Approach to damping effects of fairings on VIV of flexible cylinder with three-dimensional numerical simulations

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Abstract. Vortex-induced vibration (VIV), which will lead to the fatigue damage on coastal structures like flexible cylinders, has attracted considerable attention. The object of this study is to investigate the VIV phenomenon of flexible cylinder and the damping effects of fairings which could be used as vortex-suppressing devices, with three-dimensional (3D) fluid-structure interaction considered. In this paper, a 3D finite element (FE) model is established with the cylinder simulated by solid elements, the fluid field by the Arbitrary Lagrangian Eulerian (ALE) scheme, and the pipe surface by the moving boundary. Non-uniform mesh in different region and dynamic mesh in the area around the pipe are used to better simulate the pipe and the fluid field. The feasibility of the model is verified. The correlations between the vibration amplitude and the vortex shedding frequency are depicted, with the lock-in phenomenon captured. The displacement responses of the pipe are calculated with the damping effects considered, which is reflected not only in reducing the vibration amplitude of the pipe, but also in raising the flow rates for the lock-in region. This study will provide useful theoretical reference for vibration control of the VIV phenomenon of flexible cylinder.

1. Introduction

Vortex-induced vibration (VIV) with large amplitude is one of the most concerned problems of coastal structures, especially for cylinders with high aspect ratio and large flexibility. The VIV phenomena are prone to occur due to the excitations of ocean flow and will cause fatigue damage of cylinders. Thus it is of urgent requirement in vibration control of cylinders to prevent the fatigue damage. The fairing is a kind of vortex-suppressing devices used to control the vibration by disrupting the vortex shedding on the near-wake region of bluff bodies and delaying the interaction between the separated shear layers.

The VIV, because of its influence on the safety of the structure, has attracted a lot of attentions. Numerous studies on fluid-structure coupled vibration of cylinders or risers have been carried out experimentally and numerically. Experiments on VIV of flexible risers in different flow regime have shed light on the phenomenon of VIV and the principal of its occurrence. Huera-Huarte and Bearman [1-2] studied the VIV of a flexible riser with aspect ratio of 94 and tested the variations of displacement amplitude and average drag versus reduced velocity.

Compared with experimental research, theoretical simulation is more powerful in revealing the mechanism of physical phenomenon, and is resource- and time-saving in prediction and analysis of VIV [3]. There are primarily three strategies of theoretical analyses for VIV of cylinders according to different models of structure. The first is two-dimensional (2D) mass-spring-damping system (MSDS)
combined with the wake oscillator model [4]. The second strategy of theoretical models is based on beam theory, in which the cylinder is considered as a beam and the fluid loadings are simulated using two typical methods. One is the quasi-3D approach, in which the hydrodynamic forces are evaluated in 2D strips while the dynamic response of the cylinder is accomplished through a 3D structural calculation [5]. The other is to simplify the hydrodynamic forces as a distributed transverse loading along the pipeline [6-7]. The third one is the 3D system which can simulate the vibration of the cylinder structure, the action from the fluid field, and the fluid-structure interaction process [8-9].

In this paper, a FE model is established for the cylinder with solid elements and the fluid field with ALE scheme. The dynamic behaviors of the cylinder, including the lift and drag coefficients, and the effects of flow direction are comprehensively investigated. Special emphasize is thrown on the damping effects of fairings with different size. The results may offer a useful reference for fairing design in cylinder vibration control.

2. Numerical model

To establish a rational simulation on the VIV phenomenon of cylinders including the damping effect from fairings, both the structure and the fluid field should be well simulated, as well as the fluid-structure interactions. A 3D FE model is developed for the cylinder with solid elements and the ALE scheme is adopted for simulation of fluid field.

2.1. Computational theory for fluid-structure coupling vibration on cylinders

In order to describe accurately the dynamic behaviors of cylinders and the fluid-structure interaction, a 3D FE model is used to simulate the cylinder with solid element, and the ALE scheme to simulate the fluid field. The equations of motion for the cylinder in FE format may be expressed as

\[ \begin{align*}
M\dddot{X} + C\dddot{X} + K\dddot{X} &= F_{X} \\
M\dddot{Y} + C\dddot{Y} + K\dddot{Y} &= F_{Y}
\end{align*} \]

where \(X\) and \(Y\) are the displacement vectors of the cylinder, \(M\), \(C\) and \(K\) are respectively the mass matrix, damping matrix and stiffness matrix, and \(F_X\) and \(F_Y\) are respectively the vectors of hydrodynamic forces on the cylinder in downstream and transverse directions. The over-dot indicates the derivative with respect to time \(t\), with \(X\) and \(Y\) denoting the velocity vectors, while \(\dddot{X}\) and \(\dddot{Y}\) denoting the acceleration. The hydrodynamic force on the surface of the cylinder is obtained from the integration of the hydrodynamic pressure of the fluid.

The continuity equation and N-S equation for incompressible fluid flows could be expressed as:

\[ \frac{\partial u_i}{\partial x_i} = 0 \]

\[ \frac{\partial}{\partial t} \left( \rho u_i \right) + \sum_{j=1}^{3} u_j \frac{\partial}{\partial x_j} \left( \rho u_i \right) = - \frac{\partial p}{\partial x_i} + \sum_{j=1}^{3} \frac{\partial^2}{\partial x_j^2} (\mu u_i) \]

where \(x_i\) \((i = 1, 2, 3)\) is the Cartesian coordinate, \(u_i\) the velocity of the flows in the \(x_i\)-direction, \(\rho\) the fluid density, \(p\) the pressure, \(\mu\) the dynamic viscosity.

As the ALE scheme is applied to simulate the fluid field around the cylinder. The influence of the vibrating cylinder on the fluid field is treated as a moving boundary in the CFD model. The velocity on the boundary surface of the moving mesh \(\hat{u}_i\) is considered, so the Eq.(4) could be corrected as

\[ \frac{\partial}{\partial t} \left( \rho u_i \right) + \sum_{j=1}^{3} \frac{\partial}{\partial x_j} (u_j - \hat{u}_j) \left( \rho u_i \right) = - \frac{\partial p}{\partial x_i} + \sum_{j=1}^{3} \frac{\partial^2}{\partial x_j^2} (\mu u_i) \]
According to Reynolds averaged method, flow rate consist of time-averaged term and fluctuating term:

\[ u = U + u' \]  \tag{6}

where \( U \) is the time-averaged term which is expressed as

\[ U = \frac{1}{T} \int_{t}^{t+T} u dt \]  \tag{7}

So Eq.(3) and Eq.(5) could be changed into Reynolds-averaged Navier-stokes (RANS) equations:

\[ \frac{\partial U_i}{\partial x_i} = 0 \]  \tag{8}

\[ \frac{\partial}{\partial t} (\rho U_i) + \sum_{j=1}^{3} \left( U_j - \hat{U}_j \right) \frac{\partial}{\partial x_j} (\rho U_i) = -\frac{\partial P}{\partial x_i} + \sum_{j=1}^{3} \frac{\partial^2}{\partial x_i \partial x_j} (\mu u_i \bar{u}_j) \]  \tag{9}

where \( \bar{u}_i \bar{u}_j \) is the time-averaged value of \( u'_i u'_j \), which can be solved by eddy-viscosity model:

\[ \rho u_i u_j = \mu t \left( \frac{\partial U_i}{\partial x_j} + \frac{\partial U_j}{\partial x_i} \right) - \frac{2}{3} \rho k \delta_{ij} \]  \tag{10}

where \( \mu t \) is the vortices viscosity, \( k \) the turbulence kinetic energy. The RANS equations associated with equations of motion are solved by commercial software COMSOL.

The finite volume method is utilized as the discretization method. The incompressible second-order format is utilized as the pressure interpolation format, and the pressure-velocity coupling method is SIMPEC. The transient separation solver is used in this paper. The first-order implicit discretization is used for the transient term, the second-order upstream upwind scheme for the convective term, and the central difference scheme for the diffusion term.

2.2. Model and mesh of the computational domain and the fairing

The cylinder is simulated using 3D solid finite element and the fluid field is simulated using CFD model (Figure. 1). The support conditions at both ends of the suspension are assumed clamped. For the sake of improving computational efficiency, only half of the cylinder is simulated in the model using the symmetrical condition.

The computational domain for fluid field is assumed to be a cuboid (40D×16.65D×20D) (Figure. 1). The inlet boundary is assumed 10D from the cylinder to ensure the fluid a uniform flow when it reaches the surface of the cylinder. The flow velocity on the inlet is \((u_x, u_y, u_z)=(u, 0, 0)\), in which \( u \) is considered a constant. The outlet surface is arranged 30D from the cylinder and supposed to be a free boundary so that the vortex shedding behaviors behind the cylinder could be well presented and the hydrodynamic pressure be well considered. The top and bottom surfaces are both placed 10D from the cylinder so that the fluid field located in the region above and below the cylinder could be well simulated. The top surface is assumed to be an open boundary, which means the pressure on the top surface is the same as the atmospheric pressure outside. The bottom and the two side surfaces considered non-slip boundaries, which means tangential velocity \( u_z = 0 \), that is, the relative velocity between the flow and the wall is zero. The surface of the cylinder is treated as a moving boundary so that the influence of the vibrating cylinder on the fluid field could be considered accurately. The FE mesh work for the fluid field and the cylinder are shown in Figure. 2. The key parameters of the cylinder are shown in Table. 1.
Figure 1. calculation model and boundary conditions.

Figure 2. FEM mesh work for the fluid and the cylinder.

Table 1 Key parameters of the cylinder.

| parameters        | symbol | unit | value |
|-------------------|--------|------|-------|
| External diameter | $D$    | m    | 0.03  |
| Thickness of wall | $d$    | m    | 0.003 |
| Length            | $L$    | m    | 1     |
| Aspect ratio      | $L/D$  | -    | 33.3  |
| Bending stiffness | $EI$   | Nm² | 7.2   |
| Mass              | $m$    | kg/m | 2.0   |
| Mass ratio        | $m^*$  | -    | 2.83  |
| Natural frequency | $f_n$  | Hz   | 6.69  |

The model and the FE mesh of the cylinder equipped by the fairing are shown in Figure 3, in which the fairing is assumed to be attached on the surface of the cylinder and $L_f/D$ is a dimensionless parameter referring to the geometrical size of the fairing. The meshing discretization for the FE model on the cylinder with a fairing is roughly the same as that on the bare cylinder.
2.3. Validation of the model

The lift ($C_l$) and drag ($C_d$) coefficients reflect the variation of the fluid-structure interaction due to the VIV phenomenon. The case $Re=200$ has been considered in the following as it corresponds to the highest Reynolds number for which the wake remains laminar (Figure. 4). The results exhibit that after the oscillation of $C_d$ and $C_l$ become steady, the maximum and minimum values for $C_d$ are 1.43 and 1.27, and 0.51 and -0.51 for $C_l$ (Figure. 4). These results are compared with the lift and drag coefficients in literature (Table 2). The present results of $C_l$ are smaller compared to the results of the other scholars [10-11]. The cause may be the influence of number of diameters of the far field boundary and the dimensional time step. But in general, as the relative error is acceptable, the VIV calculation model could be validated through this comparison.

Table 2 Comparisons of lift and drag coefficients.

| Resource                  | $C_{d,\text{max}}$ | $C_{d,\text{min}}$ | $C_{l,\text{max}}$ | $C_{l,\text{min}}$ |
|---------------------------|--------------------|--------------------|--------------------|--------------------|
| Present                   | 1.43               | 1.27               | 0.51               | -0.51              |
| Zhang and Dalton (1997)   | 1.28               | 1.22               | 0.54               | -0.54              |
| Lecointe and Piquet (1989)| 1.33               | 1.25               | 0.60               | -0.60              |

3. The VIV response of the pipeline

To study the VIV dynamic behaviors of the cylinder, the FE model established is utilized for calculating of the dynamic responses of the cylinders. The displacement responses of the cylinder are obtained and depicted in Figure. 5, due to uniform flows with different velocities. In Figure. 5, $U_r$ is...
the reduced velocity of the flow with the definition of \( U_r = U/f_n D \), in which \( U \), \( f_n \) and \( D \) are the flow rate, the natural frequency and the diameter of the cylinder, respectively. \( X/D \) here is the dimensionless downstream displacement with \( Y/D \) the dimensionless transverse displacement.

From Figure. 5, it can be seen that, for various reduced velocity in the range of 1 - 14, the dimensionless transverse displacement \( (Y/D) \) increases firstly and then decreases. The phenomenon of “beat” is captured at \( 5 \leq U_r \leq 7 \). The dimensionless downstream displacement \( (X/D) \) increases with the reduced velocity. The parametric analysis shown in Figure. 5 demonstrates the significant influence of the flow rate on the dynamic responses of the cylinder.

In order to systematically investigate the effect of the flow rate on the VIV responses of the cylinder, the variations of dimensionless displacement amplitude versus the reduced velocity are depicted in Figure. 6. The amplitude of dimensionless transverse displacement \( (Y/D) \) increases firstly and then decreases with the increasing of the reduced velocity, with the peak observed when \( U_r \) is 6.
The VIV phenomenon with higher amplitude occurs at $5 \leq U_r \leq 7$. The amplitude for dimensionless downstream displacement ($X/D$) increases with the reduced velocity, which shows the positive correlation between the dimensionless downstream displacement ($X/D$) and the flow rate. The effect of reduced velocity on the vortex shedding frequency is also depicted in Figure 6, in which $f^*$ is the dimensionless vortex shedding frequency with the definition $f^* = \frac{f}{f_n}$. $f$ and $f_n$ are the vortex shedding frequency and the first order natural frequency of the cylinder. When the dimensionless vortex shedding frequency is close to 1, it means the synchronization between the vortex shedding and the vibration of the cylinder, which introduces the lock-in phenomenon. Here the lock-in occurs when $5 \leq U_r \leq 7$, and this is the reason for the occurrence of VIV responses of the cylinder with high amplitude as the red curve shows in Figure 6.

Figure 6 The dimensionless displacement amplitude and the dimensionless vortex shedding frequency.

4. Damping effects of fairings on VIV displacement responses

VIV with large amplitude is usually regarded as a great threat on the safety of cylinders. Here the fairing is utilized as a damper and corresponding damping effects are investigated through the displacement responses of the cylinder. The time histories of the displacement responses in three cases are depicted in Figure 7, with the responses curves of the bare cylinder also presented for comparison.

When the water flow is at low reduced velocity like $2 \leq U_r \leq 8$ (Figure 7a-b), the transverse displacement amplitude of the cylinder equipped with fairings decreases significantly compared with those of the bare cylinder, especially when $U_r = 6$ with which the maximal amplitude of the cylinder is observed as Figure 6(a) shows. In addition, the damping effect increases with the increase of geometric size in Case II and Case III. It is well known that the ocean flow is always within the range of low flow rates like $2 \leq U_r \leq 8$, which also involves the lock-in region. Hence the damping effects of fairings captured on displacement amplitude with low flow rates are of great importance for size selection for dampers on the cylinders for vibration control.

To furtherly quantify the damping effects of the fairings on the transverse displacement responses, the amplitude of transverse displacement responses versus geometrical sizes of the fairings are depicted in Figure 8. It is clear that the damping effects of fairings are reflected not only in reducing the vibration amplitude in low flow rate, but also in alternating the lock-in region into the reduced velocity range with higher flow rates. As mentioned before, the ocean flow is always within the range of low flow rates like $2 \leq U_r \leq 8$, thus it is of great significance for damping effects on raising the flow rates for the lock-in region. The results of VIV responses in Figure 7-8 have demonstrated the damping effects of fairings on VIV displacement responses from various aspects and have facilitated the application of fairings on cylinders.
Figure 7. Time histories of the displacement responses with fairings. (a): $U_r=2$, (b): $U_r=6$, (c): $U_r=8$ and (d): $U_r=14$. 
5. Conclusions

As a concerned problem of coastal structures, VIV phenomenon of the cylinder in uniform flow is simulated in this paper via a 3D FE model with fluid-structure interaction well considered. The damping effects of fairings which are used as vortex-suppressing dampers are investigated as well. For the bare cylinder, the correlations between the vibration amplitude and the vortex shedding frequency are depicted, with the lock-in phenomenon captured. When the cylinder is equipped with the fairing, the damping effects of fairings on VIV displacement responses are reflected not only in reducing the vibration amplitude in low flow rate, but also in alternating the lock-in region into the reduced velocity range with higher flow rates. Considering that the ocean flow is always within the range of low flow rates, it is of great significance for damping effects on raising the flow rates for the lock-in region, and the applicability of fairings in controlling the VIV phenomenon has been demonstrated.

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