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Selected Elementary Criteria for Evaluating Propeller-Induced Surface Force Excitation [and Discussion]

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Phil. Trans. R. Soc. Lond. A 1991 **334**, 319-337
doi: 10.1098/rsta.1991.0017

Selected elementary criteria for evaluating propeller-induced surface force excitation

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This paper is a review and assessment of existing simple prescriptions that can be used to evaluate hull geometries for limiting problems with propeller-induced hull surface force excitation. A general description is provided for the categories of excitation and of the pertinent propeller–hull–wake parameters that are involved. Available elementary criteria and estimation formulas useful for making preliminary judgments on the acceptability of a given arrangement are discussed in some detail. Critiques of the applicability of these criteria are offered in light of their use with example ships, and their usefulness in implementing early stage design for vibration avoidance.

Introduction

Over the past 20 years there has been increased concern about problems of propeller-induced ship hull vibration and noise; and these include nuisance for the crew, structural damage due to fatigue or propeller blade erosion that sometimes accompanies these difficulties, troublesome environment for machinery, and reduced allowable r.p.m. and thus reduced operational speed. Useful predictive schemes for judging propeller–hull geometries are needed for both early preliminary design evaluation and also for later, more thorough studies, with detailed structural arrangements.

Problems associated with propeller-induced ship vibrations have become especially vexing on certain types of mainly single-screw merchant ships. Restrictive demands on the stern geometry or propeller–stern arrangements have led to hull geometries that produce heavy and steep wake velocity distributions. Ships with these wakes have also been equipped with increasing amounts of horsepower-per-shaft, although greater power alone does not necessarily bring on increased danger of vibration difficulties. The types of ships usually affected are tankers, roll-on–roll-off (RO–RO) ships, container ships, and other bulk and product carriers. A useful outline of the broad scope of ship vibration response characteristics and the types of magnitudes of typical vibration problem areas is presented in the paper by Johannessen & Skaar (1980). A monumental collection of work on the character of ship vibration, outline of various sources of excitation, and engineering guidance for the avoidance of problems is presented in the Bureau Veritas monograph (1987).

From experience with excessive vibrations on many ships, there are published elementary criteria for estimating and/or evaluating limiting levels of propeller excitation. This paper is concerned mainly with a review of these criteria.

Phil. Trans. R. Soc. Lond. A (1991) **334**, 319–337
Printed in Great Britain

Components of propeller-excited vibration

The operation of a marine propeller in a non-uniform flow in the proximity of the hull surface gives rise to oscillating forces and moments that act to excite vertical, horizontal, torsional, and longitudinal vibrations of the hull and the propulsion machinery. Propeller-induced periodic forces occur at simple harmonic components of the blade passing frequency, that is, at the blade rate and integer multiples of the blade rate.

There are two general categories of propeller exciting forces and moments that cause hull vibration. (1) Bearing or shafting forces and moments comprise transverse forces and moments (vertical and athwartships) and the fluctuating thrust and torque that are developed on the propeller as it operates in a non-uniform wake. The latter are transmitted to the hull through the propulsion shafting. (2) Surface forces are distributed oscillating loads associated with the propeller-induced fluctuating pressure and velocity fields. They act on the hull surface in a sometimes wide vicinity of the propeller's location. Blade-fanning surface forces occur even without a non-uniform propeller inflow, but they can be greatly accentuated by a wake and especially by unsteady cavitation. This review is concerned mainly with the preliminary estimation of the surface forces, or more typically the estimation of some indicator of their magnitude.

It appears that the propeller-induced surface forces are the most sensitive to the propeller–hull arrangement (tip clearances vertically and longitudinally) of all the exciting force components identified. Once cavitation appears, the surface forces become dominant and especially so at the very highest speeds.

Surface forces are the spatially integrated unsteady loads associated with the travelling unsteady pressure footprint created by the passing blades. Under cavitating conditions the fluctuating sheet cavity volumes on the blades are efficient producers of relatively large pressure amplitudes, especially during the phase of cavity collapse. More importantly, the phase angles associated with the cavity collapse-induced pressure peaks are spatially rather constant over a wide vicinity of the propeller, longitudinally and laterally. This tends to produce a large spatially coordinated, integrated surface force, which for most ship afterbodies has a big vertical component.

There is an additional important effect of steep circumferential wake gradient, such as occurs behind a single-screw ship. The violence of blade cavity collapses can be markedly aggravated by rapid local changes of flow angle of attack on blade sections, which in turn can magnify further the magnitude of the unsteady pressure pulses produced. Thus the combination of heavy blade sheet cavitation and steep wake velocity gradients produce the worst potential for excessive surface force excitation.

In the few instances where there have been estimates made of the blade rate force magnitudes of both the vertical propeller-induced surface force $(F_z)_S$ and the vertical bearing force $(F_z)_B$, the surface force is clearly the larger, whenever intermittent cavitation effects are accounted for. For example, for the case of the single-screw lake freighter with a conventional four-bladed propeller discussed by Reed *et al.* (1981) the estimated ratio of blade rate vertical bearing force-to-surface force was found to be $(F_z)_B/(F_z)_S = 0.12$ at the full power condition. From the group of calculated results for 17 example merchant ships given by Skaar & Raestad (1979) the predicted ratio for $(F_z)_B/(F_z)_S$ using computation schemes developed at Det norske Veritas (DnV) come out in the range 0.0002 to 0.39, with an average of 0.08.

Since the onset of some amount of cavitation near the tips or leading edge of propeller blades is the rule rather than the exception, it is fair to conclude that the surface force excitation will always be a major factor of the propeller-excitation of ship vibration.

Review of available criteria

This section is a summary of the available practical criteria that may be used to assess the acceptability of propeller–hull arrangements from an excitation point of view. Practical criteria are rules of thumb or sometimes self-contained recipes for making judgments at the preliminary design stage. The Report of Propeller Committee for the 15th International Towing Tank Conference (ITTC) of 1978 and that of the 16th ITTC of 1981 have provided a useful core of information for this review. The simple criteria are organized here into four categories: pressure amplitude limits, surface force amplitude limits, wake quality and cavitation factors, and vibration level limits. These prescriptions can be divided into two types: (*a*) criteria for evaluation or judging whether a troublesome condition could exist and (*b*) techniques for estimating either the magnitude of certain point pressure pulse amplitudes or surface force amplitudes.

Pressure amplitude limits

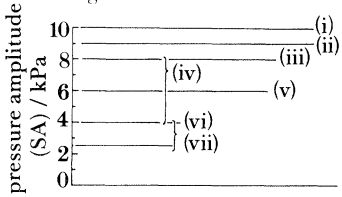
From experience with model experiments and observations full scale there is a body of information about the magnitudes and distribution of surface pressure amplitudes. These can be grossly correlated with the vibration tendency of the ship involved.

1. The simplest prescription is a single-point specification of acceptable limiting blade rate pressure amplitude on the hull directly above or very near the propeller. A sampling of recommended limiting pressure amplitude and the source author include 9 kPa by Huse (1972); 10 kPa by Okamoto (ITTC 1978); 4–8 kPa for merchant ships, and 2.5–4 kPa for Navy ships by Weitendorf (ITTC 1978); 6 kPa for ships with flat counters, and 8 kPa for conventional afterbody ships by Wills *et al.* (1979); 8 kPa by Raestad (ITTC 1978); 4 kPa by Suhrbier (ITTC 1978); and qualified ranges of sensible values from 8 to 12 kPa outlined by Volcy (ITTC 1978). Figure 1 is a summary graph of various suggested pressure level limits given as single values from various sources.

2. From SSPA Johnsson (1975), Lindgren & Johnsson (1977) and Johnsson (1983) have established an empirical formula for the allowable hull surface pressure above the propeller tip, based on a limiting value of the representative blade rate vertical vibration velocity of 5 mm s^{-1} r.m.s. at the fantail. This applies to single- and twin-screw merchant ship arrangements, and is used in conjunction with figure 2. In its most recent form, it is a dimensional formula:

$$2(\Delta p)_{\text{allow}} = (6.25)\,(\nabla/N_{\text{p}}/D^2K_{\text{L}}K_{\text{B}})\,(a_x/a_z), \tag{1}$$

where $(\Delta p)_{\text{allow}}$ is the allowable blade rate pressure amplitude (Pa); ∇ is the volume of displacement (m^3); N_{p} is the number of propellers; D is the propeller diameter (m); L is the ship length (m); a_x is the horizontal clearance from propeller blade mid chord, measured forward to point on hull $0.8R$ above the axis (m); a_z is the vertical tip clearance from propeller to hull (m); K_{L} is the length correction factor ($L/140$ for figure 1



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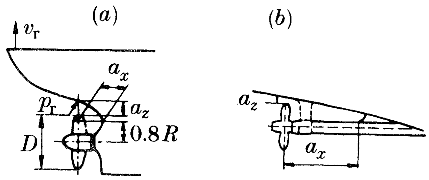

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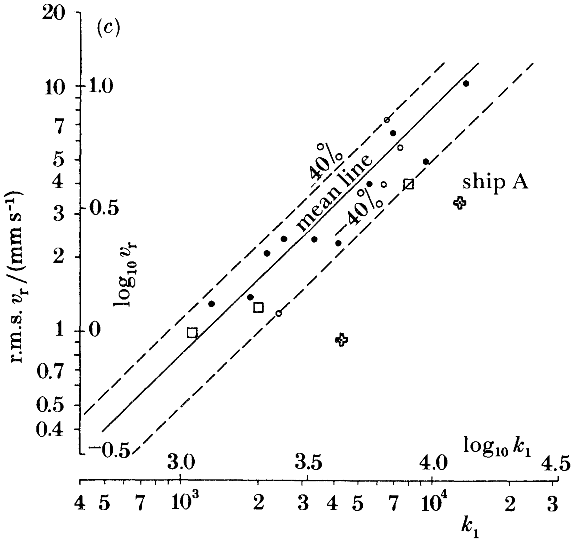

Figure 1. Summary of recommended limits for blade rate pressure amplitudes induced on the hull above propeller tip. (i) Okamoto (1978); (ii) Huse (1972); (iii) Lloyds Register (1978), NSMB (1978), Wills *et al.* (1979), Ward (1983), Raestad (1978); (iv) Weitendorf (1978) (Merchant ships); Wills *et al.* (1979) (for ships with flat counters); (vi) Surbier (1978); (vii) Weitendorf (1978) (Navy ships).

Figure 2. The SSPA criterion for induced blade rate pressure amplitude–vertical stern vibration. (From Lindgren & Bjärne 1980, Johnsson 1983.) SSPA data: ●, fully loaded single screw; ○, ballast single screw; □, conventional twin screw. $k_1 = 10^3(2p_r)D^2(N_p/\nabla)(a_z/a_x)K_L K_B$. (*a*) Clearance for single-screw ships; (*b*) clearance for twin-screw ships.

$L \leqslant 140$ m, 1.0 for $L \geqslant 140$ m; K_B is the proximity to free surface factor (0.03 $(10^3 b_z/L)+0.65$ for $10^3 b_z/L \leqslant 11.7$, 1.0 for $10^3 b_z/L \geqslant 11.7$; b_z is the submergence depth of pressure gauge location or pressure calculation point.

The SSPA criterion is presented in figure 2, modified from Lindgren & Bjärne (1980). It shows a dimensional correlation of the representative vertical vibration velocity of the ship hull as a function of the pressure amplitude on the hull above the propeller tip in terms of $\log_{10} v_r$ plotted against a pressure amplitude factor k_1. Here, the vibration velocity v_r refers to the hull girder vibration velocity (mm^{-1} s) (r.m.s.) at the extreme end of the main deck. The value of peak-to-peak blade rate pressure fluctuation on the hull above the propeller tip, denoted in figure 2 as $2p_r$, must be determined from experiment or by estimation methods.

Actually the correlation shown in figure 2 can be used two ways: either to check the expected level of hull girder vibration given the value of blade rate pressure fluctuation; or inversely, with the specification of some limiting level of hull vibration, the allowable magnitude of hull surface pressure fluctuation above the propeller tip can be determined.

3. Björheden (1979) has offered an empirical criterion for the recommended allowable value of pressure amplitude, based on limiting the vertical girder vibration level at the stern to 4 mm s^{-1} r.m.s. The formula provides a crude account of the hull girder bending vibration response, and thus depends on the hull girder depth D_H and ship displacement.

4. For DnV, Holden (1979) and Holden *et al.* (1980) have assembled a large amount of full-scale data covering the hull surface pressure amplitudes, hull girder vibration, powering performance, and propeller–hull geometry for 72 merchant ships (container,

RO–RO, tanker ships) and has used a regression analysis approach to obtain empirical estimating formulas for determining induced surface pressures based on what has emerged as the most significant parameters.

(*a*) The blade rate pressure amplitude induced by a non-cavitating propeller was found to be mainly dependent upon blade thickness,

$$\Delta p_0 = (0.01245)\,\rho n^2 D^2((t_\mathrm{m}/D)^{1.33}/Z^{1.53})\,(d/R)^{-\kappa}, \tag{2}$$

where Δp_0 is the pressure amplitude (kPa); t_m/D is the blade thickness-to-diameter ratio at $0.7R$; Z is the number of blades; d is the distance from the $r/R = 0.9$ position to the fieldpoint on the hull; R is the propeller radius;

$$\kappa = \begin{cases} 1.8+(0.4)d/R, & (d/R) \leqslant 2, \\ 2.8, & (d/R) > 2. \end{cases}$$

(*b*) For a cavitating propeller, the blade rate pressure amplitude may be greatly amplified and can be estimated from

$$\Delta p_{Z_\mathrm{c}} = (9.8\times 10^{-5})\,\rho n^2 D^2 (J_1 - J_\mathrm{M})\,(f_2/\surd\sigma)\,(d/R)^{-\kappa_1}, \tag{3}$$

where Δp_{Z_c} = pressure amplitude (kPa); $(J_1 - J_\mathrm{M})$ is the change in advance ratio with respect to the minimum inflow velocity to propeller (see Holden 1979); f_2 is the blade tip loading parameter; σ is the cavitation number based on peripheral speed at $r/R = 0.7$;

$$\kappa_1 = \begin{cases} 1.7-(0.7)\,(d/R), & d/R \leqslant 1, \\ 1.0, & d/R > 1. \end{cases}$$

The resultant pressure fluctuation amplitude at blade rate acting on the local hull surface near the propeller depends in detail upon the phase angles of the non-cavitating and cavitating propeller contributions (see Holden *et al.* 1980); but a rough estimate of the blade rate pressure excitation amplitude is given by

$$\Delta p_Z = \surd(\Delta p_0^2 + \Delta p_{Z_\mathrm{c}}^2). \tag{4}$$

Most often the Δp_0 contribution is so small compared with the cavitating propeller part that it is negligible and Δp_{Z_c} alone is a good approximation to the pressure fluctuation excitation level.

(*c*) For a cavitating propeller, Holden *et al.* (1980) has presented an estimating formula for fluctuating pressure at twice blade rate, Δp_{2Z_c} in a form similar to equation (3).

These formulas serve as estimators for the expected levels of point pressure fluctuations produced by an operating propeller. Some idea of the accuracy associated with their use is discussed by Holden (1979) and Holden *et al.* (1980) using the statistical properties of the estimated values of Δp_Z and Δp_{2Z_c} compared with the full-scale measurements.

DnV has expressed its current recommendation or criterion for the acceptable pressure level from the point of view of fatigue cracking in the afterpeak area in the references by Holden *et al.* (1980) and Johannessen & Skaar (1980). The allowable or limit values for peak fluctuating pressures are $\Delta p_\mathrm{tot} < 16$ kPa for all frequencies (total amplitude) and $(\Delta p)_Z < 8$ kPa for the blade rate component.

Surface force amplitude limits

Similar to the efforts with surface pressure estimates, there have been several attempts to provide guidance for checking the limiting surface force induced by a propeller, usually based on an estimate of the fluctuating vertical force determined over a reference area of the hull surface.

1. For SRI, Takahashi (1976, 1975) has developed an estimate for the cavitating propeller-induced vertical fluctuating force amplitude $(F_z)_S$ acting on a flat surface of dimensions $D \times D$ centred above the propeller location. The formula takes the form of a dimensional ratio (t m^{-2}):

$$(F_z)_S/D^2 = (0.358)\, K^0_{P(M)}\, K_A\, K_C\, P/ND^3, \tag{5}$$

where $K^0_{P(M)}$ is the maximum value of K_P directly over the tip, for non-cavitating propeller; $K_P = 2\Delta p/K_Q \rho n^2 D^2$ is the pressure fluctuation-to-torque coefficient ratio for double amplitude $2\Delta p$ at blade rate; K_Q is the torque coefficient $(= Q/\rho n^2 D^5)$; K_C is a factor dependent on the distribution of induced surface pressure and phase angle (Takahashi 1976); P is the metric horsepower per shaft; N is the propeller r.p.m.; D is the propeller diameter (m); K_A is a factor for magnification effects of wake non-uniformity and cavitation number upon the unsteady pressure (Takahashi 1976).

Application of this approach is based on comparison of results from the formulas and observations of full-scale ship vibration performance. Takahashi (ITTC 1978) has suggested an evaluation criterion based on vertical force amplitude limit of

$$(F_z)_S/D^2 = 6\text{–}7\ \text{kN m}^{-2} \quad (0.61\text{–}0.714\ \text{t m}^{-2}). \tag{6}$$

2. Mano *et al.* (1978) have provided a formula for estimating the magnitude of the blade rate vertical surface force produced by a cavitating propeller, on a patch of hull surface $D \times D$. Estimates from the formula, together with measurements of vibration levels on many ships have been used to produce the limiting surface force diagrams such as that given in figure 3 for ships with block coefficient $C_B < 0.65$. These results indicate that for a hull shape of a given block coefficient, a larger displacement (and thus the length and other dimensions) coincided with a smaller acceptable $(F_z)_S/D^2$ ratio.

3. Yamaguchi (1977) has also provided an estimating formula for the blade rate vertical induced force amplitude on a flat surface over the propeller based on a square integration area of length D on each side. Correlations of estimates from this formula have been made with full-scale shipboard measurements of vibration to produce the diagram given in figure 4 for $C_B < 0.65$.

Wake quality and cavitation factors

Other types of empirical vibration criteria have been explored that never deal directly with estimates of either fluctuating pressure amplitudes or with reference area force amplitudes, but rather are concerned with overall properties of the wake, average cavitation numbers and propeller loading.

1. van Gunsteren & Pronk (1973) have tried to delineate limit regions of safe or troublesome operation with regard to ship vibration in terms of cavitation inception using plots of two different cavitation numbers against thrust loading coefficient. In the plotting of the database points, the only problem cases of propeller-excited vibration are single-screw ships and it is difficult to understand the relationship between the contours of onset of cavitation and the boundaries of vibration trouble for single- and twin-screw ships.

2. Fitzsimmons (1977) has suggested an approach based on a wake factor and a cavitation number, with limit regions determined empirically from the plotting of data for both satisfactory and unacceptable ships from the vibration point of view. The two parameters used are

$$w_\Delta = \Delta w/(1-\bar{w}), \tag{7}$$

$$\sigma_{nt} = 2(p_{\text{atm}} + \rho g h_t - p_v)/\rho(\pi n D)^2, \tag{8}$$

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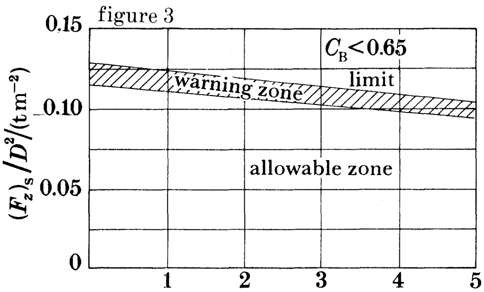


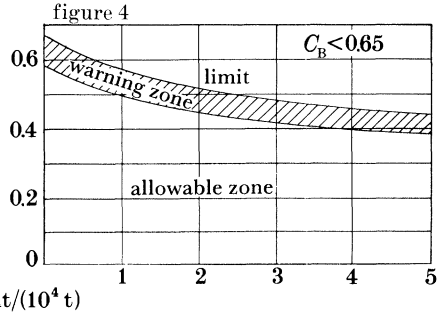
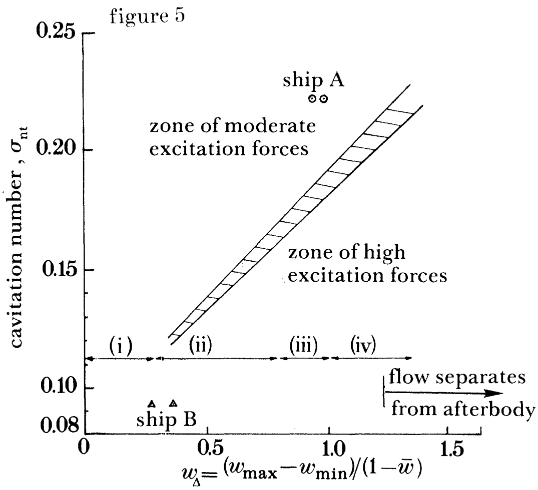


Figure 3. Limits of acceptable reference area surface force for $C_B < 0.65$, for use with estimate by Mano *et al.* (1978).

Figure 4. Limits of acceptable reference area surface force for $C_B < 0.65$, for use with estimate by Yamaguchi (1977).



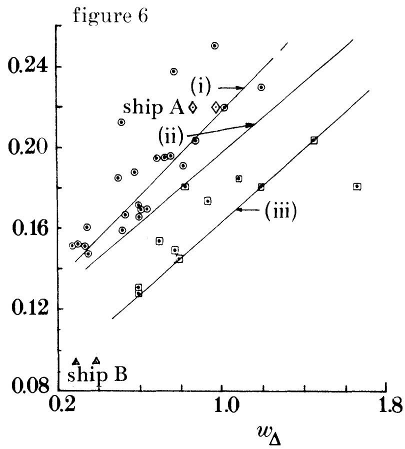


Figure 5. Fitzsimmons cavitation criterion for wake non-uniformity. (From Fitzsimmons 1977.) (i) Twin-screw form; (ii) bulb-extreme U form; (iii) moderate U/V form; (iv) extreme V form.

Figure 6. Summary of predictions of propeller-induced vibration problems based on model wakes (Fitzsimmons Plot). (From Rutherford 1979.) (i) Low levels of vibration from cavitation; (ii) onset of increasing vibration from cavitation; (iii) heavy vibration from cavitation. ⊙, Ships with low levels of vibration from cavitation; ⊡, ships subject to vibration from cavitation.

where $\bar{w}$ is the circumferential mean wake at a characteristic radius; $\Delta w = (w_{max} - w_{min})$, taken at the characteristic radius; h_t is the depth of propeller tip; w is the wake fraction; p_v is the vapour pressure. To apply this criterion, the calculated point is simply entered into the plot of figure 5. Values falling below the cross-hatched region are likely to correspond to vibration problems.

3. Rutherford (1979) has built upon the concept of the Fitzsimmons criterion by presenting a quantity of ship hull–propeller information and using the trends of plotted data to suggest new limit regions in the graph of σ_{nt} against w_Δ. Discussers of Rutherford's work have also added their suggestions of where the limit regions should be located. An example of a revised wake factor diagram (Fitzsimmon Plot) is included here in figure 6.

4. Huse (1974) has offered a four-part package of recommendations involving

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properties of ship maximum wake fraction values and the width of the characteristic wake distribution intended to help avoid ship vibration problems.

5. Odabasi & Fitzsimmons (1978) have incorporated the features of Huse's recommendations and the wake factor concept of Fitzsimmons to form a wake quality criterion with five parts.

(*a*) The maximum wake measured inside the angular interval $\theta_B = 10 + 360/Z$ centred at the wake peak, and in the range $0.4R$ to $1.15R$ around the top of the wake should satisfy

$$w_{\max} < 0.75 \quad \text{or} \quad w_{\max} < C_B, \tag{9}$$

whichever is less.

(*b*) Maximum acceptable wake peak should satisfy $w_{\max} < 1.7\bar{w}_{0.7}$, where $\bar{w}_{0.7}$ is the mean wake at radius $0.7R$.

(*c*) The width of the wake peak should not be less than the θ_B given above.

(*d*) For acceptable vibration performance, plot of parameters σ_{nt} and w_Δ should lie in the zone of moderate excitation of the Fitzsimmons plot of figure 5.

(*e*) For propellers susceptible to cavitation, the local wake gradient per axial velocity for points inside the angular interval θ_B centred at the wake peak, and in the radial range $0.7R$ to $1.15R$ should satisfy the condition

$$G_w = \frac{1}{r/R}\left|\frac{dw/d\theta}{(1-w)}\right| < 1.0, \tag{10}$$

with θ in radians, w is the local wake fraction value in the region being considered.

6. Ward (1983) has assembled elements of design guidance for vibration avoidance, incorporating features of Huse (1974), Fitzsimmons (1977), and Odabasi & Fitzsimmons (1978) with some further additions and amplifications. Listed here are the elementary parts of the design criteria outline described by Ward, to be added to the features of Odabasi & Fitzsimmons criterion.

(*a*) The waterline maximum angle of run φ_m (or hull water line ending half-angle) should be limited as follows: $\varphi_m \leqslant 23°$ for fine forms and $\varphi_m \leqslant 28°$ for full forms. The maximum angle of run along a streamline should be limited by $(\varphi_m)_s/C_B \leqslant 30°$. Applicability of these particular criteria elements seems to be confined to single-screw and twin-skeg ships.

(*b*) With unsteady pressure amplitudes on the hull above the propeller estimated by empirical means, computed by potential flow analysis, or measured in model experiments Ward's recommended limits for avoiding hull girder and local vibration are $(\Delta p)_Z \leqslant 8$ kPa and $(\Delta p)_{2Z} \leqslant 4$ kPa.

7. Jonk & van der Beek (1983) of MARIN have suggested the use of a parameter to indicate the difficulty of matching a propeller with a given ship. The 'difficulty index' I_d is defined as

$$I_d = \{T + (0.61)[(0.01205)ND^3V_s(\varphi_{m(0.8)} + 29)\}/(h_t + 10)D^2, \tag{11}$$

where T is the thrust (kgf); N is the propeller r.p.m.; D is the propeller diameter (m); V_s is the ship speed (knots); $\varphi_{m(0.8)}$ is the hull waterline ending half-angle at $0.8R$ at 12 o'clock position (deg); h_t is the depth of water above tip (m).

Based on the evaluation of several ships, Jonk & van der Beek (1983) have stated that ships with an I_d larger than 740 are likely to display unacceptable vibration levels. This may be applied to single-screw ships or twin-skeg arrangements for which there is a clear cut way to determine the hull waterline ending half-angle $\varphi_{m(0.8)}$.

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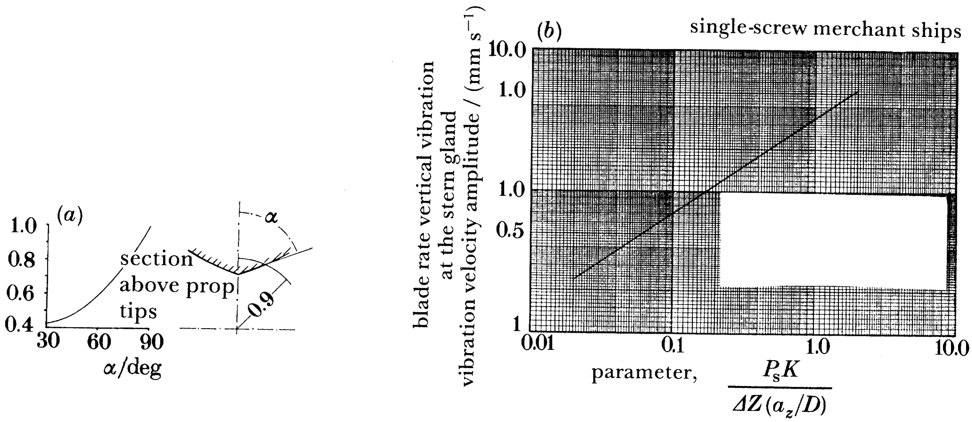

Figure 7. Empirical relationship between blade rate vertical vibration and selected ship characteristics. (From Meek *et al.* 1979.) (*b*) Single-screw merchant ships.

Vibration level limits

1. Some British Ship Research Association efforts in the area of measured propeller-induced ship vibrations have been discussed by Ward & Willshare (1976) and by Meek *et al.* (1979). Both these references have given versions of an empirical estimating scheme for the blade rate vertical vibration level at the stern gland of single-screw merchant ships. A dimensional similarity parameter container the power-to-displacement ratio is defined by Meek *et al.* as

$$P_S K / \Delta Z(a_z/D), \tag{12}$$

where P_S is the shaft power (kW); Δ is the displacement (t); Z is the number of blades; a_z/D is the propeller vertical tip clearance ratio; K is the hull shape factor.

Figure 7, taken from Meek *et al.*, shows a line fit of full-scale data for the blade rate vertical vibration velocity amplitude (mm s^{-1}) plotted against the parameter noted above, and with a legend defining the hull section shape factor K.

One possible use of this diagram would be to choose an acceptable vibration velocity level, and then solve for a limiting value of tip clearance ratio corresponding to the other ship particulars. How this could be trusted without any information about the wake variations or their influence on the response is very uncertain.

2. The SSPA correlation plot of figure 2 can be used to estimate the level of vertical vibration response, say at blade rate (BR) frequency, given the blade rate unsteady pressure amplitude over the tip (estimated, computed, or measured on model scale). The correlation parameter (Johnsson 1983) is given in the legend of figure 2.

Exercise of criteria

To test the applicability of these criteria, they have been applied in several examples of ships for which some corroborating data exists. Two examples have been checked extensively.

Ship A. Single-screw, moderate size, 20-knot supply tanker with narrow V/U-sections aft, a conventional clearwater stern arrangement, and a seven-bladed skewed propeller.

Ship B. Large, high-speed, twin-screw, open stern ship with exposed shafts and V-

Table 1. *Summary of results from elementary criteria for evaluating ship A*

(*a*) *Pressure amplitude limits and/or estimates*

source	estimated pressure amplitude over tips/kPa	recommended or allowable pressure amplitude/kPa	remarks
Holden (1979; Holden *et al.* 1980)	$(\Delta p)_Z = 7.37$		Estimated $(\Delta p)_Z$ is low
Johnsson (1983)		$(\Delta p)_{\text{allow}} = 4.73$	too low allowable value
Björheden (1979)		BR: $(\Delta p)_{\text{recomm}} = 1.45$	very low allowable value
Several authors		single-value limits of BR component	see figure 1; model test value falls high in range; but hull girder vibration was not a problem here

(*b*) *Surface force amplitude limits*

source	estimated reference area vertical surface force	recommended or allowable limit	remarks
Takahashi (1976) (reference area = $D\times D$)	$(F_z)_S/D^2 = 0.676\ \text{kN m}^{-2}$	6–7 kN m^{-2}	well under troublesome level
Mano *et al.* (1976) (reference area = $D\times D$)	$(F_z)_S/D^2 = 0.104\ \text{t m}^{-2}$	0.105–0.115 t m^{-2} (warning zone)	use figure 3; just inside allowable zone
Yamaguchi (1977) (reference area = $D\times D$)	$(F_z)_S/D^2 = 0.082\ \text{t m}^{-2}$	0.42–0.495 t m^{-2} (warning zone)	use figure 4; very low force level; inside allowable zone

(*c*) *Wake quality and cavitation factors*

source	parameters	remarks
Fitzsimmons (1977) Rutherford (1979)	$\sigma_{nt} = 0.22$ $w_\Delta = 0.87$–0.99	figures 5, 6; plots in zone of moderate excitation
Huse (1974)	$w_{\max} = 0.83$	wake violates $w_{\max} < 0.75$
BSRA Scheme:		wake violates $w_{\max} < C_B$
Odabasi & Fitzsimmons (1978); Ward (1983)	$(1.7)\bar{w}_{0.7} = 0.359$ $\theta_B = 61.4°$	wake violates $w_{\max} < (1.7)\,\bar{w}_{0.7}$ width of wake (120°) is greater than θ_B; *OK* σ_{nt}, w_Δ plot as moderate excitation
	$G_w = \dfrac{1}{r/R}\left\|\dfrac{\mathrm{d}w/\mathrm{d}\theta}{1-w}\right\| = 2.1$–$1.1$ in $r/R = 0.7$–1.15	wake gradient factor G_w exceeds 1.0 in range of interest; potential trouble
	$\varphi_{m(0.8)} = 19°$ $\varphi_{m(0.8)}/C_B = 31.8°$	waterline ending half-angle factor exceeds 30°; potential trouble
	measured maximum (model) $(\Delta p)_Z = 9.5$ kPa $(\Delta p)_{2Z} = 3.27$ kPa	BR pressure amplitude exceeds 8 kPa 2BR pressure amplitude less than 4 kPa

Table 1 (*cont.*)

source	parameters	remarks
Jonk & van der Beek (1983)	difficulty index is 655	difficulty index lies below the trouble level of 740

(*d*) *Vibration level limits and/or estimates*

source	parameters	remarks
Ward & Wilshare	$P_s K/\Delta Z(a_z/D) = 0.193$	from figure 7
Meek *et al.* (1979)	inferred vibration velocity $v_v = 1.08\ \mathrm{mm\ s^{-1}}$	predicted value is too low
SSPA Correlation Johnsson (1975); Lindgren & Johnsson (1977); Johnsson (1983)	measured (model) maximum $2(\Delta p) = 19.0\ \mathrm{kPa}$ $k_1 = 4.1 \times 10^4$	from figure 2
	inferred vibration velocity: $v_r = 10.35\ \mathrm{mm\ s^{-1}}$ (r.m.s.)	predicted value is too high

strut supports, hull form similar to a container ship, and six-bladed skewed propellers.

Ship A is a type similar to those covered by the databases used to establish the available criteria described in the paper. It has a deep, spiky wake that represents a general category of ship that has become known for tendencies toward noisy and vibration-prone operation. Ship B, with its open stern arrangement has an upwardly sloped after body and a relatively mild wake shadow in way of the propeller discs. It is rather dissimilar to the ships used for databases supporting the elementary criteria.

Some particulars on the hull geometry, propeller, propulsion characteristics, and wake for each of the ship examples are given in Appendixes A and B. Calculations have been carried out for each case using the simple formulas and prescriptions outlined earlier, and the prediction results are summarized in table 1 for ship A and table 2 for ship B.

Discussion and conclusions

Some idea of the quality of the answer from each of the engineering prescriptions discussed is shown in the scorecard presentation of table 3. Overall, the mixed results from the examples do not inspire solid confidence in any of the elementary criteria, especially in terms of revealing the nature of specific excitation problems. Probably we cannot expect any of these simple criteria to predict the peculiar difficulties of ship A, for instance, which had high blade rate pressure amplitudes, with no problems with blade rate hull girder vibration, but unacceptably high inboard airborne noise.

In the criteria presented, propeller blade skew was not addressed at all. Skew can have a large, beneficial effect by reducing induced pressure amplitudes. Information on the isolated effect of blade skew on pressure pulses has been included in the pressure pulse estimation scheme of Johnsson (1983).

Certain parts of some of these criteria are promising and provide useful indications.

1. Pressure amplitude formulas of Holden (1979) give fairly believable results, for the blade rate component. Skew effects should be added, however.

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Table 2. *Summary of results from elementary criteria for evaluating ship B*

(*a*) *Pressure amplitude limits and/or estimates*

source	estimated pressure amplitude over tips/kPa	recommended or allowable pressure amplitude/kPa	remarks
Holden (1979; Holden *et al.* 1980)	$(\Delta p)_Z = 14.3$		estimated $(\Delta p)_Z$ is too high
Johnsson (1983)		$(\Delta p)_{\text{allow}} = 18.8$	very high value; large ship predicted with high tolerance
Björheden (1979)		$(\Delta p)_{\text{recomm}} = 7.14$	
several authors		single-value limits	see figure 1 middle of range

(*b*) *Surface force amplitude limits*

source	estimated reference area vertical surface force	recommended or allowable limit	remarks
Takahashi (1976) (two props, each reference area $D \times D$)	$(F_z)_S/D^2 = 4.18$ kN m^{-2}	6–7 kN m^{-2}	below troublesome level
Mano *et al.* (1978) (two props, each reference area $D \times D$)	$(F_z)_S/D^2 = 0.18$ t m^{-2}	0.105–0.115 t m^{-2} (warning zone)	use figure 3; plots well into zone of vibration problems
Yamaguchi (1977) (two props, each reference area $D \times D$)	$(F_z)_S/D^2 = 1.16$ t m^{-2}	0.385–0.45 t m^{-2} (warning zone)	use figure 4; plots in zone of vibration problems

(*c*) *Wake quality and cavitation factors*

source	parameters	remarks
Fitzsimmons (1977); Rutherford (1979)	$\sigma_{nt} = 0.0946$ $w_\Delta = 0.28$–0.38	figures 5, 6; plots in indeterminant region at low end of w_Δ range
Huse (1974)	$w_{\max} = 0.55$	wake satisfies $w_{\max} < 0.75$
BSRA Scheme:		wake satisfies $w_{\max} < C_B$
Odabasi & Fitzsimmons (1978)	$(1.7)\,\bar{w}_{0.7} = 0.09$	wake violates $w_{\max} < (1.7)\,\bar{w}_{0.7}$
Ward (1983)		
	$\theta_B = 70°$	width of wake (*ca.* 130°) is greater than θ_B; *OK*
		σ_{nt}, w_Δ values plot in indeterminant region, doubtful indication
	$G_w = \dfrac{1}{r/R}\left\lvert\dfrac{dw/d\theta}{1-w}\right\rvert = 0.45$–$0.5$ in $r/R = 0.7$–1.15	wake gradient factor G_w is less than 1.0; mild wake
	measured maximum (model) $(\Delta p)_Z = 6.72$ kPa	BR pressure amplitude less than 8.0 kPa
	$(\Delta p)_{2Z} = 4.34$ kPa	2BR pressure amplitude exceeds 4 kPa

Table 2 (*cont.*)

(*c*) *Vibration level limits and/or estimates*

source	parameters	remarks
SSPA Correlation Johnsson (1975); Lindgren & Johnsson (1977); Johnsson (1983)	Measured (model) maximum $2(\Delta p) = 13.44$ kPa $k_1 = 2.23 \times 10^3$ inferred vibration velocity $v_v = 1.82$ mm s^{-1} (r.m.s.)	from figure 2 predicted value is low

2. The Fitzsimmons plot (figures 5 and 6) seems to give a reasonable idea about the trend of possible excessive hull girder vibration, but not about allied problems such as the high propeller-excited inboard airborne noise of ship A.

3. The wake gradient factor of Odabasi & Fitzsimmons seems to point in the right direction for identifying wakes with potential for making trouble. Alone, it is insufficient to predict occurrence of excitation problems because no account is provided for the important influence of propeller blade geometry.

4. The basic idea of the SSPA correlation seems to work. It attempts to account for the important influence of ship size on the gross vibration response due to fluctuating hull pressures above the propeller. The slope of the correlation curve of figure 2 may be generally applicable, but the actual values of the correlation mean line do not match the sample database results for Navy auxiliary (cargo) ships very well.

The work involved with this paper was sponsored by the Ship Hydromechanics Department of the David Taylor Research Center and by the Ship Performance and Hydromechanics Block Program of the Naval Sea Systems Command. Ms Melanie Joy provided the excellent typing of the typescript.

Appendix A. Example ship A: single-screw, U/V form, 20 knot tanker

Selected particulars for ship A are:

Hull, power, speed	Propeller, operating conditions
$L_{OA} = 180.3$ m	$D = 6.4$ m
$L_{WL} = 170.8$ m	$Z = 7$ blades
$B = 26.8$ m	$A_E/A_0 = 0.771$
$D_H = 14.94$ m	$\theta_S = 45°$
$T_M = 9.6$ m	$(P/D)_{0.8} = 1.14$
$T_A = 9.45$ m	$N = 100$ r.p.m.
$\Delta = 26813$ t	$J = 0.797$
$\nabla = 26138$ m^3	$K_T = 0.291$
$C_B = 0.597$	$C_T = 1.167$
$a_z/D = 0.292$	$f_Z = 11.667$ Hz
$a_x/D = 0.55$	$w_T = 0.23$
$\varphi_{m(0.8)} = 19°$	$w_{max} = 0.83$
$P = 17900$ kW	$T = 1392$ kN
$V_S = 21.5$ knots	

Model experimental results for the propeller-induced, blade rate unsteady hull pressures for ship A are displayed in figure 8, for the case of full power speed of

Table 3. *Scorecard showing if criterion gives correct indication*

item	example ship case A	example ship case B
pressure amplitude limits and/or estimates		
Holden (estimate)	BR estimate low	BR estimate high
Johnsson/SSPA	no	mixed
Björheden	no	—
single value	mixed	yes
surface force limits		
Takahashi	yes	yes
Mano *et al.*	~ yes	no
Yamaguchi	yes	no
wake quality and cavitation factors		
Van Gunsteren & Pronk	yes	no
BSRA (Fitzsimmons, Odabasi, Ward)	mixed	mixed, indeterminate
Jonk & Van der Beek	yes	—
vibration level limits and/or estimates		
Ward & Willshare; Meek *et al.*	no (low)	—
Johnsson/SSPA	no (high)	no

21.5 knots. In this case, the maximum blade rate hull pressure amplitude occurs over the tips, on the centreline. At this same point, the first three blade rate harmonic components are $(\Delta p)_Z = 9.5$ kPa, $(\Delta p)_{2Z} = 3.27$ kPa, $(\Delta p)_{3Z} = 1.63$ kPa.

The full-scale representative vibration velocity amplitude, measured on the centreline of the main deck, near the rudder post, at 100 r.p.m. is $v_V = \pm 3.3$ mm s^{-1} (r.m.s.).

This ship did not have a propeller-excited hull girder vibration problem, but had a problem with excessive unsteady surface force excitation that showed up as heavy inboard airborne noise.

Appendix B. Example ship B: twin-screw, open-stern, high-speed cargo ship

Selected particulars for ship B are:

Hull, power, speed	Propeller, operating conditions
$L_{OA} = 229.8$ m	$D = 7.01$ m
$L_{WL} = 222.5$ m	$Z = 6$ blades
$B = 32.61$ m	$A_E/A_0 = 0.758$
$D_H = 20.32$ m	$\theta_S = 33°$
$T_M = 11.58$ m	$(P/D)_{0.8} = 0.934$
$\Delta = 48900$ t	$N = 144.2$ r.p.m.
$\nabla = 47660$ m^3	$J = 0.799$
$C_B = 0.567$	$K_T = 0.14$
$a_z/D = 0.296$	$C_T = 0.558$
$a_x/D = 3.17$	$f_Z = 14.42$ Hz
$c_t/D = 0.233$	$w_T = 0.075$
$P = 75950$ kW	$w_{max} = 0.55$
$V_S = 28.3$ knots	$T = 2004$ kN (each)

figure 8

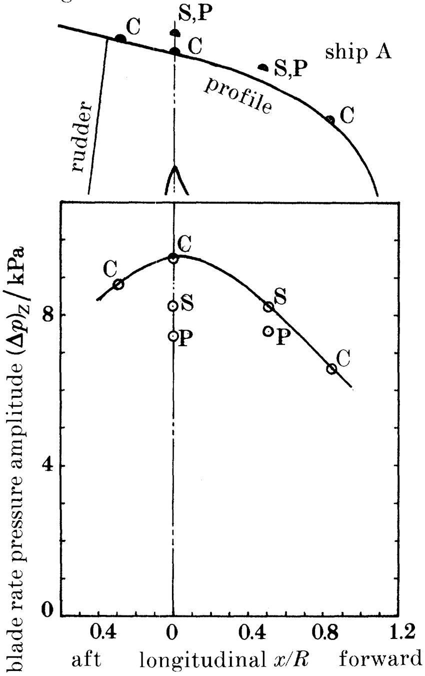

figure 9

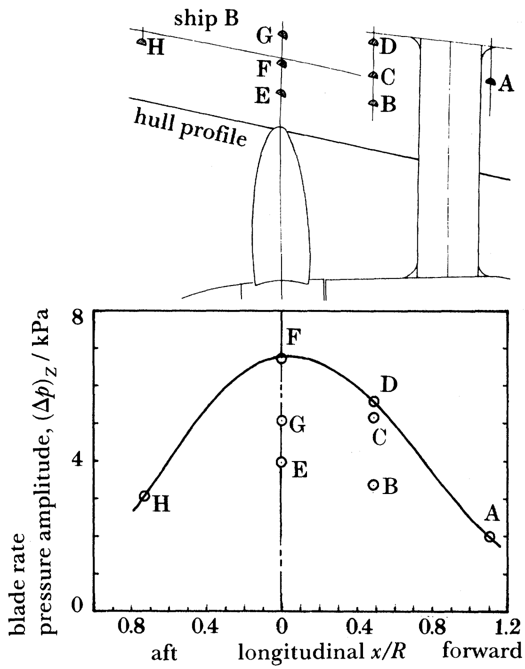

Figure 8. Longitudinal distribution of model experiment blade rate pressure amplitudes for ship A. Pressure gauge location: C, centreline; S, starboard side; P, port side.

Figure 9. Longitudinal distribution of model experiment blade rate pressure amplitudes for ship B.

Model experimental results for the propeller-induced blade rate unsteady hull pressures for ship B are displayed in figure 9, for the full power condition. In this case, the maximum blade rate pressure amplitude occurs in line with the tips, at the point of closest approach of the tips to the hull (point F of figure 9). At this same point, the first three blade rate harmonic components are: $(\Delta p)_Z = 6.72$ kPa, $(\Delta p)_{2Z} = 4.34$ kPa, $(\Delta p)_{3Z} = 2.89$ kPa.

The parent (and very similar) ship design for ship B has been operated many years with an old propeller design that produces about double the levels of pressure amplitude indicated above. No complaints of vibration or noise have been reported with that ship.

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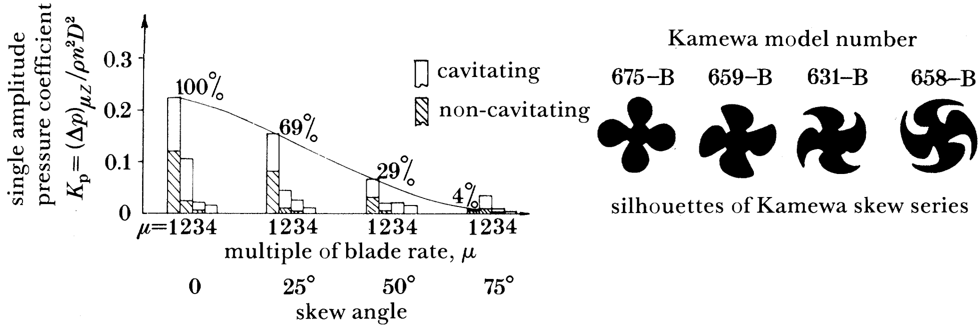

Figure 10. Effect of blade skew on propeller-induced pressure amplitudes on hull surface above propeller for a simulated single-screw merchant ship for non-cavitating and cavitating operation. (From Björheden 1981.)

Discussion

B. O. Wall (*Ministry of Defence, Bath, U.K.*). The criteria reviewed in the paper contained no explicit reference to the effects of propeller skew, which I believe to be an important parameter. The unfavourable characteristics associated with single-screw vessels are in my view associated with stern shape rather than the number of shafts. Single-screw frigate designs, for example, were not noticeably worse than twin-screw designs in terms of propeller-induced vibration.

M. B. Wilson. Mr Wall points out that propeller skew does not appear to be properly accounted for in the criteria reviewed. Skew is a very important feature and design option for marine propellers. Skew provides several benefits. It has the effect of widening the characteristic propeller cavitation bucket or region of cavitation-free operation, and it leads to large reductions of all the unsteady bearing loads (see Boswell 1971; Cumming *et al.* 1972). Another important effect is the reduction of the amplitudes of unsteady pressure pulses induced on a hull surface near the propeller. This reduction with skew occurs for non-cavitating propeller-induced pressures (see Nelka 1974) and in the presence of blade sheet cavitation as well. Figure 10 shows the variation of induced surface pressure amplitudes with increasing skew angle, for several blade rate harmonics, obtained from tests with a series of model propellers with otherwise identical characteristics.

For use with the elementary criteria, the drop-off of blade rate pressure amplitude, for example, could be applied as a skew factor correction to the estimated pressure amplitude obtained from Holden's (1979) formulas given in the paper as equations (2), (3), and (4).

Johnsson (1983) shows a similarly varying recommended correction curve for the effect of skew on estimated pressure amplitude on a surface near a cavitating propeller. It is based, in part, on the data of Björheden (1981).

Problems with propeller-excited vibration and noise are dependent mainly on the character of the wake, the propeller geometry and magnitude of thrust loading, and the blade clearances and geometry of the nearby hull shape. It is an observation that there are only rare instances of propeller-induced vibration problems occurring with twin-screw and multiple-screw ships. The important characteristic is the quality of the wake. Most older twin-screw ships (with open stern and V-struts and inclined shafts) have relatively mild wakes in terms of velocity gradient and magnitude of the velocity defect. The troublesome variety of single-screw ships either have closed

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sterns or involve some variant of a clearwater stern. But the governing characteristic is a large velocity-defect wake with very steep circumferential gradients. Twin-skeg ships tend to produce a pair of single-screw-type wake profiles.

Another important factor is that for twin- and multiple-screw ships the thrust loading C_T is generally lower on each propeller because the total required thrust is shared. Vibration problems often go hand in hand with high thrust loading.

The type of stern of a typical single-screw frigate ship (or any combatant type) is the open variety, with gradually upwardly sloped buttocks. The wake of such a ship is usually very mild. Therefore, unless the tip clearances are unreasonably small, the frigate-style hull wake-propeller combination is very rarely troublesome.

N. J. Smail (*Kent, U.K.*). In as comprehensive a treatment of the subject as this, it would have been appropriate to have included an assessment of those high-frequency vibrations that arise in the propeller blade itself, as distinct from the major ones generated, at relatively low frequency, by the varying wake distribution over the propeller disc and by proximity of the sternframe. I refer, for example, to leading edge or trailing edge vibration, the phenomenon once known as 'singing'. In the early days of high-efficiency propeller design one major manufacturer offered a Guarantee Against Singing, which defined it as a propeller excited vibration 'severe enough to affect seriously the health and comfort of passengers and crew'. In the light of our present concern, would that now have to include 'the safety of the ship'? Although, admittedly, the principal effect of these vibrations is experienced by their transmission through the shafting, where suitable damping can be arranged, they are nevertheless a contributor to the total pattern of hull vibration. They can be minimized in the blade design by such measures as increasing blade skew, but has the highly skewed propeller eliminated them altogether?

There are now available a variety of wake flow enhancing gadgets, and other devices, for fitting to the after-body of a ship – ducts, fins, vane wheels, Z drives, contrarotation and controllable pitch (CP) – and their interaction with forces acting on the hull. A CP propeller, for instance, working in an off-design condition will cause unpredicted disturbance of the wake pattern as well as negating the fuel economy and efficiency considerations on which the propeller design was based.

M. B. Wilson. The short answer is that skew has little effect on propeller singing noise. Skew is not the design feature which works against the occurrence of singing. Modern propeller blades are usually designed with an anti-singing trailing edge. This consists of a non-symmetric bevel or curved wedge shape cut at the trailing taper of the blade foil section all along the downstream edge of each blade. Blade skew alone is of secondary importance to this type of noise problem.

On the question of the effects of flow-modifying devices and propeller types, I include some notes. (*a*) Ducts located upstream of the propeller (e.g. Mitsui Integrated Duct or Hitachi Zosen Nozzle) have been shown to help reduce vibration problems somewhat, but only for very full form ships such as tankers, etc., with very severe initial wake patterns. (*b*) A large variety of fins and turning vane concepts have been proposed and applied full scale. The technical literature on this subject alone is very extensive by now. Most of these devices work by a partial 'filling in' or directing of flow into the slowest flow regions of the wake, thereby reducing the extremes of the inflow velocity field. This makes the net variation of local angle of attack at the blade sections smaller, and thus reduces the extremes of the unsteady sheet cavity volumes and the severity of cavity collapse. Generally the effect is to reduce the unsteady excitation pressure levels, although there have been instances where no fin seems to work well enough. (*c*) Vane wheels typically work to reduce the peak pressure amplitudes on the hull above the main rotor, because the thrust loading has been lowered from the levels without the vane in place. The thrust loading is shared, since the free-wheeling vane takes up some of the thrust. The net effect is usually beneficial. (*d*) If the Z-drive mentioned by Mr Smail means an orientable thruster, operation of such a propulsor at a large yaw angle to the ahead-direction could introduce severe vibration problems, if the ship speed were high enough. The mechanism would be the exaggeration of unsteady blade cavitation in the poor wake field. (*e*) Use of contrarotating propellers would ease the tendency toward vibration problems because the thrust loading on each rotor would be lower than on a single equivalant propeller. (*f*) Operation of a controllable pitch propeller in a sufficiently off-design mode could aggravate the severity of unsteady blade cavitation and thus the tendency toward excessive excitation.

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