MECHANICAL DESIGN OF A BIRADIAL TURBINE

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Abstract

The oceans are an important natural resource which, when properly exploited, can make a relevant contribution to the electricity supply in countries with large coastal areas. This topic is the basis for this thesis, aiming at the execution of the mechanical design of a turbine for wave energy conversion – the biradial turbine.

The projected turbine has the particularity of being axially compact with a biradial rotor input and output flow. Peak efficiency may exceed 80%. This work starts from a preconceived configuration, elaborating a process of concept generation and selection, which is followed by a stage of material selection, taking into account the operating environment and functional requirements. Next, the dimensioning process begins. At this stage the Eurocode 3 must be taken into account when designing the rotor blade elements [1]. This phase is followed by the static and fatigue design of the turbine shaft, using a theoretical basis of mechanical design and some design approximations. These are then validated through numerical analysis. Finally, there is also the process of designing the chassis and the adjacent components, which is done computationally due to the size and complexity of the structure.

The mechanical design of such a device is dependent on a set of variables, such as the loads used during the design phase. In fact, the considered loads are decisive, as they affect the course of the project. A conservative approach is a key to ensuring structural integrity.

Keywords: Wave energy, Oscillating water column, Biradial turbine, Mechanical design, Conceptual development, Dimensioning.

1. Introduction

The world's energy consumption has been increasing over the years. It is now double what it was forty years ago. According to data from the International Energy Agency, in 2013, about 85% of the world's energy consumption came from the use of fossil fuels [2]. The great problem is that the natural resources producing these fuels do not regenerate at the rate they are consumed. This means that their availability is compromised for future generations. Furthermore, the use of non-renewable energy sources causes gas emissions that produce a greenhouse effect.

The large potential of wave energy has been recognized over the years. However, it was only after the 1973 oil crisis, that countries with the required geographical conditions and energy import needs chose wave energy in governmental programs and research and development institutions [3]. The topic of this thesis stems from work performed by Instituto Superior Técnico (IST) Wave Energy Group and mainly deals with the mechanical design of a biradial turbine to equip a floating system, based on the oscillating water column principle. The aim of this project is to respond to the following needs: planning and general design of a wave energy conversion device; selection of building materials, taking into account the marine environment where the turbine will operate; design of the subcomponents: rotor, stator and turbine shaft; design of the structure that allows support and integration of the turbine in the floating system (called chassis and adjacent components); drafting of the technical drawings.

2. The Biradial Turbine

The biradial turbine is a self-rectifying impulse turbine that is very compact axially. Air input and output in the rotor is performed radially, so the flow is centripetal and centrifugal (hence the term biradial turbine).

The rotor is symmetrical in relation to the perpendicular plane to its rotational axis (Figure 1 a)), being surrounded by two rows of guide vanes (stator) disposed symmetrically in the inlet and outlet ducts (Figure 1 b)).

Computational simulations (CFD) show that the average efficiency over time, in a random wave environment, can exceed 70%, providing that rotational speed is appropriately controlled [4].

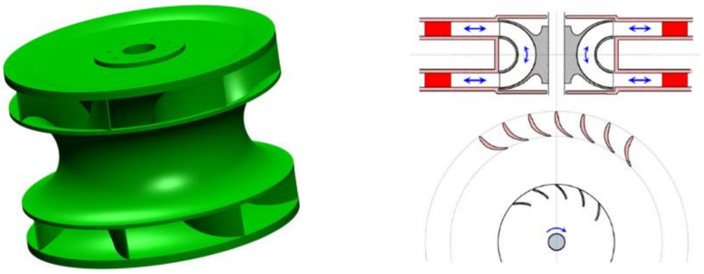

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Figure 1 – a) Conceptual model of the biradial turbine rotor; b) Flow directions in the rotor [5]

3. Project Specifications

The project is a process composed of several iterative phases in the course of which the designer makes decisions based on pre-defined specifications.

The intended features of the turbine, given by IST Wave Energy Group, are shown in Table 1.

Table 1 – The main intended features of the turbine

Rated electric power [kW]	25 to 30
Nominal rotating speed [rpm]	1500
Runaway rotating speed [rpm]	6000
Rotor diameter [mm]	490
Rotor overall height [mm]	387,25
Number of rotor blades	11
Thickness of rotor blades [mm]	6 to 8
Diameter of the outer circular row of the stator [mm]	2280,60
Diameter of the inner circular row of the stator [mm]	1930,80
Number of guide vanes in each stator area	64 (outer row); 64 (inner row)
Outer diameter of the integration structure in the floating system [mm]	2800

The project's main requirements are as follows:

- The turbine must be able to operate on a vertical axis (installed in a floating system) or on a horizontal axis, with some adjustments to the basic configuration;
- The turbine rotor and its attached equipment, including the electrical generator and shaft, should be able to be removed quite easily from the interior of the structure housing these components, without shaft alignment problems during its replacement.

4. Conceptual Development

Initial concept

The IST Wave Energy Group initially suggested the idea to conduct the turbine study (Figure 2). During the project, the initial configuration was gaining new contours, greater complexity and robustness and was subject to several significant changes.

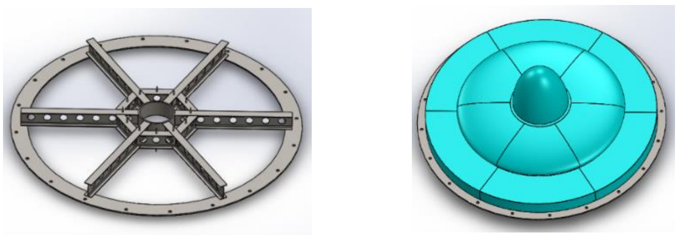

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Figure 2 – a) Chassis of the initial concept; b) Configuration of the device in the initial concept

Final concept

This concept arises from the need to ensure the structural integrity of the chassis and adjacent components that operate in very severe load conditions. The previous concept has some limitations (checked by computational analysis) as it is not robust enough to support the loads involved.

It then proceeds to identify some relevant components related to this final concept (Table 2).

Table 2 – Identification of components and material selected for turbine development

Reference	Designation	Assigned material
1	Device integration disk 1	AISI 316 L
2	Device integration disk 2	AISI 316 L
3	Radial central beam	AISI 316 L
4	Inner central reinforcement plate 2	AISI 316 L
5	Inner central reinforcement plate 1	AISI 316 L
6	Outer central reinforcement plate 2	AISI 316 L
7	Outer central reinforcement plate 1	AISI 316 L
8	Upper stator blades connecting disk	AISI 316 L
9	Upper stator outer blade row	AISI 316 L
10	Upper stator outer structural blade row	AISI 316 L
11	Upper stator inner blade row	AISI 316 L
12	Fitting disk in upper stator blades	AISI 316 L
13	Upper stator radial reinforcement beam	AISI 316 L
14	Upper stator outer reinforcement plate 1	AISI 316 L
15	Upper stator outer reinforcement plate 2	AISI 316 L
16	Upper stator inner reinforcement plate 1	AISI 316 L
17	Upper stator inner reinforcement plate 2	AISI 316 L
18	Disk supporting bushing	-
19	Generator supporting flange	AISI 316 L
20	Generator cover threaded flange	AISI 316 L
21	Reinforcement columns supporting the generator	AISI 316 L
22	Generator supporting flange	AISI 316 L
23	Generator	-
24	Linear guidance system	-
25	Pneumatic actuator	-
26	Shaft coupling	-
27	Safety valve	AISI 316 L
28	Support cone	AISI 316 L
29	Bottom base	AISI 316 L
30	Conical shaft support	AISI 316 L
31	Generator cover	Fiberglass reinforced polyester (FRP)

The chassis is the central structure of the device, whose primary function is to support all turbine components and allow their integration into a floating device or a fixed oscillating water column system, through the device's integration disks. There are eight UPE beams that are radially supported and bolted on these disks (Figure 3). There are also two rows of reinforcement plates and the inner row is also a central cylinder that protects the rotor. This beam section was chosen as it can alleviate the weight of the structure and has constructive advantages.

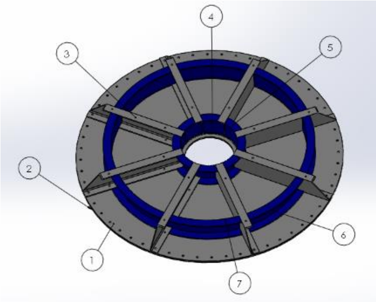


Figure 3 – Chassis

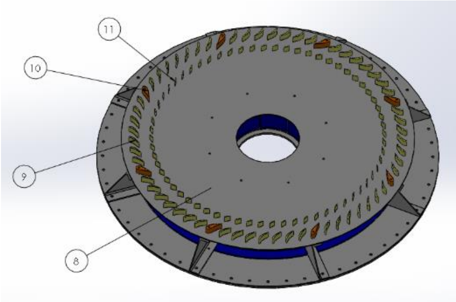
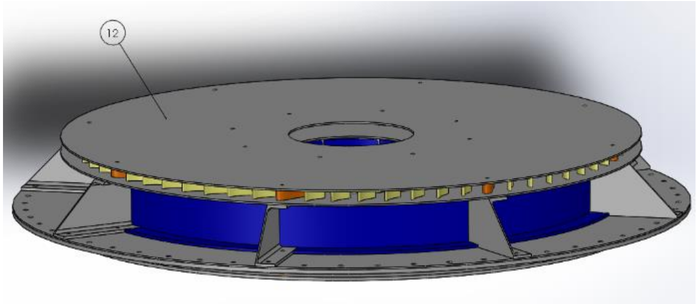


Figure 4 – Upper stator blades and base disk of the blades

Eight blades were designed with a mainly structural function and are crossed by a connection element (Figure 7). The stator blades should be welded to the base plate (connecting disk, reference 8 in Figure 4) and, subsequently, fitted into another disk (reference 12 in Figure 5). These two disks form the stator air flow channel.



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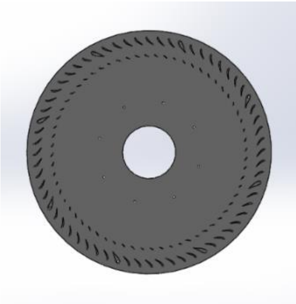

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Figure 5 – Fitting disk in the upper stator blades: a) Installation on the device; b) Detailed representation

Reinforcement beams and plate rows are installed to strengthen the structure of the stator channel disks (Figure 6). Stator guide vanes are also installed at the lower part of the turbine for the same purpose. The structure of the lower stator channel must also be reinforced, following the same procedure.

Support bushings should be installed for blade fitting disks in both turbine parts (upper and lower), crossed by a threaded element to reinforce the structure.

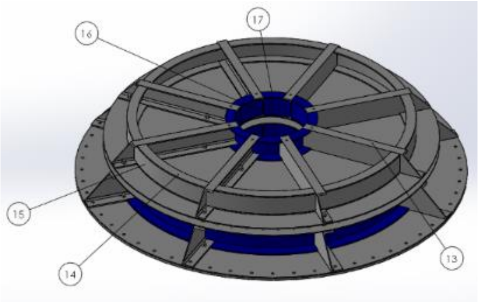


Figure 6 – Reinforcement structure of the upper stator channel

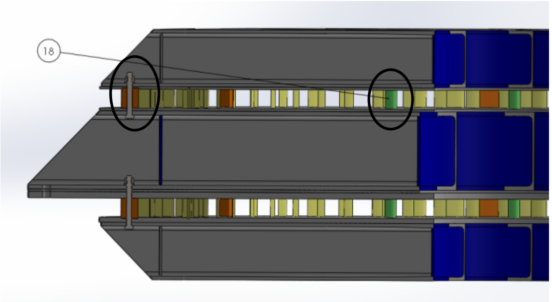
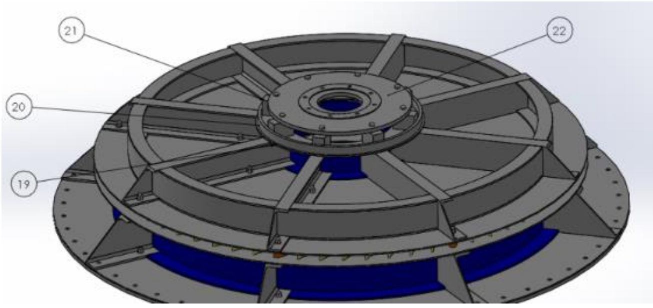


Figure 7 – Detail of the structural blades and the bushing of the stator

The generator is installed with a plate supported on reinforcement columns (Figure 8). The whole assembly generator + shaft + rotor may be removed and replaced using the bolted connection between the components.



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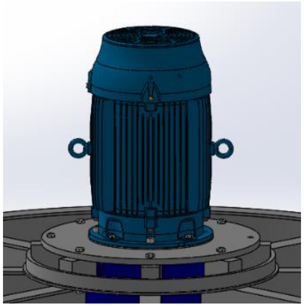

b)

Figure 8 – Generator support structure: a) Identification of components; b) Generator in position

Figure 9 shows the unit to be removed from the interior of the chassis for eventual maintenance. This whole unit is based on a support structure for the cone attached to the shaft. Such structure should be fixed to the turbine base (Figure 10).

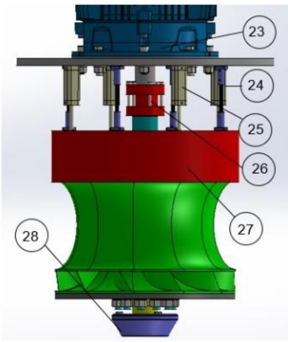

Figure 9 – Configuration of the device for removal and replacement on a floating system

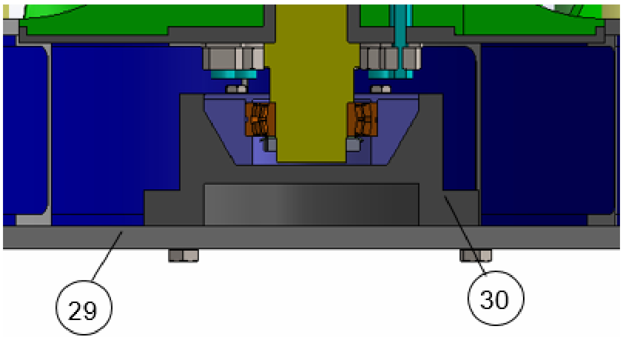

Figure 10 – Details of the shaft's conical support system

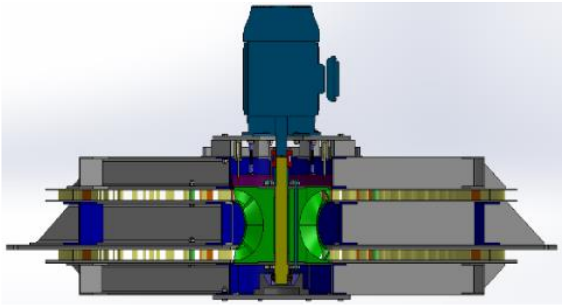

Figure 11 – Overall view of the shaft's conical support

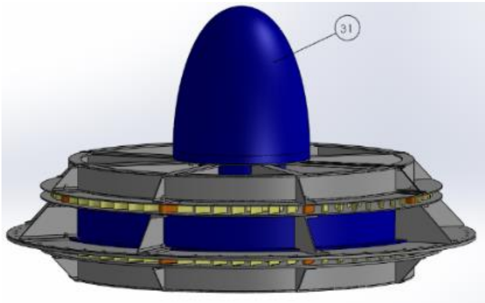

Figure 12 – Overall view of the turbine

5. Dimensioning Process

This design stage is very important as it deals with the dimensioning of the turbine's main components and connection elements. Extreme loads to which the device may be subjected must be taken into account when calculating its dimensions. The worst situation, in terms of loads on the device, is when the turbine is installed in a buoy. The following procedures and results presented are therefore for vertical axis configuration. Table 3 shows the load values used in the dimensioning process.

Table 3 – Extreme load cases (load values used in the dimensioning process)

Description	Value	Unit
Maximum pressure in the chamber ($P_{máx}$)	2,50	bar
Minimum pressure in the chamber ($P_{mín}$)	-1	bar
Maximum vertical acceleration (g_z)	13,20	m/s^2
Maximum horizontal acceleration (g_h)	18,50	m/s^2
Maximum rotational speed (pitch and roll) (ω_{pr})	0,38	rad/s
Nominal rotational speed (N_{nom})	1500	rpm
Runaway rotational speed (N_{sobr})	6000	rpm
Wave period (T_{ondas})	9,30-20	s

<u>Dimensioning of the rotor blades elements</u>

The connection elements of the rotor blades cross the whole length of the blades, ensuring their correct positioning during turbine operation (Figure 13). The design method involves identification of loads operating in the blade mass center (C_M), in other words, loading resulting from the motion of the buoy (Figure 14 and Table 4). Aerodynamic forces caused by the air flow on the blade are not taken into account as kinematic loading considered makes the process adequate.

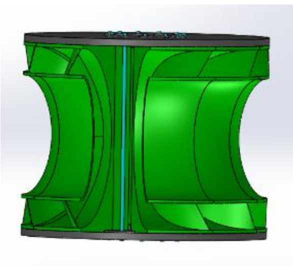

Figure 13 – Elements representation

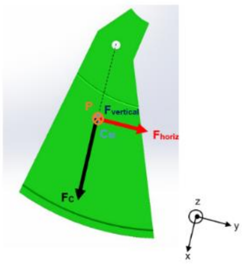

Figure 14 – Loads acting on the mass center of the blade

Table 4 – Identification of the loads used in elements design process

P	Force due to the mass of the blade
F_c	Centrifugal force induced by the rotor rotational speed
F_{horiz}	Force due to horizontal acceleration
$F_{vertical}$	Force due to vertical acceleration
C_M	Mass center of the blade

In the design process, the combined use of Eurocode 3 and DNVGL-OS-C101 standard [6] implies that the general design criteria was based on the limit state concept. This means that the dimensioning consists of checking the safety of the structure in terms of limit states in comparison with the states in which the structure is driven by the combination of the loads to which is subjected. Elements are designed to guarantee safety in accordance with the ultimate limit state criteria. The final results for the diameter of the elements, taking into account the pressure inside the chamber (not shown in Figure 14), are presented in Table 5. The intended final diameter is 20 mm.

Table 5 – Final results for the element diameters

Dimensioning type	**Final result** [mm]
Shear	18,99
Bearing	8,56
Tension	0,76

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Shaft dimensioning

The shaft ensures torque transmission, which is a result of converting kinetic energy from the flow into mechanical energy, in the generator shaft. The project starts with static yield design method, which requires knowledge of the component's critical section (section subjected to the greatest loads). That loads may be quantified by knowing the forces applied in each section of the shaft (figures 15 and 16).

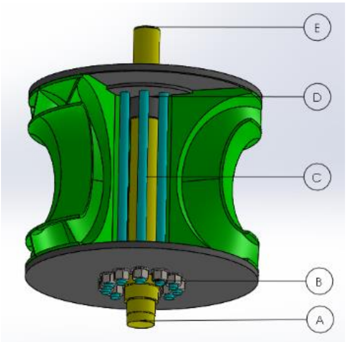


Figure 15 – Main sections of the shaft

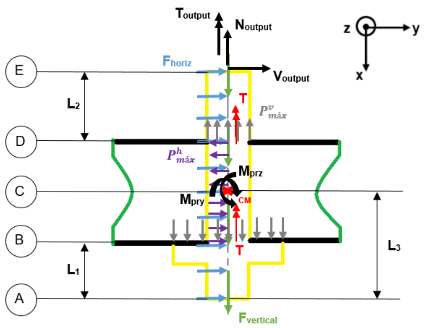


Figure 16 – Loads placed on the shaft and identification of the shaft's main sections

Sections of the shaft are subjected to loads resulting from horizontal and vertical acceleration, respectively, F_{horiz} and $F_{vertical}$. Pitch and roll movements (rotation of the device around the y and z axis) cause bending load on the shaft. So, it is crucial to quantify the torque on the shaft $(M_{pry}\ e\ M_{prz})$, and considered to be applied in the mass center of the shaft, corresponding to the section C. The minimum diameter for the shaft's critical section is strongly affected by the value of such loads. The impact of the pressure in the chamber is introduced as an approximated way, but safe and conservative. Simultaneous action of two pressure loads is considered in the contact area between the shaft and the rotor (most frequently required by these loads), a horizontal load $P^{h}_{máx}$ and a vertical load $P^{v}_{máx}$. Keys must be installed in the shaft for the rotor rotating motion be transmitted to the turbine shaft, i.e., for both to rotate as a single component. The entire length of the shaft, between sections B and E, is considered to be under torsional load, and its value is regarded as the maximum mechanical torque supported by the generator. After quantifying the loads on the shaft, it can be concluded that the critical section of the shaft is located in section C. Shaft material is ductile, which allows von Mises criteria to be used to calculate design load $\overline{\sigma_{VM}} \equiv \bar{\sigma}$. This criteria is used to determine the shaft's diameter for the critical section, which is approximately 42 mm.

A component subjected to cyclical loads requires fatigue-dimensioning to avoid this failure mode. The shaft turbine is designed using the Goodman criteria as failure criteria. The calculations performed provide a value of 60 mm for the minimum diameter of the critical section.

Chassis and adjacent components dimensioning

The complexity of the chassis and its adjacent components implies that a computational analysis must be used for dimensioning. The assembly shown in Figure 17 is simplified to make these analyses feasible. So, two areas of interest should be studied - the upper and lower stator. The lower stator (Figure 18) is the area most affected by the maximum pressure in the chamber (2,5 bar). The upper stator is the area most affected by the minimum pressure in the chamber (-1 bar). Each one of these areas is analyzed separately, using the same approach.

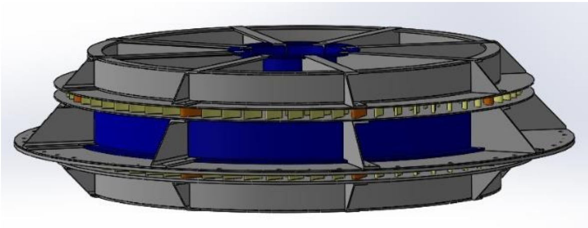


Figure 17 – Chassis and adjacent components

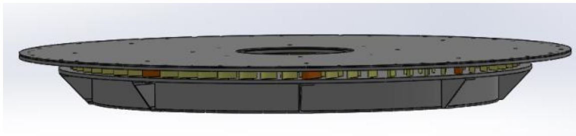


Figure 18 – Lower stator

The structure's geometric features make the 3D mesh the best choice for its modelling. Therefore, four-node 30 mm tetrahedral elements are used. The dimension used is the smallest processed by the computer. When designing the stator blade rows, the reliability of the results is always low due to the excessive dimension of the elements used.

According to Figure 19, the maximum stress value registered at the lower stator is 236,56 MPa. Although this is very close to the material yield stress value, it is justified by the fact that there are several geometric factors that lead to a localized increase in stress (Figure 21). In the central cylindrical zone, the radius of concordance/rounded corners between the elements must be taken into account to reduce stress. In the stator blades area, it is important to note that in this analysis, its connection to the disks is not progressive, that is, it does not take into account any fillets or other local geometrical changes to reduce stress at this location. These factors are important from the point of view of the production process and their implementation is fundamental at this stage to avoid any service failures. By analyzing Figure 19, the general structure is affected by stress below the allowable stress limit. Maximum stress values are only registered at locations shown in Figure 21.

By implementing the aforesaid solutions to reduce stress, the structure can be considered suitably dimensioned, if a 20-mm thick fitting disk in the lower stator blades is used. The maximum radial deformation is 0,348 mm and the maximum axial deformation is around 2,218 mm (Figure 20). Maximum values are acceptable, since they are lower than 1/3 of the components thickness.

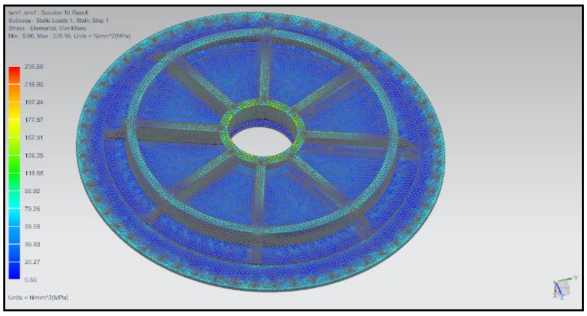


Figure 19 – von Mises stress distribution in the lower stator

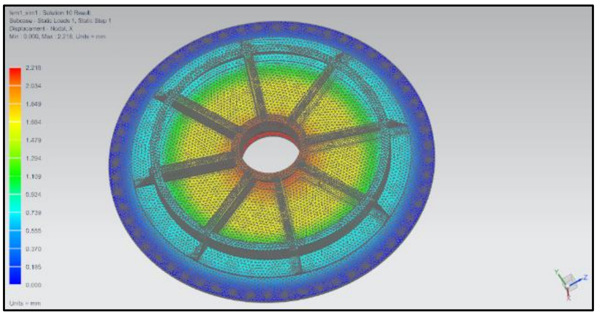
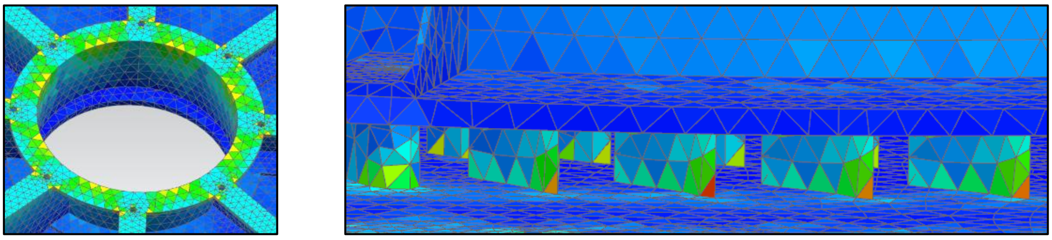


Figure 20 – Axial deformations for the lower stator



a) b)

Figure 21 – Location of stress concentration areas: a) Central cylindrical zone; b) Connection of the blades to the disks; stator blade edges

6. References

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