Noise and Vibration
Investigation of
Composite Shaft

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Abstract:
Experimental and numerical analyses of a composite tubular shaft in order to investigate its sound radiation properties for a vibratory torque excitation were performed. Experimental and numerical analyses in order to establish material properties of the laminate were performed.

Keywords:
Composite material, Composite Shaft, Propulsion Shaft, Macromechanics, Laminate theory, Experimental analysis, FEM, BEM, Vibration, Sound radiation.
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Karlskrona, December 1998

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1. Notation

\begin{align*}
A & \quad \text{Extensional stiffness matrix} \\
E & \quad \text{Young’s modulus} \\
G & \quad \text{Shear modulus} \\
i,j & \quad \text{Number} \\
k & \quad \text{Number of the ply} \\
N & \quad \text{Number of plies} \\
Q & \quad \text{Stiffness matrix} \\
\bar{Q} & \quad \text{Transformed reduced stiffness matrix} \\
t & \quad \text{Thickness} \\
T_k & \quad \text{Transformation matrix} \\
Z & \quad \text{Thickness of laminat} \\
\alpha & \quad \text{Fibre orientation in global coordinate system} \\
\gamma & \quad \text{Shear strain} \\
\epsilon & \quad \text{Strain} \\
\theta & \quad \text{Angle in cylindrical coordinate system} \\
\nu & \quad \text{Poisson’s ratio} \\
\rho & \quad \text{Density} \\
\sigma & \quad \text{Stress} \\
\tau & \quad \text{Shear stress} \\
\end{align*}

Indices

\begin{align*}
k & \quad \text{Number of the ply} \\
T & \quad \text{Transpose} \\
\end{align*}
\(x, y, z\) \hspace{1em} x, y and z direction in global coordinate system

\(1, 2, 3\) \hspace{1em} 1, 2 and 3 direction in local coordinate system
2. Introduction

When composite shaft propulsion of ships came into more common use, it was noticed that uncharacteristic noises were audible from the shaft line and the propellers. For the specific ship studied in this work, this phenomenon is experienced in the speed range of 650 - 950 rpm at zero pitch. Above 950 rpm, or when the pitch is above 40%, the noise do not occur. It is also noticed that the noise is more severe on the starboard side compared to the port side.

The aim of this work is to simulate the sound radiation of the composite propulsion shaft when excited by a vibratory torque, and give suggestions of suitable methods of removing the unwanted noise from the shaft.

This involve a theoretical investigation of the sound radiation characteristics of the composite shaft and an experimental investigation in order to verify the theoretical model.

To model the composite material the work also includes a theoretical and experimental study of a test specimen, cut out from the composite shaft.
3. Composite Shaft Propulsion

The propulsion is delivered by four engines, two engines per shaft line. Two stages, a belt pulley stage and a worm gear stage makes the speed reduction. Figure 3.1 shows the propulsion arrangement.

![Figure 3.1. Propulsion arrangement.](image)

The engines are 4-stroke high-speed diesel engines in V-8 configuration. The speed range is 650-1800 rpm. The belt ratio is 2.018. The propeller is a cycloidal 5-blade controllable pitch propeller with an incorporated worm gear. The worm gear ratio is 7.625. Figure 3.2 shows the dimensions of the composite shaft.

![Figure 3.2. Dimensions of composite shaft (measures in mm).](image)
4. Composite Shaft Material

The material in the shaft is a laminated composite, which means that it consists of several layers of composite. The composite consists of a matrix and a fibre, which is epoxy (matrix) and glass (fibre). Each layer is referred to as a ply.

4.1 Laminate Build-up

The laminate consists of forty-four layers of antisymmetric plies, which are orientated in \( \alpha = \pm 30^\circ \) relative to the \( x \)-axis according to figure 4.1.

![Figure 4.1. Part view of stack sequence and definition of global coordinate system (x, y, z) and local coordinate system (1, 2, 3).](image-url)
Each ply is built-up by glass reinforced epoxy, and has a thickness of 0.25 mm. A filament-winding process achieves the antisymmetric build-up. In this process the glass-fibre reinforcement is fed through an epoxy resin bath and then wound on a mandrel according to figure 4.2.

![Figure 4.2. Filament-winding process.](image)

**4.2 Laminate theory**

By knowing the micromechanical property of the ply, the property of the laminate is derived by using the laminate macromechanic theory, according to reference [1].

**4.2.1 Material properties of the laminate**

As mentioned before, in the manufacturing of composite laminates several plies are assembled on top of each other but with different orientation to the global coordinate system. Thus, in order to describe the behaviour of the laminate one must know the behaviour of each ply described in the global coordinate system \((x, y, z)\) and not in its local coordinate system \((1,2,3)\). The stiffness of the ply in the global coordinate system is found by using a
transformation matrix. Assuming that the local coordinate system is rotated an angle $\alpha$ to the global coordinate system, (see figure 4.1), the transformation matrix is

$$
T_k = \begin{bmatrix}
\cos^2 \alpha & \sin^2 \alpha & 2\sin \alpha \cos \alpha \\
\sin^2 \alpha & \cos^2 \alpha & -2\sin \alpha \cos \alpha \\
-\sin \alpha \cos \alpha & \sin \alpha \cos \alpha & (\cos^2 \alpha - \sin^2 \alpha)
\end{bmatrix} \tag{4.1}
$$

It should be noticed that transformation has nothing to do with material properties but merely a rotation of stress.

The relation between stress and strain for an orthotropic material can be described as a generalised Hooke’s law. For plane stress we can write

$$
\begin{pmatrix}
\tilde{\sigma}_1 \\
\tilde{\sigma}_2 \\
\tilde{\sigma}_{12}
\end{pmatrix} =
\begin{bmatrix}
Q_{11} & Q_{12} & Q_{16} \\
Q_{12} & Q_{22} & Q_{26} \\
Q_{16} & Q_{26} & Q_{66}
\end{bmatrix}
\begin{pmatrix}
\tilde{\epsilon}_1 \\
\tilde{\epsilon}_2 \\
\tilde{\epsilon}_{12}
\end{pmatrix} \tag{4.2}
$$

where $Q_{ij}$, the so-called reduced stiffness, are

$$
Q_{11} = \frac{E_1}{1 - v_{12} \cdot v_{21}} \tag{4.3}
$$

$$
Q_{12} = \frac{v_{12} \cdot E_2}{1 - v_{12} \cdot v_{21}} = \frac{v_{21} \cdot E_1}{1 - v_{12} \cdot v_{21}} \tag{4.4}
$$

$$
Q_{22} = \frac{E_2}{1 - v_{12} \cdot v_{21}} \tag{4.5}
$$

$$
Q_{66} = G_{12} \tag{4.6}
$$

To be able to use the reduced stiffness matrix the following reciprocal relation have to be considered

$$
\frac{v_{12}}{E_1} = \frac{v_{21}}{E_2} \tag{4.7}
$$
By using the abbreviation

\[
\bar{\mathbf{Q}} = \left[ T_k \right]^{-1} \cdot \mathbf{Q} \cdot \left[ T_k \right]^{-1}^T
\]  \hspace{1cm} (4.8)

we get the stress-strain relations in \( xy \)-coordinates, where the bar over the \( \bar{Q}_{ij} \) matrix denotes that we are dealing with the transformed reduced stiffness instead of the reduced stiffness, \( Q_{ij} \).

By using (see figure 4.3)

\[
A_{ij} = \sum_{k=1}^{N} (\bar{Q}_{ij})_k (z_k - z_{k-1})
\]  \hspace{1cm} (4.9)

, which is the so called extensional stiffness matrix,

\[\text{Figure 4.3. Cross-section of laminate and relations for computing the extensional stiffness matrix.}\]

we get \( E_x, E_y \) and \( G_{xy} \) for the laminate as follows

\[
\nu_{21} = \frac{A_{12}}{A_{11}}
\]  \hspace{1cm} (4.10)
and

\[ v_{12} = \frac{A_{12}}{A_{22}} \]  

(4.11)

then

\[ E_x = \frac{A_{11}(1-v_{12}v_{21})}{t} \]  

(4.12)

\[ E_y = \frac{A_{22}(1-v_{12}v_{21})}{t} \]  

(4.13)

\[ G_{xy} = \frac{A_{66}}{t} \]  

(4.14)

The following properties for the ply

- \( E_1 = 38.6 \) Gpa
- \( E_2 = 8.27 \) GPa
- \( G_{12} = 4.14 \) GPa
- \( v_{12} = 0.26 \)
- \( \alpha = \pm 30^\circ \)
- \( t = 11 \) mm
- \( N = 44 \)

gives

- \( E_x = 21.837 \) GPa
- \( E_y = 9.147 \) GPa
- \( G_{xy} = 9.134 \) GPa
- \( v_{xy} = 0.27 \)
- \( v_{yx} = 0.65 \)

for the laminate.

All calculations are performed in MatLab.
4.3 Tensile test of $E_x$

To confirm that the theoretical material properties, given for the ply and calculated for the laminate are correct, tensile tests of test specimens are made. The test specimens are cut out from a shaft with the same properties as the one in the ship. This tests were carried out by Kockums AB Karlskronavarvet. The tests were only made for $E_x$ due to the difficulty of fastening the test specimens in the tensile testing machine for the $E_y$- and $E_z$- direction.

4.3.1 Experimental Procedure

The test specimen was fastened in a tensile test machine. On the test specimen a strain gauge was applied, this was done in such way that only the strain in the $E_x$-direction of the specimen was registered by the gauge. The machine exerted a tensile force to the test specimen and the strain was simultaneously measured by the strain gauge. The test continued until the specimen snapped. The test was carried out with two different test specimens.

4.3.2 Results of Tensile Test

The tensile tests gave $E_x = 24.2$ GPa in comparison to $E_x = 21.8$ GPa for the theoretical model. That is, $E_x$ for the test specimens was approximately 11% stiffer than calculated. Complete results of the tests are found in appendix A.
5. Acoustic Theories

5.1 Sound in structures

When a structure is vibrating, sound radiation may occur if the structure can impart kinetic energy to the surrounding fluid. The energy takes form as pressure waves, i.e. sound, from the vibrating structure. The structure's ability to couple to the fluid decides the amount of energy that is radiated. As long as the speed of the propagating waves are smaller than the speed of sounds in all directions of wave propagation in the fluid there is no sound radiation. On the other hand, if the speed of the propagating waves is higher than the speed of sound, waves will propagate out in the fluid. This can be heard as a sound. Internal damping will decrease the amplitude of the vibration and in some way lower the sound level [2] [3].

5.2 Wave types

There are several wave types in solids but the most important type of waves in a thin tubular cylinder is flexural waves. Flexural waves and bending waves are the same. Figure 5.1 shows a bending wave.

Figure 5.1. Bending wave.

These waves are of great importance in sound radiation studies because they have very good connection to the surrounding fluid. Other wave types in solids are quasi-longitudinal and transverse shear waves but these waves do not have equally good connection to the surrounding fluid. They can still...
radiate sound but flexural waves radiate more easily, according to for example reference [2].

5.3 Boundary Element Method

For studying sound pressure in the fluid the Boundary Element Method is used, as the numerical method. The boundary between the structure and the fluid is divided into a number of elements. The pressure and velocity distributions over these elements are approximated by assumed shape functions and unknown variables. A system of equations is determined, where the different elements and the unknown variables interact with each other acoustically. Furthermore, an additional boundary condition must be applied; the fluid-structure interface to describe the behaviour of the structure. This can be done by prescribing pressure values, velocity values or acoustic impedance, or by describing the structure by the Finite Element Method. The system of equations is solved for the unknowns with present structural loads and existing acoustical sources in the fluid. Thereafter it is possible to explicitly calculate the acoustical properties in the fluid. This and a further discussion are found in reference [4].
6. Introductory Experiment

In this test a comparison was made of natural frequencies between an experimental and a numerical modal analysis of a test specimen. This was made in order to get acquainted with the Finite Element software I-DEAS and its laminate module, and to verify the material properties of the laminate.

6.1 Experimental Analysis of Test Specimen

In this experiment the natural frequencies of a test specimen was measured. The test specimen was cut out from a shaft with similar properties as the shaft of the ship.

6.1.1 Experimental Equipment

The measurement was performed using a Dynamic Signal Analyser of type Hewlett Packard System HP-3567A, an accelerometer and a modal hammer.

6.1.2 Test Specimen

The test specimen was cut out from a tubular shaft with similar properties as the ones in the ship. It was cut in such a manner, that it could be seen as a slim beam. Figure 6.1 shows the measures of the specimen.
6.1.3 Experimental set-up

Figure 6.2 shows a principal sketch of the measurement set-up. The modal hammer excites the specimen and the accelerometer measures the acceleration of the surface. Weak suspension strings support the specimen, which was done to ensure that no restraints were applied to the specimen.
Figure 6.2. Principal sketch of experimental set-up.

The modal hammer and the accelerometer were connected to the HP-system. The HP-system collects the measured signals and was set to calculate and display natural frequencies, transfer functions and phase angles.

6.1.4 Measurement Procedure

In this experiment the natural frequencies, mode shapes and damping ratios were determined for the first five natural bending modes in the y-direction of the test specimen. To get a comprehension of the mode shapes, the acceleration was measured in five different points along the centre of the specimen, according to figure 6.3.

Figure 6.3. Five measuring points along the x-axis.
The accelerometer was attached in point five and the modal hammer was used to excite the specimen at point 1-4. Each point was excited three times and an average of the signals was used.

6.2 Numerical Analysis of Test Specimen

I-DEAS Master Series 5 is used to numerically calculate the natural frequencies and mode shapes for the first five bending modes in the $y$-direction of the test specimen.

6.2.1 Building the Laminate

The laminate model is built in I-DEAS Laminate Module. The laminate model is built by first defining the ply properties and then defining the thickness of the ply and its angle in the laminate. Finally the number of plies in the laminate is specified. Then I-DEAS computes the properties of the laminate. The following material properties are used for the ply

\[
\begin{align*}
\rho &= 2100 \text{ kg/m}^3 \\
E_1 &= 44 \text{ GPa} \\
E_2 &= E_3 = 11 \text{ GPa} \\
G_{12} &= G_{23} = G_{13} = 5 \text{ GPa} \\
\nu_{12} &= \nu_{23} = \nu_{13} = 0.26 \\
\alpha &= \pm30^\circ \\
t &= 11 \text{ mm} \\
N &= 44
\end{align*}
\]

6.2.2 Finite Element Model

The specimen is modelled with 20 mm eight-node parabolic quadrilateral shell elements. The elements are assigned the same properties and thickness as the test specimen. Figure 6.4 shows the mesh of the specimen.
The ten first natural modes of the specimen are computed, then the result is exported to I-DEAS Model Response. In Model Response an excitation set is created and the specimen is excited in the same way as in the experimental set-up. The damping ratios from the experimental test are assigned to the numerically computed mode shapes. Then the five first bending modes, in the $y$-direction, are examined.

### 6.3 Results

The experimental natural frequencies and damping ratios for the five first bending modes in the $y$-direction of the test specimen are shown in table 6.1.

<table>
<thead>
<tr>
<th>Mode</th>
<th>Frequency [Hz]</th>
<th>Damping Ratio</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>79</td>
<td>0.015</td>
</tr>
<tr>
<td>2</td>
<td>218</td>
<td>0.006</td>
</tr>
<tr>
<td>3</td>
<td>435</td>
<td>0.005</td>
</tr>
<tr>
<td>4</td>
<td>717</td>
<td>0.005</td>
</tr>
<tr>
<td>5</td>
<td>1076</td>
<td>0.005</td>
</tr>
</tbody>
</table>
The numerically calculated natural frequencies for the five first modes in the y-direction of the test specimen are shown in table 6.2.

Table 6.2. Numerically calculated natural frequencies of the test specimen.

<table>
<thead>
<tr>
<th>Mode</th>
<th>Frequency [Hz]</th>
<th>Damping Ratio</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>81</td>
<td>0.015</td>
</tr>
<tr>
<td>2</td>
<td>220</td>
<td>0.006</td>
</tr>
<tr>
<td>3</td>
<td>429</td>
<td>0.005</td>
</tr>
<tr>
<td>4</td>
<td>717</td>
<td>0.005</td>
</tr>
<tr>
<td>5</td>
<td>1051</td>
<td>0.005</td>
</tr>
</tbody>
</table>

Figure 6.5-6.7 shows a comparison between experimental and numerical natural frequencies and transfer functions for measuring point 1-3 of the test specimen. Due to symmetry, only points 1-3 are displayed.

Figure 6.5. Natural frequencies and transfer function for measuring point 1.
Figure 6.6. Natural frequencies and transfer function for measuring point 2.

Figure 6.7. Natural frequencies and transfer function for measuring point 3.
The experimental and the numerical natural frequencies correlate well, which figure 6.8 illustrates. The transfer functions are satisfactory for the first three modes, but for the last two modes there are quite big differences between the experimental and numerical values. Since the analysis of the shaft will be performed in the interval 0-520 Hz, the differences of the two last modes can be disregarded.

![Figure 6.8. Comparisons between experimental and numerical natural frequencies.](image_url)
7. Structural Dynamic Full-scale Measurements

Several full-scale measurements have been performed by Kockums AB Karlskronavarvet on the shaft line. The measured data are sound pressure, vibratory torque and vibration of the shaft line bearings at different speeds (650-1000 rpm).

7.1 Measured Signals

The signals needed for further analysis is sound pressure and vibratory torque. Figure 7.1 and 7.2 show measured time signals of sound pressure and vibratory torque for 805 rpm and zero pitch. It should be noticed that the y-axis is displayed without any unit.

![Figure 7.1. Torque at 805 rpm and zero pitch.](image-url)
7.2 Processing of Measured Signals

To be able to use the measured signals, a FFT is performed. This is done to convert the signals from time domain to frequency domain. To get a good understanding of the frequency contents, waterfall analyses are performed for the signals. These analyses are displayed in Figure 7.3 and 7.4. It should be noticed that the y-axis is displayed without any unit.

*Figure 7.2. Sound pressure at 805 rpm and zero pitch.*
Figure 7.3. Waterfall analysis of measured torque.

Figure 7.4. Waterfall analysis of measured sound pressure.
The frequency peaks in the waterfall analysis that make up vertical lines are the natural frequencies of the system. The peaks that makes up inclined lines defines the revolution dependent frequencies. As can be seen in figure 7.3 and 7.4 there is a correlation between the torque and sound pressure. It should be noticed that there is a relation at 100 Hz, but on the other hand there is no relation at 200 Hz.

7.3 Excitation Frequencies

There are several excitations, such as engine revolution, engine-firing rate, shaft revolution, propeller unit revolution, propeller blade frequency and worm gear frequency. Their different frequencies are presented in table 7.1.

Table 7.1. Different frequencies that may excite the shaft. The frequencies are calculated for the minimum and maximum revolutions.

<table>
<thead>
<tr>
<th>Excitation</th>
<th>650 rpm</th>
<th>1000 rpm</th>
</tr>
</thead>
<tbody>
<tr>
<td>Engine rev.</td>
<td>$f = \frac{n}{60}$</td>
<td>10.8 Hz</td>
</tr>
<tr>
<td>Engine firing rate</td>
<td>$f = \frac{n}{60} \cdot 4$</td>
<td>43.3 Hz</td>
</tr>
<tr>
<td>Cylinder firing rate</td>
<td>$f = \frac{n}{60} \cdot \frac{1}{2}$</td>
<td>5.4 Hz</td>
</tr>
<tr>
<td>Shaft revolution</td>
<td>$f = \frac{n}{60} \cdot \frac{1}{2.018}$</td>
<td>5.4 Hz</td>
</tr>
<tr>
<td>Propeller unit revolution</td>
<td>$f = \frac{n}{60} \cdot \frac{1}{2.018} \cdot \frac{1}{7.625}$</td>
<td>0.7 Hz</td>
</tr>
<tr>
<td>Propeller blade frequency</td>
<td>$f = \frac{n}{60} \cdot \frac{1}{2.018} \cdot \frac{1}{7.625} \cdot 5$</td>
<td>3.5 Hz</td>
</tr>
<tr>
<td>Worm gear freq.</td>
<td>$f = \frac{n}{60} \cdot \frac{1}{2.018} \cdot \frac{1}{7.625} \cdot 61$</td>
<td>43.0 Hz</td>
</tr>
</tbody>
</table>

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8. Finite Element Model

The model is built in I-DEAS Master Modeler, the meshing is done in I-DEAS Meshing and finally the FE-model is solved in I-DEAS Model Solution. For further description see reference [5]. The shaft consists of three different parts, a cylindrical shell and two caps.

8.1 Shell

To be able to use the laminate created in I-DEAS Laminate module it is necessary to use eight-node parabolic quadrilateral thin shell elements for the laminated shell. The element length is 105 mm, which is the largest length possible to use with maintained convergence. The shell is assigned a thickness of 11 mm, which is the thickness of the laminate in the shaft. The laminate is built in the same manner as described earlier and with the same material properties as in section 6.2.1. The outer diameter is 212 mm and the length is 3540 mm. Figure 8.1 shows the meshed shaft.

Figure 8.1. The meshed shaft.
8.2 Cap

Both eight-node parabolic quadrilateral thin shell elements and ten-node parabolic tetrahedron solid elements make up the cap. The cap is built in three different parts that are meshed separately and then merged together. Measures of the cap are shown in figure 8.2. Figure 8.3 shows the FE-model of the cap.

![Figure 8.2. The end caps (measures in mm).](image)
Finally the shaft and caps are merged together. Figure 8.4 shows the assembled shaft.

8.3 Assembled shaft

Finally the shaft and caps are merged together. Figure 8.4 shows the assembled shaft.
9. Solving the Model in I-DEAS

For solving the problem several steps are performed. The steps are presented in figure 9.1. For a further description of the procedure see references [5] and [6].

![Figure 9.1. The steps for calculating the model.](image)
9.1 Boundary Conditions

After the model is meshed, the boundary conditions are applied to the model. The shaft is clamped at one end and at the other end the shaft must be able to rotate around its own axis. Applying a cylindrical coordinate system makes it possible to clamp one end and apply a torque at the other. The shaft is free in $\theta$ and fixed in the other degrees of freedom in the cylindrical coordinate system according to figure 9.2. The definition of clamped is that it is fixed in all degrees of freedom.

![Figure 9.2. The shaft is clamped at one end and the other end is free in $\theta$ and fixed in the other degrees of freedom in a cylindrical coordinates system.](image)

9.2 Model Solution

The natural frequencies and mode shapes for the shaft are calculated numerically in Model Solution.
9.3 Model Response

The shaft is excited in Model Response by four forces at the end where rotation is allowed. The forces create a torque in the shaft. The results from model response are mode shapes created by the excitation. The mode shapes are exported to Vibro-Acoustics. All the forces have the same size but different directions. The forces and their directions are presented in figure 9.3. The damping ratio is set to 0.008 in all modes.

![Figure 9.3. Excitation with four forces.](image)

9.4 Vibro-Acoustics

Vibro-Acoustics is the module in I-DEAS that calculates the sound pressure in the fluid. Vibro-Acoustics uses the Boundary Element Method for calculating sound pressure in the surrounding fluid. The sound pressure in the fluid is presented in domain points. The domain points surround the BE-model and are placed 1 m from the shaft in a circle according to figure 9.4. The acoustic loads in Vibro-Acoustics are the mode shapes that are calculated in Model Response.
9.5 MatLab

The result in I-DEAS is exported to Universal Files. The Universal Files are then converted to MatLab files. Then MatLab is used to analyse the results.
10. Results and Discussion

The analyses include numerical calculations of natural frequencies, mode shapes and sound pressures. Two shafts are analysed, the tubular shaft of the ship and a stiffer shaft with the same measures. Three different analyses are performed; two for the tubular shaft of the ship and one for a stiffer shaft. The waterfall analyses show that there is no sound radiation above 500 Hz, therefore the upper limit of the analysis is set to 515 Hz.

10.1 Results of computation

The natural frequencies are calculated between 1 and 1000 Hz, because although the report is limited to 515 Hz, mode shapes above 515 Hz may still affect the mode shapes from the excitation range anyway. The natural frequencies are presented in table 10.1. The mode shapes can also be viewed in appendix B.

<table>
<thead>
<tr>
<th>Mode</th>
<th>Natural Frequency [Hz]</th>
<th>Modeshape</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>51.3</td>
<td>Bending</td>
</tr>
<tr>
<td>2</td>
<td>134.6</td>
<td>Bending</td>
</tr>
<tr>
<td>3</td>
<td>139.8</td>
<td>Breathing</td>
</tr>
<tr>
<td>4</td>
<td>249.9</td>
<td>Bending</td>
</tr>
<tr>
<td>5</td>
<td>361.8</td>
<td>Bending</td>
</tr>
<tr>
<td>6</td>
<td>388.9</td>
<td>Bending</td>
</tr>
<tr>
<td>7</td>
<td>432.9</td>
<td>Breathing</td>
</tr>
<tr>
<td>8</td>
<td>511.5</td>
<td>Membrane</td>
</tr>
</tbody>
</table>

Table 10.1. Natural Frequencies.
The FE-model is excited with four forces according to figure 9.3. Each force is 2000 N and constant from 1 to 520 Hz.

Table 10.2 shows the frequency and amplitude of the sound pressure.

*Table 10.2. Sound pressure with constant force.*

<table>
<thead>
<tr>
<th>Mode</th>
<th>Frequency [Hz]</th>
<th>Sound Pressure [dB]</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>50</td>
<td>74</td>
</tr>
<tr>
<td>2</td>
<td>136</td>
<td>79</td>
</tr>
<tr>
<td>4</td>
<td>248</td>
<td>95</td>
</tr>
<tr>
<td>6</td>
<td>388</td>
<td>99</td>
</tr>
</tbody>
</table>

The only natural mode shapes that give sound are the bending and breathing modes. The four peaks in figure 10.1 are only bending modes at natural frequencies. The sound pressure gets higher as the natural frequency increases. Some natural frequencies tend to radiate more sound than others do.

*Figure 10.1. Sound Pressure Level.*
10.2 Results of Computation with Excitation by a Measured signal

The FE-model is also excited with a measured signal. The signal is a vibratory torque, the signal is transformed to force to make excitation possible. The signal is presented in figure 10.2. The signal is recorded in the time domain and is transformed to the frequency domain by using FFT in MatLab.

![Figure 10.2. Exciting force.](image)

The shaft radiates sound from several frequencies when the measured vibratory torque excites it. It should also be noticed that the numerically calculated natural frequencies might differ from the natural frequencies in the shaft. This is due to a stiffer suspension in the FE-model than in the ship. This will alter the sound pressure when the measured torque excites the shaft. The calculated sound pressures are presented in figure 10.3 and table 10.3. The peak of the sound pressures tallies very well in frequency with the natural frequencies.
Figure 10.3. Sound Pressure.

Table 10.3. Sound pressure with measured exciting force.

<table>
<thead>
<tr>
<th>Mode</th>
<th>Frequency [Hz]</th>
<th>Sound Pressure [dB]</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>50</td>
<td>71</td>
</tr>
<tr>
<td>2</td>
<td>138</td>
<td>75</td>
</tr>
<tr>
<td>4</td>
<td>248</td>
<td>100</td>
</tr>
<tr>
<td>6</td>
<td>387</td>
<td>92</td>
</tr>
</tbody>
</table>
10.3 Results of computation of a stiffer shaft

The sound radiation of a stiffer shaft is also analysed. This shaft has a laminate thickness of 15 mm. When the laminate gets thicker the stiffness increases, which also makes the natural frequencies increase. Table 10.4 presents the natural frequencies of the stiffer shaft. The shaft is built and analysed in the same way as the 11 mm shaft. The purpose of analysing a stiffer shaft is to investigate if the sound radiation will decrease. The natural frequency is calculated between 1 and 1000 Hz. As before only frequencies below 515 Hz will be considered. All mode shapes are assigned a damping ratio of 0.008. Table 10.4 shows the natural frequencies of the stiffer shaft.

<table>
<thead>
<tr>
<th>Mode</th>
<th>Natural Frequency [Hz]</th>
<th>Mode Shape</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>49.4</td>
<td>Bending</td>
</tr>
<tr>
<td>2</td>
<td>129.8</td>
<td>Bending</td>
</tr>
<tr>
<td>3</td>
<td>144.2</td>
<td>Breathing</td>
</tr>
<tr>
<td>4</td>
<td>241.6</td>
<td>Bending</td>
</tr>
<tr>
<td>5</td>
<td>355.5</td>
<td>Bending</td>
</tr>
<tr>
<td>6</td>
<td>378</td>
<td>Bending</td>
</tr>
<tr>
<td>7</td>
<td>439.4</td>
<td>Breathing</td>
</tr>
</tbody>
</table>

Four constant forces of 2000 N each, in the frequency range 1 to 530 Hz, excite the shaft. The forces are applied to the shaft according to figure 9.3. The only modes that create sound are the natural breathing and bending modes, as for the tubular shaft in the ship, according table 10.5 and figure 10.4.
Table 10.5 Sound pressure.

<table>
<thead>
<tr>
<th>Mode</th>
<th>Frequency [Hz]</th>
<th>Sound Pressure [dB]</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>46</td>
<td>76</td>
</tr>
<tr>
<td>2</td>
<td>126</td>
<td>73.5</td>
</tr>
<tr>
<td>3</td>
<td>233</td>
<td>99</td>
</tr>
<tr>
<td>4</td>
<td>366</td>
<td>91.5</td>
</tr>
<tr>
<td>5</td>
<td>521</td>
<td>100</td>
</tr>
</tbody>
</table>

Figure 10.4. Sound Pressure.
10.4 Discussion of Computed results

In all three analyses there are audible sound pressures for a human ear. Some of the peaks in the figures 10.1, 10.3 and 10.4 radiate very high sound pressures. Some of them exceed the limit of impairment of hearing. One should remember that the damping ratio is constant. This is not really accurate, because the damping ratio is frequency dependent. The sound pressure will alter with the damping ratio. Low damping ratios will give higher sound pressures and high damping ratios will give lower sound pressures. Making the shaft slightly stiffer showed no decrease in sound pressure. The sound pressure is almost the same as for the shaft in the ship. The only thing that is accomplished is that the natural frequencies are altered. There is a need for further investigations.

The sound level and the amount of excitation depend on each other. According to figure 10.5 and 10.6 there is a resemblance between the measured and the numerically calculated sound pressure. The measured and the calculated sound should theoretically be a match, but due to the simplification of the numerical model the natural frequencies differ.

![Figure 10.5. Sound Pressure from fullscale measurement.](image)
Figure 10.6. Sound Pressure from numerical fullscale excitation
11. Further Research

There are some ideas that can be performed to remove or decrease the noise from the shaft line.

11.1 Stiffer Shaft

If the shaft is made stiffer the natural frequencies will be altered. The stiffness of the shaft must increase so that the natural frequencies are moved above the current excitation frequencies. The analysis of the stiffer shaft shows that a small change of the thickness is not sufficient.

By altering materials, or widely increase the thickness of the composite shaft, adequate stiffness would be obtained. Installing metal bars inside the shaft could also increase the stiffness.

11.2 Shaft Damping

The natural breathing and bending modes tend to radiate sound in the analyses. These modes can be damped if metal rings are placed where the modes have their largest amplitude. An example is shown in figure 11.1.

*Figure 11.1. Shaft with metal rings for damping the bending modes.*
The dimensions, number of rings and where they are placed must be analysed. This report is not covering this matter.

11.3 Excitation Damping

When changing the gear ratio of the worm gear and the belt pulley stage, the excitation frequencies are altered.
12. Conclusions

The aim of this work was to find out if it was possible to simulate the sound radiation of a composite propulsion shaft excited by a vibratory torque. The calculated result was compared to the measured. Suggestions of suitable methods of removing the unwanted noise from the shaft were also given.

The study was performed in the frequency range 1 to 500 Hz. The reason to only investigate the frequencies within this range was that the results from tests have shown that there was no sound radiation outside these frequencies. Material properties of the laminated shaft were also established through analytical, experimental and numerical tests. The results of these tests were corresponding satisfactorily.

The BEM in I-DEAS Vibro-Acoustic performed the sound calculations. The results were presented as frequency spectra versus radiated sound pressure.

The FE-model was probably stiffer than the shaft in the ship. This was due to the boundary conditions, and the thin shell elements.

The damping ratio of the shaft was set to 0.008. This affects the amplitude of the sound radiation and was a possible source of error.

One important conclusion of this work was that only natural bending and breathing modes tend to radiate sound. This was true for both the shaft in the ship with 11 mm laminate thickness and the stiffer shaft with 15 mm laminate thickness.

The stiffer and the softer shaft give about the same sound pressure in the surrounding gas.

The numerical calculation shows that the shafts radiate sound that was audible for a human ear.
13. References


5. SDRC, “I-DEAS On-Line Smart-View”, I-DEAS Master Series 5.0, Structural Dynamics Research Corporation

14. Appendices

A. Tensile Test of $E_x$

Test type: Tensile
Operator name: Hans Forsman
Sample Identification: D73327T
Interface Type: 4200 Series
Machine Parameters of test:
- Sample Rate (pts/sec): 10.000
- Crosshead Speed (mm/min): 2.000
- Extensometer switch value: 4.0000% offset

Laminatuppløsning Fiberlindet ±30 grader
Matris Epoxi
Efterhårdning Förfäring av vax
Överlåt Provbi från en drivaxel Aplied Comp.

Dimensions:

<table>
<thead>
<tr>
<th></th>
<th>Spec. 1</th>
<th>Spec. 2</th>
</tr>
</thead>
<tbody>
<tr>
<td>Width (mm)</td>
<td>11.900</td>
<td>20.330</td>
</tr>
<tr>
<td>Thickness (mm)</td>
<td>10.140</td>
<td>9.8900</td>
</tr>
<tr>
<td>Ext. gauge len (mm)</td>
<td>25.000</td>
<td>25.000</td>
</tr>
<tr>
<td>Spec. gauge len (mm)</td>
<td>100.00</td>
<td>100.00</td>
</tr>
</tbody>
</table>

Out of 2 specimens, 0 excluded.
Sample comments: Bitarna slipade till rektangulär form

<table>
<thead>
<tr>
<th>Specimen Number</th>
<th>Load at Max. Load</th>
<th>Stress at Max. Load</th>
<th>% Strain</th>
<th>Modulus between (0.2-0.5%)</th>
<th>Load/width</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>16.38</td>
<td>302.5</td>
<td>2.011</td>
<td>23.50</td>
<td>3037.</td>
</tr>
<tr>
<td>2</td>
<td>71.07</td>
<td>352.5</td>
<td>2.272</td>
<td>26.63</td>
<td>3496.</td>
</tr>
</tbody>
</table>

Mean:
53.73 327.5 2.141 24.17 3276.

Standard Deviation:
24.53 36.8 .185 .94 310.

Mean - 2.00 * Sdv:
4.67 254.0 1.772 22.28 2656.

Mean + 2.00 * Sdv:
401.0 2.911 26.05 3897.
B. Mode Shapes

First Natural Frequency.

Second Natural Frequency.

Third Natural Frequency.
Fourth Natural Frequency.

Fifth Natural Frequency.

Sixth Natural Frequency.
Seventh Natural Frequency.

Eighth Natural Frequency.