



GUIDANCE NOTES ON

SHIP VIBRATION

FEBRUARY 2018

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Foreword (1 February 2018)

The American Bureau of Shipping recognizes the overall ship vibration as an important measure to ensure the habitability, safety and functionality of the vessels. The *ABS Guidance Notes on Ship Vibration* have been developed to provide users with specific guidance on the design, analysis, measurement procedures and criteria in order to achieve the goal of limiting the ship vibration to an acceptable level. In the text herein, this document is referred to as “these Guidance Notes”.

The design and construction of the hull, superstructure, and deckhouse of a steel vessel are to be based on all applicable requirements of the *ABS Rules for Building and Classing Steel Vessels (Steel Vessel Rules 2006)*. Specifically, for the Container Carriers over 130 meters in length, the *ABS Steel Vessel Rules* require the consideration of vibratory responses of hull structures, as applicable (5-5-3/13.1). For the LNG Carriers, the *ABS Steel Vessel Rules* require special attention to the possible collapse of membrane due to hull vibration (5-8-4/4.2). In conjunction with the propulsion shaft alignment, the *ABS Steel Vessel Rules* require the consideration of propulsion shaft vibrations (4-3-2/7). For the cargo and passenger vessels, ABS provides optional classification notations for crew habitability and passenger comfort (*ABS Guide for Passenger Comfort on Ships* and *Guide for Crew Habitability on Ships*). Also ABS provides Condition Monitoring Program for machinery vibration (7-A-14/5.1.2 of the *ABS Rules for Survey After Construction (Part 7)*).

These Guidance Notes provide practical guidelines on the concept design to assist ship designers to avoid excessive shipboard vibration at an early design stage. These Guidance Notes also assist with the finite element analysis (FEA) based vibration analysis procedure to calculate the vibration response and evaluate the design at detail design stage. The analysis procedure represents the current analysis practice in ABS. These Guidance Notes also offer guidelines on the vibration measurement procedure at sea trials and the acceptance criteria on vibration limits based on the international standards and the practice in ABS.

The 2018 edition provides guidance on quick local vibration analysis methods for ships.

These Guidance Notes become effective on the first day of the month of publication.

Users are advised to check periodically on the ABS website www.eagle.org to verify that this version of these Guidance Notes is the most current. Comments or suggestions can be sent electronically to rsd@eagle.org

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SECTION 1 General

1 Introduction

With the increase of ship size and speed, shipboard vibration becomes a great concern in the design and construction of the vessels. Excessive ship vibration is to be avoided for passenger comfort and crew habitability. In addition to undesired effects on humans, excessive ship vibration may result in the fatigue failure of local structural members or malfunction of machinery and equipment.

These Guidance Notes are to provide users, specifically shipyards, naval architects, and ship owners, with practical guidance on the concept design to avoid excessive ship vibration at an early design stage. If simple procedures are followed with insight and good judgment in the concept design stage, then the difficult countermeasures and corrections at the subsequent design stages may be avoided in most cases.

These Guidance Notes also assist with the finite element analysis (FEA) based vibration analysis procedure to predict the vibration response and evaluate the design in detail design stage. The vibration analysis procedure represents the most current analysis practice in ABS. These Guidance Notes also offer guidelines on the vibration measurement procedure during the sea trials and the acceptance criteria on vibration limits based on international standards and practice in ABS.

3 Application

These Guidance Notes are applicable to the vessels of all lengths.

5 Scope

These Guidance Notes provide overall guidelines on ship vibration excited by the main engine and propeller. In these Guidance Notes, the following subjects are considered:

- i) Concept Design
- ii) Vibration Analysis
- iii) Measurements
- iv) Acceptance Criteria

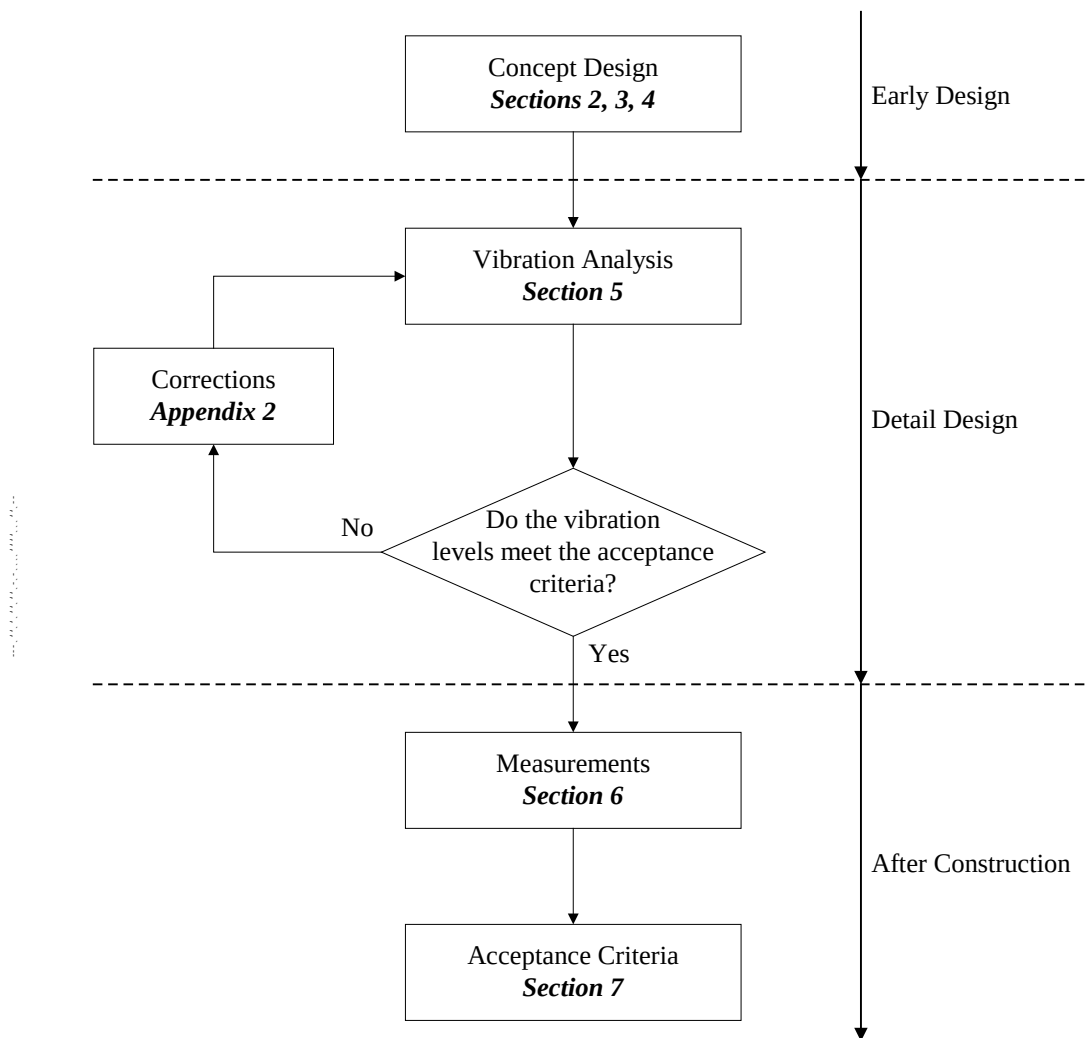
The concept design in Sections 2, 3 and 4 provides users with immediate, direct, and concise guidance in effectively dealing with ship vibration in the *concept design* stage. In attempting to provide a sound and no-nonsense guidelines, these Guidance Notes identify the most serious problem areas that have caused difficulties to the industry, and concentrate on those areas. In the concept design, local vibration is not addressed because detail information is not usually available in the early design stage. Instead, the concept design is focused on those areas that have been known to be of critical importance in avoiding harmful ship vibration.

The vibration analysis in Section 5 provides the FE-based vibration analysis procedure based on first principles direct calculations. The FE-based vibration analysis is recommended to evaluate the design during the detail design stage. If found necessary, the local vibration is to be addressed in the detail vibration analysis. The analysis procedure provides guidelines on FE modeling, engine and propeller excitation, and free and forced vibration analysis.

For the assessment of ship vibration performance, the actual vibration levels at the most critical locations are to be measured and evaluated during the sea trials. Section 6 provides guidelines on the vibration measurement procedure on the instrumentation, measurement conditions and locations, data processing and reporting. Section 7 provides acceptance criteria on the vibration limits for human comfort and habitability, local structures and machinery based on international standards and practice in ABS.

The shaft alignment and torsional vibration are not directly addressed in this document. For the requirements of the shaft alignment and torsional vibratory stress, refer to 4-3-2/7 of the *ABS Steel Vessel Rules*. The overall procedure for ship vibration assessment recommended in these Guidance Notes is shown in Section 1, Figure 1.

FIGURE 1
Overall Procedure for Ship Vibration Assessment





SECTION 2 Concept Design

1 Introduction

Concept design is where the vibration avoidance process must begin. It is clear that if the vibration problems, repeatedly identified by experience as the most important, are addressed at the earliest design stage, ultimately serious problems, involving great cost in correction efforts, may be avoided.

The focus is on planning for vibration early at the *Concept Design* stage, where there has been no development of details. If as much as possible can be done in concept design with the simple tools and rules of thumb available at that level, it will help to avoid major vibration problems. The major potential problems may often be present in the crude concept design definition. Just identifying and addressing those potential problems in terms of the minimal technology available at the concept design stage is considered very important to the success of ship design.

Sections 2 through 4 provide guidelines on the concept design. Some of these guidelines are presented in *Principles of Naval Architecture, Chapter 7* (SNAME, 1988) and the SNAME granted permission to be included in this document.

3 Design Considerations

The four elements of importance in ship vibration are:

- Excitation,
- Stiffness,
- Frequency Ratio, and
- Damping

It is noted that any of the following contribute to vibration reduction:

- Reduce exciting force amplitude, F .* In propeller-induced ship vibration, the excitation may be reduced by changing the propeller unsteady hydrodynamics. This may involve lines or clearance changes to reduce the non-uniformity of the wake inflow or may involve geometric changes to the propeller itself. Specifics in this regard are addressed in Section 3.
- Increase stiffness, K .* Stiffness is defined as spring force per unit deflection. In general, stiffness is to be increased rather than decreased when variations in natural frequency are to be accomplished by variations in stiffness. It is not a recommended practice to reduce system stiffness in attempts to reduce vibration.
- Avoid values of frequency ratio near unity; $\omega/\omega_n = 1$ is the resonant condition.* At resonance, the excitation is opposed only by damping. Note that ω/ω_n can be varied by varying either excitation frequency ω or natural frequency ω_n . The spectrum of ω can be changed by changing the RPM of a relevant rotating machinery source, or, in the case of propeller-induced vibration, by changing the propeller RPM or its number of blades. ω_n is changed by changes in system mass and/or stiffness; increasing stiffness is the usual and preferred approach. Specific measures for resonance avoidance in ships are addressed in Section 4.
- Increase damping, ζ .* Damping of structural systems in general, and of ships in particular, is small; $\zeta \ll 1$. Therefore, except very near resonance, the vibratory amplitude is approximately damping independent. Furthermore, damping is difficult to increase significantly in systems such as ships; ζ is, in general, the least effective of the four parameters available to the designer for implementing changes in ship vibratory characteristics.

5 Concept Design Approach

Four elements were identified in the preceding as being influential in determining ship vibratory response, and their relationship to vibration reduction was addressed. While quantification of all four elements is required in calculating the vibration response level, acceptable results may consistently be achieved with reasonable effort by focusing attention in concept design on two of the four elements. The two of the four elements of importance are *excitation* and *frequency ratio*. The achievement in design of two objectives with regard to these elements has resulted in many successful ships:

- Minimize dominant vibratory excitations, within the normal constraints imposed by other design variables, and
- Avoid resonances involving active participation of major subsystems in frequency ranges where the dominant excitations are strongest.

Unlike vibration response, the excitation and frequency ratio elements involved in these objectives may be predictable with useful reliability for making design judgments. For the most troublesome case of propeller excitation, detailed hydrodynamic calculation procedures in conjunction with model testing have been established, at least to the level of reliable relative predictions in the post-concept, preliminary design stage. Natural frequencies involving the ship hull and its major subsystems are usually predictable using judicious modeling and modern numerical structural analysis methods. Accuracy levels achievable in predictions of propeller and engine excitation and of sub-system natural frequencies are generally considered to be reliable to consistently achieve the two objectives cited above.

The detailed calculations and experiments required to avoid resonance and to minimize excitation are usually performed by engineering analysis groups or model basins, and are usually not the immediate responsibility of the ship designer. If addressed early enough while basic changes in a design can still be made, excessive vibration may be avoided effectively. The main functions of the ship designer in this regard are:

- To establish a concept design to serve as the subject of the detailed investigations,
- To judge whether detailed analysis is warranted as a further step.

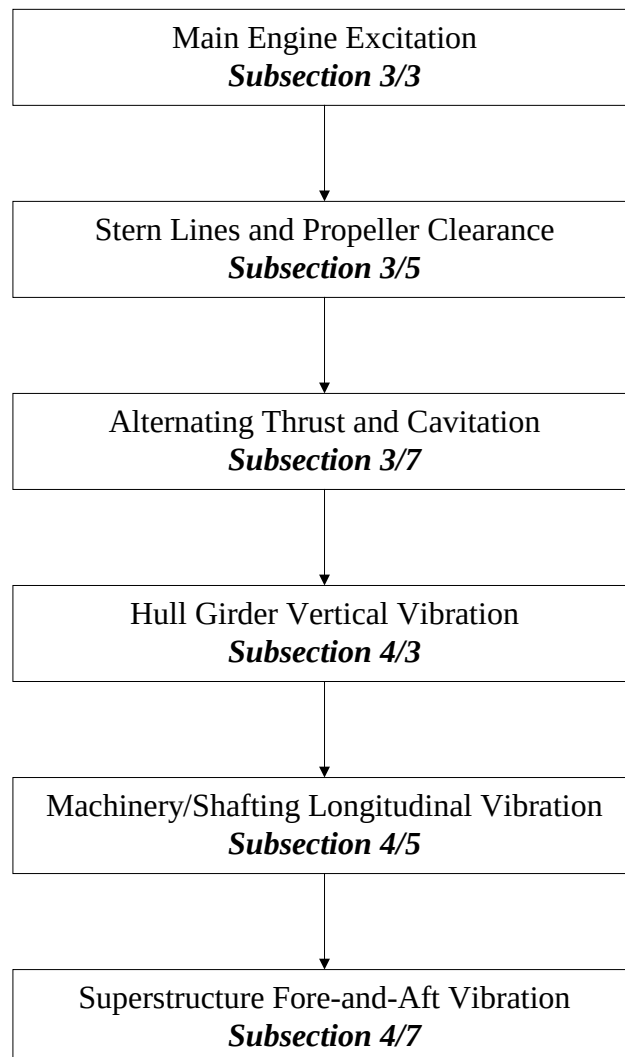
The quality of the concept design will be reflected in the number of detailed iterations, if any, that are required for achieving an acceptable final design. For the purpose of establishing high-quality concept designs, which may require time consuming calculations and model testing, the designer needs both guidance as to the areas of the design likely to be in most need of attention, and some simple methodology for identifying the critical aspects. Experience has shown that attention to vibration in concept design of large ships can usually be paid to the following items:

- i) Hull girder vertical vibration excited by the main engine.
- ii) Main machinery/shafting system longitudinal vibration excited by the propeller.
- iii) Superstructure fore-and-aft vibration excited by hull girder vertical vibration and/or main propulsion machinery/shafting system longitudinal vibration.

A myriad of local vibrations, such as hand-rails, antennas, plating panels, etc., may be encountered on new vessel trials in addition to these three. But local problems usually involve local structural resonances and often considered as minor problems, as the correction approach by local stiffening may be easily achievable.

Section 3 and Section 4 provide general guidance on the methodology of established effectiveness in dealing with the three critical items cited above in the concept design stage. Section 2, Figure 1 shows the important areas to be considered during concept design. The concept design checklist is summarized in Appendix 4.

FIGURE 1
Items to be Considered During Concept Design



SECTION 3 Excitations

1 Introduction

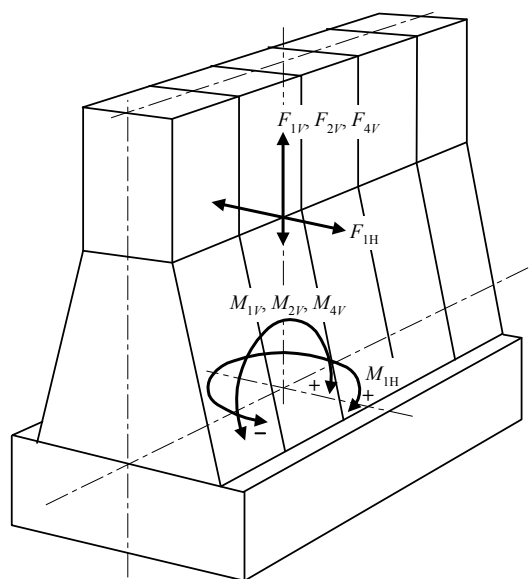
It is appropriate that the principal vibration exciting sources be addressed first, since with high excitation levels excessive vibration can occur almost independently of system structural characteristics. In general, the major sources are the low-speed diesel main engine and the propeller. Gas turbines are generally considered to give less excitation than diesel engines. Thus, in this section, attention is paid to the excitation of the low-speed main diesel engine.

3 Low-speed Main Diesel Engine

Significant progress has been made in recent years by engine manufacturers in reducing vibratory excitation, largely by moment compensators installed with the engine. Steps to be taken by the engine manufacturer are to be addressed in main engine specifications for a new vessel. In this regard, it is important for the shipyard or owner to be technically knowledgeable on this issue. Diesel engine vibratory excitation can be considered as composed of three periodic force components and three periodic moment components acting on the engine foundation. Actually, the periodic force component along the axis of the engine is inherently zero, and some other components usually balance to zero depending on particular engine characteristics.

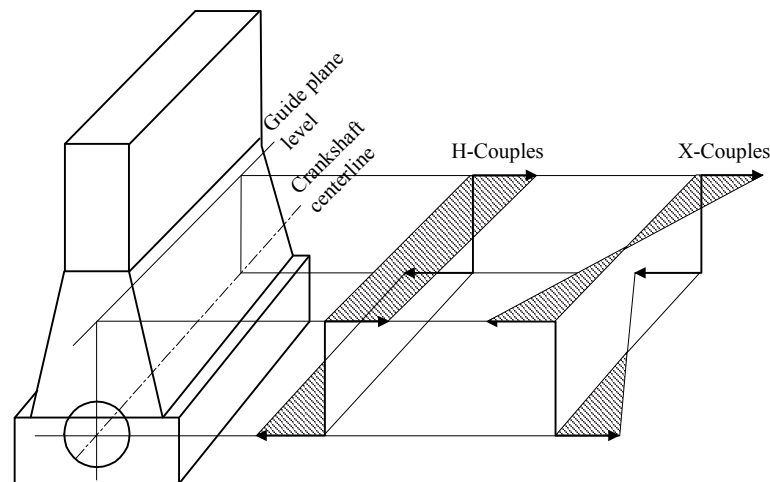
Two distinctly different types of forces can be associated with the internal combustion reciprocating engine. These are: (a) gas pressure forces due to the combustion processes (guide force couples) and (b) inertia forces produced by the accelerations of the reciprocating and rotating engine parts (external forces). Section 3, Figure 1 shows the typical external force and moments acting on a diesel engine.

FIGURE 1
External Forces and Moments



Guide force couples acting on the crosshead result from transverse reaction forces depending on the number of cylinders and firing order. The guide force couples cause rocking (H-couples) and twisting (X-couples) of the engine, as shown in Section 3, Figure 2. The engine lateral vibration due to the guide force couples may cause resonance with the engine foundation structure. A possible solution at concept design stage may be a consideration of lateral stays (top bracings), connecting the engine's top structure to the ship hull.

FIGURE 2
Guide Force Couples



The vertical force and moment, which are of primary concern with regard to hull vibratory excitation, and the transverse force and moment as well, are due to unbalanced inertial effects. For engines of more than two cylinders, which is the case of interest with ships, the vertical and transverse inertia force components generally balance to zero at the engine foundation. This leaves the vertical and transverse moments about which to be concerned. The values of the moment amplitudes are usually tabulated in the manufacturer's specification for a particular engine.

The majority of low speed marine diesels currently in service have 6 cylinders or more. Therefore, the 2nd order vertical moment M_{2v} is generally considered to contribute the most to the hull vibration. However, depending on the specific number of cylinders, the first order or higher order moment can be as large as the second order moment. In that case, further consideration is to be given to the first or higher order moment.

A hull girder mode up to the third or fourth can have a natural frequency as high as the twice-per-revolution excitation of the 2nd order vertical moment. Hull girder modes higher than the first three or four have diminishing excitability and may be of less concern. The following steps are therefore recommended in concept design:

- i) The 2nd order vertical moment M_{2v} is the diesel engine excitation of most concern. The potential danger is in resonating one of the lower hull girder vertical modes with a large 2nd order vertical moment. The value of M_{2v} is to be requested from the potential main engine manufacturer, as early as possible.
- ii) Power Related Unbalance (PRU) values may be used to determine the acceptable level of M_{2v}

$$PRU = \frac{M_{2v} \text{ [N-m]}}{\text{EnginePower [kW]}}$$

TABLE 1

<i>PRU</i>	<i>Need for Compensator</i>
Below 120	Not likely
120-220	Likely
Over 220	Most likely

Further attention is recommended in cases where PRU exceeds 220 N-m/kW. The action recommended at the initial engine selection stage may be either change of engine selection or installation of moment compensators supplied by the engine manufacturer. Otherwise, the vertical hull girder response is to be checked by calculation within an acceptable level without installation of compensators.

- iii) The engine lateral vibration due to X-type and H-type moments may produce excessive local vibration in the engine room bottom structure depending on the engine frame stiffness and engine mounting. The installation of lateral stays on the engine room structure is to be addressed at the early design stage.

5 Hull Wake

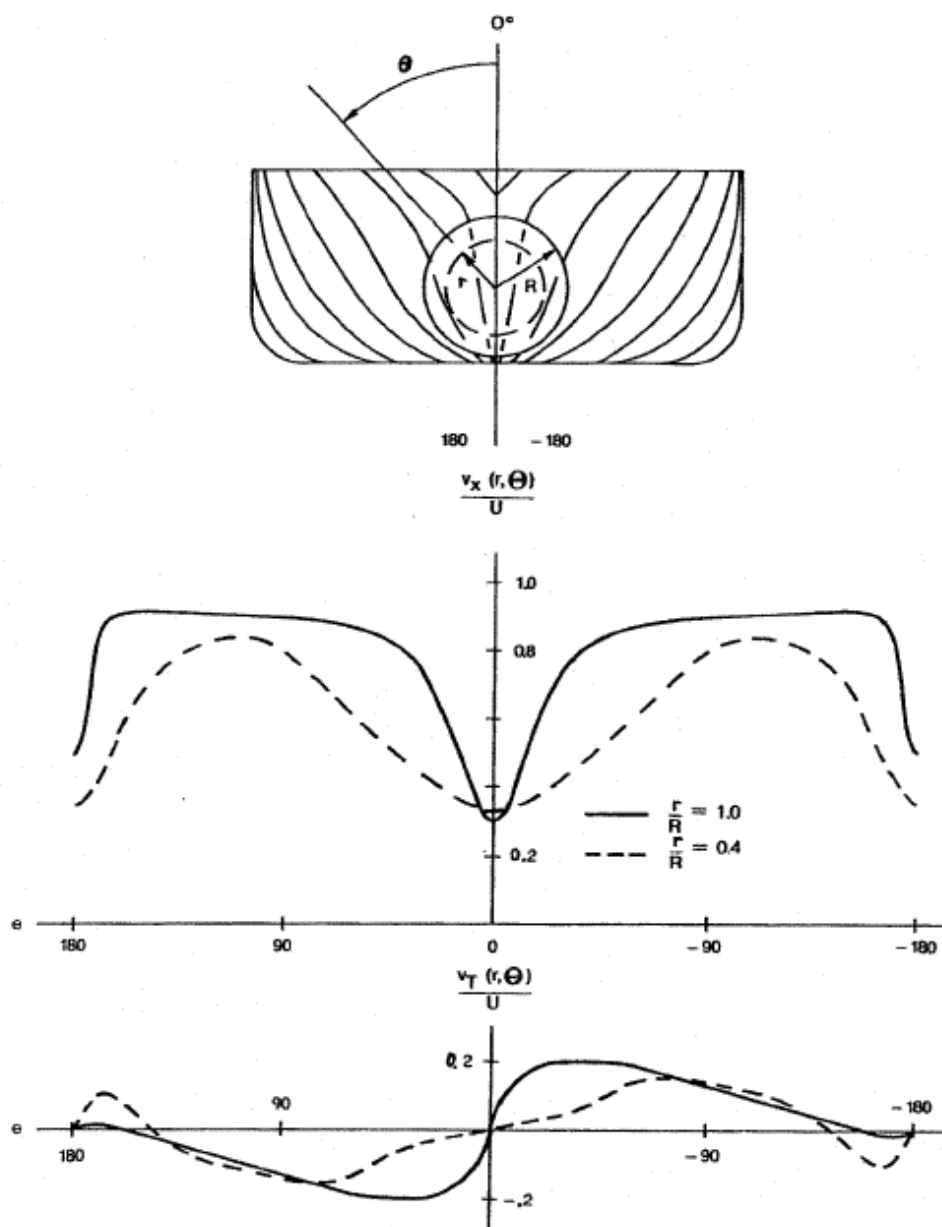
Hull wake is one of the most critical aspects in avoidance of unacceptable ship vibration. Propeller-induced vibration problems in general start with unfavorable hull lines in the stern aperture region, as manifest in the non-uniform wake in which the propeller must operate. Unfortunately, propeller excitation is far more difficult to quantify than the excitation from internal machinery sources. This is because of the complexity of the unsteady hydrodynamics of the propeller operating in the non-uniform hull wake. In fact, the non-uniform hull wake is the most complicated part; it is unfortunate that it is also the most important part. Propeller-induced vibration would not be a consideration in ship design if the propeller disk inflow were circumferentially uniform. Any treatment of propeller excitation must begin with a consideration of the hull wake.

For engineering simplification, the basic concepts allow for the circumferential non-uniformity of hull wakes, but assume, for steady operation, that wake is time invariant in a ship-fixed coordinate system. Nominal wake data from model scale measurements in towing tanks are presented either as contour plots or as curves of velocity versus angular position at different radii in the propeller disc. Section 3, Figure 3 shows the axial and tangential velocity components for a typical conventional stern merchant ship.

The position angle, θ , on Section 3, Figure 3 is taken as positive counterclockwise, looking forward, and x is positive aft. The axial wake velocity v_x and tangential wake velocity v_T are dimensionless on ship forward speed, U . Note that the axial velocity is symmetric in θ about top-dead-center (even function) and the tangential velocity is asymmetric (odd function). This is a characteristic of single screw ships due to the transverse symmetry of the hull relative to the propeller disk; such symmetry in the wake does not, of course, exist with twin-screw ships.

The wake illustrated above represents one of the two characteristically different types of ship wakes. The flow character of the conventional skeg-stern is basically waterline flow; the streamlines are more or less horizontal along the skeg and into the propeller disk. The flow components along the steep buttock lines forward of the propeller disk are small. The dominant axial velocity field of the resultant wake has a substantial defect running vertically through the disk along its vertical centerline, at all radii.

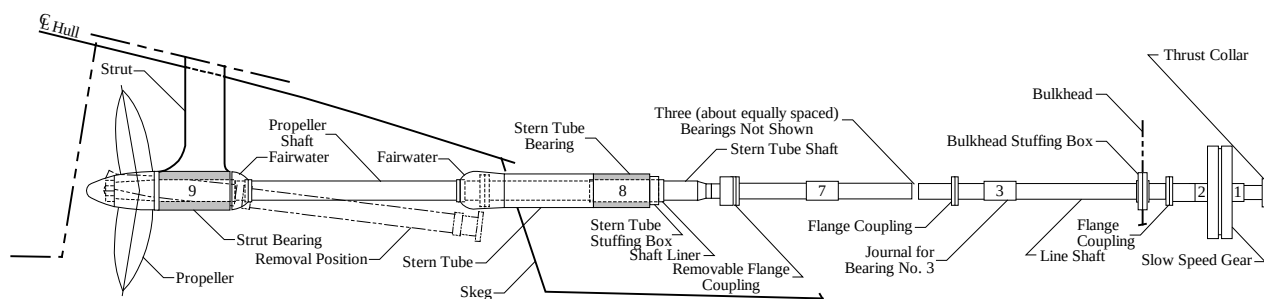
FIGURE 3
Nominal Wake Distribution for a Typical Merchant Ship (DTMB Model 4370, $C_B = 0.6$)



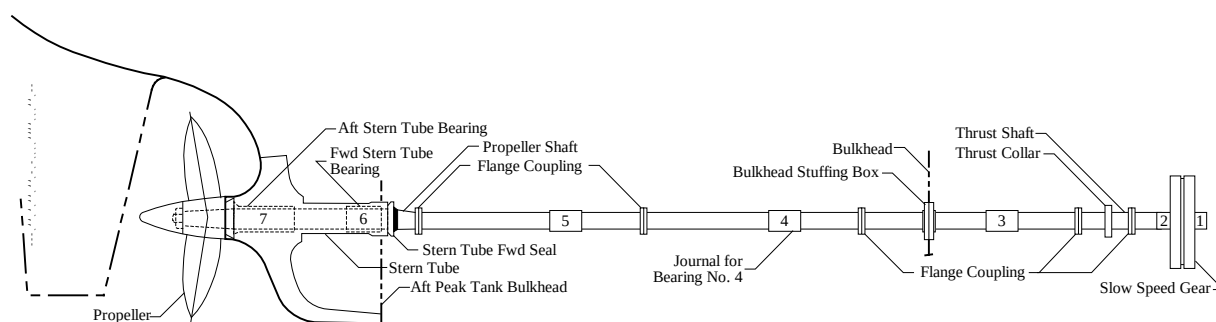
This defect is the shadow of the skeg immediately forward. The tangential flow in the propeller disk, being the combination of the component of the upward flow toward the free surface and any disk inclination relative to the baseline, is much smaller.

The idealization of this wake, for conceptual purpose, is the two-dimensional flow behind a deep vertical strut placed ahead of the propeller. In this idealization, the axial velocity distribution is invariant vertically, and any tangential velocity distribution (due to disk inclination) is asymmetric about the vertical disk axis. This basic characteristic is exhibited in the Section 3, Figure 3.

FIGURE 4
Alternative Shafting Arrangements:
Open Strut Stern (upper); Conventional Skeg Stern (lower)



Shafting Arrangement with Strut Bearing



Shafting Arrangement without Strut Bearing

A characteristically different wake flow is associated with the *strut or barge-type* stern, the upper of Section 3, Figure 4, which has a broad counter above the propeller disk and minimal irregularity immediately forward. The engine is further forward with this stern-type to accommodate finer stern lines needed to minimize wave resistance in high speed ships, although an open strut stern would be beneficial for vibration minimization at any speed (provided the buttocks lines are not too steep). The flow character over this type of stern is basically along the buttock lines, versus the waterlines. Some wake non-uniformity may be produced by appendages forward, such as struts and bearings or by shaft inclination, but the main wake defect, depending on the relative disk position, will be that of the counter boundary layer overhead. In this case, assuming minimal shaft inclination, substantial axial wake again exists, but only in the top of the disk. Generally, only the blade tips penetrate the overhead boundary layer, and the axial wake defect occurs only at the extreme radii near top-dead-center, rather than at all radii along the vertical centerline, as in the case of the conventional single screw stern. Just as in the case of the conventional stern, the tangential disk velocity with the strut stern will generally be small; the vertically upward velocity ratio through the propeller disk will have average values on the order of the tangent of the sum of the buttock and shaft inclination angles. The idealization in the case of the barge stern, as a sequel to the vertical strut idealization of the wake of the conventional stern, is a horizontal flat plate above the propeller. Here, the degree of axial wake non-uniformity depends on the overlap between the propeller disk and the plate boundary layer. The tangential (and radial) wake components are due entirely to the shaft relative inclination angle in this idealization, as the flat plate boundary layer produces only an axial defect.

5.1 Hull-Propeller Clearance

The distinction between the two basically different wake types is useful in understanding the importance of clearances between the propeller blades and local hull surfaces. First of all, it is helpful to consider the hull surface excitation by the propeller as composed of two effects:

- i) *Wake effect.* The effect of changing the wake inflow to the propeller according to a specified propeller relocation, but with the propeller actually fixed in position relative to the hull.
- ii) *Diffraction effect.* The effect of changing the propeller location relative to the hull, but with the wake inflow to the propeller held fixed.

It is a common misconception that the cruciality of propeller-hull clearances has to do primarily with the diffraction effect. To the contrary, analysis shows that for wake inflow held invariant, propeller-induced excitation level is relatively insensitive to near-field variations in propeller location. It is the high sensitivity of propeller blade pressure and cavitation inception to the variations in wake non-uniformity accompanying clearance changes that dictates the need for clearance minima. In general, the wake gradients become more extreme as propeller-hull clearances are decreased.

The conventional and strut-stern wake types are to be considered in light of the above fact. It is probable that too much emphasis is often placed on aperture clearances in conventional stern single-screw ships. From the point of view of vertical clearance, there is no significant boundary layer on the usually narrow counter above the propeller with this stern type. Furthermore, from the point of view of the vertical strut idealization of the skeg, the axial velocity distribution would be invariant with vertical disk position. The critical item with vertical tip clearance in the conventional stern case seems to be the waterline slope in the upper skeg region. Blunt waterline endings can result in local separation and substantially more severe wake gradients in the upper disk than suggested by the simple strut idealization; a “blunt strut” idealization then would be more appropriate. Fore-and-aft clearances in the conventional stern case are generally less critical than the vertical clearances. Wakes attenuate very slowly with distance downstream. While increasing the fore-and-aft clearances between the blade tips and the skeg edge forward certainly acts to reduce the wake severity, the reduction will be marginally detectable within the usual limits of such clearance variation. An exception would exist in the case of separation in the upper disk due to local waterline bluntness. The closure region of the separation cavity in that case exhibits large gradients in axial velocity.

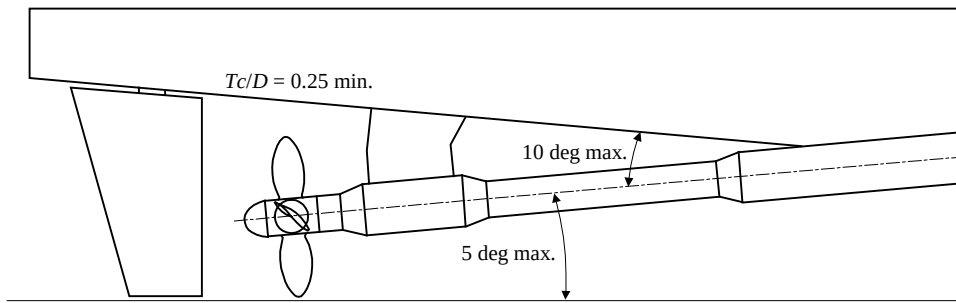
On the other hand, for the broad and flat countered strut stern vessel, the vertical tip clearance is a much more critical consideration. A relatively uniform wake will result if the propeller disk does not overlap the overhead boundary layer and the shaft inclination is moderate. This is the condition, in general, achieved on naval combatant vessels; the usual practice in naval design is a minimum vertical tip clearance of one-quarter propeller diameter. Vibration problems are almost unheard of on naval combatants.

Some wake non-uniformity on strut stern ships results from shaft struts and from the relatively high shaft inclinations often required to maintain the 25 percent overhead tip clearances. With proper alignment to the flow, shaft struts produce highly localized irregularities in the wake which are generally not effective in the production of vibratory excitation. The main effect of shaft inclination is a relative up-flow through the propeller disk, although separated flow from forward off of a highly inclined propeller shaft can produce a serious axial defect. The cavitation that can result at the 3 and 9 o'clock blade positions has shown to be of concern with regard to noise and minor blade erosion, but the hull vibratory excitation produced has not been found to be of much significance for strut stern ships.

The minimum vertical tip clearance of 25 percent of a propeller diameter is more or less accepted as the standard in commercial practice as well as naval. Consistent with the preceding analysis, the following lists the recommended configuration for stern arrangements in the order of preference for avoidance of excessive propeller induced vibration:

- i) *Single/Twin Screw Strut Stern*
 - The minimum vertical tip clearance is not to be less than 25% of the propeller diameter.
 - The shaft inclination angle relative to the baseline is not to be more than 5 degrees.
 - The shaft inclination angle relative to the buttocks angle of the counter is not to be more than 10 degrees. Refer to Section 3, Figure 5.

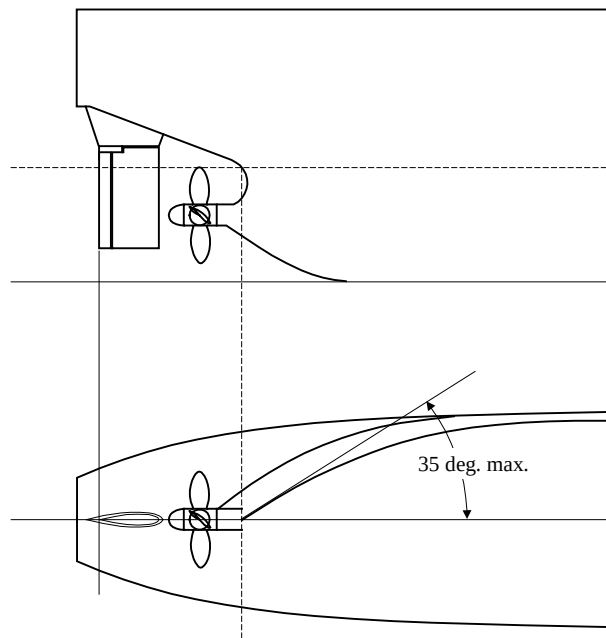
FIGURE 5
Open Strut Stern Arrangement



ii) *Conventional Stern with Skeg and Bossing.*

- The waterline angle from the vertical centerplane at the entrance to the aperture just forward of the top of the propeller disk is not to exceed 35 degrees. Refer to Section 3, Figure 6.

FIGURE 6
Conventional Skeg-Stern Arrangement



- With regard to the propeller tip clearance, the conventional skeg-stern ships are less critical than the strut-stern ships, as discussed earlier. In commercial practice, a minimum vertical tip clearance on order of 25% of propeller diameter and forward clearance of 40% of propeller diameter are often employed as a usual practice.
- iii) In the event that these limits cannot be achieved in concept design, it is recommended that the decision be made to proceed on to model testing and/or direct calculation for confirming or establishing stern lines.

7 Propeller

With an unfavorable wake, propeller compromises are then usually required to achieve compensation. Two types of excitation are of primary concern in conjunction with the three main items identified as critical in Subsection 2/5.

- i) Alternating thrust exciting longitudinal vibration of the shafting and machinery, and
- ii) Vertical pressure forces on the stern counter exciting hull and superstructure vibration.

7.1 Alternating Thrust

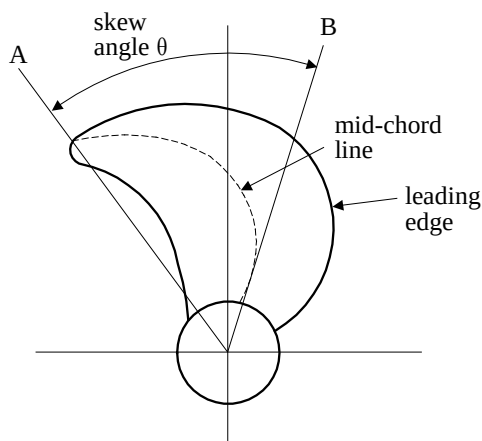
Alternating thrust, the excitation for longitudinal vibration of the shafting/main machinery system, occurs at blade rate frequency (Propeller RPM $\times$ Blade number N) and its multiples. The fundamental is usually much larger than any of its harmonics, however. Alternating thrust is produced by the blade number circumferential harmonic of the hull wake. This suggests that the higher the blade number the better, since the wake harmonic series does converge. However, around the typical blade number of 4, 5, and 6, the wake harmonic series convergence is not well organized, and is not a primary consideration. In fact, with the wake of a conventional single-screw stern, lower alternating thrust favors an odd-bladed propeller. This is because of, for an even bladed propeller, the line-up of opposite blades with the characteristic wake spike along the vertical center-plane above and below the propeller axis. This wake characteristic was discussed in Subsection 3/5 on hull wake. With strut-stern ships there is typically little blade number bias on the basis of wake.

Misconceptions exist about the effectiveness of propeller blade “skew” in reducing vibratory excitation. Skew is the tangential “wrapping” of the blades with radius. Positive skew is in the angular direction opposite to the direction of rotation. For the case of the conventional single screw ship wake, it has been discussed that the shadow of the vessel skeg produces a heavy axial wake defect concentrated along the disk vertical centerline. The blades of conventional propellers ray-out from the hub (i.e., the blade mid-chord lines are more or less straight rays emanating from the hub centerline). Such “unskewed” blades abruptly encounter the axial velocity defect of the conventional stern wake at the top and bottom-dead-center blade positions. The radially *in-phase* character of the abrupt encounter results in high net blade loads and radiated pressure.

A more gradual progression of the blades through the vertical wake defect is accomplished by curving the blades. Different radii enter and leave the wake spike at different times; cancellation results in the radial integrations to blade loads and radiated pressure, with the result of potentially significantly reduced vibratory excitation.

Percentage skew is the blade tip skew angle, relative to the blade ray through the mid-chord of the propeller hub section, divided by the blade spacing angle (e.g., for 5 blades the blade spacing angle is 72 degrees; therefore, a tip skew angle of 72 degrees would be 100% skew). See Section 3, Figure 7, where ray ‘A’ is passing through the tip of blade at mid-chord line and ray ‘B’ is tangent to the mid-chord line on the projected blade outline.

FIGURE 7
Maximum Skew Angle



Highly skewed propellers may be designed to have efficiency equivalent to propellers with minimal skew, but there have been problems with blade strength and flexibility. The modern trend has been a movement back toward more moderate positive tip skew, but with negative skew in the blade root regions. Here the percent skew may be defined in terms the total angle enclosed by the angular maxima of the blade leading and trailing edges. This provides blades with more balanced skew distributions, and highly *swept* leading edges. The radial load cancellations are achieved as effectively and blade strength is less problematic. The balanced skew is beneficial to limiting blade spindle torque with controllable pitch propellers, and is used widely with naval combatant vessels.

Skew will inherently work less effectively with strut-stern wakes since the axial velocity defect tends to be concentrated more in the outer extreme radii. The more radially uniform distributions of the conventional stern case are not available with which to achieve as high a degree of *dephasing* and radial cancellation. Of course, the strut stern vessel is in less need of propeller design extremes, as vibration problems are already largely eliminated by the stern form selection, provided proper clearances are incorporated.

Care must be taken in incorporating skew, particularly in replacement propellers, that adequate clearances between the blade tips and the rudder be maintained. As the blades are skewed in the pitch helix the tips move downstream, closing-up blade tip to rudder clearances. The consequences can be increased hull vibratory forces transmitted through the rudder, as well as rudder erosion caused by the collapsing sheet cavitation shed downstream off the blade tips as they sweep through the top of the propeller disk. The recourse is to incorporate warp into the blades along with the skew. Warping is a forward raking of the skewed blades back to (and sometimes beyond) the propeller disk. It is equivalent to skewing the blades in the plane of the disk rather than in the pitch helix.

It is noted that skew (and/or sweep) has a beneficial effect in reducing the effects of vibration-producing unsteady sheet cavitation, even when such cavitation may be concentrated in the blade tips. The blade curvature is thought to result in a radially outward flow component in the vicinity of the blade tips which tends to sweep the cavity sheets into the tip vortex, where the critical collapse phase occurs more gradually. Unsteady sheet cavitation is found to have minimal effect on alternating thrust, but can dramatically amplify the hull pressure force components. This is addressed specifically in 3/7.3.

If the guidelines offered on stern lines with regard to hull wake are adopted, then moderate and acceptable alternating thrust will result with propellers properly sized for propulsion considerations. This is without the need, particularly at the concept design stage, for detailed analysis to quantify alternating thrust level further along. Such quantification for detailed analysis is considered in Section 5.

Some ranges of the blade-rate alternating thrust amplitude are given below to serve as a reference for engineering decision making at the concept design level. The values are given as percentages of steady ahead thrust, and *assume that due care has been taken with stern lines and that the propeller is properly sized for proper propulsive performance*:

	<i>Alt Thrust/Steady Thrust</i>
Conventional Stern, moderately skewed/swept	0.02-0.06
Strut Stern	0.005-0.03

More details on propeller bearing forces and moments including alternating thrust for 20 real ship cases are given in Section 5. There is an outstanding issue here regarding proper sizing of the propeller, which is needed in the trade-offs at the concept design level. The old Burrill (1943) cavitation criterion is still useful in this regard. As stated, alternating thrust is not very sensitive to blade cavitation, but limiting the steady cavitation to accepted levels assures that the propeller is not overloaded from propulsive considerations, which is recommended as necessary to achieve the alternating thrust ranges listed above.

The Burrill chart is Section 3, Figure 8, which is a plot of a thrust loading coefficient at the 0.7 propeller blade radius, τ_c , versus the blade cavitation number, σ , also at the 0.7 radius.

$$\tau_c = \frac{T}{A_p q_T} \quad \text{and} \quad \sigma = \frac{p_a + \rho g h - p_c}{q_T}$$

where

- T = propeller thrust in kN
- A_p = propeller projected area in m^2
- q_T = $\frac{1}{2} \rho V_R^2$ in kPa
- V_R = relative velocity of water at 0.7 radius in m/sec
- p_a = absolute atmospheric pressure in kPa
- h = depth of the propeller axis in m
- p_c = cavitation cavity absolute pressure in kPa
- ρ = water density in ton/m^3

The relative velocity V_R at 0.7 radius is:

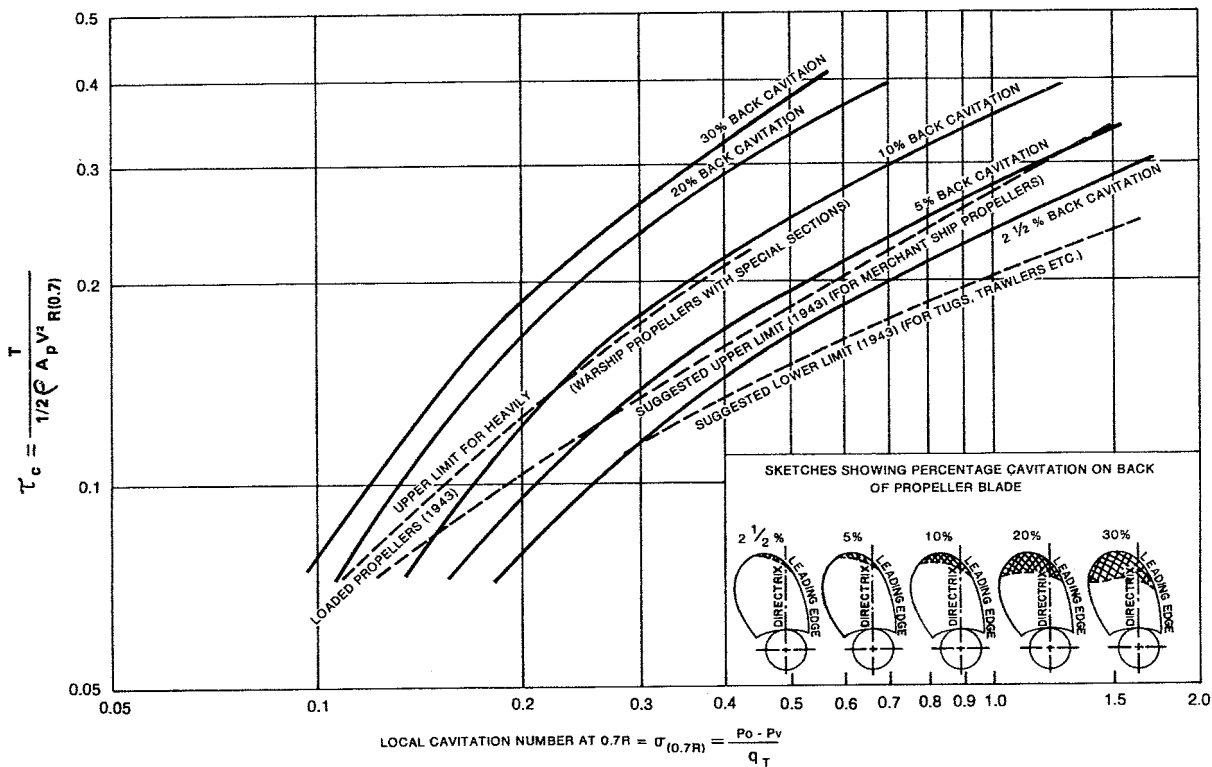
$$V_R(0.7R) = U(1-w) \sqrt{1 + \left(\frac{0.7\pi}{J} \right)^2}$$

where

- U = ship speed
- w = wake fraction
- J = design advance coefficient

The cavity pressure is often taken to be pure vapor pressure at the ambient water temperature, but for developed cavitation as is the case here, versus inception, p_c is higher due to dissolved air in the water; p_c around 6-7 kPa is often assumed. The curves of Section 3, Figure 8 correspond to the degree of steady cavitation development. The 5% back cavitation line was suggested as the limit for merchant ship propellers back in 1943. However, for modern commercial ship propellers, which are universally designed with aerofoil-type blade sections, the 10% back cavitation line is probably more appropriate. But either the 5% or 10% lines, considering the cavitation developments shown, may serve the purpose to avoid overloading the propeller in steady propulsion.

FIGURE 8
Burrill Cavitation Inception Chart



An example of the use of the Burrill chart for this purpose is as follows:

- Take a single screw ship with a delivered power $P_d = 22,380$ kW and a max rated speed $U = 22$ knots, corresponding to a thrust, $T = 1,400$ kN. The propeller diameter has been set at 6 meters consistent with tip clearance maxima. The $PRPM = 120$ and the wake fraction is estimated as $w = 0.2$. The pitch ratio at the 0.7 radius, $P/D]_{0.7R} = 0.9$. This gives an advance coefficient, $J = 0.755$ and resulting q_T of 399.3 kPa. For the hub centerline depth $h = 4.5$ m, $p_a = 101.3$ kPa, and $p_c = 6.9$ kPa, $\sigma = 0.35$. If the 5% back cavitation line is selected, $\tau_c \cong 0.16$. With the disk area $A_0 = \pi D^2/4$, the required projected area ratio would be:

$$\frac{A_p}{A_0} = \frac{4T}{\pi \tau_c q_T D^2}$$

- Substituting the values, $A_p/A_0 = 0.775$. It is preferable to work with the developed area ratio, A_d/A_0 , where A_d denotes propeller developed area with zero pitch. Taylor's approximate formula gives the projected area in terms of the developed area as:

$$\frac{A_p}{A_d} \cong 1.067 - 0.229 \frac{P}{D} \Big]_{0.7R}$$

Then the required developed area ratio is:

$$\frac{A_d}{A_0} = 0.90$$

- The projection here is then, with a good wake, as to be achieved by following the provided recommendations in these Guidance Notes, the subject propeller blades may not be heavily loaded in the mean, and the alternating thrust is to be in approximately direct proportion. As an alternative, if it was decided to use a less conservative approach and accept steady cavitation to the 10% back cavitation at $\sigma = 0.35$, $\tau_c \cong 0.2$. Then the alternative developed area ratio would be:

$$\frac{A_d}{A_0} = 0.72$$

Of course, the more conservative propeller, with the larger blades, will have lower propulsive efficiency. The following is recommended in concept design, in addition to the preceding stern lines recommendations relative to hull wake, for achieving not excessive alternating thrust from the propeller design consideration:

- Perform the Burrill cavitation calculation demonstrated above with a blade area ratio as needed to limit the steady cavitation up to 10% back cavitation.
- For a conventional stern, a 5-bladed propeller is favored in reducing the alternating thrust, unless any harmful resonant vibration is anticipated on superstructure, shafting system or local structures.
- In general, incorporate blade skew of no more than 50%, or if a blade with a highly swept leading edge is adopted (negative skew in the root), use no more than 25 degrees of tip skew (for either stern type). For a highly skewed propeller design other than the foregoing, refer to 4-3-3/5.5 of the *ABS Steel Vessel Rules*.

7.3 Hull Pressure Forces

The dominating excitation for ship hull vibration, related to the third of the three critical items identified in the preceding, is propeller cavitation-induced hull surface pressure forces. If intermittent blade cavitation does not occur to a significant degree, then the main excitation of the hull is via the shafting system and the main engine, as already discussed. These sources will cause minor problems relative to, at times, the intense vibration from an intermittently cavitating propeller. It was in the mid-70's that propeller blade unsteady cavitation, triggered by the wake non-uniformity, was found to be the main culprit in most of the ship vibration troubles.

The sheet cavitation expands and collapses on the back of each blade in a repeating fashion, revolution after revolution. The sheet expansion typically commences as the blade enters the region of high wake in the top part of the propeller disk. Collapse occurs on leaving the high-wake region in a violent and unstable fashion, with the final remnants of the sheet typically trailed out behind in the blade tip vortex. The sheet may envelop almost the entire back of the outboard blade sections at its maximum extent. For large ship propellers, sheet average thicknesses are on the order of 10 cm, with maxima on the order of 25 cm occurring near the blade tip just before collapse.

The cavitation, while of catastrophic appearance, is usually not deleterious from the standpoint of ship propulsive performance. The blade continues to lift effectively; the blade suction-side surface pressure is maintained at the cavity pressure where cavitation occurs. The propeller bearing forces (i.e., alternating thrust) may be largely unaffected relative to non-cavitating values for the same reason. The cavitation may or may not be erosive, depending largely on the degree of *cloud cavitation* (a mist of small bubbles) accompanying the sheet dynamics. The devastating appearance of fluctuating sheet cavitation is manifest consistently mainly in the field pressure that it radiates, and the noise and vibration that it thereby produces. The level of hull surface excitation induced by a cavitating propeller can be easily an order of magnitude larger than typical non-cavitating levels. Vertical hull surface forces due to intermittent cavitation typically exceed propeller bearing forces by large amounts.

The cavitation-induced hull surface force is, like the bearing forces, composed of the fundamental blade-rate frequency and its harmonics. However, unlike the character of the alternating thrust discussed in the preceding, the harmonics of the cavitating surface force typically converge very slowly. The fundamental, at PRPM times blade number, is usually largest, but twice and three times blade-rate of the same order typically exists, and can resonate structure and generate noise well above the normal excitation frequency range.

Cavitating vibratory force reduction is achievable with propeller design refinements, as sheet cavitation dynamics is sensitive to blade geometric, as well as wake, detail. As previously noted, blade skew can be beneficial, and tip loading can be reduced to reduce cavitation extent and intermittency by local blade tip pitch reduction, as well as changes in blade tip chord distributions. But measures of this type do reduce propeller efficiency and thereby compromise propulsive performance.

The occurrence of propeller blade intermittent cavitation cannot be ignored in attempts to control propeller induced hull vibration. The issue here is again primarily a hull wake issue. With an unfavorable wake, characterized as exhibiting high gradients in the upper disk, or even a separation pocket, the resulting large cavity volume variations can produce hull surface force amplitudes of up to 30-40% of the steady thrust, with significant levels out to several harmonics of blade-rate frequency. But even with a good wake, as described in the preceding, the conventional stern ship will still exhibit a wake defect in the disk over the depth of the skeg, and some intermittent cavitation will be unavoidable. A vertical surface force with a blade-rate amplitude of 15% of steady thrust is probably a conservative reference for a conventional stern ship with a “good wake” as achievable by the stern lines guidance of the preceding section.

With strut stern vessels with ample tip clearances and low relative shaft inclination, intermittent cavitation, or any cavitation at all, might be avoided entirely. This is largely achieved in the case of radiated noise sensitive naval combatant vessels.

Prediction of the cavitating hull forces is much more complicated, with many more factors contributing, than is alternating thrust, which, as stated, is more or less insensitive to cavitation. Cavitating hull force analysis is certainly not an activity for concept design. However, with any conventional stern ship of modern power level, a propeller cavitation/hull pressure assessment program is to be initiated in the preliminary design stage. For the direct calculation of the propeller-induced hull surface force, refer to the vibration analysis procedure described in Section 5.

At concept design, which is the scope of interest here, the same analysis as in the preceding subsection, using the Burrill criteria for 5-10% steady back cavitation, is to be applied to provide reasonable protection against under-sizing the propeller blades from the standpoint of unsteady cavitation. The example of the preceding subsection on alternating thrust is therefore proposed as applicable here as well.