T O P I C A L   R E P O R T

BC-TOP-1

Revision 1

CONTAINMENT BUILDING
LINER PLATE DESIGN REPORT

PREPARED BY:

T. E. Johnson

B. W. Wedellsborg

MASTER

APPROVED BY:

L. G. Hinkelman

L. G. Hinkelman

H. W. Wahl

H. W. Wahl

DISCLAIMER

This report was prepared as an account of work sponsored by an agency of the United States Government. Neither the United States Government nor any agency Thereof, nor any of their employees, makes any warranty, express or implied, or assumes any legal liability or responsibility for the accuracy, completeness, or usefulness of any information, apparatus, product, or process disclosed, or represents that its use would not infringe privately owned rights. Reference herein to any specific commercial product, process, or service by trade name, trademark, manufacturer, or otherwise does not necessarily constitute or imply its endorsement, recommendation, or favoring by the United States Government or any agency thereof. The views and opinions of authors expressed herein do not necessarily state or reflect those of the United States Government or any agency thereof.

DISCLAIMER

Portions of this document may be illegible in electronic image products. Images are produced from the best available original document.

INTRODUCTION

This topical report provides design calculations and analytical techniques for the following containment liner plate system components:

- Part I   Liner Plate and Anchorage System
- Part II   Main Steam Pipe Liner Plate Penetration
- Part III   Polar Crane Bracket

The design criteria for these items are given in BC-TOP-5.

PART I

LINER PLATE AND ANCHORAGE SYSTEM

TABLE OF CONTENTS

Section

- 1.0 Summary
- 2.0 Description of Problem
  - 2.1 General
  - 2.2 Liner Plate Details
    - 2.2.1 Cylinder
    - 2.2.2 Dome
    - 2.2.3 Floor
    - 2.2.4 Junctions
  - 2.3 Problem Areas
- 3.0 Factors Affecting the Liner Plate and Anchorage System
  - 3.1 Initial Inward Curvature of the Liner Plate
  - 3.2 Variation of Anchor Spacing
  - 3.3 Variation in Plate Thickness
  - 3.4 Variation in Liner Yield Stress
  - 3.5 Variation of Poisson's Ratio and Modulus of Elasticity of the Liner Plate
  - 3.6 Variation of Concrete Modulus of Elasticity and Anchor Stiffness
  - 3.7 Cracking or Crushing of Concrete in the Local Liner Plate Anchorage Zone
  - 3.8 The Tolerance of the Liner Plate and Anchorage System
- 4.0 Loads to be Considered in the Liner Plate Design and the Strains Resulting From These Loads
  - 4.1 Individual Loads and Corresponding Liner Strain
    - 4.1.1 Creep and Shrinkage
    - 4.1.2 Prestress Combined with Concrete Creep and Shrinkage

TABLE OF CONTENTS
(Continued)

	4.1.3	Deadload
	4.1.4	Earthquakes, Wind, Tornadoes, and Hydrostatic Loads
	4.1.5	Vacuum Loads
	4.1.6	Pressure
	4.1.7	Temperature
4.2		Load Combinations
5.0		Detailed Analysis
5.1		General - Tangential Shear Load and Displacement of the Liner Plate Anchors
	5.1.1	Effects of All Anchors and Panels on Anchor 1 and Panel A
	5.1.2	General Description of Necessary Information and Analysis Techniques
	5.1.3	Modification of Test Results for Use in the Analysis
	5.1.4	Effects of Internal and External Pressure on the Liner Plate Anchors
	5.1.5	Detailed Analysis of Anchorage System
5.2		Effects of Moment on the Anchor
5.3		Effects of Radial Force on the Anchor
5.4		Effects of Longitudinal Loads on the Anchor
5.5		Longitudinal Buckling of the Liner Plate and Anchorage System
Appendix A		Bent Plate Tests
Appendix B		Liner Plate Anchorage Tests

1.0 Summary

The report gives a thorough description of the liner plate together with figures illustrating all the applicable details. Problem areas and factors affecting the liner plate and anchorage system design are covered in great detail. The loads considered in the liner plate design are discussed and the approximate strains in the liner plate resulting from these loads are stated.

The occurrence of the loads and conditions imposed upon the liner plate in the analysis are highly improbable. The liner plate was subjected to the effects of prestress, concrete creep and shrinkage, deadload, maximum hypothetical earthquake, accident thermal gradients, and accident pressure. The accumulative total strain in the longitudinal direction was $-1,616\mu$ in/in, and $-1,788\mu$ in/in in the hoop direction. In order to have these strains convert to maximum stresses, it was assumed that the liner plate would have a uniaxial yield stress of at least 53.7 ksi. This resulted in a hoop stress of -74.5 ksi and a longitudinal stress of -71 ksi. The analysis then assumed that a panel with initial inward curvature would be adjacent to a panel with initial outward curvature. The panel with outward curvature was assumed to be 16% thicker than the panel with inward curvature. All of the preceding conditions lead to a very large and highly improbable membrane load of 21.6 kips/in. The form of the solution was to remove a 1" strip of the liner plate in the hoop direction. At this point, the benefits of biaxial stiffening were entirely neglected. The panel with inward curvature was assumed to have a maximum inward deviation of $1/8$ ". The solution assumed that the concrete shell moves radially inward so that the membrane force of 21.6 kips/in. would exist. Since the panel with inward curvature acts primarily in bending and the panel with outward curvature acts in membrane compression, the anchor between these two panels will tend to displace toward the panel with inward curvature due to the difference in stiffness of the two panels. The movement is resisted by the panel with inward curvature together with the anchor embedded in concrete. Also, as the anchor displaces, the membrane load will tend to reduce. As this happens, other anchors tended to become unbalanced with respect to force and a method was used which considers the effects of all anchors. In solving the problem, test results were used which gave the load vs. displacement curves of both the concrete anchor and the panel with inward curvature, or bent plate. The data used for the bent plate did not consider any benefit from biaxial stiffening which will result in the real structure. The data used for the anchor did not consider the benefits gained by having concrete with a higher modulus of elasticity than that of the tests.

To simplify the solution, an imaginary force was calculated which considered the effects of all anchors acting upon the anchor under consideration so that only a single anchor needed to be analyzed. At this point, a membrane load was calculated to simulate the effects of internal pressure which tend to reduce the load on the anchor. This value was then subtracted from the previous imaginary force. Referring to Case II of Section 5.0, the liner plate anchorage system was then subjected to a self-relieving type of load with an initial force of 26.37 kips/in. The anchorage system was then allowed to displace tangentially until equilibrium was reached. Due to the fact that both the anchorage system and the bent plate went over yield, a plastic solution was used which considered the non-linear force vs. displacement relationship of both the bent plate and the anchor. After reaching equilibrium, it was found that the anchor had displaced .0318 inches and had developed a force of 4.41 kips/in. The energy used up in the anchor in obtaining equilibrium was calculated to be .1039 kips-in/in. The total energy available in the anchor was defined as .541 kips-in/in. This value is the lowest total energy observed in the testing of the anchorage system. The safety factor pertaining to the effects of tangential loads was the total energy available divided by the energy used up in obtaining equilibrium. In the previous case, the safety factor is 5.2. Since the force on the anchor is 4.4 kips/in, and the embedment length is 3", the concrete will have an average stress of 1470 psi; but in reality, as the anchor starts to displace, the stresses on the inner face of the concrete shell will increase rapidly until yield is reached, and the stresses further away from the liner plate will increase until a balanced condition is obtained. Due to the fact that the anchor is required to displace .0318 in. as predicted by the analysis, it is expected that there will be some local crushing of the concrete within the first 1/2 inch of the concrete. In the report other cases have been analyzed with various concrete properties.

As the panel with inward curvature deforms inward due to anchor movement, the panel will be subjected to flexure, and moments will develop at the anchor attachment points. The flexural stress or strain from these moments is also self-relieving since the panel will tend to have limited deformation. As an upper limit, a plastic hinge will form. The weld will then be subjected to both tangential shear and moment. The amount of rotation under accident conditions is not of the magnitude of the rotation found during testing. During the tests, summarized in Appendix B, the anchor weld proved its ability to withstand both pure shear strain in conjunction with rotational strain due to bending.

The anchor to liner plate attachment weld will also be subjected to strains in the longitudinal direction. These longitudinal strains may approach yield, but relative to the strains occurring in the tangential direction, they are of a very small magnitude and when combined with all other strains, this will only produce a maximum principal shear strain slightly greater than the tangential shear strain.

The anchorage system may at times be subjected to axial pullout loads. These pullout loads will have a negligible effect on the weld. The area of concern is the anchorage into the concrete. The tensile load will be resisted by bond on one side of the anchor and shear on the other, but even without bond and shear, the pullout would have to overcome a large friction force imposed by shear load. Since the expected maximum pullout force is approximately 310 lbs/in., radial pullout should not be a problem due to the 1500 lbs/in. capacity of the system. The possibility of longitudinal buckling of the liner plate and anchorage system was considered in the analysis and it was found that in the worst condition, a force of 560 lbs/in. was needed to prevent longitudinal buckling. Since the system should have a capacity of 1,500 lbs/in., longitudinal buckling is not considered to be a problem.

The conditions and resulting loads and displacements mentioned previously, will only occur at very isolated locations in the liner plate, if at all due to the improbability of having a simultaneous occurrence of all imposed conditions. The forces and displacements found during operation are very small relative to those occurring during an accident condition. Since the accident condition is a one-time occurrence, and the nature of the loads imposed upon the liner plate are self-limiting, it is only reasonable to allow the occurrence of large strains in the anchorage system relative to yield.

In conclusion, the report verifies through the use of analysis and test results that a failure of the liner plate anchorage system will not occur under an accident condition in conjunction with an earthquake even when all other worst conditions are assumed to occur simultaneously.

2.0 Description of Problem

2.1 General

Prestressed concrete and reinforced concrete reactor buildings which house Pressurized Water Reactors presently use carbon steel liner plates. The liner plate functions as a gas-barrier to prevent uncontrolled release of fission products from the reactor building, during operation and also in the unlikely event of an accident, which would release large amounts of fission products into the reactor building. In the event of such an accident that is considered credible, the temperature inside the reactor building is predicted to rise to approximately 280° with a pressure of about 55 psig. At this time, the liner plate is called upon to limit leakage to approximately .1 to .2% per 24 hours, of the contained weight of gas so that fission product leakage is also limited. Due to the required high leak-tight integrity of the liner plate, specific care must be given to its design, fabrication, and erection. In general, the following report will discuss the liner plate details; factors affecting the liner plate such as load conditions; detailed theoretical analyses and sample calculations; applicable test results; and the results of parametric studies.

2.2 Liner Plate Details

(See Figure 1 for details of the liner plate)

2.2.1 Cylinder

The cylinder is fabricated from 1/4" ASTM A-442 plate together with ASTM A-36 longitudinal stiffener angles welded to the liner and also horizontal ASTM A-36 channels, angles, and flat bars attached to the liner plate and the longitudinal angles. The rolled structural shapes and flat bars both stiffen the liner plate during erection and concrete placement and anchor the liner plate to the hardened concrete.

2.2.2 Dome

The dome is similar to the cylinder except it only has angle stiffeners which also act as anchors; the attached angles are oriented approximately in the hoop direction.

2.2.3 Floor

The floor is composed of 1/4" plate welded to embedded beams. The floor is covered with an

FIG.-1

18" concrete slab with a leak-chase system over all liner plate seams which are composed of 1/4" butt welds.

2.2.4 Junctions

At the junctions where the cylinder intersects the dome and base slab, horizontal channels or angles are attached as anchors near the location of the maximum changes in meridional curvature.

2.3 Problem Areas

A section of the cylinder, after liner plate erection and the placing of concrete, is shown in Figure 2. The majority of the loads imposed upon the liner (which will be discussed in Section 4) can be thought of as resulting from an inward radial movement and shortening of the concrete shell relative to the liner plate. The relative movement causes compressive membrane loads on the liner plate. If a panel of the liner plate should have an inward curvature resulting from fabrication and construction, such as illustrated in Figure 2, then this panel will deform inward since it has low stiffness relative to a panel with outward curvature, and the anchorage system will be subjected primarily to a shear load. The analysis of this situation will be covered in Section 5. If the liner plate had and maintained a perfectly straight or curved geometrical shape, the liner anchors would never be required to withstand shear loads parallel to liner plate surface. The anchor will also be subjected to a moment, radial load and longitudinal strain; these items will also be discussed in Section 5.

The basic unit of the cylindrical liner plate is composed of stiffened panels whose length to width ratio is 8 to 1. Figure 3 shows a possible deformation pattern for such a panel of liner plate. Due to that deformation pattern, the stiffening angles will be subjected to some torsion. The torsion will, however, be relatively minor since it is a function of the slope of the curve shown as $\Delta X / \Delta Y$. Also, the analysis to be done for Figure 2 is based on removing a strip of liner as shown in Figure 3 which is at the most critical location, and the solution involving the strip will not consider the resistance given by adjacent strips. This condition is much more severe on the anchorage system than any secondary torsional effects in the real system, and therefore, this problem will not be considered further.

FIG. - 2

FIG. - 3

FIG. - 4

The situation shown in Figure 4, which is a longitudinal buckle of the cylinder, will be discussed further in Section 5. The discontinuities shown in Figure 1 are in equilibrium when the liner plate is in compression and even local yielding of the concrete at the discontinuity cannot impair the integrity of the liner plate. If the concrete would be in tension at one of the discontinuity points, the liner plate integrity would still not be in jeopardy because, due to the amount of steel reinforcing used at discontinuities, cracking is widely distributed and any pulling away of the liner plate would only lead to flexural liner strains of low magnitude. As shown in Figure 1, there are anchors at these discontinuity points. The main function of the anchors is to stiffen the liner plate for erection and not to anchor the liner plate, even though they will serve in this capacity. Since the dome, which is shown in Figure 1, is composed of flat stiffened support panels supported by trusses for erection purposes, the situation is not nearly as severe as that shown in Figure 3, since all panels should have less relative variation in curvature than that of the cylinder, and therefore the analysis given for Figure 2 will also cover the dome.

3.0 Factors Affecting the Liner Plate and Anchorage System

3.1 Initial Inward Curvature of the Liner Plate

The fact that the liner plate will at some locations have initial inward curvature adjacent to panels that are flat or have outward curvature, is one of the most important items to consider when designing a liner plate anchor. The inward curvature in one panel enables the liner plate to relieve membrane loads leading to an unbalanced load condition at the anchor point and inducing loads into the anchor system. The larger the initial inward displacement, the greater is the ability of the panel to displace inward leading to greater loads imposed upon the anchor system. Due to this fact, the amount of inward curvature is limited to not more than 1/8" during fabrication and erection of the liner plate. Inward curvature of 1/8" is very likely to occur in some liner plate panels since the anchors are relatively close together and the plate is almost flat between the anchor points. As an example, if a plate with a radius of curvature of 58' has anchors spaced at 15", then the theoretical arc-to-chord distance will be only .04", and minor external applied fabrication and erection loads will tend to snap the panel through, leading to inward curvature.

3.2 Variation of Anchor Spacing

A variation in the anchor spacing will have the following effect on the liner plate anchorage system. If the anchors are further apart than the specified dimension, then a panel between anchors with inward curvature will have less stiffness than a panel with the specified dimension. This condition will lead to more shear load on the anchor. Due to the relatively simple task of fabricating anchors to the specified spacing, any small variation in spacing is not considered to have any appreciable effect on the anchorage system.

3.3 Variation in Plate Thickness

Due to standard rolling tolerances, 1/4" plate will have a thickness variation within the limits of +16% and -4%. The only effect of having a plate which is 4% under the theoretical thickness is that the plate would have a lower stiffness than the theoretical plate and there would be a slight increase in loading on the anchor. A plate which is thicker than the theoretical plate is advantageous as long as the excess thickness is constant throughout a large area, because a panel with inward curvature would be stiffer, and therefore, the anchor would be subjected to a decrease in load. In general, the smaller the ratio

of anchor spacing to thickness, the lower the loads in the anchorage system. It must be pointed out here that very low anchor spacing to plate thickness ratios tend to be unnecessary and quite uneconomical. The most undesirable condition is when the panel with inward curvature is thinner than the adjacent panels with outward curvature. The membrane loads imposed upon the panel with inward curvature are for this case greater, and the anchorage loads increase. In the analysis, a panel with outward curvature which is +16% over the nominal thickness will be considered adjacent to a plate with inward curvature of nominal thickness. The preceding condition is highly improbable and therefore, it is not necessary to consider the case of a plate which is -4% under the nominal thickness.

3.4 Variation in Liner Yield Stress

The liner plate, in general, is subjected to a strain input, as shown in Figure 2, and the stress of the liner plate will increase linearly as a function of the concrete radial displacement only until the yield of the steel material is reached. After this point, strain will increase but the stress in the liner plate will remain essentially constant since the strains will exceed yield but will by no means approach ultimate. Therefore, with a fixed strain input, the upper bound for the state of stress in the liner plate will be the yield stress of the material. As an example, consider the strain in the liner plate to be .0015 in/in, then if the material yield stress is 30 ksi, the stress level will not exceed 30 ksi. If the material would have a yield stress of 60 ksi, then the stress level would not exceed 45 ksi since to reach 60 ksi, a strain of .002 in/in. would be required. Liner plates are just the opposite of normal structures; a low value of yield is advantageous since the low yield will limit the stress and force on the liner plate and the anchors. This phenomenon may be easily visualized by considering a liner plate made of rubber, since obviously this system will result in very small loads into the anchorage system.

3.5 Variation of Poisson's Ratio and Modulus of Elasticity of the Liner Plate

Relative to other items which are subjected to variation, Poisson's ratio and Young's modulus of the liner plate will have very little significance on the performance of the liner plate and its anchorage system. As an example, approximately 52% during operation and 88% during the accident, of the load on the anchorage system is from the effects of thermal-gradients through the wall of the

reactor building. Thermal strains are first calculated and then converted to thermal stress. In converting from strain to stress, the reciprocal of $(1-\nu^2)$ enters the expression and a variation of Poisson's ratio (ν) has very little effect upon the calculated stress value.

Variations in the modulus of elasticity of steel are not expected to have an appreciable effect upon the design of the liner plate, but the calculated value of thermal stress will vary with any change in modulus of elasticity and the amount of stiffness in a panel with inward curvature will also be affected by any variation in the modulus of elasticity of the liner plate. Due to the fact that only minor variations in the modulus of elasticity of steel are expected, this item will not be considered further.

3.6 Variation of Concrete Modulus of Elasticity and Anchor Stiffness

Since the anchorage system relies on both displacement and load-carrying capability, both of these items must be considered in evaluating the capability of the anchorage system. Displacement and force may be considered simultaneously by working in terms of energy which is really the same as being able to predict the point at which the system will be in equilibrium on its force vs. displacement curve. Consider Figure 5, which is similar to a typical load vs. displacement curve for an angle, channel, etc., embedded in concrete which resists shear forces parallel to the plate surface.

FIG.-5

The total energy available in the anchorage system can be defined as the entire area (ABCD) under the curve. When subjected to a particular load, the point (B) on the curve at which the liner plate anchorage system is in equilibrium may be predicted. The area under the curve (ABE) up to this point may be considered the energy used up in obtaining equilibrium.

Dividing the total area (ABCD) by the equilibrium area (ABE) will result in a safety factor based on the ultimate energy capacity of the system. A variation in the modulus of elasticity of the concrete will affect the anchorage system as follows. When working in the linear range (AB) of the curve, the slope of the curve (which is also the spring constant of the system) will be a function of the modulus of elasticity of the concrete. The force in the anchorage system is equal to the displacement times the spring constant. The energy in the linear range of the curve is defined as one half (displacement) x (force), and substituting it may be noted that the energy equals one half (spring constant) x (displacement.)² As the concrete modulus reduces, the spring constant reduces, but displacement increases. And since the energy involves the displacement squared, this results in more energy being required for equilibrium for low modulus of elasticity concrete.

3.7 Cracking or Crushing of Concrete in the Local Liner Plate Anchorage Zone

A liner plate anchorage system must have sufficient ductility and force carrying capability in order to accommodate the imposed loads. In general, embedded shear-carrying devices used in conjunction with concrete, exhibit the ability to deform when subjected to loads. When a shear-carrying device is loaded, the stress distribution exhibits a very high peak at the concrete surface, but this situation is quickly eliminated due to local yielding with the stresses further from the surface increasing in magnitude until an equilibrium condition is reached. Since concrete does not have the ability to greatly yield and develop high strains, crushing of the concrete takes place. The crushing of the concrete is necessary when testing a system to ultimate. When an anchorage system has sufficient safety factors, the local yielding of concrete will be of a very minor nature and certainly not an area of concern since it is only the concrete's way of redistributing stress to obtain maximum load carrying capability.

3.8 The Tolerance of the Liner Plate and Anchorage System.

Typical tolerances used in liner plate fabrication and erection are as follows:

- 3.8.1 The deviation in 15" between liner stiffeners shall not exceed plus or minus $1/8$ " when referenced to the theoretical surface.
- 3.8.2 A template 15' in length shall not show deviations greater than plus or minus $3/4$ ". In the event that the template crosses a weld seam, the previous tolerance may be increased to plus or minus 1".
- 3.8.3 In the longitudinal direction of the cylinder, a straight edge shall not show deviations greater than plus or minus $1/2$ ".
- 3.8.4 The slope of the cylindrical wall shall not exceed $1/240$ within 10' panels.
- 3.8.5 The misalignment of seams shall not exceed 10% of the plate thickness but for $1/4$ " plate, the misalignment shall not exceed $1/16$ ".
- 3.8.6 The location of liner plate anchors relative to each other shall not deviate from the theoretical dimension by more than plus or minus $1/4$ ".

4.0 Loads To Be Considered In The Liner Plate Design And The Strains Resulting From These Loads

4.1 Individual Loads and Corresponding Liner Strain

4.1.1 Creep and Shrinkage

As the concrete dries out, it will tend to shrink, which, in general, will result in compressive strains in the liner plate. The expected value of shrinkage strain before prestressing is approximately -100μ in/in. The total shrinkage after prestressing will be combined with prestressing and given in a later section. No appreciable concrete creep is expected before prestressing and concrete creep will be combined directly with prestress in a later section.

4.1.2 Prestress Combined with Concrete Creep and Shrinkage

The most critical situation is near end of life when the concrete has maximum creep and shrinkage. At this time, the dome liner will experience strains in the hoop and meridional direction of approximately -200μ in/in. At the mid-height of the cylinder, the hoop strains will be approximately -334μ in/in., and the meridional strain will be -69μ in/in. The preceding low value is due to the fact that the prestressing force in the hoop direction is much greater than that of the meridional direction.

4.1.3 Deadload

The strain due to deadload at the mid-height of the cylinder is approximately -30μ in/in in the meridional direction. The peak strain is in the meridional direction at the base of the cylinder and is approximately -45μ in/in.

4.1.4 Earthquakes, Wind, Tornadoes, and Hydrostatic Loads

No appreciable hydrostatic loads are expected on the structure. In structures such as a reactor building housing Pressurized Water Reactors, the effects of high winds and tornadoes are negligible compared to the effects of earthquakes due to the large mass and stiffness of the structure. The strains from the maximum hypothetical earthquake

are expected to vary from approximately 0 to +76u in/in from the top of the structure to the base in the meridional direction, whereas the strain in the hoop direction should vary from 0 to +23u in/in.

4.1.5 Vacuum Loads

The design vacuum pressure is 3psi. This pressure will generate a membrane hoop strain of -8.5u in/in and a membrane longitudinal strain of -3.0u in/in. A flexural strain of 180u in/in. in the liner plate will also exist between the anchors.

4.1.6 Pressure

The strain expected under accident conditions in the hoop and meridional direction in the dome should be approximately +100u in/in. The strains at the mid-height of the cylinder should be approximately +50u in/in in the longitudinal direction and +147u in/in in the hoop direction.

4.1.7 Temperature

During the operating condition, the thermal strains in the cylinder and dome are expected to be approximately -325u in/in in both the hoop and meridional directions. This is based on a thermal gradient of 100°F on the inside face and 0°F on the outside face.

During the accident condition, the thermal strains are expected to be approximately -1500u in/in in the cylinder and the dome in both the hoop and meridional directions. These strains are based on the thermal gradient going from 280°F on the inside face to 0°F on the outside face with a very rapid drop-off in temperature within the first few inches of the inside face.

4.2 Load Combinations

The load combinations are shown in detail in the Detailed Analysis section. The normal operating condition is analyzed together with an accident in conjunction with the maximum hypothetical earthquake.

A protracted shutdown and a cold-weather start-up may yield conditions slightly more severe on the liner plate than the operating condition, but these conditions will not approach the accident condition with respect to harmful effects on the liner plate.

5.0 Detailed Analysis

5.1 General - Tangential Shear Load & Displacement of the Liner Plate Anchors

All of the major strains imposed on the liner plate such as concrete shrinkage and creep, prestressing and thermal effects may be considered as an inward movement of the shell as shown in Figure 2.

As the shell moves inward, all the panels will be subjected to membrane compression. But since Panel A has initial inward curvature, it will displace an additional amount (W) inward due to flexure. Since Panel A is not as stiff as the other panels, a tangential movement will occur, thus loading the anchor. As the tangential movement occurs at Anchor 1, the load in Panel B will drop off until Anchor 1 is in equilibrium. If only Panel A and Panel B are considered, then at equilibrium of Anchor 1, Anchor 2 will not be in equilibrium. To bring Anchor 2 in equilibrium, additional displacement will occur at Anchor 1. The effects of all anchors must be considered when analyzing Anchor 1.

5.1.1 Effects of All Anchors and Panels on Anchor 1 and Panel A

First consider the equilibrium of Anchor 1 after being subjected to the loads from Panel B. Figure 6 shows the model of Anchor 1.

Description of Terms:

K_C : Spring constant 1st Anchor with concrete .
 K_C : Spring constant of other Anchors with concrete.

K_{BR} : Spring constant of Panel A or bent plate

K_{RR} : Imaginary spring constant simulating load drop-off or relaxation of Panel B with Anchor 2 held fixed.

δ_{II} : Displacement of Anchor 1 due to unbalance at Anchor 1.

N : Initial membrane force in Panel B

FIG. -6

For equilibrium of Anchor 1 then

$$\delta_{11} (K_C + K_{B.R.} + K_{R.R.}) = N \quad \text{And}$$

$$(1) \quad \delta_{11} = N / (K_C + K_{B.R.} + K_{R.R.}) \quad \text{And the force in the Anchor is}$$

$$(2) \quad F_{11} = \delta_{11} K_C$$

Now since Anchor 1 has moved there is an unbalance at Anchor 2 equal to $\delta_{11} K_{R.R.}$ and the following Figure 7 will be used to illustrate how this unbalance affects Anchor 1

FIG.- 7

In Figure 7 δ_{22} is the displacement of Anchor 2 due to an unbalance at Anchor 2 and δ_{12} is the displacement of Anchor 1 due to an unbalance at Anchor 2. Figure 8 shows Anchor 2 with a single spring used to represent Anchor 1.

FIG.- 8

In Figure 8

$$K_{T1} = \frac{(K_C + K_{B.R.})(K_{R.R.})}{(K_C + K_{B.R.} + K_{R.R.})}$$

For equilibrium of Anchor 2 Then

$$\delta_{22} (K_C + K_{T1} + K_{RE}) = \delta_{11} K_{RE} \text{ And}$$

$$(3) \quad \delta_{22} = \delta_{11} K_{RE} / (K_C + K_{T1} + K_{RE})$$

Displacement at Anchor 1 due to a movement at Anchor 2

$$(4) \quad \delta_{12} = \frac{\delta_{22} (K_{RE})}{(K_C + K_{BE} + K_{RE})} \quad \text{Substitute (3) into (4) and simplify}$$

$$(5) \quad \delta_{12} = \frac{\delta_{11} (K_{RE})^2}{[(K_C + K_{BE})(K_{RE}) + (K_C + K_{RE})(K_C + K_{BE} + K_{RE})]}$$

By combining equation (1) and (5) the total displacement at Anchor 1 due to movement at Anchor 1 and 2 may be obtained

$$(6) \quad \delta_{11} + \delta_{12} = \frac{N}{(K_C + K_{BE} + K_{RE})} \left[1 + \frac{(K_{RE})^2}{(K_C + K_{BE})(K_{RE}) + (K_C + K_{RE})(K_C + K_{BE} + K_{RE})} \right]$$

Now since Anchor 2 has moved, there is an unbalance at Anchor 3 equal to $\delta_{22} K_{RE}$ and Figure 9 may be used to find how this unbalance affects Anchor 1.

FIG. - 9

Using the same approach which was used previously, the expression for the displacement at Anchor 1 due to movement of Anchors 1, 2 and 3 is found to be

$$(7) \quad \delta_{11} + \delta_{12} + \delta_{13} = \frac{N}{(K_C + K_{BR} + K_{RR})} \left\{ 1 + \frac{K_{RR}^2}{(K_C + K_{BR})(K_{RR}) + (K_C + K_{RR})(K_C + K_{BR} + K_{RR})} \right\} \left[1 + \frac{K_{RR}^2}{(K_C + K_{BR})(K_{BR}) + (K_C + K_{RR})(K_C + K_{BR} + K_{RR})} \right] \left\{ \frac{1}{K_C + K_{BR} + K_{RR}} \left[\frac{K_{RR}[(K_C + K_{RR})(K_{RR}) + (K_C)(K_C + K_{BR} + K_{RR})]}{(K_C + K_{BR})(K_{RR}) + (K_C + K_{RR})(K_C + K_{BR} + K_{RR})} \right] \right\}$$

TERM "A"

Since Term "A" in equation 7 is always larger than 1.0 for the range of variables involved, it is conservative to assume that the equation for the displacement at Anchor 1, which considers the movement of an infinite number of anchors is as follows:

$$(8) \quad \delta = \sum_{n=1}^{n=\infty} \delta_{1n} = \frac{N}{(K_C + K_{BR} + K_{RR})} \left\{ 1 + \sum_{n=1}^{n=\infty} \left[\frac{K_{RR}^2}{(K_C + K_{BR})(K_{RR}) + (K_C + K_{RR})(K_C + K_{BR} + K_{RR})} \right]^n \right\}$$

A very convenient way to handle the problem and reduce it to a one anchor solution is as follows: Let N' be an imaginary load which simulates the effects of all anchors causing movement at Anchor 1

$$(9) \quad N' = N \left\{ 1 + \sum_{n=1}^{n=\infty} \left[\frac{(K_{RR})^2}{(K_C + K_{BR})(K_{RR}) + (K_C + K_{RR})(K_C + K_{BR} + K_{RR})} \right]^n \right\}$$

Then the single anchor solution and model are as shown in Figure 10

FIG.-10

$$(10) \quad \delta = \frac{N'}{(K_c + K_{BR} + K_{RR})}$$

Total displacement of Anchor 1 in the linear range

$$(11) \quad F_c = \delta K_c$$

Force in anchor

$$(12) \quad F_{BR} = \delta K_{BR}$$

Force in bent plate

5.1.2 General Description of Necessary Information and Analysis Techniques

In order to evaluate the load and displacement of the anchor shown in Figure 10, the following information is needed:

- (1) A load vs. displacement curve of the anchor embedded in concrete as shown in Figure 11
- (2) A load vs. displacement curve of the bent plate as shown in Figure 12

FIG. 11

FIG. 12

After obtaining the information shown in Figures 11 and 12 by testing or any other reliable method, the anchor may be analyzed.

Figure 12 shows all information needed for the solution.

FIG. - 13

When the required anchor displacement is low enough so that the load vs. displacement curves for the anchor and the bent plate are in the linear range then the following elastic solution may be used :

Elastic Solution

$$\delta = \frac{N'}{(K_C + K_{BR} + K_{RR})} \text{ and } F_C = \delta K_C \quad F_{BR} = \delta K_{BR}.$$

When the required anchor displacement is great enough so that the displacement is in the nonlinear range of one of the curves, then the following Plastic Solution must be used.

Plastic Solution

For ease in visualizing this solution an example problem will be used.

Consider the following anchor (as shown in Figure 14) with only the concrete resisting the imposed load. Since the load N' is self limiting, $K_{R\&}$ will simulate the load decrease as displacement occurs. Also included in Figure 14 is the information applicable to the anchor and relaxation characteristics.

FIG. - 14

Let $N' = 28 K/in$ and use an elastic solution to see if the anchor is in the plastic range:

$$\delta = \frac{28}{(200 + 500)} = .04" \quad \text{Since } \delta \text{ is greater}$$

than $.025"$, the solution must consider the plastic range. When a displacement of $.025"$ is reached, the unbalanced load (F) is as follows:

$$F = N' - \delta_e (K_c + K_{R\&}) = 28 - .025 (200 + 500) = 10.50 K/IN.$$

After a displacement of .025" is reached, then only plate relaxation is available to reach equilibrium and the additional displacement (δ') is:

$$\delta' = F / K_{REL} = 10.50 / 500 = .021" \quad \text{The total displacement is } \delta = \delta_e + \delta' = .025 + .021 = .046"$$

The energy used up in obtaining equilibrium in the anchor is:

$$E = 1/2 (5) (.025) + (5.0) (.046 - .025) = .1675 \text{ K/in.}$$

This energy is shown as the shaded area under Anchor Curve in Figure 14. The total energy in the anchor is defined as the total area under the Anchor Curve. The total energy is:

$$E_T = 1/2 (5) (.025) + (5.0) (.150 - .025) = .6875 \text{ K/in.}$$

The safety factor may be defined as:

$$S.F. = E_T / E = \frac{.6875}{.1675} = 4.1$$

5.1.3 Modification of Test Results For Use In The Analysis

The tests performed to determine the characteristics of the bent plate used a material which had a yield strength of 43 ksi. Since data will be needed for yield strengths both higher and lower than 43 ksi, the original data will be modified as shown below in Figure 15. The shifting technique used should be fully valid since with an initial eccentricity of 1/8" the plate is primarily in bending and the curve is never shifted high enough to approach the buckling load. See Appendix A for test results.

FIG. - 15

FIG. - 16

FIG. - 17

Figures 16 and 17 show the data which will be used for the properties of the bent plate.

The test performed to determine the anchor characteristics had a concrete modulus of elasticity of 2.67×10^6 psi at the time of testing. Since a higher modulus of elasticity is expected in the real structures, the anchor load vs. displacement curves will be modified as shown in Figure 18. It is assumed that the anchor spring constant changes linearly as a function of the change in modulus of elasticity. For test results see Appendix B.

FIG.- 18

Figures 19, 20 and 21 show the data which will be used for the properties of the anchors. It may be noted in Appendix B that the least value of total energy was .541 K/in/in. and this occurred with a maximum force of 5.97 K/in. To be conservative, the total energy will be assumed to be .541 K/in/in. and the maximum force will be limited to 5.00 K/in.

FIG. - 19

FIG. - 20

FIG. - 21

5.1.4 Effects of Internal and External Pressure on the Liner Plate Anchors

Internal pressure results due to a loss of coolant accident. This pressure will tend to reduce the shear load on the anchorage system as shown in Figure 22. Also, the pressure will provide restraint against any minimal pullout forces exerted on the anchorage system during the accident.

FIG. - 22

External pressure is exerted from a vacuum load which results from the difference in pressure on the inside and outside of the reactor building. The vacuum pressure acts in the opposite direction of that shown in Figure 22.

The following analysis will illustrate how the pressure may be converted to an equivalent axial load. This axial load will be very convenient when analyzing individual anchors. The applicable differential equation for loading conditions shown in Figure 22 is as follows:

$$(13) \quad \frac{EI d^4 w}{dx^4} + \frac{Nd^2 w}{dx^2} + \frac{Nd^2 \Delta}{dx^2} = -q$$

FIG.-23

Using the following relationships and Figure 23

$$W = W_m \sin^2 \frac{\pi}{a} x \quad \Delta = \Delta_m \sin^2 \frac{\pi}{a} x$$

An approximate solution for this problem is as follows:

(14)

$$W_m = \frac{\Delta_m N - q a^2 / 2 \pi^2}{\left(\frac{4 \pi^2 E I}{a^2} - N \right)}$$

The axial load equivalent to the pressure effect may be found by the following:

$$(15) \quad \Delta_m N'' = q a^2 / 2 \pi^2$$

$$N'' = \frac{q a^2}{2 \pi^2 \Delta_m}$$

5.1.5 Detailed Analysis of Anchorage System

As a review, a detailed outline of the method of analysis will be given below:

1. Combine the strains from all loads being considered and determine the two-dimensional state of strain. Using the largest of the strains, determine the uniaxial yield stress that must exist if the strains are to be converted to stress by use of Hooke's Law. Convert strains to stress and determine the load per inch. This portion is highly conservative

due to the fact that a strip of the shell is being analyzed, but the imposed load comes from a biaxial state of stress. Using a uniaxial yield to determine the bent plate characteristics is also conservative since it neglects the stiffening and increase in yield due to a biaxial state of stress.

2. Determine the load vs. displacement curves for the anchor and bent plate which are applicable to the problem based on the concrete modulus and steel yield strength.
3. Determine the spring constant for the plate relaxation based on $K_{RE} = AE/a$
4. Find the force N' which simulates the effects of all other anchor movements use Equation 9.
5. If internal or external pressure is acting on the liner plate, then the pressure load will be converted to an equivalent axial load (N'') by use of Equation 15 and combined with N' .
6. Do an Elastic Solution of the anchor to determine if either the anchor or bent plate is over yield. If either or both is over yield, then do a Plastic Solution.

The following cases will be analyzed for their affects on the Anchorage System.

- Case I - Most probable worst condition to which the anchor is subjected during operation with $E_c = 5.4 \times 10^6$ psi and $\sigma_y = 32$ ksi.
- Case II - Most probable worst condition to which the anchor is subject in the event of a loss of coolant accident with $E_c = 5.4 \times 10^6$ psi and $\sigma_y = 53.7$ ksi.
- Case III - Same as Case II but $E_c = 7.0 \times 10^6$ psi
- Case IV - Same as Case II but $E_c = 2.67 \times 10^6$ psi

Case III and IV are being analyzed to show how the variation of concrete modulus affects the anchorage system.

STRAIN TO STRESS CONVERSION

Case I

$$E = 30 \times 10^3 \text{ ksi} \quad \nu = .3$$

Operating Condition With Vacuum

$$\sigma_x = \frac{E}{(1-\nu^2)} (\epsilon_x + \nu \epsilon_y)$$

State of stress and strain in liner at mid-height

Summary of Strains

	Source	Hoop	Longitudinal
Hoop	1. Prestress w/ creep & shrinkage	-334u	-69u
-760u in/in.	2. Dead load	+10u	-30u
-30.0 ksi	3. Earthquake max. hypo.	±11u	±33u
	4. Thermal 100°F Inside 0°F Outside	-325u	-325u
	5. Initial shrinkage before prestress	-100u	-100u
	TOTAL	-760u	-491u

Miscellaneous Information:

1. Uniaxial yield strength of material is at least 32 ksi
2. Plates adjacent to the panel with initial curvature are 16% over standard thickness.

Plate thickness:

$$h = \frac{1}{4} (1.16) = 0.29 \text{ in.}$$

Maximum membrane stress

$$\sigma_x = \frac{E}{(1-\nu^2)} (\epsilon_x + \nu \epsilon_y) = \frac{30000}{(1-0.30^2)} (760 + 0.3(491)) 10^{-6} =$$

$$\sigma_x = 30.0 \text{ K/in}^2 < \sigma_y = 32.0 \text{ K/in}^2;$$

Maximum Membrane Force:

$$N = h(\sigma_x) = 0.29(30.0) = 8.72 \text{ K/in}$$

Assumed concrete modulus of elasticity $E = 5.4 \times 10^3$ ksi with an anchor spring constant in the linear range of $K_c = 270$ K/in/in. (See Figure 19).

The bent plate will have a spring constant of $K_{BR.} = 130$ K/in/in in the linear range (See Figure 16).

The plate when relaxing will have a spring constant of:

$$K_{REL.} = AE/a (1.16) = \frac{(.25)(30 \times 10^3)(1.16)}{15} = 580 \text{ K/in/in}$$

Find the force to put on the first anchor which simulates the effects of all other anchors on the first anchor:

$$N' = N \left\{ 1 + \sum_{n=1}^{n=\infty} \left[\frac{K_{REL.}^2}{(K_c + K_{BR.})(K_{REL.}) + (K_c + K_{REL.})(K_c + K_{BR.} + K_{REL.})} \right]^n \right\}$$

Using $N = 8.72 \text{ K/IN}$. $K_{REL} = 580 \text{ K/IN/IN}$ $K_C = 270 \text{ K/IN/IN}$. $K_{BRE} = 130 \text{ K/IN/IN}$.

$$N' = 8.72 \left\{ 1 + \sum_{n=1}^{n=\infty} \left[\frac{580^2}{(400)(580) + (850)(980)} \right]^n \right\} = 8.72 \left\{ 1 + \sum_{n=1}^{n=\infty} (.316)^n \right\} = 12.76 \text{ K/IN.}$$

Find the additional axial load which simulates a 3 psi vacuum:

$$N'' = \frac{q a^2}{2 \pi^2 \Delta_m} = \frac{(.003)(225)}{(2)(9.85)(.125)} = .281 \text{ K/IN.}$$

$$N_T = N'' + N' = 12.76 + .28 = 13.04 \text{ K/IN.}$$

Anchor Problem

Use Figure 16 and 19 for the bent plate and anchor properties.

Elastic Solution

$$\delta (K_C + K_{BRE} + K_{REL}) = N_T \quad \delta = \frac{N_T}{(K_C + K_{BRE} + K_{REL})} = \frac{13.04}{980} = .0133''$$

Since the yield of the bent plate is exceeded, a Plastic Solution must be used:

Anchor	Bent Plate	Relaxation	Load
$270 (\delta)$	$1.60 + 60 (\delta - .0123)$	$580 (\delta)$	$= 13.04$

Solving the above equation $\delta = .0134''$

Check equilibrium:

Anchor:	$F_C = 270 (.0134)$	$= 3.62 \text{ k/in}$
Bent Plate:	$F_{BRE} = 1.60 + 60 (.0134 - .0123)$	$= 1.67 \text{ k/in}$
Relaxation:	$F_{REL} = 580 (.0134)$	$= 7.78 \text{ k/in}$

$$\sum \text{Forces} = 13.07 \approx 13.04 \text{ Within slide rule accuracy.}$$

The energy used up in obtaining equilibrium in the anchor is:

$$E = 1/2 (3.62) (.0134) = .0242 \text{ K-in/in}$$

The safety factor is:

$$\text{S.F.} = E_T/E = \frac{.541}{.0242} = 22.4$$

Case II

Accident Condition

State of stress and
strain in liner
at mid-height

Summary of Strains

	Source	Hoop	Longitudinal
Hoop -1788u in/in.	1. Prestress with creep & shrinkage	- 334u	- 69u
-74.5 ksi	2. Dead load	+ 10u	- 30u
Longitudinal -1616 -in/in	3. Earthquake Max. hypo.	± 11u	± 33u
-71.0 ksi	4. Thermal 280°F Inside		
Misc. Information	0°F Outside	-1500u	-1500u
1. In order to get the above stresses, then a uniaxial yield stress of $\sigma_y = (1788) 30 = 53700$ psi	5. Intial shrinkage before prestress	- 100u	- 100u
must exist.	6. Accident pressure	+ 147u	+ 50u
2. Plates adjacent to the panel with initial curvature are 16% over thickness	TOTAL	-1788u	-1616u
3. Internal pressure is 55 psi Maximum membrane force $N = (74.5) (1/4) (1.16) = 21.6$ K/in.			

Assumed concrete modulus of elasticity $E = 5.4 \times 10^3$ ksi with an anchor spring constant in the linear range of $K_c = 270$ K/in/in. (See Figure 19)

The bent plate will have a spring constant of $K_{BP} = 130$ K/in/in. in the linear range (See Figure 17).

The plate when relaxing will have a spring constant of:
 $K_{RP} = AE/\Delta (1.16) = 580$ K/in/in.

Find the force to put on the first anchor which simulates the effects of all other anchors on the first anchor by using equation 9 together with: $N = 21.6$ K/in/in. $K_{RP} = 580$ K/in/in. $K_c = 270$ K/in/in.
 $K_{BP} = 130$ K/in/in.

$$N' = 21.6 (1.452) = 31.4 \text{ K/in.}$$

Find the force which must be subtracted from N' to simulate the effect of a 55 psi internal pressure:

$$N'' = \frac{q a^2}{2 \pi^2 \Delta_m} = \frac{(.055)(225)}{(2)(9.85)(125)} = 5.03 \text{ K/in.}$$

$$N_T = 31.4 - 5.03 = 26.37 \text{ K/in.}$$

Use Figures 17 and 19 for the bent plate and anchor properties -

Try an Elastic Solution:

$$\delta = N_T / (K_c + K_{BP} + K_{RP}) = 26.37 / 980 = .0270''$$

Since both the yield of bent plate and anchor are exceeded, a Plastic Solution must be used:

Plastic Solution

Assuming the displacement is less than .0321" (See Figure 17), then the following equilibrium equation may be written:

$$\underbrace{4.2 + 12.3 (\delta - .015)}_{\text{Anchor}} + \underbrace{3.0 + 60 (\delta - .0231)}_{\text{Bent Plate}} + \underbrace{580 (\delta)}_{\text{Relaxation}} = \underbrace{26.37}_{\text{Load}}$$

Solving the above equation $\delta = .0318''$

Check equilibrium:

$$\text{Anchor: } F_C = 4.2 + 12.3 (.0318 - .0150) = 4.41 \text{ K/in.}$$

$$\text{Bent Plate: } F_{BR} = 3.0 + 60 (.0318 - .0231) = 3.52 \text{ K/in.}$$

$$\text{Relaxation: } F_{RR} = 580 (.0318) = 18.50 \text{ K/in.}$$

$$\sum \text{ Forces} = 26.43 \approx 26.37 \text{ Within slide rule accuracy}$$

The energy used up in obtaining equilibrium in the anchor is:

$$E = 1/2 (4.2)(.015) + 4.2 (.0318 - .015) + (4.41 - 4.2) (.0318 - .015)/2$$

$$E = .1039 \text{ K-in/in}$$

The safety factor is:

$$\text{S.F.} = E_T/E = \frac{.541}{.1039} = 5.2$$

Case III

Accident Condition:

All data is the same as Case II except the concrete modulus is $E_c = 7 \times 10^6$ psi. The anchor spring constant is $K_C = 350$ K/in/in. (See Figure 20).

Find the force to put on the first anchor which simulates the effects of all other anchors on the first anchor by using equation 9 together with: $N = 21.6$ K/in, $K_{RR} = 580$ K/in/in., $K_C = 350$ K/in/in., $K_{BR} = 130$ K/in/in.

$$N' = 21.6 (1.363) = 29.5 \text{ K/in.}$$

The force which must be subtracted from N' to simulate the effect of a 55 psi internal pressure: $N'' = 5.03$ K/in.

$$N_T = 29.5 - 5.03 = 24.47 \text{ K/in.}$$

Based on Case II it is known that a Plastic Solution is necessary and the displacement is less than .0321"

Plastic Solution

$$\overbrace{4.2 + 11.75(\delta - .012)}^{\text{Anchor}} + \overbrace{3.0 + 60(\delta - .0231)}^{\text{Bent Plate}} + \overbrace{580(\delta)}^{\text{Relaxation Load}} = \overbrace{24.47}^{\text{Load}}$$

Solving the above equation $\delta = .0288$ "

Check Equilibrium:

$$\text{Anchor} : F_C = 4.2 + 11.75(.0288 - .012) = 4.39$$

$$\text{Bent plate} : F_{BP} = 3. + 60 (.0288 - .0231) = 3.42$$

$$\text{Relaxation} : F_{RP} = 580 (.0288) = 16.70$$

$$\sum \text{Forces} = 24.51 \approx 24.47$$

The energy used up in obtaining equilibrium in the anchor is:

$$E = 1/2 (4.2)(.012) + 4.2(.0288 - .012) + (4.39 - 4.2)(.0288 - .012)/2$$

$$E = .0974 \text{ K-in/in}$$

The safety factor is:

$$S.F. = E_T / E = \frac{.541}{.0974} = 5.56$$

Case IV

Accident Condition:

All data is the same as Case II except the concrete modulus is $E_c = 2.67 \times 10^6$ psi. The anchor spring constant is $K_C = 135$ K/in/in. (See Figure 21).

Find the force to put on the first anchor which simulates the effects of all other anchors on the first anchor by using equation 9 together with: $N = 21.6$ K/in/in., $K_{RP} = 580$ K/in/in., $K_C = 135$ K/in/in., and $K_{BP} = 130$ K/in/in.

$$N' = 21.6 (1.795) = 38.8 \text{ K/in.}$$

The force which must be subtracted from N' to simulate the effect of a 55 psi internal pressure: $N'' = 5.03$ K/in.

$$N_T = 38.8 - 5.03 = 33.77 \text{ K/in.}$$

Try an Elastic Solution to find the approximate value of

$$\delta = 33.77 / (135 + 130 + 580) = .04"$$

Plastic Solution

$$\begin{array}{ccccccc} \text{Anchor} & & \text{Bent Plate} & & \text{Relaxation} & \text{Load} & \\ \hline 4.2 + 16.35(\delta - .031) & + & 3.54 - 35.6(\delta - .0321) & + & 580(\delta) & = & 33.77 \end{array}$$

Solving the above equation $\delta = .0455"$

Check equilibrium:

$$\text{Anchor} : F_C = 4.2 + 16.35(.0455 - .031) = 4.44 \text{ K/in.}$$

$$\text{Bent plate} : F_{B\cancel{P}} = 3.54 - 35.6 (.0455 - .0321) = 3.06 \text{ K/in.}$$

$$\text{Relaxation} : F_{R\cancel{L}} = 580 (.0455) = 26.40 \text{ K/in.}$$

$$\sum \text{ Forces} = 33.90 \approx 33.77$$

The energy used up in obtaining equilibrium in the anchor is:

$$E = 1/2(4.2)(.031) + 4.2 (.0455 - .031) + (4.44 - 4.20)(.0455 - .031)/2$$

$$E = .128 \text{ K-in/in}$$

The safety factor is:

$$\text{S.F.} = E_T / E = \frac{.541}{.128} = 4.23$$

The results of Cases I to IV are summarized below:

Condition	Case I Operating	Case II Accident	Case III Accident	Case IV Accident
Concrete Modulus (psi)	5.4×10^6	5.4×10^6	7.0×10^6	2.67×10^6
Anchor Spring Const. K/in/in.	270	270	350	135
Anchor Disp. (in.)	.0134	.0318	.0288	.0455
Anchor Force K/in.	3.62	4.41	4.39	4.44
Safety Factor	22.4	5.20	5.56	4.23

5.2 Effects of Moment on the Anchor

Due to the inward deflection of a panel with initial inward curvature a moment will exist at the anchor as shown in Figure 22.

FIG. 22

Since both the angle and the plate adjacent to the deflected panel are capable of carrying moment, the moment M will distribute. Figure 23 shows the model that was used to evaluate the distribution of M .

FIG. 23

The analysis indicated that approximately 50% of M is taken by the angle.

Assuming that the deflected curve may be approximated by the following relationship:

$$W = W_m \delta \sin^2 \frac{\pi x}{a} \quad \text{Then using } M = EI \frac{d^2 W}{dx^2} \quad \text{the end}$$

moment will be: $M = 2W_m \left(\frac{\pi}{a}\right)^2 EI$

Using Case II of the previous analysis the axial displacement was found to be: $\delta = .0318$ " with a total displacement of $2\delta = .0636$ " when including both ends. Now the information given in Appendix A may be used to get an approximation of W_m . Then $W_m = .20$ " and the moment is found to be:

$$M = 2(.20) \left(\frac{\pi}{152}\right)^2 (30 \times 10^3) / (768) = .685 \text{ K-in/in.}$$

The maximum moment at the start of yield will be:

$$M = S_e \sigma_y = 1/6 (1/4)^2 53.7 = .560 \text{ K-in/in.}$$

The moment of .685 K-in/in. is not a true moment but it may be used to evaluate the approximate amount of strain in the weld which attaches the angle to the liner plate. As stated previously

the moment into the weld and angle is approximately 50% the applied moment; therefore a moment $M' = .50(.685) = .342$ K/in/in. will act on the weld. Since the attachment weld is a 3/16 - 4 - 12 intermittent fillet weld, the previous moment may be increased by a factor of 3 and then the weld may be considered as a continuous fillet. Considering the following sketch, the force in the weld is:

$$3/8 F = 3(.342)$$

$$F = 8(.342) = 2.74 \text{ K/in.}$$

Under the previous force, it is possible for the weld to reach yield in this direction, especially when combined with the other forces. This is not particularly of major concern when considering the two situations shown below:

Typical
From Tests
(See Appendix B)

Predicted
Worst Condition For
 $E_c = 5.4 \times 10^6$ psi

Since the strains from the predicted case are considerably lower than those obtained during testing, there is no possibility of a failure.

5.3 Effects of Radial Force on the Anchor

Figure 24 shows a panel with inward curvature adjacent to a panel with outward curvature and also the deflected shape the panels would take if the anchor did not restrain against this condition.

FIG. 24

As a conservative solution for evaluating F consider Figure 25 and evaluate the force needed to form plastic hinges at A, B and C.

FIG. 25

$$M_B = Fa/2 - M_p \quad \text{Let } M_B = M_p \quad \text{then } F = \frac{4M_p}{a}$$

$$M_p = \sigma_y \delta_p = (74500) \left[\frac{1}{4} \left(\frac{1}{4} \right)^2 (1) \right] = 1.16 \text{ in-K/in}$$

$$F = 4/15 (1.16) = 310 \text{ lbs./in.}$$

The force F is relatively small compared to the capacity of the anchorage system. The resistance against pullout will be developed through shear and bond as shown below. Both shear and bond should ultimately develop at least 250 psi.

Capacity of System

$$R = 250(3+3)(1) = 1500 \text{ lbs/in.}$$

5.4 Effects of Longitudinal Loads on the Anchor

Due to longitudinal strains in the concrete, the liner plate will be subjected to loads in the same direction. Since the liner plate loads result primarily from concrete creep and shrinkage, prestress and thermal gradients through the wall, the weld attaching the anchor to the liner plate, will in reality be subjected to a self limiting strain. The longitudinal strains which were given in the previous Cases I to IV are very small compared to the shear strains resulting from tangential loads. Under accident conditions the tangential strain is predicted to be approximately 25% of ultimate while the longitudinal strain is approximately at yield. The longitudinal and tangential strains will create a slightly higher maximum principal strain. But due to the much greater tangential strain relative to the longitudinal strain, the effect of the longitudinal strain should have a negligible effect on the ultimate capacity of the anchorage system.

5.5 Longitudinal Buckling of the Liner Plate and Anchorage System

The following discussion will pertain to the situation shown in Figure 4. The following analysis will derive an approximate expression which may be used to determine the amount of radial force which must be developed in the anchorage system to prevent longitudinal buckling. In this situation, longitudinal buckling may be defined as a breaking away of the anchorage system so that a portion of the liner plate and anchorage system will displace radially inward. Figure 26 shows a cross section of the liner plate together with a model of the system.

FIG. 26

The general differential equation applicable to this problem is:

$$\frac{EI d^4 W}{dx^4} + \frac{Nd^2 W}{dx^2} + \frac{Nd^2 \Delta}{dx^2} = -q$$

Since the load imposed in the liner plate is transmitted from concrete strain, it is not really an end load but a load distributed along the length. Assuming the following relationships, an approximate solution may be found.

$$\text{Let } W = W_m \sin\left(\frac{2\pi x}{L}\right) \quad \Delta = \Delta_m \sin\left(\frac{2\pi x}{L}\right) \quad N = \int_0^L n dx$$

and solving the differential equation using an average value of N:

$$N_{av} = \frac{\int_0^L n x dx}{L} = \frac{nL}{2} = \frac{N}{2}$$

$$W_m = \frac{N \Delta_m - \frac{qL^2}{\pi^2}}{\frac{8\pi^2 EI}{L^2} - N}$$

For displacement to be zero, then:

$$\Delta_m N = qL^2 / \pi^2 \quad \text{or} \quad q = \frac{\Delta_m \pi^2 N}{L^2}$$

For $\Delta_m = \frac{1}{2}$ " : Maximum allowable initial displacement

$N = (71000)(3.13) = 223000$ lbs: (Maximum expected stress during accident) x (Area shown in Figure 26)

$L = 44$ " : Shortest restrained length.

$$q = \frac{(.5)(9.85)(223000)}{1960} = 560 \text{ lbs/in.}$$

Since the force carrying capacity of the anchorage system in the radial direction is at least 1500 lb/in (as shown in Section 5.3) there is no possibility of longitudinal buckling.

APPENDIX A

Bent Plate Tests

The purpose of the tests was to determine the load vs. displacement relationship in an axially loaded plate with large initial eccentricity. Three specimens made of ASTM A36 material were tested for Bechtel Corporation by Anamet Laboratories, Inc., Berkeley, California. The specimen had a tensile yield strength of 43 ksi. The length of plate used was 15" which simulates the anchor spacing. The initial eccentricity or lateral displacement of the plate was 1/8". The plates were pre-curved to approximately a sine squared curve. Figure A-1 shows a sketch of the test setup.

FIG. A-1

The plates were loaded in the axial direction. As the load was applied, displacement readings were obtained in the axial and lateral directions. Figure A-3 shows the axial displacement vs. load curve for the average of the three specimens tested. Figure A-2 shows the lateral displacement vs. load curve for the average of the three specimens tested. The axial displacement value must be divided by two when using this data in the liner plate analysis since the analysis considers the plate being loaded from both ends simultaneously with movement at both ends. Included is the report from Anamet Laboratories pertaining to this testing.

FIG. A-2

FIG. A-3

LABORATORY CERTIFICATE

Anamet Laboratories, Inc.

2827 SEVENTH STREET . BERKELEY 10, CALIFORNIA . THORNWALL 1-5771

ANALYTICAL
CHEMICAL
METALLURGICAL

HIGH TEMPERATURE
APPLIED RESEARCH
PHYSICAL TESTING

September 29, 1969

LABORATORY NUMBER: 969.256
SAMPLE: Three (3) 1/4" Plate Specimens
DATE SUBMITTED: September 23, 1969
REPORT TO: Bechtel Corporation
9th Floor Section C-14
50 Beala Street
San Francisco, California
Attn: Mr. Ted Johnson

R E P O R T

Three (3) 1/4" plate specimens, 6" x 15", of A36 material were bent $1/8" \pm 1/32"$ (see attached sketch) prior to compressing them in the longitudinal direction. Load verses deflection was measured at three (3) locations; to determine the lateral movement at both edges of the specimen at the bend and the verticle movement as the specimen compressed. This data is attached.

A tensile specimen was also prepared and tested from one of the plate specimens. The results of this test is also attached.

Respectfully submitted,
ANAMET LABORATORIES, INC.

By S. J. Johnson

LABORATORY CERTIFICATE

Anamet Laboratories, Inc.
2027 SEVENTH STREET • BERKELEY 10, CALIFORNIA

Lab. No. 960.256

TENSILE TEST

Dimensions of Specimen:	0.493 x 0.245 Inches
Area:	0.121 Square Inches
Tensile Strength:	65300 Psi
Yield Strength @ 0.2% Offset:	43000 Psi
Elongation in 2" Gage:	33%

Page One

LABORATORY CERTIFICATE

Hammett Laboratories, Inc.
2027 SEVENTH STREET • BERKELEY 10, CALIFORNIA

Lab. No. 969.256 continued

PLATE #1 - BENT PLATE TEST

<u>Gage 1</u> <u>(inches)</u>	<u>Gage 2</u> <u>(inches)</u>	<u>Gage 3</u> <u>(inches)</u>	<u>Load</u> <u>(in pounds)</u>
0.02	0.014	0.0245	5400
0.04	0.035	0.037	9000
0.06	0.055	0.035	11400
0.08	0.075	0.038	13200
0.10	0.094	0.047	14700
0.15	0.143	0.040	16600
0.20	0.197	0.056	17500
0.25	0.239	0.063	17000
0.30	0.288	0.069	16300
0.40	0.386	0.084	14600
0.50	0.484	0.101	13200
0.60	0.586	0.122	11800
0.70	0.688	0.146	11300
0.80	0.788	0.171	9800
0.90	0.889	0.200	9200
0.990	0.985	0.234	8500

Page Two

LABORATORY CERTIFICATE

Anamet Laboratories, Inc.
2027 SEVENTH STREET • BERKELEY 10, CALIFORNIA

Lab. No. 960.255 continued

PLATE #2 - BENT PLATE TEST

<u>Gage 1</u> <u>(inches)</u>	<u>Gage 2</u> <u>(inches)</u>	<u>Gage 3</u> <u>(inches)</u>	<u>Load</u> <u>(in pounds)</u>
0.02	0.012	0.013	5700
0.04	0.019	0.020	7800
0.06	0.026	0.027	9700
0.08	0.034	0.036	11200
0.10	0.043	0.047	13000
0.15	0.080	0.084	15500
0.20	0.120	0.072	15500
0.25	0.159	0.077	16700
0.30	0.200	0.084	16300
0.40	0.293	0.097	16000
0.50	0.392	0.113	13500
0.60	0.491	0.133	12200
0.70	0.592	0.155	11200
0.80	0.697	0.180	10200
0.90	0.795	0.207	9400
1.00	0.897	0.238	8600

Page Three

LABORATORY CERTIFICATE

Anamet Laboratories, Inc.
127 SEVENTH STREET • BERKELEY 10, CALIFORNIA

Lab.No. 969.256 continued

PLATE #3 - BENT PLATE TEST

<u>Gage 1</u> <u>(inches)</u>	<u>Gage 2</u> <u>(inches)</u>	<u>Gage 3</u> <u>(inches)</u>	<u>Load</u> <u>(in pounds)</u>
0.02	0.018	0.008	5300
0.04	0.036	0.012	7900
0.06	0.054	0.015	9900
0.08	0.074	0.019	11900
0.10	0.091	0.022	13300
0.15	0.134	0.028	15900
0.20	0.178	0.034	16900
0.25	0.224	0.040	16800
0.30	0.271	0.048	15900
0.40	0.370	0.063	14200
0.50	0.470	0.082	12800
0.60	0.570	0.103	11450
0.70	0.673	0.131	10600
0.80	0.774	0.155	9700
0.90	0.875	0.185	9000
1.00	0.974	0.217	8400

Page Four

LABORATORY CERTIFICATE

Anamet Laboratories, Inc.

2827 SEVENTH STREET • BERKELEY 10, CALIFORNIA • THORNWALL 1-5771

ANALYTICAL
CHEMICAL
METALLURGICAL

HIGH TEMPERATURE
APPLIED RESEARCH
PHYSICAL TESTING

$$L = 15''$$

Bent Plate Specimen

Dim: $\frac{1}{4}'' \times 6'' \times 15''$

Not to scale

APPENDIX B

Liner Plate Anchorage Tests

The following information is a summary of the past concrete anchorage tests that are fully explained in Reference 1.

The specimens consisted of 1/4" A36 plate with an attached A36 1/4" x 3" x 2" angle welded to the plate with a 3/16" 4-12 intermittent fillet weld. The welding rod used was an E6010 electrode. The angles were embedded in concrete blocks and when the concrete reached a strength of approximately 5,000 psi, the specimens were tested. Figure B-1 shows a sketch of the test setup.

FIG. B-1

The test was performed by loading the plate which then exerted a shear load upon the angle embedded in the concrete. Table B-1 summarizes the test results.

TABLE B-1
TEST RESULTS

Specimen Test No.	Spring Constant (k/in/in)	Maximum Load (k/in)	Max. Displace- ment (in.)	Total Energy Available (in.-lbs/in.)
I.	131.7	4.94	.313	1263.
II.	123.5	4.45	.625	900.
III.	151.5	6.16	.250	722.
IV.	132.7	5.97	.125	541.

Reference 1: "Reactor Building Liner Plate Anchorage Tests for Arkansas Nuclear One" - Supplement No. 11 of the Arkansas Nuclear One PSAR.

- TEST I, ANGLE LEG UP, CENTER WELD DOWN
- - - TEST II, ANGLE LEG UP, CENTER WELD UP
- TEST III, ANGLE LEG DOWN, CENTER WELD DOWN
- - - TEST IV, ANGLE LEG DOWN, CENTER WELD UP

FIG. B-2

PART II

MAIN STEAM PIPE LINER PLATE PENETRATION

CONTENTS

1.0	General	1-1
2.0	Main Steam Piping System Properties	2-1
2.1	Derivation of Main Steam Piping System Properties	2-1
3.0	Penetration Assembly Loading	3-1
3.1	Calculation of Loads	3-3
3.2	Rupture Loading Acting Upon Nozzle	3-9
4.0	Nozzle Proportioning	4-1
4.1	Design Parameters	4-1
4.2	Loading Condition	4-1
4.3	Nozzle Design Thickness	4-2
5.0	Nozzle Stress Analysis	5-1
5.1	Nozzle Plastic Capacity	5-1
5.2	Detailed Analysis of Nozzle Section (1)	5-4
5.3	Detailed Analysis of Nozzle Section (2)	5-5
6.0	Flanges and Gussets	6-1

7. Flanges	7-1
7.1 Concrete Bearing	7-1
7.2 Flange Bending and Shear	7-2
8. Gussets	8-1
8.1 Loading	8-1
8.2 Concrete Bearing	8-1
8.3 Gusset Bending	8-2
 Appendix A	 A-1
References	

LIST OF FIGURES

Fig. No.

1-1	Main Steam Pipe Penetration Assembly
3-1	Loading Conditions I, II, and III Rupture Loads
3-2	Loading Conditions IV and V Steady State Operating Temperature Distribution
5-1	Combined Thermal Stress Diagram
7-1	Flange Bearing
7-2	Flange Plastic Analysis
8-1	Gussets
8-2	Gusset Stress Distribution
A-1	Plastic Stress Distribution in Circular Cylindrical Shell Subject to Axial Load, Shear, Bending and Torsion

Table No.

LIST OF TABLES

3-1	Summary of Loading Conditions

Symbols and Notations

A_o	Steam Pipe Flow Area
A_p	Plastic Shear Area
a	Inside Radius of Flange
b	Outside Radius of Flange
C	Coefficient
C_D	Ratio of Penetration Resistance to Peak Rupture Force
d_n	Mean Nozzle Diameter
d_p	Average Diameter of Steam Pipe
d_p'	Outside Diameter of Steam Pipe
F	Force
f_c	Concrete Compressive Stress
f_{cmax}	Maximum Concrete Stress
f_c'	Ultimate Compressive Strength of 6" x 12" Concrete Cylinder at 28 Days
f_v	Shear Stress in Weld
f_s	Bending Stress in Weld
K	Thrust Coefficient
L_1, L_2, L_3	Axial Distances from Section (0) to Sections (1), (2) and (3)
M	Bending Moments applied to Penetration Assembly
M_p	Plastic Moment Capacity
M_u	Ultimate Moment Capacity
m, m'	Bending Moments per unit length
N, N_o	Axial Force
P	Rupture Force

p	Steam Pipe Operating Pressure
R, r	Radii
S_p	Plastic Section Modulus
T, T_o, T_1	Torsional Moments
T_u	Ultimate Torsional Moment Capacity
T_y	Torsional Moment at Yield
t	Thickness of Plate
t_n	Thickness of Nozzle
t_p	Thickness of Steampipe
U	Strain Energy of Distortion
V, V_o, V_L, V_R	Shear Force
V_p	Plastic Shear Capacity
V_u	Ultimate Shear Capacity
σ_u	Ultimate Stress
σ_y	Yield Point Stress (NOMINAL)
σ_r	Radial Stress
σ_z	Axial (Longitudinal) Stress
$\sigma_I, \sigma_{II}, \sigma_{III}$	Principal Stresses
σ_ϕ	Hoop (Tangential) Stress
$\tau, \tau_{xy}, \tau_{\phi z}, \tau_{zr}, \tau_{r\phi}$	Shear Stress
τ_u	Ultimate Shear Stress
θ, ϕ	Angle to Neutral Axes
ω	Angle
c, e, g, h, j	Dimensions

1.0 GENERAL

The following illustrates the design of a main steam pipe containment penetration assembly in accordance with the criteria given in Reference 1, BC-TOP-5.

The penetration assembly includes the nozzle, flanges or reinforcing plate on the inside face of the containment and attached anchors such as the gusset plates.

The main steam piping and the flued head details are not within the scope of this example with respect to design.

1.1 Loading

The following loadings were considered in this example in the analysis of the penetration assembly.

(a) Steam Pipe Rupture Loads

Three steam pipe rupture configurations which will impose maximum loading conditions upon the penetration assembly were examined. (The loading conditions are denoted I, II, and III)

(b) Steam Pipe (Normal) Operating Load

(c) Steady State Operating Thermal Load

This load encompasses any transient thermal loads which may be imposed on the penetration assembly during its lifetime.

Loading Condition IV constitutes the Thermal Load (c) combined with any of the steam pipe rupture loadings. The loading is applied as a "cyclic" load where the number of cycles is 10.

Loading Condition V constitutes the steam pipe operating load (b) in conjunction with the Thermal Load (c). The loading is applied as a cyclic load. The number of (cycles) transients used in the design is 100.

1.2 Design Criteria

The steam pipe is assumed to be able to reach ultimate strength at the junction to the flued head when subject to rupture loading.

The ultimate strength of the pipe is reached when the energy of distortion of the steam pipe material* reaches its ultimate value at the full cross-section.

The capacity of the penetration assembly (nozzle, flange, and gussets) however, is based on the material reaching its plastic capacity and (conservatively) following Tresca's yield criterion.

* Based on v. Mises Failure Criterion.

For normal operating loading and for the analysis for cyclic operation, the ASME Code, Section III (Reference 5) design criteria have been adhered to.

The first part of this design example involves the determination of maximum loads transmitted from the piping system to the nozzle. The piping system is assumed unrestrained, allowing it to develop its ultimate strength. Due to the nature of the rupture loading, it is assumed to be applied instantly. The second part of this example involves the penetration assembly design. Since the rupture loading is assumed to be impulsive in nature, the penetration assembly is designed to have a resistance of approximately 1.2 times the peak impulse load and to displace to about three times yield. Bechtel topical report BN-TOP-2, Reference (2) illustrates these techniques. For orientation, the final design configuration is shown in Figure 1-1.

FIGURE 1-1 MAIN STEAM PIPE PENETRATION ASSEMBLY

2.0 MAIN STEAM PIPING SYSTEM PROPERTIES

(a) Outside diameter	$d_p' = 28 \text{ in}$
(b) Thickness	$t_p = 1\frac{1}{2} \text{ in}$
(c) Average diameter	$d_p = 26\frac{1}{2} \text{ in}$
(d) Pressure	$p = 1300 \text{ psig}$
(e) Temperature	$T = 570^\circ\text{F}$
(f) Ultimate strength (for material SA 155 KC 70)	$\sigma_u = 70 \text{ ksi}$

2.1 Derivation of Main Steam Piping System Properties

(a) Steam Pipe flow area:

$$A_o = \frac{\pi}{4} [d_p - t_p]^2 = \frac{\pi}{4} [26.5 - 1.5]^2 = 490 \text{ in}^2$$

(b) Steam Pipe force for postulated rupture*:

$$P = K(pA_o) = (1.25)(1.3)(490) = 797 \text{ k}$$

(c) Plastic section modulus:

$$S_p \approx (t_p)(d_p)^2 = (1.5)(26.5)^2 = 1050 \text{ in}^3$$

(d) Plastic shear area:

$$A_p \approx 2(t_p)(d_p) = 2(1.5)(26.5) = 79.5 \text{ in}^2$$

*K is the "thrust - coefficient" which is a function of rupture orifice geometry and type of flow. Here $K = 1.25$. See BN-TOP-2, Reference 2.

3.0 PENETRATION ASSEMBLY LOADING

The following loading conditions based on conservative assumptions are considered:

- (1) Loading Condition I: Maximum bending and shear from rupture loads
- (2) Loading Condition II: Maximum torsion, bending and shear from rupture loads
- (3) Loading Condition III: Maximum axial force and bending from rupture loads
- (4) Loading Condition IV:
Operating thermal load in combination with any of the rupture loads (Loading Conditions I, II, or III.)
- (5) Loading Condition V:
Steam Pipe Operating Load plus Operating Thermal Load.

Loading Conditions IV and V satisfy the ASME, Section III Code requirements for cyclic operation.

The first three loading conditions are caused by postulated rupture in the main steam pipe, located and oriented as to cause maximum loadings in the penetration assembly. See Figure 3-1 for a summary of the Loading Conditions I, II, and III which are based on the following three assumptions:

- (1) Ultimate capacity of the steam pipe is reached at Section (0).
- (2) Material satisfies v. Mises Maximum Energy of Distortion Criterion.
- (3) The effect of pressure in the main steam pipe downstream of the rupture location has been disregarded in reducing the ultimate strength.

3.1 Calculation of Loads

The development of v. Mises Maximum Energy of Distortion Criterion which evaluates the ultimate strength is found in Chapter 6 of Reference 3. For the conservative case of a pipe assumed to have only longitudinal membrane stress σ_z , and membrane shear τ_{xy} , the strain energy of distortion becomes:

$$U = c(\sigma_z^2 + 3\tau_{xy}^2) \quad \text{for biaxial stress condition.}$$

The strain energy for uniaxial stress condition is

$$U_1 = c(\sigma_z)^2.$$

Failure occurs according to v. Mises Theory when U reaches ultimate:

$$U_u = c(\sigma_u)^2, \text{ or when } U_u = c(\sigma_z^2 + 3\tau_{xy}^2) \quad \text{or} \\ \sigma_u = \left(\sigma_z^2 + 3\tau_{xy}^2 \right)^{\frac{1}{2}} \quad (3-1)$$

The stresses σ_z and τ_{xy} may be expressed in terms of a bending moment (M) and a shear (V) acting on a pipe section.

$$\sigma_z = \frac{M}{S_p} : \quad \tau_{xy} = \frac{V}{A_p \sin \theta} \quad (3-2)$$

where $S_p = t_p (d_p)^2$ is the plastic section modulus, and

$A_p = 2t_p (d_p)$ is the plastic shear area of the pipe.

(2θ) is the angle bounded by the planes of zero shear strain and may be determined from:

$$\frac{\sin \theta}{\left[\frac{\pi}{2} - \theta \right]} = \frac{V}{T} \left[\frac{d_p}{2} \right] \quad (3-3)$$

where $\frac{V}{T}$ is the shear-torsion ratio. For derivation of formulas (3-2) and (3-3) see Equation A-10 in Appendix A.

Insertion of (3-2) into Equation (3-1) gives:

$$\left[\frac{M}{M_u} \right]^2 + \left[\frac{V}{V_u} \right]^2 = 1 \quad (3-4)$$

where $M_u = (\sigma_u)(S_p)$ is the assumed ultimate moment capacity and

$$V_u = \left[\frac{\sigma_u}{\sqrt{3}} \right] \left[A_p \right] \sin \theta \quad \text{is the assumed ultimate shear capacity.}$$

This is the shear-bending interaction equation for the steam pipe with a given shear-torque ratio of $\frac{V}{T}$. For a given $\frac{V}{T}$ value, θ is found from (3-3) and substituted into (3-4).

For combined bending and axial load a similar interaction formula applies.

From Equations A-5 and A-7 in the Appendix the following expressions are developed:

$$N = A_p (\phi) (\sigma_z)$$

$$M = S_p (\cos \phi) (\sigma_z)$$

$$N_u = A_p \left(\frac{\pi}{2} \right) (\sigma_u)$$

$$M_u = S_p (\sigma_u)$$

Since the shear is zero (σ_z) will be equal to (σ_u) and the interaction equation is:

$$\left[\frac{M}{M_u} \right]^2 + \sin^2 \left\{ \frac{\pi}{2} \left[\frac{N}{N_u} \right] \right\} = 1 \quad \text{or} \quad (3-5)$$

$$\frac{M}{M_u} = \cos \left\{ \frac{\pi}{2} \left[\frac{N}{N_u} \right] \right\} ; \quad (3-6)$$

Forces and moments exerted by the ruptured steam pipe upon Section (0) may now be derived from Equations (3-3) and (3-4) or (3-6).

3.1.1 Load Condition I

Shear Force $V_o =$ Rupture Force $P = 797$ k

Torsion $T_o = 0$;

$\sin \theta = 1$;

Equation (3-4) becomes:

$$\left[\frac{M_o}{(70)(1,050)} \right]^2 + \left[\frac{V_o}{\frac{70}{\sqrt{3}}(79.5)} \right]^2 = 1 \quad \text{or}$$

$$\left[\frac{M_o}{73,500} \right]^2 + \left[\frac{V_o}{3213} \right]^2 = 1;$$

Insertion of $V_o = 797$ k results in:

Bending $M_o =$ 71,200 in-k.

3.1.2 Load Condition II

Shear force $V_o =$ Rupture force $P = 797$ k

Bending $M_o =$ Zero

$$\sin \theta = \frac{V_o}{A_P \sigma_u / \sqrt{3}} = \frac{797}{(79.5)(70/\sqrt{3})} = 0.248$$

$\theta = 14^\circ 21' \approx 0.250$ radians. From Equation (A-8) of Appendix A:

$$\text{Torque } T_o = \left[\frac{\sigma_u}{\sqrt{3}} \right] (S_P) \left(\frac{\pi}{2} - \theta \right);$$

$$T_o = \left(\frac{70}{\sqrt{3}} \right) (1,050) [1.57 - 0.25] = \underline{\underline{56,000 \text{ in -k}}}$$

3.1.3 Load Condition III

Axial force $N_O =$ Rupture force $P = \underline{797 \text{ k}}$

Shear Force $V_O = \underline{\text{Zero}}$

Torsion $T_O = \underline{\text{Zero}}$

From Equation (3-6)

$$\frac{M_O}{M_u} = \cos \left\{ \frac{\pi}{2} \left[\frac{N}{N_u} \right] \right\}$$

where $M_u = \sigma_u (S_p) = 70.0 (1,050) = 73,500 \text{ in-k};$

$$\text{and } \frac{\pi}{2} \left(\frac{N}{N_u} \right) = \frac{\frac{\pi}{2} (797)}{\frac{\pi}{2} A_p (\sigma_u)} = \frac{797}{79.5(70.0)} = .143;$$

so that

$$M_O = 73,500 \cos (1.43) = \underline{\underline{72,740 \text{ in-k}}}.$$

The loading conditions are summarized in Table 3-1 for the pipe rupture loads.

3.1.4 Load Condition IV

This is a combined loading due to nozzle axial temperature gradient acting concurrently with any one of Loading Conditions I, II or III. A heat transfer analysis using Bechtel Program ME620, Reference 4, established the steady state operating temperature distribution shown in Figure 3-2.

The temperatures are based on a mean steam temperature of 600°F. The temperature at the nozzle-flued head interface (Section (1)) is 450°F and the temperature at the containment wall (Section (2)) is 200°F.

This loading occurs only once but ten cycles are considered in the design.

3.1.5 Load Condition V

The loading condition consists of the steady state operating thermal load derived above for Load Condition IV (Figure 3-2) plus the following operating loads:

Axial Force $N_o = 50 \text{ k}$

Bending Moment $M_o = 17800 \text{ in-k}$

Shear Force $V_o = \text{Zero}$

Torsion $T_o = \text{Zero}$

These loads vary for each piping system. For illustration of the design technique the above loads were assumed.

A maximum of 100 cycles is considered for this loading condition.

3.2 RUPTURE LOADING ACTING UPON NOZZLE

Following are the relations between the forces and moments exerted by the ruptured steam pipe at Section (0), and the forces and moments resisted by the nozzle at Section (1), (2), or (3):

$$V = C_D V_O$$

$$T = C_D T_O$$

$$N = C_D N_O$$

$$M = C_D [M_O + V_O L]$$

where V , T , N and M denote forces and moments at Sections (1), (2), or (3), and L denote the distance from Section (0) to the nozzle section considered.

C_D is equal to the penetration resistance divided by the peak calculated rupture force, moment and torque acting upon the penetration assembly. The ratio results from the fact that the applied loads are impulsive in nature. For a complete explanation see Reference 2. For the penetration assembly design, the system will be allowed to strain to approximately 3 times yield which results in $C_D = 1.20$.

FIGURE 3-1   LOADING CONDITIONS I, II, AND III
RUPTURE LOADS

FIGURE 3-2   LOADING CONDITIONS IV AND V
 STEADY STATE OPERATING TEMPERATURE DISTRIBUTION

TABLE 3-1
SUMMARY OF LOADING CONDITIONS I, II AND III

Loading Condition	ITEM Description of Loading	Moments At Section (0)		Forces At Section (0)		Moments At Section (1)		Forces At Section (1)	
		Bending	Torsion	Shear	Axial	Bending	Torsion	Shear	Axial
		in-k	in-k	k	k	in-k	in-k	k	k
I	Pipe Rupture Located to Produce Max. Bending	71,200	---	797	--	97,000	---	956	--
II	Pipe Rupture Located to Produce Max. Torsion	---	56,000	797	--	11,480	67,200	956	--
III	Pipe Rupture Located to Produce Max. Axial Force	72,740	---	--	797	87,000	---	---	956

For location of Sections see Figure 3-1.

TABLE 3-1 (CONTINUED)

SUMMARY OF LOADING CONDITIONS I, II AND III

Loading Condition	ITEM	Moments At Section (2)		Forces At Section (2)		Moments At Section (3)		Forces At Section (3)	
		Bending	Torsion	Shear	Axial	Bending	Torsion	Shear	Axial
	Description of Loading	in-k	in-k	k	k	in-k	in-k	k	k
I	Pipe Rupture Located to Produce Max. Bending	112,750	--	956	--	120,000	---	956	--
II	Pipe Rupture Located to Produce Max. Torsion	27,250	67,200	956	--	34,600	67,200	956	--
III	Pipe Rupture Located to Produce Max. Axial Force	87,000	--	--	956	87,000	---	--	956

For location of Sections see Figure 3-1.

4.0 NOZZLE PROPORTIONING

4.1 Design Parameters

An average nozzle diameter of 46 inches will be used in the design together with the following information:

- (1) Material ..... SA516 Grade 70
- (2) Yield stress of nozzle at 450°F $\sigma_y = 31.80$ ksi
0.9 of nominal yield stress $0.9\sigma_y = 28.60$ ksi
- (3) Yield stress of nozzle at 200°F $\sigma_y = 34.60$ ksi
0.9 of nominal yield stress $0.9\sigma_y = 31.10$ ksi
- (4) Cantilever Arms ...
 $L_1 = 1'-0"$ ..... $L_2 = 2'-4\frac{1}{2}"$ ..... See Figure 3-1.
- (5) Thickness of Concrete Wall ..... $H = 3'-6"$
- (6) Concrete compressive strength $f'_c = 5.0$ ksi

4.2 Loading Condition

Loading Condition II, which usually governs the nozzle design, will be treated in this example.

- (a) Ultimate capacity of steam pipe in "pure" torsion:

from Equation (A-8) in Appendix A:

$$T_u = \frac{\pi}{2} (t_p) (d_p)^2 (\tau_u) = \frac{\pi}{2} (t_p) (d_p)^2 \frac{\sigma_u}{\sqrt{3}} \quad (4-1)$$

- (b) Plastic capacity of nozzle in "pure" torsion:

$$T_y = \frac{\pi}{2} (t_n) (d_n)^2 (\tau_y) = \frac{\pi}{2} (t_n) (d_n)^2 \left(\frac{.9\sigma_y}{\sqrt{3}} \right) \quad (4-2)$$

This is a special case of the Tresca Theory with $\sigma_z = 0$;

(see Equations (5-0) and (5-1)).

4.3 Nozzle Design Thickness

The nozzle will be designed to yield capacity for a loading based on the ultimate capacity of the steam pipe, multiplied by C_D :

$T_y = C_D(T_u)$ ..... from which the trial nozzle thickness t_n is obtained.

$$t_n = C_D \left[\frac{d_p}{d_n} \right]^2 \left[\frac{0.577 \sigma_u}{0.450 \sigma_y} \right] t_p \text{ ..... or}$$

$$t_n = 1.2 \left[\frac{26.5}{46.0} \right]^2 \left[\frac{0.577 \times 70.0}{0.450 \times 31.8} \right] \times 1.5 = 1.68''$$

However, use $t_n = 1\frac{7}{8}$ inches to account for possible combinations.

5.0 NOZZLE STRESS ANALYSIS

A stress analysis is performed for nozzle junction Section (1) and Section (2) at the face of the wall, to check if the shears and moments caused by Loading Conditions I, II, and III are within limits of the nozzle plastic capacity as given by the Interaction Equations (5-4) and (5-5).

An elasto-plastic stress analysis for both pipe rupture loads and operating loads combined with steady state operating thermal loads is subsequently performed to determine if the resulting stress intensities are within the limits established by the design fatigue curves of the ASME Section III Code (Reference 5) for the postulated number of cycles.

5.1 Nozzle Plastic Capacity

The derivation is based on Tresca's Yield Criterion:

$$\sigma_I - \sigma_{II} = 0.9\sigma_y \quad (5-0)$$

where $(\sigma_I - \sigma_{II})$ is the difference in principal stresses (or stress intensity):

$$\sigma_I - \sigma_{II} = \left[(\sigma_z)^2 + 4(\tau_{xy})^2 \right]^{\frac{1}{2}} \dots \text{by Tresca} \quad (5-1)$$

The axial stress (σ_z) and shear stress (τ_{xy}) may be expressed in terms of the bending moment (M), and the shear (V) which act at the section:

$$\sigma_z = \frac{M}{S_p} \quad \text{and} \quad \tau_{xy} = \frac{V}{A_p \sin \theta} \quad (5-2)$$

where (S_p), the plastic section modulus = $t_n (d_n)^2$
 (A_p), the plastic shear area = $2t_n d_n$. The angle (θ) to the planes of zero shear strain is determined from the shear-torsion ratio $\left(\frac{V}{T}\right)$ using the following expression (derived in the appendix as Equation (A-10)).

$$\frac{\sin \theta}{\left(\frac{\pi}{2} - \theta\right)} = \left(\frac{V}{T}\right) \left(\frac{d_n}{2}\right) \quad (5-3)$$

Substitution of Equations (5-2) into (5-1) and equating to (5-0) yields the following "Shear-Bending Interaction" expression for the nozzle:

$$\left(\frac{M}{M_p}\right)^2 + \left(\frac{V}{V_p}\right)^2 = 1 \quad (5-4)$$

where $M_p = 0.9\sigma_y (S_p)$ = Assumed yield capacity in bending.

$V_p = 0.45\sigma_y (A_p) \sin \theta$ = Assumed yield capacity in shear.

The applied moment (M) and shear (V) will not exceed the yield capacity of the nozzle if the expression on the left hand side of Equation (5-4) is equal to or less than unity.

The axial force-bending interaction expression for the nozzle is (see Equation 3-6):

$$\frac{M}{M_p} = \cos \left\{ \frac{\pi}{2} \left(\frac{N}{N_p} \right) \right\}; \quad (5-5)$$

5.1.1 Nozzle (Plastic) Section Properties

The "flued head to nozzle" junction (Section (1)) and the section at the face of the wall (Section (2)) are analyzed in the following:

Material..... A 516 Grade 70

Thickness ..... $t_n = 1.875"$

Outside Diameter ..... 48"

Average Diameter ..... $d_n = 46.125"$

Plastic Section Modulus .. $S_p = (t_n)(d_n)^2 = 3990 \text{ in}^3$

Plastic Shear Area ..... $A_p = 2(t_n)(d_n) = 173 \text{ in}^2$

Yield Stress at Section (1) $\sigma_y = 31.8 \text{ ksi}$

(Temperature ... 450°F)

Yield Stress at Section (2) $\sigma_y = 34.6 \text{ ksi}$

(Temperature ... 200°F)

Plastic Moment Capacity at Section (1) ... $M_p = S_p(.9\sigma_y) =$
 $3990(0.9)(31.8) = 114,200 \text{ in-k}$

Plastic Shear Capacity at Section (1) .... $V_p = A_p(.45\sigma_y) =$
 $173(0.45)(31.8) = 2,476 \text{ k}$

Plastic Moment Capacity at Section (2) ... $M_p = S_p(.9\sigma_y) =$
 $3990(0.9)(34.6) = 124,100 \text{ in-k}$

Plastic Shear Capacity at Section (2) .... $V_p = A_p(.45\sigma_y) =$
 $173(0.45)(34.6) = 2,693 \text{ k}$

5.2 Detailed Analysis of Nozzle Section (1)

5.2.1 Loading Condition I

$$M = 97,000 \text{ in-k} \quad V = 956 \text{ k} \quad T = 0 \text{ (See Table 3-1)}$$

Substituting into Equation (5-4):

$$\left(\frac{97,000}{114,200} \right)^2 + \left(\frac{956}{2476} \right)^2 = .72 + .15 = .87 < 1.00$$

5.2.2 Loading Condition II

$$M = 11,480 \text{ in-k} \quad V = 956 \text{ k} \quad T = 67200 \text{ in-k}$$

$$\therefore \frac{V}{T} = 0.0143 \dots \text{ from which } \sin \theta = 0.388$$

$$V_p = 2476 \sin \theta = 2476 (.388) = 960 \text{ k.}$$

Substituting into Equation (5-4):

$$\left(\frac{11,480}{114,200} \right)^2 + \left(\frac{956}{960} \right)^2 = 0.01 + 0.99 = 1.00$$

5.2.3 Loading Condition III

$$M = 87,000 \text{ in-k} \quad N = 956 \text{ k}$$

Substituting into Equation (5-5):

$$\frac{87,000}{114,200} \leq \cos \left(\frac{956}{4950} \right)$$

$$\text{Since } M_p = 114,200 \text{ in-k}$$

$$\begin{aligned} \text{and } N_p &= \frac{\pi}{2} (A_p) (.96_y) = \frac{\pi}{2} (173) (.9) (31.8) \\ &= \frac{\pi}{2} (4950) \end{aligned}$$

or

$$.765 \leq \cos (.193)$$

or

$$.765 \leq .980$$

5.2. Loading Condition IV

The thermal stresses are much higher at Section (2) than at Section (1). Therefore, only Section (2) will be analyzed for illustration.

5.2.5 Loading Condition V

Only Section (2) will be analyzed.

5.3 Detailed Analysis of Nozzle Section (2)

5.3.1 Loading Condition I

$M = 112750 \text{ in-k}$ $V = 956 \text{ k}$ $T = 0 \text{ (Table 3-1)}$

Substituting into Equation (5-4):

$$\left(\frac{112750}{124100}\right)^2 + \left(\frac{956}{2693}\right)^2 = .82 + .13 = .95 < 1.00$$

5.3.2 Loading Condition II

$M = 27750 \text{ in-k}$ $V = 956 \text{ k}$ $T = 67200 \text{ in-k}$

$\sin \theta = .388$ as for Section (1).

$$V_p = 2693(.388) = 1045 \text{ k}$$

Substituting into Equation (5-4):

$$\left(\frac{27750}{124100}\right)^2 + \left(\frac{956}{1045}\right)^2 = .05 + .84 = .89 < 1.00$$

5.3.3. Loading Condition III

$$M = 87000 \text{ in-k} \quad N = 956 \text{ k}$$

$$\phi = .244 \text{ as for Section (1).}$$

Substituting into Equation (5-5):

$$\frac{87,000}{124,100} = \cos \left(\frac{956}{5400} \right)$$

since:

$$N_p = \frac{\pi}{2} (A_p) (.9 \sigma_y) = \frac{\pi}{2} (173) (.9) (34.6) = 5400 \text{ k;}$$

so that

$$.705 \leq \cos(.177) \quad \text{or}$$

$$.705 \leq .985.$$

5.3.4 Loading Condition IV

(Steady State Operating Thermal-Plus Rupture Load)

The method used in calculating stress intensity follows NB-3222 of Reference 5. The loading condition is considered as a cyclic load with the number of cycles $n = 10$ and the limit on the range of primary and secondary stress intensity is established by using the simplified elastic-plastic analysis described in NB-3228.3 of Reference 5.

The highest stressed (strained) region of the nozzle (the "extreme" fibers) is considered.

(a) STRESSES DUE TO PIPE RUPTURE, LOADING CASE I

The stresses calculated in the following are indicated stresses which are derived from the calculated strains by multiplication by the modulus of elasticity.

(Bending Moment)

The forcing function (the pipe rupture loads) is assumed to be a step impulse.

The idealized (resistance) response function or idealized force displacement function of the nozzle is shown above.

The forcing function is in this case a moment:

$$M_F = M_O + V_O L$$

$$\underline{M_F} = 71200 + 797(28.5) = \underline{94000 \text{ in-k}}$$

as obtained from Table 3-1.

The maximum resisting moment capacity of the nozzle at Section (2) is when subject to shear ($V=797$).

$$M_R = M_P \left[1 - \left(\frac{V}{V_P} \right)^2 \right]^{\frac{1}{2}}$$

(See Interaction Formula (5-4))

The plastic moment capacity for zero shear is

$M_p = 124000$ in-k, and the plastic shear capacity for zero moment is $V_p = 2693$ k. (See Section 5.1.1)

Therefore,

$$M_R = 12400 \left[1 - \left(\frac{797}{2693} \right)^2 \right]^{\frac{1}{2}} \quad \text{or}$$

$$\underline{M_R = 119000 \text{ in-k.}}$$

Yielding of the "extreme" nozzle fibers start when the strain ϵ reaches ϵ_y .

The "full" capacity is assumed reached when $\epsilon = \epsilon_y'$.

The strain ratio:

$$\frac{\epsilon_{y'}}{\epsilon_y} = \frac{S_p}{S_e} = \frac{4}{\pi}$$

The maximum strain (ϵ) is reached when the strain energy absorbed by the nozzle is equal to the work done by the forcing function (rupture load), or when:

$$\epsilon(M_F) = (\epsilon - \epsilon_y') M_R + \frac{1}{2}(\epsilon_y') M_R$$

or when:

$$\epsilon = (\epsilon_y') \frac{\frac{M_R}{M_F}}{2 \left(\frac{M_R}{M_F} - 1 \right)}$$

For this case

$$\frac{M_R}{M_F} = \frac{119000}{94000} = 1.27$$

and

$$\epsilon = (\epsilon_y') \frac{1.27}{2(1.27 - 1)} = 2.34 \epsilon_y' ;$$

$$\epsilon = 2.34 \left(\frac{4}{\pi} \right) \epsilon_y = 2.34(1.27) \epsilon_y = 3.0 \epsilon_y$$

The "indicated" axial stress (σ_z) at the "extreme" fiber is:

$$(\sigma_z) = 3(.9)(34.6) = 93500 \text{ psi.}$$

The tangential shear stress will be approximately zero at the peak flexural stress location.

The diagram below shows the stresses in the "extreme fiber" of the nozzle at Section (2) due to Load Condition I (worst case) resulting from pipe rupture.

INDICATED STRESSES
AT
NOZZLE SECTION (2) DUE TO PIPE RUPTURE

(B) indicates bending stress ksi

(S) indicates shear stress ksi

(b) Stresses Due to Steady State Operating Thermal Load

The temperature distribution given in Figure 3-2 was used as input for Bechtel Program ME200 (Reference 6), resulting in the (axisymmetrical) nozzle stress distribution plotted in Figure 5-1. The figure illustrates that the stresses are greatest at the containment wall. Since the program assumes elastic deformations, all stresses will be "indicated stresses" rather than actual. The maximum indicated thermal stresses at Section (2) are shown below.

Nozzle

B Indicates Bending Stress (ksi)

M Indicates Membrane Stress (ksi)

S Indicates Shear Stress (ksi)

OPERATING THERMAL STRESSES AT NOZZLE SECTION (2)

(c) Combined Stresses due to Pipe Rupture and Steady State Operating Thermal Load

Combining the stresses which were derived above result in:

Outside Face Axial Stress.. $\sigma_z = +93.5 - 70.1 = +23.4$

Outside Face Hoop Stress .. $\sigma_\phi = -24.6 + -21.0 = -45.6$ ksi

Outside Face Shear Stress . $\tau_{z\phi} = 0 + 0 = 0$ ksi

Outside Face Shear Stress . $\tau_{zr} = 0 + 6.5 = 6.5$ ksi

Inside Face Axial Stress .. $\sigma_z = +93.5 + 70.1 = 163.6$ ksi

Inside Face Hoop Stress .. $\sigma_\phi = -24.6 + 21.0 = -3.5$ ksi

Inside Face Shear Stress .. $\tau_{z\phi} = 0 + 0 = 0$ ksi

Inside Face Shear Stress . $\tau_{zr} = 0 + 6.5 = 6.5$ ksi

COMBINED STRESSES DUE TO
 PIPE RUPTURE AND STEADY STATE
 OPERATING THERMAL LOAD AT NOZZLE SECTION (2)

The principal stresses σ_I , σ_{II} , and σ_{III} are derived from formulas found in Chapter 8 of Reference 3:

$$(\sigma)^3 - (I_S)(\sigma)^2 + (II_S)(\sigma) - (III_S) = 0$$

where:

$$(I_S) = \sigma_r + \sigma_\phi + \sigma_z$$

$$(II_S) = \sigma_r \sigma_\phi + \sigma_\phi \sigma_z + \sigma_z \sigma_r - \tau_{\phi r}^2 - \tau_{\phi z}^2 - \tau_{zr}^2$$

$$(III_S) = \sigma_r \sigma_\phi \sigma_z - \tau_{\phi z}^2 \sigma_r - \tau_{zr}^2 \sigma_\phi - \tau_{r\phi}^2 \sigma_z + 2\tau_{r\phi} \tau_{\phi z} \tau_{zr}$$

Insertion of the stresses derived for the inside face result in:

$$(I_S) = (0) - (3.5) + (163.6) = 160.1$$

$$(II_S) = (0) + (-3.5)(163.6) + (0) - (0) - (6.5)^2 = -615.5$$

$$(III_S) = (0) - (0) - (6.5)^2(-3.5) - (0) + 2(0) = 148.5$$

The equation

$$(\sigma)^3 - 160.1(\sigma)^2 - 615.5(\sigma) - 148.5 = 0;$$

has the solution:

$$\sigma_I = 163.9 \text{ ksi}$$

$$\sigma_{II} = -0.3 \text{ ksi}$$

$$\sigma_{III} = -3.5 \text{ ksi.}$$

The maximum stress intensity is:

$$\sigma_I - \sigma_{III} = 167.4 \text{ ksi.}$$

(d) Simplified Elastic-Plastic Analysis

Definition of Terms:

S_n = Range of Primary and Secondary Stress Intensity

S_n' = Range of Primary and Secondary Stress Intensity
excluding Thermal Bending Stresses.

S_a = Alternating Stress allowable from Figure J-9-1 (Reference 5)

S_a' Calculated Alternating Stress

S_m = Stress Intensity Value

m = Material Parameter

n = Material Parameter

The analysis follows NB3228.3 of Section III of the ASME Code (Reference 5). The range of primary plus secondary stress intensity is:

$$S_n = \sigma_I - \sigma_{III} = 167400 \text{ psi}$$

The range of primary plus secondary stress intensity excluding thermal bending is:

$$S_n' \leq 0 + 24580 = 24580 \text{ psi} < 3S_m = 3(23100) = 69300 \text{ psi}$$

(Thermal Membrane)

$$S_a \leq \frac{1}{2}(167400) = 83700 \text{ psi.}$$

The stress intensity value:

$$S_m = 23100 \text{ psi}$$

and the material parameters:

$$m = 3 \quad \text{and} \quad n = 0.2$$

for SA516 Grade 70 Material at 200°F.

are found from Table I-1.1 and NB 3228.3(b) of Reference 5.

$$\text{Check if } 3S_m < S_n < 3mS_m$$

$$3(23100) < 167400 < 3(3)(23100)$$

K_C is determined from the formula in NB 3228.3(b) of Reference 5.

$$K_C = 1.0 + \frac{1-n}{n(m-1)} \left(\frac{S_n}{3S_m} - 1 \right) = 1.0 + \frac{1-0.2}{0.2(3-1)} \left[\frac{167400}{3(23100)} - 1 \right] =$$

$$3.84$$

$$S_a' = K_C(S_a) = 3.84(83700) = 322000 \text{ psi.}$$

Entering the design fatigue curve (Figure I-9-1 of Reference 5) with the number of thermal cycles required ($N = 10$) results in an alternating stress value:

$$S_a = 600,000 \text{ psi, therefore } S_a' < S_a.$$

5.3.5 Loading Condition V

(a) Stresses Due to Operating Load from Steam Pipe

$$N = 50 \text{ k} \quad M = 17800 \text{ in-k}$$

(See paragraph 3.1.5)

This loading satisfies NB 3221 of Reference (5). The elastic stresses in the "extreme" fibers are:

$$\sigma_z = \frac{N}{A} \pm \frac{M}{S_E} = \frac{50}{272} \pm \frac{17800}{3133} = 0.18 \pm 5.69 = \begin{cases} 5.87 \text{ ksi} \\ -5.51 \text{ ksi} \end{cases}$$

$$S_m = 23.1 \text{ ksi for SA 516 Grade 70 material at } T = 200^\circ\text{F}.$$

(b) Stresses Due to Thermal Load

The stresses are derived and shown in paragraph 5.3.4(b).

(c) Combined Stresses from Steam Pipe Operating and Thermal Loads on Nozzle

This loading is considered as cyclic with number of cycles $N = 100$. Stress intensity limits must satisfy NB - 3228.3 of Reference 5.

Combining the stresses derived above results in:

$$\text{Axial Stress } \sigma_z = 70.13 + 5.87 = 76.0 \text{ ksi}$$

$$\text{Hoop Stress } \sigma_\phi = -24.58 + 21.04 = -3.5 \text{ ksi}$$

$$\text{Shear Stress } \tau_{rz} = 6.51 + (0) = 6.5 \text{ ksi}$$

The principal stresses are found from:

$$(\sigma)^3 - (I_S)(\sigma)^2 + (II_S)(\sigma) - (III_S) = 0$$

where

$$(I_S) = \sigma_r + \sigma_\phi + \sigma_z = (0) + 76.0 - 3.5 = 72.5$$

$$(II_S) = \sigma_z \sigma_\phi - \tau_{rz}^2 = -(76.0)(3.5) - (6.5)^2 = 308.3$$

$$(III_S) = -\tau_{rz}^2 (\sigma_\phi) = +(6.5)^2 (3.5) = +147.9$$

The solution is:

$$\sigma_I = 76.6 \text{ ksi}$$

$$\sigma_{II} = -.08 \text{ ksi}$$

$$\sigma_{III} = -3.3 \text{ ksi}$$

The maximum stress intensity is

$$S_n = \sigma_I - \sigma_{III} = 76.6 - (-3.3) = 79.9 \text{ ksi.}$$

The range of primary plus secondary stress intensity exceed $3S_m$.

$$S_n = 79900 > 3S_m = 3(23100) = 69300 \text{ psi.}$$

The procedure outlined in NB 3228.3 of Reference (5) is followed again:

The range of primary plus secondary stress intensity excluding thermal bending is:

$$S_n' = 5510 + 24580 = 30090 \text{ psi}$$

(Operating Membrane) + (Thermal Membrane)

$$S_a \leq \frac{1}{2}(79900) = 40000 \text{ psi}$$

$$S_n' = 30090 < 3S_m = 69300 \text{ psi.}$$

Determine the multiplication factor K_C :

Since $3S_m < S_n < 3(m)(S_m)$

$$3(23100) < 79900 < 3(3)(23100)$$

$$K_C = 1.0 + \frac{1-n}{n(m-1)} \left(\frac{S_n}{3S_m} - 1 \right)$$

where the parameters m and n are $m = 3$ $n = 0.2$ and

$$K_C = 1.0 + \frac{1-n}{0.2(3-1)} \left(\frac{79900}{69300} - 1 \right) = 1.32$$

$$S_a' = K_C(S_a) = 1.32(40000) = 52800 \text{ psi.}$$

Then, entering the design fatigue curve figure I-9-1 of Reference 5 with the required number of thermal cycles ($N=100$), gives the allowable alternating stress value:

$$S_a = 200000 \text{ psi} > 52800 \text{ psi.}$$

FIGURE 5-1

COMBINED THERMAL
STRESS DIAGRAM

6.0 FLANGES AND GUSSETS

6.1 Flanges

The flanges are designed to carry the total bending moment and the axial force by bearing against the concrete.

6.2 Gussets

The gussets are designed to carry the total torsional moment and the vertical shear force by bearing against the concrete.

6.3 Loading Distribution

The peak concrete bearing stress is derived assuming a triangular stress distribution with zero stress at the outer edge of the flange or gussets, and maximum at the nozzle. For this type of calculation an allowable bearing is taken as f'_c .

The thickness of the flanges and gussets is determined assuming a uniform concrete bearing stress distribution.

7.0 FLANGES

7.1 Concrete Bearing

The area of the flanges must be sufficient to limit the peak concrete bearing stress to the allowable value.

A triangular concrete stress distribution has been assumed (see Figure 7-1) which is proportional to the distance ($r \sin \theta$) to the neutral axis, peaking at the inside edge ($r=a$) and zero at the outside edge ($r=b$) of the flange.

Equation (7-1) expresses the assumed concrete bearing stress distribution:

$$f_c = f_{cmax} \left[\frac{b-r}{b-a} \right] \sin \theta \quad (7-1)$$

Integration over the full area of the flange gives:

$$\begin{aligned} M &= 4 \int_{r=a}^b dr \int_{\theta=0}^{\frac{\pi}{2}} f_c r^2 \sin \theta d\theta ; \quad \text{or,} \\ M &= \frac{4f_{cmax}}{(b-a)} \int_a^b r^2 (b-r) dr \int_0^{\frac{\pi}{2}} \sin^2 \theta d\theta ; \quad \text{or} \\ M &= \frac{\pi f_{cmax}}{(b-a)} \left\{ \frac{b}{3} (b^3 - a^3) - \frac{1}{4} (b^4 - a^4) \right\} ; \end{aligned}$$

from which:

$$f_{cmax} = \frac{12}{\pi} \frac{M}{[(b^2 + a^2)(b+a) - 4a^3]} \leq f'_c;$$

7.1.1 Section (3) Analysis

Loading Condition I governs:

Insertion of the bending moment $M = 120,000$ in-k, and radius $a = 24$ "; $b = 41$ " results in:

$$f_{cmax} = \left(\frac{12}{\pi} \right) \left[\frac{120000}{(41^2 + 24^2)(41+24) - 4(24)^3} \right] = 4.98 \text{ ksi} < f_c'$$

The allowable concrete bearing stress is equal to $f_c' = 5.0$ ksi.

7.2 Flange Bending and Shear

The thickness of the flanges must be sufficient to provide a flange plastic capacity equal to or greater than the maximum imposed bending moment and shear.

Uniform concrete bearing stress distribution is assumed since this results in what are considered to be upper limit flange bending moments (see Figure 7-1).

Equation (7-2) expresses the concrete stress distribution:

$$f_c = \left(\frac{3}{4} \right) \frac{M}{(b^3 - a^3)} \quad (7-2)$$

Loading Condition I governs:

$M = 120000$ in-k (See Table 3-1)

$$f_c = \left(\frac{3}{4} \right) \frac{120000}{[(41)^3 - (24)^3]} = 1.63 \text{ ksi.}$$

7.2.1 Flange Plastic Bending Analysis

The flange is supported at the nozzle and by the gussets and is free at the outer edge. When it is subject to a uniform load (f_c) of sufficient magnitude to cause yielding, the formation of a simple pattern of yield lines (shown in Figure 7-2) which will result in uniform bending moment distribution, is assumed.

The bending moments m and m' per unit length of the yield lines will reach the plastic capacity (m_y) along the full length of the yield lines:

$$m \rightarrow m' \rightarrow m_y = (\sigma_y) (S_p) = (\sigma_y) \left(\frac{1}{4}\right) (t)^2.$$

The yield lines separate the flange sector into segments I and II as indicated in Figure 7-2.

Equilibrium of Segment I requires:

$$(m+m')(9.4) = \frac{1}{6} (f_c) (9.4) (9.7)^2;$$

or

$$m = m' = \frac{1}{12} (f_c) (9.7)^2 = (7.84) (f_c); \quad (7-3)$$

Equilibrium of Segment II requires:

$$(m+m')(17.0) \cong \frac{1}{6} (f_c) (9.7) \times (6.60)^2 + \frac{1}{2} (f_c) (7.3) \left[\frac{1}{2} (6.60+8.05) \right]^2$$

or

(7-4)

$$m = m' = (2.08) (f_c) + (5.77) (f_c) = (7.85) (f_c);$$

From (7-3) and (7-4) one obtains the same value for (m):

$$m = m' = (7.85) (f_c); \quad \text{or}$$

$$\underline{m} = \underline{m'} = (7.85) (1.63) = 12.8 \text{ in-k/in.};$$

(a) Segment II Plastic Analysis.

The total bending moment in the edge of flange segment II is:

$$M = m' (b-a) \quad \text{or}$$

$$M = 12.8(17) = 217.6 \text{ in-k}$$

The plastic bending moment capacity is:

$$M_p = (.9\sigma_y) (S_p) = (.9\sigma_y) \left(\frac{1}{4}\right) (b-a) (t)^2$$

where σ_y is the nominal yield stress with a minimum value of 34.6 ksi at 200°F.

For a flange thickness $t = 1\frac{1}{2}$ in. then:

$$M_p = 0.9(34.6) (.25) (17) (1.5)^2 = 298 \text{ in-k.}$$

Triangular Bearing
Stress Distribution

Uniform Bearing
Stress Distribution

FIGURE 7-1 FLANGE BEARING

FIGURE 7-2 FLANGE PLASTIC ANALYSIS

The total shear acting along the edge of Segment II is equal to the load on that segment:

$$V = (1.63) \left\{ \frac{1}{2}(9.7)(6.6) + \frac{1}{2}(7.3)(6.6 + 8.05) \right\} = 140 \text{ k.}$$

The plastic shear capacity is:

$$V_p = (.45\sigma_y)(A_p) = (.45\sigma_y)(b-a)(t) \quad \text{or}$$

$$V_p = (.45 \cdot 34.6)(17)(1\frac{1}{2}) = 397 \text{ k.}$$

Using the shear-bending interaction Equation (5-4) derived earlier:

$$\left(\frac{M}{M_p} \right)^2 + \left(\frac{V}{V_p} \right)^2 = 1$$

results in:

$$\left(\frac{217.6}{298} \right)^2 + \left(\frac{140}{397} \right)^2 = .53 + .13 = .66 < 1.00$$

Total shear transmitted from the flange segments II to gusset for Loading Condition I is:

$$2V = 2(140.0) = 280.0 \text{ k}$$

Since the uniform concrete bearing stress (Equation 7-2) and the shear acting along the edge of Flange Segments II is directly proportional to the imposed bending moment (M) the total shear transmitted from the Flange Segments II to the gusset may, for Loading Condition II, be obtained by proportioning from Loading Condition I:

$$2V = 280 \left(\frac{34600}{120000} \right) = 80.7 \text{ k}$$

$$f_s = \frac{2V}{(b-a)t} = \frac{280}{(17)(1\frac{1}{2})} = 13.2 < .45(34.6) = 15.6 \text{ ksi}$$
$$f_s = \frac{80.7}{(17)(1\frac{1}{4})} = 3.8 \text{ ksi for Loading Condition II.}$$
$$f_v = f_s \text{ (see figure above).}$$

Segment I is less severely loaded than Segment II.

8.0 GUSSETS

The radial gusset plates carry the torsional moment and radial shear force.

8.1 Loading

Loading Condition II governs.

Shear $V = 956k$. Torque $T = 67,200$ in-k.

Maximum reaction on one gusset (see Figure 8-1):

$$F = \left(\frac{\pi}{2}\right) \left(\frac{V}{16}\right) + \frac{T}{16D}$$

or

$$F = \left(\frac{\pi}{2}\right) \left(\frac{956}{16}\right) + \left(\frac{67200}{16 \left(48 + \frac{2(17)}{4}\right)}\right) =$$

$$F = 94.0 + 74.4 = 168.4 \text{ k.}$$

8.2 Concrete Bearing

The area of the gussets must be sufficient to limit the peak concrete bearing stress to the allowable.

A triangular stress distribution has been assumed (see Figure 8-2(a)), which peaks at the inside edge at the nozzle and is zero at the top.

The peak concrete stress is:

$$f_{cmax} = \frac{6F}{(b-a)^2} \quad \text{or}$$

$$f_{cmax} = \frac{6(168.4)}{(17)^2} = 3.50 \text{ ksi} < f_c' = 5.0 \text{ ksi for Loading Condition II.}$$

8.3 Gusset Bending

The thickness of the gussets must be sufficient to provide a gusset plastic capacity equal to or greater than the maximum bending moment and shear.

Uniform concrete bearing stress distribution was assumed.
(Figure (8-2(b))).

The concrete bearing stress is:

$$f_c = \frac{F}{\frac{1}{2}(b-a)^2} = \frac{168.4}{\frac{1}{2}(17)^2} = 1.17 \text{ ksi} \quad \text{for Loading Case II.}$$

Gusset plastic analysis yield lines are shown in Figure(8-2(c)).
Equilibrium of Segment I or II requires:

$$\left(\frac{F}{2}\right)\left(\frac{1}{3}\right)\left(\frac{b-a}{2}\right) = m\left(\frac{b-a}{2}\right) + m'(b-a)$$

or

$$m = m' = \frac{F}{18} = \frac{168.4}{18} = 9.36 \text{ in-k/in.}$$

The bending moment along the edge at the nozzle is:

$$M = (m)(b-a) = (9.36)(17) = 160 \text{ in-k.}$$

The plastic moment capacity of the gusset is:

$$M_p = (.9\sigma_{yp})(S_p)$$

$$M_p = 0.9(34.6)\left(\frac{1}{4}\right)(17)(1.25)^2 = 207 \text{ in-k.}$$

The tangential shear along the nozzle is equal to the load on Segment I:

$$V_R = F = 168.4 \text{ k.}$$

The longitudinal shear along the nozzle is:

$$V_L = 80.7k \text{ (See paragraph 7.2.1(a) Page (7-6).)}$$

The resulting shear is:

$$V = \sqrt{(V_R)^2 + (V_L)^2} = \sqrt{168.4^2 + 80.7^2} = 187k.$$

The plastic shear capacity of the gusset along the edge is:

$$V_p = (.45\sigma_y)(A_p)$$

$$V_p = (.45\sigma_y)(b-a)(t)$$

or

$$V_p = .45(34.6)(17)(1.25) = 331 k.$$

The shear bending interaction equation derived earlier must result in:

$$\left(\frac{M}{M_p}\right)^2 + \left(\frac{V}{V_p}\right)^2 \leq 1,$$

$$\left(\frac{160}{207}\right)^2 + \left(\frac{187}{331}\right)^2 = 0.60 + 0.32 = 0.92 < 1.00.$$

The combined bending and shear is within the plastic capacity of the gusset.

FIGURE 8.1 GUSSETS (REACTIONS $\vec{F}$ SHOWN FOR LOADING CONDITION II.)

Concrete Bearing
Triangular Stress
Distribution Assumed

(a)

Gusset Bending
Uniform Stress
Distribution Assumed

(b)

Plastic Analysis
Yield Lines Assumed

(c)

FIGURE 8-2 GUSSET STRESS DISTRIBUTION

APPENDIX A

DERIVATION OF EQUILIBRIUM EQUATIONS

FOR A CIRCULAR CYLINDRICAL SHELL

IN THE PLASTIC RANGE:

For the main steam pipe, and/or penetration nozzle being subject to general load: axial force $\bar{N}$, Shear $\bar{V}$, Bending $\bar{M}$, and Torsion $\bar{T}$, a stress distribution as shown on Figure A-1 will be assumed to exist when the shell is stressed to its "Plastic capacity".

FIGURE A-1 PLASTIC STRESS DISTRIBUTION IN CIRCULAR CYLINDRICAL SHELL SUBJECT TO AXIAL LOAD $\bar{N}$, SHEAR $\bar{V}$, BENDING $\bar{M}$ AND TORSION $\bar{T}$.

The equilibrium requirements of the shell are expressed by the following equations :

$$\bar{N} = 4 \int_{\theta}^{\phi} (\sigma_z) (Rt) (d\omega); \quad (A-1)$$

$$\bar{V} = 4 \int_{\theta}^{\phi} (\tau_{xy}) (Rt) (\cos \omega) (d\omega); \quad (A-2)$$

$$\bar{M} = 4 \int_{\frac{\pi}{2}}^{\frac{\pi}{2}} (\sigma_z) (R^2 t) (\sin \omega) (d\omega) \quad (A-3)$$

$$\bar{T} = 4 \int_{\theta}^{\phi} (\tau_{xy}) (R^2 t) (d\omega); \quad (A-4)$$

which result in:

$$N = (A_p) (\phi) (\sigma_z) \quad (A-5)$$

$$V = (A_p) (\sin \theta) (\tau_{xy}); \quad (A-6)$$

$$M = (S_p) (\cos \phi) (\sigma_z); \quad (A-7)$$

$$T = (S_p) \left(\frac{\pi}{2} - \theta \right) (\tau_{xy}); \quad (A-8)$$

where:

(A_p) the plastic shear area is 2(t)(d);

(S_p) the plastic section modulus is (t)(d)²;

(φ) The angle to the plane of zero axial stress is determined from Equations (A-5) and (A-7):

$$\frac{\cos(\phi)}{(\phi)} = 2 \left(\frac{M}{Nd} \right); \quad (A-9)$$

(θ) The angle to the plane of zero shear stress is determined from Equations (A-6) and (A-8):

$$\frac{\sin(\theta)}{\frac{\pi}{2} - \theta} = \frac{1}{2} \left(\frac{V}{T} \right) (d); \quad (A-10)$$

The plastic shear area (A_p) is obtained from (A-2) for integration over the full cross-section:

$$A_p = \frac{|\bar{V}|}{(\tau_{xy})} = 4R(t) = 2(t)(d)$$

The plastic section modulus (S_p) is obtained from (A-3) for integration over the full cross-section:

$$S_p = \frac{|\bar{M}|}{(\sigma_z)} = 4R^2(t) = (t)(d)^2$$

REFERENCES

- (1) Bechtel Topical Report, BC-TOP-5, Revision 1, "Prestressed Concrete Nuclear Reactor Containment Structures", December, 1972.
- (2) Bechtel Topical Report, BN-TOP-2, "Design for Pipe Rupture Effects", September, 1972.
- (3) S. Timoshenko and J. N. Goodier, "Theory of Elasticity", Second Edition, McGraw-Hill, N.Y., 1951.
- (4) Bechtel Program ME 620, "Heat Conduction".

The finite element technique coupled with a step-by-step time integration procedure is used to determine the temperature distribution within plane or axisymmetric body as a function of time when subjected to step-function temperature or heat flux inputs.

- (5) ASME Section III - "ASME Boiler and Pressure Vessel Code, Section III, Division 1, 1971, Nuclear Power Plant Components".
- (6) Bechtel Program ME 200, "Shells of Revolution Stress Analysis Program".

This program provides the solution for axisymmetric shells with axisymmetric loading. The structure is modeled as a series of interconnected cone frustrums. By using a classical solution for each cone frustrum and establishing the compatibility of the interconnected cones the solution is obtained.

PART III

POLAR CRANE BRACKET

10.0	Tension Member	10
11.0	Compression Member	12
11.1	Check for Punch-Out	13
12.0	Baseplate	15
12.1	Maximum Bending Moment	15
12.2	Maximum Shear	16
References		

CONTENTS

1.0	General	1
2.0	Summary	1
3.0	Materials	1
4.0	Design Method	1
5.0	Loading	2
5.1	Loading at Crane Girder Centerline	2
5.2	Maximum Load on One Web	3
5.3	Loading at Face of Wall	3
6.0	Supports	3
7.0	Initial Design of Support Bracket Box Beam	3
7.1	Width of Box Beam	4
7.2	Depth of Box Beam	4
7.3	Flange Thickness	4
7.4	Web Thickness	4
7.5	Web Stiffeners	5
7.6	Combined Shear and Tension Limitations	6
8.0	Final Design Analysis of Crane Support Bracket	6
8.1	Maximum Flange Tensile Stress	7
8.2	Maximum Shear Stress in Flanges and Webs	7
9.0	Liner Plate and Ribs	8

LIST OF FIGURES

Fig. No

1	Polar Crane Support Bracket
2	Dimensions, Loading and Coordinates
3	Liner Reinforcing Plate and Ribs
4	Tension Member
5	"Punch-Out"
6	Baseplate

FIGURE 1 POLAR CRANE SUPPORT BRACKET

5.0 LOADING

The loading shown on Figure 2 is a typical loading for a polar crane support bracket including impact.

5.1 Loading at Crane Girder Centerline

Vertical load	$F_z = 612 \text{ k}$, $F_z = 392 \text{ k}$ (Maximum vertical load into single web)
Radial Load	$F_x = \pm 71.8 \text{ k}$
Tangential Load	$F_y = \pm 70.8 \text{ k}$
Torque	$T_x = \pm 486 \text{ in-k}$

It is assumed that these forces and moments act simultaneously.

The torque is caused by the vertical load:

$$F_z = F_{z1} + F_{z2} = 445 + 167 = 612 \text{ k};$$

$$T_x = (F_{z1} - F_{z2}) (e) = (445 - 167) (1.75) = 486 \text{ in-k}.$$

1.0 GENERAL

The following illustrates the design of a polar crane bracket in accordance with the criteria given in Reference 1, BC-TOP-5. The polar crane is supported on a system of girders which are supported by a series of brackets that are attached to the containment shell.

2.0 SUMMARY

The polar crane bracket was designed by the techniques shown in this example. For orientation the final design configuration is shown on Figure 1.

3.0 MATERIALS

Crane Support Bracket	A 36	$F_y = 36 \text{ ksi}$
Thickened Liner Plate	A 516	$F_y = 38 \text{ ksi}$
Containment Wall Concrete		
Specified Compressive Strength		$f'_c = 5.0 \text{ ksi}$

4.0 DESIGN METHOD

Working Stress Design

In accordance with Section 1.5 of Reference 2 and Section 8.10 of Reference 3 and Bechtel Topical Report BC-TOP-5 Reference 1.

FIGURE 2 DIMENSIONS, LOADING AND COORDINATES

5.2 Maximum Load on One Web at Crane Girders Centerline

Vertical Load $F_z = 392 \text{ k}$

(This case governs web crippling. See Section 5.1)

5.3 Loading at Face of Wall

Vertical Load $F_{z_{web}} = 612 \text{ k}$

Radial Load $F_x = 71.8 \text{ k}$

Tangential Load $F_y = 70.8 \text{ k}$

Torque $T_x = 486 + 70.8 (25) = 2,250 \text{ in-k}$

Bending Moment $M_y = 612(26) + 71.8(25) = 17,700 \text{ in-k}$

Bending Moment $M_z = 70.8(26) = 1,840 \text{ in-k.}$

6.0 SUPPORTS

The box beam bracket is cantilevered from the wall. Lateral support of compression flange at wall only.

7.0 INITIAL DESIGN OF SUPPORT BRACKET BOX BEAM

SECTION AT
FACE OF WALL

SECTION AT CENTERLINE
CRANE GIRDER

7.1 Width of Box Beam

The width of the box beam is set equal to $b_F = 24"$ in order to accomodate the crane girders.

7.2 Depth of Box Beam

The depth at face of wall is assumed $d = 50"$ (distance to C. G. of flanges). The depth at the crane beam centerline is assumed $d = 36"$ (distance between C.G. flanges).

7.3 Flange Thickness

The flange thickness is determined from the loading at face of wall

$$t_F = \frac{M_y}{(d)(b_F)(0.6F_y)} = \frac{17700}{(50)(24)(0.6 \times 36.0)} = 0.68 \text{ in.}$$

Use $t_F = 1"$.

Projecting flange width to thickness ratio:

$$\frac{b_P}{t_F} = \frac{6}{1} = 6 < \frac{95}{\sqrt{F_y}} = 15.8; \quad (\text{AISC 1.10.3})$$

(AISC 1.9.1.2)

$$\frac{b_F'}{t_F} = \frac{12}{1} = 12 < \frac{239}{\sqrt{F_y}} = 40; \quad (\text{AISC 1.10.3})$$

(AISC 1.9.2.2)

7.4 Web Thickness

The web thickness is determined from the loading at centerline of crane girders.

$$t_w = \frac{V}{(d)(2)(0.4F_y)} = \frac{612}{(36)(2)(0.4 \times 36)} = 0.59 \text{ in.}$$

use $t_w = 3/4"$

For A-36 steel, a limiting value of $\frac{d}{t_w}$ with no reduction

in flange stress equals $\frac{d}{t_w} = \frac{760}{\sqrt{F_b}} = \frac{760}{\sqrt{22}} = 162$ (AISC 1.10.6)

$$\frac{d}{t_w} = \frac{50}{3/4} = 67 < 162;$$

7.5 Web Stiffeners

The need for web stiffeners is checked:

7.5.1 Intermediate web stiffeners:

The allowable shear stress in plate girders for 36 ksi specified yield stress steel when stiffener spacing (a) to web depth (d) is more than 3 and depth to web thickness ratio (d/t_w) is less than 70, is $F_v = 13.0$ ksi. (AISC Table 3-36, Appendix A 5.95.)

$$f_v \leq \frac{F_z}{2dt_w} = \frac{612}{2(36)(3/4)} = 11.3 < F_v = 13.0 \text{ ksi};$$

No intermediate web stiffeners are required.

7.5.2 Web Stiffeners Under Crane Girder

Check for web crippling in accordance with AISC 1.10.10.1.

$$f_{sw} = \frac{R}{t_w(N+2k)}$$

$$f_{sw} = \frac{392}{(3/4)(24+2(1))} = 20.1 < 27.0 \text{ ksi.}$$

Web stiffeners are not required.

The compressive stress (f_{sw}) in the web must not exceed:

$$f_s = (5.5) \frac{10,000}{\left(\frac{d}{t_w}\right)^2} \quad \text{in accordance with AISC 1.10.10.2.}$$

$$\frac{a}{d} > 3.0; \quad \frac{d}{t_w} \leq \frac{36}{3/4} = 48.$$

$$f_s = 5.5 \frac{10,000}{(48)^2} = 23.9 \text{ ksi.}$$

$$f_{sw} = 20.1 \text{ ksi} < f_s = 23.9 \text{ ksi.}$$

7.6 Combined Shear and Tension Stress Limitations

$$\text{use } F_b = 29.7 - 13.5 \left(\frac{f_v}{F_v} \right); \quad (\text{AISC 1.10.7})$$

Section at Wall:

$$f_v = \frac{F_z}{2dt_w} = \frac{612}{2 \times (50) (3/4)} = 8.2 \text{ ksi;}$$

$$F_v = 13.0 \text{ ksi;} \quad (\text{Reference 2, Appendix A.5.95 Table 3-36})$$

$$F_b = 29.7 - 13.5 \left(\frac{8.2}{13.0} \right) = \underline{\underline{21.2 \text{ ksi}}}$$

where F_b is the allowable tensile bending stress in the web.

8.0 FINAL DESIGN ANALYSIS OF CRANE SUPPORT BRACKET

CROSS-SECTION AT FACE OF WALL

8.1 Maximum Flange Tensile or Compressive Stress

$$f_{sx} = \frac{F_x}{A_x} + \frac{M_y}{S_y} + \frac{M_z}{S_z}$$

$$f_{sx} = \frac{71.8}{2(24) + 2(3/4)(49)} + \frac{17700(25.5)}{(1/12)(2)(3/4)(49)^3 + (24)(2)(25)^2} + \frac{(1840)(12)}{(1/12)(2)(24)^3 + (3/4)(49)(2)(6)^2} =$$

$$f_{sx} = 0.591 + 10.10 + 4.46 = 15.2 \text{ ksi} < F_b = 21.2 \text{ ksi}$$

8.2 Maximum Shear Stress in Flanges and Webs:

$$\text{Flange: } v_{yz} = \frac{T_x}{S_x} + \frac{F_y}{A_y} \quad (\text{Torsional shear derived from Reference 4, paragraph 51, page 204})$$

$$\text{Web: } v_{yz} = \frac{T_x}{S_x} + \frac{F_z}{A_z}$$

where:

Flange:

$$S_x = b_F' (b_F' t_w + d t_F);$$

Web:

$$S_x = d (b_F' + d t_F);$$

Flange:

$$S_x = (12) [(12)(3/4) + (49)(1)] = 696 \text{ in}^3;$$

Web:

$$S_x = (49)[(12)(3/4) + (49)(1)] = 2842 \text{ in}^3;$$

Flange:

$$V_{yz} = \frac{486 + 70.8(25.5)}{696} + \frac{70.8}{2(1)(24)} = 3.29 + 1.48 = 4.77 \text{ ksi} < F_v$$

Web:

$$V_{yz} = \frac{486 + 70.8(25.5)}{2842} + \frac{612}{2(3/4)(49)} = .81 + 8.33 = 9.14 \text{ ksi} < F_v$$

(Reference 2, Appendix 5.95 Table 3-36) $F_v = 13.0 \text{ ksi}$

9.0 LINER PLATE AND RIBS

See Figure 3 for reactions.

The vertical shear $V = F_z = 612 \text{ k}$ is transferred from the webs of the bracket box beam to the liner reinforcing plate and ribs which in turn transfer it to the baseplate and into the concrete.

The maximum compressive stress in the liner reinforcing plate and ribs is:

$$f_s = \frac{V}{A} = \frac{612}{24(1) + 8(3/4)(2)} = \frac{612}{36} = 17.0 \text{ ksi}$$

$$< .60F_y = .60 \times 36.0 = 21.6 \text{ ksi},$$

(See Section 1.5.1.3.4, Reference 2.)

The torsional moment (see Section 5.3) is transferred to the wall by the ribs bearing against the concrete.

The maximum concrete bearing stress is low:

$$f_c = \frac{T_x}{\frac{1}{6} b(d)^2} = \frac{2250 \times 6}{8(50)^2} = .675 \text{ ksi.}$$

FIGURE 3
LINER REINFORCING PLATE AND RIBS

10.0 TENSION MEMBER (See Figure 4)

The loading acting at the face of the wall is shown in Section 5.3. Since the shear-reaction is assumed to be located at the center of the baseplate less than $e = 10"$ behind the face of the wall liner, the maximum tension in the tension member is:

$$T = \frac{M_y + F_z(e)}{d} + F_x \quad \text{or}$$

$$T = \frac{17,700 + 612(10)}{50} + 71.8 = 547.8 \text{ k}$$

Allowable Concrete Bearing Stress:

$$f_c = 0.50(\phi)(f'_c) = 0.50(0.70)(5.0) = 1.75 \text{ ksi};$$

(Reference 3, Sections 8.10.3 and 9.2.1.4).

Required Bearing Area:

$$A = \frac{T}{f_c} = \frac{547.8}{1.75} = 312 \text{ in}^2$$

Try 14 x t x 2'-0" Plate.

$$\text{Net area } A = 24(14-1) = 312 \text{ in}^2$$

Bending Moment:

$$m = 1/2(f_c)(a)^2 = 1/2(1.75)(6.5)^2 = 37.0 \text{ in -k/in}$$

where a is the cantilever span of the bearing plate.

Required Thickness of Bearing Plate:

$$t = \sqrt{\frac{6(m)}{f_s}} = \sqrt{\frac{6(37.0)}{0.75(36.0)}} = 2.86"$$

Use $2 \frac{7}{8}"$ thickness. (Reference 2, Section 1.5.1.4.3)

Shear checked:

$$f_v = \frac{547.8}{(2)(24)(2\frac{7}{8})} = 3.97 \text{ ksi} < F_v;$$

FIGURE 4 TENSION MEMBER

11.0 COMPRESSION MEMBER

$$\text{Maximum compression } C = \frac{23800}{50} = 476 \text{ k}$$

Try 12 x t x 2'-0" plate.

$$\text{Bearing Area : } A = 12(24) = 288 \text{ in}^2$$

Bearing Stress:

$$f_c = \frac{476}{288} = 1.65 < 1.75 \text{ ksi}$$

Bending Moment:

$$m = 1/2(f_c)(a)^2 = 1/2(1.65)\left(\frac{12-2.62}{2}\right)^2 = 18.1 \text{ in -k/in}$$

Required Thickness:

$$t = \sqrt{\frac{6(m)}{(f_s)}} = \sqrt{\frac{6(18.1)}{0.75(36.0)}} \approx 2.00"$$

Use 2" plate.

11.1 Check for Punch-Out

(See Figure 5)

11.1.1 Tension Member $V = 548 \text{ k}$.

Nominal shear stress in concrete for two-way action is:

$$v = \frac{V}{b_o d}$$

according to Sections 11.10.1(b), 11.10.2 and 11.10.3 of Reference 3 where b_o is the periphery of the critical shear section;

$$b_o = 2 [(14+d) + (24+d)] = 2 (41+51) = 184 \text{ in.}$$

and d is the embedment length of the tension member

$d = 27 \text{ in.}$ See Figure 1.

Therefore:

$$v = \frac{V}{b_o d} = \frac{548,000}{(184)(27)} = 115 \text{ psi} < (0.5) 4 \sqrt{f'_c}$$
$$= 2 \sqrt{5000} = 141 \text{ psi}$$

according to Sections 8.10.3 and 11.10.3 of Reference 3.

11.1.2 Compression Member $V = 476 \text{ k.}$

The nominal shear stress in the concrete is:

$$v = \frac{V}{b_o d}$$

where the periphery of the critical section b_o is 180 in. and d is the concrete depth or "cover" which is 2'-3" (see Figure 5), so that:

$$v = \frac{476}{(180)(27)} = 98 \text{ psi} < 141 \text{ psi.}$$

12.0 BASEPLATE

The vertical shear $V = 612$ k is transferred from the baseplate into the concrete underneath by bearing. The baseplate is supported on four sides and a uniform bearing stress distribution has been assumed. The allowable concrete bearing stress is:

$$f_c = 1.75 \text{ ksi.}$$

(Reference 3, Sections 8.10.3 and 9.2.1.4)

The required bearing area is:

$$A = \frac{F_z}{f_c} = \frac{612}{1.75} = 350 \text{ in}^2.$$

Use a 15" x t x 2'-0" plate with a net area

$$A = 24(15) = 360 \text{ in}^2 \text{ so that}$$

$$f_c = 1.75 \frac{350}{360} = 1.70 \text{ ksi.}$$

12.1 Maximum Bending Moment

If the center segment of the base plate is conservatively considered as simply supported on all four edges and the end segments are cantilevered, the maximum bending moments are approximately:

$$m = 1/2 f_c c^2 = 1/2 \times 1.70 \times 6.0^2 = 30.6 \text{ in-k/in (Reference 5, page 133).}$$

$$m = 0.07 f_c a^2 = 0.07 \times 1.70 \times 12^2 = 17.2 \text{ in-k/in}$$

$$\frac{b}{a} = \frac{15}{12} = 1.25$$

Required plate thickness:

$$t = \sqrt{\frac{6m}{f_s}} = \sqrt{\frac{6 \times 30.6}{0.75 \times 36.0}} = 2.60 \text{ in.}$$

Use $2\frac{5}{8}$ " thick plate

12.2 Maximum Shear

$$v = \frac{F_z}{(b)(t)} = \frac{612}{24(2.625)} = 9.7 \text{ ksi}$$

FIGURE 6 BASEPLATE

REFERENCES

- (1) Bechtel Topical Report BC-TOP-5, Revision 1, "Prestressed Concrete Nuclear Reactor Containment Structures", December, 1972.
- (2) American Institute of Steel Construction Specification - Steel Construction Manual, 7th Edition.
- (3) American Concrete Institute (ACI) Standard Building Code Requirements for Reinforced Concrete (ACI 318-71), Reported by Committee 318.
- (4) R. J. Roark, "Formulas for Stress and Strain", Fourth Edition, 1954, McGraw-Hill, N. Y.
- (5) Timoshenko, "Plates and Shells", First Edition, 1940, McGraw-Hill, N. Y.