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PUBLIC NOTICE BY THE
UNITED STATES NUCLEAR REGULATORY COMMISSION'S
ADVISORY COMMITTEE ON REACTOR SAFEGUARDS

WEDNESDAY, MARCH 30, 1988

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UNITED STATES NUCLEAR REGULATORY COMMISSION

ADVISORY COMMITTEE ON REACTOR SAFEGUARDS

In the Matter:)
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STRUCTURAL ENGINEERING)
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Wednesday
March 30, 1988

Malibu Room
Pacifica Hotel
6161 Centinela Blvd.
Culver City, California

The above-entitled matter came on for hearing, pursuant to notice, at 8:30 a.m.

BEFORE: DR. CHESTER P. SIESS, CHAIRMAN
Professor Emeritus, Civil Engineering
University of Illinois
Urbana, Illinois

ACRS MEMBERS PRESENT:

DR. PAUL G. SHEWMON
Professor, Metallurgical Engineering Department
Ohio State University
Columbus, Ohio

DR. DAVID A. WARD
Research Manager on Special Assignment
E.I. du Pont de Nemours & Company
Savannah River Laboratory
Aiken, South Carolina

ACRS COGNIZANT STAFF MEMBER:

Elpidio Egne

NRC STAFF PRESENTERS:	Page No.
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PRESENTATIONS BY:	
Sam W. Tagart Jr., P.E. Technical Specialist Nuclear Systems and Materials Department Electric Power Research Institute 3412 Hillview Avenue Palo Alto, California	27
William English General Electric Company Nuclear Energy Business Operations Structural Analysis Services 175 Curtner Avenue San Jose, California	62
Sampath Ranganath, Ph.D. General Electric Company Nuclear Energy Business Operations Manager, Structural Analysis Services 175 Curtner Avenue San Jose, California	111

NRC CONSULTANTS:

Spencer Bush

Everett Rodabaugh

March 30, 1988
8:30 a.m.

-- PROCEEDINGS --

CHAIRMAN SIESS: Good morning. The meeting will come to order.

This is a meeting of the ACRS Subcommittee on Structural Engineering, and present today, starting on my right is Paul Shewmon, Dave Ward, and we have two consultants: Mr. Rodabaugh and Mr. Bush.

Today we will review and discuss the EPRI NSRC Piping and Fitting Dynamic Reliability Program PFDRP--unpronounceable.

The cognizant ACRS staff member for the meeting is Elpidio Egne, who is seated on my left.

The rules for participation by the public at today's meeting was announced as part of the notice published in the Federal Registry on March 14. It says here that the meeting is being conducted in accordance with provisions of the Federal Advisory Committee Act for the government, and the Sunshine Act, and we've received no written statements from members of the public nor any request to make oral statements.

These microphones are not working. He said he might be able to get them fixed during the break. We are a small enough group that I think the--if the people sitting out here want to move up a little bit they can, but let's just try to speak loudly enough to be heard. I'm a little hard of hearing, so I may be the test for the volume level, and please give your name the first time that you speak so that the recorder can get it.

Any of the Subcommittee members or the consultants have any opening remarks that they would like to make at this stage?

[No response.]

Just for the record, I would like to point out that we had the opportunity yesterday to visit the two sites at which tests are being made at Blue Tech on the system's test, and at Anco on the component tests, and there will be essentially no repeat of what we learned at those visits. We will concentrate today on a brief review, I think, of the program, but then we will concentrate on the test results and the analyses and something on what is being considered for changes in the ASME Code on the piping.

We will start off with Dan Guzy from NRC Research. We have both NRC and EPRI represented and Mr. Guzy and Mr. Tagart will lead off this morning.

Dan.

MR. GUZY: Let me just emphasize some of the things that Professor Chet said about the presentation. We will be covering the summaries of the tests that we saw yesterday of the systems and the component tests, plus a little more on the specimen tests that were discussed briefly yesterday, and Bill English of General Electric will handle that.

Sam Ranganath of General Electric will talk about our concepts now for changes to the ASME Code, but the point is I will begin off with a brief talk of the structure of the program, status, a little about what I consider the highlights of what has happened from the programmatic point of view.

Sam Tagart will talk a little more about the technical overview and perhaps give a little more of an industry perspective on why they are doing this program.

Before I begin, I would like to point out that this program is called the EPRI/NRC Piping and Fitting Dynamic Reliability Program. The reason that EPRI has the top billing is--well, actually two reasons: one is they are contributing more money to this; but more importantly they have the lead in the planning of the program.

The NRC has been involved in this program from the beginning--from the beginning of the testing and the analysis, but the lead, in terms of structuring this program, the credit goes to EPRI. The NRC recognizes this is a good thing to be involved with, and we got involved after it had been pretty much planned out.

As far as the program, I would like to start off by talking about what the emphasis of this program is, so there is no uncertainty with what we are trying to do here. The emphasis of the program is the design of piping components for dynamic inertia loads. The key words are design--we are talking about design rules, not so much inspection rules. Piping components, we are looking at the stress rules, or the rules for designing elbows and tees of the piping system, not so much supports and say nozzles, but that would be considered in our design roles.

Also inertia loads: one of the chief objectives was to provide a more rational set of rules for dynamic inertia loads because that seemed to be an area of concern. We will address other types of loads though, too, such as anchor motion loads.

The objectives of the program have been from the beginning to identify clearly what the dynamic failure mechanisms and failure levels are for piping systems under dynamic loads. It is important to know what the level is for the large cycle failure and how do they fail so that we can develop more rational rules for preventing the failure.

Also, we are interested in gathering high level response information so we can know more about what happens in the area regime of a failure, in terms of parameters such as damping, ductility, deamplification, we are lacking information in this area, and this program is providing some valuable information for things not only in this program, but for future programs, as far as bench marks and data that we can use later on.

And, the key, final product of this program will be a recommendation for changes to the ASME Code. We are talking about changes to the design rules themselves, as given in subsections MB, MC, and D, stress allowables for Class 1, 2, and 3 piping.

CHAIRMAN SIESS: Dan, you use the term non-linear response in there. This may be partly semantics, but it is the way that I think about it.

When I look at this, I am really looking at the inelastic response. Now, I admit that inelastic is non-linear but nonlinear is not necessarily inelastic.

MR. GUZY: Right.

CHAIRMAN SIESS: And, I think the inelastics, that's where your large damping comes from, not from a nonlinear.

MR. GUZY: Okay.

CHAIRMAN SIESS: But, that is your thrust. It is the inelastic.

MR. GUZY: Yes.

We lack information in the very high levels. Maybe the emphasis should be on high levels rather than nonlinear, but that is where we didn't have much information.

I have got a list of the cast of characters, or some of the cast of characters that are involved in this program, and many you met yesterday. I would like to highlight the people that you didn't meet yesterday. Y.K. Tank is also from EPRI. He is program manager for the tests, for the Anco and the systems test for the EPRI, and of course Sam Tagart is overall program manager for the EPRI program. A lot of credit for the development of the program comes from--the credit should be given to Sam.

I am the NRC person, Dan Guzy, and responsible for the program in terms of what research programmatic responsibilities are.

From General Electric, I have listed some of the people--not all of the people, but the main program manager is Bill English, who you will hear from later. A person who is not here today, but has been heavily involved in the analysis is Henry Hwang. Sam Ranganath, who you will hear from later on, is involved in developing the

Code rule changes, and Ed Swain is another General Electric person who has been heavily involved in these Code changes also. There are other people from GE that I have not listed.

From Anco Engineers you have met Kelly Merz, who is here today; also Paul Ibanez you haven't met, at least at these meetings, and is involved in the program.

ETEC, you met Mr. Devita yesterday, Ron Johnson and a cast of thousands, I guess, at ETEC yesterday.

A key part of the program, but not represented by an individual today is the specimen tests. The main person for that has been Roy Williams, formally of General Electric of Schenectady. Now he has his own company called Material Characterizations Lab, and he is the one responsible for the specimen tests, and Bill English will talk about those tests some today.

There also have been several consultants involved with the program who have reviewed it and have given a few suggestions for changes in the program, and these included Everett Rodabaugh, who is here today; Bob Kennedy; Don Landers; Bob Cloud; Doug Munson; Stan Moore from Oakridge; Bob Bosnak has also served, he is from NRC and has served as a consultant; and Verne Severud.

[Slide]

Okay, the program is structured into eight tasks, and I will just briefly go through what they are. We had a Task 1--and who is involved in it--Task 1 is the program plan development. This has been the primary responsibility of General Electric in San Jose.

The pipe component test, which you saw yesterday, is at Anco, and they are responsible for all of the 41 tests that they have run.

The pipe system testing has been split into a number of organizations. The main seismic and hydrodynamic tests have been conducted at ETEC. You saw the results of these yesterday. The point I would like to make is the system 1 and the system 2 tests, the red and green tests that you saw yesterday, have been in integral part of this program; however, some of the earlier tests, the demonstration tests, have been part of the NRC's contribution to this program, although it is not formally a part of the program. There is a distinction between the tests--maybe it is just a paper distinction.

The water hammer test that you saw yesterday, of course, was being conducted at ANCO, so that all of these together consist--comprise of Task 3.

The other tasks--okay, the specimen tests at Schenectady are a separate set of tasks which we will hear about today.

The remaining part of under GE's responsibility.

They have done the analysis of the tests. They have taken the data and ETEC and ANCO and MCL have supplied them and looked at this data and summarized it, and you will hear more about that today. They have also been charged with developing--to identify and justify new design rule changes based on these test results.

And, the final reports are General Electric's responsibility, although I think ANCO and ETEC have reports--okay, so that GE is in charge of the final reports and they will be draft reports that will be supplied by GE to EPRI. The final reports will be EPRI reports not General Electric, however the NRC has information, all of the data, and we just--the burden of publication is not on us for this one.

[Slide]

Okay, as far as the status and schedule, the program itself, in terms of doing anything other than program planning began in the spring of 1985, three years ago. All the testing now has been completed except for the retest of System 1 which you saw ready to go at ETEC yesterday, so all of the component tests have been completed, the water hammer test, and the specimen test, so having been completed--some of these very recently have been completed, but they are all finished now.

The process of evaluating this data and developing

Code rules ongoing you will hear about where we are today, but it has not been finalized yet, so we are still working with the data and drawing conclusions to make recommendations.

The program itself will formally end in June of this year. General Electric being the main contractor, that is when their role is over. Other than writing reports, most of the other subcontractors are completed.

Okay, then when the final recommendations are made to the program, of course they will be reviewed by EPRI and NRC--at least in NRC Research, and then will begin the Code revision process through the various organizations that will be involved in the Code changes.

And finally, as I mentioned, the reports will be published probably sometime this year, I imagine. That is the EPRI reports.

[Slide]

Some of the key points that I would like to make from perhaps more of a programmatic point of view. This is a formal EPRI/NRC research program. We have a formal agreement on it. There have been five review meetings that have been held with the program managers and consultants. The most recent one was less than a month ago. This is our way of getting input, by getting everybody together, getting input on direction and what the results mean, and it has had an impact on--particularly in the component tests with--there have been some changes made, suggestions as we reacted to data as it has been coming in.

All along there has been interactions with the ASME and the PVRC standards groups, in a number of ways. First of all, we have been giving presentations to everybody at the meetings, and also a number of the members, the people who have been involved with this program directly, are also in this core group, so there is direct involvement by many of the members.

CHAIRMAN SIESS: Excuse me, Dan.

Does PVRC write standards?

MR. GUZY: Well, they--what do they do? Write recommendations?

MR. BUSH: They write recommendations basically.

CHAIRMAN SIESS: And, where are they implemented? In ASME?

MR. BUSH: Yes.

What we do, for example, is I would write a letter and transmit it to say Roger Reedy, the chairman of section 3, suggesting an implementation of a given action. That is the mechanism. We are not a formal standards writing body as such. PVRC is the reason they do it...[voice fades out of hearing range]...certainly do it, but it ends up going directly into the code.

MR. GUZY: There has been some activities that have had an impact on the Code, the Code cases. As a-- using the data from the tests we have to date, a Code case--a Class 1 Code case effecting BBOB allowables has been approved through the Code system, essentially this gives relaxation for inertial load requirements on OB and overloads at B level--

CHAIRMAN SIESS: What does it do to the change? What dominates the design? OBE versus--

MR. GUZY: --this would--if implemented, this would make the SSE dominant, essentially of less importance than OB.

There is a similar Code case--class--that should be Code case, not class--for Code case, not class, for class 2 and 3 piping that is up to the main committee now in the ASME codes. I believe that section 3 has one more committee, or two more committees.

CHAIRMAN SIESS: Help me a little again.

I don't think class 1, 2, 3. The only thing I think, from what I deal with, is seismic category 1 or not seismic category 1.

Does 2 or 3 make any sense in that classification? Or, is it something else?

MR. GUZY: They are both subsets of that classification. They are all seismic category 1. Class 1 frankly requires a more rigorous fatigue analysis--

CHAIRMAN SIESS: Okay, but--

MR. GUZY: --so you don't have that in the--

CHAIRMAN SIESS: --2 and 3, in this sense, is category 1 piping?

MR. GUZY: Yes.

CHAIRMAN SIESS: It is just a different type of analysis?

MR. GUZY: Right.

CHAIRMAN SIESS: Okay.

MR. GUZY: And, there are different rules that have to be changed. The class 2 and 3 rules are pretty much identical, as far as the design part. Class 1--

CHAIRMAN SIESS: In terms of the plant, how do you decide whether something is class 1, 2 or 3?

MR. GUZY: Class 1 has to do with the pressure boundary--primary system pressure boundary--

CHAIRMAN SIESS: Primary system pressure boundary.

MR. GUZY: --main loops in the surge lines and the recirculation loops.

Class 2 and 3 are other piping than category 1. The distinction between 2 and 3, I think, is more of an inspection--maybe that is sort of an arbitrary type of thing.

CHAIRMAN SIESS: So there is an isolation valve between class 1 and class 2 and 3?

MR. GUZY: Right.

MR. BUSH: However, for clarification, the utility has a considerable say in the pipe. There is one utility that has a system they call class 2, and it doesn't necessarily say it will be the same--[voice fades out of hearing range]

CHAIRMAN SIESS: Mr. Bush, we can't hear you.

MR. GUZY: I think the point here is that most of the piping that would be required is class 2 and 3. We pay a lot of attention to class 1 and sometimes, even in this program--

MR. BUSH: Don't say that, Dan. That's not true.

CHAIRMAN SIESS: In a PWR--

MR. BUSH: Most of the piping--you say safety related, because don't say "safety related" because that is not true.

MR. GUZY: Yes, yes, that is what I meant. I am sorry.

MR. BUSH: Okay.

Most of the piping now is--

MR. GUZY: What I am trying to say is there is a lot more class 2 and 3 systems than class 1 systems.

CHAIRMAN SIESS: All right, but in a PWR, steam lines are what?

MR. GUZY: Class 2.

CHAIRMAN SIESS: Class 2.

MR. GUZY: Well, PWR is class 2.

CHAIRMAN SIESS: Well, PWR is steam lines, and are class 2.

MR. GUZY: All right.

CHAIRMAN SIESS: And a BWR, what is class 1N?

MR. GUZY: It would be the--

CHAIRMAN SIESS: Like a turbine stop valve? No?

MR. GUZY: Isolation valve.

MR. BUSH: Inaudible.

COURT REPORTER: Mr. Chairman, Mr. Enge, I can't hear Mr. Bush at all. I can't even see him.

CHAIRMAN SIESS: That's all right. It wasn't important.

MR. GUZY: If there is any point to be made here, it is we are concentrating--there is a lot of emphasis on class 1 piping, and even programmatically we have GE who is our main contractor; however, we plan to address class 2 and 3 piping, and this is probably where we will get the biggest relief in snubbers and snubber reduction, et cetera.

Other than the Code cases there is also an activity that just started with the PVRC, and is a task frequent functionality criteria. NRC has a requirement on piping functionality which we think that the results from this program can support changes to, and that will be something that PVRC will make a recommendation on, and perhaps the NRC itself will take care of the standard change on that.

Publications, there has been a number of papers that have been presented, and will be presented, in like the Pressure Vessel Piping Journal and SMRT. There have been four semi-annual progress reports and you all should have received the last one from this program, and that will be the last progress report. The next set of reports will be the final reports, and again will be issued by EPRI.

CHAIRMAN SIESS: Dan, let me ask you a slightly unrelated question, as far as this particular thing is concerned: the National Research Council Report on the NRC research program placed a considerable emphasis on peer review, and in the response to that report the NRC research seems to have said that the way to get peer review is to publish in referee journals.

Now, I notice here that you have got papers and journals--I assume that SMRT is more or less referring, although I question it sometimes, whether they ever threw anything out, but you have your panel of consultants. Which serves as peer review in your mind?

MR. GUZY: Personally, I think the consultants do a better function of peer review than having it published; however, it gets to a wider audience by having it published in the journals.

CHAIRMAN SIESS: Do you think publishing in a journal is a peer review? It is sort of after the fact, isn't it?

MR. GUZY: I personally--I think it is valuable, but I think it is more valuable to have the right people review it, in a more formal setting.

CHAIRMAN SIESS: Well, once you have said "the right people" I won't ask you whether you think this is a peer review setting.

MR. GUZY: As far as consultants, I am convinced that these are the best people we could get to review the program. I mean, nobody is not on that list that should be on the list. I think it is an impressive list of people involved--

CHAIRMAN SIESS: Now, whose job is it to see that they are listened to? Yours and Sam's?

MR. GUZY: Yes, sir.

CHAIRMAN SIESS: Okay.

MR. GUZY: Let's see.

Maybe since we are talking about this, there is planned to be three--I think three main papers that will come in the near future that will try to summarize and explain what this program has been about, and what the findings are, and that will be--and there are three important papers that are planned that will be, you know, supported by the reports but this is our way of introducing to the world what we are really doing.

CHAIRMAN SIESS: Very good.

MR. GUZY: At the end of the program, we realize that there is a lot of information here that is valuable that we don't even know about yet, and we plan to do the best we can of storing this information. It is all available to the NRC, but we would like to make it available to everybody else and not throw anything away, and so there are some plans to archive those ANCO and ETEC tests at the NE Center in Charlotte, I believe, also the information, the data, we will try to do the best we can to save all of that.

[Slide]

One more slide I would like to present on is just talking in terms of what the NRC's perspective of this program has been.

The main thrust of why we are in this comes from the activities of the piping review committee of many years ago. The piping review committee looked at piping design and they identified a number of concerns about overdesigning for dynamic loads, especially inertial loads.

In their reaction to this they made a number of recommendations; however, because of the state of the knowledge or information data at the time, recommendations were mainly in the response areas, and of these the most significant, the one that has paid out perhaps the best has been the damping.

So, their immediate recommendations for changes were addressed more to response, because they didn't have data; however, the realized that they needed failure data and one of--the highest, the A category priority items that the piping review committee recommended for research has been to do pipe tests and this program was mentioned by name as something we should be involved with, so NRC research and NRC is involved in this program because our piping review committee recommended it to us, and I think it was a wise thing to do.

We've had a number of interactions with the NRC staff and we will continue to have that. There has been much information sent informally. There has been presentations and video tapes to staff, to people interested in what was going on. There has also been meetings, formal meetings, on other subjects where results from these tests have been quoted in terms of what's happening with piping? How is piping going to fail? In particular there was a formal meeting for the staff and people from standards group on damping code case N-411--

CHAIRMAN SIESS: When you say "staff," you are staff. You mean "other staff."

MR. GUZY: I mean the whole staff, and not just research.

CHAIRMAN SIESS: You mean NRR--

MR. GUZY: I mean NRR and the licensing people and people that now inspect the projects are involved in piping, too--the people outside of the Nicholson Lane Building.

There was also a presentation on this class 1 code case that I mentioned before, and we presented information from this program in support of these other changes.

We've given presentation at our information meetings at Gettysburg year to year, and then perhaps the first really formal presentation of staff, solely on this subject, was given last September when we gave detailed briefing of the results and where we were heading at the time and criteria development.

Today is the first meeting with the ACRS. We are interested in your comments on the program, and any suggestions or comments you may have on the program once you hear us out today.

We also plan to have future meetings with the NRC staff, particularly licensing people, and particularly in terms of the criteria of changes that we probably will be recommending. Many of the--all of the standard groups will be involved, the ASME representatives from the NRC. We like to have feedback from the staff before--get some direction from the staff before we actually start becoming involved as an NRC representative to the ASME, so we plan to have meetings this spring with the staff to present what we have developed.

In terms of how the regulatory changes go, the Code case, such as the stress allowable Code cases are endorsed formally through revisions of R.G. guide 1.84--

CHAIRMAN SIESS: That is the one that periodically comes up?

MR. GUZY: Yes.

CHAIRMAN SIESS: And, updates the reference--

MR. GUZY: And, the status now of the Code case, the class 1 Code case N-451 is published and will be treated in the next revision, revision 26. That is R.G. guide 1.84, so that will go through the formal NRC endorsement process for that Code case. The other Code case will probably be in the next revision to R.G. 1.84.

[Slide]

However, the changes we are talking about today will be to the Code itself, and the way the NRC endorses the ASME Code changes is through 10 CFR 50.55A and we essentially incorporate specific addenda and revisions to the Code as they come about.

CHAIRMAN SIESS: And, that again, you do periodically?

MR. GUZY: Do that periodically, and it will be done through the regulations, so that is the formal process for endorsing the changes we will be hearing about today.

Also, there are changes that we may make to the standard review plan, particularly in its functionality area, also I think the information from this program will have an impact on many other things we do in the piping area; perhaps not in a--as a more explicit way for providing backgrounds, supports a lot of conclusions people have made, for instance in seismic margin studies, or PRA's, seismic inertia loads are generally not considered important if we don't review them.

In contrast to the way that we may have reviewed plants in the past--

CHAIRMAN SIESS: Wait a minute.

You said in seismic margins they are not considered important?

MR. GUZY: --not considered important. Piping inertial loads are generally not even--in piping systems--

are not considered important because of the experience, the piping experience--

CHAIRMAN SIESS: Oh, okay, simply based on piping experience and--

MR. GUZY: --because of the piping experience, plus findings from this kind of a program. This supports piping experience data--

CHAIRMAN SIESS: Oh, okay, yes.

MR. GUZY: --showing that your margins are much greater than other things in the plant that will--

CHAIRMAN SIESS: Yes, okay, but now you just mentioned fragility in passing, and that is a big area of business these days, making PRAs, and certainly there somewhere they have got to put a fragility in there, don't that?

MR. GUZY: They--

CHAIRMAN SIESS: Or, do they just--

MR. GUZY: --they do in the piping area, but they only do in at very high level earthquakes. Sometimes you can dismiss that without developing it in the piping.

CHAIRMAN SIESS: Okay.

MR. GUZY: Seismic margin areas, I was going to contrast to--in the old SEP program you had to reevaluate everything, a lot of the effort was involved with piping analysis. In the seismic margin approach now has taken from the SEP program, although they recognize that all of this piping reanalysis is not necessary. They are concentrating on things of importance such as the seismic ankle motions or systems interactions, and not worry so much about seismic inertial loads.

This program supports other data and experience showing that we can, in the global formal safety scenario we can downplay seismic inertial loads.

CHAIRMAN SIESS: That explains why--

MR. WARD: Well, it is--

CHAIRMAN SIESS: --couldn't find any failures.

MR. GUZY: That's right.

MR. WARD: --but a lot of that has been taken credit for already though--

CHAIRMAN SIESS: Yes, that is what he is saying.

MR. GUZY: We are supporting--

MR. WARD: --and this is confirmatory, actually.

[Slide]

MR. GUZY: Okay, I have one other slide in my package on piping resource that I would like to hold on for later.

CHAIRMAN SIESS: Okay.

MR. GUZY: I would like to, at this point, take any questions.

Yes.

MR. SHEWMON: I am very pleased, as a metallurgist who makes his living sort of on the fact that metals are different from glass, seeing these things designed to take credit for some of the plasticity that is inherent in the metal that we've paid for and built it out of, but, as you do this I am some concerned about the fact that there is no consideration of castings and the fact that they might have different plastic properties than the wrought material and the test consisted only of wrought material.

Is there a basis on this that it was just so much more convenient to work with wrought material? Or the Code says that castings always must have appreciably lower stresses? Or that the Code has been able to ignore it because they do elastically and the plastic properties don't enter, or what?

MR. GUZY: I think--maybe somebody else would like to speak to this, but--

MR. SHEWMON: I ask you, but you can pass it off. That is your advantage to--

MR. GUZY: --it is my understanding--okay, but my understanding is that, you know, we are looking for the majority of the piping in the plant, and the majority of the piping, to my understanding, does not use cast fittings.

MR. SHEWMON: Well, but it is not just the piping. It is the components in the elbows--

MR. GUZY: That is what I was talking about.

MR. SHEWMON: --and the valve bodies and the pump housings, these things often are, and the fact that if you design your plants with the strongest elements, it is the weak link that is going to rise up and bite you, so saying the majority of it is piping, probably isn't the right way to look out for failures.

MR. GUZY: Maybe somebody else can address that, but it is my understanding that the majority of the fittings, elbows, and tees, et cetera, were wrought and not cast.

MR. SHEWMON: There is a lot of cast that comes out there. I don't know whether the majority of it is or not, but if you change the Code--

MR. GUZY: The Code will not address that at this time, because we don't have the data right now to do--

CHAIRMAN SIESS: The Code will be limited to wrought material?

MR. GUZY: Wrought material, yes.

MR. SHEWMON: So the Code will distinguish between wrought and cast--

MR. GUZY: Yes.

MR. SHEWMON: --in this case?

MR. GUZY: That's right.

MR. BUSH: Are you sure of that? I don't think that. I am not quite sure of that.

MR. GUZY: Well, maybe later on we will talk about this, but our data, since we have not tested, particularly the ratcheting specimens, we have to limit our recommendations now to things we've tested or somehow address this later.

CHAIRMAN SIESS: Can you use the MCL type material test to make a bridge between the material properties and the observed behavior of components and systems?

MR. GUZY: Yes.

MR. TAGART: We have not, at this point done that.

We had a long discussion on this point at our last review meeting, about potential restrictions of the first round of rules that we are going to recommend for the Code. We do expect to have caveats, restrictions, whatever you want to call them, with respect to the application of these rules to the materials.

We have not completed the process of identifying exactly which materials that will not be specifically included; however, comments like the ones that you are providing will be helpful in our--as an input to that consideration.

CHAIRMAN SIESS: Well, there are two ways to limit materials: one is by naming them; and the other is by tying it to properties, measurable properties, and I gather from what you say it will be by name rather than by the other.

MR. TAGART: Yes.

We would plan to do it by name, because we have identified four materials in the materials testing work, and one of the primary purposes of the materials test is to bridge the gap between room temperature tests and elevated temperature tests.

None of the tests that you saw in the last two days were run at elevated temperatures, and the materials tests were intended to bridge the gap between room temperature tests and the elevated temperature tests, for the important failure phenomenon that occurs in the materials, fatigue ratchet.

MR. GUZY: Anything further?

CHAIRMAN SIESS: Any other questions?

[No response.]

No, let's proceed.

MR. TAGART: Good morning, ladies and gentlemen. I welcome this chance to provide the introduction and overview of this program.

The things that I am going to talk about for the next few minutes involve a brief history of the Code rules relative to piping. I am going to summarize what we believed we knew in 1985 when we started this program. I would like to take a few minutes to explain a very simply way of understanding why piping is so resistant to dynamic and seismic type loadings, and I would like to summarize what we know now near the completion of the program.

Also, I would like to discuss the challenges and the opportunities that we think are available as the result of this program.

[Slide]

This is not a complete history of piping, but I wanted to highlight some of the things that I think--what we've done in perspective.

The earliest date here is 1952 when the Markl fatigue tests were introduced into the B31.1 Code. The basis of these tests were semi-static, that is they were slowly fatigued to determine when various kinds of piping components would fail in a leaking manner. They became the basis for the detailed rules and the B31.1 Code.

In 1963 the nuclear pressure vessel rules were introduced into the SME Code where static and fatigue type loads were considered. The Markl work was the forerunner of the static treatment and the low-cycle heat treatment of loads for pressure vessels.

In 1968 nuclear piping rules were introduced into the ASME Code, which involved static, dynamic, and fatigue loads.

To put a footnote on dynamic--and I want to emphasize that the basis for the dynamic loading was that the effects of dynamic loads were handled by static failure criteria; that is, it was recognized that there would be dynamic loads, but the criteria for failure under those loads would be the same as if those loads had been applied staticly. That was a simplification, made at the time. If we had had the results of this research available then we would have done that differently.

In approximately 1975, Japanese research, which was aimed at confirming whether--particularly whether the D-level stress levels in the ASME Code, which goes somewhat beyond the elastic limit, were acceptable and safe. In the process of evaluating these D-levels stress limits, they identified large dynamic margins; however, their focus was not to find out exactly how large they were but simply to establish whether the ASME rules for level-D were acceptable and safe.

They also identified fatigue ratcheting, the swelling of the pipe, as an important part of the failure mode.

In 1982, a PVRC program under the leadership of Spencer Bush was initiated to improve nuclear piping, and one of the major things that came out of that was the Code case N-411, which allowed people to begin removing snubbers.

In 1985--Dan has already mentioned--the 1061 piping recommendations, and simultaneously with that was the beginning of this program.

Now we are about to complete these dynamic tests and we will have a basis for new rules for the ASME Code.

[Slide]

To summarize what we knew in 1985:

No. 1. We knew that dynamic margins were large, but we were uncertain as to exactly how large they were, so the emphasis of these tests was to take them all the way to failure, and there was considerable thought and effort put into selecting test facilities that would produce failures in a relatively few number of load applications, and our target was no more than five time history earthquakes being applied to the specimen to produce failure; and, at the same time we were planning to use more or less normal pressure loads. Pressure loading was not exaggerated, only the dynamic loading.

No. 2. The fatigue failure mode for reversed dynamic loading is ratcheting and fatigue, not static collapse. We knew this from the Japanese research.

MR. SHEWMON: Is static collapse what Chet would call "net section collapse" sometimes? Or, what is static collapse?

MR. TAGART: Static collapse--

CHAIRMAN SIESS: It is collapse.

MR. TAGART: --is the--

MR. SHEWMON: It is not failure. It is the walls coming together?

MR. TAGART: No.

What I mean by static collapse here is that if you plot the load deformation behavior of the structure, the deformation starts becoming large with small increases in the load. It is well beyond the elastic--

CHAIRMAN SIESS: The curve you showed--somebody showed yesterday--went down versus the one that went up.

MR. SHEWMON: Went up, yes.

MR. TAGART: The ASME Code has some definitions of collapse that suggest collapse occurs before you start going down, so it depends on whose terms you are using as to what it means. I think it means something slightly different to civil engineers.

CHAIRMAN SIESS: Would that be different if you left static out?

MR. TAGART: It think it is not quite as clear without the word "static" and I am suggesting static collapse as a term used in this program to distinguish it from an incremental collapse. Incremental collapse, as you saw in the films yesterday, could involve step-wise collapse of the structure. We want to distinguish static collapse from a one-application of load to cause collapse, between that which occurs with many applications of load where it moves slowly.

CHAIRMAN SIESS: I am still trying to understand.

Frequently we take repeated loadings and find that they can be enveloped by a single monatonic static load.

Are you saying that the static collapse would be that monatonic loading, and that the dynamic collapse that you are talking about would not be enveloped by that? Do you visualize what I am talking about? You know, I will draw you a curve, and it will look like this--[drawing curve in the air]--they look like this, a static curve, and a monatonic loading would be right in the upper bound of it, and this is in certain types of things, not piping necessarily, but things that I know about. Is that a distinction you are making?

MR. TAGART: It is very hard to make a general distinction in that way; for instance, in our materials tests we clearly see what you are talking about. We plot on a diagram what happens to the reverse loading, and at the same time we can look what the comparative material behavior is of a uniaxial specimen just being pulled. There one can see a relationship between a incremental collapse and a single collapse.

CHAIRMAN SIESS: Are you saying that you do not get that kind of relationship in piping?

MR. TAGART: Well, we see it, yes, but there is a complication in piping, because piping is a fairly complicated structure, even for example, understanding an elbow--

CHAIRMAN SIESS: Yes.

MR. TAGART: --the elbow is one of the more difficult things to understand because it is a complex structure and each material in it behaves in a certain way differently.

CHAIRMAN SIESS: But, I couldn't envelop the dynamic incremental collapse with the static collapse curve?

MR. TAGART: I don't know.

CHAIRMAN SIESS: It would be wonderful if you could!

MR. RODABAUGH: Isn't the answer "yes," Sam?

CHAIRMAN SIESS: I don't think you can.

MR. RODABAUGH: The answer is "yes" because that is a much bigger envelop than--

CHAIRMAN SIESS: No. I mean match it.

MR. BUSH: No.

CHAIRMAN SIESS: I mean, if I draw the envelop for the ratcheting of the incremental collapse that that would agree. I think it would fall well below the static collapse.

MR. TAGART: It is not identical, though, and I think the behavior of the structure--it is not even identical for the materials test.

CHAIRMAN SIESS: Okay, then I think I understand what you are saying there.

MR. TAGART: All right.

Now, the third point here is a conclusion that we reached in 1985 as a result of some preliminary thinking about this program. We felt quite strongly that fatigue ratcheting was the mode of failure, but we didn't feel that that was well known in the industry, and we felt that we could support that kind of conclusion analytically but we felt such a demonstration would not be convincing, and we concluded that experimental evidence, plus an engineering understanding of those experiments would be necessary to effect a change at this point in time, and of course, the observation that many people came to was that nuclear plants have too many snubbers.

Now, I would like to take a few minutes to describe a simple explanation of why piping is so resistant to dynamic loads. I am going to talk about a picture that appears in that fourth simi-annual report, and I'll show you what the picture looks like first, and then go back to this diagram.

It is this page in the fourth semi-annual on page 3-208, where we are describing an amplification versus frequency. The standard kind of thing that appears in textbooks on force vibration analysis.

But, before I discuss that I want to discuss the assumptions here that go into one of the curves. This diagram is the force on a single degree of freedom spring versus the deflection of the single degree of freedom spring.

So, this is the deflection--[referring to the drawing]--this is the stiffness times the deflection, or the force. And, this is the simplest model showing complete plasticity, assuming that there is some value at which the structure become elastoplastic. At this point it unloads, become elastic again, goes into reverse plasticity and absorbs energy through this loop.

The component tests are very close to a single degree of freedom system. The major complication that could be added to better understand it would be to put a slope on this curve right here, and to make it elastic and then strain hardening in both areas. But, for the moment, let's look at perfectly plastic.

One of the simplest models that we can use, which is an approximate dynamic analysis--

CHAIRMAN SIESS: That has a mean stress on it, obviously?

MR. TAGART: --no.

CHAIRMAN SIESS: Then it is not at zero.

MR. TAGART: This is adjusted so that the diagram centers around this point here, but there is no mean stress in this. Sam Ranganath will talk a little later about what happens when we add mean stress to this.

MR. SHEWMON: This is an A cycle taken after you have reached a steady state.

MR. TAGART: This is a steady state behavior with no mean stress.

CHAIRMAN SIESS: Okay.

MR. TAGART: Now, then we--

CHAIRMAN SIESS: I guess that I am still having a problem.

How do you get to this? You have to start at zero when you go up on that slope and get plastic, and then it settles down to this loop then?

MR. TAGART: Okay, let me describe the problem--

CHAIRMAN SIESS: This is the Nth cycle.

MR. TAGART: This is sinusodial exicitation of a single degree of freedom system after it gets through its transient. It settles into some steady state.

CHAIRMAN SIESS: Okay.

MR. TAGART: And, it is driven hard enough that it becomes elastoplastic.

CHAIRMAN SIESS: In both directions?

MR. TAGART: In both directions, right.

CHAIRMAN SIESS: Do your component tests become plastic in both directions?

MR. TAGART: Yes.

CHAIRMAN SIESS: Then I don't get a ratchet out of it, do I?

MR. TAGART: In this case, no.

CHAIRMAN SIESS: Okay.

MR. TAGART: If you add a mean stress, you will.

CHAIRMAN SIESS: Okay.

MR. TAGART: In this model, there are two things done to the simple equation of motion: one is to approximate the damping by the energy absorbed in this loop.

The second is to put a reduced stiffness in the single degree of freedom, which is the slope of this line, and this makes the solution nonlinear; that is, you now don't know the deflection before hand, and the equation that will solve this for a sinusodial motion has to be solved by some trial and error technique.

It is not an exact solution to the problem because the motion is not sinusodial, purely sinusodial when it goes to the elastic plastic, but this makes a very simple explanation of what goes on, so that is the assumption that goes into the elastic plastic model, and the diagram that is in the report shows the results.

The solid curve is a curve at two percent damping--

CHAIRMAN SIESS: I am having a problem.

That is a response spectrum, in effect?

MR. TAGART: Yes.

It is a steady state response spectrum for a single degree of freedom system--

CHAIRMAN SIESS: And, it is the amplification of what?

MR. TAGART: It is the amplification of mass relative to the ground.

CHAIRMAN SIESS: Its acceleration?

MR. TAGART: It is the--either the displacement or acceleration. The assumption here--

CHAIRMAN SIESS: What's "A"? It says C over A in there?

MR. TAGART: Okay.

"A" is the amplitude of the input motion. See in the box there--

CHAIRMAN SIESS: Okay, okay--

MR. TAGART: --"A" is the amplitude of the sinusodial displacement.

CHAIRMAN SIESS: --I've got it. All right. I've got it now. That is a displacement versus--

MR. TAGART: Right, it is a displacement response.

CHAIRMAN SIESS: It is a displacement response, yes.

MR. TAGART: Right.

This curve is for the two percent damping case if it were elastic.

The assumption for the picture that I just described: it is labeled "Tagert's Model Lambda=Zero" meaning no strain hardening, is this dotted curve right here, and what it tells us is that we get a frequency shifting from this point back to this point, the softening effect, and an enormous lowering of the peak, as you know. This peak goes way up here, so you get an enormous reduction.

This particular case here is pictured for five times the yield stress--

CHAIRMAN SIESS: Wait a minute, wait a minute. Five times the yield stress? Or yield strength?

MR. TAGART: Yield strength, yield strength.

CHAIRMAN SIESS: Okay.

MR. TAGART: I'm sorry.

CHAIRMAN SIESS: Does that frequency shift correspond to that dotted line you had on the previous figure?

MR. TAGART: Yes.

CHAIRMAN SIESS: Okay.

MR. TAGART: An exact solution of this same problem, not an approximation, is also shown on the diagram by what's called the numerical solution for Lambda to equal zero. It is this curve right here.

Now, that solution is conservative compared to the exact solution; however, when you put a little strain hardening into it, in the exact solution it brackets the approximation, so the approximation is very close to the reality of what goes on in this single degree of freedom test, and here we see the effect of the energy absorption which completely chops off this high resident peak, and it shifts the response to the left on this diagram.

Another very interesting thing that one can observe from this diagram--which we did not strongly observe in any of our tests--is that there is a region in here where elastic analysis will underpredict the response regardless of what damping you put in it, and that's one of the reasons why--

CHAIRMAN SIESS: It is the frequency shift.

MR. TAGART: --yes, it is because of the frequency shift. That is one of the reasons why we were less enamored with the idea of making changes by controlling just the damping. The damping certainly has an important effect because it chops off this peak, but the frequency effect is also more important, and we think probably the easier way, the more straight forward way, is to handle both of these effects in linear analysis by changing the allowable stress, rather than by trying to deal with the damping.

MR. BUSH: Sam, how do you handle the strain softening aspects, or do you simply--

CHAIRMAN SIESS: A little louder, Spence.

MR. BUSH: I was asking how you handle the strain softening aspect, or do you simply cut it off at Lambda equals zero?

MR. TAGART: The lambda equal zero model is a--do you mean if the slope were actually to go down rather than up?

Well, strain softening, there are two ways to think about strain softening. There is strain softening in a single cycle, or there is strain softening where successive cycles may have lower stress range than they had in the earlier cycles.

We've addressed that by our materials tests. We selected four different materials in the materials test: one to be strain hardening; one to be strain softening; and two to be more or less neutral.

We are concerned about the problem of predicting the ratchet. I think the subject of strain hardening and softening is most important in the area of how much ratcheting will occur in each cycle? And, I think Sam Ranganath is going to cover this in a little bit of detail to explain what conclusions we've drawn to date on the ratcheting. I hadn't planned to talk about that at this part of the discussion.

CHAIRMAN SIESS: Fine.

MR. TAGART: What I think is useful about this diagram is that there is a simple physical way to explain why under the steady state response--and I think you saw yesterday in these experiments--that you reach some kind of a steady state. Now, most of those tests were seismic inputs. We only had a test where we've done sinusodial inputs, and we will be able to more directly compare the applicability of this model to those sinusodial inputs, but it is a relatively easy thing for us to put a non-sinusodial input into this single degree of freedom model and get comparisons, and we have done that, so we've made a lot of progress to understanding those compliment tests by simply looking at a single degree of freedom elastoplastic model. That is the point that I wanted to get across with this diagram.

[Slide]

I would like to give an overview of what we think we know now, as the result of this program. We believe that we know why static collapse does not occur-- it does not generally occur, in the dynamic loading situation. The previous diagram is an elastoplastic system. If any system could collapse as a single degree of freedom system, that one would collapse and it generally does not, and there is a good explanation as to why it doesn't.

We know also that there are certain types of dynamic loads that can collapse the piping. As we saw yesterday, we were able to produce large deformations in the water hammer cases where the load holds up long enough to allow the pipe to mold, so we don't want over generalizing results. Our objective is to make Code changes but not to overstate the case relative to certain kinds of dynamic loads.

CHAIRMAN SIESS: So, the type that can cause it is one that isn't too dynamic?

MR. TAGART: That's right. It behaves more like a static load.

We know how to approximately predict the component results from first principles. I haven't discussed the ratcheting, but that will be discussed.

We know that there are some limitations of linear dynamic analysis, that previous diagram showed the key area where there may be some concern about that.

We have clarified concepts of apparent damping. I haven't discussed that much, but if we look at the damping that is available, the equivalent damping that is available in that single degree of freedom system, it is very, very large, on the order of 40, 50 percent is available in that single degree of freedom system, with sinusodial inputs. But, we need to distinguish between the what we call--Henry Hwang has termed true damping, and apparent damping--and this word should be true--[referring to the slide]--I am sorry for the error here, instead of "time" this should be true damping.

Piping systems are fundamentally resistant to seismic and other dynamic loads because the true damping is very high at ductility of as low as three, and as a matter of fact, if you will look at that little degree of freedom model, which is not an exact solution, it actually maximizes the damping at a value of three.

MR. SHEWMON: Can you tell me what a dynamic ductility of three means to a stress strain curve?

MR. TAGART: It means if the single degree of freedom system were a mass hung on a tensile bar, then the yielding of the structure and the yield of the material would be the same, and if we had a elastoplastic material and the deformation which occurred was three times the collapse load, that is what I mean by ductility of three.

CHAIRMAN SIESS: Not three times the yield?

MR. TAGART: Well, in that case it would be the same.

CHAIRMAN SIESS: That is--

MR. SHEWMON: This is three times the strain of the yield; is that right?

MR. TAGART: Well, this is three times the deformation at which the structure collapses.

CHAIRMAN SIESS: On this curve, isn't it the ratio of that distance to that?

MR. TAGART: Yes, yes.

CHAIRMAN SIESS: The ratio of--

MR. SHEWMON: Thank you.

MR. TAGART: We believe that we understand ratcheting. Ratcheting very simply is a lack of symmetry through the cycle, and the presence of a mean load means that when you have half of a cycle, a mean load is adding to one direction of plasticity and in the opposite direction it may be subtracting, and therefore there is a net accumulated plastic strain, or deformation, when one completes a cycle in the presence of mean loading.

CHAIRMAN SIESS: Now, you talk about mean loading--

MR. TAGART: Mean stress.

CHAIRMAN SIESS: --it is mean stress--

MR. TAGART: Right.

CHAIRMAN SIESS: Because that mean stress can be a pressure induced stress--

MR. TAGART: Yes.

CHAIRMAN SIESS: It doesn't have to be a load induced stress.

MR. TAGART: That's right.

CHAIRMAN SIESS: Okay.

MR. TAGART: All right.

Typically, in the--and most of the applications we are talking about pressures is the dominant behavior, although weight is also an important consideration, and we've seen this in our experiments. The influence of the weight is a very strong effect in how much ratcheting will occur.

CHAIRMAN SIESS: You mean simply the weight producing a longitudinal stress in the pipe, or whatever you test?

MR. TAGART: Yes, yes.

CHAIRMAN SIESS: Just gravity.

MR. TAGART: Yes.

[Slide]

I would like to spend a couple of minutes on the opportunities and challenges that present themselves as the result of completing this research. We will have a significant Code margin reduction proposed, as a result of this program, that's not necessarily going to be easy. As I have discussed this with Dan Guzy, he tells me we shouldn't expect that this is going to happen one week after we make our proposal. It--I almost hesitate to say this--but it may take a year or more for these results to get into the Code, and some of the things that we see that the regulator will have to look at is managing the prior and future Code changes.

A couple of important things, the Code case N-411 which got us going relative to snubber reduction--

CHAIRMAN SIESS: What is 411?

MR. TAGART: Code case N-411 is the one that increased the damping to five percent--

CHAIRMAN SIESS: Okay.

MR. TAGART: --at low frequency, and two percent at high frequency.

CHAIRMAN SIESS: Okay.

MR. TAGART: The Code case N-451 which was just recently passed, which took the operating basis earthquake out of equation--yes.

MR. SHEWMON: Would you put some words on significant Code margin reduction? Would that come later?

MR. TAGART: Okay, it will come later, but I think it is a good point to bring it up now.

We are thinking of effectively increasing the allowable stress in the Code somewhere in the order of 50 to 100 percent, and we haven't decided exactly how much that should be right now.

MR. SHEWMON: Double?

MR. TAGART: Up to doubling it, at least 50 percent higher than--

MR. SHEWMON: That is the seismic component of the stress? Or the Code?

MR. TAGART: The dynamic--the total stress equation that covers the combination of pressure stress and the inertia stress from dynamic loading that is of a reverse type, either seismic or reversed other type that behaves like seismic.

CHAIRMAN SIESS: But, if you are going to increase the total allowable, how are you going to take care of the range and ratio between seismic and other stress? An element that has very little seismic stress, do you have some other equation that governs that?

MR. TAGART: Yes.

CHAIRMAN SIESS: Okay.

MR. TAGART: There are other equations that--

CHAIRMAN SIESS: This would be the only equation that includes the seismic stress?

MR. TAGART: Right.

CHAIRMAN SIESS: Load combination--

MR. TAGART: Seismic or other dynamic stress.

CHAIRMAN SIESS: All right, I got it.

MR. TAGART: One of the activities that is going on is handling the independent support motion with SRSS and it is currently the subject of research.

We have simplified static methods that are now being seriously considered by the Code. We have non-linear methods which we know are going to be coming along, seismic anchor motion modifications, and I mention here possibly designed by rules. We believe that the results of this research program are conducive to eventually producing design by rules in piping, as opposed to design by analysis, particularly for the seismic effects. We are not recommending that at this time, but we think it is a fruitful potential improvement in the future, and our approach, or our recommendation will be to work with the current rules that are designed by analysis, make those changes which are appropriate as the result of those rules, and at some point in the future explore and examine the possibility of great simplification in the piping design process for nuclear plants. That is for the future.

I would like to show you something that addresses a method to optimize piping design. I think that is one other real opportunity here. In the past the approach to making changes to the Code has involved trading one conservatism against another, and that's a viable approach, but it doesn't allow us to optimize the piping system, and I would like to show you an approach that we have examined at EPRI in the last year and what I've shown here is a probabilistic approach to optimizing piping design, and this is what decision analysts call "an influence diagram."

[slide]

A circle on this diagram represents something that we are uncertain about. A square represents something that we have control over, that we can made a decision about. So, at the top of the diagram we focused on snubber reduction efforts.

What we did in this program was attempted to say if we temporarily removed all requirements for piping design, and we could trade off the pluses and minuses relative to how many snubbers we would put into a nuclear power plant, how many would we put in. We had complete freedom to do it, and we could make the decision on the basis of cost, safety, or both--

CHAIRMAN SIESS: Knowing what you know now.

MR. TAGART: --knowing what we know now, yes.

CHAIRMAN SIESS: The early plants didn't have any--

MR. TAGART: Yes.

CHAIRMAN SIESS: --knowing what they did then.

MR. TAGART: Right.

This model is an ambitious model, and it was done by one of our contractors with the help of a second one who was very familiar with probabilistic risk analysis, so we have things in here about the core melt. How does the pipe failure influence the core melt? And, here you see in this original diagram, we had things like water hammer, the seismic loading--

CHAIRMAN SIESS: Sam.

MR. TAGART: Yes.

CHAIRMAN SIESS: What's the significance of the two circles with a lot of arrows out of them that don't do anywhere?

MR. TAGART: It means they connect to a lot of other ones, and--

CHAIRMAN SIESS: Okay.

MR. TAGART: --for instance--

CHAIRMAN SIESS: And, you don't know which ones.

MR. TAGART: We could connect them to almost all of them.

CHAIRMAN SIESS: Okay.

MR. TAGART: It would make the diagram so confusing to put all of those arrows in there, we are saying the design and construction errors can--are pervasive through the whole diagram. You can have things go wrong, with what you know about anything going on in this diagram.

Similarly, we have this influence of NRC regulations on very many parts of this diagram, in other words there are regulations that effect how each one of these things are done, and we don't know how those regulations are going to change in the future.

The study that we did simplified this diagram a little bit, and although it would take too much time to discuss the complete implications of this study, I would like to show you some of the results.

CHAIRMAN SIESS: Sam, it seems to me that if you are going to do a true optimization you have got to know almost everything about everything.

MR. TAGART: That's right.

CHAIRMAN SIESS: It is obvious that you don't.

MR. TAGART: That's right. You are very uncertain. That is why these are put in circles.

CHAIRMAN SIESS: Yes, so it has to be a probabilistic optimization?

MR. TAGART: That's correct.

And, what we attempted to do here is a picture as close to reality as we know it today. We recognize that some parts of it are very uncertain, but we want to give the best expected values for each one of the variables in the diagram.

CHAIRMAN SIESS: Okay.

MR. TAGART: The results of that study, cn a very simple system, extrapolated to an entire plant, and the costs of an entire plant is a diagram that looks like this.

[Slide]

This is where we stand right now, and typically this might be a plant with, let's say, a thousand snubbers. We are plotting here the number of snubbers removed, versus expected lifetime costs of the plant.

CHAIRMAN SIESS: Percentage.

MR. TAGART: And, for different costs involved in maintaining snubbers, we have different curves.

This is a very high cost snubber, $7000 per snubber for the lifetime, annual maintenance cost for the snubbers.

This is a considerably lower cost, and this is a very low cost.

And, what you see here, of course, as you can well imagine--and we have put the safety costs in this picture, as well. You will notice down here, we said if we get a core melt there is some large cost associated with that core melt. For example, in this case, $20 billion, and in this case $5 billion, and of course it changes the picture.

So, this is one diagram that puts the safety picture and the cost picture on the same diagram. It shows, for example, that we should be, regardless of what the cost of snubbers are, we should be moving in the direction that we are moving, and that is to remove snubbers.

If the cost is very, very low, we see a diagram that reaches some minimum point here and then starts coming back up again. If the cost is very high, and we were optimizing on cost, it would come down.

[Slide]

Now, I would like to show you a diagram that shows what happens if you forget the cost, and simply optimize on the safety question. Here is a diagram that says the lifetime probability of core melt due to pipe failure, and here is where we stand right now with a large number of snubbers in the plant. It says that--and or two probabilities or earthquake--

CHAIRMAN SIESS: What earthquake is that, that you are putting the probability on?

MR. TAGART: This is the probability of the SSE.

CHAIRMAN SIESS: Well, that won't cause any damage--well, I guess it is some probability.

MR. TAGART: Okay.

The reason that it causes some problem is that this model says I could have degraded piping. I could have piping with cracks in it before the earthquake comes.

CHAIRMAN SIESS: Well, now, suppose you did, and a three SSE at 10 to the minus 5--two SSE at 10 to the minus 5, what would it look like?

MR. TAGART: Well, it would change the picture and I don't want to speculate about how it would change it.

I want to tell you about the results that we have.

CHAIRMAN SIESS: What pipe failure frequency are you taking for that earthquake? You said degraded piping, so you must be some way putting in--

MR. TAGART: We have a very crude model for the degraded piping.

What we have done, essentially, is look at the piping as if it can fatigue all the way to a failure point, with a very simple model. We have a very simple fracture mechanic's model. The input is not the failure rate of the piping. The input is part of the equation to describe the lifetime of the piping as a function of the thermal loads, the seismic loads, and how big a crack it has.

CHAIRMAN SIESS: And, you are assuming that the earthquake is 1/22nd cycle like you have? No after shocks?

MR. TAGART: We are not into that level of detail.

CHAIRMAN SIESS: But, you are assuming some kind of an earthquake, at so many cycles, and on the basis of what you know that that combined with other things will cause fractures.

MR. TAGART: Yes, yes.

CHAIRMAN SIESS: Okay.

MR. TAGART: So this model admits the possibility that the earthquake--and in fact a very crude assumption was made here. For example, we said if a pipe can get to a failure point by leakage, that 15 percent of the cases would break before leak.

CHAIRMAN SIESS: Okay.

MR. TAGART: Here again, we see that the right direction to move is to remove snubbers, and these curves turn up sharply, which was a bit of a surprise when we first saw that. We thought that perhaps the diagram would look like this, and come to a point which was below this point so that if we removed all of the snubbers we'd be better off than if we kept some in.

This is because--and it is dotted here, and I would emphasize that this is very tentative in nature--it is because of the degraded pipe question, and we have another program, the IPERC program which is coming along to examine degraded pipe. We focused on sound pipe. The degraded pipe program tells us--answers questions about the uncertainty of unsound piping, and I believe the long term optimization of piping depends on our looking at both of these programs.

I think it is clear at this point that very large numbers of reduction will both improve the costs and improve the safety.

CHAIRMAN SIESS: It would seem that--let's take not the 80 percent point, but your 90 percent point--it would seem to me that it would make a difference as to which snubbers were included in that 10 percent window?

MR. TAGART: That's true, and what we are assuming here is that we take out the right ones, too. We are taking them out in some systematic way--

CHAIRMAN SIESS: So, the 20--

MR. TAGART: --so we take the right ones.

CHAIRMAN SIESS: --percent you are leaving in are some of the most crucial ones--

MR. TAGART: Yes.

CHAIRMAN SIESS: --and when you start taking those out the risk goes up.

MR. TAGART: Right.

CHAIRMAN SIESS: Okay.

MR. TAGART: And, if one wanted to think about further optimizing it, some of the passive restraint devices could be substituted for those few that really do us some good.

MR. RODABAUGH: Sam, what is it about--is it possibility of snubber malfunction that makes this curve go down?

MR. TAGART: Yes, that's it, exactly.

The snubbers themselves can malfunction, they increase the stress, and that is why these curves go down there.

MR. BUSH: And, that depends on the types of snubbers.

MR. TAGART: Yes.

MR. WARD: Then are these curves--let's see, the last figure, are those consistent with the previous figure?

MR. TAGART: Yes.

MR. WARD: I mean it is the same?

MR. TAGART: They are the same.

MR. WARD: So the break off is what you see on the previous one?

MR. TAGART: Exactly, right.

MR. WARD: What would that previous one look like with, if you are talking about a new plant capital costs of snubbers? The same?

MR. TAGART: Which curve would we pick?

MR. WARD: Yeah.

MR. TAGART: It probably is this middle one for the cost is more like what we would expect.

MR. BUSH: Sam, a question on that curve.

I don't know what the maintenance cost are that it has, because one of the problems, obviously, with the snubbers is the effect on outage time.

MR. TAGART: Yes.

MR. BUSH: Which has a, spread over several years, it doesn't take many days of outage time to increase to bias the costs considerably.

MR. TAGART: Yes. I think we have not considered a lot of outage time in this, and if outage time got to be a big factor, these number of course would move more in this direction.

CHAIRMAN SIESS: This is sort of routine maintenance, then.

MR. BUSH: You are talking about $100 million roughly, as a zero baseline on that conservative model.

MR. TAGART: Yes.

CHAIRMAN SIESS: And the snubbers that you can't test, I assume you are leaving in?

MR. TAGART: Pardon me?

CHAIRMAN SIESS: The big snubbers that you can't test, I assume those are some that you are leaving in?

MR. TAGART: Well, here we are talking about piping.

Some of those big snubbers are on--

CHAIRMAN SIESS: Oh, that's right--

MR. TAGART: --peak generators.

CHAIRMAN SIESS: --they are on peak generators.

MR. TAGART: With that, I would like to--I guess we are at the break?

CHAIRMAN SIESS: Yes.

MR. TAGART: Anymore questions?

[No response.]

CHAIRMAN SIESS: Any questions for Sam before we take a break?

[No response.]

Okay, let's take about 10 to 15 minutes for a break, and the audio systems man may be able to fix the microphones.

[Recess: 9:50 a.m. to 10:10 a.m.]

CHAIRMAN SIESS: All right, you may proceed.

MR. ENGLISH: My name is Bill English. I am from General Electric.

I would like to take just a couple of minutes to discuss with you the portion of the program that I will be talking about today, in a few moments, to describe some of the objectives that we focused on through the program, but the bulk of the time I would like to spend talking about the component test, system test, specimen, and analysis avoidance test.

CHAIRMAN SIESS: Please keep in mind that we have had a pretty good description of the tests themselves.

MR. ENGLISH: Right, right, okay. I will try to go through those portions of the presentation very fast, and if I am going too slow, you just speed me up.

CHAIRMAN SIESS: All right.

Well, I may ask you to just skip some of these, since we've seen them.

MR. ENGLISH: Okay.

Today I will be talking about the component tests of Anco, the system tests at ETEC and Anco, the specimen fatigue ratcheting tests at MCL--actually started at the General Electric turbine technology lab with Sumu Ykowa [sic.] and Roy Williams, and GE got out of the materials testing business at that particular location, and the lab was transferred over to--Roy Williams actually bought the equipment and continued the tests, so we didn't lose much time on part of the program.

And, then task 5 was the analysis of test and design rules, which was done with GE in San Jose.

[Slide]

As you've probably heard a number of times, a major objective of the program was to try to take advantage of current dynamic margins that we've seen in metal piping systems, and to devise new, more realistic ASME Code rules, but in addition to that we wanted to determine what the actual failure mechanism was in piping. It is not believed to collapse at the same point now as fatigue ratcheting.

We wanted to measure pipe damping as a function of different stress levels, with the system configurations and frequency inputs. We wanted to determine how big an earthquake the piping systems could tolerate without failure.

[Slide]

We would like to develop a lab specimen to quantitatively predict fatigue ratcheting, and ultimately plan to suggest changes to the standard review plans, regulations and codes to account for the margin that we have in piping of the dynamic loading, and ultimately we would like to be able to simplify the piping dynamic analysis.

[Slide]

We felt the focus of the testing program was really on the component tests. The most severe loading was in the component tests. It had the most instrumentation, 28 to 30 to 32 channels of instrumentation, as compared to only 80 channels on the system. The component behavior for a number of different components could be demonstrated, and we could determine what the actual failure modes were, show that functionality was not compromised. We could use the component test results to predict the system test behavior and we could calibrate the design rules.

The main function of the system tests then was to confirm what we've learned in the component tests, to confirm that a single component doesn't actually collapse, the load redestributes, that the mode of failure is not collapse, that it is fatigue ratcheting, some kind of incremental kind of form of failure. It confirms the functionality of the piping system, that the pipes actually get bigger in diameter rather than smaller, and tend not to restrict the flow. It helps to design rules and margins and it provides a lot of benchmark analysis for benchmarking some of the computer programs used in piping analysis.

The specimen tests, on the other hand, are a very simple method of demonstrating ratcheting, enabling us to evaluate many different materials at minimal costs as compared to the component systems tests, and enables us to determine the effects of temperature.

MR. WARD: Bill, why aren't those--I mean Paul Shewmon asked earlier about the--

CHAIRMAN SIESS: Speak up, please.

MR. WARD: --Paul asked earlier about the, you know, seems to be a remaining kind of major uncertainty with the cast materials.

Why wasn't the answer given that the specimen tests are going to give a lot of information about that? Won't they?

MR. ENGLISH: Well, part way through the program this issue about cast materials came up, and we discussed it in the context of advising the component test makers, and at every review meeting we discussed the component tests matrix to decide what we would like to change, and it seemed at those meetings when the question of castings came up that there weren't enough of them to warrant changing the component test program to include a casting, and maybe you couldn't draw a significant conclusion from one cast component.

The obvious way to look at this in some detail would be in the specimen test program, but that was already established, and maybe at some later date we can get at cast materials.

As Sam pointed out, we would probably restrict the rules at this point in time to exclude casting.

MR. BUSH: I may comment that if that is what-- [voice fades]-- on the basis that it would just make it too difficult from a design point of view.

MR. ENGLISH: Just briefly, you probably had most of this information on the component tests, but the major objectives were to show once and for all that collapse was not the failure mode of the components, to measure the ratcheting and cycles to failure.

We had 41 components in the program, all six inch in diameter, scheduled 10 through 40, with various combinations of elbows, tees, reducers, and in the original plan it was devised with Dr. Kennedy's help, was to put the peak of the input at about a half hertz below the component natural frequency, such that the component plastic input would be driven up to a higher peak value, and at Anco we were able to drive the sleds at the maximum capability of the sleds.

We wanted to get the fatigue ratchet crack to develop in two to three of these 20 second seismic inputs, any longer than that we felt would be pretty much just a fatigue test, so this was a target. Ultimately we eliminated schedule 80 because it took too many inputs at the Anco table capacity to generate a crack, and originally we had planned to do some schedule 160 testing, and that became obvious early in the program that we wouldn't be able to crack those components, so we focused on schedule 10, 40 and 80.

CHAIRMAN SIESS: In selecting the two or three inputs, you gave no consideration to what the probability was of a seismic input of a given duration--

MR. ENGLISH: No.

CHAIRMAN SIESS: --of frequency?

MR. ENGLISH: No.

We selected--I'll show you later. We just picked a typical seismic time history in a BWR. In a BWR--

CHAIRMAN SIESS: Yes, I know what you did. I was just asking.

MR. ENGLISH: No.

We reviewed that with Dr. Kennedy and we selected one time history and we used it for all of the tests, so we had one common basis--

CHAIRMAN SIESS: No, but my question you didn't understand.

When we have an earthquake it is one time history--

MR. ENGLISH: Right.

CHAIRMAN SIESS: --or D earthquake. It may be longer than 20 seconds, depending on where it is--

MR. ENGLISH: Right.

CHAIRMAN SIESS: --and there may or may not be after shocks, and the question of now what happens to a second earthquake when it has been damaged by the first one? And, I wondered if that was a consideration in the two or three--

MR. ENGLISH: No.

CHAIRMAN SIESS: --or just getting enough cycles?

MR. ENGLISH: No.

We just put in the same earthquake two or three times, with no consideration that the subsequent ones might be different in frequency counts.

CHAIRMAN SIESS: Okay.

[Slide]

MR. ENGLISH: The next two or three pages provide a capsule summary of the 41 component tests, and I didn't want to go over all of these with you, but only to describe what these headings are, and you can refer to them and ask any questions as we go along.

We show the number of the test here, the type of component, the material--as you can see, it is either carbon steel or stainless steel--and the schedule of the pipe. The residual strain, it was a cumulative ratchet strain that we measured in the component at the completion of the test.

In some cases we have no data because early in the program the high elongation of the gauges tended to come off of the component before the test was completed, and later we put scratch marks on the components so to be sure of getting a measurement of the cumulative strain.

The pressure is the internal pressure in the component, and typically the pressure is selected such that it creates a stress of 1 S of M loop stress in the component which is the ASME Code limit. Some of the lower stresses were to get immediate data points that were less than the Code limit.

The load direction is either in plane or out plane, typically . The ratio here of the dynamic moment, that is the measured moment in the test compared to the static limit moment, and you can see in most cases that we actually exceed the static limit moment without causing collapse of the component.

There are three types of loads: SSE is a seismic load. We used mid-frequency loads on a couple of components, and water hammer loading on a couple of components, and we had two static tests, but most of them were seismic tests.

The peak-to-peak cyclic strain is the maximum strain that was measured on the exterior surface of the component, at what we believe to be the maximum strain location, or high stress location of the component.

Now, the input times level D, this is the number of times the ASME Code faulted limit that the input represented. That is, if we took the input from the sled, did a linear response spectrum analysis with two percent damping, 15 percent broadening--just like you would have to do in a typical piping analysis--and calculated the stress in the elbow: in case 1 it would be 15 times the ASME Code allowable.

The number of times histories is the number of times histories that were input--20 second times histories at full amplitude of the sled before failure was in induced, or before we stopped the test.

Typically, we would not go more that five inputs. NF means that we got no failure. FR means there was a fatigue ratchet failure.

Without going into any detail on these different tests, there are a couple that I might point out that were significant. If you look at 6, 7, and 8 on the first page, that was an attempt to determine the effect of mean stress on fatigue ratchet failures. You see that they are all stainless schedule 40 elbows, and test 8 has zero internal pressure, test 7 has a 1000 psi, and test 6 has 1700.

And, if you will look over to the far right, under the number of time histories required to fail the component, you can see that as the pressure increases it took less input times to fail the component, so in effect, as we all knew, mean stress had some effect on the actual failure of the component.

There was one other component that I might point out that we introduced as a result of recommendations from Ev Rodabaugh. It took some convincing to convince everyone that we couldn't collapse the piping as the result of dynamic loading, and Everett suggested that one component test, Test 37, which would probably convince him, would be if we took a very low frequency component--in this case it was 1.4 hertz--and as you understood I am sure, the lower the frequency you get the closer you get to just a static collapse test--so we tested a component at 1.4 hertz with zero internal pressure--and with no internal pressure you don't get any stiffening effect from the pressure--and with that test we felt that was the bounding test of this whole program to show that collapse was not a credible mode of failure.

And, in this test it was somewhat excessive, in the sense that we had a very large weight stress--10,000 psi in this particular component, not the typical--in the normal operating reactive plane, and because of space limitations at Anco it couldn't--to get the frequency low, they had to put a large weight on the inertia arm, rather than bringing the inertia arm along, so we had a very large weight stress, and even with that we could not induce a collapse failure. We got what is called ratchet buckling incrementally because the large weight stress it tended to bend significantly.

CHAIRMAN SIESS: Did you convince Mr. Rodabaugh?

MR. ENGLISH: I hope so.

MR. RODABAUGH: Yes, that did it.

CHAIRMAN SIESS: Good.

MR. ENGLISH: And, the final page of this series is just more or less a legend that indicates what the terminology is.

[Slide]

I am sure you probably have seen this. Kelly's--at Anco yesterday, he probably showed you this, where we use the interia arm to--

CHAIRMAN SIESS: Yes, that we heard about.

MR. ENGLISH: --okay.

[Slide]

This was a time history that is input on almost all of the tests. It is representative of a RPD steam nozzle, typical PDW plant.

[Slide]

You can see the response spectrum of that time history as we tune the component just slightly to the right of the peak, with the input such that it softens as it is driven up.

[Slide]

This is the mid-frequency input. It is representative of a safety relief valve discharge on a BWR. It is a two-second duration. It is interesting that there is a small 7 hertz peak in this response spectrum, when the bulk of the responses we've seen in the components is the result of this peak, and practically none from the 30 hertz portion of the spectrum.

We found that if we tilt it out at this small peak here, and only applied true mid-frequency input to the component, we have negligible response.

[Slide]

This is just to give you an example of the kind of data that we get from Anco's use of four channels out of 28 that we would be examining. On a typical component test we spread the time histories out considerably more than the Anco data. We are able to look at the cycle.

I show this one just to show that you can see that as the relative displacement from the top of the sled to the bottom increases, those are these peaks, and you can see the ratcheting actually occurring in the elbows. When the peaks are small there is no ratcheting. When the peaks increase again, the ratcheting is back up to a point where, in most of these cases, it will shake down.

[Slide]

Now, this slide shows that as you increase the sled acceleration for these component tests, the cyclic strain initially increases relatively fast, and then tends to level off and become asymptotic somewhere between two, three, four percent cyclic strain. We believe that is because the component becomes so plastic up in this region that the plastic energy is absorbed by damping in the plastic deformation.

CHAIRMAN SIESS: How did you draw that curve? Whoever drew it didn't make it become asymptotic.

MR. ENGLISH: Well, maybe I shouldn't use the word asymptotic. It does tend to show that it doesn't increase--

CHAIRMAN SIESS: Well, I suspect that it might, but that looks like that these squares fit.

MR. SHEWMON: Is what you have there, is cyclic strain the same thing as the rise in the mean strain increases?

MR. ENGLISH: No, it is a cyclic peak-to-peak strain.

MR. SHEWMON: Okay.

MR. ENGLISH: We distinguish between cyclic strain and cumulative strain. Cumulative is the rise, and I'll show you that in the next slide.

MR. SHEWMON: Okay.

MR. ENGLISH: Okay.

[Slide]

This is the ratcheting strain or the cumulative strain that we measure on the exterior surface of the component, and these are the number of high input runs that the component was subjected to, and it shows that the majority of the ratcheting strain occurs in the very first time history input. It also shows the effect of pressure increasing, the internal pressure, the cumulative ratcheting strain tends to increase.

CHAIRMAN SIESS: Does the end point represent failure?

MR. ENGLISH: Yes.

MR. SHEWMON: What is the Code allowable pressure in this piping?

MR. ENGLISH: 17--well, these are two different kinds, two different schedules. This is schedule 40; this is schedule 10. The Code allowable is 1700 for this schedule 40, and 800 for the schedule 10.

MR. SHEWMON: Okay, yes.

MR. ENGLISH: This is a plot that shows that as the cyclic strain on the exterior surface of the component increases the damping increases.

We are talking, the SSE level D is down in this low region here. These are much higher strains than would be permitted to have at level D. But, also--even though we don't have great deal of data--it appears that the damping--for a given strain, the damping is greater in these thicker pipes than in the thin scheduled 10 pipe, which implies that maybe more energy is being absorbed in the thick-walled pipe for a given surface strain than in a thin-walled pipe.

Now, these damping values here are strictly material damping, and you have a lot greater damping in a piping system of insulation, gaps, sliding, friction.
[Slide]

This is a typical hysteresis loop that in this case just shows moment versus displacement for one of the components, and you can see that the curve flattens out which would tend to indicate that you have reached some kind of limit moment, but the curve reverses before the displacement can actually cause physical collapse of the component.

CHAIRMAN SIESS: What is that vertical line over on the right?

MR. ENGLISH: It has no significance.

MR. RODABAUGH: That's their plotter.

MR. ENGLISH: That's the plotter, right.

CHAIRMAN SIESS: That's the plotter, okay.

MR. ENGLISH: Yes.

CHAIRMAN SIESS: And, they forgot to put the zero in--

MR. ENGLISH: Okay.

[Slide]

Now the next couple of slides show the effect of the--or the comparison of the linear elastic analysis that we do with two percent, five percent damping--

CHAIRMAN SIESS: Excuse me.

MR. ENGLISH: --yes.

CHAIRMAN SIESS: Could you go back to that previous slide for just a minute.

MR. ENGLISH: Yes.

CHAIRMAN SIESS: That is the one that I was looking for.

MR. ENGLISH: Okay.

CHAIRMAN SIESS: Where does it start?

MR. ENGLISH: Somewhere down in here. I am not sure that it is very clear.

MR. RODABAUGH: I think it is a little bit above the end of that diagonal line, right across from zero. You had your pencil almost on it.

MR. ENGLISH: Down in here?

MR. RODABAUGH: Down a little bit.

MR. ENGLISH: Here.

MR. RODABAUGH: Now, up that diagonal, there.

CHAIRMAN SIESS: What about that little peculiar--

MR. ENGLISH: Well, it strain hardens as you go along.

CHAIRMAN SIESS: What is that little peculiar--

MR. ENGLISH: This was one of the better looking ones. The rest of the looked real strange. I don't think we can--

CHAIRMAN SIESS: I put the zero axis on it. I couldn't figure out which one it started at.

MR. ENGLISH: Yes.

CHAIRMAN SIESS: The first small loop is practically all over in the upper left quadrant.

Okay, go ahead.

MR. ENGLISH: All right.

[Slide]

As Sam indicated earlier, the elastic analysis can be non-conservative at ratios of the peak of the seismic input to the natural frequency of plus of one, and so we investigated that and at some of the component tests found that for two percent, five percent, damping with peak broadening we were able to conservatively predict the moment, compared to the measurements, so I think the peak broadening used in the calculations probably insures that the elastic calculations that we've done on these component tests are conservative.

CHAIRMAN SIESS: No, I'm sorry.

This is peak?

MR. ENGLISH: This is the peak of the input--

the frequency of the peak of the input divided by the natural frequency of the component.

CHAIRMAN SIESS: Okay, I see.

MR. ENGLISH: And, this is not linear. These are just three data points.

CHAIRMAN SIESS: I see.

MR. ENGLISH: Probably could be better represented with a bar chart.

[Slide]

This is a similar calculation showing the displacement versus the frequency ratio.

[Slide]

From the component tests we--

MR. RODABAUGH: I would like to make a point, I think for the benefit of--

CHAIRMAN SIESS: Speak a little louder, will you please.

MR. RODABAUGH: --some of the other committee members.

In some of your other tests, don't you have measurements that would not be conservatively predicted by two percent damping, or two percent broadening--displacement is what I am thinking about.

MR. ENGLISH: Yes.

The displacement is the one that--the moments are--

MR. RODABAUGH: The moments are okay.

MR. ENGLISH: --always conservative, yes, but some of the displacements were a little bit screwy at times.

MR. RODABAUGH: Well, I think that is an important point.

I didn't see it on the graph here, is what I am saying.

MR. ENGLISH: I had to cut down the presentation to something within the time constraints.

MR. RODABAUGH: Yes.

MR. SHEWMON: Well, is your point, Ev, that the designer also specifies the maximum displacement, and that this is used and important?

MR. RODABAUGH: The displacement could be important in the sense that--as we were discussing yesterday--you have a motor operated valve with some cable, a certain amount of slack, now as you took off snubbers, for example, the displacement would increase. You would like to know what that displacement is so that you can look at your cable, electrical cable, and see whether it has got enough slack in it.

MR. SHEWMON: Okay.

MR. RODABAUGH: And, there are many other examples of that type of thing--

MR. ENGLISH: Well, as Sam pointed out that is an area of non-conservatism down there and some of the calculations showed that we were okay, and others it was questionable.

CHAIRMAN SIESS: Put that last figure back on.

MR. ENGLISH: Okay.

CHAIRMAN SIESS: It is interesting, and I don't quite understand it.

This is the ratio of the applied frequency to the natural frequency--

MR. ENGLISH: Yes.

CHAIRMAN SIESS: --and that I understand, and that is the displacement.

MR. ENGLISH: Yes.

CHAIRMAN SIESS: Now, one of those curves is the measured displacement?

MR. ENGLISH: Right.

CHAIRMAN SIESS: And, then the others are the computed displacements--

MR. ENGLISH: At different dampings.

CHAIRMAN SIESS: --elastic analysis, single degree of freedom--

MR. ENGLISH: Yes, that is it.

CHAIRMAN SIESS: --at different damping, okay.

Now, I understand, thank you.

MR. ENGLISH: Okay.

We are not implying that we are--that elastic analysis is good across the board. As Sam pointed out, that's an area of concern which we are still trying to decide whether to use peak broadening, or what is supposed to be done. In these couple of cases, it looked like we had done a pretty good job. We haven't looked at every single case by any means.

[Slide]

So the observations from the component tests are that the dynamic load reversal, we believe, is what prevents collapse. The seismic loads behave more like secondary than primary, and the ratchet failure loads are much greater than the SSE. The ratcheting doesn't impair functionality, the diameters tend to increase, rather than decrease.

The damping for large dynamic loads, bigger than the SSE is certainly greater than the reg guide would permit. The amplified high frequency SRV loads cause negligible response to the component.

So, the bottom line we were trying to show from these component tests was that failures were not collapse type failures as the current Code indicates they might be, but rather fatigue and fatigue ratcheting types of failures.

[Slide]

The system tests main objectives were more confirmatory to confirm that the failure mode was was as observed on the component tests, using either three or four sleds, and confirmed the effects of low- and mid-frequency loadings. To determine a system damping for a number of different kinds of configurations we looked at systems with balanced stress, high stress everywhere, unbalanced stress, different time histories inputs at different sleds, with or without snubbers and struts.

We also wanted to confirm that the functionality was not violated, compromised, and conformed the design rules and margins that we'd observed--

CHAIRMAN SIESS: Now, when you say "functionality" you are limiting yourself to the pipe?

MR. ENGLISH: Yes.

CHAIRMAN SIESS: And, not the valve operators.

MR. ENGLISH: Yes, yes, that is right.

CHAIRMAN SIESS: Okay.

MR. ENGLISH: Okay.

[Slide]

The major focus on this program has been piping, and it has been indicated that it would be nice to look at supports, too, but--

CHAIRMAN SIESS: And, the valves were there simply to provide loadings.

MR. ENGLISH: And, to get a little bit more information--it was kind of a, you know, piggy back type test.

CHAIRMAN SIESS: Right.

MR. GUZY: I think we should point out in that system 1 test, there was a valve that we operated--

CHAIRMAN SIESS: Yes, yes.

MR. GUZY: --and we do have limited information on that.

CHAIRMAN SIESS: That was the piggy back.

MR. GUZY: Right.

MR. ENGLISH: So, you have seen this. There is an operational hanger and an operational valve. This pressure vessel has a vesselette that simulates--

CHAIRMAN SIESS: We saw that.

MR. ENGLISH: --and it has an R over T ratio that is typical of a reactive pressure vessel, even though it is very small.

[Slide]

I think you have seen this stress summary--

CHAIRMAN SIESS: No, we haven't.

MR. ENGLISH: The only thing I wanted to--

CHAIRMAN SIESS: No, we haven't seen that.

MR. ENGLISH: --oh, you haven't seen this?

Okay, this was a pre-test calculation, mainly to try to identify the location of where a failure would likely occur, and to make sure that the stresses in this particular system are relatively high, and relatively uniform throughout the system.

This was a high-stress system. Stresses were calculated using ASME Code techniques, and we identified this short radius elbow as the highest stress location. The 1 means the highest stress; 2 means the next highest stress location.

So, the Code calculation, in fact, was successful in predicting the failure location for this particular test.

MR. BUSH: Bill, can I ask you a question?

MR. ENGLISH: Yes.

MR. BUSH: It is kind of a follow-up on something that I said earlier.

Now, in this instance you had a--I think this is the one where they had a valve--

MR. ENGLISH: Yes, correct.

MR. BUSH: --okay, and on the basis of either a pump or a valve on the inherent thickness--or inherent stiffness, it is necessary to provide the function. In other words, you can't have the body of the pump weaving all around or it won't pump water, and the valve the same thing is true, so as the result they are much, much thicker than would be calculated on the basis of Code allowable.

It would seem to me that on a simple analytic basis you could pretty well establish that the body of a pump or of a valve, whether cast or wrought--and most of them are cast, and whether stainless or ferritic--probably is not a factor--

MR. ENGLISH: Because the stress is so much lower.

MR. BUSH: --because the stresses are so low that you can basically--it is just like a rigid object sitting there, and you could probably, in a relatively straightforward fashion dismiss them, which would remove about 95 percent of my concern with regard to the system, because I worry if you kind of ignore pumps and valves, because there an awful lot of valves in this pipeline system.

MR. ENGLISH: I might point out that this valve did not lose pressure integrity through the whole test, and it was identified as a high stress location. And, in any event it saw G loadings well in excess of what its rating is.

[Slide]

These are the various runs that I am sure that Spence discussed with you yesterday. We looked at uniform input, and we looked at independent support motion input.

[Slide]

These are the tables, all the way up to full table capacity.

[Slide]

And, I think you probably saw the fail component, the crack initiated in the elbow bent center, and propagated around the elbow, which is where the ASME Code calculations had predicted it would occur.

[Slide]

This is a summary of system test 1 at that short radius elbow. You can see, depending upon whether you assume two percent damping or five percent damping, the stress was a large number times the total allowable fault conditions. The stress at the SSE input was about half a level D limit. There was no ratchet strain at that level. At three to five times the level D limit we only had a quarter percent ratchet strain, and no ratchet displacement. The staff was questioning us back in September about how much ratchet displacement in the piping system as the result of these large dynamic loads, and it wasn't until we got up to very large loads, up to half the table capacity, that we got any significant ratchet displacement in the piping system.

CHAIRMAN SIESS: Why was the ratchet strain larger at half than full?

MR. ENGLISH: This is the strain during that run, and the component failed early.

MR. SHEWMON: What are the units on these first set of numbers up there?

MR. ENGLISH: These?

MR. SHEWMON: Yes.

MR. ENGLISH: They are non-dimensional, just a multiple times level D.

CHAIRMAN SIESS: So it is a multiple of level D, okay.

MR. ENGLISH: Yes.

CHAIRMAN SIESS: So, the SSE was 8/10ths?

MR. ENGLISH: Yes, 8/10ths of level D, right.

MR. SHEWMON: Now, then I guess I am--the inputs known, and if you assume a five percent damping, then it is 24 X D--

MR. ENGLISH: Right.

MR. SHEWMON: --and if you assume at two percent, it is 42--

MR. ENGLISH: Right, yes.

MR. SHEWMON: --okay, I see.

MR. ENGLISH: It is how you calculate it, really determines what the ratio is.

I think the other significant thing here is that at these relatively low stress levels the damping wasn't any higher than the N-411 would permit you to use today, even though, again, this is material damping, and damping at the plant would be considerably higher than that.

CHAIRMAN SIESS: What would you call relatively low? The SSE? Well, even the five SSE--

MR. ENGLISH: Yes.

CHAIRMAN SIESS: Yes.

MR. ENGLISH: These two.

CHAIRMAN SIESS: Yes.

MR. ENGLISH: This is not relatively, it is relatively low compared to the full table.

I think the one message from these system tests that it just takes one hell-of-a-lot of load, seismic load, to break a piping system, or even to damage it.

[Slide]

I did have a couple of slides here that again Spence may have shown you yesterday. This is the strain gauge on the elbow that failed, and it shows that the half table run, the ratcheting that occurs, and in fact the ratcheting continued on up after the sensor departed the scene.

[Slide]

And, this is the accompanying ratchet displacement at the top of the piping system. You can see these large swings of 14 inches, or so, peak to peak, the top of the piping system, and when it finally came to rest after a half-table input, the final displacement was only about an inch from where it started initially, so that ratchet displacement of these piping systems doesn't seem to be of concern.

The zero is right here in the middle.

CHAIRMAN SIESS: Computers are funny that way.

MR. RODABAUGH: That one ratcheting figure here that you showed us, this is gauge failure?

MR. ENGLISH: Yes.

MR. RODABAUGH: Then the sensor apparently kept saying it was a high strain, even though it failed?

MR. ENGLISH: That is what it looked like.

MR. RODABAUGH: Yes, okay.

CHAIRMAN SIESS: At least it shows a high resistance.

[Slide]

MR. ENGLISH: This is again an attempt to show that the linear elastic analysis that we do in piping gives conservative results, especially conservative at the high input levels. We calculated the moments. The moments are what we use to calculate the allowable stresses-- or calculate the stresses in the piping system.

Once again, as Ev has pointed out earlier, the displacement calculations don't have that much conservatism in them, but they are a pretty good approximation of the displacement at low inputs, and somewhat conservative at the high inputs--at least with the specter that we use.

MR. SHEWMON: Now, is moment, the calculated moments that you have here is something you get from a stress--a strain you would get out of a strain gauge, or what?

MR. ENGLISH: These moments are calculated--yes, they are.

MR. SHEWMON: Okay.

MR. ENGLISH: They are an adjacent part of the component that remains elastic. We had a big thick component adjacent to the elbow that stays elastic.

MR. SHEWMON: I see.

MR. ENGLISH: Okay.

[Slide]

A system two was a system of unbalanced stress, and it used a fabricated nozzle rather than a forged nozzle, and it had a snubber, and all of the valves in this system are simulated there, they are no operational valves.

Four sleds were used in system 2, rather than three, and we used stainless steel as the material. I don't know whether they mentioned to you yesterday that we were attempting to simulate 316 nuclear grade material, and we did that by selecting 316 L with 316 mechanical properties which is the least expensive way to try to simulate 316 nuclear grade.

MR. BUSH: [out of hearing range]

MR. ENGLISH: Well, it was low carbon and high strength. It was hand picked for the components.

MR. BUSH: Yes, but did you use ELC grade? Or did you use the 316?

MR. ENGLISH: The 316 L, hand selected--

MR. BUSH: Okay, I misunderstood, so what you really had was what is in the upper part, the right hand, so you are closer to what you'd expect to get--

MR. ENGLISH: Right.

MR. BUSH: --in the increased strength because of that.

MR. ENGLISH: Yes.

[Slide]

This is a stress summary for a system 2, and you can see this nozzle was the area of high stress. It was a factor of two higher than anyplace else in the piping system, and again the ASME Code calculations correctly predicted the location of the failure in this system.

Again then--

MR. BUSH: And, that nozzle must have been a fairly stiff one so that the load was pretty much translated to the interface between the nozzle and the vessel?

MR. ENGLISH: Yes, yes.

[Slide]

These are the inputs that are essentially the same as we used in system test 1, with the exception of the sine sweep input, and it turns out that the fatigue usage introduced into the nozzle was significantly increased by this sine sweep input, as compared to even the full table input. Sinusodial input has a much more marked effect on the tee uses than the random nature of the seismic inputs.

CHAIRMAN SIESS: You get more maximum cycles.

MR. ENGLISH: And, in fact, I am sure they indicated to you yesterday, they started to see surface cracking right after the sine sweep input.

[Slide]

And, this is just to show you that the cracks actually occurred on the side of the fabricated nozzle out at this--adjacent to the well, the vessel interface.

CHAIRMAN SIESS: What was ISM correleted?

MR. ENGLISH: That is--

CHAIRMAN SIESS: To the input side?

MR. ENGLISH: --independent, oh, okay.

MR. ENGLISH: Inputs in the different sleds, and we had an inface, outface.

CHAIRMAN SIESS: Okay.

MR. ENGLISH: Again, I wanted to show you this relationship.

[Slide]

The relationship between the local ratchet strain and the high stress component and the net residual displacement in the piping system, such that even though we get a large residual, local strain, ratcheting strain on the component, the piping system itself comes back to its initial starting position in this particular case.

CHAIRMAN SIESS: Go back one.

[Slide]

MR. ENGLISH: This is the ratchet strain in that nozzle that failed.

MR. ENGLISH: Okay.

Now, what is the big step in there?

MR. ENGLISH: You have this high displacement, you immediately get large ratcheting, local ratchet strain, but that doesn't translate into a large displacement of the piping system. The piping system, even though it is vibrating, it could be 16 inches peak to peak, it still comes back to rest because the load is redistributed and there is no permanent deformation of the piping system.

CHAIRMAN SIESS: Okay.

MR. ENGLISH: Okay.

[Slide]

This again is an attempt to show that linear elastic analysis is very conservative at the high load level and reasonably good at the low load level, again with peak broaden in all of these calculations. That is a displacement in the moment calculations.

[Slide]

So the observations from the system test, I think, could be summarized as followed: We confirmed that fatigue ratcheting again was the failure mode, as we expected. Failure loads are much greater than the SSE. The functionality was not impaired. Damping was very large, much greater than R.G. at these levels that we obtained failure.

MR. SHEWMON: Now is the Reg Guide the same as what was called the Code case N-411?

MR. ENGLISH: No, that is five percent damping. They are a little different.

MR. SHEWMON: Oh, okay.

MR. ENGLISH: The damping is also bigger than the Reg Guide--

MR. SHEWMON: Yes.

MR. ENGLISH: --also than the Code case--

MR. SHEWMON: I was just wondering if the Reg. Guide had caught up with the Code case yet?

MR. ENGLISH: It is catching up, but it is not there yet.

And, the amplified safety relief valve loads are very small. When we filtered out that small seven hertz peak we got practically no response.

I think, again, the bottom line is that piping is the result of--in seeing these tests--just extremely difficult to fail, and I am sure you all realize by now that the building will fall long before the piping system is going to fail.

MR. SHEWMON: I am not so sure of that.

MR. BUSH: Well, the great Alaska earthquake pictures.

CHAIRMAN SIESS: Well, I'll admit, you found more buildings that fall down during earthquakes than you ever found piping that failed.

MR. ENGLISH: Yes.

[Slide]

I've separated out the water hammer testing because we've come to somewhat different conclusions from the water hammer testing than from the seismic and low frequency testing.

We have two component tests that are really two small loops, about 50 long, six inch in diameter, carbon steel and they have been tested with and without supports.

CHAIRMAN SIESS: Well, we saw both of those.

MR. ENGLISH: You saw both of those yesterday?

The mini-system test is considerably longer and it has supports, branches, vessels, thin pipes.

[Slide]

And, these are different kinds of loads that we have conducted--Anco has conducted--simulated steam hammer, hard system acoustic test, the water slug, and various gravity pressures.

MR. BUSH: In the steam hammer, you are--

CHAIRMAN SIESS: Little louder, Spence, we can't quite hear you.

MR. ENGLISH: Steam hammer is more like the hard systems test, you have a pipe that is just full of air and you pressurize it up stream at 1000, 2000 psi, compress the diaphragm and watch the reflective wave in air, rather than in water. It is not steam, but it is a gas rather than a liquid.

MR. BUSH: And, the loads from that steam hammer?

MR. ENGLISH. The lowest of the three categories of loading.

[Slide]

Now, you stop me if you have seen all of this.

CHAIRMAN SIESS: Well, we've seen the next two slides, yes.

MR. ENGLISH: Okay.

[Slide]

Okay, these are the preliminary observations from the water hammer test, and I say they are preliminary because the second mini-systems test was just finished yesterday, or the day before.

But, at any rate, as the result of what we know already, the water slug definitely the primary type loading that can cause collapse of piping systems, and we would probably say in equation 9, and in fact maybe all of these water hammer loads should be in equation 9 with some kind of a relief on the allowable.

MR. SHEWMON: Equation 9?

MR. ENGLISH: It is a piping design, sort of a fundamental piping design equation which is based on collapse of the component.

MR. RODABAUGH: I will write you later, but I will reserve judgment on that, because I am not so sure that water hammer is that much different than earthquake.

MR. ENGLISH: Good, glad to hear that.

Well, I think the steam hammer and the hard test certainly behave more like secondary loading. They didn't collapse the pipe, even though the limit moment was exceeded and so we tend to be more inclined to categorize those as secondary.

CHAIRMAN SIESS: Which was the loading that picked that pipe up, up against the--

MR. ENGLISH: That is the water slug test.

CHAIRMAN SIESS: That is the water slug test.

MR. ENGLISH: Yes.

MR. SHEWMON: I'm interested to see Everett's letter on the water slug test.

MR. RODABAUGH: What I am thinking about, Chet, is remember this is a loop that went all around that building, and the end, when you got down to the end was free.

MR. ENGLISH: Right.

MR. RODABAUGH: Now, if you compare that with your elbow test, where you had that long, I think, but very, very short length of pipe when compared to this huge loop.

MR. ENGLISH: Yes.

MR. RODABAUGH: I would like to think about it a bit more before I said that there is really that much difference between the two.

MR. ENGLISH: All right.

MR. BUSH: Are you arguing about the primary versus the secondary, or about the test per se?

MR. RODABAUGH: Well, both.

I think other people have brought up the point that if you had a pipe that long you would surely have a second anchor someplace.

MR. ENGLISH: Yes, yes, this was definitely more severe than you would expect to see in a power plant. We have had that comment made by others, that in the power plant the piping terminates at a heat exchanger pump, or vessel, or something. It doesn't just hang out in the breeze like that one did.

MR. RODABAUGH: Or a containment--

MR. ENGLISH: Or containment, yes.

MR. RODABAUGH: --and it breaks there.

MR. ENGLISH: Right.

[Slide]

We also found that supports could tolerate dynamic loads up to ten times more than a rated load, and it looks like, based on the telephone conversations yesterday with Anco, that piping can tolerate transient pressures maybe one to two times the burst pressure without actually rupturing.

We had a section of schedule 10 pipe put in the second mini-system test and we were unable to break that with the loads that they tested.

I think, at this point, that the basic rule is still to design to avoid water hammer.

CHAIRMAN SIESS: When you say two times the burst pressure, that is the static burst pressure?

MR. ENGLISH: Yes.

CHAIRMAN SIESS: And, the design pressure would be at what ratio of that?

MR. ENGLISH: The burst pressures are, I think, around 4700 psi, and design pressure would be 800.

CHAIRMAN SIESS: So, there is that kind of a factor in there?

MR. ENGLISH: Yes.

MR. BUSH: Let me ask an embarrassing question.

When you consider a straight section 10, if you would have put a section 10 elbow at that first location, what do you think would have happened?

MR. TAGART: Do you mean schedule 10?

MR. BUSH: I'm sorry, not section, schedule.

MR. ENGLISH: I don't think it would have ruptured.

MR. BUSH: Well, I am much less optimistic than you are.

CHAIRMAN SIESS: Are you making calculations that will enable you to answer that kind of a question? Or, is this strictly empirical?

MR. ENGLISH: I would say that it is pretty much empirical at this point.

[Slide]

Incidentally, there is a typo in your handout, this 1 dash was left out, and there was another typo.

I have forgotten where it is.

CHAIRMAN SIESS: What is that?

MR. ENGLISH: One to two. We know the pressure is higher than the burst pressure, but we don't know, you know, we are just getting the information over the phone at this point, so perhaps we shouldn't have even put that in there. We haven't seen the data yet.

[Slide]

And, on this systems test 2, the damping down at the bottom of the page, there are a couple of errors there, particular, the five and a five and a 22, that should be 5.235.

[Slide]

Okay, the specimen test, as I indicated earlier, that the objective was to develop a lab specimen that could give us some quantitative evaluations of fatigue ratcheting with mean stress.

We also wanted to be able to correlate the behavior specimen with the components, and extrapolate conclusions from the four test materials--the four materials that were tested to other piping materials, and as Roy indicated in one of the progress reports, that two of the materials are non-strain, softening or hardening. One is strain softening, and one is strain hardening, and two are stable, and we wanted to investigate the fatigue ratcheting effects at 550, which you could really on do practically with specimen tests.

[Slide]

This is a matrix of the specimen test--excuse me, I'm sorry.

MR. SHEWMON: All those materials that you used for strain hardening in the tensile tests, so when you say they are strain softening, you are implying after a certain number of cycles and certain kind of a test.

Would you define that a little bit better? How may cycles?

MR. TAGART: It is a cyclic fatigue test.

MR. SHEWMON: Yes.

MR. TAGART: You do a low-cycle fatigue test, and the stress range increase for a given strain range as you progress--

MR. SHEWMON: And, this is a distress with--low cycle is anything under 1000 cycles of failure, as I recall.

So, you are in that range with a lot of cycles, is that it?

MR. TAGART: Well, it varied, depending on what strain range you put on it initially, but they are all characterized--when we say softening we mean that after some initial hardening it begins to lower the stress range with the number of cycles.

CHAIRMAN SIESS: Now, these are controlled strain tests?

MR. TAGART: Yes.

CHAIRMAN SIESS: So the stress--

MR. ENGLISH: Now there are 140 component tests here, of which 48 were--excuse me?

CHAIRMAN SIESS: Forget it, go ahead.

MR. ENGLISH: The majority of the tests were two-bar tests.

[Slide]

This is a two-bar milimodel where you have two bars with rigid ends, you apply a fixed-mean load and you cycle one of the two bars, let's say by heating, such that that bar is in compression with other bars in tension, and it turns out in this testing program the bar that is being heating is simulated by a computer and the testing is done on a single-uniaxial specimen.

Most of the testing in the program, as you can see, is done using this two-bar technique, and then the verification test done with beams and pressurized pipe, to show that the two-bar tests are conservative.

CHAIRMAN SIESS: Now, the two-bar test are not two bars.

MR. ENGLISH: It is just one bar.

CHAIRMAN SIESS: I see.

MR. ENGLISH: It is just this bar with the testing machine, computer controlled to simulate the effects of the second bar on this first bar.

CHAIRMAN SIESS: What is the object of the two-bar test, originally?

MR. ENGLISH: It is a simple way of measuring ratcheting, mean stress.

And, that two-bar test was verified with--

CHAIRMAN SIESS: Before you had computer controlled testing machines?

MR. ENGLISH: Yes.

[Slide]

The rectangle beam specimen looks like this and it is tested with an applied axial load and an alternating bending load, and that fits into this machine, in this region here these springs apply the static load and then the full-point bending is done by the actuator here.

CHAIRMAN SIESS: Is that a strain control?

[General discussion by several speakers.]

MR. RODABAUGH: Chet asked the question: is this strain controlled, and he said, "Yes."

CHAIRMAN SIESS: A little louder, Ev, please.

MR. RODABAUGH: You asked the question: is it strain controlled? And--

MR. ENGLISH: I guess it--is it load control?

MR. RODABAUGH: --no, it is some of both, is the point.

MR. RANGANATH: It is initially strain controlled.

MR. RODABAUGH: The bending is strain controlled, but see those big springs on the end--

MR. RANGANATH: That is prime mean load.

MR. RODABAUGH: --yes, it is load controlled.

MR. ENGLISH: So, it is attempting to simulate the ratcheting.

[Slide]

The observations from the specimen tests to date are that the two-bar test is a conservative representation of the ratcheting on cyclic life. The beam and pipe specimens confirmed that the two-bar test is conservative. We found that if you have controls on the cumulative ratchet strain, that is if you don't permit significant large ratcheting to occur, you get in fact a fatigue type failure, and that with controls on the ratchet strain the mean stress, then the temperature doesn't effect the cyclic fatigue life.

And, we observed some cyclic creep in these tests that were done at MCL, and the creep occurs in the testing which is done at about two minutes per cycle. It is a very slow test. We've identified some creep in these tests, but we don't think it is going to be present in the seismic test.

CHAIRMAN SIESS: This is creep in the axial direction?

MR. ENGLISH: Yes.

MR. SHEWMON: Are the basis for these conclusions written up in a paper that has been submitted, yet?

MR. ENGLISH: No.

MR. TAGART: No, not yet.

MR. ENGLISH: The whole program will be written up soon.

MR. TAGART: This will be one of three papers.

MR. SHEWMON: Okay, I would like to see this one.

CHAIRMAN SIESS: What rate are these run at?

MR. ENGLISH: These are run at a rate of 1000 times faster frequency--higher frequency than these tests, so we think the cyclic creep that has been observed in these low frequency tests probably won't be present in any significant degree in the seismic tests.

CHAIRMAN SIESS: How many cycles?

MR. ENGLISH: Well, these tests are two minutes per cycles, and the seismic tests are like eight cycles per second.

CHAIRMAN SIESS: And, now many cycles did you get per failure?

MR. ENGLISH: In these tests?

CHAIRMAN SIESS: In these specimen tests?

MR. ENGLISH: Oh, gosh, it depends on the strain range, but hundreds, thousands.

CHAIRMAN SIESS: Hundreds, yes.

MR. BUSH: You don't distinguish whether you are talking about--on that last bullet, on room temperature or elevated temperature, or both?

MR. ENGLISH: Both.

MR BUSH: It is both. I thought it might be, but I wasn't sure.

MR. ENGLISH: Okay.

MR. RODABAUGH: Bill, you have one other type of test, the pipe test, pressurized pipe test.

MR. ENGLISH: Pressurized pipe test?

MR. RODABAUGH: The last in your list of specimen tests.

MR. ENGLISH: Oh, oh, yes. I didn't show that picture. I don't have a cross-sectional drawing, but it is just a small piece of steel pipe, and it failed at the grips where the pipe is being held by the testing machine.

MR. RODABAUGH: I was mentioning the rubberband aspect of it.

MR. ENGLISH: Oh, yes.

It turns out in the specimen test, the pressurized pipe test, the rubberbands on the exterior--oh, you have already heard about that?

CHAIRMAN SIESS: No, no, go ahead. I would like to hear it again.

MR. ENGLISH: I am not sure--

MR. RANGANATH: It is on page 374.

MR. SHEWMON: On 374?

MR. RANGANATH: On 3-74.

MR. ENGLISH: It is pretty interesting to see that a rubberband could actually cause deformation in a pressurized pipe.

CHAIRMAN SIESS: Well, is that good or bad?

MR. ENGLISH: I'm not sure.

[General discussion]

MR. BUSH: It is surprising.

MR. TAGART: Damping is very difficult to predict exactly, and ratcheting is very difficult to predict exactly.

CHAIRMAN SIESS: Does that conclude your presentation?

MR. ENGLISH: Yes.

CHAIRMAN SIESS: Are there any further questions?

[No response.]

MR. ENGLISH: Sorry to run over.

CHAIRMAN SIESS: No, no, you were right on schedule.

We started late. We are doing real good, unusually good.

Okay, Mr. Ranganath.

MR. RANGANATH: Thank you.

[Slide]

I am going to briefly cover potential design rule changes that we may look at as the result of the data that we have generated here.

I want to emphasize the word "potential" because we are still evaluating it, and what I am going to talk about, it kind of gives you a flavor of the direction we are going in. We may in fact change some of the exact numbers that we have, either recommendations, that I will explain here.

Now, I am going to talk about three items, kind of like what were the conclusions from these component and system tests, and regards to design rule development? What did we learn from the specimen fatigue and ratcheting tests? And, I also am going to give you a little more of a description of the single degree of freedom model analysis that we did as it relates to the inelastic dynamic response as well as ratcheting.

And, from that I will try to make some conclusions relative to the elastic analysis, make some proposals on rule changes, and briefly remark on what the long-term goal might be in terms of this proposed design rule changes.

CHAIRMAN SIESS: Let me interrupt you for just a minute.

MR. RANGANATH: Yes.

CHAIRMAN SIESS: Could you at some point give us a little picture of what the design rules on now, for example, what equation 9 is, and how it relates to equations 1 through 8?

I am sure that some of the people at the far end of the table are quite familiar with this, and some in the middle may be.

MR. RANGANATH: Yes, I am going to do that.

[Slide]

Now, this has been said many times: the component system test, we just couldn't fail the thing, even though we went to well in excess of what the level D allowables might be, and in fact we could not have any collapse effect, so no limit load--

CHAIRMAN SIESS: Level D is faulted?

MR. RANGANATH: --level D is faulted--

CHAIRMAN SIESS: That is everything you can think of?

MR. RANGANATH: Right, right.

CHAIRMAN SIESS: Thrown in there?

MR. RANGANATH: Right.

CHAIRMAN SIESS: That is the highest level?

MR. RANGANATH: That is correct.

CHAIRMAN SIESS: It doesn't get any worse than D, then?

MR. RANGANATH: Yes.

CHAIRMAN SIESS: Okay.

MR. RANGANATH: We didn't see any limit load failures, and again maybe this is the time to talk about the equation 9. Equation 9 just says that in effect we can call it without confusing the issues. There are some constants, M over Z, less than some allowable, call it here 1.5 S of M.

What it is then, this is the pressure stress, okay, this is the actual pressure stress, this is the bending moment that includes the seismic weight and so on, and these are generally multiplied by a figure which was determined in large part by some tests done by Markl in the beginning, and Ev Rodabaugh himself lead to a lot of these indices. These are called B-2 indices.

CHAIRMAN SIESS: Are they factored greater or less than one?

MR. RANGANATH: Some are less than one, but most of them are greater than one.

CHAIRMAN SIESS: Now, just to let you know what level you are talking to, what is S of M?

MR. RANGANATH: S of M is what's called as a design stress intensity that established by the ASME Code. It is generally quoted to the lower of two-thirds of the yield strength, or one-third of the ultimate.

So, anyway, it says you have a factor of three margin on pressure for ultimate strength.

CHAIRMAN SIESS: Well, they call it a stress intensity, and the units are stress?

MR. RANGANATH: That's correct. It is nothing to do with the stress intensity factor.

CHAIRMAN SIESS: It is a stress?

MR. RANGANATH: Right.

CHAIRMAN SIESS: It is an inelastic strain.

MR. RANGANATH: Right.

So, and again these are all done on a pseudo-elastic basis so if you look at--

CHAIRMAN SIESS: Then 1.5 S of M, might be the yield stress?

MR. RANGANATH: For carbon steel it is. For stainless steel it is a little higher than the yield strength, but as a rule it is--

MR. RODABAUGH: Since Chet asked about level D, specifically, you have been talking about SSE, why don't you put a 3 out there?

MR. RANGANATH: Right, okay.

This is for what is called a design condition, and 3 S of M is what is called as the level D.

CHAIRMAN SIESS: Design, in terms of seismic would be OBE?

MR. RANGANATH: That is correct.

CHAIRMAN SIESS: But, there are some other things also that are in there, design and--

MR. RANGANATH: Right, yes, it will be the design pressure plus all of the loads.

CHAIRMAN SIESS: Yes.

MR. RANGANATH: Now, when we did get failure, failure was predominantly due to a combination of fatigue and ratcheting, so that accounts for emphasizing that we ought to be focusing on those.

But, even when we did get failures by fatigue and ratcheting, we were able to have the number of cycles it took to cause failure was well in excess of what you would, for example, expect in one level D event, like many of the tables that Bill showed you had two, three, four times full cycles of this level D transient, so that shows that you have got a lot of margin even there, too.

Analysis of the test shows that elastic prediction are generally conservative for response spectrum analysis with peak broadening for up to five percent damping. You know, you recall that Bill showed you this comparison of displacement and moment, and--

CHAIRMAN SIESS: I am not sure that I buy the two or three cycles, the two or three imputs, as well in excess of an SSE.

That could be an SSE plus one good after shock, or two after shocks, and if I recall we've had earthquakes where the second shock was much greater than the first, and I think there have been three well up in there. Now, there can be considerable time between the first and the second--

MR. SHEWMON: A year.

CHAIRMAN SIESS: --not this one, as one was in December and one was in January.

[laughter]

Now, if we had the first earthquake, obviously the plant would be shut down, but the second one could bust some pipe and they may be required to move the KE except there would be a lot less, but again we frequently get an after shock within a day or two, and sometimes within hours. It usually is smaller than the first one.

But, again, I am, you know, we are sort of--

MR. RANGANATH: Yes, but the key though is that we didn't get--all of the core calculations are based on primary loads. If you apply the the same load enough number of cycles, yes we can expect cracking, but the magnitude of the loads themselves you can tolerate numbers well into excess of--

CHAIRMAN SIESS: Well, if you talk about that, yes, the magnitudes.

MR. RANGANATH: Yes.

[Slide]

You also heard Bill's conclusions relative to the fatigue ratchet cycles, the testing that was done, and ratcheting occurs when you have a combination of primary means stress, and cyclic dynamic stress.

And, here are some conclusions that we can get from Dan Miller's original model which was originally intended for thermal stresses. It turns out that these are applicable even for mechanical loading with seismic inputs, but the two-bar and the bent tests showed in addition to the normal ratcheting predictions that you would get, some time dependent behavior, which caused us some concern in the beginning when we looked at it. The cycles were, as Bill pointed out, relatively very low frequency when compared to the earthquake's and since then we have done some additional work--Roy Williams has done some additional work at higher frequencies, and what he found was that indeed the extent of the time difference in cyclic deformation was much lower, so that gives us reason to believe that the time different type of behavior was more symptomatic of the very low frequencies, and is the kind of behavior that you would not expect at high frequencies.

MR. RODABAUGH: How high is higher?

MR. RANGANATH: He has done this at .5 cpm, and he went to ten times that frequency, which is five cycles per minute, and whereas the real earthquake you would be looking at five to ten hertz.

CHAIRMAN SIESS: What are you calling time dependent? What he called creep?

MR. RODABAUGH: Sam, I think there is something wrong with your first bullet equation, and your combination of mean stress and cyclic, dynamic stress, your bounds are not complete, but I'll write you a letter about it.

MR. RANGANATH: We can talk further about that, okay?

MR. RODABAUGH: Okay.

MR. RANGANATH: All right.

[Slide]

Failure is either by fatigue or excessive ratcheting. He observed some situations--

CHAIRMAN SIESS: You are still talking about the specimen tests?

MR. RANGANATH: --beg you pardon?

CHAIRMAN SIESS: You are still talking about the specimen tests?

MR. RANGANATH: I am still talking about the specimen test, where he had, after he had enough number of ratchet cycles, he found that in fact he could neck this specimen, even though he was doing fatigue testing, when he had enough ratcheting accumulated the failures became more like a rupture failure in a tension test, and so he had kinds of failure.

One, was where the classic symptoms of some nice, fine, smooth, shiney surface of fatigue failure, whereas the others were loss of ductility, rupture type of failures, okay.

So, what he found was when he excluded all of the data points where the failures occurred by necking and loss of ductility, he found that the data points fell very nicely on fatigue curve that is close to the mean data curve that is used in the ASME Code.

So, what it is saying is that as long as you don't allow your ratchet strain to get out of control, whereby might get failure by necking and excessive deformation, the ratcheting in itself does not have substantial impact on fatigue cracking.

CHAIRMAN SIESS: The ASME curve is a high cycle fatigue?

MR. RANGANATH: It is a combination of a low and high cycle, but they were all done with zero means stress and certainly did not have ratcheting.

So, this is an important conclusion because what it is saying is, well, ratchet is something to be worried about, as long as you have reasonable confidence that the total ratchet strain is small, you can still use the fatigue rules for prediction.

[Slide]

I am going to very quickly go over the single degree of freedom model analysis.

[Slide]

Some of it Sam has already covered, so I really don't have to go into in any detail. This is the standard single degree of freedom system, and we put the structural damping here, and if you were looking at elastic behavior, the kind of response spectrum that most of you--all of you are probably aware of--is what you'd get here.

CHAIRMAN SIESS: That is the one that he had to cut off down there by five.

MR. RANGANATH: That's correct, right.

Now, what we then did was we said the components are well described by single degree of freedom system, so what would happen if you do elastic plastic analysis and here is the elastic plastic analysis.

[Slide]

This is where we now did an analysis based on the elastic perfectly plastic behavior. This is the-- Sam Tagart described it, it is his model--and essentially what we did was we came up with an effective stiffness and an effective damping based on a hysteresis slope like this, and we performed the evaluation of the response of the system in an elastic manner.

To compliment that, we also did an elastic plastic model. This was a numerical model where we modeled the material by a bilinear spring, with two slopes. We assumed kinematic hardening, and we also later on applied a static force, in order to simulate what would happen in a ratchet type of condition, so this is what you would call as a-- with the limitation of bilinear stress strain curve, and the assumption of kinematic hardening what you call as an exact solution to the problem.

CHAIRMAN SIESS: Your "P" is his lambda?

MR. RANGANATH: That is correct, that is correct, yes. Lambda is a non-dimensional slope.

And again--

CHAIRMAN SIESS: Is kinematic hardening, of what English was talking about as a strain hardening fatigue curve sort of a behavior, or what?

MR. RANGANATH: The kinematic hardening is what I mean as the unloading and subsequent loading. It operates between these two lines.

MR. TAGART: It is the conservative description of hardening when talking about--there are two models talked about--isotropic hardening, and kinematic hardening. Isotropic hardening assumes that the whole--

MR. RANGANATH: The isotropic--

MR. TAGART: --expanse increases with cycles, and the kinematic ones does not increase with cycles.

MR. RANGANATH: The kinematic--

MR. SHEWMON: So, there is no work hardening?

MR. RANGANATH: --well, there is work hardening, but what it is what you gain on the tensile portion of the curve, you lose--in effect, it accommodates the--

MR. SHEWMON: It is strain hardening, but it doesn't change with number cycles.

MR. RANGANATH: Right, that is correct, so that is the assumption that was made in the elastisisa most materials, maybe after about ten cycles the initially show hardening, where you get increase in both tension and compression, but after a few cycles you don't get this continuous increase, and this is what you would have. [Slide]

So, again then to very quickly go over--Sam already went through this thing. What it is is here we are plotting the relative amplification. In non-dimensional terms this could be viewed as a strain in the spring, and this is dimensional frequency, and one thing we see very clearly is if you account for elastic plastic behavior, there is substantial reduction in the amplification. The peak strains are significantly lower.

And, what you do see, though, is the sift in the peak and that goes with the fact that the slope corresponding to plastic behavior, the effect of stiffness, is somewhat lower so you can see that there is a shifting to the left.

And, depending upon the extent of deformation that you allow, or in other words the ductility that you can tolerate, you see that if you go to higher ductility, in fact, the peaks are lower, but there is a region in the response spectrum where the elastic analysis can be somewhat non-conservative, and this is something we have talked about earlier.

CHAIRMAN SIESS: What is that less than one by elastic?

MR. RANGANATH: Well, what he is saying is he is non-dimensionalizing this with the--

CHAIRMAN SIESS: Oh, okay, I see.

MR. RANGANATH: --and he is saying that is less than one.

CHAIRMAN SIESS: Right.

MR. RANGANATH: So, that kind of gave us a good understanding, Sam's model couple with an exact analysis, and now we feel like we understands what happens in terms of the elastic behavior, inherently the higher the strain the more the damping, and therefore it acts like a self-limiting type of behavior.

Now, the second thing we did was to see if we could predict what would happen in terms of ratcheting.

[Slide]

You can see we used the same model with the elastic plastic behavior, you know, and in relation to that we applied a static force to simulate the mean stress, if you will, say due to pressure or dead weight in the system.

Again, we found, for example this is typical response here, this is the support motion. This F = .5 means that we applied a force equal to one half of the yield force on the spring, and L = .1 meaning the slope of the plastic portion is one-tenth the slope of the elastic portion.

So, with that, you can see the strain--dimensional strain, in fact, goes up very rapidly, and after awhile you end up just cycling once you obtain a mean value.

Now, if you look a the same thing--

CHAIRMAN SIESS: What was the mean value here?

MR. RANGANATH: This is what--this is steady ratchet strain.

CHAIRMAN SIESS: No, you said you applied a mean stress.

MR. RANGANATH: The mean stress was one-half of the yield.

CHAIRMAN SIESS: And, so this went into yield in both directions when you applied the cyclic?

MR. RANGANATH: Yes, I think it will show you here.

[Slide]

CHAIRMAN SIESS: Okay, sure.

MR. RANGANATH: This is the--this shows, for example, it is kind of a--well, we felt like we understood the model much better once we started looking at it from the simple spring mass system.

What it is, is the mean stress act like an asymmetric on the stress strain hysteresis load, so all you do, as long as you shift the hysteresis load so that it can support this mean stress, and now you have a symmetrically heavy load over the--so this hysteresis load in fact moves up along the cyclic stress change--

CHAIRMAN SIESS: You are only yielding in one direction, now?

MR. RANGANATH: There is yielding in the reverse direction, also, you can see.

Some of these data points--

CHAIRMAN SIESS: Very little, though.

MR. RANGANATH: There is some yielding, yes, but as you can see, more of it, of course, is on this side.

CHAIRMAN SIESS: This is dimensionless? What is it? Ratio to yield?

MR. RANGANATH: This is dimensionless, right.

CHAIRMAN SIESS: But, what is the ratio?

MR. RANGANATH: This is ratio to the yield strain.

CHAIRMAN SIESS: So, I look at the bottom part of that and I am not reaching yield. It looks more like a--

MR. RANGANATH: Well, this would be the elastic unloading and then--

CHAIRMAN SIESS: Well, how far does it go? That is what I can't tell.

I take one of the early cycles--

MR. RANGANATH: Un-huh.

CHAIRMAN SIESS: --and when you went up you moved over about a foot on that screen, and when you came down you essentially went very little--

MR. RANGANATH: Yes, yes, there is less yielding on the compression side than on the tensile side but there is yielding.

CHAIRMAN SIESS: Even though the stress is not yield?

MR. RANGANATH: That is correct. What it is, is you can yield on the compressive stress at a lower stress level because of this Bauchelder [sic.] effect that we are talking about.

CHAIRMAN SIESS: All right, when I get over--

MR. RANGANATH: Yes, kinematic.

CHAIRMAN SIESS: --to this right-hand side, is that one big loop there.

MR. WARD: Trace the loop on the right-hand side.

CHAIRMAN SIESS: I couldn't tell whether--

MR. RANGANATH: I think it is kind of hard to-- I think it is hard because it goes up like this and then back--

CHAIRMAN SIESS: Okay.

MR. RANGANATH: --and after awhile it just starts-- you can see these dark lines are when you no longer had a shift in the hysteresis loop it kept revealing itself, so this is the final hysteresis loop.

CHAIRMAN SIESS: The other thing that bothers me is why, if you have an applied mean stress, you still got your line going through the origin--

MR. RANGANATH: Now, this is what--here is the where we--here is the applied--

CHAIRMAN SIESS: Okay, that is your static point--

MR. RANGANATH: Right.

CHAIRMAN SIESS: --I see.

MR. RANGANATH: Right, right.

CHAIRMAN SIESS: You got up to there and then put the other one in--

MR. RANGANATH: And, now we are cycling this--

CHAIRMAN SIESS: And, what is the one that doesn't come down all of the way?

MR. RANGANATH: Here?

CHAIRMAN SIESS: The first unloading.

MR. RANGANATH: Oh, I think that was more--it is a plot and--we use Lotus to pick up points, so it is an aberration caused by the--we didn't pick up every point because there are so many points to calculate.

CHAIRMAN SIESS: That one should come down a little.

MR. RANGANATH: Right, that is correct.

So what it is showing is, this nice, simple, strain-stress model can in fact--we can understand ratcheting as it happens in a structure when you have primary mean stresses in seismic type of loading.

[Slide]

But, we did find that the predicted ratchet--

CHAIRMAN SIESS: Excuse me, excuse me.

Go back, because you have got something defined on that previous one. The diagonal line, sloping line, would be E of P.

MR. RANGANATH: Right.

What we found out was that the--you can make a prediction of the cumulative ratchet strain by just taking your applied mean stress and divide it by the plastic slope of the stress-strain curve.

CHAIRMAN SIESS: Is that the slope of the--

MR. RANGANATH: Right, this is the plastic slope of the stress-strain curve.

CHAIRMAN SIESS: Oh, that is the plastic slope of the stress-strain curve.

MR. RANGANATH: Right.

CHAIRMAN SIESS: It is almost parallel to the envelop.

MR. RANGANATH: It is parallel.

CHAIRMAN SIESS: It is parallel?

MR. RANGANATH: It is parallel.

CHAIRMAN SIESS: Okay.

MR. RANGANATH: So, we found out this was kind of a phenomenal observation that we could come up with a prediction for the hysteresis and it turned out that the Dan Miller Model which was based on thermal stresses gave us essentially the same kind of prediction.

CHAIRMAN SIESS: So, if I look at the top line there that would be what I would get in a static monatonic test?

MR. RANGANATH: This would be--yes, yes, if you used a, you know, a cyclic stress strain.

CHAIRMAN SIESS: Okay.

MR. RANGANATH: Okay.

[Slide]

Now, when we--

MR. SHEWMON: You might have to run it through several cycles.

MR. RANGANATH: Right, that is correct.

You see, because you may have a yield material that has somewhat lower yield strain.

[General discussion]

Right, but anyway, these are some of the conclusions then from the single degree of freedom system model. We felt that the single degree of freedom system model was somewhat conservative because we applied axial loads of tremendous stress, the real piping is subjected to more lending behavior, and the whole answer is, of course, a strong functional, how you compute this plastic slope in a bilinear model, depending upon which part of the stress-strain you match you can get different answers, but the usefulness of this is in terms of understanding what's happening in the ratcheting condition.

CHAIRMAN SIESS: Does this thing get very difficult when you take tri-linear or multi-linear curves?

MR. RANGANATH: I think--we did all of this on a personal computer, so I think we can--

CHAIRMAN SIESS: You have an equation, then you can--

MR. RANGANATH: I think so. We can do them much better.

And, I said, well, what do we do in terms of predicting ratchet strains for all of the component tests.

[Slide]

So, what we did was we wanted to plot all of the data that we had where we had ratcheting, and in many of these cases we didn't quite get the ratchet strain but we have enough number of data points that we use to predict the condition under which ratcheting would occur.

[Slide]

The next plot--it is kind of a busy plot--shows then the ratchet threshold. This was for carbon steel. Let me take a minute to explain it.

Here we are plotting the mean stress, that is the pressure stress, plus the strain times the Young Smartings [sic.], you know, okay, so this is, in a sense, if you ignore the mean stress, [illegible] 's almost all the seismic stress, or seismic strain, if you will, and these were several runs, so the axis really doesn't have any meaning. It is more like a bar chart.

CHAIRMAN SIESS: These are several different specimens?

MR. RANGANATH: Several different tests, or in some, several different runs on one specimen, you know.

And, what the engineer who did this analysis did was he looked at the actual records, and find the point when he got ratcheting, so these pluses are shown just the onset of ratcheting. Below the plus would mean he didn't see any ratchet; above that, you know, he saw some ratchet.

Then, the squares are points where--

CHAIRMAN SIESS: That is the strain at which he saw?

MR. RANGANATH: That is right, that is correct.

So, the squares are where the was ratcheting. All we wanted to do was, we wanted to find out--not only wasn't it enough to just say no ratcheting, we followed and concluded that we can tolerate some ratcheting, so what we said was--I took arbitrarily ten percent. I concluded that ten percent was the strain level that I could live with, for example, in a level D faulted effect. For a

Level B normal operation I may settle for this one, that is, I don't want to see ratcheting under normal conditions--

CHAIRMAN SIESS: Ten percent of what?

MR. RANGANATH: Ten percent strain.

CHAIRMAN SIESS: Oh, ten percent strain?

MR. RANGANATH: Ten percent strain--ratchet strain.

CHAIRMAN SIESS: Oh, I see.

MR. RANGANATH: So, this would be a faulted condition. The idea being if you have a faulted event, a one-time occurrence, then it is okay to have some ratcheted strain.

CHAIRMAN SIESS: And, the pluses are much, much less than ten percent?

MR. RANGANATH: The pluses, we could not get any--just the onset of ratcheting.

The open squares are less than ten percent, and the solid squares are greater than ten percent--again, ten percent is somewhat of an arbitrary number that I just use for the purpose of the discussion here.

And, I concluded that the ten percent I could live with it, if you had a one-time event.

And what the cycles show is they tell you how much ratchet increment occurs per cycle. I made the assumption that in a typical SSE event a very conservative number is to say there are 50 cycles, so I said 10 cycles of the highest--ten seconds at the highest stress strain and 500--so I came up with 50 cycles as the maximum number of peaks cycles that you could get. I think it is a conservative number, but that is what I use in making this plot.

So, then this told me that if I wanted to look at condition--first of all, we didn't get any limit load failures, so all we have concern about is ratcheting, and if you allow your stress to be below 2 Sm Y for example, then you would not get ratcheting, say for level b type of conditions; and if you allow your stress to be less than say 4 Sm Y then you would get some ratcheting, but the ratchet of strain would not be large enough to cause any concerns on structural integrity due to the faulted event.

And, again, the numbers 2 and 4 may in fact change over time, but this is the concept that I am proposing.

CHAIRMAN SIESS: Now, I don't recall having seen the number of cycles in the component tests. It is always giving the number of inputs. Did any of the component fail at less than 50 cycles?

MR. TAGART: Well, they are different amplitudes. I think 50 is a very large number for any of those component tests.

I would say that probably it is 50 is the upper limit of what we've seen in the component tests. We took five component tests and said you had ten maximum amplitudes equivalent cycles in those tests. That adds up to--

CHAIRMAN SIESS: Is that about what you got? About ten for an input?

MR. TAGART: Close to the maximum limit. I'd say ten was the upper limit.

CHAIRMAN SIESS: That is what I didn't know.

MR. TAGART: It is more like six, would be a better number.

CHAIRMAN SIESS: Yes, I see.

MR. BUSH: Sam, would you interpret your horizontal axis. I don't understand it.

MR. RANGANATH: All right.

The horizontal axis really does not have any meaning. They are all--that was plotted as bar charts, you know. These were all different runs, that's all. This was one set, where he ran it at half sled range, and then full range, and so on, but in itself the X axis does not have any meaning.

CHAIRMAN SIESS: Are those all of the tests?

MR. RANGANATH: We, in many cases, could not get meaningful data because the strain gauges came off.

He looked at all--many of them, and picked those where he had good data that went all the way to the higher stresses.

CHAIRMAN SIESS: Each plus is a different test?

MR. RANGANATH: Each test maybe a different run in a test.

Remember, they kept increasing the amplitude at one elbow, with a lower forcing function and they kept increasing it, so each point is a different run there.

CHAIRMAN SIESS: Okay.

MR. RANGANATH: Now, I did the same thing.

[Slide]

This was for carbon steel, but I express it in terms of Sm Y--

MR. RODABAUGH: Sam, before you say it, for the benefit of the subcommittee, when you divided by Sm Y, what Sm Y were you using?

MR. RANGANATH: I use the Code minimum Sm Y, okay. This material probably has Sm Y levels that were higher than the Code minimum value.

To that extent it can be construed as the somewhat non-conservative, but when you also recognize that when this thing is going through the cyclic behavior it doesn't take a whole lot of time before it strain hardens itself, so I don't know that that distinction is that critical.

MR. RODABAUGH: I think that you would be better off to use your best estimate of Sm Y from your material mill test reports.

CHAIRMAN SIESS: Would that vary with the different tests?

MR. RODABAUGH: Yes.

MR. RANGANATH: Yes, it would vary for different tests, and--

CHAIRMAN SIESS: If the materials were enough different that the actual yield stress, compared to the specified minimum would vary?

MR. RANGANATH: I would say most cases the specified minimums were much lower than the actual--

CHAIRMAN SIESS: No, that is not my point.

If you went in with each of these specimens, and used the specimen--used the actual yield stress, measured yield stress, was the ratio of measured yield stress, the specified minimum--or Code stress as you call it--would that be fairly constant, or would these different heats--

MR. RANGANATH: I would guess it is constant, but I will let Kelly answer it. He probably has a better idea.

MR. MERZ: Some of the specimens were taken from the same heat, okay. The mill test is for that heat, okay.

In terms of elbows, tees, pipe, I would say there is probably 20 different heats, okay, all with probably about 10 KSI above the specified minimum, approximately.

CHAIRMAN SIESS: Mill test.

MR. MERZ: Yes, above the mill test.

Now, that is the mill test, of course, is for the specimen before it is forged, not after it is forged, really. It is for the piece of pipe that it is forged from, usually.

Now, Mr. Rodabaugh can probably answer that more correctly.

CHAIRMAN SIESS: Since all you ever know, in designing, is the Code specified minimum, the question then remains is this a representative sampling of the variations you could get between the specified minimum properties and the actual properties?

MR. RANGANATH: All right--

MR. RODABAUGH: Yes, that is the question, Chet.

Their materials, rather typically, were about 20 percent higher yield strength, for example--20 or 25--but if you look at statistics on the materials, like stainless steel, you will find some small percent, one percent, is right at the minimum, even a little percent below minimum.

CHAIRMAN SIESS: What I was thinking is, that suppose you went back and took say mill tests, which are not the right answer, they are not the properties the material is formed into the elbow, but supposed you took the mill tests that did fluctuate, and you did this and your scatter decreased? Let's just be real optimistic and say, you know, you have got all of these in a very narrow band, what would you do with them? You still wouldn't be able to use--

MR. RODABAUGH: I think, Chet, the very next slide is going to be the key to my thought here.

Eventually these stress limits are going to be shown in terms of Sm Ep. Now, that is a minimum. If it turns out that--

CHAIRMAN SIESS: What is that?

MR. RANGANATH: S of M is a design stress intensity.

MR. RODABAUGH: Design stress intensity, which may be either based on yield strength, or ultimate strength.

In general, the Code philosophy is to base your design on the minimum properties. Now we have got some tests which are tests of components that had higher than minimum properties. A rather straightforward completely defensible way to evaluate the data is to adjust it to the difference between what you tested and the Code values, then you have a very straightforward four-story and you don't have to go back and say, " Well, if we had a minimum, we still think we are in our stato--

MR. RANGANATH: You are right, and in fact one of the action items that I have is to redo these results in terms of the actual--

CHAIRMAN SIESS: Then you can decide whether you want a five percent cutoff or a ten percent value.

MR. RANGANATH: And, again that is--the ten percent that I took was arbitrary--

CHAIRMAN SIESS: Not on the stress, the yield stress.

MR. RANGANATH: Yes.

Now, here is the same plot for carbon steel that was based on S of M.

[Slide]

And, again this is showing about 6 S of M may be okay for Level D; maybe 3 S of M is okay as opposed to the current limit for level B, which is 1.8 S of M, you know, so again it indicates there is some room for improvement.

[Slide]

I'll show you--

CHAIRMAN SIESS: You mean the solid squares then to represent level D?

MR. RANGANATH: Rather than ten percent, right--no, solid squares are the stress amplitude that I have to have--

CHAIRMAN SIESS: Yes, but I then you said you thought 6 was good enough for--I thought you said level B?

MR. RANGANATH: Level D, right.

CHAIRMAN SIESS: Level?

MR. RANGANATH: D, faulted.

CHAIRMAN SIESS: And, 3 for what?

MR. RANGANATH: Normal, level B--no upset requirements, upset, upset.

So you can do the same thing again. For stainless steel we kind a somewhat fewer points on stainless steel, and again the same comments that Ev brought up apply here, too.

This is expressed in terms of S of Y, and here we have the same thing expressed in the terms of S of M.

MR. SHEWMON: Tell me again what you have been telling me for ten minutes.

I don't understand--we've gotten an S mean, plus some small, I hope, adjustment to the elastic treatment--

MR. RANGANATH: Oh, no.

All we did was S mean is the pressure stress.

MR. SHEWMON: Okay.

MR. RANGANATH: The strain amplitude is the strain amplitude that was measured, from constrain gauges, you know. It was multiplied by E, right, to get the pseudo-elastic stress.

MR. SHEWMON: I understand that.

Then this is the ratio of that times the design stress--

MR. RANGANATH: Right.

Maybe it would be a better way to look at it would be in terms of S of Y. That means four times S of Y.

MR. SHEWMON: Well, but there is always as many pluses above your lines as there are below, so I don't quite see any threshold for ratcheting.

[General discussion]

CHAIRMAN SIESS: You can get the open squares for his bottom line margin, that is ten percent--less than ten percent, and the solid squares for his top ones.

MR. SHEWMON: Right, right.

MR. RANGANATH: See, we have--let's look at no ratcheting at all.

That means I should not have any squares. I did have two squares here. When we went back to check it the strain was very small. It was almost close to no ratchet, so I am saying--on the level B conditions I wouldn't want to see any ratcheting, and that is saying that maybe 2 S of Y is a good limit for level B.

For level D, I will say, yes, I can tolerate some ratcheting, and I just arbitrarily picked ten percent in 50 cycles, and that saying if I am below 4 S of Y--

MR. SHEWMON: Okay, fine.

MR. RANGANATH: --I wouldn't get ten--

MR. SHEWMON: Thank you.

CHAIRMAN SIESS: Now, you know that line, you could have drawn at five, couldn't you?

MR. RANGANATH: That is what Sam Tagart also asked me, you know, and I can do that, and again I just picked ten for this case. We can do that. I think it would be one of those numbers that will have to check with our consultants and come up with something that is acceptable.

So, with that we sent 3 S of M for level B, and 6 S of M for level D, okay.

[Slide]

So, let's take a look at design rule changes that we might propose. And, one design rule that--this is the pressure, remember I told you--showed you the equation PD over 4P plus M over Z, where M is the earthquake moment. Now, right now, that includes the seismic loading also, and these are the current limits. It is the lesser of this or this, that is what we have today.

And, what we might change is--

CHAIRMAN SIESS: Now, 3 S of M, if it is two-thirds S Y, that is what the right-hand side governs?

MR. RANGANATH: Yes, close to.

CHAIRMAN SIESS: Okay.

MR. RANGANATH: In some cases, you know, it is a little different because it is the lower of one-third of alternate, or two-thirds of yield, and so this changes.

CHAIRMAN SIESS: So that is equal really to ultimate or two times the yield, if I look at level D?

MR. RANGANATH: Yes.

CHAIRMAN SIESS: Right?

MR. RANGANATH: Yes.

So, that is where we are today. That includes the seismic loads also.

What we might look at in terms of a proposal for new Code limits, you don't hav to do any special analysis for ratcheting, and so on, provided you make these limits for, you know, with the new proposal, 3 S of M to 6 S of M, 3 for level B, and 6 for level D. And, again this is something that we would probably have to reiterate on, but this is the direction we are going to.

CHAIRMAN SIESS: And, 6 is two times ultimate?

MR. RANGANATH: Right.

CHAIRMAN SIESS: Okay.

MR. RANGANATH: And, again the numbers we are talking about are pseudo-elastic stress.

CHAIRMAN SIESS: Oh, yes.

MR. RANGANATH: It is measures of strain.

CHAIRMAN SIESS: I know, it is a measure of strain caused by--

MR. RANGANATH: Yes, and the non-seismic loading portion will still be limited by the current Code, so weight, water hammer, and pressure, and so on, would still be controlled by this.

So, you know, how can we implement it in the Code? One thing we were looking at was we have some options, and this again we will have to decide on it with some more interpretation. Should be make a blanket change in the Code? Or, should we be more restrictive in terms of a Code case approach where we say you can use these rules provided you use response spectrum analysis, provided you use peak broadening, provided you don't use damping in excess of five percent, provided you don't have castings, and so on, so that may be one way to look at it.

[Slide]

And, right now we will not need any firm decisions on which way to do, the above spells it out.

[Slide]

And, finally, looking at it from the long term viewpoint, where, you know, do we go from here? Clearly the current Code approach has been shown to be very conservative, and it is kind of does not make sense, and one of things that the limits on the primary stresses, on equation 9, lead to excessive snubbers.

Snubbers, and so on, very fancy analysis, and we can probably reduce design costs, hardware costs, and really other potential problems associated with snubbers--failing and so on--can be reduced, and so I think we can get a simple, more cost effective system--

CHAIRMAN SIESS: Now, you had a caveat before about the response spectrum analysis, so you are not going to change the analysis costs there much?

MR. RANGANATH: Oh, okay, in the long term, that kind of let's me up to the third bullet--

CHAIRMAN SIESS: Let's stay up at the second bullet.

I am trying--

MR. RANGANATH: I agree.

CHAIRMAN SIESS: --to visualize, given those Code changes in stresses, what would be different in the plant? What would be different the design? What would be different physically in the plant?

They would have fewer snubbers and steel pipe supports--

MR. RANGANATH: Right.

CHAIRMAN SIESS: Would the pipe sizes be different?

MR. RANGANATH: No.

CHAIRMAN SIESS: The schedule wouldn't be different?

MR. RODABAUGH: No.

CHAIRMAN SIESS: That is based on pressure, so it would be mainly supports and snubbers--

MR. RANGANATH: Right.

CHAIRMAN SIESS: --and the maintenance that goes with snubbers--

MR. RANGANATH: That is correct.

CHAIRMAN SIESS: --and the problems that go with supports and so forth?

MR. RANGANATH: That is right.

MR. GUZY: It should be noted that the designing of the piping system, support design, is the major factor in this, so you have less support design--

CHAIRMAN SIESS: Yes, I know.

MR. RANGANATH: And, in the long run--

CHAIRMAN SIESS: Thousands--

MR. RANGANATH: --as I--

CHAIRMAN SIESS: --you didn't finish your last bullet there.

MR. RANGANATH: Yes, I will get to that in just a second.

As you can see here, if you look at the elastic plastic response they are almost getting to a factor of 1, you know, so what this is saying is--and a lot of the experts like Bob Kennedy are saying, you know, the old way of doing things where we just did it in a static manner, and in fact is okay, and this elastic plastic--

CHAIRMAN SIESS: An equivalent static.

MR. RANGANATH: --analysis has shown that that is okay.

CHAIRMAN SIESS: An equivalent static.

MR. RANGANATH: --right, equivalent and static method.

CHAIRMAN SIESS: At the workshop on Appendix B that was held a couple of years ago, somebody proposed an equivalent static. Was it Bob Kennedy?

MR. TAGART: I am not sure about that meeting, but equivalent static has always been in the Code. The problem with it is it has always taken a 1.5 factor peak of the response spectrum as the equivalent static--

CHAIRMAN SIESS: Well, that is not equivalent static, then--

MR. TAGART:--and that is just way too conservative--

CHAIRMAN SIESS: Yes.

MR. TAGART: --and recently there is a proposal by John Stevenson to introduce an equivalent static approach that is more realistic.

CHAIRMAN SIESS: Thank you, that is who it was.

And, then somebody suggested, you know, simplify that and put a little more effort into steam generator supports, and vessel supports, in places where the consequences would really be serious.

It was Stevenson at that meeting, I guess.

Well, I tell you, there are real benefits in getting back to that sort of an approach, because you have got a heck of a lot better feel for what--

MR. RANGANATH: Bring in the understanding.

CHAIRMAN SIESS: --you are doing, yes, I mean this stuff all comes out of the damn computer and if there is a mistake then nobody has any feel for it.

MR. SHEWMON: Tell me what drives snubbers now? Is it displacement? Or does decrease mean stress? Or what?

MR. RANGANATH: Stress-strain reduction is the one that drives, although--

MR. TAGART: Equation 9 right now, which is a stress equation on the pipe.

MR. SHEWMON: And, so if you put a snubber in you decrease the dynamic stress?

CHAIRMAN SIESS: You also change the frequency, too, don't you? Shift off of the peak.

MR. SHEWMON: Now, the frequency didn't come into that equation, as he drew it?

CHAIRMAN SIESS: No, but in the analysis it does.

MR. SHEWMON: Yes.

CHAIRMAN SIESS: For smaller pipes, it just support it to get it above 33 hertz, and there is no amplification.

MR. RANGANATH: That is the last one, some kind of a design by rule test, Sam was talking about, and the whole idea is mainly these experts believe there is a very simple ways of designing piping that is quite effective and still maintain the safety margins.

CHAIRMAN SIESS: Well, again, don't we have some evidence from SKRUG [sic.] that at best was design equivalent static?

MR.TAGART: Yes, there is a lot of evidence to that effect.

CHAIRMAN SIESS: What causes the failure?

MR. TAGART: The only problems that would cause failures is where you have seismic anchor motions that weren't accounted for in the design.

CHAIRMAN SIESS: Yes, but you can do those by equivalent static and probably better than anything else can, and the anchors are something else.

MR. GUZY: Also, connections with the piping, when we talk about threaded piping, then there may be a problem.

CHAIRMAN SIESS: Okay.

MR. SHEWMON: To go back and try to understand these two curves, I find they come in pairs, where the data points are the same but the answers change.

MR. RANGANATH: That is correct.

MR. SHEWMON: Or the numbers on the left axis--

MR. RANGANATH: Right, we have the same data point, and in one we express it as a ratio of the yield strength, and the other way we express it--

MR. SHEWMON: Okay, fine.

MR. RANGANATH: --as the ratio of the design stress--

MR. SHEWMON: He just defines strength--

MR. RANGANATH: Thank you.

CHAIRMAN SIESS: Well, I have got on comment about the proposed Code changes: lots of luck!

It is hard to change it, but I think everything we know says that we are awfully conservative in what we are doing, and not necessarily that we are getting a conservative power plant as the result of it, and that kind of bothers me, and if there is some way of arriving at a better design that is just as safe I sure we could do it.

Gentlemen, that concludes the presentations that are being made to us, except Dan had one slide left he wanted to bring in at the end. I am not sure if he forgot about it or not.

MR. GUZY: It was just to talk about the rest of the research that we are doing.

From a research point of view, I just sort of wanted to put this into context along with some other things that we are doing in piping.

First of all, we are developing a new piping research plan that will include--the type of work we talked about today--response design rules for piping, as well as cracked piping, so that there will be a forthcoming NUREG, and the pad has only addressed, like the degree of piping program now, only it will be more comprehensive and will include things we are doing that use to be in the old cycling plant--that should be familiar to you.

In addition to this work, which 1 consider the most important and certainly the major part of our research in the design area, we are still doing work in the piping response method area. We have done work in the past on damping. We are supporting work that is being done now at Vectal [sic.] and through EPRI on the new damping criteria, and there will be some forthcoming actions on the Code bodies to revise this Code case, and hopefully to make it easier to use. Right now only half the plants can qualify for using the damping criteria--

CHAIRMAN SIESS: Why?

MR. GUZY: Because the NRC in their endorsement of Code case N-411 put a number of limitations on it, for instance, you have to use a modern spectra, ground spectra. If you use a--

CHAIRMAN SIESS: Oh, yes.

MR. GUZY: --and that can't be used in the conservative.

Also, there is activity now in the ISM response method area, and we are doing some regulatory and Code work on that, hopefully to resolve it and bring some of this research to a head, a change in what our current position is.

There is also doing some work on the treatment of high frequency modes, how you can--with response analysis, and better treatment of closely spaced modes response analysis, and there is work in the nonlinear response area, supportive work at federal. There is some--a number of small projects in response to this area, so are continuing to do work in these areas.

I think in the response margin methods areas, it is not as big as it was a couple of years ago.

CHAIRMAN SIESS: Now, these two middle bullets, is that going to involve experiments?

MR. GUZY: We can use some experiments as benchmarks, but primarily analytical.

One of the advantages of these most recent system tests is that we do have physical evidence for support, and in the past there hasn't been that many piping benchmarks that included that, so this is good.

CHAIRMAN SIESS: The high frequency thing, I guess I will never quite understand the concern about some of the high frequency because of such small energy content.

MR. GUZY: That's--I mean, our program, the test program is showing that they are not of much concern. If we could make it go away, using these results, then we wouldn't have to worry about it, but on paper high frequency modes do pose a problem at least in licensing cases, and there may be some better way to combine--it has to do with correlation of the modes and the plants, and maybe there is some analytical way we can do that through research.

Okay, my point was that piping response methods, we have done work--a lot of it recommended by the piping review committee--that work is kind of winding down, but there still are things that we are doing.

We have a major effort now at Oakridge, research is sponsoring now at Oakridge on nozzle flexibility and design, and there has been--it is having some impact now on some of the Code activities, and I think it will have a future impact.

There has also been work that EPRI has sponsored for this area. There are some definite Code activities going on now, in the nozzle area, to provide some relief. The point is that when we change the piping stress rules then nozzles, or supports we are talking about here, in the nozzle area we are doing something, and the support design will be a new area, and we are just trying to get our hands around this now.

There is some PBRC activities, we will make recommendations, and improve the support design. EPRI is having a workshop on support design every month--

CHAIRMAN SIESS: Pipe supports?

MR. GUZY: Pipe support, pipe support design. There is--

CHAIRMAN SIESS: How do they determine the force?

MR. GUZY: How to design what you know of the forces, okay, so that this is--

CHAIRMAN SIESS: That is a structural engineering problem that I thought the steel people had solved a number of years ago.

MR. BUSH: They have to a degree, in fact, the PBRC effort would end up looking at a package with the suggestion that effectively what you do is remove NF from the Code, which represents a tremendous load, particularly for inspection and so forth.

CHAIRMAN SIESS: It seems to me that you've got two choices on support design: you design them to take the loads without yielding; and the other is you design them to yield and absorb energy.

MR. GUZY: That is the--

CHAIRMAN SIESS: The first one people ought to know how to do, and the second one is a little bit more of a problem.

MR. GUZY: There are a number of areas that are involved in support design, one is the concept of supports that are used, you know, when they fail they'd better fail first.

What I am trying to note is that we are planning on doing research in this area and there are a number of recommendations to be made.

CHAIRMAN SIESS: Yes.

MR. GUZY: The use of piping experience data, we are doing a test through Oakridge in this area, research is, and we are following a project that EQE had done for EPRI, which followed the project that Don Stevenson had done for us. We are still trying to use the piping experience data from SCRUG [sic.]--essentially the same plants as SCRUG [sic.] and essentially to try and bring it into more of a regulatory process.

MR. BUSH: That sounds like you are limiting it pretty much to seismic response?

MR. GUZY: Yes.

MR. BUSH: And not just piping experience--

MR. GUZY: Yes, there is some element of what the operating conditions were and how it is inspected and if you can show that you are enveloped by industrial plants, then you can feel more comfortable.

In terms of degraded piping, or the IPIRG program, it does have an element of--

CHAIRMAN SIESS: What is IPIRG?

MR. GUZY: --it is the International Degrading Piping Program, but I don't know what that--

CHAIRMAN SIESS: Okay.

MR. GUZY: But, it will include an element of dynamic testing, in fact, there is some simple tests now set up for essentially dynamic failed piping with known cracks in it--

CHAIRMAN SIESS: Degraded piping.

MR. GUZY: Degraded piping, yes.

Then the last bullet is piping liability studies. Sam showed you earlier some things that EPRI has sponsored. We, NRC, may become more involved in this also. We see that way of integrating new piping information, like what we've just talked about, the program, and say information

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on the degraded piping will be--probably could be most economically once we know the basis data on it and--

CHAIRMAN SIESS: Now, is that PRA related?

MR. GUZY: --I don't know how probabilistic it will be.

I see us as maybe improving our full commercial type program. Sam would see it as an extension of the things he was talking about this morning.

CHAIRMAN SIESS: Now, you skipped one.

MR. GUZY: I skipped one? Oh, cumulative effects of piping criteria changes, this is something that the licensing staff has asked for for a long time and we had a hard time getting our arms around it.

The way it is envisioned now would be sort of a response margins approach where you would show how you would trade these, and we've had a hard time getting this off of the ground, and my feeling is if we can effectively use the information from this test program we may not have to do that.

CHAIRMAN SIESS: Well, I think that is very important, because I think that anything we can do to improve piping design, reduce some of the problems we have with snubbers, is excellent, but we have gotten an awful lot of comfort out of the margins we have.

Now, we are finding the margins are tremendously large, but let's don't get the margins down to the point where when somebody wants to up an earthquake hazard, we get another earthquake in the eastern U.S., now our comfort has disappeared, and that comfort is hard to qualify. It is a fairly important aspect of this, and we make all of these changes and somebody needs to take a look and say, "Now, okay, where are we now?"

I don't know if that is easy to do, but--

MR. GUZY: I think the last project, the piping reliability studies, which would attempt to do that--

CHAIRMAN SIESS: I think that will tie in.

MR. BUSH: Dan, could I have a comment, because--

CHAIRMAN SIESS: You don't have margin in there, as such, but that is what I am thinking.

MR. BUSH: It depends on what you put in the last one, because as I visualize it what you really need is something--a cut across, at a minimum--three of those: piping experience data, the cumulative effect of changes, and the piping reliability study, and somehow they have to be integrated.

MR. GUZY: Yes, and I think--

MR. BUSH: And, if they don't, if they aren't integrated, you may not accomplish what you need.

MR. GUZY: Yes, I think that ideally, if we can get something going that we should include--especially the piping experience, I think that is something that has to be brought in more than it has been in the past.

CHAIRMAN SIESS: Would the piping reliability studies take the cyclic approach?

MR. GUZY: I think they would take--cyclic, maybe like Sam was showing this morning, maybe some frequency of core melt type of--

CHAIRMAN SIESS: Well, yes, okay.

That approach just emphasized the uncertainties.

MR. TAGART: Quantifies and deals with the uncertainties.

MR. GUZY: So that is briefly what we are doing in my branch.

CHAIRMAN SIESS: Now, this is all--what's current, and what's for the future?

MR. GUZY: Well, I would--let's see--everything is sort of current, except for support design, which should happen fairly soon. The cumulative effects, which we had not started and the piping reliability studies, which Sam has already started--

CHAIRMAN SIESS: And, these are all withstanding the budget cuts?

MR. GUZY: Yes.

CHAIRMAN SIESS: I think that concludes the presentations.

Are there any questions or last words?

[No response.]

I would hope that Mr. Rodabaugh, who has threatened you with a letter will provide us with a copy, and I would appreciate anything from Spence Bush in the way of comments on the meeting that you can pass on to the rest of the committee.

MR. GUZY: Will the committee be making recommendations? Or what will you do?

CHAIRMAN SIESS: You haven't asked for any. We don't now have any particular input into the budget. If it comes up, obviously, we would be prepared.

MR. GUZY: We've heard your concerns over cast versus stainless steel, and that will be addressed in the program--

CHAIRMAN SIESS: What we will do is I will probably make a brief report at the next full committee meeting, and if they are interested in hearing more about this as a committee we might ask for some presentation at a future committee meeting.

I think it might wait until it gets to some regulatory action, you know, if we brief the full committee on what's going on now, and the regulatory action comes two years from now, we start over.

I think the subcommittee should be kept abreast, but to try and keep the full committee addressed in advance doesn't always work. Then other things come up with them.

Thank you, gentlemen. It has been very fine presentations. I think you covered a lot of territory and answered the questions we had today, and I say lots of luck with the Code changes.

Adjourned: 12:30 p.m.

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REPORTER'S CERTIFICATE

STATE OF CALIFORNIA)
) ss.
COUNTY OF VENTURA)

I, PRISCILLA PIKE, an official hearing reporter for the State of California, do hereby certify that the foregoing pages 1 through 162, inclusive, constitutes a true and correct transcript of the matter as reported by me.

I FURTHER CERTIFY that I have no interest in the subject matter.

WITNESS my hand this 2nd day of April,in the County of Ventura, State of California.

Priscilla Pike

Priscilla Pike
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Ventura, California 93001
(805)568-7770

~~Sam Tagart~~
Dan Guzy —

PIPING AND FITTING DYNAMIC RELIABILITY PROGRAM

EMPHASIS

DESIGN OF PIPING COMPONENTS FOR DYNAMIC INERTIAL LOADS

OBJECTIVES

IDENTIFY DYNAMIC FAILURE MECHANISMS AND LEVELS

PROVIDE HIGH-LEVEL NONLINEAR RESPONSE DATA

DEVELOP IMPROVED ASME CODE DESIGN RULES

PIPING AND FITTING DYNAMIC RELIABILITY PROGRAM

EPRI	SAM TAGART, Y. K. TANG
NRC	DAN GUZY
GE (SAN JOSE)	BILL ENGLISH, HENRY HWANG, SAM RANGANATH, ED SWAIN
ANCO ENGINEERS	PAUL IBANEZ, KELLY MERZ
ETEC	RON JOHNSON, VINCE DEVITA
MCL	ROY WILLIAMS
CONSULTANTS	E. RODABAUGH, R. KENNEDY, D. LANDERS, R. L. CLOUD, D. MUNSON, S. MOORE, R. BOSNAK, L. SEVERUD

PIPING AND FITTING DYNAMIC RELIABILITY PROGRAM

TASK 1: PROGRAM PLAN DEVELOPMENT (GE)

TASK 2: PIPE COMPONENT TESTING (ANCO)
- 41 FAILURE TESTS OF ELBOWS, TEES, ETC.

TASK 3: PIPE SYSTEM TESTING
- PFDRP "SYSTEMS 1 & 2" TESTS (ETEC)
- OTHER SYSTEM TESTS (ETEC)
- WATERHAMMER SYSTEM TESTS (ANCO)

TASK 4: SPECIMEN FATIGUE RATCHETING TESTS (MCL)
- 140 SPECIMENS, DIFFERENT MATERIALS & TEMPERATURE

TASK 5: - ANALYSIS OF TESTS AND DESIGN RULES (GE)

TASK 6: - IDENTIFICATION AND DEVELOPMENT OF ALTERNATIVE DESIGN RULES (GE)

TASK 7: - EVALUATION OF ALTERNATIVE DESIGN RULES (GE)

TASK 8: - PROJECT FINAL REPORTS (GE)

PIPING AND FITTING DYNAMIC RELIABILITY PROGRAM

STATUS AND SCHEDULE

PROGRAM INITIATED IN SPRING OF 1985

ALL TESTING COMPLETED (EXCEPT RETEST OF SYSTEM 1)

FINAL ANALYSES AND CRITERIA DEVELOPMENT UNDERWAY

PROGRAM ENDS JUNE 1988 WITH DRAFT FINAL REPORTS

INITIATION OF REVISIONS TO ASME CODE NB/ND/NC-3600 IN 1988

EPRI TO PUBLISH FINAL REPORTS

PIPING AND FITTING DYNAMIC RELIABILITY PROGRAM

COOPERATIVE EPRI/RES RESEARCH AGREEMENT

FIVE REVIEW MEETINGS WITH PROJECT MANAGERS AND CONSULTANTS

INTERACTIONS WITH ASME AND PVRC STANDARDS GROUPS

- PRESENTATUONS AT MEETINGS
- MEMBERSHIP ON GROUPS BY PFDRP PARTICIPANTS
- ASME CODE CLASS N-451
- CLASS 2 & 3 DYNAMIC ALLOWABLE CODE CASE
- PVRC TASK GROUP ON PIPING FUNCTIONALITY

PUBLICATIONS

- PAPERS IN JOURNALS AND SMiRT
- FOUR SEMI-ANNUAL PROGRESS REPORTS
- FINAL REPORTS TO BE ISSUED BY EPRI

ARCHIVING OF TEST SPECIMENS AT NDE CENTER

PIPING AND FITTING DYNAMIC RELIABILITY PROGRAM

NRC PERSPECTIVE

PIPING REVIEW COMMITTEE

- RAISED CONCERNS ABOUT OVERCONSERVATISMS IN INERTIAL LOAD DESIGN
- REGULATORY CHANGES LIMITED TO RESPONSE CRITERIA (E.G., DAMPING)
- IDENTIFIED HIGH PRIORITY NEED FOR FAILURE TESTS (NUREG 1061 VOLS. 2 & 5)

PFDRP PRESENTATIONS TO NRC

- INFORMATION DISTRIBUTED, VIDEO TAPES SHOWN
- BACKGROUND PRESENTATIONS AT CODE CASE N-411 AND N-451 MEETINGS
- WATER REACTOR SAFETY INFORMATION MEETINGS
- 9/11/87 FORMAL BRIEFING TO STAFF
- 3/30/88 MEETING OF ACRS SUBCOMMITTEE ON STRUCTURAL ENGINEERING
- FUTURE MEETINGS WITH NRC STAFF

REGULATORY CHANGES

- R.G. 1.84 ENDORSES ASME CODE CASES
- 10 CFR 50.55A INCORPORATES SPECIFIC CODE ADDENDA & REVISIONS
- S.R.P. CRITERIA FOR FUNCTIONALITY CRITERIA, ETC.
- PFDRP RESULTS WILL PROVIDE "FAILURE MARGINS" DATA FOR OTHER REGULATORY ACTIONS

OTHER RES PIPING DESIGN RESEARCH

FORTHCOMING RES PIPING RESEARCH PROGRAM PLAN (NUREG-1222)

- PIPING RESPONSE METHODS
  - DAMPING
  - INDEPENDENT SUPPORT MOTION (ISM) METHOD
  - HIGH FREQUENCY, CLOSELY SPACED MODES
  - NONLINEAR RESPONSE PREDICTION
- NOZZLE FLEXIBILITY AND DESIGN
- SUPPORT DESIGN
- PIPING EXPERIENCE DATA
- CUMULATIVE EFFECT OF PIPING CRITERIA CHANGES
- IPIRG
- PIPING RELIABILITY STUDIES

PIPING AND FITTING
DYNAMIC RELIABILITY PROGRAM
EPRI INTRODUCTION AND OVERVIEW

FOR

ACRS SUBCOMMITTEE ON
STRUCTURAL ENGINEERING

PACIFICA HOTEL
CULVER CITY, CALIFORNIA

MARCH 30, 1988

PRESENTED BY

SAM W. TAGART JR.
EPRI

TOPICS

- BRIEF HISTORY OF CODE RULES
- WHAT WE KNEW IN 1985
- SIMPLE ANALYSIS EXPLAINING NO STATIC COLLAPSE
- SUMMARY OF WHAT WE KNOW IN 1988
- THE OPPORTUNITIES AND THE CHALLENGES

HISTORY OF PIPE DESIGN

- 1952 - MARKL FATIGUE TESTS FOR B31.1 (SEMI STATIC)
- 1963 - NUCLEAR P.V. RULES (STATIC AND FATIGUE LOADS)
- 1968 - NUCLEAR PIPING RULES (STATIC, DYNAMIC* AND FATIGUE LOADS)
- 1975 - JAPANESE RESEARCH SHOWS LARGE DYNAMIC MARGINS AND FATIGUE RATCHET FAILURE MODE FOR PIPING (NO COLLAPSE)
- 1982 - PVRC PROGRAM TO IMPROVE PIPING
- 1985 - NUREG 1061 NRC PIPING RECOMMENDATIONS
- 1988 - EPRI/NRC PIPING DYNAMIC TESTS (BASIS FOR NEW RULES)

* EFFECTS OF DYNAMIC LOADS WERE HANDLED BY STATIC FAILURE CRITERIA (CONFIRMED BY GREENSTREET TESTS)

WHAT WE KNEW IN 1985

- PIPING DYNAMIC MARGIN WAS LARGE BUT UNCERTAIN
- PIPING FAILURE MODE FOR REVERSED DYNAMIC LOADING IS RATCHETING AND FATIGUE (NOT STATIC COLLAPSE)
- REDUCTION OF PIPING CODE MARGINS REQUIRED CONVINCING EXPERIMENTAL EVIDENCE PLUS ENGINEERING UNDERSTANDING
- MODERN NUCLEAR PLANTS HAD TOO MANY SNUBBERS

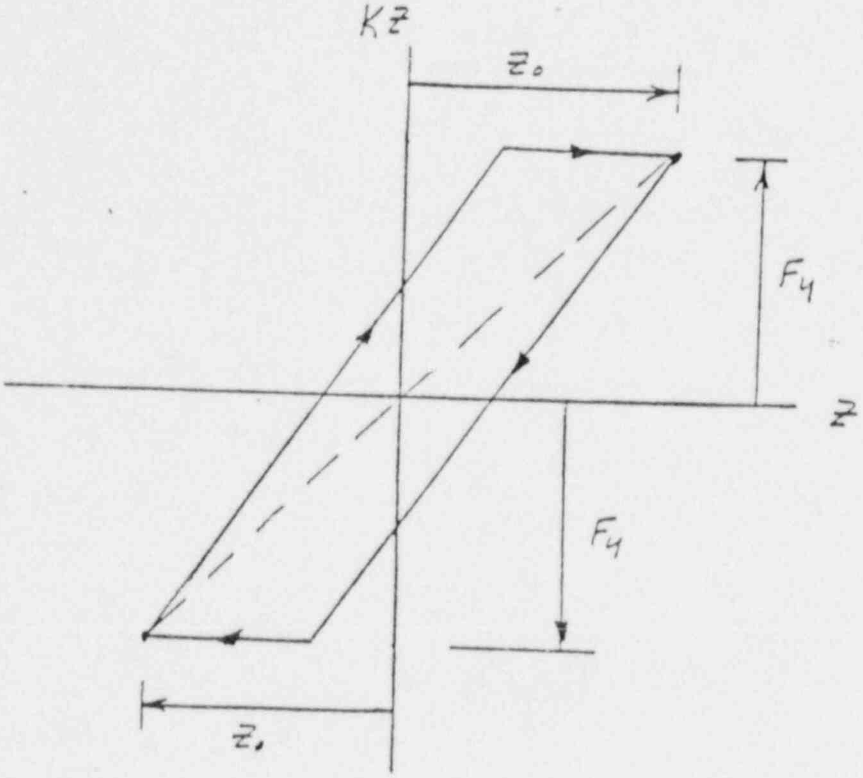

Figure 2

Restoring Force In
Spring During 1 Cycle

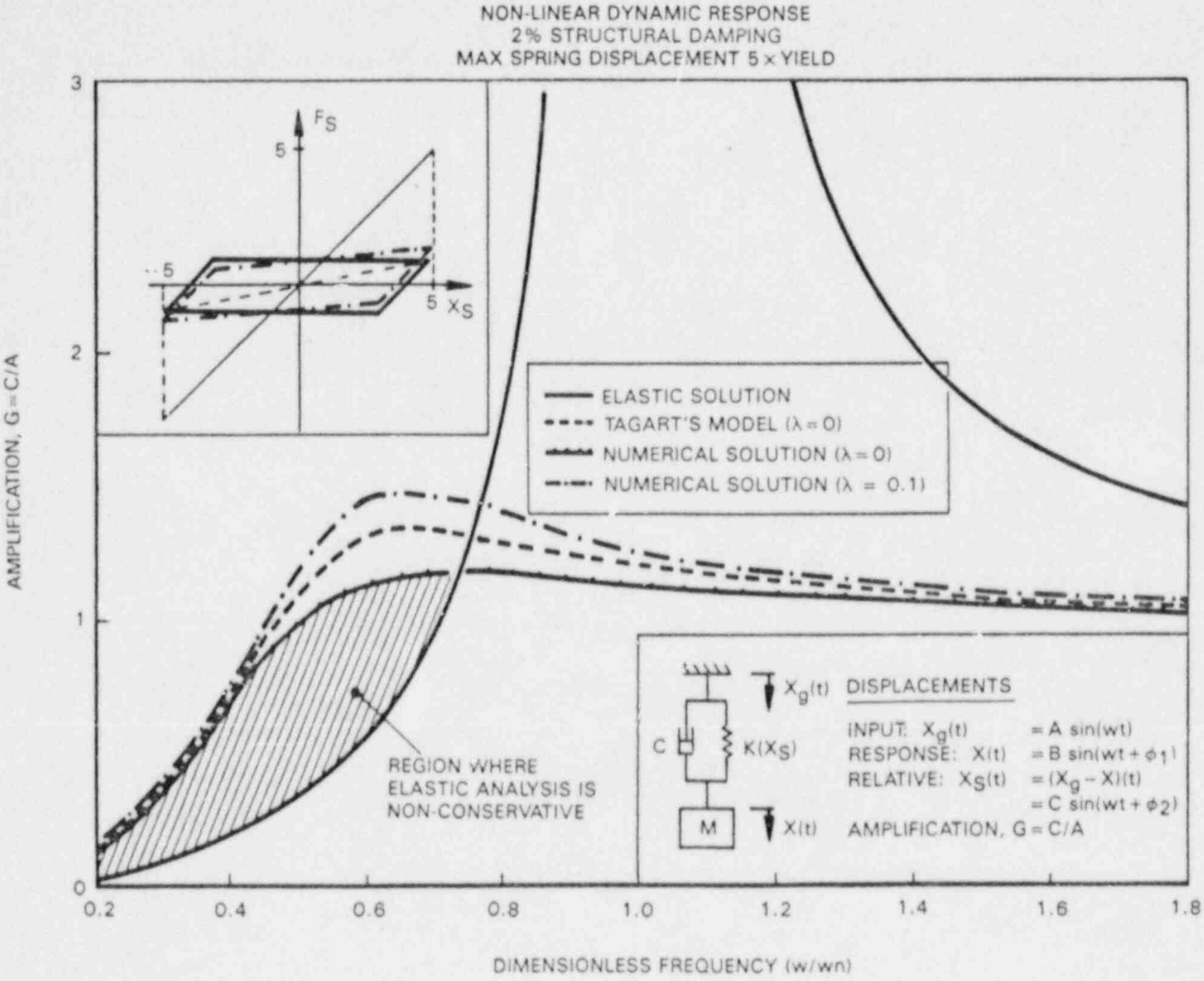

Figure 3.5-8. Dynamic Amplification for Elastic and Elastic-Plastic Systems

WHAT WE KNOW IN 1988

- WHY STATIC COLLAPSE DOES NOT GENERALLY OCCUR
- WHAT TYPES OF DYNAMIC LOADS CAN COLLAPSE PIPING
- HOW TO APPROXIMATELY PREDICT COMPONENT TEST RESULTS FROM FIRST PRINCIPLES
- LIMITATIONS OF LINEAR DYNAMIC ANALYSIS
- CLARIFY CONCEPTS OF APPARENT DAMPING
- WHY SOUND PIPING SYSTEMS ARE SO FUNDAMENTALLY RESISTANT TO SEISMIC OR OTHER CYCLIC-TYPE DYNAMIC INPUTS ("TIME" DAMPING IS VERY HIGH AT MODERATE DYNAMIC DUCTILITY OF 3)
- HOW TO UNDERSTAND RATCHETING

OPPORTUNITIES AND CHALLENGES

- SIGNIFICANT CODE MARGIN REDUCTION PROPOSAL (THIS PROGRAM)

- MANAGING PRIOR AND FUTURE OTHER CODE CHANGES

    N-411
    N-451
    ISM WITH SRSS
    SIMPLIFIED STATIC ANALYSIS
    NON-LINEAR METHODS
    SAM MODIFICATIONS
    DESIGN BY RULES

- METHOD TO OPTIMIZE PIPING DESIGN FOR SAFETY AND COSTS

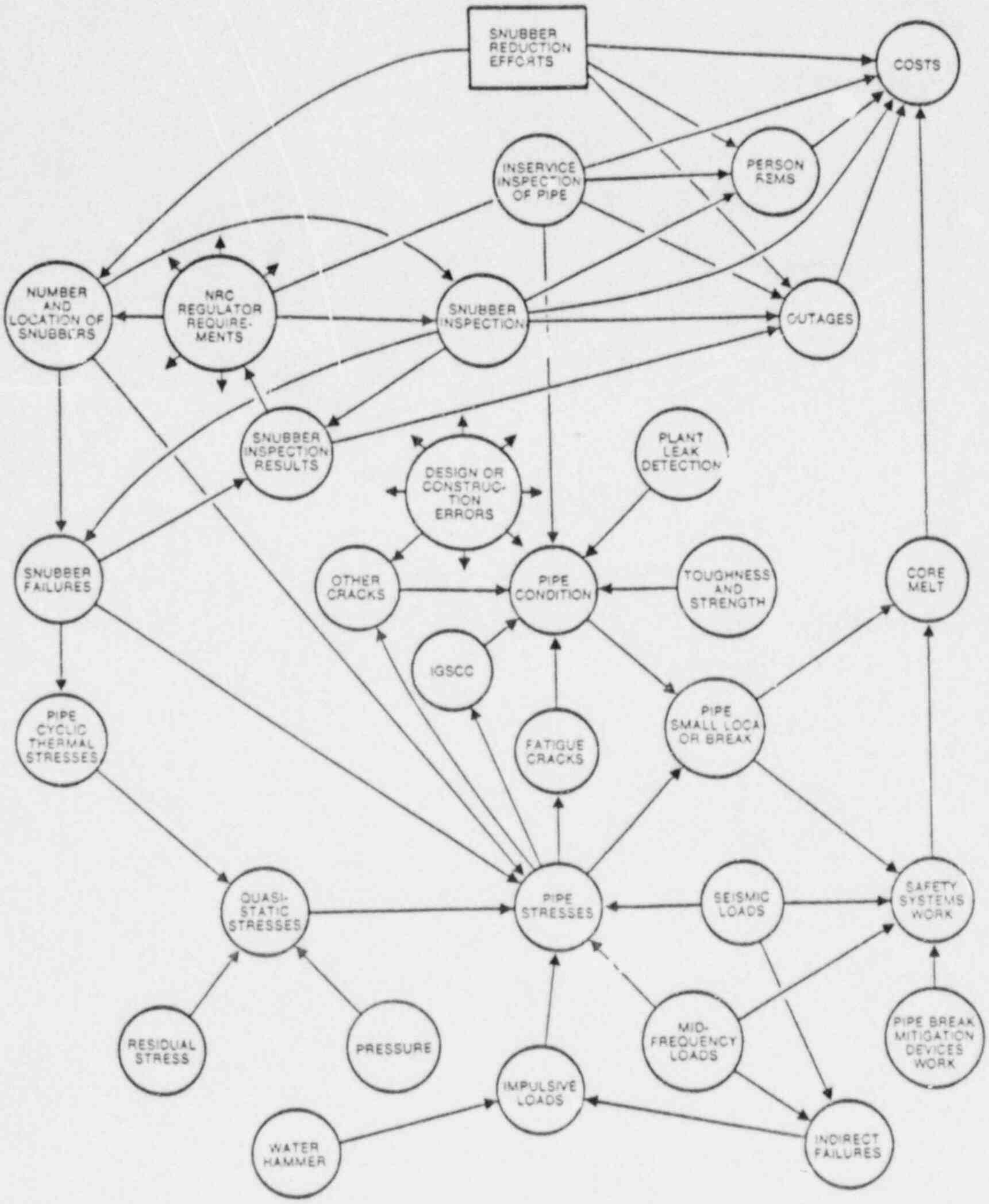

Figure 1. Snubber Reduction Model as Originally Conceived

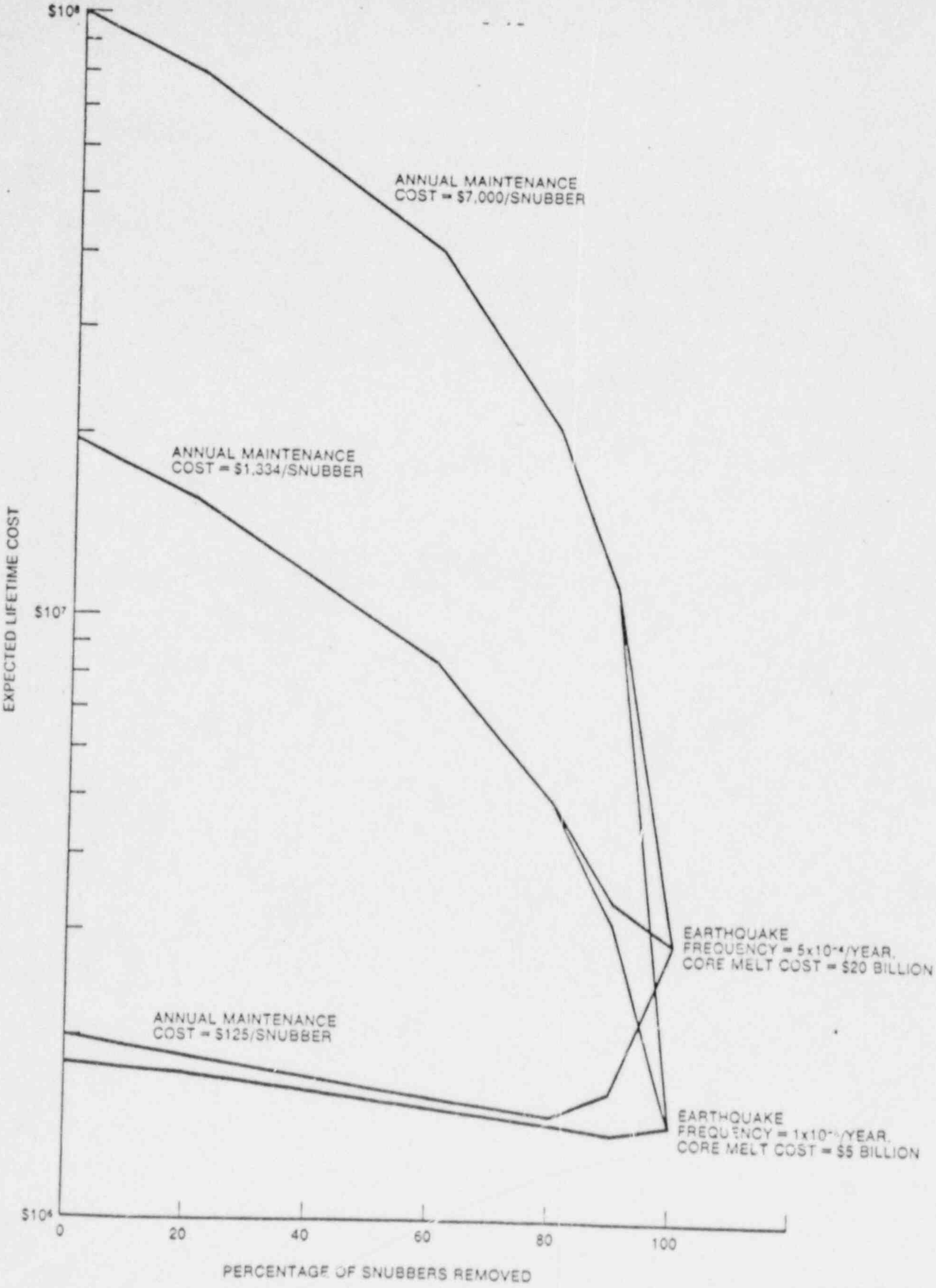

Figure 15. Expected Total Cost for an Entire Plant

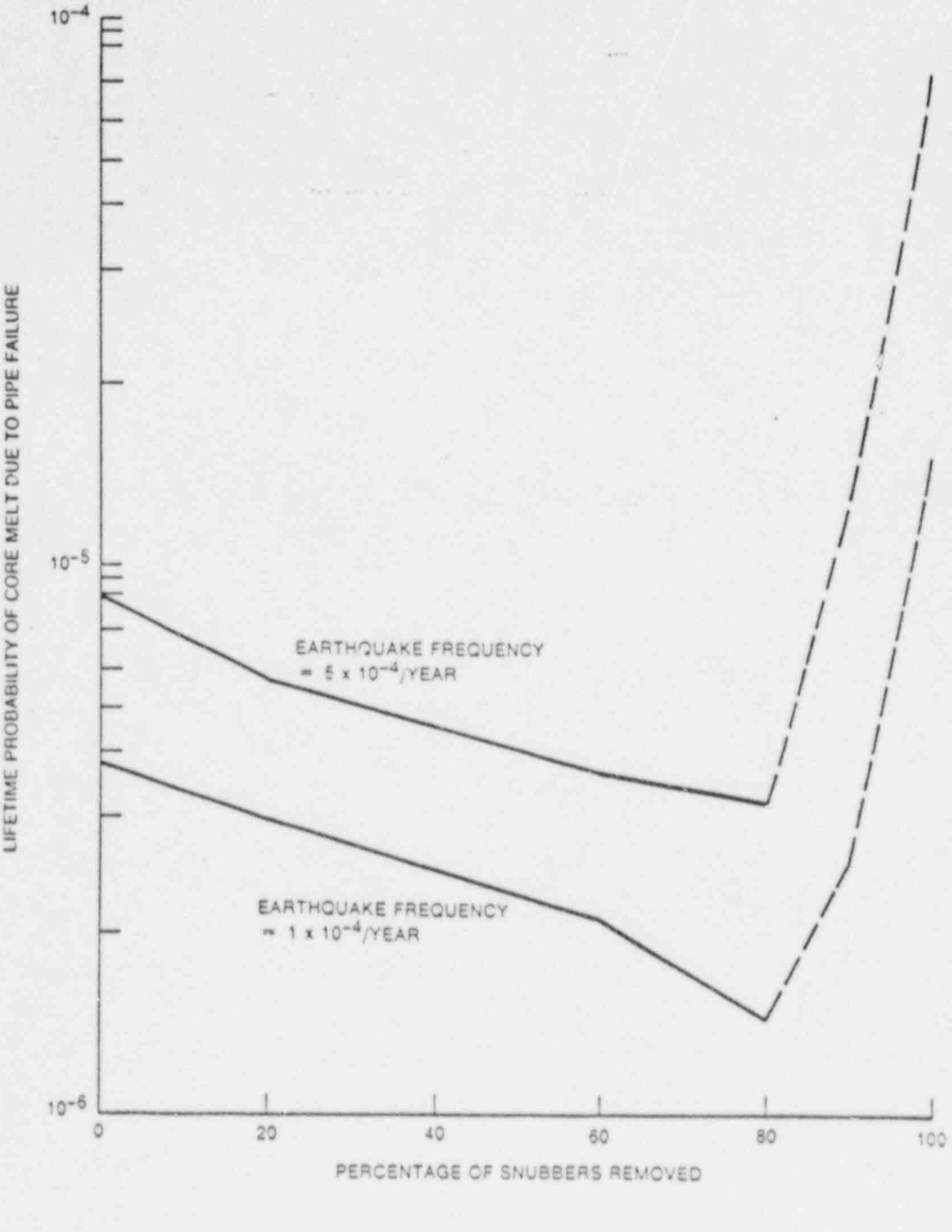

Figure 14. Core Melt Probability as a Function of Snubber Reduction (Results Extrapolated to an Entire Plant)

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POTENTIAL DESIGN RULE CHANGES

S. RANGANATH
GE NUCLEAR ENERGY

OUTLINE

- REVIEW OF CONCLUSIONS FROM TEST PROGRAM
    - COMPONENT AND SYSTEM TESTS
    - SPECIMEN FATIGUE & RATCHETING TESTS
    - RESULTS OF SDOF MODEL ANALYSIS
- POTENTIAL DESIGN RULE CHANGES
    - VALIDITY OF ELASTIC ANALYSIS
    - PROPOSED RULES AND TECHNICAL BASIS
    - STRATEGY FOR IMPLEMENTATION
- LONG TERM GOALS

CONCLUSIONS FROM COMPONENT AND SYSTEM DYNAMIC TESTS

- Component and system tests show that seismic loads well in excess of Level D limits can be tolerated.
- No limit load failures occurred in any of the component and system tests. This confirms that current Code stress limits providing margins on limit loads may be overly restrictive.
- Test failures involve a combination of fatigue and/or ratcheting suggesting that Code rules should consider this. Even when failure did occur, the number of cycles was well in excess of that in typical seismic loading events.
- Analysis of tests shows that elastic predictions are generally conservative for response spectrum analysis with peak broadening for up to 5 percent damping.

CONCLUSIONS FROM FATIGUE-RATCHET SPECIMEN TESTS

o RATCHETING OCCURS WHEN THE COMBINATION OF PRIMARY MEAN STRESS AND CYCLIC DYNAMIC STRESS EXCEEDS THE YIELD STRENGTH FOR POSITIVE MEAN STRESS.

  - TIME <u>INDEPENDENT</u> RATCHET STRAIN DETERMINED FROM MILLER MODEL

  - $\epsilon_R = S_{MEAN} / E_P$ FOR BILINEAR STRESS - STRAIN CURVE WITH KINEMATIC HARDENING

o TWO BAR AND BEND TESTS SHOW TIME <u>DEPENDENT</u> RATCHET STRAIN FOR THE LOW FREQUENCY (0.5 CPM) TESTS.

  - RATCHET STRAIN PER CYCLE DEPENDS ON MEAN STRESS, CYCLIC STRESS, AND TEMPERATURE

o PRELIMINARY DATA SUGGEST THAT TIME DEPENDENT RATCHET IS LESS SIGNIFICANT AT HIGHER FREQUENCIES.

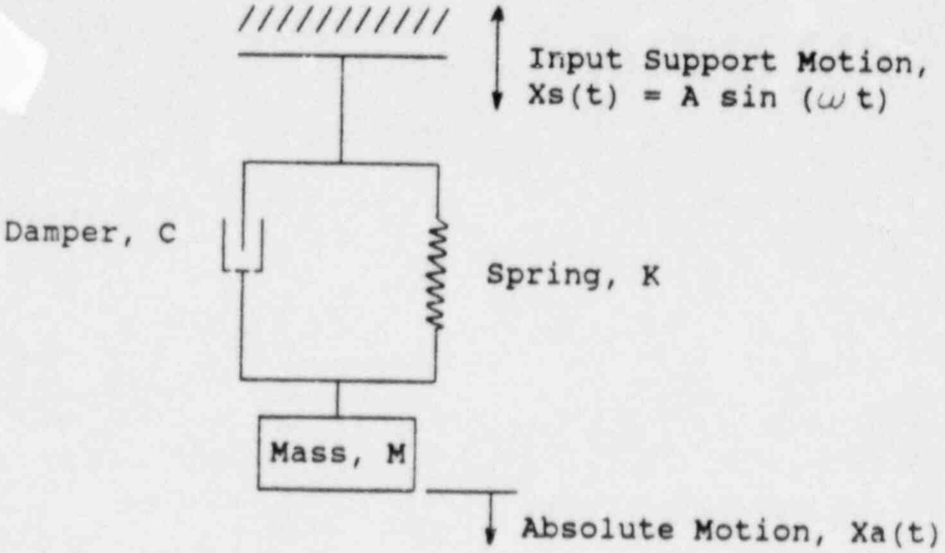
FATIGUE-RATCHET RESULTS (CONTINUED)

- FAILURE IS EITHER BY FATIGUE OR BY EXCESSIVE RATCHET STRAIN LEADING TO NECKING AND SUBSEQUENT RUPTURE.
- WHERE FAILURE WAS BY FATIGUE, THE DATA POINTS FALL ON THE MEAN FATIGUE DATA CURVE REGARDLESS OF THE RATCHET STRAIN.
- THUS, AS LONG AS THE CUMULATIVE RATCHET STRAIN IS NOT EXCESSIVE (SAY 5% - 10%) THERE IS NO SIGNIFICANT EFFECT ON CYCLIC FATIGUE LIFE.

SINGLE DEGREE OF FREEDOM (SDOF)
MODEL ANALYSIS

THE ELASTIC SDOF SYSTEM

WITH INPUT SUPPORT MOTION



Relative Motion, Xr(t) = Xa(t) - Xs(t)

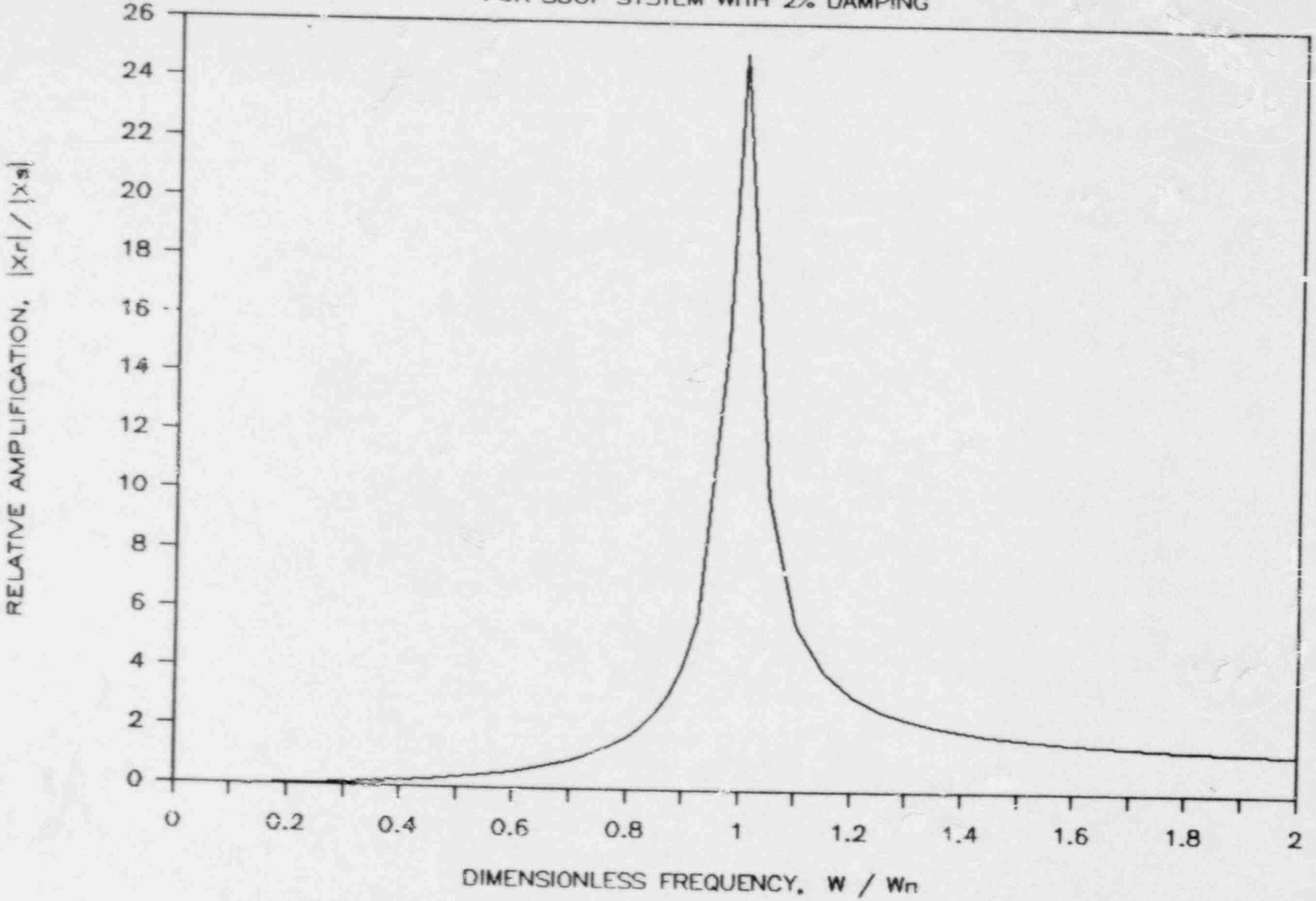
Two types of Displacement to be studied:

1) **Absolute Displacement of Mass** - Analogous to the motion of piping system components. Important in determining accelerations and velocities for loads on pipe mounted equipment.

2) **Relative Displacement between Mass and Support** - Analogous to relative displacements or deflections of piping components. Important for determining strain in piping components.

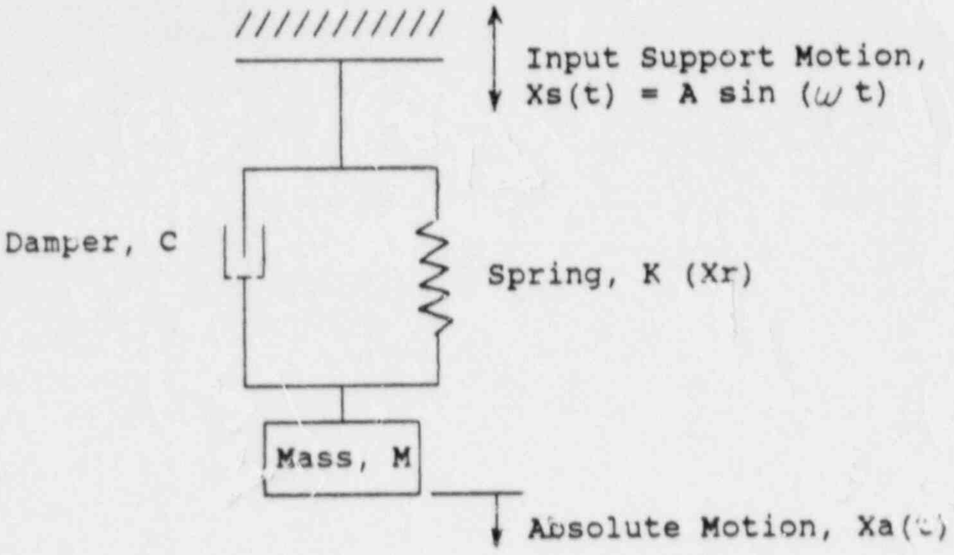
RELATIVE ELASTIC DYNAMIC RESPONSE

FOR SDOF SYSTEM WITH 2% DAMPING



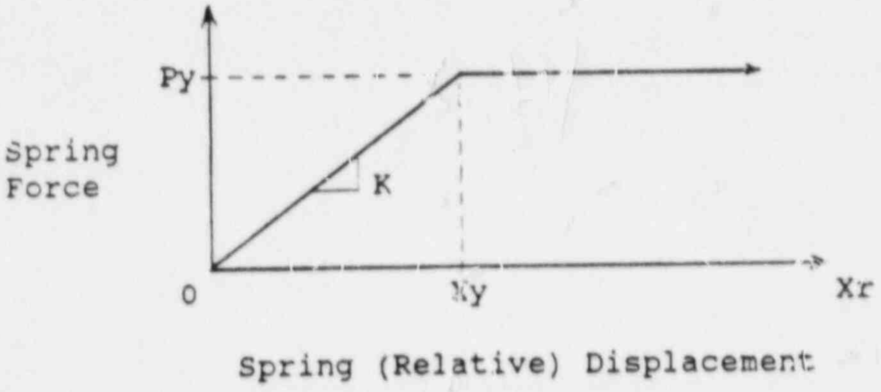
ELASTIC-PLASTIC SDOF MODELS

Modified Elastic Model - Developed by Sam Tagart



Relative Motion, $X_r(t) = X_a(t) - X_s(t)$

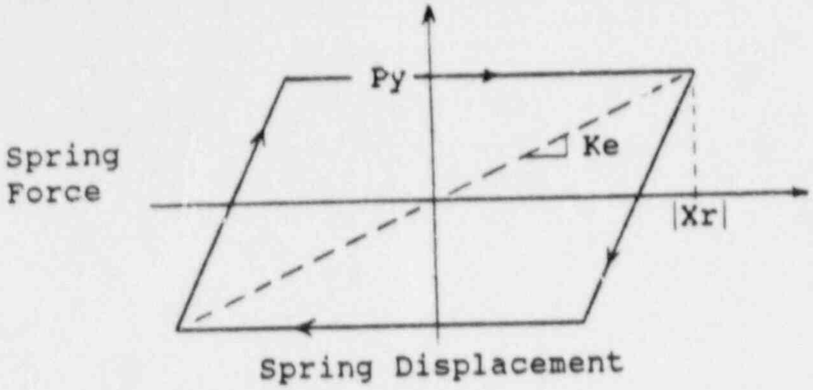
Assume Elastic - Perfectly Plastic Spring



- Model Elastic-Plastic system as elastic system with reduced stiffness and increased damping as shown on following page.

MODIFIED ELASTIC SYSTEM

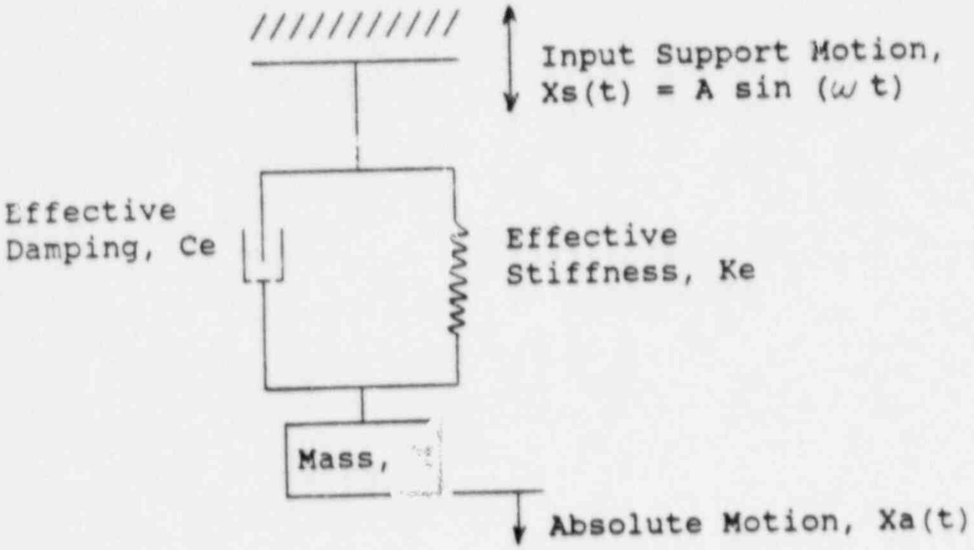
Spring Force-Displacement Hysteresis



- Stiffness reduced to account for plasticity. Effective stiffness calculated to give yield force at maximum relative displacement.

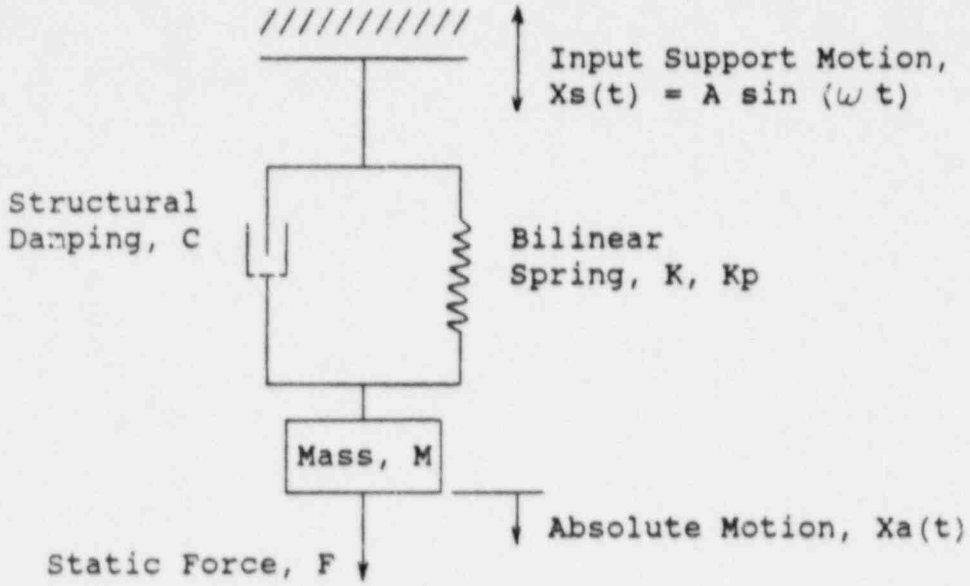
- Damping increased to account for irreversible work. Effective damping calculated to give same irreversible work per cycle as elastic-plastic system.

Modified Elastic Model

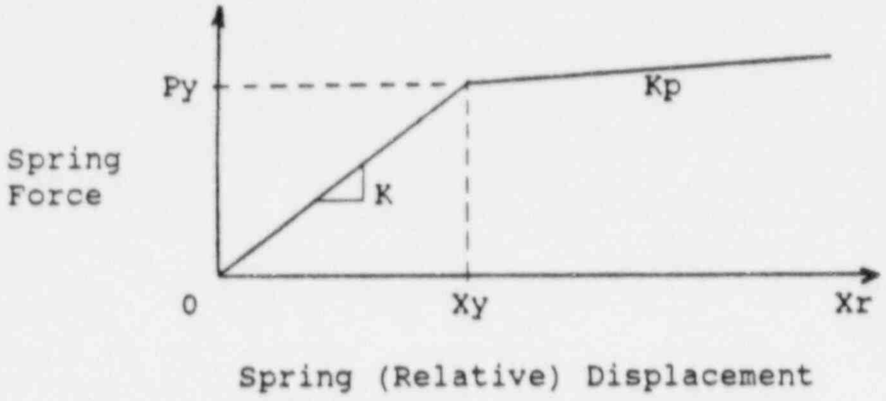


ELASTIC-PLASTIC SDOF MODELS

'Exact' Numerical Elastic-Plastic Model



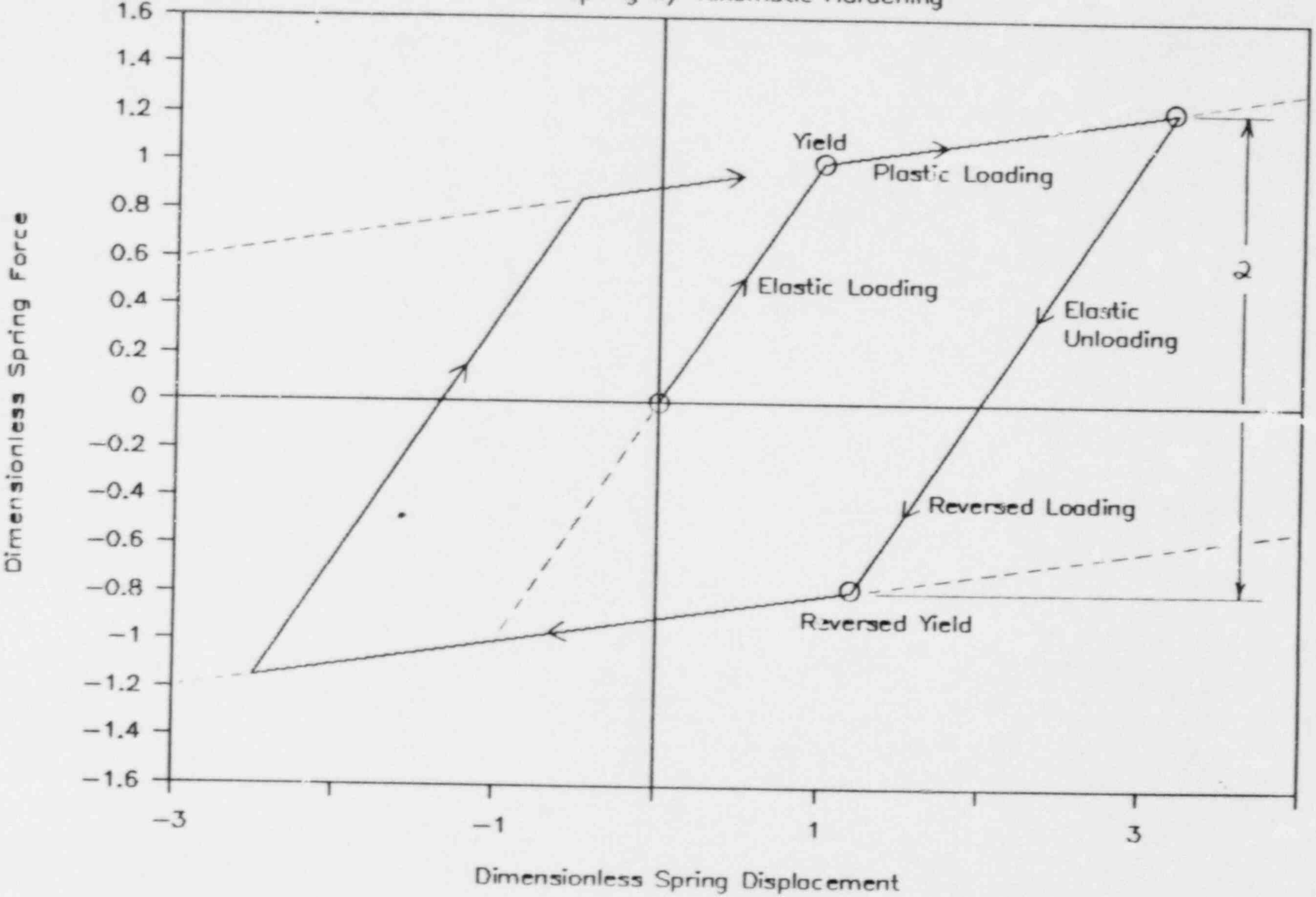
Assume Bilinear Spring - Includes strain hardening

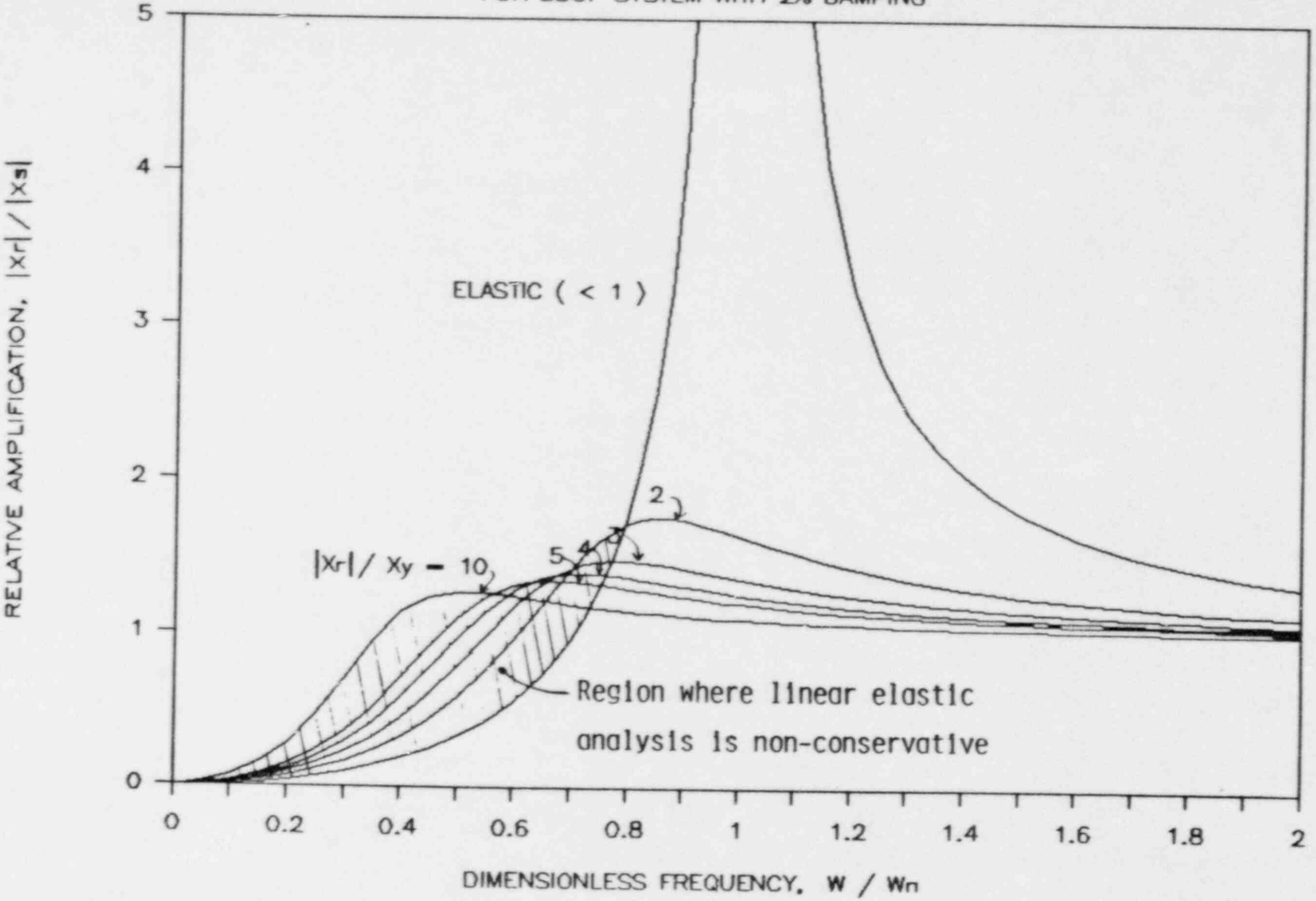


- Assumed Kinematic Hardening as shown on following page.
- Static force is included such that ratchetting may be simulated.
- 'Exact' time history solution using numerical solution.

Spring Force—Displacement Cycle

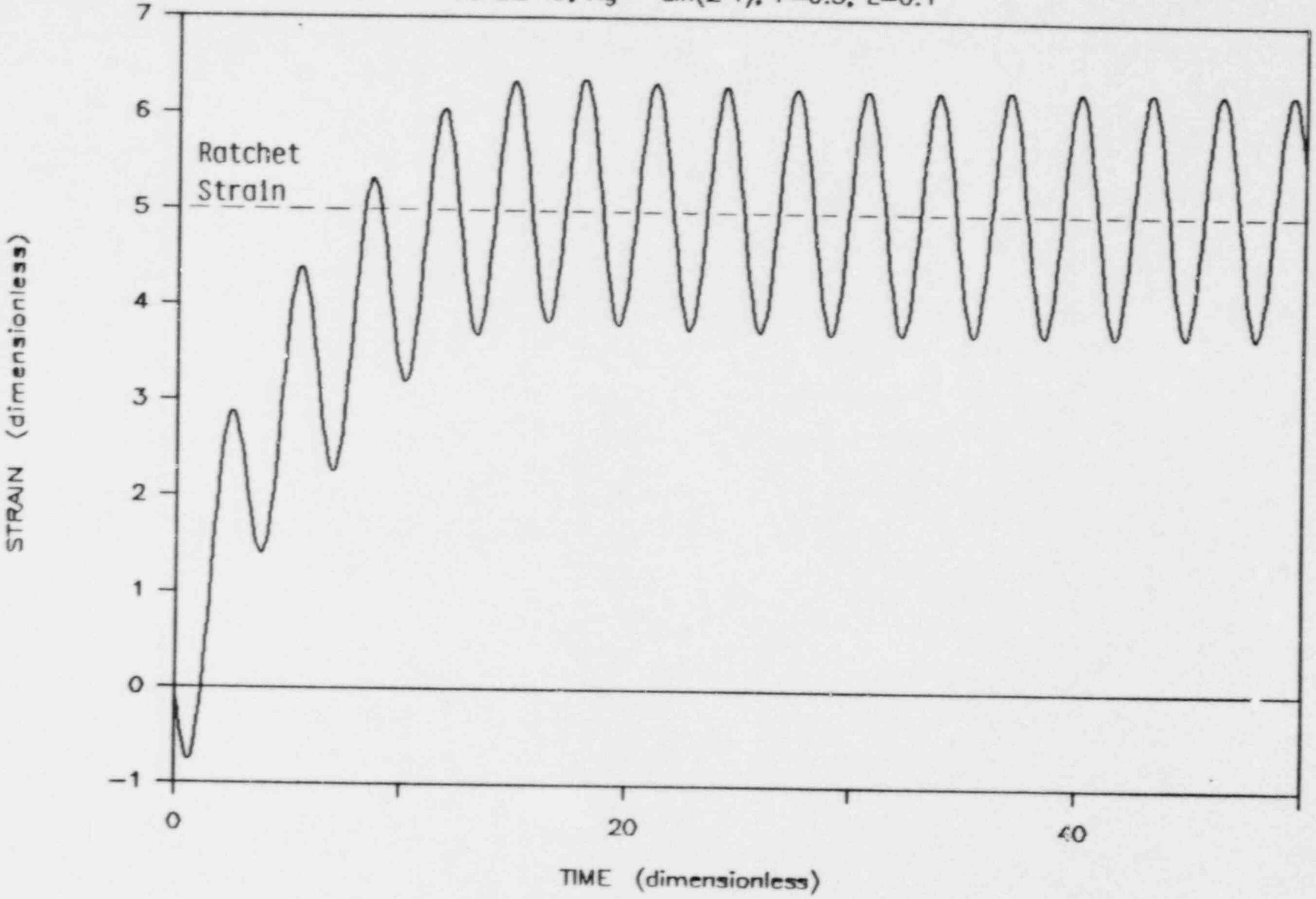
Bilinear Spring w/ Kinematic Hardening





RELATIVE EL-PL DYNAMIC RESPONSE

FOR SDOF SYSTEM WITH 2% DAMPING



SPRING STRAIN vs TIME

CASE 19. Xg - sin(2 T). F-0.5. L-0.1

-14-

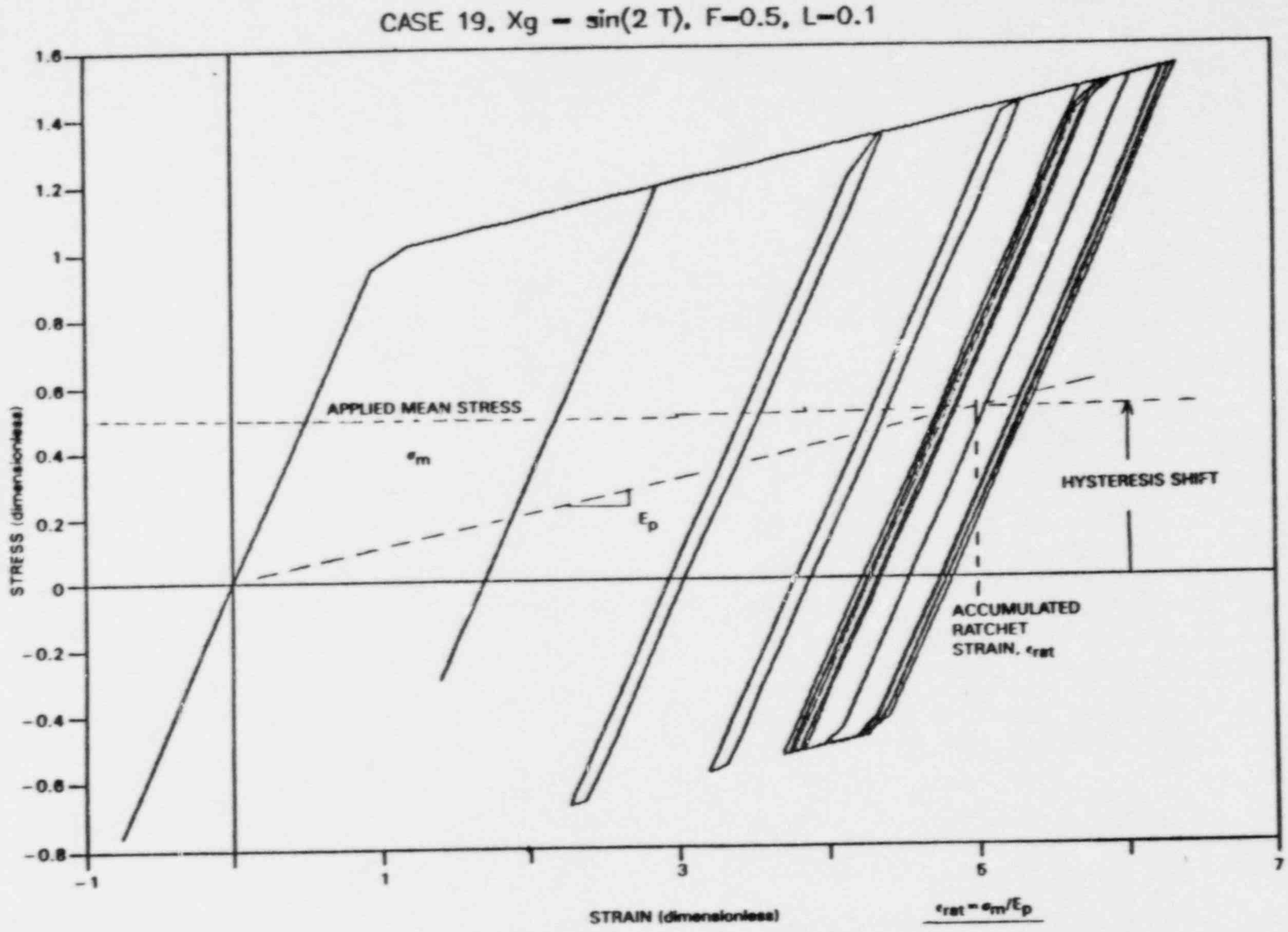


Figure 3.5-9. Arrest of Ratcheting Due to Stress-Strain Hysteresis Shift, Case 19

CONCLUSIONS FROM SINGLE DEGREE OF FREEDOM (SDOF) SYSTEM ANALYSIS

o SDOF EVALUATIONS SHOW THAT ELASTIC ANALYSIS MAY NOT BE CONSERVATIVE FOR APPLIED FREQUENCIES BELOW THE NATURAL FREQUENCY.

o PEAK BROADENING MAY BE NECESSARY TO ASSURE THAT ELASTC ANALYSIS IS CONSERVATIVE. THIS ACCOUNTS FOR THE SHIFT IN NATURAL FREQUENCY WITH PLASTICITY.

o RATCHETING OCCURS WHEN $S_{MEAN} + S_{DYN} \geqq S_Y$

o CUMULATIVE RATCHET STRAIN $E_R = S_{MEAN} / E_P$

o SDOF ANALYSIS INDEPENDENTLY PREDICTS THE SAME CUMULATIVE RATCHET STRAIN AS THE MILLER MODEL FOR BILINEAR KINEMATIC HARDENING.

o SDOF MODEL MAY BE OVER CONSERVATIVE COMPARED TO RESULTS OF COMPONENT AND SYSTEM TESTS:

  - GROSS SECTION YIELDING ASSUMED INSTEAD OF LOCAL YIELDING IN BENDING

  - VARYING E_p IN THE ACTUAL STRESS STRAIN CURVE INSTEAD OF LOWER CONSTANT E_p IN THE BILINEAR MODEL

  - HIGHER MATERIAL YIELD STRENGTH DUE TO STRAIN HARDENING

RATCHET CRITERION BASED ON RESULTS OF COMPONENT & SYSTEM TESTS

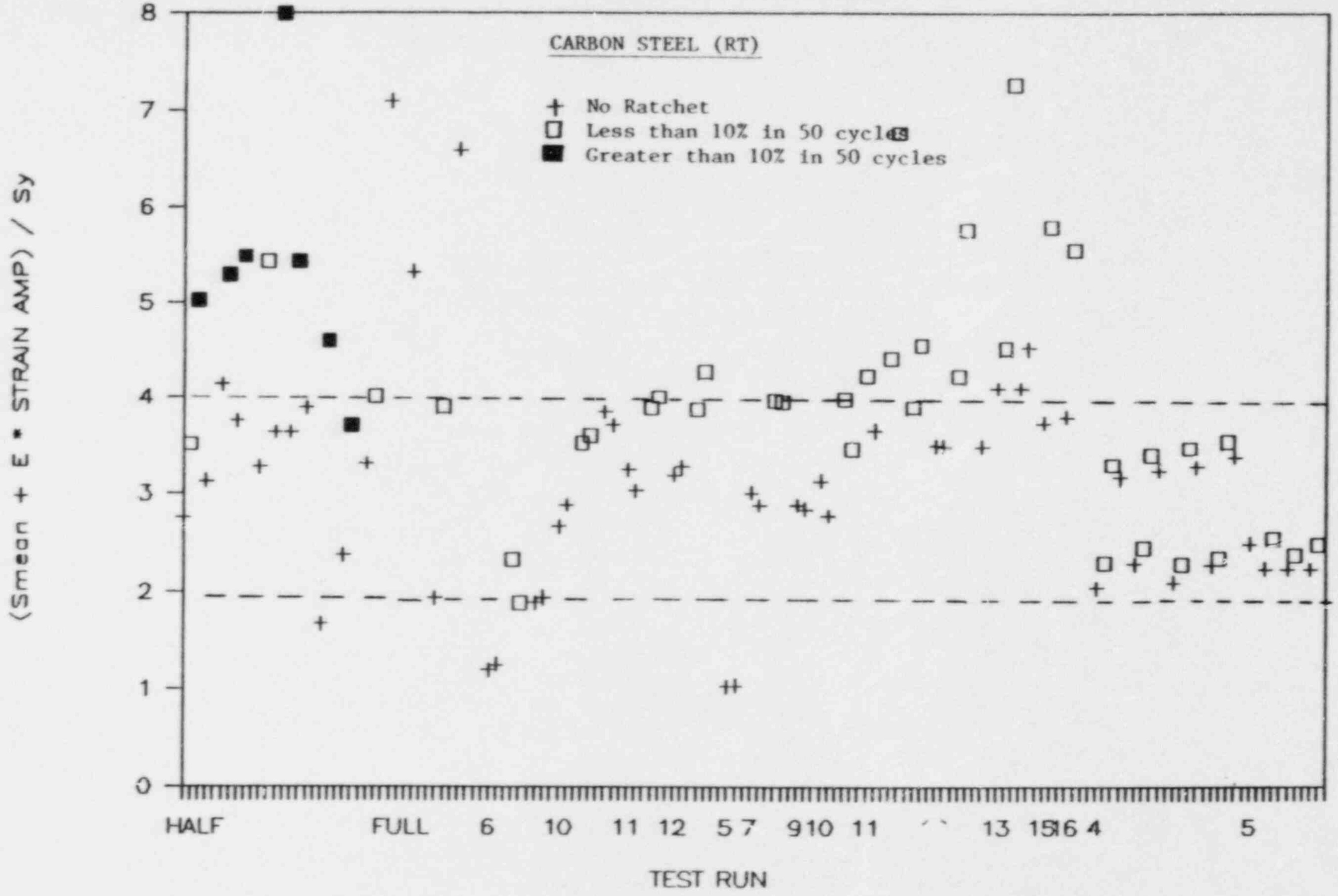
- WHERE GOOD DATA ARE AVAILABLE, THE MEASURED STRAINS CAN BE USED TO DETERMINE THE STRESS LEVEL BELOW WHICH THERE IS NO RATCHETING.

- FOR TYPICAL MEAN STRESS VALUES (0.5 S_M) SIGNIFICANT RATCHETING WAS NOT OBSERVED FOR STRESS AMPLITUDES BELOW APPROXIMATELY 6 S_M FOR BOTH CARBON STEEL AND STAINLESS STEEL AT ROOM TEMPERATURE.

- THIS STRESS VALUE MAY BE USED AS THE STRESS LEVEL BELOW WHICH SPECIAL FATIGUE OR RATCHETING ANALYSIS IS NOT NECESSARY.

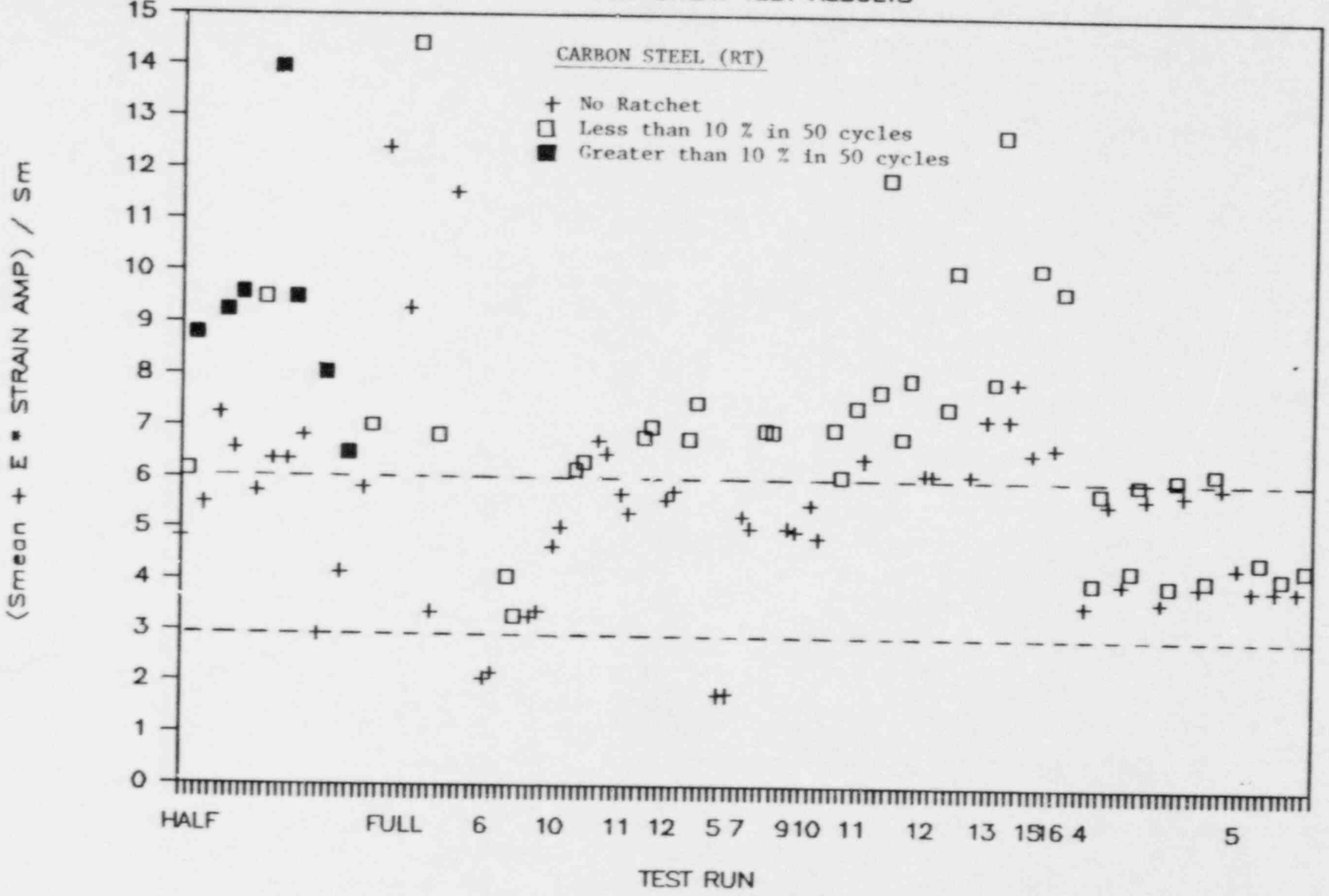
RATCHET THRESHOLD

FROM COMPONENT TEST RESULTS



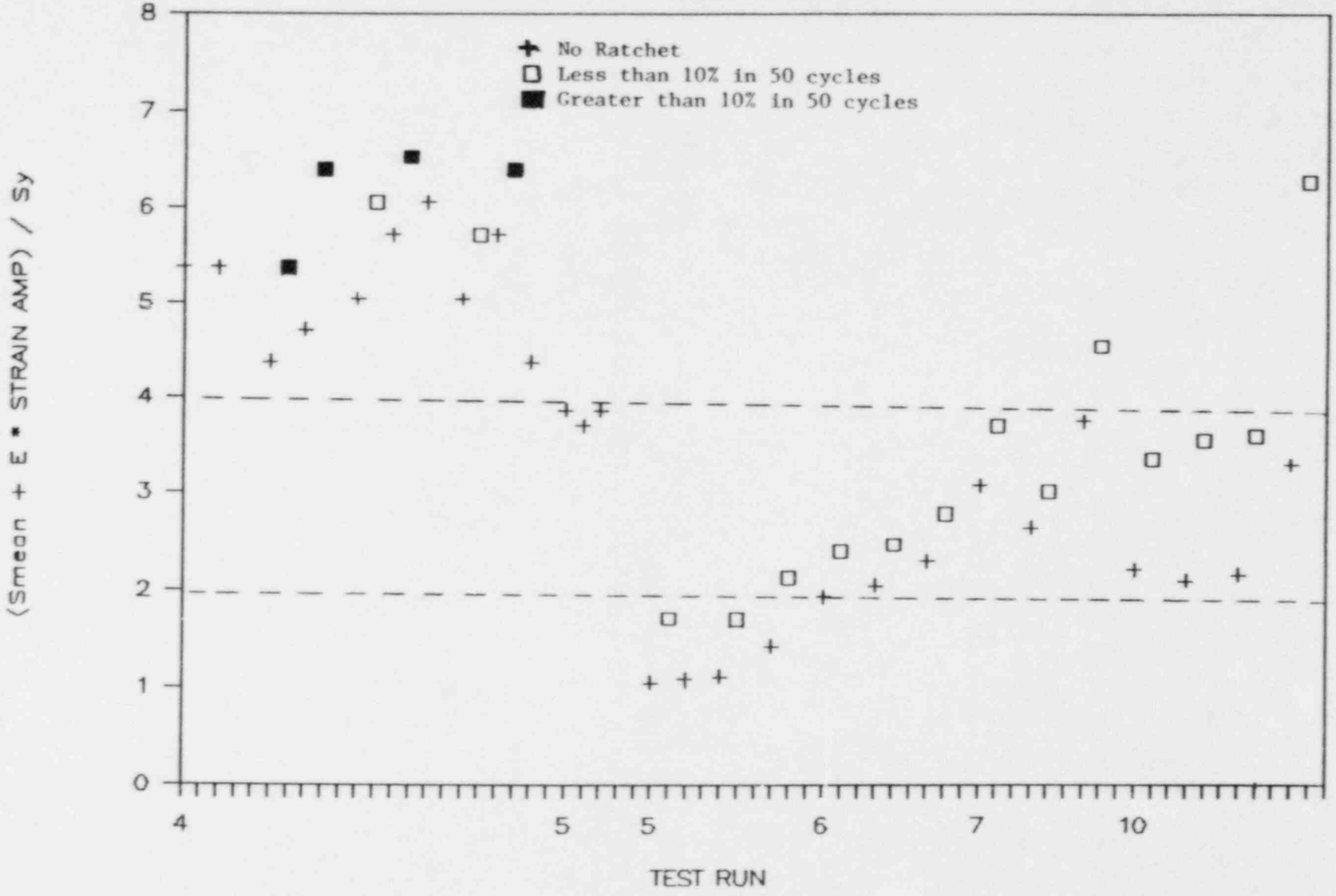
RATCHET THRESHOLD

FROM COMPONENT TEST RESULTS



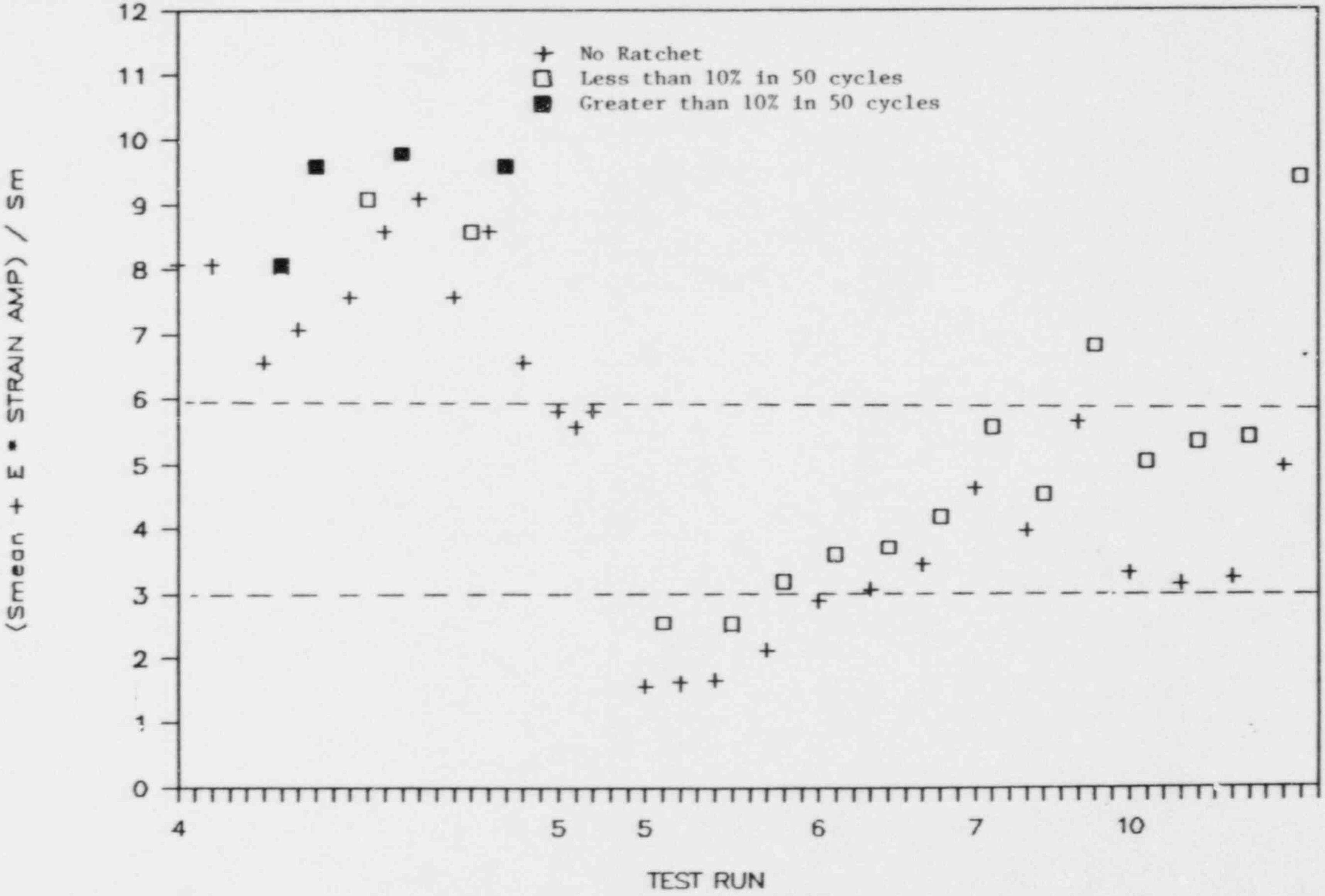
RATCHET THRESHOLD

FROM STAINLESS STEEL COMPONENT TESTS



RATCHET THRESHOLD

FROM STAINLESS STEEL COMPONENT TESTS



POTENTIAL DESIGN RULE CHANGES

- **PRESENT CODE LIMITS**

PRESSURE + EARTHQUAKE STRESS (EQ. 9) LIMITS

	LESSER OF		
LEVEL B	1.8 S_M	OR	1.5 S_Y
LEVEL C	2.25 S_M	OR	1.8 S_Y
LEVEL D	3.0 S_M	OR	2.0 S_Y

FATIGUE ANALYSIS FOR LEVEL B; NO SPECIFIC RATCHETING CONTROLS.

- **PROPOSED NEW CODE LIMITS**

PRESSURE + EARTHQUAKE STRESS (EQ. 9) LIMITS

	LESSER OF		
LEVEL B	3.0 S_M	OR	2.0 S_Y
LEVEL C	4.5 S_M	OR	3.6 S_Y
LEVEL D	6.0 S_M	OR	4.0 S_Y

FATIGUE ANALYSIS FOR LEVEL B; NO SPECIFIC RATCHETING CONTROLS.

STRATEGY FOR CODE IMPLEMENTATION

SHORT TERM GOALS

- o PROPOSE FOR INCLUSION IN CURRENT NB - 3000 REQUIREMENTS. WITH THIS APPROACH IT MAY BE DIFFICULT TO IMPOSE RESTRICTIONS ON ANALYSIS APPROACH.

- o INCLUDE NEW RULES IN A CODE CASE; IMPOSE SPECIFIC RESTRICTIONS ON HOW IT IS USED:
    - RESPONSE SPECTRUM ANALYSIS
    - PEAK BROADENING TO BE INCLUDED
    - DAMPING NOT IN EXCESS OF 5 PERCENT

- o USE NON-MANDATORY APPENDIX APPROACH

LONG TERM GOALS

- o TEST DATA SHOW THAT PIPING COMPONENTS CAN TOLERATE LOADS WELL BEYOND CURRENT CODE LIMITS
- o THE CURRENT DESIGN CRITERIA MAY BE ADDING TO COSTS SIGNIFICANTLY WITHOUT COMMENSURATE SAFETY BENEFITS
    - \+ SUBSTANTIAL DESIGN/ANALYSIS COSTS
    - \+ INCREASED HARDWARE COSTS, E.G., SNUBBERS, PIPE SUPPORTS
    - \+ INCREASED MAINTENANCE COSTS
    - \+ POTENTIAL FOR INCREASED STEADY STATE STRESSES IF SNUBBERS 'LOCK UP'
- o STATIC ANALYSIS MAY IN FACT BE ACCEPTABLE IN MOST CASES RESULTING IN SIGNIFICANT SIMPLIFICATION
- o LONG TERM DESIGN ANALYSIS GOAL SHOULD BE TO IMPLEMENT REALISTIC STRESS LIMITS WITH SIMPLER ANALYSIS METHODS

PIPING AND FITTING DYNAMIC RELIABILITY PROGRAM

- OVERALL PROGRAM STRUCTURE
- OBJECTIVES OF PROGRAM
- PFDR TEST PROGRAMS
    - COMPONENT TESTS
    - SYSTEM TESTS
    - SPECIMEN TESTS

OVERALL PROGRAM STRUCTURE

EPRI/NRC PIPING AND FITTING DYNAMIC RELIABILITY PROGRAM (PFDR)

- TASK 1: PROGRAM PLAN DEVELOPMENT - GE-SJ
- ▷ TASK 2: PIPE COMPONENT TESTING - ANCO
- ▷ TASK 3: PIPE SYSTEM TESTING - ETEC, ANCO
- ▷ TASK 4: SPECIMEN FATIGUE RATCHETING TESTS-MCL
- ▷ TASK 5: ANALYSIS OF TESTS AND DESIGN RULES - GE-SJ
- TASK 6: IDENTIFICATION AND DEVELOPMENT OF ALTERNATIVE DESIGN RULES AND REGULATIONS - GE-SJ
- TASK 7: EVALUATION OF ALTERNATIVE DESIGN RULES - GE-SJ
- TASK 8: PROJECT FINAL REPORTS - GE-SJ

OBJECTIVES OF PROGRAM

MAJOR OBJECTIVE: DEVELOP AN IMPROVED, REALISTIC AND DEFENSIBLE SET OF PIPING DESIGN RULES FOR INCLUSION IN ASME CODE

1. DETERMINE ACTUAL FAILURE MECHANISM FOR PIPING SYSTEMS AND COMPONENTS
    - A) LOW FREQUENCY LOADS
    - B) MID FREQUENCY LOADS
    - C) HIGH FREQUENCY IMPULSIVE LOADS

2. MEASURE PIPING SYSTEM DAMPING FOR VARIOUS STRAIN LEVELS OVER A LARGE RANGE OF FREQUENCIES

3. DETERMINE INFLUENCE OF SUPPORT FAILURE ON PIPING SYSTEM RESPONSE

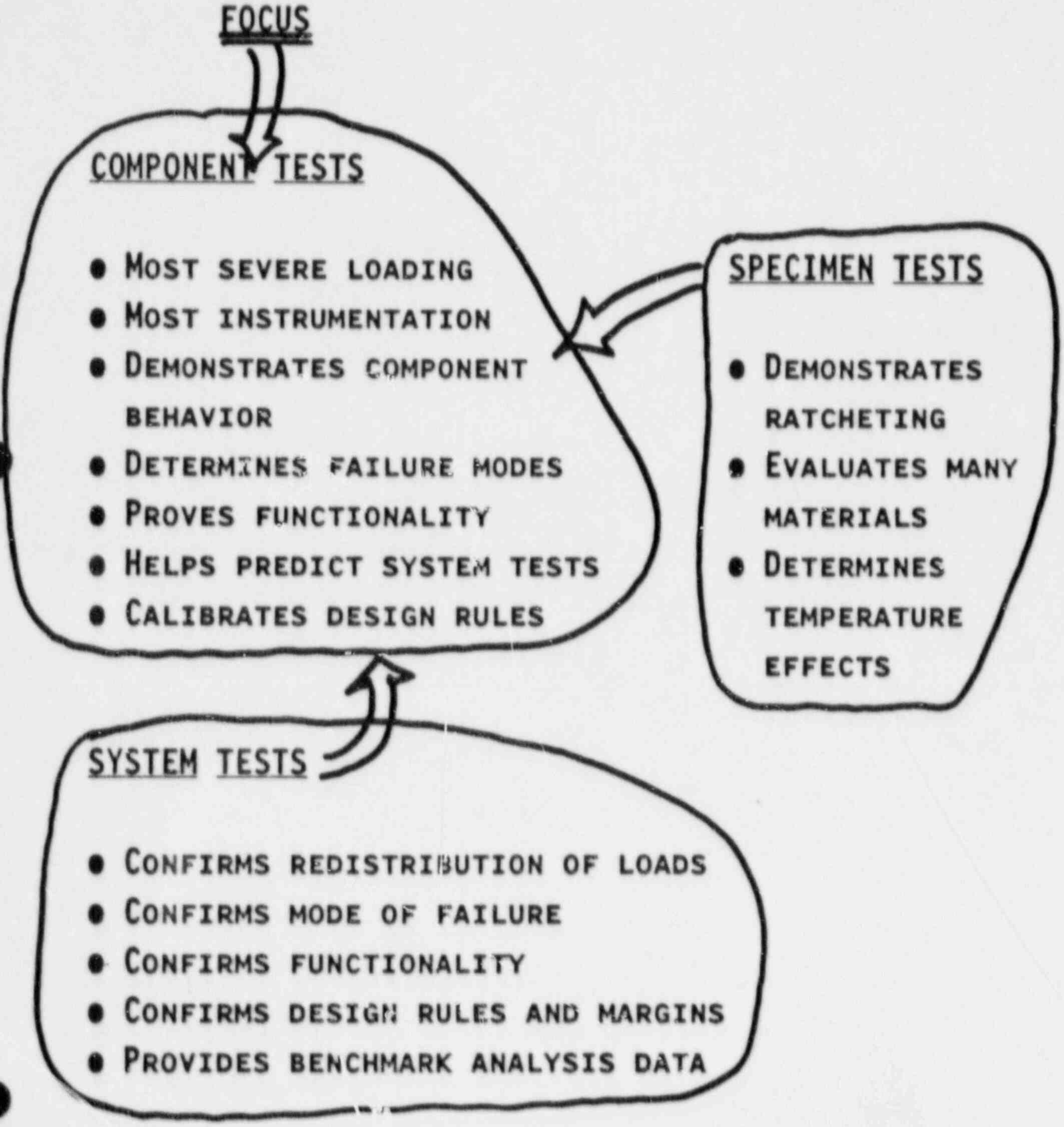
4. SHOW EFFECT OF LOW FREQUENCY INPUT TO PIPING FROM BUILDINGS SUBJECTED TO LARGE AMPLITUDE EARTHQUAKES

5. DEMONSTRATE THAT PIPING COMPONENTS AND SYSTEMS CAN TOLERATE EARTHQUAKES MUCH LARGER THAN SSE WITHOUT PIPE FAILURE

OBJECTIVES OF PROGRAM (CONTINUED)

6. DEVELOP LABORATORY PROCEDURE FOR QUANTITATIVE EVALUATION OF FATIGUE-RATCHETING
7. QUANTIFY ECONOMIC BENEFITS OF NEW DESIGN RULES TO UTILITIES BY APPLICATION TO TYPICAL NUCLEAR PIPING SYSTEM
8. SUGGEST CHANGES TO SRP AND RG WHICH REFLECT INHERENT DYNAMIC MARGINS IN PIPING
9. DEVISE SIMPLIFIED METHODS FOR ACCOUNTING FOR PLASTIC DEFORMATION, AND FATIGUE RATCHETING
10. SIMPLIFY PIPING SYSTEM DYNAMIC ANALYSIS

PFDR TEST PROGRAMS



FOCUS

COMPONENT TESTS

- Most severe loading
- Most instrumentation
- Demonstrates component behavior
- Determines failure modes
- Proves functionality
- Helps predict system tests
- Calibrates design rules

SPECIMEN TESTS

- Demonstrates ratcheting
- Evaluates many materials
- Determines temperature effects

SYSTEM TESTS

- Confirms redistribution of loads
- Confirms mode of failure
- Confirms functionality
- Confirms design rules and margins
- Provides benchmark analysis data

COMPONENT TESTS

OBJECTIVES

- DETERMINE FAILURE MODE (S) UNDER DYNAMIC LOADING
- MEASURE RATCHETING AND CYCLES TO FAILURE
- DEVELOP ENGINEERING UNDERSTANDING OF COMPONENT BEHAVIOR

COMPONENTS TO BE TESTED (6 IN. DIA., SCH 10, 40, 80)

ELBOWS, TEES, REDUCERS, NOZZLES, SUPPORT CONNECTIONS

PLAN FOR COMPONENT TESTS

- INPUT PEAK AT 0.5 HZ BELOW COMPONENT NAT. FREQ.
- ANCO SLEDS OPERATED AT MAXIMUM EXCITATION

DESIRED RESULTS

- FATIGUE RATCHET CRACK IN 2 - 3 SEISMIC INPUTS

ACTUAL RESULTS

- SCH. 10: CRACKED IN 1/2 TO 3-1/2 SEISMIC INPUTS
- SCH. 40: CRACKED IN 1-1/2 TO 3-1/2 SEISMIC INPUTS - OPTIMUM
- SCH. 80: CRACKED IN 5 TO 9 SEISMIC INPUTS

INVESTIGATING BEHAVIOR AT INTERMEDIATE LOAD LEVELS

PIPING AND FITTING DYNAMIC RELIABILITY PROGRAM
COMPONENT TEST SUMMARY

NO	TYPE	MAT SCH	RES STR % (1)	PRESS	LOAD DIR /SIZE	DYN MOM LIM MOM	LOAD TYPE	P-P CYC STRAIN ∞	INPUT X LEVEL D	NO TH	FAIL MODE
1	Elbow	CS 80		1500	I-P	1.21	SSE	2.5%(1)	15	5	NF
	(Retest)	80		2600	I-P	1.21	SSE	1.5%(1)	15	0.5	FR
2	Elbow	CS 80		1500	O-P	1.04	SSE	1.4%(1)	15	5	NF
	(Retest)	80		2600	O-P	1.04	SSE	1.4%(1)	15	4.5	FR
3	Elbow	SS 10	3.5	400	I-P	2.36	SSE	2.4%(1)	21	3.5	FR
4	Elbow	CS 40		1000	I-P	1.[illegible]	SSE	2.0%(1)	18	2.5	FR
5	Elbow	CS 40	13.8	1700	I-P	2.06	SSE	2.0%(1)	21	3.5	FR
6	Elbow	SS 40	16	1700	I-P	2.00	SSE	2.0%(1)	19	3.5	FR
7	Elbow	SS 40	9	1000	I-P	1.80	SSE	2.0%(1)	23	4.5	FR
8	Elbow	SS 40	1.5	0	I-P	1.80	SSE	2.0%(1)	24	5	NF
9	Tee Fix-2	SS 40	8	1700	O-P	2.50	SSE	2.2%	21	1.5	FR
10	Tee Fix-2	SS 40	6.5	1000	O-P	2.40	SSE	2.2%	21	2.5	FR
11	Tee Fix-2	SS 10	3	400	O-P	1.00	SSE	1.9%	16	0.5	FR
12	Tee Fix-2	SS 40	11	1700	I-P	2.30	SSE	2.2%	27	2.5	FR
13	Short Elb.	CS 40	6	1000	I-P	2.30	SSE	1.9%(1)	22	2.5	FR
14	Tee Fix-2	CS 40	10	1700	O-P	2.46	SSE	2.2%(2)	18	1.5	FR
15	Reducer	SS 40	18	1700	8x4	1.18	SSE	13%(3)	13	5	FR
16	Reducer	SS 40	2.5	1700	8x4	1.72	SSE	3.3	30	0.5	FR

PIPING AND FITTING DYNAMIC RELIABILITY PROGRAM

COMPONENT TEST SUMMARY

NO	TYPE	MAT SCH	RES STR % (1)	PRESS	LOAD DIR /SIZE	DYN MOM / LIM MOM	LOAD TYPE	P-P CYC STRAIN OD	INPUT X LEVEL D	NO TH	FAIL MODE
17	Short Elbow	CS 40	2.5	1000	TOR	N/A	SSE	2.5%(1)	20	3	FR
18	Reinforced Fab. Tee	CS 40		1000	8x4		SSE			0.3	FR
19	Elbow	CS 40		2300	I-P		SSE				
20	Nozzle	SS 40		1000	12x4		SSE				
21	Guide Lug	CS 40	5	1700	8x6 Circ. Mom.		SSE			0.4	FR
22	Guide Lug	SS 40		1700	8x6 Circ. Mom.		SSE			0.4	FR
23	Strut	CS 40	1.5	1000	I-P	2.3	SSE	2.1	N/A	5	NF
24	Elbow	CS 40		1000	I-P	1.0	Static Closing	2			Collapse
25	Elbow-Mid	SS 10	4	800	Mix	6.0	RV2 Mid	1.4	27	7	NF
26	Elbow	CS 40		1700	I-P		Sineswp			8	FR
27	Tee Fix-1	SS 40	8	1700	O-P		Mid + Sine			9	NF
28	Waterhammer	CS 40		1700 1000	I-P I-P	2.7 3.1	Solid Wh Water Slug	2.2		3 3	NF Collapse

PIPING AND FITTING DYNAMIC RELIABILITY PROGRAM
COMPONENT TEST SUMMARY

NO	TYPE	MAT SCH	RES STR % (1)	PRESS	LOAD DIR /SIZE	DYN MOM LIM MOM	LOAD TYPE	P-P CYC STRAIN (%)	INPUT X LEVEL D	NO TH	FAIL MODE
29	Waterhammer	CS 40		1700	1-P/ Strut	2.7	Solid Wh	0.55		3	NF
				1000	1-P/ Strut	2.8	Water Slug	0.40		3	NF
30	Elbow 1.4 Hz	SS 10		0 410	1-P 4 Hz 1-P 1.3 Hz		SSE	1.36 2.0		3	FR
31	Elbow 4.1 Hz	SS 10		410	1-P		Sineswp + SSE			3.5	FR
32	Elbow	SS 40		1700	1-P	3	Static Opening	0.5		3	NF
33	Pipe 15 Hz	CS 40		1000	N/A	1.1	Sineswp	1.1		10	NF
34	Pipe 6 Hz	CS 40		1000	N/A	1.8	Sineswp	2.16		4	FR
35	Elb High Wt	CS 40		1700	1-P	1.65	SSE	3.4%(1)	18	5	FR
36	Tee Fix-1	CS 40		1700	Thru-run		SSE			0.5	FR
37	Elb 1.4 Hz	SS 10	4	0	1-P	1.03(4)	SSE	2.0%(1)	10	2	RB
38	Tee Fix-1	CS 40	20	1700	O-P	1.92	SSE	2.7%(3)	20	3.6	FR
39	Tee Fix-1	SS 40	10	0	O-P	1.84	SSE	3.4%(3)	11	4	NF
40	Reducer	SS 40	9	0	8x4	1.2	SSE	3.3	2	2	RB
41	Elbow	CS 40		1000	1-P		SSE & SINE				

SYMBOLS:

WH = Water Hammer
I-P = In-Plane
O-P = Out of Plane
Fix-1 = Single End Fixed
Fix-2 = Both Ends Fixed
NO TH = Number of high level input test runs to cause failure
FR = Fatigue ratcheting failure
NF = No failure
RB = Ratchet Buckling
FRT = Fatigue ratcheting failure and followed by ductile tearing
Residual Strain = Measured by 2 inch scratch marks
Input X Level C = Calculated stress using linear response spectrum analysis, 2% Damping, ± 15% broadening and actual sled input. Use the calculated stress, (B_2 M/2), divided by Level D allowable, $3S_m$ to determine multiple of Level D allowable.

NOTES:

(1) For all the elbows, the measured strains are on the outside surface. For strain on inside surface, multiply the values by 1.5:

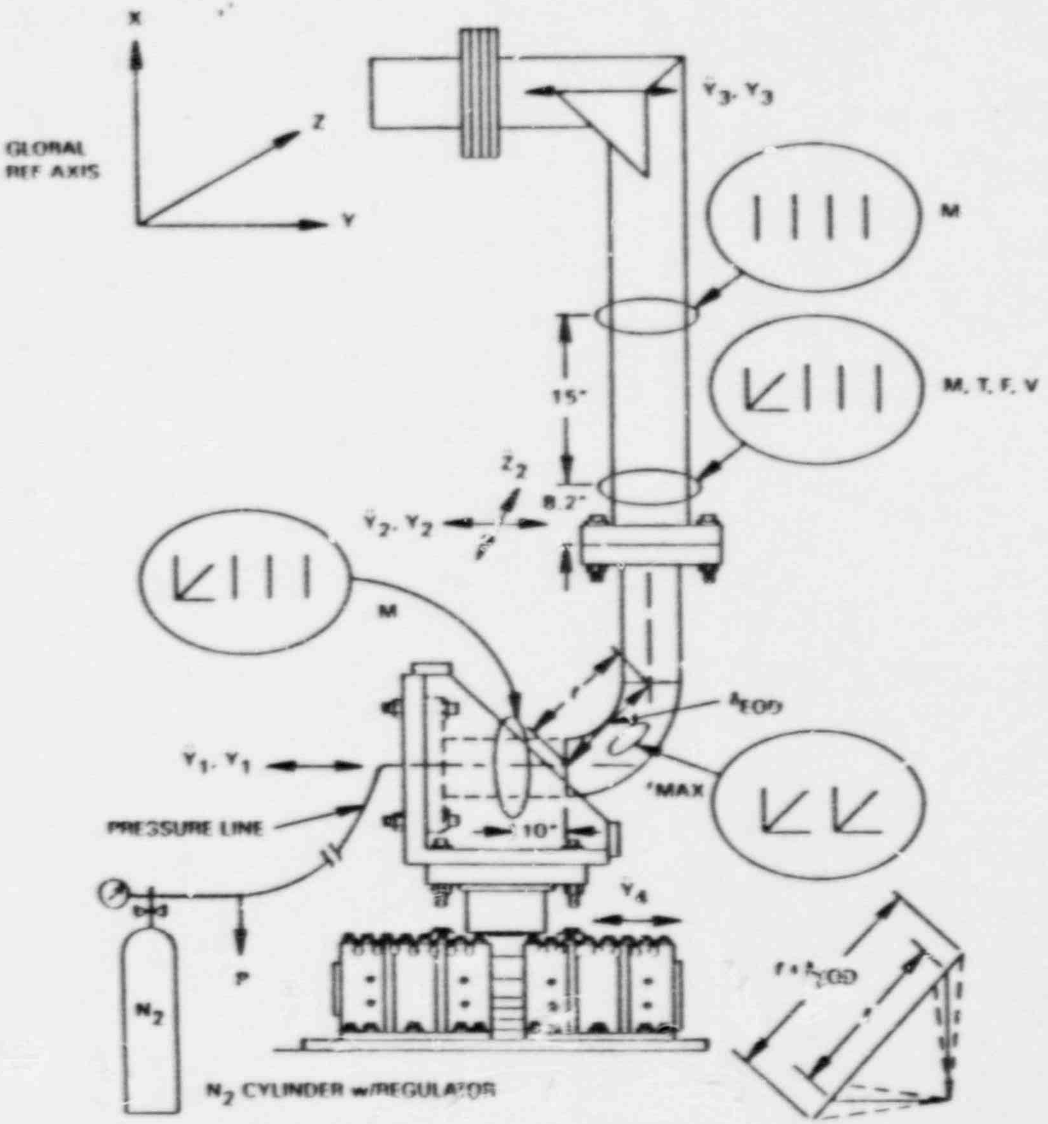
The inside surface is 1.388 times of the outside, principal strain is 1.072 time of circumferential strain and the circumferential strain is 1.01 time of average strain over gage length 1/16". 1.338 x 1.072 x 1.01 = 1.50. For 2" scratch mark, the factor 1.01 is increased to 1.52 and the multiplication is 1.388 x 1.072 x 1.52 = 2.26.

For ratchet strain on the inside surface, a factor of 2.0 over the tabulated values.

(2) Gage failed too early, there is almost no data, use previous similar test run data.

(3) Gage filed too early, it is not known if peak values have been obtained.

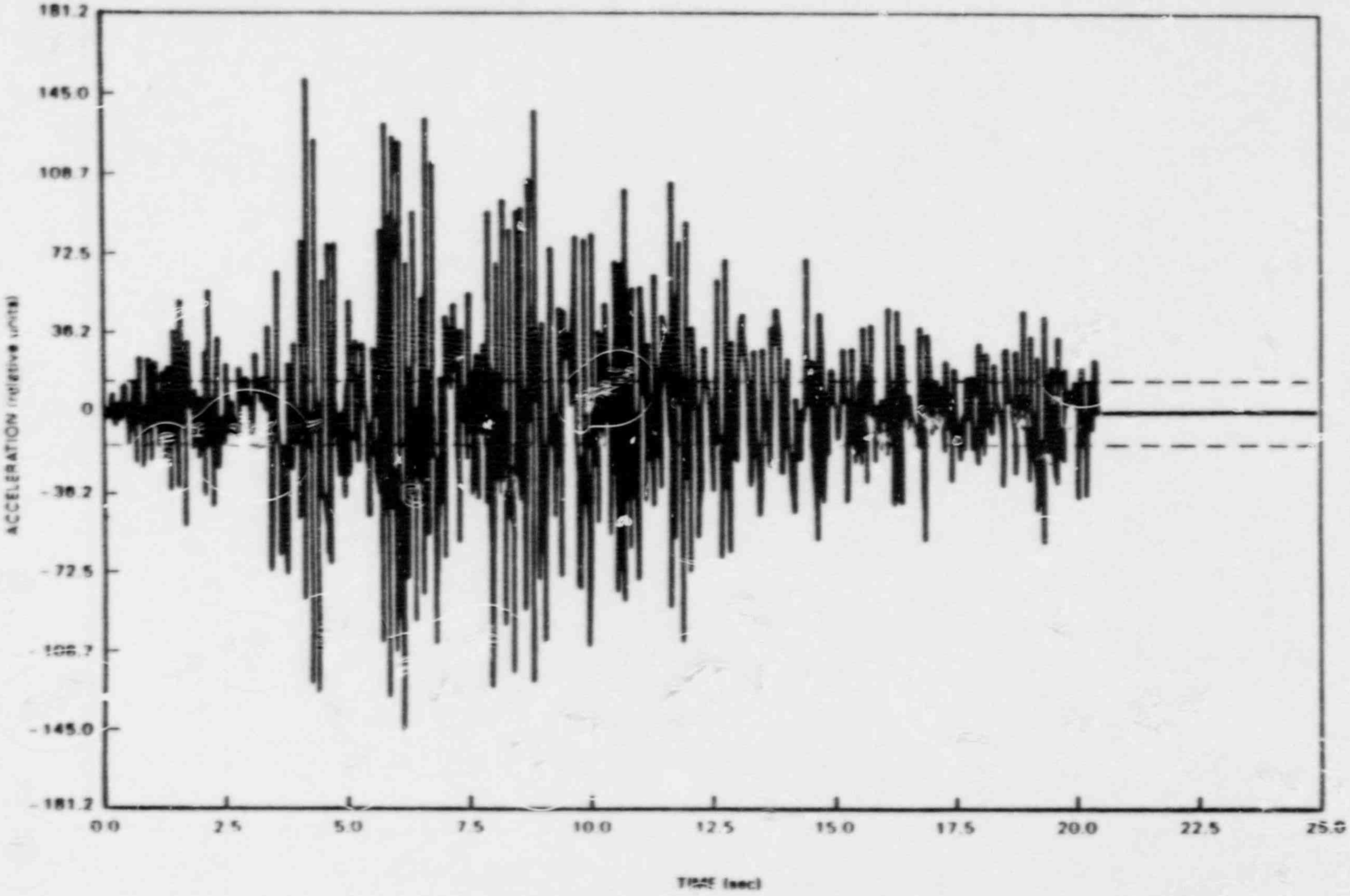
(4) Weight stress over 1000 psi.



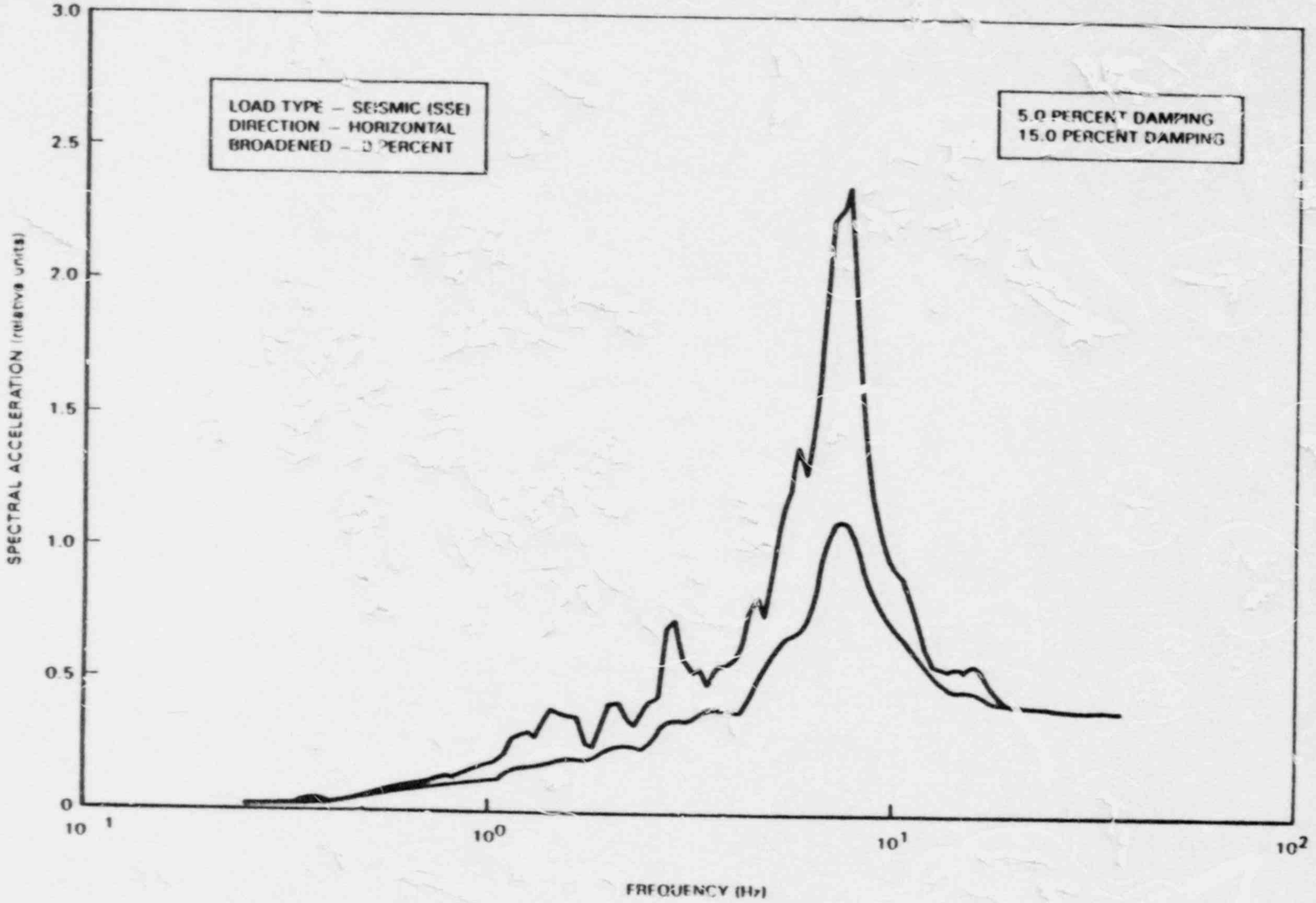
SYMBOL	STRAIN GAGE ORIENTATION
∠	ROSETTE ORIENTATION
|	AXIAL ORIENTATION

SYMBOL	DESCRIPTION	DATA CHANNELS
$\ddot{Y}_1$	ACCELERATION	1
$\ddot{Y}_2, \ddot{Z}_2$	ACCELERATION	2
$\ddot{Y}_3$	ACCELERATION	1
$\ddot{Y}_4$	ACCELERATION	1
Y_1	HORIZONTAL DISPLACEMENT	1
Y_2	HORIZONTAL DISPLACEMENT	1
Y_3	HORIZONTAL DISPLACEMENT	1
δ_{EOD}	ELBOW OPENING/ CLOSING DISPL	1
P	PRESSURE	1
M	MOMENT FORCE RESULTANT	18
ϵ_{MAX}	PEAK STR	2
	TOTAL	28

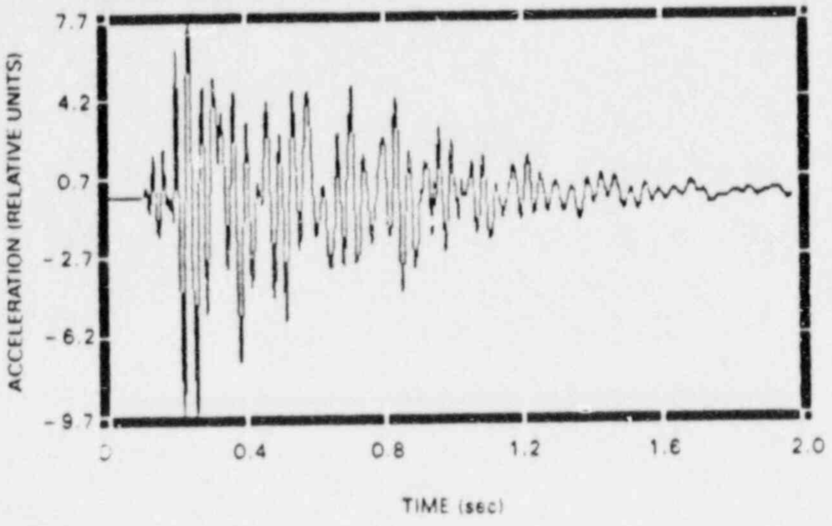
Location of Transducers for In-Plane Elbow Component



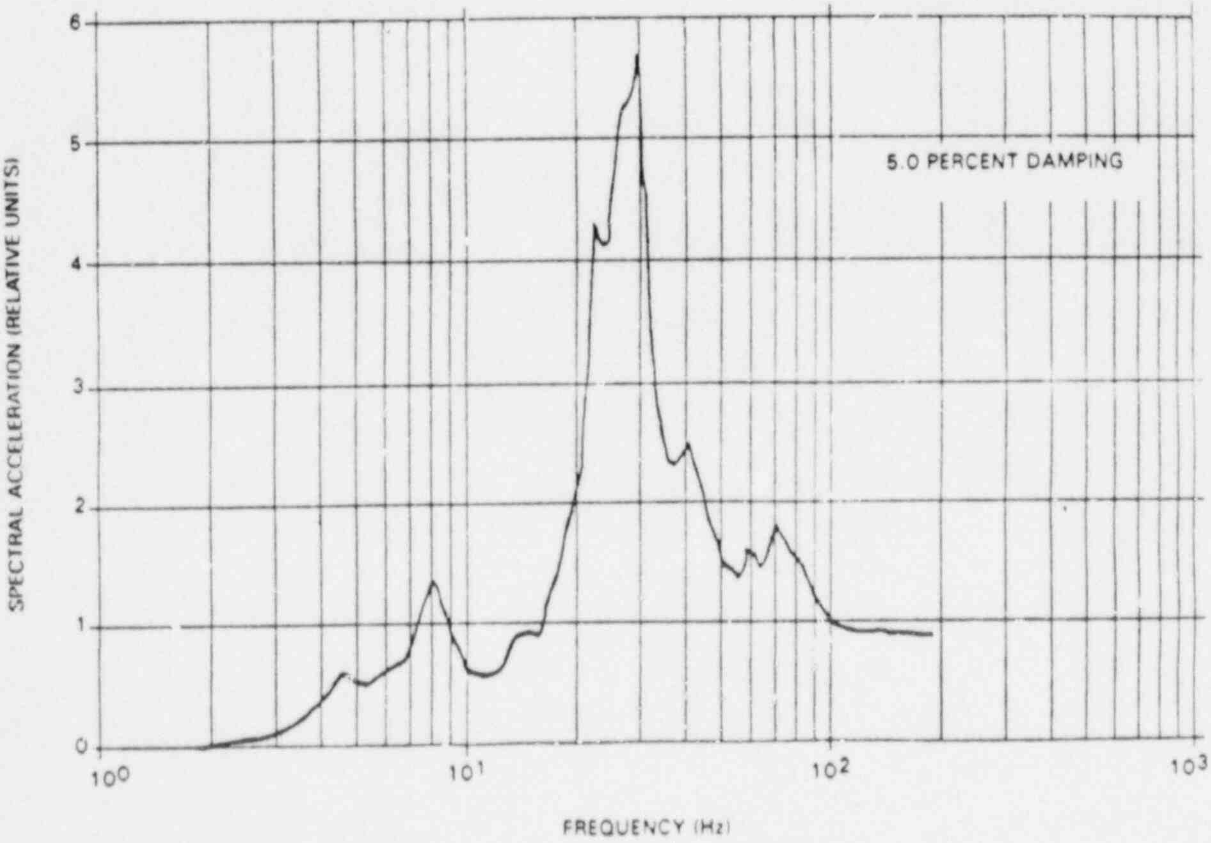
Input Motion Time History



Input Motion Response Spectra



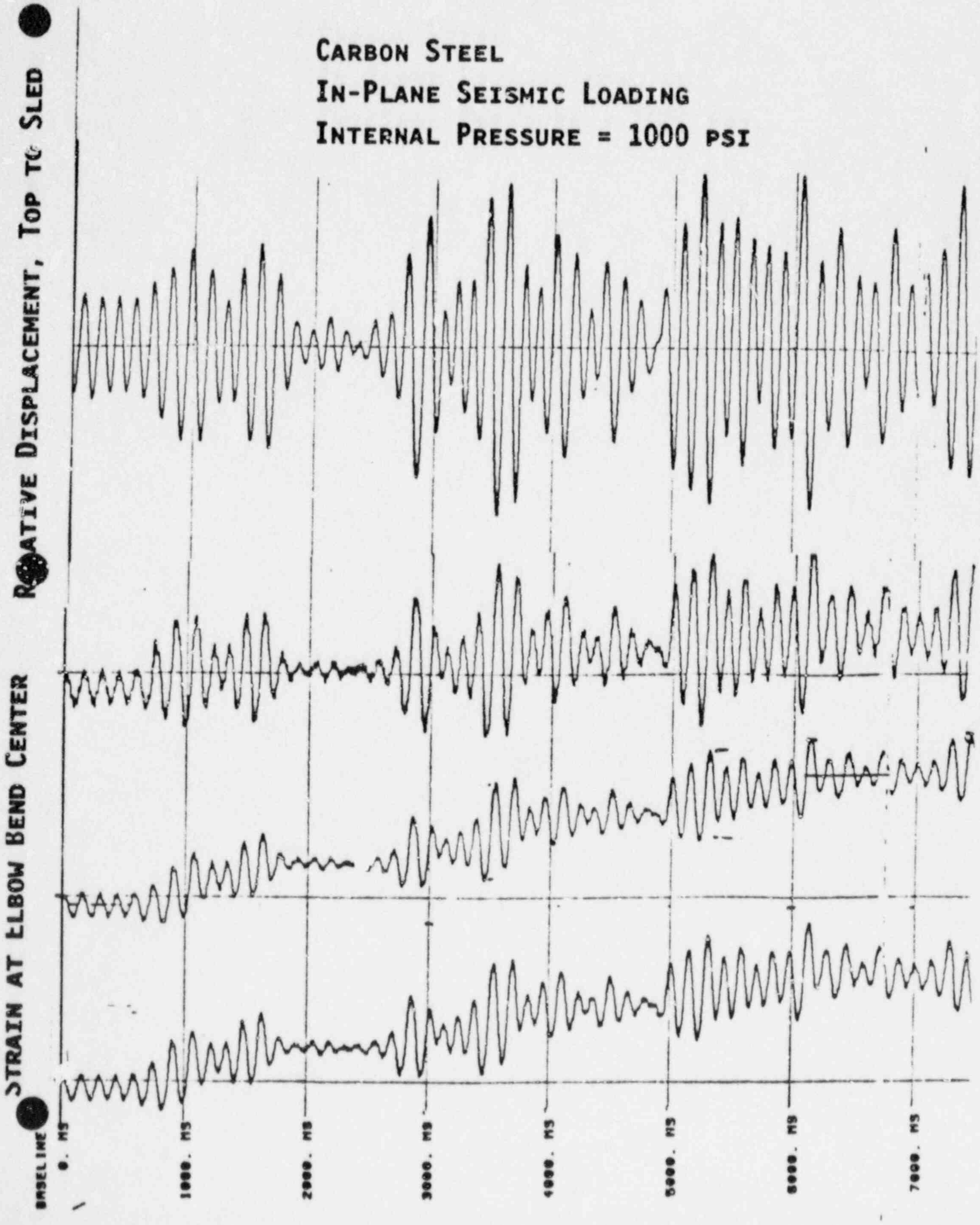
Mid-Frequency Input Motion Time History (SRV)

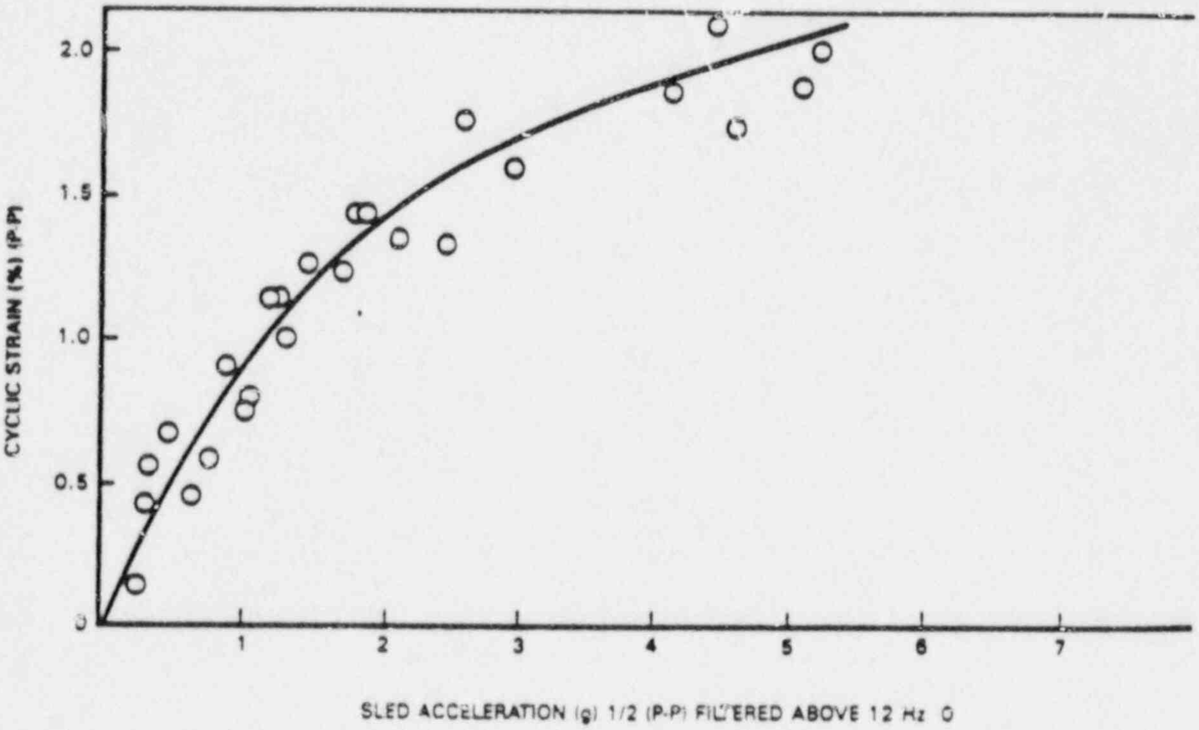


Mid-Frequency Input Motion Response Spectra (SRV)

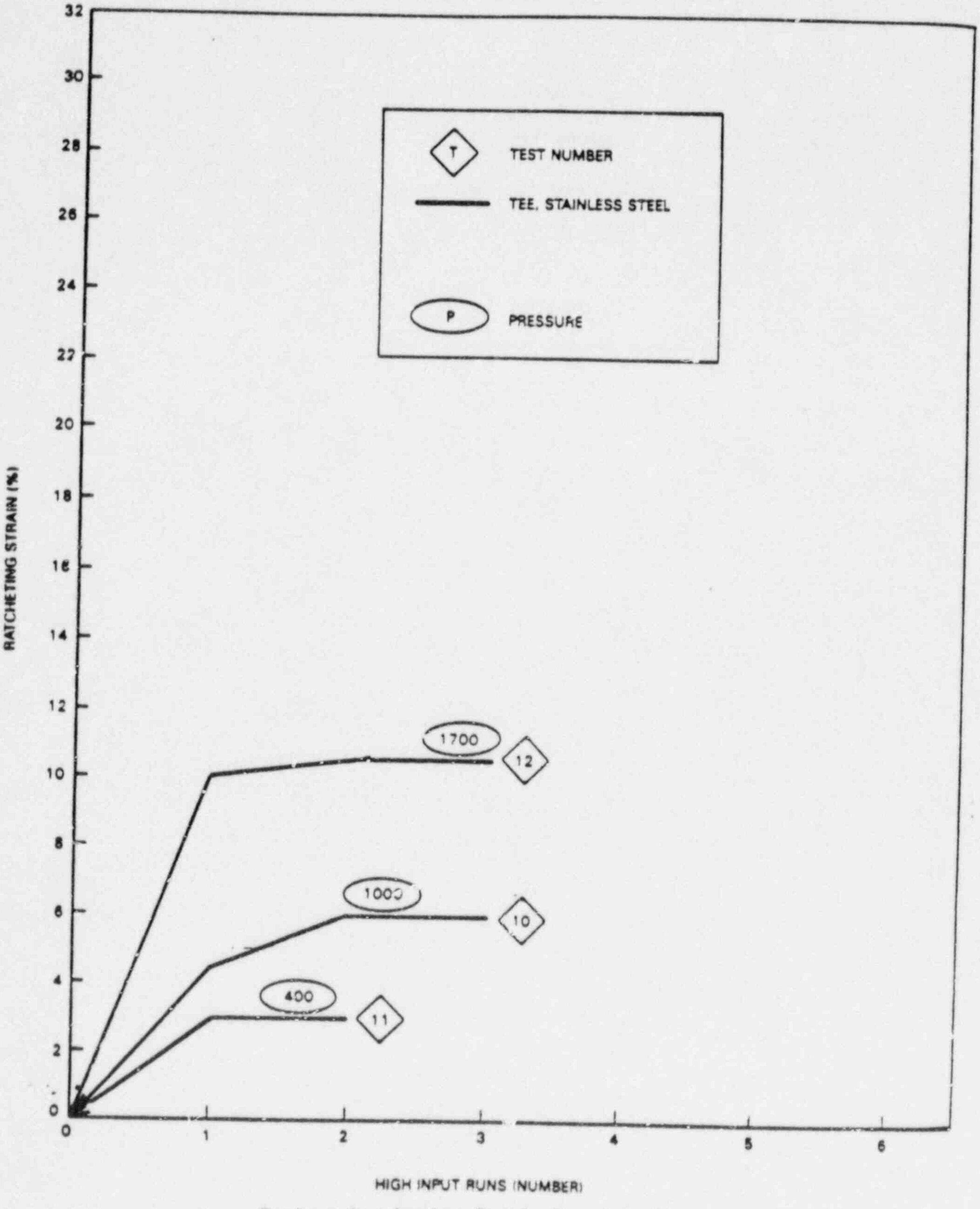
TEST 4 - ELBOW

CARBON STEEL
IN-PLANE SEISMIC LOADING
INTERNAL PRESSURE = 1000 PSI

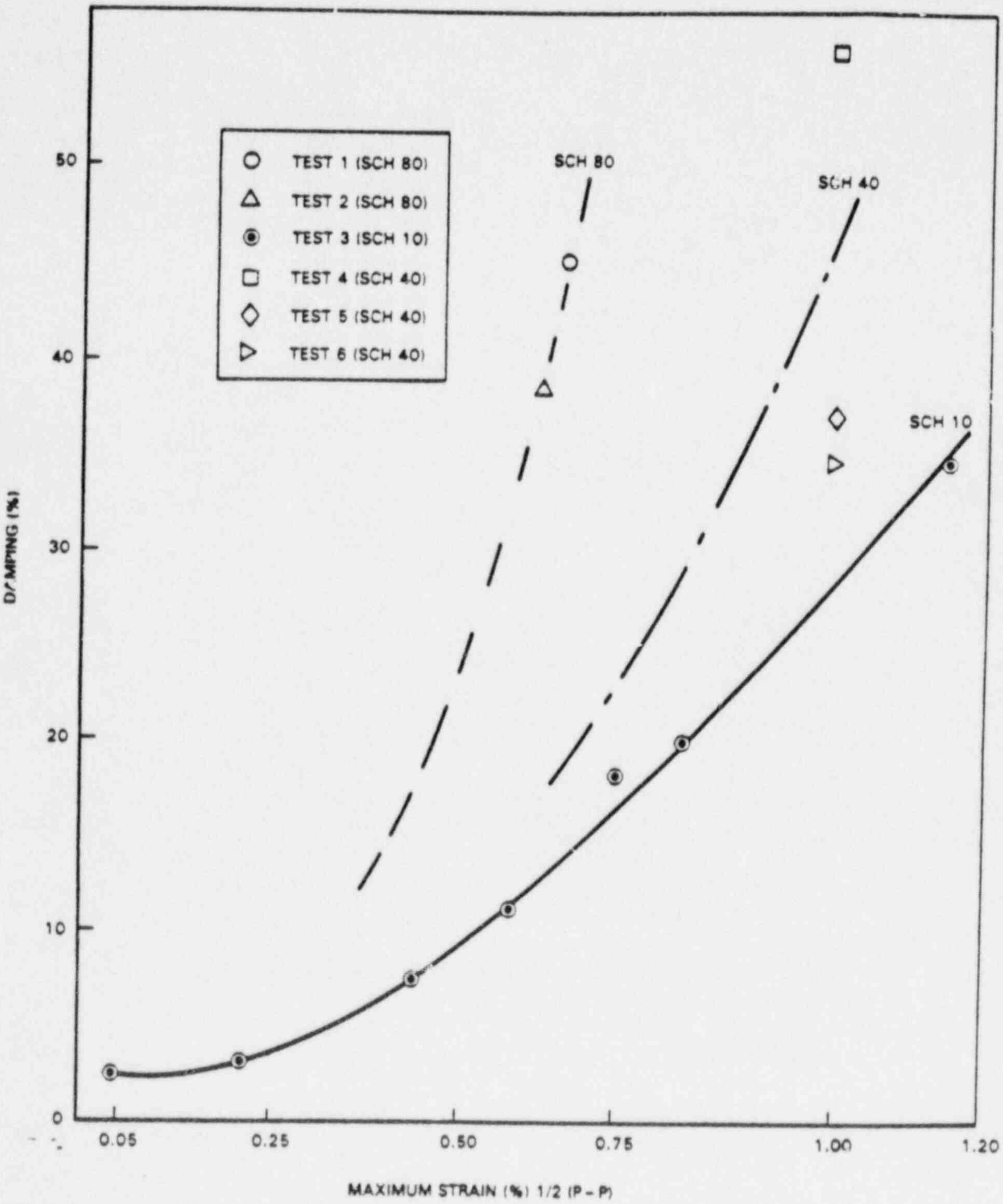




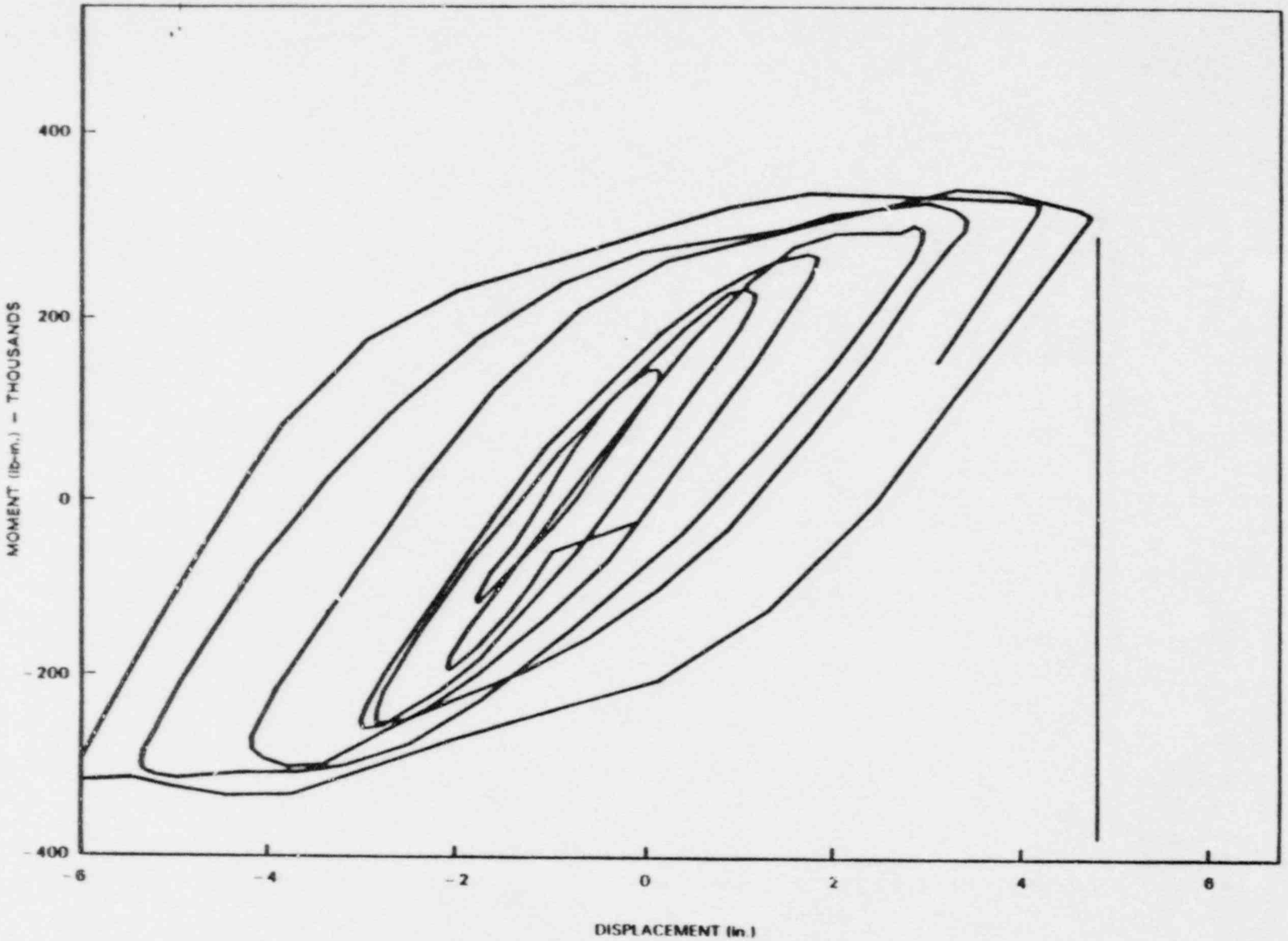
Measured Elbow Strain Versus Input Acceleration
(Test 3 Schedule 10 Elbow)



Fatigue Ratcheting Strain Based on 2-in. Wide Marks

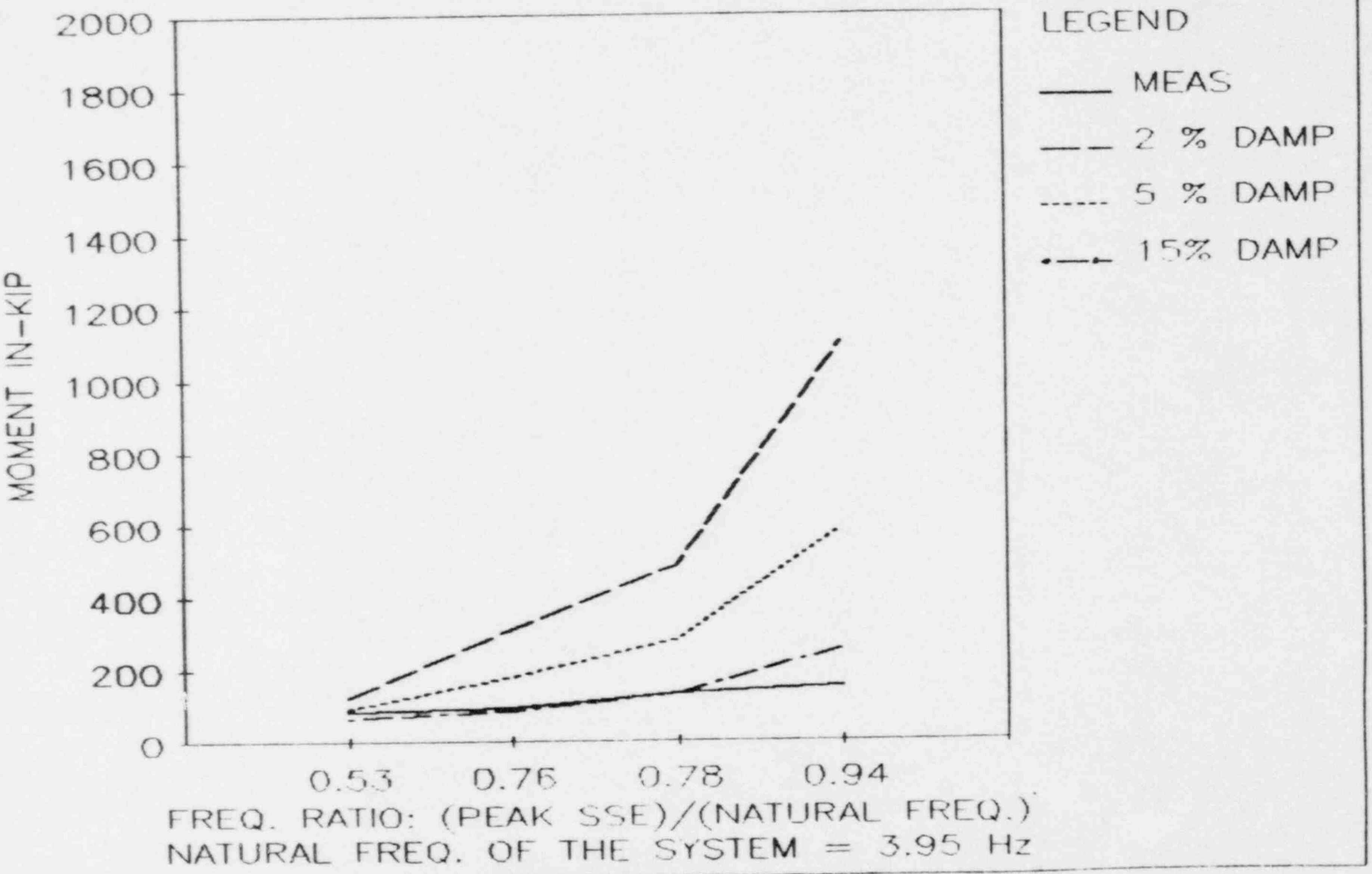


Equivalent Damping Versus Maximum Strain

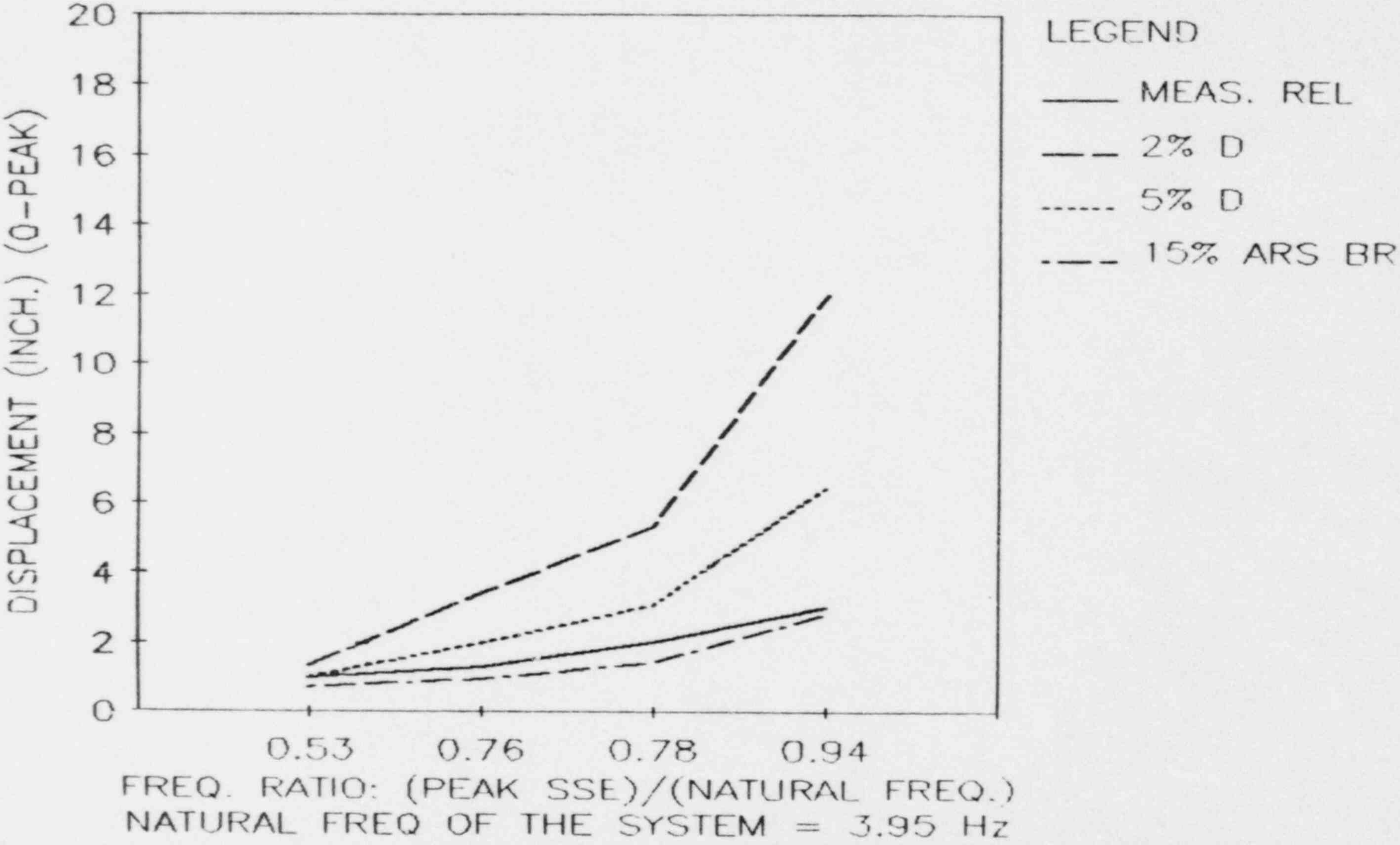


Test 39, Run 4 Moment vs Displacement

T31 6 IN. ELBOW SCH 10 IN–PLANE 410 PSI
CALC MOMENT VERSUS MEASUREMENTS



T31 6 IN ELBOW SCH 10 I-P 410 PSI
DISPLACEMENT VERSUS MEASUREMENT



OBSERVATIONS FROM COMPONENT TESTS

- DYNAMIC LOAD REVERSAL PREVENTS COLLAPSE
- SEISMIC LOADS BEHAVE LIKE SECONDARY NOT PRIMARY
- RATCHET FAILURE LOADS > SSE
- RATCHETING DOES NOT IMPAIR FUNCTIONALITY
- DAMPING FOR LARGE DYNAMIC LOADS > R.G. 1.61
- AMPLIFIED HIGHER FREQUENCY SRV INERTIA LOADS CAUSE SMALL RESPONSE

BOTTOM LINE

FAILURES ARE CHARACTERIZED BY

FATIGUE AND/OR RATCHETING

NOT STATIC COLLAPSE

OBJECTIVES OF SYSTEM TESTS

CONFIRM FAILURE MODE (3 OR 4 SLED INPUTS)

CONFIRM EFFECTS OF LOW AND MID FREQUENCY LOADS

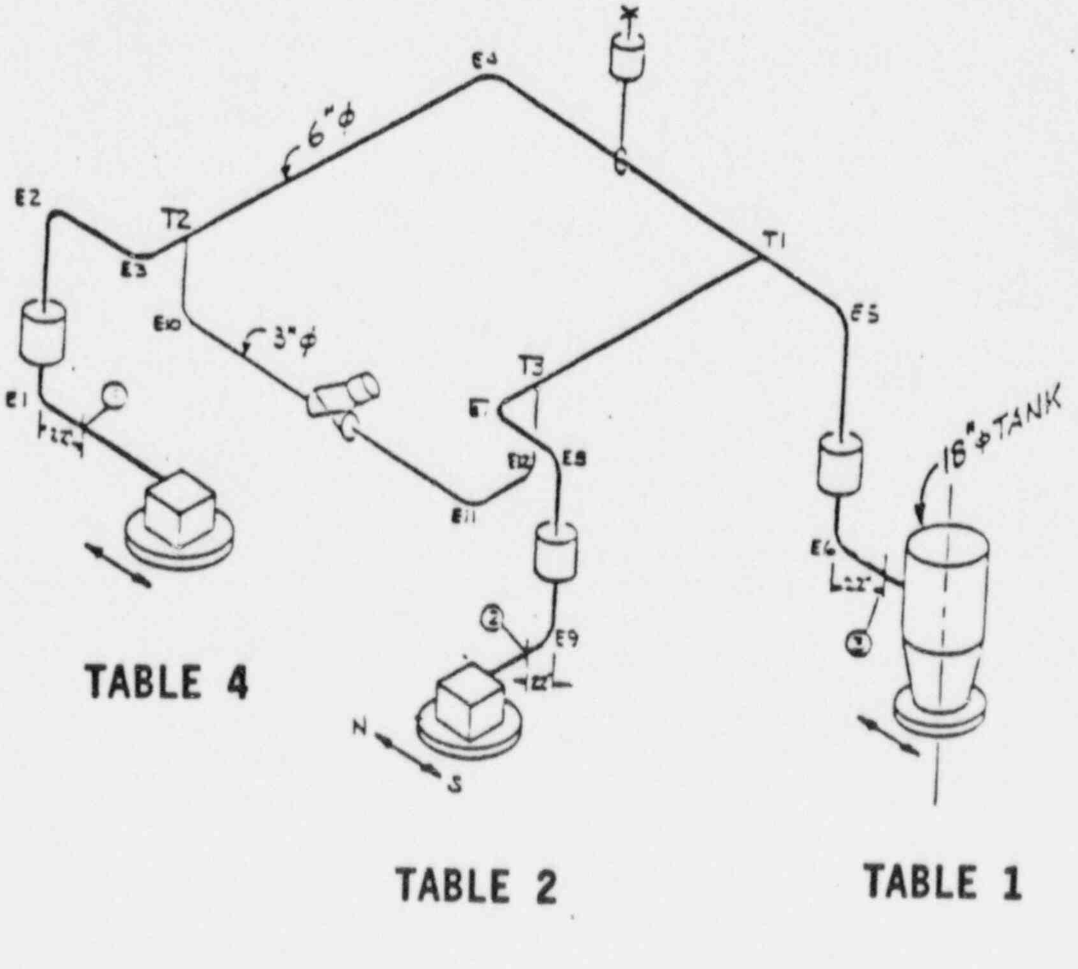
DETERMINE SYSTEM DAMPING
- BALANCED SYSTEM STRESS
- UNBALANCED SYSTEM STRESS
- DIFFERENT TIME HISTORIES
- W & W/O SNUBBER AND STRUT

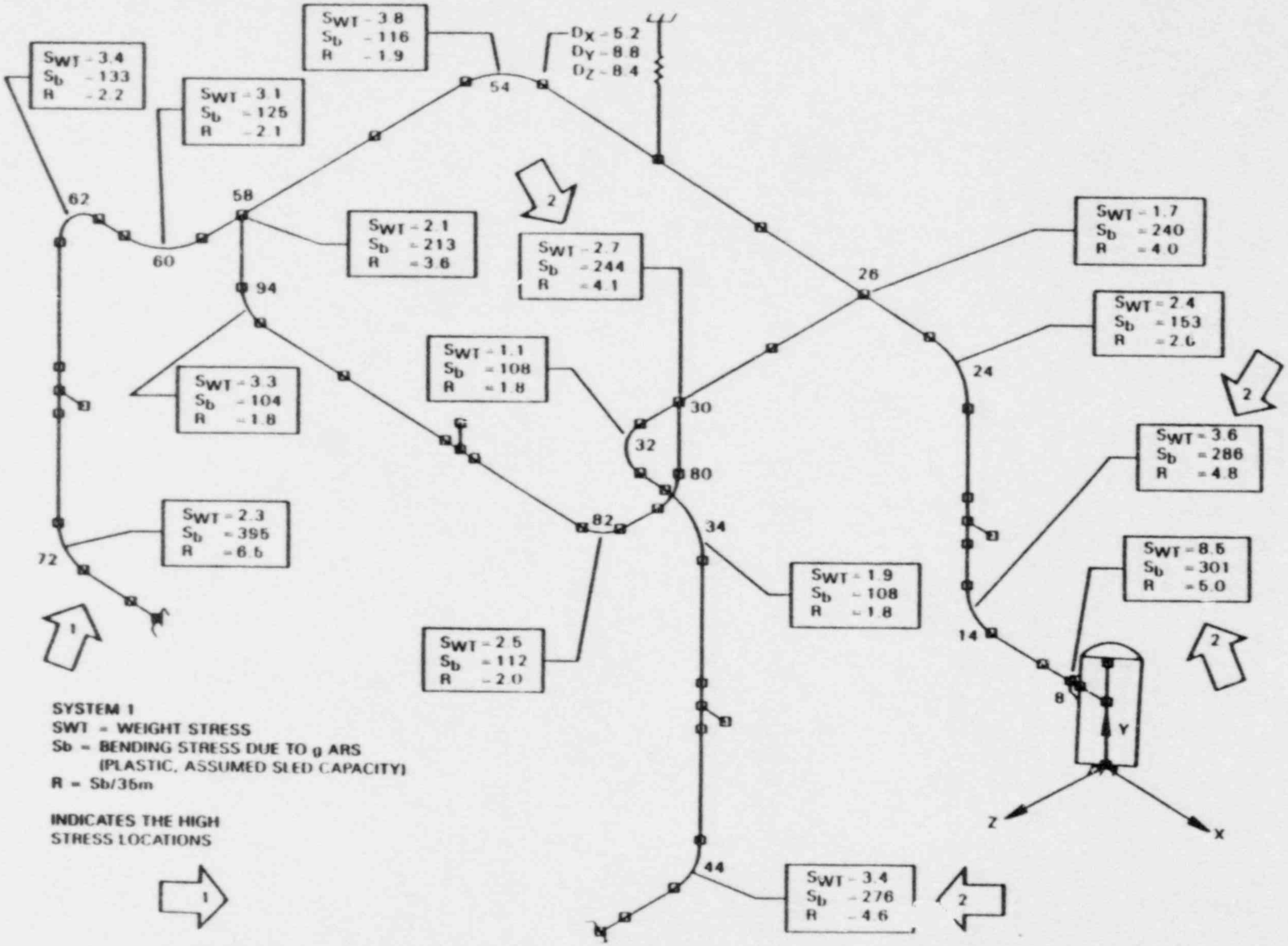
CONFIRM FUNCTIONALITY

CONFIRM DESIGN RULES AND MARGINS

SYSTEM TEST 1

PWR COMPONENT COOLING WATER
THREE SLEDS
CARBON STEEL A106B
INTERNAL PRESSURE = 1000 PSI





SYSTEM 1
SWT = WEIGHT STRESS
Sb = BENDING STRESS DUE TO g ARS
(PLASTIC, ASSUMED SLED CAPACITY)
R = Sb/3Sm

INDICATES THE HIGH
STRESS LOCATIONS

Stress Summary for System 1

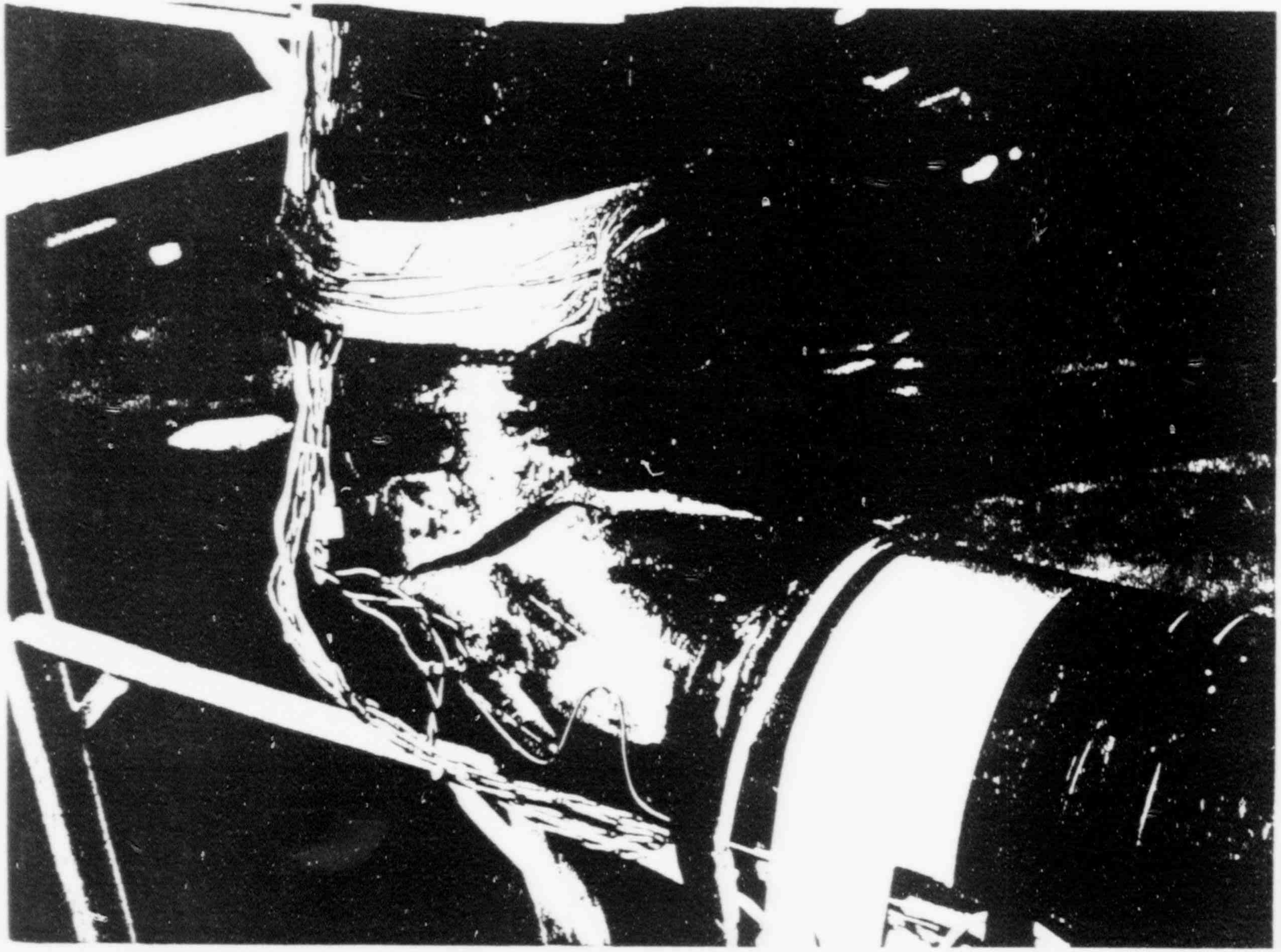
PIPE SYSTEM TESTING

PERFORMED TEST RUNS FOR SYSTEM TEST 1

- OBE .. UNIFORM
- SSE .. ISM
- 5XSSE

- 1/2 TABLE CAPACITY (33 SSE) ..UNIFORM
- FULL TABLE CAPACITY (52 SSE)

- MID FREQUENCY . UNIFORM, FULL TABLE



Side View of Cracked Elbow Near Table 4

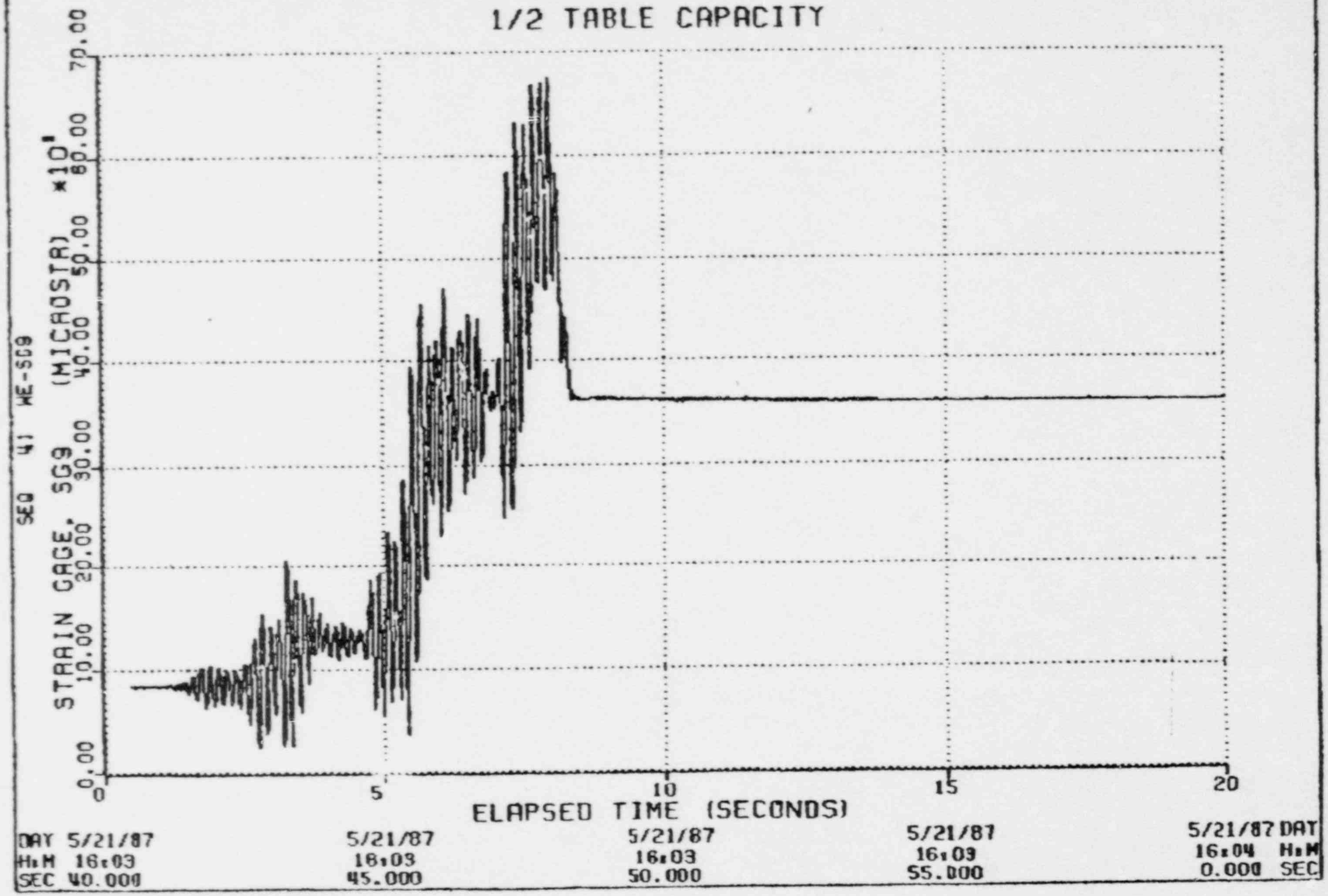
Side View of Cracked Elbow Near Table 4

SYSTEM TEST 1 - TEST RESULTS AT FAILURE LOCATION

PARAMETER	SSE	5SSE (7SSE)	HALF (33SSE)	FULL (52SSE)
INPUT X LEVEL D				
2% DAMPING	0.8	5.3	27.0	42.0
5% DAMPING	0.5	3.1	16.0	24.0
(BROAD, SLED ARS)				
MEASURED MOMENT, IN-KIP	66	330	725	717
CYCLIC STRAIN, P-P, %	0.09	1.2	3.7	4.4
RATCHET STRAIN, DURING RUN, %	0	0.44	8.5	2.4
RATCHET DISPLACE, IN. FOR 20 SEC RUN	0	0	1.0	4.5
DAMPING, %	4.2	6.4	23	-

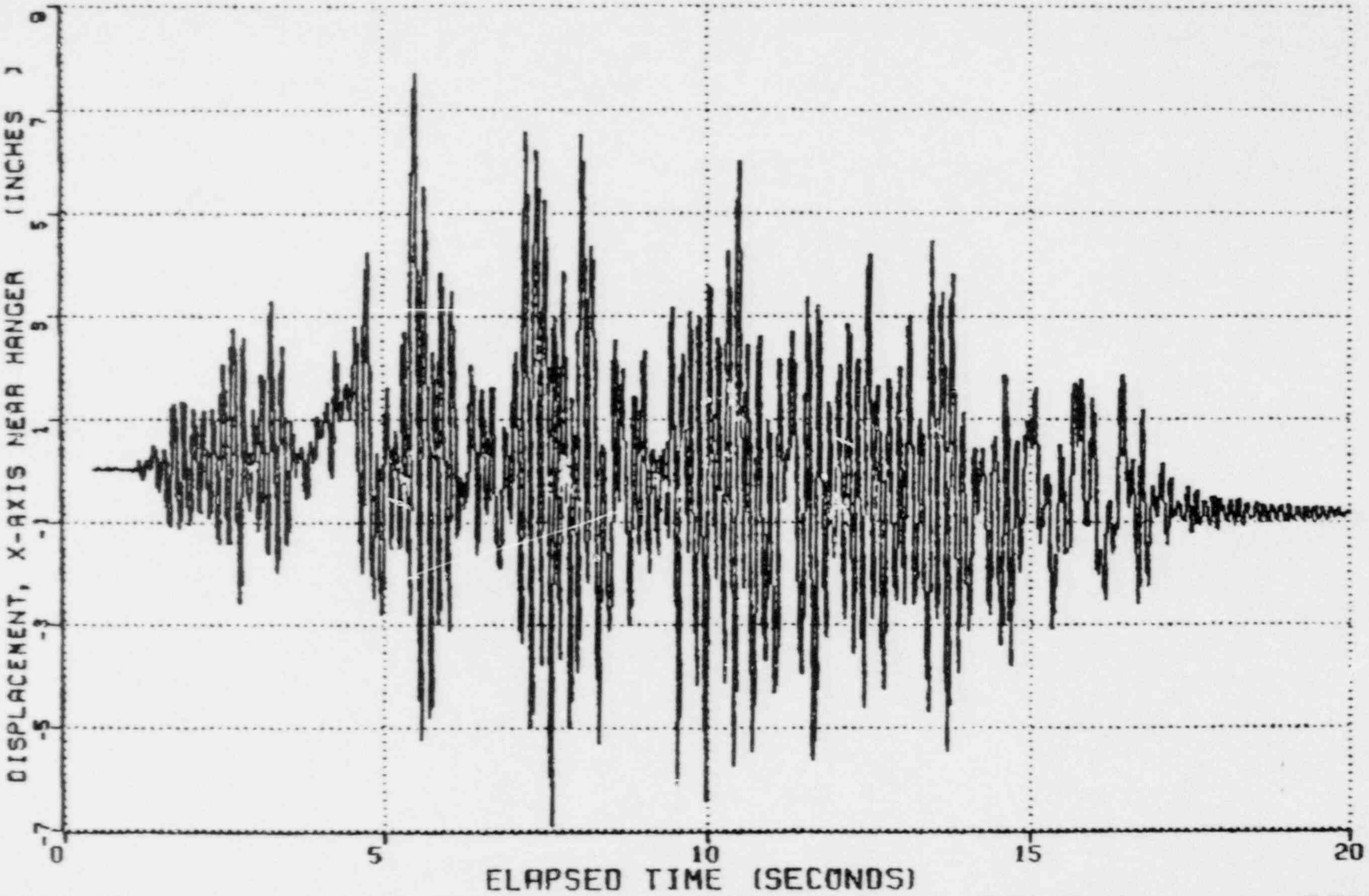
ETEC DATA PL 5/21/87

TTF REAL-TIME PLOT
1/2 TABLE CAPACITY



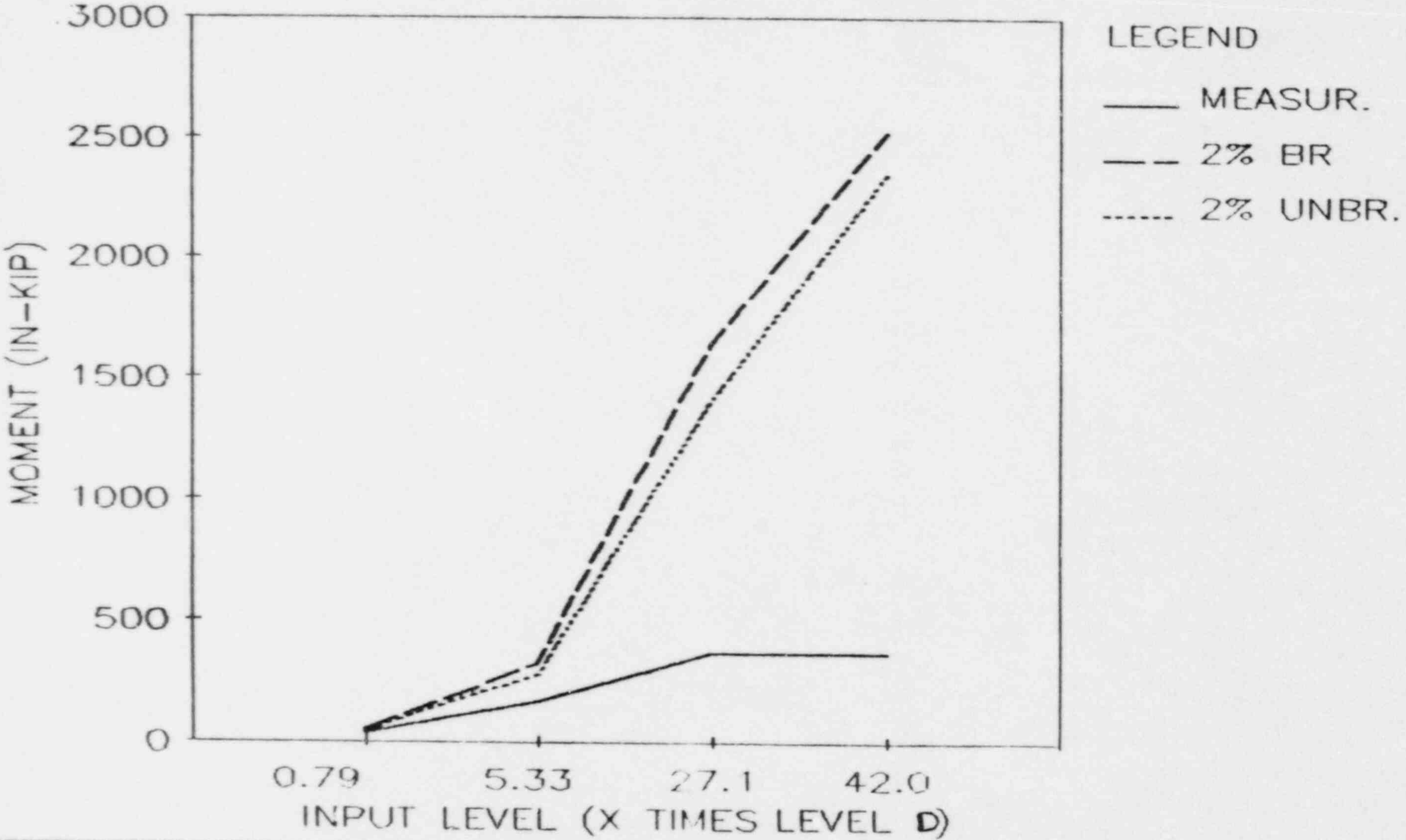
ETEC DATA PL 5/21/87

TTF REAL-TIME PLOT
1/2 TABLE CAPACITY

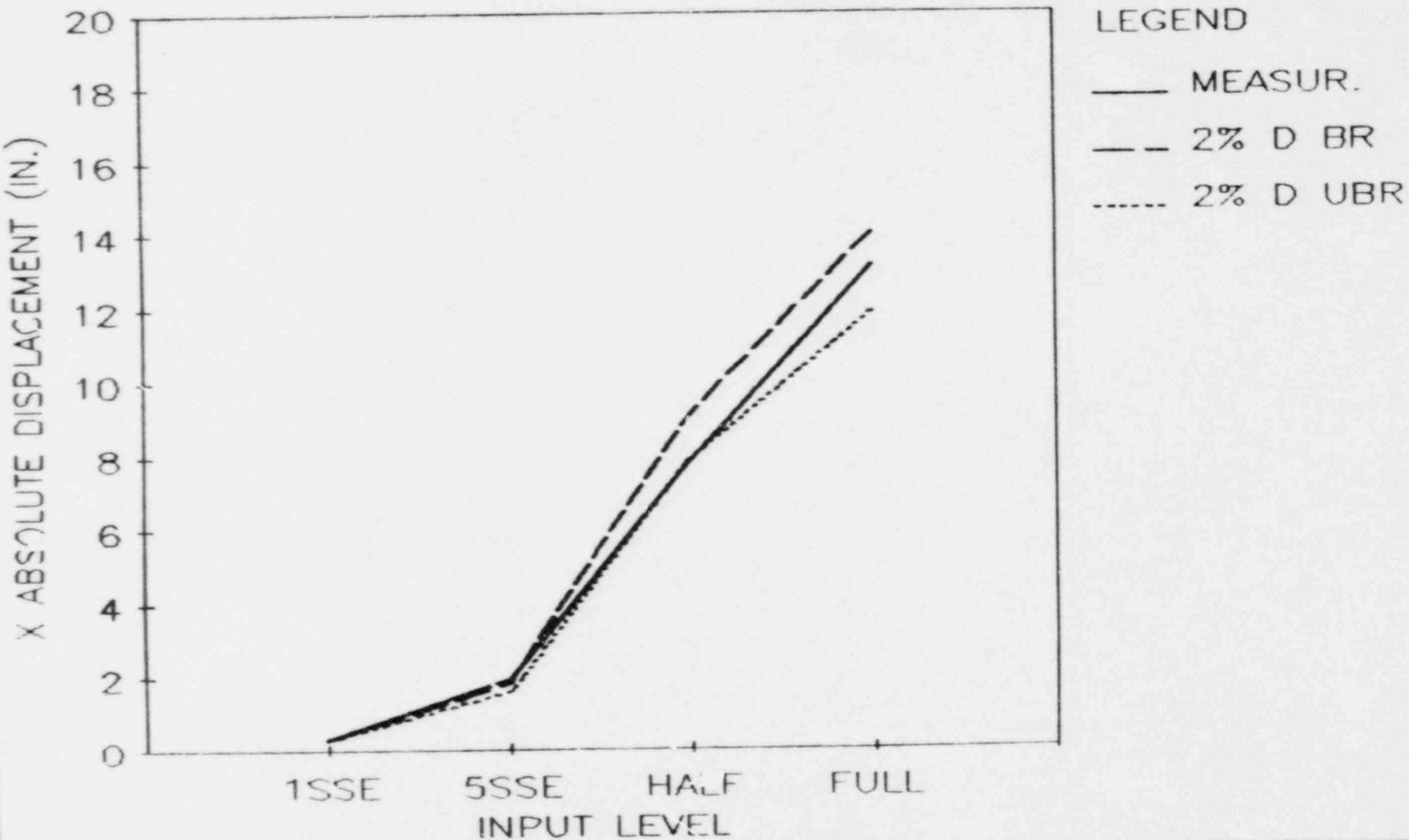


DAY	5/21/87	5/21/87	5/21/87	5/21/87	5/21/87	DAY
H:M	16:03	16:03	16:03	16:03	16:04	H:M
SEC	40.000	45.000	50.000	55.000	0.000	SEC

SYSTEM 1 TEST DATA CORRELATION MOMENT VERSUS INPUT LEVEL 2% DAMPING SLED INPUT ARS

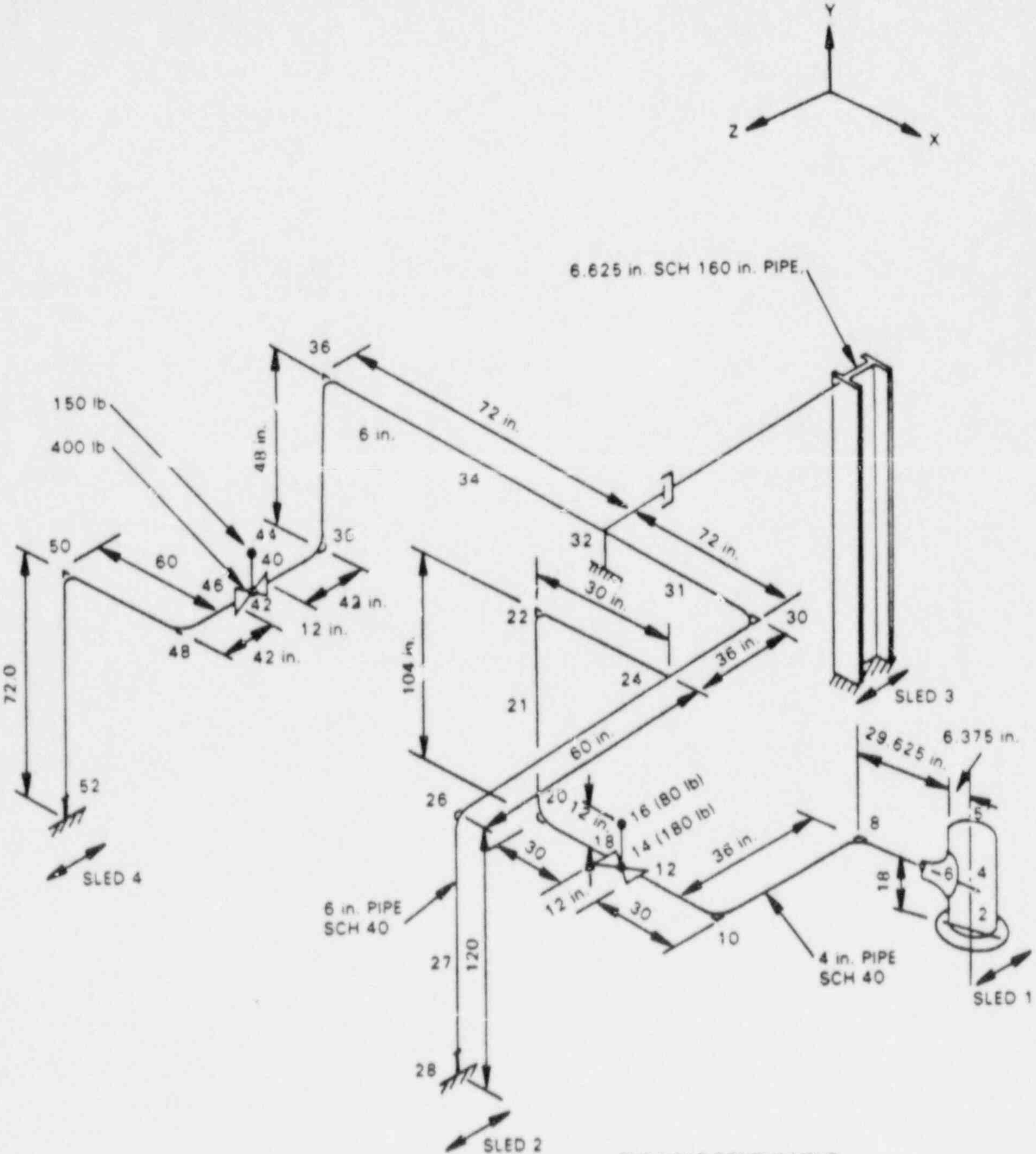


SYSTEM 1 TEST DATA CORRELATION
ABS X DISPLACEMENT VERSUS INPUT
2 % DAMPING SLED INPUT ARS



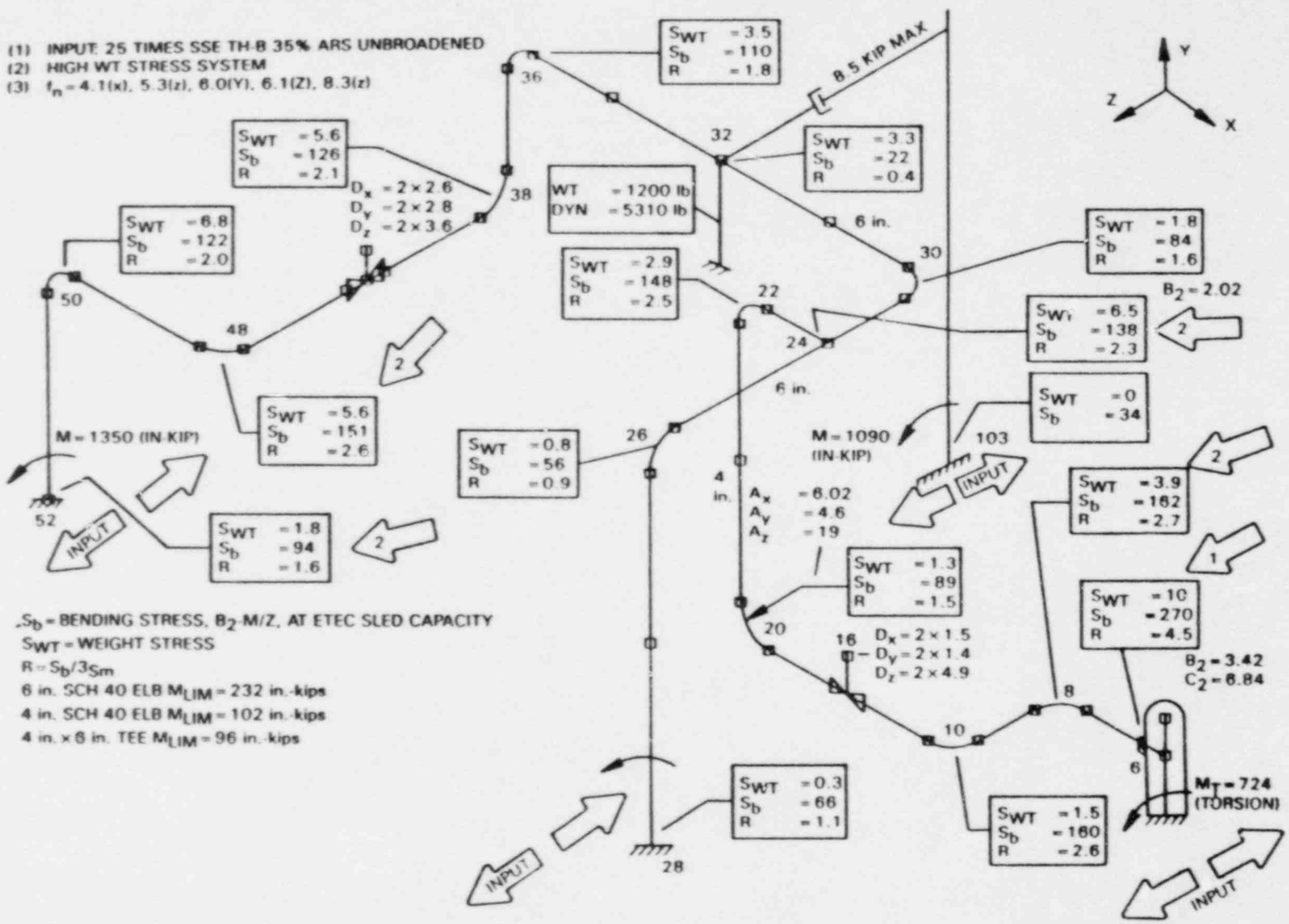
SYSTEM TEST 2

BWR RHR SYSTEM
FOUR SLEDS
STAINLESS STEEL 316L
INTERNAL PRESSURE = 1000 PSI



RHR NEAR CONTAINMENT

(1) INPUT: 25 TIMES SSE TH-B 35% ARS UNBROADENED
(2) HIGH WT STRESS SYSTEM
(3) f_n = 4.1(x), 5.3(z), 6.0(Y), 6.1(Z), 8.3(z)

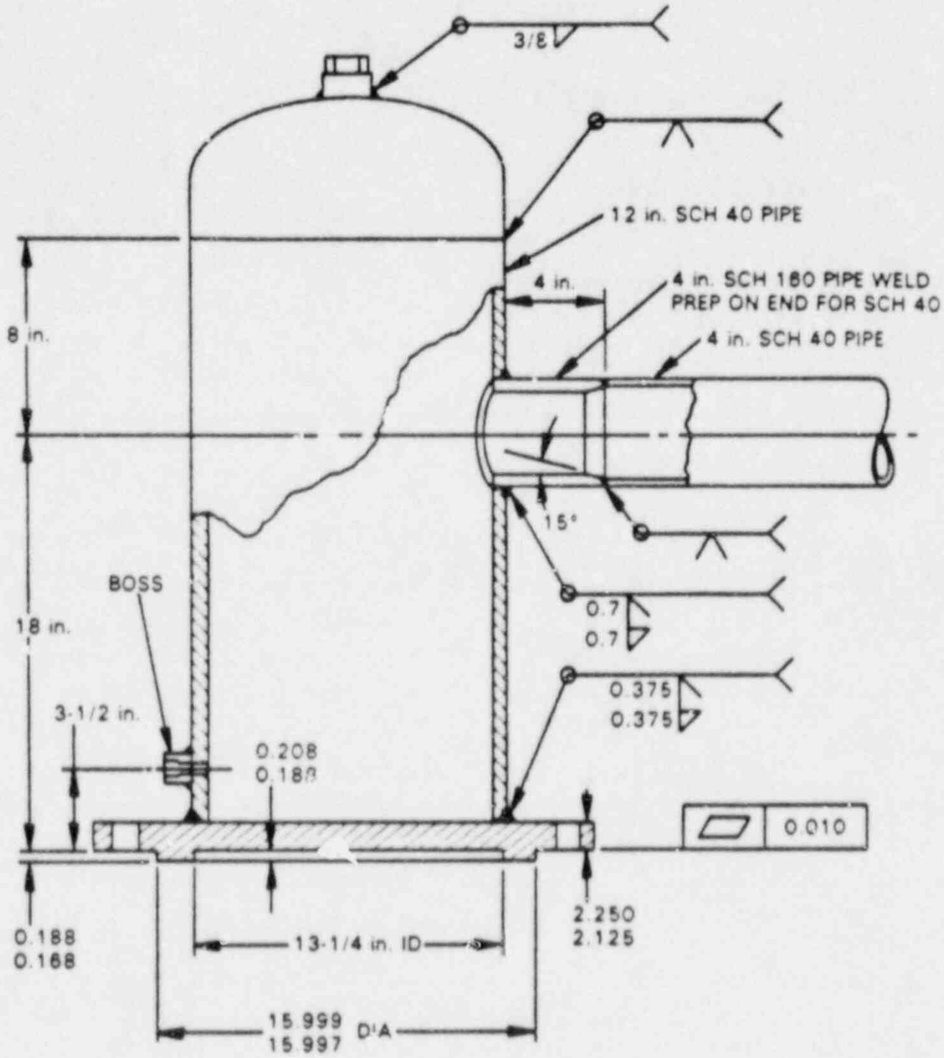


S_b = BENDING STRESS, B_2-M/Z, AT ETEC SLED CAPACITY
S_{WT} = WEIGHT STRESS
R = $S_b/3S_m$
6 in. SCH 40 ELB M_{LIM} = 232 in.-kips
4 in. SCH 40 ELB M_{LIM} = 102 in.-kips
4 in. × 6 in. TEE M_{LIM} = 96 in.-kips

System 2 Stress Diagram

TEST RUNS FOR SYSTEM TEST 2

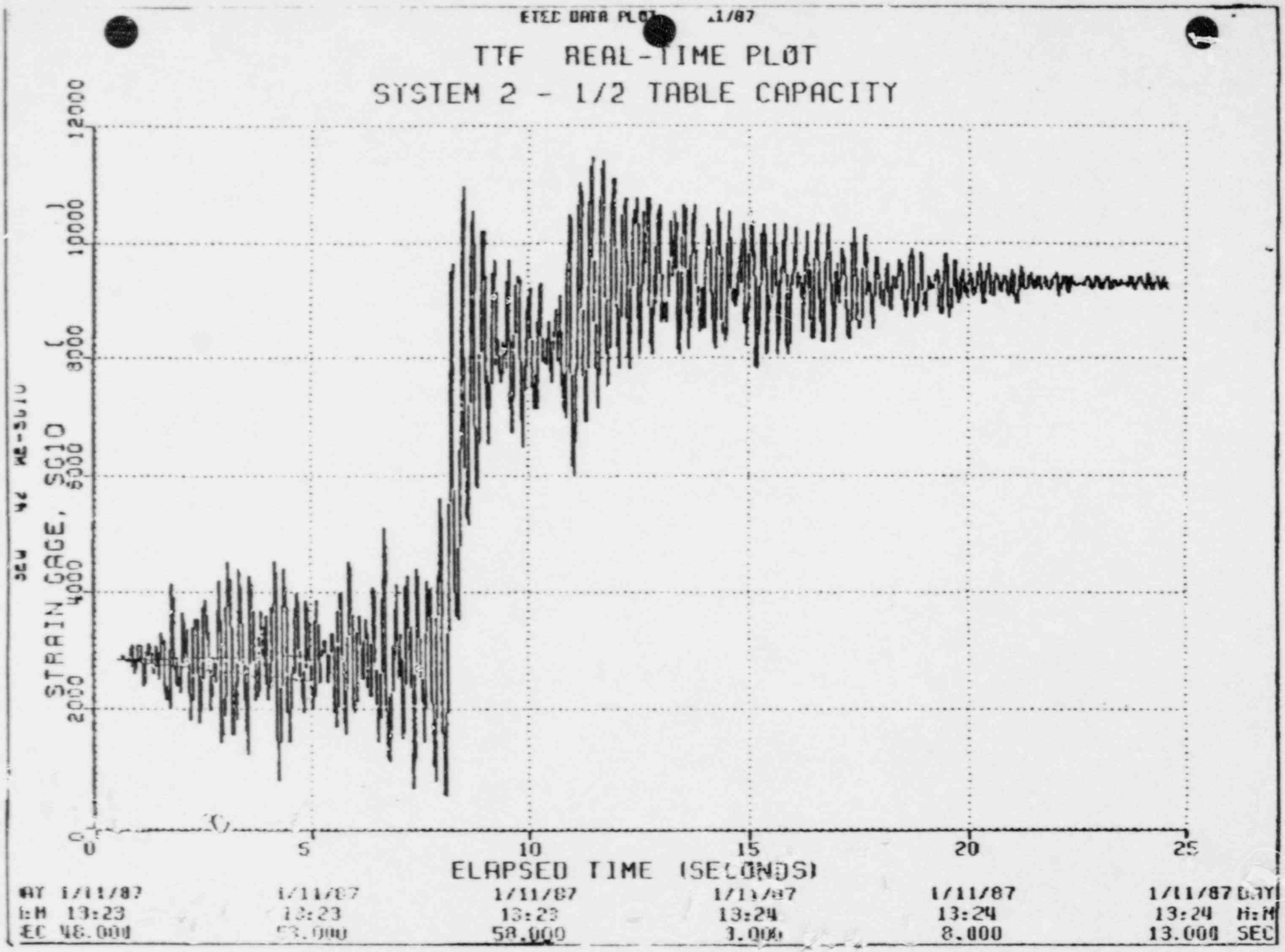
- SSE .. UNIFORM
- 2 x SSE .. ISM-CORRELATED
- 5 x SSE .. ISM-UNCORRELATED
- SINE SWEEP
- HALF TABLE
- FULL TABLE .. UNIFORM
- MID-FREQUENCY



Fabricated Nozzle Test Detail

SYSTEM TEST 2 - TEST RESULTS AT FAILURE LOCATION

PARAMETER	SSE	5SSE	HALF (9 SSE)	FULL (18 SSE)
INPUT X LEVEL D				
2% DAMPING	1.0	6.3	11.0	21.0
5% DAMPING	0.8	4.9	8.0	15.0
(BROAD, SLED ARS)				
MEASURED MOMENT, IN-KIP	39	119	156	235
CYCLIC STRAIN, P-P, %	0.21	0.77	0.96	2.8
RATCHET STRAIN, DURING RUN, %	0.07	0.18	0.65	2.1
RATCHET DISPLACE, IN. FOR 20 SEC RUN	0	0	0	2.1
DAMPING, %	5.0	5.0	22	-



ETEC DATA PLOT .1/87

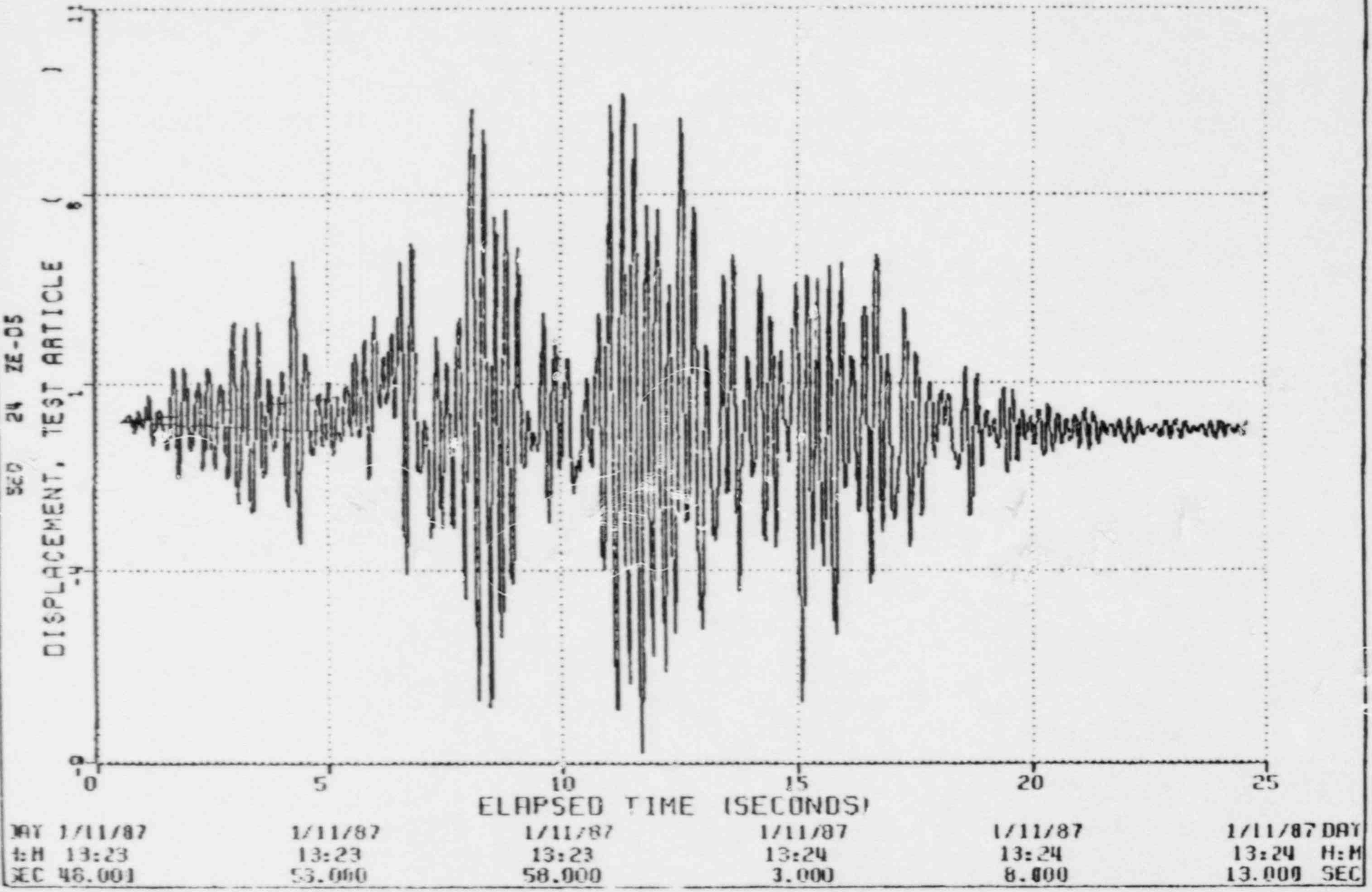
TTF REAL-TIME PLOT

SYSTEM 2 - 1/2 TABLE CAPACITY

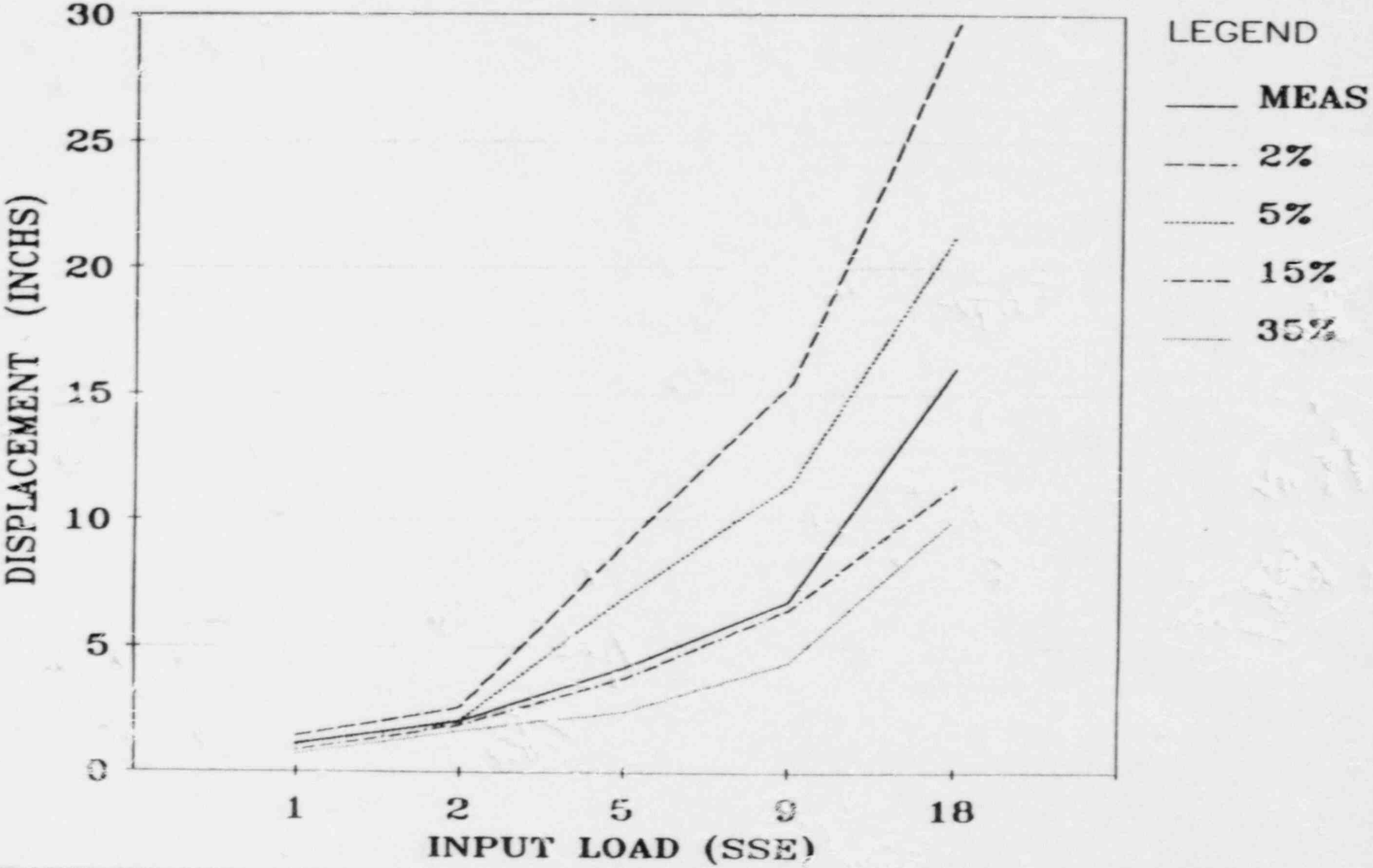
ETEC DATA PLOT 11/87

TTF REAL-TIME PLOT

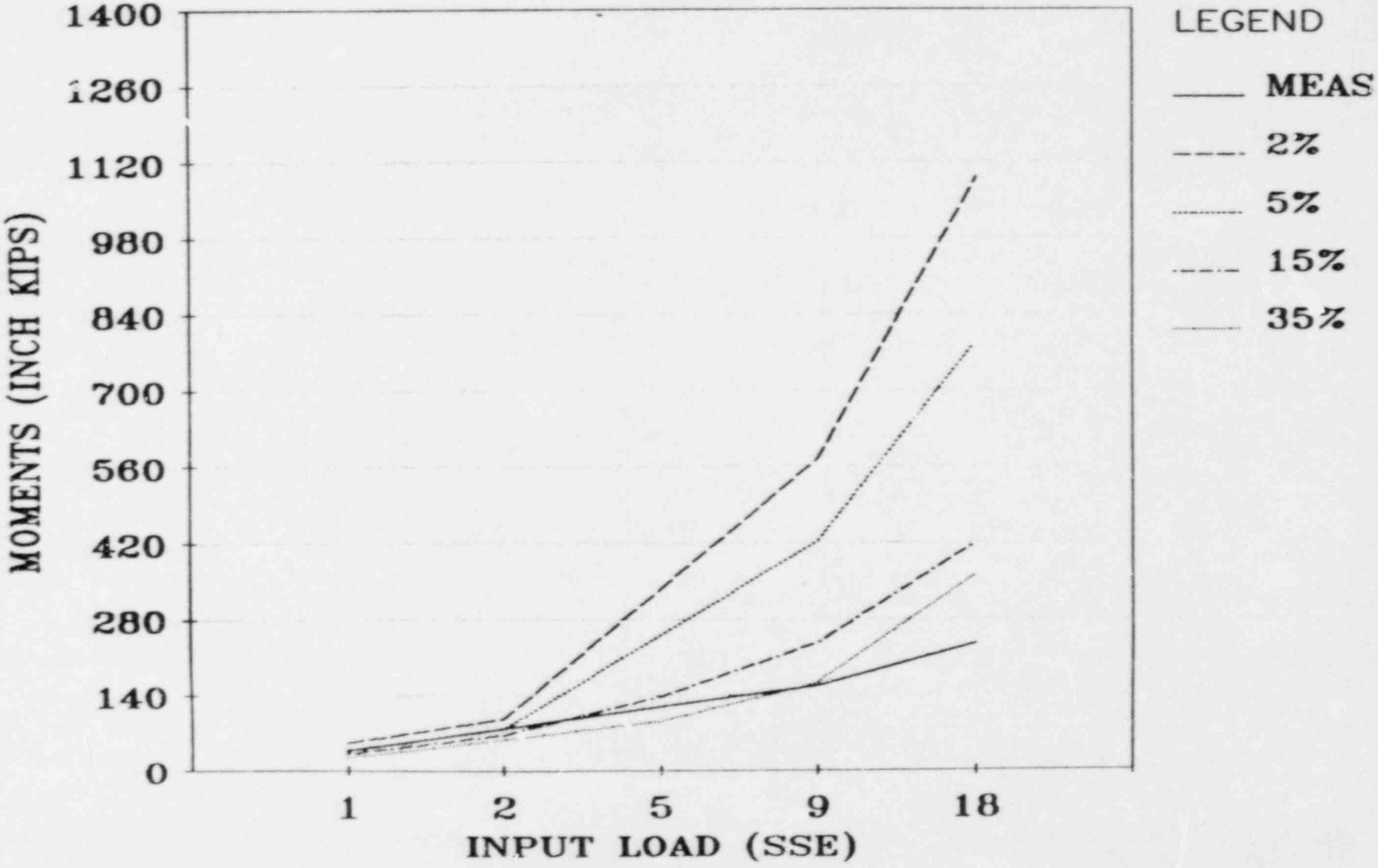
SYSTEM 2 - 1/2 TABLE CAPACITY



DISPLACEMENT MEASUREMENTS VS CALCULATIONS
EPRI SYSTEM TEST #2
NODE 20W (BROADENED ARS)



MOMENT MEASUREMENTS VS CALCULATIONS
EPRI SYSTEM TEST #2
NODE 6 (BROADENED ARS)



OBSERVATIONS FROM SYSTEM TESTS

• FATIGUE RATCHET FAILURE MODE	-	CONFIRMED
• FAILURE LOADS > SSE	-	CONFIRMED
• FUNCTIONALITY UNIMPAIRED	-	CONFIRMED
• DAMPING > R. G. 1.61	-	CONFIRMED
• AMPLIFIED SRV LOADS SMALL	-	CONFIRMED

BOTTOM LINE

EXTREMELY DIFFICULT TO FAIL PIPING WITH DYNAMIC LOADS

WATER HAMMER TESTING

COMPONENT TESTING

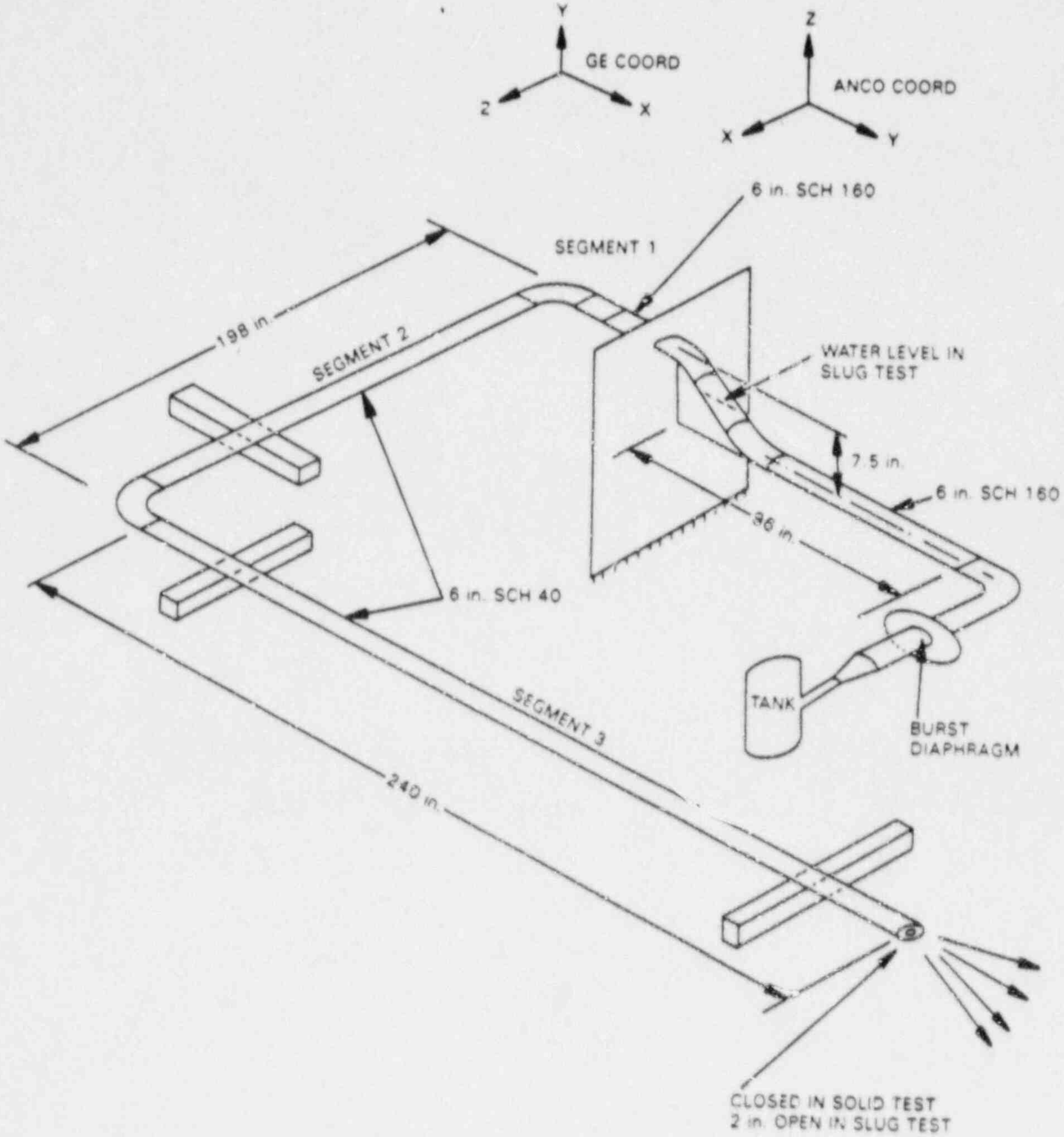
- TWO SMALL LOOPS, TESTS 28 AND 29
- 6-IN PIPING SYSTEMS
- CARBON STEEL, SCH. 40
- WITH AND WITHOUT SUPPORTS

SYSTEM TESTING

- TWO LONGER LOOPS, MINI-SYSTEMS 1 AND 2
- 3-IN PIPING SYSTEMS
- CARBON STEEL, SCH. 40
- SUPPORTS, BRANCHES, SIMULATED VESSEL, THIN PIPE

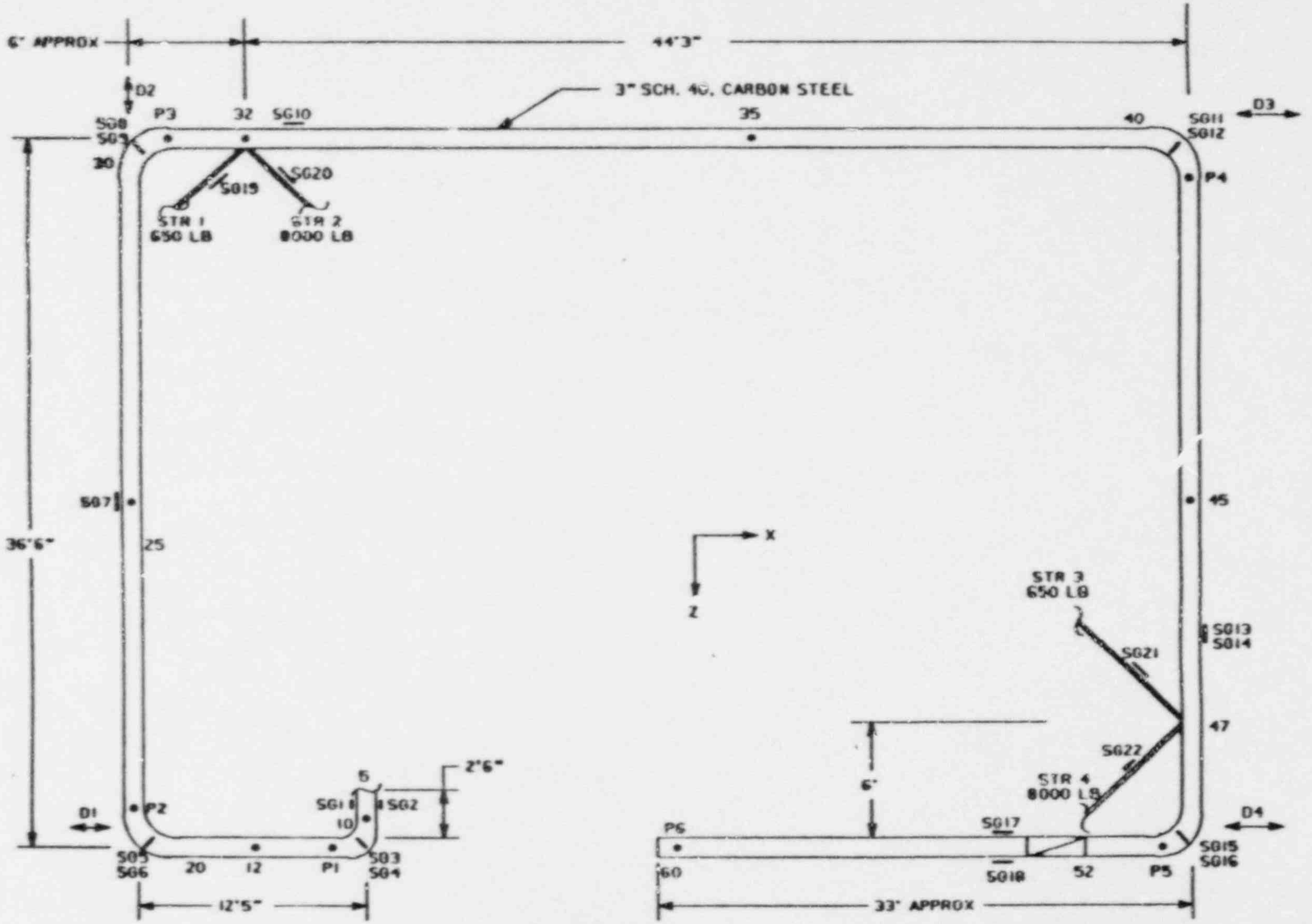
LOADING CONDITIONS FOR WATER HAMMER TESTS

- SIMULATED STEAM HAMMER TEST
- HARD SYSTEM ACOUSTIC TEST
- WATER SLUG TEST
- VARIOUS PRESSURES FROM 150 TO 2000 PSI



Test 28 Water Hammer Test Configuration

MINI SYSTEM I WATER HAMMER



INSTRUMENT	CHANNEL
P1 - P6	6
SG1 - SG 24	22
D1 - D4	4
TOTAL	32

TMP.TZE.HAMMER

PRELIMINARY OBSERVATIONS FROM WATER HAMMER TESTS

- WATER SLUG CAUSES PRIMARY LOADING - CAN COLLAPSE PIPING
- STEAM HAMMER AND HARD TESTS MORE LIKE SECONDARY LOADING - DID NOT COLLAPSE PIPING
- SUPPORTS CAN TOLERATE LOADS 10 X RATED LOAD W/O FAILURE
- PIPE CAN TOLERATE TRANSIENT PRESSURES 2 X BURST PRESSURE W/O FAILURE

BASIC RULE: DESIGN TO AVOID WATER HAMMER

OBJECTIVES OF SPECIMEN TESTS

- DEVELOP LAB SPECIMEN TO EVALUATE FATIGUE RATCHETING WITH MEAN STRESS
- CORRELATE SPECIMEN BEHAVIOR WITH COMPONENT BEHAVIOR
- EXTRAPOLATE CONCLUSIONS FROM 4 TEST MATERIALS TO OTHER PIPING MATERIALS
- INVESTIGATE FATIGUE RATCHETING EFFECTS AT TEMPERATURE (550 DEGREES F)

Modified Test Matrix

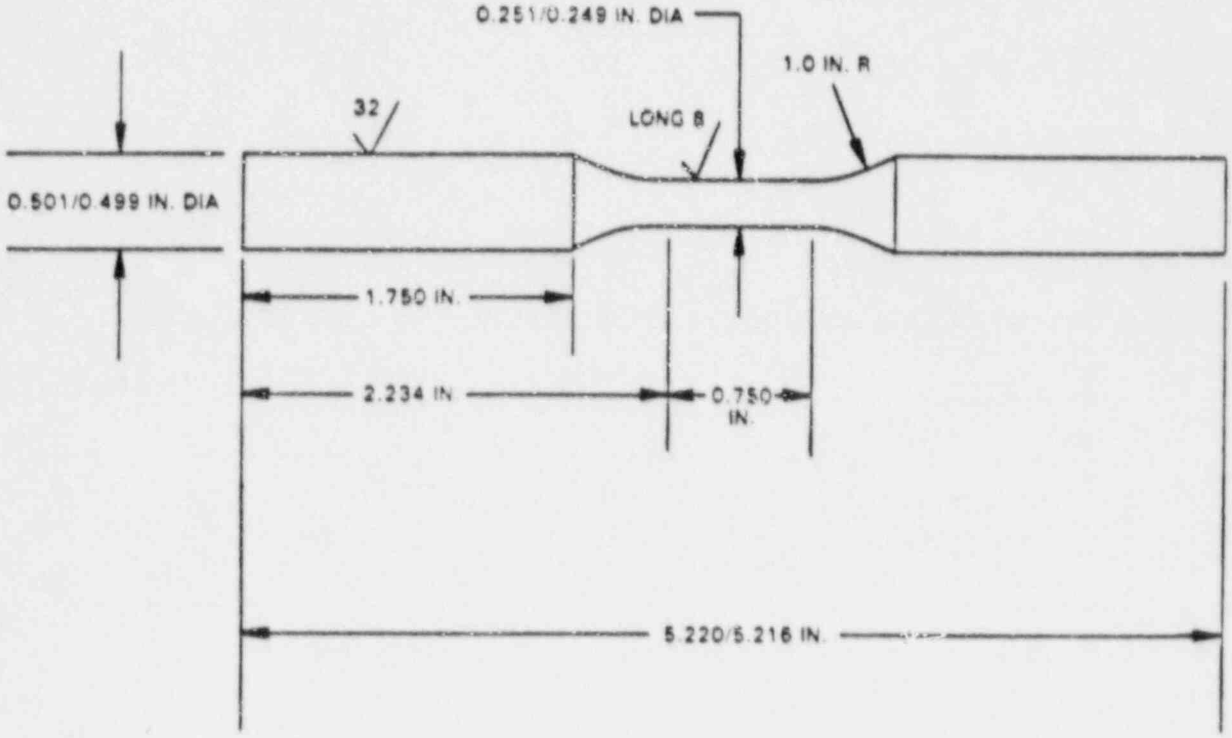
Test Type	Temp F	No. of Mat'ls	No. of Tests	Purpose
Baseline	RT	4	5	Same as original program
Baseline	550	4	4	Find high temp properties
Two Bar Low Mean	RT	4	5	Find effect of low mean stress on all materials
Two Bar High Mean	RT	4	5	Duplicate tests with high mean stress
Two Bar Low Mean	550	4	4	Determine effect of temp on low mean stress test
Two Bar High Mean	550	4	4	Duplicate temp tests for high mean stress
Two Bar Rate Effects	RT	2	8	Investigate strain rate effects on the amount of ratchetting
Ten/Bend smooth	RT	2	4	Verify Ratchetting on 2 materials
Ten/Bend notched	RT	2	2	Test same 2 material for notch effects
Press Pipe	RT	2	2	perform four point bend tests on pressured pipe

Materials Tested

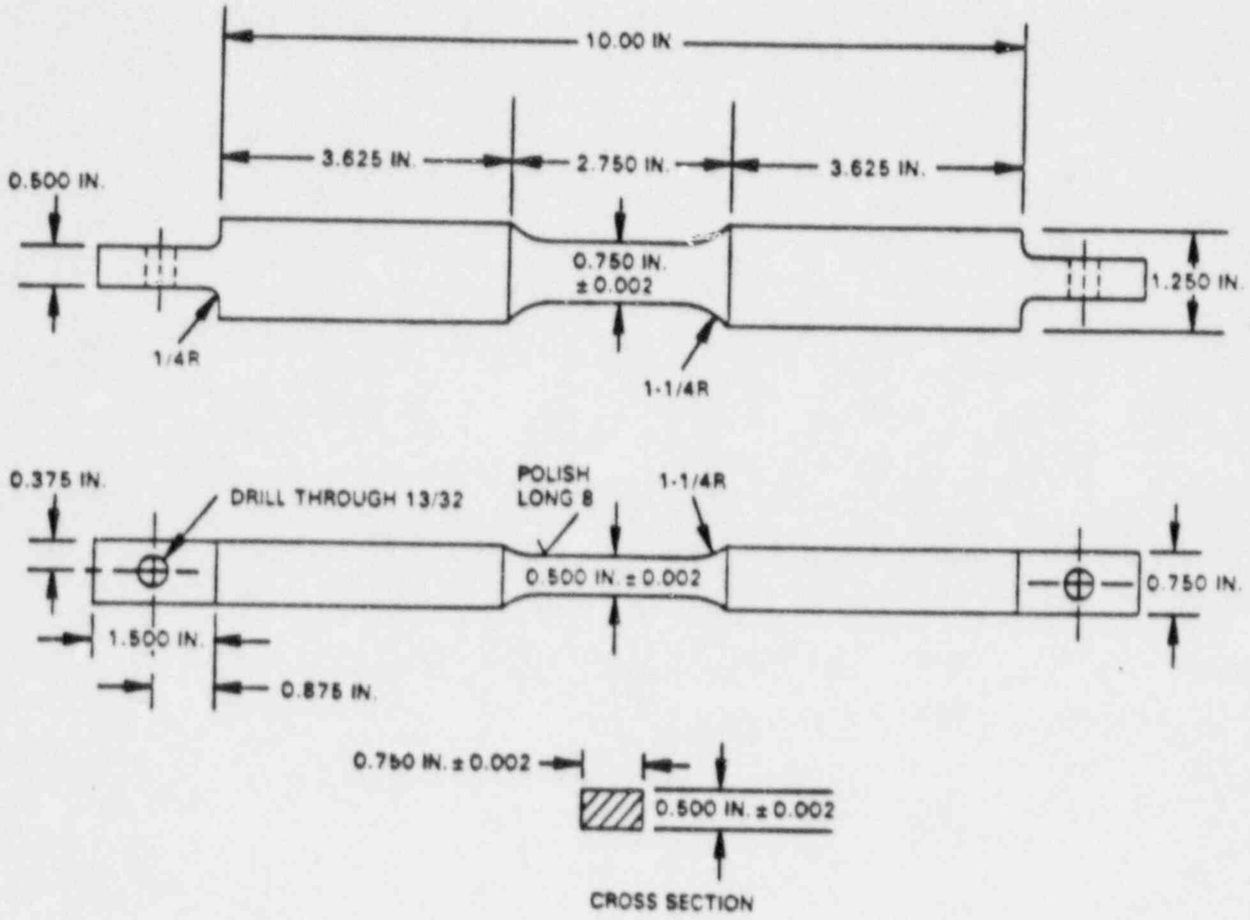
Material 1	A333	Grade 6 Carbon Steel
Material 2	A358	Type 304 Stainless Steel
Material 3	A387	Grade 22 Class 2 Steel
Material 4	A533	Grade B Class 3 Steel

Notes:

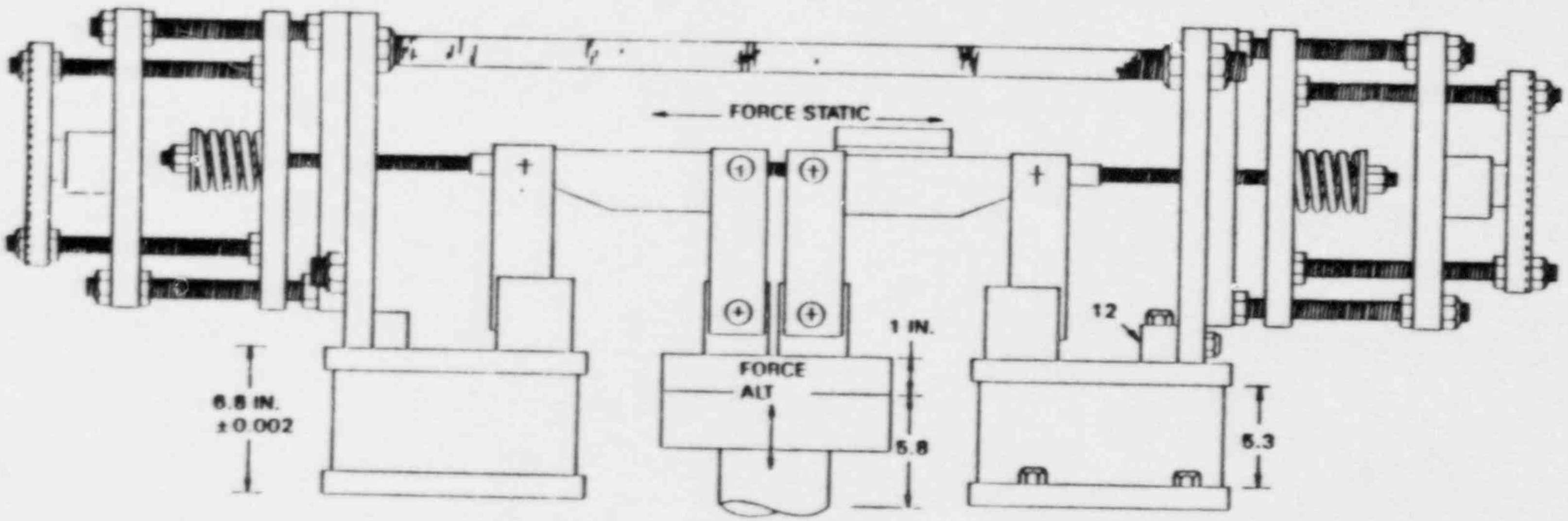
1. When two materials are to be tested they are A333 Carbon Steel and A358 Type 304 Stainless Steel.
2. Number of tests are for each material to be tested.



Uniaxial Test Specimen



Verification Specimen



Test Fixture

OBSERVATIONS FROM SPECIMEN TESTS

- 2-BAR TEST CONSERVATIVELY ESTIMATES EFFECTS OF RATCHETING ON CYCLIC LIFE
- BEAM AND PIPE SPECIMENS CONFIRMED 2-BAR TEST RESULTS
- WITH CONTROLS ON CUMULATIVE RATCHET STRAIN, MEAN STRESS AND TEMPERATURE DID NOT AFFECT CYCLIC FATIGUE LIFE
- CYCLIC CREEP OBSERVED IN LOW FREQUENCY SPECIMEN TESTING MAY NOT BE PRESENT IN HIGH FREQUENCY SEISMIC LOADING