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Dynamics of a Flexible Mobile Flywheel Energy Storage System – Theoretical Modeling and Analysis

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Abstract

Flywheel energy storage systems (FESSs) based on active and passive magnetic bearings (MBs) have a huge energy storing potential in mobile applications. Under such operating conditions the FESS is exposed to gyroscopic forces, which can be handled by mounting the FESS in a passive gimbal. The purpose of this article is to investigate the coupled dynamics among the flexible bodies: gimbal, housing, and rotor in a FESS. This is carried out by theoretical modal analysis of each body and the FESS by utilizing the finite element (FE) method. Modeling of the bodies are carried out in ANSYS MAPDL using SOLID186 elements. The model of the FESS has been built by assembling the FE models of the bodies, representing the MB forces by linear springs using the MATRIX27 element. The first six natural frequencies and mode shapes have been identified for the gimbal, housing, and rotor occurring in the range of 195-801 Hz, 2882-4580 Hz, and 1346-6504 Hz respectively. The model of the FESS indicates 23 natural frequencies in the range 0-2000 Hz. Decoupled vibrations are noticed, such as isolated vibrations corresponding to the fifth mode shape for the gimbal and the first bending mode for the rotor. Coupled vibrations are also observed between the gimbal, housing, and rotor for which the active MB controllers must be designed taking such dynamics into account.

1 Introduction

Today’s modern society is still highly dependent on fossil fuels in terms of oil and natural gasses [7]. These are typically extracted from the ground by drilling. Often, this is carried out offshore where drillships are used in locations with very deep water and where drilling depths of up to 12.000 meters are needed [8]. During the drilling process, huge amounts of energy are sometimes wasted which are desired to store instead. The risks of explosive gas emissions and fire hazards in a potential battery used to store the energy on the drillships should be minimized, which can be achieved by storing the excess energy as kinetic energy in a flywheel. These type of batteries are called flywheel energy storage systems (FESSs).

The amount of kinetic energy stored is increasing quadratically with rotational speed of the flywheel and the life time is expected to be up to 20 years for such systems [3]. To compete with other battery types, the rotational speeds of FESSs must be very large and the friction and wear should be very low to achieve a high energy density. This can be achieved by implementation of magnetic bearings for supporting the flywheel instead of traditional ball bearings or hydrodynamic based bearings.

Support of the flywheel in the axial direction can be achieved by permanent magnetic bearings and active magnetic bearings can be utilized for radial support of the flywheel. When the FESS is installed in a mobile setting like a ship, it will be imposed to large gyroscopic forces, which in most cases are difficult to handle for active magnetic bearings. In [9] and [10] it is suggested how manoeuvring, vibrations, and gyrodynamics imposed to the FESS can be removed by implementing a passive gimbal mount. To investigate the coupled dynamics of a passive gimbal mounted rotor supported by active magnetic bearings, a test-rig has been designed at the Technical University of Denmark.

The flywheel energy storage system considered consists of a tilting frame, a gimbal, a housing, and a rotor. The gimbal frame is implemented into the test-rig with the sole purpose of removing gyroscopic forces, which is desired if the FESS described is to be implemented in mobile applications. Furthermore, magnetic bearings are mounted inside the housing to keep the rotor levitated and positioned in the centre. The energy is supplied to the system...
by blowing air on a compressor wheel mounted on the top of the rotor and thus, the energy is stored in the rotor which acts as the flywheel. A picture of the test-rig can be observed in figure 1 where only the top of the rotor can be seen and the compressor wheel is not mounted.

A mathematical model has been developed of the FESS in [3] where the multi-body dynamics of the gimbal, housing, and rotor is coupled to the forces from the magnetic bearings. In this model the parts have been assumed rigid. However, results [4] indicates that there might be an influence on the dynamics from the fact that the parts are flexible and not rigid. The purpose of this article is to investigate the flexibility effects on the dynamics of the FESS and the coupled vibrations between the flexible bodies through a theoretical modal analysis.

2 Finite Element Model

Theoretical results in terms of natural frequencies and mode shapes are determined by setting up mathematical models for the considered structures. These mathematical models are designed by utilizing the finite element method through the commercial program ANSYS. The finite element model of the FESS consists of individual FE models of the gimbal, housing, and rotor which are assembled together. During the investigation of the flexibility effects, the FESS will only be modeled with the rotor levitated but not rotating. Thus, gyroscopic forces are not considered. Furthermore, damping is neglected and contact modeling between the assembled parts in the housing and rotor is not taken into account. It is assumed throughout the report that all materials are isotropic and homogeneous. The main materials used in the model are aluminum and steel. In table 1 Young’s modulus, $E$, Poisson’s ratio, $\nu$, and the density, $\rho$, for the steel and aluminum types considered are listed provided from [1],[2].

<table>
<thead>
<tr>
<th>Material</th>
<th>$E$ [GPa]</th>
<th>$\nu$</th>
<th>$\rho$ [kg/m$^3$]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Aluminum</td>
<td>71</td>
<td>0.33</td>
<td>2770</td>
</tr>
<tr>
<td>Steel</td>
<td>210</td>
<td>0.3</td>
<td>7830</td>
</tr>
</tbody>
</table>

Table 1: Material properties for aluminum and steel

In the following the FE model of the gimbal, housing and rotor as well as the final global model of the FESS is considered. The tilting frame will not be taken into account. All parts are modeled using the SOLID186 element type, which is a quadrilateral 20 node solid element with 3 DOF: translation in $x$, $y$, and $z$-direction. The inter-part connections are modeled using linear springs through the MATRIX27 element type, which is an arbitrary element.
in ANSYS that can be used to relate the movement of two nodes to each other by defining the element type as a
stiffness matrix. When analyzing the gimbal, housing, and rotor individually, they are assumed in a free condition.
Moreover, the global model of the FESS is assumed clamped at the locations where the gimbal is mounted to the
tilting frame (see figure 1). Further information regarding the FE models of the gimbal, housing, rotor, and FESS
can be found in [6].

2.1 Gimbal
In order to represent the gimbal model using FEM, a simplification of the physical structure is carried out. Bolt
holes for the gimbal parts and connecting bolts are neglected. These are instead represented by contact surfaces.
In figure 2a the simplified structure of the gimbal is shown, consisting of four side parts (blue), four corner parts
(red), and eight contact regions (green). Furthermore, the global coordinate system for the model is shown. It
should be noted that two different corner parts are used, the ones connecting the gimbal to the housing and the
ones to the tilting frame with reference to figure 1, however, their geometries are close to similar.

![Simplified gimbal structure](image1)

![Sketch of spring directions in a contact region](image2)

**Figure 2**: Contact regions in gimbal

The contact regions are modeled using linear springs relating the translation DOF in the coincident nodes from
the corner parts and side parts. One spring is used for each translational DOF as shown on figure 2b. Here \(k_n\)
is the spring stiffness assigned to the spring relating the translational DOF in the normal direction to the contact
surface. The spring stiffnesses \(k_p\) and \(k_z\) are assigned to the springs relating the translational DOF in the
parallel direction and global \(z\)-direction respectively. The spring stiffnesses are determined by adjusting them to obtain a minimum sum of squares fit between the six first theoretical and experimental obtained natural frequencies. Details on the experimental obtained natural frequencies can be found in [1] and the obtained stiffnesses are:

\[
k_n = 2 871 000 \text{ N/m} \quad k_p = 3 065 000 \text{ N/m} \quad k_z = 1 748 000 \text{ N/m}
\] (1)

A convergence study has been carried out to assure that a suitable amount of elements is used to describe the
gimbal and the FE model consists of 7098 elements with 98892 DOF.

2.2 Housing

The housing consists of multiple parts assembled together with contact surfaces perpendicular to the axial
direction. A picture of the CAD model of the housing can be observed in figure 3a. In order to simplify the
physical structure of the housing the following assumptions are made. The active magnetic bearings (AMB)
actuator geometries are approximated as cylinders. The ball bearings can be described as steel cylinders. The
permanent magnetic bearings (PMB) are made only of aluminum. The bearings are perfectly connected to the
housing parts. The housing can be modeled as one solid part and the mounting holes for the house parts can be
neglected. Based on these assumptions the simplified geometry of the housing can be observed in figure 3b as a
section view.
On the figure the global coordinate system is shown which is aligned with the global coordinate system for the gimbal model. Furthermore, the ball bearing volumes, AMB volumes, and aluminum volumes are designated. The overall material properties for the AMB volumes are estimated by volume fractions of copper and steel from the CAD model. A convergence study has been carried out and the housing FE model consists of 8568 elements with 133023 DOF.

2.3 Rotor

The rotor consists of a steel shaft where aluminum hubs and stacked steel sheet hubs (identical to the AMB stator steel component) are pre-fitted onto. Furthermore, the PMB rotor part is mounted by a bolt. A cross-sectional view of the CAD model of the rotor can be observed in figure 4a. In order to describe the physical structure of the rotor, it is modeled as one solid part and the compressor wheel is neglected. The simplified geometry of the rotor is shown in figure 4b.
PMB rotor part and therefore, the rotor is symmetric around the \((x,y)\)-plane w.r.t. the global coordinate system. A convergence study of the rotor FE model yielded 5816 elements with a total of 74283 DOF to be sufficient.

2.4 Global model

The FE models of the gimbal, housing and rotor are assembled into the global FE model of the FESS. This is carried out by implementing the ball bearings, magnetic bearings, and the steel rods connecting the housing to the gimbal. In order to simplify the inter-part connections, the connection of the gimbal corner part to the surroundings is regarded as clamped, the ball bearings and magnetic bearings are represented by linear springs, and the steel rods are perfectly connected to the housing. Based on these assumptions, the physical FESS is simplified such that it can be represented by the mechanical model illustrated in figure 5. Here the springs in the gimbal corner part holes represent the ball bearings, and the springs between the housing and rotor represent the magnetic bearings.

![Figure 5: Mechanical model of the FESS](image)

The steel rods are modeled using SOLID186 elements, and a suitable mesh is assured by a convergence study. The ball bearings are modeled by linear springs in the radial direction and axial direction. Four connection points in the radial direction are chosen at every 90 degrees. This is carried out with the purpose of "spreading out" the ball bearing effect into the corner parts. Furthermore, this connection is carried out at three points in the axial direction as shown on figure 6.

![Figure 6: Ball bearing spring arrangement](image)

The ball bearing stiffnesses are not available and therefore, to make sure that these do not affect the results, a convergence study has been carried out yielding a stiffness of \(k_{BB} = 1 \times 10^{11} \text{ N/m}\). The ball bearing stiffnesses can then be adjusted to fit experimental results for future aspects. The AMBs are implemented in the same manner as the ball bearings, however, only with one supporting point for each AMB and are only supporting the rotor in the radial direction with a spring stiffness from [5] to be:

\[
k_{r,AMB} = 284/2 \text{ kN/m}
\]  

The PMB is modeled by springs connected between a node in the bottom of the housing model and a node in the PMB part on the rotor model by linear springs with stiffness:

\[
k_{z,PMB} = 15 \text{ kN/m} \quad k_{r,PMB} = -12 \text{ kN/m}
\]  

422
It should be noted that the stiffness obtained for the PMB cannot change. This is not the case for the AMBs, as this
depends on the controller settings. However, fixed stiffnesses for the AMB springs are used in the model.

3 Results

Theoretical natural frequencies and corresponding mode shapes for the gimbal, housing, rotor, and global FESS
model have been obtained and are represented in the following. Furthermore, experimental results are available
for the gimbal and rotor, which the theoretical results are compared to.

3.1 Gimbal

In table 2 the experimental and theoretical obtained natural frequencies for the gimbal and the discrepancies
between the two are listed. The experimental results are obtained from [1].

<table>
<thead>
<tr>
<th>Freq. no.</th>
<th>$f_1$</th>
<th>$f_2$</th>
<th>$f_3$</th>
<th>$f_4$</th>
<th>$f_5$</th>
<th>$f_6$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Experimental [Hz]</td>
<td>193.4</td>
<td>384.9</td>
<td>426</td>
<td>767-778</td>
<td>767-778</td>
<td>~801</td>
</tr>
<tr>
<td>Theoretical [Hz]</td>
<td>195.4</td>
<td>380.5</td>
<td>426</td>
<td>772</td>
<td>775</td>
<td>801</td>
</tr>
<tr>
<td>ERROR [%]</td>
<td>1.03</td>
<td>1.14</td>
<td>0</td>
<td>0.65-0.77</td>
<td>0.39-1.04</td>
<td>0</td>
</tr>
</tbody>
</table>

Table 2: Experimental and theoretical natural frequencies for gimbal

It is observed from the discrepancies that a good agreement between the experimental and theoretical results
are obtained. This indicates that the FE model represents the behaviour of the real physical structure well. The
theoretical obtained mode shapes for the third and fifth natural frequency for the gimbal is shown in figure 7.

![Modal Solution 1](image1)  ![Modal Solution 2](image2)

(a) $f_3 = 426$ Hz  (b) $f_5 = 775$ Hz

Figure 7: Third and fifth theoretical mode shape for gimbal

3.2 Housing

In table 3 the theoretical obtained natural frequencies for the housing are listed. As indicated by the table
experimental results are not available.

<table>
<thead>
<tr>
<th>Frequency</th>
<th>$f_1$</th>
<th>$f_2$</th>
<th>$f_3$</th>
<th>$f_4$</th>
<th>$f_5$</th>
<th>$f_6$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Mode type [Hz]</td>
<td>bend.</td>
<td>bend.</td>
<td>tor.</td>
<td>bend.</td>
<td>bend</td>
<td>long.</td>
</tr>
<tr>
<td>Theoretical [Hz]</td>
<td>2882</td>
<td>2903</td>
<td>3741</td>
<td>4299</td>
<td>4316</td>
<td>4580</td>
</tr>
</tbody>
</table>

Table 3: Theoretical natural frequencies for housing
It is observed that the first natural frequency for the housing is rather large and therefore, the flexibility effects for the housing might not become of interest in the global FE model of the FESS. Due to a lack of experimental results, the housing model was not adjusted further. In figure 8 two selected mode shapes for the housing are illustrated.

\[
\begin{align*}
\text{(a)} & \quad f_1 = 2881.9 \text{ Hz} \\
\text{(b)} & \quad f_3 = 3740.5 \text{ Hz}
\end{align*}
\]

Figure 8: First and third theoretical mode shape for housing

From the figures, it is observed that the mode shapes behave in the same manner as the first two bending modes for a beam in free condition. Furthermore, it is observed that the spacers in the top of housing appear to be the most flexible part of the housing.

### 3.3 Rotor

In table 4 the experimental and theoretical obtained natural frequencies for the rotor and the discrepancies between the two are listed. The experimental results are obtained by frequency response functions (FRF) based on experimental data, where the response of the rotor at different points is measured by a accelerometer under impact from a hammer with the input measured by a force transducer.

From table 4 it is seen that fairly low discrepancies are obtained, however, they are not in same order of magnitude as for the gimbal. Better theoretical results might be obtained if inter-part contact were modeled as well. In order to do so, more detailed experimental results are needed to justify these modifications.

<table>
<thead>
<tr>
<th>Frequency</th>
<th>( f_1 )</th>
<th>( f_2 )</th>
<th>( f_3 )</th>
<th>( f_4 )</th>
<th>( f_5 )</th>
<th>( f_6 )</th>
</tr>
</thead>
<tbody>
<tr>
<td>Experimental [Hz]</td>
<td>1237</td>
<td>1237</td>
<td>3023</td>
<td>3023</td>
<td>4797</td>
<td>4797</td>
</tr>
<tr>
<td>Theoretical [Hz]</td>
<td>1347</td>
<td>1347</td>
<td>3224</td>
<td>3224</td>
<td>3644</td>
<td>6504</td>
</tr>
<tr>
<td>Error [%]</td>
<td>8.81</td>
<td>8.81</td>
<td>4.19</td>
<td>4.19</td>
<td>32.39</td>
<td>35.58</td>
</tr>
</tbody>
</table>

Table 4: Experimental and theoretical natural frequencies for rotor

It should be noted that the FE model of the rotor has mainly been updated according to the first natural frequency, since this frequency were verified by two experiments: based on the before mentioned FRF and under excitation of the rotor using the AMBs. In figure 9 the first two mode shapes for the rotor is shown.
As for the case with the housing, the first two bending mode shapes look very similar to the first two bending modes for a slender beam.

3.4 Global Model

In table 5 the theoretical obtained natural frequencies for the global model of the FESS are listed. It is observed that a total of 23 natural frequencies are occurring in the interval from 0 to 2000 Hz.

<table>
<thead>
<tr>
<th>Freq.</th>
<th>f₁</th>
<th>f₂</th>
<th>f₃</th>
<th>f₄</th>
<th>f₅</th>
<th>f₆</th>
<th>f₇</th>
<th>f₈</th>
<th>f₉</th>
<th>f₁₀</th>
<th>f₁₁</th>
<th>f₁₂</th>
</tr>
</thead>
<tbody>
<tr>
<td>[Hz]</td>
<td>0</td>
<td>0</td>
<td>10.10</td>
<td>26.33</td>
<td>27.79</td>
<td>59.75</td>
<td>64.25</td>
<td>68.51</td>
<td>68.87</td>
<td>111.4</td>
<td>270.8</td>
<td>365.1</td>
</tr>
<tr>
<td>Freq.</td>
<td>f₁₃</td>
<td>f₁₄</td>
<td>f₁₅</td>
<td>f₁₆</td>
<td>f₁₇</td>
<td>f₁₈</td>
<td>f₁₉</td>
<td>f₂₀</td>
<td>f₂₁</td>
<td>f₂₂</td>
<td>f₂₃</td>
<td>-</td>
</tr>
<tr>
<td>[Hz]</td>
<td>504.4</td>
<td>641.8</td>
<td>801.0</td>
<td>1086</td>
<td>1095</td>
<td>1226</td>
<td>1238</td>
<td>1347</td>
<td>1347</td>
<td>1396</td>
<td>1693</td>
<td>-</td>
</tr>
</tbody>
</table>

Table 5: Theoretical natural frequencies for the global model of the FESS in the range of 0-2000 Hz

The first two natural frequencies occur at 0 Hz. The first mode is a rigid body mode for the housing and rotor rotating together about the ball bearing axis. The second mode is the rotor rotating about the centre axis, which is a movement the magnetic bearings allow. In figure 10 a few selected mode shapes are shown.

In general, it is observed that a lot of the mode shapes appearing in the interval 0-2000 Hz are non-rigid body motions, which means that the flexibility effects play a role in the mode shapes. This is especially true for the gimbal, steel rods, and rotor, whereas the housing behaves more as a rigid body. Therefore, a more simple rigid model of the housing might be sufficient in the global model.

The mode shape corresponding to the sixth natural frequency is shown in figure 10a, where the gimbal and housing is vibrating together with the rotor almost standing still. Here the flexibility effects of the bodies do not play a large role in the mode shape. However, the flexibility effects are clearly seen on the 11th mode shape in figure 10c. Here a kind of bending mode for the steel rods is observed where the housing is twisted around the global z-axis. This mode clearly illustrates a scenario of coupled vibrations where energy is exchanged between the housing and gimbal. As illustrated by this mode shape the steel rods tends to be the most flexible part in the FESS. Furthermore, a strong coupling between gimbal and housing is often seen in the mode shapes.
Figure 10: Selected theoretical mode shapes for global model of the FESS

(a) $f_6 = 59.75$ Hz

(b) $f_{10} = 111.4$ Hz

(c) $f_{11} = 270.8$ Hz

(d) $f_{14} = 641.8$ Hz

(e) $f_{17} = 1095$ Hz

(f) $f_{20} = 1347$ Hz
In general, the coupling between the housing and rotor is weak. As an exception, mode shape 10 shows coupled vibrations between the housing and rotor. In contrast to this, mode shape 20 serves as a great example of decoupled vibrations. Here the rotor is solely vibrating alone with a natural frequency and mode shape corresponding to the first natural frequency and mode shape of the rotor itself in free condition as shown on figure 9a. "Isolated" mode shapes are not only observed for the rotor only but also the gimbal. This is seen in the 14th and 17th mode shape where only the gimbal is vibrating. These two modes look similar to the third and fifth mode shape for the gimbal model shown in figure 7. However, the frequencies are a lot higher which indicates that the vibrations are still affected by the coupling to the other parts and the implementation of the BC.

Further details on the results and illustrations of all the mode shapes can be found in [6]. Finally, it should be mentioned that in order to verify the mode shapes and natural frequencies obtained by the FE models, a experimental modal analysis should be carried out.

4 Conclusion and Future Aspects

Theoretical natural frequencies and mode shapes have been obtained for the gimbal, housing, and rotor individually. Six natural frequencies have been identified in the range of 0 to 800 Hz for the gimbal with the first natural frequency occurring at 193.4 Hz. Furthermore, a maximum discrepancy of 1.14% between the theoretical and experimental natural frequencies were obtained, which indicates that the FE model of the gimbal represents the physical structure well.

A FE model of the housing has been developed and six theoretical natural frequencies have been identified in the range of 0-4580 Hz with the first natural frequency occurring at 2882 Hz. However, the flexibility effects of the housing do not appear to be relevant in the interval from 0 to 2000 Hz in the global model due to the high natural frequencies for the housing. The FE model of the rotor indicates the first natural frequency to occur at 1347 Hz, which yields a discrepancy of 8.81% compared to the experimental obtained first natural frequency. However, further adjustments of the rotor were not carried out due to lack of experimental data.

The theoretical results for the global FE model of the FESS indicate that the flexibility effects have an influence on the dynamics of the FESS, since some of the components are not behaving as rigid bodies in the mode shapes. Furthermore, coupled vibrations also occur between the flexible bodies for which the AMBs must be properly designed taken these into account.

A FE model of the FESS has been designed, however, the FE model should be validated according to experimental results. Therefore, experimental modal analyses (EMAs) should be carried out to determine the natural frequencies and mode shapes experimentally. It is suggested that an EMA is carried out for the gimbal, housing, and rotor separately so the components in the FESS can be validated individually and an EMA of the FESS for validating the global model of the FESS.

Furthermore, aspects such as reducing the model in terms of number of DOF could be of relevance in order to simplify the model. Finally, gyroscopic forces could be introduced by simulating the FESS under operational conditions, that is, by giving the flywheel/rotor an angular velocity. This will affect the natural frequencies and mode shapes obtained for the system. The FE model of the FESS under operational conditions should be validated by experimental results as well. Therefore, a operational modal analysis should be carried out.

REFERENCES


