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Reference Piping Design Stress Analysis

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ABSTRACT

A history of the piping stress analysis is presented, tracing the work done from the conceptual stage to the final reference design layout. The piping stress analysis of the final reference design piping layout is also presented.

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Table of Contents

I. INTRODUCTION

The purpose of this report is to provide a detailed outline of the piping thermal and primary stress analysis of the 5 Kwe Reactor T/E Unmanned System Test.

Initially the thermal analysis was performed on the primary piping system of the reference conceptual design layout. It investigated the reference design and also explored alternate design configurations. The analysis was limited to the primary piping system from the reactor to the sixteen (16) thermoelectric converters. The stress analysis was performed under start-up conditions (see TSR-652-340-002) because this transient cycle caused the most severe differential thermal movements between the body and the piping.

After the reference design concept became finalized, a more detailed thermal stress analysis and primary stress analysis was performed on all the piping systems, both primary and secondary. The primary load analysis is based on the system of loadings that would occur in space. No gravitational, seismic, support or component weight forces were considered, therefore, only the pressure thrust of the expansion joint was taken into account.

II. STRUCTURAL CRITERIA

For the piping thermal analysis, the structural criteria of the Power Piping Code ANSI B31. 1. 0 and Section III Class 1 high temperature code case 1331. 5 of ASME BPVC Section III were considered for application to the piping systems.

The pertinent differences between the Power Piping Code (ANSI B31. 1. 0) and the ASME, Boiler and Pressure Vessel Code, High Temperature Code Case 1331-5, are tabulated in Figures 1, 2 and 3. Basically, the high temperature code case calculates stress indices for elbows (C_2) that are double that of the Power Piping Code (i). This means that calculated stresses under the high temperature code case will be much greater than those calculated under the power piping code for the same piping configuration. Also, due to a more stringent approach, the allowable stress intensity levels under the code case are considerably lower than the allowable stress ranges calculated under the power piping code. For Type 316 austenitic stainless steel piping at 1200°F, the allowable stress range S_A (B31. 1. 0) = 25, 200 psi and the allowable stress intensity Sq (1331-5) = 11, 250 psi. In this report, stress analysis is performed per the MEL-21 computer program. All stress levels that are shown at elbows and nozzles are directly related to the allowable stress range S_A per the Power Piping Code B31. 1. 0.

The primary load analysis made use of the ANSI B31. 1. 0 section on Additive Stresses. The sum of the longitudinal stresses due to pressure, weight and other sustained loads are not to exceed the allowable stress in the hot condition, S_H. The values of the longitudinal pressure stress, S_{LP} and S_H are tabulated in Figure 4. Since conditions in space are considered only, the stress due to all other sustained loads is S_B, the expansion joint pressure thrust. The value of S_B must be tabulated to establish the maximum allowable stress caused by the pressure loading from the expansion joint. The results of the tabulations for S_B are shown in Figure 4.

THERMAL -

POWER PIPING CODE ANSI B31.1.0

EXPANSION STRESS $S_E = \sqrt{S_b^2 + 4 S_t^2}$

WHERE $S_b = \dfrac{\sqrt{i M_b^2 + i M_t^2}}{Z}$

AND $i_{ELBOWS} = \dfrac{0.9}{h^{2/3}}$
(STRESS INDICES)

ALLOWABLE STRESS RANGE, S_A

$$S_A \geq f\,(1.25\,S_c + 0.25\,S_H)$$

$$\boxed{\begin{array}{l} S_A\,(316\ \text{ST. STL. @ } 1200^\circ F) = 25{,}200\ \text{PSI} \\ S_A\,(\text{HASTELLOY-N @ } 1200^\circ F) = 32{,}500\ \text{PSI} \end{array}}$$

CYCLING ACCOUNTED FOR

WHERE $f = 1$ FOR 7000 CYCLES OR LESS

$f = .5$ FOR 10^5 CYCLES OR MORE

LEGEND

S_b = RESULTANT BENDING STRESS, PSI

M_b = BENDING MOMENT, IN. LBS.

M_t = TORSIONAL MOMENT, IN. LBS.

Z = SECTIONAL MODULUS OF PIPE, $IN.^3$

h = FLEXIBILITY CHARACTERISTIC

i = STRESS INTENSITY FACTOR

f = STRESS RANGE REDUCTION FACTOR

S_c = COLD STRESS, PSI

S_H = HOT STRESS, PSI

FIGURE 1

THERMAL (CONT'D)

NUCLEAR CODE - SECT. III ASME BOILER & PRESS. VESSEL PLUS HIGH TEMP. CODE CASE 1331-5

PRIM. & SEC. STRESSES

$$S_N = C_1 \frac{P_o D_o}{2t} + C_2 \frac{D_o M_i}{2I} + \ldots\ldots \leq 3 S_M \quad (T < 800°F) \quad \left(\begin{array}{c}\text{SECT. III} \\ \text{ASME} \\ \text{BPVC}\end{array}\right)$$

WHERE $C_{2\,ELBOWS} = \frac{1.95}{b^{2/3}}$ $(b = h)$
(STRESS INDICES)

ALLOWABLE STRESS INTENSITY, S_g

$$S_g \geq \underbrace{P_L + \frac{P_B}{K}}_{(PRIM.)} + \underbrace{Q}_{(SEC.)} \quad (1331\text{-}5)\ (T > 800°F)$$

WHERE $Q = Q_{DISCON.} + Q_E$

& Q_E IS SIMILAR TO S_A (FIG. 1)

$$\boxed{S_g\ (316\ \text{ST. STL.}\ @1200°F) = 11{,}250\ \text{PSI}}$$

CREEP-FATIGUE EVALUATION IS REQ'D.

PER $\sum_{j=1}^{P} \left(\frac{n}{n_d}\right) + \sum_{k=1}^{g} \left(\beta \frac{t}{T_d}\right)_k \leq D$

FOR LEGEND, SEE FIG. 2A

FIGURE 2

LEGEND FOR FIG. 2

S_N = CALCULATED PRIM. & SEC. STRESSES, PSI
C_1, C_2 = SECONDARY STRESS INDICES
P_o = DESIGN PRESS.
D_o = OUTSIDE PIPE DIAMETER
t = NOMINAL PIPE WALL THK.
I = MOMENT OF INERTIA
M_i = RESULTANT MOMENT LOADING
S_M = ALLOWABLE STRESS AT TEMP.
b = FLEXIBILITY CHARACTERISTIC
P_L = LOCAL PRIMARY MEMBRANE STRESS
P_B = PRIMARY BENDING STRESS INTENSITY
Q = SECONDARY STRESS INTENSITY

FIGURE 3

PRIMARY LOADS

$$S_H \geqq S_{LP} + S_B$$

WHERE

S_H = BASIC MATL. ALLOWABLE STRESS AT THE HOT TEMP.

S_{LP} = LONGITUDINAL PRESSURE STRESS

S_B = STRESS DUE TO ALL OTHER SUSTAINED LOADS

AND

S_H = 6950 PSI TP316H A312 @ 1200°F

AND

$$S_{LP} = \frac{Pd^2}{D^2 - d^2} = \frac{(30)(1.46)^2}{(1.5)^2 - (1.46)^2} = 535 \text{ PSI}$$

WHERE

P = INTERNAL DESIGN PRESSURE
D = PIPE O.D.
d = PIPE I.D.

THEREFORE

$\underline{S_B}$ = 6950 − 535 = <u>6415 PSI</u>
MAXIMUM ALLOWABLE

FIGURE 4

III. THERMAL STRESS ANALYSIS

Part A

This part of the thermal analysis reflects the work done in selection of the reference design.

Part B

This part of the thermal analysis reflects the analyses done after finalization of the reference design.

Part A

1. Primary Piping System

An in-depth thermal stress analysis of the primary piping outflow system from the reactor to the T/E converters was made to ascertain the validity of the use of expansion joints and Type 316 stainless steel piping in the reference design. Various configurations (Figures 5 through 9) without the use of expansion joints were first considered. In each case, the maximum thermal stress level exceeded the allowable stress range per the power piping code B31. 1. 0. The reference design was then analyzed (Figures 10 through 13) utilizing expansion joint spring rates in increments of 100 lb/in from 100 lb/in to 400 lb/in, and tabulated in table form (Figure 14) and graph form (Figures 15 through 17). Results indicate that the reference design is satisfactory for spring rates up to a maximum of 400 lb/in providing the nozzle reactions at the reactor and the T/E converters do not exceed the equipment design limits. The thermal movement of the body at the reactor outlet (.25") reflects the use of a lockalloy radiator.

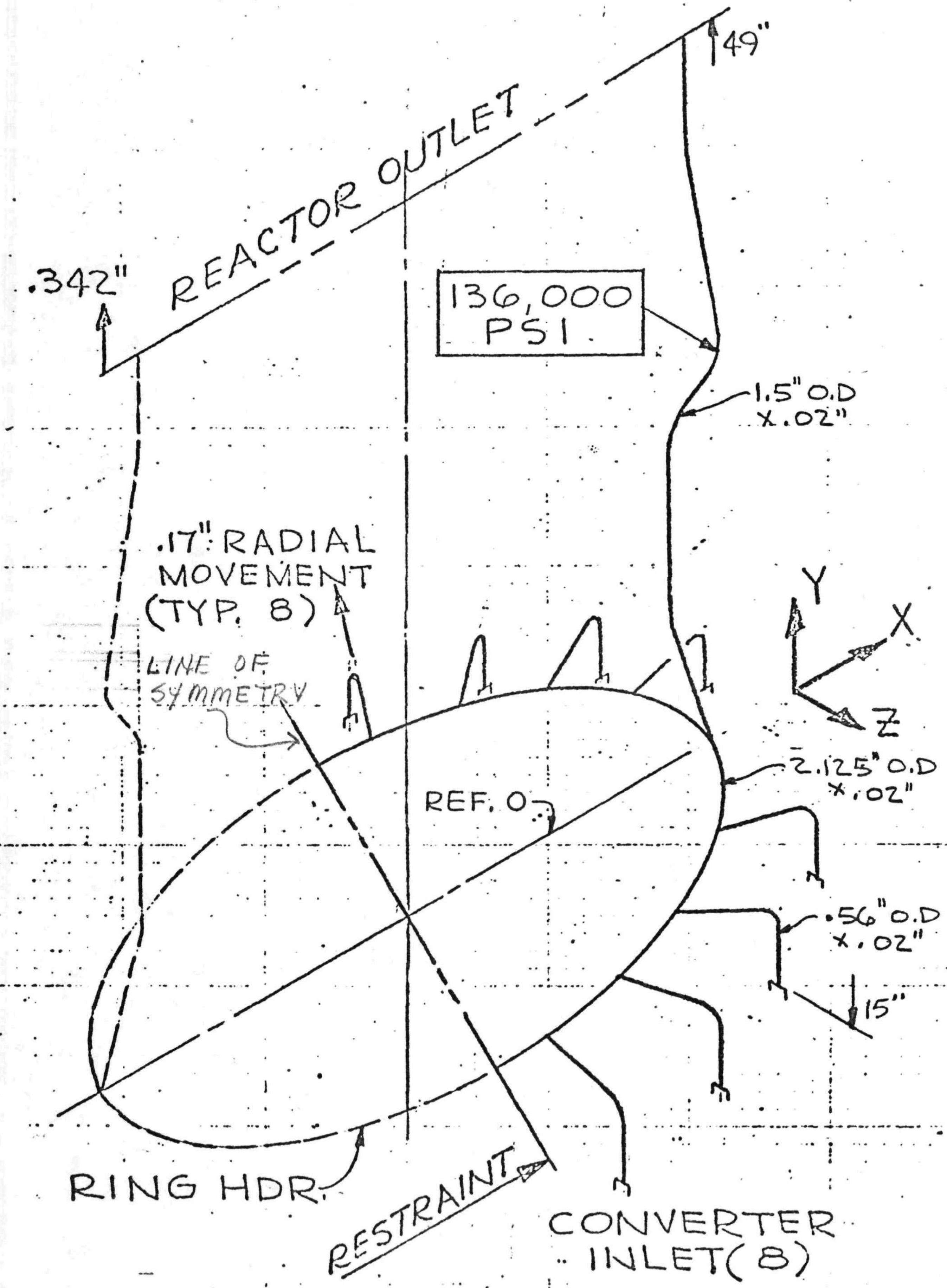

49"
REACTOR OUTLET
.342"
136,000
PSI
1.5" O.D
X.02"
.17" RADIAL
MOVEMENT
(TYP. 8)
LINE OF
SYMMETRY
Y
X
Z
2.125" O.D
X.02"
REF. O
.56" O.D
X.02"
15"
RING HDR.
RESTRAINT
CONVERTER
INLET (8)

FIGURE 5

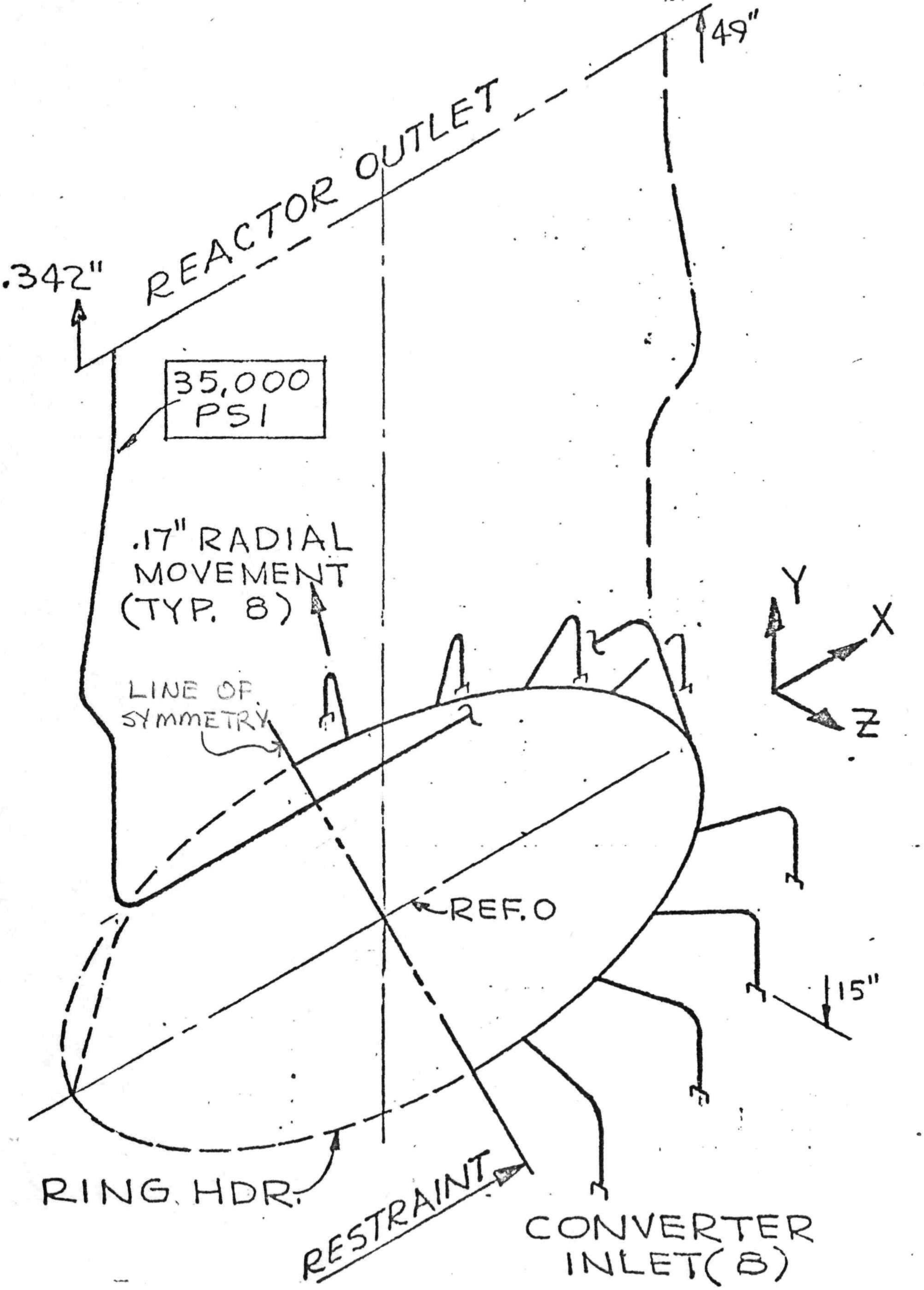

.49"
REACTOR OUTLET
.342"
35,000 PSI
.17" RADIAL MOVEMENT (TYP. 8)
Y
X
Z
LINE OF SYMMETRY
REF. 0
15"
RING. HDR.
RESTRAINT
CONVERTER INLET (8)

FIGURE 6

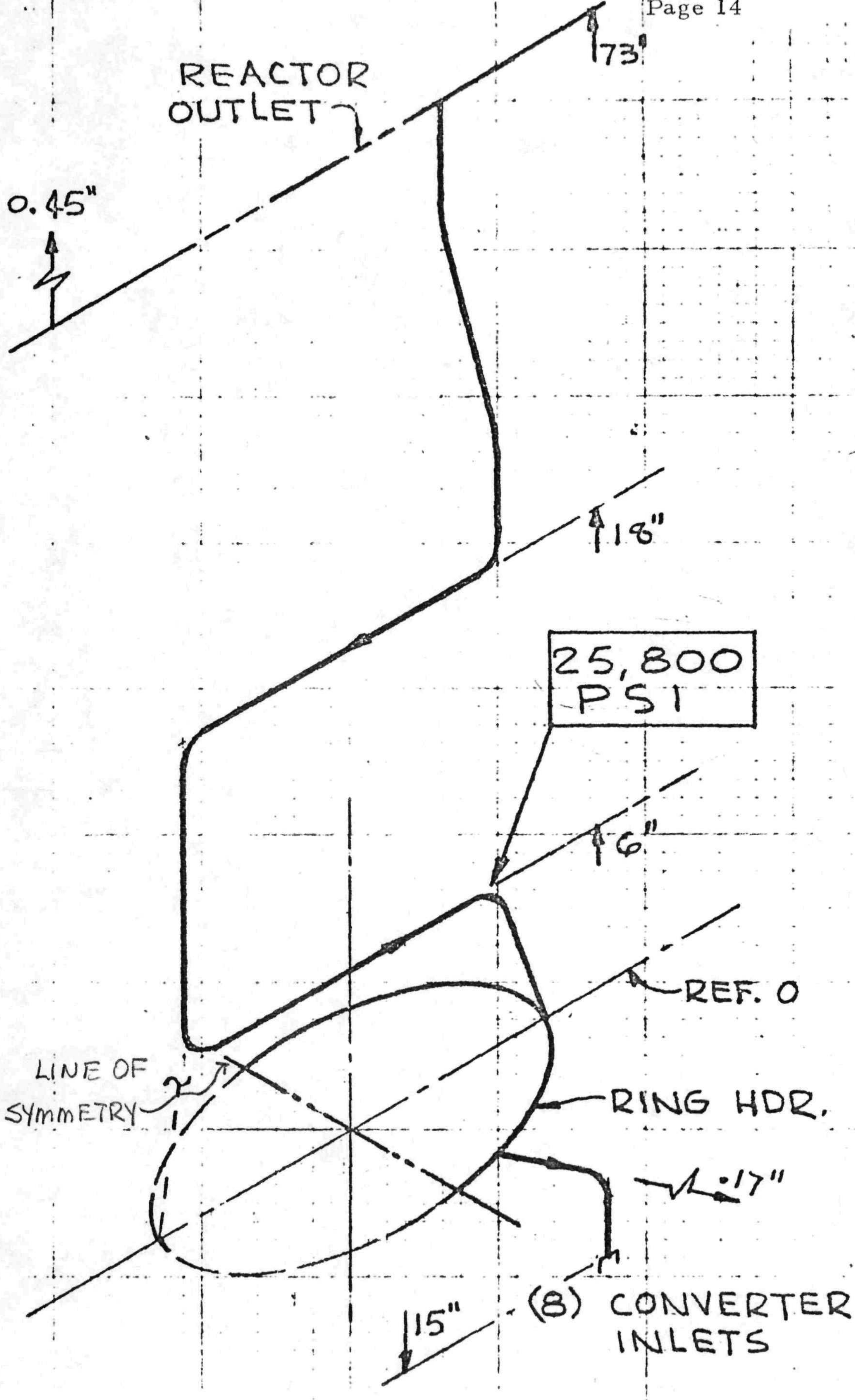

73"
REACTOR OUTLET
0.45"
18"
25,800 PSI
6"
REF. 0
LINE OF SYMMETRY
RING HDR.
.17"
15"
(8) CONVERTER INLETS

FIGURE 7

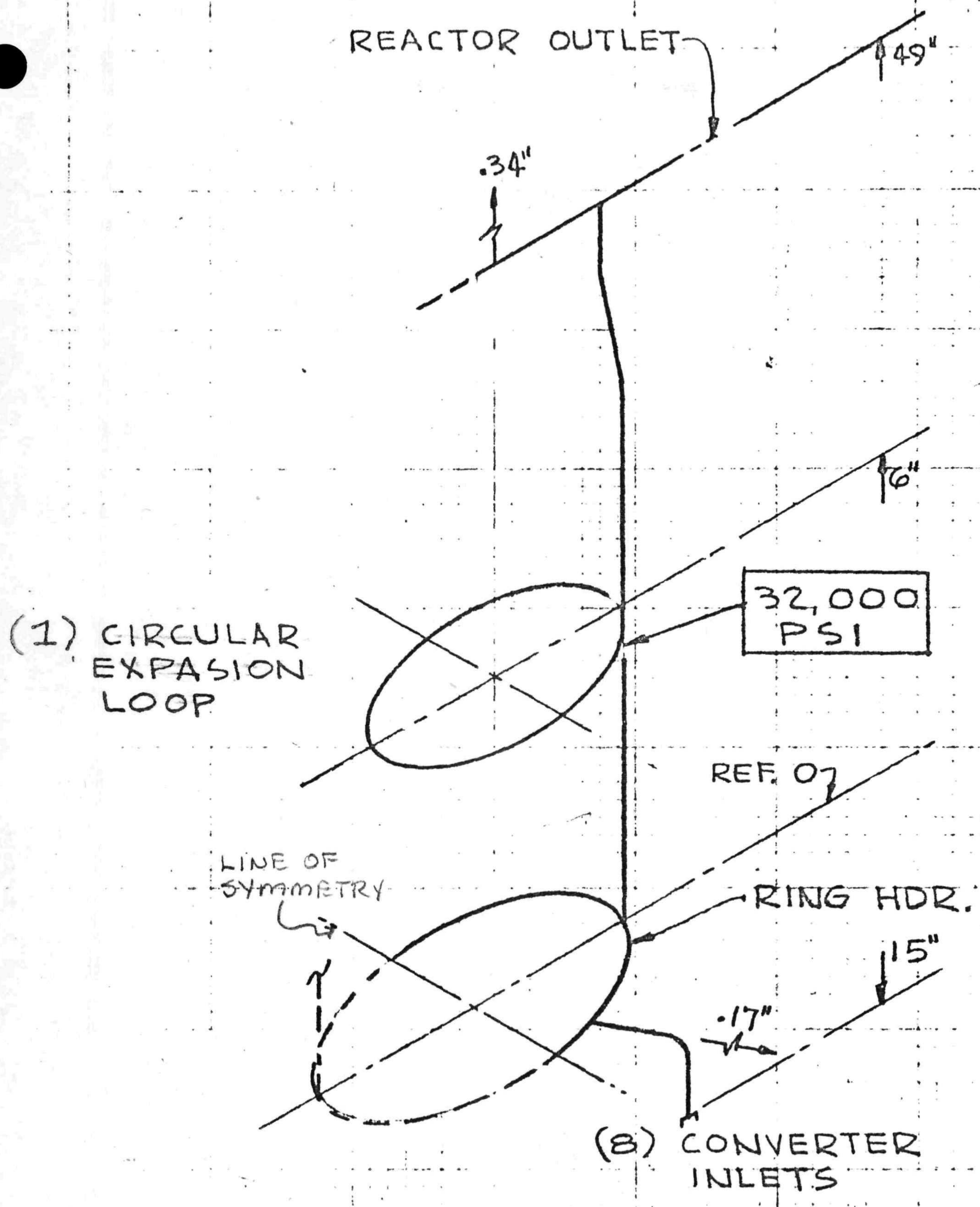

REACTOR OUTLET
49"
.34"
6"
(1) CIRCULAR EXPASION LOOP
32,000 PSI
REF. 0
LINE OF SYMMETRY
RING HDR.
15"
.17"
(8) CONVERTER INLETS

FIGURE 8

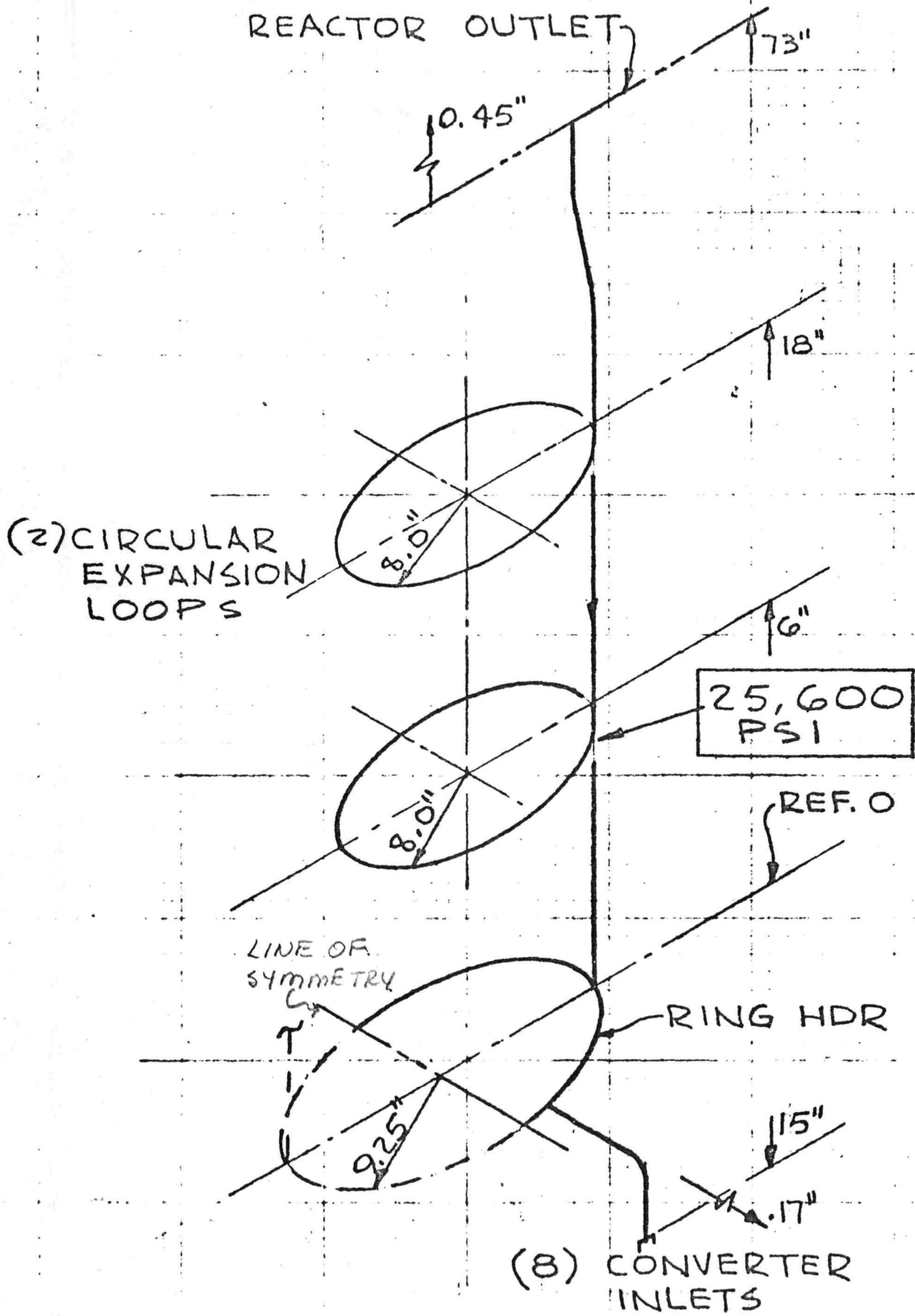

REACTOR OUTLET
73"
0.45"
18"
(2) CIRCULAR EXPANSION LOOPS
8.0"
8.0"
6"
25,600 PSI
REF. 0
LINE OF SYMMETRY
RING HDR
9.25"
15"
.17"
(8) CONVERTER INLETS

FIGURE 9

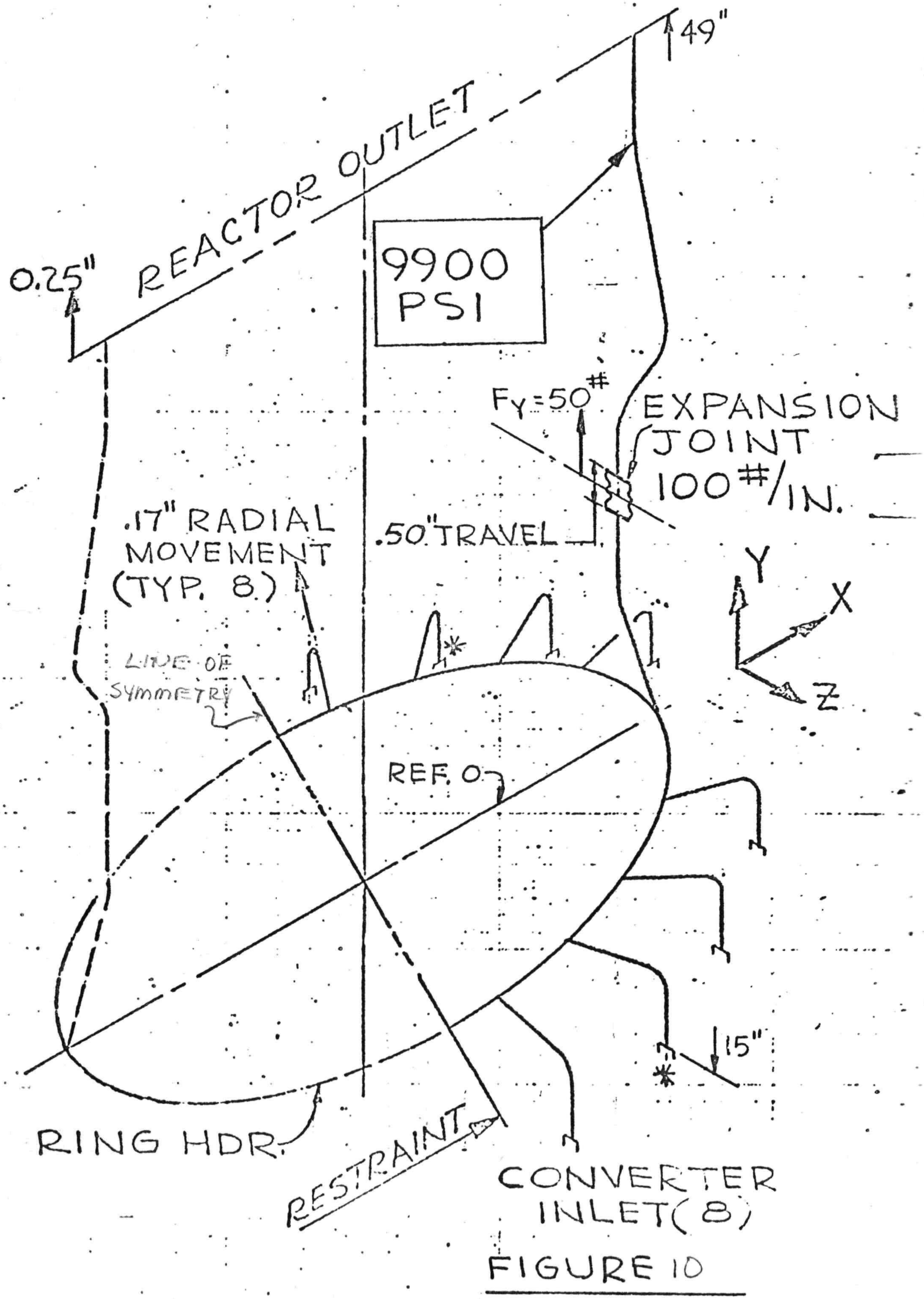

49"
REACTOR OUTLET
0.25"
9900
PSI
Fy=50#
EXPANSION
JOINT
100#/IN.
.17" RADIAL
MOVEMENT
(TYP. 8)
.50" TRAVEL
Y
X
Z
LINE OF
SYMMETRY
REF. O
15"
RING HDR.
RESTRAINT
CONVERTER
INLET (8)

FIGURE 10

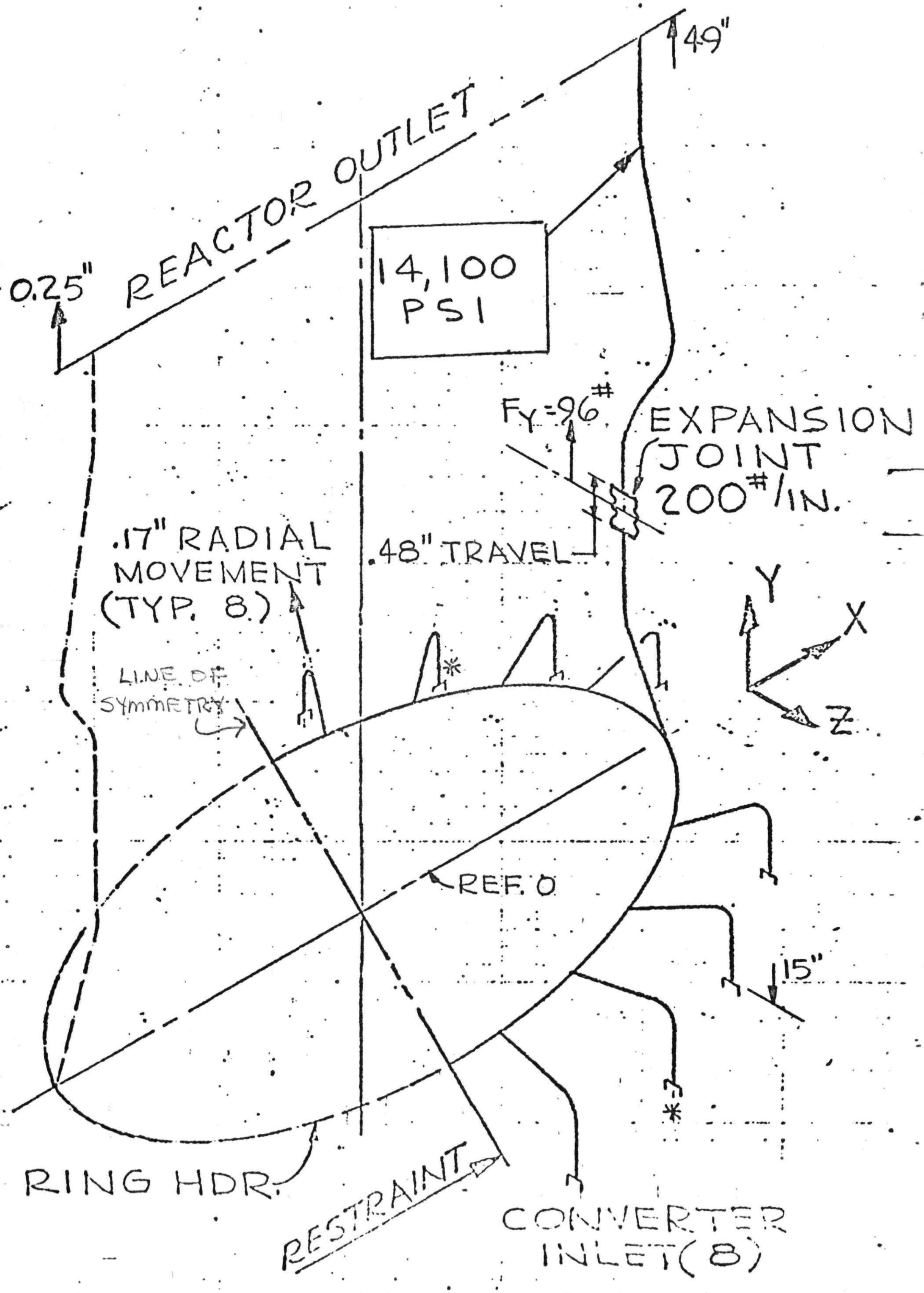

49"
REACTOR OUTLET
0.25"
14,100
PSI
FY=96#
EXPANSION
JOINT
200#/IN.
.17" RADIAL
MOVEMENT
(TYP. 8)
.48" TRAVEL
Y
X
Z
LINE OF
SYMMETRY
REF. 0
15"
RING HDR.
RESTRAINT
CONVERTER
INLET (8)

FIGURE 11

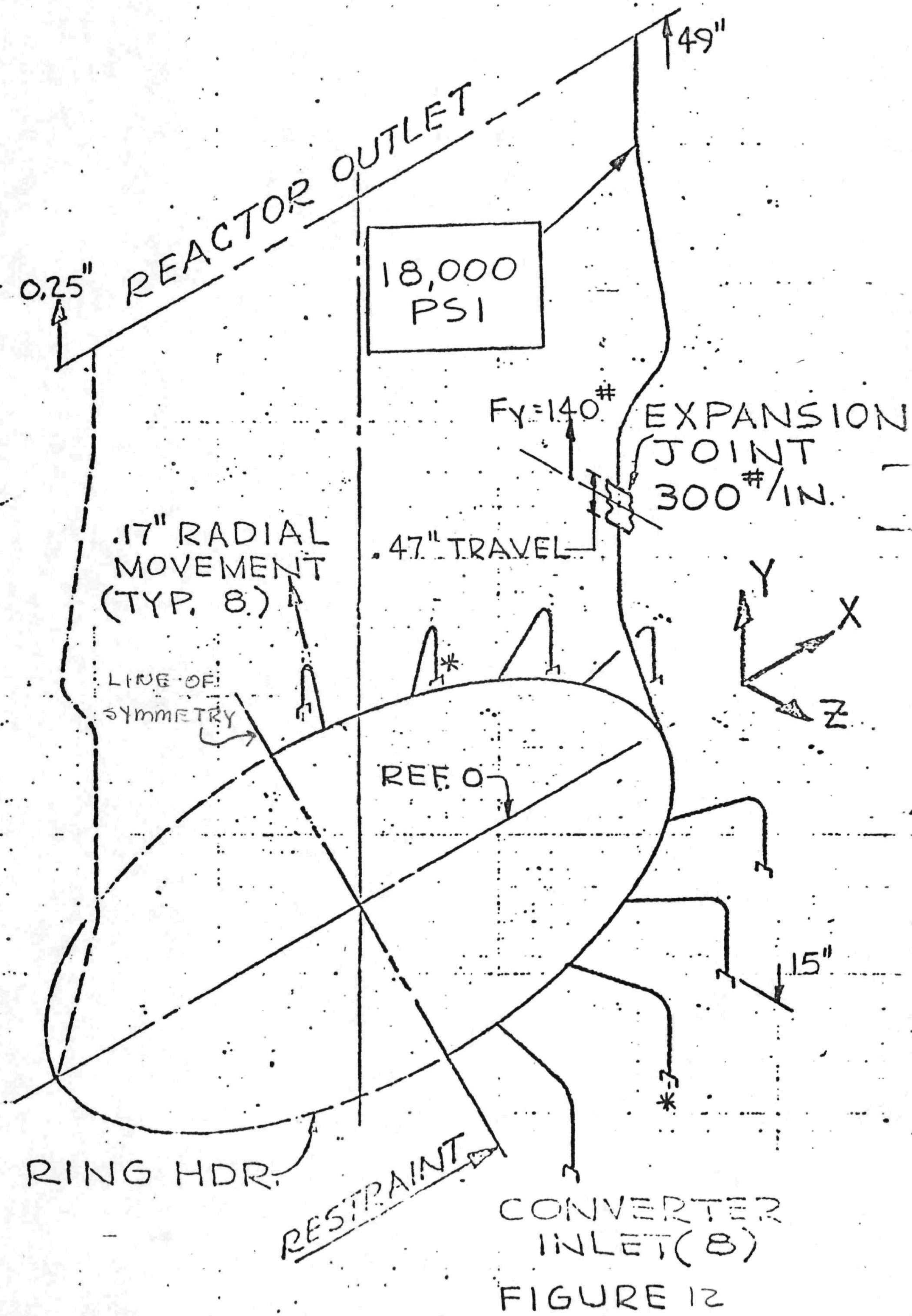

49"
REACTOR OUTLET
0.25"
18,000 PSI
Fy=140#
EXPANSION JOINT 300#/IN.
.17" RADIAL MOVEMENT (TYP. 8.)
.47" TRAVEL
Y
X
Z
LINE OF SYMMETRY
REF. 0
.15"
RING HDR.
RESTRAINT
CONVERTER INLET (8)

FIGURE 12

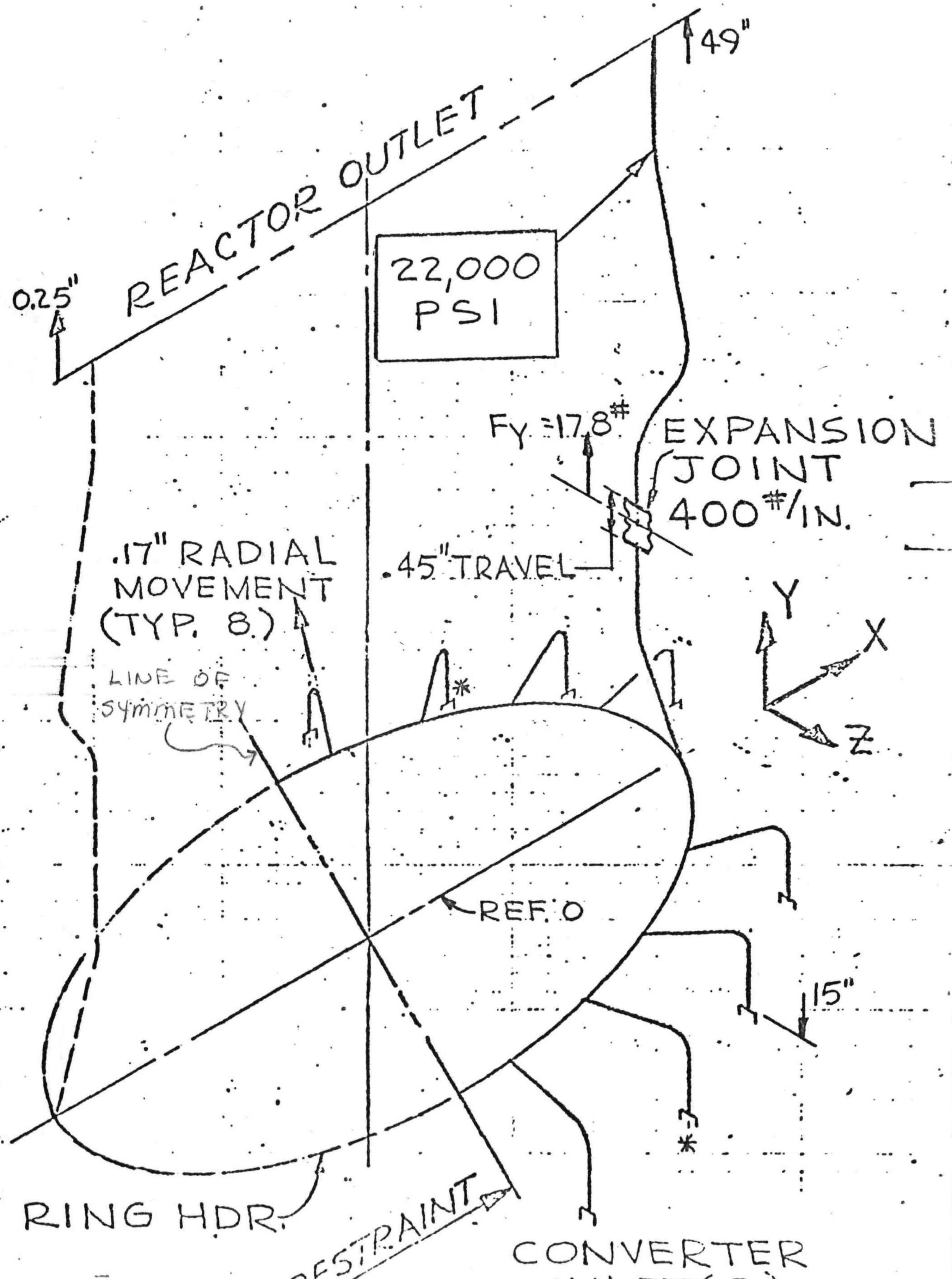

49"
REACTOR OUTLET
0.25"
22,000 PSI
F_Y = 17.8#
EXPANSION JOINT 400#/IN.
.17" RADIAL MOVEMENT (TYP. 8.)
.45" TRAVEL
Y
X
Z
LINE OF SYMMETRY
REF. 0
15"
*
RING HDR.
RESTRAINT
CONVERTER INLET (8)

FIGURE 13

MAXIMUM NOZZLE REACTIONS AT COVERTERS (MARKED *) AND AT REACTOR OUTLET

	STRESS, PSI		FORCE (Fy), LBS		MOMENT, IN-LBS	
SPRING RATE, #/IN	REAC.	CONV.	REAC.	CONV.	REAC. Mz	CONV. Mx
100	4000	7000	50	-11	128	25
200	5650	8250	96	-17	172	31
300	7100	9400	140	-21	230	34
400	8450	10,400	178	-24	274	37

FIGURE 14

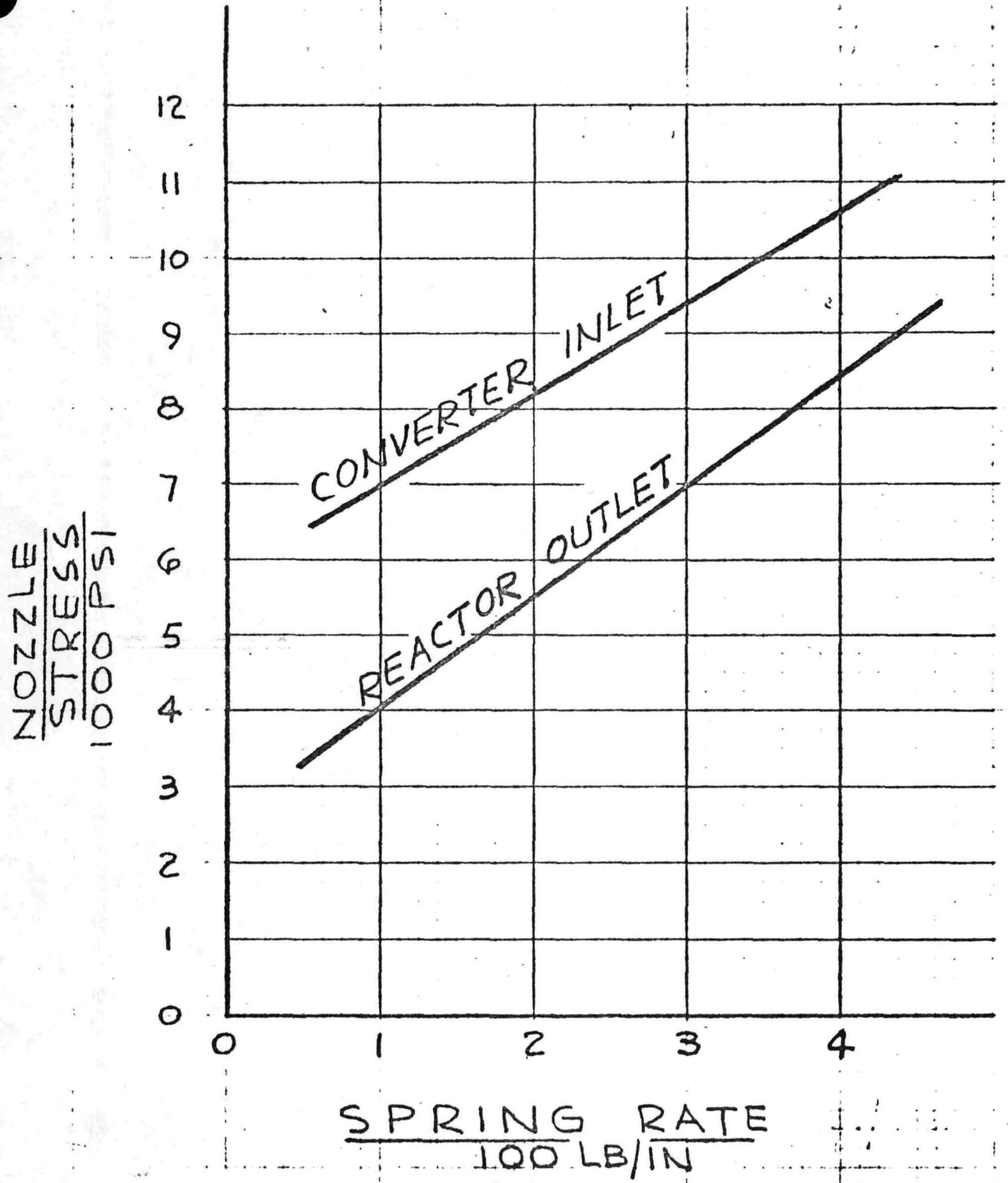

CONVERTER INLET
REACTOR OUTLET
NOZZLE STRESS / 1000 PSI
0
1
2
3
4
5
6
7
8
9
10
11
12
0
1
2
3
4
SPRING RATE / 100 LB/IN

FIGURE 15

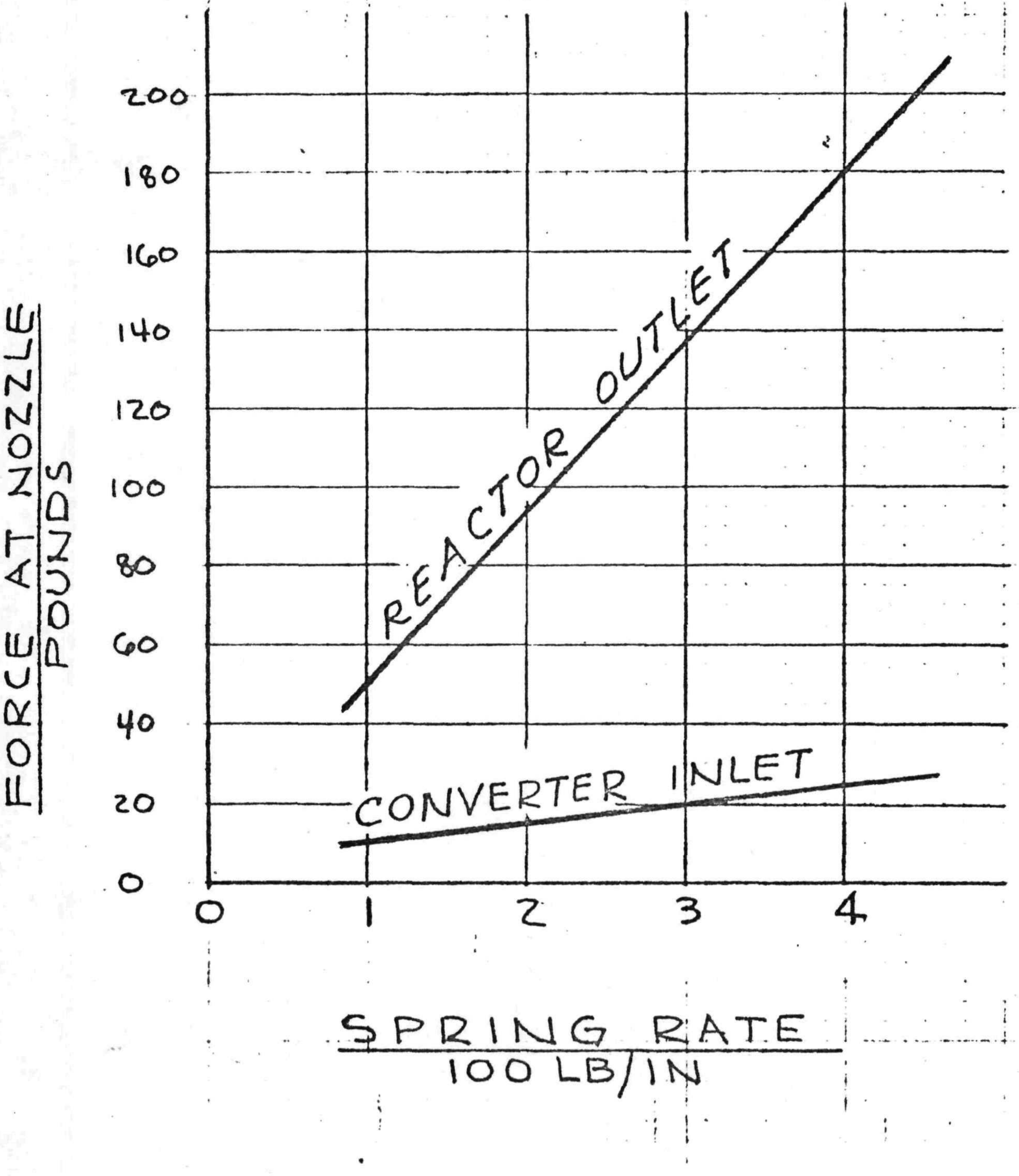

FORCE AT NOZZLE
POUNDS
200
180
160
140
120
100
80
60
40
20
0
REACTOR OUTLET
CONVERTER INLET
0
1
2
3
4
SPRING RATE
100 LB/IN

FIGURE 16

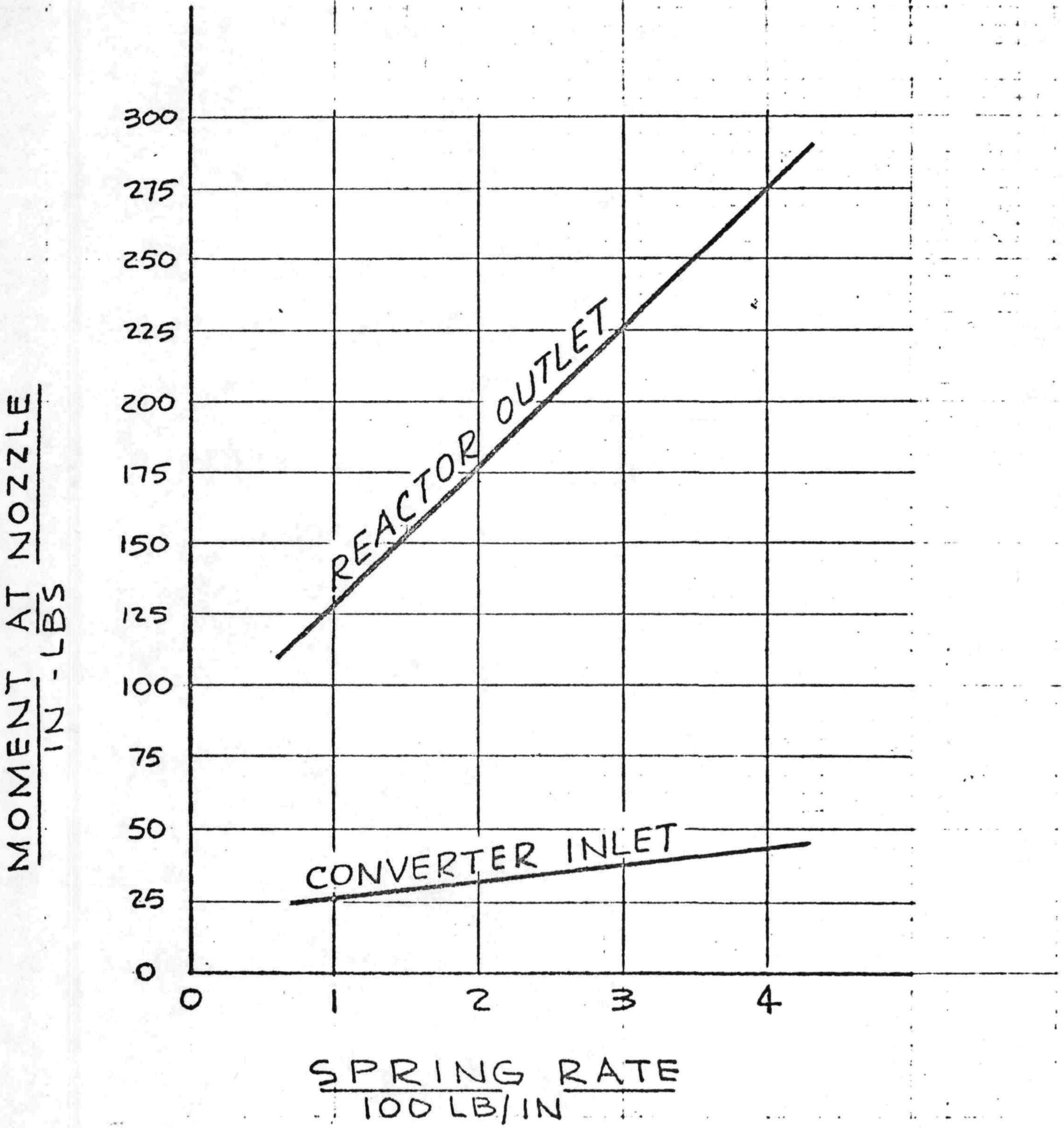

MOMENT AT NOZZLE
IN - LBS
300
275
250
225
200
175
150
125
100
75
50
25
0
REACTOR OUTLET
CONVERTER INLET
0
1
2
3
4
SPRING RATE
100 LB/IN

FIGURE 17

2. DISSIMILAR METALS

A backup study, utilizing Hastelloy-N and pipe loops in place of 316 stainless steel and expansion joints, is shown on Figures 18 through 21. Figure 22 shows the results of using a Hastelloy N half-loop on the down-comer and a 316 stainless steel ring header and brach runouts to the converters. Figures 19 and 20 show the results of utilizing all Hastelloy-N material in the primary outflow piping from the reactor to the converters. Conditions A and B do not apply to the reference design because they assume a radiator of stainless steel - aluminum alloy material. Conditions C and D, as summarized in Figure 20, do apply to the reference design, and show that, for an allowable stress range S_A of 32,500 psi at 1200°F (Figure 1), it is necessary to change the access panel material from aluminum alloy to stainless steel, and to maintain the access panel at a temperature of 1200°F. Also, Figure 21 shows that the half-loop design is satisfactory if an aluminum alloy access panel is maintained at a temperature of 600°F.

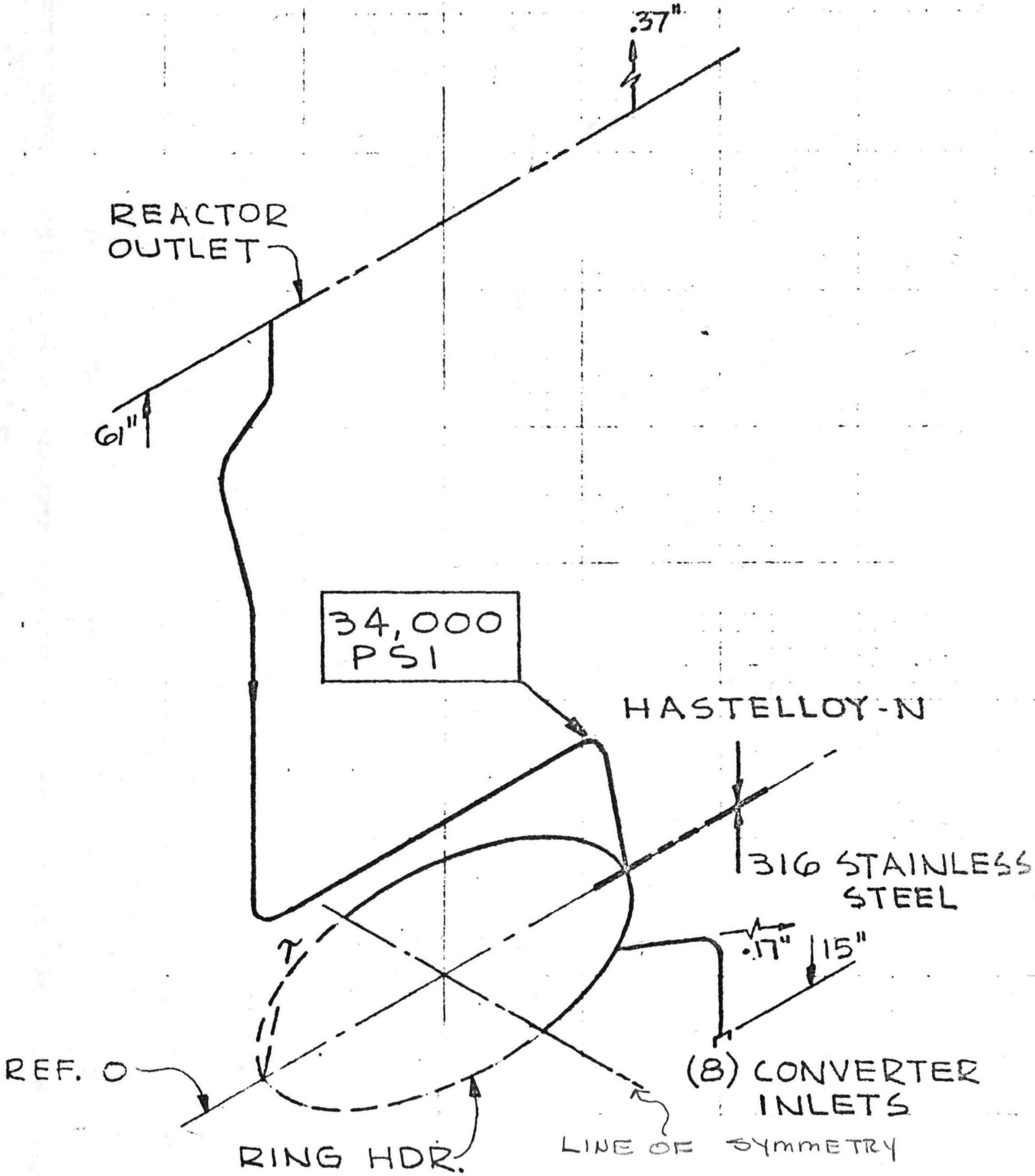

.37"
REACTOR
OUTLET
61"
34,000
PSI
HASTELLOY-N
316 STAINLESS
STEEL
.17"
15"
REF. O
(8) CONVERTER
INLETS
RING HDR.
LINE OF SYMMETRY

FIGURE 18

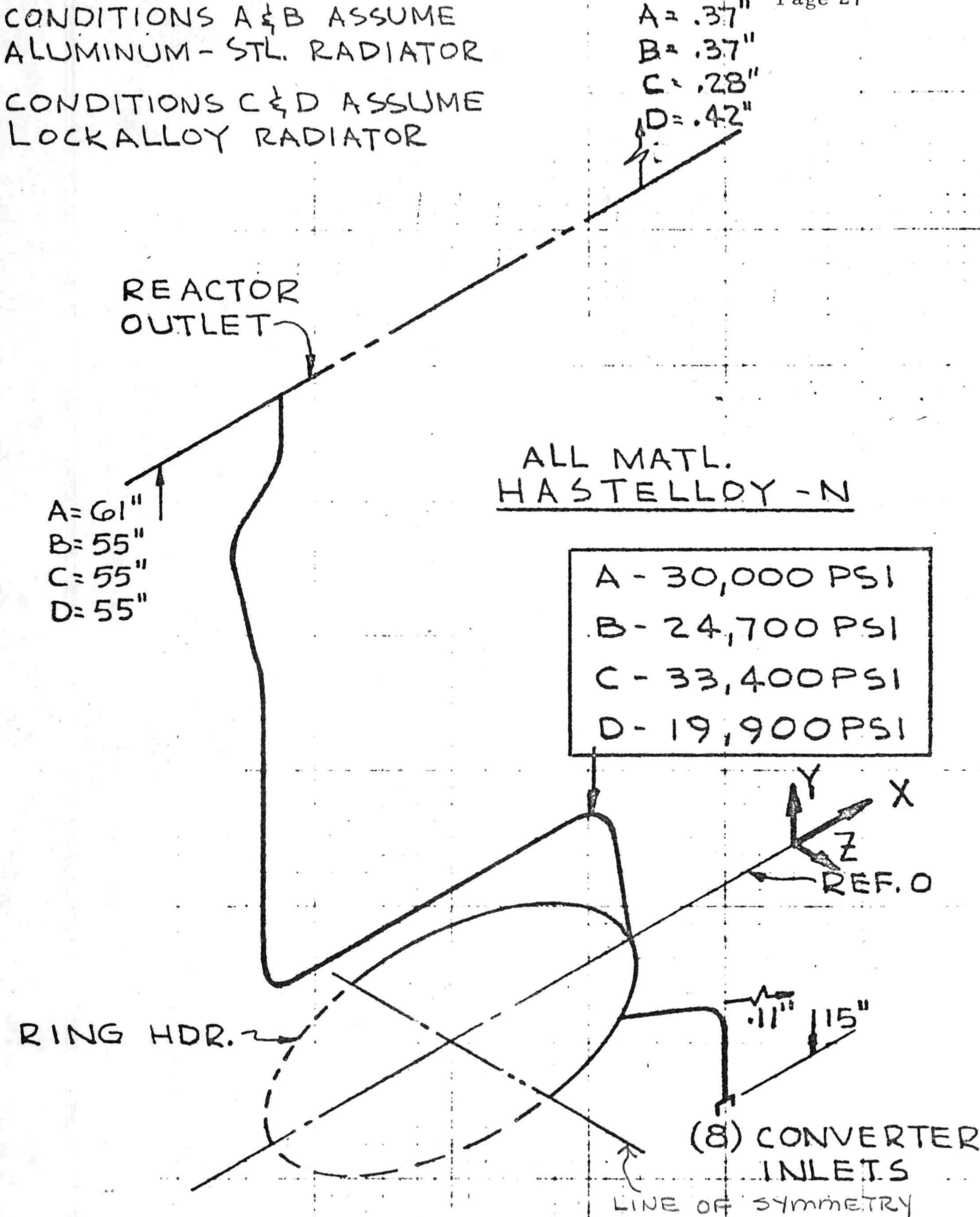

CONDITIONS A & B ASSUME
ALUMINUM - STL. RADIATOR
CONDITIONS C & D ASSUME
LOCKALLOY RADIATOR
A = .37"
B = .37"
C = .28"
D = .42"
REACTOR
OUTLET
A = 61"
B = 55"
C = 55"
D = 55"
ALL MATL.
HASTELLOY - N
A - 30,000 PSI
B - 24,700 PSI
C - 33,400 PSI
D - 19,900 PSI
Y
X
Z
REF. 0
RING HDR.
.11"
.15"
(8) CONVERTER
INLETS
LINE OF SYMMETRY

FIGURE 19

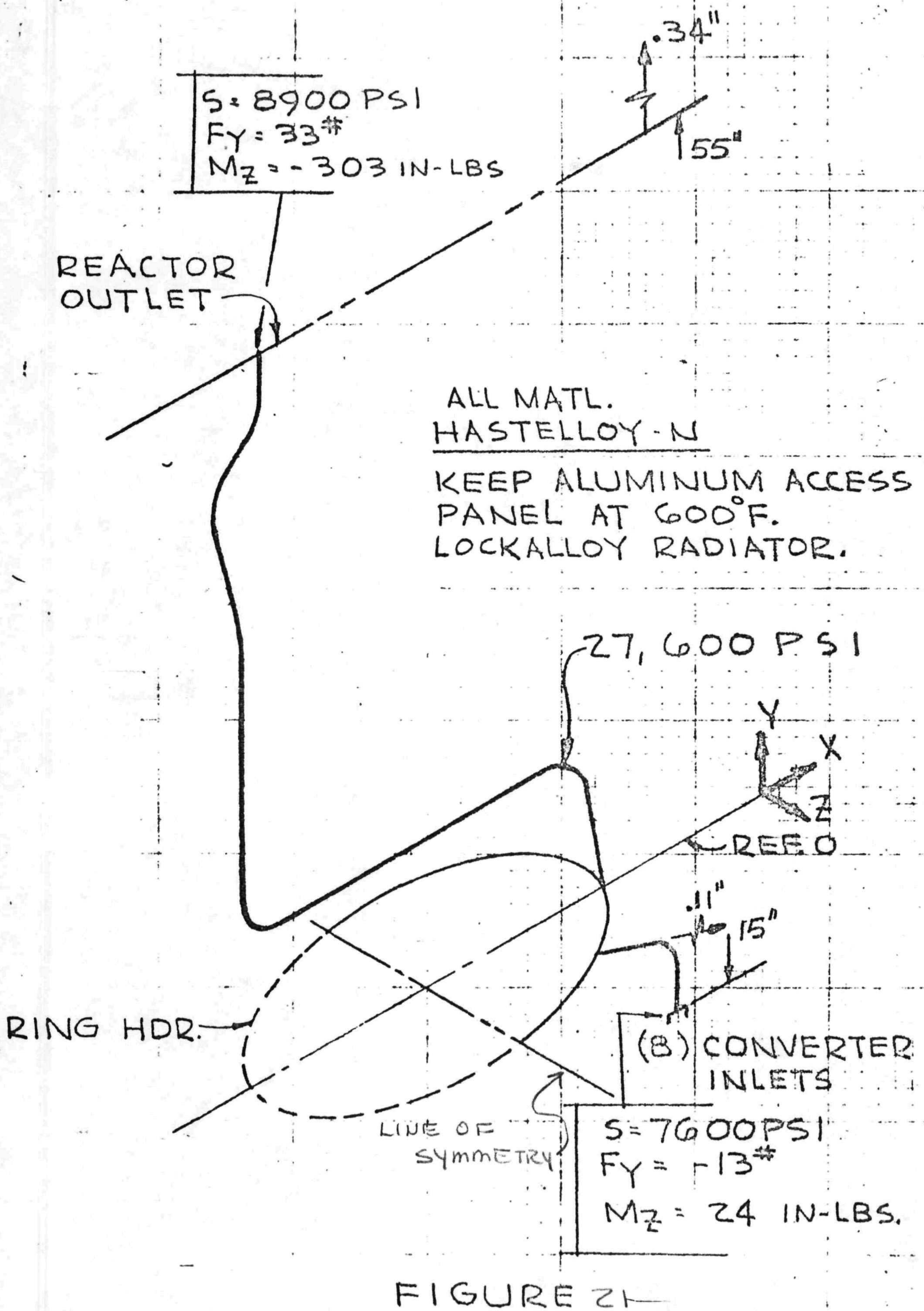
ALL HASTELLOY-N MATL.

CONDITION C - LOWER T/E BASKET ASSY. 6" TO ACCOMODATE LOOP, LOCKALLOY RADIATOR, STARTUP CONDITIONS.

CONDITION D - SAME AS C EXCEPT KEEPING ACCESS PANEL HEATED TO 1200°F.

	REACTOR OUTLET		CONVERTER INLET (MAX. REACTIONS)	
	C	D	C	D
STRESS, PSI	10,400	7000	7800	7300
FORCE, (F_Y) POUNDS	40	24	16	10
MOMENT, IN-LBS.	M_Z - -353	M_Z - -236	M_X - 25	M_X - 19

FIGURE 20



.34"
S = 8900 PSI
FY = 33#
MZ = -303 IN-LBS
55"
REACTOR
OUTLET
ALL MATL.
HASTELLOY-N
KEEP ALUMINUM ACCESS
PANEL AT 600°F.
LOCKALLOY RADIATOR.
27,600 PSI
Y
X
Z
REF 0
.11"
.15"
RING HDR.
(8) CONVERTER
INLETS
LINE OF
SYMMETRY
S = 7600 PSI
FY = -13#
MZ = 24 IN-LBS.

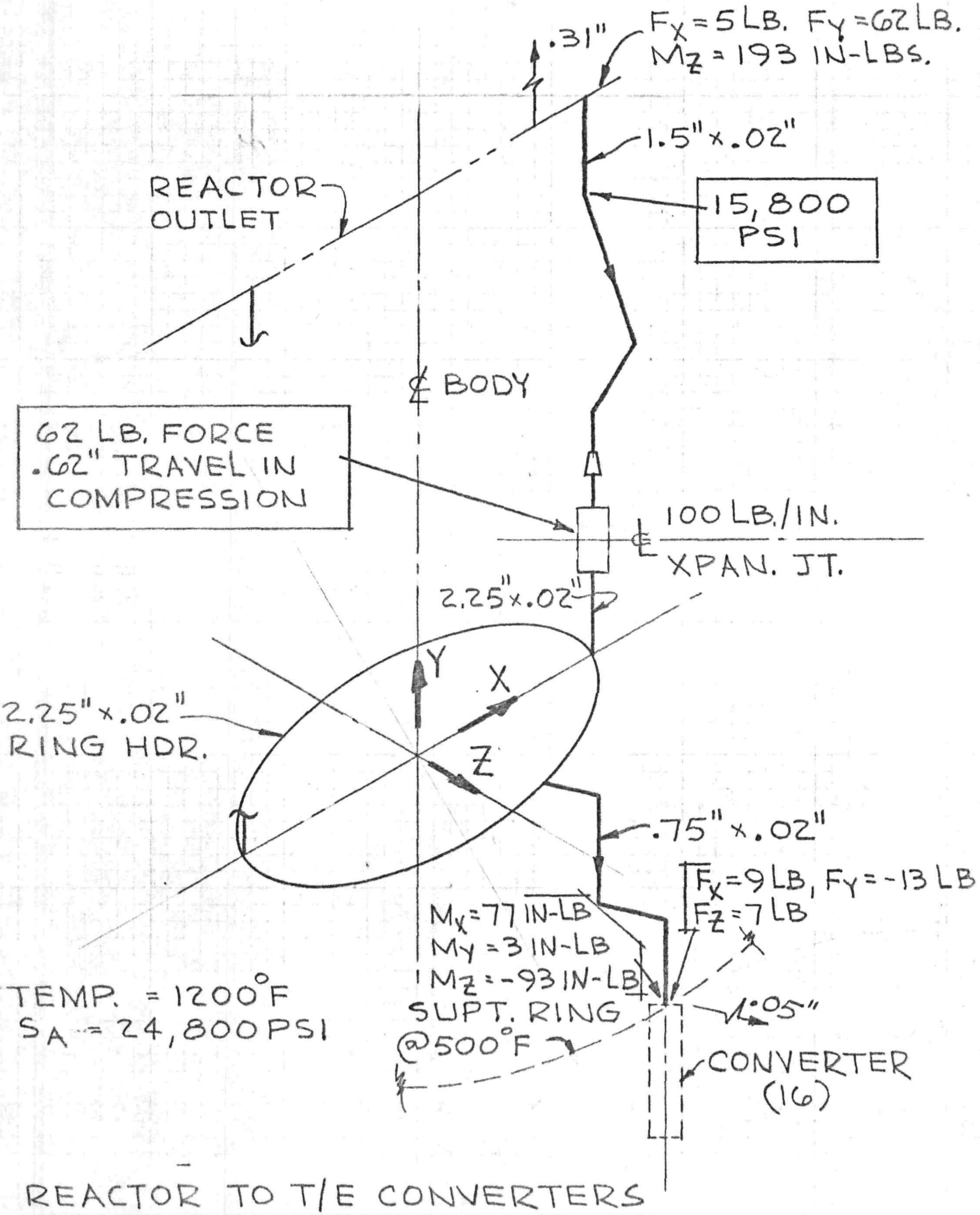
FIGURE 21

Part B

1. Reactor to T/E Converters

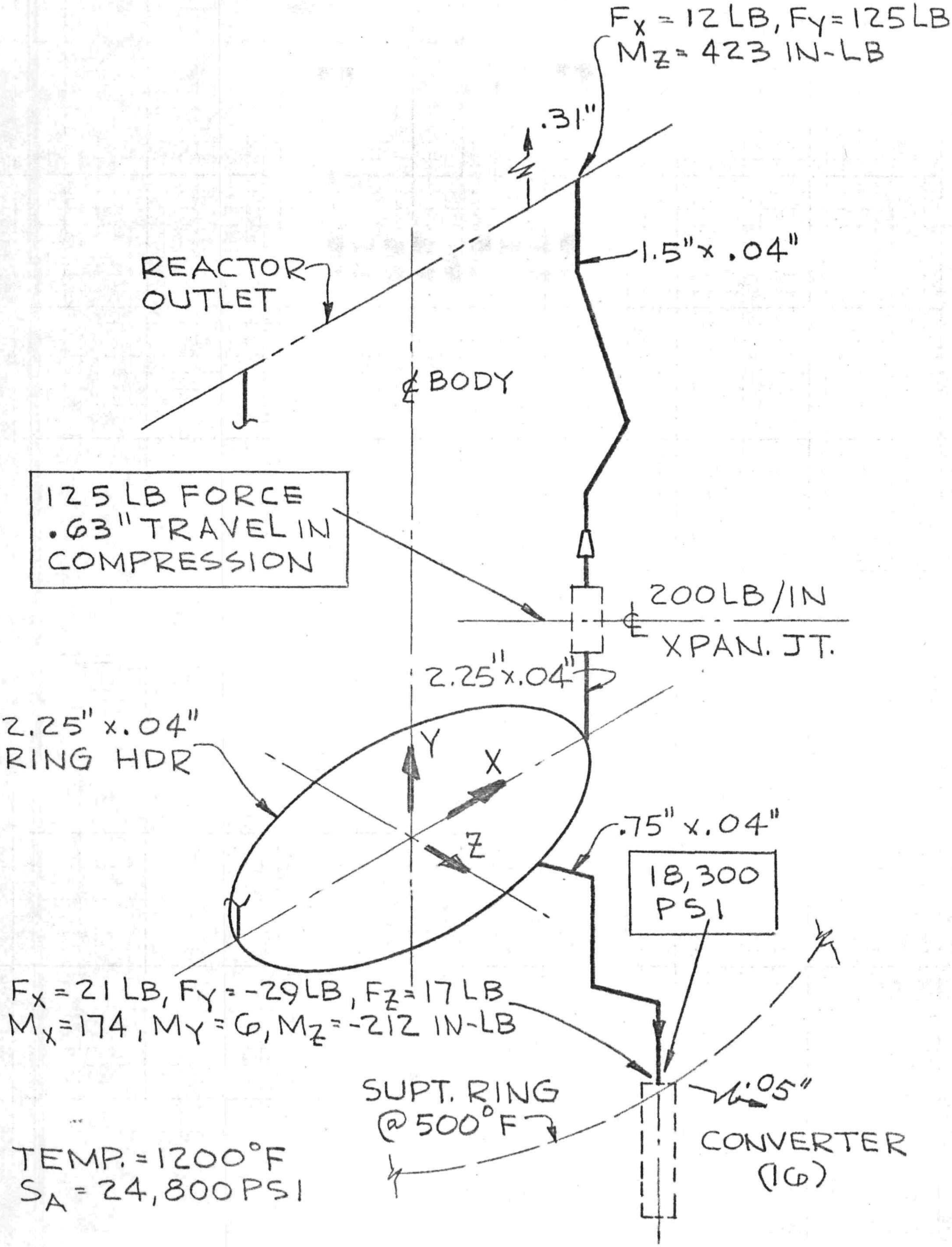
This section deals with the thermal reference design layout. Figure 22 shows the results of these calculations for an expansion joint with a spring rate of 100 lb/in. and a 0.02 in. wall piping system. Thermal stresses for 100 lb/in. and 200 lb/in. expansion joints were also calculated for a 0.04 in. wall piping system (Figures 23 and 24). When the expansion joint was finalized to a spring rate of 220 lb/in., the thermal bending stresses were again calculated based on this spring rate (Figure 25).

In the preceding calculations the expansion joint was modeled as a sleeve-type expansion joint. It was then modeled with a different mathematical configuration as a component (bellows) and the stresses recalculated (Figure 26).

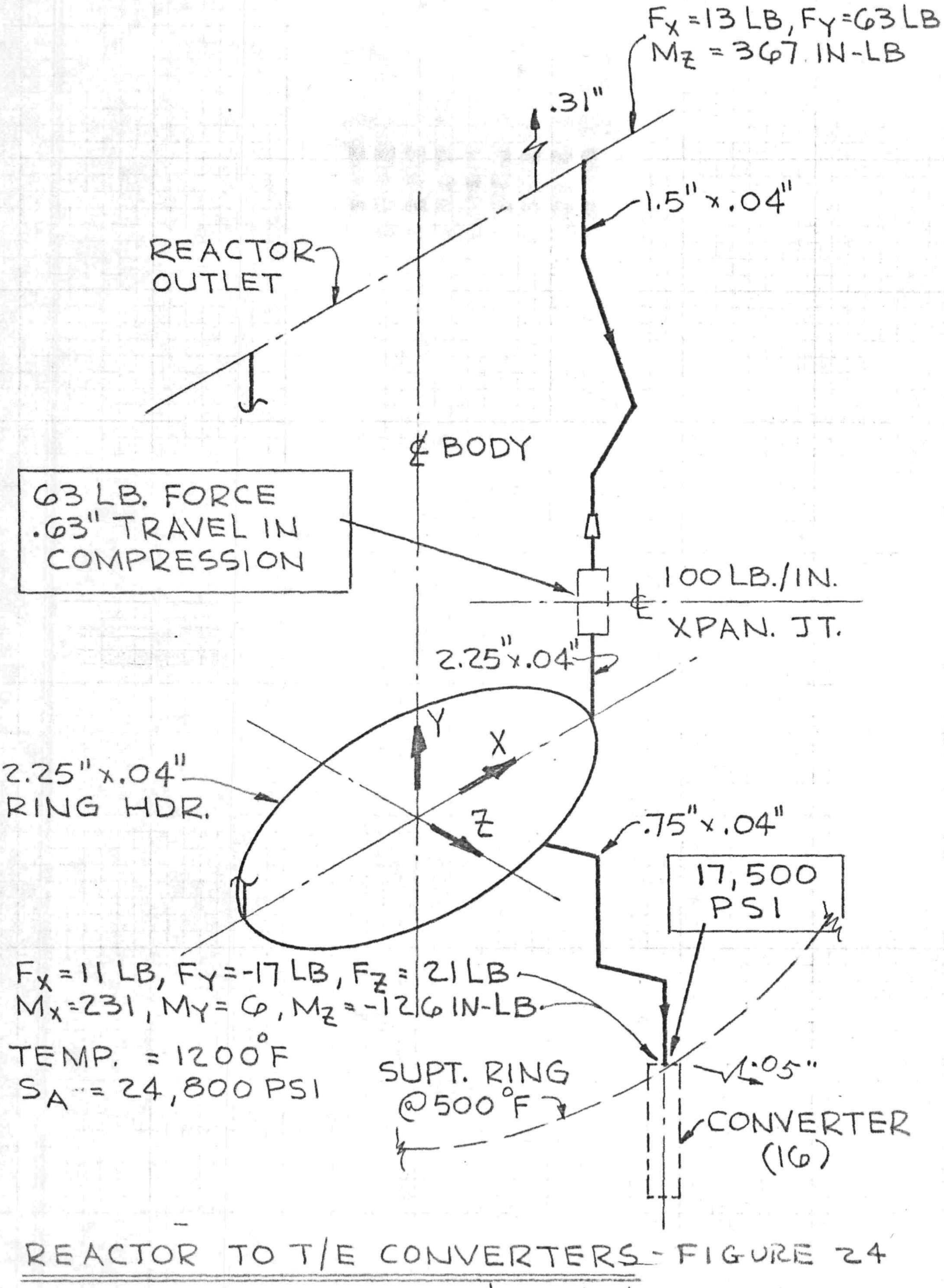


REACTOR TO T/E CONVERTERS

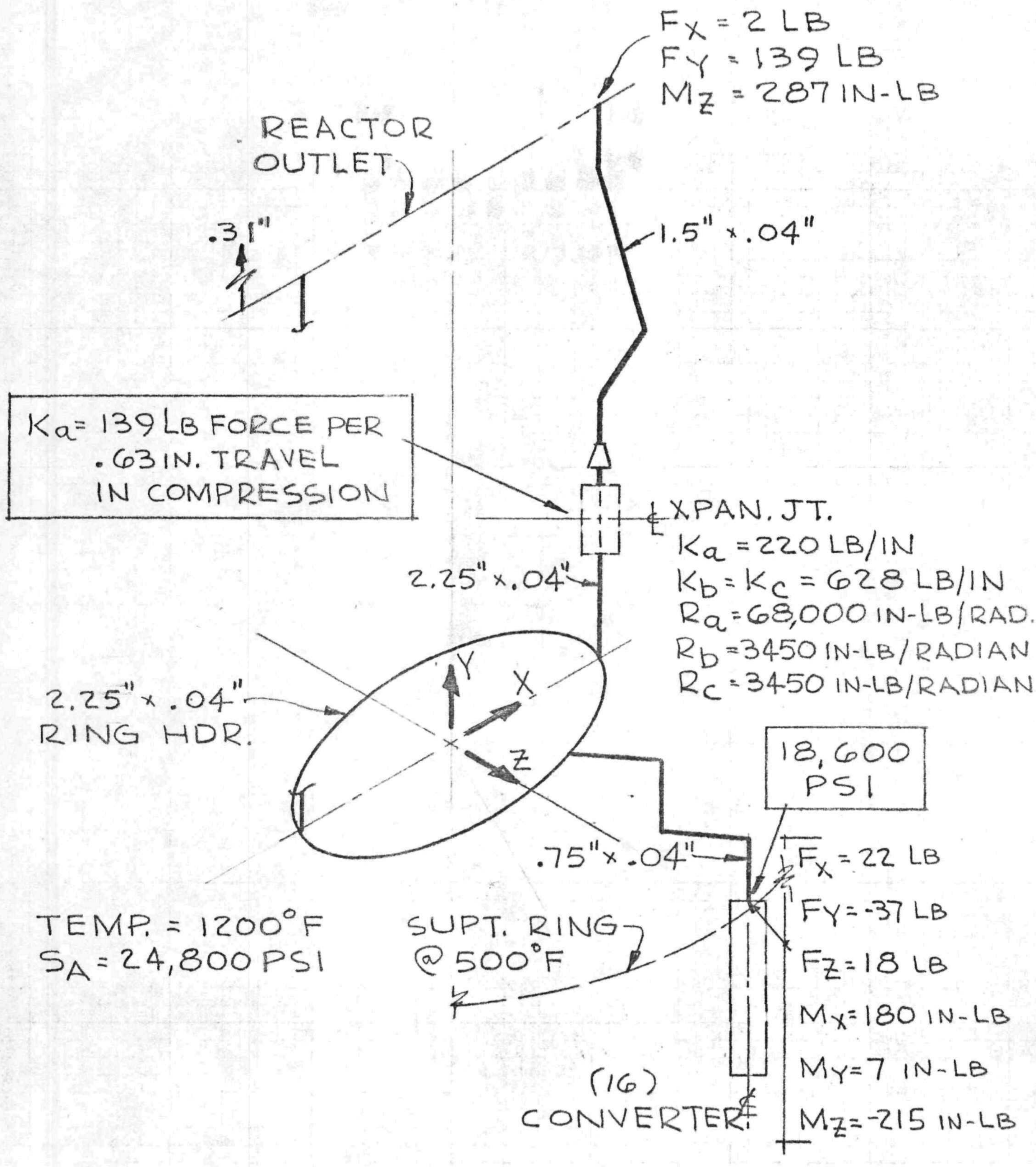
REF. DESIGN - FIGURE 22



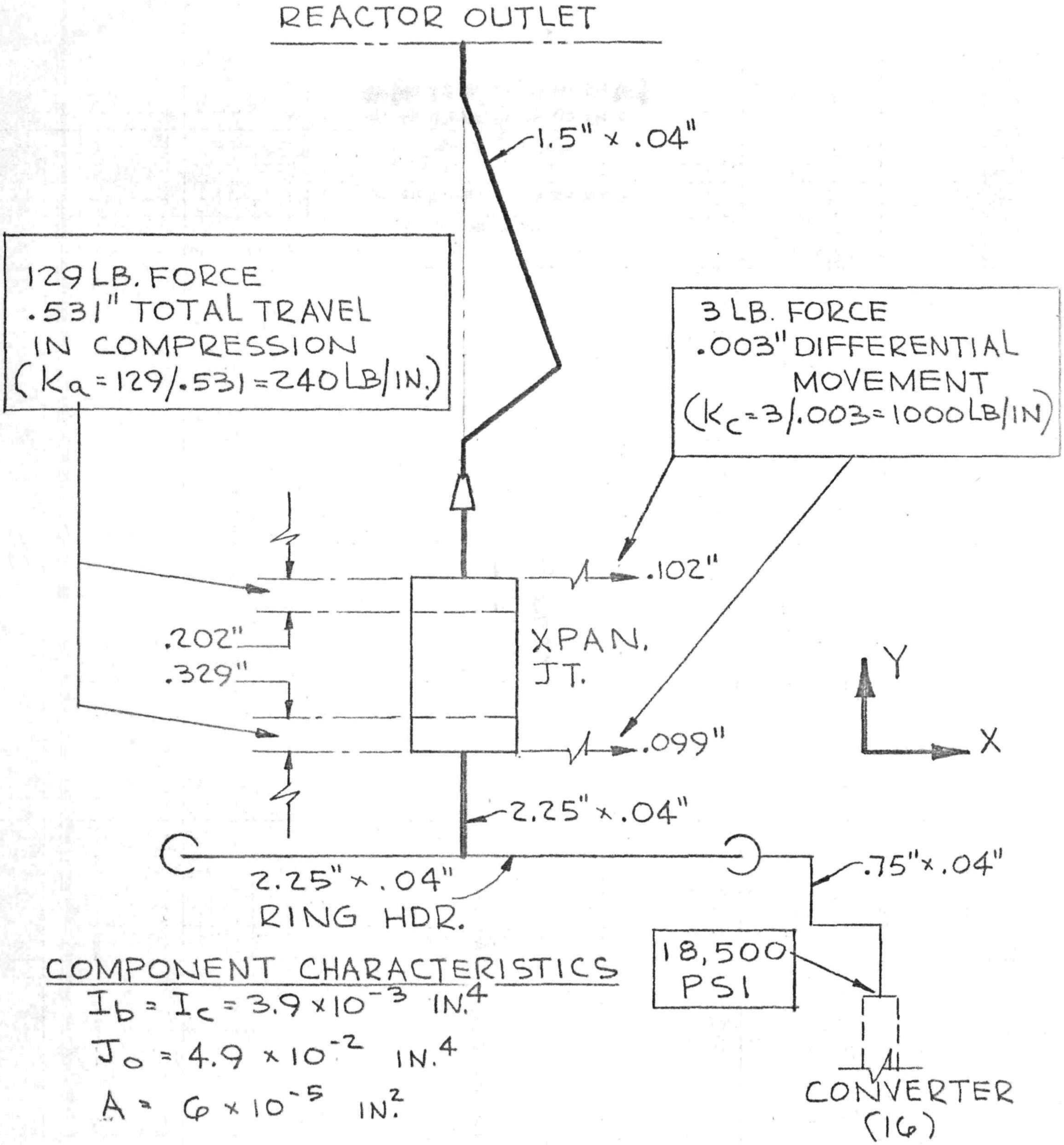
REACTOR TO T/E CONVERTERS - FIGURE 23
WITH .04" WALL PIPE & 200 LB/IN XPAN. JT.



REACTOR TO T/E CONVERTERS - FIGURE 24
WITH .04" WALL PIPE & 100 LB/IN XPAN. JT.



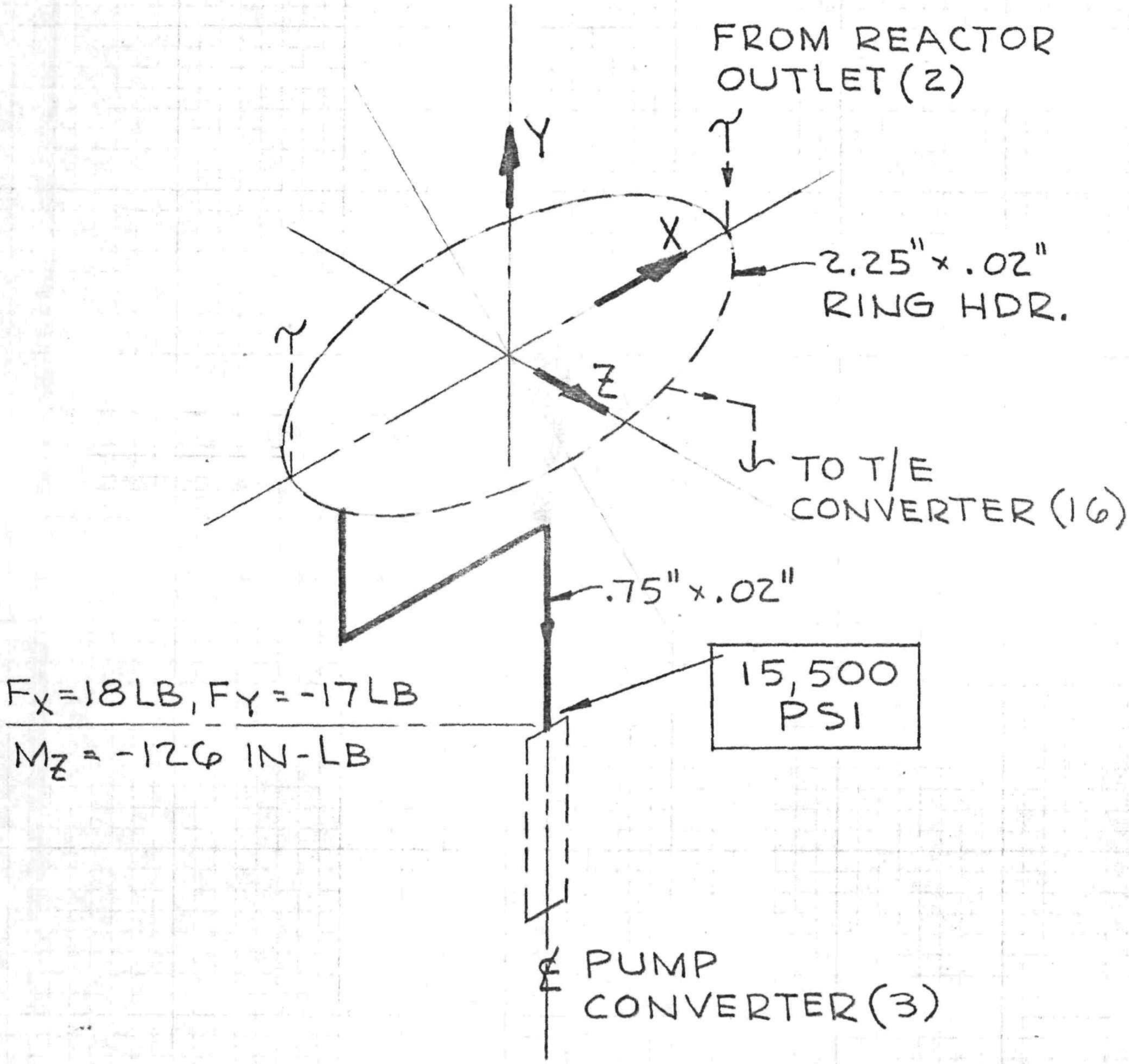
REACTOR TO T/E CONVERTERS WITH CALCULATED XPAN. JT. SPRING RATES — FIGURE 25



REACTOR TO T/E CONVERTERS
WITH XPAN. JT. MODELED AS
A COMPONENT - FIGURE 26

2. Reactor Outlet to Pump Converters

Figure 27 shows the results of the bending stresses. This system was modeled separately, because it would have been quite cumbersome to try to incorporate it into the main system.

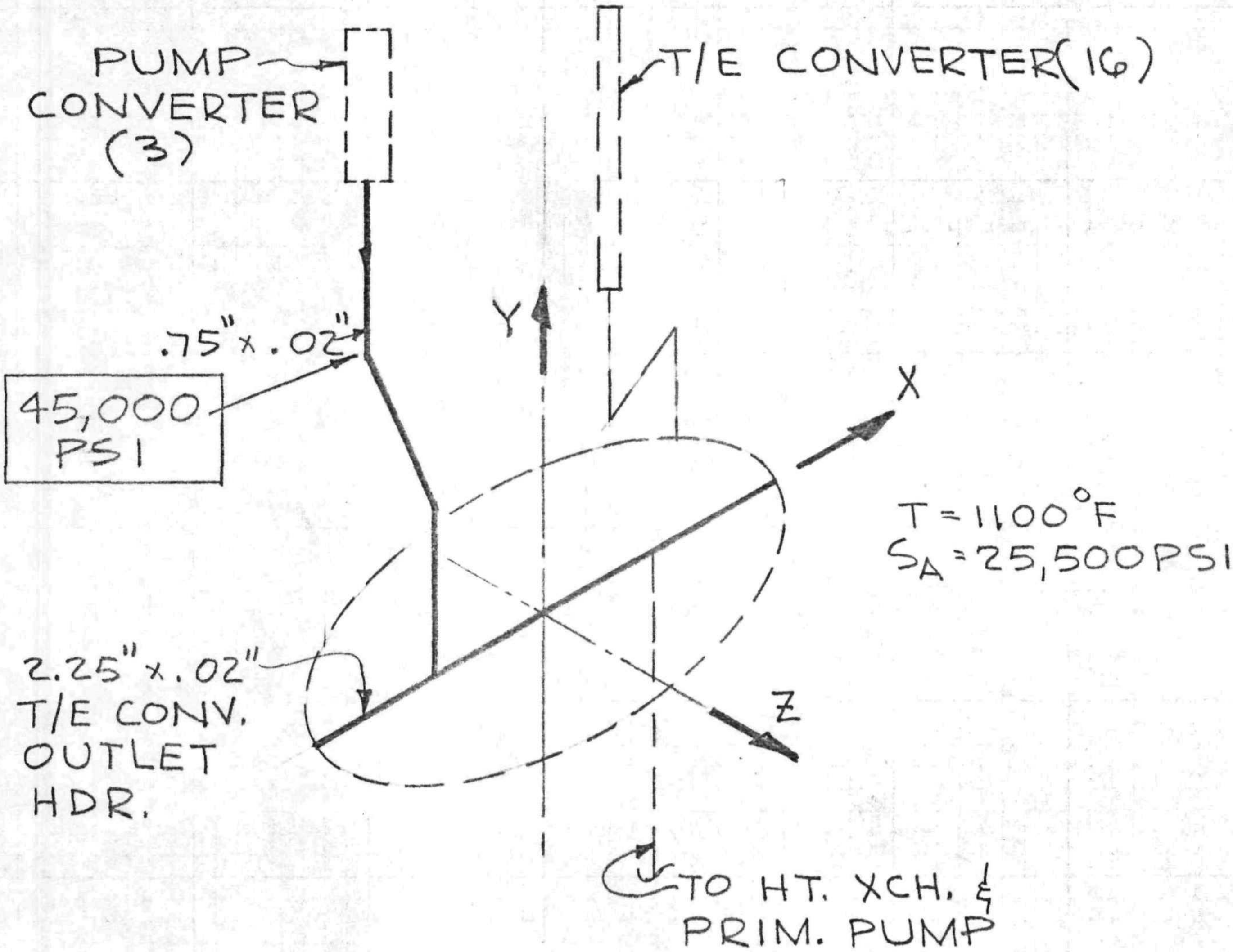


REACTOR OUTLET TO PUMP CONVERTERS

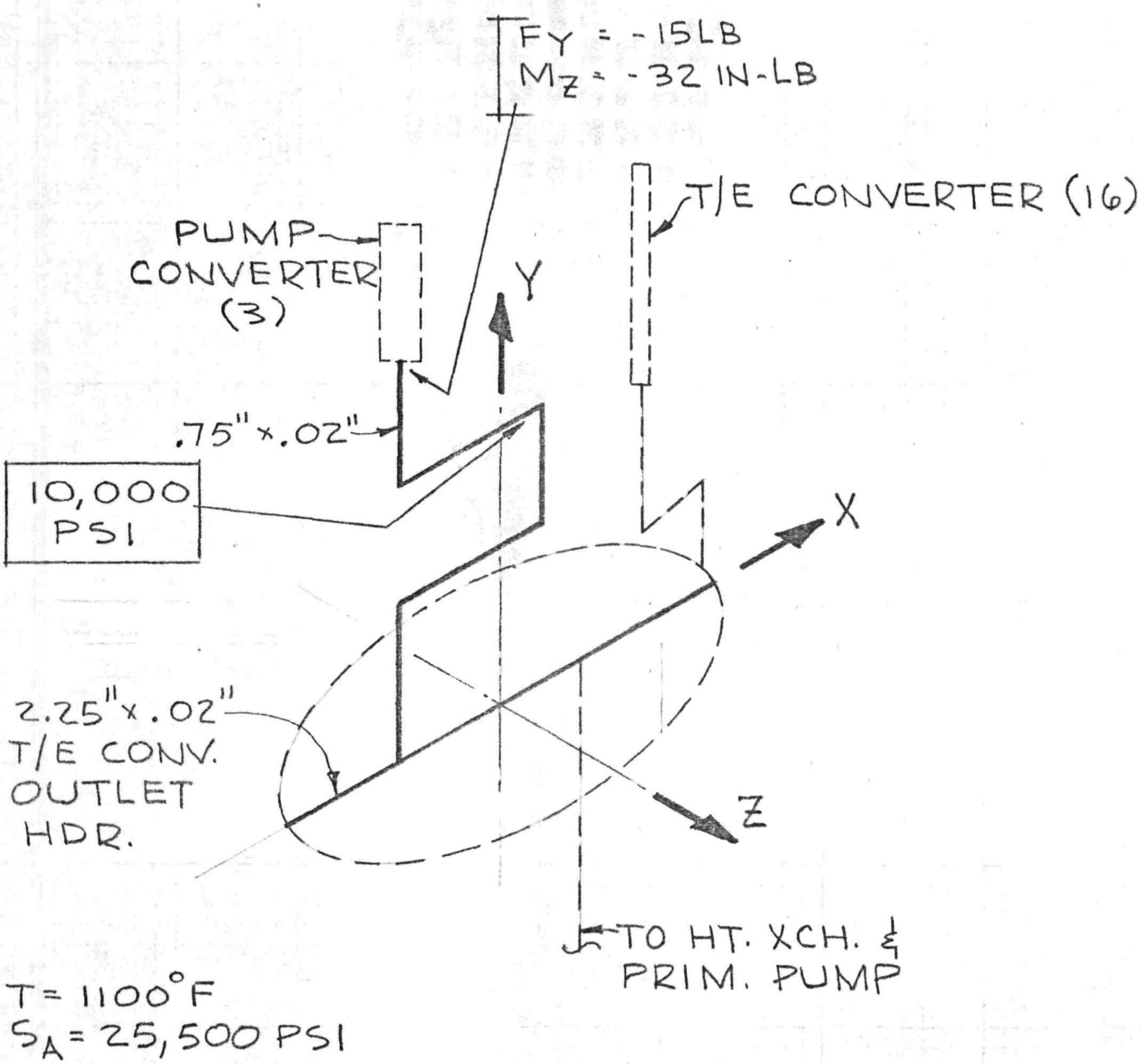
FIGURE 27

3. Pump Converters to Heat Exchanger to Primary Pump

A half-loop was added to this system in an attempt to reduce the bending stresses to a value below the allowable, S_A, of 25,500 psi. This method did not work (Figure 28). A full-loop was added which reduced the bending stresses to 10,000 psi which was less than 'S_A' (Figure 29). The final reference design for this system incorporates the full loop as recommended by this analysis.



PUMP CONVERTERS TO HT. XCH. TO PRIM. PUMP WITH HALF-LOOP ADDED - FIGURE 28



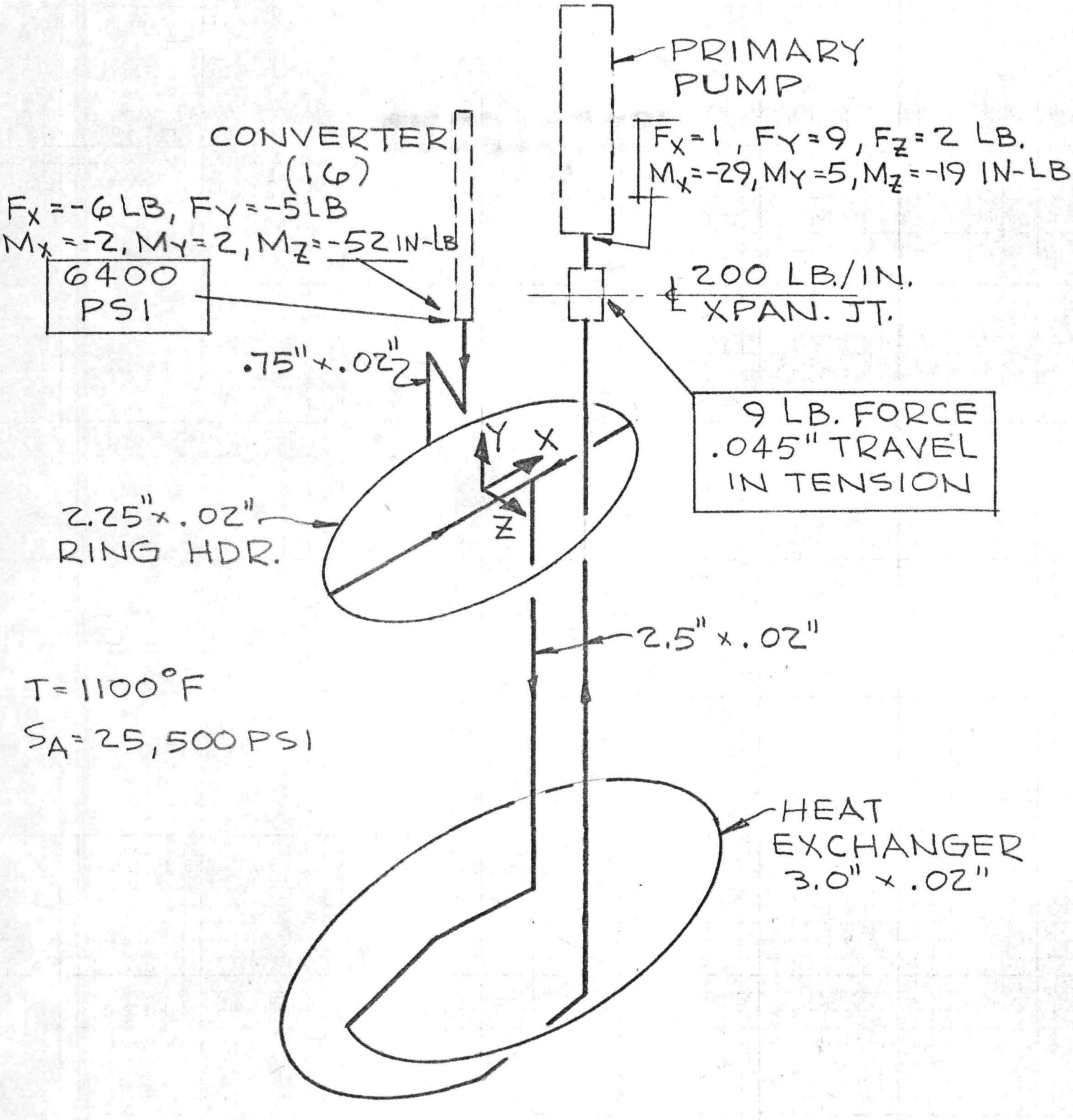
PUMP CONVERTERS TO HT. XCH. TO PRIM. PUMP

WITH FULL LOOP ADDED - FIGURE 29

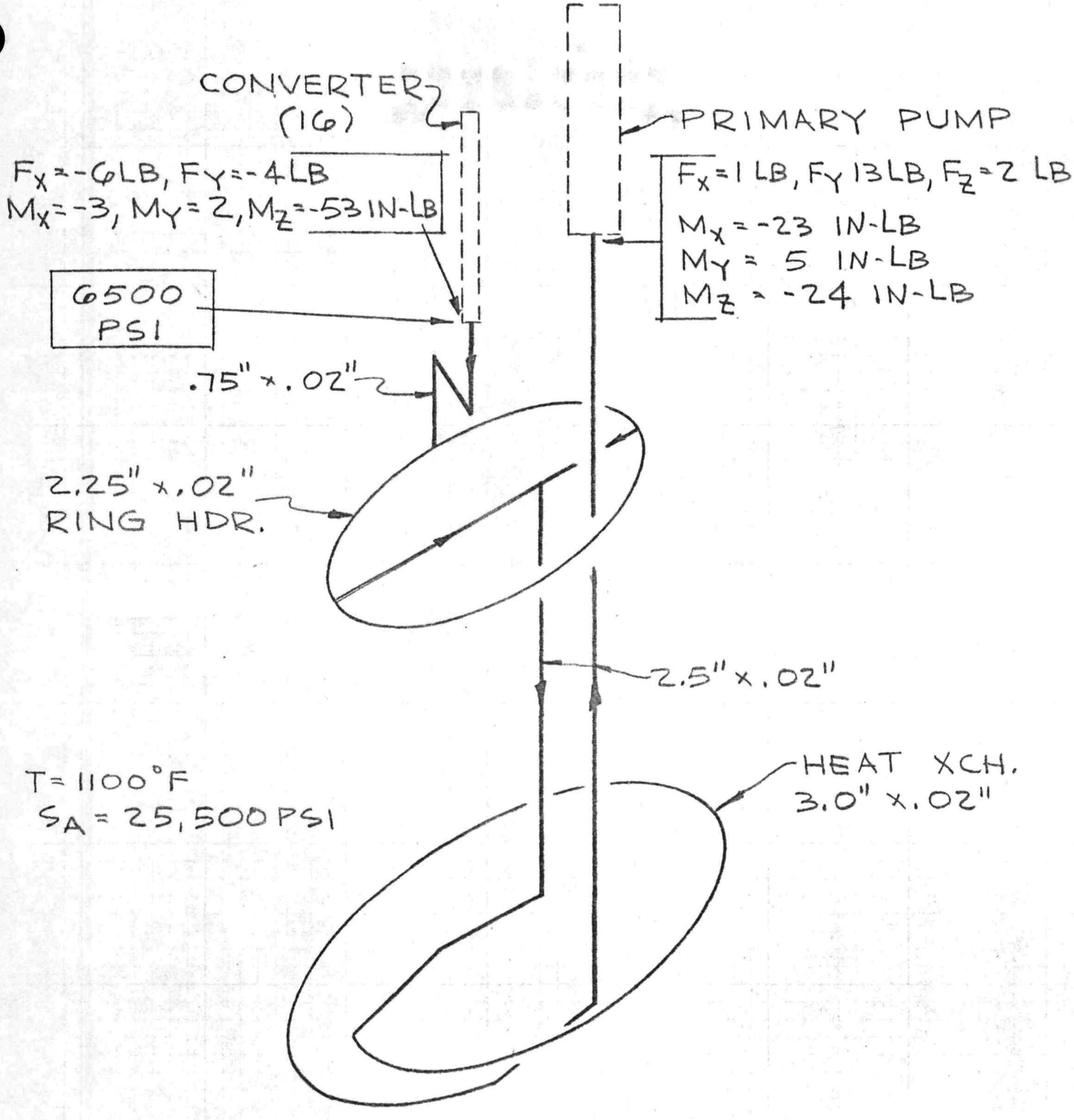
4. T/E Converters to Primary Pump

Figure 30 shows the bending stresses in the piping system when a heat exchanger is added and an expansion joint with a spring rate of 200 lb/in. is used. When the expansion joint was removed there was little change in the bending stresses (Figure 31).

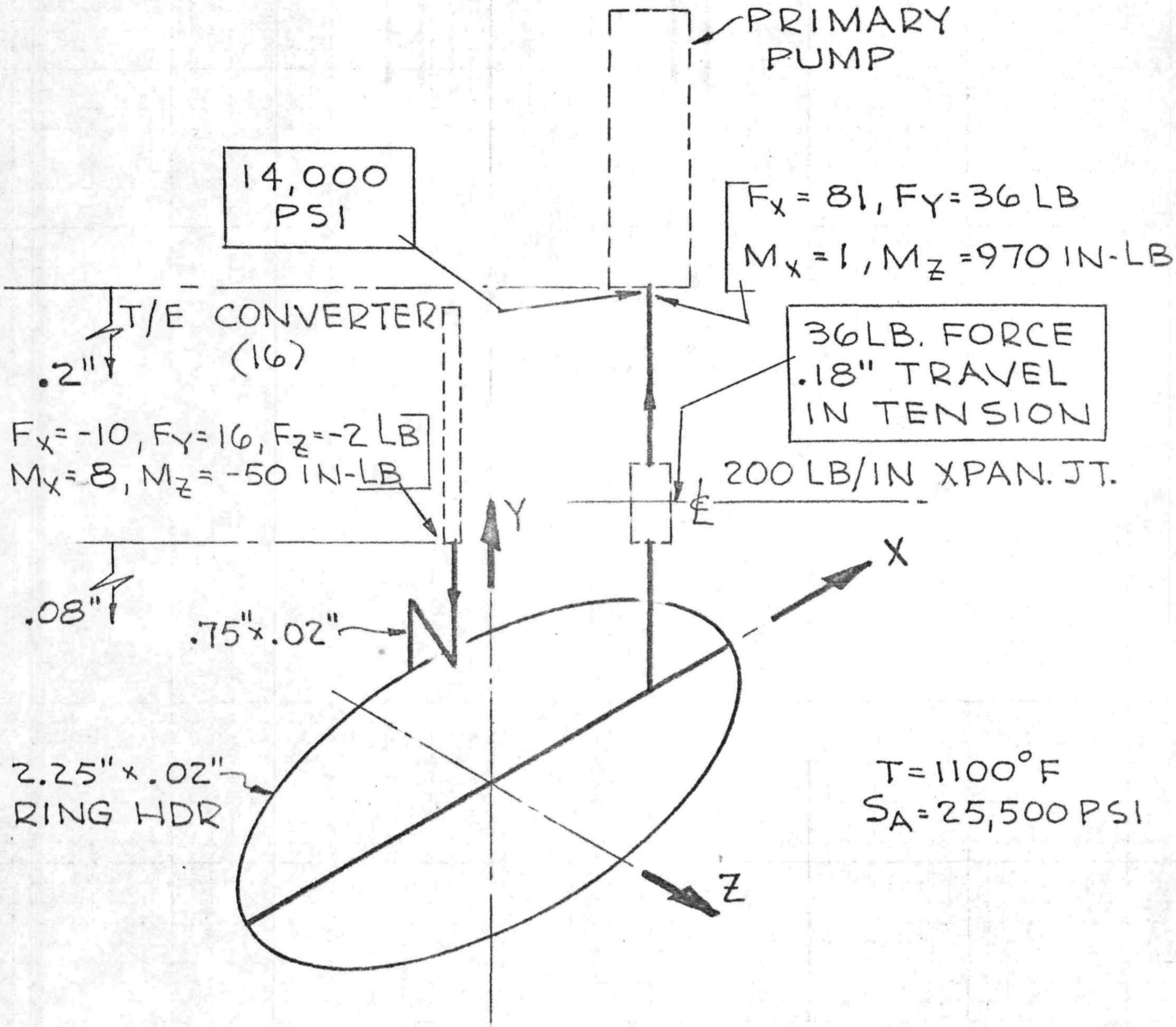
Figure 32 shows the thermal bending stresses for the piping system with the heat exchangers omitted. This configuration represents the final reference design concept.



T/E CONVERTERS TO PRIM. PUMP — FIGURE 30
WITH HEAT XCH. & 200 LB/IN XPAN. JT.



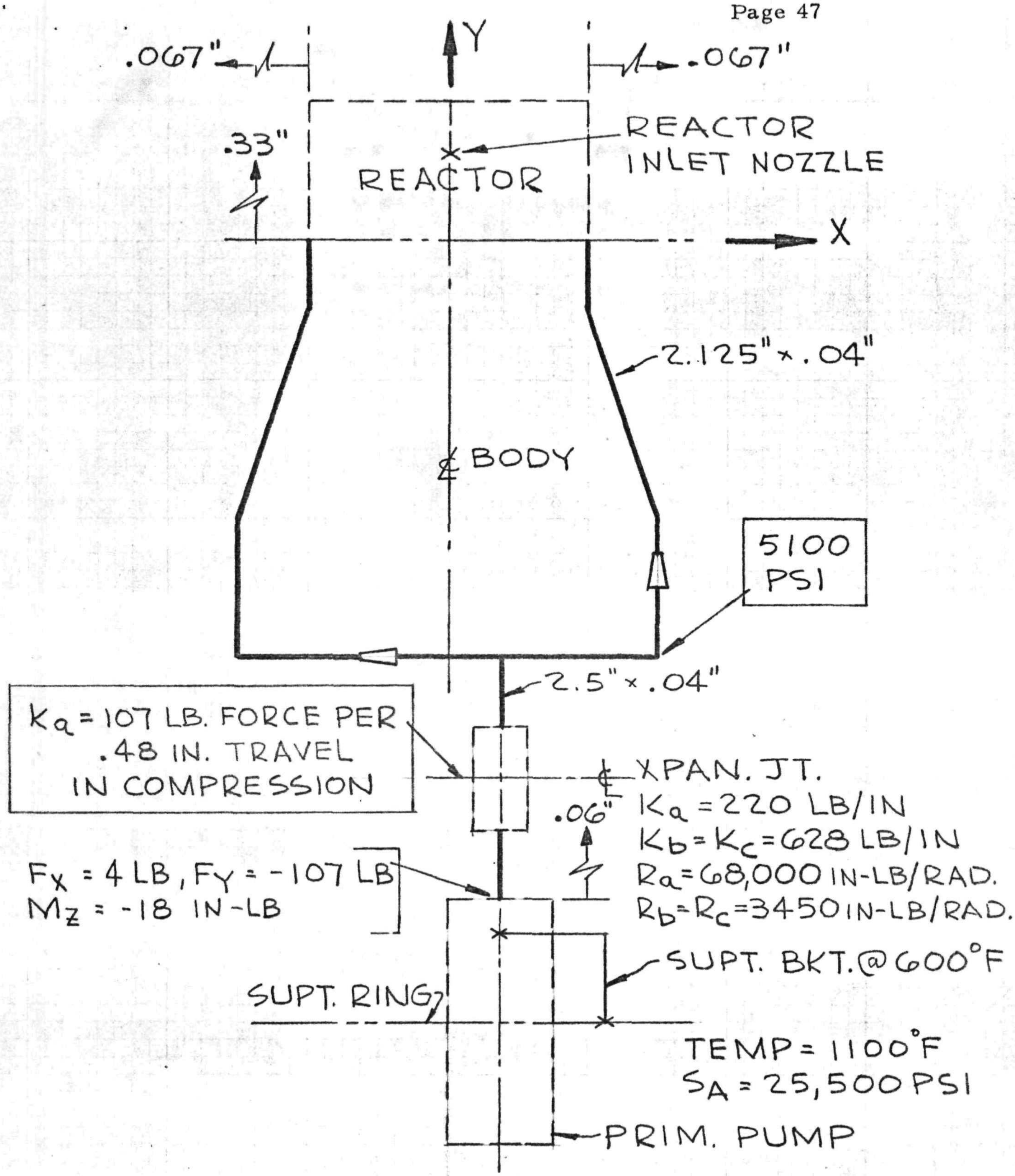
T/E CONVERTERS TO PRIM. PUMP WITH HEAT XCH. — FIGURE 31



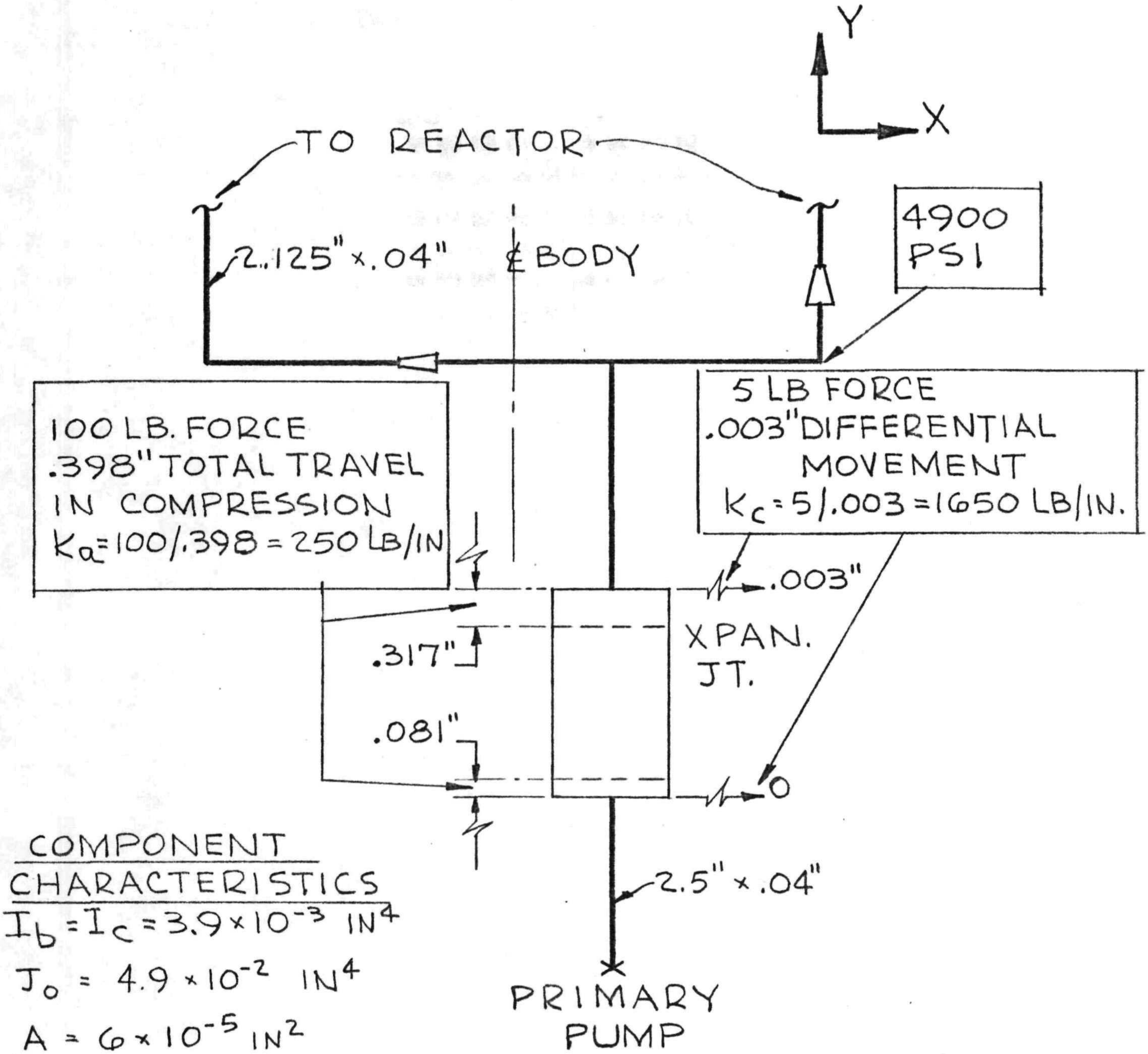
T/E CONVERTERS TO PRIMARY PUMP - FIGURE 32

5. Primary Pump Discharge to Reactor

Figure 33 shows the 0.04 inch wall, piping system with an expansion joint of 200 lb/in. The bending stresses do not exceed the allowable, S_A = 25,500 psi. In this analysis the expansion joint was modeled as a sleeve - type expansion joint. The bending stresses were re-calculated when the value of 220 lb/in was chosen as the spring rate of the expansion joint (Figure 34 and 35). In these calculations the expansion joint was modeled as a component. This modeling gives values for the differential movement of the ends of the expansion joint.



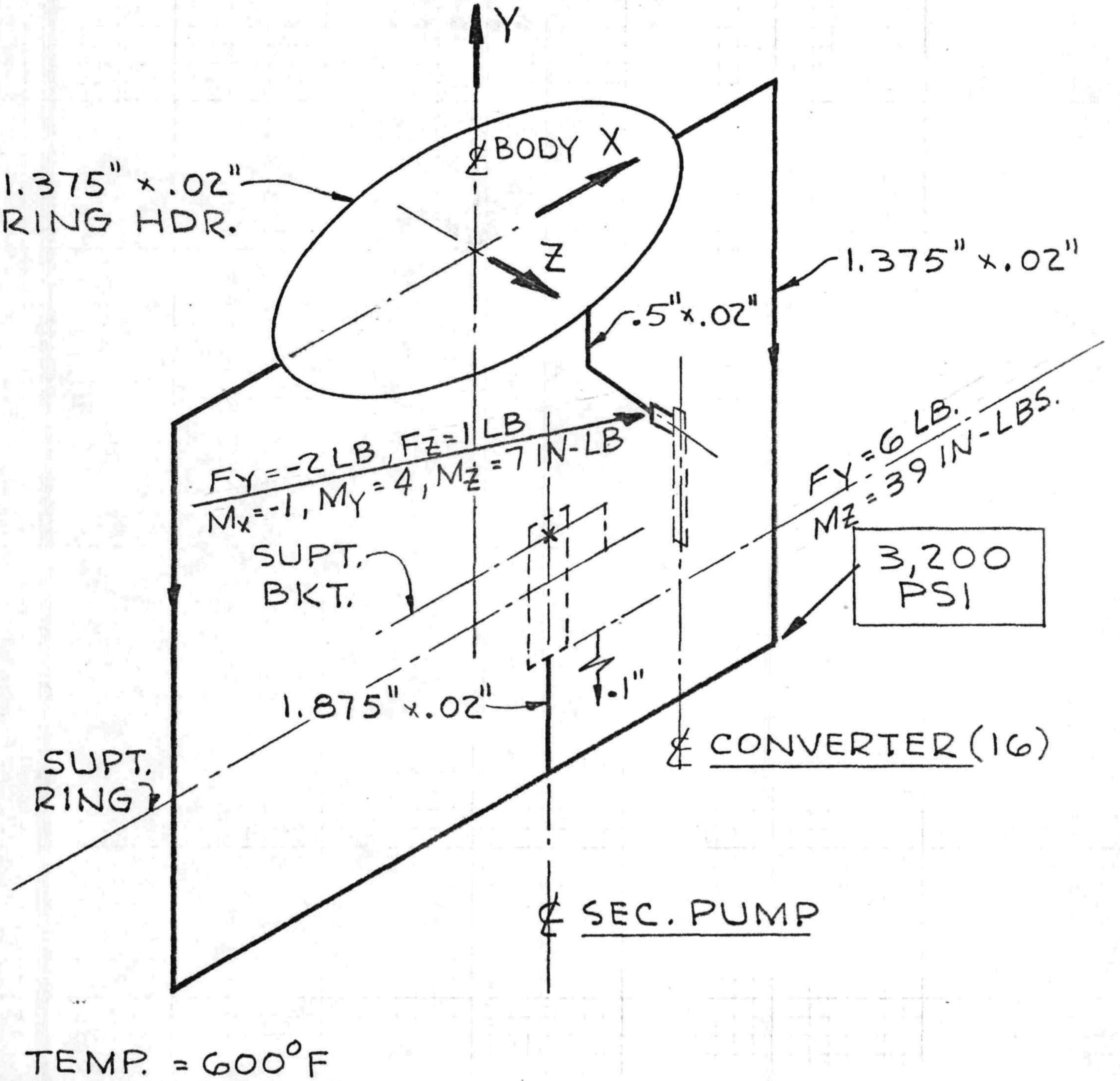
PRIM. PUMP DISCH. TO REACTOR WITH CALCULATED XPAN. JT. SPRING RATES — FIGURE 34



PRIMARY PUMP DISCH. TO REACTOR WITH XPAN. JT MODELED AS A COMPONENT - FIGURE 35

6. T/E Converters to Secondary Pump Inlet

The secondary piping system bending stresses are shown in Figure 36. Problems existed between the relative expansion of the stainless steel piping system and the radiator which was an aluminum alloy. The coefficient of thermal expansion was much higher for the aluminum alloy (2219) than for the stainless steel at 600°F. After much research, lockalloy was chosen as the material for the radiator because it had the same coefficient of thermal expansion at 600°F as stainless steel, thereby, eliminating the problem of relative expansion of the secondary heat rejection system.

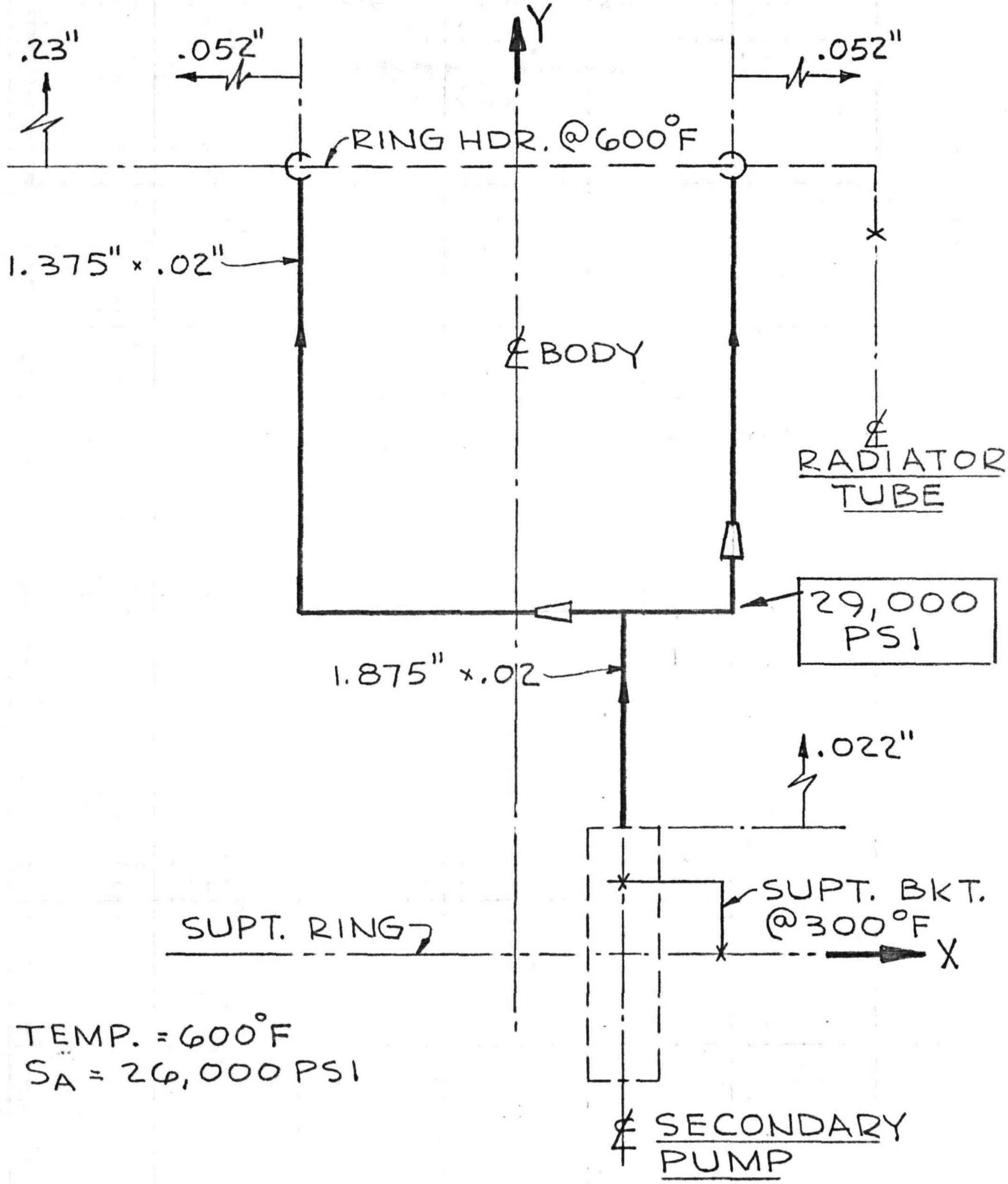


TEMP. = 600°F
S_A = 26,000 PSI

T/E CONVERTERS TO SEC. PUMP INLET - FIGURE 36

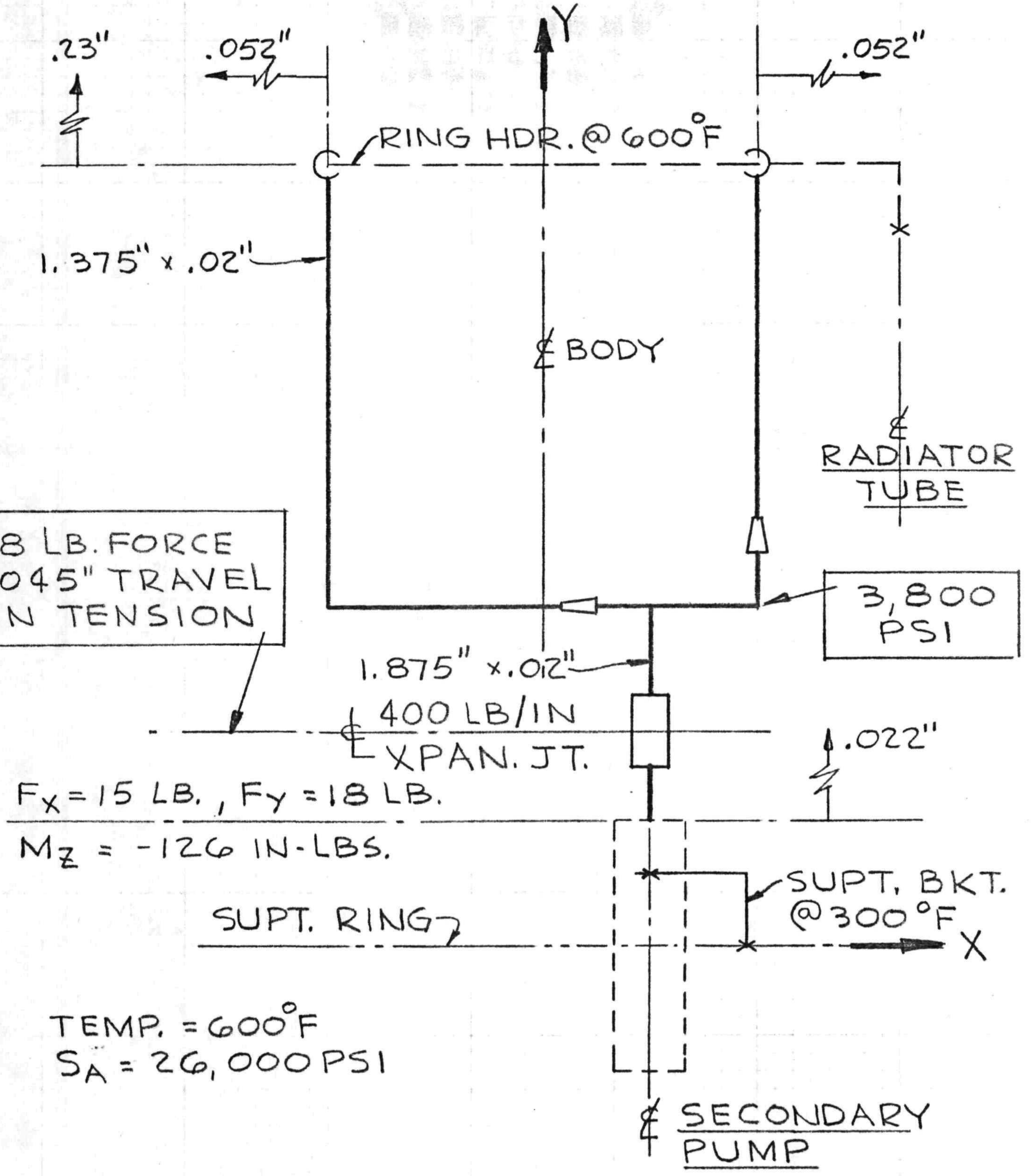
7. Secondary Pump Discharge to Radiator

Figure 37 shows the piping system bending stresses. This particular configuration was not acceptable because the stresses exceeded the allowable, $S_A = 26,000$ psi. In Figure 38, a 400 lb/in expansion joint was selected arbitrarily. This reduced the bending stresses to values below S_A. Another method using offset was tried (Figure 39). This offset reduced the stresses but it was physically impossible to incorporate it into the design. None of these configurations have been added to the reference design as yet.

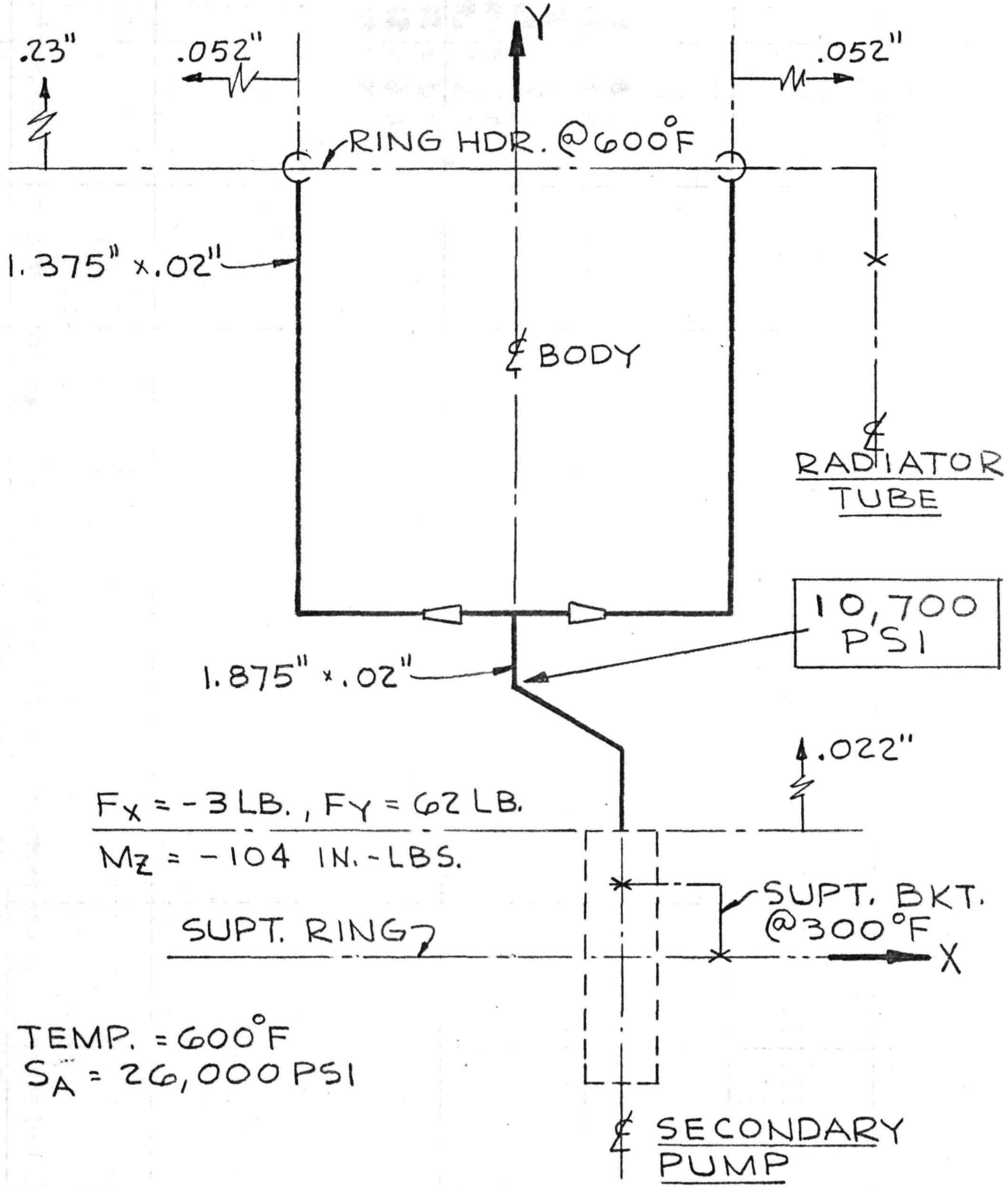


SEC. PUMP DISCH. TO RADIATOR

REF. DESIGN - FIGURE 37



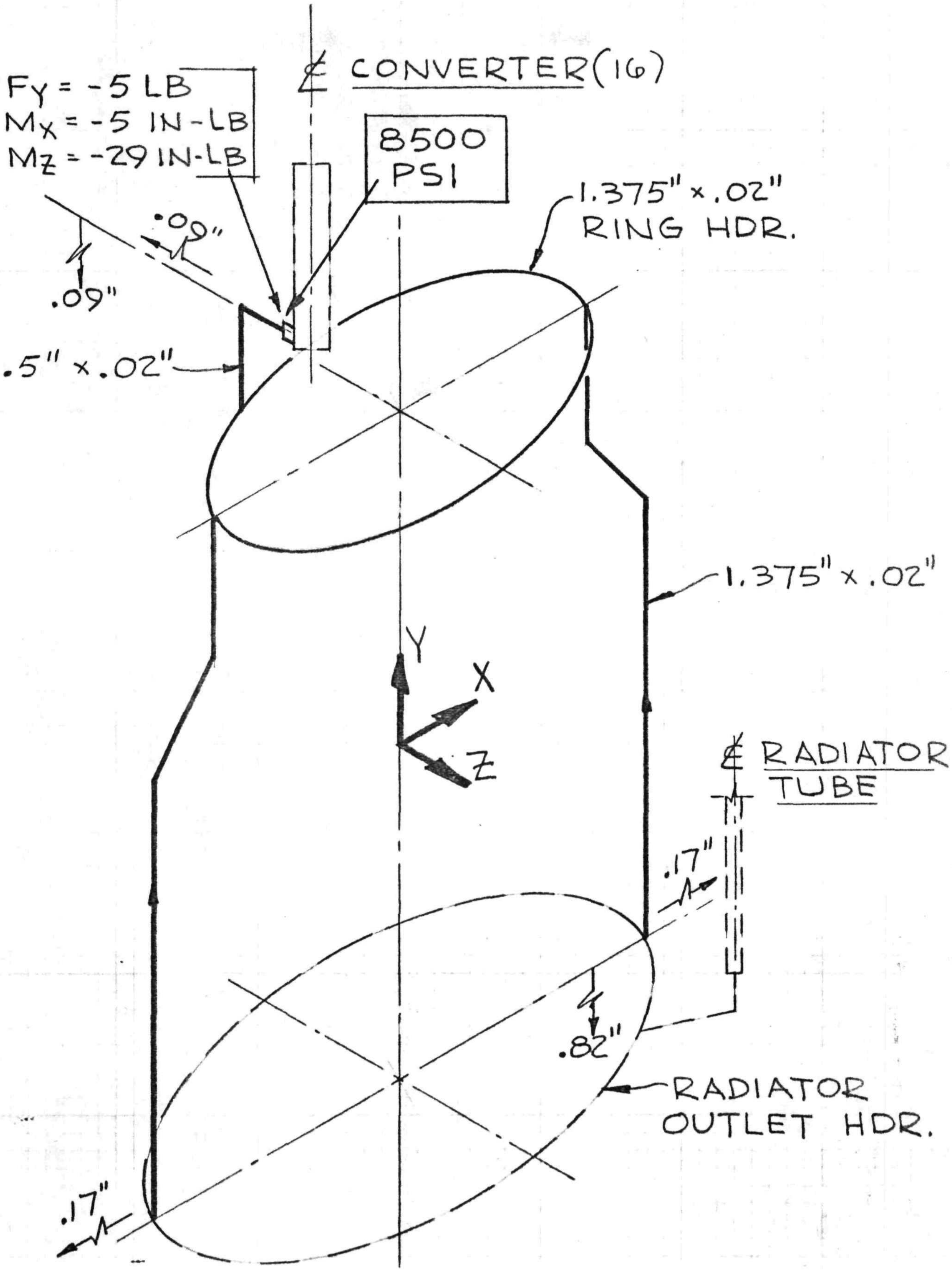
SEC. PUMP DISCH. TO RADIATOR
WITH ADDED XPAN. JT. - FIGURE 38



SEC. PUMP DISCH. TO RADIATOR WITH REVISED PIPING (1) - FIGURE 39

8. Radiator to T/E Converters

Figure 40 shows the bending stresses for the secondary system. Because lockalloy was chosen for the radiator, problems of relative expansion were solved.



RADIATOR TO T/E CONVERTERS - FIGURE 40

IV. PRIMARY STRESS ANALYSIS

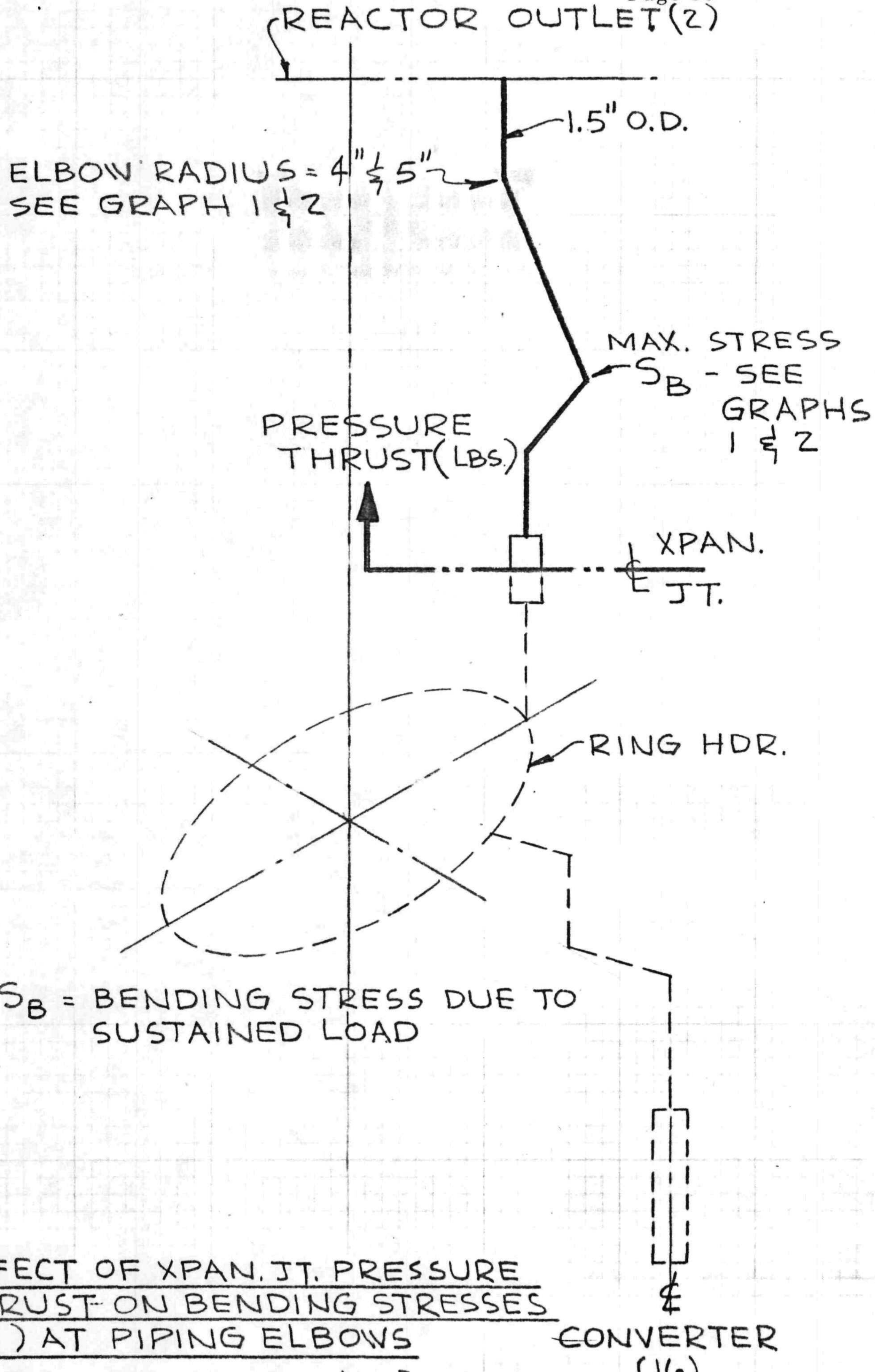
A. Effects of Expansion Joint Thrust Pressures on the Primary Piping System

Based on the calculations in the thermal analysis it was determined that an expansion joint would be necessary to relieve the bending stresses in the piping system. The primary bending stresses for corresponding pressure thrusts for four and five-inch bend radii were calculated (Figures 41 through 43).

The maximum allowable primary stress for the piping system is 6415 psi, (Figure 4). Using this value, the maximum pressure thrusts for different wall thicknesses at the elbow of maximum stress were found.

1. Expansion Joint to Reactor Outlet - Figure 41

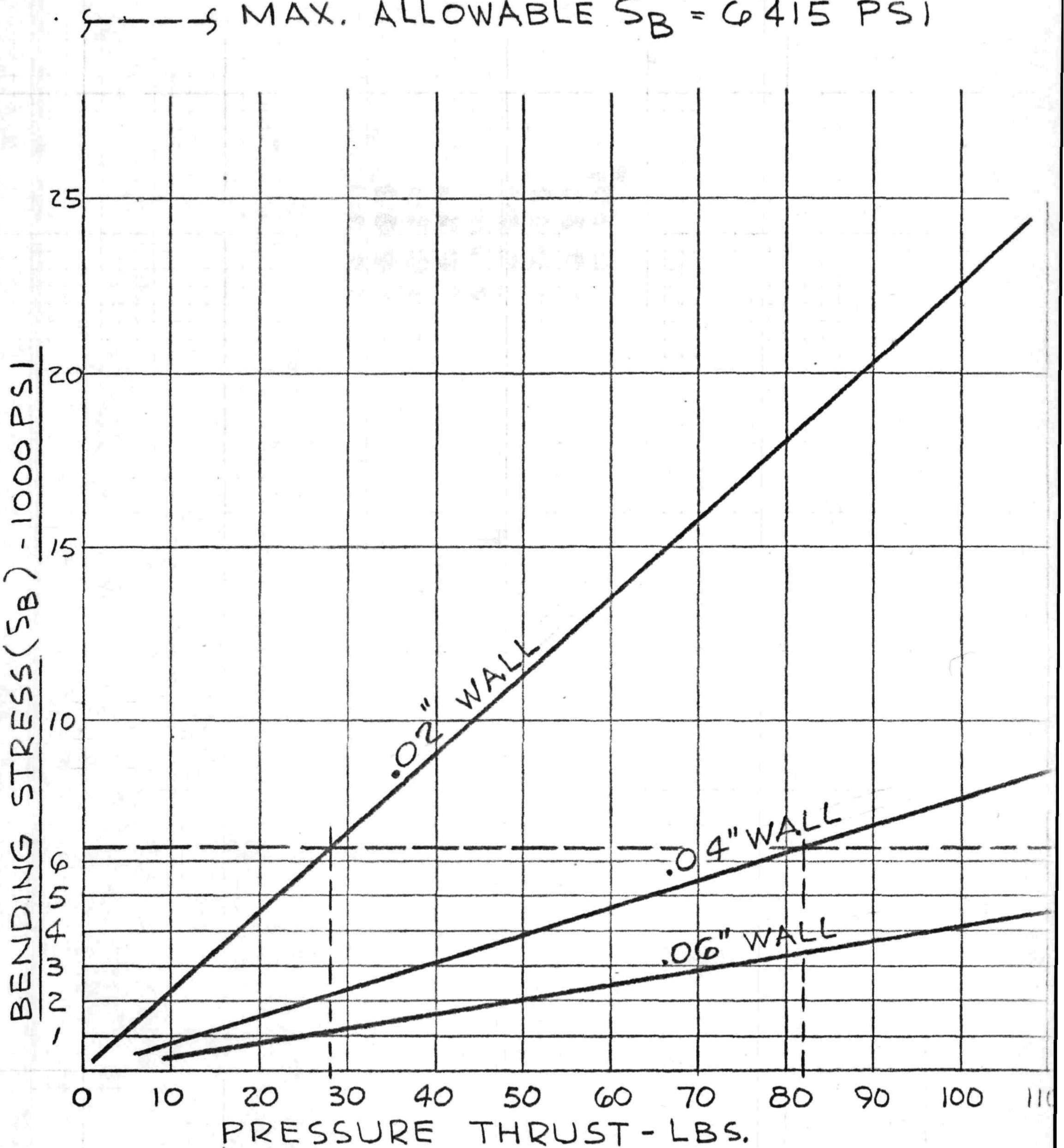
For a four-inch elbow bend radius, the maximum pressure thrusts that 0.02-inch and 0.04-inch piping can sustain are 28 pounds and 82 pounds respectively. The 0.06-inch wall thickness was off the graph (GRAPH 1). For a five-inch bend radius, the maximum pressure thrusts for 0.02 inches and 0.04 inches were 33 pounds and 102 pounds, respectively (GRAPH 2).



EFFECT OF XPAN. JT. PRESSURE THRUST ON BENDING STRESSES (S_B) AT PIPING ELBOWS

(SEE GRAPHS 1 & 2)

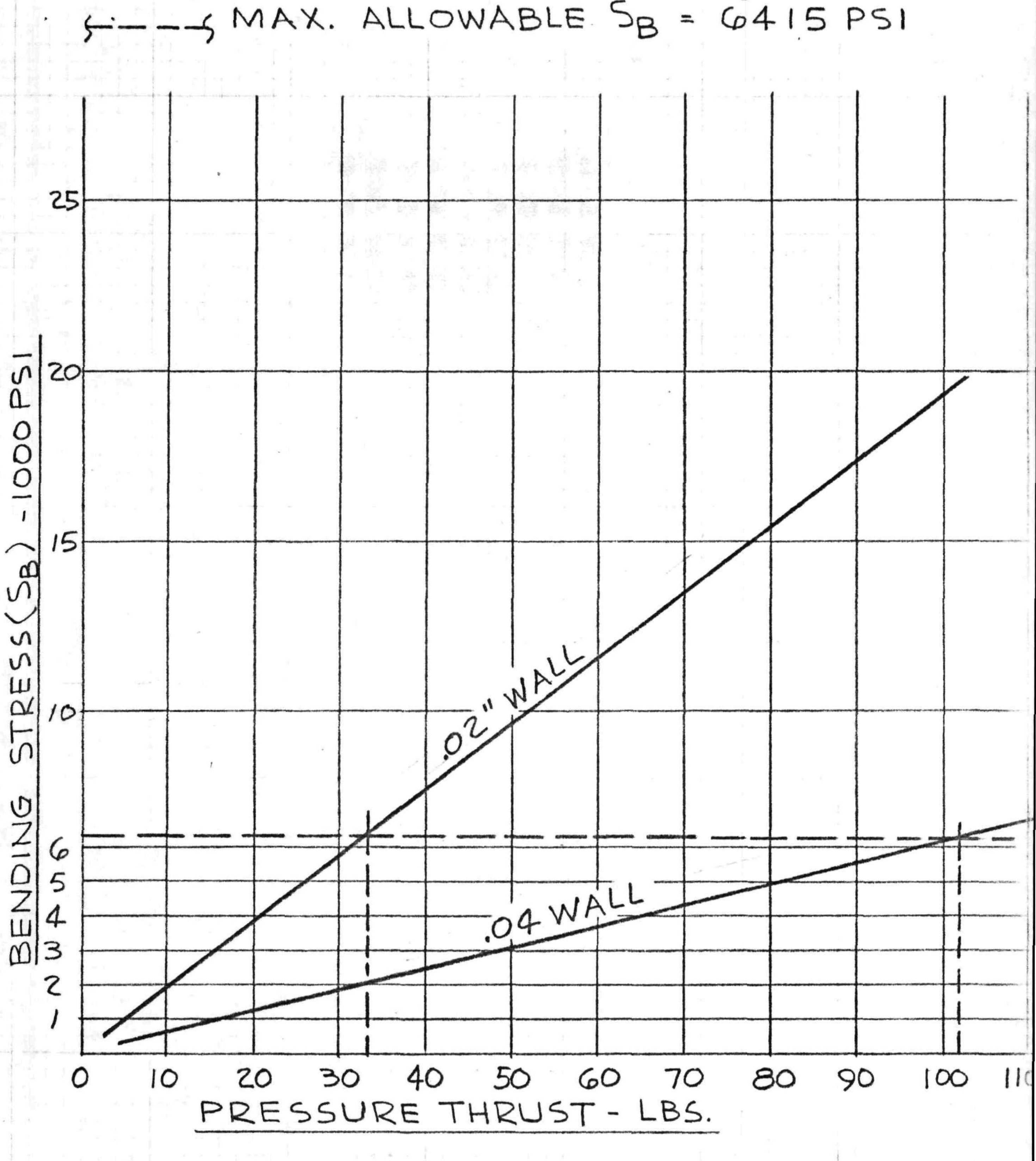
FIGURE 41



GRAPH 1

EXPANSION JT. PRESSURE THRUST (LB) VS. BENDING STRESS DUE TO SUSTAINED LOADS (S_B - 1000 PSI) AT ELBOW OF MAX. STRESS & 4" BEND RADIUS

REACTOR OUTLET TO XPAN. JT.



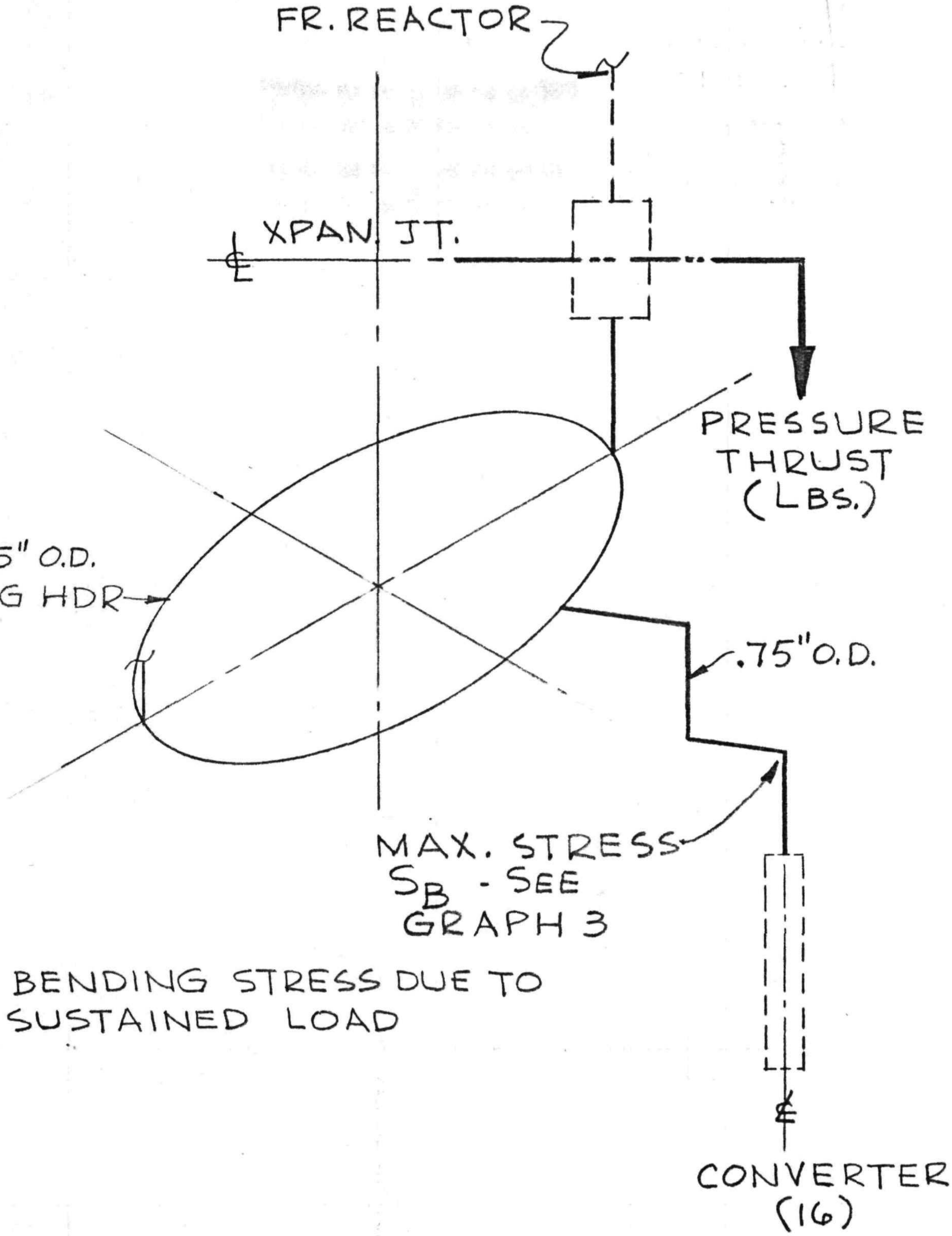
GRAPH 2

EXPANSION JT. PRESSURE THRUST (LB) VS. BENDING STRESS DUE TO SUSTAINED LOADS (S_B - 1000 PSI) AT ELBOW OF MAX. STRESS & 5" BEND RADIUS

REACTOR OUTLET TO EXPAN. JT.

2. Expansion Joint to T/E Converter Inlets - Figure 42

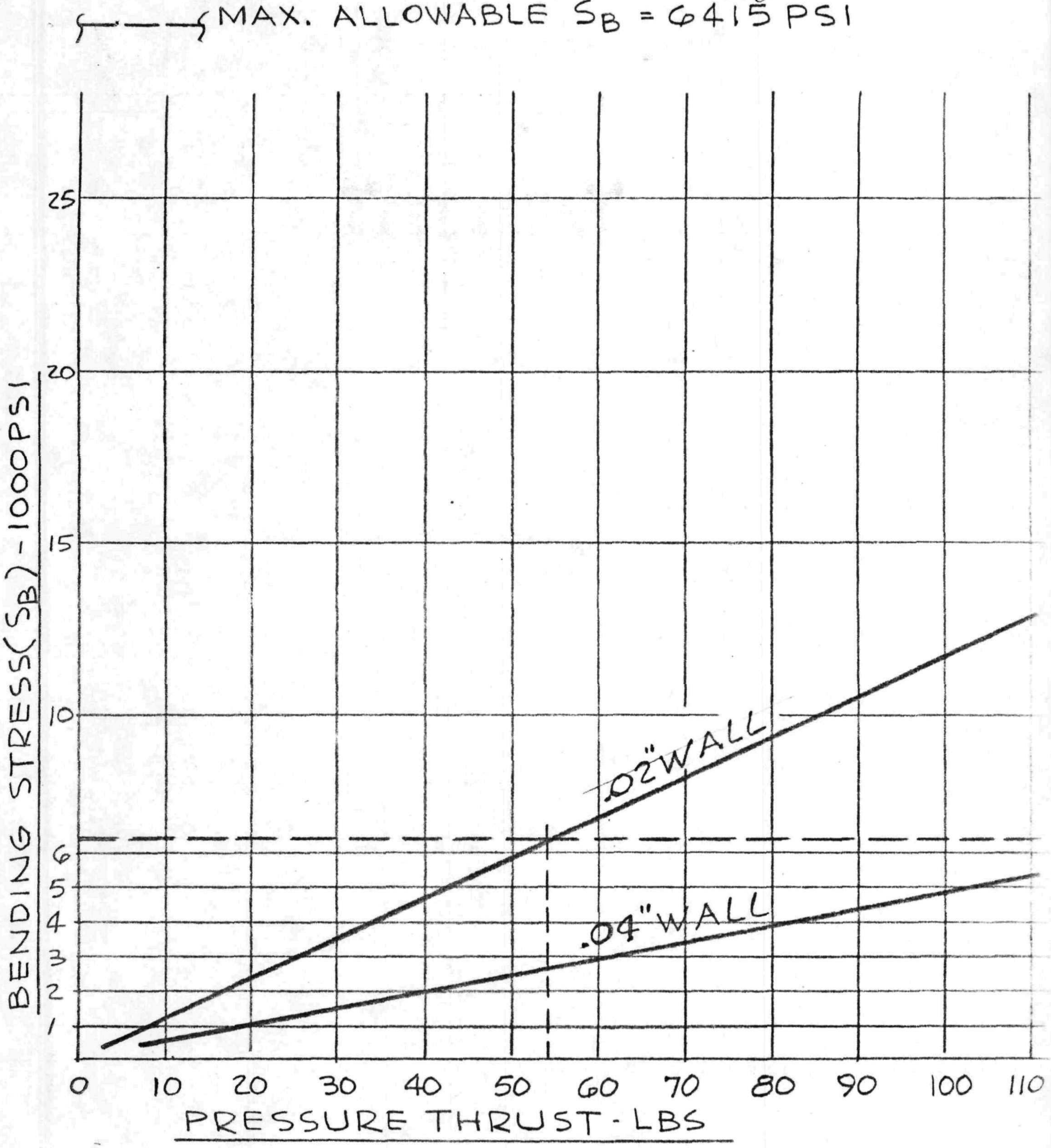
The maximum pressure thrust for a 0.02-inch wall was determined to be 54 pounds. The point of intersection of the 0.04-inch wall pipe and the S_B line was approximately 120 pounds (GRAPH 3).



EFFECT OF XPAN. JT. PRESSURE THRUST ON BENDING STRESSES (S_B) AT PIPING ELBOWS

FIGURE 42

SEE GRAPH 3



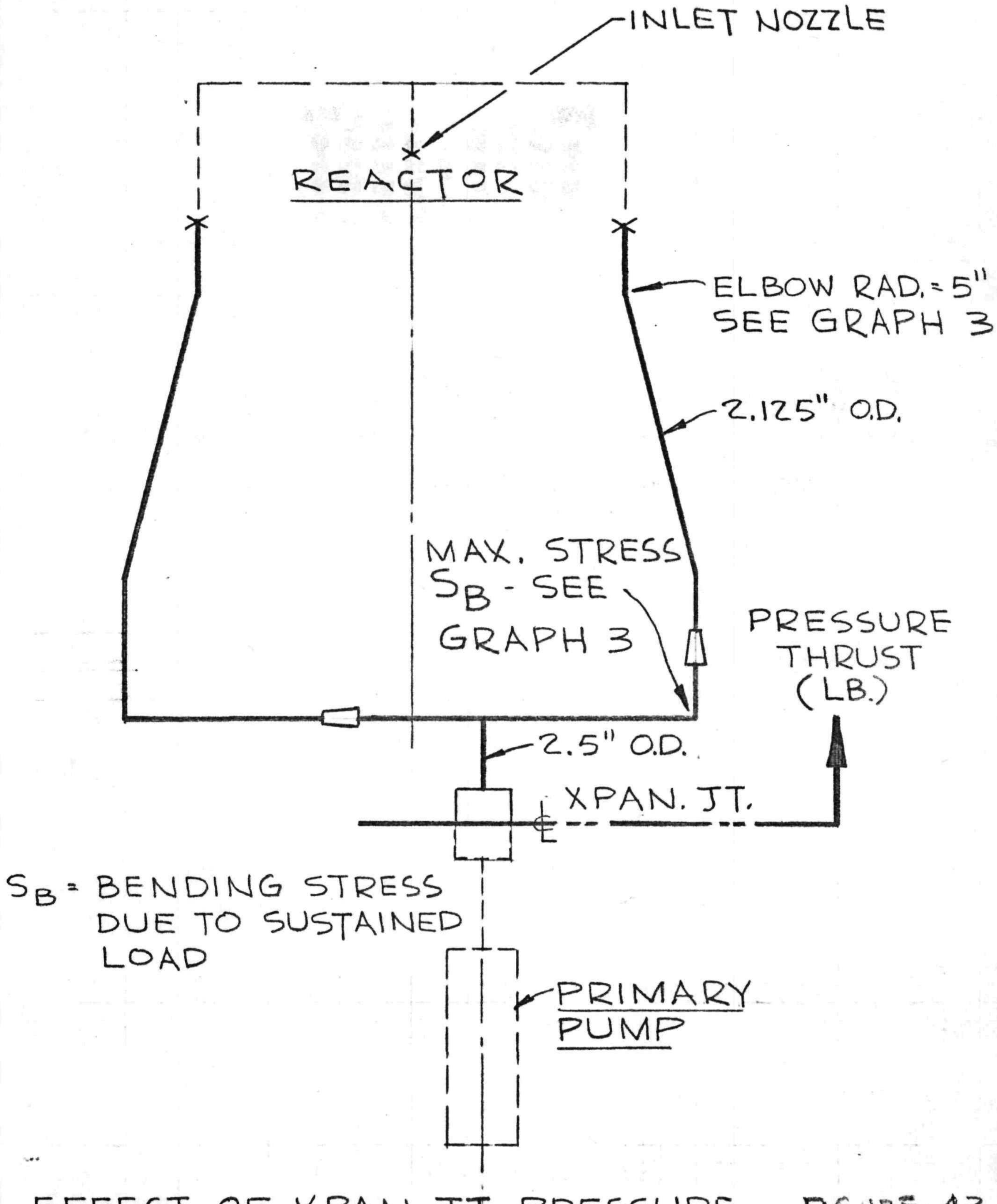
GRAPH 3

EXPANSION JT. PRESSURE THRUST (LB) VS. BENDING STRESS DUE TO SUSTAINED LOADS (S_B - 1000 PSI) AT ELBOW OF MAX. STRESS

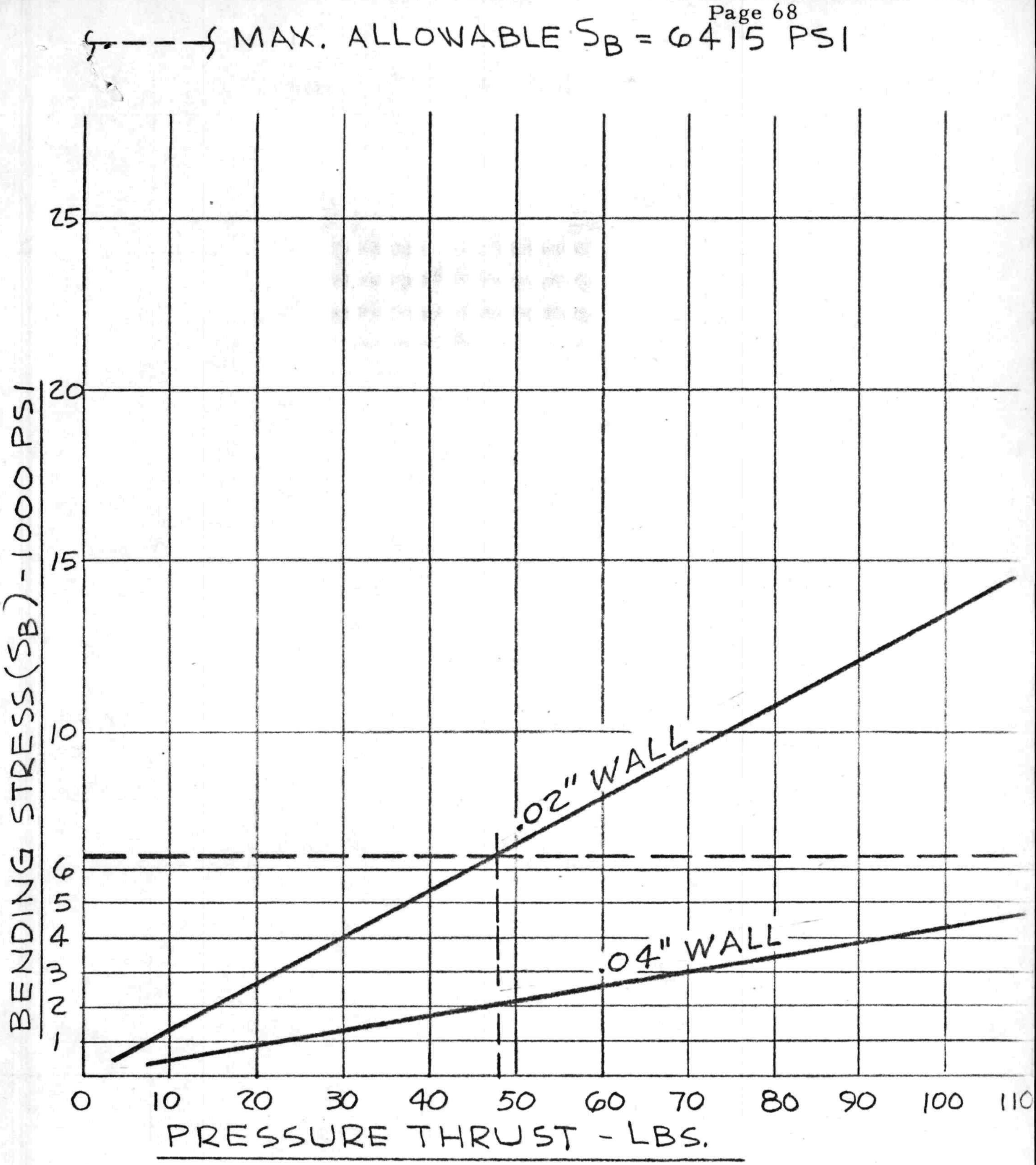
XPAN. JT. TO T/E CONVERTER INLETS

3. Expansion Joint to Reactor - Figure 43

For a five-inch bend radius, the maximum pressure thrust for a 0.02-inch wall was determined to be 48 pounds. The point of intersection of the 0.04-inch wall line and the S_B line was off the graph (GRAPH 4) but the maximum pressure thrust is approximately 140 pounds.



EFFECT OF XPAN. JT. PRESSURE THRUST ON BENDING STRESSES (S_B) AT PIPING ELBOWS OF 5" RADIUS (SEE GRAPH 4) FIGURE 43



GRAPH 4

EXPANSION JT. PRESSURE THRUST (LB) VS. BENDING STRESS DUE TO SUSTAINED LOADS (S_B - 1000 PSI) AT ELBOW OF MAX. STRESS & 5" BEND RADIUS

PRIM. PUMP DISCH. TO REACTOR