Structure Design and Assessment of a Floating Foundation for Offshore Wind Turbines

Ye, Q

http://hdl.handle.net/10026.1/14897

10.1115/IOWTC2019-7594
ASME 2019 2nd International Offshore Wind Technical Conference
American Society of Mechanical Engineers

DRAFT
Proceedings of the ASME 2019
2nd International Offshore Wind Technical Conference
IOWTC2019
November 3-6, 2019, St. Julian's, Malta

IOWTC2019-7594

STRUCTURE DESIGN AND ASSESSMENT OF A FLOATING FOUNDATION FOR OFFSHORE WIND TURBINES

Qi Ye[1], Shanshan Cheng, Boksun Kim, Keri Collins
University of Plymouth, Plymouth, UK

Gregorio Iglesias
University of Plymouth, Plymouth, UK
University College Cork, Cork, Ireland

ABSTRACT
This paper summarizes the assessment of the structural analysis and design of a floating foundation for offshore floating wind turbine (FWT) based on DNVGL standard and Eurocode in terms of economy and reliability. The wind loads are calculated using empirical equations. The wave loads are obtained and verified using various methods including hand calculation, AQWA and Flow-3D. It is found that the shell thickness could be reduced significantly by introducing the stiffeners (stringer or ring), which can decrease the weight of the hull and lower the cost. While DNVGL and Eurocode yield similar design solutions if using plane shell structures, Eurocode significantly underestimates the buckling resistance of stiffened cylindrical shells.

Keywords: floating foundation, structural design, shell structure, buckling resistance, wind loads, wave loads.

SYMBOLS
The following symbols are used in flow chart in section 4:

$\sigma_{a,Sd}$	design membrane stress in the longitudinal direction due to uniform axial force
$\sigma_{m,Sd}$	design membrane stress in the longitudinal direction due to global bending
$\sigma_{x,Rd}$	meridional design buckling stress
$\sigma_{xE,A}$	design value of the meridional stress due to axial force
$\sigma_{xE,M}$	design value of the meridional stress due to bending
$\sigma_{x,Ed}$	design value of meridional stress
$\sigma_{h,Sd}$	design membrane stress in the circumferential direction
$\sigma_{\theta,Ed}$	circumferential design stress
$\sigma_{\theta,Rcr}$	elastic critical circumferential buckling stress
$\sigma_{\theta,Rk}$	circumferential characteristic buckling stress
$\sigma_{\theta,Rd}$	circumferential design buckling stress
$\sigma_{j,Sd}$	design equivalent von Mises' stress
τ_{Sd}	design shear stress tangential to the shell surface
$\tau_{x\theta,Ed}$	design value of shear stress
$\tau_{x\theta,Rcr}$	elastic critical shear buckling stress
$\tau_{x\theta,Rk}$	shear characteristic buckling stress
$\tau_{x\theta,Rd}$	shear design buckling resistance stress
Z_l	curvature parameter
Z_s	curvature parameter
C	reduced buckling coefficient
C_θ	circumferential buckling factor
C_τ	shear buckling factor
f_E	elastic buckling strength
f_{Ea}	elastic buckling strength for axial force
f_{Em}	elastic buckling strength for bending moment
f_{Eh}	elastic buckling strength for hydrostatic pressure, lateral pressure and circumferential compression
$f_{E\tau}$	elastic buckling strength for shear force
f_{ks}	characteristic buckling strength of a shell
f_{ksd}	design buckling strength of a shell
$\bar{\lambda}_s^{\,2}$	reduced shell slenderness
ω	length parameter
Δw_k	characteristic imperfection amplitude
α_x	meridional elastic imperfection reduction factor
$\bar{\lambda}_\theta$	circumferential shell slenderness parameters
$\bar{\lambda}_{\theta p}$	circumferential plastic limit relative slenderness
$\bar{\lambda}_\tau$	shell slenderness parameters of shear

[1] Contact author: qi.ye@plymouth.ac.uk

$\bar{\lambda}_{\tau p}$ shear plastic limit relative slenderness

$\bar{\lambda}_x$ meridional shell slenderness parameters

$\bar{\lambda}_{xp}$ meridional plastic limit relative slenderness

k_x meridional buckling interaction parameter

k_θ circumferential buckling interaction parameter

k_τ shear buckling interaction parameter

χ_x meridional buckling reduction factor

χ_θ circumferential buckling reduction factor

χ_τ shear buckling reduction factor

1 INTRODUCTION

As energy consumption increases globally and environmental issues threaten the quality of life, new sustainable ways of energy generation are actively being researched and promoted. As a type of renewable energy, wind power has advantages in terms of duration, reserves and distribution. Compared with onshore wind power, offshore wind power has advantages in terms of the magnitude, quality and reliability of the resource [1]. Although offshore wind energy has demonstrated great potential, it needs to cut down cost significantly in order to be competitive with conventional energy types. Given the fact that the cost of foundation for the offshore FWT accounts for around 30% of capital expenditures [2]. It is essential to optimize the design and fabrication of the support structures in order to lower the cost.

To promote the stability and structural reliability of offshore wind turbines, research studies have been carried out on the dynamic response [3-6] due to environmental loads and structural performance due to static and dynamic forces. The dynamic response affects the structural performance to a great degree, especially for FWT, and can be studied by using physical and numerical modelling under operational and extreme conditions. As for the structural design of the foundation, the study mainly focuses on geometric optimization to reduce the structural mass [1, 7-11]. Based on the design guideline of offshore structure, finite element method and optimization algorithm, the thickness and mass of material could be reduced. Shell structures are usually used in offshore structures, which have complex details such as stiffeners and connection joints. In FWT construction and installation, the labour cost and towing vessel account for a large proportion in the total cost; therefore, the structural design including details design can be investigated to simplify the structure and reduce the construction and maintenance cost.

The aim of this work is to assess the structural design of a floating foundation for offshore wind turbines based on DNVGL [12] and Eurocode [13] in terms of economy and reliability.

2 PROBLEM DEFINITION

The WindFloat [14] developed by Principle Power is used as the concept of FWT in this study. It has a semisubmersible floating foundation composed of three columns to provide buoyancy to support a wind turbine. The three columns have the same geometric profile and are connected by the pontoons and bracings. The complex loading features of the columns should be considered in structural design. A 5 MW wind turbine is designed to be installed on one of the columns of the floating foundation and the main parameters of this wind turbine are shown in Table 1 and Table 2. The structure of column, pontoon, bracing and mast is a shell structure with ring and stringer stiffeners to provide sufficient local and global buckling stiffness to the component.

Figure 1. The WindFloat concept [15]

The wave and wind loads [16] play important roles in structural analysis, so the effect of wave and wind loads on the turbine and hull should be paid more attention. After that, the material, strength, concept and details of structures are produced to resist the effect of different load combinations. To meet the engineering demands for strength and stability, different ultimate limit states such as plastic limit, cyclic plasticity, buckling and fatigue are considered in structural design. In this study, the wind loads and wave loads applied on the structure are analyzed separately to obtain their respective contributions of shear force, axial force and bending moment. The buckling resistance of steel shell structures is investigated and the related industry criteria, DNVGL standard and Eurocode, are compared to develop a better and more economical design.

Table 1. WindFloat main dimensions (unit: metre) [14]

Column diameter	10.7
Length of column	28
Length of heave plate edge	13.7
Column centre to centre	56.4
Pontoon diameter	1.8
Bracing diameter	1.2
Mast diameter	8
Mast length	88
Rotor diameter	126

Table 2 5 MW turbine characteristics [14]

Rotor mass	135 tonne
Nacelle mass	294 tonne
Mast mass	425 tonne
Mast diameter	8 metre
Mast length	88 metre
Rotor diameter	126 metre

3 ENVIRONMENTAL LOADS

As the FWT moves from shallow waters to deep water, wind speeds generally increase and weather conditions usually worsen further [17]. In this study, wind loads and wave loads are calculated separately based on the environmental conditions of Humboldt Bay. A 100 year return wave is considered as an extreme event in structural design purposes, which is shown in Table 3. The critical load combinations including permanent load, variable functional load and environmental load are considered in structural analysis and design.

Table 3. 100 year storm [14]

Significant wave height	13.5 m
Peak period	17 s
Wind speed at 10 m elevation	25.9 m/s

3.1 Wind loads

It is crucial to assess the wind loads so that every wind-exposed part of the FWT should be considered including the rotor, the mast (or tower) and the part of the columns above water level. The rotor thrust can be estimated by the following expression [18],

$$F_{thrust} = \frac{1}{2} \cdot \rho \cdot C_T \cdot A_{rotor} \cdot U_{10}^2 \quad (1)$$

where ρ is the density of air, U_{10} is the far-field 10-minute mean wind speed, C_T is the thrust coefficient and A_{rotor} is the swept area of the rotor. The swept area is $A_{rotor} = \pi R^2$, where R is the rotor radius.

The wind force on the surface of a structural member, for example mast, pontoon and column above water level part, may be calculated according to [18]:

$$F_W = CqS \sin \alpha \quad (2)$$

where C is the shape coefficient, q is the basic wind pressure or suction, S is the projected area of the member normal to the direction of the force and α is the angle between the direction of the wind and the axis of the exposed member or surface.

The mean wind speed of different components is defined by the wind speed profile which can be expressed as [18]:

$$U(z) = \frac{u^*}{k_a} \ln \frac{z}{z_0} \quad (3)$$

where u^* is the friction velocity, k_a is von Karman's constant, z is the height and z_0 is a terrain roughness parameter.

The mean wind speed in the height of the centre of the rotor is 34.3 m/s and the thrust coefficient is assumed to be 0.08 according to DNVGL-ST-0119 [19]. The shear force and bending moment due to wind loads and the self-weight of the wind turbine would be transferred to the top of the column.

3.2 Wave loads

According to DNVGL-RP-C205, the sectional force f_N on a fixed slender structure in two-dimensional flow normal to the member axis is given by:

$$f_N(t) = \rho(1 + C_A)A\dot{v} + \frac{1}{2}\rho C_D D v|v| \quad (3)$$

where v is the fluid particle (waves and/or current) velocity, $\dot{v}$ is the fluid particle acceleration, A is the cross-sectional area, D is the diameter or typical cross-sectional dimension, ρ is the mass density of fluid, C_A is the added mass coefficient (with cross-sectional area as reference area) and C_D is the drag coefficient.

A regular linear wave is used to describe the wave form function. In order to verify the wave loads due to the extreme event (13.5 m significant wave height, 17 s period) on the foundation of the FWT during operation, three methods are used in this study including hand calculation, AQWA and Flow-3D. In this extreme case of 100 year storm, the wave length is 450 m which is longer than 5D (53.5 m). The drag force and inertia force are dominant in wave induced loads so Morison's load formula is applicable. AQWA is a commercial simulation tool which is used to calculate 1st order wave loads based on linear potential flow theories. Computational Fluid Dynamics (CFD) Flow-3D is a CFD (Computational Fluid Dynamics) software. In the comparison of wave loads, linear wave, inviscid & incompressible fluids and irrotational flow are used in AQWA wave model. On the other hand, linear wave, viscid & incompressible fluids and irrotational flow are used in Flow-3D wave model.

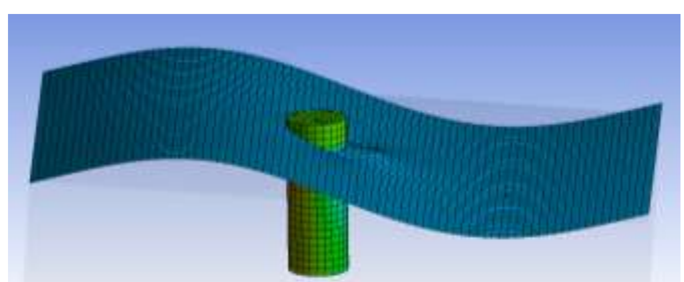

Figure 2. Wave loads calculation in AQWA

Figure 3 shows the comparison of the total wave loads applied to a single column of WindFloat hull by using Flow-3D, hand calculation and AQWA. As can be seen in this graph, the maximum wave loads are consistent, and approaching to about 4000 kN on the surface of the shell structure. In ultimate limit state design of structure, the maximum wave loads are calculated only in hand calculation, which should be checked by the maximum value of the computational results. The result of hand calculation is slightly more conservative than that of AQWA. Compared with the result of AQWA, the mutational wave loads exist in the first three period in Flow-3D. This is because the extreme wave relatively unstable in the beginning, which induces break waves on the surface of the structure. After that, the wave loads tend to be stable. The function of total wave loads and time of AQWA is a simulated harmonic curve. Because the model of AQWA was based on linear wave theory, assumptions of ideal fluid. While in Flow-3D analyses, assumptions of viscid and incompressible fluids and irrotational flow were used and the absolute value of the minimum wave loads is smaller than the maximum wave loads. In consequence, Flow-3D gives a larger wave loads than AQWA but overall differences in results are not significant in the stable stage of wave propagation.

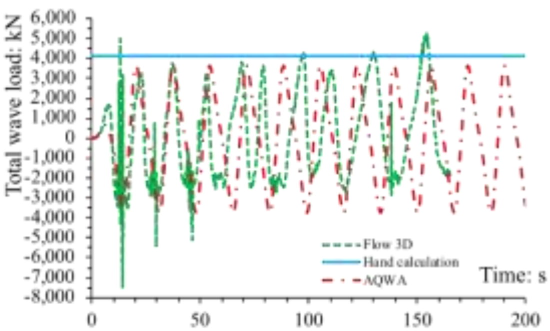

Figure 3: Comparison of the total wave loads of three methods

3.3 Load factors and combinations

The different characteristic loads (wind loads, wave loads, self-weight, etc.) are combined by different factors and combinations to calculate the design load in ultimate limit state design. According to DNVGL-ST-0119 [19], the following load factors combinations should be taken into account:

Table 4. Load factors and combinations for normal condition design [19]

Load factor set	Load categories					
	G	Q	E		D	P
			Consequence class			
			1	2		
(a)	1.25	1.25	0.7		1.0	0.9/1.1
(b)	1.0	1.0	1.35	1.55	1.0	0.9/1.1

G: permanent load (mass of structures, permanent ballast, equipment and hydrostatic pressure, etc.)
Q: variable functional load (loads on access platforms, weight of variable ballast, etc.)
E: environmental load (wind loads, wave loads, current loads, etc.)
D: deformation load (temperature loads, creep loads, etc.)
P: prestressing load

In this study, deformation and prestressing loads are ignored, and only permanent, variable functional and environmental loads are considered.

4 STRUCTURAL DESIGN CODES

The three columns of the floating structure of the WindFloat are defined as the cylindrical shell. In this section, two guidelines DNVGL-RP-C202 and EN 1993-1-6 for shell structure design are introduced, which will be used to determine the geometry of the columns in the next section.

4.1 Unstiffened shell structure design

4.1.1 DNVGL-RP-C202

According to DNVGL-RP-C202, the buckling modes of the unstiffened circular cylinders to be checked are: 1) shell buckling and 2) column buckling.

To meet the stability requirement of a structure, the design load effect does not exceed the design buckling strength. The stability requirement subjected to axial force, bending moment, circumferential compression and shear is given by:

$$\sigma_{j,Sd} \leq f_{ksd} \tag{4}$$

where $\sigma_{j,Sd}$ is the design equivalent von Mises' stress and f_{ksd} is the design buckling strength of a shell.

The reduced buckling coefficient can be used to consider geometrical and material imperfections provided by:

$$C = \psi\sqrt{1 + \left(\frac{\rho\xi}{\psi}\right)^2} \tag{5}$$

The coefficients ψ, ξ and ρ are buckling coefficients. In order to present the design processes more clearly, the flow chart of shell buckling calculation based on DNVGL-RP-C202 is shown in Figure 4.

The cylinders would be more unstable as the length increase so the maximum load effect the column can support should be determined before it buckles. Hence, except the local buckling introduced in the front part, buckling of the entire column should be considered as well. The column buckling strength should be assessed if

$$\left(\frac{kL_c}{i_c}\right)^2 \geq 2.5\frac{E}{f_y} \tag{6}$$

where k is the effective length factor, L_C is the total cylinder length, $i_c = \sqrt{I_c/A_c}$ is the radius of gyration of cylinder section, I_C is the moment of inertia of the complete cylinder section (about weakest axis), A_C is the cross sectional area of complete cylinder section.

According to the guidelines, besides the column buckling strength, the stability requirement also need to be met. The combined action due to axial compression and bending moment need to be calculated with the buckling strength and the cross-section capacity need to be checked. The stability requirement for the column could be calculated as:

$$\frac{\sigma_{a0,Sd}}{f_{kcd}} + \frac{1}{f_{akd}}\left[\left(\frac{\sigma_{m1,Sd}}{1-\frac{\sigma_{a0,Sd}}{f_{E1}}}\right)^2 + \left(\frac{\sigma_{m2,Sd}}{1-\frac{\sigma_{a0,Sd}}{f_{E2}}}\right)^2\right]^{0.5} \leq 1 \tag{7}$$

where $\sigma_{a0,Sd}$ is the design axial compression stress, $\sigma_{m,Sd}$ is the maximum design bending stress about given axis, f_{akd} is the design local buckling strength, f_{kcd} is the design column buckling strength and f_{E1}, f_{E2} are Euler buckling strength. The Euler buckling strength is used to calculate the maximum stress that the column can bear before it buckles.

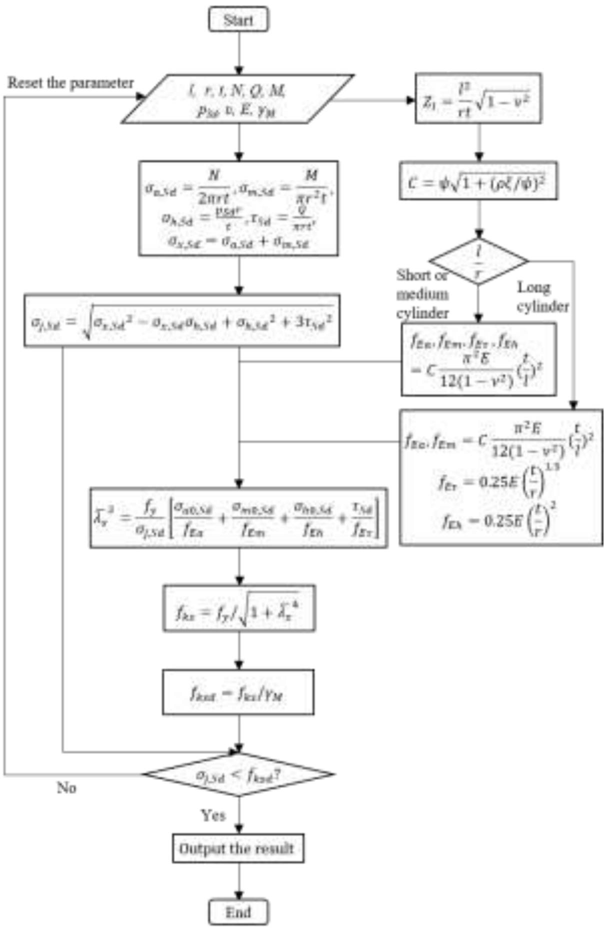

Figure 4. Flow chart of shell buckling calculation

4.1.2 EN 1993-1-6

The following modes should be checked: 1) Meridional (axial) buckling, 2) Circumferential (hoop) buckling, 3) Shear buckling, 4) Meridional compression with coexistent internal pressure, 5) Combinations of meridional compression, circumferential compression and shear.

Cylinders need be checked against meridional shell buckling if they satisfy:

$$\frac{r}{t} > 0.03\frac{E}{f_{yk}} \tag{8}$$

The flow chart of meridional buckling calculation is shown in Figure 5.

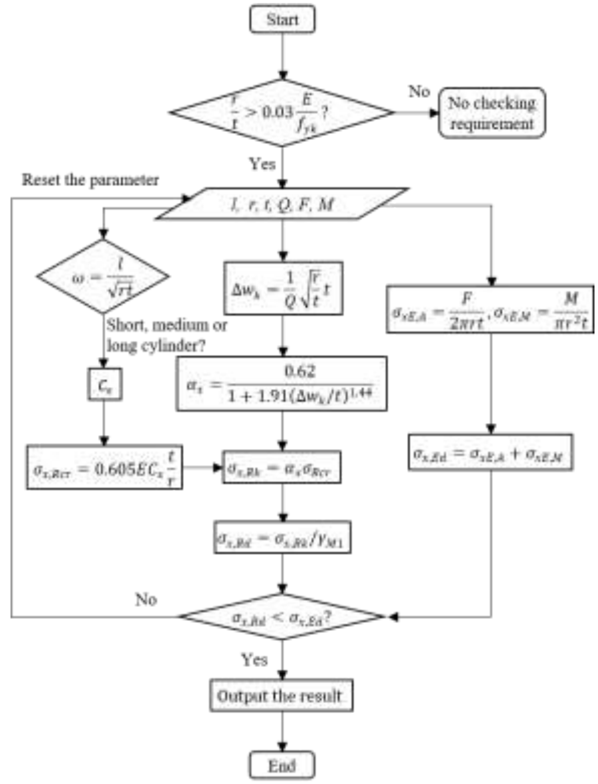

Figure 5. Flow chart of meridional buckling calculation

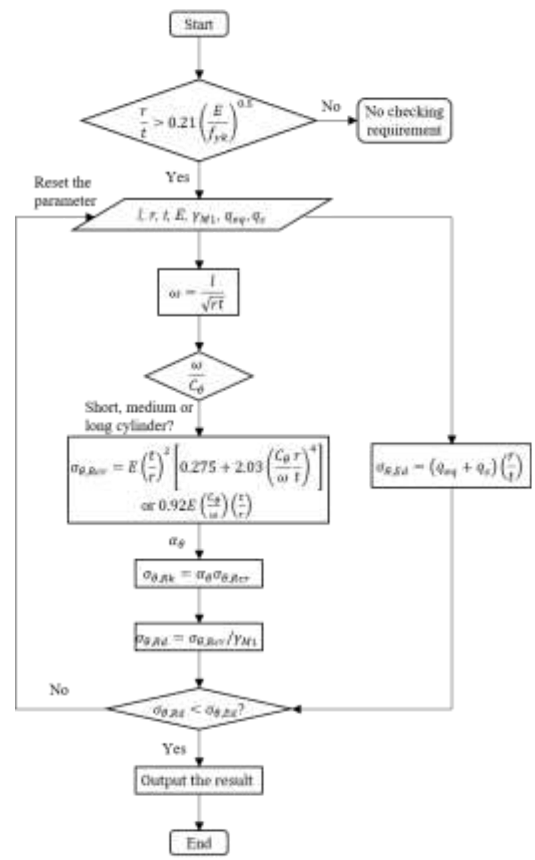

Figure 6. Flow chart of circumferential buckling calculation

Cylinders need be checked against circumferential shell buckling if they satisfy:

$$\frac{r}{t} > 0.21\sqrt{\frac{E}{f_{yk}}} \tag{9}$$

The flow chart of circumferential buckling calculation is shown in Figure 6.

Cylinders need be checked against shear shell buckling if they satisfy:

$$\frac{r}{t} > 0.16\left(\frac{E}{f_{yk}}\right)^{0.67} \tag{10}$$

The flow chart of shear buckling calculation is shown in Figure 7.

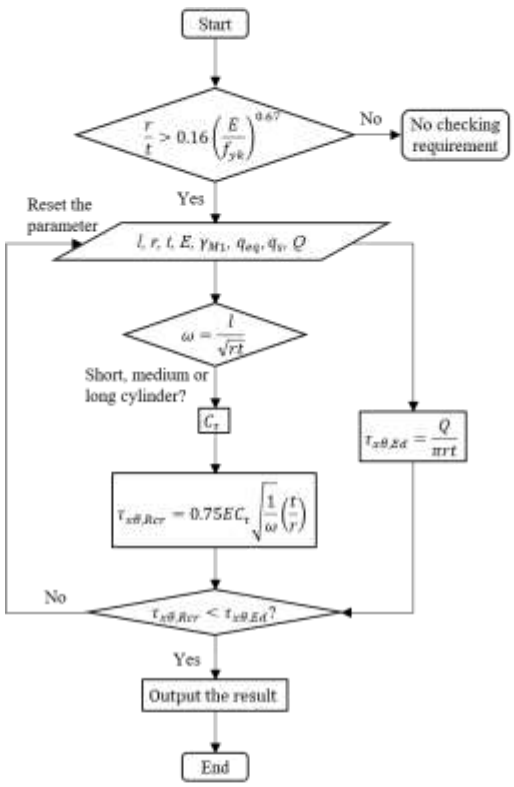

Figure 7. Flow chart of shear buckling calculation

In this case, the cylinder is exposed to the extreme environmental conditions leading to meridional compression, circumferential compression and shear. So the buckling interaction of the three membrane stress should be checked. The flow chart for buckling interaction checking is provided in Figure 8.

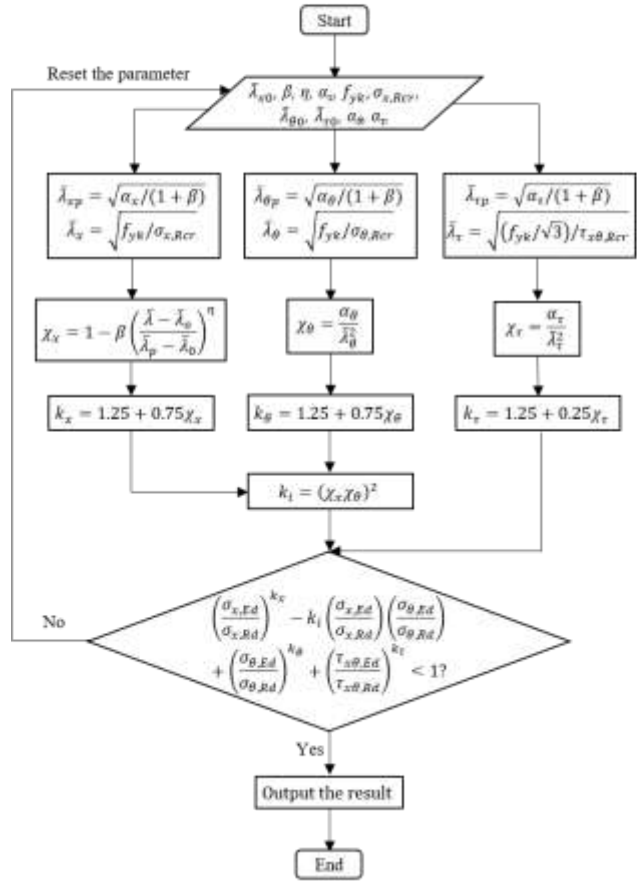

Figure 8. Flow chart for buckling interaction checking

4.2 Stiffened shell structure design

Shell buckling is usually the major failure mode of a shell structure, due to its large slenderness. A relatively thick structure is usually required to reach the design resistance for a plane shell structure compared with the stiffened structure. However, this may consume unnecessary steel materials, which disadvantages the floatability and sustainability of the floating FWT. Stiffeners may be essential for some large scale floating structures.

4.2.1 DNVGL-RP-C202

In DNVGL, the shell cylinder may be stiffened by longitudinal stiffeners and/or ring frames illustrated in Figure 9. The following buckling modes should be checked: a) shell buckling (unstiffened curved panels), b) panel stiffener buckling, c) panel ring buckling, d) general buckling, e) column buckling. In this case, the panel ring is provided to avoid the general buckling so the checking of general buckling can be ignored. Shell buckling and column buckling have been explained in 4.1.1 and the others are described in this section.

The panel stiffener is assumed to bear the longitudinal force subjected to axial force and bending moment with shell panel, so it is permissible to replace the shell thickness by the equivalent thickness. The checking procedure is similar to the shell structure without stiffener but the some coefficients are changed because of the change of equivalent thickness. Additionally, the

Geometric requirement of web and flange should be checked to prevent the local buckling of stiffeners. Calculation procedure of panel stiffener buckling is provided in Figure 10.

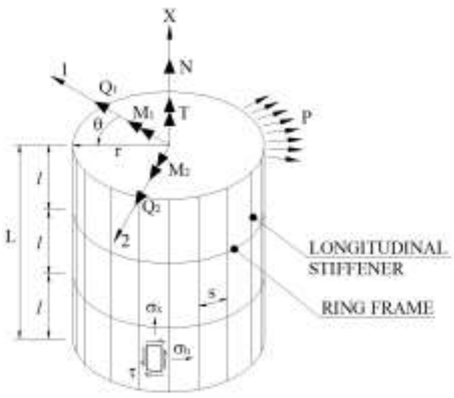

Figure 9. Stiffened cylindrical shell [17]

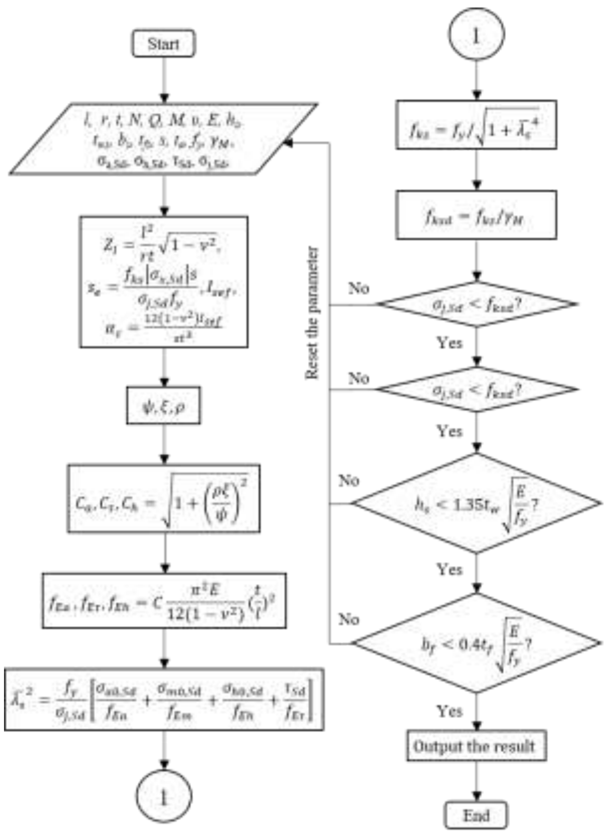

Figure 10. Flow chart of panel stiffener buckling calculation

On the basis of DNVGL-RP-C202, the main role of panel ring is assumed to bear the circumferential force subjected to lateral pressure. To avoid the panel ring buckling mode, the effective moment of inertia of the ring section should not be less than $I_R=I_x+I_{xh}+I_h$ (see Figure 11). I_x, I_{xh}, I_h represent the minimum requirement of the moment of inertia of ring frames subjected to axial compression with bending, shear and lateral pressure respectively. It has been calculated that I_h accounts for a large proportion (around 74%) in I_R.

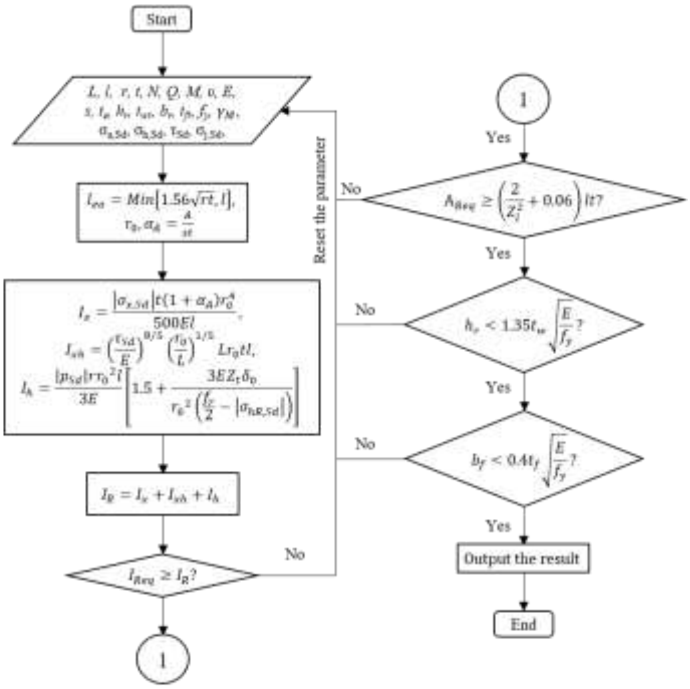

Figure 11. Flow chart of panel ring buckling calculation

4.2.2 EN 1993-1-6

EN 1993-1-6 considers three case for ring stiffener on cylindrical shell: 1) Ring stiffened cylinder: radial force on ring, 2) Ring stiffened cylinder: axial loading, 3) Ring stiffened cylinder: uniform internal pressure. The design of longitudinal stiffeners have not been provided in this standard. Similar with the other codes EN 1993-4-1 [20] and EN 1993-4-2 [21], this three standards are normally used to design tank and silos which contain liquid and solid without external pressure. However, the floating foundation of wind turbine is surrounded by seawater that means it would be impacted by extreme wave loads. On the other hand, the panel ring is welded on the external surface according to Eurocode but it is on the internal surface based on actual case. Whether EN 1993-1-6 can be used to design the orthogonally stiffened shells remains to be discussed.

5 CASE STUDY

The design of the column supporting the tower with wind turbine is challenging and critical. In this section, unstiffened and stiffened shell structure of the column are considered in structural design based on the DNVGL standard and Eurocode and the design results will be compared. The columns are designed with grade S275 structural steel. The yield strength is 275 Mpa; Young's modulus is 210 Gpa; Poisson's ratio is 0.3.

5.1 Unstiffened shell structure design

An Excel design spreadsheet has been developed to calculate the structural scantlings of the single column of the WindFloat according to DNVGL-RP-C202 and EN 1993-1-6.

The calculation procedure follows the flow charts given in this paper. To obtain the minimum shell thickness, the other geometry (length of column: 28 m, column diameter: 10.7 m) and material parameter remain unchanged. The design load cases and their components are shown in Table 5.

Table 5 Summary of load combination

Load case	1.0G+1.0Q+1.35E		1.25G+1.25Q+0.7E	
	Wind	Wave	Wind	Wave
Axial force	23600 kN		29400 kN	
Shear	1200 kN	3000 kN	620 kN	1550 kN
Bending	36540 kNm	20000 kNm	19000 kNm	10200 kNm

G: permanent load; Q: variable functional load; E: environmental load (wind loads, wave loads).

As for DNVGL rules, in order to bear the maximum von Mises' stress $\sigma_{j,Sd}$ = 11.25 Mpa induced by combination loads effect, the thickness of the shell is 76 mm providing the design buckling strength of a shell for f_{ksd} = 11.4 Mpa. On the other hand, the interaction check for the combined membrane stress state for axial force, shear force and bending moment should be carried out following Eurocode. The minimum thickness of the shell is 81 mm and the result of the interaction check is 0.964 which is smaller than 1.

The results of the two standards are rather similar, and the two recommended methods are semi-empirical methods considering the effect of geometric imperfections. DNVGL uses the reduced buckling coefficient C and the reduced shell slenderness $\bar{\lambda}_s^2$ to calculate the buckling strength of a shell. By contrast, Eurocode proposes the imperfection reduction factor α and the buckling reduction factor χ to analyse the design value of stress. One of the reasons why the shell thickness due to Eurocode is bigger than that due to DNVGL is the definition of the factor C_θ which is used to calculate the elastic critical circumferential buckling stress $\sigma_{\theta,Rcr}$. There is no specific recommendation for buckling factors for long cylinders C_θ so a conservative value 0 has been used in this research to define buckling stress $\sigma_{\theta,Rcr}$.

5.2 Stiffened shell structure design

An Excel spreadsheet using Solver Add-in function has been developed to calculate the structural scantlings. Table 6 summarizes the scantlings optimisation results of the external shell and internal stiffeners (Ring and stringer). Not only the weight of structural steel but also the fabrication and transportation of the FWT should be considered in cost analysis. Case 1 considers the mass of shell structure only and case 2 considers the costs in the fabrication analysis shown in Table 7, such as labour welding and filler material. Labour welding is assumed that a welder can complete 2 metres of 5 mm fillet weld per hour. The mass of 5 mm fillet is assumed as around 0.356 kg/m. Of course, the spread depends on the actual conditions and location.

As can be seen from Table 6, the thickness of shell, stringer web and stringer flange of Case 1 is much smaller than that of Case 2. The philosophy of optimised structural scantlings based on Case 1 considering mass reduction only is to decrease the steel thickness and to contribute the strength of material as much as possible. As a result, with the compact and thin stiffeners, the structure seems a mesh structure which has many connections.

According to Table 8, although the cost of structural steel is cut down to 62,490 € (62,490 kg) in Case1, which is only half cost of Case 2, the cost of labour welding and filler material are surprising. When the fabrication of connection has been considered, the cost is dropped sharply from 888,090 € to 188,370 € although the cost of structural steel is double.

Table 6. Optimised results of the structural scantlings (unit: mm)

	Case 1	Case 2
Shell thickness t	5.7	15.0
Distance of ring frames l	2165.8	2840.2
Stringer web width h_s	24.6	70.8
Stringer web thickness t_{ws}	0.66	1.9
Stringer flange width b_s	14.1	13.4
Stringer flange thickness t_{fs}	0.6	5.5
Spacing of stringer s	49.8	741.0
Ring web width h_r	435.0	464.1
Ring web thickness t_{wr}	11.7	12.4
Ring flange width b_r	11.7	12.4
Ring flange thickness t_{fr}	0	0

Table 7. Costs considered in the fabrication analysis [22]

Cost voice	Specification	Unit cost
Steel plate	Structural steel	1 €/kg
Labour welding	-	20 €/h
Filler material	Solid wire	2 €/kg

Table 8. Costs considered in the fabrication analysis (unit: €)

Cost item	Case 1	Case 2
Steel	62,490 (62,490 kg)	127,910 (127,910 kg)
Labour welding	680,000 (34,000 h)	49,800 (2,490 h)
Filler material	145,600 (72800 kg)	10,660 (5330)

Total Cost	888,090	188,370

5 CONCLUSIONS

This paper has discussed the wind and wave loads calculation, the design process of two standard (DNVGL-RP-C202 and EN1993-1-6) and the detailed shell structure design of a floating foundation for an offshore wind turbine.

Wind and wave loads are calculated based on the DNV standard. It is found that the results of the total wave loads by using Flow-3D, hand calculation and AQWA are similar, approaching to about 4000 kN on the surface of the shell structure.

The design process of two standard by using flow chart are provided in this paper. In DNVGL-RP-C202, the design equivalent von Mises' stress $\sigma_{j,Sd}$ is compared with the design buckling strength of a shell f_{ksd}. In comparison, the interaction check for the combined membrane stress state is carried out in EN 1993-1-6.

By introducing the stringer and ring stiffeners, the shell thickness is reduced sharply from 76 mm to 15 mm based on DNVGL guidelines. Eurocode provides slightly more conservative solutions compared with DNVGL: the minimum shell thickness is 81 mm without internal stiffeners if using Eurocode.

REFERENCES

[1] Hussein, K.R., et al., *Structural design of a floating foundation for offshore wind turbines in red sea*. Analysis and Design of Marine Structures, Marstruct 2013, ed. C.G. Soares and J. Romanoff. 2013, Boca Raton: Crc Press-Taylor & Francis Group.

[2] Tyler Stehly, P.B., Donna Heimiller, and George Scott, *2017 Cost of Wind Energy Review*. 2017, National Renewable Energy Laboratory: Golden, USA.

[3] Cheng, Z., et al., *Power performance and dynamic responses of a combined floating vertical axis wind turbine and wave energy converter concept.* Energy, 2019. **171**: p. 190-204.

[4] FrankLemmer, K.l., WeiYu,DavidSchlipf, PoWenCheng, *Optimization of floating offshore wind turbine platforms with a self-tuning controller*, in *ASME 2017 36th International Conference on Ocean*. 2017: Trondheim, Norway.

[5] Ruzzo, C., et al., *Progress on the experimental set-up for the testing of a floating offshore wind turbine scaled model in a field site.* Wind Engineering, 2016. **40**(5): p. 455-467.

[6] Zhou, S., et al., *Directionality Effects of Aligned Wind and Wave Loads on a Y-Shape Semi-Submersible Floating Wind Turbine under Rated Operational Conditions.* Energies, 2017. **10**(12): p. 2097.

[7] Chew, K.-H., et al., *Optimization of Offshore Wind Turbine Support Structures Using an Analytical Gradient-based Method.* Energy Procedia, 2015. **80**: p. 100-107.

[8] Niklas, K., *Strength analysis of a large-size supporting structure for an offshore wind turbine.* Polish maritime research, 2017. **24**: p. 156-165.

[9] Kim, D.H. and J.K. Paik, *Ultimate limit state-based multi-objective optimum design technology for hull structural scantlings of merchant cargo ships.* Ocean Engineering, 2017. **129**: p. 318-334.

[10] Natarajan, A., M. Stolpe, and W. Njomo Wandji, *Structural optimization based design of jacket type sub-structures for 10 MW offshore wind turbines.* Ocean Engineering, 2019. **172**: p. 629-640.

[11] Lefebvre, S. and M. Collu, *Preliminary design of a floating support structure for a 5MW offshore wind turbine.* Ocean Engineering, 2012. **40**: p. 15-26.

[12] DNVGL-RP-C202, *DNVGL-RP-C202 Buckling strength of shells*. 2017, DNV GL.

[13] EN1993-1-6, *en.1993.1.6.2007 shell structures*. 2007, European Committee Standardization: Brussels.

[14] Roddier, D., C. Cermelli, and A. Weinstein, *Windfloat: A Floating Foundation for Offshore Wind Turbines.* Omae 2009, Vol 4, Pts a and B, 2010: p. 845-853.

[15] https://cleantechnica.com/2011/12/06/2-mw-windfloat-towed-to-atlantic-site-off-portuguese-coast/.

[16] Liu, L., et al., *Structural Strength Analysis of a Tri-Floater Floating Foundation for Offshore VAWT.* Journal of Ocean University of China, 2018. **17**(4): p. 753-762.

[17] Bilgili, M., A. Yasar, and E. Simsek, *Offshore wind power development in Europe and its comparison with onshore counterpart.* Renewable and Sustainable Energy Reviews, 2011. **15**(2): p. 905-915.

[18] DNVGL-RP-C205, *DNVGL-RP-C205 Environmental conditions and environmental loads*. 2017, DNV GL.

[19] DNVGL-ST-0119, *DNVGL-ST-0119 Floating wind turbine structures*. 2018, DNV GL.

[20] EN1993-4-1, *en.1993.4.1.2007 Design of steel structures silos*. 2007, European Committee Standardization: Brussels.

[21] EN1993-4-2, *en.1993.4.2.2007 Design of steel structures tank*. 2007, European Committee Standardization: Brussels.

[22] Quintana, J.A., *Alternative Installation Methods for offshore wind substation*, in *Department of Mechanical and Aerospace Engineering*. 2016, University of Strathclyde: University of Strathclyde Engineering.