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UNITED STATES OF AMERICA

NUCLEAR REGULATORY COMMISSION

In the Matter of:

QUESTIONS ON SUMMARY DISPOSITIONS
FILED BY TEXAS UTILITIES ON
COMMANCHE PEAK

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UNITED STATES OF AMERICA

NUCLEAR REGULATORY COMMISSION

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In the matter of: :

TEXAS UTILITIES GENERATING COMPANY, *et al.* : Docket Nos. 50-445, 50-446

(Comanche Peak Steam Electric Station, Units 1 and 2) :

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7735 Old Georgetown Rd.
Room 1713
Bethesda, Maryland

Wednesday, August 8, 1984

Hearing in the above-entitled matter convened at 9:20 a.m.

APPEARANCES:

On behalf of the Applicants:

PAUL CHEN
WILLIAM A. HORIN, ESQ.
ROBERT IOTTI
JOHN FINNERAN
DAVID WADE

On behalf of the NRC Staff:

J. BRAMMER
HANK FLECK
DAVID TERAO
JOHN FAIR
SPOTSWOOD BURWELL

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P R O C E E D I N G S

MR. BURWELL: My name is Spot Burwell. I am the project manager for Comanche Peak of the NRC. The meeting today, the purpose of the meeting today is to afford the staff an opportunity to ask questions on some of the summary dispositions that were filed by Texas Utilities during the months of June and July. With that, we have proposed a general agenda dealing with four items as the lead off item on summary disposition. These items are in order. The motion for summary disposition on U-bolt. The moticn for summary disposition on force distributions in actual restraints, motions for summary dispositions on local displacements and stresses, and the motion for summary disposition on wall to wall floor to ceiling supports.

If there are no further comments, between the members present, we will proceed to the first agenda item concerning U-bolts.

MR. IOTTI: Spot. Did I miss something or are you including the stability which was tenatively listed as being part of the agenda in the U-bolt general studies?

MR. BURWELL: Oh. Good point. It was not my attention to exclude anything, any of the others. What I am doing with the agenda I proposed right now was to

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just establish the first full item that I have got, okay. We will get around to the stability and a number of the others.

MR. FLECK: Spot, should I announce who I am?

MR. BURWELL: Yes, if you would please.

MR. FLECK: I am Hank Fleck from E-Tech. I have some general and also some specific questions on the U-bolt. Why don't I just go through them, and the ones that are general can be provided later with some answers or whatever. The specific ones, we can address as we go along. Turning to the affidavit portion of ascension data U-bolts, page 11, the A36 carbon steel will relax to a state where at preload stresses will be appoximately 1/2 yield. In the report, I really have not seen any material data specific to this material that identifies how the material reacts. I would like to get some clarification, either with tests or analysis of the material itself that shows the relaxation is approximately 1/2.

MR. IOTTI: There are no, you know A36 is kind of, the garbage in the news.

MR. FLECK: Right. I tried to find some livelihood reports and I didn't...

MR. IOTTI: The only data that I was able to find, I brought it along for another reason, but I

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happen to have it with me. I went back. There is a calculation from the STM60. I'm sure you are familiar with this document, okay. There are several materials in this calculation which have properties which are similar to A36 steel, in fact, they could well be A36 steel as far as I know. Out of this whole table, there are about five or six of them, that are very close to the specification that STM puts in for the calculation of the properties of A36.

Unfortunately, the data is scant, and we have talked to a lot of the materials experts on that also. We have got from them that there is simply not much data available with relaxation of A36.

MR. FLECK: I ran into that same problem.

MR. IOTTI: So, what I did was went back and found anything that could be as close as possible to A36, and I found material number 2 here is about as close as I can make it.

MR. BURWELL: May I interrupt, would you move the mic over closer to you?

MR. IOTTI: I am quoting a particular document right now which is entitled <u>Calculation of Stress Relaxation Data for Engineering Alloys</u>. It is ASTM data series publication DS60. This document is rather large, but the only section that is of concern here is the one

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that pertains to carbon steels. What this shows is that there is very little data applicable to A36 type steels, but there is some. In particular, as you look through this calculation at or near room temperature, it certainly would be below the 4/10's of the nothing temperature of the material which is kind of critical. We have certain material which exhibit a loss of stress, I should call it. We will approximately have within the first hour, after being strained to 2/10's of yield, not to 2/10's yield of the percent, which is...strain constitutes to yield.

We believe this characteristic exists because of any material in this relaxation is really micro strain relaxation conversion from lasting to, you know, lasting type strain which occurs as you go through the proportional limit someplace along the U-bolts. Okay, it is not universal to any other U-bolt, but because of that, and because of the way that these stresses are measured, you, in fact are reducing stresses from strained measurements. That can lead you to a, to the kind of a conclusion that you are nowhere near the proportional limit would in fact, at some locations of the U-bolt you have exceeded the proportional limit of the material. You may be cycling back and forth to that proportional limit and this is what is causing some of

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this relaxation.

This material was not cycled in loading, this material number 2 that I am referring. So, this was uni-axial strain, and it was strain to 2% of the strain, 2/10's of a percent strain. It exhibited a loss of stress to about half of its original value in the first hour. After ten hours, it was essentially still at the same point.

MR. FLECK: Do they also include maybe what the key facts that you made, or may not lose from the fact that you are relaxing?

MR. IOTTI: Well, there is no mention of that. I did find another material, material 20 which was cycled. Now, the problem with material 10 is it will cycle at a temperature of about 900 degrees Fahrenheit. Nevertheless, the cycle did not seem to have as profound an influence in strain. That is why I was let to believe why we wouldn't have much of a problem. The other thing, is we did do cyclic testing here, so we can make some conclusion from that.

Let me get ahead of myself. One of the reasons I ultimately believe in this half of a yield is for the seismic tests that we have run, full seismic tests, originally they have come back to me telling me they had loaded 7,000 lbs. That 7,000 pounds would have

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been just about half of the stresses calculated. You know, on a linear basis. They were affording me about a 10-15% relaxation. I can't beat. The fear is shot. Are you sure you have that. We confirmed that the load, in fact was 8500 lbs., and that is where you get the 10-15% relaxation. So, there is some additional evidence to us. This material behaves almost like a 3 or 4 type stainless steel.

MR. FLECK: I wonder, could we get some documentation on that for the amount that you have?

MR. IOTTI: Sure. I can send you the whole history. I can send you the original data from Westinghouse which afforded 7,000 lbs. What you see here is actually the revised data that was prompted by my question to them. I said, look. It can't be right. If I am correct, you can't be. And, as you see, the data was then subsequently changed and they found the load to be much higher.

So, there is no hard and fast evidence of that. On the other hand, we also ran to make sure, because we couldn't understand some instances why were getting relaxation, for instance, on the load of the U-bolt. If the U-bolt stays totally elastic, don't exceed the proportional limits, why should we have a relaxation. We ran a finite element analysis, just do

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the free load, where the U-bolt modeled as elastic elements as well as the pipe.

As it turns out, as you load the U-bolt right at the deflection point, your bending stress out of the outer fiber exceeds to proportional limit. Everytime you do so then you load or preload in a cyclic mode, you actually go through the proportional limits; which indicates that that is probably where you are getting the relaxation even though you may be at reasonably low total stresses. This was the curve, the stress strength curve we used for the computer model. We only ran free load. It wasn't something you were aware of, I just did it last week.

Essentially, this is how much loading you get. So, when you attempt to cycle about this, you don't start up.

MR. FLECK: Right. In most material you will tend to do that. In some materials, you will tend to lose some of the fatigue problem. In other words, you don't get a free ride. Somewhere along the line, you are going to pay for it.

MR. IOTTI: Right.

MR. FLECK: Do you feel that most of the relaxation is occurring in material? Could some of it be slopped in the threads?

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MR. IOTTI: There has been some of that for the preload that we have been using in the plan. We don't expect, really, any relaxation.

For the type of testing that we did, most of the relaxation was in the material thread, or in the thread. There was one particular test where, in the four inch schedule 160, we actually exceeded yield in the thread area. Clearly, we had a lot of relaxation at that point.

MR. BURWELL: A point of clarification...You said, in the material as oppossed to the thread. You mean, in the bend of U-bolt as opposed to the...

MR. IOTTI: It is eternal to the material. It is actually a conversion of elastic strain the material has into some interelastic strain which does not recover easily. Essentially, you have lost.

MR. FLECK: Well, if I read Bob right, he could take a tension specimen and demonstrate the same thing. Unfortunately, this material was something that most people haven't tested. I tried to find data like that in the library that I could say yes, I believe it, or no, I don't have enough data to believe it or disbelieve it.

MR. IOTTI: The problem with this material is there is so much of it. It is so different. A36, has

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got a large carbon spread all the way up to .25 percent. You get up in the maganese and the silicon again, you can be up to the 15 percent, or you could be down to as little as .2. Each of these materials behaves slightly different.

MR. FLECK: The other thing that you have mentioned now, as you have indicated, in all probability, you don't believe you will be even with the torques that applied will even be close to you?

MR. IOTTI: Well, that is correct. You see, the way we ultimately will end up at the plant after the inspection to make sure that the minimums are to be acquired. That minimum torque has been determined by finite element analysis to be high enough to cause the problem of relaxation.

So, we have to be careful. We can't be too high in the freeload, because, if we are too high in the freelcad the material will relax down to where it wants to be to start with. So, our concern was if it relaxes down to where it wants to be, do we have sufficient preload maintained as pure stability.

Our answer, the finite element analysis was the test. We indicated that that would be the case.

MR. FLECK: Okay. I guess then, I would like to see two things. One, the limited data you have on

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that, and also verification that the maximum applied torque would be on the yield under whatever, 90% of yield. So, the question would disappear as far as to whether A2A, or A36 whatever batch you had, whether it was the worst or the best wouldn't really have a concern as far as relaxation because you are never really going to be close to yield.

MR. IOTTI: The problem is you do relax if you are not. That is the point that I want to make. In reality, that is not really the case. What is happening, I will use this as an illustration. The typical plot of financial data A36. That is from the Bethlehem Steel. We are due to get some more, but I haven't received it yet, because there is quite a spread. What happened is all the stresses that we deduce here by finite element analysis by test, in particular by test. Make certain assumptions. If you are smart, in the finite element you actually monitor the behavior to proportional limits. So, the finite element turns out to be, perhaps, more accurate than the test. In the test, what is generally done is assumed that the yield strenght is 36,000, and the yield strength by the finish is 2,000 of a percent of strain. So, you can draw the line in between.

Then, you deduce you are at half the stress

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from the strain. From reality, that strain would put you way up to the proportional limit. So, you in fact are getting relaxation before you think you are. One of the reason, one of the reasons this halfpoint comes out is that when you go to the strain that causes the half point below the proportional limit, I mean you are not getting that...

When I say half the yield strength, I mean in the neighborhood. It could be 40%, it could be 60%.

MR. FLECK: The material, too, could be anything from 36 up to 60.

MR. IOTTI: That is correct. In fact, the material number two, the one that I quoted you has a yield strength of about 52. It drops to about 26 in one hour. What I wanted to do, and I didn't have enough data to bring for this meeting. I am still expecting it, and if I get it, I will forward it to you. It is actually plot all of the A36 and show by test if you take the strain that corresponds to half the stress along the true stress strain curve, whether in fact you have billowed the proportional limits.

The only theory that I see as holding water is the microstrain conversion from elastic to interlastic. We are not in a crete region, not that I can see. We are at too low a temperature for that. So,

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this is the only cause that we can think of would make the material relax at those low temperatures and at these low loads.

In addition, what I will try to do is, there is actually data from all of the tests, that if you read it in a certain manner will tell you, the material had in fact been stretched over half of the yield. You know, this is relaxed down to where this would again be half a yield. Every one of these tests, where the tests at any point unit was over a half a yield, ultimately that brought...

MR. FLECK: That would be good to see then.

MR. IOTTI: I'll send that to you. Some of it is in these plots, but they are not that specific.

MR. TERAO: Unfortunately, I guess what Paul was mentioning, do you have enough data there to show the spread so you can get a statistical basis, yes what we have would be a minimum or whatever. I don't know whether with the amount of testing you have done...

MR. IOTTI: We have done quite a bit of testing, remarkably in spreading data. Now, we have to be careful now in talking about spreading data. With regard to proving that everytime we see that yield at any point that we experience a relaxation which was proportional to how much over the half a yield we were.

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We have a fair amount of data. I can all put it on one sheet. Some of it comes from the freeload test, some of it from the thermal cycling test, some of it comes from the seismic tests.

MR. TERAO: I would have one question here, now. What means would you have to establish for that particular U-bolt what yield of that A36? So, there we would have a big unknown.

MR. IOTTI: That's the bit unknown. There is nothing I can do about it. Well, I may be able to phrase back to the heat of that particular U-bolt through the ITT Grinnel.

MR. TERAO: If that is possible, that would be...

MR. IOTTI: I think we can do that. We think the yield is around 42, this material. We asked, but we don't actually have the data.

MR. TERAO: Is that ascertable material?

MR. IOTTI: Yes.

MR. TERAO: So, you could go back to ITT and have them ascert, tell you what potential is allowable in the yield.

MR. IOTTI: That is correct.

MR. TERAO: We would need that for a base.

MR. IOTTI: None of these are NPSI's. The ones

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that we tested, none of them were, they all came up from ITT, right?

MR. TERAO: They came from the site.

MR. IOTTI: But were they provided by ITT?

MR. TERAO: I don't know. We have the data.

MR. FLECK: I just have one question. When you mentioned the variability in the carbon content, I believe you said it varys from about .2 to .8%.

MR. IOTTI: No. No. Not the carbon. Hold on, I can tell you. ASTM specifications give you certain limits. Carbon can be a maximum of .26 percent. BUt, it could be as low as .04. Maganese can go as high as .8, and silicon, again, can be as high as .15. Anything in between can be construed to be 36 type material. In addition, there are other impurities that are permissible. So, it is truly garbage material. I think that is one reason why very little data is available, when you test. You test this A36 with the next A36.

MR. FLECK: I don't know whether I would characterize that as garbage material or not. I think there is a problem with this particular non-headed bolt that it originally was formed under the A307 spec, and then for under the A307 spec which I would characterize more as a garbage spec and A36 for non headed bolts then he said for anchorages would have to be

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requalified for A36, which has much more requirements say or tensile tests or chemical requirements than A307.

MR. IOTTI: Well, when I meant garbage. You are right, I shouldn't use that word. To me the material has so much variability. It is almost impossible to test on a global manner to determine what it is going to test aside. Because you would have to essentially test every chemical composition.

MR. TERAO: Why do you say this was made for A307?

MR. FINNERAN: A307 is the base expect for bolts.

MR. TERAO: Not bolts, U-bolt material.

MR. FINNERAN: I couldn't hear what you were saying.

MR. FINNERAN: A307 is the base expect for bolts. There is also the requirement, A307 for non-headed bolts which need upgraded to the tensile and chemical requirements of A36.

MR. FINNERAN: Our U-bolts, I don't think are made from a bolt spec. They are made from A36 rod. They are not made to a bolt spec. It is an A36 rod that has been threaded and formed to a U-bolt shape.

MR. TERAO: In either case, whether they were

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formed, I guess I was think in terms of the Richmond inserts also. They have A36.

MR. FINNERAN: No. That is A36 rod that has been threaded also.

So, neither the notes in the Richman inserts or he U-bolts are made to an A307 spec.

MR. TERAO: I just noticed there was some confusion in the record about the difference between A307 and A306. I thought maybe we could clarify that a little, maybe we have to some extent.

MR. IOTTI: There is really no difference in material per se. It is more a difference in requirement.

MR. TERAO: A difference in testing, right.

MR. BURWELL: Off the record a moment.

(Off the record discussion.)

MR. FLECK: Okay, well I don't have anything further until we get some data to look at. This is Hank Fleck again from E-Tech. Dave, the next question I have is on page 25 and 26 here, unless Dave has got some questions prior to that.

MR. WADE: No.

MR. FLECK: I guess I am a little confused on one thing when you read the deposition and then you go back to the finite element or the test data trying to

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see how the statements relate to facts. Here is one area that I really got a little confused, I guess. In my crete test. This indicates that crete is not a problem, yet I look at your test data in figure 25, and it shows for 24 hours the U-bolts lost a preload of 1-2% Well, that is hard to believe that crete is not a problem. In one day you lose 1%. This is primary crete, and the secondary crete would have a different rate. But, not knowing what the primary and secondary crete rates would be for this material, just taking that one percent in one day, that means in 100 days I have lost all preload just through the crete.

So, I dispute that crete is insignificant when the test results and the party who reported it shows 1-2% loss.

MR. IOTTI: Okay. Did you look at the next figures on the one that was also heated up at the same temperature and shows, in fact that it gains strength?

MR. FLECK: The next one, let's see. Let's take a look and see.

MR. IOTTI: I guess you are referring to figure 25 which is 4". If you look at figure, and it's true for the 4".

MR. FLECK: Okay, let me preference one thing Bob. Four inch, the hottest one. The next one for the

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ten inch is around 180,200. I wouldn't expect anything.

MR. IOTTI: Given the fact that I am talking about he U-bolt temperature at 250, 32" carbon steel.

MR. FLECK: There I have a question. What do you mean by apparent strain? In other words, were these strain gauges temperature compensated?

MR. IOTTI: Yes, they were.

MR. FLECK: Then, I kind of look out at the test for the 32" and I can't make head or tail of it.

MR. IOTTI: That's a problem too.

MR. FLECK: Something was lost there.

MR. IOTTI: The system has variability.

MR. FLECK: I don't know whether the gauges, whether the gauges were lost to get a statistical analysis to get a max/min, that is fine. But, for crete tests it doesn't show me anything. So, I go back to the four inch pipe which definitely shows a trend. Now, again, not having enough basic data on A36, I would assume if it is a reasonably good carbon that crete wouldn't occur until about 5-550.

Now, I look at A307, basically that is cut off in the ANSI B31 at 400, which said crete has probably occurred after that.

MR. IOTTI: Well, we obviously owe you more explanation than this one. The reason we made the

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statement, and we will have to get to some of the raw data. We believe what caused this load was the relaxation and not crete.

MR. FLECK: And not crete? Okay, I would have to look at...To do a crete test in 24 hours doesn't really show you anything. I would expect you would have to do a crete test for a month if you wanted to find it.

MR. IOTTI: We had a problem with that, of course. The problem with that is simply enough, we didn't have 6 months to do it.

MR. FLECK: I understand that, and I will say that for the time you had it, I think you put out a reasonably good report, but there are some conclusions you are drawing from it that I can't buy. That is one of them.

MR. IOTTI: Okay. I guess I will have to give you more data to enable you either to conclude with us we are correct or definitely say you guys don't know what you are talking about.

MR. FLECK: Yes. For the ten inch, I wouldn't expect to see any load. Because the temperatures, when you are in the 400 degree piping area, I have concern. I would like to get some more backup on that.

The other thing, what are your nuts on that?

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Are they 307 material? They would tend to start relaxing at 400 decrees, crete?

MR. IOTTI: I don't remember what those were, I will have to check.

MR. FLECK: I think you owe us a little more than to agree with you or disagree with you. That is not enough to really conclude much test.

MR. IOTTI: Let me make a note here to check on the material on the nuts.

MR. FLECK: If it did become a problem, then I come back to maybe you need inservice inspection. That would take care of your problem. That is what you would hang your head on.

MR. IOTTI: That is not where we are hanging our hat. Crete is not a problem. Would you give us that for the time being, that that is the result, and then ultimately we will be able to mention that this business of relaxing to this half is not required.

MR. FLECK: Right. But, until we have enough data to reasonable conclude that, I don't think at this point we could agree with you that this inspection is not required.

MR. IOTTI: We are in the process of putting together more elaborate.

MR. FLECK: I realize that your time span is

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very condensed.

MR. BURWELL: Do you find a submittal on preservices, I mean in-service inspection?

MR. IOTTI: No.

MR. BURWELL: Pre-service inspection, I guess.

MR. IOTTI: No.

MR. FLECK: I am going to, until this is resolved, request for a service inspection. If this is resolved, then that would disappear. If not, there would be an in-service inspection requirement and the timespan for it would have to be decided whether it is a year, eighteen months, or six months, and for what temperature pipes that this would be required for.

Then, I switch to page 59, and I believe, that is a typo. So, we can eliminate that very quickly. Your table asks for the 32" mainsteam. The allowable of 1.5S, 13KSI. I don't believe that is correct.

MR. BURWELL: That is on page 59?

MR. FLECK: Fifty nine of the affidavit.

MR. IOTTI: No. That is a typo.

MR. FLECK: Again, when it came to an obvious typo, I put it down. That is no big...

MR. IOTTI: When we take a break, I will go back to the original stuff and see if I can find the acutal value.

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MR. FLECK: I guess I am really confused what the 29 is there too. That might be part of the typo too.

MR. IOTTI: Which one?

MR. FLECK: The 32SM29.

MR. IOTTI: Mainsteam?

MR. FLECK: Yes. What the 29 represents. There is no standard gauge for it. I think that is a typo also.

MR. IOTTI: Maybe that was the allowable?

MR. FLECK: Yeah. I think that is right. You might want to verify that.

MR. IOTTI: I don't know how that 13 got in there. I brought most of the original stuff with me just in case.

MR. CHEN: At the top of page 126, 29.1.

MR. IOTTI: The 29 has got to be the allowable.

MR. FLECK: Yeah. I think 29 is the allowable.

MR. IOTTI: What I am puzzling is where the 13 came from.

MR. CHEN: 13.26 which is the SM divided by 29.1.

MR. IOTTI: .13.

MR. FLECK: Again, I don't think it is any big

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concern, it was an obvious, and it didn't look like it was correct.

On attachment 2, then, page 4 and 5, the same thing as before on the crete test. I would like to get...

MR. IOTTI: Attachment two...

MR. FLECK: Item E. I had that problem too, when you guys were on the phone flipping back, Monday.

MR. IOTTI: Okay. Statement of material facts.

MR. FLECK: Okay. That is the crete tests, and that tied into what we said before. So, additional data. Okay, on page 7 of that attachment, I guess in reading all of this, I am still wondering when I read it, and I keep going through it looking for certain items. I don't see anywhere where it is spelled out what the minimum freeload requirement would be to keep the cross bar with enough friction to keep it stable.

MR. IOTTI: Well, I believe we gave you some...Well, first of all, we have got to define stability. We are getting ahead of ourselves. When we talk stable here, we are talking stable in the truest sense. The U-bolts and bark, friction holder...For that, I think we provided some figures in the affidavit of he minimum freeload for the pipe size, for instance, which were tested and analyzed. I believe for the 4" it

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was 25 foot pounds. For the ten inch, would believe after all of this time I don't remember. I can't believe that.

I thought I had this memorized by now. Twenty five, fifty foot pounds for the 10" and 250 foot pounds, that is on page 74 of the affidavit.

MR. FLECK: As I say, I went back and forth.

MR. IOTTI: Well, it is understandable where you may miss one here or there.

MR. FLECK: I see 25 foot pounds. Now that is the value that the mechanic is talking to, right?

MR. IOTTI: No. This would be the minimum value that we will make sure exists in each, in the U-bolts.

MR. FLECK: So, these values in Table T represent the minimums.

MR. IOTTI: This would be the minimum, not that they are not there now, but they would be the minimum after we get through with our pre-service inspection in torquing up. Now, there was also a table 2, I believe. Letters and tables, we should be totally confused now.

MR. FLECK: It is around 25 or 24.

MR. IOTTI: No. That's...Right here. Right after, keep going to the end of that would be Table 1.

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Those represents the torques that are present now in the field. There are some instances, that you can see, where the torques are below the minimum value.

MR. FLECK: For the 4"?

MR. IOTTI: Seven and one half. In those instances it is possible for you to have a seismic event that the U-bolt and the bar would rattle. We have had that happen to us in the test. But, it also still carried the load which is really the function of the support.

MR. FLECK: I guess we are committed to make sure it doesn't rotate, even though we don't want to get back into...

MR. IOTTI: Well, I'm not so sure that that committment was ever made. We are committing right now.

MR. FLECK: You are. You are retorqued to 380. That it shouldn't rotate...

MR. IOTTI: That is a committment that we are making because we think it is a hell of a lot simpler than trying to fight it, okay.

MR. FLECK: Yes, I agree.

MR. IOTTI: But, we don't necessarily agree that it is necessary. We just committed to do it.

MR. FLECK: Well. Now that you are committed to do it though, I feel that it is my responsibility to

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make sure you have established minimum torque values as well as maximum torque values to accomplish one not exceeding the one half a yield, and also providing enough frictional force to present it from yielding. That is what I was concerned with there.

MR. IOTTI: So minimum values were quoted to you.

MR. FLECK: In table P, in this I guess they were established probably through the finite element.

MR. IOTTI: A combination of the finite element and the test.

MR. FLECK: These were reasonable minimums and not the absolute minimums. You can go through an iteration in the finite elements.

MR. IOTTI: In some instances, it may actually differ. We have got down as low as nine pounds minimum. We said it could be stable, but the test said hey, maybe we ought to stay around 25. The reason for that is, we have noted that testwise, for instance we were always stable when the U-bolt would contort well.

At 25, we were getting the perfect contort, so we said we are going to use 25, regardless of whether the finite element tells us nine.

MR. FLECK: So, table P would be a minimum.

MR. IOTTI: We're talking conservatism.

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MR. FLECK: When the fellow in the field torques, he is going to be given a max and a minimum in tolerance to establish.

MR. IOTTI: We have given, to the best of our knowledge, we haven't established that it is going to be done. I believe it is going to be given the minimum. You cannot be below the minimum.

MR. FLECK: Okay.

MR. IOTTI: I don't think it is necessarily going to be given the maximum.

MR. FLECK: So the minimum would have a 10% allowable for the field person torque?

MR. IOTTI: The reason that we don't perceive it is necessary for us to specify maximum is from a sample you will notice we have had no guerillas out there, literally, in torquing this thing way up. Even if there may have been a guy who took it, it has relaxed back down to where we wanted it to be anyway.

MR. FLECK: Well that depends on, again, getting more data on A36, and if it does relax and damage due to fatigue is insignificant, then the minimum value is the one you want to put the tolerance above. I would agree with that.

Okay, then the next question I would have would be in your attachment 3, the finite element.

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MR. IOTTI: Yes.

MR. FINNERAN: On page 11, figure, Roman numeral III-1, it shows .75 thickness, but in the enclosure for what the certified drawing Rev 3RC-1-018-016-GCB/R, it shows a quarter of an inch thickness. I would like to have some justification of why the difference.

MR. IOTTI: Those were actual measurements taken in the test.

MR. FLECK: Yes, but the drawing says it should be a quarter of an inch.

MR. IOTTI: Which drawing?

MR. FLECK: That's the drawing for the four inch, the one that was included in the submittal. You will see where I have yellow lined. Now, is the certified drawing correct? If so, the tests would have to be redone. Or then, is the certified drawing wrong?

MR. IOTTI: Why should the tests be redone. You are jumping to a lot of conclusions. First of all, what are we trying to establish in the test?

MR. FLECK: Okay.

MR. IOTTI: First of all, what we wanted to do was make sure the finite element matched the test, because that is the only way you can correlate the bolts.

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MR. FLECK: Right, agreed.

MR. IOTTI: We are not saying that that test, in order to find that element are representative of every support in the plant has a four inch U-bolt.

MR. FLECK: Alright. If the cross piece is a quarter of an inch in stiffness you had between the pipe and the U-bolt and cross beam is going to change immensely unless your freeload is going in the pipe. I was led to believe, I guess, that was the enclusure here. Therefore, I would assume all four inch scheduled pipe are a quarter of an inch thick for the crossbeam.

MR. FINNERAN: I think we might have an error in his drawing or something.

MR. FLECK: I would probably agree with you.

MR. FINNERAN: We do not have a quarter inch plate on the this torque for that.

MR. FLECK: I guess then, I would come back to say what does, then a certified drawing really mean? This comes to another question, I guess.

MR. FINNERAN: I think you may have a bad drawing, would be my first guess.

MR. FLECK: I would probalby agree with you, but I would like to have that verified, tracked.

MR. IOTTI: We can verify what the minimum

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plate is. It was high, I believe it was 3/4, but I just don't want to jump to the conclusion tat all of them are 3/4 for the 4".

MR. FLECK: Yeah. I don't want to have to jumpt to that conclusion either, and this leaves me to believe maybe, then, the certified drawing is incorrect. But then, it also brings out a question in my mind, I say, are all four inch pipes standard for the pipe plant U-bolt, or could some of them be a quarter.

MR. IOTTI: No. They are not.

MR. FLECK: That we would need some verification for.

MR. IOTTI: Nor are all 4" pipes schedule 160.

MR. FLECK: I agree. I kind of expected the drawing to agree with the test though. I guess that was my first subtelty.

MR. IOTTI: This is not right. It should be 3/4.

MR. FLECK: I used the magnifying glass too, I thought my eyes were wrong.

MR. IOTTI: Well, the drawing is obviously wrong here because we measured that just to make sure.

MR. FLECK: Well, my concern would be what is in the field, what you have tested and what finite

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element model is represented as one thing, what is in the field is...

MR. FINNERAN: Well, actually we have the clamp mechanism fabricated from this drawing.

MR. FLECK: Then it was done wrong.

MR. FINNERAN: You have a bad drawing, that is all.

MR. FLECK: Well, then I would wonder what the drawing system is really doing to you guys.

MR. FINNERAN: No. I just think you have a drawing that was copied badly, or something.

(Pause)

MR. FLECK: Okay. Then on page 21, it looks like another slight error there. The figure 4, Roman Numeral IV-3 shows the strut attachment 90° from what the drawing MS-1001-005-S72R. I think that is insignificant.

MR. IOTTI: That is in fact, the strut got attached the wrong way. My recollection is in the field and in the test.

MR. FLECK: Then, I would, I think for your test it makes a small difference but...

MR. FINNERAN: It was intentional for him to allow him to use the mechanism test.

MR. IOTTI: Otherwise, the test.

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MR. FLECK: I guess what you are saying is the drawing is right but in order to get the test fixture in you had to change it.

MR. FLECK: Okay. No problem. I was just curious after seeing the other just what the problem may have been.

MR. IOTTI: In the finite analysis we tried to...

MR. FLECK: I don't think it would make any significant difference one way or the other. I was just curious why the drawing and the test were different and that explanation was adequate.

Okay, on page 29, I think I know what you mean but I would like to get a further clarification just so I am thinking the same path that you are. When, the preload was, excuse me. That is the preload cynching due to U-bolts, due to torque on the U-bolts. This was done by imposing a negative temperature on the U-bolt which resulted in the shorting of the lathe. This then, how do you accomplish it physically in the program? Did you shorten it by alpha delta pi?

MR. IOTTI: Yeah. You actually put it in and the program does it automatically. There is an eternal program within the temperature of the U-bolt. There is two ways you could do it. You could largely

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apply the load.

Westinghouse said this worked. We tried it, but independently, we get the same answer.

MR. FLECK: So, actually, if the program would just take alpha delta ti it would do it. I kind of thought that, but I wanted to...

MR. IOTTI: The key here is you only want to do it in the straight portions. Otherwise you get ficticious answers.

MR. FLECK: Yeah. You get an radio that really doesn't correspond.

MR. IOTTI: We ran independent tests upon the loads just to make sure that Westinghouse was totallly right.

MR. FLECK: So, within the NASTRAN printout, you could see physically that you were in the axial, or straight portion of what the axial preload should have been?

MR. IOTTI: Yes. As a matter of fact, it was tuned to make sure that it matched the preload of the tests so we could correlate it.

MR. FLECK: Okay.

MR. IOTTI: Don't be surprised if the agreement on the preloads is so spectacular to be good.

MR. FLECK: Well, that is true. And, in order

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to understand some of it, I have to know how the boundary cases of this particular case are put in. That was one thing that I just wanted to get clarification so I would know how in tune or how tuned it was, and how close it should be.

Okay, page 42 I had a question in te same area here. I guess what I had a question or hard to accept is the statement where it says stresses measured in the test and calculated for the 32" pipe crosspiece and U-bolt are compatable,

MR. IOTTI: Comparable.

MR. FLECK: Comparable, rather. I guess really, in looking at the results of the 32" pipe I would say that they are the worst that I have ever seen. htings don't correlate very well with span gauge and test. The other pipes do, and I just can't buy that statement, I guess. The 32", you explain, you did have problems in teh fabrication received. So, I just look at that statement adds no meaning to the test results. I just take issue with that statement being due to the fact that the 32" hardware...

MR. IOTTI: Do you take issue with the fact of the stresses alone?

MR. FLECK: No. I do not.

MR. IOTTI: That's maybe, we put it in other

words. What we wanted to get across is hey, they are so low.

MR. FLECK: Okay, I guess the word is compatable then.

MR. IOTTI: We wanted to make sure that it wasn't a failure of the strain gauges that causes the low stresses in the test. The finite element analysis predicts the same thing, that they are very low.

MR. FLECK: Yeah. I think it was worded that way. I wouldn't have taken any issue. Just a clarification of when you look at the test data it doesn't really give you that same fealing. They are small, we will definitely agree with that.

Then, starting on page 54, where the ground work is layed for what the allowable stresses would be, either I didn't take enough time or I need further clarification of equasion 9 and 12 equal to 3SM.

MR. IOTTI: Would you like to explain what you mean by clarification. What you are driving at?

MR. FLECK: I went from 54 on to the numbers.

MR. FLECK: There are two methods, as you represent here.

MR. FLECK: I may have gotten lost in the two methods, to be quite frank with you.

MR. IOTTI: The problem is that the code does

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not provide the right guidance. So, we are to invent some new acceptance criteria. There are two ways we could proceed. If you proceed a traditional way, you dump all of the secondary stresses into equasion 12 and 13, and in fact, since the code doesn't address these local secondary stresses, you ignore them. You pass. Well, what does that tell you? Have you really passed? That is the second approach. We took the standard equasion 9, and then all of the secondary stresses. Most of them are secondary into equasion 12. The code tells you not to consider them anyhow. Then we said, well what makes sense. What we really want to do here is mae sure the material does not fail. We don't want any gross failure or anything that could lead us until possibly a big problem later on.

What we elected to do is to take the secondary stresses that are bending secondary stresses and consider them as if they were primary local membrane and put them into equasion 9. T.is is the front part. Because of that, we thought it was appropriate to increase the allowable equasion to 3. Because, we felt that this might be building settling, with a one time load. That is why you see in equasion 12, approach from the 3 years.

MR. FLECK: When you use equasion 9 this way,

would you use the C in the C's in the normal moments in the pipe or would you peak in the C's?

MR. IOTTI: This where we used the B and C's.

MR. FLECK: I guess there I haven't digested this enought to really feel that we should use the C in the C's for the regular pipe moments with the 3SF, or whether we should use the B. So, again, I have been unfortunate. I have been on jury duty and haven't had time to digest some of the more subtle things here.

MR. IOTTI: As you go through these, once again, we are not claiming...We didn't feel it was appropriate for us to say we know better than the code people. So, we took both approaches to make sure that both instances in our own mind they are okay. It is the combination in the both that led us to believe we don't have a problem as oppossed to just the first one of the second.

MR. FLECK: Well, maybe this might be an area that we have to do more homework too on. But, it may still be an open issue. At this point, I don't think I could stand firm either way. Unfortunately, the numbers that you have provided here, I can't backtrack enough to see what the differences may or may not be. The stress table that you provide in 72 on doesn't...

MR. IOTTI: The only way you can backtrack is

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go back to the normal stresses in the pipes and then change the C & C's. I think I would ask Westinghouse to do that. I would ask them to reuse the C&C one time. They said they were still okay.

MR. FLECK: I would probably see it that way, because I get the impression we are doing it both ways.

MR. IOTTI: Well, I asked them, because I had the same concern.

MR. FLECK: But then, I see only one way and then I see well...I go along with it to a certain point an if you are using equasion 9 with primary and secondaries, which you are combining all the effects, well then, the piping moments should have the highes C in he C's to include primary and secondary effects if that were the case. But, as I say, we may have to do more homework on that.

MR. IOTTI: One of the reasons too, that ultimately I believe Westinghouse decided to stay with the B & the C's was they weighed the moment was included in the, at the location of the support. It was not by acutal stress analysis by Gibbs and Hill, by an equivalent method which was already considerably comparative to the Gibbs and Hill calculation.

MR. FLECK: Okay. That would be nice to see the comaprison of the conservatism there, because I

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realize you are not going to go through every pipe and get the true moment at that spot and use it.

MR. IOTTI: I believe someplace in here we nad included that spot, because I could foresee that question coming up. Didn't I provide a table which actually provides the Gibbs and Hill calculation, maximum calculated...I would have sworn I had.

(End of tape.)

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MR. FLECK: You have a lot of data there. There's a possibility coming back and forth that once I have a point that I want to have a question, it may have been answered later and I missed it when I went back.

But I'd like to see a little more there and in numbers so I can really get a feel for just what's happening there. And when I get to page 59...

MR. IOTTI: I have those numbers because I actually have the... We asked Gibbs and Hill to go back and search for these type pipes, what the maximum stresses were, bending stresses and, you know, just total stresses.

And what we then did, compared those against the ones that Westinghouse had been calculated, these hand calculation or method used in the industries, and we determined that in all cases Westinghouse was being conservative, and sometime by almost too much of a margin.

MR. FLECK: Because the other thing I can really foresee here with setting a precedent that I'm not going to put the burden on as far as what the ASME code may or may not consider it, and it would be better, I think we'd have a complete set of data that we can look at, see just how it is.

MR. IOTTI: Well, we try not to attempt to set a precedent here, okay?

MR. FLECK: Right, I understand that.

MR. IOTTI: We said if the code people couldn't come up with anything over many, many years, we weren't going to be able to do that in two months. Nevertheless, we had to do something. We just couldn't say here are our results.

MR. FLECK: I agree. The only thing right now, as I say, we may have to do more homework to agree or disagree with you on a firmer ground, and right now I need clarification.

I can't really, from what the data is presented here, go one way or the other. I need some more input, I guess, is what I'm saying.

MR. IOTTI: I can have... No problem for us to give you the comparison, for instance, between the Westinghouse calculated normal pipe stresses and the Gibbs and Hill. We have... I thought it was included here and I guess...

MR. FLECK: It may, and maybe I didn't...

CHAIRMAN BURWELL: You will give us that comparison then?

MR. IOTTI: Sure. I have it. I may even have it with me.

CHAIRMAN BURWELL: Off the record.

(Off the record.)

MR. FLECK: Okay, this is Hank Fleck again, and I have another question on page 59, and what I would like is to get some clarification for the total stress intensities.

I tried to go back to the tables that were presented for the stress summary tables on page 68 to 89, but I don't believe there's enough information for me to back track to see how these numbers were obtained.

MR. IOTTI: Would it be helpful if we actually gave you an example, how you...

MR. FLECK: Yes, it would be very helpful.

MR. IOTTI: ...go from one to the other? Okay.

MR. FLECK: And on that same page then, I would need an explanation on how those total stress intensities were divided into equation 9 and into an equation 12. I can't follow the breakdown.

MR. IOTTI: Let me make a note. Actually, I was looking... For reason I knew that this was going to be your next question.

MR. FLECK: We're on the same wave length.

MR. IOTTI: I actually had a worked out example and I appear to... I've got to look through

all my files now.

MR. FLECK: Okay, look.

MR. IOTTI: But I want to make a note...

MR. FLECK: As long as we can get something there.

MR. IOTTI: Give example on how... So what we'll do, we'll pick one of them, okay. I think that's the one that I have.

MR. FLECK: Fine. If I can follow one of them...

MR. IOTTI: And we'll... You have to go back to the tables, you know, for the maximum intensity.

MR. FLECK: Yes. I went to the tables...

MR. IOTTI: We'll tell you which table to get, how to combine them, okay, and how you derive all the language.

MR. FLECK: I did that and I got lost and then I figured...

MR. IOTTI: Well, I did the same thing many times.

MR. FLECK: Part of the tables may have left out some of the information that I needed, so if you could have an example...

MR. IOTTI: No, all the information is here, but it's in about four places.

MR. FLECK: Okay.

MR. IOTTI: Okay, so you've got to go everywhere. Give an example on how to derive value of total stress intensity and also how that is broken down into equation 9 and equation 12.

MR. FLECK: Yes. And that would give me a better feeling for your 3-SM for equation 9.

MR. IOTTI: Okay.

MR. FLECK: That, I believe, is what your allowables are based on there and if I can get something I can follow, I can either agree or disagree on a more reasonable basis.

MR. IOTTI: That's it, but I think I need to polish it up.

MR. FLECK: Okay, fine. I'd like something that I...

MR. IOTTI: This was one of the examples I was...

MR. FLECK: Yeah.

MR. IOTTI: ...referring to.

MR. FLECK: Well, if I can follow what you're doing, I think I'll understand your criteria a little better and have a feel for whether I think we're in the same ballpark or whether additional questions would generate from that.

MR. IOTTI: Maybe... Let me just, for instance, pick this one. I'm trying to recall now this one which was... This is the... had a marginal pressure stress of 4.8 in equation 9 (inaudible) 12.45.

You've got to look at the inside surface and at the outside surface and you've got to go to the table that give you these two piece of information.

CHAIRMAN BURWELL: Can you speak up a little bit?

MR. IOTTI: Can we go off the record because I'm going to provide this on the record, okay? I'm just...

MR. FLECK: Giving me a little preliminary reference.

MR. IOTTI: I want to make sure that what I'm going to send him, it's acceptable to him so I'd like to just have a short discussion off the record to tell him what it is that I'm going to send him and see if he needs more.

CHAIRMAN BURWELL: Okay, let's go off.

(Off the record.)

CHAIRMAN BURWELL: While we were off the record Dr. Iotti showed Mr. Fleck a hand calculation. They discussed the analysis performed on the hand calculation.

With Dr. Iotti's explanation of the hand calculation, which he will provide on the record, why, then that appeared to satisfy Mr. Fleck, subject to its review, of course.

MR. FLECK: Right.

CHAIRMAN BURWELL: Now, with that, I'll turn the meeting back. Mr. Fleck, I believe you were discussing...

MR. FLECK: Right. The next question I have would be on page 62, which refers to the evaluation of the transient stress and the damage factors as shown. Here again, I would like some clarification in terms of what cycles were used.

In other words, how the numbers were derived and back-up input for that. You could come up with a useage factor of .04 and .06, respectively, and I'd like to see what your assumptions were as far as the cycles assumed and what have you.

MR. IOTTI: We have all of that. I was wondering whether we actually put it in the affadavit.

MR. FLECK: I don't believe it was there, but again, you may find that it is and I have overlooked it.

MR. IOTTI: Yeah, for instance, it tells you on page 69 through 71 is a little bit more explanation as to how the fatigue analysis was considered and it

tells you what kind of cycles we looked at.

MR. FLECK: Sixty-nine?

MR. IOTTI: Sixty-nine through seventy-one.

CHAIRMAN BURWELL: Sixty-nine through seventy-one of the affadavit.

MR. IOTTI: Of the affadavit.

MR. FLECK: Oh, of the affadavit.

CHAIRMAN BURWELL: Ah, well, how are we going to label...

MR. IOTTI: It included all of the design transients for the plant's, you know, specification, plus the maximum plant induced stresses, the... And then it computed how many cycles one could, in fact, tolerate.

For instance, for the maximum alternating stresses we were computing we decided that we would have about 22,000 allowable number of cycles. For the number of heat-up and cool-down cycles we used 200, and for the seismic 200 cycles were also considered, 20 significant cycles per earthquake.

MR. FLECK: I guess the only thing, it resolved that crete could be a problem. You would have to do a crete damage accumulation on that, too. But it looks like you'd have plenty

MR. IOTTI: Well, our conclusion was that

we didn't have crete so we didn't consider it. That's something that's still open, as far as you're concerned.

MR. FLECK: Right. All right. Then another general question I have...

MR. IOTTI: Now, do you have sufficient information or do you need more on the fatigue?

MR. FLECK: That should give me enough to go back to see whether the cycles are reasonable and everything else.

MR. IOTTI: If you need more... Mr. Burwell, how do we handle... If he needs additional information, does he come through you or could he call me directly? It may be more expedient because I know the difficulty he must have reading through this because I wrote it and I know the difficulty I had it, I had with it.

So I can see where one reading it would have difficulty, and on occasion a quick phone call that says, "Look, I'm not understanding. What am I reading here? Can you refer me to a page?" can save him a lot of time.

CHAIRMAN BURWELL: I quite agree, and if I can get Mr. Fleck to write me a hand-written note explaining that you called and what the subject was and any major conclusions. Just a brief summary.

MR. FLECK: We normally have a telephone

format that we use when we...

CHAIRMAN BURWELL: That'll be fine.

MR. FLECK: ... and I could transmit that to both you and to Bob so the misunderstandings could be limited.

CHAIRMAN BURWELL: Serve as to confirm the conversation.

MR. IOTTI: Whatever I send him, of course, I'll send a copy to you and to everybody. I'm just trying to save time.

MR. FLECK: Okay, then I have another very general question on page 72, 78 and 84.

MR. IOTTI: This is still that attachment?

MR. FLECK: Right.

MR. IOTTI: Seventy-two, oh, these are all the tables.

MR. FLECK: Yeah. The wording now comes in, partial pre-load, and I just had no idea or feel what is partial pre-load. What does it mean? What's your definition of it? Why is it included is what I really have from reading the information, having decided what the significance of it is.

MR. IOTTI: Oh. The pre-load as defined in our test was the maximum pre-load that we could expect to ever see in the field, okay?

MR. FLECK: Okay.

MR. IOTTI: And so we specify for Westinghouse that where they ran the test and when they did the final (inaudible) they should use this upper bound value pre-load. That's called the pre-load.

The partial pre-load is a lower value which is generally typical of the minimum, or a little bit higher than the minimum value that we, that we choose to flow.

We did this to try and find the sensitivity of the results of the pre-load.

MR. FLECK: I thought that might be the case, but I didn't see a definition and I think that's what I needed. You might want to include it.

MR. IOTTI: I'll clarify it. Partial pre-load, clarify what we mean.

MR. FLECK: Okay. Okay, this has been answered already. Okay, now we get to the test report, and on page 54 all of a sudden I see the test was conducted with 600 foot pounds of caulk and the specified caulk that 2C had was 240 foot pounds.

MR. IOTTI: The explanation is simple for the difference. A 240 foot pound we could do anything. Not only that, we couldn't get that U-bolt to even come close to the pipe. We had gaps all over.

MR. FLECK: Okay. This leads to a question. That's what I thought the problem was. Then I wondered in the field is 240 foot pounds adequate. If the test showed that you need 600 foot pounds to even accomplish the kick, so to speak, why is 240 adequate?

MR. IOTTI: Well, one of the reasons we need that to accomplish the fit is that the U-bolt came to the testing facility incorrectly sized. We had to spread it out at the testing facility, and that hasn't happened in the field, to the best of my knowledge.

MR. FINNERAN: No.

MR. FLECK: I thought that was the case, but I'd like to get that firm in my mind that the article you tested wasn't typical and you had to do something to bring it into its sub-crushing and load.

MR. IOTTI: That's correct. That's correct. The other thing is that the 250 foot pound, which is the minimum value that we plan to have for that size pipe in the field, is arrive at from the final element analysis.

MR. FLECK: And that would provide your friction resistance you needed.

MR. IOTTI: With sufficient friction, right.

MR. FLECK: Okay. And I guess on page 52 and 53 I had a question that I think we've answered already.

What is the apparent strain, and that is really temperature compensating.

MR. IOTTI: We have all of the raw data. We can give you the raw data plus the explanation. I mean these are reams and reams.

MR. FLECK: Well, normally, when I've been involved in strain gauge testing, we either specified we want the bridge wide for temperature compensating...

MR. IOTTI: These are temperature compensated.

MR. FLECK: And apparent has been a new term that I wanted to make sure that my definition and your definition were the same. Okay, then there's a typo error on page 60, I believe.

The strain gauge leg S-5 on gauge 1, or leg 1, rather, and S-10 do not agree with the sketch on Sketch 5 on the preceding page, 58. The S-5 appears to be on leg 2.

MR. IOTTI: Be careful, those are thermal couples. Those are not strain gauges.

MR. FLECK: The temperatures?

MR. IOTTI: Yeah. Those are temperatures measurement. For the strain gauges you have to go to the higher one.

MR. FLECK: Okay. Then I'm curious why are thermal couples on... and you're plotting pre-load

versus cycles. Wouldn't you want to have strain gauges?

MR. IOTTI: Yeah. Well, we do have strain gauges also.

UNIDENTIFIED SPEAKER: Hank's on page 58. I think you're on page 59.

MR. IOTTI: Oh, okay. Well, he said the preceding page so...

MR. FLECK: Oh, okay, I'm sorry.

MR. IOTTI: ...I went to page 59.

MR. FLECK: Page 58 is the Sketch 5...

MR. IOTTI: Oh, okay.

MR. FLECK: ...and page 60 is where the Leg 1 shows a S-5. And it doesn't appear to be... S-5 is on Leg 1 and it shows an S-10 that doesn't appear to be on Leg 2.

MR. IOTTI: I'll have to go back and reconfirm that. It may be a typo, but I don't know enough to be sure.

MR. FLECK: I think it may be, yeah.

MR. IOTTI: I don't want to give you information that may turn out to be wrong.

MR. FLECK: Okay. And page 66, I believe there's another typo, middle of the page, Leg 2 gauge...

MR. IOTTI: Just a minute, Hank. If I don't

write it down, I'll forget it. Okay.

MR. FLECK: On page 66, in the middle of the page, Leg 2, paren, gauges S-4, S-11, end paren, equal 5316 pounds. I believe that should be...

MR. IOTTI: That's 3516.

MR. FLECK: I believe it should be 3516. And on page 65 and 68 we come back to the creep, and that was mentioned before. The other thing I noticed there on page 66 you have a different ambient temperature for your initial reading and the last ambient was 14.4 degrees higher where you have a TATB effect because the pipe is stainless and the clamp is carbon steel.

So you would really have a greater difference in the loss of pre-load than is what's shown here. So it would tend to hurt you more as far as for the creep caution.

MR. IOTTI: Would you mind going over that again because...

MR. FLECK: Okay. Due to the fact you have two different materials, when you start at ambient and don't end at the same ambient, you have, in effect, raised the pre-load.

If they were both the same materials and you went from a high, a low ambient to a high, it would have no significance. But the carbon steel does not expand

at the same rate as the stainless. Therefore, the stainless pipe is growing, which is the same thing as shortening the raise which adds to pre-load.

MR. IOTTI: That's correct.

MR. FLECK: Therefore, the values that you have here would even show a greater loss than what you had when you're quoting the 1%.

MR. IOTTI: Yeah, you pre-loaded it slightly because you say your final ambient is slightly higher.

MR. FLECK: And in general, that could be anywhere from 50 to another 100 pounds out of the...

MR. IOTTI: Well, I can do the calculation quickly enough.

MR. FLECK: Right. But what it does, it adds to the fact that you're losing more load after 24 hours, and this is the creep test. Originally I thought when I looked at it that possibly you would have better agreement.

But, no, you even have worse agreement than a 1% loss. It's slightly greater.

MR. IOTTI: That's possible. I'll have to go back to the raw data. This is going to be all part of the answer.

MR. FLECK: The creep, right.

MR. IOTTI: We're talking about a total of

what? Five degree difference in ambient temperature.

MR. FLECK: Fourteen point four degree difference. The ambient originally was 77 and the final was...

MR. IOTTI: Ninety-one point something.

MR. FLECK: Ninety-one point four is your final ambient and your original ambient is 77 degrees. On page 55.

MR. IOTTI: Yeah, got it.

MR. FLECK: So again, the creep should be, I think, elaborated as far as whether it does or does not exist. From the test data it appears it does exist in a magnitude such that you want to have in-service inspection, and until that's verified that could be the open item.

MR. IOTTI: Well, yeah, we'll try and provide you as much answers as we can. We certainly don't want to do in-service inspection.

MR. FLECK: I understand your concern there, but...

MR. IOTTI: Well, it's not just our concern. The problem is that there's many other plants which are in the same boat.

MR. FLECK: Yeah. And if we can establish better creep data, I think we can come to a very logical

engineering conclusion.

MR. IOTTI: Well, what I may also do if I have a little bit more time is to try and get additional creep data on A-36 if anything else like that exists, you know.

MR. FLECK: You may have to establish it.

MR. IOTTI: That's been the problem with this material.

MR. FLECK: Right. But our concern would be that we have closed that loop and that if there's any doubt, in-service inspection would be the only answer to it, which would mean in time you'd prove that either we're right or you're right by checking the torque on the bolts.

MR. IOTTI: That may be a one-time in-service inspection.

MR. FLECK: That's very true.

MR. IOTTI: If the first one you pass, then obviously there's no creep.

MR. FLECK: That might be your logical out. The only thing would be what time interval we set for it, whether it's six months, a year.

MR. IOTTI: We still have to establish that from the data.

MR. FLECK: Right.

MR. IOTTI: So I have to go back to the data.

MR. FLECK: Right.

MR. IOTTI: And the data scatter because what you here see is a curve.

MR. FLECK: Right.

MR. IOTTI: Really, you need to go back to the data scatter.

MR. FLECK: Then my last question, on page 88, is a very general question. The natural frequency of the test...

MR. IOTTI: Wait a minute, Hank. Let us get that down. No, I've got it on pre-loads.

MR. FLECK: Yeah, I'm rushing. Slow me down.

MR. IOTTI: Well, I know we're all anxious to get this over and done with. Page 77?

MR. FLECK: Eighty-eight. Just from a general curiosity, from an engineering standpoint, I wonder where the 75 hurts basically would be found. Would it be found in the pipe, the specimen, the test rod or whatever.

What I would like to do is kind of get a feel for what was really getting the most damage from the test, whether or not I should even be concerned with it is immaterial.

MR. IOTTI: What was getting the most damage

from the test was the test apparatus. Neither the pipe nor the...

MR. FLECK: Because I went through some rough calcs. For that span you'd be way over this frequency, and for the cross member you're way over the frequency and...

MR. IOTTI: Well...

MR. FLECK: I guess what I'm getting at...

MR. IOTTI: What did you assume is your boundary condition, though, because, see, that's a problem.

MR. FLECK: Yeah, fine.

MR. IOTTI: The reason that we... By hand calculation you're operating 100 hurts, okay. You have a natural frequency of that span, but by the time you put the boundary condition, which aren't really fixed, you're now back to something of the order... Seventy-five hurts was determined by actually scanning through it, finding the maximum (inaudible).

MR. FLECK: I agree, and that's a good way. But then to be meaningful for me, whether I put a lot of emphasis on this data, is to find out what was the weak link, and if it wasn't the U-bolt, then I don't care.

MR. IOTTI: The weak link was not the U-bolt

and this is because we couldn't drive.

MR. FLECK: I really don't care about the data. I realize this.

MR. IOTTI: At this frequency we just couldn't drive the force through it.

MR. FLECK: Right. So I would have looked at it as, again for my own engineering idiosyncracy, as well, the frequency was 75, but it wasn't in the area of the test specimen.

Therefore, I really concluded what you concluded before. You didn't have enough force to get any meaningful data from it.

MR. IOTTI: So really all of our conclusion is essentially predicated on the 9 hurts test.

MR. FLECK: Right. So I kind of looked at that portion of the test and I wiped it out of my mind as being meaningful or not.

MR. IOTTI: Did he bring the tape?

UNIDENTIFIED SPEAKER: I do not have the tape with me.

MR. IOTTI: We took a movie of the test, of this particular test, that actually shows the...

MR. FLECK: What frequencies have you finished?

MR. IOTTI: Well, we took it at both. You really don't see a thing.

MR. FLECK: Yeah, you probably wounldn't.

MR. IOTTI: But the 9 hurts, you see things.

MR. FLECK: You used a strobe then when you took the movies so you could...

MR. IOTTI: It's a very slow speed. Very slow speed. It shows it. It's very vivid.

CHAIRMAN BURNELL: I can have the tapes in here overnight if you want to see it.

MR. FLECK: It's just my own engineering curiosity, like anything...

MR. IOTTI: We have a TV that we could show it on? It's one of these video cassettes.

UNIDENTIFIED SPEAKER: A VHS, isn't it?

MR. FLECK: It would be nice for me to get more of a feel for it

MR. IOTTI: Yeah, it's a VHS.

UNIDENTIFIED SPEAKER: Be here tomorrow?

MR. FLECK: Yeah, I'll be here tomorrow. Again, when I made some quick calcs I just said, well, it's not in the area that I'm concerned, or if it is I better find out...

MR. IOTTI: No, we can have it sent over.

CHAIRMAN BURNELL: We could get access to one if you all need it.

MR. IOTTI: Because I'll tell you, it certainly

drives home a point. Nothing like seeing it.

MR. FLECK: Yeah. If it's available, I think I'd appreciate seeing it. Whether we had the time for it would be another thing. But my own curiosity just said, "Gee, it can't be the pipe. It can't be the cross piece. What was the thing you were driving?"

I understand what you're saying and if we say, well, the test really didn't... You didn't have enough energy to drive what you wanted to drive. And if you did, it also concludes that the frequencies are so high an earthquake would never drive it.

MR. IOTTI: They can't... They can't drive it.

MR. FLECK: Yeah. That kind of summarizes the areas that I had specific questions. I have one general question, though. I noticed you do have pipes under four inches, and I don't know what you plan to do or whether the finite element for the two-inch or the one-and-a-half-inch pipe sizes you have, but I do have a concern knowing that these pipe sizes and the standard gauges or schedules you'd have on it would be such that they would be much stiffer than the four-inch, much being at least a factor of one and a half over it.

And I'm wondering what would be done to alleviate any thoughts that now the pipe is greatly stiffer than the clamp, are we getting enough pre-load

in the clamp or is it dissipating itself.

MR. IOTTI: Well, the...

MR. FLECK: Do you have any plans to do any?

MR. IOTTI: We have two plans. One of our original plans was to use the same pre-load there as for the four-inch, but we probably will abandon that plan.

But we do have the finite element model and we would use the finite element model to guide us in establishing what kind of a pre-load we need to establish for the smaller lines.

MR. FLECK: Okay. I would definitely like to see the last of your plans activated so you pin down the fact that now you're, you're getting in a realm where that small pipe is probably going to be appreciably stiffer than the clamp cross piece member, and the finite element model, I think, verifies.

Your test is reasonably close for the U-bolt at least and for your friction, and I would accept the additional finite element...

MR. IOTTI: Our biggest concern and the reason why we would run the finite element is really more from the standpoint of the pipe than we do for the U-bolt and the... There are simplified methods that we can utilize and we've kind of benchmarked against the

finite element. It can tell you what the U-bolt will do and what the backing plate will do, but it's not so simple to determine what the pre-load will do to your pipe and that's why we would rather use the finite element in that regard.

CHAIRMAN BURWELL: Would you move the mike a little closer to you, please?

MR. IOTTI: Either that or I move closer to the mike, one of the two. So that's, that's our current plan. We haven't quite determined which one of the two will follow, but we're leaning toward the latter, because the reason we developed all of the finite element models to be able to interpolate and extrapolate from the test data.

MR. FLECK: Okay. Well, I would probably feel that that should be included, just so we see that the values that you're going to talk it to will still satisfy your Table P and also provide rotational resistance.

And I think you have the tools to do it with so it's just a question of when you present it.

MR. IOTTI: Well, I'm not sure that we had any intent to presenting the values at this point in time, other than say here are the tools that we would use to determine at what value we would torque all of

the various pipes in the plant. Now, obviously, there will be a set of minimum values that will have to be agreed upon before Tutco goes by and inspects this 380 U-bolts and torques them down or, you know, if they're higher, they're not going to touch them.

But if they don't have that torque, they're going to be torqued up to those minimum values.

MR. FLECK: Okay, but you would expand your Table P to include all pipe sizes then? That's your minimum torque.

MR. IOTTI: I'm not thoroughly sure that we would do so in terms of a table. It may be a curve as a function of all the pipe schedule and size, yeah.

MR. FLECK: But the person in the field would have something to go by.

MR. IOTTI: Would have something. Oh, yeah. He's not just going to walk without...

MR. FLECK: That's my only concern, how you will present it.

MR. IOTTI: That will be made available.

MR. FLECK: Is the way you feel is easiest for them to read.

MR. IOTTI: What I was trying to say is that it's not our intent to augment to the affadavit or supplement the affadavit. This would be an independent

type chart or figure developed to serve as guidance for the people in the field when they were torqued.

MR. FLECK: All right, as long as the affadavit, which states that Table P would be expanded...

MR. IOTTI: We would prefer not doing that because we don't see what purpose that would serve. We've already stated we're committing to arriving at minimum pre-load value so I think the Commission to certainly audit us...

MR. FLECK: Right.

MR. IOTTI: ...in terms that says "do you have such a figure? Do you have such a table?" They can come in and verify that for each pipe size there will be a guidance given to the, to the people. What do you call it? The inspector or torquer or...

CHAIRMAN BURWELL: Both.

MR. IOTTI: Both. Now, we have a torquer and an inspector.

MR. FLECK: Then I have no further comments to that. Dave may have some additional comments he might want to include, but those were the areas where I felt either I needed some information or things were...

MR. IOTTI: Before we leave this topic, or at least before we leave your particular comment, could I summarize? I have 11 items that I think I owe you.

MR. FLECK: Okay.

MR. IOTTI: Okay. And the first one is that you would like to have additional raw data, in particular the spread of the data and the specific heats of the U-bolts that were used in the tests so that we know what their yield strength were and so forth to confirm the validity of this relaxation down to something of the order of one half of the yield strength. Okay.

The second one is to, which is fairly pervasive, is to provide more information on the creep test of the 4-inch pipe. Okay?

MR. FLECK: Right.

MR. IOTTI: And also corollary to that is to check on the materials of the nuts. Okay? The third one is to... Well, I think that one is resolved. This was that typo on page 59 that should be 29.13, so I think you can write down 29.13.

Then we are supposed to check on the accuracy of that drawing that shows the thickness of the back-up plate for the 4-inch pipe to be a quarter inch instead of a three-quarter that was used.

We'll provide you some rewording on page 42 of that Attachment 3, that's that Westinghouse report, to explain really what we mean when we say the stresses on the 32-inch mainstream are negligible and why it's

really not that important. The test didn't show whether it was accurate or not. The finite element confirmed that.

The next one is what would happen if we were to use the C indices instead of the B indices for the elbows as part of the whole calculation with regard to the first approach that we took where we used the 3S for both equation 9 and equation 12.

The next thing is I'm to give you an example on how to derive the values of the total stress intensities from the various results of the finite element analyses and how to break the total stresses down into an equation 9 and equation 12 stress intensity.

Clarify what we mean by partial pre-load, although I think I gave you the answer. I might as well provide that in writing. There's a discrepancy between Figure 21 on page 60 and... that shows the location of the strain gauge 5 and 10 on perhaps the opposite leg, so where they actually are that we have to clarify.

There's a typo on page 66 in the test report which I'll correct, but you already know that it's juxtaposition of the first two figures. And finally, the effect of having a different ambient temperature at the beginning and at the end of the creep test and what effect that would have on loss of pre-load.

That's, of course, tied in with the whole question on creep. That's the only thing I have.

MR. FLECK: That agrees with what I have, too.

CHAIRMAN BURWELL: Okay, any further questions on U-bolts?

MR. TERAO: Well, I didn't have any questions on this particular submittal on U-bolts, but I wonder if we could go back to our previous submittal on the U-bolts acting as two-way where there was test data.

I just want one thing clarified that apparently ties in, as long as we're on U-bolts. On Attachment 1...

MR. IOTTI: Give me one second, Dave. Let me get my file on U-bolts acting as two-way restraints. Okay. Attachment 1?

MR. TERAO: Attachment 1.

MR. IOTTI: Okay.

MR. TERAO: Back in Figure A-13.

MR. IOTTI: A?

MR. TERAO: Right, A-13.

MR. IOTTI: Those are not valid anyhow. That's just one that they're supposed to give us an ambient fact figure. Thirteen or thirty?

MR. TERAO: A-13.

MR. IOTTI: Thirteen.

MR. TERAO: It's the normal load versus

elongation.

MR. IOTTI: Yes.

MR. TERAO: Okay, there on I guess Figure A-13, A-14 and A-15 and also Figures A-28, A-29 and A-30...

MR. IOTTI: Well, 28, 29, 30, I'm going to have to replace for you because something had malfunctioned when these figures were plotted, and we've been expecting the correct figures from ITT Grenile.

MR. TERAO: Okay.

MR. IOTTI: A-13, 14 and 15 are correct, to the best of our knowledge.

MR. TERAO: Okay. I guess these are the flocks. They're on a different scale and the horizontal scale, it's not clear exactly what (inaudible) are.

MR. IOTTI: There is no horizontal scale. Now, the first ones, 13, 14 and 15 I believe are on... You can see it's .5, 1.5... I mean 1 and 1.5. If you look at Figure A-13?

MR. TERAO: Yes.

MR. IOTTI: Very, very...

MR. TERAO: Yes, I see that.

MR. IOTTI: You've got to really, you've got to really look, okay?

MR. TERAO: It's handwritten in there. It's only on the first sheet.

MR. IOTTI: Right. The other ones, I believe, are still the same scale, but we'll confirm that for you. But the ones that are definitely off and they don't make any sense is the 29, 30 and 31. So (inaudible).

MR. TERAO: Would the scale have changed on the 28, 29 and 30?

MR. IOTTI: No way of knowing what the scale even was. The thing malfunctioned and we're supposed to get the correct figures.

MR. TERAO: Okay, that's all I wanted to know there.

CHAIRMAN BURWELL: Let's see, you're going to replace Figures what? 28?

MR. IOTTI: Twenty-nine and thirty.

CHAIRMAN BURWELL: Will replace A-28, A-29 and A-30.

MR. FINNERAN: We'll replace if we can... It may be that the data is not reliable at all and that there isn't anything to replace it with.

CHAIRMAN BURWELL: Okay, fine. But you will either confirm that we should use that or we shouldn't?

MR. IOTTI: Can we go off the record for a second?

CHAIRMAN BURWELL: Off the record.

(Off the record.)

CHAIRMAN BURWELL: Okay, so you will provide us something on those three figures.

MR. IOTTI: Those three figures, yes.

CHAIRMAN BURWELL: And furthermore, you will confirm the scale, the absistance (ph) scale...

MR. IOTTI: For 13, 14 and 15.

CHAIRMAN BURWELL: ... on Figures A-13, A-14 and A-15. Okay. Thank you. That particular discussion was held on a motion for summary disposition regarding U-bolts acting as two-way restraints. I just wanted to get that into the record to make sure... make sure which motion we were discussing.

With that, the next item I had was the motion on forced distributions and axial restraints. That word is axial, a-x-i-a-l. Axial restraints. Okay, the document we are discussing was filed on July the 9th, 1984, and this concerns the applicant's motion for summary disposition regarding allegations concerning consideration of forced distribution in axial restraints.

MR. FLECK: Okay, this is Hank Fleck again. I think I might have some questions there. One of them basically deals on page 7 of the affadavit, and it's a question that's generic, I guess.

In this case the paragraph that's referenced in this 1974 code, NF Article 3231.1 defines... and

it defines the subject matter in not too explicit terms. It deals with primary and second.ry stresses, and in this case it's been, I guess, adopted that this paragraph indicates that for the axial struts and/or snubbers receiving additional load from corginal (ph) restraint from the pipe would be treated as a secondary.

I take issue with that and I look at the 1980 code for clarification of what is meant for primary and secondary, realizing the '74 code has a lot of ambiguous areas.

And I submit the table from the 1980 code, and it's NF 3623.2-1. It's the elastic analysis categories and stress limit factors for Class 1, 2, 3 and MC linear type supports designed by analyses/piping supports.

Primary stresses has a Note 5. That Note 5 indicates for service levels A, B, C and D stresses induced by restraint of free-end displacement and anchor motions of pipes shall be considered as primary.

So my issue here is the secondary stresses generated here would, by my definition and the 1980 code definition, be primary stresses. And I guess this would be an area of dispute or contention.

MR. IOTTI: Hank, could you give me that table number again? Okay, so service level B allowable is

being increased over what it used to be.

MR. FLECK: Right. So the treating of or interpretation of the '74 code is noc in existence anymore in the '80 code. And I realize that interpretation has been going back and forth on what is primary and what is secondary.

Instead of wasting our time here to dispute how the code was worded in '74, I just looked to the '80 code as clarification of that. I know you are bound to the '74 code, but as an engineer, I look at what I would consider would be reasonable and the '80 code has clarified in my mind how that type load should be treated.

And I submit then if I look at a snubber that could be in this position, or two snubbers, if I use three times the A level, I'd be way over the failure load of the snubber because the fault load established by test is 10% of the test value.

So I really have a concern here that it should be treated primary, and I realized in your summary, your analysis, you used a conservative approach to determine the resistance of the dual supports, but I have no feel for whether you, if you used a realistic analysis, whether that would reduce such that you wouldn't exceed loading and snubbers.

And that... I come back to the '84 code as

far as for... or the '80 code as far as for the clarification of that interpretation.

MR. IOTTI: Well, here is, of course, a reasonably gray area that's to be discussed because implicit in this statement is that there is a concurrence that the rotational constraint of this particular (inaudible) needs to be modeled in the most conservative fashion.

That in itself can probably precipitate a discussion that can go on for a long, long time. Applicant's position is that the modeling of the restraints as a single axial restraint is being, in general, an industry practice and it's appropriate.

MR. FLECK: Well, industry practice is changing, though. I believe the plants in Arizona have modeled it differently now.

MR. IOTTI: Well, that's fine, but like everything else, you have to go back to when this was done, at the time when the design was undertaken. Now, we're not disputing the fact that we ought to review the present design of the plan versus what might be, you know, a good engineering view as to whether the support is capable of accepting a load or it is not capable of accepting a load.

And we have attempted to do that. But in doing

so we felt we could permit ourself a little bit more latitude, number one. We are developing the additional loads in a conservative fashion.

MR. FLECK: Not in all cases. In a lot you are, yes.

MR. IOTTI: In all case.

MR. FLECK: You mean you're not using the computer program?

MR. IOTTI: Even if it is in the computer program, you're modeling it in a conservative fashion.

MR. FLECK: Maybe if you elaborate, I can...

MR. IOTTI: Well, you're doing a response spectrum analysis in any case. You're not doing a time history. If you were to do a time history and if the piping is flexible enough, you'd find virtually no correlation between the axial load and the torsional load.

If your piping is rigid, however, you do. Okay? So that, first of all, tells you that unless you have a system that's rigid enough where you get coupling of the axial and the out of plane bending due to the stiffness effects of the system itself, where inertia is not a consideration, you're likely to find that when the maximum load occurs axially, your torsional load may be very little at that time.

MR. FLECK: It may, but it may not.

MR. IOTTI: I have just defined for you... We have actually ran time histories.

MR. FLECK: Okay.

MR. IOTTI: We have done a most simplified system and that's why I can make the statement that I cannot categorically tell you that they're not correlated. What we did, we looked at the point and we imagined that there was a restraint at that point.

So you input the earthquake at that point and at other points of the simplified system, and depending on whether the system is flexible enough so that at the restraint what you get is inertial effect and you find that correlation between the axial time history of the response and the out of plane bending time histories of the response are very, very, it's very, very low.

That would indicate that the peaks don't occur simultaneously to the point where I could even show you if you had a stiff enough system, the system couples through the stiffness of the members that join at that particular axial restraint, in which case the correlation could be as high as one.

So when the peak axial occurs, you'll also get the peak rotation. In that instance, yes, you would have to then add the two. Okay? So...

MR. FLECK: Okay. Now, what I'm... Sorry, go ahead.

MR. CHEN: This is Paul Chen. Can you hear me up there? Coming back to this business of the criteria for treating, you know, the loads as either primary or secondary, I think in the previous hearing Bruce said that the 1980 code, I think, clarifies the position of the 1974 code.

MR. FINNERAN: We'll make some statements about that and we'll let you... The code is not always clarified. Sometimes they're taking conservative approaches because of practices that they've identified, and that's what I would like to address.

MR. TERAO: Well, before we get into code philosophy and explaining what the code is trying to say, perhaps you could clarify some easier questions, and that is when you say that you're going to take credit for this three times the allowable increase, specifically, what part of the support are you going to be taking credit for?

Are you going to be taking credit for this in snubbers?

MR. IOTTI: You are taking credit on each of the two legs of the support, or the support itself.

MR. TERAO: All right, now...

MR. IOTTI: Well, we can go through it support by support as to where the weak link existed.

MR. TERAO: Okay. I think what we'd like to do is try to find out where you're going to take credit for this because if you're saying that you're going to take credit for the three times the allowable in the struts and snubbers, we definitely take exception to that because those are compliment standard supports and the NF criteria for 3231.1 is for any supports that are designed by linear elastic analysis.

MR. IOTTI: I understand.

MR. TERAO: Okay. So we would not accept that for any compliment standards so...

MR. FLECK: Or if we had an idea how many of these exceed the primaries.

MR. IOTTI: There weren't that many, but... And in some instances if we had used just level A or the, you know, the 1974 edition of level B and C allowables, we would exceed them.

If we agree that the proper way to model it is by putting 100% of the load onto one leg of the support.

MR. FLECK: Well, I guess the other thing, not knowing how you modeled and it sounds like possibly you used generic and not actual loops that were concerned

where they exceed loads. If you put in the proper stiffness of the pipe, which is internally just due to the pipe size, there are a lot of things we could do to reduce the load.

MR. IOTTI: We could put local effects in pipes. We could put local effects on the trunion. All of those were not included. Okay?

MR. FLECK: I guess what I'm looking for is one, I would like to have a definite feel of how many and where and look at them in pacific locations because I would probably say I agree with the '80 code's interpreation and not the '74.

MR. IOTTI: There's a total, there's a total of 36 out of however many of these supports we gave you, which is what? 198? Which would fail or would be considered to have failed the criterion of the code of 1974, assuming that you just take '74 in the strictest interpretation.

Everything is a primary strut, okay? So that's however many. Now, the next question is how many of these are snubbers? One, two, three, four, I'm telling you that the snubber type, that doesn't mean the snubber's the weak link.

Just telling you these approximately half, I would say, are snubbers. One, two, three, four, five,

six, seven, eight, nine, ten, eleven, twelve, thirteen, fourteen, fifteen, sixteen, seventeen, eighteen. Exactly half.

I was looking at this as to whether, where we had the remark that indicated where it failed. Well, we can make that... we know where the weak links are. We know that for some of these the weld turned out to be a weak link.

I can't make the general statement... I don't recall Peter saying that any of the standard support itself, where they're in trouble.

MR. FLECK: So the axial then was, may not be the area where you had to use three times?

MR. IOTTI: That may not be. We need to go back and see if we have a complete list.

MR. FLECK: I think that would be helpful before we argue and argue which interpretation is right. If you know where you're critical, and if it's axial members, then I feel strongly then.

MR. IOTTI: The crucial thing in our interpretation is that we rely on the ductility of the support, permit us to have that. That's what permit us to make that interpretation.

MR. FLECK: But an axial member o ductility could give you a compression stability problem or it

could give you a tension failure. You don't have the leeway of plastic bending.

MR. IOTTI: I understand.

MR. FLECK: Okay.

MR. IOTTI: So far as... You're right. If our problem is one of the supports which in fact does not bend or plasticize or whatever, then the whole argument is not valid because we're relying on that ability to absorb unilasticaly some portion of this load.

MR. TERAO: Well, one thing I do want to point out, the reason why we brought up the snubbers and compliment supports is because these type of trapeze supports are, you might call them a hybridization of compliment standards and structural steel.

Now, the component... I mean the structural steel aspects, where you do use a linear elastic analysis, I could understand your arguments to use three times the increase in stress for those portions, but as far as the compliment standard, there really is no basis for using that. The increase for those compliment standard (inaudible).

MR. IOTTI: I don't know how much time we want to take up, but chances are we have all of the information right here. Everyone of the support has gone through and identified where the failure occurred.

MR. FLECK: I think it would be helpful, not here maybe, but to provide that so we can either do battle or agree.

MR. TERAO: Yes, but it's... I'm saying that even a good compliment standard is not very easy to say where the failure occurred. Are you saying from tests or from analyses?

MR. IOTTI: No, no. There we would consider... Well, we're discussing failure. It's not failure here that we're discussing, it's whether you exceeded the recommended LCD values, okay, or, you know, per the normal level or whether we take three times the LCD value and so forth.

We're not talking about the component actually failing, per se, okay, although I guess conceivably that could be, could be the case. But...

CHAIRMAN BURWELL: This is Spot Burwell. I wonder if we couldn't handle this more efficiently if you people just tabulated that information.

MR. IOTTI: Well, the information is already tabulated. What we need to do is in the remarks column add explanation where, that are a little bit more lucid than these cryptic notes that we have.

CHAIRMAN BURWELL: Have we the information you have in your hand?

MR. IOTTI: Pardon?

UNIDENTIFIED SPEAKER: No.

CHAIRMAN BURWELL: Okay, we do not have the table that you have in your hand?

MR. IOTTI: No, you do not. This is...

CHAIRMAN BURWELL: Okay, fine.

MR. FLECK: That would have to be edited to be a little more presentable, I believe.

MR. IOTTI: And even this information that I have in my hand, unless we're willing to take time off and sit around the table and go through one-by-one...

MR. FLECK: No, I don't think...

MR. IOTTI: ...it's simpler for us to add on the "remark" columns where, which particular portion of the particular support could... A lot of these supports, of course, passed.

MR. FLECK: Yeah, I think you see our concern.

MR. IOTTI: Well, the other thing that we now want to... If you're going to quote us the '80, then, then we consider ourself free to also take advantage of the fact that, yes, the '80 code, it prescribed that you not treat it as primary but it permitted an increase in the allowable for level B and level C.

MR. FLECK: But you don't have a three times allowable...

MR. IOTTI: No, I understand.

MR. TERAO: Let's stay away from the '80 code.

MR. IOTTI: I understand.

MR. TERAO: I don't think I want to get into that, but we can get into an argument regarding primary and secondary stresses from the sense that you are regard...

END OF TAPE

MR. TERAO: You're going to take credit for it as a secondary stress. Now, we could take exception to that because what you're doing in that sense is, is, yes, I would agree that the pipe rotation is self-limiting but you could also argue that the pipe, even its displacements are self-limiting.

So, so, it's the same...

MR. IOTTI: Well, it's the (inaudible). You, you have to understand it is not only self-limiting but it's very, very small.

MR. TERAO: Well, okay, but that's the arguments that are currently going through the Piping Review Committee right now in the NRC is whether or not...I'm sorry, not Piping Review Committee, the PVRC.

MR. IOTTI: The PVRC, yeah.

MR. TERAO: Whether or not seismic stresses and pipes should be considered primary or secondary. So, from that standpoint, you're making the same type of arguments with your rotations.

MR. IOTTI: That's correct.

MR. TERAO: And from that standpoint we cannot accept it at this time. We understand the arguments. We understand why you're saying it's secondary, but at this point, it really has not been either resolved either in the industry or, specifically, not in the NRC to

accept that philosophy.

MR. IOTTI: Yeah, but you understand that we're making that argument at the same time and it's a complimentary argument to say, look, we don't really agree we even ought to model it this way, but if you force us to model it this way, okay, then, then it's something that we also ought to consider.

We, we also have evaluated this by our normal procedures, modeling a single axial restraint and evaluate it against the allowable of the code, the '74 code. In all instances, we're okay.

So, here, here, we're getting away from the design practice that was being used at Comanche Peak, and we're trying to defend an existing design. Will it, in fact, fail or not? We're not designing a system here. What, we're trying to prove to you whether the system could, in fact, fail or not.

And I think that it's appropriate in that instance to say, okay, suppose that you did have this rotation? What would...if you have a sufficiently ductile support there that can displace, you know, strain sufficiently to absorb that, what that to me means that the support isn't failing, you know, provided I can cycle it through and not, you know, and shake down to a lasting connection.

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See, we're coming from that point of view. We're not trying to say this is the method that we would use to design the system, far from it. If we were to start from scratch and, and, and design it with 100%, you know, 100% split, we would design it per, per the code. You'd have a little bit more massive support. But we're trying to defend an existing design.

MR. TERAO: All right. Well, let's go back to a statement you made very early where you said this was a standard engineering practice to model axial restraints with two snubbers as a single restraint in the piping model. I, I would agree to that.

In fact, as you pointed out in your affidavit, CYGNA even stated this, too. But, first of all, I'd like to point out that CYGNA configuration was not a trapeze type. It was strictly a, a pipe with two trunnions (Ph.) on it. It was not a trapeze type.

And second of all, standard industry practice is to model lugs or trunnions and single axial restraints, but then there are other considerations, design considerations that are made to account for any unbalanced loading. And we pointed this out to CYGNA, too.

But, but the trapeze configuration, especially the one that's welded to the pipe, is a very, is unique

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to Comanche Peak. I've never seen any design like that before. So, as far as using standard industry practice to model that particular restraint, I would have to take caution of that particular support configuration, mainly because even though you're angular rotations are small, the displacements where the struts attach to the beam itself may be large because of the large radius.

MR. IOTTI: Okay. I understand, but Gibson Hill and correct me if I'm wrong here, but where the letter arm was long enough. you know, where they had long trunnions, that Gibson Hill did not make the assumption of modeling it as a single axial restraint, either.

They did consider the fact that you would have the split, is, is where you're, you're talking relatively short trunnions that we're discussing here.

MR. TERAO: Well, that definitely should have been in the write-up because in the write-up there is no mention of taking credit or modeling in longer arms. And that is where the major concern would be, is in the trapeze with the longer arms. There has to be some...

MR. FINNERAN: What we're saying, what we're saying is that in some cases the analysts have chosen to model these in to the analysis. I don't know what their criteria for doing so was or even if they have a

written criteria. I think that's a matter of looking at the arms and at the rotations that the piping analysis would predict and make a decision that we ought to model these in as, as restraints.

And they have done so.

MR. FLECK: Yeah. Some of the plants, the 79-14, they've done that.

MR. FINNERAN: Well, we've done that.

MR. FLECK: We've done it on (inaudible).

MR. FINNERAN: Again, the original anaylsis, the piping person doesn't know whether you're going to have a single strut or double strut, and he puts it in as a restraint in one direction.

Then the 79-14 always comes back as built (inaudible).

MR. FINNERAN: That's precisely what happened with us. When they did see in our 79-14 copies of the drawings and the supports were supplied by the analysts.

MR. FLECK: So, that's, that's where the fact would come back where the piping analyst now would have to evaluate whether or not there is a sufficient tordinal (ph) restraint for that size pipe, and he would have to model it in, then.

MR. FINNERAN: In fact, for some of ours,

they have decided that it would be significant and have modeled them in.

MR. TERAO: Yeah. Well, I think that's part of our problem is, I understand when you keep saying some of, some of the time they do it. I think what we want to know is, we would be concerned where they don't do it.

If there's no written criteria, I agree that it's up to the engineer to, for him, for himself to make that decision, but on the other hand, when you're, when you end up with restraints that are different in the industry, what one engineer does, another engineer may not.

And we're in this last phase now. We're in the as-built check, where this is the, this is the last chance they have to take account of any of these problems.

Now, just for the fact that you say you have seen it done, I don't think that's going to fly as far as the final as-built check. We have, we have to understand where there could be a problem.

MR. IOTTI: Well, I don't see what else we could do other than give you the complete list of everyone of them and tell you, you know, what's been done or as the case may be, what hasn't been done.

And for the cases where it hasn't been done, is how does that support...

(CHATTER)

MR. BURWELL: Gentlemen, there's an awful lot of conversation going on amongst the different members, and I guess I was guilty of that, too. Where are we? Are we not at a place that you could submit the table, I assume? Does that clear up your statement, Dr. Chen?

DR. CHEN: No, I think we have a little bit more in the way of a request, and we're trying to formulate that at this point.

Can we just take a break?

MR. BURWELL: Okay. Off the record.

(OFF THE RECORD)

MR. BURWELL: I believe Mr. Terao has a statement to make.

MR. TERAO: We just had a discussion and I think in general what we're saying at this time is we can't accept a summary disposition the way it's written, and we're going to need quite a bit more information before we can attempt to start addressing where we feel the real problems are because at this point, we don't know exactly how big the concern is. We don't know to what extent our concerns are even

valid.

The first thing that we really need is, is a listing, let's say a quantification of the different types of axial restraints that are in the plant. Your Figure 1 which shows the Type 1, Type 2 and Type 3 restraint are, we might start at that point. And if you can at least tell us how many in Unit 1 and also in Unit 2 there are, that would at least provide a starting point.

What I'd like to discuss is a little bit of our concerns with each of those types and perhaps in that sense you might be able to address those concerns from a generic standpoint on how it was addressed in design.

Okay. Starting with the Type 1, this is the trunnion with the trapeze, using either snubbers or struts. Our concern here, as pointed out earlier, is the fact that the snubbers and struts are not allowed by three times the design allowable increase. And what we would like to know is what the impact of the rotation can be on the snubbers and struts as far as overloading them.

And I would also like to point out that the rotation has to be combined with the axial loading. I, I understand your point of the conservatism involved

but then, again, with, even with the standard response spectra (ph) analysis, we don't accept the decoupling of the rotation from displacements.

So, I, I understand the conservatism involved, but I can't...

MR. FINNERAN: Why do you say that that allowable is not applicable to standard components? As I recall the '74 code, that paragraph applies to, and I wish I had a copy of the code...

MR. TERAO: I have that section, if you like. The '74 code says you're allowed three times the allowable design limits of Appendix 17-2000.

MR. FINNERAN: Right.

MR. TERAO: Appendix 17-2000...

MR. FINNERAN: Those, those design limits in Appendix 17-2000...

MR. TERAO: Are design limits based on linear elastic analysis. The component standard supports are not designed using the linear elastic analysis.

MR. FINNERAN: Aren't, aren't the methods for determining the capacity of standard component supports spelled out in 17-2000?

MR. TERAO: I'm sorry, I, I don't understand. What was the question?

MR. FINNERAN: The methods for determining

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the allowable loads for standard component supports are spelled out in Section 17-2000. So,...

MR. TERAO: No, they're spelled out in NF.

MR. FINNERAN: That's what I want to check.

MR. FLECK: What is that paragraph? I don't have the...17-2000.

MR. FINNERAN: 17-2000 is what we need.

MR. FLECK: Yeah. It's all right.

MR. FINNERAN: I think you're right. You're right. And I withdraw my comment.

MR. TERAO: One area with these trapeze type supports that we would like addressed is, first of all, if you can tell us what the maximum rotations can be or typical rotations and what is the worst case length of the trapeze itself. Then determining what the displacement can be at the support itself.

One thing you might want to take credit for or at least consider is the loss motion in the snubbers themselves. In other words, if the angular rotation of the trapeze is less than the, the loss motion of the snubber, of course, there is no additional loading due to rotation, but that would not apply to struts. With struts there is really no gaps.

MR. IOTTI: Haven't seen one yet (inaudible) but that's okay. I understand.

MR. TERAO: All right. One thing we have not discussed before but one of my primary concerns was that with this particular arrangement, that was not addressed in the summary disposition, are the local stresses on the pipe. And I think this is one of the major concerns with this design.

MR. IOTTI: I thought that we had addressed... oh, I guess not. We did this for the lug.

MR. TERAO: There was a Gibbs and Hill analysis would show that there is no increase in general bending stresses, but I don't believe the concern was even the general bending stresses and the piping system. It's the local stresses...

MR. IOTTI: Secondary and local.

MR. TERAO: ...induced into the pipe. And it's my opinion at this time that probably this Type 1 trunnion will give you the most problems of the, of the three. And I would like to see the write-up, specifically, addressing this trunnion in detail.

With the other two trunnion types, this is the Type 2 trunnion which is simply a two trunnions, 180 degrees apart with snubbers or struts. This was addressed by CYGNA.

One area that we'd like addressed there, in particular, is what is the maximum length of that

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trunnion?

MR. IOTTI: Aren't you really interested in what is the maximum deflection as opposed to the maximum length?

MR. TERAO: Yes, basically.

MR. IOTTI: This is Bob Iotti. I was asking Mr. Terao whether, what he's really interested in is, in fact, the maximum deflection at the location of the support on the trunnion as opposed to how long the trunnion is.

Because if there is no rotation, it really doesn't matter how long it is.

MR. TERAO: But the reason why I, I asked for the length is because in looking at Type 1, you may be able to determine what is a typical or maximum bounding rotation of the pipe, in general. And once you know what that would be, you could at least determine, knowing Type 2, what the...if you know what the trunnion length is, then you can determine what the displacement is.

MR. IOTTI: Oh, I don't think we're going to do that, Dave. The, the pitfall for doing that is that you're going to be mixing a very large rotation which occurs for a very short trunnion, with a very long trunnion where there may not be a rotation. And you,

and you're creating for yourself a problem that does not exist. So,...

MR. TERAO: Well, okay. That, that may be true. I...

MR. IOTTI: We're already, we think, conservative and we're adding further conservatisms on and that could lead us into answers that are incredible.

MR. TERAO: I guess what I'm trying to do is shorten the process. Of course, the alternative is to look at every one of these and what we're trying to do is at least screen out where the pro lems can be and then we can focus on those that we feel there is concern.

But I, I agree with you. What we are really interested in is the displacement at the attachment of the snubber or strut to the trunnion.

UNIDENTIFIED SPEAKER: Would you want the rotation at the same time because that's what's causing the displacement?

MR. IOTTI: Well, what you will, what you realize is that in order for us to give you that information, we have to look at every one of them in any case to determine where, where are the rotations, how large are they, and, in fact, pick the maximum.

This is what we did for the trouble trunnion;

we identified every single one of them in Unit 1 and (inaudible).

MR. TERAO: I see.

MR. IOTTI: Okay. So, we'll just do the same thing for, for the other types and give you all that information.

MR. TERAO: Fine, that would be the best way.

MR. IOTTI: If you have that concern, we want to lay your concern. We're not trying to minimize the work. We're trying to satisfy you that there is no problem in the plant.

MR. TERAO: And as far as the Type 3 type trunnion, it's basically the same concerns as the Type 2, but I would like to point out, again, that, that what CYGNA had looked at was mainly Type 2 and Type 3. And there's been a lot of discussion regarding Type 2 and Type 3, but what they're, what has not been discussed in detail is the Type 1 trunnion. And I think this is probably the worst case trunnion.

MR. IOTTI: Okay. We'll have to provide you with that information. Obviously, we don't have that with us today.

(PAUSE)

(CHATTER)

MR. HORIN: Spot, could we go off the record?

MR. BURWELL: Off the record.

(OFF THE RECORD).

MR. BURWELL: During the break, we had a discussion of Figure 1 of the different types of trunnions on the support. As a result of that discussion, I believe Mr. Horin has a statement to make.

MR. HORIN: During the break we discussed the information that the staff has requested with respect to the different types of supports identified in a figure included with our affidavit regarding the axial restraints.

And the staff has questions with respect to each of the types of supports identified in that figure. And applicants intend to provide information to the staff to respond to their questions.

And there is an outstanding question which we have not resolved but will resolve outside of this meeting as to the precise scope of the motion and the appropriate scope of the staff's response to the motion.

I think at this point we should just move on to the next set of questions.

MR. BURWELL: I think I need to make one statement, and I think that is that the staff will, of course, search out, pursue its concerns on this matter to a final conclusion.

And with that, I think we're ready to move on to the next question.

(PAUSE)

MR. TERAO: Okay. Well, I guess most of our concerns were with Type 1 restraints, and we did have some questions about the Type 2 and 3, but I think we did describe what we wanted addressed earlier this morning on the Type 2 and 3 as far as the rotations and displacements.

MR. IOTTI: You mean the rotation and the displacement and my understanding is that you also wanted to have the detail out of all of those that we looked at as to whether, where, what the weak link was.

MR. TERAO: Right, as far as...

UNIDENTIFIED SPEAKER: The ones that would have exceeded the normal allowance?

MR. TERAO: The ones that would exceed the primary of the axial involved.

UNIDENTIFIED SPEAKER: Right.

MR. IOTTI: I just wondered, maybe we could

even make the copy because that exists already?

(CHATTER)

MR. IOTTI: Because we have all of the rotations. These are, of course, the free rotations except in those few instances where we went back and remodeled the whole thing and also we have the actual rotation when you put the rotation constraint in.

MR. TERAO: Okay. Aside from that, we have no questions on the lug portion of the write-up.

MR. IOTTI: Meaning that's acceptable or not acceptable?

MR. FLECK: Right. It would be...as far as what I have seen, I have no questions that would question what you've done.

THE REPORTER: Would you speak up a little bit, gentlemen?

MR. FLECK: Should I repeat that?

THE REPORTER: Yes.

MR. FLECK: On the lug portion of this item, I have no additional questions. So, I would be satisfied with what has been presented.

MR. TERAO: I did have a few clarifying questions. Maybe you could...on one of your tables, Table 4, I wasn't quite sure of some of your notations there.

MR. IOTTI: Table 4?

MR. TERAO: On Table 4. There's...in the fourth column, there's an L value. Could you describe what that L value is?

MR. IOTTI: That's the length of the arm.

MR. TERAO: That's what I thought it was. And these, when you call these trapeze supports, are these...this is not the single trunnion trapeze support. These are the double trunnion?

MR. IOTTI: These are the double trunnion.

MR. TERAO: Okay. I guess...

MR. IOTTI: Tied to a tree.

MR. TERAO: I've never referred to those as trapeze types. So, I was a little confused. So, L is the length of the trunnion itself; correct?

MR. IOTTI: I think it's from the center line of the pipe because the computer would measure it from the center line of the pipe.

MR. TERAO: From center line of pipe to where the support is attached or to the end of the trunnion?

MR. IOTTI: To where the support would be attached.

MR. TERAO: And just a clarification on Table 2.

MR. FINNERAN: I think that L is the length of the, the coupling length.

MR. TERAO: The total.

MR. FINNERAN: It's the coupling length between the compression and the...

MR. TERAO: Length between the two supports.

MR. IOTTI: Let me confirm that for you, Dave, okay?

MR. TERAO: Oh, okay.

MR. IOTTI: But I believe...let me just confirm it to make sure. I'll tell you in a second. L is the total length of the space between supports. Okay.

So, it would be twice that dimension I just gave you.

MR. TERAO: All right. Thank you.

MR. IOTTI: Let's see, Table 2, you said?

MR. TERAO: Yes. On Table 2 it says change of adjacent support loads for rotation of constraint and on rotation constraint analysis.

Could you just explain just briefly what, what this table is?

MR. IOTTI: Okay. When you have the piping run, some of those supports will have a rotational resistence and some of them do not. When we reanalyze

we distinguish between the affect on the support, the trapeze type support itself and also the other supports.

What, what this table is intended to show is that the other supports are not affected. If there is an affect of modeling in the rotation constraint, it's strictly on the trapeze type supports.

MR. TERAO: Okay. So, it's the affect on the system of the trapeze type supports?

MR. IOTTI: Right.

MR. TERAO: Okay.

MR. IOTTI: That's why you see three tables. One is on the piping, one is on the other supports and then the third one is the actual supports (inaudible).

MR. FLECK: Okay. So, when you say that these are the supports, immediately following or preceding...

MR. IOTTI: All others (inaudible) whatever they are.

MR. FLECK: That's what I thought.

UNIDENTIFIED SPEAKER: Excuse me. I'm not, I'm not...I don't know if it's a problem with the air conditioner here, but I'm having trouble hearing you. Maybe you could turn the mikes (inaudible) would be fine.

MR. TERAO: All right. That's it for axial restraints.

MR. BURWELL: No further questions on (inaudible)? All right. The next item I have is local displacements and stresses.

MR. IOTTI: Before we go on to the local displacement and stresses part, irrespective of what is being, what would be decided on the Type 1, I have an action item on Type 2 and 3 that I'm to provide to you as soon as possible. And this is the rotations, the deflection, the lengths for each one of the type 2 and 3 supports and the finishing of which of weak links, you know, which one of those that would be computed to exceed the old allowable values (inaudible).

Is that the only one that you have?

MR. TERAO: Yes.

MR. IOTTI: Okay.

MR. CHEN: Is that basically going to be an extension of that table (inaudible)?

MR. IOTTI: It is the table that I have. It's just that we want to have a last opportunity to go back and provide some clarifying statement to it. And the table exists already.

MR. BURWELL: Okay. I have the local displacements and stresses.

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(PAUSE)

MR. BURWELL: Okay. I believe this is yours, isn't it, John?

MR. BRAMMER: Yeah, are we ready to go? I'm John Brammer from ETEC. And most of my comments are on Finneran's affidavit and attachments that go with it.

And to start out with, on Page 3 I'd like to clarify the allowables that are used. Some of this has gone on in the past that I've been cognizant to. One thing I noticed, it says applicants have utilized the conservative allowables employed by Gibbs and Hill for assessing local pipe stresses.

Well, based on the affidavits, I assume this is Equation 11 allowables?

MR. FINNERAN: Well, actually, it's a whole set of, of allowables. It's Equation 8 and 9 and 11 allowables. Actually, they have...yeah.

MR. BRAMMER: Okay. But in the calculations, you did, in your attachments you used 11 all the time?

MR. FINNERAN: Yes.

MR. BRAMMER: For the pipe and 3SM for the, the structure.

MR. FINNERAN: Except that the equations for the assessment of these stresses have been adjusted as

Gibbs and Hill adjusted them in the allowables that they have set up to assess the piping.

Our edition of the code...I don't know if this is in the hearing yet or not, but our edition of the code does not give any direct guidance or in C for assessing these localized stresses in the piping.

So, Gibbs and Hill had come up with some modified equations to use to assess the local stresses and the p.ping. And that's the ones that we used. It's the document that they've always used in...so, we used those values.

MR. BRAMMER: Okay. Now, but in your calculations, didn't you use Equation 11 allowables?

MR. FINNERAN: These modified equation 11 allowables, okay.

MR. BRAMMER: Okay. Then another question on this. You refer to ASME Code Section NF 32-31A, governs the consideration of these types of stresses box frames and so forth.

Well, according to that article, it says but not thermal or peak stresses shall be limited to three times the stress limits.

And this is what we're talking about, isn't it, thermal and peak stresses?

o? Well, you explain it.

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MR. CHEN: Just off the record.

(OFF THE RECORD).

MR. BURWELL: Okay.

MR. BRAMMER: Okay. Then the next bunch of comments I have are on Page 5. You say the results of that analysis demonstrated, even when including local stresses included in the frame from the thermal expansion of the pipe are those...all stresses in the frames are less than code allowables.

Well, I have some questions on your analysis on that that we'll get to a little later. So, I can't really buy that statement at, at this time.

MR. IOTTI: Which statement, I'm sorry?

MR. BRAMMER: The first paragraph up there more or less.

MR. IOTTI: On Page 5, where...

MR. BRAMMER: Are less than...

MR. FINNERAN: Are you in the affidavit?

MR. IOTTI: Yes, he is.

MR. BRAMMER: No, I'm still on, on your... we can go to the affidavit now and I'll my discussion on that...

MR. IOTTI: The affidavit.

(CHATTER)

MR. BRAMMER: Oh, yeah, the affidavit. Excuse me,

yeah.

UNIDENTIFIED SPEAKER: Page 5, about four lines down from the top?

MR. IOTTI: We're in the motion so I guess... I think you're reading from the motion.

MR. BRAMMER: Well, let me read it. This is the affidavit.

UNIDENTIFIED SPEAKER: Page 4.

MR. IOTTI: Yeah, I thought that was really domething different?

MR. BRAMMER: Okay.

MR. IOTTI: And you say you cannot buy that?

MR. BRAMMER: Not...I have some questions on...

MR. IOTTI: We'll get into it in the analysis, I gather?

MR. BRAMMER: Yeah, in the analysis.

UNIDENTIFIED SPEAKER: Could you just explain on that same statement what class we're talking about there? What class piping? Is that Class 1 or Class 2 and 3?

MR. FINNERAN: This particular frame is on Class 2 and 3 piping.

MR. BRAMMER: Okay. Then there's a statement ...

MR. TERAO: Are you done, Dave, or you have...

MR. IOTTI: Yes.

MR. TERAO: So, are you saying there's no frames on Class 1 piping?

MR. IOTTI: Yes.

MR. BRAMMER: Okay. And then you have a statement here, I should note that these loads and stresses would be greater than those encountered in the other supports of this type because of the higher temperature of this pipe and the fact that the pipe is stainless steel and the greater thickness of the pipe affording less flexibility and thus employing greater loads.

I think that that's too general a statement. It looks to me like you do have a thicker pipe. Right, it is stiffer. If you go to a thinner pipe, it's less stiff but it also will take a lot less load.

I think it depends on the individual...

MR. FINNERAN: I, I think I disagree with that. The load generated is based upon the (inaudible) flexibilities between the piping and the support.

MR. BRAMMER: That's right, yeah, and there's no doubt about that.

MR. FINNERAN: The stiffer the pipe, the

higher the load that would be generated.

MR. BRAMMER: But with a thicker pipe, the pipe could take a higher load without being over-stressed by ...

MR. IOTTI: I think he's not disagreeing with the loads being higher. He's disagreeing with the stresses being greater.

MR. BRAMMER: No, I know. The stresses could be higher.

MR. IOTTI: The load is related to the stiffness and the capability of the (inaudible) and the stresses also related to the thickness of the pipe.

MR. BRAMMER: Right, right.

MR. IOTTI: The two kind of go hand in hand. And we have convinced ourselves that this statement was true. So, we'll have to...

MR. BRAMMER: Also, the stiffness of the box frame will vary between supports, too, and that will have an affect on it. So, I think this is a little over-generalized here.

MR. IOTTI: Well, is there any specific action item that you want from us with regard to this? Do you want us to further justify why the statement was made by example?

MR. BRAMMER: Yes. Okay, then,okay. On

Page 6 it's the same type of statement under the last sentence on your answer; by taking this conservative approach, we have also effectively bounded the remaining supports on low temperature lines.

Again, you've done one support and I don't think you can do it with one support because, again, it depends on the pipe size, the pipe thickness, support stiffness.

MR. IOTTI: I guess we'll answer that one the same way as the former one.

MR. BRAMMER: Yeah.

MR. IOTTI: We'll give you more than one example. Okay. And try to pick the two bounding cases.

MR. BRAMMER: Okay. Then on Page 8, okay. Again, I have some questions on the attachments on the analysis, so that the statement at the, to the answer at the bottom of Page 8, I'm not sure that I can agree with at this time.

MR. BURWELL: Well, I take it we will go into these questions of yours when we get over in the later...

UNIDENTIFIED SPEAKER: (Inaudible).

MR. HORIN: What is the specific statement he had a question on?

MR. BRAMMER: Their, their results also

demonstrated all stresses in the frame and (inaudible) were far below the allowables used by CYGNA.

And then the same, the same is true up in Answer A up at the top, as shown in the attached analysis, Attachment C, D, and E, inclusion of the thermal expansion affects of the pipe with other loads and assessment of the anchor result in no over-stressed conditions.

MR. IOTTI: I'm a little confused as to why you quarrel with the, on Page 8. All we're saying here is the CYGNA analyzed the similar supports, and what they found for that support was that their stresses were below their allowables.

MR. BRAMMER: Yeah. I, I guess what I'm really questioning is at the top of the page.

MR. IOTTI: Oh, okay.

MR. BRAMMER: On the answer there.

MR. IOTTI: I, I may be looking in the wrong, wrong place.

MR. BRAMMER: Yeah, okay. Okay. On Page 9, these stresses are right, in the last answer on Page 9. These stresses are assessed by each of the support design organizations on a case by case basis when deemed appropriate by the engineer.

Is there some screening criteria or something

that you have or is it just each individual's judgment that goes into that?

MR. FINNERAN: Well, each, each group has a little different approach to them. Let me take just a second here to make sure where we are.

(PAUSE)

MR. FINNERAN: ITT, as we state later on Page 10, ITT uses the AWS approach in 10.5.1.

MR. BRAMMER: Yeah, that's on the ones they decide to analyze, I take it.

MR. FINNERAN: I think they...well, I can't answer that one, specifically, because I, I think they assess them pretty often in their calculations. They have an engineering standard that they developed, utilizing the AWS approach. And it's utilized quite a bit. So, it's...there probably is some judgment exercised by the engineers whether to always do the calculations or not if the loads are very well, but the calculations are done quite often.

In the case of NPSI, in their design guidelines they have some general statements about using reenforcing plates if deemed necessary by the engineer. And there's not very much precise guideline beyond that.

In the case of PSE, we really don't have anything written down like ITT or NPSI has that would

serve as guidance for the engineer as to when he should do them and when he should not. So, I would say in those cases it,it is left up to the judgment of the engineer.

UNIDENTIFIED SPEAKER: Do you have a problem with that?

MR. BRAMMER: No, I just was curious as to what, what their criteria was.

Okay. On Page 11 where you're talking about this tube to tube ratio, where CASE said it was less than .4, looking at your drawing, the dimensions of that, that end bracket is 4 by 2-7/8 inches. So, depending on which way the bracket sits on the tube steel, they could be right or...

MR. FINNERAN: I don't know where you got your dimension, but the dimension was 4-1/2. The dimension that runs across the tube steel on the bracket was 4-1/2.

MR. BRAMMER: Okay, in one direction, but it's 2-7/8 in the other direction.

MR. FINNERAN: Yeah, well, the direction we're talking about here as far as the ratio is across the tube steel.

MR. BRAMMER: It's set so that it is across and not turned the other way?

MR. FINNERAN: That's correct. The 4-1/2 dimension is across the tube steel, across the 8 inch width of the tube steel.

MR. BRAMMER: Okay.

MR. FINNERAN: If you...don't we have...isn't that attached?

MR. BRAMMER: Yeah, yeah, that's where I got it.

MR. FINNERAN: I could show it. I could, I could make that clear for you if you wish.

MR. BRAMMER: The bracket is, the picture of the bracket is attached.

MR. FINNERAN: Yeah, I could show you.

MR. BRAMMER: There's also a page out of the vendor's catalog, I believe. You can see that the dimension we want is that dimension in and out of a page, okay.

So, if you drew a picture of the bracket, here's the bracket, and that would be looking at it like this, okay. Okay. And if you go to the vendor's catalog sheet or this bracket, you see that the dimension we want is d). You mention d)?

UNIDENTIFIED SPEAKER: Yeah.

MR. BRAMMER: Okay. You want that dimension.

MR. FINNERAN: Okay, right.

MR. BRAMMER: And if you go...I've circled d) for this number one (inaudible) and it's 4-1/2 inches.

MR. FINNERAN: Okay. This is the one I was curious about because that's 2-7/8...

MR. BRAMMER: Yeah.

MR. FINNERAN: ...as near as I can read it.

MR. BRAMMER: Yeah. Well, that wouldn't be a ratio. I don't know what you would ratio it to because that's along the whole...

MR. BRAMMER: Yeah, because that doesn't matter there. Okay. Then a question on this Table 1. You have a list of stresses there. What, what loads are included when you determine these stresses? Does this include the local stresses due to thermal expansion in the, in the zero gap supports or the whole bit or is it just...I guess...

THE FINNERAN: It seems to be the design loads for the sports themselves, that we have the design loads that we have gotten from the packing analysts to assess these particular supports.

MR. BRAMMER: So, it doesn't include any stresses or loads that you would get due to this zero clearance wall to wall analysis that you do? It's just your basic stresses?

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MR. FINNERAN: I guess I'm not sure where you're coming from on that question.

MR. BRAMMER: I think the question is, do these stresses include the stresses due to constraint of free end displacement?

MR. FINNERAN: Yes. These are, these are the, these, I believe, are the Level C loads for these supports. They come from the piping analysis and by our design...we do. We do.

MR. IOTTI: Okay, but you don't have to.

MR. FINNERAN: No, we don't have to, but, but our particular design spec includes for Level C, the loading, the SSE load plus thermal plus deadweight.

MR. CHEN: Plus the affects of a constraint of the...the constraint of the support.

MR. FINNERAN: Oh, no, no, not those. Not those.

MR. CHEN: Yeah, okay. Yeah.

MR. FINNERAN: You're talking about the radial, the radial thermal expansion of the pipe. No, those loads wouldn't be in here. Let's see, what is Table 1? Why do we reference that?

MR. BRAMMER: I think it's referenced on Page 12.

MR. FINNERAN: Where?

MR. BRAMMER: Page 12, about two-thirds of the way down, the first paragraph.

MR. FINNERAN: Okay. So, that's...what I told you was correct, that the loads of these supports were assessed (inaudible) the loads that came from the piping analysis.

As far as the restrain of (inaudible) displacement, it would be the axial expansion of the piping that the piping analysts do. That constraint of (inaudible) displacement is in these loads.

Now, we're talking here about stresses induced and tube steel walls from, from attachments to the tube steel, I believe. And I really don't see what connection a radial expansion with piping would have to those loads.

MR. BRAMMER: Well, it adds, it adds extra ...this is...it adds extra loads to your tube steel; right?

MR. IOTTI: Got radial expansion?

UNIDENTIFIED SPEAKER: Depends on a relative stiffness.

MR. IOTTI: Depends on a relative stiffness. I, I suspect that most pipe being far more flexible than the ones (inaudible) restrain it or (inaudible)

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very little to that restraint.

MR. BRAMMER: Well, yeah, we've gone through a bunch of stuff here on zero clearance and determine loads due to zero clearance.

MR. ICTTI: Yeah.

MR. BRAMMER: Okay. Now, these loads, it seems to me, should be added to the, to the loads that were used to calculate the stresses in Table 1 to get the whole picture.

MR. FINNERAN: Well, I don't believe that any of these supports in Table 1 are on the nature of a box frame around the pipe.

MR. CHEN: Okay. That may be I was...

MR. FINNERAN: All these are, these are like a canterlever (ph) tube steel beam with a bracket attached to the wall of the tube steel and then the strut coming down with a clamp around the pipe.

So, those radial expansion affects from the piping wouldn't, wouldn't have any affect on these attachments.

MR. BRAMMER: Okay. On...are you saying, John, that there would not be configuration where you had a box frame around a pipe?

MR. FINNERAN: That's right. I don't think any of these are that kind in Table 1. What we're

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assessing here is attachments to the wall of the tube steel like rear brackets.

MR. BRAMMER: Okay. On Page...

MR. FINNERAN: I think every one of these are rear bracket type attachments, just about.

MR. BRAMMER: On Page 13, what's the basis of this 16,1/16th inch deflection guideline? That, it doesn't seem to me that that tells you much about the stiffness of a support.

(END OF TAPE 3)

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MR. CHEN: I think this was something which was, this arose in the hearing before, this is the significance of the 116 is the (inaudible) criteria and I think the position of this package is really, the signficance of this report is really what matters and not what, the deflection of the support is under the (inaudible) is.

MR. FINNERAN: You got to look at what we're addressing here. We're addressing an allegation by Mr. Doyle, that related to the calculation of deflections. The allegations in Mr. Doyles deposition are all centered around deflection calculations, and that's all we're addressing here.

MR. TERAO: Just to ease Johns' mind I just want to point out that there's a seperate issue on generics. The difference is that the staff is addressed and John Fair has addressed that point there, as far as differences used for the support, so I think we should keep that seperate from what this particular issue.

MR. BRAMMER: So there's a 6th of an inch in the generics difference will be covered in that, is that what you're saying?

MR. TERAO: Yes.

MR. BRAMMER: Okay, on page 14 down at the bottom of the page, you say the actual difference of the support is 1.347 times 10 to the 6th pounds per inch. Given that the generic, and this is at the top of page 15 now, given

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that the generic stiffness for this support is five times ten to the 6th pounds per inch, the actual stiffness is well within the acceptable range of stiffness, given that the deflection is slightly more then 1/16th. I don't understand that, it looks like your actual stiffness is about a quarter of the generic stiffness, and that's an acceptable range.

MR. IOTTI: Well again to answer y ır question, you have to place it in two contexts, where does Mr. Dolyes allegation come from, and what his concerns are, and what we were able to prove to ourselves through a different item of work which eventually was involved in his affidavit on generic versus actual stiffness. To us, that work, is you were to review that affidavit it would show that the difference which is less then a factor of five, in stiffness there's not much difference in this pounds, at this range. So it's in that context that you see the statement that that is an acceptable range. Mr. Doyle is also concerned with the 1/16th of an inch because he took that 1/16th of an inch guideline ot be a sacred cow that could not be exceeded in any case.

So youhave to address the fact, ofyou know, so what if you exceed 1/16th of an inch by a slight amount, there's nothing magic about 1/16th of an inch to begin with.

MR. BRAMMER: Yeah, it doesn't seem like it means much to me in any place.

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MR. IOTTI: No but apparently Mr. Doyle misinterpreted that as being an inflexible rigid rule that if you exceeded 1/16th of an inch, something catastrophic is going to happen to you, so this is why that statement was made, if you care to add to it. But it really has to be read almost together with that other affidavit otherwise it can't stand on it's own.

MR. BRAMMER: Is this something that Fair is going to handle, I guess?

MR. TERAO: Yes.

MR. BRAMMER: Okay, well then I guess my next comments are on the attachment A. The first question I have is on the temperture distribution, you just assume a linear temperture distribution across the tube steel and assumed that that went the whole length of the support.

Well I'm not sure that that is conservative or not because actually you have line to line contact on the tube steel and by the time you get off to the corners you're quite a way away from where your contact is so the temperture would seemingly drop quite a bit, which could effect your loads considerably.

MR. FINNERAN: Well keep in mind we're not trying to do an exact analysis of this thing. We're trying to do an approximate analysis by manual methods, so we've made some approximations and some simplifing assumptions.

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MR. BRAMMER: I'm just a little concerned on some of these it may be a little over simplifing in this.

MR. FINNERAN: What was your comment again?

MR. IOTTI: Oh, I understand, at the line contact this tends to overestimate the temperture in the pipe in the region near the line of contact, you agree with that, your question is, as you move away from the line of contact and towards the edges of the box frame, this particular approach may, in fact, underestimate, over estimate the temperture and under estimate the restraining effect. Let's see I think we did some work to prove to ourselves that this was not that far off, I have to dig it up, because I had the same question, at one time on it. I would agree that it's not linear, but there's two compensating effects. One you are over estimating the temperture near the edges, okay, and that means you are under estimating the restraining effect, on the other hand you are also under estimating the temperture at the line of contact in this instance, and I don't recall but on our computer we ended up with something which tended to say the two in fact, were canceling out.

MR. BRAMMER: Well, if you could show me something.

MR, IOTTI: I'll have to go back and dig up these calculations because this was not done just cavalierly as this, 'cause I had questioned that approximation also, myself. So make an action item, I'll just have to dig it up.

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Peter may have it, we may want to call him.

MR. BRAMMER: Then you have a statement on the same page, that the air film insullation effect is considered actual tube temperture, actual tube temperture is much higher thus this approach is very conservative. Well that seems to be in conflict with your analysis, because what you did in the analysis is you subtracted deflection of the tube ex-spansion deflection from the tube from the net. Okay so if you have a higher temperture you would be subtracting more so the net deflection would be less, and you would have a lower load.

MR. IOTTI: That's correct, that's why we say it's conservative, 'cause we're over estimating the load by neglecting that we're over estimating.

MR. BRAMMER: Well that's not the way, the actual tube temperture will be much higher, 'cause then the thermal expansion would be much higher and you would be subtracting more.

MR. IOTTI: Correct, so the load would go down, from the standpoint of load this approach is conservative, from the standpoint of expansion of the tube it's not. That's the way.

MR. FINNERAN: The problem here, you've got to look at it as the interface between the tube and the pipe, and if the tube heats up less, andit has more of a constraining

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effect on the pipe and has higher load.

MR. FLECK: The colder it is the higher the load.

MR, IOTTI: Wait just a minute I just want to make a note you need that additional calculation which shows the delinear distribution of tube steel across and along the tube steel, is reasonably appropriate.

MR. BRAMMER: Then down in item two here where you, okay, where you calculated the pipe wall stiffness, this formula is for a thin walled vessel and your R/T ratio is not in the range of thin walled vessels.

MR. FINNERAN: Well I don't know, if you can say it's for a thin walled vessel, this particular.

MR. BRAMMER: Well that's what Rorch says.

MR. FINNERAN: Well he says for an R/T greater then ten, I don't know whether he catagorizes that as a thin walled vessel, he just puts that as a peramiter.

MR. BRAMMER: Well that's what the formula in his table, that I understood.

MR. FINNERAN: Well I'm looking at the formula and I don't see any reference to thin walled vessel, it just gives the R/T ratio.

MR. BRAMMER: Yeah, but when he gives the criteria of his formulas they are for thin walled vessels with R/T of greater then ten.

MR. FINNERAN: Well all I can state to that is again

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where we were trying to do an approximation here.

MR. IOTTI: Yes, but we have some for the thicker vessel too. I don't have Rorch with me, that's not the one I used. Oh, I thought you were referring to the tube steel.

MR. BRAMMER: No, we're talking about the pipe, the pipe wall stiffness.

MR. IOTTI: No wonder I couldn't connect the two.

MR. FINNERAN: I guess Johns' question goes to justification of the use of this formula for R/T of 6.06.

MR. CHEN: Well I'll tell you where we were, in looking at the Rorch formula's we looked at equation nine.

MR. BRAMMER: Yeah, that's for line contact, right.

MR. CHEN: Well no, the only difference I think in equation eight and equation nine, is equation nine is for closed in, you can see in the case of equation nine here, it's for the closed in pipe, of length L, and for equation eight it's open ended, length L.

MR. BRAMMER: As I recall for equation eight, the L/R ratio was equal to 8, 18 or greater, equation nine should give you the same results, according to the footnote on the equation here. But again it's for thin walled vessels.

MR. CHEN: Well I don't see the same qualifer on equation nine that, that he has put on equation eight, as far as the R/T ration.

MR. IOTTI: It's a fact it goes through several

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R/T's. Down to as low as 15. Well we'll get you the answer on that.

MR. FINNERAN: Well our answer, actually the reason for choosing eight was that it looked like it more closely approximated piping being open ended then it did just closed ended, so that's why we chose the equation.

MR. BRAMMER: Well I don't argue with the reason why you chose that if it's applicable to this thick a walled vessel, I'd like to have some justification for using it.

MR. FINNERAN: Well I'll get back to you because I know we did something we'll just have to dig it up.

MR. BRAMMER: Okay, on page 5 this same thing applies again,

MR. IOTTI: Yeah, it would apply throughout. That's the one that we couldn't find the one for a thick walled, the one for localized stresses, it may have been that we ended up with all of it for consistency, rather then mix and match.

MR. BRAMMER: Okay, then on page 7, maybe this has something to do with the Gibbson Hill allowable that you were talking about earlier, for equation nine, it looked to me like you could of used a higher (inaudible) at 1.8 sh, but not greater then 1.5sy.

MR. FINNERAN: Well that's why we said, in the affidavit that we used the conservative Gibbson Hill allowables

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for this approach.

MR. BRAMMER: But that comes out of the Gibbson Hill?

MR. FINNERAN: Yes, it comes out of the Gibbson Hill procedure, and they've taken a conservative approach to establishing allowables for those stresses.

We need to give you a copy of this Gibbson Hill appraoch.

MR. BRAMMER: Is that the, the same would be true for equation nine E on page 8.

MR, FINNERAN: Yes the same thing.

MR. BRAMMER: Okay, I have a question here the thermal stresses is determined with a calculated P load of 31814 pounds which you came up with, if you use that in all these other values you put in, it doesn't meet equation 10, it meets equation eleven and even your reduced value of equation eleven, but the termal stresses don't meet equation 10.

MR. FINNERAN: Well as I explained to you we used the Gibbson Hill approach, it doesn't have an equation 10 in their approach. I'm not sure if it explains why, but ...

MR. BRAMMER: Well according to the code it's okay to go with equation eleven, but youhave to meet equation 10 too, I mean you have to meet the requirements for equation 10.

MR, FINNERAN: I guess I don't have an answer for you on that one.

MR. TERAO: Is this calss two and three?

MR. FINNERAN: Yes.

MR. TERAO: I don't believe the code says you have to meet equation ten.

MR. IOTTI: I thought you could go to equation 11, could I see the code?

MR. FINNERAN: Well that would explain why there's no equation 10 in the Gibbson Hill. I think it's the other way around, I think it's the other way around from what you said. Maybe we should go off the record here.

MR. BRAMMER: I guess we can say that that was answered satisfactorily.

MR. BURWELL: During the record we pulled out the code and determined that the criteria was that you need to meet either equation ten or eleven and that has sinced been confirmed by Mr. Brammer.

MR. BRAMMER: Alright attachment B is next, and the only comment I have on that is that the temperture distribution is different then what was assumed here, the loads could change, but even I, ran a quick calculation and used seventy degrees for the temperture and the load only went up to 852 so, there's no problem.

Okay, then attachment C, the formula in Rorch by the way it's written out is for a small concentrated load, this doesn't appear to be to me a small concentrated load.

the load is spread over a considerable area. I'm not sure how applicable that formula is again, due to that fact.

MR. FINNERAN: Well I hope you can appreciate the effort of trying to access these things without going to a full blown finite element analysis of the thing.

MR. BRAMMER: Sure I understand that, but that doesn't seem that that is really applicable to this situation. You know, you've got two eight inch pipes pushing down on it.

MR. IOTTI: Well let's include that as part of the laundry list that we'll provide explanations why we think it's a reasonable approximation, even though it's not precise, the approach.

He has questioned the applicability of the formula used to determine the stiffness of the pipe which was distracted from table 31, page 8 of Oroch and Young, because that formula is apparently, is for a localized load, whereas, one could question whether the load, the local loads here transmitted to the pipe in question is, in fact, localized since it's transmitted by a set of four eight inch pipes, and we are going to provide an explanation as to why we feel it's still appropriate to use that formula for the purposes of this calculation.

MR. BRAMMER: Okay, then another question I had on that when you were applying this stiffness formu in

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determining your wall thickness you included the wall thickness of the pipe, plus the thickness of the doubler or saddle underneath of that.

MR. FINNERAN: Yep.

MR. BRAMMER: I realize this sketch is not to scale but it looks like the doubler is very small, it doesn't extend much beyond the pipe that's loading it.

MR. FINNERAN: Well on calculating this particular stiffness, wouldn't that be a conservative approach.

MR. BRAMMER: It would be a conservative approach in calculating the stiffness but when you come to determine the local stresses, if you use that double thickness it would be very un nservative.

MR. FINNERAN: Well would it be unconservative or would it not? I think the basis for the equation that we used to determine the localized stress came from the Beam and Elastic foundation theory, and I think the points, of high concentrated stress in this case would exsist in the area of the pipe where it was reinforced with that(inaudible).

MR. BRAMMER: Well what you used is the equations from this (inaudible) 107.

MR. FINNERAN: Yeah, and that's, those equations were developed from Elastic Beam, I mean Beam and Elastic Foundation Theory.

MR. BRAMMER: Okay, but it seems to me that this

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doubler, if this sketch is anywhere near being close is part of the trunyan(ph) it's not part of the skin.

MR. FINNERAN: It's welded to the pipe.

MR. BRAMMER: Well yeah, but it's very local, you may get your maximum stress in the pipe just at the edge of the doubler and the doubler and ends.

MR. FINNERAN: In our looking at the theory of Elastic, Beam and Elastic foundation theory, the, the stresses peaked right under the externally applied load, and then drop off very quickly, away from that point, or we wouldn't agree that the maximum stress point would be out there.

MR. BRAMMER: Well I guess I'd like to see how big this doubler is, how far it extends. You're right the stresses do drop off very fast, but again this looks to me like it's part of the loading mechanism and not the resisting mechanism.

MR. FINNERAN: Alright, by the fact that they're welded together you can't ignore that fact.

MR. BRAMMER: Well, right, right. But if the doubler is part of the load that's applying the pipe then the peak stresses would be right at the edges of the doubler.

MR. IOTTI: Yeah, as far as the Beam and Elastic foundation, that would also be those results.

MR. FINNERAN: You'd like to see physically how the pad is attached.

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MR. BRAMMER: Yeah, some sort of justification for doing it that way. Okay, on page 4, I think maybe you were a little unconservative, the calculations are not shown for the lower obstructogen(ph) but it appears that the effect on the stiffness of the two 23 inch long, 18 inch diameter pipe cantilevel beam were neglected. That just would lower your loads quite a bit really.

MR. FINNERAN: Well that would be conservative. You said unconservative.

MR. BRAMMER: No, I meant to say conservative. Yeah it dropped down 30% or better. Okay, well then on page 5 I had the comment about, again the doubler for saddle thickness was included in determining the local pipe stresses and I guess you're going to give me something to justify that.

Again, okay, I guess that takes care of that one. Okay, now on attachment D, again I have the same comment as I had on, same formula right, and saddle thickness is again included in the wall thickness. That's about it on that one.

MR. IOTTI: Well I just wish, maybe can we go off the record for a second.

MR. BRAMMER: Oh, on page 7, just, this is on D still I was wondering if the, are the local stresses due to dead weight, siesmic thermal included in equation 11 stresses

stress where you come up with 30,502 PSI?

MR. FINNERAN: Yes, we've taken those stresses and added them in. If you read this, paper by Gibson Hill, we use these modified equations and those are included in those equations.

MR. BRAMMER: Okay, Attachment E, again this one does not, it's again a question with the stiffness calculation of the pipe being used. Two equations, it doesn't fall within the thin walled criteria and the load is applied over a, on a six inch pipe with a five inch pipe which I wouldn't consider a concentrated load at al_, that pretty well covers the pipe. So, it doesn't look like this formula applies again, for two reasons, and again the saddle thickness was included in the overall thickness.

MR. FINNERAN: I think this one does meet the R/T greater then ten criteria.

MR. BRAMMER: Well you say on page 2, it's 5.13. That's when you include the saddle thickness and the...

MR. FINNERAN: Oh, I'm sorry. That's corrent. That should be answered when we answer the other one.

MR. BRAMMER: Then you go to page four where you check the local stresses with this WRC-107, these peramiters don't fall within the limits of WRC-107. I'm not sure that you're conservative or unconservative when you come up with stresses that are out of the range of the curves.

MR. FINNERAN: Would you like us to address that?

MR. BRAMMER: Yes. Okay, on page 6, you have FB over capital FB 2725, and that equation there, isn't this 2725 that's only the part of the load that from the stiffness calculations, shouldn't there be other loads included in that, in a normal ...?

MR. IOTTI: Could you repeat your question again please, I'm confused, on page 6.

MR. BRAMMER: On page 6, the first equation there, or the second one down, you've got a small fb over a capital FB, equals 2725. That 2725 load is the load from the stiffness, or stress from the stiffness calculations. It seems that there should be other stresses or loads included in that.

MR. FINNERAN: This old calculation is under an area that, look back on page 5, it says check extra members stress. Okay? So, on page 5 in the middle of the page, right above A, it says check extra member stress. So what we're calculating is the additional stress here due to these thermally generated forces, and quite possibly it could be a full explanation of the other stresses from the other loads should of been included there, but this thing says, it's characterizing that as only the extra stresses.

MR, IOTTI: Well then it's no problem for us to go

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back and give you the remaining stresses.

MR. BRAMMER: Then I guess by the same token where you have the double asterick down there where it says extra loading exsisting analysis you have 4,355 pounds, I was curious as to where that came from because your calculated load was 5,908 pounds.

MR. FINNERAN: Well I don't know the orientation of this structure from our attachment here.

MR. BRAMMER: Well that's it I really couldn't follow it from about page 5 on, I didn't, I just didn't know exactly what you were doing, but the loads didn't seem consistent.

I guess what I'd like is a little more explanation from what is happening from page 5 to page 6.

MR. FINNERAN: Okay, it looks like the exsisting analysis on this support, the loadings on it FZ, it looks like that is a load in the same direction as the 5,980 pounds, calculated, right?

MR. BRAMMER: This 4,355 pounds you mean?

MR, FINNERAN: Right. So if that's the case, and I believe it is, if you got an interaction back on page 6, for 5,000 pounds of .108, interaction for 4,000 is going to be something less then that, and the total of the two would still be much less then 1.

MR. BRAMMER: Okay, then when you go to this sketch

on page 7, I can't hardly check any of those loads.

MR. FINNERAN: Those are the loadings on the base plates, on the base plates. Right?

MR. BRAMMER: Right, but they should include, somewhere they should include the sum of this 5908 and 4355 right?

MR. FINNERAN: Yeah, I guess we owe you an explanation or a clarification.

MR. BRAMMER: There's nothing, F, X, Y or Z doesn't add up to 9,000 there.

MR. FINNERAN: Yeah.

MR. IOTTI: Let's take a look there, yeah that's a neat trick, how did we get 5149 plus 4355 equals 6104?

MR. FINNERAN: That's 1749

MR. IOTTI: 1749.

MR. FINNERAN: Yeah.

MR. IOTTI: Those are the loads that you see reflected in page 7. The ones on the bottom of page 6 which are the total loadings.

MR. FINNERAN: Well the key here is in the interpretation of the note on the bottom of page 6, and (inaudible) explained it to me at ome time, and it's been so long ago that I've forgotten it, so we're just going to have to get an explanation.

R. BRAMMER: Yeah, maybe you could try to clarify

this whole section through page 10 because I couldn't really follow it.

MR. FINNERAN: Alright.

MR. BRAMMER: Okay, then on page 11 where you solve for F8 Prime stress there 4240 divided by 7.95 which is the area of a five inch support pipe you have a load that you calculated that's 5,900 pounds plus this exsisting load, doesn't seem like everything is included in that baby either. By the same token I'm not sure where some of moments came from for the equation just below it, but, if part of the load was left out of that one, it, there may be some of it left out of the (inaudible) calculation.

MR. IOTTI: We'll just go back and provide the explanation for this entire calculation.

MR. FINNERAN: Yeah, that's what my note says to do.

MR. BRAMMER: Okay, we'll that's really all I have.

MR. BURWELL: Would you all like to take a break at this point in time, or keep going.

MR. BRAMMER: Do you have anything to add Paul?

MR. BURWELL: Did I head a yes or a no? Yes Okay, off the record, break.

(Recessed)

MR. BURWELL: Do we need to sum up on the last point?

MR. IOTTI: If you want I have a list of action

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items that I jotted down this information that we need to provide to the staff and their consultant for the local displacement and stresses.

The first action item is on the affidavit, where we are making a statement that on the basis of calculations that we have done it should be obvious that the loads and the stresses are conservative with respect to any of the other similar type frame supports and we need to further justify why the stresses would be greater. I think we've concurred that the loads are, but the stresses it's not immediately obvious.

The second action item I have is that we have to provide the calculations, which show that the linear distribution of temperture in the tube steel across and along the tube, is a reasonably appropriate approximation to make from the standpoint of conservatism.

Then I have several action items which are related to the same topic in reality and that is the use of a formulas from Rorch and Young, which is really a formula for pipes which have ratios of, radius to thickness is excess of 10 or 15, whatever, I don't remember which, it's ten, and some of the example problems that were solved those formulas have been applied for pipes which have smaller R over T ratios and we have to justify why it is still appropriate to use those formulas.

We are to provide a copy of the Gibson Hill approach on integral welded attachments for ASME code 2 and 3 pipes and that document is called the verification procedure, making a copy and providing it.

In addition there's another general comment which is similar in nature to the use of the Rorch formula is that in some instances when calculating the stresses on the pipe, the pad of the particular support was added as part of the thickness, and the stress was calculated through that thickness and we need to justify why that might be appropriate.

So those comments apply to attachment A, B, D and E, on attachment E, in addition there is a question that the calculated stresses fall outside the curves of the welding research council bulletin 107 and whether it's appropriate to use that particular document for the conclusions that we reach, and we have to justify that.

Finally we are to provide a, I don't want to call it an expanded attachement, we need to provide an explanation in the calculation that comprises attachement E to permit the reviewer to follow through the calculations as to where the numbers come from and so forth.

Those are the only action items that I have on the local displacement of stress, have I left any out?

MR. BRAMMER: No that agrees with my list.

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MR. BURWELL: With that I would propose that we go on to the motion for summary disposition of case allegation relating to differential displacement of large frame wall to wall, and floor to ceiling pipe supports. This motion was filed on June 22nd, 1984. I would hope that we are able to complete that today. In any event, let's give it a shot.

END OF TAPE.

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CERTIFICATE OF PROCEEDINGS

This is to certify that the attached proceedings before the NRC COMMISSION

In the matter of:

QUESTIONS ON SUMMARY DISPOSITIONS FILED BY TEXAS UTILITIES ON COMMANCHE PEAK

Date of Proceeding: August 8, 1984

Place of Proceeding: Bethesda, Maryland

were held as herein appears, and that this is the original transcript for the file of the Commission.

Joe Newman
Official Reporter - Typed

[signature]
Official Reporter - Signature

ORIGINAL

UNITED STATES OF AMERICA

NUCLEAR REGULATORY COMMISSION

In the Matter of:

QUESTIONS ON SUMMARY DISPOSITIONS
FILED BY TEXAS UTILITIES ON
COMMANCHE PEAK

Location: Bethesda, MD Pages: 146-209

Date: August 8, 1984

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UNITED STATES OF AMERICA

NUCLEAR REGULATORY COMMISSION

- -x

In the matter of: TEXAS UTILITIES GENERATING COMPANY, et al. (Comanche Peak Steam Electric Station, Units 1 and 2)	Docket Nos. 50-445 50-446

- -x

7735 Old Georgetown Rd.
Room 1713
Bethesda, Maryland

Wednesday, August 8, 1984

Hearing in the above-entitled matter convened at 9:20 a.m.

APPEARANCES:

On behalf of the Applicants:

PAUL CHEN
WILLIAM A. HORIN, ESQ.
ROBERT IOTTI
JOHN FINNERAN
DAVID WADE

On behalf of the NRC Staff:

J. BRAMMER
HANK FLECK
DAVID TERAO
JOHN FAIR
SPOTSWOOD BURWELL

MR. BURWELL: John, is this yours?

MR. BRAMMER: I'm going to defer to Dr. Chen on this one. I think he has more background on it than I do.

DR. CHEN: From a technical point of view, in addition to what's already on the record, am I correct, but the only thing that's added in here is the conclusion regarding the adequacy of the..what's this? ..15 supports that you looked at.

MR. FINNERAN: I would say is what the affidavit does is pretty much like we characterized it. It's the expansion of what's in the record, to include all of the wall to wall, floor to ceiling supports for all three pipe support organizations.

DR. CHEN: I think up to this point, floor to ceilings were addressed in the hearings. The ones in the service order tunnel.

MR. FINNERAN: I think we described in here which ones were previously addressed.

DR. CHEN: And on page 7 I think, of the motion, it says subsequently applicants have reviewed all unit one and common safety related piping supports and determined that there are 26 wall to wall and floor to ceiling supports.

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Fou- have slip joints, no, seven have slip

joints, and four have small spands. Okay. And fifteen have, the remaining have been evaluated and found to be acceptable, considering the potential for differential size and displacement.

So, the only thing new, here is, other than what's in the record already, is the fact that you looked at fifteen others and found them to be satisfactory, is that correct?

MR. FINNERAN: Yes.

DR. CHEN: There's also one other item and this has to do with the wall of the slab, wall to wall to floor. The only request that I would have here would be for you to characterize how close you were to the other allowables, I guess for these fifteen. Do you know how close you were to the allowables?

What is acceptable, what is the maximum? For the worst case, how close were you to the allowable? Were you 50% or what, do you have that number? Or can you get that information?

MR. FINNERAN: Yeah, we have the number.

DR. CHEN: I mean, can you get that information for us?

MR. FINNERAN: Uh-huh.

MR. IOTTI: This is for only, Dr. Chen, is this only for the fifteen, or also for the wall to ceiling,

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and ceiling to floor?

DR. CHEN: Just the fifteen. Yeah, I think the others don't need..they have small spans and they have slip joints, so you don't need to do anything with those. I think these fifteen with the bounding, and what I'm looking for is what the maximum is of these fifteen.

MR. FINNERAN: Okay. I guess my impression is, why, since this is an assessment of all the ones we have.

DR. CHEN: Well, I just wanted to get a feel for how close you were to the allowables, that's all. That one would bound them all, is that correct? The worst one here, would be the bounding case for all of them?

MR. FINNERAN: That's correct, that's all you got.

DR. CHEN: That's all I want. You know, that's just a, that's the only information I need. I want the worst one.

Okay, on page 9, also the motion, you talk about differential seismic deflections for wall to slab supports. It's mentioned there that you looked at three of them, and they are mentioned there. They're identified on page 8, of the affidavit. The question

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that I have there is, can you give me some idea of how many of these kind of supports you've got in the plant?

MR. FINNERAN: Well, there's more than these three, and that's about the only exact answer that I can give you. We have never attempted to count them, nor do I really have any kind of feel for how many there are.

We think that these three supports that we analyzed are very representative of the, any of the other supports that would be from wall to ceiling or floor to wall.

DR. CHEN: Can you give me some idea of how much along the wall or, do you have those drawings here?

MR. FINNERAN: I think I have those drawings here, but I'm not, it's not possible to bring all the back-up material.

DR. CHEN: CC107, SWN?

MR. FINNERAN: Yes, I have the 132703 one here, and it's, it extends, this is the drawing here, and this one extends from the wall to about seven feet out in the span of the slab. I don't know how far down that goes. That's 804 to 806, that's a little over two feet, from there to there.

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DR. CHEN: Okay.

MR. FINNERAN: For that particular one, that span out into that floor slab is..I think that whole tunnel is probably maybe about 12 feet wide or something like that.

DR. CHEN: Okay. Is that one just down from two of the four that we talked about, in the hearings, John?

MR. FINNERAN: Pardon?

DR. CHEN: Is that one in the same tunnel that..

MR. FINNERAN: This is one of the four we talked about, where we cut off the column legs. That's the 132703 one.

DR. CHEN: Would it be easier, John, if I said that you would just provide us with those?

MR. FINNERAN: Yes. We can do that. I was hoping to find those here.

MR. IOTTI: Do you want the drawings for these three supports?

DR. CHEN: Right.

MR. FINNERAN: I don't have them.

MR. BURWELL: Do the three supports cover a different pipe sizes or, did you pick them because they are large pipe sizes, or, you said they were

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representative of wha'.'s in the plan. I just wanted, was wondering...

MR. FINNERAN: Well, the service water one is a gain frame, and typically, several of these types supports that do go from wall to ceiling, because of picking up more than one pipe, they have a tendency to do that sometimes.

DR. CHEN: What size pipes are those?

MR. FINNERAN: I think those are ten inch pipes..service water pipes on there. Yeah, those are ten inch pipes. And that particular one there are four of them. Now, I've forgotten your question.

DR. CHEN: What do they spend on pipe sizes, different pipe sizes..

MR. BURWELL: I was looking for why you think the three selected are representative of worst case things, and I think one of the points that you brought out..that gang hangers would likely be more of a, would demonstrate larger displacements that would say, single hang..single to double hangers.

MR FINNERAN: Well, the gang hanger would tend to have a longer span, so it might tend to have to attach from a wall to a ceiling. As far as the representative nature of the rest of them, I would say that the dimensions of these, and that seems to be

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the characteristics that should be looked at, represent the deminsions of the other supports that are of this nature. The spans, how far out in the slab, or how far down the wall they extend.

DR. CHEN: One of the analysis that you submitted, way back when, consisted of a prop cantilever. There was a structural steel member, then cantilevered from the wall, like there was a prop from the floor up to that, to that steel member. Is that correct?

MR. FINNERAN: Well, you would have to get more specific than that, I'm afraid. These service water frames, they extended from the wall, and they also had a column that came from the floor, a prop that also went all the way up to the ceiling. And what we ended up doing with those, and you're well aware, we cut the columns off, so they wound up then, being a ceiling to wall type of support, rather than a floor to ceiling. That might be the ones that you're thinking about.

DR. CHEN: No, what I'm thinking of is, you have the wall here, and you had a cantilever sticking out..and there was a prop, here, that went to the floor.

MR. FINNERAN: Oh, that. That was one of the moment restraints, that was one that gives and held, moment restraints.

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There was a pretty detailed analysis of that.

DR. CHEN: The only remaining question goes towards spots, was one of the spots raised earlier, but are representative of these three, really, we need to address that.

MR. FINNERAN: Well, we can provide some more information along that line, when we provide you the drawings.

DR. CHEN: I have no other questions.

MR. BURWELL: Okay. We have a request outstanding on that, I guess, for the drawings, and some explanation.

MR. IOTTI: There is a total of three that I have. If we could keep going, and maybe we should stop. The first one I have is for the fifteen wall to wall and floor to ceiling. Which one is closest to allowables, and how close is it?

MR. BURWELL: That's correct.

MR. IOTTI: And the second one is to provide the drawings for these three supports, and the third one is justify how representative they are.

MR. BURWELL: Yes.

MR. IOTTI: That's all I've got.

MR BURWELL: With that, it is now..okay, I have a vote to go, continuous stability I think, and

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with that I'll turn the questions over to Block.

UNIDENTIFIED SPEAKER: Well, this might take awhile, so, it's not necessary that we cover it all tonight, today.

MR. TERAO: This is on stability, and I guess before we get into the affidavit, in reading your response on stability, I'm not sure exactly what you're trying to say, but, perhaps, as an opening question, I should just ask, is there a current problem, now, with instability of any pipe supports?

MR. FINNERAN: No, I don't think there's any pipe supports, that there is a problem with stability.

MR. TERAO: Okay. Has there ever been a problem with instability?

MR. FINNERAN: Potential instability. The box frames that we talked about. There were modifications made to them to improve the potential instability, aspects of them. And there were modifications made to the..some of the main steam supports, which had gaps in them.

MR. TERAO: Well, I guess my question is more to do with the way it's worded in the write up. I don't think you ever stated, even that there was a problem with instability. I think what you've said

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is that you've improved stability on these supports, but I'm not sure if you ever, are saying that ever had a problem with stability, instability, or whether or not you just felt that the stability could be improved, but you never did have a ·oblem.

MR. FINNERAN: Well, I think the bottom line to our testimony on these supports, would have been that we believe that these supports would have behaved in a stable manner, and that they would have served their function. They would have functioned, they would have carried the load that they were going to carry. But, there was a question of those stability. And, in leiu of trying to prove that they would have behaved in a stable manner is it is more prudent for us to modify the ones for which there was a question of stability.

MR. TERAO: All right. From the very start of this issue, on instability or stability, I've always had a problem with trying to define exactly what Case meant by instability,and what the applicant has meant by instability, what SIGNA has meant by instability, and even what the Sit and the NSE stamp has meant by instability.

It's always been a fundamental problem, and until we get this thing, until we get the definition

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well defined, you see the problem is that if we can't agree on a definition, then we can't agree that there is ever a problem. So, maybe we should spend a little bit of time trying to define what instability is. Let me just point out, that when I first heard about the problem of instability, the two classical definitions of instability were, of course, buckling and the second is the inverted pendulum type of instability where the classical definition of that is of a system is displaced slightly from it's equilibrium position, does it tend to return to it's original position, or does it tend to displace further, when the disturbance is removed.

If it tends to return to it's original position, the system is stable,and if it tends to displace further, if the system is unstable.

Now, but those two definitions, when we looked at the Walsh/Doyle allogations on instability, it didn't appear that that was what Walsh and Doyle were talking about in their case of instability.

The way I understand their definition of instability it really..well, it seems to have come about from modeling of using the STRUDL program. And I believe Mr. Walsh defines stability, in one sense, as equivalent to pushing a chain. And Mr. Doyle has

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always explained it in terms of three bi-linkages. But, when one thinks in terms of the STRUDL modeling, I believe Mr. Doyle made some reference to when they modeled some these supports in STRUDL, they ended up with very large rotations and displacement. In other words, it really could not be well modeled into STRUDL.

Now, I'm not saying that that means the support is necessarily unstable, but, it does point out either that there is a modeling problem, or that something is not correct with that support. That it cannot be properly modeled using STRUDL analysis.

So, with INSIGNA, has also defined instability in terms of buckling and also in the sense of plastic hinge, in another words, the total collapse of a system. And I still don' t believe that that is what Walsh and Doyle were talking about.

Maybe at this point I should ask you for a clarification of your definition of instability, and then I could probably put in what I think is, what we mean by instability, then we could proceed from there.

MR. FINNERAN: Well, I think that, I think I would have, as far as a definition of instability, I would have to agree with what Dr. Bjorkman said in the hearings, that there are different kinds of instability and he characterized it, I believe, pretty much along

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the lines of structural instability, and I think that that is the collapse of a column. And, lets say, in compression, when you exceed the collapse load of the column. And rigid body instability. Which would be akin to the pushing the chain or Mr. Doyle's three bar linkages. That type of instability would be where the support may not be able to take any load at all.

I think Walsh and Doyle, and maybe it's easier to...my impression of what they're talking about when they say instability, are the supports that we've addressed here, the box frame on a single strut, where the question of whether the box frame will rotate around the pipe, and thus you have the three bar linkage he's talking about. The two pins of the strut and then let's say a third pin that represents the cability of the box frames that rotate around the pipe.

That, if that support weren't perfectly aligned, and if there weren't some mechanism to preclude the rotation of the frame, around the pipe, then that's a rigid body type of instability. That support potentially can carry no load in compression. The same thing with a strut, a backing plate, and let's say, a u bolt with a gap, i.e. the mainsteam support that Mr. Doyle had in his deposition. The same type of phenomenon, rigid body type of instability. Then I

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think the other type of instability that Walsh and Doyle were talking about, related to what you said about the STRUDL modeling, is that, actually, you could take a strut and a clamp, and model that thing in the STRUDL, and it's going to be unstable as far as STRUDL is concerned, because it's looking at the load that you're applying on it, it's just an externally applied load. From some source.

And it's not taking into account the restraining effect of the pipe. The support is actually attached to a pipe, that will not allow the support to swing out of plane, and become unstable, as STRUDL.. there's nothing in STRUDL to model that restraining effect of the pipe.

So, I think that's the instability as you had related to STRUDL modeling. That they're talking about. I really think that the real stibility issue that they were concerned with is the rigid body type of the three pin or the three bar linkage, as they call it, related to the single strut or snugger (ph) supports, and the piping plate and the u bolt that had a thermal cap, and the same thing, single strut snugger support that had a box frame around the pipe.

MR. TERAO: Going back to the STRUDL modeling, when you said that the model in a strut with a clamped into

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STRUDL, that would also appear to be unstable for the STRUDL, unless you could some how account for the piping modeled in there with the stiffness.

But, with these supports that were considered to have potential instability, what you're saying, then, is that if that support were allowed to rotate around the pipe, in other words, if it has no clamping action, then that is what you defined to be rigid body instability? Okay, I could agree to that definition. In fact, when I look at the different so-called classical definition of a stability, there is the, if you look at a ball, on certain elevations, if you have a ball in a valley, so to speak, that's a stable situation. If you have a ball on a peak, that's an unstable situation, and if you have a ball on a plateau, that's a neutral stability.

All right, now, when you refer to a rotation of the clamp around the pipe, I agree that it's not unstable from, in other words, it's not equivalent to the ball on the peak, but it is equivalent to a ball on a plateau. Hope this makes any sense. In other words, it's not really unstable, but, on the other hand, it's not stable, either.

MR. FINNERAN: Its not carrying loads. It dosen't carry loads.

MR. TERAO: So, the concern, and this is why

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I would agree with you, the concern is where you could have a support or a clamp around the pipe, that it's not performing a clamping function, and therefore, the clamp can move or rotate to an undesireable position or to an unanalyzed position.

MR. FINNERAN: Okay.

MR. TERAO: Now, that is the way that I'm going to be defining instability, in the sense that instability is the potential for support to shift to an undesirable position, which has not been analyzed.

Now, that dosen't mean that it's necessarily a safety concern, and it dosen't necessarily mean that it's unstable in a classical sense. But, it only means that it is not controlled.

In other words, it would be equivalent to a clamp that has loose bolts, and in that sense, we would not accept the clamp with loose bolts, or you wouldn't accept a clamp that dosen't completely clamp around the pipe, and that could slide along the pipe. Okay.

So, if we can agree to that definition of instability, then I think that we can proceed with something else.

MR. FINNERAN: I would say one thing. Our primary concern is to address the allogations of

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Mr. Walsh and Mr. Doyle in this area, and I also, I thought that we were trying to address some of this confusion about stability and the different kinds. In fact, Chairman Block specifically asked that we do that.

If you look at the Doyle allogations on, back in his deposition, on stability, the only real issue on stability, that he is addressing, is the rigid body mode. If you go back and look at, I think it's Section 4, of his attachments, where he's given examples of the stability issue that he's talking about.

And he's talking about the three bar linkage in all cases, either with box frames or ..

MR. IOTTI: With one exception. He kept bringing up, which is purely imaginary, that of the u bolt, with no pipe through it, where, if you let it go, it will fall. He kept bringing it up and we kept asking, you know, what's the purpose of that support if there is nothing to support? But, that one he kept bringing up, so I'm just pointing out that that was also his idea of instability.

MR. TERAO: But, I think, my problem with trying to address this particular summary of his position, was trying to decide if we are going to be addressing stability strictly from the Walsh, the way Walsh and Doyle

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have tried to characterize it, or perhaps the way other people have tried to characterize it, or whether or not there is any question and a potential safety concern from, not necessarily the allogations themselves, but from the design that was being addressed through the allogation.

In the sense that, Walsh and Doyle may not have clearly established on the record, exactly what their concern is with instability, but they may have had a valid concern with the design, itself, and whether or not that design can be considered acceptable.

And that is the way that I would like to at least, address this issue from a staff standpoint.

MR. IOTTI: We're not entirely sure what you're saying, David, but, the affidavit addresses two parts. The front part is really intended for Judge Block. He specifically requested applicants to provide a definition of the various types of stability. Whether in fact, all of them would be addressed or not, as to the latter part of the affidavit, specifically addresses the stability issues raised by Case.

So, from the standpoint of Case, we're addressing the stability issues that they have raised. And Judge Block wants an education on stability. And then, of course, we have to comply. But, a lot of

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this information is provided in direct reply to his questions, that were raised on, both on that telecon and also on these subsequent memo and order. So, we couldn't very well ignore those. I always agree with what you're saying, either...

MR. TERAO: But, I also recognize what the Board is asking the staff to look at it, too. So, with that, maybe we should get into the affidavit, itself. I think comment, and answer a few questions on the affidavit, itself.

On page 7, of your affidavit, the question was what is standard industry practice regarding consideration of stability of piping systems, and one part of your answer states that it's not necessary, apparently, to consider stability, because during the normal design process, the piping designers achieve a system which will stay within specified deflection, and thus, will be incapable of a type of instability illustrated in figure 1C.

I guess I take exception to that statement. The deflection limits, themselves, do not necessarily assure piping system stability. First of all, deflection limits are intended for the pipe supports, to keep the supported component within, it's intended to assure the operability and functionality of the supported

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component. I'm not sure how you concluded that the deflection limit can assure stability of the piping system. So, perhaps you might want to address that a little bit more.

MR. FINNERAN: Well, the only thing that is meant by this particular statement is, that we talked about earlier, the fact that when you have a support, such as is shown, in figure 1C, say of a strut with a clamp, but that support, in and of itself, isn't unstable support, in and of itself. But, you attach with a snort (ph) clamp, to a piping system in which there is another support, in the other direction down stream, that can prevent the side sway that would make it an unstable support..

MR. IOTTI: Yeah, but the answer stands as it is. Everywhere in the piping system, within prescribe deflection limits, and if you can assure function ability by the system, by your definition, you're not unstable. You cannot be. Now, implicit in that is that the designer has recognized that the support behaves as he has modeled it. That's where the question of instability may creep in. But, if you, in fact, have modeled it properly, and if the piping system, and, of course, the support system is everywhere within the specified deflection limit, how

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can it be unstable?

MR. TERAO: Well, that's very simple. If you had a system with a long riser, a tall riser, and if you only supported it from the bottom, and the riser is not considered stable, unless, of course, as John has pointed out, added supports, horizontal supports at the top.

MR. IOTTI: But, if it weren't stable, you would calculate that it deflected out past the prescribed deflection limits. If you stay within the prescribed deflection limits, either the analysis has computed that it can stay within the deflection limits without the additional supports, or he will add the supports.

MR. TERAO: No, it's unstable from, in a horizontal sense. If it moves sideways.

MR. IOTTI: But, then he would violate the limit sideways.

MR. TERAO: But it's not shown in the analysis.

MR. IOTTI: How do you know that? The deflection is shown in all directions of the analysis, and the limits are prescribed in all directions. I think that we're getting hung up on semantics. Where the problem might arise, is, if in fact, it's mismodeled, and the applicant, the analysis has believed he's modeled it correctly, but, in reality the system behaves differently

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than it has been modeled. For instance, a typical example is these box supports. When the analysis models it as if the support were not rotating, he in fact, believes that that support will keep that point of the pipe fixed in space, or within a small deflection. In reality, the support by rotating around the pipe, permits the displacement to be much higher than what the analysis believes it to be.

So, what the model would show you, the analytical model will show you is that you have a stable system, but the real model may show instability at that point, in that sense.

MR. TERAO: Well, I guess what I'm referring to is some of the fundamental rules of hanging a piping system. In that sense, I'm not saying it's unstable, but, I'm saying that you may not have a totally stable system if you support a riser, only from the bottom. I agree it's common sense...

MR. IOTTI: Not many people would do so. On the other hand, there maybe instances where you want to do that, in which case you would do a very accurate analysis to make sure that if you only support it at the bottom, you don't have a bucking problem, nor do you have a matter of displacement problems.

MR. TERAO: But, that is the type of system

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stability that was referred to in the SIT report, that should be reviewed by piping designers.

MR. IOTTI: And it is, it is.

MR. TERAO: Which brings me to the next comment. I believe, on page 7..

MR. IOTTI: Well, I guess, before we go there, on this page 7, you said that the stability of piping is not explicitly, but that dosen't mean that implicity is not addressed. That, you know, one dosen't explicitly go through, and on a piece of paper say, you know, this portion of the piping system is stable and that portion of the piping system is stable, but the piping analysis, when he performs all of his analysis, that's very far that this piping system is stable, that everywhere deflections are within what he wants to have. So, implicitly he does the stability review.

MR. TERAO: Yes, I will agree that the piping system can only be evaluated for stability by looking at the configuration and the locations of the supports. But, you have to look at the system as a whole to determine the piping system stability. But, I don't think that is really what the issue is, here, with the unstable supports. I don't believe that was what the issue was.

I agree that is what the SIT addressed, and I

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agree that when you look at it from that perspective, that there is no problem, and there shouldn't be any guidelines, because that is common sense ...

MR. FINNERAN: Judge Block asked us to address all the stability issues that have been talked about, and this was one of them. I think he had alittle point of confusion in his mind, about this system stability thing, and relating it to the (inaudible) so we tried to put it in perspective.

MR. IOTTI: We actually restated, if you will read the question, here. It says, what is the standard and practice regarding consideration of (inaudible) piping systems, and maybe we should have, as a whole, maybe we should have added that. So, that answer strictly pertains to the piping system as a whole, not to stability of supports, in itself. And, it was again, for the purpose that John just stated, to clarify whatever misconception or lingering doubts that Mr. Block may have had.

MR. TERAO: It's just that maybe the last statement, you say in summary, the piping analsys assures stability of the piping system by limiting deflection, which negates any need to assist stability separately, and I took issue to hat.

MR. IOTTI: I see. The wording is ..okay.

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MR. TERAO: I'm saying that he should look at the configuration, and not only rely on his analysis.

MR. BURWELL: Could we go off the record just a moment?

(Off the Record)

MR. BURWELL: Back on the record. Mr. Finneran was called to the phone so we went off the record and took a brief recess.

There was some discussion during that period, and I guess, would one of you gentlemen restate the gist of that, please?

MR. IOTTI: I don't know whether David or myself, should restate it. My understanding is that the contention, here, is on the last sentence of the answer that appears on page 7, of the affidavit, where we state that in summary, the piping analsys assures the stability of the piping system by limited deflection, which negates any need to assess stability separately.

That could be misconstrued to mean that the piping analysis only looks at deflection. That is not the case. He also looks at configuration. But, typically, it's the combination of assuring that a configuration is proper and the deflection limits are self imposed by the analysist themselves, are met everywhere. Aed if the modeling and the actualy system are one and the

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same, that that insures that the system, as a whole, is stable.

MR. BURWELL: Before you go on, I want to clarify that some point. When you say, he looks at the configuration, are you talking about him looking at the configuration of the pipe, plus all of the support from one anchor point to the next anchor point, or stated another way, I think, at the pipe stress problem in it's total, or are you speaking of configuration as being related to only one support, and it's a piping for (inaudible)

MR. IOTTI: You would look at the overall configuration, probably from anchor point to anchor point..

MR. BURWELL: The configuration that he is studying for stability, is it overall system stability and not necessarily the stability of the, of each individual support.

MR. IOTTI: That, and I refer, that is the piping analysis that does so. Okay? The support designer is something else. The support designer has to provide a support which fulfills the function which the piping analysist has asked that support to fulfill.

MR. BURWELL: But, then, is the piping stress analysis reviewing and reaffirming that each support does perform the function that he needs to assure the

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accuracy of the stress analysis?

MR. FINNERAN: I think we stated in the affidavit that that is not the piping analysists job. That it is the support designer's job. Provide a support that will provide the function that the piping analysist wants, at that point. And that's what we say in the affidavit.

I think, personally, I still think what we've stated here is correct. That the analysist looks at figuring the supports for the stress problem. He knows that he wants to get enough supports on there in all directions, that it will be stable, and if he puts that thing in the computer program, and he controls the deflection, he will have assessed the system's stability.

He wouldn't be able to invert his stiffness matrix in the piping analysis program if it wasn't stable. And just what we say right there, if he has not provided the proper multi directional support required by the patten configuration, he won't be able to run the computer program.

MR. FLECK: That's true when you mean computer program.

MR. FINNERAN: That's true. This be a addle (ph) pipe or..I'm talking about piping analysis, super pipe, or new pipe..all of those, I believe, I'm

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I believe they all involve the production of stiffness matrix for the piping system, and if he has a direction for the piping that's not supported in a particular direction, there, he'll get a zero term on the diagonal lease, and on his stiffness matrix,and it will blow out.

MR. TERAO: Well, I take exception to the fact that you're saying that the deflection of the supports that assure the stability of the piping system?

MR. FINNERAN: No, I'm not saying that's a question on supports.

MR. IOTTI: He's talking about total deflection of piping systems. That's what I tried to clarify.

MR. FINNERAN: I don't think we said..we're not talking about the deflection of the supports. This is a specified deflection limits for the pipe.

And all points along the pipe.

MR. FLECK: Yeah, I think that's where I, that's why I asked that question.

MR. FINNERAN: No, this is not the deflection of the supports.

MR. FLECK: We're talking STRUDL and then we're backing to piping.

MR. IOTTI: This entire question relates to piping, but not to the support itself.

MR. TERAO: Okay. In that sense, I don't

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object to your statement as strongly, but I do, but I still do object if you are saying that the piping designer does not look at his system or stability, and only looks at the analytical results which may show movement of the pipe. Just from a very basic engineering principle, going back to my riser supporter from the bottom, that may show up to be quite stable in the weight analysis, and may show up to be quite stable in the thermal analysis, but your thermal analysis can tip the riser over, so you can't have an unstable system. It's not accounted for when you combine the two.

MR. FINNERAN: You don't think that controling the deflection limits on that riser would dilute that?

MR. TERAO: No, no. The deflection limits will not account for ..

MR. FINNERAN: I'm talking about in all directions. The deflection limits in all directions.

MR. TERAO: What I'm saying is, your deflection limits does not assure that the off set due to the thermal extension of the pipe will not affect the stability from the weight analysis. The weight analysis does not, does not have the deflections of the pipe as shown in the thermal analysis.

MR. IOTTI: What John started, and never quite

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finished is that, when he considers the configuration, what you said he should do in a good engineering sense, that's the time when the analysist choses where he wants the supports placed. It's at that time which, where he implicitly he consideres the stability issues.

In other words, if he sees a long riser, he will place intermediate support in that riser, so that he dosen't have the problems that you're referring to. But, that's not when he's finished, because those support may not be adequate to meet certain deflection criteria that he has imposed. So, he performs then, the analysis, and the combination of having done the implicit consideration of the stability question of the overall system, from anchor to anchor. Plus the fact that he now has an answer that says I meet my deflection criteria everywhere, is what gives him the assurance that he has a stable piping system.

The configuration analysis, if you so will, is done at the time when the analysist has an unsupported backing system, and he has to contend with how to support it. That's where the implicity goes through those considerations. He's the one that select the location of the support.

MR. TERAO: That maybe true, and it maybe true that the original person who selected the original

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location, did take that into account. But, as you know, as you go into the field, and you may not be able to add a support at the top of the riser, you may not, may be only to put it at the bottom of the riser, and supports get shifted around and skewed and so, someone has to look at support configurations at the end to assure that the system is still stable.

MR. IOTTI: It's either the same analysist or that information back from the field is relayed back to the analysist that says, is it still okay from the piping system standpoint. I'm talking now, both in terms of global stability, global stability being the stability of a piping system. Not local stability of a particular support. That information is relayed back to the analysist or some other analysist, which then has to review it and still accept it.

So, it's again looked at.

MR. TERAO: I think that goes back to the Spot's question about who is responsible for that? Who..

MR. IOTTI: The piping analysist only wants to know whether that support is being moved from here to there, and whether that support still satisfies the function for which he originally, that he originally specified that support should accomplish. In other words,

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if you said I need an axel restraint, here, he wants that support wherever it's being moved to, to be again an axel restrain, or he has to go back to the support engineers and say, now, okay, change the support type. I want something different, now. It is the support designer responsibility to make sure that the support, wherever he designed it, fulfills the function that the piping analysist calls for.

MR. TERAO: I'm still only talking about the piping system stability.

MR. FLECK: I think that's what he's elaborating. If you have a run riser and no support, therm at loose there, the weights there, could you ever have that?

MR. FINNERAN: Well, the analysist would see that in his deflection analysis.

MR. FLECK: Which deflection analysis?

MR. IOTTI: No, he wouldn't see that, because you don't combine thermal and with that weight in that fashion. He says, if this thing is tall enough or long enough, weight alone, under a different deflection, could cause a buckling problem, if you don't have support someplace. Or at the proper..

MR. FLECK: Now you have a huge moment from that, say, it's an 18inch riser, and it rotates one half a degree, you're going to have a big moment due to dead

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weight, would the piping analysis person normally have put in a side support there or not? It's a pipe problem, not a support problem.

DR. CHEN: Is it true, John, that, if that is stress problems, that as (inaudible)they get an estimate that would show what the analysist gets an estimate that would show is the runner piping from anchor to anchor, and together with that he gets a drawing of all the supports. And the drawings reflect the thermal movements are specified. Only at support points. If you don't have a support there, there's no displacement specified. The pipe is not going to give displacements to the length of the pipe.

MR. IOTTI: Oh, sure it can. Sure it can. At every point that we want to.

MR. FLECK: But, would he transmit that to the support people? Only places..

MR. IOTTI: He would only transmit to the support people what the support people are interested in.

MR. FLECK: So, if the piper forgot to put a snubber on a high riser, the pipe is the person, the piper would be the one that has created the problem, not the support people.

MR. IOTTI: But, on the other hand, if he

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has forgotten, and if that dosen't hurt him, then you don't need it. If it does need it, the piping analysis will tell him, hey, you need something here.

MR. FLECK: If he had did a seismic analysis and found out he's got a, or a thermal..what Dave is saying, the thermal he could do. There's no problem. But, now the pipe is displaced, due to this large riser, displaced a foot, say, horizontally. Now, you have a moment due to that weight, that's never included in the weight. I can't forsee that happening if he did a seismic because he would have to put a snubber up there, or he would never be able to take what's equivalent to a side load of .2 or .3 Gs, depending on your..

MR. TERAO: But, the snubber can bottom out, that's the point. Come late and come thermal.

MR. FLECK: Well, not from weight. From thermal it could, because the snubber would have to be horizontal.

MR. IOTTI: Well, let's be factual. Are we imagining a problem? I have never been in too many plants where you see this long risers without any actual supports along. You know, supporting the weight, intermediate. Okay?

MR. TERAO: If he'd done his job right, he'd have to take a snubber and put it in there.

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MR. IOTTI: Well, not always. Sometimes, yeah, you put a snubber, of course, and typically, you would design your support in such a way, even if you have a very large thermal expansion, you design it so that at the end of the thermal expansion..

MR. TERAO: Or he might just have a regular strut there..but, I think that's what Dave is, has a question about. Yes, that's basically, what I disagreed with, was your conclusion. Because when I read that, it seems to go against the grain of some of the basics of just piping design. That you are relying only on deflection, as shown in your analysis, to assure the stability of the system. That is what I'm disagreeing with.

MR. IOTTI: And what I tried to clarify is it is not just deflection. It's also configuration. That the typical analysist looks at the configuration. Where I have difficulty now, how does he look at it explicitly, he does not. I mean, he looks at it. But, then, you really rely on an individual's judgement and his experience that says this configuration may lead me to a problem and I need support here and there.

MR. TERAO: I see this contradiction, here. On the one hand, you're saying, yes they do, and on the

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other hand, John is saying, no, they don't have to. I guess, in that sense, I'm a little disturbed because I don't know, if you were to say, of course they do, this is common sense engineerying, everyone does that. But,..

MR. IOTTI: I don't think that's what John is saying. They do do it, and I don't think John is saying that they are not doing it. You're saying that he's now, are you telling me that they don't have to do it?

MR. FINNERAN: No, that's not what I'm saying.

MR. IOTTI: I didn't think that's what he's saying. That's..

MR. TERAO: And that's why I'm taking exception to this last statement, which negates any need to assess stability separately, that that is what I'm taking exception to. If you want to change that statement, you may.

(END OF TAPE)

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MR. IOTTI: We will have to, well, if you take that sentence in conjunction with the prior sentence, in addition, if the total support scheme does not provide the proper multi-direction of support required by the composite configuration, the way I read that, that says implicitly the piping analyst does look at the configuration. Cause, he's the one that selects where he wants to support, he's the one that selects what kind of support and what type of function the support has to exercise. And then, as he goes through his analyses and then confirms it, all his deflections is where he wants them to be, then he has the stability of the system that he desires. Whether we used the improper wording there at the end, I guess I'll have to talk to John and see how we might change it.

MR. TERAO: Maybe I just wanted to clarify where I disagree. Perhaps, I'm reading into this a little too deep then what you're trying to say, but, but,

MR. IOTTI: Well, I think you're reading it a lot more strictly then we usually do.

MR. FINNERAN: I was really trying to say what I said before in relation to 1C. We're talking about this type of stability where other supports that are unstable by themselves because they're figured in a system with neighboring supports in a certain direction that

controlling these deflections of the piping system will make the system stable. In that limited sense, this is really what this is saying.

MR. IOTTI: Well, look, we can go back and actually go back to the beginning as to how the normal pipe analyst does the normal. Everyone that I've ever known, regardless of whether they're are in a (inaudible) with Gibbson Hill and Stolen Webster or Bechtel does despite economists. Normally the mechanical people...

MR. FINNERAN: I think he agrees with that. He's just taking issue with this statement which was made in the narrow context of this in this Figure 1C.

MR. TERAO: Right. Maybe I'm looking at it not even from the sense of Walsh Doyle allegation but from a sense of piping engineering. I'm looking at it from a broad sense not only from what the allegation is saying. Because I want to clarify that. Because I don't think the allegations are really specifically aimed at system instability, they're aimed at pipe support instability.

MR. IOTTI: But, Mr. Bloch asked for us to also explain what system instability was because Dr. Chen brought it up here in the Hearing and I think Bloch got totally confused by all of these stability questions raised right and left.

MR. TERAO: Well, I don't blame him to tell you the truth.

MR. IOTTI: Actually, this year was pretty black and white, to be honest with you. I don't really understand why we were getting so confused on this.

MR. WADE: I think the whole issue came down to Walsh Doyle wanting us to consider each individual pipe support, without the piping. And our whole contention was, you must consider the piping there as a system with a standard way of doing analysis, that guarantees that the system is stable. The individual support by itself doesn't matter.

MR. IOTTI: Yes it does. Okay. That's the whole issue that I'm again raising. Because the support must perform the function that the piping analyst called for.

MR. WADE: But their contention was, it's was a 3-pin connection.

MR. FINNERAN: Aw, that's ridiculous, isn't it?

MR. WADE: That's where they started this whole thing. Support must perform the function that the puying analyst has called for.

MR. FINNERAN: If you've got no pipe there, you don't need the support to start with.

MR. WADE: Our whole contention and all we've

said, I think, is that when you take that whole system properly analysed, it's a stable system regardless of this 3-pin connection business.

MR. TERAO: Yes, and maybe I should continue with that train of thought because with the example that you gave us, 1C, 1A, D, and C, I would agree with what you saying there 100%.

MR. FINNERAN: Well, that's really the narrow context of my statement at the bottom of page 7.

MR. TERAO: Okay. I'd better continue. I would agree with what you're saying here with regards to system stability, provided that the support is performing its function, as Dr. Iotti has said, and providing that the clamp is performing its function too. But if the clamp can rotate then I don't agree that you have a system instability but you have a support instability, which again is deficiency but not necessarily a safety concern. And, I think that is the area that I would like to focus in on. Not piping system stability but on this particular, on a particular support potential instability.

MR. IOTTI: Fine.

MR. TERAO: As long as we're on the same wave length I don't think we'll have a problem.

MR. IOTTI: We are.

MR. TERAO: Okay. So let's take the first type of support. Apparently, you discussed four different types of unstable, potentially unstable supports, and provided either reasons why they are not potentially unstable any more, and the reasons for them becoming potentially unstable.

The first type of support is the box frame with single struc or snubbers. The modifications are shown on page 13 where it is saying that you may either use an index lug, or additional struc, or a cinched up U- bolt.

First of all, is this type of support, this type 1 type of support, including those box frames with and without thermal gap, between the pipe and the support?

MR. FINNERAN: Okay. First John will have to illuminate on that a little bit.

MR. TERAO: Are these box frames with single strucs or snubbers? Do they have zero gap and a gap, if there is one?

MR. FINNERAN: These supports generally, I believe, would have zero clearance between the pipe and the box frames.

MR. TERAO: Did you ever any of these with a gap? When you say that there is 12 supports of this kind I wasn't sure whether you're saying that that included the gap, or did not include a gap.

MR. FINNERAN: Well, these box frames with single structure snubbers were all the result of modifications made by the field engineers. It's possible that some point in time that they may have put a box frame around the pipe if it had a clearance but at some point, all these box frames were made with zero clearance between the pipe and the boxes.

MR. TERAO: I think that was my first question, whether or not there existed any, whether there still exists any of the box springs with a gap. I believe in your Attachment, it doesn't have a number, sheet 101, Request for Information, Comanche Peak Site, where it showed a single st c with a box frame and it looks like it had a gap around it. I wasn't sure.

MR. FINNERAN: I think that's just the drafting of that particular time, or the sketching of it.

MR. TERAO: So, you're saying that there are none with gaps that would exist, at this time?

MR. FINNERAN: No. Not without some of these modifications. I think more important than whether there's any with gap left, is that there are none left without and specifications made, regardless of whether they had a gap or not.

MR. IOTTI: All of these box frames and single strucs have been modified.

MR. TERAO: Right. But when you said there were 12 supports of this kind, and what your discussion is focusing around zero clearance box frame and the pipe. You say there's 12 supports of this kind. I wasn't sure if that excluded frames with a gap or whether it was implicit that there does not exist any frames with a clearance. And that's what I'm trying to clear up.

MR. FINNERAN: I don't think any of these had a clearance, or have a clearance on them now.

MR. IOTTI: Is the total number of box springs, with or without clearance, whether they might of had one, or whether it was intended or not. Is it 12? Is that the total? Is that the total sample? Box frames?

MR. FINNERAN: That's the total box frames that have single structures. That's it. That's the total.

MR. BURWELL: Let me just get our tenses straight. There were 12, now there are zero. Thank you.

MR. IOTTI: Now there are zero. Now there are bona fide box frames.

MR. FINNERAN: There might still might be a box frame on a single struc or snubber but it will have some other modifications.

MR. IOTTI: That's what I say. Whatever there

are now, they're modified.

MR. FLECK: So none of them can rotate, whatever their modifications.

MR. TERAO: So, now the question whether or not we will accept the modifications.

MR. CHEN: Let me just ask one question. In addition to the 12 are there any other single strucs that have, other than zero clearance, that have not been modified?

MR. IOTTI: There are only 12.

MR. TERAO: Okay. With respect to the index lugs, at this time, I can't accept the index lugs. This might be easier explained with the frame design, your Type 3.

Okay, let me explain what my problem is with the index lugs. As you pointed out earlier, concern with the support instability was the ability for the strength to rotate around the pipe. What would prevent the frame from sliding along the length of the pipe?

MR. FINNERAN: You talking specifically about index lugs?

MR. TERAO: Yes.

MR. FINNERAN: If you want to address index lugs, or even additional struc, you both, I know you're going to say, it's a cinch, Donald, but the first two

modifications, can slide along the length of the pipe.

MR. TERAO: I don't think so.

MR. IOTTI: I have an answer to that one, actually. As it turned out, from a test data, it indicated that unless you were also able to rotate about the pipe the tendency, during a seismic oxidation, was to pull back to align yourself further, as opposed to go out, and that happened in all instances. And you can actually rationalize why that is so. It's just a mechanical advantage that forces you to go to the region of least resistance. So, if it slides, it slides in a favorable manner. But the key there is you must prevent rotation, because the moment you present rotation you can actually gain an unfavorable mechanical advantage that makes the least resistance position away from you.

MR. FLECK: Are the lugs welded long enough on the pipe that the rotation is guaranteed?

MR. IOTTI: That I don't know. I don't know the details but I presume that the index lug would be such that the frame cannot rotate about.

MR. TERAO: Maybe I didn't quite catch what your argument was.

MR. IOTTI: Well, as it turns out, if you have let's say something that's misaligned, and you now have seismic exertation, unless the frame can rotate

about the pipe, what the seismic then will do will pull you back in a better aligned position, where you lose if anything, or are lined better with your pipes.

MR. TERAO: Thank you.

MR. IOTTI: You'll see that in the U bolt, also. And that happened every time and you can actually sit down with a piece of paper and prove to yourself that must be the case. The moment this frame can begin to rotate you can actually go to a lever arm, which is more favorable, that pushes the frame away from you, in which case the frame will slide along the pipe and, if you want to call it, in an unstable manner, you would do so.

MR. TERAO: Wasn't that testing done by exciting the support itself?

MR. IOTTI: No. Through the struc, the very bottom.

MR. TERAO: Well, that's what I mean. Exciting the support itself. In other words, the exertation was coming from the support, right? But, I guess my problem would be,

MR. IOTTI: But that's where the exertation comes from. It's a seismic event. Oh yes. Exertation comes through support, not from the pipe.

MR. TERAO: All right. There's two sources of exertation. One is,

MR. IOTTI: The pipe reacts to the exertation. If you have some sloff, you will have some back from the pipe, I agree.

MR. TERAO: I think what we have to talk about a little is the physical behavior of a plant during a seismic event. You have two basic movements acting on your pipe. One is the relative displacement, relative excitation of coming in through the support from the wall of the building. It's the relative movement between the building and the pipe.

But, you have another excitation, due to the building moving itself. This is your global movement of the building. And these can be quite large, in the order, maybe a dimension peak of 3 inches, which is small but in other plants it could be quite a few inches larger. If the plant is moving, globally several inches, back and forth,

MR. IOTTI: So, it's got support, moving globally, several inches, back and forth.

MR. TERAO: But not along the length of the pipe.

MR. IOTTI: What happens if the support moves with the floor or the wall?

MR. TERAO: No, it's the inertia movement of the support along the length of the pipe. The building

shakes, the pipe and the box frame around the pipe is moving together with the building, and when the building backs up, the pipe backs up with it because of all the support on it. The frame keeps on moving in that direction.

MR. IOTTI: I understand. You don't have to say nothing further. Now we're talking about the inertia of the frame relative to the building, inertia that pipe, and it's likely to be much less, the frame itself.

MR. TERAO: Well, I agree that the inertial vote is very small but it's enought to move it along the length of the pipe, unless you have a positive clamping.

MR, IOTTI: What I was actually thinking was the opposite because it's very small and it reacts very quickly, while the other one doesn't. If anything, that's what can make it slide.

I don't have an answer for you in that regard. I hadn't thought about it, but it is, at least, theoretically possible, that if you have the pipe actually moving and swaying and the frame lags behind it because its inertia, well, you still have the whole base of the support also moving the same way because that's the whole building. Then you have to look at the relative motion of the support itself.

MR. TERAO: That's right. But it's a 10-10

support so, basically what you have.

MR. IOTTI: In that, actually is a pin roll support, with a roll of the pipe.

MR. TERAO: What do you mean by roll?

MR. IOTTI: Well, it's essentially capable of sliding up there. You can't pin it.

MR. TERAO: Oh yes, but what I'm referring to is the effect that you're basically getting of that support. And the frame is that, the same type of effect you would see from swinging chandeliers.

MR. IOTTI: No, no. I don't agree with that. Because, first of all, in a seismic event this thing is also moving up and down, and sideways, in addition to just axelly. If you have pur axel movement, I would only see it your way, but since this is now making contact at various points, during a seismic event, you don't have the same thing.

MR. TERAO: Without the pipe, you would have the chandelier.

MR. IOTTI: Oh, yeah.

MR. TERAO: What you're saying is that, it is reacting against the edge of the plant. I guess my problem from this point is that there's so much vibration going on that the plant can literally walk.

MR. IOTTI: This is why I said if it walks,

unless it can also rotate, it walks, it will tend to walk back towards the position of the lineman because that's the minimal management position.

MR. TERAO: That was based on your testing where the exertation was coming only from the support itself.

MR. IOTTI: Right. But really, when you think about it, what is the difference? Suppose we'd even had moved the pipe, the same thing would have happened. See, the support is still be excited in the proper manner. Support is on the floor and you're shaking it up, down, and also, in this direction. Don't forget, we put in the load at an angle, within conial angle, so we could excite both the axel, the vertical and the lateral. So the support was being excited in three, in three directions. And what we found, sure enough, is that if you don't prevent rotation and you can also imagine the pipe moving at the same time, for all we know, for it was moving, incidental. And it was also being excited, actually to a far lesser degree, but laterally. You can imagine if you start rotating, and we'll show you the picture, it would walk away but if you can't rotate, it actually pulls it back. Then it sits. Once it finds its position where it likes to be, it'll sit there. And, unless the earthquake radically changes, it isn't going to move from there.

MR. IOTTI: That's what I'm saying. Implicit in this is you have a certain length to look that permits it to slide that distance back. We can actually prove mathematically whether you're right or not, if we wanted to do a following on this.

MR. TERAO: No, no. Now, that is what I am trying to avoid if we can't reach any resolution on this but, if I prove this mathematically,

MR. IOTTI: Let me ask John a question that maybe makes this whole thing mute. Do we have any modifications?

MR. FINNERAN: We have one of the index lug.

MR. IOTTI: If you had said zero, we could have stopped right there.

MR. FINNERAN: Well, we have one of them. That's why it's on the table. If we didn't have any, it wouldn't be in the (inaudible).

MR. FLECK: How long, what's the lug length?

MR. FINNERAN: Well, this particular support as a matter of fact, is on a vertical run of pipe, right below a penetration into the force lab.

MR. FLECK: Oh, so it has to have a vertical.

MR. FINNERAN: There are lugs on the pipe to hold the support to keep it from moving down the support and it can't go up the support because it,

MR. IOTTI: So, then it can't slide anyhow. So it's a non-problem, because it can't slide to begin with.

MR. FLECK: I think I remember that. It's right by the floor level. Right under the floor. The floor level won't permit th..t excess dimension of the frame to drop through it. Do you have a support number on this?

MR. IOTTI: I think that may be the easiest way to dissolve it. Just give you the drawing of that support and you can take a look and satisfy yourself. It won't rotate, nor will it slide.

MR. FLECK: Otherwise, I guess, if you wanted to at least eliminate the problem, you put the lugs with another vertical (inaudible) that comes out horizontally from the pipe to stop it from going, ..

MR. IOTTI: Well, you can put a lug and then a larger log that contains the frame.

MR. TERAO: That doesn't necessarily mean that stirrings have been put to bed. What line is this on? Do you recall?

MR. FINNERAN: I believe, it is, let's see. CTR.

MR. TERAO: Do you have the support number chere?

MR. FINNERAN: Yes. I think it's one of the,

support in the OR disc, I believe. I think it's CT1008008F22K.

Chatter.

MR. IOTTI: At this point, I guess the best way to proceed is to look at the drawing.

MR. TERAO: Yes, I'd like to look at the drawing.

MR. FINNERAN: This particular one actually has two snuggers coming into the frame. So it's not a single.

MR. TERAO: How's it going to rotate?

MR. IOTTI: I don't believe that particular frame would function in a stable manner. We just decided to do something extra. I think I stated that back in September 82 Hearings. First thing about Dr. Cole, that we were talking about one of the Dole's frames. One that had a struc in, tow strucs coming to the front. Neither he nor I thought that was unstable frame. Regardless, we modified the frame.

MR. FINNERAN: I think it also gets into the question of safety concern. Is there a potential safety concern? What could actually happen to this type of arrangement?

Of course, I'd be hard pressed to tell you or to tell anyone why there is a problem here, but, on the

other hand, because of the unusual design it would be very difficult for you to prove conclusively that the design is working too. You get into arguments, such as, impacting, if the frame is excited in a vertical direction. If it is indeed stopped by a, by the lugs then you could have the frame then basically bouncing on the lugs. You could get some type of loading that way that is not accounted for.

There is just two many uncertainties with this type of design.

MR. IOTTI: The frame is still a contact frame. That's my understanding. Is that right?

MR. TERAO: What I'm trying to get at, is this design, even in an excellent additional struc,

MR. FINNERAN: It's very difficult to show analytically that the design is going to work. You would have to do some type of non-linear analysis, maybe a time history analysis, to show conclusively that,

MR. TERAO: Why do you think this struc wouldn't work? (inaudible) and I think you can show analytically that frame will not slide down the pack. If this support is performing its function it has a load on the detension or depression. If it doesn't have that load on it, it's not going to go anywhere.

MR. IOTTI: That's why I need the drawing and

the next time at the site look at the support itself; see what type of gaps, what type of rattling the frame could have.

MR. FINNERAN: What one?

MR. TERAO: The index lugs.

MR. FINNERAN: I hate to say it, but sometimes it might even come down to an engineering judgment of whether or not that support is going to work, cause you may not even be able to show analytically without some extremely expensive analysis for this single frame. It may not even be worth it for a single frame. I don't understand why a simpler fix wasn't initated. But that's water over the dam.

I guess what I'm saying, all right, we have one of these index lugs, but still the basic problem comes down with the additional struc modification where you get into an area where the clamping mechanism around the pipe is not a controlled factor. It's not like a typical clamp where you can tighten the bolts, and assure through the tightening of the bolts, that the clamp is going to perform its function.

This is one of the basic problems I've had with these designs from the start. The type of clamping mechanism that you're relying on, there's either friction, or the type heating up in these box frames.

It is not controlled. There's just too many other physical factors that can affect that clamping mechanism. Therefore, the box frame itself can lose its clamping function.

This goes back to my initial definition of what do I mean by unstable? By instability? What I'm talking about is whether a clamp or support can lose its function and become ineffective and possibly move to an undesirable or unanalysed position.

MR. IOTTI: Well, you'll agree that one could not rotate. Here, your concern would be with sliding only.

MR. TERAO: Yes.

MR. FINNERAN: I think you can show on additional struc that analytically that frame will not slide.

MR. IOTTI: It might slide, but how far can it go, because the struc has a finite length? The difference is when we test, like we did it, is when it started sliding, the testing apparatus compensates and increases the stroke until we were unable to increase the stroke any further, then we couldn't trust it with the load. But the floor, which transmits the seismic event, will not do so. And so, finally it will slide but it's controlled in terms of sliding by the presence

of the pipe, and the fact that you had to find that length struc.

Well, what he's saying, you'll have a hell of a time proving it analytically. I may even compare with him. It comes right down to someone's engineering judgment. Just proving it analytically will cost you more than modifying the damn frame.

MR. TERAO: So, I don't want you to take my signals as meaning I want more analysis done.

MR. IOTTI: No. I understand. I know when analysis is essentially, you're running cross the law of diminishing returns. There's no sense in doing it. We think if you were to look at the plan your engineering judgment would, in all likelihood, concur. I really don't see where they would slide. Then, I guess I would make the same statement as I made on the previous one. If it slides, I think it would slide back into the position where it's been. It's more on line rather then less on line.

MR. TERAO: You see the difficulty that we're in right now to show this analytically would be too expensive.

MR. IOTTI: And you probably (inaudible) the results anyhow.

MR. TERAO: That's true. You could question

more of the analytical techniques, and go on endlessly. On the other hand, to show that it's a problem I can't do that either. You can't even show that this thing can fail. You can make many assumptions but many of the assumptions you make are just as invalid.

MR. IOTTI: The key is the very statement that you made. Will it, even it moves, will it retain the function? In other words, would it be able to move sufficiently so that it cannot transmit the load? Or would it change the configuration of the note of the pipe? Don't forget. This is all seismic stuff. That's what we're talking about.

Let's say, it moves an inch even. Would an inch make a difference in the piping analysis? If the support were an inch downstream or upstream, whereever it's located, provided it can still carry the load?

That's the kind of engineering judgments that we have to use with these supports. The only way you can exercise is by actually looking at it.

MR. TERAO: I think my basic concern here is still the fact that from an operational loading, whether or not these supports can move. Suppose it moves to a position before the seismic event? I'm not saying that it can move just only during the seismic event, but there's enough vibration in the plant on some of your

piping systems.

MR. IOTTI: There has been vibration in the piping system already at that plant. I've been there. I've been going through the pre-op testing. You would have seen if there had been something flagrant, you would have seen it.

We tend to forget that but this plant has gone through (inaudible) one and have you noticed that anything of this sort has happened, or moved?

MR. FINNERAN: Well, these particular. I know one of these double structures in on the compound (inaudible) system and that thing has actually been in operational testing for over a year.

MR. IOTTI: So, you know, if you want we can take a look at it and measure it before and after. Maybe we should have data before, where it was. What do you think, Bill?

MR. HORIN: I don't think it's moved. I don't think it's moved at all.

MR. TERAO: And that system vibrates.

MR. CHEN: How many of these do you have?

MR. FINNERAN: I know the total of all three of these is 12.

MR. TERAO: You get the additional struc is a modification, what I would confirmed is this original

type 1 frame, box frame with single struc snuffers, are there other additional double struc frames that were always considered to be stable from the very start.

MR. FINNERAN: Oh, possibly. I don't know.

MR. BURWELL: I don't understand where that question is going.

MR. IOTTI: He feels he is not in agreement that this is a proper modification. That any other frame that we may not have included in this list because it had double strucs and would therefore consider stable. He would then question those also.

MR. BURWELL: I would agree but I don't think he should reach this conclusion on a double struc frame. To say that the system is stable--there's many uncertainties involved here. Perhaps the frames on the component cooling water had enough friction on it so it wouldn't move but suppose there are other frames where you don't have as much friction. In other words, it's not controlled. You don't have a controlled method of clamping that frame around the support, I mean around the pipe.

MR. TERAO: So, in that sense, I would find it very difficult to agree with this modification when I don't physically know what is out there in the plant.

MR. HORIN: Even if you assume the worse case

scenario that you think could create some instability, what exactly do you think would happen that would cause it to be unstable? Is it simply sliding up and down on the pipe?

MR. IOTTI: That's the only thing that could happen in this particular instance.

MR. TERAO: Well, let me ask you this? Let's not talk about these clamps at all but let's just talk about a standard clamp on a snubber. You're running an in-service inspection and you found that the clamp had moved several inches down the line, would you thank that was a major problem? Would that be identified as a deficiency?

MR. IOTTI: Well, not necessarily a deficiency. It would be identified. I don't know whether it's a deficiency or not.

MR. TERAO: It has to be corrected.

MR. IOTTI: It has to be determined unless it can be proved that it doesn't matter.

MR. WADE: Move it back to its original position and (inaudible) the bolts.

MR. IOTTI: Not always, Dave. Let me correct you. Sometimes you find it had moved because to begin with it was wrong and where it moved to is where it ought to be. So the fix is not to take it back. It's

to leave it where it is.

MR. WADE: If we move it where it ought then they would not have known it moved.

MR. TERAO: Well, not because they noted where it was to begin and they thought it was correct where it was.

MR. IOTTI: We assist by how much one could physically slide. I think that's the key. You can't go too far.

MR. TERAO: We could go with Dave, take him to look at all of these frames and have him assist for himself. Even if it slides. How far can it go? Maybe it can only move an inch. Now, maybe an inch is important. I don't know what the answer is. I'm not saying. Maybe an inch is terrible. But it also may be, your span may be 20 feet long and you move one inch on either side of the span. That is less than the tolerance that you normally supply when you install the support in.

MR. IOTTI: This is the kind of consideration that we have to look at. Clearly, it's only a sliding consideration. Forget about the other things that you brought up, which are valid. Possible impacting. That's true. It's possible.

MR. TERAO: Now, I agree. The difficulty here

is not that we're saying these are safety concerns but the difficulty is trying to address these summary dispositions on whether or not these modifications are acceptable. That is a totally different ballgame because I can say I don't believe there's any safety concern. But on the other hand, I can't say in my Summary Disposition that I think these modifications are acceptable.

MR. IOTTI: Well, you must say what you agree to be right. What else can we do? We're at am impasse here. Clear, that we do believe that this is a proper fix. If you don't agree, we may have to modify it further. There comes a point in time where we're not foolish either. It's a lot cheaper for us to go fix it then try to fight you on this. We're not talking a large number.

MR. WADE: I think its important that you come and look at it. I really do.

MR. TERAO: The difference is I can go look at it and say it's no problem. Another person can go look at it and say there is a problem.

MR. WADE: I think any practical, sound professional engineer would have to agree.

MR. BURNWELL: Okay. I think we have reached a good break point here and so with that, let's close for

the day and we will come back tomorrow morning at nine o'clock and continue with our discussions on stability. Thank you.

(Meeting adjourned at 6:10 p.m.)

CERTIFICATE OF PROCEEDINGS

This is to certify that the attached proceedings before the NRC COMMISSION

In the matter of:

QUESTIONS ON SUMMARY DISPOSITIONS FILED BY TEXAS UTILITIES ON COMMANCHE PEAK

Date of Proceeding: August 8, 1984

Place of Proceeding: Bethesda, Maryland

were held as herein appears, and that this is the original transcript for the file of the Commission.

Joe Newman
Official Reporter - Typed

Joe Newman
Official Reporter - Signature