Numerical Simulation of Vortex-induced Vibration for A Real Size Drilling Riser System with Auxiliary Lines

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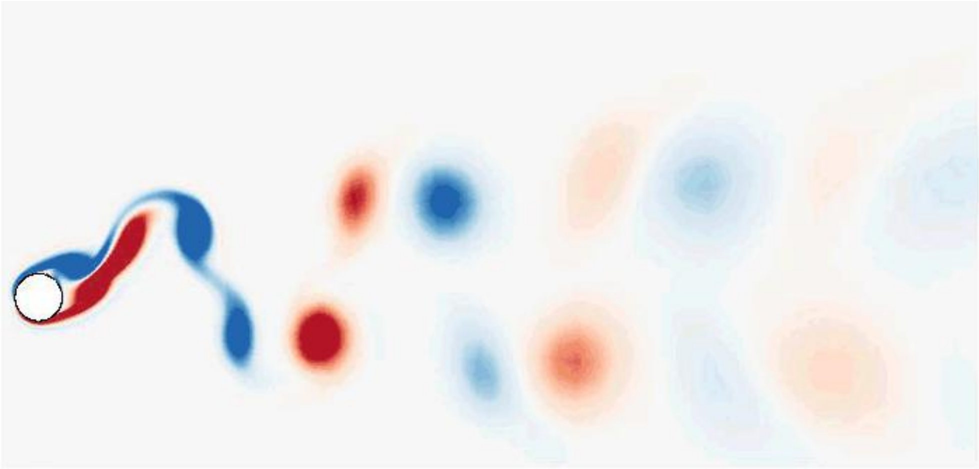
Introduction

- When the vortex shedding frequency is close to the natural frequency of structures, vibrations could be induced, which is called vortex-induced vibration(VIV).
- Fatigue failure may be caused by VIV, like risers in drilling platforms.

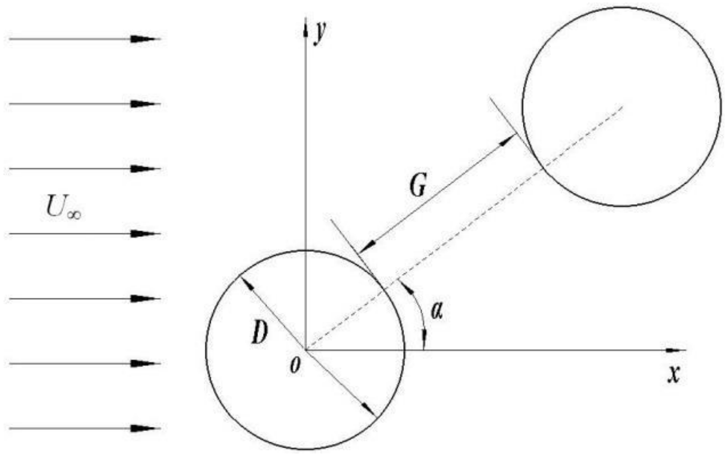


Vortex-induced vibration

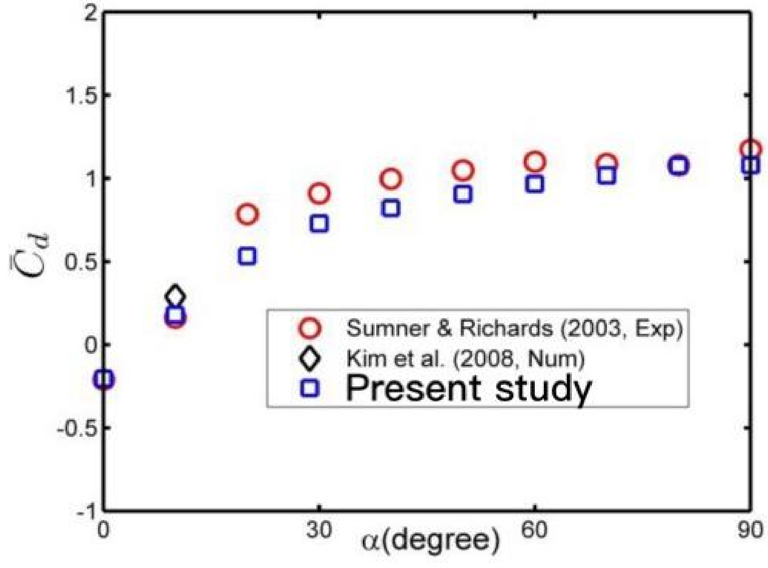
Drilling platforms

Introduction

- Flow past two equal diameter static cylinders at $Re=3.2\times10^4$.
- $G/D=1$, $\alpha=0°\sim90°$, k-ω turbulence model
- The mean drag coefficient changes with the angle of attack.



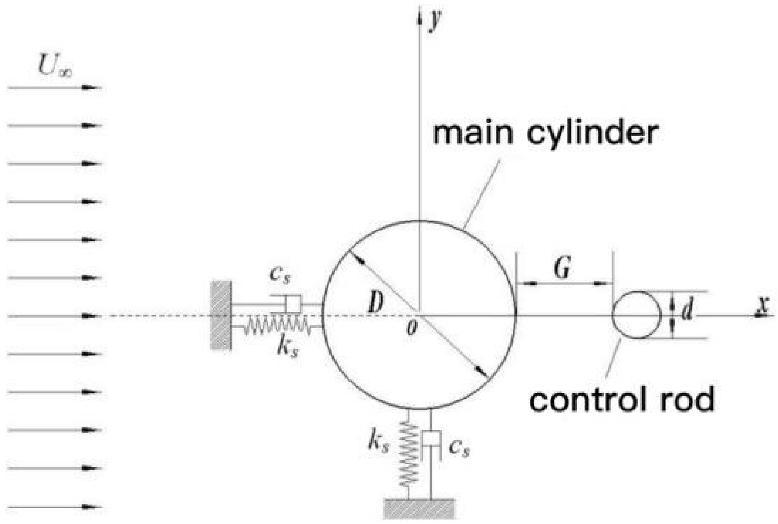
The model for flow past double circular cylinders in Wu (2017).



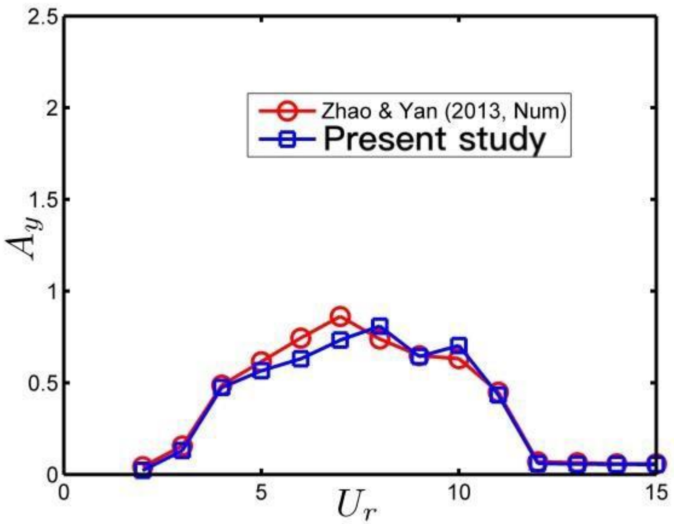
The mean drag coefficient of downstream cylinder in Wu (2017).

Introduction

- Tandem cylinders can vibrate in in-line and cross flow directions synchronously at Re=250 based on the upstream cylinder.
- $m^*=2$, $c_s=0$, $d/D=0.2$, $G/D=0.2$
- Different branches (initial branch, upper branch and lower branch) were obtained.
- $A_{ymax}=0.9$ (Ur=7.5)



The model for VIV of two circular cylinders in Wu (2017).



Cross flow vibration amplitude in Wu (2017).

Introduction

- The vortex-induced vibration (VIV) study of real-size drilling riser system has not been seen in literature yet. Here we further simulate the VIV response for this real-size drilling riser system in service in the South China Sea by the secondary development of OpenFOAM platform based on pimpleDyMFoam.

Drilling riser with auxiliary lines

Contents

Present targets

- Simulate the vibration response amplitudes of the main cylinder in various incoming angles of attack with reduced velocities.

- Study the characteristics of vibration response of the riser system with auxiliary lines.

Contents

Formulation and numerical method

- The analytical form of the governing equations of flow of viscous incompressible fluid can be expressed as follows, i.e. RANS model:

$$\frac{\partial U_i}{\partial x_i} = 0$$

$$\rho \frac{\partial U_i}{\partial t} + \rho \frac{\partial}{\partial x_i}\left(U_j U_i\right) = \mu \frac{\partial^2 U_i}{\partial x_j \partial x_j} - \frac{\partial p}{\partial x_i} + \frac{\partial \tau_{ij}}{\partial x_j}$$

- Eddy viscosity model (EVM) was used here, the Reynolds stresses were computed with the Boussinesq expression:

$$\tau_{ij} = \mu_t \left(\frac{\partial U_i}{\partial x_j} + \frac{\partial U_j}{\partial x_i} \right) - \frac{2}{3} \rho k \delta_{ij}$$

$$\mu_t = \frac{\rho k}{\omega}$$

Formulation and numerical method

- And k-ω turbulence model was used here:

$$\frac{\partial(\rho k)}{\partial t}+\frac{\partial(\rho k U_i)}{\partial x_i}=\frac{\partial}{\partial x_i}\left[\left(\mu+\frac{\mu_t}{\sigma_k}\right)\frac{\partial k}{\partial x_i}\right]+P_k-\beta^*\rho k\omega$$

$$\frac{\partial(\rho\omega)}{\partial t}+\frac{\partial(\rho\omega U_i)}{\partial x_i}=\frac{\partial}{\partial x_i}\left[\left(\mu+\frac{\mu_t}{\sigma_\omega}\right)\frac{\partial\omega}{\partial x_i}\right]+P_\omega-\beta_1\rho\omega^2$$

$$P_k=2\mu_t S_{ij}S_{ij}-\frac{2}{3}\rho k\frac{\partial U_i}{\partial x_j}\delta_{ij}$$

$$P_\omega=\gamma_1\left(2\mu_t S_{ij}S_{ij}-\frac{2}{3}\rho\omega\frac{\partial U_i}{\partial x_j}\delta_{ij}\right)$$

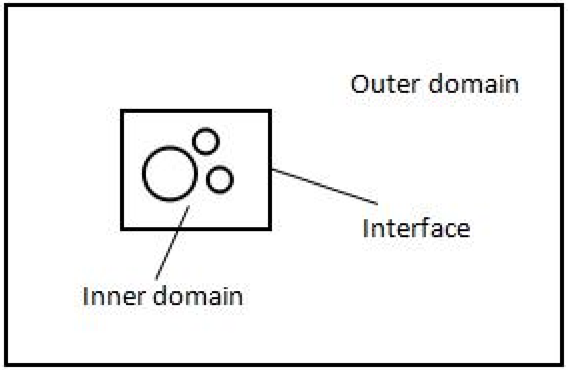
- The cylinders are rigid connected, i.e. all cylinders move synchronously. Spring oscillator model was used here:

$$m_s\ddot{u}+c_s\dot{u}+k_s u=F$$

Formulation and numerical method

- Dynamic mesh solution module：Inner and outer domains. In the inner domain, the grid and the cylinder system move synchronously, which ensures that the grid around the cylinder system remains unchanged during the movement. The mesh deformation of the outer domain is obtained by solving the Laplacian equation based on the displacement of the interface.

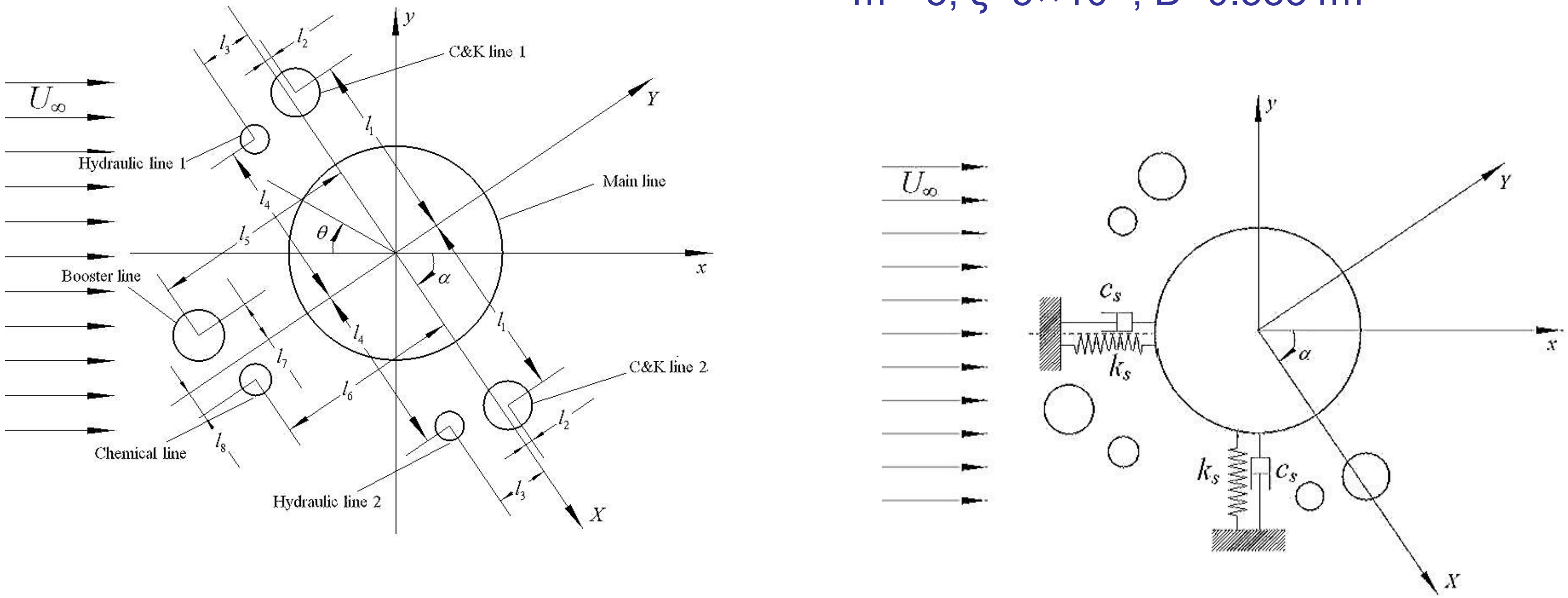
- Quantity of mesh: 9.2×10^4; computational domain: -15D~50D in in-line, -15D~15D in cross flow.



The division of the computation domain

Formulation and numerical method

- In the real size drilling riser system, the auxiliary lines have different diameters and distribute asymmetrically around the main line.
- The flow is investigated based on the different incidence angles at $Re=3.5\times10^4$.The model can vibrate in the in-line and cross flow directions simultaneously.
- $m^*=3$, $\zeta=5\times10^{-4}$, $D=0.5334$m



Sketch of the model for a real drilling riser system

Formulation and numerical method

D	Cylinder diameter
U	Free stream velocity
ρ	Fluid density
m^*	Mass ratio($m^*=4m_s/(\pi\rho D^2)$)
ζ	Damping ratio
Re	Reynolds number(Re=UD/v)
Ur	Reduced velocity ($U_r=U/f_nD$)
A_y	Cross-flow dimensionless amplitude
$A_{y,rms}$	Cross-flow dimensionless RMS amplitude
α	Incoming angle of attack
θ	Circumferential angle of the main line
m_s	Mass of the riser system
k_s	Stiffness coefficient of the riser system
c_s	Damping coefficient of the riser system
G	Gap between cylinders
k	Kinetic energy
ω	Turbulent dissipation rate
μ	Coefficient of kinetic viscosity

Main symbol description

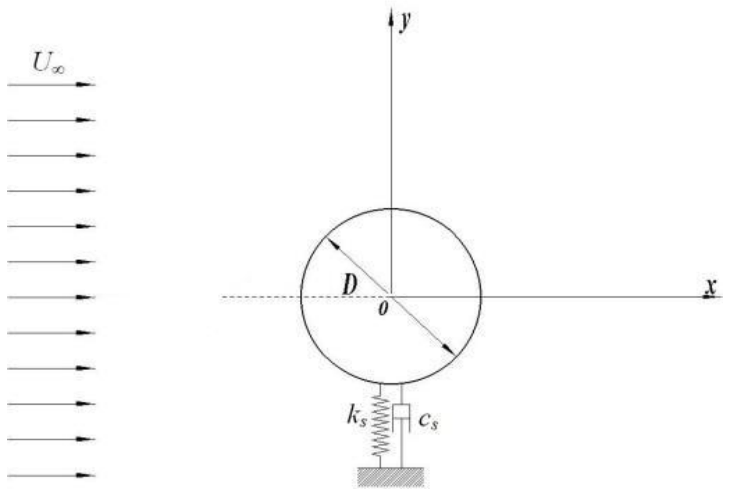
C&K line	0.285D
Hydraulic line	0.119D
Booster line	0.238D
Chemical line	0.125D
l_1	0.973D
l_2	0.12D
l_3	0.212D
l_4	0.902D
l_5	0.928D
l_6	0.857D
l_7	0.154D
l_8	0.163D

The diameters of lines and geometrical parameters in the riser system

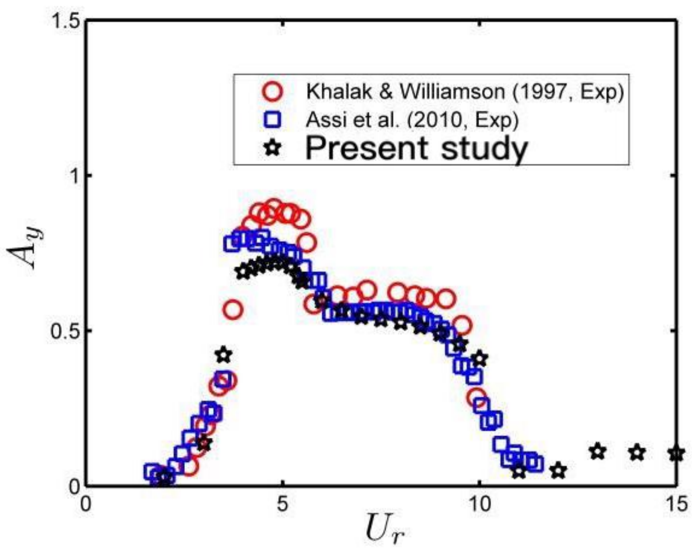
Formulation and numerical method

Validation of present numerical method

- Single cylinder can vibrate in cross flow direction.
- m*=2.4, ζ=0.0059, Re=10000, K-ω turbulence model
- Different branches (initial branch, upper branch and lower branch) were obtained.
- A_{ymax}=0.78 (Ur=5.0)



The model for VIV of single cylinder with one degree of freedom

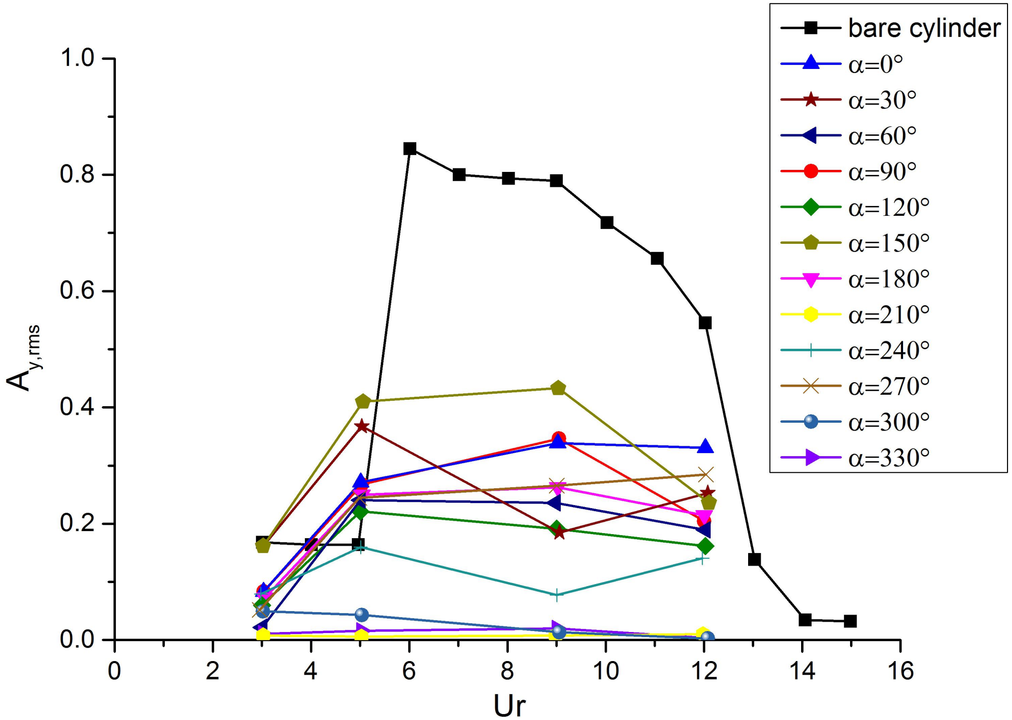


Amplitude response

Contents

Results and discussions

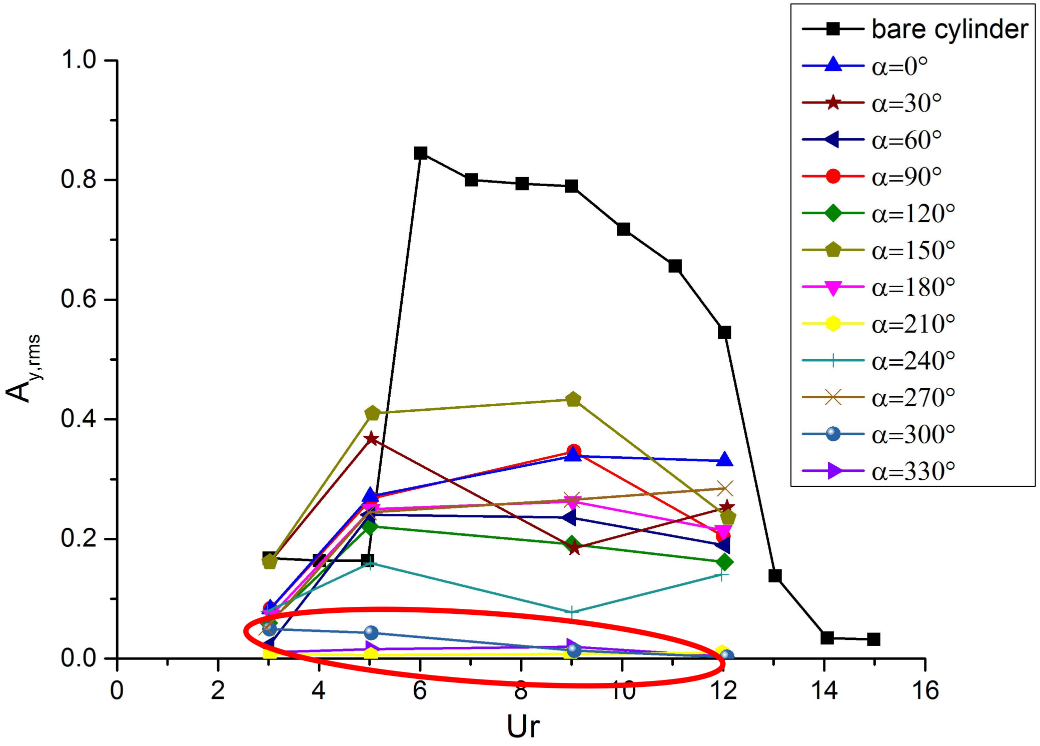
- The auxiliary lines can effectively reduce the vibration amplitudes in most reduced velocities, and the effect is greatly related to the incoming angle of attack.
- At each incoming angle of attack, the vibration amplitude of the riser system varies with the changing reduced velocity.



Amplitude response for riser system and bare cylinder

Results and discussions

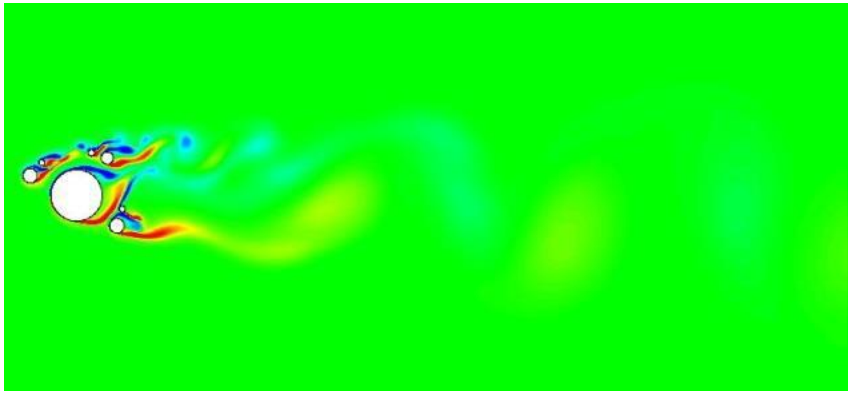
- The upper branch of the riser system does not appear at any angle of attack, and its amplitude is much smaller than that of a single cylinder, especially at ***210°***, ***300°*** and ***330°***, the amplitudes are almost close to zero.



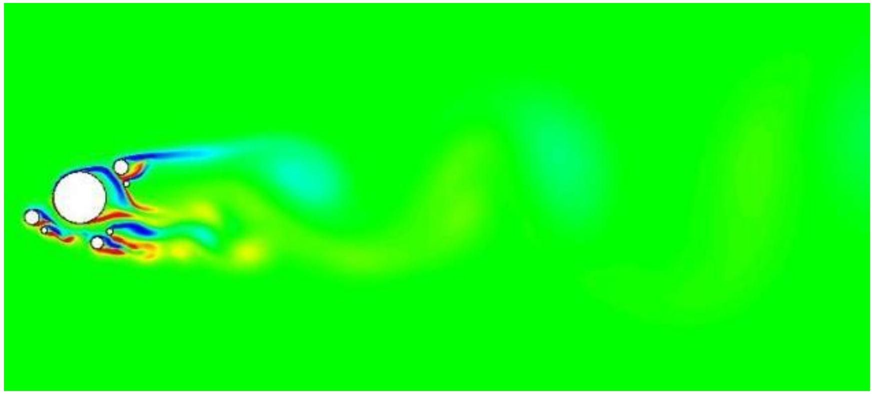
Amplitude response for riser system and bare cylinder

Results and discussions

- The vortices near the riser system are affected by multiple auxiliary lines and the vortices are irregularly distributed, so the vibration responses are suppressed.
- At 210° and 330°, the shear layer on the main line is confined to a very small area by the downstream auxiliary lines, and disappears quickly due to the interaction, so obvious discrete vortices in the riser system wake cannot be observed and the vibration is very weak.



α=210°, Ur=3

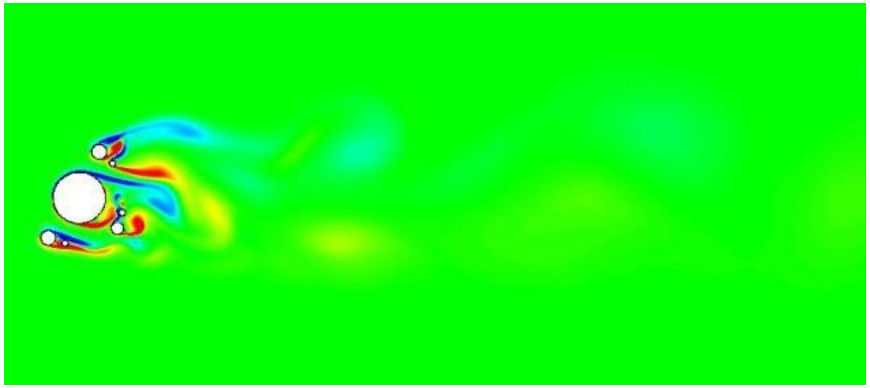


α=330°, Ur=3

The vorticity contour at the reduced velocity corresponding to the maximum amplitude of the system under different position angles

Results and discussions

- At α=300°, the shear layer on the underside of the main line cannot be discharged due to the influence of the downstream auxiliary lines. Due to the weak interaction between the shear layers on both sides of the main line, the discharge of the shear layer on the other side is also affected. Therefore, the vortex-induced vibration amplitude of the riser system is very small.



α=300°, Ur=3

The vorticity contour near the riser system

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Conclusions

- **The auxiliary lines can effectively reduce the vibration amplitudes, and the effect is greatly related to the incoming angle of attack.**
- **At each incoming angle of attack, the vibration amplitude of the riser system varies with the changing reduced velocity.**
- **The main line can not shed vortex due to the clamping of the downstream auxiliary lines, so the vibration responses are suppressed.**

Thank you