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International Working Group on Fast Reactors
Specialist's Meeting on LMFBR-Reactor-Block
Aseismic Design and Verification

Bologna, Italy
October 12th - 15th, 1987

Seismic Qualification of Piping Sytems Based on Strain Criteria

by

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PAPER NR. 29

Abstract

Typical LMFBR piping is characterized by elevated temperature and low pressure levels. Taking into account operational conditions only these characteristics demand for and allow flexible piping design. The overestimation of the damage potential of seismic loading by e. g. improper failure criteria usually contradicts operational needs producing the known result of excessive "snubberism" and reduction of operational margins.

As a matter of fact, due to its transiency seismic loading is essentially secondary provoking the natural design requirement ductility instead of stiffness and rigidity - i. e. exclusion of failure by strain control instead of stress control - and thus avoiding the LMFBR typical competition between operational needs and seismic qualification.

The design requirement ductility needs judgement mechanisms, i. e. suitable load descriptions, allowed strain levels and strain evaluation tools. A simplified method for strain range estimation and the underlying basic ideas are roughly outlined. The status of verification and experience gained so far is described. The results achieved suggest that the qualification of piping based on ductility requirement controlled by strain criteria is not out of reach.

Contents

1 Introduction

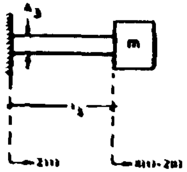
It is well known and often bemoaned that the seismic design on stress criteria basis especially in the case of LMFBR piping interferes with operational conditions asking for flexibility rather than rigidity. On the other hand, due to transiency seismic excitation induces essentially secondary loading so that seismic resistancy could rely on the requirement "ductility" not necessarily in competition with operational needs. Therefore, it is not very astonishing that there are world wide efforts towards a better understanding of failure mechanisms due to seismic loading which this paper is intended to deliver a contribution to.

2 Some Comments on the Secondary Character of Seismic Loading and the Significance of Ductility

It is easy to understand that the work per unit mass done by a transient excitation must be finite. A typical primary load, e. g. dead weight, is potentially able to do infinite work on a structure which is not designed to withstand on a force or stress basis. In this sense seismic loads are secondary. For a component built from ductile materials it is fairly obvious that the proof of seismic resistancy can be obtained by showing that the energy fed to the system can be absorbed without material failure, i. e. by the judgement of maximum strain. These facts are often interpreted in the following way: there exists a "static-dynamic margin" which can be used to allow higher stresses or higher dampings. However, it is simple to see that such a margin does not exist in general and that therefore the design requirement "energy absorption capability" or "ductility" needs special instruments for control. For the following example consisting of two extremely trivial structures ideally elastic-plastic material is assumed.

Structure 1: Two truss elements of Area A_i and lengths l_i carrying a rigid mass m

Structure 2: One truss element of Area A_3 and length l_3 carrying the rigid mass m



The connection between the structures is described by the following equations:

$$A_3 = A_1 \tag{1}$$

$$\frac{A_3}{l_3} = \frac{A_1 A_2}{A_2 l_1 + A_1 l_2} \tag{2}$$

For every choice of l_1, if only l_1 is smaller than l_3, there exists a pair A_2, l_2 with $A_2 > A_1$ so that the equation (2) is fulfilled. It is a simple task to establish the following statements.

- Due to equations (1), (2) and because of $A_1 < A_2$ the two structures cannot be distinguished with respect to force/displacement relation, so that the governing equations for the motion of mass m do not differ neither in the elastic nor in the plastic regime.
- Under the assumption of elastic material behaviour both structures reveal the same stress behaviour (equation (1) and $A_1 < A_2$).

Nevertheless, it is clear that structure 2 is more ductile than structure 1. This can be measured by comparing the hysteresis work $H_i(\varepsilon)$ dissipated by structure i as a function of strain range $\varepsilon > \varepsilon_y$:

$$\frac{H_1(\varepsilon)}{H_2(\varepsilon)} = \frac{l_1}{l_3} \to 0 \text{ for } l_1 \to 0 \tag{3}$$

or for given hysterisis work H the ratio of plastic strain ranges $\varepsilon_{pl,i}$ needed by structure i is evaluated to be

$$\frac{\varepsilon_{pl,1}}{\varepsilon_{pl,2}} = \frac{l_3}{l_1} \to \infty \text{ for } l_1 \to 0 \tag{4}$$

According (3) and (4) a static-dynamic margin does not exist in a general sense. The ductility of a structure must be identified explicitely.

3 Outline of a Method to Support Ductility Design

The lesson learned from chapter 2 is that a design based on ductility requirements controlled by strain limitation is not possible without special non-linear information about the component to be designed. Full non-linear F(inite) E(lement) computations in the time domain - although in principle able to take into account all non-linear information implicitely - are not useful to accompany a design process. There is a deeper reason for that than a mere effort argument: a time history FE-computation for a complete structure delivers <u>one</u> singular event. There is no possibility given

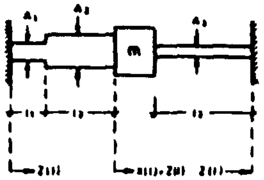
- to characterize the excitation, i. e. the class of time histories to be taken into account for ductility proofs
- to check a design measure in a manageable way.

The straight forward remedy consists in an approach based on simplified substitute structures analogously to strengh design controlled by stress limitation. In this well known case the component is projected on a set of linear 1-DOF-systems connected together by a stress superimposition rule. To support the ductility design the substitutes must reflect the following phenomena:

- load limit leading to plastifying
- selectivity with respect to excitation time histories
- geometric hardening effects
- strain concentration or equivalently energy absorption capability in the plastic regime

The example of chapter 2 shows that it is important that these phenomena are described by parameters dependent on strain, i. e. the failure relevant variable. It is felt that this principle is constituent for any substitute approach.

The most simple structures revealing sufficient complexity to serve as substitutes consist of three massless truss elements of areas A_i and length l_i carrying a rigid mass M:



$A_1 < A_2,\ l_1 \ll l_3$

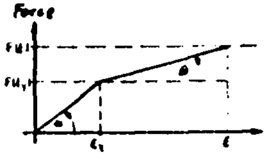
If elastic-plastic material is assumed these structures form a four-parametrie set R(ω, ρ, s, v) with respect to strain answer:

ω = elastic natural frequency

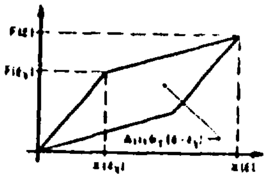
ρ = acceleration to induce unity stress

s = geometrie hardening coefficient

v = volume coefficient



$$s = \frac{tg\,\beta}{tg\,\alpha}$$



$$v = \frac{A_1 L_1 \sigma_y (\varepsilon - \varepsilon_y)}{\frac{\varepsilon - \varepsilon_y}{\varepsilon_y} F(\varepsilon_y) X(\varepsilon_y)}$$

The meaning of s is obvious, v compares the actual hysteretic work with a normalized ampleness of hysteresis.

Fundamental for the projection of a given component into the set $R(\omega, \rho, s, v)$ is the concept of <u>quasistatic process</u>. If ω_i is a natural frequency and ϕ_i the correspondant mode shape induced by the force distribution F_i, than the quasistatic process $0 \rightarrow \alpha F_i$ in ϕ_i-direction is defined to be a force time-history $t \rightarrow f(t)F_i$ so that $0 < \dot{f}(t)$ is arbitrarily small and $\max f(t) = \alpha$. Especially $0 \rightarrow \alpha_i(\varepsilon)F_i$ is the process to induce the strain ε within the component.

The quasistatic processes $0 \rightarrow \alpha_i(\varepsilon)F_i$ deliver functions

$$\varepsilon \rightarrow s_i(\varepsilon) = \frac{\dfrac{\alpha_i(\varepsilon)}{\alpha_i(\varepsilon_y)} - 1}{\dfrac{\varepsilon}{\varepsilon_y} - 1}$$

$$\varepsilon \rightarrow v_i(\varepsilon) = \frac{H_i(\varepsilon)}{(\dfrac{\varepsilon}{\varepsilon_y} - 1)\alpha_i^2 \quad (\varepsilon_y)\omega_i^2}$$

Here $H_i(\varepsilon)$ is the energy dissipated by the process. $s_i(\varepsilon)$ and $v_i(\varepsilon)$ can easily be shown to generalize the concepts of s and v. If now $\hat{\rho}$ is defined to be the acceleration delivering unity stress within the component assuming RSS-superimposition of modal stresses, then there is defined a projection

$$\mathit{Component} \rightarrow (\varepsilon, \omega_i) \rightarrow R(\hat{\rho}, \omega_i, s_i(\varepsilon), v_i(\varepsilon))$$

For ε the allowable strain range and for a given excitation the maximum strain range experienced by a substitute $R(\hat{\rho}, \omega_i, s_i(\varepsilon), v_i(\varepsilon))$ is an estimation of the maximum strain range experienced by the projected component. Non-linear computations for the whole component are restricted to be quasistatic while time history computations are only necessary for the substitutes.

4 <u>Status of Verification, Experiences</u>

The methodology outlined in chapter 3 has been checked against non-linear FE computations of simple structures (2 DOF-Systems, geometrically simple piping arrangements), experiments and computations on a one inch piping configuration carried out by Westinghouse Hanford Company (U.S. Department of Energy Contract No. DE-AC06-76FF02170) and two of a serie of seismic experiments carried out by KWU (Kraftwerk Union) on a geometrically complex piping system consisting of DN 100 main and DN 50 branch piping. In all cases the strain range estimation has been conservative with a maximum overestimation factor of 2. Most experience could be gained by the recalculation of the two KWU-experiments:

	KWU-Experiment V57 (Korinth TH)	KWU-Experiment V78 (Konvoi TH)
Linear elastic Stress (Response Spectrum)	4.6 σ_y	5.0 σ_y
max. strain range, ANSYS computation	0.90 %	0.90 %
max. strain range measured	1.2 %	0.75 %
max. strain range estimated (chapter 3)	1.8 %	1.3 %

The computational effort for strain estimation needed for the recalculation of maximum strain range using the substitute approach amounted to 2 to 3 times the effort for a response spectrum calculation of stresses.

5 Conclusions

Although there remain open questions especially with respect to excitation characterization and material properties the mehodologies developed and experiences gained so far suggest that the seismic qualification of piping based on ductility requirements controlled by strain criteria is not out of reach. Although the computational effort will naturally rise the benefit with respect to a better balance between operational and seismical design is worth further development and completion. A synopsis of German Efforts to support the adequate LMFBR-piping design is contained in "Nuclear Engineering and Design 96 (1986)".

Seismic Analysis Method of MONJU Liquid Metal Pipings
and
Verification with Seismic Tests on Model Pipings

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PAPER NR. 30

abstract

The heat transport system pipings of the prototype LMFBR "MONJU" are thin walled pipings with large diameter and are supported by pipeclamps and mechanical snubbers, therefore the piping system may have nonlinear dynamic characteristics during a seismic event due to gaps in the support structures and local deformation of the thin walled piping. For this reason it is necessary to develop an equivalent linear modelling method for these structures in order to apply the response spectrum method which is generally adopted as the seismic analysis method at present to these system.

Vibration tests were performed using full scale models for pipe supporting structures of the primary heat transport system piping and a 1/4 scale model piping system for the hot leg piping system of the prototype LMFBR "MONJU". The purpose of these tests was to investigate the effect of nonlinear characteristics of a pipe supporting structure with gaps on pipe vibration behavior experimentally and analytically.