

Root Cause Determination of Excessive Hull Vibration On a First-In-Class Tugboat Design

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In August of 2002, Azima DLI was tasked to collect vibration data on a new tug design being built in Seattle for use as a North Puget Sound escort tug. The tug's main duty is to escort tankers through the narrow channels of the San Juan Islands. It is designed to apply large forces to steer or brake the tanker in the event of an emergency. To meet these requirements, the 129'-6" long tug has a unique hull shape as shown in Figure 4. The vessel has a large skeg at the bow to provide lift, and a relatively wide and shallow hull. In indirect mode the hull acts similar to a kite, resulting in line pulls of over 150 tons at ten knots. This is double the static pull which the vessel's propulsion system can achieve. The vessel is propelled by a pair of Voith-Schneider cycloidal propellers, each driven by a 3,600 horsepower Caterpillar diesel. These propellers give precise control of both the thrust magnitude and direction.



Data were being collected in hopes of determining the root cause of excessive hull vibration. To do this, several testing and analytical techniques were utilized. The DOE (design of experiments) encompassed operating factors of ballast level, speed, and structural points. In all cases, structural point data was collected simultaneously from a matrix of conditions decided upon before each sea trial.

With the data provided by Azima DLI, naval architects on the project were able to determine a course of action that would provide an operational solution that was agreeable with respect to both time and budget.

On the initial sea trial, a high level of speed dependent vibration had been detected during operation and structural resonance near full power was suspected by the naval architects involved in the project. Azima DLI was contracted to quantify the vibration levels at several structural points, determine the mode shape and specify forcing frequencies found during tug operation. To record digital data from all points simultaneously, two 16 channel DAT (Digital Audio Tape) recorders were utilized. For most tests, 24 accelerometers were powered by two 12 channel ICP amplifiers that also conditioned the accelerometer signals before sending them on to the DAT. Optical tachometer data was collected on each run from both of the propulsion drive shafts. The tachometers were located between the turbo coupling to the 3600 series Caterpillar diesel engines and the input to each Voith cycloidal drive unit. Once the 24 channels of simultaneous test data had been saved to the DAT, the data were analyzed and disseminated in the following three ways.

To quantify speed dependent displacement at different locations around the ship, a set of speed plots for point locations and ballast levels were created using MATLAB. The axes ranges of all the plots were kept constant for easy comparison between points and test conditions. See Figure 1.

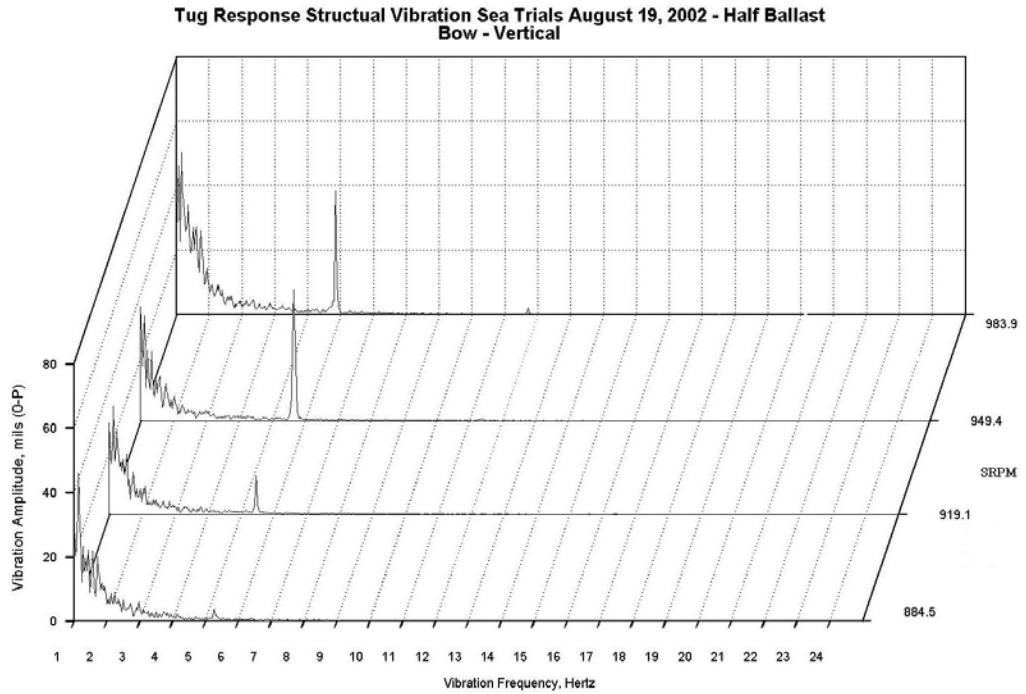


Figure 1

To identify forced as well as resonant responses at each point, MATLAB was also used to create 3D speed rundown plots. These plots clearly show the relationship of speed dependent energy inputs and their interaction with the natural frequency of the structure at each point. For example, the forced elements occur on the plot as diagonal stripes, whereas the natural frequencies appear as stripes parallel to the Z or Shaft RPM axis. When a forced input crosses a natural frequency, the resonant response can be seen on the Y axis of the plot. A good example of this occurs at ~5 Hz and ~1000 SRPM. The height of the peak as well as its color denote the severity. In this case, the cycloidal drive blade rate crosses the fundamental natural frequency of the hull and the large red peak is the resonant response in Gs RMS. See Figure 2.

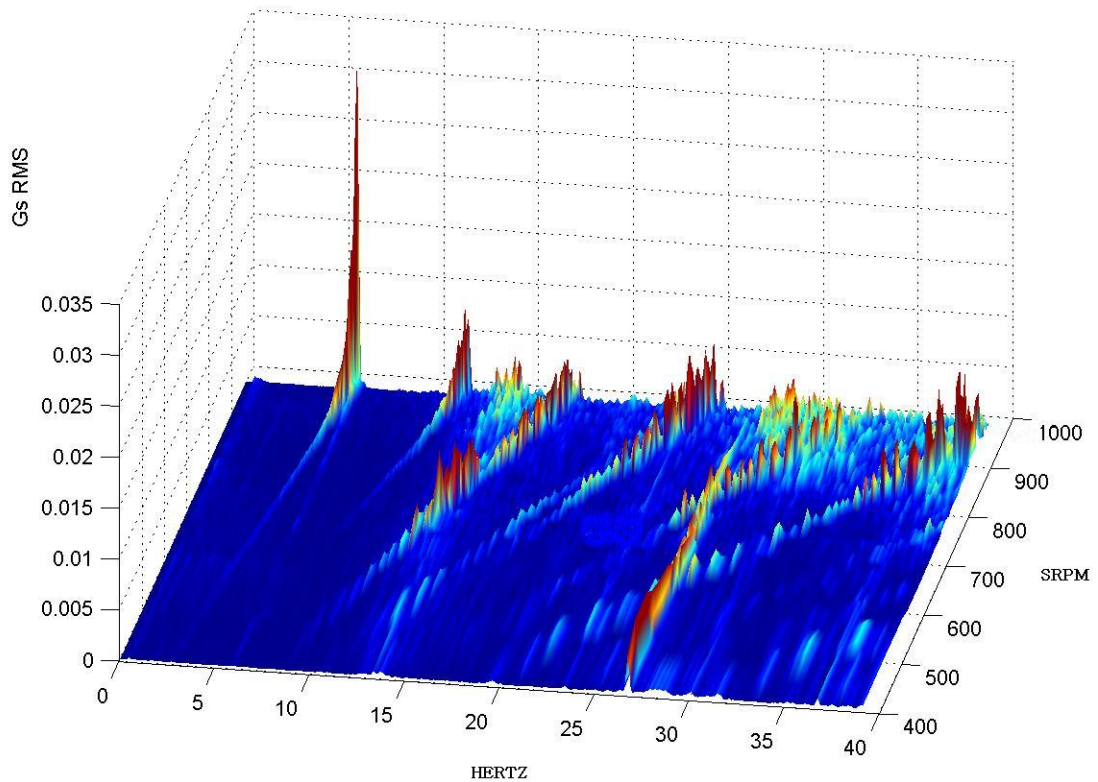


Figure 2

ME'scope from Vibrant Technologies was used to create an animated three dimensional model of the structural points for each set of conditions. These animated models allowed us to view the movement of the points relative to each other at discrete frequencies. The most interesting frequency range was found to be at full power and excited by the Voith drive system. This frequency ranged between around 5.4-6.0 Hz depending on ballast conditions. For example, the animated structure shown in Figure 3 below has the bow point (1) in the upper left corner. The other structural locations are labeled accordingly throughout the ship with the stern point (23) at the right. The dotted lines are the at-rest values and the solid lines represent the deviations as the structure moves through its animation cycle. Notice the discrete animation frequency window in the upper right-hand corner with the animation frequency set to resonance at 5.45 Hz.

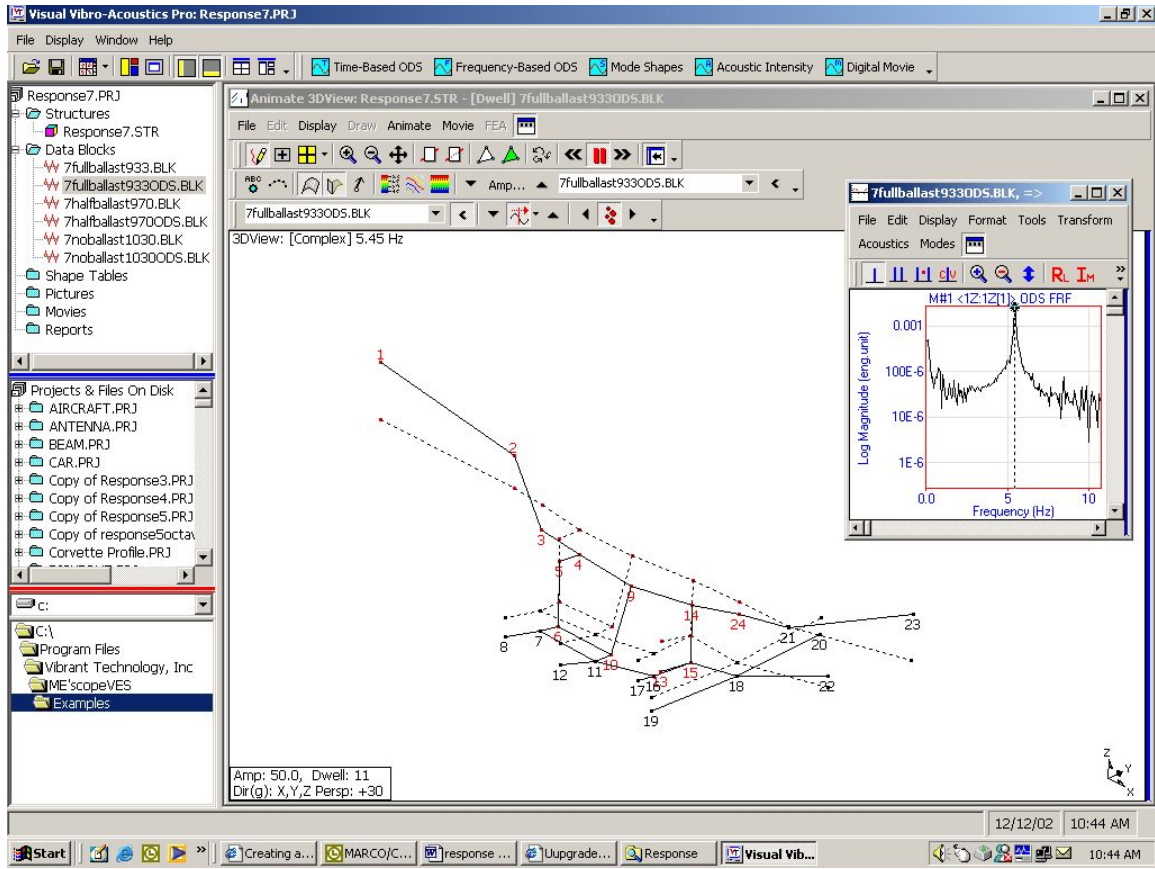


Figure 3

In all cases, the data were collected while underway, later analyzed at Azima DLI's offices and then disseminated out to the interested parties electronically: the graphs in .jpg format and the animations in .avi format.

In the first test, triaxial sensors were located symmetrically about the centerline of the vessel. With 24 channels available to collect simultaneous data, this provided us with only eight triaxial measurement points. After reviewing the animated and graphical data from the first test, it became apparent that the movement at resonance was symmetrical around the centerline of the hull. It was also apparent that the vertical components of the vibration were the only components of interest for this project. These results allowed us to concentrate the 24 points of simultaneous data on half the hull and extrapolate that into 48 points encompassing the whole entire for a much higher resolution and understanding of the structural dynamics than was originally anticipated.

Following the second test, several areas were identified that needed more points to provide still higher resolution animations. These included the engines and engine mount areas, the Voith cycloidal drive and associated foundation, and the engine room stanchion interfaces. Subsequent tests were performed with the 24 points concentrated in these areas, keeping some points redundant from earlier tests so that relative analyses could be made.

Later, strain gages were installed and strain data collected and analyzed to determine relative contributions of certain structural elements to the effective section modulus of the hull. This was important, as the section modulus is one of the primary determinants of the natural frequency of the hull. With such a complex structure, it was helpful to see which components in the hull were contributing to the effective section modulus. Gages were located amidships at the top of the side of the main deckhouse, the top of the bulwarks, the cabin sole, and the top and bottom of the longitudinal bulkheads.

In all, there were nine sets of distinct data taken during six separate sea trials. Each set of data contained several runs with different operating conditions including speed rundowns. Analysis of all graphical and animated data from all tests showed the resonance of the hull being excited by the blade pass frequency of the Voith cycloidal drive units. With this conclusion, two potential options were available as solutions:

1) **Change the blade pass frequency.**

To do this and maintain the same running characteristics of the boat (torque, speed, maneuverability), a new cycloidal drive unit would need to be installed with a different number of blades.

2) **Change the natural frequency of the hull.**

To do this, the structure would need to be modified in such a way so that the new stiffness, mass and effective cross sectional modulus would shift the natural frequency above the drive unit's excitation frequency.

The first option was known to be very expensive and have a six month lead time. To check the viability of the structural changes, the second option, the project's naval architects contracted for a fine mesh finite element model of the ship (See figure 4). Once built, the model was tuned with the previously acquired empirical data. Different modification variants were then modeled and the natural frequency was evaluated. The modifications chosen were: the boxing-in of the bulwarks, the addition of a heavy keel doubler, the addition of two stanchions in the engine room, and the addition of two stanchions in the generator room. This was an addition of 30 tons of steel. The model showed these modifications shifting the natural frequency of the hull higher by about 10% for the half ballast condition. In this case, the natural frequency would shift to 6.26 Hz from 5.66 Hz for the as-built hull. At 6.26 Hz, the natural frequency is approximately 7% higher than the maximum Voith bladerate excitation frequency. Given the response curve measured in the tests, it was determined that this was sufficient to meet the vibration goals. These modifications were completed and the sea trials resumed.

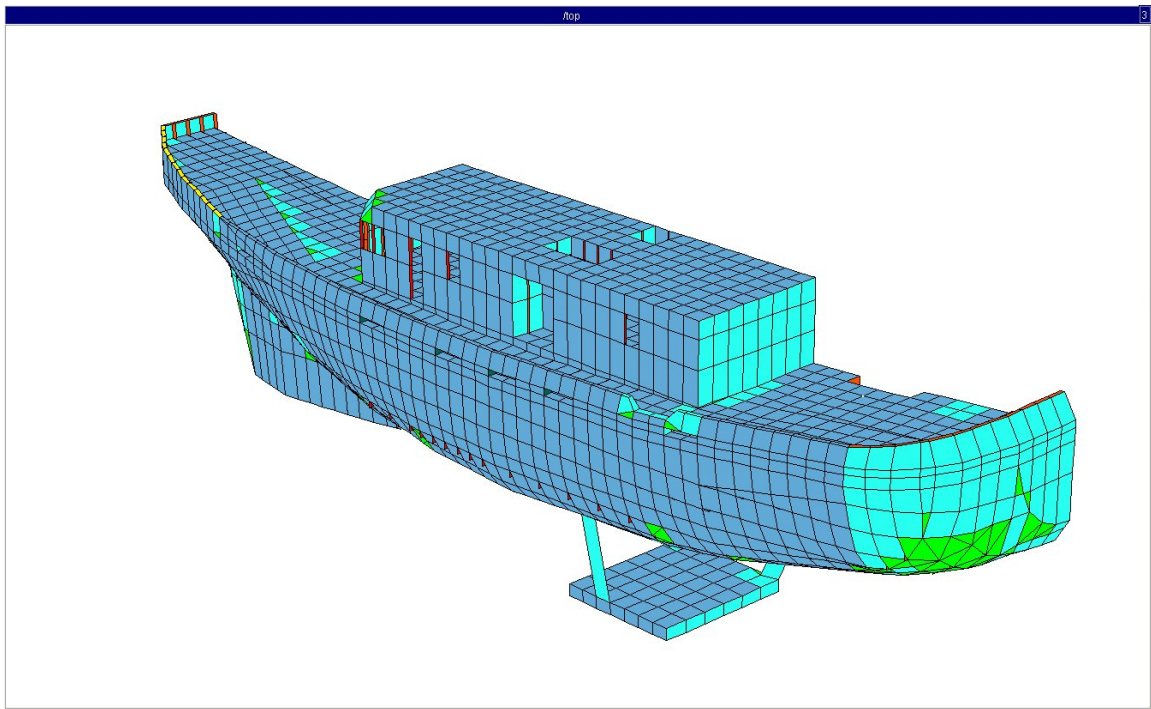


Figure 4

On December 26, 2002, the final vibration sea trial was conducted using a DCX (4 channel analyzer) running signal AQ and ME'scope software. Vibration levels were quantified at two previously measured locations, three different ballast conditions and three different operating speeds for each ballast condition. The modifications were successful. Vibration levels at half ballast and at a top speed of 15 knots (1000 RPM) were reduced at the bow from 39 mils to 12 mils (0-peak). See Figure 5. . Midship vibration levels were about one half of those at the bow. This operational vibration level was acceptable to both the ship's builders and operators.

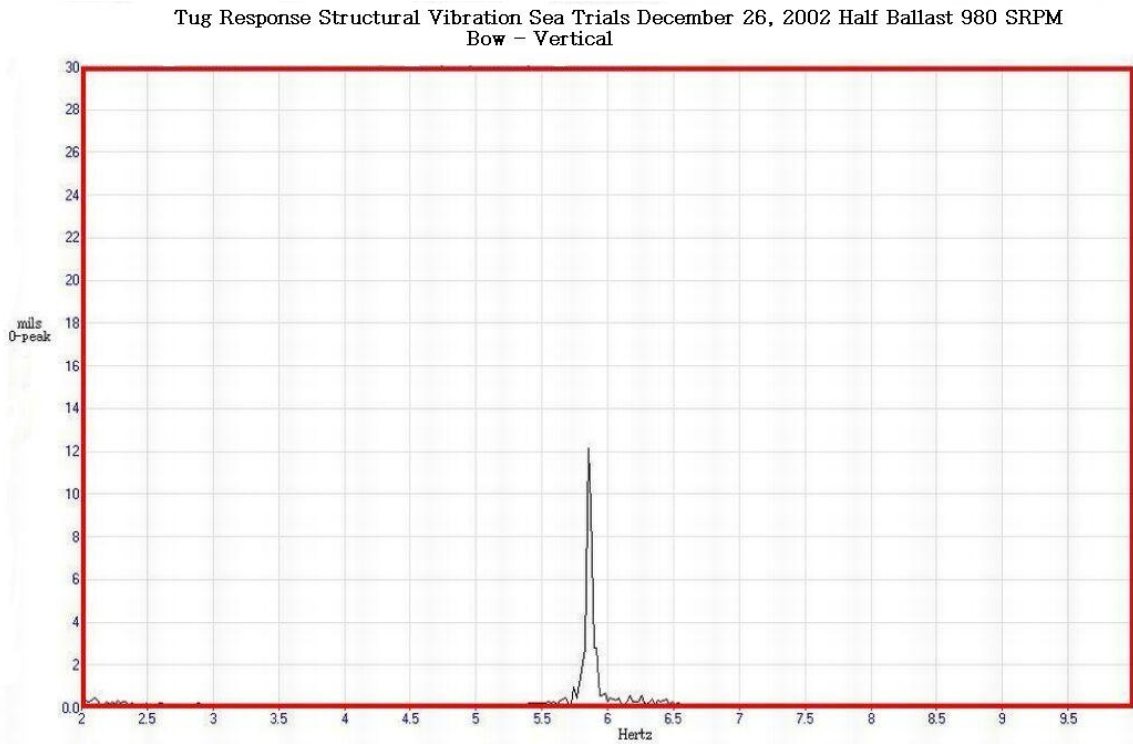


Figure 5

Special Thanks to John Van Buskirk, Crowley Vessel Management Services.