Ref. Ares(2023)4562544 - 30/06/2023

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POWER STORAGE IN D OCEAN

D3.1. Design Report of the KESS dimensioning, rotodynamic analysis and calculation of the machine side converter.

Document Information	
Deliverable	Design Report of the KESS dimensioning, rotodynamic analysis and calculation of the machine side converter.
Lead Beneficiary	ANTEC
Type:	R -
Work Package	WP3 ESS 2: KESS Kinetic Energy Storage System (Flywheel) for marine applications.
Date	30/06/2023

Funded by the European Union

This project is funded by European Climate, Infrastructure and Environment Executive Agency (CINEA), under the powers delegated by the European Commission under grant agreement Project: 101096457 – POSEIDON - HORIZON-CL5-2022-D5-01.

Dissemination level

Dissemination Level	
PU: Public	
SEN: Sensitive, limited under the conditions of the Grant Agreement	X

History

Version	Date	Reason	Revised by
1	17/05/2023	First draft	
2	09/06/2023	Final Version	
3	26/06/2023	Peer reviewed	Ana Orduña (CTN)
4	30/06/2023	Final version submitted	Ivan Felis (CTN)

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1. INTRODUCTION

Flywheel energy storage systems (FESS) use electric energy input which is stored in the form of kinetic energy. The inertia allows the flywheel to continue spinning and the resulting kinetic energy can be converted to electricity. A flywheel is a disc with its weight concentrated toward the outer circumference. When the disc is spin, it stores energy by virtue of its angular momentum. To improve the efficiency of the system, the rotating mass should have as low friction losses as possible.

The two main advantages of the KESS energy storage method, compared to other systems, are the absence of carbon emissions and faster response times. Moreover, it has an attractive energy density, high efficiency, and low standby losses, periods of many minutes to several hours.

On the other hand, their main disadvantage is the limited energy storage time of a few minutes, which makes the flywheels only suitable for quick and timely applications. Flywheels are therefore mainly used for regulating and optimizing systems, rather than for ensuring long-term autonomy like batteries, for example.

As flywheels spin, heavy flywheels store more energy than lighter ones. However, heavier flywheels accelerate slower. Lighter flywheels create less inertia at launch, but accelerate quicker through the rpm band.

Flywheels are a mature technology, developed for many different conditions. However, this is not the case for marine applications. Main problem for marine application is the movement of the boat and the forces related with that movement. The analysis of these conditions will be held in task 2. KESS ONBOARD INTEGRATION REQUIREMENTS

To size the flywheel, the energy and power of the machine must be taken into account, starting from these initial specifications, both the revolutions per minute of the flywheel and its weight must be calculated. 3. KESS DESIGN

Concerning the consortium and the tasks distribution.

There is a long and deep experience of CIEMAT in developing both categories of flywheels, fast and slow, although with more activity in slow ones. The applications for which they were mainly developed (railway traffic energy management in substations) are more suitable for this type of KESS, even from an economic perspective.

CYCLOMED has capabilities for the analysis and consideration of marine and in-vessel conditions to be used in the design of the KESS.

ANTEC, industrial partner, has experience in the design and manufacture of electromagnetic systems for different applications such as energy generation and storage, among others.

The leader of the Work Package is ANTEC and the different tasks are led as follows: analysis of marine and ship conditions by Cyclomed, design and calculation of the system by Ciemat and, finally, manufacturing by ANTEC.

2. KESS ONBOARD INTEGRATION REQUIREMENTS

2.1. INITIAL CONSIDERATIONS

KESS systems have a high level of TRL for terrestrial applications, however, few developments have been carried out in the maritime environment. KEES are based on storing energy (E) in a rotating mass (flywheel). The maximum stored energy can be defined by selecting the moment of inertia and the rotational speed, although they are not independent variables and are related through mechanical constraints. The energy density that can be stored is limited by the elastic limit of the material used. There are two types of flywheels depending on their rotational speed: "slow flywheels", spinning below say 15.000 rpm) and made from high strength steel (alternative other metals also) and "fast flywheels", typically spinning above 30.000 rpm and made from fiber. Slow flywheels achieve high values of energy because they are dense while fast flywheels achieve high values of energy because the rotating speed is high.

POSEIDON project for the waterborne KESS is based on the Slow Flywheel category specifically addressed to this application where issues related to the ship environment can be crucial. In this case, slow flywheels are selected as the technological solution because the geometry is not a problem and the shape is optimal from the point of view of maximum energy density, since the material is isotropic. In addition, the raw material used for slow flywheels is cheaper and the manufacturing processes include less stringent requirements in terms of tolerances, since dynamic balancing is less severe. Finally, the electric machine can be simpler than in the case of fast flywheels, since they do not need to use magnetic bearings for rotation. Finally, although the weight of the slow flywheels is heavier than that of the fast flywheels, the extra-weight associated to this category is fully acceptable for the application in which the flywheel weight is only a tiny fraction of the ship deadweight.

The objective during these first months of project execution has been to design a switched reluctance machine to achieve the target power. The length of the machine (cross section geometry is already closed) will determine the torque and the power will be calculated from this torque and the flywheel rotation speed whose value is known. Likewise the length of the flywheel will be designed to reach the target energy, once the cross section has already been defined. Once the geometry is fixed, a rotodynamic analysis will be performed at this stage to evaluate the eigenfrequencies according to the Campbell Diagram. This phase will be mainly performed at CIEMAT with ANTEC collaboration.

2.2. MARINE CONDITIONS

The marine environment is harmful for a lot of machines and systems, it has a lot of humidity, salinity and lot of restrictions depending on which kind of boat the system will be installed. In the case of kinetic energy storage systems (KESS), all these conditions affect to the design, but being a system based on a big rotary mass, the biggest challenge in the marinization of the system is how to deal with the movement of the boat and the gyroscopic forces that appear in the rotary system due to the boat movement.

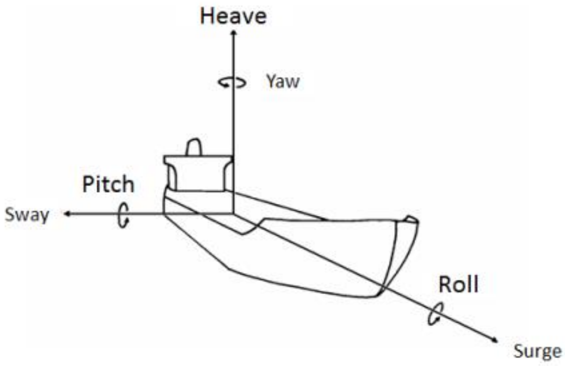


Figure 1. Boat movements

Boat movements and the forces related with those movements can be separated in 2, linear movement with linear forces and angular movements with gyroscopic forces. The movements of the boat are not the same for every case, it depends on the geometry of the boat, the waves and sea conditions and the load of the boat, but it can be generalized as the bigger the vessel, the higher stability and lower accelerations. Also, the higher waves, higher accelerations in the boat.

In the POSEIDON project, a passenger ferry with the normal operation between island in the Mediterranean Sea will be the final target of the project. Based on the boat type and the sea conditions, two different conditions will be differentiated, the first one will be the nominal operation condition and the second will be the extreme condition.

Figure 2. BALEARIA Boat for POSEIDON project

2.2.1. GYROSCOPIC FORCES

The wave disturbance induces rotational movement in the boat, roll and pitch mainly, as the flywheel in anchored to the boat, the movement of the boat induce precession motion in the flywheel. The precession is the change in the orientation of the rotational axis in a rotational system, in this case, the change in the flywheel rotational axis. The precession is the cause of the gyroscopic torque in rotational system.

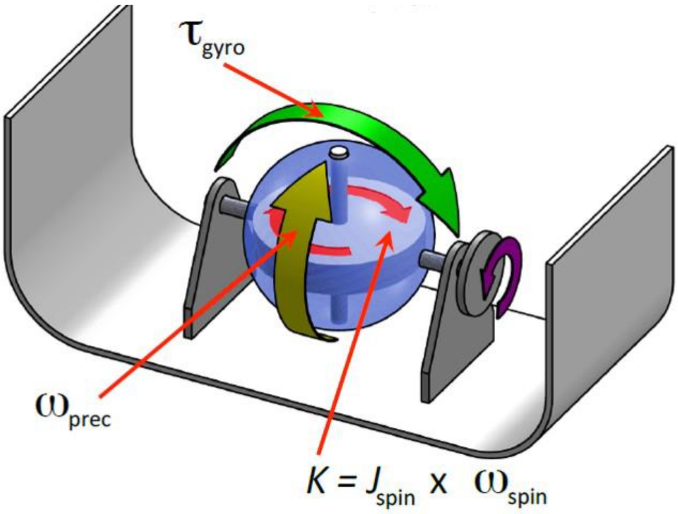


Figure 3. Schematic image of gyroscopic forces.

The gyroscopic torque is described with the following equation.

$$\tau_{Gyro} = J_{spin} * \omega_{spin} * \omega_{prec}$$

Where τ_{Gyro} is the gyroscopic torque, J_{spin} is the polar inertial moment of the flywheel, ω_{spin} is the rotational speed of the flywheel and ω_{prec} is the rotational speed of the boat.

2.2.2. NOMINAL OPERATION CONDITION

The nominal operation of the system will be on a ferry boat between islands. In this specific conditions, small waves up to 1.5m height are expected as normal condition.

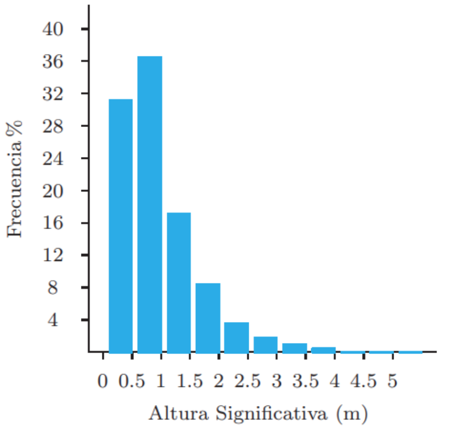


Figure 4. Historic waves amplitude between Balearic Islands (from genuary. 1985 to april 2022).

Since the final geometric characteristics of the boat are not defined, conditions from "*Measuring vessel motions using a rapid-deployment device on ships of opportunity*" will be used to estimate the boat motions. In this study the motion of a passenger ferry boat is studied but the average waves are higher than in the POSEIDON project, 2 to 3.5 meters against 1 to 1.5 meters height in our sea conditions.

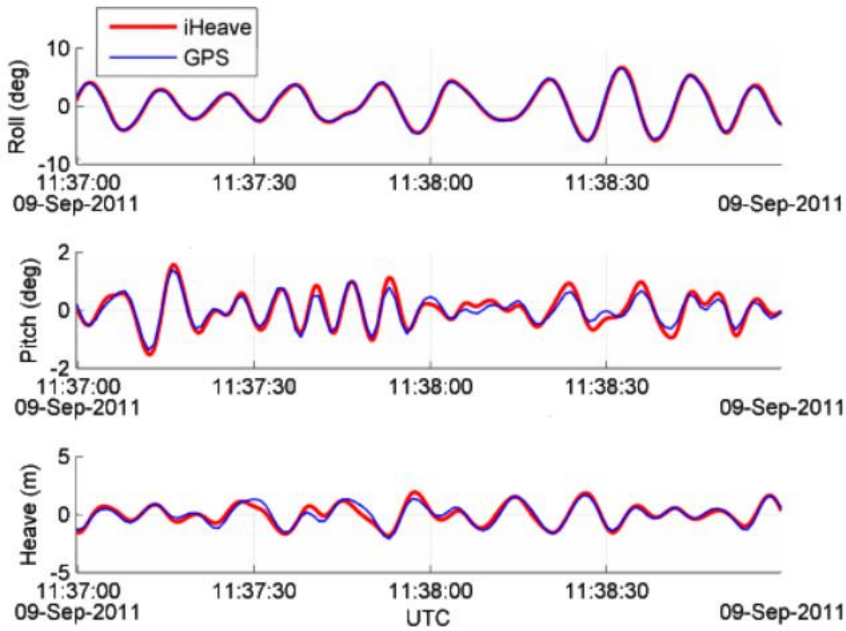


Figure 5. Boat movement in "Measuring vessel motions using a rapid-deployment device on ships of opportunity"

From the data obtained, a roll angular velocity of 2 deg/sec and a heave acceleration of 0.5 m/s^2 are estimated. This movements will be used in the following sections to calculate the forces in the flywheel and evaluate all the possible solutions.

2.2.3. EXTREME CONDITIONS

The extreme conditions or worst case should be delimitated by regulations and governments, but there isn't any specific regulation for flywheel or similar machines onboard. The only limitation found in the regulation is with the classification societies with the following table.

Installations, components	Angle of inclination (degrees) **(1) (4)**			
	Athwartship		Fore and aft	
	static	dynamic	static	dynamic
Main and auxiliary machinery	15	22,5	5	7,5
Safety equipment, e.g. emergency power installations, emergency fire pumps and their devices Switch gear, electrical and electronic appliances **(3)** and remote control systems	22,5**(2)**	22,5 **(2)**	10	10

(1) Athwartship and fore-and-aft inclinations may occur simultaneously.
(2) In ships for the carriage of liquefied gases and of chemicals the emergency power supply must also remain operable with the ship flooded to a final athwartship inclination up to a maximum of 30°.
(3) No undesired switching operations or operational changes are to occur
(4) Where the length of the ship exceeds 100m, the fore-and-aft static angle of inclination may be taken as 500/L degrees, where L is the length of the ship, in metres, as defined in Pt B, Ch 1, Sec 3, [2.1.1]

Table 1: Regulation of angle of inclination for machinery and safety equipment on a boat

According with the table, the flywheel must be designed for a maximum angle inclination for 22.5 degrees.

In *"Offshore Application of the Flywheel Energy Storage"*, similar project with a worse case is presented, with a smaller boat and higher waves. This project presents their worst case with the following characteristics:

- Roll movement: +/- 45 deg in 12 sec
- Pitch movement: +/- 12 deg in 9 sec
- Heave acceleration: 0.4 g
- Surge acceleration: 0.25 g

The previous characteristic will be used as extreme conditions as the boat movement presented in this project are more severe than the requirement of the classification societies.

3. KESS DESIGN

3.1. POWER AND ENERGY TARGETS

3.1.1. ENERGY

Kinetic energy storage systems are based on storing energy in a rotating mass, according to the equation E = ½ ·J · w2, where J is the mass moment of inertia and w its angular velocity. The acceleration and deceleration of the mass is achieved by injecting and extracting power from an electric machine whose moving part rotates in solidarity (mechanical coupling) with the flywheel. The power of the electric machine determines the power of the storage system. In this way, power and energy are decoupled in the flywheels.

Regarding the design target energy exchangeable by the flywheel at rated power (upper operating speed range), this is set at **9MJ (2.5kWh)**. On the other hand, this interchangeable energy at rated power is usually set to be half of the total storable energy (18 MJ from zero to maximum speed). However, a minimum operating speed is set for flywheels, which in this case is set at 40% of the maximum speed, corresponding to a usable energy of 15 MJ. The material to be used for the manufacture of the flywheel will be high strength steel of about 650MPa, so if the design energy is 9MJ and if a safety coefficient of 30% is given to the maximum allowable stress, the mass M of the flywheel must be:

$$M = \frac{1}{0.7} \cdot \frac{\rho \cdot E}{0.606 \cdot \sigma_M} = \frac{1}{0.7} \cdot \frac{7800 \cdot 15}{0.606 \cdot 650} \cong 400kg$$

where:
ρ: Density of steel (g/dm^3)
E: Energy required (MJ)
σ_M: Maximum specific stress (MPa)

Another parameter that needs to be fixed and that has to do with the amount of energy the flywheel is capable of storing is the maximum rotational speed. Due to possible limitations of the control platform to be used and the maximum switching frequency of the power semiconductors, the target maximum design speed is set at 13000rpm. Once the mass and target maximum speed have been defined, it is necessary to determine the dimensions of the flywheel (w x h x l). A limiting factor in this regard is the dimension of commercial high strength steel bars. After probing the market, bars up to 600mm in diameter have been found. Knowing the radius, solid cylindrical shape, maximum speed and energy, the required machine length is calculated according to the following formula. The resulting length is 160 mm, with a polar moment of inertia of 16 kg·m^2:

$$E = \frac{1}{2} \cdot J \cdot w^2 = \frac{1}{4} \cdot V^2 \cdot M = \frac{1}{4} \cdot \pi \cdot \rho \cdot V^2 \cdot R^2 \cdot H$$

On the other hand, the diameter of the surface where the rotor is to be strapped must meet two main requirements: 1) be large enough so that sufficient interference can be achieved by mounting it by thermal difference and 2) be small enough so that the available rotor area for the magnetic flux path is not reduced too much. For this last requirement, a maximum diameter of 120 mm.

By means of ANSYS (finite element software) modelling of the rotor profile in 2D axisymmetric and 3D, the most suitable shape for the design is being sought. Once the maximum stress in the centre of the flywheel is known by applying the analytical results of the elasticity theory, it will be tried to locate and eliminate any possible dangerous stress concentration in the flywheel.

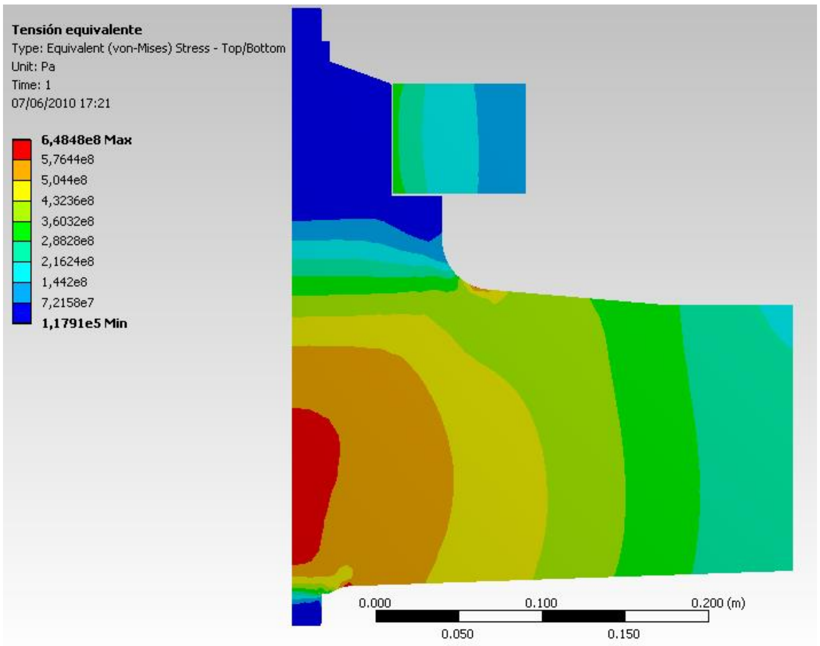


Figure 6. Axisymmetric model of the levitation flywheel with maximum stress of 648MPa.

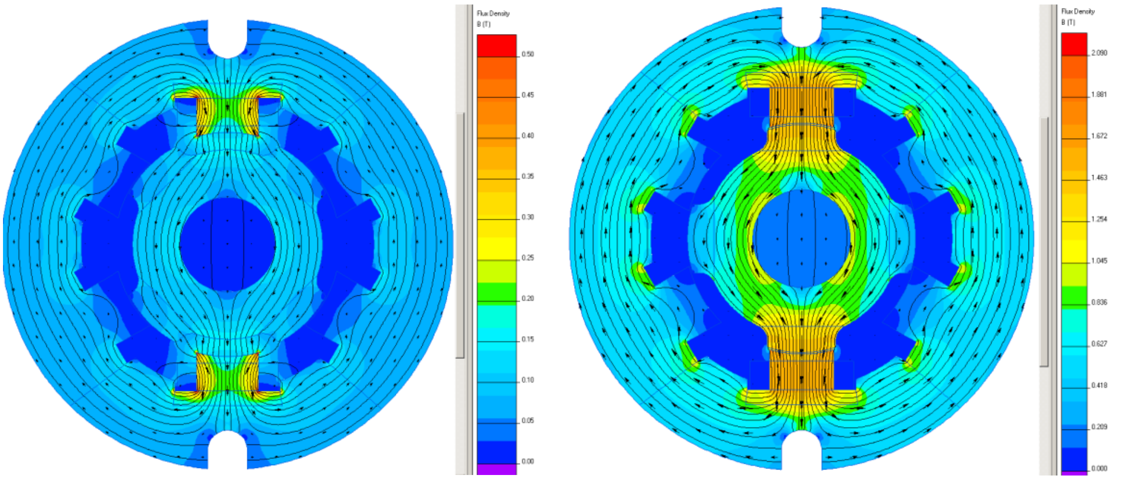
Regarding flywheel balancing, both static and dynamic balancing is required. Dynamic balancing makes it possible to determine the possible effective variations in the degree of balancing as a function of deformation at supercritical speeds according to ISO 11342. The purpose of balancing is to ensure that the radial forces of inertial origin supported by the bearings will be maximum: 1400 N.

3.1.2. POWER

One of the specific objectives of the POSEIDON project is the construction of 3 ESS (SMES, Supercapacitors and Flywheel) in the power range of 20-200kW. Regarding the design target power of the KESS, it is set at **25kW**. This value, associated to the electric machine, is selected based on CIEMAT's experience in the design of machines for this type of applications and because it is considered an appropriate value for a unitary module. The modularity of the storage system allows the parallel connection of several modules in case of needing to increase the power in future applications.

Among the different electrical machines available on the market, switched reluctance machines and permanent magnet machines are the most appropriate for this type of application. In this case, the switched reluctance machine has been chosen because it has lower losses than the magnet machine in the absence of torque, no demagnetization problems at high temperatures, lower cost and does not depend on rare earths. Among the possible configurations of the switched reluctance machine, a 6/4 machine was chosen because for high-speed applications a machine with a very high number of poles would imply high switching frequencies and higher torque ripple.

From the minimum speed at which the SRM operates at rated power, the maximum torque of the machine is determined. With the value of this torque and the rotor radius, the required magnetic field as a function of the SRM height is calculated using finite elements. The figures show the magnetic flux lines in the machine when the rotor poles are aligned with the stator poles and when they are misaligned.



(a) *(b)*

Figure. Magnetic field lines in the machine (a) for the 0° position (poles not aligned) and (b) for the 45° position (poles aligned) for the rated current in the coils.

3.2. ROTODYNAMICS AND SIZING

The rotodynamic study in a flywheel is very important, it studies the behavior of rotary systems. Rotodynamic can determinate critical rotational speed of the system and its necessary to avoid resonance in the operational speed of the flywheel. In this study the stiffness and characteristic of the flywheel will be evaluated to determinate the critical speed of the flywheel.

The stiffness of a KESS is determined by the stiffness of the bearings and the stiffness of the housing. The bearing stiffness is not constant, it varies according to the load and the size of the bearing, also the maximum admissible load and maximum speed of the bearing varies according to the size, but the life of the bearing is determined by the relation of load in operation conditions and maximum admissible load.

3.2.1. FLYWHEEL FORCES

First step in the rotodynamic study is to determinate the loads in the flywheel, on one side, the operational loads according to the normal operation condition, and in the other side the maximum forces according to the extreme conditions.

In the normal operation condition, the main forces that are applied in the flywheel are gyroscopic forces, inertial load and weight. The gyroscopic and inertial forces are in the same direction, radial, but the weight load is in the axial direction.

Figure 7. Radial force in the bearing for each rotational velocity

In the picture, the radial forces in the bearing depending on the speed of the flywheel are represented. Gyroscopic forces are 4 times higher than inertial forces at operational speed (8000-13000 rpm).

In the extreme conditions, the most relevant movements are the roll movement and the heave acceleration, the first one will affect as a gyroscopic force in radial direction and the heave acceleration in the axial direction.

Roll movement in extreme conditions is represented as a sinusoidal movement with +/- 45º in a period of 12 sec, in this movement profile, the maximum roll speed is **23.5 deg/s.** With this maximum roll speed, the radial force is **21500 N**. In the axial directon the heave acceleration of 0,4 g with the mass of the system of 400kg, the axial force is **1560 N.**

3.2.2. BEARINGS

The bearings are a critical part in the KESS, they support the flywheel to the housing and allow it to spin. Some change in the characteristic of the bearing will affect to the behavior of the system, so every change must be analyzed.

The original bearing configuration has a hybrid deep groove ball bearing. This type of bearings has a very good characteristic in terms of electrical insulation, speed capability and friction, these properties are the most important in our application. The life of the bearing with original bearing configuration, due to the gyroscopic forces, has been reduced to **787 hours**. This life expectancy is not acceptable in this application, because this short life of the bearing would cause to many maintenances stops to change the bearing of the flywheel.

Two different strategies are presented to solve the problem of the bearing life, first of all is change the configuration to a double bearing configuration, this configuration has the disadvantage that the speed has to be reduced to 80% due to put 2 bearing together. The second strategy is changing the bearing to a bigger bearing of the same family (hybrid deep groove ball). The result from both strategies are shown in the table:

	Model	**6207 (current)**	**6208**	**6209**
Geometry	d (mm)	35	40	45
	D (mm)	72	80	85
	B (mm)	17	18	19
Properties	Limit load (kN)	**25,5**	**30,7**	**33,2**
	W limit (RPM)	13000	11000	11000
	Bearing life (h)	**787**	1623	**2053**
Doble bearing	Bearing life (h)	**4326**	8923	11285

Table 2. Properties of the bearings

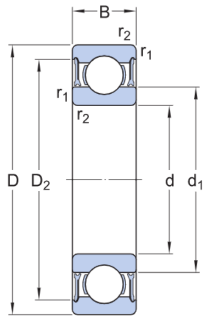
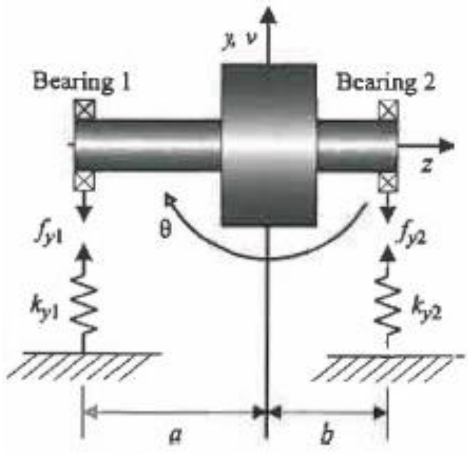


Figure 8. Schematic image of the bearing

It can be seen in the table that both strategies have limit load higher than the load in the extreme condition (21,5 kN) and also both configurations have a significative increase in the life of the bearing, 4326 hours and 2053 hours. Both strategies are valid in term of maximum load and bearing life. The next step is to evaluate all system together to evaluate the rotodynamic of the system and chose the best option.

3.2.3. ROTODYNAMIC

In the rotodynamic analysis a rigid flywheel and independent bearings are considered to evaluate the behavior of the system. The stiffness of the bearing is load dependent and the supplier provide the stiffness matrix for our load case for both bearings.



$$k_{xT} = k_{x1} + k_{x2},$$
$$k_{xC} = -ak_{x1} + bk_{x2},$$
$$k_{xR} = a^2k_{x1} + b^2k_{x2},$$

Figure 9. Schematic image of the flywheel (left), equation of stiffness of the system (right)

From this stiffness matrix of the system and the kinematic equation associated with these degrees of freedom, the following characteristic equation arises.

$$\omega^4 \mp \left(\frac{I_p}{I_d}\right)\Omega\omega^3 - \left(\frac{k_R}{I_d} + \frac{k_T}{m}\right)\omega^2 \pm \left(\frac{k_T I_p}{mI_d}\right)\Omega\omega + \frac{k_R k_T - k_C^2}{mI_d} = 0$$

Equation 1. Rotodynamic equation of the system

For each velocity value, four roots are calculated that correspond to the four curves of the Campbell diagram. FEM analysis was performed to evaluate the rotodynamic and Campbell diagram of the system, first of all, a simplify model, with the flywheel and rotor, was studied and after analyze the first result a more detailed model with the housing was studied.

With the simplified model, it can be seen in the 2-bearing configuration with 6207 model, left picture, that has the first critical speed around 8500 RPM, inside the range of operation but in the configuration with the bigger bearing, 6209 model, there are 2 critical speeds in the operation range, 9000 RPM and 11000 RPM.

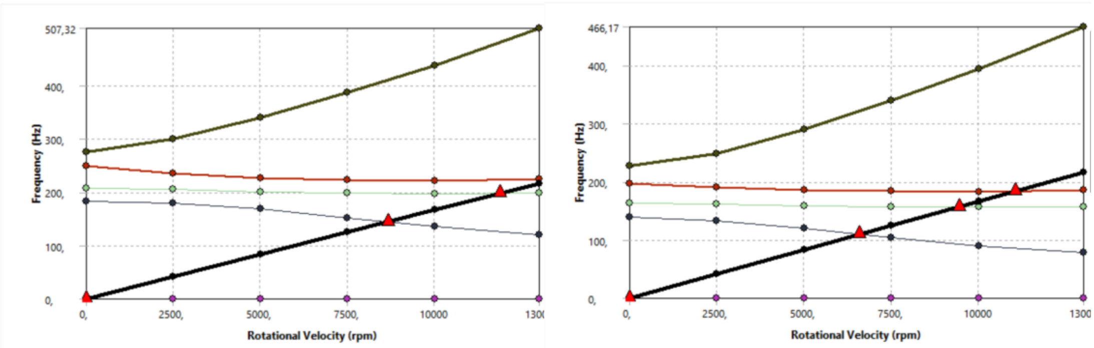


Figure 10. Campbell diagram of 2-bearing configuration (left) and Campbell diagram of bigger bearing configuration (right)

Once the simplified model is evaluated, the 2-bearing configuration is a better option because only one critical speed is in the operation range. This configuration is studied with the detailed model, including the electrical machine and the housing of the flywheel.

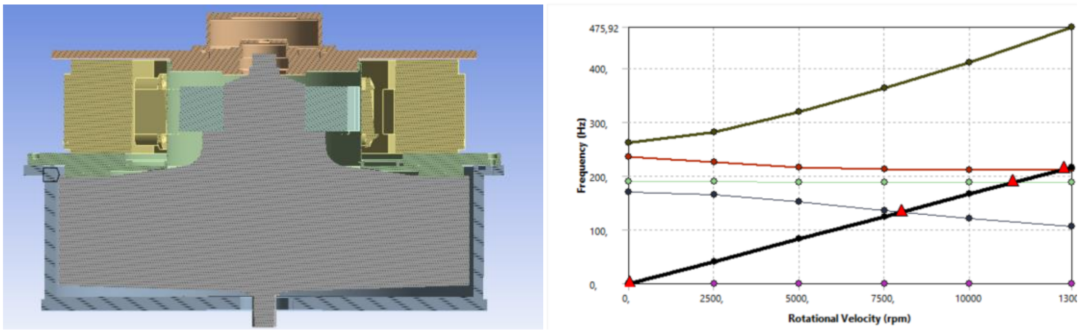


Figure 11. Detailed FEM model for rotodynamic analysis (left) and Campbell diagram of the model (right)

The results of the detailed model can be seen in the picture X. The results show that all the critical speed is out of the operational range, between 8000 and 10400 RPM.

The conclusion of this study is that the best configuration for a marine environment is the 2-bearing configuration with the current bearing model, 6207. The final limit load of this configuration is **51 kN**, much higher than the maximum load in our application, 21,5 kN, and the life of the bearing is more than **4300 hours**, enough to avoid having to make excessive maintenance stops. Finally, the operation range of the KESS will be between **8000 RPM and 10400 RPM**, with this range of operation the final energy storage is **9,7 MJ** of energy.

3.3. POWER ELECTRONICS (CONVERTER) AND CONTROL

The Switched Reluctance Machine (SRM) is characterized by the need to operate in conjunction with power electronics that control the sequential activation of each of its phases. The operating principle of a SRM is based on minimizing the energy of a magnetic circuit, i.e. reducing the reluctance of the circuit. Figure shows an example of a 6/4 SRM in which the phase A coils are energized with a supply current. Following the principle of minimum reluctance search of the magnetic circuit the rotor tries to rotate counter clockwise until this position is reached (position of maximum alignment). When the minimum energy of phase A is reached, the magnetic forces try to keep the rotor in this position in compliance with the above principle. At that moment the power supply of phase A must be turned off, energizing the next phase, making the rotor rotate. Once this switching of the phases is done, continuous motion is generated in the SRM.

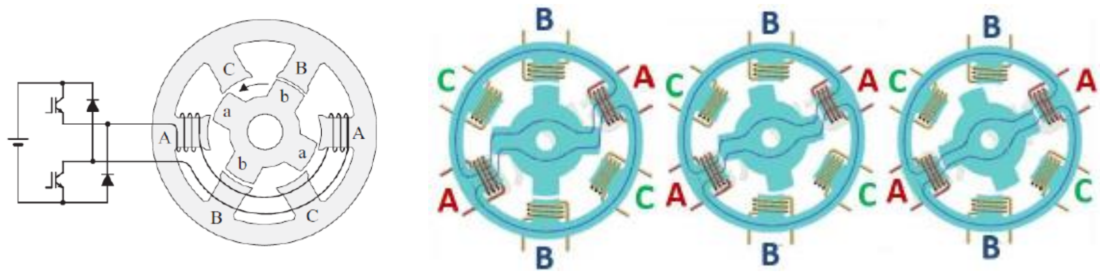


Figure 12. Principle of operation of a SRM 6/4.

As mentioned at the beginning of this point, the electrical machine requires a power converter that allows the switching on and off of the phases. The topology of the converter is shown in the figure. The converter allows three different voltage levels to be applied to each of the phases of the machine: $+U_{DC}$ to increase the current or $-U_{DC}$ or 0V to decrease the current. In this way it is possible to regulate the level of current through the phases, and thus control the torque and power exchanged by the machine. Another important aspect of the converter is that it must be bidirectional in power to allow charging and discharging of the kinetic storage system.

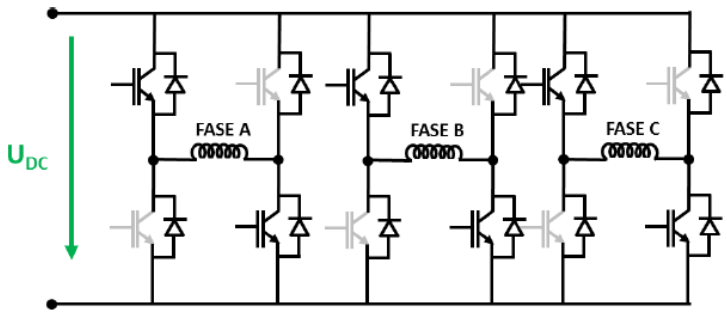


Figure 13. Schematic diagram of three-phase, 3-bridge H-bridge converter used to power the SRM.

In order to calculate the waveform of the currents through the phases of the machine and the power semiconductors, calculate the power exchanged by the machine, validate the control strategy of the converter and select the commercial semiconductors of the converter, a dynamic model of SRM has been developed in Matlab-Simulink. This model also allows the evaluation and calculation of the effect of certain parameters on the dynamic response of the storage system, such as: saturation of the electrical machine, high rotation speed, phase activation and deactivation angles, etc.

For the SRM model, the machine torque equation (T(θ,i)) and the expression that determines the evolution of the current (di/dt) (as a function of the position of the rotor with respect to the stator and the voltage applied in the phase) is used. The expression of the torque and the evolution of the current for an unsaturated SRM is shown in the following equation:

$$T(\theta, i) = \frac{1}{2} \cdot \sum_{k=1}^{m} i_k^2 \cdot \frac{\partial L_k(\theta)}{\partial \theta}$$

$$\frac{di}{dt} = \frac{U - R \cdot i - f.e.m.}{L(\theta, i)}$$

In order to know the evolution of the mechanical torque when the SRM is saturated, an analysis with finite elements is made. In this analysis, the evolution of the mechanical torque is calculated based on the position of the rotor poles with respect to the stator poles.

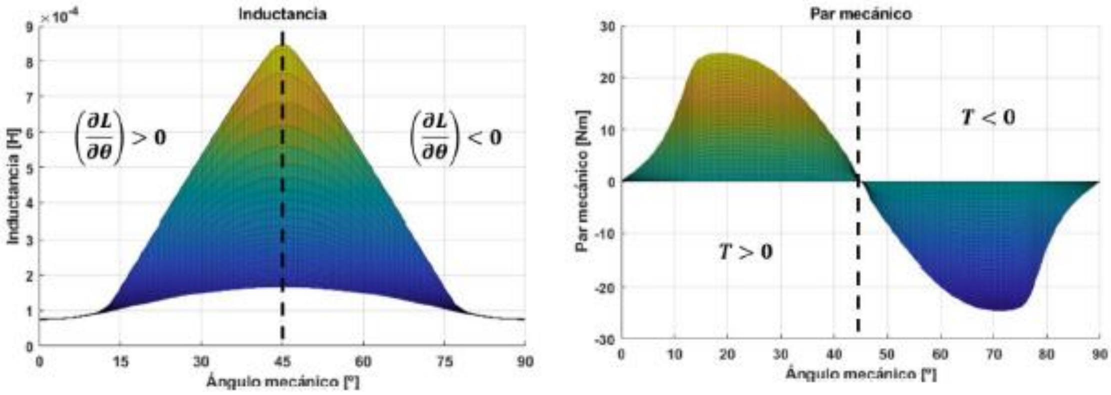


Figure 14. Characteristic curves of SRM: Inductance and mechanical torque as a function of the position of the rotor pole

The model implemented in Matlab-Simulink allows knowing the dynamic behaviour of the SRM. This model is made up of 3 main parts, which are represented in the following figure.

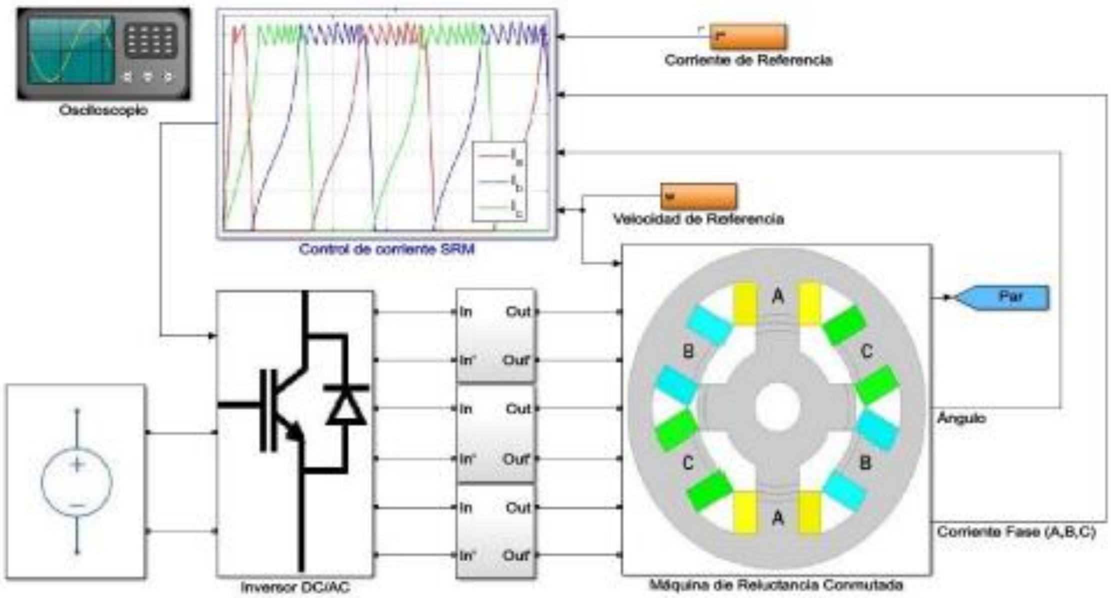


Figure 15. Scheme of the SRM model developed in Matlab Simulink.

The main parts of the model are the power electronic converter plus the DC-link, the control of the power electronic converter and the switched reluctance electrical machine plus the flywheel.

The power converter includes the topology shown in figure 14 and the DC-link is modelled with a constant value DC voltage source. In the control stage, the current hysteresis band control strategy is implemented. Finally, the machine model includes the dynamic tables of the machine flux as a function of the current through the phases and the position of the rotor relative to the machine stator. These flux values are calculated in the previous finite element analysis. From these values and the voltage applied to the phases, the current waveform in the operating speed range is known, as shown in figure 17. Furthermore, the machine force is also estimated as a function of current and position by finite element analysis. With this force value and the instantaneous speed, the mechanical power developed by the machine is calculated. The electrical power is calculated by multiplying the DC-link voltage by the measure of the current (I_{DC}).

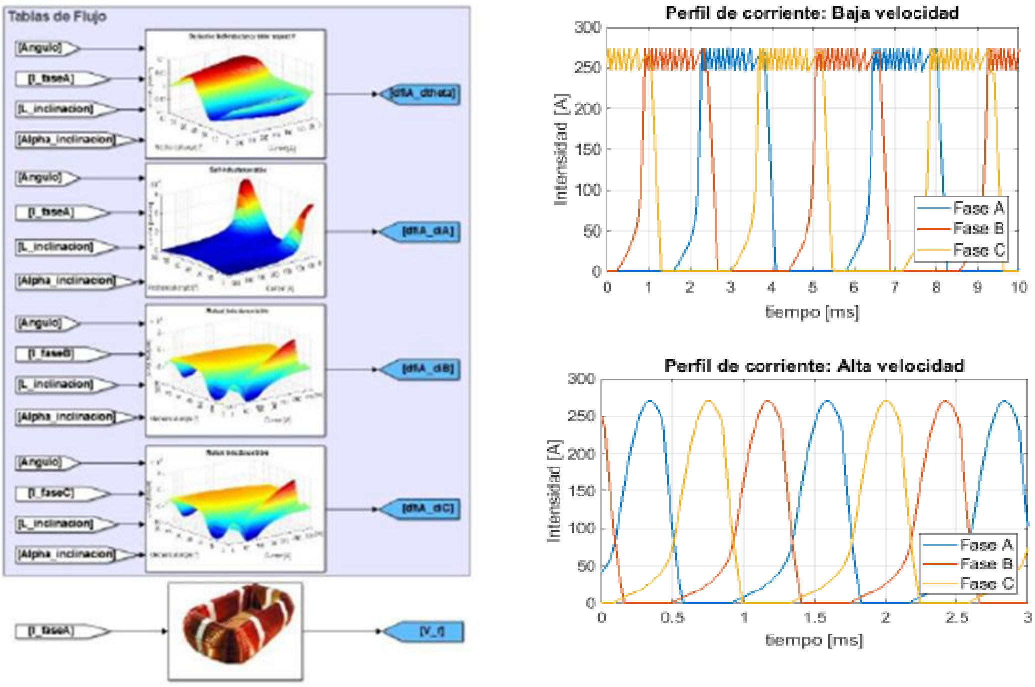


Figure 16. Magnetic Flux link as a function of angle and current. Low speed current control (hysteresis band) and high-speed current control (monopulse).

Figure 18 shows a simplified schematic of the electrical machine, the power electronic converter and the control platform (DSP from Texas Instruments) and the signals exchanged between the different parts.

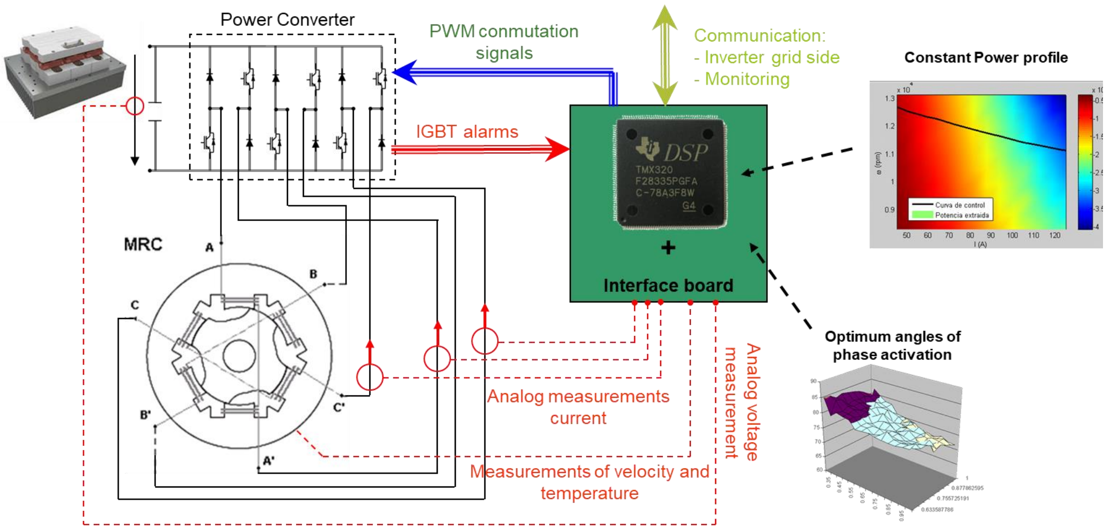


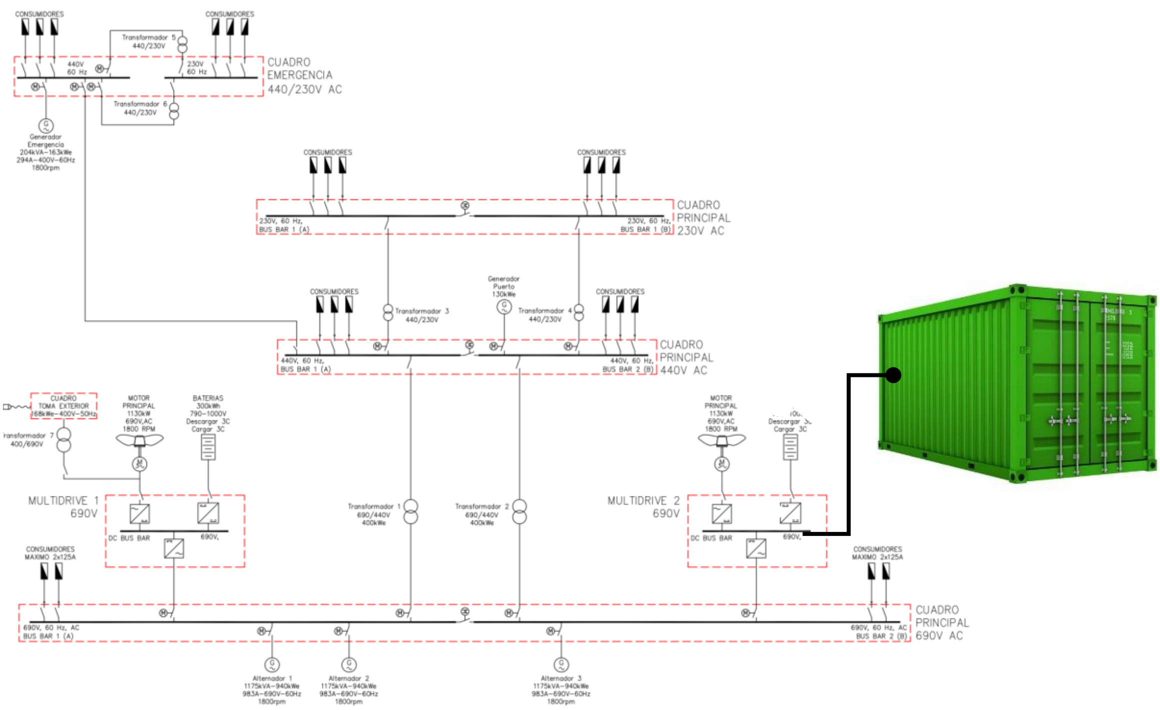
Figure 17. Simplified diagram showing the power stage of the kinetic storage system, the control platform and the signals exchanged between them.

4. ASPECTS OF THE INTEGRATION ON-BOARD

Integration the technologies on board a vessel presents significant challenges due to space restrictions and specific requirements such as humidity and temperature. In order to address these challenges, a preliminary version of the main requirements has been developed with the support of BALEARIA. Additionally, a scheme has been created to illustrate the general layout of the vessel and highlight the integration points for the various technologies.

The main results of this preliminar study are shown below.

- Scheme of the connections on-board: we present the single-line diagram of one of the lines in BALEARIA's vessel, as well as the connection point for the container's power supply, where the technology could be located.

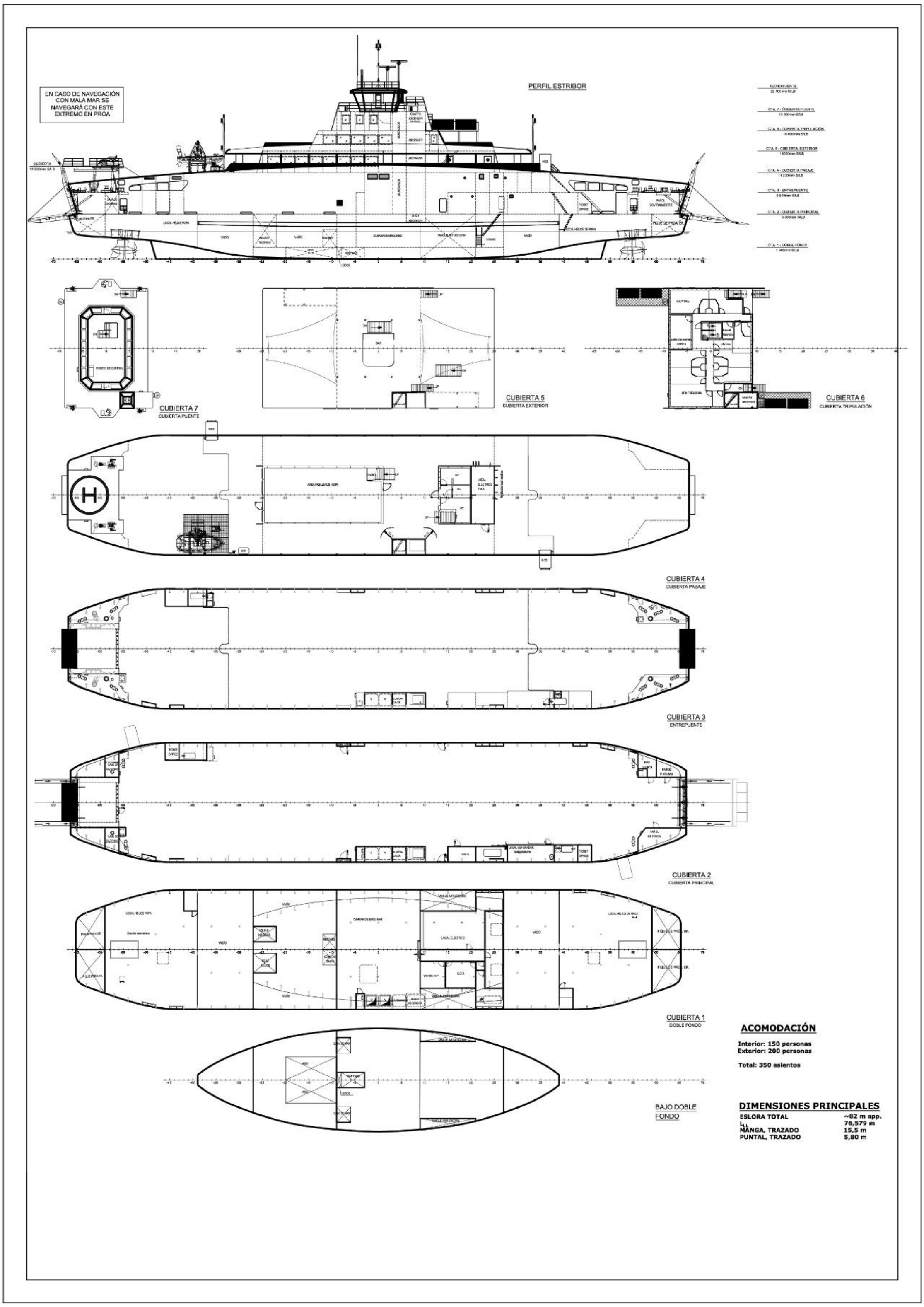


- Requirements provided by BALEARIA: although WP5 is specifically dedicated to studying the overall suitability of each FRESS technology for ship integration and, in particular, the integration of KESS and EESS in the container, the following initial specifications should be taken into account.

Requirement	Specification
Size	20/40ft
SSCC	Bureau Veritas
Current supply	
Localization	Garage
Voltage	According to the single diagram
Loading/Unloading means	By its own means, with a wheeled platform for easy loading and unloading

Table 3. Initial requirements that FRESS have achieve to the integration on-board, provided by BALEARIA.

- General layout of the vessel:



5. CONCLUSIONS AND OUTPUTS

During this first task t3.1 "modelling, design study and adaptation of a KESS for waterborne conditions", a study has been carried out to define the dimensions of the flywheel in order to achieve the set energy target. This energy is determined by the operating speed range (8000-10400rpm) and the shape and dimensions of the flywheel. The useful energy calculated in the analysis carried out in the indicated speed range was 9.7mj. Figure 19 shows the dimensions of the flywheel that allow the required energy to be achieved. in addition, during this task 3.1 a fem analysis has also been carried out to calculate the force that the electrical machine is capable of exchanging, once the geometry of the cross section has been fixed. A study was carried out to determine the dimensions of the electrical machine to achieve the target power, in this case 25kw. The force developed by the machine as a function of the current allows the torque to be calculated and, once the speed is known, it can be determined whether the machine design is capable of exchanging the required power. This is an iterative process with FEM analysis in which some parameters of the machine are varied until the target power is reached. figures 19 and 20 show the dimensions of the electrical machine that allows the target power to be reached.

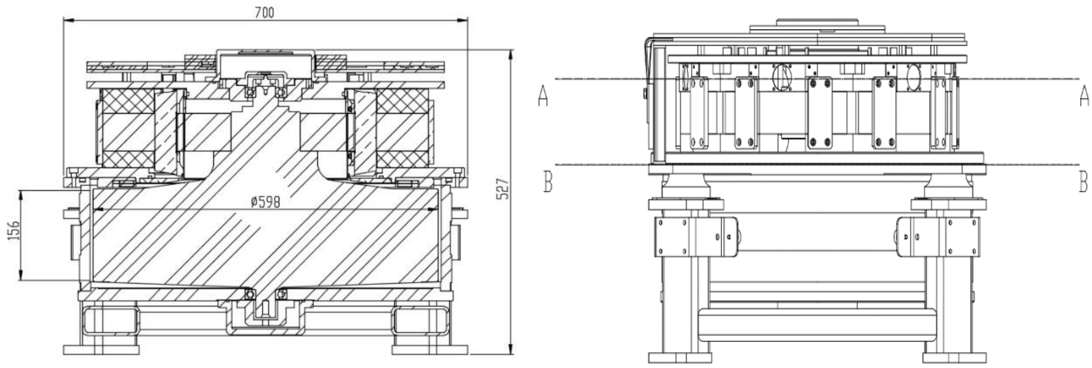


Figure 19. Cross-section of the kinetic storage system showing the flywheel and the electrical machine and schematic of the assembly.

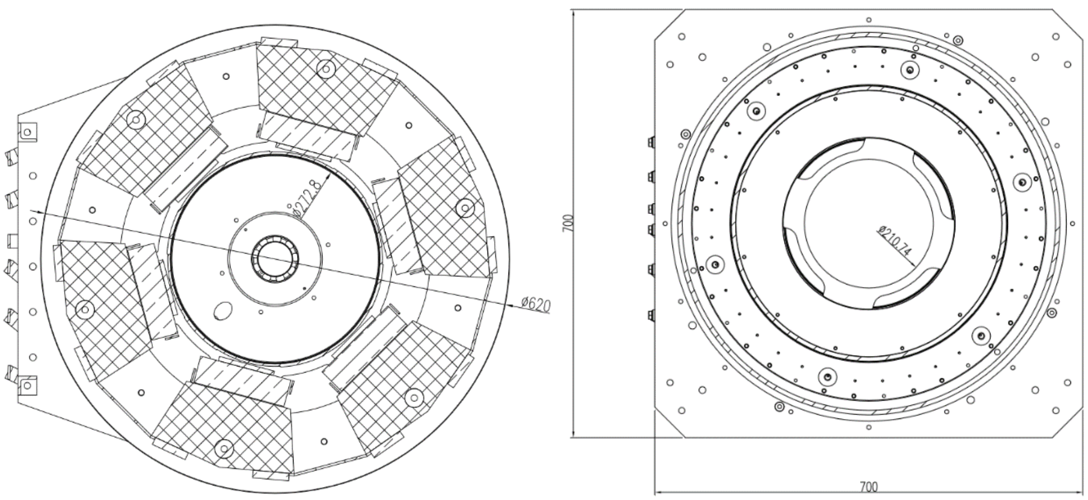


Figure 20. Cross-section A-A and B-B showing parts and dimensions of the KEES.

Finally, a rotodynamic study has been carried out to determine the stresses that the bearings will undergo due to the inertial force, as a consequence of the inertia of the flywheel itself, and the gyroscopic force, determined by the sea conditions. The axial force has been added to the inertial and gyroscopic force in order to find out, by means of a finite element analysis, which are the most suitable bearings for this application. From the analysis it can be concluded that the best option for the system designed in a marine environment is the **two-bearing** configuration. In the operating speed range and with the bearing model chosen (ball bearing model 6207) they allow more than **4300 hours** of operation, which is considered an acceptable maintenance time.