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## DEVELOPMENT AND DESIGN VERIFICATION OF A FLOATING TIDAL POWER UNIT

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### ABSTRACT
This paper presents how the design verification process for a tidal current power plant was carried out, and points out the challenges needed to be addressed in the technical assessment phase of the concept.

The rules and regulations found suitable for the design process are presented.

The paper focus mainly on what analysis is required to validate the capacity of the structure. The structure is exposed to a harsh offshore environment. The relative importance of the different loads and load effects is shown and discussed. The need for hydroelastic coupled analysis is demonstrated.

The work shows that interactions between the main components are of large importance. There are interaction effects between moorings and structure as well as between turbine blades and structure. In fact the fully integrated system needed to be analysed as an integrated system, hence the need for coupled analysis.

Although the structure was designed to derive energy from tidal currents, wave induced forces showed as important as currents for the structural integrity of the unit.

The analysis demonstrated the versatility of the used FE analysis program AquaSim. This program is developed specifically to carry coupled analysis with hydroelastic interactions between structure and fluid. The FE program is in wide use within the Aquaculture industry. It is also in wide use in the offshore industry including moored systems and towed seismic equipment.

### INTRODUCTION
Due to the soaring prices of energy, there is at present a strong interest in new concepts for production of renewable energy. Several sea based concepts are investigated for wind, wave and current energy. At the 2006 OMAE, focus were drawn to offshore wind energy, where among them Nielsen et al (2006) showed floating offshore wind turbines, and Buck et al (2006) showed wind farms in conjunction with aquaculture farms.

This paper deals with tidal energy concepts. Large currents from tidal variations are observed at several locations around the world, as seen in Figure 1. This means that there is a strong potential for tidal energy world wide.

**Figure 1 Areas with large tidal currents**

A wide range of concepts have been proposed to extract the vast amount of energy present in tidal currents. Most are based on regular turbine systems, whereas some are more non-conventional like the one from Tidalsails (2007), seen in Figure 2.

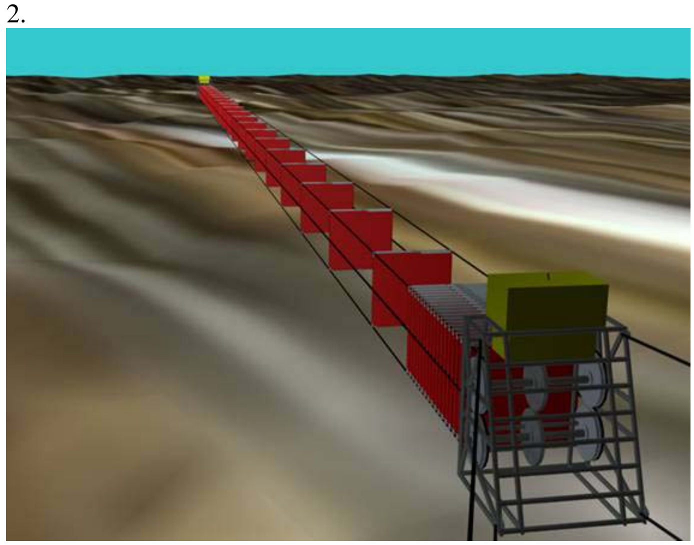

**Figure 2 A non-conventional concept for a tidal current power plant. (Tidalsails 2007)**

A more conventional concept has been introduced by Hammerfest strøm (Hammerfest 2007), see Figure 3.

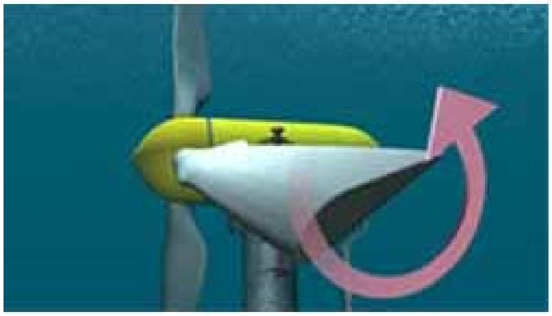

**Figure 3 A sea bottom attached concept (Hammerfest 2007)**

This paper presents the engineering challenges and the analysis tool used to carry out a design and design verification for another novel concept, the "Morild 1" as shown in Figure 4. This is a concept owned and developed by HTET (2007).

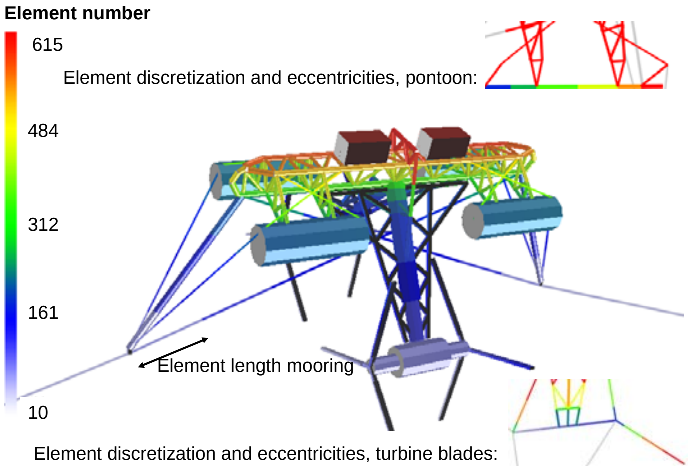


**Figure 4 Element discretization beam and bar element. 1 beam element per chord/brace in truss structure. Eccentricities are generally modelled by beams.**

The power plant is a moored structure with 4 turbines. Forces introduced to the turbine blades need to find their way through the structure and moorings down to the soil. Hydroelasticity, nonlinearities, interactions between turbine-blades and structure and interactions between structure and moorings are typical phenomena to be accounted for. This sparked the need for a coupled analysis where both the hydrodynamic loading and the structural response are calculated in the time domain for a given environmental condition (waves, current and wind).

Since this was such a novel structure, no rules or regulations had been developed explicitly for this type of structure. This meant that suitable design criteria were derived from the Offshore and Aquaculture industry. For environmental loads NS 3490-4 (2002) (wind) and DNV-OS-E301(2004) were used as a basis. For capacity checks, the NORSOK (1998) and NS 3472 (2001) were relevant for steel structure, supplemented by DNV Class Note 30.1(2004) and DNV RP 202 (2000) for buckling, DNV Class Note 30.7 (2003), DNV RP 203 (2001) and IIW(1997) for fatigue. For making a testing program, DNV-OS-C401 (2004) was resorted to. For the moorings, DNV-OS-E301(2004) was applicable both for loads and capacity. In addition, NS 9415 for the Aquaculture industry was resorted to. This is further outlined in DNV(2005).

From the very beginning of the project it was apparent that coupled analysis was needed to obtain the response of this system being exposed to nonlinear load effects and a hydroelastic response pattern. Note in particular the special mooring arrangement seen in Figure 4. Coupled analysis was pin pointed as a need for the offshore industry at OMAE 2005. The system consisted of 3 separate parts usually being assessed separately (Turbine, main structure and moorings). The challenge was to cover all design sensitive interactions between these components. The project chose AquaSim (Aquastructures 2006a-b, Berstad et al 2004) as the tool for the analysis. This is described in the next section. From this analysis tool, the relevant response parameters such as stress level and motions,, could be derived from time domain simulation. The analysis tool has been developed through many years of research and verification testing with the bases founded through PhD studies at NTNU. The program was developed because of a need for an analysis tool that was able to incorporate several individual floating objects integrated in one numerical model. To the project's knowledge, no other analysis tools were able to carry out this type of coupled analysis. Hence the project did not compare results to other tools.

Having established design criteria and an analysis program, a program for testing and quality assurance of production needed to be worked out. However, as applicable standards for this was found, the project's challenge was reduced to establish the scope of this activity.

Regardless of all the interaction effects, the turbine and turbine blades represented a large design issue. They needed to cope with a rather hostile design environment. The challenge was to keep the light enough and still strong enough for the given environment.

As seen, the project was faced with a large range of challenges, but this paper focuses mainly on the challenges regarding the analysis and design verification.

## THEORETICAL BASIS FOR AQUASIM

The AquaSim program is based on the finite element method. It utilizes beam and shell elements with rotational DOF's, as well as membrane elements and bar elements with translatory DOF's only. Geometric nonlinearities are accounted for in all element types, so the program handles large structural deformations. The program is based on time domain simulation, where it is iterated to equilibrium at each time instant. Both static and dynamic time domain simulations may be carried out. Features such as buoys, weights, hinges and springs are included in the program.

The basic idea of the FE analysis program is to establish equilibrium between external loads acting on the structure at a given time instant, and internal reaction forces.

$$\sum F = \mathbf{R}_{ext} + \mathbf{R}_{int} = 0 \tag{1}$$

where $R_{ext}$ is the total of the external static forces acting on the structure at a given time instant, and $R_{int}$ is the internal forces. The structure is discretized to a finite number of degrees of freedom (DOF's). Equation 1 is then discretized as

$$F^{idof} = \mathbf{R}_{ext}^{\ idof} + \mathbf{R}_{int}^{\ idof} = 0, \quad idof = 1, N_{dof} \tag{2}$$

where $N_{dof}$ is the discrete number of DOF's the structure has been discretized into. The current element program deals with strongly nonlinear behaviour both in loads and structural response. In order to establish equilibrium, the tangential stiffness method is used. External loads are incremented to find the state of equilibrium. Having established equilibrium in time step ***i-1***, the condition for displacement ***r***, step ***i***, is predicted as

$$\Delta\mathbf{R}^{i}(\mathbf{r}_{i-1}) = \mathbf{R}_{ext}{}^{i}(\mathbf{r}_{i-1}) + \mathbf{R}_{int}{}^{i-1}(\mathbf{r}_{i-1}) = \mathbf{K}_{t}^{i-1}\Delta\mathbf{r} \quad (3)$$

where $K_t^{i-1}$ is the tangential stiffness matrix at configuration ***i-1***. The external load is calculated based on the configuration of the structure at ***i-1***. This gives a prediction for a new set of displacements ***(j=1)***. Based on Equation 3, a prediction for the total displacement $r_{(j=1)}$, is found as

$$\bar{\mathbf{r}}_{j=1} = \mathbf{r}_{i-1} + \Delta\mathbf{r} \quad (4)$$

Based on this estimate for new displacements, both external and internal forces are derived based on the new structural geometry, and the residual force is put into the equation of equilibrium as follows

$$\Delta\mathbf{R}(\bar{\mathbf{r}}_{j}) = \mathbf{R}_{ext}{}^{i}(\bar{\mathbf{r}}_{j}) + \mathbf{R}_{int}{}^{i}(\bar{\mathbf{r}}_{j}) = \mathbf{K}_{t}{}^{i}\Delta\mathbf{r} \quad (5)$$

Note that both the external and internal forces will vary for each iteration due to the strong hydroelastic nature of the fluid structure interaction. Equation 5 is solved for the displacement $\Delta r$. Incrementing ***j*** with one, the total displacement is now updated as

$$\bar{\mathbf{r}}_{j} = \bar{\mathbf{r}}_{j-1} + \Delta\mathbf{r} \quad (6)$$

Now if Δr, found from Equation 6, is larger than the tolerated error in the displacements, Equation 4 is updated ***(j = j+1)*** and Equation 5 is solved based on the new prediction for displacements, this is repeated until $\Delta r$ is smaller than a tolerated error, then

$$\mathbf{r}_{i} = \bar{\mathbf{r}}_{j} \quad (7)$$

***i*** is increased with one, and Equation 4 is carried out for the new load increment.

At the default configuration, the program works like this: Static analysis is used to establish static equilibrium including buoyancy. Secondly, current loads are applied, and wind and wave loads added. (Still static analysis.) Then dynamic analysis commence. Waves are introduced with the first wave used to build up the wave amplitude. Both regular waves and irregular waves may be simulated. Waves are assumed to be sufficiently described by linear wave theory. Inertia and damping are accounted for in the wave analysis, meaning that mass and damping are accounted for in the equations of equilibrium. The Newmark-Beta scheme is applied for the dynamic time domain simulation (e.g. Langen and Sigbjørnson 1979). Note that the above equations imply using the Euler angles for rotations. This is just a simplification for easy typing. For rotational DOF´s AquaSim uses a tensor formulation for the rotations as outlined in e.g. Eggen (2000).

Wave loads may be derived using the Morison formulae (Morison et al 1950) or using diffraction theory.

The diffraction theory used in AquaSim is a form of "strip theory" (e.g. Salvesen et al 1970), but in this case hull forces are derived by direct pressure integration over the mean hull surface. Diffraction loads may be applied to beams or bars. In this case linearized values for diffraction, added mass and damping are derived at the mean wetted position. Added mass and damping are linearized at the peak period in the wave spectrum . The Froude Kriloff part of the hydrodynamic pressure is applied at the actual location of the component. Wave interaction between separate components is not accounted for.

When the Morison formula is used, the cross flow principle is applied for beams and bars (see e.g. Faltinsen 1990). This load term is quadratic with respect to the relative velocity between the undisturbed fluid and the structure, both the mass of the structure as well as added mass in the cross sectional plane is accounted for. Due to the large deflections occurring, the added mass is nonlinear.

The above presented algorithm represents a practical approach to simulate this type of integrated structures, given the size of the structures and current computer capabilities.

## PROGRAM VERIFICATION

The analysis program, AquaSim, has undertaken the following verification scheme: Analyses have been carried out on a wide range of computational cases where results have been compared to handbook formula or other programs, see Aquastructures (2006a). Model tank testing have been carried out and compared to analyses, see Berstad et al (2004). The program has been compared to accidents where the capsize origins were known (Aquastructures 2003 and 2005). In addition, experience has been obtained during several years where the program has been the most widely used for calculation of the structural integrity of fish farm systems in Norway. These systems in general consist of moorings, stiff structure and nets responding to wave and current in a strongly hydroelastic manner.

## CASE STUDY 1

This paper presents two case studies. Case study 1 is an FE model of the fully integrated system, established and analysed by AquaSim. The model is seen in Figure 4. As seen from Figure 4, the concept is moored with two moorings.
Component wise, the parts in the system are named as shown in Figure 5.

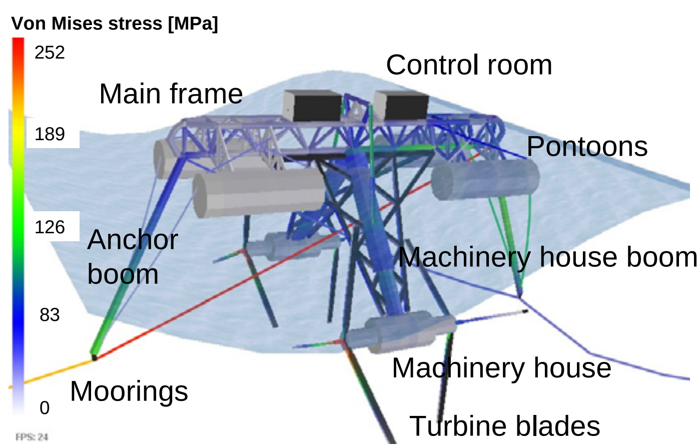


**Figure 5 Components in the system**

The load bearing system shown in Figure 5 works as follows: The current velocity leads to the torque bending moments in the turbine blades which is collected as electricity in the generators located in the Machinery house. The thrust force and other forces and moments introduced in the turbine blades needs to find their way through the main structure where the forces are beard by the moorings and buoyancy in the pontoons. In addition to forces introduced from the fluid to the turbine blades, forces are also introduced from the fluid directly to the structure. The most important of such forces are wave forces introduced to the pontoons.

**Current loads vs. wave induced loads**

Figure 6 shows Von Mises stress (effective stress) in the structure, turbine blades and moorings when only current is applied. Figure 7 and Figure 8 show the stress level at two separate time instances in waves..

As seen in Figure 6 through Figure 8, the stress level at joints in the main structure is tree-doubled when waves are introduced and the forces in the moorings are doubled. Figure 6 through Figure 8 show that although the facility is designed for obtaining energy from tidal currents, waves introduce so much extra forces in the structure that it becomes the main design load. This means that optimum design for tidal power plants will depend as strongly on the design waves as current velocity.

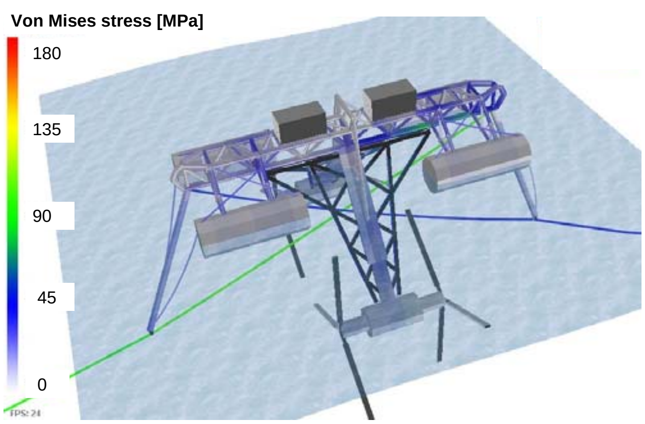


**Figure 6 Stress level as the structure is exposed to a current velocity of 2 mps**

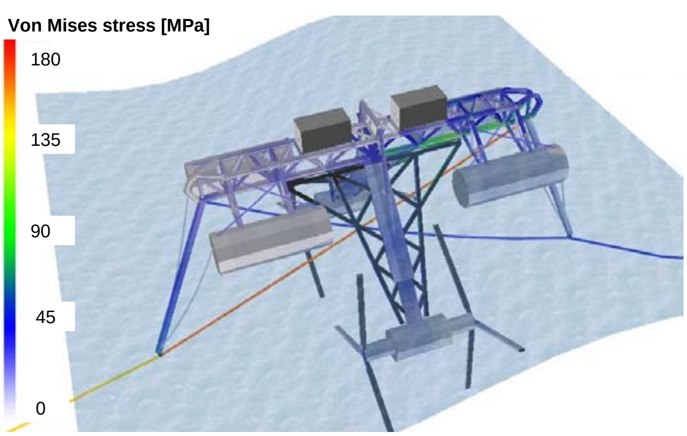


**Figure 7 Stress level as the structure is exposed to a current velocity of 2 mps and waves of 4 meters. One particular time instant**

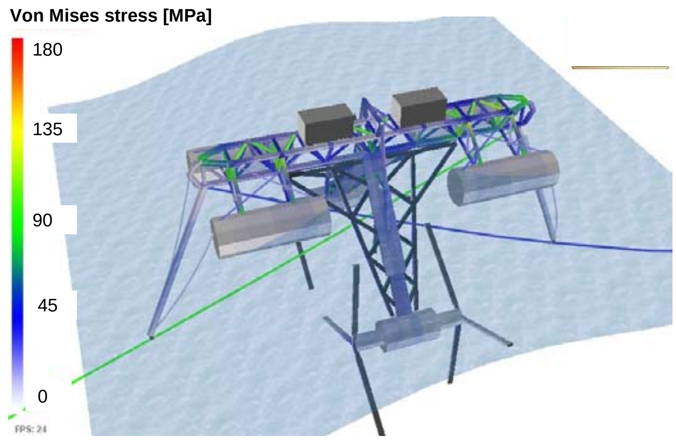


**Figure 8 Stress level as the structure is exposed to a current velocity of 2 mps and waves of 4 meters. Second time instant**

**Load formulation on the turbine blades**

Consider the turbine blades. Define a coordinate system where the *z*- axis is vertical and the *x*- axis runs perpendicular to the plane where the turbine blades are located, as shown in Figure 9. Assume a uniform current velocity. In each separate turbine blade there will be introduced a lift force leading to a torque moment acting as a torsional moment in the spindle. There will also be a drag force introducing a thrust force acting as an axial force in the spindle.

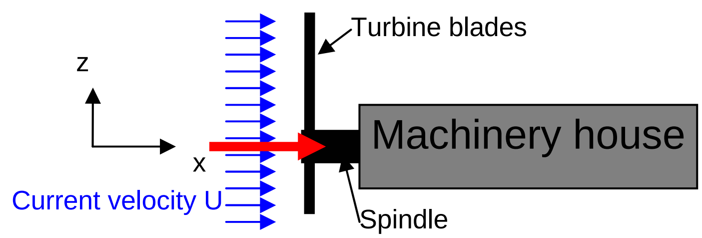


**Figure 9 Nominal current velocity to the turbine blades**

This condition corresponds to an ideal condition. Due to symmetry of forces into the spindle, there will be no bending moments in the spindle (apart from torque). This does however represent a strong simplification as shown in Figure 10.

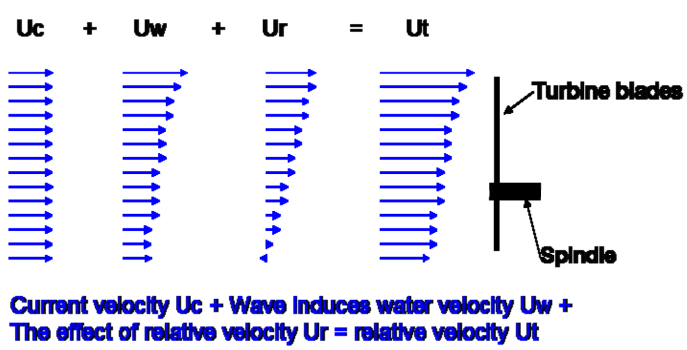


**Figure 10 Relative velocity experienced for the turbine blades.**

Figure 10 shows how the turbine blades are not only subjected to a steady varying uniform current load, but also to wave induced fluid motions and fluid motion caused by the wave induced motions of the structure. This effect obviously works both ways so that the calculated motions of the structure will be different if the effect is accounted for or not. In addition to the loads shown in Figure 10, the project also considered turbulence in the current (DNV 2005b). This load component showed to introduce the largest uncertainty with respect to fatigue.

In the first phase of the project, CFD analysis was carried out on the turbine blades (Marintek 2005). They found the turbine forces as a resulting thrust force component in the spindle for the structural engineers to use. This force acts in the x- direction as defined in Figure 9. It soon became apparent that a refined approach should be applied. This was done by applying the thrust force as a drag force over the turbine blades. These two approaches are compared in this paper.

The system is as seen in Figure 5..The structure is subjected to the environmental data given in Table 1. The case study compares the two alternatives for modeling the thrust force as outlined above. These are denoted as:

1. Thrust as point load: This means the structure is modeled and loads are applied in terms of drag loads on the different components and hydrodynamic loads on the pontoon. The loads on the turbine blades from the CFD analysis are modeled as a constant node load into the spindle as shown in Figure 9 with red arrow.
2. Thrust as drag: In this case the loads on the structure apart from the turbine blades are modeled in the same way, but instead of using a node load for the thrust force on the blades, a drag area is applied to the turbine blades with the cross flow principle, so that the loads at each cross section is in proportion with the relative motion between fluid and structure squared.

The two load formulations give the same results when a uniform current of 2.0 meters per second (m/s) is applied. For both cases the stiffness, mass and added mass are the same, and the turbine blades do not rotate during the simulations but are fixed in the position as seen in e.g. Figure 7.

The combination of waves and current given in Table 1, introduces forces and motions in the structure. Figure 7 shows a time instant in the AquaSim simulations. In the present analysis regular waves are applied.

**Table 1 Key data for case study**

| Parameter | Value |
|---|---|
| Current velocity [m/s] | 2 |
| Wave height [m] | 4 |
| Wave period [s] | 5.66 |
| Nominal thrust force as point load [kN] | 30 |

Figure 11 shows axial force in the spindle (see Figure 9). As seen from this figure, the forces vary strongly when the thrust force is modeled by drag forces over the turbine blades, while they are steadier when modeled as a point load. This shows that the interaction between turbine blades and the structure is of large importance in the design.

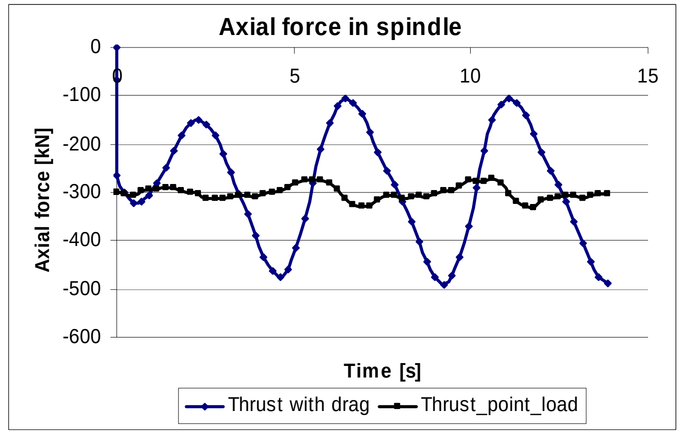


**Figure 11 Axial force in spindle (- means compression)**

Figure 12 shows the longitudinal motion of the spindle. As seen from this figure, the motion amplitudes are much smaller for the case where the thrust force is modeled by drag over the turbine blades. This confirms the importance of interaction between current and waves.

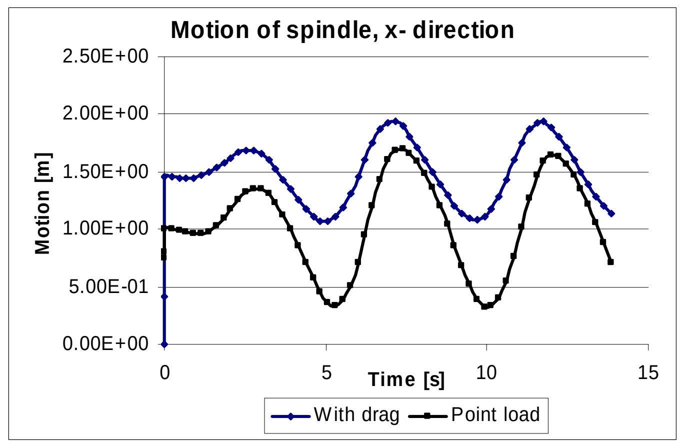


**Figure 12 motion of spindle in x- direction**

Figure 13 shows the vertical motion of the spindle as a function of time. As seen from this figure, the motion amplitude is about the same, but the results with drag are shifted slightly upwards. In both cases, the turbine blades have been modeled with a drag area of 0.3 m to account for motions in the y-z plane. However, modeling the thrust by drag introduces a pitch since the back and forward turbine blades are modeled in the opposite direction, as seen in e.g. Figure 7. This shows that varying the angle of the turbine blades may be of importance. This is further considered in case study 2.

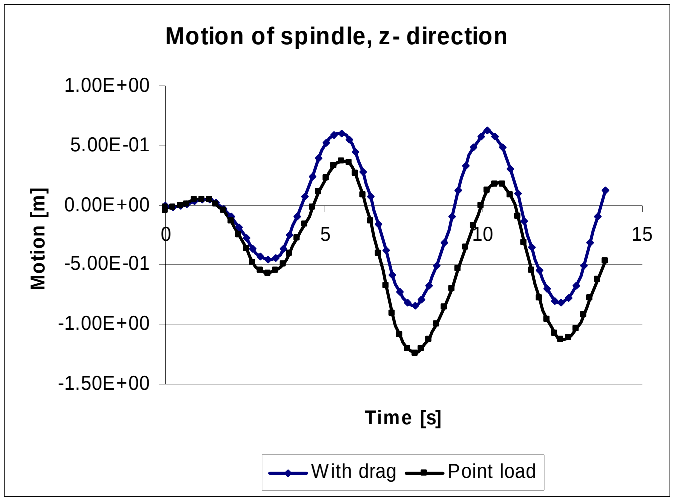


**Figure 13 Motion of spindle in z- direction (Vertically)**

Another parameter which is of large importance for the design of turbine blades and spindle, is the bending moments. Figure 14 shows the bending moments in the spindle about the y- axis as a function of time for the two alternative algorithms. As expected, the load formulation is of large importance. To investigate this effect the drag formulation should be applied.

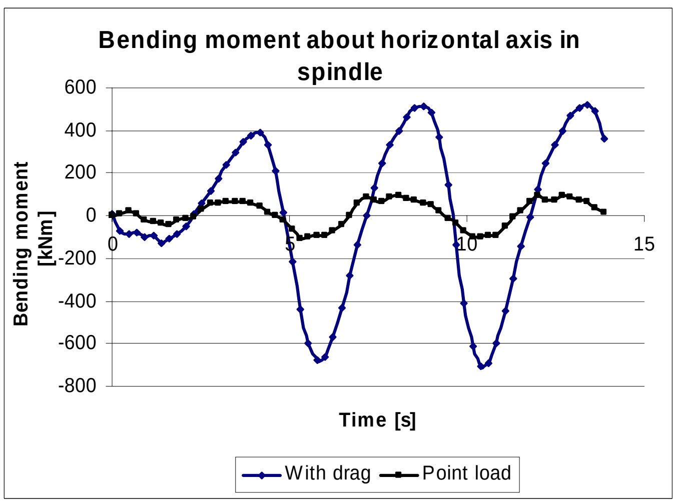


**Figure 14 Bending moment about y-axis**

## CASE STUDY 2

Case study 2 is carried out on a bottom fixed turbine. This case is shown in Figure 15. The x- axis runs along the spindle and the z- axis points upwards.

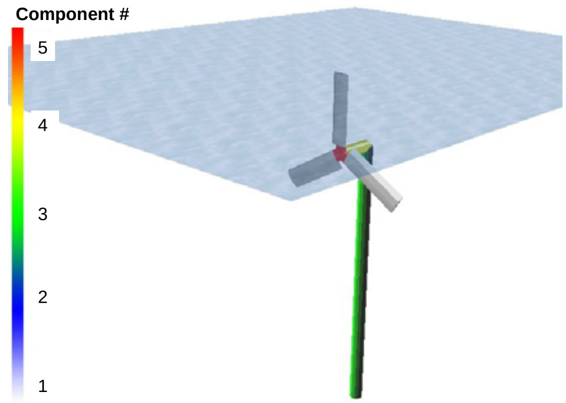


**Figure 15 Analysis case with bottom fixed turbine with rotating turbine blades.**

In Figure 15 the turbine blades are seen in grey with inner parts red, the spindle is seen in yellow. The column fixing it to the bottom is seen in green. Figure 16 shows the rotation after some time of simulation in waves. In the figure, the blades have rotated more than 2 full rounds on the spindle.

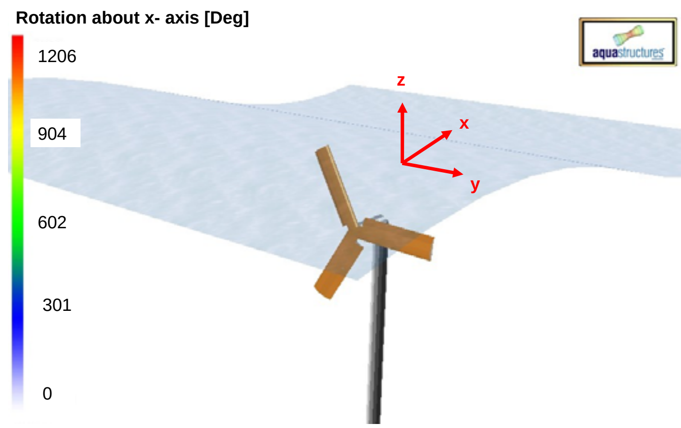


**Figure 16 Rotation shown by colors. In this case the blades have rotated about 900 degrees, meaning more than 2 full turns.**

The lift effect on the turbine blades has been modeled in a simplified manner by using elements transverse to the radial direction skewed in an angle, which is the pitch angle. The key data for the case is given in Table 2. In this table rpm means rotations per minute.

**Table 2 Key date for case study**

| Case data | |
|---|---|
| # of turbine blades | 3 |
| Blade radius [m] | 10 |
| Pitch angle [deg] | 67.4 |
| Current velocity [m/s] | 4 |
| Wave height [m] | 3 |
| Wave period | 8 |
| Wave direction is along the x-axis | |
| Blade height [m] | 1 |
| Generator resistance [kN/rps*] | 100 |

| | |
|---|---|
| *rps = radiants per second | |
| Average rotation with current 4 m/s [rpm] | 8.107 |
| Average rotation with current 4 m/s [rpm] | 8.107 |

The generator has been modeled by a damper linear to the velocity. The mass and added mass of the turbine blades have been neglected in the analysis and the weight of the blades are neutral in water. The wave field is assumed to ride on the current field.

Figure 17 shows the rotation velocity (number of rotations per minute, rpm) as a function of time. As seen from the figure, the results show that with no waves, the rotational velocity is constant, as suspected, while when waves of 3 meters are introduced, the rotational velocity starts to vary. As seen from the figure, it varies rather symmetrical around the mean value. The variation period is equal to the wave period of encounter

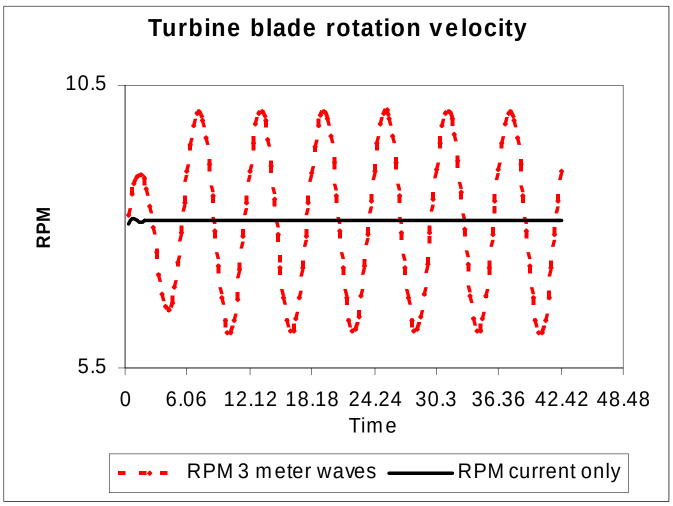


**Figure 17 RPM with 4 mps current.**

Figure 18 shows thrust forces and bending moment in the spindle for the present case study with waves of 3 meters. Figure 19 shows the same, but in this case the turbine blades are restrained from rotating. As seen from the figures, both cases give approximately the same variation in the thrust forces while the latter case have a much more periodic behavior for the bending moments. This is because of the turbine blades in the latter case are fixed in the initial position.

Inspecting Figure 18 and Figure 19, it is seen that on a rough scale much of the spindle moments are picked up in the latter model. However as seen from the figures, including rotations increases the moments. Approximately 50 % increase in moment about the horizontal axis, and approximately 200% for the vertical axis. Hence this effect should be considered when this part of the structure is designed. This was not done for the case study 1 in this paper. As seen from the figures, however, much of the damping introduced to the main structure is picked up with by assuming that the forces to the turbine blades act as drag forces on the blades rather than using a point load into the spindle. Considering structural details far away from the turbine blades, this is probably sufficient whereas for the spindle a refined approach should be considered.

It should be noted that accounting for rotation of turbine blades significantly increases the computing time and convergence difficulties. This may be a subject for further studies.

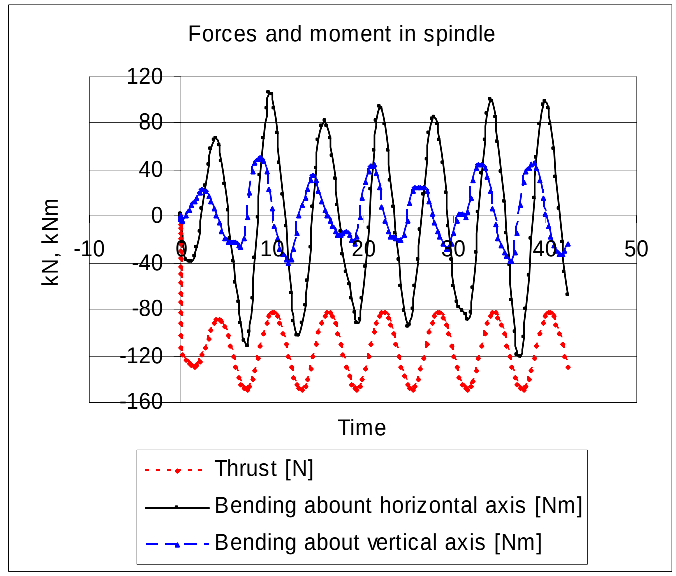


**Figure 18 Results with rotating turbine**

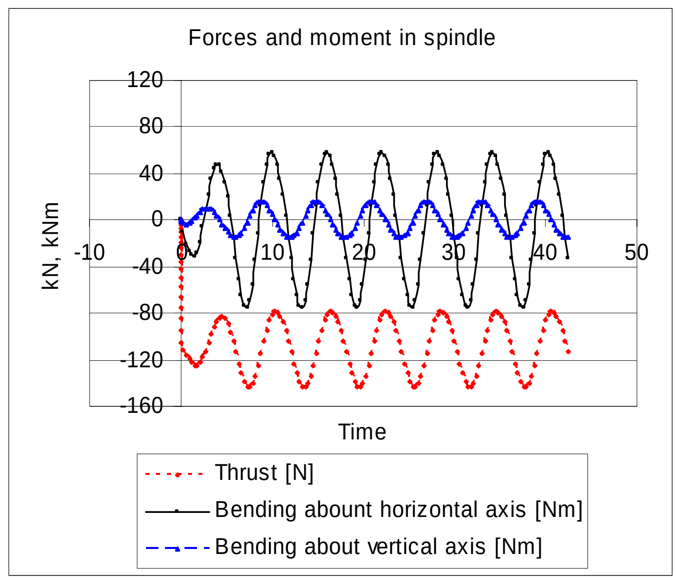


**Figure 19 Results with fixed turbine**

## CONCLUSIONS

As part of a design verification process, analysis of a facility for producing renewable energy from sea currents has been carried out.

The results show the importance of coupled analyses to find the structural response so that interaction effects between the main parts of the structure are accounted for in a proper manner.

The calculations showed that although current is what the energy is taken from, waves are even more important for the design forces in the structure.

For calculation of the structural motions and internal forces from sea loads, the damping introduced by the turbine blades should be accounted for. In general, accounting for the interaction will reduce motions but increase forces.

Also, the interaction to the mooring system should be accounted for when structures are as the ones investigated in the present case.

Taking account for the rotation of turbine blades will also influence results, but does increase computing time. For the spindle and areas close to the turbine blades, it should however be accounted for. This may be a subject for further studies.

The AquaSim analysis program proved very useful for this case study. In particular its capability to account for fluid structure interaction proved suitable for this case study with its hydroelastic response. This versatility was expected due to previous use for a wide range of offshore structures with a hydroelastic response pattern.

Based on the work described above, the project entered a review phase to calculate the cost benefit of the concept as well as diversity with respect to environmental conditions.

## ACKNOWLEDGMENTS

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