FLUID-STRUCTURE INTERACTIONS

Jong Chull Jo
Korea Institute of Nuclear Safety, Republic of Korea

Keywords: Fluid-structure interaction(FSI), hydrodynamic mass, hydrodynamic damping, weakly coupling, strongly coupling, one-way separate analysis, two-way coupled analysis, flow-induced vibration(FIV), steam generator(SG), thermal-hydraulic analysis, modal analysis, fluid-elastic instability, finite element method (FEM), lattice Boltzmann method (LBM), three-dimensional analysis

Contents

Summary

Fluid-structure interaction is a multifaceted physics problem occurring in a system where

flow of a fluid causes deformation of a solid structure which, in turn, changes the boundary condition of the fluid problem. For instance, flow of air around an airplane wing causes deformation of the wing which subsequently causes the flow pattern of the surrounding air to change.

The overall aim of this Chapter is to help the readers understand the phenomena and mechanism of fluid-structure interaction and get a fundamental knowledge and information for the analysis of fluid-structure interaction problems occurring in pressure vessels and piping systems.

1. Introduction

1.1. Concept of Fluid-Structure Interaction

A flexible solid structure contacting a flowing fluid is subjected to a pressure which may cause deformation in the structure. As a return, the deformed structure alters the flow field. The altered flowing fluid, in turn, exerts another form of pressure on the structure with repeats of the process. This kind of interaction is called Fluid-Structure Interaction (FSI).

For examples, a fluid flowing either inside or outside of pipes or vessels exerts steady or oscillatory pressure on the wetted surface of pipes or vessels, which may deform or vibrate them. Another one is that the flow of air around an airplane wing causes the wing to deform, and as the wing deforms it causes the air pattern around it to change. In a broad sense, fluid-structure interaction covers such subjects as aero-elasticity, hydro-elasticity, flow-induced vibration, thermal deformation, etc. The pressure vessels and piping systems, which are typical structures involved in the power and process industries, are stationary except for their vibratory motions or thermal deformations unlike in aero-elasticity. In general, the velocities of flowing fluid are much lower than those encountered in aero-elasticity.

The fluid-structure interaction problems in pressure vessels and piping systems involve diverse mechanical failure mechanisms of stationary structures in a flowing fluid, moving structures immerged in a flowing or quiescent fluid, or stationary pipes or vessels containing cold or hot flowing fluid, such as flow-induced vibration, water hammer, thermal deformation and fatigue.

In general, a fluid-structure interaction system is classified as either strongly or weakly coupled.

- ***Weakly coupled fluid-structure system***

Let’s consider the following two typical cases of fluid-structure interaction system. If a structure in the flow field or containing flowing fluid deforms slightly or vibrates with small amplitude, it will affect negligibly the flow field because of the relatively low pressure. Even if significant thermal stresses in the solid may be induced by thermal gradients in the flow field, the flow field may not be greatly affected if the resulting deformation of the solid is too small. These fluid-structure interaction systems are called weakly coupled systems if a thermo-fluid deforms a structure while the deformed structure hardly alters the flow field.

Typical examples of weakly coupled fluid-structure interaction systems encountered in the area of pressure vessels and piping are small thermal deformation or low amplitude-vibration of a pipe conveying a fluid and turbulence induced vibration of shells or tubes. For any of these systems, it is assumed that the force acting on the fluid due to the structural motion can be linearly super-imposed onto the original forcing function in the fluid.

- ***Strongly coupled fluid-structure system***

On the other hand, the fluid-structure systems are called strongly coupled systems if alteration of the flow field due to large deformation or high amplitude-vibration of the structure can not be neglected. In such strongly coupled fluid-structure systems in which large structural deformation or displacement results in a significant alteration of the original flow field, both altered and original flow fields can not be linearly super-imposed upon each other. Two typical examples of the strongly coupled fluid-structure interaction system encountered in the area of pressure vessels and piping are vortex-induced vibration and fluid-elastic instability of heat exchanger tubes.

1.2. Brief History of Fluid-Structure Interaction

This sub-section briefly introduces the history of fluid-structure interaction, which has been previously reported by Au-Yang, M. K. (2001).

In 1828, the concept of hydrodynamic mass (or added mass) was proposed first by Friedrich Bessel who investigated the motion of a pendulum in a fluid. He found out that a pendulum moving in a fluid had longer period than in a vacuum even though the buoyancy effects were taken into account. This finding meant that the surrounding fluid increased the effective mass of the system. Thereafter, in 1843 Stokes performed a study on the uniform acceleration of an infinite cylinder moving in an infinite fluid medium and concluded that the effective mass of the cylinder moving in the fluid increased due to the effect of surrounding fluid by the amount of hydrodynamic mass equal to the mass of fluid it displaced. It was known that this finding produced the concept of fluid-structure interaction.

In the 1960s, some designers of nuclear reactor systems found that the hydrodynamic mass of a structure in a confined fluid medium resulting from the fluid-structure interaction was much larger than that for the structure in an infinite fluid medium which was equal to the mass of fluid displaced by the structure.

The investigation of fluid-structure interaction as in the form known to engineers working in the area of pressure vessels and piping systems is considered to have begun in the 1960s. In 1966, Fritz and Kiss performed a study on the vibration response of a cantilever cylinder surrounded by an annular fluid, which is known to be the pioneering study of fluid-structure interaction for power plants. From the early 1970s to the late 1980s, a lot of investigators studied the dynamics of interaction between fluid and elastic shell systems including pipes, tubes, vessels, and co-axial cylinders. Based on the results of such extensive studies, American Society of Mechanical Engineers (ASME) Boiler Code Sec. III Appendix 1400 "Dynamics of Coupled Fluid-Shells" was published.

Nowadays, various techniques for simulating the strongly coupled fluid-structure systems numerically are under development as the Computational Fluid Dynamic (CFD) analysis technique evolves rapidly.

1.3. Purposes of this Section

The purposes of this section are to help readers understand the phenomena of fluid-structure interaction and to provide them with a fundamental knowledge and information for the analyses of fluid-structure interaction problems occurring in the pressure vessels and piping systems.

To this end, the mathematical formulation of a simple fluid-structure interaction problem is introduced. Then, the essential fundamental knowledge needed for analyzing the general fluid-structure interaction problems, including the concepts of two approaches to the analysis of fluid-structure interaction problem, hydrodynamic mass, fluid coupling, and hydrodynamic damping is described. Finally, in order to help the readers understand readily how to analyze coupled fluid-structure interaction problems, two examples of solutions either by the one-way separate approach or the two-way coupled approach are illustrated.

2. Mathematical Formulation of a Simple Fluid-Structure Interaction Problem

As a simple illustration of a typical fluid-structure interaction problem, let's consider a situation where a flexible cylindrical tube generating heat, of which the thermal deformation is negligible, is subjected to external cross flow of a cold fluid as shown in Figure 1.

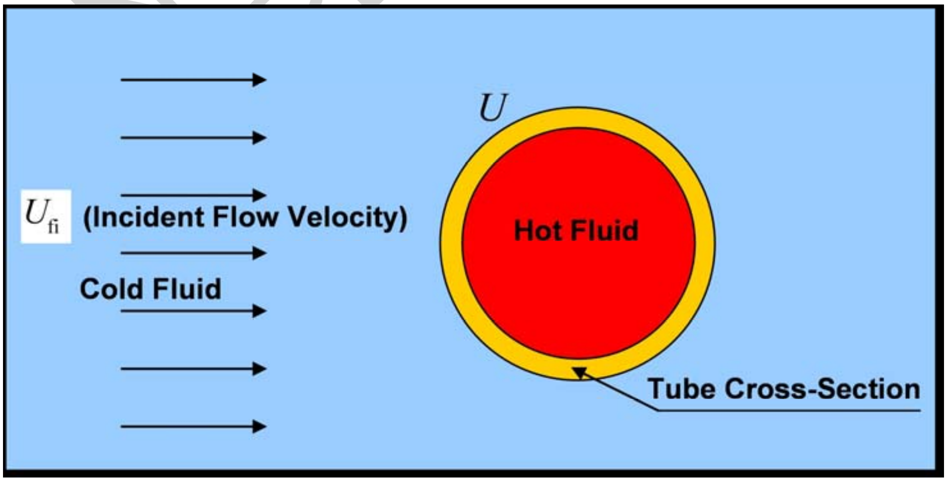

Figure 1(a). A stationary flexible cylindrical tube in an externally cross flowing fluid: Analysis model

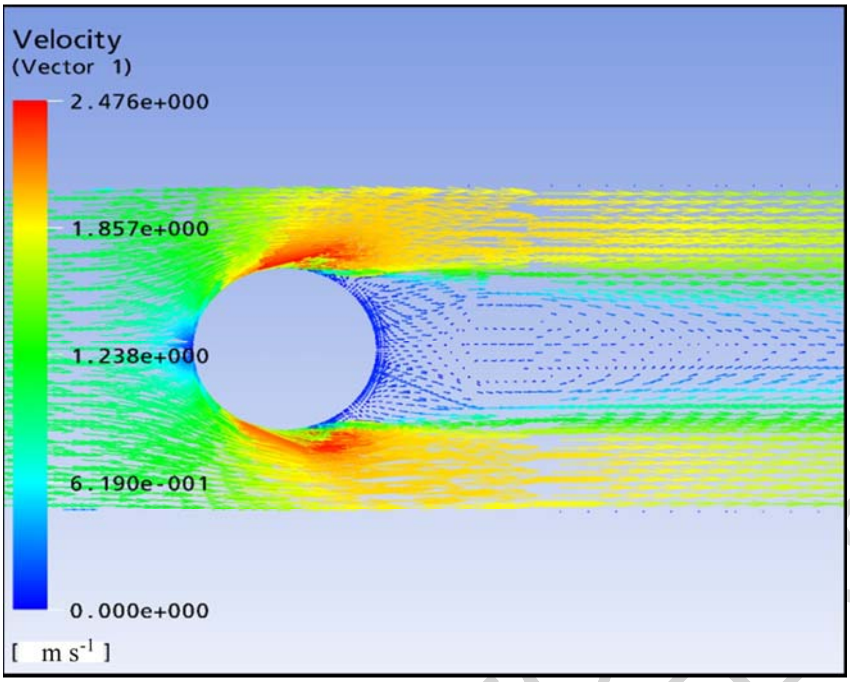

Figure 1(b). A stationary flexible cylindrical tube in an externally cross flowing fluid: Typical instantaneous flow field around the cylinder subjected to cross-flow in a rectangular duct

2.1. Fluid Domains

The instantaneous equation of mass, momentum and energy conservation in a stationary frame governing the flow fields inside or outside the tube whether they are laminar or turbulent can be given as:

Continuity equation

$$\frac{\partial \rho}{\partial t} + \nabla \bullet (\rho \mathbf{U}) = 0 \tag{1}$$

Momentum equations

$$\frac{\partial \rho \mathbf{U}}{\partial t} + \nabla \bullet (\rho \mathbf{U} \otimes \mathbf{U}) = \nabla \bullet (-p\delta + \mu(\nabla \mathbf{U} + (\nabla \mathbf{U})^{\mathrm{T}})) + \mathbf{S}_{\mathrm{M}} \tag{2}$$

Energy equation

$$\frac{\partial \rho h_{\mathrm{tot}}}{\partial t} - \frac{\partial p}{\partial t} + \nabla \bullet (\rho \mathbf{U} h_{\mathrm{tot}}) = \nabla \bullet (\lambda \nabla T) + S_{\mathrm{E}} \tag{3}$$

where

$$h_{\mathrm{tot}} = h + \frac{1}{2}U^2, \; h = h(p, T) \tag{4}$$

where ρ, t, U, p, δ, μ, S, h, λ, and T represent the density, time, velocity, pressure, Kronecker delta function, viscosity, source term , sensible enthalpy, thermal conductivity, and temperature of the fluid, respectively. In addition, the term $\mathbf{S}_{\mathrm{M}}$ denotes the sum of interfacial forces acting on fluid. The operator $\otimes$ is a tensor product and $(\nabla\mathbf{U})^{\mathrm{T}}$ is the transpose of the matrix $\nabla\mathbf{U}$.

For better understanding of above equations, the mathematical notation is described below.
Considering a Cartesian coordinate system (x, y, z) in which $\mathbf{i}$, $\mathbf{j}$ and $\mathbf{k}$ are unit vectors in the three coordinate directions, ∇ is defined as

$$\nabla = \left[\frac{\partial}{\partial x}, \frac{\partial}{\partial y}, \frac{\partial}{\partial z}\right]$$

For a general scalar function $\varphi(x, y, z)$, the gradient of φ is defined as

$$\nabla\varphi = \left[\mathbf{i}\frac{\partial}{\partial x} + \mathbf{j}\frac{\partial}{\partial y} + \mathbf{k}\frac{\partial}{\partial z}\right]\varphi$$

For a vector function $\mathbf{U}(x, y, z) = U_x\mathbf{i} + U_y\mathbf{j} + U_z\mathbf{k}$ where

$$\mathbf{U}(x, y, z) = \begin{bmatrix} U_x \\ U_y \\ U_z \end{bmatrix},$$

The Cartesian components of gradient of $\mathbf{U}$ are expressed as

$$[\nabla\mathbf{U}] = \begin{bmatrix} \frac{\partial U_x}{\partial x} & \frac{\partial U_x}{\partial y} & \frac{\partial U_x}{\partial z} \\ \frac{\partial U_y}{\partial x} & \frac{\partial U_y}{\partial y} & \frac{\partial U_y}{\partial z} \\ \frac{\partial U_z}{\partial x} & \frac{\partial U_z}{\partial y} & \frac{\partial U_z}{\partial z} \end{bmatrix}$$

and the divergence of $\mathbf{U}$ (div $\mathbf{U}$) is defined as

$$\nabla \bullet \mathbf{U} = \frac{\partial U_x}{\partial x} + \frac{\partial U_y}{\partial y} + \frac{\partial U_z}{\partial z}$$

The tensor product of two vectors, $\mathbf{U}$ and $\mathbf{W}$ operator, is defined as

$$\mathbf{U} \otimes \mathbf{W} = \begin{bmatrix} U_x W_x & U_x W_y & U_x W_z \\ U_y W_x & U_y W_y & U_y W_z \\ U_z W_x & U_z W_y & U_z W_z \end{bmatrix}$$

The transpose of a matrix is formed by turning all the rows of a given matrix into columns and vice-versa. Thus, the transpose $\mathbf{U}^{\mathrm{T}}$ of the matrix $\mathbf{U}$ is written as

$$[\nabla \mathbf{U}]^{\mathrm{T}} = \begin{bmatrix} \frac{\partial U_x}{\partial x} & \frac{\partial U_y}{\partial x} & \frac{\partial U_z}{\partial x} \\ \frac{\partial U_x}{\partial y} & \frac{\partial U_y}{\partial y} & \frac{\partial U_z}{\partial y} \\ \frac{\partial U_x}{\partial z} & \frac{\partial U_y}{\partial z} & \frac{\partial U_z}{\partial z} \end{bmatrix}$$

The Kronecker delta function i.e. the identity matrix is defined as

$$\delta = \begin{bmatrix} 1 & 0 & 0 \\ 0 & 1 & 0 \\ 0 & 0 & 1 \end{bmatrix}$$

2.2. Solid Domain

Heat conduction equation

The only mode of heat transfer in the solid domain of tube wall is the heat conduction expressed as

$$\frac{\partial \rho c_{\mathrm{p}} T}{\partial t} = \nabla \bullet (\lambda \nabla T) + S_{\mathrm{E}} \tag{5}$$

where ρ, c_{p}, λ, and S_{E}, respectively, are the density, specific heat capacity, thermal conductivity, and heat generation of the solid.

Equation of structural dynamics

The equation of motion for a point-mass-spring-viscous damping system with external forces is given by

$$M\ddot{x}+C\dot{x}+Kx=F(t) \tag{6}$$

where x, $\dot{x}(=\partial x/\partial t)$ and $\ddot{x}(=\partial^2 x/\partial t^2)$ denote the vibration amplitude of the point, the 1st order-partial derivative and 2nd order-partial derivative with respect to time t, respectively. M, C, K and F represent the mass, damping coefficient, spring (stiffness) constant, and externally applied forces, respectively.

For a damped structure system comprised of continuous masses such as beams, plates, tubes, rods, shells, etc., which are subjected to a set of external forces, the motion of equation can be described by

$$[M]\{\ddot{\mathrm{X}}\}+[C]\{\dot{\mathrm{X}}\}+[K]\{\mathrm{X}\}=\{F\} \tag{7}$$

where $[M]$, $[C]$, and $[K]$ are the matrices of mass, damping, and stiffness, $\{\mathrm{X}\}$ is the vector of the generalized coordinates, and $\{F\}$ is the vector of the external forces.

2.3. Coupling of the Equations for Fluid and Structure

The equations for fluid and structure can be coupled by the following fluid-structure interfacial boundary conditions.

$$\mathbf{U}\bullet\mathbf{n}=\dot{\mathbf{X}}\bullet\mathbf{n} \tag{8}$$

$$\mathbf{U}\times\mathbf{n}=\dot{\mathbf{X}}\times\mathbf{n} \tag{9}$$

$$\lambda_{\mathrm{f}}(\nabla T\bullet\mathbf{n})_{\mathrm{f}}=\lambda_{\mathrm{s}}(\nabla T\bullet\mathbf{n})_{\mathrm{s}} \tag{10}$$

where $\mathbf{n}$ is the unit normal to the surface. The vector operator $\times$ is defined as

$$\mathbf{U}\times\mathbf{W}=\begin{vmatrix}\mathbf{i} & \mathbf{j} & \mathbf{k}\\ U_x & U_y & U_z\\ W_x & W_y & W_z\end{vmatrix}=(U_yW_z-U_zW_y)\,\mathbf{i}+(U_xW_z-U_zW_x)\,\mathbf{j}+(U_xW_y-U_yW_x)\,\mathbf{k}$$

The 1st interfacial boundary condition given by Eq. (8) is that the fluid velocity normal to the surface of structure must be equal to the normal component of the velocity of structure. This velocity or acceleration component normal to the interface is directly related to the fluid force exerting on the surface of structure and the distortion of the flow field.

The next one expressed as Eq. (9) means the no-slip condition at the interface that both fluid and solid velocity components parallel to the surface of structure must be matched each other.

The last one given by Eq. (10) is for the requirement of heat balance at the interface that both heat fluxes at the interfacial boundary obtained from the equations for fluid and solid domains must be equal.

3. Analysis Methods for Fluid-Structure Interaction Problems

As was discussed in the previous section, even the governing equations with the interfacial boundary conditions for a simple fluid-structure interaction problem are too complex to address with analytical approach. Thus, in most cases, fluid and structure systems have been solved separately up to the present. However, there are special problems which are too complex and strong in coupling to solve the two systems separately. For those cases, numerical methods such as the computational fluid dynamic (CFD) and computational structural dynamic or mechanics (CSD or CSM) techniques are essential in predicting the behavior of such coupled systems. Recently several numerical approaches to solve such highly complex non-linear problems are being developed.

3.1. One-way Separate Analysis Method (Hydrodynamic Mass and Damping Method)

In a weakly coupled fluid-structure system, the fluid flow field altered by the motion of structure is negligible so that we can assume the motion of structure does not affect the fluid flow field. The total pressure (or force) acting on the fluid is obtained by linearly superimposing the induced pressure (or force) acting on the fluid due to the motion of structure onto the original pressure (or force) in the fluid.

Another example of the weakly coupled fluid-structure is the case where significant thermal stresses in the pipe conveying a hot fluid are induced by thermal gradients in the flow field while the flow field is not greatly affected since the resulting deformation of the pipe structure is so small. As a result, CFD and CSD (or CSM) solutions for both temperature distributions and thermal deformation, respectively, are allowed to be calculated independently by transferring data from the CFD to CSD solutions in one-way as is shown in Figure 2.

As mentioned previously, the complete sets of equations of fluid dynamics, heat transfer, and structural dynamics with interfacial boundary conditions for mathematically formulating various practical coupled fluid-structure problems are generally too complicate to solve either by any analytical or numerical methods.

For this reason, during the past 30 years and over most coupled fluid-structural dynamic problems have been solved by simplifying them as weakly coupled problems provided that a hydrodynamic mass and a hydrodynamic damping are added to the original mass and damping of each structure under consideration, respectively, to take into account the effect of fluid-structural interaction appropriately.

These hydrodynamic mass and damping terms to be used for the one-way analyses of the coupled fluid-structural dynamic problems can be calculated or measured separately. This will be discussed further in the following chapters.

Till now, this approach has been applied to most practical problems encountered in the area of pressure vessels and piping systems including especially shell-and tube heat exchangers at power and process plants. It only employs computer programs which can be developed without difficulty and/or experimental or test data where available so that it does not require it to take into account the effect of fluid-structure interaction directly.

To solve the weakly coupled fluid-structural dynamics problems two alternative formulations of the hydrodynamic mass and damping method are available: the generalized hydrodynamic mass and the full hydrodynamic mass matrix methods. The first formulation is for estimating the coupled fluid-shell frequencies when the uncoupled, in-air natural frequencies of the individual cylindrical shells are known from measurement, separate finite-element analysis or numerical solution of the characteristic equation. Often, the full hydrodynamic mass matrix needs to be formulated for subsequent dynamic analysis by the finite element method. This hydrodynamic mass is derived from a pressure.

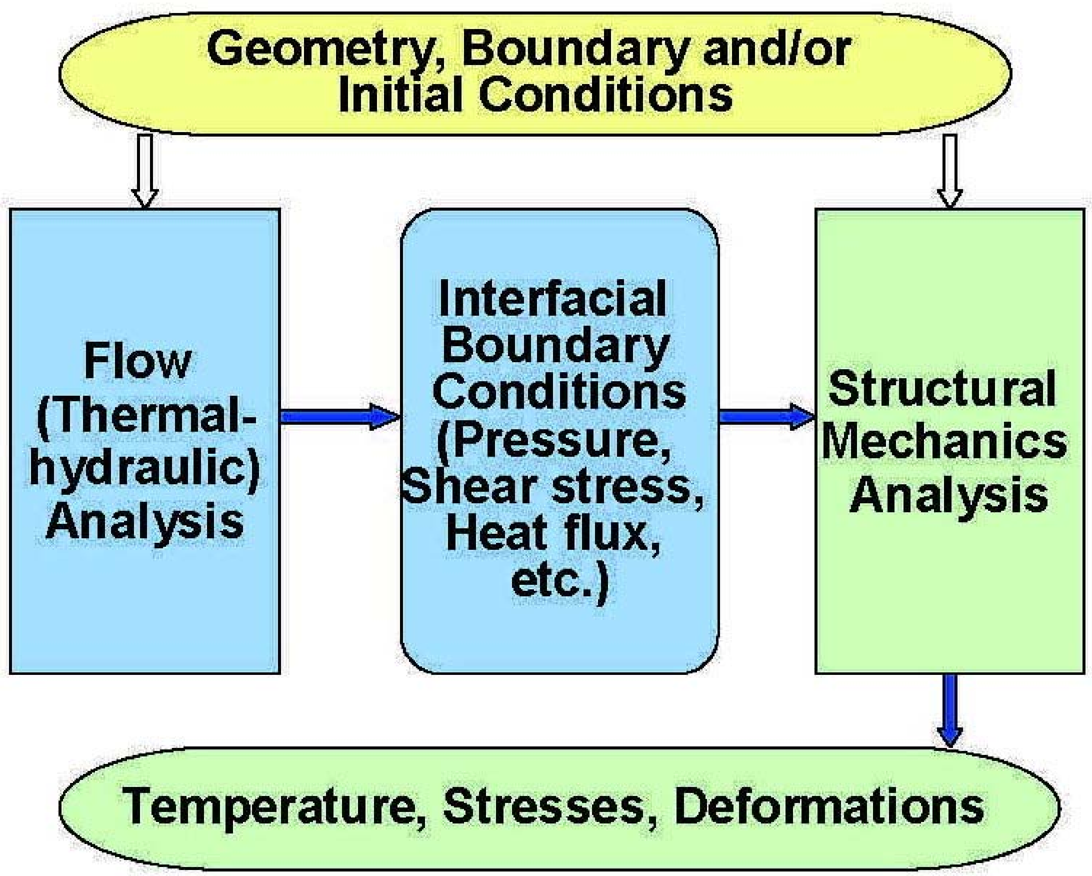

Figure 2. One-way separate analysis method

-
-
-

Bibliography

Au-Yang M.K. (2001). *Flow-Induced Vibration of Power and Process Plant Components, A Practical Workbook.* 478 pp. New York, USA, ASME Press. [This book provides the practicing engineers with a consolidated source to obtain information on the most common flow-induced vibration problems in power and process plant components.]

Blevins R.D. (1994). *Flow-Induced Vibration.* 2nd ed., 477 pp. Malabar, Florida, USA, Krieger Publishing Company. [This book provides with analytical tools for the analysis of the vibrations of structures exposed to fluid flow.]

Brennen C.E. (1982). *A Review of Added Mass and Fluid Inertial Forces.* CR 82.010, 50 pp. Naval Civil Engineering Laboratory. Port Hueneme, California, USA. [This report provides knowledge and information concerning the evaluation of the forces imposed on a body in a fluid due to acceleration of either the body or the fluid.]

Chen S. S., Wambsganss, M. W., Jendrzejczyk, J. A. (1976). Added mass and damping of a vibrating rod in confined viscous fluids. *ASME, National Conference of Applied Mechanics*, University of Utah, Salt Lake City, Utah, 5 p. [This paper addresses analytical and experimental studies on a cylindrical rod vibrating in a viscous fluid enclosed by a rigid, concentric cylindrical shell.]

Conners H.J. (1981). Flow-induced vibration and wear of steam generator tubes. *Nuclear Technology* **55,** 311-331. [This paper introduces a fundamental concept of flow-induced vibration and wear of steam generator tubes.]

Jo J.C., and Shin W.K. (1999). Fluidelastic instability analysis of operating nuclear steam generator U-tubes. *Nuclear Engineering & Design* **193**, Issue No.1-2, 55-71. [This paper presents the one-way analysis approach for predicting the fluidelastic instability of U-type steam generator tubes.]

Kwon Y.W. and Jo J.C. (2008). 3-D Modeling of Fluid-Structure Interaction with External Flow Using Coupled LBM and FEM. *ASME Journal of Pressure Vessel Technology* **130**, 021301, 1-8. [This paper presents the two-way analysis approach for simulating the fluid-structure interaction using the coupled LBM and FEM.]

McAdams W. H., Woods W. K., and Bryan R. L. (1942). Vaporization inside Horizontal Tubes- II. Benzene-oil Mixtures. *Trans. ASME* **64**, 193-200 [The paper presents the experimental procedure and the results of experiments on two-phase viscosity and density using benzene-oil mixtures.]

Paidoussis M.P. (1998). *Fluid-Structure Interactions, Slender Structures and Axial Flow.* Vol. 1, 572 pp. San Diego, California, USA, Academic Press. [This book deals with the dynamics of slender, mainly cylindrical or quasi-cylindrical, bodies in contact with axial flow, either containing the flow or being immersed in it, or both.]

Pettigrew M. J. and Taylor C. E. (2003). Vibration analysis of shell-and-tube heat exchangers: an overview-Part 1: flow, damping, fluidelastic instability. *Journal of Fluids and Structures* **18**, 469-483. [This paper presents an overview of the damping and fluidelastic instability prediction for heat exchanger

tubes.]

Biographical Sketch

Dr. Jong Chull Jo is a mechanical engineer who graduated from Hanyang University, Seoul, Korea in 1979, and obtained his M.S. and Ph. D. degrees from the same university in 1981 and 1985, respectively. Currently, he is a technical consultant of the OECD Nuclear Energy Agency in the area of nuclear safety and regulation and concurrently is affiliated as a principal researcher with the Korea Institute of Nuclear Safety (KINS), Daejon, Korea for which he has been working since 1986. Before that, he worked as a full time lecturer and an assistant professor of mechanical engineering department at Induk College, Seoul for 5 years.

His job for over the past two decades relates to the safety regulation of nuclear reactors including inspection, licensing review, establishment of regulatory requirements, and resolution of reactor safety issues. He was Head of Safety Issue Research Department at KINS and the manager of the national project for development of a new reactor licensing framework. He served as a member of the national R&D projects evaluation committee.

He has been serving as Chair of the ASME Pressure Vessels and Piping (PVP) Division FSI Technical Committee since 2008 and also as an associate editor of the ASME Journal of Pressure Vessel technology since 2009. In addition, he has been serving as Chair of the FSI Division of the Korean Society of PVP since 2004. He has been invited as a peer reviewer of contributing papers for several archival journals. He has been a member of the Korean Society of Mechanical Engineers since 1981, and a member of the Korean Nuclear Society since 1986.

Dr. Jo has published over 50 technical journal papers and over 100 conference proceeding papers. He received 'Korean Prime Ministerial Citation' for recognizing contribution to the promotion of science and technology in 1994 and 'Korean Presidential Citation' for contribution to development of science and technology in 2004.