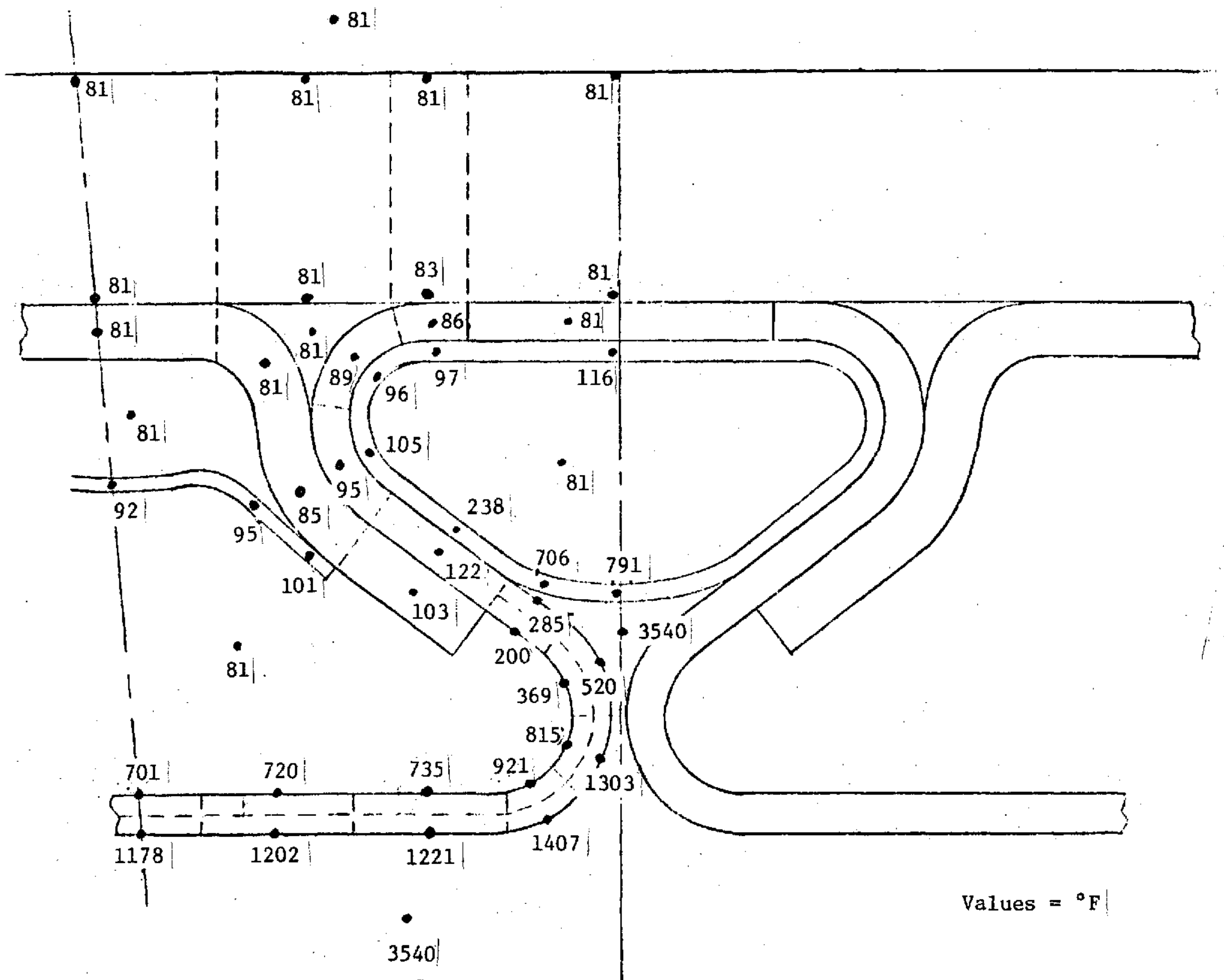


Figure 7. Temperature Distribution at 4000 psia after 12 Seconds at 5 Inches Upstream of Nozzle Section.

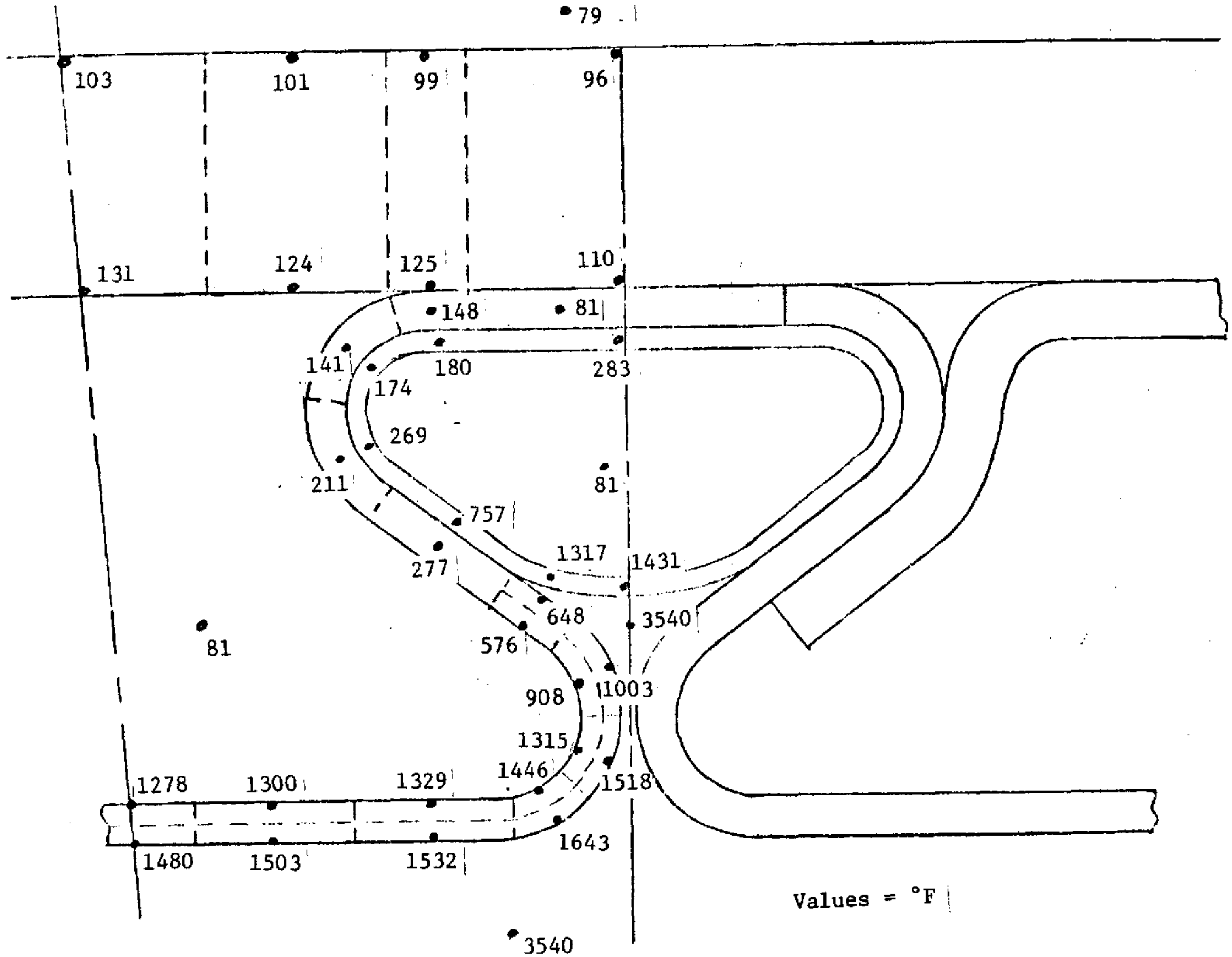


Figure 8. Temperature Distribution at 600 psia after 120 Seconds Near Nozzle Section Beyond Rail.

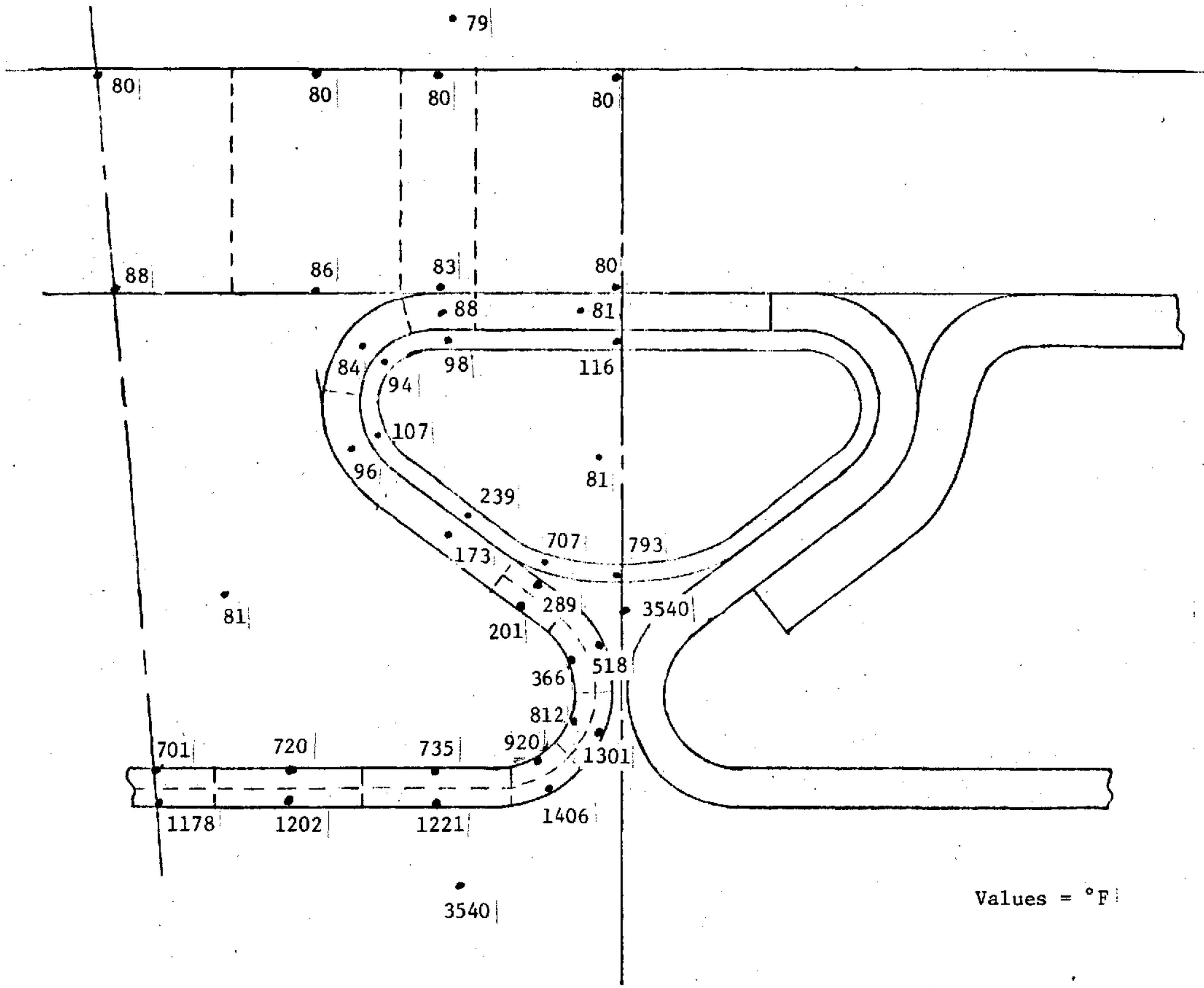


Figure 9. Temperature Distribution at 4000 psia after 12 Seconds Near Nozzle Section Beyond Rail.

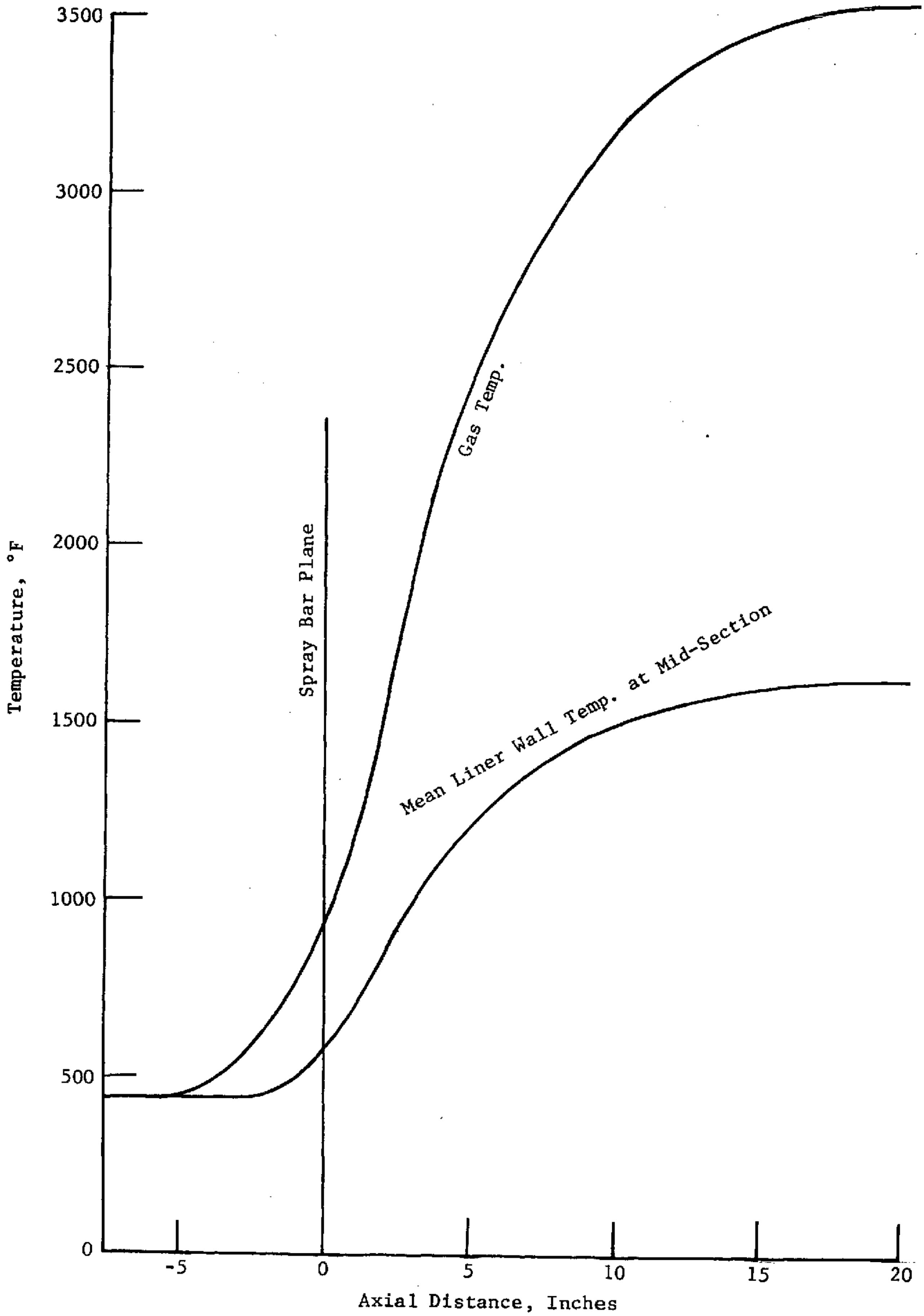


Figure 10. Axial Temperature Distribution Near Spray Bar.

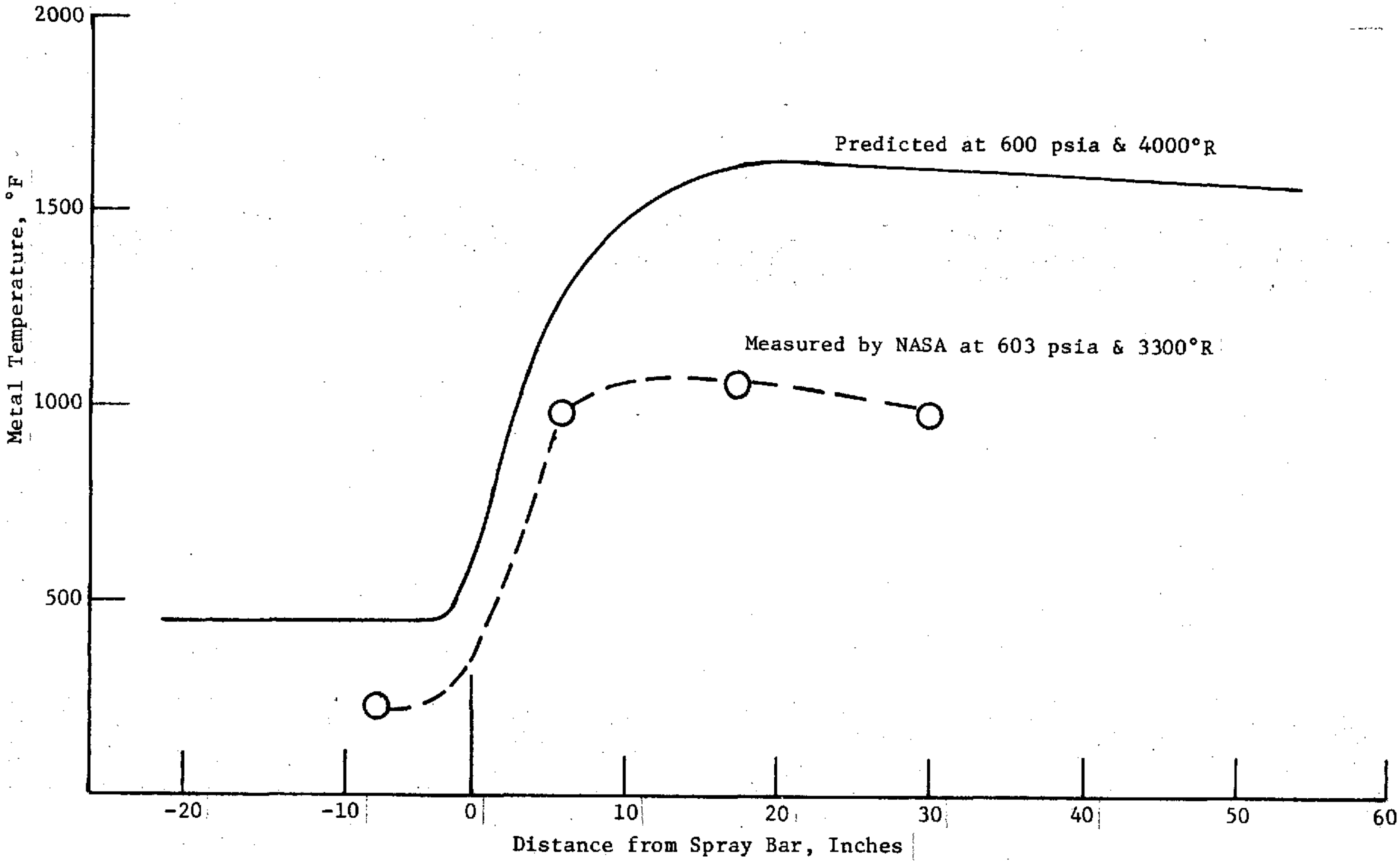
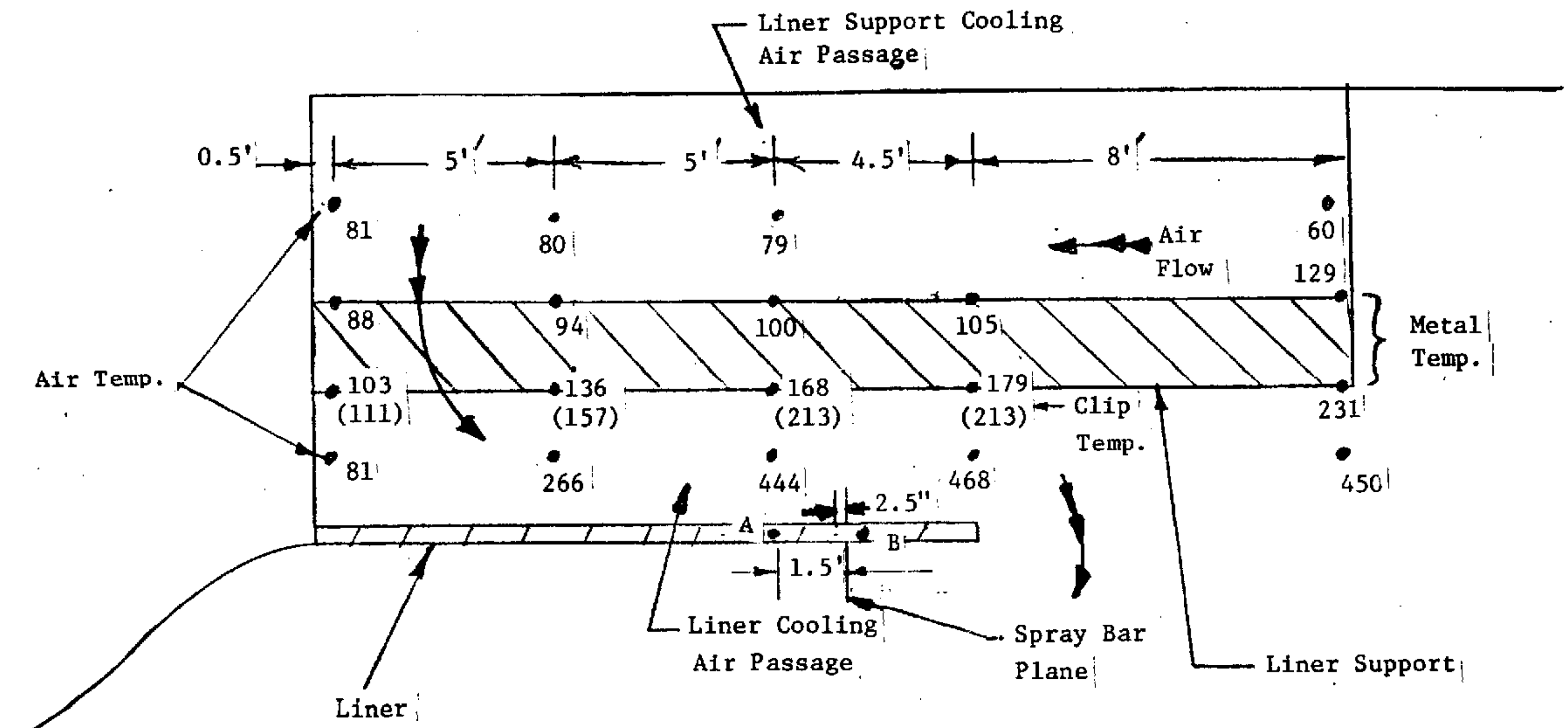


Figure 11. Comparison of Measured & Predicted Liner Axial Temperature Distribution Near Spray Bar.



Temperatures in °F

A = Maximum Radial Temperature Gradient
B = Maximum Axial Temperature Gradient

Figure 12. Liner and Support Axial Temperature Distribution.

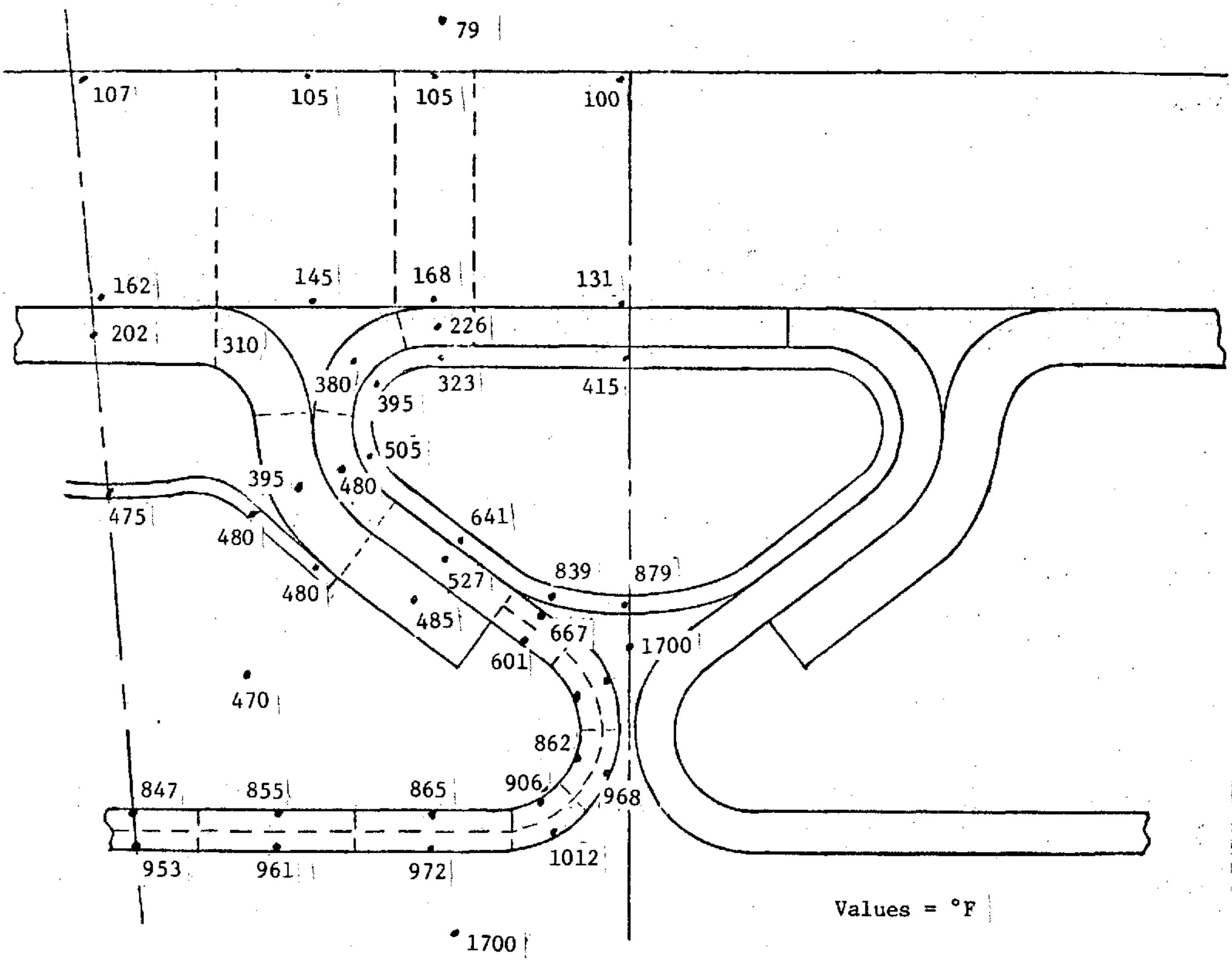


Figure 13. Temperature Distribution at 600 psia and 4000°R after 120 Seconds at 2.5 Inches Downstream of Spray Bar.

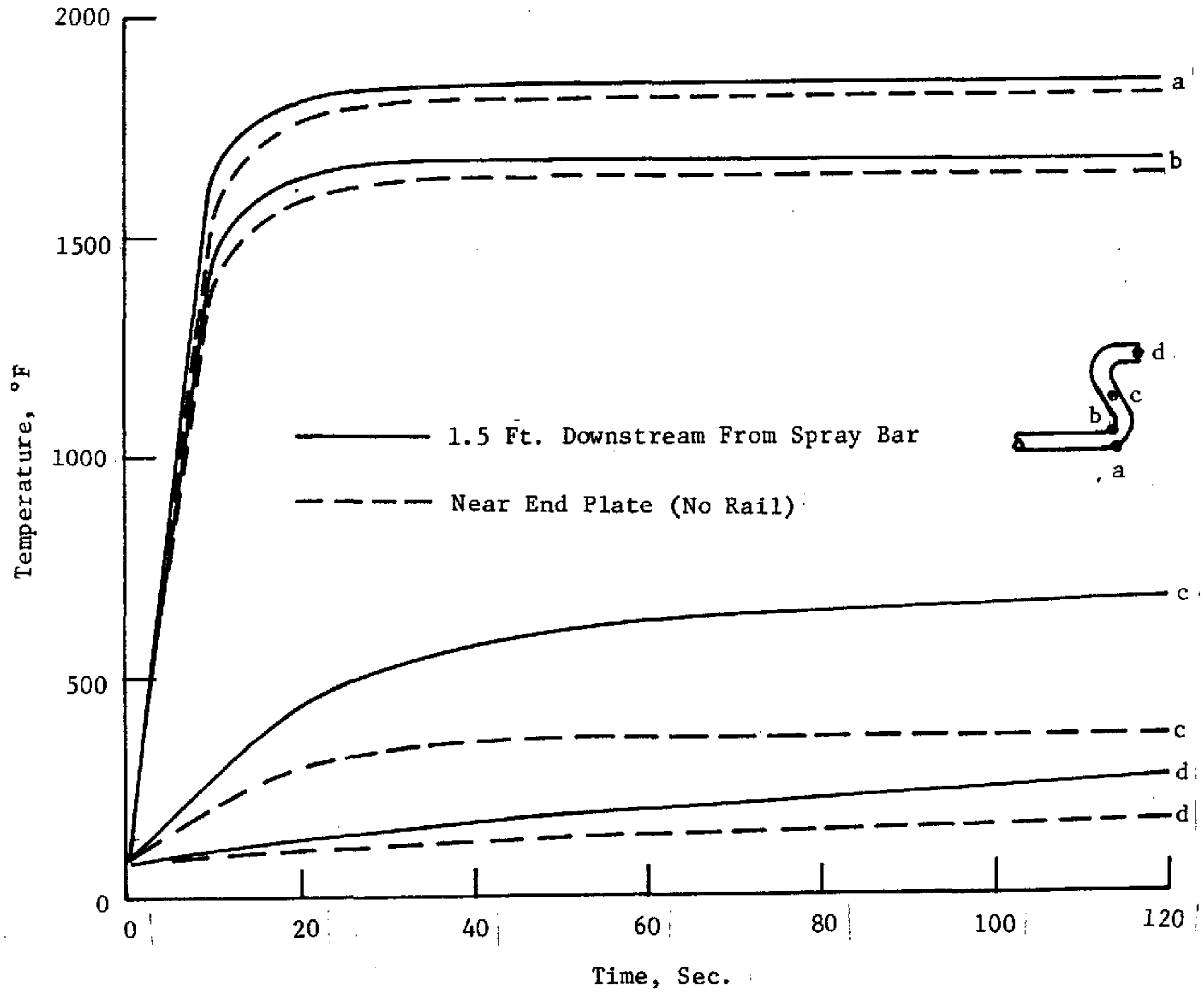


Figure 14. Liner Transient Temperature History at 600 psia & 4000°R

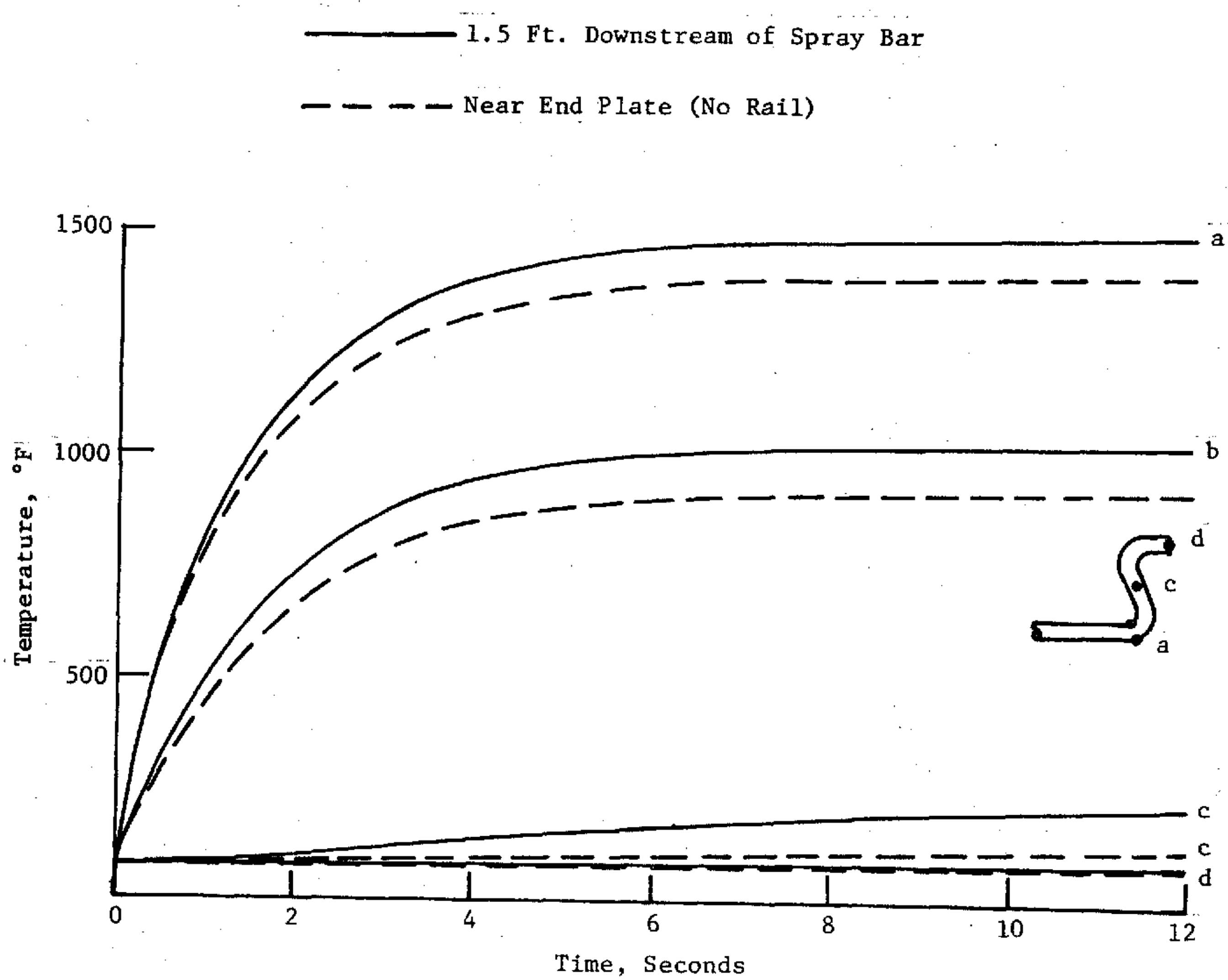


Figure 15. Liner Transient Temperature History at 4000 psia & 4000°R.

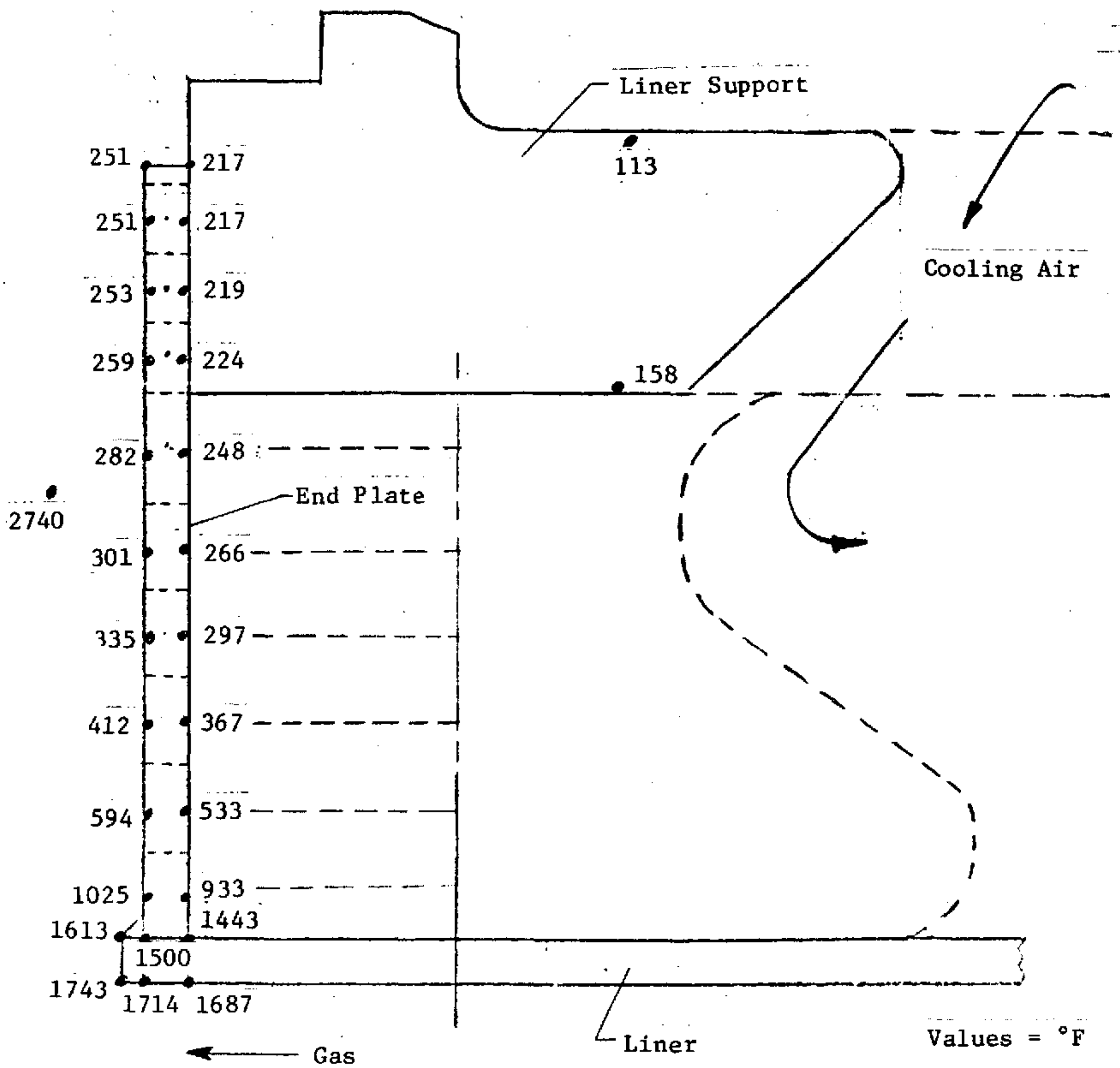


Figure 16. End Plate Temperature Distribution at 600 psia and 4000°R after 120 Seconds.

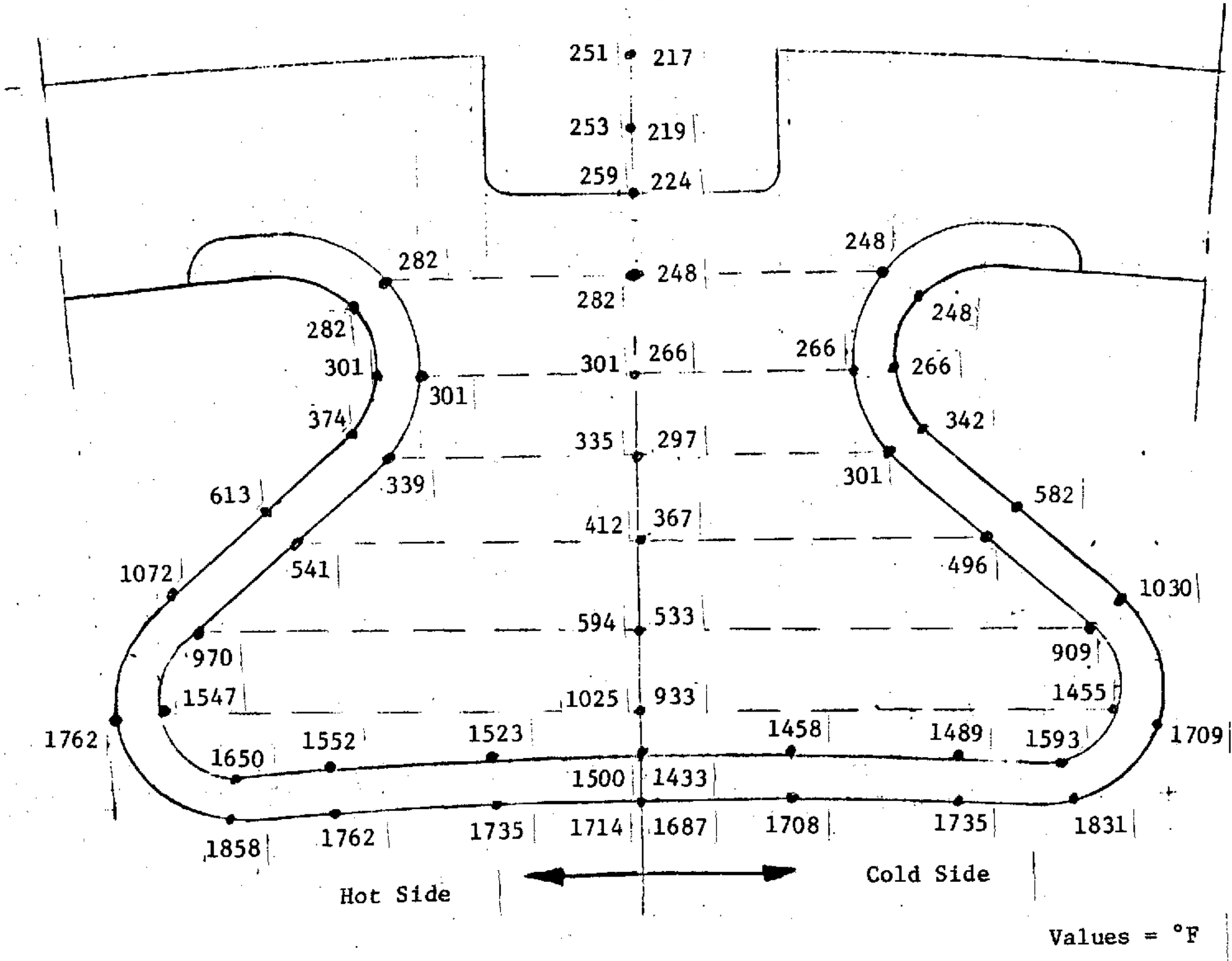


Figure 17. Liner & End Plate Temperature Distribution at 600 psia and 4000°R after 120 Seconds.

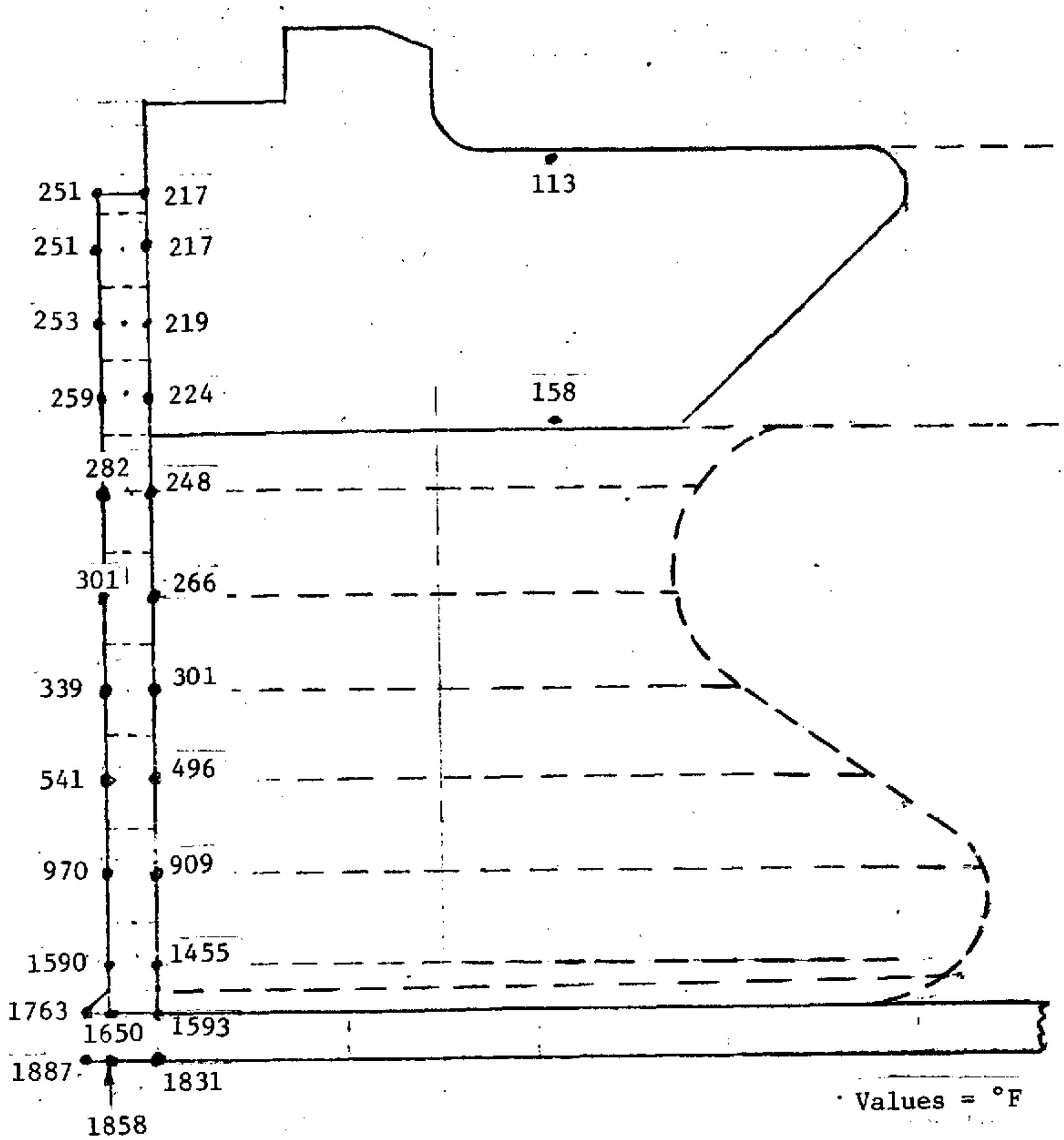


Figure 18. End Plate Temperature Distribution at 600 psia & 4000°R after 120 Seconds.

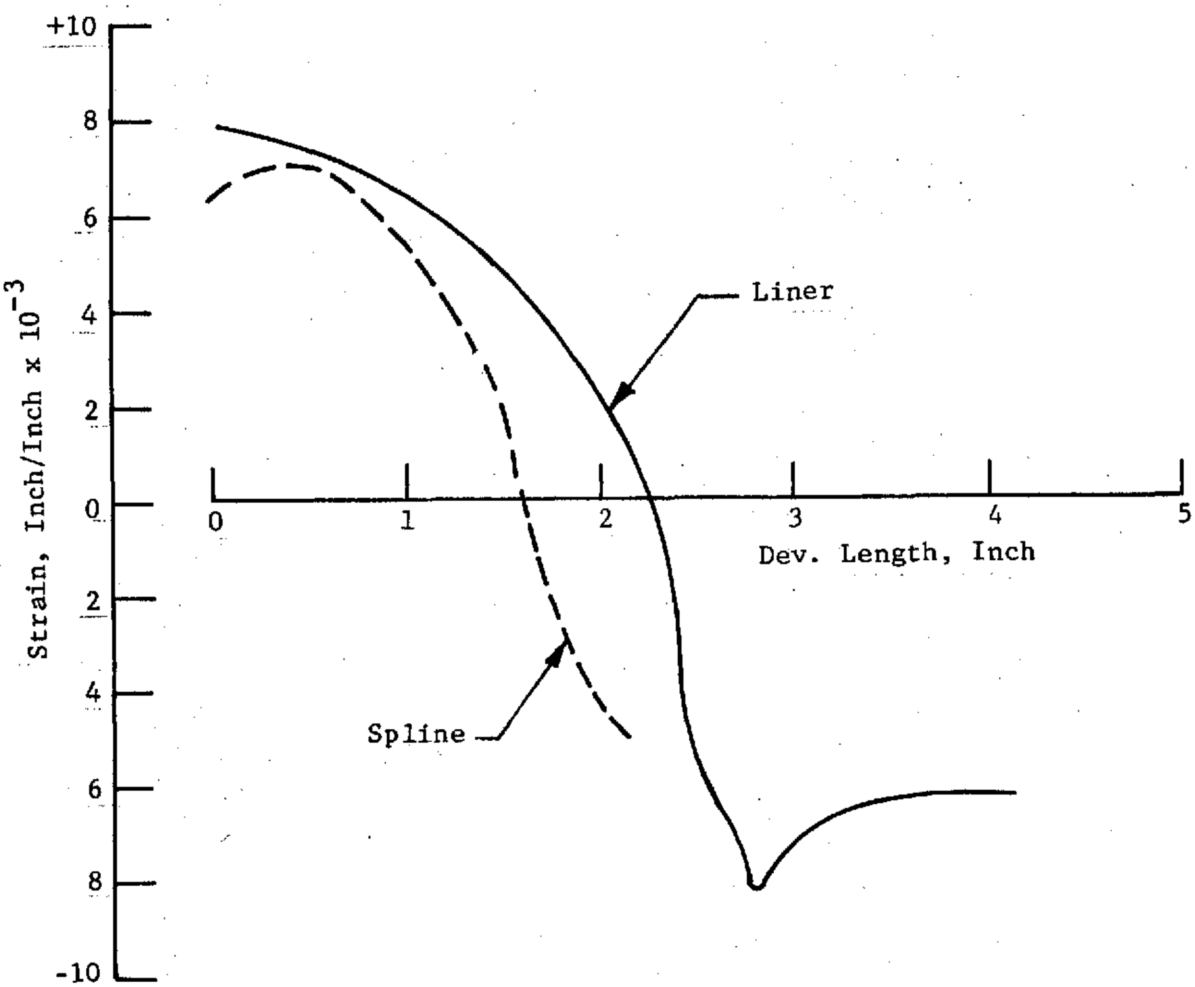


Figure 19. Liner Radial Thermal Gradient Strain Distribution at 600 psia and 4000°R After 120 Seconds at 1.5 Ft. Downstream of Spray Bar.

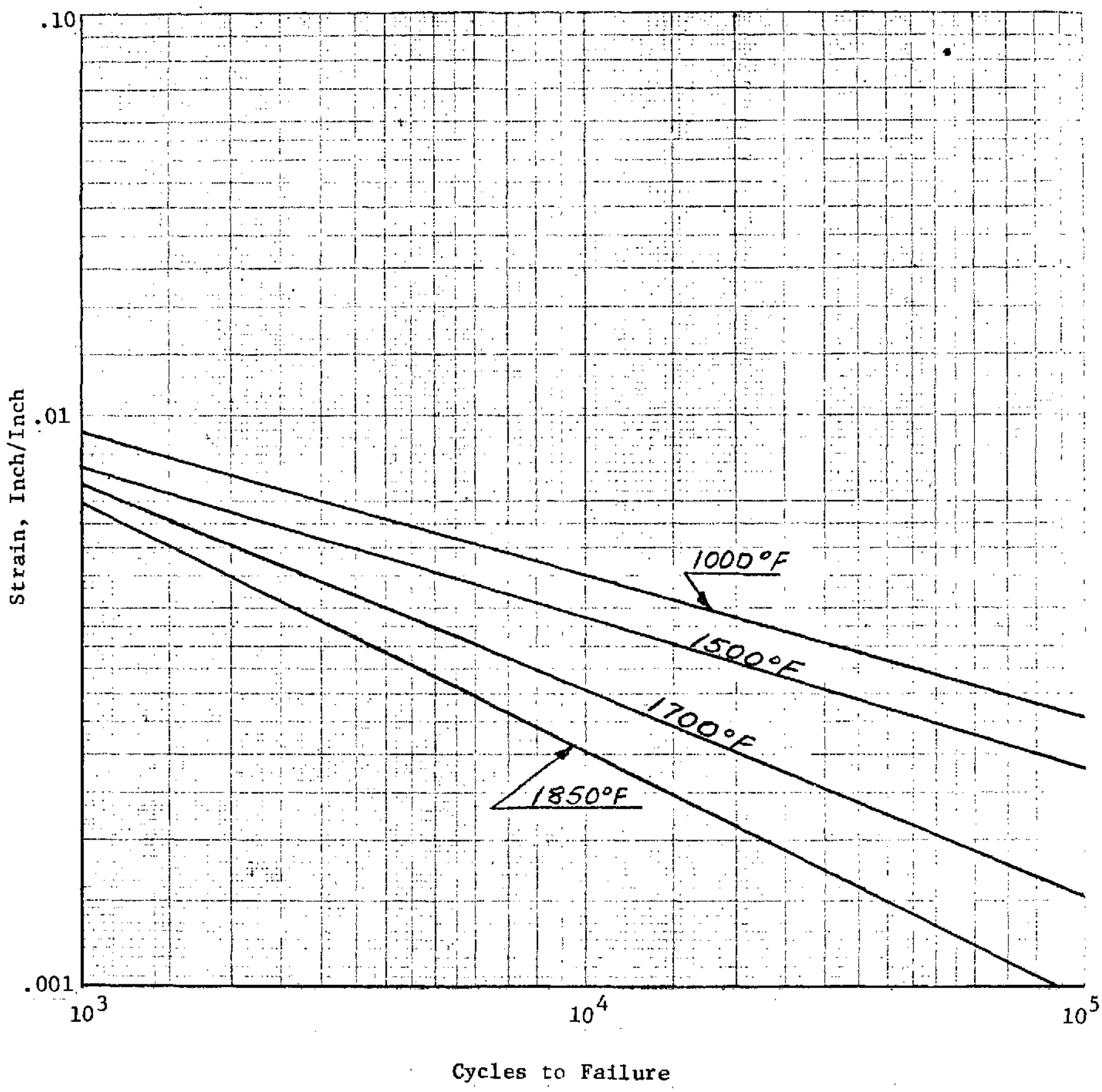
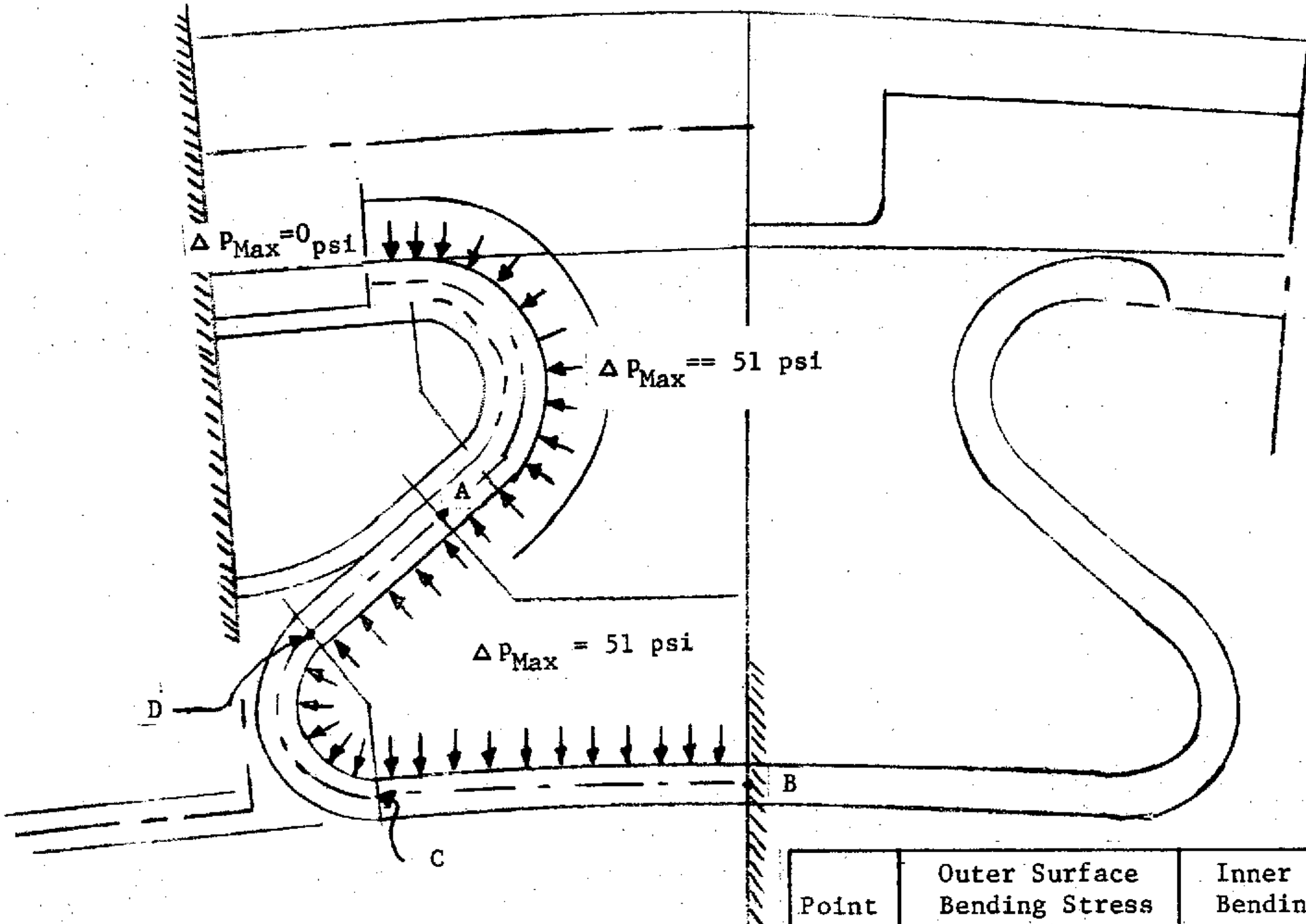
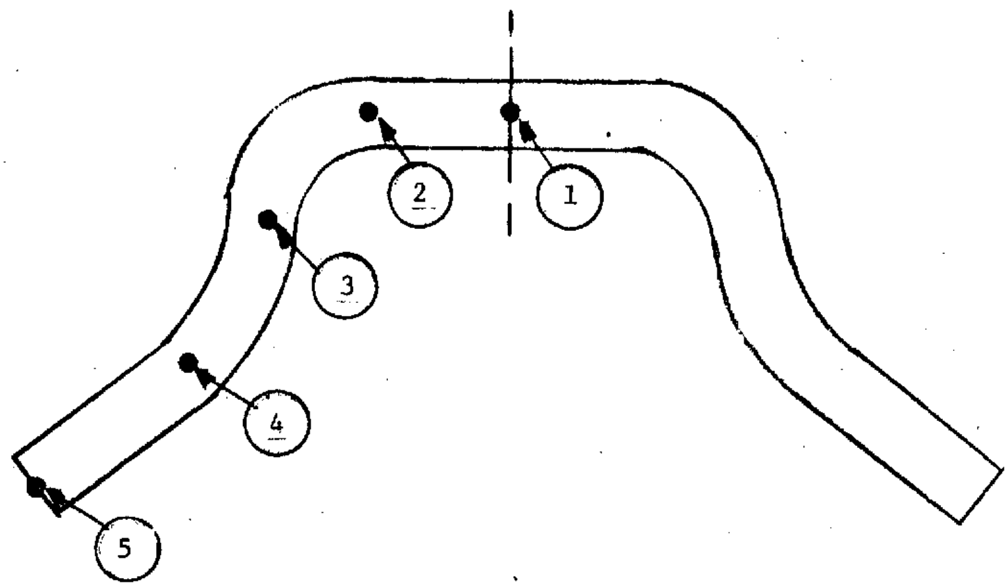


Figure 20. Low Cycle Fatigue Life of Hastelloy X.



Point	Outer Surface Bending Stress	Inner Surface Bending Stress
A	+ 11,676 psi	- 11,410 psi
B	+ 9,379	- 8,454
C	- 4,721	+ 5,493
D	- 117	+ 386

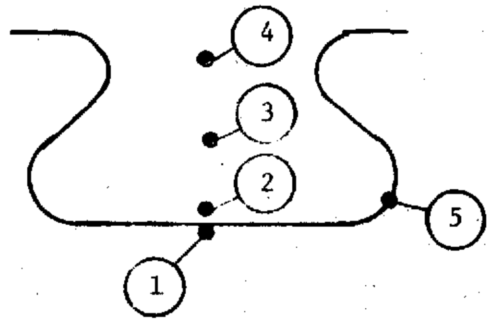
Figure 21. Liner Pressure Stress Distribution at 4000 psia & 2500°R.



Location	Temp., °F	Stress, psi	Allowable* Stress, psi
1	202	49,520	91,500
2	292	30,130	90,500
3	420	3,030	88,800
4	532	- 19,994	88,800
5	668	- 47,656	86,800

*80% of Yield Strength

Figure 22. Rail Radial Thermal Gradient Stress Distribution at 600 psia and 4000°R after 120 Seconds at 1.5 Ft. Downstream of Spray Bar.



Location	Transverse Strain in./in. x 10^{-6}	
Radial Temperature Gradient		
1	- 2988	
2	- 1472	
3	+ 3581	
4	- 3675	
Thickness Gradient		
2	425	
3	251	
4	193	
Compatibility		
	Inside	Outside
5	+ 2271	- 1068
Total Strain		
1	-	- 2988
2	- 1047	- 1897
3	+ 3840	+ 3822
4	- 3482	- 3868
5	+ 1224	- 4056

Figure 23. End Plate Thermal Gradient Strain Distribution at 600 psia and 4000°R After 120 Seconds.

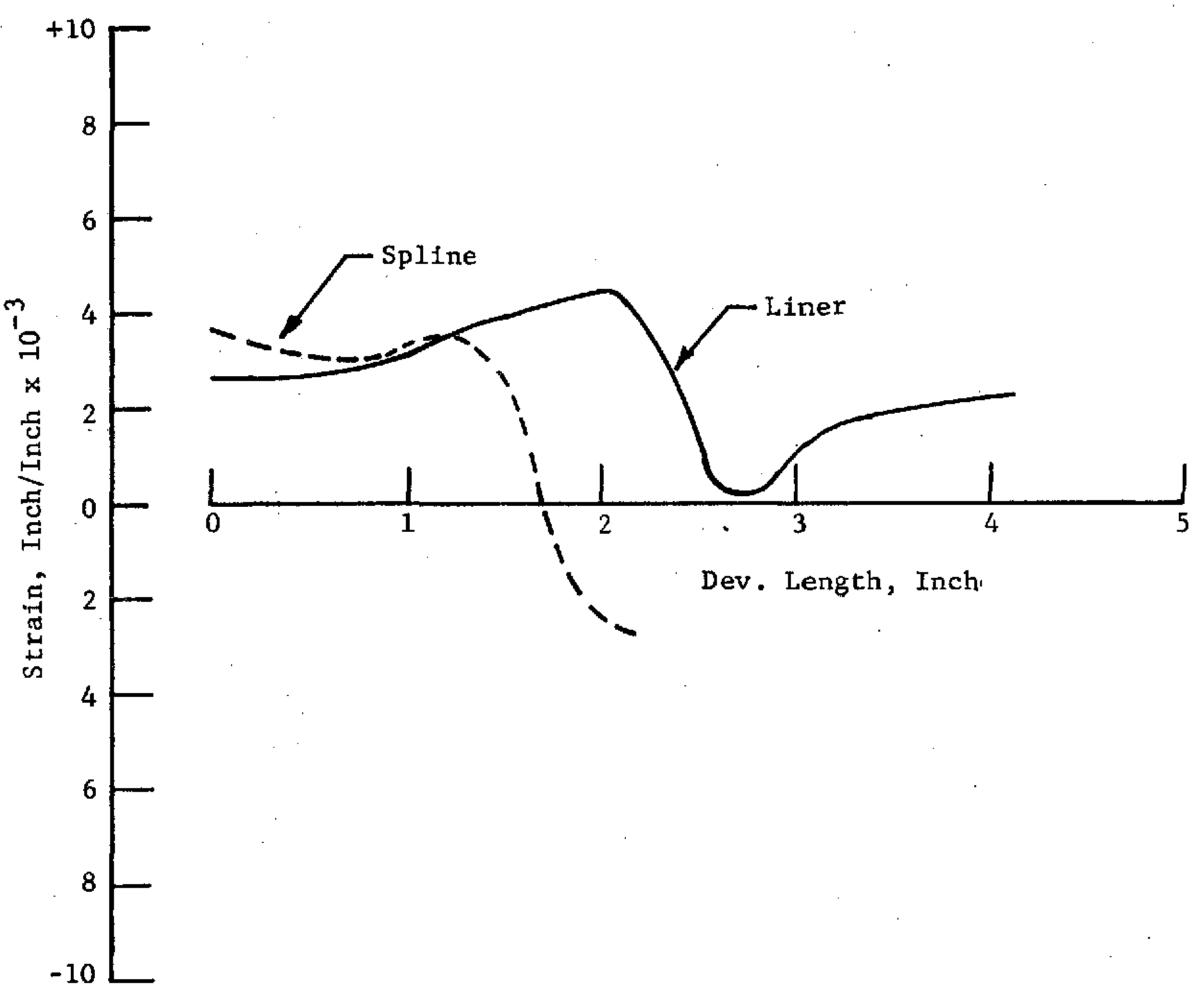


Figure 24. Liner Thermal Gradient Strain Distribution at 600 psia & 4000°R at End Plate Joint.

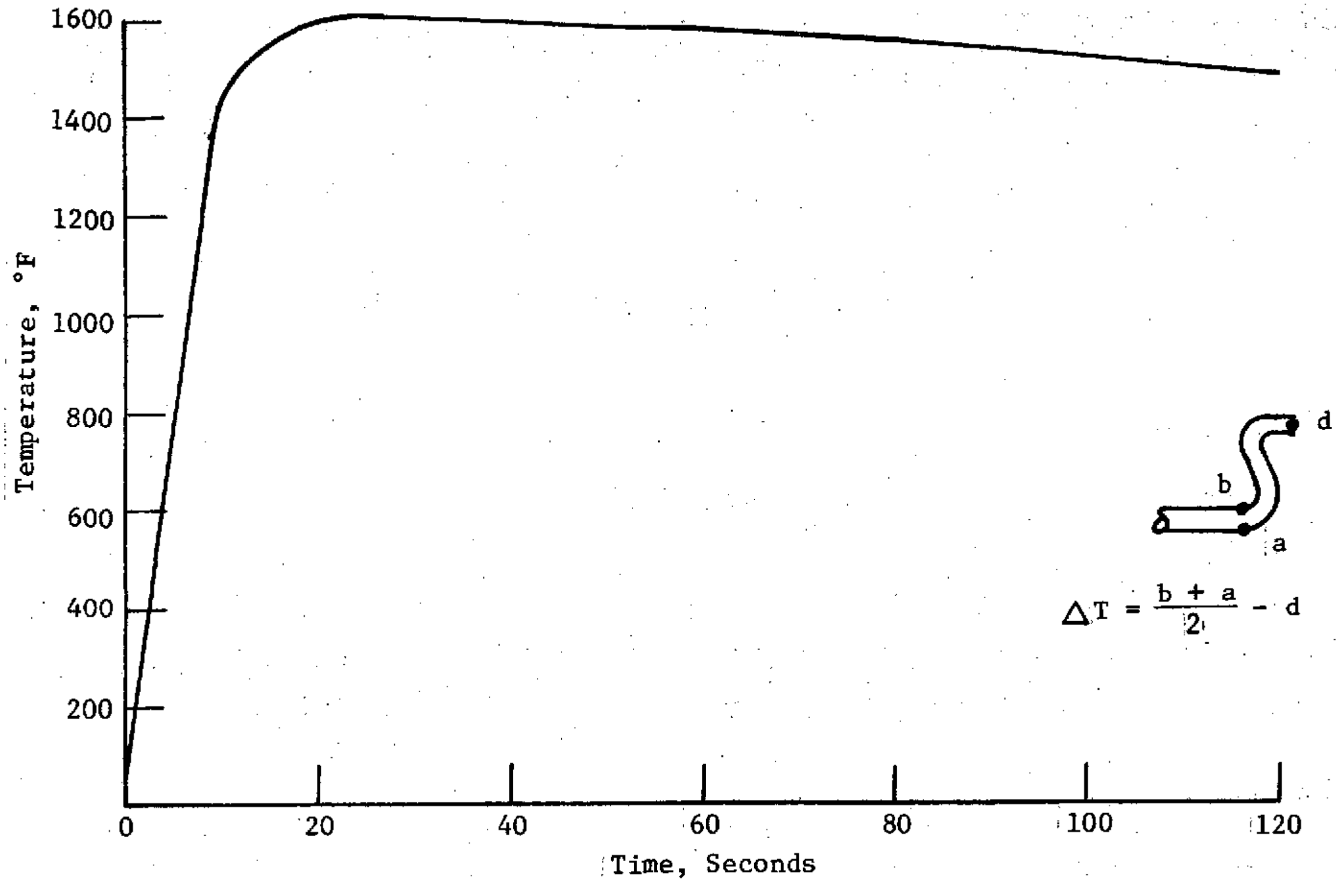


Figure 25. Liner Average ΔT Transient Analysis at 600 psia and 4000°R at 1.5 Ft. Downstream of Spray Bar.

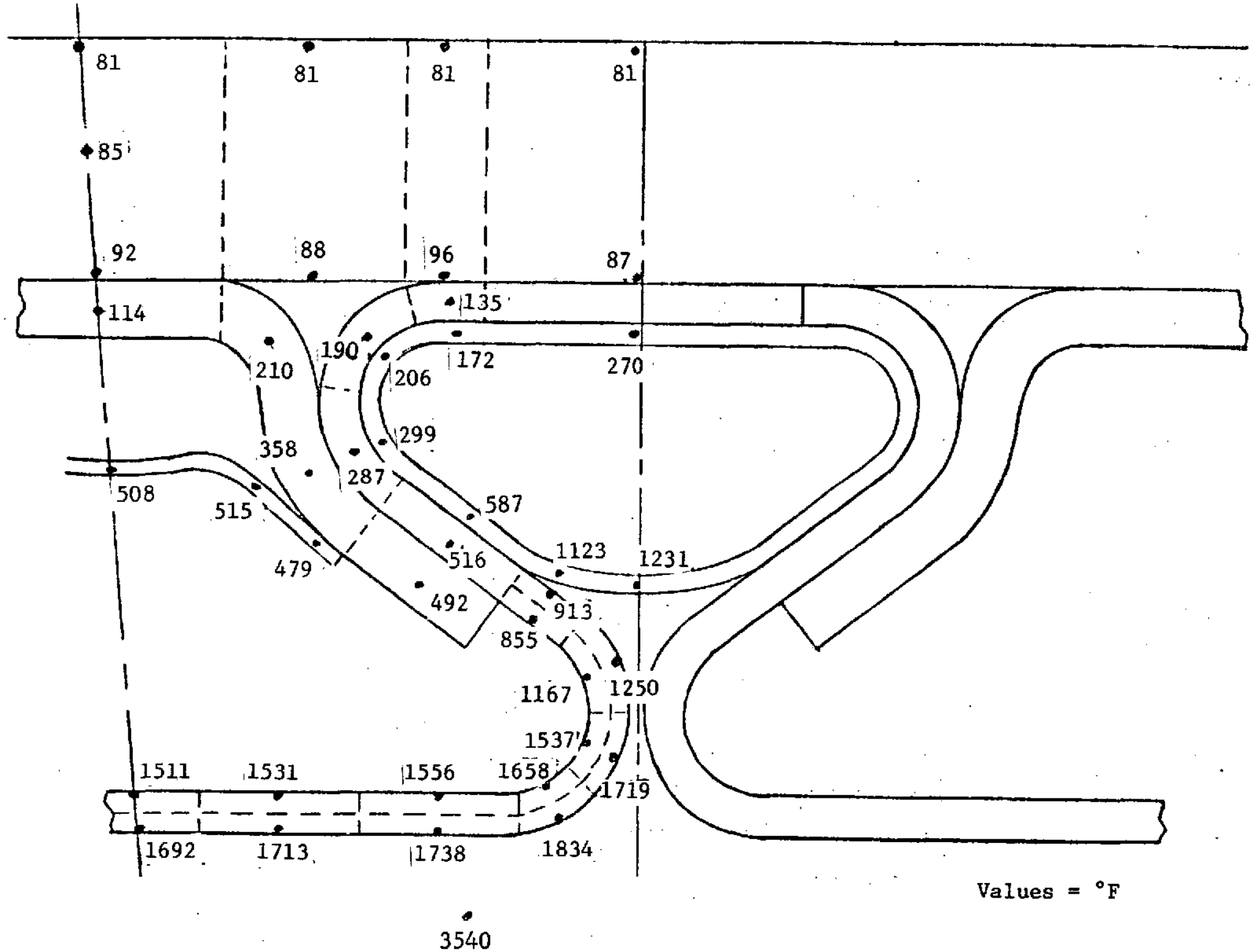


Figure 26. Liner Temperature Distribution at 600 psia
after 30 Seconds at 1.5 Ft. Downstream of Spray Bar.

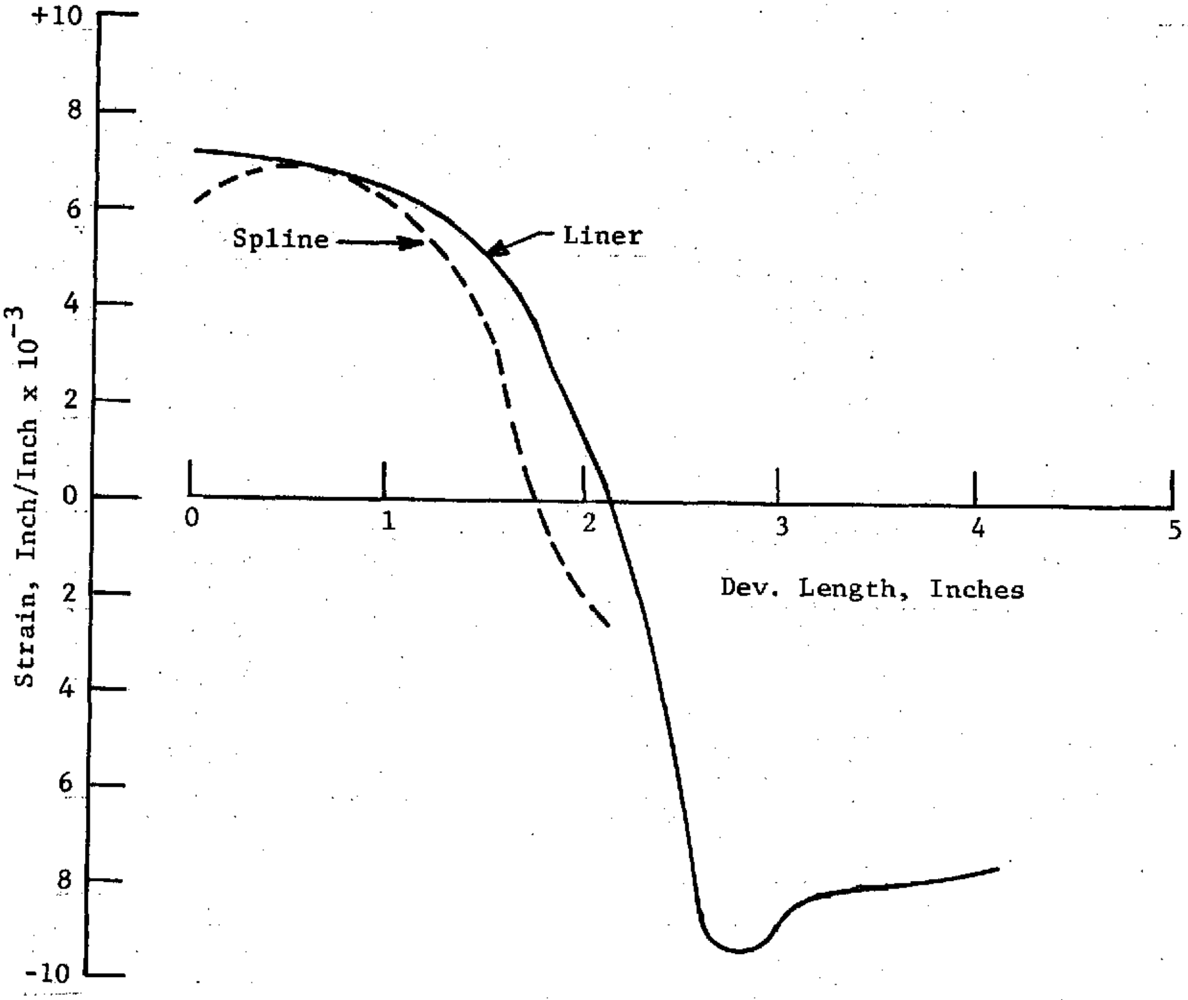


Figure 27. Liner Thermal Gradient Strain Distribution at 600 psia and 4000°R after 30 Seconds at 1.5 Ft. Downstream of Spray Bar.

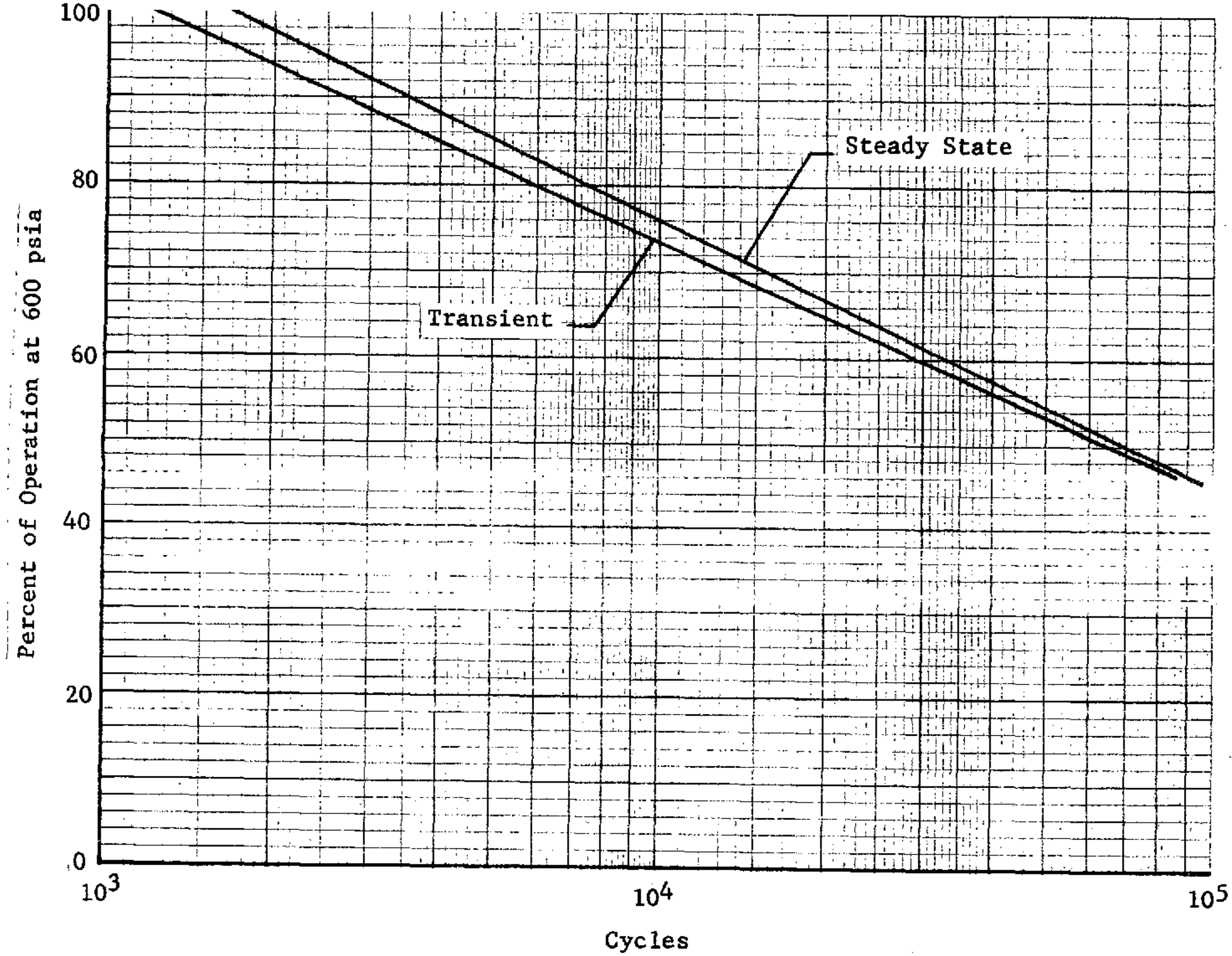


Figure 28. Estimated Liner Service Life at 4000°R for Distribution of Gas Pressure.

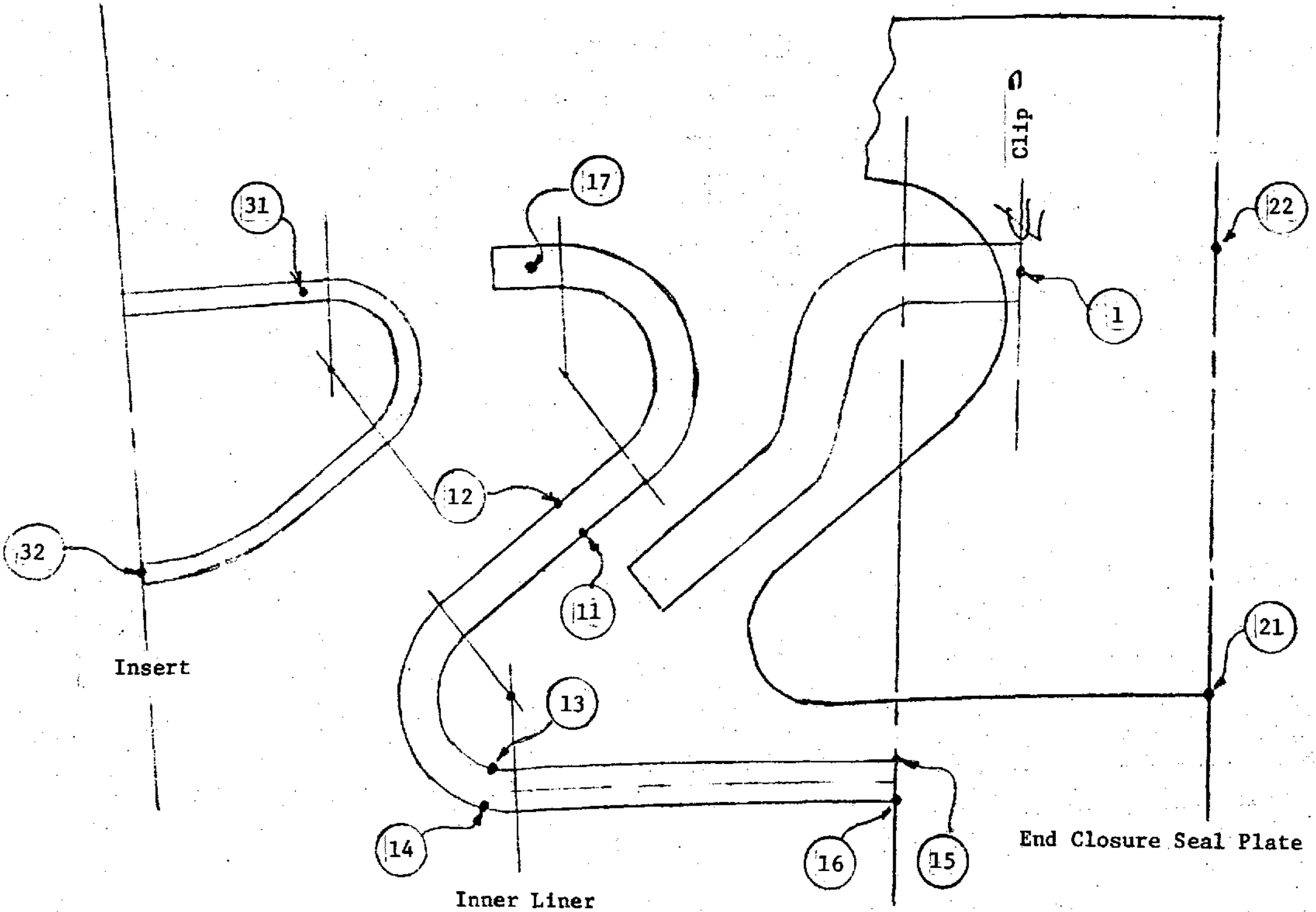


Figure 29. Stress Analysis Locations.

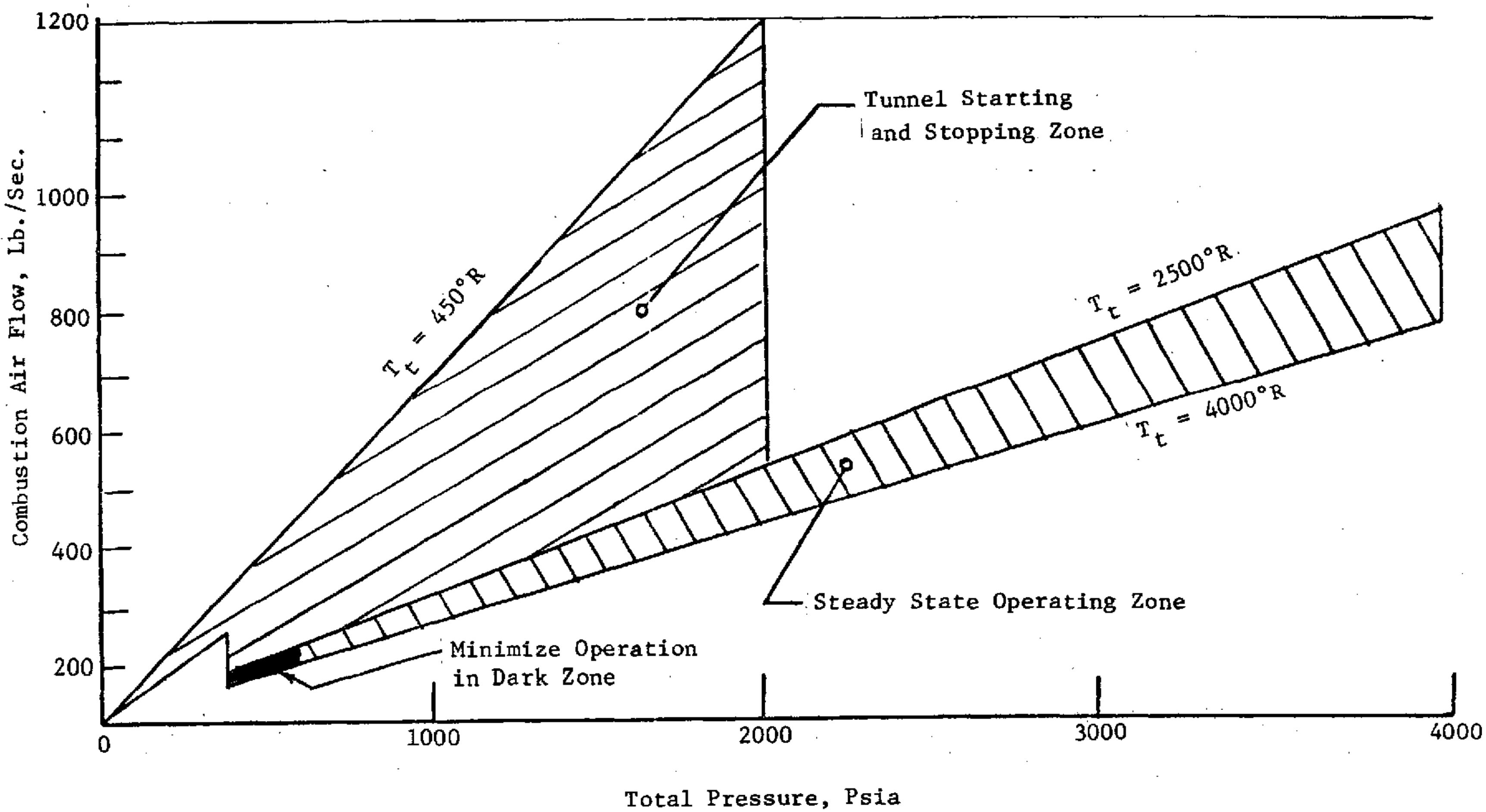
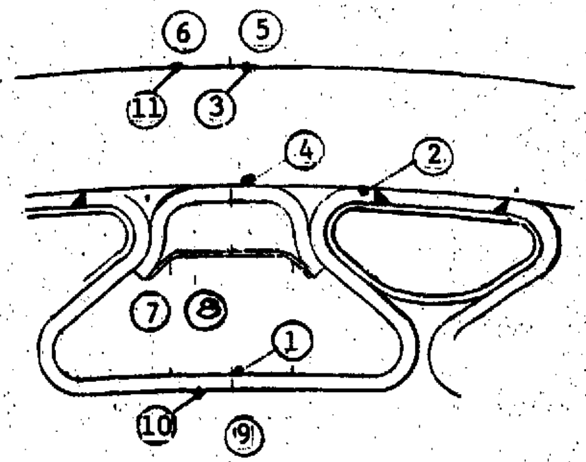


Figure 30. Operating Envelope for Omega Liner.



Location	Type	Qty.	Axial Spacing Inches	Remarks
1	T_{Metal}	16	6	6" U/S 24" D/S of Spray Bar
2	T_{Metal}	8	18	18" U/S 36" D/S of Spray Bar
3	T_{Metal}	8	60	Full Length
4	T_{Metal}	4	60	U/S of Inner Liner
5	$T_{air}/P_{s_{air}}$	2/2	-	Entrance
6	$T_{air}/P_{s_{air}}$	2/2	-	Exit
7	$T_{air}/P_{s_{air}}$	2/2	-	Entrance
8	$T_{air}/P_{s_{air}}$	2/2	-	Exit
9	P_{air}	2	-	Entrance
10	Strain Gage	12	-	6" U/S of Spray Bar
11	Strain Gage	6	-	6" U/S of Inner Liner

U/S = Upstream D/S = Downstream

Figure 31. Liner Instrumentation Summary.

APPENDIX A

T-Bar Liner Design and Failure Summary

The present liner design arrangement used in the NASA Facility and shown in Figure A-3 consists of a 3/16 inch thick Inconel 600 cylinder with a smooth 36 inch ID and an OD surface with axial fins to promote cooling. The fins have a height of 3/16 inch, a width of 1/16 inch, a pitch of about 1/4 inch and run axially for the full length of the liner. The liner is cantilevered from a bolt attachment to the nozzle approach section at the combustor exit station. The liner has 32 axial T-bars equally spaced around the OD which are guided for the length of the liner in channels on the liner support. The liner support is a 1/2 inch thick AISI 304 cylinder with a 39-3/8 inch ID and a 14 foot length. The liner support is cantilevered from a bolt attachment to the outer housing at 11.33 feet upstream of the fuel distributor rings. The support has 16 equally spaced axial T-bars which are guided in channels on the ID of the barrel or outer housing. The laminated outer housing is made of ASTM A20-56 and has a 44.5 inch ID, an average thickness of 4.5 inches and a length of 30 feet.

The failure history of the present liner design indicates that preferential cracking occurred along the T-bar weld attachment, mostly in the vicinity of the 7 o'clock location (viewed from upstream) at approximately one to three feet downstream of the fuel distributor rings. This axial location was also noted to have the maximum air-side metal temperature of 1170°F measured on the liner during tests at gas conditions of 3300°R and 600 psia. These gas conditions are substantially below the limits of the required operating envelope and do not represent the severest conditions for the liner. The liner has also been tested at other conditions including gas temperature up to 3640°R and pressures up to 3360 psia which resulted in lower liner temperatures.

Examination of the failed liner segments revealed many axial cracks along the welds which attach the T-bars to the OD of the liner and short axial cracks with multiple fatigue nuclei in the roots of the fin grooves adjacent to the T-bar welds. Both types of cracks progressed radially through the wall. Fracture examination revealed the transgranular nature of the T-bar and the fin root cracks. Both types are typical of cracks produced by high cycles and high mechanical loads. The multiple nuclei found in the fracture surfaces also are indicative of extremely high stresses.

It was observed that cracks occurred with a greater frequency in a liner configuration where radial interference would readily occur between the liner T-bars and the liner support during the test.

Except for the cracks in the butt weld (which joins the ends of a flat plate rolled into a cylinder) at the 11 o'clock location, the failure history reveals a pattern of liner cracking at the angular locations of the fuel distributor ring support spokes. The distributor is cantilevered from two axial fuel supply pipes. The spokes rest on metal pads welded to the I.D. of the liner. These pads had been added after the liner showed the effects of severe pounding from the spokes.

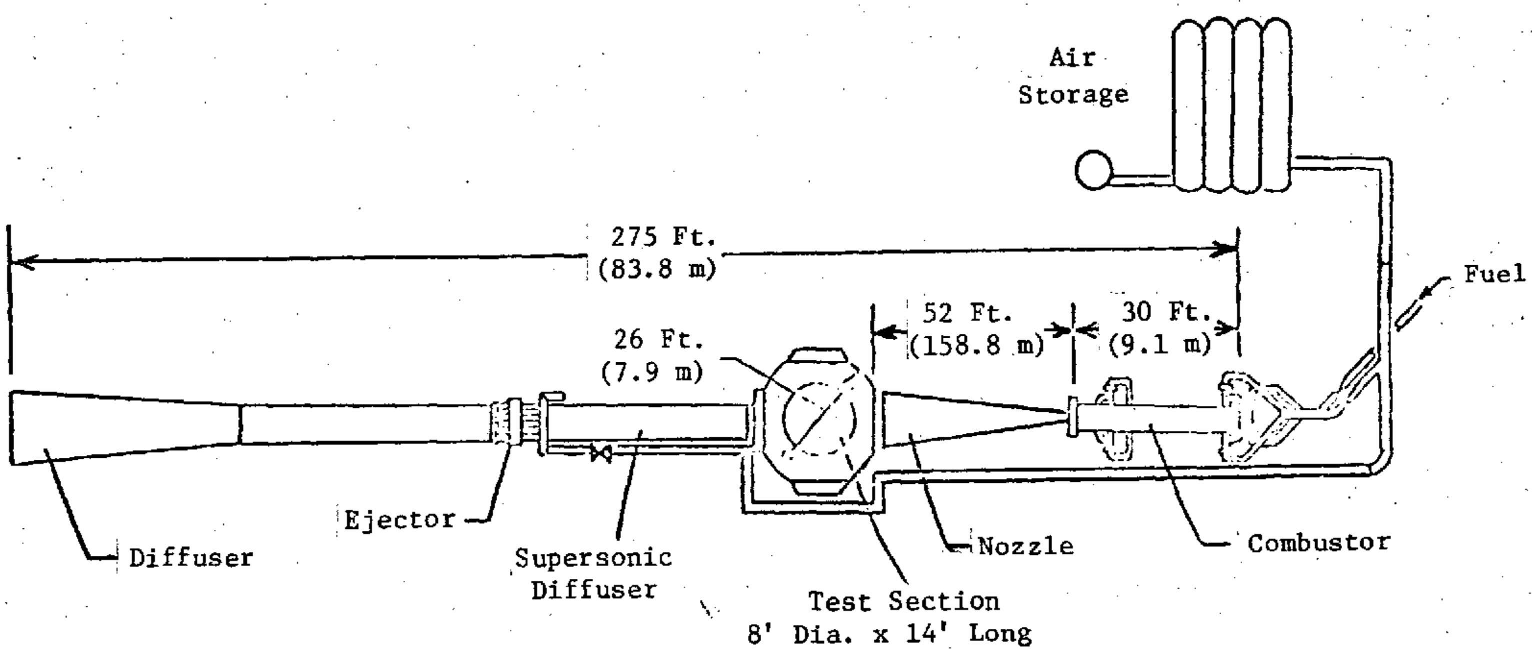


Figure A-1. Sketch of the 8-Foot HTST.

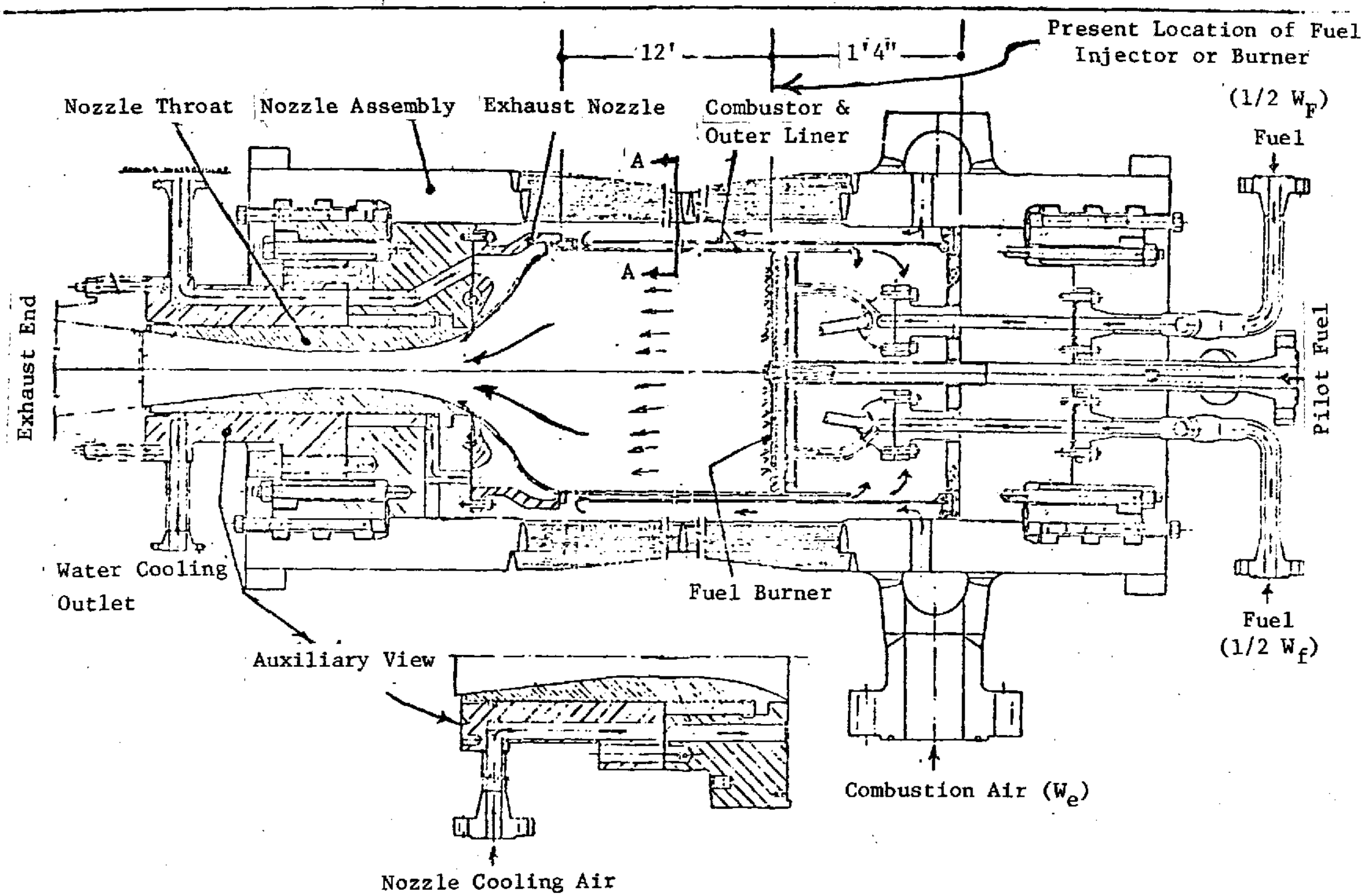


Figure A-2. Layout of High Pressure Combustor.

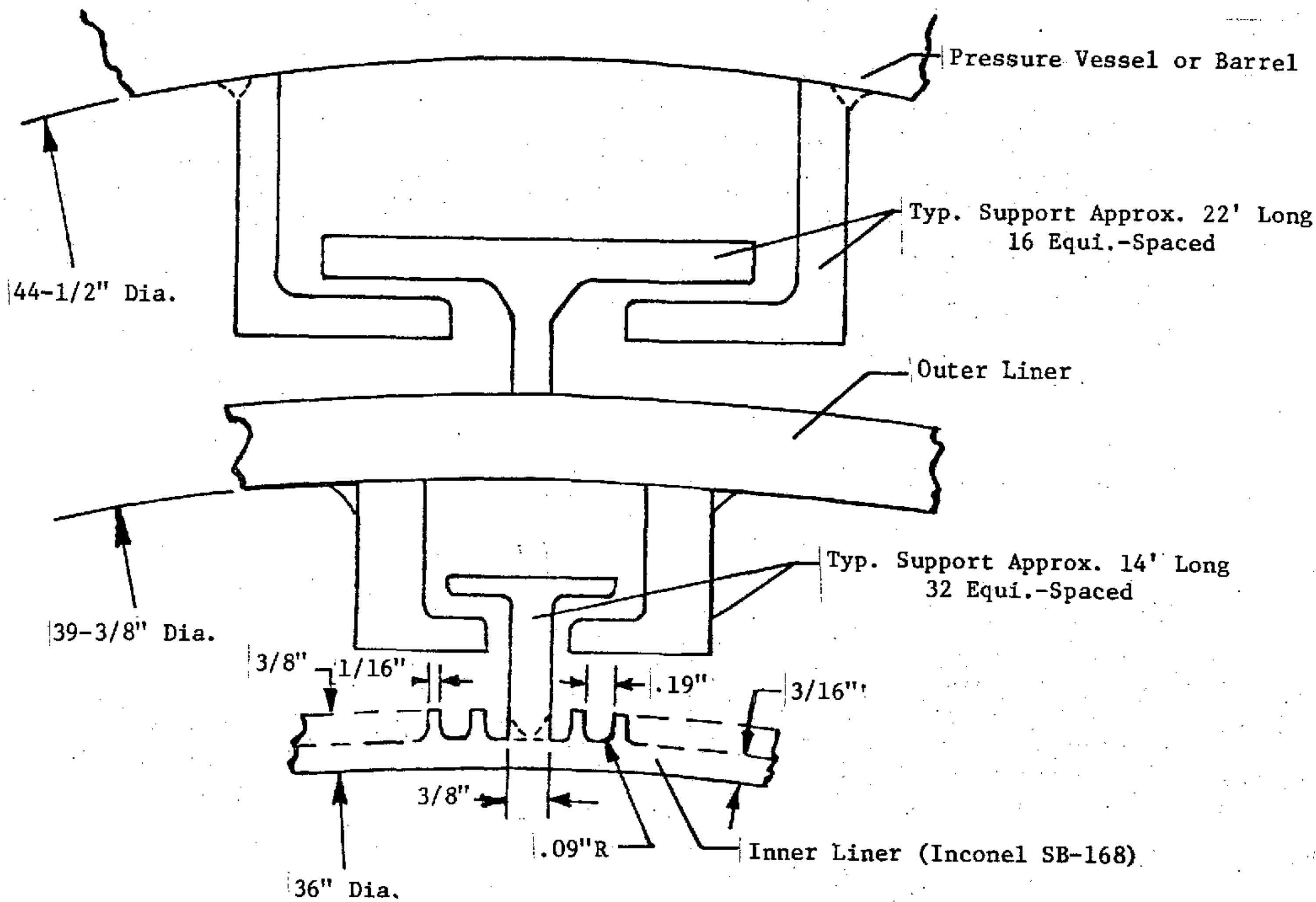
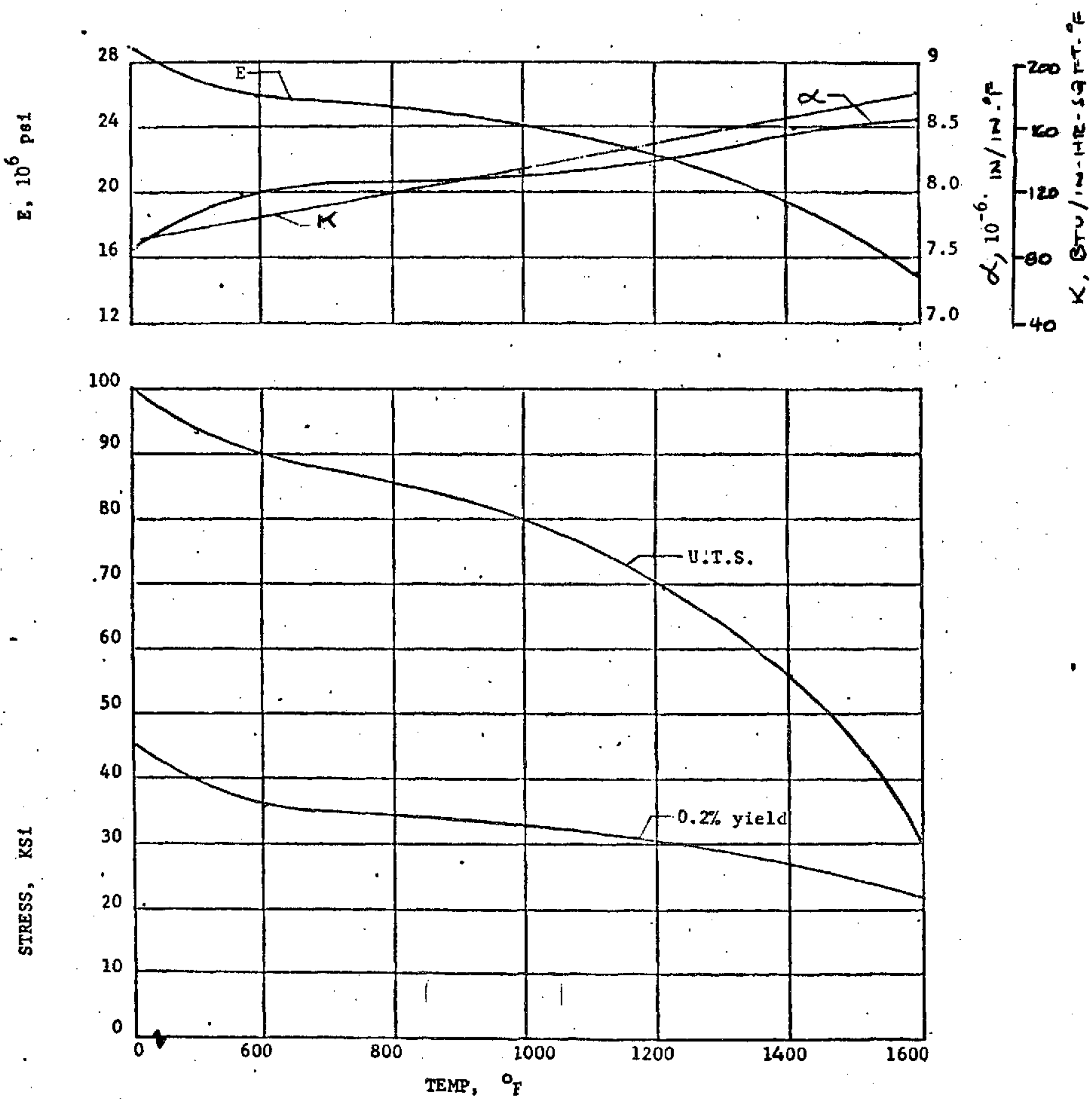


Figure A-3. T-Bar Liner Design Arrangement.



Ref. High Temperature Alloys
& Investment Cast Steels.
Haynes Technical Data Manual

Figure A-4. Hastelloy X Material Properties

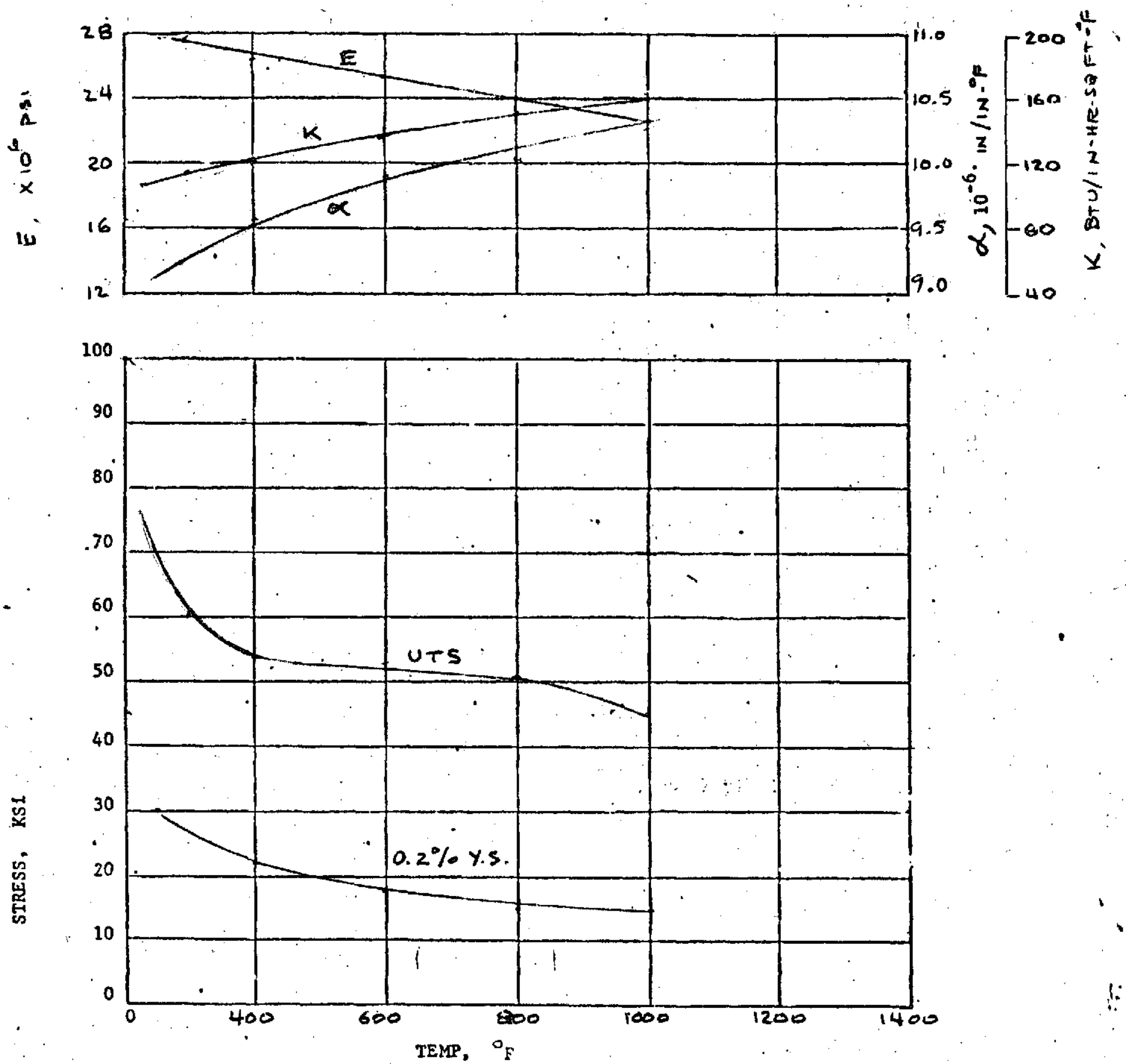


Figure A-5. AISI 304 Material Properties

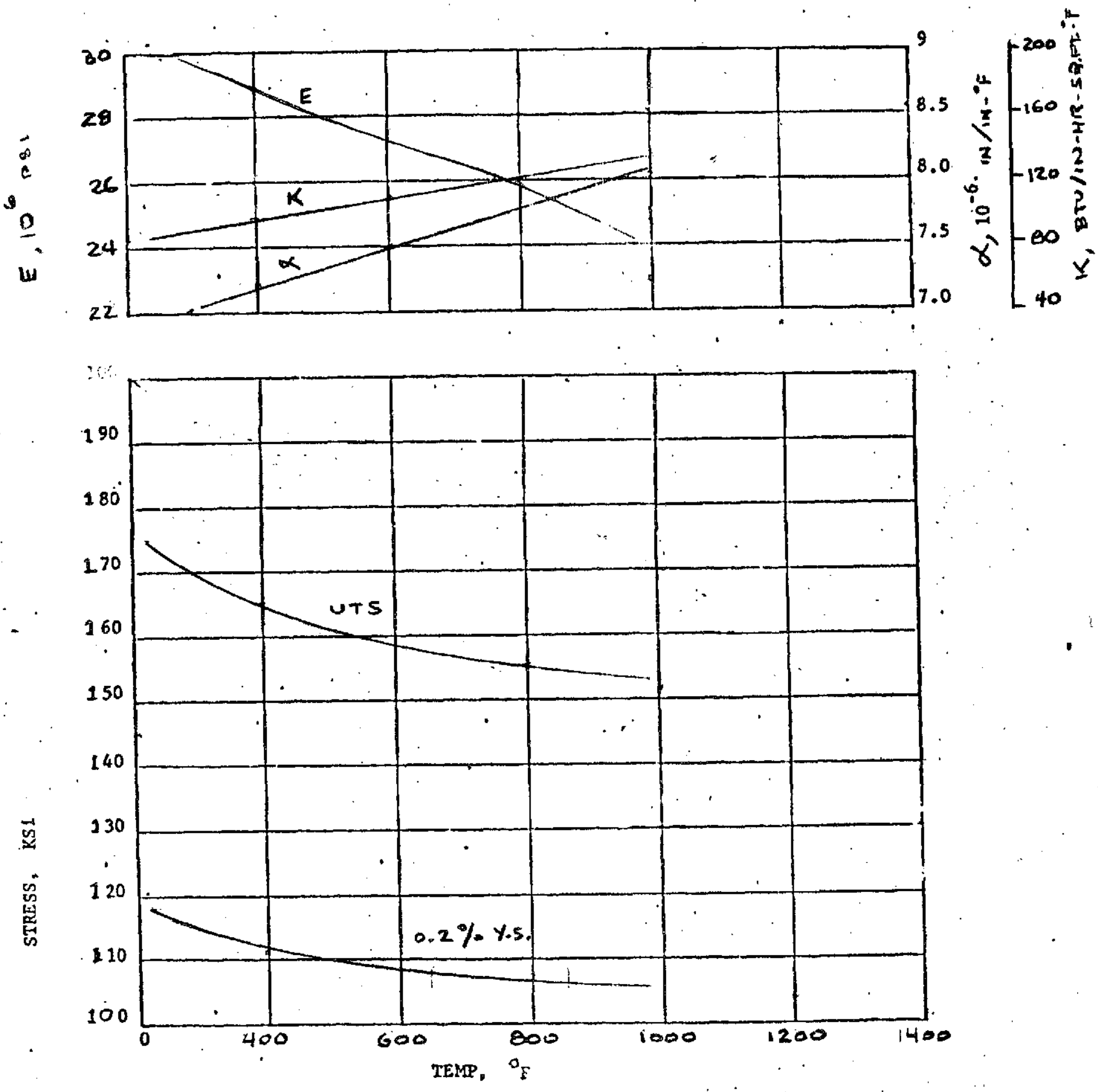


Figure A-6. Inconel X Material Properties