

⟨**Technical Note**⟩

Fluidelastic Instability Characteristics of Helical Steam Generator Tubes

**Jong Chull Jo, Myung Jo Jung, Woong Sik Kim, Young Hwan Choi,
and Hho Jung Kim**

Korea Institute of Nuclear Safety
19 Guseong-dong, Yuseong-gu, Daejeon 305-338 Korea
mjj@kins.re.kr

(Received January 12, 2004)

Abstract

This study investigates the fluidelastic instability characteristics of helical steam generator type tubes used in operating nuclear power plants. To obtain a natural frequency, corresponding mode shape, and participation factor, modal analyses using various conditions are performed for helical type tubes. Investigated are the effects of the number of turns, the number of supports, and the status of the inner fluid on the modal and fluidelastic instability characteristics of the tubes, which are expressed in terms of the natural frequency, the corresponding mode shape, and the stability ratio.

Key Words : fluidelastic instability, steam generator helical tubes, modal analyses, mode shape, participation factor, stability ratio

1. Introduction

Advanced nuclear power reactors are currently under development worldwide, and some innovative designs are now ready for construction. One advantage of these new reactors will be the easy implementation of advanced design concepts and technology. Drastic safety enhancement can now be achieved by adopting inherent safety characteristics and passive safety features. Economic benefits can also be reaped through plant system simplification and modularization, and through the reduction of construction time.

SMART (System-integrated Modular Advanced

Reactor), a small sized integral type PWR is an advanced type of reactor being developed in Korea. All the major primary components are contained in a single pressurized vessel. The in-vessel self-controlling pressurizer is one of the advanced design features. The system pressure is passively adjusted by partial pressure of steam and nitrogen gas filled in the pressurizer, in accordance with variations in pressure and temperature of the primary coolant. The control element drive mechanism has a very fine-step maneuvering capability, to compensate for the core reactivity change caused by fuel depletion during normal operation. The modular type once-through SG

(steam generator) has an innovative design feature, with helically coiled tubes to produce superheated steam at normal operating conditions.

There are twelve identical SG cassettes, which are located on the annulus formed by the reactor pressure vessel and the core support barrel. Each SG cassette is of once-through design, with a number of helically coiled tubes. The primary reactor coolant flows downward in the shell side of the SG tubes, while the secondary feedwater flows upward in the tube side.

The helical-type tubes adopted for SMART may have a totally different behavior from that of U-tubes, which are used in typical PWR [1, 2]. This necessitates a study on fluidelastic instability of the helically coiled tubes, including vibration characteristics, to assure the structural integrity of such tubes during normal operation.

For many years, the problem of steam generator tube rupture (SGTR) has been one of the most significant safety issues in operating nuclear power plants worldwide. This is because leakage due to SGTR has such serious implications, including the possible direct release of radioactive fission products to the environment and the loss of coolant.

Tube vibration excited by dynamic forces of external fluid flow in nuclear steam generators may either initiate such mechanical damages on intact tubes as fretting-wear and fatigue, which

may eventually result in severe tube failures, or may accelerate the growth of pre-existing flaws or cracks caused by stress corrosion in the tubes. Even less significant dynamic forces of external fluid flow exerting on a tube, which do not cause any actual damage to the intact tube, may lead to excessive vibration, either resulting in fatigue failure of the tube with pre-existing flaws (cracks) or growing flaws originally due to stress-corrosion, or resulting in failure of the tube due to fretting-wear. Therefore, with regard to nuclear safety it is very important to assess the potential for SG tube failures due to fluidelastic instability and to take the necessary preventive measures for minimizing the probability of SG tube failures in operating plants. A fluidelastic instability analysis, beginning with a modal analysis, can provide an assessment of the potential for such SG tube failures.

This study investigates the fluidelastic instability characteristics of helical steam generator tubes. Modal analyses are performed by finite element modeling of the tubes, using various conditions. This investigation examines the effects of the number of turns, the number of supports, and the status of inner fluid on the modal and fluidelastic instability characteristics of tubes, which are expressed in terms of the natural frequency, corresponding mode shape, and stability ratio.

Table 1. Geometric Description and Material Properties

Parameter (mm)	Type A	Type B	Type C	Type D	Type E
Wire diameter	10				
Wire thickness	15				
Coil diameter	422	338	282	198	142
Full height	1104	1150	1104	1104	1150
Number of turns	8	10	12	16	25

2. Analysis

2.1. Modal Analysis

Modal analyses using a commercial computer code ANSYS 7.0 [3] are performed to find the vibration characteristics of a tube. Several different kinds of finite element models are developed, according to the coil diameter, the full height, the number of turns (helix angle), and the number of support points (Table 1).

Finite element models are developed using the elastic straight pipe elements (PIPE16) for the helical tube and 3-D point-to-point contact elements (CONTAC52) between the support and the tube. PIPE16 is a uniaxial element capable of tension-compression, torsion, and bending. CONTAC52 represents two surfaces that may maintain or break physical contact, or may slide relative to each other. The element is capable of supporting only compression in the direction normal to the surfaces and shear (Coulomb friction) in the tangential direction. The finite element model consists of 1280 PIPE16 elements for the helical tube and 65 CONTAC52 elements between the support and the tube for 8 support points of Type A, as shown in Fig. 1.

The boundary conditions at the two ends of the tube are fixed. To simulate that the nodes of the tube at the support locations are free to move in the longitudinal direction, contact elements are used between the support and the corresponding tube locations, with the support node fixed.

The Block Lanczos method is used for the eigenvalue and eigenvector extractions to calculate 50 natural frequencies. It uses the Lanczos algorithm, where the Lanczos recursion is performed with a block of vectors. This method is as accurate as the subspace method, but faster. The Block Lanczos method is especially powerful when searching for eigenfrequencies in a given

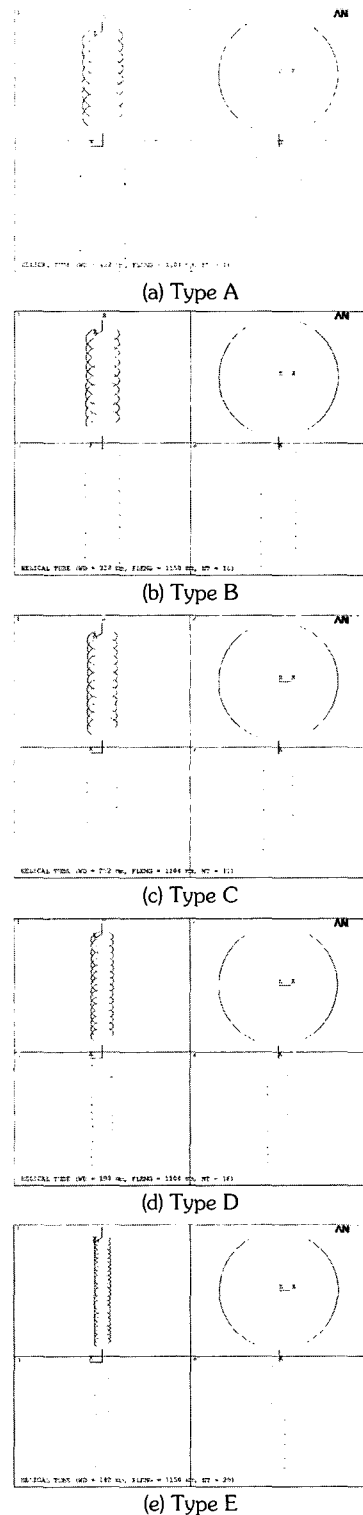


Fig. 1. Finite Element Models of Helical Tube

part of the eigenvalue spectrum of a given system. The convergence rate of the eigenfrequencies will be about the same when extracting modes in the midrange and in the higher end of the spectrum as when extracting the lowest modes.

2.2. Fluidelastic Instability Analysis

The critical velocity to initiate fluidelastic instability was formulated by Connors [4] for the simple case of a tube bank subjected to uniform cross flow over the entire length of the tubes. The formulation of fluidelastic instability proposed by Connors is a semi-empirical correlation, fitted by experimental data, and is expressed in terms of a dimensionless flow velocity called a reduced velocity $v_{c,n} / f_n d$ and a dimensionless mass-damping parameter $2\pi\zeta_t m_t / \rho d^2$ as follows:

$$\frac{v_{c,n}}{f_n d} = C \left(\frac{2\pi\zeta_t m_t}{\rho d^2} \right)^{0.5} \tag{1}$$

where $v_{c,n}$, C , f_n , ρ and d are the critical velocity of the n th free vibration mode, the fluidelastic instability coefficient (or the Connors' constant), the natural frequency of the n th mode, the shell-side fluid density, and the outer diameter of tube, respectively. Also, ζ_t and m_t are the total damping ratio and the total mass per unit length of the tube.

The total damping ratio in a two-phase flow is the sum of viscous damping, support damping, and two-phase damping, and these may be determined either from available measured data or by empirical expressions. Because of the great difficulty involved with conducting the relevant experiments, only limited data on the damping in two-phase flows are available at present. For water or wet steam, Au-Yang [5] recommended a mean damping ratio of 0.015 for a tightly supported tube and of 0.05 for a loosely supported tube.

The total effective mass of a tube surrounded by

a fluid consists of three components: mass of the tube material, mass of fluid in the tube, and added mass (or hydrodynamic mass) of fluid displaced by the tube. The third component of the effective tube mass is affected by the proximity of other tubes in the tube bundle, and it is bounded by the pitch pattern for maximum and minimum by the triangular and square pitch, respectively.

The fluidelastic instability coefficient C is a function of the tube arrangement and the ratio of tube pitch p over the outer diameter of tube d . Mean values for the onset of instability can be established by fitting a semi-empirical correlation to the experimental data [6]. For the entire mass-damping parameter range, a mean value of $C = 3.3$ was recommended by Pettigrew and Gorman [7] and by Paidoussis [8]. Also Yetisir and Pettigrew [9] used $C = 3$ for $p / d \geq 1.47$ and $C = 4.76 (p / d) / p + 0.76$ for $p / d < 1.47$ as a bounding design guideline.

The critical velocity $v_{c,n}$ is related to the gap velocity v_g between the tubes, which is based on the tube pitch and the diameter, as applied to the approach or free stream velocity v_∞ . The gap velocity in the fluid region is defined as

$$v_g = \frac{p}{p-d} v_\infty \tag{2}$$

For most practical shell-and-tube type heat exchangers, including SGs, the tube bundles comprise multi-span tubes and only partial portions of the tubes may be exposed to cross flow. The onset of fluidelastic instability of those multi-span tubes partially subjected to cross flow can be predicted by several approaches. It has been indicated that the equivalent velocity approach based on mode shapes is valid and is the simplest method to use [10]. Equation (1) was originally extended by Eisinger and Juliano [11], to be used in the equivalent velocity approach in the fluidelastic instability analysis for tubes partially subjected to cross flow.

The fluidelastic instability for tubes partially exposed to cross flow can be evaluated by the comparison of the critical velocity $v_{c,n}$ with the effective cross flow gap velocity v_{ge} , which is a uniform cross flow velocity equivalent to the actual non-uniform normal-to-tube cross flow gap velocity distribution along the tube length $v_g(x)$, where x denotes the distance along the tube with full length from the hot side tube end.

$v_{ge,n}$ is a variable dependent on the free vibration mode, as in the case of $v_{c,n}$. The value of $v_{ge,n}$ equivalent to $v_g(x)$ can be determined by weighting the n th mode shape as follows:

$$v_{ge,n}^2 = \frac{\frac{1}{\rho_o} \int_0^l \rho(x) v_g^2(x) \varphi_n^2(x) dx}{\frac{1}{m_o} \int_0^l m_i(x) \varphi_n^2(x) dx} \quad (3)$$

where $\varphi_n(x)$ is the n th mode shape function, $\rho(x)$, $m_i(x)$ are the shell-side fluid mass and total tube mass densities along the tube, and ρ_o , m_o are the corresponding average densities.

The stability ratio $R_{s,n}$ is defined by the ratio of $v_{ge,n}$ over $v_{c,n}$ as given by

$$R_{s,n} = \frac{v_{ge,n}}{v_{c,n}} \quad (4)$$

where $R_{s,n}$ indicates the stability ratio for the n th vibration mode. The maximum value among the stability ratios for all the vibration modes of a specified tube is used as the criteria to assess the potential instability of the tube. If the maximum value of a stability ratio R_s is smaller than unity, the tube is fluidelastically stable. Otherwise, it is unstable and its vibration amplitude becomes divergent rapidly as R_s increases beyond unity, which means that v_{ge} should be less than v_c for all modes during a normal operation.

3. Results and Discussion

Modal analyses for several kinds of finite

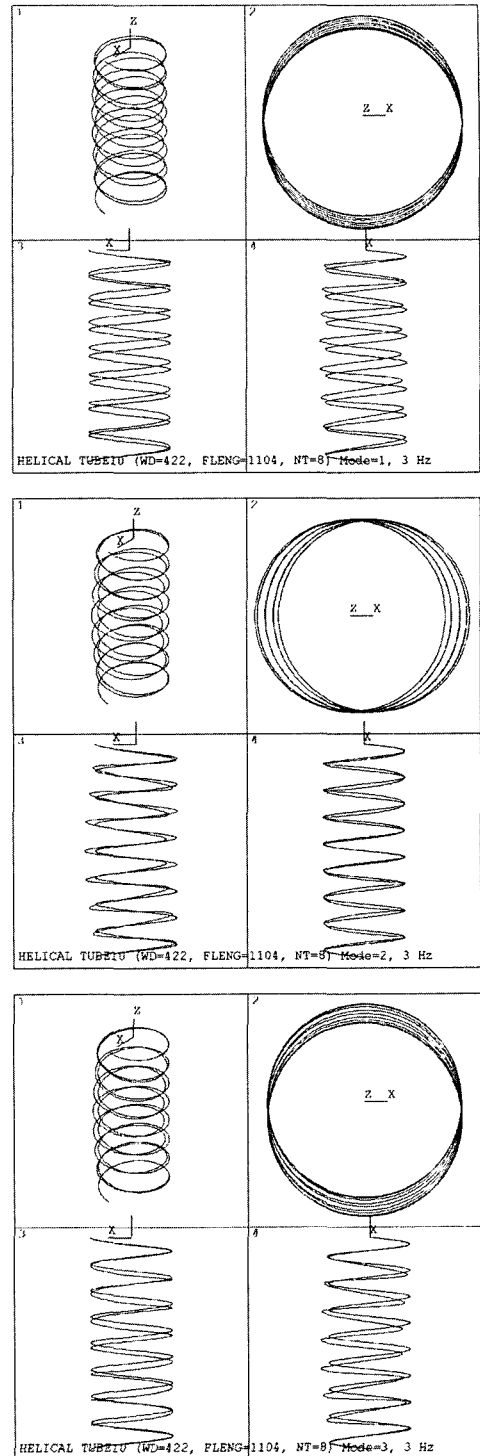


Fig. 2. Typical Mode Shapes of Helical Tube Without Supports

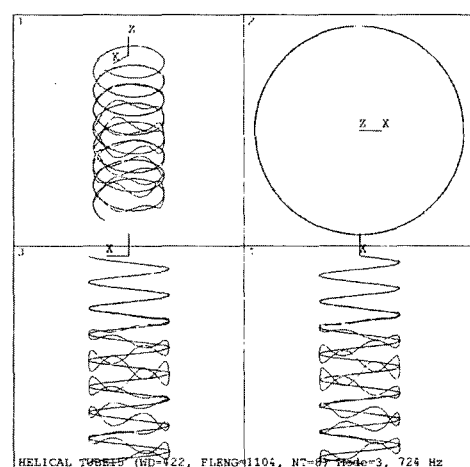
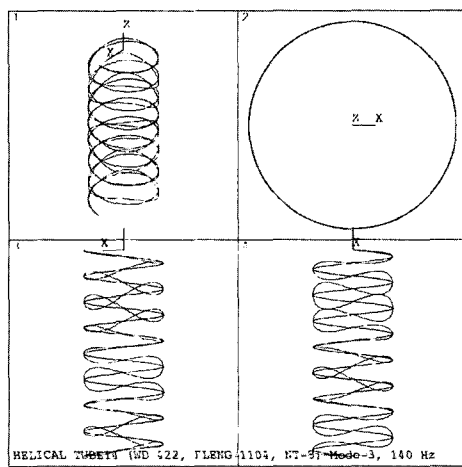
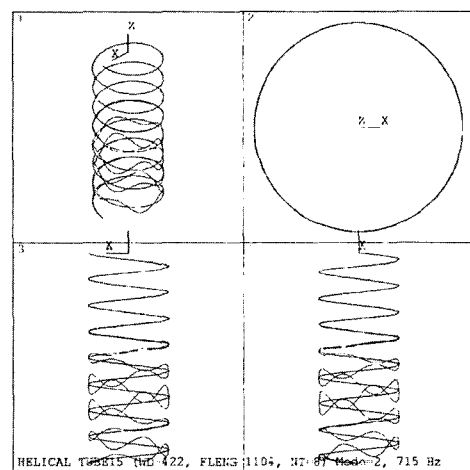
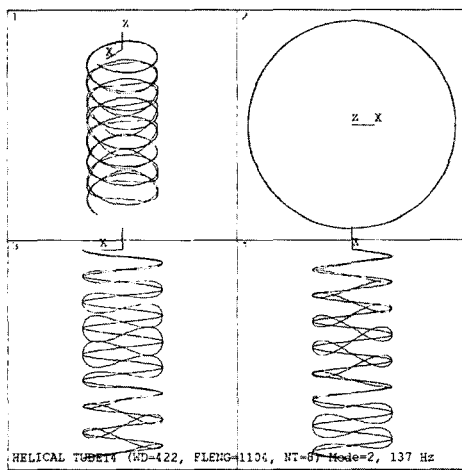
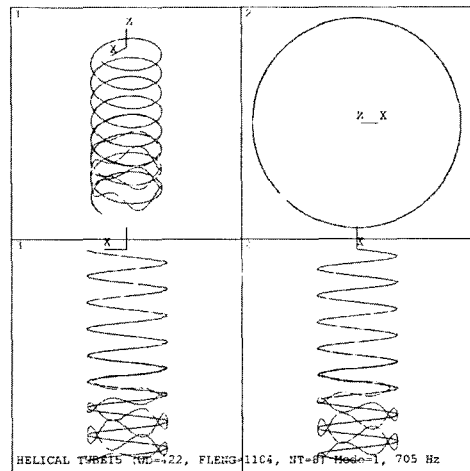
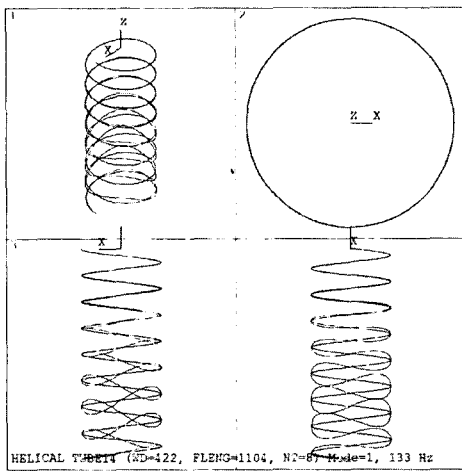


Fig. 3. Typical Mode Shapes of Helical Tube with 4 Supports

Fig. 4. Typical Mode Shapes of Helical Tube with 8 Supports

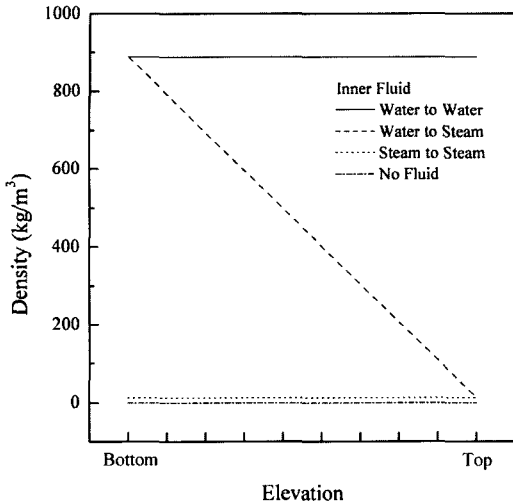


Fig. 5. Density Distribution of Inner Fluid Along the Tube

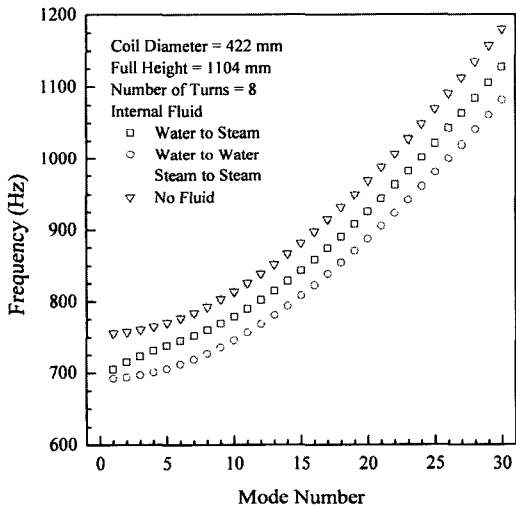


Fig. 6. Natural Frequency Variations w.r.t. the Status of Inner Fluid

element models are performed and typical mode shapes are shown in Figs. 2 through 4, which show that local modes, rather than global modes, appeared with the increasing number of supports.

The effective mass density distribution of the inner fluid along the entire tube is assumed to be water to steam, from the bottom to the top of the tube, as shown in Fig. 5, which was simplified

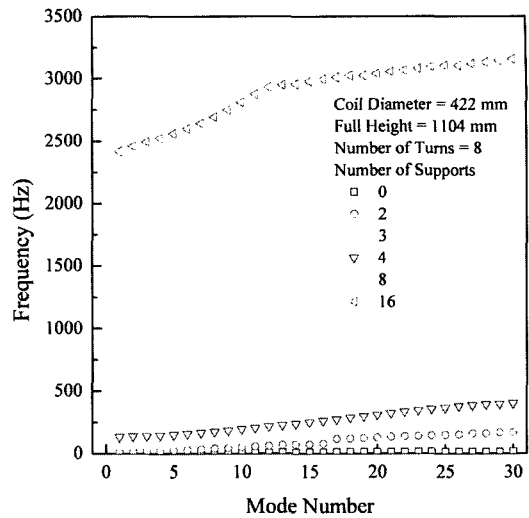


Fig. 7. Natural Frequency Variations w.r.t. the Number of Supports

from the thermal-hydraulic analysis [12]. This mass density is used to find the vibration characteristics. In addition, the effect of the inner fluid density is investigated by comparing the frequencies between three kinds of steam quality of the inner fluid: water to water, water to steam, and steam to steam. The natural frequency variations are shown in Fig. 6 with respect to the quality of the inner fluid. The resulting natural frequency comparisons between the qualities of the inner fluid indicate that the frequencies of the water-to-steam case are in the middle of those of the steam-to-steam case, which have almost the same values as that of a case with no inner fluid. As the inner fluid is superheated, the frequencies of the tube increase, providing a larger safety margin against instability. Therefore the inner fluid is assumed to be water-to-steam conservatively.

The support plays a major role in keeping the tube from moving freely in any direction. A tube without a support is too flexible and has very low frequencies of less than 27 Hz for the first 30 modes, resulting in critical problems of fluidelastic instability, because the stability ratio is inversely

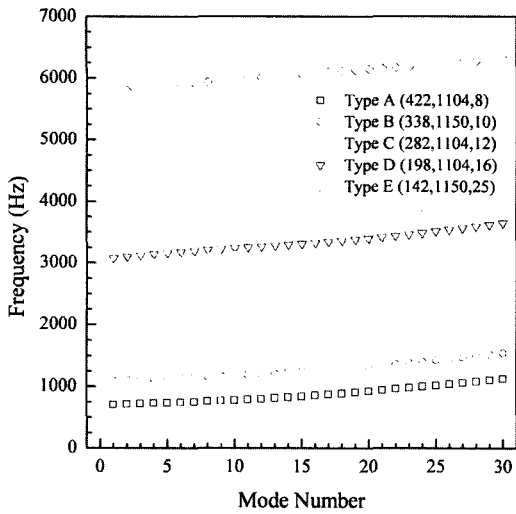


Fig. 8. Natural Frequency Variations w.r.t. Tube Type

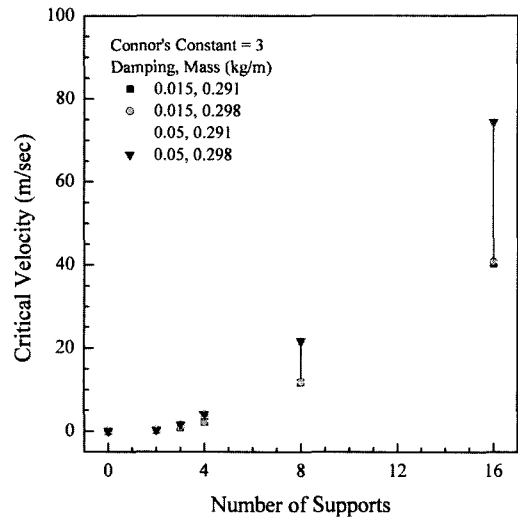


Fig. 9. Critical Velocity with Respect to the Number of Supports

proportional to the frequency, as shown in Eqs. (1) and (4). Several supports are installed in the circumferential direction, and the effect of supports on the frequencies is investigated by comparing the frequencies between those tubes with supports with those tubes without supports. Support points each turn are 2, 3, 4, 8 and 16 in the circumferential direction. Their natural frequency variations are shown in Fig. 7. The inclusion of supports increases the natural frequencies of the first mode significantly, from 2.68 Hz with zero support to 705.3 Hz with 8 supports. Therefore 8 supports for each turn are expected to be enough to avoid fluidelastic instability by providing high frequencies. In addition, fewer than 8 supports are recommended for tubes with a coil diameter of less than 422 mm, because it is complicated to install supports that need many welding points in a difficult working space.

Five different types of helical tubes, as shown in Table 1, are chosen to investigate the stability. Modal analyses are performed and their natural frequencies are shown in Fig. 8.

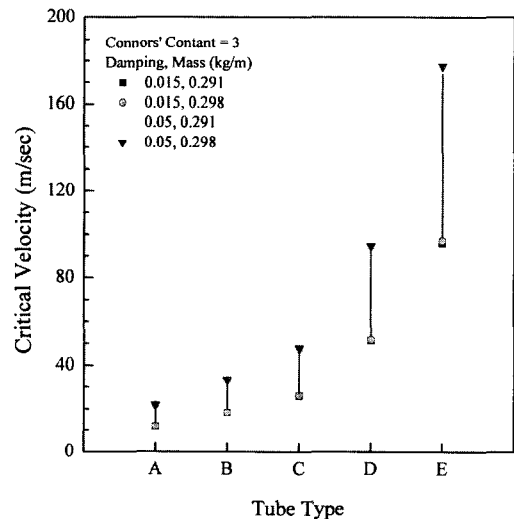


Fig. 10. Critical Velocity with Respect to Tube Type

The critical velocity for the first mode is calculated from Eq. (1) and is summarized in Figs. 9 and 10. For the tube to be fluidelastically stable, the stability ratio defined in Eq. (4) should be less than unity, which means that the gap velocity should be less than the critical velocity. Therefore, the allowable gap velocity is less than the minimum value of the critical velocity shown in

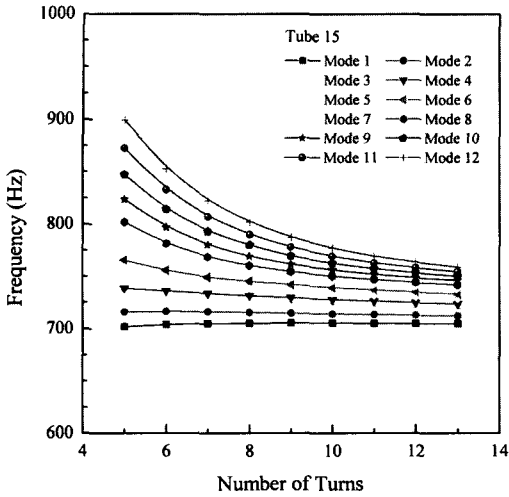


Fig. 11. Natural Frequency Variations w.r.t. the Number of Turns

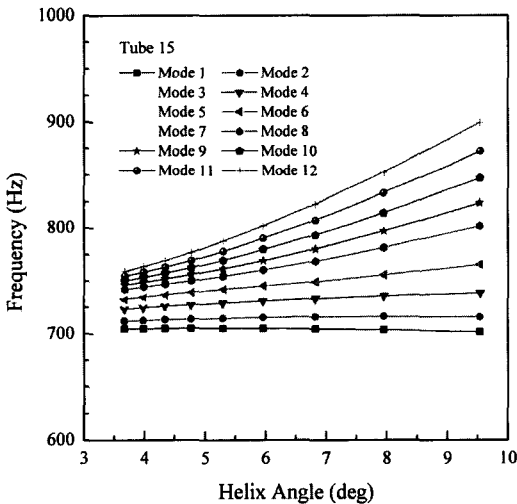


Fig. 12. Natural Frequency Variations w.r.t. Helix Angle

Figs. 9 and 10. As shown in Fig. 9, the allowable gap velocity is less than 1.0 m/sec for less than or equal to 3 supports and, therefore, more than 4 supports are recommended to avoid fluidelastic instability. In addition, Fig. 10 shows that Type A is the least desirable tube for fluidelastic stability.

The effect of damping on the critical velocity can be predicted from Eq. (1), where the critical

velocity is proportional to the square root of the total damping ratio. Therefore, a more accurate estimation of the total damping ratio needs to be made to predict fluidelastic instability.

The variation of frequencies versus the number of turns and the helix angle is given in Figs. 11 and 12, respectively. As the number of turns increases, or as the helix angle decreases for the same height, the total length of the tube increases and the stiffness of the system decreases when all the other properties are kept constant. The frequencies, except for the first several modes, are reduced, and this is particularly pronounced in the higher modes.

4. Conclusions

To investigate the vibration characteristics of a helical tube, modal analyses for various conditions, such as the status of inner fluid, the number of turns, and the number of supports were performed. The effects of the modal characteristics on fluidelastic instability were addressed. Based on the analyses performed, the optimal number of supports is found to be between 4 and 8 for each turn, to avoid fluidelastic instability. Type A is the least desirable tube for fluidelastic stability. In addition, with an increasing number of turns, the natural frequencies of the higher modes decrease significantly with the first several modes maintained almost the same.

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