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The Effect of Internal Flow on Vortex-Induced Vibration of Marine Riser
Riser의 內部流體 흐름이 소용돌이로 인한 Riser 動的反應에 미치는 影響

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Abstract□Combining Iwan-Blevin's model into the approximated form of the nonlinear model derived for the dynamic analysis of the riser system with the inclusion of internal flow, current-vortex model is developed to investigate the effect of internal flow on vortex-induced vibration due to inline current. The riser system includes a steadly flow inside the pipe which is modeled as an extensible or inextensible tubular beam. Galerkin's finite element approximation are implemented to derive the matrix equation of equilibrium for the finite element system. The investigations of the effect of internal flow on vibration due to inline current are performed according to the change of various parameters such as top tension, internal flow velocity, current velocity, and so on. It is found that the effect of internal flow on vibration due to vortex shedding can be controlled by the increase of top tension. However, careful consideration has to be given, in design point, in order to avoid the resonance band occuring near vortex shedding frequency, particularly for the long riser.

要 旨: Riser 內部의 流體흐름을 包含해서 動的解析을 하기위해 誘導된 非線形 모델의 近似化한 形態에 Iwan-Blevin의 모델을 結合함으로써 흐름-소용돌이 모델이 開發되며 開發된 數學的모델을 解析함으로써 面內 潮流흐름에 따라 形成된 面外 過流로 인한 riser의 振動에 內部 流體흐름이 미치는 影響에 관해 調査하였다. Riser 內部 流體흐름은 一定한 流速分析을 가진 正常流로 假定하며 riser官은 신축성 혹은 비신축성 관상빔으로 看做된다. 誘導된 모델에 Galerkin의 有限要素近以法을 적용함으로써 數値解析을 위한 모델을 開發하였다. 官內部 流體 흐름이 riser의 소용돌이로 인한 振動特性에 미치는 影響을 上部 引長力, 內部流體 흐름 혹은 潮流速度 등과 같은 여러 影響要因 등을 變化시키면서 調査하였다. 數値解析 결과 內部流體 흐름으로 인한 影響을 줄이기 위하여 riser의 上部에 引長力을 riser의 許谷內力 限度내에서 增加시키는 方法이 있으나 vortex shedding 으로 인해 形成되는 resonance band를 피하기 위해 設計 觀點에서 細心한 注意가 要求된다. 특히 길이가 긴 riser에 대한 細心한 注意가 要求된다.

1. INTRODUCTION

Stuctures shed vortices in a flow. The vortex street wakes tend to be very similar regardless of the geometry of the structure. As the vortices are shed from fist one side and then the other, surface pressures are imposed on the structure. The oscillating pressures cause elastic structures to vibrate and generate vortex-induced forces. The vibration induced in elastic structures by vortex shedding is of practical importance because of its potentially destructive effect on pipelines of marine riser. Vortex-induced forces may excite the riser in its normal modes of transverse vibration. When the vortex shedding frequency-approaches the natural frequency of a marine riser, the cylinder takes control of the shedding process causing to be shed at a frequency close to its natural frequencies. This phenomenon is called vortex shedding "lock-in" or syncronization. Under "lock-in" coditions, large resonant oscillations occur and

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lift forces are amplified due to increased vortex strength and spanwise correlation along the cylinder. This is attributed to a substantial increase in in-line drag under these conditions. Large responses in both directions give rise to oscillatory stresses. If these stresses persist, significant fatigue damage may occur.

The investigations of the vortex-induced vibration of structures are primarily conducted in a laboratory environment. Observations and measurements are made on both rigid (Feng, 1968; Sarpkaya, 1977) and flexible (Ramberg and Griffin, 1974; Every *et al.*, 1982) cylinder vibrating transvrsely in uniform cross-flows. Emphasis is placed on the characteristics of the shedding process, particularly in the region of lock-in. In the field, excitation of many real structures near a structural natural frequency has also exhibited a locked-in response. However, Alexander (1981) concludes on the basis of field observations that a locked-in condition or standing wave profile apparently does not occur in a series of tests on long wire towed in the ocean. For vortex suppression, Grant (1977) attaches a fairing to the riser. Whitney and Chung (1981) makes an analysis of vortex-induced riser vibration in a constant current by applying modal method with an empirical lift coefficient. Iwan and Blevins (1974) proposes the two dimensional, coupled wake oscillator model. In this model, the effects on both uniform and depth-dependent current velocity are investigated. Using coupled wake oscillator model proposed by Iwan and Blevins, Nordgren (1982) treats vortex-induced vibrations numerically. In his paper, a hybrid finite element method is introduced with deflection and moment as basic variables. Variation of tension with time is accounted for. Rajabi *et al.* (1984) describe an analytical model to predict the dynamic response of a riser in regular waves or in current to vortex shedding-induced forces. Patrikalakis and Chryssostomidis (1985, 1986) develops a method for the theoretical prediction of the lift responses of a flexible cylinder in a unidirectional constant current. The theoretical prediction is based on information derived from experimental results involving rigid cylinders forced to oscillate sinusoidally to uniform stream. The prediction of lock-in vibration on flexible cylinders in a sheared flow is made theoretically by Vandiver (1985). Later, Vandiver and Jong (1987) investigates the coupled relationship between in-line and cross-flow vibration. Kim *et al.* (1986) investigates the vortex-induced vibration response of long cables subjected to vertically sheared flow in two field experiments. Iwan and Jones implements a solution based on the traveling wave nature of the response in order to avoid the limitations associated with the model solution under the assumption of total spanwise correlation in the fluid forces or responses.

Although the vortex-induced vibrations have been studied continuously and widely up to now, the system with the inclusion of internal flow inside pipe is not considered. In other words, the effect of internal flow on vortex-induced vibrations are not investigated in their works. The question still remains to solve how the internal flow affects the vortex-induced vibration responses. Chen (1992) handles the question only in the part of his paper. Hong (1994) investigates the effect of internal flow on vortex-induced vibrations by using oscillator model.

In summary, the objectives of the present study are 1) to develop a mathematical model for the analysis of vortex-induced vibration with the inclusion of internal flow and 2) to examine the effect of internal flow on vortex-induced vibrations. Iwan-Blevin's model (1990), which is one of the self-excited oscillator models, is implemented as a vortex shedding model and Galerkin's usual finite element processes are adopted to solve mathematical model.

2. GOVERNING EQUATION

The riser system can be modeled as a long tubular beam connecting the drilling platform with the wellhead at the seabed and is composed of rigid pipes. A tensioning system is installed on the drilling platform and applies a tension at the top of the riser.

This tension provides part of the support required to keep the riser tight and prevent buckling or collapse. The ring space between the riser and the drill string circulates the drilling mud. It exerts on the riser static pressure force, Coriolis and centrifugal

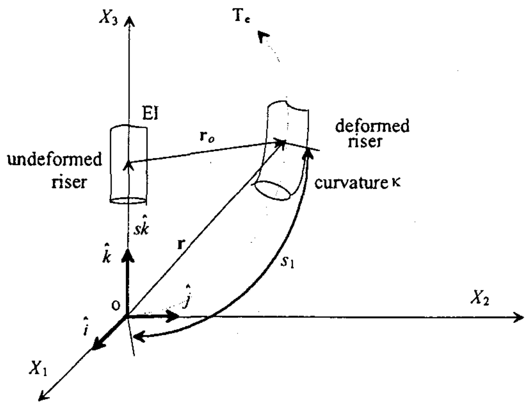

Fig. 1. Position vector at a poin on the riser centerline.

forces due to the riser's local rotation, and vertical and torsional frictional forces. And also, the source of external forces exerted on the riser are the ocean currents.

2.1 Approximate Governing Equations and B.C.s

The main aspects of the curved geometry of the system are depicted in Fig. 1. The space curve is identified with the central axis of the riser in the deformed state, with the position vector, as shown in Fig. 1, represented in the following equation.

$$\mathbf{r}=\mathbf{r}_o+s\hat{k}=x_1\hat{i}+x_2\hat{j}+(s+x_3)\hat{k} \qquad (1)$$

where $\mathbf{r}_o$ is the deformation vector at any point on the riser in the undeformed state. In the riser problem in an intermediate water depth, the deformed arclength S_1 can be approximately identified with the undeformed arclength S for inextensible riser and the nonlinear vector equation can be used except the fact that prime denotes differentiation with respect to S (Nordgren, R.P., 1974). Further, if we notify the approximation used to obtain the fluid acceleration due to internal flow (Hong, N.S., 1994), we can have the following equation:

$$m_t\ddot{\mathbf{r}}+2m_fV_I\dot{\mathbf{r}}'+m_fV_I^2\mathbf{r}''+(\mathrm{EI}\mathbf{r}'')'' \\ =[(T_e-\mathrm{EI}\kappa^2)\mathbf{r}']'-\mathrm{H}(\mathbf{r}'\times\mathbf{r}'')'=\mathbf{q} \qquad (2)$$

where prime denotes the differentiation with respect to the undeformed arclength s.

For the extensible riser, the actual tension is given by Hook's law as

$$\mathrm{T}=\mathrm{EA}\varepsilon_t,\ \varepsilon_t=(\mathbf{r}'\cdot\mathbf{r}')^{1/2}-1 \qquad (3)$$

For nearly vertical riser, we introduce a rectangular Cartesian coordinate system with the x_3 axis vertically straigth upward. And then, with small deflection and rotation, i.e., x_1', $x_2'\ll1$ and $s+x_3\approx s$, the curvature and the strain along centerline are given by

$$\kappa=[\{x_1''(1+x_1'^2)^{-3/2}\}^2+\{x_2''(1+x_2'^2)^{-3/2}\}^2]^{1/2} \\ \approx(x_1''^2+x_2''^2)^{1/2} \qquad (4)$$

$$\varepsilon_t=x_3'+(x_1'^2+x_2'^2)/2 \qquad (5)$$

In the above equation, the nonlinear terms in bracket have been retained in order to include the effect of lateral deflection on tensions.

Restricting attention to risers whose deformation lies wholly in a single plane and dropping nonlinear terms, the vector equation (2) reduces to

$$m_t\ddot{x}_1+2m_fV_I\dot{x}_1'+m_fV_I^2x_1''+(\mathrm{EI}x_1'')'' \\ =(T_ex_1')'+\mathrm{H}x_2'''=q_1 \qquad (6)$$

$$m_t\ddot{x}_2+2m_fV_I\dot{x}_2'+m_fV_I^2x_2''+(\mathrm{EI}x_2'')'' \\ -(T_ex_2')'-\mathrm{H}x_1'''=q_2 \qquad (7)$$

$$m_t\ddot{x}_3-T_3'=q_3 \qquad (8)$$

Most riser problems in intermediate water depth, longitudinal vibration is unimportant and we may neglect the longitudinal inertia term in Eq. (8). Further, it is convenient to include the hydrostatic effects of internal and external fluid pressures by defining effective weight per unit length w and effective tension T_e as

$$\mathrm{w}=\mathrm{w}_r+\gamma_iA_i-\gamma_oA_o \qquad (9)$$

$$T_e=T-p_iA_i+p_oA_o \qquad (10)$$

where w_r=riser weight per unit lenght γ_i, γ_o=specific weight of internal and external fluid, p_i, p_o=internal and external static pressure of riser, and A_i, A_o=internal and external area of riser.

Equation (9) leads to $q_3=-\mathrm{w}$. Equation (8) can then be integrated to give

$$T_e=\mathrm{TTR}-\int_0^l \mathrm{w}\ ds \qquad (11)$$

or

$$T_e = TTB + \int_0^l w \; ds \tag{12}$$

where TTR and TTB are, respectively, the top tension and the bottom reaction tension.

Thus, the effective tension is independent of the horizontal deflections and then, with no torsion, Eqs. (6) and (7) becomes

$$m_t\ddot{x}_1 + 2m_f V_i\dot{x}_1' + m_f V_i^2 x_1'' + (EIx_1'')'' - wx_1' - T_e x_1'' = q_1 \tag{13}$$

$$m_t\ddot{x}_2 + 2m_f V_i\dot{x}_2' + m_f V_i^2 x_2'' + (EIx_2'')'' - wx_2' - T_e x_2'' = q_2 \tag{14}$$

Equations (13) and (14) represent the linear model and can be also derived by applying Hamilton's principle (Chen, B.C.M., 1992).

The boundary conditions associated with linear model can be found by modifying the direction cosines of nonlinear boundary condition (Hong, N.S., 1994) into 0, 0 and 1, and are given as following:

$$\begin{aligned} F_{1u} &= -K_1 x_{1u} \\ F_{2u} &= -K_2 x_{2u} \\ F_{3u} &= TF - K_3 x_{3u} + TTR \end{aligned} \tag{15}$$

where

$$TF \approx \rho_w g\pi D_o^2(S_w - z)/4 - \rho_m g\pi D_i^2(S_m - z)/4 \tag{16}$$

the subscript u indicates the upper end of the riser, TTR is the tension applied at top of the riser by the tensioning system and K_1, K_2, K_3 are spring constants supplied by the restoring boundary force. Equations (15) shows that that the fluidic tension at the top of the riser always acts in the tangential diection. The nonlinearity of the boundary conditions disappear in linear model because the three direction cosines become 0, 0 and 1, respectively.

2.2 Current-Vortex Model

In this section, the mathematical model of vortex-induced vibration with internal flow is presented using the two dimensional coupled wake oscillator model proposed by Iwan and Blevins (1974) for the Reynolds number of 10^3 to 10^5. There are many vortex excitation model such as harmonic model, wake oscillator model, etc. Of those models, Iwan-Blevin's model has an advantage as a numerical model. This model has a nature of self-excited vortex shedding which means that the fluid behavior may be modeled by a simple, nonlinear, self-excited oscillator. Attention is confined to plane vibrations ($x_2=0$) transverse to a steady current in direction x_2. In-line vibration is treated in the usual way and no coupling is considered in this model (see Fig. 2).

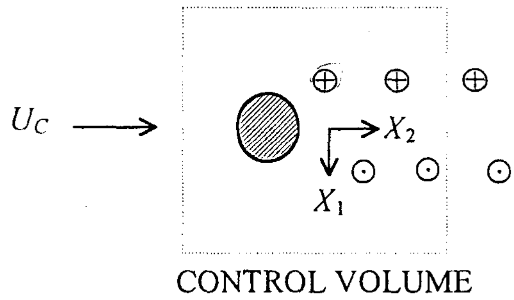

Fig. 2. Control volume containing a cylinder shedding vortices.

First, a so-called hidden flow variable w is defined as a fluid motion transverse to a steady current and it, in turn, produces transverse force to cylinder. This transverse force, which induces the motion of the cylinder is a forcing component in the oscillator equation. The net force per unit length on cylinder is evaluated using the momentum equation for the control volume containing cylinder as shown in Fig. 2 and is given by

$$q_1 = a_4\rho_w D_o u_c(\dot{w} - \dot{x}_1) \tag{17}$$

and the fluid oscillator equation is

$$\ddot{w} + \omega_s^2 w = (a_1' - a_4')u_c\dot{w}/D_o - a_2'\dot{w}^3/(u_c D_o) + a_4' u_c\dot{x}_1/D_o \tag{18}$$

where ω_s is the vortex shedding frequency given by $\omega_s = 2\pi S u_c/D_o$ and Strouhal number S is taken as equal to 0.20.

On the basis of the results of various vortex experiments for subcritical Reynolds numbers, Iwan and Blevins took

$$\begin{aligned} &a_1' = a_1/a_o,\ a_2' = a_2/a_o,\ a_4' = a_4/a_o \\ &a_o = 0.48,\ a_1 = 0.44,\ a_2 = 0.20,\ a_4 = 0.38 \end{aligned} \tag{19}$$

Later, Blevins considered the spanwise coupling along the riser because portions of the riser span are not in resonance for nonuniform flow (1990). In order to apply the model to nonuniform flow, the span of the riser is divided into segments that are within the resonance band the remainder, which is outside the resonance band.

Define a parameter $p(x_3)$ that defines the spanwise region of the resonance band:

$$p(x_3)=\begin{cases} 1, & \text{if lock-in band, } (\alpha f_s<f_i<\beta f_s) \\ 0, & \text{if no lock-in} \end{cases} \tag{20}$$

where α and β specify the lock-in band and $f_s=2\pi/\omega_s$ is the stationary cylinder vortex shedding frequency. Typically $\alpha=0.6$, $\beta=1.4$ lightly damped, large amplitude motion. The parameter $p(x_3)$ will step between 0 and 1 along the span, depending on the current profile.

The evaluation of the net force on cylinder under the consideration of the spanwise coupling is obtained from the modification of Eq. (17) and is given by

$$q_1=a_4\rho_w D_o u_c(\dot{w}-\dot{x}_1)p(x_3)-\frac{1}{2}\rho_w u_c D_o C_D\dot{x}_1(1-p(x_3)) \tag{21}$$

Since the model derived by Blevin and Iwan is based on the assumption of small amplitude, this model is applied to the approximated equation of motion (13), which gives

$$m_t\ddot{x}_1+2m_fV_I\dot{x}_1'+m_fV_I^2x_1''+(EIx_1'')''-wx_1'-T_ex_1'' = a_4\rho_w D_o u_c(\dot{w}-\dot{x}_1)p(x_3)-\frac{1}{2}\rho_w D_o C_D\dot{x}_1(1-p(x_3)) \tag{22}$$

From above equation, we can recognize that the riser system with internal flow will respond resonantly to vortex shedding along the part of its span in the lock-in band and the system oscillation will be damped by the external fluid out of that part.

3. FINITE ELEMENT MODEL

The governing equations derived under the assumption of small deformation, that is, transverse vibration and in-line dynamic response of current-vortex model are modeled using finite element method. The usual finite element procedure is applied.

The mathematical model of current induced vibration can be obtained simply by combining Eq. (14), (18) and (22):

$$m_t\ddot{x}_1+2m_fV_I\dot{x}_1'+m_fV_I^2x_1''+(EIx_1'')''-wx_1'-T_ex_1'' = a_4\rho_w D_o u_c(\dot{w}-\dot{x}_1)p(x_3)-\frac{1}{2}\rho_w D_o C_D\dot{x}_1(1-p(x_3)) \tag{23}$$

$$\ddot{w}+\omega_s^2 w=(a_1'-a_4')u_c\dot{w}/D_o-a_2'\dot{w}^3/(u_cD_o)+a_4'u_c\dot{x}_1/D_o \tag{24}$$

$$m_t\ddot{x}_2+2m_fV_I\dot{x}_2'+m_fV_I^2x_2''+(EIx_2'')''-wx_2'-T_ex_2'' = q_2 \tag{25}$$

Those three equations represent the transverse vortex induced vibration, the fluid oscillator equation about the hidden flow variable w, and the riser vibration in the current direction. The forcing term q_2 in the right side of Eq. (25) can be derived by only taking the drag term due to current and given as,

$$q_2=-\frac{1}{2}\rho_w C_D D_o|\dot{x}_2-u_c|(\dot{x}_2-u_c) \tag{26}$$

The weak form of the equations are produced in multiplying the residual by a sufficiently smooth test function, integrating the product by parts, and equating the result to zero. The details in derivation are not given and only the results are written here as followings:

$$\int_0^l[m_t\ddot{x}_1\bar{x}_1+2m_fV_I\dot{x}_1'\bar{x}_1-m_fV_I^2x_1'\bar{x}_1'+(EIx_1'')\bar{x}_1''+(T_ex_1')\bar{x}_1'-a_4\rho_wD_ou_cp(x_3)\dot{w}\bar{x}_1+\{a_4\rho_2D_ou_cp(x_3)+\rho_wu_cD_oC_D(1-p(x_3))\}\dot{x}_1\bar{x}_1]dx_3+m_fV_I^2x_1'\bar{x}_1|_0^l+\{-(T_ex_1')\bar{x}_1+(EIx_1'')'\bar{x}_1-(EIx_1'')\bar{x}_1'\}|_0^l=0 \tag{27}$$

$$\int_0^l\{\ddot{w}\bar{w}+(a_4'-a_1')(u_c/D_o)\dot{w}\bar{w}+\omega_s^2w\bar{w}+(a_2'/(u_cD_o))\dot{w}^3\bar{w}-a_4'u_c\dot{x}_1/D_o\bar{w}\}dx_3=0 \tag{28}$$

$$\int_0^l[m_t\ddot{x}_2\bar{x}_2+2m_fV_I\dot{x}_2'\bar{x}_2-m_fV_I^2x_2'\bar{x}_2'+(EIx_2'')\bar{x}_2''+(T_ex_2')\bar{x}_2'-\frac{1}{2}\rho_wC_DD_o|u_c-\dot{x}_2|(u_c-\dot{x}_2)\bar{x}_2]dx_3+\{m_fV_I^2x_2'\bar{x}_2-(T_ex_1')\bar{x}_1+(EIx_1'')'\bar{x}_1-(EIx_1'')\bar{x}_1'\}|_0^l=0 \tag{29}$$

Implementation of Hermite polynomials as basis functions yields the matrix dynamic equilibrium equations constructed in terms of the unknown deformations at node. For the construction of the matrix equation, the usual processes are applied and the resulting matrix equations are:

$$[\hat{M}_{ij}]\{\ddot{x}_{1j}(t)\}+[\hat{C}_{ij}+\hat{C}_{fij}]\{\dot{x}_{1j}(t)\}+[\hat{T}_{ij}-\hat{C}_{eij}+\hat{K}_{ij}]\{x_{1j}(t)\} =[\hat{F}_{ij}]\{\dot{w}_j(t)\} \quad (30)$$

$$[\overline{M}_{ij}]\{\ddot{w}_j(t)\}+[\overline{C}_{ij}]\{\dot{w}_j(t)\}+[\overline{K}_{ij}]\{w_j(t)\}=[\overline{F}_{ij}]\{\dot{x}_{1j}\}-\{\overline{f}_j\} \quad (31)$$

$$[\tilde{M}_{ij}]\{\ddot{x}_{2j}(t)\}+[\tilde{C}_{ij}]\{\dot{x}_{2j}(t)\}+[\tilde{T}_{ij}-\tilde{C}_{eij}+\tilde{K}_{ij}]\{x_{2j}(t)\}=\{\tilde{f}_j\} \quad (32)$$

where,

$$\hat{M}_{ij}=\tilde{M}_{ij}=\int_{\Omega e} m_t N_i N_j dx_3,\ \hat{C}_{ij}=\tilde{C}_{ij}=\int_{\Omega e} 2m_f V_I N_i N_j' dx_3$$

$$\hat{C}_{fij}=\int_{\Omega e}\{a_4'\rho_w D_o u_c p(x_3)+\frac{1}{2}\rho_2 u_c D_o C_D (1-p(x_3))\}N_i N_j dx_3$$

$$\hat{T}_{ij}=\tilde{T}_{ij}=\int_{\Omega e} T_e N_i' N_j' dx_3,\ \hat{C}_{eij}=\tilde{C}_{eij}=\int_{\Omega e} m_f V_I^2 N_i' N_j' dx_3$$

$$\hat{K}_{ij}=\tilde{K}_{ij}=\int_{\Omega e} EI N_i'' N_j'' dx_3,$$

$$\hat{F}_{ij}=\int_{\Omega e} a_4'\rho_w D_o u_c p(x_3) N_i N_j dx_3$$

$$\tilde{f}_j=\int_{\Omega e}\frac{1}{2}\rho_w D_o C_D |u_c-\dot{x}_{2h}|(u_c-\dot{x}_{2h}) N_j dx_3$$

$$\overline{M}_{ij}=\int_{\Omega e} N_i N_j dx_3,\ \overline{C}_{ij}=\int_{\Omega e}(a_4'-a_1')(u_c/D_o) N_i N_j dx_3$$

$$\overline{K}_{ij}=\int_{\Omega e}\omega_s^2 N_i N_j dx_3,\ \overline{F}_{ij}=\int_{\Omega e} a_4'(u_c/D_o) N_i N_j dx_3$$

$$\overline{f}_j=\int_{\Omega e}\{a_2'/(u_c D_o)\}\dot{w}_h^3 N_j dx_3 \quad (33)$$

Eqs. (30) and (31) represent the vortex induced vibration which characterizes self-excited oscillator. Once the forcing due to well-formed vorticities acts on riser, the velocity of riser itself acts again to fluid particle as a forcing. Eq. (32) presents in-line vibration due to current. In those equations, the system matrix $\hat{M}_{ij}$ and $\tilde{M}_{ij}$ represent the system mass, $\hat{C}_{ij}$ and $\tilde{C}_{ij}$ Coriolis force terms, $\hat{C}_{fij}$ and $\overline{C}_{ij}$ fluid damping terms, $\hat{T}_{ij}$ and $\tilde{T}_{ij}$ stiffness terms due to effective tension, $\hat{C}_{eij}$ and $\tilde{C}_{eij}$ stiffness terms due to centrifugal force, $\hat{K}_{ij}$ and $\tilde{K}_{ij}$ stiffness terms due to bending of pipe, $\hat{F}_{ij}$ and $\overline{F}_{ij}$ self-excited forcing terms due to the vortex shedding generated by inline current, and $\overline{M}_{ij}$ and $\overline{K}_{ij}$ represent mass and stiffness term of fluid oscillator respectively. And also, the system vector $\tilde{f}_j$ represents the nonlinear drag damping due to current and $\overline{f}_j$ represents the nonlinear forcing term in fluid oscillator equation. The matrices representing Coriolis force term, which cause the distortion of the response of a riser, are skew symmetric, while the others are symmetric.

Form the inspection of coefficient matrices and vectors in (33), it may be recognized that the drag force in inline vibration equation and the fluid force in fluid oscillator equation have nonlinear term. In other words, there is no highly nonlinear terms in the equation to be solved, so that simple iteration can be applied for the convergence of solution. Thus, the application of Newmark method combined with the simple iteration is enough to solve the system.

4. THE EFFECT OF INTERNAL FLOW ON VORTEX INDUCED VIBRATION

Vortex-induced vibration due to inline current has been formulated by modifying Iwan-Blevin's model to include the effect of internal flow and a computer program CO DEV has been developed to solve it. The transverse vibration due to vortex shedding with the inclusion of internal flow is estimated and the effect of internal flow on the riser vibration is investigated.

The time simulation of the displacement at the middle node of a riser is presented in Fig. 3. The presence of internal flow in this case, which has a very high top tension, causes a slight increase in amplitude and a slight shift in phase. Figure 4 presents the trajectory of displacement at this middle node according to the lapse of time. The trajectory shape is modified more in the transverse direction than in the inline direction. This is partly due to the high tension applied at top of riser. As top tension eases, the effect of internal flow on the riser responses, particularly on inline displacement, should become more pronounced.

The optimal riser tension can be determined by minimizing the combined stress of the axial stress resulting from internal tension and the bending stress from lateral deflection. Thus, although high top tension tends to reduce the riser displacement and the associated bending stress, there is a limit to its

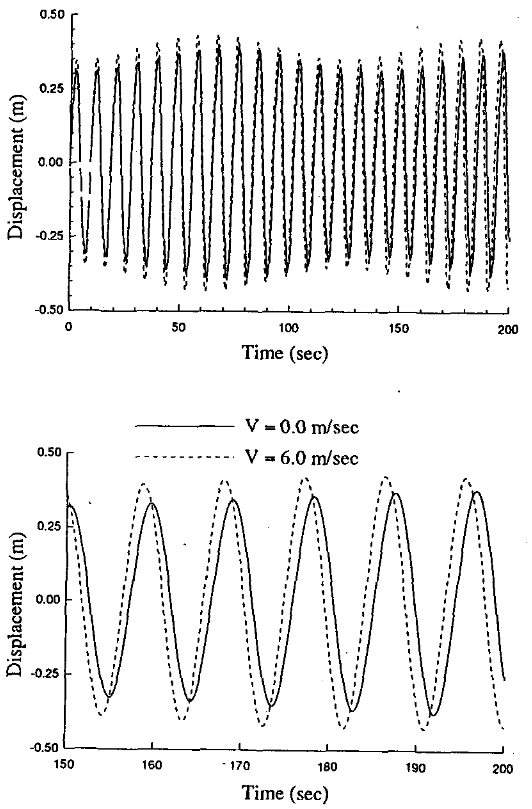

Fig. 3. The effect of internal flow on displacement at the middle of 152 m riser (current velocity U=0.32 m/sec, top tension=1550 kN, TR=1.2, w=3.86 kN/m).

application beause the increase in tensile stress may cut the benefit on the reduction in bending stress. Since direct stress increases with water depth, top tension has to be eased for deep water application. It is known from previous investigations that the effect of internal flow on riser dynamics is not significant under high top tension. However, the question is whether the internal flow effects should be considered for deep water case for the reason given above.

For comparison purpose, the same riser in a 152 m of water depth as used in previous examples but with very low top tension is used here. Computations are then performed for three types of tension (high, middle, and low), each with three internal velocities (null, high, and very high). Figure 5 shows the effect of internal flow on maximum displace-

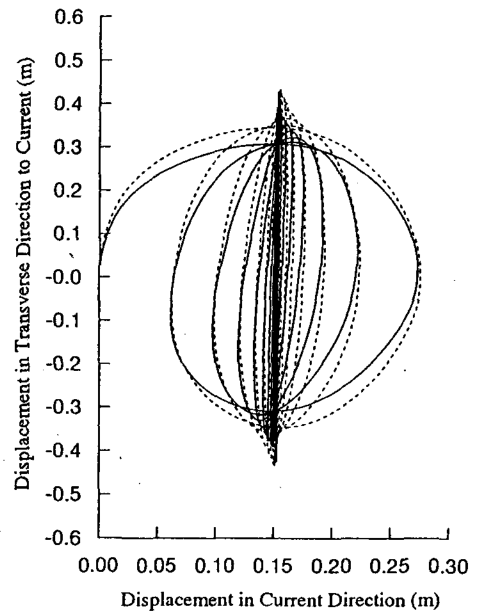

Fig. 4. The effect of internal flow on the trajectory at the middle of 152 m riser according to the lapse of time (current velocity U=0.32 m/sec, top tension=1550 kN, the other properties are the same as example of Fig. 3).

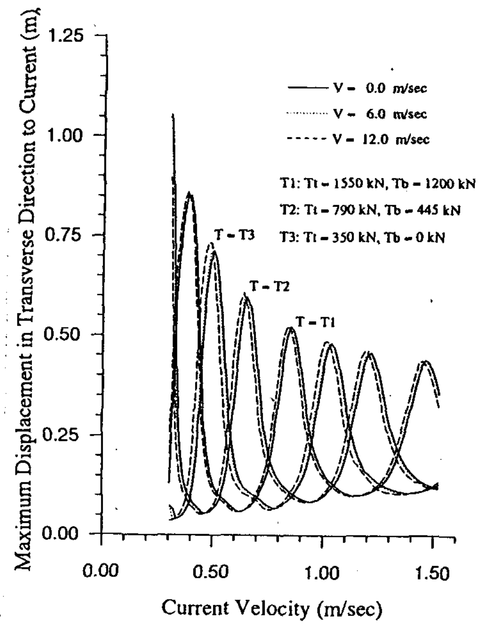

Fig. 5. The effect of internal flow on maximum displacement vortex direction for 152 m riser. Riser properties except top tension are the same as example of Figure 3.

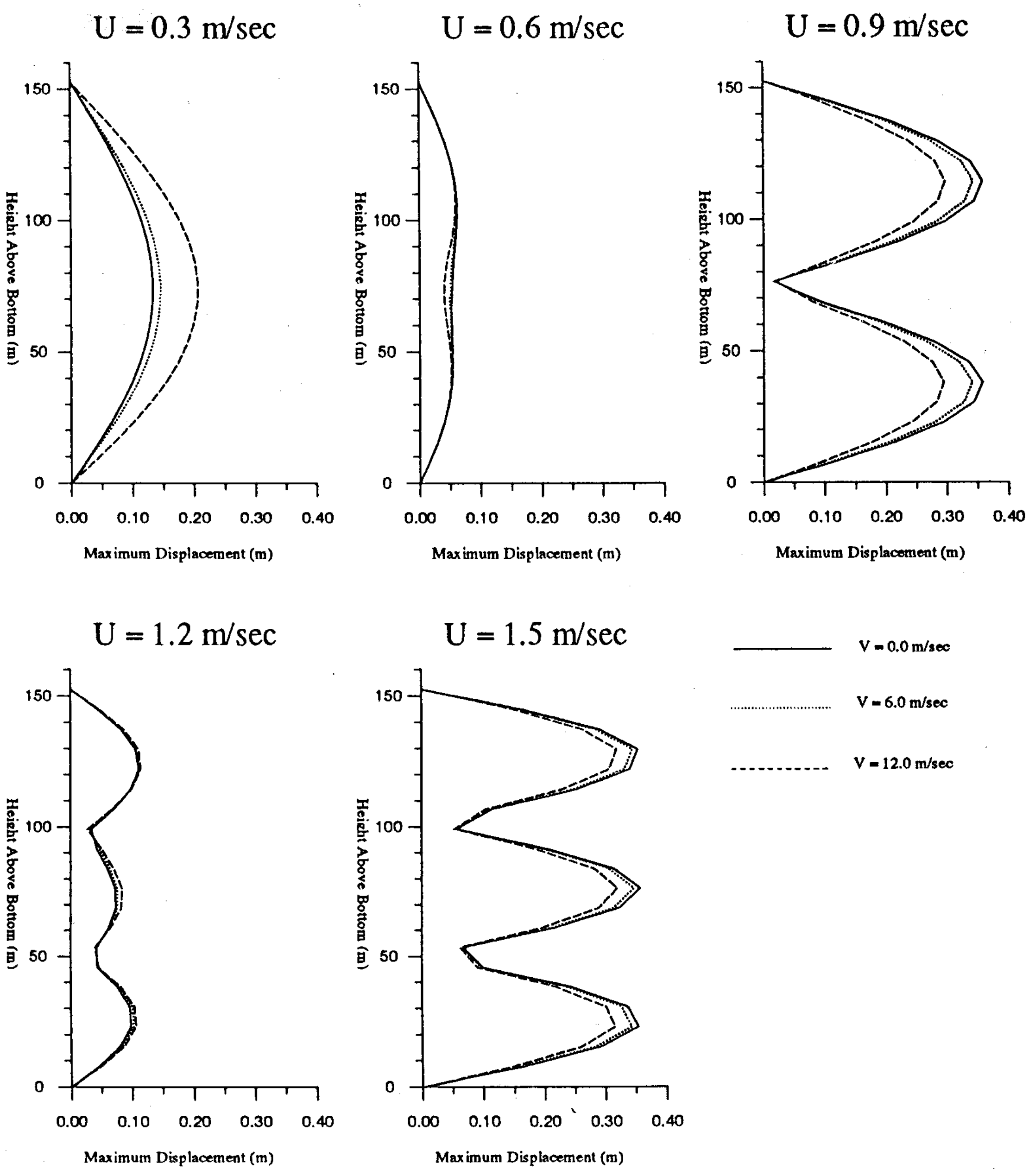

Fig. 6. The effect of internal flow on maximum displacement envelope in vortex direction for 152 m riser at some current velocity (top tension=1550 kN, bottom tension=1200 kN, riser properties except tension are the same as example of Figure 3, top condition is semi-restrained).

ment in the transverse direction, i.e., the direction that the forces due to vortex shedding act. Since the vortex model proposed by Iwan and Blevin has proven to work well in certain range of Reynolds number, current velocities selected in the present computations are also determined within this Reynold number range. As shown in the figure, the presence of internal flow causes only a slight shift of the resonant bands as well as a slight increase in peak values. However, for non-peak positions,

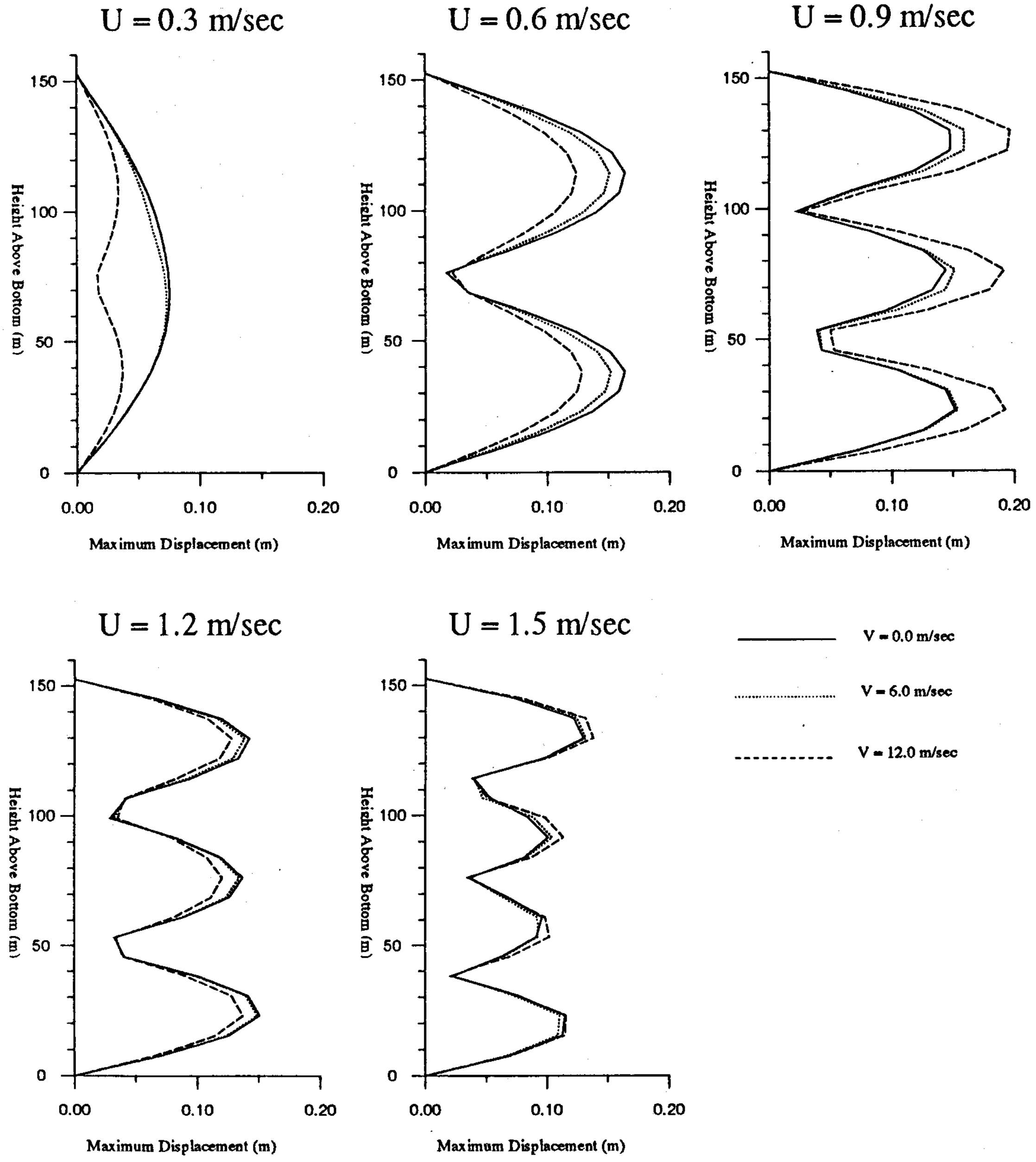

Fig. 7. The effect of internal flow on maximum displacement envelope in vortex direction for 152 m riser at some current velocity (top tension=350 kN, bottom tesion=0 kN, riser properties except tension are the same as example of Figure 3, and top condition is semi-restrained).

the effect of internal flow on displacement amplitude is much larger. Following the curve of any set one sees that internal flow causes both amplitude amplification and reduction depending upon the flow velocity. This is clearly significant in riser in determining the optimum internal flow velocity. In other word, displacement amplitudes are either augmented or reduced depending on where the current velocity magnitudes as shown in this figure. This situation is also shown in Figs. 6 and 7. Also, as internal current velocity increases, the modal shape will also shift from lower to higher modes. For instance, for the case presented in Fig. 6, the riser deflection shape changes from lst to 2nd mode when U=0.6 m/sec and, again, changes to 3rd mode when U=0.9 m/sec.

In summary, the effect of internal flow on vortex-induced vibration is not negligible, in particular, when the top tension is low as the cases found in deep water.

5. CONCLUSIONS AND FURTHER STUDY

A mathematical model for computing vortex-induced vibrations of riser system with the inclusion of internal flow is developed together with numerical solutions and computer programs. From the results of sample computations, the effects of internal flow on vibrations due to vortex shedding are examined and the following conclusions are drawn:

1) The effect of internal flow on vortex-induced vibration is not negligible, in particular, when the top tension is low as for the cases in deep water.

2) For the system under harmonic loadings such vortex shedding, the resonance bands of displacement are affected slightly and the changes become more visible under low tension. Therefore, the effect has to be considered carefully in design point.

3) The maximum effect on displacement is at locations where the slope is zero, while the maximum effect on rotation occurs not only at bottom but also at inflection points.

In this study, Iwan-Blevin's model, which is one of the self-excited oscillator models, has been implemented as a vortex shedding model to examine the effect of internal flow. In that model, the span of riser is divided into segments that are within the resonance band and the remainder, which is outside the resonance band. However, the numerical application of the spanwise extent of resonance band is not feasible due to the distortion of mode shape resulting from the mixed derivative term even though it is expected to be small. Thus, we need to develop the more refined current-vortex model to include the depth-dependent resonance described above. Besides, the nonlinear coupling between in-line and transverse vibration and the extension of the applicable range of Reynolds number may be required for the development of the refined model to investigate the effect of internal flow more precisely.

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