ATTACHMENT 1

UNITED STATES OF AMERICA
NUCLEAR REGULATORY COMMISSION

BEFORE THE ATOMIC SAFETY AND LICENSING BOARD

In the Matter of	)	
	)	
TEXAS UTILITIES ELECTRIC COMPANY, *et al*.	)	Docket Nos. 50-445 50-446
	)	
(Comanche Peak Steam Electric Station, Units 1 and 2)	)	(Application for Operating License)

AFFIDAVIT OF ROBERT C. IOTTI AND JOHN C. FINNERAN, JR. REGARDING CINCHING DOWN OF U-BOLTS

We, Robert C. Iotti and John C. Finneran, Jr., being first duly sworn, hereby depose and state as follows:[1]

(Iotti) I am employed by Ebasco Services, Inc. as Chief Engineer of Applied Physics. In this position, I am responsible for directing analytical and design work in diverse technical areas, including analyses of the response of piping and support systems for dynamic events, including earthquakes. I have been engaged by TUECO to coordinate and oversee the technical activities performed to respond to the Board's Memorandum and

1 Except as otherwise indicated, each Affiant attests to all parts of this affidavit.

Order of December 28, 1983. A statement of my educational and professional qualifications is attached to Applicants' letter of May 16, 1984 to the Licensing Board.

(Finneran) I am the Pipe Support Engineer for the Pipe Support Engineering Group at Comanche Peak Steam Electric Station. In this position, I oversee the design work of all pipe design organizations for Comanche Peak. I have previously provided testimony in this proceeding. A statement of my professional and educational qualifications was received into evidence as Applicants' Exhibit 142B.

Q. What is the purpose of this Affidavit?

A. The purpose of this Affidavit is to respond to CASE's concerns regarding the practice of cinching down U-bolts. These concerns have been summarized by the Board and can be categorized into the following four areas:

> 1. The acceptability of cinching down U-bolts as a resolution of the potential instability of non-rigid U-bolt supports (see the December 28, 1983 Memorandum and Order at 27 and 33, and the February 8, 1984 Memorandum and Order (Reconsideration) at 20).
>
> 2. The use of SA-307 (or SA-36) steel in U-bolts which are cinched-down considering the ASME Code prohibition against the use of such material in friction type connections (see December 28, 1983 Memorandum and Order at 28 and 33, and February 8, 1984 Memorandum and Order (Reconsideration) at 22-4). A fair restatement of this second concern is that there needs to be reasonable assurance that adequate clamping force can be produced and

maintained by the U-bolt connection to prevent rotation of the U-bolt cross piece assembly around the pipe.

3. The forces and stresses that are induced in the U-bolt itself (see December 28, 1983 Memorandum and Order at 33-41, and February 8, 1984 Memorandum and Order (Reconsideration) at 24-5).

4. The local (and global, if any) stresses induced in the pipe by the cinching down practice (see December 28, 1983 Memorandum and Order at 33-41, and February 8, 1984 Memorandum and Order (Reconsideration) at 25-6).

Q. In overview fashion, how have you responded to these concerns?

A. To address each of these concerns, Applicants committed to provide the following (see Applicants' Plan to Respond to Memorandum and Order (Quality Assurance for Design) at 5-6 (items 3, 4 and 5)):

"3. Provide evidence that the use of U-bolt cinching is appropriate to eliminate potential local instability without introducing adverse effects in the piping and the U-bolt itself."

"4. Provide evidence that there are no adverse long-term effects from U-bolts caused by heat-up and cooldown and related friction on the pipe."

"5. Provide evidence of the acceptability of stresses on pipes caused by thermal expansion in local areas around cinched U-bolts."

The primary method used to obtain this information is testing. However, Applicants have also performed finite element analyses[2] of the tested configurations. The reason for also having finite element analyses performed for the tested configurations are two-fold:

1. Data obtained by tests are limited to locations where strain gauges are placed. A tool is needed to obtain information at other locations and to interpret the data obtained by test. This tool is the finite element model of the tested configuration, which is correlated to the test data at the locations where data are directly available.

2. A model is needed to predict (with good confidence) the behavior of configurations which are different from those being tested, i.e., different pipe size, U-bolt size, preload, etc. The finite element model, once it is verified against the test results, can be used for this purpose. Moreover, it can also be used to verify the adequacy of other, more simplistic models, which are developed to assess U-bolt loads, pipe stresses, and stability questions.

Q. Before describing the test program, its results, and the results of the finite element analyses, are there any items of apparent misunderstanding that you wish to clarify?

A. Yes. We would like to clarify an apparent misunderstanding by the Board. On page 28 of its December 28, 1983 Memorandum and Order, the Board states:

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2 These finite element analyses employ idealized but realistic models of the piping and U-bolt cross piece assemblies, subdivide the models into many "finite elements" and theoretically predict the states of stress and strains in each of the elements.

> "The fact that this material [SA-307[3]] was incorporated into the U-bolts is not surprising, since they were not initially designed to be cinched down and to develop friction forces to hold the pipe."

The Board is apparently under the impression that no U-bolts at CPSES were initially intended to be cinched down. This is not the case. A significant number of U-bolt supports at CPSES were always intended to be cinched down. On only a relatively small number (less than 15) was the initial design changed such that U-bolts were cinched down because of potential pipe support instability. It should be noted that there are other U-bolt supports at CPSES which are not cinched down, e.g., U-bolts on rigid frames used as one- or two-way supports.

For the Board's information, Table 1 provides a partial list of the cinched-down U-bolts at CPSES. Considering normal, upset and emergency loads, the ten highest loaded U-bolt supports for each pipe size are given in this table, except for the smaller and the larger pipe sizes which have less than 10 such supports.

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3 Even though the Board refers to SA-307 material, the designation of the U-bolt material is SA-36. Applicants recognize that the material is the same in any case, with A-307 being the designation employed for headed bolts.

There is another item which we wish to clarify with the Board. This item deals directly with the second concern, i.e., the use of SA-36 U-bolts where SA-307 bolts (similar to SA-36) are prohibited by ASME Code Table XVII-2641.1-1 Note 1. We do not take issue with the fact that the material employed, SA-36, is known to relax[4] under applied thermo-mechanical loads. We do, however, take issue with the Board's apparent characterization of the U-bolt/cross piece connection as a friction connection, and therefore, with the Board's implied conclusion as to the applicability of Note 1 of ASME Table XVII-2461.1-1 (quoted below) to this connection.

> "Friction type connections loaded in shear are not permitted. The amount of clamping force developed by SA-307 is unpredictable and generally insufficient to prevent complete slippage."

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4 Relaxation is here denoted as a characteristic of certain materials which when stressed to certain levels will not maintain that level, but will "relax" to a lower stress level. For instance, a bar loaded to an initial stress of 40,000 psi and then held at constant strain and temperature may after a time period have a remaining stress of only 30,000 psi. This time dependent stress reduction is called stress relaxation. The total strain remains fixed, but a part of the elastic strain is replaced with inelastic strain. It should be noted that stress ralaxation stops after a material reaches a certain level of stress, e.g., for material such as SA-36 this level is approximately 1/2 of the yield stress.

This note is clearly intended for a friction type connection in which the load, as transferred in shear, is reacted by the friction between the two surfaces bolted together. In the case of the U-bolt (as shown in Figure 1), the load is intended to be carried by the bolt in tension or by the cross piece in compression, and not in shear. Further, slippage between the U-bolt nut and the surface of the cross piece (either plate or built-up tube steel) as would occur if F_s exceeds $2f + \mu F_p$ (<u>See</u> Figure 1) is perfectly tolerable provided there is sufficient tension in the U-bolt to maintain contact between the pipe and the U-bolt and the pipe and the cross piece, which is needed to keep the U-bolt/cross piece assembly from rotating (slipping) about the pipe. Significantly, the U-bolt/cross piece connection is <u>not</u> a friction type connection, and is not intended to be loaded in shear. Obviously, it could be loaded in shear under U-bolt lateral or axial[5] loads, but in this instance it is inconsequential whether it acts as a friction or a bearing connection. Accordingly, we do not see the direct relevance of the ASME Note (quoted above) to the U-bolt "clamp" configuration used at Comanche Peak.

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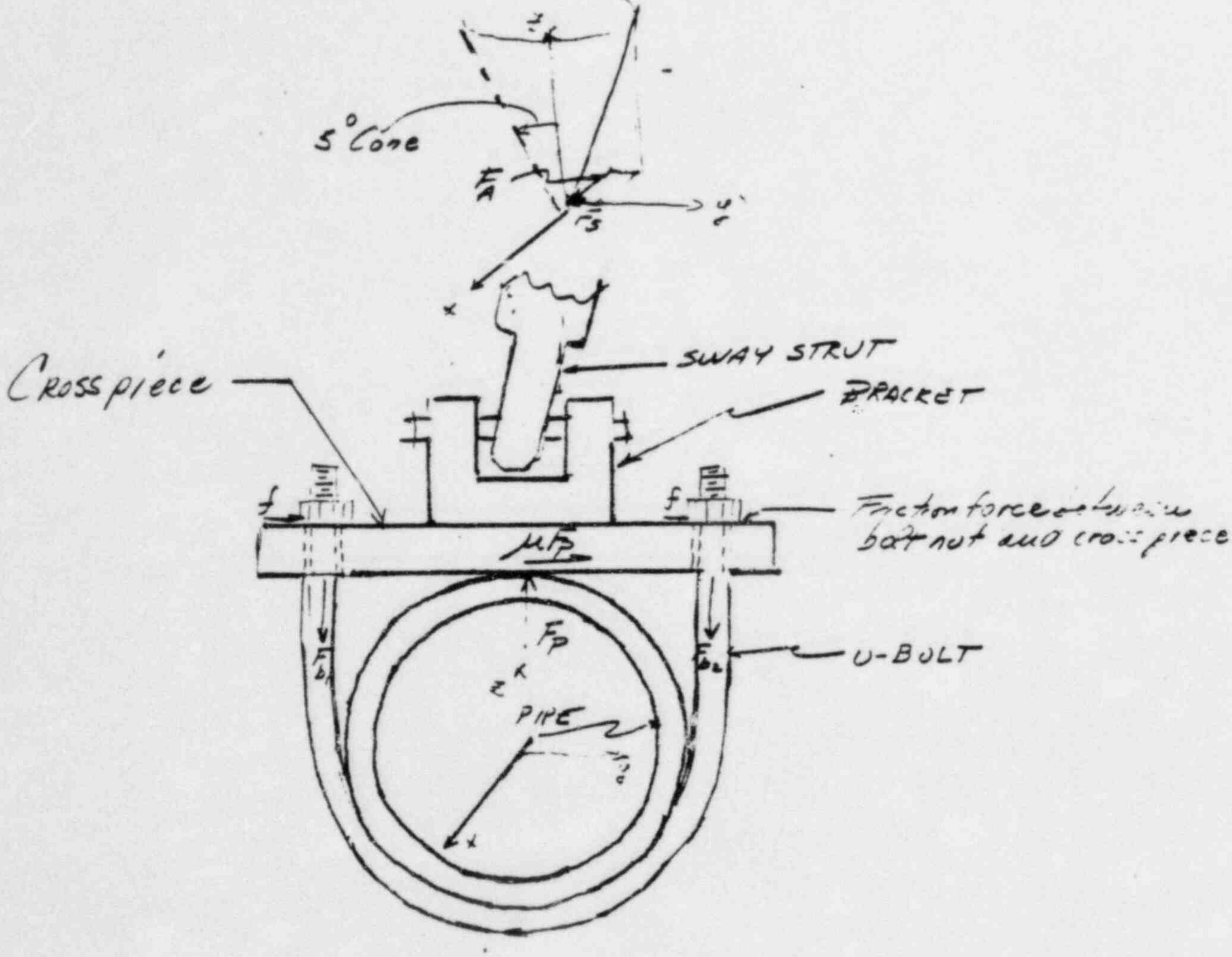
5 Lateral direction is defined as the direction normal (perpendicular) to the pipe axis and parallel to the plane of the U-bolt. Axial direction is the direction of the pipe axis.



F_{b_1}, F_{b_2} = Tension in U-bolt legs

F_p = Interface force between pipe and cross piece

F = Externally applied load

F_N = Normal component of F

F_S = Lateral component of F

F_A = Axial Component of F

FIGURE 1

We disagree that the use of SA-36 threaded rods is in contradiction to the ASME Code, and hence that their use is a design error. As we have already stated, we do agree that the relaxation[6] characteristics of this material are a concern only to the extent that a sufficient clamping force must be maintained to insure contact between pipe and U-bolt/cross piece. The test program specifically addresses this question along with others.

Q. Describe the objectives of the testing program.

A. The test program had seven overall objectives:

1. To determine the relationship between applied torque and the tension developed in the U-bolt;

2. To determine the force required to initiate slip between the U-bolt or cross piece and the pipe as a function of preload (applied torque);

3. To determine the load (and stresses) induced in the pipe and the cross piece as a function of preload;

4. To determine the additional loads (and stresses) induced in the pipe, the cross piece and the U-bolt by differential thermal expansion of the pipe with respect to the U-bolt/cross piece, and internal pipe pressure;

5. To determine the additional load (and stresses) induced in the pipe, U-bolt and cross piece by application of mechanical loads, which

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6 Relaxation has been previously defined (see note 4). Here, however, Applicants wish to inform the Board that stress relaxation does not continue but ceases after the material reaches a certain level of stress. For a material like SA-36, the level of stress at which stress relaxation ceases is in the range of one half of the yield stress.

are applied in both directions of intended constraint. This is done to verify any preloading effects in the U-bolt/cross piece assembly;

6. To determine method and extent of U-bolt material stress relaxation (i.e., loss of preload) under applied mechanical load, thermal cycling and normal plant vibration, to verify whether for the expected torque levels (preload), material relaxation would result in sufficient loss of preload to impair the capability of the U-bolt assembly to function as intended; and

7. To determine whether the U-bolt assembly would be stable under a seismic event.

Q. In that the objectives of the tests rely on the preload (torque applied to the bolts) in the field, how did you determine the range of preloads to use?

A. Until now, the Board's sole information regarding the preload level that would exist in the field was based on the Brown and Root Design Change Notice No. 1, dated October 8, 1982, to Construction Procedure No. 35-1195-CPM 9.10 Rev. 8, which states:

> "When U-bolts are specified on the design document as not having any clearances, the U-bolt shall be snug tight so that the U-bolt cannot be moved by hand . . . . Snug tight is defined as the tightness attained by a few impacts on an impact wrench or the full effort of a man using an ordinary spud wrench."

This process has been interpreted by Mr. Doyle as resulting in approximately 800 in-lbs. of torque (May 4, 1983 Surrebuttal Testimony of Mr. J. Doyle, at 11-12). Mr. Doyle

made no attempt to quantify whether this value of torque is applicable to all pipe sizes and all pipe schedules. Applicants have done so.

To determine the range of torques which exists in the field, Applicants inspected the torque of a randomly selected representative sample of cinched down U-bolt supports. The results of this sampling are summarized in Table 2. This data was used to determine the range of torques to be applied to each of the test specimens. From the data, Applicants established that for the four inch pipe, tests should be conducted with preload varying from 5 to 60 ft-lbs. Similarly, a torque range of 10 to 100 ft-lbs. was established for the 10-inch pipe tests, and a 20 to 240 ft-lb. range was established for the 32-inch pipe tests. In all cases, the upper value equals or exceeds the torques measured in the field for that particular pipe size. The 240 ft-lbs. was selected for the 32-inch specimen, even though it is considerably higher than the maximum value measured in the field for similar diameter pipes. However, since the torques noted in the field were progressively higher on larger pipe sizes, this was viewed as a conservative approach and reasonable upper bound.

Q. Is it possible that there might be considerably higher torques applied to U-bolts in the plant than those which you have described and were used in the tests?

A. We consider that this likelihood is very remote. For the smaller lines, as is discussed later in this Affidavit, initially high torque values that would stress the U-bolt above 1/2 of the yield stress would have decreased as the material relaxed to a state where the stresses are about 1/2 of the yield.[7] For the small lines (below 10 inches), then, the upper value of the torque that should be present in the field is that which corresponds to a U-bolt load which is stressed to about 1/2 yield.

The maximum torque achieved in the torquing process for 10-inch pipe would place the U-bolt at about the 1/2 yield stress. Above the 10 inch size, we would not expect the torquing process itself to result in values significantly above 1/2 yield stress. Accordingly, while relaxation may, in rare instances, be a factor in larger pipes, generally torque will not be so high that significant relaxation will occur.

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7 The value of 1/2 of yield stress is not a precise figure and should be interpreted to denote a level of stress in the general neighborhood of 1/2 of the yield stress, at which level of stress material stress relaxation stops.

Hence, the effective torque for all pipe sizes will be the lesser of the value corresponding to a U-bolt stress of half-yield or the value achieved by a man with a torque wrench or impact wrench. Applicants believe that 240 ft-lbs. is a reasonable upper bound for the torque achievable by a man with a torque or impact wrench. On occasion, higher torques may be measured because painting introduces a shear resistance to torquing the U-bolt nut. However, these higher torques would not correspond to the actual preload in the U-bolt, which was achieved by torquing with the clean (not painted) threads.

Q. Would you describe the tests that have been conducted and summarize the results?

A. A total of seven separate tests have been performed. The description of each test, including purpose, test configuration, instrumentation and results is provided in Attachment 1. Four specimens have been tested. They are: a 4 inch Schedule 160 pipe with a 1/2 inch U-bolt; a 10 inch Schedule 40 stainless steel pipe with a 3/4 inch U-bolt; a 10 inch Schedule 80 carbon steel pipe also with a 3/4 inch U-bolt; and a 32 inch pipe (same size and schedule as the main steam line) with a 2 3/4 inch U-bolt. We conclude from these tests that the U-bolt cross piece essentially can

perform effectively as a clamp provided that sufficient preload is established in the U-bolt. A brief summary of the test results is set forth below:

(1) Torque versus Preload Test

The objectives of this test were twofold. The first objective was to establish the relationship between torque applied to the U-bolt nuts and the resulting tension in the U-bolt as a function of pipe size. This information is needed to fully establish whether the particular connection behaves like the traditional bolted connection in which the tension in the bolt is given by the following linear relationship:

$$t = KTD$$

where K is a constant (assumed to be 0.2 by Mr. Doyle, see CASE Findings at IV-14), t is the applied torque, D is the bolt diameter, and T is the tension in the bolt. The second objective of the preload test was to determine the strain in the pipe as a function of preload. This latter information is used to verify the adequacy and accuracy of the finite element analysis models.

The results of the torque versus preload test indicate that a roughly linear relationship exists between the torque imparted to the U-bolt nut and the tension developed in the U-bolt. While the test does confirm the approximate

linearity of the formula t = KTD, it indicates that the values of K vary between 0.22 and 0.35 for the type of threads present in the U-bolts. Thus, for this type of bolted arrangement, more torque is generally needed to develop the same tension in the bolt than would be estimated by using the conventional formula t = .2TD to which Mr. Doyle refers. It is noted that there is also a slight variation from linearity (which appears to increase with U-bolt size) within the range of specified torque values for each U-bolt size. This variation is not significant for the range of torque values which exist in the field.

In addition, the test reflects that maximum pipe strains (and stresses) caused by preload are generally found in the circumferential direction, are compressive in nature, and occur generally right below the cross piece. In the 10-inch schedule 40 stainless steel pipe, strains similar in magnitude are also seen near the U-bolt contact area, but in this instance they are longitudinally oriented. The magnitude of the stresses varies from pipe size to pipe size and, of course, varies with preload. The relationship between maximum stresses and preload is nearly linear for all pipes. Refer to Figures 5 through 8 of Attachment 1 for the maximum pipe stresses caused by preload alone.

(2) Friction Test

The objective of this test was to determine the force on the U-bolt needed to cause slippage between the U-bolt/cross piece assembly and the pipe. Since the direction of slippage (which is of primary interest for stability) is in the plane of the U-bolt, i.e., rotation around the pipe, the slip force is determined by applying a tangential load, i.e., in the plane of the U-bolt, to the cross piece, shown in Photograph 5 of Attachment 1.

The friction test produced two results. The first result is the force required to cause slippage between the U-bolt support assembly and the pipe in the plane of the U-bolt (i.e., the force that produces rotation about pipe axis). This force is that required to overcome the friction developed between the pipe and the cross piece at the line of contact between the two, plus the friction developed between the pipe and the U-bolt. The latter friction force develops asymmetrically around the U-bolt pipe contact area. The lateral force applied to the cross piece produces a moment which causes unequal tension in the two U-bolt legs with correspondingly unequal normal forces and friction developed in their respective contact areas with the pipe.

From this test, the lateral forces as a function of preload which overcome asymmetric friction were determined. This lateral force required for slippage is indicative of whether the U-bolt/cross piece will rotate (slip) under the application of seismic forces. The seismic force applied through the strut can vary in inclination about a 5° cone from the strut axis. This 5° cone is allowed by CPSES's maximum permissible offset, which includes effects due to installation tolerances, thermal and seismic motion and pipe rotation. Therefore, there can be a lateral force (tending to rotate the U-bolt/cross piece around the pipe), axial force (tending to move the assembly along the axis of the pipe) or combined axial and lateral force acting on the assembly which equals the sway strut earthquake force times the sine of the inclination angle. The maximum value of this lateral or axial force is 8.7 percent of the sway strut force. (See Figure 1 for descriptive explanation of the forces mentioned above.)

If the lateral force required for slippage exceeds the maximum value of the lateral component of the sway strut force, then the U-bolt assembly cannot slip and rotate about the pipe. The U-bolt assembly would then be laterally stable, i.e., capable of transmitting and receiving the

applied loads. This stability issue is the one upon which the Board focused its attention, i.e., rotation of the assembly around the pipe.

Although the assembly would not slip and rotate about the pipe under these circumstances, a slight roll with no slippage will occur to balance the moment created by application of the lateral component of the seismic force to the cross piece. This roll in no way impairs the stability of the assembly. It is the roll which creates the asymmetry in the tension of the U-bolt legs and the consequent frictional force asymmetry, and it corresponds to the physical behavior of the assembly which wants to align with the line of application of the force.

In addition to the rotation about the pipe, insufficient axial friction force between the U-bolt assembly and the pipe would permit motion of the U-bolt assembly along the pipe axis. As will become apparent in later sections of this Affidavit, this axial motion, when occurring without rotational motion, can only result in an axial movement of the assembly which reduces the strut inclination angle. This motion would continue until the inclination angle reduces to a value at which the axial component of the seismic force equals the axial, frictional resistance force. This conclusion, which is borne out by

the vibration and seismic test results, provides further assurance that axial motion alone does not cause concern with stability.

The second result of the friction test is the determination of the coefficient of friction which exists for the typical U-bolt/cross piece pipe assemblies. The coefficient of friction is determined to vary between 0.12 and 0.225 for stainless steel pipes and 0.19 to 0.52 for carbon steel pipes. The variation in the friction coefficient could not be correlated to surface conditions. With the exception of the 10-inch stainless steel specimen, no polishing of the contact surface between the pipe and the assembly was observed after all the testing. This is strongly indicative of little if any surface yielding at the contact points. Some surface polishing was observed on the 10-inch stainless steel pipe used in the tests. In fact, an increase in the coefficient of friction for this specimen was observed when the friction test was repeated utilizing a new U-bolt placed on an untested surface of the pipe.

(3) Load Distribution/Strain Measurement Test

The objective of this test was to determine the stiffness of the assembly. The test was performed to provide information to refute Mr. Doyle's allegations that in addition to the preload, one should add thermal expansion

and mechanical loads directly to the support/pipe connection. Applicants do not disagree that thermal expansion loads are additive to the preload, but ran this test to demonstrate that the U-bolt/cross piece pipe connection behaves as a typical preloaded joint, whereby the total mechanical external loads are not directly additive to preload.

This test was conducted on the 10 inch stainless steel pipe/U-bolt assembly only. The choice of the test specimen was predicated on the fact that this specimen results in the most flexible connection of those tested, and thus, is the least likely to behave as a preloaded joint.

The test results indicate that for pretorquing levels ranging from 33 ft-lbs. to 100 ft-lbs. (these levels encompass most of the torques that are present in the plant on 10-inch piping), the behavior of the pipe assembly closely resembles that of an ideal preloaded joint. For the latter, no increase in compressive or tensile loads would be experienced until the applied load exceeds the preload. Figures 17 through 20 of Attachment 1 indicate a relatively shallow slope of the U-bolt load/applied load relation (for the ideal preloaded joint the slope would be zero). The slope of the line is approximately the same for the joint

acting in tension as in compression. This indicates that regardless of the direction of externally applied mechanical load, this load is not directly additive to the preload.

For instance, from Figure 17 of Attachment 1, an externally applied load of 7200 lbs. on a U-bolt assembly initially preloaded to 9600 lbs. causes only a 3600 lb. increase in load, i.e., U-bolt final load is 13,200 lbs. and not 16,800 lbs. as one would obtain by directly adding the external load to the preload.

Thus, although the three mechanisms referred to by CASE on page IV-8 of their Findings are considered, namely preload, thermal induced loads, and mechanical loads, only two are directly additive (preload and thermal). The percentage of mechanical load addition depends on the relative stiffness of the U-bolt/cross piece and pipe.

Measurement of the strains (stresses) registered by strain gauges placed in the cross piece during this test confirms the validity of the strains measured in the U-bolts. Finite element analyses of the cross piece (with the bracket welded on the two sides parallel to the long side of the plate) confirm that the plate behaves essentially as a cantilever. Correlation of the stresses that would result from application of the load applied by the U-bolt (U-bolt tension) with those measured via the strain gauges is good.

For instance, the test stress derived from the cross piece strain gauge readings corresponding to a U-bolt tension of 6,200 lbs. (per leg) is 17.1 ksi. Attachment 2 provides the model and results of a finite element analysis of the cross piece. The model and analysis employed are linear. This model was executed for an arbitrary load of 20 lbs. and hence the results are for a load equal to 20 lbs. in each leg of the U-bolt. Stresses resulting from higher (or lower) loads can be computed by linear ratioing. Thus, the stress computed from the finite element analyses, when a 6200 lb. load is applied, is 16.4 ksi.[8]

(4) Thermal Cycling/Thermal Gradient Test

The objectives of this test were twofold. One objective was to determine the additional load on the support and pipe (and resulting stresses) caused by differential thermal expansion of the pipe with respect to the U-bolt. (The results of testing for this objective are set forth in Attachments 2 and 3 and were used in the finite element analyses.) The second objective was to assess the relaxation of the U-bolt preload caused by long-term

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8 This finite element analysis was performed prior to a final precise measurement of the cross piece dimensions. As a result, some dimensions are not exactly equal to the dimension of the cross piece utilized in Attachment 3. The small differences in dimension do not affect the conclusion of this study.

temperature cycling to determine whether material relaxation effects would reduce the preload to the extent that slipping of the U-bolt/cross piece can occur.

In this test, the specimens were cycled between room temperature and the maximum operating temperature. The maximum operating temperatures are 560°F, 250°F, 250°F, and 560°F for the 4 inch, two 10 inch, and 32 inch pipe specimens, respectively.

Results of this test indicate that the temperature distribution in the U-bolt is not uniform, regardless of whether the assembly is insulated or not. Temperature distributions achieved in the U-bolts at steady state are reported in Attachment 3. This Attachment reports on the results of the finite element analyses conducted in parallel with the testing program. The temperature distributions in the U-bolts are used as input to the finite element analyses.

Figures 21 through 24 of Attachment 1 show the loss of preload resulting in the U-bolts when the specimen assemblies are cycled between room and maximum temperatures. The thermal cycling was terminated after 10 cycles, as it became evident that the U-bolt had relaxed to a final stress state after a few cycles. The results of the thermal cycling tests indicate that the 4-inch specimen had relaxed

from the maximum preload value (corresponding to a torque level of 60 ft-lbs., which is the maximum measured in the field) to a value which is approximately 64 percent of that value.

For the 10-inch schedule 40 stainless steel specimen, the relaxation is small, if any, as shown in Figure 22 of Attachment 1. Regarding the behavior of the 10-inch schedule 40 specimen, further explanation is necessary. As the specimen was heated during cycle one, the preload immediately dropped to about 4000 lbs. from the initial value of 4500 lbs. This behavior is contrary to what was expected, (i.e., additional thermal load of approximately 700 lbs. per leg should have increased the preload from 4500 to 5200 lbs. per leg). This behavior is indicative of an initial fit-up which caused the initial preload to revert to the lower preload corresponding to a better fit as soon as the heatup started. Problems with fit-up are also evident by the sudden relaxation of preload in one of the legs during the cycling. Thus, essentially no thermal cycling relaxation was experienced by the 10-inch schedule 40 specimen. This is not unexpected since the maximum temperature of the specimen is 250°F.

For the 32-inch specimen, the relaxation cannot be determined from the data (see Figures 23 and 24 of Attachment 1).

The 4-inch specimen is subjected to far more severe thermal cycling than the 10-inch specimen. This specimen is cycled from 107°F to 560°F, with an initial maximum preload which induces stresses in the U-bolt body which are approximately 89 percent of the yield strength.[9] After the final thermal cycle at ambient temperature, the stresses in the U-bolt are reduced to about 54 percent of yield. This is also not unexpected. Materials with the characteristics of A-36 exhibit relaxation characteristics at low temperatures (low temperatures being defined as below 600°F) which cause the stresses in the material to drop to about 1/2 of yield.[10] Also, these materials exhibit the characteristic that most of the strain relaxation occurs very soon after the high stress is created. For instance, the C, Mn, Si steels (of which U-bolts are made) tested for

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9 This value is based on 36 ksi minimum yield of SA-36 steel.

10 If the material initial stress state is above a nominal 1/2 yield stress, the material would relax even without thermal cycling, but probably at a slower rate. If the material initial stress state is below the nominal 1/2 yield stress, thermal expansion stresses might place it in a state of stress above 1/2 yield, from which the material would relax.

relaxation at initial strains corresponding to yield strains, relaxed to almost half that strain in the first hour and remained at that level after 100 hours.[11]

This means that the maximum relaxation of each specimen, or conversely, the minimum preload that would remain in field applications, can be predicted with reasonable assurance. At the temperature and stress level existing in the 10-inch schedule 40 stainless steel specimens (250°F and 16 ksi, respectively), no relaxation was expected and none was noted (see remarks on fit-up.) This is also true for the 32-inch pipe for which the stress data are not conclusive, but certainly indicative of low stresses. Further information regarding the relaxation characteristics of the specimens is provided by the creep and vibration tests.

(5) Creep Test

The objective of this test was to determine whether long-term temperature exposure could result in material relaxation so that preload would be decreased or lost.

The creep test was conducted on all three specimens following completion of the thermal cycling test. The specimens were maintained at their peak temperatures for

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11 "Compilation of Stress-Relaxation Data for Engineering Alloys," ASTM Data Service Publication DS-60.

over 24 hours. The results show that after the initial relaxation achieved during the thermal cycling test, no further relaxation occurred. This indicates that at these temperatures creep is not a concern.

(6) Accelerated Vibration Test

The objective of this test was to determine whether normal vibration levels in the plant could cause material relaxation, and consequently, loss of preload. In order to simulate 40 years of accumulative effects of piping vibration, this test was run as an accelerated vibration test utilizing vibratory forces varying in frequency from 5 to 200 Hz at an amplitude equal to the maximum expected OBE force for the pipe tested (4000 lbs.) as well as at lower forces (1000 and 1500 lbs.). The time duration of this test combined with the amplitude of the vibratory (sinusoidal) force resulted in an overall energy input to the test specimen exceeding by orders of magnitude the energy that would be induced by an earthquake[12] (both operating basis and design basis earthquake). An initial 4000 lb. test was run in excess of 30 seconds, two tests were run with a 1000

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12 Applicants recognize that the energy imparted to the piping/U-bolt assembly during the 30 second duration of a design basis earthquake is larger than the energy imparted during any 30 seconds of the test. Applicants therefore also performed a seismic test of the 10" Sch. 40S pipe which utilized a force equivalent to that of the SSE. This test is described later in the Affidavit.

lb. force for 2.5 minutes and a 1500 lb. test was run for 270 minutes as described in more detail later. This test simulates conditions far more severe than expected in the plant for normal vibration levels.

The major results of this test are noted below. Insufficient assembly preload will permit the assembly to rotate about the pipe and also to walk axially along the pipe axis in either direction with respect to the location of the strut. This can occur because the low preload permits the asymmetric tension in the U-bolt legs to relax sufficiently upon application of a sufficiently large compressive external load within the 5° inclination that the frictional force between the U-bolts and the pipe, and/or friction between the cross piece and the pipe, is overcome by the lateral or axial component of the externally applied force. This permits slippage of the U-bolt and/or cross piece along an erratic helical path (if the applied external load is inclined both laterally and axially). The axial motion away from the strut, i.e., the motion that would lead to an increase in the angle of inclination, can only occur if the assembly is permitted to rotate. With rotation, the lateral component of the force is increased as the size of the angle increases, and the axial component at the new

location of the strut can also be higher, causing the cross piece to move away from the application of the compressive force.

With rotation inhibited by sufficient preload, any axial motion will be in a direction toward the applied load and will occur during the pull portion of the cycle at the cross piece contact point. The direction of travel is preordained since slip occurs axially only during the pull portion of the cycle. (During the push portion, sufficient friction is created at this point to resist slip by the component of the sway strut force increasing the cross piece/pipe contact load.) This behavior is verified by the accelerated vibration and the seismic tests.

In preparation for the full accelerated vibration test, a constant amplitude sinusoidal force of 4000 lbs. was applied to the pipe for a period exceeding 30 seconds with the frequency of excitation being varied from 5 to 10 Hertz. (These frequencies are at or near the peak in the response spectrum of CPSES.) This pretest estimates the seismic excitation imposed by an OBE, since the maximum force from an OBE for a 10-inch pipe would be 4000 lbs. The pretest was interrupted to avoid damage to the hydraulic actuator in the test machine. At 20 ft-lbs. preload, the assembly was seen to rotate about and walk along the pipe. Although the

assembly was still capable of transmitting the same load, approximately 4000 lbs., (thus acting as a stable support for 30 seconds or more) Applicants consider this to be insufficient preload. This was confirmed by a second two and one half minute pretest conducted at a sinusoidal force of 1000 lbs. having a frequency sweep from 5 to 200 Hertz at 2 octaves[13] per minute. At the 20 ft-lb. preload, although the assembly was acting as a stable support, it still rotated and walked, although in a slower manner. The pretest was then rerun with the preload torque increased to 35 ft-lbs. The input sinusoidal force had an amplitude equal to 1000 lbs. with the same frequency sweep. No motion of the assembly was observed during the 2 1/2 minute pretest.

To run the official acceleration vibration test, the preload torque was increased to 50 ft-lbs. The official accelerated vibration test was run at 1500 lbs. (swept from 5 to 200 Hz and back at 2 octaves per minute) for 270 minutes. The assembly was observed to move initially axially (in the direction that reduced the strut angle),

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13 An octave is a doubling of frequency. Thus, from an initial frequency of 5 Hz, the frequency of the excitation will be 10 Hz in 30 seconds, 20 Hz in one minute, 80 Hz in 2 minutes and 160 Hz in 2.5 minutes (200 Hz takes slightly more than 2.5 minutes).

then to stay in place for the duration (except for a sudden, but inconsequential cocking in the latter part of the test). No rotation was observed.

At the end of the test, the applied sinusoidal force was increased to 4000 lbs. With this preload, the assembly was also vibrated for a period of 30 seconds with the frequency being swept from 5 Hz to 200 Hz and back to 5 Hz at a rate of 20 octaves per minute. This rapid sweep was done to avoid damage to the hydraulic actuator. This last accelerated test conservatively bounds the seismic excitation imposed by an OBE, since not only is the force (4000 lbs.) equal to that which results from an OBE for 10-inch pipes, but the frequency range of the OBE is narrower than that swept in the test. Moreover, for the OBE, the forces at the high frequencies are much lower than the 4000 lbs. which was applied for all frequencies. No rotation of the assembly was noted during these 30 seconds, but there was the same axial motion of about 1/16 inch toward the strut that occurred during the 270 minute test. This motion reduced the angle of inclination.

The observed behavior confirmed that the assembly is stable at that preload torque, and also confirmed the theorized behavior of the clamping action.

At the end of the official accelerated vibration test, the relaxation of the preload was also measured. The initial preload stress (based on U-bolt body area initial strain = 311 micro inch/inch) was equal to 9020 psi. Significant relaxation was not expected since the initial stress was below half of the yield stress. After the initial repositioning of the assembly, which reduced the preload, no further decrease in preload was observed, indicating that the vibration per se had no effect on relaxation.

(7) Seismic Test

The objective of this test (an auxiliary test to the accelerated vibration test, noted above) was to test the effect on the assembly of the peak SSE force, 7000 lbs. Although the overall energy inputted in the tested system in the official accelerated vibration test (discussed above) is clearly much greater than that inputted by the design bases earthquake (SSE), Applicants wanted to test the effect of peak force on the specimen assembly and also the effect of inputting an energy rate comparable to or in excess of that expected from the SSE. Therefore, a 39 second test with a sinusoidal force applied at a frequency of 9 Hz (roughly corresponding to the peak of the CPSES floor response spectrum) was also run to simulate the maximum response to

the SSE. Although the sinusoidal force was to have a magnitude of 7000 lbs., in reality the average force magnitude reached a maximum of 9500 lbs. initially, decreased linearly to 8600 lbs. after 21 seconds and remained at 8600 lbs. throughout the remainder of the test. (See Figure 1 of Addendum 1 to Attachment 1.) Preload was applied via a 50 ft-lb. torque.

The test was to have been repeated at the resonant frequency of the test specimen (established to be at about 75 Hz by sweeping the frequency range with a lower amplitude force).[14] However, at this higher frequency, the hydraulic actuator was incapable of transmitting the required 7000 lbs. of force (due to play in the strut connections exceeding the displacement output of the actuator at that frequency). The maximum force output of 75 Hz was approximately 1300 lbs. The SSE energy at this frequency is neglibible. While the 1300 lb (75 Hz) test was aborted after 4 seconds, it provided useful additional qualitative information that the U-bolt assembly behaves stably. Prior to running the nominal 7000 lb. test, three trial tests were run to adjust test equipment and instrumentation. Although

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14 The combination of the test run at a frequency corresponding to the peak of the floor response spectrum and of the test run at the resonant frequency of the test specimen would encompass the worst situation that can be encountered in the field.

none of the three trial tests can be considered official, we note that the performance observed was in all cases consistently the same as that in the official test. Thus, what was observed in the official test was systematic behavior of the assembly. In the official test, the assembly did not rotate, but moved axially toward the strut (confirming the theoretically predicted behavior) a distance of about 1/2 of an inch. After that, it remained in place. The same motion had been observed for the three trial tests.

Although the 75 Hz test was also unofficial, it was observed to result in no rotational motion and barely perceptible axial motion (toward the strut).

At the end of the nominal 7000 lb., 9 Hz test (actual force in excess of 8600 lbs.) the preload value in each leg of the U-bolt was measured to determine whether strain relaxation had taken place. Strain relaxation was expected since the applied force coupled with the initial preload stresses the U-bolt to above 1/2 of yield stress. In this instance, with an initial preload of 4484 lbs. in both legs of the U-bolt and an applied external force of 8600 lbs., the peak tension in the U-bolt would be approximately 6600 lbs. (see Figure 17 of Attachment 1), resulting in a stress at the threaded area which is about 10 percent over 1/2 of

yield.[15] Since prior to stabilization of the applied force, a higher amplitude force was seen, more relaxation could be expected and was in fact seen during the initial stabilization period. The remaining preload, measured after completion of the test indicates relaxation of approximately 12-13 percent to a stress level below 1/2 of yield stress.

Q. What do you conclude from the results of the tests regarding CASE's allegation concerning instability?

A. We conclude that the U-bolt/cross piece assembly can perform effectively as a clamp provided that sufficient preload is established in the U-bolt. (It should be noted that a clamp also requires preloading.) We further conclude that even if the preload level was insufficient, but still present in some amount, the U-bolt support would vibrate, but still be capable of supporting the necessary loads, thus behaving "stably."

To provide further assurance that the preload on all affected cinched down U-bolts is adequate, Applicants will conduct a 100 percent inspection of the torque of all such U-bolts (380). At the time of the inspection, to remove questions regarding stability, Applicants will assure that

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15 Using a yield stress equal to 36 ksi and recognizing once again that the 1/2 yield stress level at which relaxation ceases is not a precise figure, but a more imprecise range of stresses near the 1/2 yield stress at which relaxation would stop.

such U-bolts are torqued to levels at which the assemblies will be stable in the absolute truest sense, i.e., no rotation, and axial movement, if any, is toward the strut. (To check the torque of a U-bolt requires essentially no more effort than torquing the U-bolt to prescribed values.) The levels to which U-bolts will be torqued are set forth in Table P of this Affidavit.

The results of the tests conducted for vibration and for seismic response confirm the stability of the assembly when preloaded to these values. These values are set forth later in this Affidavit.

Q. What are your conclusions from the results of the tests regarding the stresses in the U-bolt and piping?

A. The only conclusions that can be derived from the test program regarding the piping stresses are limited to data obtained during the preload test. The test results indicate no unacceptable stresses in the pipes for the preload conditions. Measurement of the strains on the pipe from which stresses can be obtained, is limited to locations near but not precisely at the location where peak stresses are expected to occur, i.e., under the cross piece and U-bolt at the points of contact.

Because of this limitation, Applicants' plan to determine such stresses relies on measurement during tests of strains (stresses) at selected locations along the pipe axis and around the circumference near the cross piece and the U-bolt, to serve as reference data points for the finite element models. The stress distribution in the pipe resulting from the application of preload, thermal expansion, internal pressure, and externally applied loads, is developed using the finite element models which will be described later. Detailed discussion of the stresses in the pipe is thus deferred to later sections of this Affidavit, which summarize the results of the finite element analyses.

Several points need to be made regarding the potential high stresses in the U-bolt. First, the preload applied in the test is larger than that expected in the field.[16] Hence, actual field stresses will be lower. Second, if initially the bolt is preloaded to stresses exceeding approximately one-half the yield stress, the material will relax, and the preload will consequently drop until a stress

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16 The preload applied in the test is larger than that expected presently in the field. However, for large pipes such as the 32 inch tested specimen, Applicants plan to increase the preload over that which is now present, to ensure adequate margin for stability. For the larger pipes and large diameter U-bolts, however, the U-bolt stresses are low. For instance, stresses in the 2 3/4 inch U-bolt of the 32 inch specimen preloaded with 250 and 500 ft-lbs. are approximately 1000 and 2000 psi, respectively.

state of about 1/2 of yield is achieved. The deformation required for this relaxation is negligible[17] and sufficient preload still exists to assure the stability of the support. The minimum preload necessary to assure stability is below the preload level which would exist when the stress _s about 1/2 yield. For the 4-inch specimen, for instance, the preload torque that would correspond to stresses equal to 1/2 yield is approximately 30 ft-lbs., and the minimum torque conservatively estimated to be for stability is about 25 ft-lbs. (finite element analyses would predict about 10 ft-lbs.), (*see* later discussion of results of finite element analyses). Third, normal plant vibration will not significantly affect the preload once it has relaxed to the "final" condition. Here, final denotes the condition for which the stress in the U-bolt is about 50 percent of the yield stress and no further relaxation takes place. Fourth, seismic loads would not affect the "final condition" preload (as is evident from the accelerated vibration and seismic tests)[18]. Fifth, tests conducted by Applicants have demonstrated that there is adequate margin between yield and

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17 Deformation is negligible since total strain is not altered. Stress relaxation occurs because part of the elastic strain is converted to inelastic strain.

18 Had the seismic test been run at 7000 lbs. (maximum seismic load for 10 inch line) little if any relaxation would have occurred.

failure of the U-bolts.[19] There is thus no concern with failure for stresses that might be initially above the yield stress.

The stresses in the U-bolts were measured during all of the tests. With the specimens torqued at the highest preloads (60 ft-lbs., 100 ft-lbs., and 240 ft-lbs. for the 4", 10" and 32" pipe, respectively), the maximum stresses in the U-bolts occurred during the thermal cycling test. First, we discuss the results for the 4-inch pipe. During torquing, the stress in the U-bolt for the 4-inch pipe reached 35,360 psi (near yield) in the U-bolt shank. (Within the first hour or so, due to relaxation, these stresses would have decreased to about 1/2 yield.) During torquing, the threaded area of the bolt exceeded yield slightly; yielding of the threaded area was noted in the test. (However, due to the self-limiting nature of the load, this would have no adverse impact on the material.)

After initial material relaxation (if any, since the initial state of stress depends on actual preload and if preload is low there will be no relaxation) application of a mechanical load in the U-bolt tensile direction such as

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19 See Affidavit of R.C. Iotti and J. C. Finneran, Jr., attached to Applicants' Motion for Summary Disposition of CASE's Allegations Regarding One-Way U-bolts Acting as Two-Way Restraints (May 23, 1984).

might occur during an earthquake would add some load. Because the joint behaves as a preloaded joint, and because the pipe is very stiff (4" Schedule 160), most of the load goes into the pipe and little into the U-bolt. (The finite element analysis conducted for a compressive load confirms this joint behavior). The increment in load experienced by the U-bolt would cause some further relaxation which would reduce the stress to 1/2 of yield stress again.

Secondly, we discuss the 10-inch pipes. For the 10" Schedule 40 stainless steel pipe, U-bolt stresses measured in the shank for preload conditions corresponding to the maximum torque (100 ft-lbs) were 12,838 psi. No appreciable increase in load resulted from thermal cycling. (Actually, there was a decrease indicating release of some form of mechanical binding). Externally applied loads, such as seismic loads, would increase the stress when directed so that the U-bolt tension is increased. For the maximum postulated external load (7000 lbs.), the stress would increase by about thirty percent, placing it near 1/2 yield.

The 10" Schedule 80 carbon steel pipe was not tested during thermal cycling since the behavior of the corresponding stainless steel specimens is more severe. U-bolt stresses in the 10" Schedule 80 pipe test were measured

to be 17,164 psi in the shank (22,650 psi in the threaded area), when the preload corresponds to a torque of 100 ft-lbs.

Finally, for the tests of 32" pipe, the preload value (240 ft-lbs. torque) placed negligible stresses in the 2-3/4 inch U-bolt.

Data on stresses in the cross piece are available only for the 10-inch and 32-inch tests. The geometric configuration of the 4-inch pipe prevented placement of strain gauges in the cross piece locations where their readings could be correlated to the readings of strain gauges placed on the U-bolt.

For the 10- and 32-inch cross piece specimens, no stresses exceeding 23,000 psi were observed. The stress of 23,000 psi occurred in the 10-inch stainless steel specimen cross piece when the maximum mechanical load was added (as a pull) to the maximum preload. (It should be noted that since joint acts as a non-ideal preloaded joint, the external mechanical load is not fully additive to the preload.) These stresses are less than the allowable, 0.75 times the yield stress.

In summary, we conclude that application of the maximum torques to the U-bolt pipe assemblies can potentially result in high but acceptable local pipe stresses and can further

result in high stresses in the U-bolts. Test results were not intended to provide, by themselves, sufficient information to assess the significance of the pipe stresses. We defer discussion of that aspect to later sections of this Affidavit. We do not believe that the stresses produced in the U-bolts present a concern. To begin with, high stresses occur only if large preload values are applied (i.e., near the maximum used in the test) to small diameter U-bolts. Large preload values are generally not present in the plant supports, nor are they needed to assure stability of the supports under seismic excitation (see note 15 in reference to large pipes and large diameter U-bolts). In those instances where high preload torques may be initially present, the characteristic relaxation behavior of the material employed (A-36) will reduce the preload value, and hence, the stresses in the U-bolt, to acceptable levels. Moreover, tests conducted for Applicants have demonstrated that there is adequate margin between yield and failure of the U-bolts. For instance, these tests showed that for the 1/2-inch U-bolt employed for the 4-inch specimen, the margin is about 2 to 1. Therefore, there is no concern with possible failure of the U-bolt even if it were to be initially pretorqued to the largest values noted in the field.

With regard to the cross piece, no unacceptable stresses have been noted. However, there was no test of the 4-inch pipe corresponding to the conditions in which the U-bolt experiences the largest load. Finite element analysis of this configuration indicates that bending stresses in the cross piece are determined by stresses in the U-bolt. Initial high preload values on the U-bolt could produce stresses in the cross piece that exceed allowable values on an elastically calculated basis. However, since U-bolt stresses ultimately revert to one half of yield, the cross piece would not have stresses above allowables at that point.

Q. Please restate why Applicants performed finite element analyses in addition to the tests.

A. The reasons for the development of the models and the execution of the finite element analyses have already been stated, but can be briefly summarized again: (1) without a theoretical model, explanation of results from tests may be impossible, (2) a theoretical model is needed to extend the information provided by test into the location of the tested specimens where the test provides no information, and (3) a theoretical model, verified by test comparison, must be used for any predictions/conclusions that one may have to make on configurations which are not tested.

Q. Please describe the finite element analyses used in your evaluation.

A. Each assembly tested was modeled utilizing MSC NASTRAN Version 63. This computer code was chosen because it is universally recognized and accepted by industry as having the capability of providing analytical solutions that accurately characterize the local stress, gap, friction effects, and plastic material behavior (if any) that are important for assessing the pipe and U-bolt assembly stress, and the support stability. A detailed description of the finite element models developed for each test configuration and of the modeling technique used is provided in Attachment 3.

Q. Why is not all the information essential to respond to the Board's concerns available from the tests?

A. Some of the information can be derived directly from the test program. For instance, the Board's concern regarding forces and stresses that are induced in the U-bolt itself can be obtained directly from the test results. The forces and stresses in the U-bolt which result from torquing, torquing plus thermal expansion, and externally applied loads are direct outputs of the preload, thermal cycling, and loading portions of the test program.

On the other hand, to unequivocally answer the Board's concern with pipe stresses and how they are influenced by cinching the U-bolt and related stresses, a mix of information derived from test and analyses is required. Where the peak stresses occur, they cannot be measured. Also, it is not feasible to completely cover the pipe specimen with strain gauges (inside and out) to obtain the stress distribution in the pipe. In this instance, the finite element model is used to extend and complete the test data. Finally, there are concerns which can only be answered by test. Examples of these concerns are the relaxation characteristics of the assembly under long term vibration, thermal cycling, and preload. The thermal cycling, creep and accelerated vibration tests have provided answers to these concerns. No analytical tool could have done it.

Finally, test results are far more persuasive than any theoretical model with regard to demonstrating behavior. If the test shows that the assembly does not move, then it is impossible to argue that it did. This is different than having a static, theoretical model predict that it should not move. Endless arguments would ensue over the correctness of the prediction.

To answer the Board's concern on stability, we have conducted a test on assembly stability and have also correlated the prediction of the static model (finite element) to the dynamic results so that the Board would have confidence that the stable or unstable behavior of the assembly can be correctly assessed.

Q. Can you describe the objectives and results obtained from the finite element analysis program?

A. The finite element analysis program was performed (1) to determine if the pipe would slip, thereby creating an unstable support condition when the hanger support was subjected to the preload, thermal, pressure and mechanical loads that could be expected in the Comanche Peak hanger assemblies; and (2) to calculate pipe and pipe support stresses that could be expected to be experienced at Comanche Peak and assess their significance. In order to perform the above evaluations, it was necessary to evaluate the U-bolt support-piping assembly, using finite element analysis, for the four loading conditions noted below:

(1) Preload;

(2) Preload + Thermal;

(3) Preload + Thermal + Pressure;

(4) Preload + Thermal + Pressure + Strut Applied Load (Push).

The thermal loading condition was based on temperatures associated with the normal operating condition. The pressure loads were based on normal operating pressure. The applied loads evaluated (representing the external mechanical loads) were generally higher than the loads that the support strut would be expected to carry. The strut loads noted below in Table A were dependent on the line size considered.

TABLE A

Mechanical Loads on Sway Strut

4" Sch. 160	2,000 lbs.
10" Sch. 40S	10,000 lbs.
10" Sch. 80	10,000 lbs.
32" Sch. Main Steam	100,000 lbs.

This load was applied at the maximum permissible offset of 5° from the U-bolt axis parallel to the U-bolt legs. The preload values, based on torque values appropriate for each pipe size, are as given in Table B below.

TABLE B

Maximum and Minimum Analysis Torque Values

4" Sch. 160	Max. value 60 ft-lbs. Min. value 9 ft-lbs.*
10" Sch. 40S	Max. value 100 ft-lbs. Min. value 46 ft-lbs.*
10" Sch. 80	Max. value 100 ft-lbs. Min. value 11 ft-lbs.*
32" Main Steam	Max. value 240 ft-lbs.**

* These minimum values are different from the minimum values employed for the test. They are values derived from iterative finite element analyses per-formed to help define lower bound values required for stability. The finite element analysis predicts that the assemblies would behave stably a these and even lower values. (No optimization was done.)

** No minimum value analysis was conducted for the 32 inch model.

The results of the analyses are given in Attachment 3. The maximum stresses in the U-bolt legs for the four cases considered were determined to be the following for the different torque values.

TABLE C

Maximum Stresses in U-Bolts

4" Sch. 160	60.5* ksi
10" Sch. 40S	22 ksi
10" Sch. 80	27.5 ksi
32" Main Steam	7.5 ksi

* Calculated on an elastic basis. Actual stresses will be lower and includes the preload, thermal expansion restraint, external mechanical load and radial pressure expansion. Stresses are calculated in the threaded area.

These stress values are based on the U-bolt threaded area and compare generally very favorably with test results. Deviations are explainable and are due to differences between real configurations and idealized representations, e.g., fit-up, relaxation, and out of roundness. The loading case resulting in the highest U-bolt leg stress was Preload + Thermal + Pressure. When the push load was applied, the leg forces were unloaded.

The stresses in the pipe for the maximum preloads evaluated for each pipe size were obtained from the finite element model. The maximum preload torque values were used since this loading level produces the highest stresses in

the pipe. Tabulated below in Table D are the maximum pipe stress intensities for the four load cases evaluated without the effect of mechanical piping stresses.[20]

TABLE D

Maximum Calculated Stress Intensities*

Load Case	4" Sch. 160 (ksi)	10" Sch. 40S (ksi)	10" Sch. 80 (ksi)	32" Main Steam 32MS (ksi)
Preload (P)	26.1	48.5	29.9	4.9
P + Thermal (T)	39.3	60.6	34.9	21.6
P + T + Pressure (Pr)	42.9	58.6	32.9	34.4
P + T + Pr + Push	44.8	73.4	44.4	47.2

* Excludes external mechanical load.

It is evident from these results of the elastic finite element analysis that elastically calculated stresses can exceed yield. These stresses occur locally at the point of contact between the cross piece with the pipe, if the U-bolt is torqued beyond a certain value, which is generally in excess of that which can be expected in the field and is always above the value which is required for stability.

To investigate the effect of plastic deformation resulting from excessive U-bolt torquing, the elastic finite element model was modified to include the nonlinear material

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20 The reader can determine how the mechanical piping stresses (i.e., those due to all effects other than the clamping action) affect the values of the stress intensities given in this table, from the table given on p. 59 of Attachment 3. From this Attachment total maximum stress intensities are 64.14, 74.21, 54.69 and 47.17 ksi for the 4", 10" Sch 40S, 10" Sch 80 and 32" pipes, respectively.

behavior. The finite element model chosen for this analysis was that for the 10" Schedule 40S pipe. An elastic/plastic analysis on this pipe/U-bolt assembly would therefore be the most indicative of the effects that local yielding would have on the conclusions reached regarding assembly behavior and distribution of stresses. The results of the elastic/plastic analysis show for an approximate 100 ft-lb. preload, small plastic deformations of only those finite elements adjacent to the line of contact between the pipe and cross piece. The distribution of stresses for the elastic/ plastic analysis is the same as for the elastic analysis and is discussed in Attachment 3. This result confirms what has been observed in the test, namely that no visible yieldings of the pipe take place. (Therefore yielding, if any, is inconsequential.) (Refer to pp. 17-18 of the Affidavit and the observation that little if any surface yielding (polishing) was noted after all tests were completed.)

Q. Does the ASME Code provide direct guidance regarding acceptance criteria for local stresses induced by external attachments such as these U-bolt clamp assemblies?

A. We are not aware of any direct quantitative guidance provided by the ASME Code regarding this issue. However, the ASME Code provides qualitative guidance in regard to

consideration of the effects of local external attachments (Sections NB-3645, NC-3645, ND-3645, and NF-3121). CYGNA, in its oral testimony presented in the May hearings corroborates our position. This qualitative guidance given in the Code leaves to the designer the responsibility for quantifying the acceptance criteria.

Q. In view of this lack of direct guidance, what did you use as acceptance criteria?

A. Applicants do not believe that it is proper on their part to establish firm acceptance criteria for industry regarding this situation. This does not mean that we are unmindful of our responsibilities to satisfy the intent of the ASME Code as expressed in NB-3111, NB-3624.1, NB-3645 and the corresponding articles in section NC and ND. We have in fact formulated acceptance criteria for CPSES to meet the intent of the ASME Code. The bottom line, however, is that the testing program and finite element analyses have demonstrated that cinching of U-bolts as done at CPSES and generally by the industry produce no adverse effects on piping and supports for the range of pretorque values which are either representative of the worst conditions encountered at the plant or required to ensure stable behavior of the U-bolt assembly.

In what follows, therefore, we will establish the logic of our interpretation of the intent of the ASME Code as expressed in the acceptance criteria which we have adopted, by showing how the conclusions reached using these acceptance criteria match those derived from the test and finite element analysis. The following pages provide information from the results of the finite element analyses to enable interested parties to follow the logic of the conclusion reached by Applicants. To understand how the values given in several tables are derived, it is necessary for the reader to refer back to Attachment 3.

Q. What acceptance criteria have you adopted to assess the results of the finite element analyses?

A. To answer this question we need to define the type of stresses which have been measured and/or calculated by finite element analyses and compare them to our interpretation of the ASME Code.

Of the possible stress-type classifications, the only possible controversial one is that of the stresses caused by preload. Clearly preload is a one-time applied and maintained mechanical load. Tests as well as finite element analyses have shown that preload stress has some of the

characteristics of a secondary stress.[21] Finite element analyses and tests indicated that very small deformation at the contact area between the pipe and the cross piece result in relief of the local stress without affecting the stresses in the adjoining areas. Therefore, the stress[22] it is clearly local. Finite element analyses show that the locally high stress reduces to the general level of stress within one meridional (along the circumference) element on either side of the contact area.

The finite element analyses further indicate that the stress is composed of both a membrane and a bending component with the latter being predominant. The membrane portion is a secondary mean stress which the ASME Code defines as the <u>local primary membrane stress</u>, and which the Code includes in equation (9), Section III. The bending portion is a secondary bending stress which the ASME Code does not single out, but includes in the general category of the secondary stresses of equation (12) Section III.

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21 Secondary stress is defined as a normal stress or a shear stress developed by the constraint of adjacent material or by self-constraint of the structure. The above characteristic of a secondary stress is that it is self-limiting.

22 By stress here, we mean the stress caused by the preload proper and also stresses which result from the presence of the preloaded U-bolt assembly such as radial thermal and pressure expansion constraint stresses and stresses induced locally by external mechanical loads at the line of contact.

One way to look at the intent of the Code is to classify the entire local stress (membrane plus bending) as the primary local membrane stress to be inputted in a modified equation (9) with allowable stress intensity limits increased to $3S_m$ to account for the presence of the local bending stress (which would have to be limited to $3S_m$ by itself or in combination with the primary membrane stresses). One can infer from the Code that for application of a one-time load, a doubling of the allowable limits of equation (9) is permissible without inclusion of secondary bending stresses.[23] Since we are including secondary bending stresses in equation (9) (the secondary bending stress is by far the major component of the stresses caused by preload while the primary local membrane is very small), and do not further raise the allowable limit over that which can be inferred from the Code for a one-time load application, we are conservative with respect to what the Code intent would allow.

An alternative approach which we believe to be not as conservative, is to place the secondary bending stress into equation (12), and to evaluate the remaining, small local

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23 For example, equation 10a of the Code in Class 2 Rules for single, nonrepeated anchor movement allows a stress equal to $3S_c$, where S_c is the material stress allowable for cold conditions.

membrane stress with the regulation equation (9) (allowable limit = $1.5S_m$). Applicants have chosen to examine both alternatives in their evaluation of the pipe stresses.

Evaluation Using the First Alternative (See p. 66 for second alternative.)

Equations (9) and (12) are employed here, since the maximum piping stress due to the applied loadings is considered to be composed of primary and secondary stress. In the evaluation of the pipe, it is necessary to separate the primary and secondary stresses. The classification of the stresses is given below:

- Pressure membrane stress - primary stress
- Preload - local primary membrane stress (includes secondary bending)
- Thermal and pressure pipe growth restriction - secondary or peak stress
- Pipe hanger load (mechanical loads) - primary stress
- Pipe hanger thermal load - secondary stress

The rules of Subsection NB may be used to qualify a Class 1, 2, 3 or NNS (non-nuclear safety) piping component if the designer is willing to comply with all aspects of ASME Code Section NB. Subsections NC, ND and the ANSI B31.1 piping code evaluate primary and secondary stresses only by limiting the principal stresses caused by pressure and moment loading in the pipe. By evaluating stress

intensities (as opposed to principal stresses), detailed localized stresses, and the effects of cyclic loadings (fatigue), one is clearly enveloping the intent of sections NC and ND of the ASME Code and ANSI B31.1 by an NB evaluation.

Based on the above, the evaluation of acceptability of pipe stresses induced by the U-bolt pipe support is generally addressed using Section NB3600 of the code. NB-3600 (which governs the design and qualification of piping systems) gives little specific guidance to the method that should be used to evaluate stresses similar to those caused by the U-bolt pipe support. However, using equation (9) and equation (12) of NB-3600, an assessment of these stresses can be made since they provide a means of evaluating the stresses caused by the loading applicable to piping in the vicinity of the U-bolt. A total stress limit equal to $3S_m$ (where S_m is an allowable stress intensity for the material defined by the Code) is used for each equation.

The loading considered for each of the code equations used are given below:

1. Equation (9) of the Code with a $3S_m$ limit.

This equation must consider (a) primary membrane pressure stress[24] (b) piping moments at the hanger location due to pipe deadweight and seismic load, (c) maximum stress due to preload of the U-bolt, and (d) stresses due to the applied hanger load. The applicable stress limit is $3S_m$ since preload of the U-bolt is a one-time applied load (e.g., similar to building settlement) and includes local, through-wall bending (secondary bending).

2. Equation (12) of the code with a $3S_m$ limit.

This equation must consider (a) thermal stresses due to the restriction of pipe radial thermal growth at the U-bolt, (b) stresses due to the restriction of pipe radial pressure growth at the U-bolt,[25] and (c) piping moments at the hanger location due to longitudinal thermal expansion of the pipe.

Tabulated below in Table E are the $3S_m$ stress limits for the pipes in question.

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24 For convenience, these pressure stresses resulting from restraint of radial growth due to pressure expansion have also been included in this equation, although such stresses would be secondary. They have also been included in equation (12) so that these stresses have been accounted for twice.

25 See previous note.

TABLE E

Pipe Size	$3S_m$ at Normal Operating Temperature
4" Sch. 160	50.52 ksi
10" Sch. 40S	60 ksi
10" Sch. 80	60 ksi
32" Main Steam	58.26 ksi

A conservative estimate is obtained for the mechnical stresses in the pipe resulting from sources other than the U-bolt assembly itself. This is necessary since all local sources of load must be considered when comparing the pipe stress state at the U-bolt location to the above stress limit. Because of the multitude of supports involved in this evaluation, individual piping moments at the support locations, calculated from detailed stress analyses, have not been used. Rather, they have been considered in a general manner. The piping stress due to other mechanical loads has been conservatively estimated assuming that the stress was at its maximum allowable limit ($1.5S_m$). This code allowable stress level ($1.5S_m$) is used for the following three reasons:

1. SSE induced stresses need not be included in the secondary or peak stress evaluation.

2. SSE piping seismic levels are in the same order of magnitude as those associated with the OBE (operating basis earthquake) stresses.

3. In general, the stresses in the pipe will not be at the code allowable.

The stress calculated using equation (9) that results from the maximum pipe moment has been determined by subtracting the pressure stress from the allowable stress ($1.5S_m$) and multiplying by the ratio of stress index associated with high stress points (e.g., elbows). The maximum equation (9) moment stress is given in Table F below for the different pipe sizes.

TABLE F

Maximum Equation (9) Moment Stress

PIPE STRESS	ALLOWABLE $1.5S_m$	EQ (9) PRESSURE STRESS	EQ (9) MOMENT STRESS AT PIPE HANGER	EQ (9) TOTAL STRESS AT PIPE HANGER
4" Sch. 160	25.26 ksi	4.8 ksi	12.146 ksi	16.95 ksi
10" Sch. 40S	30 ksi	4.4 ksi	6.048 ksi	10.45 ksi
10" Sch. 80	30 ksi	2.6 ksi	9.23 ksi	11.83 ksi
32" MS 29	13 ksi	7.1 ksi	6.7 ksi	13.80 ksi

The equation (12) maximum mechanical piping stress is determined in a similar manner; allowable stress ($3S_m$) times the maximum ratio of stress indices values. The maximum equation 12 moment stress values at the pipe hanger locations are noted below.

TABLE G

Maximum Equation (12) Moment Stress

PIPE SIZE	ALLOWABLE STRESS	EQUATION 12 PIPING MOMENT STRESS
4" Sch. 160	50.52 ksi	22.49 ksi
10" Sch. 40S	60 ksi	10.63 ksi
10" Sch. 80	60 ksi	15.15 ksi
32" MS	58.26 ksi	13.34 ksi

The total stress intensity for each of the piping sizes evaluated is given in Table H.

TABLE H

	PRELOADED TORQUE	APPLIED STRUT LOAD	TOTAL STRESS INTENSITY
4" Sch. 160	60 ft-lbs.	2,000 lbs.	64.14 ksi
10" Sch. 40S	100 ft-lbs.	10,000 lbs.	74.21 ksi
10" Sch. 80	100 ft-lbs.	10,000 lbs.	54.69 ksi
32" MS	240 ft-lbs.	100,000 lbs.	47.17 ksi

Splitting the above total stress intensity into primary (equation 9) and secondary (equation 12) stress intensities results in the following (Table I).

TABLE I

Maximum Primary and Secondary Stress Intensities

	EQ. 9 (PRIMARY STRESS INTENSITY) (ksi)	EQ. 9 ALLOWABLE (ksi)	Eq. 12 (SECONDARY STRESS INTENSITY) (ksi)	EQ. 12 ALLOWABLE (ksi)
4" Sch. 160	31.60	50.52	32.54	50.52
10" Sch. 40S	60.61	60	13.6	60
10" Sch. 80	38.15	60	16.54	60
32" MS	30.57	58.26	16.6	58.26

As can be seen from a comparison of the above maximum stress intensities to the equation (9) and equation (12) allowable stresses, the 10" Sch. 80, 32" MS, and 4" pipes meet the stress criteria established herein. The 10" Sch. 40S pipe essentially meets all of the allowable limits.

Q. In the preceeding discussion you have used "general" piping moments at the support location. How do you know that they are representative of the actual piping moments? Moreover, in the presentation of the mechanical stresses reported in Attachment 3, it is stated that these general piping moment stresses are "realistic" values. Please compare the Code primary piping moment stresses (NB, NC and ND Equation 9) and secondary thermal piping moment stresses (NB Equation 12 and NC and ND Equation 10) to actual, randomly selected, computer piping analysis stresses.

A. In Table 3, summaries of stresses associated with straight runs of pipe for 4, 10 and 32 inch pipe are given. The piping lines and locations were randomly selected by Gibbs & Hill. The maximum stresses from this summary calculated using Equation 9 stresses for each of the pipe sizes given are compared to the Attachment 3 mechanical pipe stresses. This comparison is given in Table J.

TABLE J

Comparison Between Equation (9) Stresses Computed by Attachment 3 Method and Stresses Calculated by Gibbs & Hill

Attachment 3 Stresses

Pipe Size	Mechanical Primary Pipe Moment Stress (ksi)	Pressure Stress (ksi)	Primary Piping Moment Plus Pressure (ksi)
4" Sch. 160	12.146	4.8	16.95
10" Sch. 40	6.05	4.4	10.45
10" Sch. 80	9.23	2.6	11.83
32" MS	6.7	7.1	13.8

Gibbs & Hill Attachment - Stresses

Pipe Size	Randomly Selected Maximum Primary Piping Moment Plus Pressure (ksi)
4"	7.37
10"	7.063
32"	10.6

As seen from comparing the primary piping moment stresses presented in Attachment 3 with the randomly selected maximum primary pipe stresses presented in Table 3, the Attachment 3 stresses are all higher. The maximum (NC & ND) pipe stresses, calculated using equation 10, compiled at random straight run piping analysis locations (see Table 3) are compared to the Attachment 3 Code (NB) Equation 12 stresses in Table K below.

TABLE K

Comparison Between Attachment 3 Code (NB) Equation 12 Stresses and Gibbs & Hill Calculated Stresses

Attachment 3 Stresses

Pipe Size	Equation 12 Stress (ksi)
4" Sch. 160	22.49
10" Sch. 40	10.63
10" Sch. 80	15.15
32" Sch. MS	13.34

Gibbs & Hill Table 3 Code (NC & ND) Equation 10 Stresses

Pipe Size	Equation 10 Stress (ksi)
4" Sch. 160	3.6
10" Sch. 40	1.5
32" Sch. MS	5.0

From the above tables, it can be seen that the Attachment 3 values calculated using equation 12 are higher than the randomly selected equation 10 stresses compiled by Gibbs & Hill. Therefore, the primary and secondary stresses used in Attachment 3 (Code Equations 9 and 12) are conservative when compared to Gibbs & Hill's randomly selected cases.

Q. In Attachment 3, "realistic" mechanical pipe stresses were determined based on ASME Class 1 rules; how will these stresses vary considering ASME Class 2 and 3 rules?

A. Using Code Equations 9 and 12 requires that the mechanical stresses in the pipe resulting from sources other than the U-bolt assembly be included. Because of the number of hangers involved, this information was not supplied to Westinghouse for evaluation purposes.

In order to conservatively estimate the magnitude of these mechanical stresses, a generic procedure was developed and is discussed in Attachment 3. In the development of these mechanical stresses, it is assumed that the pipe is stressed to its maximum allowable value at a high stress intensification point (e.g., elbow). It is assumed that the piping moment which results in this maximum allowable stress also occurs at the U-bolt hanger locations. To obtain stresses due to piping moments, Class 1 rules are used in Attachment 3. If Class 2/3 rules are used to determine the piping moment stresses, the mechanical pipe moment stresses would be different since the stress inches (stress intensification factors for Class 2 and 3) are different.

Given below in Tables L and M is a summary of the mechanical pipe stresses based on ASME Class 2/3 (NC and ND of Section III) rules compared with those used to assess the pipe stresses in Attachment 3 based on Class 1 rules (NB of Section III).

TABLE L

Deadweight and Seismic Piping Moment Stresses

Pipe Size	Material	Class 1 Rules (ksi)	Class 2/3 Rules (ksi)
4" Sch.160	Stainless	12.146	14.3
10" Sch. 40	Stainless	6.05	8.6
10" Sch. 80	Carbon	9.23	11.2
32" MS	Carbon	6.7	9.1936

As seen from the above comparison, the difference in primary pipe moment stress using Class 1 and Class 2/3 rules is at most only 2.5 ksi, which is less than five percent of the $3S_m$ allowable limit used for piping stress assessment.

TABLE M

Thermal Piping Moment Stresses

Pipe Size	Material	Mechanical Pipe Class 1 Rules (ksi)	Moment Stresses Class 2/3 Rules (ksi)
4" Sch. 160	Stainless	22.49	26.5
10" Sch. 40	Stainless	10.63	10.7
10" Sch. 80	Carbon	15.15	12.29
32" MS	Carbon	13.34	13.02

Only the 4" Schedule 160 pipe has a calculated stress that is higher (18%) following Class 2/3 rules than for Class 1 rules. The Equation 12 piping moment stresses for the 10" Schedule 40, 10" Schedule 80 and 32" MS are lower or comparable to the Equation 10 stresses developed using Class 2/3 rules.

In conclusion, the mechanical pipe moment stresses used in Attachment 3 are conservative or yield only slightly lower (<8% based on $3S_m$) stresses than those based on Class 2/3 rules. The mechanical pipe moment stresses given in Attachment 3 are adequate estimates for the mechanical pipe stresses to be used in the assessment of the local pipe stress state in the vicinity of the U-bolt piping support assembly.

Q. You have stated that Applicants adopted two alternate approaches to evaluate the acceptability of pipe stresses. In the first alternate approach, described previously, the total preload stresses were considered as a primary stress in NB code Equation 9 with a $3S_m$ limit. What is the effect on the piping stress evaluation if a $1.5S_m$ limit is used for Code Equation 9 and only the primary membrane portion of the U-bolt preload, push and pressure stress is considered? In other words what conclusions would you reach if you follow the second alternate approach? Here we present the result of Applicants' evaluation using the second alternative acceptance criteria (see p. 54 for first alternative).

Evaluation Using Second Alternative

A. To determine the primary membrane portion of the U-bolt preload, push and pressure stress, the stress state at the inside and outside of the pipe element surface is averaged. To these average stresses, the mechanical piping stresses were added. The resulting total stress state is calculated using Code Equation 9 and compared to the allowable stress ($1.5S_m$). The results are given in Table N below.

TABLE N

Pipe Size	Eq. 9 Stress (ksi)	$1.5S_m$ Allowable (ksi)
4" Sch. 160	20.99	25.26
10" Sch. 40	18.44	30.00
10" Sch. 80	15.33	30.00
32" MS	11.50	29.13

As seen from the above stress allowable tabulation, Code Equation 9 defined stresses are less than the $1.5S_m$ stress allowable. To complete the evaluation of the piping stress, Code Equations 10 and 12 (if Equation 10 is exceeded) must be evaluated using an allowable of $3S_m$. Code Equation 10 calculates the stress intensity which occurs in mechanical or thermal loadings which take place as the system goes from one load set, such as pressure, temperature, moment and force loading to any other load set which follows it in time. Loads which are noncyclic in nature need not be considered in Code Equation 10. The

maximum Equation 10 primary plus secondary stress intensity range for the four pipes evaluated is tabulated below in Table O.

TABLE O

Pipe Size	Eq. 10 Stress (ksi)	Eq. 10 Allowable ($3S_m$)
4" Sch. 160	50.8	50.52
10" Sch. 40	44.33	60.00
10" Sch. 80	39.83	60.00
32" MS	43.58	58.26

As seen from the above comparison, only stresses for the 4" Schedule 160 pipe slightly exceed (0.5%) the $3S_m$ limit. Two items must be noted concerning the conservatism of the stress state calculated using Equation 10 for the 4" Schedule 160. They are:

1. The dead weight stress of the pipe (1 to 1.5 ksi) is included. This should be removed from Equation 10.

2. Conservative values for the mechanical piping stresses (primary 12,146 ksi and secondary 22.49 ksi) have been used. The conservatism was previously demonstrated by comparison with piping moment stresses calculated and randomly selected by Gibbs & Hill.

If the conservatism reflected by these two items were removed, the Equation 10 stress for the 4" Schedule 160 pipe would be below the $3S_m$ limit. Since the Equation 10 stress allowable ($3S_m$) is not exceeded for the four pipe sizes, it is not necessary to consider stresses calculated using NB Code Equation 12.

In conclusion, as demonstrated from the above, the piping stresses in the vicinity of the U-bolt piping assembly will be acceptable.

Q. Please discuss the effect of U-bolt cinching on the fatigue usage factor associated with the pipe.

A. To ascertain further the possibility of failure under cyclic stresses caused by the clamping action, a fatigue assessment was performed to determine the possibility of initiating a crack in the pressure boundary at the vicinity of the U-bolt. This assessment included a calculation for Class 1 piping and a calculation for Class 2 and 3 piping. The fatigue calculation for Class 1 piping utilized the U-bolt stresses from the 4" Schedule 160 case. The calculation for Class 2 and 3 piping utilized the worst case peak stresses from the four pipe/U-bolt assemblies evaluated.

For the Class 1 piping, locations having the relative highest usage factors, excluding U-bolt considerations, were identified so that the maximum overall usage factor including the effect of the U-bolt could be determined. These locations were the 10" accumulator injection line and the 3" normal charging line. These two Class 1 piping locations represent the highest straight run fatigue usage factor (due to extremely severe thermal transient loadings) of all the Class 1 auxiliary piping.

The fatigue calculation performed included all design transients for these sections and added the maximum clamp induced stresses as represented by the 4" Schedule 160 case.

For the charging and accumulator lines the final usage factors considering U-bolt local stresses were .04 and .06, respectively. Based on an ASME Code allowable usage factor of 1.0 or 0.1 for pipe break requirements, the usage factor requirements are met.

For the Class 2 and 3 fatigue assessment, a simplified worst case fatigue calculation was performed.

In this calculation, a conservative peak stress (S_P) was determined using the maximum total stress intensity without preload. The effect of preload is not included since it is noncyclic and not required to be considered per the code in a fatigue evaluation.

The 4" Schedule 160 pipe has the largest peak stress:

$$S_P = 50.57 \text{ ksi}$$

Based on this stress, the shakedown criteria (allowable = $3S_m$ = 50.52 ksi) is met recognizing that S_P also includes deadweight stress which is noncyclic. Therefore, the simplified elastic/plastic damage factor equals one (Ke = 1.0). The alternating stress (S_{Alt}) is:

$$S_{Alt} = \frac{Ke\ S_P}{2} = 25.3 \text{ ksi}$$

The allowable number of cycles associated with S_{Alt} is 22,000 (Code Figure I-9); Ultimate Tensile Strength = UTS $<$ 80 ksi).

To determine the number of loading cycles, the worst of the following two cases were considered. For Case 1, 200 cycles were considered based on 200 cycles of heatup and cooldown, or for Case 2, 200 cycles were considered, based on 10 earthquakes, 20 significant cycles per earthquake.

The incremental usage factor is:

$$Ui = \frac{200}{22,000} = .009$$

The fatigue assessment just described was based on both a representative Class 1 method and a Class 2 and 3 method. It provides a measure of the fatigue damage that would be expected at the U-bolt location. This assessment indicates that the integrity of the pressure boundary, based on fatigue considerations, would not be significantly affected by the localized U-bolt effects. This assessment is over and above the ASME Code requirement for Class 2 and 3 piping components. The Code does not require any specific fatigue usage factor calculation for Class 2 and 3 piping other than controlling stress levels by the factor f.

Q. How would you interpret the results from the tests and finite element analyses regarding the stability of the U-Bolt assembly?

A. Stability of the U-bolt assembly is an issue that was addressed using the "push" applied load. For a "pull" load, the U-boat assembly, if it slipped, would be self-aligning and therefore not a stability concern. Stability for a "push" force requires that sufficient friction exists between the cross piece and pipe, and the U-bolt and pipe to "balance" an applied load which is not directed through the pipe centerline. Attachment 3 addresses the method by which the U-bolt assembly balances the "push" load. The results given in the report show that the U-bolt assembly must be evaluated as two coupled assemblies (U-bolt and cross piece) rather than one assembly which behaves rigidly. This is because slippage can occur at the pipe and cross piece interface and not at the interface of the U-bolt and pipe.

The results from the finite element analyses (using the minimum torque values) indicate that the U-bolt/pipe assembly will be stable. The minimum preload value used in the analyses is not the absolute since a determination of the absolute minimum value is an iterative process. No attempt was made to further define the absolute minimum torque value which could assure stability since the reported values are already generally representative of the lower bound value found in the field (with the exception of

the value of the 32-inch pipe). The minimum preload value that would insure stability can also be lower than the values given by finite element analysis since the pipe can roll on the cross piece until sufficient normal forces are developed between the U-bolt and pipe, along with friction forces, to balance the cross piece moment.

This physical mechanism in the cross piece and U-bolt interaction, not represented in the finite element analyses, would be self stabilizing. If the U-bolt assembly was pushed in the circumferential, direction, the tension would increase in one U-bolt leg and decrease in the other one resisting the applied load. As long as the load is not applied to the cross piece at an angle which exceeds the inverse tangent of the friction coefficient, the cross piece/U-bolt assembly will not slide. As stated previously this mechanism is not incorporated in the finite element model and therefore the minimum torque values determined by finite element analysis are larger than those that would be needed in reality.

From the finite element analyses performed and reported in Attachment 3 and observation of the tests, torque values to which cinched down U-bolts will be torqued during the previously mentioned inspection were established

for various sizes (Table P). (For other sizes, torque values in line with those set forth in Table P will be established.)

TABLE P

Torque Values for Inspection

Pipe Size	Torque Value
4" Sch. 160	25 ft-lbs.*
10" Sch. 40S	50 ft-lbs.
10" Sch. 80	50 ft-lbs.
32" Main Steam	250 ft-lbs.

* 25 ft-lbs. is a compromise between 9 ft-lbs. at which the finite element analysis predicts that the 4" assembly would be stable and the 35 ft-lbs. at which the test indicated percent contouring of the U-bolt to the pipe was achieved. Torquing of the U-bolts in the field indicates that for this size lines and U-bolt diameters, perfect contouring of the U-bolt to the pipe occurs at 20-25 ft-lbs.

The U-bolt piping support assembly can be stable for smaller torque values. A good practice for U-bolt installation is to insure that there is a good fit ("Snug" tight) between the U-bolt and pipe at torque values given above.

Q. Since the torque value for inspection (Table P) torque value for the 32-inch pipe is larger than the value at which the analyses for the piping stresses have been conducted, would there be an adverse effect on those stresses from this increase?

A. No. The piping stresses will remain within the established allowable since the piping stresses due to preload are small in the 32" main steam pipe (4.9 ksi). Therefore, the total primary stress would be about 36 ksi, which is still below the established allowable.

Q. What overall conclusion would you draw from the testing and analyses program?

A. As mentioned previously, the Applicant committed to an analytical and testing program that would address the Board's following concerns:

> 1. U-bolt cinching may not be an appropriate means of eliminating potential local instability without introducing adverse effects in the piping and the U-bolt itself.
>
> 2. There may be adverse long-term effects from U-bolts due to heat-up and cooldown and related friction on the pipe.
>
> 3. Stresses on the pipes caused by thermal expansion in local areas around cinched U-bolts may not be acceptable.

These concerns were addressed within this Affidavit, and it was shown that cinching of the U-bolts is a viable method of eliminating potential local instability of the U-bolt strut supports. There are no adverse effects in the piping or U-bolts themselves from the cinching process. This statement is based on the fact that the U-bolt is generally cinched to a preload value which is less than the

value used for test and analysis. Therefore, stresses existing in the field will be less than those obtained by the finite element analyses, which are shown to be acceptable, and also will be less than those obtained during testing, both of which were based on the highest value of pretorque.

With regard to stability, there is some concern that if the pretorque level is insufficient, i.e., the U-bolt is not snug tight, that the assembly might work loose and walk. This doesn't mean that the support is unstable in the sense of not being able to carry load. (The test program showed, for example, that when the 10-inch Schedule 40S assembly was torqued to 20 ft-lbs., the U-bolt/cross "walked" both axially and radially, but was still transmitting and carrying load.) However, as previosuly noted, Applicants will inspect every U-bolt on a single strut or snubber (a total of 380) and assure that each U-bolt is torqued to the torque values set forth in Table P.

The specific points that were made pertaining to the Board's concerns demonstrating that cinching of the U-bolts were the following:

1. Issue - U-bolt cinching will eliminate potential support instability.

Response - It was demonstrated by both testing and finite element analyses that when the U-bolt is cinched to a minimum preload value, the support will be stable.

2. Issue - Cinching of the U-bolt will cause adverse effects in the piping and U-bolt itself.

   Response - Stress results obtained from the finite element analysis of the U-bolt support piping assembly associated with the anticipated support and piping loads as well as recommended preload values are within acceptable limits. The support as well as pipe will not experience any gross distortion or loss of function.

3. Issue - There may be adverse long term effects from U-bolts due to heat up and cooldown.

   Response - Both test programs and analyses have demonstrated that this is not a valid concern.

Robert C. Iotti

Sworn to before me this 22d day of June, 1984.

Notary Public
My Commission Expires May 31, 1987

John C. Finneran, Jr.

Sworn to before me this 22d day of June, 1984

Notary Public
My Commission Expires May 31, 1987

DIAMETER = 1/2"

TABLE 1

SUPPORT #	SS / CS	TYPE U-BOLT	# U-BOLT	Allowable Loads N	E	Actual Loads N	E	% of Allowable N	E
-WP-X-AB-182-007-3	SS	PUH 004	1	1090	1450	21	42	1.9	2.9

DIAMETER - 3/4

Support #	SS / CS	Type U-Bolt	# U-Bolt	Allowable Loads N	Allowable Loads E	Actual Loads N	Actual Loads E	% of Allowable N	% of Allowable E
1-RC-1-RB-044A-002-3	CS	PUH 060	1	1220	1623	T 18	T 18	1	1
1-CS-1-RB-008-009-2	SS	PUS 006	1	485	645	T 7	T 11	1	2
-CS-1-RB-019-013-2	SS	PUS 006	1	485	645	T 272	T 303	56	4
-DD-1-SB-003-011-3	SS	PUH 006	1	3620	4815	T 58	T 77	2	2
-FS1-2108-01-04 102-003-2	CS	PUH 006	1	1220	1623	T 59	T 118	5	7
1-MS-1-RB-006-018-2	SS	PUS 006	1	485	645	T 50	T 100	10	15
1-MS-1-RB-034-009-2	SS	PUH 006	1	1220	1623	T 17	T 34	1	2
1-PS-1-SB-003-001-2	SS	PUS 006	1	485	645	T 31	T 62	6	1
C-1-133-700-C82K	SS	PUH 006	1	1090	1450	295	271	27	18
-1-143-701-C82K	SS	PUH 006	1	1090	1450	197	201	18	13

DIAMETER = 1"

SUPPORT #	SS / CS	TYPE U-BOLT	# U-BOLT	Allowable Loads N	Allowable Loads E	Actual Loads N	Actual Loads E	% of Allowable N	% of Allowable E
H-BR-1-SB-003-002-3	SS	PUH 010	1	3620	4815	379 T	529 T	10	11
H-CC-1-SB-046B-009-3	CS	PUH 010	1	3620	4815	15 T	19 T	.4	.4
CS-1-162-703-C42K	SS	PUH 010	1	3230	5713	22	32	.68	.5
RC-1-903-712-C51K	SS	PUH 010	1	3230	4296	147	211	4.6	4.

Diameter = 1½"

Support #	SS / CS	Type U-Bolt	# U-Bolt	Allowable Loads N	Allowable Loads E	Actual Loads N	Actual Loads E	% of Allowable N	% of Allowable E
-CH-1-SA-026-013-3	CS	DUH 015	1	3620	4815	3^T	6^T	.08	.1

DIAMETER = 2"

SUPPORT #	SS / CS	TYPE U-BOLT	# U-BOLT	Allowable Loads N	Allowable Loads E	Actual Loads N	Actual Loads E	% of Allowa… N	…
H-CC-1-RB-002-001-3	CS	PUS 020	1	1220	1623	T 141	T 186	11	11…
1-CC-1-RB-005-006-3	CS	PUH 020	1	3620 585	4815 778	339 T 130 L	394 T 152 L	9 22	8… 19…
1-CC-1-RB-031-005-3	CS	PUS 020	1	1220	1623	T 13[illegible]	T 194	11	12…
1-CS-1-AB-127-011-2	CS	PUS 020	1	1220	1623	T 162	T 221	13	1…
H-CS-1-AB-040-009-2	SS	PUH 020	1	3620	4815	T 614	T 781	17	16…
1-DD-1-DG-004-011-3	CS	PUH 020	1	3620	4815	T 513	T 590	14	12…
1-DO-1-DG-013-030-3	CS	PUH 020	1	3620	4815	T 648	T 814	18	17…
1-MS-1-RB-101A-008-2	CS	PUH 020	1	3620	4815	T 469	T 555	13	1…
1-SW-1-SB-001B-006-3	CS	PUS 020	1	1220	1623	T 200	T 229	16	[illegible]
CS-1-104-718-C42K	SS	PUH 020	1	3230	5713	397	405	12	7…

DIAMETER = 3"

ICTT I Item 2

SUPPORT #	SS / CS	TYPE U-BOLT	# U-BOLT	Allowable Loads N	Allowable Loads E	Actual Loads N	Actual Loads E	% of Allowables N	% of Allowables E
CC-1-230-002-C53R	CS	PUS 030	1	2260	3006	860	1134	38	38
CC-1-271-008-C53R	CS	PUS 030	1	2260	3000	773	813	34	27
CS-1-049-703-A52R	SS	PUS 030	1	2260	3000	1942	2389	86	80
CS-1-074-039-S42R	SS	PUS	1	2260	3006	790	1000	34.9	33.2
CS-1-075-701-A42R	SS	PUS	1	2260	3006	853	1007	37.7	33.5
CS-1-242-005-S52R	SS	PUS	1	2260	3006	897	1047	39.6	34.8
CS-X-004-702-A33R	SS	PUS 030	1	2260	3005	692	709	30	24
CC-1-052-009-C41K	SS	PUH 030	1	5420	7209	3479	4960	64.19	68.8
RH-1-005-010-C42K	SS	PUS 030	1	2260	3006	874	1472	38.67	48.9
SI-1-176-002-S42R	SS	PUH 030	1	5420	7208	3643	5029	67	70

DIAMETER = 4"

SUPPORT #	SS / CS	TYPE U-BOLT	# U-BOLT	Allowable Loads N	Allowable Loads E	Actual Loads N	Actual Loads E	% of Allowables N	% of Allowables E
AF-1-097-026-S33R	CS	PUS 040	1	2260	3006	1337	1433	59	48
CS-1-076-003-A42R	SS	PUS	1	160	213	52	53	46.3	35.4
MS-1-026-004-S72R	CS	PUS 040	1	2260	3005	1653	1979	74	66
MS-1-026-008-S72R	CS	PUH 040	1	7540	10028	2568	3544	34	36
MS-1-027-002-S73R	CS	PUS 040	1	2260	3005	682	763	36	30
MS-1-027-037-S33R	CS	PUS 040	1	2260	3006	1155	1265	51.1	42.0
MS-1-028-035-S33R	CS	PUS 040	1	2260	3006	2229	2560	98.6	85.1
RC-1-018-016-C81R	SS	PUS 040	1	2260	3006	1793	1997	79.34	66.4
SI-1-178-008-S33R	SS	PUS 040	1	2260	3013	735	1001	33	33
SI-1-300-006-S22R	SS	PUS 040	1	2260	3013	886	1064	39	35

DIAMETER = 6"

SUPPORT #	SS / CS	TYPE U-BOLT	# U-BOLT	Allowable Loads N	Allowable Loads E	Actual Loads N	Actual Loads E	% of Allowables N	% of Allowables E
AF-1-025-005-S33R	CS	PUS 060	1	3620	4814	781	1003	22	21
CC-1-007-037-A63R	CS	PUS 040	1	2260	3006	1566	1710	69	57
CC-1-105-016-E33R	CS	PUS 030	1	3620	4815	663	887	18	18
CC-X-013-003-A43R	CS	PUS 060	1	3620	4815	1006	1141	28	24
CC-X-023-007-A43R	CS	PUH 060	1	7540	10028	2871	3227	38	32
CC-X-023-012-A43R	CS	PUH 060	1	7540	10028	3252	3631	43	36
CS-2-239-407-A42R	SS	PUS	1	3620	4814	1863	2025	51.5	42.1
CS-2-239-408-A42R	SS	PUH	1	7540	10028	2083	2409	27.6	24
CS-2-240-707-A42R	CS	PUS	1	3620	4814	1713	1925	47.3	40
RC-1-147-007-C81K	SS	PUH 060	2	7540	10028	4095	4515	27.16	22.5

DIAMETER = 8"

SUPPORT #	SS / CS	TYPE U-BOLT	# U-BOLT	Allowable Loads N	Allowable Loads E	Actual Loads N	Actual Loads E	% of Allowables N	% of Allowables E
AF-1-011-005-S33R	CS	PUH 080	1	7540	10028	3756	4730	50	47
C-2-049-706-A33K	CS	PUS	1	3620	4814	1185	1515	32.7	31.5
CS-2-063-412-A42R	SS	PUS	1	3620	4814	1174	1527	32.4	31.7
RH-1-020-003-S22K	SS	PUH 080	1	7540	10028	2285	2833	30.30	28.25
RH-1-063-007-S22K	SS	PUS 080	1	3620	4815	1226	1448	33.87	30
RH-1-064-005-S22R	SS	PUH 080	1	7540	10028	2945	3426	39.1	34.16
RH-1-064-006-S22K	SS	PUS 080	1	3620	4815	1179	1481	32.57	30.76
RH-1-064-010-S22R	SS	PUS 080	2	3620	4815	3078	3129	42.5	32.49
RH-1-064-011-S22R	SS	PUS 080	1	3620	4815	3319	3915	91.69	81.31
SI-1-090-008-C41K	SS	PUS 080	2	3620	4815	2559	3780	35.35	39.25

DIAMETER = 10"

SUPPORT #	SS / CS	TYPE U-BOLT	# U-BOLT	Allowable Loads N	Allowable Loads E	Actual Loads N	Actual Loads E	% of Allowables N	% of Allowables E
PH-1-024-007-S22R	SS	PUS 100	1	5420	7209	4496	5462	82.95	75.77
-1-079-007-S42K	SS	PUH 100	1	9920	13194	6682	8585	67.36	65.1
-1-093-009-S42K	SS	PUH 100	1	9920	13194	4561	6033	46.	45.73
W-1-129-040-A43K	CS	PUS 100	1	5420	7208	2363	2551	44	31
W-1-129-070-S43R	CS	PUS 100	1	5420	7208	2244	2914	41	40
W-1-129-071-S33R	CS	PUS 100	1	5420	7208	2164	2961	40	41
W-1-132-043-A43R	CS	PUS 100	1	5420	7208	4808	4825	89	67
W-1-173-029-A43R	CS	PUH 100	1	9920	13194	4114	4421	42	33
W-2-129-014-A43R	CS	PUS	1	9920	13193	6199	7101	62.5	53.8
W-2-129-018-A43R	CS	PUS	1	3620	4814	1821	2192	50.3	45.5

DIAMETER = 12"

SUPPORT #	SS / CS	TYPE U-BOLT	# U-BOLT	Allowable Loads N	Allowable Loads E	Actual Loads N	Actual Loads E	% of Allowables N	% of Allowables E
?C-2-126-010-F43R	CS	PUS	1	7540	10028	1960	2315	26	23
?T-1-008-004-S22K	SS	PUS	1	7540	10028	3704	5391	49.1	53.8
?C-1-069-002-C41R	SS	PUH 120	2	9920	13194	9090	12747	45.82	48.31
?H-1-001-007-C41K	SS	PUH 120	1	9920	13194	4590	7061	46	54
?H-1-004-006-S42K	SS	PUH 120	1	9920	13194	7610	9711	76.71	73.6
?F-X-032-018-F33R	SS	PUS 120	1	7540	10028	4665	9643	62	96
?1-1-031-041-Y32K	SS	PUS 120	1	7540	10028	3479	4439	46.14	44.27
?1-2-031-433-A32R	SS	PUS	1	7540	10028	2311	4342	30.6	43.3
?1-2-322-400-A42R	SS	PUH	1	9920	13193	3662	7376	36.9	56
?A-X-004-703-A73R	SS	PUH 120	1	9920	13194	5459	6631	55	50

DIAMETER = 14"

SUPPORT #	SS CS	TYPE U-BOLT	# U-BOLT	Allowable Loads N	Allowable Loads E	Actual Loads N	Actual Loads E	% of Allowables N	% of Allowables E
PC-1-135-004-C51K	SS	PUH 140	2	9920	13194	4772	7118	24.05	26.97
PC-1-135-010-C41K	SS	1½ Ø Rod as U-Bolt	2	55728	74118	14112	18656	12.66	12.58

DIAMETER = 16"

SUPPORT #	SS / CS	TYPE U-BOLT	# U-BOLT	Allowable Loads N	Allowable Loads E	Actual Loads N	Actual Loads E	% of Allowable N	% of Allowable E
C-1-158-702-A43R	CS	PUH 160	2	9920	13194	7119	8608	36	37.
C-2-158-409-A43R	CS	PUH	1	9920	13193	5376	6322	27	24
T-1-013-022-S42K	SS	FIG. 137	1	7540	10028	5898	9486	78.2	94
T-1-013-417-C82K	SS	PUS	2	7540	10028	1657	2416	22	24
T-1-014-015-S32K	SS	Special U-Bolt 1½ Ø Rod	1	23260	30900	4687	6783	20.2	22
T-1-014-022-S22K	SS	PUS	1	7540	10028	975	1395	13	14
H-1-007-004-S22K	SS	PUH 160	1	9920	13194	4035	5087	40.68	38.5
H-1-007-007-S22K	SS	PUS 160	1	7540	10028	1606	2023	21.3	20.1
H-1-008-005-S22K	SS	PUH 160	1	9920	13194	3462	4321	35.	32.
SI-1-326-002-S32R	X	PUS 160	2	7540	10028	14728	18458	49	46

DIAMETER - 18"

SUPPORT #	SS / CS	TYPE U-BOLT	# U-BOLT	Allowable Loads N	Allowable Loads E	Actual Loads N	Actual Loads E	% of Allowable N	% of Allowable E
FW-1-018-021-C72K	CS	PUS 180	2	9920	13194	7554	11987	38	45
SI-1-030-001-S32R	CS	Special U-Bolt 1½ Ø Rod	2	56498	75143	21253	23215	18.80	15.
SI-1-030-004-S32R	CS	PUS 180	1	9920	13194	2711	3140	27.33	23.

DIAMETER = 24"

Support #	SS / CS	Type U-Bolt	# U-Bolt	Allowable Loads N	Allowable Loads E	Actual Loads N	Actual Loads E	% of Allowable N	% of Allowable E
?C-1-008-006-A33R	CS	PUS 240	2	9920	13194	6266	8272	32	31
?C-1-020-001-A33K	CS	PUS 240	1	9920	13194	5200	6486	52	49
?C-1-028-701-A33R	CS	PUS 240	1	9920	13194	4927	6177	50	47
?C-1-051-700-A43K	CS	R21" ① PUS240 ②	2	9920	13194	8775	12927	44	49
?C-2-043-715-A33R	CS	PUS	1	9920	13193	5996	7267	60.4	55
?C-2-087-702-A33R	CS	1½ Ø SA-36	1	23260	30900	9473	10985	40.7	35.
?51-1-029-031-Y32R	SS	PUS 240	2	9920	13194	9751	11335	49	43
?51-1-029-033-Y32R	SS	PUS 240	2	9920	13194	9731	12476	49.05	47.
?51-1-029-701-S32R	SS	PUS 240	1	9920	13194	8527	9603	86	72.
?SW-1-010-700-A33R	CS	PUS 240	1	9920	13194	9321	9327	94	71

"DIAMETER = 30"

Support #	SS / CS	TYPE U-BOLT	# U-Bolt	Allowable Loads N	Allowable Loads E	Actual Loads N	Actual Loads E	% of Allowable N	% of Allowable E
SW-1-001-700-F33K	CS	PUS	2	9920	13194	14733	20633	74	78
SW-1-001-701-F33K	CS	2" Ø U-Bolt	2	41380	55035	26895	37580	32.5	34
SW-2-012-701-F33K	CS	1½ Ø SA-36	2	23260	30900	26117	31695	56.1	51.[illegible]

DIAMETER - 32"

Support #	SS / CS	TYPE U-BOLT	# U-BOLT	Allowable Loads N	Allowable Loads E	Actual Loads N	Actual Loads E	% of Allowable N	% of Allowable E
MS-1-001-003-S72R	CS	137SN III	1	67000	89120	55142	41388	82	47
MS-1-001-004-S72R	CS	137SN III	1	118742	157927	57009	94901	48	60
MS-1-001-005-S72R	CS	137SN III	1	147470	196136	72133	39912	49	20
MS-1-002-003-S72R	CS	137SN III	1	118742	157927	57609	45255	49	29
MS-1-002-005-S72R	CS	137SN III	1	147470	196136	73887	42973	50	2[illegible]
MS-1-003-003-S72R	CS	137SN III	1	118742	157927	56689	44549	48	28
MS-1-003-005-S72R	CS	137SN III	1	147740	196136	73584	42423	50	2[illegible]
MS-1-004-003-S72R	CS	137SN III	1	118742	157927	59548	47777	50	30
MS-1-004-005-S72R	CS	137SN III	1	67000	89120	30191	42784	45	48

TABLE 2

Observation of Torques in U-Bolts

Pipe Size	No. of U-bolts Sampled	Torque Range (ft-lbs.)
3/4"	4	4-8.5
1"	2	5-5.5
2"	14	4.5-20
3"	15	9-30
4"	31	7.5-60
6"	16	10-75
8"	18	8-70
10"	14	10-83
12"	8	26-87
14"	4	19.5-95
16"	13	25-126
18"	3	[illegible]5-117
20"	2	65-108
24"	9	30-85
26"	1	22
30"	4	55-112.5
40"	2	61-82.5

TABLE 3 - STRESSES IN STRAIGHT PIPE SEGMENTS

Gibbs & Hill, Inc. Job No. 2323 Client TUGCO/CPSES UNIT 1

Subject STRESSES ON STRAIGHT PIPE - 4, 10 & 32 INCH DIAMETER

Calculation Number — Sheet No. —

Revision	Original Issue	Date	Rev.	Date	Rev.	Date	Rev.	Date	Rev.	Date
Checking Method #		X		X		X		X		X
Preparer	[illegible]	5/14/84								
Checker										

RANDOM SELECTION OF STRESS RESULTS ON STRAIGHT PIPE

PROB. AB-1-	PIPE SIZE	SCH. t (in)	MATL	P psi	SYSTEM	NODE	NORMAL & UPSET STRESSES EQ.8	EQ. 9	EQ. 10	EQ. 11
19A	4	80 .337	SA 312 TP304	1750	SI	6	5938	7370	597	6535
37Z	4	40 .237	SA 312 TP304	325	CT	1503	1673	2367	2817	4490
52U	4	40 .237	SA 312 TP304	150	CS	1001	863	1323	1529	2392
64C	4	40 .237	SA 106 GR.B	150	CC	841	1276	2789	3557	4832
11A	10	40 .365	SA 312 TP304	50	AFW	65	869.	3773.	693.	1561
70	10	40 .365	SA 312 TP304	600	RH	2245	4875.	7063.	1034	5908.
163	10	40 .365	SA 312 TP304	325	CT	14	2405.	3761.	14.	2419.
67Z	10	40 .365	SA 106 GR.B	150	SW	643	1521	3344	1541	3062
4	32	32" OD t=1.25"	SA155 KCF70	1185	MS	44	7866	8947	5018	12844
2	32			1185	MS	23	8282	9098	2173	10456
1	32			1185	MS	12	7772	10309	1958	9731
3	32			1185	MS	93	8389	10617	1817	15207

REPORT NO. EQ&T-EQT-860

REVISION 0

COMANCHE PEAK STEAM ELECTRIC STATION

U-BOLT SUPPORT/PIPE TEST REPORT

WESTINGHOUSE ELECTRIC CORPORATION
P.O. BOX 355
PITTSBURGH, PA 15230

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5/17/84
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Table of Contents

I. INTRODUCTION AND SCOPE

Tests were carried out by Westinghouse under contract with Texas Utilities Generating Company (TUGCO) to investigate friction and stresses in piping/U-bolt support configurations in use at the Comanche Peak Steam Electric Station. More specifically, in accordance with TUGCO's request, tests were performed to investigate the following:

a. The range of friction coefficients developed between the pipe and the U-bolt support while the U-bolt is preloaded to specified levels.

b. The effects of hanger size and pipe material on the friction coefficient for a constant U-bolt preload.

c. The stresses and strains generated in the U-bolt.

d. The stresses and strains generated in the pipe.

e. The extent of long term effects such as normal piping vibration and cyclic thermal expansion on U-bolt preload.

To address the above items, Westinghouse developed a test program which consisted of six separate tests as shown below:

1. Torque versus preload test
2. Friction test
3. Load distribution/strain measurement test
4. Thermal cycling/thermal gradient test
5. Creep test
6. Normal vibration simulation test

The tests were performed under test conditions defined by TUGCO at the facilities of Westinghouse Advanced Energy Systems Division (WAESD) in Large, Pennsylvania during the time frame of March 1984 through May 1984 in accordance with procedures developed by Westinghouse and approved by TUGCO. These tests are described and the results are presented in Section V.

II. OBJECTIVE

The test program had two basic objectives: (1) Confirm whether a nonslip condition exists between the U-bolt support and the piping and (2) Provide test data to facilitate the correlation of test results with analytical models of the U-bolt support/piping.

III. TEST ITEMS

Test items were provided by Texas Utilities Generating Company and were delivered to the Westinghouse Advanced Energy Systems Division Engineering Laboratories located in Large, Pennsylvania. The following test specimens were provided by TUGCO for use in the test program:

Pipe Size	Material	Length	Number of U-Bolt Supports
32"	Carbon Steel	10' (10'-7")*	1
10"	Stainless Steel	10' (9'-9 3/4")*	2
10"	Carbon Steel	10' (9'-8 1/4")*	2
4"	Stainless Steel	10' (9'-9 1/2")*	2

U-bolts were identified by a unique number for their use on a specific pipe size prior to the initiation of the test program. (See Appendix II)

Insulation, struts, pins and clevises were also provided by TUGCO so that test configurations could simulate as closely as possible the actual field installation configurations.

IV. SUMMARY

Six separate tests were performed under this program as identified below:

Torque versus preload test: Preloads produced in the U-bolts were determined for the four pipe test specimens as a function of U-bolt torque. Pipe stresses resulting from U-bolt preloads were also determined. Results from this testing are presented in Section V.A., pages 6 through 23.

Friction test: The forces required to cause the U-bolt supports to slip on the pipes were determined for the four pipe test specimens at various increments of U-bolt torque. The coefficients of friction between the U-bolt supports and pipes were also determined over a defined range of U-bolt torques. Results from this testing are presented in Section V.B., pages 24 through 40.

Load distribution/strain measurement test: U-bolt loads and crosspiece stresses were determined for given U-bolt torques and support loads on the 10" stainless steel pipe test specimen. Results from this testing are presented in Section V.C., pages 41 through 49.

Thermal cycling/thermal gradient test: The effects of thermal cycling at elevated temperature on U-bolt preload were investigated on the 4" stainless, 10" stainless and 32" carbon steel test specimens. Results from this testing are presented in Section V.D., pages 50 through 63.

Creep test: The effects of high temperature as a function of time on the U-bolt preload were evaluated on the 4" stainless, 10" stainless and 32" carbon steel test specimens. Results from this testing are presented in Section V.E., pages 64 through 71.

Normal vibration simulation test: The effects of piping vibration on U-bolt preload were evaluated on the 10" stainless steel pipe test specimen. Results from this testing are presented in Section V.F., pages 72 through 76.

* Actual test specimen lengths are shown in parentheses.

The friction test was performed to confirm whether a nonslip condition exists between the U-bolt supports and the piping. The other five tests provided results used as a basis for development of analytical models of the U-bolt supports and piping.

V. TEST DESCRIPTION AND RESULTS

V.A. Torque Versus Preload Test

1.0 Purpose

The purpose of this test was: (1) measure the preload produced in the U-bolt below the U-bolt support crosspiece as a known amount of U-bolt nut torque was applied, and (2) measure the stresses in the pipe caused by the preload in the U-bolt. This testing was performed on four test specimens (pipes) using three designs of U-bolt type supports as specified and provided by TUGCO.

Pipe Test Specimens

32 inch carbon steel pipe
10 inch stainless steel pipe
10 inch carbon steel pipe
4 inch stainless steel pipe

U-bolt Designs

32 inch U-bolt type pipe support
10 inch U-bolt type pipe support*
4 inch U-bolt type pipe support

2.0 Test Configuration (see Photographs 1 through 4)

The four pipe test specimens described in Section 1.0 had a mounting plate welded to each end. These plates were then fastened to two floor mounted brackets using high strength fasteners. The carbon steel test specimens had a Steel Structures Painting Council No. 2 finish applied on a four foot section in the center of the pipe. Strain gauges were installed on the U-bolt, pipe and support crosspiece. (Note: No strain gauges were installed on the crosspiece of the 4" specimen.) The U-bolt support was then installed in the center of the test pipe with the nuts hand tightened.

* This design of U-bolt support was used on both the 10 inch carbon and 10 inch stainless steel pipes.

3.0 Instrumentation

Instrumentation used for these tests is identified below.

1. Strain gauge signal conditioning and readout devices
2. Seventeen strain gauges

The strain gauges were installed on each test specimen prior to testing (see Sketches 1 and 2). The above instrumentation was calibrated prior to use.

4.0 Loading Devices

Loading devices used for these tests are identified below.

1. Torque wrench

The torque wrench was calibrated prior to use.

5.0 Load Application

The load induced into the test specimen was due to the U-bolt nut torque. No other loads were applied. Tests were performed at laboratory ambient environmental conditions.

6.0 Test Description

After a specimen had been installed and the instrumentation calibrated, testing commenced. The test steps were the same for the four test specimens. The testing variable was the U-bolt nut torque range. The torque ranges for the U-bolts were defined by TUGCO as:

4 inch test specimen: 5 to 60 foot pounds in increments of 5 foot pounds.

10 inch test specimens: 10 to 100 foot pounds in increments of 10 foot pounds.

32 inch test specimen: 20 to 240 foot pounds in increments of 20 foot pounds.

With the test specimen U-bolt nuts hand tight, the strain gauges were zeroed. The U-bolt nuts were then torqued to the lowest value in the ranges defined above. Readings were recorded from each strain gauge.

When strain gauge data had been recorded, the U-bolt nuts were alternately (torque Leg 1 and then Leg 2) torqued to the next increment. Again the strains were recorded. This process was repeated for each test specimen up to and including the maximum torque as defined above.

7.0 Data Reduction

The test output consisted of U-bolt and piping strains. Using the strain data, U-bolt preloads were determined. The analytical method for determining U-bolt preload is contained in Appendix I. Longitudinal and circumferential pipe stresses were also determined using the elastic stress-strain relationships contained in Appendix I and the strain readings obtained from the pipe strain gauges.

8.0 Data Description/Presentation

Test output was reduced and plotted so results could be presented graphically. U-bolt preload versus bolt torque was plotted for each of the four specimens in Figures 1 through 4. From the piping strain measurements, the maximum longitudinal or circumferential stresses in the pipes were determined. These values are plotted versus the U-bolt preload or bolt torque for each of the specimens in Figures 5 through 8.

9.0 Results

Results show that U-bolt preload increases proportionally to bolt torque. Preloads associated with the maximum applied bolt torques are:

- 4 inch stainless steel specimen: 5918 pounds at a bolt torque of 60 foot pounds. See Figure 1.

  A conforming fit between the U-bolt support and the pipe was achieved at a U-bolt torque of 35 foot pounds. Conforming fit is defined as the absence of visual gaps between the U-bolt support and the pipe.

- 10 inch carbon steel specimen: 8057 pounds at a bolt torque of 100 foot pounds. See Figure 2.

  A conforming fit between the U-bolt support and the pipe was achieved at a U-bolt torque of 70 foot pounds.

- 10 inch stainless steel specimen: 5444 pounds at a bolt torque of 100 foot pounds. See Figure 3.

  A conforming fit between the U-bolt support and the pipe was achieved at a U-bolt torque of 50 foot pounds.

- 32 inch carbon steel specimen: 4651 pounds at a bolt torque of 240 foot pounds. See Figure 4.

  A conforming fit between the U-bolt support and the pipe was not achieved during the Torque versus Preload Test even at the bolt torque of 240 foot pounds. A conforming fit

between the U-bolt support and the pipe was achieved during the Friction Test at a bolt torque of 1170 foot pounds.

Preload variation between the two legs of each U-bolt ranged from 16% for the 4 inch stainless steel specimen to 26% for the 32 inch carbon steel specimen.

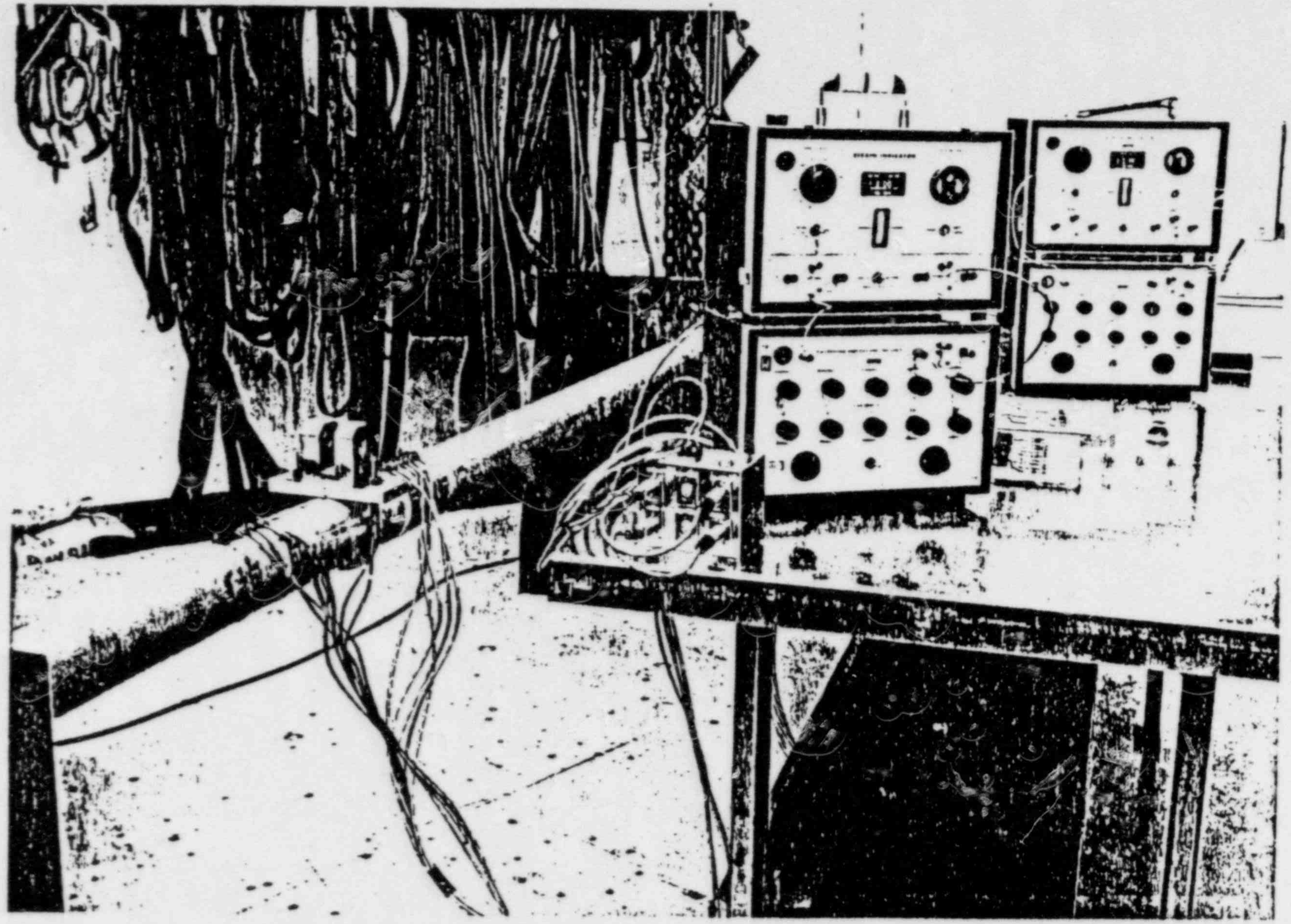
The pipe stresses associated with the maximum applied bolt torques were:

- 4 inch stainless steel specimen: The maximum stress was circumferential with a value of -7217 psi measured at strain gauge S8. See Figure 5.

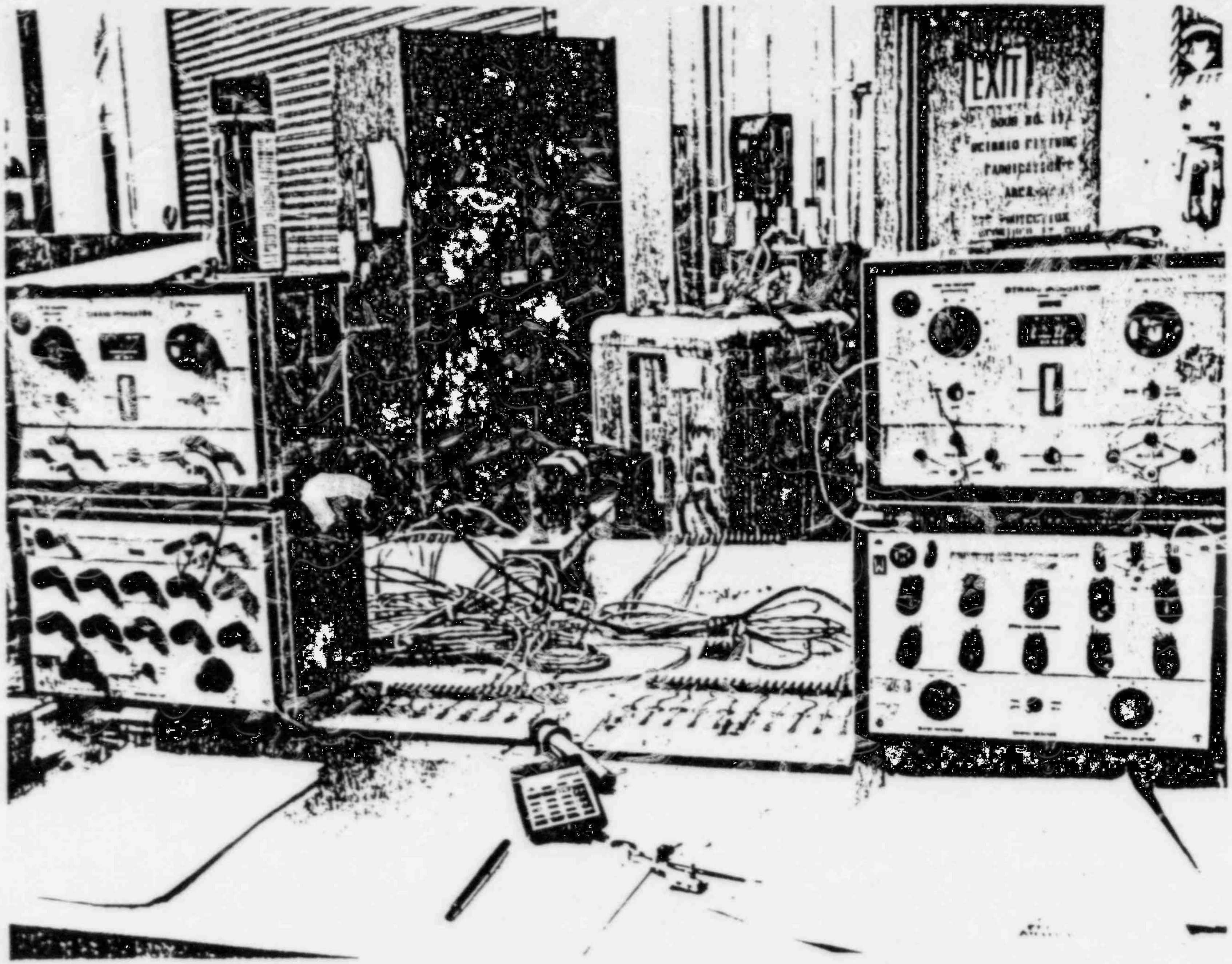
- 10 inch carbon steel specimen: The maximum stress was circumferential with a value of -8063 psi measured at strain gauge S9. See Figure 6.

- 10 inch stainless steel specimen: The maximum stress was circumferential with a value of -19,386 psi measured at strain gauge S12. See Figure 7.

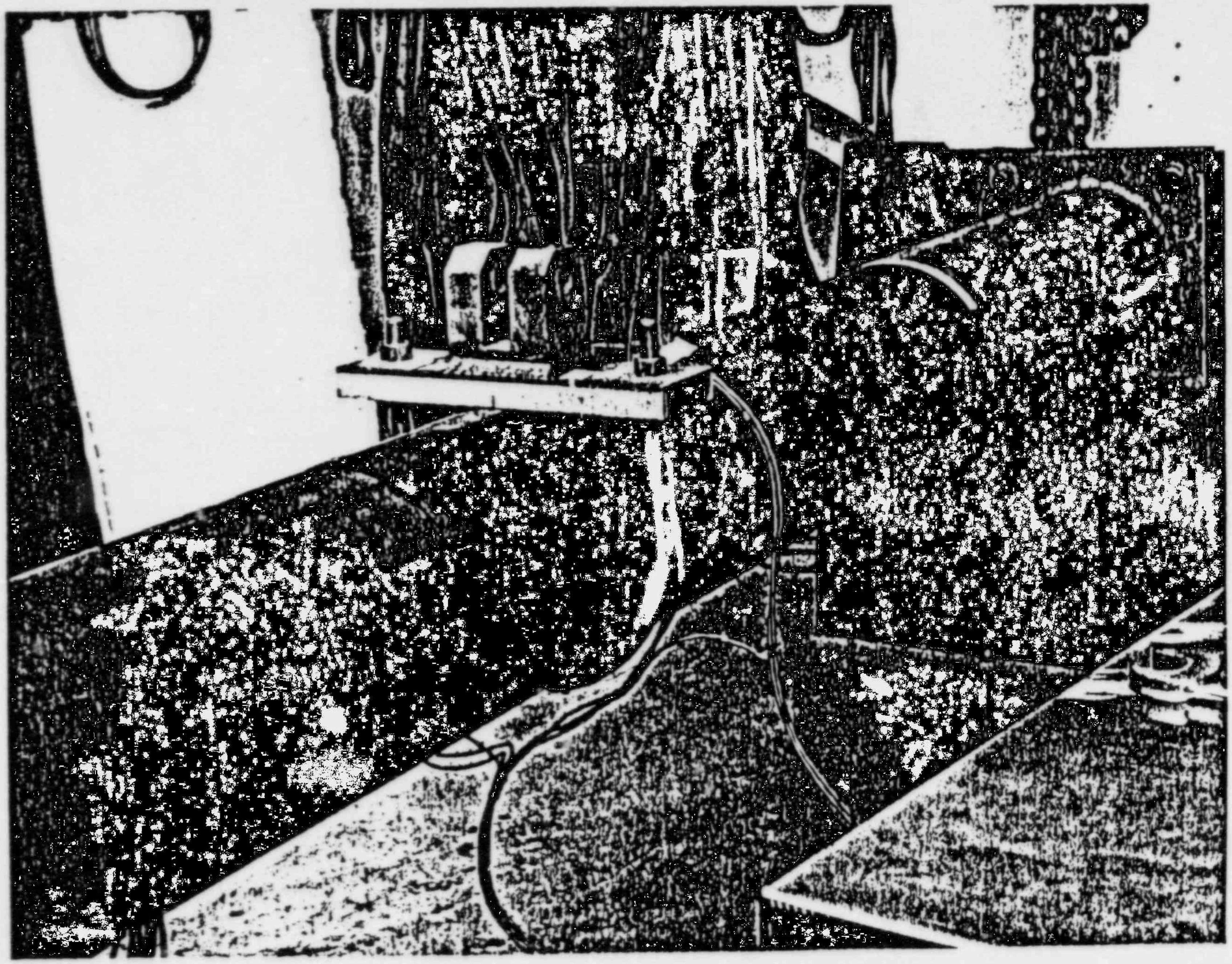
- 32 inch carbon steel specimen: The maximum stress was circumferential with a value of -1562 psi measured at strain gauge S9. See Figure 8.



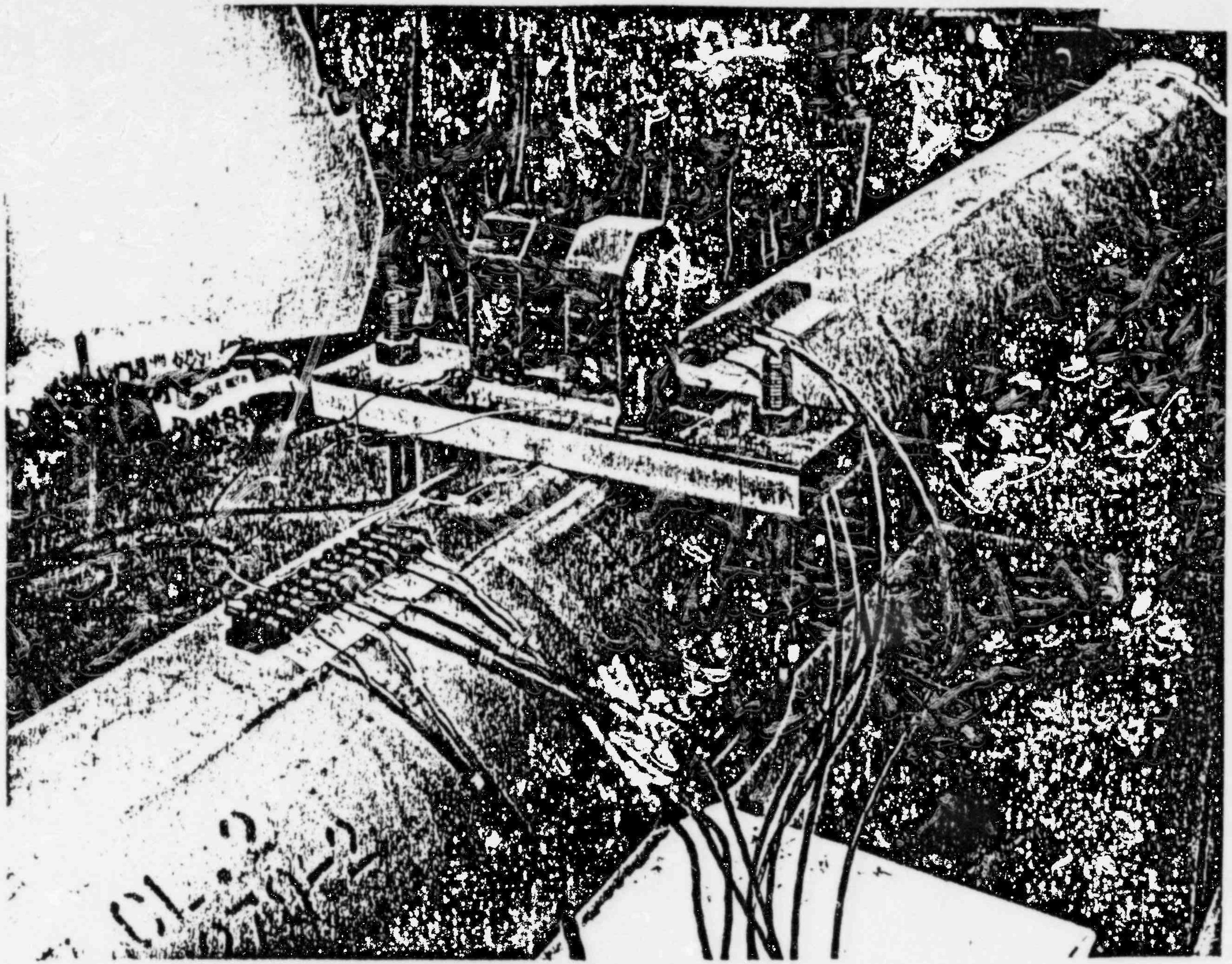
TORQUE VERSUS PRELOAD TEST
4" STAINLESS STEEL SPECIMEN
PHOTOGRAPH 1



TORQUE VERSUS PRELOAD TEST
INSTRUMENTATION
PHOTOGRAPH 2



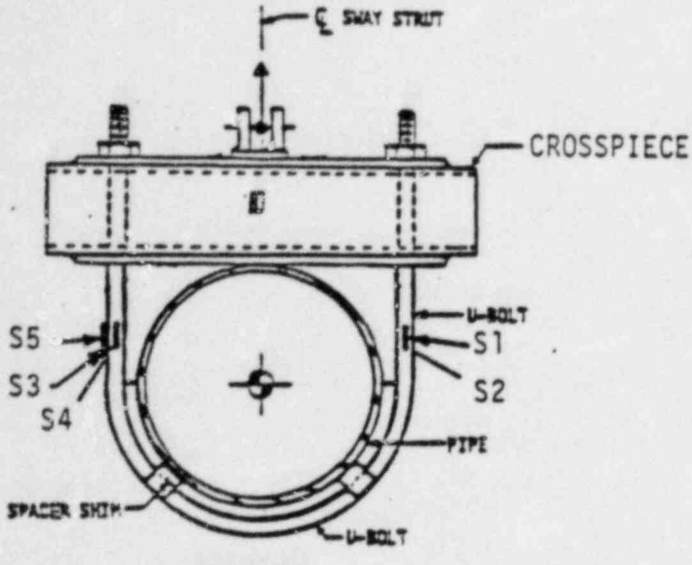
TORQUE VERSUS PRELOAD TEST
10" CARBON STEEL SPECIMEN
PHOTOGRAPH 3



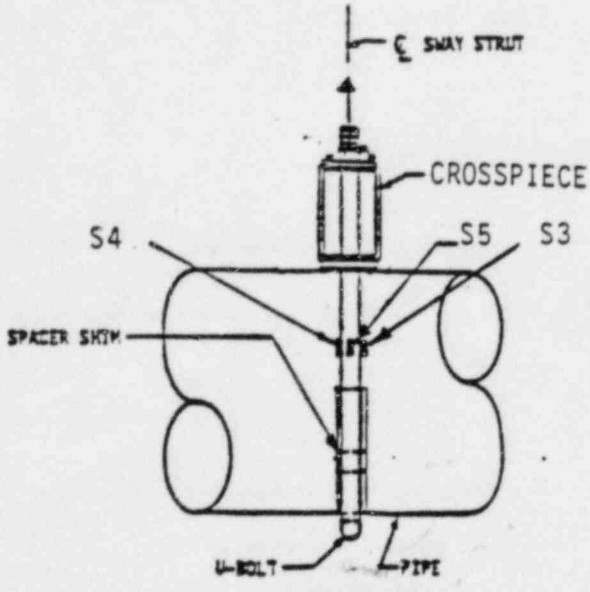
TORQUE VERSUS PRELOAD TEST

10" STAINLESS STEEL SPECIMEN

PHOTOGRAPH 4



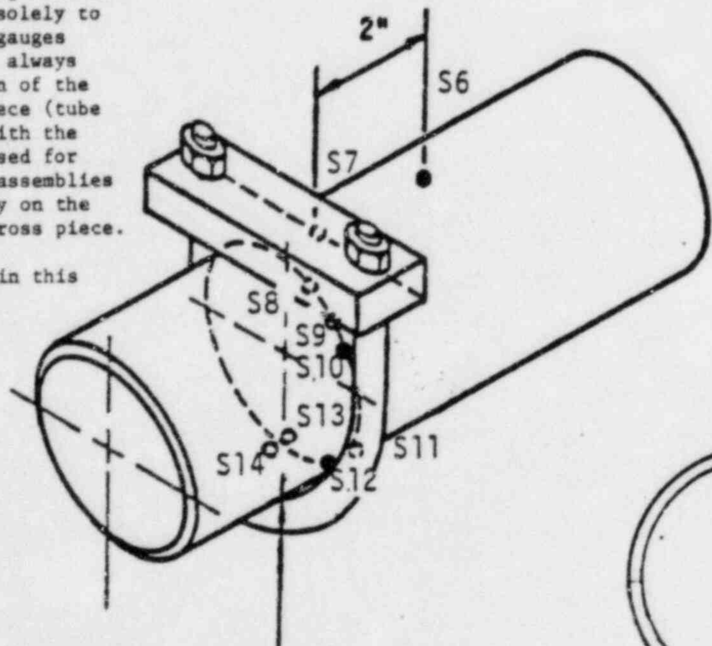
SECTION VIEW

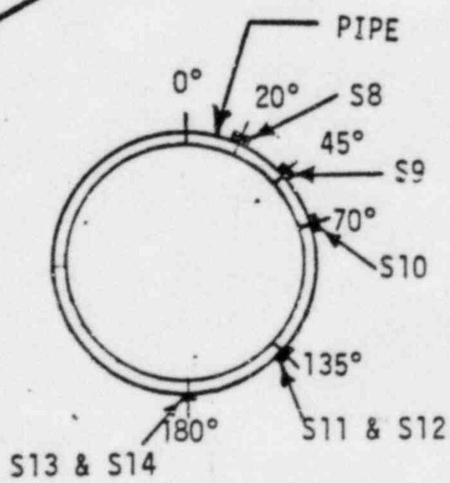


ELEVATION VIEW

The pipe, U-bolt, cross piece configurations shown in this sketch are provided solely to illustrate the location of strain gauges and/or thermocouples. They do not always illustrate the actual configuration of the assemblies. The built-up cross piece (tube steel plus plate) and the U-bolt with the load spacer curved plate is only used for the 32" assembly. The 4" and 10" assemblies consist of a U-bolt placed directly on the pipe and a thick steel plate for cross piece.

This note applies to all sketches in this document.



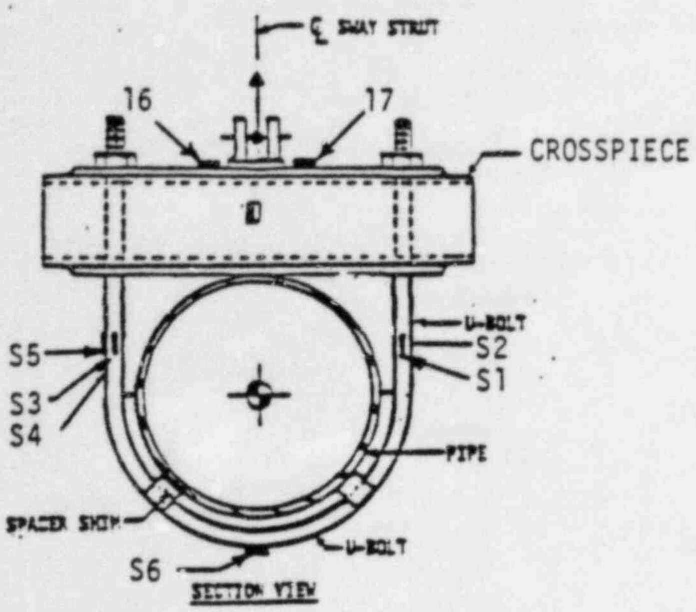


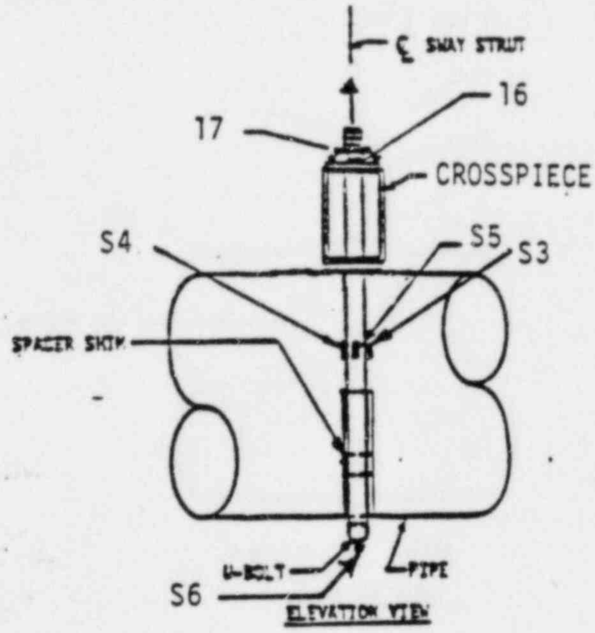
STRAIN GAUGE NUMBER	ACTIVE GAUGE LENGTH
S1-S14	0.25"

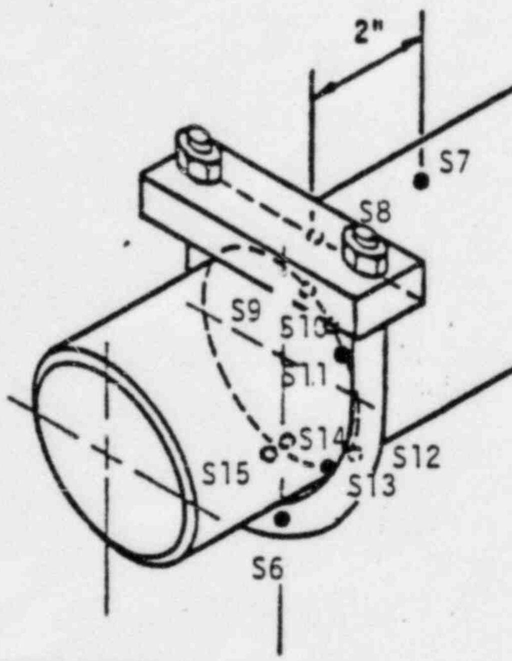
TORQUE VS. PRELOAD TEST

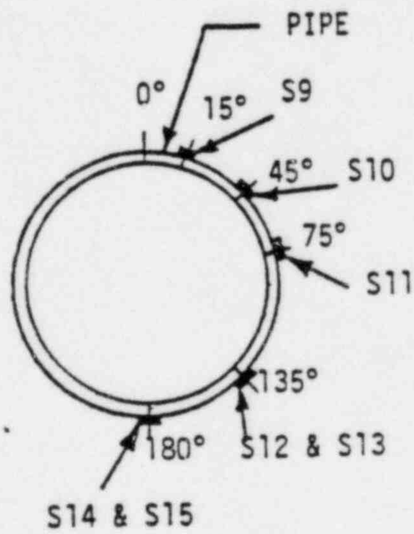
4" SPECIMEN STRAIN GAUGE LOCATIONS

SKETCH 1









STRAIN GAUGE NUMBER	ACTIVE GAUGE LENGTH
S1-S17	0.25"

TORQUE VS. PRELOAD TEST

10" AND 32" SPECIMENS STRAIN GAUGE LOCATIONS

SKETCH 2

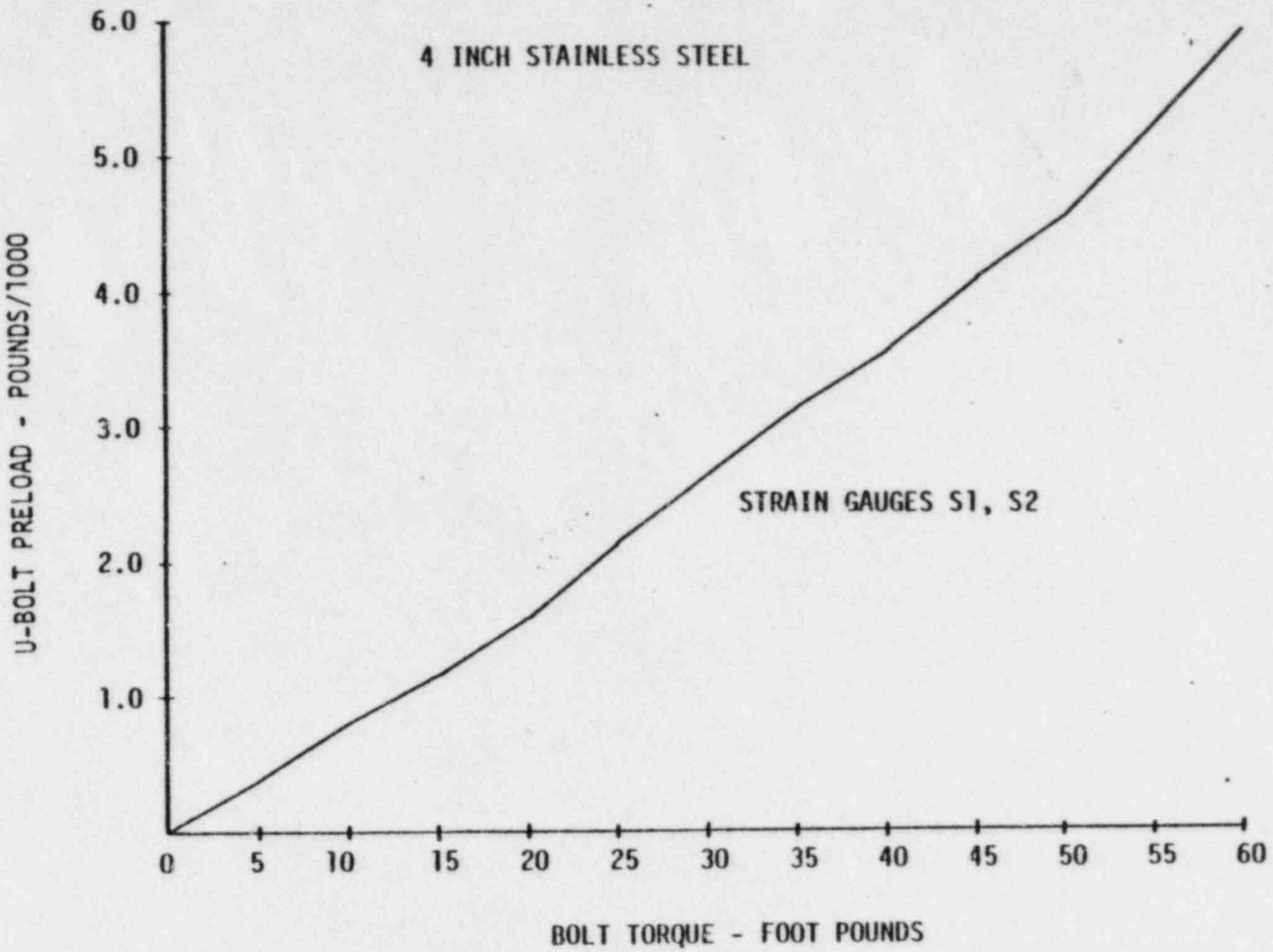

4 INCH STAINLESS STEEL
STRAIN GAUGES S1, S2
U-BOLT PRELOAD - POUNDS/1000
6.0
5.0
4.0
3.0
2.0
1.0
0
5
10
15
20
25
30
35
40
45
50
55
60
BOLT TORQUE - FOOT POUNDS

FIGURE 1

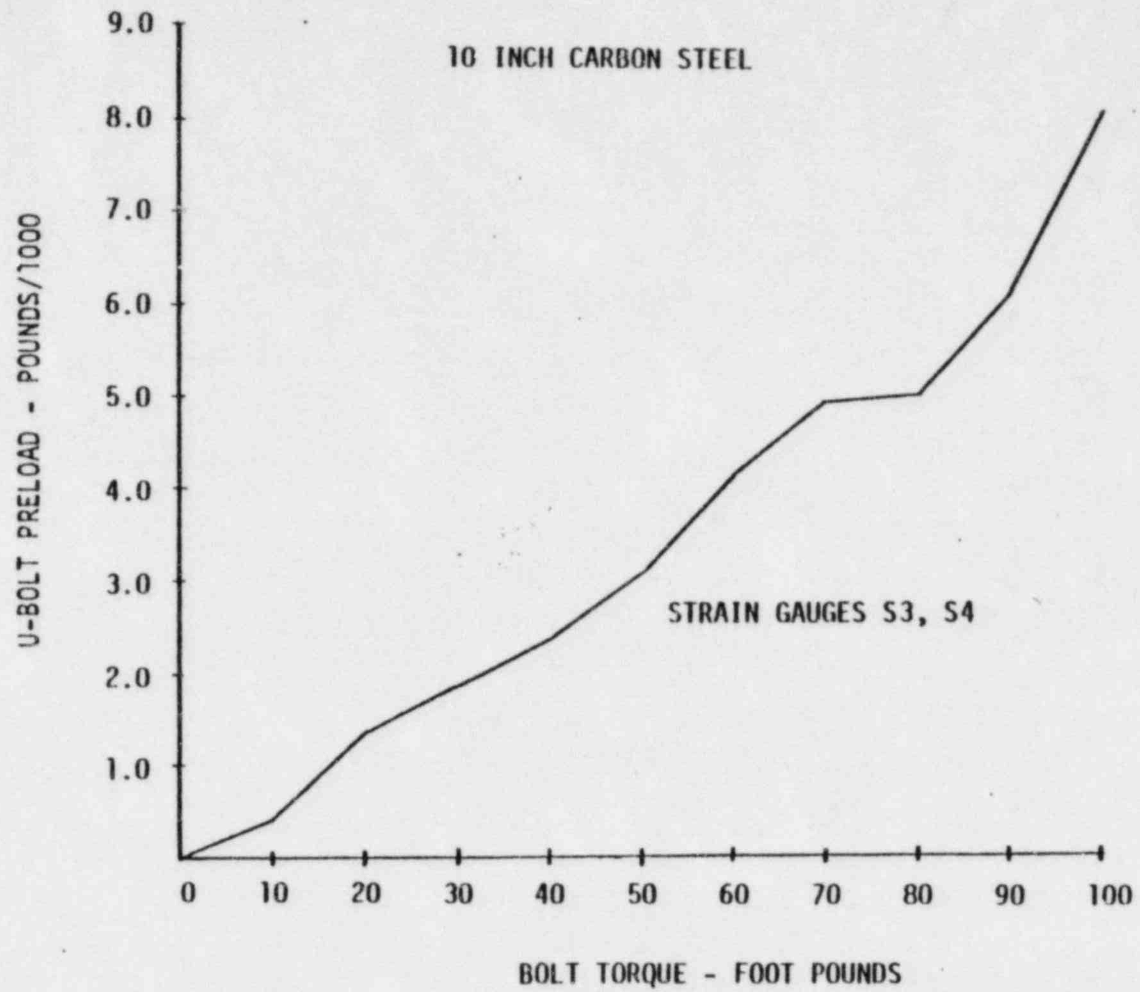

10 INCH CARBON STEEL
STRAIN GAUGES S3, S4
U-BOLT PRELOAD - POUNDS/1000
9.0
8.0
7.0
6.0
5.0
4.0
3.0
2.0
1.0
0
10
20
30
40
50
60
70
80
90
100
BOLT TORQUE - FOOT POUNDS

FIGURE 2

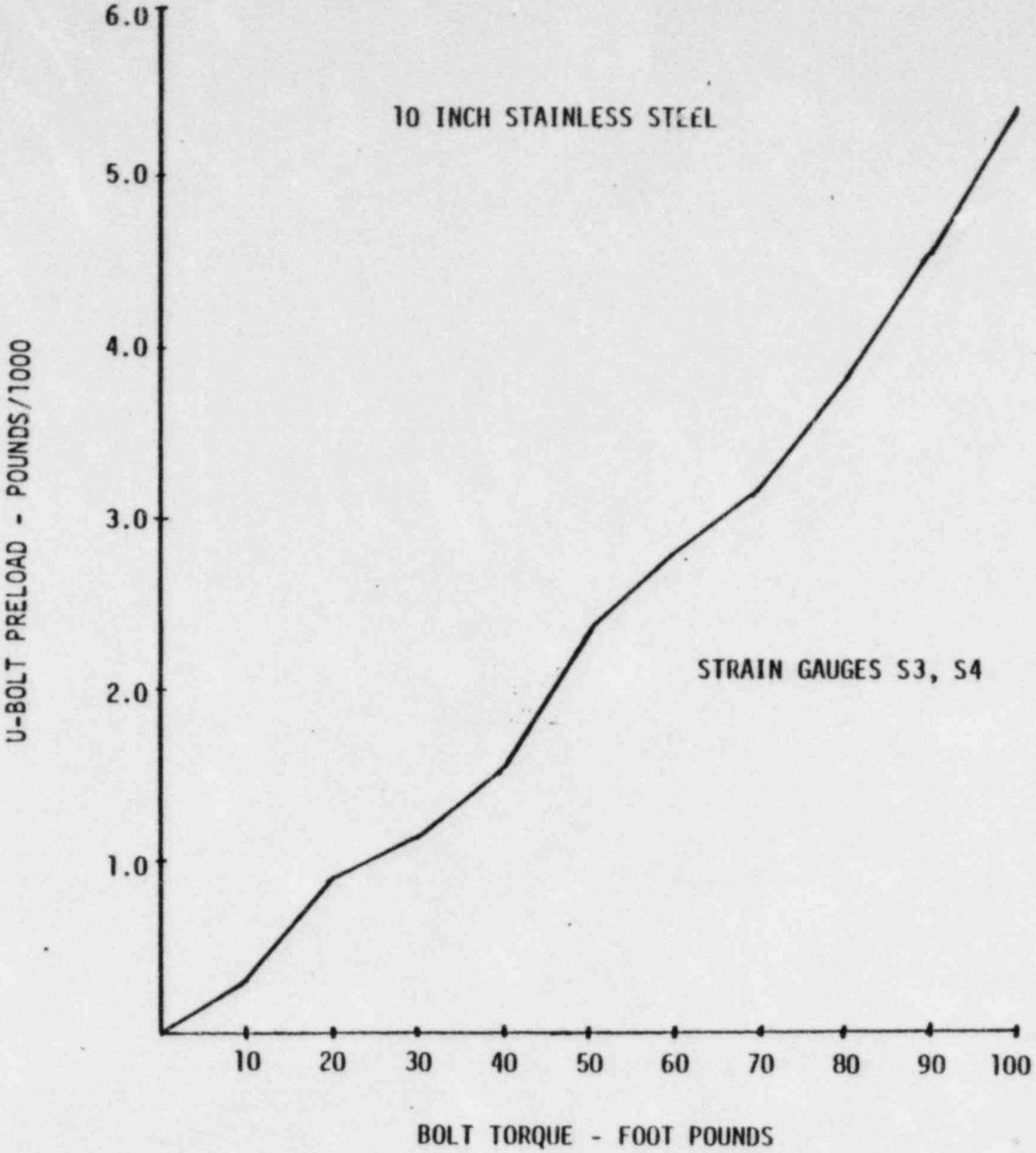

10 INCH STAINLESS STEEL
STRAIN GAUGES S3, S4
U-BOLT PRELOAD - POUNDS/1000
6.0
5.0
4.0
3.0
2.0
1.0
10
20
30
40
50
60
70
80
90
100
BOLT TORQUE - FOOT POUNDS

FIGURE 3

32 INCH CARBON STEEL

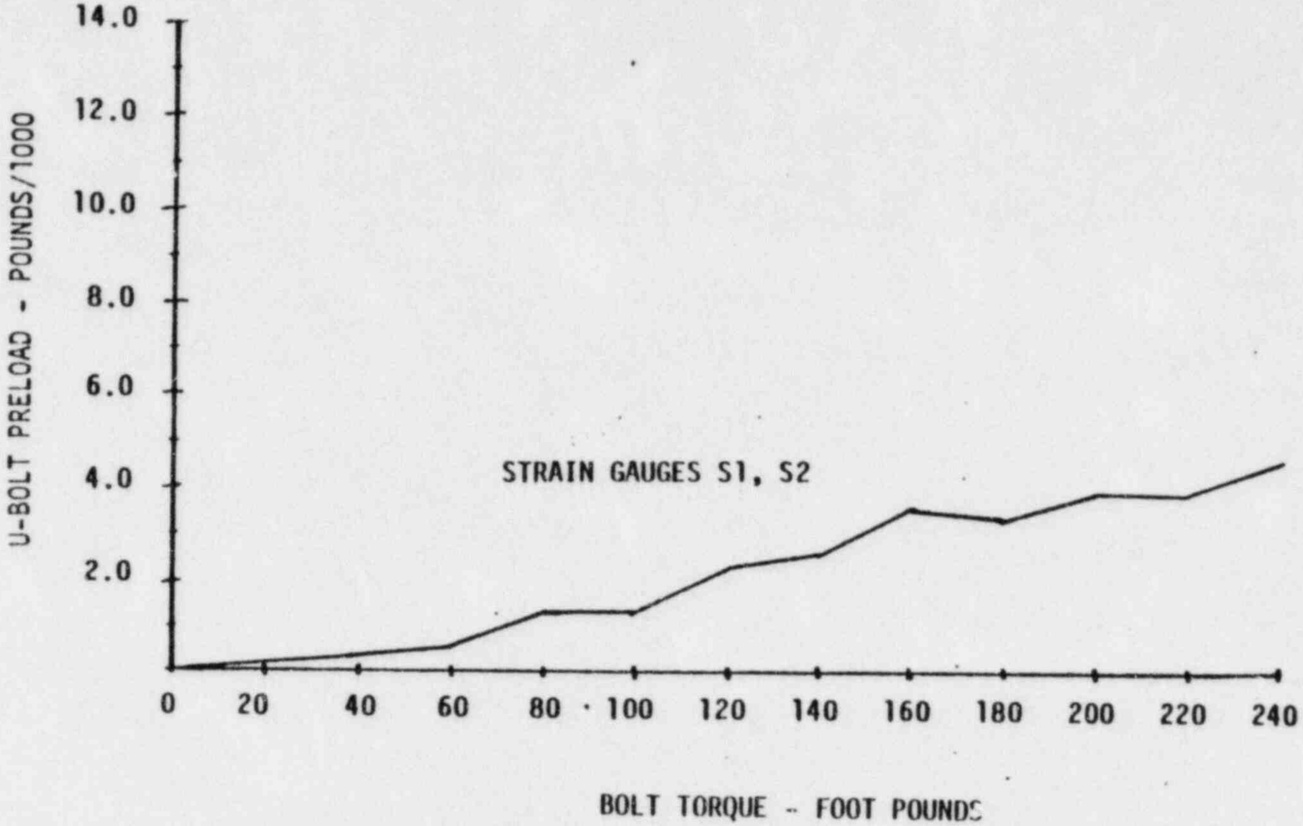


FIGURE 4

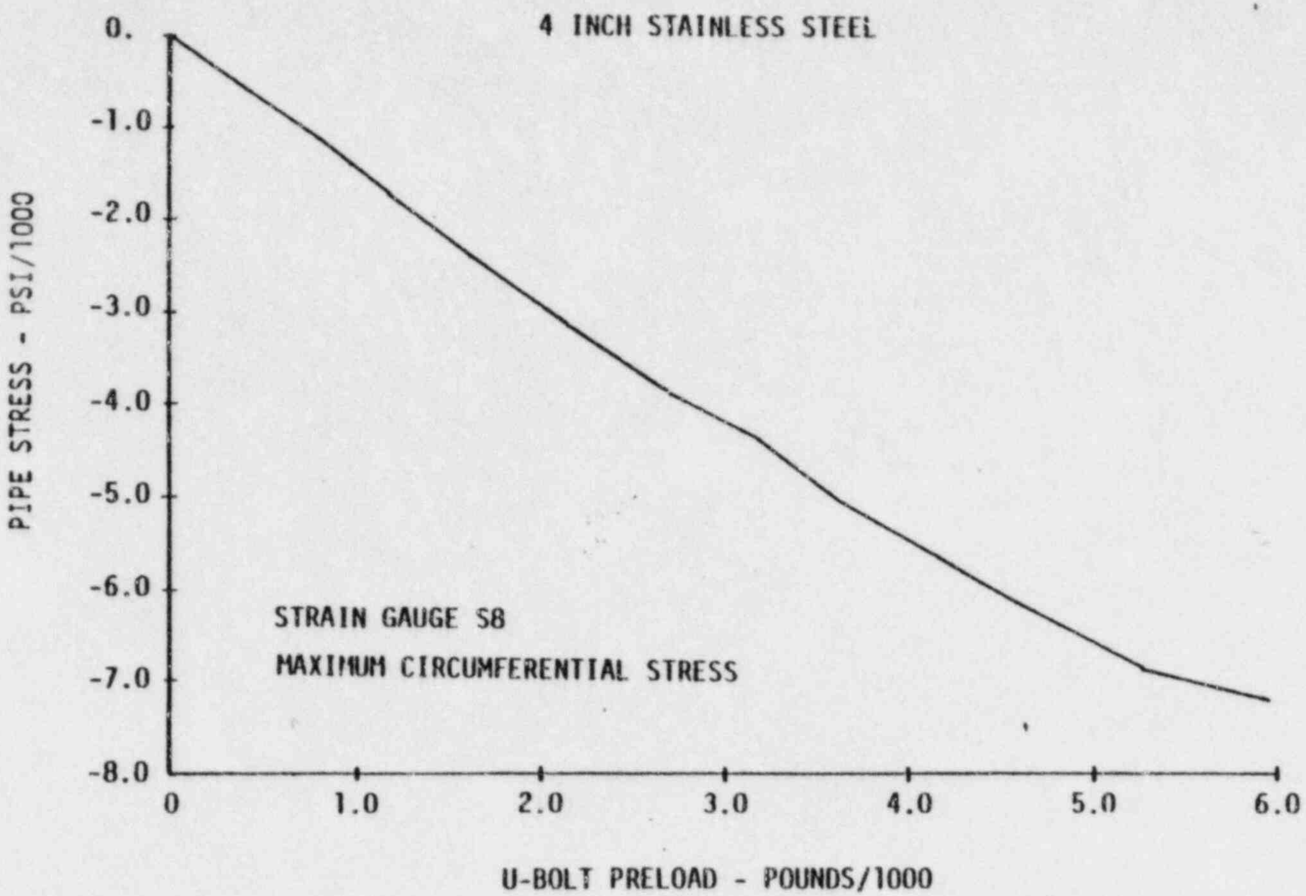

4 INCH STAINLESS STEEL
PIPE STRESS - PSI/1000
0.
-1.0
-2.0
-3.0
-4.0
-5.0
-6.0
-7.0
-8.0
STRAIN GAUGE S8
MAXIMUM CIRCUMFERENTIAL STRESS
0
1.0
2.0
3.0
4.0
5.0
6.0
U-BOLT PRELOAD - POUNDS/1000

FIGURE 5

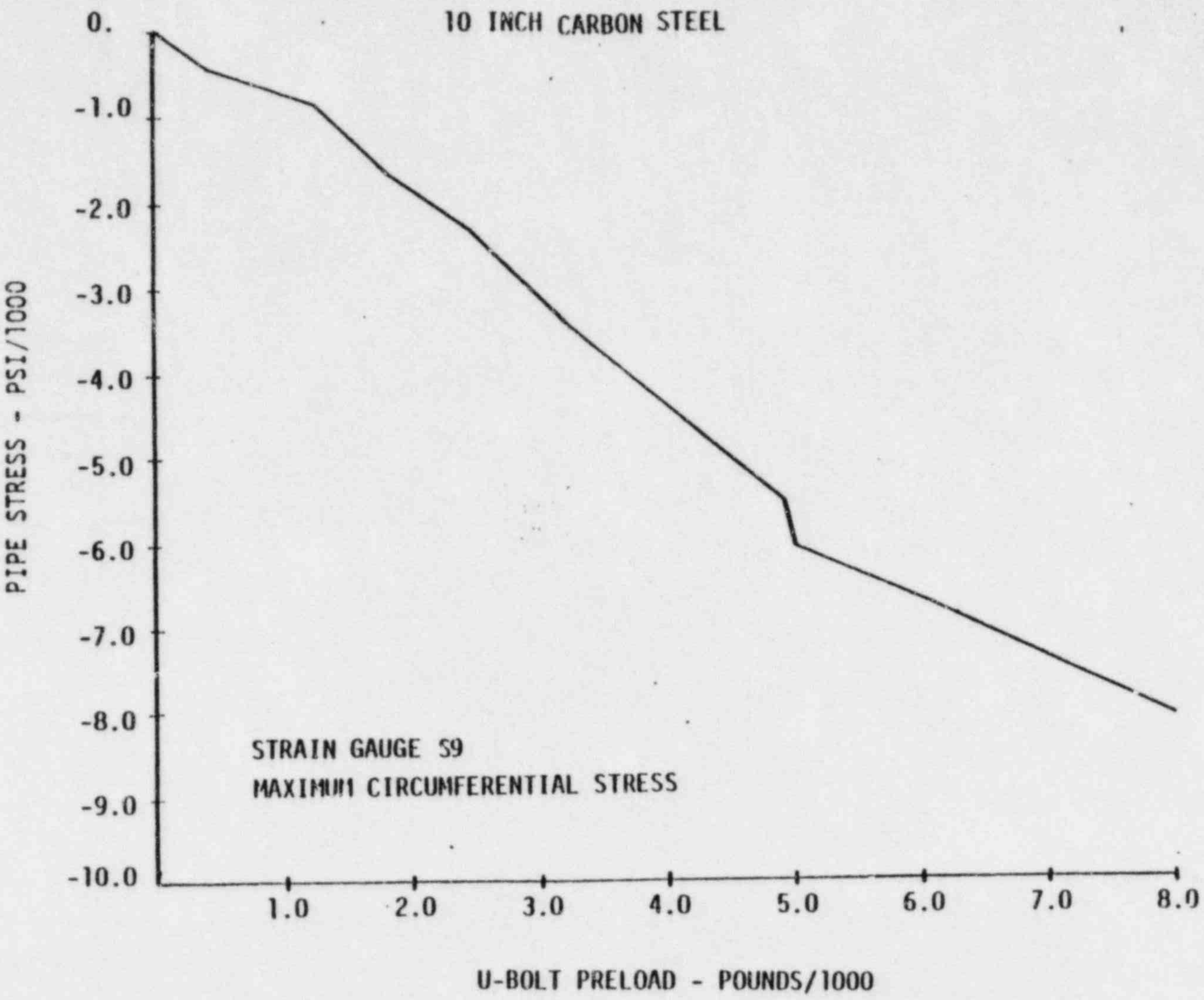

10 INCH CARBON STEEL
STRAIN GAUGE S9
MAXIMUM CIRCUMFERENTIAL STRESS
PIPE STRESS - PSI/1000
0.
-1.0
-2.0
-3.0
-4.0
-5.0
-6.0
-7.0
-8.0
-9.0
-10.0
1.0
2.0
3.0
4.0
5.0
6.0
7.0
8.0
U-BOLT PRELOAD - POUNDS/1000

FIGURE 6

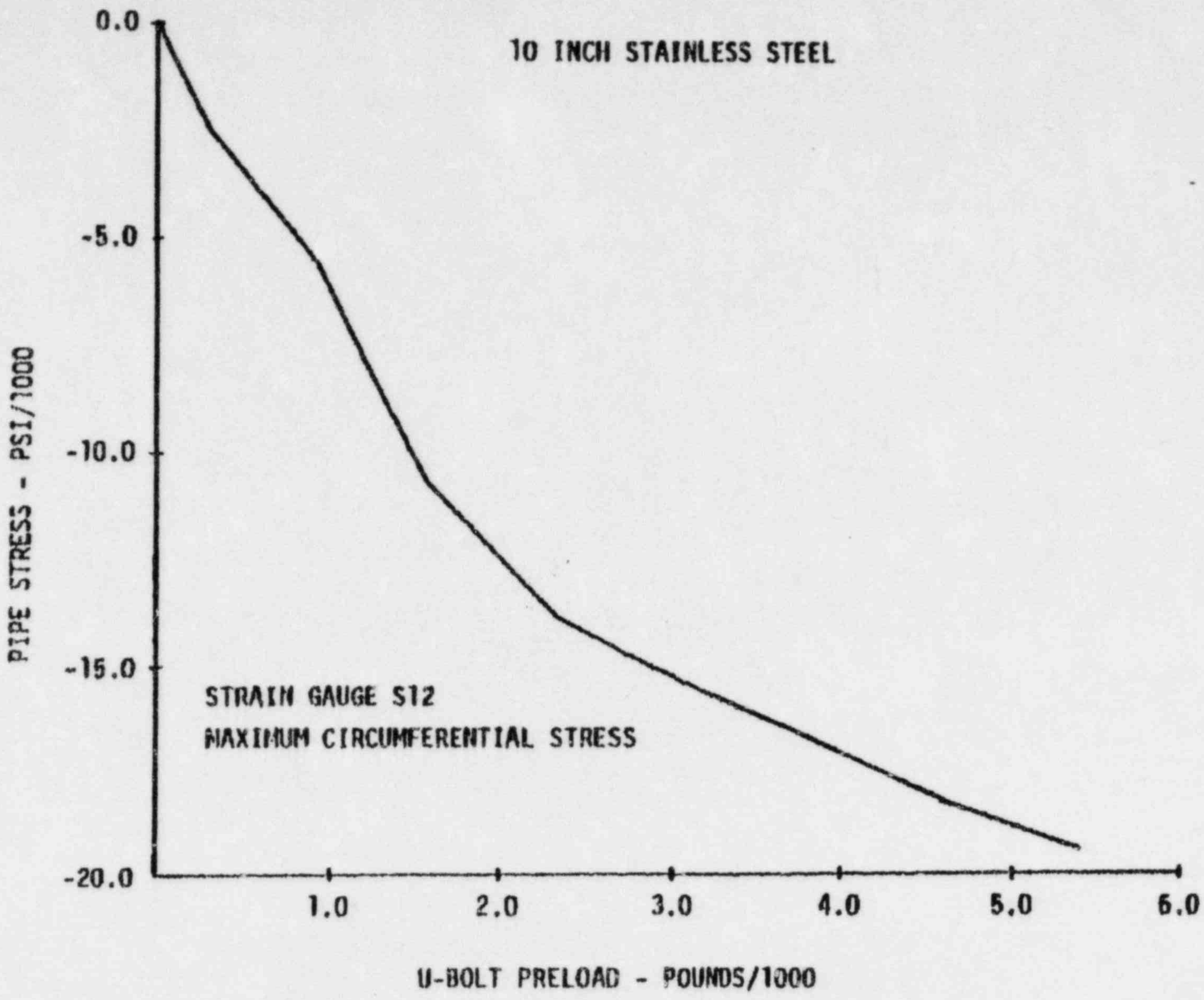

10 INCH STAINLESS STEEL
STRAIN GAUGE S12
MAXIMUM CIRCUMFERENTIAL STRESS
PIPE STRESS - PSI/1000
0.0
-5.0
-10.0
-15.0
-20.0
1.0
2.0
3.0
4.0
5.0
6.0
U-BOLT PRELOAD - POUNDS/1000

FIGURE 7

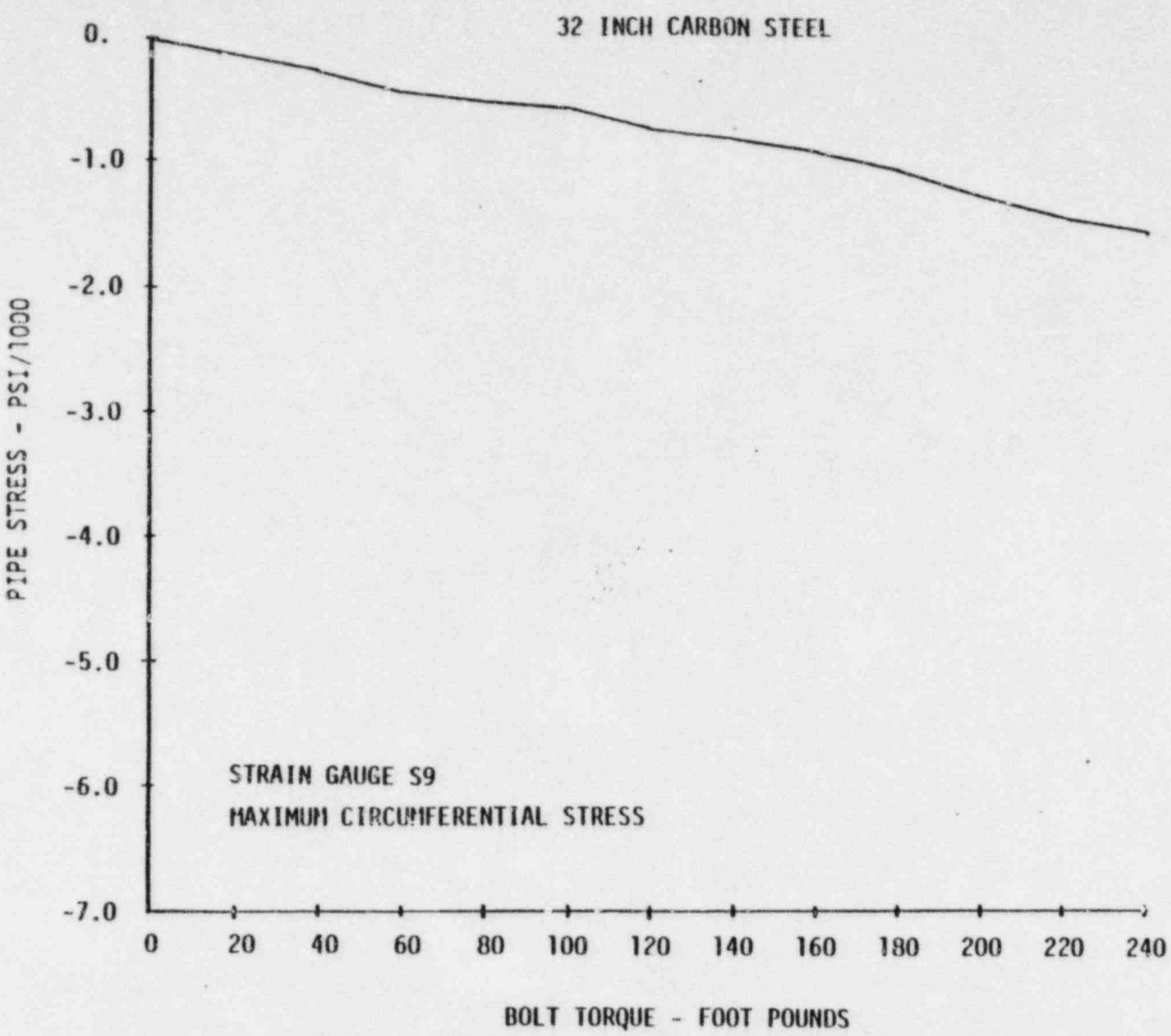

32 INCH CARBON STEEL
0.
-1.0
-2.0
-3.0
-4.0
-5.0
-6.0
-7.0
PIPE STRESS - PSI/1000
STRAIN GAUGE S9
MAXIMUM CIRCUMFERENTIAL STRESS
0
20
40
60
80
100
120
140
160
180
200
220
240
BOLT TORQUE - FOOT POUNDS

FIGURE 8

V.B. Friction Test

1.0 The purpose of this test was to measure the coefficient of friction between the U-bolt type support and pipe while the torque in the U-bolt was varied from a minimum to a maximum level. The maximum U-bolt torque level was defined as the one that could develop a U-bolt thread stress of ninety percent of yield* during the test. This limiting criterion was established to preclude yielding of the U-bolts. This permitted use of the same U-bolts in subsequent tests. The data obtained from this test was also used to define the relationship between the force that causes the U-bolt suppport to slip on the pipe and U-bolt torque. Tests were performed on four test specimens (pipes) using three designs of U-bolt type supports as specified and provided by TUGCO.

Pipe Test Specimens

32 inch carbon steel pipe
10 inch stainless steel pipe
10 inch carbon steel pipe
4 inch stainless steel pipe

U-Bolt Designs

32 inch U-bolt type pipe support
10 inch U-bolt type pipe support
4 inch U-bolt type pipe support

Note: Two (2) U-bolt type supports were tested on both 10 inch pipes and the 4 inch pipe. The second support was used to evaluate polishing phenomena. Only one 32 inch specimen was available for testing. The effects of polishing on the results obtained for this specimen are discussed in Section 9.0 below.

*Yield was defined as minimum yield for A-36 i.e. 36,000 psi.

2.0 Test Configuration (See Photographs 5 through 7)

The four pipe test specimens described in Section 1.0 had a mounting plate welded to each end. These plates were then fastened to two floor mounted brackets using high strength fasteners. The carbon steel test specimens had a Steel Structures Painting Council No. 2 finish applied on a four foot section in the center of the pipe. Strain gauges were installed on the U-bolt and crosspiece. The U-bolt was then installed in the center of the test pipe with the crosspiece perpendicular to the floor. The U-bolt nuts were initially hand tightened. A loading apparatus consisting of a hydraulic cylinder and a load cell was then installed.

3.0 Instrumentation

Instrumentation used for these tests is identified below.

1. Strain gauge signal conditioning and readout devices
2. Load cells
3. Magnetic tape recorder and brush recorders
4. Eight strain gauges

Strain gauges were installed on the U-bolt support (see Sketch 3). The above instrumentation was calibrated prior to use.

4.0 Loading Devices

Loading devices used for these tests are identified below.

1. Torque wrench
2. Hydraulic cylinder

The torque wrench was calibrated prior to use and the hydraulic cylinder load was controlled by the above referenced load cells.

5.0 Load Application

The test specimens were subjected to two types of loading during this test. Torque loads were applied in increments to the U-bolt nuts and the hydraulic cylinder/load cell was used to apply a load at the crosspiece clevis hole centerline tangentially to the pipe and parallel to the crosspiece. Tests were performed at laboratory ambient environmental conditions.

6.0 Test Description

After a specimen had been installed and the instrumentation calibrated, testing commenced. The test steps were the same for the four test specimens. The testing variables were the U-bolt nut torque increments and the load applied at the clevis to cause

slippage between the U-bolt support and pipe. The torque increments were defined by TUGCO as:

4 inch test specimen: 5 foot pound increments.

10 inch test specimens: 10 foot pound increments.

32 inch test specimen: 20 foot pound increments.

With the test specimen U-bolt nuts hand tight, the strain gauges were zeroed. The hydraulic cylinder/load cell was connected and positioned so that it was not loading the U-bolt in either the upward or downward vertical directions. The load cell reading was then zeroed.

The U-bolt nuts were then torqued to the first incremental value and the U-bolt strain recorded. With the recording equipment activated, the hydraulic cylinder load was increased until slippage occurred. Slippage was defined as a negative change in slope of the load time history.

The U-bolt strain in the leg opposite the loading was checked against the following maximum levels:

4 inch test specimen: maximum strain = 839 μ inches per inch.
10 inch test specimens: maximum strain = 877 μ inches per inch.
32 inch test specimen: maximum strain = 964 μ inches per inch.

The above strains correspond to U-bolt thread stresses equal to 90% of the material yield.

If the above maximum strains were not reached, the U-bolt nuts were loosened and the pipe support returned to its original position. The U-bolt nuts were then alternately retorqued to the next incremental torque value and the load applied until slippage occured. This sequence was repeated until the maximum strain levels were reached.

Once the maximum U-bolt strain levels were reached the U-bolt was loosened and returned to its original position. The U-bolt nuts were then retorqued to the same value and the friction test was repeated two more times. This procedure was performed to establish data repeatability.

After completing these tests, the U-bolt support was removed and a second U-bolt support was installed on the pipe several inches away from where the original had been positioned. The friction testing was repeated with the U-bolt preloaded to a strain value equal to one-half of the value at which the repeatable tests were performed. This was done to evaluate polishing phenomena. Polishing is defined as the smoothing of the U-bolt support/piping contact surfaces due to wearing action.

7.0 Data Reduction

The test output included U-bolt strains and maximum slip friction forces. This information allows for a comparison of the maximum slip friction force versus bolt torque and also determination of the coefficient of friction between the pipe and the U-bolt support. The analytical method for determining the coefficient of friction between a U-bolt support and pipe is presented in Appendix I.

8.0 Data Description/Presentation

Test output was reduced and plotted so results could be presented graphically. Maximum slip friction force, that is the force applied to the support at which friction was overcome, was plotted versus bolt torque for each of the four test specimens in Figures 9 through 12. Coefficients of friction were determined and plotted versus bolt torque for each of the four specimens in Figures 13 - 16.

9.0 Results

Results determined for each test specimen are as follows:

- 4 inch stainless steel - Coefficients of friction ranged from .12 to .193. Slip friction forces ranged from 270 pounds at 15 foot pounds of U-bolt torque to 1050 pounds at 35 foot pounds of U-bolt torque. See Figures 9 and 13. Polishing was evaluated by performing a final friction test on a virgin pipe section. Results indicate that this specimen did not experience polishing while subjected to the specified test conditions.

- 10 inch carbon steel - Coefficients of friction ranged from .196 to .279. The trend indicates that the coefficient of friction increases along with bolt torque, and consequently the normal forces between the pipe and support. Slip friction forces ranged from 320 pounds at 10 foot pounds of U-bolt torque to 4725 pounds at 160 foot pounds of U-bolt torque. See Figures 10 and 14. Polishing was evaluated by performing a final friction test on a virgin pipe section. Results indicate that this specimen did not experience polishing while subjected to the specified test conditions.

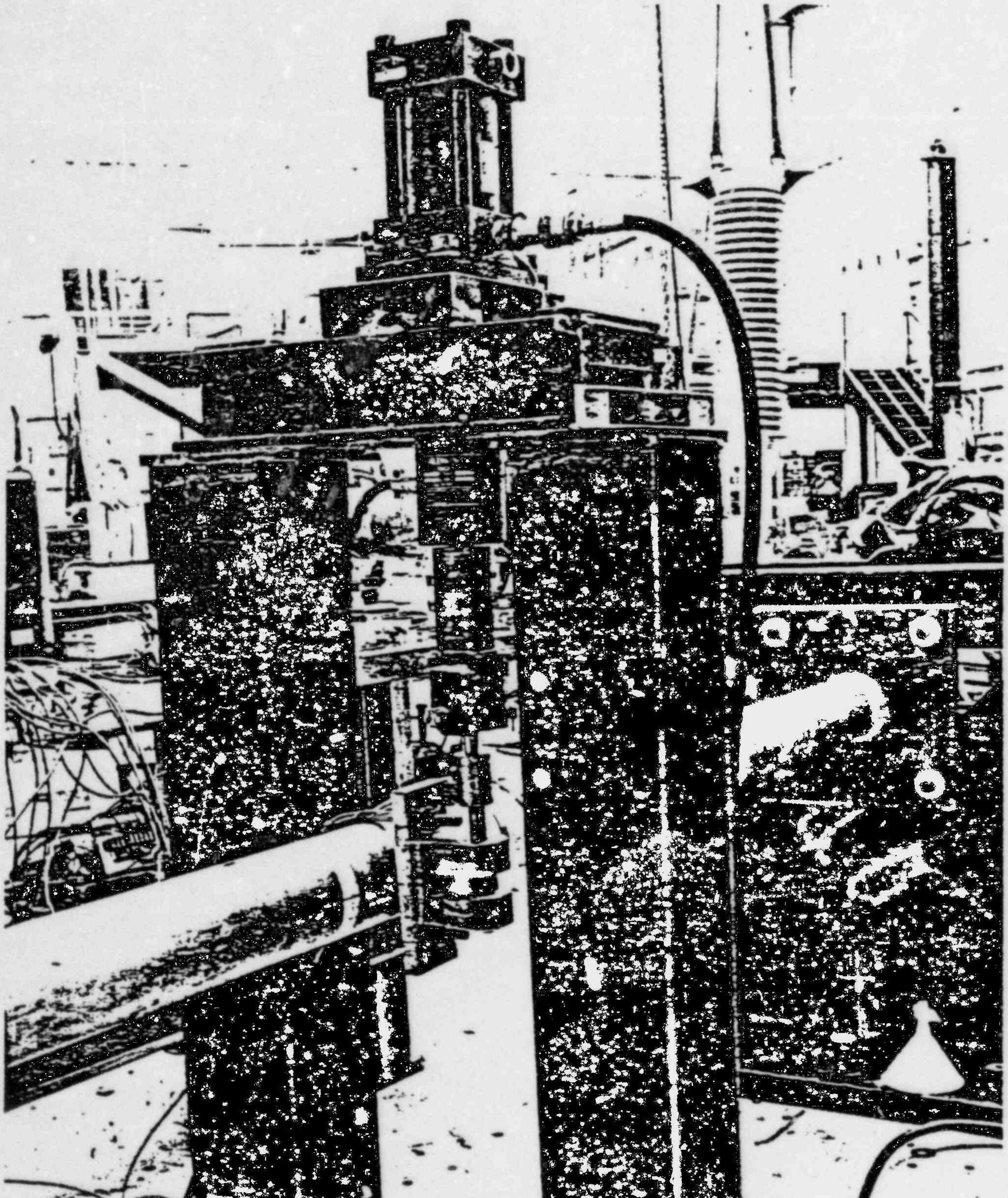
- 10 inch stainless steel - Coefficients of friction ranged from .160 to .225. There is a slight trend for the coefficient of friction to decrease as the bolt torque increases. This is attributed to the polishing phenomena occurring between the pipe and support due to the repeated test steps. Polishing was also verified by performing a final friction test on a virgin pipe section. This test yielded a higher coefficient of friction, .231, than did the

polished section at comparable strain levels. Slip friction forces ranged from 330 pounds at 10 foot pounds of U-bolt torque to 3500 pounds at 130 foot pounds of U-bolt torque. See Figures 11 and 15.

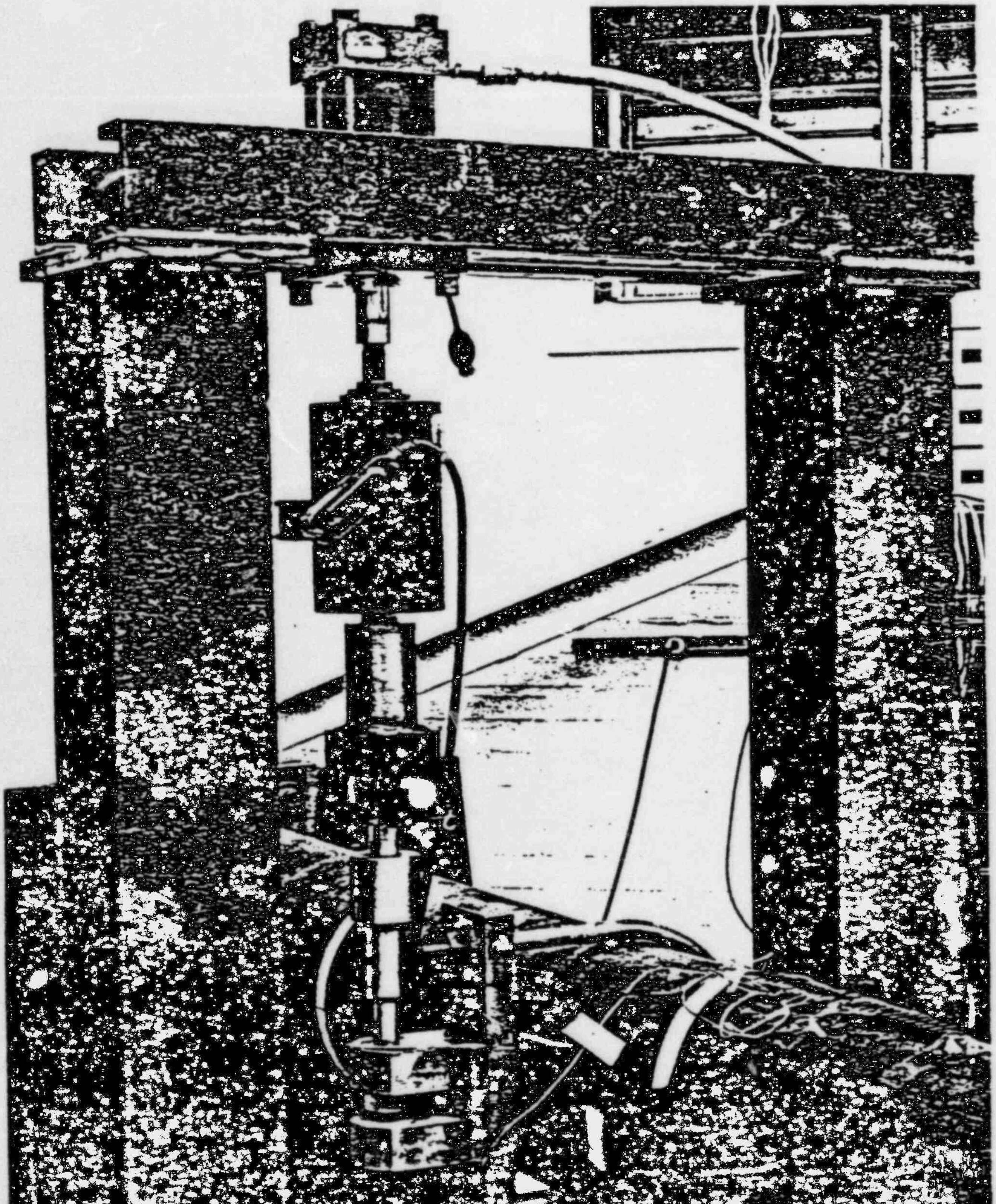
- 32 inch carbon steel - Coefficients of friction ranged from .220 to .516. Slip friction forces ranged from 3200 pounds at 200 foot pounds of U-bolt torque to 11,000 pounds at 1170 foot pounds of U-bolt torque. See Figures 12 and 16. Data obtained at bolt torques of less than 200 foot pounds was not included in the results. At these low torque levels the dead weight of the support caused normal forces between the U-bolt support and piping that were far more significant than the normal forces caused by the bolt torques. As a result, unrealistic coefficients of friction in excess of 1.0 were determined. From the results of the 10 inch carbon steel specimen where friction coefficients ranged from .196 to .279, it was verified that coefficients of friction exceeding 1.0 were unrealistic.

Because of the load capability limitations of the torque wrench, the maximum strain of 964 inches per inch was never achieved.

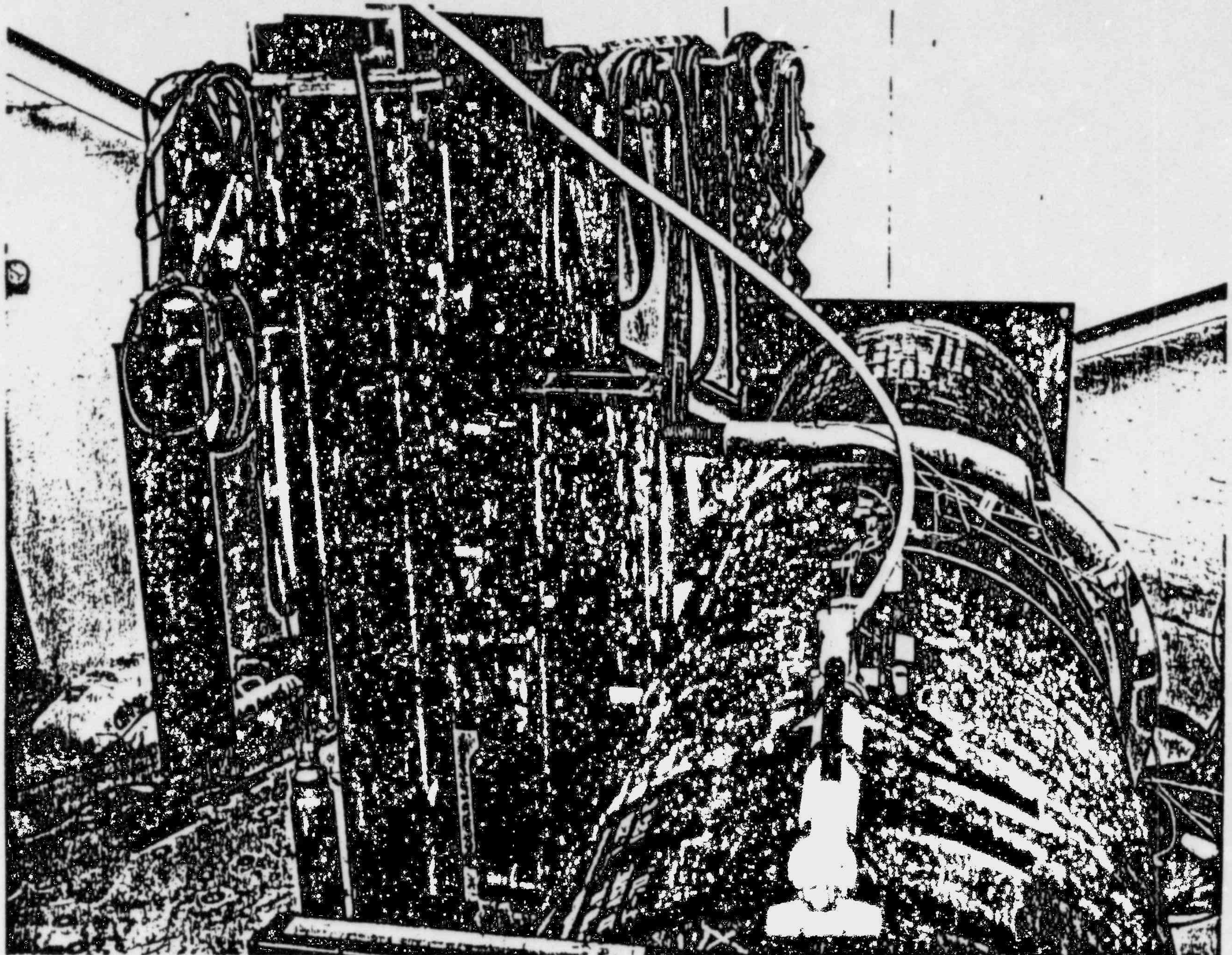
The trend was for coefficients of friction to decrease with increasing bolt torque. Even though a final friction test to evaluate polishing could not be performed on this specimen, the decrease in coefficients of friction was characteristic of the polishing phenomena.



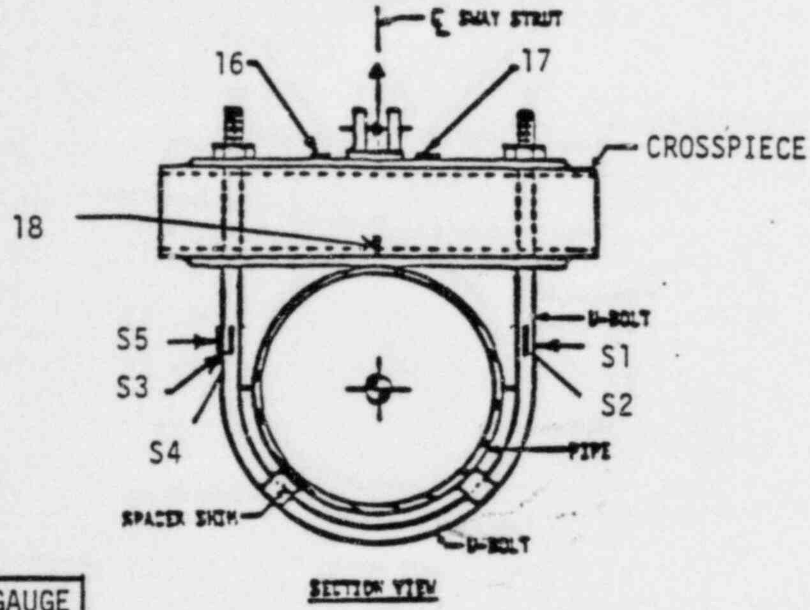
FRICTION TEST
4" STAINLESS STEEL SPECIMEN
PHOTOGRAPH 5



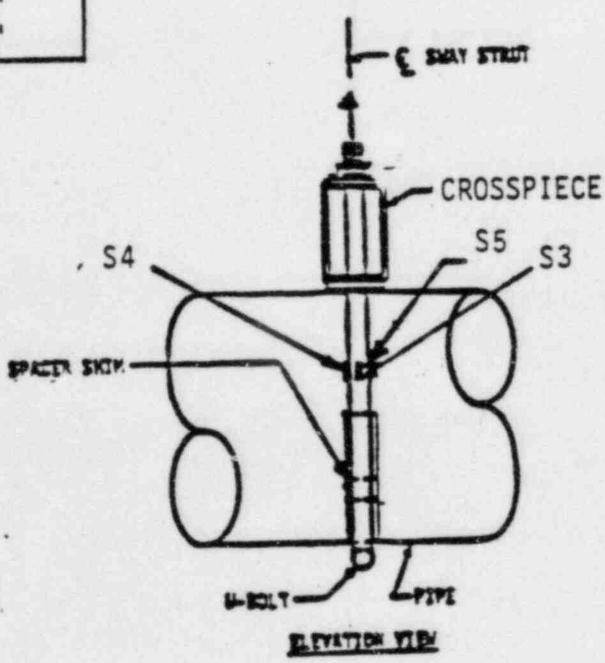
FRICTION TEST
10" CARBON STEEL SPECIMEN
PHOTOGRAPH 6



FRICTION TEST
32" CARBON STEEL SPECIMEN
PHOTOGRAPH 7



STRAIN GAUGE NUMBER	ACTIVE GAUGE LENGTH
S1-S5 S16-S18	0.25" 0.25"



FRICTION TEST

STRAIN GAUGE LOCATIONS

SKETCH 3

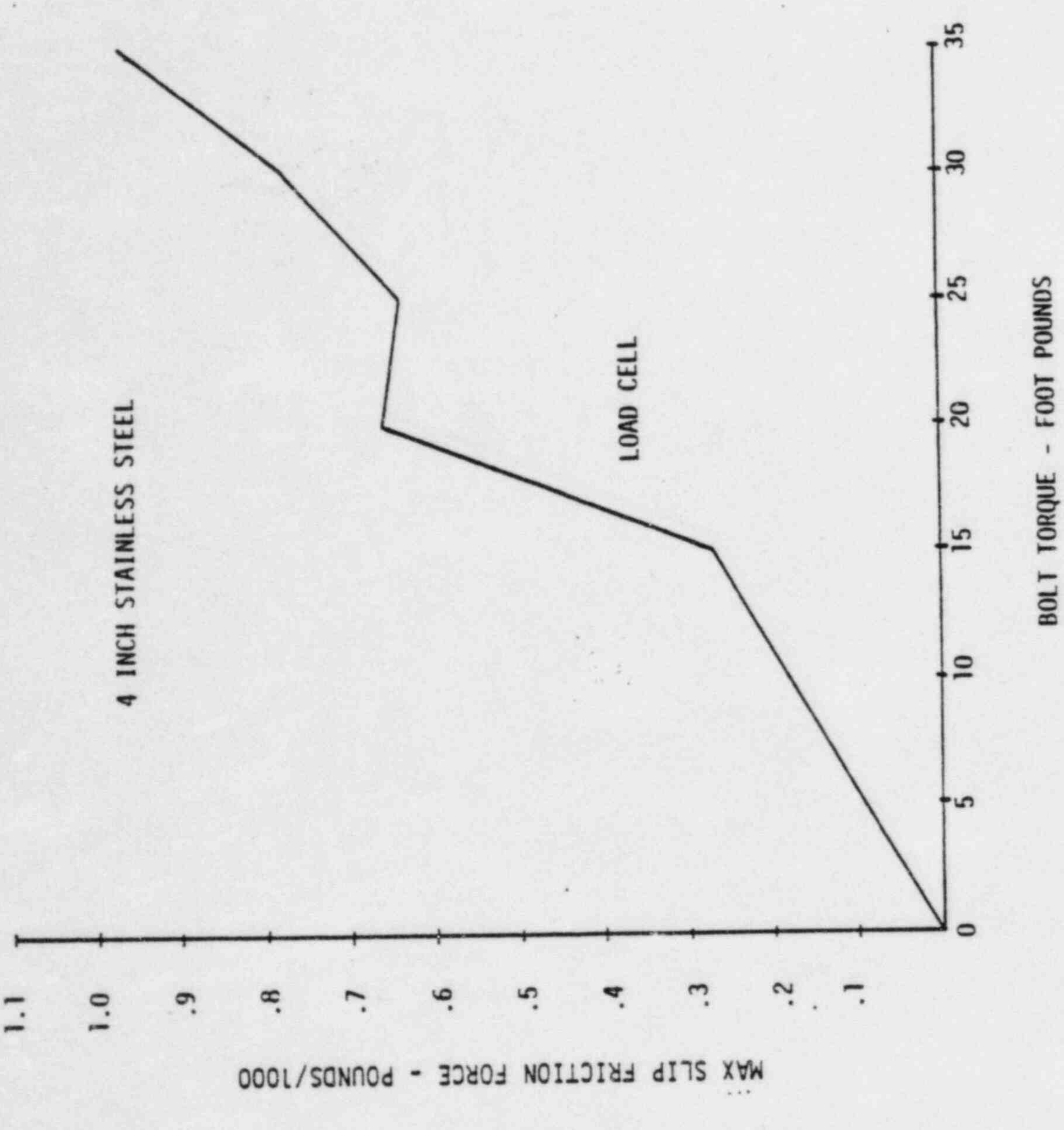
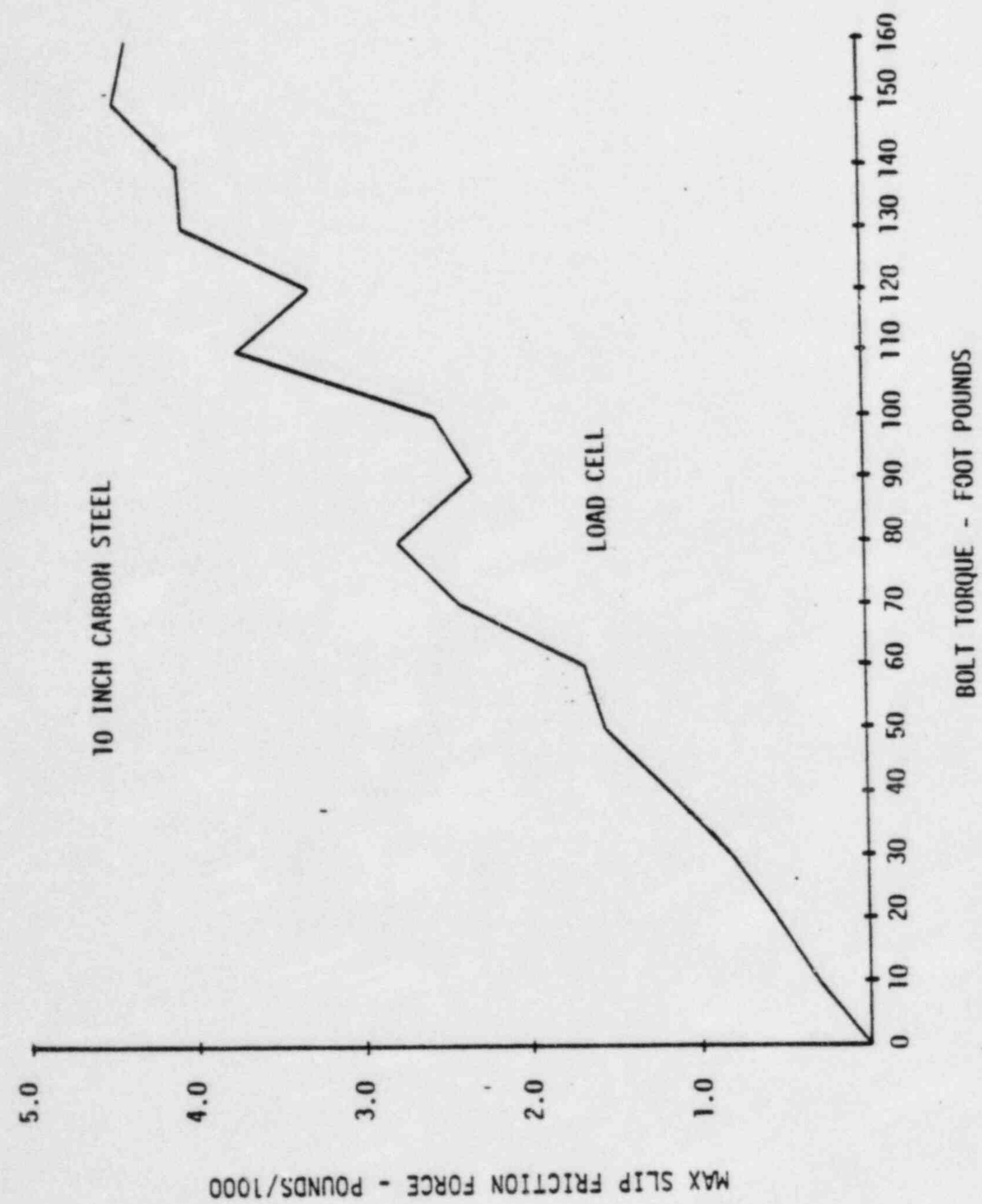

4 INCH STAINLESS STEEL
LOAD CELL
MAX SLIP FRICTION FORCE - POUNDS/1000
1.1
1.0
.9
.8
.7
.6
.5
.4
.3
.2
.1
0
5
10
15
20
25
30
35
BOLT TORQUE - FOOT POUNDS

FIGURE 9



10 INCH CARBON STEEL
LOAD CELL
MAX SLIP FRICTION FORCE - POUNDS/1000
5.0
4.0
3.0
2.0
1.0
0
10
20
30
40
50
60
70
80
90
100
110
120
130
140
150
160
BOLT TORQUE - FOOT POUNDS

FIGURE 10

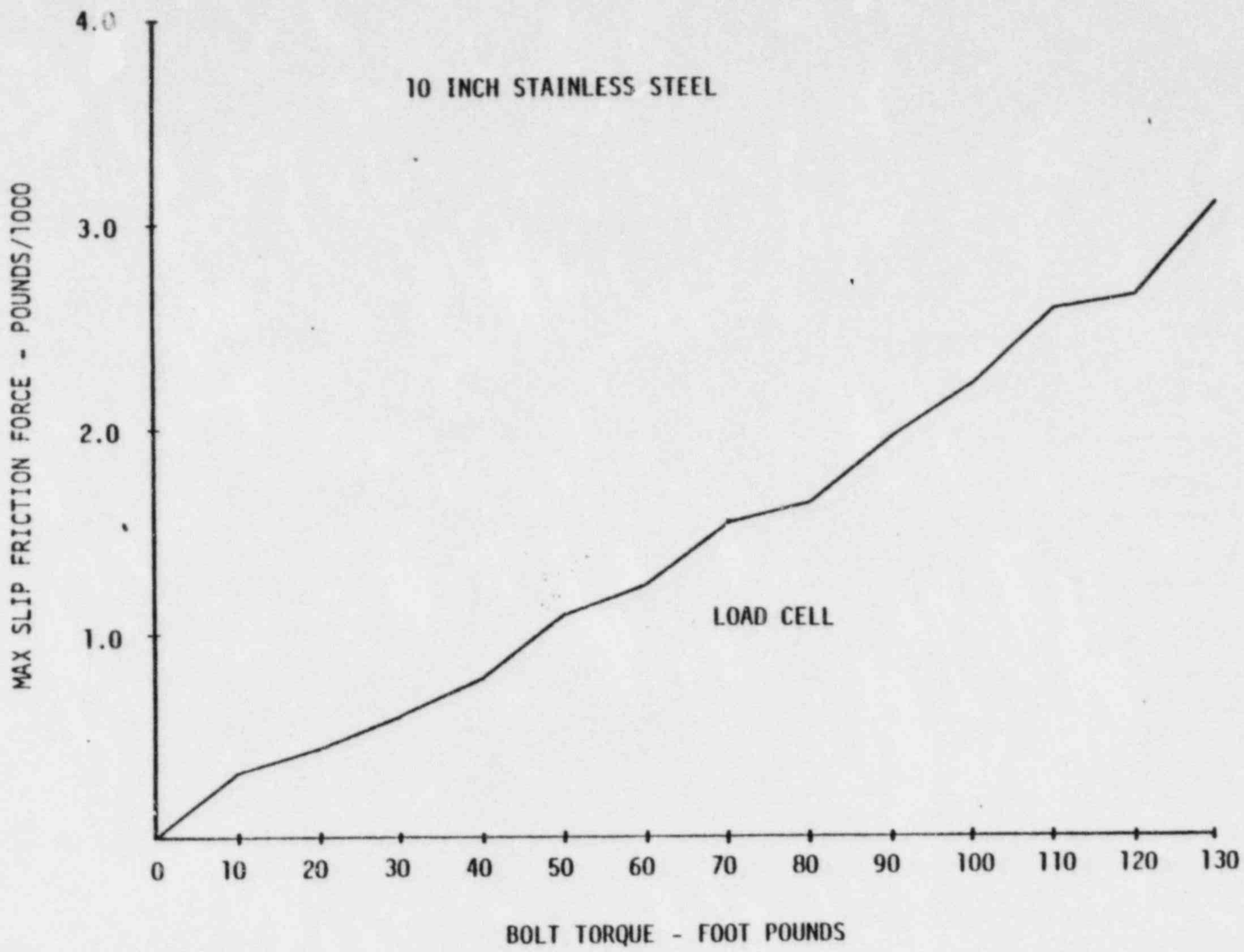

10 INCH STAINLESS STEEL
LOAD CELL
MAX SLIP FRICTION FORCE - POUNDS/1000
4.0
3.0
2.0
1.0
0
10
20
30
40
50
60
70
80
90
100
110
120
130
BOLT TORQUE - FOOT POUNDS

FIGURE 11

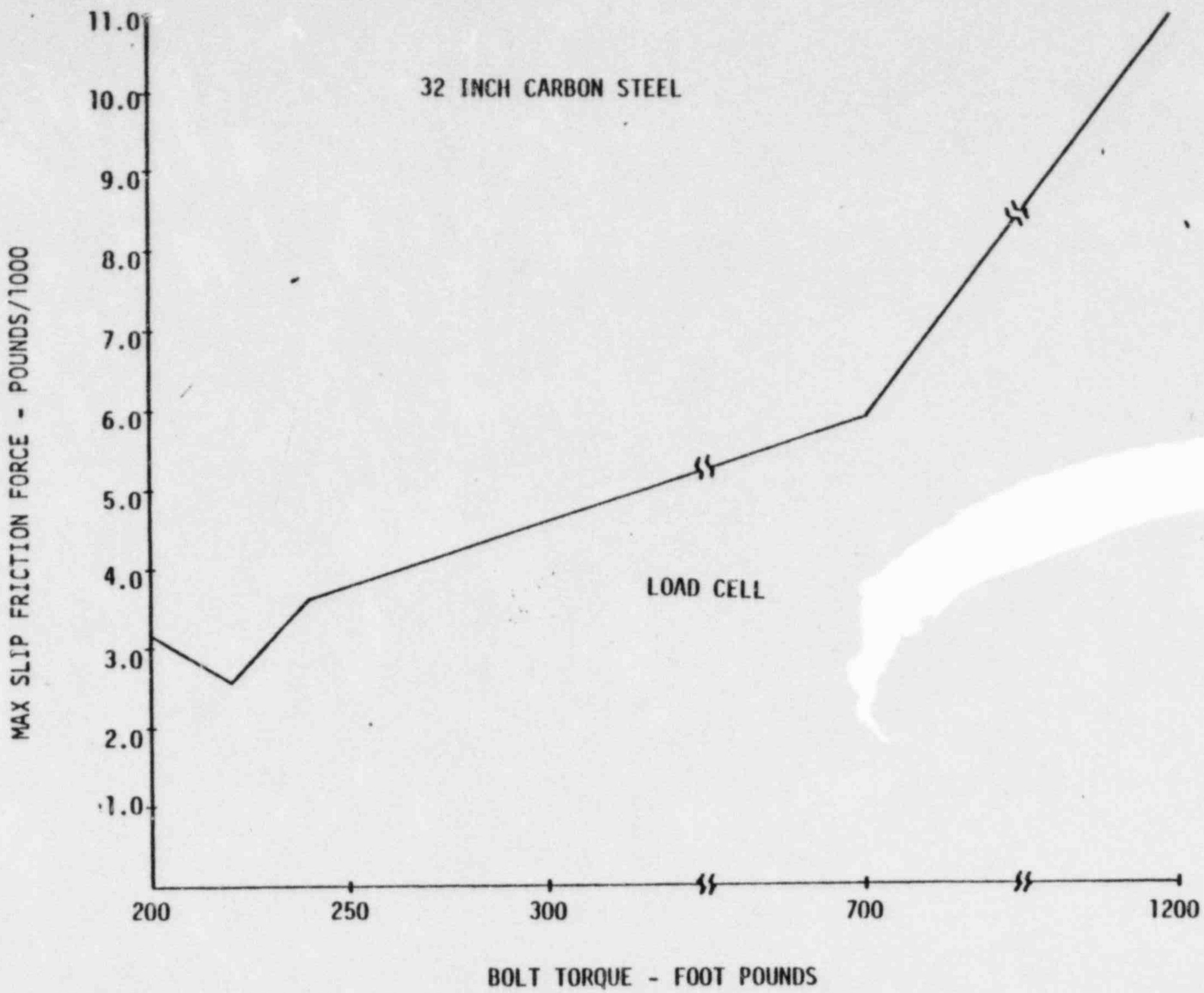

11.0
10.0
9.0
8.0
7.0
6.0
5.0
4.0
3.0
2.0
1.0
32 INCH CARBON STEEL
LOAD CELL
200
250
300
700
1200
MAX SLIP FRICTION FORCE - POUNDS/1000
BOLT TORQUE - FOOT POUNDS

FIGURE 12

4 INCH STAINLESS STEEL

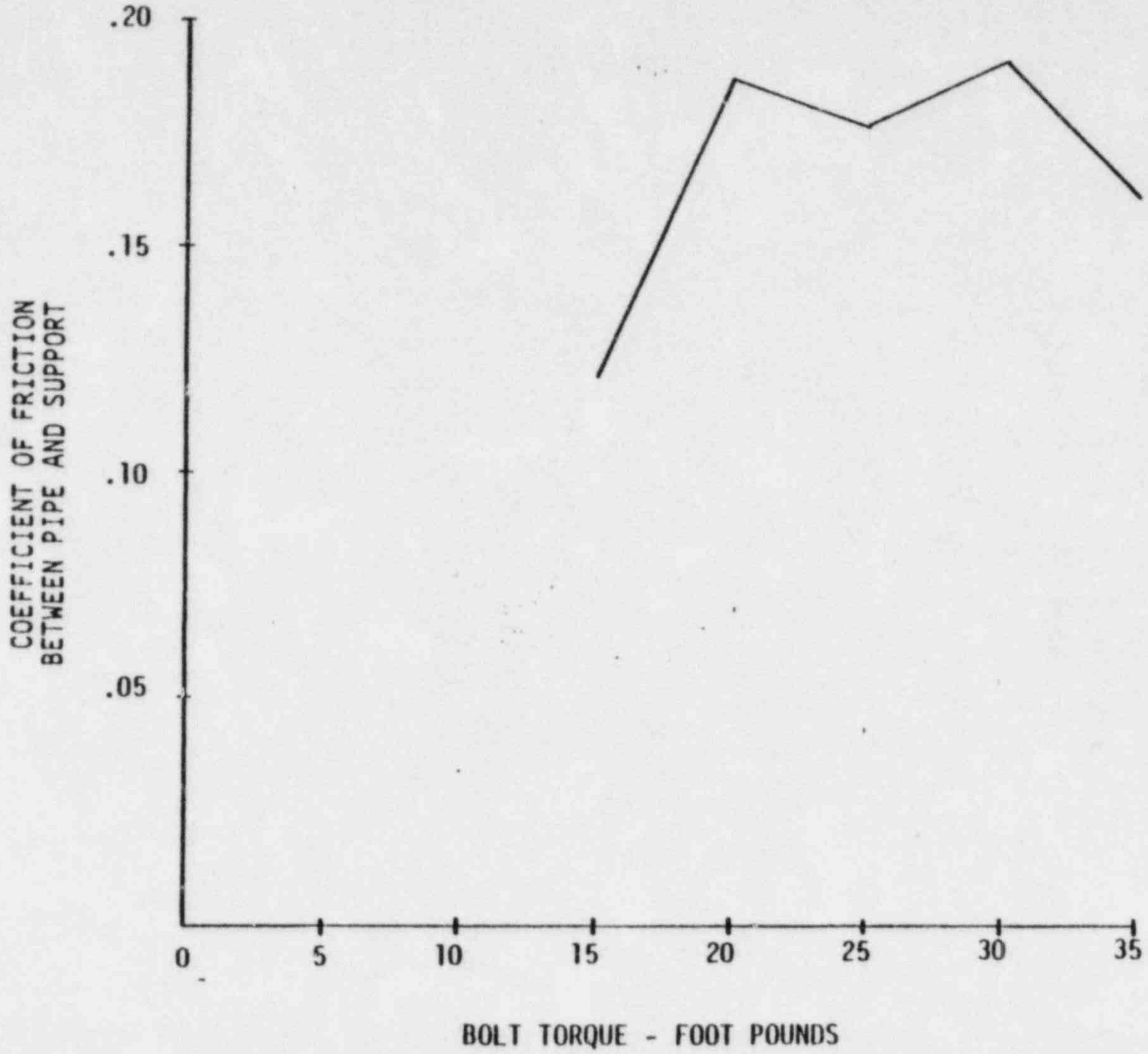


FIGURE 13

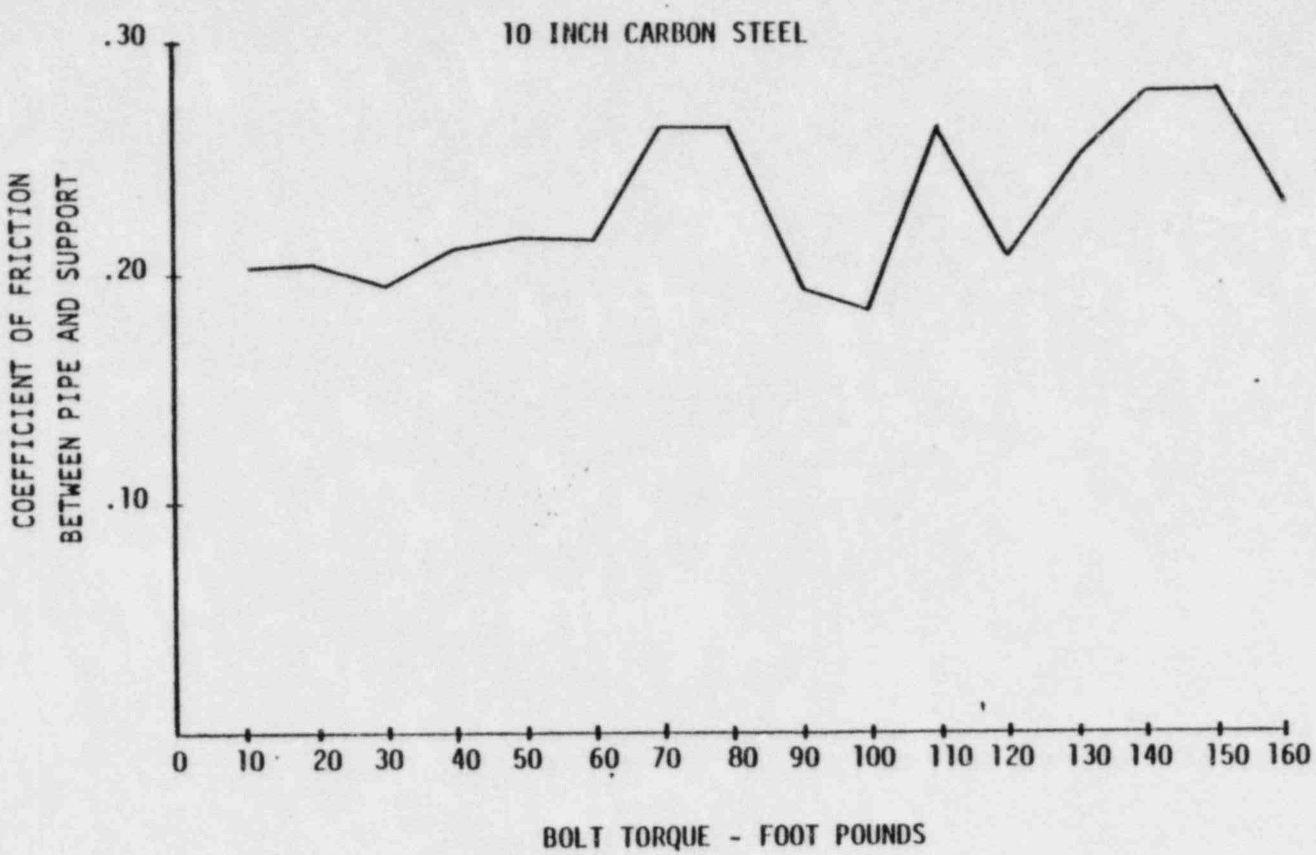

10 INCH CARBON STEEL
.30
.20
.10
COEFFICIENT OF FRICTION
BETWEEN PIPE AND SUPPORT
0
10
20
30
40
50
60
70
80
90
100
110
120
130
140
150
160
BOLT TORQUE - FOOT POUNDS

FIGURE 14

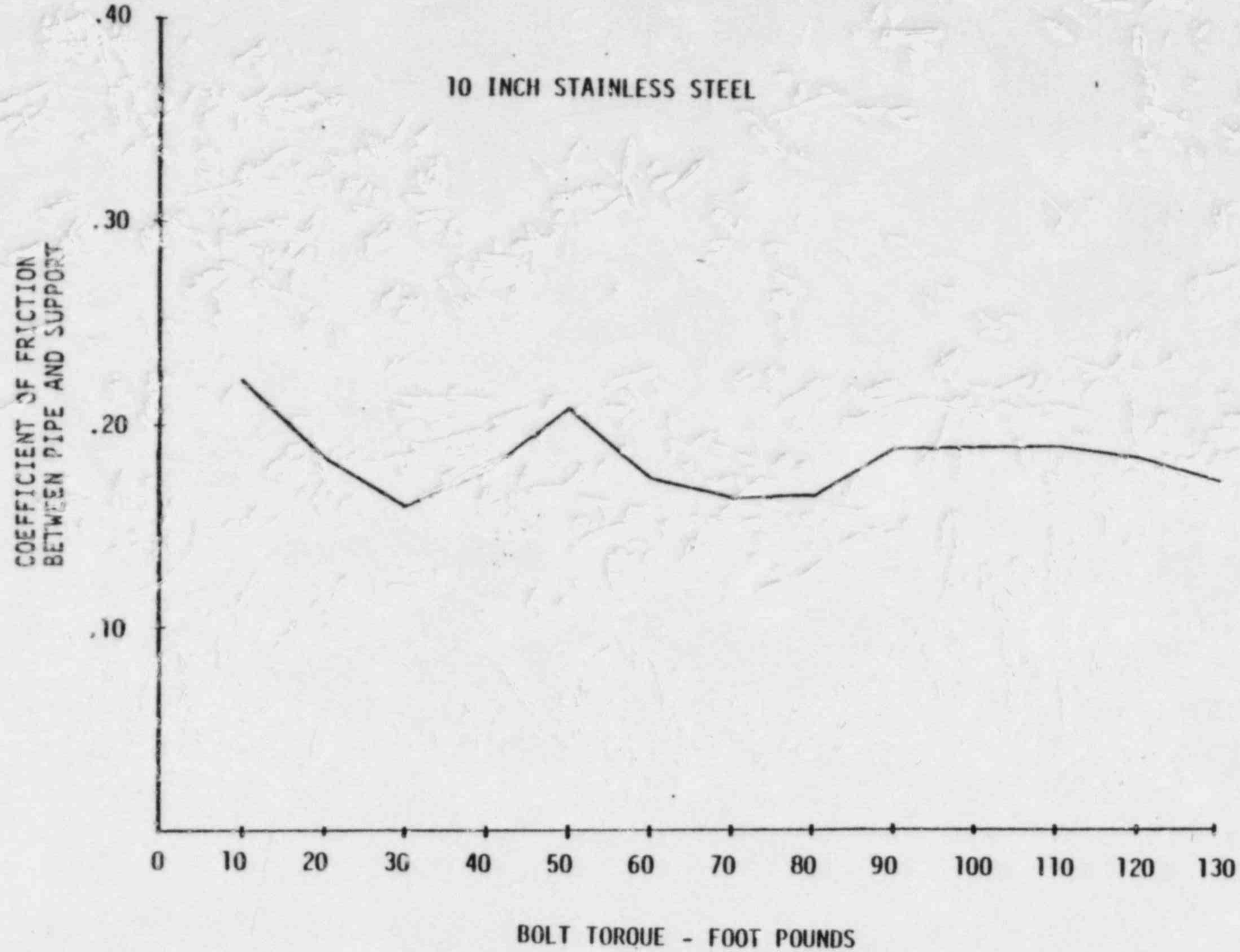

10 INCH STAINLESS STEEL
COEFFICIENT OF FRICTION BETWEEN PIPE AND SUPPORT
.40
.30
.20
.10
0
10
20
30
40
50
60
70
80
90
100
110
120
130
BOLT TORQUE - FOOT POUNDS

FIGURE 15

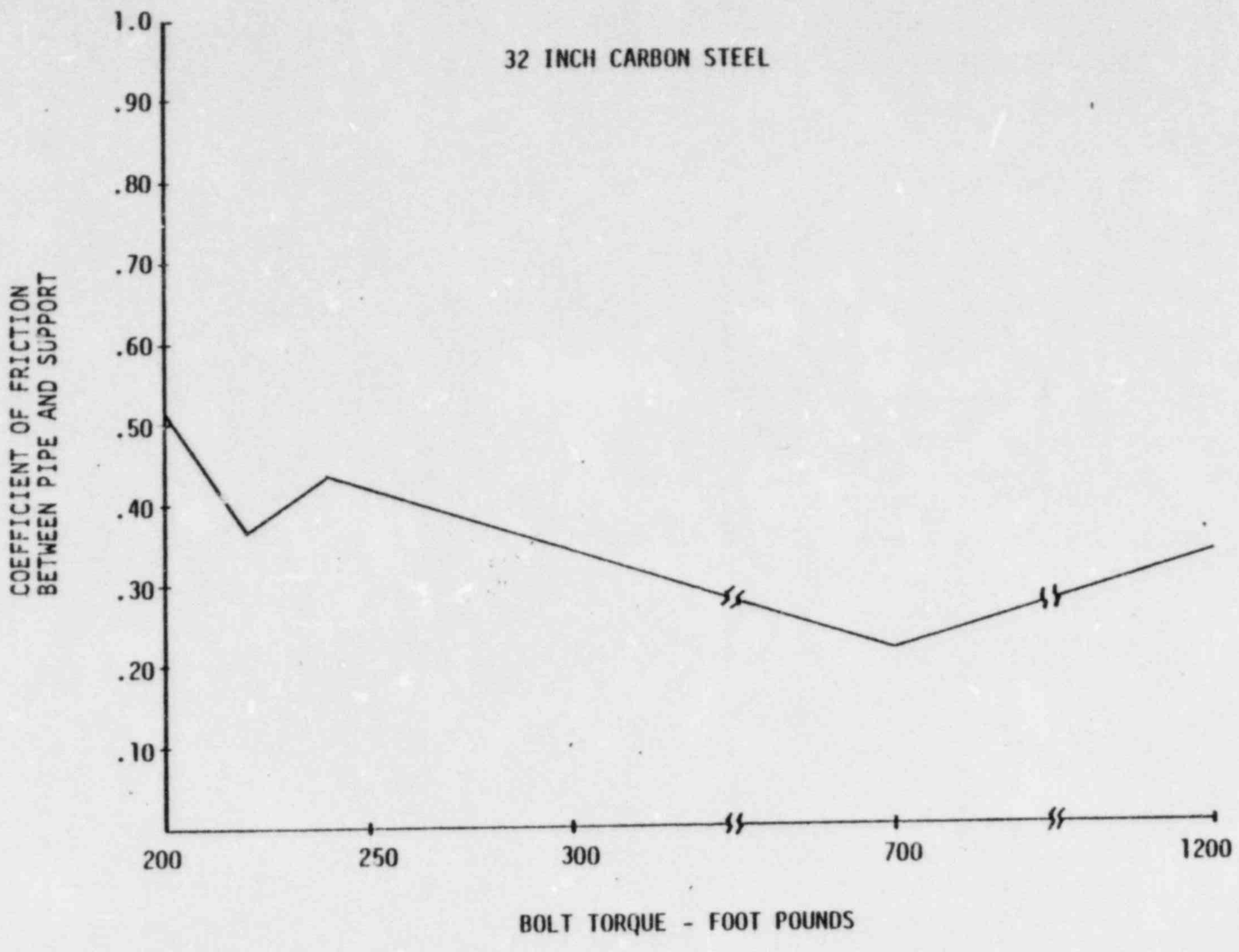

32 INCH CARBON STEEL
COEFFICIENT OF FRICTION
BETWEEN PIPE AND SUPPORT
1.0
.90
.80
.70
.60
.50
.40
.30
.20
.10
200
250
300
700
1200
BOLT TORQUE - FOOT POUNDS

FIGURE 16

V.C. Load Distribution/Strain Measurement Test

1.0 Purpose

This test was designed to determine the load in the U-bolts and crosspiece stresses for given U-bolt nut torque values and support loads. With the U-bolt nuts torqued to three different values and the U-bolt support subjected to four different values of support loads at each value of torque, strains in the crosspiece and U-bolt were measured. This test was performed on the 10 inch stainless steel specimen as specified and provided by TUGCO.

2.0 Test Configuration (See Photograph 8)

The test specimen had mounting plates welded to each end. These plates were then fastened to two floor mounted brackets using high strength fasteners. Strain gauges were installed on the U-bolt and support crosspiece. The U-bolt support was then installed in the center of the test pipe with the crosspiece parallel to the floor. The U-bolt nuts were hand tightened. A loading device consisting of a hydraulic cylinder and load cell, which provided loading in the vertical upward and downward directions, was installed.

3.0 Instrumentation

Instrumentation used for this testing is identified below.

1. Strain gauge signal conditioning and readout devices
2. Load cells
3. Eighteen strain gauges

The strain gauges were installed on the test specimen prior to testing (see Sketch 4). The above instrumentation was calibrated prior to use.

4.0 Loading Devices

Loading devices used for this test are identified below.

1. Torque wrench
2. Hydraulic cylinder

The torque wrench was calibrated prior to use and the hydraulic cylinder load was controlled by the above referenced load cells.

5.0 Load Application

The test specimen was subjected to two types of loading during this test. Torque loads were applied to the U-bolt nuts and the hydraulic cylinder/load cell was used to apply a vertical upward

and downward load at the U-bolt support clevis. Testing was performed at laboratory ambient environmental conditions.

6.0 Test Description

After the specimen had been installed and the instrumentation calibrated, testing commenced. The loading apparatus was used to induce the following loads on the test specimen as specified by TUGCO:

7208 lbs.
5406 lbs.
3604 lbs.
1802 lbs.

These loads were applied both vertically upward and downward at each U-bolt torque value. The U-bolt was torqued to the following values as specified by TUGCO:

100 foot pounds
67 foot pounds
33 foot pounds

These torque values were applied in increments of 10 foot pounds prior to each load test. The U-bolt strain gauge readings and crosspiece strain gauge readings were recorded at each of the three torque levels without the support loads applied and then recorded again with the support loads applied.

7.0 Data Reduction

The test output included U-bolt and crosspiece strains. This information was used to determine the load in the U-bolts and crosspiece stresses for each U-bolt nut torque and support load. The analytical methods for determining U-bolt load and crosspiece stresses are contained in Appendix I.

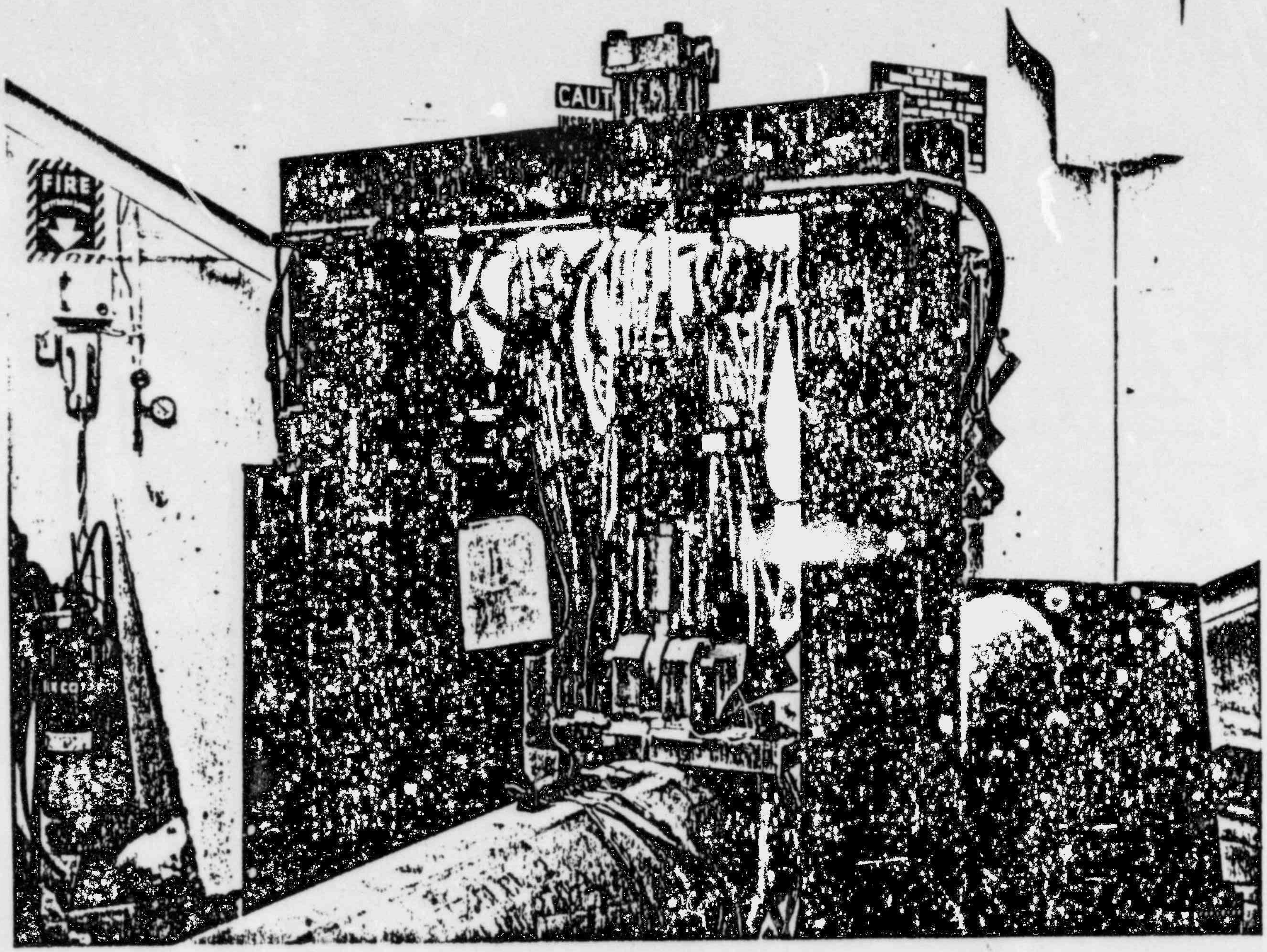
8.0 Data Description/Presentation

Test output was reduced and plotted so results could be presented graphically. U-bolt load was plotted versus support load for bolt torques of 33, 67 and 100 foot pounds. Results due to the support load applied vertically upward are shown in Figure 17. Figure 18 contains results for the support load applied vertically downward. Maximum crosspiece stresses were plotted versus support load for bolt torques of 33, 67 and 100 foot pounds. These results are included in Figures 19 and 20 for the support loads applied vertically upward and downward, respectively.

9.0 Results

With the vertical load upward the U-bolt load increased as the support load increased, thus adding to the U-bolt preload. Conversely, the U-bolt preload decreased as the support load was applied in the vertically downward direction. The maximum U-bolt load occured with a bolt torque of 100 foot pounds and an upward support load of 7208 pounds. The resultant U-bolt load was 8327 pounds. See Figure 17. This was an increase of 1858 pounds over the unloaded condition at 100 foot pounds of torque. The decrease in U-bolt load due to a 7208 vertical downward support load with the bolts torqued at 100 foot pounds was 2306 pounds. The resultant U-bolt load was 4163 pounds versus the 6469 pounds for the unloaded condition. See Figure 18.

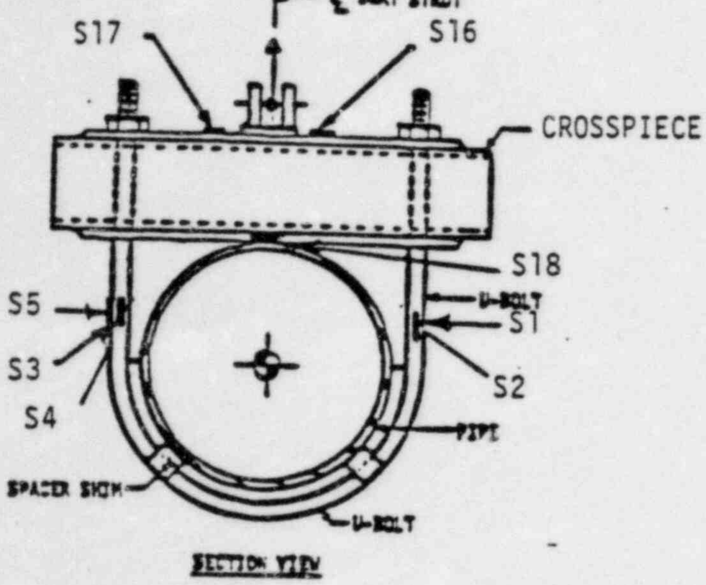
The maximum crosspiece stress occurred with a bolt torque of 100 foot pounds and an upward support load of 7208 pounds. The resultant crosspiece stress was 22,487 psi. See Figure 19. This was an increase of 5,384 psi over the unloaded condition at 100 foot pounds of torque.

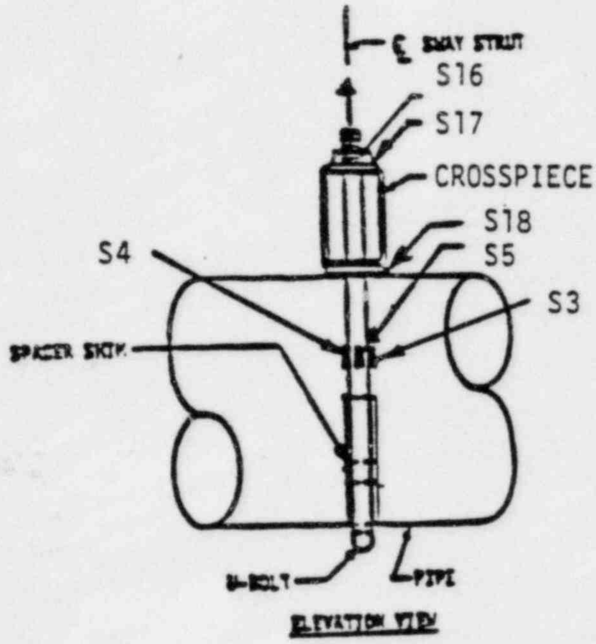


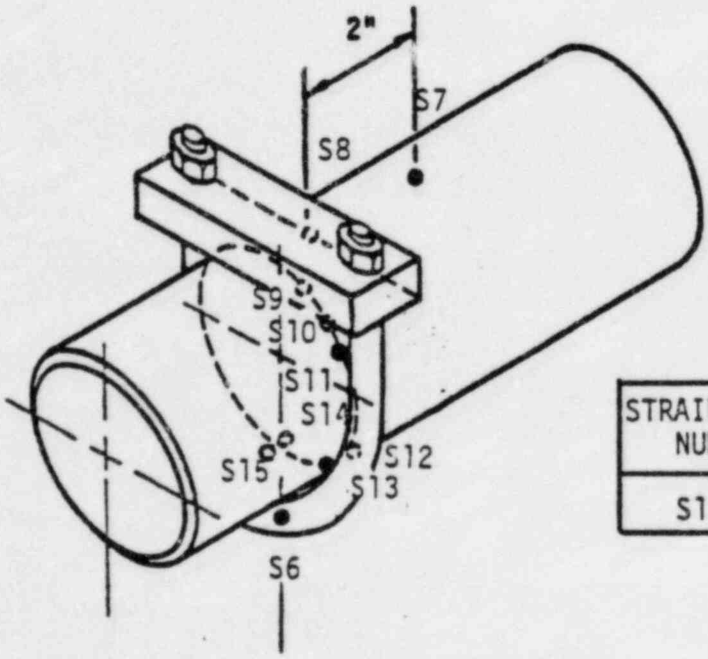
LOAD DISTRIBUTION/STRAIN MEASUREMENT

10" STAINLESS STEEL SPECIMEN

PHOTOGRAPH 8







STRAIN GAUGE NUMBER	ACTIVE GAUGE LENGTH
S1-S18	0.25"

LOAD DISTRIBUTION/STRAIN MEASUREMENT TEST
STRAIN GAUGE LOCATIONS

SKETCH 4

10 INCH STAINLESS STEEL

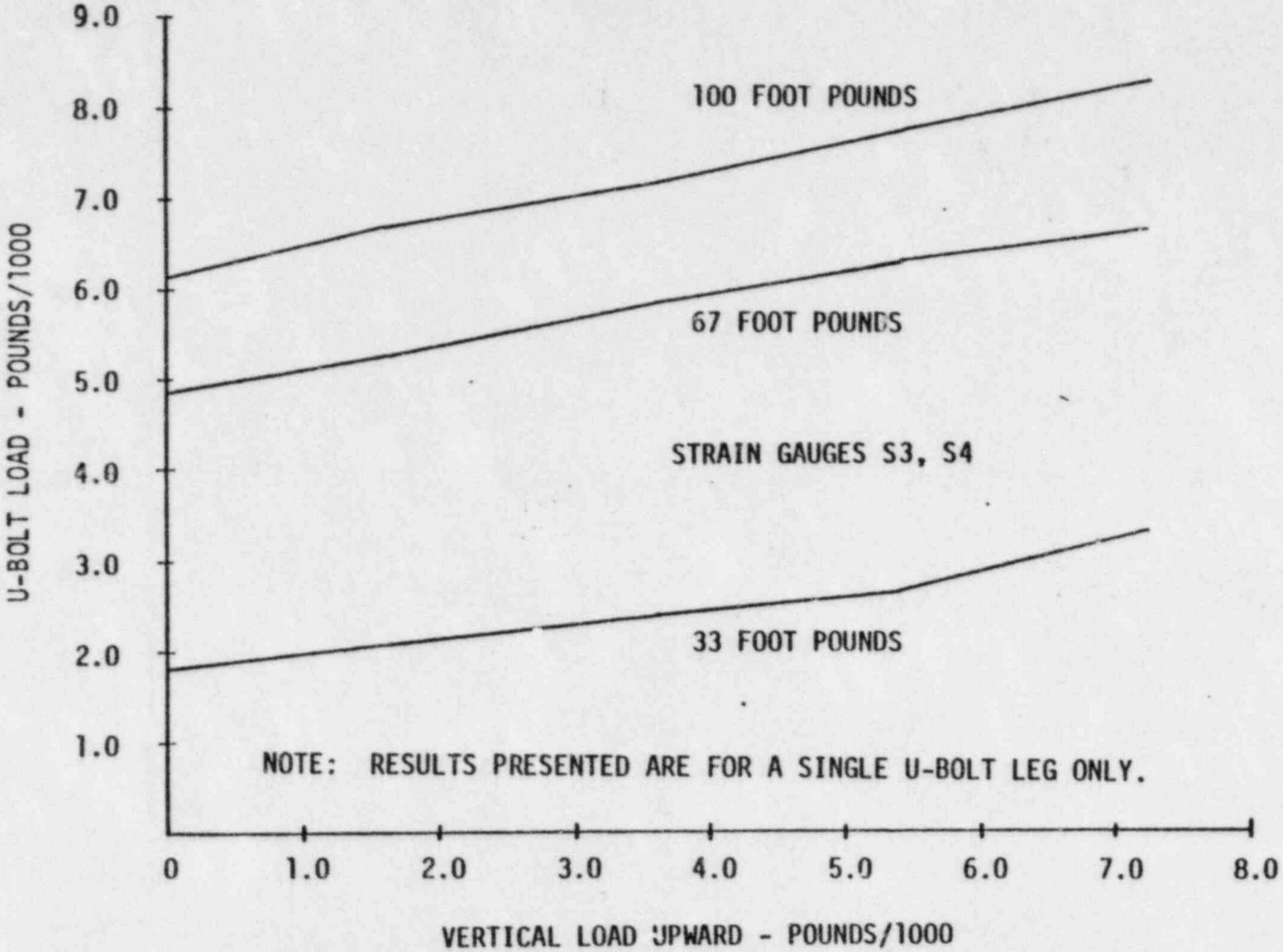


FIGURE 17

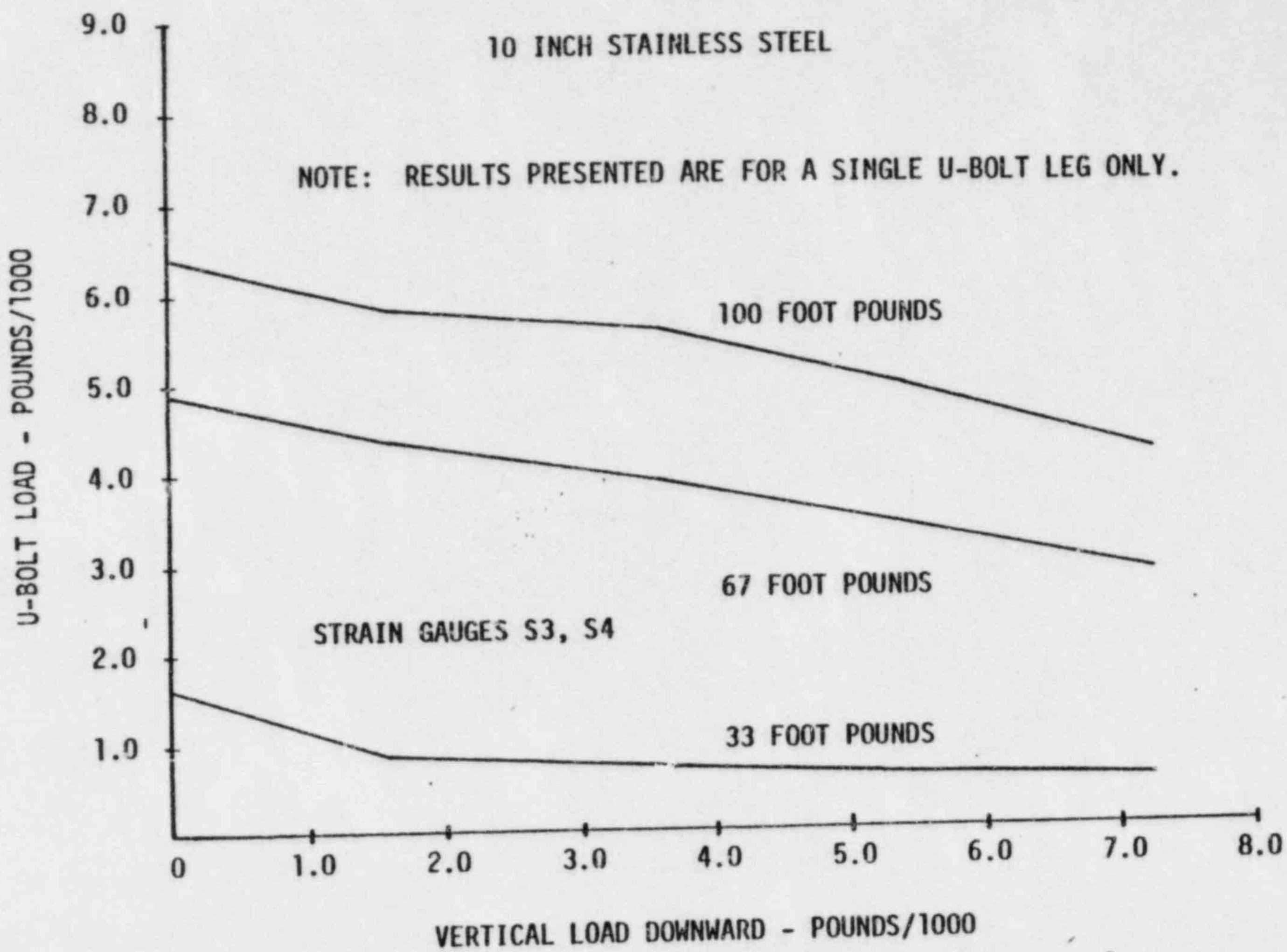
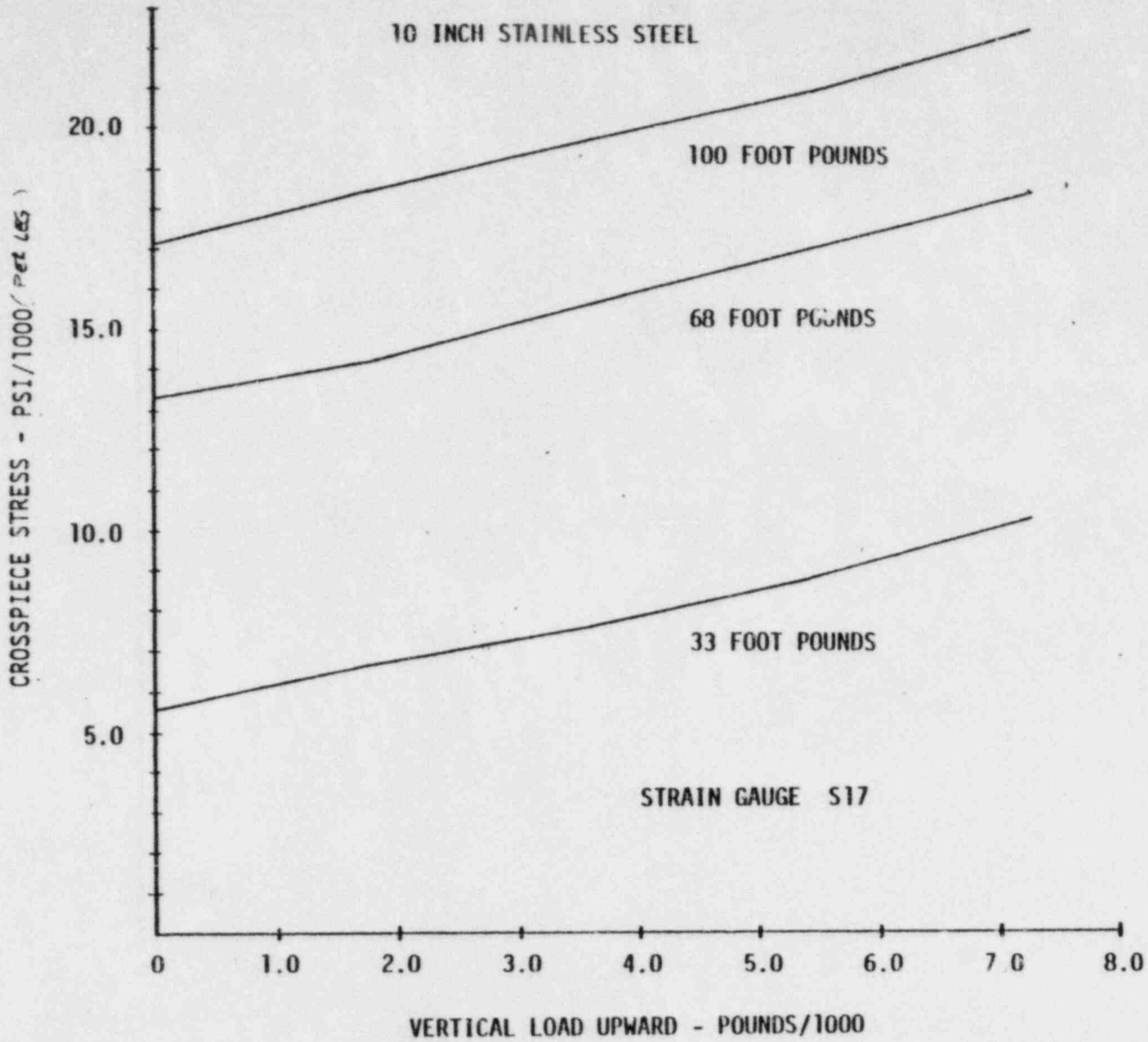
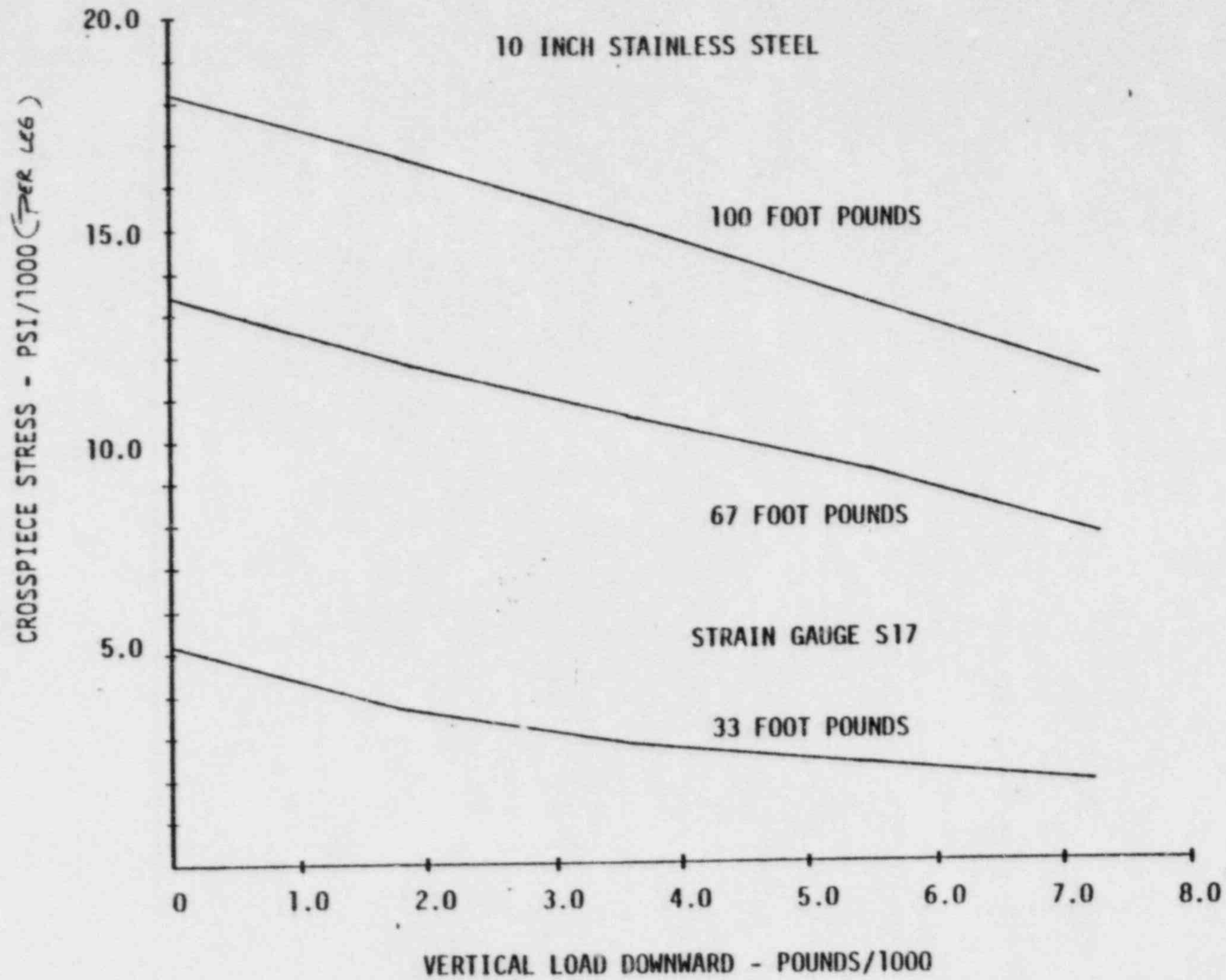

10 INCH STAINLESS STEEL
NOTE: RESULTS PRESENTED ARE FOR A SINGLE U-BOLT LEG ONLY.
100 FOOT POUNDS
67 FOOT POUNDS
STRAIN GAUGES S3, S4
33 FOOT POUNDS
U-BOLT LOAD - POUNDS/1000
9.0
8.0
7.0
6.0
5.0
4.0
3.0
2.0
1.0
0
1.0
2.0
3.0
4.0
5.0
6.0
7.0
8.0
VERTICAL LOAD DOWNWARD - POUNDS/1000

FIGURE 18



10 INCH STAINLESS STEEL
100 FOOT POUNDS
68 FOOT POUNDS
33 FOOT POUNDS
STRAIN GAUGE S17
CROSSPIECE STRESS - PSI/1000(Per LBS)
20.0
15.0
10.0
5.0
0
1.0
2.0
3.0
4.0
5.0
6.0
7.0
8.0
VERTICAL LOAD UPWARD - POUNDS/1000

FIGURE 19



10 INCH STAINLESS STEEL
100 FOOT POUNDS
67 FOOT POUNDS
STRAIN GAUGE S17
33 FOOT POUNDS
20.0
15.0
10.0
5.0
CROSSPIECE STRESS - PSI/1000 (PER LEG)
0
1.0
2.0
3.0
4.0
5.0
6.0
7.0
8.0
VERTICAL LOAD DOWNWARD - POUNDS/1000

FIGURE 20

V.D. Thermal Cycling/Thermal Gradient Test

1.0 Purpose

The purpose of this test was to investigate the effect of thermal cycling at elevated temperatures on U-bolt preload. This test was performed on three test specimens (pipes) using three designs of U-bolt type supports as specified and provided by TUGCO.

Pipe Test Specimens

32 inch carbon steel pipe, insulated
10 inch stainless steel pipe, uninsulated
4 inch stainless steel pipe, insulated

U-bolt Designs

32 inch U-bolt type pipe support
10 inch U-bolt type pipe support
4 inch U-bolt type pipe support

2.0 Test Configuration (See Photographs 9 through 11)

The three pipe test specimens described in Section 1.0 had a mounting plate welded to each end. These plates were then fastened to two floor mounted brackets using high strength fasteners. The carbon steel test specimen had a Steel Structures Painting Council No. 2 finish applied on a four foot section in the center of the pipe. Strain gauges were installed on the U-bolt. Thermocouples were installed on the U-bolt, crosspiece and pipe. The U-bolts were then installed with the nuts hand tight. The 4 inch and 32 inch test specimens were insulated. Strip heaters were installed inside the total length of the pipe to provide the internal heat source.

3.0 Instrumentation

Instrumentation used for these tests is identified below.

1. Strain gauge signal conditioning and readout devices
2. Thermocouple signal conditioning devices
3. Five strain gauges
4. Eleven thermocouples
5. Temperature data recorder

See Sketches 5 and 6 for locations of strain gauges and thermocouples. The above instrumentation was calibrated prior to use.

4.0 Loading Devices

Loading devices used for these tests are identified below.

1. Torque wrench
2. Strip heaters

The torque wrench was calibrated prior to use.

5.0 Load Application

The test specimens were subjected to two types of loading during this test. Torque loads were applied to the U-bolt nuts and the strip heaters were used to provide the thermal cyclic loads.

6.0 Test Description

After each specimen had been installed and the instrumentation calibrated, testing commenced. The testing method was the same for the three specimens. The only variables were the U-bolt nut torques and maximum test temperatures. These values were specified by TUGCO.

Nut Torques:

4 inch test specimen: 60 foot pounds; in increments of 5 foot pounds.

10 inch test specimen: 100 foot pounds; in increments of 10 foot pounds.

32 inch test specimen: 600 foot pounds; in increments of 20 foot pounds.

Maximum Test Temperatures:

4 inch test specimen: 560°F
10 inch test specimen: 250°F
32 inch test specimen: 560°F

With the test specimen installed, the U-bolts were tightened to the above torque values. With the test specimen at ambient temperature, the recording equipment was activated. The strip heaters were then used to bring the test specimen to the test temperature. After maintaining the peak temperature for one hour, the strip heaters were deactivated to allow the test specimen to cool to the laboratory ambient temperature. This thermal cycling was repeated a total of ten times for each specimen as specified by TUGCO.

7.0 Data Reduction

The test data recorded included U-bolt strains and temperatures of the pipe, U-bolt and crosspiece. Using the strain data, U-bolt preloads were determined. The analytical method for

determining U-bolt preload is contained in Appendix I. This information was used to evaluate the effect of thermal cycling on the U-bolt preload.

8.0 Data Description/Presentation

Test output was reduced and plotted so results could be presented graphically. U-bolt preloads were plotted versus thermal cycle for each of the specimens while at maximum pipe temperature. This information is presented in Figures 21 through 24.

9.0 Results

- 4 inch stainless steel

  The U-bolt preloads at ambient temperature (105°F) with a torque of 60 foot pounds prior to cycle 1 were:

  Leg 1 (Gauges S2, S5) = 5903 pounds.
  Leg 2 (Gauges S3, S10) = 6292 pounds.

  The U-bolt preloads at ambient temperature (107.5°F) with a torque of 60 foot pounds prior to cycle 10 were:

  Leg 1 (Gauges S2, S5) = 3271 pounds.
  Leg 2 (Gauges S3, S10) = 4529 pounds.

  The U-bolt preloads on the 4 inch stainless steel specimen, while at a pipe temperature of 560°F and a torque value of 60 foot pounds, dropped during the first few cycles but stabilized at approximately 7000 pounds for Leg 1 and 6400 pounds for Leg 2. See Figure 21.

  Since strain gauges are sensitive to temperature, direct strain readings may include temperature induced or "apparent" strains. The portion of the strain readings that are "apparent" were determined by calibrating the gauges over a wide temperature range. "Apparent" strains were determined for the strain gauges used on the 4 inch stainless steel specimen by calibration testing. The "apparent" strains were combined with the direct strain readings to assure that accurate strains were determined over the entire temperature range.

  The 4 inch stainless steel specimen required 2 hours to reach the maximum pipe temperature (560°F) during each cycle.

- 10 inch stainless steel

  The U-bolt preloads at ambient temperature (78.2°F) with a torque of 100 foot pounds prior to cycle 1 were:

Leg 1 (Gauges S4, S11) = 4515 pounds.
Leg 2 (Gauges S8, S1) = 4503 pounds.

The U-bolt preloads at ambient temperature (76°F) with a torque of 100 foot pounds prior to cycle 10 were:

Leg 1 (Gauges S4, S11) = 3209 pounds.
Leg 2 (Gauges S8, S1) = 3516 pounds.

The U-bolt preloads on the 10 inch stainless steel specimen, while at a pipe temperature of 250°F and a torque value of 100 foot pounds, dropped slightly during the first few cycles but stabilized at approximately 3500 pounds and 3900 pounds for Legs 1 and 2, respectively. See Figure 22.

Apparent strains were determined for the strain gauges used on the 10 inch stainless steel specimen by calibration testing. The apparent strains were combined with the direct strain readings to assure that accurate strains were determined over the entire temperature range.

The 10 inch stainless steel specimen required 20 minutes to reach the maximum pipe temperature (250°F) during each cycle.

o 32 inch carbon steel

The U-bolt preloads at ambient temperature (75.7°F) with a torque of 600 foot pounds prior to cycle 1 were:

Leg 1 (Gauges 13, 18) = 15,158 pounds.
Leg 2 (Gauges 15, 17) = 15,675 pounds.

Because of the large mass of the 32" pipe test specimen there was a temperature lag between the pipe and the U-bolt support during cooldown between cycles. As a result, the U-bolt support never reached ambient temperature coincident with the pipe. Therefore, while the pipe had reached ambient temperature between cycles as specified by the test procedure, U-bolt preloads at ambient temperature prior to cycle 10 could not be determined.

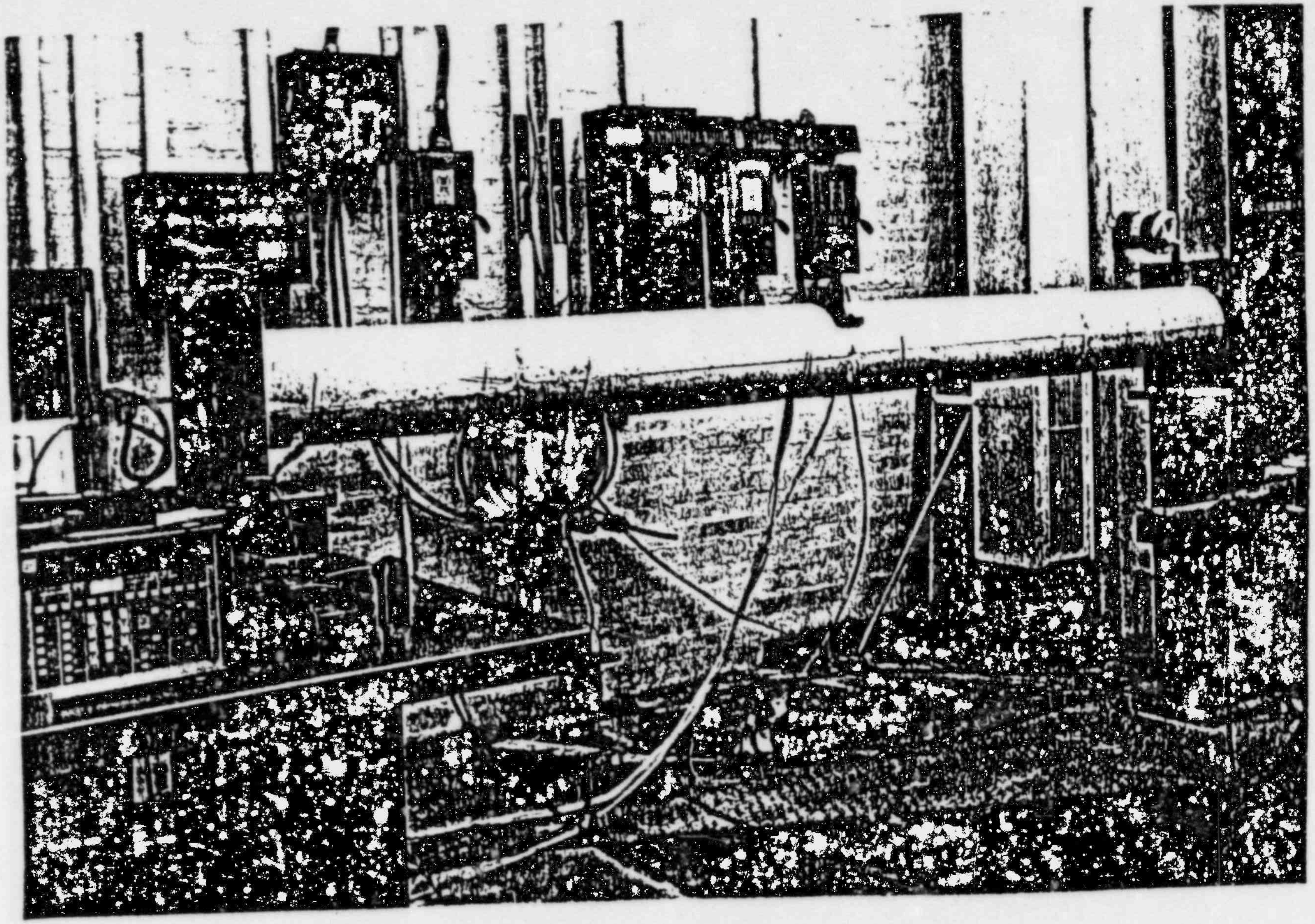
Apparent strain data was not directly available for the gauges used on the 32 inch specimen. Since the same type of gauges were used on the 10 inch specimen, a statistical evaluation was performed and a confidence interval was determined for apparent strain at a 95 percent confidence level. This confidence interval was applied to the apparent strain curves obtained for the 10 inch specimen so that apparent strain ranges were established for use with the 32 inch specimen.

Apparent strain is a function of gauge type and material to which the gauge is attached. Both the 10 and 32 inch U-bolts are of the same material. As noted previously, the same type of gauges were used on both. U-bolt size has no effect on apparent strain.

The U-bolt preload on Leg 1 (Gauges 13, 18) of the 32 inch carbon steel specimen, while at a pipe temperature of 560°F and 600 foot pounds of torque, went from 30,144 pounds during cycle 1 to 29,799 pounds during cycle 10. The difference between the highest and lowest recorded values over the course of 10 cycles was 2756 pounds. See Figure 23.

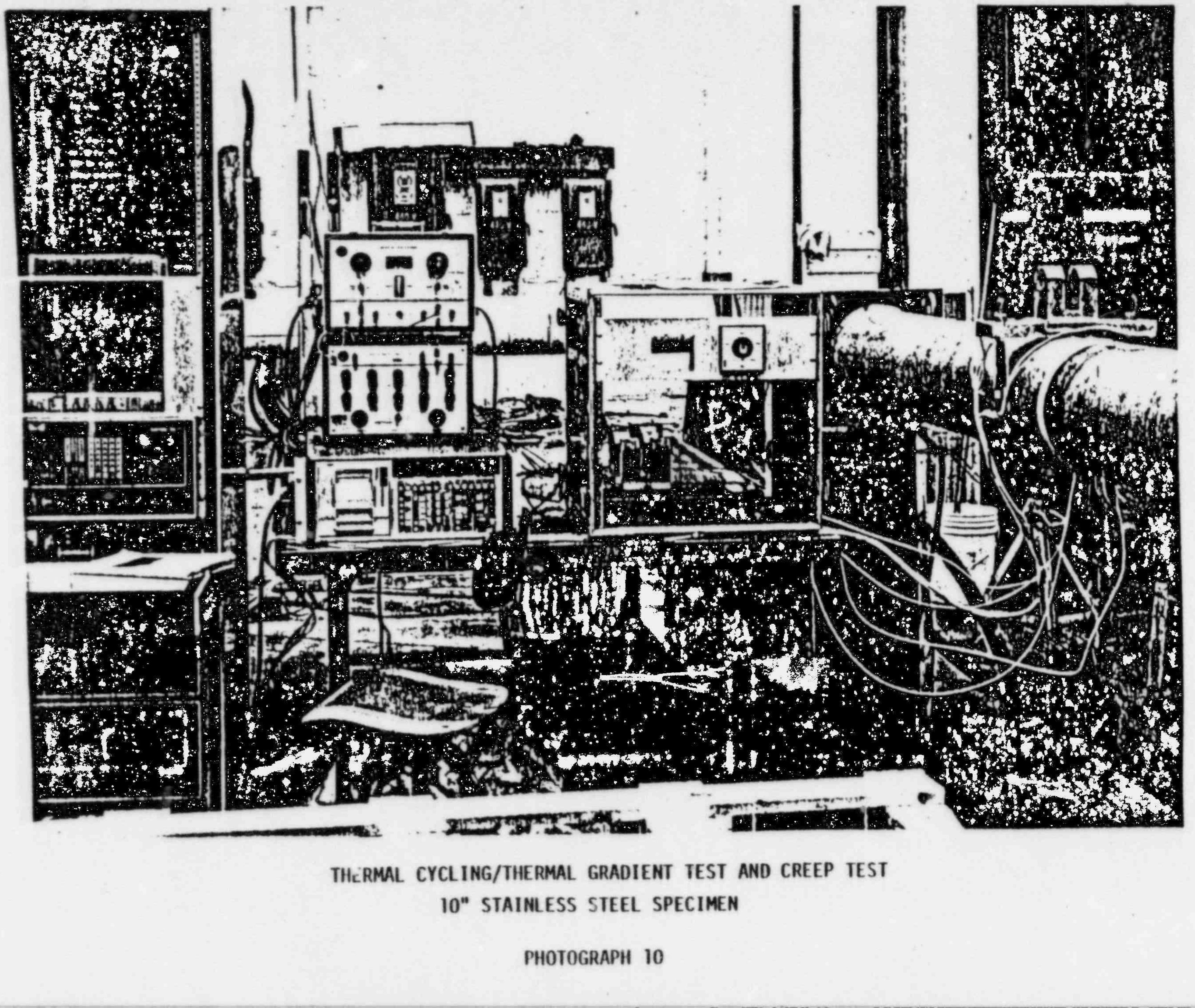
The U-bolt preload on Leg 2 (Gauges 15, 17) of the 32 inch carbon steel specimen, while at a pipe temperature of 560°F and 600 foot pounds of torque, went from 25,579 pounds during cycle 1 to 24,373 pounds during cycle 10. The difference between the highest and lowest recorded values over the course of 10 cycles was 3359 pounds. See Figure 24.

The 32 inch carbon steel specimen required 4 1/2 hours to reach the maximum pipe temperature (560°F) during each cycle.



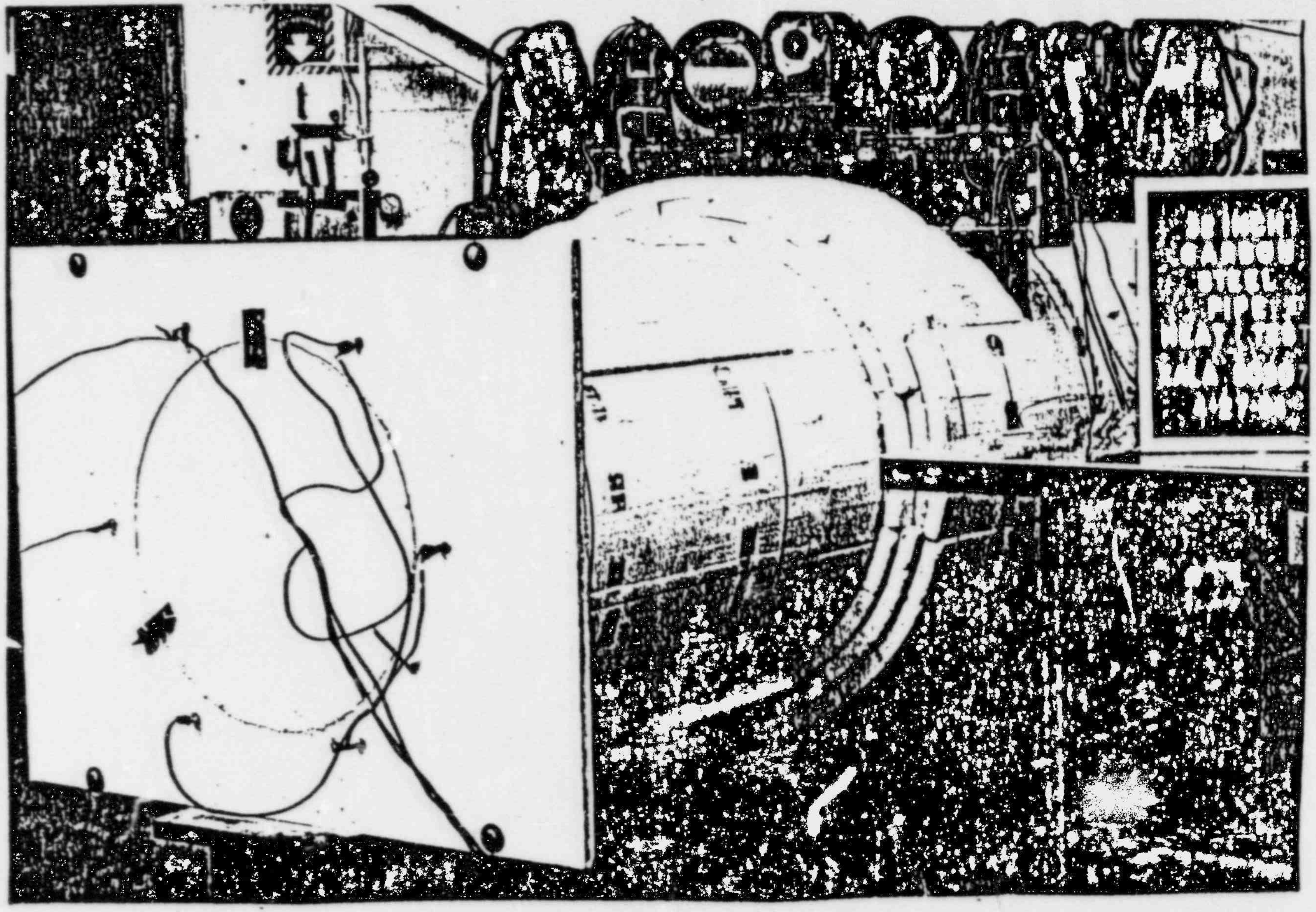
THERMAL CYCLING /THERMAL GRADIENT TEST AND CREEP TEST

4" STAINLESS STEEL SPECIMEN



THERMAL CYCLING/THERMAL GRADIENT TEST AND CREEP TEST
10" STAINLESS STEEL SPECIMEN

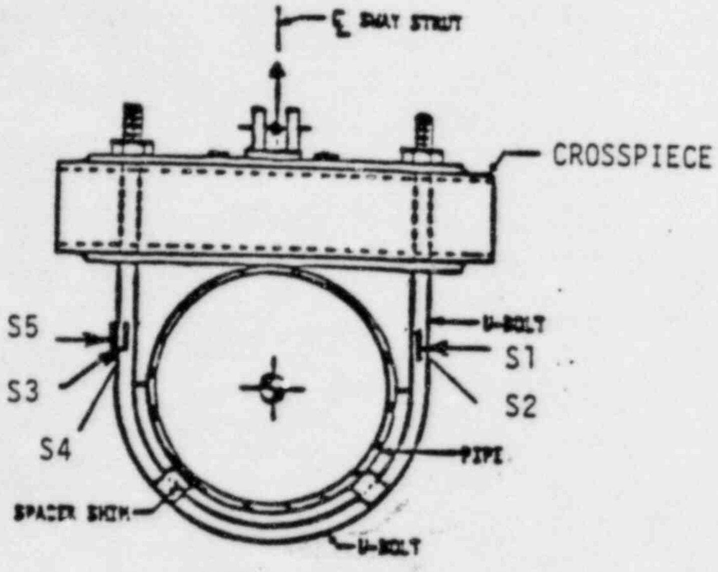
PHOTOGRAPH 10



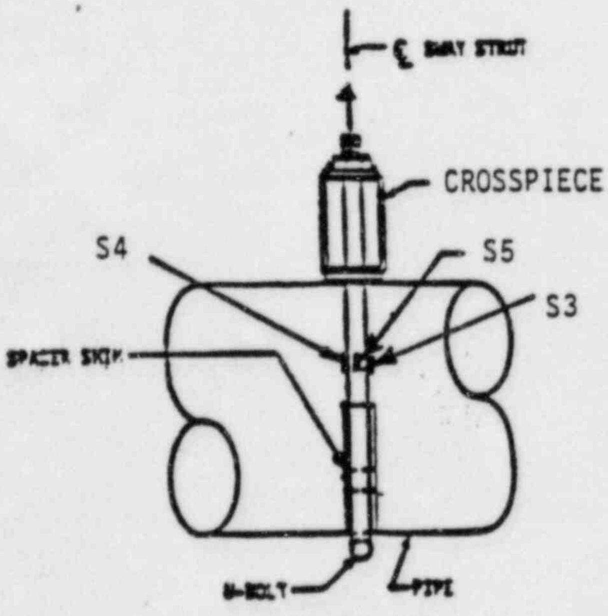
THERMAL CYCLING /THERMAL GRADIENT TEST AND CREEP TEST

32" CARBON STEFL SPECIMEN

PHOTOGRAPH 11



SECTION VIEW

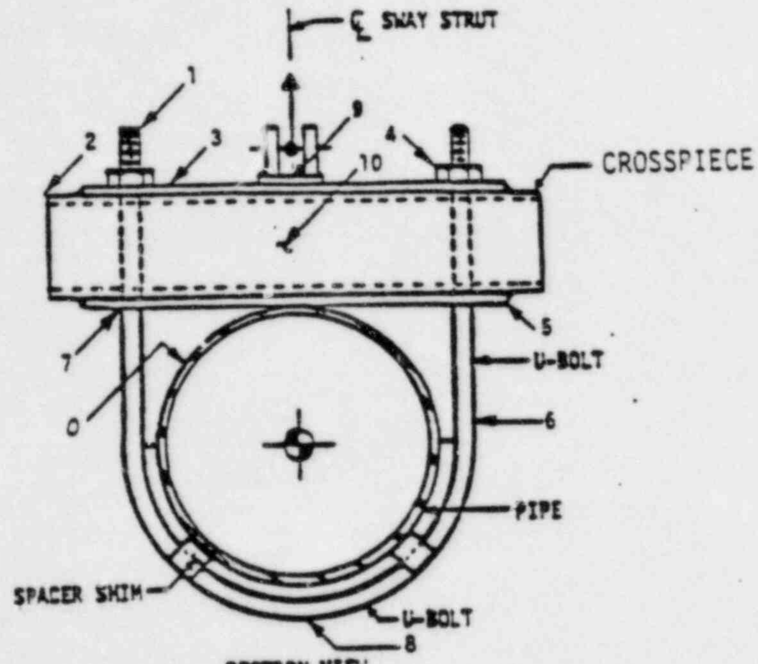


ELEVATION VIEW

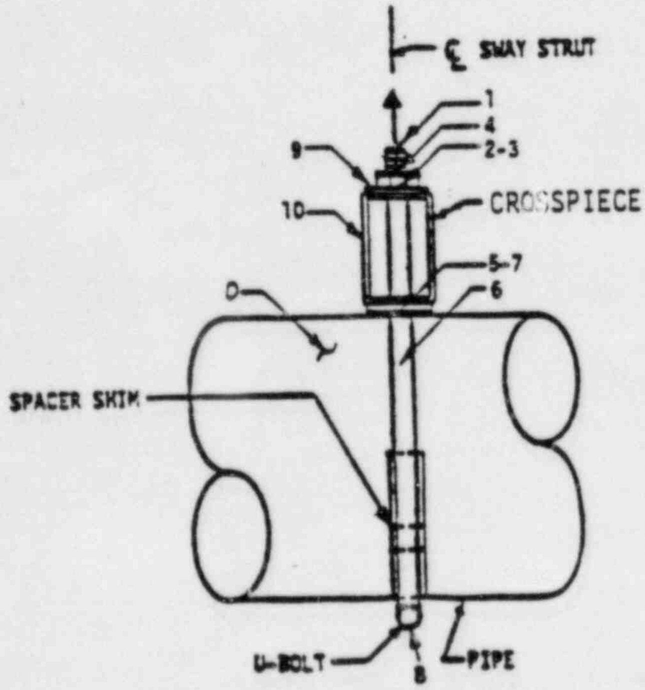
STRAIN GAUGE NUMBER	ACTIVE GAUGE LENGTH
S1-S18	0.40" FOR 4" AND 10" SPECIMENS 0.80" FOR 32" SPECIMEN

THERMAL CYCLING/THERMAL GRADIENT TEST
AND CREEP TEST
STRAIN GAUGE LOCATIONS

SKETCH 5



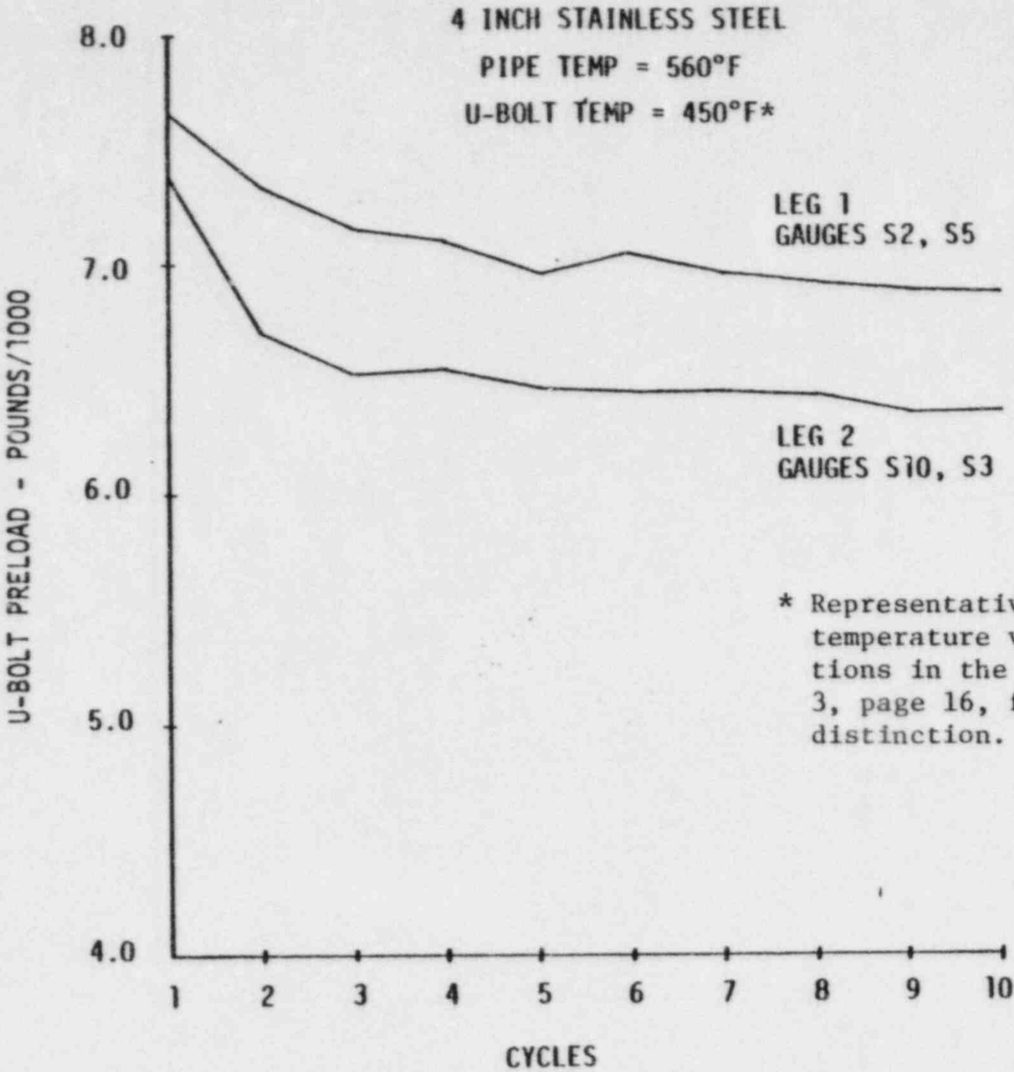
SECTION VIEW



ELEVATION VIEW

THERMAL CYCLING/THERMAL GRADIENT TEST
AND CREEP TEST
THERMOCOUPLE LOCATIONS

SKETCH 6



* Representative temperature. Actually, temperature varies at different locations in the U-bolt. Refer to Attachment 3, page 16, for actual temperature distinction.

FIGURE 21

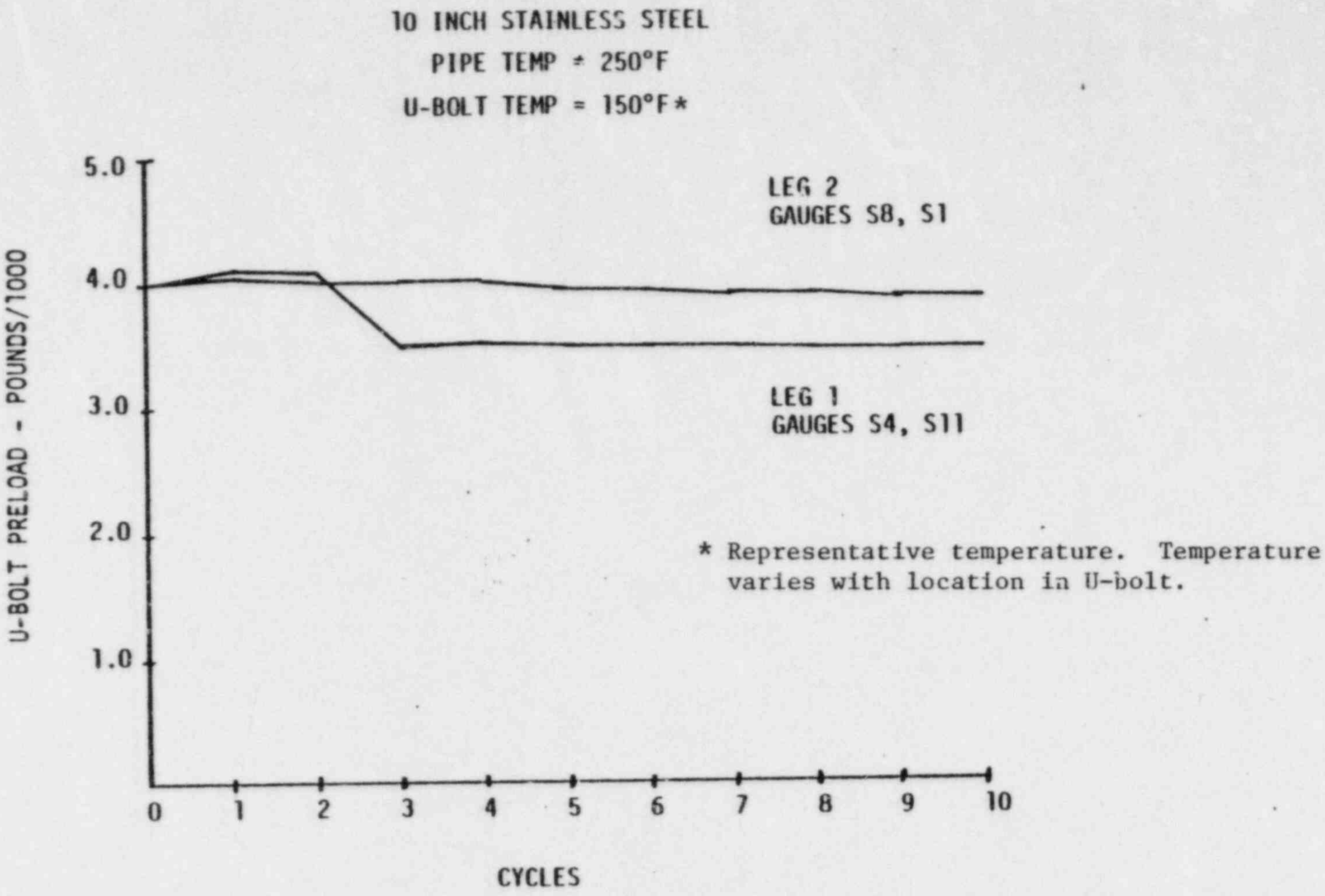
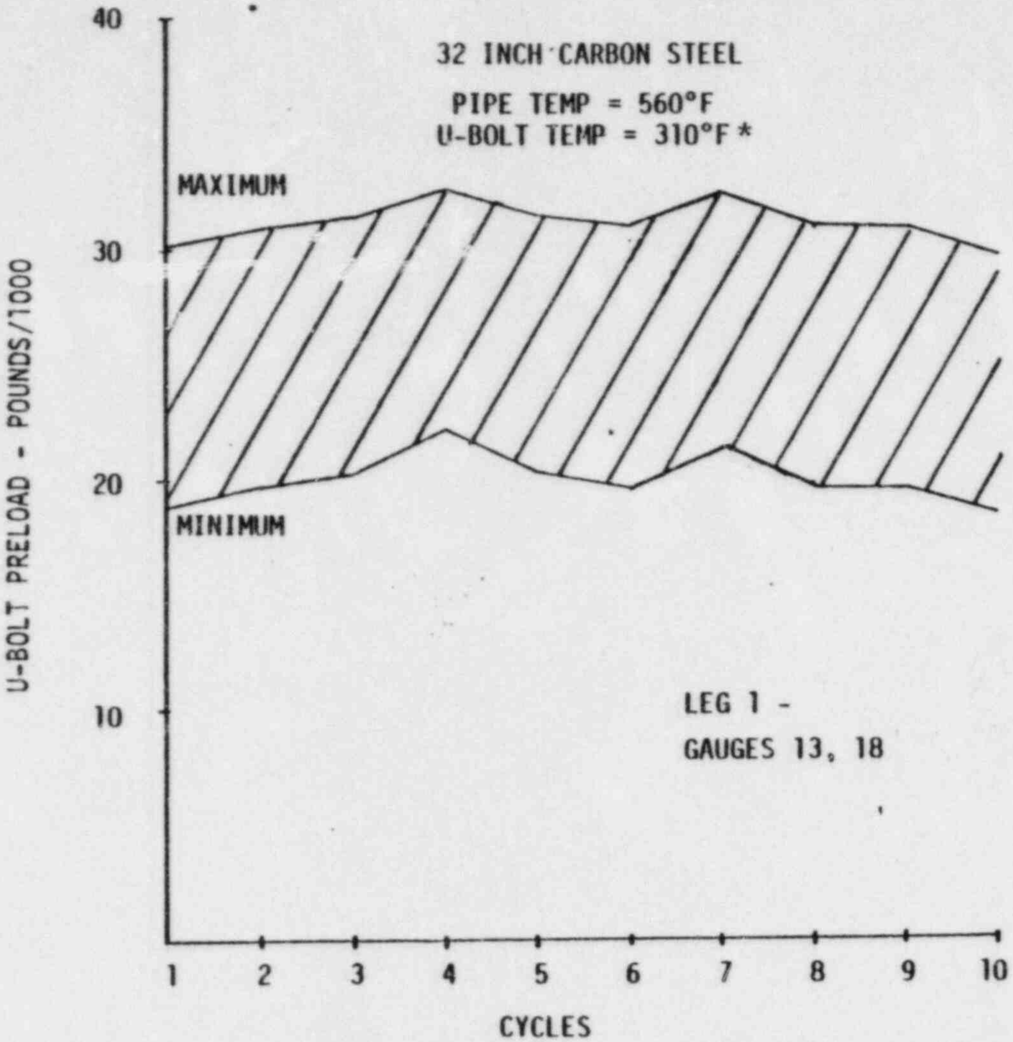
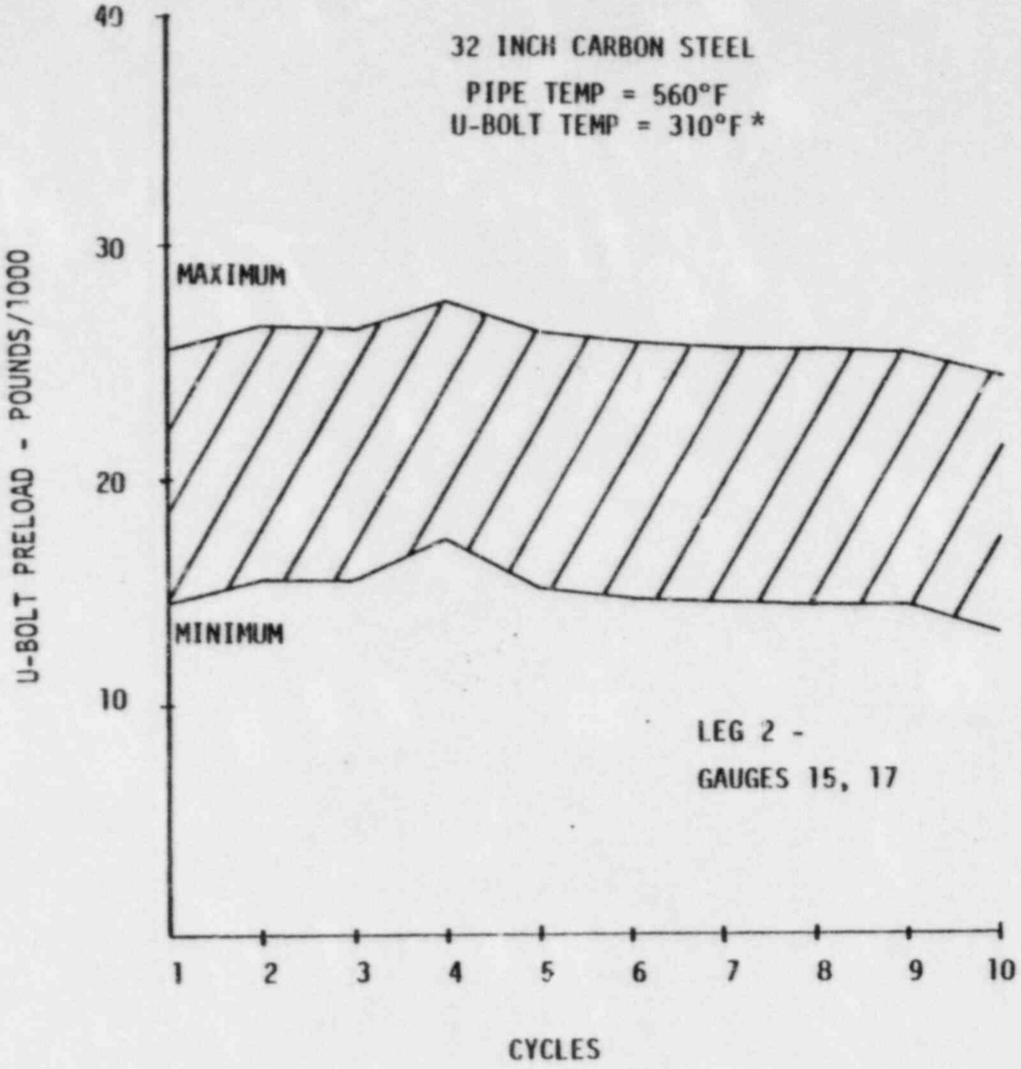

10 INCH STAINLESS STEEL
PIPE TEMP ≐ 250°F
U-BOLT TEMP = 150°F*
U-BOLT PRELOAD - POUNDS/1000
5.0
4.0
3.0
2.0
1.0
LEG 2
GAUGES S8, S1
LEG 1
GAUGES S4, S11
* Representative temperature. Temperature varies with location in U-bolt.
0
1
2
3
4
5
6
7
8
9
10
CYCLES

FIGURE 22



* Representative temperature. Temperature varies at different locations in the U-bolt. Refer to Attachment 3, page 16, for actual temperature distribution.

FIGURE 23



* Representative temperature. Temperature varies at different locations in the U-bolt. Refer to Attachment 3, page 16 for actual temperature distribution.

FIGURE 24

V.E. Creep Test

1.0 Purpose

The purpose of this test was to evaluate the effect of high temperature on the U-bolt preload as a function of time. This test was performed on three test specimens (pipes) using three designs of U-bolt type supports as specified and provided by TUGCO:

Pipe Test Specimens

32 inch carbon steel pipe, insulated
10 inch stainless steel pipe, uninsulated
4 inch stainless steel pipe, insulated

U-bolt Designs

32 inch U-bolt type pipe support
10 inch U-bolt type pipe support
4 inch U-bolt type pipe support

2.0 Test Configuration (See Photographs 9 through 11)

The Creep test was performed immediately after the Thermal Cycling test with the same test configurations as that used during the Thermal Cycling test.

3.0 Instrumentation

Instrumentation required for this test is identified below.

1. Strain gauge signal conditioning and readout devices
2. Thermocouple signal conditioning devices
3. Five strain gauges
4. Eleven thermocouples
5. Temperature data recorder

See Sketches 5 and 6 for strain gauge and thermocouple locations. The above instrumentation was calibrated prior to use.

4.0 Loading Devices

Loading devices used for these tests are identified below.

1. Strip heaters

5.0 Load Application

The test specimens were subjected to two types of loading during this test. Torque loads applied to the U-bolt nuts prior to the

Thermal Cycling/Thermal Gradient Test were maintained. Strip heaters were used to provide the thermal load.

6.0 Test Description

After each specimen had been installed and the instrumentation calibrated, testing commenced. The testing method was the same for all three specimens. The U-bolt nut torques were maintained from the Thermal Cycling test. The maximum test temperature was 560°F for both the 4 and 32 inch specimens and 250°F for the 10 inch specimen.

With the test specimen at the maximum test temperature, U-bolt strains were recorded at periodic time intervals over a 24 hour period. After 24 hours, the strip heaters were deactivated and the test was concluded.

7.0 Data Reduction

The test data recorded included U-bolt strains and temperatures of the pipe, U-bolt and crosspiece. Using the strain data, U-bolt preloads were determined. The analytical method for determining U-bolt preload is presented in Appendix I. This information was used to evaluate the effect of elevated temperature on the U-bolt preload as a function of time.

8.0 Data Description/Presentation

Test output was reduced and plotted so results could be described graphically. U-bolt preloads were plotted versus elapsed time at maximum temperature for each of the test specimens. This information is presented in Figures 25 through 28.

9.0 Results

The variation in preload over the 24 hour time period at maximum temperature was minimal for all three test specimens.

o 4 inch stainless steel - The maximum preload variation was 85 pounds, or 1.2% of the initial value of 6728 pounds at maximum pipe temperature. See Figure 25.

  The U-bolt preloads determined at ambient temperature (77°F) and 60 foot pounds of torque prior to creep test heat up were:

  Leg 1 (Gauges S2, S5) = 4871 pounds.
  Leg 2 (Gauges S10, S3) = 4410 pounds.

  The U-bolt preloads determined at ambient temperature (91.4°F) and 60 foot pounds of torque after creep test completion and cooldown were:

Leg 1 (Gauges S2, S5) = 4854 pounds.
Leg 2 (Gauges S10, S3) = 4398 pounds.

The 4 inch stainless steel specimen exhibited no loss of ambient condition U-bolt preload as a result of the creep test.

- 10 inch stainless steel - The maximum preload variation was 103 pounds, or 2.7% of the initial value of 3862 pounds at maximum pipe temperature. See Figure 26.

  The U-bolt preloads determined at ambient temperature (75.8°F) and 100 foot pounds of torque prior to creep test heat up were:

  Leg 1 (Gauges S8, S1) = 3625 pounds.
  Leg 2 (Gauges S4, S11) = 3587 pounds.

  The U-bolt preloads determined at ambient temperature (66.9°F) and 100 foot pounds of torque after creep test completion and cooldown were:

  Leg 1 (Gauges S8, S1) = 3593 pounds.
  Leg 2 (Gauges S4, S11) = 5316 pounds.

  The 10 inch stainless steel specimen exhibited a small loss of ambient condition U-bolt preload as a result of the creep test.

As described in Section 9.0 of the Thermal Cycling test description, apparent strains were determined for the gauges used on the 4 and 10 inch stainless steel specimens by calibration testing. The apparent strains were combined with the direct strain readings to assure that accurate strains were determined at the maximum temperatures.

- 32 inch carbon steel - The maximum preload variance was 517 pounds or 3.8% of the initial value of 13,436 pounds at maximum pipe temperature. This is based on the lower limit of the 95 percent confidence level for Leg 1. See Figure 27.

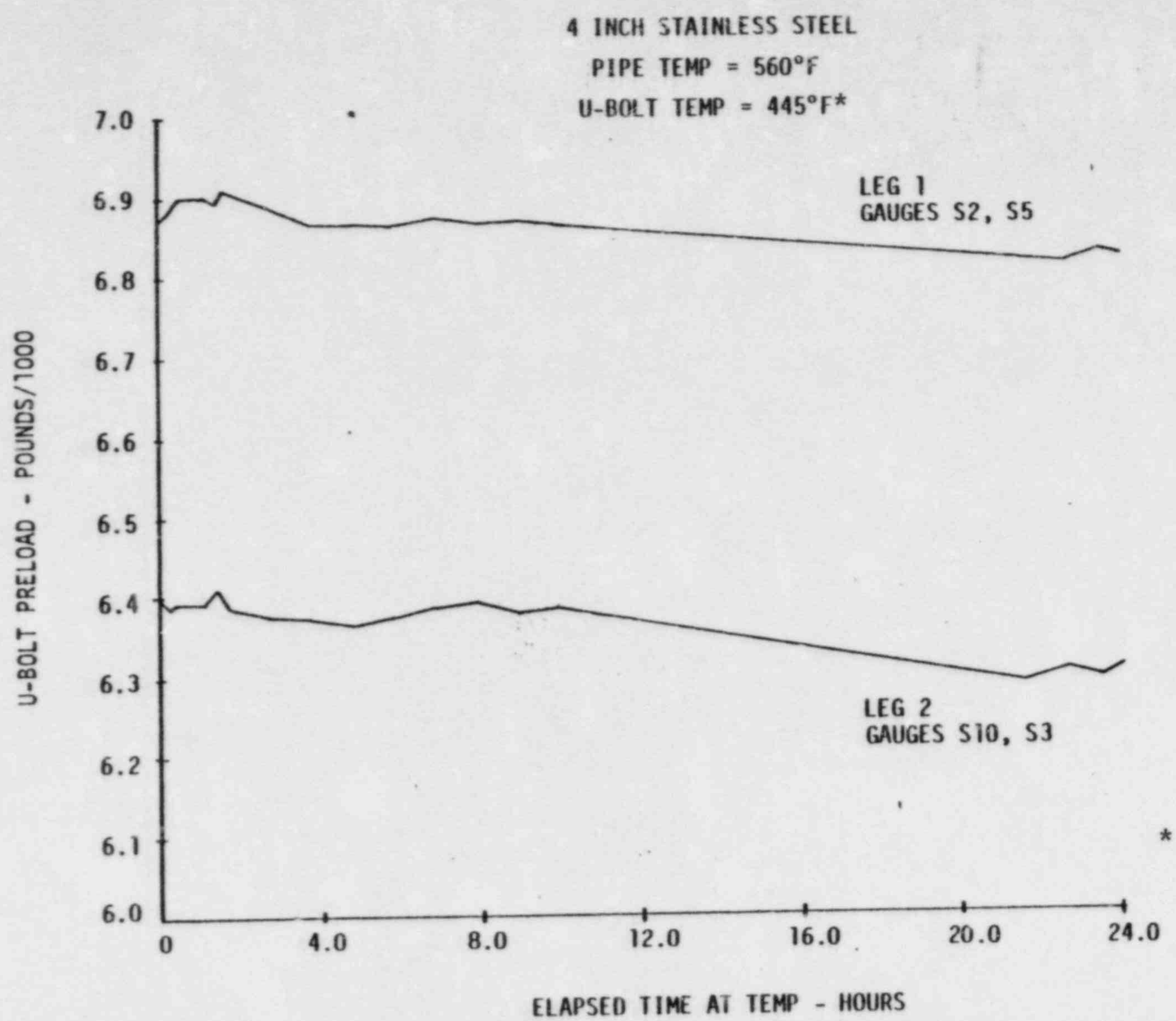
  The U-bolt preloads determined at ambient temperature (86.7°F) and 600 foot pounds of torque after creep test completion and cooldown were:

  Leg 1 (13, 18) = 8526 pounds.
  Leg 2 (15, 17) = 9904 pounds.

Because there was no cooldown between the Thermal Cycling and Creep testing, U-bolt preloads could not be determined at ambient temperature prior to Creep Test heat up.

Apparent strain data was not directly available for the gauges used on the 32 inch specimen. Since the same type of gauges were used on the 10 inch specimen, a statistical evaluation was performed and a confidence interval was determined for apparent strain at a 95 percent confidence level. This confidence interval was applied to the apparent strain curves obtained for the 10 inch specimen so that apparent strain ranges were established for use with the 32 inch specimen.

Apparent strain is a function of gauge type and material to which the gauge is attached. Both the 10 and 32 inch U-bolts are of the same material. As noted previously, the same type of gauges were used on both. U-bolt size has no effect on apparent strain.



* Representative temperature.

FIGURE 25

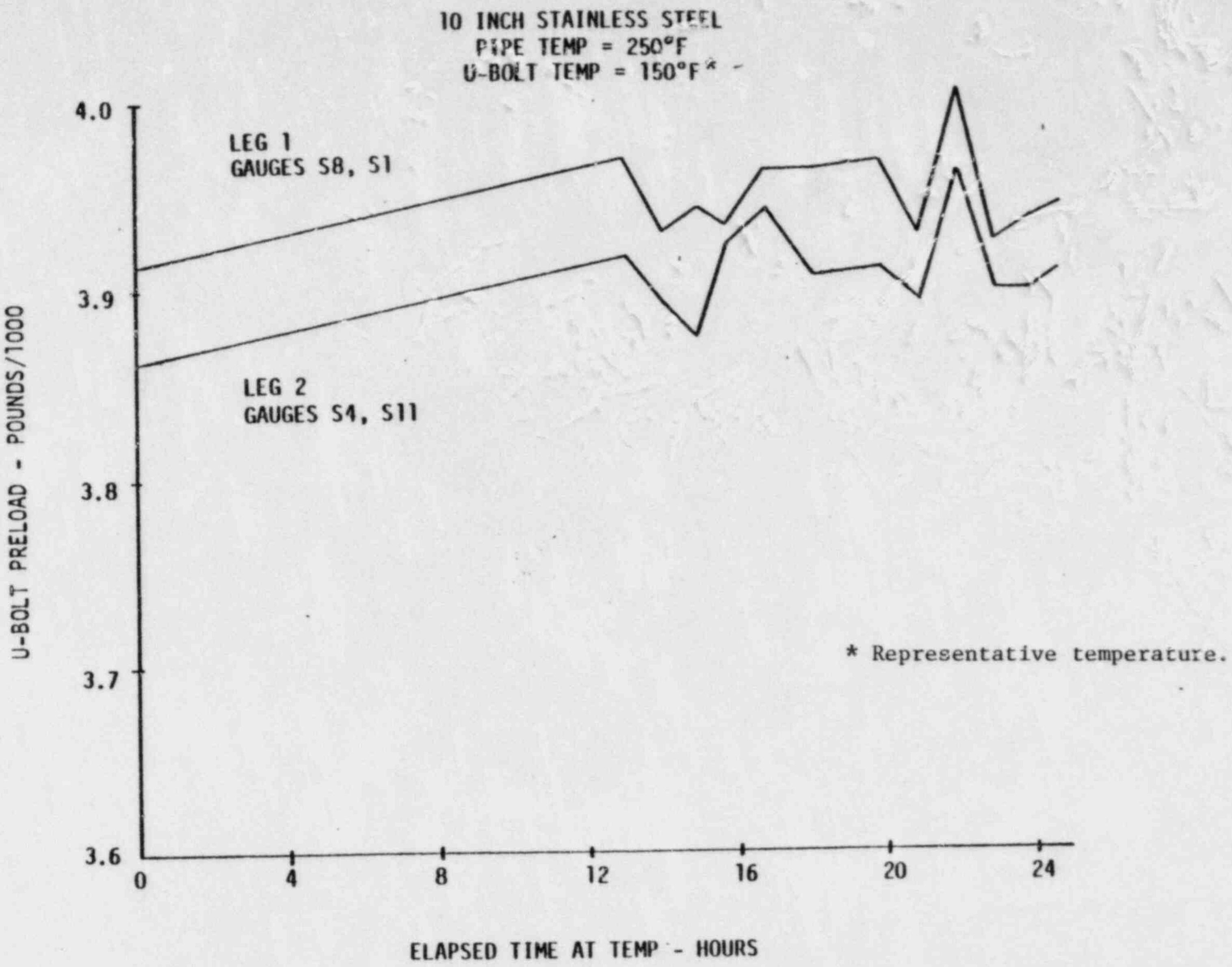
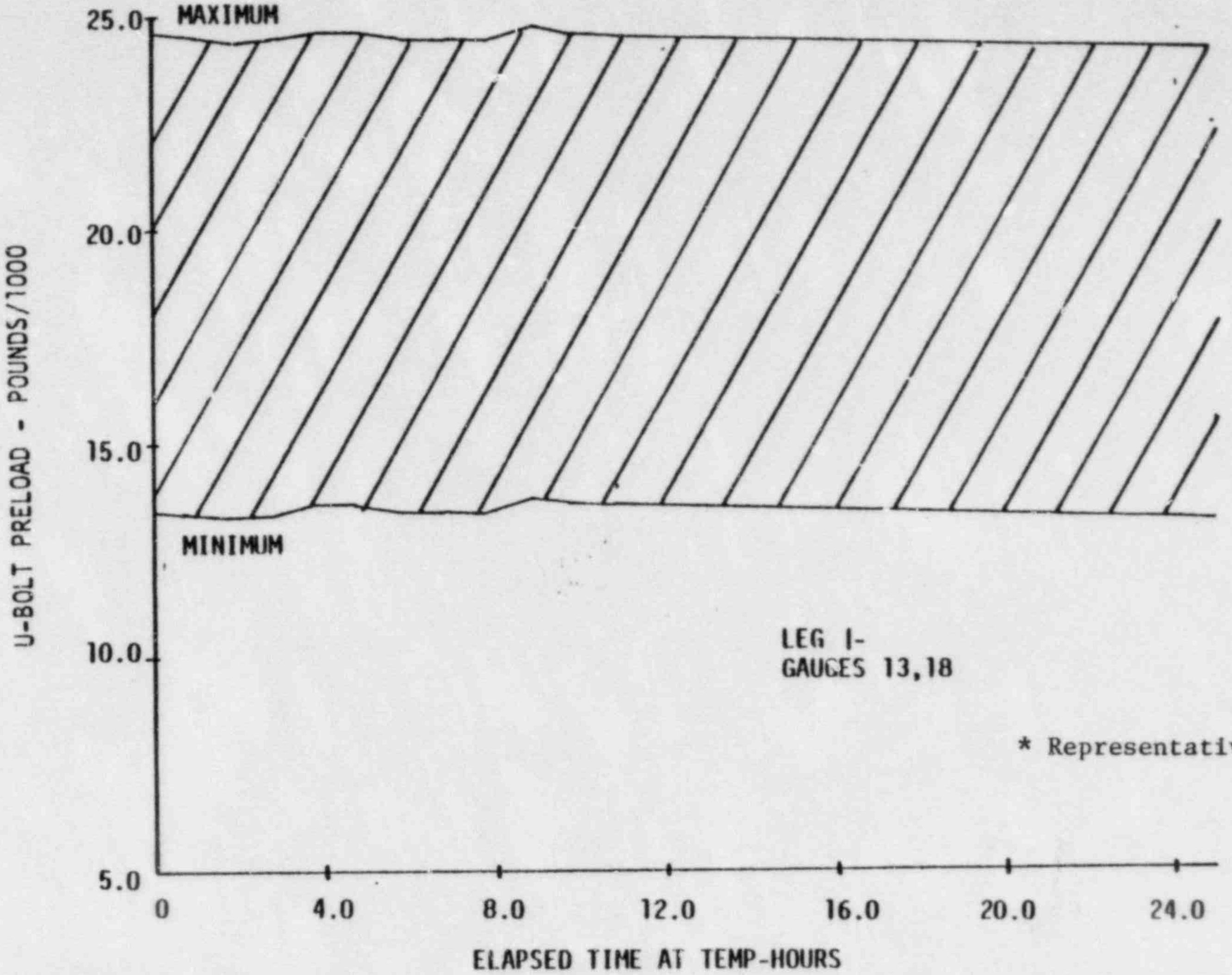

10 INCH STAINLESS STEEL
PIPE TEMP = 250°F
U-BOLT TEMP = 150°F *
LEG 1
GAUGES S8, S1
LEG 2
GAUGES S4, S11
U-BOLT PRELOAD - POUNDS/1000
4.0
3.9
3.8
3.7
3.6
0
4
8
12
16
20
24
* Representative temperature.
ELAPSED TIME AT TEMP - HOURS

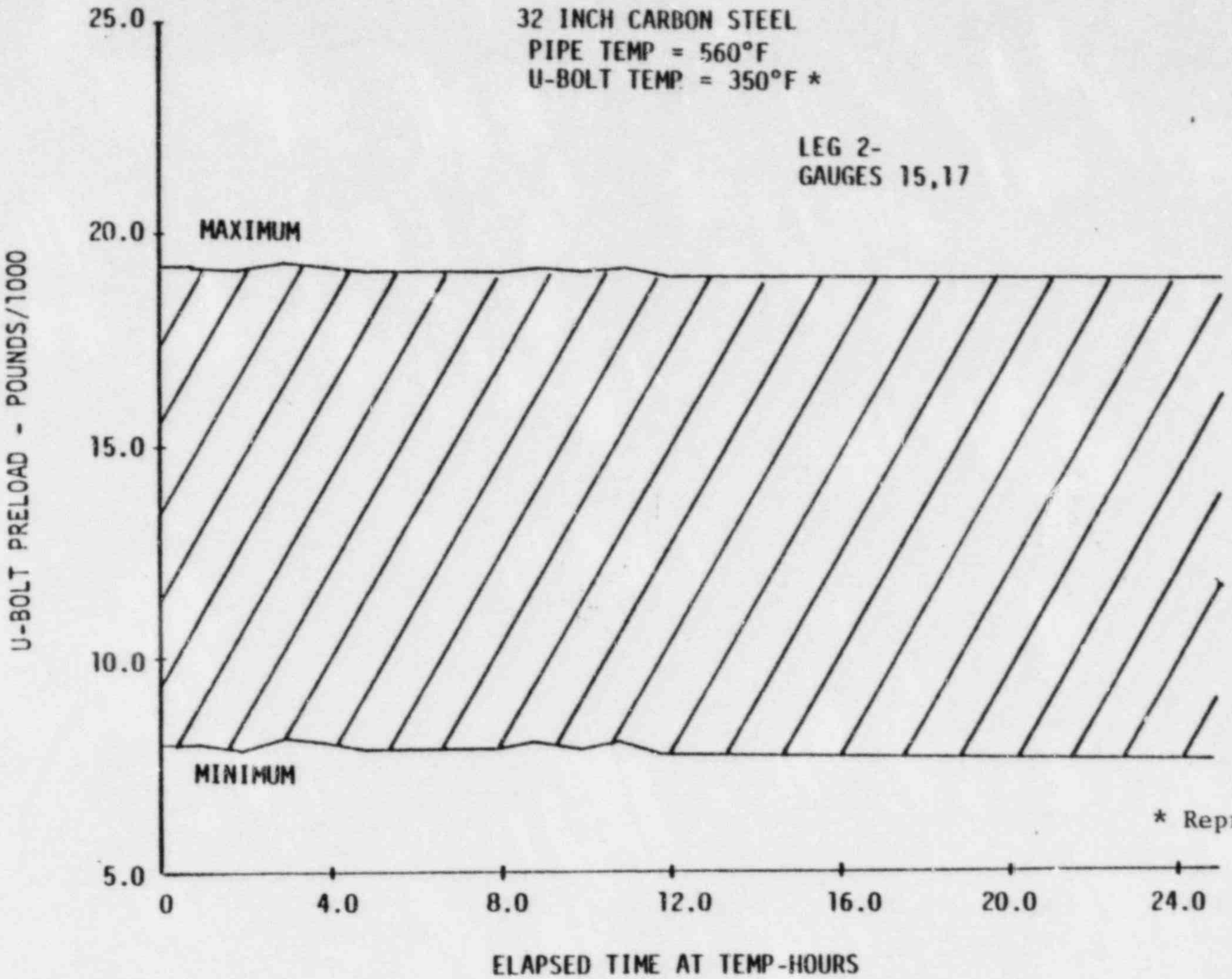
FIGURE 26

32 INCH CARBON STEEL
PIPE TEMP = 560°F
U-BOLT TEMP = 350°F*



* Representative temperature.

FIGURE 27



* Representative temperature.

FIGURE 28

V.F. Normal Vibration Simulation Test

1.0 Purpose

The purpose of this test was to evaluate the effects of piping vibration on the U-bolt preload. The U-bolt was preloaded to the value of torque that provided a conforming fit between the U-bolt and pipe. This test was performed on the 10 inch stainless steel pipe specimen as specified and provided by TUGCO.

2.0 Test Configuration (See Photograph 12)

The pipe test specimen described in Section 1.0 had a mounting plate welded to each end. These plates were then fastened to two floor mounted brackets using high strength fasteners. Strain gauges were installed on the U-bolt. The U-bolt support was then installed in the center of the test pipe with the crosspiece perpendicular to the floor. The U-bolt nuts were hand tightened. An electrodynamic shaker was fixtured so that a forcing function could be induced to the U-bolt support clevis via a strut. The strut was arranged such that the pipe test specimen experienced triaxial loading.

3.0 Instrumentation

Instrumentation required for this testing is identified below.

1. Strain gauge signal conditioning and readout devices
2. Five strain gauges

The strain gauges were installed on each test specimen prior to testing. See Sketch 7 for location of strain gauges. The above instrumentation was calibrated prior to use.

4.0 Loading Devices

Loading devices used for this test are identified below.

1. Torque wrench
2. Electrodynamic shaker

The torque wrench was calibrated prior to use.

5.0 Load Application

The loads experienced by the test specimen were due to U-bolt nut torque and the electrodynamic shaker. Testing was performed at laboratory ambient environmental conditions.

6.0 Test Description

After the specimen had been installed and the instrumentation calibrated, testing commenced. With the U-bolt located at the center of the pipe with the crosspiece perpendicular to the floor, the nuts were torqued to 50 foot pounds. With the test specimen in the unloaded condition, recording equipment was activated. The electrodynamic shaker was then activated to induce a sinusoidal load of 1500 pounds on the specimen as specified by TUGCO. The electrodynamic shaker was calibrated to assure the load magnitude was 1500 pounds. The frequency varied from 5 to 200 to 5 hertz at a rate of 2 octaves per minute. The test duration was 270 minutes. This is equivalent to 90 minutes of vibration in each of the three axes. IEEE 382-1980 was used as a basis for establishing test parameters. IEEE 382-1980 is an industry accepted standard for simulating piping vibration in a plant.

7.0 Data Reduction

The test data recorded included U-bolt strains. Using the strain data, U-bolt preloads were determined. The analytical method for determining U-bolt preload is presented in Appendix I. This information was used to evaluate the effect of vibration on the U-bolt preload.

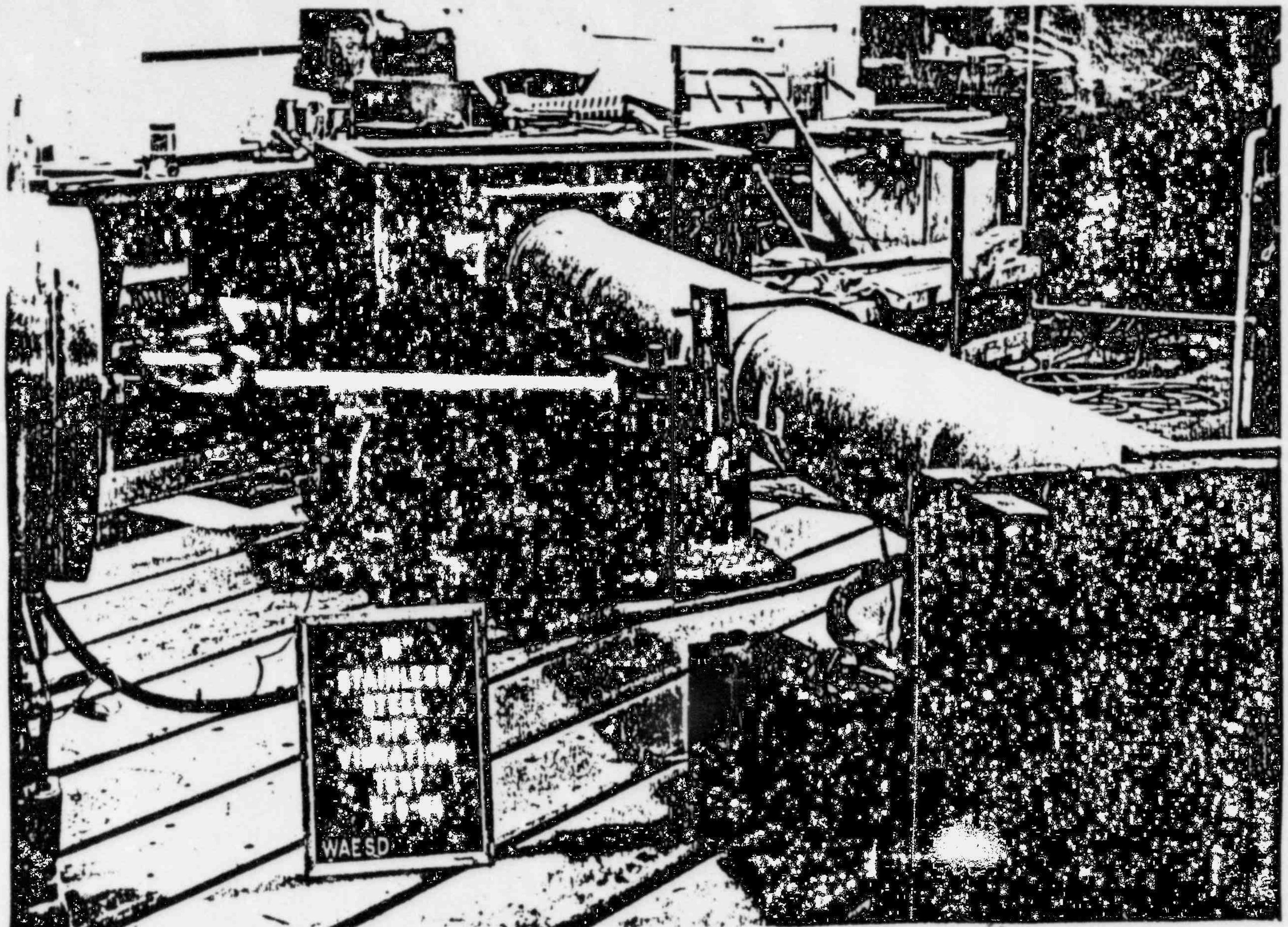
8.0 Data Description/Presentation

Test output was reduced and plotted so results could be presented graphically. U-bolt preload was plotted in Figure 29 as a function of test time at vibration.

9.0 Results

The initial preload resulting from the U-bolt nut torque of 50 foot pounds was 4285 pounds in both Leg 1 (Gauges S1, S2) and Leg 2 (Gauges S3, S4). During the first few seconds of vibration, the U-bolt support moved in the axial pipe direction slightly (< 1/16 inch) but repositioned itself to a location where it remained during the rest of the test. This repositioning resulted in a decrease of U-bolt preload of approximately 600 pounds.

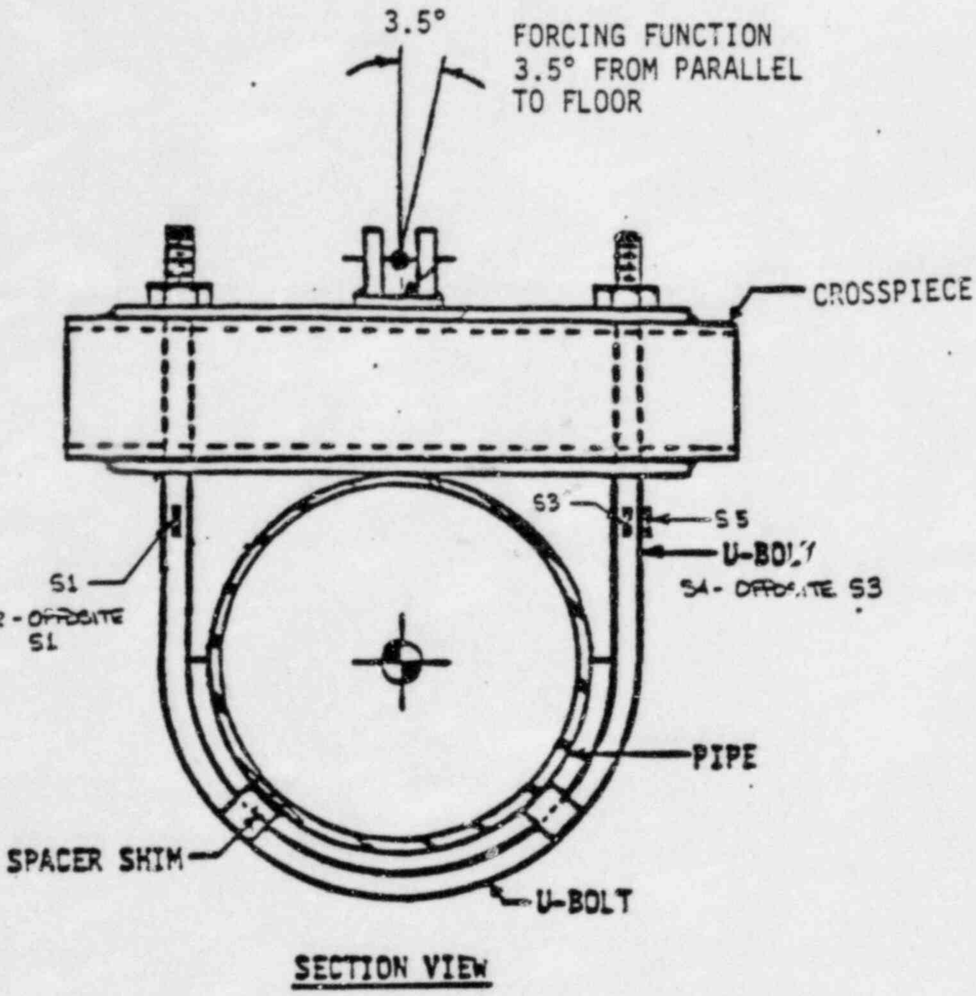
The maximum variation in preload, after repositioning occurred, was 1083 pounds in Leg 1 and 621 pounds in Leg 2. The changes in preloads based on data recorded 5.25 minutes into the test and 270 minutes into the test were 660 pounds in Leg 1 and 564 pounds in Leg 2. See Figure 29.



VIBRATION TEST

10" STAINLESS STEEL SPECIMEN

PHOTOGRAPH 12



SECTION VIEW

VIBRATION TEST
STRAIN GAUGE LOCATIONS

STRAIN GAUGE NUMBER	ACTIVE GAUGE LENGTH
S1-S5	0.25"

SKETCH 7

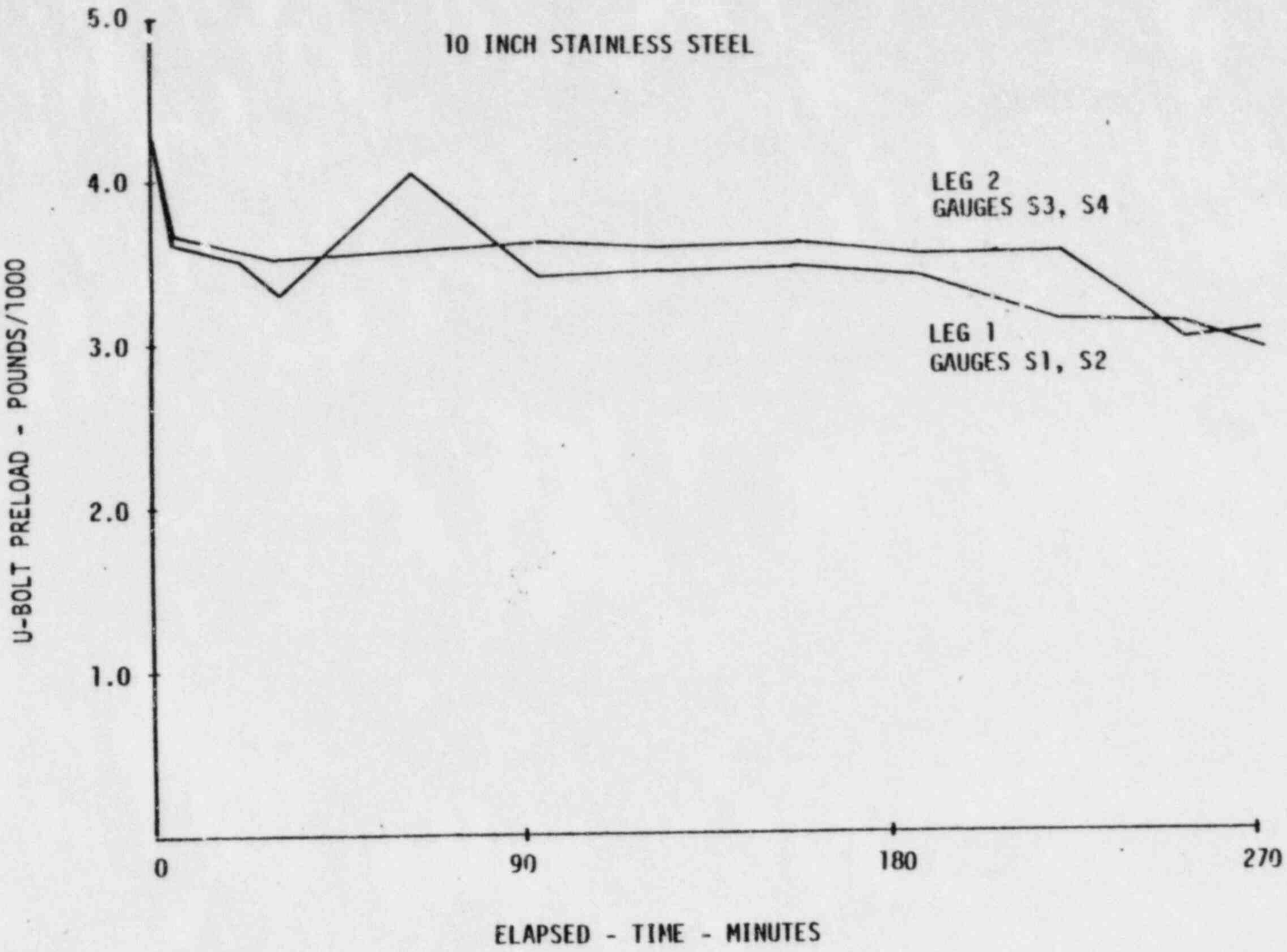

10 INCH STAINLESS STEEL
U-BOLT PRELOAD - POUNDS/1000
5.0
4.0
3.0
2.0
1.0
0
90
180
270
LEG 2
GAUGES S3, S4
LEG 1
GAUGES S1, S2
ELAPSED - TIME - MINUTES

FIGURE 29

APPENDIX I

ENGINEERING CALCULATIONS

Following are the engineering calculations used to reduce test data and graphically present results.

A. U-bolt preload (or load)

Preload (or load) = microstrain x Ex bolt shank area

Where: E = Young's modulus = 29×10^6 lb/in^2
and: bolt shank areas are
4 inch specimen = .196 in^2
16 inch specimen = .442 in^2
32 inch specimen = 5.94 in^2

Strains were recorded as test output. Stains recorded were in units of 10^{-6} in/in. Stains of 10^{-6} are defined as microstrain.

B. Pipe and crosspiece stresses

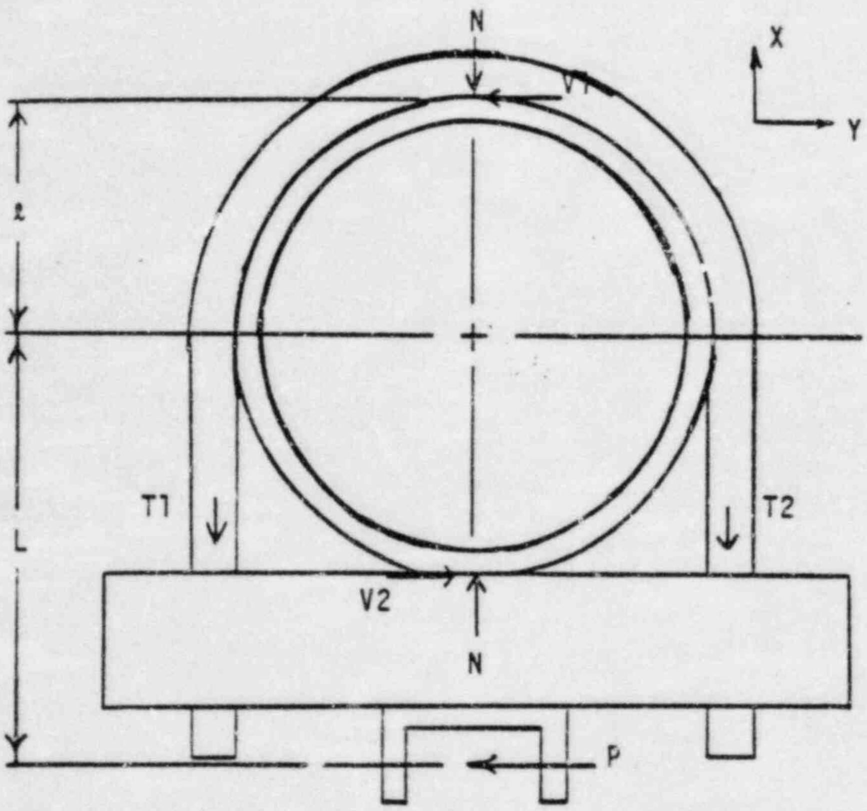
Stresses were obtained from the elastic stress-strain relationship:

$\sigma = \varepsilon E$

Where: ε = microstrain
E = Young's modulus
= 28.3×10^6 lb/in^2 for stanless steel
= 27.9×10^6 lb/in^2 for carbon steel

C. Friction coefficient

The coefficient of friction between the U-bolt support assembly and the pipe is determined as follows:



Summing moments about the pipe centerline:

$$PL = \ell\,(V1 + V2)$$

$$\frac{PL}{\ell} = V1 + V2$$

Summing forces in the Y direction:

$$P = V2 - V1$$

By combining the last two expressions:

$$P\left(1 + \frac{L}{\ell}\right) = V2$$

Upon determination of V1 and V2, the average coefficient of friction between the U-bolt support and pipe is expressed as:

$$\mu = \frac{1}{2}\left(\frac{V1}{N} + \frac{V2}{N}\right)$$

Where: $N = T1 + T2$

P = Slip force

μ = Coefficient of friction

D. 95% confidence level determination for apparent strains of 32" specimen

$$\text{Max, Min} = \overline{X} \pm \frac{S}{\sqrt{n}}\, t\, \frac{\alpha}{2}$$

Where: $\overline{X} = \text{average} = \sum_{i=1}^{n} Xi/n$

$S = \text{standard deviation} = \left[\sum_{i=1}^{n} (Xi - \overline{X})^2/(n-1)\right]^{1/2}$

n = number of samples = 4

$t\,\frac{\alpha}{2} = 3.182$ for 95% confidence level

E. Friction test strain criteria

Maximum strains measured on the bolt shank were determined such that resulting stresses in the threads are 90% of material yield.

For A36 material, $S_y = 36,000$ psi

$$.9S_y = 32,400 \text{ psi}$$

First determine strains that would cause shank stresses of 32,400 psi.

$$\varepsilon = \frac{\sigma}{E} = \frac{32,400}{27.9 \times 10^6}$$

$$\varepsilon = 1161\ \mu \text{ inches/inch}$$

Where: ε = microstrain
E = Young's modulus

Since there is a difference in area between bolt shank and thread, the microstrain that causes a stress of 32,400 psi can be determined using a ratio of the shank and thread areas.

Shank areas		Thread areas	
1/2 inch:	.196 in^2	1/2 inch:	.1419 in^2
3/4 inch:	.442 in^2	3/4 inch:	.334 in^2
2 3/4 inch:	5.94 in^2	2 3/4 inch:	4.93 in^2

The shank microstrain that would cause a thread stress of 32,400 psi is determined by:

$$\text{microstrain} = 1161 \times \frac{\text{Thread area}}{\text{Shank area}}$$

The required microstrains are:

1/2 inch: 839 μ inches/inch
3/4 inch: 877 μ inches/inch
2 3/4 inch: 964 μ inches/inch

APPENDIX II

U-BOLT IDENTIFICATION

Following is a summary of which U-bolts were installed on the test specimens for each specific test.

Torque versus Preload

32 inch carbon steel pipe:	32 inch U-bolt - since only one 32 inch U-bolt was supplied, it was used on all testing conducted on the 32" carbon steel specimen.
10 inch stainless steel pipe:	10 inch U-bolt #1.
10 inch carbon steel pipe:	10 inch U-bolt #1.
4 inch stainless steel pipe:	4 inch U-bolt #1.

Friction

32 inch carbon steel pipe:	32 inch U-bolt.
10 inch stainless steel pipe:	10 inch U-bolt #2 - Polishing test used 10 inch U-bolt #1.
10 inch carbon steel pipe:	10 inch U-bolt #4 - Polishing test used 10 inch U-bolt #3.
4 inch stainless steel pipe:	4 inch U-bolt #2 - Polishing test used 4 inch U-bolt #1.

Load Distribution/Strain Measurement

10 inch stainless steel pipe:	10 inch U-bolt #2.

Thermal Cycling/Thermal Gradient

32 inch carbon steel pipe: 32 inch U-bolt.

10 inch stainless steel pipe: 10 inch U-bolt #4.

4 inch stainless steel pipe: 4 inch U-bolt #1.

Creep

32 inch carbon steel pipe: 32 inch U-bolt.

10 inch stainless steel pipe: 10 inch U-bolt #4.

4 inch stainless steel pipe: 4 inch U-bolt #1.

Vibration

10 inch stainless steel pipe: 10 inch U-bolt #3.

The U-bolts were identified by a unique number for their use on a specific pipe size prior to the initiation of the test program.

ADDENDUM 1 TO

REPORT NO. EQ&T-EQT-860

REVISION 0

COMANCHE PEAK STEAM ELECTRIC STATION

U-BOLT SUPPORT/PIPE TEST REPORT

WESTINGHOUSE ELECTRIC CORPORATION

P.O. BOX 355

PITTSBURGH, PA 15230

Mark Kamenic 6/5/84

PREPARED BY: MARK KAMENIC

J.M. Snider 6-5-84

REVIEWED BY: J.M. SNIDER

6/5/84

APPROVED BY: L.I. WALKER

Seismic Loading Simulation Test

1.0 Purpose

The purpose of this test was to evaluate the effects of seismically induced piping vibration on U-bolt preload. This test was performed on the 10 inch stainless steel pipe specimen as specified and provided by TUGCO.

2.0 Test Configuration (See Photograph 1)

The pipe test specimen described in Section 1.0 had a mounting plate welded to each end. These plates were then fastened to two floor mounted brackets using high strength fasteners. Strain gauges were installed on the U-bolt. The U-bolt support was then installed in the center of the test pipe with the crosspiece perpendicular to the floor. The U-bolt nuts were hand tightened. A hydraulic shaker was fixtured so that a forcing function could be induced into the U-bolt support clevis by a strut. The strut was arranged such that the pipe test specimen experienced triaxial loading. The strut was instrumented with strain gauges to form a load cell.

3.0 Instrumentation

Instrumentation required for this testing is identified below.

1. Strain gauge signal conditioning and readout devices
2. Magnetic tape recorder
3. Six strain gauges

The strain gauges were installed on the test specimen prior to testing. See Sketch 1 for location of strain gauges. The above instrumentation was calibrated prior to use.

4.0 Loading Devices

Loading devices used for this test are identified below.

1. Torque wrench
2. Hydraulic shaker

The torque wrench was calibrated prior to use.

5.0 Load Application

The test specimen experienced U-bolt preload and seismically induced vibration resulting from the U-bolt nut torque and hydraulic shaker, respectively. Testing was performed at laboratory ambient environmental conditions.

6.0 Test Description

After the specimen had been installed and the instrumentation calibrated, testing commenced. With the U-bolt located at the center of the pipe with the crosspiece perpendicular to the floor, the nuts were torqued to 50 foot pounds. This value of torque provided a conforming fit between the U-bolt and the pipe. Seismic loading simulations were then performed. The first simulation was performed using a sinusoidal input at 9 hertz. The input force as a function of test time is shown in Figure 1. The second simulation was performed using a maximum sinusoidal input of 1300 pounds at the test specimen natural frequency. Using the strain gauges installed on the strut, the hydraulic shaker was calibrated to assure the proper load magnitudes were achieved.

The first seismic loading simulation was performed at 9 hertz with full amplitude input lasting 39 seconds. A magnetic recorder was used to record the U-bolt strains as well as the input load from the strut load cell gauges.

The second seismic loading simulation was conducted at the natural frequency of the pipe test specimen in the fixture. The natural frequency was determined by exciting the pipe with a 1000 pound sinusoidal input and sweeping upward from 5 hertz until the natural frequency was found. An accelerometer mounted on the U-bolt support crosspiece was used to identify the natural frequency. Once the natural frequency had been determined, an input level of 7000 pounds was to be introduced at the system natural frequency. However, at the system natural frequency the hydraulic actuator was not capable of transmitting the 7000 pound force due to movement in the strut connections (resulting from design and manufacturing tolerances) which exceeded the displacement output of the actuator. As a result, the maximum input level that was achievable from the actuator was used for this test. This level, 1300 pounds, was applied at the natural frequency. The minimum duration of this test was defined as 300 cycles divided by the natural frequency in cycles per second. A magnetic tape recorder was used to record the U-bolt strains as well as the input load from the strut load cell.

7.0 Data Reduction

U-bolt strains and load input were recorded. Using the strain data, U-bolt preloads were determined. The analytical method for determining U-bolt preload is presented in Appendix I of EQ&T-EQT-860, Revision 0. This information was used to evaluate the effect of seismic level piping vibration on the U-bolt preload.

8.0 Data Description/Presentation

Test output was reduced and plotted so results could be represented graphically. The input force was plotted in Figure 1 as a function of test time for the 9 hertz seismic loading simulation test. This figure shows the input forces in the push (U-bolt compressive) and pull (U-bolt tensile) directions in addition to the average force magnitude. U-bolt preload is shown in Figure 2 as a function of test time for the 9 hertz seismic loading simulation test.

9.0 Results

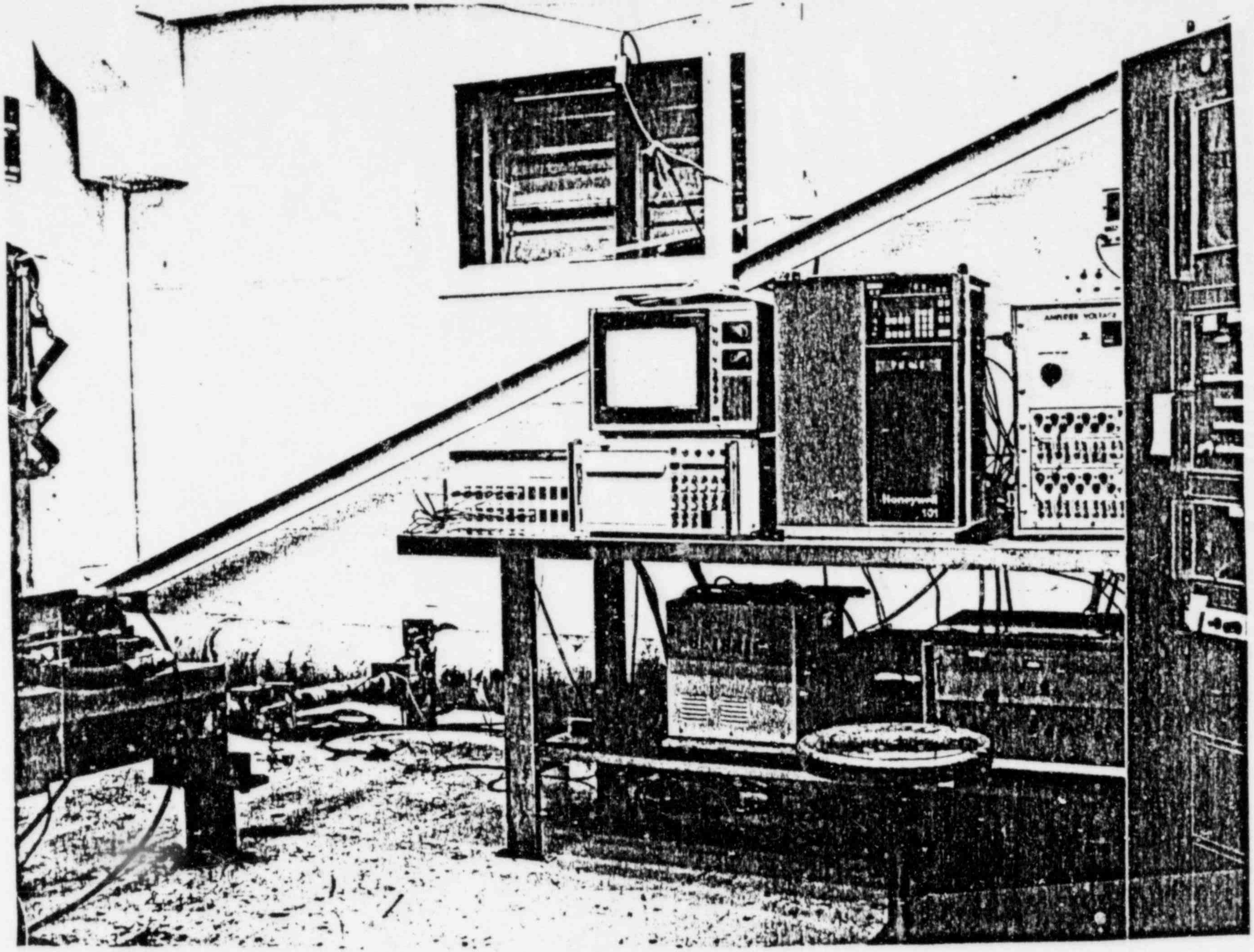
The initial preload resulting from the U-bolt nut torque of 50 foot pounds prior to initiation of any vibratory input was 4484 pounds in both Leg 1 (Gauges S1, S2) and Leg 2 (Gauges S3, S4). Four seconds elapsed from the time vibration was initiated until the full amplitude input was reached. During that time the U-bolt preloads decreased to 4375 pounds and 4426 pounds in Legs 1 and 2, respectively. These values are indicated at the zero time point on the abscissa of Figure 2 and represent the U-bolt preload at the time the vibratory input reached full amplitude.

After completion of the seismic loading simulation test (with no external cyclic load applied), strains remaining in the U-bolt corresponded to a preload of 3914 pounds in each of the two legs. Thus a relaxation of 570 pounds from the original 4484 pound preload occurred in both U-bolt legs. The U-bolt support was observed to move approximately one-half inch in the axial (toward the strut) direction. No rotational motion was perceptible. Prior to running this test, three trial tests were run to adjust test equipment and instrumentation to obtain the desired input. Although none of the three trial tests can be considered official, their observed behavior was the same as that of the official test.

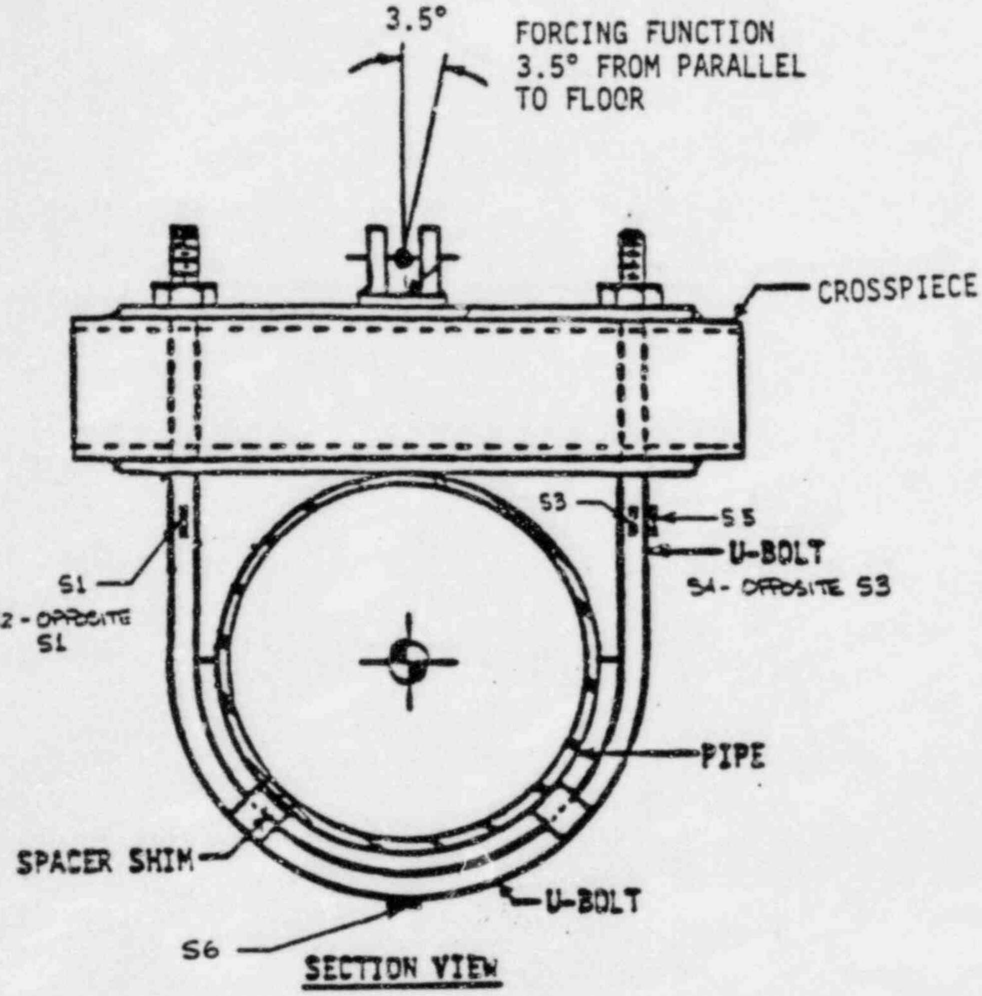
The natural frequency of the test specimen as mounted in the fixture was found to be 75 hertz. As stated in Section 6.0, a seismic test was to be performed at the test specimen natural frequency using an input load magnitude of 7000 pounds. However, at this higher frequency, the hydraulic actuator was not capable of transmitting the required 7000 pounds of force because the movement in the strut connections resulting from design and manufacturing tolerances exceeded the displacement output of the actuator at 75 hertz.

The maximum actuator force output at 75 hertz was 1300 pounds. The duration of the 1300 pound, 75 hertz test was in excess of four seconds. During the 75 hertz test, the U-bolt support was observed to exhibit barely perceptible axial motion (toward the strut) and no rotational motion.

Because the desired 7000 pound load was not achievable during this test, the test was considered unofficial. For this reason no other test data is presented.



SEISMIC LOADING SIMULATION TEST
10" STAINLESS STEEL SPECIMEN
PHOTOGRAPH 1

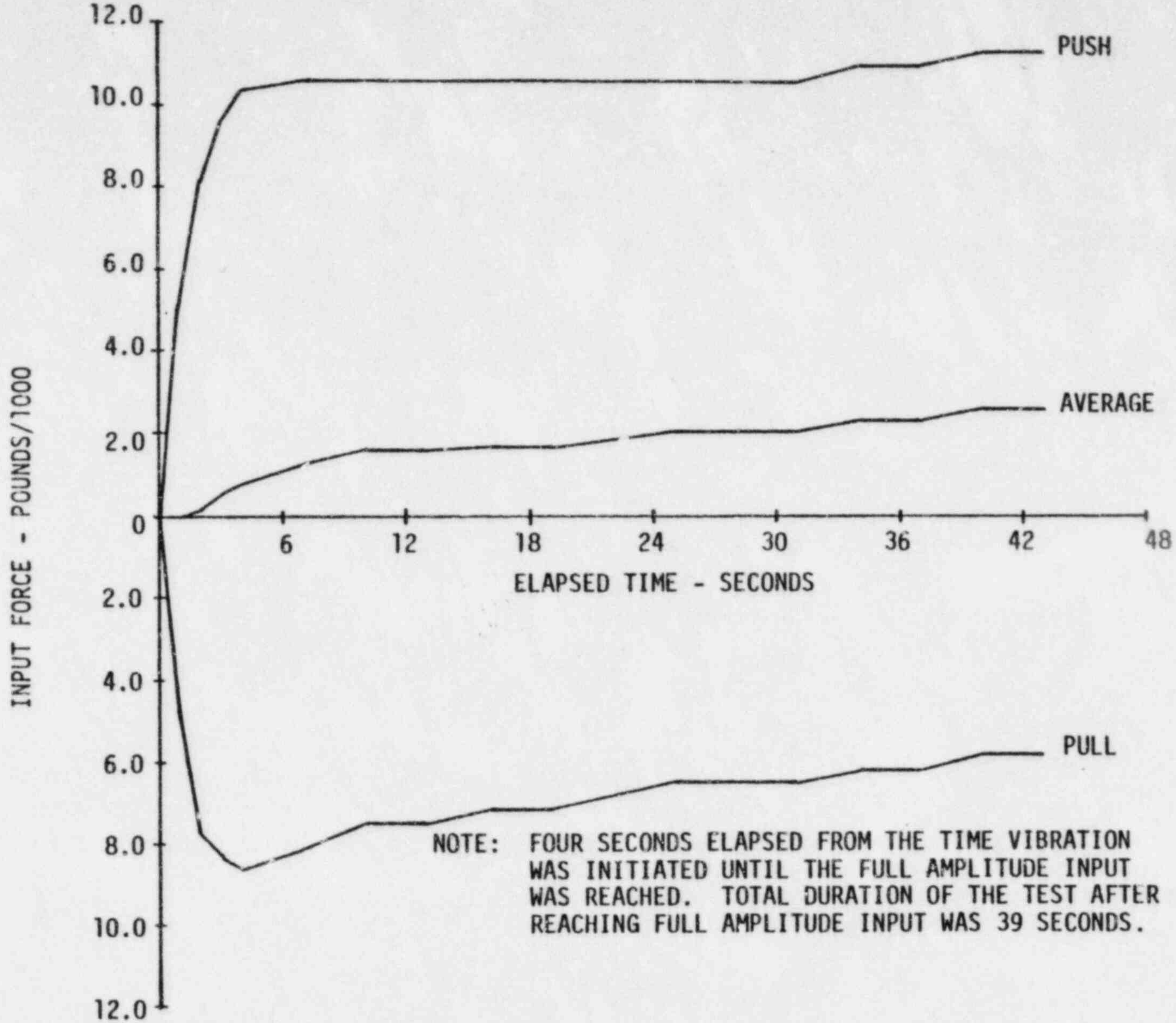


SEISMIC TEST

STRAIN GAUGE LOCATIONS

STRAIN GAUGE NUMBER	ACTIVE GAUGE LENGTH
S1-S6	0.25"

SKETCH 1


INPUT FORCE - POUNDS/1000
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10.0
8.0
6.0
4.0
2.0
0
2.0
4.0
6.0
8.0
10.0
12.0
6
12
18
24
30
36
42
48
ELAPSED TIME - SECONDS
PUSH
AVERAGE
PULL
NOTE: FOUR SECONDS ELAPSED FROM THE TIME VIBRATION WAS INITIATED UNTIL THE FULL AMPLITUDE INPUT WAS REACHED. TOTAL DURATION OF THE TEST AFTER REACHING FULL AMPLITUDE INPUT WAS 39 SECONDS.

FIGURE 1

10 INCH STAINLESS STEEL

SINE DWELL AT 9 HERTZ

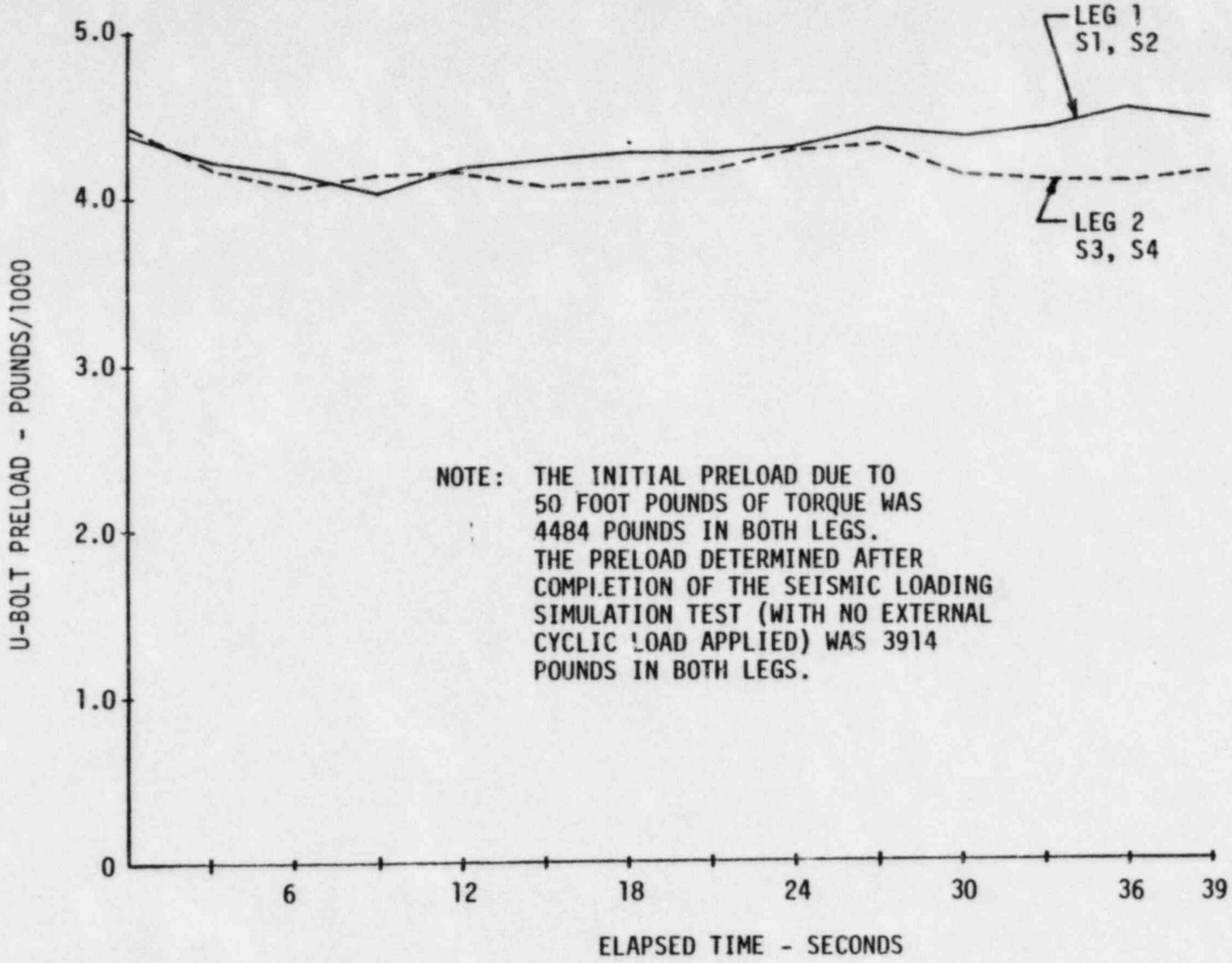


FIGURE 2

ATTACHMENT 2

EBASCO SERVICES INCORPORATED

BY A.T. DATE 4.12.84 SHEET ____ OF ____

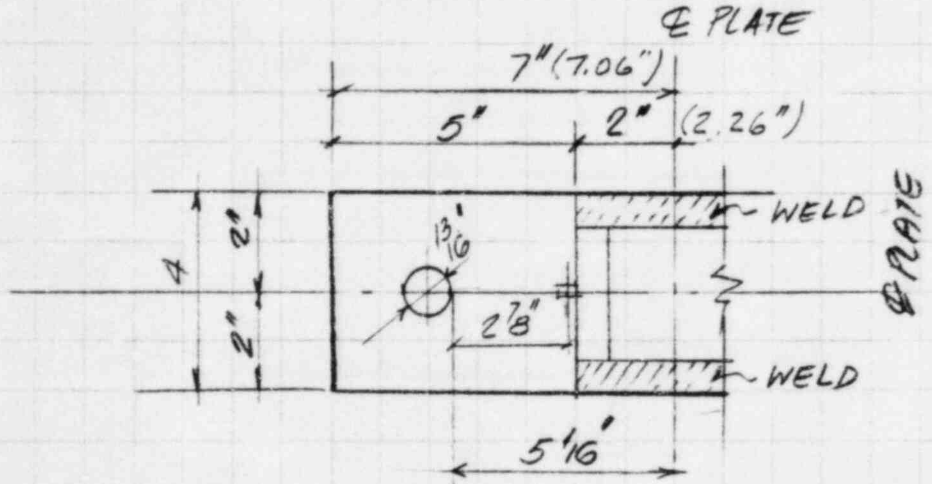
CHKD. BY [signature] DATE 4/12/84 OFS NO. ________ DEPT. NO. ____

CLIENT ________

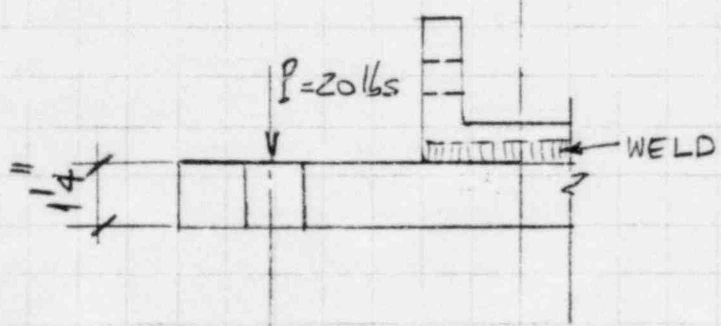
PROJECT ________

SUBJECT FINITE ELEMENT MODEL OF 10-INCH PIPE CROSS PIECE

DIMENSIONS OF CROSS PIECE ARE APPROXIMATE - EXACT DIMENSIONS, MEASURED AFTER ANALYSIS ARE SHOWN IN PARENTHESIS WHERE DIFFERENT.

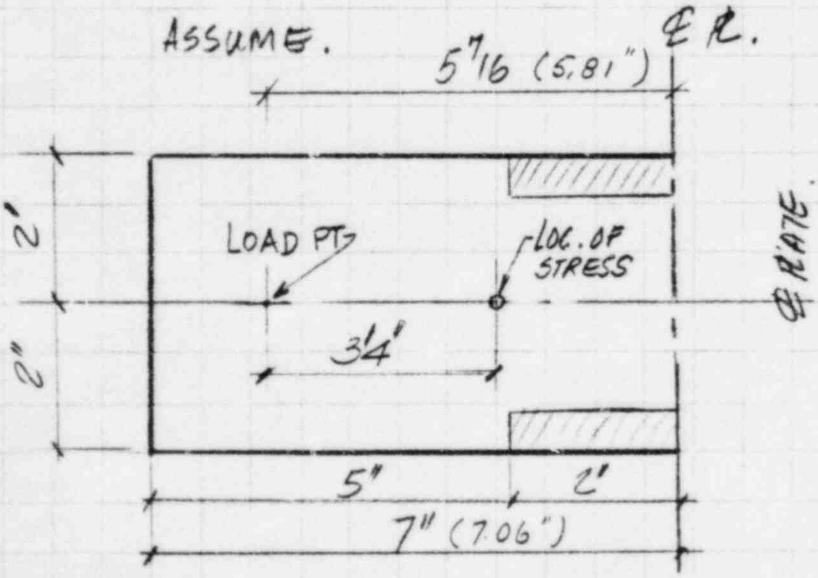


PLAN



SECTION.

ASSUME.



EBASCO SERVICES INCORPORATED

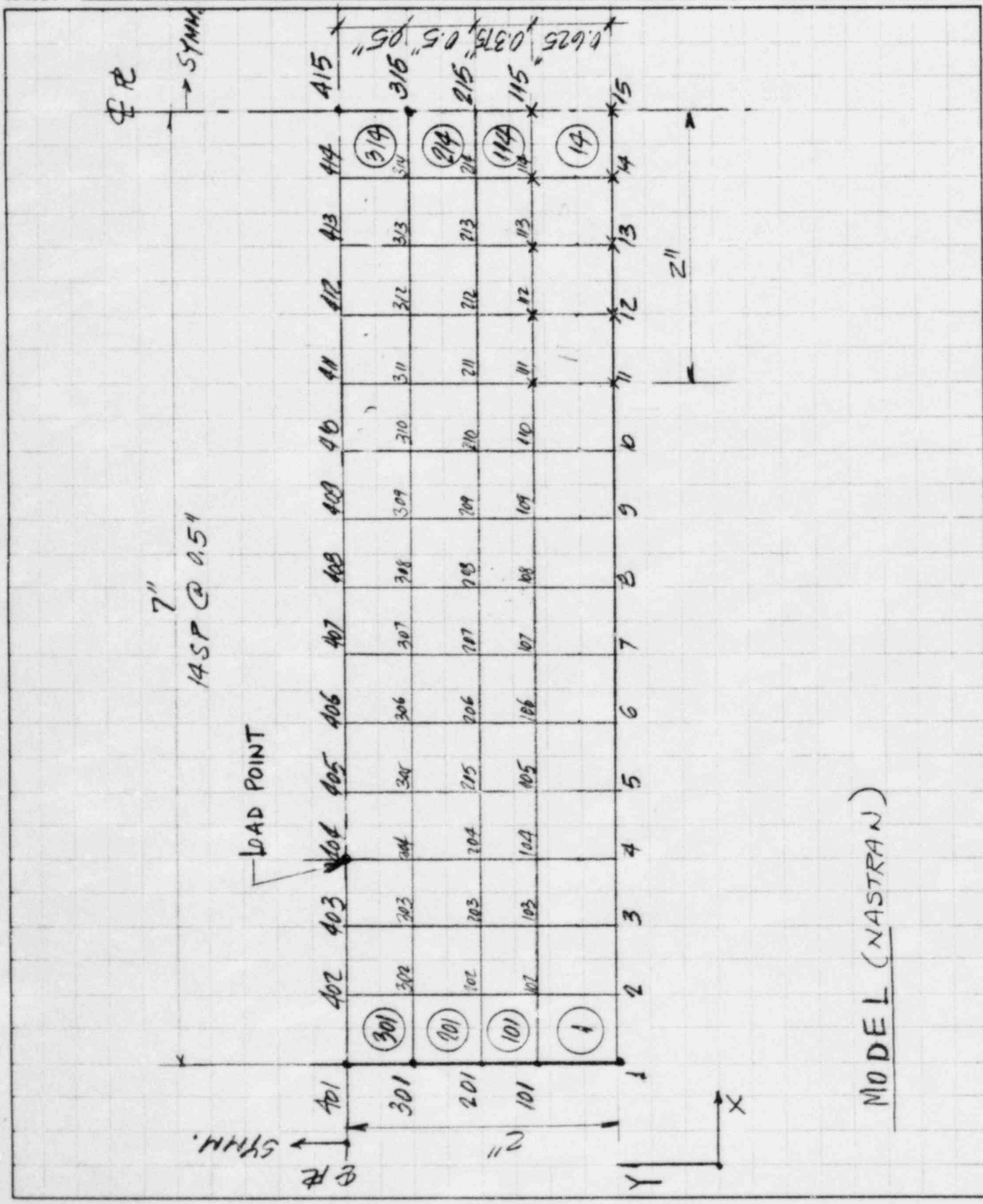
BY A.T DATE 4.12.84 SHEET ____ OF ____

CHKD. BY [signature] DATE 4/12/84 OFS NO. ____ DEPT. NO. ____

CLIENT ____

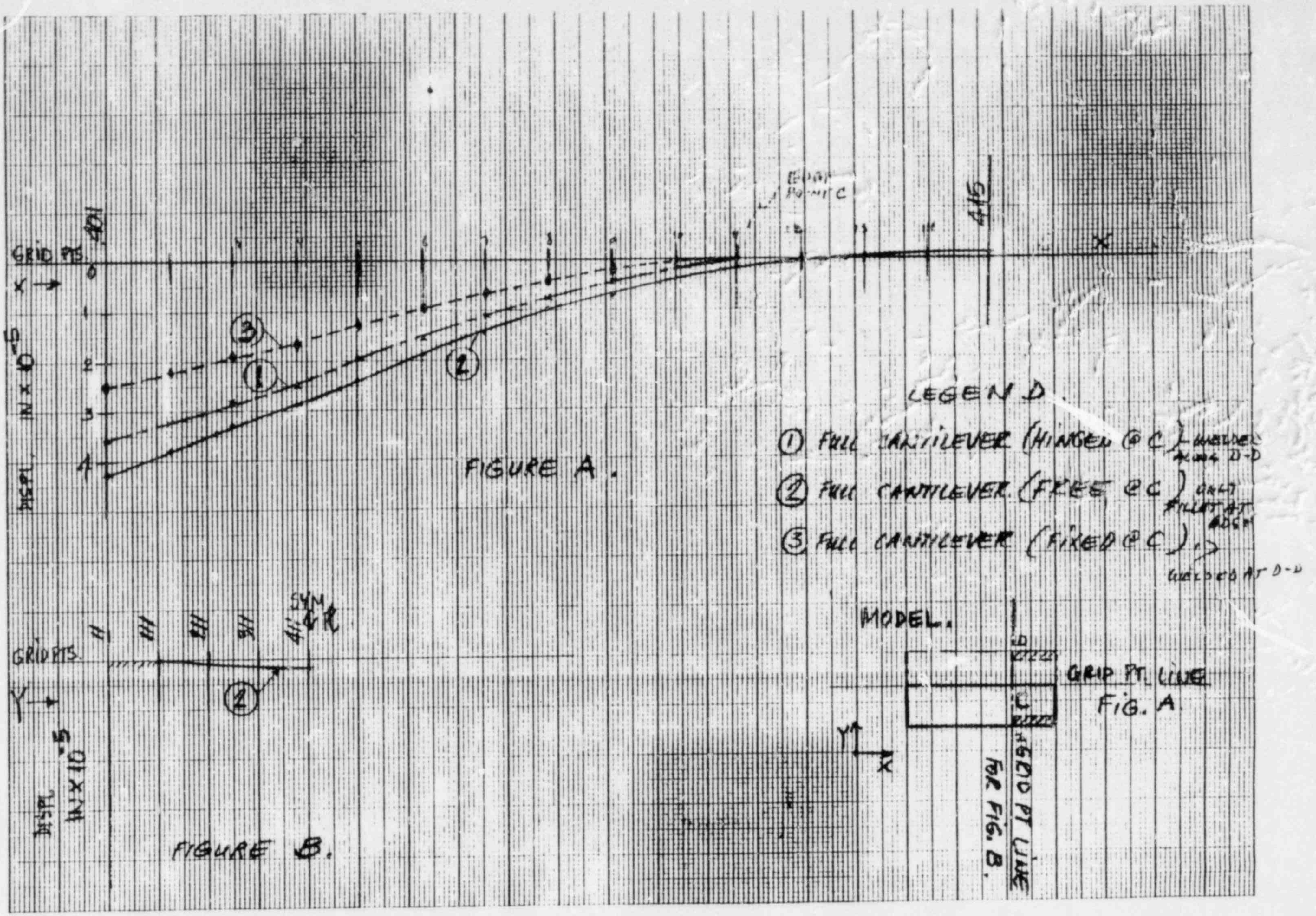
PROJECT ____

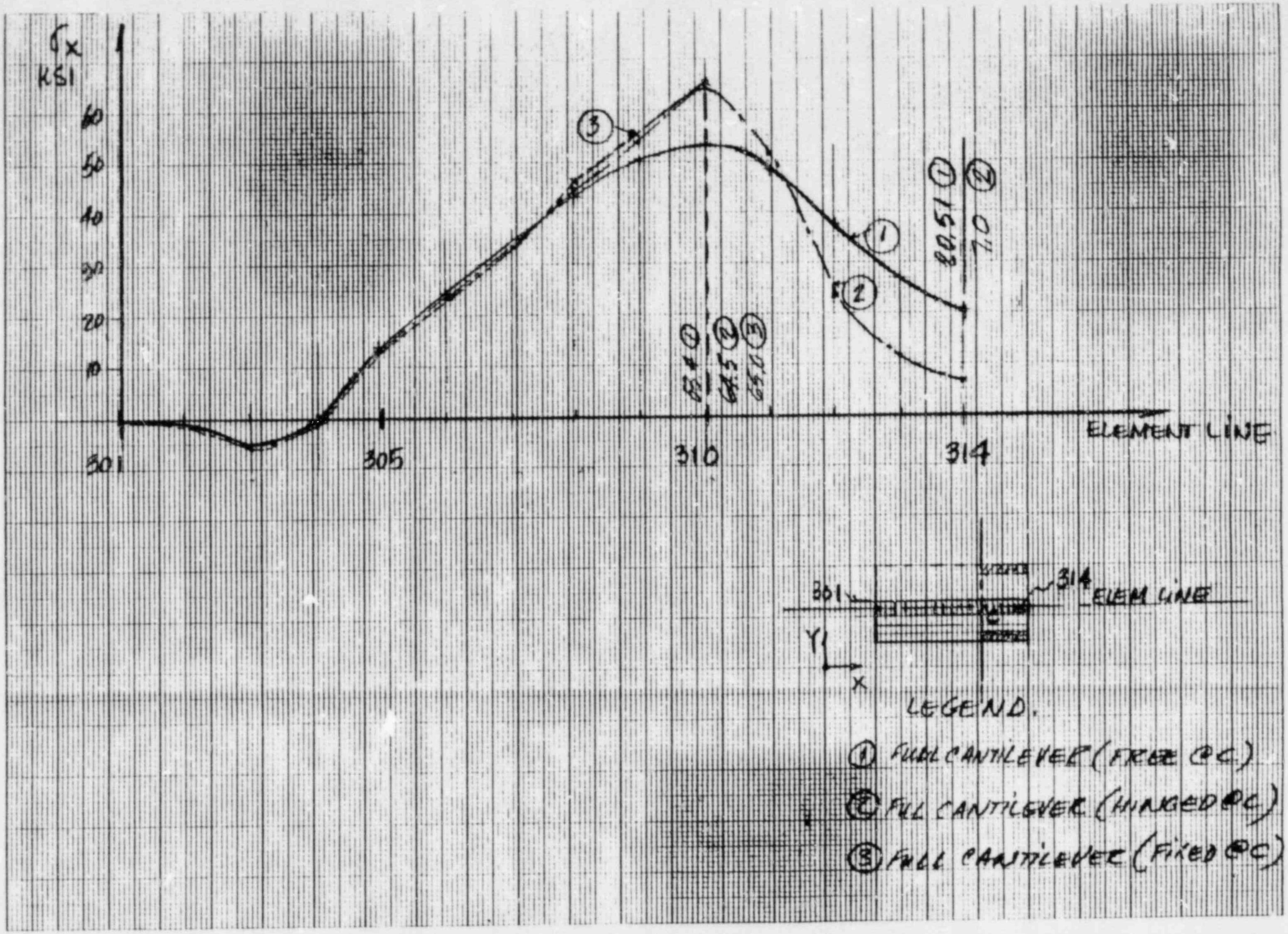
SUBJECT ____



K+E 10 X 10 TO THE CENTIMETER
KEUFFEL & ESSER CO.

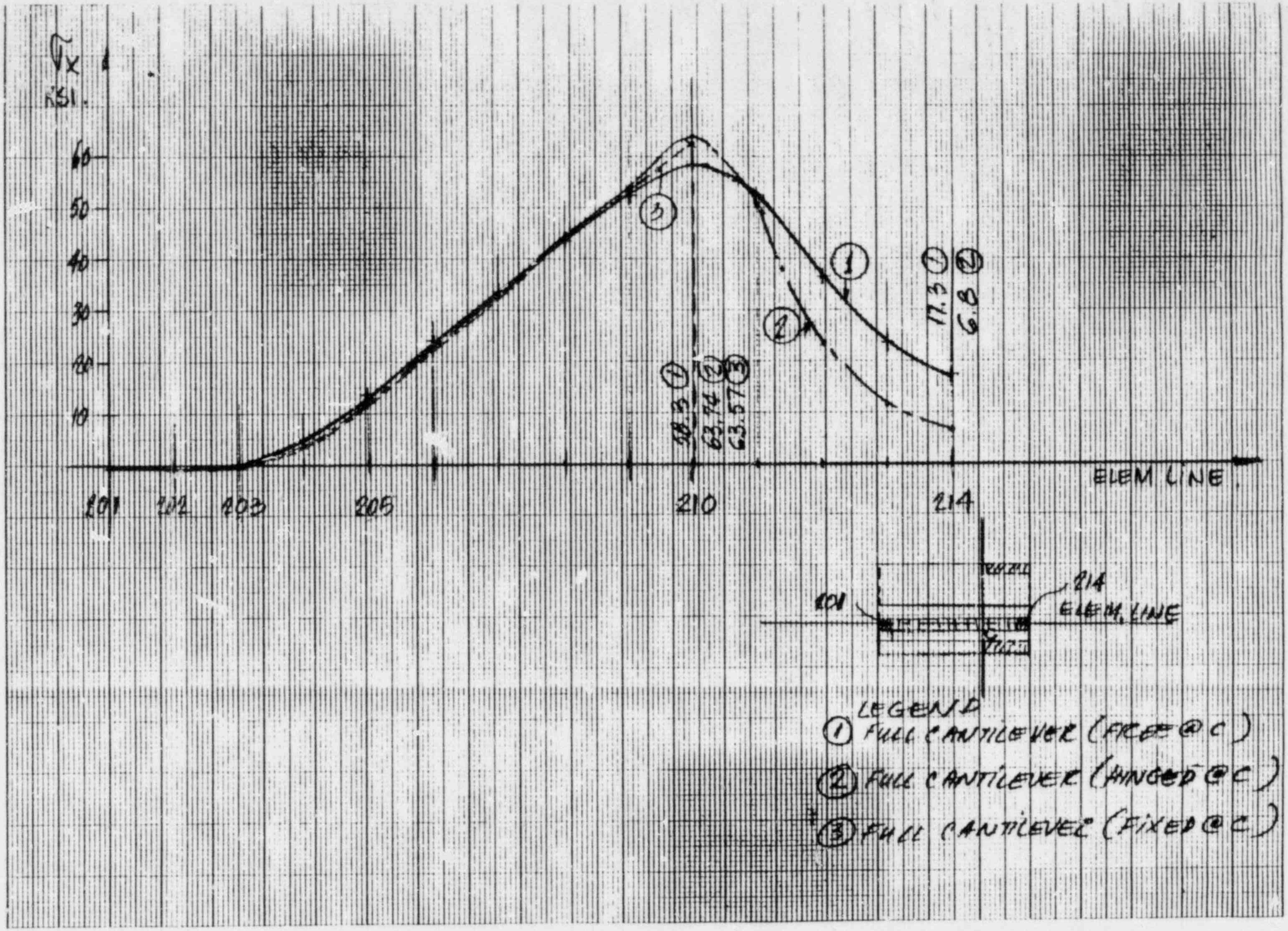
46 1513





σx KSI
60
50
40
30
20
10
301
305
310
314
ELEMENT LINE
3
1
2
80.51 (1)
7.0 (2)
65.0 (3)
64.5 (2)
301
314
ELEM LINE
Y
X
LEGEND.
(1) FULL CANTILEVER (FREE @C)
(2) FULL CANTILEVER (HINGED @C)
(3) FULL CANTILEVER (FIXED @C)

46 1513



KSI
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50
40
30
20
10
201
202
203
205
210
214
ELEM LINE
58.3 ①
63.74 ②
63.57 ③
17.3 ①
6.8 ②
201
214
ELEM LINE
LEGEND
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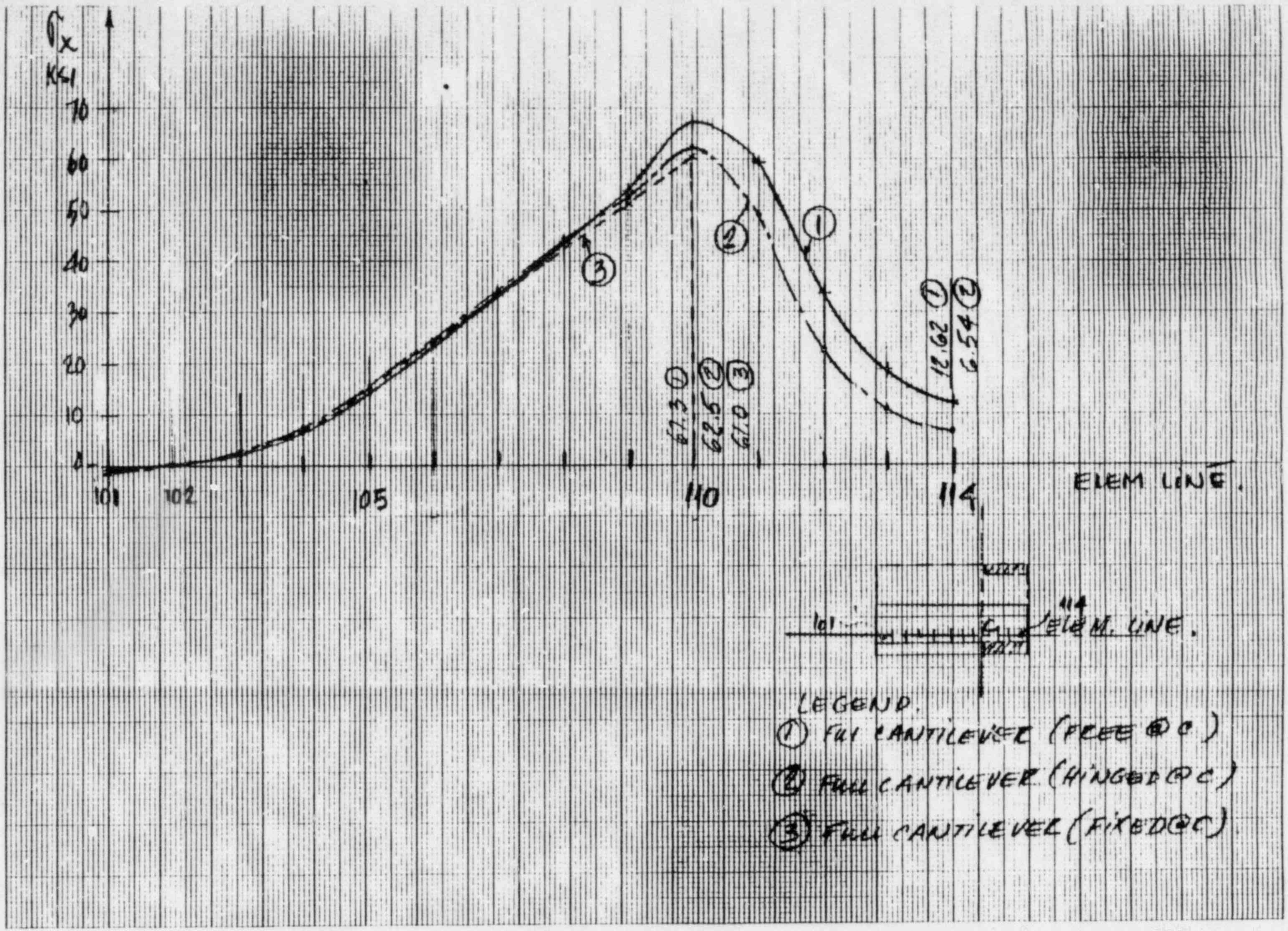
② FULL CANTILEVER (HINGED @ C)
③ FULL CANTILEVER (FIXED @ C)



σx
KSI
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50
40
30
20
10
0
101
102
105
110
114
ELEM LINE.
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62.5 ②
61.0 ③
12.62 ①
6.54 ②
①
②
③
101
114
ELEM. LINE.
LEGEND.
① FULL CANTILEVER (FREE @ C)
② FULL CANTILEVER (HINGED @ C)
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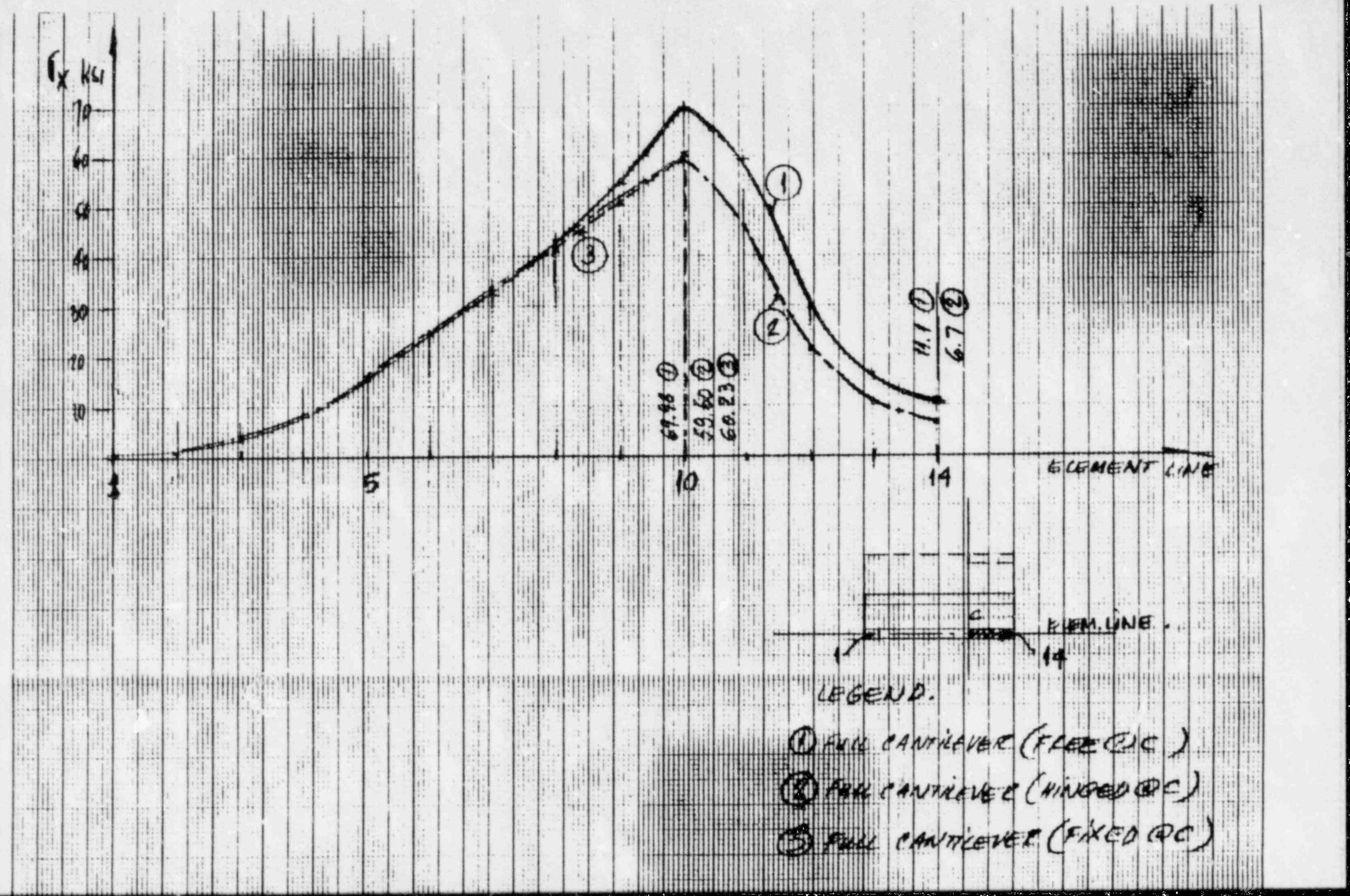
σx ksi
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60
10
5
10
14
ELEMENT LINE
69.98 ①
59.60 ②
60.23 ③
11.1 ①
6.7 ②
ELEM. LINE
1
14
C
LEGEND.
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② FULL CANTILEVER (HINGED @ C)
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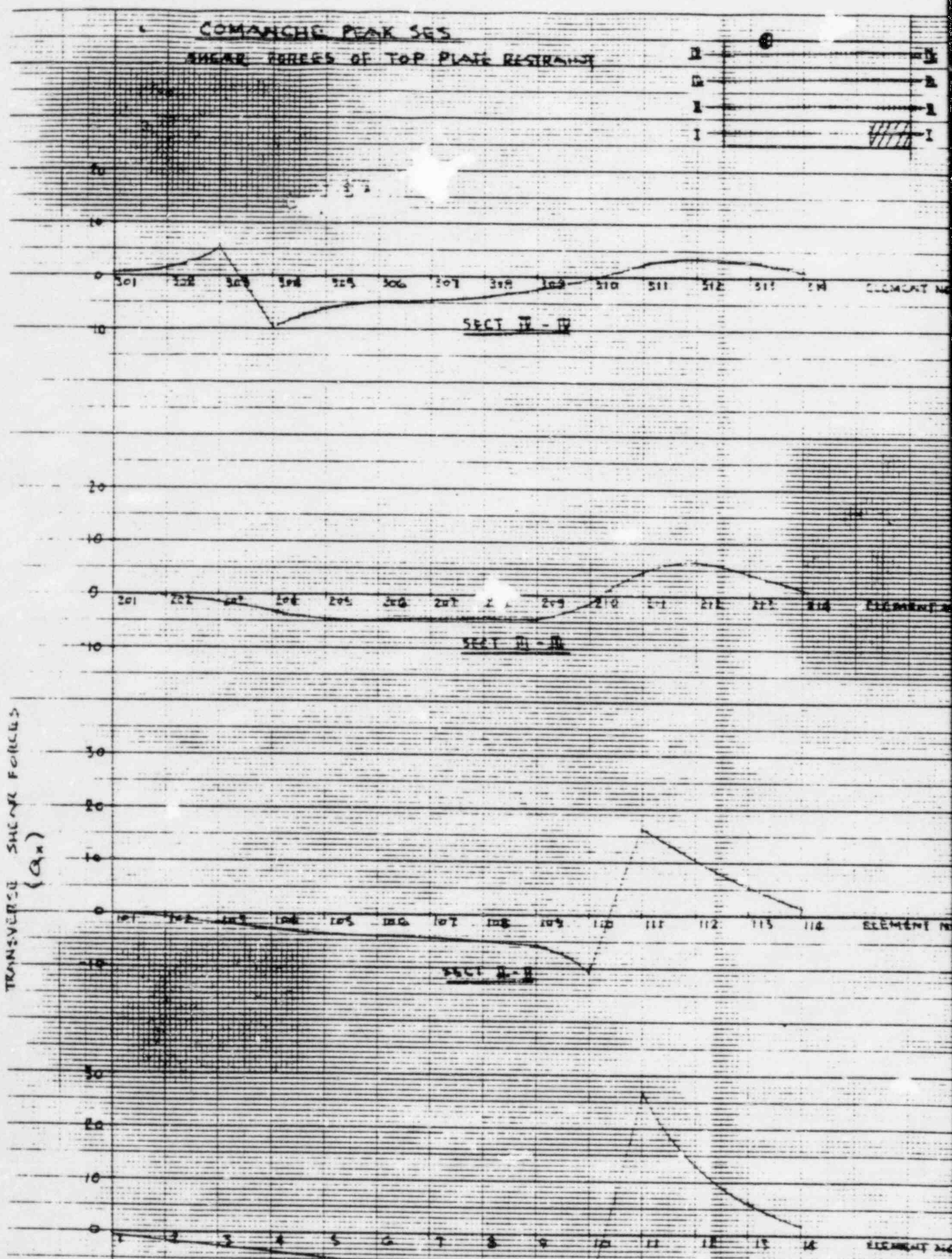


COMANCHE PEAK SES
SHEAR FORCES OF TOP PLATE RESTRAINT
SECT IV - IV
SECT III - III
SECT II - II
SECT I - I
ELEMENT NO
TRANSVERSE SHEAR FORCES (Qx)

ATTACHMENT 3

COMANCHE PEAK STEAM ELECTRIC STATION

U-BOLT FINITE ELEMENT ANALYSIS

WESTINGHOUSE ELECTRIC CORPORATION
P. O. BOX 355
PITTSBURGH, PA 15230

June 12, 1984

AUTHORS

M. J. FISHER

W. S. LAPAY

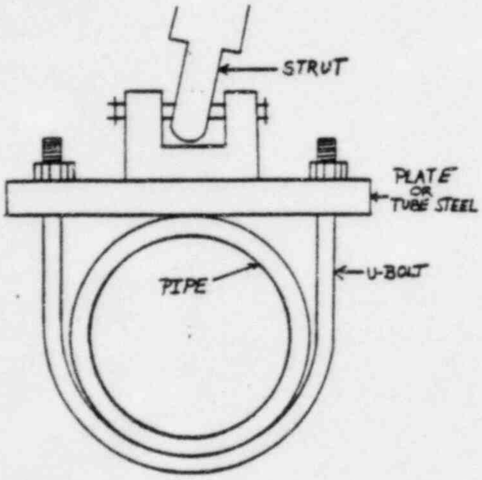
COMANCHE PEAK U-BOLT ANALYSIS REPORT

TABLE OF CONTENTS

COMANCHE PEAK U-BOLT ANALYSIS REPORT

I. INTRODUCTION

During the Atomic Safety and Licensing Board hearing for Comanche Peak Unit No. 1, concerns were introduced regarding the pipe support assembly shown below.



Specifically, the concerns are that sufficient testing and analysis has not been conducted to ensure that:

1. Adequate frictional forces exist at the pipe/pipe support interface to balance a moment created when the U-bolt legs are not parallel to the strut so that the U-bolt strut assembly support is stable.

2. Pipe and pipe support stresses are within acceptable limits.

Texas Utilities Generating Company (TUGCO) requested that Westinghouse perform a finite element analysis and testing to investigate these concerns.

The tests performed are discussed and results presented in Westinghouse Report Number EQ&T-EQT-860, Revision 0, "Comanche Peak Steam Electric Station U-bolt Support/Pipe Test Report." The finite element analyses are discussed herein.

The intent of the analysis program was (1) to determine if the pipe would slip, thereby creating an unstable support condition when the hanger support was subjected to the preload, thermal, pressure and mechanical loads that would be expected in the Comanche Peak hanger assemblies and assess its significance, and (2) to calculate pipe and pipe support stresses that could be expected to be experienced by the Comanche Peak U-bolt support assemblies. To obtain the analytical information necessary to reach the conclusions related to the scope of the analysis program, finite element models of the U-bolt, cross piece and strut attachment, and pipe were developed. These models are discussed in Section IV. They were verified by comparison to analytical data as well as test data. This is discussed in Sections IV and V. The results from the analysis program related to slip are discussed in Section VI and the pipe stress conditions in Section VII.

II. SUMMARY AND CONCLUSIONS

The finite element analysis performed on the U-bolt pipe support assembly was performed to evaluate: (1) the stability of the U-bolt support system with respect to sliding, and (2) to evaluate the stress intensities in the piping caused by the cinching of the U-bolt. In order to perform the above evaluations, finite element analysis models were developed to analyze the U-bolt piping assembly for four loading conditions:

1. Preload
2. Preload and thermal
3. Preload and thermal and pressure
4. Preload and thermal and pressure and strut applied load (Push)

From these loading conditions, it is possible to determine the forces in the legs of the U-bolt, the stresses in the pipe, and the minimum torque required to maintain stability against sliding when the U-bolt assembly is subjected to all of the loadings with the push force having a 5° permissible offset from the vertical axis.

Four U-bolt support piping assemblies were studied consistent with the configurations tested. They are (1) 4" Sch 160, (2) 10" Sch 40S, (3) 10" Sch 80, and (4) 32" Main Steam piping systems. The finite element models developed were shown to provide results that are in agreement with the test results. The analytical results showed similar magnitudes and stress trends in the vicinity of the location of strain gauges in the

test. There were differences between the tests and analytical results; however, this is due to the difference in the fitup between the pipe and U-bolt in the test and in the finite element model. The difference in fitup is attributed to the pipe out-of-roundness and tolerance in the U-bolt bend. The finite element analysis model was not intended to model details such as gap nonlinearities but rather to evaluate as-designed conditions.

The U-bolt leg force summaries are given in Tables II-1, through II-4. As seen from these tables, as one would intuitively expect, 1) the U-bolt preload torque is indicative of the U-bolt leg tension, 2) the thermal expansion of the pipe increases the U-bolt leg tension, 3) the pipe expansion due to internal pressure increases U-bolt leg tension, and 4) the applied hanger "push" load decreases the U-bolt leg tension.

Recommended torque values are given below based on consideration of the maximum and minimum preload values evaluated in the finite element analysis. The stress in the U-bolt, based on the threaded area, will be under yield at these preload values.

	Recommended torque value
4" Sch 160	25ft-lb
10" Sch 40S	50ft-lb
10" Sch 80	50ft-lb
32" MS	250ft-lb

Further, it is necessary to torque to these values to assure proper U-bolt fitup and therefore stability of the U-bolt assembly. Hanger stability at these preload torques was demonstrated by the analyses, reported herein. As a guide to assuring that the U-bolts will function properly, they should be installed with torque values equal to the recommended values so that a "snug" uniform fit is maintained between the pipe and U-bolt.

The upper bound torque value is in the ultimate, (1) the maximum torque to which the piping can still be shown within acceptable stress limits, and (2) a value that will not yield the crosspiece, U-bolt, or pipe locally to such an extent that preload is lost and the hanger loses stability. This upper torque range value was not determined herein.

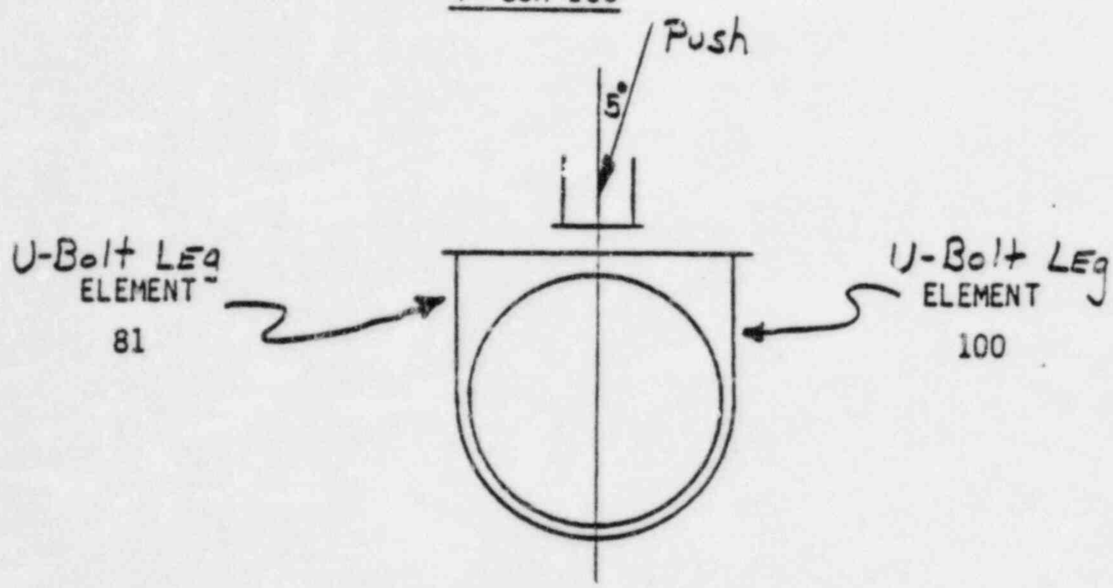
The stress state in the pipe with maximum preloads, thermal, pressure and push, as defined in Tables II-1, to II-4, was found to be acceptable.

It is concluded that the U-bolts can be cinched to prevent sliding, thereby assuring the transfer of loads (force, moments) to the pipe and maintaining the stability of the U-bolt support system. The pipe will not have stresses that exceed acceptable limits if preload torquing is restricted to the recommended values.

U-BOLT LEG FORCE SUMMARY

TABLE II-1

4" SCH 160

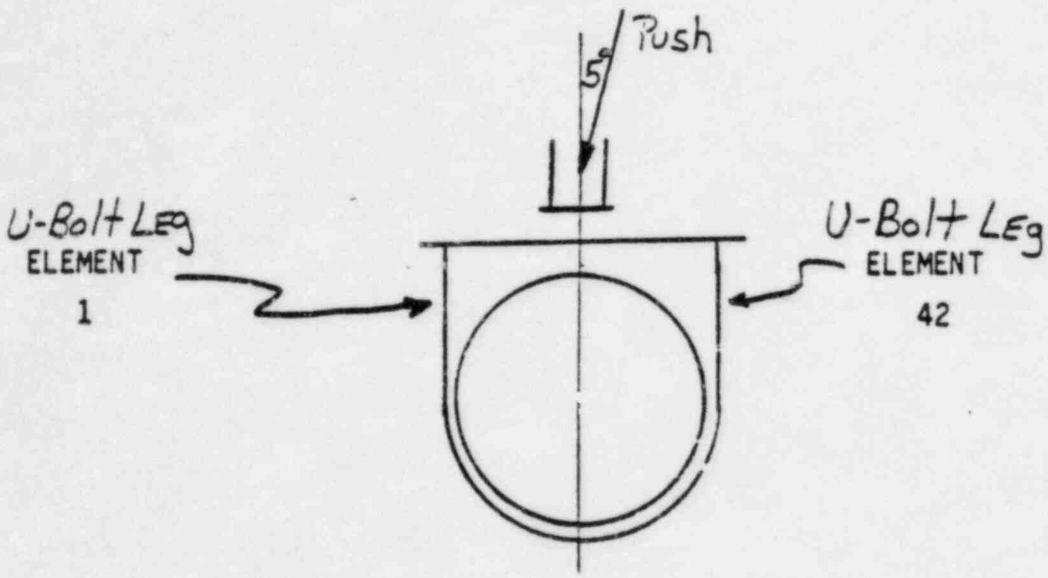


Loading Condition	Tension for Element 81	Tension for Element 100
Preload (60 ft-lbs.)	5.41 kips	5.41 kips
Preload + Thermal	8.31 kips	8.31 kips
Preload+Thermal+Pressure	8.58 kips	8.58 kips
Preload + Thermal Pressure + Push at 5° (2,000 #)	8.21 kips	8.33 kips
Minimum Preload (9 ft-lb.)	.80 kips	.80 kips
Minimum Preload+Thermal+ Pressure+Push at 5°	2.80 kips	2.89 kips

U-BOLT LEG FORCE SUMMARY

TABLE II -2

10" SCH 40S

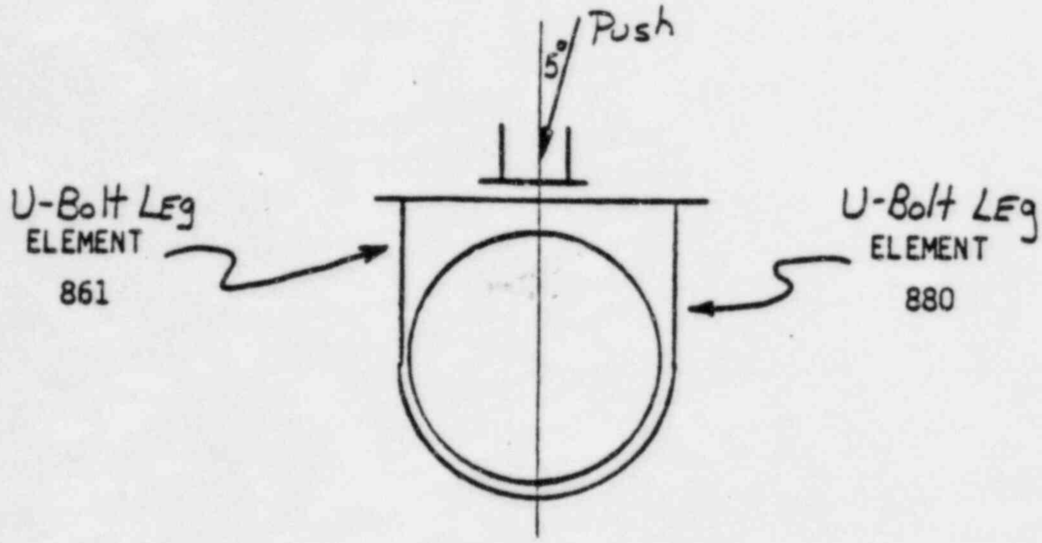


Loading Condition	Tension for Element 1	Tension for Element 42
Preload (100 ft-lbs.)	5.62 kips	5.64 kips
Preload + Thermal	6.91 kips	6.93 kips
Preload+Thermal+Pressure	7.22 kips	7.23 kips
Preload + Thermal Pressure + Push at 5° (10,000 #)	4.43 kips	4.78 kips
Minimum Preload (46 ft-lb.)	2.56 kips	2.57 kips
Minimum Preload+Thermal+ Pressure+Push at 5°	1.44 kips	1.78 kips

U-BOLT LEG FORCE SUMMARY

TABLE II-3

10" SCH 80

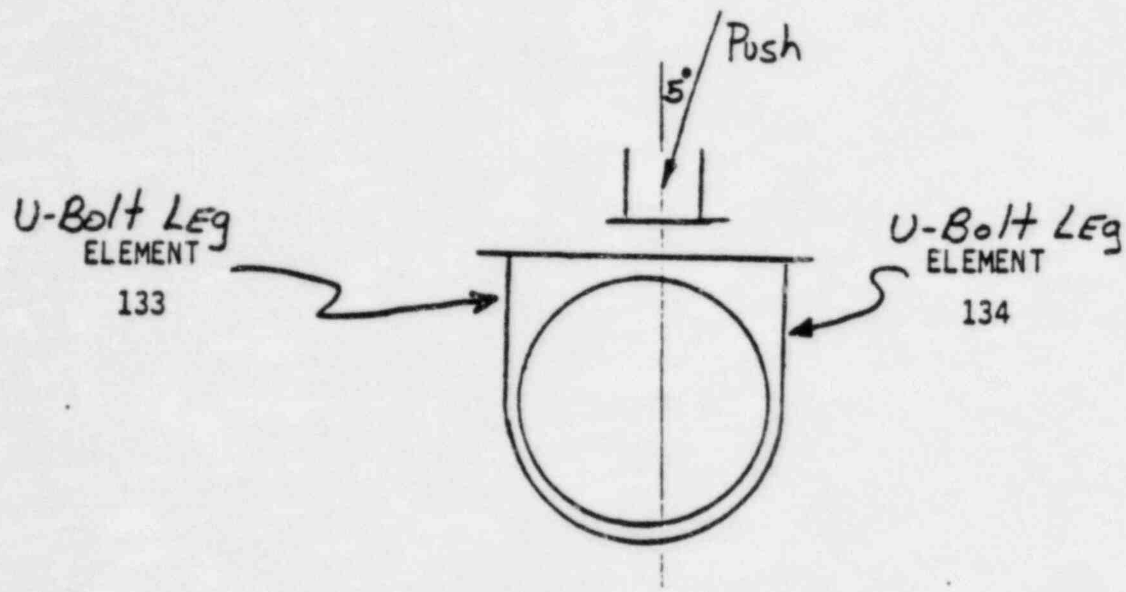


Loading Condition	Tension for Element 861	Tension for Element 880
Preload (100 ft-lb.)	7.51 kips	7.54 kips
Preload + Thermal	8.91 kips	8.91 kips
Preload+Thermal+Pressure	9.19 kips	9.19 kips
Preload + Thermal Pressure + Push at 5° (10,000 #)	7.68 kips	8.11 kips
Minimum Preload (11 ft-lb.)	.83 kips	.83 kips
Minimum Preload+Thermal+ Pressure+Push at 5°	.09 kips	.57 kips

U-BOLT LEG FORCE SUMMARY

TABLE II-4

32" Main Steam



Loading Condition	Tension for Element 133	Tension for Element 134
Preload (240 ft-lb.)*	6.04 kips	6.05 kips
Preload + Thermal	27.44 kips	27.45 kips
Preload+Thermal+Pressure	37.13 kips	37.15 kips
Preload + Thermal + Pressure + Push at 0°	12.16 kips	12.21 kips
Preload + Thermal + Pressure + Push at 5° (100,000 #)	8.13 kips	16.23 kips

* Because U-bolt leg force and pipe stresses are small for preload only, no attempt was made through analysis to exactly duplicate the 240ft-lb test preload. The analysis results are closer to a 380ft-lb preload. See stress results, test gage K, p. 39. However, the conclusion reached for the 240ft-lb (as discussed later) remains valid since the U-bolt leg forces for the various load combinations would only be about 2.2 kips lower at the 240ft-lb preload than the 380ft-lb preload. Thus, there would still be sufficient margin to maintain preload in each U-bolt leg under any load combination.

III. INPUT TO ANALYSIS

Westinghouse was requested by Texas Utilities to perform a finite element analysis of each of the four hanger assemblies shipped to the WAESD Engineering Laboratory (located at the Westinghouse Advanced Energy Systems Division, Large, Pennsylvania) for testing. As input for the development and verification of the finite element models Westinghouse used the following information:

1. Hanger Drawings MS-1-001-005-S72R, Rev. 4 (Appendix I)
   RH-1-024-007-S22R, Rev. 3 (Appendix I)
   RC-1-018-016-C81R, Rev. 7 (Appendix I)

These drawings were supplied by Texas Utilities. Dimensions from these drawings were used in developing the models. They were verified and supplemented as necessary by measurements of the principal dimensions on the test specimens used in the testing program. This was done so that the model used in the analysis corresponded to the actual support assembly used in the test. No verification was performed for secondary effects on dimensions (out of roundness, bowing, etc.) since the finite element model represents the ideal situation. See Figure III-1 through III-4 for dimensions used in the analyses.

2. Additional information about the piping is tabulated below. Pipe size, schedule, material, normal operating temperature and pressure were provided by TUGCO.

Pipe Size	Schedule	Material	Temp. °F	Pressure psi
4"	160	A312 Grade 304	559	2485
10"	40S	A312 Grade 304	210	600
10"	80	SA 106 Grade B	210	600
32" (O.D.)	T = 1.45"	SA 155-KCF70	557	1285

4 INCH

PIPE/U-BOLT ASSEMBLY

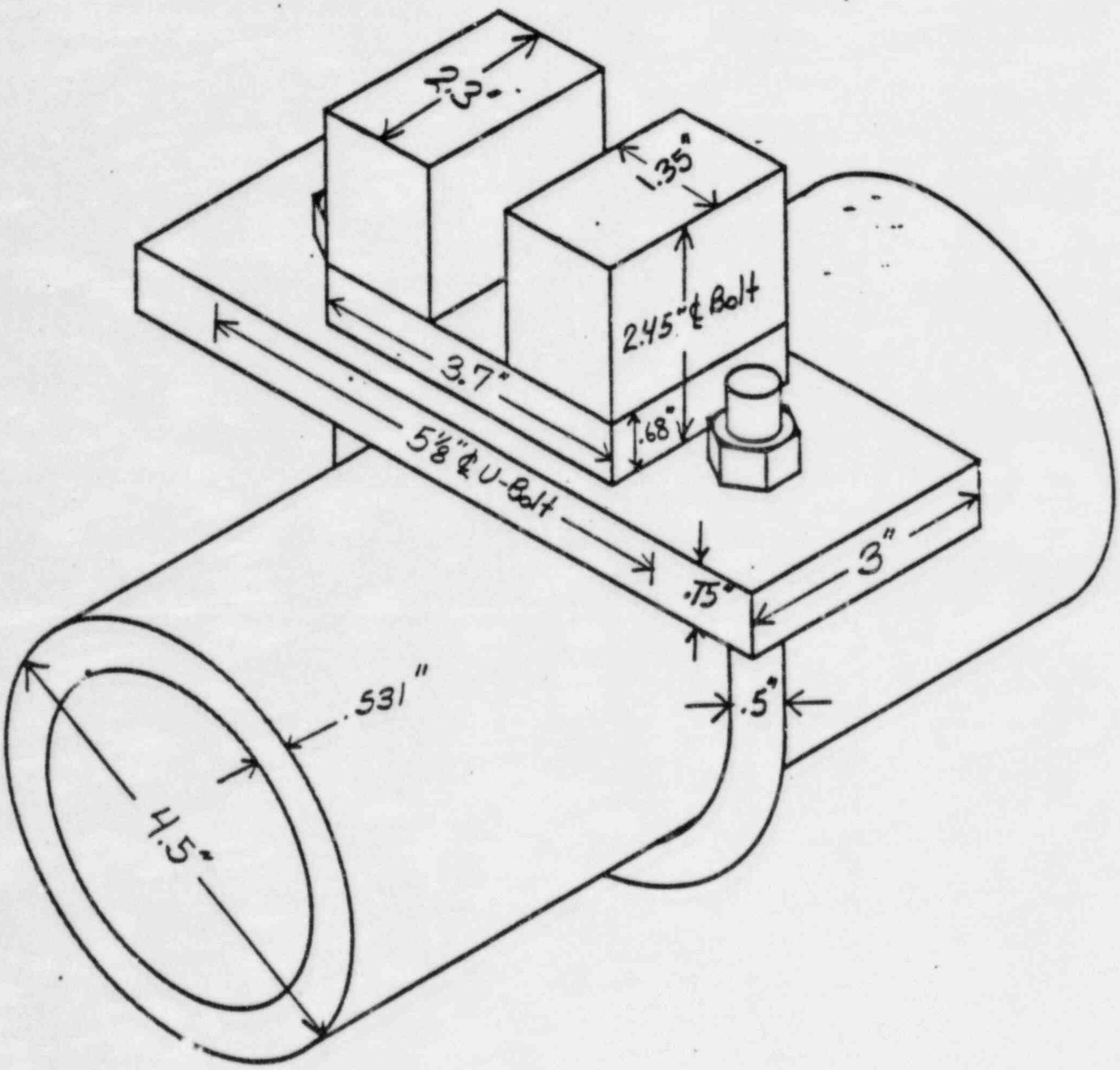


FIGURE III-1

10 INCH STAINLESS

PIPE/U-BOLT ASSEMBLY

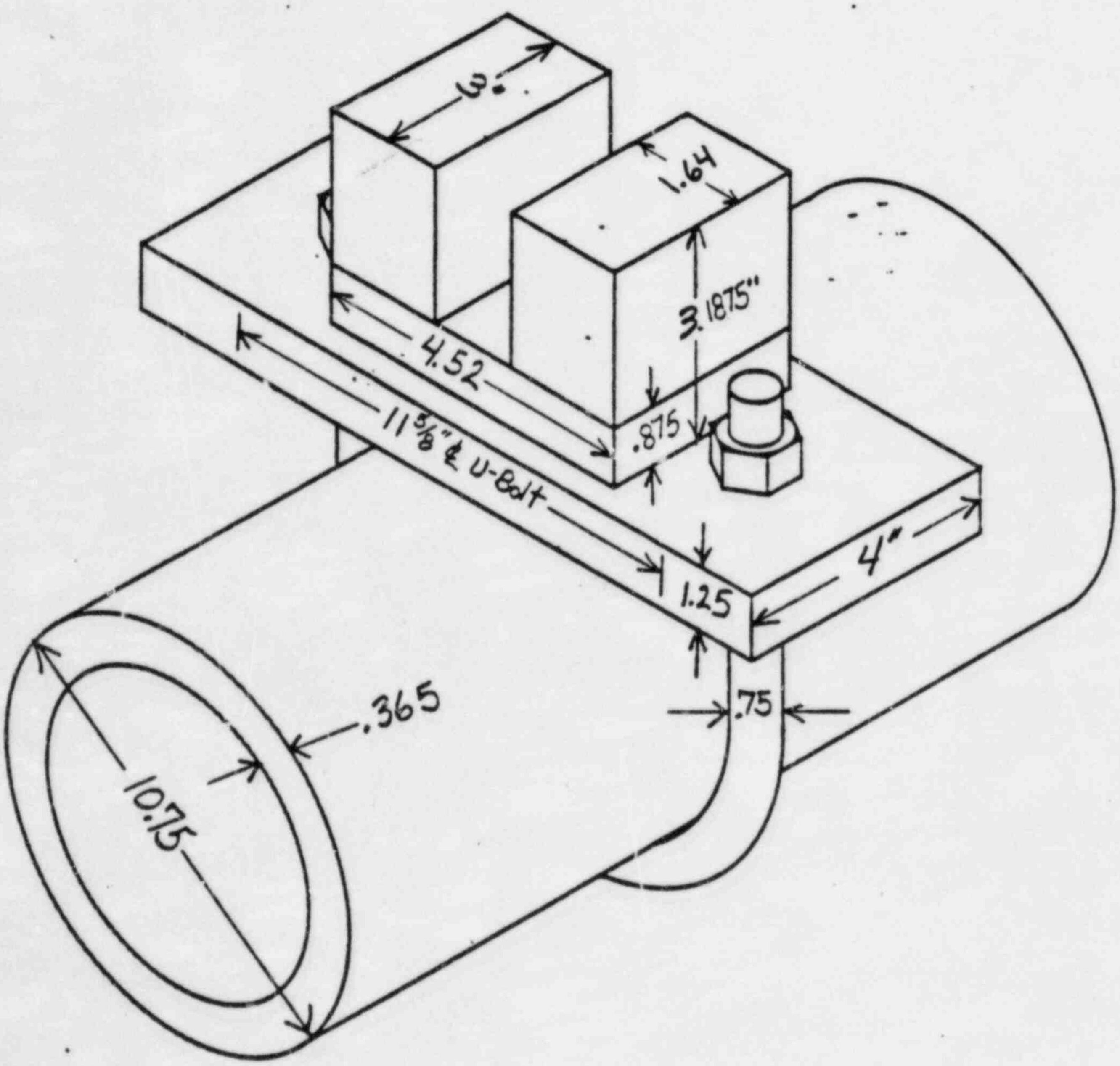


FIGURE III-2

10 INCH CARBON

PIPE/U-BOLT ASSEMBLY

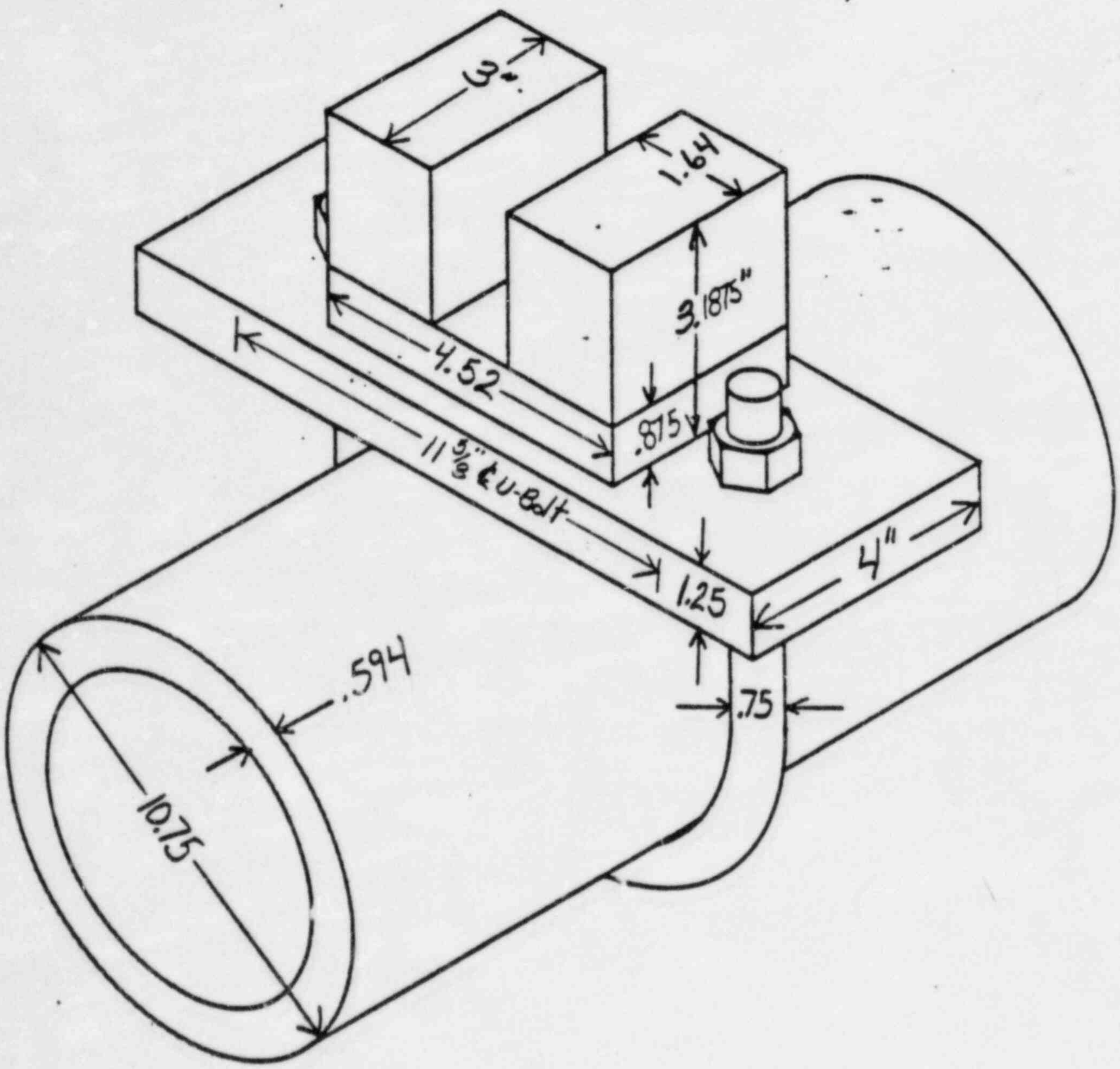


FIGURE III-3

32" MAIN STEAM

PIPE/U-BOLT ASSEMBLY

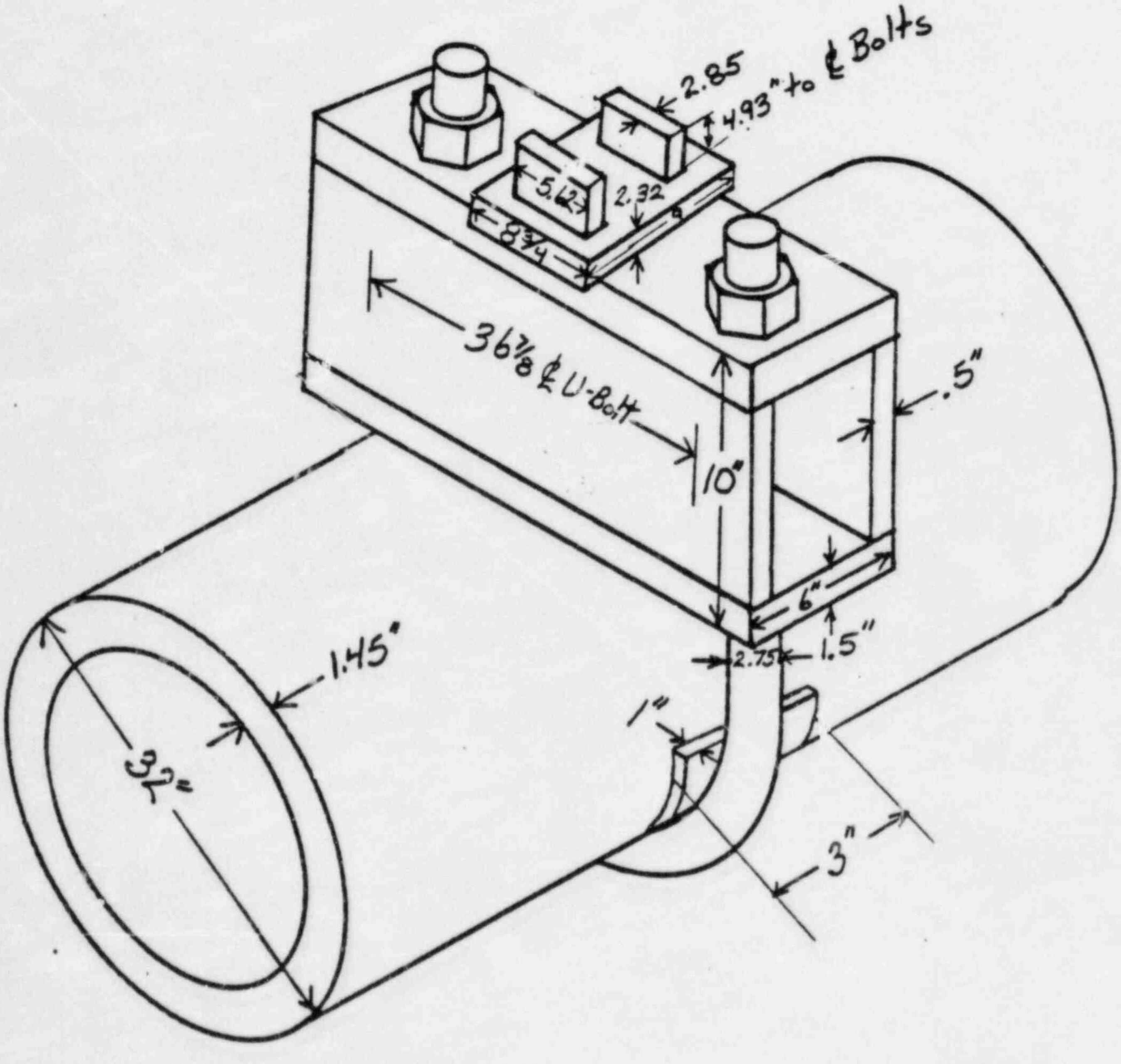
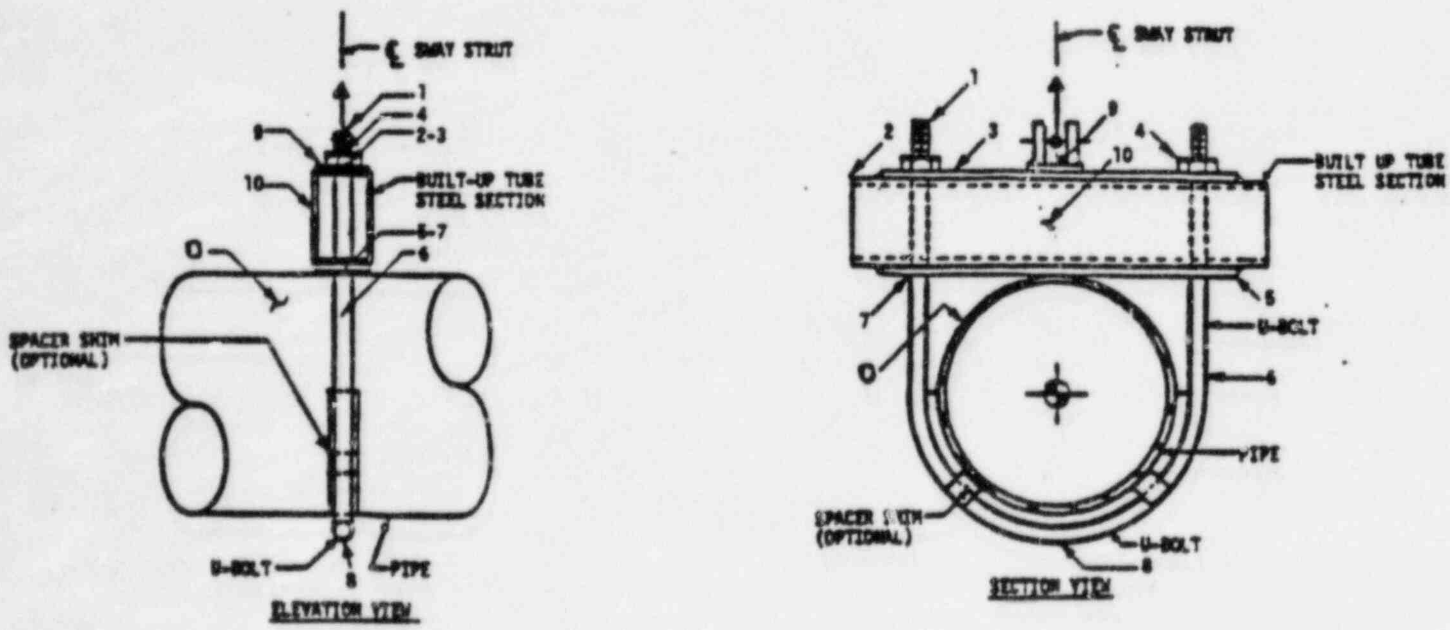


FIGURE III-4

Pipe support materials were obtained from the pipe support drawings in item 1 above.

3. Information from the preload test, load distribution test and thermal cycling test given in the "Comanche Peak Steam Electric Station U-bolt Support/Pipe Test Report," Report Number EQ&T-EQT-860, Revision 0, were compared to the analytical results to verify the analytical models.

4. Steady state pipe support temperature distributions were obtained from tests conducted at WAESD engineering laboratory. The thermocouple temperature distributions for the 4" and 32" pipe supports are listed below. The thermocouple temperature distributions were used in the analysis to obtain pipe and pipe support stresses resulting from the restriction of the thermal expansion of the pipe by the U-bolt assembly.



The above diagrams are for the 32" pipe/U-bolt assembly thermocouple locations; the 4" pipe/U-bclt assembly has thermocouples at the equivalent 32" assembly locations.

THERMOCOUPLE LOCATION	4" PIPE SUPPORT THERMOCOUPLE READING	32" PIPE SUPPORT THERMOCOUPLE READING
0	563°F	563°F
1	304°F	149°F
2	326°F	150°F
3	342°F	166°F
4	320°F	150°F
5	329°F	174°F
6	442°F	350°F
7	332°F	185°F
8	497°F	440°F
9	343°F	171°F
10	367°F	248°F

Thermal distributions for the 10" pipe supports were conservatively assumed to be at ambient (70°F) since the maximum line temperature is 210°F and the line is uninsulated.

5. TUGCO advised that the maximum permissible misalignment between the strut and U-bolt legs to be used in the analyses was 5 degrees, see Figure III-5.

6. The strut loads used in the finite element analysis are as follows:

4" 2000 lb.

10" Sch. 40 10,000 lb.

10" Sch. 80 10,000 lb.

32" M.S. - 100,000 lb.

These values were determined to be reasonable maximum loads in comparison to loads compiled by ITT Grinnell, and agreed to be Texas Utilities.

7. U-bolt gap condition was defined by Texas Utilities, see Figure III-5.

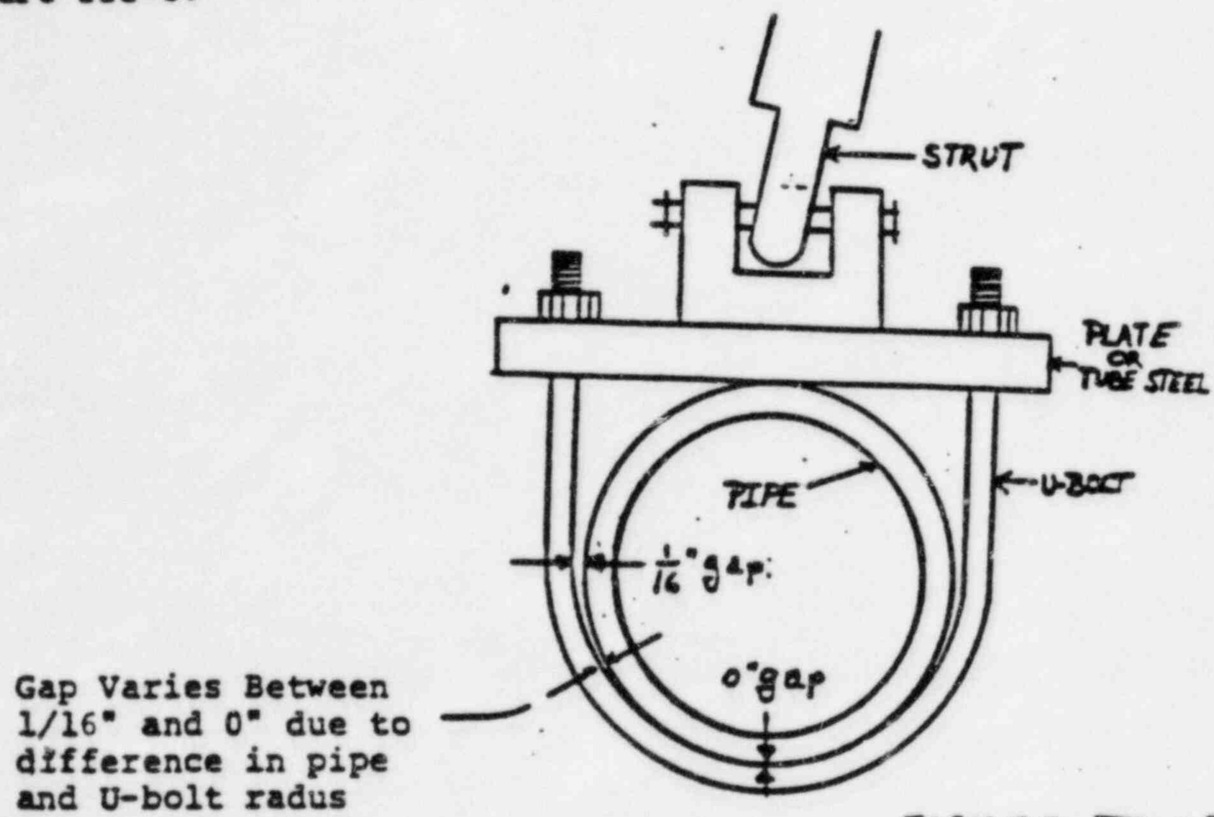


FIGURE III-5

A summary of the pipe/U-bolt finite element analysis loadings is given below:

Load CASE	4" sch 160	10" sch 40S	10" sch 80	32" M.S.
Preload Torque	60 Ft.-lb.	100 ft.-lb.	100 ft.lb.	240 ft.-lb.
Thermal Pipe Temp.	559°F	210°F	210°F	557°F
Hanger Temp.	304-497°F	70°F	70°F	149-440°F
Pressure	2485 psi	600 psi	600 psi	1285 psi
Strut Load	2000 lb.	10,000 lb.	10,000 lb.	10,000 lb.

IV. ANALYTICAL MODEL DESCRIPTION AND ANALYSIS TECHNIQUE

Analysis Model Description

TUGCO requested that Westinghouse perform finite element analyses for four U-bolt assemblies. Each assembly was modeled on MSC NASTRAN, Version 63, a widely used and verified finite element code. NASTRAN was chosen since it is a computer code that is accepted and recognized by industry as having the theoretical capability to provide analytical solutions that reflect the local stress, gap, friction effects, and plastic material behavior that are important in assessing pipe and U-bolt support assembly stress and support stability issues.

The following NASTRAN elements were chosen for the U-bolt analysis model based on their compatability with the desired non-linear solution and solution accuracy, see Figures IV-1 through IV-4 and Appendix II for more detail.

ITEM	NASTRAN ELEMENT	ELEMENT SPACING
Pipe	QUAD4	Circumferentially one QUAD4 element was used to represent each 10° of the pipe wall, i.e., 36 elements were used for the pipe circumference. Longitudinally, near the area of interest, i.e., the U-bolt, the element aspect ratio was ~1 and increased with the distance from the U-bolt.

This Figure is applicable to 4", 10" SCH 40S, and 10" SCH 80 pipe support finite element models.

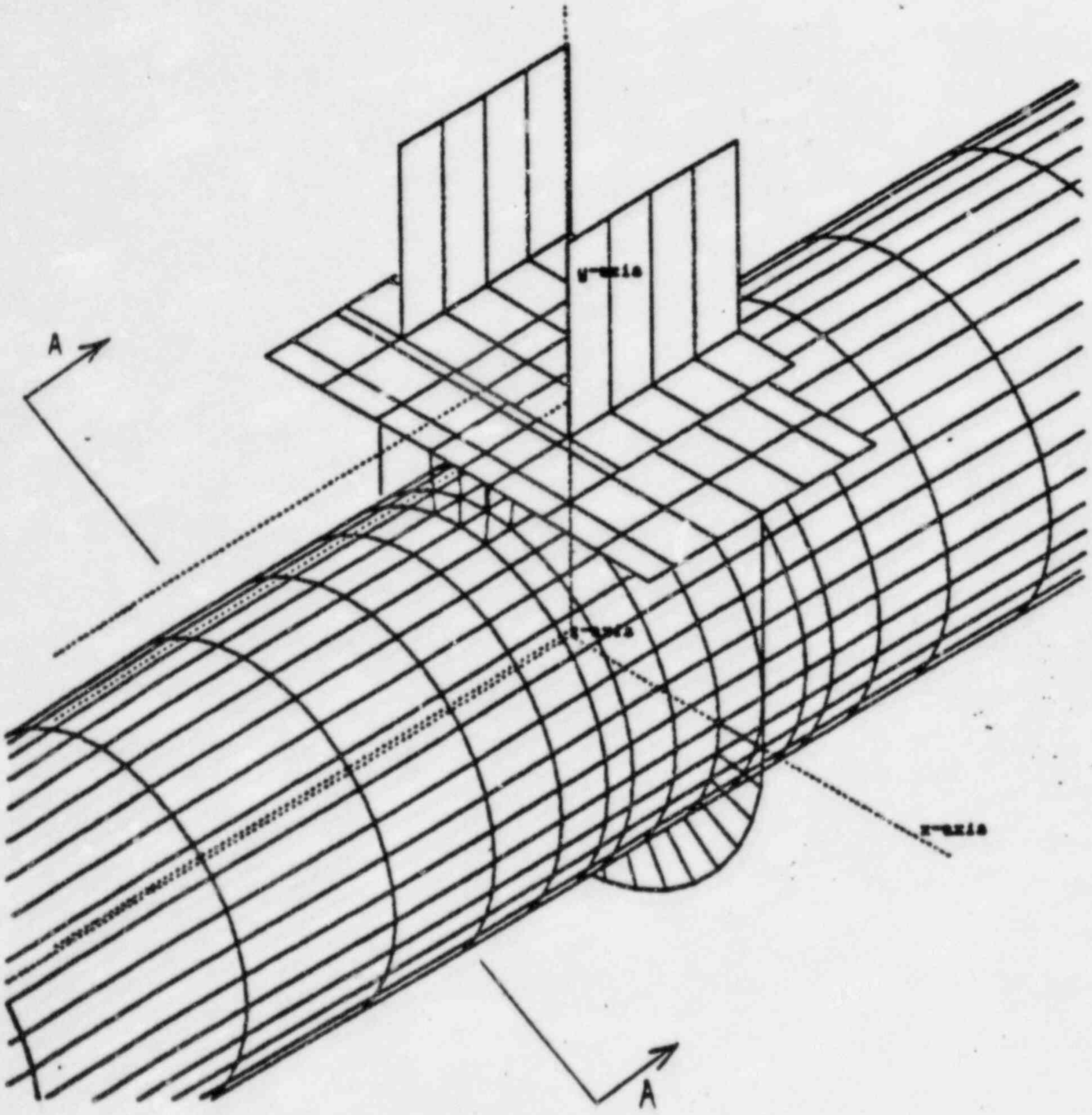
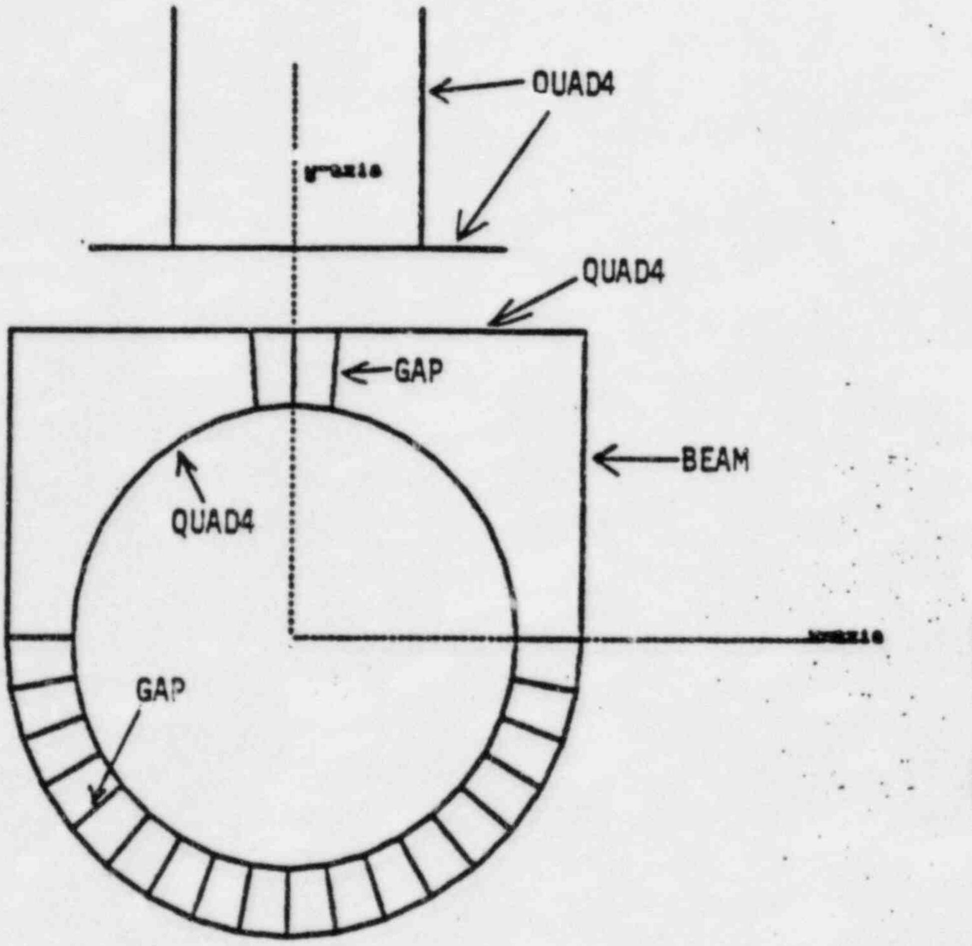


FIGURE IV-1

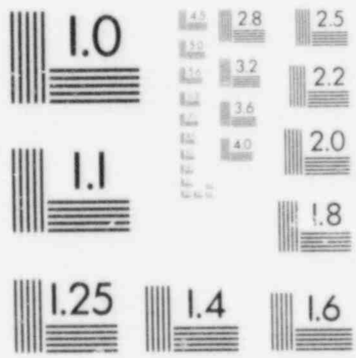
This Fiure is applicable to 4", 10" SCH 40S, and 10" SCH 80 pipe support finite element models.



SECTION A-A

FIGURE IV-2

IMAGE EVALUATION TEST TARGET (MT-3)



150mm

6"

This Figure is applicable to the 32" pipe support model.

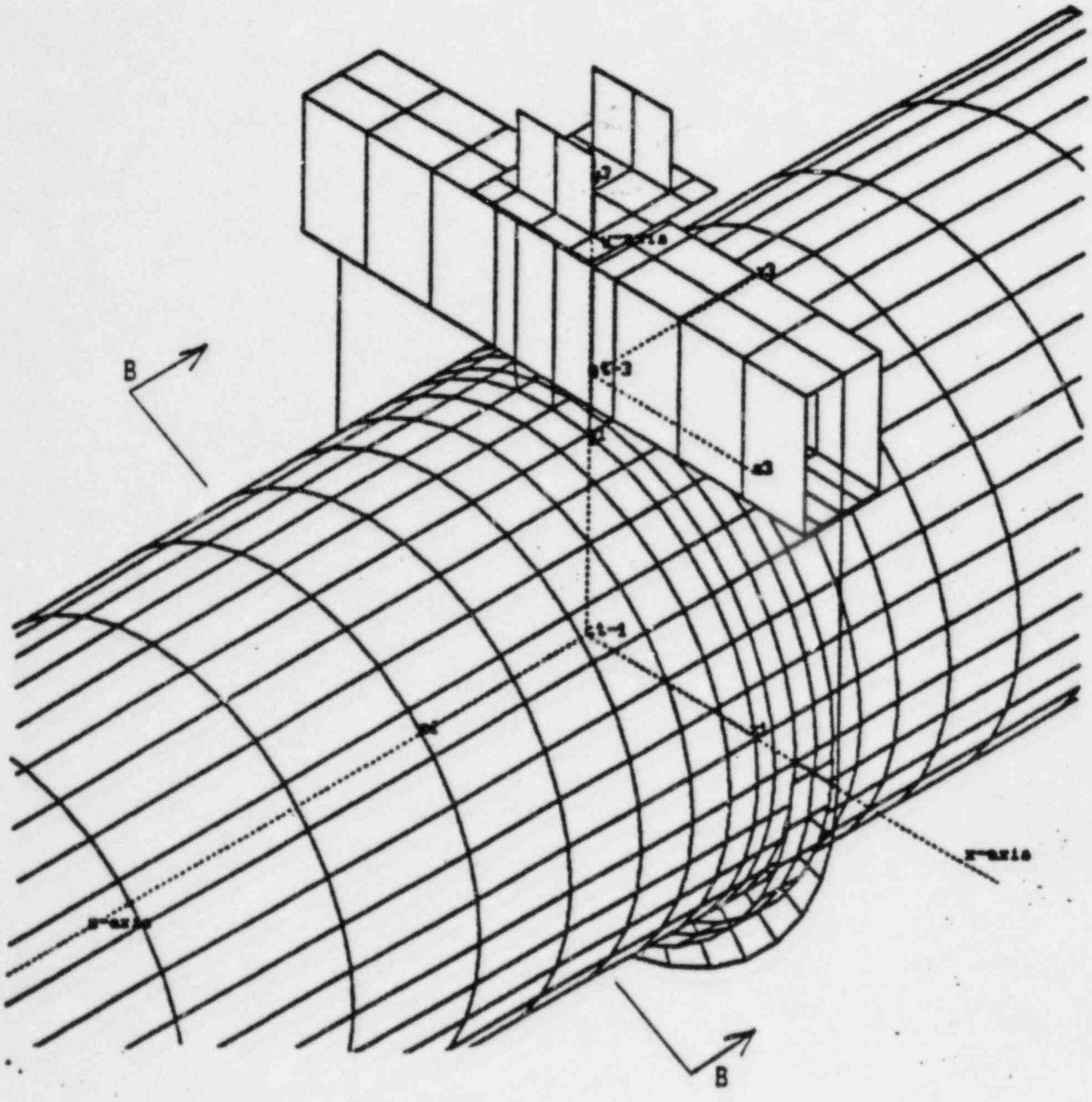
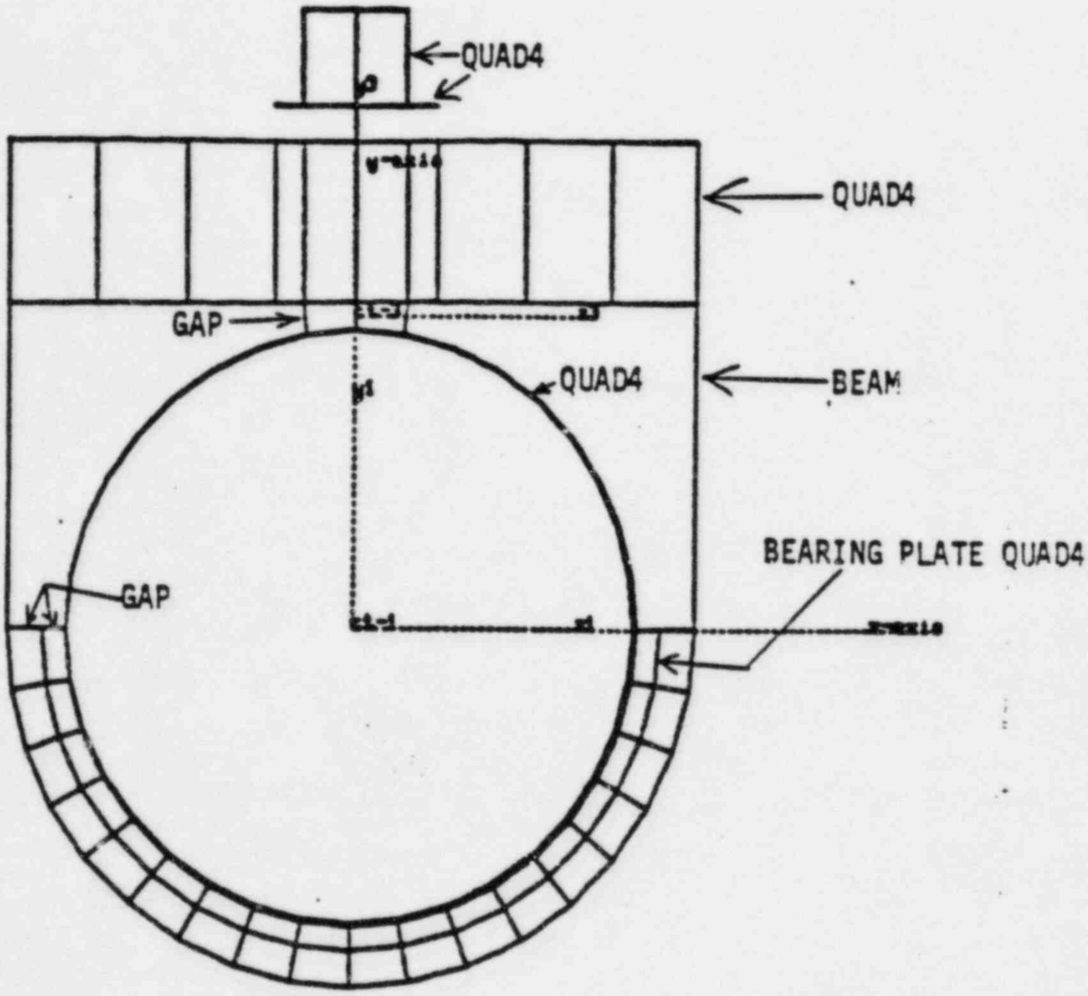


FIGURE IV-3

This Figure is applicable to the 32" pipe support model.



SECTION B-B

FIGURE IV-4

U-bolt	Beam	The curved portion of the U-bolt has one beam element for each 10° arc.
Gap/ Friction	Gap	Gap elements were used in the 4", 10" stainless and 10" carbon steel pipe/U-bolt models at two locations (1) between the pipe and cross piece and (2) between the pipe and U-bolt.

Three rows of gap elements were used between the pipe and cross piece. The inner row represents the point contact between the pipe and the cross piece (gaps closed) in the unloaded condition. The two outer rows of gap elements connect the pipe nodes 10° from the top of the pipe to the cross piece and are gapped for the distance between the pipe outside diameter and cross piece at this location.

Gap elements connecting the pipe and U-bolt were spaced at 10° intervals. In the unloaded condition, the bottom gap

(at the apex of U-bolt) is closed and the gaps vary from 0" to 1/16" at the pipe center line elevation.

Gap* elements were used in the 32" model at three locations (1) between the pipe and cross piece, (2) between the pipe and bearing plate, and (3) between the bearing plate and pipe.

Gaps between the pipe and cross piece in the 32" model are the same as for the 4" and 10" models.

Gaps between the pipe and bearing plate in the 32" model are spaced and gapped as in the 4" and 10" models with one exception. Instead of one gap element every 10°, there is a row of gap elements the width of the bearing plate.

---

* Gaps in the main steam model were represented with beam elements.

Gaps between the U-bolt and the bearing plate in the 32" model are spaced at 10° intervals, and all gaps are closed in the unloaded condition.

A friction coefficient (μ=.16) is specified for all gaps on all models. This is of course only active if the gap is closed or closes.

Cross Piece	QUAD4	The element spacing used is defined by the spacing of the gap and rear bracket elements.

The analytical model discussed above is used for elastic non-linear gap and friction effect solutions. In order to assess the effect of plastic material behavior on preload and pipe stresses, non-linear material behavior was included in the 10" Sch 40 pipe model, as discussed further in Section VII. The remainder of the 10" Sch 40 pipe/U-bolt model for the plastic analysis was unchanged.

Adequacy of Modeling Technique

Three methods were used to demonstrate that the models are (a) accurate for the idealized situation, and (b) representative of the real condition existing in the field. Item (a) was addressed by:

1. Comparison of element modeling size to recommended modeling limits.

2. Comparison of analytical results obtained from the model to results calculated by other methods.

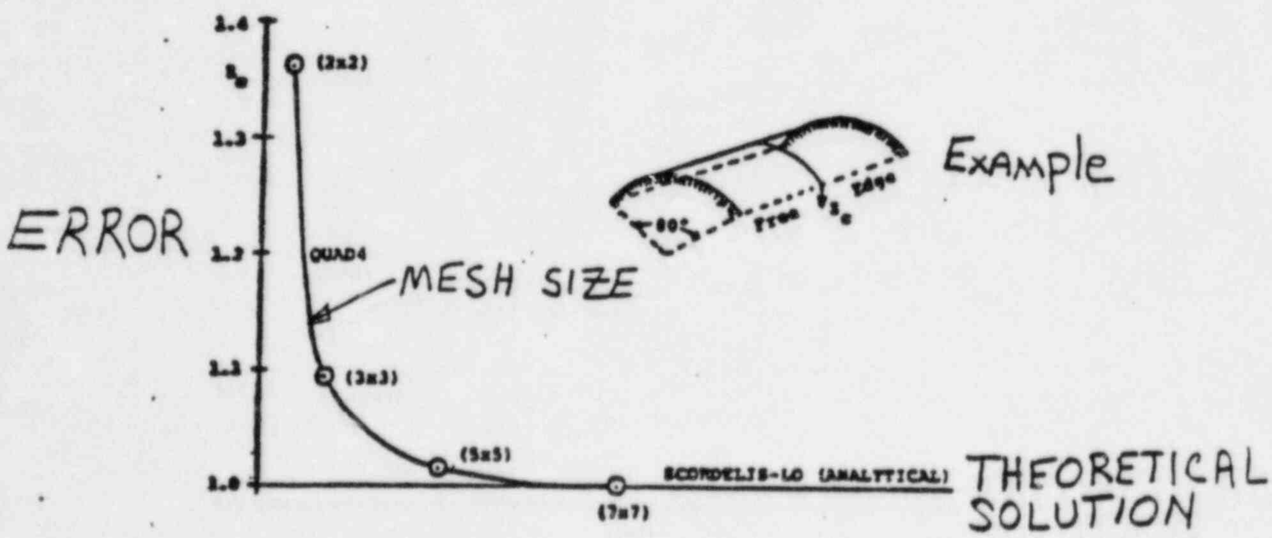
Item (b) was addressed by:

3. Comparison of analytical results to rest results.

Items 1 and 2 are discussed below, Item 3 is discussed in the next section.

1. Element Size

Cf primary importance to the solution accuracy of this model is the element selection and element circumferential spacing which represents the pipe wall. The following information reproduced from the MSC NASTRAN Application Manual, Volume II, confirms that the selection of QUAD4 elements at 10 degree circumferential intervals to represent the pipe wall is appropriate.



A 5 x 5 mesh of QUAD4 plate elements, one element every 16 degrees yields a result within approximately one percent of the theoretical result.

Although the QUAD4 is not a shell element in the strict sense, it should be noted that excellent results can be obtained when the angle subtended by individual elements is limited to some 10 degrees.

Additional examples and user information on the QUAD4 element are presented in the MSC/NASTRAN Application Manual. Some of the important features of the QUAD4 are indicated below.

-- QUAD4 may be used as a membrane element, bending element, or as a combined membrane-bending element.

-- The accuracy of the QUAD4 element is effectively independent of aspect ratio.

From the above, it can be stated that the element size used in the finite element model will yield results that are in close agreement with the theoretical solution.

2. Comparison of Analytical Results

The U-bolt bending moment calculated with the finite element analysis model was compared to hand calculations. Assuming the pipe is infinitely rigid, as the U-bolt is preloaded the gap between the U-bolt and the pipe will close with no pipe distortion. The theoretical moment resulting from the change in U-bolt curvature would be:

$M = EI\ (1/R_2 - 1/R_1)$

$E = 27.9 \times 10^6$ psi

$I = \pi D^4/64$ (D is the U-bolt diameter)

R_1 and R_2 are the U-bolt radii of curvature before and after preload

The U-bolt moments calculated with the finite element model and the theoretical U-bolt moments calculated by the above equation are compared below.

PIPE SIZE	ANALYSIS MOMENT IN-KIPS	DIAMETER U-BOLT INCHES	R_2 INCHES	R_1 INCHES	THEORETICAL MOMENT IN-KIPS
4"	.818	.5	2.5	2.5625	.835
10"-40S	.738	.75	5.75	5.8125	.810
10"-80	.847	.75	5.75	5.8125	.810
32"-MS	13.890	2.75	18.375	18.4375	14.449

As seen from the above table, the moments calculated with the finite element analysis model compare favorably to those calculated with the theoretical U-bolt bending moment equation.

Analysis Technique

The Westinghouse analysis of the pipe/U-bolt structure included four load cases which represent the loading conditions for the pipe/U-bolt assembly. A description of each load case and the analysis simulation technique used to evaluate the effect is given below.

LOAD CASE	SIMULATION TECHNIQUE	NASTRAN LOADING
Preload/"Cinching" due to torque on U-bolt nuts	Torquing of the U-bolt nuts shortens the U-bolt legs (straight portion). Preload was therefore simulated by imposing a negative temperature on the U-bolt legs which results in shortening of the U-bolt legs.	TEMPRB
Radial expansion of the pipe from internal pressure	To simulate the effect of pipe internal pressure, internal pressure was defined on the inside surface of each pipe element.	PLOAD 2
Thermal expansion of the pipe/pipe support	To simulate the thermal load, the pipe wall was specified to be at the normal operating temperature and, the pipe	TEMPRB TEMPP1

	support was given the temperature distribution determined from the test thermocouple readings.	
Applied strut load *	Engineering and construction tolerances are such that the maximum strut misalignment which results in pipe torsion is 5 degrees, see Figure IV-1. Maximum strut forces were applied as shown to determine pipe stresses and checked for slippage.	FORCE

*All four finite element models had 10 ft. of pipe length (corresponding to test specimens) with pinned end conditions. Therefore the applied strut load, at pipe midspan, would create a moment equal to PL/4. An equal and opposite moment was applied to the ends of the beam so that at the pipe midspan, the moment would be zero. This was done so that the results from the finite element analysis would reflect only the local effect of the U-bolt. The magnitude of piping moment stresses is determined in section VII of this report.

V. COMPARISON OF ANALYTICAL RESULTS TO TEST RESULTS

In this section, the analytical results are compared to the test results obtained from the tests performed at the WAESD Engineering Laboratory. Three test conditions are compared. They are preload, thermal, and load distribution. The other tests performed (torque versus preload, friction test and creep) were not used for comparison since they were performed to provide data on physical properties and behavior.

1. Preload

Test

The pipe, U-bolt and cross piece were instrumented with strain gauges at the locations shown in Figures V-1 and V-2. The U-bolt nuts were torqued to the maximum value specified by Texas Utilities and strain readings were taken and converted to stress.

Analysis

The U-bolt legs were "shrunk" using the previously described technique until the analytical U-bolt leg force corresponded to the preload test leg force. A stress comparison was then performed between test and analytical stresses.

Table 1 provides the strain gauge versus analytical element correlation. Tables 2, 3, 4, and 5 correlate stresses from the test and analysis for the 4", 10" stainless, 10" carbon, and 32" pipe/U-bolt assemblies, respectively.

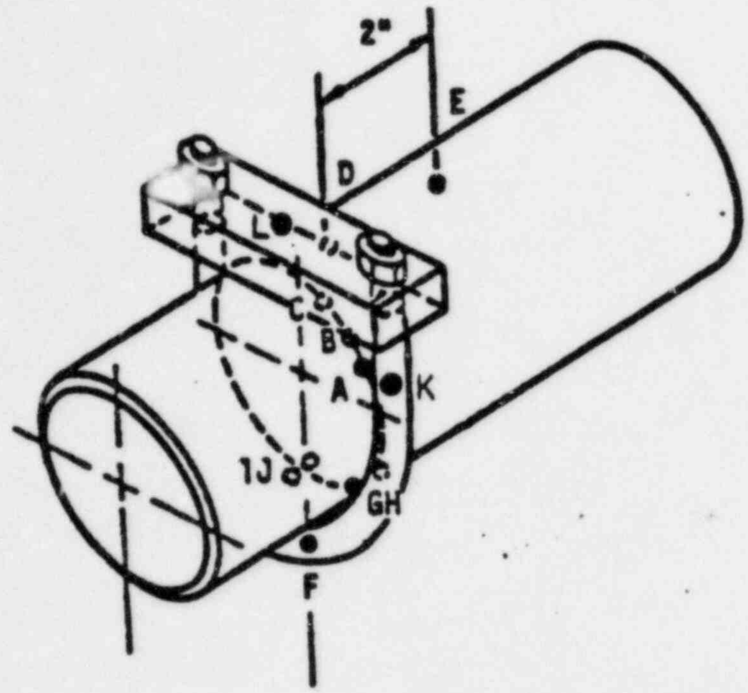


FIGURE V-1

PIPE INSTRUMENTATION

*A. Circumferential mounting on pipe as close as possible to U-bolt contact with pipe.

*B. Circumferential mounting on pipe midway between cross piece and U-bolt contact points.

*C. Circumferential mounting on pipe as close as possible to point of cross piece contact with pipe.

*D. Longitudinal mounting on pipe as close as possible to cross piece contact with pipe.

*E. Longitudinal mounting on pipe two inches from D.

*F. Longitudinally mounting on U-bolt shank opposite cross piece.

G. and *H. Longitudinal and circumferential strain measurements 45 degrees below horizontal center line on pipe as close as possible to the U-bolt contact area.

I. and *J. Longitudinal and circumferential strain measurements on pipe opposite cross piece as close as possible to the U-bolt contact area.

K. Longitudinal strain measurement on U-bolt shank. Two gauges were placed on each U-bolt shank 180 degrees apart. The strain readings were averaged to remove the effects of bending.

L. Longitudinal strain measurement on cross piece as close as possible to strut rear bracket.

* Actual strain gauge locations are detailed on Figure V-2.

TESTING STRAIN GAGE LOCATIONS

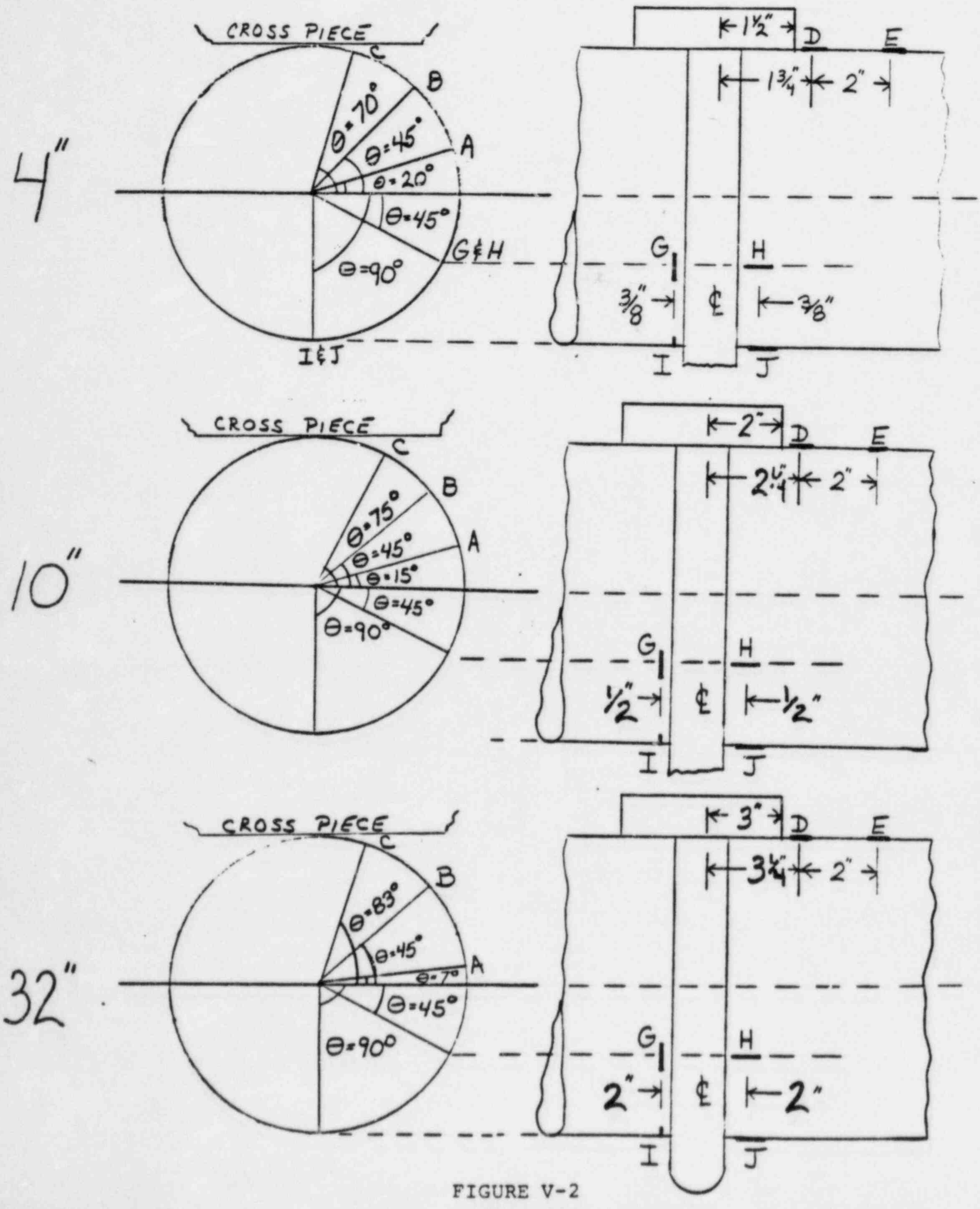


FIGURE V-2

TABLE 1

Analysis Elements Corresponding to Strain Gauge
Locations on Previous Page

Test Gage	4" Schedule 160 Model Element	10" Schedule 40S Model Element	10" Schedule 80 Model Element	32" Main Steam Model Element
A	152/151	155	78	215
B	154	152	82	219
C	157/156	149	85	223
D	734	723	590	295
E	806	759	626	295
F	Not used			
G	145	125	73	210
H	145	125	73	210
I	141	129	69	206
J	141	129	69	206
K	81	3	861	133
L	Not used/4" test	1132	7	60

NOTE: Two elements are given where the strain gage location falls between elements.

TABLE 2

4 Inch Pipe/U-Bolt Assembly

Comparison of Test Results to Analysis Results for Max Preload
Max Preload = 60 ft-lb.

Test Gage	Test Strain ϵ (μin/in)	Test Stress lb./in^2 $\sigma = \epsilon E$	Analysis Stress (lb./in^2)
A	+158	4,424	10,014/9,026
B	+ 86	2,408	3,413
C	-255	-7,140	-8,009/-16,265
D	-107	-2,996	-3,878
E	+ 53	1,484	1,020
F	Not used on 4" test		
G	-15	- 420	-10,410/-1066
H	-184	-5,152	-14,519/-5100
I	- 95	-2,660	-9,827/-3815
J	-209	-5,852	-12,151/-7769
K	+951	26,628	27,544
L	Not used on 4" test		

$E = 28 \times 10^6$ lb./in^2

Note: Two element stresses are given where the strain gauge falls between elements.

TABLE 3

10 Inch Schedule 40S Pipe/U-Bolt Assembly

Comparison of Test Results to Analysis Results for Max Preload

Max Preload = 100 ft-lb.

Test Gage	Test Strain ϵ (μin/in)	Test Stress lb./in^2 $\sigma = \epsilon E$	Analysis Stress (lb./in^2)
A	+425	11,900	13,568
B	+486	13,608	11,536
C	-561	-15,708	-20,040
D	-270	-7,560	-20,561
E	-10	-280	-8,764/-6,135
F		Not used on 10" test	
G	-275	-7,700	-10,212
H	-685	-19,180	-13,684
I	+30	840	-6,489
J	+70	1,960	- 8,816
K	+458	12,838	12,726
L	+579	16,212	16,702

$E = 28 \times 10^6$ lb./in^2

TABLE 4

10 Inch Schedule 80 Pipe/U-Bolt Assembly

Comparison of Test Results to Analysis Results for Max Preload

Max Preload = 100 ft-lb.

Test Gage	Test Strain ϵ (μin/in)	Test Stress lb./in^2 $\sigma = \epsilon E$	Analysis Stress (lb./in^2)
A	+213	5,964	8,910
B	+200	5,600	6,261
C	-289	-8,092	-14,170
D	-24	-672	-8,291
E	20	560	-2,094
F		Not used on 10" test	
G	-76	2,128	-8,665
H	-92	-2,576	-10,875
I	-47	-1,316	-5,755
J	-44	-1,232	-6,888
K	+613	17,164	17,008
L	+754	+21,112	23,678

$E = 28 \times 10^6$ lb./in^2

TABLE 5

32 Inch Pipe/U-Bolt Assembly

Comparison of Test Results to Analysis Results for Max Preload

Max Preload = 240 ft-lb.

Test Gage	Test Strain ϵ (μin/in)	Test Stress lb./in^2 $\sigma = \epsilon E$	Analysis Stress (lb./in^2)
A	0	0	1,113
B	+15	+420	789
C	-56	-1,568	-4,859
D	-11	-308	-493*
E	4	+112	-493*
F		Not used on 32" test	
G	0	0	-1,170
H	0	0	-1,446
I	-3	-84	-1,075
J	0	0	-1,208
K	+23	+644	+1,018
L	+25	+700	+1,242

$E = 28 \times 10^6$ lb./in^2

*One Analysis element encompasses both these strain gauge locations.

DISCUSSION

Given below is a comparison of the test and analytical results for each pipe size evaluated.

4" Pipe Sch 160

Good agreement[1/] exists between the measured and calculated stresses at B and C. Stresses measured at test points A, G, H, I and J do not compare as well with those calculated with the finite element analysis model. This is due to the difference between the pipe and U-bolt boundary conditions (U-bolt fit-up). In the test, gaps existed between the U-bolt and pipe which did not completely close due to out-of-roundness of the pipe and tolerances in the U-bolt bend. This resulted in point contacts between the pipe and U-bolt which cannot be simulated exactly in the finite element analysis. The stress in the U-bolt (point K) is comparable in both the test and analysis. Good agreement between stresses exists at points D and E. In general, there are differences between the test and finite element preload analysis, but they are within the accuracy of (1) the location of the strain gauges, (2) the finite element location with respect to the strain guages and, (3) the simulated fit-up between the U-bolt and pipe in the analysis.

1/ Agreement between tests and analysis will be considered good, if the finite element analysis stress within (±) 10 degrees of the test strain gauge in the circumferential direction matches the test stress. Longitudinal stress agreement will be considered good if the finite element stress matches the test stress within one longitudinal analysis element of the test strain gauge location.

10" Pipes Sch 40 and 80

Good agreement exists between measured and calculated stress at locations A, B and C. There are differences near the contact point of the cross piece and pipe, but this is due to the large increase in stress over a short distance (small angle). Preload test stresses for points G, H, I and J do not compare well to calculated stresses at the same point. This is again attributed to the difference in U-bolt fit-up between the test and the finite element analysis model. Analytical results at points D and E do not agree with the test results. The differences are attributable to the following:

First, the same cross piece was used on both the 10" Sch 40 and the 10" Sch 80 preload test and it was observed to be "bowed". The edges of the cross piece, in the direction that bears on the pipe, are approximately 1/64" higher than the cross piece centerline. This is probably due to welding of the rear bracket. The deflection of the pipe in the analysis at the contact point of the cross piece is approximately 1/32" for the 10" Sch 40 pipe. Therefore, the effect of "bowing" would be to concentrate the contact force at the center of the plate which would lower the stresses at points D and E making the test and analysis comparison more favorable.

The second observation which would also result in forces being concentrated under the center of the plate, thereby reducing the calculated stresses at points D and E, is the fit-up of the U-bolt. If the unloaded gap condition in Section III is

assumed between the pipe sidewalls and the U-bolt, contact forces concentrate at the edges of the cross piece plate. If the U-bolt is assumed to conform to the pipe in the unloaded condition, the bearing force under the cross piece will be concentrated at the center of the cross piece plate, thereby reducing the calculated stress at points D and E. This observation is supported by the analysis performed for the boundary conditions described. The fit-up of the U-bolt and pipe was observed to more closely approximate conformance to the pipe in the unloaded condition than the 1/16" gaps at the pipe sidewalls input in the analysis.

32" Pipe

The stresses measured in the test and calculated for the 32" pipe, cross piece, and U-bolt are comparable. They are very low, almost immeasurable. This is due to the fact that the U-bolt preload torque was very small (240 ft-lb). It sho ld be noted however that the radius of curvature of the main steam line U-bolt was too small, as shipped to the WAESD Engineering Laboratory, to fit on the pipe. The U-bolt legs were therefore "spread" at the Laboratory so that fit-up to the pipe could be accomplished. This spreading resulted in pipe/U-bolt fit-up not consistent with the U-bolt fit-up as modeled. Specifically, (1) point contact existed in the unloaded position between the pipe sidewalls and U-bolt, and (2) the apex of the U-bolt which should have been the only contact point in the untorqued condition had a gap which did not close even at the maximum preload torque value.

2. THERMAL

Test

Thermocouples were placed on the pipe/U-bolt assembly at the locations defined previously in Section III, and high temperature strain gauges were placed on the U-bolt legs. All four U-bolt supports were assembled and insulated in accordance with Texas Utilities' instructions. Strip heaters were then sealed in the pipe and heated until the pipe skin reached the specified normal operating temperature. The skin temperature was held constant until the pipe hanger reached equilibrium. Thermocouple readings were recorded at equilibrium and U-bolt leg strain readings were recorded and converted to force.

Analysis

The pipe elements were given temperatures equivalent to the normal operating temperature defined by Texas Utilities and used in the test. The pipe hanger thermal distribution was input from thermocouple data recorded during the thermal test. The forces in the U-bolt legs were calculated and are compared to test results in the following table.

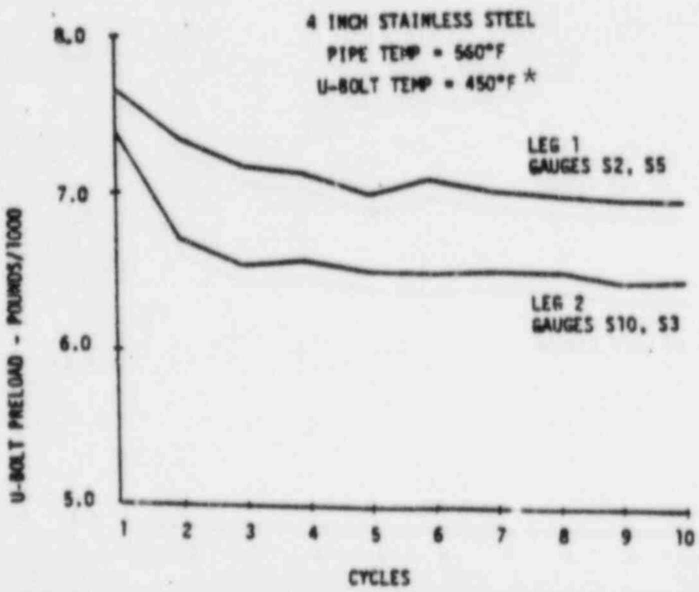
PIPE SIZE	THERMAL TEST FORCE	THERMAL ANALYSIS FORCE
4"	1.7 kips	2.9 kips
10" stainless	-.5 kips	1.3 kips
10" carbon	Not tested	1.4 kips
32" Main Steam	10 kips	21 kips

Comparison of 32" Pipe Thermal Test and Analysis

As stated previously, the actual test fit-up between the pipe and U-bolt for the 32" pipe pipe/U-bolt assembly was totally different than that for the 32" finite element analysis. Therefore, differences in results would be expected.

Comparison of 4" Pipe Thermal Test and Analysis

The measured forces were 70 percent smaller than those calculated with the finite element analysis model. During the preload test, high stresses and some yielding in the U-bolt and hanger cross piece were noted for the 60 ft-lb preload torque. This observation is substantiated by the finite element analysis that was performed. See Stress Tables in Section VIII. In addition to increasing the hanger loads and stresses, the elevated temperature would decrease the yield point of the hanger material. The expected result would be that more yielding would occur and the increase in U-bolt leg load predicted by a linear finite element analysis would not be duplicated in the test. This conclusion is supported by the figure below developed from the test results.



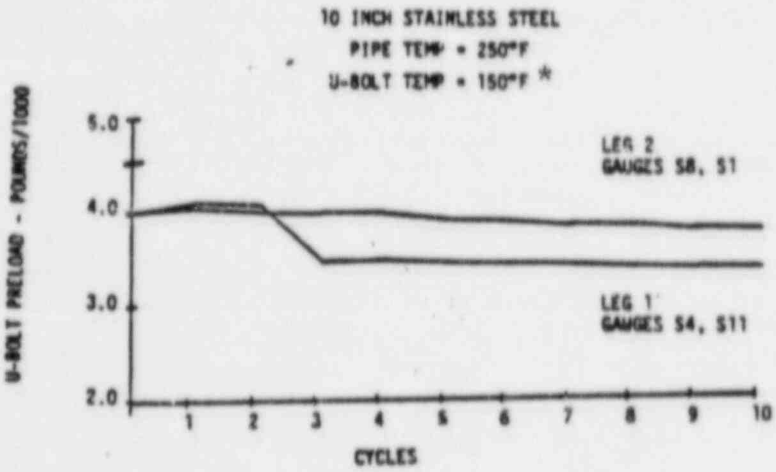
* Representative temperature. Refer to page 16 for actual temperature distribution in the U-bolt.

The sharp decrease in the U-bolt preload during the first few thermal cycles indicates the U-bolt assembly did not behave elastically during the thermal test; i.e., yielding occurred.

Comparison of 10" Stainless Steel Pipe Thermal Test and Analysis

Although the elastic finite element model predicts piping stresses above the pipe material yield point test results shown below indicate linear material behavior.

* Representative temperature.



Because U-bolt preload as a function of thermal cycles is essentially constant for both U-bolt legs, the difference between measured and any calculated thermal loads is attributed to hanger "binding". Both U-bolt legs were torqued at ambient temperature to approximately 4,500 pounds. At temperature (pipe = 250°F) the U-bolt leg forces decreased to 4,000 pounds but no further decrease was noted for the second and third thermal cycles. Leg 1 decreased approximately another 600 pounds between thermal cycle 3 and 4. These step decreases in U-bolt load indicate some

form of mechanical binding existed in the hanger assembly following the 100 ft-lb preload torque which was relieved during thermal cycling. The results between test and analysis are therefore not comparable because of this phenomenon.

3. LOAD DISTRIBUTIONS

Test

In this test, the strut was given an applied force ("push"), see figure III-5. This test was performed on the 10" stainless pipe only. The test consisted of torquing the U-bolt nuts to 100 ft-lb and then pushing on the rear bracket with a force of 7,000 lb. The test resulted in a 2,000 lb. reduction in U-bolt preload.

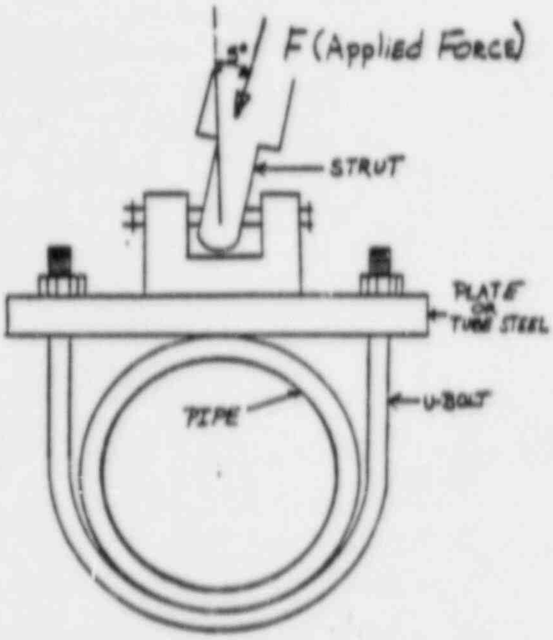
Analysis

In the analysis corresponding to this test, a preload of 100 ft-lb and a 10,000 lb. push force was applied to the rear bracket. The analysis resulted in a 2,700 lb. reduction in the U-bolt preload value. This is equivalent to a reduction in U-bolt preload of 1890 lb. for a 7000 lb. push force.

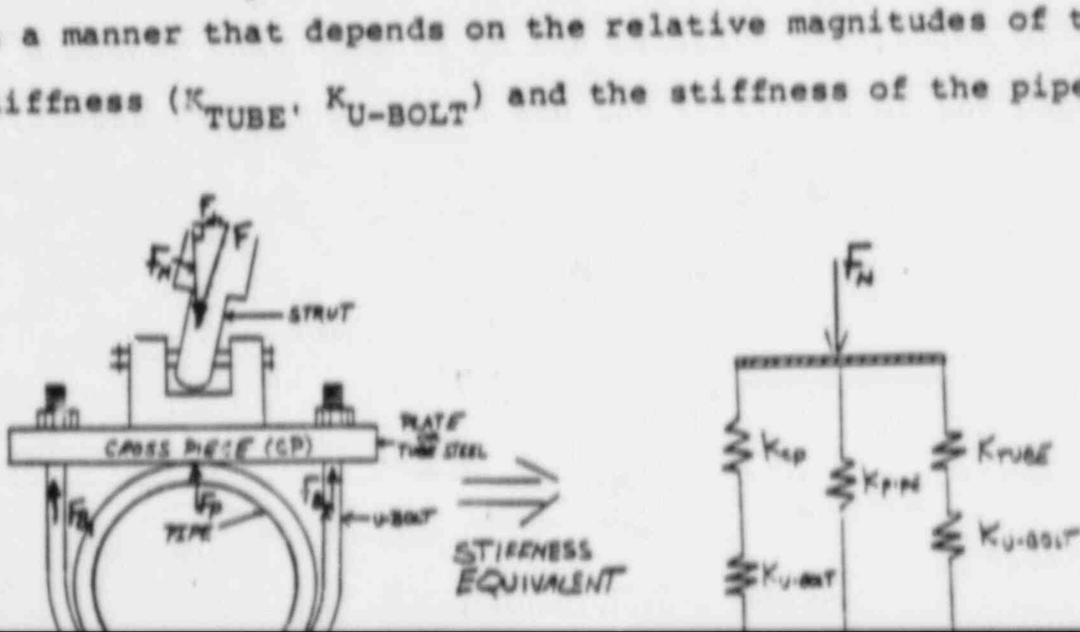
As can be seen from the above comparison of test and analysis the finite element model closely predicts the test behavior.

VI. BEHAVIOR OF PIPE/U-BOLT ASSEMBLY FOR "SLIP"

This section describes the characteristics of the U-bolt assembly when subjected to a push load which produces a moment about the pipe centerline.



The stiffness equivalent diagram below can be used to show that the normal force (F_N) for an externally applied load (F) will be distributed between F_p (pipe contact force) and F_B (U-bolt leg force) in a manner that depends on the relative magnitudes of the hanger stiffness (K_{TUBE}, K_{U-BOLT}) and the stiffness of the pipe (K_{PIPE}).

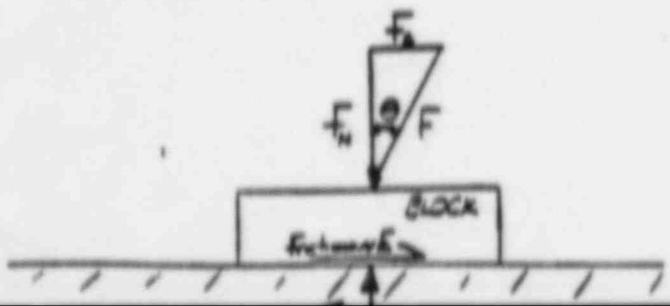


If the pipe is very stiff compared to the hanger, the external force component (F_N) will increase the pipe contact force (F_P) by essentially F_N with no decrease in the U-bolt preload value (F_B). If the pipe is not very stiff compared to the hanger stiffness, the external force component will be split between an increase in F_P and a decrease in the U-bolt preload F_B which depends on the relative stiffnesses of the hanger and pipe. The change in F_B and F_P due to the applied external force component (F_N) is tabulated below for the four pipe/U-bolt models evaluated. A minus (-) sign indicates a decrease in load; a plus (+) sign indicates an increase in load.

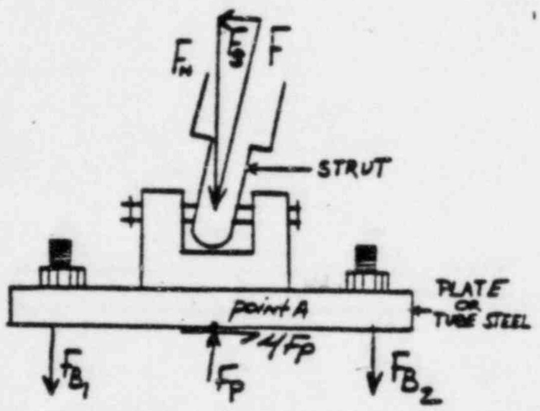
		Change Resulting From FN	
PIPE SIZE	F_N	ANALYSIS F_B	ANALYSIS F_P
4"	2,000 lb.	- 184 lb.	+ 1,630 lb.
10" Stainless	10,000 lb.	- 2,615 lb.	+ 4,770 lb.
10" Carbon	10,000 lb.	- 1,295 lb.	+ 7,410 lb.
32" Main Steam	100,000 lb.	-24,600 lb.	+50,800 lb.

The distribution between F_B and F_P of the applied external force component determines to a large extent the ability of the hanger to resist. This is explained below.

For the simple example below, it can be shown that the block will not slide until the force (F) is applied at an angle (θ) which exceeds $\tan^{-1}\mu$ where μ is the coefficient of friction, i.e., F_S is equal to μF_N at the initiation of slip.



The cross piece for the pipe support assembly in question also obeys the same principle.



From the free body diagram of the cross piece shown above, it can be seen that the cross piece will not "slide" until the applied strut force component (F_S) exceeds μF_P. Using the minimum coefficient of friction determined by the testing program (μ = .16) and F_P* from the analysis, the following information is given and conclusions reached regarding "slippage" of the hangers.

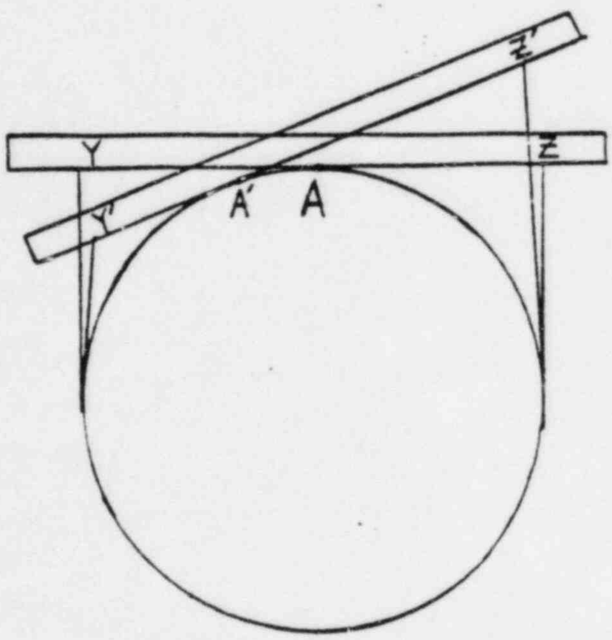
PIPE SIZE	F_S	F_P*	μF_P	"SLIP" OCCURS
4"	174 lb	15,280 lb	2,444 lb	No
10" Stainless	871 lb	19,210 lb	3,073 lb	No
10" Carbon	871 lb	25,790 lb	4,126 lb	No
32" Carbon	8710 lb	105,000 lb	16,800 lb	No

* F_P is the calculated load due to preload, thermal pipe expansion pressure pipe expansion and push.

As can be seen from the table above, F_S is in all cases is much smaller than μF_P and therefore slippage of the cross piece will not occur.

Although all four hanger cross pieces can be seen not to "slip", the external force component (F_S) creates a moment about point A which must be balanced by a difference in U-bolt leg tensions (F_{B1} and F_{B2}).

The difference in U-bolt leg tensions is balanced by two methods, the friction between the U-bolt and the pipe. The second and a mechanism of resistance that would be in effect if the friction capacity between U-bolt and pipe is exceeded. The mechanisms of resistance would be as diagramed below.



It has previously been demonstrated that the cross pieces will not "slide" on the pipe. However, the cross piece will "roll" on the pipe surface until a difference in leg tensions exists

sufficient to balance the moment created about point A by force F_S. In other words, it can be shown that as the cross piece rolls, point A moves to A', point Y moves to Y', and point Z moves to Z', and that the distance from Y' to Z' (around the pipe) is greater than the distance Y to Z. The increase in length creates additional tension in the U-bolt. The cross piece will continue to "roll" until (friction forces, μFn) sufficient normal forces between the U-bolt and pipe are created to balance the cross piece moment created by F_S.

VII. ACCEPTANCE CRITERIA

The rules of Subsection NB given in the 1979 ASME Boiler and Pressure Vessel Code, Section III, (referred to as code hereafter) may be used to qualify a Class 1, 2, 3 or non-nuclear safety piping components if the designer is willing to comply with all requirements of Subsection NB. Subsections NC, ND and the ANSI B31.1 piping code provide criteria for evaluating primary and secondary stresses only by limiting the pressure plus principal stresses caused by moment loading in the pipe. By evaluating stress intensities (as opposed to principal stresses), detailed localized stresses, and the effects of cyclic loadings (fatigue), one is clearly enveloping the intent of NC, ND and ANSI by an NB evaluation.

Based on the above, the discussion for acceptability of pipe stresses induced by the U-bolt pipe support will generally be addressed herein using NB-3600 of the code. NB-3600, which governs the design and qualification of piping systems gives little specific guidance to the method that should be used to evaluate stresses similar to those caused by the U-bolt pipe support. However, equations 9 and 12 of NB-3600 may be used to assess the piping stresses as discussed herein. This will provide a means for evaluating piping stresses in the vicinity of the U-bolt caused by the loading assoc[illegible]ed with the U-bolt.

In applying code equations 9 and 12 to assess the significance of the piping stresses it is necessary to classify the stresses that exist in the pipe due to the different loading conditions associated with the U-bolt. These piping stresses are classified following ASME nomenclature.

Using code equations 9 and 12 requires that the mechanical stresses in the pipe resulting from sources other than the U-bolt assembly be included. Because of the number of hangers involved, this information was not supplied to Westinghouse for evaluation purposes.

In order to conservatively estimate the magnitude of these mechanical stresses, a generic procedure was developed and is discussed herein. In the development of these mechanical stresses, it is assumed that the pipe is stressed to its maximum allowable value at a high stress intensification point (e.g., elbow). It is assumed that the piping moment which results in this maximum allowable stress also occurs at the U-bolt hanger locations. The maximum piping stress is assumed to be at the stress limit of 1.5 S_m. 2/ This code allowable stress level is used instead of the higher Level D limit (3 S_m) for the following three reasons.

1. SSE induced stresses need not be included in the secondary or peak stress evaluation.
2. SSE piping seismic levels are in the same order of magnitude as those associated with the OBE (operating basis earthquake) stresses.
3. In general, the stresses in the piping will not be at the code allowable.

2/ S_m is the allowable stress intensity defined by the Code.

Therefore, based on the above, using a 1.5 S_m limit will produce mechanical piping stresses that are realistic.

In the assessment of the piping stresses, the allowable stress used for code Equations 9 and 12 is 3 S_m. It is realistic to use this allowable for evaluation of primary stresses in the vicinity of the U-bolt since the preload stresses in the pipe are predicted to be very high by elastic analysis due to local stress concentrations. In reality, these stresses will not result since small insignificant plastic deformations will occur in the region of the high stress concentration resulting in stress redistribution in a small local area of the pipe. Further, a fatigue evaluation was performed in which this high stress state is considered.

In the sections that follow, the piping mechanical stresses are determined and the piping stress state in the region of the U-bolt support assessed.

<u>Code Classification of Stresses</u>

Pressure Membrane Stress - Primary Stress

Preload - Local Primary Membrane Stress

Thermal Pipe Growth Restriction - Secondary or Peak Stress

Pressure Pipe Growth Restriction - Secondary or Peak Stress

Pipe Hanger Load - Primary Stress

<u>Applicable Code Equation Stresses</u>

From the above stress definitions and code requirements, the applicable code equations for these stresses and limits can be selected.

1. Equation (9) of the code.

This equation must consider (a) primary membrane pressure stress, (b) piping moments at the hanger location due to pipe deadweight and Seismic, (c) stress due to preload of the U-bolt and (d) stresses due to the deadweight and seismic portion of the applied hanger load.

2. Equation (12) of the code.

This equation must consider (a) thermal stresses due to pipe radial thermal growth restriction, (b) pressure growth restriction at the U-bolt, and (c) piping moments at the hanger location due to pipe longitudinal thermal expansion.

Development of Mechanical Pipe Stresses

Tabulated below are the 3 S_m stress limits for the pipes in question.

PIPE SIZE	3 S_m AT NORMAL OPERATING TEMPERATURE
4" Sch 160 STAINLESS	50.52 KSI
10" Sch 40 STAINLESS	60 KSI
10" Sch 80 CARBON	60 KSI
32" MS CARBON	58.26 KSI

All sources of load must be considered when evaluating the pipe stress state at the U-bolt location. Because of the number of hangers involved, detailed information was not supplied to Westinghouse for the piping moments at the hanger locations. However, these moments are considered in a general manner herein. Generic piping stress intensification values are developed in the sections that follow.

1. Piping Moment Stresses for Equation 9

Piping stresses caused by sources other than the U-bolt can be approximated assuming that (1) the pipe meets the service level limit of 1.5 S_m (pressure + moment), (2) pipe supports are not located at high stress index locations and (3) the high stress index locations generally control the piping system stress qualification (elbows, tees, branches, etc.). The magnitude of moment stress can be determined at the hanger locations assuming (1) the maximum system stress is at a long radius elbow (high stress intensification), (2) the long radius elbow is stressed to the code allowable (1.5 S_m) and (3) the piping moment at the pipe hanger is the same as the piping moment at the long radius elbow. The result of this evaluation is presented below.

PIPE SIZE	ALLOWABLE STRESS FOR SERVICE LEVELS A AND B	LONGITUDINAL PRESSURE STRESS	EQ. 9 PIPING MOMENT STRESS AT PIPE HANGER*
4" Sch 160	25.26 ksi	4.8 ksi	12.146 ksi
10" Sch 40	30 ksi	4.4 ksi	6.0477 ksi
10" Sch 80	30 ksi	2.6 ksi	9.23 ksi
32" Main Stm.	29.13 ksi	7.1 ksi	6.7 ksi

* Piping Moment Stress = (Allowable Stress - Pressure Stress) x Ratio of Stress Indices

2. Piping Moment Stresses for Equation 12

The qualification of a piping system for thermal growth moment stresses is known to be controlled by the elbows. This is due to the high stress index at elbows and the fact that the piping thermal expansion moments are generally highest at elbows. Assuming again that (1) the maximum system stress is at a long radius elbow (high

stress intensification), (2) the long radius elbow is stressed to the code allowable (3 S_m) and (3) the moment at the pipe hanger is the same as the long radius elbow, the magnitude of the piping moment stress can be determined at the hanger locations.

PIPE SIZE	ALLOWABLE STRESS	EQ. 12 PIPING MOMENT STRESS AT PIPE HANGER*
4" Sch 160	50.52 ksi	22.49 ksi
10" Sch 40	60 ksi	10.63 ksi
10" Sch 80	60 ksi	15.15 ksi
32" Main Stm.	58.26 ksi	13.34 ksi

* Piping Moment Stress = Allowable Stress x ratio of stress indices.

EVALUATION OF PIPING STRESSES

From the finite element analysis performed for the four different U-bolt assemblies, the highest stressed piping element (see Stress Tables In Section VIII) correspond to the same relative location, directly under the cross piece.

At this location, the circumferential and longitudinal stresses correspond to the major and minor principal stresses (i.e., no shear stress). The longitudinal, circumferential, and the major and minor principal stresses at the inside and outside pipe surface at this location are tabulated below for the maximum load case (Preload + Thermal + Pressure + Push).

		(ksi) Long.	(ksi) Circ	Principal Stress (ksi) Maj	(ksi) Minor
4" Sch 160	(inside)	10.49	44.79	44.78	10.50
	(outside)	-26.65	-34.07	-26.63	-34.08
10" Sch 40	(inside)	10.77	72.71	72.71	10.77
	(outside)	-48.02	-73.46	-48.02	-73.46
10" Sch 80	(inside)	10.24	43.15	43.15	10.23
	(outside)	-30.22	-44.38	-30.22	-44.38
32" M.S.	(inside)	19.58	47.17	47.22	19.52
	(outside)	-31.01	-34.10	-30.89	-34.22

(negative is compressive stress)

It is evident from the values given above that the longitudinal and circumferential stress are similar to the principal stresses.

Adding the longitudinal pressure effects results in the following:

		Long. ksi	Circ. ksi
4" Sch 160	(inside)	15.29	44.79
	(outside)	-21.85	-34.07
10" Sch 40	(inside)	15.17	72.71
	(outside)	-43.62	-73.46
10" Sch 80	(inside)	12.84	43.15
	(outside)	-27.62	-44.38
32" M.S.	(inside)	26.68	47.17
	(outside)	-23.91	-34.10

The relationship between the longitudinal and circumferential stresses and the stress intensity is illustrated in Figure VII-2.

It can be seen from Figure VII-2 that the primary and secondary piping moment stresses, which only affect the longitudinal stress, will not increase the stress intensity provided:

1) The longitudinal stress does not exceed the circumferential stress.
2) The longitudinal stress does not decrease to the extent it changes sign, in which case the stress intensity would be the absolute sum of the circumferential and longitudinal stresses.

The mechanical piping stresses developed earlier for the primary (Equation 9) and secondary (Equation 12) effects must be included with the longitudinal stresses. The total stress intensity for each of the piping sizes evaluated are:

	TOTAL STRESS INTENSITY
4" Sch 160	64.14 ksi
10" Sch 40	74.21 ksi
10" Sch 80	54.69 ksi
32" M.S.	47.17 ksi

Splitting the above total stress intensity into primary (equation 9) and secondary (equation 12) stresses results in the following:

	Eq 9 ksi	Eq 9 Allowable ksi	Eq 12 ksi	Eq 12 Allowable ksi
4" Sch 160	31.60	50.52	32.54	50.52
10" Sch 40	60.61	60	13.6	60
10" Sch 80	38.15	60	16.54	60
32" M.S.	30.57	58.26	16.6	58.26

As can be seen from a comparison of the above maximum stress intensities to the equation 9 and equation 12 allowable stresses, the 10" Sch 80, 32" MS, and 4" pipes meet the stress criteria established herein. The 10" Sch 40 pipe essentially meets all of the allowable limits.

Further, the following elastic/plastic analysis and test results are used to demonstrate the acceptability of this pipe/U-bolt assembly for existing stress states.

Because the acceptance stress criteria used (3 S_m) exceeds the material yield strength, the magnitude of yielding must be addressed, and the effect on U-bolt preload determined. Two methods were used to investigate the yield effect on preload. First, the nonlinear material properties (stress-strain curve) were input into the finite element analysis, and second, a test was performed at the WAESD Engineering Laboratory.

The elastic/plastic finite element analysis was performed on the 10" Sch 40 pipe/U-bolt assembly for the preload load case (100 ft-lb). The 10" Sch 40S pipe was selected for this analysis because the stresses in this assembly were the highest, and therefore would yield the highest strains. The stress-strain curve input is should in Figure VII-1. The results from the analysis showed that yielding occurred only locally around the contact point of the pipe and cross piece. Further, the strains were very small and do not significantly affect the linear stress distribution within the pipe. This is seen below in the stress comparison table.

TEST GAUGE	ELASTIC ANALYSIS PSI	PLASTIC ANALYSIS* PSI
A	13568	15358
B	11536	14068
C	-20040	-17567
D	-20561	-19674
E	-8764/-6135	-9232/-6536
F	---	---
G	-10212	-10667

* See footnote next page.

H	-13684	-14152
I	-6489	-7265
J	-8816	-9677
K	12726	14335
L	16702	18640
U-Bolt	5.62 kips leg tension	6.33*kips leg tension

By comparison of the above elastic and elastic/plastic analysis results, it can be seen that for the preload case, the difference in deformation between the elastic analysis, that shows stresses equal to twice yield locally under the cross piece, and the elastic/plastic analysis, that shows yielding and redistribution of the stress under the cross piece, is negligible.

This analytical result is substantiated by test results. The test was conducted by preloading the hanger to 100 ft-lb, and then pushing with a 7000 lb force. The loss of preload as a result of the 7000 lb applied push load was an accurate indication of the plastic deformation encountered. The U-bolt leg forces before the load was applied were 5393 lb and 6469 lb, when the load was removed the leg loads were 5060 lb and 6168 lb, respectively. The 300 lb loss in leg force is an indication of very small plastic deformation.

It is possible, under certain conditions, for progressive radial distortions of the pipe wall to occur at pipe clamps. This would occur only if two conditions existed.

1. Maximum pipe stress intensity exceeded 3 S_m and

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* Slightly higher (13%) U-bolt leg tension applied than in the elastic analysis. No attempt made thru iterations to make them identical since they are close in magnitude.

2. The clamp is periodically retorqued.

This is not a problem for the typical U-bolt type restraint because U-bolts are not periodically checked for preload and, therefore, not retorqued.

A fatigue assessment was performed to determine the possibility of initiating a crack in the pressure boundary at the vicinity of the U-bolt.

This assessment included a calculation for Class 1 piping and a calculation for Class 2 and 3 piping. The calculation for Class 1 piping utilized the U-bolt stresses from the 4" SCH 160 case. The calculation for Class 2 and 3 piping utilized the worst case stresses from the other three cases.

For the Class 1 piping, two locations were chosen for the evaluation. These locations were the 10" accumulator injection line and 3" normal charging line, assuming the U-bolt was located somewhere between the first check valve and the reactor coolant loop nozzle.

These two locations represent the most difficult section to qualify for fatigue, (due to extremely severe thermal transient loadings) of all the Class 1 auxiliary piping. In this evaluation, several load cases were modified to conservatively include the clamp induced stresses.

This evaluation included all design transients as well as the clamp induced stresses. For the charging and accumulater line the usage factors were .04 and .06 respectively. Based on an ASME code allowable of 1.0 or .1 for pipe break requirements, the usage factor requirements are met.

For the Class 2 and 3 assessment, a simplified fatigue calculation was made using the worst case data. Based on maximum stress, excluding preload (which is not a cyclic event), the 32" main steam line was selected. Based on a simplified calculation, the incremental usage factor was less than .01. This assessment was based on a representative Class 1 method. It provides a measure of the fatigue damage that would be expected at the U-bolt location. This assessment indicates that the integrity of the pressure boundary, based on fatigue consideration, would not be significantly affected by the localized U-bolt effects.

From the above discussion, it can be concluded that the pipe stresses induced in the pipe due to cinching of the U-bolts will not exceed acceptable limits. Further, the loss of preload due to any yielding in the pipe material will be small as demonstrated by the test and plastic analysis performed.

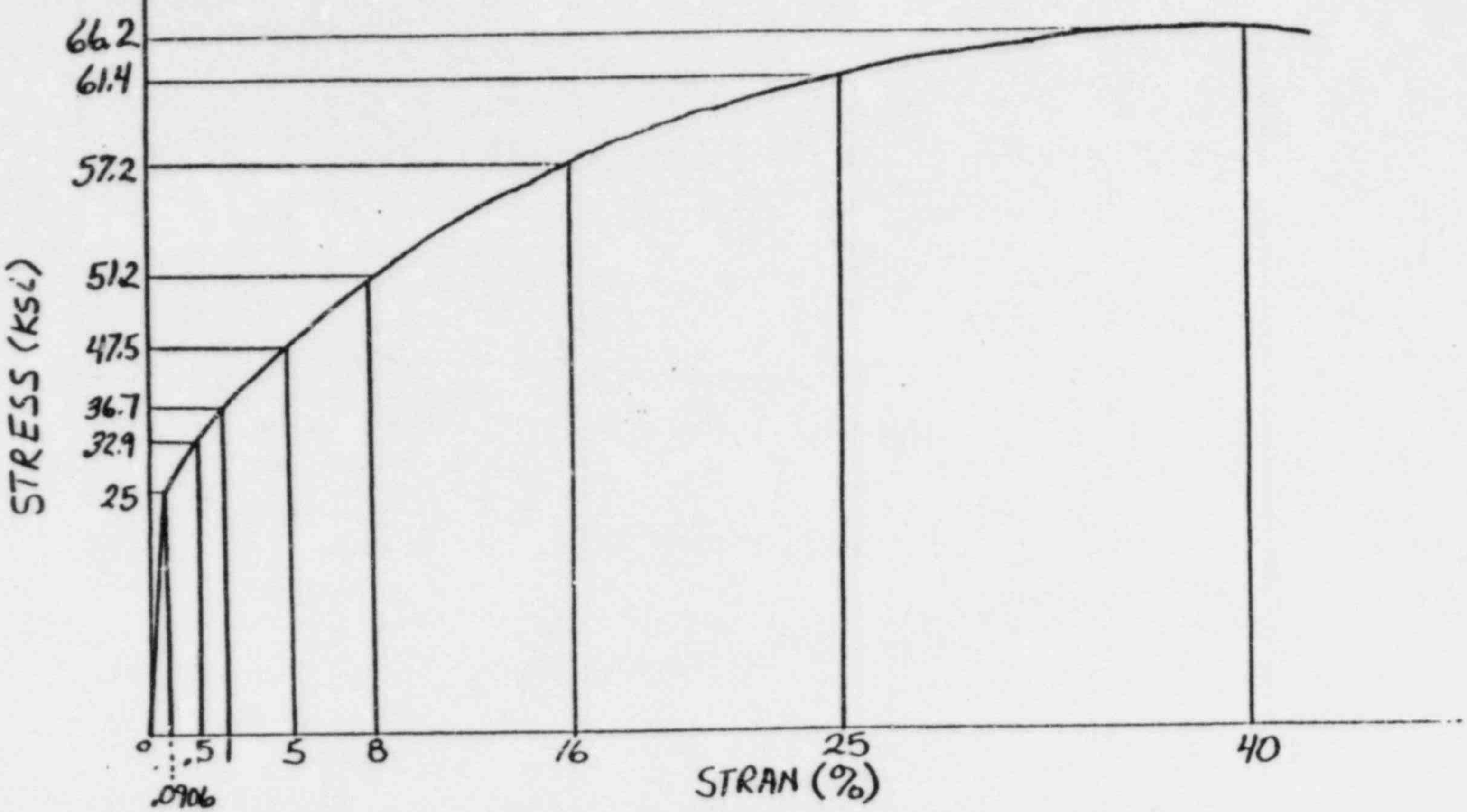
Developed Using

$$\sigma = \sigma_0 (\varepsilon)^N$$

$$N = \frac{\ln\left(\frac{66.2}{25}\right)}{\ln \frac{.4}{9.06 \times 10^{-4}}} = .16$$

$$\sigma_0 = \frac{\sigma_u}{\varepsilon_u^{\,N}} = 76.7$$

$$\sigma = 76.7(\varepsilon)^{.16}$$



Stress-Strain Curve SS304 At 210°F

Figure VII-1

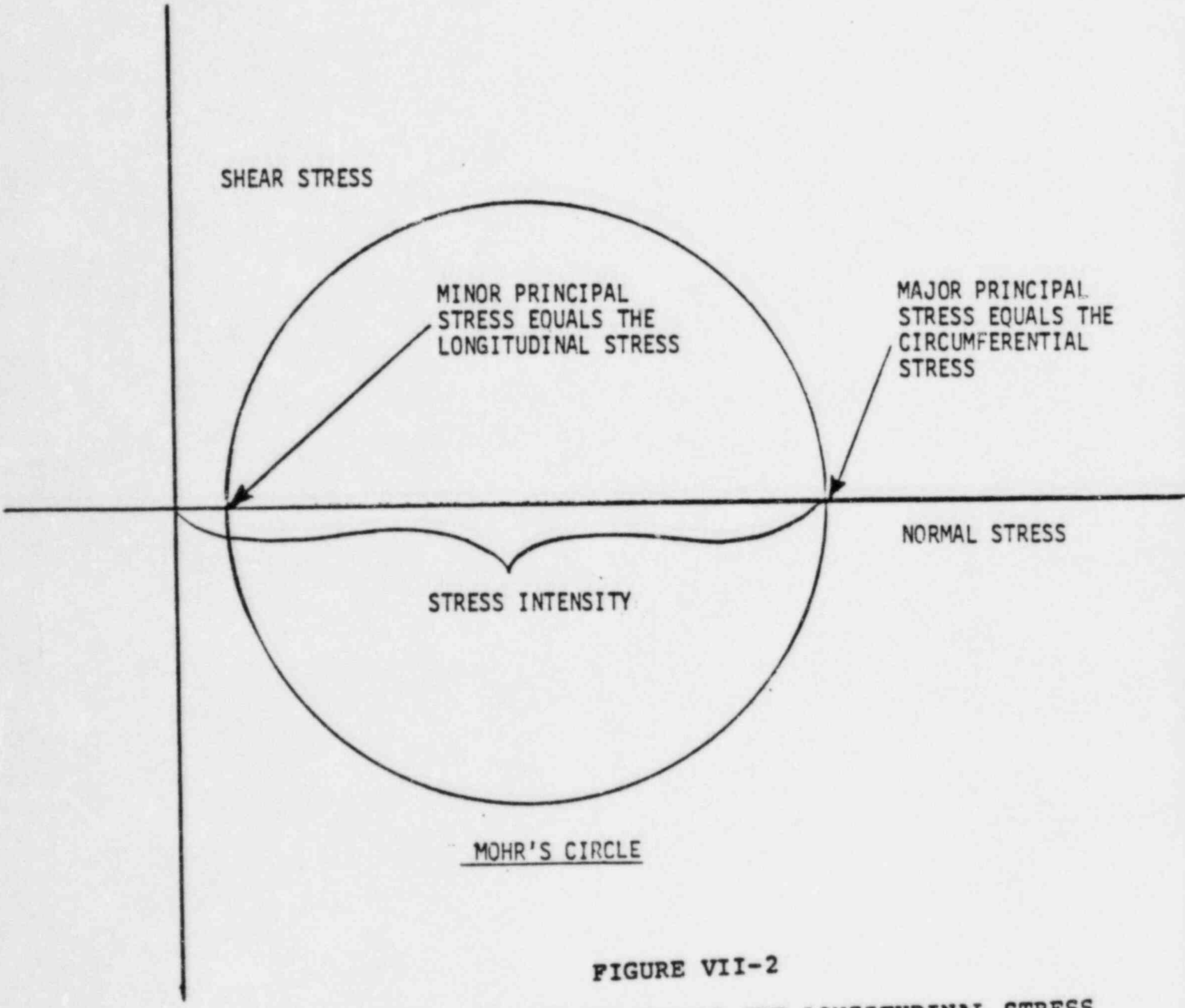


FIGURE VII-2

MINOR PRINCIPAL STRESS EQUALS THE LONGITUDINAL STRESS

VIII. STRESS SUMMARY TABLES

This section presents the stress results from the four load cases defined in Section IV.

Terms used are defined below.

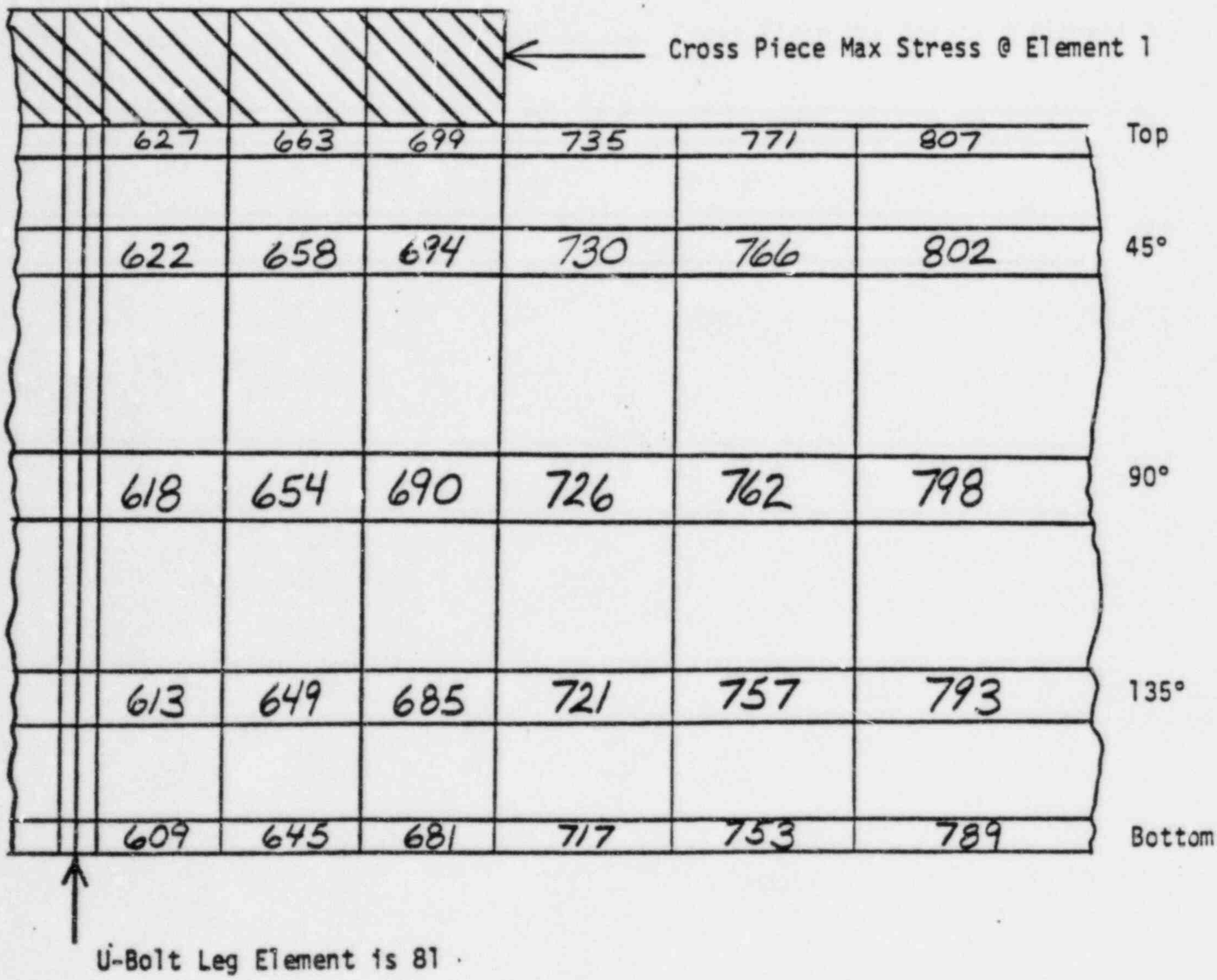
MAX Longitudinal Stress: The longitudinal direction of the pipe is parallel to the pipe centerline; for the cross piece it is parallel to the long axis of the cross piece, and for the U-bolt is parallel to the U-bolt centerline. The maximum longitudinal stress (at the inside or outside surface) at the centerpoint of the analysis element is tabulated.

MAX Circumferential Stress: The circumferential stress is in a direction parallel to the pipe circumference; for the cross peice it is parallel to the shortest axis of the cross piece. The maximum circumferential stress (at the inside or outside surface) is tabulated at the centerpoint of the analysis element.

MAX Stress Intensity: This is the maximum absolute difference between the major prinicpal stress or minor principal stress and zero. The maximum surface stress (inside or outside) at element center point is tabulated. Note, that because max surface stresses were tabulated for the longitudinal and circumferential stresses the stress intensity cannot in all cases be derived from the longitudinal and circumferential stresses tabulated.

4" Sch 160

STRESS SUMMARY TABLE ELEMENT DIAGRAM



NOTE: Not to scale, for relative location of stress summary elements only.

SIDE VIEW OF PIPE

4" Sch 160

Preload

Analysis Element	Max Longitudinal Stress lb./in^2	Max Circum. Stress lb./in^2	Max Stress Intensity lb./in^2
627	-13480	-26091	26091
663	-13646	-24736	24736
699	-10669	-20639	20644
735	-3879	-13042	13047
771	441	7770	7787
807	1021	4198	4201
622	2921	-8363	8396
658	2423	-7392	7674
694	1728	-5973	6573
730	852	-4006	5049
766	406	-1602	3411
802	-343	216	1486
618	8610	-14544	14552
654	7112	-12789	12834
690	5405	-10828	10875
726	3645	-8923	8946
762	1461	-6510	6511
798	-985	3941	3947
613	-10410	-14519	14535
649	-1066	-5100	5833
685	-1967	-1824	4640
721	1533	1071	3886
757	741	730	2629
793	229	232	1422
609	-9827	-12151	12153
645	-3815	-7769	7776
681	-2165	-5590	5889
717	-1746	5756	6169
753	-1189	5476	5484
789	965	3810	3815
1 (Cross Pc.)	-34187	3838	34187
81 (U-Bolt)	27544	——	——

4" Sch 160

Preload + Thermal

Analysis Element	Max Longitudinal Stress lb./in^2	Max Circum. Stress lb./in^2	Max Stress Intensity lb./in^2
627	-19970	-39305	39305
663	-20250	-37204	37204
699	-15779	-30939	30947
735	-5510	-19338	19348
771	918	11174	11199
807	1578	5915	5919
622	4930	-13667	13706
658	4018	-12023	12373
694	2790	-9661	10426
730	-1325	-6454	7680
766	-638	-2568	4934
802	-590	339	2077
618	12011	-22022	22050
654	9965	-18753	18888
690	7686	-15433	15564
726	5256	-12563	12626
762	2137	-9105	9108
798	-1369	-5507	5518
613	-14775	-20100	20179
649	-1141	-8065	8414
685	-2999	-2946	6596
721	2351	1840	5598
757	1198	1236	3742
793	400	377	2020
609	-13728	-17332	17334
645	-4498	-10389	10396
681	-2393	-7170	7600
717	-2207	7300	8142
753	-1770	7346	7356
789	1336	5289	5297
1 (Cross Pc.)	-50841	5714	-50841
81 (U-Bolt)	42320	----	----

4" Sch 160

Preload + Thermal + Pressure

Analysis Element	Max Longitudinal Stress lb./in^2	Max Circum. Stress lb./in^2	Max Stress Intensity lb./in^2
627	-20675	42905	42905
663	-21146	41286	41286
699	-16266	36879	36879
735	-5526	28795	28795
771	968	20854	20854
807	1616	15399	15399
622	5075	15120	15120
658	4160	14510	14510
694	2911	13637	13637
730	1396	12413	12413
766	-652	10792	10792
802	-599	9600	10199
618	12433	23645	23645
654	10345	22256	22256
690	8005	20834	20834
726	5482	19210	19210
762	2235	16628	16628
798	-1416	13580	13955
613	-15730	14803	15730
649	-1130	11666	12796
685	-3161	11296	14457
721	2455	11120	13273
757	1243	10523	10523
793	405	9642	9642
609	-14403	9228	14403
645	-4613	11944	11944
681	-2379	15176	15176
717	-2214	16829	16829
753	-1809	16887	16887
789	1382	14747	14747
1 (Cross Pc.)	-52255	5607	52255
81 (U-Bolt)	43685	---	---

4" Sch 160

Preload + Thermal + Pressure + Push

Analysis Element	Max Longitudinal Stress lb./in^2	Max Circum. Stress lb./in^2	Max Stress Intensity lb./in^2
627	-26648	44786	44786
663	-27411	43125	43125
699	-22498	38753	38753
735	-10773	29820	29820
771	-2929	21973	24902
807	-3567	15274	18523
622	-6045	15419	15419
658	-5625	14790	14790
694	-5045	13895	14801
730	-4430	12596	15024
766	-3912	10904	14816
802	-4290	9603	13893
618	11641	24341	24341
654	9525	22929	22929
690	7132	21459	21459
726	4531	19767	19767
762	-3158	17070	17070
798	-2433	13862	13862
613	15003	15043	15043
649	2745	11903	11903
685	5540	11471	11471
721	5055	11252	11252
757	3783	10551	10551
793	2874	9705	9705
609	12220	10173	12220
645	3159	12824	12824
681	1879	15808	15808
717	2913	17629	17629
753	3552	17004	17004
789	5787	15598	15598
1 (Cross Pc.)	-52824	55[illegible]3	52824
81 (U-Bolt)	41797	---	---

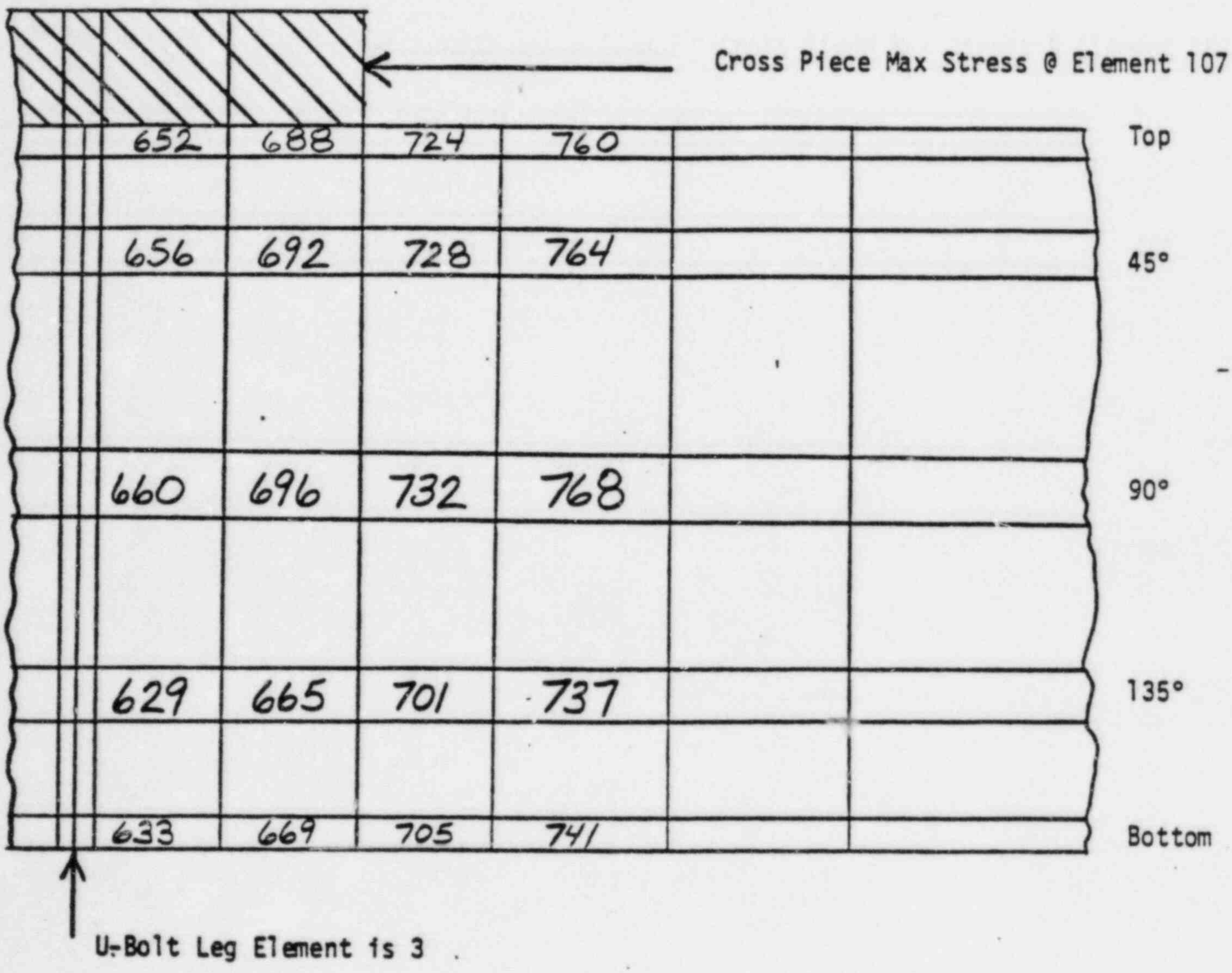
4" Sch 160

Partial Preload + Thermal + Pressure + Push

Analysis Element	Max Longitudinal Stress lb./in^2	Max Circum. Stress lb./in^2	Max Stress Intensity lb./in^2
627	-15591	24998	24998
663	-15566	23857	23857
699	-11032	21198	21198
735	-5380	16877	21435
771	-3314	14662	17631
807	-4393	11483	15876
622	-4202	11749	13791
658	-4039	11447	13812
694	-3779	11031	13819
730	-3541	10475	13854
766	-3856	9842	13698
802	-3894	9354	13248
618	-3032	15649	15649
654	-2782	15066	15066
690	-2528	14443	14443
726	-2236	13722	13722
762	-1856	12574	12574
798	-1570	11211	12609
613	6003	10560	10560
649	2608	9671	9671
685	3480	9876	9876
721	3338	9922	9922
757	2912	9697	9697
793	2589	9449	9449
609	8078	12990	12990
645	3711	12786	12786
681	3350	13068	13068
717	3677	13487	13487
753	3173	12528	12528
789	5172	12277	12277
1 (Cross Pc.)	-20235	2702	20235
81 (U-Bolt)	14275	---	---

10" Sch 40

STRESS SUMMARY TABLE ELEMENT DIAGRAM



NOTE: Not to scale, for relative location of stress summary elements only.

SIDE VIEW OF PIPE

10" Sch 40

Preload

Analysis Element	Max Longitudinal Stress lb./in^2	Max Circum. Stress lb./in^2	Max Stress Intensity lb./in^2
652	-25252	-48528	48529
688	-26833	-46576	46577
724	-20567	-37190	37300
760	-8764	25535	25913
656	8779	-14247	14280
692	7632	-13375	13642
728	5930	-11909	12529
764	3754	-9904	10916
660	10263	-14157	14194
696	9222	-12958	20700
732	7985	-11702	18916
768	6907	-10990	11310
629	-10193	-13672	13752
665	-1779	-7056	7214
701	-1434	4953	6393
737	-1334	4999	5034
633	-6487	-8826	8826
669	-3209	-4414	4414
705	-2709	-3093	5445
741	-3300	3955	5230
107 (Cross Pc.)	-16933	160	17093
3 (U-Bolt)	12726	——	——

10" Sch 40

Preload + Thermal

Analysis Element	Max Longitudinal Stress lb./in^2	Max Circum. Stress lb./in^2	Max Stress Intensity lb./in^2
652	-33218	-60636	60637
688	-34016	-57528	57595
724	-22554	-44359	44613
760	-9875	30795	31222
656	11344	-18210	18250
692	9818	-17050	17374
728	7410	-15115	15862
764	4553	-12502	13707
660	11423	-17027	17390
696	10612	-15001	15740
732	9352	-13250	14109
768	8086	-12454	13086
629	-12267	-16841	16856
665	-2850	-8036	8130
701	-2248	5208	7457
737	-1543	5738	5788
633	-10070	-13948	13948
669	-4804	-5884	5884
705	-4095	-3267	6911
741	-3434	4517	6291
107 (Cross Pc.)	-20153	191	20344
3 (U-Bolt)	15645	----	----

10" Sch 40

Preload + Thermal + Pressure

Analysis Element	Max Longitudinal Stress lb./in^2	Max Circum. Stress lb./in^2	Max Stress Intensity lb./in^2
652	-34425	58585	58589
688	-35642	56732	56911
724	-23452	49436	50182
760	-10214	-21518	40903
656	11943	24013	24037
692	10327	23023	23193
728	7781	21414	21692
764	4766	19256	19508
660	11402	21929	21930
696	10887	21546	21571
732	9680	21195	21222
768	8371	20680	20683
629	-12759	-8981	12770
665	-3117	10624	13747
701	-2467	13643	16112
737	-1582	14316	14403
633	-10557	-6608	10588
669	-5255	7577	12833
705	-4377	11465	15844
741	-3501	13253	15094
107 (Cross Pc.)	-20897	190	21087
3 (U-Bolt)	16346	——	——

10" Sch 40

Preload + Thermal + Pressure + Push

Analysis Element	Max Longitudinal Stress lb./in^2	Max Circum. Stress lb./in^2	Max Stress Intensity lb./in^2
652	-48024	-73462	73462
688	-50094	70533	70737
724	-35933	61070	62038
760	-16968	48720	53286
656	12386	28408	28468
692	10499	27111	27466
728	7431	24987	25578
764	-5410	22154	22755
660	13686	23408	23414
696	12995	23154	23158
732	12033	22897	22913
768	11017	22635	22702
629	8949	11275	11425
665	1701	12442	12454
701	2029	14188	14189
737	3438	14422	14424
633	7057	10646	10649
669	1062	12144	12144
705	638	14577	14578
741	2150	15729	15703
107 (Cross Pc.)	-14242	449	14694
3 (U-Bolt)	10026	----	----

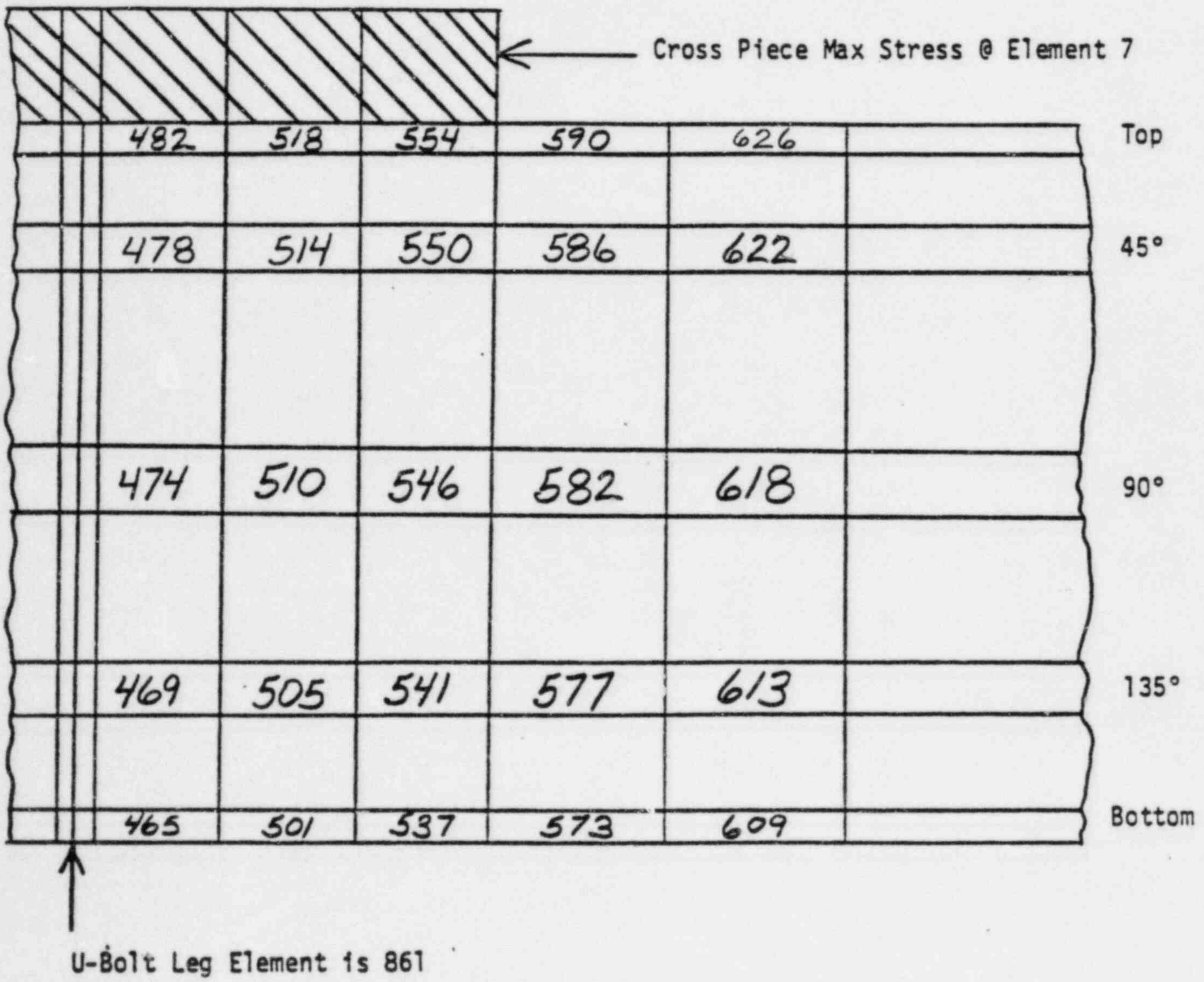
10" Sch 40

Partial Preload + Thermal + Pressure + Push

Analysis Element	Max Longitudinal Stress lb./in^2	Max Circum. Stress lb./in^2	Max Stress Intensity lb./in^2
652	-34782	52415	52418
688	-35733	50846	50977
724	-25655	44150	44756
760	-12735	35559	40018
656	7365	21878	21920
692	6115	20977	21215
728	-4931	19501	19904
764	-4330	17547	17975
660	8923	17762	17775
696	8671	17783	17806
732	8305	17828	17875
768	7845	17873	17965
629	6046	12770	12770
665	3090	12121	12125
701	3125	12165	12171
737	3486	11936	11942
633	4171	12251	12251
669	2376	12874	12874
705	2528	13776	13776
741	2952	14278	14278
107 (Cross Pc.)	-5953	340	6300
3 (U-Bolt)	3269	——	——

10" Sch 80

STRESS SUMMARY TABLE ELEMENT DIAGRAM



NOTE: Not to scale, for relative location of stress summary elements only.

SIDE VIEW OF PIPE

10" Sch 80

Preload

Analysis Element	Max Longitudinal Stress lb./in^2	Max Circum. Stress lb./in^2	Max Stress Intensity lb./in^2
482	-16270	-29008	29880
518	-17270	-29510	29520
554	-16130	-27490	27530
590	-8291	-18660	18710
626	-2094	11030	11100
478	4976	-9296	9315
514	4594	-8848	9025
550	4012	-8165	8567
586	2632	-6534	7310
622	-800	-3751	5302
474	7611	-11110	11120
510	6403	-10030	10130
546	5751	-9041	9242
582	4793	-7896	8127
618	3257	-6633	6757
469	-8840	-10900	11000
505	-2808	-6819	6900
541	-1091	-4377	4488
577	-923	-2650	2834
613	50	2249	2454
465	-6324	-7122	7173
501	-2143	-4749	4751
537	-1912	-3272	3274
573	-1481	-2634	2639
609	-2194	3283	3487
7.(Cross Pc.)	-4108	-14310	-18490
861 (U-Bolt)	17008	——	——

10" Sch 80

Preload + Thermal

Analysis Element	Max Longitudinal Stress lb./in^2	Max Circum. Stress lb./in^2	Max Stress Intensity lb./in^2
482	-19410	-34890	34890
518	-19330	-33550	33560
554	-17850	-30970	31000
590	-9655	-21460	21530
626	-2258	12650	12730
478	6040	-11390	11410
514	5493	-10750	10950
550	4720	-9848	10290
586	3020	-7808	8657
622	-964	-4412	6069
474	7230	-11570	11600
510	6933	-10730	10900
546	6555	-9801	10080
582	5545	-8658	8960
618	3766	-7398	7548
469	-10100	-12380	12390
505	-2839	-7737	7780
541	-1522	-4775	4863
577	-1255	-2791	2979
613	9	2446	2683
465	-8426	-10240	10240
501	-2424	-6388	6388
537	-2636	-4127	4128
573	-2034	-3087	4039
609	-2392	3809	4117
7 (Cross Pc.)	-4847	-15850	20730
861 (U-Bolt)	20175	——	——

10" Sch 80

Preload + Thermal + Pressure

Analysis Element	Max Longitudinal Stress lb./in^2	Max Circum. Stress lb./in^2	Max Stress Intensity lb./in^2
482	-19940	32960	32960
518	-19960	32210	32230
554	-18460	30720	30830
590	-9937	25260	25500
626	-2309	18120	18180
478	6249	12990	13000
514	5684	12640	12680
550	4884	12140	12310
586	3123	10900	11000
622	-995	8604	8652
474	7329	13620	13620
510	7084	13410	13410
546	6726	13240	13240
582	5698	12820	12820
618	3871	11880	11880
469	-10390	-7606	10400
505	-2885	5875	5876
541	-1604	6746	6758
577	-1319	7611	9004
613	8	7615	7700
465	-8832	-5672	8832
501	-2471	3636	4373
537	-2778	5070	5071
573	-2147	7125	7126
609	-2432	9001	9001
7·(Cross Pc.)	-5006	-16200	21250
861 (U-Bolt)	20803	——	——

10" Sch 80

Preload + Thermal + Pressure + Push

Analysis Element	Max Longitudinal Stress lb./in^2	Max Circum. Stress lb./in^2	Max Stress Intensity lb./in^2
482	-30220	-44380	44380
518	-31010	-43430	43430
554	-29230	41070	41310
590	16560	32650	32930
626	-5343	22890	25022
478	-5915	15580	15600
514	-5665	15100	15210
550	-5284	14440	14640
586	-4488	12770	13030
622	-3248	9659	11430
474	8729	16040	16060
510	8344	15840	15860
546	7844	15650	15670
582	6670	15190	15240
618	4559	14120	14240
469	8234	7742	8249
505	2592	7536	7542
541	767	8035	9401
577	1299	8435	8447
613	1464	8000	8050
465	5638	6329	6329
501	1708	6783	6790
537	464	7794	7805
573	1062	9521	9527
609	1833	10600	10710
7. (Cross Pc.)	4475	-14420	18955
861 (U-Bolt)	17388	----	----

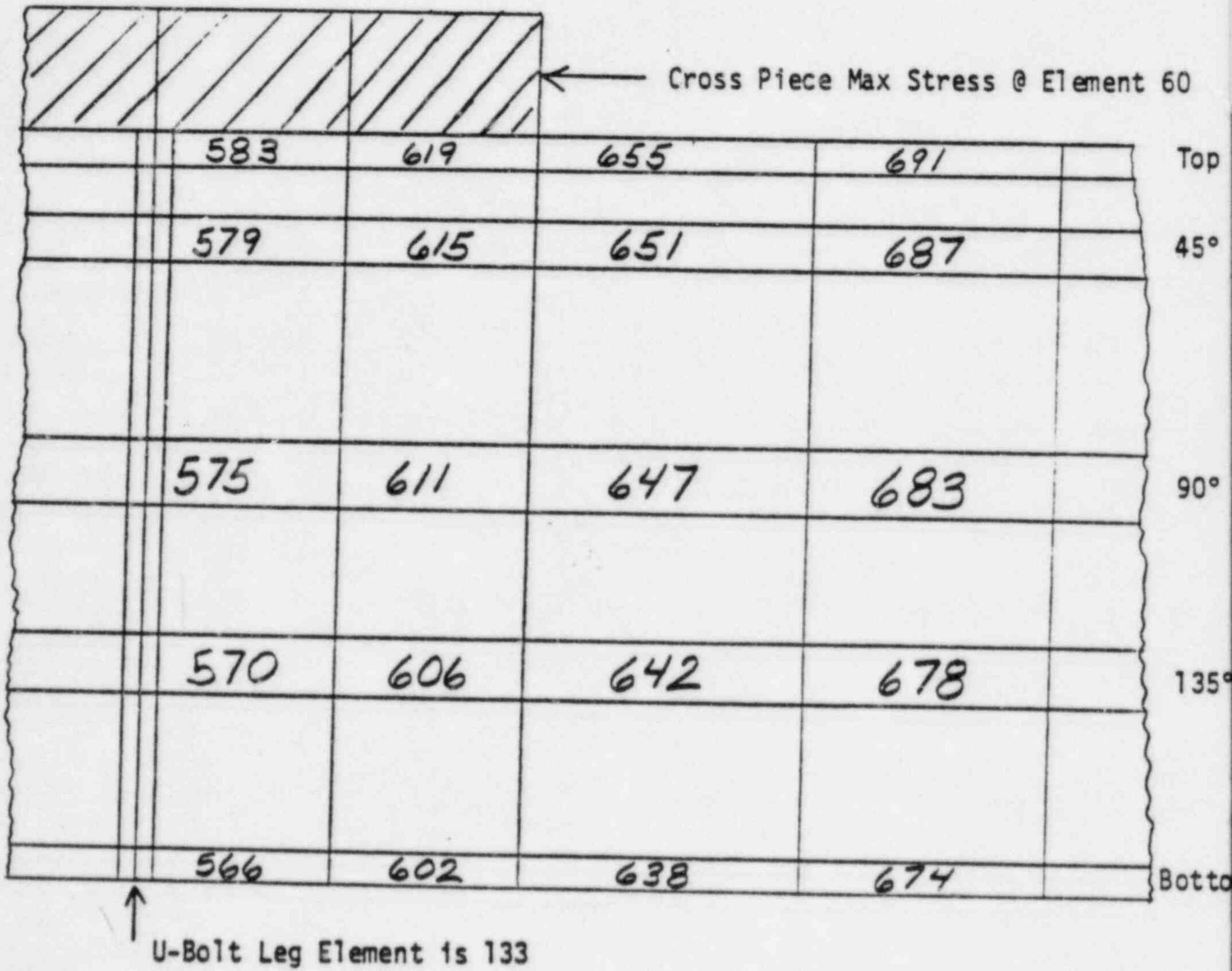
10" Sch 80

Partial Preload + Thermal + Pressure + Push

Analysis Element	Max Longitudinal Stress lb./in^2	Max Circum. Stress lb./in^2	Max Stress Intensity lb./in^2
482	-14810	21520	21530
518	-14960	21620	21630
554	-14370	20950	20980
590	-9178	17110	17140
626	-3606	12800	15012
478	-3821	9266	9285
514	-3689	9056	9111
550	-3482	8781	9082
586	-3105	8071	9269
622	-2455	6777	8995
474	2901	9358	9382
510	2821	9354	9385
546	2701	9322	9378
582	2379	9241	9324
618	1684	8978	9096
469	2114	5751	5751
505	1717	5889	5893
541	1505	5954	5957
577	1479	5967	5970
613	1390	5810	5811
465	3122	7825	7827
501	2576	7970	7975
537	2328	8071	8080
573	2440	8410	8425
609	2056	8076	8223
7·(Cross Pc.)	-518	-2546	3139
861 (U-Bolt)	1294	——	——

32" MAIN STEAM

STRESS SUMMARY TABLE ELEMENT DIAGRAM



NOTE: Not to scale, for relative location of stress summary elements onl

SIDE VIEW OF PIPE

32" Main Stream

Preload

Analysis Element	Max Longitudinal Stress lb./in^2	Max Circum. Stress lb./in^2	Max Stress Intensity lb./in^2
583	-2966	-4859	4859
619	-3412	-4918	4921
655	-2568	-4116	4140
691	-1092	-2717	2733
579	668	-1147	1148
615	633	-1113	1126
651	548	-1028	1066
687	390	-863	947
575	1001	-1154	1154
611	989	-1155	1156
647	957	-1160	1161
683	891	-1166	1168
570	-1170	-1445	1455
606	-650	-1134	1148
642	-209	-758	761
678	-120	-419	697
566	-1074	-1208	1209
602	-702	-1009	1011
638	-492	-827	830
674	-432	-691	713
60 (Cross Pc.)	-24	1242	1255
133 (U-Bolt)	1017	——	——

32" Main Steam

Preload + Thermal

Analysis Element	Max Longitudinal Stress lb./in^2	Max Circum. Stress lb./in^2	Max Stress Intensity lb./in^2
583	-12786	-21306	21306
619	-14810	-21570	21582
655	-10966	-17917	18030
691	-4249	-11542	11618
579	3664	-5991	5997
615	3487	-5833	5880
651	3050	-5439	5575
687	2227	-4660	4957
575	3469	-5418	5425
611	3420	-5235	5291
647	3295	-4865	4998
683	3030	-4348	4545
570	-4082	-4942	4945
606	-2508	-4030	4042
642	-996	-2841	2870
678	-592	-1733	1792
566	-3609	-4397	4397
602	-2117	-3518	3519
638	-867	-2459	2459
674	-1150	-1583	1710
60 (Cross Pc.)	3679	6092	6178
133 (U-Bolt)	4619	----	----

32" Main Steam

Preload + Thermal + Pressure

Analysis Element	Max Longitudinal Stress lb./in^2	Max Circum. Stress lb./in^2	Max Stress Intensity lb./in^2
583	-17149	33973	33960
619	-19883	34387	34411
655	-14676	32016	32296
691	-5588	27716	27887
579	5078	19409	19410
615	4830	19290	19299
651	4218	18981	19003
687	3066	18337	18371
575	3934	18018	18019
611	3971	17973	17973
647	4030	17993	17993
683	3901	18028	18028
570	-5105	13159	13159
606	-3058	13166	13166
642	-1101	13599	13684
678	-851	14439	15290
566	-4972	12830	12830
602	-2959	12834	12834
638	1187	13300	14160
674	-1526	14264	15790
60 (Cross Pc.)	3679	7979	8076
133 (U-Bolt)	6251	----	----

32" Main Steam

Preload + Thermal + Pressure + Push

Analysis Element	Max Longitudinal Stress lb./in^2	Max Circum. Stress lb./in^2	Max Stress Intensity lb./in^2
583	-26304	46413	46413
619	-31915	47170	47223
655	-22391	43296	43916
691	-7224	36269	36633
579	6752	22707	22713
615	6383	22517	22559
651	5477	22021	22129
687	3788	20994	21176
575	3255	20452	20453
611	3218	20454	20456
647	3116	20473	20478
683	2892	20530	20544
570	2291	15172	15172
606	1594	15184	15186
642	954	15344	15346
678	655	15604	15606
566	3497	16896	16896
602	2861	16907	16907
638	2341	17091	17091
674	2182	17471	17471
60 (Cross Pc.)	8924	3865	8926
133 (U-Bolt)	2048	——	——

APPENDIX I

FOR OFFICE AND ENGINEERING USE ONLY

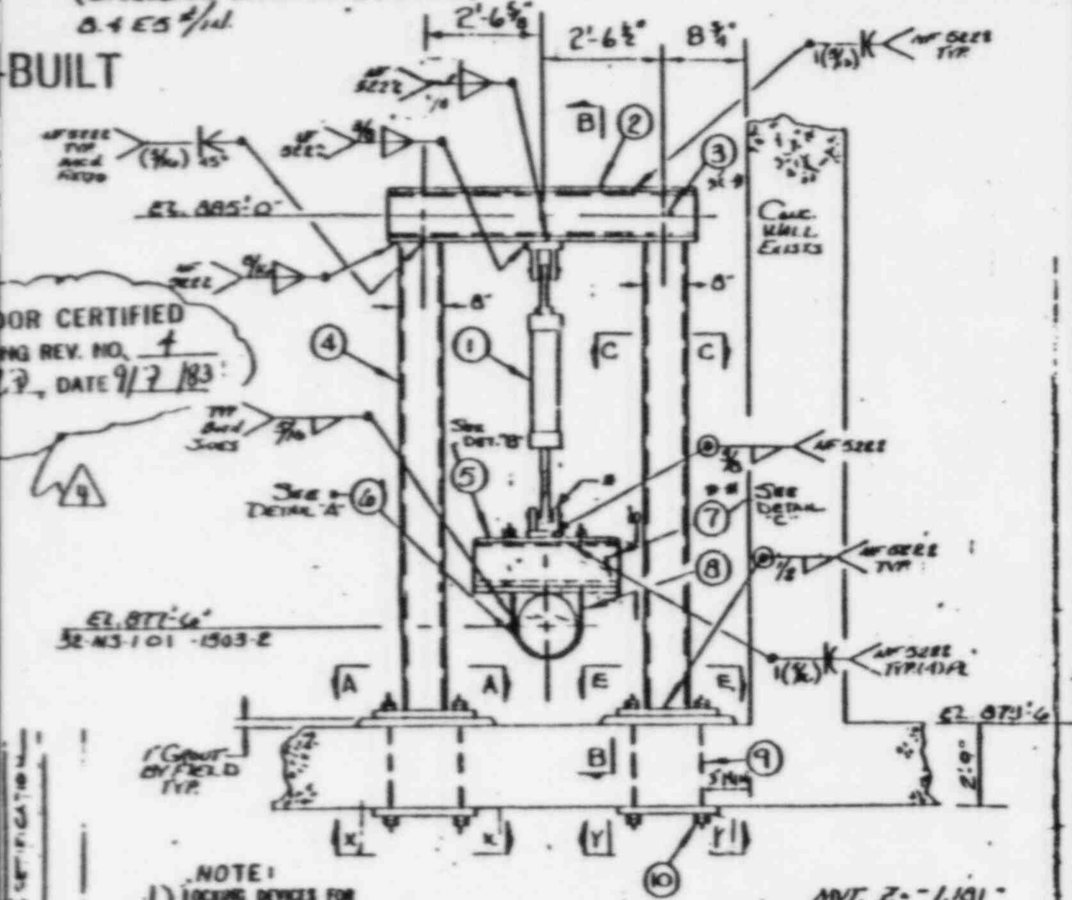
32"φ PIPE

BLUELINE *20 JULY*

Note: Structure Stiffness (Excluding Baseplate & Bolts) = 8.4 E5 #/in.

AS-BUILT

VENDOR CERTIFIED
DRAWING REV. NO. *4*
..., DATE *9/7/83*



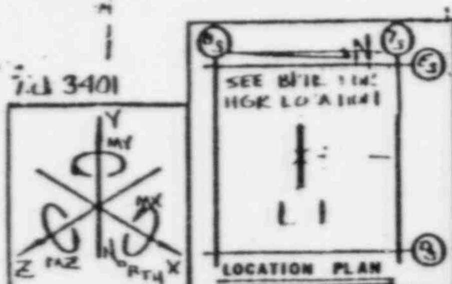
	MATERIALS & OPERATIONS	QUAN.	SHIP.		PBS	L	CSS	PRIM.	SEC.	AISC
	SEISMIC SWAY STRUT ASSEMBLY CONSISTING OF:	ONE								
1	Fig. 211N, #8 Sway Strut Assembly, Opt. #2 W = 1'- 6 3/8", Load = 39912#	1					x			
2	1 1/2" x 8" Carbon Steel (SA515 GR.65 or SA-36) Plate, 5'- 11" long, TW = 483#	2				x		x		
3	1/2" x 8" x 8" Structural Tubing (A500-76 GR.B) 5'- 11" long, TW = 280#	1				x		x		
4	3/8" x 4" x 8" Structural Tubing (A500-76 GR.B) 10'- 8 3/4" lg. TW = 500#	2				x		x		
5	1" x 6" Carbon Steel (SA515 GR.65 or SA-36) Plate, 3'- 9" lg TW= 153#	2				x		x		
6	1" x 3" Carbon Steel (SA515 GR.65) Plate, 4'- 3 13/16" long/DETAIL "A" TW = 44#	1								
7	1/2" x 6" x 8" Structural Tubing (A500-76 GR.B) 3'- 9" long, TW = 152#	1				x		x		
8	Fig. 137SN, U-Bolt, Carbon Steel (SA-36) A = 2 3/4", B = 2'- 10 1/8", C = 3'- 0 7/8", D = 2'- 7 7/8", E = 1'- 8", Developed Lgth. 10'- 1 11/16", w/Hex Nuts	1					x			
9	1 1/8" - 8UN x 3'- 9" Round Rod (SA193 GR.B7) w/ 8" Threads Both Ends	8					✓			
10	FGN-1 1/8" HEX NUTS	32					✓			
11	Carbon Steel (SA515 GR.65 or SA-36) Plate/ SECTION "A-A" 2" THK	1			x			x		
12	1" x 8" Carbon Steel (SA515 GR.65 or SA-36) Plate, 10'- 8 3/4" long, TW = 1167#	4				x		x		
13	Carbon Steel (SA515 GR.65 or SA-36) Plate/ per SECTION E-E 2" THK	1			x			x		
14	FHN-2 3/4" HVY. HEX NUTS	2					x			
15	2" C.S. R. PER SEC. "X-X" (SA36 OR SA515 GR 65)	1			x			\		
16	2" C.S. R. PER SEC. "Y-Y" (SA36 OR SA515 GR. 65)	1			x			\		
	CONT ON SHEET 4 OF 4									
	MK #MS-1-001-005-S72R									
	Apply Carbo-Zinc #11 to above mat'l except th'ds which shall be treated with a rust preventative. Approved By: DG Date: 11/7/80	QUAN.	SHIP.		PBS	L	CSS	PRIM.	SEC.	AISC.

FOR MATERIALS AND OPERATIONS SEE SKETCH NO. ______ SHEET __ OF __

NOTE:
1) LOCKING DEVICES FOR HIGH STRENGTH BOLTS ARE NOT REQUIRED PER DCA 7607
2) * FIELD TRIM BASE ON BRACKET TO A-3/4" x 11"
3) ITEM #3, 7 MAY NOT WARP

MVT. Z = -1.181"

T.I. 3401



THIRD PARTY INSPECTION YES ☒ NO ☐
CODE CLASS: *ASME III-2*

Pipe ...
I.P.D. Iso. *MS-1-30-16 R4 L1*
Data Point *1820 / ...*
Pipe Mat'l. *SA-155 KCF 70 Class 1*
Insul. *2.33"* Bldg. *SB*

BROWN & ROOT, INC.
ENGINEERS & CONSTRUCTORS

CONDITIONS	Fx	Fy	Fz	Mx	My	Mz
DESIGN	—	—	—	—	—	—
NORMAL & UPSET	—	[illegible]	—	—	—	—
EMERGENCY	—	[illegible]	—	—	—	—
FAULTED	—	—	—	—	—	—

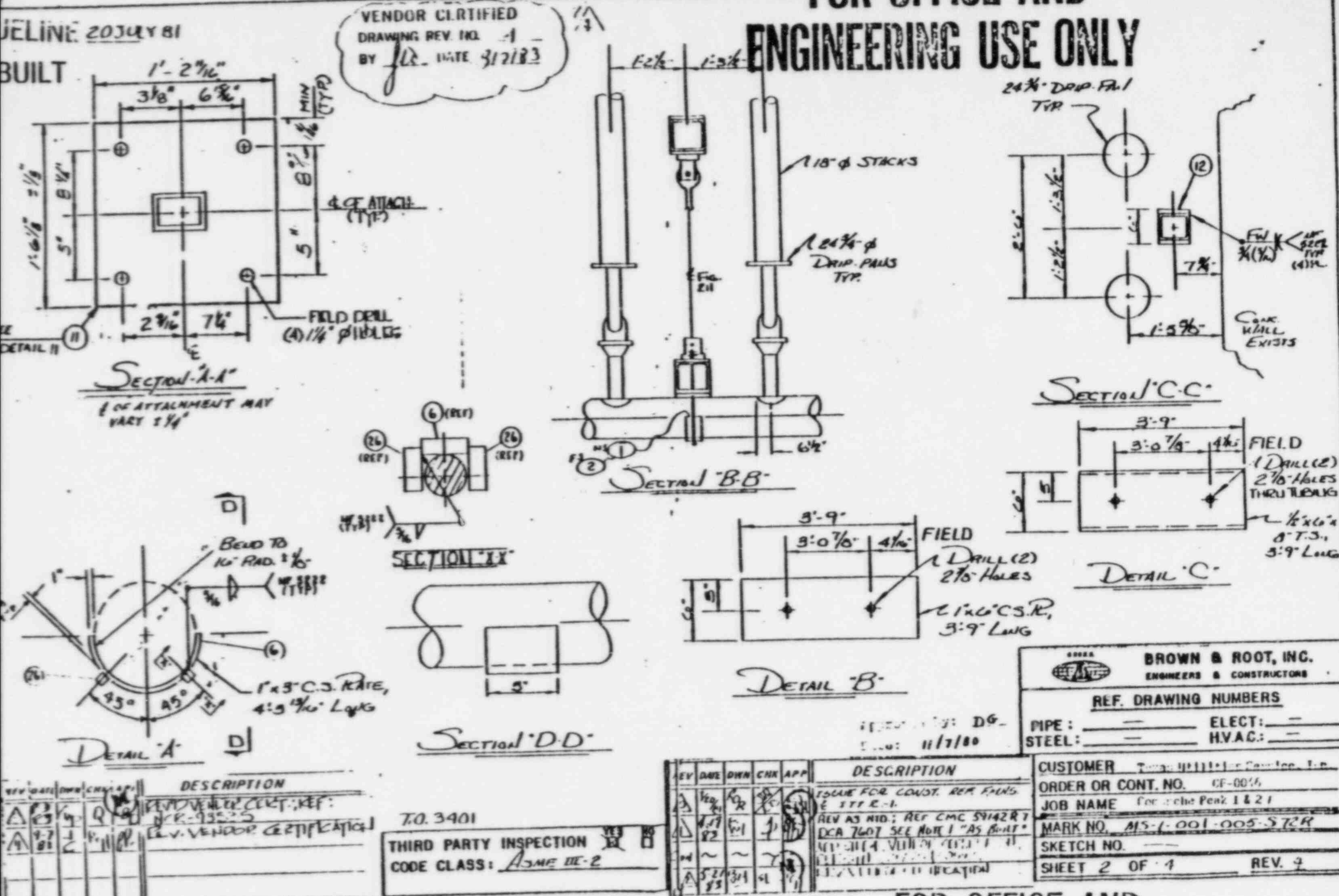
REF. DRAWING NUMBERS
PIPE: *M1-0600 R.5* ELECT: *E1-0603 R1*
STEEL: *S1-0614 R3* H.V.A.C.: *M1-0555 R4*

REV	DATE	DWN	CHK	APP	DESCRIPTION
1	11/20/81	RO2	[illegible]	M	ISSUE FOR CONST. REF. FAILURE & ... R.1
2	4-19-83	Kuy	[illegible]	M	REV AS NOTED; REF CMC 59142 R7 DCA 7607 SEE NOTE 1 "AS BUILT"
3	~	~	~	M	[illegible]
4	3-21-83	Kuy	SL	M	[illegible]

CUSTOMER Texas Utilities Services, Inc.
ORDER OR CONT. NO. CP-0046
JOB NAME Comanche Peak 1 & 2
MARK NO. *MS-1-001-005-S72R*
SKETCH NO. —
SHEET *3* OF *4* REV. *4*

FOR OFFICE AND

FOR OFFICE AND ENGINEERING USE ONLY



FOR OFFICE AND ENGINEERING USE ONLY

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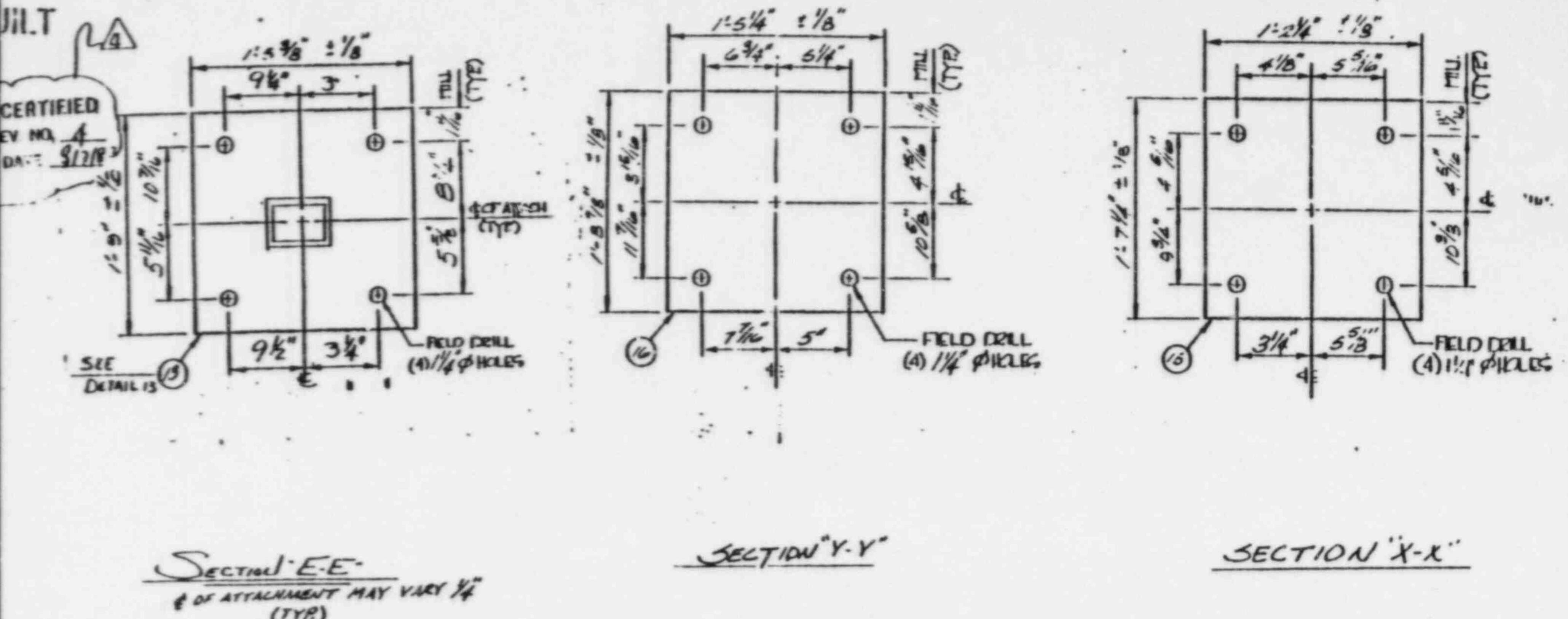
20 JUL 82



DG
11/7/80

DATE	DWN	CHK	APP	DESCRIPTION
8-? 83				REV'D VENDOR CERT. REF: NCE-9335S
9-? 83				Rev. VENDOR CERTIFICATION

T.O. 3401

THIRD PARTY INSPECTION YES ☒ NO ☐

CODE CLASS: ASME III-2

REV	DATE	DWN	CHK	APP	DESCRIPTION
1					ISSUE FOR CONST. REF. ... E-1
2				M	REV AS NTD: REF CMC 50142 ...
	~	~	~		[illegible]
4	5/83		SL		[illegible]

BROWN & ROOT, INC.
ENGINEERS & CONSTRUCTORS

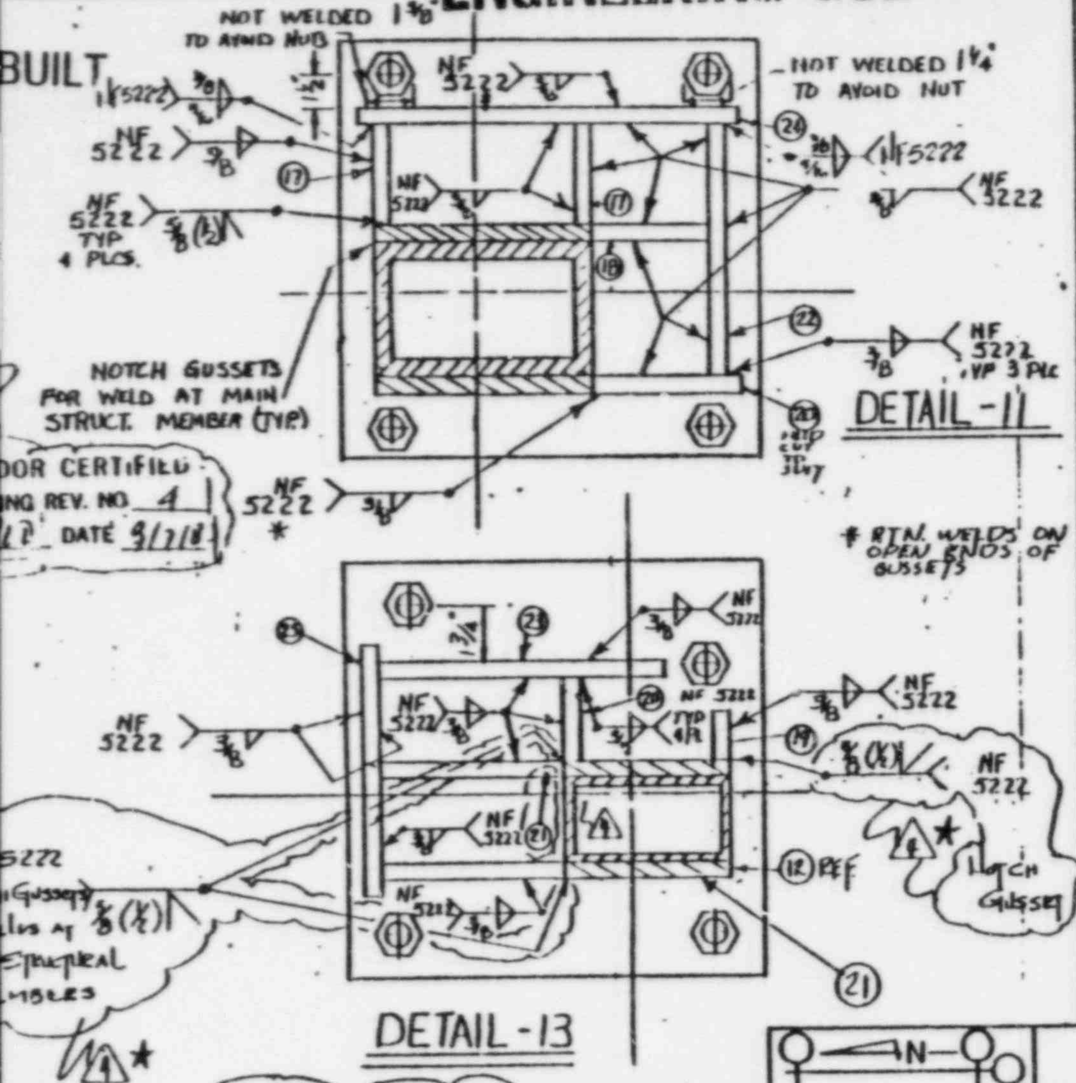
REF. DRAWING NUMBERS

PIPE: — ELECT: —
STEEL: — H.V.A.C.: —

CUSTOMER Texas Utilities Services, Inc.
ORDER OR CONT. NO. CP-0046
JOB NAME Comanche Peak 1 & 2
MARK NO. M?-1-001-005-S72R
SKETCH NO. —
SHEET 3 OF 4 REV. 4

FOR OFFICE AND ENGINEERING USE ONLY

FOR OFFICE AND ENGINEERING USE ONLY



ITEM NO.	MATERIALS & OPERATIONS	QUAN.	SHIP.		PBS	L	CSS	PRIM.	SEC.	AISC
17	PL 1" x 4" x 3" LG (SA-36/SA 515 GR 65)	2			X			X		
18	PL 1" x 3" x 3 1/2" LG (SA 36 / SA 515 GR 65)	1			X			X		
19	PL 1" x 4" x 4" LG (SA-36 / SA 515 GR 65)	1			X			X		
20	PL 1" x 4" x 5 1/2" LG (SA-36 / SA 515 GR 65)	2			X			X		
21	PL 1" x 4" x 6" LG (SA-36 / SA 515 GR 65)	2			X			X		
22	PL 1" x 4" x 8 1/4" LG (SA-36 / SA 515 GR 65)	1			X			X		
23	PL 1" x 4" x 10 1/4" LG (SA-36 / SA 515 GR 65)	1			X			X		
24	PL 1" x 4" x 1'-1 1/2" LG (SA-36 / SA 515 GR 65)	1			X			X		
25	PL 1" x 4" x 1'-2" LG (SA-36 / SA 515 GR 65)	1			X			X		
26	PL 1" x 2 1/2" x 2 1/2" (SA-36)	4								

REV	DATE	DWN	CHK	APP	DESCRIPTION
4	9-7-83				REV. VENDOR CERTIFICATION

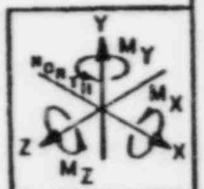
FOR MATERIALS AND OPERATIONS SEE SKETCH NO. ______ SHEET ___ OF ___

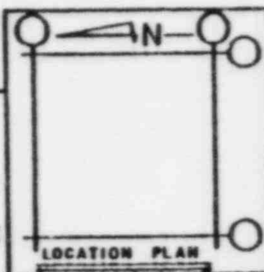
BROWN & ROOT, INC.
ENGINEERS & CONSTRUCTORS

REF. DRAWING NUMBERS

PIPE: ______ ELECT: ______
STEEL: ______ H.V.A.C.: ______

CONDITIONS	Fx	Fy	Fz	Mx	My	Mz
DESIGN	—	—	—	—	—	—
NORMAL & UPSET	—	—	—	—	—	—
EMERGENCY	—	—	—	—	—	—
FAULTED	—	—	—	—	—	—





THIRD PARTY INSPECTION YES NO
CODE CLASS: ASME III-2

REV	DATE	DWN	CHK	APP	DESCRIPTION
0					ISSUED FOR AS-BUILT; REF CMC 59142
1					[illegible]
2	5-27-83				REV VENDOR CERTIFICATION
3	8-2-83				[illegible]

CUSTOMER Texas Utilities Service, Inc.
ORDER OR CONT. NO. CP-0046
JOB NAME Comanche Peak 1 & 2
MARK NO. AC-1-001-005-S22R
SKETCH NO. —
SHEET 1 OF 1 REV. 4

G.H.I. Iso. ______
I.P.D. Iso. ______
Data Point ______
Pipe Mat'l. ______
Insul. ______ Bldg. ______

TD. # 3401

FOR OFFICE AND ENGINEERING USE ONLY

4" ϕ PIPE

FOR OFFICE AND ENGINEERING USE ONLY

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BLUELINE: TIDEC-B11

FIELD REVIEW OF SUPPORT DESIGN
AS BUILT CONDITION CPPA - 3211
DATE: REVIEWED BY: W. WILLIAMS

AS-BUILT
VENDOR CERTIFIED
DRAWING REV. NO 7

ITEM NO	NO REQ'D	DESCRIPTION	WT.	ASME OR ASTM	MIC.
1	1	TS 6"x6"x1/2"x2'-9" LG.		A500GR.B	
2	1	TS 6"x6"x1/2"x3'-2" LG.		A500GR.B	
3	1	1" C.S. PL. (SECTION "C-C")			
4	8	DSA-34-10 CONC. ANCHORS			
8	1	1/4" C.S. PL. x 7"x7" LG.			
9	1	1" C.S. PL. (SECTION "B-B")			
10	2	XRB-10 REAR BKT.			
11	1	1/4" C.S. PL. 3"x8" LG.			
12	1	PHS-040 U-BOLT			
14	1	3/4" C.S. PL. x 6"x8" LG.		SA36	
15	1	1/2" C.S. PL. x 6"x8 1/2" LG.		SA36	
16	1	SRF-10-50 STRUT			

REV	DESCRIPTION
	REVD VENDOR CERT.
	REVD VENDOR CERT.

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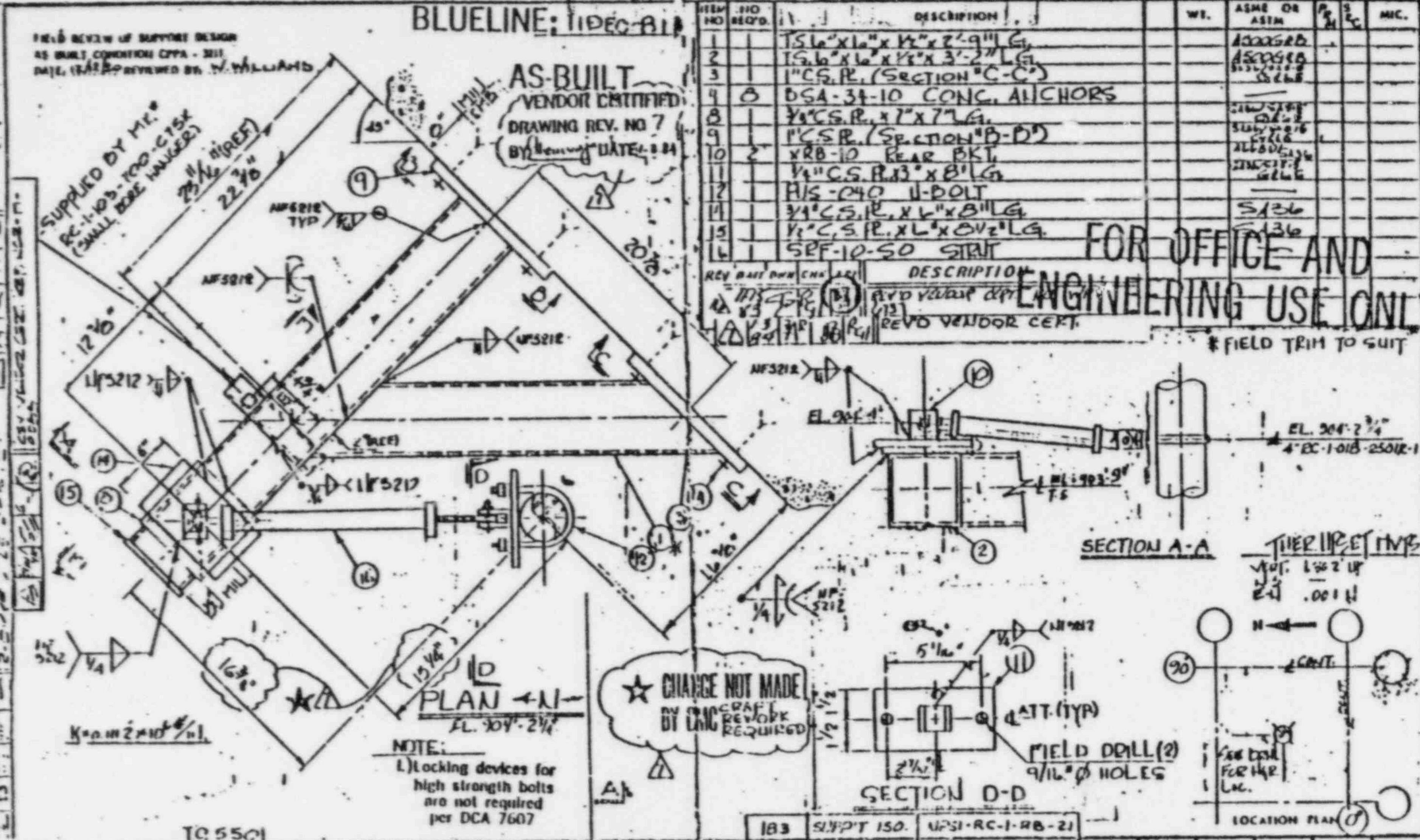
* FIELD TRIM TO SUIT



NOTE:
1) Locking devices for high strength bolts are not required per DCA 7607

183	SUPP'T ISO.	UPSI-RC-1-RB-21

REFERENCE DRAWINGS: ISOMETRIC RC-1-RB-021 REV 2; PIPING 2323-MI-0514; ELECTRICAL 2323-E1-0503 REV 4; CODE/CLASS: III/1 REV 6; FAB. ISOMETRIC RC-1-RB-21 REV 3; STRUCTURAL 2323-S1-0533; H.V.A.C. 2323-MI-0558 REV 2 / REV 4; ZONE

DRAWN	DATE	CHK'D	DATE	APPV'D	DATE
RG	2.25.81	AY	3/2/81	E.L.	3.1.81

P.O. NO. CP-0046 A.1 MFG. REL.

OWNER: TEXAS UTILITIES SERVICES INC.
PROJECT: COMANCHE PEAK UNITS NO. 1 & 2
ENGINEER: GIBBS & HILL INC.

Brown & Root, Inc.
HOUSTON, TEXAS

PRODUCTION ORDER | SERIAL NUMBER | SHEET 1 of 2

1304 | MK. NO RC-1-018-016-C81R | REV 7

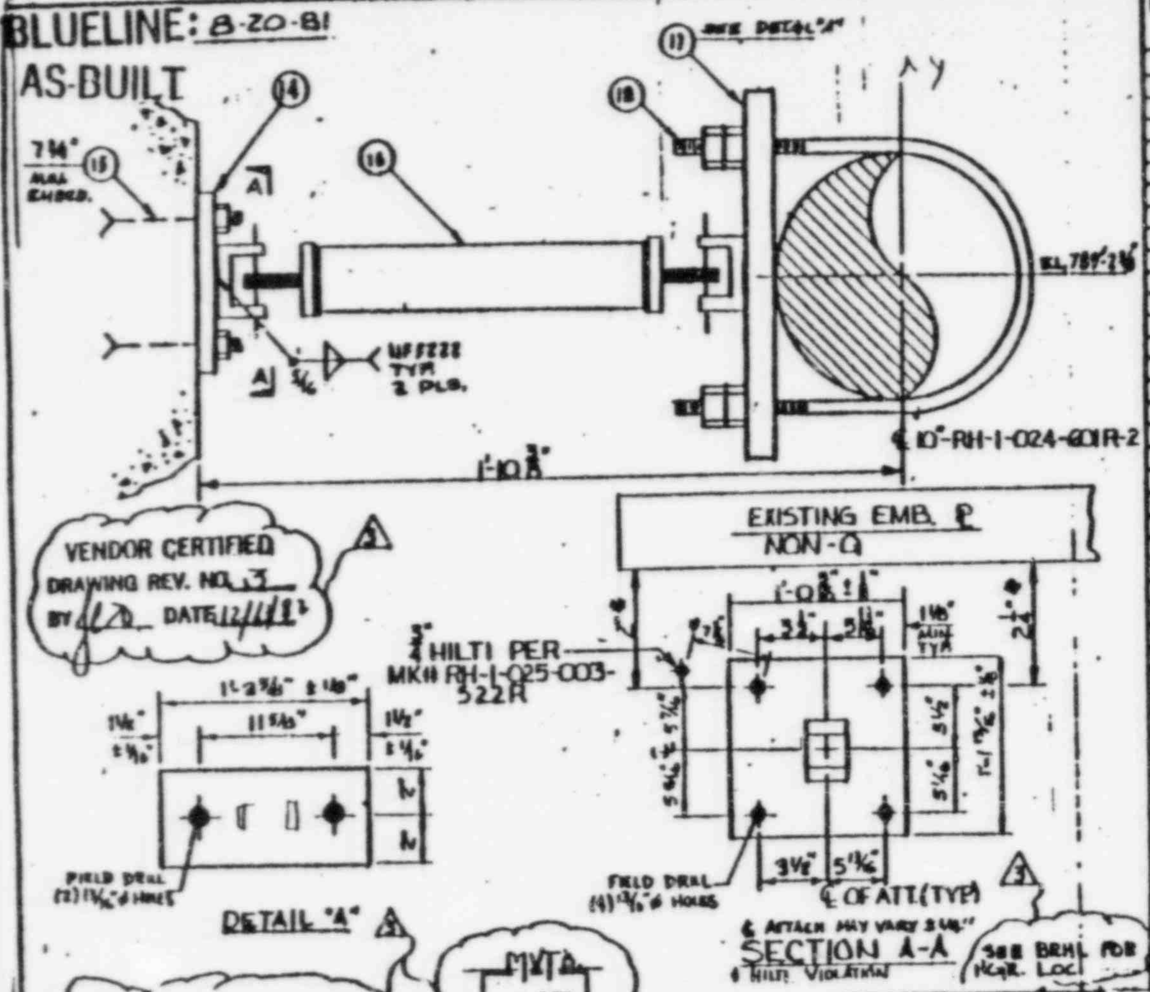
TC 5501

-95-

10" Ø PIPE

BLUELINE: 8-20-81

AS-BUILT



ITEM NO.	MATERIALS & OPERATIONS	QUAN	SHIP
14	3/4" C.S. PLATE PER SECT. A-A (24996066/34-36)	1	
15	3/4" x 10" HILTI-KWIK CONS. ANCHOR	4	
16	SNF-11-BA SWAY STRUT C-C = 01'8 5/8"	1	
17	1 1/4" THK C.S. PLATE PER DETAIL "A" (34-36)	1	
18	PBS-180 U-BOLT	1	

MARK # RH-1-024-007-S22R

PAINT: CARBO ZINC 11

FOR ENGINEERING & OFFICE USE ONLY

CPSES

FOR MATERIALS AND OPERATIONS SEE SKETCH NO. ___ SHEET ___ OF ___

BROWN & ROOT, INC.
ENGINEERS & CONSTRUCTORS

CONDITIONS	Fx	Fy	Fz	Mx	My
DESIGN / NORMAL & UPSET					
EMERGENCY					
FAULTED					

REF. DRAWING NUMBERS
PIPE: M1-0616 R-11 ELECT: [illegible]
STEEL: S1-0600 R-17 H.V.A.C.: [illegible]

BRHL ISO. BH-1-SB-09 R-2
T.P.D. ISO. RH-1-SB-09-R-1
Data Point 1136 / HAB-1-WA-RD
Pipe Mat'l. [illegible]
Insul. 1 1/2" Bldg. 8

T.O. 5801

LOCATION PLAN

THIRD PARTY INSPECTION
CODE CLASS: ASME III-2

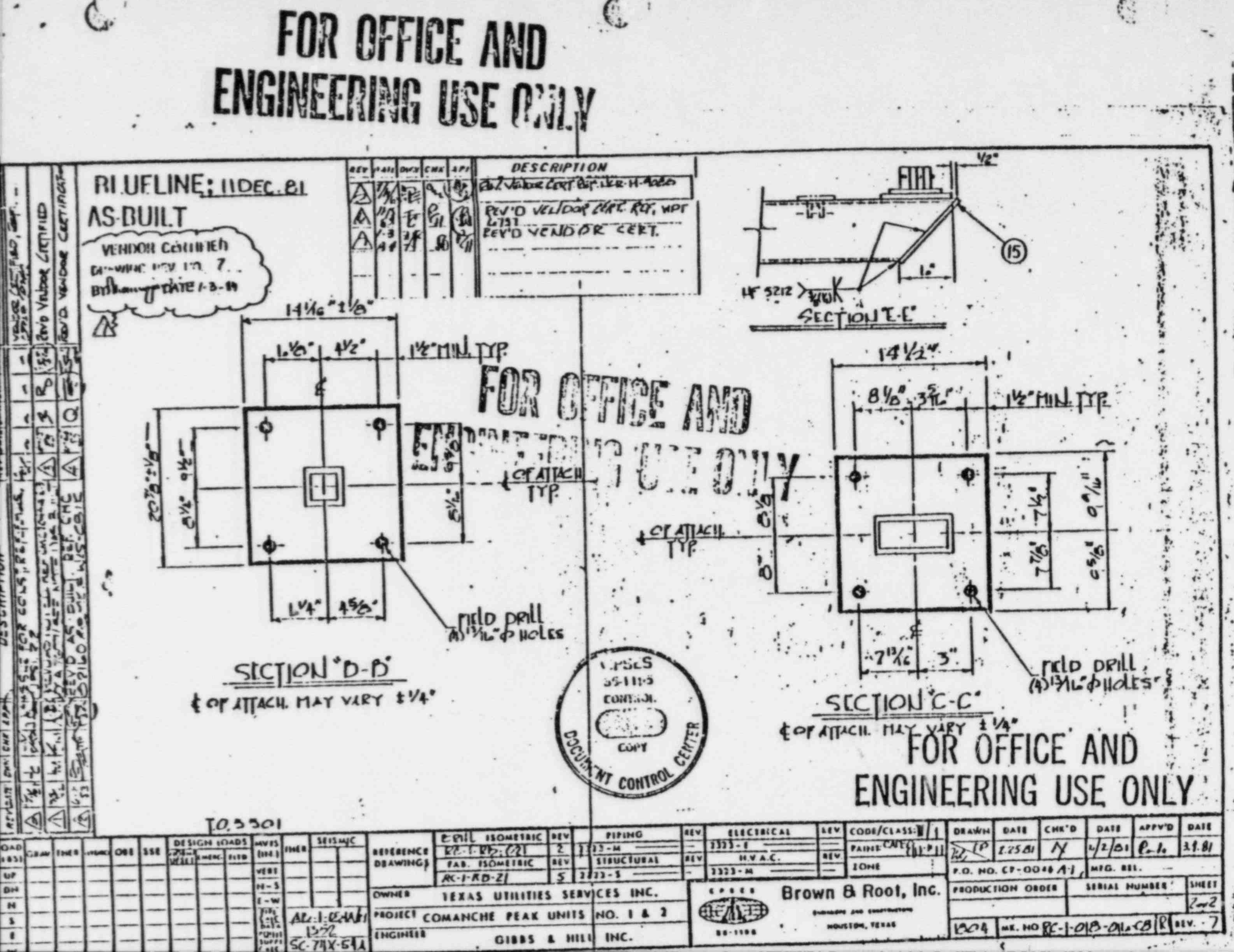
REV	DATE	DWN	CHK	APP	DESCRIPTION
0					ISSUE FOR CONST. PER CPPA 3000
1					REVD AS NOTED REF. CMHS SEE MPT. DEL. SHT. 2
2					REVD AS NOTED REF. CMC 63088, NCA, NOTE SEE NOTE 5 AS BUILT
3					VENDOR CERTIFICATION REF. GTN # 67332

CUSTOMER Texas Utilities Services, Inc.
ORDER OR CONT. NO. CP-0046
JOB NAME Comanche Peak 1 & 2
MARK NO. RH-1-024-007-S22R
SKETCH NO.
SHEET 1 OF 1 REV. 3

NOTES
1) BY ISSUE OF REV. 1 OF THIS DRAWING, THE FOLLOWING DOCUMENTS ARE VOIDED: CMC 93011 R-4

NOTE:
5) Locking devices for high strength bolts are not required per DCA 7607

FOR OFFICE AND ENGINEERING USE ONLY


FOR OFFICE AND
ENGINEERING USE ONLY
BLUELINE: 11DEC.81
AS-BUILT
DESCRIPTION
SECTION "E-E"
SECTION "D-D"
℄ OF ATTACH. MAY VARY ±1/4"
FIELD DRILL
(4) 13/16" ⌀ HOLES
℄ OF ATTACH TYP.
1 1/2" MIN. TYP.
14 1/2"
SECTION "C-C"
℄ OF ATTACH. MAY VARY ±1/4"
DOCUMENT CONTROL CENTER
COPY
FOR OFFICE AND
ENGINEERING USE ONLY
REFERENCE DRAWINGS
OWNER TEXAS UTILITIES SERVICES INC.
PROJECT COMANCHE PEAK UNITS NO. 1 & 2
ENGINEER GIBBS & HILL INC.
Brown & Root, Inc.
HOUSTON, TEXAS
DRAWN
DATE
CHK'D
APPV'D
PRODUCTION ORDER
SERIAL NUMBER
SHEET
2 of 2
MK. NO RC-1-018-016-C8R
REV. 7

APPENDIX II

Following are NASTRAN element descriptions for the three finite element types used in the finite element analysis of the U-bolt pipe support configuration. These elements include (a) the "beam" element, (b) the "QUAD4" plate element, and (c) the "gap" element.

1. "BEAM ELEMENT"

The "beam" element is a straight element which connects two grid points, and which has extensional and torsional stiffness, bending stiffness and transverse shear flexibility in two perpendicular directions.

2. QUAD4 PLATE ELEMENT

The QUAD4 element has a general nonplaner quadrilateral shape with four straight edges and four corners that are connected to grid points. Connections are made to three translational and two rotational degrees of freedom at each corner of the element. The QUAD4 element is a modified isoparameric element. It may be used as a membrane element, a bending element, or as a combined membrane/bending element. In the latter case, the user may account for coupling between membrane and bending properties. Its membrane part uses reduced order integration for in-plane shear. Its bending part has been designed to give nearly exact results when the curvature varies linearly over the surface of the element.

The QUAD4 element behaves well when its shape is irregular. There is no aspect ratio limit. Good results have been obtained with skew angles up to 45 degrees. The corner points are not required to lie in the same plane.

3. GAP ELEMENT

The gap element is used to define a gap or frictional element of the structure model. It is nonlinear element which may have

compression and shear forces when the gap is closed. Additional properties include axial stiffness, trasverse stiffness when the surfaces are in contact, and an initial preload.

Figures 3 through 6 show the gap element and the force-displacement curves used in the stiffness and force computations for this element.

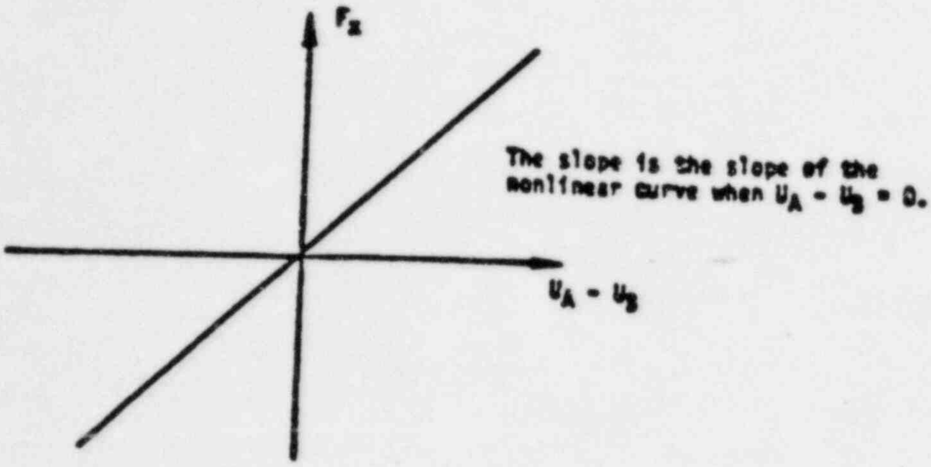
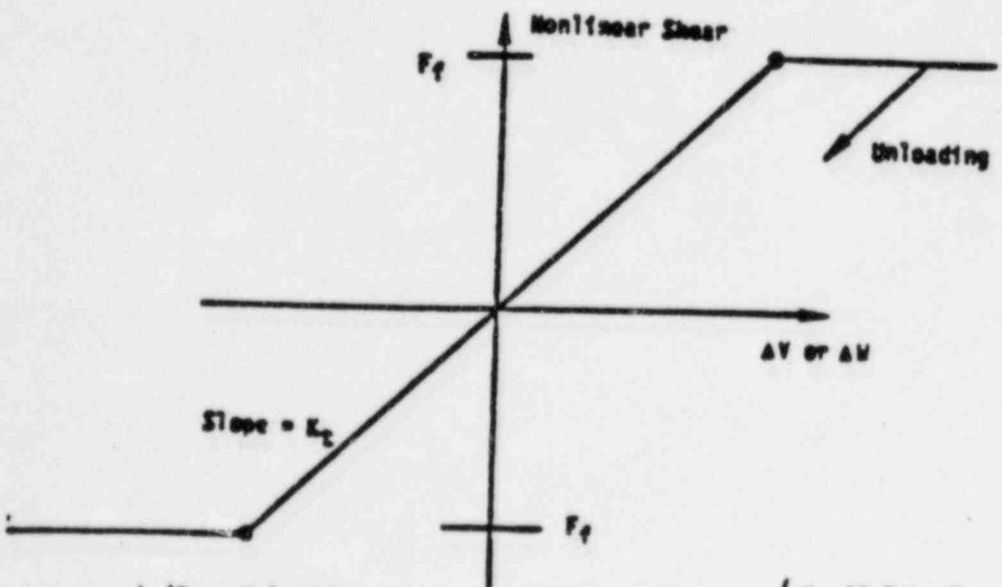


FIGURE 3: Force-Deflection Curve for Linear Analysis



where $F_f = \begin{cases} \mu(F_x - F_0) & \text{if } F_x > F_0 \\ 0 & \text{if } F_x < F_0 \end{cases}$ slope for <u>linear</u> analysis $= \begin{cases} K_t & \text{if } F_x \geq F_0 \\ 0 & \text{if } F_x < F_0 \end{cases}$

FIGURE 4: Shear Force for the GAP Element

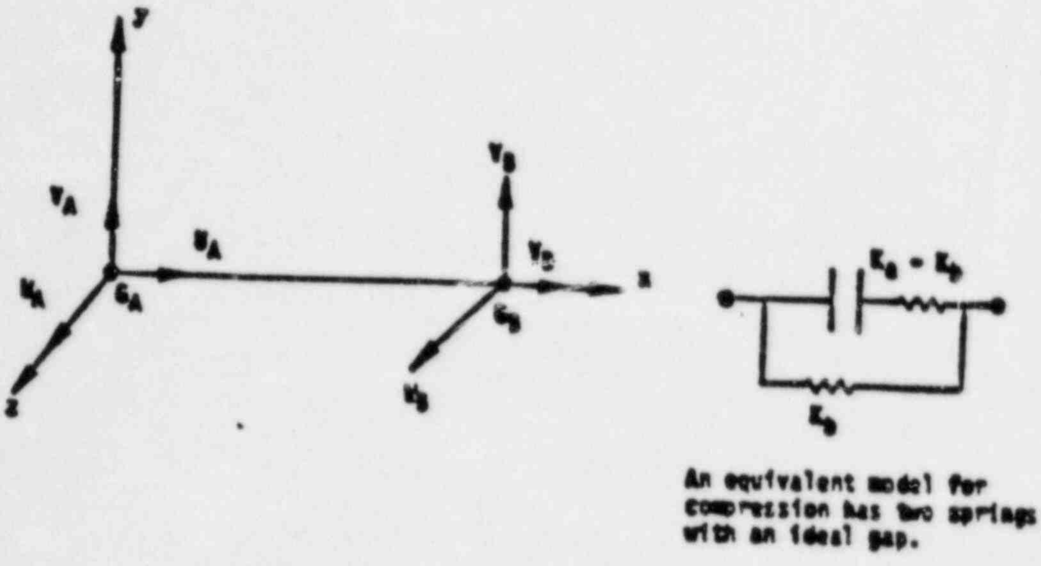


FIGURE 5: The GAP Element

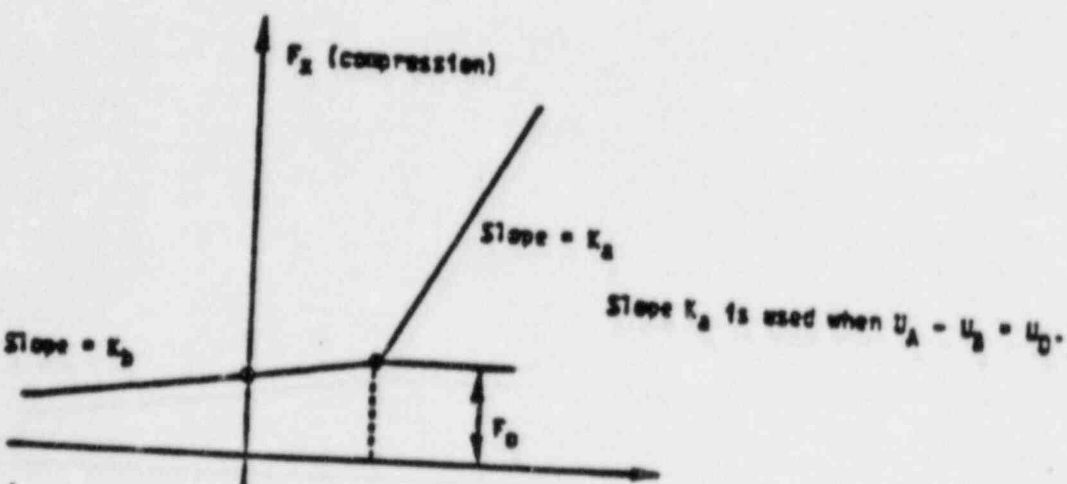


FIGURE 6: Force-Deflection Curve for Nonlinear Analysis.