Original Paper

Numerical Simulation of Vortex Induced Vibration of A Flexible Cylinder

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Abstract

Numerical simulations of vortex-induced vibration of a three-dimensional flexible Cylinder under uniform turbulent flow are calculated when Reynolds number is 1.35×104. In order to achieve the vortex-induced vibration, the three-dimensional unsteady, viscous, incompressible Navier-Stokes equation and LES turbulence model are solved with the finite volume approach, the Cylinder is discretized according to the finite element theory, and its dynamic equilibrium equations are solved by the Newmark method. The fluid-Cylinder interaction is realized by utilizing the diffusion-based smooth dynamic mesh method. Considering VIV system, the variety trends of lift coefficient, drag coefficient, displacement, vortex shedding frequency, phase difference angle of Cylinder are analyzed under different frequency ratios. The nonlinear phenomena of locked-in, phase-switch are captured successfully. Meanwhile, the limit cycle and bifurcation of lift coefficient and displacement are analyzed using trajectory, phase portrait and Poincare sections. The results reveal that: when drag coefficient reaches its minimum value, the transverse amplitude reaches its maximum and the “lock-in” begins simultaneously. In the range of “lock-in”, amplitude decreases gradually with increasing of frequency ratio. When lift coefficient reaches its minimum value, the phase difference, between lift coefficient and lateral displacement, undergoes a suddenly change from the “out-of-phase” to the “in-phase” mode. There is no bifurcation of lift coefficient and lateral displacement occurred in three dimensional flexible Cylinder submitted to uniform turbulent flow.

Keywords: Vortex-induced,dimensional,finite element,lift coefficient.

1. Introduction

Vortex-induced vibration occurs in many engineering situations, such as Cylinder bundles in steam generator or heat exchangers, deep sea risers, etc. Circular cylinder or cylinder bundle is the key components in reactor. The shedding of vortices, alternately and periodically, from a bluff object gives rise to fluctuating lift and drag forces and leads to vibrations and noise; furthermore it could cause structure failure. Therefore, it is very necessary to further investigate the vortex induced vibration of 3-D flexible Cylinder.

The practical significance of vortex-induced vibration has led to a large number of fundamental studies, such as turbulence, separation, fluid-structure-interaction, are discussed in many comprehensive reviews [1]0. The earlier research on VIV mainly relies on experiment. Many researchers have investigated the flow induced vibrations of elastically mounted rigid Cylinder, such as Feng [2], Griffin [3], Griffin&Ramberg [4], Williamson [5]6. The VIV characteristic of cylinder with high mass ratio was conducted by Feng who undertook one of the first comprehensive experimental studies of this problem. Feng’s data have only two branches (Initial and Lower). For lower mass ratio cylinder system, fairly comprehensive reviews on this VIV problem can be found in the article by Williamson. They found that, at lower Reynolds numbers (3500-10 000), the VIV system has three branches (Initial, Upper, and Lower), a much larger peak amplitude, and a broader synchronization range. Much progress has been made in the vortex formation and development, and it induced vibrations. The flow field pattern, vortex shedding process and wake vortex mode are obtained successfully.Besides experimental study, much progress has been made numerically toward the understanding of the dynamics of VIV. In this complicated problem, a lot of methods were used to solve Navier-Stokes equations, involving computational fluid dynamics (CFD) method [7]8, Finite element method (FEM) [9]10, vortex element methods (VEM) [11][12], time-marching technique [13], etc. Among these methods, The CFD method is mostly used. The circular cylinder is generally simulated by an equivalent mass-spring-damper to investigate the dynamics of flow induced cylinder vibration and the influence of cylinder vibration.
oscillation on flow field. Antoine Placzek [14] and Mittal [15] studied the flow induced vibration characteristics of circular cylinder and wake vortex structure. Evangelinos [8] did a 3-D DNS study (1-DOF) at Re=1000 for flexible cylinders. Li [16] use the space-time finite element method to investigate the VIV of a two-dimensional elastic mounted circular cylinder under the uniform flow when Reynolds number is 200. Gabbai [17] reviewed the literature on the mathematical models used to investigate vortex-induced vibration (VIV) of circular cylinders and a variety of issues concerning the vortex-induced vibration of circular cylinders were discussed. The strengths and weaknesses of the current state of the understanding of the complex fluid/structure interaction are discussed in some detail. Sarpkaya [18] made a comprehensive review of the progress made during the past two decades on vortex-induced vibration (VIV) of mostly circular cylindrical structures subjected to steady uniform flow. The critical elements of the evolution of the ideas, theoretical insights, experimental methods, and numerical models were traced systematically. Though much progress has been made during the past decade, both numerically and experimentally, the complex interaction between structure and fluid is not completely understood yet and remains to be discovered. Meanwhile, Vortex-induced vibration of an elastic cylinder is of strongly nonlinear quality [16]. However, there is a few nonlinear analysis. On the other hand, due to the complexity of the flow induced vibration of 3-D flexible Cylinder, current VIV models mostly simplify the 3-D Cylinder to a 2-D elastically mounted rigid Cylinder, thus the elastic distortion is neglected. Therefore, it cannot consider the interaction between elastic distortion of structure and fluid flow. With the equipment process parameters (flow rate, temperature, etc.) becoming higher and higher, it is necessary to construct more accurate physical models to analyze the interactions between fluid and structure as well as their nonlinear response characteristics.

The VIV model based on fully coupled approach combining the computational fluid dynamics (CFD) and the computational structural dynamics (CSD). The finite volume method, with the dynamic mesh technique to model the motion of Cylinder, is used to compute the flow field while a finite element method is implemented for the Cylinder displacement assessment. The VIV characteristics and nonlinearity of three-dimensional Cylinder under turbulent flow are investigated. Meanwhile, the interaction between Cylinder and fluid is studied numerically. Such an approach is expected to help assessing some vibration for designers.

2. Numerical Methods

2.1 CFD model

The general conclusion is that even advanced RANS (Reynolds averaged Navier–Stokes) models such as non-linear realizable and RNG types of k-ε models severely underestimate the high turbulent kinetic energy levels observed in densely packed Cylinder bundles. The LES results on the fine mesh are comparable to a DNS and experiments and reasonable agreement is still achieved with a coarse mesh [19]. The LES model can obtain satisfactory results in the turbulent flow field which RANS cannot. Thus, the fluid domain is calculated by LES in the present work. The governing equations employed for LES are obtained by filtering the time-dependent Navier-Stokes equations in physical space. Filtering the continuity and momentum equations, one obtains:

\[
\frac{\partial \bar{u}_i}{\partial x_i} = 0
\]

\[
\frac{\partial \bar{u}_i}{\partial t} + \frac{\partial \bar{u}_i \bar{u}_j}{\partial x_j} = - \frac{1}{\rho} \frac{\partial p}{\partial x_i} + \frac{\partial}{\partial x_j} \left( \mu \frac{\partial \bar{u}_i}{\partial x_j} \right) - \frac{\partial \tau_{ij}}{\partial x_j}
\]

where \( p \) is fluid density, \( \mu \) is dynamic viscosity, \( t \) is time, \( p \) is pressure. \( u_i \) (i=1,2,3) is velocity components and is Cartesian coordinates. \( \bar{p} \) and \( \bar{u}_i \) are, respectively, the filtered variable of \( p \) and \( u_i \). \( \tau_{ij} = \bar{u}_i \bar{u}_j - \bar{u}_i \bar{u}_j \) is the subgrid-scale stress defined by Algebraic Wall-Modeled LES (WMLES) approach. The WMLES approach combines a modified Smagorinsky model and with a mixing length model and with the wall-damping function. Thereby WMLES approach can be applied at the same grid resolution to an ever increasing Reynolds number.

2.2 The structural analysis model

The Cylinder is discretized according to the finite element theory [20] and for each rod a mass matrix (M) and a stiffness matrix (K) are generated. The Newmark method is used for integrating the dynamics equilibrium equations over time and to analysis transient dynamics.

\[
M \ddot{x} + C \dot{x} + Kx = F(t)
\]

where M and K are mass matrix and stiffness matrix respectively, C being the damping matrix, expressed as a proportional Rayleigh damping \( \alpha \alpha M + \beta K \). \( x \), \( \dot{x} \) and \( \ddot{x} \) are displacement, velocity and acceleration of node. The fluid loads coming from the fluid computation by CFD model takes the form of a loading vector on nodes \( F(t) \). The initial conditions in velocity and displacement are taken to be nil for the whole structures.

2.3 The dynamic mesh model

Diffusion-based smoothing is used to update a dynamic mesh. For diffusion-based smoothing, the mesh motion is governed by the diffusion equation as in equation (4):

\[
\nabla \cdot (\gamma \nabla \bar{u}_i) = 0
\]

where, \( \bar{u}_i \) is the mesh displacement velocity. On deforming boundaries, the boundary conditions are such that the mesh motion is tangent to the boundary. The diffusion coefficient \( \gamma \) in equation (4) can be used to control how the boundary motion affects the interior mesh motion and it can be a function of the cell volume \( V_i \) and is of the form \( \gamma = \frac{1}{V_i} \alpha \) and \( \alpha \) is the control parameter. The equation (4) is discretized using finite volume method, and the resulting matrix is solved iteratively. The cell centered solution for the displacement velocity \( \bar{u}_i \) is interpolated onto the nodes using inverse distance weighted averaging, and the node positions are updated according to:
\[ x_{new} = x_{old} + u \Delta t \] (5)

3. The Conditions of the Computation

Cylinder parameters: Cylinder length \( L = 0.5 \text{m} \). Outer diameter \( D = 0.01 \text{m} \). Inner diameter \( D_i = 0.0095 \text{m} \). Young modulus \( E = 1 \times 10^5 \text{MPa} \). Possion ratio \( \nu = 0.3 \). Density \( \rho_s = 6500 \text{kg/m}^3 \). Damping coefficient \( \alpha = 5.098 \). \( \beta = 0.000215 \). The Cylinder is fixed supported at both the end.

Fluid parameters: the work fluid is water. Density \( \rho = 998.2 \text{kg/m}^3 \), dynamic viscosity \( \mu = 0.001003 \text{pa.s} \). Non-dimensional inflow velocity \( U_r = U/(f_n D) = 0.5 \sim 17 \), where \( U \) is the upstream velocity and \( f_n \) is the natural frequency of the Cylinder.

Computational domain and boundary condition: the computational domain and mesh is shown in Fig. 1. All the structured grids were generated using ICEM CFD. In Fig. 1, Boundary conditions are a specified fluid inlet for the upstream border (left side in Fig. 1) and a fixed pressure at the downstream one (left side in Fig. 1). Other boundaries are symmetry planes and walls. Cylinder wall is the fluid-structure interface, and set as dynamic mesh condition.

Time Control parameters: during the calculation, the time step both in structural dynamics computation and fluid dynamics computation is 0.00025s.

![Fig. 1 Schematic of computational domain and mesh](image)

We introduce the following non-dimensional quantities for describing briefly: mass ratio \( m^* = m/(\rho D^2) \), response parameter \( G = 2\pi St^2m^*\zeta \), dimensionless time \( t^* = tU/D \), lift coefficient \( C_l = F_l/(0.5\rho U^2) \), r.m.s(root-mean-square) of the lift coefficient \( C_l R M S = F_l R M S/(0.5\rho U^2) \), drag coefficient \( C_d = F_d/(0.5\rho U^2) \), mean drag coefficient \( \bar{C}_d = F_d/(0.5\rho U^2) \), r.m.s of the drag coefficient \( C_d R M S = F_d R M S/(0.5\rho U^2) \), streamline displacement \( x/D \), lateral displacement \( y/D \), streamline amplitude \( Ax/D = x R M S / D \), lateral amplitude \( Ay/D = y R M S / D \), dimensionless vortex shedding frequency \( St = f_{vs} D / U \), reduced velocity \( U_r = U/(f_n D) \), where \( U \) is the mass per unit length of the Cylinder. \( \rho_s \) and \( \rho \) are the density of Cylinder and fluid respectively. \( \zeta \) is the damping ratio. \( F_d R M S \) and \( F_l R M S \) are the r.m.s of the drag and lift respectively. \( F_d \) is the mean of drag \( F_d \). \( x R M S \) and \( y R M S \) are the r.m.s of the streamline displacement and lateral displacement. \( U \) is the inflow velocity, \( D \) is the diameter of the Cylinder, \( f_n \) is the natural frequency of the Cylinder, \( A \) is the projective area of computation, \( t \) is time.

4. Investigation of Computational Mesh and Numerical Model

Structured, non-uniform, boundary-fitted grids were generated for the solution domain shown in Fig. 2. All the structured grids used were generated using ICEM CFD, the grid generation component of ANSYS. The O-type grid is generated around the Cylinder to ensure good quality meshes. The grid expands away from Cylinder boundary in radial-direction with the geometric expansion factor 1.08 within O-block. Away from the O-block, the grid expands with the geometric expansion factor 1.4. Fig. 2 shows the four grids used in present calculation. Table 4 provides some details of the grids, lift force, drag force, and St, including the numbers of nodes on the surface of the Cylinder and in the radial-direction, and the maximum values in the domain of the standard \( y^+ \). Table 1 indicates the four grids have a small standard \( y^+ (y^+ \approx 1) \). It should be noted that, Grid A is the finest grid, Grid B tests the influence of circumference grid nodes, Grid C tests the influence of radial-direction grid nodes, Grid D is the mesh adopted after investigating the influence of grid resolution on flow filed characteristics.

A mesh refinement study is done to verify the result independence and the accuracy of the method. A uniform flow with \( Re = 3800 \) is calculated to assess the performance of the method. The results are compared with the experimental data [21]. The comparison of main parameters is shown in Table 1. It is seen that the present result is compared to experimental data and existing models in the literature. That shows the present models are all reasonable. Hence, all computational results are obtained on the grid D mesh. Besides, in order to further validate the numerical model, a case of vortex induced vibration for a three-dimensional flexible Cylinder is computed and analyzed. Fig. 3 shows the variation of the frequency ratios \( f_{ex}/f_n \), response frequency \( f_{ex} \) to natural frequency \( f_n \), and lateral amplitude \( A_y/D \) versus reduced velocity \( U_r \). Meanwhile, the numerical results are compared with the experiment data [22]. As can be shown in Fig. 3, both the amplitude and frequency is compared to experimental data and existing models in the literature. That shows the present numerical model is reasonable.
Table 1 Details of grids used in mesh-independence tests and their fluid force

<table>
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<tr>
<th></th>
<th>Nc</th>
<th>Nr</th>
<th>y+</th>
<th>Cd_{RMS}</th>
<th>St</th>
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<tr>
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<td>0.293</td>
<td>0.875</td>
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<tr>
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<td>0.293</td>
<td>0.918</td>
<td>0.229</td>
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<tr>
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<td>1.467</td>
<td>0.906</td>
<td>0.229</td>
</tr>
<tr>
<td>Norberg{superscript}[21]</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>0.99±0.05</td>
<td>0.215±0.005</td>
</tr>
</tbody>
</table>

Fig. 2 Computational grid

Fig. 3 Cylinder response verse $U_r$: (a) frequency (b) amplitude

5. Results and Discussions

5.1 Fluid forces and amplitude

The influences of frequency ratio on VIV characteristics of three-dimensional flexible Cylinder are to be discussed by making the inflow velocity fixed and decreasing natural frequency $f_n$. A summary of several vibration parameters varying with $f_n/f_{st}$ is shown in Fig. 4. Where Fig.4(a) shows the effect of frequency ratio $f_n/f_{st}$ on $CD_{RMS}$(the r.m.s. of drag), $CD_{MAX}$(the maximum drag), $CL_{RMS}$(the r.m.s. of lift force coefficient), $CL_{MAX}$(the maximum lift force coefficient). It is seen that, the maximum and minimum peak value of drag coefficients occurs at $f_n/f_{st}=1.25$ and $f_n/f_{st}=0.56$ respectively. The maximum and minimum peak value of lift coefficients occurs at $f_n/f_{st}=1.67$ and $f_n/f_{st}=0.45$ respectively. The trends of lateral amplitude $A_y/D$ is shown in Fig. 13(b). $A_y/D$ reaches its peak value at $f_n/f_{st}=0.56$. It obviously shows that, the maximum of lateral responses are appeared at the minimum drag coefficient, and not the maximum drag coefficient as ordinarily considered. The relationship between $f_{vs}/f_{st}$ and $f_n/f_{st}$ is presented in Fig.4(c), where $f_{vs}$ represents the vortex shedding frequency ratio of vibrating Cylinder, $f_{st}$ represents the vortex shedding frequency corresponding to the fixed Cylinder, and $f_n$ is the natural frequency of flexible Cylinder. The maximum of $f_{vs}/f_{st}$ appears at about $f_n/f_{st}=1.25~2$, which indicates, in this range of $f_n/f_{st}$, the interaction effect between Cylinder and fluid flow is the most intense.

When the frequency ratio $f_n/f_{st}=0.56~2.5$, the lateral amplitude lock-in occurs. Dissimilar to the trends of vibration characteristics versus reduced velocity $U_r$, the onset of lateral response lock-in occurs at the minimum drag force coefficient (slightly larger than the minimum lift force coefficient) for various frequency ratios. In the peak value range of drag and lift force coefficient, and the frequency ratio range of lock in, the lateral amplitude is decreasing with the frequency ratio increasing. While in Fig.4, the onset of lock in occurs at the reduced velocity $U_r$ corresponding to the maximum lift force coefficient. And the amplitude is constant under lock in conditions.
5.2 Phase difference

The VIV characteristics for three-dimensional flexible Cylinder are very well characterized by plotting the phase \( \phi \) between the lift force and displacement. Fig.5 shows the phase difference \( \phi \) at different frequency ratios. When the frequency ratio is between 0.45 and 0.5, the phase between the lift force and the lateral displacement undergoes a suddenly change from out-phase to in-phase mode. The jump phenomenon of phase difference is called the "phase-switch", which is a typical nonlinear phenomenon. The phase angle \( \phi \), found as a function of time by using the Hilbert transform, are shown in Fig.5. These figures show that, when \( fn/fst=0.15 \), the phase difference is 180°. While in the transition stage, the phase angle "slips" periodically through 360° and its time history becomes disorderly. On the other hand, as \( fn/fst \geq 0.56 \), the phase difference remains close to 0°. That is to say, phase difference between the lift force coefficient and lateral displacement is in-phase mode.

5.3 Phase portrait and limit cycle

Phase portrait is a very useful tool to analyze dynamics of system. And limit cycle is one of the most important characteristics of nonlinear vibration. The limit cycles of lateral displacement and lift coefficient when \( fn/fst=0.15 \sim 5 \) are shown in Fig. 6 and Fig. 7 respectively. Fig. 8 shows the Poincare section of lift coefficient. The Cl' represents the derivative of the lift coefficient and it was calculated by the difference method with second order accurate. The period of the Poincare map is the vortex shedding period.

When the frequency ratio \( fn/fst=0.15 \) or between 0.56 and 2.5, the shape of the lateral displacement limit cycle is an ellipse. However, the shape of the lift coefficient limit cycle changes from the simple ellipse to the complex geometric figure. Meanwhile, there is only one point in Poincare section correspondingly. As frequency ratio \( fn/fst \) is any other values, the lift coefficient and lateral displacement curves become very complex. There are a lot of points in Poincare section which forms a complex situation.
However, it is not the chaos motion as it is independent of the initial condition. Thanks to several detailed analyses of limit cycle and Poincare map of lateral displacement and lift force coefficient at different frequency ratios, It is found that, under turbulent flow, there is no bifurcation of periodic solution for three-dimensional flexible Cylinder within the frequency ratio range from $f_n/f_{st}=0.15$ to $f_n/f_{st}=5$.

![Fig. 6 Limit cycle of the lateral displacement at different $f_n/f_{st}$](image)

![Fig. 7 Limit cycle of the lift coefficient at different $f_n/f_{st}$](image)

![Fig. 8 Poincare section of the lift coefficient at different $f_n/f_{st}$](image)

### 5.4 Analysis of Reynolds Number

In order to discuss the influences of Reynolds number on VIV characteristics for three-dimensional flexible Cylinder, keeping the natural frequency of Cylinder fixed and changing the inflow velocity, the effects of Re on response are investigated. Where the inflow velocity $U=0.5\text{m/s}~10.98\text{m/s}$ and its corresponding Reynolds number range is $5\times10^3~1.1\times10^5$. Fig. 9 shows the vibration parameters at different $f_n/f_{st}$. In these figures, “$U$” presents the change of frequency ratio via changing the inflow velocity (therefore changing the Reynolds number and vortex shedding frequency of fixed Cylinder $f_{st}$). While “$f_n$” presents the change of frequency ratio via changing the natural frequency $f_n$. These figures show that, the variation curves of drag force coefficient and amplitude versus frequency ratio on both cases is nearly identical. It reveals that the VIV characteristics of three-dimensional Cylinder manifest are remarkably similar in the Reynolds number range of $5\times10^3~1.1\times10^5$.

![Fig. 9 Main parameters versus $f_n/f_{st}$: (a) force coefficient (b) amplitude](image)
6. Conclusions

The variation of VIV behavior and nonlinear characteristics for flexible Cylinder versus velocity and frequency ratio is investigated by combined computational fluid dynamics and computational structural dynamics. Conclusions are drawn as follows:

Compared to high $m^*$ and $m^*$ system or low $m^*$ and $m^*$ system, a quasi-upper branch is found in the present fluid-flexible Cylinder coupling system with high $m^*$ and low $m^*$. The amplitude of three-dimensional flexible Cylinder has a broader synchronization range and it is higher than elastically mounted rigid Cylinder under VIV.

There is no bifurcation of lift coefficient and lateral displacement occurred in three dimensional flexible Cylinder submitted to uniform turbulent flow within the range of reduced velocity $Ur=0.5$–$18$ and frequent ratio $fu/fst=0.15$–$5$. The phase angle reaches zero under “lock-in”, and the dynamic behavior is a periodic motion.

In the quasi-upper branch and lower branch regime, the “lock-in” begins. In quasi-upper branch, the transvers amplitude increases with reduced velocity increasing. While in lower branch, the amplitude keeps almost constant. The drag force reaches its peak value before lift force. The lift force coefficient reaches its maximum value at the switch from initial branch to quasi-upper branch. In the peak of drag and lift force coefficient versus frequency ratio, the transverse response “lock-in” occurs. The maximum transverse amplitude takes place in the minimum drag force coefficient. The amplitude in “lock-in” decreases with frequency ratio increasing.

References