Finite Element Modeling Methods - Vibration Analysis for Ships

J. H. Spence, Noise Control Engineering, Inc.
E. A. Favini, Noise Control Engineering, Inc.
C. A. Page, Noise Control Engineering, Inc.

Accurate accounting of the vibration induced by machinery is important for the design of vessels that have low adverse impact on the crew, the ocean environment, and the vessel's structural integrity. This paper presents a summary of the findings of a project aimed at identifying appropriate methods of performing finite element analyses of ship vibration. The project focuses on predictions of local vessel vibrations in response to forced input from both hard and resiliently mounted machinery. Investigations include identification of minimum element density, model extents, boundary conditions, damping loss factors, and force input methodology, as well as other ship-specific factors such as influence of auxiliary structures, tanks, and water loading. The identified modeling methods are intended to provide a degree of model simplification that can be used with general purpose finite element software and will minimize loss of accuracy while improving the speed with which models can be created and analyzed.

KEY WORDS: Vibration, Finite Element Analysis, Vibration Modeling, Ship Vibration Prediction.

INTRODUCTION

Finite Element Analysis (FEA) is a powerful tool used for analysis of ship vibrations that result from various sources to assess 'habitability' vibration response (i.e. vibrations that would affect crew/passenger comfort) as well as vibration induced fatigue. Vibration analyses using finite element methods have become standard practice during various stages of vessel design since potential issues can be identified and remedied before construction begins. Finite element analysis is an invaluable tool which can be used to avoid serious vibration problems and save on costly remediation that might be needed on completed vessels.

However, creating finite element models is time consuming. In many cases, weeks to multiple person-months are required to create models, depending on the required model size. As a result of these time requirements such analyses are often pushed to later design stages. This can impact the available options for mitigation strategies when a vibration problem is detected. If problems are found late in the design, structural modification may not be practical and other factors such as weight and cost impacts may be onerous.

The accuracy of any finite element analysis will be subject to the modeling and analysis methodology that is used as well as the quality of the inputs. Selection of element types, mesh size, model extents, and how vibration sources are defined, among other factors will influence the model results. In many cases it is difficult to validate FEA models of vessels because measured data are limited, the measurement conditions do not line up with the modeling conditions, source forces or vibration levels are not known, budgetary considerations, and other influences.

This paper presents the findings of a project funded by the Ship Structure Committee with the objective of identifying effective and efficient methods of creating finite element models using commercially available software allowing for assessments of vibrations of local vessel structures due to excitation from shipboard machinery. The desire is to identify methods that are quick and efficient, which have been shown through validation tests to have a reasonable degree of accuracy.

This effort has been performed in two phases. In Phase I, a finite element model of a small aluminum catamaran was created with the purpose of performing sensitivity studies to various modeling parameters. These parameters include:

- Mesh density
- Element types
- Model extents
- Boundary conditions
- Effects of vessel outfitting
- Effects of localized masses and structural simplifications
- Damping

In Phase II, measurements of various types were performed on a large, steel hulled vessel (the Maine Maritime Academy’s training vessel, STATE OF MAINE, formerly the USNS TANNER). Vibration data were collected with machinery sources operating (diesel engine and smaller auxiliary items) as well as artificial sources (a modal impact hammer). Finite element models of this vessel were then created, and the measurements have been compared to the vibration response predicted by the model. Modeling approaches build on the lessons learned from Phase I.

The analyses performed in this effort generally focus on the frequency range of 1-100 Hz, which is applicable to many standard requirements for habitability vibration and structural fatigue. Examples of such standards include ISO 6954, ANSI...
S2.25, ANSI S3.18, ABS HAB/HAB+, and DNV Rules for Classification of Ships.

An overview of the vessels used for these analyses are provided in Fig. 1–2 and Table 1–2. Information on the vessel used in Phase I was provided by Kvichak Marine Industries, and information on the vessel used in Phase II was provided by the Maine Maritime Academy. Testing of the vessel used in Phase II was also facilitated by the Maine Maritime Academy and the US Department of Transportation.

Table 1: General parameters of the aluminum catamaran

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Length, Overall</td>
<td>57’ 8” (17.6 m)</td>
</tr>
<tr>
<td>Length, On Deck</td>
<td>54’ 8” (16.7 m)</td>
</tr>
<tr>
<td>Length, on Waterline</td>
<td>50’ 3” (15.3 m)</td>
</tr>
<tr>
<td>Beam, molded</td>
<td>20’ 7” (6.3 m)</td>
</tr>
<tr>
<td>Draft, approximate</td>
<td>33” (0.8 m)</td>
</tr>
<tr>
<td>Weight, LightWeight</td>
<td>50,000 pounds (22,700 kg)</td>
</tr>
<tr>
<td>Weight, Full Load</td>
<td>70,000 pounds (31,800 tonnes)</td>
</tr>
<tr>
<td>Structural Material</td>
<td>Aluminum</td>
</tr>
</tbody>
</table>

Table 2: General parameters of the STATE OF MAINE

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Length, Overall</td>
<td>499’ – ½” (152.1m)</td>
</tr>
<tr>
<td>Beam, molded</td>
<td>72’ (21.9 m)</td>
</tr>
<tr>
<td>Depth to Main Deck</td>
<td>42’ (12.8 m)</td>
</tr>
<tr>
<td>Draft (max)</td>
<td>28’ – ½” (8.5 m)</td>
</tr>
<tr>
<td>Light Ship Displacement</td>
<td>9267.29 LT (9,400 tonnes)</td>
</tr>
<tr>
<td>Structural Material</td>
<td>Steel</td>
</tr>
</tbody>
</table>

A general outline for finite element modeling methods has been developed using the results of Phases I and II, along with identification of prediction accuracy. This paper presents a summary of the findings, focusing on the recommended procedures for finite element modeling. Selected supporting information is provided for these recommendations; the full set of analysis data and results can be found in the full report for the Ship Structure Committee (Spence 2014).

Note that this study focuses on excitations from machinery sources; all measurements were performed dockside. Propulsion related excitations, (propeller, shaft, etc.) can also be a significant source of vibration on many vessels. Propeller excitation was not directly investigated as part of these efforts, though it is reasonable to expect that similar approaches to those described here can be used to model propeller and propulsion excitations. The modeling of vessel structures should be the same as when modeling machinery sources; primary differences will lie in the identification an application of forces and pressures, such as propeller pressure pulsations and alternating thrust (as described in ABS, 2006).

OVERVIEW OF RECOMMENDED ANALYSIS APPROACH

Errors, being a deviation from accuracy or correctness, are unavoidable with any modeling process, and finite element analysis is no exception. Prediction errors due to vibration modeling can be categorized as frequency errors (such as in the prediction of frequencies of peak response, corresponding to one or several modes of vibration) and magnitude errors (the ratio of corresponding predicted and measured levels). By expecting a certain degree of inaccuracy from a model, it is possible to use an analysis approach that reduces the overall error, allowing for better predictions and reduced need for excessive treatments to fix problems.

For example, assume the curves in Fig. 3 represent measured and modeled mobilities (i.e. vibration velocity referenced to a constant force) over a certain frequency range. Assume the blue curve is the measured mobility and the red and black curves are predictions from two different models. If we restrict the analysis to the exact frequency of the peak in the blue curve, there is error in the red model at that frequency and very large error in the black curve. This is true, even though all three curves can qualitatively be considered as being ‘close’. If instead we take the maximum level over a frequency range that
Machinery sources typically generate forces at specific frequencies which are often harmonically related to their rotation rate. Assessing the predicted vibration levels at a single frequency is not recommended as model errors related to those shown in Fig. 3 can lead to large discrepancies between model and measurement.

A second example is shown in Fig. 4, where the response magnitude from a fictitious source which generates a tone at a single frequency is compared to the modeled response over a 'prediction frequency range'. The prediction frequency range is centered on the actual excitation frequency, but spans over a range of frequencies which includes the peak in predicted response. In this case, the peak response in the prediction frequency range could be used as the prediction of the actual response to improve accuracy. The analysis frequency range would be chosen based on the expected error in the predicted frequency of controlling modes.

This may not be true in many situations, though when this condition occurs it will tend to lead to elevated if not excessive vibration levels.

In those cases where the structure does not contain a strong peak in response at a particular excitation frequency, using the peak predicted response will lead to higher predictions, and will often be an over-prediction of the actual vibration level. In these cases, it may be more appropriate to use an average response over the analysis frequency range rather than the maximum.

It will be shown that in the cases investigated here, this ‘average response’ method produces better correlations to measured data than taking the ‘maximum response’ over the analysis frequency range. However, for analyses of new vessels it is recommended that the maximum response approach be used, at least for initial analyses, as this will help to identify excitations near strong resonances of the structure which can lead to vibration excesses. Subsequent to this initial modeling, if it is found that there are no strong peaks in the local response of the structure the average response method may be preferred (and is possibly more accurate) as this will reduce the likelihood of over-treating the vessel.

**RECOMMENDATIONS FOR MODELING**

The following subsections provide recommendations for creating finite element models of ship vibration based on the results of the Phase I and Phase II analyses. The accuracy that is achieved by using these methods is provided later in this report.

**Meshing and Mesh Density**

In order to model the vibration response of local structures a certain level of detail of those structures is necessary in order to obtain reasonable fidelity. Since typical vessel construction uses stiffened plating, it has been assumed in this investigation that separate elements must be used for plates and stiffeners.

The minimum mesh density (i.e. largest element size or fewest number of elements) that is possible using this approach contains a single plating element between stiffeners. Higher mesh densities would result in multiple elements between stiffeners. A summary of mesh density options are provided in Table 3.

In all cases it is desirable to use elements with low aspect ratios (shape is close to square instead of long and rectangular). Furthermore, it was found that using 4-noded plate elements are preferred to using 3-noded triangular elements, though the occasional use of triangular elements will not drastically impact modeling results. Triangular elements are, practically speaking, unavoidable for real vessel geometries. Details of stiffener meshing are provided in a later section.

The investigations performed as part of this effort show that higher mesh densities generally produce improved accuracy, particularly for weaker structures (such as structural bulkheads used primarily for compartment segmentation and not for the
overall strength of the vessel) and at higher order modes of all structures. However, they also lead to longer solution times, and in extreme cases may not be possible to solve with available computing resources. It is therefore desirable to identify a balance between model size (higher mesh density, with added model creation and solution time) and the accuracy required for a given frequency range.

Table 3: Mesh Density Summary

<table>
<thead>
<tr>
<th>Mesh Density</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>Minimum Mesh Density</td>
<td>The nominal length of each side of an element is equal to the frame spacing of the vessel.</td>
</tr>
<tr>
<td>2x Mesh Density</td>
<td>The nominal length of each side of an element is equal to one-half the frame spacing of the vessel. There are generally two equally spaced elements per frame.</td>
</tr>
<tr>
<td>4x Mesh Density</td>
<td>The nominal length of each side of an element is equal to one-fourth the frame spacing of the vessel. There are generally four equally spaced elements per frame.</td>
</tr>
</tbody>
</table>

An example of the effect of mesh density taken from the Phase I results is shown in Fig. 5. Very generally speaking, it can be seen that the peaks in response (corresponding to various modes of the structure) are reduced in frequency when the mesh density is increased. This shift appears to converge with increasing mesh density. At low frequencies the models all predict responses that are close to one another, with shifts in peak frequency on the order of a few percent (in this example) and small differences in response magnitude. At frequencies corresponding to higher order local modes of the structure (above 65 Hz in this example) it is seen that the character of the models change. For example, the ‘minimum mesh density’ model loses features such as the peak in response near 70 Hz. This is one example of many from Phase I that show similar effects.

In Phase II it was found that the minimum mesh density model generally did not produce accurate results at response peaks corresponding to local modes of the structure. An example is provided in Fig. 6. (Note that the area below 8 Hz is greyed-out. This is because the measurement of mobility was found to be erroneous as a result of instrumentation limitations. This low frequency measurement error is typical of all measurement and is seen elsewhere in this report.)

The black curve in this example is the measured point mobility on the Transom measured with an instrumented impact hammer. It is seen that the minimum mesh density model (the blue curve) cannot accurately produce the peak in response seen at 40 Hz.

Fig. 5: Effect of mesh density – from Phase I analysis

The 2x and 4x density models produce better results, with a distinct peak being present in both cases. Both models produce reasonable correlation to the measurement at frequencies up to 60 Hz. The error in frequency at the 40 Hz peak is 5% for the 4x mesh and 10% for the 2x mesh. The magnitude of the response is off by a factor of 2.6 in these cases. In this specific case this error is believed to be due to in part to differences in the measured vs. modeled damping (discussed later in this paper).

Fig. 6: Effect of mesh density – from Phase II analysis

Based on the findings of many models and analyses not shown here, it is generally recommended that a “2x density” or finer mesh be used in order to obtain accurate results for local modes. It may be reasonable to use the minimum mesh density only when modeling whole body modes (being modes where large portions of the vessel move together, and resemble modes of a large beam).
A “4x density” mesh is preferred (4 elements between stiffeners) particularly when accurate modeling of higher order modes is required and when model sizes do not become prohibitively large.

An 8x mesh was investigated for one area of the vessel in the Phase II analysis and was found to improve results for higher order local modes. (8x mesh densities were not investigated in other areas of the vessel, though arguably some improvement could be gained elsewhere as well.) For this specific structure (located in the superstructure) it was seen that use of a 4x mesh resulted in 6 elements per wavelength or less at the higher order local modes of the structure. The primary benefit seen with the 8x mesh was a reduction in the error of predicted peak response frequencies. The error in the 8x mesh was minimal, and the 4x mesh produced errors that were 10% or less. The error in predicted magnitude did not change appreciably. In this case, it could be argued that by using a larger analysis frequency range, a 4x mesh could be used with acceptable accuracy in predicting peak responses, and would result in a smaller model and reduced analysis time.

Stiffener Element Types
Stiffeners can be modeled in various ways using beam elements, plate elements, or both. Beam elements incorporate the effects of the stiffener web and flange in a single element. An alternative is to separate the web and flange by using a plate element for the web and a beam element for the flange. A third option is to use plate elements for both web and flange.

By using a beam to model the entire stiffener (web and flange), the nodes that already exist on plating elements are used and no additional nodes are created. This is by far the simplest approach and leads to the smallest model size. Conversely, if a plate is used to model any part of the stiffener then additional nodes are created. These nodes must be connected to adjacent structures, will increase the size of the model, and will increase model creation and analysis time. However, the connection of the stiffener plate to an adjacent bulkhead, stiffener, floor, etc. provides additional coupling and stiffness which may be necessary in order to obtain accurate results.

Conventional stiffened plate ship construction makes use of both “large” and “small” stiffeners. For the purposes of these recommendations for finite element modeling, large stiffeners have webs that are on the order of ½ to 1x frame spacing or larger.

It is generally recommended that large stiffeners be modeled using plate elements for webs and beam elements for flanges. This is strongly recommended on curved hull sections, particularly where the stiffeners tie-in to floor plating. Minimal improvement is expected by modeling flanges with plate elements.

It was found that model accuracy is not drastically affected by modeling small stiffeners with beam elements (for the frequency range where accuracy is generally seen, discussed later). It is possible to model small stiffeners with plate webs, though it was generally seen that very minor improvements were found when this approach was used. An example is shown in Fig. 7, which shows the measured and modeled point mobility for a location on the side shell. Here it is seen that the primary effect of using plate stiffeners is to increase the frequency of peak response for a higher order local mode (near 70 Hz in this case). In this case a slight improvement in correlation between model and measurement is achieved for this single peak. Differences at frequencies lower than this peak are negligible, and the use of plate stiffeners does not improve results at higher frequencies.

![Fig. 7: Effect of mesh density – from Phase II analysis](image-url)

**Model Extents and Boundary Conditions**
It is often desirable to model only a portion of a vessel; this allows for reduced model creation and analysis time. However, care must be taken when modeling only sections of vessels, particularly when the vessels are small.

There are two items of concern when truncating a model. First, ‘false whole body modes’ of the truncated model may interfere with the analysis frequency range by producing peaks and dips in the response, confusing the results for local structural response. An example of this phenomenon is shown in Fig. 8 which is taken from the Phase I analysis of the small aluminum catamaran. The black curve is the vibration response of the full finite element model (all structures included) for a particular force/receiver pair. This curve is used as the ‘goal’ for the purpose of this comparison. The other curves show the results from models with varying degrees of truncation for the same force/receiver pair. Note also that the truncated models used pinned boundary conditions at the locations of truncation, where the full ship model had no constraints applied.

For this vessel the frequency range of predicted whole body modes extends up to 50 Hz (this is not typical for many vessels, and is due to the small size of the vessel combined with ignoring...
water loading effects). It is seen that the truncated models produce peaks in the response (corresponding to predicted whole body modes) that are completely erroneous. Above the whole body mode frequency range the models all produce similar results. This is because, for this force/receiver pair, the response is dominated by a mode at 95 Hz.

An example plot of the displacement from one of these modes is shown in Fig. 9. It is seen that the ship is acting in a similar manner to the first mode of a cantilevered beam. The frequency of this ‘equivalent beam’ is dependent on its length; for a truncated model, this choice of length is arbitrary and therefore the predicted frequency for this mode is equally arbitrary and does not correspond to a mode of the full ship model.

Identification of false modes can be achieved by changing boundary conditions. For example, both the mode shape and frequency of the mode highlighted in Fig. 9 change if the pinned constraints are removed. This change can be identified in the vibration response at specific locations and through more detailed investigations of mode and deflection shapes.

This result implies that for a model of ‘sufficient’ extent (i.e. where false whole body modes do not interfere with local mode analysis) the choice of specific boundary conditions will not impact the results. It should be acceptable to use “free” boundary conditions (no constraints) as this approach is easiest to implement.

A second concern with model truncation is model accuracy of local modes. It was found that even when false whole body modes are not problematic, the response of a given structure (such as a bulkhead) is dependent on the makeup of the surrounding structures.

An example is provided in Fig. 10, which shows the point mobility of a bulkhead exhibiting strong local resonances (taken from the Phase II analysis). Two curves are shown; one from a model containing structures surrounding the test bulkhead and one from a model that only contains the test bulkhead (with pinned boundary conditions applied to the edges of the bulkhead).

The frequency of the first peak has shifted downward by 60% for the ‘bulkhead only’ model (the deflection shape also changes significantly). This indicates that the intersecting structures are an important factor in the overall stiffness of the bulkhead and the resulting vibration response. Therefore, capturing this stiffness (and mass effects) by modeling the surrounding structures is an important aspect in obtaining accurate results.
Generally speaking, it is recommended that the model extend at least one major bulkhead/deck in each direction from the area encompassing both the source and response locations. It is also recommended that the full width of the vessel be modeled to avoid complications with symmetry and anti-symmetry boundary conditions. For cases where false whole body modes occur at frequencies of local modes, larger models should be used. In situations where it is not possible to model the full extents of the vessel, it is recommended that multiple boundary conditions be used to identify false modes.

**Damping**

In this analysis a ‘structural damping’ or ‘loss factor’ type damping has been assumed. This is due in large part to the ease with which this damping type can be applied to models and its use in commercial finite element codes. By using a loss factor damping model, the damping is proportional to the stiffness matrix and a separate damping matrix is not needed; this simplifies the calculation process. Additional details can be found in Ginsberg (2001).

It is important to recognize that multiple damping mechanisms may be at work for any given ship structure or for the vessel as a whole. The modeling process used here assumes that these mechanisms can be simplified for the purpose of analysis. In this analysis, an ‘equivalent’ damping is assumed to exist, which may encompass multiple damping phenomena.

The damping loss factors of several test structures have been analyzed using modal impact test data analyzed using LMS Test Lab Polymax software (LMS, no date). The extracted results were limited to modes that exhibited strong peaks in response for point and transfer mobilities.

The range of loss factors for all test structures was generally between 0.01 and 0.09, though for each specific structure the values were more consistent. Examples are provided in Fig. 11, which shows the levels of measured damping on three test structures. The damping value is plotted for each extracted mode.

In each example it is seen that, for a given structure, the damping values are relatively consistent across the entire test frequency range. In the case of the Steering Gear Room side shell (top plot), the values range primarily between 0.02 to 0.06, with an approximate average damping of 0.04. On the Transom (adjacent to the side shell, middle plot of Fig 11.), damping values are lower, ranging between 0.01 and 0.03 with an average value of 0.02.

The side shell and Transom have no insulation, joiner, or damping treatments, and are effectively ‘raw’ stiffened plating (both are above the waterline). Taken together, the average damping loss factor is approximately 0.03. This is in line with the recommendations of American Bureau of Shipping (ABS, 2006).

Fig. 11: Examples of change measured damping loss factors for difference structures.
The damping on the Battery Room Bulkhead (bottom plot of Fig. 11.) show higher levels of damping, with an average damping of 0.06. This structure contains insulation and a metal joiner facing. Based on these results it may be inferred that the use of joiner will increase the damping for a particular structure. However, these results may be limited to the specific joiner type used on the STATE OF MAINE. Further analysis would be required to confirm this conclusion.

A loss factor value of 0.03 was used for all models in this investigation with the exception of the Battery Room bulkhead model. These and other damping results indicate the actual level of damping is typically close to this value (for structures without joiner facings), though for any specific structure the actual damping can be slightly greater or less than this value.

This is the case on the Transom; these results show an actual, average damping that is slightly less than the value used for analysis, leading to somewhat greater errors in magnitude response (as was shown earlier in Fig. 6 for the Transom point mobility peak at 40 Hz). The higher value of damping was used to show the degree of error that would result in a typical analysis where the actual damping is not known. It is not possible to use the available results to determine which structures will have greater or lesser degrees of damping without prior testing.

It is recommended that a damping loss factor value of 0.03 be used when modeling steel ship structures. This value would apply at all frequencies for local modes where the analysis is shown to be valid (discussed later in this report). This damping value is an average level that should be appropriate for most structures. Increased levels of damping may be appropriate for structures with joiner facings. Structures with applied damping treatments should also have increased damping levels, though this may be frequency dependent. Damping for aluminum structures is expected to be similar to steel, but may vary from those measured on the steel test vessel.

**Joiner, Deck Coverings, and Insulation**

The influence of joiner, deck coverings, and insulation was investigated as part of this effort, though the results provide only a partial indication of appropriate modeling methods which are subject to further verification.

Modeling of these items was performed by adding a ‘non-structural mass’ to the affected plating elements. This approach solely accounts for their mass; it is assumed that the influence of stiffness is negligible, though this may be dependent on the material, attachment method and type of base structure.

In the Phase I analysis, the influence of ‘typical’ composite joiner facings (being a sandwich with a mineral wool core between sheets of thin sheet metal) as well as a 1” (25 mm) thick lightweight marine leveling compound were investigated for the aluminum craft and an ‘equivalent’ steel vessel. It was found that there were only small differences in response at global mode frequencies, though much larger differences are possible for local modes. These differences were greater for the aluminum vessel as compared to the steel vessel, since the change in mass from a given treatment was more significant. In all cases the changes were primarily seen on the structure where the treatment was applied.

In Phase II, one structure was tested to determine the influence of joiner facings. The STATE OF MAINE used sheet metal facings that are directly attached to the hull structure being covered. (Insulation was also located between the structure and facing.) This approach is seen on older vessels, though for newer vessels using ‘composite’ joiner panels with no direct connection to the structure being covered is common. This nuance may affect how such treatments should be modeled.

The model of the tested structure showed improved correlation to measurements when the mass of joiner and insulation was added to the structure, though this was somewhat dependent on location and frequency. Example comparisons are provided in Fig. 12; the upper plot shows the point mobility at a location centered on plating and the lower plot is for a location on a stiffener. In both cases the response is reasonable at low frequencies (above 5 Hz) and poor at high frequencies (above 60-70 Hz).

For the plating location (upper plot), the peaks in response between 30-70 Hz are more closely captured when the mass is included, indicating that the joiner has an effect on the response of the structure. A similar effect is seen for the stiffener location (lower plot), though this is primarily for the first peak near 35 Hz. The ‘with treatments’ model does not do as well in capturing the response between 40-50 Hz at this location; this may be due to a decoupling effect of the joiner, or simply model error.

As discussed in the previous section, there is an indication that structures with joiner facings have a higher damping loss factor than non-treated structures. Though again, this result is one example and further study is recommended.

Other locations were tested which contained insulation material alone (no joiner or sheet metal facing). These locations were found to have minimal change in modeled response when the mass of insulation was included, and modeling of this mass was not found to be needed in order to obtain reasonable accuracy. Based on these results, it is believed that the mass of joiner and deck coverings may need to be included when modeling the response of local structures. For this effort, the mass of both the joiner and insulation between the joiner and structure were included. However, including the mass of insulation in the model, when there is no joiner present, does not appear to be required.
Fig. 12: Examples of effect of including joiner panel mass.

These results are based on a detailed analysis of a single structure with a particular joiner arrangement. Additional investigations are recommended as different constructions may lead to different conclusions.

**Auxiliary Structures**

In many cases it is possible to ignore certain structures as they will have negligible influence on model results and are not required in order to maximize accuracy. As part of the Phase II study it was found that items such as piping and pipe supports, lighting, cableways, handrails, small antenna foundations, and similar structures could be ignored. Some example pictures of these items are included in Fig. 13.

In some cases, the omission of these items may affect the accuracy of prediction in the immediate area of the structure where they are located (i.e. at the attachment point). However, the response of nearby structures appears to be largely unaffected (at least subject to the achievable modeling accuracy seen elsewhere).

Fig. 13: Items that can be ignored when modeling local structural response. Piping (top), small pipe supports (middle), small auxiliary structures (bottom).
Conversely, inclusion of some other items was seen to be beneficial to model accuracy, if not required in order to achieve reasonable results. This includes the mass of some hard mounted auxiliary machinery. Masses were added to the model using a simplified approach either by 'smeared' the mass across appropriate plate elements or nodes or by using a single mass element with rigid connections to the stiffened plating (or foundation).

The greatest influence of machinery items was seen in the vibration response on decks adjacent to machinery. It was also found that machinery items had a significant influence at some bulkhead locations bulkheads directly adjacent to hard mounted machinery and other large masses. This result was not universal though, and in several cases the local response of structures that were several frames away from large masses was not significantly influenced by the inclusion of these masses.

Based on these results it is generally recommended that masses of auxiliary machinery items be included in the model, particularly when they are hard mounted to the vessel. In general, the added mass of machinery will be more significant when it makes up a greater percentage of the local structural mass, as would be the case on aluminum vessels as compared to steel.

It may be possible to achieve accurate results by only including masses from large machinery items such as large pumps, compressors, etc., especially if results are needed at lower frequencies (i.e. near or below the first local mode of pertinent structures), particularly if the response close to smaller machinery items is of lesser concern. Refinement of this recommendation would require further study, but may result in decisions made on a case-by-case basis. It should be possible to inspect the impact of added mass for any given model by analyzing the point or transfer mobilities of a given model section. (Note that information on machinery masses was very limited for this study, and no direct measurements of machinery mass influence could be performed.)

The results of Phase I indicate that the modeling of doors, windows, and removable deck plating can affect the predicted response spectrum. Again, these effects will be most significant in areas 'close' to the additional structure. The Phase II study indicates that the influence of doors is relatively small and localized to areas near the door. Based on these results, it is recommended that door frames be modeled (likely using beam elements) though the door itself may be neglected since it is decoupled.

In Phase I it was seen that the addition of windows had a large effect on the vessel’s vibration response, though in that case the windows provided a significant degree of stiffening to the superstructure (a large percentage of the superstructure was windows). In such cases modeling of windows will likely be beneficial, though the specific method of modeling that is appropriate could not be studied in detail. For larger vessels it would likely be acceptable to neglect windows as long as the response on the bulkheads in close proximity to windows is of lesser concern.

The influence of removable plates such as BERPs and WERPs were investigated on a limited basis in Phase I. Based on these results it is recommended that these structures be included in a vibration finite element model. It should be reasonable to model these structures as if they were continuous structure (i.e. use the same nodes at the interface with removable plates and the primary structure) though additional investigation may be needed if the attachment method is complex.

The mass of fluids in tanks was seen to produce negligible changes in the response at all receiver points (far from tanks) during the Phase I study. No tank loads were included in the modeling efforts of Phase II; it was found that even for compartments located adjacent to tanks, tank loading did not yield a significant effect.

It is expected that tank loads can influence the frequencies of whole body modes for some vessels. For this reason it is believed that the mass of fluids in tanks should be included, possibly through the use of a ‘smeared mass’ on plating. This approximation will likely have limitations at higher frequencies. As discussed above, the effects of these masses should have the greatest effect on local modes of the structures at locations where the masses are applied, and have less of an effect at other locations (other than for whole body modes).

Although cargo loads were not investigated as part of this effort, similar conclusions discussed above for tanks should apply to large cargo masses. It is generally recommended that the mass of cargo be included in the model, though their impact may be more significant for the prediction of whole body modes rather than local modes.

**Water Loading**

Hull plating that is in contact with the ocean (i.e. the “wetted hull”) is subject to additional dynamic forces. These ‘water loading’ forces effectively create an added mass on the hull plating (Veritec, 1985). The effect of water loading is a function of the vibration contour of the hull, depth of the plating in the water, and other factors. The effective mass that is added to the hull is complex and frequency dependent.

Water loading was seen in this analysis to be an important factor in achieving accurate results, primarily for structures that are in contact with the water. An example is shown in Fig. 14, which presents a comparison of measured vs. modeled point mobility on a stiffener that is part of the wetted hull in the lower Machinery Room of the vessel analyzed in Phase II. The results from two models are compared to the measurement: no added mass and with a uniform 0.58 lb/in² (400 kg/m²) added to all plates in contact with the water. (This approach to water loading is highly approximate. The specific value used was determined through iterative processes to identify a ‘good fit’ to the measurement data. This was done not to identify a proper
method for calculating water loading but to verify the added mass effects.)

All sources investigated in this study, both resiliently and hard mounted machinery, were modeled by applying forces at mounting locations. This approach implies that the machinery sources are ‘force sources’ rather than ‘velocity sources’; in other words, it is assumed that the force on the foundation is largely independent of where the unit is installed instead of the velocity being constant regardless of installation location and details. While this aspect was not studied directly as part of this effort, experience indicates that highly stiffened foundations result in lower vibration levels for the same machinery item. A force source is consistent with this finding, where the stiffer foundation has a greater impedance (reduced mobility), and the same force applied to such a foundation will result in lower vibration levels. (This assumption may not apply in all cases, but may apply to most practical marine situations.)

Various methods of force estimation were investigated as part of this effort. For resiliently mounted machinery, an “Above Mount Vibration” approach was found to produce reasonably accurate results. In this approach, the above mount vibration levels at each mount location are measured (in three orthogonal directions) and are combined with the dynamic stiffness of the mounts to produce the forces on the foundation using the equation:

\[ F = kd \]  \hspace{1cm} (1)

where \( F \) is the magnitude of the force, \( k \) is the dynamic stiffness of the resilient mount, and \( d \) is the vibration displacement magnitude. The relative phase of the acceleration at the above mount measurement points can be used directly to determine the relative phases of the forces at each mount. This approach is relatively straightforward and was seen to produce similar results as more rigorous methods of force estimation performed in this investigation.

Unfortunately the same method cannot be applied to hard mounted machinery. For hard mounted machinery an ‘Impact and Relative Response’ approach was used and found to produce reasonable estimates of force. In this approach, the vibration levels of the machinery foundation at attachment locations are measured with the machinery operating. Then, with the machinery secured, the point mobility at the same points are measured using a known force (such as an instrumented impact hammer or a shaker). The force at each point and direction is then directly calculated using the equation:

\[ F = v/Y \]  \hspace{1cm} (2)

where \( Y \) is the measured mobility with the unit secured and \( v \) is the measured velocity with the unit operating. In this approach, only the mobility and vibration pairs at the same location and direction are used to produce the force at that location and direction. Cross coupling effects are ignored, though in the investigations of Phase II these couplings were seen to be weak.

![Image](image-url)

Fig. 14: Example of added mass effect of water loading on wetted hull plating.

The ‘no added mass’ model produces a response that essentially cuts through the middle of the measured peaks and dips; the overall trend is correct but the details of peaks and dips are not. The ‘with added mass’ model does capture the first large peak at 35 Hz, and has some similar features at frequencies up to 44 Hz, though above this frequency the measurement and model diverge.

The results of other models with different values of mass loading produced improved results in specific, narrow frequency ranges, but no single added mass model was able to produce better results than the result shown above. A second test location on the plating of the wetted hull showed poorer correlation.

These results speak to both the frequency dependence of the water loading effect and its complexity. Identification of proper methods to calculate water loading effects could not be performed as part of this study.

It is recommended that more sophisticated approaches be used and/or investigated to calculate and apply water loading effects. Examples include the use of coupled boundary element / finite element methods or explicit finite element modeling of the fluid domain.

It is important to note, however, that accurate results were obtained for structures that were not in direct contact with the wetted hull, including those structures that were immediately adjacent to wetted hull areas.

**Modeling Machinery Sources**

Machinery sources typically generate tones at their rotation rate and harmonics of rotation rate; in some cases additional tones or broadband components are present.
Both approaches require performing measurements on the
machinery item, though such measurements may not be possible
in all cases. If the assumption of machinery sources being
‘force sources’ is correct, it should then be possible to estimate
the force from a given source by either directly transferring the
forces from a similar machinery item to the current situation, or
otherwise scale the forces based on differences in machinery
size or other parameters. Using this approach, it is reasonable to
expect that a database of source level information can be
developed for different sources.

When modeling resiliently mounted machinery sources in a
finite element model, the machinery item itself should not be
included in the model. For hard mounted machinery, the mass
of the machinery should be included as one or more mass
elements connected to the foundation with rigid elements.

One hard mounted source was investigated in detail during
Phase II. This was a bilge pump, which consists of the pump
itself and a motor; a picture is shown in Fig. 15. The measured
vibration phase data indicate that these items act as two separate
rigid masses. Although this result is from testing of a single
unit, it is consistent with the general makeup of the motor/pump
system since the motor is connected to the pump only via the
shaft (and the foundation). This result may be applicable to
other machinery items that are made up of two or more separate
components.

![Image of hard mounted bilge pump](image.png)

Fig. 15: Picture of hard mounted bilge pump.

Any real machinery source should be expected to have different
force magnitudes and phases at each mounting location. The
most accurate modeling results are achieved when the specific
force distribution and relative phases of forces are included in
the model. A simplified approach was also investigated, where
all forces were of the same magnitude (being the average force
at all mounts) and the same phase. This resulted in increased
levels of predicted vibration; on average the increase was a
factor of 2-3 though in some cases predicted levels were six
times greater using this approach. If detailed force and phase
distribution information is not available then the prediction
results are expected to be conservative.

**MODEL ACCURACY**

The results of model accuracy are presented below. These
results are derived from the Phase II analysis of the *STATE OF
MAINE* which compares measured vs. modeled vibration
response.

**Point and Transfer Mobility**

Several examples of modeled vs. measured point mobility
comparisons have been provided in previous sections. For the
models of the *STATE OF MAINE*, the predicted local vibration
responses capture the overall nuances of the measurements out
to frequencies of 60-70 Hz when the modeling
recommendations of the previous sections are followed (with the
exception of areas on the wetted hull). This frequency range
roughly corresponds to higher order local modes of the structure
being tested. This result also applies to comparisons of
measured and modeled transfer mobilities; several transfer
mobility examples are presented in Fig. 16.

Similar results are also seen when analyzing the predicted and
measured vibration contours. Fig. 17 provides an example
comparison of the measured and modeled operational deflection
shapes corresponding to impact testing on the side shell in the
Steering Gear Room of the *STATE OF MAINE*. Comparisons
are shown for 2x and 4x mesh densities. The vibration
distribution across the test structure is captured in both models
at frequencies of peak response near 30 and 70 Hz. (Predicted
peaks at 20 and 27 Hz are false whole body modes.) At 77 Hz
the modeled deflection shapes start to show discrepancies with
the measurement, and at 88 Hz larger discrepancies exist. This
result corresponds with the accuracy seen when comparing point
and transfer mobilities for this structure. In general, the
predicted vibration distribution was seen to be accurate when
the predicted point and transfer mobilities correlate to
measurements.

**Vibration Response to Machinery**

As discussed previously in this paper, when performing a forced
response analysis it is recommended that models be analyzed
over a frequency range in order to account for errors in
predicted mobility response. Two approaches have been
suggested: ‘average response’ and ‘maximum response’, both of
which are taken over the analysis frequency range.

Example comparisons of measured vs. modeled vibration
response to the three sources investigated in this effort are
provided in Fig. 18–19. The sources include a reciprocating
start air compressor (resiliently mounted), genset (resiliently
mounted), and bilge pump (hard mounted). Fig. 18 provides the
predicted levels using the average response method and Fig. 19
provides the same data using the maximum response method. In
all cases, the response was taken for a model with a 4x mesh
density using a frequency range of +/- 5% of the excitation
frequency.
Fig. 16: Examples of measured vs. modeled transfer mobilities.

Fig. 17: Example of measured vs. modeled operational deflection shapes at frequencies of peak response.
Fig. 18: Examples of measured vs. modeled response to machinery excitation using average response method. Start air compressor (top), genset (middle), and bilge pump (bottom). Note that bottom plot is response on wetted plate; water loading errors are believed to be the primary cause of discrepancies.

Fig. 19: Examples of measured vs. modeled response to machinery excitation using maximum response method. Locations and forces are the same as for Fig. 18.
The correlation between measurement and model for the start air compressor is very good at 30 and 60 Hz; one of two response locations is shown here, with both showing similar accuracy. At 90 Hz the prediction shows greater error; this is expected given the errors in mobility seen above 70 Hz discussed in previous sections. In this case the response locations were on the side shell directly adjacent to the compressor.

The predicted response from genset excitation shown here also has strong correlation to the measurement (note that the apparent accuracy at 90 Hz is believed to be coincidental given the errors noted previously at this frequency for this vessel). This is one of many locations which were both adjacent and ‘far’ from the source (the furthest response point where measured data was above background was the aft Machinery Room bulkhead, one deck down from the genset and roughly 40 feet (12 meters) away from the genset). The other response locations showed reasonable accuracy with errors in magnitude response typically being near or less than a factor of 2, though there are some exceptions.

The side shell below the deck where the genset is located is wetted hull plate. These response locations had good correlation (a factor of 2 or less) at many frequencies though at some frequencies greater errors were seen. These errors are believed to be due to errors with water loading approximations.

Other areas with greater errors include deck locations that are in close proximity to large machinery items. Detailed information on these machinery items was not available, and therefore there is greater uncertainty at these locations.

The response locations for the bilge pump are all on the wetted hull, and are again subject to errors in the water loading approximations. Typically, the response near 30 Hz was seen to have the best accuracy, with somewhat larger error at the 25.8 Hz peak. The correlation at higher frequencies was poorer overall.

Comparing Fig 18 to Fig. 19, it is seen that the maximum response method predicts similar though higher vibration levels (as expected). It should be noted that the transfer mobilities from the sources to the response locations did not contain large, strong peaks in response at the frequencies of interest (as opposed to the response on the Transom at 40 Hz seen in Fig. 6 and Fig. 16). Therefore, the benefit of the maximum response method was not realized in these examples — i.e. there was no forced excitation of a strong structural resonance. As a result the average response method generally had better correlation to measured data. However, as discussed previously use of the maximum response method is still recommended, at least to determine the worst-case vibration levels, as this approach will capture vibration responses which can lead to problems.

The overall accuracy that can be achieved when modeling the response of a ship structure is dependent not only on the ability of the model to capture the transfer mobility from the machinery to response locations but also on the accuracy of the forcing inputs and the methods used to analyze the results. Force estimation techniques have been discussed previously and are believed to provide reasonable accuracy as they were checked using multiple methods. A direct identification of the accuracy of these methods was not made in this study.

**Summary of Accuracy**

The results of this study have shown that with reasonable modeling effort and accurate input information the following accuracy can be achieved for a majority of structures, excluding wetted hull surfaces:

- Prediction of resonance frequency of local ship structures to within 5-10% or less.
- Prediction of local response magnitudes to known input forces and real machinery sources within a factor of 2-3, with results often being less than a factor of 2.

This prediction accuracy is, of course, subject to several limitations. It was found that this accuracy could be achieved for frequencies up to approximately 60-70 Hz for the modeled vessel, though this frequency limit depends strongly on the specific structure that is being modeled.

More generally, accuracy was often obtained for the first several local modes of a structure, determined through investigation of point mobilities. (Note that many modes will exist throughout the vessel for most frequency ranges of interest. The local modes of interest here are those that would be highly excited at a particular point on a structure by excitation at that point.) Accuracy typically worsened as the complexity of modes increased; in these cases, plating motion was seen to be significant relative to stiffener motion, and full wavelengths occurred in the plating between stiffeners. Mesh density plays a big role in determining accuracy, as the wavelength at a particular frequency will need to be sufficiently represented spatially in the model.

These results do not apply to wetted hull plating. The modeling approach used here is not practical and does not capture water loading effects accurately. More sophisticated methods are recommended, as discussed previously.

**CONCLUSIONS**

This study has shown that modeling of the local vibration response of ship structures as a result of machinery excitation is possible using finite element methods. Using the modeling and analysis recommendations provided in this report, it is generally possible to predict the magnitude of response of structures to within a factor of 2-3 from actual measurements, with results often being less than a factor of 2. Some exceptions do exist however, such as on wetted hull plating and areas in close proximity to large machinery.

Modeling accuracy will always be subject to, among other factors, mesh resolution, the need for structural and mass approximations, damping and water loading estimations, and
even accurate vessel information. However, it is possible to perform predictions which can be used to aid in the assessment of vibration levels in new builds and to provide information as to the distribution of vibration throughout a given structure to assess appropriate mitigation approaches.

The greatest uncertainty identified in this effort is the estimation of proper mass loading effects on the hull plating that is in contact with the water. This uncertainty was seen to produce large errors and limits the frequency range where accuracy can be achieved. Water loading effects are complex and frequency dependent, and could not be studied in detail as part of this effort. Methods such as coupled finite element and boundary element modeling or full finite element meshing of the water domain should be explored further. Fortunately, the impact of these errors was seen to have the greatest impact on the wetted hull plating; accuracy can still be achieved on structures directly adjacent to those in contact with the water.

Another potential source of error was seen to be in the modeling of auxiliary machinery and other localized masses. Throughout these investigations it was seen that localized masses can produce significant changes in vibration response. The significance of these changes was seen to be a function of proximity, where the greatest influences are observed at locations that are directly adjacent to these masses, with reduced effects as the response location is moved farther away. In several cases, the error of not including these masses was minimal even for locations on the same bulkhead or deck as the mass, as long as they were separates by several frames.

Other factors such as choice of element types for stiffeners, damping, joiner panel and deck treatment masses, model extents, and general accuracy of the vessel’s structural information were seen to be important in developing accurate models. Conversely, items such as piping, handrails, small foundations of antennas, and other similar details were not found to have a large impact on the model results. This is fortunate, because such details are often not available early in the design stages of a vessel and their modeling can become laborious.

ACKNOWLEDGEMENTS
The authors would like to thank Mr. Andrew Girdler of Kvichak Marine Industries for providing details of the vessel used in Phase I of this effort. The authors would also like to thank Captain Leslie Eadie, Chief Engineer Roger Lowell, Mr. Tony Margan, and the crew of the Main Maritime Academy Training Vessel STATE OF MAINE who facilitated and supported the measurements performed in Phase II of this effort, and for providing vessel details. The authors would also like to thank the Maine Maritime Academy and the US Department of Transportation for allowing these measurements to take place.

REFERENCES


