

Article

Two-Degrees-of-Freedom CFD Simulation of Aeolian Wind Energy Input to Conductors and Integrated Anti-Vibration Solutions

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Abstract

Aeolian vibrations of overhead transmission conductors under light wind excitation can cause fatigue damage at line clamps, threatening grid safety. Accurately evaluating wind energy input is key for assessing vibration intensity and designing anti-vibration devices. This paper employs CFD overlapping mesh technology to establish a two-degrees-of-freedom fluid–structure interaction model of the conductor. Through numerical simulations at different Reynolds numbers, a cubic polynomial relationship between wind energy input power and amplitude ratio is derived, and a wind turbulence intensity correction factor is introduced. Using a long-span conductor as a case study, the fitted wind energy input is substituted into the energy balance equation to calculate dynamic bending strain under four conditions: no control, only vibration dampers, only damping wires, and a combined scheme. The results show that using either vibration dampers or damping wires alone cannot reduce dynamic bending strain below 200 $\mu\epsilon$, while the combined scheme reduces the maximum strain within the critical frequency range to 113.09 $\mu\epsilon$. This verifies the effectiveness of the fitted wind energy input formula in engineering anti-vibration design.

Keywords: aeolian vibration; wind energy input power; CFD; two-degrees-of-freedom; combined anti-vibration



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1. Introduction

Overhead transmission conductors subjected to long-term outdoor environmental exposure are prone to aeolian vibrations induced by steady laminar winds. Aeolian vibration is the most common cause of conductor fatigue and strand breakage, posing a serious threat to the safe operation of power grids [1,2]. Accurately describing the relationship between wind energy input power and conductor amplitude is a key prerequisite for assessing vibration severity and designing anti-vibration devices.

Currently, wind energy input power is typically derived from classical wind tunnel test curves based on rigid circular cylinders [3,4]. However, these curves are mostly obtained from single-degree-of-freedom tests and do not account for the coupling effect

of actual conductor vibrations on the surrounding flow field [5,6]. With the development of computational fluid dynamics (CFD), numerical simulation has become an efficient tool for studying fluid–structure interaction problems [7–9]. Li et al. [10] investigated the aeolian vibration of a circular cylinder model using an “a priori amplitude assumption” and fitted wind power input expressions. Vaja et al. [11] established an analytical model for multi-resonant Stockbridge dampers of overhead transmission lines and verified the model via finite element simulation and experiments, yet the study failed to discuss vortex-induced vibration characteristics of conductors under multiple degrees of freedom. Most existing CFD studies are based on single-degree-of-freedom vibration models [12], ignoring the coupling effect between transverse and vertical vibrations. In reality, the conductor motion is a typical two-degrees-of-freedom coupled vibration characterized by a figure-eight trajectory [5]. Therefore, a more accurate two-degrees-of-freedom model is necessary to obtain the wind energy input characteristics.

Furthermore, the influence of regional wind turbulence on wind energy input should be considered [13]. The International Council on Large Electric Systems (CIGRE) recommends the use of a non-uniformity correction factor [14], but traditional wind energy input models rarely incorporate this correction, limiting their applicability to different geographic regions.

Regarding anti-vibration measures, vibration dampers are widely used due to their good performance [15,16], while damping wires, as distributed vibration absorbers, offer a broader frequency protection range and are especially suitable for long-span lines [11,17]. For long-span, large-cross-section conductors, practical operating experience indicates that a single anti-vibration measure often fails to keep dynamic bending strain below the safety limit [18,19]. However, engineering validation of combined anti-vibration schemes remains insufficient, and systematic theoretical and computational verification based on an accurate wind energy input model is lacking.

To address these issues, this study conducts two main tasks: (1) Using CFD overlapping mesh technology [5,20], a two-degrees-of-freedom fluid–structure interaction model of the conductor is established. Numerical simulations at different Reynolds numbers are performed, and a cubic polynomial relationship between wind energy input power and amplitude ratio is fitted. A wind turbulence intensity correction factor [14] is introduced to obtain a wind energy input expression that is more applicable to practical engineering. (2) Taking a long-span conductor as a case study, the fitted wind energy input power is incorporated into the energy balance method [21,22]. Dynamic bending strains are compared under four conditions: no vibration control, vibration dampers only, damping wires only, and a combined scheme. The results verify the practicability of the proposed wind energy input model in anti-vibration design and provide a theoretical basis for anti-vibration optimization of similar lines.

2. Theoretical Background and Methods

2.1. Calculation of Dynamic Bending Strain

The dynamic bending strain at the clamps of an overhead line is a key indicator for evaluating its fatigue life. The dynamic bending strain at the clamp location can be calculated based on the maximum amplitude at the antinode, A_0 . When the conductor vibrates, it can be regarded as a clamped system with a span equal to half the wavelength $\lambda/2$, where deformation near the clamping points is influenced by the conductor's bending stiffness. If the vertical reaction force exerted by the vibration wave on the suspension

point can be determined, the stress and strain at the suspension point can be calculated using the static bending stress formula [15]:

$$\sigma = \pm \frac{2\pi}{\lambda} A_0 E d_A \sqrt{\frac{T}{EI}} \quad (1)$$

$$\varepsilon = \frac{\sigma}{E} = \pm \frac{2\pi}{\lambda} A_0 C \sqrt{\frac{T}{EI}} \times 10^6 \text{ (}\mu\varepsilon\text{)} \quad (2)$$

where λ_n corresponds to the vibration wavelength under the n th-order mode, satisfying $\lambda_n = 2L/n$. The vibration amplitude of the lower-order modes accounts for the largest proportion [23]; therefore, the first-order mode function is selected for the self-damping calculation of the conductor. In Equation (2), the value of C represents the distance from the outer surface of the conductor to the neutral axis of bending. Its theoretical range generally lies between the conductor radius $D/2$ and the radius of the outermost aluminum strand $d_A/2$. The specific value is determined by the slippage state between the strands. According to the literature [24], $C = 0.35D$ is adopted, resulting in a dynamic bending strain that is on the safe side. The formula for the elastic modulus E is [25]

$$E = (0.85 \sim 0.91) \frac{E_S A_S + E_A A_A}{A} \quad (3)$$

where E_S and E_A are the elastic moduli of steel and aluminum; and A_S and A_A are the total cross-sectional areas of the steel and aluminum wire strands.

The effective bending stiffness EI of a power transmission conductor is [26]:

$$EI = 0.4 \left(\frac{\pi}{64} \right) \left\{ E_a d_a^2 \left[d_a^2 + \sum_{i=1}^{N_a} n_{ai} (d_a^2 + 2 \times D_{ai}^2) \right] + E_e \times d_e^2 \times \sum_{i=1}^{N_e} n_{ei} (d_e^2 + 2 \times D_{ei}^2) \right\} \quad (4)$$

where $E_a = 193.1 \text{ kN/mm}^2$, $E_e = 69 \text{ kN/mm}^2$ (for ACSR or AACSR conductors); d_a is the diameter of the steel core region wire (mm); d_e is the diameter of the sheath region wire (mm); n_{ai} is the number of core wires in the i -th layer of the steel core region; n_{ei} is the number of core wires in the i -th layer of the sheath region; D_{ai} is the outer diameter of the core wire in the i -th layer of the steel core region (mm); D_{ei} is the outer diameter of the core wire in the i -th layer of the sheath region (mm); N_a is the number of layers in the steel core region (excluding the central core wire); and N_e is the number of layers in the aluminum sheath region.

2.2. Energy Balance Method

The key to the energy balance method lies in determining the wind input power P_W , the power dissipated by the conductor's self-damping P_C , and the power dissipated by the vibration damping device P_D . All three are functions of the maximum amplitude at the antinode A_0 ; and solving these equations simultaneously gives A_0 . The relationship between the power values of these three factors and the conductor amplitude is shown in Figure 1.

When the conductor amplitude reaches A_a , the wind energy input power balances the power dissipated by the conductor's self-damping. If the transmission line can remain in a safe state without fatigue damage under these conditions, there is no need to install vibration dampers. In this case, the expression for the energy balance method is

$$P_W = P_C \quad (5)$$

If the transmission line is already in a state of fatigue damage at an amplitude of A_a , it is necessary to consider installing vibration dampers to shift the energy balance point from a to b. If the line is in a safe state at this point, the installation of vibration dampers is considered appropriate. In this case, Equations (5) become

$$P_W = P_C + \sum P_D \quad (6)$$

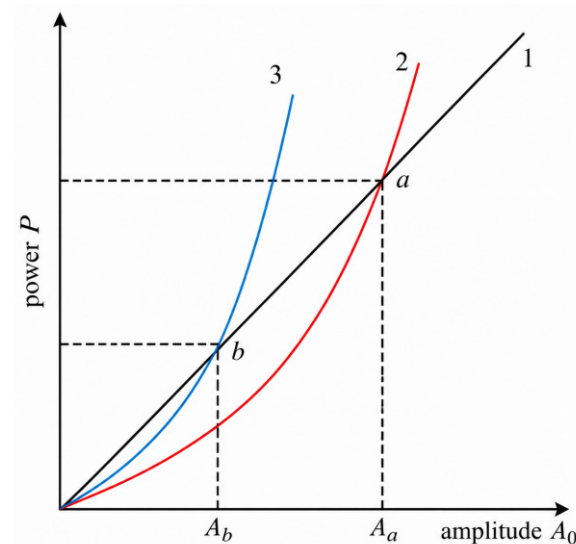


Figure 1. Curve 1 represents the input wind power; Curve 2 represents the power dissipated by the transmission line's self-damping; and Curve 3 represents the sum of the power dissipated by the transmission line and the vibration damper system.

2.3. Power Consumption Due to Self-Damping

According to the guidelines for calculating conductor self-damping power provided by the IEEE Power and Energy Society, domestic research on single-conductor self-damping often uses the self-damping power calculation expressions derived from the conductor vibration equations by Lu M L and Chan J K [21]

$$P_C = 1.07 \times 10^6 E_{eq} \sqrt{k_D k_s} K_0 L \frac{D^{4.5} f^{6+\beta} A_0^{2.5}}{V_c^5} \quad (7)$$

In the equation, E_{eq} is the equivalent elastic modulus; k_D is the equivalent diameter coefficient, which is empirically set to 0.54 for aluminum-clad steel strands (ACSR) and 0.65 for steel strands; k_s is the reduction factor for maximum bending stiffness, which can be set to 0.5; K_0 and β are parameters related to the conductor structure and material that must be determined experimentally, based on Kinectrics [27] curve fitting of self-damping data for different types of conductors, for aluminum-clad steel stranded wire (ACSR): take $K_0 = 0.0042$, $\beta = -0.4256$; for aluminum stranded wire (ACC): take $K_0 = 0.0035$, $\beta = -0.4332$; A_0 is the single amplitude of the conductor; V_c is the wave velocity in the transmission conductor, which is related to the catenary constant C , $V_c = \sqrt{Cg}$. The equivalent elastic modulus can be approximated using Equation (8)

$$E_{eq} = k_E \frac{E_S A_S^4 + E_A A_A^4}{A_S^4 + A_A^4} \quad (8)$$

where k_E is a correction factor; 0.75 for steel-core aluminum stranded wire and 0.70 for steel stranded wire.

The catenary constant C represents the shape of the curve and the tension of the transmission conductor; the larger the value, the flatter the curve; the smaller the value, the steeper the curve. It is expressed by Equation (9)

$$C = \frac{\sigma_0}{\gamma} \quad (9)$$

where σ_0 is the stress on the conductor under operating conditions (at the midspan); γ is the self-weight load ratio acting on the conductor, $\gamma = mg/A$.

2.4. Power Consumption of the Anti-Vibration Device

The relationship between the power dissipated by the vibration damping hammer and the wave belly amplitude of the conductor is expressed as [28]

$$P_{D1} = \frac{1}{4} \omega^2 \left(2\sqrt{Tm} + EI\omega^2 \sqrt{(m/T)^3} \right) \frac{1 - |\beta|}{1 + |\beta|} A_0^2 \quad (10)$$

where β is the reflection coefficient.

The expression for the energy dissipation power of the damping wire is consistent with that of the self-damping power of the transmission line, and can be written as [29]

$$P_{D2} = 1.07 \times 10^6 E_{eq2} \sqrt{k_D k_s} K_0 \frac{D_2^{4.5} f^{6+\beta} A_2^{2.5}}{V_{c2}^5} + 8.7 \times 10^4 E_{eq2}^{0.25} \sqrt{k_D} K_0 m_2^{0.75} \frac{D_2^{1.5} f^{3.5+\beta} A_2^{2.5}}{V_{c2}} \quad (11)$$

Let the contact points between the damping wire and the conductor be x_1 and x_2 , respectively. Based on the relationship that the displacements, velocities, and accelerations of the conductor and the damping wire are equal at these two points, and by combining the vibration equations of the two systems, the ratio of the damping wire's amplitude A_2 to the conductor's amplitude A_0 can be determined

$$\frac{A_2}{A_0} = \sqrt{\left[\frac{\sin \frac{n\pi x_2}{L} - \cos \frac{n\pi(x_2-x_1)}{L_2} \sin \frac{n\pi x_1}{L}}{\sin \frac{n\pi(x_2-x_1)}{L_2}} \right]^2 + \sin^2 \frac{n\pi x_1}{L}} \quad (12)$$

3. CFD-Based Wind Energy Input Power Fitting

3.1. Computational Model and Boundary Conditions

This section focuses on the AACSR/EST-500/230 steel-core aluminum stranded conductor, which is widely used in long-span transmission line projects. A three-dimensional CFD model was established using ANSYS Fluent 2024 to investigate the aerodynamic characteristics of the conductor. The SST k-omega turbulence model was employed. A turbulence intensity of 5% was specified at the velocity inlet as the inflow boundary condition. The air density and dynamic viscosity were set to 1.225 kg/m^3 and $1.789 \times 10^{-5} \text{ Pa}\cdot\text{s}$, respectively. The local turbulence quantities within the computational domain were determined by solving the transport equations of the SST k-omega turbulence model.

The computational fluid domain had dimensions of $40 D \times 20 D$, where D represents the conductor diameter of 35.2 mm. To minimize the influence of the computational boundaries on the flow around the conductor, the inlet boundary was located $10 D$ upstream of the conductor center, while the outlet boundary was positioned $30 D$ downstream. The upper and lower boundaries were each placed $10 D$ away from the conductor center. The computational domain and mesh configuration are shown in Figure 2. A minimum mesh size of $0.005 D$ was employed in the near-wall region to accurately resolve the boundary layer [20]. The unsteady simulations were performed using the pressure-based solver in

ANSYS Fluent, and the maximum Courant number was maintained below 5.66 throughout the computation.

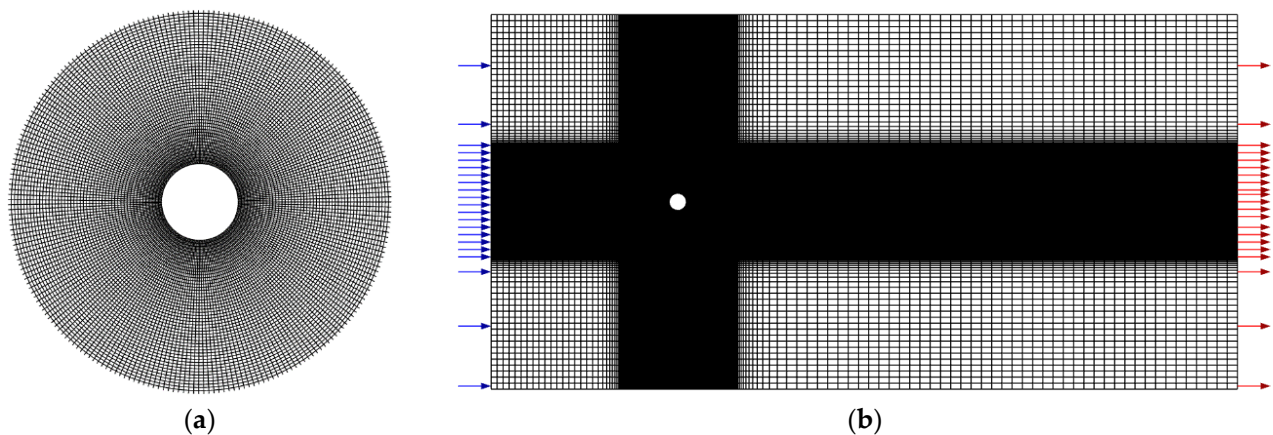


Figure 2. Schematic diagram of the computational domain and mesh for the flow field: (a) is the component mesh; (b) is the overlapping mesh.

Based on the incoming flow conditions and the near-wall grid spacing, the dimensionless wall distance y^+ is theoretically evaluated using the standard wall-shear stress relation. The resulting y^+ values are below 1.0 across the vast majority of the cylinder surface, with a minimum of approximately 0.3 at the stagnation point and a maximum of approximately 0.9 near the separation zone.

3.2. Simulation of Aeolian Vibration in a Single Conductor

Air velocity $V = 0.4979$ m/s, corresponding Reynolds number $Re = 1200$; at this point, the flow field is in a subcritical state. A massless, undamped ($\xi = 0$) cylindrical model with a mass of $m = 0.05767$ kg is used. To induce resonance in the wire, the natural angular frequency of the wire is set to $\omega = 17.773$ rad/s, which is determined from the reduced velocity $V_r = V/(fD) = 5.0$ using the prescribed inflow velocity $V = 0.4979$ m/s and conductor diameter $D = 35.2$ mm, that is, $\omega = 2\pi f = 2\pi V/(V_r D)$. CFD iterative calculations are performed using a time step of 0.002 s, and the wire reaches a steady-state vibration approximately 25 s later. The extracted velocity contour plot is shown in Figure 3. From the figure, the formation and shedding of vortex structures in the downwind region of the wire can be observed. The lift coefficient and drag coefficient curves are shown in Figure 4. Due to the periodic separation of vortices, the lift coefficient exhibits a sinusoidal oscillation with a mean value approaching zero, while the drag coefficient fluctuates slightly around 1.86; both demonstrate the oscillation patterns characteristic of Karman vortices. The displacement time histories in the Y and X directions are shown in Figure 5. Figure 6 depicts the motion trajectory of the wire under a stable vibrational figure-8-like motion mode with a double-loop characteristic.

A Fourier transform was performed on the obtained data, and the following equation was used:

$$f_s = S_t \frac{V}{D} \quad (13)$$

where f_s is the vortex-induced frequency; and S_t is the Strouhal number.

The Strouhal number S_t was calculated to be 0.204. This parameter agrees with the reference data reported in the literature by Norberg [8]. Based on the verification of the computational model's reliability, research on the wind-induced power input of a single conductor was continued.

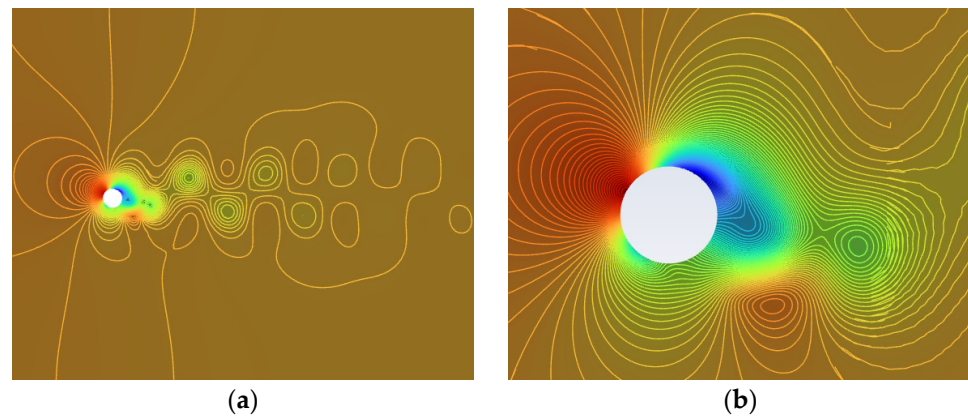


Figure 3. Velocity contour plot of the flow field when $t = 40$ s: (a) velocity contour plot of the wire's Karman vortex street; (b) velocity vector plot near the wire.

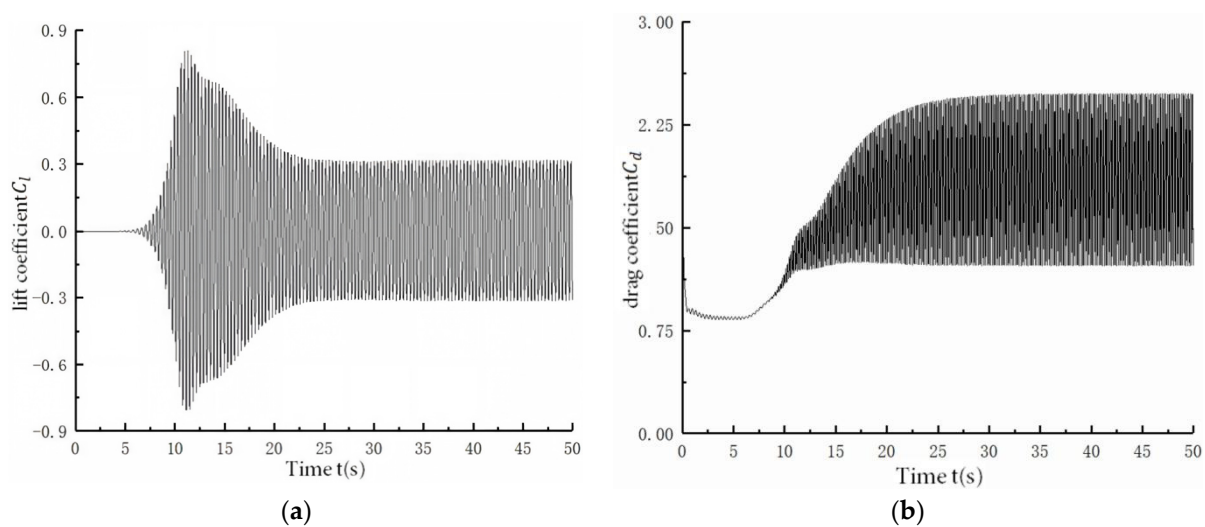


Figure 4. Time-history plots of lift coefficient and drag coefficient: (a) Lift coefficient; (b) Drag coefficient.

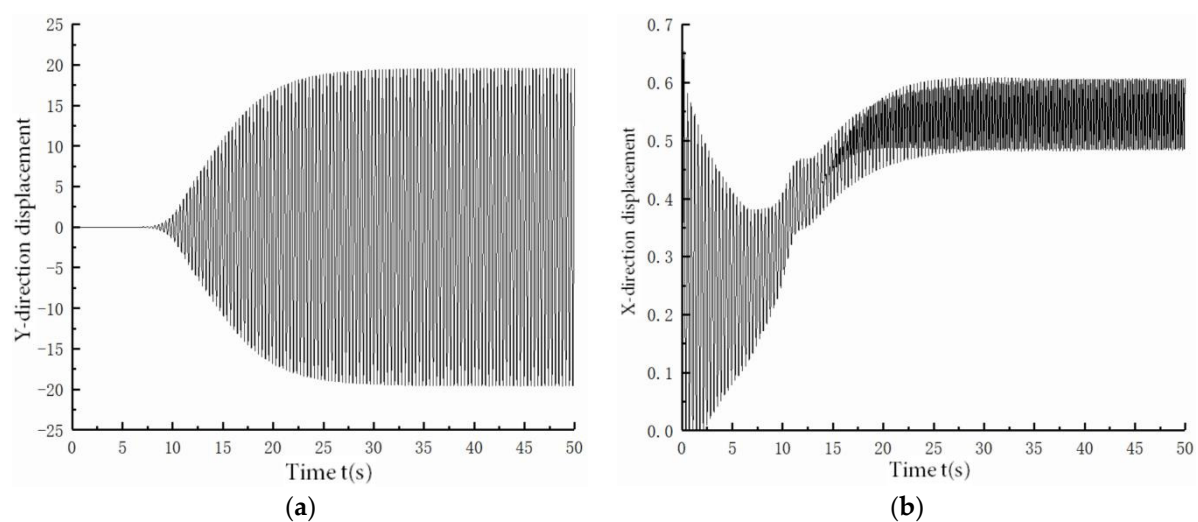


Figure 5. Y- and X-axis displacement time histograms: (a) Y-axis displacement; (b) X-axis displacement.

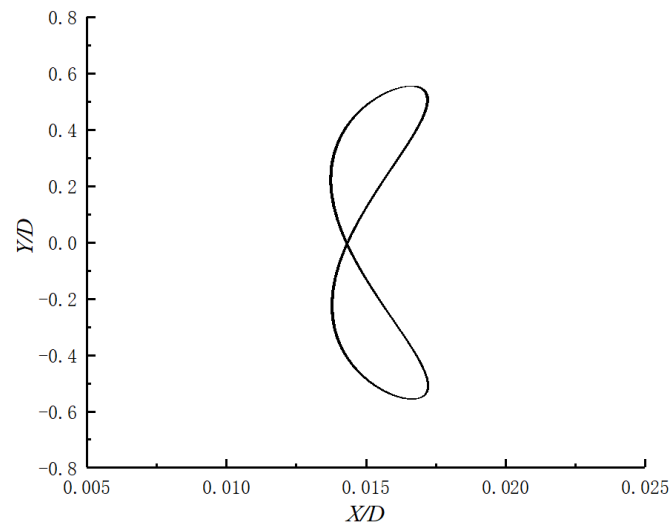


Figure 6. Trajectory diagram of two-degrees-of-freedom motion under steady-state vibration.

3.3. Wind Energy Input Power of a Single Conductor

According to the energy balance method, determining the amplitude of the wire first requires establishing the relationship between the wind energy input power and the amplitude. In this study, CFD overlapping mesh technology was employed to establish a two-degrees-of-freedom fluid–structure interaction (FSI) model of the wire, and the wind energy input at different amplitudes was directly obtained through numerical simulation.

The structural natural frequency parameters were determined based on the equivalent wind speed ($V_r = 5.0$). The lift coefficients, vertical displacement time-history curves, and motion trajectory plots were extracted for Reynolds numbers (Re) of 600, 1800, 2400, and 3000, respectively, as shown in Figures 7–10. The CFD simulations covered a range of A_0/D values from approximately 0.01 to 1. Lower A_0/D values were not considered due to the difficulty of reliably distinguishing the resulting small-amplitude vibrations from numerical noise, while higher A_0/D values (beyond ~ 1.0) were not included because the wind energy input is physically self-limiting.

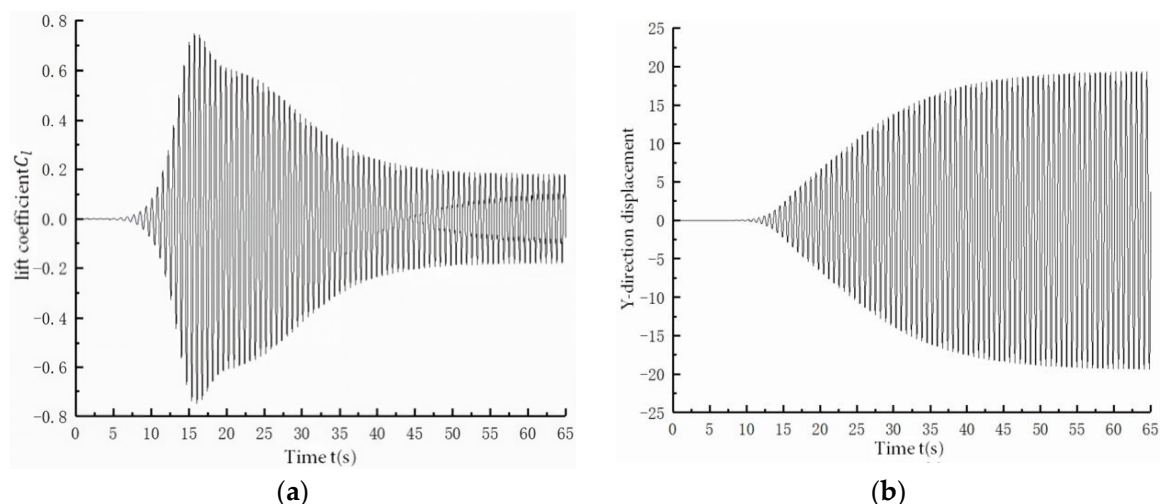


Figure 7. Lift coefficient and Y-direction displacement time history at $Re = 600$: (a) Lift coefficient; (b) Y-direction displacement.

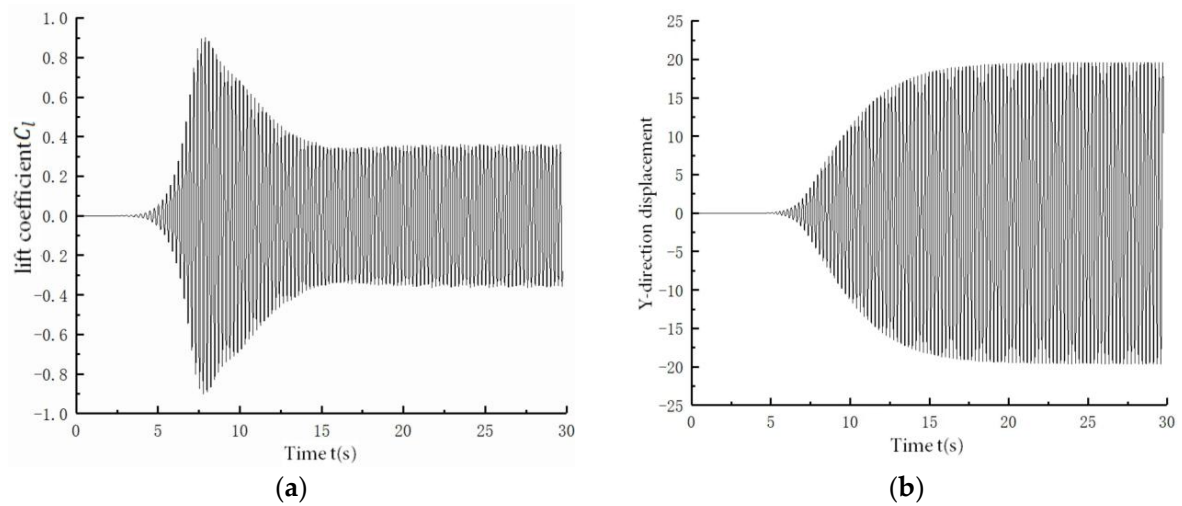


Figure 8. Lift coefficient and Y-direction displacement time history for $Re = 1800$: (a) Lift coefficient; (b) Y-direction displacement.

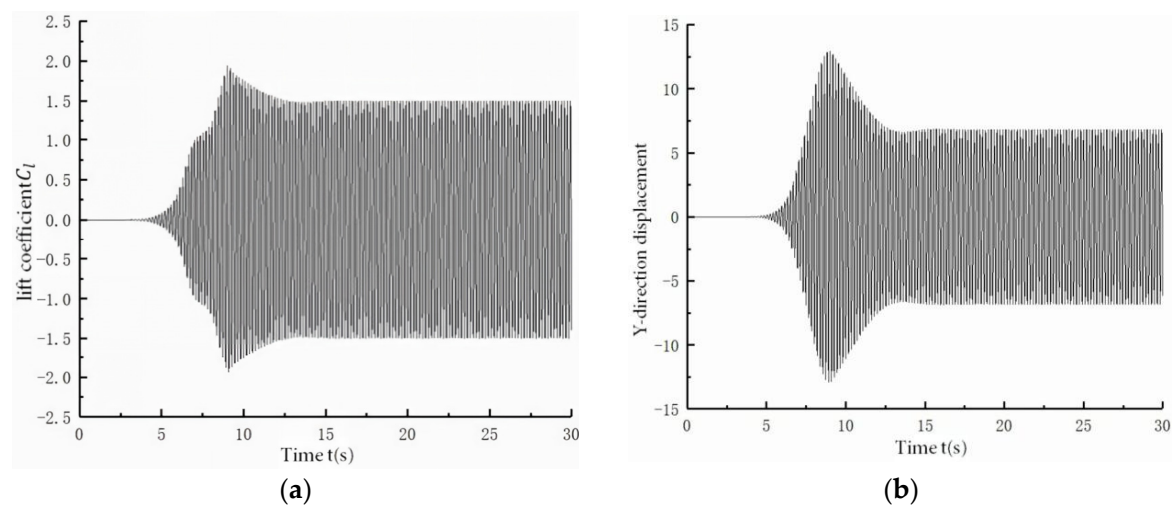


Figure 9. Lift coefficient and Y-direction displacement time history at $Re = 2400$: (a) Lift coefficient; (b) Y-direction displacement.

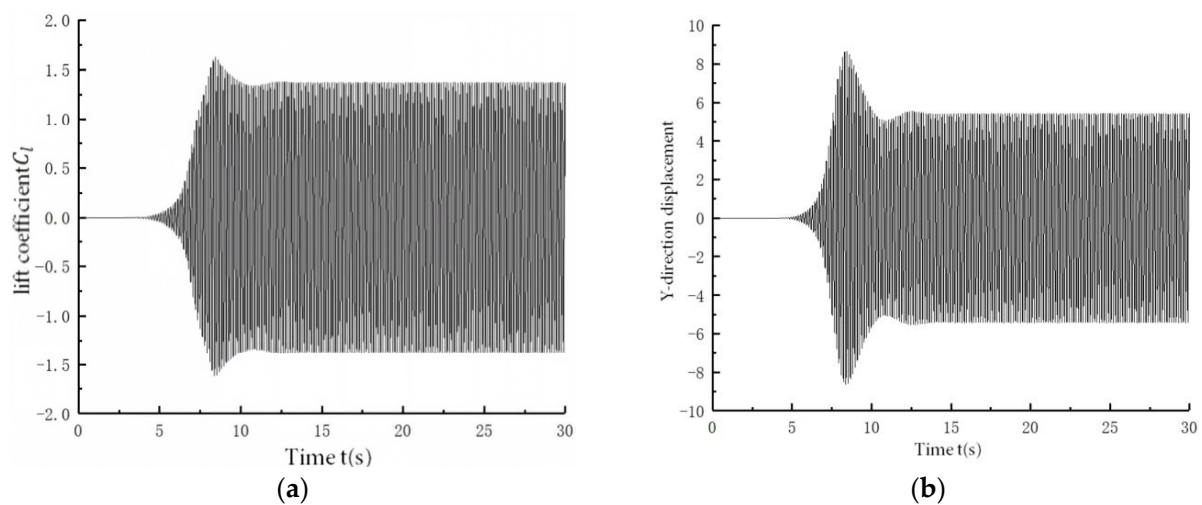


Figure 10. Lift coefficient and Y-direction displacement time history at $Re = 3000$: (a) Lift coefficient; (b) Y-direction displacement.

The lift coefficient time history curves and vertical vibration displacement data obtained from CFD simulations can be further converted into lift and velocity time history data. The average wind input power over one cycle is calculated based on Equation (14). Since the relevant data are discrete values, the trapezoidal numerical integration algorithm can be used for the calculation.

$$P_W = \frac{1}{T} \int_0^T F_l(t) \dot{y}(t) dt = \frac{\rho V^2 D}{2T} \int_0^T C_l(t) \dot{y}(t) dt \quad (14)$$

where P_W is average wind input power per unit length; T is integration time.

To facilitate comparison with wind tunnel experimental data, the input power must be converted to a dimensionless form to eliminate the effects of wire diameter and vibration frequency, as shown in Equation (15). This is also the standard expression in the energy balance method. Figure 11 shows the logarithmic distribution of wind input power under different amplitude ratios.

$$\frac{P_W}{D^4 f^3} = \frac{\rho V^2}{2TD^3 f^3} \int_0^T C_l(t) \dot{y}(t) dt \quad (15)$$

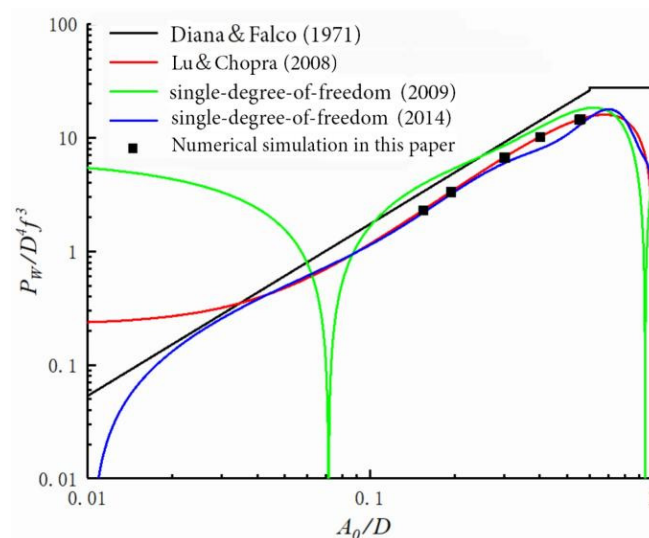


Figure 11. Wind input power curves and CFD simulation results [3,12].

The black line in the figure represents the wind input power for Diana and Falco. This result stems from earlier wind tunnel experiments using rigid cylinders to predict wind power input. It begins to decline around $A_0/D = 0.6$, taking into account the self-limiting nature of aeolian vibrations. Its fitting formula is the piecewise function [3]

$$P_W = \begin{cases} \left(\frac{A_0}{D}\right)^{2a_1} \left(\frac{A_0}{D}\right)^{a_2} e^{a_3} f^3 D^4 & A_0 < 0.6 D \\ 27.51 f^3 D^4 & 0.6 D \leq A_0 < D \\ 0 & A_0 > D \end{cases} \quad (16)$$

where $a_1 = 0.0526$, $a_2 = 1.4074$, and $a_3 = 4.0324$. The remaining three curves are the results of polynomial fitting based on CFD numerical simulations of single-degree-of-freedom vibrations reported in the literature [12]. The red, green, and blue expressions are as follows:

$$P_W = \left(-99.73 \left(\frac{A_0}{D} \right)^3 + 101.62 \left(\frac{A_0}{D} \right)^2 + 0.1627 \left(\frac{A_0}{D} \right) + 0.2256 \right) f^3 D^4 \quad (17)$$

$$P_W = (760.42 \left(\frac{A_0}{D}\right)^5 - 1915.5 \left(\frac{A_0}{D}\right)^4 + 1621.5 \left(\frac{A_0}{D}\right)^3 - 591.09 \left(\frac{A_0}{D}\right)^2 + 127.65 \left(\frac{A_0}{D}\right) - 6.6502) f^3 D^4 \quad (18)$$

$$P_W = (-29473.95 \left(\frac{A_0}{D}\right)^9 + 112307.22 \left(\frac{A_0}{D}\right)^8 - 168494.81 \left(\frac{A_0}{D}\right)^7 + 125036.41 \left(\frac{A_0}{D}\right)^6 - 46644.05 \left(\frac{A_0}{D}\right)^5 + 7312.67 \left(\frac{A_0}{D}\right)^4 + 7.58 \left(\frac{A_0}{D}\right)^3 - 62.62 \left(\frac{A_0}{D}\right)^2 + 15.28 \left(\frac{A_0}{D}\right) - 0.15) f^3 D^4 \quad (19)$$

The numerical simulation results of two-degrees-of-freedom vibration in this paper are consistent with the red curve; therefore, the curve fitted by this cubic polynomial is adopted as the wind power input. To account for the impact of varying regional conditions on wind power input, the non-uniformity coefficient for correcting wind power input, as proposed in a report from the International Conference on Large Electric Systems (ICLES), is introduced here. The expression for this coefficient is as follows:

$$\beta_W = \frac{1}{\sqrt{1 + (I_V/I_L)^2}} \quad (20)$$

where I_V is the wind turbulence intensity at the transmission line suspension point: $I_V = \sigma_V/\bar{V}$, σ_V is the standard deviation of wind speed, $\bar{V}$ is the average wind speed over the statistical time period; I_L is the locking constant, used to reflect the locking effect in light-wind vibrations, and can be taken as $I_L = 0.09$. I_V Calculated using statistical wind speed data for the site where the transmission line is located; taking Nanchang City in 2023 as an example: $\bar{V} = 2.75$ m/s, $\sigma_V = 0.254$, from which $\beta_W = 0.6975$ can be calculated. After introducing the non-uniformity coefficient, Equation (17) takes the following form:

$$P_W = \beta_W \left(-99.73 \left(\frac{A_0}{D}\right)^3 + 101.62 \left(\frac{A_0}{D}\right)^2 + 0.1627 \left(\frac{A_0}{D}\right) + 0.2256 \right) f^3 D^4 \quad (21)$$

4. Results and Discussion

4.1. Calculation Example of a Long-Span Transmission Conductor Without Vibration Dampers

To evaluate the light-wind vibration intensity of long-span conductors without vibration dampers, this section uses AACSR/EST-500/230 conductors as the subject of study, with a span of 2000 m, with operating tensions set at three levels: 20% RTS, 25% RTS, and 30% RTS. The fatigue safety limit is defined as a dynamic bending strain not exceeding ± 200 . The results of the calculations for the wave belly amplitude and dynamic bending strain of the conductor at different excitation frequencies are shown in Figures 12 and 13.

Analysis of the results in Figures 12 and 13 reveals the following:

- (1) When the excitation frequency is below 15 Hz, increasing tension enhances bending stiffness, causing a slight decrease in dynamic bending strain. However, when the excitation frequency exceeds 15 Hz, increased tension weakens the self-damping energy dissipation capacity, leading to an increase in amplitude and a significant rise in dynamic bending strain. Overall, dynamic bending strain increases with increasing tension.
- (2) As tension increases, the critical frequency range widens significantly: at 20%RTS, it is 3.45 Hz to 38.62 Hz; at 25%RTS, it is 3.93 Hz to 50.92 Hz; and at 30%RTS, it is 4.16 Hz to 66.40 Hz. The corresponding wind speed range at which vibration begins also increases accordingly.

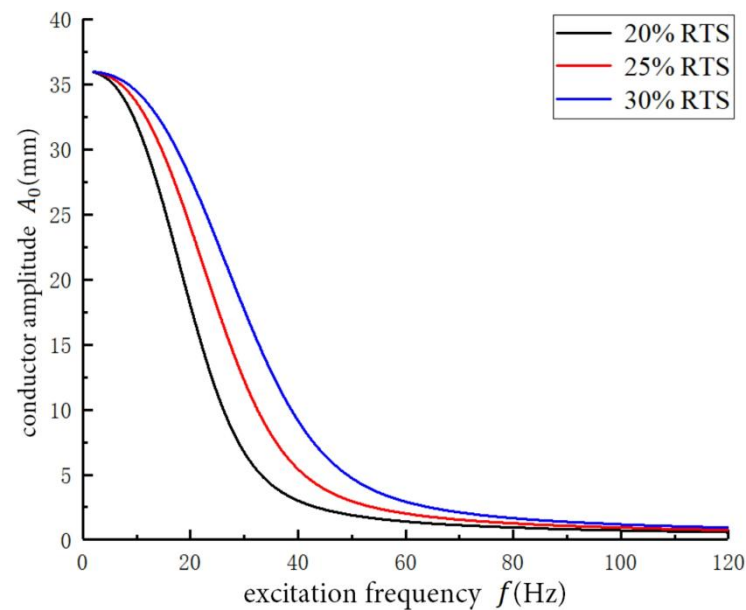


Figure 12. Wave belly amplitude of the conductor under different tension conditions.

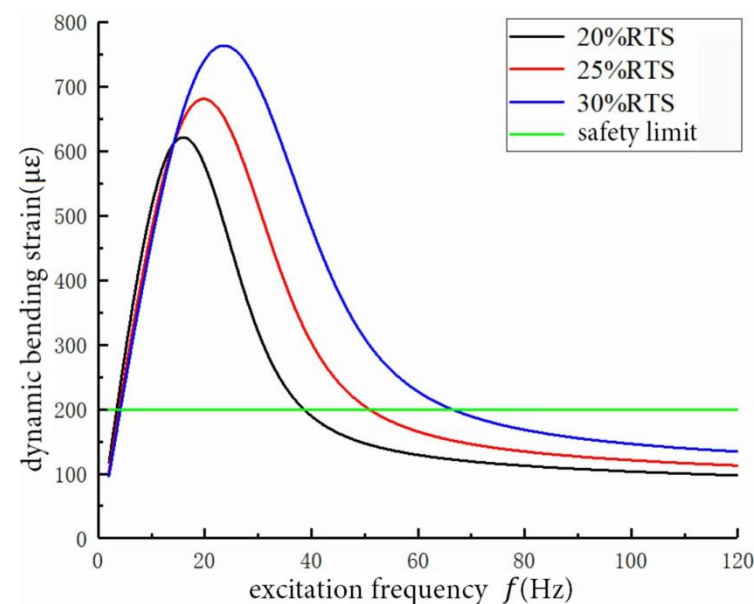


Figure 13. Dynamic bending strain of the conductor under different tension conditions.

4.2. Calculation Example After Installing Vibration Dampers

Existing design codes and research [15,30] mainly provide regulations on the number of vibration dampers for single conductors and ground wires with diameters up to 37.1 mm and spans up to 1200 m, but do not cover cases with larger diameters or longer spans, and relevant research is scarce. Therefore, this study is of significant importance for determining the configuration method of vibration dampers under such conditions.

Below, using the AACSR/EST-500/230 conductor as an example, the installation positions of the vibration dampers are determined based on the vibration wavelength of the conductor and the probability distribution of wind speeds. If subsequent calculations of vibration damper installation positions result in overlaps, equidistant installation is adopted, with symmetrical arrangement at the left and right ends. Subsequently, the micro-wind vibration of long-span transmission conductors is analyzed using the energy balance method. Based on a specific vibration damper model (FD-6) and design codes, the

number of vibration dampers to be installed for different spans (with the same tension) is determined. The vibration damper layout scheme for a 2000 m span is shown in Figure 14, where the original dimensions of the vibration dampers are magnified by a factor of 3. The calculation results are shown in Figure 15.

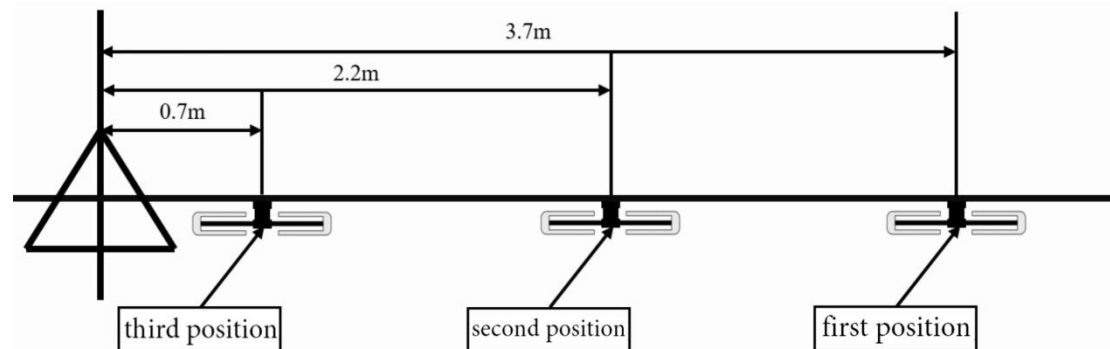


Figure 14. Schematic diagram of vibration damper arrangement at the left end for a 2000 m span (symmetrical arrangement at the right end).

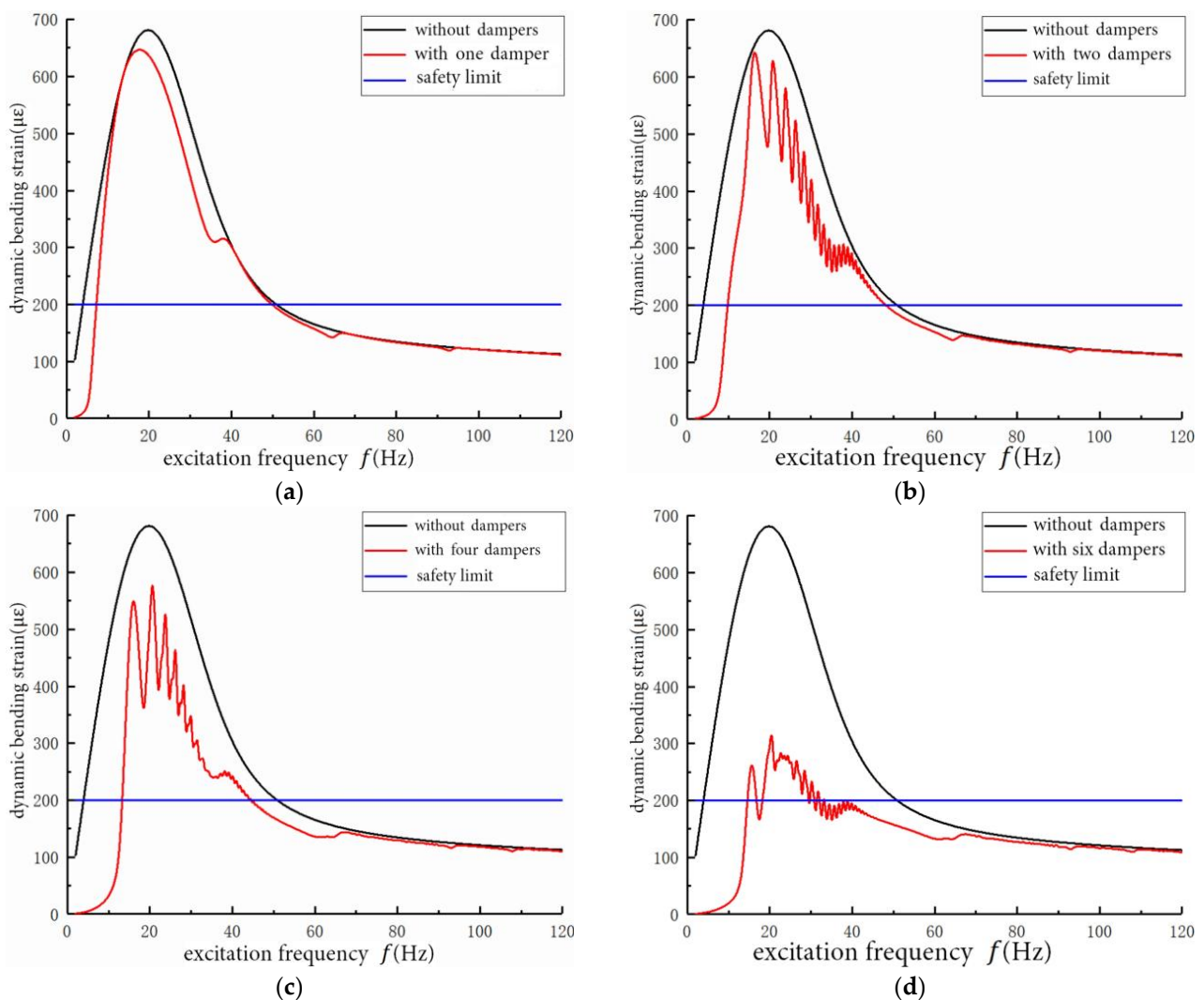


Figure 15. Dynamic bending strain of conductors under different numbers of vibration dampers (2000 m span). (a) Case with one vibration damper at the left end; (b) case with one vibration damper at each end; (c) case with two vibration dampers at each end; (d) case with three vibration dampers at each end.

Analysis of Figure 15 reveals:

- (1) Vibration dampers are highly effective at suppressing low-frequency vibrations, but their damping capacity drops sharply in the high-frequency range. When dampers are located near the wave nodes (13 Hz, 40 Hz, 70 Hz), there is almost no improvement in dynamic bending strain; when located near the wave antinodes (35 Hz, 64 Hz, 93 Hz), the strain is significantly reduced.
- (2) Increasing the number of vibration dampers can gradually narrow the critical frequency range, but the damping effect is not proportional to the number. For a 2000 m span, even after installing six vibration dampers, the maximum dynamic bending strain cannot be reduced below 200 $\mu\epsilon$, indicating that vibration dampers alone are insufficient to meet the vibration control requirements for long-span conductors.

4.3. Calculation Example After Installing Damping Wires

When the number of damping wires is small, the installation distance is calculated using the formula [19]

$$S_1 = \frac{\lambda_m}{4} = \frac{D}{800v_M} \sqrt{\frac{9.81\sigma}{g}} \quad (22)$$

$$S_1 + S_2 + S_3 = \left(\frac{1}{4} \sim \frac{1}{6}\right) \lambda_M = \left(\frac{1}{4} \sim \frac{1}{6}\right) \frac{D}{800v_m} \sqrt{\frac{9.81\sigma}{g}} \quad (23)$$

$$S_2 = S_3 \quad (24)$$

where λ_m and λ_M are the minimum and maximum vibration wavelengths within the protection frequency band; and σ is the conductor stress during operation.

In engineering practice, design schemes often draw on examples with proven operational experience as a basic template. Figure 16 shows the schematic diagram of damping cable installation. Taking the Nanjing Yangtze River Grand Crossing project as an example, its 1933 m span length employs a decreasing lace combination of 6, 5, 4, 3, and 3 m [11]. The following example uses the damping wire arrangement scheme for the Nanjing Yangtze River Grand Crossing. In this scheme, the damping wire type matches that of the conductor. Figure 17 shows the damping wire arrangement, which is symmetrically arranged on both sides. Figure 18 illustrates the variation in dynamic bending strain of the conductor when different numbers of damping wires are installed.

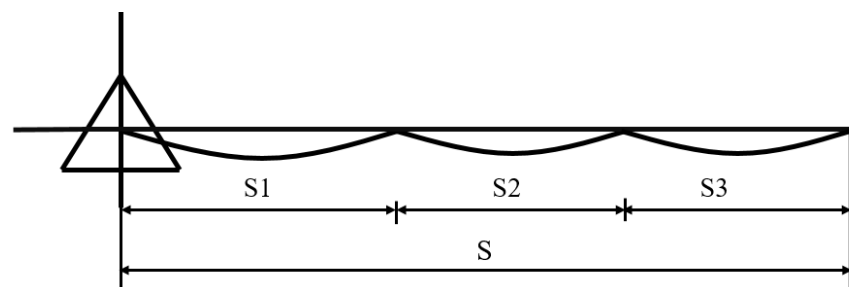


Figure 16. Schematic diagram of damping cable installation.

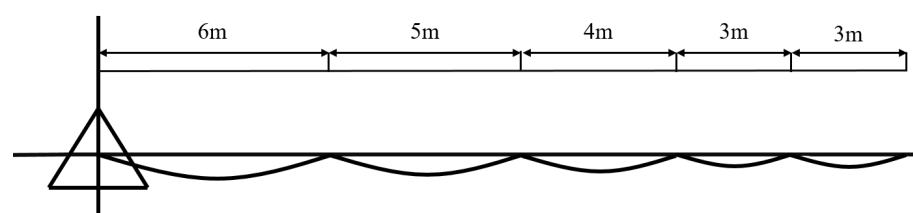


Figure 17. Damping wire arrangement scheme.

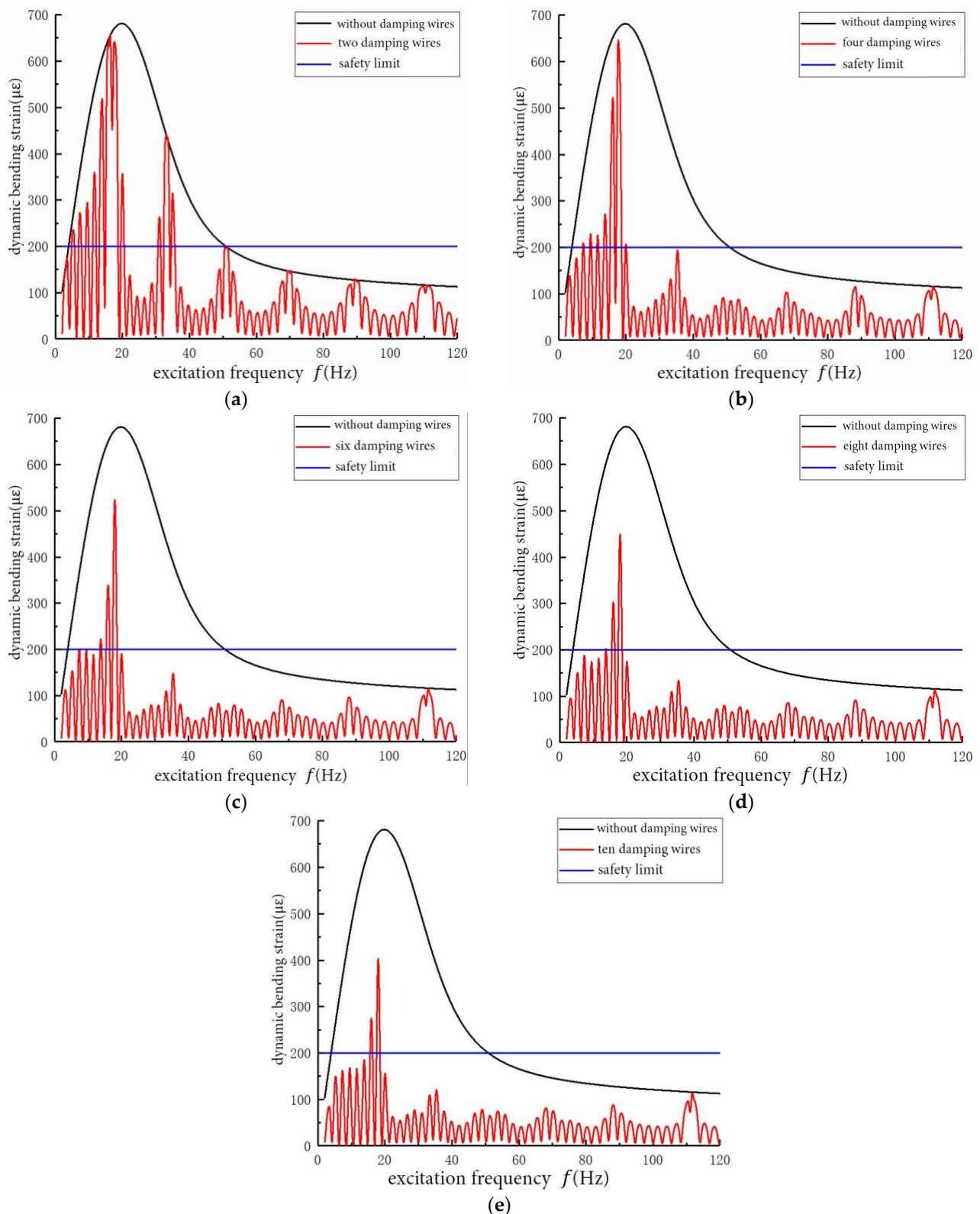


Figure 18. Dynamic bending strain of the conductor under different numbers of damping wires. (a) One damping wire at each end; (b) two damping wires per end; (c) three damping wires per end; (d) four damping wires at each end; and (e) five damping wires at each end.

As shown in Figure 18, damping wires provide effective vibration damping. Their protective frequency range is wider than that of vibration dampers, covering nearly the entire frequency band of aeolian vibrations. Furthermore, since damping wires of different

lengths have different natural frequencies, their maximum damping performance varies; however, the protective frequency range of damping wires remains in the low-frequency range. Nevertheless, the use of damping wires alone cannot reduce dynamic bending strain to below $200 \mu\epsilon$.

4.4. Calculation Example of Combined Installation of Damping Wires and Vibration Dampers

In long-span transmission line projects, a combined damping wire and vibration dampers system is commonly adopted, and practical operational experience has confirmed the vibration-damping effectiveness of this combined protection method. This approach consists of damping wire as the primary component, supplemented by vibration dampers. Since damping wires and vibration dampers each have their own advantages and disadvantages, the combined protection method allows for mutual reinforcement, thereby achieving a broader frequency response. There are two scenarios for the installation of vibration dampers: one involves installing the dampers at the outermost end of the damping wire, and the other involves installing them inside the first to third loops of the damping wire. When installing multiple vibration dampers, it is advisable to use dampers with different frequency characteristics. Figure 19 shows a schematic diagram of the damping wire and vibration damper arrangement for combined vibration damping; in the figure, the original dimensions of the vibration dampers are magnified by a factor of 3. Figure 20 illustrates the dynamic bending strain of a conductor with “3 vibration dampers + 5 damping wires” installed at each end.

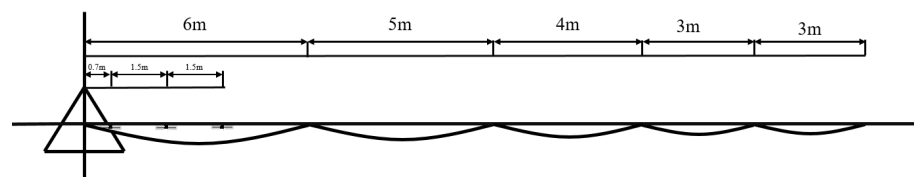


Figure 19. Schematic of combined vibration damping arrangement.

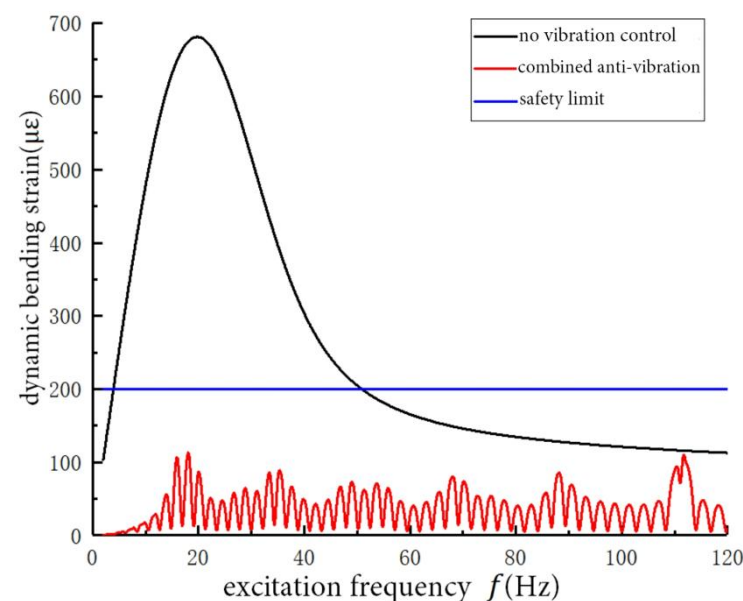


Figure 20. Dynamic bending strain of the conductor with combined vibration damping.

A comparison of Figures 13, 15 and 18 shows that using either vibration dampers or damping wires alone cannot reduce the maximum dynamic bending strain of the conductor to below $200 \mu\epsilon$. However, the vibration control scheme combining damping wires and vibration dampers reduces the maximum dynamic bending strain of the conductor to

113.09 $\mu\epsilon$ within the critical frequency range (3.93 Hz–50.92 Hz), demonstrating excellent vibration damping performance.

5. Conclusions

This paper employs CFD overlapping mesh technology to establish a two-degrees-of-freedom fluid–structure interaction model for the conductor. Through numerical simulations at different Reynolds numbers, a cubic polynomial was fitted for the relationship between wind power input and amplitude ratio, and a wind turbulence intensity correction factor was introduced to account for regional influences. Based on this, using a long-span conductor as a case study, the dynamic bending strain of the conductor was compared under scenarios with no vibration control, vibration dampers only, damping wires only, and a combined vibration control scheme, using the energy balance method. The main conclusions are as follows:

- (1) The wind energy input power curves obtained from two-degrees-of-freedom CFD simulations are consistent with the trends observed in classical wind tunnel experiments. The cubic polynomial fitting expression effectively describes the relationship between wind energy input power and the amplitude ratio. After introducing the wind turbulence correction coefficient, the model can reflect the influence of regional wind speed statistical characteristics, thereby enhancing its engineering applicability.
- (2) For long-span, large-cross-section conductors, using either vibration dampers or damping wires alone makes it difficult to control dynamic bending strain below 200 $\mu\epsilon$. Vibration dampers are primarily effective in the low-frequency range, while damping wires offer protection across a broader frequency range; however, both methods have protection blind spots when used individually.
- (3) Adopting a combined vibration control scheme of “damping wires + vibration damper” can reduce the maximum dynamic bending strain within the critical frequency range (3.93 Hz–50.92 Hz) from well over 200 $\mu\epsilon$ to 113.09 $\mu\epsilon$, demonstrating significant vibration damping effects. This conclusion validates the effectiveness of the wind energy input formula fitted in this paper for engineering vibration control design and can serve as a reference for optimizing vibration control schemes for similar power lines.

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