

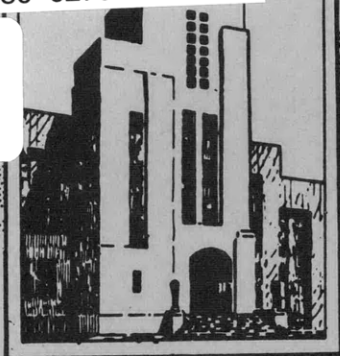
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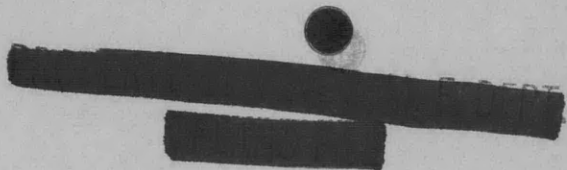


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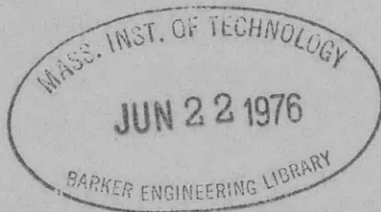
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APPLIED  
MATHEMATICS

HYDROELASTIC STUDY OF A SHIP EQUIPPED  
WITH AN ANTIPITCHING FIN

by

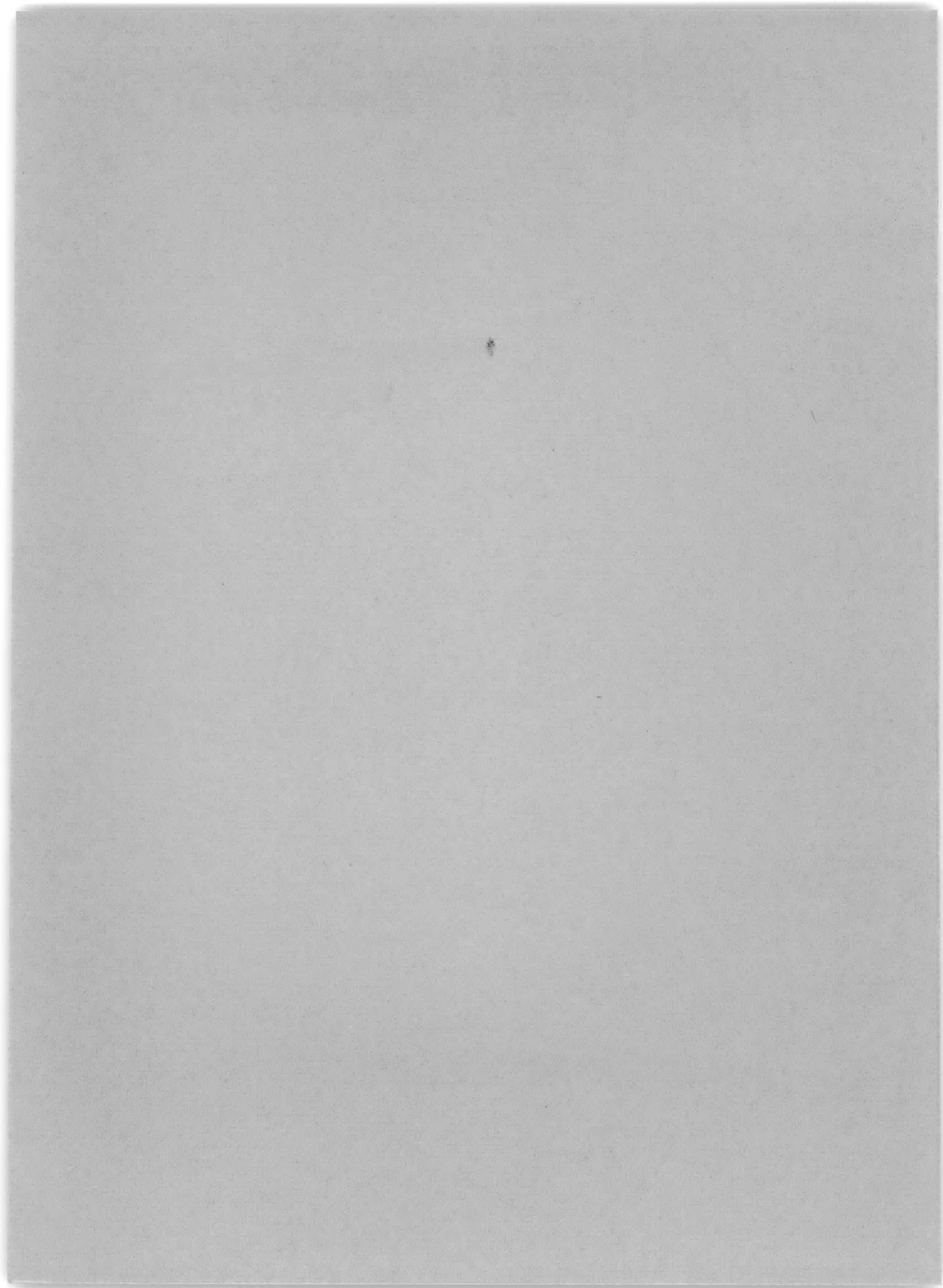
Kazuo M. Ochi



HYDROMECHANICS LABORATORY  
RESEARCH AND DEVELOPMENT REPORT

October 1962

Report 1455



**HYDROELASTIC STUDY OF A SHIP EQUIPPED  
WITH AN ANTI-PITCHING FIN**

**by**

**Kazuo M. Ochi**

**Paper presented at Annual Meeting of  
Society of Naval Architects and Marine Engineers,  
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**October 1962**

**Report 1455**



# Hydroelastic Study of a Ship Equipped With an Antipitching Fin

By Kazuo M. Ochi,<sup>1</sup> Member

This paper presents the results of an experimental study to determine the seaworthiness characteristics and induced vibrations of a ship equipped with an antipitching fin. The mechanism of occurrence and the fundamental properties of the induced vibrations are examined from the hydroelasticity viewpoint and are discussed in Part 1 of the paper. Effect of location, size, and configuration of an antipitching fin on ship motions and the induced vibrations are discussed in Part 2. The study consisted of tests on 5.5-ft and 22-ft Mariner models equipped with fins. The majority of the tests were carried out in regular waves; however, tests in irregular waves were also conducted for a specific ship speed. A fin having holes of proper size and shape (called Fin X) was found to be beneficial both for pitch and vibration reduction.

APPLICATION of a bow antipitching fin has proved practical for reducing the pitching motion and thereby maintaining ship speed at sea. Results of full-scale trials carried out on the USS *Compass Island* with fins showed reduction in pitch of approximately 10 to 15 per cent in Sea State 5, in comparison with pitch data taken on the *Silver Mariner*, a similar ship without fins [1].<sup>2</sup> Experiments made on a Mariner model in regular waves showed a reduction in pitch of 30 per cent in waves whose length was  $0.91L$  at a ship speed of 6 knots [2]. Other model tests on a carrier showed 35 per cent reduction at the synchronous speed for pitch in regular waves of length equal to ship length [3].

These results from full-scale trials and model tests have indicated the feasibility of reducing pitching motion by means of fins installed at the ship bow. However, one serious problem associated with the use of antipitching fins is that of transverse hull vibration. The intensity of these induced vibrations is so severe as to negate the value of the fin in reducing ship motion.

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<sup>2</sup> Numbers in brackets designate References at end of paper.

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As background for the present study, the history of the vibration problem which was experienced by the *Compass Island* [4] will be reviewed. The *Compass Island* had been equipped with both antipitching and antirolling fins. The area of the antipitching fin was about 1.6 per cent of the water plane area, and the fin had small tip fences. During several ocean crossings, serious transverse hull vibrations were experienced and were attributed to the antipitching fin. In order to investigate the nature of these vibrations, full-scale trials were conducted. As a result, corrective measures of adding holes through the fin and enlarging the tip fences were tried, and the alterations resulted in some reduction of the magnitude and frequency of occurrence of hull vibration. However, the buffeting and vibration experienced by the ship were still excessive at 10 and 14 knots and even at anchor in the open sea, in rough weather. After the fin was removed at the recommendation of the captain, there were no vibrations of this type at any speed up to 16 knots, even in heavy seas. Furthermore, large quantities of sea spray formerly thrown up and over the ship by the fin were absent. These experiences leave no doubt that the transverse hull vibrations were caused by the presence of the fin.

These vibration phenomena have been considered by several investigators in an attempt

to find their cause and to establish a fin configuration which would minimize the vibrations [5-7]. However, as yet a complete physical explanation of the cause and nature of this hydroelastic phenomenon has not been given.

It is the purpose of this paper to clarify the mechanism of occurrence and the fundamental properties of these vibrations and to suggest methods for minimizing them. It is also the purpose of this paper to find a fin configuration of simple shape suitable for practical application, with which minimum vibration as well as significant reduction in pitch can be expected. A thorough experimental study was carried out. The majority of the tests were made on 5.5-ft Mariner model, but supplementary tests were conducted on a 22-ft Mariner model in order to obtain the pressure distribution on the side of the bow and the vertical and horizontal acceleration distribution in the bow structure.

The paper is divided into two parts. The mechanism of occurrence of the induced vibrations and their fundamental properties are discussed in Part 1. Part 2 considers the effects of various parameters (such as fin configuration, size, and location) on ship motions and vibration.

## 1 Mechanism of Occurrence and Fundamental Properties of Induced Vibration

### Description of Experiment

#### Model and Fin Particulars

A 5.5-ft model of the Mariner was employed for the major part of the experiments. The lines of the model are shown in Fig. 1, and the characteristics of the model ship are given in Table 1. The model was made of fiberglass

Table 1 Characteristics of Model and Mariner

	Model	Mariner
Length, LOA, ft.....	5.86	563.64
Length, LBP, ft.....	5.50	528.00
Breadth, B, ft.....	0.79	76.00
Depth, D, ft.....	0.37	35.50
Draft, max, $H_{max}$ ft.....	0.31	29.75
Block, coefficient, $C_b$ .....	0.624	0.624
Prismatic coefficient, $C_p$ .....	0.635	0.635
Midship coefficient, $C_m$ .....	0.983	0.983
Water-plane coefficient, $C_w$ .....	0.745	0.745
Displacement, $\Delta$ .....	51.5 (lbs)	21,093 (ton)
Radius of gyration.....	0.24 L	0.24 L
Bulbous bow (% of midship area).....	4	4
Scale ratio.....	1	96.00

material having a Young's modulus of  $5.9 \times 10^5$  psi. It is desirable in hull-vibration tests in waves to employ a model whose construction

is similar to that of the actual ship, thereby satisfying dynamical similitude. Nevertheless, since the main purpose of this study was to clarify the basic properties of the fin-induced vibrations, it was decided (for practical reasons in construction of the model) to omit the dynamical similitude requirement. Therefore, the magnitudes of the horizontal vibrations measured in this test cannot be converted directly to full scale. However, the qualitative vibration characteristics of the model which are derived from the test results are applicable to a full-scale ship, and the magnitude of measured ship motions, vertical accelerations at ship bow, pressure on ship bow, pressure on ship bottom, and forces acting on the fin can be converted to full scale.

The natural frequencies for the first three modes of horizontal vibration of the model hull, measured in calm water, are in order, 33.8, 72.2 and 101.0 cps. Therefore, the hull natural frequency of the model increases almost linearly with increase in the mode of vibration and this tendency agrees well with that of the ship [11]. The natural frequency of the 1st mode of torsional vibration of the model hull is 97.7 cps, and this value is very close to that of the 3rd mode of horizontal vibration.

The fin used for the experiments described in this part of the paper was a simple aluminum plate of rectangular plan and cross section, and had no tip fences. It was installed at the keel line of the model. The principal characteristics of

Table 2 Fin Particulars (Fin A)

Location (aft of FP) .....	3.8 % L
Span, in.....	5.40
Chord, in.....	2.50
Aspect ratio.....	2.16
Breadth of flat bottom, in.....	0.20
Effective span, in.....	5.20
Effective aspect ratio.....	2.08
Effective area, sq in.....	13.00
Effective area/water-plane area, per cent.....	2.8

the fin (identified as Fin A) are listed in Table 2, and a photograph of the fin is shown in Fig. 2.

In addition to tests on a 5.5-ft model, supplementary tests were conducted on a 22-ft Mariner model. Particular effort was made in the latter tests to obtain the pressure distribution along the bow side as well as the distribution of the horizontal component of the vibration acceleration along the vertical centerline at the bow. Details of these tests are given in Appendix 1.

### Test Procedure

The experiments were conducted in the 140-ft basin of the David Taylor Model Basin using

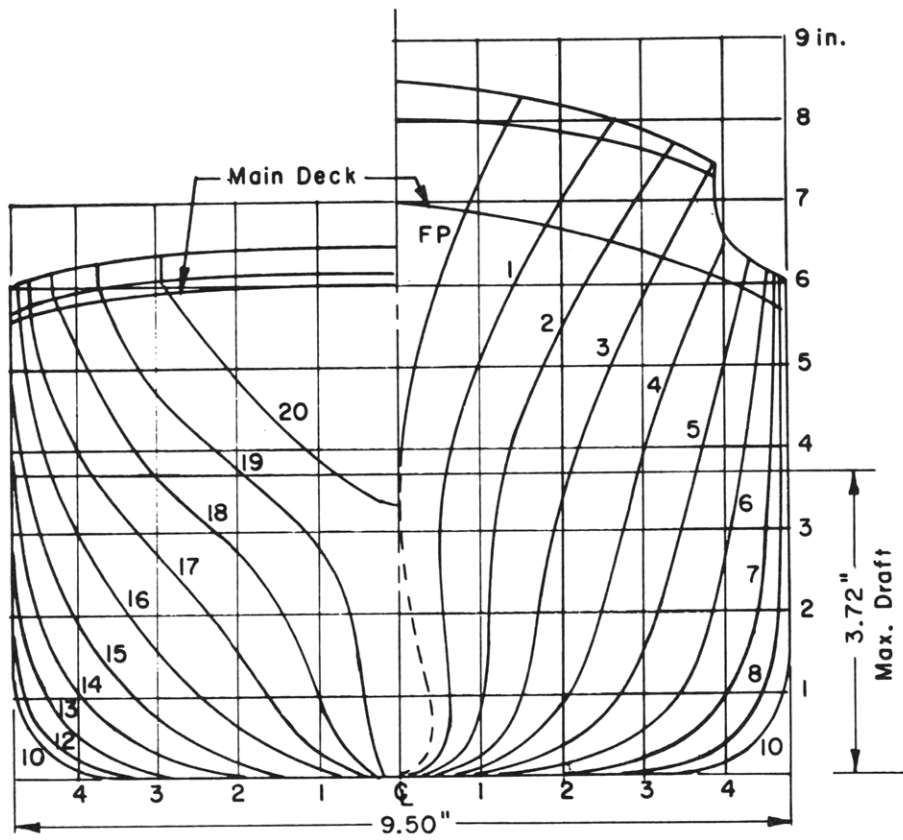


Fig. 1 Lines of a 5.5-ft Mariner Model

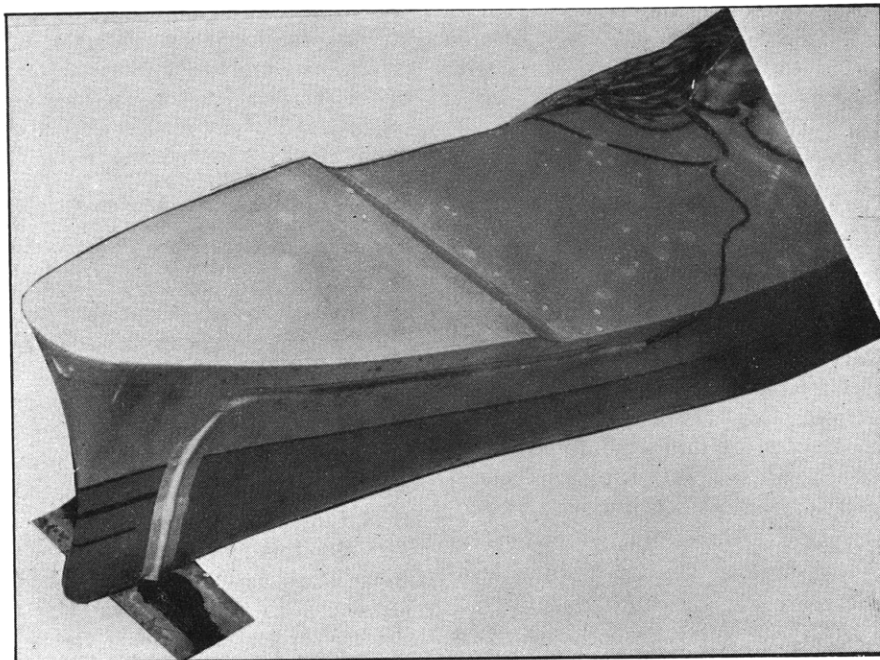


Fig. 2 Fin located at 3.8 per cent of model length aft of forward perpendicular

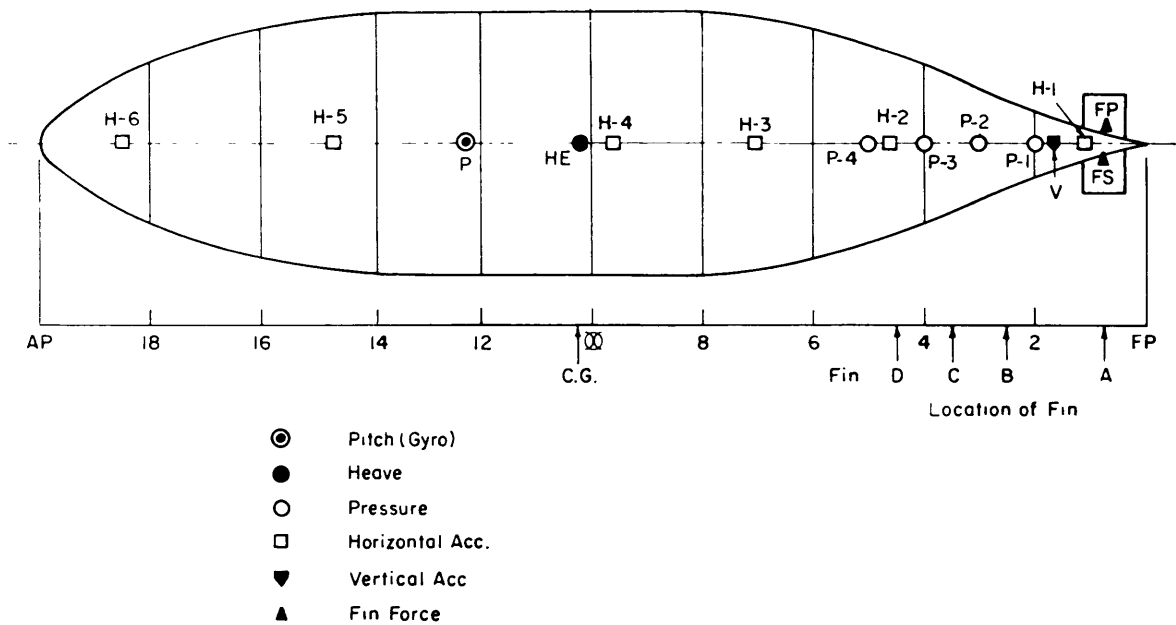


Fig. 3 Arrangement of recorders mounted on the model (5.5-ft model)

a gravity system for towing the model. The waves were generated by a pneumatic-type wavemaker, and the wave dimensions were recorded by a capacitance-type wave-height probe at one fixed point in the tank.

The model was towed at speeds ranging from 0 to 2.5 knots (0 to 24.5 knots full scale) at the maximum draft condition. The majority of the tests were carried out in regular waves of length equal to the model length ( $\lambda/L = 1.0$ ) and height of  $\frac{1}{20}$  of the wave length. Tests with other wave heights were also made at 1.27 knots (12.5 knots full scale), since this speed proved severe for this location of the fin.

In addition, a test in irregular waves was carried out at the low speed of 0.1 knot (1 knot full scale). For this test the wave height was recorded by a wave-height probe which was towed ahead of the model at model speed. The wave spectra obtained in these irregular waves are discussed in Part 2.

Resistance, pitch, and heave motion, vertical acceleration at ship bow, pressure on ship bottom, horizontal acceleration of hull, and force acting on the fin were measured. Pressures were measured at four locations on the keel line by Dynisco-type pressure gages. Horizontal vibrations of the hull were measured at six longitudinal locations along the centerline by Statham linear accelerometers. The frequency response of these meters was adequate for accurate measurement

of the phenomena under study. The intensities of the vibration were obtained in units of acceleration; however, these are expressed in terms of "arbitrary units" in the figures in this report.

Forces acting on the fin were measured by Baldwin wire strain gages  $\frac{1}{4}$  in. in length fixed at the root of the fin on both the upper and lower surfaces. Calibration of the strain gages was made outside of the tank by measuring strains produced by applying a series of pressures distributed uniformly over the fin. The location of the instrumentation used in the tests is shown schematically in Fig. 3 and a photograph of the installation is shown in Fig. 4.

#### Analyses of Experimental Results

##### Mechanism of Occurrence of Induced Vibration

*Conceivable Causes of Vibration.* The conceivable causes of vibrations induced by an anti-pitching fin will be discussed here with references to earlier studies.

First, the forces which may be considered as sources of the induced vibration are (a) force generated by vorticity, (b) force due to flow separation, (c) force associated with cavity formation and collapse, and (d) force associated with slamming. The force associated with slamming is an impact force applied on the lower surface of the fin when the ship's forefoot (and also the fin) emerges from the wave surface and reenters.



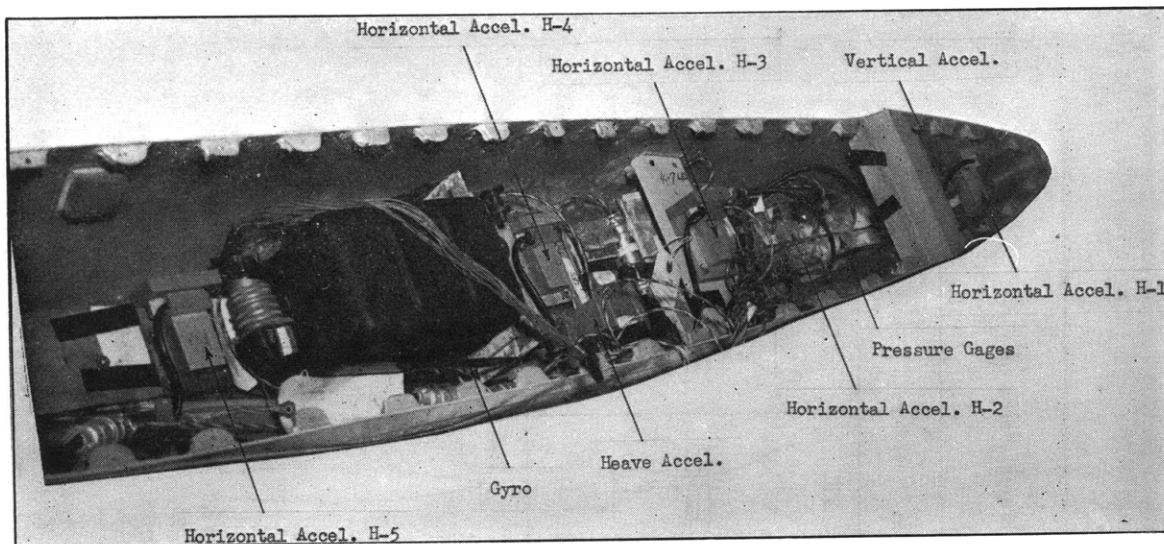


Fig. 4 Installation of recorders

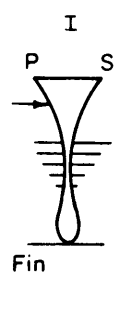
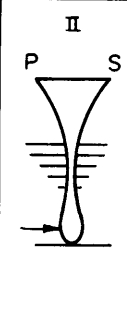
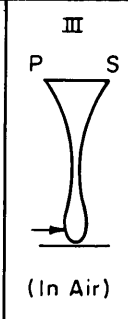
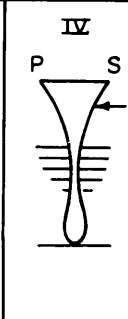
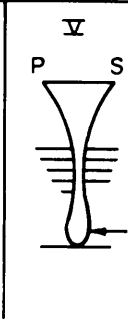
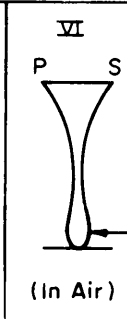
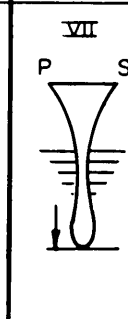
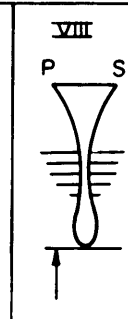
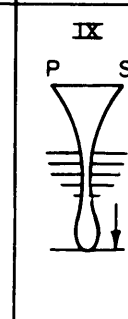
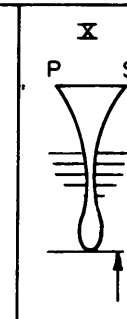
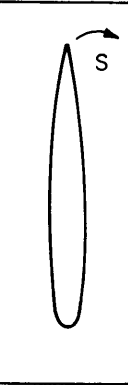
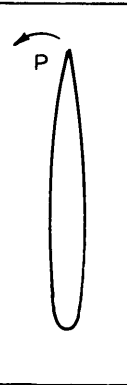
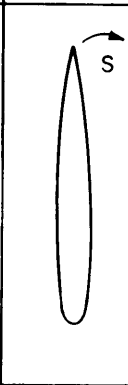
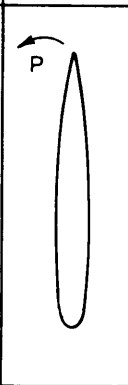
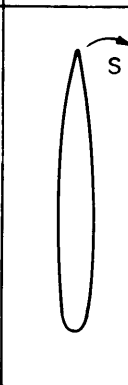





Next, the possible locations of the applied force which could induce vibration are (a) flare of bow, (b) bow side just above the fin, and (c) upper or lower surface of the fin. The combination of the force and its location will determine the mechanism of the occurrence of the induced vibrations.

Abkowitz believed that the cause of the induced vibrations was an impact force on the side of the ship hull above the fin and that this force was produced by collapse of an air bubble [8]. Goren and Pearlman came to the same conclusion [9]. Stefun and Schwartz concluded on the basis of their experiments that a lateral impact force on the flare and/or hull produced by flow separation caused the vibration [10]. Pournaras attributed the vibration to vorticity generated by fin motion [2]. In summary, all previous investigators believed the induced vibrations to be due to a lateral force applied to the ship bow flare and/or ship bow side above the fin, and the induced vibration was considered to be pure horizontal flexural vibration.

However, from an hydroelasticity point of view, there are two facts which suggest that the horizontal vibration induced by an antipitching fin is not necessarily a pure horizontal flexural vibration but a vibration initiated at least in part by a torsional vibration; specifically: (a) A lateral impact force applied to the ship bow side certainly causes a horizontal vibration if the location of the lateral force is near the center of rotation of the hull. However when the location of the lateral force is far below the center

of rotation of the hull, the force may cause a torsional vibration rather than a pure horizontal flexural vibration. (b) A horizontal vibration of the hull may be generated by applying an impact force to the fin itself. If a downward or upward force is applied to either the port or starboard fin, a horizontal vibration can be excited even in calm water, Fig. 6. Of course, a torsional vibration appears at the initial stage of impact; however, only a pure horizontal flexural vibration is apparent a few cycles after impact.

Then, the following question arises: What is the relation between the horizontal vibration of the hull and the torsion caused by the lateral force on the ship bow side and/or vertical force on the fin? To answer this question, it should be mentioned that, in the case of horizontal vibration of a ship's hull, the vibration can be excited either by a lateral force or by a *moment* about the longitudinal axis of the ship. In other words, there exists a coupling between torsional and horizontal flexural vibration. In torsional vibration of a hull, the rotation does not usually take place about the center of gravity of a particular section but about the center of rotation determined from geometric considerations. Thus, horizontal inertial forces which arise in torsion tend to produce vibration of the horizontal flexural type [12-16]. Vibration tests made on the *Gopher Mariner* indicated that torsional and horizontal vibrations occur simultaneously [11]. On the *Gopher Mariner*, the natural frequencies of the horizontal vibrations are very close to those of the torsional vibrations. For

	Horizontal Impact Force at Ship Bow						Vertical Impact Force on Fin			
	Port Side			Starboard Side			Port Fin		Starboard Fin	
Location of Force	I 	II 	III  (In Air)	IV 	V 	VI  (In Air)	VII 	VIII 	IX 	X 
Direction of Initial Deflection of Ship Bow *										

\* Referring to Ship Deck

Fig. 5 Relation between location of applied force and bow deflection obtained in calm water

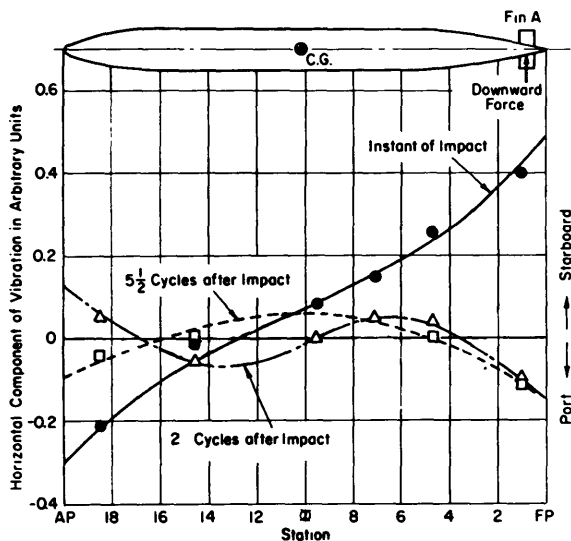


Fig. 6 Vibration pattern produced by applying an downward impact force on starboard fin in calm water

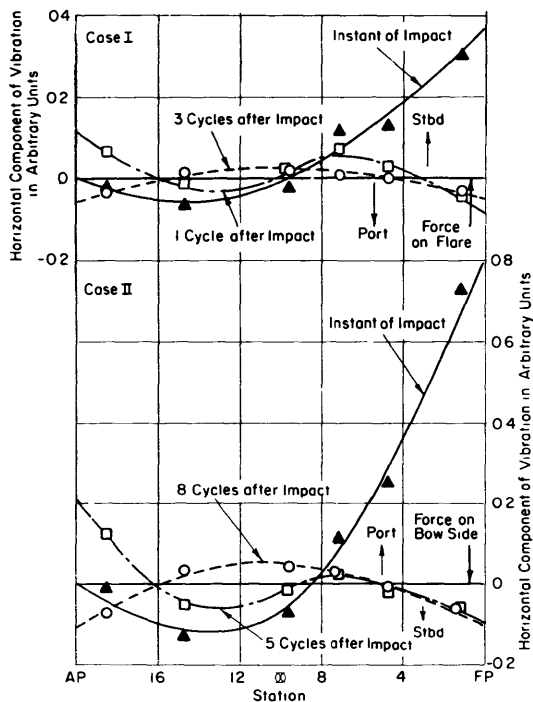


Fig. 7 Vibration pattern produced by applying an impact force on port bow flare and side in calm water

example, natural frequencies of the horizontal modes in succession, starting with the 1st mode were 135, 275 and 435 cpm, respectively, for light draft, while the natural frequencies of torsional modes in succession were 275, and about 400 cpm, respectively. Thus, the natural frequencies of the torsional modes are very

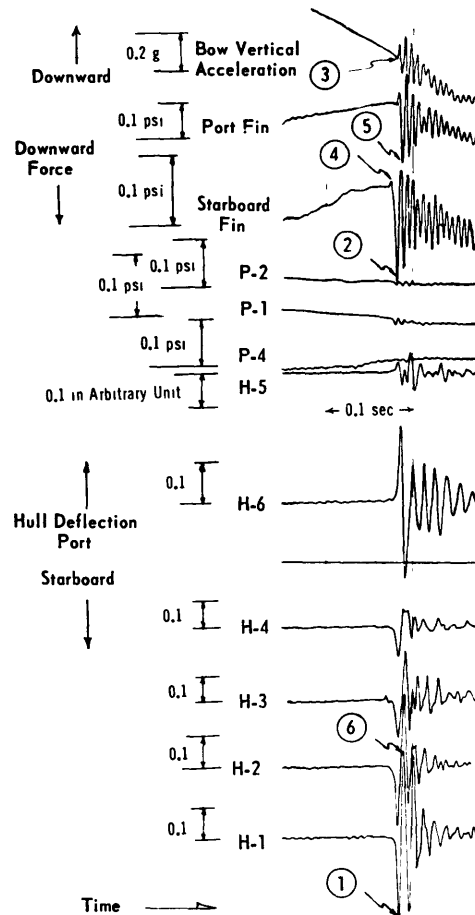
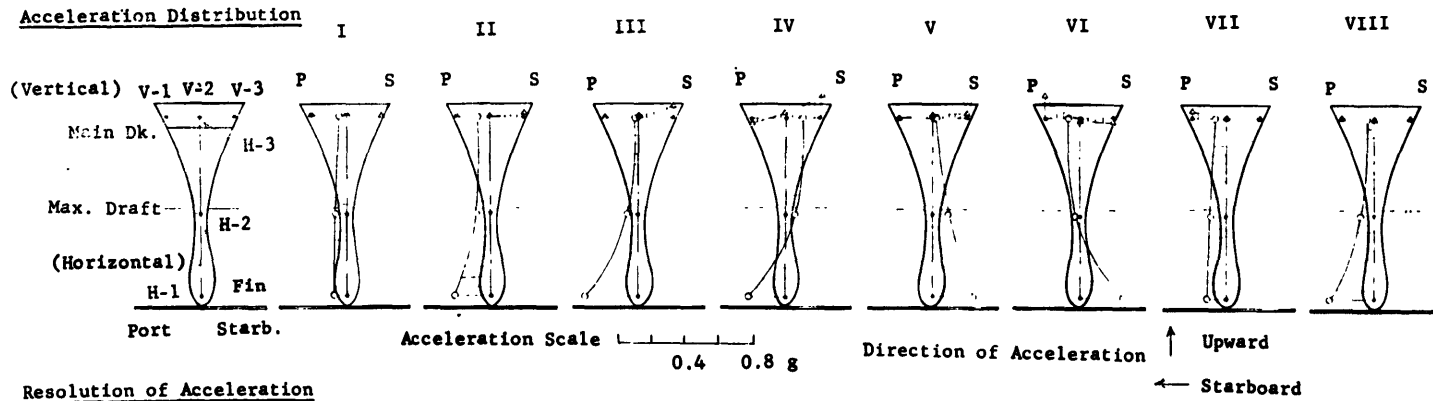


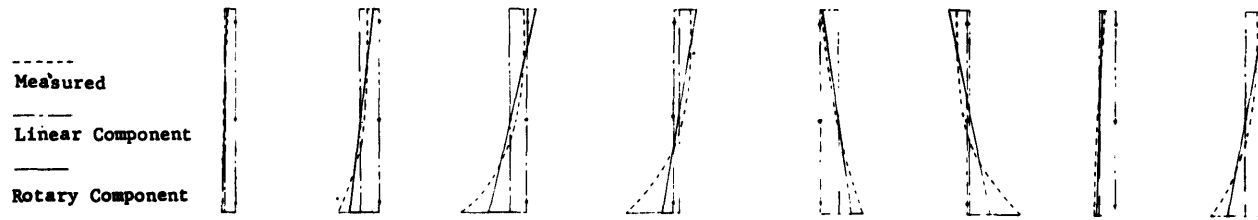
Fig. 8 Sample of record taken in the experiment.  $\lambda/L = 1.00$ ,  $b/\lambda = 1/20$ , model speed 0.66 knots (6.5 knots in full scale)

close to those of the next higher modes of horizontal flexural vibration. The fact that the horizontal and torsional natural frequencies of the Mariner-type ship are very close is not exceptional. Generally, the torsional natural frequency for the 1st mode is close to the horizontal natural frequency for the 2nd or 3rd mode [17,18]. Therefore, coupling between horizontal and torsional vibration is inevitable, and this coupling effect appears to be a factor in the serious horizontal flexural vibration of the hull. The validity of the foregoing discussion will be verified in the following sections.

*Location of Forces Producing Vibrations* The relation between the location of an applied force and the resulting bow deflection provides valuable information for determining the mechanism of occurrence of the induced vibration. For this purpose, a force was applied at various locations of the model bow as shown in Fig. 5, and the direction of the initial deflection of the bow was



Resolution of Acceleration



Pressure Distribution

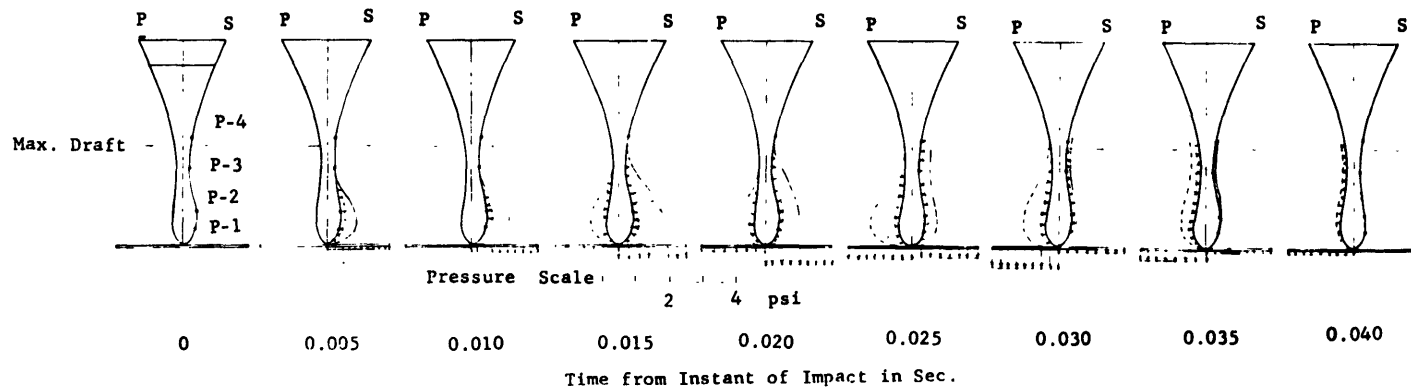


Fig. 9 Distribution of acceleration and impact pressure at section  $0.038 L$  aft of forward perpendicular (22-ft Mariner model,  $\lambda/L = 1.0$ ,  $b/\lambda = 1/20$ , speed = 1.23 knots)

determined. The tests were carried out in calm water and the deflection of the bow was measured on the deck, primarily at a location at 5.3 per cent of the model length aft of the forward perpendicular. Cases VII, VIII, IX and X in the figure are especially interesting. These cases illustrate the results of applying a downward or upward force to the fin.

One example of the test results is shown in Fig. 6. This figure shows the pattern of vibration produced by applying a downward impact force to the starboard fin (Case IX in Fig. 5). The deflection of the ship bow at the instant the force is applied is to starboard, while the stern undergoes port deflection. There is a zero deflection point midway between amidship and stern, and this pattern is typical of a torsional vibration. Higher modes of horizontal vibration are apparent even in this case. The impact force appears to set up a vibration where torsional and the 2nd or 3rd mode of horizontal vibration are coupled. However, the higher modes of vibration decay quickly since they are subject to greater damping, so finally only the fundamental mode remains.

Fig. 7 shows examples of Cases I and II in Fig. 5. Case I is for a lateral impact force applied near the flare on the port bow, and Case II is for a lateral impact force applied above the port fin. Note that torsional vibration appears to be involved at the initial stage of the vibration, even when a lateral impact force is applied above the fin or on the flare.

The foregoing experimental results suggest that the generated vibration involves torsional vibration at the initial stage, irrespective of whether the impact force is applied to the bow side above the fin, to the flare, or to the fin itself. It should be noted that the pattern of the initial stage of the vibration, recorded whenever the hull vibration occurs in waves, is similar to that shown in Fig. 6 (for examples, see Fig. 17).

Fig. 8 shows a sample record at the instant the vibration starts. The model speed for this case was 0.66 knot (6.5 knots full scale), and the ship's forefoot did not emerge. P-1, P-2, and P-4 in the figure are the pressure records measured on the ship bottom at  $0.10L$ ,  $0.15L$ , and  $0.25L$  aft of the forward perpendicular, respectively. H-1, H-2, H-3, H-4, H-5, and H-6 are the horizontal component of vibration, measured at  $0.053L$ ,  $0.232L$ ,  $0.352L$ ,  $0.480L$ ,  $0.735L$ , and  $0.928L$  aft of the forward perpendicular, respectively. Records of the vertical acceleration at the ship bow, and the forces on the port and starboard fins are also shown.

As seen in the figure, large vibrations appeared for this condition. The deflection pattern of the

hull at the instant the vibration starts is exactly the same as that resulting from an impact force to the fin in calm water, Fig. 6, since the vibration records H-1, H-2, H-3, and H-4 indicate a starboard deflection of the bow while records H-5, and H-6 exhibit a port deflection of the stern.

An important relation between the horizontal component of vibration and the force on the fin can be derived from this figure. The ship bow experiences a large starboard deflection at the instant the vibration occurs (see Mark ① in the figure). On the other hand, from Fig. 5, conceivable sources which produce a starboard deflection of the ship bow, are as follows:

Case I Lateral force applied to the port flare.

Case V Lateral force applied the starboard bow above the fin.

Case VIII Upward force applied to the port fin.

Case IX Downward force applied to the starboard fin.

The large starboard deflection of the ship bow seen in the sample in Fig. 8 should correspond to one of these four cases. Since a slow-motion movie taken in the experiments, Fig. 12, showed no water spray around the bow flare at the initial occurrence of vibration, Case I can be ruled out. Furthermore, since no force is observed on the port fin at the instant the vibration occurs, Fig. 8, the possibility of Case VIII being the source becomes slim.

There remain Cases V and IX. Now, in Fig. 8, a strong downward force is recognized on the starboard fin at the instant the vibration occurs, Mark ②. This follows precisely the general rule given as Case IX in Fig. 5. Also, a slight downward impact acceleration of the ship bow can be observed at this instant on the record of vertical acceleration at the ship bow, Mark ③. The direction of this acceleration is opposite to that observed at the instant of slamming. This verified the existence of some downward force on the fin. This force will be identified and discussed in detail in the next section.

Note that a very small upward force occurs on the starboard fin just prior to the severe vibration, Mark ④. This force results from suction on the upper surface of the fin, but it is too small to cause horizontal vibrations of the hull.

Of great importance is the time difference between the loading on the port and starboard fins. The downward force on the port fin, Mark ⑤, occurs a short time after the downward force on the starboard fin, Mark ②, in this sample. Since the downward force on the port fin produces a port deflection of the ship bow, Case VII in Fig. 5, and since it is superimposed

on the latter half-cycle of the vibration initially generated by the force on the starboard fin, an appreciable augmentation of the vibration results, Mark ⑥.

When the fin emerges from the water surface, the same principles still apply. For example, the forefoot emerges from the water surface at a model speed of 1.10 knots (10.8 knots full scale) as evidenced by the presence of a small impact pressure recorded by the foremost pressure gage. In this case, an upward force due to slamming is applied to the fin. The response of this upward force is the starboard or port deflection of the ship bow depending on which side is loaded first, and thus corresponds to either Case VIII or X in Fig. 5. A short time later, a downward force is applied to the fin and a pattern similar to that shown in Fig. 8 results.

The foregoing analysis of the phenomenon proves that a vertical force applied to the fin contributes to the vibration. Next, it must be determined whether or not a lateral impact force applied to the bow just above the fin (not to the flare) is also a factor in the vibration. This corresponds to the Case V in Fig. 5. Unfortunately, the pressure distribution along the bow could not be measured for the 5.5-ft model. For an evaluation of the contribution of the lateral force, a 22-ft model was tested (Appendix 1).

Fig. 9 shows an example of the results obtained in regular waves on this longer model. The wave length was equal to that of the model ( $\lambda/L = 1.00$ ), wave height was  $1/20$  of the wave length, and model speed was 1.23 knots (6.1 knots full scale) for this case. Thus, the speed corresponds approximately to that for the sample shown in Fig. 8. The figures in the top row show the time history of vibration acceleration distribution at the section where the fin is installed; ( $0.038L$  aft of the forward perpendicular). The figures in the middle row show the resolved linear and rotary components of the accelerations, and the figures at the bottom show the time history of the impact pressure distribution on the ship's bow sides and on the fins. The time interval between two adjacent figures is 0.005 sec (0.025 sec full scale), starting from the instant the vibration appears.

The pressures were not measured on both sides but on the starboard side of the bow only. However, the time history of the pressure on the port side may be assumed approximately equivalent to that on the starboard, since the time histories of the pressures on the port and starboard fins are nearly equal at corresponding instants in their cycles. For example, in Fig. 9, pressure variation with time on the port fin is almost equal

to that on the starboard fin with a time lag of 0.01 sec (model scale). Then, the time history of the pressure distribution on the port side of the bow may be obtained from that on the starboard side, using the same time lag as observed for the fin. The results are shown in dotted lines in the figure.

It is clear from Fig. 9 that the vibration starts as soon as an impact pressure is applied to the bow and to the fin. In this case, impact pressures start first on the starboard side and fin. Then, a short time later, 0.01 sec (model scale), impact pressures appear simultaneously on the port side and fin.

Note that the pressures acting on bow side and fin have a duration of 0.035 sec (0.17 sec full scale) from the instant of impact, and the maximum impact pressure on the bow side occurs 0.005 sec (0.025 sec full scale) earlier than the maximum impact pressure on the fin (see also Fig. 11). These values were consistent for all speeds investigated. However, the time differential of loading on the port and starboard sides is not consistent, even in regular waves. This differential will be discussed in detail in a later section.

The region of the ship's bow side where an impact pressure is applied can be estimated from Fig. 9. It is clear that the impact pressure acts between the fin and the maximum waterline.

Thus, there is no doubt that the lateral impact pressure applied to the ship's bow just above the fin is also a factor in producing the vibration. In summary, it can be concluded that the *vertical impact pressure* applied to the fin and the *lateral impact pressure* applied to the ship's bow both contribute to the vibration.

*Nature of the Induced Vibration.* In the past it has been thought generally that the vibration induced by an antipitching fin is a horizontal flexural vibration, since models and ships equipped with antipitching fins have experienced horizontal vibration. However, as discussed in the foregoing, there is always coupling between torsional and horizontal flexural vibrations of a ship. Also, the conclusion was derived that a vertical impact pressure on the fin and a lateral impact pressure on the ship's bow are both sources for producing the vibration. Therefore, the question arises as to whether the induced vibration is a pure horizontal flexural vibration or a torsional vibration. The nature of the induced vibration will be clarified in the following text.

The distribution of the horizontal component of acceleration along the centerline of the ship bow section and of the vertical acceleration on the deck level of the section is shown at the

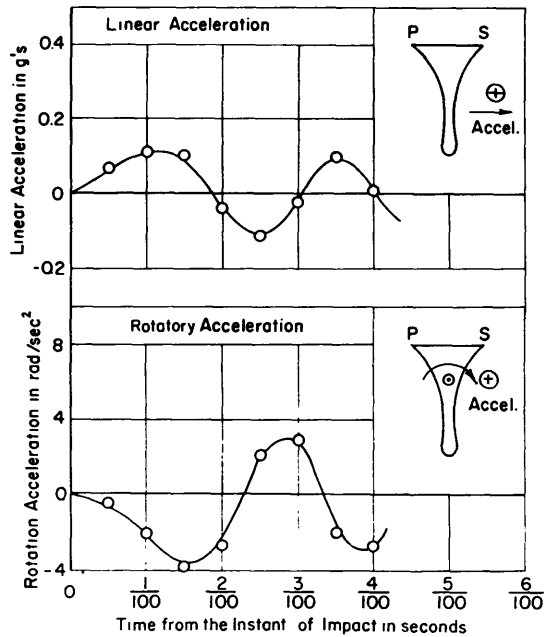


Fig. 10 Time history of linear and rotary component of acceleration (22-ft Mariner model,  $\lambda/L = 1.0$ ,  $b/\lambda = 1/20$ , speed = 1.23 knots)

top of Fig. 9 for various times after impact. These accelerations are vibration accelerations resulting from an impact and are not due to ship motions. The figures show clearly that torsion is involved in the vibration. However, from these figures, it is not possible to evaluate the relative magnitude of the horizontal (linear translation) and the torsional (rotary) components of the vibration. In order to estimate these two components, the measured accelerations were resolved by means of the least-square method discussed in Appendix 2. The results are shown in the middle row of Fig. 9. Since accelerations were measured at only three vertical positions on the centerline of the section, some errors may be involved in the analysis. However, the results show clearly that the acceleration contains a rotary component as well as a linear component.

It is assumed in the analysis that rotary motion for a particular section takes place about the center of shear of that section, and that this center of shear corresponds to the centroid of the area of the section [19]. The position of the center of shear at the model section where the accelerations were measured was 14.1 in. (28.3 ft full scale) above the base line. The center of shear for the same station on *Gopher Mariner* is 27.5 ft above the base line [19], a value very close to that obtained here.

Note that the center of rotation (center of

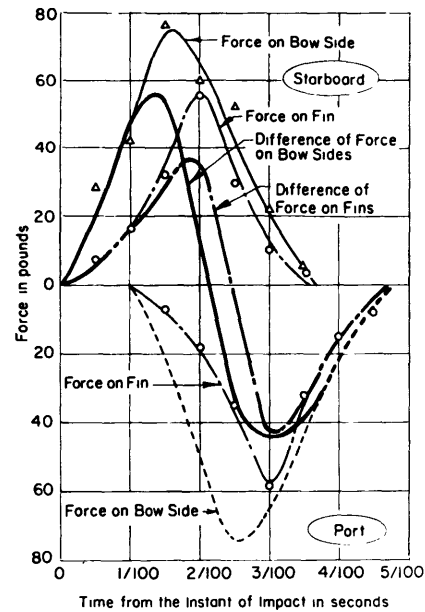


Fig. 11 Time history of forces acting on bow side and fin (22-ft Mariner model,  $\lambda/L = 1.0$ ,  $b/\lambda = 1/20$ , speed = 1.23 knots)

shear) at the section is far above the area of application of the force. This means that the applied forces produce a rotary motion about the center of rotation. However, because of the lack of symmetry of ship sections, a linear translatory motion is also associated with a rotary motion. Thus, in general, the torsional and horizontal flexural vibrations coexist for any ship. *In brief, it may be concluded that the vibration induced by an antipitching fin essentially starts as a torsional rather than a horizontal flexural vibration.*

Let us examine the relation between the linear and rotary components of the vibration in more detail. Fig. 10 shows the variation with time of linear and rotary accelerations shown in Fig. 9. Note that the natural frequencies of the horizontal flexural and torsional vibration of the model are very close. Since the model was not structurally scaled down, the natural frequencies are applicable for this model only. However, the natural frequency of the 1st mode of torsional vibration of the model, 47.5 cps, is close to that of the 2nd mode of horizontal flexural vibration, 45.4 cps. The 1st mode of the horizontal flexural vibration which remained several cycles after the impact was 21.3 cps.

One may question why the period of the first half-cycle of the vibration seen in Fig. 10 is larger than the following ones. The explanation is that the response of a system to an impact load is

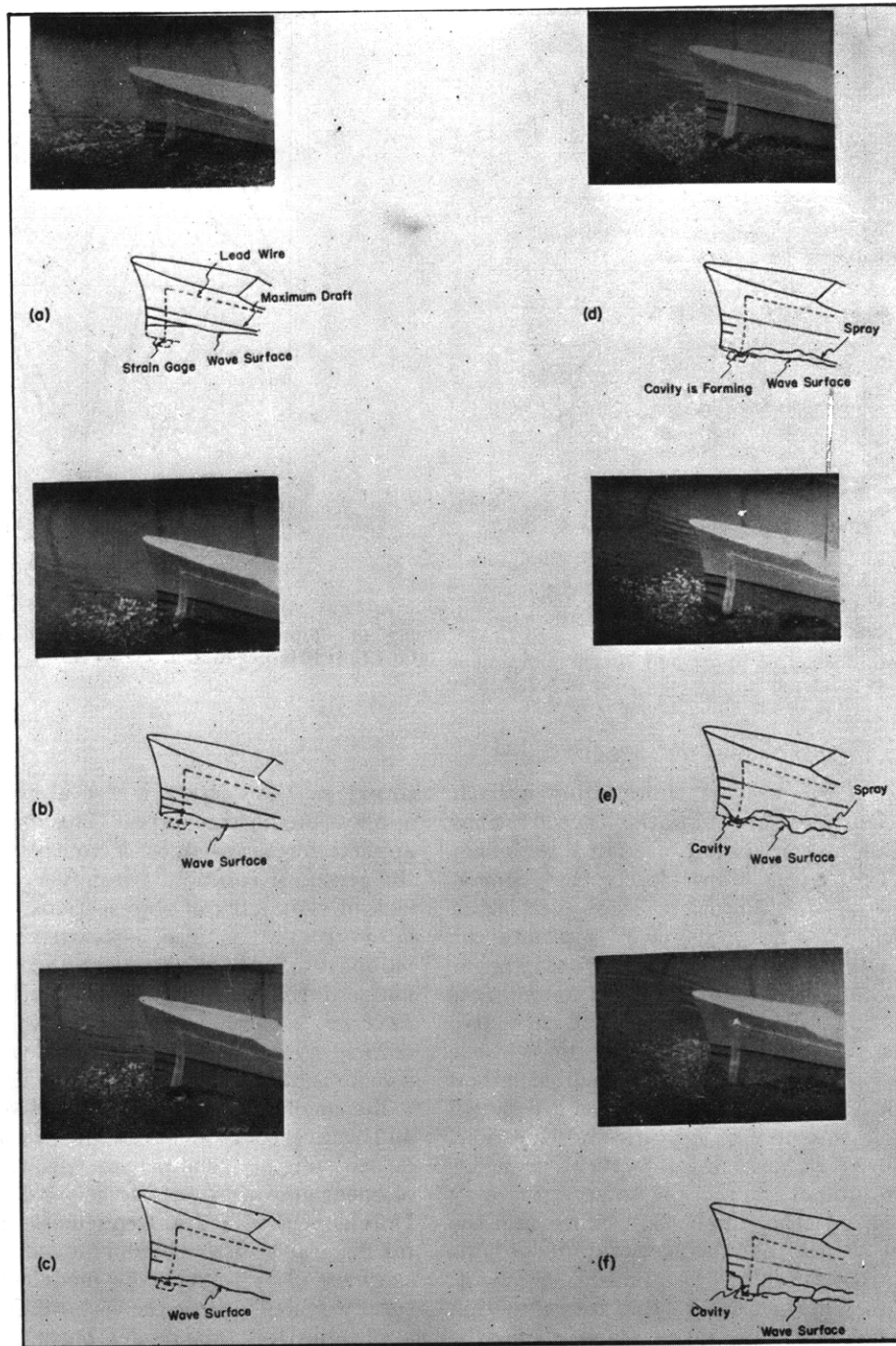


Fig. 12 Photograph duplicated from slow-motion movie.  $\lambda/L = 1.00$ ,  $b/\lambda = 1/20$   
 model speed 0.90 knots (8.8 knots in full scale)

determined by the ratio of the duration of the loading to the natural period of the responding structure. In this case, the duration of the loading was longer than the natural period of the model as can be seen by comparison of Figs.

10 and 11; hence, the response is regulated by the duration of loading. A detailed discussion of the effect of duration of impact in transient phenomena is given in [20].

In order to examine the phase relation between



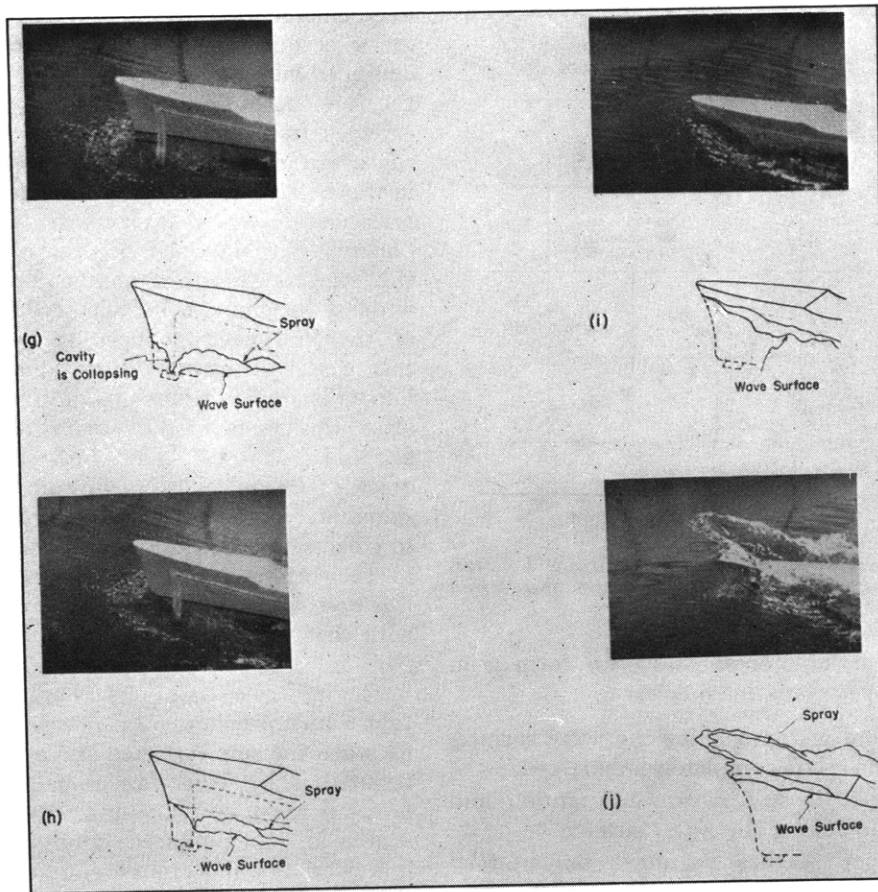


Fig. 12 (Continued)

the induced vibration and impact force, the impact pressures acting on the bow and the fin were integrated individually, and the impact forces were obtained at every instant, Fig. 11. Included also are the differences between the forces on the port and starboard sides. These differences of impact forces show roughly a sinusoidal form for both the bow side and the fin. In this case, the impact forces on the bow and fin reach their first maximum on the starboard side about 0.015 and 0.020 sec (model scale), respectively, after the start of impact, and indeed the forefoot swings to the port at this moment, accompanied by large rotation, Fig. 9, III. The impact force reaches its second maximum on the opposite side about 0.030 sec (model scale) after the instant of impact, and the bow bottom responds by a starboard deflection, Fig. 9, IV.

Another question arises; namely, how severe does the vibration become if the period (duration) of the impact force is close to the natural period of torsional vibration of the hull? This is a

problem of augmentation. Augmentation did not occur in the sample shown in the figure, since the period of the impact force is about twice that of the natural torsional vibration in this case. However, augmentation phenomena often appeared in the test and severe vibration did occur. This will be discussed in detail in a later section.

*Nature of Vibration-Inducing Force.* The discussion so far has pertained to the location of the force and nature of the hull vibration in a ship equipped by an antipitching fin. In this section, the nature of the unknown impact force will be discussed.

First, the case when the ship's fore body does not emerge from the wave surface will be examined. This is the most typical case for an actual ship, when equipped with an antipitching fin.

Fig. 12 shows several photographs reproduced from a slow-motion movie, taken with model speed of 0.90 knot (8.8 knots full scale). The fig-

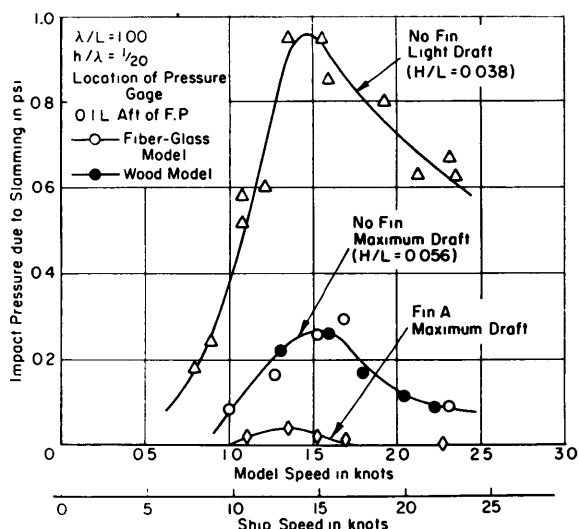


Fig. 13 Impact pressure due to slamming at location 0.10L aft of forward perpendicular versus ship speed

ure illustrates the process of cavity formation. A step-by-step explanation follows:

(a) Forefoot is approaching the wave surface, however the fin is still completely submerged.

(b) Forefoot is still in upward motion and the fin is approaching the water surface.

(c) Forefoot reaches its maximum upward position, and the submergence of the fin is a minimum. A slight disturbance of the water surface produced by the fin can be observed.

(d) Bow is starting downward, and formation of the cavity is begun.

(e) The cavity on the fin continues to develop. A small suction (upward force) appears on the fin at this instant.

(f) The cavity reaches maximum size.

(g) The cavity begins to collapse resulting in the fin being covered by water. A lateral impact force is applied to the bow side, a downward impact force is applied to the fin surface, and torsional as well as horizontal flexural hull vibrations occur at this moment. A time difference between the occurrence of the force on port and starboard sides is recognized in most cases.

(h) The collapse of the cavity is almost completed.

(i) The bow submerges and severe horizontal and slight vertical vibrations continue.

(j) The submergence of the bow continues and significant spray is observed when the water hits the bow flare.

Now, consider the case when a ship's forebody emerges from the wave surface. The forebody

may emerge from the water surface somewhat for a certain range of ship speeds even when equipped with an antipitching fin. For instance, the forefoot of the model equipped with Fin A emerges from the water surface slightly within the speed range from 11 to 16 knots full scale in wave of  $h/\lambda = 1/20$ . This is evidenced by the presence of a small impact pressure on the ship bottom, Fig. 13. Although the magnitude of this impact pressure is small compared with that without the fin, it is sufficient to cause vibration of the hull even at deep-draft condition. In this case, an upward force due to slamming impact is applied to the fin, and then immediately after this impact, the cavity appears on the fin and collapses. This process is completed quickly; however, the upward force due to slamming impact and the downward force due to collapse of the cavity can be identified clearly in the record. Also the induced vibration in this case is generally more severe in comparison with that when the fin does not emerge (see Fig. 29).

The full-scale trials on the *Compass Island* show that a high pressure (72 psi) was observed on the fin when the ship slammed and a severe vibration resulted [1]. Thus, an impact force on the fin as a result of slamming is also a significant source of the induced vibration. Especially, it may be very important for light-draft conditions, since the impact force due to slamming is significantly larger at light draft than that at deep draft, Fig. 13.

In summary, the mechanism of occurrence of the induced vibration is of the two types shown in Fig. 14, the "cavity type" and the "slam-plus-cavity type." The cavity type occurs with deep draft when the ship's forefoot does not emerge. The slam-plus-cavity type is generally associated with light-draft conditions when the ship's forefoot emerges.

A significant and common feature in both types, however, is that the impact force, due to collapse of the cavity and/or ship slamming, is applied to the bow side and to the fin. Therefore, in the design of an antipitching fin, it is of importance to minimize this impact force as well as to minimize the pitching motion.

#### Fundamental Properties of Induced Vibration

Before discussing the fundamental properties of the vibration induced by an antipitching fin, it may be well to give a review of the induced vibrations which were actually observed.

As mentioned previously, the vibration generally shows the fundamental mode of torsional and the higher modes of horizontal flexural vibration initi-




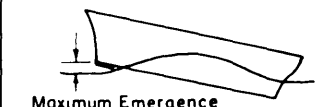

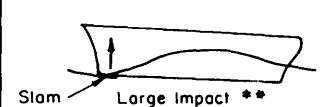
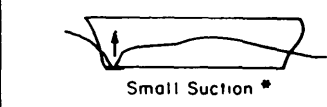
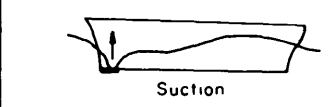
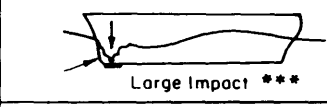
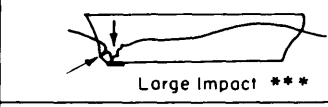


	Type I Cavity Type	Type II Slam and Cavity Type
1	Bow Upward Motion 	Bow Upward Motion 
2	Bow Approaches Wave Surface  Minimum Submergence	Bow Emergence  Maximum Emergence
3	Formation of Cavity  Cavity	Slam Impact  Slam Large Impact **
4	Fully Developed Cavity  Small Suction *	Fully Developed Cavity  Suction
5	Collapse of Cavity  Large Impact ***	Collapse of Cavity  Large Impact ***
6	Bow Submergence 	Bow Submergence 
<p>* Very Small Vibration Occurs, Occasionally  ** Severe Vibration (Seldom for Deep Draft)  *** Severe Vibration Occurs</p>		

Fig. 14 Explanatory figure of mechanism of occurrence of induced vibration

ally. However, owing to the fact that the higher modes of horizontal vibration decay at a faster rate than the fundamental mode, the latter remains and persists for some time. This vibration is created by nonsynchronous loading of the two sides of the bow-fin structure. Therefore, the vibration of the ship hull shows a complex pattern in the progressive stages of vibration. Nevertheless, it is of importance for design purposes to estimate the maximum intensity of vibration at various positions along the ship length during one cycle of encounter.

In Fig. 15, the maximum measured vibration, irrespective of time is shown rather than simultaneous values obtained at specific times. The

figure admittedly does not show the pattern of hull vibration, but serves to indicate the location along the hull where the severest vibration occurs. The intensity of the induced vibration is always maximum at the ship bow and is minimum at a position just aft of midships, independent of ship speed. The same trend was observed, irrespective of fin location.

It is of interest to know whether or not the induced vibration is of constant magnitude, and whether the initial displacement of the bow is in the same lateral direction for every cycle of encounter, when wave conditions and speed are constant. Fig. 16 shows the intensity of the vibration measured at a location of  $0.053L$  aft

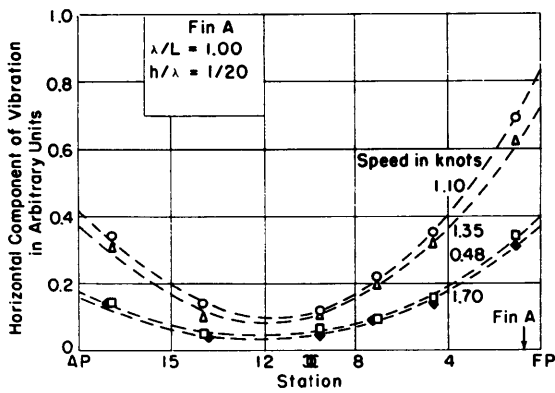


Fig. 15 Distribution of maximum intensity of induced vibration during one cycle of encounter at various ship speeds

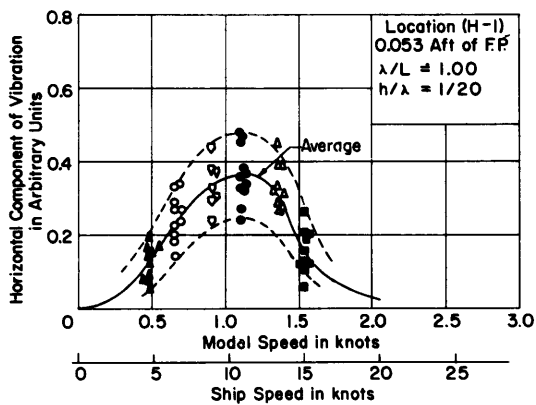


Fig. 16 Intensity of induced vibration at the instant vibration occurs, measured at  $0.053L$  aft of forward perpendicular versus ship speed at each encounter

of the forward perpendicular at the instant the vibrations starts. In the figure, for any particular speed,  $V$ , a wide variation in the intensity of the vibration can be seen from cycle to cycle even in the tests in regular waves.

Fig. 17 shows samples of the vibration pattern at the instant the vibration starts, for several cycles of encounter, at ship speed of 1.10 knots (10.8 knots full scale) with  $\lambda/L = 1.0$ . The figure clearly shows that the bow acceleration occurs randomly to port or starboard with different magnitudes, even in regular waves. This is due to a very small difference in time between the collapse of the cavity on the port fin and that on the starboard fin. This time difference was of the order of  $3/10^3$  to  $7/10^3$  sec in these model tests.

To conclude, the induced vibration is not of the same intensity and the initial bow displacement is not in the same lateral direction for each encounter, but instead they vary from cycle to

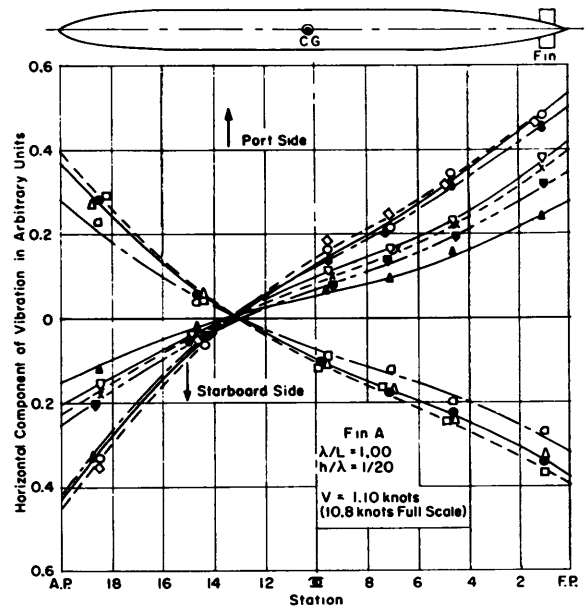


Fig. 17 Sample of vibration pattern at instant vibration occurs measured at each encounter. Ship speed 1.10 knots (10.8 knots full scale)

cycle, even in regular waves. The reason for this must depend on the random nature of collapse of the cavity on the fin. It may be expected, however, that in confused seas the intensity of the induced vibration and the direction of the initial deflection of the ship bow will be mostly governed by the unsteady motion of the ship bow due to rolling and/or yawing.

*Effect of Ship Speed.* The intensity of the horizontal vibration measured at  $0.056L$  aft of the forward perpendicular, as a function of ship speed, is shown in Fig. 18. The maximum peak-to-peak value of the vibration was taken for every cycle of encounter for a particular test condition. These values were averaged and plotted. The figure indicates that the intensity of the induced vibration is a function of ship speed. The intensity of the induced vibration becomes maximum at a certain ship speed, and no vibration appears in these regular waves at zero ship speed or at high speeds. Of course, the speed range within which the vibration occurs and the intensity are functions of configurations, locations, and size of the fin. They are also functions of wave length, height, and ship draft. However, the general tendency shown in Fig. 18 is applicable for any case.

Note that a small horizontal vibration also appears within a certain speed range without the fin. This is associated with slamming, since Fig. 13 shows that slamming occurs within this speed range. Slight slamming occurs for the

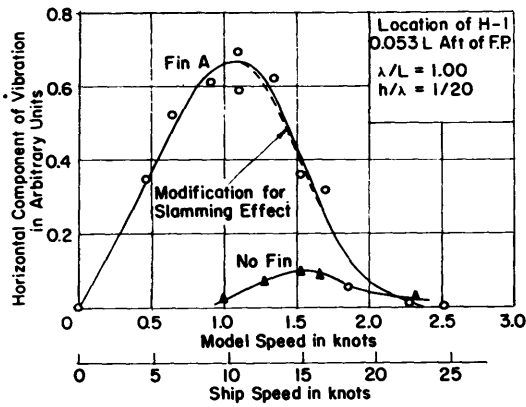


Fig. 18 Horizontal component of vibration measured at 0.053 L aft of forward perpendicular versus ship speed.  $\lambda/L = 1.00$ ,  $b/\lambda = 1/20$ . (Average of the maximum peak-to-peak value for all cycles of encounter)

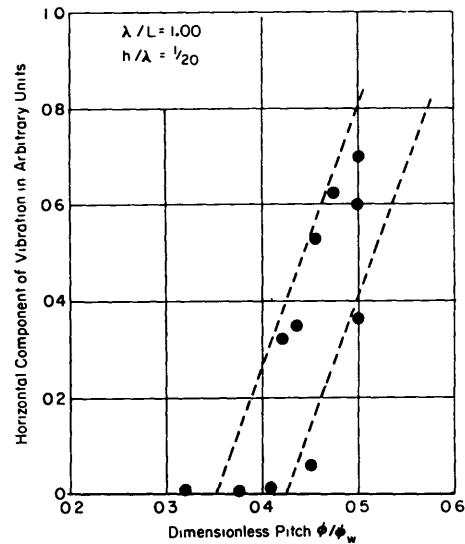


Fig. 20 Relation between pitching motion and intensity of induced vibration for Fin A

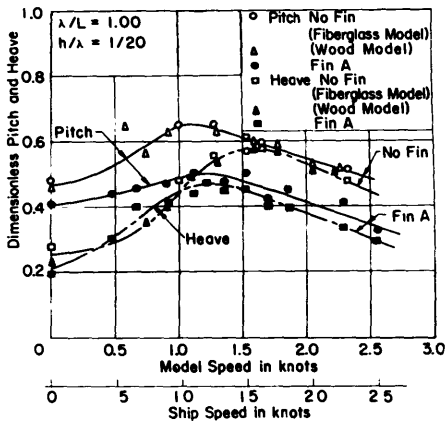


Fig. 19 Pitch and heave motions versus ship speed with Fin A and without fin

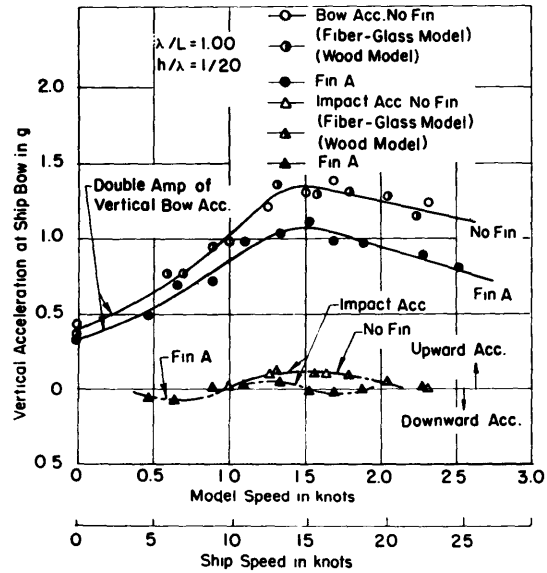


Fig. 21 Vertical acceleration at ship bow versus ship speeds

model equipped with Fin A within a specific speed range, Fig. 13, so the measured horizontal vibration for Fin A, Fig. 18, may contain a component of horizontal vibration associated with ship slamming.

To obtain the horizontal vibration due only to the fin, this component caused by slamming should be subtracted from the observed values. For this, the assumption is made that the intensity of the horizontal vibration is proportional to the magnitude of impact pressure on the ship bottom. The validity of this assumption will be discussed in a later section. Now, let the magnitudes of the impact pressure on the bottom at  $0.1L$  aft of the forward perpendicular with Fin A and with no fin be  $p_{*A}$  and  $p_{*0}$ , respectively; and let the magnitude of horizontal vibration at the ship bow due to slamming correspondingly be  $\beta_{*A}$  and  $\beta_{*0}$ .

Then, by the foregoing assumption, the horizontal vibration due to slamming,  $\beta_{*A}$ , is given by

$$\beta_{*A} = \beta_{*0} \left( \frac{p_{*A}}{p_{*0}} \right)$$

The horizontal vibration caused by Fin A is then

$$\beta'_A = \beta_A - \beta_{*A}$$

where

$\beta'_A$  = intensity of horizontal vibration caused by Fin A

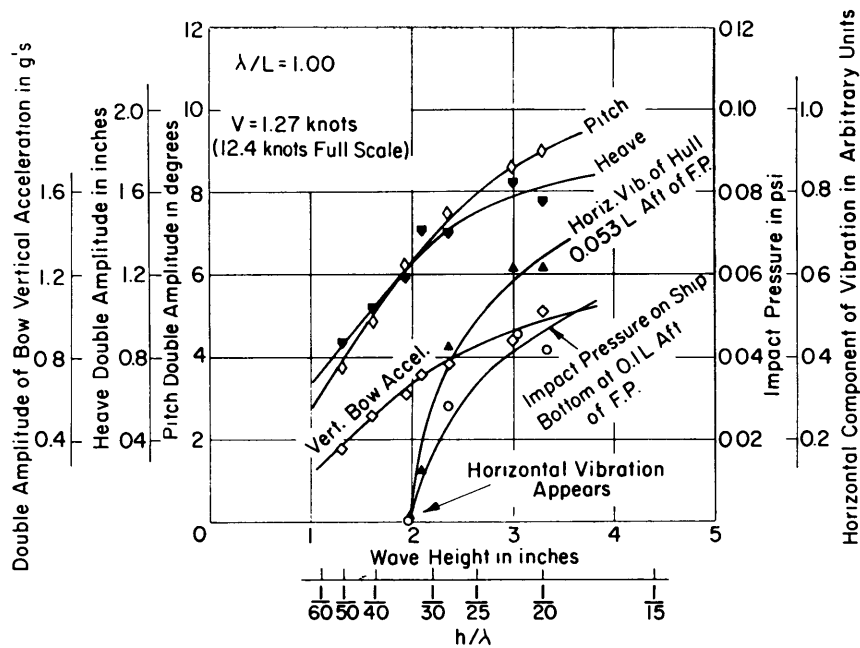


Fig. 22 Effect of wave height on pitch and heave motions, vertical bow acceleration, impact pressure on ship bottom, and intensity of horizontal component of vibration  $\lambda/L = 1.00$ , ship speed 1.27 knots (12.5 knots full scale)

$\beta_A$  = intensity of measured horizontal vibration

$\beta_{*A}$  = intensity of horizontal vibration associated with slamming

This modification for slamming effect was made at various ship speeds, and the magnitude of the horizontal vibration caused by the fin only is shown in Fig. 18. The result shows that the component of the vibration associated with slamming for the fin-equipped model is negligibly small in comparison with the severe horizontal vibration caused by the fin.

*Relation between Pitching Motion and Induced Vibration.* The full-scale trials carried out on the USS *Compass Island* equipped with an anti-pitching fin indicated that no clear relationship could be established between the pitching magnitude and that of the associated vibration. However, our data were examined to determine whether or not the intensity of the induced vibration is related to the amplitude of pitching motion.

The pitching and heaving motions, as a function of speed, for the model both without and with Fin A are shown in Fig. 19. Experimental data from tests on the 5.5-ft wood model of reference [21] are also included in the figure for comparison. The results obtained on the wood model show good agreement with those obtained

on the fiberglass model. The figure shows a significant reduction in pitch for the Mariner equipped with Fin A for all speeds, about 20 to 23 per cent at low speeds and about 30 per cent at high speeds. No reduction in heaving motion can be expected at low speeds, but about 30 per cent reduction occurs at speeds above 15 knots full scale.

From Figs. 18 and 19, the relation between the pitching motion and the intensity of the induced vibration can be obtained, and the result is shown in Fig. 20. The figure shows that the vibration occurs only when the pitching motion exceeds a certain limit, and its intensity increases linearly with increase in pitch. The same tendency is also indicated by the test results in irregular waves, as will be seen later.

Vertical accelerations at the ship bow, as a function of speed, are shown in Fig. 21. The vertical acceleration is the combination of the pitch and heave accelerations with due consideration to their phase relation. Although only slight reduction of bow acceleration is effected at low speeds by adding the fin, a significant reduction results in speeds above 15 knots full scale.

Consider the deceleration, or sudden change in bow acceleration resulting from slamming. As seen in Fig. 21, an upward impact acceleration

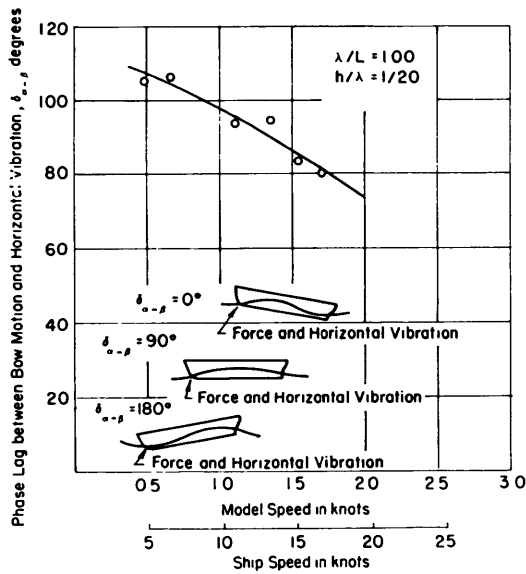


Fig. 23 Phase lag between bow motion and induced vibration versus ship speed.  $\lambda/L = 1.00$ ,  $h/\lambda = 1/20$

appears within a certain speed range for the model without fin. On the other hand, a small downward impact acceleration appears at low speeds for the model equipped with Fin A for which the induced hull vibration occurs. This is attributable to the impact force on the fin caused by collapse of the cavity, since emergence of the forefoot is not expected at these low speeds. A small upward impact acceleration also appears within the speed range in which slight emergence of the fin is evidenced, Fig. 13.

*Effect of Wave Height.* Fig. 22 shows the effect of wave height on the pitching and heaving motions, vertical acceleration at the bow, impact pressure on the ship bottom, and the intensity of horizontal vibrations measured at the bow. Tests were made in waves of  $\lambda/L = 1.0$  for various wave heights ranging from  $1/50$  to  $1/20$  of the wave length. The speed selected for the tests was 1.27 knots (12.5 knots full scale), a condition for which the vibrations had been found to be severe.

It can be seen in the figure that pitching, heaving and vertical bow acceleration increase almost linearly with increase in wave height. However, the intensity of horizontal vibration and the magnitude of impact pressure on the ship bottom do not exhibit the same trend. Instead, there exists a minimum wave height which causes vibration. This limiting value is  $1/3$  of the wave length in this case. This result is similar to that observed in ship slamming [22].

*Angle of Attack of Fin at Time of Occurrence*

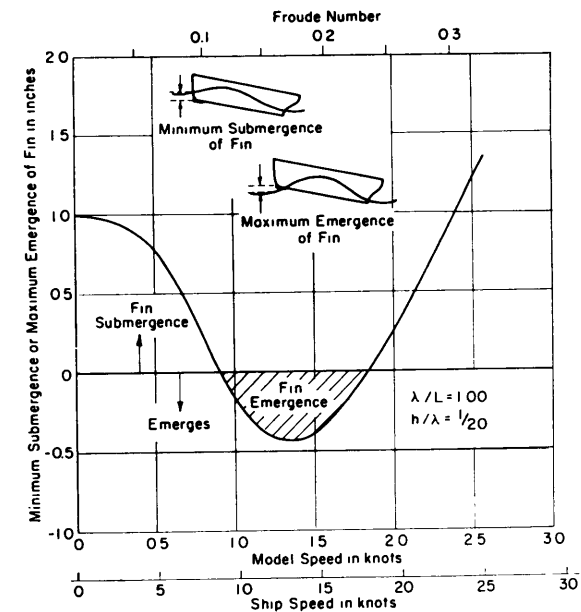


Fig. 24 Minimum submergence or maximum emergence of fin versus ship speed.  $\lambda/L = 1.00$ ,  $h/\lambda = 1/20$

*of Vibration.* The angle of attack of the fin at the initial occurrence of the vibration was obtained for various speeds by measuring the phase between bow motion and initial peak of the horizontal acceleration record. The phase angles obtained for Fin A are shown in Fig. 23. An explanatory sketch is included in the figure to assist in interpretation of the phases; that is, (a) a phase lag between bow motion and initial peak of acceleration of 0 deg means that the induced vibration occurs at the instant the bow reaches its maximum upward position, (b) a phase lag of 90 deg means that the induced vibration occurs when the attitude of the ship is horizontal, and (c) a phase lag of 180 deg means that the vibration occurs when the ship bow reaches its maximum downward position.

Now, the severest vibrations are observed at a speed 1.10 knots (10.8 knots full scale) for this fin, Fig. 18, and the phase lag for this speed, Fig. 23, is 95 deg. Therefore, the vibration occurs when the ship is almost horizontal. This result agrees well with the result observed in full-scale trials.

*Relation Between Submergence of Fin and Induced Vibration.* It has been suggested that less severe vibrations can be expected if the submergence of the fin is increased [8, 10]. However, no detailed discussion of this possibility has been made as yet.

Since the vibration is caused by collapse of a cavity which forms above the fin, the intensity

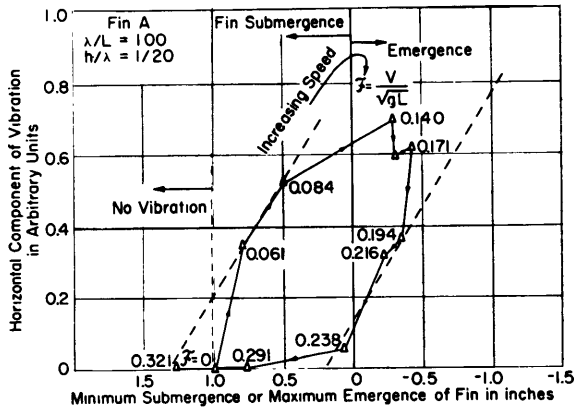


Fig. 25 Relation between minimum submergence or maximum emergence of fin and intensity of induced vibrations as parameter of ship speed

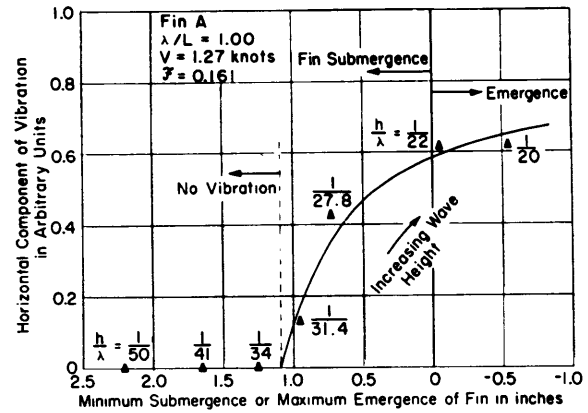


Fig. 26 Minimum submergence or maximum emergence of fin versus wave height.  $\lambda/L = 1.00$ , ship speed 1.27 knots (12.4 knots full scale)

of the vibration should be a function of submergence of the fin. If the upper limit of the fin motion is sufficiently deep, a cavity cannot form, and the vibration will not occur. That is, a certain limiting depth of the fin is required for formation of the cavity when the fin reaches its maximum upward position.

To determine this limiting depth, a simple study was made concerning the fin motion. The distance between the fin and wave surface was evaluated, permitting an estimation of the minimum submergence or maximum emergence of the fin. The case of maximum emergence is considered also, since if the fin emerges, severe vibration most certainly can be expected. The details of the analysis are given in Appendix 3. The results of the calculation for Fin A, for various speeds in waves of  $\lambda/L = 1.0$  and  $h/\lambda = 1/20$  are shown in Fig. 24. Emergence of the fin occurs within a speed range of 0.9 and 1.8 knots (8.8 and 17.7 knots full scale). Experimentally, small impact pressures were measured on the ship bottom within a speed range of 1.0 and 1.7 knots (9.8 and 16.7 knots full scale), as seen in Fig. 13. The agreement indicates that the method for evaluating minimum submergence of the fin is adequate although the theory involves some assumptions.

Now, from Figs. 18 and 24, the relation between the minimum submergence or maximum emergence of the fin and the intensity of the induced vibration can be obtained for various speeds, Fig. 25. An interesting and important conclusion can be derived from this figure. Irrespective of ship speed, no vibration appears when the submergence of the fin is greater than 1.0 in. (8 ft full scale). In other words, if the fin is so located that a minimum depth of 1.0 in. below

the wave surface is always maintained, the vibration does not occur for any model speed.

The minimum submergence or maximum emergence of the fin for various wave heights has also been estimated by the method in Appendix 3, with wave length equal to model length, and model speed of 1.27 knots (12.4 knots full scale). The relation between the minimum submergence or maximum emergence of the fin and the intensity of the induced vibration was then obtained for various wave heights. This result is shown in Fig. 26. The figure indicates that the vibration does not occur for low waves, but appears in waves whose heights are greater than  $1/33$  of their length. It also can be seen that the minimum submergence of the submerged fin for which the vibration does not occur is 1.1 in. (8.8 ft full scale). This value agrees very well with that obtained for various ship speeds, Fig. 25.

It is evident from the foregoing analysis that if the fin is always at a depth greater than 1.0 in. (8 ft full scale) beneath the moving wave surface, the cavity will not form above the fin and thus the vibration will not occur irrespective of speed and wave height. Of course this value pertains to regular waves only and a more generalized value should be determined for design purposes through tests in irregular waves. However, it will be sufficient here to note that in general a certain lower limit on the submergence of the fin is sufficient for the prevention of the induced vibration.

*Effect of Time Differential of Loading.* It was mentioned earlier in connection with the sample record shown in Fig. 8 and also in the discussion of Fig. 11 that a time difference existed between the forces on the starboard and port sides. In this section, the effect of this time



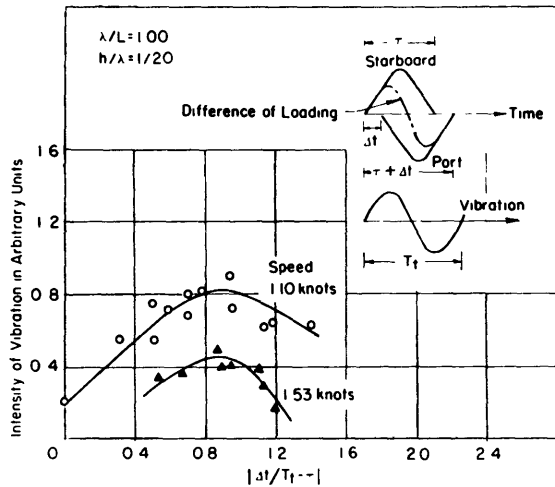


Fig. 27 Effect of time differential of loading upon intensity of induced vibration

differential in loading on the intensity of the induced vibration will be discussed in detail. Since the time differential of loadings on the fin and on the bow sides are equal, Fig. 11, the discussion will be made only on the time differential of loading on the fin.

The time difference between the forces on the port and starboard sides of the fin exists in most cases even in regular waves. In regular waves it usually ranged from  $3/10^3$  to  $7/10^3$  sec on the Mariner model. Out of a total of 700 cycles of encounter in which the vibration occurred, only a few cases were recorded in which the force was applied to both sides of the fin at exactly the same instant. The intensity of the induced vibration for these cases was always much less than that for the cases in which the force was not applied simultaneously.

In order to illustrate the effect of the time differential of fin loading on the intensity of the induced vibration, Fig. 27 was prepared. The sketch in Fig. 27 will help to explain the augmentation phenomenon. Let the duration of loading on each side of the fin be  $\tau$  and the time difference between the collapse of the cavity on the port and starboard sides be  $\Delta t$ . Then, the difference of loading on the two sides of the fin shows approximately a sinusoidal curve, and its period becomes approximately  $\tau + \Delta t$ . This assumption can be verified with the sample shown in Fig. 11. If the period,  $\tau + \Delta t$ , were to approximate the natural period of the 1st mode of torsional vibration,  $T_t$ , then the force on the port (or starboard) fin would augment appreciably the vibration initially generated by the force on the starboard (or port) fin. The sample record shown in Fig. 8 clearly illus-

Table 3 Natural Frequency and Period of Vibration Measured on SS Gopher Mariner [11]

	1st Mode	2nd Mode	3rd Mode
For maximum draft:			
Natural frequency, cpm:			
Torsional.....	—	—	—
Horizontal flexural..	120	235	395
Natural period, sec:			
Torsional.....	—	—	—
Horizontal flexural..	0.500	0.256	0.152
For light draft:			
Natural frequency, cpm:			
Torsional.....	275	400 (about)	—
Horizontal flexural..	135	275	435
Natural period, sec:			
Torsional.....	0.218	0.150	—
Horizontal flexural..	0.446	0.218	0.138

trates this case. On the other hand, if  $\Delta t = 0$ , there would be little or no vibration, as the two opposing loadings would cancel.

In Fig. 27 the intensity of the induced vibration is plotted against the ratio of  $|\Delta t/(T_t - \tau)|$ . The maximum peak-to-peak value of the vibration during one cycle of encounter indicates the intensity of vibration. As expected, the intensity of the induced vibration is nearly a maximum when the ratio is equal to 1. Also, the vibration is significantly reduced when the forces are applied to both sides simultaneously, although some small vibration is still observed.

Now consider this augmentation phenomenon on the full-scale ship. The natural frequencies and corresponding periods of the torsional and horizontal flexural vibration of the Mariner for various modes of vibration are shown in Table 3. The natural period of the 1st mode of torsional vibration is about 0.22 sec. The time difference in loading of the port and starboard sides of the fin,  $\Delta t$ , ranged mostly from  $3/10^3$  to  $7/10^3$  sec at the severest speed for the vibration for the 5.5-ft model in regular waves. These values scale to 0.03 to 0.07 sec for the full-scale ship. Also, the duration of loading,  $\tau$ , was 0.017 sec for the model, or 0.17 sec for the Mariner. Therefore, the ratio,  $|\Delta t/(T_t - \tau)|$ , ranged from 0.6 to 1.4, and it may be concluded that the augmentation phenomenon can also be expected on the Mariner.

It should be noted that although augmentation may occur for any mode of torsional and horizontal vibration, the first mode of torsional vibration is most critical, since it usually appears at the initial stage of the induced vibration.

In summary, the effect of time differential in loading on the intensity of the induced vibration appears to be very important and should be

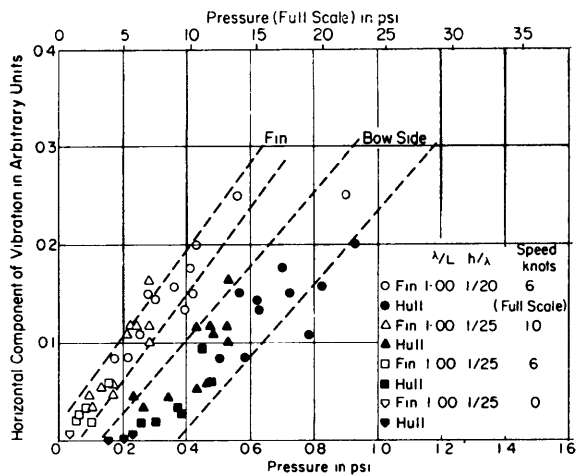


Fig. 28 Relation between intensity of induced vibration and magnitude of impact pressure on bow side and on fin (22-ft Mariner model)

considered in the design of a fin. For instance, if some device could be contrived on the fin or on the bow side by which the forces could be equalized or adjusted so as to act simultaneously on both sides, the induced vibration would be remarkably reduced.

*Relation between Magnitude of Impact Pressure and Induced Vibration.* In Fig. 28, the intensity of the induced vibration is plotted as a function of the impact pressure applied to the fin and to the bow side, as obtained from the tests on the 22-ft model. In the preparation of this figure, the cases where the collapse of cavity occurred on the starboard side prior to that on the port side are considered, since the pressures on the bow were measured on the starboard side only. The magnitude of the first pressure peak during impact is taken as a measure of the impact pressure, and the magnitude of the first peak of the vibration is taken as a measure of the intensity of the vibration. The intensity of the vibration is the result of impact pressures on both the bow side and on the fin. For example, a vibration intensity of 0.25 in the figure is caused by a pressure on the bow side of 0.90 psi and by a pressure on the fin of 0.56 psi.

The figure indicates that the intensity of the induced vibration is approximately linearly proportional to the impact pressures acting on the bow side and on the fin. However, a small impact pressure on the bow side caused by collapse of a small cavity or disturbed water flow does not produce the vibration. Thus, the vibration does not always result from the presence of the cavity above the fin, but it occurs when the generated cavity is large enough so that its col-

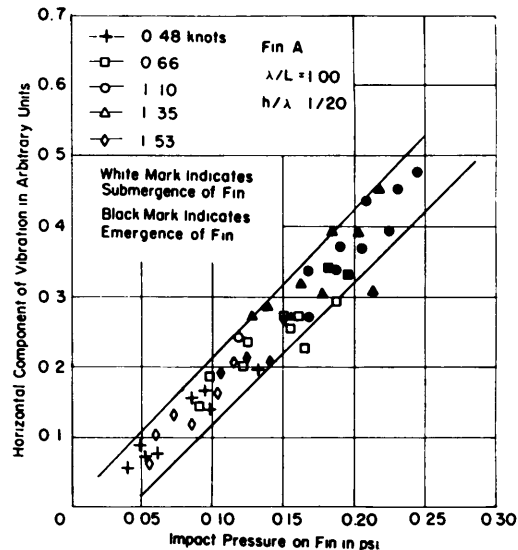


Fig. 29 Relation between intensity of induced vibration and magnitude of impact pressure on fin, (5.5-ft. Mariner model,  $\lambda/L = 1.00$ ,  $h/\lambda = 1/20$ )

lapse produces an impact pressure both on the bow side and fin. Therefore, the magnitude of impact pressure on the fin is a convenient measure of the vibration.

The relative magnitude of impact pressures on the bow side to that on the fin is of importance. As seen in Fig. 28, the impact pressure on the bow side is relatively larger than that on the fin. Particularly, the ratio of the former to the latter is high for vibrations of minor intensity. However, this ratio appears to decrease for vibrations of severe intensity. Since tests on the 22-ft model were not conducted at the severest condition for the vibration (e.g., a speed of 11 knots, full scale; in waves of  $\lambda/L = 1$ ,  $h/\lambda = 1/20$ ), a more detailed study of the relation between the magnitude of impact pressure on the fin and the intensity of the vibration was made from the test results on the 5.5-ft model. These results are shown in Fig. 29.

In the preparation of Fig. 29 for cases where the fin does not emerge from the wave surface, the magnitude of the first peak of the induced vibration is taken as a measure of the intensity of the vibration. The magnitude of the first pressure peak during impact on either the port or starboard fin is taken as a measure of the impact pressure. When the fin emerges, the magnitude of the peak-to-peak variation of the first cycle in the vibration record is taken as a measure of the intensity of the vibration. The sum of the upward and downward pressures is taken as a measure of the impact pressure.

Fig. 29 indicates that the intensity of the in-

**Table 4 Principal Characteristics of Fin A, Fin B, Fin C, and Fin D  
(Fin Location Test)**

	Fin A	Fin B	Fin C	Fin D
Location (aft of FP).....	3.8%L	12.5%L	17.5%L	22.5%L
Span, in.....	5.40	5.70	5.92	6.44
Chord, in.....	2.50	2.50	2.50	2.50
Aspect ratio.....	2.16	2.20	2.36	2.58
Breadth of flat bottom, in.....	0.20	0.50	0.72	1.24
Effective span, in.....	5.20	5.20	5.20	5.20
Effective aspect ratio.....	2.08	2.08	2.08	2.08
Effective area, sq in.....	13.00	13.00	13.00	13.00
Effective area/water-plane area.....	2.8%	2.8%	2.8%	2.8%

duced vibration is also approximately linearly proportional to the impact pressure acting on the fin at severe conditions for the vibration. This result suggests that the larger the size of the fin, the more severe the vibration would become. Generally speaking, if the size of the fin is increased, the pitching motion will be reduced and thereby less probability of occurrence of vibrations may be expected. However, if once a cavity and/or slamming occurs for the larger fin, the intensity of the induced vibration should be in proportion to the increase in size of the fin. Thus, the increase of the size of fin is favorable for reduction of pitch, but can be unfavorable for vibration reduction. Therefore, there may exist an optimum fin size when both pitching and vibration are considered; this problem will be discussed in detail in Part 2.

## 2 Effect of Fin Configuration, Size, and Location on Ship Motions and Vibrations

### Description of Experiment

Tests were first conducted for four different longitudinal locations of the fin; namely, at 3.8, 12.5, 17.5, and 22.5 per cent of the model length aft of the forward perpendicular. Shifting the location of the fin aft from the bow not only permits evaluation of the effect of location of the fin on pitch reduction, but also clarifies the properties of the induced vibration. The basic configuration of the fin used in these tests was the same as that used in Part 1. It consisted essentially of a simple aluminum plate of rectangular planform and cross section. The effective area of the fin was held constant for all four locations (2.8 per cent of the water-plane area). However, since the breadth of the flat bottom increases with the distance from the forward perpendicular, the span of the fin was increased by the same amount. Then, the effective span (difference between the original span and breadth of flat bottom) remained constant. Fin particulars are shown in Table 4.

The effect of fin size was studied, using the

**Table 5 Principal Characteristics of Fin N, Fin A, and Fin L (Fin Size Test)**

	Fin N	Fin A	Fin L
Span, in.....	4.63	5.40	6.40
Chord, in.....	2.13	2.50	2.98
Aspect ratio.....	2.18	2.16	2.15
Breadth of flat bottom, in.....	0.20	0.20	0.20
Effective span, in.....	4.43	5.20	6.20
Effective aspect ratio.....	2.08	2.08	2.08
Effective area, sq in.....	9.43	13.00	18.48
Effective area/water-plane area (per cent)...	2.02	2.80	3.97

basic fin of rectangular planform but varying the effective area. As seen in Table 5, the effective area of the fin was varied from 2 to 4 per cent of the water-plane area; however, the effective aspect ratio of the fin was kept constant at 2.08 for all three fins.

The fixed fin located at the stern was the same as that designated as Fin A in the bow-fin tests (rectangular shape with an area of 2.8 per cent of the water-plane area).

Three fins, differing in form, were used to determine the most favorable fin configuration for minimum vibration and acceptable pitch reduction, Fig. 30. In selecting these configurations, the goal was a fin whose shape was as simple as possible from practical and economic considerations.

It may be well to give a brief review of the reasoning which directed this selection. As mentioned in Part 1, the impact forces applied to the bow side and the fin by the collapse of the cavity and/or ship slamming are the primary sources of the induced vibration. Therefore, it is of greatest importance to minimize these impacts. In other words, if it is possible to prevent the occurrence of the cavity above the fin surface entirely or partially, the induced vibrations would be reduced greatly.

With these considerations, the simplest fin configurations which satisfy our purposes appear to be of two types. One is the rectangular planform type with sufficient open area in the

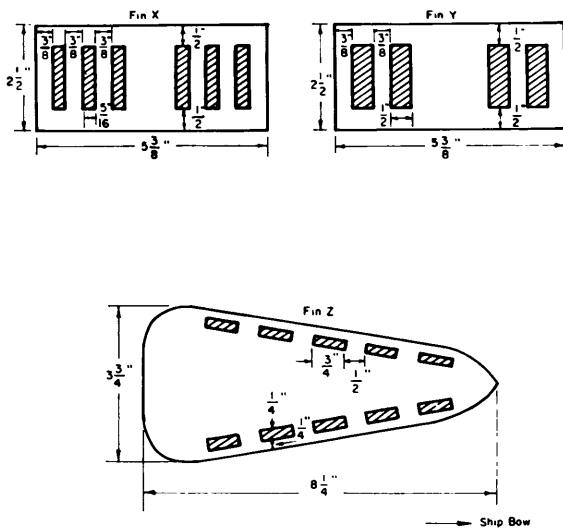


Fig. 30 Planforms of Fin X, Fin Y, and Fin Z

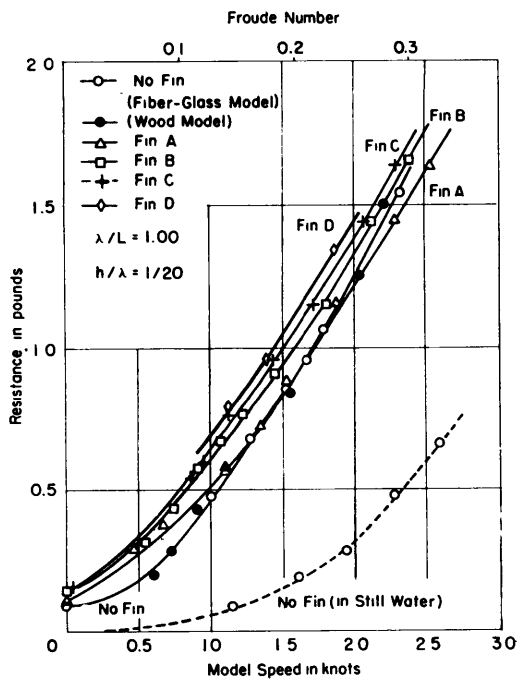


Fig. 31 Resistance of model for various fin locations

fin to prevent cavity formation. Fin X and Fin Y shown in Fig. 30 represent this type. The other type, Fin Z, has a variable span (fin width) increasing from fore to aft. Selection of this shape was based on the test results of the rectangular planform fins. These results showed that a fin located near the forward perpendicular is very effective for pitch reduction but is unfavorable for the induced vibrations. However,

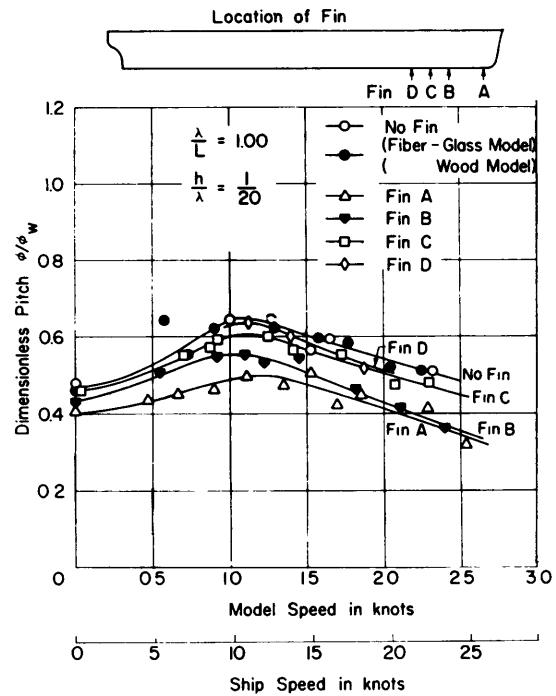


Fig. 32 Pitch motion versus ship speed for various fin locations ( $\lambda/L = 1.00$ ,  $h/\lambda = 1/20$ )

a fin located 15 to 20 per cent of ship length aft of FP is not so effective in pitch reduction but is beneficial from the vibration point of view, Table 6. Fin Z was designed to present a small area near the forward perpendicular and a relatively large area near 15 per cent  $L$  aft of FP, in an attempt to effect beneficial results for both pitch and vibration reduction.

The test procedure used in the tests mentioned here was the same as that used in the tests discussed in Part 1.

## Analyses of Experimental Results

### Effect of Location of Bow Fin

To determine the effect of the fin location on pitch reduction and vibration, tests were carried out for four longitudinal locations of the fin. These were 3.8, 12.5, 17.5, and 22.5 per cent  $L$  aft of the forward perpendicular, and the fins are designated Fin A, B, C, and D, respectively. Tests were conducted in regular waves with  $\lambda/L = 1.0$  and  $h/\lambda = 1/20$ . The primary purpose of the tests was to find the most desirable location of the bow fin from joint consideration of induced vibration and pitch reduction. However, resistance and slamming were also studied, since these factors should not be neglected in the selection of the most beneficial fin location.

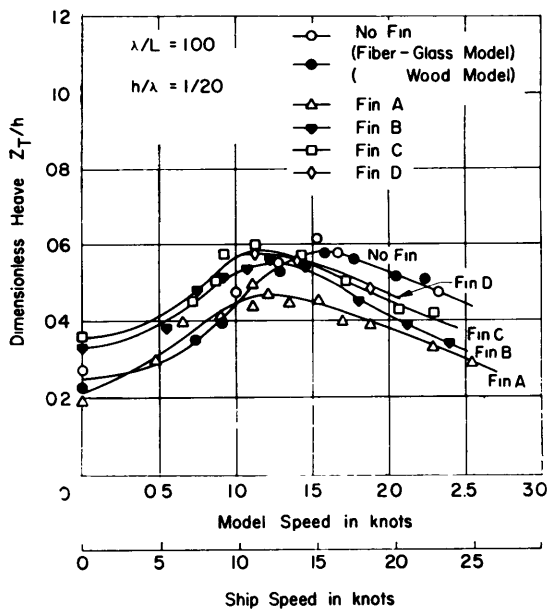


Fig. 33 Heaving motion versus ship speed for various locations of fin ( $\lambda/L = 1.00$ ,  $b/\lambda = 1/20$ )

**Resistance.** The model resistance for the various fin locations is shown in Fig. 31. There is no serious difference between the bare-hull model resistance in waves and that of model equipped with Fin A, except at low speeds. However, results for Fins B, C, and D show a tendency for the resistance to increase with distance of the fin from the forward perpendicular. This tendency is particularly evident for the speed range between 10 and 20 knots full scale.

The percentage increase in the resistance due to fin location can be estimated from Fig. 31 for various speeds. When the fin is located at 5 per cent  $L$  aft of the forward perpendicular, the maximum increase in resistance is about 12 per cent, while a significantly greater increase can be expected at all speeds for fins located aft of the 10 per cent point. It is, therefore, desirable that the fin location should not be more than 10 per cent aft of the forward perpendicular from this point of view.

**Ship Motions.** The reduction of ship motion for different longitudinal locations of the fin is of consideration since the purpose of the fin is to reduce severe pitching motion and thereby minimize the loss of speed in rough seas.

Of course, maximum reduction of pitching motion can be expected when the fin is located near the forward perpendicular. However, if the severity of the vibration induced by the fin can be lessened by locating the fin further aft, this might be worth considering, even if it results

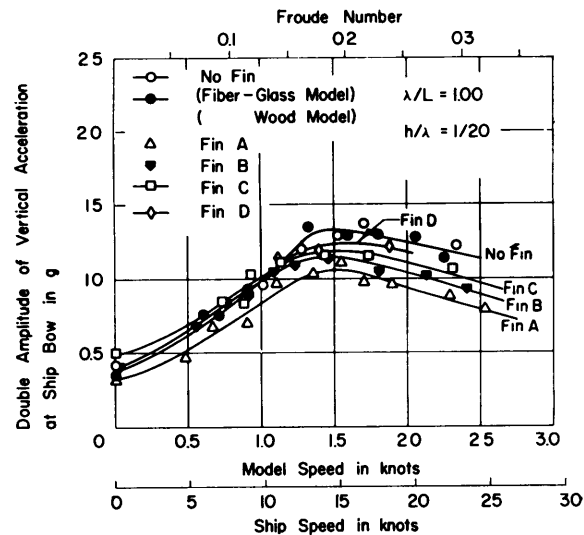


Fig. 34 Double amplitude of vertical acceleration at ship bow versus ship speed ( $\lambda/L = 1.00$ ,  $b/\lambda = 1/20$ )

in some sacrifice in pitch reduction. Therefore, the test results discussed here should be evaluated along with the results on the intensity of the induced vibration, which will be discussed in a later section.

Fig. 32 shows the pitching motion for various locations of the fin. Fin A is very effective in reducing pitch for all speeds, while Fin D is almost ineffective. It is apparent that a fin located between the forward perpendicular and  $0.1L$  aft of the perpendicular is particularly effective at speeds over 15 knots. However, at speeds below 10 knots the effectiveness of the fin decreases rapidly with increasing distance from the forward perpendicular. Also, it is evident that the effectiveness of fins located aft of 15 per cent point is small for all speeds.

Heaving motion versus speed for the various locations of the fin is shown in Fig. 33. Note that the fin, in general, is not beneficial for heaving motion at speeds below 15 knots. Only if the fin is located near the forward perpendicular is it beneficial for heave for all speeds. Although the primary purpose of the fin is not to reduce the heaving motion, this characteristic is worthy of consideration in the design of a fin.

Vertical accelerations at the bow (which are algebraic combinations of the accelerations due to pitching and heaving and their phase relation) are shown in Fig. 34. Fin A is effective in reducing vertical accelerations at the bow for all speeds, but, fins located further than  $0.10L$  aft of the forward perpendicular are not effective at speeds less than 10 knots.

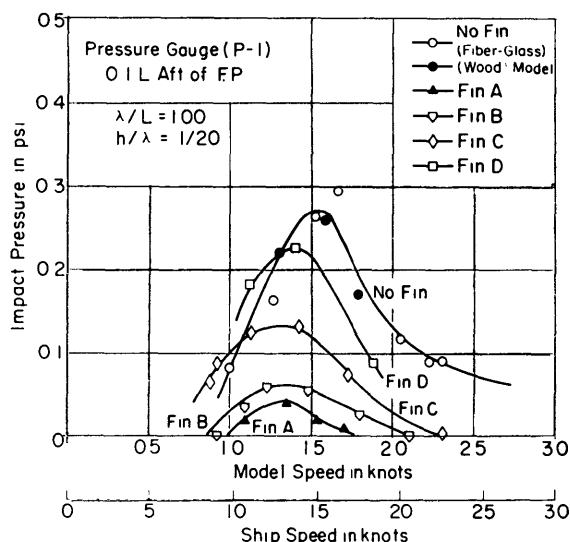


Fig. 35 Magnitude of impact pressure at 0.10  $L$  aft of forward perpendicular due to ship slamming for various locations of fin

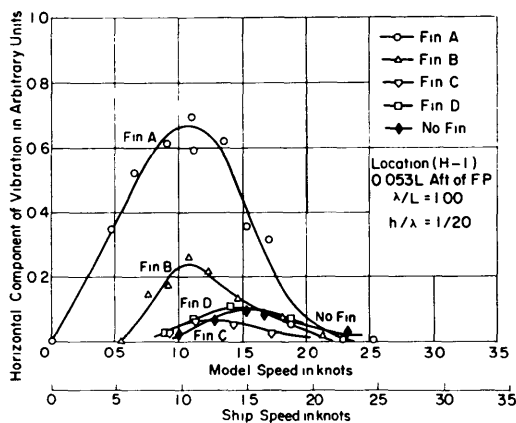


Fig. 36 Intensity of horizontal component of vibration versus ship speed for various locations of fin

In summary, it appears desirable that the location of a fin should not be further than 5 per cent of the ship length aft of the forward perpendicular from the point of view of motion reduction.

*Impact Pressures on Ship Bottom.* It is of interest to discuss the effect of fin location on the impact pressure on the ship bottom due to slamming in waves. Inasmuch as the tests were not conducted at light-draft condition, a complete discussion cannot be given. However, the general tendency can be established, since some slamming was observed even at deep-draft condition.

Fig. 35 shows the impact pressure due to slamming measured at 0.10 $L$  aft of the forward

Table 6 Optimum Fin Location for Various Parameters at a 15-Knot Ship Speed

	Location of fin aft of FP				
	FP	0.05 $L$	0.10 $L$	0.15 $L$	0.20 $L$
Resistance	Increase (%) 0	1.9	9.2	16.0	
Pitching	Reduction (%) 22.8	21.9	18.5	12.4	
Heave	31.5	20.7	12.9	8.0	
Bow Vertical Acceleration	24.5	19.0	15.5	11.8	
Slamming	85.0	85.0	81.8	67.7	
Induced Vibration	Intensity 1	0.66	0.35	0.10	0.02

perpendicular for various locations of the fin. It is apparent that installation of a fin would be helpful in reducing the severity of ship slamming, and that when the fin is located near the forward perpendicular it is particularly effective. This result could be expected, since the accompanying reduction in pitch reduces the probability of bow emergence. It appears from the figure that location of the fin not farther than 15 per cent of the length aft of the forward perpendicular is desirable from this point of view.

*Intensity of Induced Vibrations.* Fig. 36 shows the intensity of the horizontal component of vibration for various locations of the fin, plotted against ship speeds. These vibrations were measured at a location of 5.3 per cent of the length aft of the forward perpendicular. It appears that the location of the fin should be at least as far aft as the 10 per cent point for prevention of the vibration.

The effect of fin location on the intensity of the induced vibration is of course a function of ship speed. It is apparent that the effect of the location of the fin is considerable for relatively low speeds such as 5 and 10 knots, but is not so pronounced for higher speeds such as 20 knots, where the vibration is already small.

It is of interest and importance to select the most favorable location of the fin from various considerations such as ship motion, resistance, slamming, and the induced vibration. An example of such an approach is given in Table 6 for a ship speed of 15 knots. The most favorable location of the fin will be a function of fin configuration and ship speed; however, the example shown in Table 6 should serve as a guide for estimating the most favorable location for a rectangular fin. The double lines in the table show the favorable range for the fin location for each

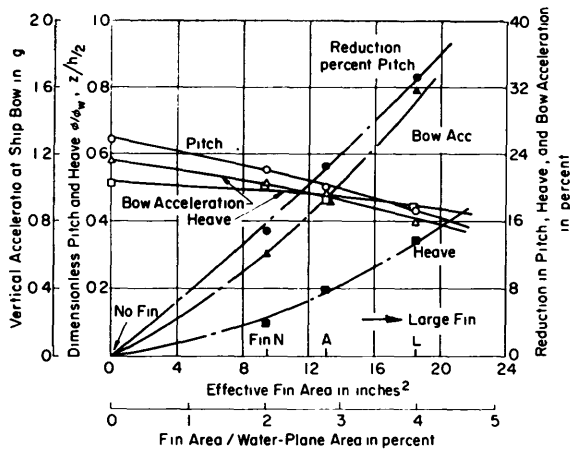


Fig. 37 Pitch, heave, and bow acceleration versus fin area.  $\lambda/L = 1.00$ ,  $b/\lambda = 1/20$ , speed = 1.10 knots (10.8 knots full scale)

of the parameters considered. Also shown is the percentage increase in resistance and the percentage reduction in pitch, heave, bow acceleration, and slamming effected by the fin. For example, the fin located at  $0.10L$  will increase resistance by 9.2 per cent, and will reduce pitching motion by 18.5 per cent, heaving by 12.9 per cent, bow acceleration by 15.5 per cent, and will reduce the intensity of slamming by 81.8 per cent. The intensity of induced vibration is given as the ratio of the intensity with the fin at a specific location to that for a fin located at the forward perpendicular.

A fin located near the forward perpendicular is most beneficial for resistance, ship motion, and slamming, but a location further aft is more beneficial for reducing vibration. Therefore, a compromise between the two is required in estimating the most favorable location of the fin. It may be concluded from the table that a fin located at  $0.1L$  aft of the forward perpendicular appears to be the most beneficial for a ship of 15 knots, if a 10 per cent increase in resistance is permissible. Horizontal vibration will still be present in this case, but its intensity will be decreased to 35 per cent of that for a fin located at the forward perpendicular.

Additional fin configurations will be studied in an attempt to effect further reduction of ship motion and lessen the induced vibration.

#### Effect of Size of Bow Fin

The effect of fin size on ship motion and vibrations was studied for three different sizes of the fin. These fins had the same effective aspect ratio but different areas, Table 5. They were all located at 3.8 per cent of the model length

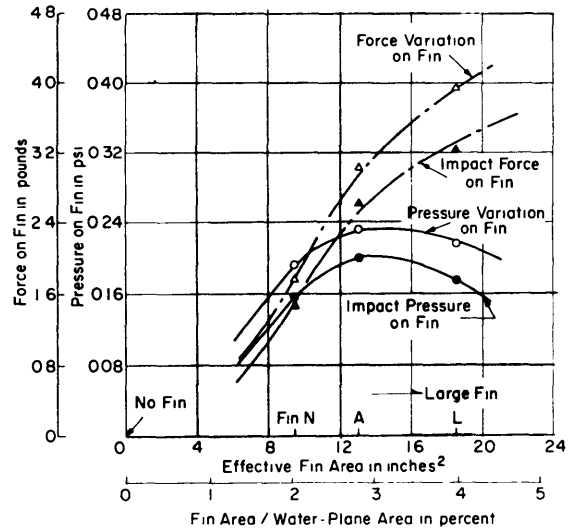


Fig. 38 Pressure and force acting on fin versus fin area

aft of the forward perpendicular. They are designated as Fin N, A, and L and their effective areas are 2.02, 2.80, and 3.97 per cent of the water-plane area, respectively.

Tests were conducted in regular waves with  $\lambda/L = 1.0$  and  $h/\lambda = 1/20$ . All tests were made at a model speed of 1.10 knots (10.8 knots full scale) because the previous test results showed that the severest pitching and vibrations were expected at this speed. Tests at other speeds were not included since the results obtained by Stefun [3] indicated that the effect of fin size on pitch reduction showed the same tendency for all speeds except very low speeds.

*Ship Motions.* Double amplitude of pitch, heave, and bow acceleration are plotted against fin area in Fig. 37. The percentage of reduction of motion is also included. Fins of larger area produce large reduction in pitch and bow acceleration, but small reduction in heave. However, all reductions appear to increase approximately linearly with increase in fin area. Stefun concluded on the basis of tests results that fins with an area greater than 2 per cent of the water-plane area did not produce reduction in direct proportion to the increase in area [3]. The discrepancy between the two sets of results is not surprising as the models used in the tests represent different types of ships, and fences were not fitted to the fins in the present tests. In general, however, it may be concluded that the effectiveness of pitch reduction is approximately linearly proportional to the fin area, since fins of unusually large area will not be used in a practical case.

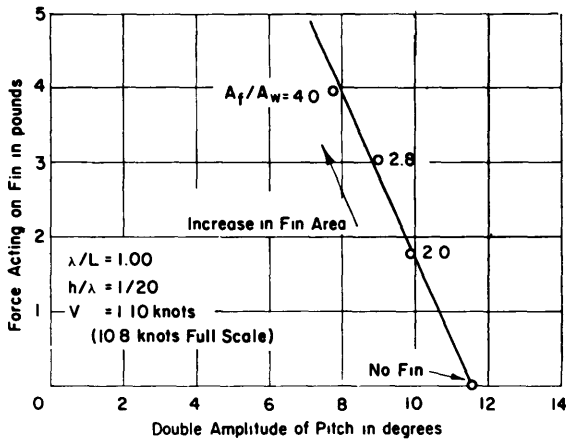


Fig. 39 Relation between pitch and force acting on fin

*Force Acting on Fin.* The pressure and force acting on the fins consist of two components; namely, the pressure (force) due to ship motion in waves, and the impact pressure (force) caused by collapse of the cavity and/or ship slamming. The latter component is directly related to the intensity of the induced vibrations.

Fig. 38 shows the relation between these pressures (forces) and fin area. Both the pressure due to ship motion and the impact pressure due to collapse of the cavity and/or slamming increase approximately linearly with increase in fin area up to an area of 3 per cent, and then the curves show a tendency to flatten. On the other hand, the forces acting on the fin (integrated pressure over the whole fin) continue to increase approximately linearly with fin area.

Pitching motion decreases with increase of fin area, since the large fin area gives large resisting moment for pitch if other conditions are equal. Fig. 39 shows the relation between pitch double amplitude and the force acting on the fin at a model speed of 1.10 knots (10.8 knots full scale). The forces due to ship motion were taken in this analysis because these are the forces, which furnish a resisting moment in pitch. As can be seen, the force acting on the fin is linearly proportional to the reduction in pitch if other condition (wave dimensions, ship speed, and so on) are equal.

*Intensity of Induced Vibration.* The intensity of induced vibration was measured at  $0.053L$  aft of the forward perpendicular for various fin sizes. These results are plotted as a function of fin area in Fig. 40. The intensity of vibration at the instant the vibration occurred was averaged over all cycles of encounter. Also, the maximum peak-to-peak variation during each cycle of

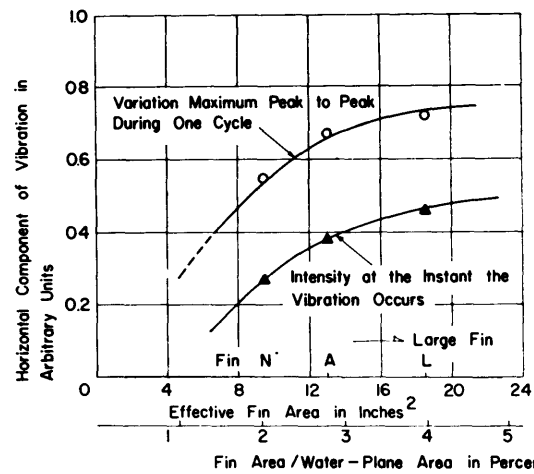


Fig. 40 Intensity of induced vibration versus fin area

encounter was averaged over all cycles. As noted in Part 1, the intensities of these vibrations varied from cycle to cycle, even under uniform speed in regular waves. However, the average of these intensities was fairly well defined. It may be said that, on the average, both the intensity of vibration at the moment the vibration occurs and the maximum peak-to-peak variation of the vibration increase with fin area.

In Fig. 41, the intensity of horizontal vibration is plotted against the impact force acting on fins of different size. The impact forces plotted in the figure refer to load on one side of the fin only. Although some scatter is apparent in the figure, the intensity of induced vibration is approximately linearly proportional to the impact force acting on the fin, irrespective of fin size. Though the data in Fig. 41 pertain to one ship speed and one wave condition only, the same trend was established in Fin A from tests at various speeds, Fig. 29, and also in irregular waves, refer to Fig. 55. Therefore it may be concluded that the intensity of induced vibrations is approximately linearly proportional to the impact force acting on the fin irrespective of ship speed, fin size, and regularity of the waves.

The relation between pitch reduction and the intensity of vibration at a model speed of 1.10 knots (10.8 knots full scale) is shown in Fig. 42. The maximum peak-to-peak value during each cycle of encounter averaged over all cycles was taken as the measure of intensity of vibration.

The figure shows that increase in fin size is favorable for reduction of pitch but is unfavorable for vibration, and the relation between pitch reduction and vibration is linear for fins of area not greater than 2.5 per cent of the water-plane



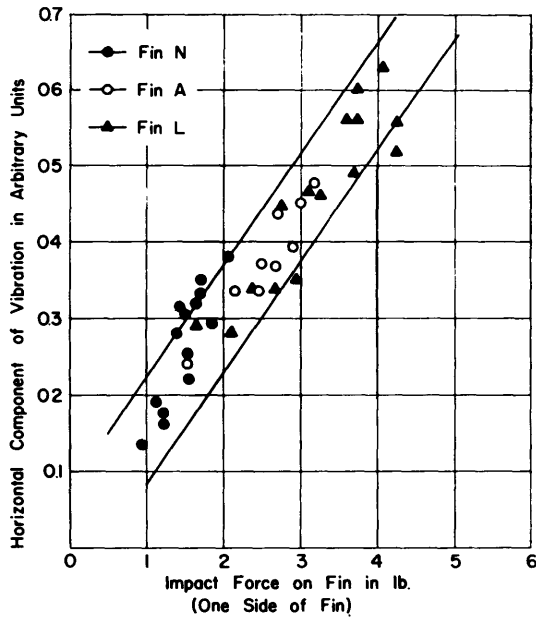


Fig. 41 Relation between intensity of vibration and magnitude of impact force acting on fin

area. Although the vibration intensity levels off for fins with areas greater than 2.5 per cent, such fins continue to produce reductions in pitch in direct proportion to fin area, Fig. 37. For example, the 2 per cent fin is twice as effective as the 1 per cent in pitch reduction and the associated vibrations are twice as severe as those for the 1 per cent fin. However, the 3 per cent fin is 3.2 times as effective in pitch reduction and the vibrations are only 2.4 times the severity of those for the 1 per cent fin. These figures increase to 4.4 and 2.6, respectively, for the 4 per cent fin. Thus, a fin of large area is relatively beneficial for both pitch and vibration reduction, although a fin of such large area would probably not be usable in practice.

#### Fixed Stern Fin

It has been said that a fixed stern fin is less favorable than a bow fin for pitch reduction [23]. However, the vibration problems associated with a stern fin can be expected to be less severe since the fin is less likely to approach the water surface. An activated stern fin, undergoing oscillatory motion so that its relative angle of attack will yield lift forces of suitable magnitude and phase, has proved much more effective in reducing pitch motion than a fixed stern fin [23]. There are certain difficulties, however, in practical application of the controlled stern fin. These include problems associated with interference

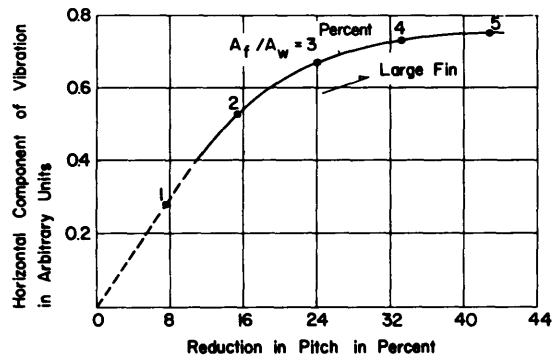


Fig. 42 Relation between pitch reduction and intensity of vibration (model speed 1.10 knots, full scale 10.8 knots)

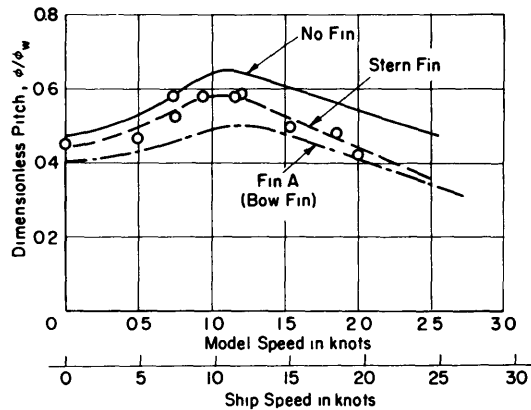


Fig. 43 Comparison of pitch motions for stern fin, bow fin, and bare hull ( $\lambda/L = 1.00$ ,  $b/\lambda = 1/20$ )

to the flow around the propeller, control of the fin movement, and prevention of fin stall for large amplitudes of oscillation. Since these difficulties have so far prevented the installation of activated stern fins on ships, the present study is limited to the fixed stern fin only. For these tests Fin A, previously used as a bow fin, was installed  $0.033L$  forward of the after perpendicular.

*Resistance and Ship Motions.* It should be mentioned at the outset that the ship resistance appears to be considerably increased by the presence of the stern fin. The increase in resistance over bare hull (without fin) was obtained in regular waves with  $\lambda/L = 1.0$  and  $h/\lambda = 1/20$ . These results are shown in Fig. 50, with those for various bow-fin configurations included for comparison. It is seen that Fin A, as a bow fin, shows 30 per cent increase in resistance at low speeds but no appreciable increase at speeds above 10 knots, full scale. On the other hand, the stern fin increases the resistance about 30 per cent at low speeds, 20

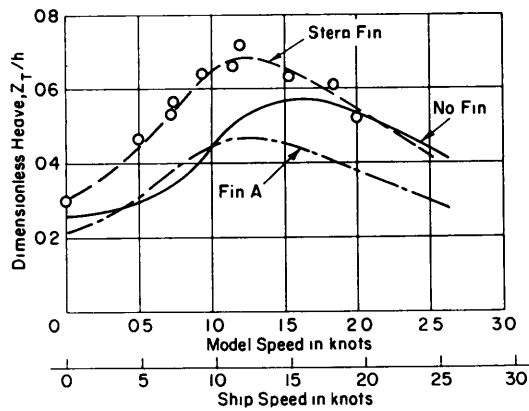


Fig. 44 Comparison of heave motion for stern fin, bow fin and bare hull ( $\lambda/L = 1.00, h/\lambda = 1/20$ )

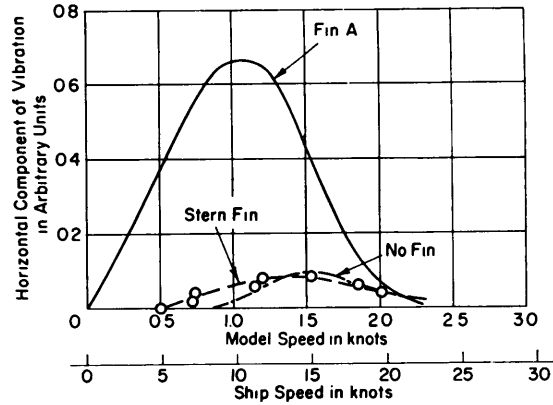


Fig. 45 Comparison of intensity of induced vibrations with stern fin, bow fin, and bare hull ( $\lambda/L = 1.00, h/\lambda = 1/20$ )

Table 7 Comparison of Motions and Intensity of Vibration for Bow and Stern Fins  
( $\lambda/L = 1.00, h/\lambda = 1/20, V = 9.6$  knots full scale)

	No fin	Bow Fin A	Bow Fin X	Stern fin	Bow Fin X and stern fin
Increase of resistance (per cent) . . . . .	—	8.2	15.2	19.0	19.5
Pitch, $\phi/\phi_w$ . . . . .	0.64	0.49	0.51	0.58	0.51
Reduction in pitch (per cent) . . . . .	—	23.4	20.3	9.4	20.3
Heave, $Z/h$ . . . . .	0.43	0.44	0.52	0.64	0.51
Reduction in heave . . . . .	—	-2.3	-20.9	-48.8	-18.6
Bow vertical acceleration ( $g$ ) . . . . .	0.98	0.82	1.00	1.05	0.98
Reduction in acceleration (per cent) . . . . .	—	16.3	-2.0	-7.1	0
Intensity of vibration (arbitrary units) . . . . .	0.02	0.66	0.10	0.06	0.06

per cent at 10 knots and 10 per cent at 18 knots, full scale.

Pitching motion in waves with the fixed stern fin is shown in Fig. 43, again with Fin A and the bare hull included for comparison. There is a reduction of only 6 per cent at low speeds, about 10 per cent at 10 knots and 18 per cent at 20 knots, full scale.

Fig. 44 compares heaving motion for the stern fin, bow fin, and bare hull. A considerable increase of heave with the stern fin can be seen at speeds up to 15 knots, amounting to about 45 per cent more than that for the bare hull. This large increase in heave, incidentally, could increase the vertical motion and acceleration at the ship bow. The experimental results show that bow acceleration with the stern fin is higher than that for bare hull at ship speeds under 12 knots, but a small reduction in acceleration is found at high speeds over 15 knots.

The foregoing discussion suggests that the fixed stern fin is not a suitable device for pitch

reduction unless proper consideration is given to its configuration as well as the angle at which it is installed. It should be mentioned, however, that if a stern fin is provided with oscillatory motion so as to generate sufficient lift forces, good results can be expected so far as pitch reduction is concerned [23].

*Intensity of Vibration and Variation of Pressure Acting on Fin.* Fig. 45 shows the intensity of induced vibrations associated with a fixed stern fin. The circles plotted are the average of the maximum peak-to-peak vibration which occurred in each cycle of encounter, measured at  $0.053L$  aft of the forward perpendicular. It is apparent that the hull vibration induced by a fixed stern fin is not serious. Although vibrations are present over a wide range of ship speeds, their magnitudes are small.

For the stern location, the intensity of vibrations at the ship bow is approximately 15 per cent more than that at the stern even though, in this case, the source of vibration is located near the

stern. For the bow location, the intensity of vibrations at the bow is approximately twice that at the stern. This indicates that severest vibration is experienced at the ship bow irrespective of the fin location.

It is also of interest to compare the magnitude of pressures acting on the bow and stern fins. As the size and configuration of the bow and stern fins were the same in these tests, a direct comparison of pressure acting on these fins can be made. These results are shown in Fig. 46. The impact pressures produced by collapse of the cavity (though no appreciable impact pressure occurred on the stern fin) were neglected and only the peak-to-peak values of the pressure on the fins produced by pitching and heaving motions were considered. The results show considerable differences in the magnitude of pressure on the two fins. The pressure acting on the bow fin is approximately three times that acting on the stern fin in these waves. This may be primarily due to the difference in the relative flow at the bow and stern fins and partially due to the difference in the variation of submergence of the fins at their locations due to the pitching motions.

*Combination of Bow and Stern Fins.* Inasmuch as satisfactory reduction in pitch cannot be expected for a fixed stern fin, it was thought that one feasible and simple method to reduce pitch might be found in a combination of bow and stern fins. To compare the resistance, ship motions, and induced vibration for combined fins with those for a single fin, a limited test was made. A particular fin configuration, called Fin X was used as the bow fin, and Fin A was used as the stern fin in this test. The characteristics of Fin X will be discussed in detail later; however, tests on this particular configuration had shown it to be an effective anti-pitching device for both the pitch and vibration reduction.

The test using the combined bow and stern fins was made in the usual regular waves with  $\lambda/L = 1.0$ ,  $h/\lambda = 1/20$ , at a speed of 0.98 knot (9.6-knots full scale).

Comparisons of resistance, pitch and heave, bow vertical acceleration, and intensity of vibration were made for the bare hull (without fin), bow Fin X, stern fin, and a combination of the Fin X and the stern fin. These are shown in Table 7. The values for the bow Fin A, which has the same configuration and size as the stern fin, are also included in the table for reference. As seen in the table, significantly greater reduction in pitch can be expected for the combination of bow and stern fins than for the single stern

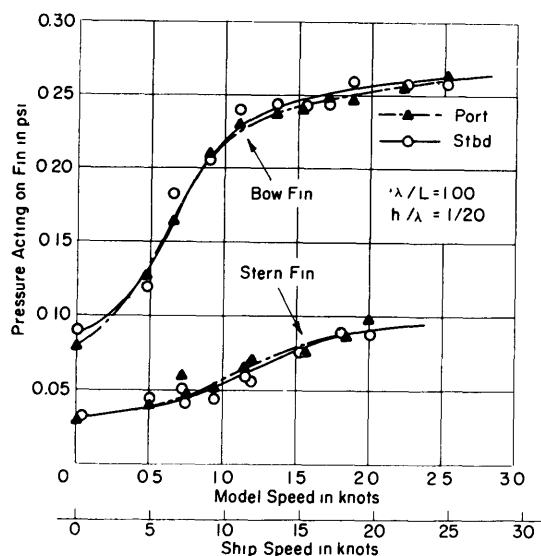


Fig. 46 Comparison of variation of pressure acting on bow and stern fins ( $\lambda/L = 1.00$ ,  $h/\lambda = 1/20$ )

fin. However, the increase in resistance is still high, and there is no appreciable improvement in ship motions over those with Fin X at the bow. Furthermore, the intensities of vibration for Fin X, stern fin, and the combined bow and stern fin are of the same order of magnitude and they are all much less than that for Fin A at the bow. It may be concluded that a combination of bow and stern fins has no significant advantage over a single bow fin of proper configuration.

#### Effect of Bow Fin Configuration

One of the most interesting phases of this study was the attempt to find the optimum fin configuration for minimum vibration and adequate reduction in pitch. A comprehensive study of the effect of bow fin configuration on pitch reduction was made by Stefun and Schwartz [10], and they suggested some configurations which satisfy, to some extent, both requirements.

However, the proposed fin configurations have associated with them certain difficulties in practical application and, on the basis of the fundamental properties found in the present study, three different fin configurations of simple form were selected and employed in the tests. The reasons for selecting these particular fins (Fins X, Y, and Z) have already been discussed.

*Ship Motions.* Fig. 47 compares the pitch motions for the various fin configurations in regular waves ( $\lambda/L = 1.0$ ,  $h/\lambda = 1/20$ ). It is seen that the pitch motions for Fins X, Y, and Z are nearly equal and that at high speeds they

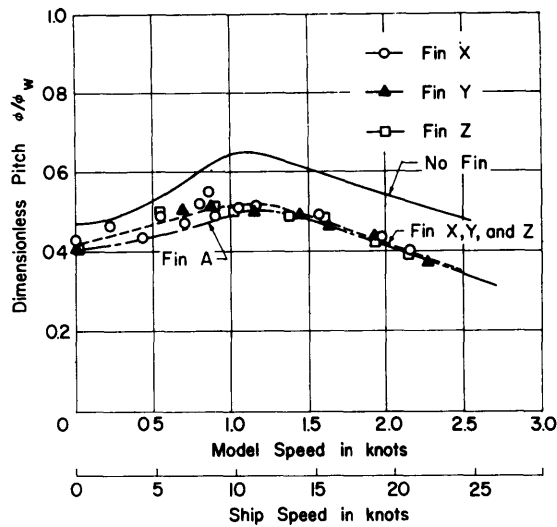


Fig. 47 Comparison of pitch motions for various fin configurations ( $\lambda/L = 1.00$ ,  $b/\lambda = 1/20$ )

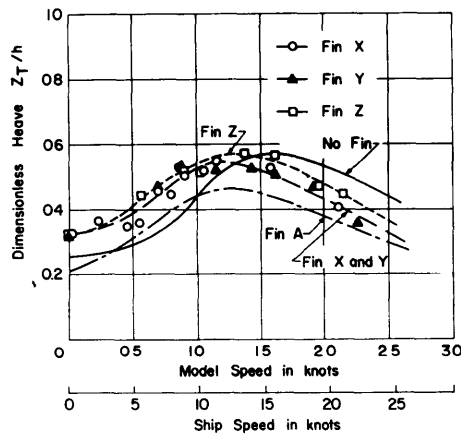


Fig. 48 Comparison of heave motions for various fin configurations ( $\lambda/L = 1.00$ ,  $b/\lambda = 1/20$ )

are comparable to those for Fin A. Although significant pitch reduction cannot be expected for these fins at speeds below 5 knots, still a reduction of over 10 per cent is realized. Fins X and Y have holes which reduce the effective area for producing resisting moment to pitch, and large pitch reduction cannot be expected at low speeds. However, the presence of these holes does not impair the fin efficiency in pitch reduction at high speeds as the water flow along the fin surface becomes strong enough to prevent strong flow through the holes. Therefore, the same order of reduction in pitch as that for the solid fin, Fin A, is obtained for these fins at high speeds.

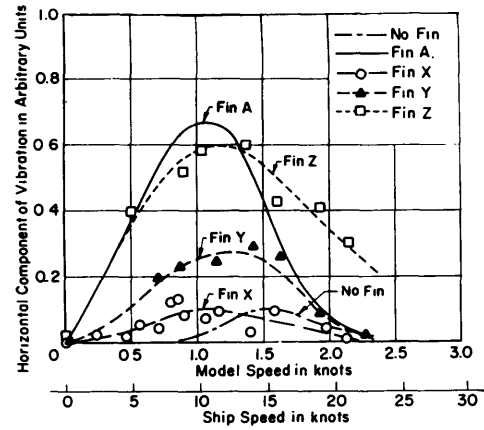


Fig. 49 Comparison of intensity of induced vibration for various fin configurations ( $\lambda/L = 1.00$ ,  $b/\lambda = 1/20$ )

The foregoing comparison of pitch reduction applies to regular waves of length equal to the model length, only, but it was later found in irregular wave tests Fin X has pitch-reduction characteristics comparable to Fin A (refer to Figs. 52 and 53).

Heaving, as a function of speed, is shown in Fig. 48. A remarkable feature is that heaving motions are generally increased at low speeds by adding the fin. Particularly, this tendency is noticeable for fins with holes. However, these undesirable large heave motions might be lessened by giving proper consideration to the section profile of the fin as well as the angle at which the fin is installed.

*Intensity of Induced Vibrations.* Fig. 49 compares the intensity of vibration at various speeds for the various fin configurations. Fins A and Z show extraordinarily severe vibrations in comparison with other fins. Considering also the magnitude of pitch and heave motions for Fin Z, this fin cannot be considered as a suitable device for pitch reduction. On the other hand, Fin X shows very good results for vibration. Although small horizontal vibrations appear within a wide range of speeds by adding this fin, the intensity of these vibrations is not serious, and hull vibrations of comparable intensity appear on the model without fins.

Although comparable pitch reduction can be expected for Fin X and Fin Y, the former will be superior to the latter as an antipitch device when the intensity of vibration is considered.

*Resistance.* Fig. 50 shows the increase in resistance produced by the various bow fin configurations in regular waves. Included also, for comparison, is the resistance increase for the stern fin. It is seen that in general the resistance

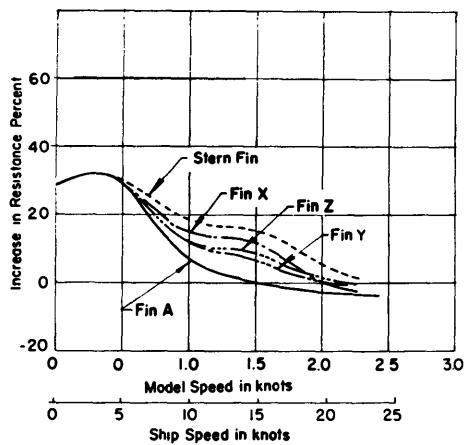


Fig. 50 Comparison of percentage of increase in resistance for various bow fin configuration and stern fin ( $\lambda/L = 1.00$ ,  $b/\lambda = 1.20$ )

is increased by adding the fins. Particularly, the increase is large for the stern fin. However, this increase in resistance might be lessened by again giving proper consideration to the fin section profile and its angle of installation.

#### Pitch Reduction and Induced Vibration in Irregular Waves

The foregoing discussion on pitch reduction and the induced vibration in regular waves is applicable for heavy swell conditions experienced at sea. While the regular waves used in the tests are not typically representative of actual seas, they approximate the smooth regular swell often encountered after a storm has passed. Then, the question arises as to the applicability of the fundamental properties obtained in regular waves to a realistic seaway. Can more frequent or more severe vibrations be expected in irregular seas than in regular waves? The answer to these questions will be obtained through statistical studies to be carried out in the new David Taylor Model Basin Seakeeping Facility. However, in order to provide some information on this subject, a limited study was conducted in irregular seas generated in the 140-ft Basin. The investigation was made using Fins A and X, at a model speed of 0.1 knot (1 knot full scale).

The frequency of the wavemaker was varied manually in random fashion over a wide range in an attempt to reproduce as nearly as possible a Neumann Sea State 7 to 8. The wave characteristics were measured by a wave probe moving with the carriage.

The spectra of the irregular waves generated for these tests actually correspond to Neumann's spectrum for a partially developed Sea State 7

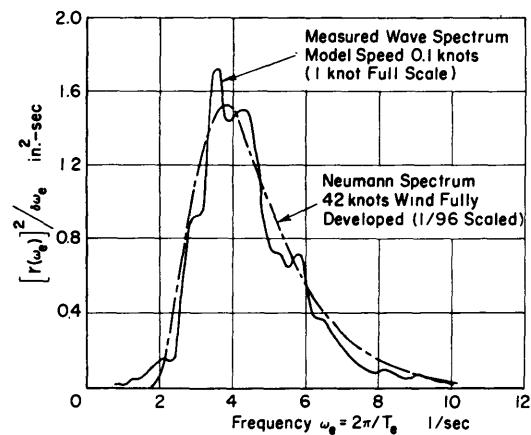


Fig. 51 Energy spectrum of irregular waves generated in 140-ft basin

for Fin A, and to a fully developed Sea State 8 for Fin X. Fig. 51 shows the spectrum for Fin X. Included also is the Neumann spectrum for a fully developed sea of 42-knot wind force. Reasonably good agreement between the measured and the Neumann's sea spectra is evidenced.

*Pitch Motion.* It is of considerable importance to compare the pitch-response amplitude operators of the Mariner when equipped with Fin A, Fin X, and bare hull in order to estimate the pitch reduction with fins in rough seas. For this purpose, tests in regular waves of various lengths were conducted on a 22-ft model at speeds of 0 and 10 knots full scale (Figs. 52 and 53). The response amplitude operators for Fins A and X, obtained from irregular wave tests at a speed of 1 knot (full scale), are also included in Fig. 52 for comparison. The black marks in the figure were taken from the tests in regular waves on the 5.5-ft model. Some discrepancy exists between the two response amplitude operators for Fin X, but good agreement can be seen for Fin A.

An interesting conclusion which can be drawn from these figures is that the pitch response for Fin X is nearly equal to that for the Fin A and that a significant reduction in pitch can be expected for both fins in comparison with the pitch for the bare hull.

Predictions of pitch motions in fully developed Sea States 5 (wind force 24 knots) and 8 (wind force 40 knots) were made for Fin A, Fin X, and bare hull at a ship speed of 0 and 10 knots, using the response amplitude operators shown in Figs 52 and 53. The results show that reduction of pitch by 20.6 per cent and 19.8 per cent can be expected on the Mariner for Fin A and Fin X, respectively, in Sea State 5, and 13.9 per cent and 12.8 per cent reduction, respectively,

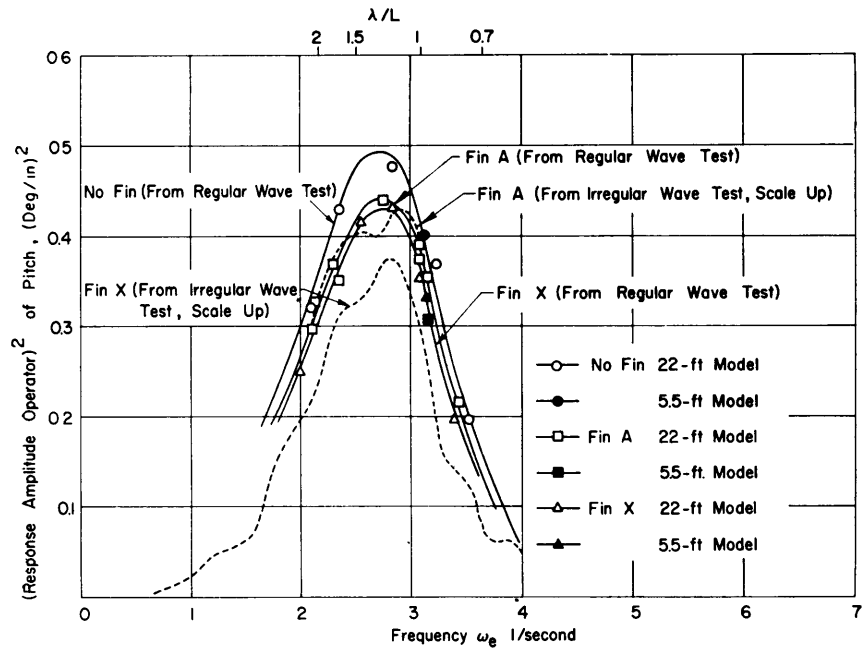


Fig. 52 Pitch response amplitude operator for Mariner model with Fin A, Fin X, and without fin (22-ft model, 0 knot speed)

Table 8 Comparison Between Fin A, Fin X and Fin Y of Probability of Occurrence of Vibration

(Ship Speed 1 Knot)

	Fin A	Fin X			Fin Y
Sea State.....	7	8	8	8	8
Wind force, knots.....	36	39	40	42	40
Development.....	Partially	Fully	Fully	Fully	Fully
No. of cycles of encounter in record, <i>N</i> .....	118	212	210	183	210
Induced vibration:					
Total no. of occurrences.....	34	25	22	21	25
Percentage of occurrence.....	28.8	11.8	10.5	11.5	11.9
Classification of intensity of vibration, $\beta$ (arbitrary units)					
0 < $\beta$ <= 0.1.....	13	19	13	13	12
0.1 < $\beta$ <= 0.2.....	6	4	5	4	9
0.2 < $\beta$ <= 0.3.....	6	2	3	3	3
0.3 < $\beta$ <= 0.4.....	5	0	1	1	1
0.4 < $\beta$ <= 0.5.....	3	0	0	0	0
0.5 < $\beta$ <= 0.6.....	1	0	0	0	0
Relative frequency of occurrence (per cent)					
No vibration.....	71.1	88.2	89.4	88.5	88.0
0 < $\beta$ <= 0.1.....	11.0	9.0	6.2	7.1	5.7
0.1 < $\beta$ <= 0.2.....	5.1	1.0	2.4	2.2	4.3
0.2 < $\beta$ <= 0.3.....	5.3	0.9	1.4	1.6	1.4
0.3 < $\beta$ <= 0.4.....	4.2	0	0.5	0.5	0.5
0.4 < $\beta$ <= 0.5.....	2.6	0	0	0	0
0.5 < $\beta$ <= 0.6.....	0.8	0	0	0	0
Measured pitch double amplitude, deg					
Significant.....	8.4	8.0	9.4	9.4	8.9
Highest in <i>N</i> -encounters.....	13.9	11.0	15.4	14.2	13.7

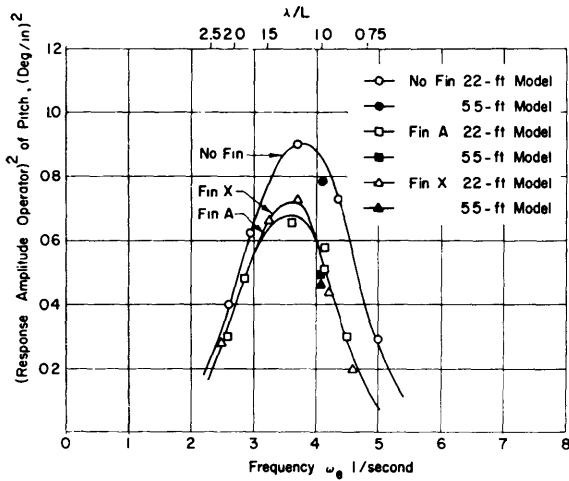


Fig. 53 Pitch response amplitude operator for Mariner model with Fin A, Fin X, and without fin (22-ft model, 10 knots speed)

in Sea State 8. Thus, the same order of magnitude in pitch can be expected for Fin X as for Fin A.

*Probability of Occurrence of Induced Vibration.* It is of interest to estimate the probability of occurrence of the induced vibration based on these irregular wave tests. The vibrations occurred 34 times in 118 cycles of encounter for the Mariner with Fin A in irregular seas corresponding to a partially developed Sea State 7 at a ship speed of 1 knot, while no vibration occurred at this speed in regular waves of height equal to  $\frac{1}{20}$  of the wave length. Thus, it is obvious that fin configurations such as Fin A cannot be used in a practical case even though it may yield satisfactory pitch reduction. Fortunately, Fin X appears to satisfy, though not perfectly, both requirements for pitch reduction and minimum vibrations.

Table 8 compares the probability of occurrence of vibrations for Fin A, Fin X, and Fin Y at a ship speed of 1 knot. Included also are the measured pitch double amplitudes for these three fins. Note that the relative frequencies of vibration for Fins X and Y are much less than that for Fin A in spite of the fact that the sea states in which Fins X and Y were tested were more severe than that for Fin A. That is, the relative frequency is only 11.2 per cent for Fin X in a fully developed Sea State 8 while it is 28.8 per cent for Fin A in the partially developed Sea State 7. Note also that not only is the relative frequency less for Fin X but also the intensity of the generated vibration is much less severe than that for Fin A. For instance, vibrations occur an average of 11

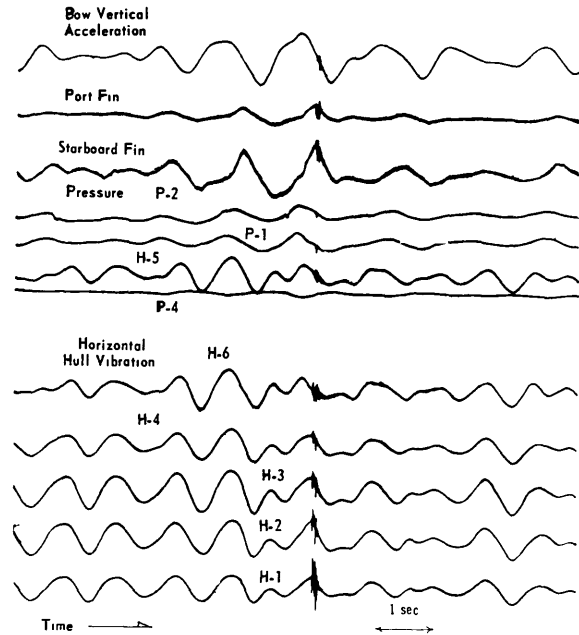


Fig. 54 Record taken in irregular wave test (5.5-ft. model, 0.1 knot speed)

times in 100 cycles of encounter for Fin X in Sea State 8; however, 7 of these 11 occurrences are vibrations of minor intensity. Inasmuch as a large size was intentionally selected for Fin X for this study, some vibration still occurred even at low speed, in irregular waves. These vibrations might be lessened by using proper size and/or by adopting a suitable sectional profile of the fin.

A remarkable feature found from the tests in irregular waves was that an impact pressure was *always* observed on the fin whenever the vibration occurred. A portion of the record taken in irregular seas is shown in Fig. 54. It is apparent that impact pressures appear on both fins and that a small impact acceleration also occurs in vertical acceleration at the bow when vibration occurs. This is a phenomenon similar to that found from tests in regular waves.

*Fundamental Properties of Induced Vibrations.* To verify that the properties of the induced vibration obtained in regular waves are also applicable in irregular seas, Fig. 55 was prepared. The figure shows the relation between the magnitude of impact pressure on the fin and the intensity of the induced vibration, as obtained in irregular waves. The envelopes of the regular wave data Fig. 29, are also shown. It is clear that the intensity of the induced vibration in irregular waves increases approximately linearly with increase of the impact pressure acting on the fin. Also, good agreement between the results in regu-

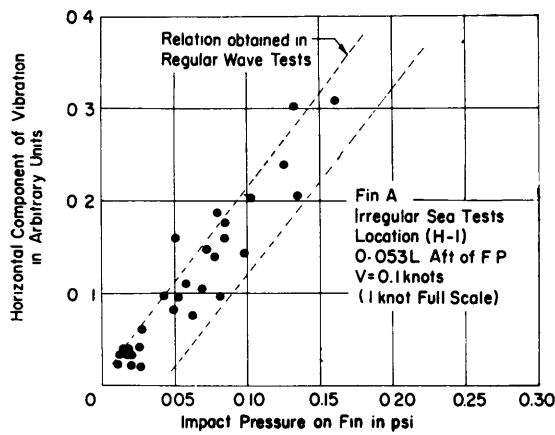


Fig. 55 Relation between magnitude of impact pressure on fin and intensity of induced vibration in irregular wave (5.5-ft. model, 0.1 knot speed)

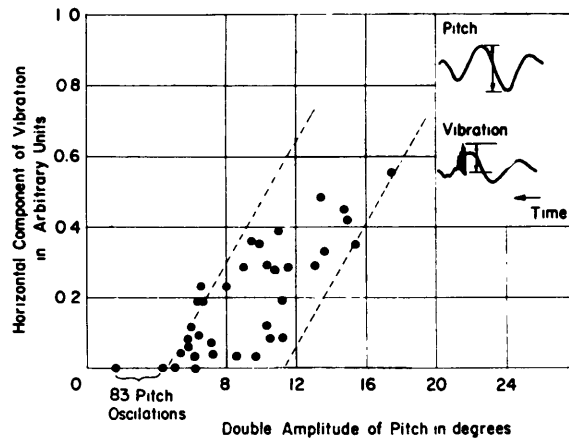


Fig. 56 Relation between pitching motion and intensity of induced vibration in irregular waves (5.5-ft. model, 0.1 knot)

lar and irregular seas can be seen from the fact that the scatter of points obtained in irregular seas is within the limits established in the regular wave tests.

The results of the full-scale trial, carried out on the *Compass Island* indicated that no direct relationship could be observed between the magnitude of pitch and that of the associated vibration. On the other hand, a linear relation between them was obtained in the present tests in regular waves as shown in Fig. 20. In an attempt to clarify these somewhat contradictory results, the intensity of the induced vibration measured in irregular waves was plotted as a function of the amplitude of pitch and the result is shown in Fig. 56. The maximum peak-to-peak variation of the induced vibration was taken as a measure of its intensity and the double amplitude of the pitching motion a half-cycle prior to the occurrence of the vibration was taken as a measure of pitching motion (see explanatory sketch in the figure). The points are scattered over a wide range; however, a general tendency of the relation between the intensity of vibration and the magnitude of pitching motion can be observed. No vibration occurs for small pitching motions, and the vibration increases with increasing of pitching motion. In short, it may be concluded that the relationship can be observed between the magnitude of pitching amplitude and the intensity of the associated vibration even in irregular seas, and that the tendency is the same as that obtained in regular wave tests.

*Prediction of Forces Acting on Fin.* The distribution of the double amplitude of hydrodynamic pressures acting on the fin was examined to determine if it is of the Rayleigh type frequently

Table 9 Comparison Between Predicted and Measured Variation of Pressure<sup>a</sup> on Fin A (Model speed 0.1 knots)

	Port fin	Star-board fin
Predicted pressure, psi:		
Average.....	0.069	0.078
Significant.....	0.110	0.125
Highest in 222 variations.....	0.192	0.216
Measured pressure, psi:		
Average.....	0.063	0.077
Significant.....	0.112	0.126
Maximum in 222 variations.....	0.222	0.230

<sup>a</sup> Variation of pressure refers to the peak-to-peak variation of pressure neglecting the impact pressure at the instant the vibration occurred.

used in wave and ship-motion analysis. Should this be the case, it is possible to estimate the magnitude of pressures acting on the fin in rough seas and the result would provide valuable information for design purposes. Hydrodynamic pressures here refer to the low-frequency pressures produced by the wave and ship motions, and they are directly related to the reduction in pitch. The peak-to-peak variations of the pressure were considered, neglecting the impact pressure at the instant the vibration occurred. Analysis was made of the record obtained in irregular waves corresponding to a partially developed Sea State 7, and at a ship speed of 1 knot.

In Table 9 a comparison is made of the measured variations of pressure on the fin and those predicted by assumption of the Rayleigh distribution. The predicted average and significant pressures agree well with the measured values for both sides of the fin. Although the result of  $\chi^2$  test is inconclusive, a comparison of predicted and observed values suggests that the assumption



of the Rayleigh distribution provides a practical tool.

The pressure due to ship motion acting on the fin in rough seas can be estimated if the response-amplitude operators are known. The response-amplitude operators for the pressures acting on the fin were obtained in the tests on the 22-ft model, and full-scale values are shown in Fig. 57. Using Neumann's spectra for Sea States 5 (24-knot wind) and 8 (40-knot wind), prediction of the magnitude of pressures acting on the fin

**Table 10 Predicted Magnitude of Pressure Variation Acting on Fin**

(Fin area = 2.8 per cent of water-plane area, ship speed = 10 knots full scale)

Sea State.....	5	8
Wind force, knots.....	24	40
Predicted pressure, psi:		
Average.....	3.2	11.9
Significant.....	5.1	19.0
Highest in 1200 variations (1 hr ship operation).....	9.6	36.0
Highest in 6000 variations (5 hr ship operation).....	10.6	39.6
Highest in 12000 variations (10 hr ship operation).....	11.0	41.0

was carried out for a ship speed of 10 knots, Table 10.

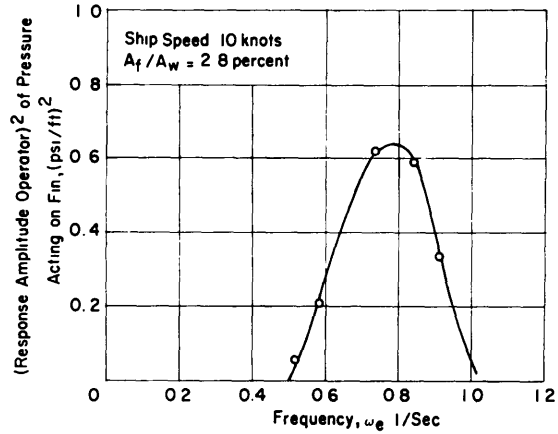
It can be seen from the table that the maximum pressure variation acting on the fin during 10 hr of ship operation at 10 knots speed in Sea State 8 is expected to be 41 psi, for a fin having an area of 2.8 per cent of the water plane. For fins having an area of 1.6 per cent of the water-plane area (the fin actually installed on the *Compass Island*), the estimated pressure for the conditions just given is 25 psi. This pressure gives a double-amplitude force of 375 tons on each side of the fin.

This magnitude of force on the fin is required for just reducing the pitching motion. In addition, a large impact force will be applied to the fin due to slamming and/or cavity collapse, and this should also be taken into consideration in the design of a fin.

#### Feasible Methods for Minimizing the Vibration

This experimental study indicates that the induced vibration may be eliminated or reduced by two approaches—the hydrodynamic and the structural:

*From the Hydrodynamic Point of View.* The cavity caused by the presence of the fin should be eliminated, since the collapse of this cavity is the source of the vibration. For this purpose, the simplest method is to use a proper fin configuration. Fin X, is an example of a suitable antipitching fin.



**Fig. 57 Response amplitude operator of pressure acting on fin for 10 knots speed**

Deep submergence of the fin is apparently a good method to eliminate the cavity. However, this method appears to be practically unrealistic, since the test results suggest that a fin submergence of at least 8 ft (full scale) is required to prevent cavity generation.

Devices by which the impact pressures could be equalized or be adjusted so as to act simultaneously on both sides of the ship bow and/or fin are also worthy of consideration.

*From the Structural Point of View.* Although it may be difficult to achieve in practice, it is desirable to design the hull so that the natural frequencies of the torsional and horizontal flexural vibrations of the hull are not close, since the coupling effect between the two types of vibration appears to contribute to serious vibration.

Structural reinforcement of the bow would be helpful to lessen the intensity of the vibration. However, even in this case effort should be made to separate the natural frequencies.

#### Conclusions

On the basis of this experimental study concerning the effect of an antipitching fin on the seaworthiness characteristics of the *Mariner* and the induced vibrations, the following conclusions are drawn:

#### Mechanism of Occurrence of Induced Vibration

1 The mechanism of occurrence of the induced vibration is of two types. One is the cavity type and the other is the slam-plus-cavity type. The former is usually associated with deep-draft condition, while the latter is generally experienced at light draft condition of the ship. A significant and common feature in both types is that the impact force, due to collapse of the

cavity and/or ship slamming, is applied to the bow side *and* to the fin. Therefore, it is of importance to minimize this impact force for elimination of the induced vibration.

2 The induced vibration is initially a torsional rather than a pure horizontal flexural vibration. Since both vibrations generally coexist for a ship, serious vibration will be produced if the natural frequencies of the torsional and horizontal flexural vibration of the hull are close.

3 The 1st mode of torsional and the higher modes (2nd or 3rd) of horizontal flexural vibration appear initially; then the higher modes decay at a faster rate than the fundamental horizontal flexural vibration, and finally the latter remains and continues for a relatively longer time than the higher mode vibrations.

4 The intensity of the induced vibration is maximum at the ship bow and minimum at a position just aft of midship for a ship equipped with a bow fin. For the fixed stern fin, the intensity of the vibration is maximum at the bow even though, in this case, the source of vibration is located near the stern.

#### ***Fundamental Properties of the Induced Vibration***

5 The vibration is usually not associated with simultaneous loading of the port and starboard sides, as there is a time differential of loading in most cases. The induced vibration is seriously augmented when this time differential plus the time duration of loading is nearly equal to the natural period of the 1st mode of torsional vibration of the hull.

6 The intensity of the vibration is a function of pitching motion. The induced vibration occurs when the pitching motion is of a certain magnitude, and its intensity increases approximately linearly with increase in pitch.

7 The induced vibration does not appear in waves of low height for any ship speed. There exists a minimum wave height which causes the vibration.

8 The induced vibration occurs at the instant the ship bow has an almost horizontal position.

9 If the fin is always greater than a certain depth beneath the moving wave surface, the cavity will not form and hence the vibration will not occur for any ship speed or any wave height. This limiting depth is about 8 ft for the Mariner in regular waves of length equal to the ship length and height of  $\lambda/20$ .

10 The intensity of the induced vibration is approximately linearly proportional to the magnitude of impact pressure on the bow side and on the fin. However, a small impact pressure applied

only to the bow side does not produce the vibration.

#### ***Effect of Location of Bow Fin***

11 A fin located in the forward 10 per cent of the length is particularly effective for reduction of pitch at speeds over 15 knots. The effectiveness in reducing pitch of fins located aft of the 15 per cent point decreases rapidly for all speeds.

12 A fin located further aft than  $0.15L$  aft of the forward perpendicular produces no appreciable vibration, while a fin at the more forward locations induces vibrations.

13 A bow fin located aft of the forward perpendicular is not beneficial for heaving motion at speeds below 15 knots. Only if the fin is located near the forward perpendicular is it beneficial for heave at all speeds.

#### ***Effect of Size of Bow Fin***

14 Fins of large area produce large reduction in pitch and bow vertical acceleration but relatively small reduction in heave. However, the reduction in pitch, heave, and bow acceleration increases approximately linearly with increase of fin area.

15 The forces acting on the fins increase approximately linearly with increase in fin area. They are also linearly proportional to the reduction in pitch if other conditions (such as wave dimensions, ship speed, and so on) are equal.

16 The intensity of the induced vibration is proportional to increase of fin area for fins whose area is less than 2.5 per cent of the water-plane area.

17 The intensity of the induced vibration is very nearly proportional to the impact force acting on the fins irrespective of the fin size.

18 Increasing the size of fins is favorable for reduction in pitch but is unfavorable for vibration. The relation between pitch and intensity of vibration is linear for fins of area less than 2.5 per cent of the water-plane area.

#### ***Fixed Stern Fin***

19 A fixed stern fin appears to be less favorable for reduction in pitch than the fixed bow fin, and also increases the resistance in wave considerably.

20 The magnitude of pressure acting on a fixed stern fin is approximately one-third of that acting on the bow fin in waves of  $\lambda/L = 1.0$ ,  $h/\lambda = 1/20$ . The induced vibration is not a serious problem for the fixed stern fin.

21 A combination of bow and stern fin does not appear to be beneficial, as a single bow fin having the proper configuration is as good as the combination.

### **Effect of Bow Fin Configuration**

22 A fin having holes of proper size and shape (Fin X) appears to be beneficial for both pitch reduction and vibration prevention; that is, the pitch response amplitude operator for this fin is nearly equal to that for a fin without holes (Fin A), but the vibration for the former is much less severe than with Fin A. However, the resistance and heaving motion for Fin X is greater than that with Fin A. These characteristics may be improved by giving proper consideration to the sectional profile of Fin X as well as the angle at which it is installed.

### **Pitch Reduction and Induced Vibration in Irregular Waves**

23 Reduction in pitch by about 20 per cent can be expected on the Mariner in a fully developed Sea State 5 of a wind force 24 knots when equipped with either Fin A or X. Also 13 to 14 per cent reduction can be expected for Fin A and Fin X, in a fully developed Sea State 8 of a 40 knot wind force.

24 The probability of vibrations with Fin X in irregular waves is much less than that with Fin A at a speed of 1 knot. Also, the intensity of the generated vibration with Fin X is much less severe than that with Fin A.

25 The fundamental properties of the induced vibration obtained in regular waves are also applicable in irregular waves. A remarkable feature found from the tests in irregular waves was that an impact pressure was *always* observed on the fin whenever the vibration occurred; no vibration occurred without impact pressure on the fin.

26 It appears that the pressure acting on the fin which is produced by the wave and ship motions and which gives a resisting moment to a ship for reducing pitch, is distributed according to the Rayleigh distribution. The predicted maximum pressure variation acting on the fin during 10 hours ship operation in Sea State 8 at 10 knots speed is 25 psi for the fin actually installed on the Mariner. This pressure gives a double amplitude of force of 375 tons on each side of the fin. In addition to this force, a large impact force will be added to the fin due to slamming and/or cavity collapse, and this should also be considered in the design of a fin.

27 Means for minimizing the induced vibration should be considered both from the structural as well as from the hydrodynamical points of view. Specifically, the cavity caused by the presence of the fin should be eliminated, and, if possible, the natural frequencies of the torsional

and horizontal flexural vibrations of the hull should be separated.

### **Acknowledgment**

The author wishes to express his thanks for the co-operation of Mrs. M. D. Bledsoe. Her discussion and help during the test and preparation of this paper are most gratefully acknowledged. Valuable technical advice was received from Dr. W. E. Cummins throughout the preparation of this paper for which the author is deeply appreciative. The continued encouragement of Mr. P. Golovato in this study and the discussions with Mr. R. T. McGoldrick in the study of the mechanism of occurrence of the induced vibration are also greatly appreciated.

Thanks are also due to Messrs. R. S. Kiley, N. G. Milihram, J. H. Hendrican and J. A. Bonilla-Norat for their active co-operation in the tests. In addition, the appreciation should be expressed to Miss N. Walker and Mr. A. Gerstenzang for their outstanding help in this work.

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## Appendix 1

### Outline of Experiment on a 22-Ft Mariner Model

In order to determine the lateral force acting along the bow side due to collapse of a cavity, and to measure the linear and torsional components of the induced vibration, tests were made on a Mariner model of 21.85 ft length. The model was made of wood, dynamical similitude considerations were omitted.

Four pressure gages were installed on the starboard side of the bow at  $0.038 L$  aft of the forward perpendicular. Three accelerometers were arranged along the vertical centerline at this location, for measurement of the horizontal component of the induced vibration. In addition, three accelerometers were located transversely at a vertical location between the main and forecastle deck levels to measure the vertical component of the vibration.

Resistance, pitch and heave motion, and force acting on the fin were also measured. The tests were conducted in regular waves of various lengths and heights.

## Appendix 2

### Resolution of Acceleration into Linear and Rotary Components

In order to resolve the horizontal component of acceleration measured along the centerline of the bow into linear and rotary components, a least-square fitting method was applied.

In Fig. 58, let the distance of the three accelerometers above the base line be  $a$ ,  $b$ , and  $c$ ,

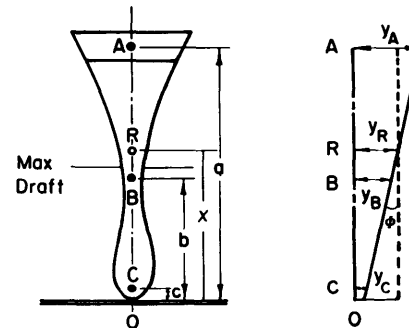


Fig. 58 Resolution of acceleration into linear and rotary components

and let the measured horizontal components of acceleration by  $\ddot{y}_A$ ,  $\ddot{y}_B$ , and  $\ddot{y}_C$ , respectively. Let  $R$  be the center of rotation at distance  $x$  above the base line, and let the linear (horizontal) and rotary accelerations by  $\ddot{y}_R$ , and  $\ddot{\phi}$  respectively. The quantities  $x$ ,  $\ddot{y}_R$ , and  $\ddot{\phi}$  are unknown.

Then, the acceleration at the points A, B, and C can be expressed by the following equations:

$$\begin{aligned}\ddot{y}_A &= \ddot{y}_R + (a - x) \ddot{\theta} \\ \ddot{y}_B &= \ddot{y}_R + (b - x) \ddot{\theta} \\ \ddot{y}_C &= \ddot{y}_R + (c - x) \ddot{\theta}\end{aligned}\quad (1)$$

where  $\theta = \tan \phi$ .

From equation (1), the following relations can be derived:

$$\begin{aligned}\ddot{y}_A - \ddot{y}_B &= (a - b) \ddot{\theta} \\ \ddot{y}_B - \ddot{y}_C &= (b - c) \ddot{\theta} \\ \ddot{y}_C - \ddot{y}_A &= (c - a) \ddot{\theta}\end{aligned}\quad (2)$$

The foregoing equation can simply be written as

$$\dot{Y}_i = A_i \dot{\theta}, \quad i = 1, 2, 3 \quad (3)$$

where  $\dot{Y}_i$  and  $A_i$  are known, while  $\dot{\theta}$  is unknown.

In applying the least-square method to equation (3), the following condition is required:

$$\frac{\partial E}{\partial \dot{\theta}} = 0 \quad (4)$$

where

$$E = \Sigma (\dot{Y}_i - A_i \dot{\theta})^2$$

Therefore,  $\dot{\theta}$  can be determined from the following relation;

$$\Sigma \dot{Y}_i A_i - \dot{\theta} \Sigma A_i^2 = 0 \quad (5)$$

Since the unknown quantities  $x$  and  $y_{\bar{x}}$  cannot be determined by the procedure given, an assumption from McGoldrick's paper [19] will be used for determining the position of center of rotation; that is, rotary motion for a particular section takes place about the center of shear of that section, and that this center of shear corresponds to the centroid of the area of the section. By introducing this assumption, and by using the value of  $\dot{\theta}$  which is determined from equation (5), the linear (horizontal) acceleration at the center of rotation can be obtained.

### Appendix 3

#### Estimation of Minimum Submergence or Maximum Emergence of Fin

The distance between the fin and wave surface in regular waves can be approximated if certain simplifying assumptions are made.

Since the fin is fixed on the keel line at the ship bow, the vertical motion of the fin relative to the water surface is equal to that of the ship bow. The vertical position of the fin relative to the mean water surface is then given by

$$r_f = u_m \cos(\omega_e t - \epsilon_{w-B}) - H \quad (6)$$

where

$u_m$  = amplitude of vertical motion at ship bow  
 $H$  = ship draft

$\epsilon_{w-B}$  = phase lag between wave and ship bow motion

Wave motion at the ship bow is given by

$$r_w = \frac{h}{2} \cos \omega_e t \quad (7)$$

where

$$\frac{h}{2} = \text{wave amplitude}$$

Therefore the relative distance between wave and fin is given by

$$r = -r_w - r_f$$

It is noted that if  $r > 0$ , the fin does not emerge from the wave surface. The foregoing equation can be reduced to

$$r = r_m \cos(\omega_e t - \epsilon_0) + H \quad (8)$$

where

$$r_m = \left[ \left( \frac{h}{2} \right)^2 + u_m^2 - h u_m \cos \epsilon_{w-B} \right]^{1/2}$$

$$\tan \epsilon_0 = \frac{u_m \sin \epsilon_{w-B}}{\frac{h}{2} - u_m \cos \epsilon_{w-B}} \quad (9)$$

We wish to evaluate the minimum submergence or maximum emergence of the fin. These are

$$r = -r_m + H \quad (10)$$

when,

$r < 0$ : Fin does not emerge from the wave surface, and equation (10) gives the minimum submergence of the fin.

$r > 0$ : Fin emerges, and equation (10) gives the maximum emergence of the fin.

For application of equation (10), the amplitude of vertical motion at the ship bow,  $u_m$ , and the phase between wave and bow motion,  $\epsilon_{w-B}$  are required. The amplitude of vertical motion at the ship bow was obtained from the measured vertical acceleration at the bow. Since the phase between wave and bow motion was not measured in the present tests, the experimental data given in [24] was used in the numerical calculations. The results of the numerical calculations for the given conditions are shown in Fig. 25.

### Discussion

**Prof. M. A. Abkowitz, Member:** I find myself in disagreement with several conclusions and analytical approaches described in the paper. This subject of pitch reduction by fins and the problem

of its associated vibrations has been one of deep interest to me for many years and has been worked on off and on in a fairly continual, though small, effort by students conducting research at the

Massachusetts Institute of Technology in association with our Ship Model Towing Tank.

Ever since 1953 this work has been in progress and the vibration problem was known to exist as far back as 1954. In my paper on this subject before the Society in 1959 [8],<sup>3</sup> clear descriptions were given as to the cause of the bow vibration associated with the use of antipitching fins. The conclusion then was that a major part of the vibration was caused by the slight time differential in the collapse of the ventilated bubble on the port and starboard sides. The collapse was the result of an instability and therefore could be initiated randomly from either side. Another conclusion of this paper was that the predominant force resulted from the pressure of bubble collapse on the side of hull and not on the surface of the fin. Subsequent research by Goren and Pearlman [9] clearly indicated the relative orders of magnitude of the pressure on the side of the hull and the torque transmitted to the side of the hull through loading on the fin. The result of this work by Goren and Pearlman clearly established that the major load causing this vibration was from the pressure on the side of the hull. So, you see, I question the statement of the author that a "... complete physical explanation of the cause and nature of this hydroelastic phenomenon has not been given."

The author presents what apparently is a strong case for the loading on the fin being a large contributor to the vibration excitation since he concludes with the following remarks: "The foregoing analysis of the phenomenon proves that a vertical force applied to the fin contributes to the vibration." This conclusion is based on the phasing of the various bow vibrations with the phasing of the loading on the fin. However, one must recall that the pressure field felt by the fin is also the pressure field felt by the side of the bow and therefore the sideways loading on the bow would also have the phasing indicated in the analysis and could just as well have contributed to the measured vibration which the author seems to single out for a vertical force on the fin. In fact, all through the paper the author keeps pointing out that at the time he measures a vertical load on the fin there is an associated impact on the hull whether this is a static load, loading in regular seas, or loading in irregular seas. This constant reference to the phasing of the load measured on the fin and the impact on the hull whether unintentional or not, leads the reader to feel that this is the absolute cause of the vibration.

<sup>3</sup> Numbers in brackets designate References at the end of the paper.

In each case where the phasing of the vertical load on the fin relative to the impact on the hull is referred to, a similar statement can be made about the transverse load on the bow and the vibration impact on the bow. Again, I must reiterate that this phasing argument is no proof at all that the fin loading contributes markedly to the bow vibration.

Another argument with which I cannot agree is where the author essentially states that because a moment applied to a structural section of non-symmetrical elastic restraints can produce both a rotational and linear vibration this necessarily means that because one obtains a combination of linear and vibrational rotations that this motion must of necessity have arrived from a moment acting on a nonsymmetrical elastic structure. So, there is questionable basis for his conclusion that the vibration induced by an antipitching fin essentially starts as a torsional rather than a horizontal flexural vibration.

Another point in the analysis which is not clear to me is demonstrated in the discussion of the effect of ship speed. The conclusions derived therefrom apparently indicate that speed does affect the fin-vibration problem. It may be from the basic physical phenomenon of water flow over the fin and hull and that speed may contribute in some way, but the results indicate that most of the effect which the author attributes to speed are due to the effects of speed on the motion of the hull in going through resonance which in turn affects the hydrodynamics around the fin. This should be clearly stated in that much different conclusions will be arrived at if one should change the wave length in which tests are carried out. Incidentally, practically all the conclusions with regard to pitch reduction and fin vibration demonstrated in the paper are for wave length equal to ship length. Research at MIT has indicated clearly that even serious vibrations occur at wave lengths longer than that of the ship, especially waves which are steeper than those indicated in the plots.

The author comes to the conclusion and indicates in the results that fin X gives less vibration than fin Y by a factor of 3 and that slotted fins, in effect, cause reduced vibrations. I cannot see from physical hydrodynamics that if pressure release through the fin by means of slots is the cause of reducing vibrations, why fin X should be so much better than fin Y, considering that at the Reynolds number at which the model tests must have been carried out fin Y must have allowed more flow through its surface. And since I am talking about Reynolds numbers, I might point out here that the scale effect of flow through

these holes based on the local Reynolds number of the model holes would be so great as to make very questionable the performance of similar geometry on the full-size ship both with regard to pitch motion, bow vibration, and hull drag. I would like to refer the author to such a problem encountered by the David Taylor Model Basin in its research with respect to flooding holes on submarines.

Since I personally lean heavily towards physical phenomenon analysis and have been taught by experience of the troubles one runs into using electronic devices such as strain gages, accelerometers, and so on, as measuring devices, I might be tempted to suggest that somewhere along the line one should test and be completely assured of the measurements coming out of the instrumentation. Along this line I should like to ask the author if he checked the transient response (not frequency response) of the various elements which he is comparing phasing of the different measurements. For instance, it takes time for the loading on the fin to deflect the fin to actuate the strain gage and eventually move the recording pen. Similarly, it takes time for the loadings to deflect the hull, strain the strain gauge and move the recording pen. It takes time for pressure to activate a pressure transducing device (and if this device should be a column of water, a relatively long time) to actuate instruments. If the time constants of these measuring devices are much greater than about 0.002 sec, then a lot of the phasing analyses done may be subject to instrumentation error.

The author comes to certain conclusions relative to the depth at which ships may be devoid of cavities forming over the fin even under the restriction of a given wave length and wave height. However, I must remind the author that there is such a thing as scaling of the atmospheric pressure which results in a cavitation number which must be used properly in scaling ventilated phenomenon. A straight carry-over of bubble collapse and to the full size using Froude scaling without the cavitation scaling may lead to errors in full-scale prediction.

In the summary of conclusions, I would like to restate conclusions 10, 17, and 25 which will be telling exactly the same interpretation of results:

10 The intensity of the induced vibration is approximately linearly proportional to the magnitude of impact pressure on the bow side and on the fin. However, a small impact pressure applied only to the fin does not produce the vibration.

17 The intensity of the induced vibration is

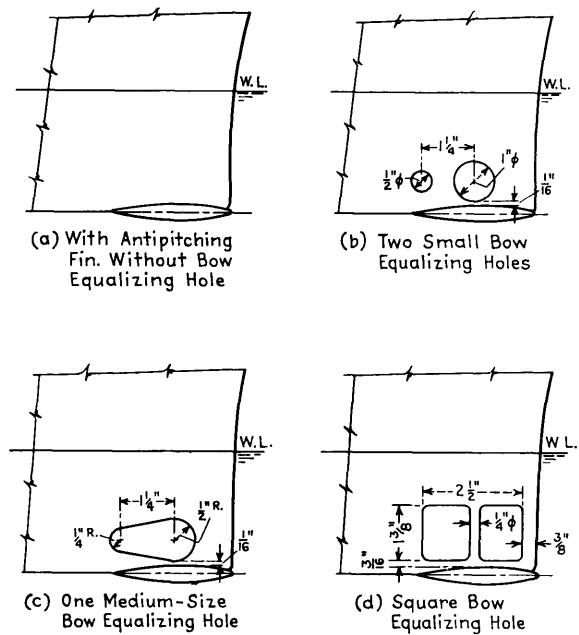


Fig. 58 Four views of a mariner bow having various pressure equalizing holes cut through the bow above the antipitching fin

very nearly proportional to the impact force acting on the bow irrespective of the fin size.

25 The fundamental properties of the induced vibration obtained in regular waves are also applicable in irregular waves. A remarkable feature found from the tests in irregular waves was that an impact pressure was always observed on the bow whenever the vibration occurred; no vibration occurred without impact pressure on the bow.

I believe that the author has done a very good research study about an interesting and valuable problem. None of my criticism has been on this phase of the project. It just happens that I disagree with the author on some of the analyses of the results of the experiments and the conclusions derived therefrom. At MIT we have been tackling the vibration problem from the point of view of reducing the impact load on the side of the hull through providing some flow and therefore pressure relief by the port and starboard sides of the upper surface of the fin. This pressure relief should enhance the probability that the bubble collapse on the port and starboard fin will occur at the same time and therefore reduce or eliminate the initial impact which causes the bow vibration. Two schemes were tried: One was to connect the upper surface of the port fin with the upper surface of the starboard fin through the thickness of the foil; the

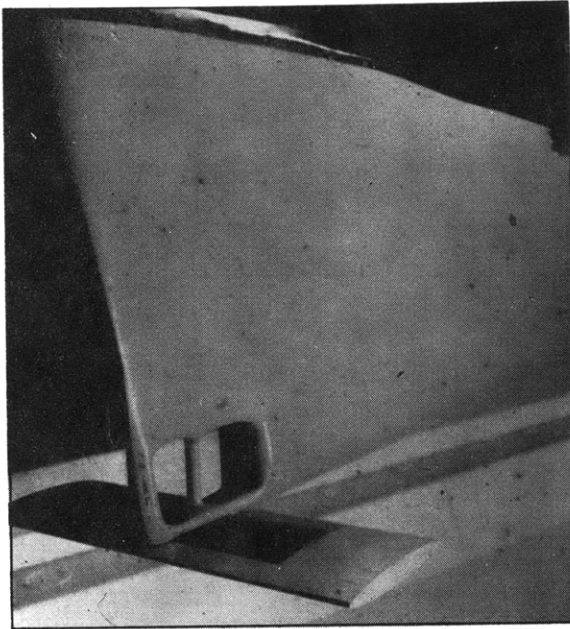


Fig. 59 Photograph showing hole D in the actual test model

other method was to increase progressively the size of the holes in the bow of the ship just above the fin location. The side holes should contribute favorably in three ways:

- 1 They would not alter the antipitching effects of the fin.
- 2 They allow cross flow and therefore pressure release between the upper surface of the port and starboard fin.
- 3 They reduce the actual area on which pressure can act.

Although we are in the midst of the model testing program, we can say at this time that these bow openings have a substantial effect in reducing bow vibrations. Fig. 58 of this discussion shows the various types of openings which were tested on the models and Fig. 59 shows the bow of the *Mariner* model with the largest opening. Of course, in any actual practice, the lines of the model will be faired on both sides of the opening. There should be no serious objection to transverse holes on the bow since several ships have been built with such holes in association with the use of bow thrusters. Another way of providing the hole and getting rid of the side area is to have a raked stem and support the foil on a bow frame which is something like the inverse of the stern-frame opening.

M. D. Bledsoe,<sup>4</sup> *Visitor*: This paper is the first

<sup>4</sup> David Taylor Model Basin, Washington, D. C.

which proposes that the vibration is basically of a torsional nature and that the coupling between the torsional and horizontal vibrations leads to the serious vibrations experienced by ships when equipped with a fin. Without this concept one fails to look to the fin itself as a direct source of the vibration but rather as an indirect source; i.e. how its presence affects the flow about the ship resulting in lateral impact to the bow side.

The author says "it can be concluded that the vertical impact pressure applied to the fin and the lateral impact pressure applied to the ship bow side both contribute to the vibration." He points out that even in case of bow impact, the problem is still a torsional one (rather than pure horizontal) since the portion of the bow to which the load is applied is below the center of rotation for that section. If the torsional moment on the ship bow is the sum of that due to fin loading and bow loading, can the author estimate the relative importance of the two types of loading in inducing the vibration? In other words, is the vibration primarily the result of impact on the fin, or on the bow or are they both equally important? For the case shown in Figs. 9 and 10, the maximum vibration (referring to that measured at the keel) occurs when the difference in bow forces is a maximum. At this same time the difference in fin forces is only about one half that of the bow force difference. If the moments associated with these forces are evaluated, one finds that the moment due to force on the bow is more than three times that of the moment resulting from fin loading. This certainly implies that the impact on the bow side is the more significant contributor to the induced vibration.

The same conclusion can be reached by examination of the data in Fig. 28, especially since the lever arm for the bow forces is larger than that for the fin forces. It does not appear, however, that slamming was involved in any of the cases shown in these figures. Considering the results of the full-scale trials of the *Compass Island* where slamming pressures of the order of 72 psi were measured on the fin, we find that a moment of 24,000 ft-tons can be expected just due to slamming alone for a fin comparable in size to the one discussed in the present paper. This moment of 24,000 ft-tons is 2.5 times the moment due to the bow and fin loading when slamming doesn't occur and over three times that due to impact on the bow alone. Thus it appears that if we consider slamming, and this must be considered in a practical case, the force on the fin is far more significant than the force on the bow side in inducing the vibrations. The author's comments in this area would be appreciated.



**Prof. E. V. Lewis, Council Member:** In general, the best approach to the problem of reducing ship pitching and heaving motion is to adopt hull characteristics in the design stage which will permit the avoidance of the critical conditions of synchronous motions at speeds as close as possible to design speed. However, even though the designer has gone as far as is feasible in this direction, every ship will at some time encounter synchronous pitching when speed and the roughness of the sea have increased sufficiently. Under such circumstances a simple damping device can be highly effective in reducing pitching motions. The paper does not bring out the fact that fixed fins located at the bow do act as damping devices, since in their bow location the proper phasing between wave excitation and ship motion is generally experienced.

The effectiveness of a bow fin in reducing pitching has been known for some time. Hence, the main contribution of this paper is in demonstrating that the design of the fin can be modified to reduce the hull vibration that has to date prevented the general adoption of antipitching fins. Fin X appears to have provided a solution to the vibration problem without seriously affecting the fin's effectiveness in reducing pitch. The fact that fin X is not so effective as fin A in reducing heave is not believed to be significant.

The model tests in irregular waves are of particular interest since they confirm the performance of the fins under more realistic conditions than are provided by regular waves. However, it should be noted that if the irregular wave patterns had been reproducible, a direct comparison of the different fins could have been made under various conditions. In this case, spectra of pitching motion and fin pressure had to be synthesized from many regular wave tests instead of being obtained directly. This indicates the very real value of equipment for generation of reproducible irregular (long-crested) waves in conventional long model tanks, such as the equipment which has been in use at the Davidson Laboratory for many years.

A great deal of the paper is devoted to a study of the nature of the vibrations of the model hull induced by the fins. Although the results are of interest, the fact that dynamical similitude was not obtained leaves some doubt regarding the statement that "the qualitative vibration characteristics of the model which are derived from the test results are applicable to a full-scale ship." Certainly even qualitative application of model vibration results to full scale must be made with caution.

One hesitates to suggest more work to some

one who has done such a comprehensive piece of research as this. But an important consideration in the actual installation of fins on ships is the avoidance of serious resistance increases. As noted in conclusion 22, the angle at which the fin is installed is an important factor to be investigated in any particular installation.

**M. D. Pearlman, Associate Member:** The detailed pressure and vibration measurements made in the bow region of the model give a clear picture of what is happening at the instant of impact. The author's systematic variations of fin size and fin location indicate the advantages which can be realized by modifying the bow fin, or relocating it, or both.

He points out, and quite rightly so, that it is not a horizontal vibration alone which manifests itself in the ship, but rather a horizontal vibration plus a torsional vibration.

He claims that the cause of this vibration is due to both the collapse of a ventilated bubble against the fin surface and the side of the bow, as well as slamming. Work done at MIT recently by Y. Goren and the writer proved that the principal contribution to vibration was, in fact, the collapse of the bubble and its impingement against the side of the bow. By means of a technique which the author did not employ, that is, attaching strain gages directly to the hull at its axis of vertical bending, we were able to estimate the horizontal impact necessary to excite the model. Also, by means of a bow-fin dynamometer mounted within the model, we were able to separate the forces acting normal to the plane of the fin and the torque which the fin transmitted to the model. A comparison of model bending strain compared to fin torque at the instant of impact showed that the model strain was of the order of 75-100 times higher than that which could have been produced by torque on the fin. From these results we can safely conclude that the collapse of the bubble directly against the side of the bow causes the vibration.

We tested a 5.5-ft Mariner type model at 24-ft ship draft using a bow fin NACA 16-012 with area equal to 5 per cent of the waterplane area. The pitch reduction obtained with this fin was of the order of 35-50 per cent and it was noticeable that at no time during our tests in regular waves did the fin come to the surface and slam. From this we conclude that slamming should not be a problem with correctly designed fins; but should the fin clear the surface and re-enter, there will, of course, be sudden loading and, therefore, vibration.

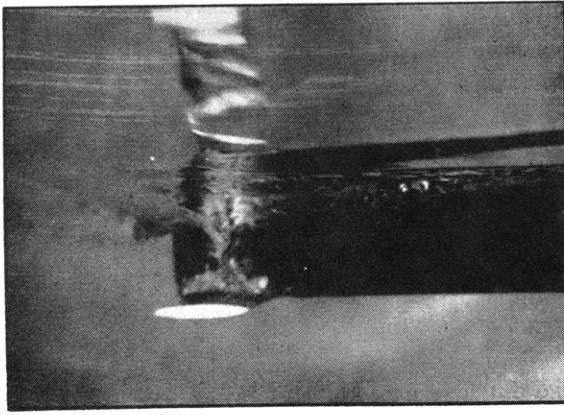


Fig. 60 Formation of bubble and ventilated shaft

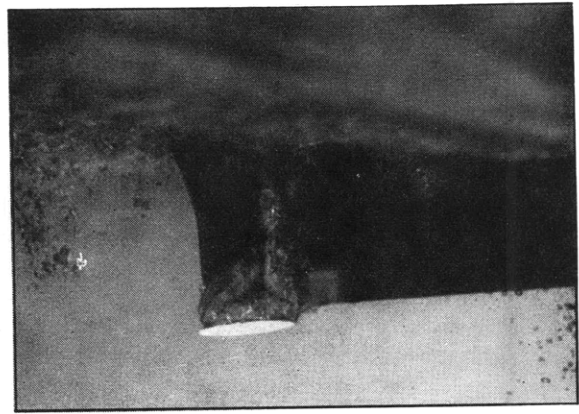


Fig. 61 Bubble just about to collapse

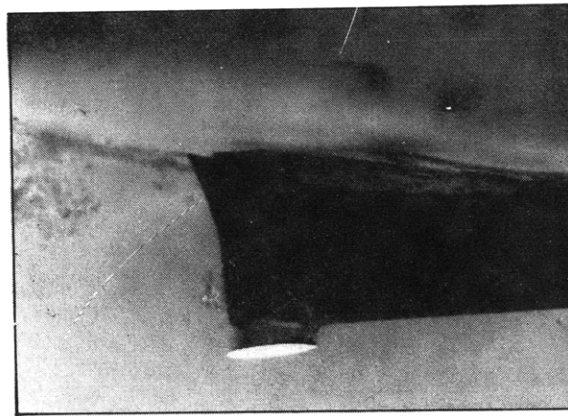


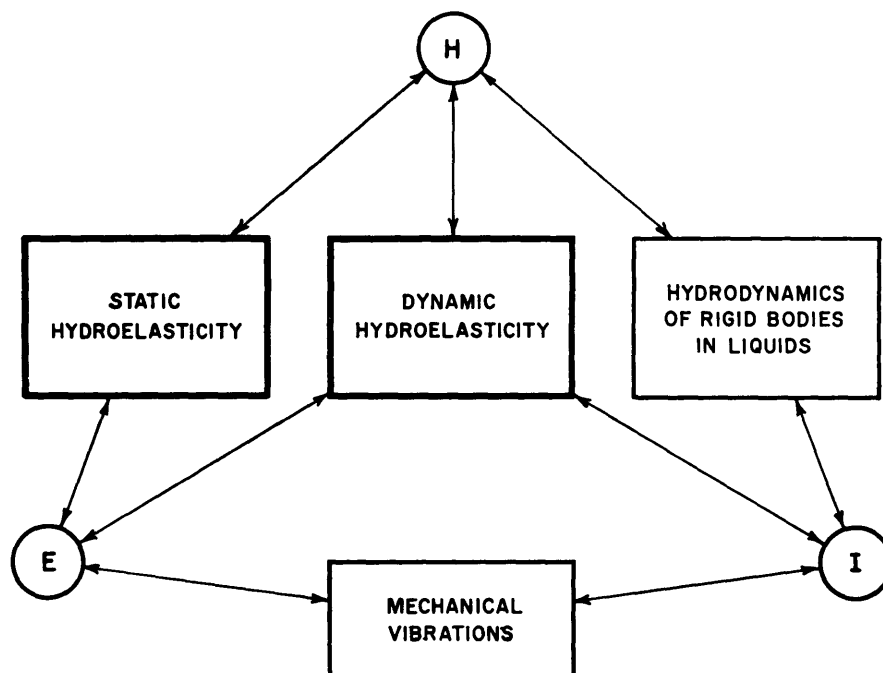
Fig. 62 Bubble just after collapse, and small wool tufts along the side of the bow indicate the direction flow when this occurs

Underwater photographs of ventilated bubble as it exists above the antipitching fin

Referring to the author's solution of the problem of eliminating vibration, and our work in this area: The author has cut slots in his fin to relieve the low pressure on the top surface of the fin and prevent formation of the cavity. He has moved the fin aft systematically with the same motive and also hopes to eliminate slamming. It is obvious that both of these methods will reduce vibration; however, at the same time, they will reduce the effectiveness of the fin, and ship motions will increase. Working on the assumption that the impact is principally against the hull, rather than on the fin, we took underwater photographs of the bubble and determined its shape and location just prior to collapse. The model has been tested with a hole the shape of the bubble cut through the hull above the fin

to allow relief of the bubble collapse impact. Rather than impacting against the hull, the shock travels through the hole in the bow and considerably reduces vibration; and at the same time, none of the pitch damping forces have been sacrificed. It seems that the success of the relief hole through the bow, in reducing vibration, proves that the vibration impact is felt against the hull, rather than on the fin. A little more research in this area should advance the state of art to the point where correctly designed fins mounted beneath suitably altered bows will be practical.

**C. J. Henry, Associate Member:** The term, hydroelasticity, derives from the analogous field aeroelasticity which is the study of the mutual



H HYDRODYNAMIC FORCES    E ELASTIC FORCES    I INERTIAL FORCES

Fig. 63 Hydroelastic triangle of forces (adapted from: A. R. Collar, "The Expanding Domain of Aeroelasticity," *Journal of the Royal Aeronautical Society*, vol. 50, August 1946)

interaction between aerodynamic forces and elastic forces, and the influence of this interaction on aircraft design. Hydroelasticity, therefore, is the study of the mutual interaction between hydrodynamic forces and elastic forces, and the influence of this interaction on ship design. Hydroelastic problems arise when elastic strains of a structure induce additional hydrodynamic forces which in turn cause additional elastic strains which will induce still greater hydrodynamic forces; that is the hydrodynamic forces due to the elastic deformations of a structure must be considered when hydroelastic phenomena are involved. The elastic response and hydrodynamic force in this paper are not related in this manner. In fact, the hydrodynamic impact would still exist even if the hull were rigid and not flexible.

The interaction between elastic forces and hydrodynamic forces is not involved; only the response of an elastic body to an impulsive force. This phenomenon in fact belongs in the field of mechanical vibrations; the study of the impact force belongs in the field of dynamics of rigid bodies in liquids. These fields are shown graphically in the hydroelastic triangle of forces, Fig. 63 of this discussion.

The intermodal coupling, which is the cause of responses in one mode due to excitation of a

different mode, can involve more than inertia coupling. For example, if the locus of the centers of shear of the cross sections of a ship is curved (in practice this locus would have very sharp bends), then elastic intermodal coupling terms arise in the equations of motion. In addition, hydrodynamic coupling terms are present. For instance, a lateral hydrodynamic force is developed due to rotation of a cross-sectional element of a ship about its center of shear. Theoretical techniques for determining these coupling terms are nearly nonexistent.

In addition, the duration and intensity of the hydrodynamic impact force due to antipitching fins cannot as yet be predicted.

As a result, the design problem of determining the elastic response of a proposed hull to impacts due to antipitching fins must be solved in the towing tank. This response must be determined to be less than the allowable limits of vibration of the hull structure for an acceptable design.

**A. Taplin, Associate Member:** This paper appears at first to be somewhat in the nature of a post-mortem, since the antipitching fin on USS *Compass Island* has long been removed. However, it does considerably more than demonstrate the mechanism that caused the objectionable

vibration; it points out what to avoid in future installations.

In the *Compass Island* analysis, the author indicates that he has used full-scale hull-vibration data from that ship and also from a sister ship, SS *Gopher Mariner*. He concludes that proper design should avoid having torsional and horizontal flexural vibration frequencies close to one another. Even airplane designers, who devote considerable effort to calculation and structural testing, are sometimes surprised by aeroelastic effects. Does the author consider that we now have the ability to design an antipitching fin that will not uncover new ways of producing hull vibration?

This paper shows the author's thoroughness and skill as an experimenter and as an analyst. It is hoped that his future investigation, using the Model Basin's new Seakeeping Facility and elastically scaled models, will add still more to our knowledge of reducing pitching motion.

**M. L. Sellers, Member:** With ship rolling motions now brought under reasonable control by the use of stabilizing fins it is only fitting and proper that concentrated effort be applied to that much greater problem, the control of pitching motion. The adverse results of violent pitching are too well known to repeat here and significant reductions of the same has always been a rather elusive objective for designers, particularly in the zone of synchronism and for waves of about the ship's length. The increasing use of long range radar at sea for military purposes has increased the necessity for reduction in motions because the effectiveness of these systems is contingent upon a stable platform. For merchant vessels, the demand for increase in average sea speeds makes any device attractive which permits continuation of these speeds in rough weather without sacrifice of other characteristics.

The failure of the few applications of antipitching fins made to date has been discouraging to the practicing naval architect and it is refreshing to know that our testing facilities are still working on solutions to the problem and papers such as this are a welcome interim progress report.

Part 1 of the paper appears to be a reasonable analysis of the vibration problem created by antipitching fins. It is only by determining the cause of trouble that a satisfactory solution can be found. Undoubtedly this aspect of the paper will be ably discussed by those more qualified to do so than the writer.

Part 2 which suggests practical applications of the findings of the first part was studied with hope

and interest. The suggested fin locations should prove of value and will be helpful. From a practical standpoint the 10 per cent of  $L$  location is almost necessary to provide clearance for anchors and it is fortunate that this position coincides with that which is most beneficial hydrodynamically. Owing to arrangements and hull configuration, the hawse pipe location at the shell is rather limited, particularly when there is a bulbous bow, and it is nice to know that bow fins can be located without too much interference with the anchor. Some skippers like to anchor with a little way on to dig the hook in which streams the chain aft a bit. While this probably wouldn't damage the fin, it will scrape off paint and increase the corrosion problem.

The superiority of fin X with slots is indicated by model tests. It would be interesting to know how fin X compares with the perforated fin which was applied to the *Compass Island* without too much success. Does the author expect fin X to be much superior to the modified *Compass Island* fin when applied full scale?

We have always been apprehensive of the strength of antipitching fins. We are constantly made aware of tremendous pitching forces when we see the forward end of bilge keels torn loose and corrugated forward bottom plating. Fins are subject to similar or greater loads which must be resisted by not only the fins themselves but also must be absorbed by the relatively thin bow sections. Structurally the *Compass Island* installation was adequate during the period of its service but it must be admitted that this service was of relatively short duration. One ship in such a limited time cannot be considered as a basis for general performance. Further, the forces mentioned near the close of the paper are far less than the reported design forces used for the *Compass Island*. These comments are given merely to sound a word of caution that we should be conservative in our design of early fin applications until sufficient service experience proves our fears to be unwarranted.

Whatever the degree of magnitude of the forces on the fins, in any case, the resultant scantlings will represent a considerable structure running into several tons. Although some of the fin configurations show promise in reducing vibration we are warned that to insure maximum results, one should design the hull so that the natural frequencies of torsional and horizontal flexural vibration are not close. If this could be accomplished, it is sure to mean some more additional weight.

In summary and with the present state of the art, it looks as if the major penalties for carrying a bow fin consist of substantial increase in still-

water resistance, increase in structural weight, and a chance of vibration problems. The gains are some increase in speed and reduction of motions when there is synchronism between the natural ship period and the waves. It is estimated for the average cargo vessel that evasive action and delay due to sea conditions can be pessimistically assumed to be no more than 5 per cent of the days at sea. It would seem on this basis that general adoption of bow antipitching fins is still a long way off for the average merchant vessel.

**R. T. McGoldrick, Member:** The author points out that the evidence on the *Compass Island* left little doubt that horizontal hull vibration resulted from the presence of the bow antipitching fins. Any explanation of this in terms of the coupling of torsion and horizontal flexure of the hull is of special interest since the identification of such coupling action for the *Mariner*-class ships was one of the major objectives of the hull-vibration investigation on SS *Gopher Mariner* [11].

On the occasion of the *Gopher Mariner* tests, torsional amplitudes of the hull were measured independently of the horizontal amplitudes while the hull was under excitation by a horizontal force applied at the stern at the main deck level. This was accomplished by recording simultaneously the vertical amplitudes (together with phase) at the port and starboard deck edges.

The theoretical calculations for SS *Gopher Mariner* indicated that there would be two pairs of coupled torsion-bending modes, each pair having the same number of nodes in both torsion and flexure but with phase relations reversed as shown in Fig. 33 of reference [11]. Although it cannot be said that this prediction was positively established experimentally on this occasion, at least the experimental results were not inconsistent with it.

Theoretical calculations (Tables 11 through 14 of author's reference [19]) indicated torsional amplitudes at the bow ranging from 0.02 rad to 0.14 rad for a flexural amplitude of 1 ft at the after perpendicular. To yield equal horizontal amplitudes at the main deck level due to flexure and torsion, the torsional amplitude for a hull depth of 44 ft would have to be about 0.05 rad per ft flexural amplitude.

In the absence of similar calculations for the *Compass Island* (a modified *Mariner* hull) and for the author's 5.5-ft model, only qualitative comparisons can be made. However, the author's contention that a vertical impact on either the port or starboard antipitching fin could result in horizontal vibration on the main deck level is well taken. Such an impact, yielding an impulsive

moment about the ship's longitudinal axis could in general excite any torsion-bending mode which is characteristic of the hull involved. Those modes in which the phase relation between torsion and flexure is such as to give reinforcing horizontal displacements at the main deck level could hence give the illusion of a horizontal vibration of the common type at that location. Such a case is illustrated in Fig. 33 (a) of author's reference [11]. Even a pure torsional vibration will appear as a horizontal vibration on the main deck along the centerline.

One of the most important considerations in the investigation reported in this paper is the degree of generality that can be assigned to the results. On both the *Compass Island* and on the 5.5-ft model (in which there apparently was no attempt made at scaling for dynamic effects), the presence of bow antipitching fins resulted in horizontal vibration. This has been ascribed to the coupling of torsional and flexural vibration of the hull. In the case of the model the impulse on the fin is reported to have produced an initial torsion of the hull which was followed by a 3-node horizontal vibration 2 cycles later and by a 2-node horizontal vibration  $5\frac{1}{2}$  cycles later (see author's Fig. 6).

It is a common experience in making anchor drop tests on ships to find that the sudden arresting of the anchor is followed by a complex pattern of vertical vibration indicating the presence of higher mode components but settling down later into the 2-node or fundamental mode of vibration which then persists for some time. This phenomenon has been ascribed to the hull-damping characteristic which shows an approximation to constant logarithmic decrements for the several modes excited. Under such circumstances the modes of higher frequency will decay at a faster rate in time than the fundamental mode. In an anchor-drop test only vertical flexural modes are usually involved and coupling with torsion of the hull is not to be expected unless an unusual bow flare is present.

When it comes to coupled torsion-bending hull modes the situation is somewhat different. If a single mode of this type were excited by an impulse, a pattern that changed as the vibration decayed would not be expected. On the contrary both the angular and rectilinear components of the displacement would decay at the same rate. Moreover, if this mode involved one node in torsion and three in flexure there would be no tendency for conversion to a 2-node mode at all. The latter should be found only if actually excited by the initial impulse. In the latter case, that is, if both the torsion-bending mode and the 2-node horizontal flexural mode were both excited, the

former (having a higher frequency) could decay at a faster rate in time. Hence, in this case the 2-node flexural vibration could persist alone after a sufficient lapse of time. According to the author's Fig. 6 the excitation involved only a vertical impulse. However, while the calculations for *Gopher Mariner* indicated negligible torsion in the mode that had two nodes in horizontal flexure, there may have been sufficient torsion in this mode in the model to cause its excitation by a vertical impulse on the fin.

Another point made by the author warrants some consideration here. He indicates the desirability of keeping the natural frequencies of the torsional and horizontal flexural modes of the hull widely separated. There are two distinct points to be noted here. When torsion and flexure are coupled in the hull due to eccentricity either of the center of mass or of the center of shear with respect to the longitudinal axis of the hull, then neither pure torsional modes nor pure flexural modes are to be expected. In other words only modes of combined flexure and torsion would exist and only one frequency would be assigned to each mode of such a type. When such eccentricities do not exist, torsion and flexure are uncoupled and then pure torsional and flexural modes should be found. Their frequencies are then independent of one another and if they happen to be the same in such a case, it is a mere coincidence. When the coupling is negligible, if the frequencies are near one another, since both torsion and flexure involve horizontal displacements at the main deck in the bow, a beat in the signal from a horizontal vibration pickup located there should be observed. In neither case, however, should there be any tendency for either a one-node torsional mode or a three-node flexural mode to settle into a two-node flexural mode.

**Dr. G. P. Weinblum, Member:** To the writer's knowledge pitch stabilization by fins was proposed in the early 1930's. Since that time he was a protagonist of this bold idea although he was fully aware of the difficulties caused by the tremendous forces needed and the possible dangers due to slamming. The practical application of the simpler proposal—fixed fins—shows once more how far the way is from intuition to actual realization. Some 15 years ago experiments made by Perelmut (USSR) on the effects of horizontal plates at the bottom of a model proved good damping qualities of the arrangement. Later systematic investigations were started by Professor Abkowitz and his school; from these and still more from the author's experiments.

The writer had much to learn about the mecha-

nism of occurrence of troublesome vibrations and of the character of these vibrations. The facts contradicted his preconceived ideas which centered in slamming as only prime mover and vertical vibrations as detrimental effect. The paper represents a fine piece of work in engineering research. By this I wish to emphasize that fundamental physical effects and the phenomenology of the problem have been classified and suggestions have been made for a further sound development. In the world literature on naval architecture such a thorough experimental investigation as presented by the author occurs rather seldom and can serve as prototype for future work in our field.

The "conclusions" are a highlight of the work. The fact that arbitrary scales have been used for the vibratory displacements indicate already that no final results can be expected from this investigation. We are looking forward to a continuation of the excellent research work.

The paper does not present many weak points for criticism. Considering the importance of the problem and the amount of work involved a dynamically similar model should have been used. Because of the vast amount of information the paper is not too readable. However, it is much more to the point to praise (beside the general merits of the paper) a lot of beautiful details like Fig. 7 showing the vibration pattern produced by an impact force, the numerous sketches devoted to the explanation of the physics of the output, and the section dealing with irregular seaway. Rarely the need for pertinent model investigations has been so clearly demonstrated as by this paper.

**Prof. J. R. Paulling, Jr., Member:** It is of interest to compare the order of magnitude of the impulsive forces and moments induced by fins with the vibratory forces and moments produced at the opposite end of the ship by the propeller. Lewis and Tachmindji,<sup>5</sup> and Stuntz, Pien, Hinterthan, and Ficken<sup>6</sup> have reported results of measurements of propeller-induced forces and moments on ship models. While the exact manner in which these quantities scale to full size is not fully understood at present, one may reasonably assume that forces scale approximately as lengths cubed and moments as lengths to the fourth power. Accordingly, using model data from these sources, it is estimated that the propeller-induced horizontal force is of the order of  $20$  to  $50 \times 10^3$  lb in amplitude while the couple referred to the shaft center line

<sup>5</sup> F. M. Lewis and A. J. Tachmindji, "Propeller Forces Exciting Hull Vibration," *TRANS. SNAME*, vol. 62, 1954.

<sup>6</sup> G. R. Stuntz, P. C. Pien, W. B. Hinterthan, and N. L. Ficken, "Series 60—The Effect of Variations in Afterbody Shape . . ." *TRANS. SNAME*, vol. 62, 1954.

is of the order  $0.5$  to  $1 \times 10^6$  lb-ft for a ship similar to the *Mariner* operating at designed speed.

Scaling the results given by the author in Fig. 11, one obtains a horizontal force on the side of the hull of about  $750 \times 10^3$  lb and a moment applied by the fin of about  $10 \times 10^6$  lb-ft. It appears that the point of application of the resultant horizontal force is roughly the same distance below the shear center of the section in each case. Thus we see that the torsional moment and the horizontal force exerted by the fin are an order of magnitude or more greater than the force and moment exerted by the propeller. Of course, the situations are not completely analogous since the former leads to an essentially transient structural response while in the latter case, because of the periodic nature of the force and moment, a steady-state vibration is produced. Moreover, the designer is able to exert somewhat more control over propeller-induced vibration by a judicious selection of RPM, number of propeller blades, and tip clearances. In the situation presently under discussion, the author has shown in what direction and to what extent the designer may go in selecting the configuration of the fin to minimize the impulsive loading while retaining adequate motion control. Further design refinements must probably be in the direction of improving the ability of the structure to withstand the fin-induced loads. For this purpose it is necessary to obtain more exactly the pressure loading applied to the hull in the vicinity of the fin. Therefore, it would appear highly desirable to continue the experiments using the 22-ft model equipped with a larger number of pressure gages in order to obtain a more complete mapping of the pressure distribution induced by the fin on the hull surface.

A second area in which further research is definitely indicated is the behavior of the fin-equipped ship in oblique seas. The author points out the importance to the phenomenon of the differential in time of collapse of the port and starboard cavities. In the present experiments which were conducted in long-crested head seas, this time differential appears to be a random quantity dependent upon slight and unpredictable departures from perfect symmetry of model and wave at the instant of collapse. Rolling motion of the model in oblique seas would introduce a pronounced dissymmetry and corresponding discrepancy in both time of collapse and magnitude of force port and starboard. Indeed, it appears possible that the time differential may be made so large under such conditions that the augmentation

effect and vibration intensity may be less severe than in the symmetric seaway.

**E. D. Hoyt, Member:** I am impelled to comment on the author's use of the term "coupled vibration" and "mode." It is my understanding that the modes of vibration, or the natural modes or normal modes, are independent motions. They are defined as those motions which can take place in such a way that all parts of the structure move with harmonic motion. The author has shown that for the ship hull, being asymmetrical as it is (or "as symmetrical as it is") apart from the horizontal plane, it is not possible to have a pure horizontal motion or a pure torsional motion. That is to say, no purely horizontal motion could be a normal mode and no purely torsional motion could be a normal mode.

Any normal mode other than the vertical mode must combine horizontal or transverse flexure and torsion, but to speak of coupling between such modes is really not possible; therefore any discussion of the frequencies of horizontal modes and the frequencies of torsional modes and of their near proximity is meaningless.

Now, it is not clear from the discussion whether when approximately equal frequencies are mentioned there are really two normal modes present or whether there is only one normal mode and that different attempts to measure the frequency resulted in different answers. This is something the answer to which would be of great interest.

The author notes that the initial motion appears to be a torsion of the bow of the ship and, of course, it is quite possible for such a motion to result from a combination of two or more normal modes excited by the initial impact. One then has an initial value problem which, as is well known, requires generally the superposition of at least two normal modes of the system, depending on the number of degrees of freedom of the system, in order to satisfy the initial condition.

**P. A. Markussen, Life Member:** From the hydrodynamic point of view the author gives three methods of reducing the vibration imposed by fins. The first is reducing the cavity; the second is the submergence, which also, of course, affects the cavity; and the third is devices by which the impact pressures could be equalized.

In the third category, would not a mechanism whereby the fin angle is altered as a function of the movement of the bow have the same effect as equalizing the pressure and with somewhat greater advantage?

## Author's Closure

Since it is apparent that there are several differences of opinion between Professor Abkowitz, Mr. Pearlman, and the author, it may be well to clarify these issues and the basis of our disagreement.

Professor Abkowitz and Mr. Pearlman mention, on the basis of their experimental results, that the collapse of a ventilated bubble against the bow side caused the vibration and that the torque due to the vertical force on the fin did not contribute to the vibration. It is the author's opinion, however, that the foregoing conclusion is incorrect. In Reference [9],<sup>7</sup> Mr. Pearlman evaluated the horizontal force by the formula which was valid only for the static loading of a cantilever beam, not for the dynamic response of a free beam to transient loads at one end. A similar static analysis was also made to evaluate the strain which could have been produced by torque on the fin. Since the phenomenon which we are discussing is a transient vibration phenomenon, the frequencies involved in the vibration, the time duration of impact force, and the coupling between torsion and bending must be considered for evaluation of the impact force from the measured strains. If these factors are not included, the analysis is erroneous and the results are meaningless. This could be the reason that an unusually large percentage of the bending strain in Mr. Pearlman's analysis was attributed to the horizontal force on the ship's bow side. Therefore, the author cannot concur with the conclusions of Professor Abkowitz and Mr. Pearlman.

The author does not recommend the strain-gage technique which Professor Abkowitz and Mr. Pearlman used in their tests to obtain the vibration characteristics of this torsional-bending phenomenon. While their technique provides a valid qualitative means for detection and determination of frequencies of either horizontal bending or torsion, it can yield no information for identifying the types of the vibration involved. It should be mentioned that without obtaining the solution of the coupled equations for torsional-bending vibration either by the normal mode or by the Laplace transformation method, the measured hull strain would not provide any information concerning the nature or type of coupled vibration which induces the strain, except the frequencies involved. Since solution of the equations involved in the foregoing methods, is rather tedious, the strain-gage

<sup>7</sup> Numbers in brackets designate References at end of closure.

technique was not used in the author's test, but a more direct approach of measuring accelerations to obtain the nature and type of vibrations was chosen instead.

Professor Abkowitz questions the reliability of the instrumentation used in the tests. The instrumentation was specifically selected and carefully checked for proper response to the transient phenomenon observed in the tests. Perhaps his concern with the reliability of the instrumentation arises from the inadequacies of the instrumentation used by Mr. Pearlman. A Sanborn recorder which has a frequency response of less than 100 cps, is certainly not a good device for investigation of the details of the time history of an impact. Details of the vibration characteristics must be studied by *simultaneous* recording of hull vibration and sources of the vibration on an oscillograph or oscilloscope either of which responds to much higher frequencies. Further, measurements of hull stresses remote from the phenomenon being studied are not likely to be reliable for precise interpretation.

Professor Abkowitz and Mr. Pearlman attribute the cause of vibration to collapse of a ventilated bubble against the bow side, and in connection with this Professor Abkowitz discusses the concept of cavitation number. If their definition of the ventilated bubble is the so-called "air-filled cavity," the collapse of the bubble may certainly take part in the generation of the vibration. However, the concept of cavitation number is not appropriate in this case. Although the author mentions briefly in the paper that collapse of the cavity is one of the causes of the vibration, it may be well to give a detailed discussion here concerning the cavity formation, its collapse and the generation of the vibration.

A close examination of the phenomenon has revealed that it is a typical example of the air-water entry phenomenon. Even when the fin approaches very near to the water surface but does not emerge, the following remarks concerning air-water entry phenomena are still applicable. There are two types of cavity formation in the air-water entry phenomenon depending on the magnitude of entrance velocity [25, 26, 27]. In the case of a low entrance velocity, the penetration of a body in the water leaves an air column behind the body, which later contracts and closes with a sharp impact at some point between the body and the water surface, thus separating into two parts or cavities. The upper cavity is open to the atmosphere and rapidly collapses due



to the influx of water from all sides. The lower cavity contains air, and the cavity pressure differs only slightly from the atmospheric pressure. This air-filled cavity also collapses by the influx of water and the so-called "reentrant jet" phenomenon results. A jet is also directed upward from the point of closure, and is clearly visible in photographs taken from above the surface.

On the other hand, in the case of a high entrance velocity, the foregoing phenomenon is preceded by sealing at the surface (so-called surface seal). Necking then results in a "vapor-filled cavity." A significant difference between these two cases is that the former is not sensitive to the cavitation number, while the latter is definitely subject to the cavitation number. The Froude number may therefore be used for scaling of the air-water entry phenomenon with a low entrance velocity, particularly for blunt bodies. Now, the cavity formation associated with an antipitching fin certainly falls into the low entrance-velocity category. Photographs shown in Fig. 12 of the paper explain the formation and collapse of the upper cavity. Mr. Pearlman's excellent underwater photographs [9] show the necking phenomenon and collapse of the air-filled cavity. The closure and subsequent collapse of the upper cavity give an impact pressure to the bow side. The collapse of the air-filled cavity provides an impact pressure to both the fin and bow side; however, the greater percentage of the pressure may be applied to the fin since a water jet appears at the instant this cavity collapses. This sequence of events appears to explain why the peak pressure appears first on the bow side and then a very short time later the peak pressure appears on the fin as shown in Fig. 11. In any case, it may be mentioned again that the concept of cavitation number is not appropriate for this phenomenon.

Professor Abkowitz suggests that an elaboration on the discussion of speed effect on vibration may be in order. He is correct that the effect of speed arises from its relation to resonant conditions for motion. As seen in Fig. 18, which shows a typical example of speed effect, the vibration becomes severest near the ship speed for which pitching becomes maximum. In addition to the tests on the 5.5-ft model, tests on a 22-ft model were conducted in a series of wave lengths. It was found, of course, that the resonant speed for pitching was dependent on wave length but the general tendency as established in wave lengths equal to ship length was maintained throughout.

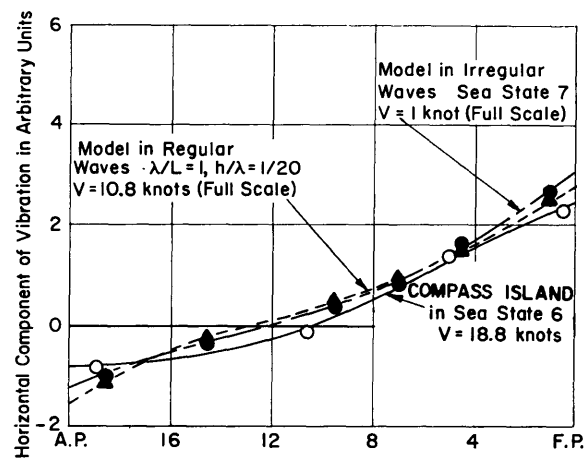


Fig. 64 Comparison of vibration pattern at instant deflection occurs measured on *Compass Island* and model

Professor Abkowitz questions the basis on which the author concludes that the vibration starts essentially as a torsional vibration rather than a horizontal flexural vibration as previous investigators had believed it to be. As is clearly mentioned in the paper, the impact force on the fin certainly produces a moment about the center of rotation of the section. Furthermore, the horizontal impact force on the bow side also results in a moment about the center of rotation, since the location of this lateral force is far below the center of rotation of the hull. Now, the resultant moment causes both torsional and horizontal flexural deflection. However, the torsional vibration is more pronounced at the initial stage than any of the modes of horizontal vibration. This has been verified by results from full-scale trials as well as model tests. For example, Fig. 64 of this closure shows a comparison of the vibration patterns obtained from the full-scale trials on the *Compass Island*, and the model tests in regular and irregular waves. These patterns were obtained at the instant the vibration appeared. The scale for the vibration acceleration is arbitrarily selected for all these curves since the purpose is not to compare the intensities of the vibration but to obtain the general character of the vibration patterns only. In the case of the models, the initial lateral deflection at the deck is known to be opposite to the direction of the applied force. Fig. 65 shows the initial deflections at the deck and at the keel. It can be seen in the figures that torsional vibration is more pronounced at the initial stage of the induced vibration. Of course, components of various modes of pure horizontal flexural

vibration are also included at this stage, yet the torsional vibration is far greater than any of the other modes of vibration. It was on this basis that the author introduced the concept of torsional-horizontal coupled vibration while earlier investigators had considered only pure horizontal flexural vibrations. However, it is emphasized here that the concept does not imply that the only or even predominant cause is loading on the fin. In fact, loads upon the bow above the fin are significant as is clearly mentioned in the paper.

An interesting discussion was brought up concerning whether or not ship slamming should be taken into consideration in the present problem. Mrs. Bledsoe suggests that slamming must be considered in a practical case, while Mr. Pearlman concludes it is not a problem. The author believes that Mr. Pearlman has derived his conclusion on the basis of his experimental results obtained with a fin area equal to 5 per cent of the water-plane area, and at low ship speeds which were not sufficiently severe for slamming [9]. If such a large fin is permissible in practical application, certainly the probability of occurrence of slamming is slim, since a large reduction in pitch (more than 40 per cent) can be expected. However, such a large fin will probably not be practically used. The results on the effect of fin size suggests that the force acting on the fin becomes very large as the fin area is increased. The fin installed on the *Compass Island* was only 1.6 per cent of the area of the water plane; nevertheless, the fin structure was designed for an equivalent static load of 3000 tons. The price which must be paid for fins must be carefully considered as Dr. Cummins pointed out at the annual meeting of the Society in 1959. Also, a fin of unusually large area will increase tremendously the resistance and will affect the maneuverability of the ship. No shipowner would be willing to accept these defects. Thus, a certain limitation in fin size is required. Then, we must reluctantly admit the possibility of the occurrence of slamming for a ship equipped with a practically usable fin. Therefore, the author must agree with Mrs. Bledsoe's opinion that slamming cannot be neglected in the design of a fin.

In connection with the magnitude of impact pressure, Mrs. Bledsoe asks whether or not the relative importance of the two types of loading in inducing the vibration can be estimated; the two types of loading being the vertical impact on the fin and the horizontal impact on the bow side. To answer this question, it may be well

to show a practical example. The relative magnitude of impact force on the bow side and fin can be obtained from Fig. 11. Since these forces vary with time, the impulse must be evaluated by taking the time history of these forces into consideration. This can be done from Fig. 11 also. Then, multiplying by the lever arms about the center of rotation, the ratio of the exciting torsional impulse on the fin and on bow side becomes 1 to 3.4 on either the port or starboard side. In other words, the impact force on the fin contributes approximately 30 per cent to the induced vibration, and 70 per cent of the induced vibration is generated by the impact force on the bow side. Obviously, the impact on the bow side is the more significant contributor to the induced vibration, but the 30 per cent contributed by the impact on the fin should not be neglected. This example is the case when slamming is not involved. Should slamming occur, the fin becomes a 100 per cent contributor to the induced vibration.

Professor Lewis suggests that the resistance increased by the presence of the fin should be avoided in the actual installation of fins on ships. Certainly, the author agrees with his opinion. Although an increase in resistance at low speeds is not a serious problem, a 10 to 15 per cent increase in resistance at operating ship speeds in waves must be reduced. Reduction of the increased resistance attributable to the presence of the fin may be obtained more easily than for reducing the vibrations. As mentioned in the paper, a properly selected sectional profile of the fin and the angle at which it is installed may lessen the problem.

Professor Lewis asks whether or not the qualitative vibration characteristics of the model which are derived from the test results are applicable to a full-scale ship. To answer this question, it may be well to refer to Fig. 64. Even though the curves shown in the figure were obtained in different conditions, it can easily be seen that the vibration pattern obtained on the model is in good agreement with that obtained on the full-scale ship. Therefore, it may be safely said that the model results are qualitatively representative of those of the full scale.

The author would like to thank Dr. Weinblum and Mr. Taplin for their general comments. They mention that, considering the importance of the problem, a dynamically similar model should have been used in the tests. It is, of course, most desirable in hull-vibration tests in waves to employ a dynamically similar model.

This was stated in the paper. However, the following factors make the construction of the dynamically similar model difficult: (1) For phenomena of this type, the model must have the frequencies and damping characteristics scaled for all modes of vibration, since various modes of vibration are involved. To fulfill this requirement in practice is very difficult indeed. (2) A segmented model cannot conveniently be used, since it would not provide the accurate vibration pattern, unless the model is segmented in many parts.

On the other hand, the main purpose of this study was to clarify the nature and fundamental properties of the vibrations induced by an anti-pitching fin so that intelligent measures may be taken to reduce or eliminate them. When this has been done, use of the fin in reducing pitching motion may again be attempted. Therefore, for purpose of clarifying the basic nature of the vibrations, tests on a nondynamically similar model are still considered useful. The one disadvantage involved in this approach is that the intensity of the vibration on the full-scale ship cannot be estimated directly from the model test results. However, if some data on full-scale ships involving simultaneous measurements of the intensity of the vibrations and the magnitude of the impact pressure on the fin are or become available, the model test results given in this paper can be directly converted to those for the full-scale ship. This can be done, since a linear relationship between the intensity of the induced vibration and the impact pressure on the fin could be established irrespective of ship speed, fin size, and irregularity of sea as is shown in Fig. 29, 41 and 55.

Mr. Henry's discussion concerns the definition of the term "hydroelasticity." In reply to his remarks, it may be well to quote the phrase given in Mr. McGoldrick's paper on ship vibration [28]; that is, "Although various definitions of this term will be found in the literature, it seems sufficient to state that hydroelasticity is concerned with those problems in which water vehicles are subject to time-varying forces imposed by the water, but governed also by the elastic properties of the hull or its appendages."

Mr. Seller's opinion from the ship designer's point of view are greatly appreciated. He questions why the design force for the fin given in Table 10 is far less than the reported design forces for the *Compass Island*. It should be mentioned that the magnitude of the force given in this table is just the force required for

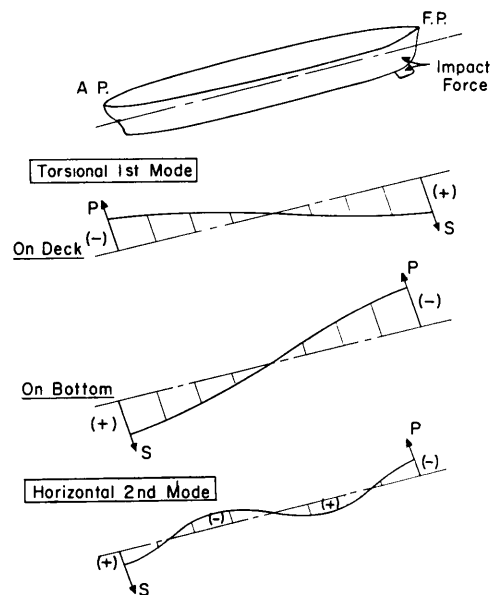


Fig. 65 Explanatory figure of phase relation between torsional and horizontal vibration at instant deflection occurs

reducing the pitching motion, and it does not include the magnitude of the impact force due to cavity collapse or slamming. Of course, the latter forces should be taken into consideration in the design of a fin. The *Compass Island* fin was designed to withstand slamming forces. Indeed, the *Compass Island* did slam in rough weather and the results showed that the design force was of proper magnitude.

An interesting discussion concerning the comparison of forces and moments produced by the fin and by the propeller was given by Professor Paulling. It is significant that the force and moment induced by the fin are an order of magnitude or more greater than the force and moment exerted by the propeller. In the case of propeller-excited vibrations, hull resonance occurs if the hull frequencies fall in the range of the operating blade frequency, and resonance is usually associated with higher modes of hull vibration. While, for the vibration induced by the fin, there exists no specific frequency corresponding to the blade frequency; instead, the time duration of loading and time differential of loading on both sides of the fin and/or bow side are the important parameters which affect the intensity of the vibration as is discussed in the paper.

Mr. McGoldrick discusses the phase relationship between torsion and bending modes. He states that generally there would be two pairs of coupled torsion-bending modes, each pair con-

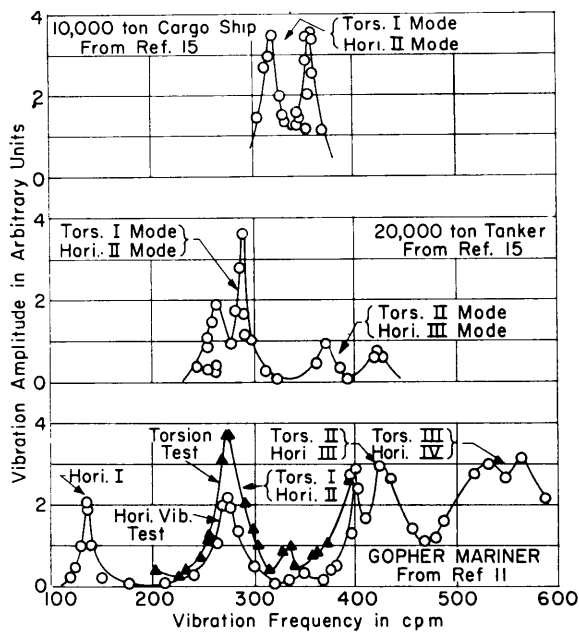


Fig. 66 Amplitude of horizontal component of vibration versus frequency measured on full-scale ships (from References [11] and [15])

sisting of the same modes in both torsion and horizontal flexural vibration but with reverse phase relations. This is a very interesting subject to consider, since the test results suggest that there is only one coupled torsion-bending mode for a ship equipped with an antipitching fin.

Fig. 65 shows a pictorial presentation of the phase relation between torsional and horizontal vibrations and was taken from Fig. 10 of the paper. The figure is for the case in which an impact force is applied to the starboard side first. In this case, the deck at the ship bow deflects to starboard due to the torsional vibration and deflects to port due to the horizontal flexural vibrations. However, the torsional components are much more predominant than the flexural component. This phase relation was found for all cycles of encounter whenever an impact force is applied to the starboard side first. The reverse phase relation is, of course, established for impact on the port side first. An interesting conclusion derived from this figure is that the combined vibration component at the forefoot and at the base of the stern is more severe than the combined vibration component at deck level for a ship equipped with an antipitching fin.

An important subject was discussed by Mr. McGoldrick, Mr. Taplin, and Mr. Sellers concerning the desirability of keeping the natural frequency of the torsional and horizontal flexural

modes of the hull widely separated. Also, Mr. Hoyt questions the number of normal modes involved in the phenomenon. The author would like to answer these questions referring to a practical example. As was discussed by Mr. McGoldrick, if torsion and flexure are coupled then only one frequency would be assigned to each mode of coupled vibration from a theoretical consideration. However, in a practical case, peaks at two discrete frequencies, not just one, are usually observed in the vibration tests on the full-scale ship.

Fig. 66 shows three examples of vibration test results obtained from full-scale trials. It should be noted that for all three ships, the independently calculated natural frequency of torsional vibration was very close to that of the horizontal flexural vibration as is usually expected. It can be seen in the figure that each mode of torsional vibration forms a pair with the next higher mode of flexural vibration and that there are two peaks for each combination. Only the 1st mode of horizontal vibration does not pair with any mode of torsional vibration since its natural frequency is not close to the frequency of any of the modes of torsional vibration. The nature of this interesting phenomenon has not been completely clarified as yet. In the case of the vibration induced by an antipitching fin, only one combination of torsional (1st mode) and horizontal (2nd mode) vibration appears. This has been verified from the results of full-scale trials. Although the fundamental mode of horizontal flexural vibration is also included from the beginning, the torsional vibration is predominant at the initial stage as is shown in Fig. 64. Now, it is of interest to point out that the frequency bands where two peaks appear are wider than the frequency bands for a single peak (1st mode horizontal vibration). Also, the amplitude of the coupled vibrations is larger than the amplitude for a single vibration. These are the features of coupled vibration when the natural frequencies of torsional and horizontal vibration are very close. If the two frequencies are widely separated, these features would not appear. This is the reason why the author mentions in the paper that separation of the frequencies is desirable although it may be difficult to achieve in practice.

Mr. Markussen asks whether or not an activated fin gives the same effect as equalizing the pressure on both sides of bow and/or fin. An activated bow fin may be more effective in reducing pitching motion if it is controlled so that its motion leads the bow motion by 90 deg. However, the activated fin probably would not equalize the pressure on both sides of the bow and/or

fin but would be helpful to maintain a deeper submergence of the fin. Additional problems such as those of power and control will be involved for an activated fin.

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