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UCRL-7679

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University of California

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ENGINEERING DESIGN OF THE TORY II-C
NUCLEAR RAMJET REACTOR

AEC RESEARCH AND DEVELOPMENT REPORT

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Nuclear Reactors for Ram-Jet
Propulsion, C-90
M-3679 (38th Ed.)

UNIVERSITY OF CALIFORNIA
Lawrence Radiation Laboratory
Livermore, California
AEC Contract No. W-7405-eng-48

ENGINEERING DESIGN OF THE TORY II-C
NUCLEAR RAMJET REACTOR

(Title: Unclassified)

Edited by C. E. Walter

November 1964

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ABSTRACT

The Tory II-C nuclear ramjet reactor is the culmination of the Lawrence Radiation Laboratory's development efforts in the Pluto program from 1957 to 1964.

This report presents details of the design and assembly of the reactor. Included are descriptions, criteria, analyses, and computations of the core components, axial and transverse reactor support systems, and reactor controls. In addition, a detailed description of the assembly of the reactor is profusely illustrated. Not included are discussions of reactor control actuators, electronic systems, etc.

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FOREWORD

This report provides detailed information on the engineering design of the Tory II-C reactor.

The data provides a record of the reactor design and assembly for the benefit of all persons directly associated with the Tory II-C reactor experiment, as well as the Atomic Energy Commission and Department of Defense programmatic personnel.

The criteria, analyses, calculations, etc., are furnished in enough detail to indicate the problems encountered and their solutions. Supplemental or related information sources are referenced throughout the text; the documents are listed in References. Many of these are internal publications, but they may be made available on request.

Since preparation of this report began, the reactor was successfully tested and the program terminated. Results of the reactor tests will be published separately. (UCRL-12069)

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ACKNOWLEDGMENTS

The design, assembly, and testing of the Tory II-C reactor was not the work of one person or even a small number of individuals. Rather, it was the result of the combined, coordinated teamwork of all personnel assigned to the Pluto project.

Similarly, this report is the product of the efforts of many of these people. Direct contributors to this publication effort are:

Section II - C. Walter, G. Patraw, W. O'Neal

Section III - W. O'Neal, E. Jackson, R. Werner, G. Patraw

Section IV - A. Miller, G. Patraw, W. O'Neal, N. Brown,
E. Jackson

Section V - J. Hovingh, E. Jackson,

Section VI - A. Williams

In addition, acknowledgment is made of the efforts of all other members of the Reactor Design Section since the beginning of the Tory II-C work in April 1960.

The guidance provided by W. B. Myers, Propulsion Division Leader, and H. L. Reynolds, Pluto Program Manager, is implicit in the engineering design presented here.

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I. INTRODUCTION

A. General Description of the Pluto Program

In 1957, the Lawrence Radiation Laboratory (LRL) was requested by the U. S. Atomic Energy Commission to develop a nuclear reactor capable of propelling a ramjet vehicle. Attention was directed to a vehicle that would travel at supersonic speeds over long distances and at very low altitudes. Ambient air would enter at the intake end and pass through the reactor, where it would be heated to high temperatures and forcibly discharged rearward through a nozzle, imparting thrust to the vehicle. Enough thrust to transport large thermonuclear warheads was desired.

To meet these ends, a program was established at LRL to design, develop, and test a series of high-power, high-temperature nuclear reactors. The first of these, Tory II-A, was successfully tested several times during 1961 at the Nevada Test Site (NTS). This reactor was designed with facility limitations in mind, but nevertheless provided pertinent information on fuel element integrity under realistic operating conditions. The engineering feasibility of the engine concept was thereby established, and invaluable design and operational experience was acquired. Not only were the desired operating conditions attained, but the reactor itself withstood extreme temperature and power levels for the desired time.

Tory II-C, the second reactor of the Pluto program, is meant to be adequate as an engine for a ramjet vehicle. Problems such as control rod operation within the reactor environment, which were avoided in the Tory II-A design, are inherent in Tory II-C.

The reactor was tested in the early months of 1964, before the Pluto program was terminated in July 1964. The tests were eminently successful; results will be published separately. Preliminary data indicates that design expectations were fully met or surpassed.

B. General Description of the Tory II-C Experiment

Following a zero power blowdown of the reactor at design air flow rate and inlet temperature, initial nuclear operation of Tory II-C was at 65% full power to check out reactor aerodynamic, heat transfer, and neutronic design predictions. One high-power run was made before the program was cancelled. At least three additional high-power runs had been planned.

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For operation the reactor was mounted into a duct system on a railroad flatcar (Fig. I-1) for movement between the test bunker and disassembly building. A remotely controlled, battery-powered prime mover was used to move the test vehicle. A fail-safe braking system on the test vehicle prevented any but intended movements.

The test facility (Fig. I-2) was designed to deliver 1900 lb/sec of air at 1060° F to the reactor for 300 seconds. This capability permits simulation of the Mach 3 flight condition at 1000 feet elevation. These conditions are met through provision of 1.2 million pounds of air stored in an extensive array of long 10-inch-diameter pipes. The air is heated by passage (in parallel) through four pebble-bed heaters containing 2 million pounds of 1-inch-diameter stainless steel balls preheated to 1200° F. The flow rate and temperature are adjustable during operation by valve manipulations. Flow meters and pressure and temperature transducers permit determination of flow rate.

Before entering the reactor, the coolant air flows through an aerodynamic grid located 20 feet forward of the reactor. Normally, this grid is choked and the air supply is decoupled from reactor transients. This also permits reliable determination of the flow rate through measurement of temperature and pressure immediately upstream of the grid. After passing through the reactor, the air enters a convergent nozzle which exhausts to the atmosphere.

The nozzle design does not adversely affect the pressure profile across the reactor, and bypasses the need for cooling a divergent section.

C. General Description of the Tory II-C Reactor

The Tory II-C reactor consists of a reflector core, structural components, and a reactor duct. (Figure I-3) The duct is a flanged circular cylinder 103 inches long and 57 inches in diameter, and contains the fueled core which is 51 inches long and 47-1/2 inches in diameter. Structural components include the support grid, tie rods, side support springs, peripheral shims, and base blocks. A summary listing of the components, materials and weights, in the reactor is given in Table I-1. Table I-2 gives the critical temperature limits established for reliable ground operation of the reactor. Material properties and reactor component designs and analyses are presented in detail in subsequent sections of this report.

Table I-1. General summary of components and materials in Tory II-C.

Component	Material	Weight	
		(lb)	(%)
<u>Reflected Core</u>		10,300	56.0
Ceramic tubes	{ BeO BeO-OyO ₂ ^a -Y ₂ O ₃ -ZrO ₂ Ni Cr-MgO		
Peripheral shims			
Nickel tubes			
Transition parts			
<u>Base Blocks</u>	F-48	450	2.0
<u>Tie Rod Assemblies</u>		700	4.0
Tie rods	{ Rene' 41 SS 410 Hastelloy C Pt-Ru		
Reflector inserts			
Cell plates			
Compression springs			
Spring adapters			
<u>Support Grid</u>	Rene' 41, Hastelloy C	900	5.5
<u>Control Rods</u>	Hafnium, Rene' 41	100	0.5
<u>Side Support System</u>		1,900	10.0
Springs	Rene' 41		
Rails, bars, and wipers	Hastelloy C		
<u>Reactor Duct</u>	Hastelloy C	4,000	22.0
		18,350	100

^aOy is uranium, highly enriched (93.5%) with U²³⁵.

Table I-2. Critical temperature limits for ground tests.

Component	Material	Critical stations	Temperature limits (°F)		Remarks
			Maximum	Minimum	
Duct 5	Hastelloy C	338 359 406 430 441	1500	-	Limit established by occurrence of carbide precipitation rather than stress.
Control rod	Hafnium (Rene' 41)	389 401	1900 (1/2 hr) 1600	- -	Limit established by oxidation and lack of strength data above 1900° F. If safety strap confidence is not desired, 1600° F temp may be violated.
Tie rods	Rene' 41	415	1700	-	Yielding can be expected above this temperature. For 30% load conditions allow 1800° F. Radiation damage may be a problem.
Support grid	Rene' 41 Hastelloy C	Aft surface -	1600 1500	- -	---
Comp springs	Inconel X	-	1200	-	Loss of compressive load.
Cell plates	410				Negligible stress.
Adapters	410				
Spacers	410	-	1435	-	
Reflector inserts	410				
Base blocks	D-40	-	2430	800	Ignition may occur above upper temp. Brittle failure possible below lower temp.
Cartridges	Cr-MgO	-	2430	-	To limit baseblock temp.
Shims	Nickel	Inside Surface. Finger Region.	1800 1300	- -	
Springs	Rene' 41	-	1400	-	Independent of operating condition
Rails	Rene' 41	-	1500	-	---
Wipers	Rene' 41	-	1400	-	---
Wiper bars	Hastelloy C	-	1500	-	Low stress.
Shroud	Rene' 41	-	1700	-	---
Nickel tubes	Nickel	-	1800	-	Negative stress.

About 47.5% of the gross reactor volume consists of hollow, hexagonal, beryllium-oxide tubes. These comprise the homogeneously fueled moderator and most of the reflectors making up the reflected core. There are approximately 465,000 tubes having a typical length of 3.92 inches; they are either unfueled or loaded to various degrees with highly enriched uranium dioxide. The hollow tubes are close-packed to provide a honeycomb pattern of about 27,000 parallel flow channels running the length of the reactor. The flat-to-flat dimension of the hexagonal fuel tubes is 0.297 inches, giving a fuel tube porosity of about 53%. Heat released in the fission of U^{235} is conducted to the channel walls and transferred by convection into the air passing through the channels. The ratio of the fueled, channel-flow area to the overall cross-sectional area of the reactor is 0.33.

The core is reflected on all sides. The forward reflector is 9.7 inches thick, while the aft reflector is 2.4 inches thick. Both are essentially composed of hexagonal beryllium oxide tubes. The aft reflector contains chromium-cermet transition pieces which manifold seven fuel tube passages into one passage. The side reflector is composed of a ring of hexagonal BeO tubes in immediate contact with the core. The ring is nominally 2 inches thick. Nickel peripheral shims, nominally 1 inch thick, surround the BeO.

The reactor is designed for flight capability at low-altitude, hot-day Mach-3 conditions for periods of 3 to 10 hours. It must sustain a thrust load of 280,000 pounds due to air flow, air pressure up to 342 psia, and lateral maneuver loads as high as 4 g. This force picture appears in a high-temperature environment where metal temperatures range to 2370° F and ceramic temperatures range to 2650° F. All cooling is accomplished with ram air, the temperature of which is 1060° F.

The forces are accommodated and the reflected core integrity maintained by side and axial support systems located in a high pressure duct. The side support system occupies the annulus between the reflected core and the duct. It consists of 32 bands of corrugated springs which exert a uniform radial compressive force on the reflected core. The irregular radial boundary formed by the hexagonal tube columns meshes with the inner surface of thick nickel shims to assure a stable geometry. The corrugated springs hold the nickel shims in place, but are flexible enough to accommodate thermal expansion of the reflected core. Transverse inertia loads on the reflected core, which would arise during missile maneuvers, are transferred to the duct by

30 support rails bolted to the inside of the duct. These rails engage "valleys" of the springs. The springs engage fingers in the nickel shims in a similar manner.

Axial loads are reacted at the front (cooler) end of the duct. Here a support grid spans the duct and is pinned to an interrupted ledge on the inside of the duct. Attached to the support grid are 121 Rene' 41 tie rods that pass through the reflected core and attach to base blocks at its rear. These base blocks are made of F-48 alloy, coated to prevent oxidation corrosion. A nominal axial compression is applied to the reflected core at each tie rod location by coil springs compressed between the support grid and the front of the reflected core. Thus, the bundle of almost one-half million parts comprising the reflected core is held in the desired position and geometry by side spring pressure and prestressed "hangers" or tie rods. All structural parts operate below 1600° F, permitting the use of super-alloys, except for the base blocks at the rear of the reflected core. Their operating temperature may be as high as 2370° F.

Nuclear control is achieved through the movement of hafnium control rods within the core. Twelve of these, called shim rods, are situated approximately 9 inches off axis, while the remaining two are closer to the reflector. One serves as a vernier rod, the other as a safety rod. All move axially within especially large tie rods.

Normal shim rod movement is mechanically restricted to less than 3 inches/second. Scram time, however, is approximately 1.5 seconds. Four actuators located within the duct move the shim rods; each actuator handles a set of three shim rods. The safety and vernier rods have separate actuators.

An independent safety system composed of six poison rods is also mounted within the ducting. This system is designed to yield a standby shutdown capability. It may be used only when the reactor is at zero or low power, because entry of these rods into the corresponding centrally located tie rods restricts coolant flow in the tie rods. The system is mechanically actuated from outside the duct to provide a high degree of reliability.

Additional information on Tory II-C neutronics, heat transfer, etc., may be found in UCRL-7272, "Tory II-C Safety Analysis Report," and UCRL-7315, "Tory II-C Data Book."

Section VII contains transverse and longitudinal cross-section drawings showing ceramic, structural, and safety system components. Also included is a detailed parts list, which is keyed to the drawings.

Fig. I-1. Tory II-C test vehicle.

Fig. I-2. Test bunker and air storage facilities.

Fig I-3. Reactor schematic.

II. CORE COMPONENTS

A. General

1. Dimensional Criteria

The overall dimensions of the reflected core are given in Fig. II-1. Sizing of the reactor was based on system performance analyses made by the LRL staff. Conclusions were tempered by the size and complexity of the air supply which would be required for ground tests. The size of the active core was chosen without a specific missile or reactor design in mind. Rather, ground rules were established for a reasonable vehicle and payload which could be propelled by 2500° F reactors of three general types at Mach 2.8, under hot day, sea level conditions. These criteria and the results of the original system analysis are stated in considerable detail in Ref. 1. Briefly, they are:

- Maximum missile diameter at reactor (performance)
- Critical mass less than 100 kg (cost)
- Maximum BeO wall temperature of 2500° F (water attack)
- Missile design Mach No. 2.8; ambient temperature of 100° F, altitude of 1000 feet (reasonable mission)
- BeO moderator and reflector material density: 96% (existing technology)
- 20% thrust margin for maneuver (reasonable mission)
- Diffuser pressure recovery 80% with 4% bleed with 80% recovery of bleed (existing technology)
- Nozzle divergence factor of 0.98 and nozzle velocity coefficient of 0.97 (existing technology)
- Core length 51.2 inches, with 7-inch front reflector and a 2-inch aft reflector (thrust relatively insensitive to core length)
- Reactor exit Mach No: 0.5 (LRL studies)
- External missile drag coefficient (with maneuver) based on maximum missile diameter: 0.168 (Air Force and LRL studies)
- Flight time, 10 hours, with 3 hours at sea level (reasonable mission)
- Radial power, flat (optimum).

Reactor dimensions, together with dimensions of the final design, are compared below:

	<u>Early 1960 chosen dimen- sions (inches)</u>	<u>Final Tory II-C (cold) dimen- sions (inches)</u>
Active core length	51.2	51.03
Active core diameter	48	47.46
Side reflector thickness	3	2.98
Core plus side reflector diameter	54	53.42
Forward reflector thickness	7	9.67
Aft reflector thickness	2	2.37
Missile diameter (max.)	56.5	57.23

The principal change was in the length of the forward reflector. The original axial power distribution resulted from uniform axial fuel loading. It was later decided to raise the power at the front end by increasing both the front end loading and the front reflector length. Raising the power at the front end permits greater heat transfer from the reactor for fixed peak wall temperature. Since the peak power density is reduced, thermal stress in the fueled tube is also reduced.

2. Reactor Type

There are three ways of reacting the axial pressure drop load on the reactor:

- (1) Load reacted at front (cool) end. This type reactor would have longitudinal tie rods and base blocks, with the tie rods attached to a support grid at the front of the reactor.
- (2) Load reacted at aft (hot) end by a large ceramic dome. This type reactor would not require any internal structural metal parts.
- (3) Load reacted along the length of the reactor. This type reactor would have transverse beams or "trays," each supporting a fraction of the reactor pressure drop.

In the original system performance analyses, all three types of reactors were considered. These studies indicated that the first two types had better performance capabilities, regardless of size.

In early 1960, the tie rod type of reactor was decided upon as the most appropriate choice consistent with the program schedule. (At the same time, the considerable potential of the ceramic dome concept was recognized and further development along this line was continued.) The technology of tie rod reactors was well understood and was merely an extension of the Tory II-A design. However, lateral loads for a flying reactor needed to be reacted in a different manner than that employed to react the ground loads of Tory II-A.

To avoid neutronically costly metal structure in the reactor, transverse support considerations required that the fuel tubes be close-packed. In this way the integrity of the matrix would be maintained by appropriately applied external forces.

3. Unit Cells

The arrangement of the cross-section of the reflected core is basically an array of hexagonal ceramic tubes. The uniformity of the array is systematically interrupted by the tie rods which extend the full length of the reactor. The tie rods replace a number of the usual ceramic tubes. Also, a ring of special ceramic tubes surrounds each tie rod. These "guard" tubes are sized to fit loosely in the array and are unfueled. Each tie rod assembly and base block is associated with a surrounding, roughly hexagonal bundle of ceramic tubes. This repetitive group of parts is called a unit cell. Each unit cell is similar in geometry and dimensions, except that those unit cells which accept control rods have larger diameter tie rods. A cross-section of neighboring standard and control unit cells is shown in Fig. II-2.

4. Peripheral Cells

Near the periphery of the reactor, unit cell geometry is modified to conform to the reflected core boundary. A typical sextant is shown in Fig. II-3. Peripheral cells each contain two tie rods. This is required because of their greater size, and because in early versions of the design of the reactor, transverse support system high friction resulted on the boundaries of the reflected core. The resulting moment on the base block associated with a single peripheral tie rod was unopposed, and

"rocking" of the base block could result. The use of two tie rods along a generally radial line in the peripheral cells tends to minimize the effect of this moment. The final design of the side support design is such that relatively low frictional forces are present.

The number of tube column positions within the peripheral shims is 27,163. Of these, 21,012 are coupled by fueled tube channels; 1200 are occupied by guard tubes around the tie rods; and 3888 are occupied by ceramic tubes in the side reflector. Less than 4% of the tube positions (1063) are occupied by the tie rods themselves.

The boundary between the active core and the side reflector was chosen to provide an equivalent core cylinder 48 inches in diameter at operating temperature. Further, the boundary was chosen to accommodate groups of seven tube positions, making it compatible with the chromium-cermet manifolding arrangement used in the aft reflector.

B. Ceramic Tubes

1. Purpose and Description

The ceramic tubes used in the reflected core comprise the bulk of the moderating, fissionable, and reflecting material of the core. These tubes are either unfueled or have one of 26 discrete fuel concentrations; with one of three different bore diameters, several lengths, and one of three slightly different across-flat dimensions for their hexagonal cross-section. The combinations result in 94 different types of fueled tubes and 23 different types of plain tubes. A total of 464,595 ceramic tubes is used. The tube cross-sections shown in Fig. II-4 were selected on the basis of:

- Ability to maintain mechanical integrity of the beryllium oxide moderator and or alloy oxide fuel while the material is generating power and transferring heat at high subsonic air flow rate;
- Ease of extrusion, coupled with "packing efficiency" in a matrix;
- Ability to vary the fuel loading within narrow radial bands in the reactor;
- Experience with Tory II-A fuel element fabrication.

The across-flat dimension of the Tory II-C fueled tubes was chosen to be the same as for Tory II-A. Not only did this yield a near optimum L/D ratio for the flow channel and acceptable thermal stress, but also permitted use of readily available inspection tools and mockup parts. Tube lengths vary from 1 to 4 inches. The upper limit was set to simplify extrusion of straight tubes. Several lengths were required to obtain a stagger of tube ends in adjacent tube columns.

The only other shape seriously considered was a multiholed hexagonal plate about 1 inch thick and 5 inches across. This plate was not adopted because of the inability to obtain uniform density, the difficulty of varying the loading radially in small bands, the difficulty of fabrication, etc.

2. Material Selection

Beryllium oxide (BeO) is the only moderator material available that can be used successfully in the hotter regions of the reactor. With reasonable porosity (50%) for coolant flow, the critical reactor diameter is in the size range of interest. BeO possesses the necessary resistance to oxidation and sufficient strength at elevated temperature. Its principal limitation is the reaction with water vapor which becomes significant only above 2500° F. (There is adequate strength above this temperature, probably to 3000° F.) The average modulus of rupture strength of BeO tubes with 98.3% theoretical density is 39,000 psi at room temperature and 15,000 psi at 2500° F. Fueled BeO can have respectively higher strengths, depending upon loading concentration and firing temperature. Additional strength data on fueled and unfueled tubes is given in Ref. 2.

The fuel used is oralloy dioxide (OyO₂). Oralloy is uranium containing 93.5% Uranium-235. This degree of enrichment is warranted by performance requirements. OyO₂ can be homogeneously dispersed in BeO, and is compatible with it when proper precautions are taken. References 3 and 4 contain discussion of the stabilization of fueled material, together with data on the characteristics of the ceramic tube material. Additional data can be found in many of the Tory II-C Quarterly Reports. Beryllium metal could have been used in the forward reflector; however, to reduce the number of design variables, BeO was used in the forward reflector as well.

3. Fabrication

The beryllium oxide used in the fabrication of the tubes was produced by the sulphate process. A very pure grade of powder referred to as UOX in the industry was used for both fueled and plain ceramic tubes. This material was purchased under an LRL specification⁵ and carefully analyzed for conformance to its requirements. Fueled tube starting ingredients included uranyl nitrate, zirconyl nitrate, and yttrium nitrate. A very careful quality assurance program was maintained to insure that these materials conformed to their specifications.⁶⁻⁸

Other raw materials required in the tube fabrication process are water, for making slurries; ammonium hydroxide, to precipitate urania, yttria, and zirconia from their nitrate solutions; and binders. These miscellaneous materials were supplied by the tube fabricator, and there was no direct LRL control on their purity.

The basic extrusion process used to fabricate the tubes was developed at LRL. The process is generally described in several papers collected in Ref. 3.

Considerable significance was given to the meaning of a "mix batch" as a means of controlling the process. A mix batch is composed of several materials which may come from various sources and have different histories; i. e., from two or more uranyl nitrate containers, several BeO shipping drums, etc. Since in a mix batch the material was homogeneously mixed, any portion of the mix batch was thought to be representative of the entire batch. Mix batch identification was retained (to the extent of permanently marking each tube) so that specific tube performance could be correlated with fabrication history. It later turned out that during assembly operations the mix batch identification number was useful in other ways. (See Section VI-B, Ceramic Assembly.)

4. Tube Characteristics

All ceramic tubes were procured under an LRL specification.⁹ In addition to the normal drawing requirements, the specification required that the tubes have certain other characteristics.

a. Composition

The concentration of impurity elements to be less than the values given in Table II-1.

The summation

$$\Sigma (c/q) \leq 16$$

for the impurity elements listed in Table II-2, where c is the analyzed concentration of the impurity element, and q is the value tabulated in the table.

Impurity elements not listed in Tables II-1 or II-2 not to be present in concentrations greater than 80 ppm.

On the average, these impurity limits were met with ease. In a few instances concentrations in excess of the maximum specified were analyzed. Neutron economy was the primary consideration in establishing the limits. In addition, a large amount of impurities could result in poor high-temperature performance.

The maximum effect on reactivity of the impurities permitted by the specification was 2 to 3%. The effect on reactivity of the impurities actually present in the ceramic tube was 0.5%.¹⁰

Table II-1. Maximum permissible impurity concentrations.

Impurity element	Concentration (ppm)*
nickel	40
chromium	50
molybdenum	70
magnesium	150
aluminum	200
silicon	200
iron	300

* ppm = weight of impurity element divided by weight of ceramic tube times one million.

Table II-2. Importance factors for impurity elements.

Impurity element	Importance, q
gadolinium	1
boron	4
samarium	5
cadmium	6
europium	8
cobalt	12
lithium	20
indium	20
maganese	25
dysprosium	30
thulium	30
hafnium	30
gold	30
rhodium	40
sodium	50
titanium	50
silver	50
erbium	50

The composition of the fueled tubes was further controlled by specifying the fuel and stabilizer concentration. The fuel concentration was specified as grams per inch of tube length rather than weight %, since the former is a more direct measurement. Twenty-six fuel loading classes differed in fuel concentration by 8%. The lightest class contained 0.0219 gram UO_2 /inch of tube (1.23%), and the heaviest class contained 0.1591 gram UO_2 /inch of tube (8.12%). The different loadings were required to obtain a flat radial power distribution and the desired axial power distribution.

Although the tolerance on fuel loading concentration was large ($\pm 4\%$), the average weight of total urania (UO_2) per unit length of tube for all fueled tubes was controlled within 1% of the nominal value of each loading concentration.

b. Density

The density of unfueled tubes was required to be greater than 2.9 g/cm^3 . The average density for each size code of plain tubes is given in Table II-3. Note that in all cases the average density is greater than 2.9 g/cm^3 .

The density, D , of fueled tubes was required to be greater than the value calculated from the relation:

$$D = \frac{2.960}{w_1 + 0.277 w_2 + 0.570 w_3 + 0.479 w_4}$$

where w_1 , w_2 , w_3 , and w_4 are the weight fractions of BeO , UO_2 , Y_2O_3 , and ZrO_2 respectively, and their sum is 1.

When the minimum density given by the above relation is plotted against the weight of fuel per inch of tube, a straight line results. This line, together with the average density data actually obtained during production (Tables II-4, 5, 6, and 7) is shown in Fig. II-5. Note that the average density is above the minimum density line in all cases.

c. Ceramic defects

Tables II-3 through II-7 indicate the quantities of tubes of the various size codes received and accepted at LRL. The reject rate was actually higher than indicated in the tables, since the tubes were inspected twice; the first time before shipment to LRL. The largest cause for rejection was for chips and cracks. Inspection for chips and cracks was by far the most time consuming, and in some instances the reasons for their presence were never determined. It was not clearly established whether or not tubes with cracks are "good" reactor parts.

Other defects looked for were inclusions and voids. Their presence was detected by radiography. The specification required initially that the ceramic tubes be free of any void or inclusion having a dimension greater than 30 mils. This was later changed to 35 mils. An appreciable fraction of the relatively few tubes rejected for these causes was saved by the latter criterion. In later stages of production, a very small percentage of the tubes produced was radiographed.

Table II-3. Inspection data summary for plain tubes.

Size code	Quantities				Avg density (g/cc)	Avg weight (g)	Avg flat (in.)	Avg bore (in.)
	Ordered	Re- ceived	Ac- cepted	Re- jected (%)				
B1	11,625	12,375	12,375	3.9	2.9497	1.741	0.2973	0.2266
C1	11,625	13,251	12,717	4.0	2.9507	2.614	.2972	.2266
E1	22,400	25,694	23,288	9.4	2.9519	3.418	.2972	.2266
J1	24,000	26,231	24,569	6.3	2.9595	6.599	.2973	.2273
F1	10	43	42	2.3	2.9424	4.128	-	-
G1	70	151	148	2.0	2.9515	4.262	.2970	.2272
H1	50	67	64	4.7	2.9555	5.098	-	-
L1	1,850	2,261	2,194	3.0	2.9518	5.958	.2971	.2272
K1	21,200	24,556	23,129	5.8	2.9523	6.788	.2972	.2277
A2	450	475	464	2.3	2.9176	4.360	.2952	.1304
E2	850	866	856	1.2	2.9287	5.859	.2950	.1295
G2	450	699	690	1.3	2.9517	7.366	.2951	.1297
H2	675	691	681	1.3	2.9227	8.765	.2946	.1295
K2	15,100	15,969	15,787	1.1	2.9380	11.729	.2951	.1302
D3	3,780	2,967	2,933	1.1	2.9315	6.247	.2962	.0931
E3	9,400	10,004	9,851	1.5	2.9175	6.508	.2059	.0935
F3	1,350	1,621	1,584	2.3	2.9241	7.900	.2958	.0928
J3	3,800	4,027	3,925	2.5	2.9420	12.561	.2960	.0928
K3	48,500	51,964	50,593	2.6	2.9390	13.082	.2960	.0928
A3	125	140	138	1.4	2.9650	4.866	.2061	.0928
G3	2,200	2,374	2,332	1.8	2.9521	8.215	.2958	.0926
L3	350	1,159	1,112	4.0	2.9436	11.402	.2960	.0930
K4	3,500	3,824	3,756	1.8	2.9373	13.035	0.2959	0.0935

Table II-4. Inspection data summary for "U" fueled tubes.
(Avg flat = 0.2971 in., avg bore = 0.2269 in.)

Loading class	Quantities			Rejected (%)	Average density (g/cc)
	Ordered	Received	Accepted		
219	5,600	7,039	6,399	9.1	3.0373
237	12,300	15,623	13,330	14.7	3.0380
256	9,600	12,196	10,036	17.7	3.0491
277	9,900	14,043	11,611	17.4	3.0453
299	11,300	15,217	13,821	9.2	3.0538
323	10,200	11,328	10,456	7.7	3.0567
350	11,300	12,717	11,653	8.4	3.0699
379	12,800	14,172	12,976	8.4	3.0784
410	11,800	13,872	12,842	7.4	3.0830
443	12,500	14,148	12,945	8.5	3.0833
479	12,500	13,670	12,688	7.2	3.1012
518	13,200	13,973	13,349	4.5	3.1159
560	14,550	15,537	14,797	4.8	3.1236
606	20,900	21,855	21,037	3.7	3.1329
656	16,650	17,897	17,150	4.2	3.1490
710	38,500	40,054	37,858	5.5	3.1590
769	6,000	6,606	6,335	4.1	3.1627
832	7,100	7,748	7,423	4.2	3.1798
901	5,000	5,065	4,845	4.3	3.1915
976	5,000	5,604	5,309	5.3	3.2114
1058	3,500	3,885	3,691	5.0	3.2293
1147	3,700	4,372	4,154	5.0	3.2551
1243	3,900	4,462	4,158	6.8	3.2709
1349	2,400	2,508	2,421	3.5	3.2965
1465	2,300	2,363	2,254	4.3	3.3185
1591	2,500	2,549	2,377	6.7	3.3434

Table II-5. Inspection data summary for "T" fueled tubes.
(Avg flat = 0.2970 in.; avg bore = 0.2268 in.)

Loading class	Quantities			Rejected (%)	Average density (g/cc)
	Ordered	Received	Accepted		
219	370	450	411	8.7	3.0258
237	750	982	889	9.5	3.0417
256	550	739	676	8.5	3.0484
277	510	839	766	8.7	3.0508
299	630	1,151	1,077	6.4	3.0573
323	560	809	770	4.8	3.0601
350	530	655	601	8.2	3.0737
379	620	753	654	13.2	3.0850
410	580	646	628	2.8	3.0554
443	640	915	881	3.7	3.0899
479	480	716	697	2.7	3.0999
518	650	1,114	1,048	5.9	3.1117
560	660	717	696	2.9	3.1247
606	1,050	1,310	1,273	2.8	3.1307
656	820	1,028	1,004	2.3	3.1433
710	2,000	2,081	2,002	3.8	3.1563

Table II-6. Inspection data summary for "S" fueled tubes.
(Avg flat = 0.2970 in.; avg bore = 0.2269 in.)

Loading class	Quantities			Rejected (%)	Average density (g/cc)
	Ordered	Received	Accepted		
219	260	388	368	5.2	3.0364
237	480	592	554	6.4	3.0394
256	380	475	452	4.8	3.0431
277	350	533	497	6.8	3.0457
299	420	753	669	11.1	3.0492
323	360	450	412	8.4	3.0591
350	360	469	453	3.4	3.0415
379	420	555	530	4.5	3.0789
410	380	442	415	6.1	3.0908
443	420	634	619	2.4	3.0945
479	360	548	517	5.7	3.1033
518	400	553	536	3.1	3.1043
560	1,190	1,292	1,264	2.2	3.1223
606	1,280	1,436	1,394	2.9	3.1347
656	1,040	1,304	1,258	3.5	3.1433
710	1,760	2,055	1,998	2.8	3.1558
769	630	686	667	2.8	3.1672
832	1,290	1,757	1,696	3.5	3.1798
901	690	863	830	3.8	3.1879
976	1,380	1,640	1,567	4.5	3.2134
1058	600	786	746	5.1	3.2308
1147	650	692	663	4.2	3.2512
1243	810	937	913	2.6	3.2702
1349	2,130	2,138	2,057	3.8	3.2939
1465	1,320	1,419	1,351	4.8	3.3164
1591	1,440	1,488	1,420	4.6	3.3431

Table II-7. Inspection data summary for "R" fueled tubes.
(Avg flat = 0.2971 in.; avg bore = 0.2270 in.)

Loading class	Quantities			Rejected (%)	Average density (g/cc)
	Ordered	Received	Accepted		
219	260	279	261	6.5	3.0352
237	510	601	549	8.7	3.0249
256	380	506	445	12.1	3.0385
277	360	518	489	5.6	3.0501
299	440	528	510	3.4	3.0588
323	380	454	396	12.8	3.0493
350	370	459	427	7.0	3.0577
379	450	555	525	5.4	3.0849
410	390	410	405	6.0	3.0829
443	440	569	546	4.1	3.0935
479	340	443	439	0.9	3.1092
518	430	563	553	1.8	3.1059
560	1,340	1,590	1,541	3.1	3.1243
606	1,430	1,973	1,908	3.3	3.1280
656	1,130	1,195	1,156	3.3	3.1755
710	1,860	2,057	1,952	5.1	3.1598
769	890	1,013	943	6.9	3.1658
832	1,610	1,779	1,706	4.1	3.1719
901	860	927	865	6.7	3.1990
976	1,800	1,866	1,829	2.0	3.2128
1058	840	1,108	1,072	3.3	3.2261
1147	830	1,308	1,043	20.3	3.2510
1243	1,070	1,268	1,213	4.3	3.2657
1349	2,750	2,959	2,857	3.5	3.2922
1591	1,700	1,743	1,684	3.4	3.3419
1465	1,640	1,939	1,857	4.2	3.3213

d. Dimensions

A close tolerance on the across-flat dimension of the tubes was required to limit the variation to be expected in the size of the assembled reactor. The shrinkage of ceramic tubes could not be controlled well enough to permit the use of as-extruded tubes. Early inspection results for ground tubes indicated that a tolerance of ± 0.0001 inch on the across-flat dimension was reasonable. Accordingly, all ceramic tubes were extruded slightly over-size and ground externally. The variation in reactor diameter was further limited by specifying that the average of individual tube across-flat dimensions in each size code fall within ± 0.0005 inch of the nominal across-flat dimension. Tables II-3 through II-7 show that the average across-flat dimension was generally within 0.0002 inch of the nominal dimension. The across-flat dimensions of guard tubes (A2, etc.) and side reflector tubes (D3, etc.) were 0.295 inch and 0.296 inch, respectively.

The effect of camber and twist of ceramic tubes in the assembled reactor is to increase its diameter and make it "spongy." From the standpoint of transverse support design and neutronics, it is necessary to have a stable, known reactor diameter. This requirement pointed strongly toward allowing very little camber and twist. Side pressure is used to support the reactor against transverse body forces, and one might consider the deflection of a tube which can be tolerated without incurring tube breakage to establish allowable camber. This amount of deflection at the center of a tube is about 3 mils. Only about 70% of 4-inch long as-extruded tubes had less than 3 mils camber in the 1961 state of the art. For ground tubes, it is practical to specify a fairly close envelope in which the tubes are required to be contained. The tube drawings therefore call for all tubes to fit inside a geometrically true hexagonal prism 0.2985 inches across-flat and of sufficient length. (For guard tubes and side reflector tubes the corresponding envelopes were 0.2965 and 0.2975 inches, respectively.) This in itself permits small tubes to have considerable camber and twist. The control of the average flat dimension as discussed previously, however, prevents making tubes on the low side of the allowable across-flat dimension range to meet the envelope requirements.

The care taken with the transverse envelope of the ceramic tubes resulted in close achievement of the expected reactor diameter. The expectation is shown in Fig. II-6.

Early in the production of ceramic tubes an assembly was made up to simulate a full size reactor cross-section.¹¹ This assembly, one tube high, is shown in Fig. II-7. Measurements of the assembly established a stacking allowance of 0.0004 inch per tube.

This allowance was included in subsequent predictions of reactor diameter, as in Fig. II-6. Active core and reflected core radii were calculated from the number of tube positions, the nominal tube area, and the 0.0004-inch build-up per tube. The spring radius was calculated from the actual part dimension in the two planes of symmetry. The thickness of ceramic was obtained by a tube count, and the distance across the shim and spring assembly was measured by large scale lofting of the side support region. The two measurements differed by only 2 mils.

The tolerance bands shown in Fig. II-6 were obtained by combining the spring and shim tolerances with the specified ± 0.0005 inch allowable on the tube average dimension. Based on the first 90,000 production tubes, the most probable tube cross-flats dimension was 0.2972 inch.

In addition to its function for determining tube stacking build-ups, the dimensional cross-section test was used to estimate load concentration factors. When radial pressure is applied to a tube matrix, certain tubes are more highly loaded than others. This is due to small dimensional differences, and results in a statistical distribution of pressure loads throughout the matrix. A load of 10 psi was placed on the cross-section, and 1068 tubes were pulled and replaced consecutively. The load required to pull the tube was recorded. The load distribution is shown in Fig. II-8. The load concentration factor (defined as the ratio of the minimum load for the upper 5 percentile to the load for the median tube) was found to be 3. This is well within the strength ability of the ceramic tubes. It was also found that if a bundle of tubes were subjected to loads beyond crushing strength, the bundle still retained geometrical stability, and no flow holes were plugged.

Upon completion of the reactor side support spring installation, the reactor diameter was determined from circumferential measurements of the spring.¹² The build-up per tube was less than 0.0007 inch per tube in the region of the active core, and less than 0.0006 inch per tube in the region of the front reflector. The larger stacking build-up present in the reactor is thought to result because tube ends are staggered axially in the reactor. The

bore diameter of the fueled tubes was selected on the basis of performance optimization studies in which thermal stress and ease of tube fabrication were parameters. The bore diameter of guard tubes (tubes surrounding the tie rods) was chosen to limit the peak wall temperature of these tubes to 2000° F. This temperature is half way between the 2500° F fueled tube wall temperature and the 1500° F tie rod temperature.

The side reflector tube bore size was selected on the basis of one constant diameter hole per tube. The use of constant hole size in the side reflector produces a temperature gradient, since the nuclear heating drops off radially. This gradient is shown in Fig. II-9. Note that the temperature of the nickel peripheral shim matches the temperature of the BeO reflector tubes quite well.

The bore diameter tolerance for the three hole sizes used in the reactor was the same in each case. The ± 0.002 -inch tolerance is consistent with fabrication by extrusion. The bore of the tube was ground only when so doing permitted acceptance of the tube.

The tolerance on the bore diameter of the fueled tubes allows the porosity of fueled tubes to vary between 52.1 and 54%, or the BeO and fuel content to vary $\pm 1.9\%$. Because of the consequence on reactivity of this variation in material, the average bore diameter for a group of tubes was specified to be within 0.0005 inch of the nominal diameter. As seen from Tables II-4 through II-7, the average diameter was held within 0.0002 inch.

Wall thickness variation was also of major concern. The tube cross-section needed to be symmetrical to a high degree, since lack of symmetry would result in variation of heat flux around the circumference of the coolant hole. This variation in power would produce two effects: (1) higher thermal stress on the thick side, and (2) a tendency for the tube to bend (power camber). If the tube is constrained by the surrounding matrix of tubes, an incremental thermal stress also results, from the latter effect.

e. Mechanical properties

Thermal stress imposes the most severe requirement on the strength of the ceramic tubes. Otherwise, the only loads are compressive and quite low, producing a stress less than 300 psi. Thus, the most convenient strength property of the ceramic tube material is its resistance to

thermal stress fracture under operational conditions. This property is statistical, due to the brittle nature of the material. It has significance when compared with thermal stress demands in the reactor. The required and available thermal stress properties are best illustrated in Fig. II-10. References 4 and 13 give additional thermal stress and strength data applicable to these ceramic tubes.

5. Thermal Stress

a. Nominal symmetrical thermal stress

Much work has been done at LRL to understand the stresses which result in a power-producing hexagonal tube. An exact analytical treatment has not yet been obtained; however, a good experimental technique¹⁴ is available. The usual analytical method assumes that the material behaves elastically. The thermal stress away from the ends of a tube is then calculated for a nominal geometry where the outer boundary of the hexagonal tube is replaced by a circular boundary of equal area. The bore diameter is unchanged. The resulting tube is referred to as the Equivalent Round Tube (ERT). According to Ref. 14, the maximum elastic stress is about 5% higher for the hexagonal geometry.

Experiments were conducted with specimens of fueled BeO, in which cracking originated in a region where only compressive stresses should exist. From this was inferred that appreciable stress relaxation can occur with this particular material. The effect of thermal stress relaxation is the presence, after shutdown, of residual stresses. The stresses have the reverse sign of the operating stress on the inner and outer surfaces. Thermal stress relaxation is beneficial in a flight reactor, and also in a ground test reactor.

Usually we are interested only in the maximum tensile thermal stress, and this stress occurs on one or the other surface of the material thickness over which the temperature difference is imposed. Intuitively, we deduce easily which surface is in tension and which is in compression.

We have come to regard that the magnitude of elastic thermal stress is directly proportional to ΔT , the total temperature difference across a characteristic thickness of material. The constant of proportionality is a group of factors which takes into consideration geometry, size, and material properties.

While it is correct to say that thermal stress is proportional to ΔT , a more fundamental formulation of the law for tangential surface thermal stress is

$$\sigma_{\theta 1} = C(\bar{T} - T_1) \quad (1)$$

Equation 1 (no external constraints present) states that the tangential thermal stress, $\sigma_{\theta 1}$, on surface 1 is proportional to the difference between the volume mean temperature, $\bar{T}$ and the temperature, T_1 , of surface 1. Equation 1 also confirms the intuitive deduction that the stress is tensile (+) on the cooler surface (when $\bar{T} > T$), and that the stress is compressive (-) on the hotter surface (when $\bar{T} < T$).

The constant of proportionality in this case involves only material properties and is given in Eqs. 2a and 2b for the case of very short and very long cylinders, respectively.

$$C = E\alpha \text{ (very short cylinder)} \quad (2a)$$

$$C = \frac{E\alpha}{1 - \gamma} \text{ (very long cylinder, away from the ends)} \quad (2b)$$

Using Eqs. 1 and 2, we can easily compute the magnitude of the elastic surface stress if the volume mean temperature and surface temperature are known, together with Young's modulus, E ; absolute thermal expansion coefficient, α ; and Poisson's ratio, γ . The laborious algebra required to determine $(\bar{T} - T_1)$ in a cylinder under various heat transfer conditions can be avoided by using the normalized curves of Fig. II-11. The values used for these curves were computed from the solution of the steady state heat conduction equation:

$$\nabla^2 T(r) = \frac{-Q}{K} \quad (3)$$

The boundary conditions, the solution for $T(r)$, the surface temperatures, and the volume mean temperature are given in Fig. II-12 for four cases of interest, in addition to the expressions for $(\bar{T} - T_a)$ and $(\bar{T} - T_b)$. The ratios of the latter quantities for various cases are plotted in Fig. II-13.

We can state the following rules regarding surface tangential thermal stresses (σ_a , σ_b) for cylinders. (Figs. II-11, II-12, and II-13.)

Cases 1 and 2

1. σ_a and σ_b are always of opposite sign, since the ratio is always negative.
2. σ_a is tensile (+) for radially inward heat flow.
3. $|\sigma_a| > |\sigma_b|$ for values of $0 < \Gamma < 1$, regardless of the direction of heat flow.
4. In the limit,
 - at $\Gamma = 1$, $\sigma_a = \sigma_b$, which is consistent with analysis of heat transfer across a flat plate
 - at $\Gamma = 0$, $\sigma_a/\sigma_b = \infty$ which is a trivial result, since it is impractical to consider heat flux on a surface of zero area
5. $|\sigma_a|$ and $|\sigma_b|$ are identical whether heat flows radially inward or outward as long as the total amount of heat transferred is the same.

Cases 3 and 4

1. σ_a and σ_b are always of opposite sign.
2. σ_a is tensile (+) for radially inward heat flow.
 σ_a is compressive (-) for radially outward heat flow.
3. The cooler surface is always in tension, and the magnitude of the stress at the cooler surface is always greater than at the hotter surface.
4. For Case 3:

$$|\sigma_b| > |\sigma_a| \text{ for values of } 0 < \Gamma < 1.$$

In the limit,

at $\Gamma = 1$, $\sigma_b = 2 \sigma_a$, which is consistent with analysis of a flat plate with heat generation

at $\Gamma = 0$, $\sigma_b = \sigma_a$, which appears inconsistent with analysis of a solid cylinder with heat generation. The inconsistency is due to the fact that the radial stress component is zero at the surfaces of a hollow cylinder, but is not at the center of a solid cylinder. A slight hole causes a stress concentration factor of 2.

5. For Case 4:

$$|\sigma_a| > |\sigma_b| \text{ for values of } 0 < \Gamma < 1$$

In the limit,

at $\Gamma = 1$, $\sigma_a = 2 \sigma_b$, which is the same expression as above, with surfaces defined in reverse
 at $\Gamma = 0$, $\sigma_a = \infty$, because of the inability to remove heat at the center of the cylinder.

In the Tory II-C reactor we have $\Gamma = 0.53$ in the fueled tubes, and from Fig. II-13, curve C, we find that $\sigma_a = -2.3 \sigma_b$. Therefore, if complete stress relaxation occurs, when the temperature difference is removed and the tube is cooled to room temperature, the highest residual tensile stress would be 43.5% of the design operating tensile stress.

The highest residual tensile stress in guard tubes and side reflector tubes (if these also crept) would be relatively much less.

b. Asymmetrical thermal stress

The thermal stress σ , arising from wall thickness variation was treated as a limiting case. It was assumed that the effect of the diametral tolerance of a concentric hole on thermal stress was greater than the effect due to a hole whose axis is offset by the same amount as the unilateral hole diameter tolerance. Equation 4, for the effect of concentric hole radius tolerance on thermal stress, was derived from the basic expression for thermal stress.

$$\frac{\partial \sigma}{\sigma} = \left[\frac{\frac{2}{1-\Gamma} \left(1 + \frac{\Gamma \ln \Gamma}{1-\Gamma} \right) - \Gamma}{\frac{\ln \Gamma}{1-\Gamma} + \frac{3-\Gamma}{2}} \right] \left(\frac{\partial \alpha}{\alpha} \right) \quad (4)$$

where all symbols have been previously defined.

Equation 4 is plotted in Fig. II-14.

For fueled tubes, using the curve of Fig. II-14 and adverse tube tolerances, (wall thickness variation ± 0.0035 inches), the thermal stress may be 18% higher than in a nominal tube. Greater wall thickness variation was permitted for tubes which operated in regions where the thermal stress was not as great.

c. Power camber

The power camber effect in the ceramic tubes due to wall thickness variation is treated in a simplified manner. The hot side (thick

wall of the tube) is assumed to have its inner radius, a , smaller than nominal. Similarly, the cold side (thin wall) is assumed to have a larger than nominal inner radius. The free stream air temperature difference (ΔT_{HC}) between outside opposite sides of the tube is given by

$$\Delta T_{HC} = \left[\Delta T_{wall} + \Delta T_{film} \right]_{hot \text{ side}} - \left[\Delta T_{wall} + \Delta T_{film} \right]_{cold \text{ side}} \quad (5)$$

The value of ΔT_{wall} is generally less than 10% of ΔT_{film} , so that Eq. 5 is simplified:

$$\Delta T_{HC} \approx 1.1 (\Delta T_{hot \text{ film}} - \Delta T_{cold \text{ film}}). \quad (6)$$

This is not true for tubes in the front and rear reflector, but power there is low, and power camber and thermal stress are negligible.

A general expression for ΔT_{film} is obtained from the expression for the convection film coefficient, h :

$$h = \frac{Qa}{2\Delta T_{film}} \left[\frac{1}{\Gamma} - 1 \right] \quad (7)$$

Equation 7 is substituted in Eq. 6 with appropriate values of a and Γ . Then, assuming that h and b are the same for both sides,

$$\Delta T_{HC} = \frac{1.1 (2\epsilon)}{a} \left[\frac{1 + \Gamma}{1 - \Gamma} \right] \Delta T_{film} \quad (8)$$

where ϵ is the eccentricity of the hole. The amount of power camber is given by

$$y = \frac{L^2}{8x} \propto \Delta T_{HC} \quad (9)$$

where the new symbols are:

y = power camber

L = length of tube

x = effective width of the tube.

Combining Eqs. 8 and 9 gives

$$y \propto L^2 \Delta T_{film} \left(\frac{\epsilon}{a} \right)$$

This result shows that if a given amount of power camber is allowable, the eccentricity permitted is a function of tube length and film temperature drop. Accordingly, only full length fueled tubes were required to meet additional wall thickness specifications. For these tubes, it was required that the difference in thickness of opposite walls at the center of the tube be less than 0.004 inch. The general full length fueled tube wall thickness specification permits wall thickness anywhere to vary over a range of 0.007 inch. For other tubes a wall thickness variation of 0.010 inch is permitted.

Tailoring the wall thickness specification in this manner minimized tube rejection rate without compromising tube performance.

C. Front Reflector

1. General

The forward reflector is essentially a 9.7-inch long continuation of the active core, the major exception being that the ceramic tubes are unfueled BeO. The reflector is keyed to the active core by several hundred hexagonal cross-section nickel tubes extending axially into both core and reflector. The radial support of the reflector is identical to that of the active core; a continuation of the corrugated springs which extend to the front face of the reflector and act radially inward on peripheral (nickel) shims which fit into the array. The axial support of the front reflector consists of coil springs mounted between the cell plates at the reflector front face, and the support grid. Each unit cell is thus supported by a 150-pound spring fitted around the tie rod and located to the cell plate by adapters. This support prevents the reflector and whole reactor, for that matter, from forward movement during forward axial g loads, but permits small axial movement resulting from differential thermal and tensile movements of reactor, tie rods, and grid.

2. Cell Plates

The cell plates are hexagonal-shaped 3/8-inch thick stainless steel plates about 5 inches wide, with a hole pattern matching the reflector tube hole pattern. The cell plates hold the tubes in place and protect them from the impact of foreign material. In Section VI, Reactor Assembly, Figs. VI-33 and VI-36 show the standard cell plates in place.

3. Radial Alignment

Space is provided around each tie rod in the reflector for radial movement of the tie rods during differential thermal expansion of core, reflector, and grid. This space resulted from removal of the ring of guard tubes which is present in the active core around each tie rod. The surrounding tubes are held in place radially by rings machined on the outside diameter to fit into the hexagonal tube array.

Due to the large temperature difference between the fueled core and the front reflector, an interlocking arrangement is provided to prevent radial misalignment of flow channels. The nickel tubes fitting between the core/front reflector interface form a shear continuity so that the reflector moves out radially as far as does the core in spite of the lesser thermal expansion of the reflector. Figure II-15 shows the forward reflector/active core interface region.

D. Aft Reflector

1. General

The 2.4-inch thick aft reflector consists of 1-inch long (average) hexagonal tubes of unloaded BeO and chromium-cermet transition cartridges, each of which channels seven air channels into one large channel that matches the base block hole pattern (see Fig. II-16). As in the entire active core, the cartridges and reflector tubes are keyed to the core radially by mutual overlapping of different lengths. The cartridges extend radially out to the side reflector where they are replaced by longer BeO reflector elements. The base block hole pattern changes to accommodate the multiple smaller holes exhausting from the side reflector. Figure VI-2 shows a cartridge being installed in the reactor.

2. Cartridge Material Selection

The environment imposed on the chromium-cermet cartridge components is severe in several aspects. High-pressure, high-velocity air is channeled through the flow passages for many hours in a missile flight. Compressive stresses as high as 500 psi may be present continuously.

Thermal and transient tensile stresses exceeding the yield strength are repeatedly applied in a ground test program. The cartridges are

surrounded by other materials which react chemically with chromium at these temperatures. A very high neutron flux is present for many hours. Despite various penalties, the advantage of ductility and resulting continued integrity of chromium-cermet material is considerable.

Requirements for the flow-channeling components dictated the geometry shown in Fig. II-16. To keep pressure loss low in the transition from seven holes to one hole, the seven must angle into each other, producing feather edges at the intersection. This requirement alone precluded the use of refractory metals requiring the application of oxidation-resistant coatings, since such coatings cannot be successfully formed on edges of radius less than about 0.020 inch. Analysis of the transition geometry shows high elastic thermal stresses due to internal neutronic heating and from transients during start-up and shutdown of the reactor. Since the stresses exceed the strength of the strongest ceramics, a ductile metal was required. Of the several chromium alloys examined, chromium-5% MgO (Chrome-30) was the only available material which retained some room-temperature ductility after exposure to air at 2300° F.

3. Material Properties

Tensile properties of Chrome-30 are shown in Table II-8 and Fig. II-17. Compressive creep properties are shown in Fig. II-18.

Table II-8. Tensile strength of Chrome-30.

Test conditions	Ultimate strength (psi)	0.2% offset yield strength (psi)	Elongation, (%)
Room temp	45,000	23,000	16
Room temp, after 10 hours at 2300° F in air	30,000	20,000	2.5
2300° F	3,500	2,000	25

Figure II-19 shows the oxidation rate of Chrome-30 in ambient, dry air. At high pressure (200 psi) the rate is not significantly different. The low rate of oxidation is partly due to the adherent coating which forms and acts as a barrier against further diffusion to the metal surface.

During the development and testing of Chrome-30 components, it was discovered that the material reacts chemically with many other metals and ceramics at temperatures above 2000°F. The resulting bond, both of chrome to chrome and chrome to nonchrome, is often stronger than the parent material. The solution to the problem of Chrome-30 in contact with Chrome-30 is to develop a 0.001-inch chrome oxide coating prior to assembly. Although there is still some degree of bonding between the chrome oxide layers, it is a weak bond which is easily sheared. To prevent bonding to other ceramics and metals, a plasma-sprayed 0.005-inch layer of aluminum oxide is applied to the ends of the cartridges and adjacent reactive surfaces. The alumina-alumina interface does not bond with contact pressures up to 500 psi.

Details of machining, fabrication, and nondestructive testing of the parts are presented in Refs. 15, 16, 17, and 18.

E. Side Reflector

1. General

The side reflector zone (Fig. II-20) is a 3-inch thick radial continuation of the active core, consisting of nickel shims and unloaded BeO tubes with dimensions similar to those of the fueled tubes, except for a smaller hole size and slightly reduced across-flat dimension. The function of the side reflector is to reflect neutrons back into the active core and to provide a temperature transition from the 2500°F active core to the 1400°F nickel-base alloy side support system. The nickel shims surrounding the unloaded BeO tubes are part of the side reflector, since their neutron reflecting ability is about equal to that of BeO at that location. The shims (Fig. II-21) are machined on the inside to match the hexagonal array of the reflector tubes, and on the outside to fit into the side support springs.¹⁹ Cooling is by axial flow passage holes which match the general hole pattern of the reactor. Exhaust is through the base block holes. The shims are keyed radially to each other by stainless steel keys which fit into slots machined in the shim edges. (Fig. II-20 and Fig. VI-28).

2. Loads

The peak load on the peripheral shims is 47.5 lb/inch at the toe, the point of contact of the shim and the side support spring. (Fig. II-20,

point A). This load would occur only during a 4-g missile maneuver. The 10-hour load is 12.1 lb/inch.

3. Shim Temperature

The results of two-dimensional heat conduction calculations of nickel shim No. 6 for the Mach = 3.0 operating condition show that the maximum material temperature in the shim is 1580° F, and that 52.3% of the total heat generated in the shim is rejected into the side support region. This amount of heat requires a total air flow of 66 lb/sec between the shim and the spring, in order that the maximum desired material temperature of 1300° F is not exceeded in the spring. These calculations also show that the average distance between the shim and the spring should be a minimum of 0.100 inch. This spacing provides a cooling margin of 15%, to account for various sources of error in the analysis. The design air flow which provides the above cooling margin is 76 lb/sec.²⁰ Figure II-22 shows the temperature distribution of the hottest (NO. 6) shims. The hole size was determined by neutronic and thermodynamic considerations .

4. Friction and Abrasion

There will be sliding between shims and springs and side reflector tubes. If the resulting friction stresses are high, the nickel, (being softer than the Rene' 41 side support springs) will yield. If the stress between nickel and BeO tubes is great enough, the tubes may break transversely, and thus become shorter. No impairment of performance would result.

5. Mechanical Stress

Based on the aforementioned (E.2) shim loads, the highest mechanical stress, 3480 psi, would be in the middle of the thinnest shims (Fig. II-20, point B). This assumes that only the shims adjacent to rails will take the load, making it double the average load. The mechanical stress at the contact point of the shim finger and the spring (Fig. II-20, point A) is 2530 psi.

6. Thermal Stress

The maximum thermal stress is estimated to be less than 40,000 psi. It results from the temperature gradient shown in Fig. II-22, plus the thermal stress due to nuclear heating. The stress would yield out with a plastic deformation of 0.3%. The ductility of nickel 200 is sufficient to permit many more than the probable number of cycles of strain. In addition, the material will probably self-anneal somewhat at operating temperatures.

7. Radiation Damage

The annealing temperature of nickel is about 1500° F. It is expected that self-annealing will minimize radiation damage.

8. Material Properties

Nickel is the logical choice for shim material since it acts as a good reflector, has the required ductility to yield out thermal stress, is machinable, and has the required oxidation resistance. The creep rupture and tensile properties of nickel 200 are shown in Tables II-9 and II-10. These are the properties of annealed material. Annealing was performed on the peripheral shims to relieve stresses resulting from machining.

Table II-9. Creep-rupture properties of nickel 200 for 3- and 10-hours.

Temperature (° F)	Stress for 3-hour rupture			Stress for 10-hour rupture		
	Max (ksi)	Min (ksi)	Avg (ksi)	Max (ksi)	Min (ksi)	Avg (ksi)
1000	31.5	29.5	30.5	30.0	28.0	29.0
1100	24.5	23.5	24.0	23.0	22.0	22.5
1200	18.5	17.5	18.0	15.5	14.5	15.0
1300	13.5	12.5	13.0	11.0	10.0	10.5
1400	8.9	8.5	8.7	7.6	7.2	7.4
1500	6.4	6.0	6.2	5.4	5.0	5.2
1600	4.9	4.7	4.8	4.3	4.1	4.2
1700	4.1	3.9	4.0	3.4	3.2	3.3
1800	3.4	3.2	3.3	2.9	2.7	2.8

Table II-10. Tensile properties of nickel 200 (two tests per condition).

Test temperature (° F)	Ultimate tensile strength (ksi)	Yield strength (0.2% offset) (ksi)	Reduction of area (%)	Elongation (%)
Room temp	70.1	15.4	87	41
100	69.3	15.9	84	39
200	70.4	15.1	83	41
300	72.4	15.5	81	40
400	75.6	16.3	80	38
500	77.8	14.4	78	40
600	70.1	13.8	78	42
700	58.0	14.0	84	48
800	47.9	13.6	92	46
900	30.1	11.4	95	48
1000	33.4	10.8	97	49
1100	27.3	8.62*	96	56
1200	21.8	8.65	96	56
1300	16.9	7.40	99	62
1400	12.7	6.32	99	68
1500	9.63	5.32	99	64
1600	7.12	3.67	100	94
1700	5.43	2.88	100	100
1800	4.54	2.15	100	94

* one test only.

Fig. II-1. Reactor core and duct dimensions.

Fig. II-2. Standard and control unit cells, looking downstream.

Fig. II-3. Peripheral unit cell geometry, looking downstream.

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Fig. II-4. Ceramic tube dimensions.

Fig. II-5. Fueled tubes density curves. Nominal cross-section and fuel composition.

Fig. II-6. Reactor radii tolerances at 70°F.

Fig. II-7. Simulated reactor cross-section, one tube high.

Fig II-8 Cumulative distribution of pull forces.

Fig. II-9. Radial temperatures, side reflector region, at reactor structural design condition.

Fig. II-10. Predicted thermal stress demands and characteristics of fueled tubes.

Fig. II-11. Temperature differences vs porosity, for cylinders.

	(1) Heat Transfer, Cooled @ b	(2) Heat Transfer Cooled @ a	(3) Heat Generation, Cooled @ b	(4) Heat Generation Cooled @ a
B C @ a	$dt/dr = -q/K$	$t = 0$	$dt/dr = 0$	$t = 0$
B C @ b	$t = 0$	$dt/dr = q/K$	$t = 0$	$dt/dr = 0$
$t(r)$	$\frac{qa}{2K} \ln(b/r)^2$	$\frac{qb}{2K} \ln(r/a)^2$	$\frac{Qa^2}{4K} \ln(r/b)^2 + \frac{b^2 - r^2}{2a^2}$	$\frac{Qb^2}{4K} \ln(r/a)^2 + \frac{a^2 - r^2}{2b^2}$
T_a	$\frac{qa}{2K} \ln m$	0	$\frac{Qa^2}{4K} - \ln m + (m-1)$	0
T_b	0	$\frac{qb}{2K} \ln m$	0	$\frac{Qb^2}{4K} \ln m - \frac{m-1}{m}$
$\bar{T}$	$\frac{qa}{2K} - \frac{\ln m}{m-1} - 1$	$\frac{qb}{2K} - \frac{m \ln m}{m-1} - 1$	$\frac{Qa^2}{4K} - \frac{\ln m}{m-1} + \frac{m-3}{2}$	$\frac{Qb^2}{4K} - \frac{m \ln m}{m-1} - \frac{3m-1}{2m}$
$\bar{T} - T_a$	$\frac{qa}{2K} - \frac{m \ln m}{m-1} - 1$	$\frac{qb}{2K} - \frac{m \ln m}{m-1} - 1$	$\frac{Qa^2}{4K} - \frac{m \ln m}{m-1} - \frac{m+1}{2}$	$\frac{Qb^2}{4K} - \frac{m \ln m}{m-1} - \frac{3m-1}{2m}$
$\bar{T} - T_b$	$\frac{qa}{2K} - \frac{\ln m}{m-1} - 1$	$\frac{qb}{2K} - \frac{\ln m}{m-1} - 1$	$\frac{Qa^2}{4K} - \frac{\ln m}{m-1} + \frac{m-3}{2}$	$\frac{Qb^2}{4K} - \frac{\ln m}{m-1} - \frac{m+1}{2m}$

a = inside radius
 b = outside radius
 $b^2/a^2 = m$

$t(r)$ = temperature at r
 T_a = temperature at $r = a$
 T_b = temperature at $r = b$
 K = thermal conductivity

q = heat flux at input surface
 Q = heat generation rate per unit volume
 $T = \frac{2}{b^2 - a^2} \int_a^b T r dr$

Fig. II-12. Temperature relationships in hollow cylinders.

Fig. II-13. Ratio of surface tangential stress vs cylinder porosity.

Fig. II-14. Thermal stress change for change in hole radius vs porosity.

Fig. II-15. Active core interface in forward reflector. Standard unit cell.

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Fig. II-16. Cermet transition cartridge.

Fig. II-17. Tensile strength of Cr-5MgO.

Fig. II -18. Compressive creep of Cr-5MgO at 500 psi.

Fig. II-19. Oxidation in dry air of Cr-5MgO.

Fig. II-20. Side reflector elements.

Fig. II-21. Nickel shims.

GAS TEMPERATURES
IN CHANNELS, °F

$T_1^o = 1430$
 $T_2^o = 1466$
 $T_3^o = 1464$
 $T_4^o = 1466$
 $T_5^o = 1468$
 $T_6^o = 1483$
 $T_7^o = 1480$
 $T_8^o = 1481$
 $T_9^o = 1486$
 $T_{10}^o = 1490$
 $A T_B^o = 1302$
 $B T_C^o = 1232$
 $C T_D^o = 1198$
 $D T_E^o = 1235$
 $E T_F^o = 1305$
 $A \bar{T}_F^o = 1249$

NOTES: 1. SURFACE AHGF IS ADIABATIC
 2. $x/L = 0.75$

Fig. II-22. Temperature distribution of No. 6 shim, 3° sector.

III. REACTOR SUPPORT (AXIAL)

A. General

The cooled tie rod axial support system was chosen as a logical follow-on from the proven Tory II-A support system. The stresses and temperatures of the tie rods and F-48 alloy base blocks can be accurately predicted. Successful tests on representative parts have been made with realistic environments.

The axial support system (Fig. I-3 and III-1) is quite simple. The pressure-drop force of the reactor is transmitted to the base blocks. The tie rod flange on the aft end of the tie rod picks up the load from the center of the base block and transfers it to the tie rod. The load is thus carried forward to the threaded nut on the front end of the tie rod. The nut transfers the load to the beam-like forward support grid. The grid is supported on a flange which is machined out of the duct; so the load is transferred to the duct.

Figure VI-1 shows the tie rods at the aft (base block) end of the reactor; Fig. VI-40 shows the forward (support grid) end.

B. Support Grid

1. Introduction

The support grid assembly (Fig. III-2, III-3) is a circular grid structure fitted to the reactor duct just forward of the reactor core. It transmits the axial load from the 121 tie rods of the reactor core to the reactor duct. It also supports and positions the reactor control rods and control rod actuating mechanisms. The support grid assembly is about 73% porous at the grid plate; the remainder is about 85% porous, to allow the ram air to get through to the reactor core with negligible pressure loss and uniform distribution. Streamlining was not required because of the low Mach numbers in this area.

Figure I-3 shows the position of the support grid assembly in the reactor. The support grid assembly consists of a grid plate, 12 transverse beams, grid ring, ring pins, beam spacers, and fastenings. The grid ring (Fig. III-4) is notched to locate the support the transverse beams. The grid plate bolts to the ring and beams. The grid plate transfers the tie rod axial load to the beams, and laterally supports the beams to prevent buckling. Axial deflection of the grid is controlled by adjusting the height of the shorter beams.

Beam spacers increase the stability of the two bent beams. The beams transfer the axial load to the grid ring, which transfers the load to the duct through matching ledges. The ledges are grooved to allow the flow of cooling air between the support grid and reactor duct. Ring pins radially locate the support grid in the duct, and prevent forward movement of the reactor core during assembly and handling. The grid ring is fabricated of Hastelloy C, and the remaining parts are of Rene' 41.

The support grid structure is deflection limited (0.163-inch maximum axial deflection). The maximum stresses, including the theoretical stress concentration factors, are 30% less than the yield and creep strengths. The peak material power density is 0.67 MW/cu ft, and all cooling is with 1060° F ram air. To allow considerable testing, the grid was designed for 35 seven-minute cycles; ten at maximum conditions.

The final design is the result of a comparative analysis of many different designs, and was chosen for its simplicity and superior structural reliability. The basic design loads for the grid are (1) at operating conditions, 140 tons axially directed to the rear and 4 g's radially, and (2) for handling conditions, 30 tons (5 g's on the reactor core) axially forward. Because these loads differed greatly, the philosophy was to design the grid simply and reliably for the basic load and then modify it for the other loads. The basic load capability of this structure does not depend on welds, bolts, or other fasteners, yet the total weight is only 885 pounds. This same basic design could be lightened to about 600 pounds for a flight reactor and still retain a high degree of structural reliability.²¹⁻²³

2. Design Analysis

The basic design requirements are to axially support the reactor core and to radially support the control guide system, without affecting the air flow distribution prior to entering the core. Minimum weight was not a requirement. Deflection while supporting the reactor pressure drop load was limited to 0.180 inch overall and about 0.035 inch between adjacent tie rods. These limits were set to minimize axial displacement between adjacent base blocks.

The support grid is the middle link of the three items that guide the control rods. The other two are the slider and way assembly and the reactor core. Hole locations in the grid plate for the control tie rods are chosen to minimize control rod binding during operation.

The basic purpose of the support grid is to support the reactor axially against the flow pressure drop loads. For the reactor structural design condition² the loads are:

$$F_a \equiv \text{Average tie rod load} = 2340 \text{ lb}$$

$$F_T \equiv \text{Total axial load} = 280,000 \text{ lb}$$

These loads are based on a reactor pressure drop of 120 psi. The pressure drop estimate was later reduced to 113 psi, including a grid screen. Because of differential expansion and friction, not all tie rods are loaded evenly. Most of the core pressure drop load is carried by the outer tie rods, as shown in Fig. III-46. The average tie rod load was conservatively assumed to exist at each tie rod location.

The lateral load which would be expected during a 4-g missile maneuver is:

$$F_R \equiv \text{Total Radial (side) load} = 4,140 \text{ lb}$$

Investigation of other loads indicated the possibility that a tube bundle associated with a "hot" tie rod could stick during reactor cooldown. This would result in one or a few tie rods loading the support grid. A reasonable estimate of this load (based on 12 psi side support pressure and a 0.5 coefficient of friction) is 3,000 pounds. This load could conceivably be higher, so the design F_S was 6000 pounds.

Handling loads are based on 5-g acceleration of the reactor assembly (Ducts 4 and 5) in any direction.

$$F_{hr} \equiv \text{Total Radial Load} = 5,170 \text{ lb}$$

$$F_{hT} \equiv \text{Total Axial Load} = 60,000 \text{ lb}$$

(based on no friction between core and duct).

In the forward direction, the ring pins resist the weight of the core and support grid:

$$F_{hF} = F_{hT} + 5(885) = 64,430 \text{ lb.}$$

3. Material Selection

The principal support grid material requirements are:

- 1) Sufficient strength and durability under operating conditions,

- 2) availability in required shape and size,
- 3) economical fabrication.

In addition, the material is preferred to have a high modulus of elasticity and a low coefficient of thermal expansion, and be moderately ductile within operating temperature range.

The main load-carrying components of the support grid (beams) are designed for minimum fabrication requirements. These minimum requirements are drilling and tapping the forward edge and machining the ends and grooves of the straight and bent beams in the final heat-treated condition. The highly stressed areas of the beams are smooth and do not have stress raisers; therefore, ductility is not a prime requirement. The strength and temperature requirements of the braces and ring are much less than the beams; therefore, these items can be fabricated from readily machinable and weldable alloys.

A preliminary heat transfer analysis indicated that the beams should be 1/2 inch or less in thickness to maintain reasonable thermal gradients and limit the maximum temperature below 1600° F. Based on this 1/2 inch thickness and 0.125 inch mechanical deflection, the maximum stress in a center beam would be 40,200 psi and the height would be about 7.4 inches. Material studies on other components, mainly the reactor duct, indicated that Rene' 41 and Hastelloy C were the two most likely materials. In this case, Hastelloy C does not have sufficient strength and would require larger beams (> 9 inches high).²² This would increase the possible thermal gradients and the reactor length. The possible single tie rod overload condition also indicated that a material of superior strength was preferred for the grid plate. A high cobalt content (11%) would not be neutronically objectionable in this application.

As a result of these considerations, Rene' 41 was chosen for the beams and grid plate. Hastelloy C was chosen for the grid ring because of its thermal compatibility with Rene' 41 and its rolling and welding characteristics.

Rene' 41 is a wrought age-hardening nickel base alloy especially suited for service temperatures between 1200 and 1800° F. The support grid requirements are not severe compared to its capabilities. The 1950° F stabilizing solution heat treatment was employed to prevent the possibility of strain age cracking and to develop maximum ductility.

Table III-1 lists 1400° F tensile properties of Rene' 41 for the material used to fabricate the support grid. See References 24, 25, and 26 for additional properties.

Hastelloy C properties are reported in detail in Section IV -H.

Table III-1. Tensile properties of Rene' 41 at 1400° F

Material: 7/8 in. and 5/8 in. plate from Tory II-C reactor support grid
Heat treatment: 1950° F 1/2 hr A.C., 1400° F 16 hr
Hold time at temperature: 30 min
Strain rate: 0.000083 in./in./sec to yield strength, then
0.00125 in./in./sec to rupture.

Spec. No.	Ultimate tensile strength (ksi)	Yield strength (ksi)		Elongation in 1 inch (%)	Young's modulus (10 ⁶ psi)	Reduction of area (%)
		0.02% offset	0.2% offset			
980H	150.0	100	115	13	24.2	13.1
981H	152.0	102	114	15	24.1	20.6
982H	147.0	105	117	16	23.0	22.1
983H	144.0	104	118	11	23.7	15.9
984H	152.0	107	126	7	22.2	-
985H	143.0	97	110	15	23.6	16.0
987H	140.0	103	115	17	24.1	35.9
988H	142.0	106	119	15	23.9	34.6
989H	145.0	111	122	15	23.9	30.9
990H	152.0	107	121	12	23.8	12.6
991H	154.0	110	125	10	24.0	13.9
992H	168.5	111	122	15	24.1	28.7
993H	153.0	98	116	16	24.9	34.0
994H	154.0	112	123	16	24.0	27.8
995H	139.0	105	116	18	22.0	33.9
898H	154.0	103	115	11	26.9	20.0
899H	166.0	89	126	13	26.5	21.0
900H	167.0	107	123	14	27.5	28.0
901H	168.0	-	117	14	24.1	31.0

4. Nuclear Heating:

Nuclear heating densities of the support grid and other components forward of the core were evaluated with the aid of the Monte Carlo codes YOGI and YOGI-N.²⁷⁻²⁹ Both core gamma heating and heating by gamma rays generated by the (n, γ) process in the forward region were considered. The total nuclear heating density distribution in the forward reactor region is given in Fig. III-5. The heating density values are given for full density material, normalized to 1 MW/ft³ of average hot active core power density of the respective components located on axis. For the structural design point condition (Mach 3.0) and 1 hour of reactor running time (assuming 90% of infinite saturation for fission product decay), we have:

Power generated = 489 MW

Hot active core volume = 54.04 ft³

$\therefore$ Average power density = 9.05 MW/ft³

The radial and axial dependence of nuclear heating (including approximate uncertainty) in the support grid for structural design point condition are given in Fig. III-6.

The core gamma energy absorption distribution in the forward reactor regions was analyzed with the gamma YOGI code, using both rectangular and cylindrical geometries. In addition, the effects of smeared and discrete representations of the support grid and cell plate material were investigated.

It was shown by comparison that smearing the support grid beams uniformly increased the average heating density by about 10%. This apparently is due to streaming past the beams in the discrete model. Another considered effect is the nonhomogeneous character of the cell plate region, where most of the material is concentrated in line with the support grid beams. This shields the beams from the head-on radiation to some extent, and reduces the beam heating by about 4%.

A number of toroidal geometry Monte Carlo problems were run to determine the effect of the finite cylindrical geometry on the infinite slab approximation. It was shown that an average reduction factor of 1.9 is realized when finite cylindrical (actual) geometry is considered in the smeared homogeneous model, as compared to the infinite slab geometry. This effect was confirmed analytically.

The analysis of the heating generated by neutron captures in the forward regions was performed in two steps: first, the neutron absorption distribution in the cell plate assembly and support grid was calculated by means of the YOGI N code. This distribution was then used as gamma ray source input to the YOGI code which yielded a spatial distribution of neutron capture gamma ray heating. The results thus consist of two independent distributions, one from each of the considered (n, γ) source regions. They are shown in Fig. III-5.

Since neither of the two Monte Carlo codes can represent a rectangular array in cylindrical geometry, it was necessary to simulate the support grid beams by a series of concentric equidistant cylindrical rings. This model was constructed such that a total mass was conserved, and that the thickness and the depth of the beams was the same as in the actual configuration. Although this model does justice to some aspects of the analysis (e.g., corrects self-shielding treatment of the beams), the coaxial nature of the model tends to symmetrize the spatial distribution of the flux. Also, the concentration of more than actual mass per unit volume in the central region of the support grid overestimates the absorption density in that region. An additional difficulty was presented by the inability of the code to treat all of the support grid beams simultaneously. This led to the necessity to superimpose results from a series of problems, treating a few beams at a time. This eliminated the coupling effect between beams to some extent. However, in view of the complexity of the support grid configuration, the analysis provided by the neutron Monte Carlo code is superior to other available methods, which would be laborious, time-consuming, and less exact.

The spatial and energy distribution of the neutron leakage at the forward front reflector face was obtained from results of ANGLE calculations. The model used in the gamma ray Monte Carlo problems, to determine the spatial distribution of the (n, γ) gammas, is the same as the one used in the neutron analysis. The cell plate assembly was smeared and the support grid was represented by concentric rings.

5. Convection Coefficients

All cooling of the support grid is by the ram air flowing inside the duct. For the structural design point condition:

Flow rate in open duct = 1814 lb/sec

Flow through grid (excludes flow through the tie rods) =
1720 lb/sec

Temperature = 1063° F

Pressure = 340 psia

Open duct mass flow rate = 0.73 lb/sec in²

Grid plate mass flow rate = 0.98 lb/sec in²

Beam area mass flow rate = 0.86 lb/sec in²

Equations 1 and 2 for laminar and turbulent flat plate flow were used to determine appropriate convection coefficients. They are applicable for low Mach numbers and moderate temperature differentials between fluid and wall.

$$h_x = 0.332 \text{ Re}^{0.5} \text{ Pr}^{1/3} k/x, \text{ laminar flat plate flow} \quad (1)$$

$$h_x = 0.0296 \text{ Re}^{0.8} \text{ Pr}^{1/3} k/x, \text{ turbulent flat plate flow} \quad (2)$$

Results of these equations for the support grid are shown in Fig. III-7. The transition region between laminar and turbulent flows varies greatly. The transition region shown in Fig. III-7 is based on a high turbulence level in the approaching stream. The curve from Eqs. 1 and 2 is applicable for the grid plate and beams and the inside diameter of the grid ring. The outside diameter of the grid ring was analyzed in Section IV, Reactor Duct. Results are included in Fig. III-7.

The sections of the grid plate between the beams are essentially square beams in crossflow. The local film heat transfer coefficient around cylinders and other shapes in flow of air normal to the axis varies greatly (see Ref. 30, Fig. 9-4) and is highest on the forward and aft surfaces. Much of the available experimental data are based on total heat flow to or from a tube and are intended to indicate average film coefficients, which is applicable here. Hilpert,³¹ proposes a correlation for square cross sections:

$$\text{Nu}_d = 0.0921 (\text{Re}_d)^{0.675} \quad (3)$$

Equation 3 is based on air flow experiments with square cross sections. The correlation is intended to be applicable for Reynolds numbers between 5,000 and 100,000. It is also based on a low turbulence level in the approaching stream. The Reynolds number for the grid plate (300,000) exceeds the recommended range for Eq. 3. There is also the tendency for the numerical constant to decrease as the Reynolds number increases (based on comparing circular tube data). Assuming that the high turbulence level in the approaching

stream cancels this effect, a conservative estimate of the average film coefficient for these webs is:

$$\bar{h}_d = 300 \text{ Btu/hr-ft}^2\text{-}^\circ\text{F}.$$

This result is shown on Fig. III-7 for comparison with flat plate flow. Good agreement is noted.

6. Heat Transfer

At any one location, the nuclear heating value is constant and both surfaces are cooled about the same. Assuming no axial conduction, the wall and maximum internal temperatures are determined from Eqs. 4 and 5.

$$t_s = t_a + \Delta t_f = t_a + \frac{qx}{2h} \quad (4)$$

$$t_m = t_s + \Delta t_w = t_s + q \frac{(x/2)^2}{2k} \quad (5)$$

where

t_a = air temperature = 1060° F

t_s = surface temperature

t_m = maximum internal temperature

x = thickness

The support grid steady state temperature distribution for nominal conditions is shown in Fig. III-8. Equations 4 and 5 were also used to estimate the temperatures in the grid ring lug area. For assumed extreme conditions the maximum support grid temperature would be about 1353° F.

Axial conduction near the trailing edge accounts for approximately 1% of the power removed, and will not have much effect on the maximum material temperature.

The transient response of different support grid sections are about the same prior to nuclear heating. This is indicated by the fairly uniform thickness and film coefficients. There is a difference, however, between the response of the support grid and the reactor duct. Results for a severe transient condition (step input of 1060° F air at full flow) are shown in Fig. III-9.

7. Stress Analysis

a. Grid plate

Reactor pressure drop loads are transferred from the tie rods to the grid plate webs and thence to the beams. The webs are held together by links which cover the forward face of the beams. Pertinent grid plate definitions are given in Fig. III-10. The stresses in the webs and links are functions of the end moments of the webs and torsion in the connecting links. The magnitude of the moments and torsion are dependent upon each other and from web to web. The distance between beams and location of tie tubes on the webs vary. Since the grid plate is symmetrical only about the center, complete solution for the moments and torsion must consider seven webs and seven links (neglecting peripheral tie rods). This could easily be solved with a computer, but since the grid plate is overdesigned (thickness was based on stuck tube bundle (STB) condition) sufficient accuracy is achieved with a few simple conservative assumptions.

By inspection, the area between the center and first row of tie rods is seen to have the highest stresses (see Fig. III-10). The assumption of completely restrained ends makes M_2 about 28% greater than M_1 . We will assume $M_1 = M_2 = M_a$, and solve for M_a . We will then increase M_a a proportional amount for an estimate of M_2 . We have also assumed the load is applied to the center of the tie rod base instead of each side. For the center bay (from basic moment and deflection equations) we have at $X = 0$:

$$EIy'' = -M \quad (6)$$

$$EIy'_1 = \frac{M\ell}{2} - \frac{F\ell^2}{16} \quad (7)$$

Since $M_1 = M_2$ we can analyze a link of length $g/2$ with:

$$\theta = -y'_1 \quad (8)$$

$$\tau = \text{torque} = \frac{1}{2} M \quad (9)$$

Torsion on link: $\tau = \iint \psi \, dx \, dy = k_1 G\theta(2/g)e^3 d$, for a rectangular bar.

For $d/e = 1.5$ $k_1 = 0.196$ $k = 0.848$ (Ref. 32).

From above,

$$\theta = \frac{\tau g}{2k_1 G e^3 d} \quad (10)$$

From Eqs. 7 through 10, letting $\alpha = gEI/2k_1 Ge^3d$, our result is:

$$M(\alpha + \ell) = \frac{F\ell^2}{8} \quad (11)$$

for design conditions:

$$\alpha = 1.7 \quad \alpha + \ell = 5.368 \quad \text{and}$$

$$M_a = M_1 = 0.314 F_a \text{ in. lb} \quad (12)$$

Since this is about 65% of the fully constrained moment, M_2 is about 18% $> M_1$ so:

$$M_2 \approx 0.37 F_a \text{ in. lb} \quad (13)$$

From Eqs. 8, 9, and 10

$$y' = \frac{\alpha M}{EI2} \quad (14)$$

$$y'_1 = 0.45 \times 10^{-6} F_a \quad y'_2 = 0.53 \times 10^{-6} F_a \quad (15)$$

For the STB condition we assume no link rotation at adjacent webs. The analysis is same as above except that the link length is g .

$$\alpha = 3.4 \quad \alpha + \ell = 7.068$$

$$M_1 = 0.238 F_S \text{ in. lb} \quad (16)$$

$$y'_1 = 0.68 \times 10^{-6} F_S \quad (17)$$

Maximum stress (shear) in the link is:

$$\begin{aligned} \sigma_s &= 2KG\theta e \\ \sigma_s &= 39,000 \text{ psi for STB condition} \end{aligned} \quad (18)$$

Stress distribution on the aft web surfaces for design and STB conditions is shown in Fig. III-11. At points p and v (Fig. III-10) there is a concentrated stress area caused by the through holes ($K_p \approx 1.5$ and $K_v \approx 2.5$). The maximum stress (at v) for aft bay No. 1 web surface is:

$$\begin{aligned} \sigma_v &= 38,000 \text{ psi for design condition.} \\ \sigma_v &= 120,000 \text{ psi for STB condition.} \end{aligned}$$

Thermal stresses in the grid plate are negligible.

b. Beams

The stresses in the beams are mainly due to the reactor pressure drop loads, and they are much lower or negligible during handling

or in the event of stuck tube bundle conditions. In this analysis the effects of friction and stiffness of the grid plate are neglected. The thickness, depth, and length ratios are such that elastic beam formulas are applicable. For simply supported uniformly loaded beams we have:

$$M = \text{maximum moment} = WL/8 \quad (19)$$

$$\sigma = \text{maximum bending stress} = M/Z = 0.75WL/bh^2 \quad (20)$$

$$\delta = \text{deflection at any point} = Wx/24EIL (L^3 - 2Lx^2 + x^3) \quad (21)$$

$$\delta_m = 5/384 \times WL^3/EI \text{ at center} \quad (22)$$

The short beam depths were adjusted to yield the same deflection the center beam has at the same radius from center. The calculated beam bending stresses and load deflections are summarized in Table III-2. Grid plate bolt holes cause a stress concentration factor of about 2.0 on the forward (compressive) beam surfaces.

Table III-2. Support grid. Beam mechanical stresses and deflections.

Beam depth h	Deflection at center (in.) δ	Maximum stress (psi) σ	Total uniformly distributed load per beam* (lb) W	Length (in.) L
7.75	0.113	38,200	28,800	53.08
7.75	0.106	38,000	29,500	51.50
7.320	0.0904	34,600	25,600	48.20
7.090	0.0705	32,600	25,200	43.25
6.240	0.0520	27,500	18,800	37.81
4.560	0.0264	19,300	9,800	27.45

*Based on a pressure drop of 120 psi.

Maximum thermal stress (tension on surface) for the aft area of the center beam is:

$$\sigma_t = \frac{\alpha_a E_q b^2}{12k(1 - \gamma)} = 9500 \text{ psi} \quad (23)$$

c. Ring

The ring transfers the loads from the beams and distributes them to the duct. The beam support does not uniformly distribute the load around the ring, but concentrates the loads at the ends of the long beams. In this area, the maximum stresses in the ring are:

Bearing stress between longest beam and ring	48,000 psi
Bearing stress between grid-to-duct teeth	12,000 psi, nominal case 36,000 psi, poor contact between teeth
Shear stress in ring due to eccentricity of beams and teeth	4,000 psi
Tensile stress on aft ring surface due to poor contact between teeth (i.e., 3 teeth not in contact)	11,000 psi

d. Spacers

The two bent beams (No. 5) are kept from rotating by close fitting slots in the ring, shear of grid plate bolts, and two spacers near the aft surface (see Fig. III-12). For a conservative analysis, each individual item was assumed to resist the total offset load (moment) of the beam. Results of such an analysis are:

Shear in grid plate bolt	34,400 psi
Tension in round spacer	5,100 psi
Compression in rectangular spacer	5,400 psi
Tension in spacer bolt	27,600 psi
Shear in grid ring bolt	13,000 psi

e. Bolts

Preload tensile stress in the bolts at room temperature is approximately 60,000 psi (σ_a). During transient warm-up conditions the bolt temperature may lag the plate by 100° F, and this would increase the stress by about 24,000 psi (σ_b).

The grid plate bolts do not transfer operating loads, but they are stressed during reactor operation. The ends of the webs tilt (as indicated in paragraph 7. a.) and induce tensile and bending stresses in the bolts. For normal conditions the maximum stress is 5,500 psi (σ_c) in bending and tension is

increased by about 11,000 psi (σ_d). For the stuck tube bundle condition these stresses would increase to 35,000 psi (σ_e) in tension and 18,000 psi (σ_f) in bending.

Small deflections, mechanical and thermal, between the grid plate, beams, and ring are not resisted by shear in the bolts. Friction is the primary resistance. Accurate clearance holes in the grid plate allow about 0.008-inch radial movement. This is sufficient for a 45° F temperature differential between the grid plate and ring. Temperature differences this high are unlikely. If all of this deflection (0.008 inch) were transmitted to the bolts, the bending stress would be about 120,000 psi (σ_g).

During handling of the reactor, the most severely stressed bolts are the 34 grid plate-to-ring bolts. Their average tensile stress, based on F_{hT} and static analysis, is 46,000 psi, but the stress in the bolts associated with the longest beam reaches 84,000 psi (σ_h). Accurate impact analysis was not practical.

Maximum combined tensile stresses in the bolts are as follows:

Room temperature (handling conditions)

$$\sigma = \sigma_a + \sigma_h = 144,000 \text{ psi}$$

During reactor run, first part of plateau

$$\sigma = \sigma_a + \sigma_b + \sigma_d \pm \sigma_c = 95,000 \pm 5500 \text{ psi}$$

During reactor cool down (stuck tube bundle condition)

$$\sigma = \sigma_a + \sigma_e \pm \sigma_f = 95,000 \pm 18,000 \text{ psi}$$

f. Ring pins

The ring pins support forward and radial handling loads and the radial grid loads during operation, and allow for differential expansion. The aft reactor pressure drop load would deform the pins if the grid-to-duct teeth are not properly mated. A 0.003-inch tooth gap for about 5 inches each side of a pin could result in 180 ksi bending and 45 ksi shear stresses in the pin. With larger gaps, this stress could approach 120 ksi in shear and 300 ksi in bending.

During normal conditions the pins support the dead weight of the support grid with a shear stress less than 1,200 psi. With 4-g vibration this stress would increase to about 5,500 psi. For handling conditions a radial or forward shock would induce respectively about 6,800 psi or 28,000 psi shear stress.

Maximum Hertz stress would be about 22,000 psi. These impact stresses are also based on static load analysis.

8. Deflections

The axial deflection of the support grid is due to a combination of loads and thermal gradients. Deflections due to loads are given in Table III-2 and were verified by tests (see paragraph 11 below). The temperature distribution shown in Fig. III-8 increases the deflection of the longest beam 0.050 inch. This results in a maximum support grid deflection of 0.163 inch, and deflection differences between adjacent beams, starting at center, of 0.015, 0.020, 0.030, 0.035, and 0.035 inch.

Radial deflections are essentially the difference between tie rod positions of the reactor core and the support grid. For design purposes, the tie rod positions in the core at room temperature assembly were calculated by statistical fuel tube stacking effects. This resulted in considering the effective tube across flats dimension to be 0.298 inch instead of 0.297 inch. To minimize control rod binding, two extreme cases were considered: (1) structural design (1063° F inlet air), and (2) maximum reactor temperature with 600° F inlet air conditions. During both runs the reactor core expands about the same in the area where the tie rods are constrained ($X/L > 0.16$). Radial expansion of the core at the vernier control tie rod (17.3 radius) is shown in Fig. III-13. The expansions of the support grid for the two conditions are also indicated. Pre-bending the tie rod 0.030 inch outward at assembly decreases the possibility of binding during the 600° F run, and equalizes it for both conditions.

Differential thermal radial expansions between the beams, ring, and grid plate are very small, and not constrained except for friction.

9. Elastic Stability

The elastic stability of the support grid was not analyzed in detail, but its reliability was insured by a few conservative estimates. Neglecting the stiffening effects of the grid plate, the critical buckling load on the longest beam is about 24,600 pounds (design load is 28,800 pounds on this beam). Friction between the plate and compressive surface of the beam (backed up by bolts) increases this critical load many times. This friction is sufficient to

resist lateral movement even if the beam curves laterally more than 0.5 inch. Buckling of the bent beams (No. 5) is prevented by the ring slots, grid plate bolts, and aft spacers (see paragraph 7.d.).

10. Vibration

The axial natural frequency of the support grid, assuming a uniform load from the tie rods, was estimated to be about 43 cps. This is about the same as the complete axial support system natural frequency.³³ Damping by friction between the core and duct indicates that a uniformly distributed pressure pulse > 5 psi at the natural frequency would be required to excite the system.

Lateral natural frequency of the center beam was estimated to be greater than 300 cps when the grid is unloaded, and much greater than 300 cps when loaded. Structural tests confirmed this.

11. Testing

When the design of the support grid was finalized, the question of proof testing arose. A load test of the service grid at design temperature would have been very costly. It was therefore decided to do two complementary tests: (1) a hydrostatic, room temperature test of the actual service grid,³⁴ and (2) a hot load test of a simulated grid.³⁵ Both tests were performed, and gave satisfactory proof of the design and fabrication adequacy of the support grid.

a. Support grid hydrostatic test³⁶

The support grid was assembled in the reactor duct, and brass load buttons were located in each tie rod hole as shown in Fig. III-14. An aluminum plate just smaller than the duct inner diameter was placed over the buttons. A rubber sheet was then placed over the aluminum plate and bonded to the wall. A pressure dome was bolted to the forward end of the reactor duct (see Fig. III-15). Figure III-16 shows some of the test instrumentation.

The grid is deflection-limited, so the test pressure was determined by multiplying the design pressure (120 psi) by the ratio of the elastic modulus for Rene' 41 at ambient temperature and at 1200° F. This yields a test pressure of 145 psig, or a load of about 2900 pounds per tie rod hole. The test pressures

used were 29, 58, 87, 116, and 145 psig. Pressurization was accomplished manually by use of pressure and bleed valves. All data was read on a strip chart and punched onto tape.

The instrumentation employed was of four types: strain gages, deflection gages, stresscoat, and pressure gages. Table III-3 gives the instrumentation list, and Figs. III-17 through III-20 show their locations.

Table III-3. Support grid test instrumentation.

Gage No.	Description	Location	Range	Remarks
1	Y-axis strain	Grid plate top	0 to -333 in./in.	See Fig. III-17
2	X-axis strain	Grid plate top	"	"
3	Y-axis strain	Grid plate top	"	"
4	Y-axis strain	Grid plate top	"	"
5	Y-axis strain	Grid plate top	"	"
6	Axial strain	Beam F bottom	0 to 1265 in./in.	See Fig. III-20
7	Axial strain	Beam G bottom	"	"
8	Axial strain	Beam H bottom	"	"
9	Axial strain	Beam J bottom	"	"
10	Axial strain	Inner grid stiffener	0 to 333 in./in.	"
11	Axial strain	Outer grid stiffener	"	"
12	Circum strain	Support ring aft edge	"	See Fig. III-18 (90°)
13	Circum strain	Section 5 outside aft of duct teeth	"	See Fig. III-18 (0°)
14	Axial strain	Section 5 outside aft of duct teeth	"	"
15	Circum strain	Section 5 outside aft of duct teeth	"	See Fig. III-18 (90°)
16	Axial strain	Section 5 outside aft of duct teeth	"	"
17	Circum strain	Section 5 outside aft of duct teeth	"	See Fig. III-18 (180°)
18	Axial strain	Section 5 outside aft of duct teeth	"	"

Table III-3. (Continued)

Gage No.	Description	Location	Range	Remarks
19	Circum strain	Section 5 outside fwd of duct teeth	0 to 333 in./in.	See Fig. III-18 (0°)
20	Axial strain	Section 5 outside fwd of duct teeth	"	"
21	Circum strain	Section 5 outside fwd of duct teeth	"	See Fig. III-18 (90°)
22	Axial strain	Section 5 outside fwd of duct teeth	"	"
23	Circum strain	Section 5 outside fwd of duct teeth	"	See Fig. III-18 (180°)
24	Axial strain	Section 5 outside fwd of duct teeth	"	"
25	Circum strain	Section 5 inside fwd of duct teeth	"	See Fig. III-18 (0°)
26	Axial strain	Section 5 inside fwd of duct teeth	"	"
27	Circum strain	Section 5 inside fwd of duct teeth	"	See Fig. III-18 (90°)
28	Axial strain	Section 5 inside fwd of duct teeth	"	"
29	Circum strain	Section 5 inside fwd of duct teeth	"	See Fig. III-18 (180°)
30	Axial strain	Section 5 inside fwd of duct teeth	"	"
31	Axial deflection	Center line of Beam F	0 to 0.11 in.	See Fig. III-19 (0.150-in. gage)
32	Axial deflection	4 in. from $\mathcal{C}_L$ on Beam F	0 to 0.10 in.	See Fig. III-19 (0.150-in. gage)
33	Axial deflection	8 in. from $\mathcal{C}_L$ on Beam F	0 to 0.09 in.	"
34	Axial deflection	11 in. from $\mathcal{C}_L$ on Beam F	0 to 0.07 in.	See Fig. III-19 (0.100-in. gage)
35	Axial deflection	14 in. from $\mathcal{C}_L$ on Beam F	0 to 0.05 in.	"
36	Axial deflection	18 in. from $\mathcal{C}_L$ on Beam F	0 to 0.03 in.	See Fig. III-19 (0.050-in. gage)
37	Axial deflection	Grid center on Beam E	0 to 0.10 in.	See Fig. III-19 (0.150-in. gage)
38	Axial deflection	Grid center on Beam D	0 to 0.09 in.	"

Table III-3. (Continued)

Gage No.	Description	Location	Range	Remarks
39	Axial deflection	Grid center on Beam C	0 to 0.07 in.	See Fig. III-19 (0.100-in. gage)
40	Axial deflection	Grid center on Beam B	0 to 0.05 in.	"
41	Axial deflection	Grid center on Beam A	0 to 0.03 in.	See Fig. III-19 (0.05-in. gage)
42	Axial deflection	Right side Beam B	"	"
43	Axial deflection	Left side Beam B	"	"
44	X-axis strain	Grid plate top	0 to 333 in./in.	See Fig. III-17
45	Y-axis strain	Grid plate top	"	"
46	Circum strain	Section 5 inside fwd of duct teeth	"	See Fig. III-18 (3° 45')
47	Axial strain	Section 5 inside fwd of duct teeth	"	"
48	Circum strain	Section 5 inside fwd of duct teeth	"	See Fig. III-18 (93° 45')
49	Axial strain	Section 5 inside fwd of duct teeth	"	"
50	Circum strain	Section 5 inside fwd of duct teeth	"	See Fig. III-18 (183° 45')
51	Axial strain	Section 5 inside fwd of duct teeth	"	"
P ₁	Pressure	Vessel	0 to 145 psi	

The results of the hydrostatic test are shown in Figs. III-21 through III-31. Many of the smaller strains are omitted because they were quite low. The hysteresis on the load curves is due to friction between the grid plate and the beams.

b. Simulated support grid test

A simulated grid consisting of two beams was designed and fabricated. This test section could conveniently be tested at temperature with minimum compromising of true conditions.

The test section is shown in Fig. III-32. Two beams corresponding to the central grid beams carried the load. The center of the grid plate picked up the main control tie rod loads. It overhung the beams and picked up loads which compensated for other tie rod loads. The load was distributed by a system of whiffletrees which led to three hydraulic cylinders. The cylinder loads were controlled by a marginator. The test section was encased in a furnace consisting of a long aluminum cylinder packed with castable alumina and other insulation. Two silicon carbide heating elements ran the length of the assembly. Figure III-33 shows the assembly with the top half of the furnace off. Figure III-34 shows the whiffletree arrangement in place.

Three tests were performed with this arrangement. The first was at ambient temperature; stress, deflection, and vibration data were obtained. The second was at design temperature (1250° F), and deflection and temperature data were obtained. The third was at design temperature under steady load for 8 hours. Deflection and temperature data were obtained.

Transducer locations during the tests are shown in Fig. III-35. The following instrumentation was used:

- Seven dial gages, 0 to 0.25 inch, in 0.001 increments.

- Seventeen strain gages, temperature compensated with recorded readout.

- Thirteen thermocouples, chromel-alumel with recorded readout.

- Three accelerometers, with readout on trigger oscilloscope.

In Test 1 the top half of the furnace was removed. Loads were applied starting at 1/2 design load, continuing to design load and 1-1/2 design load, and then decreasing through design load, 1/2 design load, and zero load. Design load was 2400 pounds per tie rod for a total load of 90,000 pounds. Strain and deflection were measured and recorded between all load changes. The grid section was struck with a rubber mallet in three orientations at each load, and accelerometer data was recorded to obtain natural frequency.

In Test 2 the grid section was heated to design temperature (1300° F maximum) by the furnace, and the loading schedule used previously was repeated. Deflection was measured at each load.

In Test 3 the grid section was again heated to design temperature. Design load was applied and held for 8 hours. Deflection was measured every 15 minutes.

After testing, the test arrangement was disassembled and the grid section was inspected for straightness and cracks. The grid section was sound in all respects.

Figures III-36 through III-43 and Tables III-4 through III-7 show the results obtained from the simulated support grid test. The results confirm calculations. Vertical (axial) vibration results indicate very high natural frequency. Probably there was insufficient input energy to excite the fundamental mode. Of interest is the hysteresis displayed in both the strain and deflection curves.

Table III-4. Deflection data (in thousandths of an inch) for ambient temperature test.

Load	Deflectometer No.						
	1	2	3	4	5	6	7
0	0	0	0	0	0	0	0
1/3	14	26	44	47	47	43	14
2/3	30	66	92	95	95	86	30
Full	44	99	121	144	142	128	46
2/3	35	76	104	110	110	99	37
1/3	21	43	61	63	63	57	22
0	0	0	1	1	3	4	4

Table III-5. Stress data (in pounds per square inch) for ambient temperature test.

Strain gage No.	Load				
	1/3	2/3	Full	2/3	1/3
2	-840	-1225	-2025	-1435	-912
3	950	1890	2900	2000	840
4	-420	-280	-315	-280	-315
5	6560	13250	19700	14000	7180
6	9650	21750	29000	20800	10850
7	13100	30000	40200	29000	15200
8	15350	35210	47300	34500	17950
9	16200	37900	50900	37400	19750
10	16300	38200	51700	37800	20050
11	14500	30000	45150	32900	17550
12	70	350	595	420	210
13	840	1190	1820	315	-140
14	-195	-315	-455	-280	-70
15	350	700	1150	770	526
16	-12650	-27300	-41500	-32200	-18000
17	442	600	725	505	473
18	-1635	-2020	-2240	-1925	-2170

Table III-6. Vibration data (cycles per second) for ambient temperature test.

	Accelerometer No.			
	1	2	3	4
Vertical strike no load	228	2000		1030
Horizontal strike no load	-	-	870	-
Vertical strike 1/3 load	1450	1370	1090	1925
Horizontal strike 1/3 load	1400	1360	-	1875

Table III-7. Deflection data (in thousandths of an inch) for 8-hour, 1300° F test.

Load	Deflectometer No.						
	1	2	3	4	5	6	7
0	0	0	0	0	0	0	0
1/3	16	34	46	47	47	40	14
2/3	34	73	100	102	101	90	30
Full	53	115	157	163	160	143	47
2/3	41	88	119	123	122	107	37
1/3	25	36	73	75	75	64	22
0	3	6	5	6	5	3	2

The simulated support grid represents closely the operational support grid. Proof of high temperature adequacy of the actual support grid rests on this premise. Similarity is present if the deflections measured in the hydrostatic test of the support grid and those measured in the simulated support grid (at ambient temperature in each case) are identical. Referring to Figs. III-21 and III-36, it is seen that the slopes of the deflection curves differ by less than 2 mils per 1000 pounds of tie rod load. In addition, it can be shown that the beam deflection curve conforms to simple beam theory with a distributed load in both cases. Simple beam theory gives the deflection curve:

$$y = -\frac{1}{24} \times \frac{WL^3}{EI} \left(\frac{x}{L} \right) \left[1 - 2 \left(\frac{x}{L} \right)^2 + \left(\frac{x}{L} \right)^3 \right]$$

or for our case, with P = load per tie rod,

$$y = -1.28 \times 10^{-4} P \left(\frac{x}{L} \right) \left[1 - 2 \left(\frac{x}{L} \right)^2 + \left(\frac{x}{L} \right)^3 \right]$$

Figure III-44 shows the excellent agreement between the above expression and deflections measured on the simulated support grid. Maximum deflection of the support grid during the hydrostatic test was 0.114 inch at 145 psi and ambient temperature (see Fig. III-23). This deflection is in excellent agreement with the theoretical deflection at operating conditions: 0.113 inch at 120 psi and 1200° F (see Table III-2).

The hysteresis shows in both cases, because of friction between the plate and beams. With increasing load, the beam tends to stiffen. During unloading, the forces are reversed, tending to keep the deflections high. This phenomenon is helpful in damping mechanical vibrations.

12. Reliability

Factors of safety are given in Table III-8. These factors are conservative because most stresses are bending, compressive, or concentrated bending and do not indicate failure when compared to the yield strength of a ductile material. Impact stresses and strengths are based on static and yield values, but the overall result is still conservative. The table indicates the high structural reliability achieved in the design of the support grid.

Table III-8. Factors of safety.

Condition	Location	Stress (ksi)		Stress type	Material yield strength (ksi)	Relative minimum factor of safety
		Theoretical	Test			
Structural design point	Grid plate-link	11	11 and 12.5	Shear	87	14
	Grid plate-web	22	-	Bending (t)	145	6
	Grid plate at hole	38	-	Bending (t) and SC (t)	145	3.8
	Center beam-aft	38.2	32 and 38	Bending (t)	140	3.7
	Ring-beam support	48	-	Bearing	> 50	1.05
	Ring-tooth surface	<36	-	Bearing	> 50	1.4
	Ring-aft surface	11	14	Bending (t)	32	2.3
	Spacer ring bolt	34.4	-	Shear	87	2.5
	Spacer beam bolt	27.6	-	Bending (t)	145	5.2
	Grid plate bolts	95	-	Bending (t)	145	1.5
Stuck tube bundle	Ring pins	5.5	-	Impact-shear	87	16
	Grid plate-link	39	-	Shear	87	2.2
	Grid plate-web	66.3	-	Bending (t)	145	2.2
Handling	Grid plate-web at hole	120	-	Bending (t) and SC (t)	145	1.2
	Grid plate ring bolts	144	-	Impact-tensile	155	1.1
	Ring pins	28	-	Impact-shear	93	3.1

t = Tensile bending.
SC = Stress concentration.

The useful life of the support grid at structural design point conditions would be more than 1000 hours based on 0.5% creep strength. Satisfactory performance was demonstrated for 8 hours.

Nuclear radiation would not decrease the indicated structural reliability of the support grid. Detectable changes in the mechanical properties of Rene' 41 would occur during 10 hours of reactor running time. Irradiation would increase the yield strength and decrease the rupture strength and ductility. The maximum support grid fast neutron dose for 10 hours is about 2×10^{18} nvt. For this dose the 50-hour rupture strength of Rene' 41 is reduced from 132 ksi to about 113 ksi. Accompanying percentage elongation ranges are 4 to 9% and 1 to 2%, respectively.³⁷

13. Fabrication

Several methods of fabricating the grid plate were investigated. The primary concern was to discover economical methods of removing 75% of 7/8-inch-thick Rene' 41 plate. Typical samples were fabricated using chem-milling and electro-chem milling methods. Chem-milling left very rough edges and instability of the process was questionable since leakage under the masking might occur, resulting in scrap plate. Electro-chem milling was fast and left a clean, finished surface but required a separate anode for each different shaped section to be cut out. There were more than sixty different sections in the grid plate, so this method was not economical for fabricating only one item. The method finally used was end-milling with a tape-controlled machine. Required fabrication procedures are delineated in Ref. 38.

The simulated support grid was fabricated prior to the actual grid, to verify the adequacy of fabrication procedures and methods. A summary of the procedure used to machine the grid plate follows.

The Rene' 41 plate used was the largest available plate at the time.

It was rolled from an 875-pound ingot to a 7/8-inch-thick octagon.

The octagon measured about 55 inches across flats.

The plate was completely inspected dimensionally and by ultrasonic and penetrant methods. All surface rolling laps were removed. The plate was initially about 1/4 inch out of flatness. It was flattened satisfactorily on a hydraulic press.

The plate was heat treated at $1950 \pm 25^\circ \text{F}$ for 30 minutes and then air cooled. Next it was aged for 1/2 hour at 1400°F . This short aging treatment was intended to put the material in its most machinable condition by increasing its hardness and avoiding the gummy solution heat-treated material.

The plate was surfaced, then rough-machined by end milling using a tape-controlled mill. Each pass was about 0.10 inch deep, and after cutting half way through, the plate was turned over and finished from the opposite side (see Fig. III-45).

The plate was then aged at 1400°F for 15.5 hours. All round holes were drilled and surfaces ground.

Beam and ring fabrications were typical for the alloys concerned. There were some problems encountered in surfacing the Rene' 41 beams. There was too much wear during Blanchard grinding, so single-point planning was substituted.

All finished parts were completely inspected dimensionally and with fluorescent penetrant. Tensile specimens were made from parent material for each part. Two specimens from each beam and eight specimens from the grid plate were tensile tested at 1400°F .

The beams were set into the ring and aligned, and the grid plate clamped into position. Grid plate bolt holes were then match-machined. After final cleaning and inspection the structure was assembled, making sure there was uniform clearance around each bolt.

The next assembly step consisted of installing the support grid into the aft reactor duct. This was a critical phase because of the stringent teeth mating and ring pin match-machining requirements. This phase is discussed in Section IV-H.

C. Tie Rod Assemblies

1. Forces and Stresses

The pressure drop for the structure limit condition of 121 psi times the unit cell area of 19.785 in^2 gives a load of 2393 pounds on each standard tie rod. The cell plate spring load of 95 pounds is additive, giving a total load of about 2500 pounds. This is the average load for all tie rods. Due to differential expansion between the core ceramic tubes and the tie rods,

the maximum load is 3500 pounds, and appears on tie rods 25 inches out from the center of the core. The 1170° F temperature of these peripheral tie rods is lower than the 1530° F temperature of the inner tie rods, so the peripheral tie rods maintain higher stresses. Figure III-46 shows the loads and temperature of tie rods at different positions in the core, and Fig. III-47 shows the tie rod design temperature distribution.

The hottest tie rods, in the center of the core, have greater thermal growth axially, so they carry less load than the cooler ones.

The tie rod aft flange is insulated from the base block by five Hastelloy C washers, as shown in Fig. III-48. Analysis shows the maximum flange temperature at that point to be 1485° F. The platinum, 5%-ruthenium insulating cup will reach a temperature of 2150° F in the skirt portion, and 1775° F where it contacts the base block. The Hastelloy C washers reach a maximum temperature of 1765° F.

The fittings, springs, and cell plates at the forward end of the reactor run about 100° F hotter than the inlet air temperature, to a maximum of 1200° F. All parts have at least one surface cooled by moving air.

Tie rod stress and strength are shown in Fig. III-49. The weakening effect of neutron irradiation reduces the safety factor for a 3-hour structural limit condition to, in the worst case, 1.4. These figures are conservative for the Nevada Test Site runs of Tory II-C, since less irradiation will be experienced than in the Fig. III-49 tests.

The compressive stress of the Hastelloy C insulating washers is 6,800 psi for the highest loaded tie rod. Compressive creep tests of 2000° F Hastelloy C at 10,000 psi showed no measurable creep, and no harmful oxidation. A similar low compressive stress occurs in the platinum cup.

The cell plate springs operate at 57,800 psi at the hot, nominal position. The load produced is 94 pounds, relaxing to 85 pounds after one hour at 1150° F. This gives a 1.2 safety margin over the 71-pound load required to slide the reactor aft on the rails, to keep the core in contact with the base blocks. Figure III-50 shows the load relaxation of standard and control cell plate springs.

2. Cell Plates

The maximum cell plate load (5 g's forward) occurs during handling. This produces a 500-pound load on the plates at low temperatures.

The spring retainers (used only prior to operation of the reactor) form a load circuit around the front support springs, so that the 5-g forward load is transmitted directly to the front support grid. A maximum bending stress of 21,500 psi is produced in the cell plates by this load. Room temperature yield strength is 85,000 psi for 410 stainless steel.

The 1150° F load of 95 pounds produced by the cell plate springs causes a bending stress of 20,500 psi in the cell plates. The 3-hour stress rupture strength is 28,000 psi. (The 20,500 psi stress is a concentrated stress around the holes and would creep out to a lower stress in 3 hours.)

The 1200° F yield strength of 410 stainless steel is 35,000 psi.

3. Tie Rod Nut

The shear stress in the 410 stainless steel standard tie rod nuts is 4660 psi for a 3500-pound load. In the control tie rod nut the maximum stress is 11,400 psi. Tests of these nuts at temperature have shown their strength to be adequate.

4. Tie Rod Brazed Joints

J8100 alloy was used to vacuum braze the forward and aft flanges. The shear stress in the forward joint is 6,400 psi, and 2,080 psi in the aft flange joint. Figure III-51 shows the braze strength of cylindrical joints made with J8100. In all tubing tests conducted, the joint never failed. The strength of the brazed joint was, in all cases, greater than the ultimate strength of the tubing.

D. Base Blocks

1. Function and Environment

The base blocks are structural members that retain the ceramic tubes within the reactor core against the axial pressure drop. This is accomplished by using a large number of base blocks, each supporting a longitudinal bundle or cluster of ceramic tubes. Since the fueled tubes are not mechanically interlocked, failure of any one base block could constitute failure of the entire reactor. Failure is defined as physical separation of the base block from its tie rod, or some mode which permits displacement of the fueled tubes within the reactor.

The base blocks are hexagonal plates, perforated to provide passage of the reactor exit air. A central hole receives the base block retaining tie rod, and the load is transmitted via the tie rod to the support grid. The loading on the base block, in its simplest analysis, is that seen by a centrally supported, uniformly loaded, hexagonal plate. The loading produces bending stresses which are tensile on the upstream face and compressive on the downstream face.

The physical location of the base blocks in the exhaust section of the reactor (Fig. I-3) exposes them to severe conditions of temperature, pressure, and gas flow. The operating environment considered for design was:

Operating temperature (nominal)	2180° F
Operating temperature (extreme)	From ambient to 2430° F
Operating pressure	180-220 psia
Gas flow rate	150-250 lb/sec ft ²
O ₂ content of gas stream	22 w/o
Gas velocity	Mach 0.4
Design life	10 hours
Number of cycles	10-20
Axial pressure (load-producing)	145 psi

2. Material Selection

The material chosen for the base blocks was the columbium-base alloy F-48. Nominal composition of this alloy is 79% Cb, 15% W, 5% Mo, and 1% Zr.

F-48 was developed for use as a structural material having superior strength, particularly in creep exposure, in the temperature range of 2,000 to 2,500° F. Melting and fabricating techniques for producing sheets of this material had been previously developed in the Air Force re-entry vehicle program. Pilot quantities of metal were produced in this program, and it became reasonable to assume availability of reliable plate material for Pluto program reactor application.

A preliminary program to evaluate the alloys F-48, Cb 1 Zr, 90 Ta 10W, Cb 10Ti 10Mo 0.1C, and several other candidate alloys was conducted. F-48 was chosen after giving due consideration to high-temperature strength, oxidation resistance, creep stress, fabricability, and elongation.

An extensive program to further evaluate F-48 as an engineering material for use in a high-temperature air-flow environment ensued. Tensile and rupture strengths were determined up to 2500° F, and high-temperature flow tests were performed to study the oxidation behavior of the basic metal. In addition, a process was developed for applying an oxidation-protection coating to the alloy.

3. Material Fabrication Method

Consolidation of the F-48 alloy is accomplished in a vacuum arc furnace. The alloying metals (columbium, tungsten, molybdenum, and zirconium) are initially combined in powder form, mixed thoroughly, compacted hydrostatically, and sintered in hydrogen to approximately 85% density. The sintered form, a round bar about 2 inches in diameter, becomes the electrode during vacuum arc melting.

After vacuum casting, the sides and ends of the ingot are machined to produce a cylindrical billet. At this stage the billet is inspected for composition and soundness. A sample of metal is taken from either end for chemical analysis; fluorescent dye penetrant and ultrasonic inspections are conducted for surface and internal cracks; and X-rays are taken to check for homogeneity.

Billets to be processed further are sealed in a jacketing can of sacrifice metal. This step is necessary to avoid contamination of the F-48 by air during subsequent processing at high temperatures. The can is about 1/8 inch thick. Although the can is attacked freely by oxygen in the air while at the high processing temperatures, sufficient thickness is provided to last through the final operation.

Breakdown of the canned billet structure is accomplished by extrusion. Upset forging has not been successful with F-48 because of excessive cracking at the unconfined sides of the forged piece. The high side compressive restraint offered by a die and container during extrusion appears to be necessary for successful initial breakdown. The billet is preheated to approximately 3000° F in an argon atmosphere. It is then elevated on a loading table into alignment with the container of the extrusion press, and extruded.

After extrusion the bar is approximately 2-1/8 × 4 inches in cross-section. At this point, a radiographic inspection is conducted to determine soundness and homogeneity. If it is to be used for further processing, the bar is stress-relief annealed.

Plates of F-48 are then produced from segments of the extruded bar by cross-rolling. The original thickness of 2-1/8 inches is reduced to approximately 1-3/8 inches in five passes through the rolls. Thickness is reduced about 9% with each pass. Rolling temperatures are maintained at approximately 2400° F by reheating the plate in argon between passes.

After reduction to final thickness the plate is stress-relieved for one hour at 2200° F, and the remaining sacrifice metal cladding is removed by pickling in a solution of nitric and hydrofluoric acids. After light machining to true up the surfaces, final inspection is performed, and the plate blank is ready for machining into a base block.

4. Material Properties

a. General

An investigation of the properties of F-48 was made for LRL.³⁹ To eliminate heat-to-heat variation in the properties determined, all specimens were taken from a single heat, Heat 110. Subsequently, heat-to-heat variations were studied.

Tensile specimens in longitudinal, transverse, and 45° directions were tested at temperatures ranging from 75 to 2500° F. All elevated temperature tensile tests were conducted in a vacuum greater than 0.1 mm of mercury. Load extension curves were recorded autographically. Nominal strain rates of 0.005 per minute were maintained through the 0.2% offset yield strength, and increased to 0.05 per minute thereafter. Yield strength (0.2% offset), ultimate tensile strength, elongation in 1 inch, and reduction in area were recorded.

Creep specimens were tested at 2200° F in a vacuum greater than 0.1 mm of mercury. Using optical readout techniques, deformation values were obtained directly from extensometers attached to the shoulders of specimens. Four specimens were tested in longitudinal, transverse, and 45° directions at each temperature. Stress levels at 2200° F ranged from 10,000 to 30,000 psi, whereas those at 2500° F ranged from 3,500 to 15,000 psi. Time to reach 0.5 and 1.0% creep strain was recorded for all of the specimens. Several of the specimens tested at the higher stress levels were allowed to proceed to rupture.

Impact specimens were tested at temperatures between 300 and 800° F in a Tinius Olsen testing machine with a maximum capacity of 200 inch-pound and a striking velocity of 11.37 fps. The specimens were clamped into a steel adapter block, and the entire assembly was heated in an electrical-resistance furnace to approximately 20° F above the desired test temperature. The assembly was then transferred to the testing machine, and the pendulum was released when a thermocouple in contact with the specimen indicated that the specimen had cooled to the desired test temperature.

Linear-thermal-expansion measurements were made in a recording dilatometer between room temperature and 2500° F. The specimens were protected with a vacuum of about 2×10^{-5} mm of mercury. Expansion versus temperature curves were automatically recorded. Specimens were cylinders 1/8 inch in diameter and 3 inches long.

Thermal conductivity measurements were made of temperatures between 150 and 1650° F, using a longitudinal, steady-state, comparative method.⁴⁰ Briefly, the method consists of heating the end of the specimen; measuring (with five thermocouples spaced along the specimen) the temperature gradient produced; and determining the rate of heat flow through the specimen by means of a metal standard (Armocotron) of known thermal conductivity attached to the cold end of the specimen. Radial heat flow was minimized by an insulating layer of bubbled alumina, and an encircling guard tube adjusted to a temperature as near as possible to that of the specimen. Measurements were made in steps of increasing temperatures to the maximum temperature, after which at least one equilibrium measurement was obtained at a lower temperature. This duplicate measurement is a check on thermocouples and changes which may have taken place in the specimen. The specimens were cylinders 7/8 inch in diameter and 5-1/4 inches long, and were protected by a vacuum of approximately 2×10^{-5} mm of mercury.

b. Tensile properties

The tensile properties of Heat 110 between room temperature and 2500° F are illustrated in Fig. III-52. Some anisotropy is apparent. Strengths tend to be highest in the longitudinal direction and lowest in the 45° direction at a given temperature. The ductility tends to be lower in the transverse direction than in either the longitudinal or 45° direction.

A rapid increase in strength and decrease in ductility with decreasing temperature is evident between approximately 700° F and room temperature. This is indicative of a ductile-to-brittle transition characteristic of all body-centered-cubic metals. The low ductility (at room temperature) of F-48 in the lightly-worked, partially recrystallized condition indicates a ductile-to-brittle transition temperature range with a lower limit at about room temperature.

c. Creep properties

The creep behavior of Heat 110 in the longitudinal, transverse, and 45° directions at 2200° F is illustrated in Fig. III-53.

d. Impact behavior

The impact behavior of Heat 110 over the ductile-to-brittle transition temperature range is illustrated in Fig. III-54. The impact transition curves in the 45° and transverse directions are characterized by a more gradual decrease in impact energy with decreasing temperature than is the case in the longitudinal direction. Thus, impact resistance above approximately 450° F is somewhat superior in the longitudinal direction. There is little apparent difference in impact resistance between the 45° and transverse direction.

A considerable amount of scatter is evident in the data in the temperature range of decreasing impact energy. This is characteristic of the behavior of body-centered-cubic metals tested in the transition range.

e. Thermal expansion

Figure III-55 describes the thermal expansion of F-48 (Heat 110) between 200 and 2500° F. Coefficients of thermal expansion are greatest in the longitudinal direction and lowest in the 45° direction. Differences between the transverse and 45° directions are relatively small. This anisotropy is greatest at low temperature.

f. Thermal conductivity

The thermal conductivity of F-48 (Heat 110) between 250 and 1650° F is illustrated in Fig. III-56. Some anisotropy is evident at temperatures below approximately 750° F. At these temperatures, thermal conductivity is

greatest in the transverse direction, and least in the longitudinal direction. Above 750° F this anisotropy disappears.

5. Effect of Material Fabrication Variables

Plates prepared by extrusion and rolling from 6- and 8-inch diameter ingots (Heats 111 and 112 respectively) were evaluated to determine the effect of initial ingot size on mechanical properties. These ingots were extruded and rolled to final dimensions without annealing. Thus, the plate fabricated from 8-inch diameter ingots was worked to a greater degree than plate fabricated for 6-inch diameter ingots. Evaluation consisted of a determination of the tensile and impact transition temperatures in the longitudinal, transverse, and 45° directions.

a. Tensile transition behavior

The tensile properties of F-48 fabricated from 6- and 8-inch ingots in the transition temperature range are illustrated in Fig. III-57. The strength of the 8-inch plate is slightly higher, most likely due to more extensive work hardening and the lower degree of recrystallization. As in the detailed study of Heat 110, strength was highest in the longitudinal direction for both plates.

Ductility over this temperature range was generally highest in the longitudinal direction. The 8-inch plate generally showed better ductility at a given temperature and direction than did the 6-inch plate. This was particularly true in the transverse and 45° directions at 200 and 300° F. The improvement in ductility with increased ingot size is consistent with the expected decrease in transition temperature with increased working reduction. Such behavior is characteristic of molybdenum and tungsten or other niobium base alloys containing these elements.

b. Impact transition behavior

Figure III-58 illustrates the impact transition behavior of 6- and 8-inch plates in the longitudinal, transverse, and 45° directions. Again, as in the detailed evaluation of Heat 110, impact resistance was best in the longitudinal direction for both plates at a given temperature. The 8-inch plate offered better impact resistance than the 6-inch plate in all directions and at all temperatures. Greatest overall improvement was noted in

the transverse and longitudinal directions. It is also apparent that, although the impact resistance in the transverse and 45° directions was not appreciably different for the 6-inch plate, an 8-inch plate had decidedly inferior impact resistance in the 45° direction between approximately 400 and 600° F. This result was to be expected, as more extensive working would tend to maximize anisotropy in the 45° direction, relative to either the longitudinal or transverse directions.

Compared to impact behavior earlier for Heat 110 (6-inch plate), this 6-inch plate material (which was uniformly 50% recrystallized) exhibited somewhat higher transition temperatures (see Fig. III-54). However, the 8-inch plate exhibited significantly lower transition temperature than determined for Heat 110.

c. Heat-to-heat property variations

Triangular corner sections of plate from sixteen heats were used to assess heat-to-heat variations in mechanical properties. Evaluation consisted of tensile tests at room temperature and 2200° F, and creep tests at 2200° F.

The room temperature tensile properties of the seventeen heats examined (including Heat 110) are given in Table III-9. There is a fairly wide scatter in strength values among the various heats. For example, Heat 152 exhibited the lowest average yield strength (approximately 92,500 psi) and Heat 174 had the highest yield strength (105,200 psi). However, all but one heat exhibited average yield strengths within approximately 5% of the average value for all of the heats (98,600 psi). The exception was Heat 174, which exhibited a yield strength appreciably greater than that of any of the others.

Table III-9. Heat-to-heat variations in room-temperature tensile properties of F-48 (transverse direction).

Heat	Plate	0.2% offset yield strength (1000 psi)	Ultimate tensile strength (1000 psi)	Elongation (%)	Reduction in area (%)
152	1-34	95.4	106.4	14	33.0
		89.7	99.5	6	10.5
154	1-39	97.8	103.8	4	8.8
		95.5	102.5	2	5.6
155	—	96.3	117.4	4	7.1
		100.3	107.3	11	29.5
156	2-26	97.3	108.7	8	13.1
		97.1	97.1	0	0
157	—	94.4	107.8	6	16.4
		98.2	114.4	6	9.0
158	2-4	96.8	107.7	8	16.1
		96.8	106.1	22	44.3
159	1-23	96.5	105.0	1	1.7
161	1-44	103.2	106.4	2	2.4
		100.0	105.8	4	7.2
162	1-43	101.7	114.2	12	25.0
		100.0	111.7	4	16.1
163	1-47	99.4	110.2	8	20.5
		104.4	108.2	2	2.5
166	1-31	94.0	114.3	14	18.3
		93.1	113.4	13	17.1
174	1-55	105.2	117.3	10	12.1
		105.2	118.2	12	17.8
175	1-32	101.2	113.4	8	14.8
		100.5	116.8	12	23.8
215	1-10	101.1	108.3	2	4.0
		99.7	103.2	4	12.9
216	1-11	97.3	106.1	8	12.9
		98.8	103.2	2	2.4
217	1-61	100.2	109.7	16	44.3
		101.8	104.5	0	4.1
110	—	96.2	108.4	2.0	2.0
		93.6	105.8	1.0	2.1
Average		98.6	108.7	7.3	14.3

Tensile properties at 2200° F are given in Table III-10. Considerable scatter is again evident in strength values. For example, Heat 166 had the lower average tensile strength (40,100 psi) and Heat 174 had the highest average tensile strength (49,600 psi). Heats exhibiting relatively high room temperature strength generally also showed relatively high strengths at 2200° F. Although a large amount of scatter is evident in the ductility, no heat showed a tendency toward brittleness at 2200° F.

Figure III-59 illustrates the most probable stress-time curves for 1% creep strain of the different heats for comparison purposes. Similar to what was observed for tensile properties, the creep curves of fourteen of the heats are grouped into a well-defined scatter band. In particular, Heat 174 showed markedly superior creep resistance. By excluding this heat, the scatter in creep behavior is reduced to a much more acceptable value.

6. Mechanical Stress in Base Blocks

To find base block stresses analytically, the stresses in a solid plate are determined and converted into those that would be seen in a perforated plate. The conversion of solid plate stresses into perforated plate stresses can be accomplished by Horvay analysis which, in essence, provides a set of equivalent elastic parameters based on porosity.

The base block analyzed was an A type as shown in Fig. III-60. The flat surface area of one side of this plate (as a solid) is 20.11 in². A circular solid plate having this area was considered to represent a solid base block. Thus, the radius of the circular plate is 2.53 inches. This plate was considered to (1) have a uniform pressure over one entire surface, and (2) be supported over a concentric circular area of radius r_0 . The r_0 dimension equals the radius of the tie rod cup.

The equations used to determine the basic solid plate stress (Ref. 41, Table 10, Case 11) assume (1) the plate is flat, of uniform thickness, and of homogeneous isotropic material; (2) the thickness is not more than one-fourth of the least transverse dimension, and that maximum deflection is not more than one-half the thickness; (3) all forces are normal to the plane of the plate; and (4) the plate is nowhere stressed beyond the elastic limit.

Table III-10. Heat-to-heat variations in 2200° F tensile properties (transverse direction).

Heat	Plate	0.2% offset	Tensile	Elongation	Reduction
		yield strength (1000 psi)	strength (1000 psi)	(%)	in area (%)
152	2-17	39.9	41.0	15.0	15.7
		37.0	40.8	29.0	38.7
154	1-41	40.8	42.3	18.0	28.2
		39.2	41.7	12.0	21.9
155	1-20	41.8	46.6	15.0	44.8
		41.8	44.9	20.0	42.4
156	1-36	36.9	40.1	6.0	6.6
		40.0	43.2	30.6	33.5
157	1-24	39.4	42.5	26.0	59.0
		39.9	43.6	13.0	43.4
158	2-2	35.6	41.2	30.0	70.6
		38.6	43.1	26.0	29.8
159	1-15	38.8	42.2	14.6	31.7
		38.0	42.0	23.9	34.2
161	2-20	38.9	43.1	24.0	37.5
		44.9	47.1	30.0	56.5
162	1-45	40.7	43.8	24.0	35.8
		34.4	41.4	28.0	64.1
163	1-35	37.1	41.5	18.0	26.3
		39.0	43.7	24.0	36.3
166	1-18	37.2	41.0	16.0	28.5
		33.9	39.2	14.8	19.1
174	2-12	46.9	49.5	—	70.3
		41.3	49.6	40.0	77.0
175	1-46	37.1	43.5	16.0	17.2
		43.7	48.1	20.0	46.1
215	1-4	34.5	41.8	20.0	45.8
		40.0	43.6	24.0	44.9
216	1-14	37.5	45.6	36.0	65.1
		37.6	44.0	22.0	47.1
217	1-33	43.6	45.7	—	45.2
		41.3	45.9	16.0	23.9
110	—	38.8	42.5	23.0	38.0
		42.6	46.0	10.3	37.4
Average		39.2	43.5	21.6	40.2

At center: Max.

$$\sigma_r = \sigma_t = \frac{-3W}{2\pi mt^2} \left[(m+1) \log \frac{a}{r_0} + \frac{1}{4} (m-1) \left(1 - \frac{r_0^2}{a^2} \right) \right] \quad (1)$$

At 1st tangential ligament: $r < r_0$

$$\sigma_{t_1} = \frac{-3W}{2\pi mt^2} \left[(m+1) \log \frac{a}{r_0} + \frac{1}{4} \left(1 - \frac{r_0^2}{a^2} \right) \left\{ (m-1) - \frac{(m+3)r^2}{r_0^2} \right\} \right] \quad (2)$$

At 1st radial ligament: $r > r_0$

$$\sigma_{r_1} = \frac{-3W}{2\pi mt^2} \left\{ (m+1) \log \frac{a}{r} + \frac{1}{4} (m-1) \left[1 - \frac{r_0^2}{a^2} + \frac{r_0^2}{r^2} \right] - m + \frac{(3m+1)r^2}{4a^2} \right\} \quad (3)$$

The appropriate values to use in Eqs. 1, 2, and 3 are:

W = 2800; total distributed load, pounds

m = 4; reciprocal Poisson ratio

a = 2.53; plate radius, inches

t = 1.110; plate thickness, inches

r_0 = 0.60; tie rod cup radius, inches

r = 0.394; radial distance to center of first tangential ligament, inches

= 0.682; radial distance to center of first radial ligament, inches

Figure III-60 shows the points taken for stress determination. Application of these values to Eqs. 1, 2, and 3 gives the following results:

At the center 2150 psi

At first tangential ligament 1950 psi

At first radial ligament 1100 psi

The stress distribution as a function of radial distance for the solid circular plate is shown in Fig. III-61.

In converting the solid plate stresses to perforated plate stresses, new elastic constants E^* and γ^* are used. That is, it is considered that a perforated plate will deform as if it were a solid plate of effective elastic modulus:

$$E^* = (1 - \gamma^*) \frac{2\bar{h}}{2R} \times E,$$

where $2\bar{h}/2R$ is the "solidity" of the plate; the ratio of mean ligament width to center-to-center span of the holes.

With γ^* defined as the effective Poisson ratio,

$$\gamma^* = 1 - \frac{12 h'^3/R^2 \bar{h}}{1 + (16.2 + 7.2\nu)h'^3/R^2 \bar{h}}$$

Values of $2\bar{h}/2R$ for several points are shown in Fig. III-62. These values provide the stress concentration factors shown in Fig. III-63.

The stresses for the perforated plate, using these stress concentration factors, become:

First tangential ligament: 1950 psi $\times$ 3 = 5850 psi

First radial ligament: 1100 psi $\times$ 2.15 = 2750 psi

The 5850 psi stress value obtained for the first tangential ligament is conservatively high, since it does not take into consideration the resistance to compressive deformation offered by the relatively cool (1500° F) tie rod. If the tie rod is assumed to offer significant resistance, then the base block must approach the behavior of a solid plate in the central region, or that of a plate having a $2\bar{h}/2R$ that is not much less than unity. The stress concentration factor is then reduced from 3 to 2, and the stress level becomes:

First tangential ligament (modified): 1950 psi $\times$ 2 = 3900 psi

As given previously, the least values for the seventeen heats tested for yield strength and ultimate tensile strength are 33,900 psi and 39,200 psi, respectively, at 2200° F. The 10-hour, 2200° F, 1% creep strength of F-48 is 15,000 $\pm$ 3,000 psi. Safety factors are therefore appreciable.

The B base block differs from the A type analyzed above in that it has a more favorable load distribution, and consequently a lower stress (see Fig. III-64). This more favorable load distribution is brought about by the fact that B base blocks are those designed to accommodate control tie rods, and thus are supported at a larger radius. That is, r_0 is equal to 0.824 inches instead of 0.60 inches, as in the case of type A base blocks; and r , the radial distance to the centerline of the first tangential ligament is 0.796 inches instead of 0.394. The resulting perforated plate stress is, for example, computed to be:

First tangential ligament: 3780 psi

7. Protective Coating

The columbium alloy F-48 was chosen primarily because of its high strength at elevated temperatures. Although its strength level was well in line with the projected design concept, full utilization of the metal (10-hour life, for example) could not be realized without providing a protective coating. F-48, as with all columbium alloys, suffers the loss of useful structural metal by conversion to the oxide Cb_2O_5 when exposed to high-temperature air. The addition of high-pressure and high-velocity air flow accelerates this reaction. Also, the base metal undergoes property changes due to intrusion of oxygen below the metal surface. This is of particular consequence at low temperature, where the metal becomes embrittled.

A state-of-the-art survey of protective coatings was initially made. Information and data were acquired in this survey by communicating directly with organizations actively engaged in the coating business. The survey indicated that coatings fall into three general categories according to their method of application:

1. Hot-dipped or slurry-dipped coatings.
2. Flame-sprayed coatings.
3. Vapor or pack deposition coatings.

Each coating type had specific advantages and limitations. All were in the early stages of development, and none had had extensive testing, either statically or in a simulated air flow environment. No major work had been done in obtaining a coating/substrate match, or in studying the effect of one upon the other.

It was decided that the pack deposition process would be most compatible with the complex geometry of the block. The development program required consisted of two phases; an environmental test program, and a parameter study in the coating processes. The eventual product of the interrelating of these two efforts was a coating suitable for the Tory II-C environment.

The coating process which evolved is summarized in Refs. 2, 42 through 47. Further information is not given here owing to the higher security classification involved.

8. Static Proof Test

To verify the ability of the base block to sustain loads at high temperature in an oxidizing atmosphere, two static proof tests were performed.

First, a coated base block was uniformly loaded in air with 20,750 pounds for 3 hours at 2300° F. Post-run inspection revealed that permanent deformation of the block was approximately 0.2 inch. No cracks due to the test were observed. The coating remained intact and functional. The load applied was 7.4 times design load. The maximum computed elastic stress, allowing for the presence of a tight centering pin, was 28,800 psi, or five times the design stress.

Another coated base block was then uniformly loaded in air with 4200 pounds for 10 hours at 2300° F. Post-run inspection revealed permanent deformation of about 1/16 inch. No cracks were observed, and the coating remained intact and functional. The load applied resulted in a stress 1-1/2 times the design stress. Figure III-65 shows the block after testing.

Figure III-66 depicts the test arrangement used in both tests. (Photographed after the tests.) The entire assembly was located in the test furnace. A water-cooled ram with a silicon carbide tip rested on two other silicon carbide pieces which increased the load area to 17.3 square inches. The bottoms of these two pieces were hexagonal. The lower piece rested on aluminum powder contained on the sides by two joined pieces of nickel. The nickel pieces had a hexagonal interior profile and circular exterior. To keep the alumina powder from falling through the base block holes, a 1/8-inch thick hexagonal platinum plate with an alumina flame spray was laid over the base block. The alumina powder was counted on to act as a semifluid and transmit uniform load to the base block, irrespective of relative motions of the block, nickel, platinum, and ram. The base block was supported by a conical section of silicon carbide whose upper area simulated the standard tie rod end. An alumina pin, glued to the cone, fitted the base block center hole as a positioner. By successive blocks of silicon carbide and alumina bricks, the load was ultimately carried to the furnace hearth.

The ram load was controlled by a programmed hydraulic servo system. A load cell read the load to ± 100 pounds. Temperatures were monitored by two platinum-platinum rhodium thermocouples imbedded in the nickel pieces. One was read on a recorder; the other on a sensitive ($\pm 2^\circ$ F) potentiometer.

The thermocouples were pre- and post-run calibrated. Dial gages were arranged to give total deflection of the ram, excluding furnace motion.

The procedure was to raise the temperature of the test assembly to 2300° F, and then linearly increase the load in 3 minutes. The load was held at 20,750 pounds and 4200 pounds for 3 and 10 hours in the first and second tests, respectively. The load was then linearly decreased to zero in 3 minutes. Dial gages were read every 2 minutes and the temperature (potentiometer) as often as necessary.

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Fig. III-1. Standard tie rod assembly.

Fig. III-2. Support grid assembly, front view.

Fig. III-3. Support grid assembly, rear view.

Fig. III-4. Support grid ring.

Fig. III-5. Total nuclear heating density distribution in forward axial regions.

Fig. III-6. Nuclear heating distribution, Mach 3.0.

Fig. III-7. Support grid convection coefficients, Mach 3.0.

Fig. III-8. Support grid steady state temperature distribution.

Fig. III-9. Support grid transient temperatures.

Fig. III-10. Grid plate nomenclature.

Fig. III-11. Stress distribution in webs.

Fig III-12 Support grid beam spacers

Fig III-13 Control tie rod radial position

Fig. III-14. Support grid assembled in reactor duct for hydrostatic test.

Fig. III-15. Support grid hydrostatic test.

Fig. III-17. Gage locations, support grid plate.

Fig. III-18. Gage locations, reactor duct teeth.

Fig. III-19. Displacement gage locations, support grid beams.

Fig. III-20. Strain gage locations, support grid beams.

Fig. III-21. Deflection at center of longest beam, hydrostatic test.

Fig. III-22. Deflections as a function of radial distance at 145 psig, (3000 lb/tie rod), hydrostatic test.

Fig. III-23. Maximum beam deflections, hydrostatic test.

Fig. III-24. Strain vs pressure, Gage 6, hydrostatic test.

Fig. III-25. Strain vs pressure, Gage 4, hydrostatic test.

Fig. III-26. Strain vs pressure, Gages 44 and 45, hydrostatic test.

Fig. III-27. Strain vs pressure, Gages 2 and 3, hydrostatic test.

Fig. III-28. Strain vs pressure, Gage 12, hydrostatic test.

Fig. III-29. Strain vs pressure, Gages 29 and 30, hydrostatic test.

Fig. III-30. Strain vs pressure, Gages 15 and 16, hydrostatic test.

Fig. III-31. Strain vs pressure, Gages 27 and 28, hydrostatic test.

Fig. III-32. Support grid test section.

Fig. III-33. Support grid test section, top view.

Fig. III-34. Support grid test section, side view.

Fig. III-35. Transducer locations for support grid test.

Fig. III-36. Deflection at center of beam, ambient temperature.

Fig. III-37. Beam deflection as a function of distance from center, ambient temperature, load increasing.

Fig. III-38. Stress at outer fiber of beam as a function of distance from center, ambient temperature, load increasing.

Fig. III-39. Deflection at center of beam, 1300° F.

Fig. III-40. Beam deflection as a function of distance, 1300° F, load increasing.

Fig. III-41. Beam deflection, 8-hour test, 1300° F (2400 lb per tie rod).

Fig. III-42. Approximate span temperature distribution, 1300° F.

Fig. III-43. Permanent deformation in simulated support grid after 8-hour, 1300° F test.

Fig. III-44. Comparison of theoretical deflection with experimental data from simulated support grid test (3000 lb per tie rod).

Fig. III-45. Grid plate during rough machining.

Fig. III-46. Tie rod loads and temperatures for structural limit conditions.

Flow channel diam	0.576 in	Exit Mach number	0.536
Porosity	0.7391	Peak wall temp	1522 $^{\circ}\text{F}$
Inlet total temp	1065 $^{\circ}\text{F}$	Power to air	34.1 kW
Exit total temp	1230 $^{\circ}\text{F}$	Average material power density	5.72 MW/ ft^3
Inlet total pressure	342.3 psia	Flow rate	0.7224 lb/sec
Exit total pressure	275.4 psia	All dimensions at	60 $^{\circ}\text{F}$

Fig. III-47. Central tie rod, structural design condition.

Fig. III-48. Tie rod aft flange at base block.

Fig. III-49. Tie rod stress and strength.

Fig. III-50. Load relaxation of cell plate springs (1150° F).

Fig. III-51. Braze strength of J8100 joints.

Fig. III-52. Tensile properties of F-48 (Heat 110).

Fig. III-53. Creep behavior of F-48 at 2200° F (Heat 110).

Fig. III-54. Impact resistance of F-48.

Fig. III-55. Thermal expansion of F-48 (Heat 110).

Fig. III-56. Thermal conductivity of F-48 (Heat 110).

Fig. III-57. Effect of ingot size on the tensile transition behavior of F-48.

Fig. III-58. Effect of ingot size on the impact transition behavior of F-48.

Fig. III-59. Probable creep strengths of F-48 at 2200° F.

Fig. III-60. Base Block A stress analysis.

Fig. III-61. Circular plate stress distribution.

Fig. III-62. Base block stress analysis (Horvay).

Fig. III-63. Horvay curves, stress concentration factors vs porosity.

Fig. III-64. Type B base block.

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Fig. III-65. Base block after 10-hour, 2300° F load test.

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Fig. III-66. Base block load test arrangement.

IV. REACTOR SUPPORT (TRANSVERSE)

A. General

The side support system of the reactor must perform two basic, related functions:

Supply a clamping pressure sufficient to maintain the stability of the array of tubes in the reflected core of the reactor.

Transmit transverse loads between the reactor pressure vessel and the reactor core.

The clamping and support functions may be provided separately or combined into a single system. Of the various schemes proposed, three different approaches are described below.

- a) Separate functions. Clamp the bundle of tubes by a corset-like flexible member and support the resulting rigid assembly in two planes; one at the front of the core, and one at the rear.
- b) Pure combination. Support the core by springs pushing radially inward on the surface of the tube bundle, and radially outward on the inner surface of the duct. Initially preload the springs to produce the clamping effect. Following the initial preload, the springs can react transverse acceleration loads by means of differential deflection among the various springs. (A design using this concept was called the "buggy spring" system.)
- c) Impure combination. Wrap the core with circumferential springs in tension, and utilize the shape of these springs to provide supporting surfaces distributed along the entire length of the reactor.

The Tory II-C design utilizes the side support scheme of (c), above.

Figure IV-1 shows a typical section of the side support system, which includes the circumferential square wave springs, support rails, wiper bars and wiper springs, and the reactor duct.

The square wave springs (corrugated springs) are fabricated of Rene 41. One wave length of the square wave shape is a convolution, of which there are 120 in the 360° of arc. Except for two narrow springs, each spring is about two inches wide, and there are 32 springs axially. Each 360° spring is made up of six segments which dovetail together as shown in Fig. IV-2.

Detailed discussion of the spring criteria, design, matrix stability, and fabrication is presented in subsequent portions of this section. Also discussed are the rails and wiper bars, wiper springs, the shroud assembly, and the reactor duct. Additional details and photographs may be found in Section VI, Reactor Assembly.

B. Matrix Stability

1. Theoretical

The core is a bundle of hollow hexagonal tubes held in a stable configuration by elastically strained circumferential springs which exert a radially inward clamping pressure. The clamping pressure is called p_o , as explained in Fig. IV-3.

If the core bundle is held in a position with the longitudinal centerline vertical, a very small value of p_o is sufficient to keep the tubes in a proper group. If the core is then laid over into a horizontal position, the value of p_o must be increased. If the core is caused to accelerate transversely through space so as to augment the effect of gravity, the value of p_o must be increased still further. Should p_o be inadequate, the structural integrity of the bundle could not be maintained.

One of the modes of failure of the array would be an opening up or "clam-shelling" at the bottom of the circular cross-section, the term bottom corresponding to the convention that weight loads point downward. Along with the spreading apart at the bottom, there would be a wedging action at right angles, as shown in Fig. IV-4. The regularity of this wedging action would depend upon friction between tubes and the resulting tendency to produce uneven sliding, accompanied by rotation.

Another mode of failure of the array would be by sliding along a jagged shear plane as shown in Fig. IV-5. For the worst condition, the load is assumed to be directed at approximately 60° to the assembly vertical. This is one of three worst load orientations for shear because the greatest amount of friction is required to prevent sliding.

In Ref. 48, particularly pages 28, 29, 30, and 42, the core pressure required to prevent either separation or sliding is studied in detail. (Similar discussion may be found in Ref. 49.) The results of these and later studies indicate:

Let

p_o = core pressure, psi

R_c = radius R_c of Fig. IV-3, inches

γ = average weight density, lb/in³, within circle of radius R_c

n = transverse acceleration of system expressed as a multiple of 1 g.

If the pressurization is such that $p_o > 2 n \gamma R$, or, if the acceleration is such that $n < p_o / 2 \gamma R$, on the average, separation and sliding action are completely avoided. The term on the average, is used to draw attention to the presence of load concentrations in the array. That is, the generalized conclusions being given here refer to a perfect array of perfect ceramic tubes. The actual load on any given tube may be above or below the average as a result of minute dimensional variations. It should also be said that the conclusions listed here refer to any two-dimensional slice of the core, although, because of stagger, any given slice may help or receive help from its neighboring slice.

The criterion stated in the preceding paragraph will completely prevent, on the average, the loss of compression among tubes. That is, it will permit compression to approach zero at the very bottom of the array. A considerable amount of acceleration load is permissible beyond this point, so that it is reasonable to use the criterion:

$$p_o > n \gamma R,$$

or

$$n < \frac{p_o}{\gamma R}.$$

Such a design permits the action of Fig. IV-4 to work its way up to the center of the core. However, as pointed out in Ref. 48, p. 43, the tubes in the upper half of the core all remain in compression. Figure IV-6 illustrates the distribution of compressive load in the upper half. The distortion of the array is limited by the existence of total compression in the upper half, plus partial compression combined with wedging in the lower half.

As indicated in Fig. IV-6, the tangential component of compressive stress along the upper half of the vertical centerline is assumed to be distributed linearly. For equilibrium, the tangential pressure at the top is four

times the radial pressure, where $n = p_o/\gamma R$. Thus, under load, there is a kind of elliptical pressure distribution at the topmost point, with pressure varying from p_o radially to $4 p_o$ tangentially.

With reference to the case of $n = p_o/\gamma R$, consider the pressure distribution in the side regions at the level of the core center. If the array were glued so that it were possible to transmit tension throughout the lower half of the core at $n = p_o/\gamma R$, there would just begin to develop at the sides a situation like what is shown at the bottom of Fig. IV-4. In the absence of the hypothetical tensile stresses, a small amount of wedging is thought to occur at the sides as well as at the bottom.

In Ref. 48, pages 29 and 30, it is shown that, for a large ratio of k_t to k_r and for $n = p_o/\gamma R$, an equivalent friction factor of unity is required to prevent shear displacements. These shear displacements would occur first at the extreme right and the extreme left, as shown in Fig. IV-5. In Ref. 48, p. 11, Eq. 1, there appears an expression relating the equivalent friction factor, f_{hex} , to the conventional surface friction factor, f . For $f_{hex} = 1$,

$$f = \frac{1 - \frac{1}{\sqrt{3} f_{hex}}}{\frac{1}{\sqrt{3}} + \frac{1}{f_{hex}}} = \frac{1 - \frac{1}{\sqrt{3}}}{\frac{1}{\sqrt{3}} + 1}, \text{ or } f = 0.268.$$

Friction measurements have indicated a coefficient of friction, f , of 0.3 or greater between tubes. Therefore, the criterion $p_o = n\gamma R$, should prevent shear separation.

The appropriate value of γR derives from

$$\gamma = 0.0887 \text{ lb/in}^3 \text{ (axial average)}$$

$$\text{and } R_c = 25.594 \text{ in.}$$

Therefore, $\gamma R = 2.270 \text{ psi}$.

Using the criterion of $p_o = n\gamma R$, and the specification that $n = 4$, we have:

$$\begin{aligned} p_o &\geq 4 \times 2.270 \\ &\geq 9.08 \text{ psi (axial average)} \end{aligned}$$

According to Ref. 50, p. 32, the reactor "as built" is estimated to have a p_o of $10 \pm 2 \text{ psi}$ at room temperature. Therefore, the clamping pressure in

the reactor provides an acceptable margin of safety against dislocations within the array of ceramic tubes for a 4-g acceleration. At the design temperature for core and spring material, the expansions are such that (coincidentally) the same p_o results.

Below are described the results of an experimental test of matrix stability. One conclusion drawn is that $p_o > 1.4 n\gamma R$ is sufficient to preclude discernible dislocation even in a tube matrix whose lateral support is essentially tangential. This is a lesser requirement than the theoretical value of $2 n\gamma R$ for complete suppression of all dislocations. The difference between the theoretical factor of 2 and the experimental factor of 1.4 lies in the word "discernible." At any rate, one conclusion that may be drawn is that requiring $p_o > 2n\gamma R$ is overly conservative.

Another result of the experiment described below is that dislocations become undesirably large if p_o falls below $0.67 n\gamma R$.

It appears then from the experiment that the region of adequate core pressurization is given by

$$1.4 n\gamma R > p_o > 0.67 n\gamma R.$$

Therefore, the choice of $p_o \geq 1.0 n\gamma R$ is corroborated as one which is not excessively conservative, but which is expected to provide a reasonable margin of safety.

Actually, it is desirable to obtain additional experimental information on the mode of failure of the tube matrix when the design inertia load is exceeded sufficiently to cause a clearly discernible failure.

Two such experiments were being designed and planned before the Pluto program was terminated. One was a drop test; the other was a centrifuge test. One of the plans called for high-speed photography of the failure sequence.

2. Experimental

A matrix model was tested to check the theory of core instability⁴⁸ and to determine minimum side support pressures required to suppress all dislocations. For a highly tangential system, this pressure was found to be:

$$p_o = 1.4 n\gamma R.$$

As indicated earlier, instability may be evidenced by (1) separation of core from side support, (2) separation between individual tubes, or (3) translational slipping due to shear.

Deflection of a rigid body is expressed as:

$$\delta = \frac{n p_{ch}}{k_t (1/\eta + 1)} .$$

Criteria for stability are given by:

$$\Psi > C \frac{2\eta - 1}{\eta + 1} , \text{ for tube separation;}$$

$$\Psi > \frac{1}{\eta + 1} , \text{ for side support-core separation; and}$$

$$\Psi > \sqrt{\frac{(\eta/f_{hex})^2 + 1}{\eta + 1}} , \text{ for translational slipping:}$$

where

k_r = Radial spring constant	lb/in ³ (psi/in.)
k_t = Tangential spring constant	lb/in ³ (psi/in.)
R = Matrix radius	in.
γ = Density of matrix	lb/in ³
n = Inertial load in gravities	dimensionless
p_o = Initial preload	lb/in ²
$\eta = k_t/k_r$ = Suspension ratio	dimensionless
p_{ch} = Characteristic pressure = γR	lb/in ²
$\Psi = \frac{p_o}{n p_{ch}}$ = Pressure ratio	dimensionless
δ = Static deflection	in.
C = Constant	dimensionless
f_{hex} = Equivalent friction coefficient	dimensionless

The dimensionless quantities, Ψ and η , may be used independent of the size of the model or the actual core. The model chosen was a 6-inch radius core containing 5-inch long steatite tubes of flat dimension equivalent to the actual tubes. Twelve aluminum shims made the transition from the rough hexagonal exterior of the core to the side support. The side support was modeled by helical compression springs for the radial support, and by cantilever sheet springs for the tangential support.

The model in the horizontal position was pressurized by a given set of radial springs. The tangential springs were set at a given rate by varying the free length. The model was then turned vertical. Deflection measurements were taken and visual inspection of the tube array was made. This procedure was repeated for many sets of Ψ and η .

In examining the matrix model, it was possible to see and feel several types of organized tube looseness or dislocations. While each type of dislocation contains some of the characteristics of other types, the differences are sufficiently distinct to permit the naming of five different stereotypes. In build-ups with a large value of η , that is with largely tangential support, Type 1, 2, or 3 dislocations (as defined below) were observed, depending on the pressure ratio, Ψ . With mainly radial support, Type 5 (and occasionally Type 4) was observed.

The first three types are various stages of tension cracking. The fourth combines tension cracking with slip planes, and the fifth combines top separation with slip planes. These stages of dislocation may be defined more precisely as:

- Type 1 One or two radial cracks at the bottom with about 1/16 inch separation. The cracks extend only a few tubes into the matrix.
- Type 2 Several radial cracks extending about 3 inches into matrix. General looseness at bottom of matrix.
- Type 3 Radial cracks, general looseness, some wedging.
- Type 4 Radial cracks, wedging, slip planes, overall looseness.
- Type 5 Top separation, loose at top, slip planes.

Table IV-1 gives the types of instability observed for nineteen sets of test parameters. The exact transition from one type of instability to another is difficult to detect; the blending of phenomena takes place over a wide range of Ψ . However, at high values of η the transition from a stable matrix to Type 4 instability can be seen. Figures IV-7, IV-8, IV-9, and IV-10 show this transition at $\eta = 35.5$. Type 5 instability is shown in Fig. IV-11. Figure IV-12 is a closeup of the tube pattern which illustrates the combination of a slip plane with a wedging action. In studying these figures, it is necessary to examine the intertube boundaries in close detail.

Figure IV-13 is a plot of η vs Ψ . On it are drawn the three stability fractions and the test points. At high η , the tension crack stability line is given experimentally by

Table IV-1. Matrix model core stability test.

η	Ψ	Observed condition of matrix
35.5	1.40	Stable
35.5	1.22	Type 1 instability
35.5	1.05	Type 2 instability
35.5	0.77	Type 3 instability
15.0	0.70	Type 3 instability
7.68	0.63	Type 3 instability
4.44	0.56	Type 3 instability
1.88	0.35	Type 4 instability
14.2	0.70	Type 3 instability
14.2	0.52	Type 4 instability
6.0	0.26	Type 4 instability
0.417	0.47	Type 5 instability
0.125	0.47	Type 5 instability
3.67	5.30	Stable
3.67	1.00	Stable
1.48	10.5	Stable
1.48	0	Types 4 and 5 instability
0	10.5	Stable
0.125	0	Type 5 instability

$$\Psi = 0.7 \frac{2\eta - 1}{\eta + 1} .$$

As $\eta \rightarrow \infty$, $\Psi \rightarrow 1.4$: and for gross stability,

$$\Psi \rightarrow 0.67.$$

Thus, for the reactor with

$$\gamma R = 2.27 \text{ psi}$$

and

$$n = 4 \text{ g},$$

we require

$$\begin{aligned} p_O &= 1.4 \times (4 \times 2.27) \\ &= 12.7 \text{ psi, for tension crack stability} \end{aligned}$$

or

$$\begin{aligned} p_o &= 0.67 \times (4 \times 2.27) \\ &= 6.1 \text{ psi, for gross stability.} \end{aligned}$$

Early in the design stage, a slice test assembly was constructed. It consisted of a 10-inch-thick full size matrix of steatite tubes surrounded by a side support system. This assembly is shown in Fig. IV-14. The side support system used was an early concept and consisted of "buggy" springs on pressure pads. This system applied a pure radial force to the core, as opposed to the actual design system used, which was a highly tangential force system (i. e., η large).

Tube pull tests, vibration tests, and gravity shock tests were performed on this assembly. The shock tests proved to be the most useful in determining core/side-support dynamics. The test consisted of dropping the assembly from heights of 8 to 16 inches on cardboard honeycomb sized and shaped to produce a 2- to 4-g triangular shock pulse. Strain gages on the buggy springs and accelerometers on the core were displayed on a trigger oscilloscope.

Plans for complete testing of the final design side-support/core system, using actual BeO for all tubes, were being formulated at the termination of the Pluto program. An ideal 10-hour proof test would consist of the assembly located on a variable speed centrifuge and heated to design temperature. Again, dynamics would have been provided by drop tests.

C. Design Considerations

1. General

When a spring is expanded elastically at assembly, and then stretched further but still elastically as core temperature rises, strain energy is stored in the material of the spring. During reactor start-up, a combination of thermal expansion of the spring and reduction in Young's Modulus of the spring material may actually result in a temporary reduction of strain energy and of spring tension.

It was initially decided that for the side support design elastic behavior is required. Plastic girdles have been proposed for advanced applications,^{51, 52} and interesting experiments pursuant thereto have been performed.

In the final design, the elastic strain energy is mainly in flat plate bending. About one-half of the material is subjected to the maximum bending moment; the other half to a cantilever beam type of bending moment distribution. The fact that a large proportion of the material is highly stressed means that the material is efficiently utilized.

Consider the question of the desired flexibility of the spring. Suppose several alternate designs are compared, all of which produce the same core pressure at full operating temperature. This is taken as a common design point for comparison because the greatest need for core pressurization occurs when the core is most expanded.

Referring to Fig. IV-15, we see that the energy stored in the spring and the core pressurization at assembly (in a low temperature environment) are functions of spring rate. Curves (a) represent a flexible spring, and curves (c) a stiff spring. The mutual intersection of all curves at the assembly condition is a characteristic of this design, and should not be concluded to be a general law. The area under each dashed line is proportional to the strain energy in that spring at assembly. The relationship of strain energy to area is more complex at other conditions because of the multiple effects of temperature.

From the standpoint of minimizing total strain energy, one might be tempted to select a very stiff spring. However, there are two considerations which direct one toward a flexible spring. First, it is desirable to maintain a reasonable core pressure and reasonable spring stress at all operating and storage conditions. Second, a flexible spring generally involves a larger quantity of material than a stiff spring, and may therefore be superior in terms of strain energy per unit volume. For a given allowable stress, this may be the deciding factor. Flexibility costs weight and space for a given maximum load, but the amount of material involved may be necessary, to limit stresses.

The discussion immediately above assumes a fixed value of efficiency of utilization. In addition, the criterion of minimum space has not yet been discussed. It is clear that the space occupied by the side support system is an important parameter in overall system design. A small space requirement is desirable to minimize the frontal area of the vehicle. On the other hand, if, because of payload size or other reasons, the vehicle drag should

be relatively insensitive to reactor diameter, the active core should be made as large as possible to enhance performance. In designing a reactor for a particular missile system, the optimization of the side support system must include the total effect of the side support radius and airflow requirements on the overall thrust-drag ratio. A considerable amount of work has been done in the whole area of spring optimization.⁵³ Also of interest in future designs is work reported on in Ref. 54.

Design considerations which limit the choice of configuration include:

Expansion and contraction of the core and of the duct must be accommodated without overstressing the ceramic tubes or other core components, the duct, or the intervening structure. The core may expand as much as 0.36 inch on the radius compared to the room temperature condition. The duct may expand as much as 0.3 inch on the radius, but sometimes out of phase with the core because of a longer time constant.

Relative axial motions must be permitted as well as the radial.

The arrangement of material must be such as to permit adequate flow of cooling air, and the airflow pattern must be conducive to heat transfer to minimize flow rate.

The structure must not be overly sensitive to dimensional tolerances.

The support must limit the transverse motion of the core as a whole because misalignment or bending of control rods and tie rods must be minimized. In addition, resonance between the free bending modes of the vehicle and the rigid body mode of the core on its support should be avoided, even though friction in the system is expected to be adequate for vibration control.

Weight should be minimized. As a corollary, if possible, any material introduced should have neutronic worth.

Frontal area should be minimized. However, the effect on overall system drag is a complex function of reactor area.

Fabrication ease and cost should be optimized.

2. Transverse Load

Based on the foregoing general considerations, more specific design criteria can be set forth. The core and side support system must be capable of a 4-g transverse load. This is documented in Ref. 55, p. 4.

The 4-g load is the constant, steady-state load considered equivalent to the summation of flight-type loads. It includes a small amount of magnification due to the dynamic response of the core as a unit sprung mass, and a 1-g dead load due to gravity. Flight loads are made up of several components, as described in Refs. 56 and 57. These components are:

<u>Component</u>	<u>Approximate period</u>	<u>Approximate frequency</u>
1. Terrain avoidance	6 seconds	---
2. Vehicle bending due to gusts	---	9 cps (first mode) 20 cps (second mode)
3. Vehicle bending due to store ejection	---	9 cps (first mode)

In the corrugated spring, the stress component to be evaluated in terms of endurance stress is that due to the bending moment created by the rail reaction. In Fig. IV-16, a Soderberg curve is constructed. For conservatism, the endurance limit is taken as one-half the 10-hour creep-to-rupture limit. The stresses of 65,000 psi and 7050 psi are taken from Section D. 3; Spring Stress. The stress of 7050 psi is obtained from the 1610 psi value for $\sigma_{B, rail}$ at 1 g. It is assumed that the alternating stress is due to a 3-g load reversal; i. e., 4 g's minus the 1 g of gravity. Then a stress concentration factor of 1.46 is applied. That is, $7050 = 1610 \times 3 \times 1.46$.

The construction of Fig. IV-16 indicates a margin of safety of at least 31%, based on alternating stresses versus endurance limit.

3. Eccentricity and Misalignment

It is necessary to limit the gross movement of the core relative to the duct to prevent excessive strain or jamming of tie rods and control rods. A limit of 0.1 inch was established as the permissible movement of the core centerline with respect to the duct centerline. This could be a pure translation or a combination of rocking and translation. In any case, the maximum relative motion at any axial position should not exceed 0.1 inch.

As built, the reactor exhibits a 1-g static deflection of 0.027 inch as described in Ref. 50, p. 35. Most of this is due to clearance between rails and spring. Additional g-loads will produce deflection at the rate of about 0.0015 inch per g, for a 4-g total of 0.032 inch.

4. Avoidance of Resonance

Gust loads and store ejection give rise to missile vibration. As a result there are vibratory inputs to the reactor duct at approximately 9 cps and 20 cps. While there is considerable damping in the system comprising the core, the side support, and the duct, it is still desirable to avoid designing a resonance into the system. In particular, it is desirable that the rigid body mode of the core on its side support be above 20 cps.

The resistance of the side support system is nonlinear. Because of clearance between the rails and the spring convolutions, and because of the use of slightly undersized side reflector tubes (to prevent arching), the effective side support spring rate is very low for about the first 0.025 inch. Calculations show that there is appreciable stiffening as transverse load is increased. The effective spring rate at high loads has been estimated as being equivalent to 0.0015 inch per g; therefore, the rigid body natural frequency for small amplitude vibration superimposed upon a large steady load is approximately 80 cps.

To summarize, the rigid body natural frequency of the core upon its support can vary between zero and 80 cps, depending upon the steady state load and the amplitude of input vibration. The zero figure is derived from the theoretical possibility of oscillating within the rail-spring clearance during a zero g maneuver. It is possible, then, for the system to resonate with the input due to vehicle body bending modes. However, because of the nonlinearity of the system, a buildup of amplitude through resonance is self-limiting because the effective natural frequency moves up out of the range of input frequency as amplitude increases, but still within a permissible value. The presence of friction helps greatly to control the response.

5. Interference and Separation Between Rails and Spring

To accommodate relative radial expansion between core and duct, the system employs a sliding relationship between the duct rails and the radially disposed sides of the spring convolutions. It is important that the rails and spring do not disengage during a transient resulting in a cold core and a hot duct.

Also, it is important that the duct rails do not bottom out against the spring. (This would occur before the spring could strike the inner surface of the duct wall.)

To prevent either interference or separation, it is necessary for the combination of duct temperature and core temperature to lie within a certain envelope as shown in Fig. IV-17. The desired operating conditions actually do lie within this envelope.

D. Side Support Springs

1. Fabrication

The corrugated springs were fabricated in 60° arcs from Rene' 41 sheet stock that had been rolled and ground to a thickness of 0.150 ± 0.001 inch. There were three main forming operations. Using male and female dies, sheet strips cut from the stock were first-formed to a rounded contour, and then coined to produce flatter flats and sharper corners. At this point the pitch line of the corrugations was straight. In a later operation the springs were shaped to the circumference of a circle at the spring free radius. The step-by-step fabrication process is given below.

- 1) The annealed stock was Rene' 41 sheet, 36 X 96 inches, rolled and then ground to proper thickness of 0.150 ± 0.001 inch.
- 2) Strips of the proper width were square-sheared from the sheet stock and edge-ground and buffed.
- 3) The strips were first-formed by means of a male and female die. At this point there were no sharp corners. The appearance was as shown in Fig. IV-18a.
- 4) The first-formed strips were solution-annealed, followed by a water quench.
- 5) The annealed strips were coined to produce flat sides and proper corners, as in Fig. IV-18b.
- 6) The parts were Zyglo inspected.
- 7) Some of the convolutions were over-formed in the coining operation; so a straightening operation was performed using a wedge, as in Fig. IV-18c.
- 8) The ends were trimmed to size.
- 9) Another Zyglo inspection was made.
- 10) Each spring was shaped to a curved center-line (Fig. IV-18d).
- 11) The springs were annealed and final heat-treated.

- 12) The springs were ground to exact width.
- 13) The connecting dovetails (Fig. IV-2) at the ends were formed by the Elox process.
- 14) The dovetails were inspected.
- 15) The springs were deburred and polished. External corners were rounded to a 0.030-inch radius.
- 16) A final inspection was made, including Zyglo.

2. Core pressure

To compute the pressure applied to the core by the corrugated springs, there are several possible approaches. The method used below depends upon an experimental measurement of the circumferential spring rate of the corrugated spring. Referring to Fig. IV-3, we have

$$T = p_o R_c, \quad (1)$$

where T is the circumferential pull of the spring in pounds per axial (parallel to reactor centerline) inch, and p_o and R_c are as defined previously. In deriving Eq. 1, it is assumed that the shims are not in circumferential contact with one another.

To examine how p_o changes with R_s , replace R_c in Eq. 1 with $R_s - Z$ (defined in Fig. IV-3), and solve for p_o :

$$p_o = \frac{T}{R_s - Z} \quad (2)$$

If R_c and Z are changed as a result of a change in core temperature but the temperature of the spring does not change, the pressure changes with the radius R_s as follows:

$$\frac{dp_o}{dR_s} = \frac{dT/dR_s}{R_s - Z} - \frac{T}{(R_s - Z)^2} \left(1 - \frac{dZ}{dR_s} \right) \quad (3)$$

or

$$\frac{dp_o}{dR_s} = \frac{1}{R_c} \frac{dT}{dR_s} - \frac{T}{R_c^2} + \frac{T}{R_c^2} \frac{dZ}{dR_s} \quad (4)$$

It can be shown that for values of p_o and R_s of interest, the last two terms in Eq. 4 are small compared to the first term, and may safely be neglected. If further use of Eq. 4 is to be made, it is necessary to be sure of the proper interpretation of dR in each term; for example, the effects of spring temperature must be considered. Equation 4 is therefore simplified to:

$$\frac{dp_o}{dR_s} = \frac{1}{R_c} \frac{dT}{dR_s} \quad (5)$$

Multiply the denominator by R_s/R_s . Then

$$\frac{dp_o}{dR_s} = \frac{1}{R_c R_s} \frac{dT}{(dR_s)/R_s} \quad (6)$$

The quantity, $\frac{dT}{(dR_s)/R_s}$, may be interpreted as the spring rate of the corrugated spring in the circumferential direction (not the tangential spring constant, k_t , referred to previously); since it may be written as $\frac{dT}{d \text{ arc}/\text{arc}}$. This rate is in units of pounds of pull force per axial inch per inch per inch of circumferential elongation or $\frac{\text{lb}/\text{in.}}{\text{in.}/\text{in.}}$.

Reference 58 reports the results of pull tests on several spring specimens consisting of five convolutions each, with straight rather than curved centerlines. The reported results include the value of $\frac{dT}{dL/L}$ at various temperatures. Using an expression like Eq. 6, the radial spring rate, dp_o/dR_s , has been computed for various spring temperatures and plotted in Ref. 2, p. III-54. In computing the curve, information regarding the variation with temperature of Young's Modulus was included. With this curve the mechanical core pressure can be computed if the spring temperature and the difference between R_s and the free radius of the spring at that temperature are known.

In Fig. IV-19 are shown the radii as functions of axial location along the core for the reactor structure limit condition. Figure IV-20 shows mechanical core pressure as a function of axial location for four cases. These cases are (1) the reactor structure limit condition (2) the condition during startup when spring and core are at a uniform temperature of 1200° F, (3) the condition at assembly at room temperature, and (4) the situation when stored at a uniform temperature of -65° F.

The following example illustrates the method of calculating a point at $x/L = 0.7$ on the curve for reactor structure limit condition.

From the known temperatures at this location for this condition and the appropriate thermal expansion coefficients it is determined that:

$$R_s = R_c + Z = 27.448$$

The average spring temperature at $x/L = 0.7$ is known to be 1300°F , and the mean coefficient of thermal expansion at that temperature is $8.4 \times 10^{-6}/^\circ\text{F}$. Since the free radius of the spring at 70°F is 26.728, the free radius at 1300°F can be calculated:

$$\begin{aligned}\text{Free radius at } 1300^\circ\text{F} &= 26.728 [1 + 8.40 \times 10^{-6} (1300-70)] \\ &= 27.004\end{aligned}$$

Then

$$p_0 = k_R (R_s - \text{free radius at } 1300^\circ\text{F})$$

where

$$k_r = dp_0/dR_s, \text{ as explained above.}$$

From Ref. 2, p. III-54,

$$k_r = 28.7 \text{ psi/in.}$$

Then

$$\begin{aligned}p_0 &= 28.7 (27.448 - 27.004) \\ &= 12.7 \text{ psi.}\end{aligned}$$

As shown in Fig. IV-20, p_0 averaged over the full length of the reflected core for the structure limit condition is 11.4 psi.

The values of p_0 and spring stress at the structure limit condition are considered critical for three reasons:

- 1) The inertial loads that must be resisted by the core as a clamped bundle of tubes are greatest at the structure limit condition.
- 2) The core pressure, p_0 , is less at the structure limit condition than at other operating conditions with comparable inertia loads.
- 3) As spring temperature increases, the material strength (10-hour creep-to-rupture) decreases at least as fast as the applied stress. This argument disposes of transient conditions as well as other steady state conditions.

Another example illustrates the effect of spring temperature on spring stress and p_0 . For the cold inlet condition, the spring is colder than in the structure limit condition, while the fueled tubes are just as hot. As shown

below, p_o will be greater, as will the stresses in the spring. However, it can be shown that the strength of the spring material increases faster than the applied stress as spring temperature decreases. Free radius of the spring at $70^\circ\text{F} = 26.728$.

As before, at $x/L = 0.7$, $R_s = R_c + R_z = 27.448$, and the average spring temperature is 900°F and $\alpha = 7.4 \times 10^{-6} (^\circ\text{F}^{-1})$.

$$\begin{aligned}\text{Free radius at } 900^\circ\text{F} &= 26.728 [1 + 7.4 \times 10^{-6} (900-70)] \\ &= 26.892\end{aligned}$$

and from Ref. 2 at 900°F ,

$$k_r = 31.5 \text{ psi/in.}$$

Then

$$\begin{aligned}p_o &= 31.5 (27.448 - 26.892) \\ &= 17.5 \text{ psi}\end{aligned}$$

The ratio of this p_o to the structure limit condition p_o is $17.5/12.7 = 1.38$. Thus the spring stress is 38% higher, but material strength at 900°F is more than 38% higher.

The minimum core pressure, p_o , occurs at a uniform reactor temperature of 1200°F . For uniform temperature, p_o decreases with rising temperature because the spring expands faster than the core and, in addition, its Young's Modulus decreases. Above 1200°F spring temperature, it is expected that the core will be hotter than the spring. Using the same method as above to calculate p_o , a value of 7.9 psi is obtained for the reactor uniformly at 1200°F . This pressure is acceptable, since the uniform temperature represents (at worst) a transient condition encountered in ground tests in the absence of large maneuver loads.

3. Spring Stress

Using the value of p_o at $x/L = 0.7$ for the structure limit condition, the spring stress can be computed. The maximum bending moment in the spring results from

$$M_{\max} = Th \tag{7}$$

but

$$\sigma_b = \frac{M_{\max}}{I/c}$$

Substitute $p_o R_c$ for T from Eq. 1 and $t^2/6$ for I/c . (This is all on the basis of one axial inch.) Then, neglecting stress concentrations:

$$\sigma_b = \frac{p_o R_c h}{t^2/6} \quad (8)$$
$$\sigma_b = 6 \times 12.7 \frac{25.594 \times 0.5}{0.15^2} = 43,300 \text{ psi}$$

Thermal stress and bending stress due to the interaction among shim, spring, and rail in taking transverse loads must be added to the above result. These two additional stresses are reported in Ref. 59, pages 9-11. Thermal stress is estimated as $\sigma_{\text{thermal}} = 1250 \text{ psi}$.

Bending stress due to rail load, assuming that the rails share the load according to a sinusoidal distribution, is estimated as:

$$\sigma_{B, \text{ rail}} = 6450 \text{ psi for a 4-g inertia load factor}$$

$$\sigma_{B, \text{ rail}} = 1610 \text{ psi for a 1-g inertia load factor.}$$

As pointed out in the reference, $\sigma_{B, \text{ rail}}$ is opposed to the major bending stress due to spring stretch. Thus, neglecting σ_B , one is conservative except for fatigue damage effects which, as shown earlier (Fig. IV-16), may be tolerated.

Adding the thermal stress to the major bending stress, we have:

$$\sigma = 43,300 + 1,250 = 44,550 \text{ psi.}$$

This is the extreme fiber stress in the circumferential portions of the corrugated spring. Stresses in the radial legs are less, with zero stress near the half-way point. In the reentrant corners a stress concentration factor is applied. Photoelastic analysis of the spring shape gives a stress concentration factor of 1.46, which results in the highest stress in the corners of the spring:

$$\sigma_{\text{corner}} = 44,550 \times 1.46 = 65,000 \text{ psi.}$$

In a ductile material such a localized stress could relieve itself by creep; however, at 1250° F, Rene' 41 is partly into the hot-short region.

The phenomenon of creep will cause the corrugated springs to partially relax their tension, and thereby partially relax the core bundling pressure, p_o , over a period of time. Partly because of the complexity of

the stress distribution in the spring and partly because of the scarcity of creep data for Rene' 41 at the time of design and manufacture of the parts, it was decided to rely on the results of a load relaxation test performed with test parts closely simulating the actual spring. The test⁶⁰ consisted of holding the test piece at a constant deformation at a given temperature for 10 hours at 1300° F, and recording the load as a function of time. The test part contained spring convolutions of the same shape as those of the real spring. It was shown that the fixtures contributed a virtually negligible amount to the overall relaxation.

The test results indicated the following:

Core bundling pressure corresponding to the initial load in the test part	Relaxation in 10 hours
10.1 psi	7.0%
12.8 psi	5.5%
15.2 psi	5.3%

At the higher loads the absolute amount of creep was greater than at the lower loads, but the percent relaxation was less.

While the effect of creep in the spring is generally to reduce the load in the spring, it is necessary to examine areas of localized high stress that could conceivably creep to rupture before the general level of load is significantly reduced. The most highly stressed region is the corner. At 1300° F, Rene' 41 10-hour creep rupture strength is approximately 90,000 psi.² The margin of safety is then:

$$\text{m. s.} = \frac{90,000 - 65,000}{65,000} = 38.5\%$$

The extent of radiation damage was not fully evaluated. Its effect over a 10-hour flight mission would be to increase the creep rate and the load relaxation rate. (Ref. 61, p. 27.) Additional material studies were required to evaluate the consequences on the corrugated spring design.

4. Spring Dimensional Tolerances

There is a tolerance of ± 0.020 inch on the free radius of the spring. Since the value of k_r is usually about 30 psi/inch, p_o may vary from the nominal by something like 0.6 psi, as a result of dimensional tolerances

in the spring. Small variations of spring temperature from those calculated have a comparable effect.

E. Rails and Wiper Bars

To transmit the reactor gravity or inertial loads to the duct, 30 support rails are bolted to the inside of the duct. The rails are located every 12 degrees, as indicated in Fig. IV-1. Figures VI-5 and VI-34 show the installation. Seventeen separate pieces, each with a central stud, are bolted to the duct end to end, to form one rail. The nut is torqued to give a known preload in the stud, to prevent the rail from loosening because of temperature differences between the rail and the duct.

The maximum design load transmitted by the rails is a 4-g maneuver load. This causes a maximum load of 180 pounds per rail, resulting in a possible 49,000 psi stress in the rail stud if all preload is lost. The maximum temperature anticipated is 1350° F, occurring in the stud portion. A test with rail loads cycled from 50 to 200 pounds for 500,000 cycles at 1300° F resulted in no deterioration of the rail.

The preload bending stress in the arms is 63,000 psi. The maximum temperature expected for the structure limit case is 1300° F.

Between each pair of rails are three wiper bars, as shown in Fig. IV-1. Figures VI-6 and VI-7 show the wiper bar assembly and installation. The bars support the wipers against the reactor pressure drop load over the area of the bar and wipers for the wing wiper bar. The load is 131 pounds. This produces a shear stress in the tie-down bolt of 1230 psi. The highest stress in the bar results from bending of the bars radially. No mechanism to cause this bending is postulated, but if the center section were bent inward 0.3 inch while the ends were held to the duct, a stress of 31,000 psi in the bar would result.

A high-temperature (1400° F) blowdown of a full-length wing wiper bar with shroud resulted in no malfunction of any parts. Vibration tests of a wing wiper bar showed no adverse resonances when vibrated at frequencies typical of operation.

F. Wiper Springs

1. General

Due to the different thermal capacitances of the core and the duct, the annulus dimension between these components varies during transient reactor operation. Provision must be made for a cold-duct/hot-core condition. If the additional space that opens up as the duct subsequently expands is not filled up, the flow rate through this region would be three times the desired rate. Tripling the amount of side support coolant, however, would severely decrease the thrust from the reactor.

The problem consists of finding a flow-blocking device that will operate throughout the regions shown in Fig. IV-21. The obvious solution seemed to be a rollover-type spring, because of the large expansion differences that had to be accommodated. The radius of curvature of a beam under load is linearly related to the applied moment and hence to the stress in the beam. Thus, as the spring shown in Fig. IV-22 is compressed, it begins to flatten out, and the stresses in the flattened portion cannot exceed a predetermined maximum stress.

There are two types of stress to consider in these springs. The first is due to the compression resulting from variations in annulus spacing. The second is due to reaction against the induced pressure drop loads. These loadings result in requirements which are contradictory.

In a stiff spring the pressure drop stresses would be slight, but the stresses due to annulus compression would be high. In a soft spring the opposite would result. There is, then, an optimum stiffness resulting in a minimum stress. Normally one would select an intermediate position where the stresses due to each condition were equal. There was, however, one further design condition.

Deflection-induced stresses such as those resulting from core-duct expansion differences tend to be self-relieving. If a certain stress is exceeded, they tend to yield until the stress is reduced to a point at which yielding no longer occurs. On the other hand, stresses due to supported

loads are due to a force balance and will not generally disappear with yielding, until the structure has collapsed to a more stable configuration. It was decided to take advantage of this built-in safety mechanism by designing the springs so that lower stresses would result from the pressure drop than from the deflection. This would also permit greater manufacturing tolerance in the radius of curvature, the principal means of stress control.

2. Design Conditions

The spring should be capable of being pressed flat from an initial height of 3/4 inch and should normally operate at half this height. Rene' 41 was selected as the spring material because of its high strength at a temperature of about 1300° F.

The maximum allowable stress was established at 90,000 psi. Further, the design was to be such that the pressure drop-induced stresses were to be approximately half the deflection-induced stresses. The pressure drops were to be such that the annulus pressure would everywhere be equal to or greater than the static pressure in the fueled tubes at a corresponding axial position; so that the annulus pressure everywhere would augment the pressure provided by the corrugated springs. Further, due to the lower temperature in the forward reflector, the corrugated spring pressure in this region is only about half that in the other portions of the reactor. The annulus pressure should be kept somewhat higher here to augment the spring pressure even further.

3. Spring Configuration

Three spring configurations were considered. The first is shown in Fig. IV-23a. Because no additional holes were desired in the duct, it was necessary to support the wipers by an unsupported wiper bar sufficiently flexible to follow the barreling of the duct. This configuration resulted in a net radially inward thrust across the spring which pulled the flexible wiper bar away from the duct and permitted large quantities of air to sneak through between the bar and the duct.

The second configuration used the same spring reversed in direction. In this case, any spring soft enough to stand the annulus compression is too soft to support the required pressure drop. The spring simply rolls away from the flow.

The axial profile of the spring finally selected is shown in Fig. IV-23b. It is formed of two arcs of the same radius, the asymmetry arising from the requirement of tangency at the point of attachment. The free unloaded height of the spring is 0.750 inches. This is more than the normal annulus spacing, so the spring is always under some preload. The stress distribution along the spring length as a function of annulus spacing is shown in Fig. IV-24.

During the initial stages of compression the shape of the stress curve is as shown by ABC. This continues to shift until the distribution is that shown by ADE. At this point the stress at D is such that the spring begins to roll over, flattening against the annulus walls. The value of this stress is determined by the spring thickness and curvature. Subsequent compression provides a stress curve such as AFGHE until the limit AIJKE is reached, at which time the spring is almost flat along its entire length. There are, in addition, stresses resulting from the pressure drop across the spring. In general, the distribution of this component is as shown in Fig. IV-25.

The stresses resulting from the pressure drop load and deflection were approximated by suitable superposition of formulas 22, 23, and 24 in Table III of Ref. 41. The results are not given here for the simple reason that they are not particularly good. This required a trial and error solution, since the springs tended to flatten out, redistributing the loads and stresses. The trial and error solutions were carried out to a point where the calculational uncertainties matched those due to the manufacturing tolerances already described.

4. Experimental Analyses

Eight temperature-compensating strain gages were applied to a spring steel wiper. Four of these were placed at 0.625-inch intervals, with the first immediately adjacent to the fastening screw. The other four were likewise placed on 0.625-inch centers with the first one 0.625-inch from the free end. These gages were numbered 1 to 8 in sequence, the gage at the screw being number 1 and that nearest the free end number 8. Table IV-2 gives the strain data obtained. (Gage number 6 was damaged in handling.)

In the first test the wiper was placed in a small duct and compressed to 0.406 inch. The gages were set at zero for this position, and low

Table IV-2. Wiper stress test data.

Gage No.	1	2	3	4	5	7	8
Press. drop psi	Flow-induced strains (in./in.)						
0	0	0	0	0	0	0	0
2.43	-21	-81	+84	+87	+34	-91	-182
3.92	-119	-103	+174	+172	+58	-149	-299
5.60	-251	-73	+299	+240	+55	-230	-422
7.49	-417	-10	+459	+287	+40	-373	-623
9.12	-533	+1	+577	+365	+16	-465	-760
10.37	-660	+50	+713	+415	-8	-577	-918
11.86	-833	+145	+907	+491	-58	-722	-926
0	-39	+59	+22	+83	-32	-53	-61
Gap (in.)	Displacement strains (in./in.)						
0.710	0	0	0	0	0	0	0
0.629	+464	+225	-83	282	-457	-155	-75
0.596	+663	+340	-77	-350	-650	-341	-174
0.568	+785	+448	-40	-381	-784	-541	-291
0.493	+1179	+714	-23	-551	-1183	-955	-519
0.450	+1391	+993	-13	-661	-1469	-1237	-637
0.406	+1459	+1311	+57	-769	-1743	-1547	-787
0.344	+1477	+1507	+141	-639	-1769	-2163	-1061
0.300	+1515	+2067	+241	-965	-2257	-2635	-1267

Notes: Gages were set at zero in position for the flow test. No gage was broken before the test.
 Gages were C6-121-120.
 Gage factor: $2.03 + 1/2\%$.
 Gages were applied with Eastman 910 and coated with DC140 Solastic.

conductivity water (LCW) was passed through the duct to provide selected pressure drops up to 11.88 psi. The latter was measured by a mercury-water manometer. The volume of water required and the characteristics of the plumbing were such that the maximum drop obtainable with the LCW system was 11.88 psi at the LCW pumping station. Plastic windows permitted observation of the spring during this test. The results were quite consistent with the deflections observed.

The spring was then removed from the fixture and compressed in stages between two plates. This gave the second set of data on Table IV-2.

The stresses corresponding to some of the deflections have been plotted in Fig. IV-26. The stresses due to some of the pressure drops have been plotted in Fig. IV-27. In Fig. IV-28 the stresses due to the 11.88 psi pressure drop and the 0.344-inch gap deflection have been plotted separately and together. This corresponds pretty closely to the maximum stress case expected. A higher pressure drop (about 16 psi) is expected across the last wiper in line, but this wiper is purposely doubled, so the stresses will be lower there.

For the purpose of clarity curves have been drawn through associated points, but some of the details could easily be argued with. In general, the results plotted in these figures agree with those calculated. The strongest deviations occur in Fig. IV-26, and it is almost certain that these result from deviations from design curvature in the specimen tested. It will be seen that there is only one point of troublesome stress about 1-1/2 inches from the free end. In this particular spring the stress reaches 95,000 psi. Since 70,000 psi is relievable (i. e., deflection-caused), a slight amount of yielding will occur here. However, this is quite near the free end, where it will have a negligible effect on position or performance. The important components due to the pressure drop nowhere exceed 30,000 psi, so there is an adequate margin of safety.

While the stress characteristics were somewhat amenable to calculation, the pressure drop characteristics were not. Consequently, a full-length model representing one repeatable section of the side support system was built. Air flow tests of this model were then used to determine the wiper widths.

The cross-section of the flow passage tested is shown in Fig. IV-29. Section C represents the rail or wiper bar. There are basically two flow

passages: A, representing that above the rail or wiper bar; and B, representing that adjacent to the duct. There are actually two A-type passages to contend with since the rail is 0.100 inch higher than the wiper bar.

The test procedure followed is described in Ref. 62. The instrumentation scheme is shown in Fig. IV-30.

Room temperature low pressure tests were the easiest to perform, so the similarity conditions which needed to be employed were studied.

For complete dynamic similarity in adiabatic compressible fluid flow, Mach number and Reynolds number similarity is required. For pressure drop calculations and experiments, Mach number similarity is by far the most important. This may be seen in a qualitative way by examining the differential equation characterizing one dimensional adiabatic compressible flow:

$$\frac{dp_o}{p_o} = - \frac{K M^2}{2} 4f \frac{dx}{D} \quad (1)$$

Substituting $f = 0.046/(Re)^{0.2}$, we have:

$$\frac{dp_o}{p_o} = - (0.092) K \frac{dx}{D} \left(\frac{M^2}{(Re)^{0.2}} \right) \quad (2)$$

Although the situation being modeled is not adiabatic, the general dependence given in Eq. 2 persists for nonadiabatic flow. This has repeatedly been verified by experiments at LRL and elsewhere. An attempt was made to simulate both Mach number and Reynolds number, with extreme care being taken to simulate Mach number. Reynolds number can be written:

$$Re = \frac{V \rho D}{\mu} = \frac{\dot{w}}{A} \frac{D}{\mu} = \frac{\dot{w} D}{\beta D^2 \mu} = \frac{\dot{w}}{\beta D \mu} \quad (3)$$

The Sutherland equation for air viscosity is given within 2% by:

$$\mu = \frac{C_1 T^{1/2}}{1 + C_2/T} \quad (4)$$

when the constants C_1 and C_2 are properly chosen. Substituting Eq. 4 into Eq. 3, we have:

$$Re = \frac{\dot{\omega} (1 + C_2/T)}{\beta DC_1(T)^{1/2}}$$

Mach number can be written:

$$\begin{aligned} M &= \frac{V}{\sqrt{KgRT}} = \frac{\dot{\omega}}{\rho A \sqrt{KgRT}} = \frac{\dot{\omega} RT}{\rho A \sqrt{KgRT}} \\ &= \frac{\dot{\omega}}{\rho A} \frac{T^{1/2}}{K^{1/2}} \sqrt{\frac{R}{g}} \end{aligned} \quad (6)$$

Let subscript 2 represent the model, and subscript 1 the prototype. The condition for dynamic similarity is:

$$\frac{M_1}{M_2} = \frac{Re_1}{Re_2} = 1 \quad (7)$$

Substitution of Eqs. 5 and 6 in 7, with appropriate subscripts gives:

$$\frac{p_2}{p_1} = \frac{(1 + C_2/T_1) T_2 K_1^{1/2}}{(1 + C_2/T_2) T_1 K_2^{1/2}} \quad (8)$$

During the flow tests Eq. 6 was used to determine the flow rates required for Mach number similarity. Pressures were made to conform as closely as possible to those determined by Eq. 8, which is the condition for Reynolds number similarity as well.

Application of Eq. 8 shows that dynamic similarity exists with entrance pressures of 173 in. Hg abs and exit pressures of 121 in. Hg abs through entrance conditions of 158 in. Hg abs to an exit of 111 in. Hg abs. This pressure range accounts for the temperature variation of the air in reactor operation.

Initially a lengthy series of tests was run, in which the passage was kept constant in cross-section along its length, and in which all of the springs were of constant width. In this series of tests only one passage (A or B of Fig. IV-29) was used at a time, the other being filled with putty. (Later it was found that there was little cross-flow between these channels.) After a series of spring widths had been tested in this fashion, shims were used to vary the passage height (to simulate gap size at different operating temperatures), and the sequence was repeated.

Flow rates were measured by means of a choked nozzle, and axial pressures were obtained by photographing a manometer bank. The photographs were later read with the aid of a low power microscope. During each test, data was taken at 3 to 5 different conditions bracketing the pre-determined conditions. At the end of each test the flow was stopped with the downstream valve. No-flow pressure data served to detect leaks in the large number of lines, since a leak would produce an obviously impossible pressure differential.

The results were then computed as a dimensionless loss parameter ($\Delta P/PM^2$), as a function of spring width and annulus height.

It became evident that the pressure drop caused by the springs was a function not only of spring geometry, but of spring position and spring sequence. Consequently, the testing program ultimately became a series of trial and error size selection at each axial position. During the final tests the actual annulus profile for the reactor structural limit condition was mocked up by the use of varying shim heights, and the surface corresponding to that of the duct was sandblasted to give a finish simulating that of the duct. (The effect of sandblasting on pressure drop was negligible.)

In general, the pressure drops varied approximately as the square of the free stream Mach number. This is shown on Fig. IV-31, where relative pressure drop is plotted as a function of Mach number for one size spring. The data are for a set of constant width springs (0.400 inch) in a constant area duct, and represent three separate runs, each run giving 18 data points. Each point represents the loss across one spring; being, in order, the loss from A to C, from B to D, from C to E, etc., as defined in Fig. IV-30.

When the data of Fig. IV-31 is replotted (Fig. IV-32) as a dimensionless pressure drop parameter ($\Delta P/PM^2$) versus axial position in the channel, a strong dependence of pressure drop on axial position is noted. The deviation of individual data points indicate that the general trend of this data is close to the predicted experimental error.

In general, this pattern changed slightly with each set of springs, but showed the same degree of repeatability for that set. Removing the springs and replacing them had a negligible effect, as did changing the individual spring sequence. As a result, the springs themselves could be eliminated as the source of the disturbance. The entire test section, including pressure taps, was carefully checked to see if any dimensional variation could be found;

the results were negative. It is evident from the sequence of pressure drop readings A-C, B-D, C-E, etc., that adjacent data points nowhere involve the same tap; yet there is evidently a rather continuous phenomena evident from tap to tap. This eliminates differences in tap geometry and position as a source. Finally, the static pressure check completing each test eliminated instrumentation as a source of error.

The shape of the pattern would suggest the possibility of organ pipe resonance. However, the pressures observed and the geometry involved indicated a frequency of about 300 cps and a noise level of 130 to 150 db. If indeed present, this should have been quite audible, but undue noise was not detected.

The explanation for the pressure drop variation seems to lie in variations in the flow pattern around the springs and associated turbulence. In Fig. IV-30, let the last wiper on the right represent the last wiper in the passage. In the case of the wing wipers (passage B), there is a passage below the wipers to insure adequate cooling of the duct. Additional pressure probes showed that for most of the wipers the greater part of the flow was passing through this passage instead of around the wipers. The pressure at F, for example, was less than that at E. However, in the case of the last wiper, the air must pass into a plenum where the pressure is that of G, where there is no section corresponding to F further downstream. At this point most of the air passes around the wiper rather than under it, requiring, as it turns out, a narrower wiper than would be expected. A similar effect requires a narrower wiper at the inlet.

Time was not available for the extensive testing required to further investigate this phenomenon. It was therefore decided to determine the average dimensionless loss parameter for various sized springs, and use this as a starting point to select the actual spring sizes by a series of trial and error solutions. Figure IV-33 shows the average dimensionless loss parameter for the narrow spring found on the top of the rails and wiper bars. The data for the 0.415-inch spring height appear more consistent than that for the rest, since a greater number of points is averaged in these particular data. Figure IV-34 shows the loss parameter for the wing wipers for one particular spring height.

From this point on, the procedure was the trial and error sequence mentioned earlier. The annulus area was varied axially by the use of shims and plaster covered by a thin alumina sheet, as shown in Fig. IV-35. The annulus areas were determined by considering the relative temperatures of the core and duct. These are tabulated in Table IV-3, and plotted in Fig. IV-36.

Table IV-3. Experimentally determined annulus areas.

Axial station (x/L)	Annulus between corrugated spring and duct (in.)	Rail flow area (in. ² × 1/2)	Bar flow area (in. ² × 1/2)	Wing flow area (in. ² × 1/2)
0	0.6342	0.1215	0.1495	0.4151
0.1	0.6481	0.1253	0.1533	0.4217
0.2	0.5610	0.1010	0.1292	0.3809
0.3	0.5419	0.0957	0.1238	0.3743
0.4	0.5326	0.0931	0.1211	0.3704
0.5	0.5130	0.0876	0.1156	0.3616
0.6	0.4902	0.0813	0.1092	0.3512
0.7	0.4763	0.0774	0.1054	0.3447
0.8	0.4727	0.0765	0.1044	0.3429
0.9	0.4793	0.0781	0.1061	0.3459
1.0	0.4939	0.0822	0.1102	0.3523

Note: The areas have been multiplied by 1/2 to conform to a convention used in heat transfer analysis.

One other observation made during the series of tests concerned the flow distribution around the springs. Each spring is essentially a double orifice, crossing the flow channel twice. The length of the portions crossing the channel is unequal. In the case of the bar and rail wipers, the ratio of the pressure drops across these is essentially the inverse ratio of these lengths. This is to be expected, since the entire flow must pass around the sides of these wipers. In the case of the wing wipers, however, the pressure drops are approximately equal. This indicates that a large portion of the flow passes under the springs in the space provided by the standoffs. These conclusions are quite important from the standpoint of heat transfer. This has been calculated on the basis of a uniform flow

profile across the channel. Due to the presence of the wipers, the flow in the center of the passage will actually be reduced in some regions, enhancing heat transfer effects. This is of particular value in the case of the duct surface, where cooling is most critical.

The dimensionless loss parameters were plotted for various sized springs in each position. These were then compared with the desired parameter for that location to select the wiper width. In interpreting the results, it must be kept in mind that the losses across each wiper are approximately 10 psi, so that a 10% error in this parameter results in about a 1 psi error at that position. Deviations of twice this value are hardly dangerous if compensated for by adjacent wipers.

In assuming the radial pressure loads on the reactor, one must also consider that the pressure drop does not occur as abruptly as indicated. This is because the wiper does not contact the spring at a mathematical point, but over an area approximately 1 inch long in the axial direction. This was witnessed in the water flow tests. Furthermore, there is a mechanical load of from 3 to 5 psi equivalent imposed at this point by the mechanical reaction of the wiper against the core. The actual pressure gradients in the annulus will appear more like A than like B in Fig. IV-37.

Figure IV-38 and Table IV-4 give the wiper dimensions finally selected. Table IV-5 gives the dimensionless loss parameters corresponding to these positions. It should be noted that the wing wiper widths of Table IV-4 are 0.075 inch narrower than the tests would predict. A design change for fabrication purposes was made after the tests had been completed, but this had a negligible effect on wiper performance. The loss parameters presented in Table IV-5 correspond to calculations based on the areas of Table IV-3.

Figures IV-39, 40 and 41 show the static pressures calculated for the three side support regions. These do not show the effects shown in Fig. IV-37.

Of the hundreds of sets of flow data available, the more pertinent sets are presented in Tables IV-6 through IV-12. Additional data may be found in the appendix of Ref. 63.

Run N-2 represents a flow test of the wipers in the small passage above the rails. Run R-1 represents a similar test on the passage immediately above

Table IV-4. Side support wiper dimensions selected.

Wiper No.	x/L	Wiper width		Wiper on rails	
		Bar	Wing	x/L	Width
1	0.047	0.285	0.675	0.059	0.345
2	0.142	0.285	0.800	0.155	0.345
3	0.237	0.315	0.900	0.250	0.345
4	0.332	0.310	0.900	0.345	0.315
5	0.427	0.320	0.950	0.440	0.315
6	0.522	0.275	0.925	0.535	0.315
7	0.618	0.295	0.950	0.643	0.290
8	0.713	0.315	0.950	0.738	0.290
9	0.808	0.315	0.975	0.833	0.290
10	0.903	0.320	0.950	0.928	0.290

Table IV-5. Dimensionless loss parameters for wiper dimensions selected.

Wiper No.	F_{rail}^*	F_{bar}^{**}	F_{wing}^{**}
1	1.53	2.33	1.79
2	2.40	2.39	1.81
3	2.07	2.13	1.85
4	1.63	1.89	1.73
5	1.80	1.94	1.82
6	1.73	1.87	1.83
7	1.44	1.74	1.80
8	1.31	1.67	1.78
9	1.26	1.66	1.83
10	1.20	1.73	1.90

* Direct experimental values

** interpolated from experimental results $\pm 5\%$

$F = \Delta P_{n-(n-1)} / P_n M_n^2 = \text{dimensionless loss parameter}$

$P = \text{static pressure at } n$

$\Delta P = \text{static pressure drop from station } n \text{ to the following station}$

$M^2 = \text{mach number squared based on the assumption of uniform flow across the channel at station } n$

$n = \text{the } n\text{th station is defined as the nearest even } 1/10\text{th } (x/L) \text{ position ahead of the wiper-core control point.}$

Table IV-6. Run N-2.*

Wiper No.	A-passage wiper width (in.)	B-passage wiper width (in.)	Pressure tap No.	A-passage static press. (in. Hg abs)	B-passage static press. (in. Hg abs)
1	0.345	Blocked	1	175.21	-
2	0.345	"	2	172.73	-
3	0.345	"	3	169.03	-
4	0.315	"	4	163.99	-
5	0.315	"	5	159.43	-
6	0.315	"	6	153.97	-
7	0.290	"	7	147.82	-
8	0.290	"	8	141.66	-
9	0.290	"	9	135.21	-
10	0.290	"	10	128.51	-
			11	121.06	-

* Plenum pressure 72.86 in. Hg abs.
 Plenum temperature 69° F.
 Effective nozzle diameter 0.3516 in.

Table IV-7. Run R-1.*

Wiper No.	A-passage wiper width (in.)	B-passage wiper width (in.)	Pressure tap No.	A-passage static press. (in. Hg abs)	B-passage static press. (in. Hg abs)
1	0.295	Blocked	1	175.84	-
2	0.325	"	2	172.49	-
3	0.315	"	3	168.70	-
4	0.325	"	4	164.35	-
5	0.325	"	5	158.67	-
6	0.295	"	6	154.89	-
7	0.315	"	7	149.36	-
8	0.315	"	8	143.58	-
9	0.315	"	9	137.44	-
10	0.315	"	10	130.50	-
			11	122.69	-

* Plenum pressure 87.08 in. Hg abs.
 Plenum temperature 68° F.
 Effective nozzle diameter 0.3516 in.

Table IV-8. Run V-1.*

Wiper No.	A-passage wiper width (in.)	B-passage wiper width (in.)	Pressure tap No.	A-passage static press. (in. Hg abs)	B-passage static press. (in. Hg abs)
1	Blocked	0.750	1	-	177.64
2	"	0.900	2	-	173.76
3	"	1.000	3	-	169.68
4	"	1.000	4	-	164.70
5	"	1.000	5	-	158.94
6	"	1.050	6	-	155.44
7	"	1.050	7	-	148.08
8	"	1.050	8	-	141.36
9	"	1.050	9	-	134.32
10	"	1.025	10	-	127.25
			11	-	120.02

* Plenum pressure 69.00 in. Hg abs.
 Plenum temperature 65° F.
 Effective nozzle diameter 0.7011 in.

Table IV-9. Run W-1.*

Wiper No.	A-passage wiper width (in.)	B-passage wiper width (in.)	Pressure tap No.	A-passage static press. (in. Hg abs)	B-passage static press. (in. Hg abs)
1	0.295	0.750	1	172.61	172.60
2	0.325	0.900	2	169.06	169.16
3	0.315	1.000	3	164.81	165.52
4	0.325	1.000	4	160.55	161.04
5	0.325	1.000	5	156.02	156.58
6	0.295	1.050	6	151.22	152.42
7	0.315	1.050	7	145.34	146.14
8	0.315	1.050	8	139.11	139.95
9	0.315	1.050	9	133.14	133.72
10	0.315	1.025	10	127.06	127.16
			11	120.78	120.62

* Plenum pressure 86.72 in. Hg abs.
 Plenum temperature 76° F.
 Effective nozzle diameter 0.7011 in.

Table IV-10. Run X-3.*

Wiper No.	A-passage wiper width (in.)	B-passage wiper width (in.)	Pressure tap No.	A-passage static press. (in. Hg abs)	B-passage static press. (in. Hg abs)
1	0.295	0.750	1	172.65	172.65
2	0.325	0.900	2	169.12	169.23
3	0.315	1.000	3	164.91	165.69
4	0.325	1.000	4	160.61	161.31
5	0.325	1.000	5	156.07	156.79
6	0.295	1.050	6	151.20	152.65
7	0.315	1.050	7	145.29	146.17
8	0.315	1.050	8	139.12	140.14
9	0.315	1.050	9	133.35	133.81
10	0.315	1.025	10	127.27	127.40
			11	120.97	121.01

* Plenum pressure 86.1 in. Hg abs.
Plenum temperature 68° F.
Effective nozzle diameter 0.7011 in.

Table IV-11. Run Y-1.*

Pressure tap No.	A-passage static press. (in. Hg abs)	B-passage static press. (in. Hg abs)
1	142.46	142.46
2	-	141.50
3	140.83	140.97
4	140.30	140.35
5	139.79	139.88
6	139.29	139.33
7	138.57	138.63
8	137.96	138.00
9	137.36	137.36
10	136.79	136.87
11	136.33	136.41

* Plenum pressure 86.23 in. Hg abs.
Plenum temperature 62° F.
Effective nozzle diameter 0.7011 in.

Table IV-12. Run Y-3.*

Pressure tap No.	A-passage static press. (in. Hg abs)	B-passage static press. (in. Hg abs)
1	167.21	167.21
2	-	155.35
3	151.13	151.99
4	147.37	147.63
5	144.23	144.61
6	140.73	140.75
7	135.83	136.05
8	130.55	130.95
9	126.19	125.87
10	121.26	122.12
11	116.85	117.49

* Plenum pressure 113.63 in. Hg abs.
Plenum temperature 62° F.
Effective nozzle diameter 0.9861.

the wiper bars. Run V-1 represents the test on a typical wing wiper passage. Run W-1 represents a test on a combined wing wiper and bar wiper passage to check for crossflow effects. Run X-13 is a similar test on the same section as the W-1 run after the fixture had been completely disassembled and the duct portion sandblasted. It shows that the effect of sandblasting is negligible, and that the test results are quite repeatable. Run Y-1 is a test on the same section tested in run X-3 with all the wipers removed. It shows that without the wipers the same flow rate occurs at an overall pressure drop only 1/8 that required when the wipers are present. Conversely, the data for Run Y-3 shows that without the wipers, the flow rate is 2.75 times that with the wipers for the same overall pressure drop.

A final test was performed with a more exact model. A spare reactor wiper bar complete with bar and wing wipers and shroud was placed in a stainless steel passage. Core barreling was simulated by stainless steel shims over cast alumina. The three passages involved (one bar wiper and two wing wiper passages) were exact to the point of having corrugated spring separations and wiper bar and rail feet protruding into the flow on either side.

In cold flow tests this arrangement duplicated the results shown for Run X-3 to within 1 psi. Two tests were performed at 1400° F for periods of 15 minutes. Results duplicated the predicted curve within 2 psi at a flow rate that was within 5% of that predicted. No damage to the wipers could be detected.

G. Shroud Assembly

The shroud, shown in Fig. IV-42, prevents the cool side support air (1300° F) from mixing with the reactor hot exhaust air (2150° F). The duct is thus kept cool back to within 3 inches of the duct support ring by flow channel separation. The exhaust air from the shroud will not mix significantly with the main flow for distances less than 15 t, where t is the thickness of the shroud flow passage (1.25 in.). This film cooling adequately protects the exposed duct, support ring, and nozzle flange. Figure VI-53 shows the last shroud piece being assembled into the reactor.

The shroud load is a radially outward pressure force resulting from a 3 psi pressure differential between the reactor exhaust plenum and the annulus. The load on the largest shroud piece is about 18.5 pounds. The load of one shroud set acting on the bar is 109 pounds.

The maximum temperature of the shroud pieces, shroud wiper spring, and shim skirt is 1625° F for the structure limit condition. The wiper bar is below 1500° F. The higher stressed part of the shroud wiper spring is below 1450° F.

Figure IV-43 gives the shroud, spring, and skirt stresses versus the pressure drop across the shroud and Rene' 41 yield strength. A 3 psi ΔP is expected. The spring and skirt stresses are moderate for as much as 10 psi pressure drop across them. Similarly, the bar stress versus the pressure drop across the shroud and the yield strength of Hastelloy C is given. Bar stress is moderate.

H. Reactor Duct

1. Introduction

The reactor duct consists of two sections; the forward reactor duct (Duct 4) and the aft reactor duct (Duct 5). The aft reactor duct is 103 inches long and contains the reactor core. It is a thin-walled, completely

machined pressure vessel penetrated by 600 holes for the reactor side support system and about 80 holes near each end for instrumentation. (Fig. IV-44.) Loads imposed by the reactor side support system are relatively small; but the combination of size, temperature, pressure, thermal stress, and stress raiser conditions imparts severe requirements.

The reactor pressure drop load is transferred from the support grid to the duct via ninety slotted lugs. Bending moments in the duct are minimized by distributing this load with only a small eccentricity of the lugs. The lugs are slotted to allow cooling of the duct and support grid in this area. The smaller support grid radial and forward loads are transmitted to the duct by 12 radial pins.

The forward reactor duct (Duct 4) is 60 inches long and contains the reactor control and poison systems (Fig. IV-45). It is accurately aligned to Duct 5, and its penetrations for various connections are reinforced (Fig. IV-46). Analysis of Duct 4 is not reported in detail because its environmental requirements are much less severe than those of Duct 5, and its factors of safety are greater.

The wall thickness of the reactor duct sections varies as the environmental requirements. It is thin (0.437 inch) in the areas of high nuclear heating and thicker (0.5 inch to 0.625 inch) in areas used for miscellaneous support (Fig. IV-47). The thick bolted flanges were chosen mainly for their reliability, but they also provide desirable remote assembly-disassembly capabilities.

The basic design philosophy was to guarantee safe operation under all foreseeable operating conditions. On this premise the duct wall thickness in the reactor area was increased (from 0.375 inch to 0.437 inch) for greater reliability in case of increased pressure due to reactor flow blockage or supply air valve failure. Surfaces of all holes through the duct wall were chamfered to decrease the possibility of crack formation. Duct 5 was fabricated from eleven rolled and welded plates and two flange forgings. Approximately 1000 linear inches of welds were positioned to miss the 680 holes.

2. Load Requirements

The primary loads that affect the reactor duct design are shown in Fig. IV-48 for the structural design point condition.² The step in the curve

at Station 351 is the reactor support grid load, and the one near Station 425 is due to the wiper bar attachment. The circumferential loads are due to the static pressure within the duct.

Axial bending loads due to the duct support system, and circumferential bending due to inertial loads transmitted through the side support rails are negligible. Flange bolt loads are induced loads which are discussed in paragraph H. 5., Stress Analysis.

3. Materials

Hastelloy C was chosen²¹ as the duct material, giving due consideration to the needs of a ground test reactor and the basic philosophy of extreme reliability. In addition, flight requirements were part of the material selection considerations.

A unitized missile design in which the reactor duct is also a section of the air frame and missile skin was assumed. There are two main differences between ground test and flight-type reactor ducts: (1) a flight-type duct will not have external flanges, and (2) it will be externally cooled during flight. External cooling will remove $1/3$ to $1/2$ of the heat deposited in the reactor duct. The resultant temperature distribution in the duct wall would result in 60% lower thermal stress. Also, the maximum metal temperature could be about 100°F less than for a ground test duct. The flight-type duct must resist maneuver, boost, and thrust loads, but the magnitude of these loads of the duct is small compared to the pressure load. Due to the possible higher temperatures and cyclic operating conditions, the material requirements for the ground test duct are more severe than the flight-type duct. Because of these differences, the flight-type reactor duct might appear different in design and might be made of a different material.

a. Material requirements

The basic material requirements for the reactor duct are:

- Sufficient strength at operating temperature.
- Ductile (not hot short* in operating regions).
- Readily weldable with high weld efficiency at operating temperature.
- Available in required shapes and forms.
- Prefer no heat treatment after forming and welding.
- Readily machinable.

These requirements are discussed in turn below.

The strength is a function of loads, stress raisers, ductility of material, wall thickness, and axial temperature distribution in the duct wall. Preliminary analyses using assumed high values (1.0 mW/ft^3) of internal heat generation indicated the membrane stress and thickness requirements vs maximum temperatures of the reactor duct (Fig. IV-49). Prospective alloys were compared against this curve. The strength properties of interest are yield strength and creep rupture strengths of up to 3-hour duration. Reasonable safety factors based on yield strength would be 1.5 (0.2% offset) and about 1.0 based on 1.0% creep for 3 hours.

Ductility is a highly desirable material property in the reactor duct design because of thermal stress and stress concentrations. This also implies that the material should not be "hot short."

There are many holes through the duct wall for the side support system. A theoretical stress concentration factor of 2.5 is caused by these holes. In brittle materials this localized stress is a direct measure of the likelihood of failure. In ductile materials this stress concentration can usually be neglected.

* A generally ductile material is hot short if it exhibits a large decrease in ductility within a specific temperature range. This is a noticeable dip in the tensile elongation vs temperature curve. Total elongation in creep rupture tests may be less than in short-time tensile tests. Therefore, short-time tensile elongations indicate only generally the magnitude of the hot short condition.

In the case of brittle material fracture, the magnitude of the stress is a direct measure of the likelihood of failure. In such cases, the elastic thermal stress is very significant. In ductile materials, a single application of thermal stress will not generally result in fracture. Thermal stress cracks may occur in ductile materials because of repeated applications of the thermal stress resulting in cyclic plastic strain.

Many thermal-stress problems involve ductile materials for which the material behavior ceases to be elastic long before fracture occurs. Results of an elastic thermal stress analysis are inaccurate for regions beyond the elastic limit and are of very little significance as a criterion of fracture in ductile materials. Unlike stresses caused by sustained mechanical loads where resultant forces and moments must be balanced by definite average stress and stress moment resultants, thermal stresses can easily self-relieve.

The size of the reactor duct requires that it be made up of several plates with both girth and longitudinal weld joints. The maximum load is a hoop force across the longitudinal joints. A reinforced (thicker section) joint would suffer greatly from increased temperature because of internal heat generation; therefore, the material must be readily weldable with high joint efficiency at operating temperatures.

Heat treatment after forming and welding of this large, thin-walled vessel could possibly distort it a great deal. The control of this distortion could be a major engineering problem, especially when using precipitation-hardening alloys. For ease of fabrication it is preferable to use a material that does not require heat treatment after forming and welding. As it turned out, the reactor duct was solution-annealed after forming and welding, and no appreciable distortion occurred.

The nature of the flanges, support grid attachment, and side support system requires the duct to be readily machinable, especially for turning and drilling operations.

b. Material properties

There are several alloys that have high strengths in the required temperature range (1250° F to 1450° F). The precipitation-hardening

(PH) nickel-base alloys are the strongest; the other high-temperature alloys such as the cobalt-base, solid solution nickel-base, and some of the complex alloys are in approximately the same strength class.

All of the alloys in the PH nickel-base group are age-hardened with co-present Al and Ti and owe the major portion of their high temperature strength to the Ni_3Al , Ni_3Ti , and $\text{Ni}_3(\text{Ti}, \text{Al})$ precipitates. Typical alloys available in plate form are: Hastelloy R-235, Inconel X, Inconel W, Udimet 500, Rene' 41, some Nimonics, M-252, and Waspaloy. The general characteristics of this group of alloys in the temperature range of interest are:

The base material, in the fully heat-treated condition, easily meets the minimum strength requirements based on membrane stresses.

All are hot-short in the required temperature range. Minimum elongations as low as 2% have been reported for some of these alloys. Generally, the minimum elongation range for this group varies from 4% to 10%.

They can be fusion-welded, but the high-temperature strength of welds, after heat treatment, is somewhat below that of the heat-treated wrought material. Rupture weld efficiencies at elevated temperatures are between 60% and 90% of the base material. These welds usually display hot-short characteristics and are sensitive to strain cracking. Heat treatment after welding is required, and it is recommended that these welds not be located at positions of high stress.

All require a solution of stress-relieving heat treatment after forming and welding. Most require between one and three aging heat treatments to develop maximum strength.

They are more difficult to machine than some of the solid-solution alloys. They can be machined, but require special techniques and over-powered machines.

The high cobalt-base alloys would absorb more neutrons and stay "hotter" for longer periods of time, but this effect is not considered in this analysis. Of the two such alloys available in plate form, L-605 does not have sufficient strength in the required temperature range, and J-1570 (a precipitation-hardening alloy) has the same characteristics as the PH nickel-base alloys.

The heat-resistant complex iron-base alloys, like A-286 and 19-9DL, rapidly lose strength above 1200° F. A-286 is a precipitation-hardening alloy and is hot-short in the 1200° F to 1300° F range. Other complex alloys, such as N-155, do not quite meet the strength requirements.

There are several solid-solution nickel alloys available in plate form (Inconel and Hastelloys B, C, and X). Inconel and Hastelloy X do not have sufficient strength. Hastelloy B is slightly inferior to Hastelloy C in strength, ductility, and stability. Hastelloy B is also affected by a Ni and Mo beta phase transformation at about 1300° F, which has a great effect on creep ductility. Hastelloy C meets the strength, ductility, and fabricability requirements.

The approximate safety factors based on hoop stresses and preliminary maximum wall temperatures, using Hastelloy C, are given in Table IV-13.

Table IV-13. Approximate safety factors.

Wall thickness (in.)	Max temp	Ground test conditions		Flight conditions		
		Based on 0.2% yield ¹	1% creep for 3 hr ¹	3-hr rupture ²	1% creep for 3 hr ¹	3-hr rupture ²
0.250	1300° F	1.3	1.3	1.7	1.6	2.0
0.313	1370° F	1.6	1.25	1.8	1.7	2.1
0.375	1475° F	1.8	1.2	1.7	1.7	2.3

¹Average data.

²Minimum data.

Hastelloy C, with a very stable matrix, has superior strength under rapid heating conditions.⁶⁴ This stability under rapid heating conditions will increase the reliability of the reactor duct.

Hastelloy C ductility is always greater than 20% based on tensile strength of wrought material. The significant difference between yield and ultimate strengths, along with the high elongation, indicates good formability and fabricability at room temperature.

It is readily weldable with elevated tensile and rupture weld strengths within the base material strength ranges. Heat treatment after welding does not increase strength but improves ductility and resistance to crack propagation. Many fabricators have had experience with Hastelloy C. It work-

hardenes during machining, but not as much as most of the super-alloys. It is machinable if slow speeds, heavy cuts, and sharp tools are used.

There are also several specifications available. Wrought plate and welding wire are covered by Military, ASTM, and ORNL specifications, and welding by Military, ASME, ASTM, and ORNL specifications.

This material has a high density ($\rho = 0.323 \text{ lb/in}^3$) due to the 4% W, and therefore tends to absorb more gamma rays. Its maximum internal heat generating rates are expected to be about 10% greater than Hastelloy R-235.

It should be noted that early in the program the maximum loads (pressure) were well defined and overloads were not likely to occur. The maximum wall temperature was not well defined, so reliability on over-temperature was of prime concern. With increasing temperatures the yield strength for Hastelloy C is fairly even, but it decreases rapidly for some of the other high temperature alloys (Fig. IV-50). The rupture strengths of Hastelloy C and other alloys are compared in Fig. IV-51.

c. Hastelloy C property evaluation

After selecting Hastelloy C for the duct material, an extensive evaluation program was begun to verify its properties with respect to the reactor duct requirements. Optimum fabrication procedures were also determined. The tests consisted of tensile, creep, rupture, cyclic creep, bend, impact, hardness, and microstructure. The variables were test temperature, heat treatment, weld procedure, and material melt. More than 800 specimens were tested during the various material evaluation and certification tests for all duct sections.⁶⁵⁻⁶⁹

All but a few of the many tensile specimens exceeded the minimum specified yield and ultimate strengths of the ASME Unfired Pressure Vessel Code, case No. 1194. Various heat treatments had negligible effects of yield and ultimate strengths, but had marked effects on ductility (Fig. IV-52 and IV-53).

Since ductility was the primary item affected, guided bend tests were used to determine effects of welding, heat treating, and operating temperatures. Base material bend results for 2 hours at various temperatures are shown in Table IV-14. Across-weld guided bend results are shown in Table IV-15. These results indicated that the low-heat as-welded specimens were quite ductile, but aging at 1500° F for 1 hour markedly decreased this

Table IV-14. Effect of temperature history on room temperature hardness and ductility of Hastelloy C. (Room temperature guided bend tests and hardness data.)

Temperature* (2 hr @temp, water quenched)	Hardness (Rockwell B)	% Elongation in 2 in.	Bend angle (deg)
As received	94.5	20	180
1000	98.5	20	180
1200	99	20	180
1300	97	20	180
1400	96	20	180
1500	100	20	180
1600	96.5	20	180
1700	101	18	138
1800	101.5	10	38
1900	103	8	27
2000	101.5	6	29
2100	100.5	20	180
2225	98	20	180
2225**	95.5	20	180

*Average of three tests for each condition.

**Temperature and quench cycle repeated four times prior to room temperature tests.

ductility. Solution heat treating high-heat welds made them quite ductile, but subsequent aging affected the ductility of specimens solution heat treated at temperatures below 2130° F.

All-weld metal charpy impact tests further indicated (Table IV-16) that the recommended solution heat treatment (2225° F and water quench) would yield the most reliable material.

All the creep and rupture test results (Fig. IV-54 and IV-55) for plate were within the advertised sheet metal values. This includes base and all weld metal specimens (as-welded and solution heat treated). The cyclic creep results were also within the ranges shown in Fig. IV-54.

Table IV-15. Across-weld room temperature guided bend tests.

Post-weld heat treatment		% Elongation in 1 in., across weld	Bend angle (deg)
Solution heat treatment*	Aging		
As welded (L)	none	18-21	180
"	12 hr @ 1200° F	20	180
"	1 hr @ 1500° F	13	55
"	12 hr @ 1200° F + 1 hr @ 1500° F	5- 8	22-50
"	12 hr @ 1300° F	12	52
As welded (H)	none	12-16	46-60
"	12 hr @ 1200° F + 1 hr @ 1500° F	6-12	32-49
2050° F	none	18-20	180
2050° F	1 hr @ 1500° F	16-20	65-180
2100° F	none	20-22	180
2100° F	1 hr @ 1500° F	9-21	40-180
2125° F	none	18-20	180
2125° F	1 hr @ 1100° F	20-23	180
2125° F	1 hr @ 1400° F	21-24	180
2125° F	1 hr @ 1500° F	10-18	41-69
2150° F	none	18-24	180
2150° F	1 hr @ 1500° F	20-24	180
2220° F	none	20-21	180
2220° F	1 hr @ 1100° F	20-21	180
2220° F	1 hr @ 1400° F	20-22	180
2220° F	1 hr @ 1500° F	20-21	180

* Water quenched after SHT.

L Low heat welding.

H High heat welding (used also on all SHT specimens).

Table IV-16. All-weld metal charpy impact tests at room temperature.

Post-weld heat treatment		Hardness (Rockwell C)	Energy (ft-lb)
Solution heat treatment	Aging		
As welded	none	27-27.5	14.75-17.75
"	4 hr @ 1300° F	26-29.5	16 -19
"	4 hr @ 1500° F	34-36.5	4.5 - 8.5
2125° F	none	21.3-23	28.5 -30
2125° F	1 hr @ 1500° F	22.6-23	19 -20
2220° F	none	25.7-32	60 -67
2220° F	1 hr @ 1500° F	27.5-28.2	40 -64

4. Thermal Analysis

Thermal reliability concerns the structural reliability of the duct and the differential expansion effects with other components. Some of the parameters which affect duct temperatures are: nuclear heating rates, reactor power level, reactor air flow rate and inlet temperature, side support annulus air temperature, changes in side support gap caused by differential expansion between duct and core, ambient temperature, dimensional tolerances of the assembly, material properties, and convection on the duct outside surface caused by wind or induction by reactor exhaust. Exact values of most parameters are not known, so most analysis results are presented in curve form.

a. Nuclear heating

Nuclear heating rates have a large influence on the reliability of the aft reactor duct. The curves discussed in this paragraph summarize the nuclear heating analysis based on the structural design point condition (Mach 3.0) and 1 hour of reactor running time. (Analysis details are given in Ref. 29 and 70 through 74.) For the 1-hour operating time, 90% of infinite saturation for fission product decay is assumed.

Power generated = 489 mW

Active hot core volume = 54.04 ft³

∴ Power density = 9.05 mW/ft³

Heating values for other power levels after 1 hour can be determined from the curves by a direct ratio of power levels.

Figure IV-56 indicates the nominal value of incident nuclear heating rate on the duct inside surface, and the gradient through the wall near the reactor center. Figure IV-57 indicates the range of effective total absorption coefficient, and the nominal and estimated extremes for the average nuclear heating rate in the duct wall. Figures IV-58 and IV-59 indicate the nominal and estimated extremes for nuclear heating distribution in the forward and aft flange sections. Nuclear heating rates in Duct 4 are much less than in Duct 5.

b. Convection coefficients

The heat transfer film coefficients vary a great deal from inside to outside, and along the length of the reactor duct.

The following analysis considers each region separately (Fig. IV-60), and is based on the structural design point condition.

For most cases outside the reactor core, the Dittus-Boelter equation for fully developed turbulent pipe flow is applicable.

$$\frac{hD}{k_b} = 0.023 \left(\frac{GD}{\mu_b} \right)^{0.8} Pr^{0.4} \quad (1a)$$

this can be reduced to:

$$h = (492) \frac{\bar{h}(G)^{0.8}}{D^{0.2}} \quad (1b)$$

where

$$\bar{h} = 0.023 \frac{k}{(\mu_b)^{0.8}} (Pr)^{0.4}$$

In general, the above equations are conservative, because the turbulence caused by internal structures found in the duct tends to increase the Nusselt numbers. For high heat flow rates, the Sieder and Tate correction factor can be applied to the above equations.

$$C.F. = \left(\frac{\mu_b}{\mu_w} \right)^{0.14} \quad (2)$$

Region A: Inside surface of essentially open duct forward of the reactor support grid:

Using Eq. 1 and the values

$$D = 56.250 \text{ in.}$$

$$t_a = 1063^\circ \text{ F}$$

$$P = 340 \text{ psia}$$

$$W = 1814 \text{ lb/sec}$$

$$G = 0.73 \text{ lb/sec in}^2$$

$$\bar{h} = 0.586$$

the film coefficient is found to be

$$h = 100 \text{ Btu/hr ft}^2 \text{ }^\circ \text{ F.}$$

Region B: Region just forward of support grid. Air flow in this region is disturbed by the screen support ring and instrumentation conduits. The screen ring lets a little air flow between it and the duct, but the conduits almost exclude air from scrubbing the duct wall. It is estimated that the film coefficient in this area is negligible ($0 < h < 20 \text{ Btu/hr ft}^2 \text{ }^\circ \text{ F}$).

Regions C and D: Flow between support grid ring and duct. Flow in this region is like the entrance of a tube, and greater heat-transfer coefficients may be expected than for fully developed turbulent flow. Since mass flow in this region is estimated (assumed to be the same as the open duct), a coefficient for fully developed flow is conservatively used. The bases for this flow estimate are (1) the general configuration of the support grid assembly in the duct, (2) the support teeth being about 50% porous and chamfered on forward and aft surfaces, and (3) negligible calculated pressure losses at average mass flow. From Eq. 1 with $t_a = 1063^\circ \text{ F}$ and $G = 0.73 \text{ lb/sec in}^2$, the film coefficients are:

$$\text{Region C: } D_C = (2) 0.93 = 1.86 \text{ inches} \quad h_C \simeq 197 \text{ Btu/hr ft}^2 \text{ }^\circ \text{ F}$$

$$\text{Region D: } D_D = 0.84 \text{ inches} \quad h_C \simeq 232 \text{ Btu/hr ft}^2 \text{ }^\circ \text{ F}$$

Region E: Flow along duct between support grid and reactor core. Flow in this region is about the same as in the open duct (Region A), and may be slightly higher because of the high flow in the side support annulus. The support grid ring and teeth create a high turbulence level and the film coefficient is estimated to be 40% higher than the nominal fully developed flow value. Therefore, it is reasonable to expect:

$$h_E > 140 \text{ Btu/hr ft}^2 \text{ }^\circ \text{ F}$$

Region F: Flow along duct in reactor side support annulus. Flow in this region is influenced by many items, including varying flow area, variable material and air properties, and turbulence caused by the general configuration of the side support system (Fig. IV-61). As indicated in a previous section, the wiper springs were designed to allow the air to contact a maximum of the duct surface.

Based on the duct-to-core gap being the same as the room temperature value (radial expansion of duct and core are about equal), the estimated wing wiper flow passage total flow rate is 84 lb/sec. Thus the flow per passage is 0.7 lb/sec. With the air at 1100° F:

$$h_{F_b} = 405 (1.4) = 560 \text{ Btu/hr ft}^2 \text{ } ^\circ \text{F}$$

in area "b" of Fig. IV-61. The factor 1.4 accounts for additional turbulence and sandblasted (420 rms) duct surface

$$h_{F_a} = 730 \text{ Btu/hr ft}^2 \text{ } ^\circ \text{F}$$

in area "a," based on $2/3 W_W$ flowing under the wipers.⁶³ It should be noted that film coefficients tend to increase with smaller gaps and decrease with longer gaps.

Region G and H: Flow in and aft of duct shroud annulus. The side support air flow continues through the annulus between the duct and shroud:

$$W = 185 \text{ lb/sec}$$

$$\text{gap} = 1.0 \text{ inch; so } D = 2 \text{ inches}$$

$$t_{\text{air}} \simeq 1300^\circ \text{F}$$

Therefore:

$$h = 270 \text{ Btu/hr ft}^2 \text{ } ^\circ \text{F}.$$

The aft part of the shroud tapers away from the duct. This decreases the film coefficient to 200 Btu/hr ft² °F. Aft of the shroud, the duct wall is protected from the hot reactor exhaust air by the film cooling annulus air for at least 10 inches.

Convection on the outer surface of the duct can be either natural convection enhanced by wind, or induced by the nozzle exhaust air. Natural convection would be negligible in this case but wind may have an effect, depending on the atmospheric conditions during the reactor test. The wind

velocity would most likely be between 0 and 30 mph. Induction by the exhaust will probably have the most effect, but the magnitude of this effect can only be estimated.

The effect of support structures, brackets, and instrumentation conduits on the outer surface of the duct make a rigorous analysis prohibitively difficult. Convection will probably have a small effect near the ends of the reactor duct because of the relatively low heat flow rates and distribution of instrumentation brackets. Near the center of the duct the effect can be quite noticeable. The rail nuts create a rough surface on the outside of the duct and tend to increase convection. The characteristic length to use in the convection equation is questionable, but the results will be within 30% if we let $x = 20$ inches (x could conceivably approach 4 inches or 50 inches). Figure IV-62 indicates the approximate convection coefficient versus air velocity (wind or induction) for the reactor duct. The curve is based on:

$$h_m = 0.037 k_b \frac{\mu_a^{0.8}}{\gamma_b} \frac{Pr^{0.33}}{x^{0.2}} \frac{\mu_b^{0.14}}{\mu_W} \quad (3)$$

with

$$t_b = 70^\circ \text{F} \text{ and } t_w = 1250^\circ \text{F}.$$

Then

$$h_m = 0.43 (\mu_a)^{0.8} \text{ Btu/hr ft}^2 \text{ }^\circ \text{F}.$$

It is reasonable to assume that an equivalent wind of 30 mph would exist during operations.

c. Thermal radiation coefficients

Radiant heat transfer between the duct and side support system is negligible during steady state, as both are at about the same temperature. As the duct heats up above 1000°F , thermal radiation to the atmosphere becomes important.

An analysis was made of maximum duct temperature versus emissivity. The results are shown in Fig. IV-63. Since machined Hastelloy C has a low emissivity ($\epsilon = 0.3$), methods to increase it were studied. Oxidizing at high temperature would increase the emissivity but could possibly distort the duct. The most reliable method found was sandblasting the surface with aluminum oxide grit. Tests indicated that this would insure an emissivity

greater than 0.7 for the first hot power run.^{75, 76} Oxidation during a power run would increase the emissivity value. As Fig. IV-63 indicates, sand-blasting decreases the maximum duct temperature about 60° F, and thereby increases its reliability.

The air temperature and nuclear heating rates used in preparing Fig. IV-63 are valid, but the inside film coefficient is low by about a factor of two. The revised film coefficients just discussed decrease the effect of emissivity on duct temperature in this area. The effects of emissivity on duct temperatures just forward of the core are shown in Fig. IV-64.

Accordingly, the aft reactor duct outside surface was sandblasted between Stations 355 and 426. Both ends were left as machined, since expected temperatures there were lower.

d. Heat transfer

The basic equation for heat conduction at any given axial position of the duct, assuming constant thermal conductivity, is given by Eq. 4.

$$\alpha \nabla^2 t = \frac{\partial t}{\partial \tau} - \frac{q(\vec{r}, \tau)}{\rho C} \quad (4)$$

Using cartesian coordinates, for a thin-walled cylinder with negligible axial and circumferential conduction, Eq. 4 reduces to:

$$\alpha \frac{\partial^2 t}{\partial x^2} = \frac{\partial T}{\partial \tau} - \frac{q(x, \tau)}{\rho C} \quad (5)$$

The nuclear heating density (Fig. IV-65) varies exponentially through the wall.

$$q(x) = q_1 e^{-\mu_a x} \quad (6)$$

For steady state conditions, Eq. 5 reduces to:

$$\frac{d^2 t}{dx^2} = - \frac{q_1 e^{-\mu_a x}}{k} \quad (7)$$

Integrating twice and using arbitrary surface temperatures for boundary conditions, the temperature at any point in the wall is given by

$$t(x) = t_1 + \frac{q_i}{\mu_a^2 k} \left[1 - e^{-\mu_a x} - e^{-\mu_a \ell} + 1 \right] - (t_i - t_o) \frac{x}{\ell} \quad (8)$$

Equation 8 need not be used in analyzing thin-walled regions ($\ell < 0.8$ inches and $\mu_a \approx 0.53 \text{ cm}^{-1}$). There the nuclear heating density is assumed to vary linearly through the wall, and an average heating density is used. The error due to these simplifications is only about 2%. Thus, a heat balance across the wall gives the equations:

$$t_3 = t_1 + \Delta t_{1-2} + \Delta t_{2-3} = t_5 + \Delta t_{5-4} + \Delta t_{4-3} \quad (a)$$

$$t_1 + \frac{\bar{q}_a(a)}{h_1} + \frac{\bar{q}_a(a)^2}{2k} = t_5 + \frac{\bar{q}_b(b)}{h_5} + \frac{\bar{q}_b(b)^2}{2k} \quad (b) \quad (9)$$

$$t_4 = t_1 + \frac{\bar{q}_a(a)}{h_1} + \frac{\bar{q}_a(a)^2}{2k} - \frac{q_b(b)}{2k} = \frac{\bar{q}_b(b)}{h_5} + t_5 \quad (c)$$

where

$$h_5 = h_c \text{ th}_r \quad h_r = \frac{\epsilon \sigma T_4^4 - T_5^4}{T_4 - T_5} \rightarrow \frac{\sigma \epsilon T_4^4}{t_4 - t_5}$$

$$\bar{q}_a = q_i - m(a) \quad \bar{q}_b = q_o + m(b)$$

$$m = \frac{q_i - q_o}{2\ell} = \frac{q_i - \bar{q}}{\ell}$$

Equations 9 are solved for various values of h_c or ϵ .

This approach is applicable for steady state temperatures in Regions E, F, and part of G. For Regions C and D an adiabatic outer wall is assumed, so Eq. 9 reduces to:

$$t_3 = t_1 + \frac{\bar{q}(\ell)}{h_1} + \frac{\bar{q}(\ell)^2}{2k} \quad (10)$$

Two methods were used to analyze transient temperatures of the thin-walled sections of the duct. Both methods are based on negligible internal resistance ($k \rightarrow \infty$). The first method is one-dimensional and applicable for hand calculations. It is based on an adiabatic outer wall and a step input of internal heating and convection.

From the heat balance across the wall:

$$qV = V\rho C \frac{\partial t}{\partial \tau} + hA (t - t_f) \quad (11)$$

with

$$\theta = t - t_f, \quad d\theta = dt, \quad \text{and } M = \frac{hA}{V\rho C} = \frac{h}{\ell\rho C}$$

t_f = convective air temp

t_1 = initial wall temp when $\tau = 0$

giving,

$$\frac{d\theta}{d\tau} + M\theta = \frac{q}{\rho C}$$

Integration by Laplace transforms gives:

$$t(\tau) = \left[\frac{q}{M\rho C} + t_f \right] (1 - e^{-M\tau}) + t_1 e^{-M\tau} \quad (12)$$

This equation is applicable to Regions C, D, E, part of G, and for estimates on Region F.

For detailed analyses of the reactor side support structure, which includes Region F of the duct, in the steady state and transient conditions a Fortran computer code for the IBM 7094 was developed. (Ref. 51 and 77 through 80.) The cross-sectional geometry was represented by means of a 58-zone model including 12 flow channels. The duct wall was divided into 4 slices, and allowances were made for heat transfer on the outer surface. Calculations were performed for each of 21 axial stations along the length of the reactor.

Two methods were also used to analyze transient temperatures in the thick-walled sections (flanges) of the duct. Both methods considered thermal conductivity constant, exponential decay of nuclear heating, and outside surface heat transfer. For hand calculation the one-dimensional Schmidt graphical method was used. Nuclear heating was accounted for by adding in the local adiabatic internal heating temperature rise for each time increment.

The other method included axial conduction and analyzed the flange, hub, and duct support ring but did not include the effect of the joint. It was a Tiger V program conducted on a Philco 2000 digital computer.⁸¹ The Tiger code is a transient three-dimensional heat transfer program which

allows heat transfer coefficients and internal heating rates to vary with time and space. The results of both methods for the flange near the joint were in good agreement.

The typical operating program used for most transient analyses is shown in Fig. IV-66. Small changes in the program will have a small effect on the transient temperature behavior of the duct.

Most of the duct reaches steady state during the first few minutes at full power, but flange temperatures are transient throughout the run. Both flanges (Regions A and G) heat up mainly by convection on the inside surface. In the forward flange hub (Region B) the primary mode of heat transfer is conduction and nuclear heating. Outside thermal radiation tends to cool both flanges, and thermal radiation from the duct shroud increases the heat flux to the aft flange area. The nominal temperature distribution in the aft flange at the end of the power plateau (Fig. IV-66) is shown in Fig. IV-67. The nominal transient temperature behavior is shown in Fig. IV-68 for the forward flange, Fig. IV-69 for the aft flange, and Fig. IV-70 for the aft flange hub. The peak temperature of the forward flange is about 200°F less than that of the aft flange.

The steady state temperature distribution for a section through the support grid attachment regions (C and D) is shown in Fig. IV-71. Between the support grid and core (Region E) and aft of the core (Region G), outside convection and thermal radiation became important. Peak Station 357 wall temperature versus outside surface emissivity for various cases is shown in Fig. IV-64. These results are based on no axial conduction, but axial conduction tends to lower some of the temperatures in this area (Fig. IV-72).

In Region E nuclear heating, thermal radiation, and inside and outside convection have large effects on temperatures. Steady state temperature distribution through the duct wall at its hottest position for several cases is shown in Fig. IV-73. The most extreme case (not shown in Fig. IV-73) with $h_1 = 400 \text{ Btu/hr ft}^2 \text{ }^\circ\text{F}$, has a peak temperature less than the 1550°F maximum allowable temperature. The transient response to the program of Fig. IV-66 at the center of the duct is shown in Fig. IV-74. The time constant of this region at the beginning of full power is about 30 seconds. Also shown in Fig. IV-74 is the response at Station 430 (aft of the core, where the duct wall is thicker). The steady state nominal axial temperature

distribution for the duct wall is shown in Fig. IV-72. Transient temperatures for Duct 4 are shown in Fig. IV-75.

5. Stress Analysis:

a. Flanges

Flange stresses are functions of the gaskets, bolts, flange proportions, internal pressure, and temperature gradients. The stresses are a direct function of the flange moments. These moments are caused by the offset of the clamping (bolt) force, gasket reaction, and main shell.

Slight leakage was allowable so gaskets with a relatively low pre-load requirement were used.⁸² High strength bolts⁸³ were used with a minimized bolt circle. The flange hub taper yielded about equal stresses at each end.

As pressure is applied to the joint, bolt stresses decrease due to flange rotation. Transient thermal gradients cause the flanges to expand between bolt seating faces. This increases the stress in the cooler bolt. This increased bolt load increases the flange moment and causes it to rotate more, which decreases the bolt tensile stress and increases bolt bending stress.

This thermally induced mechanical load can cause bolt failure and increase the flange stress. To minimize this effect, bolts with a low elastic constant (long and thin) are preferred. The transient temperature gradients in the flange wall create high thermal stress but these self-relieve.

The flange stress analysis is based on the Taylor-Waters formula⁸⁴⁻⁸⁶ for the basic stresses. Thermally induced bolt stresses are based on a rigid flange having no rotation. This conservatively high bolt load was then used to solve for maximum flange stresses.

The maximum thermally induced bolt stress is 60,000 psi (nominally) at both joints and 90,000 psi for the extreme case. The high bolt stress condition occurs earlier in the aft joint, Station 441, ($\tau \approx 12$ minutes) than in the forward joint Station 338, ($\tau \approx 14$ minutes) because of the latter's more severe environment.

The nominal flange stresses (psi) are summarized below.

	Forward flange Station 338			Aft flange Station 441		
	Preload ¹	Pressure only	Pressure and thermal	Preload ¹	Pressure only	Pressure and thermal
Bolt stress (average)	33,000	32,000	90,000	33,000	32,000	90,000 ²
Longitudinal hub stress	6,600	22,000	28,700	6,600	15,600	25,600
Radial flange stress	1,300	4,400	5,750	1,300	3,100	5,100
Tangential flange stress	970	3,300	4,300	970	2,350	3,850

¹ Gasket seating load, 2x minimum required.

² Does not include bending stress, which is about 15,000 psi maximum.

b. Lug area

The lugs which support the grid are regarded as rigid. Each lug carries part of the support grid load as an axial load. Since this load has an eccentricity with respect to the middle surface of the duct, the resultant loading on each lug is an axial force and a pitching moment. Also, friction exists between the faces of the lugs and the mating surfaces on the support grid. Differential thermal expansion between the duct and the support grid induces a radially directed friction force on each lug face. This force is directed outward during a fast start-up, and inward during a fast shut-down of the reactor. In steady state the friction force is probably negligible.

The stresses caused by each of the above three loading mechanisms were separately calculated and then linearly superimposed.

The lug moments are assumed to be uniformly distributed circumferentially and to be carried equally by the sections of the duct lying forward and aft of the lug zone (Fig. IV-76). A stress concentration factor to account for the discreteness of the loadings is applied, however. V is a running shear reaction which enforces the requirement that there be no radial displacement. One can calculate (Ref. 41, p. 271):

$$\begin{aligned}
 M_x &= 1/2 M_o \theta \\
 V_x &= 1/2 M_o \phi/\chi \\
 N_\theta &= R. M_o/\chi^2 \xi.
 \end{aligned}
 \tag{13}$$

Where χ is the attenuation length; Θ , Ψ , Φ and ξ are the well-known cylinder loading functions; and N_θ is the circumferential load. For the present case:

$$\chi = 4 \sqrt{\frac{R^2 h^2}{3(1 - \gamma^2)}} = 2.525 \text{ inches} \quad (14)$$

$$\left. \begin{aligned} \Phi &= e^{-x/\chi} \left[\cos \frac{x}{\chi} + \sin \frac{x}{\chi} \right] \\ \Psi &= e^{-x/\chi} \left[\cos \frac{x}{\chi} - \sin \frac{x}{\chi} \right] \\ \Theta &= e^{-x/\chi} \cos \frac{x}{\chi} \\ \xi &= e^{-x/\chi} \sin \frac{x}{\chi} \end{aligned} \right\} \quad (15)$$

The stresses induced by M_x , V_x , N_θ are calculated by:

$$\left. \begin{aligned} \sigma_x^o &= \frac{6}{h^2} M_x \\ \sigma_x^i &= -\sigma_x^o \\ \sigma_\theta^o &= \frac{N_\theta}{h} - \gamma \sigma_x^o \\ \sigma_\theta^i &= \frac{N_\theta}{h} + \gamma \sigma_x^o \end{aligned} \right\} \quad (16)$$

Here R = mean radius of cylinder, h = wall thickness and the superscripts o and i designate respectively the outer and inner surface fibers of the duct wall. The stresses which are given by Eq. 16 apply to the forward section of the duct. Their counterparts in the after section have the opposite sign.

An average M_o of 1690 lb-in. is applied to each lug, based on the average axial load and an average eccentricity. To account for the discreteness of the loading, a stress concentration factor of 1.60 is applied to the inner fibre and 1.25 to the outer fibre stresses. The stress concentration factor is assumed to decay linearly to 1.0 in one attenuation length. The modified bending stresses calculated in this manner are shown in Fig. IV-77 and IV-78.

At the lugs, the axial grid load is applied discretely to the circumference of the duct. For this analysis, a sinusoidal axial loading to the hoop which contains the lugs is applied. This loading is normalized for a zero resultant and has a peak value of χ_o defined below.

For this purpose consider the cylinder "unrolled" into an "infinite" flat plate (the factor $1/h$ converts to a plate of unit thickness). Each lug carries $1/a$ of the total load F_1/h and is equal to the area under the curve for length L with the datum at $x = \chi_o$.

$$\frac{F_1}{ah} = \frac{1}{2} (2\chi_o) \left(\frac{2\pi R}{a} \right) \quad (17)$$

$$\chi_o = \frac{F_1}{2\pi R h}$$

The stresses for a sinusoidal loading of amplitude (Eq. 17) are found using the theory of elasticity. The stresses in the forward section are:

$$\left. \begin{aligned} \sigma_{xx} &= \frac{F_1}{4\pi R h} \left[\frac{1+\gamma}{2} \cdot \frac{ax}{R} + 1 \right] e^{-ax/R} \cos \frac{ay}{R} \\ \sigma_{yy} &= \frac{-F_1}{4\pi R h} \left[\frac{1+\gamma}{2} \cdot \frac{ax}{R} - \gamma \right] e^{-ax/R} \cos \frac{ay}{R} \\ \sigma_{xy} &= \frac{F_1}{8\pi R h} \left[(1+\gamma) \frac{ax}{R} + (1-2) \right] e^{-ax/R} \sin \frac{ay}{R} \end{aligned} \right\} \quad (18)$$

The stresses in the aft section are the negative of these. Substituting the numerical values:

$$\frac{a}{R} = \frac{96}{28.375} = 3.383 \quad \gamma = 0.30$$

$$\frac{F_1}{4\pi R h} = \frac{259,500}{4\pi(28.375)(0.5)} = 1500$$

into Eq. 18 and noting that $ay/R = 0$ at the center of the lug, and $ay/R = \pi/2$ at the edge of the lug, the following stresses result (along the axial center line of the lug):

$$\begin{aligned}\sigma_{xx} &= 1500 [2.195 \times 1] e^{-3.383x} \\ \sigma_{yy} &= -1500 [2.195 \times -0.30] e^{-3.383x} \\ \sigma_{xy} &= 0.\end{aligned}$$

Along the edge of a lug,

$$\begin{aligned}\sigma_{xx} &= \sigma_{yy} = 0 \\ \sigma_{xy} &= [750 \quad 4.39 \times +0.70] e^{-3.383x}\end{aligned}$$

These stresses are plotted in Fig. IV-79. Note that they attenuate rapidly away from the lugs. They come into being only because of the discreteness of the axial lug loading and must be superimposed on the mean tensile stress in the duct. For the section forward of the lugs this is:

$$\sigma_{x, i} = \frac{F_1 T}{h} = \frac{(640,500)/2\pi R}{(0.5)} = 7180 \text{ psi}.$$

Differential expansion between the lug zone and the support grid may produce a radial component of lug loading due to friction. The coefficient of friction, f , may be 0.6. Thus,

$$V_{\max} = \frac{(2\pi R F_1) f}{2\pi R} = 873 \text{ lb/in.}$$

This load, V , may be directed either inwardly or outwardly. Assume $V/2$ acts on each half of the duct in the lug region. Since the slope must vanish at $x = 0$,

$$\begin{aligned}\omega' &= 0 = -\frac{Mx}{D} + \frac{1/2 Vx^2}{2D} \\ \therefore M &= \frac{1}{4} Vx.\end{aligned}\tag{19}$$

Equation 19 gives the required moment at the junction of the two halves in the lug region. The moment, shear, and circumferential load are given as a function of x by Eq. 20.

$$\begin{aligned}M_x &= \frac{1}{4} Vx \Psi \\ V_x &= \frac{1}{2} V \Theta \\ N_\theta &= \frac{RV}{2x} \Phi\end{aligned}\tag{20}$$

From which the stresses are found to be:

$$\sigma_x^O = 24 M_x$$

$$\sigma_x^i = -\sigma_x^O$$

$$\sigma_e^O = \sigma_e^i = 2 \Phi$$

Numerical values are given in Table IV-17 for V directed outwardly.

Table IV-17. Duct outside wall stress due to radial lug load, during warm-up.

x/χ	σ_x^O	σ_θ^O (psi)
0	13,220	982
0.4	4,714	863
0.8	-123	624
1.2	-2,270	383
1.6	-2,746	192
2.0	-2,371	65
2.4	-1,694	-6
2.8	-1,027	-36
3.2	-506	-42
3.6	-164	-36
4.0	+25	-25

c. Reactor area

In the reactor area there are membrane, thermal, and discontinuity stresses. The membrane stresses are functions of the pressure loads given in Fig. IV-48, and are calculated by basic methods. The side support holes increase the circumferential membrane stresses by 6% and the longitudinal membrane stress by 11%, from the standpoint of minimum load path.

Thermal stress varies in direction during a run. During warm up ($\tau \approx 10.7$ min) a peak of about 18,600 psi is reached (tension on outside surface). In the nominal steady state condition the stress is about 23,000 psi tension on the inside surface and 13,000 psi compression on the outside surface at $x/L = 0.6$.

High discontinuity stresses are caused by the side support holes. The theoretical elastic stress concentration factor is 2.6 in the circumferential direction and zero in the longitudinal direction. A detailed discussion of stress concentration factors appears in Ref. 21.

The membrane stress distribution is shown in Fig. IV-80. Surface stress distribution (excluding thermal stress) for the forward part of the aft reactor duct is shown in Fig. IV-81 and IV-82. The effects of radial support grid loads during fast transients are included in Fig. IV-80. They are not included in Fig. IV-81 because of their lesser importance in the circumferential direction.

Combined elastic circumferential stresses (mechanical and thermal) are shown in Fig. IV-83a for steady state and a selected transient condition. There may be some yielding on the outside surface during transient warm-up conditions and on the inside surface during steady state. The magnitude of this tensile yielding is low and would not result in compressive yielding when the loads and thermal gradients are removed. Combination of mechanical and thermal stresses with appropriate stress concentration factors gives a maximum elastic stress of 113,000 psi in the edge of a hole at $x/L = 0.6$. Simple superposition can be justified only for perfect elasticity of the material. In nonlinear viscoelastic, plastic, or viscoplastic materials temperature and load stresses are interdependent and are therefore affected by both load and temperature history. For the reactor duct, yielding will not substantially affect the stress distribution; therefore, Fig. IV-83b indicates the areas of circumferential yielding in an appropriate manner. Combination of all shell stresses in the axial direction are well below the yield strength.

Maximum stress in the forward reactor duct is much lower than in Duct 5. The circumferential membrane stress is 15,000 psi, and the longitudinal flange stress is less than 28,700 psi. Stress concentrations were minimized by reinforcing most duct penetrations. Operating temperature, of course, is much lower.

6. Deflection

The lateral deflection of the duct between supports and the radial deflections caused by the reactor core load are negligible. Out-of-roundness of the duct could affect the reactor side support system and support grid

attachment. Maximum out-of-roundness of Duct 5 after fabrication was specified to be ± 0.030 in. on diameter when in the vertical position. Out-of-roundness caused by the duct support system was minimized by using thick support rings between flanges and tangentially attaching four support arms to each ring. Internal pressure greater than 80 psi tends to decrease the residual out-of-roundness of the duct in the reactor area.

Mechanical and thermal radial growth of the duct in the reactor area for steady state conditions is shown in Fig. IV-84.

Steady state differential expansion between the support grid ring and the duct is negligible. This expansion would approach 0.035 inch on the radius for a severe transient condition. (Step input of 1060°F air at 1800 pps.) A radial gap between the grid and duct of 0.060 inch was allowed so that radial constraint between them is limited to the frictional component of the support grid axial load (paragraph 5, above).

7. Testing

The preferable proof test of the reactor duct would duplicate operating cyclic conditions, loads, transient and steady state temperatures, and temperature distributions. Since this would be prohibitively expensive and difficult, a series of simpler tests was specified to verify structural analyses and reliability. Three special material tests were performed on relatively small structural specimens, as well as numerous basic materials tests. These were followed by hydrostatic tests of the completed ducts.

A test was conducted to determine the number of cycles to failure in a 1/4-inch diameter Hastelloy C specimen when subjected to a maximum bending strain 1.3 times that at the yield point. With the test conducted at 1000°F , the specimen was intact after 220,000 cycles.⁸⁷ Type 304 stainless steel pass-through glands were used to route the many thermocouple and pressure leads out of the reactor ducts. The differential thermal expansion between these items theoretically would tend to loosen them, so the effect of thermal cycling on the glands (assembled in a Hastelloy C test section) was checked.⁸⁸ Gland specimens 1/2 - 14 NPT were torqued to various values and cycled 5 times for 10 minutes at 1000°F each cycle. Specimens pre-torqued to 60 in.-lb loosened after two cycles. Specimens

pre-torqued to 70 ft-lb (840 in.-lb) required 45 to 90 ft-lb break-away torque after completing 5 cycles.

Most of the reactor duct is basically in a biaxial stress condition with stress concentrations due to the instrumentation and side support penetrations. Because of the combination of cyclic temperature, thermal stress, and pressure stress conditions, along with the work hardening and aging characteristics of Hastelloy C, the possibility of crack formation in the region of the stress raisers was suspected. A test was devised to indicate crack formation and propagation characteristics. This test indicated that cracks might form but would not propagate rapidly. Ultimate strength of the cracked specimen was 80% of the minimum specified for normal material, however it was still 2.2 times greater than the maximum duct membrane stress. The tests used a flat, wide, (Hastelloy C, 1/4 inch thick by 2.75 inch wide) uniaxial test specimen with a 3/8 inch diameter hole in the center. This specimen accurately simulated the geometry very near the duct holes. Crack formation and propagation characteristics could easily be observed. The specimens had sharp corners at the hole surfaces and a stress concentration factor of 2.75, as opposed to chamfered corners and a stress concentration factor of 2.6 in the actual duct.

The specimen was first heated in air to test temperature in about 30 minutes with an induction coil. One load cycle consisted of applying a tensile load at 3000 lb/min, holding at 3000 lb for 5 minutes, and then removing the load for 30 seconds. The load cycles were repeated as many as 40 times.

At the end of the cyclic tests some specimens were heated to the test temperature and loaded at 6000 lb/min to failure. The results are given in Table IV-18.

Finally, both reactor duct sections were hydrostatically proof-tested. Duct 4 was tested once and Duct 5 on three separate occasions. Duct 5 was tested prior to drilling the holes through the shell, after final machining, and in conjunction with the support grid test. Details of the tests are covered in Refs. 34, 36, and 89 through 91.

Stresscoat and strain gages were mounted in critical areas of the duct to obtain quantitative information. The circumferential-to-axial load ratio was simulated by applying a compressive axial mechanical load with hydraulic jacks. (Note: The reactor duct is not a closed pressure vessel.) The

Table IV-18. Crack formation and propagation test results.

Specimen No.	Temp (°F)	Cyclic load ¹ (ksi)	No. of cycles	% Elongation ²	Rupture load ¹ (ksi)	Results
1	1500	25	40*	1	-	No cracks
2	1500	30	20*	14	-	See note 3
3	1400	25	40*	3	48.8	See note 4
4 ⁵	1400	30	40*	3	-	No cracks
		20	20	-	-	No cracks
5 ⁵	1400	25	40*	2	-	No cracks
	1400	20	20	-	78.4	No cracks
6	1350	43	30	-	-	See note 3

Notes:

* During first 20 cycles both load and temperature were cycled.

¹ Load = average stress at minimum area of basic specimen.

² Approximate elongation measured at room temperature at the end of the first 20 cycles for 1-inch gage length containing the hole.

³ Tests were stopped because of extreme hole out-of-roundness. No other defects were found.

⁴ About 15 cracks were found at the edge of the hole after the first 20 cycles. These cracks were about 0.010 inches long. After 40 cycles these cracks had doubled in length and width and extended about 0.040 inches deep along the hole surface (see Fig. IV-85). Additional smaller cracks were observed on the opposite side. This specimen was then loaded to failure at 6000 lb/min and 1400° F. It failed in a ductile manner at 28,100 lb (48,800 psi).

⁵ Surfaces were sandblasted to about 300 rms.

resultant axial test loads were about 4% lower than the design loads for Duct 4 and the forward joint of Duct 5, and were about 12% high for the aft joint of Duct 5. Since the hydrostatic test was performed at room temperature, a high test pressure (about 600 psi) was used. Considering the strength ratio at room temperature and operating temperature (about 1.5), this pressure was equivalent to about 130% of the operating pressure. Bolt preloads used during the tests (25,000 to 26,000 lb) were much higher than those planned for reactor operation, (about 10,000 lb). The test preloads were approximately equal to the theoretical combined mechanical and thermal operating loads (27,000 lb); therefore, the tests were good indicators of actual operating loads.

Some results for a few points of particular interest are shown in Figs. IV-86 through IV-94. These results were in good agreement with the analyses, including the complex stress distribution in the lug area.

8. Reliability

Thirty-four thermocouples were installed at various positions on the aft reactor duct to insure its thermal reliability during maximum power runs. Table IV-19 lists the limit thermocouple reading for nominal loads (Fig. IV-48) based on a 1.5 factor of safety and actual minimum yield strength data. As the table indicates, limit temperatures are 200° F to 250° F higher than expected nominal values, and 50° F to 125° F higher than extreme values.

The reactor duct has three areas of high indicated stresses: longitudinal flange hub stress, lug fillet stress, and shell hole tangential stress. The ducts were fabricated to their most ductile condition to insure that these concentrated stresses would not cause failure. When the range radial and tangential stresses have nominal factors of safety it is reasonable to allow the longitudinal hub stresses to approach the yield or creep strength of a ductile material. The ASME U. P. V. C. allows this hub stress to approach the specified minimum yield strength. If an overload or extreme thermal gradient in the flanges induces high loads, the flange and bolts may yield slightly. This does not cause a gross failure but may affect remote disassembly or assembly.

The stress raiser cyclic test verified that the shell hole tangential stress was acceptable.

Table IV-20 indicates factors of safety based on nominal temperatures, nominal loads, 0.2% offset yield strength, and membrane stresses, unless otherwise stated.

Table IV-19. Temperature limits for Duct 5 thermocouples.

Parameter	Parameter description	Station or x/L location	Limit temp °F
05503	ODWO wiper bar	0.57	1550
05504	ODWO rail	0.57	1550
05508	ODWO wiper bar	0.78	1550
05510	ODWO wiper bar	0.78	1550
05511	ODWO rail	0.78	1550
05512	ODWO rail	0.78	1550
05521	IDW	359	1500
16501	on rail nut	0.57	1550
16504	on rail nut	0.78	1550
16502	on rail nut	0.57	1550
16503	on rail nut	0.78	1550
05501	ODWO wiper	0.3	1500
05502	ODWO wiper	0.3	1500
05505	ODWO wiper bar	0.57	1550
05506	ODWO rail	0.57	1550
05507	ODWO wiper	0.78	1550
05509	ODWO wiper	0.78	1550
05513	ODWO wiper bar	0.94	1550
05514	ODWO wiper bar	0.94	1550
05522	IDW	359	1500
05940	IDW	435	1500
05519	IDW	359	1500
05520	IDW	0.99	1500
05941	IDW @ flange	440	1500
05942	IDW @ flange	440	1500
05943	IDW @ flange	440	1500
05932	IDW	431	1500
05933	IDW	431	1500
05934	IDW	431	1500
05935	IDW	435	1500

Table IV-19 (Continued).

Parameter	Parameter description	Station or x/L location	Limit temp °F
05936	IDW	435	1500
05937	IDW	435	1500
05938	IDW	435	1500
05939	IDW	435	1500

ODWO Outside duct wall opposite.

IDW Inside duct wall.

Table IV-20. Factors of safety.

Location	Factor of safety		
	Actual material test data	Minimum specified strength	Verified in hydrostatic test
Duct 4 shell	2.2	2.1	1.3
Duct 5 Sta. 360	1.7	1.6	1.3
Duct 5 Sta. 364	1.5	1.4	1.3
Duct 5 Sta. 400	1.5	1.4	1.3
Duct 5 Sta. 422	2.1	2.0	1.9
Duct 5 Sta. 431	3.2	3.0	1.9
Duct 5 forward flange*	2.1	1.8	1.3
Duct 5 aft flange*	2.0	2.0	1.9

* Stress based on 1/2 (longitudinal hub stress + radial flange stress) and material test data of actual forging.

In the reactor area, creep of the duct can affect the reliability of the reactor side support system. To insure the reliability of the reactor, creep of the duct should be limited to 0.5% total elongation. Creep will be most noticeable in the area near Sta. 400 and will be negligible in other areas. For nominal conditions (1300° F average temperature through wall) the life limit would be 10 hours. Life would decrease to 4.5 hours for an extreme case (1400° F average temperature through wall).

Nuclear radiation will not decrease the structural reliability of the reactor duct sections. Detectable changes in the mechanical properties of metals do not appear until the integrated fast neutron flux reaches a level of about 10^{19} neutrons/cm².

The expected maximum fast neutron dose at full power for the duct is about 3×10^{17} neutrons/cm² hr. For damage to be observed, it would take 30 hours of operation at full power.

The primary effect would be an increase in yield strength and a decrease in ductility as indicated below.⁹²

Property	Irradiation	
	none	4×10^{19} fast neutrons/cm ²
Tensile strength	138 ksi	142 ksi
% elongation	20	12
Brinell hardness	209	209

9. Fabrication

Hastelloy C has been produced for about 30 years. Most applications have been in the chemical, piping, and pressure vessel fields. Fabrication is less difficult than for other nickel and cobalt base high temperature alloys. Yet, to insure the highest quality, considerable fabrication development work was done with respect to the reactor duct. This development mainly consisted of materials certification, selection of weld procedures, and attempts to upgrade the minimum strength while retaining maximum ductility.

Tensile strength and ductility data for Hastelloy C is scattered over a wide band. (Fig. IV-52 and IV-53.) Attempts to increase the yield strength and maintain high ductility after 1 hour at 1500° F were not successful;

therefore, weldments were heat treated to yield the most stable and ductile properties.

Hastelloy C can be welded by many methods but SIGMA and TIG methods yield the best quality welds. Table IV-21 indicates two different SIGMA procedures. For use in the as-welded condition, the low heat method is preferred. If the component can be solution heat treated the high heat procedure can be used. The latter method was used with the following sequence.

- 1) Weld groove prepared by machining (no grinding allowed).
- 2) Tack weld, then root pass on opposite side.
- 3) Penetrant inspection.
- 4) Add filler passes on first side.
- 5) Back machine weld groove (no grinding).
- 6) Penetrant and radiographic inspection.
- 7) Apply root pass on second side.
- 8) Penetrant inspect.
- 9) Finish welding.
- 10) Penetrant and radiographic inspection (also after machining).

Specimens from each heat of plate and each forging ring were chemically analyzed and tensile tested at room temperature and at 1100° F.⁶⁹ Specimens were tested in the as-received and solution heat treated and water quenched condition. Weld wire was tested by taking a sample from each spool (about 10 pounds from each spool) and fabricating an across-weld bend specimen. The specimen was solution heat treated, water quenched, and aged for 1 hour at 1500° F. The weld bead was required to withstand a 90° bend without cracking. All material was thoroughly inspected by ultrasonic and dye check techniques.

Flanges were produced from vacuum arc remelted Hastelloy C ring-rolled forgings. The flange size required a very large ingot, and vacuum remelt was used to keep the impurities at a minimum. Flanges in the as-received condition met minimum specified strength and were ductile (about 50% elongation). After subsequent solution heat treatment and water quenching many of them failed (by 6%) to meet the 1100° F minimum specified yield strength, but ductility was increased (60% to 70% elongation). The higher strength flanges were used in the reactor duct sections.

Table IV-21. Welding procedures.

<u>High Heat</u>						
<u>Root Pass</u>						
Current	.	.	.	.	.	290 amps
Voltage	.	.	.	.	.	33 volts
Wire feed	.	.	.	.	.	1000 in./min
Back-up gas	.	.	.	.	.	20 ft ³ /hr argon
First pass ground from opposite side and die checked. The second pass, welded on the opposite side, also considered a root pass, so the above conditions apply.						
The remainder of the welding performed under the following conditions:						
Current	.	.	.	.	.	220 amps
Voltage	.	.	.	.	.	32 volts
Wire feed	.	.	.	.	.	800 in./min
The following conditions apply on all passes:						
Interpass temperature	.	.	.	.	.	300° F max
Shielding gas	.	.	.	.	.	50 ft ³ /hr helium 25 ft ³ /hr argon
Trail gas	.	.	.	.	.	25 ft ³ /hr argon
Travel speed	.	.	.	.	.	33 in./min filler passes 20 in./min root pass
<u>Low Heat</u>						
The following conditions apply to all passes:						
Wire feed	.	.	.	.	.	550 in./min
*Travel speed	.	.	.	.	.	35 in./min
Arc voltage	.	.	.	.	.	31 volts
Current	.	.	.	.	.	165 to 180 amps/dc
Polarity	.	.	.	.	.	Reverse
Base rod length	.	.	.	.	.	1/2 inch
Cup to work distance	.	.	.	.	.	5/16 inch
Shielding gas	.	.	.	.	.	75 ft ³ /hr helium 25 ft ³ /hr argon
Trail gas	.	.	.	.	.	25 ft ³ /hr argon
Interpass temperature	.	.	.	.	.	300° F max
Power supply	.	.	.	.	.	Flat slope

*Travel speed = 20 in./min for root passes.

Radiographic inspection complied with the ASME Boiler and Pressure Vessel code, Sect. VIII, for unfired pressure vessels, except that Sections m2 and m3 were changed as follows:

Any type of crack or zone of incomplete fusion or penetration shall be unacceptable.

Porosity in excess of that shown as acceptable by the standards given in Appendix IV of Section VIII of the ASME Boiler and Pressure Vessel code for Unfired Pressure Vessels shall be judged unacceptable.

The general procedure used in fabrication of the duct was:

- 1) Roll and weld plates and flanges.
- 2) Inspect.
- 3) Solution heat treat at $2220^{\circ}\text{F} \pm 25^{\circ}\text{F}$ for 2 hours, then water quench.
- 4) Inspect.
- 5) Finish machine.
- 6) Inspect (radiographic, penetrant, and all critical dimensions).
- 7) Hydrostatic proof test.
- 8) Penetrant inspection.

Mating of the support grid to the aft reactor duct required very close control. Uniform contact was required to minimize the lug stresses. Also, good contact between lugs prior to match machining the twelve grid pins was required so the pins would not be over-stressed. The general procedure^{36, 93} used to obtain uniform contact was:

- 1) Apply Prussian blue to duct lugs.
- 2) Lower support grid into upright aft reactor duct.
- 3) Center the support grid in the duct.
- 4) Apply 4-ton axial load.
- 5) Remove grid and check contact.
- 6) Apply plastic gage on lugs not indicating contact.
- 7) Repeat 2nd through 5th step.
- 8) Grind lug surface of support grid as indicated by plastic gages.
- 9) Repeat procedure to verify uniform contact.
- 10) Match machine grid pin holes.

Remaining assembly procedures are discussed in subsequent sections.

Fig. IV-1. Typical section of the side support system.

Fig. IV-2. Corrugated spring dovetail.

NOTES:

1. p_o IS THE SAME AS THE INTERACTION PRESSURE BETWEEN SHIMS AND TUBES AT $R = R_c$.
2. THE SHIMS DO NOT CONTACT CIRCUMFERENTIALLY.
3. IN THIS FIGURE, THE LATERAL INERTIA LOADS ARE ASSUMED TO BE ZERO, AS THOUGH THE ASSEMBLY WERE QUIESCENT WITH A CENTERLINE OF THE CYLINDER VERTICAL WITH RESPECT TO THE EARTH'S SURFACE.

Fig. IV-3. Corrugated spring clamping action.

Fig. IV-4. Wedging action of core.

Fig. IV-5. Sliding action of core.

THIS FIGURE ILLUSTRATES
THE CASE OF

$$p_o = n\gamma R$$

$$\text{or } n = \frac{p_o}{\gamma R}$$

RADIAL PRESSURE IN
THIS BOTTOM REGION
MAY BE SLIGHTLY INCREASED
AS A RESULT OF WEDGING SHOWN
IN FIG. IV-4.

Fig. IV-6. Pressure distribution.

Fig. IV-7. Stable matrix ($\eta = 35.5$ $\Psi = 1.40$).

Fig. IV-8. Type 1 instability ($\eta = 35.5$ $\Psi = 1.22$).

Fig. IV-9. Type 2 instability ($\eta = 35.5$ $\Psi = 1.05$).

Fig. IV-10. Type 3 instability ($\eta = 35.5$ $\Psi = 0.77$).

Fig. IV-11. Type 5 instability ($\eta = 0.125$ $\Psi = 0$).

Fig. IV-12. Slip plane with wedging.

Fig. IV-13. Side pressure (Ψ) vs suspension ratio (η) for various degrees of stability.

Fig. IV-14. Slice test assembly.

* DURING SHUT-DOWN, IN A GROUND TEST, THE CORE PRESSURE WOULD REACH LOWER MINIMA.

Fig. IV-15. Core pressure at various spring conditions for various spring stiffnesses.

Fig. IV-16. Soderberg diagram.

Fig. IV-17. Fueled tube, duct temperature operating envelope.

Fig. IV-18. Fabrication steps for corrugated springs.

Fig. IV-19. Spring radii as functions of axial location.

Fig. IV-20. Core pressures as functions of axial location.

Fig. IV-21. Duct and core annulus regions.

Fig. IV-22. Spring action.

Fig. IV-23. Spring configurations.

Fig. IV-24. Spring stress distribution as a function of annulus spacing.

Fig. IV-25. Spring stress distribution as a function of pressure drop.

Fig. IV-26. Results of compression stress tests.

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Fig. IV-27. Results of pressure drop stress tests.

Fig. IV-28. Stress components due to pressure drop and compression.

Fig. IV-29. Flow passage cross section for test.

Fig. IV-30. Test instrumentation scheme.

Fig. IV-31. Pressure drop as a function of Mach number.

Fig. IV-32. Pressure drop as a function of axial position.

Fig. IV-33. Loss parameter for rail and wiper bar springs.

Fig. IV-34. Loss coefficients for wing wipers.

Fig. IV-35. Annulus area model.

Fig. IV-36. Annulus height as a function of axial position.

Fig. IV-37. Annulus pressure gradients.

$$x'/L = 0 \text{ (a STA. 361.312)}$$

$$L = 63.075$$

$$D = 3.188 \text{ (a STRUCT. LIMIT CONDITIONS)}$$

Fig. IV-38. Wiper dimensions.

6-6 11-3 14

Fig. IV-39. Wing wiper pressure drop at structure limit condition.

Fig. IV-40. Bar wiper pressure drop at structure limit condition.

Fig. IV-41. Rail wiper pressure drop at structure limit condition.

Fig. IV-42. Shroud assembly.

Fig. IV-43. Shroud, spring, and skirt stresses as functions of pressure drop and yield strength.

Fig. IV-44. Aft reactor duct (Duct 5).

Fig. IV-45. Reactor control and poison systems (Duct 4).

Fig IV-46. Forward reactor duct (Duct 4)

Fig IV-47. Section view of reactor ducts 4 and 5.

Fig. IV-48. Aft reactor duct load distribution, reactor structural design condition.

Fig. IV-49. Preliminary estimate of duct material requirements.

Fig. IV-50. Material requirements and average yield strengths (0.2% offset) of several alloys.

Fig. IV-51. Material requirements and average 10-hr rupture strengths of several alloys.

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Fig. IV-61. Flow in the side support annulus.

Fig. IV-62. Approximate external heat transfer film coefficient for duct, at reactor structural design condition.

BASED ON PRELIMINARY ESTIMATES:

$$h_i \text{ (INSIDE SURFACE)} = 285 \text{ Btu/m ft}^2 \text{ } ^\circ\text{F}$$

$$t_i \text{ (INTERNAL AIR TEMP)} = 1110^\circ\text{F}$$

$$\bar{q} = 0.75 \text{ mW/ft}^3$$

$$t_o \text{ (OUTSIDE ATM TEMP)} = 100^\circ\text{F}$$

Fig. IV-63. Maximum temperatures (Duct 5, Sta. 396) vs external emissivity (no external convection).

Fig. IV-64. Duct temperature just forward of the core vs external emissivity for various heating rates and expected internal film coefficients.

Fig. IV-65. Duct wall heat transfer nomenclature.

Fig. IV-66. Typical reactor run, reactor structural design condition.

Fig. IV-67. Temperature distribution in Duct 5 aft flange (end of power plateau, nominal case).

Fig. IV-68. Transient temperatures, Duct 5, Sta. 339, forward flange.

Fig. IV-69. Transient temperatures, Duct 5, Sta. 440, aft flange.

Fig. IV-70. Transient temperatures, Duct 5, Sta. 436, aft flange hub.

Fig. IV-71. Section through Duct 5, Sta. 351.5.

Fig. IV-72. Steady state Duct 5 wall temperature.

Fig. IV-73. Steady state temperature distribution through Duct 5 wall at Sta. 399 ($x/L = 0.6$).

Fig. IV-74. Transient temperature, Duct 5, Sta. 399 and 430.

Fig. IV-75. Duct 4, Sta. 288 and 328, transient temperatures.

Fig. IV-76. Lug moments, forward and aft of lug zone.

Fig. IV-77. Stresses in Duct 5, caused by grid load moments (inside surface).

Fig. IV-78. Stresses in Duct 5, caused by grid load moments (outside surface).

Fig. IV-79. Membrane stress caused by axial lug loading.

Fig. IV-80. Duct 5 membrane stress distribution.

Fig. IV-81. Longitudinal stress distribution in forward part of Duct 5.

Fig. IV-82. Circumferential stress distribution in forward part of Duct 5.

Fig. IV-83. Combined elastic circumferential stresses and approximate form of plastic strain around hole at $x/L = 0.6$.

Fig. IV-84. Mechanical and thermal radial expansion of Duct 5, reactor structural design condition.

Fig. IV-85. Stress raiser cyclic test; specimen 3 after 40 cycles.

Fig. IV-86. Duct 4 structural test; stress vs internal pressure, gages 17-18 and 19-20. (Bolt preload = 25,300 lb.)

Fig. IV-87. Duct 5 structural test; stress vs internal pressure, gages 19-20. (Bolt preload = 25,890 lb.)

Fig. IV-88. Duct 5 structural test; stress vs internal pressure, gages 20-21. (Bolt preload = 25,890 lb.)

Fig. IV-89. Duct 5 structural test; stress vs internal pressure, gages 25-26. (Bolt preload = 25,890 lb.)

Fig. IV-90. Duct 5 structural test; stress vs internal pressure, gages 23-24.

Fig. IV-91. Duct 5 and support grid assembly test; strain vs pressure, gages 15 and 16.

Fig. IV-92. Duct 5 and support grid assembly test; strain vs pressure, gages 21 and 22.

Fig. IV-93. Duct 5 and support grid assembly test; strain vs pressure, gages 27 and 28.

Fig. IV-94. Duct 5 and support grid assembly test; strain vs pressure, gages 46 and 47.

V. REACTOR CONTROL

A. Control Rods

1. Design Description

The control rod assembly consists of the control rod and the safety strap. The control rod (Fig. V-1) is made from six pieces of $0.040 \times 0.777 \times 63$ -inch hafnium sheet, bent and spot welded together to form an asterisk-shaped section about 1 inch in diameter. At one end of the rod the fins of the asterisk are tapered to fit a truncated cone in the slider to which it is attached. At the other end of the control rod the fins are capped by semi-circular sections. This gives the control rod a snug fit in the tie rod, to reduce the amplitude of possible vibrations.

The safety strap (Fig. V-2) is made from a piece of $0.050 \times 0.375 \times 62.3$ -inch Rene' 41 sheet welded to a 0.313 o. d. $\times 0.054$ wall $\times 4.562$ inch Rene' 41 tube. The tubular end of the safety strap is threaded to a nut which is held in the slider. The sheet end of the safety strap has a hole to facilitate remote disassembly. The purpose of the safety strap is to prevent the control rod from blowing through the core in the event of mechanical failure in the control rod.

Fourteen control rods are positioned in the core as shown in Fig. V-3. Twelve of the control rods, termed shim rods, are used for coarse adjustment of reactor power. Two control rods are vernier rods for fine adjustment of reactor power. All the shim rod assemblies have about the same worth. The worths of both vernier rod assemblies are the same, but less than the shim rods since the vernier rods are farther (radially) from the center of the reactor.

2. Material Selection

The control rod material must be a good neutron absorber, withstand the high reactor temperature and oxidizing environment without loss of functional ability, be readily fabricable, and be available at reasonable cost. Several neutron absorbers, including boron compounds, iridium, hafnium, and rare earth oxides were considered. The boron compounds were rejected because the (n, α) absorption reaction causes high internal

heating rates in the control rods. Iridium was rejected because of fabrication difficulties. The rare earth oxides were rejected because of lack of fabrication experience. Hence, by elimination, hafnium was selected as the control material for the Tory II-C reactor.

The safety strap material must be readily fabricable and must withstand high reactor temperatures without loss of functional ability. Rene' 41, Hastelloy R-235, and Inconel X were considered as materials for the safety strap. Rene' 41 was selected because of its superior strength at elevated temperatures.

3. Material Properties

a. Physical and mechanical

Hafnium has a melting point of 4032°F and a boiling point of 5600°F . Below the allotropic transformation temperatures of 3200°F , hafnium has a hexagonal close-packed crystal structure and a density of 0.474 pounds per cubic inch. Above 3200°F the crystal structure of hafnium is body centered cubic. Hafnium is an excellent neutron absorber, with a thermal absorption cross-section of 105 barns and a resonance integral of 2000 barns. Specific heat and thermal conductivity of hafnium as a function of temperature are given in Refs. 94 and 95 respectively. Data from these references has been plotted in Fig. V-4.

Figures V-5, V-6, and V-7 show the mechanical properties of hafnium as a function of temperature. These properties were determined for material purchased under LRL specifications.⁹⁶

Specimens from hafnium sheet purchased for control rod fabrication were oxidized in air at various temperatures to 1800°F for 1 to 3 hours. The specimens were tensile tested at either ambient or elevated temperatures, using the apparatus and techniques described in Ref. 97.

The room temperature yield and ultimate strengths and the total elongation of the test specimens as a function of oxidizing temperature for oxidizing times of 1 and 3 hours are shown in Fig. V-8. The yield and ultimate strengths and the total elongation of the test specimens oxidized at the test temperature for 3 hours as a function of temperature are shown in Fig. V-5.

From Figs. V-5 and V-8 we may conclude that:

The room temperature ultimate tensile strength remains nearly constant for specimens oxidized at temperatures to 1800° F for either 1 or 3 hours.

The room temperature ductility decreases rapidly for oxidizing temperatures above 1500° F for oxidizing times greater than 1 hour.

The room temperature yield strength remains nearly constant for specimens oxidized at temperatures to 1800° F for either 1 or 3 hours.

Oxidation of hafnium for 3 hours does not markedly affect the high temperature properties, except for a decrease in ductility at temperatures above 1700° F.

b. Oxidation

Dynamic oxidation tests were performed by flowing air heated directly with a hydrogen burner through 1/2-inch-thick hafnium specimens. These specimens have the geometry shown in Fig. V-9.

The specimens were weighed before the test. They were then preheated to a steady state temperature of about 800° F. The hydrogen burner was ignited and the air brought up to test temperature, pressure, and Mach number through the specimen. Excess oxygen was added to maintain the same weight percent of oxygen in the combustion products as in air. The specimens were held at test condition for 45 minutes and then cooled down using ambient air. A specimen was removed from the test assembly and replaced with an unused specimen. The test cycle was then repeated until one specimen had spent 225 minutes in the test environment. The test temperature was, in general, either 1500 or 1700° F. One test was performed at 3000° F. No autoignition of the specimen resulted. Air flow through the specimens was at Mach 0.335 and 300 psia.

The weight gain, w , as a function of time, t , for specimens with as-received surfaces is given by the relations*

*The appropriate units are grams, minutes, and inches.

$$w = 0.0090 t^{0.45} @ 1700^{\circ} \text{F}$$

$$w = 0.0059 t^{0.36} @ 1500^{\circ} \text{F}$$

The decrease in hole diameter Δd caused by hafnium oxide buildup at 1700°F can be fairly well represented by the relation*

$$\Delta d = 4.49 \times 10^{-6} t^{0.968}$$

Static oxidation tests of hafnium were also performed. Small (about 1 gram, 1 to 2 cm²) samples of hafnium, containing less than 3% zirconium, were heated in static air at 1200 to 2200^oF, for periods up to 24 hours. Measurements were taken manually at intervals on an analytical balance having one suspension extending into a platinum-wound furnace. The furnace contained a clean, dense alumina muffle. Pretreatment of the samples was varied as indicated by the designations below:

R = cleaned with an organic solvent - "as-received"

A = annealed at 1500^oF at $< 10^{-4}$ mm Hg

P = pickled in an HF-HNO₃-H₂O solution

AP = annealed, followed by pickling.

Figure V-10 shows the test results. Some differences in results due to pretreatment were observed. No weight gain was obtained on an as-received sample heated at 1200^oF for a period of 24 hours. All samples heated at higher temperatures indicated an initial linear oxidation rate until a layer, corresponding to 34 microns thick, was obtained. Beyond this thickness an essentially parabolic law was followed. As-received samples cleaned only in acetone or hexane, showed anomalous behavior in that higher oxidation rates were observed at lower temperatures. Part of this behavior may be ascribed to residual strain on the sheet from which the samples were taken. Samples annealed at 1650^oF in vacuum and pickled to remove any surface layer, showed normal oxidation behavior. These latter samples are believed to represent the behavior of the metal in air.

Gadd and Evans⁹⁸ tested hafnium specimens machined to $3/4 \times 3/4 \times 0.060$ from 0.080 inch sheet which had been hot rolled at 800^oC and annealed at 800^oC for 1500 minutes. Their results were not entirely consistent with the findings at LRL.

*The appropriate units are grams, minutes, and inches.

In another series of tests, specimens were heated to various temperatures for fixed times. The specimens were pickled, placed in a furnace with a muffle, and brought to a specified temperature of 1500 or 1800° F at atmospheric pressure or 200 psi. The weight gain per unit surface area and superficial hardness were determined for each specimen after exposure. Bend tests were also run on each sample. Several of the specimens were placed in Hastelloy R-235 or Rene' 41 rings to determine the compatibility of these superalloys with hafnium.

The results of these tests are shown in Table V-1.

Superficial hardness varied in a random manner. There was no trend with respect to temperature, pressure, or holder material. The weight gain per unit surface area increased rapidly above 1700° F. This is shown in Fig. V-11, in which the limited data indicates an increase in oxidation rate with increased pressure. There was no marked increase in oxidation rate by using Hastelloy R-235 and Rene' 41 holders over using alumina holders at 1500° F. However, above 1700° the metallic holders seemed to increase the oxidation rate over the alumina holder.

The data spread at 1800° F may be due to the difference in surface preparation of specimens 13 and 14. Errors are due to edge effects and the spalling of oxide from the specimen before the final weighing. The weight gain is so small because of the short test times that the sensitivity of the weighing apparatus becomes a factor.

The bend test caused failure in the specimens which had been heated above 1700° F. Only one specimen heated to 1700° F or less failed in the bend test. The remainder of the specimens were bent to an included angle just under 90° without failure.

c. Radiation damage

Sufficient irradiation of hafnium causes an increase in room temperature tensile strength and a decrease in ductility. If, however, the irradiation is performed at a sufficiently high temperature, much of the radiation damage is annealed out. For irradiation to 3.7×10^{21} nvt, the annealing temperature is about 1200° F.⁹⁹

Table V-1. Hafnium oxidation test results.

Specimen	Test temp (° F)	Test environment and facility	Total wt gain (g)	Surface area (in ²)	Unit wt gain (mg/cm ²)	Superficial hardness	Bend breaking load	Bend angle (deg)
1	NT	NT	NT	NT	NT	88.2	DBN	84
2	1500	1 atm	0.0032	0.683	0.73	86.0	DNB	88.5
3	1500	200 psi	0.006	0.609	1.53	88.5	DNB	86.5
4	1600	1 atm	0.0054	0.736	1.21	88.2	DNB	88
5	1700	1 atm	0.0079	0.700	1.75	87.8	DNB	88.5
6	1500	200 psi R-235	0.005	0.594	1.31	88.0	DNB	85
7	1750	200 psi R-235	0.054	0.753	11.2	76.9	8.4	173
8	1500	200 psi Rene' 41	0.0042	0.746	0.87	88.2	DNB	87
9	1750	200 psi Rene' 41	0.0208	0.627	5.14	91.7	21.0	167.5
10	1500	200 psi	0.0053	0.630	1.30	87.5	DNB	86.5
11	1500	200 psi	0.006	0.612	1.52	89.0	DNB	86.5
12	1500	1 atm	0.006	0.811	1.22	88.0	132.8	108
13	1800	1 atm	0.0441		9.80	85.0	27.6	173.5
14	1800	1 atm polished	0.0227		5.15	93.0	24.4	175

NT = Not tested.

DNB = Did not break.

All test times are for 3 hours.

4. Temperature

The temperature of the control rod is important because of its effect on strength, oxidation rate, elasticity, and friction coefficient. Control rod temperature must be analyzed in terms of the thermodynamics of the cooling airflow. The internal heating of the control rod results from neutron absorption and gamma ray attenuation. The rod is cooled by airflow along its surface. As a result, there are both transverse and axial temperature gradients in the control rod material, and a strong axial gradient in the cooling airstream. In addition, the internal heating of the rods varies with their control position in the reactor. For conservative temperature analysis, the control rod is assumed to be fully inserted into the reactor operating at full power.

Figure V-12 shows the nuclear heating distribution of the minimum reactivity control rod normalized to core power density as a function of axial position. Using an average core power density of 9.55 mW/ft^3 , the maximum power density in the control rod is 29.0 mW/ft^3 . The nuclear heating in the control rod comes from three sources which contribute as follows:

Gamma from the core	41.7%
Internal (n, γ) source	52.0%
Tie rod (n, γ) source	6.3%

Since the axial temperature variation per unit length of the rod is small compared to the temperature gradient across the blade thickness, axial conduction is neglected. The temperature distribution in the control rod is then given by:

$$T(x) = \frac{q}{2K} (\ell^2 - x^2) + T_a$$

where:

q is the internal heating rate

K is the thermal conductivity

ℓ is the half-thickness of the blade

x is the distance from the center of the blade

T_a is the surface temperature of the blade.

The maximum temperature in the control rod is located at the adiabatic wall between the sheet section which forms the blades. Thus, at the tip of

the fully inserted control rod, with appropriate numerical values, the rise above surface temperature in the blade is 45° F.

The axial variation of the surface temperature of the control rod is shown in Fig. V-13 (taken from Ref. 2). The maximum surface temperatures of the control and safety strap are, respectively, 1790° F and 1440° F. The surface temperature of the safety strap is essentially that given by the curve marked T° (inside control rod). Thus, the maximum internal temperature of the control rod is 1835° F.

The transient temperature of the tip of the fully inserted control rod may be found from the heat balance

$$\tau \frac{dT}{dt} = \frac{P(t)}{hA} - \left[\frac{\dot{w}(t)}{\dot{w}_0} \right]^{0.8} \left[T - T_{go}(t) \right]$$

where: τ = time constant
 h = film coefficient
 A = heat transfer area
 $P(t)$ = time dependent reactor power
 $\dot{w}(t)$ = time dependent reactor flow rate
 T = control rod temperature
 $T_{go}(t)$ = time dependent inlet gas temperature
 $\dot{w}_0$ = steady state reactor flow rate.

A typical reactor run with respect to $P(t)$, $\dot{w}(t)$, and $T_{go}(t)$ is shown in Fig. V-14. The temperature of the tip of a fully inserted control rod as a function of time for the specified $P(t)$, $\dot{w}(t)$, and $T_{go}(t)$ is also shown in the figure.

5. Force Analysis

There are four major forces that act on the control rod during reactor operation. The greatest force (about 600 pounds) is due to deceleration of the 6-pound control rod during a scram. A possible deceleration for vernier rods is 100 g's and for shim rods 50 g's. There is also a bending force due to reactor barreling from the nonuniform temperature profile and side displacement of the reactor resulting from maneuver loads. The barreling of the reactor bends the control tie rod and, therefore, the inserted control rod. The control rod extends about to the middle of the reactor where the barreling effect is greatest, placing the greatest

bending stress of about 3370 psi at the rod attachment. Another force results from viscous drag from the high airflow rate along the rod. This produces about 279 pounds at the attachment. Finally, the pressure profile along the rod of 200 psi at the downstream end to 350 psi at the upstream end produces a force of 26 pounds on the attachment end section of the rod.

The reactor side support design minimizes the amount of lateral deflection which results from maneuver and gust loads. The estimated maximum deflection for this support system is 0.060 inch. The maximum axial temperature gradient (producing the greatest barreling of the reactor) is present when the reactor operates at peak wall temperature with 600° F inlet air. The control tie rod most severely affected is located 17.25 inches from the center of the reactor. The cross-sectional moment of inertia of the control rod is 0.01 in⁴.

The loads on the control rod due to tie rod bending caused by side displacement and axially varying radial thermal expansion of the reactor are shown in Fig. V-15. To simplify analysis, the radial displacement of the tie rod as a function of position in the reactor was assumed to be linear. Also, it was determined that for the side displacements and expansions expected, the control rod was constrained by the tie rod only at the leading end.

Two leading end geometries were considered:

- 1) Snug End. This configuration constrains the centroid of the leading end of the control rod to follow the centroid of the tie rod.
- 2) Loose End. This configuration allows the side of the leading end to follow the internal wall of the tie rod.

The deflection, y , of the control rod as a function of position in the core and position along its length is given by Eq. 1.

$$y = \left(\frac{x}{L}\right)^2 \left\{ \left[3 - \left(\frac{x}{L}\right) \right] \frac{y_0}{2} - \frac{WL^3}{48EI} \left[2\left(\frac{x}{L}\right)^2 - 5\left(\frac{x}{L}\right) + 3 \right] \right\} \quad (1)$$

Where the nomenclature is defined in Fig. V-15, with y_0 being given by the relation

$$y_0 = A \left(\frac{x_0}{\ell} \right) + B \quad (2)$$

Also, the leading end load, P_o , is given by:

$$P_o = C \left(\frac{x_o}{l} \right) + D \quad (3)$$

In all three equations, $0 < \frac{x_o}{l} < 0.64$. The coefficients A, B, C, and D in Eqs. 2 and 3 are functions of reactor axial temperature gradient, side displacement, leading end geometry, and control rod weight. Values of these coefficients are given in Table V-2. The coefficients in Table V-2 and Eq. 3 were used to plot the leading end load as a function of control rod position. The results are shown in Fig. V-16.

It is noted that the load does not vary greatly. As stated previously, the design with the snug end was chosen. The corresponding load is 2.55 pounds.

Table V-2. Coefficients for Eqs. 1, 2, and 3.

Tie rod displacement	Leading end configuration							
	Snug end				Loose end			
	A	B	C	D	A	B	C	D
Max radial thermal expansion	0.203	0.060	0.212	2.31	0.203	-0.030	0.212	2.22
Max radial thermal expansion plus 0.060 in. side displacement	0.352	0.060	0.367	2.31	0.352	-0.040	0.367	2.21
Max radial thermal expansion minus 0.060 in. side displacement	0.156	0.030	0.163	2.28	0.156	-0.060	0.163	2.19

If the coefficient of friction between the tie rod and the control rod is 1.0, there is a force opposing the motion of the control rod of 2.55 pounds.

When the reactor is scrammed, the control rod will be loaded, as shown in Fig. V-17. The control rod should not buckle under the deceleration load which occurs at the end of a scram.

The energy stored in the control rod is equal to the work done by the external forces and can be represented by:

$$\int_0^L EI \left[\frac{d^2 y}{dx^2} \right]^2 dx = p_{yo} + \int_0^L dx \int_0^x g \frac{w}{L} \left[\frac{dy}{dx} \right]^2 dx + \int_0^L P_f \left[\frac{dy}{dx} \right]^2 dx \quad (4)$$

where

g is the acceleration of gravity

E is the modulus of elasticity

I is the cross-sectional moment of inertia

f is the coefficient of friction between control rod and tie rod

W is the weight of the control rod

and the other symbols are as defined previously.

The deflection at a position along the control rod (Eq. 1) does not include the effect of the friction force fP on the end of the rod since this load is small compared to the inertial forces.

From Eq. 4, assuming constant properties,

$$g = \frac{L}{W} \left\{ \frac{EI \int_0^L \left[\frac{d^2 y}{dx^2} \right]^2 dx - P \left[y_0 + f \int_0^L \left[\frac{dy}{dx} \right]^2 dx \right]}{\int_0^L dx \int_0^x \left[\frac{dy}{dx} \right]^2 dx} \right\} \quad (5)$$

The critical buckling acceleration is obtained by setting $P = 0$ in Eq. 5.

$$g_{\text{crit}} = \left\{ \frac{\int_0^L \left[\frac{d^2 y}{dx^2} \right]^2 dx}{\int_0^L dx \int_0^x \left[\frac{dy}{dx} \right]^2 dx} \right\} \quad (6)$$

Combining Eqs. 1 and 6, the critical acceleration is:

$$g_{\text{crit}} = \frac{EI}{W} \left\{ \frac{\frac{3}{L^2} \left[A \left(\frac{x_0}{l} \right) + B \right]^2 + \frac{36}{5} \left[\frac{WL^2}{48EI} \right]^2}{\frac{3}{8} \left[A \left(\frac{x_0}{l} \right) + B \right]^2 - \frac{9}{70} \left(\frac{WL^2}{48EI} \right) L \left\{ \left[A \left(\frac{x_0}{l} \right) + B \right] - L \left(\frac{WL^2}{48EI} \right) \right\}} \right\} \quad (7)$$

For $EI = 10^5$ lb - in², $L = 66$ inches, and $W = 6$ pounds. Using Eq. 2, Eq. 7 becomes:

$$g_{\text{crit}} = \frac{30.6 y_0^2 + 9.49}{y_0^2 - 0.123 y_0 + 0.0443} \quad (8)$$

The acceleration that will buckle the control rod is shown in Fig. V-18 as a function of leading end deflection, y_o . The deflection of the control rod as a function of position of the leading end of the control rod in the core for the two leading end geometries considered and several reactor side displacements is shown in Fig. V-19.

The maximum critical buckling acceleration (237.5 g's) occurs when the deflection of the tie rod at the leading end of the control rod is 0.07 inch. The minimum critical buckling acceleration (133 g's) occurs at a tie rod deflection of 0.285 inch. Thus, the control rod will not buckle under the expected 100-g maximum scram deceleration load.

The aerodynamic drag on the control rod and tie rod channel is given by:

$$F = p_2 A_2 (1 + \gamma_2 M_2^2) - p_1 A_1 (1 + \gamma_1 M_1^2) \quad (9)$$

where

p is static pressure

γ is ratio of specific heats

M is mach number

A is flow area

1 is condition at aft end of control rod

2 is condition at beginning of control rod considered.

Since $A_1 = A_2$

$$F = A \left[p_2 (1 + \gamma_2 M_2^2) - p_1 (1 + \gamma_1 M_1^2) \right] \quad (10)$$

Values of p and M are obtained from a FLOSS III output for a heated control rod and tie rod assembly.¹⁰⁰ For the worst case the control rod is fully inserted and the inlet air is 1060°F. The tie rod is located at a radial position of 8.5 inches. The center of the control rod is assumed to have no flow.

The aerodynamic drag force on the control rod is found by multiplying Eq. 10 by the ratio of the periphery of the control rod to the total periphery of the tie rod and control rod. When flow through the center of the control rod is considered, the pressure drop along the length of the control rod decreases. This reduces the aerodynamic load on the control rod from that calculated. The aerodynamic drag force on the control rod with no flow through the center is 42.8 pounds. The static pressure difference between

the ends of the control rod, acting on the ends, gives an additional component of 26.1 pounds. Thus, the total aerodynamic and pressure drop load on the control rod with flow down the center is 68.9 pounds. This results in a low tensile stress in the control rod of 373 psi.

As indicated previously, there is friction between the control rod and the control tie rod as the control rod is moved. This friction places an additional load on the control rod actuator.

Tests to determine the coefficient of friction between hafnium and Hastelloy R-235 bars at 1300, 1500, and 1700°F were conducted. The hafnium surface was cleaned ultrasonically before the test. The surface finishes of the hafnium and the R-235 were, respectively, 16 and 3 micro-inches (not typical in the reactor). The hafnium bar was sandwiched between two stationary Hastelloy R-235 parts, and the bearing pressure was 10 psi. The loads required to move and maintain motion of the hafnium at a constant speed were measured. Tests were performed after 2- and 5-hour soaking times at test temperature. The coefficient of friction determined from these tests as a function of temperature is shown in Fig. V-20.

6. Stress Analysis

There are two kinds of stresses in the control rod; mechanical and thermal. The mechanical stresses are additive along the axis, and are caused by the four major forces acting on the control rod described above. The normal stress is about 70% of maximum because scram stress will not normally be present. The bending of the control rod due to reactor barreling causes the greatest tensile stress, and has its maximum effect at the attachment. This bending stress is twice as large at the attachment as the scram stress.

The maximum scram stress at the attachment is:

$$\sigma_{\text{scram}} = \frac{F_{\text{scram}}}{A_{\text{control rod}}} = \frac{600}{0.185} = 3250 \text{ psi}$$

The maximum bending stress due to tie rod displacement occurs at the position of maximum moment. With reference to Fig. V-15, the moment, including the effect of the weight of the control rod, is:

$$M(x) = p(L - x) - \frac{W}{2L}(L - x)^2 \quad (11)$$

Differentiating Eq. 11 and setting equal to zero:

$$\frac{dM}{dx} = -p + \frac{W}{L} (L - x).$$

The maximum moment is found to exist at

$$x = L (1 - p/W) \quad (12)$$

It is quite clear that Eq. 12 will apply only to certain values of p/W since certain impossibilities result. For example, if $p/W = 0$, the maximum moment occurs at $x = L$, and this is known to be not so. This situation is resolved by substituting Eq. 12 in Eq. 11, obtaining

$$M_D = \frac{p^2 L}{2W}. \quad (13)$$

By determining the moment at the builtin end from Eq. 11:

$$M_B = (p - \frac{W}{2}) L. \quad (14)$$

If the ratio M_B/M_D is plotted for various values of W/p , it can be observed that the moment obtained by using Eq. 14 is greater than the moment from Eq. 13 if $W/p > 2.414$ or $W/p < -0.414$.

At this point in the discussion it is well to note that p is a strong function of W . As noted previously, $p_{\max} = +2.55$ pounds. It can be shown that $p_{\min} = +1.64$ pounds. Thus, since $W = 5.4$, $W/p = 3.3 > 2.414$.

Using Eq. 14 then, the maximum bending stress is:

$$\sigma = \frac{MC}{I} + \frac{(p - W/2) LC}{I} = \frac{67.4 (0.5)}{0.01} = 3370 \text{ psi.}$$

The aerodynamic stress is:

$$\sigma = \frac{F}{A} = \frac{69}{0.1845} = 373 \text{ psi}$$

for airflow down center of control rod.

Thermal stresses are caused by heat generation within the control rod. Maximum thermal stress occurs near the leading end of the control rod which, at its maximum insertion, is located near the middle of the reactor. Minimum thermal stress occurs at the attachment to the slider.

The thermal stress in a control rod blade as a function of internal heat generation rate and sheet thickness (blade half thickness) is:

$$\sigma = \frac{E \alpha q_o \ell^2}{3K(1 - \nu)} \quad (15)$$

where:

σ is the thermal stress

E is the modulus of elasticity

q_o is the internal heat generation rate

ℓ is the blade half thickness

K is the thermal conductivity

α is the coefficient of thermal expansion

ν is the Poisson's ratio.

Equation 15 is plotted in Fig. V-21 for hafnium with E, α , and K corrected for the average temperature associated with each internal heating rate and blade half thickness. For this purpose the surface temperature was assumed to be 1060° F.

The thermal stress as a function of axial position of a minimum reactivity 0.080 inch blade thickness control rod fully inserted at full power is shown in Fig. V-22.

The maximum thermal stress of 1700 psi occurs near the leading end of the control rod. The attachment end of the control rod has a thermal stress of 125 psi, assuming no decrease in heating rate upstream of the front reflector. Thus, thermal stress at the attachment is small compared to the stress caused by a scram operation.

The safety strap carries no load during normal reactor operation. However, if a control rod should fail at the attachment without losing any momentum while at peak velocity (15 ft/sec) at the end of a scram, energy which would need to be absorbed by the safety strap is 21 ft-lb. Under these conditions the maximum elastic stress in the safety strap, using conventional notation, is found to be:

$$\sigma \sqrt{= \frac{2 UE}{A \ell}} = 116,000 \text{ psi}$$

This stress is tolerable in a Rene' 41 strap operating below 1500° F.

7. Testing of Completed Parts

The attachment end of a control rod assembly was heated to 1200° F and cycled five times between 0 and 1000 pounds force (5400 psi). A crosshead motion rate of 1 inch per minute was used. This stress corresponds to the total stress at the attachment for a 50-g scram deceleration load. The total permanent set at the attachment after five cycles was 0.0160 inch. The attachment was then loaded to failure with a crosshead motion rate of 1 inch per minute. The attachment failed at 3800 pounds (21,800 psi). This agrees with the ultimate strength of hafnium at 1200° F which is about 21,500 psi, as shown in Fig. V-5. (Figure V-23 shows the attachment after failure.)

The attachment strength represents a safety factor of 3.1 for a control rod with a 100-g scram deceleration load, and 3.8 for a control rod with a 50-g scram deceleration load.

The effect of high velocity air on a control rod inserted in a tie rod was to "lift" the control rod. Maximum vibration amplitude was less than ± 0.004 inch. This data was obtained in a test in which 50° F air at the rate of 4.5 lb/sec was passed through the tie rod. The tie rod was deflected to simulate several degrees of reactor core barreling.

Two safety straps were each impacted five times with 40 foot-pounds of energy by dropping a 6-pound weight 6.6 feet. One strap was in the fully aged condition; the other was in the fully solution heat treated condition. The temperature of each strap increased linearly from 1100° F to 1700° F with distance, from a point 29 inches from the threaded attachment to the broad portion (leading end) of the strap. The remainder of the safety strap was held at 1100° F throughout the test. Neither strap failed during the tests. The overall lengths of the straps before and after testing were in agreement within 1/32 inch.

A control rod and strap assembly in a tie rod deflected to simulate various degrees of reactor barreling was tested, using a vernier actuator and slider system as the drive mechanism. Tests were run at 70° F with no flow and at 1350° F, 360 psia, with flow. Linear motion of this assembly was cycled at frequencies between 2 and 30 cps with amplitudes of 0.4 inch point to point, and subjected to step inputs between 0.2 and 20 inches point

to point. The total number of cycles in both tests was 835. The only effect noted after the tests was plastic deformation on the leading end of the control rod, as shown in Fig. V-24. The diametral wear on the leading end appeared to be 0.002 to 0.005 inch.

The control rod together with slider and way (Section V-B) were impacted several times at room temperature and at 1100°F, using a Hyge machine. The control rod was guided in a simulated tie rod. These tests were designed to investigate effects of high acceleration and high deceleration on the control rod.

Table V-3 gives the control rod velocity, kinetic energy, force, and stress at impact for various runs. During some of the runs no data was taken because of other equipment malfunction.

A safety strap was broken after Run 18, and an entire new control rod assembly was installed. The failure was determined to result from faulty fabrication of the strap. No permanent elongation or cracks were observed on any other control rod assemblies. There was some local deformation at the control rod attachment to the slider, but the control rods remained completely serviceable. As expected (Fig. V-18), the control rod assembly did not buckle during 48-g accelerations.

8. Failure Modes

The two important modes of failure of the control rod are overstressing and oxidation. At temperatures above 1700°F, the oxidation rate of hafnium in static air increases rapidly. This decreases the room temperature ductility of hafnium, with a corresponding increase in notch sensitivity. With decreased ductility, flaws in the control rod may result in failure during low temperature operation. Oxidation is not likely to be a problem, however, because maximum control rod temperature of 1837°F will occur rarely during full vernier rod insertion at full power. Figure V-25 shows the strength of hafnium and the stress on the control rod at various axial points along the fully inserted rod. Figure V-26 shows the safety factor based on ultimate strength as a function of axial position along the control rod. These graphs show a minimum safety factor of 2.5 located 50 inches from the attachment.

Table V-3. Control rod impact tests.

Run No.	Temp, °F	Control rod impact velocity, ft/sec	Control rod kinetic energy, ft-lb	Control rod force, lb	Control rod stress, 10^{-3} psi
1	RT	24.2	54.6	8,980	48.7
2	RT	-	-	-	-
3	RT	-	-	-	-
4	RT	44.6	185	16,520	89.5
5	RT	-	-	-	-
6	RT	-	-	-	-
7	RT	24.3	55.0	9,000	48.8
8	RT	-	-	-	-
9	RT	7.26	4.91	2,690	14.6
10	RT	10.14	9.58	4,760	25.8
11	RT	18.45	31.7	6,840	37.1
12	RT	-	-	-	-
13	RT	15.14	21.3	5,610	30.4
14	RT	23.6	51.8	8,750	47.4
15	RT	-	-	-	-
16	RT	-	-	-	-
17	RT	-	-	-	-
18	1125*	-	-	-	-
19	1100	-	-	-	-
20	1050	40.7	154	13,400	72.5
21	1050	-	-	-	-
22	1150	-	-	-	-
23	1150	-	-	-	-

RT = Room temperature.

*Safety strap broken on disassembly after Run 18.

-(No data because of other equipment malfunction.)

The safety strap can fail only if the control rod fails by over-stressing during a scram deceleration. The factor of safety of a safety strap based on ultimate strength is 1.12.

9. Fabrication and Inspection

Two potential vendors each fabricated a preliminary control rod for test purposes and fabrication experience. The rods were fabricated using resistance welding techniques, which produced acceptable joints.

Control rods for the reactor were fabricated in accordance with Ref. 101 from hafnium purchased under Ref. 96. The fabricator sheared and inspected the hafnium sheet for flatness and width. The hafnium strips were placed in a forming die which was heated to 750° F for 6 hours. The strip remained in the die for 10 minutes before forming to a 58° included angle. The formed strips were vapor blasted and vapor degreased. They were then pickled in a solution of nitric and hydrofluoric acid until 0.0015 to 0.002 inches total material was removed from the strip surfaces. The formed strips were inspected for cracks, using fluorescent penetrant, and straightened for assembly. Six formed strips were assembled into the control rod configuration and inspected for fit and straightness. The assembly was spot welded from one end, placing spots 0.1 inch apart. The welded assembly was stress-relieved in a vacuum for 1 hour at 1350° F, and then straightened and finish machined. The final control rod was dimensionally and fluorescent penetrant-inspected, degreased, and shipped.

The safety strap was fabricated from Rene' 41 rod and sheet in the 1975° F water-quenched condition. The tube and strap were TIG welded together in an argon atmosphere. The safety strap was heat treated at 1950° F for 30 minutes and cooled in hydrogen to 1100° F in 5 minutes, then aged at 1400° F for 16 hours and cooled in hydrogen.

Safety straps were inspected for surface cracks and weld quality. Each safety strap was inspected for cracks with a fluorescent penetrant per AMS 2645. Photomicrographs were taken of the weld section of random safety straps to assure 100% weld penetration.

B. Reactor Control Guide System

1. Description

The reactor control guide system consists of the majority of the components located in Duct 4 (see Fig. VII-4). These components support, align, and comprise the attachments between control rods and control actuators. The guide system is designed to allow the control actuators to be installed or removed by remote handling methods, but the remaining components inside Ducts 4 and 5 would be very difficult to replace remotely.¹⁰² Each control actuator can be removed from its mounting plate by undoing three bolts and the rack-to-slider connector. This will automatically disconnect all electrical, pneumatic, and other connectors at the interface joint (Sta. 282.75 Fig. VII-4). Two dowel pins locate the actuators at this joint.

The life requirements imposed on this system were the same as for the control actuators: 30 hours operating time, with 10 hours under nuclear conditions. All components were to be completely reliable with no maintenance during this life-time. Table V-4 summarizes the requirements, and they are further delineated in Refs. 103 and 104.

All components are inside the duct where radiation levels are relatively high and where they are subjected to oxidation by 1060° F reactor ram air. Basic support of the various components is provided by the relatively stiff duct rather than by cantilevering the system from the reactor support grid. The latter method should be considered again, however, in the design of an actual flight system.

The guide system components plus the other associated components (see Figs. VII-5 and IV-45) create quite a complexity of structures to be mounted in the airstream just forward of the reactor. Placement of these components is with respect to the fourteen or possibly eighteen control rods with their 40+-inch stroke and the six L-ring poison rods with their 65-inch stroke (Fig. V-3). In order to insure proper placement and support of all components, a full size mockup structure was fabricated. This structure is shown in Figs. V-27 and V-28.

Precise alignment and assembly of the system was required. This was accomplished using optical and other alignment methods, combined with

Table V-4. Guide system design requirements.

I. Normal Operating Requirements

1. Stroke	40 inches, controlled. Overtravel: 3 in. for rod "full in" direction, and 2 in. for rod "full out" direction.	
2. Environment	30 reactor runs similar to Fig. IV-66.	
3. Misalignment	Rack deflection due to thermal and load misalignment with respect to actuator centerline shall not be more than: (a) 0.004-in. radius at rod "full out" position. (b) 0.075-in. radius at rod "full in" position.	
4. Airflow load	0 to 35 lb per control rod, varying with reactor power and flow rates	
5. Weight, including slider, couplers, and rods	Vernier	Shim
	11 lb min	28 lb min
	20 lb max	55 lb max
6. Friction load, total max	25 lb	75 lb
7. Scram insertion time for 40 in. stroke, minimum	0.25 sec	1 sec
8. Maximum acceleration	±50 g	±25 g
9. Endurance		
8-in. p-p amplitude	R. T.	10 ⁵ cycles at 1 cps
	H. T.	10 ⁴ cycles at 1 cps
1-in. p-p amplitude	R. T.	10 ⁵ cycles at 10 cps
	H. T.	10 ⁴ cycles at 10 cps
Scrams (40-in.)	R. T.	100
	H. T.	50
Total travel	166,000 ft	24,000 ft

II. Extreme Nonrupture Requirement

The maximum force capability of an actuator is the static scram force (air pressure on piston, no motor load) and is 3000 lb. This load can be imposed on the guide system only after a control rod has failed in buckling.

match machining and doweling at assembly (Figs. V-29 and V-30). It took 6 months to assemble the system using the procedures detailed in Ref. 105.

This system is quite reliable with respect to the intended application. Structurally, it was designed and tested to loads much greater than normal operating loads, including scrams. The usable wear life of the bearings is at least four times the life requirement, which is at least three times the 10-hour reactor flight requirement.

2. Component Design

a . Shim slider assembly

The shim slider assembly consists of the four shim sliders (two types) bearings, and four shim ways. The shim slider supports and guides the control actuator rack and transfers the loads from a single rack to three control rods. (Sliders were designed to accept four control rods.) The slider must operate over a 40-inch stroke plus 5 inches of over travel, with low resistance (friction) loads. The sliders are required to be supported and actuate around the 24 possible control and poison rods (Fig. VII-6). The frontal face of the slider is porous, to allow convection cooling and to minimize thermal gradients.

The shim ways support and guide the shim sliders, and are part of the slider bearing surfaces. They were also designed for convective cooling; large holes increase the heat transfer film coefficients and decrease fabrication problems.

Operations in high-temperature air (1060°) with high nuclear fluxes and 30 hours operating life with no maintenance, combined with the long stroke and impact loads, posed severe problems for the bearings. Non-lubricated rolling bearings were first analyzed in detail, but their integrity could not be assured. The rolling members (ball or roller) could not be adequately cooled in the required environment. Sliding bearing were considered next. After studying various sliding materials, it was determined that one of the impregnated "high-temperature" graphites would satisfactorily meet all the requirements. Results of a basic wear test of the three grades of graphite versus several metals were used in finalizing the bearing design. The slider-and-way bearing configuration is a double "V" system, and was chosen for its stability with the various control rod loads (Figs.V-31 and V-32).

Three (four possible) shim control rods are coupled in two different arrays to the slider. The basic slider-and-way bearing configuration meets the requirements of both the arrays (Fig. VII-6). Each slider has sixteen bearing pads that are adjusted at assembly for 50% uniform contact and an average 0.002-inch gap. A pad consists of a thin HT 56 graphite disc restrained in a Type 416 stainless steel cup by swaging.

The peak stress in a slider for the 3,000 pound load condition is 35,000 psi tensile bending on a wing next to the control rod boss. This load is the maximum which the actuator can apply. All critical requirements (vibration, wear life, strength, and impact) were met by the shim sliders during thorough testing.

b. Vernier slider assembly

This assembly consists of the vernier slider, bearings, and vernier support tube. The vernier slider supports and guides the vernier actuator rack and transmits the load to a single control rod. The peak stress for the 3,000 pound load condition is 20,000 psi tensile bending on a forward spoke.

The vernier support tube supports and guides the cylinder-like vernier slider and has a funnel on the forward end to allow air to cool the slider, control rod, and inside of tube (Figs. V-28 and V-33). The forward end of this tube is rigidly bolted to the vernier actuator mounting plate. The aft end is supported by the reactor support grid with two slip-fit pins. Tube diameter was dictated by location of the control actuator position transducer and the desire to keep the fundamental natural frequency above 50 cps. The tube was grooved to prevent slider rotation during assembly, normal operation, or disassembly.

The bearings are fully restrained graphite rings. Their velocities are higher than the shim slider bearings, but the loads are much lower. Their reliability was verified by the basic bearing and vernier slider operational and impact tests.

Most vernier sliders used in the proof tests were fabricated from castings; but because all the castings contained flaws of various degrees, the operational reactor sliders were fabricated from welded plates. The operational sliders, being flaw free, have higher factors of safety than those indicated by the structural test.

c. Center support tube

The center support tube supports the four shim ways and internally supports the six L-ring poison rods (Fig. V-34). It is bolted rigidly to the actuator mounting structure and has two slip-fit pins into the reactor support grid to allow for axial thermal expansion. Its fundamental natural frequency for transverse vibration is 90 to 100 cps. Stresses are negligible. The rows of large holes (2-inch diameter) along each poison rod are for insuring reliable convective cooling of the poison rods.

d. Actuator mounting plates

These plates (three types) are the basic support and alignment for the control actuators, and they contain all the pneumatic and electrical connections for the actuators. The design is based on providing sufficient space for the various connections; therefore, the stresses and deflections are negligible. The plates are bolted (six bolts) and doweled (two dowels) to the actuator mounting structure after alignment with respect to the reactor support grid (Figs. V-29 and V-33).

e. Actuator mounting structure

This structure supports the six actuator mounting plates, the two vernier support tubes, and the center support tube (Fig. VII-6). It is designed for rigidity in all directions, and transfers the actuator and air loads to the duct via six radial pins. The pins keep the structure centered during thermal expansion.

For a worst case condition of maximum airflow load and all actuators scrambling simultaneously, the structure deflects 0.010 inches axially with a peak stress less than 1000 psi. Stresses and deflections were verified by testing the mockup structure.

f. Actuator support structure

This structure loosely supports the control actuators at the forward end. The actuator ends have a slip fit in the holes provided in the structure. Four radial pins keep the structure centered and transfer the loads to the duct (Fig. 30).

Mounted on the forward surface of this structure is a two-mesh Inconel screen (Fig. V-35). The purpose of this screen is to catch debris that might be contained in the ram air. Maximum stress in the structure is 5300 psi tension due to the 3800 pound screen load during maximum flow.

3. Materials

Hastelloy X was chosen as the basic structural material for the guide system because of its reliability at elevated temperatures (1200° F), its thermal compatibility with other components, and its ease of fabrication. The coefficient of thermal expansion of Hastelloy X is between that of the control actuators (Inconel X), and the reactor support grid (Rene' 41) and ducts (Hastelloy C). Because of the probable impact loads, ductility was a prime requirement, and strength was of secondary importance. Some tensile and bend testing on cast material was done by LRL, and the results generally agreed with the advertised values.

The actuator mounting plates were machined out of solid plates. The actuator mounting structure and the actuator support structure were fabricated from weldments. The center support tube and the vernier support tubes were fabricated from rolled and welded plates. The shim ways were fabricated by welding two U-shaped bars together. The vernier sliders were fabricated from castings and weldments. The shim sliders were fabricated from welded castings. All components were solution heat-treated after welding.

4. Aerothermo Analysis

The purpose of this analysis was to calculate heat transfer film coefficients, airflow effects, and temperatures of the control guide system. The analyses were based on the reactor structural design condition.²

a. Airflow analysis

Convective coefficients were computed for the various shapes involved, using the following correlations.

Flat plates and cylinders with longitudinal flow. Equations 1 and 2 are for turbulent and laminar flow, respectively, and are applicable for low Mach numbers and moderate temperature differentials between the fluid and wall.

$$h_x = 0.0296 (k/x) Re^{0.8} Pr^{1/3} \quad \text{Turbulent flat plate flow,} \\ \text{local coefficient.} \quad (1)$$

$$h_x = 0.332 (k/x) Re^{0.5} Pr^{1/3} \quad \text{Laminar flat plate flow,} \\ \text{local coefficient.} \quad (2)$$

The transition region between laminar and turbulent flow varies from Reynolds Number of 50,000 to 3×10^6 , depending upon the degree of turbulence in the approaching airstream and the degree of roughness of the heat transfer surface. The transition region indicated by the area between curves 1 and 2 in Figs. V-36 and V-37 is for a high turbulence level in the approaching stream and is applicable in this case.³⁰

Pipe flow. Equation 3 is applicable for low Mach numbers and moderate temperature differentials between the fluid and wall. The laminar entrance region is of negligible length in this application, and the flow is turbulent.

$$h_d = 0.023 (k/d) Re^{0.8} Pr^{1/3} \quad (3)$$

Cylinders and squares in crossflow. The local film heat-transfer coefficient around cylinders (and other shapes) in flow of air normal to the axis varies greatly (Fig. 9-4, Ref. 30).

Much of the available experimental data is based on total heat flow to or from a tube, and was intended to indicate average film coefficients, which are sufficient for this analysis. Equation 4 is based on airflow experiments by R. Hilpert.³¹

$$Nu_d = 0.43 + C Re_d^m \rightarrow C Re_d^m, \text{ for high Reynold's Nos.,} \quad (4)$$

where the coefficients C and m are given by:

Cross Section	Re_d	C	m
Cylinder	40,000-400,000	0.0239	0.805
Square	5,000-100,000	0.0921	0.675

Equation 4 for cylinders agrees fairly well with the results of thirteen other experimenters (Ref. 106 p. 259). Equation 4 is based on a low turbulence level in the approaching stream. The various components upstream create a high turbulence level which increases the cross-flow film coefficients by as much as 35%.

Aft surface of obstructed item. For the sides and rear surfaces of a plate where the approaching airstream is obstructed by an object, in this case a control actuator, Eq. 5 is applicable.¹⁰⁷

$$h = 0.20 (k/L) \text{Re}^{2/3} \quad (5)$$

where

L = actuator diameter = 5.25 inches.

Accurate analytical determination of the flow distribution and pressure drop around the components in Ducts 3 and 4 was not feasible. Estimates of individual component pressure drops and resultant flow distributions were made, however. Primary mode of pressure loss is due to drag. The drag force for each component was estimated from Eq. 6.

$$F_D = C_D A \left(\frac{\rho V^2}{2g} \right) \quad (6)$$

where C_D is drag coefficient for standard shapes based on estimated Reynolds number. At several axial stations the applicable drag forces were summed up, and pressure losses for the zone were calculated. The total pressure loss in Ducts 3 and 4 was approximately 1.5 psi, excluding screen pressure drop, of which approximately 90% was due to drag losses around the control actuators. The flow rate in the center support tube was somewhat less or about the same as the general duct area. This is due to the relatively rough surface caused by guide tube brackets, and crossflow caused by the porous center support tube (Fig. V-34). The flow rate in the vernier tube was estimated in a conservative manner.

The total pressure loss there was composed of a drag loss occurring just upstream of the tube entrance, entrance losses to the funnel and slotted tube, friction loss within the tube, and drag losses around the slider.

Each loss was expressed in terms of the equivalent velocity head. The total loss was then set equal to the 1.50 psi loss in the main duct, and the velocity calculated. Very conservative drag coefficients were used in this analysis. In this manner the flow rate in the vernier tubes was computed to be at least 25% of the average flow rate in the duct. Subsequent measured values were between 52 and 67% of the average flow rate in the duct.

Convection coefficients for relatively open duct areas based on Eqs. 1 through 4 are plotted in Fig. V-36. Coefficients for inside the vernier tube are plotted in Fig. V-37 based on 25% of average flow. These results are applied in the following manner, depending on the local configuration of the component.

Actuator mounting and support structures: Curves 1 and 2, Fig. V-36.

Shim slider: Curves 1, 2, 3 and 4_b, Fig. V-36.

Conduits and other tubing: Curve 4_a, Fig. V-36.

Actuator mounting plates: Curves 1, 2 and 4_b, Fig. V-36.

For portions in direct paths of oncoming airstream.

(For sides and rear portions where the approaching air is obstructed by a control actuator, the film coefficient is 230 Btu/hr ft² °F, equation 5.)

Center support tube: Curves 1 and 2 for outside, and Curve 3 for inside, Fig. V-36. (Conservative for inside, since each hole starts a new boundary layer.)

Shim way: Curves 1 and 2 for outside, and Curve 3 for inside, Fig. V-36. (Again, each hole starts a new boundary layer.)

Guide tubes: The slotted tubes, brackets and proximity of the center support tube create a complex arrangement. Curve 3, Fig. V-36, using the center support tube diameter, yields a conservative coefficient.

Vernier tubes: Curve 1, Fig. V-36 applies to the outside, and Curve 3, Fig. V-37, applies to the inside.

Vernier slider: Curves 1 and 2, Fig. V-37, are applicable.

A regular Karman vortex sheet is observed only in the range of Reynolds numbers from about 60 to 5000.¹⁰⁸ At higher Reynolds numbers there is complete turbulent mixing. At full flow and temperature the largest diameter subject to vortex effects is 0.014 inch. Therefore, vortex vibration of components inside Duct 4 would occur only during low flow when dynamic bend is quite low.

b. Nuclear heating

Nuclear heating densities of components forward of the core were evaluated with the aid of Monte Carlo codes YOGI and YOGI-N.²⁷

The total nuclear heating density (50% uncertainty) forward of the support grid is given in Fig. V-38. The heating density values are given for full density material, normalized to 1 mW/ft^3 of average core power, of the respective components located on axis. (The design average core power is 9.05 mW/ft^3 .)

c. Heat transfer

Heat transfer analyses were made using the methods described for the aft reactor duct (Section IV-H) and the program shown in Fig. IV-66. Most transient analyses were based on infinite conductivity for the relatively thin components.

The portion of an actuator mounting plate located aft of an actuator was analyzed using a two-dimensional transient heat transfer computer code. The thick plate had an adiabatic forward surface, varying film coefficient on the sides and aft surface, and exponentially decaying nuclear heating rate. The transient temperature range for this area of a mounting plate is given in Fig. V-39, curve B. Curve A is for the thinnest (smallest time constant) part of a mounting plate. The area between the curves represents the temperatures of the remaining parts of a mounting plate.

The average transient temperature for the actuator mounting structure, vernier support tube, and center support tube is given in Figs. V-40 and V-41. Steady state temperatures are given in Table V-5.

Table V-5. Steady state temperatures of guide system components. (Structural design condition.)

Component	Temperature range, °F
Actuator mounting structure	1080 - 1100
Actuator support structure	1065 - 1080
Center support tube	1070 - 1095
Vernier support tube	1085 - 1120
Vernier slider	1120 - 1150
Shim slider	1065 - 1090
Shim way	1070 - 1090

The transient temperature distributions were used to insure sufficient clearance for differential thermal expansion from Duct 4 to vernier support tube and center support tube, shim slider to way, vernier slider to support tube, Ducts 3 and 4 to actuator mounting and support structures, and control actuators to Duct 3. Thermally induced stresses between the actuator mounting plates and structure were also checked.

5. Testing

a. Basic bearing tests

These tests were made to obtain friction, oxidation, and wear data for three high-temperature impregnated graphites sliding over two types of metal surfaces. They were based on the intended application and a compilation of data obtained from literature and material suppliers. The test procedure differed from most previous graphite wear and friction tests because of the intermittent contact between the graphite and the base metal. Three graphite wear specimens, equally spaced from the axis of rotation, rubbed against the disc specimen. This setup allowed the metal wear surface to oxidize during the test.

The tests were carried out in three phases. Phase I consisted of a series of tests designed to select the most promising of the three graphites and the most promising metal surface. The bearing loads were 90 psi, and the speeds, temperature, and travel were as shown in Fig. V-42. These conditions are based on the shim slider requirements. Figure V-43 summarizes the graphite wear results. Midway during the wear tests the graphite specimens were oxidized in 1200° F air at 350 psia flowing about 100 fps over the graphite specimen for a series of 5-minute cycles. Weight loss results are shown in Table V-6. The cyclic oxidation had a negligible effect on the wear and friction properties of the graphites tested. Chrome-plated Hastelloy X was only slightly better than Hastelloy X for overall wear and friction values.

Phase II was a series of metal-to-metal tests to determine the seizure and friction characteristics of the metal retainers for the graphite. The retainer metal, 416 annealed stainless steel, was tested with Phase I conditions against Hastelloy X and chrome-plated Hastelloy X. Neither galling nor high friction was encountered during this phase.

Table V-6. Weight loss due to oxidation.

Specimen No.*	Initial wt, g	Final wt, g	Loss, %
<u>Oxidized 5 cycles</u>			
1P	1.4088	1.3941	1.04
2P	1.3892	1.3747	1.04
3P	1.4069	1.3890	1.27
1S	1.4339	1.4068	1.89
3S	1.4169	1.3952	1.53
1M	1.6342	1.6291	0.31
2M	1.6220	1.6150	0.43
3M	1.6177	1.6130	0.29
<u>Oxidized 10 cycles</u>			
4P	1.4087	1.3895	1.36
5P	1.4139	1.3942	1.39
6P	1.4190	1.3986	1.44
7P	1.4156	1.3993	1.15
9P	1.4058	1.3840	1.55
4S	1.4328	1.4122	1.44
5S	1.4237	1.4038	1.40
6S	1.4260	1.4059	1.41
<u>Oxidized 20 cycles</u>			
3P1	1.4141	1.3940	1.42
3P2	1.4169	1.3963	1.45
3P3	1.4215	1.3986	1.61
3P7	1.3859	1.3771	0.63
3P8	1.3944	1.3857	0.62
3P9	1.3930	1.3839	0.65

*Specimen No. designations P, S, and M refer to the three suppliers.

Phase III was to obtain further data on the graphite selected from Phase I. Phase I and high speed, low load (vernier slider) conditions were used in Phase III. Typical wear results for this graphite are shown in Fig. V-42, and typical friction results are shown in Fig. V-44. See also shim slider endurance test.

Preoxidation runs using 56 HT and No. 741 graphites were characterized by a squeaking noise that varied in intensity from moderate to severe in the average temperature range 90 to 550° F. As the temperature approached the maximum in this range, the squeak intensity diminished rapidly. In the average range of 440 to 800° F, no noise was noticeable with normal laboratory background noise. The squeaking returned as the temperature passed out of the silent range and rapidly grew in intensity until the step terminated at 1200° F. In testing the EY-105 graphite, the intensity of the squeak did not reach the high level of the 56 HT or No. 741 graphites, nor was there any appreciable period of silent running.

After oxidation, the graphite generally ran without the squeak during the room temperature runs. This squeak returned as the temperature was increased from 90 to 300 or 400 ° F, stopped, and commenced again at approximately 600° F, continuing until the run ended. Squeak during the 1200° F, 10 ft/min step occurred during all the tests. At 350 ft/min and 1200° F, chatter and squeak were characteristic.

The reasons for the squeaking noted during these tests are not fully understood. This squeaking may have been a consequence of the graphite lattice structure, test apparatus, or test specimens. During the subsequent shim slider tests there was some intermittent squeaking during the 1200° F runs. The basic apparatus and specimens were quite different, but the bearing surface (way) was oxidized as the slider moved.

The test results indicate that 56HT graphite sliding over Hastelloy X will produce a satisfactory slider/bearing combination for the guide system application. Test details are reported in Refs. 109 and 110.

b. Shim slider vibration test

A shim slider and way assembly was tested in a simulated vibration environment. The test assembly is shown in Fig. V-45. Input loading was monitored on the mounting fixture near the center support tube

attachment. The assembly was tested in two directions at 2, 5, and 10 g's between 10 and 2000 cps; at 9.3 g RMS random with a band width between 50 and 2000 cps; and at 30 g's from 500 cps to 2000 cps, for a total of 31 runs.

All runs lasted for 30 minutes except the 30-g test, which was for 15 minutes plus 10 minutes at each resonance. The assembly performed satisfactorily during all the tests. Even the 5-g vibration is the most expected in service.

At the first natural frequency of the center support tube (between 90 and 100 cps) there was an amplification factor of about 20 at the pinned end of the center support tube. Inspection revealed that the pins had been incorrectly equipped with slip fits at both ends. When this was corrected the amplification factor reduced to about 5. (The first transverse natural frequency of the control rods in their full-out position is near 32 cps.) After the completion of all tests the graphite bearings were intact, not chipped, and only slightly worn.

c. Shim slider endurance test

The endurance test verified the dimensional stability of the system due to cyclic load and cyclic thermal conditions as well as satisfactory graphite bearing wear, friction, and reliability. For the endurance test, a shim slider was mounted on a shim way in a radiant heat oven (see Fig. V-46) with the appropriate loads applied. Four dummy control rods were used to simulate the normal friction and inertia loads for most of the testing. A dead weight over a pulley simulated the control rod pressure drop load for Mach 3.0 condition for almost half of the test. The slider was moved by a hydraulic actuator to simulate both positions and loads expected.

At the end of all testing, bearing wear ranged from 0.0003 to 0.0011 inch, averaging 0.0006 inch for all 16 bearings. Friction values did not change during the test. Room and low temperature ($< 800^{\circ}\text{F}$) friction results agreed with the basic bearing test. However, for temperatures above 800°F the friction values were much lower than the basic bearing test results, but higher than the advertised values.

During the test, the shim slider bearings were shimmed at room temperature to provide a 10 to 15-mil clearance between each bearing and the shim way. After several hundred feet of slider motion, all bearings showed signs of having approximately 20% average contact area. The endurance test was conducted under the following conditions:

Sinusoidal stroke mode: With the dummy control rods attached as the load to the shim slider.

1) Room Temperature:

- a) ± 4 inch stroke, 0.1 cps for one hour
- b) ± 0.5 inch stroke, 1.0 cps for one-half hour

2) Rising Temperature: Temperature of the assembly was increased linearly in the oven from room temperature to 1200°F in about 2 hours. (Long transients were used to minimize the temperature differentials caused by the radiant heat oven.) During this time, at 15-minute intervals, the following stroking was performed.

- a) ± 4 inch stroke, 0.1 cps for 6 minutes
- b) ± 0.5 inch stroke, 1.0 cps for 2 minutes

3) Elevated Temperature: ($1200 \pm 20^{\circ}\text{F}$)

- a) ± 4 inch stroke, 0.1 cps for 40 minutes
- b) ± 0.5 inch stroke, 1.0 cps for 16 minutes

4) Decreasing Temperature: Temperature of the assembly was decreased linearly in the oven from 1200°F to room temperature in about 2 hours. During this time, at 15-minute intervals, the following shaking was performed.

- a) ± 4 inch stroke, 0.1 cps for 6 minutes
- b) ± 0.5 inch stroke, 1.0 cps for 2 minutes

5) Change Load: The above conditions were repeated for ten complete thermal cycles. Alternate cycles used a constant load device in lieu of the dummy control rods.

Scram mode: The weight of the control rods was adjusted until the deceleration load on the slider at the end of the actuator scram was 1100 pounds.

1) Room Temperature:

- 40 inch scram, 50 times

2) Elevated Temperature:

- a) 250° F, 40 inch scram, 15 times
- b) 450° F, 40 inch scram, 15 times
- c) 650° F, 40 inch scram, 15 times
- d) 850° F, 40 inch scram, 15 times
- e) 1050° F, 40 inch scram, 15 times
- f) 1200° F, 40 inch scram, 15 times

d. Shim slider experimental stress analysis

The structural reliability and the location and magnitude of highly stressed areas of a type B shim slider mounted on a shim way was verified by testing. Static loads were individually applied to each control rod boss and resisted by a dummy rack connector. In the first part of the test the slider was loaded in compression and tension with an intermediate load (1000 pounds). Brittle lacquer techniques were used to determine direction and location of principal strains. This information was used to locate the biaxial strain gages that verified the stress magnitudes. Deflections were also measured.

In the second part of the test the slider was loaded in compression to the maximum capability of an actuator (3000 pounds). Results agreed fairly well with analysis. Distribution of maximum stresses for the 3000-pound compressive load conditions are shown in Fig. V-47. Impact stress results are reported below.

e. Shim slider assembly qualification tests

All four shim slider, bearing, and way assemblies finally installed in the reactor were proof-tested. Each assembly was tested for one thermal cycle of the shim slider endurance test conditions (see above). The first assembly completed the test quite satisfactorily. After the second assembly was tested, its upper bearings were deeply grooved, and the way had about a dozen small projections on the upper bearing surfaces and none on the lower. Examination of these projections revealed they were a nickel base alloy. It was theorized that these projections were a result of imbedded grinding particles in the bearing surface (finish was about 6 rms) or from loose small particles being melted in the radiant furnace. The second assembly was repaired and successfully retested.

The way for the third assembly was heated to 1200° F and cooled (but not stroked) and was still smooth. Extra precautions were used in cleaning the test assembly and apparatus; but after testing, the bearings were again grooved, and the way had about the same number of projections as the second assembly. This assembly was repaired and also successfully retested. The same precautions were used on the fourth assembly, and it completed its first test successfully.

f. Vernier slider structural test

A reject vernier slider casting with repair welds and crack, porosity, and void flaws in it was tested to verify stress analysis and to determine ultimate load capability and mode of failure. The slider was first loaded to 2500 pounds compression with eight strain gages distributed around it. The results agreed quite well with the analysis. The slider was next loaded at 80,000 lb/min in tension. It yielded at 18,400 pounds (0.2% offset), and essentially ruptured at 29,900 pounds. The flaws did not appear to influence the failure mode or magnitude.

g. Vernier slider operational test

A vernier slider assembly was tested in the LRL Hot Air Blowdown System (HABS) in conjunction with a vernier actuator and reactor control rod. The test consisted of 1148 cycles (actuator strokes) with a total operating time of 134 minutes. It included three thermal cycles with a run time of 53 minutes while the slider assembly was at 1200° F. The vernier slider operated successfully. The graphite bearings and Rene' 41 guide pin showed only slight signs of wear. See Ref. 111 for details.

h. Control system impact tests

A series of 23 impact tests was performed on the control system parts which were severely loaded during scram. The test configuration simulated the reactor assembly except that the pneumatic snubber was deleted. This configuration is quite pessimistic since it simulates an open or ruptured high-pressure air line to the pneumatic snubber with high-pressure air available to the scram piston.

The test consisted of accelerating the assembly to a peak velocity with a 6-inch-stroke Hyge machine, coasting for 6 inches, and then impacting on a solid stop. The assembly is shown in Fig. V-48. The tests were run in three phases: (1) vernier sliders at room temperature, (2) shim sliders at room temperature, and (3) shim sliders, coupling, and control rods at 1100° F. Data consisted of acceleration, peak velocity, kinetic energy, bolt elongation, and impact loads. Maximum peak velocities were three times the normal scram velocities. Impact loads were approximately 40 times the normal scram loads. Test details are reported in Ref. 112.

The test results were as follows:

- 1) Rack and snubber spring: No indication of failure was noted during entire test series.
- 2) Scram piston bolt: The original bolt broke at relatively low input velocities. The bolt was redesigned and withstood an average of four relatively high impacts without fracture. The bolt elongated approximately 0.1 inch per impact.
- 3) Slider coupler: No indication of failure in the coupler except for the rack bolt. Several materials, heat treatments, and designs were tested. Based on these results a new bolt was designed.
- 4) Vernier slider: After successfully withstanding repeated impacts it was concluded that the cast sliders were satisfactory for reactor use even though penetrant inspection showed minor cracks. The welded slider showed no signs of failure or cracks and were slightly better than the castings. Bearings cracked during testing, but remained completely serviceable.
- 5) Shim slider: Three sliders were tested with no sign of gross failure. After repeated impacts there were indications of cracks, but the structural reliability of the part was not impaired. There was some dimensional change, but the part remained completely serviceable. The graphite bearings showed no sign of failure.
- 6) Control rod: Rods did not buckle during 48-g acceleration and did not rupture on impact. There was some deformation at the slider connection, but the rods remained completely serviceable. There was no sign of cracks in the control rods.

i. Control system flow tests

The reactor control system was subjected to one cold and three hot (about 900° F) air blowdowns at full flow prior to mating the system to the reactor. Fourteen pitot-static probes were distributed on the dummy support grid to measure and verify the flow distribution through Duct 4.

The flow distribution results are shown in Fig. V-49. The range of values indicate good agreement between runs. The results indicate the flow in the vernier tubes was higher than expected, and therefore the temperature gradients during reactor runs are less than expected.

After the first hot air blowdown the control system was disassembled for actuator replacement. Many small, loose, metal particles were found in the duct at each constriction. The only damage noted was small nicks on the shim ways. The ways were then honed, and some of the severely scratched graphite slider bearings were replaced. Bearing clearances were not affected by the scratches. After the last hot air blowdown, just prior to mating the controls system to the reactor core, the bearings were inspected and found to be in excellent condition.

C. L-Ring Poison Rods

1. Description

Six L-ring poison rods are so named because they are located in tie rods L-1 through L-6, as shown in Fig. V-3. Their worth in reactivity is approximately that of six control rods. Thus the L-ring rods can be used as additional poison during transportation of the reactor and as an emergency shutdown system in the event of failure of the control system.

The L-ring poison rods are fabricated from 0.40-inch o. d. $\times$ 0.020-inch wall $\times$ 46-inch Hastelloy X tubing filled in helium with sixty-one-0.350-inch diameter $\times$ 0.75-inch high cylindrical enriched boron carbide pellets as shown in Fig. V-50. The tubing is welded gas tight. The downstream end of the L-ring poison rod is rounded to permit easy insertion into the reactor tie rods. The upstream end is a threaded attachment which connects the safety rod to the drive mechanism. Each rod weighs about 0.75 pounds.

The L-ring poison rods are fully withdrawn from the tie rods during reactor operation. They can be inserted 45 inches into the reactor at the end

of a reactor test. Insertion of the L-ring poison rods at full power would cause catastrophic failure of the reactor because the L-ring tie rods could not be adequately cooled.

2. Material Selection and Properties

Enriched boron (B^{10}) carbide was selected as the nuclear poison. The material has a high macroscopic thermal neutron absorption cross-section (about 92 cm^{-1}) and is stable at high temperature. Hastelloy X was chosen for the tubes containing the poison pellets because it has excellent weldability, with high strength and ductility at elevated temperatures. The oxide coating formed on heat-treated Hastelloy X has a low coefficient of friction in contact with cobalt or nickel based alloys such as Haynes 25, Rene' 41, and Hastelloy R-235.

Boron carbide (B_4C) has a melting point of 4460°F and a density of 0.09 lb/in^3 . At room temperature, it has a compressive strength of 414,000 psi and a modulus of elasticity of 65.3×10^6 psi. The tensile strengths of B_4C for several temperatures are:

<u>Temperature, $^\circ\text{F}$</u>	<u>Modulus of Rupture, psi</u>
70	50,000
1200	42,000
2000	35,000
2600	23,400

The coefficient of linear thermal expansion of B_4C is 2.5×10^{-6} inches per inch per $^\circ\text{F}$. The specific heat at room temperature is 0.22 Btu/hr ft $^\circ\text{F}$. The thermal conductivities of high density (2.5 g/cc) B_4C at room temperature and 1500°F are 18 and 8 Btu/hr ft $^\circ\text{F}$ respectively.

Boron carbide will change dimension when exposed to a high neutron flux for a long time, because the lithium and helium formed has large atoms which must be accommodated within the B_4C crystal lattice. Enriched boron carbide poison pellets were irradiated to 9×10^{16} thermal nvt in either air or helium environments in the Livermore Pool Type Reactor (LPTR) to determine if exposure to low integrated neutron flux would cause dimensional instability. The temperature of the pellet irradiated in air was between 1560 and 1650°F , while the temperature of the pellet irradiated in helium

was between 1345 and 1470° F. There was no apparent dimensional change in the pellets irradiated in either air or helium environments to 9×10^{16} nvt.

3. Temperature

Heat in the poison pellets comes from two sources:

Gamma from core	25%
Internal (n, α) source	75%

The heat in the poison rod tube is deposited from the interaction of external gamma radiation with Hastelloy X.

The maximum heating rate at the downstream tip of the poison rod in the Hastelloy X tube is 0.143 mW/ft³. The total internal heat generation rate in the boron carbide poison pellet is 0.153 mW/ft³. These heating rates are averaged over poison rod tip volume under normal steady state reactor operating conditons with the poison rods in their retracted position.

For this amount of heating, and a cooling convection coefficient of 130 Btu/hr ft² ° F, with inlet reactor air at 1065° F, the maximum temperature in the poison rod (at center) is 1110° F. The tube itself runs about 10° F cooler.

4. Force and Stress Analysis

The major force that acts on the poison rod during reactor operation is an external pressure force caused by the difference between the external pressure (346 psia) and the internal pressure (46 psia) on the tube. Hence, the net pressure acting on the poison rod tube is 300 psi.

The critical external pressure which will collapse the tube is about 16 times this value.

The outer fiber bending stress on a circular tube under external pressure is given by:

$$\sigma = \frac{r_o/t}{p - p_{crit}} \left\{ p^2 - p p_{crit} \left[1 - 6 \left(\frac{r_o}{t} \right) \left(\frac{\Delta r}{r_o} \right) \right] \right\}$$

where

r_o = tube radius

t = tube wall thickness

$$p_{\text{crit}} = \text{buckling pressure} = \frac{E}{4(1 - \nu^2)} \left(\frac{t}{r_o} \right)^3$$

P = gross external pressure

Δr = maximum initial deviation from a circle.

For a maximum initial deviation from a circle of 0.002 inch and an external pressure of 300 psi, the bending stress in the poison rod tube is 5690 psi.

The maximum velocity of the poison rod during insertion is 7.2 feet per minute. Thus the maximum kinetic energy of the safety rod during insertion is 1.7×10^{-4} ft-lb. The force in the poison rod required to absorb this energy is 7.04 pounds, should the poison rod be stopped instantaneously. The stress in the poison rod tube due to this force is about 295 psi. The maximum thermal stresses in the poison pellet and in the tube were found to be about 400 and about 800 psi respectively.

The poison rod stresses during normal reactor operation are small compared to the strength of Hastelloy X. Under normal conditions the safety factors in the circumferential and axial directions are 12 and 74 respectively. If the poison rod is inserted into the reactor core operating above some minimum power level, failure would occur. The poison rod would block the air flow through a tie rod, so the tie rod and poison rod would overheat.

5. Testing

The L-ring poison rods were tested as part of the overall L-ring poison system discussed in Section V-D. The test conditions are shown in Table V-7.

Table V-7. Number of insertion and withdrawal test cycles.

	Room temp	Temperature °F		
		365	735	1100
No load	685	-	-	10
Load	586	6	6	25

Each cycle consisted of full insertion into and withdrawal from a simulated tie rod. An axial 20-pound compression load was applied during insertion and a 40-pound tension load during withdrawal. Upon completion of the test the poison rods were visually inspected. No defects were found.

6. Fabrication

The enriched boron carbide pellets were formed by hot pressing boron carbide powder. The poison rod tube was welded and drawn from air-melted Hastelloy X. All metal parts were heat treated by soaking at 2150° F for 30 minutes and quenching in water. The tube was filled with the boron carbide pellets. The assembly was completed by attaching the threaded adapter and bullnose to the tube by means of TIG welding.¹¹³

A quality assurance program was followed to insure mechanical integrity of the safety rods. The safety rod tubes were inspected for compliance with Refs. 113 and 114. The tubes were inspected for defects, using fluorescent penetrants per AMS 2645 and eddy currents. Grain size and tensile properties of tube samples were determined by photomicrographs and tensile tests. After fabrication was completed, the poison rod was tested for leaks using a helium leak detector.

D. Poison Rod Drive System

1. Description

The poison rod drive system (Fig. V-51) is used to insert the L-ring poison rods in the reactor while at zero or low power levels. This poison system ensures safety for personnel or the reactor in the bunker area and during transportation on the NTS railroad. In addition, it provides checkout capability of each control actuator, singularly or in pairs, but not with simultaneous withdrawal of all control rods from the reactor. The poison rods are not to be inserted into the reactor when the reactor is at appreciable power. Accidental insertion at high power would result in gross failure of the poison rod and the tie rod, and consequently the reactor.

During reactor operation these rods are supported in the airstream just forward of the reactor. If used, they are inserted into the standard tie rods in the L-ring, as shown in Fig. V-3.

The drive system consists of six flexible cables routed through the duct wall (between the control actuators) to the poison rods. The flexible cable guide tubes are supported and aligned by the center support tube. By allowing the flexible cable to enter the tie rod during insertion, the poison rod length was minimized.

Each cable is driven by a control box mounted outside the forward reactor duct (Duct 4). A drive wheel within each control box is especially hobbled to mesh with the helix of the flexible cable. Three control boxes are on a common shaft with a dc-drive motor.

Accidental removal of the poison rods while personnel are near the reactor is prevented by a manually operated lock located on the drive assembly package. An orange pennant is attached to the lock to indicate whether the system is locked or unlocked.

The drive box is equipped with limit switches to prevent over-travel in both directions. A mechanical stop device on the drive assembly prevents over-travel in case of limit switch failure.

A potentiometer geared to each drive shaft together with a calibrated voltmeter, provides position indication accurate within 2 inches.

The poison rods are long enough to extend 45 inches into the reflected core. When inserted, these 0.400-inch diameter rods choke off nearly all the cooling air to these tie rods.

In the event one or two shim control actuators are inoperable, the following procedure would be used to prevent a major reactor failure:

Insert all operable shim and vernier rods. The reactor power will immediately begin to drop.

Maintain a high inlet air temperature to obtain maximum possible benefit from the negative temperature coefficient of reactivity.

Maintain a high rate of reactor airflow to cool the reactor down to inlet air temperature as soon as possible. (Inlet air temperature is a safe temperature for the tie rods.)

Reduce airflow to 100 pps when reactor temperature approaches the inlet air temperature. This reduces the pressure drop across the reactor and the resultant load on the tie rods.

Insert the poison rods.

Reduce inlet air temperature and complete reactor shutdown and cooldown.

Insertion of the poison rods while the reactor is at an appreciable power level is not a design condition. Increased temperature of the poison rods and tie rods will result, but failure will depend upon the maximum temperatures obtained. Safe insertion results when maximum temperatures

are equal to or lower than each part's maximum design temperature. No failures occur and the reactor can be rerun. Marginal insertion results when temperatures exceed design limits and result in minor failures. The poison system would shut down the reactor, but the reactor might not be reusable. Gross failure results when the temperature attained in the poison rod or tie rod causes melting or rupture of these components. The reactor could, but might not shut down under this condition.

The maximum temperatures obtained during insertion depend upon the reactor power level (flow rate and reactivity) prior to insertion and the transient characteristics of the power level with respect to rod insertion. Since the cooling air flow rates are almost negligible, the maximum temperatures are mostly a function of the transient gamma and neutron absorption rates.

2. Guide Tube and Brackets Design

Haynes 25 was selected for the guide tube because of its high temperature (1200° F) wear characteristics, and for its thermal expansion compatibility with the Hastelloy X center support tube. A heat transfer analysis of the guide tube shows that with a nuclear heating rate of 0.09 mW/ft³ a wall temperature of 1110° F results.

A structural analysis of the 0.410-inch inside diameter 0.060-inch wall tubing, with 70% area removed for cooling holes, was made. Anticipated loads were from cable buckling. For restrained ends and 3-inch spacing between supports, analysis indicated a load of 139 pounds would cause a 0.001-inch deflection of the guide tube. Subsequent testing revealed that actual loads are only about 1 pound.

The resonant frequency of the guide tube is 237 cps, which is far above that of the reactor core, the duct, and other adjacent larger components.

Differential expansion between the Hastelloy X guide tube brackets and the Haynes 25 guide tube was also evaluated. The expansion, based on 1100° F for the tube and 1040° F for the bracket, reduced the minimum room temperature clearance of 0.001 inch on the diameter by 25%.

Differential expansion (axially) between the Hastelloy X center support tube and the Haynes 25 guide tube was determined to be 0.250 inch, based

conservatively on a guide tube temperature of 1300°F and support tube temperature of 1100°F. This difference was used in spacing the guide tube and support brackets.

3. Flexible Cable and Fitting Design

A transient heat transfer analysis for the cable within the duct was made by utilizing an existing computer program.¹¹⁵ The spiral-wound cable was treated as a solid core with an equivalent diameter. The transient solution included nuclear heating, airflow over the conduit, radiant exchange between cable and conduit, and conduction through the air gap. It did not consider leakage air through the conduit or conduction due to the cable touching the conduit. Nuclear heating values used covered the range of 0.032-0.076 mW/ft³,²⁹ depending upon cable location.

Increasing the nuclear heating for the entire length of cable from the minimum to the maximum of the range increased the cable temperature 42.3°F. The maximum temperature of the cable interior for a typical reactor run is indicated to be 1150°F (Fig. V-52). A film coefficient of 300 Btu/hr-ft²-°F, from Fig. V-36, Curve 4a, was used.

Again, Haynes 25 was selected for the cable and fitting material on the basis of its wear characteristics. The high-temperature dry film lubricants are proprietary lead oxide base coatings of the cable and conduit vendor. The cable was sized for a 40-pound poison rod withdrawal load. This load represents the maximum possible load that could occur under the remote possibility that the poison rods would be withdrawn against full air flow.

4. Conduit and Conduit Bracket Design

A thermal stress analysis of the conduit and control box is described in detail in Ref. 116. The conduit within the duct was evaluated to have sufficient flexibility and curvature so that high expansion stresses are not thermally induced in that portion of the conduit. For the conduit exterior to Duct 4, between the duct and the control box, the analysis indicated that axial clearance is necessary. A slip fitting at the control box was incorporated in the final design. This reduced all forces and stresses to very low values. For the conduit between the control box and the square spent

cable tube, the analysis indicated excessive forces on the conduit fittings. These forces were due to the time lag in thermal expansion of the conduit caused by a heat shield between the conduit and the duct. The forces were reduced by placing springs between the spent travel tubes and their support brackets. These springs allowed the spent travel tubes to move slightly yet remain firmly in their proper location. The spent travel tubes have a square cross-section to prevent the flexible cable from rotating about its axis rather than move axially.

Haynes 25 was selected for the conduit within the duct and for the conduit between duct and control box, again, because of its high-temperature wear characteristics. The conduit between the control box and the spent travel tube was made of stainless steel because of its wear resistance at 500° F.

The aerodynamic stability of the conduit within the duct was analyzed. The frequency of induced vibrations due to the Karman vortices is 800 to 1100 cycles per second. The maximum force on the 0.425-inch o. d. conduit was estimated to be 0.87 lb/in.

The natural frequency of the conduit with cable running through it was determined experimentally. An instrumentation system consisting of a sensitive capacitance probe, a transducer power supply and preamplifier, and a dual beam oscilloscope, was used to determine the fundamental natural frequency. The conduit was plucked, and the resulting waveform observed on the oscilloscope. Excitation at frequencies other than the fundamental was accomplished by a speaker voice coil assembly attached to the conduit by a thin wire and excited by a beat frequency oscillator amplified by a public address type audio amplifier. Resonant frequencies were observed at 78, 205, 405, 420, 650, 790, 1040, 1230, and 1280 cps.

The conduits within the duct are supported approximately in the middle. Three conduits were supported by the bracket shown in Fig. V-53. Springs mounted on these brackets permit thermal expansion, yet hold the conduits snugly in place against inertia loads. This support bracket also serves to decrease the resonant frequency of the conduits by reducing the length between supports, and adds rigidity to the spans between duct wall and guide tubes.

Hastelloy X was chosen for the bracket material for complete compatibility with the Hastelloy X support structure upon which the brackets are mounted.

A temperature analysis of the spent tube storage tube was made in a manner similar to that made for the control box (see below). The model is depicted in Fig. V-54. Based upon a 5-minute, full power run, the spent storage tubes reach a maximum temperature of 450° F. Accordingly, stainless steel was selected for the spent travel tube because of its wear resistance and corrosion resistance.

5. Control Box Design

A simplified heat transfer analysis was made that included radiant heat transfer and nuclear heating. The control box was considered to be insulated. The model is depicted in Fig. V-55. A heat transfer analysis of Duct 4 indicates an equilibrium wall temperature of 1060° F, reached after approximately 10 minutes of warmup and 5 minutes at full power. The radiant heat exchange to the control box was computed assuming the duct at 1060° F and the control box at 300° F during the entire reactor run, including warmup. The nuclear heating value of 0.05 mW/ft³ for the control box²⁹ was also included. The maximum temperature arrived at in this manner is approximately 380° F. An emissivity of 0.30 was used for all surfaces, and convection was everywhere neglected.

Further analysis was required to determine the effect of leakage air on temperature of the control box. Air at 350 psig leaks from the duct through the opening between cable and conduit to the control box, and through leakage ports to atmospheric pressure. The effect that this leakage air might have on the carbon-graphite retainer in the control box backup was of concern, so a test was devised to determine the leakage rate and note the erosion effects of the high temperature air. As a result of these tests, six 3/32-inch holes were drilled in the conduit ahead of the control box. Tests on the modified system showed that only a negligible flow of air reached the control box.

Stainless steel was selected for the control box housing and all major components except for the carbon-graphite retainer.

6. Drive Assembly Design

The drive assembly is shown schematically in Fig. V-56. Its components are made from several stainless steels, Hastelloy X, Inconel X,

and Rene' 41 as appropriate from the standpoint of wear, temperature, and atmospheric corrosion.

Because of the differential expansion between the duct and mounting plate, the mounting plate had to be supported from a single point. The four bolts near the corners of the mounting plate provide gross lateral stability only. Clearances around these bolt holes provide for differential expansion between mounting plate and duct, and for positional tolerance of holes in mounting plate.

Shear stress, based upon a 4-g vibration inertia load, in the 1-1/4-inch diameter mounting bolt is less than 500 psi. Much lower vibration loads are expected. The support bolt screws into a boss which, in turn, is welded to Duct 4. The stresses in the boss and weld were analyzed and found to be negligible. Similarly, elsewhere stresses are very low and the minimum factor of safety is 3 for the splined drive shaft.

7. Drive Motors and Electric Circuit Design

A maximum temperature of approximately 250° F is computed for the electric drive motor, based upon a 10-minute warmup period and a 5-minute full power run. The analysis is conservative in that the same radiant heat rate was assumed for the motor armature as for the drive assembly support plate, whereas the armature should see very little radiant heat. The motor (class C insulation) is rated at 428° F for 5-minute operation and for 60 nonoperating minutes at that temperature. There is only a remote possibility that the motor would be required to operate at 250° F.

Electric drive motors, limit switches, position indicating potentiometers, and wiring were subjected to conditions far more severe than would be expected (see acceptance tests below). All components functioned satisfactorily with two exceptions:

- 1) An electrical connection to a limit switch broke due to fatigue from excessive bending after 300 room temperature cycles. The wiring was rearranged so that bending took place over a greater length of wire. No further wiring problems were encountered.
- 2) The position indicator failed after the first thermal cycle in the acceptance tests. Subsequent investigation indicated that this

particular potentiometer was mechanically defective. Subsequent testing verified the general reliability of these potentiometers.

8. Acceptance Tests

An extensive series of acceptance tests was performed. These tests were designed to determine reliability of the L-ring poison system components by simulating actual operating conditions including temperatures, loads, and cyclic conditions. The following considerations were of special interest:

The effect of cable buckling in the guide tube on wear, friction, and galling.

General wear and galling tendencies of the cable, conduit, drive wheel, and spent travel tube.

Tendency of carbon-graphite retainer guide in drive box to wear, chip, or break.

Quantity and effect of air leakage through conduit and location of leakage.

All parts of the system were tested at temperatures up to the maximums indicated below:

<u>Quantity</u>	<u>Description</u>	<u>Max temp, ° F</u>
1	Ganged control box assembly	400
3	Flexible cables	1100
3	Conduit within duct	1100
3	Conduit between duct and control box	400-1100
3	Conduit between control box and spent travel tube	500
3	Spent travel tube	500
3	Guide tubes	1100
3	Poison rods	1100
1	Complete drive mechanism assembly; consisting of motor, reduction gear, mechanical stop mechanism, position indicator, limit switches, etc.	400

The acceptance test procedure is summarized as follows:

- a) Perform 500 insertion cycles at room temperature.
- b) Disassemble and visually inspect cables, guide tubes, and control box for signs of wear, galling, chipping, etc.
- c) Perform 500 additional insertion cycles at room temperature.
- d) Reinspect as above.
- e) Perform 10 thermal cycles, each thermal cycle consisting of:
 - 3 insertion cycles at room temperature
 - 3 insertion cycles at 1/3 maximum component temperature
 - 3 insertion cycles at 2/3 maximum component temperature
 - 5 insertion cycles at maximum component temperature.

The last two thermal cycles are conducted without grease seals in the motor. In addition, the last thermal cycle is conducted without limit switches in the motor circuit, to test the mechanical stop.

Half of the acceptance tests were done at full load. No load was applied in the remainder of the tests. A full load consisted of a 20-pound compression load during insertion and a 40-pound tension load during withdrawal. Each insertion cycle involved a 65-inch travel of a simulated poison rod, in each direction.

The 1000 room-temperature insertion cycles were successfully completed. Motor current remained constant at 0.40 amp at no load, and 1.8 amps at 40 pounds tension load. Applied voltage was a constant 20 volts. No signs of wear were evident.

During the second thermal cycle the motor stalled at 30 volts and 3.4 amp. The short exposure to the high temperature had oxidized the coating. (The oxidation effect is analogous to adding sand to an oil or grease lubricant.) Further testing was done with the same coating except that it was burnished after application. The burnishing produced very little improvement. The lead oxide base coating was then tried. This coating had been used successfully by the vendor on numerous previous applications at 1100° F. It has a higher coefficient of friction than does the original selection in the unoxidized condition.

The lead oxide coated conduits were soaked at 1100° F for 2 hours before renewed testing began. The loaded maximum temperature cycles were spaced 1/2-hour apart so that the cable was soaked at 1100° F for over

2 hours. Two hundred and eighty additional cycles were run; 182 at no load, room temperature; 76 at full load, room temperature; 10 at no load, 1100° F; and 12 at full load, 1100° F.

After 1300 cycles, no signs of wear or other defects were observed in any parts of the system. Although the final torque, 800-in-lb., was almost twice that originally expected, the motor was capable of handling its load. The test results indicated that the motor should be operated at 50 volts rather than at 20 volts as originally planned. This will minimize the possibility of stall and reduce the insertion time to 46 seconds at the normal no-load condition.

Fig. V-1. Control rod.

Fig. V-2. Control rod safety strap.

- STANDARD TIE ROD
- L-RING POISON ROD
- * CONTROL (SHIM) ROD
- CONTROL (TIE) ROD
- ✱ CONTROL (VERNIER) ROD

Fig. V-3. Frontal view showing locations of poison rods, control rods, and tie rods.

Fig. V-4. Thermal properties of hafnium as functions of temperature.

Fig. V-5. Strength and elongation of hafnium as functions of temperature.

Fig. V-6. Dynamic moduli and Poisson's ratio for hafnium, as functions of temperature.

Fig. V-7. Mean coefficient of linear thermal expansion of hafnium, as a function of temperature.

Fig. V-8. Room temperature tensile strength of oxidized hafnium as a function of oxidizing temperature.

Fig. V-9. Hafnium blowdown specimen dimensions.

Fig. V-10. Static oxidation of hafnium.

Fig. V-11. Weight gain per unit surface area for hafnium (3 hours at temperature).

Fig. V-12. Nuclear heating of minimum reactivity control rod as a function of axial position.

Inlet total temp	1065 F	Exit Mach number	0.5187
Exit total temp	1435 F	Peak wall temp	1792 $^{\circ}\text{F}$
Inlet total pressure	342.3 psia	Power to air	20.33 kW
Exit total pressure	265.3 psia	Average material power density	14.171 MW/ft ³
		Flow rate	0.1910 lb/sec
		All dimensions at	60 F

Fig. V-13. Control rod (inserted 37.4 in.). Reactor structural design condition.

Fig. V-14. Typical reactor run.

Fig. V-15. Control rod loads due to tie rod bending.

Fig. V-16. Leading end control rod loads as a function of control rod position.

Fig. V-17. Control rod scram loads.

Fig. V-18. Control rod acceleration as a function of leading end deflection.

Fig. V-19. Control rod deflection as a function of leading end position.

Fig. V-20. Coefficient of friction as a function of temperature.

Fig. V-21. Thermal stress as a function of internal heat generation rate.

Fig. V-22. Thermal stress as a function of axial position of minimum reactivity control rod. (Fully inserted, full power.)

Fig. V-23. Failed control rod joint attachment.

Fig. V-24. Plastic deformation on leading end of control rod.

Fig. V-25. Stress as a function of position along a fully inserted control rod (end of scram with full power).

Fig. V-26. Safety factor, based on ultimate strength as a function of axial position.

Fig. V-27. Control system mockup structure.

Fig. V-28. Mockup in vernier tube area.

Fig. V-29. Overall guide system assembly, showing optical alignment method.

Fig. V-30. Alignment check of actuator support structure.

Fig V-31 Shim slider in picture above shown engaged with way below

Fig. V-32. Shim slider and way installation.

Fig. V-33. Guide system assembly.

Fig V-34. Center support tube area

Fig. V-35. Actuator support structure screen.

Fig. V-36. Approximate heat transfer film coefficients for various components forward of reactor.

Fig. V-37. Approximate heat transfer film coefficients for inside vernier tube and vernier slider.

Fig. V-38. Axial dependence of average nuclear heating density in nickel base alloys, upstream of support grid.

Fig. V-39. Average transient temperature, actuator mounting plate.

Fig. V-40. Average transient temperature, actuator mounting structure and midsection of center support tube.

Fig. V-41. Average transient temperature, midsection vernier support tube.

Fig. V-42. Cumulative wear vs distance traveled for 56HT graphite specimens sliding over various metal surfaces. Before and after oxidation.

Fig. V-43. Comparison of total wear for each type of graphite tested.

Fig. V-44. Variation of the coefficient of friction with temperature for 6HT graphite wear specimen sliding over various metal surfaces.

Fig. V-45. Shim slider vibration test setup.

Fig. V-46. Shim slider endurance test setup.

Fig. V-47. Experimental shim slider stresses (3000-lb compressive static load).

A SHIM SLIDER TEST

B VERNIER SLIDER TEST
SLIDER ASSEMBLY

Fig. V-48. Control system impact test assembly.

Fig. V-49. Airflow in vicinity of reactor controls as measured by 14 pitot-static probes (small solid circles). Values are fraction of average air velocity based on runs 78, 79, 81, and 82. View looking upstream from Sta. 338.0.

Fig. V-50. L-Ring poison rod assembly.

Fig. V-51. L-Ring poison rod system.

Fig. V-52. Cable interior temperature during a typical reactor run.

Fig. V-53. Conduit bracket.

Fig. V-54. Spent tube storage model.

Fig. V-55. Control box model.

(A) ELECTRIC DRIVE MOTOR	(L) GEAR SHAFT EXTENSION AND RETAINER (LOWER)
(B) GEAR SPEED REDUCER	(M) END SUPPORT BRACKET
(C) MOTOR-REDUCER MOUNTING BRACKET	(N) LOWER "V" SLIDE
(D) COUPLER-MANUAL SAFETY STOP	(O) UPPER "V" SLIDE
(E) GANGED TELEFLEX CONTROL BOXES	(P) MICRO-SWITCH SUPPORT PLATE
(F) AUTOMATIC SAFETY STOP, DRUM	(Q) SYSTEM MOUNTING PLATE
(G) POTENTIOMETER	(R) CONTROL BOX MOUNTING BRACKET
(H) POTENTIOMETER COUPLER AND SHAFT EXTENSION (UP)	(S) MICRO-SWITCH SUPPORT BRACKET (2)
(I) POTENTIOMETER MOUNTING BRACKET	(T) MICRO-SWITCH (2)
(J) POTENTIOMETER GEAR, THIN	(U) SPLINED SHAFT
(K) POTENTIOMETER GEAR, THICK	

Fig. V-56. Drive assembly.

VI. REACTOR ASSEMBLY

A. General

Although the reactor core has been described in detail previously, it would be well to review its geometry, for better understanding of assembly problems.

The ceramic components are arranged in the shape of a circular cylinder contained in a circular duct. Within the duct, the core has two support systems; axial and transverse.

The axial pressure drop load is transmitted to the duct via a support grid, by means of superalloy tie rods which extend through the ceramic core and secure to base blocks (Fig. VI-1). Because each periphery base block is secured by 2 tie rods, there are 121 tie rods for the 91 base blocks.

A transition cartridge is provided in the aft reflector to reduce every seven fueled tube flow channels to one channel through the base block. Figure VI-2 shows a cartridge being installed. Transition pieces were not used in the side reflector region, so the periphery base blocks have both large and small diameter holes.

The convenient geometry of the base block profile was extended to the ceramic core, resulting in an array of parts known as unit cells. A unit cell is a system of flow channels, (ceramic parts in cross section) with 18 ceramic parts in each channel stacked longitudinally on a base block, and a tie rod in the center of the cell. There are 3 unfueled parts at the inlet end, 14 fueled parts in the center, and one unfueled part and the transition cartridge at the exit end.

The average ceramic part is 4 inches long, a convenient fabrication length. However, there are three other fueled lengths and three different unfueled lengths, making seven lengths in all. In addition, while fueled tubes have a 0.227-inch diameter hole, the unfueled parts have 0.227-, 0.130-, and 0.093-inch holes, depending on where the part exists. A system of letters designates length, and, in the case of unfueled parts, a number designation for hole size is used (Fig. VI-3). These geometrical designations, fabrication batch numbers, and fuel loadings in grams of or alloy per inch are printed on three of the six flats of the hexagonal tubes.

To avoid shear planes at tube interfaces, a three-tube stagger system was used with approximately 1/2 inch between adjacent tube ends (Fig. VI-3). Although the tube pattern is broken up at the transition zone the stagger is still needed, so a two-length cartridge system is used. Thus, any tube column can be identified by its unit cell number, the stagger number (1, 2, or 3) and the cartridge number (A or B). A tube can be located by an additional number for its position from the forward reflector interface, beginning with 1. Tube stagger arrangements can be seen in Fig. VI-4.

The transverse support system at the periphery is also called the side support system. To prevent the core from opening up under maneuver side loads (missile system) a radial pressure was needed. The design adopted consists of 1020 nickel shims which terminate the ceramic core and are machined to fit the tube profile. The average shim length is 4 inches and there are 60 in a layer. A corrugated spring 2 inches wide by 0.150 inches thick with 1.00-inch high convolutions engages the nickel shims and exerts a radial pressure through the shim into the core. The core is supported within the duct by attaching to the duct wall a series of rails (Fig. VI-5) which engage the spring at every fourth convolution. There are 30 such rail sets, each longitudinal set made up of 17 short rail sections, to provide support when the duct is in any circumferential position.

There were other items which complicated assembly of the system. For example, 140 thermocouple junctions had to be embedded in the ceramic parts within the core, with about 20 more on the duct and in the side support region. In addition, to grade the pressure drop in the annulus between the duct and the core to more nearly match the fueled tube pressure gradient, 90 wiper bars with sheet metal springs were provided. The springs ride on the corrugated springs and impede the airflow. At the aft part of 60 of the wiper bars, heat shields (forming the shroud) extend from the base block to within 3 inches of the aft reactor duct end face (Figs. VI-6 and VI-7).

B. Ceramic Assembly

The principal assembly objective was to insure that the reactor core would be correctly loaded. A major complication was the fuel loading

design, which provided for both axial and radial grading of fuel, to obtain a constant radial temperature at any station. After considering possible techniques, it seemed most appropriate to build the core horizontally in modular form, a proven (Tory II-A) method to which the unit cell concept would lend itself readily. The horizontal attitude was chosen because the technicians could work more readily, and it presented the flat of the hex in a stable position. In this position the tubes could be placed easier, and each layer could be photographed from above. It was this decision which later led to the printing of all of the pertinent information on the flat of the tube.

To build unit cells horizontally, drawings had to be prepared, showing each layer in the cell, and all of the geometric and neutronic information necessary to assemble that layer. There were 3500 such layers, each differing from the others by either geometry or loading, or both. The series of layer drawings constituted the 91-chapter map book; each chapter being a complete unit cell. All of the geometrical, hardware, and thermocouple information was worked into these layers. Examination of the cross-section map showed that in the later stages, when the cells would be fitted together vertically, a sawtooth engagement would exist. Therefore, certain tube columns were left out, to be placed after adjacent cells were together (in a vertical attitude).

The front reflector/fuel interface presented a new problem. A shear plane existed here except for special nickel tubes placed in 1800 scattered columns. These tubes, 1 inch long, extend $1/2$ inch on either side of the plane. Also, the hardware surrounding the tie rod has the first piece (reflector insert) extending $1/2$ inch on either side of the plane. The thin rings and long tube reflector inserts above the first piece resulted from consideration of the interaction of the cold reflector and the hot core. Because the core would expand more than the reflector, a tilt zone would be required. Notice in Fig. VI-4 that the 3.925-inch long tubes in the front reflector are constant across the core. These tubes tilt during operation because the fueled tubes on the right are hotter, while the ones to the left are at the same temperature. The nickel tubes and reflector inserts pin the two shear planes, but permit the tubes between them to tilt.

Completed unit cells were to be assembled vertically. All the unit cells would weigh about 12,000 pounds, and could not be handled at once until secured in the duct. This posed the problem of getting the assembled unit cell from the horizontal to the vertical in a stable condition for handling. To obtain this stable condition, round rods were placed in periphery tubes and extended through a plate at the top to the base block. The map book indicated where these side support rods were to be placed. This support system presented problems, however. The side support rods could not pass through either the cartridge or the base block because the base block profile only approximated the cell profile. (The large hole resulted in a smooth scalloped base block profile.) A side pressure was needed to keep a tight unit cell during subsequent handling, so the side support bars were cambered slightly. To maintain proper orientation, a special assembly base block and cell plate system was needed.

The assembly cell plate adopted was a four-layer system. The first plate served as a stripper plate when the side support rods were pulled out of the core after cell placement in the core. The second plate served as the puller for the rods. The third plate was the orientation plate, and the fourth plate was a keeper. The rods had a 2-inch camber over the 63-inch length. At the lower end, the rod was stepped down to go into holes in the base block. The upper end had a flag brazed to it to engage the orientation plate and keep the camber such that the rod would be straightened by the main body of tubes exerting an inward force on the cell.

After each map book chapter had all of the geometric information required on all of the layers, the fuel loading information had to be transferred from the cross-section maps to the map book. In Fig. VI-8, the man in the foreground is using a straightedge on a piece of the cross-section corresponding to the layer he has before him. Each column is numbered so that direct correspondence can be maintained. He is writing letters for loading identification rather than the actual loading value. The use of letters permits master map book pages to be used for future reactors which have the same loading boundary geometry, even if the actual fuel content were to shift in value. The man at the upper left is checking the master map book page before prints are made. The man at the upper right is checking the prints after the corresponding fuel loading has been stamped

according to the letter information appearing on the print. The little card in his left hand is the complete loading key, relating letters to fuel content. Tube length identification is also printed on each layer page, so that the operator will have that information also. The cabinets contain the 91 binders for all of the map book chapters.

Unit cell assembly was performed in six hoods specially designed for the purpose. A view of the assembly area is shown in Fig. VI-9. Each pair of hoods had one jib boom for removing the cell from the hood, and two camera booms for taking photographs of each layer after assembly. Each hood had 1350 cfm of air exhausted through a common system, to obviate the beryllium toxicity problem. Unfortunately this resulted in a rather high noise level, making conversation, even between adjacent boxes, quite difficult. To minimize distractions, individual hood communication systems were used. The clerks, on the high platforms, had the appropriate map book chapters before them. Operators had duplicate copies of the appropriate layers before them at the hoods.

The procedures developed concentrated on eliminating errors in the construction of the cell. After the operator had placed girth cords in grooves in the assembly tray, the clerk would tell the operator what fuel class and how many were needed at the appropriate station. The operator would take the parts from the bin in the rack in front of him and read the class and length designations and the batch number back to the operator. While the parts were being placed in the layer, the clerk would record the batch number on his copy of that layer page. This cross talk served to keep the assemblers alert, and to insure that all batches were logged as used. In addition, from his vantage point, the clerk could see where the parts were taken from and placed, as a visual check.

After a layer was assembled, the floor supervisor would be called to the hood. He would remove the long clip board from in front of the operator and place it next to the tray. He would check the layer against the page and then take a photograph of the layer by means of an electrical switching system. The photograph would show both the layer and the map book page.

The 35-mm cameras used were bench focused and fixed in that position. Lighting was supplied by a specially developed power pack and strobe lights. Light intensity was checked with a light meter for each

photograph. High contrast copy film was used, and the slides were reverse projected (Fig. VI-10). Each slide was carefully scanned for assembly errors, using the map book print as the master.

As cells were completed the girth cords were tied, to keep the bundle close-packed, and the cell was tilted vertically by rotating the tray (Fig. VI-11). The jib boom hook was designed to couple with the assembly tie rod nut, and was spring loaded to permit gentle handling. Figure VI-12 shows a suspended cell being lowered into a plastic bag used for containment. The multi-layered cell plate is visible at the top of the cell. The three eye bolts extended to the third lamination, and were later used to pull out the side support rods. The forward reflector is clearly seen, and the nickel tubes and the fueled core/reflector interface are obvious. At the bottom of the cell the engagement of the aft reflector and the fueled tubes can be seen. The metal tubes next to the assembly base plate were to provide the hole size required for the side support rods.

All the transition cartridges used in this part of the assembly were later replaced with actual reactor hardware (Figs. VI-1 and VI-2). In the box in the foreground are the assembly "clusters," made up of Tory II-A parts and metal tubes. To provide an exact replica, and to avoid excessive tolerance buildup, BeO tubes were cut to size and cemented together in jigs (Fig. VI-13). The various types may be seen in the prints hanging on the glove box. The technician is shown checking a cluster on the air gage. As most of the Tory II-A tubes did not have holes large enough to accept the side support rods, and as their quantity was limited, metal tubes were used in all peripheral clusters for the support rods. The ceramic tubes provided a stable matrix and could be cut to precise lengths, so that the metal tubes were only fill-ins which accommodated the side support rods.

Errors found in the film reading were corrected in the cell assembly area. The cell was placed in a jig which then rotated the cell to the horizontal. Figure VI-14 illustrates this process. Notice the assembly base block which would be removed if a correction were being made. The device on the left is a tube extractor which is inserted down a column. When the spring loaded center rod of the extractor is released, a rubber grommet is compressed, increasing its diameter. The tube, in resisting this expansion, is secured to the extractor by friction between the grommet and the inside surface of the tube. The column can then be withdrawn or inserted.

The cells were stored in aluminum cans on platforms seen in the background of Fig. VI-14. Six cans were secured on each of fifteen platforms. The entire unit was removed from the cell assembly area and transported to the reactor assembly area when called for.

The arrangement of six completed cells on a platform did not present a criticality problem. However, because the core itself contains 63 kilograms of enriched uranium oxide, the core was assembled in the containment vault in which the critical experiment would be made. The 'six-pack' was transported by truck from the cell assembly area to the core assembly vault.

The cells remained on their respective platforms until needed, except for the determination of base counts. Each cell was removed from the platform, still encased in the aluminum can, by a mobile floor crane. The cell was taken to a shielded area and a BF_3 counter was used to determine the cell base neutron count (Fig. VI-15). At specific core configurations, the base count sum of the assembled cells was compared to the count of the assembly. The core multiplication was determined by dividing the count of the assembly by the count of the sum.

The reactor assembly table was in a 30-ft $\times$ 30-ft pit 10 ft deep, in the center of the vault. A platform was placed over a plenum which exhausted an enclosure in which the core was built up. The table height was a function of the distance from the base blocks to the aft flange of the reactor duct. This allowed subsequent installation of the duct. A large blower exhausted the enclosure. Because no method was feasible to exhaust the blower to the atmosphere without violating the integrity of the vault, the blower exhaust emptied into the cell itself after passing through an absolute filter.

A large circular plate (Fig. VI-16) on six shock-mounted columns served as the table. This arrangement provided a semi-rigid support which allowed the table to be vibrated after all the unit cells were in place. In the center of the table were seven precisely located buttons. From the center hole, six radial center lines were scribed on the table surface. On these lines various tangential lines indicated the theoretical positions of the various cell rings. Figure VI-16 shows the first cell being placed. The crane inserted the cell through an accordion door. (Before placement the plastic bag was stripped off.) A probe was lowered through a housing secured to the crossed beam structure seen above the enclosure and below the scaffolding.

The probe accurately located the top of the cell over the center button of the table. The probe end was machined for a snug fit with the tie rod nut, and the probe came to rest against the cell plate itself. Notice the filters in the enclosure sides. When all doors were closed, these filters admitted air to the exhaust system. With the doors open, they essentially acted as part of the wall. Normally only two doors were opened at one time to insure adequate ventilation through the open doors. The enclosure was supported on six pulleys which were guided on a track cut in the platform center plate.

Figure VI-17 shows the first two cells in place, and Fig. VI-18 shows the center seven cells. Installed on the probe housing is a Christmas tree arrangement of eyes to which all of the remaining cells will be attached. Note the special nut at the top of the second cell (right side) in Fig. VI-17. A special counting probe was inserted in the assembly tie rod of this cell for use as an internal neutron monitor. The external counter rested directly below the main assembly table. Failure to nest was easily determined by visual inspection, and each cell was manipulated until no gaps were visible at mating surfaces.

At times, to expedite assembly, two cells at a time were placed. Because of the sawtooth nature of the mating surfaces, full tube columns were left out during the cell assembly process. Placing two cells at a time (twinning) often eliminated the need to leave out columns. Figure VI-19 shows this twinning process. The first cell was brought into the enclosure, and after the bag was stripped off, the cell was secured to the enclosure lid by means of the snap hooks shown above the cells. The second cell was brought in, the bag stripped, and the girth cords cut. The second cell was nested against the first, and a bar was secured to the two tie rod nuts. Screws in the bar, acting horizontally against the nuts, forced the cells together at the top. The floor crane then lifted both cells by the bar. A second thin bar was secured to the base blocks. It was necessary to move the assembly up and out next to the door to install this base block bar. Figure VI-20 shows the assembly base block region at a later stage, immediately after the reactor duct had been tilted horizontally. This picture shows the interlocking profiles of the assembly base blocks and the twinning bars.

The pie-shaped segment of the enclosure lid (Fig. VI-18) provided access for removal of the side support rods. Three screws secured the second and third assembly cell plates to the fourth. The first plate was secured to the system by eyebolts whose studs were screwed into the second and third plates. Thus, when the three screws were removed, the top three plates were free of the cell, leaving only the keeper plate. A grappling hook arrangement attached to the overhead crane was lowered into the pit and through the enclosure opening. A plastic bag was placed before the hooks were secured so that the rods were enclosed at all times. The rods in a cell were removed once the cell had been surrounded by all of its neighbors. Neutronically, it was desirable to remove the rods as soon as possible, so that the true worth of the system was always known.

The completed ceramic core is shown in Fig. VI-21. Notice first that all of the rods have been removed. The last plate of the assembly cell plate sets remain, as well as all of the tie rod nuts which kept the ceramic tubes down during the stripping process. Particular notice should be given to the peripheral cell plates which have two systems of circumferential holes. The inner set of holes corresponds to the last ring of fuel tubes. Full size side support rods were placed here for safety reasons. In the very small holes in the side reflector region were placed needle rods, the ends of which can be seen in Fig. VI-22. These small rods also occur on the sides of the cell into the fuel zone, but were not very useful for containment purposes. Great care had to be exercised to keep these small hole columns in position during handling. To aid in this problem a system of aluminum shingles (Fig. VI-21) was placed on the outside surface of the cell. These shingles provided a more stable system and also yielded a nonceramic periphery.

Banding strips were used to girth the core before removing the stripper cell plates and nuts, exposing the ceramic face of the core. The outside cell plates secured the tops of the shingles. The missing key columns were placed at this time. Fueled tube columns were inserted using the tube extractor mentioned previously.

The reflector tube columns presented another problem because the hole was so small the extractor could not be used. Instead, the tubes were strung on copper wire. The bottom end of the wire was melted to form a globule large enough that the tubes would not pass. The strung columns

could then be pushed into the open space; with considerable difficulty, however. The matrix was very loose in the periphery, and an adjacent tube slightly out of alignment would catch on the column being installed and hang it up. The process would have gone smoother had the core been squeezed before these columns were placed.

A 58-inch micrometer was used (Fig. VI-23) to determine the core diameter. As expected, it was found to be $1/4$ inch oversize. Before the side support system could be installed, it was necessary to reduce the core diameter. Compression rings, each containing a rubber tire, were lowered over the core (Fig. VI-24), and rubber sheets were used to fill in between the core and the rings. The tires were inflated to 15 psi, and the table was vibrated at a resonant frequency, using pneumatic units attached to the table (Figs. VI-21 and VI-25). The tire pressure was released and then brought up to 20 psi. While in this condition (but with the vibrators turned off) the core was rebanded, as the first bands were now very loose. Upon removal of the compression rings, the core diameter was found to be only $1/32$ inch oversize; within allowable limits.

C. Hardware Assembly

The side support springs were assembled on two tables adjacent to the core assembly area. An internal ring provided support hangers for the six spring sections, (which form a complete circumferential spring) and held the assembled sections in slight tension to keep the dovetails mated (Fig. VI-26). An external ring consisting of guide V-blocks and screw-actuated fingers was lowered over the spring, with the fingers passing down through the corrugations (Fig. VI-27). The fingers were moved radially inward, and cam locks closed the fingers on the spring. The screws then pulled the fingers back, expanding the spring radially. Sequencing was important, so as not to overstress the spring in expanding it. The diameter was increased to $1/2$ inch over the free position; about $1/8$ inch over the expected installed dimension. Some drag was expected in lowering the spring over the shims, so the ring also had adjustable push fingers which were set against the top of the spring on every corrugation.

During the spring expansion process the lower set of aluminum shingles was removed from the core. The first set of nickel shims was

placed and held in place by a large rubber band, after which the rectangular shim keys were installed. These keys were between 5 and 10 mils undersize, so shim stock was used to attain the desired fit.

A large dolly was used to raise the spring expander ring over the enclosure and to lower the ring onto a pull-down ring (Fig. VI-28). This pull-down ring was supported on three columns which were moved vertically by three chain-driven acme screws. (The ring had been leveled and raised to the shim level.) The assembly was moved downward slowly until the spring rested against the supports on the first shim blocks. The screws were loosened, releasing the spring against the shim, and the expander ring was removed with the dolly and returned to a table to receive the next spring. The second spring, however, had its final position at the joint between the first and second shim layers. The spring was pulled down until it was 1/4 inch above its final position. This space was to allow the lower finger of the pair to release. The next shims were placed and the keys installed and shimmed as before. The shims were then spaced vertically, to allow for thermal expansion during the reactor operation. Two campher wafers, 0.015 inch thick, were used to space each shim block (Fig. VI-28). The spring was then released against the core and the push fingers (between the pull fingers) reset against the top of the spring. The pull-down ring was then pulled down by the columns until the spring was 0.010 inch above the first spring. Figure VI-28 shows the shims and springs built up almost to the front reflector. Notice the aluminum tabs which insured a 0.010-inch gap between springs. (Again, to allow for uniform thermal growth.)

The side support system in the tilt zone required the use of dumbbell-shaped keys, to interlock the shims (Fig. VI-29). The entire assembly of interlocked shims for the tilt zone, was assembled on a jig (seen behind the shims in Fig. VI-29) which simulated the core. Spacers between the top and middle shims stabilized the shims, and Fig. VI-30 shows the last shim set just before it was pushed in radially to close the loop. An L-shaped shroud in three segments was placed on the outside of the shims, and the shroud segments were locked together. Then spacers were pushed down between the shims, forcing them out against the shroud, making the assembly slightly larger than the ceramic core. The assembly was then placed over the core and lowered until the shroud could be

supported on the pull-down ring. The dumbbell shims were pushed against the ceramics, the shroud removed, and the next spring placed in the usual manner. This spring embraced the lower dumbbell shim and the shim below it. The next spring was a short spring and ended even with the top of this lower dumbbell shim. The campher wafers were then installed. The next two springs were standard length, which put the last of the pair at the top shear plane. Another set of wafers was placed. Notice in Fig. VI-31 how short the upper dumbbell shim is. The next spring was a standard length, and covered the joint between the upper dumbbell shim and the next shim which is standard length. The last shim was also standard, but the last spring was short to end the core. In Fig. VI-31, the needle rods can be seen still in place, to keep the peripheral ceramic tubes in position. After the last spring was installed, the needle rods were removed (Fig. VI-32).

The gaps between springs in the tilt zone area are larger than elsewhere in the core. This may be seen in Figs. VI-32 and VI-33, which also show the many thermocouple leads protruding from the core. Note in Fig. VI-33 that standard operational cell plates have been installed, replacing assembly cell plates. Peripheral plates were installed after the core and the duct were mated.

At this point in the assembly, shim skirts were secured with screws to the aft (bottom) shim. The skirts provide a means of sealing the side support region cooling air from the base block region. (This air is needed to cool the duct.)

The next phase of assembly was to place the duct over the core. The process was complicated by the rail system on the inside of the duct (Figs. VI-5 and VI-34). Each rail was designed as a "leaf spring," with a stud in the center of the rail passing through the duct wall. When the nut was torqued down, it deflected the rail by pulling against the feet on either end. The deflection positioned the rails so that they behaved as integral parts of the duct. The tongue and groove interlocking of the rails can be seen in Fig. VI-34. (Wiper springs have been bolted to the top of the rails.) A thermocouple to monitor the duct temperature next to the rail feet can also be seen in the figure.

The tolerances of the side support system dictated the use of careful procedures to align the complex segmented rail system. Accordingly, a

male-female gage system was devised to set the rails in the duct according to the as-built core. The male gage (Fig. VI-32) was raised and lowered and the bars set so that it just passed through the spring convolutions. It was then mated with the female gage shown in Fig. VI-35, and the female gage was set to it. The female gage was lowered into the duct after all of the rails had been assembled but not torqued. As the gage passed down, the rails were snugged. A transit was used to make sure the gage did not twist and that the rails were plumb. When the gage could pass freely, the segments were all torqued and checked again.

Figure VI-36 shows the duct just prior to mating with the reactor. On the lower flange of the duct are two machinist's levels, mounted at right angles to each other. On the upper flange, the lifting ring is secured to a precision hydraulic jack. This unit was used to indicate load, and to move the load up or down in very small increments. The dial indicator on the left of the unit (Fig. VI-37) measured the shaft movement.

Once the rails had engaged the spring convolutions, the levels and the balance system indicated changes in attitude or drag between the core and duct. The large pipe in the foreground of Fig. VI-38 was a safety stop, to prevent more than set increments of descent at a time. Because the duct adds reactivity due to its reflecting effect, sudden increases were obviously undesirable. The smaller rods were intended to pull down the duct if necessary, but they were not needed.

The thermocouple system was assembled in conjunction with the axial support grid installation. The support grid was set on stands above the duct flange, and a screen and its support ring was laid on the grid (Fig. VI-39). Notice in Fig. VI-33 the short wires with cylinders about 1 inch long attached. Each cylinder is half of the mechanical coupler device used to join primary coaxial thermocouple wires to stainless sheathed extension wires which extend across the grid and out the duct wall. The wires were routed through the proper support grid passages and out through the screen. The connectors were mated, and the leads checked electrically. During the process the assembly tie rod extension bars were placed, together with the axial pressure springs. (These springs put 130 pounds force each between the support grid and the core. The springs were pre-compressed, and keepers were used until the grid was placed and locked.)

When the extension wires had all been secured and all the rods and springs placed, the grid was lowered and secured in position against the duct flange by 12 radial pins. In some areas the grid was deflected by tightening up on the tie rod nuts enough to get the radial pins in. After all of the pins and tie rod nuts were installed, the thermocouple leads were routed across the grid and out through holes in the duct wall (Fig. VI-40). In Fig. VI-37, the leads may be seen coiled around the duct, with a few of the grid pins below them. The unused holes in the duct were for instrumentation installed later.

The lift fixture was installed as for the duct installation, and the duct assembly was eased slowly away from the main assembly table, using the hydraulic jack control. A support ring was secured to the aft flange of the duct after removal of the duct from the pit. This ring was used to support the assembly in the duct handling vehicle cradle. Figures VI-41 and VI-42 show the duct being placed in the handling vehicle. The cradle cap in the right center was closed after the ring was supported on vertical support rollers (cradle vertical).

After the duct was secured to the cradle, the large ball screws were used to tilt the assembly to the horizontal position (Figs. VI-43 and VI-44). BF_3 counters were secured to each horizontal beam of the cradle assembly, to monitor the reactor during the remaining assembly process.

The next process was operational tie rod and base block assembly. Because the assembly tie rods were contaminated with BeO , a hood was attached to the forward flange of the reactor duct (Fig. VI-45). Figure VI-46 shows pipes attached to the hood to house the assembly tie rods as they were pushed out. Control tie rods differed from standard tie rods in that the forward end of the control tie rod had interrupted external threads. These threads just fit in the space left in the matrix by the missing ceramic tubes.

The base blocks and the tie rod insulating cups used with control unit cells were designed to provide clearance passages for these thread sections. Hence, the control tie rods could also be inserted axially. Assembly of the insulating washers and cup on a control tie rod is shown in Fig. VI-47. The outside diameter of the standard tie rod is constant except for the aft flange. Therefore, the cup and washers were of simpler design, and were easier to install.

The next step was to place a poison rod in the operational tie rod assembly. Because the poison rod itself was core length (60 inches), a piece of threaded bar stock was used to make it the tie rod length (78 inches). A screw secured a button to the poison rod, and a shoulder on the button fit into the tie rod end. An adapter was screwed into the assembly tie rod, and after the base block was placed on the tie rod (Fig. VI-48) the operational and assembly tie rods were joined by screwing the poison rod into the adapter. After the connection was made, the operational tie rod was pushed and pulled (gently) into the core. Just prior to the end of tie rod travel, the base block was fitted into its neighbors.

At this point, assembly was complicated by the fact that the operational transition cartridges were not yet available. Perforated aluminum screens were placed between the base block and the BeO metal tube paste-up cartridge to prevent loss of these pieces because of the large hole in the base block. The special transition rings around all control tie rods were available, however, and they were installed (Figs. VI-49 and VI-50). Notice the poison rods protruding from the assembly tie rods, and the buttons on all of the operational tie rods, indicating poison rods in place. The core was poisoned at all times by 119 of the 121 possible poison rods. Figure VI-51 shows the support grid with all of the operational tie rods and nuts in place. Nylon buttons were used to lock the poison rods in place at the grid.

Assembly operations were interrupted at this point for cold critical experiments, which confirmed closely the neutronic predictions. The only result affecting assembly operations was that all Hastelloy R 235 tie rods were replaced by Rene' 41 tie rods, for adjustment of core criticality. (This was an anticipated change.) The tie rod placement process was reversed to remove the R 235 rods, after which the Rene' 41 rods were placed.

After the experiments were over the cartridges were exchanged. Figures VI-1 and VI-2 show this process in the final stage in one cell. The aluminum screen has been removed, all of the BeO metal tube clusters have been removed, and the cermet transition cartridges placed in all but the center area. The tie rod was pulled back by placing the assembly tie rod extension and the nut coupler on the front of the operational tie rod. This allowed the base block to be pulled back 14 inches. The tolerances were such that the assembly pieces came out a little hard, but the operational

cartridges went in easily. When all of the cartridges had been installed in each cell, the base block was replaced and the tie rod reset.

A tool consisting of a T-handle with an outside sleeve was used to torque the tie rods. The handle had a protrusion which entered a slot in the tie tube. The handle was used to twist the tie rod for alignment. The sleeve had two diametral pins which engaged the tie rod nut. The process was one of gradually tightening the nuts in sequence across the whole face of the grid, until all of the axial compression spring keepers but 11 had been freed, and all of the nuts were in positions to secure the set screws. On the control tie rods, six positions were available corresponding to the cut-away sections of the threads, but the standard rods had only one machined flat. The nut had a flange on the grid face with four cut-outs, and the grid had one tapped hole. The torqued assembly, therefore, had to be such that one screw in the cylindrical part of the nut corresponded to the flat on the rod, while a flange cut-out corresponded to the tapped hole in the support grid. The procedure was obviously one of trial and error. The axial compression ring keepers were removed as they were freed. Eleven keepers were retained so that the core could not move with respect to the grid during transit. These keepers were later removed at the Nevada Test Site.

A junction box system was secured to the duct just aft of the forward flange (Fig. VI-52). The thermocouple wires were passed through packing glands and the glands screwed into the duct wall. The leads were then routed to the various boxes and prepared for the terminal strips.

The last operational items to be placed in the duct were the wiper bar assemblies. These assemblies were of two types. One type was simple, consisting of a bar with single wipers like those on the rails in Fig. VI-34. In Fig. VI-6, the more complicated assembly is shown. The bar with the single spring was used as a base. Sheet metal standoffs and wide U-shaped springs were secured by two screws to the outboard side of the bar (Fig. VI-7). In addition, the shroud parts were attached to the aft end of the bar. In Fig. VI-5, note that the left hand edge of the U-spring is notched. This was to prevent interference with the rails. The wiper spring assemblies were installed by means of a shoehorn (Fig. VI-53) which pushed the assembly against the core, until the bar was in its final axial

position. The shoehorn was then removed, allowing the assembly to be pushed back against the duct by the compressed wiper springs. The drag force was such that even greased assemblies required a 100-pound force to pull them into place. In Fig. VI-7, note the deflection fixture for guiding the springs downward. In Fig. VI-53 the bent up section is just about to engage the shim skirts. This spring seals the annulus air from escaping to the base block side of the shroud, but allows motion between the core and the duct by riding on the shim skirts. The single wiper bar assembly was installed in the center convolution between rails, and the two double wiper bar assemblies were installed on each side. Two screws were used to secure each double wiper bar to the duct wall. The single wiper bar assemblies were easily installed, with one screw each to secure them to the duct wall.

D. Shipping and Test Vehicle Assembly

Upon completion of reactor assembly at Livermore, the handling vehicle was raised and its wheels installed. The self-powered vehicle was maneuvered out of the vault, after which the vehicle was towed to another building for loading. An inspection was made of all the flow holes for possible blockage.

Fifteen hundred auxiliary poison rods were installed in vacant holes to provide a system which would remain subcritical even if flooded and reflected by water (Fig. VI-54). Each rod had a stopper in a hole in the base block, and a short piece of rubber tubing on its end. The tubing was pressed on by the shipping cover which forced the stoppers firmly into the holes. This held the rods firmly in the flow channels during the shipment.

The duct was rigged for shipment by locking the support ring to the aft cradle both vertically and horizontally. The cradles of the vehicle were secured at the forward end to the cradle rails by tightened chains. Vertical jacks from the frame to the forward flange cover lifted the duct off of the roller cradle, and the forward cradle cap was raised to allow clearance between it and the duct. Cross-tie braces from the frame to the cover stabilized the forward end of the duct. A shipping cover was installed over the duct, and secured on the inside to the forward and aft cradle caps.

In Figs. VI-55 and VI-56 the loading scheme is shown clearly. A steel ramp was assembled to the trailer wheel housing. The trailer-mounted

winch pulled the vehicle up the ramp by a cable secured to the tow bar (center of picture) and a system of wooden wedges was used to bring the vehicle down to the trailer bed from the wheel housing. Two longitudinal braces and four lateral braces were used (just inside the axles). The vehicle jacks used to support the load rested on two cross channels. A support placed under the front frame member provided a pivot point for any forward motion.

The tractor-trailer system used was chosen because of the "soft" ride it presented. The trailer had 16 tires, and the tractor rear axles were torsion bar suspended. The tires had reduced air pressure of 60 psi. The handling vehicle frame was monitored for road shock loads by three accelerometers, and the duct was also monitored by three accelerometers. Their signals were recorded on magnetic tape. No significant vibrations over 1 g were recorded.

The trip to the Nevada Test Site was uneventful, except for a dust storm which necessitated completion of the journey in darkness. Upon arrival, the vehicle was off-loaded in the main hot cell of the Pluto Disassembly Building in the reverse manner from which it was loaded. Alignment of this vehicle with the rail platform was accomplished using an optical bench system. Brackets were set and secured in the floor, so that under remote conditions the vehicle could be reset to the rail platform if moved.

The rail platform system is a facility for aligning and engaging all of the remaining duct sections to form the complete reactor duct assembly on the test vehicle. A rail platform provided support for dollies on which Duct 4 (control duct), Duct 3 (inspection duct), and Ducts 1 and 2 (inlet ducts) each rest on roller cradles. (Right foreground of Fig. VI-57.) Drives on the dollies provided lateral and vertical adjustment, and a full length ball screw provided axial positioning. Load cells on each of the roller cradles on the dollies provided the necessary overload information for adjustment. The handling vehicle also had load cells for weighing loads imposed on the aft reactor duct. No alignment capability was incorporated in the handling vehicle for the duct mating process.

A fixture was mounted on the aft flange of Duct 4 to support the control rods. After the protection screen was placed on the support grid,

a gasket was taped to the forward flange of Duct 5. Duct 4 was eased toward Duct 5 until the control rods were mated with their respective tie rods. The duct continued to be moved aft and was adjusted to avoid binding of the control rods. When the two flanges were 6 inches apart, measurements of the flange gap were used to indicate misalignment. When the flanges were closed, the 56 flange bolts were installed and set. Ducts 1, 2, and 3 had been mated to each other earlier. The control actuators are supported at their upstream end in a grid at the forward end of Duct 3. When the engagement of Ducts 4 and 5 with Ducts 1, 2, and 3 began, the forward ducts were eased to the closed position with Duct 4.

Figure VI-57 shows the method of duct transfer to the test vehicle. A large strongback was designed so that the pickup point could be adjusted under remote conditions. This strongback was secured to the two support rings, using the empty support arm holes. The duct was lifted straight up from the handling rail platform and moved across the cell to a position over the Support Arm Remover. A special device was used for remotely removing and replacing all support arms. (Both sets of support arms would have to be removed from the duct assembly for remote maintenance and disassembly operations.)

The duct assembly was lifted again and moved down the cell to the correct position over the test vehicle. It was lowered into the support stands, the pillow-block caps were closed and secured, and the strongback was removed. Electrical and pneumatic lines were secured to the duct, and the test vehicle was ready for movement to the test pad. Additional features of the test vehicle are given in Refs. 2 and 117.

Fig. VI-1

Fig VI-2

Fig. VI-4.

Fig. VI-5.

Fig. VI-6.

Fig. VI-7.

Fig. VI-8.

Fig VI-9

Fig VI-10

Fig VI-11

Fig. VI-12

Fig. VI-13.

Fig. VI-14.

Fig. VI-15.

Fig. VI-16.

Fig. VI-17.

Fig VI-18

Fig. VI-19.

Fig. VI-20.

Fig. VI-21.

Fig. VI-22.

Fig. VI-23.

Fig. VI-24.

Fig. VI-25.

Fig. VI-26.

Fig. VI-27

Fig VI-28

Fig. VI-29.

Fig. VI-30.

Fig. VI-31.

Fig. VI-32.

Fig. VI-33.

Fig. VI-34.

Fig. VI-35.

Fig. VI-36.

Fig. VI-37.

Fig. VI-38.

Fig. VI-39.

Fig. VI-40.

Fig. VI-41.

Fig. VI-42.

Fig. VI-43.

Fig. VI-44.

Fig. VI-45.

Fig. VI-46.

Fig. VI-47.

Fig. VI-48.

Fig. VI-49.

Fig. VI-50.

Fig. VI-51

Fig VI-52

Fig. VI-53

Fig. VI-54.

Fig. VI-55.

Fig. VI-56.

Fig VI-57

VII. PARTS LIST

(Asterisks following drawing numbers indicate that there are also modification drawings. See Figs. VII-1 to -6 for part location.)

Item No.	Name and description	Drawing number	Number required
1	<u>Shim No. 1</u> Standard shim opposite flats	62-111532	84
2	<u>Shim No. 2</u> Adjacent standard shim	62-111533	168
3	<u>Shim No. 3</u> Adjacent standard shim	62-111535*	168
4	<u>Shim No. 4</u> Adjacent standard shim	62-111537*	168
5	<u>Shim No. 5</u> Adjacent standard shim	62-111538	169
6	<u>Shim No. 6</u> Adjacent standard shim	62-111540	84
7	<u>Shim No. 1 - Second & Fourth</u> Second shim in line, opposite flats	62-112796-1	6
8	<u>Shim No. 1 - Second & Fourth</u> Fourth shim in line, opposite flats	62-112796-2	6
9	<u>Shim No. 2 - Second & Fourth</u> Adjacent second shim	62-112819-1	12
10	<u>Shim No. 2 - Second & Fourth</u> Adjacent fourth shim	62-112819-2	12
11	<u>Shim No. 3 - Second & Fourth</u> Adjacent second shim	62-112807-1	12
12	<u>Shim No. 3 - Second & Fourth</u> Adjacent fourth shim	62-112807-2	12
13	<u>Shim No. 4 - Second & Fourth</u> Adjacent second shim	62-112814-1	12
14	<u>Shim No. 4 - Second & Fourth</u> Adjacent fourth shim	62-112814-2	12
15	<u>Shim No. 5 - Second & Fourth</u> Adjacent second shim	62-112799-1	12
16	<u>Shim No. 5 - Second & Fourth</u> Adjacent fourth shim	62-112799-2	12

Parts List (Continued)

Item No.	Name and description	Drawing number	Number required
17	<u>Shim No. 6 - Second & Fourth</u> Second shim, opposite points	62-112791-1	6
18	<u>Shim No. 6 - Second & Fourth</u> Fourth shim, opposite points	62-112791-2	6
19	<u>Shim No. 1 - Aft</u> Last shim in line, opposite flats	62-112793*	6
20	<u>Shim No. 2 - Aft</u> Adjacent last shim	62-112817-1*	6
21	<u>Shim No. 2 - Aft</u> Opposite hand of -1 adjacent last shim	62-112817-2*	6
22	<u>Shim No. 3 - Aft</u> Opposite hand of -2 adjacent last shim	62-112809-1*	6
23	<u>Shim No. 3 - Aft</u> Opposite hand of -1 adjacent last shim	62-112809-2*	6
24	<u>Shim No. 4 - Aft</u> Opposite hand of -2 adjacent last shim	62-112811-1*	6
25	<u>Shim No. 4 - Aft</u> Opposite hand of -1 adjacent last shim	62-112811-2*	6
26	<u>Shim No. 5 - Aft</u> Opposite hand of -1 adjacent last shim	62-112797-2*	6
27	<u>Shim No. 5 - Aft</u> Opposite hand of -1 adjacent last shim	62-112797-2*	6
28	<u>Shim No. 6 - Aft</u> Last shim, opposite points	62-112788*	6
29	<u>Standard key</u> First shim key (short)	62-116716-1	60
30	<u>Standard key</u> Last shim key (medium)	62-116716-2	60
31	<u>Standard key</u> Standard shim key (long)	62-11716-3	360
32	<u>Pivot key</u> Second & Fourth shim key	62-113256	59

Parts List (Continued)

Item No.	Name and description	Drawing number	Number required
33	<u>Taper key</u> Second & Fourth key	62-116718	1
34	<u>Rivet</u> Shim rivet at spring joint	63-102791	1128
35	<u>Spring segment</u> Standard spring	62-109268-1	180
36	<u>Spring segment</u> Spring at fourth shim elevation	62-109268-2	6
37	<u>Spring segment</u> First spring	62-109268-3	6
38	<u>Support rail</u> Bolted to duct 5, 8, 11, 13, 16	62-127095-4*	150
39	<u>Support rail No. 1</u> First rail bolted to duct 1	62-127093	30
40	<u>Wing wiper No. 5</u>	64-102027	2
41	<u>Support rail nut</u>	Esna-Rene' 41 No. LH3636-064 3/8 - 24 NF	510
42	<u>Wiper bar ass'y</u> Long "D"	64-104549	2
43	<u>Wiper bar ass'y</u> Long "A"	64-104543	28
44	<u>Wiper bar ass'y</u> Long "C"	64-104547	2
45	<u>Wiper bar ass'y</u> Long "B"	64-104545	28
48	<u>Grid plate</u> Tie rod sheet	62-109561 62-103338	1
49	<u>Ring</u> Grid ring under edge of plate	61-133958	1
50	<u>Beam No. 1</u> Longest beam	62-108279	2
51	<u>Beam No. 2</u> Adjacent beam	62-108281	2
52	<u>Beam No. 3</u> Adjacent beam	62-108285	2

Parts List (Continued)

Item No.	Name and description	Drawing number	Number required
53	<u>Beam No. 4</u> Adjacent beam	62-108286	2
54	<u>Beam No. 5</u> Adjacent beam	62-108288	2
55	<u>Beam No. 6</u> Adjacent beam	62-108289	2
56	<u>Outer spacer</u> Bolted to grid ring and beam No. 6	62-108275	2
57	<u>Inner spacer</u> Bolted between beams No. 5 and No. 6	62-108278	2
58	<u>Bolt 1/4-28 U. F. × 1-1/8 lg.</u> Bolts for grid plate and ring	-	90
59	<u>Bolt 5/16-24 U. F. × 1-5/16 lg.</u> Socket head spacer bolt	-	4
60	<u>Bolt 5/16-24 U. F. × 1-1/16 lg.</u> Socket head spacer bolt	-	2
61	<u>Nut 5/16-24 UNF-3B</u> Spacer nut	-	2
62	<u>Ring pin</u> Radial pin through duct into ring	62-127080-1	12
66	<u>Nut, tie rod std.</u>	L16B9984 62-100804	103
67	Machine procedure alloy Rene' 41 brazed ass'y including thd. adapter and aft flange	L16B9533 L16B9553 L16B6393-2 L16B4062-2	103
69	<u>Screw - No. 10-32 NFX 1/4 lg.</u> Socket head screw through nut into rod. (Control tie rod included.)	Mat'l A286	121
70	<u>Screw - No. 10-32 NFX 3/4 lg.</u> Socket head screw through nut into grid plate grid screen Tie rod included	Mat'l A286	125
72	<u>Cup insulator type "A"</u> Plat-ruthenium skirt to insulate aft flange from exit gas	L16B403 Machining L16B9523	103

Parts List (Continued)

Item No.	Name and description	Drawing number	Number required
73	<u>Washer, insulator type "A" prime</u> Forward insulator washer with chamfered outer edge	L13C6983-1	103
74	<u>Washer, insulator type "A" prime</u> Next two insulators	L13C6983-2	206
75	<u>Washer, insulator type "A"</u> Aft insulator washer with chamfered inner edge	L13C6973-1	103
76	<u>Washer, insulator type "A"</u> Adjacent (plain) washer	L13C6973-2	103
77	<u>Spacer-top standard tie rod</u> Sleeve touching grid plate	62-100862	103
78	<u>Spacer-standard tie rod</u> Next two sleeves	62-100858	206
79	<u>Spring adapter - forward (std.)</u> Forward spring collar std. tie rod	61-914917	103
80	<u>Spring adapter - aft (std.)</u> Aft spring collar std. tie rod	61-914916	103
81	<u>Compression spring "A"</u> Standard tie rod spring	63-107552	103
82	<u>Spring retainer "A" (std.)</u> Shipping spacer bridging spring	61-914979	103
83	<u>Reflector insert "A" key block</u> First insert in line	62-117000	103
84	<u>Reflector insert "A" tie rod</u> Short sleeve	62-116971	1236
85	<u>Reflector insert "A" core face</u> Vented key block	62-116116	103
86	<u>Insert "A"</u> Last reflector insert in line	62-111893	103
87	<u>Nut, tie rod</u> Large tie rod nut	62-100009 62-100004	18
88	<u>Machine procedure alloy</u> <u>Rene' 41</u> Brazed ass'y including forward collar and aft collar	62-111514	12

Parts List (Continued)

Item No.	Name and description	Drawing number	Number required
93	<u>Cup insulator, type "B"</u> Plat-ruthenium skirt to insulate aft flange from exit gas	L16B4233	18
94	<u>Washer insulator, type "B" prime</u> Forward insulator washer with chamfered outer edge	L13C7003-1	18
95	<u>Washer insulator type "B" prime</u>	L13C7003-2	72
96	<u>Lock washer insulator</u> Forward washer keep insulator cup and base block caps in line	62-111043	18
97	<u>Washer insulator, type "B"</u> Aft insulator washer with chamfered inner edge	L13C6993-1	36
98	<u>Washer insulator, type "B"</u> Adjacent (plain) washer	L13C6993-2	36
99	<u>Spacer - control tie rod</u> Sleeve touching grid R.	62-100054-2	18
100	<u>Spacer - control tie rod</u> Next two sleeves	62-100054-1	36
101	<u>Spring adapter forward (control)</u> Forward spring collar control tie rod	61-914949	18
102	<u>Spring adapter - aft (control)</u> Aft spring collar control tie rod	61-914948	18
103	<u>Compression spring "B"</u> Control tie rod spring	63-107554	18
104	<u>Spring retainer "B" (control)</u> Shipping spacer bridging spring	61-215075	18
105	<u>Reflector insert "B" key block</u> First insert in line	62-116970	18
106	<u>Reflector insert "B" tie rod</u> short sleeve	L13C4112	216
107	<u>Reflector insert "B" core face</u> Vented key block	L13C4123	18

Parts List (Continued)

Item No.	Name and description	Drawing number	Number required
108	<u>Insert " B"</u> Last reflector insert in line	62-111895	18
109	<u>Cell plate " A"</u> Standard cell	L16B1424*	43
110	<u>Cell plate " B"</u> Control cell	L13C4514*	18
111	<u>Cell plate " C"</u> Periphery	L13C6836*	6
112	<u>Cell plate " D"</u> Periphery	L13C6846*	6
113	<u>Cell plate " E"</u> Periphery	L13C6856 *	6
114	<u>Cell plate " F"</u> Periphery	L13C6816*	6
115	<u>Cell plate " G"</u> Periphery	L13C6826*	6
116	<u>Base block " A"</u> Standard block	L12E9444*	43
117	<u>Base block " B"</u> Control block	L16B2784*	18
118	<u>Base block " C"</u> Periphery block	L16B7966*	6
119	<u>Base block " D"</u> Periphery block	L16B7976*	6
120	<u>Base block " E"</u> Periphery block	L16B7986*	6
121	<u>Base block " F"</u> Periphery block	L16B7996*	6
122	<u>Base block " G"</u> Periphery block	L16B8006*	6
123	<u>Fueled ceramic tube</u> 0.227 i. d. $\times$ 1.962 lg.; R	62-102879-1	22,604
124	<u>Fueled ceramic tube</u> 0.227 i. d. $\times$ 2.463 lg.; S	62-102879-2	19,298
125	<u>Fueled ceramic tube</u> 0.227 i. d. $\times$ 2.933 lg.; T	62-102879-3	10,479
126	<u>Fueled ceramic tube</u> 0.227 i. d. $\times$ 3.925 lg.; U	62-102879-4	240,947

Parts List (Continued)

Item No.	Name and description	Drawing number	Number required
127	<u>Ceramic tube</u> 0.227 i. d. X 1.000 lg. unfueled B1	62-102881-1	11,088
128	<u>Ceramic tube</u> 0.227 i. d. X 1.500 lg, C1	62-102881-2	11,124
129	<u>Ceramic tube</u> 0.227 i. d. X 1.962 lg. unfueled E1	62-102881-3	21,081
130	<u>Ceramic tube</u> 0.227 i. d. X 3.780 lg. unfueled J1	62-102881-4	21,015
131	<u>Ceramic tube</u> 0.227 i. d. X 2.375 lg. unfueled F1	62-102881-5	3
132	<u>Ceramic tube</u> 0.227 i. d. X 2.463 lg. unfueled G1	62-102881-6	60
133	<u>Ceramic tube</u> 0.227 i. d. X 2.933 lg. unfueled H1	62-102881-7	33
134	<u>Ceramic tube</u> 0.227 i. d. X 3.425 lg. with chamfer ends - unfueled L1	62-102881-8	1,623
135	<u>Ceramic tube</u> 0.227 i. d. X 3.925 lg. with chamfer ends - unfueled K1	62-102881-9	20,115
136	<u>Ceramic tube (medium bore)</u> 0.130 i. d. X 1.462 lg. unfueled A2	62-102883-1	408
137	<u>Ceramic tube (medium bore)</u> 0.130 i. d. X 1.962 lg. unfueled E2	62-102883-2	792
138	<u>Ceramic tube (medium bore)</u> 0.130 i. d. X 2.463 lg. unfueled G2	62-102883-3	402
139	<u>Ceramic tube (medium bore)</u> 0.130 i. d. X 2.933 lg. unfueled H2	62-102883-4	608

Parts List (Continued)

Item No.	Name and description	Drawing number	Number required
140	<u>Ceramic tube (medium bore)</u> 0.130 i. d. X 3.925 lg. unfueled Ke guard tube	62-102883-5*	13,999
141	<u>Ceramic tube (small bore)</u> 0.093 i. d. X 1.875 lg. unfueled D3	62-105506-1	2,610
142	<u>Ceramic tube (small bore)</u> 0.093 i. d. X 1.962 lg. unfueled E3	62-105506-2	8,877
143	<u>Ceramic tube (small bore)</u> 0.093 i. d. X 2.375 lg. unfueled F3	62-105506-3	1,275
144	<u>Ceramic tube (small bore)</u> 0.093 i. d. X 3.780 lg. unfueled J3	62-105506-4	3,525
145	<u>Ceramic tube (small bore)</u> 0.093 i. d. X 3.925 lg. unfueled K3	62-105506-5	42,297
146	<u>Ceramic tube (small bore)</u> 0.093 i. d. X 2.462 lg. unfueled A3	62-105506-6	108
147	<u>Ceramic tube (small bore)</u> 0.093 i. d. X 2.463 lg. unfueled G3	62-105506-7	2,058
148	<u>Ceramic tube (small bore)</u> 0.093 i. d. X 3.425 lg. with chamfer ends - unfueled L3	62-105506-9	300
149	<u>Ceramic tube (small bore)</u> 0.093 i. d. X 3.925 lg. with chamfer ends - unfueled K4	62-105506-9	3,228
150	<u>Tube - nickel</u> 0.230 i. d. X 1.000 lg. with tapered end - nickel	62-113682	1,944
151	<u>Tube - nickel mod.</u>	63-114814	61
152	<u>Tube - notched</u> Small bore guard	62-129478	4,446
153	<u>Screen - grid</u>	64-101327	1
154	<u>Support ring</u>	64-101330	1
155	<u>Washer - flat - No. 10 cres</u>	-	4

Parts List (Continued)

Item No.	Name and description	Drawing number	Number required
156	<u>Poison rod ass'y</u>	62-110593	121
165	<u>Transition tube type "B"-2</u> <u>(1-3/8 in.)</u>	62-120173*	52
166	<u>Transition tube type "B"-2</u> <u>(7/8 in.)</u>	62-120174*	56
167	<u>Transition tube type "B"-1</u> <u>(1-3/8 in.)</u>	62-120178*	52
168	<u>Transition tube type "B"-1</u> <u>(7/8 in.)</u>	62-120176*	56
169	<u>Transition ring No. 1</u>	62-109494*	10
170	<u>Transition ring No. 2</u>	62-109496*	8
171	<u>Connector retainer bolt</u> Holds electrical connector in mounting plate	62-121010	18
172	<u>Control rod ass'y</u> Hafnium	62-115610	18
173	<u>Control system safety strap ass'y</u> Retains broken control rods Brazed ass'y	62-128477	14
174	<u>Safety strap nut</u> Screws on to safety tube	62-128454	14
175	<u>Control rod retaining ring</u> Fixes C. R. end in sliders and safety tube in C. R.	62-128436	14
176	<u>Actuator mounting structure</u> Main alignment and support of central and safety system	62-119646	1
177	<u>Retainer pins</u> Supports and aligns act. mtg. (6)	62-127083-1	10
178	<u>Retainer bolts</u> Bolts retainer pins to duct 2/pin	Bolts, 3/8-24 UNF x 1-1/16 lg. MS 2000 6H6	20
179	<u>Actuator mounting plate "A"</u> Aligns, supports and contains all connections for top and bottom actuators	62-115295	2
180	<u>Actuator mounting plate "B"</u> Aligns, supports and contains all connections for side actuators	62-116159	2

Parts List (Continued)

Item No.	Name and description	Drawing number	Number required
181	<u>Actuator mounting plate "C"</u> Aligns, supports and contains all connections for vernier actuator	62-117814	2
182	<u>Actuator mounting plate bolts</u> Through actuator plate into mtg. structure 2/plate	Bolt, 3/8-24 UNF $\times$ 1-9/16 lg. MS 2000 6H14	36
183	<u>Actuator mounting plate-dowels</u> Locate actuator plate on mtg. structure 2/plate	Dowel pins 1/4 diam nom. $\times$ (1/4 lg.) 410 or 416 cres MS 16555-49	12
184	<u>Center support tube</u> Supports shim way and safety system	62-115294	1
185	<u>Center support tube-bolts</u> Alignment pin for C. S. T. flange into mounting structure	Bolts, 3/8-24 UNF $\times$ 1-1/16 lg. MS 20006 H6	8
189	<u>Center support tube-dowel</u> Alignment pin for C. S. T. to shim ways	Dowel pins, 1/4 diam nom. $\times$ 1-1/4 lg. -410 or 416 cres MS 16555-49	2
190	<u>Vernier support tube</u> Support and guide vernier slider	62-121813	2
191	<u>Vernier support tube bolts</u> Through V. S. T. to mounting plate "C"	Bolt, 3/8-24 UNF $\times$ 2-3/16 lg. MS 20006 H24	8
192	<u>Vernier support tube dowel</u> Index V. S. T. to mounting plate "C"	Dowel pins, 1/4 diam nom. $\times$ 1-1/4 lg. -410 or 416 cres MS 16555-49	4

Parts List (Continued)

Item No.	Name and description	Drawing number	Number required
193	<u>Center support tube alignment pin</u> Aligns C. S. T. with support grid	62-128741	2
194	<u>Center support tube alignment pin retaining screw</u> Through alignment pin into C. S. T.	Screw, 1/4-28 UNF $\times$ 1/2 lg. MS 24678-20	2
195	<u>Vernier support tube alignment pin</u> Aligns vernier support tube with support grid 2/V. S. T.	62-128737	4
196	<u>Vernier support tube alignment pin retaining screw</u> Through alignment pin into vernier support tube	Screw, 1/4-28 UNF $\times$ 1/2 lg. MS 24678-20	4
197	<u>Vernier slider body</u> Piston in vernier support tube weldment	63-118662	2
198	<u>Vernier bearing</u> Piston ring	62-122951	4
199	<u>Bearing retainer</u> Back up ring for vernier bearing	62-122955	4
200	<u>Bearing retainer bolt</u> Through bearing retainer to vernier slider body	Screw, 1/4-28 UNF $\times$ 1/2 lg. MS 24678-20	24
201	<u>Shim slider "A"</u> Top and bottom shim slider	62-115603	2
202	<u>Shim slider "B"</u> Side shim slider	62-114915	2
203	<u>Shim slider bearing ass'y</u> Bearing assemblies for all shim sliders	62-121005, consisting of 62-121007 and 62-121004	64
204	<u>Slider washer (shim)</u> Between shim slider and bearing assemblies	-	64

Parts List (Continued)

Item No.	Name and description	Drawing number	Number required
205	<u>Bearing retainer nut</u> For shim slider bearing assemblies	Hex castle std. light duty 416 stainless steel 1/4-28 UNF-2B	64
206	<u>Coupler No. 2 ass'y</u> Connector between actuator and slider	64-103351	6
207	<u>Shim way</u> Shim slider bar	62-106359	4
208	<u>Way bolt</u> Through shim way into bosses on center support tube	Bolt - 5/16-24 UNF x 5/8 lg. MS 20005 H Grip lg. - 1/8	48
209	<u>Way washer (shims)</u> Between shim way and bosses	-	48
212	<u>Actuator support structure</u> Supports forward end of actuators in duct No. 3	62-122903	1
213	<u>Safety rod</u>	62-114370 (layout) 62-119024 (safety rod)	6
214	<u>Guide tube</u> Finned and slotted tube for guiding safety rod and cable	62-129127	6
215	<u>Guide tube clamp</u> Fastens guide tube to center support tube	62-128984 (machining) 62-128722 (casting)	42
216	<u>Bolt - guide tube clamp</u> Through bracket center support tube	62-128705	96
217	<u>Guide tube clamp forward</u> Thd. end bracket between guide tube and interior conduit	62-129109 (machining) 62-128987 (casting)	6
218	<u>Interior conduit</u> Between guide tube and bulk-head connector	Teleflex	6

Parts List (Continued)

Item No.	Name and description	Drawing number	Number required
219	<u>Exterior conduit</u> Between bulkhead connector to control box assembly	Teleflex 7-40275	6
220	<u>Cable</u> From control box ass'y to control rod	Teleflex 21731	6
221	<u>Spent tube conduit</u> Between spent travel tube and control box, includes fittings	21724 21725 21726 Teleflex	6
222	<u>Spent travel tube</u> Contains and prevents rotation of cable, includes fittings	Teleflex 21727	6
223	<u>Safety system ass'y</u>	63-118675	1
225	<u>Driveol box ass'y</u> Ganged drive boxes for flexible cable	63-108210	2
226	<u>Locking ass'y, manual</u> External lock	63-116710 63-116708	2
227	<u>Nut - guide tube brackets</u> Through bracket and center support tube	Per AN121577 material Inconel X per AMS 5667	96
228	<u>Guide pin</u> On vernier slider, prevents rotation	62-122947	2
229	<u>Cable ass'y</u> C ₁ and C ₂	64-104973	6
249	<u>Rake ass'y</u> Delta rake PT10 - 0	64-107307	1
250	<u>Delta rake</u> PT10 - 180	64-107275	1
254	<u>Reactor duct aft (weldment)</u> Duct No. 5	L16B3635	1
255	<u>Duct - forward reactor</u> Duct No. 4	61-215614	1
259	<u>Rake, Pt4</u> Duct No. 3	64-101073	2

Parts List (Continued)

Item No.	Name and description	Drawing number	Number required
260	<u>Rake gasket</u>	61-125010	2
261	<u>Rake gasket retainer</u> Pt2 and Pt4 Pt0	61-215009 62-106374	2
262	<u>Rake bolt</u>	3/8 - 24 NF × 1-1/4 soc. hd. capscrew	12
265	<u>Packing gland ass'y</u>		
266	<u>Rake Pt0 and Pt2</u>	64-101076	4
275	<u>Support rail No. 2</u>	62-127095-5	30
276	<u>Support rail No. 3</u>	62-127095-3	30
277	<u>Support rail Nos. 6-14</u>	62-127095-2	60
278	<u>Support rail No. 9</u>	62-127095-1	30
279	<u>Support rail Nos. 4-12</u>	62-127098-1	60
280	<u>Support rail Nos. 7-15</u>	62-127098-2	60
281	<u>Support rail No. 10</u>	62-127098-3	30
282	<u>Support rail No. 17</u>	63-107519	30
283	<u>Chrome cermet cartridge</u> <u>(7-1) std. transition</u> 0.875 long	62-116987-1 63-108731	578
284	<u>Chrome cermet cartridge</u> <u>(7-1) std. transition</u> 1.375 long	62-116987-2 63-108731	590
285	<u>Chrome cermet cartridge</u> <u>(7-1) "A" cut back transition</u> 0.875 long	62-116988-1 63-108727	4
286	<u>Chrome cermet cartridge</u> <u>(7-1) "A" cut back transition</u> 1.375 long	62-116988-2 63-108727	8
287	<u>Chrome cermet cartridge</u> <u>(7-1) "A" transition</u> 0.875 long	62-116990-1 63-108730	208

Parts List (Continued)

Item No.	Name and description	Drawing number	Number required
288	<u>Chrome cermet cartridge</u> <u>(7-1) "A" transition</u> 1.375 long	62-116990-2 63-108730	218
289	<u>Chrome cermet cartridge</u> <u>(7-1) std. cut back transition</u> 0.875 long	62-116991-1 63-108728	748
290	<u>Chrome cermet cartridge</u> <u>(7-1) std. cut back transition</u> 1.375 long	62-116991-2 63-108728	740
291	<u>Wiper bar ass'y</u> Long	63-107561 ACF 3-721353	60
292	<u>Wiper bar ass'y</u> Header rail	ACF 3-721346	60
293	<u>Wiper bar ass'y</u> Stand off bar forward	ACF 3-721348	60
294	<u>Wiper bar ass'y</u> Stand off bar	ACF 3-721347	60
295	<u>Wiper bar ass'y</u> Stand off bar aft	ACF 3-721349	60
296	<u>Wiper bar ass'y</u> Wiper bar bracket long	ACF 3-721350	60
297	<u>Wiper bar ass'y</u> Nut hex - (special)	ACF 3-721345	1320
298	<u>Wiper bar ass'y</u> Machine screw pan head special	ACF 3-721344	1440
299	<u>Wiper bar ass'y</u> Short	63-107557 ACF 3-721353	30
300	<u>Wiper bar ass'y</u> Header rail	721346	30
301	<u>Wiper bar ass'y</u> Stand off bar forward	721348	30
302	<u>Wiper bar ass'y</u> Stand off bar	721347	30
303	<u>Wiper bar ass'y</u> Stand off bar aft	721349	30

Parts List (Continued)

Item No.	Name and description	Drawing number	Number required
304	<u>Wiper bar ass'y</u> Wiper bar bracket short	721351	30
305	<u>Wiper bar ass'y</u> Nut, hex - (special)	721345	720
306	<u>Wiper bar ass'y</u> Machine screw - pan head (special)	721344	660
308	<u>Shroud</u> Shroud section forward	63-104379-2	60
309	<u>Shroud</u> Shroud section center	63-104378-1 63-104378-2	60 60
310	<u>Shroud</u> Shroud section aft	63-104381-1 63-104381-2	60 60
311	<u>Wiper</u> Shroud wiper	63-104382	60
312	<u>Screw</u> Shroud wiper screw	8-32 x 3/8 MS 24674-7	120
313	<u>Rivets</u> Shroud rivets - round head	3/16 x 5/8 MS 204535-6-10	180
314	<u>Bolt</u> Shroud bolt	63-104373	180
316	<u>Shim skirt</u> Sub ass'y	63-104387-1	60
317	<u>Shim skirt</u> Sub ass'y opposite hand	63-104387-2	60
318	<u>Shim skirt</u>	63-104377	120
319	<u>Block shim skirt</u>	63-104375	120
320	<u>Rivet</u>	AN420-6-8	240
321	<u>Screws</u> Shim skirt screws	8-32-3/8 soc. hd. MS 24674-8	240
323	<u>Rail wiper</u> For rail 12-14-15-17	63-108763-10	120

Parts List (Continued)

Item No.	Name and description	Drawing number	Number required
324	<u>Rail wiper</u> For rails 7-9-10	63-108763-11	90
325	<u>Rail wiper</u> For rails 3-4-6	63-108763-12	90
326	<u>Screws</u> Rail wiper screws	NAS 1101-C-06-H-4	600
327	<u>Wing wiper</u> No. 1 wiper (see items 42-45)	63-114257-1	60
328	<u>Wing wiper</u> No. 2 wiper See items 42-45	63-114257-2	60
329	<u>Wing wiper</u> No. 6 wiper See items 42-45	63-114257-3	60
330	<u>Wing wiper</u> No. 3 and No. 4 wiper right of rail See items 42-45	63-114442-1	60
331	<u>Wing wiper</u> No. 3 and No. 4 wiper left of rail See items 42-45	63-114442-2	58
332	<u>Wing wiper</u> Nos. 5, 7, 8 and 10 right of rail See items 42-45	63-114447-1	88
333	<u>Wing wiper</u> Nos. 5, 7, 8 and 10 left of rail See items 42-45	63-114447-2	88
334	<u>Wing wiper</u> No. 9 wiper right of rail See items 42-45	63-114449-1	30
335	<u>Wing wiper</u> No. 9 wiper left of rail See items 42-45	63-114449-2	30
336	<u>Wiper</u> Wiper support See items 42-45	63-114245	538

Parts List (Continued)

Item No.	Name and description	Drawing number	Number required
337	<u>Bar wiper</u> No. 6 wiper See items 42-45	63-114248-1	90
338	<u>Bar wiper</u> No. 1 and No. 2 wiper See items 42-45	63-114248-2	180
339	<u>Bar wiper</u> No. 7 wiper See items 42-45	63-114248-3	90
340	<u>Bar wiper</u> No. 4 wiper See items 42-45	63-114248-4	90
341	<u>Bar wiper</u> Nos. 3, 8, and 9 wiper See items 42-45	63-114248-5	270
342	<u>Bar wiper</u> No. 5 and No. 10 wiper See items 42-45	63-114248-6	180
343	<u>Aft bolt for long wiper bar</u>	63-112712	60
344	<u>Standard bolt for long wiper bar</u>	63-112515-1	81
345	<u>Long bolt for long wiper bar</u>	63-112515-2	4
346	<u>Connector bracket for wiper bar bolt</u>	63-110515-3	2
347	<u>Wing wiper No. 5</u>	63-114447-4	28
348	<u>Wing wiper No. 5</u>	63-114447-3	30
349	<u>Wing wiper No. 5 mod.</u>	64-102033-1	2
350	<u>Wing wiper No. 5 mod.</u>	64-102033-2	2
351	<u>Wing wiper No. 5 mod.</u>	64-102025	2
352	<u>Support - wiper</u> (mod. "B")	64-104551	118
353	<u>Support - wiper</u> (knotch)	64-102032	2
354	<u>Support - wiper</u>	64-102046	2
355	<u>Wiper bar bolt</u>	63-1125154	3

Parts List (Continued)

Item No.	Name and description	Drawing number	Number required
356	<u>Bolt</u> 6 unit disconnect	64-104930	8
357	<u>Nut</u> Sleeve fitting	63-102025-1	10
358	<u>Nut</u> Castle shear	No. 10-32 NF-3 AN 320-3 shroud	180

Fig. VII-1. Longitudinal section through Duct 5.

Fig. VII-2. Section A-A, Duct. 5.

Fig. VII-3. End view, forward end of Duct 5.

Fig. VII-4. Longitudinal section through Ducts 3 and 4.

Fig. VII-5. Section A-A, Duct 4.

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Fig. VII-6. End view, aft end of Duct 4.

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