SEAKEEPING

by

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Introduction

The phenomenon known as springing has been observed for many years on Great Lakes bulk carriers and more recently on oceangoing ships. Full-scale tests on board the Great Lakes ore carrier E.L. Ryerson revealed the persistent occurrence of high-frequency stress variations that corresponded to the natural frequency of a 2-noded vertical hull vibration.

The ship data accumulated to date have not been sufficient to draw any definite conclusions, and efforts have been directed toward a theoretical solution to the problem. One of the proposed theories (1) is based entirely on the assumption that springing is the result of 2-noded hull vibration excited by component waves of the sea spectrum that correspond in frequency to the natural frequency of hull girder vibration. The theory indicates that springing depends to a large extent on the deflection shape, the structural damping and the longitudinal distribution of added mass. Since estimates of the first two of these parameters are of limited reliability and it is difficult to obtain accurate records of the waves at sea, attempts to correlate calculations of springing with full-scale observations have not been entirely satisfactory. It was therefore felt that model tests might shed some light on the subject and would serve in particular to check the presently available theories. They would further serve to indicate the sensitivity of the induced vibration to the various controlling parameters. See also (2), (3), (4) and (5).

Since it is difficult to obtain an understanding of springing from model stress measurements alone, the approach adopted at Webb Institute has been a combined model test and theoretical program. Accordingly, the model tests were designed primarily for correlation with theory, rather than to attempt to reproduce full-scale conditions exactly.

A simplification for both experimental and theoretical studies is to consider the excitation produced by one train of regular waves which is exactly in synchronism. Hence, for each speed there is one wave length to be considered. From a practical point of view one need study only the fundamental or 2-noded response of the hull structure, as it will be the dominating mode, though in theory the ship will be excited by the wave forces in all modes of vibration.

The only known previous model tests designed to measure springing were reported by Muckle (6) in his discussion of (1). The model used was that of a 500,000 ton tanker. It was built of wax and jointed amidships, with a flexure bar connecting the two halves.

It is understood that the waves produced were not short enough for resonance, and therefore the natural frequency of the model hull had to be reduced. The damping was varied by means of a simple bellows device, and a figure attached to (6) indicates that the influence of damping, particularly at resonance, is considerable.

Waves in Model Tank

A short description is given below of the actual steps taken to achieve the experimental objective. Further details and results are given in published reports (7) and (8).

The wavemaker mechanism was first tested to determine minimum wave lengths and associated amplitudes which could be physically produced in the Webb tank. It was determined that with the aid of a new linkage unit designed to raise the wavemaker wedge 3 inches above its regular mean position, excellent waves can be generated along the entire length of the tank with periods as low as 0.308 sec., i.e., 5.85 inches length, and wave heights of 0.15 in. This represents a wave length-to-height ratio, λ/h, of about 40, which is ideal for regular wave testing of linear phenomena. The above ratio $\lambda/h = 40$ was maintained throughout most of the tests, allowing proportional increase of wave height with wave length. A few tests in steeper waves were also made to check linearity. The ability of waves with $\lambda/h = 40$ to excite the model was verified under various speed conditions, and springing at the natural frequency of the model was obtained, as well as at lower frequencies closer to the natural pitch or heave frequencies.

For the final tests, in an effort to improve the quality of the waves generated by the wavemaker, the hydraulic system was overhauled and extra stiffening applied to the frame supporting the plunger. The prime purpose of the extra stiffening was to eliminate the transverse wave system which has been observed to develop in the vicinity of the wavemaker, particularly when large wave amplitudes were called for. The substantial amount of extra weight applied to the plunger required the readjustment of the balancing springs attached to the plunger.

Experiments were also performed to determine the ideal depth of immersion of the plunger, involving adjustments in the pressure of the entire system as well as the insertion of a linkage unit between the piston drive and the plunger frame. Changes of up to 4 inches above the plunger's mean regular position were tested, and an ideal location was chosen, using as criterion the quality of the short regular waves generated and the amount of transverse wave system observed. It was felt that, overall, a slight improvement was observed in the wave systems generated.

Particular attention was given to the conditions under which transverse waves were becoming noticeable. It was concluded that the 1/4-1/2 in. clearance between the plunger and the tank wall, roughly 7 to 8 in. in extent, was the prime cause for the transverse waves, and under resonance conditions, i.e., when the wavemaker was set to produce waves of roughly 7 to 8 in. length, the greatest interference of transverse waves was observed. Fortunately, the creation of these interfering waves required time which was usually in excess of the duration necessary to carry out a test run. Furthermore, limiting the wave amplitudes to 0.2 in. or less eliminated the difficulties and resulted in remarkably good wave trains throughout the length of the tank.

Model

The model of Stewart J. Cort was built of wood having a thickness of roughly 3/4". The model was built in two parts, connected by an aluminum bar, with a separation of one-eighth inch (1/8") amidships to allow for model deflections. The weights of the two halves were approximately equal, and the total weight of the two parts was 20.85 lbs. The general characteristics of the model are given in Table 1.

Three loading conditions were determined, based on full-scale weight distribution curves supplied by the designers.

Table 1

Model Characteristics

Scale	144:1
Length overall, in.	83.32
L.B.P., in.	82.38
Breadth, in.	8.71
Depth, in.	4.08
Distance from F.P. to cut, in.	41.19
Weight, lb.	20.85
L.C.G. from midships, in.	0.04 (aft)
Gyradius, in.	25.915

Both the center of gravity and the radius of gyration of the model were measured. The gyradius was determined by swinging each half of the model, as well as the connected full model, in a lateral plane, i.e. in yaw, which is a simplification adopted in the absence of a pitching frame. The gyradius depends mainly on the longitudinal distribution of mass, when ballast weights are concentrated along the center line of the model and the vertical and lateral dimensions of the model are similar. Hence in this case the differences in the gyradius as obtained from yaw and from pitch are negligible. Ballast in each half was adjusted to give the LCG and gyradius specified in (6).

Four different aluminum connecting bars of varying thicknesses, cut from a section of 0.51 x 1.18 in., were used to connect the two halves of the model. Each bar was fitted with strain gages to monitor bending moments at midship. Beam thicknesses varied between 0.25 - 0.69 in., providing a range of natural frequencies for the model between 17-50 rad/sec. (model scale).

Each connecting bar was instrumented by mounting a full bridge strain gage installation which was connected to a Honeywell 1508 Visicorder through a Honeywell 119B Strain Gage Amplifier System. Calibration tests of the bar were conducted while the model was hanging in air supported at the center of gravity of each half. Moments of magnitude up to 60 lb. -in. were applied in both hog and sag conditions, and a linear calibration was obtained for positive and negative moments. 50 lb. -in. was found to be approximately equivalent to a 3-in. trace.

The model was then connected to a towing dynamometer usually used for ship motion measurements, which provided freedom to heave, pitch and roll while restraining surge, sway and yaw. The restraint in surge was not expected to affect the results, because of the high frequency of oscillation at which the tests were to be carried out. The attachment of the dynamometer to the model was not at the exact location of the CG but was at a slightly higher and further aft location because of practical constraints in using a pre-constructed model. This, however, is expected to have little or no effect on the results, as the motion experienced by the model was negligible until wave length exceeded $\lambda/L = 0.3$ (i.e., $L/\lambda<3$).

In addition to the bending moment strain gages, three wave wires were also installed. The first was located roughly 10 ft. from the wavemaker, the second was mounted on the model, two inches ahead of the bow, and the third was mounted on the carriage abreast the bow and about 1.5 ft. to starboard. While the first wire measured the wave trains at a stationary point, the others represented the waves as seen by the model, since little or no motion was experienced by the bow. The wave trace recorded by the model-mounted wire indicated the frequency of encounter of the model in the wave system.

Tests were carried out in regular head seas covering the range of $14>L/\lambda>0.5$, recording the bending moment response at midship as well as the wave pattern at a fixed point in the tank and as seen by the moving model. Forward speed was determined by recording the time taken by the model to travel a certain known distance down the tank. The test runs were concentrated on the combinations of wave length and forward speed required to obtain resonant response conditions. The speed range covered was 0 - 4.0 ft./sec., corresponding to 0 - 28 knots full-scale. The high speed was necessary in order to attain resonant conditions with the stiffest bar at relatively lower values of L/λ.

In order to cover the total response range of the model, waves of $L/\lambda<3$ were also generated. In these long waves low-frequency wave bending response was experienced and recorded for several conditions. One condition of special interest was one in which both high and low-frequency response were recorded simultaneously. This particular condition represents a realistic case often encountered by a ship at sea.

Results

Experimental and theoretical results have been presented elsewhere (7) (8) (9).

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