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SCALE MODELING FLOW-INDUCED VIBRATIONS OF REACTOR COMPONENTS

MASTER

by

T. M. Mulcahy

ARGONNE NATIONAL LABORATORY

BASE TECHNOLOGY

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T. M. Mulcahy

Components Technology Division

June 1982

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NOMENCLATURE

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A	Area of beam cross section
C_T	Transverse structural wave velocity
E	Elastic modulus
f	Fluid force/length
F	Force dimension
G	Shear modulus
g	Acceleration of gravity
I	Beam cross section area moment of inertia
J	Beam cross section area polar moment of inertia
K	Beam stiffness
L	Length dimension
ℓ_f	Fluid boundary length
ℓ_s	Structural length
m_s	Structural mass/length
n	Ratio of model to prototype length scales
p	Pressure
P	Static pressure
Q_v	Void fraction-two phase flow
Q_w	Quality-two-phase flow
S	Surface tension
t	Time
V	Flow velocity
V_c	Velocity of sound in fluid
x	Axial coordinate
y	Displacement of structure

Other

ε	Strain
η, η_m, η_j	Damping loss factors
ν	Kinematic viscosity
ρ_s	Structural mass density
ρ_f	Fluid mass density
ρ_g	Gas density
ω	Circular frequency
θ	Temperature

Subscript

m	Denotes model parameter
p	Denotes prototype parameter

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ABSTRACT

Similitude relationships currently employed in the design of flow-induced vibration scale-model tests of nuclear reactor components are reviewed. Emphasis is given to understanding the origins of the similitude parameters as a basis for discussion of the inevitable distortions which occur in design verification testing of entire reactor systems and in feature testing of individual component designs for the existence of detrimental flow-induced vibration mechanisms. Distortions of similitude parameters made in current test practice are enumerated and selected example tests are described. Also, limitations in the use of specific distortions in model designs are evaluated based on the current understanding of flow-induced vibration mechanisms and structural response.

INTRODUCTION

The current state-of-the-art knowledge of the fluid-excitation mechanisms responsible for the flow-induced vibrations (FIV) is not well enough developed to rely solely upon analytical response predictions to implement component design, especially for the complex geometries prevalent in industrial application [1,2]. Scale-model testing is employed as another means of determining component resonse for use both in the determination of new designs as well as in design verification. At this relatively early time in the development of prediction methods, more validity is afforded test results than analysis results if disparities occur. However, care must be exercised in unquestioned reliance upon test results, since they are only as meaningful as the validity of the scaling relations employed to design the test.

In practice, the violation or distortion of some of the scaling relations in model design and testing is inevitable. Typically testing at a reduced scale, to minimize costs, does not allow satisfaction of all similitude parameters simultaneously. Just as often, important parameters are overlooked because of the lack of understanding of the physical problems. Procedures have been defined [3,4] to identify and correct for distortions in scale modeling, but they usually involve multiple tests which, although feasible for small simple systems, are uneconomical for large complex systems.

Usually a single test design is sought where "conservative" distortions are made. Conservative distortions include disregard of physical paramters which do not significantly influence the fluid excitation mechanism under study, and the deliberate distortion of some of the significant parameters to produce a scale model more likely to undergo detrimental vibrations than the prototype.

Of course modeling practices evolve as more knowledge is gained about basic fluid/structure interaction. The recognition that the dynamic response of architectural structures could be greatly affected by the boundary layer turbulence developed by the upwind landscape has been appreciated only recently [5], and has made much previous data and testing methods obsolete. The purpose of this work is to review current scale modeling similitude parameters employed in FIV test designs, and discuss conservative modeling distortions commonly made in practice with particular reference to their limitations. The discussions will not include rotating machinery, which tends to develop a separate literature from the other components: heat exchangers, reactor vessels, piping, etc.

MODEL TESTING PHILOSOPHY

With very little known about the structure and fluidelastic mechanism under investigation, general scaling relations can be derived which are only useful for producing model designs which are exact geometric replicas of the prototype, usually at a reduced scale size. Such scaling relations are based on identification of all possible influential independent physical parameters and their reduction into a lesser number of dimensionless parameters by use of the Pi Theorem of Dimensional Analysis [3,6]. The equality of the dimensionless similitude parameters in the model and prototype lead to the desired scaling relations. Because the inclusion of irrelevant physical parameters is inconsequential, while missing an important one can lead to invalid test results, many similitude parameters may exist with conflicting requirements which can only be satisfied by a model duplicating the prototype at full scale model.

Usually some knowledge of the physical phenomenon exists, and many of the irrelevant parameters and scale model requirements can be eliminated. At a level of knowledge where the governing differential equations and boundary conditions can be formulated, a further reduction in the number of scaling relations can be expected by identification of those dimensionless parameters which only occur in combinations. With such information, if only a single fluidelastic mechanism is under investigation, a model may be constructed which is quite simple, considerably cheaper than the prototype, but physically quite different in appearance than the prototype. However, the model is only able to make accurate predictions for those characteristics simulated in the model. In general, the more fluidelastic mechanisms under simultaneous study, the more the model will have to duplicate the prototype because of competing scale modeling requirements. In turn, the cost of the model approaches the cost of the prototype as the general applicability of the model is increased. Of course, when enough information is known to enable a direct solution to the governing equations, numerical simulation, rather than experimental modeling usually is a more cost effective method of analysis.

Evidently the purpose of the test will greatly affect the scaling relations, the model simplicity, and the test costs. Recognizing that flow-induced vibrations cannot be completely eliminated, generally two types of tests are performed in reactor design [7,8,9]. Early in the design process individual feature tests are performed for those components in the system having a high known potential of experiencing severe vibrations due to strong excitation mechanisms. In a feature test, the component is isolated from the system using the best available information on expected flow fields and structural response. If an active mechanism is found, the component is redesigned. After completion of the design, a design verification test is performed to determine if the response of all the components of the system, due to the remaining weak excitation mechanisms, satisfies the design criteria. Of course, the existence of unforeseen strong excitation mechanisms is determined, also.

Because single components are being tested for the existence of specific excitation mechanisms, the scaling relations for a feature test can be made as specific as allowed by the existing state-of-art knowledge. As discussed earlier, more specific scaling relations imply less complex models and cheaper test costs. Very general scaling relations, sophisticated models, and high test costs can be expected for design verification testing because many components are being tested for many excitation mechanisms, each of which may have competing scale modeling requirements.

Although prototype testing will not be discussed here, a general understanding of its purpose is helpful in placing model testing in perspective. Both preoperational and operational tests may be performed in the prototype reactors; however only selected tests are performed because instrumentation lifetime and accessibility is limited while costs can be prohibitive. Typically, the first of each generation of reactors is instrumented to assess the validity of analysis and scale model tests results for components deemed most critical to reactor operation and safety. Also, these tests may be used to assess the existence and magnitude of excitation mechanisms which can only be simulated in the prototype. Post operation inspection of components for wear, fatigue, and other damage is the last step in an FIV test cycle.

Before proceeding with a discussion of model testing scaling relations, some inherent limitations on similitude in model testing will be discussed. Some of these limitations if not physically insurmountable, make testing in the prototype equally attractive on the basis of cost.

INHERENT LIMITATIONS IN REACTOR SYSTEM MODEL TESTING

Almost by definition of a model test of a reactor system or component, the physical phenomenon associated with the fission process is not duplicated, but it must be otherwise taken into account where necessary and possible. For FIV testing, the effects on material properties and changes in structural geometry are of most concern.

Except in the immediate vicinity of the fission process in the reactor core, the temperature and radiation fields can be regarded as having gradual gradients and as steady state for purposes of FIV testing. Utilizing the best available field measurements and/or calculations, spatial variations in the structural material properties, the adiabatic test temperature, and model fabrication techniques can be employed to simulate the temperature and radiation effects on the prototype material properties. At least, a conservative distortion of the prototype material properties can be attempted. This process may require considerable mathematically modeling of the dynamic response of the prototype and experimental model.

The components in the immediate vicinity of fission process are the fuel rods grouped into bundles, assemblies, or strings, depending upon the manufacturer. Because of the modular nature of these configurations, a requirement for refueling, each new generation of fuel configurations is tested in full scale feature tests and often in operation in existing compatible reactors [10]. Except for wear, few FIV problems have been associated with fuel rods. Even the wear has not been an insurmountable problem because of the replaceability of the fuel.

Scale modeling of spatial variations in the heat-transfer fluid properties which are affected by temperature gradient is not possible without simulation of the heat sources. For FIV testing of single-phase fluids, heat sources do not require simulation. A homogeneous temperature for the fluid is chosen which produces a conservative distortion of the fluid viscosity. This is usually possible because of the insensitivity of most FIV mechanisms to wide variations in viscosity. Selection of the proper ranges of viscosity will be addressed later in the discussion of Reynolds number.

The need for simulation of heat sources for testing in two-phase heat-transfer fluids remains questionable, and it is not done often because of the difficulty of such simulations [11]. Adiabatic gas-liquid mixtures, mainly air and water, can be employed to simulate different two-phase flow regimes (bubbly flow, slug flow, annular flow, and mist flows), Fig. 1(a)-(f), but they cannot simulate transition phases and the simultaneous existence of different phases which can occur in the prototype, Fig. 1(g), nor local boiling effects. There is some evidence that local boiling effects [11] are not important, but general conclusions cannot be drawn because of

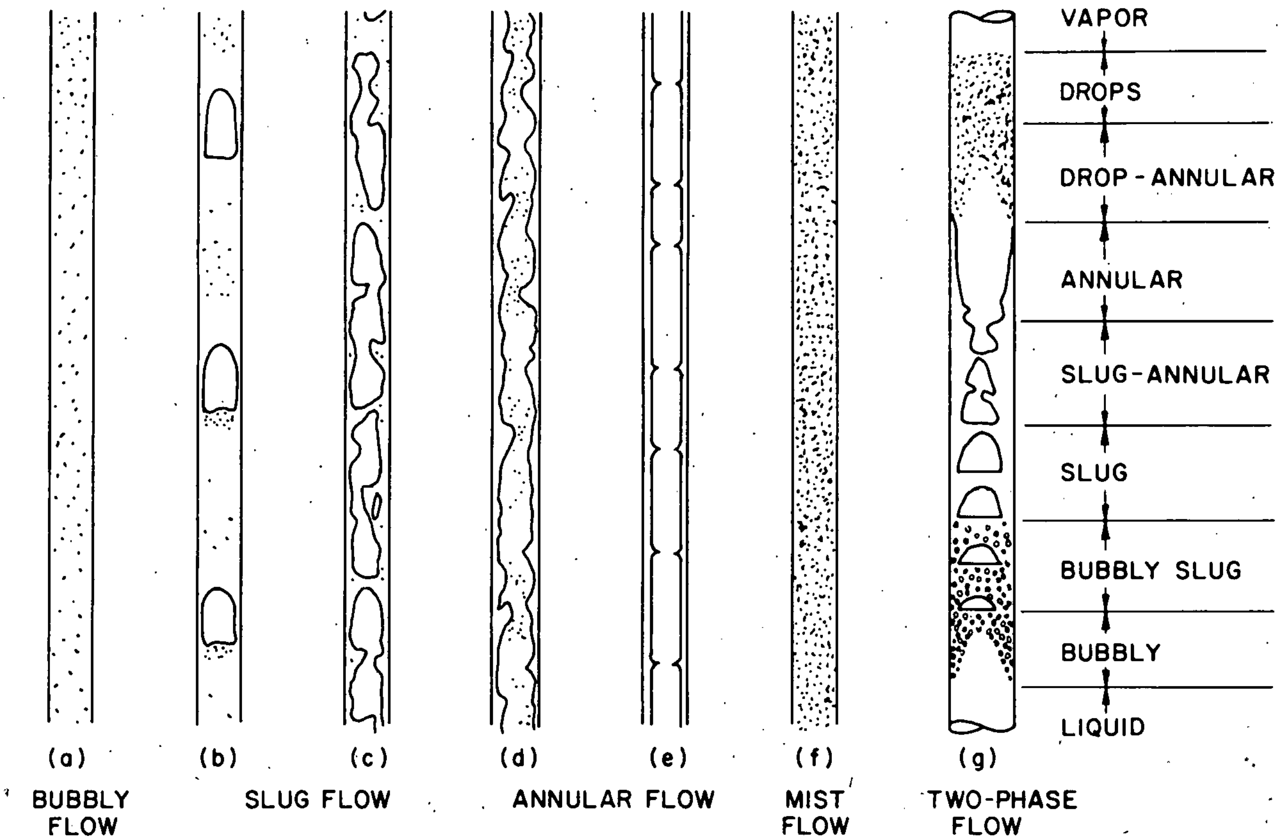

Fig. 1. (a–f) Flow Regimes in Adiabatic Two-Phase Flow and (g) Approximate Sequence of Two-Phase Flow Regimes in a Vertical Tube Evaporator (Adapted from [11])

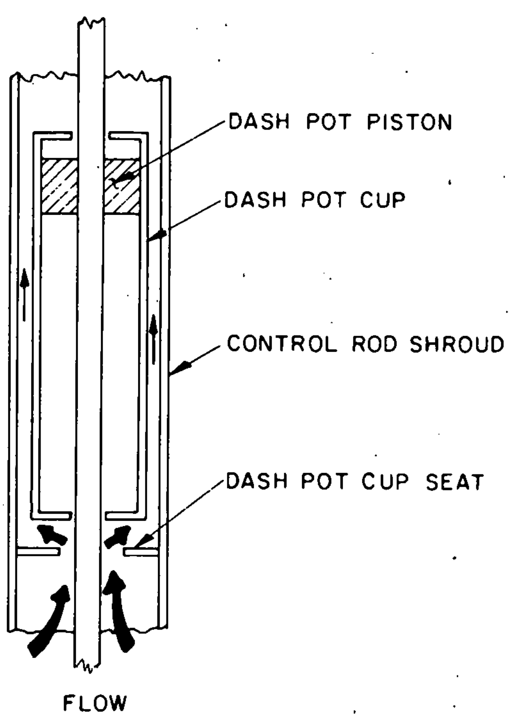

Fig. 2

Flow Lifting Dashpot Cup Off Seat

the limited number of test conditions and observations. To date, no known strong FIV excitation mechanisms have been associated with transition phases and local boiling. Further discussion will be restricted to model simulation with adiabatic gas-liquid mixtures, although fundamental studies of two-phase FIV excitation mechanisms are not available to justify fully this simplification. Clearly fundamental studies on the effects of local boiling and phase transitions on FIV are indicated.

The significant size changes which occur in the structure due to thermal expansion and radiation swelling must be understood for simulation in the model. Since the relative expansion of components is a prime concern in structural design, the creation of indeterminate structures which produce large loads and high stresses are avoided by design with few exceptions. Normally the structural connections allow for relative motion between components and/or supports, unless interference fits are specifically desired. As a result, the main experimental or mathematical modeling difficulty created is the definition and simulation of component restraints and damping at the connections, both of which can affect greatly the vibration frequencies and response amplitudes.

Joined components may respond independently of each other or as if they were part of the same structure depending upon the clearances between the components, the relative expansion between components, the static deflection of one component with respect to the other, and the amplitude of the vibrations. In general each joint must be considered individually. Where they are ill-defined, and usually nonlinear, a conservative distortion of the prototype joint must be designed for the model. Where conservatism in the model design is in question, the simulation of several joints in a series of tests may be required. Simulation of joints is a major consideration in model design and the subject will be returned to again, both in the general discussion of scaling relations which follows and the specific examples presented later.

Keeping the inherent limitations discussed above in mind, the scaling relations for acceptance tests will be considered first. They are more general than those of feature tests and will be somewhat similar for all tests. Since each feature test design depends uniquely on the available knowledge of the specific flow mechanism and structure under investigation, the scaling relations usually are unique to each test and can only be discussed by example.

ACCEPTANCE TESTING SCALE MODELING RELATIONS

The independent structural parameters which are most likely to affect flow-induced vibrations are listed (1-16) in Table 1, and essentially are the same as those given for most fluid-structure interaction problems [9, 12-15] where complete similarity between model and prototype is required. Some usual dependent parameters are listed (1-4). Note the basic dimensions of the parameters given in Table 1 in terms of force F, length L, and time T. As the influence of each parameter on FIV is discussed, the need for an almost entirely geometrically similar structure will become more apparent.

One of the most important parameters is the geometry of the structure where interaction with the fluid creates significant fluid excitation or damping forces. If much is known about the effect of the flow shape upon the fluid excitation mechanism under study, then one can be selective in which part of a component's geometry needs to be made similar to the prototype. Where the effect of the flow shapes is unknown, as is often the case in the study of whole reactors systems, the geometry should be made similar. Thus, it is not unusual to have models which are nearly exact replicas of the prototype at a chosen reduced geometric scale [1,7-9]. When a geometrically similar model is constructed, the single reference length ℓ_f of Table 1 is sufficient for geometry characterization.

The importance of the flow shape should not be underestimated. Sometimes apparently small details in the geometry can create fluidelastic excitation mechanisms which would otherwise not occur. For example, changes in the trailing edge geometry of thin plates [16] and circular rods [17] in parallel flow can significantly accentuate or damp the active fluid excitation mechanism. The detrimental wear of instrument lines due to a relatively small amount of core bypass flow was not identified [18] until several plants were in operation. The bypass flow was channeled through relatively small holes in the fuel assembly support plate but were sufficiently close to the instrument lines to create a jet buffeting problem. Another example of the effects of small details in geometry will be given later in the discussion of a feature test designed to investigate fluidelastic mechanisms associated with flow from fuel assembly nozzles.

Flow-induced vibrations rarely involve inelastic behavior of the structural material, thus the elastic moduli are the important material stiffness parameters effecting stress levels, wave propagation, vibration frequency, and vibration amplitude. Although the elastic moduli may be different throughout the structural system, reference values E and G are sufficient to characterize material stiffness. Where Poisson's ratio effects can be considered unimportant, only one of the parameters needs to

Table 1. Physical Parameters for Acceptance Testing of Geometrically Similar Structures

INDEPENDENT PARAMETER

	System	Distributed Parameter	Reference Parameter	ℓ_f	Dimension
1.	Structure	Flow Shape	Length	ℓ	L
2.	Structure	Stiffness	Elastic Moduli	E,G	F/L^2
3.	Structure	Mass	Density	ρ_s	FT^2/L^4
4.	Structure	Material Damping	Energy Loss Factor	η_m	-
5.	Structure	Connection Damping	Energy Loss Factor	η_c	-
6.	Fluid	Flow Field	Velocity	V	L/T
7.	Fluid	Mass	Liquid, Gas Density	ρ_f, ρ_g	FT^2/L^4
8.	Fluid	Viscosity	Kinematic Viscosity	ν	L^2/sec
9.	Fluid	Mixture	Void Fraction (Quality)	$Q_v(Q_w)$	-
10.	Fluid	Surface Strength	Surface Tension	S	F/L
11.	Fluid	Compressibility	Velocity of Sound	V_c	L/T
12.	External	Weight	Acceleration of Gravity	g	L/T^2
13.	External	Load Magnitude	Force	F_e	F
14.	External	Periodic Loads	Frequency	ω_e	1/T
15.	External	Boundary Movement	Displacement	y_b	L
16.	External	Periodic Motion	Frequency	ω_b	1/T

DEPENDENT PARAMETERS

	System	Distributed Parameter	Reference Parameter	ℓ_f	Dimension
1.	Structure	Deformation	Displacement	y	L
2.	Structure	Deformation	Strain	ε	-
3.	Structure Fluid	Motion	Frequency	ω	1/T
4.	Fluid	Surface Loading	Pressure	p	F/L^2

be simulated. Although the structural mass is usually unimportant for static strength analyses, it is equally as important as material stiffness in determining dynamic structural response. Again a single reference value ρ_s is sufficient for geometrically similar structures.

Structural material damping of vibrations is frequency dependent [19], and thus is nearly impossible to simulate exactly in geometrically similar scale models where the structural frequencies are different in the model and the prototype. However, the material damping normally is dominated by joint damping η_j and, where dense heat transfer fluids are employed, fluid damping may be relatively large. In any case, exact simulation of η_m is not attempted but conservative distortions of the total damping are sought as discussed previously.

Fluid forces developed by shearing the fluid element depends upon the viscosity of the fluid. Since viscous effects are dominated in high velocity flows by turbulence effects, often the effects of kinematic viscosity ν can be neglected for heat transfer systems which promote flow turbulence. However, as shall be discussed later, cases exist even in apparently highly turbulent flow where viscous effects are important.

Of course, the flow field is the energy source for the FIV excitation mechanism. If geometrically similar flow geometry exists, then a single reference flow velocity V is sufficient to characterize the flow field. If similar geometry does not exist everywhere in the model, then much must be known about the flow field. At the component under study the correct mean-flow, turbulence intensity, and turbulence length scale, all of which are known to effect to varying degrees most fluid excitation mechanisms, may have to be simulated. Since knowledge of such flow detail is usually not known a priori, and production of turbulent flows is not a straightforward task, models which are everywhere geometrically similar are employed more often than not.

The mass of the fluid is not only important in determining the magnitude of fluid excitation forces, but dense fluids can interact with structural motion sufficiently to cause relatively large differences between frequencies of structures in air and in the dense fluid, as well as coupling the motion of structures which are not mechanically connected [20]. Both liquid and gas phase reference densities, ρ_f and ρ_g, are important if two-phase flow exists, whereas only ρ_f or ρ_g is important for single phase flow.

For two-phase flow, the type of flow regime which exists greatly determines the strength of the fluid excitation mechanism. The parametric vibrations [21] associated with slug flow (see Fig. 1) appear to be potentially the most detrimental mechanism. The flow regime which is present depends fundamentally on the liquid's surface tension S and the relative amount of liquid to gas, whether measured by volume ratio Q_v or weight ratio Q_w [11].

Fluid compressibility is important to simulate where sound propagation or standing acoustic waves are expected to create significant excitation mechanisms. Certainly simulation of the velocity of sound V_c in single phase gas systems is needed because of the many associated failures which have occurred [22].

When the flow is interacting with all or part of a structure which responds as a mechanism subject to its own weight, or when free surface waves in the fluid are of importance, then the external acceleration of gravity g must be considered a fundamental parameter, also.

Two parameters which are often controlled in the prototype, temperature θ and static pressure P, do not appear directly in Table 1, although they effect many of the parameters listed. In model tests, θ and P often are chosen to give the optimum simulation of the parameters in Table 1.

Utilizing the theory of Dimensional Analysis [6], the dimensionless similitude parameters governing the independent physical parameters of Table 1 can be formulated. For a given number of independent physical parameters, the number of similitude parameters is unique, but the similitude parameters are not since they can be multiplied together and raised to powers in any combination. A typical set is given in Table 2 (1-14). The dependent parameters are given in (1-4). Equating the similitude parameters between model and prototype gives the scaling relations for test design.

Even for the case of complete geometric similitude and a single phase heat transfer fluid, all of the scaling relations in Table 2 cannot be satisfied in practice, except by testing in the prototype. For instance, to maintain similitude according to the fluidelastic parameter (1) and the fluid mass ratio (2) in Table 2, the ratio of the model to prototype flow velocity $(V)_m/(V)_p$ is required to be $(C_T)_m/(C_T)_p$ where $C_T = \sqrt{E/\rho_s}$ is the transverse wave velocity in the structure; while to satisfy the Reynolds number (3) in Table 2 would require the velocity ratio to be $(1/n)(\nu)_m/(\nu)_p$, where $0 \leq n \leq 1$ is the ratio of model to prototype length scales. In practice, fluid and structural materials are not available which can satisfy both these velocity ratio requirements at other than close to full geometric scale where the cost of model construction is high. Fortunately in most reactor systems, turbulent flow is dominant and Reynolds number dependences are few. However, even in predominantly turbulent flows Reynolds number effects may exist (see next section) which must be accounted for, at least by conservative distortions.

Table 2. Similitude Parameters for Geometrically Similar Models

Independent Similitude Parameter

	Ratio of	Importance
1. $E/(\rho_f V^2)$	Strain Energy to Fluid Kinetic Energy	Fluid Structure Interaction
2. ρ_s/ρ_f	Structural to Fluid Density	Liquids
3. $V\ell_f/\nu$	Fluid Inertia to Viscous Forces: Reynolds Number	Laminar Flow Effects
4. η	Component Energy Dissipation to Strain Energy: Energy Loss Factor	Vibrations
5. $V_c^2\rho_s/E$	Speed in Component to Speed of Sound in Fluid: Mach Number	Acoustic Wave Propagation
6. $g\ell_f/V^2$	Gravitational Force to Inertia Force: Froude Number	Surface Waves, Mechanisms
7. E/G	Elastic Extensional to Shear Modulus	Poisson Ratio Effects
8. Q_v	Gas to Liquid Volume: Void Fraction	Two-Phase Flow
9. $\rho_f V^2 \ell_f/\sigma$	Inertia Force to Surface Tension Force: Weber Number	Two-Phase Flow
10. ρ_f/ρ_g	Liquid to Gas Density	Two-Phase Flow
11. $F_e/(\rho_f V^2 \ell_f^2)$	External Force to Fluid Inertia Force	Support Excitation
12. $\omega_e \rho_f/V$	Load Frequency to Flow Periodicity	Support Excitation
13. y_b/ℓ_f	Boundary Displacement to Length	Support Excitation
14. $\omega_b y_b/V$	Boundary Velocity to Flow Velocity	Support Excitation

Dependent Similitude Parameter

1. y/ℓ_f	Displacement to Length	Function Limitation
2. ε	Elongation	Function Limitation
3. $p/(\rho_f V^2)$	Fluid Pressure to Velocity Head	Input for Analysis
4. $\omega\ell_f/V$	Frequency to Velocity Reduced by Length	Input for Analysis or Fatigue Design

Simultaneous simulation of the fluidelastic parameter, the structural to fluid density, and the Froude number, (6) in Table 2, is impractical as well, since Froude number simulation requires a velocity ratio of $\sqrt{n}$. However, components subject to gravity effects are normally few in number in internal flow systems and easily identified. Thus neglect of the Froude number in acceptance testing design is feasible, relegating the investigation of any questionable gravity effects to special tests.

CONSERVATIVE DESIGN CONSIDERATIONS

Some of the conservative distortions made in scale models have been indicated in the previous section. Others exist. All conservative distortions must be justified on the basis of the current understanding of the state-of-the-art, but the process is not necessarily straightforward. Often valid reasons exist both for and against distortion of particular similitude parameters. The usual dilemma is that theories are proposed or new data becomes available for idealized or specific geometries which indicate that a similitude parameter is or is not important, but it is not clear whether those special cases are applicable to slightly different configurations. In other words, a complete fundamental understanding of many FIV mechanisms does not exist, and until such time, the designer will have to decide on a case by case basis whether each distortion is justified. As an aid to the designer, selected information based on the current understanding of the state-of-the-art will be presented for each of the similitude parameters of Table 2.

The fluidelastic parameter and ratio of structural to fluid mass, (1) and (2) in Table 2, are most important parameters in FIV testing and normally are not intentionally distorted. However, justification for testing with model fluids slightly more dense than required by the scaling relations is reasonable. The greater driving energy of a denser fluid should lead to a conservative distortion. But care must be exercised with dense liquids, because the structure and fluid natural vibration modes may be strongly coupled and significant changes in frequencies could occur [20]. This could result in significant changes in response amplitude or deny the existence of a fluidelastic instability which would occur for the correct density fluid. Similar arguments can be made for the employment of less stiff structures and higher flow rates than required by the scaling relations. The maximum flow velocities in tests are routinely chosen higher (~25%) than required to insure that the structure is not on the threshold of an instability at normal operating flow rates.

Because simultaneous simulation of the fluidelastic parameter and Reynolds number nearly always requires a full scale test, the distortion of Reynolds number is nearly always considered because of the significantly lower costs of reduced scale testing. Due to the normally high turbulence of flow in reactor components, more often than not Reynolds number can be distorted without a significant distortion in structural response. The situations where simulation of Reynolds number must be considered usually can be identified with flow through small passages, sometimes called leakage flows; flow through valves, orifices, or other flow control devices; flow separating from a component; and components which are exposed to flow excitation over part of their bounding surface but are highly confined by narrow fluid filled gaps where any flow is due only to motion of the component.

In leakage flows through narrow passages, the wall surface resistance losses can be significant and depend upon Reynolds number. The movement of a component may modulate the amount of flow (energy) losses, or the area of the component over which the losses occur. In any case, whenever more energy is derived from the flow by a component in a cycle of motion than is dissipated, self-excitation is possible [23,24,25]. Flow control devices designed to modulate flow losses through variable flow constriction are subject to similar self-excitation [26] and Reynolds number sensitivity of the discharge coefficient can be expected. However a complete understanding is not available, since the effects of the modulation of the discharge coefficient as a function of flow constriction is still under study.

The most common Reynolds number dependent flow separation which occurs in reactor system components is the formation of the wake behind circular tubes in cross flow, resulting in vortex shedding. The character of the wake and the associated fluid forces may be considerably different depending upon whether the boundary layer before separation is fully laminar, fully turbulent, or in transition [21]. For uniform, nonturbulent cross flow over isolated stationary cylinders with smooth surfaces, large ranges of Reynolds number can be defined over which fluid forces can be assumed insensitive to changes in Reynolds numbers, thus making the design of distorted Reynolds number tests feasible. However, complications arise in design because of nature of the vortex shedding dependence upon Reynolds number and its sensitivity to other parameters.

An intermediate transition range of Reynolds numbers exists where vortex shedding becomes incoherent and relatively ineffectual as an excitation source, at least in comparison to the ranges above and below where most coherent, strong vortex shedding occurs. Obviously then, test Reynolds numbers at maximum flow velocities should be in the correct range, and for a conservative design not in the transition range. Unfortunately, avoiding the transition range, if indeed it exists in practical flows, is not straightforward. The bounding Reynolds numbers for the large ranges are known to vary, up to an order of magnitude, with surface roughness, nonuniformity in flow, turbulence levels, and amplitude of component motion [27-29]; and all are the subject of current research. Thus, particular attention should be given to the current research results when prototype Reynolds numbers occur above the transition range, 2×10^6 and above, to avoid testing a reduced scale model in the transition range. An approximately quarter scale model acceptance test has been performed [30] wherein the effects of a wide variation, factor of three, in Reynolds number was investigated by heating and pressurizing the test fluid. Even though many instances of crossflow, under varying conditions were present, no changes in response amplitudes of more than twenty percent were observed; nor were there any changes in the character of the component's responses.

For tubes in closely packed bundles, vortex shedding usually is not the dominant excitation mechanism [42], and tube response is generally insensitive to Reynolds number. The vortex shedding from other component shapes may have a Reynolds number sensitivity, if the point of boundary layer separation is variable or reattachment can occur as in airfoil stall or stop sign flutter [31,32]. Otherwise sharp edged components eliminate any dependence by fixing the boundary layer separation point and creating a fully developed turbulent wake.

The necessity to allow for relative thermal expansion between adjacent components and to optimize space usually results in narrow fluid filled gaps between many of the reactor components and/or their supports. Often the fluid is not flowing except due to component motion, and viscous damping, which is sensitive to Reynolds numbers, is provided to the component. The total loss factor for a mode of vibration of a component η, similitude parameter (4) of Table 2, can be considered to be the sum of that due to viscous damping η_v, material damping η_m, and connection damping η_c. Normally η is different for each vibration mode. Since narrow liquid (dense fluids) filled gaps can provide relatively large amounts of viscous damping, their effect on total model damping should be investigated whenever Reynolds number is not simulated. Model response results may be unconservative, because a distorted lower Reynolds number in a model test implies more viscous damping may occur for the model than the prototype.

In general more damping in the model than in the prototype implies smaller response in the model than in the prototype. Although a distortion in damping is not directly related to an excitation mechanism, the effect on a component's linear vibration response, for example, varies as the inverse of the loss factor η when the component is driven by a deterministic load at a resonant frequency. For wideband random excitation the response in a vibration mode varies as $1/\sqrt{\eta}$. Also self-excitation would be less likely to occur in a model with more damping than the prototype, since more energy would be dissipated by the model. Whether forced or self excited, an unconservative test may result. Forced vibration model test results can be corrected with theory if the damping is known for the model and prototype, but only avoidance of such distortions or a special design to yield smaller damping in the model than in the prototype can guarantee identification of self-excitation mechanisms [12]. Thus obtaining estimates of η for components expected to be more heavily damped in a model than in the prototype is most important toward achieving a conservative model test design.

Returning to consideration of liquid filled gaps, the identification of those with potential for providing significant viscous damping usually is apparent by inspection. However, information for estimating viscous damping is available for only a few idealized cases [33-36]. Unfortunately the results from the relatively more developed lubrication theory are not usually applicable, because the gaps are not so small that fluid inertia effects can be neglected. In fact, fluid viscosity can effect not only the apparent structural damping but the apparent structural mass as well.

The inability to simulate and estimate damping is not limited to viscous liquid effects. As has been mentioned, material damping is frequency dependent and cannot be simulated at reduced scale, but it is relatively small. More importantly, simulation of joint damping is not necessarily accomplished by reproducing an exact geometric scale model from the same materials [37,38]. In general, simulation of component damping cannot be accomplished, but conservatively distorted damping factors can be produced by careful design of each joint using the estimates from the available theory and measurements made on typical structures [39-41].

Simulation of acoustic wave propagation is conceptually straightforward, if the same structural material and fluids are employed in the model and the prototype. Investigation of component vibrations due to pump and flow noise has not been emphasized for systems with liquid and two-phase coolant flows because they are not usually associated with significant vibration problems. Acoustic excitation has been found to be a major source of excitation in gas systems [22], primarily when the plenum size and a sound wave-length are similar. Destructive standing waves result [42].

Since wave action is limited in internal flow systems to plenums, where wave suppressor plates are usually employed, the only remaining gravity sensitive excitation mechanisms would involve components essentially without stiffness and free to move subject to their own weight and impinging flow. Few such components exist in the internal flow systems of reactor components.

One example [44] was a dashpot cup free to move, within the constraints of stops, along a control rod drive line in the vertical direction. The function of the dashpot is to slow movement of the control rod during reactor scram. The mass of the cup was not negligible and was designed to simulate lateral vibrations of the control rod drive line. However, the weight of the cup was ~ 3 times lighter than required for Froude number simulation. During one-quarter scale model flow-induced vibration testing, the cup was found to be alternately lifted off its seat and dropped by the flow in the shroud containing the dashpot. Subsequently, modifications to the mass of the dashpot cup were made to conservatively simulate the Froude number and the excitation mechanism was not present.

The test parameters in Table 2, (8-10), concern simulation of two-phase flow. No attempts at simulation of two-phase flow in large scale FIV tests are known. One main difficulty is the need for simultaneous simulation of single-phase flow at one point along a flow path and two-phase flow at another point using liquid-gas mixtures. Nearly all FIV model tests of two-phase flow systems have been performed with single phase fluids. Single-phase mass densities and fluid velocities have been chosen to simulate the two-phase flow kinetic energy or momentum [11]. These distortions would appear conservative for homogeneous bubbly or mist flow regimes, but certainly not necessarily conservative for the more discrete slug and annular flow. The question of whether a conservative test can be designed using single-phase fluids thus becomes a question of justifying the absence of significant discrete two-phase flows.

SPECIALIZED SIMILITUDE PARAMETERS

The similitude parameters given in Table 2 are valid for any elastic continuum, whereas very specific, well understood types of elastic structures are employd in reactor system fabrication. The advanced state-of-the-art of structural analysis allows specialization of the scaling relations for each type of structure: beams, plates, and shells; and for each mode of deformation: bending, shear, torsion, and stretching. Often a structural type deforms predominantly in one mode, say a beam in bending, and considerable specialization of the scaling relations can be achieved [3,44]. With specialization of the scaling relations, often simplifications in the model design are possible. Essentially the structural theory can be used to identify the important geometric and material parameters and justify neglect of others; a form of conservative distortion. Of course identification of the correct structural type and mode of deformation by the test designer is the crucial step in any simplification of the model design, and the process can only proceed on a case by case basis; bearing in mind the model can make predictions only for those characteristics simulated in the model.

As an example, consider a uniform beam structure, of characteristic axial length ℓ_s, subject to a constant flow velocity V flow over a characteristic length ℓ_f. Assume the primary mode of deformation is bending which results in transverse vibrations where rotary inertia [45] can be neglected. Thus, the mass of the beam/unit length m_s and the cross sectional area moment of inertia I are most important independent parameters, as are 2, 4-8, and 11 of Table 1. One set of independent similitude parameters derivable from Dimensional Analysis [6] is given in Table 3, but others exist. As discussed above, often modeling similitude requirements can be simplified based on specific knowledge. In the case at hand for instance, the governing equation of motion for the beam can be written and nondimensionalized in terms of the similitude parameters identified in Tables 2 and 3 to give

$$\frac{\partial^2}{\partial x'^2}\left(\frac{EI}{\rho_f V^2 \ell_s^4}\frac{\partial^2 y'}{\partial x'^2}\right) + \frac{m_s}{\rho_f \ell_s^2}\frac{\partial^2 y'}{\partial t'^2} = f'\frac{\ell_f}{\ell_s}$$

where $x' = x/\ell_s$ is axial distance normalized by a structural length ℓ_s, $y' = y/\ell_s$ is lateral displacement normalized by the same distance, $t' = tV/\ell_f$ is time normalized by a combination of flow velocity V and fluid length ℓ_f, and $f' = f/(\rho_f V^2 \ell_f)$ is the applied fluid force per unit length normalized by the velocity head. The ability to write a nondimensionalized equation of motion shows that simulation of the combination of parameters 1 and 4 in Table 3,

Table 3. Specialized Independent Similitude Parameters for a Beam in Bending

Similitude Parameter	Comment
1. $E/(\rho_f V^2)$	Other Forms: $K_b/(\rho_f V^2)$, $V/\omega_r \ell_f$
2. $m_s/(\rho_f \ell_f^2)$	Structural to Added Fluid Mass per Length
3. η	Loss Factor
4. I/ℓ_s^4	Cross Section Area Inertia to Structural Length
5. ℓ_s/ℓ_f	Structural to Fluid Lengths
6. $V\ell_f/\nu$	Reynolds Number
7. V/V_c	Mach Number

$EI/(\rho_f V^2 \ell_s^4)$, is important and there is no need to prescribe simulation of the parameters individually: a utilitarian simplification for model design.

The term EI/ℓ_s^4 is often termed a stiffness K_b, as noted in the comments. For other modes of beam deformation, the fluidelastic parameter can be generated by use of the appropriate stiffness K. For torsion $K_t = GJ/\ell_s^4$ and for axial (stretching) deformation $K_a = EA/\ell_s^2$, where J is the cross section area polar moment of inertia and A the area of the cross section. The additional important geometric parameters are J/ℓ_s^4 and A/ℓ_s^2. In the case of torsion, the mass moments of inertia must be simulated, also. If all modes of the beam deformation require simulation, then a completely geometrically similar model probably is more practical to construct.

Specialized fluidelastic similitude parameters can be formulated [46] for plates and shells based on stiffness and inertia terms defined by available structural theories. Also, the structural theories can lead to specialized similitude parameters for acoustic wave propagation [47]. Again, geometrically similar models often are more practical to construct, especially when the mode of deformation and effects of structural joints are not clear.

EXAMPLE FEATURE TESTS

Feature tests are usually performed to determine whether a strong fluidelastic excitation mechanism is created by a particular design feature and, sometimes, to determine data for input to analysis of similar future designs. Each test is often motivated by known occurrences of FIV associated problems in similar designs. Many times the feature tests are performed to learn why a problem occurred in an operating reactor, so as to be able to avoid future occurrences or determine a fix for the problem.

Feature tests rarely receive wide documentation because of the inherent proprietary nature of the subject matter, however an indication of the substantial number performed can be appreciated by review of information in the NRC Safety Analysis Reports required for each reactor and available on the public docket. Summaries of the results of feature tests often appear in specialty conferences, many of which have been reviewed in the recent IAHR/IUTAM Symposium on Practical Experiences with Flow-Induced Vibrations [55]. The first example feature test is taken from a contribution to this symposium.

Early in the design phase of the Clinch River Breeder Reactor (CRBR), several components were identified as having a high potential for FIV, based on experience with similar reactor designs. One of the components was the massive Flow Collector hanging from the cover plate of the CRBR reactor vessel over the downstream end of the fuel assemblies, as shown in Sketch 1 of the Flow Collector Chart, Fig. 3. There was concern that the relatively high velocity coolant flow from the fuel assemblies could force the essentially separate Flow Collector to vibrate. To understand the concern, a knowledge of the structural and flow geometry is required.

The structure of the Flow Collector is essentially an inverted frame: four columns (~ 6 m long) attached to the reactor cover plate at one end and fixed to two plates at the other end. The Outlet Modules are individually attached between the two plates. The coolant flow from groups of nine (hexagonally packed) fuel assemblies is collected by a single Outlet Module. Thirty-five Outlet Modules constitute the flow channels of the Flow Collector in the upper plenum. The flow exiting from each fuel assembly nozzle jets into corresponding individual holes in a grid plate attached to the upstream end of the Outlet Module, Sketch 2 of the Flow Collector Chart.

The conditions for generating an alternating force appeared to be present. Because the fuel assemblies are relatively stationary, lateral movement of the grid plate, in the fundamental Flow Collector vibration mode, would alter the locations where the jet flow from the fuel assembly nozzle impinges. If the lateral movement results in an increase in fluid force on the Flow Collector in the direction of motion, then the potential for self-excited vibrations is high. Since analytical prediction of the

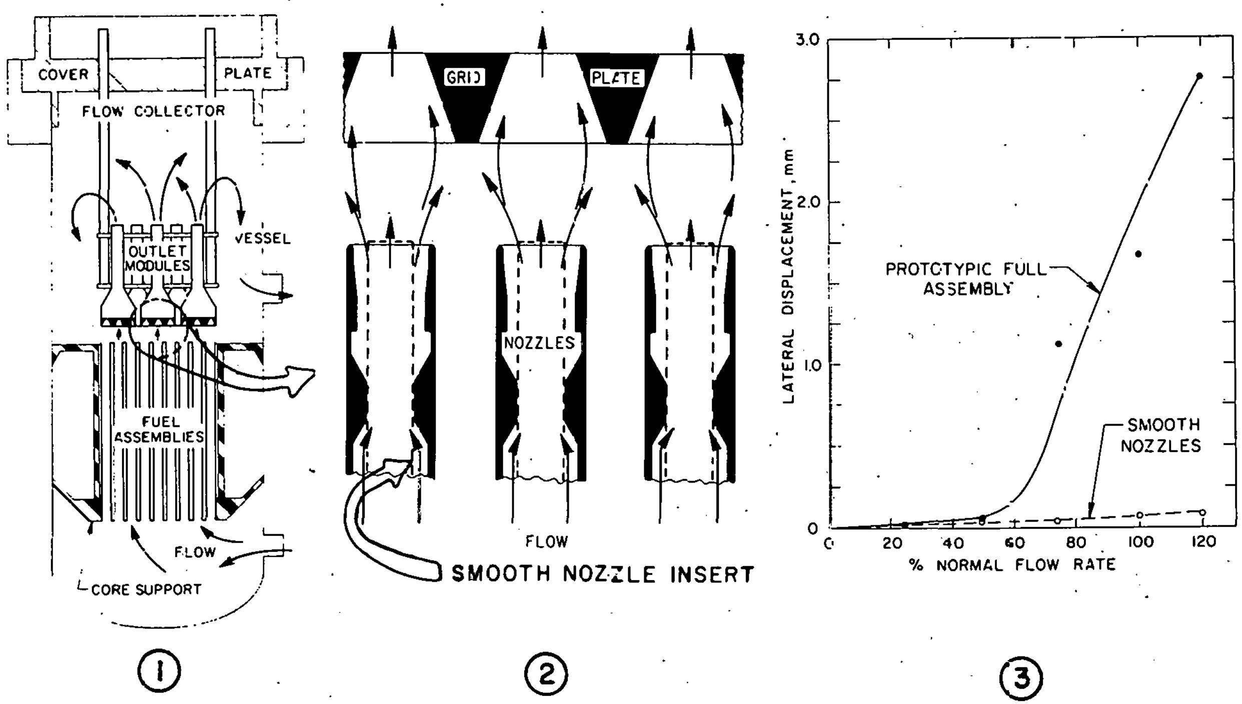

Fig. 3. Flow Collector Chart

variation in fluid forces with lateral movement for such complicated geometry would be difficult, if not impossible, model testing was performed [48].

To assess the potential for self-excitation, a one-third scale model of a single, elastically-supported Outlet Module, Fig. 4, was tested. Obviously, this represents a gross distortion of the Flow Collector, but the distortions were considered conservative and adequate to test for the existence of an excitation mechanism. The mass of the scale model was determined by maintaining the prototypic ratio of contained fluid mass to structural mass, conceptually equivalent to (2) in Table 2. The structural stiffness of the model was determined by maintaining the prototypic ratio of structural potential energy to fluid kinetic energy, conceptually equivalent to (1) in Table 2. The flow rates were made as large as the flow facility would permit in order to minimize distortion of Reynolds number: to one-tenth that of the prototype. Since the excitation mechanism was expected to be dependent mainly on the dynamic pressure, the distortion of Reynolds number was considered unimportant.

As constructed, the scale model actually represented a Flow Collector with every Outlet Module identical and subject to the same flow field. For simulation of prototype Outlet Module flows having a kinetic energy larger than the average kinetic energy for the entire Flow Collector, the test results would be conservative; whereas Outlet Module flows having less than the average kinetic energy were adjusted for the model test to simulate at least the average kinetic energy of the prototype Flow Collector. As shown in Fig. 3, a fluidelastic excitation wsa found to exist. An additional result of the test was enlightening with respect to the importance of simulating small details of the flow geometry. When a smooth pipe was employed for a flow nozzle, instead of the complex prototypic internal flow nozzle geometry, no excitation was observed for any of the test parameters. In the final design of the Flow Collector, the grid plate concept was eliminated and the lateral motion was limited by keying it to the Core Support structure.

During the early 1970's and late 1960's several pressurized reactor internal designs experienced failures [1,2] of the large shell structures (e.g., core barrels, thermal shields) which form, often with the pressure vessel wall, the relatively narrow cylindrical annulus which directs the flow from the several inlet nozzles near the top of the vessel to the bottom of the reactor vessel to enable the heat transfer to flow upward through the core. See Fig. 5. The failure usually consisted of a large movement of the shell due to fatigue of connections. Some of the vibrations leading to fatigue were thought to be flow-induced with the driving forces due to fluid momentum changes associated with impingement on the shells of the inlet nozzle jets, switching between the jets issuing from different nozzles, the

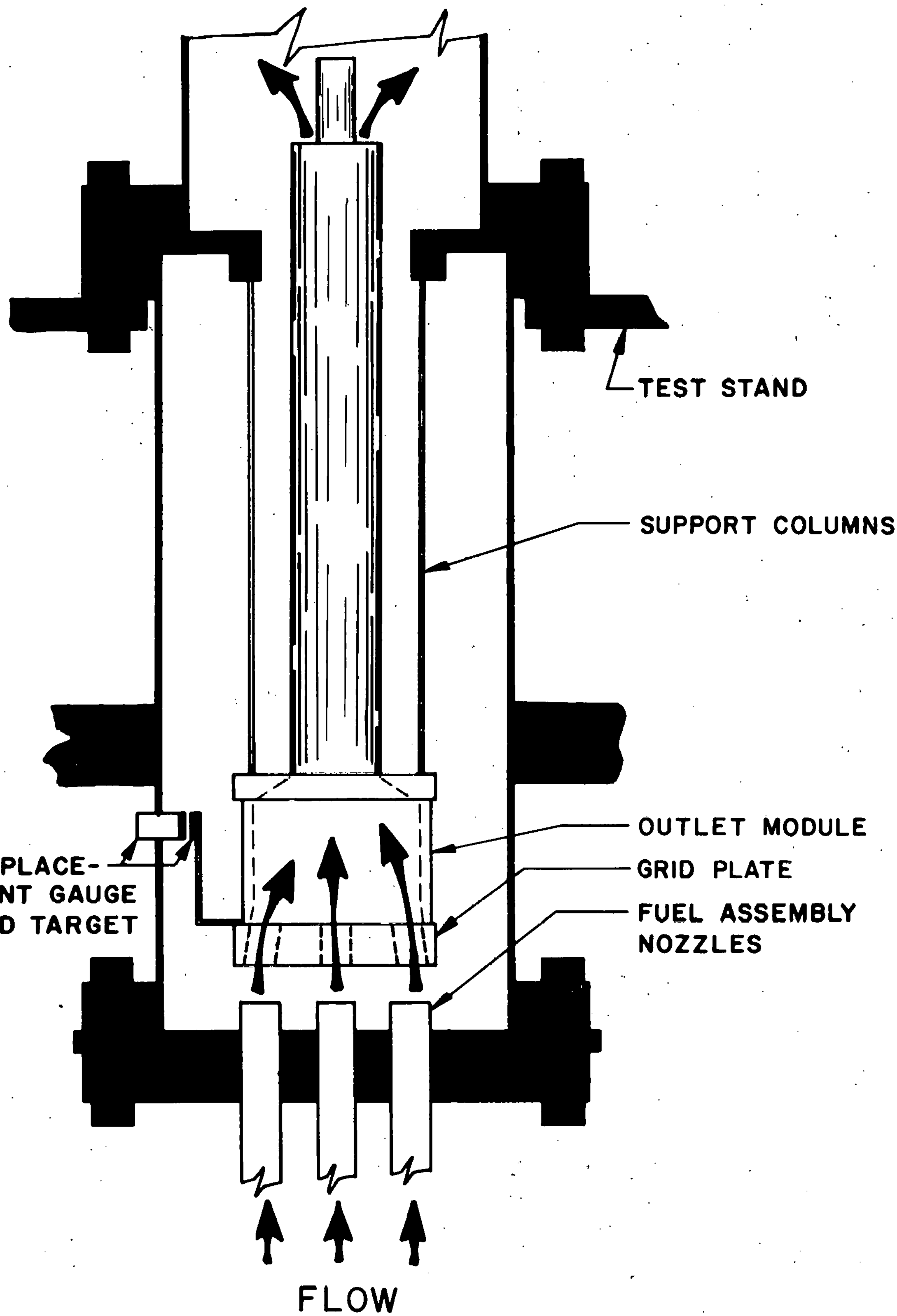

Fig. 4. Distorted Scale Model of Flow Collector

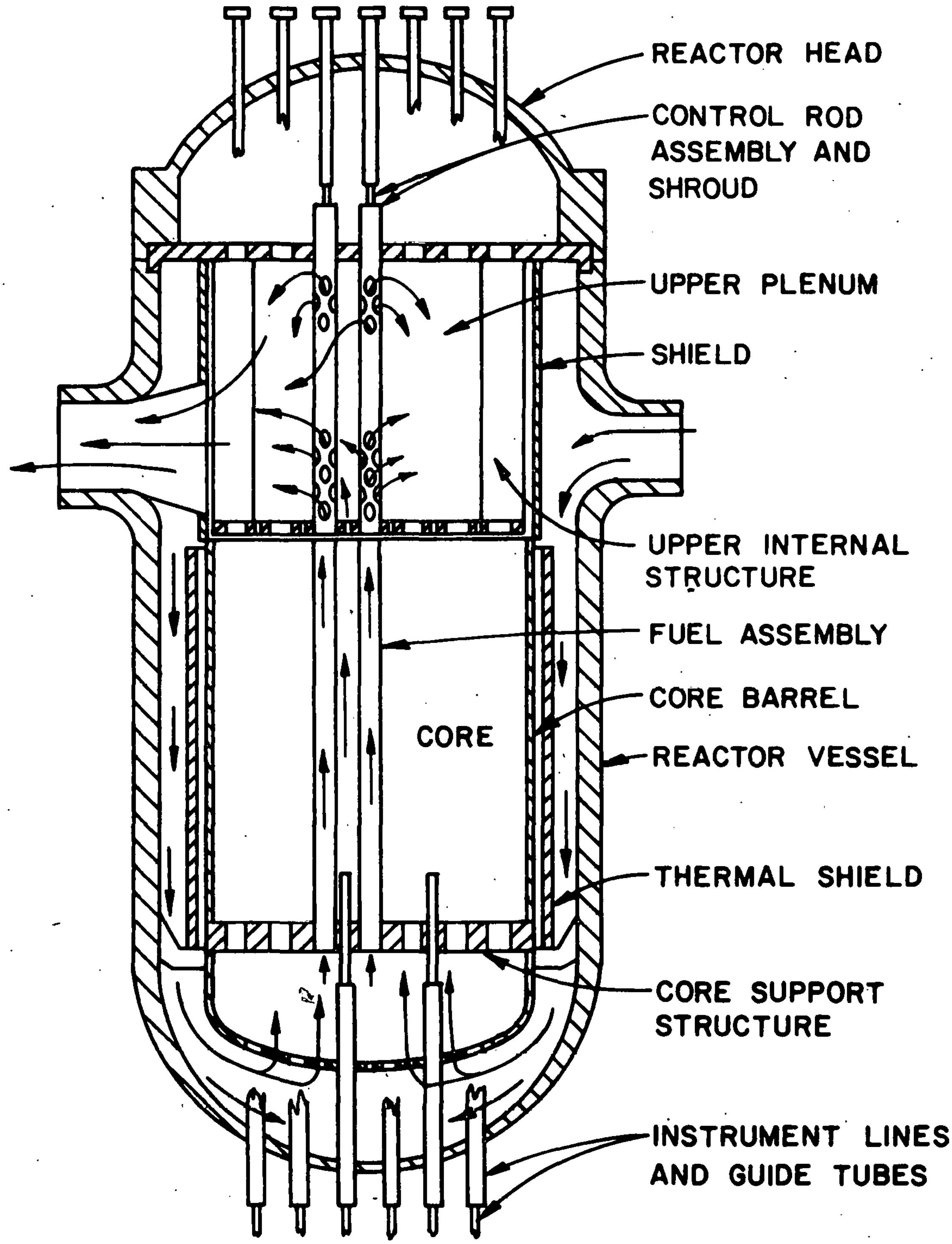

Fig. 5. Schematic of Typical Components and Flow Paths in a PWR

highly disturbed and turbulent flow paralleling the shell surfaces in the annular region, or some combination of the preceding. As a result of the failures, considerable effort has been made to eliminate design features causing the problems, to develop methods to predict the response, and to assess design adequacy via preoperational and scale model testing.

The design of one new generation of reactors underwent thorough analysis, scale model testing, and preoperational testing to assure the adequacy of the shell structure forming the inner wall of the annular flow region [49]. Included in the design process was scale model testing of the core barrel and the inner wall shell structure at 1/24 geometric scale [50]. In this design the so-called shield of Fig. 5, at the level of the inlet and outlet flow nozzles, and the core barrel are one continuous shell cantilevered from the reactor head flange and vessel without any other significant supports to the reactor vessel. A main purpose of the scale model test was to evaluate the relative susceptibility to FIV of two inner wall shell structures. One was a complete cylindrical shell Thermal Shield, as shown in Fig. 5, and supported at each end. The other inner wall was composed of Neutron Pads which are widely spaced segments of a cylindrical shell, like the thermal shield, with each segment separately attached to the core barrel. Since movement of the core barrel could be strongly coupled to the thermal shield by the narrow intervening fluid filled annular gap, the segmental neutron pads supported at the ends and sides directly to the core barrel could be expected to be less susceptible to vibrations.

Similitude of the mass ratio parameter, (2) in Table 2, and the fluid-elastic parameter, (1) in Table 2, were satisfied in the scale model by employing prototypic structural materials and testing with the prototypic fluid and flow rates. Insensitivity to Froude number, (6) in Table 2, was claimed since the large flow rates substantially diminished any effects of natural convective flow in the annulus and no surface waves were present; the model was in fact tested in a horizontal position. Also, all the structures were welded or securely bolted together thus eliminating any Froude number sensitive motion of a structure subject to only its own weight and the fluid flow or opening of structural joints due to component weight. No attempt was made to simulate the characteristic pressure pulsations of the prototypic pump because of a lack of associated problems in the past. The other distorted similitude parameter, Reynolds number, (3) in Table 2, was argued not to affect the predominant flow excitation mechanism identified as the pressure fluctuations due to turbulent parallel flow along the shell wall in the annular region.

Other distortions to aid in fabrication were made and justified. The core was not simulated except for its mass effects on the motion of the core barrel. The dynamics of the pressure vessel wall was not simulated, except to make it relatively stiff as it is in the prototype. Local shell thicknesses and stiffness were modified in places, but always the important vibrational natural frequencies and mode shapes were maintained according to mathematical analysis of the prototype design and measurements of other scale models. The lower plenum below the core was not simulated, nor was reverse flow back through the core, since the primary flow excitation being studied was that of turbulent flow through the annular region.

The important natural frequencies and mode shapes of the model shells in air and water were determined and found to be in good agreement with scaled results measured in other models and the prototype. Although damping was not measured directly, and could be expected to be flow dependent for parallel flow [10], scaled RMS displacements of selected points were found to be in good agreement with measurements in the prototype. Also the neutron pad model was found to respond less than the thermal shield.

A more elaborate 1/8-scale hydroelastic model, simulating nearly all internal reactor components except the fuel assemblies, has been constructed by another manufacturer to study, among other components, the core barrel, the thermal shield, and Neutron Pads [51,52]. The scale model is typical of those employed in design acceptance testing. Modeling procedures were similar to those discussed in the 1/24 scale model, but an effort was made to produce various pumping circuit noises. In this case, not only were component displacements measured, but surface pressures due to acoustic and turbulent flow were measured, distinguished, and utilized in analytical models to predict response. The procedures followed are extensively documented [51-54]. Essentially linear modal analysis models were constructed utilizing natural frequency, mode shape, and damping data obtained in free vibration tests of the components in air and still water. The forced response due to the measured surface pressures were then calculated. The predictions compared well with the model response. Similarly successful prediction procedures were established for prototype components. Of particular interest, the flow turbulence in the annular region again was found to be the main source of excitation of the thermal liner, also dominating acoustic noise effects.

CONCLUDING REMARKS

The construction of true models [3], simulating all aspects of fluid-structure interaction, is impractical. At most, adequate models [3] which are able to predict selected characteristics of fluid-structure interaction are constructed. Most likely the model design must employ conservative distortions of selected similitude parameters, particularly structural damping loss factors and Reynolds number, because of an inability to simulate them at reduced scale where economy of testing can be realized. Similitude parameters and scaling relations for construction of true models are well established; however the construction of adequate models with conservative distortions requires much judgment by the test designer, based on the current state-of-the-art knowledge of flow-induced vibrations.

Several types of FIV model tests and similitude parameters have been reviewed with the intention of providing a basis for future model test designs. Since each test design tends to be unique, based on simplifications and specializations deemed valid by the test designer, the need for and the limitations on employing conservative distortions and specialized similitude parameters was discussed in particular. In these discussions, the current FIV state-of-the-art was found to be under development in many instances. Recognition that knowledge of FIV mechanisms is rapidly advancing places model testing in perspective: the validity of model test results, for other than a true model, can reflect, at most, the best engineering judgment currently available and is no better than currently available theory but probably is easier to implement for complex structural geometries.

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