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WAVE EXCITED VIBRATION: SEGMENTED MODEL DESIGN

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ABSTRACT

A thorough experimental technique for resonant two-node wave excited vibration (springing) towing tank tests is described, with emphasis on the design of the model. The only type of model that can be resorted to is the multisegmented one, both because it is the only one which can achieve a natural vibration profile and also yield the longitudinal distribution of wave exciting and response forces. It is the integration of the two latter times the profile along the length, that will determine the vibratory bending moments and deflections. Hence, this experimental technique is the only acceptable one, since a two halves model provides an artificial, unrealistic, "two straight lines" profile.

The frequency and other structural dynamic properties of the model can be scaled from any desired actual ship. The larger the number of model segments, the better the representation of the full scale case will be. The model is of a "fish skeleton" type with the backbone consisting of a long rectangular beam running along the centerline and representing to scale the structural dynamic properties of the ship. Bending moments and deflections in waves are measured on it. Simple bars, running along the breadth of the model in place of the "other bones", provide the only connection of

the backbone to each segment and act as dynamometers measuring the exciting and response forces; the former in waves with the model prevented from vibrating by using a stiff backbone and the latter in calm water with the model excited in vibration mechanically, based on the linearization of the problem. These are all done on the same model hull using backbones of appropriate stiffness, due to the convenient interchangeability of the backbone. The required experimental work is thus greatly reduced and simplified.

Calculated two-node vibration profile and natural frequency, including sprung segment effects, agree excellently with experimental values.

This type of model design can also be used for other environmentally excited vertical vibratory phenomena, such as whipping of the hull girder due to slamming or bow flare.

1. INTRODUCTION

The results of the tests performed regarding resonant two-node vibratory bending moments and deflections, wave exciting forces and response forces, that is added mass and damping, were reported in (1). There it was shown that excellent agreement is obtained between the measured bending moments and deflections and the calculated ones, when the independently measured wave exciting and response forces are used as inputs to the calculations. Thus, the experimental results verified the linearization of the problem and showed that springing can be fully simulated experimentally with reliability.

The present paper deals with the experimental technique and, mainly, with the design of the model. Multisegmented models were used previously, such as in (2) - (4), but, mostly, for the quasi-static frequency regime and, then, only for the bending moments and deflections, i.e. the results of the phenomena. General descriptions of principles relating to multi-segmented models were given in (5) and (6) and a model, on which only exciting forces for springing were measured, was described in (7). Here the purpose is to provide an experimental technique and one and the same model on which, for the frequency regime of fundamental vibration, all aspects of the phenomenon could be measured; that is, in addition to bending moments and deflections, wave exciting and response forces, i.e. added mass and damping, as well, including the longitudinal distribution of response and exciting forces.

2. GENERAL DESCRIPTION OF THE EXPERIMENTAL TECHNIQUE AND MODEL

It is self evident that in order to be able to ascertain the distribution of exciting and response forces a segmented model must be used. The need for a segmented model is also imposed by the fact that, the frequency of a model in one piece would be too high, and it would not be possible to use such a model, as it would be very stiff to resonate with waves, which could be generated in a tank (19). Further, the question of what material would be more

suitable to use for a geometrically similar one piece model, in which the natural frequency would also be scaled, was examined in (8) from the structural point of view. It was concluded that, even if different scaling factors were used for the main dimensions and for the skin thickness, for the range of materials available the required scaling relation $K_L = \sqrt{K_E \cdot K_t}$ could not be satisfied, while retaining a workable model skin thickness. It is, therefore, seen that, by a "happy coincidence", a whole model could not have been constructed even from the structural point of view, let alone the impossibility of making it resonate with waves.

Additionally, a segmented model can achieve a "natural" normal mode shape, instead of a "two straight lines" one, which is the case for a model cut in two halves. It was made clear in (1) that this is an important consideration, because the results depend on the integration of various quantities together with the non-dimensional profile along the length of the model, an integration which is very sensitive to slight changes in the profile. Thus, the only resort is the segmented model.

As hull vibrations are here of concern, the model must represent adequately the basic ship properties which affect such vibrations, that is bending stiffness, displacement and length. In order to simplify the experimental procedure, it is highly desirable to use the same model design for bending moments, as well as wave exciting and response forces tests. During the bending moment tests the actual bending behavior must be simulated and the model flexes under the influence of the waves. To measure exciting forces, the model must be artificially stiff so that it does not vibrate and the exciting forces are not consumed in producing the vibration.

The first requirement can be met by providing a "backbone" beam, which represents to scale the structural bending stiffness properties of the model. The bending moments must be measured on this beam. The model segments are then connected to the beam to provide geometric similarity. The second requirement is to prevent the beam from flexing and the exciting forces have then to be measured on the connections of the model segments to the beam, there being one such connection only on each segment, i.e. one force "flow path". Thus, each segment provides the net force, or integrated pressure over its wetted surface.

A design satisfying both of these requirements consists of a solid rectangular "backbone" metal beam running along the center line of the model and located, vertically, symmetrically about the scaled position of the neutral axis. Its dimensions, i.e. its EI, are determined on the basis of the scaling law for the natural frequency, the latter being proportional to $\sqrt{EI/\Delta L^3}$. In order to achieve the largest moment of inertia about a horizontal axis with the least amount of material, the larger cross-sectional dimension of the beam is oriented in the direction of the depth of the model. The cross-members connecting the segments to the beam are located at the mid-length of each segment[1]. They are cylindrical metal

[1]Except, perhaps, for the aftmost and foremost segments depending on any limitations imposed by the hull shape.

rods, running transversely at the height of the neutral axis from port, through holes in the metal beam, to starboard. They are clamped by flat bar clamps, both at port and starboard, screwed onto angled brackets attached to the wooden model segments. Thus, the rods connect the beam to the segments without restraining the beam from flexing vertically, while at the same time providing restraint against any unwanted "snaking". To help visualize the set up, one can think of it as a "fish skeleton", with the main bone being the metal "backbone" beam and with the cross-members representing the other bones (except that they are oriented perpendicular to the backbone). The metal beam is strain gaged amidships with foil strain gages to measure the bending moments.

For the exciting force tests, the metal beam is replaced by a much deeper, i.e. stiffer, one with the result that the natural frequency of the model is higher and it does not vibrate under the influence of the waves. The natural frequency of the model with this stiffer backbone would have to be at least four times larger than the highest frequency of encounter during the tests; so would also have to be the natural frequency of each segment. In this case then, the rods connecting the segments to the beam deflect statically relative to the beam, and, being instrumented with foil strain gages, they measure the exciting force on each segment. A plan view of the model is given in fig. 1.

At the same time, this set up provides for the measurement of the response forces on the cross-members using the same beam as that used for the bending moment tests, but with the model excited mechanically in calm water.

From the point of view of obtaining accurate lengthwise distributions of exciting and response forces, the larger the number of segments the better. This depends on practical considerations, such as the length of the model, which in turn depends on the size of the towing tank available and ease of ballasting of and access to each segment. In order to afford easy access for interchanging the beams, and since only the beams represent structual properties, the model segments are open girth extending only up to the molded depth line. The model is made of wood and then cut into segments. Two removable wooden straps at the top of each segment connect the port and starboard sides to help prevent any twisting of the segments.

The aftmost and foremost segments extend outside the length between perpendiculars and it is sufficient for the cuts (gaps), located at the forward part of each segment, to be 0.125 in wide. They were sealed with 1/64 in thick nitrile rubber and all purpose plastic tape for further waterproofing. The seals were glued to the wood using a glue made of a mix of Aerolite 306 powder glue and a GBP.X Aerolite hardener. A 3/8 in fold was allowed in each gap so that the seals provided no constraint between segments. Thus, each segment is, effectively, independently attached to the beam. In previous cases, where segmented models were used with the gaps sealed with flexible materials, such as in (3), no effect of the slits was found on the measurements. The same situation applies here as well. For example, the measured fundamental frequency of the model with the seals made only of the nitrile rubber and

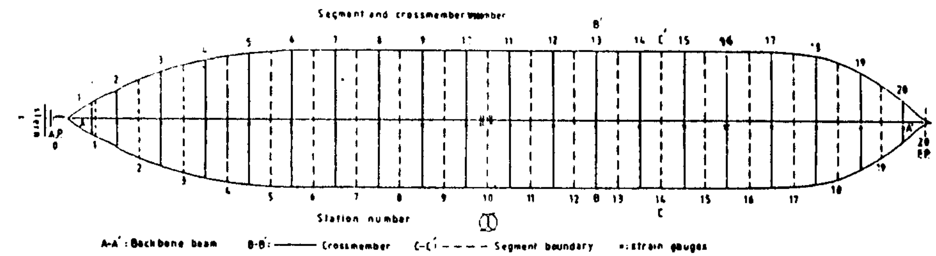

Fig. 1 Model plan view at N.A.

then made with both the rubber and tape agreed within 0.6% to each other. In addition, theoretical predictions of this frequency, which, of course, do not take into account the existence of the seals, agree excellently with the experimental values, as seen in the next section. Though it must be obvious, it is better to state explicitly that the ends of the backbone beams at the bow and stern were not attached to the hull. Each segment is attached to the beams only through the cross-members.

To ensure that the ballast distribution is accurate, the ballasted model was tested on an inertia table and the required pitch radius of gyration was obtained, together with the LCG of each segment, which in most cases was very close to the mid point of each segment. Thus, it was ensured that the longitudinal weight distribution was accurate. (The vertical weight distribution was not duplicated exactly since one was interested in motions in the vertical plane running along the ℄ ; in addition practical considerations such as the location of the backbone beams and cross-members prevented it from being attained. However, this does not influence the experiments).

3. SCALING OF THE MODEL, ITS FUNDAMENTAL NATURAL FREQUENCY AND THE FREQUENCY OF EACH SEGMENT

From the point of view of being able to operate with as long waves as possible, it is desirable to have as long a model as possible. On the other hand it is also desirable to avoid any shallow water effects in order to eliminate any interference with the results of the experiments. Thus, the consideration limiting the main dimensions of the model in any tank is, automatically, the maximum model draught consistent with the maximum possible tank water depth in order to avoid shallow water effects, that being 1/5 of the water depth. This ratio yields the scaling factor and the other dimensions of the model follow from the corresponding ones of the full scale vessel.

As is well known (1), the natural frequency of the model must be scaled from that of the full ship according to a particular law, sometimes referred to as Cauchy's Law, i.e. $(\omega_{em}/\omega_{es})^2 = L_s/L_m$

It is derived from Froude scaling, the equation for frequency of encounter and the fact that $L_m/\lambda_m = L_s/\lambda_s$.

At resonance the frequency of encounter equals the natural frequency of the ship or model. And since the frequency of encounter of the model is to be the same fraction of its natural frequency as the frequency of encounter of the ship is to the ship's natural frequency, together with the fact that the natural frequency varies as $\sqrt{EI/\Delta L^3}$, taking only bending effects into account and with Δ proportional to L^3, the ratio of model to ship bending stiffness varies as the fifth power of the length ratio, i.e.

$$(EI)_m/(EI)_s = (L_m/L_s)^5$$

Thus, from the full scale I one can find the I and, hence, the cross-sectional dimensions of the backbone beams.

In the particular case of the VLCC model used in (1), the backbone representing the full scale value would have a natural frequency of 4.13 Hz and a moment of inertia of 0.679 in^4. Now it must be noticed that the above does not take into account shear effects. These are very small for the two node mode and are usually neglected. However, if they are included, the scaling variation would be $[L_m^2/(EI)_m]+1/(KAG)_m=(L_s/L_m)^3[((L_s)^2/(EI)_s)+(1/(KAG)_s)]$. For a box section with beam and depth those of the full scale VLCC of (1), using an average uniform thickness derived from its plans, the I of the backbone beam of the model would be reduced to 0.661 in^4. Now it must also be clear that such a slender beam as used in the model would exhibit much less shear deflection than the actual ship's girder. Thus, in the model the term $1/\sqrt{1+r_s}$ for she shear correction to the natural frequency would not obtain and the model's frequency would be higher than the expected value by this amount. In order to compensate for this effect, it was assumed that $\sqrt{1+r_s} = 1.1$ and the decrease in frequency was "enforced" by reducing the I of the backbone beam by $(1.1)^2$ so that, finally, its I =0.546 in^4. With the maximum speed of the full scale vessel at 16 Kts the maximum scaled speed of the model would be 2.99 ft/sec and, using the frequency of encounter relation, it would resonate with a wave of 16.33 in at the maximum speed and a wave of 3.59 in at zero speed, for the 120 ft long towing tank used (1).

Two other more flexible beams, B1 and B2, were designed specifically so that they would resonate with longer waves at zero speed, one with a 6 in wave and the other with a 10 in one. The final I's for these two cases, going through the considerations mentioned above, were 0.326 in^4 and 0.196 in^4. All three beams were designed to be 0.5 in wide so that the same connecting cross-members could be used in all cases. Their depth was varied to obtain the different I's. Their characteristics are summarized below:-

	f nat'l Hz	I req'd. in^4	b in	d in	I act. in^4	f nat'l, due to fresh water, Hz
B1	2.44	.196	.5	1.65	.187	2.47
B2	3.22	.326	.5	1.99	.326	3.26
B3	4.13	.546	.5	1.33	.536	4.18

The column before last gives the actual I achieved during construction and the last column gives the frequency values due to the less actual model weight when floating in the fresh tank water.

All of the above considerations assume that the factor C in the equation $\omega = C\sqrt{EI/\Delta L^3}$ is the same for both parent and model. Only the inability of the backbone beams to simulate shear has been taken into account. However, the simple construction of the model, which is intended to duplicate the main characteristics of the wave induced vibration and not all the structural details of the ship, differs from the full scale in other aspects, which influence the value of the two node natural frequency. These are, whether LBP of LOA should be used in calculating frequencies and the fact that in the parent LOA flexes, while in the model the flexure is over the extent of the beam only, which is about equal to LBP, the actual E value which for the model was judged to be 29.7 x 10^6 psi, while the exact of the parent was not known and was presumed to be 30 x 10^7 psi, and the effect of the actual I, when the beams were constructed, versus the required one. In the particular case considered here, another factor was the uncertainty range of the frequency of the full scale. Taking the limits of the effects of these considerations, which will produce the lowest natural frequency values, the frequencies of the model for each beam case will be reduced to 2.16, 2.91, and 3.69 Hz for B1, B2 and B3 correspondingly.

Two other factors, which will also further reduce the frequency, are the variation in I along the parent vessel length, which is not depicted in the model, where the backbones are of a uniform cross-section, and the effect of the superstructure bridge, which is also non-existent in the model. The effects of these two considerations may create a reduction to the model's natural frequency of up to 2% and 5% correspondingly (1).

The nature of the model, a beam with attached masses, lends itself to the application of Rayleigh's method for calculating the frequency by simply treating the segments as masses rigidly attached to the beam. A first shot calculation in this way, assuming a sinusoidal profile and an added mass for each segment equal to the weight of the segment, (admittedly rough, but, it is emphasized, only a first shot calculation), resulted in frequencies equal to 2.04, 2.69, and 3.48 Hz for B1, B2, and B3 correspondingly. If these values are compared to those quoted just above, it can be seen that the two sets of values are within the 7% reduction of the "variation of I along L" and "superstructure" effects; that is the previous values with an additional reduction of not more than 7% will be very close to those predicted by the "rough" application of Rayleigh's method. This is to be expected, since it is the various factors mentioned above, which cause the model to differ from the actual ship structure, and render it in the form of a beam with attached masses.

The proper calculation of the natural frequency of the model depends on knowledge of the added mass, the measurement of which was an aim in (1). However, for the purpose of calculation of the natural frequency, it was assumed that the knowledge provided by the theoretical two-dimensional added mass calculations, coupled with appropriate J factors to allow for three dimensional effects, was sufficient. The 2D values were calculated by (9), which is based on Frank's method (10) and three different J factors were used, i.e. Andersson and Norrand's, Kumai's and Townsin's (1).

Rayleigh's method was applied to the model (potential energy of backbone beam = kinetic energy of backbone beam + kinetic energy of the segments; i.e. only the backbone flexes) using segmental added masses derived as described with each J value and using a non-dimensional profile given in (11). The results are given below:

J	And. + Norr.	Kumai	Townsin
B1	2.10	2.12	2.13 Hz
B2	2.75	2.78	2.80
B3	3.53	3.56	3.57

It can be seen that the differences between the frequency values, derived from the three different J's are small. This was to be expected as, though the added mass distribution for each J is quite different, the frequency depends more on the total added mass than it does on its lengthwise distribution. In addition, estimates of frequency using these J's, particularly Townsin's one, have been used extensively yielding good results. Townsin's J is, in fact, based on experiments measuring natural frequencies in air and water, where all the difference in frequency is attributed to the total added mass. Thus, it should be expected to yield good results for frequency calculations. (In contradistinction it is the distribution of added mass which is important when calculating bending moments in the context of this work, as explained earlier).

In order to check the frequency calculations further and to obtain an exact calculated non-dimensional profile, which it was desirable to have because the one given in (11), as pointed out in (1), does not satisfy exactly the boundary conditions, a digital method described in (12) and based on (13) was programmed. It is a difference equation method satisfying from segment to segment the shear force, bending moment, slope and deflection equations. The method was run with all three J's mentioned above. The results, given below, are very close to those of Rayleigh's method in conjunction with the non-dimensional profile of (11) given just above.

J	And. + Norr.	Kumai	Townsin
B1	2.10	2.13	2.12 Hz
B2	2.75	2.80	2.78
B3	3.53	3.57	3.56

The very small difference between the two sets of results indicates that, simply, one method verifies the other. The difference between the profiles for each beam resulting from the difference equation method is minute, as expected, because all three beams are uniform and the weight distribution of the model varies very little from beam to beam.

The difference equation method was used to check whether factors like shear, LOA versus LBP, and the variation of the cross-section of the beams in the aftmost segment due to limitations imposed by the hull shape would have any appreciable effects on the natural frequency values. As expected, the differences caused in natural frequency values were found to be insignificant.

Further, it is to be noticed that both sets of results given above in tabular form, i.e. Rayleigh's and the difference equation one are close to the two sets of frequency values for B1, B2 and B3 given earlier, i.e. that based on the "first shot rough calculation" and that resulting after consideration of all the factors peculiar to this model design (see p. 7). Notice that the first three of these result from calculating the frequency of the model independently of the full scale ship and based on first principles only, while the fourth results from scaling the fundamental frequency of the full scale ship and taking into account the characteristics peculiar to the present model design. This close agreement provides an additional means of cross checking and validating the procedures followed herein.

The final and important consideration with respect to calculating the fundamental frequency of the model, which is peculiar to the present model design, since it is composed of one beam and attached masses, has to do with whether the segments are, indeed, "rigidly" attached as assumed in the above Rayleigh and finite difference methods, or are "sprung" from the metal beam through the connecting cross-members. The latter is a more realistic possibility and it remains to determine which segments are sprung and how much they affect the model's overall two node natural frequency. This is a relative matter and it depends on the natural frequency of each segment, when the point of its attachment to the beam is held fixed.

A means, very suitable to the construction of the model, for checking this is afforded by Dunkerley's method as described in (14). Briefly, the method gives the "net" fundamental frequency of a "total" structure in terms of the natural frequencies of its component parts (the frequency for each part being calculated independently of the other parts), in our case, the backbone beam and each segment. The accuracy of the method depends on the assumption that in beam vibrations the natural frequencies of the three node and higher modes are often considerably greater than that of the fundamental. Thus, the effect of these higher modes may be omitted and the fundamental frequency of the "total" structure ω_1 is given by

$$1/\omega_1^2 = 1/\omega_{11}^2 + 1/\omega_{22}^2 + 1/\omega_{33}^2 + \dots$$

where ω_{11}, ω_{22} and so on are the natural frequencies of each "mass" separately in the absence of the other masses. The method can be extended to any number of degrees of freedom and, in the equation above, ω_{11} would be the frequency of the backbone beam alone and each of the other terms would represent the natural frequency of each segment.

In ships, the frequencies of modes higher than the fundamental are less, in proportion to the frequency of the fundamental mode, than they are in a simple free-free uniform beam (1). For this reason Dunkerley's method has been used here with a correction of 7% to compensate for the effects of higher modes. This correction is derived from applying the method to a uniform free-free beam for which the ratio of the frequencies of higher modes to that of the fundamental is known. (Data on the frequencies of higher modes for the full scale vessel was not available to attempt to estimate a more accurate correction factor).

In order to determine which segments are effectively sprung and which are not, Dunkerley's method can be used in conjunction with Rayleigh's for a given model segment. Starting with the beam alone and attaching first one segment, (the sequence of attachment is immaterial and the results will be the same) one can calculate by Rayleigh's method the "net" frequency of the "total" structure, i.e. that of the beam plus the segment (the added mass is included in the total mass of the segment used). Then, using Dunkerley's equation one can find the "frequency" of that segment, i.e. (for the first segment only):

$$1/\omega_{22}^2 = 1/\omega_1^2 - 1/\omega_{11}^2$$

where ω_1 is the "net" frequency of the "total" structure found by Rayleigh's method and ω_{11} is the frequency of the backbone beam alone. This ω_{22} can then be taken as the frequency the mass would have to have to be considered as "rigidly" attached. If then the relation $\omega_{22} = \sqrt{K_{22}/m_{22}}$ is used, together with the mass m_{22} of the segment, K_{22} can be found and this would then be the "effective spring for the mass to be attached rigidly". The process can be repeated in sequence for all segments. Comparing then the "effective spring for rigid attachment" to the actual spring constant of each connecting cross member derived as that for a clamped-clamped beam, the segment is considered to be sprung, when the latter is less than the former.

In this way it is found that relatively few out of the twenty segments are "sprung". They are concentrated near the nodes of the model vibrating as a whole, because in that region the "effective spring for a mass to be considered rigidly attached" is higher due to the fact that the motion of the beam in the two node mode is the least. If then the frequencies of each of these "sprung" segments are used in Dunkerley's equation, with the frequency ω_{11} being that of the "total" rigid model (i.e. beam plus twenty segments) derived from either Rayleigh's or the difference equation method, the final two-node natural frequency of the model will be calculated. The results are given below for the profile from (11).

	B1	B2	B3
J Andersson and Norrand			
Rayleigh	2.10	2.75	3.53 Hz
Dunkerley	2.09	2.54	2.81
J Kumai			
Rayleigh	2.12	2.78	3.56
Dunkerley	2.14	2.62	2.92
J Townsin			
Rayleigh	2.13	2.80	3.57
Dunkerley	2.12	2.59	2.88
Experimental	2.15	2.52	2.92

It is seen that the "sprung" masses have no real effect on the frequency of B1, i.e. the most flexible beam, whose "Rayleigh" type of frequency (all segments rigidly attached) is the furthest away from the natural frequencies of the segments alone, as expected according to (15), while the effect increases as one goes to B3, where

it is about 18%. These values are compared with the experimentally derived natural frequencies and the agreement is very good.

A further refinement in the method can be made as follows: When the frequency with all segments rigidly attached is calculated, either by Rayleigh's or the difference equation method, all segments are taken into consideration. Then, when Dunkerley's method is used, including only the "sprung" segments, it may seem that the "sprung" segments are taken into account "twice". This is not really so, and it can be seen as follows: To be strict one can first determine the frequency of the model including only the segments that are found to be rigidly attached (in the sense of the "effective spring for rigid attachment" described above). Then, to the frequency thus determined, Dunkerley's method can be applied, including, of course, only the "sprung" segments. Thus, the latter are taken into account only "once". The results are given below and it can be seen, as it might be expected, that the differences are very small. This is simply because, when in the first step, i.e. in determining the frequency of the model including only the segments that are found to be rigidly attached, the "sprung" segments were omitted, the resulting change in the total kinetic energy of the net structure, and, hence, the change in frequency, is small, because the "sprung" segments are located near the two nodes and have, therefore, individually and in comparison to the other segments very small kinetic energies.

		B1	B2	B3
And. & Norr.	J	2.10	2.57	2.86 Hz
Kumai	J	2.15	2.64	2.97
Townsin	J	2.12	2.62	2.94
Experimental		2.15	2.52	2.92

A last consideration relates to using the "effective" mass in Dunkerley's equation for each "sprung" segment, i.e. the segment mass reduced by the factor $1/1-(\omega/\omega_0)^2$ according to (16), where ω is the natural frequency of the structure with rigid masses attached only and ω_0 is the natural frequency of the "sprung" segment. The results according to this are given below and it can be seen that the same overall agreement is obtained with the experimental values as before. This effect is, therefore, considered small. It is considered that Kumai's J gives the best agreement with experiment. The agreement for B1 and B3 is excellent and for B2 very good, given that in some cases the difference between the natural undamped frequency, which is what the theory predicts, and the natural damped and the frequency of maximum forced amplitude (or resonant one), which are what the experiments yield, may be up to 4% (1).

		B1	B2	B3
And. & Norr.	J	2.10	2.54	2.77 Hz
Kumai	J	2.14	2.62	2.89
Townsin	J	2.12	2.59	2.86
Experimental		2.15	2.52	2.92

It is seen, therefore, that in the present design of the model the possibility of segments being sprung must be taken into account. The effect of the sprung segments on the model's natural frequency will be less, the stiffer the connecting cross-members are. However, since these same members are used to measure exciting and response forces, this would minimize their "effectiveness" in this respect, because the signals during the response and exciting force measurements will be reduced. In order to make the calibration of these cross-members for exciting and response forces easier, it is desired (1) for them to deflect more or less statically durint the tests; this would necessitate that their natural frequencies be at least four times larger than the closest frequency of encounter used in the tests; in this case the natural frequency of B3. Apart from that, the final natural frequency of the model is not known apriori because it, in turn, depends on the natural frequency of the segments. One is, therefore, faced with a "looping" design cycle between the frequencies of the segments and the frequency of the model as a whole.

4. CONCLUSION

In conclusion, it can be stated that the design of the model to depict the characteristics of the structure, which are of importance with respect to springing, will lead to an unavoidable reduction in the final value of the natural frequency of the model from the value scaled from the full scale ship. There are two kinds of effects, which cause this reduction. The first is related to structural aspects of the full ship not duplicated in the model, i.e. those mentioned in the beginning of this section. The second is the "sprung" segments effect. Both are unavoidable. They can be compensated for if one starts with a backbone beam, which will be quite stiffer than required by scaling. A looping design procedure would be involved, in order to duplicate a given ship in a given loading condition, given the interplays mentioned above. However, this is not necessary if one is interested, only, in examining the basic phenomenon under realistic conditions. For this, the final model used would have to reflect the basic characteristics of springing for a ship, or a range of ships, which are realistic, i.e. in existence at present. For example in (1), so far as B3 is concerned, the reduction of its natural frequency to its final value, though of about 29% increases the resonant wavelength at maximum speed from 16.33 in to 24.5 in and that at zero speed from 3.59 in to 7.04 in, changes which, though in themselves considerable, are insignificant, really, with respect to the model length, with the largest L/λ ratio still equal only to 0.1531.

Therefore, the resulting model used in the tests in (1) can be viewed as corresponding to a 280,000 dwt tanker, more flexible than the full scale ship, or, alternatively, as representing very closely, under different scale factors, a 553,000 dwt tanker (17), or, alternatively, a 500,000dwt one (18) (provided that the two latter have the same lines as the parent vessel used herein). The important aspect is that the full agreement between theroretical and experimental natural frequencies has been established, contrary to the previously obtaining situation with respect to springing (1).

5. NOMENCLATURE

E Young's modulus. I Moment of inertia. Δ Displacement. L;LBP; LOA Length of ship, model; length between perpendiculars; overall. L_m, L_s length of model, ship. f frequency. $\omega;\omega_{em},\omega_{es}$ circular frequency; of encounter of model; ship. $\lambda m;\lambda s$ wavelength relating to model; ship $(KAG)_m$; $(KAG)_s$ shear stiffness of model; ship. r_s shear deflection correction factor. b;d breadth; depth of the backbone beam, J added mass 3D reduction factor. Scaling factors K_L, K_E, K_t, = L, E, t (skin thickness) full scale/L, E, t model, correspondingly.

6. REFERENCES

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