Comparison of acoustic-structure based one-way FSI and two-way FSI

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Abstract. In the numerical simulation of flow-induced vibration using two-way fluid-structure interaction (TW-FSI) method, it is usually limited by mesh deformation, and consumes a lot of computing resources. These problems can be avoided by adopting acoustic-structure based one-way FSI (ASOW-FSI) approach. The purpose of this work is to explore the applicability of ASOW-FSI, and to analyze the similarities and differences between the two dynamic response analysis methods. A series of numerical calculations are carried out for the elastically mounted rigid cylinder at low Reynolds number. By systematically comparing the results calculated by these two methods, we found that ASOW-FSI can accurately predict the response in the range of partial reduced velocity, even closer to the experimental results than TW-FSI method. The main reason for the difference between the two methods in the whole reduced velocity range is that the ASOW-FSI method can not predict the "lock-in". Finally, the mechanism of "lock-in" in vortex-induced vibration (VIV) is analyzed by forced oscillation. The results show that the structural vibration has the ability to dominate the vortex shedding under certain conditions. If the natural mode of the structure is excited at this time, "lock-in" will occur.

1. Introduction

FSI is a science produced by the intersection of fluid mechanics and solid mechanics. It involves many engineering fields, such as civil engineering, aerospace, fluid machinery, ocean engineering, and so on. Therefore, it has received extensive attention in the academia. In the past decades, experimental methods and numerical calculation methods have been developed continuously, which laid a foundation for the study of FSI.

The numerical methods for FSI problems include strong-coupling method and weak-coupling method. The weak-coupling method divides the computational Domain into flow field and structure field. First, the independent solutions are obtained, and then the coupling solutions are obtained by the interaction of the coupling surfaces. Weak-coupling method can satisfy the accuracy requirement, and the computational efficiency is much higher than that of strong-coupling method, so it has been widely used. When the weak-coupling method is used, the data can be transmitted bidirectionally and unilaterally, which corresponds to TW-FSI and OW-FSI.

With the development of computer technology, the application of TW-FSI in scientific research is increasing. For example, Guilmineau et al.[1], Zhang et al.[2] and Gao et al.[3] used TW-FSI method to simulate the VIV of elastically mounted rigid cylinder, and captured the phenomena of "Lock-in",...
"beat" and "hysteresis". Liaghat[4] and Gauthier et al.[5] carried out numerical simulation on the hydrodynamic damping characteristics of hydrofoils. Compared with the experimental results, the reliability of TW-FSI method was verified. In theory, TW-FSI is an accurate method for calculating flow-induced vibration, but it is limited by mesh deformation and consumes a lot of computing resources. Therefore, TW-FSI is currently mainly used for dynamic stress analysis of simple structures. OW-FSI method is often used for complex structures. For example, based on the OW-FSI method, Zhou et al.[6] constructed a FSI interface model to transform water pressure into nodal load on structural surface, and studied the dynamic stress characteristics of axial-flow turbine blades. Wang et al.[7] studied the dynamic stress characteristics of the pump body and impeller of a large double suction centrifugal pump under hydraulic induction using a similar method. This kind of structural transient analysis method based on OW-FSI can effectively overcome the shortcomings of TW-FSI, but does not take into account the additional mass effect of water, so the prediction of structural dynamic response (especially of resonance phenomena) is not accurate enough. On the basis of these studies, He et al.[8] combined with acoustic-structural coupling, analyzed the dynamic response of pump-turbine runner. The results show that the ASOW-FSI method can calculate the added mass of fluid and accurately predict resonance phenomena. This method is considered to be an economical and effective method to analyze the dynamic response of complex structures. However, due to the neglect of the effect of structural deformation on fluid, it is usually used in the case of small deformation.

In this paper, the flow-induced vibration of elastically mounted rigid cylinder is calculated by using traditional TW-FSI and ASOW-FSI methods respectively, and compared with the experimental results of Khalak et al.[9] The applicability of ASOW-FSI method under large deformation conditions is discussed. The similarities and differences between the two transient calculation methods are analyzed in terms of lift characteristics, phase characteristics and vortex shape. The results show that the main reason for the difference between the two results in the whole reduced velocity range is that the ASOW-FSI method can not predict the "lock-in". Finally, the cylinder is forced to oscillate in the form of response calculated by TW-FSI. The flow field is solved and compared with the results of TW-FSI. The purpose is to explore the internal mechanism of "lock-in" in VIV.

2. Governing equations
In TW-FSI calculation, the dynamic equation of the structure is shown in eq. (1). At this time, the flow field and the structure field are solved alternately, and the data are transmitted bidirectionally. While the fluid flow excites the structure, the motion of the structure also affects the flow. Therefore, the flow-induced force \( F_{\text{f}} \) in the formula takes into account the motion of the structure.

\[
\begin{bmatrix}
M_s \\
C_s \\
K_s
\end{bmatrix}
\begin{bmatrix}
\ddot{y}_o \\
\dot{y}_o \\
y_o
\end{bmatrix}
= \begin{bmatrix}
F_{\text{f}} \\
0 \\
0
\end{bmatrix}
\]  

(1)

When using the ASOW-FSI method, the governing equation of the structure can be rewritten as eq. (2). Flow field and structure field are solved independently. Firstly, the flow-induced force is calculated under the condition of a fixed cylinder. The force is then applied to the elastically mounted cylinder.

\[
\begin{bmatrix}
M_s \\
C_s \\
K_s
\end{bmatrix}
\begin{bmatrix}
\ddot{y}_o \\
\dot{y}_o \\
y_o
\end{bmatrix}
= \begin{bmatrix}
F_{\text{f}} \\
0 \\
0
\end{bmatrix}
\]  

(2)

where \( [M_s] \), \( [C_s] \) and \( [K_s] \) are the structural mass, damping and stiffness matrices; \( [M_A] \), \( [C_A] \) and \( [K_A] \) are the added mass, added damping and added stiffness matrices; \( \{ y_o \} \) is the nodal displacement, and \( \{ F_{\text{f}} \} \) represents the flow induced force on the cylinder.

In the current work, an elastically mounted rigid cylinder is taken as the research object. What we care about is the oscillation of the entire cylinder, instead of the stress and strain inside it. Consistent with Khalak’s experiment, we restrict the downstream motion of the cylinder. Therefore, the cylinder oscillates in a single degree of freedom in the cross flow direction.

The fluid force can be assumed to be a constant at an arbitrary small time step, and is solved by Reynolds-averaged Navier-Stokes (RANS) equations, which can be expressed in Cartesian coordinates as follows.
\[
\begin{align*}
\frac{\partial u_i}{\partial t} + \frac{\partial u_i u_j}{\partial x_j} &= -\frac{1}{\rho} \frac{\partial p}{\partial x_i} + \frac{\partial}{\partial x_j} \left( 2\nu \delta_{ij} - \frac{\partial u_j}{\partial x_j} \right) \\
S_y &= \frac{1}{2} \left( \frac{\partial u_i}{\partial x_i} + \frac{\partial u_j}{\partial x_j} \right) \\
\bar{u}_i &= \nu \left( \frac{\partial u_i}{\partial x_i} + \frac{\partial u_i}{\partial x_j} \right) + \frac{2}{3} k \delta_y
\end{align*}
\]

in which, \(v\) is kinematic viscosity of fluid, \(v_t\) is turbulent eddy viscosity and \(k\) is turbulent energy. The SST-k\(\omega\) turbulence model is used for modelling the turbulence. The specific parameter can be referred to the work of Prasanth et al.\[10\]. Previous study showed that the SST-k\(\omega\) model gives a good prediction of the adverse pressure gradient flows.

Several dimensionless parameters involved in this paper are Reynolds number \(R_e\), reduced velocity \(U_r\), lift coefficient \(C_L\), mass ratio \(m^*\), dimensionless response amplitude \(A^*\), and dimensionless frequency \(f^*\), which are defined as follows:

\[
R_e = \frac{U_{\infty} D}{v}, \quad U_r = \frac{U_{\infty}}{f_s D}, \quad C_L = \frac{2F_L}{\rho U_{\infty}^2 D h}, \quad m^* = \frac{m_s}{m_d}, \quad A^* = \frac{A_y}{D}, \quad f^* = \frac{f}{f_s}
\]

where \(U_{\infty}\) is the inflow velocity, \(D\) is the diameter of the cylinder, \(F_L\) is the vortex-induced lift, \(m_s\) is the total mass of the cylinder, \(m_d\) is the mass of water which displaced by the cylinder, \(A_y\) is the amplitude of the cylinder in the cross-flow direction, \(f\) is the characteristic frequency (including response frequency, vortex shedding frequency, excitation frequency, etc.), and \(f_s\) is the natural frequency of the cylinder in still water.

3. Case setup

3.1. Computational domain and boundary conditions

This simulation is conducted in the commercial software Workbench and CFX. A mass-spring-damping system is used. The mass ratio of the cylinder \(m^*=2.4\), the damping ratio \(\zeta=0.0045\), the diameter \(D=0.01\text{m}\) and the reduced velocity ranges from 2.18 to 14.18. Different from the test, we will fix the inflow velocity at 0.1m/s, that is, the Reynolds number is 1000. By changing the spring stiffness, the natural frequency of the system can be adjusted to achieve the purpose of adjusting the reduced velocity. In this way, it is not necessary to re-determine the grid and time step at each reduced velocity. In addition, fixing the Reynolds number to a lower value can greatly reduce the amount of calculation.

![Figure 1. The schematic of computational model.](image)

Prasanth's research\[11\] showed that when the length of the cylindrical wake region \(L_d \geq 22.5D\) and the transverse flow width \(H \geq 20D\), the result can be considered to be unaffected by the calculation.
boundary. This simulation set \( L_d=30D, H=20D \), and the calculation model is shown in Fig. 1. In fact, the computational domain has a certain thickness in the \( z \) direction, but only one layer of mesh is divided in this direction. Therefore, this is equivalent to a two-dimensional numerical calculation model. The inlet is defined as the velocity boundary; the outlet is the static pressure; the two faces in the thickness direction are set as the symmetry; the boundary on both sides of the water flow are the free slip wall; the cylindrical surface is the no-slip wall.

3.2. Mesh dependency test
Hexahedral grids are used in the whole flow field. In order to obtain accurate and stable results, the meshes near the boundary layer of the cylinder are refined. A diagram of the grid can be seen in Fig. 2. The thickness of the first layer of the grid near the cylinder \( \Delta y \) is estimated by eq. (8). In CFD calculation with SST model, \( y^+\approx1 \) is usually required. Since \( D \) and \( Re \) are both determined, \( \Delta y \) can be obtained directly. All the calculation model in current work, \( \Delta y=1\times10^{-4}m \), the maximum value of the \( y^+ \) is 1.4.

\[
y^+ = 0.172 \frac{\Delta y}{D} R_y^{0.9}
\]

Figure 2. The schematic of the grid.

Four sets of grids with similar quality but different numbers of cells and four different time steps are provided for verification. Lift coefficient and vortex shedding frequency of fixed cylinder are used as parameters to evaluate grid and time step. The results are shown in Table 1. When the time step is 0.0025s and the number of cells is 19440, the CFD calculation can be performed relatively accurately and efficiently.

Table 1. The results of different grids and time steps

| Number of elements | Time Steps | \( C_L \) | \( f_{0V} \) |
|--------------------|------------|----------|----------|
| 13970              | 0.001s     | 0.991    | 2.147    |
| 16374              | 0.001s     | 0.979    | 2.161    |
| 19440              | 0.001s     | 0.970    | 2.172    |
| 23544              | 0.001s     | 0.966    | 2.175    |
| 19440              | 0.0025s    | 0.970    | 2.171    |
| 19440              | 0.005s     | 0.969    | 2.168    |
| 19440              | 0.01s      | 0.966    | 2.139    |

4. Results and discussion

4.1. Results of vortex-induced vibrations
The vortex-induced vibration of a cylinder with reduced velocity \( U_r=2.18\sim14.18 \) was calculated by TW-FSI and ASOW-FSI methods, respectively. \( A^* \) and \( f^* \) were obtained by dimensionless the resulting data after the displacement curve reaches stability, and the results are shown in Fig. 3 and Fig. 4. Since the structural response frequency is equal to the vortex shedding frequency, both of them can be characterized by \( f^* \).
Compared with the experimental data, it is clear that this method accurately predicts the VIV, including the phenomenon of "lock-in". When the reduced velocity $U_r<4.6$, $f_{V0} < f_n$, the structure vibrates at the natural vortex shedding frequency $f_{V0}$. When $U_r > 4.6$, $f_{V0} > f_n$, the response frequency is restrained near $f_n$. When the reduced velocity increases to a certain limit, the "lock-in" phenomenon disappears. On the other hand, the amplitude of the vibration with the reduced velocity is approximately consistent with the experiment, but there is a large error in the left and right ends of the "lock-in" region. In fact, the dimensionless response amplitude of the cylinder can be divided into three branches—the initial branch, the upper branch, and the lower branch. But the numerical simulation results only show the initial branch and the lower branch. This can also be observed in the work of Guilmineau et al. [1] and Zhang et al. [2]. Guilmineau believes that the initial conditions of the numerical simulation will affect the prediction of the upper branch. The results of Zhang showed that the excitation of the upper branch is related to the Reynolds number, and the upper branch can be predicted at a large Reynolds number. In general, TW-FSI is considered to be an accurate numerical method for predicting flow-induced vibration.

The ASOW-FSI method has high accuracy in the non-"lock-in" range. In the initial branch of the experiment, the structural response can be predicted accurately even if the amplitude is very large. On the right side of the "lock-in" region, the prediction of the vibration amplitude is obviously closer to the experimental value than that of TW-FSI. However, in the "lock-in" range, this method fails.

The variation of wake vorticity and lift coefficient with reduced velocity is shown in Fig. 5 and Fig. 6. At the tail of the fixed cylinder, the wake presents a typical 2-single-vortex (2S) mode, that is, one vortex is released on both sides of the cylinder in each oscillation period. However, in the two-way coupling calculation, different reduced velocities correspond to different vortex shapes. In the initial branch, the vortex is a 2S mode (e.g. $U_r=2.18$). In the "lock-in" region, the vortex presents a 2-vortex-pair (2P) mode, which releases two vortices on a side of the cylinder every half cycle (e.g. $U_r=5.46$).
and $U_r=8.73$). Although the simulation failed to excite the upper branch, a special dual-row 2p vortex was observed when $U_r=5.46$. On the right side of the "lock-in" region, the shape of the vortex is close to 2S. Unlike the typical 2S mode, the position of the vortex release moves downstream for a certain distance (e.g. $U_r=14.18$). The above vortex shedding modes were classified in the manner proposed by Williamson et al.[12].

In the AWOW-FSI calculation, the fluid-induced force applied to the structure is obtained from the flow field around the fixed cylinder. Since the inflow velocity is constant, the lift at different reduced velocity is consistent. In the TW-FSI simulation, the flow field interacts with the structure field. When the natural frequency of the structure changes, the structural response and fluid-induced force will change accordingly. In order to facilitate the analysis of lift characteristics, we can regard the fixed restraint of the structure as a spring restraint with infinite stiffness, so the natural frequency of the system is infinite, $f_{V0}/f_n=0$. It can be seen that when the value of $f_{V0}/f_n$ increases from 0 to 1, the amplitude of the lift coefficient increases; when $f_{V0}/f_n>1$, the amplitude gradually decreases. However, the prediction of TW-FSI in some reduced velocity ranges is not accurate, so whether this rule is applicable to real VIV needs further discussion.

![Figure 6. Amplitude of lift coefficient.](image)

![Figure 7. Phase angle of lift and response.](image)

Figure 7 shows the phase angle between lift and cross-flow vibration. The results show that both methods can capture the hysteresis effect in VIV. When $f_{V0}/f_n<1$, the phase angle is 0; when $f_{V0}/f_n>1$, the time-history curve of the structural response shows a lag of nearly half a period. The variation of the phase angle with $f_{V0}/f_n$ is consistent with the conclusion of Han et al.[13].

According to the above results, the excitation frequency has a great influence on the structural response, while the amplitude of the excitation force is a secondary factor. In addition, vortex shedding mode is also closely related to the characteristic frequency for the VIV of the cylinder. The main reason for the difference between the two methods in the whole reduced velocity range is that the ASOW-FSI method can not predict the "lock-in".

In order to explore the cause of the failure of ASOW-FSI method in the "lock-in" region, it is necessary to analyze the mechanism of the "lock-in". In fact, in recent years, a large number of studies[14, 15] have shown that flutter is the root cause of frequency locking. Essentially, it is a self-excited vibration phenomenon. However, the conditions and specific process of "lock-in" need to be further studied[16].

4.2. Discussion on "lock-in" mechanism

To investigate the mechanism of the "lock-in" in VIV, the cylinder was forced to oscillate in the law of response obtained by TW-FSI. Thus, the influence of structural response on flow field is clarified. The amplitude and frequency of forced oscillation are shown in Fig. 8. Figure 9∼Figure 11 display the vortex shedding frequency, lift and wake vorticity in forced oscillation, respectively. Since the TW-
FSI method did not predict the upper branch, the result of $U_r=5.46$ may be quite different from the actual situation. So it will not be considered in the subsequent analysis.

![Figure 8. Amplitude and frequency of forced oscillation.](image)

Figure 8. Amplitude and frequency of forced oscillation.

![Figure 9. Vortex shedding frequency of forced oscillation.](image)

Figure 9. Vortex shedding frequency of forced oscillation.

![Figure 10. Amplitude of lift coefficient of forced oscillation.](image)

Figure 10. Amplitude of lift coefficient of forced oscillation.

When $U_r=2.18 \sim 10.91$, the frequency of vortex shedding is consistent with that of oscillation, and the lift and vortex shape are the same as those in TW-FSI (except $U_r=4.36$). In fact, the particularity of lift and vortex shape of $U_r=4.36$ is related to the phase angle. In forced oscillation, the mechanism of phase angle between excitation force and structural displacement is completely different from that of VIV. When the oscillation frequency is greater than the natural vortex shedding frequency, it is "in phase", otherwise it is "out of phase". Its details can be found in Bishop et al. [17].

![Figure 11. Wake vorticity in forced oscillation.](image)

Figure 11. Wake vorticity in forced oscillation.
Based on the above results, it can be inferred that when the vibration frequency $f$ is close to the natural vortex shedding frequency $f_{V0}$ or the amplitude is large, the structure vibration has the ability to dominate the vortex shedding. This rule can well explain the cause of "lock-in" in VIV:

When $U_r \approx 2.18 \sim 4.36$, $f_{V0} / f_n < 1$, the natural mode of the structure can not be excited. The cylinder vibrates according to $f_{V0}$, and the dominant ability of the structure does not play a role. When $U_r = 6.54 \sim 10.91$, $f_{V0} / f_n > 1$, the natural mode of the structure is more easily excited. The response frequency and vortex shedding frequency are limited to near $f_n$, meanwhile, the vortex mode and phase angle are changed.

Finally, the situation of $U_r > 10.9$ is analyzed. In the forced oscillation, the vortex shedding frequency is equal to $f_{V0}$, and the lift amplitude is also consistent with that of the fixed cylinder. In addition, the wake vortex has a typical 2S mode. All these results indicate that the structure has little effect on vortex shedding. It can be inferred that in this reduced velocity range of the actual VIV, the vortex shedding frequency should be equal to $f_{V0}$. This feature has not been predicted in current two-way coupling simulation. Therefore, the accuracy of TW-FSI was lower than that of ASOW-FSI method in the range of large reduced velocity.

5. Conclusion
In order to compare the similarities and differences between the ASOW-FSI and TW-FSI methods in the numerical calculation of flow-induced vibration, a series of simulations were carried out, and the following conclusions were obtained.

ASOW-FSI can accurately predict the structural response in the non-"lock-in" region, even closer to the experimental results than TW-FSI method. The main reason for the difference between the two methods in the whole reduced velocity range is that the ASOW-FSI method can not predict the "lock-in".

When the vibration frequency $f$ is close to the natural vortex shedding frequency $f_{V0}$ or the amplitude is large, the structure vibration has the ability to dominate the vortex shedding. If the natural mode of the structure is excited at this time, "lock-in" will occur. This result provides a supplement to the research on the mechanism of "lock-in".

In general, the ASOW-FSI method has high accuracy for forced vibration, and can be applied to the case of large deformation. The disadvantage is that it cannot predict self-excited vibrations such as "lock-in". Therefore, it is necessary to judge whether there is the possibility of self-excited vibration before using this method.

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Nomenclature

| Symbol | Description |
|--------|-------------|
| $A^*$  | dimensionless response amplitude |
| $C_L$  | lift coefficient |
| $D$    | diameter of the cylinder (m) |
| $f$    | characteristic frequency (HZ) |
| $f^*$  | dimensionless frequency |
| $f_n$  | natural frequency of the cylinder in still water (HZ) |
| $f_{V0}$ | vortex shedding frequency of the fixed cylinder (HZ) |
| $m^*$  | mass ratio |
| $R_e$  | Reynolds number |
| $U_r$  | reduced velocity |
| $\Delta \phi$ | Phase angle between lift and cross-flow vibration (°) |
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