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◉ 論 文

Effect of Flexibility Variations on Ship Responses

Young-Sub Kwon*

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강성변화의 선체응답에의 영향

권 영 섭

Key Words : Hydroelasticity analysis(유체탄성 동역학적 해석), Flexibility(유연성), Stiffness (강성), Impact Slam-induced Bending Moment(충격 선수 굽힘모멘트), Momentum Slam-induced Bending Moment(모멘텀 굽힘모멘트), Steady State Responses(정규응답), Transient Responses(과도응답)

초 록

선체응답의 변화를 고찰, 계산 결과를 요약하였다. 이에 대한 계산은 유체 동력학적 해석을 이용하였으며 본 이론의 합리성을 아울러 지적하였다. 선체의 유연성을 증가시킴으로서 충격선두 굽힘 모멘트는 줄지만, 모멘컴 굽힘 모멘트는 일반적으로 증가함이 나타났다.

1. Introduction

The last couple of decades have seen the relatively sudden demand for larger, faster and different types of ships. In addition, the trends to lighter scantlings brought about by improved coatings, high-strength steels, better knowledge of loads, etc must be considered. These changes have, naturally, resulted in structural and hydrodynamic characteristics which were beyond the range of past experiences. These, in turn, have resulted in some unexpected features in ships' responses to static, as well as, dynamic loads. All these changes, as can be surmised, brougth about new problems or, more correctly, problems that were not so important in the past have to be considered and tackled. For instance, the experience of the, so called, 'springing' phenomenon gave rise to questions on whether the structure of a ship could still be treated as a rigid-body or not and the validity of the existing methods[1,2].

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* Research Student, dept. of Mech. Eng., Brunel Univ., Uxbridge, Middx., UB8.3PH, U.K.

The implications to ship designers and researchers were that more exact theories and refinement of the analytical design procedures were necessary in order to comply with the fundamental principles of the ship response mechanism and the consequent features of the changes. In this context, it may be gagued that the hydroelasticity analysis, used in this paper, is one of the most promising methods which can accommodate trends and changes in naval architecture and provide a useful and feasible design tool.

2. Effect of Flexibility Variations

One of the most imminent consequences of new trends in ships was the reduction made in the inertia of the transverse section of the hull. Fig. 1, taken from reference(3), illustrates the reduction in basic section modulus requirements for tankers which has taken place since 1959. Assuming no changes in the location of the neutral axis, the reduction in inertia is of the same order of magnitude as the reduction in section modulus. Assuming that basic ship dimensions remain unchanged, this implies an increase in flexibility and greater hull deflection. The degree of flexibility is vital to many considerations in ship design such as :

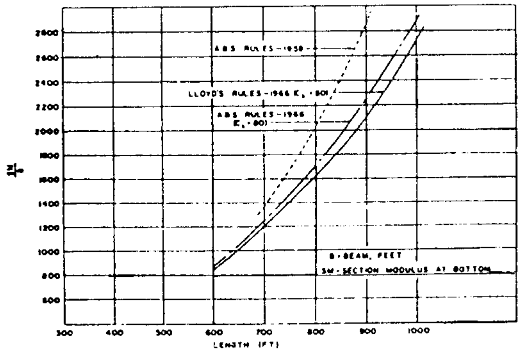


Fig. 1 Tanker basic hull girder strength requirements(ref.(3))

(a) Hull deflection
(b) Stress
(c) Metal fatigue
(d) Loads applied to non-structural components such as piping, shafting systems and usperstructure.

In ganeral, items related with deflection itself, e.g. items (a) and (d) are known to be amenable to control by careful design modification[4], although the question on the local flexibility(i.e. relative deformation of the hull) should be, justifiably, addressed[5]. In the extreme case, its effect is a lower Plimsoll Mark in the sagging condition with resultant loss in deadweight. This is a penalty which has to be accepted of loading must result in a sagging condition[6].

Metal fatigue, in which, for many years. shipbuilders have shown little interest, is a rather controversal subject of ever increasing importance. It could be argued that fatigue failures are neither as spectacular nor as catastrophic as brittle failures[7] and not all cracks are, inevitably, dangerous. However, the application of higher strength steels and the accompanying weigth reductions allowed by calssification societies changed the situation. From a fatigue point of view, these allowences, despite confliction opinions, do not provide any improvement and are too optimistic[8]. Although the topic is outside the scope of the present work, the urgent need for research on this problem is highlighted by reports of warnings and recalls given by classification societies and shipbuilders to recently-built VLCCs[9,10].

A most important consequence of the trends in hull stiffness is the change of the dynamic as well as particular high frequency responses of the primary structure, which affects the relevant loadings on the structure. For instance, one of the significant phenomena ships experienced-once size had increased-was wave-excited hull vibration or 'springing'. This was known to happen not only to Great Lakes bulk carriers but also to ocean-going cargo ships, e.g. bulk carriers and tankers[11]. As Kline[12] speculated, this was a hydroelastic problem identified as being primarily one of fundamental hull frequencies matching the encounter frequency of waves possessing sufficient energy for hull excitation. This phenomenon of wave-excited main hull vibration was mainly related to the two-node natural frequency of ship's vertical vibration-a function of hull structural properties and mass distribution-and was the subject of many investigations[13-16].

On the other hand, whereas springing is a steady state response of the ship, dynamic response due to transients such as slamming loads are relevant to hull flexibility considerations. Slamming, in head waves, causes a sudden vertical acceleration and deflection of the bow and excites flexural vibration of the hull girder, mainly in the fundamental two-node mode but also in highter modes[17]. Local damage to the shell plating and supporting longitudinals and floors will be of immediate concern. Nevertheless, one should account for the effects to the hull girder bending resulting from a slam[18].

3. Scope of Investigation

The aim of this research is to provide practical design information concerning the influence of hull bending stiffness on the steady state and the transient(slamming) responses of a ship in waves. To achieve the aim, systematic manipulation of relevant data are simulated and examined for a tanker with various speeds and seaways(see following sections to refer to descriptions on model ship and seaways). The linear hydroelasticity analysis laid down by Bishop and Price[19] has been applied, utilizing the existing numerical methods[20,22].

Accordingly, the bending stiffness of the hull structure was systematically altered by varying the cross-sectional second moment of area over the entire length of the hull as shown in Table 1. A variation in the second moment of area may also result to changes in the effective shear area. As a compromise, the shear stiffness was varied, in the present study, by the square root of the bending stiffness variation as shown in Table 1[4]. Any changes in hull steel weight are assumed to be compensated by other elements of lightweight or deadweight, so that the mass distribution remained unchanged. Only ballast condition is examined for both steady state and transient loads. In fact, this topic is much more relevant to transient rather than steady state analysis, with the exception of 'springing'.

Table 1 Bending and stiffness variations

Bending Stiffness	%	60	80	100	120	140
Shear Stiffness	%	77	90	100	110	118

Moderate slams are simulated, i.e. slamming length of 0.075L, because it is believed that for the VLCCs, in any condition, the degree of severerity of slamming experienced would not be extremely high. Details of the simulation of slamming will be presented in Sec.6.1.

3.1 Model ship

A tanker(VLCC) in ballast condition was used as a model ship in the present investigations and the hull structure is idealised as a non-uniform beam with varying properties along its length which is divided into 50 slices of the same thickness. The general characteristics and some of input data are listed in Table 2. Fig. 3 shows mass and buoyancy distributions for the ballast condition and also structural properties of the model ship. It should be noted that the ship is slightly trimmed by the stem.

The reason why a tanker has been used as the subject of the present study is, not necessarily due to safety problems in that type of ships[25,26], but merely as a convenient example with available data. It may be argued, however, that the topics considered and their results can be, in principle, applicable to some other ship types. Meanwhile, it should be helpful to refer to the eminent features of a large tanker for better perception of the present research in relation to the model ship. These are as follows :

(1) Tanker is the type of ship which has undergone significant changes which affect flexibility and, consequently, its vibratory response.

(2) The number of cracks and other types of damages, particularly in the larger tankers, are known to be severe[27]. In addition, the real problem of corrosion, despite the progress of corrosion protective technology, is quite alarming[26].

(3) Ever increasing concern with oil pollution puts increasing pressure to review the hull structure of tankers.

We may, herein, note that some of the above features may also be relevant to other types of the ship, such as,(dry) bulk carriers, which have similar dimensions and their design is driven by

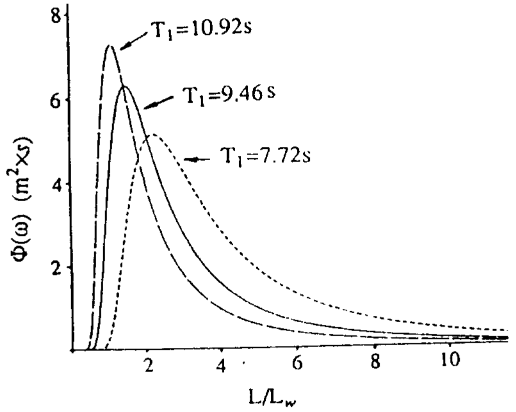


Fig. 2. The ISSC spectra(L=348.358m, $H_{1/3}$=6m)

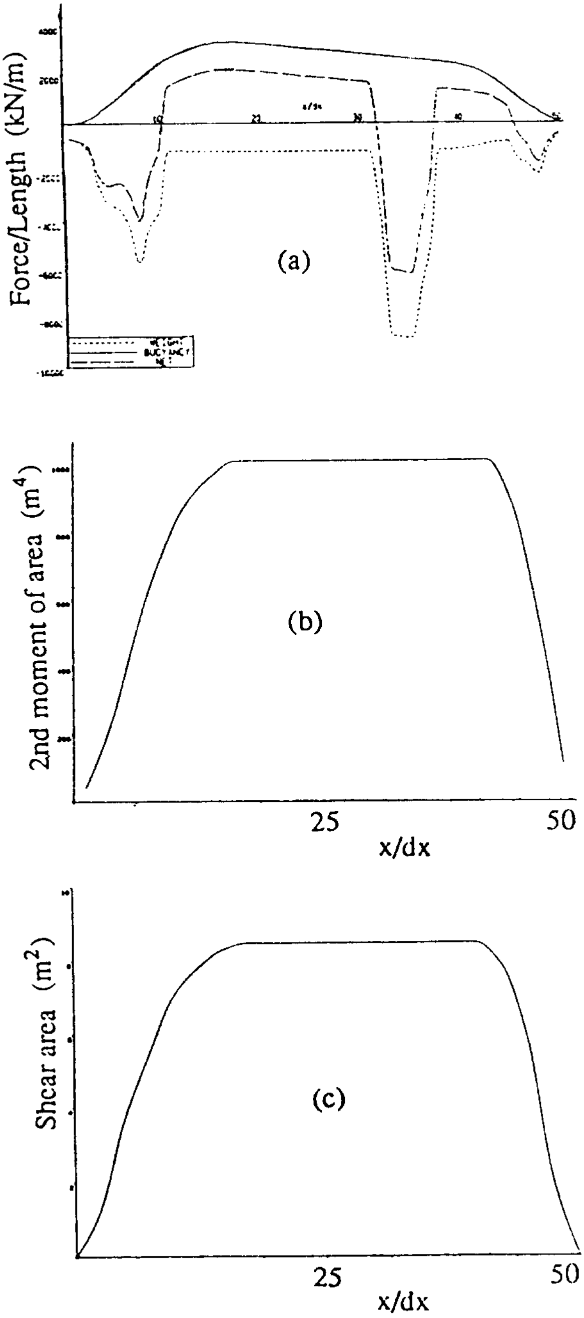


Fig. 3. Data for the tanker in a ballast condition

(a) Weight, buoyancy and net force curves

(b) Second moment of area curve

(c) Shear area curve

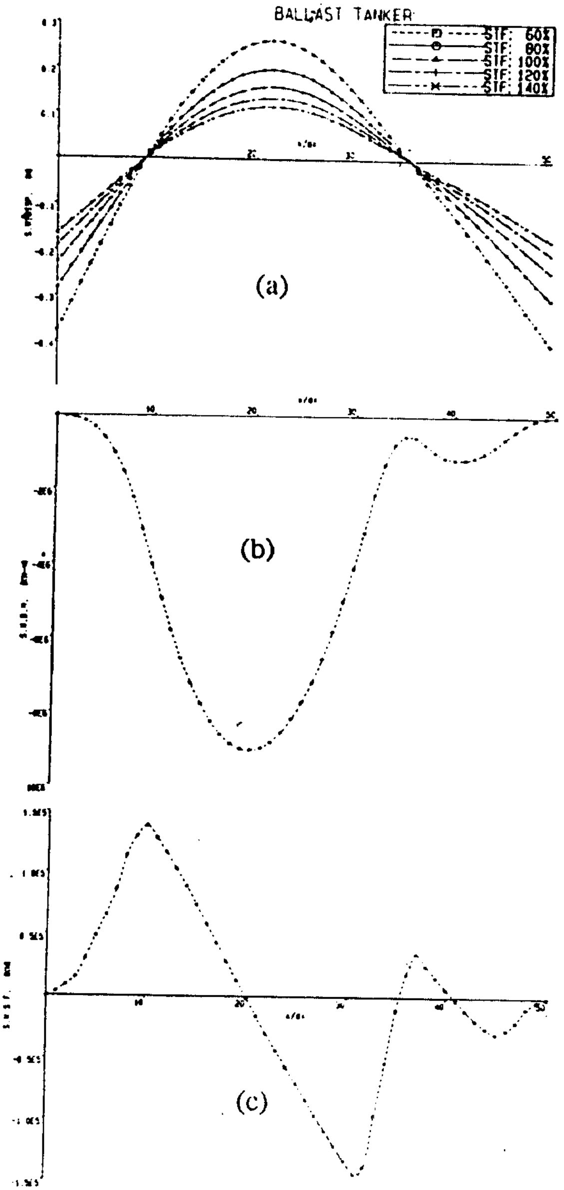


Fig. 4. Still-Water responses for a ballast tanker
(a) SWD (b) SWBM (c) SWSF

similar parameters[3].

Fig. 4 shows still-water displacements or deflections(SWD) and corresponding stillwater bending moments(BWBM) and shearing forces (SWSF). One may note, from the figure, that the hull is in hog when in ballast and the variations of displacement with the hull stiffness are distinct. (N.B. Load integration is applied for the curves of SWBM amd SWSF, whereas modal summation is used for SWD.)

3.2 Seaway

When the response in an irregular seaway is to be investigated, it is known to be practical and quite common to employ wave energy spectra representing relevant real sea states. There are, nowadays, a variety of proposed wave spectra[19] and one of the most frequently used forms is referred to as the ISSC spectrum which is used in this work.

Table 2 General characteristics and input data

Item	Value	Remarks
Principal dimensions		
Length	348.358	(m)
Beam	51.8	(m)
Draft		
(Fully loaded)	19.58	(m)
(Ballast)(*)	5.93	(m)
Displacement		
(Fully loaded)	2.79194(E6)	(kN)
Ballast)	7.63598(E5)	(kN)
Ship speed	6,8,10 (11.66,15.54,19.43)	(m/s) (Knots)
Heading angle	180	(deg.) Head sea
Sea spectra		ISSC
($H_{1/3}$)	6.0,6.0,6.0	(m)
(T_1)	7.72,9.46,10.92	(s)
Mod. of Elasticity	2.07070(E8)	(kN/m^2)
Mod. of Rigidity	8.28269(E7)	(kN/m^2)
No. of modes	5	3 distortion modes

N.B.) Data for fully loaded condition are for reference.
(*) – Trimmed by the stern, total trim = (–)3.63m

The input values of statistical parameters of the spectrum, i.e. characteristic wave period(T_1) and significant wave height($H_{1/3}$), are shown in Table 2 and the family of the spectra are illustrated in Fig. 2. The values assigned to these parameters reflect an attempt to simulate rather moderate operating conditions for this ship and examine their influence. Note that irregular waves and applied only for the steady state analysis.

4. Flexibility and Steady State Responses

4.1 Dry hull analysis

Because the changes in the hull stiffness are applied uniformly over the length of the hull, principal modes of deflection do not change as shown in Fig. 5(a). However, corresponding modal bending moments(Fig. 5(b)) and shearing forces(Fig. 5(c)) of each mode change proportional to the variations of second moment of area. On the other hand, the natural frequency of each mode changes almost proportional to the square root of the bending stiffness variation as shown in Table 3. These results of dry hull analysis may not be surprising. since they can be well compared with those of equivalent Euler beam of uniform section, the results of which are directly and exactly related to the bending stiffness by simple formula respectively. In this context, note that

Table 3 Natural frequency, ω_Y, (in rad/s) as a function of hull stiffness

Mode no.	Variation of Hull Stiffness(%)				
	60	80	100	120	140
2	5.24 (0.78)	6.03 (0.90)	6.73 (1)	7.36 (1.09)	7.94 (1.18)
3	12.45 (0.79)	14.27 (0.90)	15.84 (1)	17.25 (1.09)	18.52 (1.17)
4	23.45 (0.80)	27.15 (0.91)	29.92 (1)	32.41 (1.08)	34.60 (1.16)

N.B.) Numbers in brackets indicate fraction of natural frequencies of original hull. Note that mode number 0 and 1 correspond to rigid body motions heave and pitch respectively, i.e. $\omega_0=0=\omega_1$.

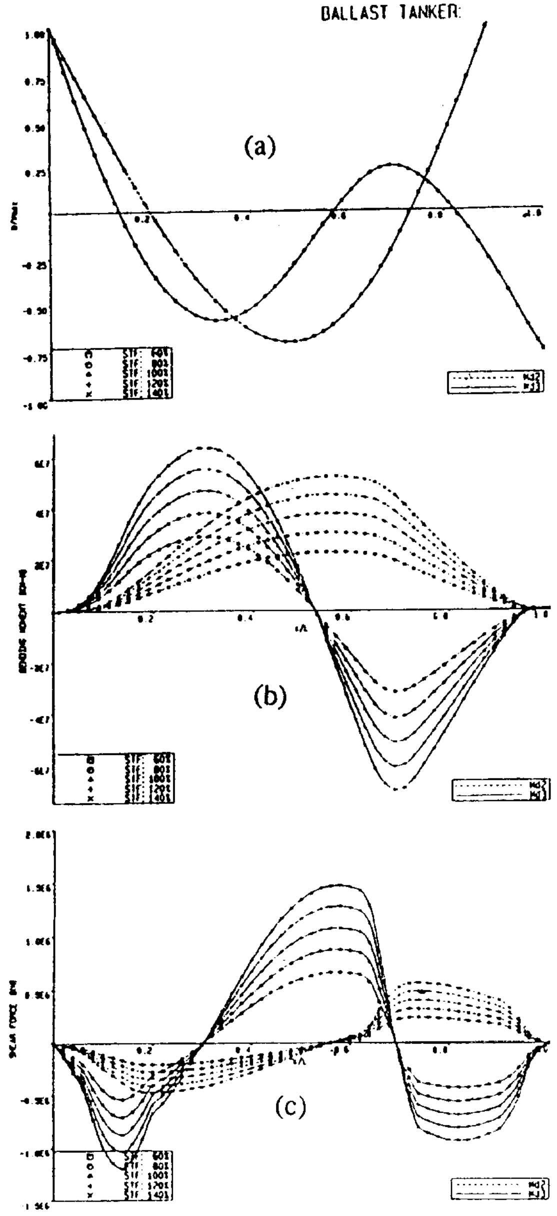


Fig. 5. Principal modes and corresponding bending moments and shearing forces

even though the variations of shear rigidity are allowed for in the present calculation, their effects are well known to be negligible compared to those of bending stiffness. Therefore, the results above are dependent on the bending stiffness.

4.2 Principal coordinates and resonant frequencies

Due to the nature of properties varied, the magnitudes of principal coordinates for the rigid modes change negligibly(, so the figures are not included in the results) ; but, as expected, distortional modes change quite notably. Moreover, in the region of small L/Lw, the magnitude of these principal coordinates are inversely proportional to the bending stiffness fractions. Naturally, in the relatively highter frequency region-including the neighbourhood of resonances-such a proportion is no more applicable(Fig. 6).

Table 4 Resonant wave frequency(in rad/s) and its ration to original

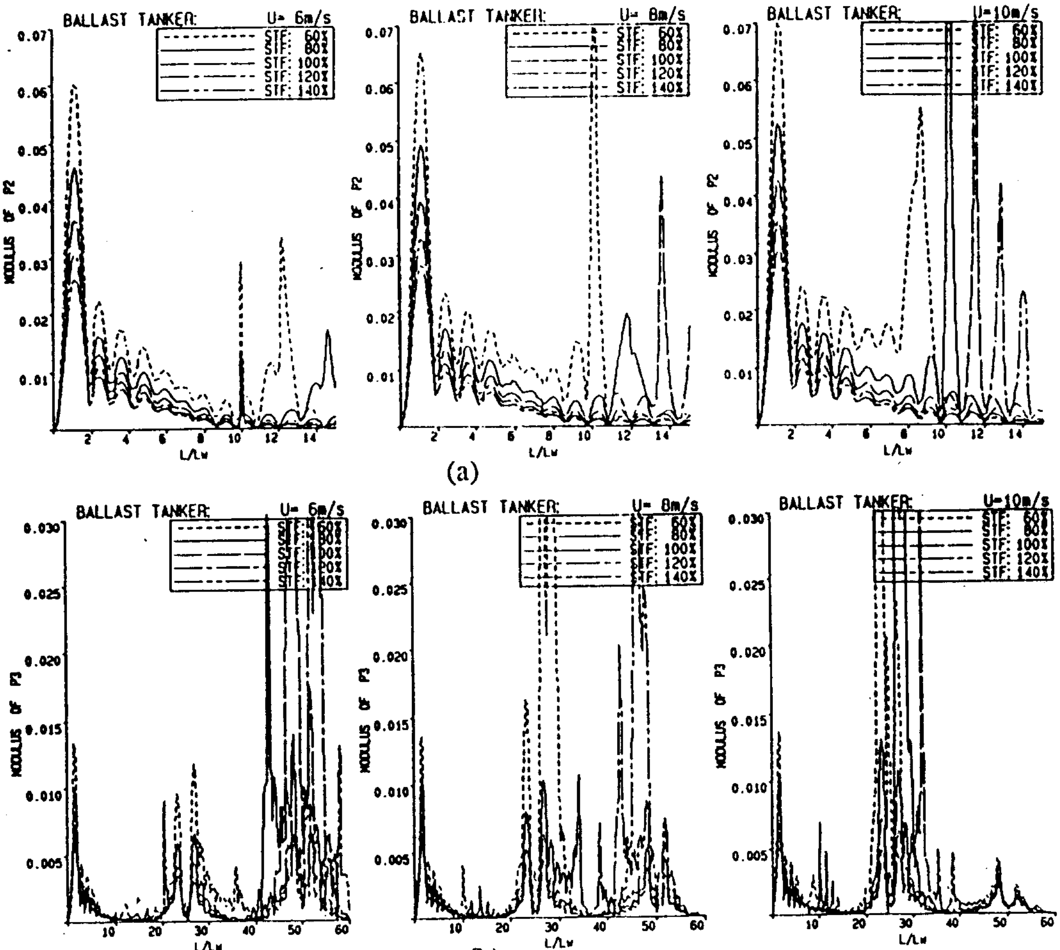
Ship speed	Mode no.	Variation of Hull Stiffness (%)				
		60	80	100	120	140
6 m/s	2	1.47 (0.85)	1.61 (0.93)	1.72 (1)	1.81 1.06)	1.90 (1.11)
	3	2.52 (0.86)	2.75 (0.94)	2.93 (1)	3.08 1.05)	3.21 (1.10)

From the curves showing the results of principal coordinate magnitudes, the characteristics of wave-induced resonance and be observed. Thable 4 shows the well known fact that decreased hull stiffness will decrease the values of L/Lw at which resonance occurs. (The effect of ship speed variation, though not our main concern at the moment, may also be noted from the figures.) In general, a reuced resonant frequency will subject the ship to resonance longer waves of greater energy, and thus, it is thought, resulting in larger bending moments. However, there and some claims that this is not as straightforward, due to the irregular nature of the sea and minor variations in the ship's speed and heading[12]. Nevertheless, although such claims are supposed to be understandable, it is perceived that there is not much room to defy the fundamental verdict of the above trend. Because of the relative importance of this wave-excited main hull vibratory response-widely known as springing-the matter is further discussed in section 4.3. It should be noticed that the resonant frequency is changed almost in proportion to the cube root of the bending stiffness fraction.

4.3 RAOs of bending moment

The differences of magnitude in the low frequencies, i.e. in the vicinity of the, so called, ship/wave matching region, are almost negligible (Fig. 7). This implies that the changes imposed into the bending stiffness(+40%, −40%) are hardly significant in the ship-wave matching region. In fact, from the variations of modal bending moment and principal coordinates in this region which show exactly opposite tendencies with each other, such a result is hardly surprising. In addition, one may note a trend of bending moment being increased slightly as the tull becomes more flexible at the wave range from L/Lw=2.5 upto just before resonance frequencies.

If we look into the responses at the wave-exited vibratory resonance (so called, springing), it is noted that the variation in the magnitude of bending moment in the vicinity of resonance due to changes in the hull stiffness is too irregular to establish a certain tendency. At this junction, a few wards can be said regarding the damping associated with ship hull vibration without going into detail. It is relevant, at least, to question whether there is a way to clarify how damping mechanism (i.e. structural and hydrodynamic damping are considered as typical damping mecha-



(a)

(b)

Fig. 6. Principal coordinates for modes 2 and 3

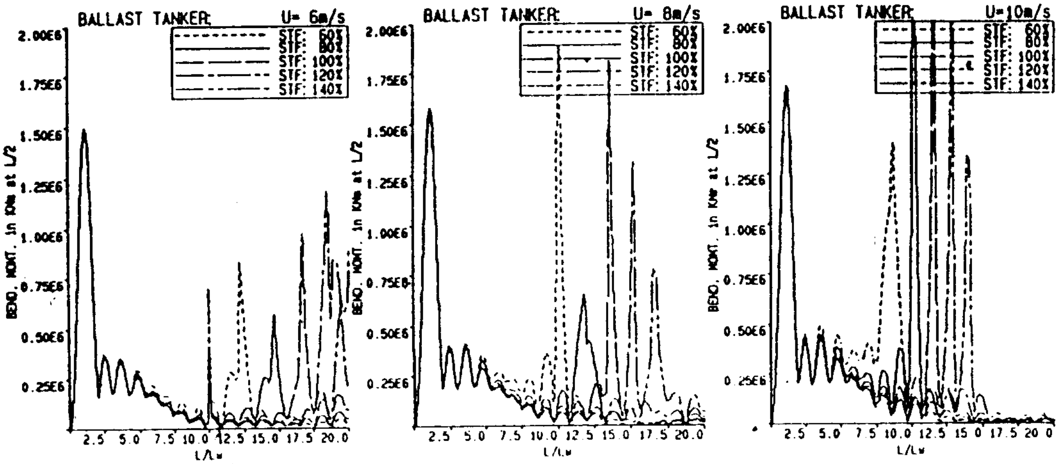


Fig. 7. Bending moments(RAO$_S$) amidships

nisms present in fluidstructure system) affects the responses in a certain wave frequency range as the ftiffness varies.

There is no doubt that the influences of hydrodynamic damping(, which should be dominant in lower wave frequencies) and structural damping are reversed with increasing wave frequency. Furthemore, a stiffer ship will get highter structural damping and smaller hydrodynmaic damping compared to a more flexible ship. As the wave frequency varies, in fact, this generally opposite trends of two main damping mechanisms would produce the resultant responese in variety between the cases of different hull stiffness. In this context, it may be noted that although the magnitude of hydrodynamic damping appears to be very small, its contribution to the total vibratory damping coefficient cannot be neglected, since the other contributions such as the structural damping are also small[11,16)].

For example, the result when U=8m/s Fig. 7, which shows, in general, that the combined damping is increased as the ship becomes stiffer, may tell us that in that condition the effect of structural damping is greater than that of hydrodynamic one. In many other cases, however, it is hardly straightforward[23,29)]. As far as springing is concerned, it would be correct that its response to the synchronous component of the sea rarely becomes well established in reality as well as in the theoretical analysis. However, if we assume that the general mechanism of occurrence of springing would be similar in various seaway conditions, it should be more relevant to quote that springing phenomenon(, not necessarily the magnitude of corresponding bending moment,) is increased by increasing hull flexibility[24,30)].

It may be concluded that the variation in stiffness would not produce any tangible difference in the magnitude of the steady state bending moment RAO in the relatively long-waves provided the resonant frequency is not very low.

4. Response spectra and RMS values of bending moment

Because of the non-tangible effects of stiffness variation on the bending moment at low wave frequencies and its significant effects at relatively higher wave frequencies, as well as resonant frequencies, selection of sea spectrum will have considerable effect on (short-term) statistical properties, such as RMS values. Accordingly, it is ea-

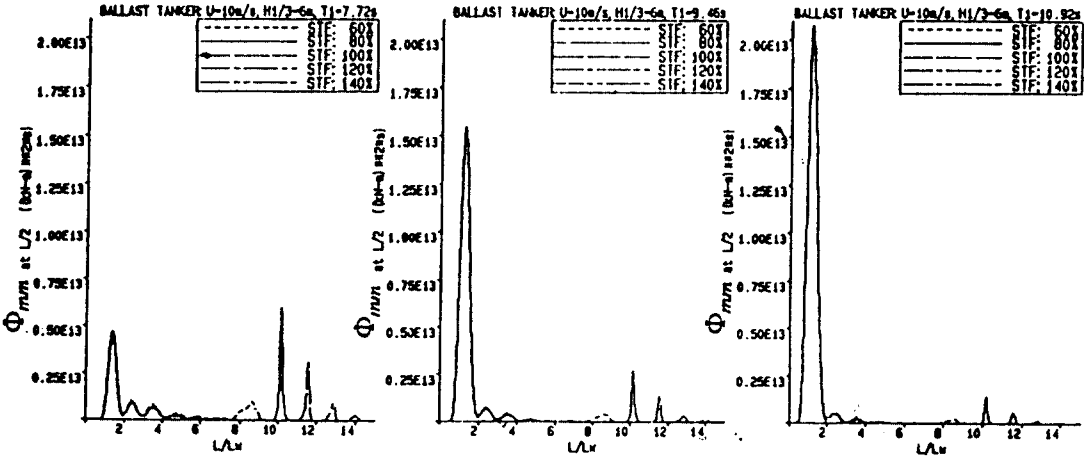


Fig. 8. Response spectra of bending moment amidships, U=10m/s

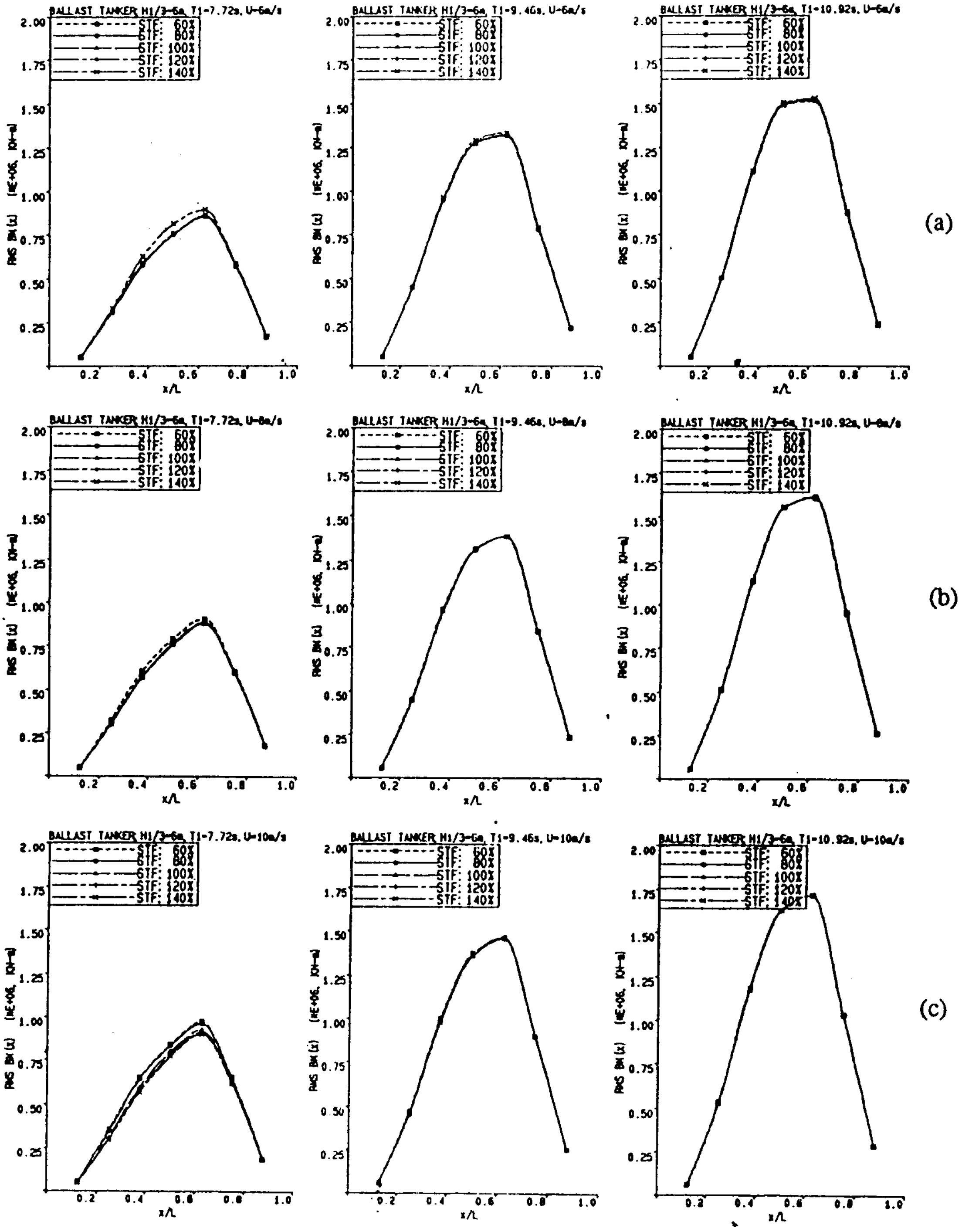


Fig. 9. RMS BM-x/L(stiffness) in ballast condition
(a) U=6m/s (b) U=8m/s (c) U=10m/s

sily seen that the characteristic period, T_1, should be small enough(i.e. the wave spectrum has greater energy in rather shorter waves), if the resonant frequency is not low enough, in order to observe any visible effects due to stiffness variation. Fig. 8 is the response spectra of bending moment amidships when U=10m/s for three wave energy spectra. (N.B. Since the curves of wave spectra and RAO_S are known, it can be rather easily surmised what the features of resultant response spectra would be. Therefore, only the results at L/2 of response spectra and demonstrated as examples.) It can be seen that the results for $T_1=7.72s$ show the most visible variations with hull stiffness. However, because of the relatively high natural (and resonant) frequencies of the hull in ballast condition, the contribution from distortion resonances would not be considerable for other greater T_1 and/or smaller speeds.

When comparing the RMS values along the hull in Fig. 9, the extreme differences in the results, as expected, arise when $T_1=7.72x$, i.e. the smallest wave period used. Note, in passing, that the largest magnitudes occur when $T_1=10.92s$, the highest period used. When U=10m/s, the result shows that(RMS) bending moment increases as the hull becomes more flexible. However, at the other lower speeds, the variations of hull stiffness do not produce any visible differences between the cases, except those of 140% and 60% of the as-built stiffness when U=6m/s and 8m/s respectively. The above trends, as can be readily seen in figures, disappear as T_1 increases.

The reason for the above is the apparent effect of 2-noded hull vibratory responses, the magnitudes of which are variable with variations of stiffness and ship's speed. Then, it is anticipated that the effect of springing could be significant on the total bending moment(31), even if its(exact) magnitude is not well devised to be clarified*). Otherwise, the steady-state bending moment seems likely to decrease very little as stiffness increases.

5. Flexibility and Transient Responses due to Slamming

5.1 Description of slamming and minimum wave amplitude

Flexibility is, of course, much relevant to the subject of slamming—a type of transient dynamic response—because both of them are related with higher modes of vibration. Thus, many relevant references can be found in the literature(e.g. refs. (1,3,4,18,32)). In the present study, the analysis applied by Bishop et al.[20,22] will be used. That is, at first, a length of emergence, which is referred to as the slamming length and represents the intensity of slam will be postulated for a ship travelling with a given speed in regular head waves of known frequency. Subsequently, the wave amplitude can be determined. Finally, the transient forces can be determined using the concepts of impact and/or momentum.

For the analysis of impact slamming, the method pioneered by Stavovy and Chuang, instead of the Ochi and Motter method, is employed, because the former is known to be more rigorous than the latter in its assumptions and formulation[21]. For the momentum slamming, which accommodates the process of continuing penetration of the hull into the wave after the initial impact and the resultant change in the local fluid momentum, the formulation of Leibowitz is used to evaluate the corresponding slamming forces. It should be noted that the duration and variation of transient

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* * In this context, the effect of ship's speed, in particular, as van Gunsteren [16] argues, seems to be difficult to be established, since the prediction of the damping coefficient which is sensitive and critical for resonance response is still based on empirical data.

forces for a momentum slam are longer and slower compared to an impact slam. This may result in different effects of hull flexibility on slamming[33]. Since physically an impact is accompanied by a change in momentum, the total transient responses as well as total responses, which include the steady state responses are illustrated.

A slamming length of 0.075L (=26.1m) and 6 different wavelengths of $L/l_w=0.543$, 0.657, 0.774, 0.905, 1.045, 1.196 are considered hereafter. The choice of wavelengths in the range $0.5<L/L_w<1.3$ is, firstly, in order to be realistic regarding occurrence of slamming and secondly because the steady state bending moments, which should not be ignores even in the investigation of slamming response, are considerable in this range. As far as the slamming length is concemed, the postulated value appears to be moderate and, as the results of the minimum wave amplitude(Fig. 10) show, is a rather appropriate input value for the investigation of such a ship.

Fig. 10 shows the minimum wave amplitudes required to attain the prescribed slamming length at the various wavelengths and speeds for each flexibility configuration of the hull in both loading conditions. These results confirm trends that as ship's seed increases the minimum wave amplitude required to attain prescribed length of emergence decreases. Naturally, such tendency would disappear and then become the other way round as the waves become shorter, say, $L/L_w>1.35$. Furthermore, the variation of the hull stiffness dose not produce any visible difference in the minimum wave amplitude for slamming as it depends on the relative rigid body motion.

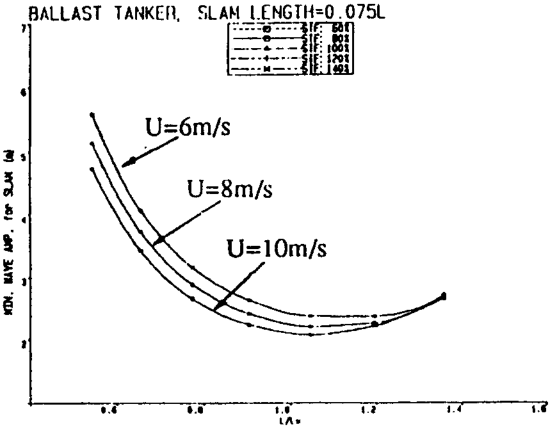


Fig. 10. Minimum wave amplitudes for slamming

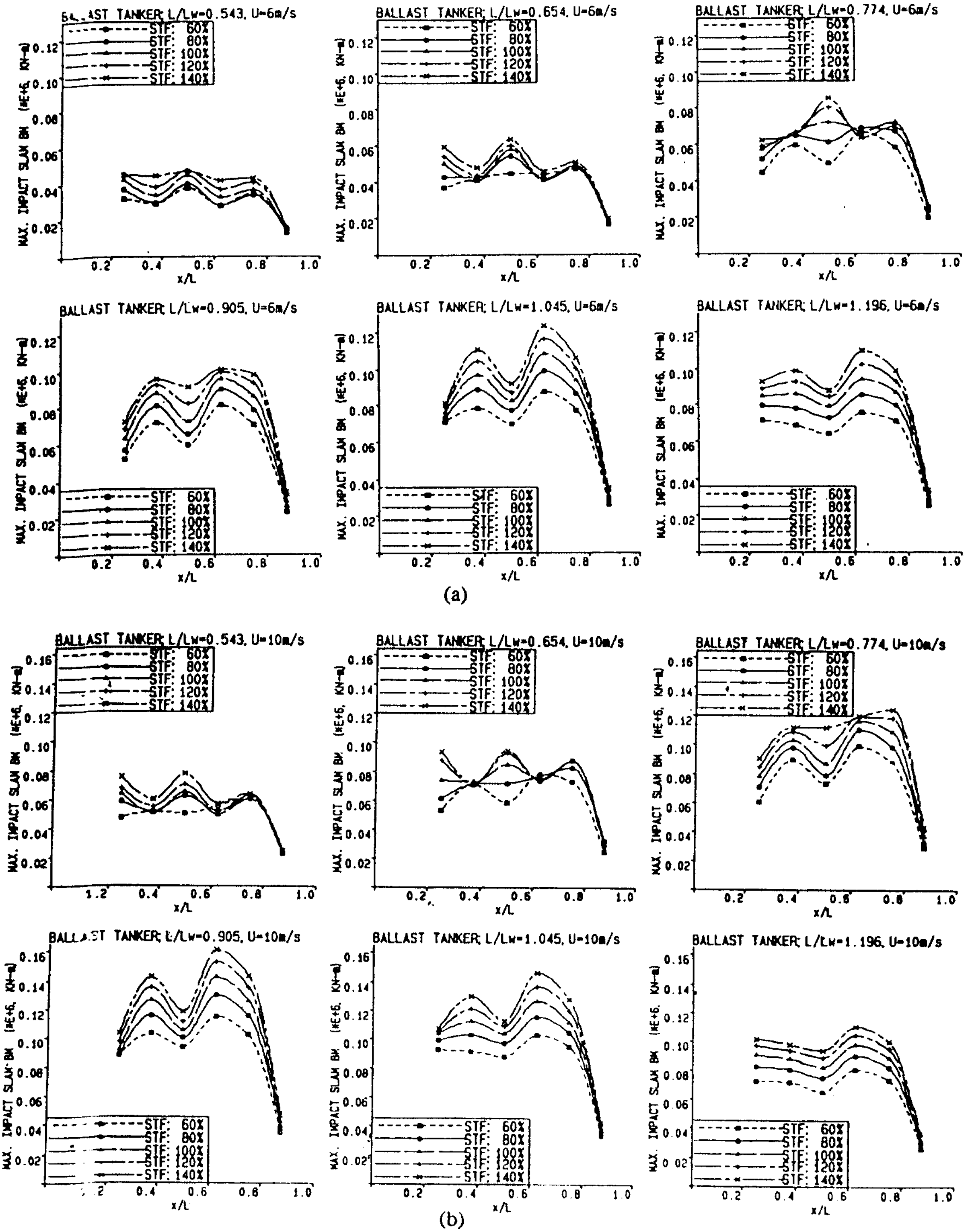
5.2 Impact slam

Fig. 11(a, b) show quite clearly that the more flexible the hull, the less the impact slam-induced bending moment is for various speeds and wavelengths along the hull. This is attributed to the fact that a more flexible body offords larger relief from the impact load. It is interesting to note the double peak locations along the hull-at around 0.38L and 0.62L—in most cases. Furthermore, for relatively stiffer ships in relatively longer waves, triple peaks, at both quarters and at midship, can be observed.

5.3 Momentum slam

One of the general feature of these results is that the momentum slam-induced bending moment has a maximum at around 0.62L for all variables considered as shown in Fig. 12(a, b). The trends of the results with the variation of hull stiffness are as consistent as that of impact slam, but in an opposite way. From these limited observations, therefore, one may speculate that increasing the stiffness is mostly beneficial as far as momentum slam-induced bending moment is concemed. Bearing in mind the much longer duration of momntum slam, the argument made by Disenbacher[33] that slams of relatively longer duration may result in contradictory conclusions seems to be justifiable.

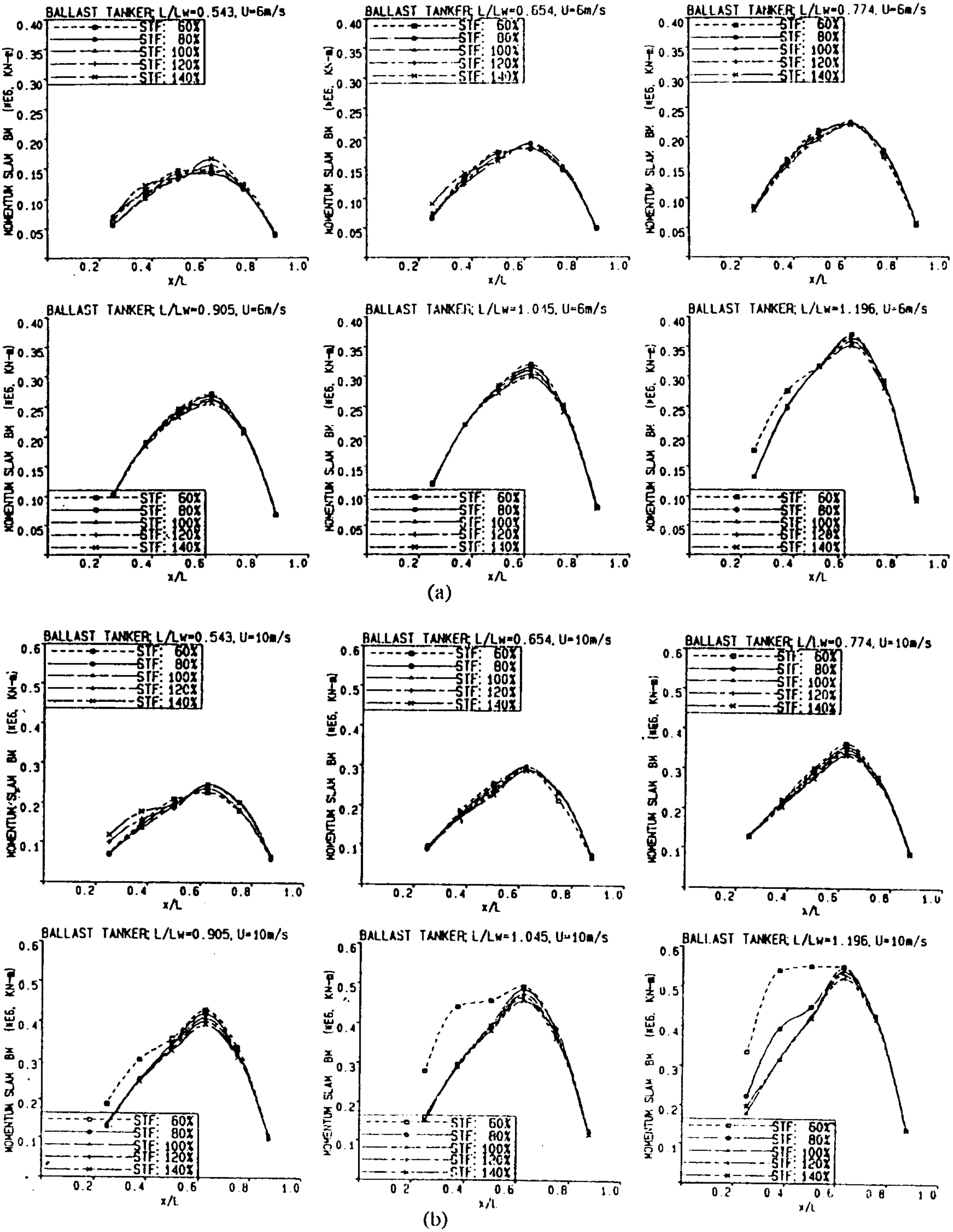
Exceptions appear to be at longer wavelengths, where the fore-mentioned benefitcan be attained



(a)

(b)

Fig. 11. Maximum impact slamming BM-x/L(stiffness)
(a) U=6m/s (b) U=10m/s



BALLAST TANKER; L/Lw=0.543, U=6m/s

BALLAST TANKER; L/Lw=0.654, U=6m/s

BALLAST TANKER; L/Lw=0.774, U=6m/s

BALLAST TANKER; L/Lw=0.905, U=6m/s

BALLAST TANKER; L/Lw=1.045, U=6m/s

BALLAST TANKER; L/Lw=1.196, U=6m/s

(a)

BALLAST TANKER; L/Lw=0.543, U=10m/s

BALLAST TANKER; L/Lw=0.654, U=10m/s

BALLAST TANKER; L/Lw=0.774, U=10m/s

BALLAST TANKER; L/Lw=0.905, U=10m/s

BALLAST TANKER; L/Lw=1.045, U=10m/s

BALLAST TANKER; L/Lw=1.196, U=10m/s

(b)

Fig. 12. Maximum momentum slamming BM-x/L (stiffness)
(a) U=6m/s (b) U=10m/s

only at and around amidships ; at the other sections, the more flexible, the more beneficial. This result(of exceptions) needs further speculation onto the effect of speed variation, because the range of the wavelengths where such a trend appears varies with it. When U=6m/s, the trend is seen at the wave-range of $L/L_W<0.654$, whereas when U=10m/s, it is $L/L_W<0.543$. That is, the trend appears of further longer waves as speed increases.

Another point worthwhile noting is that at relatively short waves the peak magnitude extends over a larger region of the hull namely(from 0.62 L) aftwards—up to 0.4L—and becomes more significant for more flexible hulls and higher speeds. Note that a speed increase results in the appearance of such a tendency in longer waves as well. Comparing the magnitude with those of impact slam, the maximum of momentum slam-induced BM is 2—3 times highter than that of impact slam. Nevertheless, even if the absolute magnitude of the momentum slam is greater than that of the impact slam, the variations due to different hull stiffness values are comparable to those of impact slam. In other words, the variations of impact slam—induced BM could also be effective on total slam response.

5.4 Total slam response

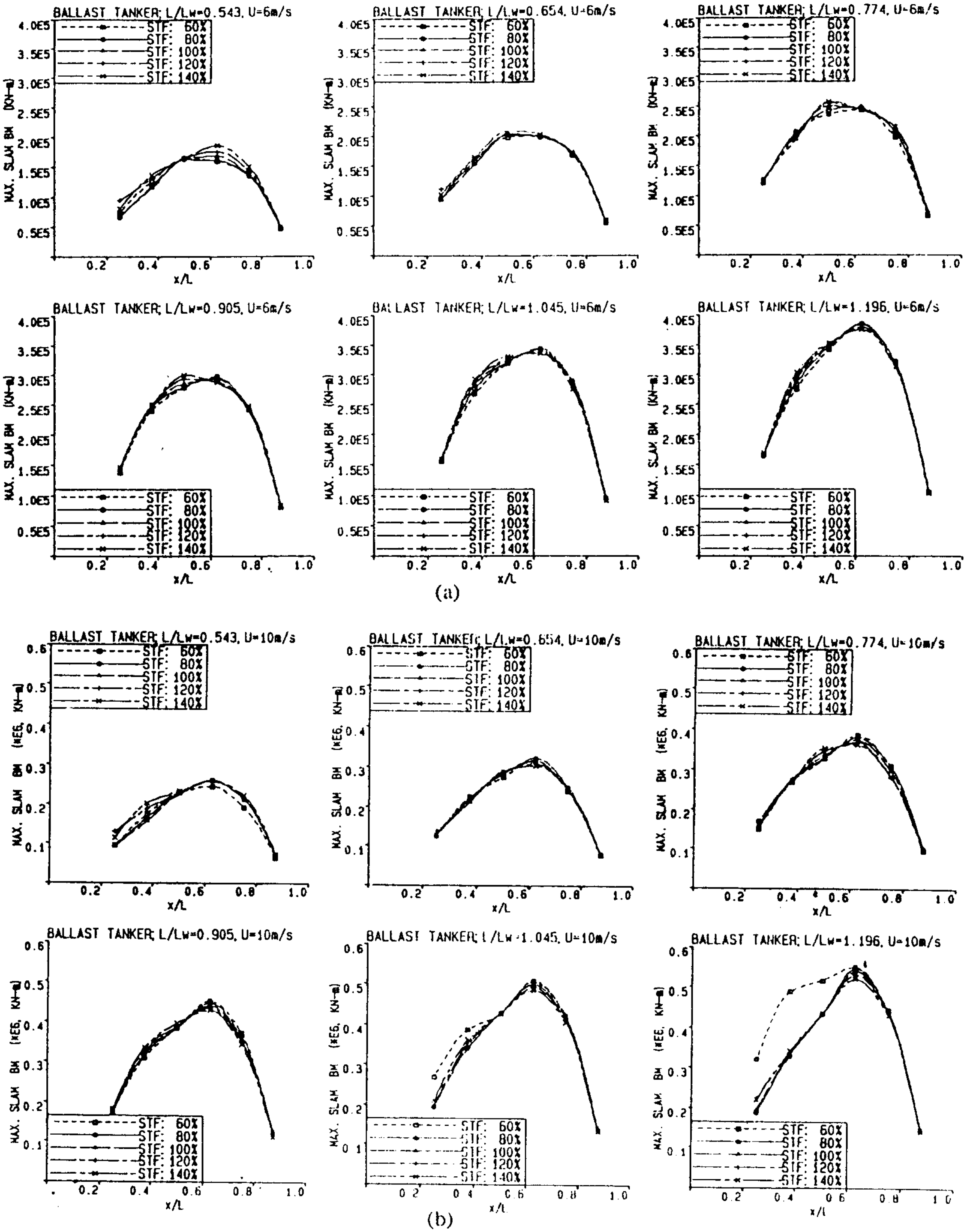
Having seen the results of impact and momentum slam-induced BM, it is clear that the momentum component provides the major contribution to the (magnitude of) total response of slam. However, as already has been noted, the effect of impact slam cannot be ignored, and this is illustrated in Fig. 13(a,b).

Although, in general, the locations of the maximum slam-induced BM along the hull are not much different compared to those of momentum slam, some exceptions, where the variation of impact slam has a decisive role into the characteristic feature of total slam response, are noted. That is, at lower speed and at wavelengths $L/L_W<1.0$, (except $L/L_W=0.543$, the longest wave) the maximum shifts towards amidships as the ship becomes stiffer. This is, of course, due to the effect of the impact slam. Furthermore, in general, at the real part of the hull, including midship area, the effect of impact slam is dominant, whereas at the fore part momentum slam is prevailig. For the exceptional cases, e.g. when $L/L_W=0.543$, it should be remembered that if the wavelength is long enough, the tendency of the variation of momentum slam along the hull, except at around midship section, is opposite to general trend i.e. the stiffer, the better. therefore, the effects of stiffness variation are the same for both impact and momentum slam components.

From the point of view of total slam-induced BM magnitudes, one may tentatively conclude that at both speeds considered a more flexible hull is disadvantageous for the fore part of the ship, say from 0.55L onwards, and advantageous for the aft part, upto 0.55L, of the thull. When $L/L_W=0.543$, however, a more flexible hull is beneficial anywhere on the hull.

6. Total Bending Moment Response

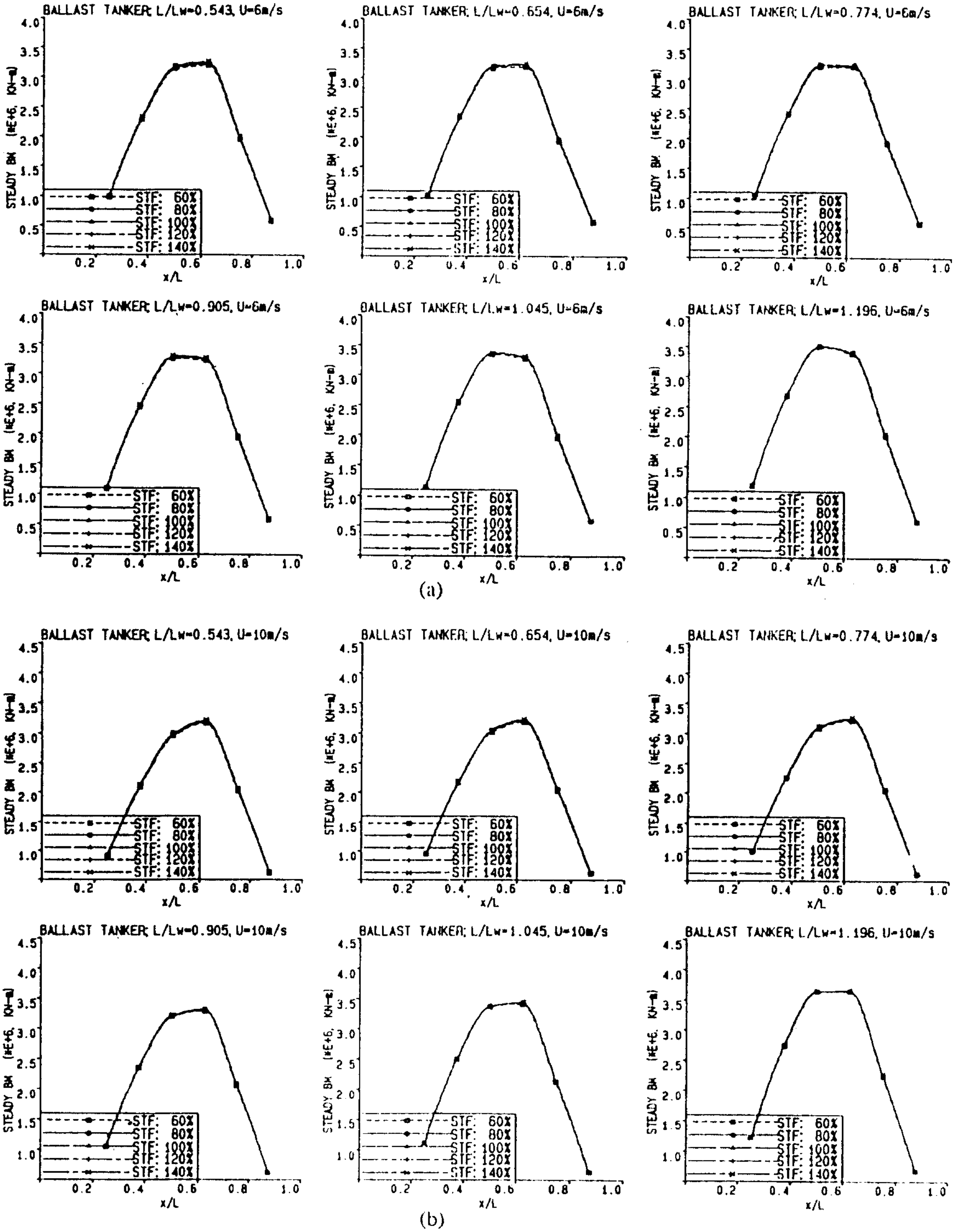
The meaning of total bencing moment here is total slam-induced maximum bending moment plus steady state bending moment only. That is to say, other effects such as still-water bending moment are not considered. To begin with, consider the steady state bending moment along the length of the hull shown in Fig. 14(a,b) for each wavelength used in the slamming response analysis. It is again noted that variation in stiffness does not produce any significant difference as far as the steady state bending moment, in the wave



(a)

(b)

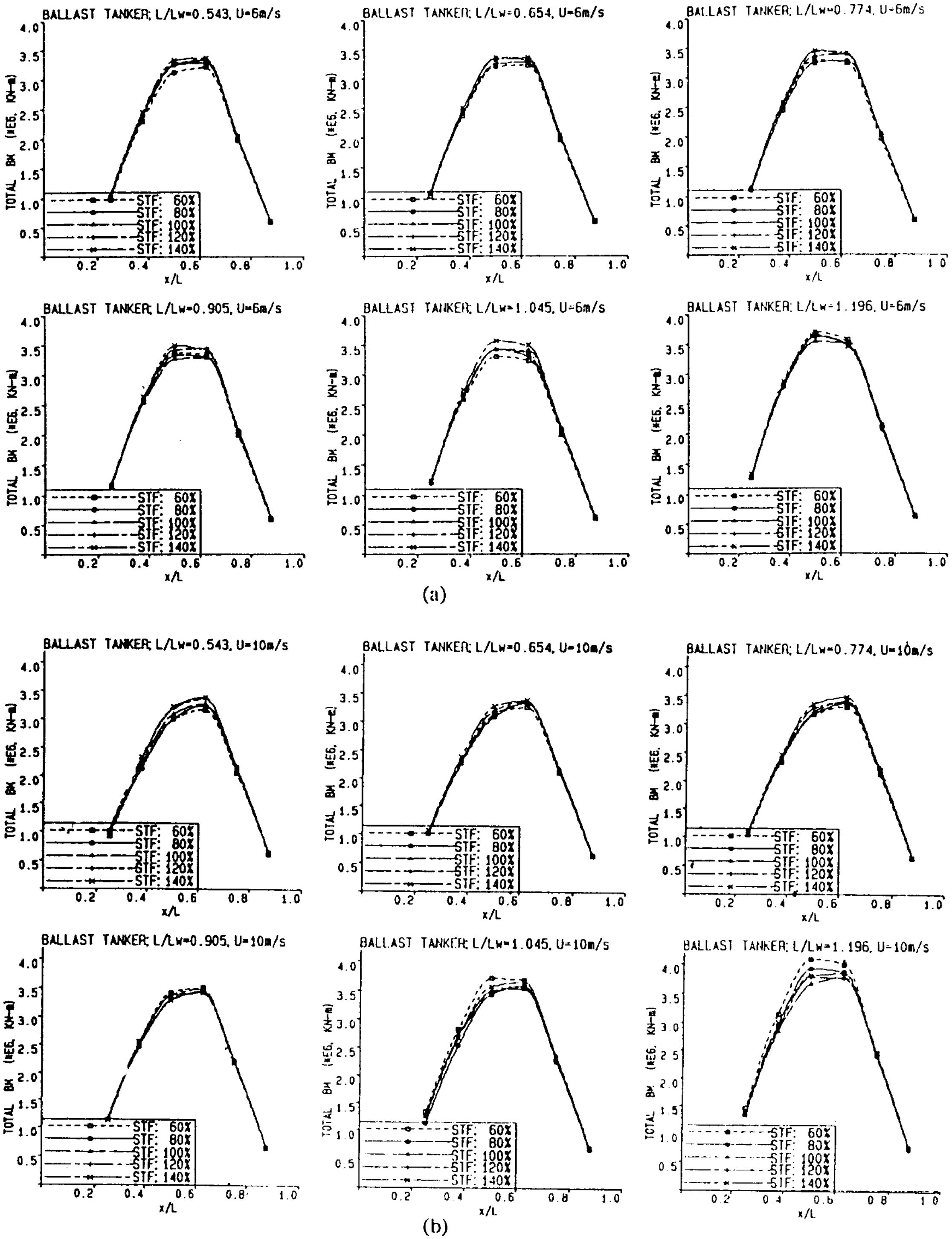
Fig. 13. Maximum slamming BM-x/L(stiffness)
(a) U=6m/s (b) U=10m/s



(a)

(b)

Fig. 14. Steady-state BM-x/L(stiffness)
(a) U=6m/s (b) U=10m/s



BALLAST TANKER; L/Lw=0.543, U=6m/s

BALLAST TANKER; L/Lw=0.654, U=6m/s

BALLAST TANKER; L/Lw=0.774, U=6m/s

BALLAST TANKER; L/Lw=0.905, U=6m/s

BALLAST TANKER; L/Lw=1.045, U=6m/s

BALLAST TANKER; L/Lw=1.196, U=6m/s

(a)

BALLAST TANKER; L/Lw=0.543, U=10m/s

BALLAST TANKER; L/Lw=0.654, U=10m/s

BALLAST TANKER; L/Lw=0.774, U=10m/s

BALLAST TANKER; L/Lw=0.905, U=10m/s

BALLAST TANKER; L/Lw=1.045, U=10m/s

BALLAST TANKER; L/Lw=1.196, U=10m/s

(b)

Fig. 15. Total BM-x/L(stiffness)
(a) U=6m/s (b) U=10m/s

range consiered.

Now, the results of total bending moment are shown in Fig. 15(a,b). Comparison of Figs. 14 and 15 clearly indicates the effect of additional (and flexibility depandent) bending moment due to slam on the steady state bending moment. The increase in bending moment due to slam is at most, 5−7%(at a ship's speed of 6m/s) and 10−15%(at 10m/s) of the steady state bending moment. These fractions are commensurate with the severity of the postulated slam. The results also reveal that the additional bending moment due to slam is more effective mainly at a portion of the hull between 0.35L−0.65L.

In order to see how the variations of slam-induced bending moment, cause by different fractions of hull stiffness, affect the existing steady state bending moment, it seems appropriate to define a transition wavelength. In the range of relatively longer waves(depending on the speed, that is to say $L/L_W<1.1$ at 6m/s and $L/L_W<0.8$ at 10m/s) the total bending moment increases with increasing stiffness. On the other hand, in the range of shorter waves, the trends become opposite. This result implies that highter speed not only allows for more slamming effect, but also makes the transition wavelength longer.

7. Conclusive Remarks

The investigation on the effect of flexibility variations on ship responses has been made for a large tanker in ballast. The most significant conclusion to be drawn from this study is that the flexibility influences certain major aspects of ship responses to slams and to vibratory resonance in the relatively high frequency range. In particular, it appears that, as far as transient repsonses due to slam are concerned, the results based on the concepts of impact and momentum slamming are quite different, and shoul be analysed together. A number of conclusions can be presented here.

(1) Increasing the flexibility of the structure reduces the natrual frequency of the hull, and thus results in the vibratory resonance at lower wave frequency. Althouh its magnitude of the response is sensitive to various factors, such as excitation, damping and ship's speed,. the possible consequence of the trend would increase the springing phenomenon as flexibility is increased.

(2) The variations of flexibility produce little changes of ship responses at low frequencies, including ship/wave matching region.

(3) The result of impact slam-induced bending moment confirms that increased bull flexibility is beneficial. However, the result of momentum slam-induced bending moment shows, although depending on wavelength, speed and hull positions, in general, the opposite trend. The latter can justifiable be drawn, bearing in mind the longer duration of the momentum slam.

(4) Total response result, comprised with total slam and steady state responses of bending moment, reveals that at relatively longer waves the total bending moment decreases with increasing flexibility. On the other hand, at shorter waves, the trand becomes opposity ; increased stiffness is beneficial.

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References

1. Kline R. G. and Daidola J. C. : Ship Vibration

Prediction Methods and Evaluation of Influence of Hull Stiffness Variation on Vibratory Response, SSC Report, SSC-249, 1974.

2. Chang P.Y. : The Effect of Varying Ship Hull Proportions and Hull Materials on Hull Flexibility, Bending and Vibratory Stresses, SSC Report, SSC-288, 1979.
3. Sellers M. L. and Kline R. G. : Some Aspects of Ship Stiffness, Trans. SNAME, 1967.
4. Evans J. H. and Kline R. G. : Effect of Hull Girder Stiffness Variations on Ship Structural Performance, Trans. SNAME, Vol.86, 1978.
5. Palermo P.M. : Discussion of Ref.4
6. Maximadji A.I. : On the Problem of Norms for Longitudinal Rigidity of Ship's Hulls made from High-Tensile Steel, Discussion to the report of ISSC Committee 3d on "Discontinuities and Fracture Mechanics", Proceedings of 3rd ISSC, Oslo, 1967.
7. D'Arcangelo A. M. : *A Guide−to sound Ship Structures,* Cornell Maritime Press, 1964.
8. Nibbring J.J. W. and Scholte H.G. : The Fatigue Problem in Shipbuilding in the Light of New Investigations, RINA, April 1975.
9. Hull cracking study "alarming", IMO News, No. 2, 1991.
10. Tankers Recalled as Cracks Found, The Sea, No. 90, Sept/Oct 1990
11. Goodman R.A. : Wave-Excited Main Hull Vibration in Large Tankers and Bulk Carriers, Trans. RINA, Vol. 113, 1971.
12. Kline R.G. : Springing and Hydroelastic Problems of Large Ships, SNAME STAR Symposia Papers, 1975.
13. Troesch A. W. : Wave-Induced Hull Vibrations−An Experimental and Theoretical Study, JSR, Vol. 28−2, Jun. 1984.
14. Hoffman D. and van Hooff, R. : Feasibility Study of Springing Model Tests of a Great Lakes Bulk Carrier, ISP, Vol. 20, 1973.
15. van Gunsteren F.F. : Springing−Wave Induced Ship Vibrations, ISP, Vol. 17, Part 195, 1970.
16. van Gunsteren F.F. : Some Further Calculations of Wave-Induced Ship Hull Vibrations, IMechE 1974, Ed. by R.E.D. Biship and W.G. Price, April 1974.
17. Hughes O.F. : *Ship Structural Design,* John Wiley & Sons, 1983.
18. Kline R. G. and Clough R. W. : The Dynamics Response of Ships' Hull as Influenced by Proportions, Arrangement, Loading, and Structural Stiffness, Proc. SNAME Spring Meeting, Montreal, 1967.
19. Bishop R. E. D. and Price W. G. : *Hydroelasticity of Ships,* Cambridge University Press, 19 79.
20. Bishop R. E. D., Price W. G. and Tam P.K.Y. : A Unified Dynamic Analysis of Ship Response to Waves, Trans. RINA, Vol. 119, 1977.
21. Belik Ö. : Symmetric Response of Ships in Regular and Irregular Waves, Ph.D thesis, Dept. of Mech. Eng., University College London, 1982.
22. Belik Ö. : Bishop R. E. D. and Price W.G. : On the Slamming Response of Ships to Regular Head Waves, RINA, W13, pp. 1−12, 1979.
23. Belgova M.A. : Discussion of Ref. 11.
24. Lewis E. V., Hoffman D., et al. : Load Criteria for Ship Structural Design, SSC Report, SSC-240, 1973.
25. KIRKI : Loss of Bow Section(from the Navigation Regulations Investigation, Report No. 33, Australian Dept. of Transport and Communications), Seaway The Journal of the Nautical Institute, July 1992.
26. Carlisle D. : Condition Evaluation and Maintenance of Tanker Structures, RINA London Branch Evening Meetings, No. 1991.
27. Bumett R. F. : Loading Calculators, Shipbuilding & Marine Engineering Intemational, March 1981.

28. Thomson W. T. : *Theory of Vibration with Applications,* Prentice-Hall, Inc., 1972.
29. Kaplan P. and Sargent T. P. : Further Studies of Computer Simulation of Slamming and Other Wave-Induced Vibratory Structural Loadings on Ships in Waves, SSC-231, 1972.
30. Lewis E. V. : Structural Dynamics of Ships, IMechE 1974, Ed. by R.E.D. Bishop and W.G. Price, April 1974.
31. Little R. S., Lewis E. V. and Bailey F. : A Statistical Study of Wave-Induced Bending Moments on Large Oceangoing Tankers and Bulk Carriers, Trans. SNAME, pp. 117−168, 1971.
32. St. Denis M. and Fersht S. N. : The Effect of Ship Stiffness upon the Structural Response of a Cargo Ship to an Impulsive Load, SSC Report, SSC-186, 1968.
33. Dinsenbacher A. L. : Discussion of Ref. 3.