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A THREE-DIMENSIONAL METHOD FOR INTEGRATED TRANSIENT ANALYSIS
OF REACTOR-PIPING SYSTEMS

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C. Y. Wang
Structural Mechanics Section
Engineering Mechanics Program
Reactor Analysis and Safety Division
Argonne National Laboratory
9700 South Cass Avenue
Argonne, Illinois 60439, U.S.A.

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A three-dimensional method for integrated hydrodynamic, structural, and thermal analyses of reactor-piping systems is presented. The hydrodynamics are analyzed in a reference frame fixed to the piping and are treated with a two-dimensional Eulerian finite-difference technique. The structural responses are calculated with a three-dimensional co-rotational finite-element methodology. Interaction between fluid and structure is accounted for by iteratively enforcing the interface boundary conditions.

A thermal transient capability has been developed. A system energy equation is used to compute the coolant temperatures due to convection. A radial heat-conduction equation is employed to establish the temperature profile throughout the pipe-wall thickness. The constitutive equation used for the thermal-mechanical stress calculation is suited for a large number of materials under various loading conditions, such as those having thermal, plastic, and viscous effects. The flow surface, which defines the purely elastic regime, can be arbitrarily small; an associated flow rule is utilized for regimes of material plasticity.

Three sample problems are presented to illustrate this method. The first one calculates the piping response under the seismic excitation. The second one validates the heat-conduction model. The third problem deals with a coupled hydrodynamic-structural-thermal analysis of a piping system. Results are discussed in detail.

1. Introduction

Safety assessment of the piping systems of nuclear reactors requires that the systems remain intact in the event of postulated accidents. To ensure the integrity of the piping system in such an event, the safety analysis should consider (a) hoop stresses arising directly from the pressure pulse inside the piping components generated by a core disruptive accident or a sodium water reaction, (b) flexural stresses, including bending in both planes, arising from the globe motion of the pipes or the supports during seismic events, (c) thermal stresses generated by suddenly entering of hot coolant into the piping system, and (d) existing stresses in the pipes due to normal-operation pressurization, thermal effects, and creep phenomena.

The structural response of a piping system to the applied and subsequently propagating pressure pulses can be visualized to consist of two distinct modes: the breathing mode of the structural components as a direct result of the internal pressure, and the flexural mode of the system as it moves in the three-dimensional space. The two modes can be adequately represented by a biaxial state of stress. Such stress is usually generated from the combination of: (1) an axial stress resulting from the axial extension and contraction of the piping components, the flexural bending of the pipes and components in two orthogonal planes, and the axisymmetric bending of the pipe walls; (2) a hoop stress resulting from the pipe-wall radial expansion or contraction; and (3) a shear stress resulting primarily from the torsion of the pipes and components.

To perform the structural analysis of piping systems generated by various loadings mentioned early, a piping analysis program was initiated in the Reactor Analysis and Safety Division of Argonne National Laboratory. The ICEPEL code [1] was developed to perform a coupled hydrodynamic-structural

analysis of piping systems in two dimensions. In this code, the Implicit Continuous-Fluid Eulerian finite difference hydrodynamics [2] is coupled to a nonlinear elastic-plastic explicit finite-element model of the piping system using thin axisymmetric shell element and co-rotational coordinate formulation [3].

Although the ICEPEL code can treat complex wave propagation in the piping system in two dimensions, it considers only the breathing mode of the pipes and components by ignoring the globe motion of the piping system. Furthermore, the code requires the centerline of all pipes and components to be fixed in space and has no thermal-stress calculational capabilities.

In order to extend the ICEPEL code to perform an integral analysis of the piping system, the structural capability has been improved significantly. A three-dimensional pipe element with eight degrees of freedom at each node was developed, using a co-rotational coordinate formulation [4,5]. Six degrees of freedom, three displacements and three rotations, describe the flexural motion in three-dimensional space. The remaining two degrees of freedom describe the axisymmetric breathing mode in terms of hoop displacement and wall bending rotation. In addition, a spring element is used to model piping supports, such as snubbers and hangers. Thus, the structural analysis considers hoop, flexural, axial, and torsion modes of the piping systems.

In this paper, the coupling of the three-dimensional structural analysis to the two-dimensional hydrodynamics are briefly addressed. Emphasis is placed on the description of the thermal-transient capability developed recently. Sample problems are given concerning with application of the resulting technique to the piping-response analyses generated by various accidental loads.

The computer code, with the capability for treating the fluid-structural-thermal interaction is referred as SHAPS (Structural Hydrodynamic Aalysis of Piping System).

2. Fluid-Structural Coupling

2.1. Inertia of the Fluid

Since the fluid is carried along with the moving pipe, the mass of the fluid should be combined with the mass of the structure, but only in some of the eight degrees of freedom. Figure 1 shows a generic pipe element in a general orientation with respect to the global coordinate system x, y, z . The translational mass of the fluid is to be combined with the translational mass of the structure in the local $\hat{y}$ and $\hat{z}$ directions. However, only the pipe mass is to be considered in the local $\hat{x}$ direction. This mass lumping scheme indicates that the fluid is free to slide along the pipe axis ($\hat{x}$ -direction), but in the transverse directions the fluid and the pipe are moved together. Similar mass coupling is to be done for the rotational mass about the local $\hat{y}$ and $\hat{z}$ axes, indicating that the pipe can twist independently from the fluid.

Because the translational equation of motion are written in the global-coordinate system, a transformation of the translational mass from the local - to the global-coordinate system must be performed. Such a transformation should ensure that the total translational mass is conserved and that the nodal kinetic energy is independent of the coordinate systems.

2.2. Hydrodynamic Force at Kinks

One hydrodynamic force contribution to the structure of a pipe system arises from elbows. An elbow is represented by a series of straight pipe segments; at each junction, the model includes "kinks" when force transfer occurs between the fluid and structure. At such kinks, two hydrodynamic

forces result: a force due to the hydrodynamic pressure within the two elements connected through the node, and a force due to the change in the direction of the fluid element.

Figure 2 shows a typical case in which a kink in the piping system is developed between elements (k-1) and (k), which are connected through node N. Consider an infinitesimal region around N to represent a control volume that moves with node N in the global reference XYZ.

Because the control volume is very small and its mass can be considered to be zero, the body, inertia, Coriolis, and centrifugal-force terms, as well as the rate of change of linear momentum inside the control volume, can be neglected. Hence, the momentum equation for the noninertial control volume can be shown to reduce to,

$$\vec{F}_S = \text{momentum rate leaving control volume} - \text{momentum rate entering control volume.} \quad (1)$$

Equation (1) is to be applied in the three global coordinates X, Y, and Z.

The left-hand side of Eq. (1) consists of two portions: the force from the pipe kink to the control volume (the reaction of the hydrodynamic force to the structure) and the force due to hydrodynamic pressures within elements (k-1) and k.

Let $F(i)$ be the global i -component of the hydrodynamic force on the node due to the kink, it can be shown that

$$F(i) = A \left[e_1^{k-1}(i) - e_1^k(i) \right] \left[P_{J+1/2} + \rho V_{J+1/2} |V_{J+1/2}| \right], \quad (2)$$

where $i = 1, 2, \text{ and } 3$ for the global X, Y, and Z directions.

2.3. Effect of Pipe Motion on Hydrodynamics

As the pipe system moves in a three-dimensional space, it carries along with it the flowing fluid. The hydrodynamic finite-difference mesh is also assumed to be carried along with the pipe system without changing its shape or its size. Nevertheless, the fluid remains free to slide along the pipe wall and the pipe remains free to twist independently of the fluid.

To account the global motion of the piping system, let us consider a control volume V bounded by a surface S and moving with an arbitrary velocity $\vec{V}^G$. The conservation of mass and momentum of the fluid flowing through V are:

$$\frac{\partial \rho}{\partial t} + (\rho \vec{v}_i^D)_{,i} = 0 \quad \text{and} \quad \frac{\partial \rho \vec{v}_i}{\partial t} + (\rho \vec{v}_i \vec{v}_j^D)_{,j} + P_{,i} = 0, \quad (3)$$

where

$$\vec{v}^D = \vec{v} - \vec{v}^G.$$

Equation (3) is written in the standard index notations. This equation indicates that only the convective terms are modified to accommodate the motion of the control volume V .

2.4. Pipe Breathing Mode

To account for the pipe-wall hoop motion in the hydrodynamics, we consider a free-slip fluid interface motion with respect to the pipe wall. The fluid interface motion is treated as equivalent to that of a free surface with an applied pressure distribution. Given the pipe-wall shell motion with respect to the pipe axis, as obtained from the structural computations, one can produce that same motion of the fluid interface with some unique pressure distribution applied to the interface. Since the viscosity of the fluid (like water or sodium) is very small, the sliding boundary condition is used at the

interface which requires that the fluid be allowed to slide along the interface, but in the normal direction, the fluid and the structure are forced to move together.

A relaxation equation is used to adjust iteratively the pressure in the fluid-structure-interface zone so that the sliding boundary condition is rigorously satisfied [1,6].

3. Thermal-Transient Analysis

3.1. Energy Equation

One capability has been introduced in the SHAPS code pertains to the treatment of the thermal shock that could be experienced by the primary piping system due to either cooling fluctuations in the core or the existing thermal gradient during the normal operating condition. Here an energy equation is developed that provides temperature distribution in the piping system. In the differential form, the energy equation derived from the piping configuration and the corresponding control volume shown in Fig. 3 is

$$\begin{aligned} \bar{\rho} c \frac{\partial T}{\partial t} = k \frac{\partial^2 T}{\partial z^2} - \bar{\rho} \bar{v} c \frac{\partial T}{\partial z} - [\bar{p} + 1/2 \bar{\rho} \bar{v}^2 - (\lambda + 2\mu) \frac{\partial \bar{v}}{\partial z}] \frac{\partial \bar{v}}{\partial z} \\ + \frac{2}{R} [h_i (T_1^W - T) - p_B u^W], \end{aligned} \quad (4)$$

where $\bar{\rho}$ is the average density at a cross section perpendicular to the pipe line; c the specific heat constant; T the temperature; k the heat-conduction coefficient of the fluid; $\bar{v}$ the average axial fluid velocity; $\bar{p}$ the average pressure; λ and μ the viscosity coefficients; R the pipe inner radius; h_i the coefficient of heat transfer at the inside surface of the wall; T_1^W the wall

temperature at the inner surface; P_B the pressure at the pipe boundary; u^W is the wall radial velocity.

The energy equation can be further written into the finite-difference form (see Fig. 3) which yields

$$\begin{aligned}
 c\bar{\rho}_{i,j} \frac{T_j^{n+1}}{\delta t} = & c\bar{\rho}_{i,j} \frac{T_j^n}{\delta t} + \frac{k}{\delta z^2} (T_{j+1} + T_{j-1} - 2T_j) \\
 & + c(\bar{\rho}\bar{v})_{i,j} \frac{T_{j-1} - T_{j+1}}{2\delta z} + (\bar{p} + 1/2 \bar{\rho}\bar{v}^2)_{i,j} \left(\frac{\bar{v}_{i,j-1/2} - \bar{v}_{i,j+1/2}}{\delta z} \right) \\
 & + (\lambda + 2\mu) (\bar{v}_{i,j-1/2} - \bar{v}_{i,j+1/2})/\delta z + \frac{2}{R_i} [h_i (T_1^W - T_j) \\
 & - p_j^B u_j^W] .
 \end{aligned} \tag{5}$$

3.2. Heat-Conduction Equation

The heat conduction model of the SHAPS code utilizes a one-dimensional radial heat conduction equation to determine the temperature profile through the wall. This equation is

$$\rho_w c_w \frac{\partial T^W}{\partial t} = \frac{1}{r} \frac{\partial}{\partial r} (k_w r \frac{\partial T^W}{\partial r}) , \tag{6}$$

where ρ_w is the density of the pipe wall; c_w is the heat capability of the wall material; k_w is the heat-conduction coefficient of the solid wall; T^W is the wall temperature; r is the radial position of the temperature zone. Note that for isotropic homogeneous materials, the thermal conductivity k_w can be factored out of the right-hand side of Eq. (6).

To obtain the temperature distribution throughout the pipe wall, Eq. (6) is written implicitly into finite-difference form with respect to the control

volume shown in Fig. 4. The resulting finite-difference heat conduction equation is

$$\frac{T_j^{n+1} - T_j^n}{\delta t} = \frac{\alpha}{r_j \delta r_w^2} \left[T_{j+1}^{n+1} \left(r_j + \frac{\delta r_w}{2} \right) + T_{j-1}^{n+1} \left(r_j - \frac{\delta r_w}{2} \right) - 2r_j T_j^{n+1} \right], \quad (7)$$

where

$$\alpha = \frac{k_w}{\rho_w c_w}.$$

Note that the finite-difference heat conduction equation applies only to the interior temperature nodes. In other words, for the finite difference temperature mesh shown in Fig. 4, Eq. (7) valids for $j = 2, 3, \dots, J-1$.

On applying the convective boundary conditions, Eq. (7) can be written, after some rearranging, in the matrix form

$$[c] \begin{Bmatrix} T_1^{n+1} \\ T_2^{n+1} \\ \vdots \\ \vdots \\ \vdots \\ T_{j-1}^{n+1} \\ T_j^{n+1} \end{Bmatrix} = \begin{Bmatrix} T_1^n + ab_1 T_f \\ T_2^n \\ \vdots \\ \vdots \\ \vdots \\ T_{j-1}^n \\ T_j^n + ab_0 T_\infty \end{Bmatrix}, \quad (8)$$

where

$$a = 2\alpha\delta t / \delta r_w^2,$$

b_1 and b_0 are the heat-transfer coefficients at the inner and outer surfaces, respectively; $[c]$ is a band matrix with band width equal to three. Alternatively, Eq. (8) is a set of linear algebraic equations. Each one consists of three unknowns and is conventionally called 3-point equation.

Recently, LINPACK [7], a new collection of machine-independent fully portable and optimized FORTRAN programs for effectively solving various types of linear systems is being implemented in the Applied Mathematics Division of Argonne National Laboratory. Two subroutines of such a package, DQRDC and DQRSL, are used to solve the linear algebraic system of the heat-conduction equations. The SHAPS code uses QR decomposition of DQRDC to change the band matrix to an upper triangular matrix and uses DQRSL to obtain the temperatures at pipe-wall nodes at every iteration.

3.3. Calculation of Thermal-Mechanical Stress

The mathematical model for the calculation of the thermal-mechanical stress is applicable to the behavior of a large number of materials under a variety of loading conditions, such as those having thermal, plastic, and viscous effects. This model is termed as "thermo-viscoplastic" even though elastic behavior is also included.

For many materials the simplifying assumptions of isotropy and a Von Mises flow criterion are justified by experimental data. Thus, for the problem involving calculations of thermal-mechanical stress the basis of the approach is to solve the flow condition [8]

$$\phi = \bar{\sigma} - H(\bar{\epsilon}^i, \dot{\bar{\epsilon}}^i, T) = 0, \quad (9)$$

in which $\bar{\sigma}$ and H denote the effective stress and flow stress, respectively; $\bar{\epsilon}^i$ and $\dot{\bar{\epsilon}}^i$ are the effective inelastic strain and its rate; T is the

temperature. Before solving Eq. (9) it is necessary to compute the stress tensor $\underline{g}$. If $\underline{c}(T)$ denote the elasticity tensor, the stress is given by

$$\underline{g} = \underline{c} \underline{e}^e = \underline{c} (\underline{e} - \underline{e}^t - \underline{e}^i), \quad (10)$$

where $\underline{e}^e$, $\underline{e}$, $\underline{e}^t$, and $\underline{e}^i$ denote the elastic, total, thermal, and inelastic strain tensors, respectively. The thermal-strain tensor can be further computed from

$$\underline{e}^t = \underline{\alpha}^t (T - T_0), \quad (11)$$

where $\underline{\alpha}^t$ represents the thermal-expansion tensor, T_0 is the reference temperature. The effective stress is then given by

$$\bar{\sigma} = [3/2 \operatorname{tr} (\underline{g}^d \underline{g}^d)]^{1/2}, \quad (12)$$

where a superscript "d" is used to designate the deviator and "tr" denotes the trace. The flow stress is a function of temperature, the effective inelastic strain and its rate where

$$\bar{e}^i = \int_0^s d\bar{e}^i, \quad (13)$$

for the load parameter s , which can be time; and

$$d\bar{e}^i = [2/3 \operatorname{tr} (d\underline{e}^i d\underline{e}^i)]^{1/2}. \quad (14)$$

If an associated flow rule is used, the increment in inelastic strain is given by

$$d\underline{e}^i = \underline{g} d\lambda \quad \text{and} \quad \underline{g} = \frac{\partial \phi}{\partial \underline{\sigma}} = \frac{\partial \bar{\sigma}}{\partial \underline{\sigma}}, \quad (15)$$

where, by definition, λ is a monotonically increasing scalar function of a load parameter, s ; and $\underline{g}$ is the stress gradient of the yield function.

Equation (14) can then be written as [8]

$$d\bar{\epsilon}^i = 3/2 \, d\bar{\epsilon}^i \frac{\underline{g}^d}{\bar{\sigma}}. \quad (16)$$

The governing equations are solved iteratively. A modified Newton-Raphson procedure is used to evaluate stress and strain for given total and thermal strains. Material properties are described by numerical data and linear interpolation in a two-dimensional space provides the required functions. The procedure for one calculational cycle is as follows:

- (a) Determine the stress tensor from constitutive law given in Eq. (10) and the effective stress tensor from Eq. (12).
- (b) Evaluate ϕ_N , the procedure is terminated if material responding elastically, or if the flow criterion is satisfied; that is $|\phi_N| < H_0 \epsilon$, where ϵ is a small positive number.
- (c) Solve effective-inelastic-strain rate $\dot{\bar{\epsilon}}_N^i$ for a time step Δt .
- (d) Apply the associated flow rate in the form of Eq. (16) to obtain the inelastic strain-tensor increment.
- (e) Determine the update inelastic strain tensor $\bar{\epsilon}^i$; and return to step (a).

3.4. Material Nonlinearities

Material with nonlinear stress-strain relationship can be treated in the SHAPS code. If problem involves the thermal-stress calculation generated by the temperature gradient and pipe end constraints, several temperature-dependent stress-strain curves are required as input to the computer program. Each curve is further approximated by a same number of piecewise linear

segments in the stress-strain plane. For problem doesn't involve the thermal loading only one temperature related stress-strain curve representing the material constitutive law is used. The analysis thus automatically ignores the thermal-stress calculations and considers the stress generated by the mechanical loading only.

4. Sample Problems

4.1. Piping Response to Seismic Excitation

This study deals with the SHAPS analysis of piping system response generated by the earthquake-induced support motions in terms of acceleration-time history. The results obtained with SHAPS on a simple piping system are compared with those of ANSYS (Analysis System Code) [9], a general finite element structural code.

A simple supported straight pipe 1219.2 cm long, 45.72 cm is OD, and 1.43-cm wall thickness, as shown in Fig. 5, is subjected to the modified El Centro acceleration-time history as input motions at the two support ends (nodes 1 and 11). Figure 6 shows the input acceleration history and the associated displacement history. The acceleration history has a maximum acceleration value of 0.684 g at 3.71 s. For comparison purposes, only the first 6 s of the history are used.

The responses of the system were calculated for two cases, one with fluid (fluid-filled pipe) and one without fluid. In the fluid case, the SHAPS code performed a coupled hydrodynamic-structural analysis, whereas the ANSYS code performed only a structural analysis in which the fluid mass was considered as additional mass on the pipe wall. In the no-fluid case, both codes performed structural analyses. The time steps used in both codes were 0.0005 s for the

fluid case and 0.0002 s for the no-fluid case; no damping was included in either case.

In the SHAPS code, each pipe node has eight degrees of freedom. Six of them are beam modes (three translations and three rotations), and the remaining two are shell modes (shell bending and membrane modes). In the ANSYS code, each pipe node has only six degrees of freedom, i.e., six beam modes. For consistency, the two shell modes of the SHAPS element were fixed in this study; the elimination of the shell modes improved the numerical stability of the SHAPS calculation. Because of space limitations, only the results of the no-fluid case are presented. The displacement histories of nodes 1 and 6 (the latter at midspan) are shown in Fig. 7. The smooth curve in Fig. 7 is the displacement plot of node 1; its shape is identical to the one shown in Fig. 6. The oscillating curve in Fig. 7 is the displacement history of node 6, and shows that node 6 vibrates about node 1 during seismic disturbances. The displacement histories of the intermediate nodes (not shown in the figure) have curves similar to that of node 6, but with smaller vibration amplitudes. The difference between the smooth curve and the oscillating curve is the relative displacement of nodes 1 and 6. Since the stresses of the system are determined from the relative displacements between the nodes, the relative displacement between nodes 6 and 1 is replotted in Fig. 8 on a different scale. This figure shows that the agreement between the two codes is fairly good. However, some differences do exist. These differences can be attributed to the different methods used in the calculations of mass and mass moment of inertia in the codes. For example, SHAPS uses the lumped-mass approach, whereas ANSYS uses the consistent-mass approach.

The fundamental natural frequency of the system can be obtained from Fig. 8 by calculating the number of oscillations. SHAPS gives a value of 8.50 Hz, whereas ANSYS gives a value of 8.042 Hz. The theoretical value is 8.123 Hz. Thus, the SHAPS code provides a slightly higher fundamental frequency, whereas ANSYS provides a slightly lower fundamental frequency.

4.2. Validation of the Heat-Conduction Model

To ensure the accuracy of the temperature calculations, the heat-conduction model was tested against three problems for which the steady-state analytical solutions could be readily obtained by direct integration of the heat-conduction equation. The problems considered were (1) a rectangular slab, (2) a thin-wall pipe with a thickness-to-radius ratio (t/R) equal to 0.0433, and (3) a thick-wall pipe with the t/R ratio equal to 0.433. In all three problems, the inner surface was assumed to be kept at 100°F, while the outer surface was maintained at 0°F. The thickness of the wall in these three problems was assumed to be equal to 60.96 cm (2 ft). Temperature profiles were calculated by the SHAPS code at eight equally spaced temperature nodes. All computations were carried out to 150 and 300 hr, with a time step equal to 0.5 hr.

The steady-state analytical solutions and SHAPS' transient solutions of temperatures at eight different nodes are given in Fig. 9. It can be seen that the solutions for the thin-wall pipe compare very well with the slab solutions. Such agreement means that when the pipe wall is very thin (small t/R), the effect of curvature on the temperature distribution becomes insignificant. However, in the case of the thick-wall pipe (large t/R), the temperature deviated significantly from those of the thin-wall pipe and slab, indicating a marked influence of curvature on temperature distribution. As expected, increasingly good agreement was found between the results of the

transient calculations and the respective steady-state analytical solutions as the heat period progressed toward the later stages of the 300 hr period.

4.3. Coupled Hydrodynamic-Structural-Thermal Analysis

To illustrate the coupled hydrodynamic-structural-thermal analysis, the SHAPS code was applied to the wave propagation in a water-filled elastic-plastic pipe. Figure 10 shows the mathematical model used in the analysis of a pipe, 203.2 cm long and 5.08 cm in inner radius, represented by 40 axial zones. A nonreflecting fluid boundary is assumed at the end junction of the pipe. A triangular pulse of 6-MPa amplitude and 0.5-ms duration, together with a constant coolant temperature of 300°C, are applied to the front junction of the pipe. The pipe wall thickness is 0.165 cm, with 10 pieces of temperature-dependent stress-strain curves being used to simulate the material properties of the vessel wall. Each stress-strain curve is further represented by four straight-line segments to model the material nonlinearities. The ambient temperature of the coolant in the pipe is 30°C.

The pressure-time histories at selected locations (see Fig. 10) of the pipe are shown in Figs. 11-12. As can be seen, the plastic response of the pipe wall attenuates the pressure peak and disperses the pressure pulse. We also find that only one of the pressure peaks being propagated down the pipe has a magnitude slightly higher than the yield pressure of the pipe. Figure 13 gives the radial displacement of the shell node A (see Fig. 10 for its location), near the source region. The plastic deformation is apparent from this figure. The coolant-temperature history at location 3 is depicted in Fig. 14. It indicates that the coolant temperature at the end of this run is still at 30°C. This is so because the propagation of the thermal wave is a slow process. Consequently, the stress and strain of the pipe wall at early stages are dominated by the mechanical loading. In other words, for computa-

tional time in the millisecond range, the effect of the thermal loading on the response of the structure is very small.

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Figure Captions

- Fig. 1. Local Coordinates for a Generic Pipe Element.
- Fig. 2. Typical Pipe Kink.
- Fig. 3. Mesh Configuration and Control-Volume Used for Derivation of the System Energy Equation.
- Fig. 4. Control Volume Used to Derive the Finite-Difference Heat Conduction Equation.
- Fig. 5. Straight-Pipe Segment Subjected to Support Motions.
- Fig. 6. Acceleration and Displacement Time Histories of Support Motions.
- Fig. 7. Displacement Plot of Node 1 (Smooth Curve) and Node 6 (Oscillating Curve).
- Fig. 8. Relative Displacement Histories Between Nodes 6 and 1 ($U_6 - U_1$).
- Fig. 9. Temperature Profiles Due to Heat Conduction for Various Cases of the Three Problems Studied.
- Fig. 10. Mathematical Model of an Elastic-Plastic Pipe Used in the Analysis.
- Fig. 11. Pressure-Time Histories at Locations 1 and 2.
- Fig. 12. Pressure-Time Histories at Location 4.
- Fig. 13. Radial Displacement as a Function of Time at Shell Node A.
- Fig. 14. Coolant Temperature as a Function of Time at Location 3.

Fig. 1. Local Coordinates for a Generic Pipe Element.

(a) STRUCTURE ELEMENTS

(b) FLUID CONTROL VOLUME

Fig. 2. Typical Pipe Kink.

(a) MESH CONFIGURATION

(b) CONTROL VOLUME

Fig. 3. Mesh Configuration and Control-Volume Used for Derivation of the System Energy Equation.

Fig. 4. Control Volume Used to Derive the Finite-Difference Heat Conduction Equation.

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Fig. 12. Pressure-Time Histories at Location 4.

Fig. 13. Radial Displacement as a Function of Time at Shell Node A.

Fig. 14. Coolant Temperature as a Function of Time at Location 3.