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HYBRID FINITE ELEMENT METHOD DEVELOPMENT FOR OFFSHORE STRUCTURES' CALCULATION WITH THE IMPLEMENTATION OF INDUSTRY STANDARDS

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ABSTRACT

In the design process of offshore steel structures, it is typical to employ commercial calculation codes in which simulation and evaluation of results are performed on the basis of the available standards (e.g. API, DNV, Lloyds). The modeling and solution rely on finite element methods and cover the simulation of the structure's properties along with the influence of the marine environment – sea currents, wave and wind loading, as well as the influence of vibrations, buoyancy and accompanying mass of water. Both commercial and open source mathematical modeling software which is available nowadays allows for cost effective and flexible implementation of advanced models for offshore industrial structures with high level of credibility and safety. The models can be built to suit task-specific requirements and evaluated on the basis of the selected criterial system best suited to the needs of the customer. Examples of methodology for environmental and structural model development are presented, along with simulation results covering a wide scope of data, ranging from stress and deformation to resonant characteristics and issues of technological feasibility.

Keywords: offshore structures,design calculations,sea loading,stress evaluation,FEM

INTRODUCTION

Correct designing of steel offshore structures is a responsible task due to numerous threats to human life and health, as well as to their possessions and natural environment which may result from possible failure of such a structure. The report prepared for UK Health & Safety Executive [28] reveals that in exploitation practice of drilling and production platforms with human crew, about 10 % of accidents involving bodily injury result from constructional errors, i.e. errors made at early project implementation stages.

One of basic issues related with designing a drilling rig is identifying environmental conditions to be faced on the route of its transportation and final foundation. These conditions include environmental loads acting on the rig during its transportation, foundation, and operation in extreme sea states. The information on parameters determining environmental conditions for the offshore structure can be found in standards and regulations of classification societies [1–3, 13, 16]. Taking into consideration extreme environmental conditions is necessary when calculating the strength of the structure. There are a number of methods to assess the extreme sea state parameters. These methods fall into two categories: short-term and long-term. The parameters considered in the model of structure's load include sea currents and waves, sea lichens, forced vibrations, or even the action of sea ice cap [4, 5, 7, 9, 22]. For structures with long exploitation lifetime, a very important factor is its fatigue life assessment. This refers to both stationary [1 – 3, 13, 16] and floating structures [6, 7].

For economic reasons, the entire methodology of marine structure design calculations should be time- and cost-efficient, at the same preserving acceptable representativeness. As a rule, commercial computer codes make use of beam models of steel structures, with shell models additionally introduced at nodes exposed to stress concentration and fatigue. This model architecture makes it possible to minimise computing power demand (simplicity of beam model) and to analyse maximum stresses (shell model). However, it requires combined use two types of finite elements, and offers only simplified analysis of interaction between the fluid and slender elements of the designed structure.

As an alternative to that approach, a methodology is proposed in the article which consists in developing a model entirely based on shell elements, with complete representation of the geometry of the load bearing structure, but with simplification of passive elements (risers, for instance) and the deck module to a single equivalent inertia, the action of which is passed in the way reflecting the real action of those elements. The advantages of this method include increased accuracy in specifying marine environment load application points and complete information on stress distributions in walls of pipe elements and profiles composing the supporting structure of the entire offshore structure, all this achieved at comparable demand for computing power and memory of the used computer. To improve further the economy of computations, a detailed analysis of the model of wave motion was performed and a simplified model was selected which had similar accuracy to that resulting from the theory [4, 5].

The discrete structure was modelled using a hybrid finite element method [15], which is a combination of the method of deformable finite elements and the concentrated mass method. This method is based on the assumption that parts of the marine structure which are responsible for its response to the applied load are modelled using deformable finite elements. In the supporting structure for the stationary platform, these are the shell elements which ensure complete representation of its geometry and provide opportunities for analysing stress concentrations at model nodes. The remaining parts of the structure, which are not involved in generating response to the applied load, are modelled using concentrated mass elements. These elements are not simply mass points, but masses with attributed tensors of inertia. The geometric representation of a mass point is a point, while concentrated masses can be connected with each other, and with deformable elements, elastic-damping constraint elements (like in the finite element method [24, 25, 26, 27]), elastic elements, and/or massless rigid elements. External loads can be applied directly to mass points, or via elastic-damping elements, elastic elements, or massless rigid elements. The above method of digitisation makes it possible to increase the accuracy of modelling in regions with greatest impact on the results of calculations, at the same time reducing the dimensions of the matrix describing the model of the entire structure and thus shortening the computing time, which is of high importance when performing long-lasting analyses with iterations [15].

ENVIRONMENTAL LOADS OF MARINE STRUCTURE

According to the standard API RP 2A-WSD [2], when calculating both floating marine structures and those founded on the seabed, the loads generated by the following environmental phenomena should be taken into account:
1) wind,
2) sea current,
3) sea wave,
4) ice cap,
5) waves generated by earthquakes,
6) earthquakes.

Additionally, accidental loads generated, for instance, by fire or explosion, or when the structure is hit by a ship, should also be taken into account.

In the case of slender marine structures, such as production platforms of jacket type, production pipe casings (risers), pipelines, or ropes anchoring the structure, of high importance is the flow of fluid in the vicinity of the structure. In those cases, the wave generated loads can be calculated using the Morison equation [3, 7, 29, 30], which is based on the assumption that the load of the structure coming from the fluid flowing around it is equal to the sum of forces of inertia and flow resistance.

In a slender marine structure, vortex induced vibrations (VIV) can appear when the frequency of changes of the vortex motion of the fluid (caused by waves or sea current) is in resonance with the free-vibration frequency of the structure. Therefore, pipe calculations should also include the analysis of fatigue loads coming from waves and water vortices.

WEIGHT OF STRUCTURE

The weight of the structure is calculated directly from the FEM model, based on material density and volumes of finite elements. The FEM codes automatically attribute concentrated forces and continuous loads coming from structure's weight to particular finite elements. The weight of the marine structure equipment is represented in the model by pressures and concentrated forces applied at selected nodes.

FUNCTIONAL LOADS

When analysing structure's elements, the use of variable loads provides opportunities for studying loads coming from different configurations of the transported cargo, and thus determining extreme load values. Variable loads are taken into account in FEM models as pressures applied to selected decks and tank walls.

WIND CAUSED LOADS

It is assumed that the marine structure loads caused by wind are static in nature [1, 2]. The wind thrust force is a function of wind pressure and wind thrust area [1]:

$$F_w = P_w \cdot A_w \tag{1}$$

where:
P_w – wind thrust pressure:

$$P_w = V^2 \cdot C_h \cdot C_s \tag{2}$$

V – wind speed,
C_h – height coefficient,
C_s – shape coefficient,
A_w – wind thrust area.

The definitions of height and shape coefficients are given in regulations of classification societies. The wind thrust pressure (2) is calculated based on the Bernoulli theory of fluid flow, according to which the fluid acts with the following dynamic pressure on the motionless body [1]:

$$P_w = \frac{1}{2}\rho V^2 \tag{3}$$

where:
ρ – fluid density.

In the majority of cases, the wind generated loads are applied to the structure's model in the form of concentrated horizontal forces acting at certain heights above sea level.

LOADS CAUSED BY WAVES AND SEA CURRENT

Hydrodynamic loads caused by waves, and forces of inertia caused by structure's rolling are low-frequency dynamic loads. To assess them, simplified quasi-static models can be used [1, 2, 30]. It is noteworthy that these calculations should be performed taking into account further fatigue analysis, in which the main load component is the action of waves, due to its cyclic nature.

In the case of floating marine structures, the calculations should be performed for extreme sea state parameters. These calculations take also into account forces of inertia coming from averaged motion of the structure.

In the case of stationary marine structures, the calculations are usually performed for a series of cases to:

1) accurately model the wave motion in the marine structure foundation place,
2) accurately describe the transmittance and load amplification factors.

The response of the marine structure is obtained as a result of the analysis of regular, design, and stochastic waves.

The load of the marine structure caused by the action of extreme wave [1, 2, 30] can be calculated using commercial computer codes, such as AQUA, WAMIT, etc. [7]. These calculations should take into account different wave directions, in order to assess maximum stresses related to those directions.

The force distributions over marine structure beams caused by the flow of water particles in their vicinity is calculated from Morrison's equation (4). This approach is widely used in standards [1, 2, 3, 7, 16]. It makes it possible to obtain results close to reality, provided that the dimensions of the beam's cross section are several times smaller than both the wave length and the characteristic distance between neighbouring beams [1, 2, 30]. These conditions are met for typical stationary marine structures. The continuous load per unit length of the cylinder with diameter D consist of two components [1]:

$$\bar{F} = \bar{F}_n + \bar{F}_t \tag{4}$$

where:
$\bar{F}_n$ – continuous load perpendicular to cylinder surface,
$\bar{F}_t$ – continuous load tangential to cylinder surface (parallel to cylinder axis).

Each component can be presented as a function of water particle velocity and calculated from Morrison's equation. In the direction perpendicular to the cylinder surface, the continuous load is given as [1]:

$$\bar{F}_n = \bar{F}_{Dn} + \bar{F}_{In} \tag{5}$$

where:
$\bar{F}_{Dn}$ – resistance force of normal flow:

$$\bar{F}_{Dn} = \frac{1}{2} C_{Dn} D \rho \bar{V}_n |\bar{V}_n| \tag{6}$$

$\bar{F}_{In}$ – force of inertia of normal flow [1]:

$$\bar{F}_{In} = \frac{1}{4}\pi C_{Mn} D^2 \rho \dot{\bar{V}}_n \tag{7}$$

C_{Dn} – resistance coefficient of the flow normal to the beam surface,
C_{Mn} – inertia coefficient of the flow normal to the beam surface,
ρ – water density,
V_n – normal component of the relative velocity of water particles with respect to the beam element.

In the direction tangential to the beam surface, it is assumed that the effect of the forces of inertia of the flow is negligibly small and the continuous load in this direction depends only on flow resistance [1]:

$$\bar{F}_t = \bar{F}_{Dt} = \frac{1}{2}\pi C_{Dt} D\rho \bar{V}_t |\bar{V}_t| \quad (8)$$

where:
C_{Dt} – resistance coefficient of the flow tangential to the beam surface,
V_t – tangential component of the relative velocity of water particles.

These forces can be expressed in the local coordinate system x, y, z, in which the x-axis coincides with the beam axis, while the y- and z-axes are perpendicular to the beam axis [1]:

$$\begin{aligned} \bar{F}_x &= \frac{1}{2}\pi C_{Dt} D\rho |\bar{V}_t| \bar{V}_t \\ \bar{F}_y &= \frac{1}{2} C_{Dn} D\rho |\bar{V}_n| \bar{V}_y + \frac{1}{4}\pi C_{Mn} D^2 \rho \dot{\bar{V}}_y \\ \bar{F}_z &= \frac{1}{2} C_{Dn} D\rho |\bar{V}_n| \bar{V}_z + \frac{1}{4}\pi C_{Mn} D^2 \rho \dot{\bar{V}}_z \end{aligned} \quad 9)$$

where: x, y, z are the axes of the local coordinate system fixed to the beam.

For non-cylindrical beams, the above equations consider differences in hydrodynamic behaviour of the beam in two directions: y and z, perpendicular to the beam axis. In this case, the continuous loads of the beam in the local coordinate system are given as [1]:

$$\begin{aligned} \bar{F}_x &= \frac{1}{2}\pi C_{Dt}(D_y + D_z)\rho |\bar{V}_t| \bar{V}_t \\ \bar{F}_y &= \frac{1}{2} C_{Dy} D_y \rho |\bar{V}_n| \bar{V}_y + \frac{1}{4}\pi C_{My} D_y^2 \rho \dot{\bar{V}}_y \\ \bar{F}_z &= \frac{1}{2} C_{Dz} D_z \rho |\bar{V}_n| \bar{V}_z + \frac{1}{4}\pi C_{Mz} D_z^2 \rho \dot{\bar{V}}_z \end{aligned} \quad (10)$$

where:
C_{Dy}, C_{Dz} – flow resistance coefficients in the direction of y- and z-axis, respectively, in the local coordinate system fixed to the beam,
D_y, D_z – effective dimensions of the beam's cross section in the directions of y- and z-axis, respectively, in the local coordinate system fixed to the beam.

MODELLING OF WIND WAVE

Wave is the deformation of the sea free surface, which moves from one point to the other in the given water region. For instance, after dropping a stone into the water, the free surface deformation propagates around the point at which the stone hit the water (circular wave). It is the free surface deformation which propagates, and not the water mass itself. Currently, the surface wave in water can be described using one of five theories [2, 7, 34]:
1) gravitational wind wave (linear model),
2) fifth-order Stokes wave,
3) stream function,
5) trochoidal (cnoidal) wave,
6) solitary wave.

Each of these theories enables obtaining an approximate solution to equations of flow motion within certain ranges of wave amplitudes and periods, and water depths. Fig. 1 presents parameters describing the sea wave [34].

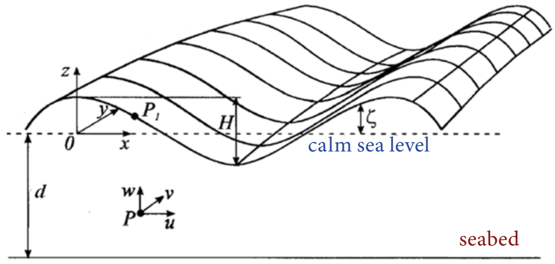


Fig. 1. Parameters describing the sea wave: H – wave height, d – water depth, ζ – free surface deflection, x-y-z – global coordinate system, u-v-w – local coordinate system fixed to fluid element P, P1 – fluid element situated on the free surface [34]

For gravitational wind waves, it is assumed that [7, 9, 34]:
1) at large water depths, the effect of surface wave action is limited to a small surface layer,
2) water displacements, velocities and accelerations in the horizontal and vertical directions are of the same order,
3) accelerations in the vertical direction are of the order of acceleration due to gravity, g,
4) the Coriolis acceleration is negligibly small due to short period of motion,
5) the motion is unsteady, i.e. [7, 9, 34]:

$$\frac{\partial}{\partial t} \neq 0 \quad (11)$$

6) the mass of water does not travel with the deformed wave (linear wave model)

The issues of water motion in wind wave can be analysed as a special case of the boundary value problem of partial differential equations [34]. The boundary conditions defined for the free surface of the wave include kinematic and dynamic conditions. For the seabed, the kinematic boundary condition is defined. Additionally, side boundary conditions can be defined, along with the condition for wave motion damping. It is assumed in the calculations that the fluid is incompressible, and the water motion is irrotational. Consequently, it can be assumed that the water flow is potential [34].

Let us consider the case of wave travelling in the direction of decreasing x values. The value of the potential is given by its real part:

$$\Phi = a\frac{\omega}{k}\frac{\cosh[k(z+d)]}{\sinh(kd)}\sin(-kx - \omega t + \theta) \quad (12)$$

The free surface deflection is affected by the real part of the potential (with the imaginary part omitted):

$$\zeta = a \cdot \cos(-kx - \omega t + \theta) \quad (13)$$

where:
a – wave amplitude,
ω – circular frequency, pulsation,
k – wave number [7]: $k = \frac{2\pi}{L}$,
t – time,
Θ – constant.

Under the sea surface, the fluid elements make orbital motions. These motions can have different orbits, depending on water depth, see Fig. 2. The water is considered deep when the water depth d is larger than half of the wave length L.

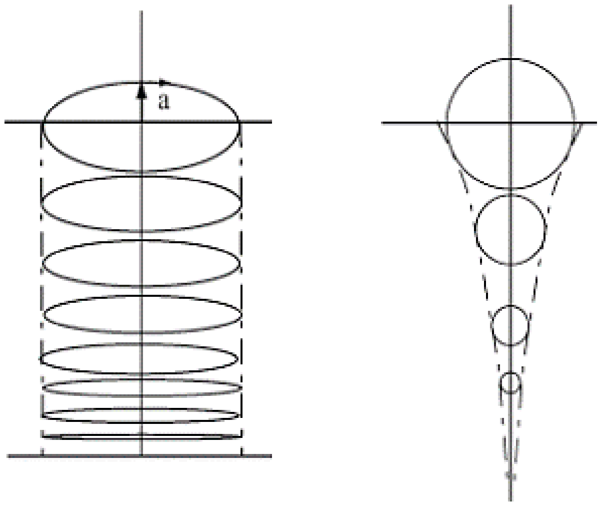


Fig. 2. Orbital motions of fluid elements: a) shallow water, b) deep water [34]

The velocity components of the fluid element in orbital motion are (Fig. 1):

$$u = -\frac{\partial\Phi}{\partial x} = a\omega\frac{\cosh[k(z+d)]}{\sinh(kd)}\cos(-kx - \omega t + \theta) \tag{14}$$

$$w = -\frac{\partial\Phi}{\partial z} = -a\omega\frac{\sinh[k(z+d)]}{\sinh(kd)}\sin(kx - \omega t + \theta) \tag{15}$$

The acceleration components of the fluid element in orbital motion are given by:

$$a_x = \frac{\partial u}{\partial t} = -a\omega^2\frac{\cosh[k(z+d)]}{\sinh(kd)}\sin(kx - \omega t + \theta) \tag{16}$$

$$a_z = \frac{\partial w}{\partial t} = -a\omega^2\frac{\sinh[k(z+d)]}{\sinh(kd)}\cos(kx - \omega t + \theta) \tag{17}$$

Let us assume that the constant θ for the wave with height H several times smaller than both wave length L and water depth d is equal to zero. Then, with the use of Eq. (12), the velocity potential can be written as:

$$\Phi \approx a\frac{\omega}{k}\frac{\cosh[k(z+d)]}{\sinh(kd)}\sin(-kx - \omega t) \tag{18}$$

Taking into consideration the definitions of wave circular frequency and wave number, the velocity potential takes the following form:

$$\Phi \approx a\frac{\frac{2\pi}{T}}{\frac{2\pi}{L}}\frac{\cosh\left[\frac{2\pi}{L}(z+d)\right]}{\sinh\left(\frac{2\pi d}{L}\right)}\sin\left(-\frac{2\pi}{L}x - \frac{2\pi}{T}t\right) \tag{19}$$

Then, assuming that the wave shape is close to a sine wave, for which the amplitude a is equal to half of its height H, we arrive at:

$$\Phi \approx \frac{HL}{2T}\frac{\cosh\left[\frac{2\pi}{L}(z+d)\right]}{\sinh\left(\frac{2\pi d}{L}\right)}\sin\left[2\pi\left(-\frac{x}{L} - \frac{t}{T}\right)\right] \tag{20}$$

Finally, the free surface deflection in the simplified free-surface model is given as:

$$\zeta \approx \frac{H}{2}\cos\left[2\pi\left(-\frac{x}{L} - \frac{t}{T}\right)\right] \tag{21}$$

Eq. (21) does not depend on sea depth d. The shape of the wave surface, which in fact depends on the sea depth, can be determined directly from the dynamic boundary condition for the free surface (full free surface model):

$$\zeta = \frac{1}{g}\frac{\partial\Phi}{\partial t}\bigg|_{z=0} = -\frac{\pi HL}{gT^2}ctgh\left(\frac{2\pi d}{L}\right)\cos\left[2\pi\left(-\frac{x}{L} - \frac{t}{T}\right)\right] \tag{22}$$

In this case, the velocity components of the fluid element in orbital motion are given by the following equations:

$$u = -\frac{\partial\Phi}{\partial x} = \frac{H\omega}{2}\frac{\cosh\left[\frac{2\pi}{L}(z+d)\right]}{\sinh\left(\frac{2\pi d}{L}\right)}\cos\left[2\pi\left(-\frac{x}{L} - \frac{t}{T}\right)\right] \tag{23}$$

$$w = -\frac{\partial\Phi}{\partial z} = -\frac{H\omega}{2}\frac{\sinh\left[\frac{2\pi}{L}(z+d)\right]}{\sinh\left(\frac{2\pi d}{L}\right)}\sin\left[2\pi\left(-\frac{x}{L} - \frac{t}{T}\right)\right] \tag{24}$$

and the acceleration components take the following form:

$$a_x = \frac{\partial u}{\partial t} = \frac{H\omega^2}{2}\frac{\cosh\left[\frac{2\pi}{L}(z+d)\right]}{\sinh\left(\frac{2\pi d}{L}\right)}\sin\left[2\pi\left(-\frac{x}{L} - \frac{t}{T}\right)\right] \tag{25}$$

$$a_z = \frac{\partial w}{\partial t} = \frac{H\omega^2}{2}\frac{\sinh\left[\frac{2\pi}{L}(z+d)\right]}{\sinh\left(\frac{2\pi d}{L}\right)}\cos\left[2\pi\left(-\frac{x}{L} - \frac{t}{T}\right)\right] \tag{26}$$

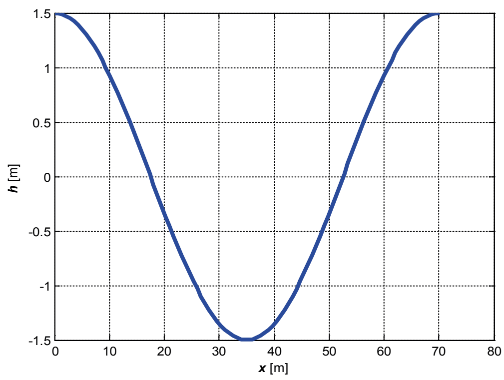


Fig. 3. Sea free surface deflection given by the simplified model, Eq. (21)

The wave models were implemented in Matlab environment with the following arbitrarily assumed parameter values:

1) wave height: $H = 3$ m,
2) sea depth: $d = 80$ m,
3) wave period: $T = 6.69$ s,
4) wave length: $L = 70.06$ m.

Fig. 3 presents the shape of wind wave free surface deflection which was determined using the simplified model given by Eq. (21). Differences between the wind wave free surface deflection shapes determined using the full model (22) and the simplified model (21) do not exceed ±2 mm, which confirms correctness of the assumptions adopted when developing the simplified model (21).

Fig. 4 shows the limiting shape of the wave velocity potential (for ¼ cycle of orbital motion of fluid element) determined for:

$$sin\left[2\pi\left(-\frac{x}{L}-\frac{t}{T}\right)\right]=1$$

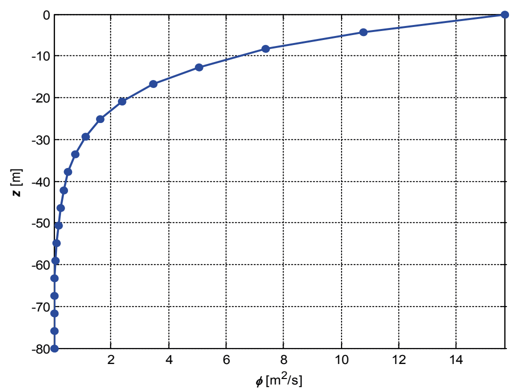


Fig. 4. Wave velocity potential for ¼ cycle of rotational motion of fluid element

Fig. 5 shows the limiting shape of velocity component u in orbital motion of fluid element (beginning of motion) determined for:

$$cos\left[2\pi\left(-\frac{x}{L}-\frac{t}{T}\right)\right]=1$$

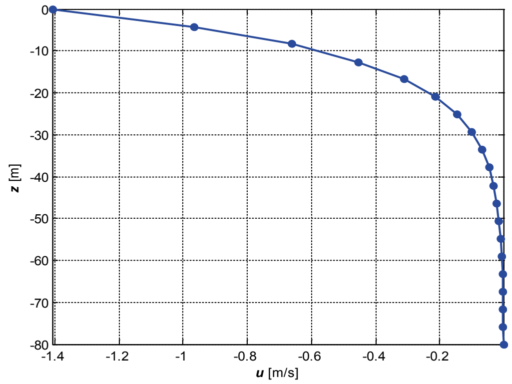


Fig. 5. Velocity component u in orbital motion of fluid

Fig. 6 shows the limiting shape of velocity component w in orbital motion of fluid element determined for:

$$sin\left[2\pi\left(-\frac{x}{L}-\frac{t}{T}\right)\right]=1$$

Fig. 7 illustrates the phenomenon of height increase of the wave with the assumed initial height L and period T when the sea depth decreases (wave passing through the shallows).

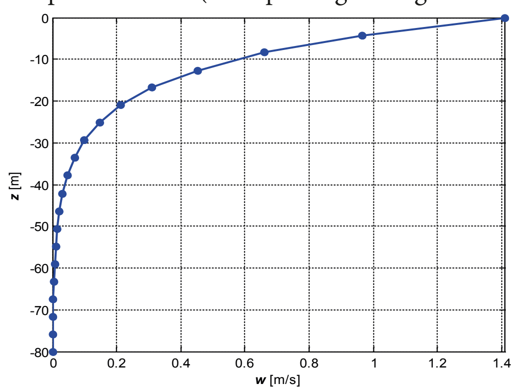


Fig.6. Velocity component w in orbital motion of fluid for ¼ cycle of rotational motion of fluid element

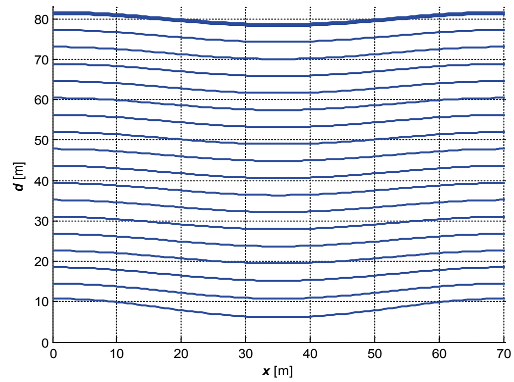


Fig. 7. Wave profile changes related with sea depth decrease

Fig. 8 shows the limiting shape of acceleration component a_x in orbital motion of fluid element determined for:

$$sin\left[2\pi\left(-\frac{x}{L}-\frac{t}{T}\right)\right]=1$$

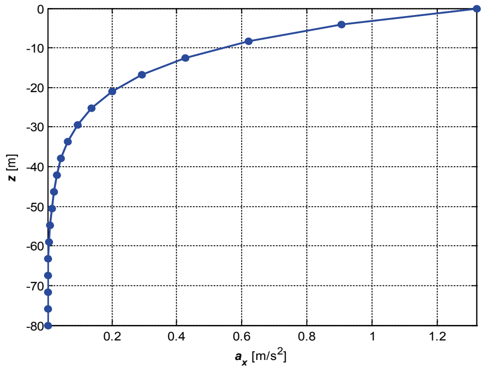


Fig. 8. Acceleration component a_x in orbital motion of fluid for ¼ cycle of rotational motion of fluid element

Fig. 9 shows the limiting shape of acceleration component a_z in orbital motion of fluid element determined for the beginning of motion:

$$cos\left[2\pi\left(-\frac{x}{L}-\frac{t}{T}\right)\right]=1$$

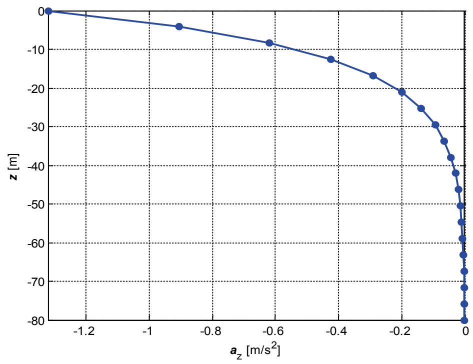
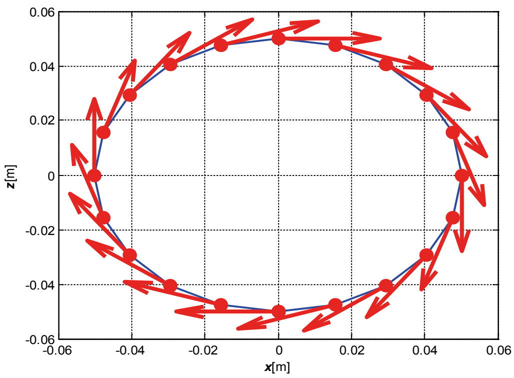


Fig. 9. Acceleration component a_z in orbital motion of fluid (beginning of rotational motion)

The motion trajectory of fluid element at depth of 37.9 m, with velocity and acceleration vectors for each element position, is shown in Fig 10, while Fig. 11 shows the trajectories at selected sea depths, starting from seabed up to free surface.



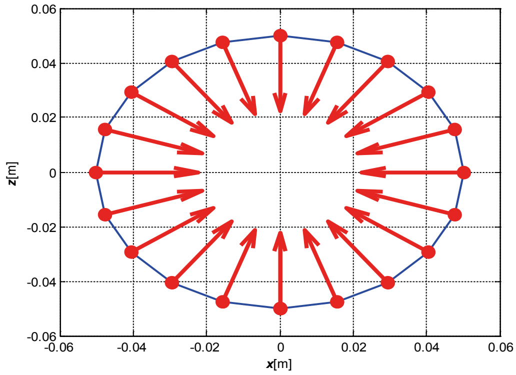


Fig. 10. Trajectory of rotational motion of fluid element at depth of 37.8947 m, with marked velocity vectors (top) and acceleration vectors (bottom)

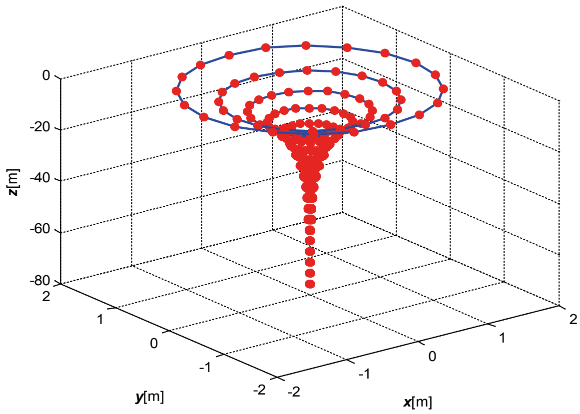


Fig. 11. Trajectories of fluid elements at selected depths

MODELLING OF MARINE STRUCTURES

Stationary marine structures are, as a rule, welded pipe structures with vertical or inclined legs connected with each other by stiffeners. The marine structure creates a support for upper, surface parts, such as pipelines, risers, and other equipment. The stages of marine structure modelling include [2, 29]:

1) determining needs of the project,
2) modelling conditions of marine environment and structure foundation,
3) working out preliminary design offers, with predicted possible places of installation, as well as conditions and costs of their execution,
4) selecting dimensions of the structure which will allow determining its load in each operating phase: well drilling, production, and well cleaning,
5) assessing the project with respect to erection technology and costs,
6) performing the analysis of load and stress distributions in the structure during its construction, transportation, and assembly in the foundation place,
7) analysing possibilities of structure's closing after termination of its operating lifetime.

The analyses of a stationary marine structure begin with determining geometric and material parameters of elements composing the structure, as well as parameters of its foundation and operation, and environmental and accidental loads.

Two basic types of analyses can be named:

1) linear analyses which aim at checking static and fatigue load limits based on regulations of classification societies (API RP2A [2], ABS [1], etc.);
2) nonlinear finite element method-based analyses which make it possible to determine the structure's response to accidental loads (such as collisions with other floating marine structures, fires, explosions, and earthquakes), and extreme environmental loads. These analyses are also performed for existing stationary marine structures.

The FEM analysis of a marine structure includes:

1) the analysis of the abovenamed environmental loads. Here, an additional factor taken into consideration is the effect of sea lichens which manifests itself in the increase of hydrodynamic load and total weight of the structure. The hydrodynamic model of the structure should include such equipment elements as drainage pipes, risers, vertical pipelines, caissons, ladders, stairs, etc. Depending on the type and amount of equipment elements, the action of waves on them should be considered. It is assumed that the equipment which is not permanently connected with the main frame of the marine structure (welded to the structure) is not a structure's element, and is only taken into account as additional load acting on the structure, a so-called non-structural member;
2) the foundation analysis, taking into consideration the interaction between the seabed soil and structure's elements;
3) the analysis of the load bearing structure and working decks.

The acting loads should be determined for each operating stage: well drilling, production, well cleaning, and their combinations. According to the standard NTS [23], the analyses of marine structures should allow to determine:

1) maximum shearing forces generated by waves and sea currents – needed for calculating dimensions of structure's reinforcements,
2) maximum restoring moments – needed for calculating dimensions of load bearing columns and the foundation,
3) maximum local forces acting on particular elements of the structure, which may occur at wave positions different than that generating the maximum global load for the entire structure.

The fatigue analysis of the marine structure should include all actions which are cyclic in nature and have relatively large frequency. This includes waves and other local phenomena caused by wave hitting against the structure, and by formation of vortices in water close to the structure.

According to the regulations of classification societies [1, 2], when the marine structure is founded on shallow water, the nonlinear wave theory should be applied. In those cases, deterministic fatigue analyses are also used. On the other hand, in cases of deep-water marine structures, the significance of dynamic phenomena is much higher than for shallow-water founded structures. In those cases, the fatigue analyses are performed for some frequency range (stochastic dynamic analyses). They require linearization of non-linear soil models. The stiffness matrix, which describes the structure's cooperation with the soil, is determined for the sea wave (with given parameters), the effect of which on the fatigue damage of the structure is most significant.

SAMPLE CALCULATIONS

The geometric model of the selected offshore structure, founded at depth of 80 m, is shown in Fig. 12. The object of the analysis was the stationary steel production platform consisting of the load bearing structure and the deck module. The geometric model for this structure was prepared and the strength calculations were performed in FEMAP environment.

The load bearing structure consists of three modules with four legs, between which a number of beams are extended in X-arrangement to increase the platform stiffness. On the upper plane of each module, guides for six production risers are mounted. The load bearing structure is the support for the deck module. The upper surface of the deck module is the production deck, below which the main technological deck is situated, with three mounted risers. Further below, there is the encased transportation deck.

The geometric model of the production platform was discretised using the earlier described hybrid finite element method, being a combination of the classic method of deformable finite elements and the concentrated mass method. The static and dynamic strength of the entire platform depends mostly on that of the load bearing structure. Therefore, this part was discretised using deformable finite elements, while the deck module and production risers were modelled using the concentrated mass method. The hybrid method was adapted for offshore structures based on the experience gained when developing models of large inland structures exposed to vibrations [15]. In Fig. 12, the structure discretised using deformable finite elements is marked green, while the deck and the production risers were reduced to concentrated masses marked yellow.

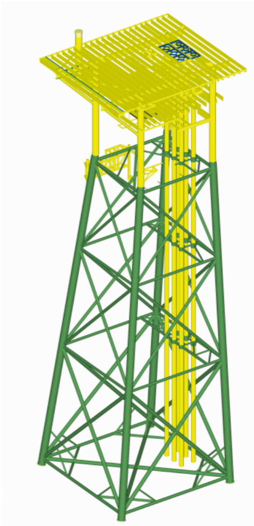


Fig. 12. Geometric model of production platform – isometric view

The deck module and the production risers were discretised using one concentrated mass, connected via two-dimensional rigid elements with the load bearing structure's legs and risers' guides. The parameters of inertia of this concentrated mass were determined by discretising the geometric model of the deck module and production risers with deformable finite elements. As a result, a complete model of the production platform was obtained, as shown in Fig. 13 presenting a net of finite elements, a symbolic representation of equivalent loads for the deck module, and constraints applied to supports of load bearing columns.

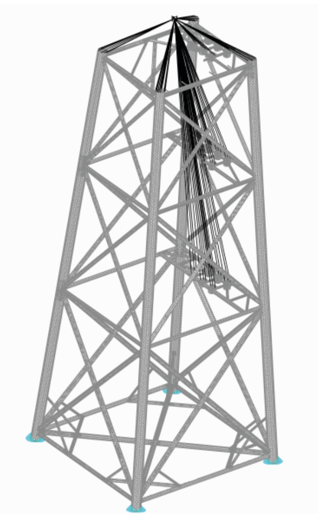


Fig. 13. Complete model of production platform

The model of marine environment was developed for the following parameters:
- wave height: $H = 3$ m,
- sea depth: $d = 80$ m,
- wave period: $T = 6.69$ s,
- wave length: $L = 70.06$ m.

The above described wave model was used for calculating the water thrust pressure acting on the production platform. The calculations took into consideration the self-weight of the structure. It was assumed that the structure' part situated under water surface is sunken, and, consequently, the buoyance force is negligibly small compared to the remaining loads of the structure. Fig. 14 shows the loads acting on the platform structure model.

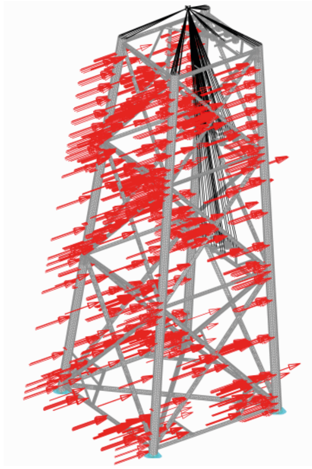


Fig. 14. Loads acting on the production platform structure model

As a result of calculations performed in FEMAP environment, the structure's deformations were obtained (Fig. 15) along with reduced stresses in deformable elements (Fig. 16, Fig. 17).

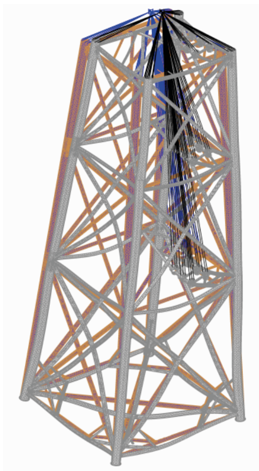


Fig. 15. Deformations of the production platform structure model

The use of deformable finite elements for discretising the load bearing structure has made it possible to calculate local stress concentrations at structure's nodes, with maximum values reaching up to 192 MPa. These stress concentrations were recorded at nodes situated close to the seabed, in the connection places of X-arranged beams with structure's legs (Fig. 17). Those places are the areas with the highest values of bending moments and forces shearing the entire load bearing structure, at simultaneous presence of notches caused by close vicinity of beams linked together at a given node.

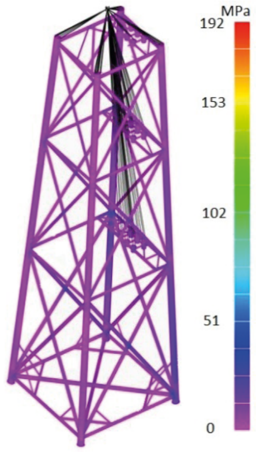


Fig. 16. Reduced stresses in deformable finite elements

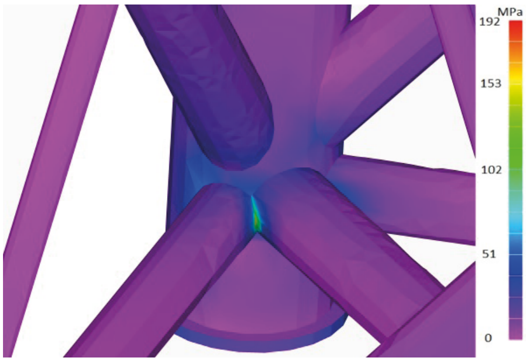


Fig. 17. Local stress concentrations at structure's node

CONCLUSIONS

The hybrid method combines together the advantages of the method of deformable finite elements and the concentrated mass method. It provides an opportunity for modelling complex structures, such as the production platform studied in the article. Deformable finite elements make it possible to analyse in detail the main part of the platform, which is the load bearing structure. In the analysed case, it allowed to determine local stress concentrations at structure's nodes. The remaining, complex platform parts, being the source of load for the load bearing structure, were discretised using the concentrated mass method, which resulted in substantial reduction in the number of finite elements used to discretise the geometric model of the deck module and production risers. In the analysed case, this reduction amounted to about 35%, while simultaneously preserving the same stress and deformation values in the model (difference by 2%).

The use of recommendations on modelling of structures and loads applied to them which are available in industry-recognisable standards and manuals makes that the obtained results can be interpreted as valuable design calculations which can be used for making responsible decisions concerning real marine platform designs. A considerable advantage of the proposed method is the possibility of its implementation in an arbitrary computing environment, (the case reported in the article was analysed in Matlab® environment), depending on the software and hardware owned by the potential user. This property would be extremely valuable in possible parallel verification of calculations performed at different marine industry centres with different computing systems to increase the safety of designed or already existing structures.

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