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ORIGINAL

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Discussion :

between :

CYGNA ENERGY SERVICES :

and :

TEXAS UTILITIES GENERATING COMPANY :

and :

EBASCO SERVICES, INC. :

- x

Ebasco Services, Inc.
Two World Trade Center
New York, New York 10048

September 13, 1984
9:30 o'clock a.m.

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Use attached dist. list

IN ATTENDANCE:

For Texas Utilities Generating Co.:

John Finneran
Project Pipe Support Engineer

For Ebasco Services, Inc.:

Dr. Robert C. Iotti
Chief Engineer

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Manager, Nuclear Licensing Department

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Project Engineer

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Boston, Massachusetts

For Citizens Association for Sound Energy:

(Telephonically)
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P R O C E E D I N G S

DR. IOTTI: Let us begin.

The purpose of this meeting is to have a discussion between CYGNA and the Applicant in regard to questions CYGNA has asked to help them come to a conclusion with regard to the Applicant's affidavit on the cinching of U-bolts.

MS. ELLIS: Was this this August 23rd letter, Dr. Iotti?

DR. IOTTI: You are correct. That is the August 23rd letter which has a 84042.015 identifier on it.

MS. ELLIS: Right. Thank you.

DR. IOTTI: Okay.

MS. ELLIS: I will be quiet except to yell if I can't hear probably.

DR. IOTTI: Okay. Very fine.

So we will proceed with our discussions.

Miss Williams, do we want to agree on a particular format or can we essentially have a free format?

I was going to suggest that we take the questions as they appear in your letter in that order and that the Applicant try to satisfy your concerns on each of those questions.

Dr. LaPay of Westinghouse is still not here. A lot of the work that led to the attachments to the affidavit was performed by Westinghouse.

So we may have to wait for some of the questions until he appears.

But on Question 1 CYGNA asked the Applicant to provide a detailed numerical breakdown on how the stresses in Tables H, I, N and O of the affidavit were obtained.

What I propose to do now is take them through a particular example.

I happened to have worked out for the four-inch Schedule 160 pipe the example that would lead them to the numbers that appear in those tables.

So with that preamble, maybe the best bet is for me to refer you to the Table on page 58 of

Attachment 3 to the affidavit.

Do you have that? That is Attachment 3.

DR. BJORKMAN: Right.

DR. IOTTI: You are there.

DR. BJORKMAN: I recognize that the stresses in Table H did come from page 59 of Attachment 3, but my --

DR. IOTTI: I know. It's not easy to get --

MS. ELLIS: Whoever was talking there I can't hear. I'm sorry.

DR. IOTTI: Mrs. Ellis, that was Dr. Bjorkman of CYGNA. He was just mumbling to himself.

MS. ELLIS: I'm sorry to have interrupted Dr. Bjorkman.

DR. BJORKMAN: What I wanted to say was I recognize that these stresses in Table H do come from page 59 of Attachment 3.

But the question really is where do the stresses then on page 59 of Attachment 3 come from.

DR. IOTTI: That is what I'm proposing to do right now.

It is not easy. In fact, it's well nigh impossible to go from one page to another without going through several iterations back and forth. So this is going to take some patience.

But we will start from the stresses that appear on page 58 of Attachment 3 to the affidavit.

These stresses are for a four-inch Schedule 160 pipe and they reflect stresses on the inside surfaces and the outside surfaces of that pipe at a particular location, essentially right beneath the backing plate.

The first point that is made on that Table is that the longitudinal stresses and circumferential stresses as predicted by the Finite Element Analysis are essentially coincident with the direction of the principal stresses.

There is minor differences. So, as far as this discussion is concerned, we can talk about principal stresses or we can talk about longitudinal and circumferential stresses in one and the same breath.

Secondly, just for point of clarity, anything that is preceded by a negative sign in terms of stresses is defined as a compressive stress. Positive stresses are defined as tensile stresses.

First of all, Dr. Bjorkman, if you go to -- if you want to find a confirmation of the maximum circumferential stress, for instance, you can go to page 71 of Attachment 3 and you will find that the 44.79 ksi that appears as the maximum circumferential stress on the inside surface on page 58 is, in fact, the maximum circumferential stress in element 627. That's the first one.

DR. BJORKMAN: Correct.

DR. IOTTI: What you will have difficulty to find is the corresponding 10.49 ksi, which is the longitudinal stress because what the Table on page 71 prints out, it's only the maximum stresses and it doesn't distinguish whether they appear on the inside or on the outside surface.

So at that point it's impossible for an

outside person to follow the derivation unless led through it by people who have done it.

So I'm not surprised at your inability to get from one point to the other.

The 44.79 ksi tensile does appear -- circumferential stress does appear on the inside surface.

Incidentally, the corresponding maximum longitudinal stress, minus 26.65 ksi, which is compressive, occurs on the outside surface, and the corresponding inside surface longitudinal would only be 10.49.

Now, you cannot know this from that. You have to go the Finite Element Analysis output.

Something we can provide you separately, we have tabulated for the elements that have the high stresses the inside and outside surface stresses. Otherwise it becomes, again, very difficult.

So what you will see in a piece of paper that later on I can give you is for the pre-load, pre-load plus thermal, pre-load plus thermal plus pressure, pre-load plus thermal plus pressure plus the push for the four-inch, ten-inch Schedule 40, ten-inch Schedule 85 and thirty-two inch. We will print out the longitudinal circumferential maximum stresses both for the inside and for the outside surfaces because these are the figures that we will be discussing at all points in time.

Well, now, that's step one.

Now, step two is, of course, how do we get from that Table on page 58 to the next Table on page 58.

I think that is fairly straightforward.

The Table on page 58 at the top of the page only refers to stresses as printed out from the Finite Element Analysis. These are local effects.

We need to add to those local effects, as far as the longitudinal stresses, the stresses that are already present due to a pressure, longitudinal pressure stress.

So all you see is, for instance, the 10.49 ksi longitudinal has now been augmented by 4.8 ksi, which represents the longitudinal pressure stress, to 15.29.

Now, in addition though to the longitudinal pressure stress there are other longitudinal stresses that need to be added.

And now I have to -- you know, at this point you're kind of lost. There is a big gap between page 58, the last table on page 58, and the table on page 59.

First of all, there is a longitudinal stress which would come from an Equation 9, the piping moment stress, which has been estimated to be plus or minus 12.146 ksi. This has been estimated on the basis of maximum stresses that could occur at the elbow and working back to the stress that might appear in the straight portion of the piping system due to the maximum stresses at the elbows.

DR. BJORKMAN: Are those --

DR. IOTTI: Those are the ones --

DR. BJORKMAN: -- the stresses found on page 56?

DR. IOTTI: Those are the stresses that are found on page 56, correct.

And 12.146 is the Equation 9 piping moment stress at the pipe hander, and the affidavit tells you how that is derived.

For further reference, on page 57 there is also the secondary piping moment stresses, which is 22.49.

Now, to arrive at the stress intensities we have to construct a Mohr circle.

And I am showing Dr. Bjorkman a Mohr circle for the particular, in this instance is the inside surface, because what we need to do is keep separating the inside surface and the outside surface.

The principal stress in the Mohr circle which coincides with the circumferential stress is 44.79.

Now, the other principal stress just due to local effect plus the longitudinal pressure stress is 15.29. That happens to be inside the circle because it is still a tensile stress.

From that we have to subtract a minus 12.146, which is the Equation 9 moment stress, and also an additional minus 22.49 from that point, which is the Equation 12 piping moment stress.

That gives us a total negative or otherwise compressive stress along the longitudinal or the second principal stress of minus 19.346 which, when added to 44.79, should give us and does, in fact, give us the 64.14 ksi that appears as that total stress intensity on page 59.

DR. BJORKMAN: Okay. That is clear.

DR. IOTTI: Okay.

DR. BJORKMAN: That is clear where that came from.

DR. IOTTI: And likewise you can derive the same numbers for all of the remainder.

Now, that essentially answers the question as to how the stresses in Table H of Attachment 3 -- I have to refresh my memory -- of the affidavit. I guess that is Table H.

DR. BJORKMAN: That is correct.

DR. IOTTI: The next portion that I will address is how the stresses in Table I are derived which then deals with how do you split the total stress intensity into a primary stress intensity and secondary stress intensity.

There is a variety of ways in which that can be done. The most straightforward way to show you this is to derive the primary stress intensity and then the secondary stress intensity is derived by subtraction from the total.

Again, let me take you through either the outside or the inside surface. It doesn't really matter.

To arrive at the primary stress intensity we have to subtract out the secondary portion of the circumferential stress which has been printed out by the Finite Element Analysis.

The way we did that was to take the case of the pre-load plus thermal, which would be given on page 59 of Attachment 3, which essentially is 39 point -- am I looking at the right number? That is the wrong page. If you go to page 69 -- I beg your pardon -- of Attachment 3, you see the case pre-load plus thermal and see a maximum circumferential as minus

9.305.

As it turns out, that happens to be on the outside surface. The only reason I'm mentioning it is at least you have a figure that you can refer to.

Now the corresponding one on the inside surface I'll give you in a second. I have to go to a different page. On the inside surface for the pre-load plus thermal would be a 32.57 tensile.

Now, that doesn't appear in any of the information that you have. We will have to make this available to you.

Now, we have to subtract -- to just arrive at the secondary portion of the circumferential stress, we will subtract from the pre-load plus thermal the stress due to pre-load alone, which appears on the page before, which on the outside surface would be minus 26.09 and on the inside surface it would be a plus 21.76.

On that basis then you would arrive at the -- let's see, on the outside surface a minus 39.06, minus a minus 26.091, would give you a primary circumferential stress of minus 20.856 ksi, because the difference that we subtracted out is 13.214, which is the difference between the 39 point -- I can't remember these numbers -- the 39.06 -- sorry, 39.3 and the 26.09.

And when you subtract that out from the -- I'm working with the outside surface. So let me work with the inside surface because I gave you all the numbers for the inside surface.

For the inside surface it would be forty-four -- first of all we calculate the difference between the pre-load plus thermal and the pre-load only. That would be 32.57 minus 21.76, we get 10.81.

Now we go back to the Mohr circle circumferential, which was 44.79. Remember that on page 58?

So 44.79 minus 10.81 gives you 33.98, which would then be the primary circumferential.

We derive the primary longitudinal in the same fashion. We take the difference between the pre-load plus thermal and the pre-load, subtract that out from our Mohr total longitudinal, and we will end up with, as it turned out, a primary longitudinal of minus 1.096.

So then the total of 33.98 minus a minus 1.096 gives a 35.1.

Let me tell you that the 35.1 is different from what you see in that Table of 31.6, and the 31.6 was an error because inadvertently what they had done is subtracted out an outside surface stress from an inside surface.

So I went back and did all of this. I rederived that number. That should be 35.1.

DR. BJORKMAN: 35.1?

DR. IOTTI: 35.1

And I checked it with Bill LaPay and he concurs.

You are not the only one confused. The people who did it tend to mix numbers on occasion.

All of this has been again checked and doublechecked.

Did you follow that, Dr. Bjorkman?

DR. BJORKMAN: Yes.

DR. IOTTI: Or should I go through it more slowly?

DR. BJORKMAN: No.

The basic reasoning is what I'm looking for. And with the additional information --

DR. IOTTI: We will give you this table. You can derive your own. You need to have this because without it -- in fact, actually I can give you the example. I can have it written up and given to you.

DR. BJORKMAN: That would even be more helpful in tracing it.

DR. IOTTI: Right.

So that then would answer the question as to how you arrive at Table I.

Needless to state, the secondary stress is just derived by subtracting the primary stress intensity from the total stress intensity.

And, of course --

MS. ELLIS: Doctor, --

DR. IOTTI: Yes, Mrs. Ellis?

MS. ELLIS: -- excuse me. This is Juanita

Ellis.

I can't hear -- I guess that was Dr. Bjorkman again. I can't hear him at all.

And you kind of have a tendency to drop your voice towards the end of your sentences.

Would it be possible -- I guess you'll be doing most of the answering on this. Would it be possible for you to maybe move a little bit closer to the sqwauk box?

DR. IOTTI: I don't think you can do that.

I will try and speak loud, Mrs. Ellis, and I have also moved closer.

MS. ELLIS: I think that would probably help a lot.

And if you would sort of repeat what Dr. Bjorkman says, I think that might be helpful.

DR. IOTTI: Let me recapitulate, and I don't want to put words in Dr. Bjorkman's mouth, but what Dr. Bjorkman was mostly interested in was not in the actual numbers but in the approach taken to derive the stresses that appear both on Table H and on Table I.

I took him through an example that shows the approach. I actually worked out some numerical numbers, and I also told Dr. Bjorkman that I will give him in writing the particular example that I worked out so that he can work the numbers for himself.

And I further told him that I would provide him with the table that lists the longitudinal and circumferential maximum stresses for all of the various loading cases both on the inside and outside surfaces because without that table he would not be able to derive any numbers on his own.

MS. ELLIS: Very good.

And will you provide us with -- any documents that you provide to them, obviously we would like to have also.

DR. IOTTI: As you know, you get everything that we give anybody. So, no doubt.

MS. ELLIS: By the way, I think that someone must have moved you a little bit closer or moved the sqwauk box. You sound much louder now.

DR. IOTTI: Yes. They are working feverishly to either move you closer or destroy the box, whichever comes first.

(Laughter.)

MS. ELLIS: It's helping. Thanks.

DR. IOTTI: Okay.

We will continue on their question which then deals with how do we derive values that appear in Table N of the affidavit, and I have to refresh my memory for a moment as to what N is.

Okay.

Table N cf the affidavit, which is on page --

MR. MINICHIELLO: 67.

DR. IOTTI: 67, right.

That table, appearing on page 67, refers to a different approach that was taken to evaluate whether the stresses resulting in the pipe from the cinching of the U-bolt would be acceptable.

This is the alternative approach.

That alternative approach requires us to evaluate the primary membrane portion of the

U-bolt, pre-load, push and pressure stress.

In order to do so, what we did we actually averaged now the inside and outside surface and, unfortunately, Dr. Bjorkman I'm sure will have a hard time remembering some of the numbers that I just ran through.

But the -- for instance, the primary circumferential stress on the inside surface after we had gone through all these numbers turned out to be 33.98 ksi. On the outside surface it was minus 20.856 ksi.

That gives us an average circumferential stress through the membrane of 6.56 ksi.

Similarly, the primary longitudinal stress on the outside surface one could calculate to be minus 27.496. We had gone through it. And on the inside surface it was this minus 1.096.

So the average on the stress intensity across the membrane is minus 14.296 ksi.

And when you put it on a Mohr circle, which is essentially a stress block, what you do is add the circumferential which is oriented in a principal stress direction and the longitudinal which is oriented in the other principal stress direction; so that the total turns out to be a 20.86.

Now, that compares to the 20.99.

Now, the reason that there are differences in the decimals is that different people round off differently. But in effect this is how the Equation 9 stress that appears in Table N is derived.

You can go through the same exercise then for the secondary stresses which are Table O where, first of all, here you have to subtract out those portions of the stresses which are non-cyclic in nature because we are comparing the stresses versus the allowables of Equation 10. Equation 10 only considers cyclic stresses.

So we have to subtract out from the stress intensity that we had derived for inside and outside surfaces that which was due to pre-load alone, pre-load not being considered a cyclic stress.

When we did so, we again generated a stress block where -- the easiest way to do this is remember the total secondary stress intensity, subtract out -- I mean the total stress intensity, subtract out the pre-load, arrive at a longitudinal stress of 50.03. Likewise for the circumferential stress, you subtract out the pre-load portion, you arrive at a minus 8.

And then when you add -- is it minus 8 or minus .8? No, minus 8.

So a minus 50.03 and a minus 8, the stress intensity in the Mohr circle would still just be a minus 50.03 because the two principal stresses are on the same side.

For the outside surface one would calculate a stress intensity of 50.03.

For the inside surface, a similar process, one would calculate a 50.6.

So again, that compares to the 50.8 which appears on page 68.

MR. MINICHIELLO: I have a question, Doctor.

DR. IOTTI: Yes.

MR. MINICHIELLO: I have a question.

The maximum stress intensity was 64 ksi.

DR. IOTTI: That's total, yes.

MR. MINICHIELLO: Total.

And you are going to subtract out effectively from that the pre-load alone stress or stress intensity.

That's what we are saying, correct?

To get to this Equation 10 stress in Table O --

DR. IOTTI: Yes.

MR. MINICHIELLO: -- you want to go back to stresses and then make up a Mohr circle, but you want to go to the stresses that made up this 64 ksi.

DR. IOTTI: Yes.

And take out of that anything that is non-cyclic in nature, which would be the primary -- you know, like the membrane stress due to pressure and the pre-load effect, if you will.

Or if you will, just take the pre-load effect out.

DR. BJORKMAN: There are times when you are talking about stress intensities when you --

DR. IOTTI: I should be talking about stress.

DR. BJORKMAN: -- should be talking about stress. We understand what you mean.

DR. IOTTI: It's the same.

That's right. I should be very specific when I talk about stress intensity and when I talk about stress.

You always start from stress and eventually construct a Mohr circle.

You can't just go and subtract out stress intensities.

DR. BJORKMAN: Correct.

DR. IOTTI: Okay.

So when I meant subtracting out the pre-load, I meant going at the beginning from the stresses, subtracting out from those stresses the portion due to pre-load and then construct a Mohr circle or a stress block.

So that concludes the explanation that I have on Question 1 of CYGNA.

As I say, there is some material that we will make available to them and we will forward a copy of that material to you, Mrs. Ellis.

MS. ELLIS: Thank you.

DR. IOTTI: I propose to go on to Question 2 if there are no further questions from CYGNA.

DR. BJORKMAN: No. We have no questions at this time.

DR. IOTTI: Okay.

Mrs. Ellis, it has been suggested that we take a break at this time to have coffee and sweet rolls. Maybe you should take a break also from listening.

We will alert you when we get back or call us when you are ready and we will not start without you being on the phone.

MS. ELLIS: Okay. About how long are you going to take?

DR. IOTTI: I would say ten minutes.

MS. ELLIS: Okay. Very good.

DR. IOTTI: Thank you.

(At 10:05 o'clock a.m. there was a recess in the proceedings.)

(At 10:15 o'clock a.m. the proceedings were resumed.)

DR. IOTTI: We are going to resume.

MS. ELLIS: Right.

DR. IOTTI: And we are going to go on to the second question.

The second question has to do with whether the pre-load is to be considered as a cyclic or non-cyclic load and whether a fatigue evaluation should, therefore, be performed that includes the effect of the pre-load or whether the pre-load should not be considered.

I guess it is CYGNA's position that the cinching of the U-bolt may be periodic for whatever reasons.

Applicants would like to state that they disagree.

I don't know where we go from here other than, I guess, just for purely academic reasons, even if one were to conduct a fatigue evaluation including the pre-load, one would find that you certainly are going to allow yourself a lot more cycles than you could ever possibly think of of cinching and uncinching the U-bolt.

And just for CYGNA's information, such a fatigue analysis would be performed in accordance with Appendix 13, Article 1153 where one would calculate the equivalent elasto-plastic factor and then increase the alternating or derive the alternating stress from the product of this particular factor times the maximum stress range that had been computed; and then, using the fatigue curves for ostenetic steel, for instance, derive the number of cycles that the specimen could be subjected to without adverse effects.

So I think the question is really an academic one, whether we agree or disagree. Even if you were to include it, you still wouldn't have any problem with fatigue.

DR. BJORKMAN: Okay.

DR. IOTTI: The third question -- if there are no -- is that sufficient?

MR. MINICHIELLO: I don't have any questions.

DR. BJORKMAN: I don't have any questions.

DR. IOTTI: The third question, we were a little surprised.

Apparently CYGNA is under the impression that the ten-inch pipe test was only conducted at either 4,000 pounds applied or something less than that.

The actual, what we call the seismic test, was really a test conducted at a single frequency of nine hertz; applied the load which was, at least at the beginning, in excess of 10,000. And after about twenty-one seconds finally got down to about 8,600 pounds.

So the load that the actual test specimen saw was in excess of the emergency load that the ten-inch pipe would see.

Maybe that wasn't clear from our explanation, but I would refer you to the seismic test, what we call the seismic test, on Attachment 1.

As you can see --

DR. BJORKMAN: Yes. I see on page 91 --

DR. IOTTI: Right.

DR. BJORKMAN: -- Figure 1.

DR. IOTTI: Right.

DR. BJORKMAN: The magnitude is there.

DR. IOTTI: So, in fact, that is why we chose to do the additional test to make sure that there would be no question of not having applied a sufficiently high load to cover the emergency case.

For your information, Mrs. Ellis, CYGNA is caucusing.

So --

(There was a brief recess in the proceedings.)

DR. IOTTI: If you have another question on that, why don't we address it when we get there.

DR. BJORKMAN: Okay.

MR. MINICHIELLO: I see your explanation in terms of the level of the load.

DR. IOTTI: Yes.

MR. MINICHIELLO: My question was in the -- came out of the affidavit where in the affidavit it seemed to imply 4 kips was the maximum in one place, where if you look at the tables, 6.7 kips was the maximum.

At the time I was trying to tie it together.

DR. IOTTI: We can certainly see where, due to the complexity of this affidavit and the many issues that have been addressed simultaneously, one can get rapidly confused in reading through it and may miss a piece of information or may not necessarily understand how that information was put in and for what purpose.

We had a very short time to put this affidavit together.

So a lot of these questions are probably in the nature of additional explanation more than any disagreement because if I were -- if I hadn't written this and somebody was asking me questions, I'd be hardpressed to understand it.

I guess you still have a concern with the seisic test with regard to the frequency, whether that's a proper or improper seismic test.

But you have an additic al question on that, so why can't -- I would suggest that we defer that to that question.

I would like to go on to Question 4 which has to do with the justification for using 250 degrees as a maximum temperature for the ten-inch pipe.

The 250 degrees, it was essentially a compromise decision that was reached.

The ten-inch lines that are employed at Comanche Peak are employed on the following systems: Containment sprays, component cooling, shield water, auxiliary feed water, safety injection, service water, residual heat removal, spent field pool cool, vent air and fire protection.

Of all of those systems the only ones that are really anywhere in the high temperature or possibly relatively high temperature range are:

Containment spray -- which would be limited to the temperature that you would see in containment during a resurge and that's less than 250.

The component cooling is less than 250.

The safety injection and the residual heat removal under certain operation could see temperatures which are in excess of 250.

Normal RHR at the beginning of the operation could be as high as 280.

You could, in fact, during normal shutdown, that portion of the line that takes suction directly from the primary system would start at 280.

The --

MR. MINICHIELLO: I thought it was 350.

DR. IOTTI: No.

350 requires some upset condition.

There are instances that you could consider where you might operate that line at 350. But the normal operation is 280 for shutdown.

MR. FINNERAN: And this line is insulated.

DR. IOTTI: Now, what we had to decide is what would be the worse: A 250 uninsulated or a 280 or whatever insulated in terms of the relative thermal expansion.

And we opted to use 250 uninsulated because it is far more representative of the vast majority of the cases that we have in the plant.

We could have gone to 280. We could also have gone to 350.

I guess maybe, Bill, you may want to make some comments.

For the ten-inch line, the thermal at those relatively low temperatures, does not represent a substantial portion of the load.

So from the Finite Element Analysis we can derive what the impact would be in the final answers if we were to increase the temperature.

Again, here, what we were trying to do is do a test and do a Finite Element Analysis to correlate the test because then the Finite Element Analysis would then become more true for determining what changes might be precipitated from any differences in conditions in the plant from what has actually been tested.

But that's the reason that we use 250.

DR. LaPAY: I can confirm exactly what Dr. Iotti has said.

Looking into those stresses due to the temperature, there is not -- it would not change the conclusions at all.

The 210 degrees is representative of the actual line temperatures, whereas the 250 degrees in the test is more of an envelope, a higher value, in our opinion.

And looking at the stability of the U-bolt to higher temperature, it would not change. It would still be a stable system.

The stresses in the pipe would still be within acceptable limits.

MR. MINICHIELLO: I just want to clarify one thing.

The portion of the RHR safety injection system that contains ten-inch piping, is that the portion taking direct suction from the primary load?

MS. ELLIS: I m sorry. I couldn't hear that part at all.

MR. MINICHIELLO: Is that the portion -- is it connected to the portion taking direct suction from the primary load?

DR. IOTTI: I believe there is -- how many of these U-bolts in that portion of the ten-inch?

MR. FINNERAN: There is only one.

DR. IOTTI: One.

MR. FINNERAN: There is only one U-bolt on RHR ten-inch line, and I don't know what part of the system.

DR. IOTTI: We are going to confirm it to you.

MR. MINICHIELLO: The only reason I'm asking that is that I thought, if it's the system that I think it is, if it's on the back side of the heat exchanger, then the 280 I believe is correct.

We have reviewed through the Gibbs and Hill analysis both sides of the heat exchanger and I think the 280 is the right number.

The reason I ask about the 350, if you are on the front side, before you hit the RHR heat exchanger, I think the maximum was 350.

DR. IOTTI: Or if you lose the heat exchanger as an upset condition, if you lost component cooling to the heat exchanger.

MR. MINICHIELLO: Yes.

MS. ELLIS: The last part faded again.

DR. IOTTI: We were discussing what maximum temperature a ten-inch line could see under certain operating conditions of the plant.

And the last sentence had to do that there is a portion of the ten-inch line which may or may not contain one of these cinched U-bolts -- we will have to confirm that -- which, if it's in front of the heat exchanger, "in front" meaning in the flow flowing into the heat exchanger, could see a temperature of 350.

That temperature could also be achieved after the heat exchanger if the component cooling water flow to the heat exchanger were to be lost.

guess, in summary, the Applicant has provided CYGNA with their explanation as to why they chose 250 as a temperature to test the ten-inch specimen.

They've further -- Dr. LaPay of Westinghouse provided an explanation as to what the consequences would be in terms of piping stresses or pre-load's inability of the cinched up U-bolt to maintain, quote, its configuration, if the temperature were to be increased to a higher temperature.

And his conclusion is that there would be no adverse consequence from that because the amount of stress caused by the thermal at this lower temperature is not very significant.

That brings me to Question 5.

Bill, could I ask you to take on Question 5 for me while I go upstairs and get another piece of paper that may help us.

DR. LaPAY: Sure. No problem.

DR. IOTTI: Question 5 has to do with the agreement between the measurements of the tests in terms of stresses, stresses deduced from the test, and those predicted by the Finite Element

Analysis.

DR. LaPAY: To respond to this question, no attempt was made to further quantify the results in what was already reported since the comparison, we considered, was reasonable in the area of significance.

Let's take the ten-inch, for example, the results as given in Attachment 3.

Looking at Attachment 3, the results are given on pages 36 and 37 for the ten-inch, referring to strain gauge locations on page 34, Figure 2.

Looking at those results, one can see in the area of significance, which is around the crosspiece, the -- in looking at strain gauge locations A, B and C, the comparison between analysis and test is reasonable and within proper sign and magnitude recognizing that the fitup or gap closure between the pipe and the U-bolt is different.

Looking at the strain gauges D and E, the test does show a drop off faster than the

Analysis, but this is explained by the fact that the crosspiece was bowed and, therefore, it was bowed where the edges were actually not -- there is about a -- it states in there I think a sixty-fourth-of-an-inch or so, that it's bowed upwards so the crosspiece is in contact prior to the edges.

DR. BJORKMAN: The crosspiece is in contact more at the centerline of the U-bolt than the outer edge?

DR. LaPAY: Correct.

Therefore, you would get a stress dropoff a lot faster.

MS. ELLIS: I couldn't hear that question.

DR. BJORKMAN: I just commented that the point of contact really is more at the centerline of the U-bolt than it is at the edge of the crosspiece, and Dr. LaLap confirmed that.

DR. LaPAY: Further, there were some studies performed that indicated that the distribution of stress within the crosspiec? is dependent upon the fitup between the U-bolt and the pipe.

Therefore, a U-bolt that is constraining the pipe, you will get a more concentrated load towards the middle than at the edge.

And it was a known fact in the test that the fitup between the U-bolt and the pipe was different than what was used in the Analysis because we could actually look and see the gaps.

Now, we further, down in the area around G, H, I and J, of course there were differences, and this was definitely recognized that there would be differences because the contact points between the Analysis and the test were different.

But this area is where the stresses in the pipe drop down considerably compared to the contact point between the crosspiece and the pipe.

Further, the sign difference, of course, could occur, that there were differences, because the magnitude of stress is quite a bit lower. They were small. So you would expect that you could possibly have a sign difference. This doesn't indicate anything of significance because they are small.

Where we do get proper boundary conditions between Analysis and test, we do get a good agreement.

Further, it's important to note that the difference between test and Analysis indicates that the Analysis results are always giving higher values which are conservative when evaluating the stresses.

So that was a benefit. That was really not a benefit so much as it demonstrates that the Analysis results would be conservative.

Therefore, based on this comparison and recognizing the differences in the boundary conditions, we felt that we had a good comparison for our purposes and no attempt was made to measure gaps in the test contact points or to refine our measurements of the stresses throughout the pipe by adding additional strain gauges, because that would be beyond the scope of what we intended to do.

DR. BJORKMAN: I have one additional question.

Do you know offhand what the gauge length of these gauges were?

DR. LaPAY: No, I don't. Unless they were given in this test, I don't know offhand.

DR. BJORKMAN: I didn't see it in there. I was just wondering as a point of interest.

DR. LaPAY: That is something we can --

MR. FINNERAN: We can provide that to you if you would like to know.

DR. BJORKMAN: Yes. It would be interesting.

MR. FINNERAN: Okay.

DR. LaPAY: We will get you that information.

MS. ELLIS: Those last couple of sentences I couldn't hear at all.

MR. FINNERAN: We are going to provide to CYGNA the gauge lengths of the strain gauges used in the test.

MS. ELLIS: Okay.

MR. FINNERAN: We will provide the same to you, Mrs. Ellis.

MS. ELLIS: Okay.

DR. LaPAY: Any other questions of 5?

MR. FINNERAN: Mrs. Ellis, we just spilled a cup of coffee, so we are just trying to clean that up.

Just a second.

MS. ELLIS: Okay.

(There was a brief recess in the proceedings.)

MR. FINNERAN: We are ready to start back up again, Mrs. Ellis.

MS. ELLIS: Okay.

DR. BJORKMAN: I think one of the reasons for just asking the question was to find out if any additional particular Finite Element tests might have been run which might have, for example, placed gaps in regions where gaps were observed in the test, to try and basically correlate some of the information.

We recognize, particularly in Regions G, H, I and J, because of the fitup problems, that you are not going to get close correlation here.

And we also recognize the problem with the crosspiece which, I believe, was mentioned in the report and that we cannot exactly duplicate in the Finite Element Analysis all of the boundary conditions exactly.

One of the purposes of the question was to just see if anything further had been done there.

DR. IOTTI: The only thing that was done that you might consider further was not along the lines that, of modelling gaps to better simulate what the test conditions were because, as you say, that can be very, very difficult.

Ebasco, independent from Westinghouse, did run, at least for the pre-load case alone, a totally elastoplastic both pipe and U-bolt analysis.

In other words, we let the material do what it wanted to verify whether, in fact, that stress pattern that we saw emerging from their analysis -- which showed a fairly significant stress reversal as you moved away within a very small region of the pipe, as you moved away longitudinally as well as circumferentially from a backing plate -- whether it was something that was a result of having done a elastic analysis and an improper Finite Element Analysis, if you will.

What, in fact, we did was to confirm that that is truly there.

And that may be one of the reasons also why in the regions where you predict low stresses from the strain gauges -- and then you also have to remember that the strain gauge is only in one direction. Of course, it will read a lower stress than what it actually has. Okay.

It is very possible that by just missing by maybe less than a half-inch in some instances the location of strain gauge versus where the stresses are predicted by the Finite Element Analysis, you can, in fact, predict the tensile stress where you have a compressor.

And that kind of us satisfied us that the Westinghouse analysis was correct; that we weren't overly concerned over the fact that we weren't simulating the very high stresses.

What it also told us though is that we didn't see as high a longitudinal stress as you did away from the backing plate either. Our stresses were lower.

So we would get better agreement.

Unfortunately, we didn't use the same size pipe as you did. We happened to model an eight-inch primarily because we knew that CYGNA had modelled an eight-inch at one time.

But the pattern is essentially the same although we will observe a more rapid decay of the longitudinal stress away from the backing plate because we modelled the plasticity between the backing plate and the pipe.

And I think that tends to give you less of an effect on the longitudinal stress as you move away from the backing plate up on the top of the pipe, if you follow what I mean, up in here.

DR. BJORKMAN: Yes.

DR. IOTTI: These dimensions -- each of these elements is like a half-inch or so and within about two inches of the edge of the backing plate.

Our longitudinal stresses went from -- let's see -- sigma y. This is top surface. Of course, this is the inside surface.

But you can see it dropped far more significantly when you arrive up here than -- theirs was still as high out here as it was right at the backing plate.

And my modelling everything elastoplastically we found that we do get some plastic deformation way up here and these stresses drop.

DR. BJORKMAN: So the --

Basically the element at the -- the finite elements, elements or element, at the point of contact of the backing plate with the shell were plasticity elements?

DR. IOTTI: It could be. They showed a very small amount of plasticity right underneath the backing plate.

MS. ELLIS: I couldn't hear that last exchange at all.

DR. IOTTI: The element right underneath the backing plate showed a very, very low amount of plasticity right underneath.

Bill, did you also run that?

DR. LaPAY: We ran a plasticity analysis too.

And I think the level of plasticity in our case didn't reach as high a magnitude where we got the dropoff.

Those reports -- our plastic analysis is reported on page 60 of Attachment 3.

And there you can see the difference between the elastic and plastic analysis, and D and E string gauge locations are about the same order of magnitude. Nothing of significant dropoff there.

DR. BJORKMAN: Which indicates that you are probably at a low level, very localized plasticity.

DR. LaPAY: Right.

DR. IOTTI: Correct.

DR. BJORKMAN: Okay.

DR. IOTTI: Anyhow, you asked whether we have done anything else. That's really the only thing that was done in addition.

It was more of a verification to provide ourselves with the assurance that the Westinghouse models were predicting the correct stresses, the correct pattern of stresses in the pipe as well as in the U-bolt and the backing plate more than anything else.

And it was done on a separate model, totally independent.

DR. LaPAY: To respond to your other question that you asked previously on the active gauge length, those are given in Attachment 1 on pages 14 and 15 with the torque versus pre-load test.

DR. IOTTI: Quarter-inch.

DR. BJORKMAN: That is fine.

DR. LaPAY: Okay.

DR. BJORKMAN: Okay. Fine.

DR. IOTTI: Want to go on to 6?

DR. BJORKMAN: Sure.

DR. IOTTI: Okay.

Six asks the question with regard to sufficiency of pre-load to preclude slippage during a seismic event after the pipe had been heated and cooled and the support loaded and unloaded.

Or, in other words, given the variability between the torque and pre-load relationship, is it not possible that the pre-load may be lowered below the level required for stability?

Our answer to that question is that we don't believe that that is possible.

We loaded the specimens for the long-term vibration test as well what we call the seismic test to a pre-load level of 50 foot pounds, although the Finite Element Analysis had indicated that we could have lowered that pre-load level and still have achieved stability.

At that particular pre-load level we really did not see or what we consider very significant relaxation at least for the specimen we tested primarily because at the pre-load level the general stress in the U-bolt is below the one-half of yield.

A lot of the tests that show significant relaxations were tests that were conducted not at that pre-load level but at a pre-load level which was double, which placed the level of stress in the U-bolt higher than one-half of yield.

Okay.

To place this in perspective, I guess, if you were to take the total percentage reduction from the 50 foot pounds pre-load and assumed that that is the same percentage in reduction load as you would get as if you started with a one hundred foot pounds pre-load -- because that's the only data we got, from the one hundred foot pounds up; we saw significant relaxation, okay -- you would end up with a total relaxation of about thirty-three percent after you vibrated.

If you take thirty-three percent out of fifty foot pounds, you end up with approximately thirty-two/thirty-three foot pounds.

We had run as part of the test a pre-test which had this thing, the specimen, only loaded to thirty-five foot pounds and that showed stability at thirty-five foot pounds.

What we did not attempt to ever do is keep lowering the pre-load until it became unstable.

We know at a certain level it would be, not unstable in the sense of not being able to transmitting and carrying a load, but at least it would walk around the pipe.

And that is an important distinction. The fact that it is walking around the pipe does not mean that it cannot transmit or carry load because we were obviously able to transmit and carry load during the test.

But we define, if it walks in any manner that we didn't consider acceptable -- in other words, in an unpredictable manner -- that we would define that as being unstable.

At thirty-five foot pounds -- Bill, is that correct? -- it did not walk at all.

DR. LaPAY: I was going to discuss that

further with one of the other questions.

DR. IOTTI: One of the things that we intend to show you is a tape that we took of the seismic test run at nine hertz as well as the fundamental frequency of the system, which was around seventy-five, I believe.

We couldn't run -- we may be getting ahead of ourselves -- we couldn't run the nominal 7,000 pounds or the 10,000 pounds at the fundamental frequency because the equipment wasn't capable of transmitting that load.

There was enough play in the equipment that you just can't get that load through.

That is why we ended up just at nine hertz.

That also ought to tell you something about what kind of frequency you can drive through these supports.

There is always some play, you know, someplace in all supports.

But what we did do, what we will show you in that videotape, is that for the level of pre-load of fifty foot pounds at the very high loads that we put into the system, the U-bolt moved in a predictable manner that you can predict by geometry and aligned itself in a more favored manner until it found its position of least energy, least potential energy, and stayed there, remained there.

And we noted that in the long-term vibration test.

So what we concluded is that, provided you cannot rotate the U-bolt around the pipe, if there is to be any ancillary movement, it is because the pipe -- the U-bolt wants to go to the distance of -- the shortest distance of the strut. It goes towards the line of action of the force, never away from it.

The moment you permit the rotation, then geometry can arise where the lever arm is such that you can actually push away; the push portion of the cycle, you can push the U-bolt away, and then the U-bolt can just rotate but also walk away from it.

We defined that condition as not conducive to good, call it, stability or functionability of the support system, even though when that happened we were still carrying the load and transmitting the load.

MR. FINNERAN: The load was still transmitted. So it still functioned.

DR. IOTTI: But we drew the line. We wanted to keep the support where it's supposed to be.

And so, when we talk about minimum pre-load level, we really don't know how low we can go.

We know that at thirty-five foot pounds for the ten-inch we were stable because that was observed by tests.

We are certainly stable at fifty.

I don't know how much lower we could go.

We know that -- the first one we ran was twenty?

DR. LaPAY: The first -- twenty foot pounds was the first.

DR. IOTTI: Twenty foot pounds it was not. It would walk away from it. It wasn't sufficient.

So somewhere between the twenty and the

thirty-five there is a point at which for that particular specimen the support would have started walking.

And we will show you the videotape at the appropriate moment or whenever we want to see it.

So, basically it is our opinion, based both on tests and on analysis, that the top of pre-load that we would apply through -- the minimum pre-load load that we would apply in the plant is high enough that, even given the relaxation, would leave you with sufficient pre-load after all of the heating and the cycling; that were you to have a seismic event it would be able to carry the load.

And this is how we arrived at the minimum pre-load, considering all of those factors.

In other words, we take into account the relaxation that we expect, the maximum relaxation that we would expect that system to see.

DR. BJORKMAN: Part of this question relates back to Question 12. We can basically wait and defer discussion until we get to Question 12.

DR. IOTTI: Yes.

In terms of how you establish the minimum pre-load and --

I'm just getting ahead of ourselves. We may have confused you unfortunately by printing out results of Finite Element Analysis that says here is the pre-load, the low pre-load. That is not the pre-load required for stability. It happened to be the lowest value of pre-load that they ran.

For some other pipes it turned out to be very low and may well be the lowest level for stability, and the system was still stable.

But for, like for that forty-six foot pounds, they could have gone much lower and still had stability. They just happened not to choose to go. They didn't optimize.

So one is led to believe that the forty-six may be interpreted as being the lower level and it's not.

DR. BJORKMAN: I think we can wait to address that when we get to that question.

DR. IOTTI: CYGNA is caucusing again, Miss

Ellis.

MS. ELLIS: Okay.

(There was a brief recess in the proceedings.)

MR. MINICHIELLO: Okay.

In Question 6, part of my question is based on, if you take a pipe, not necessarily the ten-inch, any size pipe, you heat it up, you let the system run at some normal temperature let's say for a period of time, and then you cool it down.

In a sense what I'm asking, you take out as much pre-load as you can through material relaxation.

DR. IOTTI: Yes. Or mechanical.

MR. MINICHIELLO: Whatever.

DR. IOTTI: Sure.

MR. MINICHIELLO: You put a strut -- let's call it a strut at this point. You put the strut in tension and you pre-loaded it. So you let the U-bolt relax as much as it can, and then you cool it off. Maybe you run that four or five times.

Now you have cooled it back down. The pipe is sitting there.

Another portion of the same piping system is now in operation.

Because of that you have a compressive load on the strut. The portion that the U-bolt is around is still cold. A valve is closed.

But another system is heated up such that the system wants to grow and put a compressive, a static load --

DR. IOTTI: I understand.

MR. MINICHIELLO: -- on the strut.

Part of our question, besides vibration and even under statics, are we assured that with the loads, tensile or compressive, that we have for stress, that there was sufficient pre-load.

DR. IOTTI: To answer your question, Mr. Minichiello, if the pre-load that remains on the cinched up U-bolt after it has gone through however long a thermal cycle you want or however many cycles you have, is still high enough so that when you apply a static or dynamic, if you will, push load -- because, incidentally, Westinghouse applied it statically -- so you do not lose contact, and friction -- you know, you can roll, you know, because in fact there is a roll not in terms of rotation but a roll around the U-bolt -- so that enough friction is sufficient to maintain the U-bolt from actually slipping, okay, even though you may, in fact, have lost momentarily the contact at the bottom, okay, provided that that pre-load is still sufficient to establish the couple at the U-bolt, then you will not have -- you will have sufficient pre-load.

And your question is, after you have gone through all of that cycling leave you with sufficient preload.

We think we have established that by tests because we put it through thermal cycles.

We started out -- of course, the test that you saw was predicated on the maximum pre-load. And so you would exhibit more of a relaxation than what you would will see if you started at a lower load.

The question is how much of a load remains.

And that is a function of the percentage of relaxation you can expect.

We don't expect, if you start out with a pre-load that is low enough, yes, we will get some relaxation, but not very much as you go through all of the thermal cycles, because the state of stress is such that you will not get it.

We are talking about maybe twenty percent or so.

MR. MANJOINE: Maximum thirty percent.

But that relaxation is just due to the fact that, you know, the pipe has increased in size, the modulus has reduced and the yield strength, if there are any parts up to the yield strength, goes down. The yield strength goes down thirty-five percent from room temperature.

So if you are at yield stress to begin with -- say you tightened up to yield stress -- the load will go down to the thirty-five percent.

Now when you come back to room temperature, you get the change in modulus, and you have a maximum of something like thirty percent relaxation of the load in that one shot.

DR. IOTTI: That was assuming you stressed it at yield to start with.

MR. MANJOINE: Right.

DR. IOTTI: We are not starting from that point. Our initial pre-loads will be below half a yield.

At those points you really should not see much of a relaxation at all.

So our answer to you is that, even in a static case, if you select the proper pre-load to begin with, you will always have sufficient pre-load left that, even in a static case, you can accommodate the load either -- certainly as a push. This is what I'm sure you are looking at rather than a pull. In a pull it's always stable.

MR. MINICHIELLO: Let me try to follow that line.

DR. IOTTI: Maybe you ought to use that chart.

MR. MANJOINE: Yes.

I don't know if you can see this or not.

(Indicating)

Stress versus temperature.

Here is the change in modulus effect when we are at the yield stress. That is the room temperature yield stress.

The yield stress goes down like this.

So when you come back to room temperature, you come back in a line like that.

So you have that drop in load just due to the change in yield.

And that's -- we call it relaxation. It is a mechanical relaxation because of yield.

Now, you don't get any further relaxation from that unless the stresses are up here at yield.

And then, say in a thousand hours you might -- the maximum relaxation you can get in a thousand hours due to cyclic load as well as time could be down to some point here.

Then you would still have stress at least the maximum load that you could possibly get is forty percent from the -- that's if you were at yield throughout the U-bolt from the beginning.

Now, you mentioned that you can't get up to yield throughout the whole U-bolt because when you tighten it up and you start yielding, the man who's tightening that will find out in a big hurry. And that would require a much larger foot poundage than what you are talking about.

DR. IOTTI: Yes.

Well, we have the correlation between pre-load and torque. And, of course, our pre-load would be such that we would be below -- well, let's say for the ten-inch, okay, for all of them -- our minimum pre-load would be below the top of pre-load that would put the U-bolt section, the shank, at half of yield.

MR. MANJOINE: Half of yield.

DR. IOTTI: This would be nominal yield at thirty-six. Okay.

That's all we are talking about.

MR. MANJOINE: Now, we'll define the line here at half-yield, and nothing happens. The load goes down due to the change in modulus -- going up the temperature.

But you get no relaxation for stresses below this.

So you have to get up -- clear up to this stress here to get relaxation.

And you are always way down in this region here.

So even if you maintained that load, clear out the temperature, you get no relaxation.

DR. IOTTI: One way perhaps to illustrate it better, if you look at the thermal cycle test that was done, the one the really showed significant relaxation was the one with four-inch Schedule 160 which was loaded essentially at eighty-seven percent of yield to start with.

And, of course, that relaxed very fast as you would expect initially, and then slowly relaxes, and slower and slower.

We don't really understand that first jump that we saw on the ten-inch Schedule 40. Okay.

We think there was a fitup improperly

initially. But when the test effectively started, you really didn't see much of a relaxation at all in the cycling of the ten-inch Schedule 40 because, number one, the temperatures were much lower. But most importantly also that the disstresses in the U-bolt were low.

And the same thing, of course, in the thirty-two-inch where we saw nothing at all, and that, the temperature was high. But the stresses were insignificant really in the U-bolt, the two-and-three-quarter-inch U-bolt.

So it's a combination of all of these things.

Now, I know what your next question has to be.

How do you know that you have sufficient pre-loads that you can relax and still have sufficient pre-load?

And you have this range of preloads within which the configuration is acceptable.

If it's too high, it will relax back down to the upper range.

If it's too low, you could go unstable.

Well, of course, this is what we have to give you as part of your last question, how we establish that and why we feel that within that range of pre-load we will have the stability that is required.

And incidentally, that is one of the reasons we committed to go in and check the torques of the U-bolt.

We ourself understand that if you don't have sufficient pre-load, you are going to have a problem.

We are not so much worried about the upper range. Okay?

DR. BJORKMAN: I don't know if this is the proper time to address it because again involved here is the potential variation in pre-load with full torque.

There is also the question in our mind of the actual relaxation characteristics of A36 steel.

I don't know if you want to address them now or wait until we get further on with some of the questions.

DR. IOTTI: As far as I'm concerned, we can address them now.

MS. WILLIAMS: I think we should address them now.

DR. BJORKMAN: One of the questions that I had regarding relaxation is -- I don't know which question to ask first.

DR. IOTTI: It doesn't matter.

DR. BJORKMAN: Why don't we -- I'd like to take a look, for example, at Figure 21 of the test report dealing with the four-inch pipe specimen.

I have some questions concerning that.

MS. ELLIS: Which document is that again?

DR. IOTTI: That's Attachment 1 to the affidavit, Figure 21.

MS. ELLIS: Okay. Thank you.

Do you know what page that is offhand?

DR. IOTTI: Page 60.

MS. ELLIS: Thank you.

DR. BJORKMAN: In the affidavit, in fact it was on page 24, it basically stated that, after the final thermal cycling, that the -- at ambient temperatures the stresses in the U-bolt were reduced to approximately fifty-four percent of yield.

DR. IOTTI: From an initial about eighty-seven percent of yield, yes.

It dropped about forty percent.

DR. BJORKMAN: However, what I don't see is the fact that during the ten thermal cycles I don't see the relaxation taking place during those ten termal cycles.

In other words, at the end of the tenth cycle while the U-bolt and pipe are still at temperature -- the pipe is at 560 degrees Fahrenheit, we find that the stresses have dropped down to about 34,700 psi in one length and about 32,100 psi in the second length.

These stresses are significantly above yield. They haven't come anywhere close to half-yield yet.

And I am wondering what the explanation is for the fact that they remained so high when, in fact, we would have expected them to drop down closer to half-yield by this time.

DR. IOTTI: For that same type of explanation.

The real yield is 45,000 for this material. But that's -- you know, that kind of washes out because this is on strain gauge measurements, so the two would scale out.

But what you see through the cycling -- of course, you are cycling down in this fashion, okay. And then as you return to room temperature, because you are cycling up and down, you recover some of it, and then you lose it again and then you recover it.

And it all washes out at the end of the cycle. You end up down here, no matter what.

This may be your first cycle. You come down here. You drop there. You may get some relaxation because you are above yield. So what he is talking about is dropping down to this curve.

So, if I may -- maybe I'm stealing your

thunder, Mike --

MR. MANJOINE: Go ahead. I'll sit here.

DR. IOTTI: You come down this way because yield decreases with temperature now. I guess temperature-wise we are discussing 500 degrees. So let's say that you come down here.

Now you are at the peak of the cycle temperature-wise. Now you are returning to room temperature.

Since you are above yield, yes, you may decrease. You may lose material-wise another fifteen percent, whatever it is.

And now you keep going back and forth right here.

Now, with time -- in the first cycle you may be here. When you return to room temperature, you've lost load, maybe thirty percent.

And then the next cycle you may lose some more because now the material has had a chance to relax itself being above yield.

But ultimately the worst you are ever going to get is this, forty percent of relaxation, no matter how many cycles you have.

And I think that is what you are really seeing here.

You will have noted you have dropped from eighty-seven percent of yield stress down to something like fifty-five percent, which is about a forty percent decrease, thirty-five percent decrease.

DR. BJORKMAN: At ambient temperature.

DR. IOTTI: Well, you know, at ambient -- when you ride back down to the ambient temperature, which addresses his concern now.

If you start out with a very high stress to begin with.

MR. MANJOINE: I might mention that this is not what we would call creep relaxation, normally associated with creep. You don't get into the creep range applied to the curve here of temperature -- stress versus temperature until you get up, I think, to --

DR. IOTTI: About seven hundred degrees or so.

MR. MANJOINE: -- seven hundred degrees.

Then you start to get creep, which is time dependent.

Below this thing it's really not time dependent except that it's plastic flow, and anything of flow takes time.

So these things always reach a certain limit and you can't get below that.

Now, since each test only lasts a certain amount of time, it is trying to reach this limit each time.

But eventually there is a limit to where you can go.

DR. IOTTI: Well, the first initial drop is what you see, I think, --

MR. MANJOINE: Yes. That's the first --

DR. IOTTI: -- is the first cycle, the big one.

MR. MANJOINE: The big one.

DR. IOTTI: After that it will keep trying to get closer to this last thing where it comes down to stresses which are around half a yield or so.

MR. MANJOINE: Yes.

As long as you go to a temperature that is say something like seventy percent of the yield, it will still yield -- at room temperature at seventy percent of yield it will still yield at, when it gets up to temperature. And then you'll get a little bit.

And the other thing that you got to remember is that you're only a portion of the cross-section that's yielding in most cases.

And there is a springback from the rest of the system. So you always get that.

That's why the relaxation never really gets up to that forty percent.

DR. BJORKMAN: I think what we are probably suffering from here is a lack of information of what the relaxation characteristics are of A36 steel.

Which brings me to my next question which is basically how would the relaxation characteristics of A36 steel established from

DS60, the ASTM document on stress relaxation --

MR. MANJOINE: Yes.

Well, I'll plotted it here. Here is the maximum amount of relaxation you can get if you are at the yield stress or above.

DR. BJORKMAN: Now, did this come from DS60?

DR. IOTTI: No.

MR. MANJOINE: This part down here comes from DS60 because no one runs relaxation here below, say, seven hundred degrees because they load this specimen -- if you are doing it in the lab, you load the specimen -- the whole cross-section is loaded. It yields.

When you start taking records, if you don't start taking records at one-hundredth of a second, you won't get anything because --

Usually you start taking records about after a tenth of an hour, and the relaxation is gone, and then you measure a straight line.

So if I were to run tests up there, nothing is happening.

Now if you run tests and go back to virgin time with a stiff machine -- it has to be a very stiff machine -- then you can get as much as fifteen percent on the porous material.

So I have drawn the curve. And we have done this for other materials. And we have also done some for a lot of model steels.

The thing we find is this. You get no relaxation of a virgin specimen until you initiate plastic flow. You'll get nothing until you get up to the proportional limit, which for these model steels is above the yield stress.

But once you have initiated plastic flow, then you can get plastic flow.

So you have to initiate it. And that's one of the big things. You have to yield these things before they do any relaxation.

And we find this both cyclic as well as -- we have done some -- we have done some on the ferretic steels.

This is taken from a published report.

The first time you load, you have an upper and lower yield part.

After you cycle and go to reverse stress, reverse yield, the proportion limit is one-fifth of t.e yield stress.

So you can initiate pastic flow. If you had gone through this cycle here, come down to here, and then just come up to this point right here, you can actually get some relaxation because it will go to an elastic --

DR. IOTTI: Let me remind you --

DR. BJORKMAN: There is a strain limit in test too.

MR. MANJOINE: Oh, that's a strain limit in test.

DR. IOTTI: -- the real U-bolt does not experience this stress reversal.

DR. BJORKMAN: No.

MR. MANJOINE: No, no.

DR. BJORKMAN: Exactly.

MR. MANJOINE: But you have seen this, if you are familiar with materials.

You go up to here and yield, and then you come down, just unload it.

Now you go back monotonically. You never get up there. It will start bending off and do this.

DR. BJORKMAN: Correct.

MR. MANJOINE: This is mechanical relaxation and it's a limiting thing. You can't get -- as long as you stay below that stress, nothing happens.

DR. IOTTI: That fifteen percent or so.

MR. MANJOINE: Right.

Nothing happens.

And so, you have to be in a stress from here to here to get any relaxation. If you stray below this stress -- so that's why we say if you stay below half of the yield, monotonic yield, nothing happens.

And we have actually run tests. Oak Ridge National Lab ran some tests.

Relaxation is a very interesting subject if you ever get into it.

Here's the loop, see. Go into compression, comes up to here and we get relaxation at fifteen percent.

Now, let's load -- unload twenty percent. You don't get any more relaxation even though you are going through this cycle.

In fact, if you come back to zero stress, you'll get positive -- negative relaxation. The load will go up. And so they show these loads going up in this portion of the loop.

So if you go up to like what we are doing here in the yield and it comes down in here, you get no relaxation whatsoever.

And these are tests -- this is a two-and-a-quarter Cr-1 Mo, which is --

For all of these ferretic steels, you would be able to say.

But you notice the characteristic of the loop. No upper and lower yield point once yielded.

But you have to initiate plastic flow and it has to be in the wrong direction, going into compression. And the reason is that there is a back stress exactly equal to about two-thirds of the yield. It's there. Nothing will happen until you exceed that.

So it's --

DR. IOTTI: I think part of the difficulty Dr. Bjorkman had, he really didn't have any data, and it's difficult to find any unless you get a hold of persons like Mike Manjoine on ferretic steels.

MR. MANJOINE: I've shown the kind of data you can get. This would be the worst kind of relaxation.

DR. BJORKMAN: Would it be possible for us to get a copy of this?

DR. IOTTI: I am preparing an answer to the Commission. You will get get a copy of the answer. Part of that answer will contain this.

You see, DS60 is an interesting document. But you really have to lead yourself to a conclusion by examining all of these materials at the high temperatures because there is nothing available at low temperature other than monotonic loading of one particular material --

DR. BJORKMAN: That's right.

DR. IOTTI: -- which is closing out.

But that one tells you if you are below the proportional limit, you really don't get any relaxation. That is all you can come to the conclusion. Okay?

So if you are sufficiently below that, you'll know you're okay. But it doesn't answer an awful lot of other questions other than when you start thinking of the cyclic being similar to what ostenetic steel also exhibits, then you also get some information from the ostenetic steel, which is going to lead you to believe that the one-half of yield is some magic number that you ought to shoot for.

But where I arrived at the one-half of yield even before I had DS60 was from the test data.

I kept looking at the test data and correlating where I saw significant relaxation and where I didn't. And I kept concluding that every time I was below this half of yield I wasn't seeing much relaxation. Every time I was above it

I was seeing it.

That's what got me focused on the one-half yield. And after that I started hunting for material that would essentially confirm.

DR. BJORKMAN: Well, the one-half yield seems to be something that became fairly obvious to us.

MS. ELLIS: I'm sorry. I couldn't hear that one at all.

DR. IOTTI: He said the one-half yield is something that became fairly obvious to CYGNA.

DR. BJORKMAN: For the ten-inch pipes. But it wasn't as obvious for the four-inch pipe.

And I think we were missing a link here.

MS. ELLIS: Gentlemen, you all were talking about a document, I think, a few minutes ago. I couldn't hear very well. From Oak Ridge Natinal Lab?

DR. IOTTI: No. What we will do, Mrs. Ellis, we are preparing answers to the Nuclear Regulatory Commission.

One of the topics is, of course, relaxation of A36 and you will get a copy of those answers as usual. And all of that information will be contained in that answer.

MS. ELLIS: Okay.

Thank you.

DR. BJORKMAN: I think this discussion basically clears up the discrepancy that I had here.

DR. IOTTI: Okay.

MS. ELLIS: You were talking about DS60 or BS60?

DR. IOTTI: The title is "Compilation of Stress/Relaxation Data for Engineering Alloys." And it's an ASTM publication, American Society for Testing Materials. DS, like D, 60.

It is referenced in our affidavit. I think the reference -- anyhow, that's the title.

MS. ELLIS: Okay. Thank you.

DR. IOTTI: As a matter of fact, we provided you with all of the pertinent information relating to ferretic steels. We copied that portion and sent it to you.

MS. ELLIS: Okay.

DR. IOTTI: Let's see.

Gordon, you had, I think, an additional question that had to do with pre-load.

DR. BJORKMAN: Actually, we might wait for Question No. 12.

DR. IOTTI: All right.

MS. WILLIAMS: Either that or cover it all right now. We'll just cross 12 off the list.

If we are on the subject, it might be better continuity to do that now.

DR. IOTTI: Okay. You want to jump to Question 12?

DR. BJORKMAN: Sure.

DR. IOTTI: Okay. Let's jump to Question 12 for the time being.

I guess my first answer to your question is a question. I don't like to answer questions with questions, but I have to understand where you got this table because I'm --

DR. BJORKMAN: Okay.

I developed this table myself just going through the test report and for cases where the time torque was applied to the U-bolt, I then looked at what pre-load level was attained through the actual strain measurement.

And all of these numbers can be found on the various pages in the test report indicated in the column farthest to the right.

If you don't agree with some of those numbers --

DR. IOTTI: No. I just hadn't thought about it in that fashion. That's why --

I'm a little puzzled; when you say a hundred pounds, for instance, we really never torqued -- yes, I guess we did torque at a hundred foot pounds, the ten-inch. So these are all ten-inch.

We do have data which has the scatter of the data incidentally. That may make it simpler.

DR. BJORKMAN: This is not just from one particular --

DR. IOTTI: Yes.

DR. BJORKMAN: -- test. These are from

several tests where that torque was applied.

DR. IOTTI: Right.

And likewise what you will see, what these charts are, is also from several tests. Some of them were from the torque versus pre-load test. Some of them were from the friction versus pre-load test. Some of tem were from the thermal test.

As you can see, this is forty-four inch. This is for the ten-inch where there is a lot more data. This is for the ten-inch stainless steel. And this is for the thirty points carbon steel.

So we did do -- Bill, you could help me out on this one, on the correlations between U-bolt pre-load and bolt torque. We knew we had a scattering of data.

DR. LaPAY: Yes.

DR. IOTTI: And depending on which way you want to look at it; from the standpoint of the stress, you want to use the top correlation that gives you the highest load for the corresponding torque, and from the standpoint of the opposite in terms of ability to retain pre-load, you want to use the minimum value.

Okay.

The Finite Element Analyses were essentially conducted on the basis of correlating an average value.

DR. BJORKMAN: That was another question I had, a separate question.

What was the actual relationship between pre-load and torque used in establishing the Finite Element numbers.

MS. ELLIS: What was that last question?

DR. BJORKMAN: What was the actual relationship between torque and pre-load used in establishing the value of pre-load and corresponding torque in the Finite Element Analysis?

DR. IOTTI: From the standpoint of the load, it was simply not because they matched the load as measured by the test.

And then the question is, is what is the torque to correspond to that load. Really it's the other way.

DR. LaPAY: And those are given in the Figures 1, 2, 3 and 4 in Attachment 1.

DR. IOTTI: So these were the ones that we decided were the most representative figures.

In other words, rather than the scatter of the data, if you go back then to the first figure, the pre-load versus test, those would be the nominal value of torque that the Finite Element Analysis would correlate to the load.

DR. BJORKMAN: Okay.

Can we try to correlate some of these numbers because that is what I was having some difficulty doing?

DR. IOTTI: Okay.

DR. BJORKMAN: In other words, when I looked at the Finite Element value for, let's say, for --

DR. IOTTI: Pick one. It doesn't matter.

DR. BJORKMAN: -- forty-six -- for ten-inch pipe I guess it was forty-six foot pounds.

I'm looking at the U-bolt Finite Element Analysis on page 7.

And there is a bolt tension of about 2.56 kips, which is related to a torque of forty-six foot pounds.

DR. IOTTI: Right.

Now, you should go back to that initial chart, and you should relate that kips. The 2 should give you abour forty-six or thereabouts. And that should represent the low forces.

DR. BJORKMAN: What I'm wondering is, was a straight line drawn through this data?

DR. IOTTI: No.

DR. BJORKMAN: Okay.

DR. IOTTI: The only one -- the only time we drew a straight line was for the purposes of deriving this coefficient of, you know, torque equal KTD, okay, or tension -- yes, torque equal KTD, to try and place in perspective what the coefficient K value would be.

In that sense I did try to fit lines. And what I used, as you can see right here, as a matter of fact, you can see right there the lines that I drew to bound the problem and get a range of the coefficient that you will have.

But in terms of the torque itself -- correct me if I'm wrong, Bill -- what you have tried to use is the -- you know, you match the low on the U-bolt and then to arrive at an equivalent torque. You tried to come up with the lowest torque that will give you that load.

Am I correct?

DR. LaPAY: Yes, I think so.

I was just --

DR. IOTTI: Or was it the highest torque?

DR. LaPAY: Well, I was just trying to get my own memory started.

DR. BJORKMAN: But basically the Finite Element test relationship between pre-load and torque comes from Figures 1 through 4.

DR. IOTTI: From the Figures 1 through 4.

MS. ELLIS: Could you repeat that last?

DR. BJORKMAN: Basically the relationship between torque and pre-load which was used in the Finite Element test for the Finite Element Analysis comes from Figures 1 through 4 of the

U-bolt test report.

DR. IOTTI: I would like to add, I guess, just so that we are all clear, the Finite Element Analysis does not use torque. They use load. And then having the load, you go back and derive the torque.

DR. BJORKMAN: I recognize there is no need to use torque --

DR. IOTTI: Yes. I understand. Sure.

DR. BJORKMAN: -- in the analysis.

DR. IOTTI: What do you want to do?

DR. LaPAY: I want to borrow those to refresh my memory.

Then I'll confirm what I have just said.

Bear with me a minute.

DR. IOTTI: We can go on to another question.

DR. LaPAY: Let's go on with another question and let me look at this and I ll come back.

DR. IOTTI: Or, you know, I'm sure, if he has more questions --

DR. BJORKMAN: Basically, the problem that I have here is with the great variability that I seem to obtain when I look at one hundred foot pounds of torque and the corresponding pre-load.

I see that for a ten-inch Schedule 80 carbon steel pipe I can get a little over 8,000 pounds at a hundred foot pounds of pre-load, and yet in another instance where I have the same diameter pipe, the same U-bolt, although it's Schedule 40, I can get as low as apparently 3,600 pounds of pre-load for the same level of torque.

And the problem that I have is basically, if you establish some relationship between pre-load and torque in particular for determining what your minimum torque is going to be for stability, and in an actuality, if you torque to that value, and in the field you get a --

DR. IOTTI: Lower.

DR. BJORKMAN: -- actual value that is lower.

DR. IOTTI: Well, we want to use the lower.

DR. BJORKMAN: And that's of some concern to us.

My question is, how has this been taken into account, this scatter, this variability.

The numbers from the test report, Figures 1 through 4, are not the minimum values.

Have minimum values been used? Or just how is it going to be put together, the relationship between torque and pre-load, and then the level of pre-load necessary to maintain stability?

DR. IOTTI: Okay.

I'm not so sure we will give you the full answer today because we are still in the process of determining the pre-load for all of the pipes.

But, in essence, our reply to you would be that the values that you see there are average values.

But when we establish the minimum pre-load that we will put in the field, we will go then to the top of the lower bound curves that from the test data would tell us you achieved the minimum load for the highest torque.

DR. BJORKMAN: You mentioned these curves are average values.

DR. IOTTI: Representative values.

DR. BJORKMAN: So Figures 1 through 4 are not from a single test.

DR. LaPAY: They may be the highest.

I'm looking at that. For instance, if you compare this one to Figure 1, and the --

DR. IOTTI: Well, it depends on what you are talking abcut in terms of --

MR. MANJOINE: Stress.

DR. IOTTI: What your interest was is not load so much as torque. Okay?

To us what is critical is you want to make sure that you actually forecast the lowest load that you can have for the torque.

So when I'm referring to minimum, it depends how you are going to look at those curves. The minimum is really the maximum torque for the lowest load. Okay?

DR. BJORKMAN: Let's take an example. I think it would be best to look at an example.

Let's say that for a given pipe size one establishes that you need a given level of pre-load. We have a minimum level of pre-load --

DR. IOTTI: Pre-load, right.

DR. BJORKMAN: -- that is absolutely necessary for stability.

To what value of torque will you then torque that U-bolt to ensure that that minimum level is actually obtained or something greater than that minimum level?

DR. IOTTI: It would have to be the -- let me try to define it so that it's unequivocal.

The lower bound curves, lower bound meaning those are the curves from the test data that we would predict for the same torque the lowest level of load; in other words, the curves that would be lying lower in the scatter data if you were to look at that, and you have a variety of curves all within a band, the lower band curve would be the one that would be used because that would tell you you have to put more of a torque to get the same load.

You are correct. What is important here is load, not torque.

That's what we would propose to use.

DR. BJORKMAN: Now, at the other end you also have the problem that also at that level of torque you could have obtained a level of pre-load which is more than double that in the field because of the data scatter.

MR. MANJOINE: That won't hurt you.

DR. BJORKMAN: Now the problem is --

DR. IOTTI: And I trust relaxation to help me. This is why I'm not concerned at the higher level. Relaxation is going to take care of it.

This is why we shouldn't see the high values of pre-load in the field or of torques.

If there were any at any one time -- and I made that statement in the affidavit -- relaxation will take care of itself.

So I'm worried at the low end. I'm not worried at the high end.

On occasion materials help you.

DR. BJORKMAN: Oh, yeah.

DR. IOTTI: This is one of them.

DR. BJORKMAN: That is well known.

MR. MANJOINE: I might add -- I might tell you another thing about torques.

You realize that when you torque a bolt, most of the torque goes into friction on the head or surface of the bolt and friction on the threads.

Only ten percent goes into -- on the ramp of the thread to make the load.

Now, the friction doesn't vary that much so that you usually get a pretty good curve.

But fortunately, if you have high friction and get -- for the same load you get the same torque, you get a lower load, you get less relaxation.

And so a lot of people put bolts like U-bolts, take it up to yield, which you can tell very easily because the rotation starts.

DR. BJORKMAN: You can ease it.

MR. MANJOINE: Yes. Ease it up to yield.

Same thing for automobile bolts, by the way. They always take them up to yield.

DR. IOTTI: Well, right now our intention would not be to take up to yield, but to take it up to the value that would be sufficient --

MR. MANJOINE: Sure.

DR. IOTTI: -- based on our test data.

Now, that would be my explanation to you.

I'm not worried about the upper end.

If I have a gorilla out there that overtorques, I'm far less worried about that than I am about the person not putting enough torque.

DR. BJORKMAN: You see, this is what basically has not been explained in these lower bound values.

DR. IOTTI: I thought I had in the affidavit but maybe I didn't.

I have to go back and review it.

I remember distinctly the statement -- making the statement that at the upper end relaxation will take care of me -- of the bolts.

But let me go back --

MR. FINNERAN: I think you probably talked about that but maybe not in this context.

DR. IOTTI: That may well be. Okay.

But if I have clarified now what our intent is, that's how we would arrive at that table.

DR. BJORKMAN: And what would be helpful to us is to really see what the, you know, what these lower bound curves look like.

DR. IOTTI: Well, we can send you this scattering of data and draw a lower bound curve through it which is what we would be using.

So --

DR. LaPAY: So far what I see is, what you said, is average.

DR. IOTTI: The one we represented there is average essentially.

MR. MINICHIELLO: Dr. LaPay, are these lower bound curves, the base the results of your first test, i.e., torque/pre-load, torque/pre-load, just keep measuring it, take it back down, do it again?

Is that what these lower bound curves based on?

Or are these curves based on a compilation of all the data --

DR. IOTTI: All the data.

MR. MINICHIELLO: -- in the test?

DR. IOTTI: It's all the data, to the best of my knowledge that is in the test.

MR. MINICHIELLO: In all the tests?

DR. IOTTI: In all the tests.

It includes friction. It includes thermal cycling, everything.

MR. MINICHIELLO: Then you take the lower bound of that?

DR. IOTTI: That's right.

DR. BJORKMAN: One point that I did not see on the curve was for the -- and just having looked at them quickly now -- the ten-inch Schedule 40 stainless steel pipe at a hundred foot pounds of torque, I have that you could achieve a pre-load of as low as 3,600 pounds.

And I don't know if that is shown on the graph.

DR. IOTTI: It should be. It should be.

DR. LaPAY: Which one now?

DR. IOTTI: Ten-inch, Schedule 40.

That's the one that we had the most data now.

Look at the scattering of data. I believe you would have gotten that from the space 66 thermal cycle.

So you got curves that go quite low.

Take a look at that and tell me if it's there.

One hundred pounds, you see how low you can go? There is that square, I think.

If it's not there, then I guess my retort would be where did you get yours because I think we identified everything there.

DR. BJORKMAN: The lowest value that I see on that curve --

DR. IOTTI: Forty-five.

DR. BJORKMAN: -- is a square data point at about forty-five hundred.

DR. IOTTI: Well, I guess --

DR. BJORKMAN: Let's take a look at my numbers.

DR. IOTTI: All I have to do is go and look at your numbers.

That's page 66?

DR. BJORKMAN: Page 66.

MR. MANJOINE: This number isn't after test.

DR. IOTTI: That could be because, see -- if it's the thermal cycling, it could very well be.

MR. MINICHIELLO: The average of the 3625 and the 3587.

DR. BJORKMAN: Yes. That's the average of those two numbers.

MR. MINICHIELLO: That is prior to pre-test.

DR. IOTTI: Right.

I guess you are right.

All I have to do is go back to the raw data of Westinghouse and see.

Bill, did we forget to plot that point?

DR. LaPAY: Which one?

DR. IOTTI: Page 66, the pre-test torque.

MR. MANJOINE: Was it at temperature?

DR. IOTTI: No. That was at ambient temperature.

DR. LaPAY: Page 66.

DR. IOTTI: Of course, we didn't plot the creep there. That's why you don't have it.

DR. LaPAY: Hundred foot pounds --

DR. IOTTI: Thirty-six hundred pounds, whatever, prior to creep.

DR. LaPAY: But that wasn't plotted here.

DR. IOTTI: It wasn't plotted. But we have to go back and verify that we can still stay with the fifty foot pounds because of this data point.

So I guess we will take your point and --

MS. ELLIS: Dr. Iotti?

DR. IOTTI: Yes?

MS. ELLIS: Could you just sort of summarize what just went on. I heard just bits and pieces of it.

DR. IOTTI: Well, Dr. Bjorkman pointed out to us that there was a particuar level of pre-torque at a hundred foot pounds which resulted in a load of only 3,600 pounds in the ten-inch Schedule 40 pipe.

When we look at the data that we had

accumulated for pre-load versus tension in the U-bolt, we had not included that point.

And we need to then include that point and see whether that affects any of the conclusions that we have drawn in terms of the minimum torque that we need to eseablish for the ten-inch pipes.

MS. ELLIS: Okay.

DR. IOTTI: And I can't do that right now. Obviously I need to go back and review it.

MS. ELLIS: Right.

DR. IOTTI: But he is correct. We did not include that point from the creep test in total evaluation of the bound.

MS. ELLIS: Okay.

DR. IOTTI: We will try to get you that information.

Just mumbling to myself. I'm sorry.

MS. ELLIS: Okay.

DR. IOTTI: That wasn't a statement.

Go ahead.

DR. LaPAY: I've just confirmed what you said. Those are the averages and --

DR. IOTTI: Well, I knew that.

I don't know whether there is a way to summarize this, but we understand Dr. Bjorkman's concern.

I have explained to him what Applicants intend to do, which is the intent to use the lower bound curves, that is, those curves that would predict the lowest tension in the U-bolt for the highest torque.

And then the question that remains to be assessed is that there happens to be a point that has not been included in the curves that we have that essentially bound the bolt torque versus U-bolt pre-load. And we have to go back and make sure that that point is included.

MS. ELLIS: What page was that discussed on?

DR. IOTTI: That would be -- of these questions?

MS. ELLIS: Yes.

DR. IOTTI: That is Question 12, Mrs. Ellis.

MS. ELLIS: Question 12. Okay.

DR. IOTTI: Dr. Bjorkman, is that sufficient on Question 12, or do you have additional concerns?

DR. BJORKMAN: I do have a question, but I think that it could probably be cleared up by seeing the actual curves of pre-load versus torque for all of the pipes.

The question centers around why the fact that a ten-inch Schedule 80 pipe should have such a high pre-load in comparison with the ten-inch Schedule 40 pipe when the diameter of the pipe is the same.

DR. LaPAY: You are talking Question 18.

DR. BJORKMAN: I was looking through to see if I had already asked that.

DR. IOTTI: I think so. Why is there a difference?

DR. LaPAY: That one I can't address. If you want to skip to 18 --

DR. IOTTI: It's a function of frition to a very large extent.

DR. LaPAY: It's not only that.

Let me just address Question 18 because

that's back to what you addressed earlier, Dr. Iotti, that really there is some confusion, misunderstanding, of what we mean in our affidavit by minimum torque value.

It's -- what we meant by minimum is that it is not "the minimum" torque value that has been established.

DR. IOTTI: You are referring to the Finite Element Analysis.

DR. LaPAY: Finite Element Analysis only.

It was the value, minimum value that we analyzed for.

There was no attempt to find a lower one. When we had found a stable solution that was below our given torque values, we were happy with that and we could look at the actual amount of tension that left -- was remaining in the individual U-bolts, and you could see, based on that comparison -- and those are given -- those loads are given in Attachment 3 on pages 6 through 9 -- you can evaluate the factor of safety associated with that loading.

Now, there are comments made about the forty-six foot pounds for the ten-inch Schedule 40 being different from that given for the ten-inch Schedule 80.

It happened to have been the case that, when we do the citative process, and we don't know exactly where we are going to stand -- where the solution is going to lead us until we do the analysis.

For forty-six foot pounds we had a stable solution that was -- that we were happy with.

When we happened to do the ten-inch carbon case, we happened to have used the lower pre-load.

And you can tell by what is remaining, when you look at Table 2-3, that the loads are still positive. It's a stable solution.

And comparing -- and there is a significant amount of load left in the ten-inch Schedule 40.

So it can go quite a bit lower.

There should be no attempt made to compare the forty-six foot pounds as "the" minimum load compared to the eleven.

If we went back and iterated down, we would probably approach something like the eleven foot pounds.

DR. BJORKMAN: Okay.

DR. LaPAY: Now, further on that question, you mention the statement in there on page 37 about the twenty-five foot pounds in the affidavit as being a factor of safety of one because there are statements made that the twenty-five foot pounds is the minimum torque for stability.

Now, what was interpreted there is, on page 37 -- and not to put words, but you can confirm it, Dr. Iotti -- that when we said that the twenty-five foot pounds for stability there on page 37 it was really meant as a minimum recommended value for stability.

DR. IOTTI: Yes.

DR. LaPAY: Now, if you look at the safety factor using the --

DR. IOTTI: We really expected that we would be stable at about ten foot pounds.

DR. LaPAY: Which is not --

DR. IOTTI: Maybe our wording is not correct.

DR. LaPAY: Yes.

DR. IOTTI: Twenty-five to us is the minimum recommended for stability. And that is why you see it later on again.

DR. LaPAY: Using nine as a minimum -- and I'm not saying that is "the" minimum, it could possibly be lower than that -- there you have a safety factor of 2.7 based on that.

So it's not one, but much larger.

DR. BJORKMAN: I think since the question of stability is a concern, it would have been nice in these tests to have seen the test drop back until the stability limit had been reached and basically instability had occurred maybe in rotating to know where you are or at what point instability does occur in your system.

DR. IOTTI: Well we did do that to some extent for the test -- the one specimen was subjected to the whole series of tests.

We had a limited amount of time to conduct testing and we came up with a test program that would hopefully answer as many questions in a given time as we could.

This is why for the ten-inch we know that the limit of stability is somewhere between twenty and thirty-five.

You know, to do what you are suggesting that we should have done --

DR. BJORKMAN: That was the vibration test.

DR. IOTTI: Yes.

DR. BJORKMAN: I was talking about the Finite Element Analysis.

DR. IOTTI: Oh, the Finite Element.

That would have required an optimization program essentially, to keep lowering the pre-load until such time when you found that under the applied load you could be considered to be unstable.

Or, in other words, as far as this is concerned, when these two numbers that you see on those tables, two point -- we go to Attachment 3 -- on those tables like Table 3, you see .09 kips and .57 kips. Those values are sufficiently low that you are beginning to think that you may be approaching that position of instability, okay, because you could lose contacts at that point.

But, you see, the minimum pre-load, that's eleven foot pounds.

Now, compare that with the ten-inch Schedule 40, where instead of having about .09 kips and .57 kips left, when you push it five degrees, you have 1.44 and 1.78 kips. And, of course, that is at a pre-load of forty-six foot pounds.

That leads you to believe that, if you want to lower those last two numbers down to the order of, you know, half a kip or less, or essentially zero -- this is what you are shooting for -- you could lower the pre-load quite substantially.

Those are the numbers -- am I correct, Bill --

DR. LaPAY: That is correct.

DR. IOTTI: -- that you will be looking for to look at stability.

DR. LaPAY: That is what I was pointing out earlier.

Those are the key numbers to determine --

DR. IOTTI: When some of those numbers go zero and negative, then at that pre-load level at least, the Finite Element Analysis would tell you you've got problems.

DR. LaPAY: And that's a measure of how much you had remaining.

DR. IOTTI: As you see for the four-inch Schedule, even at nine foot pounds, you got a lot remaining.

DR. BJORKMAN: Is this -- the problem is a non-linear one. I don't know the degree of non-linearity. You can't go directly --

DR. IOTTI: That's true.

DR. LaPAY: That's correct.

DR. BJORKMAN: -- from one to the other.

It may not be highly non-linear because it's just a question that maybe a few gaps opening or closing.

MS. ELLIS: I'm sorry. I can't hear.

DR. BJORKMAN: I said that the, since we do have a non-linear problem, the degree of non-linearity, however, does not appear to be too severe.

One could not directly take the results in Table 2, numbers one through four, and establish the minimum pre-load for stability.

But since the non-linearities are probably not severe because we are just dealing with a question of gaps, one could probably come very close to the minimum pre-load.

But what I'm interested in knowing basically is what kind of a margin on stability we really have.

That's really where this questicn is.

DR. IOTTI: Well, we think we generally hover at least on the factor of two and possibly more.

For instance, for a four-inch, if you look at the -- what we intend to use, which is twenty-five foot pounds versus what the Finite Element Analysis tells us we ought be using or we should be using, which is nine foot pounds, we are talking of the order of a factor of 2.5, 2.7.

For a ten-inch we know for a fact that we are least fifty over thirty-five because we actually have test data that shows at thirty-five we are stable.

But we also know that we can go lower than that.

If you look at, for instance, the ten-inch Schedule 80, it might tell us that we could be, in fact, stable at around eleven.

Again, there is where you got some uncertainty because we did run a test at twenty foot pounds which showed some instability in that ten-inch Schedule 40, not the Schedule 80.

So the Schedule 40 Finite Element Analysis would tell us that the pre-load that we need is probably higher than the ten-inch Schedule 80, not too surprisingly because of the difference in the flexibility of the piping itself.

But again here we are talking of the order of a factor of two margin between what we intend to apply and what would be the minimum required.

For the thirty-two inch, I frankly don't really know what kind of margin of safety. We expect that we have an awful lot there.

There I'm not so sure that any of these analyses even have any bearing in the final results because there the outer roundness of the pipes, the outer tolerance of the U-bolt, when you are talking these very large pipes and gigantic U-bolts aren't going to have a far more profound an effect than anything else that we have done.

And I guess I would have to state that as being an opinion, not a scientific fact.

MR. FINNERAN: Well, it's just that you have so much more surface area of contact between the U-bolt and the pipe.

DR. IOTTI: Five minute break, Mrs. Ellis. The reporter needs to reload.

MS. ELLIS: Very good.

(There was a recess in the proceedings.)

DR. IOTTI: We can go on without Bill for the time being.

MR. FINNERAN: Mrs. Ellis?

MS. ELLIS: Yes.

MR. FINNERAN: We are going to start again.

MS. ELLIS: Okay.

MR. MINICHIELLO: One of my concerns again about this lower bound and the idea -- ties in with Gordon's idea of safety factor is -- let's take, for example, page six of the test report.

DR. IOTTI: The test?

MR. MINICHIELLO: Not the test. I'm sorry. The analysis report, the Finite Element Analysis report.

If I look at the U-bolt leg four summary, we've been talking here and we have been saying with nine foot pounds the analysis predicted that we would be stable because we had a residual tensile stress in element 81 of 2.8 kips.

DR. IOTTI: Well, wait a minute.

You got to subtract from that residual the thermal eventually. I mean, you could be cold.

MR. MINICHIELLO: That's right.

DR. IOTTI: So --

MR. MINICHIELLO: That's the point I wanted to make.

DR. IOTTI: That's an obvious point.

MR. MINICHIELLO: Okay.

And that comes down -- that kip comes out to just blindly being linear.

DR. IOTTI: That's why we think the nine is the minimum because we subtract thermal that may not be there, you could end up at zero.

MR. MINICHIELLO: That's the only point I wanted to make is to be sure that when we are establishing things like safety factors we definitely look at --

DR. IOTTI: Yes.

MR. MINICHIELLO: -- pre-load plus push.

DR. IOTTI: Of course.

DR. BJORKMAN: The minimum pre-load in this case for the four-inch Schedule 160 pipe, the minimum pre-load from the Finite Element Analysis was established basically with only the push at five degrees --

DR. IOTTI: No.

DR. BJORKMAN: -- as the applied load.

DR. IOTTI: No.

I guess Bill has to be here but physically they ran the minimum pre-load plus the thermal plus the pressure. And then there they have to have enough margin left at the end that, when you subtract the thermal -- the thermal isn't there -- that final number may be close to zero.

The thermal here, if you want to use linear, is about 2.83 or so.

So it's around nine pounds maybe ten pounds, is what you would have if you were just to have the minimum pre-load plus the push.

DR. BJORKMAN: So that nine foot pounds --

DR. IOTTI: You couldn't lower this much more. And if you had much less than this 2.8, and if you take thermal out of the 2.8, you would find yourself, you know -- instead of 2.8, let's say you have 2 or 1.5, you could end up, if you had no thermal there to add to that total load, you could end up negative in which you would not be stable when you are in cold --

MS. ELLIS: Excuse me.

Who brought up that initial question. That sounded like someone besides Dr. Bjorkman.

DR. IOTTI: That was Joe Minichiello.

MR. FINNERAN: John Minichiello.

DR. IOTTI: Sorry, John.

MS. ELLIS: Okay. Thank you.

DR. IOTTI: I think we understand the question.

Our final numbers in terms of pre-load will reflect that.

DR. BJORKMAN: Just to clarify from my own point of view, the minimum pre-load is established with only a push at five degrees.

That's the only load that is applied.

DR. IOTTI: Push plus pre-load.

DR. BJORKMAN: Push --

DR. IOTTI: Plus whatever minimum pre-load.

DR. BJORKMAN: Push plus pre-load, correct.

And that establishes minimum pre-load. There are no other loads at the time.

DR. IOTTI: In terms of establishing a

minimum pre-load, yes.

DR. BJORKMAN: Yes.

MR. MINICHIELLO: That number is not reported anywhere in these analyses, however.

Just minimum pre-load plus push at five degrees, that number is not reported.

DR. IOTTI: That's correct.

DR. BJORKMAN: Minimum pre-load here is defined as the minimum load at which the test was run.

DR. LaPAY: The analysis.

DR. BJORKMAN: The analysis was run. Excuse me.

DR. IOTTI: That's correct.

It may or may not necessarily be close to the minimum which you require for functioning of the supporting in the intended manner.

MR. FINNERAN: Well, I think in the Table P where we establish minimum pre-loads, they only cover the sizes that we had tested.

So we had good correlation to do that.

MR. MINICHIELLO: Okay.

DR. IOTTI: What you naven't seen -- and that may be one of your questions -- and what we are not prepared to give you yet today is the final table for all the pipe size, the pre-loads.

We might as well tell you that right off the bat because we don't have it yet.

MS. ELLIS: Could you repeat that a little louder, Dr. Iotti?

DR. IOTTI: I was just stating that we still have not established a final tabulation that establishes the minimum pre-load -- let me not use the word "minimum" -- the pre-load that we intend to apply to the U-bolts, the cinched up U-bolts in the field.

And there will be a different value of torque for each of the pipe sizes and possibly schedules -- there may be variability with schedules -- that we will have to look at in the fields.

We have not established that final tabulation yet.

And since this is one of their questions, I was just telling them that there is no sense addressing it today because we don't have the answer.

MS. ELLIS: Okay.

DR. IOTTI: But they are essentially are being told who we intend to arrive at those numbers.

So -- Bill, while you were gone we were discussing this minimum pre-load and I was telling them that the minimum pre-load ultimately is what remains and shows stability in the Finite Element Analysis when you subtract out thermal and pressure because the system could be cold without thermal and pressure -- I mean it would be without pressure.

That was a concern that John Minichiello had.

And I told them, yes, we agree with that concern.

If we are done with this, could we take care of Finneran's question because he has an appointment at 1:30?

Could we go back to No. 7?

I don't want to cut you short, by the way. We want to address everything.

MR. FINNERAN: Let's ask them if that's -- I think we have addressed all of the --

DR. IOTTI: We addressed 12.

MR. FINNERAN: -- questions on pre-load.

DR. IOTTI: We addressed 18. And we addressed 19 by saying we are not going to address it.

So as far as I can tell, we are back to 7 now.

MR. FINNERAN: The question in 7 is what justification do we have for the limiting value of five degrees that we used in the -- for the test basically.

I think what's left out of the data that John or whoever wrote this provided is the fact that the as-built program at site actually surveys and shoots in the angularity of each of the struts in the field.

And that information is reviewed by

engineering.

So we rely on that aspect of our program to assure that we stay within the five degrees specified by the vendor for the struts.

If that's not satisfactory, it's an easy enough efffort for us to review that data for all of the particular U-bolts and tell you what the maximum angularity is on any one of them.

MR. MINICHIELLO: I was not sure whether part of your as-built program would cover that and that is why I wanted to ask this question here.

If there is someplace in writing in the as-built program where it states, when you do an as-built, you put the degree that the strut is offset on the drawing such that the engineer, when it comes back for vendor certification for any new loads or whatever, has to look at that offset and add to that the thermal offset and make sure that that is within five degrees.

That's what I wanted to get at.

MR. FINNERAN: Yes.

If you look at our as-built procedure or the as-built documents that are given to engineering, the field team that does the survey actually writes the angularity of the strut down regardless of what the angle is. They shoot it in. They measure this.

DR. IOTTI: That was discussed in the hearings as a matter of fact.

MS. WILLIAMS: It was, but maybe not quite to what we are driving at here.

I think we have seen ones that exceed the tolerance noted, like 8.6 degrees or 7. whatever.

MR. FINNERAN: Well, some of those do and by design they are okay.

MS. WILLIAMS: Right. We understand.

MR. FINNERAN: There is an angle where they are not binding in the bracket. The bracket is turned a certain way and the vendor has allowables for struts at an angle as long as you are swinging in the right way in the bracket and you are not swinging in a way that will bind in the ears.

MS. WILLIAMS: Right.

MR. FINNERAN: They actually have brackets that you can swing up to nine degrees.

So that's where you see those, that data.

MS. WILLIAMS: For some reason we were thinking that that was not being fed back to engineering necessarily.

MR. FINNERAN: Oh, yes. That's written on each survey drawing. That goes right back to engineering.

MR. MINICHIELLO: I have seen on many of the pipe's drawings angles noted.

MR. FINNERAN: No.

I'm talking about the drawings that the survey team actually takes to the field to write their information down on, and then those drawings go back to engineering. Okay?

MR. MINICHIELLO: These are markups?

MR. FINNERAN: Markups.

MS. WILLIAMS: Which may or may not be reflected --

MR. FINNERAN: Right.

MS. WILLIAMS: -- in the drawing --

MR. FINNERAN: Right.

And on occasion an engineer, when they have reviewed those, they have gone ahead and put that information on the drawing.

They have not always done that, but on occasion they have. And that is why you see three degrees two minutes sometimes on the drawing.

It's just a matter of putting that information on the drawing.

There wasn't any reason to do so, but he decided to do that.

MS. WILLIAMS: I think that is why because we're looking at the drawings --

MR. FINNERAN: Right.

MS. WILLIAMS: -- when we were doing our checks in the field.

DR. IOTTI: On to 8?

8 is the toughie today.

It's three-and-a-half degrees in both octogonal directions for thermal five degrees.

MS. ELLIS: Sorry. I couldn't hear that part.

DR. IOTTI: The question related as to

whether the strut was inclined only 3.5 degree in one octagonal direction or both.

And the answer is both.

MR. FINNERAN: Which gives you then on the diagonal a five degree ofset.

DR. IOTTI: Question 9.

The stresses randomly selected by Gibbs and Hill, please justify the Gibbs and Hill sample size. It refers to the mechanical piping stresses.

I don't have information yet from Gibbs and Hill.

My understanding of what they have done is they sampled I guess a substantial portion of the piping system of the size that we were concerned with at my request, and they selected the highest stresses out of that sample.

I had a call from Harry Mantell. He's been trying to reach me this morning probably to tell me what the total sample size was.

So I'll defer the answer to CYGNA. I'll provide you that by telephone or in writing, if you wish, what the sample size was.

Question 10 is: Justify the use of 1,500 pound amplitude in load on the normal vibration simulation test.

I have to go through a little narrative on this one because our intent was not to utilize 1,500 pounds.

Our intent was to utilize 4,000 pounds initially because, number one, 4,000 is the level of the OBE.

So we were trying to marry the long-term vibration test to the OBE test and do the whole thing at once.

Now, first of all, let me point out that the total number of cycles that the specimen sees during the long-term vibration test is at excess of ten to the sixth cycle.

So 1,500 pounds for ten to the sixth cycle is not an inconsiderable amount of energy that you are putting into that system.

The 1,500 pounds was ultimately arrived at by the level of tolerance of the equipment that we had.

At 4,000 pounds we were destroying the hydraulic actuators.

And we lowered that load until the hydraulic acutator was able to function for -- or it was perceived that it could function for as long as the test would be run.

So there is nothing magic about 1,500. We were trying to get the highest posible number.

There is something else though behind that.

When we arrived at that number, we asked ourselves what does that number represent and is it reasonable to expect that the normal level of vibration in the plants would be anywhere near that.

If you think of the vibration level in a plant normally it's of the order of a G or less, normal vibration level.

The deadweight of these pipes would give you about a thousand pounds, 1,500 pounds at 1 G.

So this is an upper estimate of what a normal vibration level could be in a power plant.

And as I say, when you sweep the

frequencies from 5 to 200 hertz at two octaves per minute you get ten to the sixth cycles or more during that five -- what was it? -- 5.4 hours.

DR. LaPAY: 270 minutes.

DR. IOTTI: 270 minutes; whatever it was that we ran this test at.

So we think that's a very severe test.

Now, the reason I'm prefacing that is that we swept all frequencies at this level.

Now, I'm getting ahead to the next question -- and I presume that question was asked by John Minichiello. It's about why is -- is a seismic test really representative of the seismic field especially when only rendered at one frequency?

Well, as you know, a seismic you are supposed to -- in terms of analysis you normally expect that it be about two hundred cycles or really a significant cycle.

We put in forty times nine -- three hundred and sixty significant cycle at 8,600 pounds at one frequency.

Other cycles that you have at other

frequencies we feel were encompassed by the previous test.

So we had to compromise.

And the reason we had to compromise is that we could never get that kind of a load with the equipment that we have into the pipe as the high frequencies.

And initially we were going to run that test as a minimum at two frequencies, one which corresponds to the frequency at which the response spectrum has its highest amplitude, which happens to be around nine hertz, and the other one at the natural frequency of the system when it was being excited, which happened to be 75 hertz.

The problem with that is that we couldn't get the force at 75 hertz through the system.

The best we could do was about 1,500 pounds.

Well, at 1,500 pounds we had already swept all frequencies anyhow.

So the combination of the two tests is what leads us to believe that we've adequately bounded any of the seismic events that the ten-inch line could see.

So I guess I have answered Question 10 and 11, or at least have attempted to answer Questions 10 and 11 at the same time.

MS. ELLIS: Could you sort of summarize that last part?

DR. IOTTI: The last part is that the combination of the long-term vibration test and what we call the seismic test, which was run at one frequency, was believed by the Applicant to be encompassing of the worse seismic excitation that the particular specimen could see in the plant.

(There was a brief recess in the proceedings.)

MS. WILLIAMS: On this particular issue I don't think there is anything additional you can provide us with today, but we do want to think about it.

And we are -- I don't think there is anything else --

DR. IOTTI: Go ahead and think about it.

MS. WILLIAMS: -- unless you want to walk back through it again. You are welcome to. But we can't think of any other questions to ask on the subject to clarify it for us at this point in time.

DR. IOTTI: I understand.

I don't want to add anything because that is just going to confuse the issue.

And I think if you think about it, you will come to the same conclusion we did.

So --

Just bear in mind that you are not testing a piece of equipment here. You are testing piping, and so the difficulty in simulating piping, okay.

There is no question that the strut sees the whole that it sees and I don't think that's your problem. It's what might be coming back from the piping. That's possibly the only thing that could possibly be not quite falling into place in your mind is that. But I know no way to put a piping system or at least an adequate portion of it in the shaker table.

So that's enough said.

We think we have bounded the problem.

On to No. 13, which is another one that Mr. Finneran will address for the Applicants.

MR. FINNERAN: Well, I guess have a problem.

DR. IOTTI: Hold it. I'm sorry.

Mrs. Ellis, are you on?

MS. ELLIS: Yes, I'm here.

DR. IOTTI: Okay. We didn't want to start without you.

MS. ELLIS: Thank you.

MR. FINNERAN: If you think about how this -- how these U-bolts are painted in the area where the nut is on the U-bolt, you have a backing plate with a U-bolt protruding through the hole in the backing plate and you screw the nut down on to that and then the retaining nut goes down on top of that, I don't see how, if you go in and back those nuts off, that you have any paint. And then those are painted.

There is no paint that gets between the interface, between the backing plate and the nut or into the threads that the nut will be on when we do the retorque again.

So I don't see how paint in this question even enters into the problem.

MR. MINICHIELLO: When you go in to do your field test, are you going to back the nuts off --

MR. FINNERAN: We have to back the nuts off and break the paint bond and then we have to completely retorque the nut back down.

There would be no way for us to go in and just put a torque wrench on the nut because we're testing the bond of the paint there as well as the --

MS. WILLIAMS: And then --

MR. MINICHIELLO: I think that's what --

MS. WILLIAMS: Yes.

MR. MINICHIELLO: I think that is what we were getting at.

MR. FINNERAN: The question is not exactly worded that way.

What we plan to do is we back the nut off, break the paint bond and then, when you torque back down, there is no paint involved in the issue.

MS. WILLIAMS: You've got clear threads.

MR. FINNERAN:: Right.

MS. WILLIAMS: Yes.

And I think that had you torqued enough in the first place, you've still got clear threads within the backing plate.

DR. BJORKMAN: In the crosspiece.

One of the reasons that the question was also raised is that I think somewhere in the affidavit it talks about the fact that the torque valve may be affected by painted versus unpainted threads.

MR. FINNERAN: Oh. That was on our field sample.

DR. IOTTI: Yes. The initial field samples we had some values that showed to be higher than what you might expect to be and that could possibly be the effect of paint.

DR. BJORKMAN: How would the field sampling differ from applying the proper torque in the field to establish the minimum pre-load?

DR. IOTTI: In precisely the way that John referred to.

In a field sampling, when they first went through, they didn't bother to take the -- you know, to back the nut and then torque it back in because they were trying to determine what was in there.

Here our intent is to put in a specific value.

So, to us we can back it off and then retorque it to the value that we want.

There we were trying to establish what might be there. And the moment we backed it off, we were invalidating what was there.

So that was the difference.

DR. BJORKMAN: Okay.

DR. IOTTI: What we said is that we couldn't really have great confidence in some of the torque values, particularly in the higher values, because paint could lead us to a wrong conclusion.

DR. BJORKMAN: So you anticipate basically that there will be no problem with painted threads in the direction of torque; in other words, even after painting these threads. There are some unpainted.

MR. FINNERAN: Well, don't forget, the nut that you are really interested in is the first one. There is another nut that is run down on top of that just as a retaining nut.

So when you take that nut off, and the one we are really interested in is the next one, you have a full nut of clear threads on top of that one. That has no paint on it either.

DR. BJORKMAN: Because it's locked -- because it's protruding through the whole crosspiece.

DR. IOTTI: That's right.

MR. MANJOINE: Yes. And when you take --

DR. BJORKMAN: And these crosspieces are not overdrilled to the point where you can get the paint in?

MR. FINNERAN: Oh, no.

DR. B[illegible]RKMAN: Okay.

MS. ELLIS: I'm sorry. All that faded out.

MR. FINNERAN: The crosspieces are not overdrilled to the point where you would get up in the threads underneath the nut and over the nut.

MS. WILLIAMS: Well, it's tight anyway. You wouldn't be able to get --

DR. IOTTI: You can't because the bolt protects the backing plate.

MS. WILLIAMS: The nut.

DR. IOTTI: The nut --

MS. WILLIAMS: Right.

DR. IOTTI: -- protects the backing plate.

It's only painted from the back end.

You follow what I'm saying?

MS. WILLIAMS: That's like saying you've got clear thread.

MR. FINNERAN: I think Gordon is thinking about paint coming up from underneath.

DR. IOTTI: Coming up from underneath. Of course

MR. FINNERAN: I don't think so.

DR. BJORKMAN: Actually maybe it would be --

I understand what nut is torqued, and I'm worrying about the paint coming in from the opposite end.

DR. IOTTI: I understand. From the plate.

DR. BJORKMAN: Correct.

DR. IOTTI: From the plate.

DR. BJORKMAN: There is no nut there.

DR. IOTTI: That's right.

So if they paint that very heavily and slop a lot of paint in there, you say that you could have threads that are covered with paint.

DR. BJORKMAN: Only in the case where you have a very large hole relative to the --

DR. IOTTI: That's correct.

DR. BJORKMAN: -- shank in this case.

DR. IOTTI: Not only that, but even if that were the case, that paint would stop where the nut then is.

Now you are backing it up. Okay? You could conceivably have maybe the first thread. If you had that hypothetical case of a very large hole with paint going through it, after you then torqued the nut back in, you may have a thread which may paint at the very end.

And that would be the worst case.

MR. FINNERAN: I doubt it seriously based on --

DR. IOTTI: There is not that much clearance between the holes of the backing plate and the U-bolt itself.

It would be with great difficulty for somebody to get paint in there.

So we don't anticipate a problem at all.

Question 15?

Really 14 and 15 kind of go together, right? -- having to do with the friction test and the variability of friction versus pre-load.

Bill, you got some stuff for that?

DR. LaPAY: Yes. We have something there that I think directly answers those.

Let's see if the copy machine was working.

This is for Question 14 which we can pass on.

And this is for Question 15.

MR. FINNERAN: Mrs. Ellis, Bill LaPay from

Westinghouse is passing out some sheets with some information on them concerning Questions 14 and 15.

We will be sure you get copies of these.

MS. ELLIS: Very good.

DR. IOTTI: That is good enough, Bill.

DR. LaPAY: Okay.

I think that addresses the Questions 14 and 15, I believe.

DR. BJORKMAN: Yes.

DR. IOTTI: Okay.

The next one is Question 16 which refers to Figure 26 of Attachment 1 to the affidavit which is the test report.

DR. LaPAY: That is one I got confirmation on the numbers too.

DR. IOTTI: Which one?

DR. LaPAY: Figure 26.

DR. IOTTI: Yes. The one that showed the pre-load actually --

DR. BJORKMAN: Increase.

DR. IOTTI: -- increase.

DR. LaPAY: Let me just say a few --

DR. IOTTI: I think before we do anything else, you ought to look at the vertical scale.

DR. LaPAY: That is exactly what I was going to say.

DR. IOTTI: Okay?

Because that's -- those are large numbers, and that's probably within the uncertainty of the measuring device.

DR. LaPAY: The variation is only 2.7 percent.

And also it's important to note that throughout the temperature wasn't constant too. There was some variations of some three to four degrees.

And that's enough to cause some slippage in binding in this situation back and forth.

DR. IOTTI: You are still trying to get that additional information?

DR. LaPAY: Oh, yes. I'm still trying to get it and I will.

DR. IOTTI: As far as we know, the gauges were compensated.

DR. LaPAY: They were compensated, yes.

MR. MONJOINE: Within at least one percent.

DR. IOTTI: Yes.

DR. LaPAY: Also it's important to note that between zero and 12 there were no points taken. There is only the zero point and then the point at 12.

So you really could have a lot of variation within that band during hat time interval.

DR. IOTTI: The other point that is not quite clear is wnat point was really zero.

DR. LaPAY: Where in the thermal cycle --

DR. IOTTI: Where in the thermal cycle was zero taken.

See, it's within -- to the best -- all we could conclude from that data is what was in the scatter of the errors that you can expect when you do this type of measurement.

In other words, it's flat. That's the best I can do

DR. LaPAY: And high temperature

compensation gauges were used.

DR. IOTTI: Here you didn't even need high temperature compensation gauges. The temperature is pretty low.

DR. LaPAY: Sure.

MS. ELLIS: I really can't hear any of what has been going on.

DR. IOTTI: Oh, we are arguing about Figure 26 which would indicate, taken literally, as you thermally cycle the -- I'm sorry. This is not a thermal cycle. This is the creep.

DR. BJORKMAN: Creep.

DR. IOTTI: Creep test --

MR. MANJOINE: At temperature.

DR. IOTTI: -- at temperature, where, in fact, the pre-load or the bolt pre-load increases as opposed to any degrees.

And what I'm trying to state is that in reality the only conclusion that should really come from this chart is that there is no change because it is not entirely clear when the zero cycle pre-load was taken.

And the next point was taken at 12 hours. And in between variations that are not recorded here are not shown.

So I guess that is about the only explanation we can offer on that.

We are trying to get additional data, raw data. But I don' think that is going to give us any more information than we have right now.

The strain gauges were temperatures compensatd.

We essentially saw no significant change in the pre-load during that period of time.

That leads us to Question No. 17 which is a typo. The factor of 2 should be there.

And I think that is the end of the questions that were written --

By the way, the factor of 2 has no effect. It's just a typo.

The calculation of the friction factors was done correctly irrespective of that.

That's the end of the written questions that we received from CYGNA. It is my understanding that they may have a few more.

I guess now we will attempt to answer whatever additional questions they may have.

MS. WILLIAMS: Should we take a couple of minutes to go through your notes?

DR. BJORKMAN: Yes.

DR. IOTTI: They are suggesting maybe a ten-minute break so that they can --

MR. FINNERAN: Organize.

DR. IOTTI: -- formulate their questions, organize their questions.

So if you want to go and get some lunch or something, Mrs. Ellis, --

MS. ELLIS: In ten minutes?

DR. IOTTI: -- we should be back here --

MR. MANJOINE: She's finished.

DR. IOTTI: Oh, you're finished lunch?

MS. ELLIS: No.

I said in ten minutes did you say?

DR. IOTTI: Well, it's quick lunch.

MR. FINNERAN: Pick a quick sandwich.

DR. IOTTI: Right.

(There was a ten-minute recess in the proceedings.)

DR. IOTTI: Okay, Mrs. Ellis.

We are ready to resume.

MS. ELLIS: Okay.

MS. WILLIAMS: The good news is we don't have further questions.

DR. IOTTI: You mean we had this long break.

MS. WILLIAMS: But -- well, you saved two hours for fifteen minutes.

DR. IOTTI: Okay.

MS. WILLIAMS: I do want one clarification.

The tabulation that you have in your book, Bob, does that include all sizes of pipe, or is that just ten-inch?

DR. IOTTI: For what? For the inside and outside?

MS. WILLIAMS: For Question 1.

DR. IOTTI: It has all: Four-inch, ten-inch Schedule 40, ten-inch Schedule 80 and thirty-two inch.

MS. WILLIAMS: Okay.

And we will get a copy of that.

DR. IOTTI: Right.

I'll put it in the mail tomorrow morning.

To whom would you like that I send it to? To yourself? To Gordon?

MS. WILLIAMS: Just send it to me because then I distribute it to everybody.

MR. FINNERAN: We'll have to send copies --

DR. IOTTI: We send out copies to everyone including you, Mrs. Ellis.

MS. ELLIS: Okay.

MS. WILLIAMS: And that's it.

DR. IOTTI: I propose that we then view the tape.

Mrs. Ellis, what we are about to do is turn on the videotape recorder.

We have recorded the actual test that was done on the ten-inch Schedule 40 stainless steel pipe subjected to a 10,000 -- you know, what we call the seismic test.

And we will be viewing the response of the specimen to both the excitation, the large excitation at 9 hertz as well as the excitation at the higher frequency but lower load.

MS. ELLIS: Okay.

I don't know if they'll want to do this, but would that be available for us to view at some point?

DR. IOTTI: Sure.

We would have to send you the tape.

Incidentally, the Commission has also viewed the same tape. So there is --

MR. FINNERAN: This was the same tape, Mrs. Ellis, that we discussed in the August 8th and 9th meeting with the NRC in Bethseda. And the viewing of that tape is on the record of that meeting.

MS. ELLIS: Okay.

Are we going to go ahead and have further questions following the tape?

MR. FINNERAN: No.

DR. IOTTI: I guess possibly. I wouldn't exclude it.

MS. ELLIS: Okay. I'll just stay on.

MR. FINNERAN: Okay.

DR. IOTTI: Okay.

THE REPORTER: I am assuming that you don't want me to record this, do you?

DR. IOTTI: I don't think -- I guess --

Let me state for the record that we don't consider it necessary for the recorder to record any of this viewing of the tape because the Commission has seen it and it's a matter of public record already.

We will resume recording if there are questions following the showing of the tape.

MS. ELLIS: Okay.

(A videotape was viewed by all those assembled.)

MR. FINNERMAN: Do you have any questions about the tape?

If not, we will go off the record.

DR. IOTTI: CYGNA has no further questions so we are ready to go off the record, Mrs. Ellis.

MS. ELLIS: Okay. Very good.

DR. IOTTI: Thank you very much for listening.

MS. ELLIS: Thank you.

DR. IOTTI: Have a good day.

MS. ELLIS: Thanks.

(At 1:35 o'clock p.m. the proceedings were concluded.)

STATE OF NEW YORK)
) SS.
COUNTY OF NEW YORK)

I, ROY A. SELENSKE, a Certified Shorthand (Stenotype) Reporter and Notary Public within and for the State of New York, do hereby certify that the foregoing Pages 1 through 155, taken at the time and place aforesaid, is a true and correct transcription of my shorthand notes.

IN WITNESS WHEREOF, I have hereunto set my name this 17th day of September, 1984.

Roy Selenske

ROY A. SELENSKE, C.S.R.