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SHAFT STRUTS

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161-1-b. Introduction

This design data sheet is issued to provide a standard procedure for the design of surface ship main and intermediate shaft struts. The types of struts considered are twin-arm radial and tangential main struts, twin-arm radial and tangential intermediate struts, and single-arm radial intermediate struts.

Data are presented only for the Elliptical, Parabolic, Hyperbolic (EPH) section. Other strut sections may be used if permitted by the Ship Specifications, but the strut properties such as section area, moment of inertia, natural frequency, and column strength would then have to be calculated separately for that strut section.

The paragraphs below discuss considerations pertinent to design of shaft struts. Although actual loadings on the struts cannot be accurately predicted, the empirical loading formulas included herein reflect probable effects of the more important variables involved in design of the ship and of the struts themselves. Loadings will be determined from these formulas unless other requirements are stated in Ship Specifications.

For most designs, struts can be proportioned from figures provided herein, without extensive calculations. Occasionally it may be desirable to make more thorough analyses of strength and stiffness. Formulas and data are provided for this purpose.

161-1-c. Definitions

Shaft strut. A support for a shaft bearing outside of the hull, consisting of a strut barrel and one or two strut arms.

Base chord. The axis of symmetry of a strut-arm cross section.

Strut-arm axis. A line, along the length of the arm, through mid-points of the base chords of the cross sections.

Neutral axis of strut arm. A line joining the neutral axes of successive sections which are subjected to longitudinal bending moments. Normally it is just forward of the strut-arm axis and parallel to it.

Vee angle. The angle between the axes of the two arms of a shaft strut.

Strut barrel. The juncture of the strut arms outside the ship, or the housing for the shaft bearing. It may be constructed by welding two castings together to form a cylinder. Its size should be the minimum which provides clearance for removal of shafting or bearings.

Strut-arm twist angle. The angle between the base chord of a strut section, at any designated point along the arm, and a plane which is both normal to a ship transverse section and passes through the strut-arm axis. The angle is measured in a plane normal to the strut-arm axis (see paragraph 161-1-h).

Strut-arm cant angle. The angular deviation of the plane defined by the strut-arm axes, from a plane transverse to the ship (see paragraph 161-1-i).

Strut palm. The inboard end of the strut arm, flattened and spread out for attachment to hull structure. Generally it is welded to longitudinal framing within the hull and the shell plating.

161-1-d. Symbols and abbreviations

Symbol	Parameter	Unit
A	section area of strut arm	square inches (in^2)
a_p	propeller overhang, or the distance from a plane through the 0.7 propeller radius at the maximum propeller blade thickness to the plane of the neutral axes of the strut arms, measured along the shaft centerline	inches (in)
a_s	distances from shaft centerline to a given section of a single-arm strut	in
a_T	distance from shaft centerline to point of intersection of neutral axes of tangential strut arms	in
a_x, a_y	distances from loads P_x and P_y, respectively, to neutral axes of strut arms	in
b	fore and aft dimension of palm	in
c	base chord of strut-arm section	in
c_L, c_T	distances from point of maximum bending stress to neutral axes for longitudinal and transverse bending, respectively	in
d	depth of palm	in
D_p	diameter of propeller	in
D_s	outside diameter of shaft	in

Symbol	Parameter	Unit
E, E_s	moduli of elasticity for strut arms and shaft, respectively	thousands of pounds per square inch (ksi)
F_a, F_w	natural frequencies of strut arms in air and water, respectively	cycles per minute (c/min)
F_c	column strength (stress at failure from axial loading)	ksi
F_y, F_{ys}	tensile yield strengths for strut arms and shaft, respectively	ksi
f_B	maximum stress from combined longitudinal and transverse bending moments	ksi
f_c	stress from axial loading	ksi
f_L	maximum stress (at trailing edge) from longitudinal bending moment	ksi
f_s	maximum shearing stress (from torsional moment)	ksi
f_T	maximum stress from transverse bending moment	ksi
f_x, f_y	stresses from unit components of load or moment, or both (subscripts correspond to those of P and M)	ksi/kip, or ksi/in-kip
G	modulus of elasticity in shear	ksi
H	shaft horsepower per shaft	horsepower (hp)
I_L, I_T	moments of inertia of cross section of strut arm about a neutral axis normal to the base chord and about the base chord, respectively	in^4
I_s	moment of inertia of cross section of shaft (about axis through centerline)	in^4
I_1, I_2, I_3	moments of inertia of cross sections of single-arm strut at the strut barrel or hub, at midpoint, and at hull, respectively (about base chords)	in^4
K_6, K_7	coefficients (subscript refers to number of figure showing values)	-
L	length of strut arm from hull to point of intersection of strut-arm axes	in
L_s	length of shaft between supports	in

Symbol	Parameter	Unit
ℓ	distance from outboard end of strut arm (attachment to barrel) to point of intersection of strut-arm axes.	in
M, M_1, M_2 etc.	applied moments	in-kip
M_L, M_T	moments acting on cross section of strut arm longitudinally and transversely, respectively	in-kip
M_x, M_y	components of applied moment associated with P_x, P_y, respectively	in-kip
N	full power shaft speed	revolutions per minute (r/min)
P, P_3, P_4 etc.	applied forces	kip
P_x	applied force at shaft centerline, normal to the plane which bisects the vee angle (for single-arm strut, normal to the plane of the axis and the base chord of the section in question)	kip
P_y	applied force parallel to bisector of vee angle	kip
P_z	applied force at shaft centerline normal to plane of strut-arm axes	kip
r_T	radius of gyration of strut-arm section about base chord	in
t	maximum thickness of strut-arm section	in
V	maximum speed of ship	knots (kt)
W_p	weight of propeller	kip
w, w_s	unit weight of strut arm and shaft, respectively	kip/in
x	distance from leading edge of strut-arm section	in
y	distance from base chord to curve of strut-arm section	in
Z_L	section modulus for trailing edge of strut-arm section under longitudinal bending moment	in^3

Symbol	Parameter	Unit
Z_S	section modulus of shaft	in^3
Z_T	section modulus of strut-arm section under transverse bending moment	in^3
θ	half of vee angle	degrees
τ	strut-arm twist angle (see paragraphs 161-1-c and 161-1-h)	degrees
Φ	strut-arm cant angle (see paragraphs 161-1-c and 161-1-i)	degrees

161-1-e. General design principles

Shaft struts must be designed for satisfactory performance in several, quite different respects. The relative importance of the various considerations depends, to some extent, upon the ship, its mission, and its characteristics. The design procedure outlined in succeeding paragraphs is based upon the following principles:

1. Struts must have adequate strength to support the shaft bearings under normal operating conditions up to full power. Loadings may result from any unbalanced centrifugal or hydrodynamic force, and from inertial forces under ship motion, as well as from gravity acting on propellers and shafting. Vibratory forces may be quite large during a crash-back maneuver. Since stresses pass through many cycles, they must be kept well below the fatigue limit of the material.

2. A suitable margin of safety must be provided to allow for possible damage to the propeller or shafting. Under such conditions, unbalanced thrust and centrifugal forces will be greatly increased. In heavy weather the propeller may be exposed to pounding by the sea, with strut loadings far in excess of normal forces and moments.

3. Struts must be rigid enough to support the shaft bearings properly. Their stiffness is a factor in determining the vibratory characteristics of the shafting. In addition, consideration must be given to possible vibration of the struts themselves.

4. The maximum practical resistance to such extraordinary loadings as grounding or underwater explosion (shock loads in accordance with reference (a)) should be provided. This is dependent primarily on the ultimate strength of the whole shaft and strut system. Stresses under normal operating conditions are not necessarily an accurate measure of ultimate strength under emergency loadings.

5. Resistance and interference with flow of water to the propellers shall be kept to a minimum. A principal cause of ship vibration is the variation in flow over the propeller disk. The struts must not unduly aggravate this condition.

6. Struts shall be designed and constructed to limit the destructive effects of erosion caused by cavitation or by rapid flow of water past the strut arms. A smooth, well faired surface is required to minimize erosion, noise, and resistance. Paragraph 161-1-v describes aids for use in fairness inspections.

161-1-f. Arrangement of strut arms

On most ships fitted with shaft struts, each shaft is supported by a main strut at the propeller and an intermediate strut. Location of the bearings is determined by design considerations for the shafting (see reference (b)). If the intermediate strut bearing is very close to the hull, this strut may consist of a single arm. More often, both main and intermediate struts are of the vee type, each having two arms of approximately equal length.

Generally the axes of the arms of vee struts are radial to the strut barrel. Sometimes, however, particularly if the angle between arms is small, it may be desirable to provide greater separation of the arms at the strut barrel, and the outer surfaces of the arms may be made tangential to the strut barrel. This introduces additional transverse bending moments on the arms. Occasionally a compromise may be made between radial and tangential arrangements.

Longitudinally, the main struts should be placed well forward on the strut barrel. This is done, in spite of increasing longitudinal bending stresses, in order to provide sufficient clearance between the strut arms and the propeller, and thus minimize vibration caused by interaction of strut and propeller blades. The normal distance between the strut arm trailing edge and the 0.7 radius at maximum propeller blade thickness should be 0.35 times propeller diameter.

Intermediate struts are generally placed so that the strut-arm axes are approximately in the transverse planes passing through the centers of the bearings. Some variation is permissible if attachment of strut palms to hull structure is thereby facilitated.

161-1-g. Vee angle selection

Vee angles near 90 degrees have maximum strength, considering loadings in all directions, other things being equal. However, somewhat smaller angles ordinarily give shorter strut arms, which is desirable from the standpoint of strength, stiffness, resistance, and economy of material. If the vee angle is too small, stresses will be high. Moreover, unnecessary constriction of flow at the hub shall be avoided.

The possibility of aeration shall be considered in selecting the location of the upper end of strut arms. It is important to keep that juncture with the hull submerged at all likely operating conditions. If the strut were to pierce the free-water surface it could result in air being drawn down, adversely affecting propeller performance. Typical submergence values are shown in paragraph 161-1-r.

An additional consideration in selecting the vee angle for main struts is the angle between propeller blades. If the vee angle and blade angle coincide, any disturbance of flow will affect two blades simultaneously and accentuate forces tending to vibrate the ship.

For many ships 65 degrees is a satisfactory vee angle.

161-1-h. Twist angle selection

For high-speed ships (20 knots or greater) the direction of flow in way of the struts will generally be determined from model tests so that twist angles can be chosen to minimize angles of attack. Figure 1 shows this twist angle, which can in general vary over the entire length of the strut arm.

For slower ships (less than 20 knots) twist angles may be approximated from study of the lines plan. Flow lines generally lie between waterlines and buttock lines, tending to converge aft because of inflow to the propeller.

161-1-i. Cant angle selection

To simplify the geometry of the design, the strut arms are usually located in a transverse plane. Because of shaft rake, clearance toward the propeller blade tips is then increased over that at the hub. Pressure of the shaft on the bearing tends to slide the barrel along the shaft, but this is not a significant factor. Occasionally the strut arms are canted to be normal to buttock lines, further increasing clearance. Figure 2 shows this cant angle. This minimizes the lengths of the arms, but increases the component of bearing pressure which is normal to the plane of the arms. Because canting the strut arms reduces their lengths, it also changes their natural frequencies. Hence, canting can be used to vary frequency.

161-1-j. Cross section of strut-arm

The EPH section is a streamlined section having satisfactory strength and cavitation characteristics. Figure 3 gives offsets in terms of maximum thickness. The equations of the EPH section are:

ellipse: $$\left(1 - \frac{x/c}{0.43613}\right)^2 + 4(y/t)^2 = 1; \quad 0 < x/c < 0.43613$$

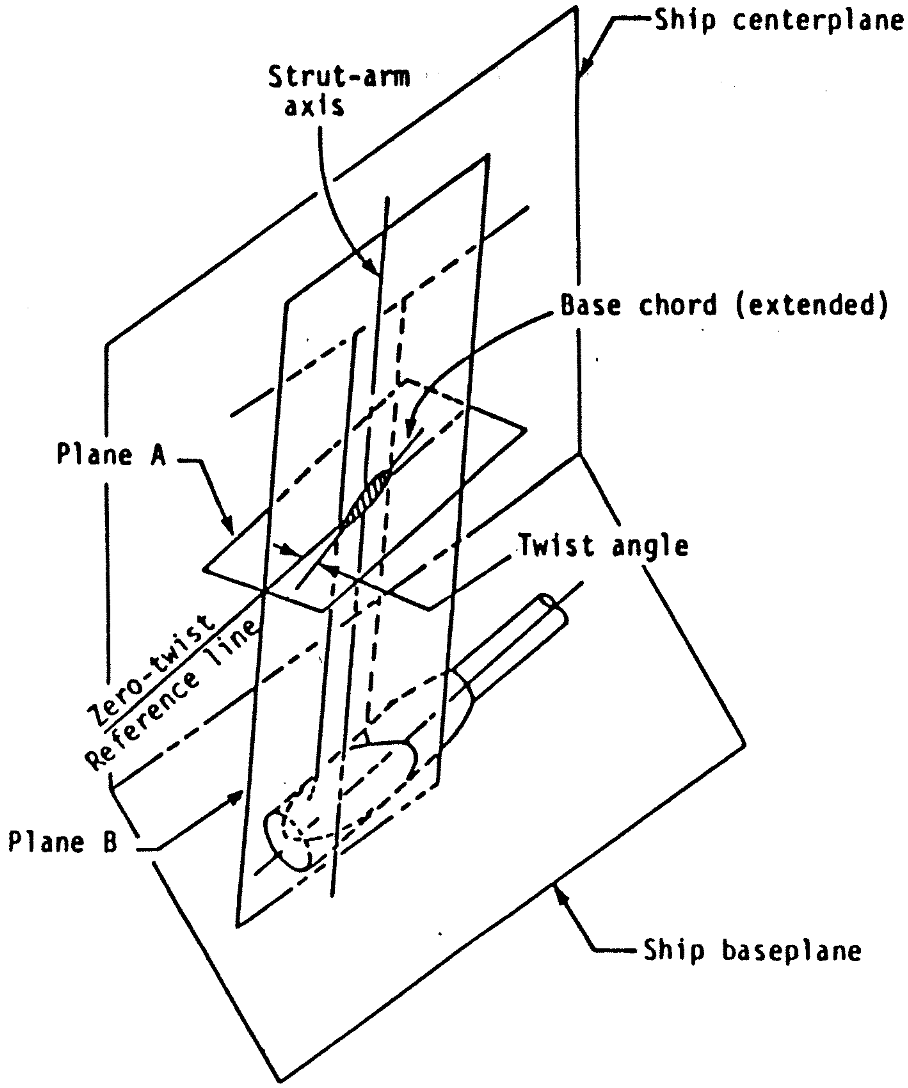
parabola: $$0.5\left(\frac{x/c}{0.43613} - 1\right)^2 + 2(y/t) = 1; \quad 0.43613 < x/c < 0.87226$$

hyperbola: $$\left(1 + \frac{1 - x/c}{0.30839}\right)^2 - 16(y/t)^2 = 1; \quad 0.87226 < x/c < 1$$

Principal properties affecting strength and rigidity of the struts, which are given by figure 4, were computed from the following formulas:

Sectional area, $A = 0.747\ ct$
Properties about base chord:
Moment of inertia, $I_T = 0.0449\ ct^3$
Section modulus, $Z_T = 0.0898\ ct^2$
Radius of gyration, $r_T = 0.245\ t$
Properties about neutral axis normal to base chord:
Location of neutral axis, from leading edge, 0.4735 c
Moment of inertia, $I_L = 0.0443\ c^3t$
Section modulus (for trailing edge), $Z_L = 0.0842\ c^2t$

Strut arms are usually made solid to minimize dimensions for a given strength. A hollow section is lighter, however, especially if it must be



1. Twist angle is measured in Plane A, which is normal to the strut-arm axis.
2. Plane B passes through the strut-arm axis and is parallel to the intersection of the ship centerplane and the ship baseplane.
3. The Zero-twist reference line is the intersection of Planes A and B.
4. Twist angle is the deviation of the strut section's base chord from the Zero-twist reference line.

Figure 1. <u>Strut-arm twist angle</u>

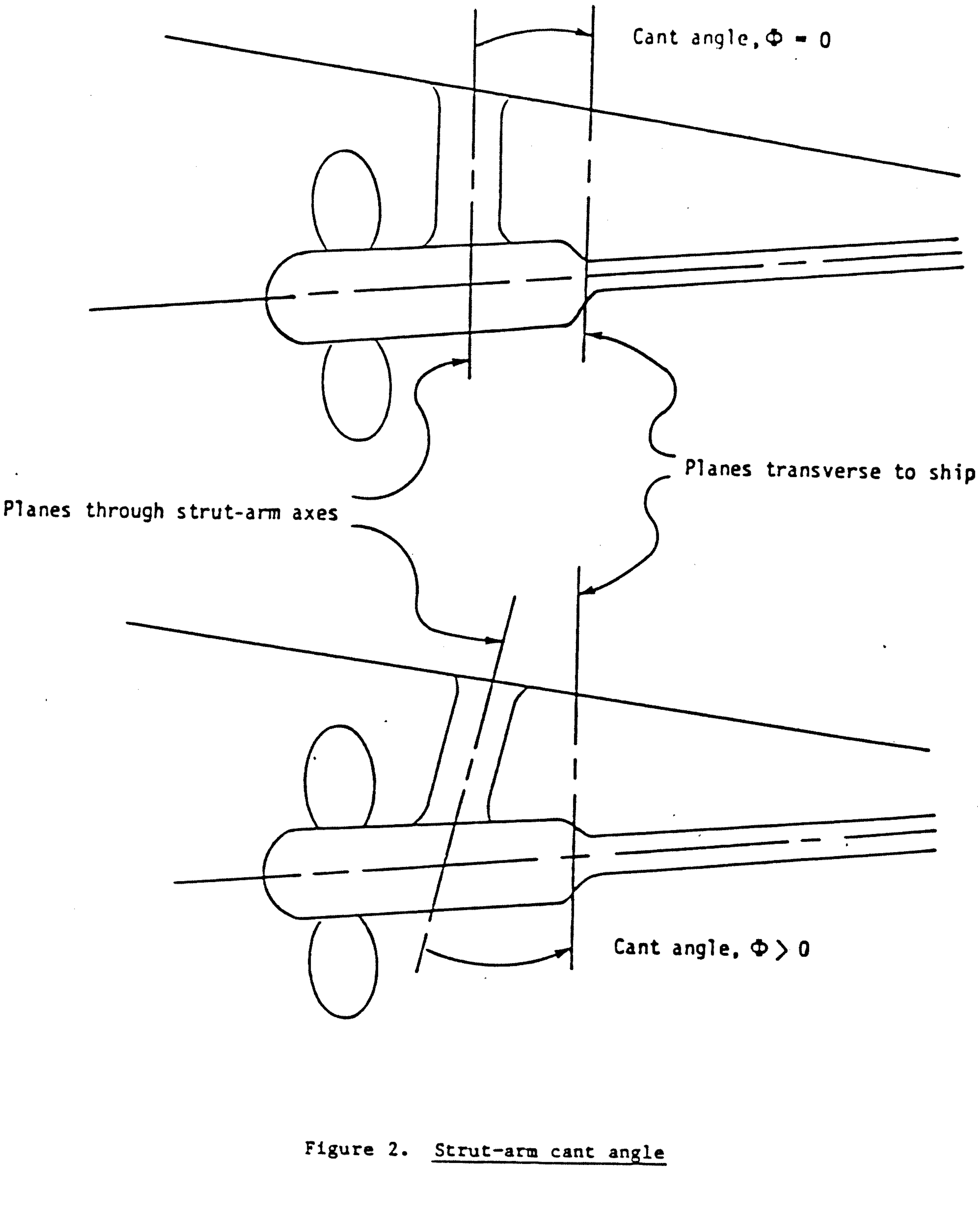
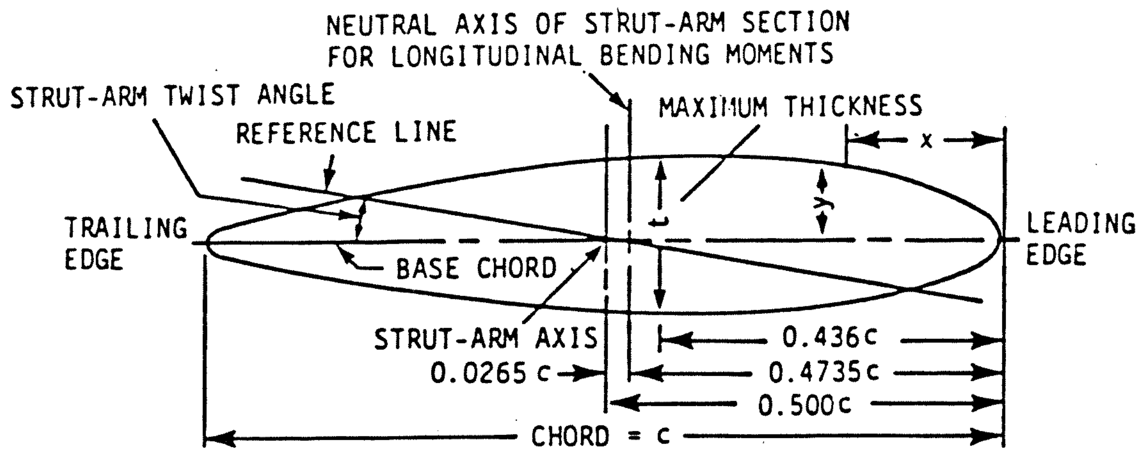


Figure 2. Strut-arm cant angle



The reference line for twist angles lies in a plane normal to a ship transverse section and is perpendicular to the strut-arm axis. If the axis is in a transverse plane, the reference line is fore and aft. Offsets for the EPH section are as follows:

x/c	y/t	x/c	y/t	x/c	y/t
.01	.1064	.35	.4902	.75	.3705
.025	.1669	.40	.4983	.80	.3260
.05	.2325	.45	.4997	.85	.2749
.10	.3186	.50	.4946	.90	.2170
.15	.3774	.55	.4830	.95	.1480
.20	.4204	.60	.4647	.975	.1027
.25	.4522	.65	.4399	.99	.0642
.30	.4750	.70	.4085		

Leading edge radius: $R_1 = 0.5732(t^2/c)$
Trailing edge radius: $R_2 = 0.2027(t^2/c)$

Figure 3. EPH strut-arm section

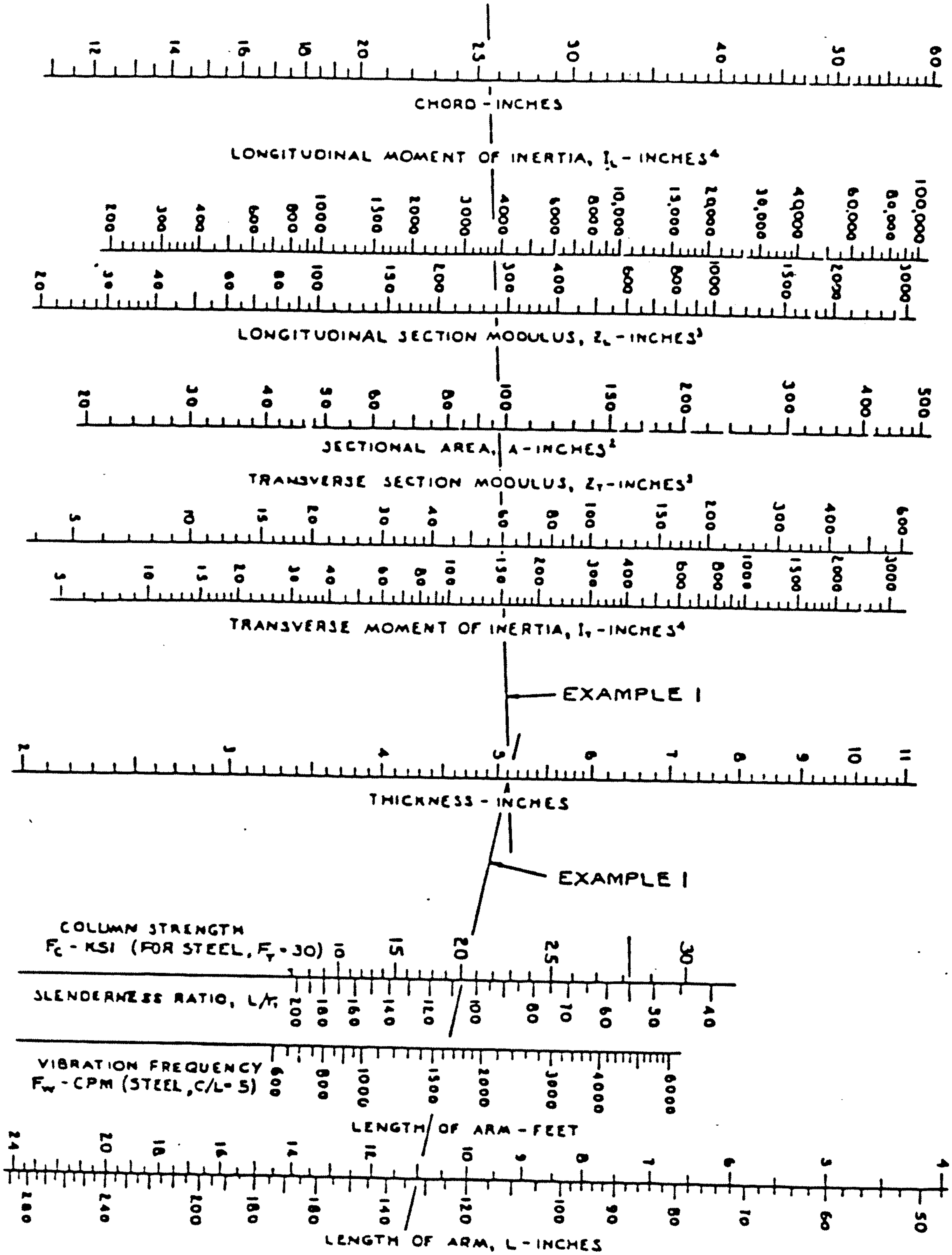


Figure 4. Properties of EPH section

designed for any considerable transverse bending moment. In designing for a specified vibration frequency, a hollow section is particularly advantageous, permitting reduced thickness as well as substantial weight savings. Hollow (air-backed) sections are subject to being crushed by underwater shock pressure waves. If strut arms are not solid, they must either be designed to withstand 3,000 lb/in^2 or be filled with an incompressible fluid.

161-1-k. Chord-thickness ratio selection

Selection of the chord-thickness ratio is a convenient way to commence sizing the strut. A good initial ratio is 5. This generally provides a reasonably-sized strut with good hydrodynamic characteristics.

As chord-thickness ratio (c/t) increases, the strut arms become lighter for a given longitudinal bending strength. Theoretically, their tendency to cavitate may also be reduced at small angles of attack. However, a thin section has less strength under transverse bending or column loads. It is more subject to erosion and corrosion damage. Furthermore, since lines of flow are changed by wave action and maneuvering, appreciable angles of attack must be allowed for, even where twist angles are accurately established for ideal conditions. A chord-thickness ratio (c/t) of at least 4.3 but no greater than 6 is required for vee struts. Radial vee struts require sections thick enough to prevent undue vibration or column failure. The arms of tangential struts must also resist considerable transverse bending moments. For a single-arm strut it is advantageous to have thickness increase with distance from the shaft, since the strut is loaded as a cantilever. The arms of vee struts are usually uniform in cross section throughout their lengths, except that fillets are provided at both hub and hull to minimize stress concentrations and improve flow. However, no strut arm, whether single- or twin-arm configuration is used, should have a chord-thickness ratio less than 4.3. This limit is established by construction considerations and because thicker sections may cause undue flow separation.

161-1-l. Stresses in radial vee struts

Any loading which is applied to the strut via the shafting and bearings can be resolved into a force normal to the shaft at the neutral axes of the strut-arms and a moment in a plane through the shaft centerline. Frictional forces tending to rotate the strut barrel are neglected. The strut arms are assumed fixed at the hull and rigidly joined at the strut barrel. For simplicity, the two strut arms are considered to be symmetrical about the bisector of the vee angle, and both twist and cant angles are initially assumed to be zero. The formulas below give stresses resulting from the various components of force and moment.

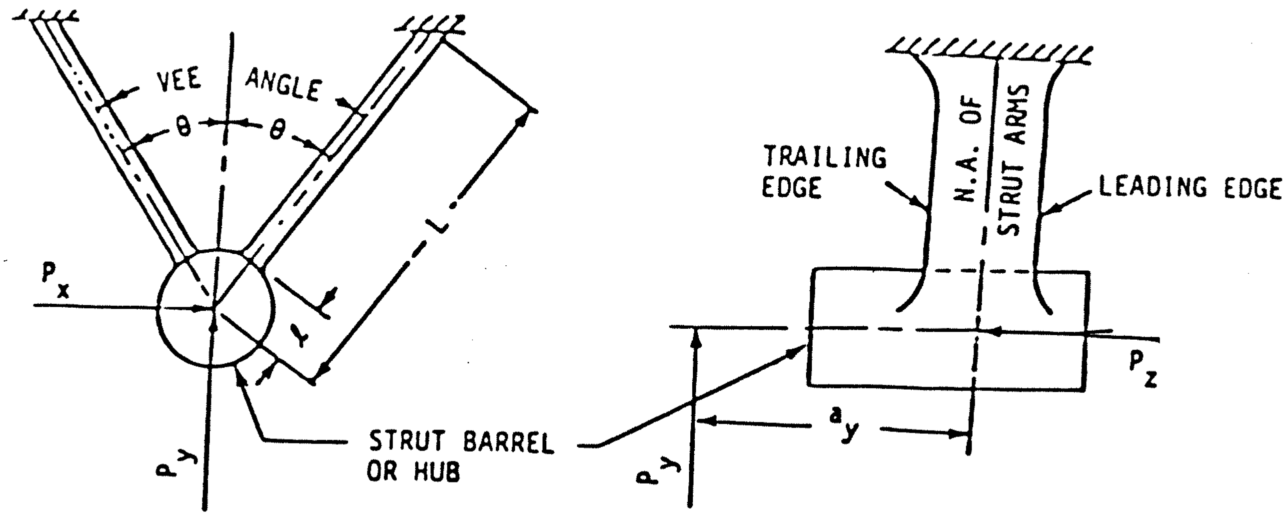


Figure 5. Loadings on radial vee strut

1. Load normal to plane bisecting vee angle

Axial stress, $f_c = P_x/2A \sin\theta$, compression, P_x acting toward arm being considered.

Longitudinal bending stress, $f_L = K_6\, M_x/Z_L \sin\theta$, compression in trailing edge at strut barrel. Figure 6 is used to determine K_6. $M_x = P_x\, a_x$, with P_x acting toward arm being considered, at a distance a_x aft of the neutral axes of the strut-arms.

$$K_6 = \frac{2 - (\ell/L) - (\ell/L)^2}{4[1 + (\ell/L) + (\ell/L)^2]}$$

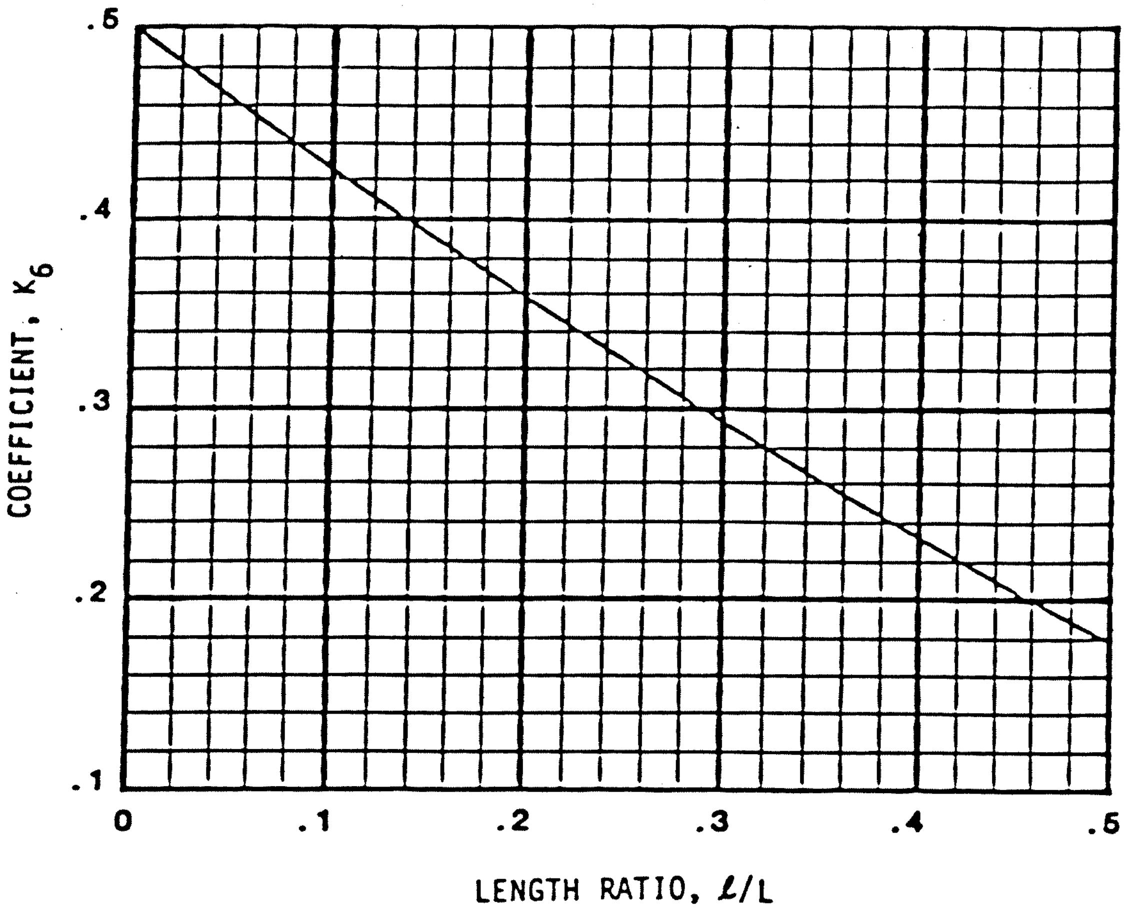
2. Load parallel to bisector of vee

Axial stress, $f_c = P_y/2A \cos\theta$, compression, P_y acting toward strut-arms.

Longitudinal bending stress, $f_L = M_y/2Z_L \cos\theta$, compression in trailing edges. $M_y = P_y\, a_y$, with P_y acting toward arms, at a distance a_y aft of the neutral axes of the strut arms.

3. Load normal to plane of neutral axes of strut-arms

Longitudinal bending stress, $f_L = P_z\, \ell/2Z_L$, compression in trailing edges at strut barrel, with P_z directed aft at the point of intersection of the strut-arm neutral axes. At hull, $f_L = P_z L/2Z_L$.



$$f_L = \frac{K_6 M_x}{2\ Z_L \sin\theta} \qquad f_T = \frac{K_6 P_x a_T}{Z_T}$$

Figure 6. Coefficient for determining longitudinal bending stress due to moment M_x, in radial and tangential struts, and transverse bending stress in tangential struts

4. Total stress

Combining the effects of the above loads:

$$f_c + f_L = \frac{P_x}{2 \sin \theta} \left(\frac{1}{A} + \frac{2K_6 a_x}{Z_L}\right) + \frac{P_y}{2 \cos \theta} \left(\frac{1}{A} + \frac{a_y}{Z_L}\right) + \frac{P_z \ell}{2Z_L}$$

If P_x and P_y are components of a force, P, of constant magnitude and variable direction in a plane parallel to that of the strut arm axes (P_z negligible):

maximum bending stress, $f_B = P \sqrt{f_x^2 + f_y^2}$

where f_x and f_y are stresses (at the same location) caused by unit values of P_x and P_y, respectively. Similarly, under moment alone:

$$f_B = M \sqrt{f_x^2 + f_y^2}$$

Under usual loadings the effects of the axial forces are comparatively small and may be neglected for design purposes. Figure 7 shows coefficients giving maximum moment on the strut section for a given moment applied to the hub. Under longitudinal moment alone:

maximum bending stress, $f_B = K_7\ M/Z_L$, where

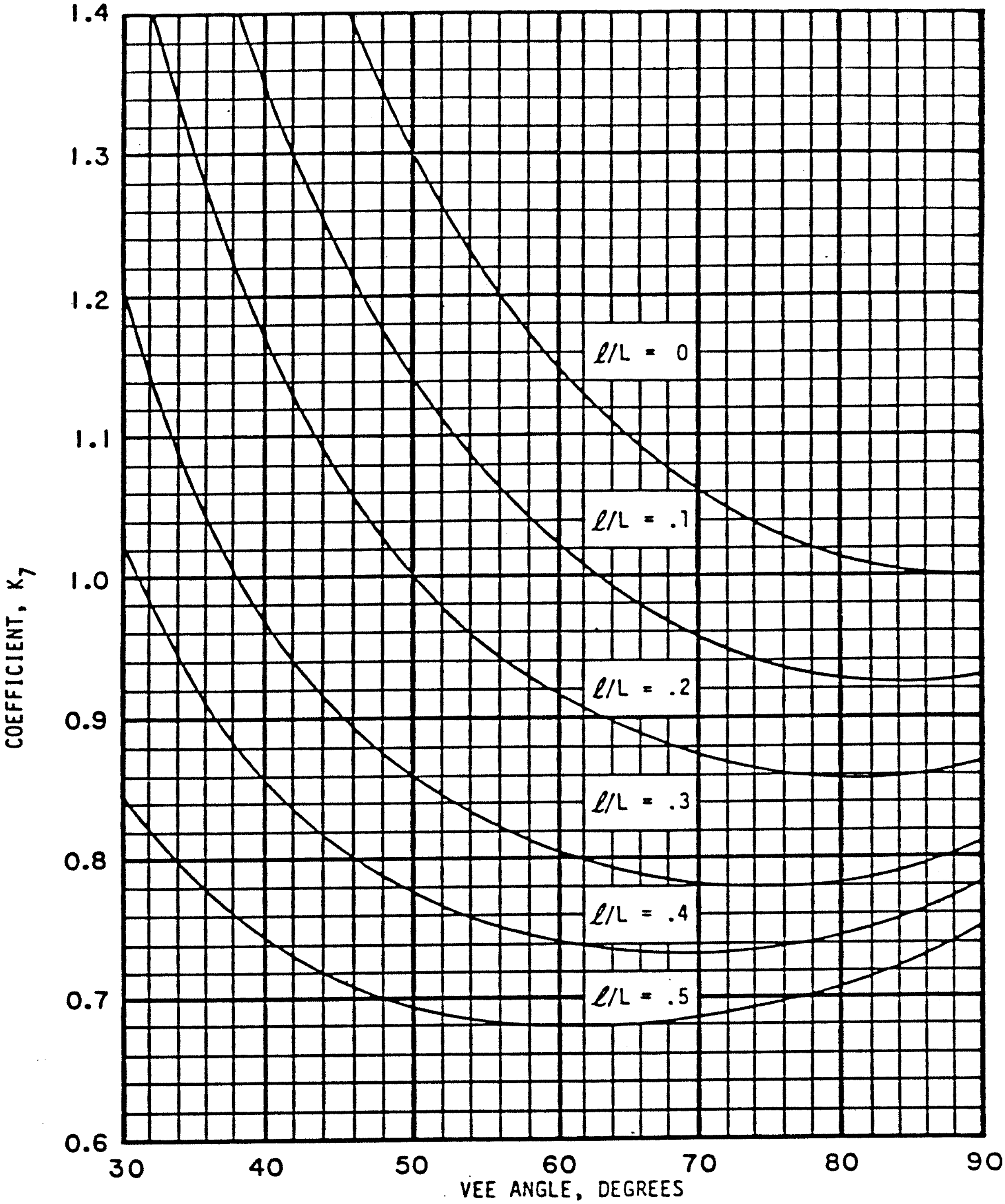
$$K_7 = \sqrt{(K_6/\sin \theta)^2 + (0.5/\cos \theta)^2}$$

5. Effect of strut twist

To examine the effect of strut twist, let the total longitudinal bending moment for the untwisted strut be M, and let the twist angle be τ at some given point along the length of the strut arm. The moment M can be resolved into longitudinal and transverse components M_L and M_T, respectively, in the twisted strut as follows:

$$M_L = M \cos \tau$$
$$M_T = M \sin \tau$$

The neutral axis of the section subject to combined moments is as shown in figure 8.



$f_L = K_7M/Z_L$ Required $Z_L = K_7M/F_y$

Figure 7. Coefficients for determining maximum moment on arm of vee strut subjected to longitudinal bending

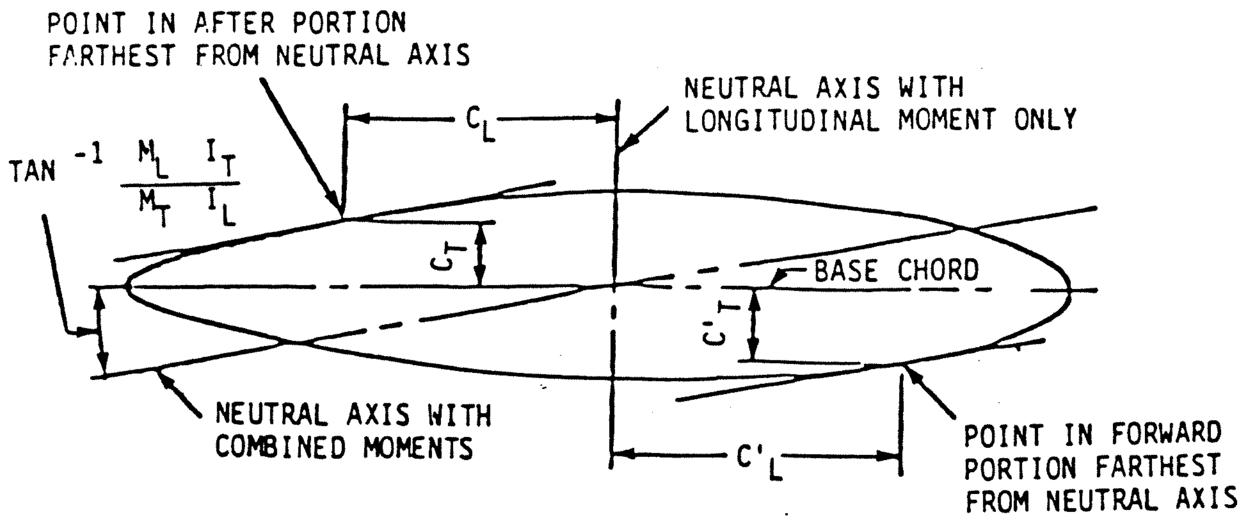
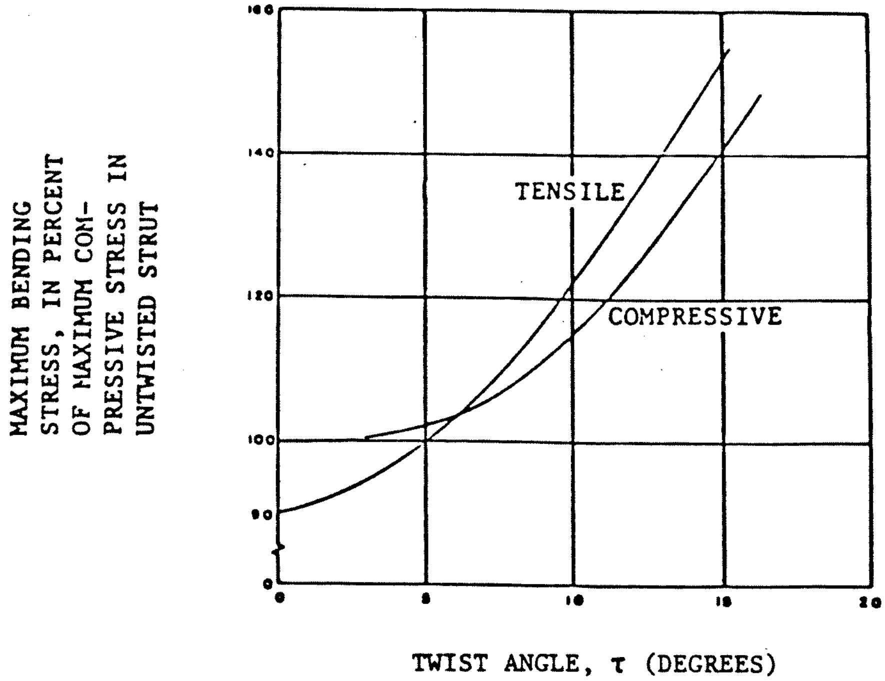


Figure 8. Strut-arm section under combined longitudinal and transverse bending moments

The angle which defines the neutral axis is a function of the ratio of the moment components and the ratio of the longitudinal and transverse moments of inertia of the section. Thus the location of the point of maximum bending stress depends upon the ratio of the moment components and the shape of the section. Figure 8 shows how points most distant from the inclined neutral axis can be determined. Alternatively, the points for maximum bending stress can be found by calculating the separate stresses due to longitudinal and transverse components of bending moment for selected points along the boundary of the section, and then adding the separate stresses, considering whether the stress components are tensile or compressive. The locations and magnitudes of the maximum bending stresses can then be found by interpolation.

A strut arm with variable twist should be strengthened by using the following procedure. This procedure is followed for twist angles of 5, 10, and 15 degrees, using an EPH section with length-to-chord ratio of 5. For an assumed load P as shown in figure 5, the maximum bending stress in an untwisted or uniformly twisted strut is a compressive stress at the trailing edge. The maximum tensile stress occurs at the leading edge, but its magnitude is about 10-percent less than the maximum compressive stress. Figure 9 shows how both the maximum compressive and tensile bending stresses increase with twist angle. Note that figure 9 shows the maximum bending stresses (whether tensile or compressive) in terms of percent of the maximum compressive stress for the untwisted strut. For twist angles of 5, 10, and 15 degrees, maximum tensile stresses occur at points about 5, 14, and 20 percent of the chord from the leading edge, respectively, and maximum compressive stresses occur at points about 95, 86, and 70 percent of the chord from the leading edge. At a twist angle of 6 degrees the maximum tensile and compressive stresses are about equal in magnitude, and are about 4-percent greater than the maximum stress in the untwisted strut.



$$\% \text{ TENSILE} = 90 + 0.30\tau + 0.38\tau^2 - 0.008\tau^3$$

$$\% \text{ COMPRESSIVE} = 100 - 0.567\tau + 0.18\tau^2 + 0.003\tau^3$$

Figure 9. Variation in maximum bending stress with strut-arm of variable twist angle

For a twist angle of 7 degrees, which is probably the largest that need be considered, the maximum stress is a tensile stress, and it is about 8 percent greater than the maximum compressive stress in the untwisted strut. As shown by the expression for total bending stress in paragraph 161-1-m, the total stress can be reduced by a given percentage by increasing both the longitudinal and transverse section moduli by that same percentage. This can be accomplished by increasing both the chord and thickness of the section by the factor $[1 + (\text{percent increase in stress due to twist}/100)]^{1/3}$. To achieve an 8-percent decrease in stress, the factor is $(1.08)^{1/3} = 1.026$.

It is not apparent that stresses due to column loading along the axis of a strut arm are affected by twist. The effect of eccentric loading would be modified slightly by twist. Although this DDS does not explicitly consider eccentricity, it may be assumed that its effects are implicitly allowed for in the empirical design loads given in paragraph 161-1-t.

6. Effect of canted strut

If a strut is canted at an angle Φ, as shown in figure 2, a load P (figure 5) introduces additional longitudinal bending moments in the strut arms, both at the strut barrel and at the hull. For calculating the longitudinal bending stress in the trailing edges at the strut barrel, using the formulas given in paragraphs 161-1-1-1. and 2., above,

$M_x = P_x(a_x + \ell \sin \Phi)$, and
$M_y = P_y(a_y + \ell \sin \Phi)$

At the hull, the additional compressive stress in the trailing edges can be calculated using the moment

$$M_L = (P_x/2 \sin\theta)(a_x + L \sin\Phi) + (P_y/2 \cos\theta)(a_y + L \sin\Phi)$$

161-1-m. Stresses in tangential vee struts

If the struts do not enter the strut barrel radially, the load normal to the bisector of the vee angle will cause transverse bending stresses. Figure 10 shows the strut configuration and loadings.

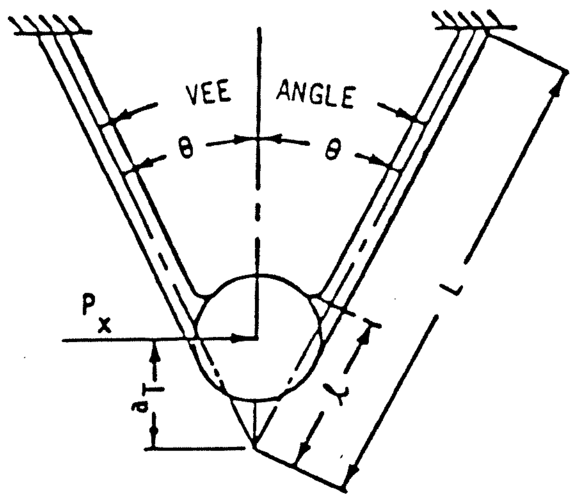


Figure 10. Loadings on tangential vee strut

At the hub, at greatest distance from base chord: maximum transverse bending stress, $f_T = K_6 M_T/Z_T$ with K_6 as shown in figure 6, and $M_T = P_x a_T$.

For axial and maximum longitudinal bending stresses, the equations of paragraph (1) are applicable.

The most adverse direction of loading can be approximated from the following relationships: transverse bending is greatest when $P_x = P$; longitudinal bending is greatest when $M_x/M_y = 2K_6/\tan\theta$.

As discussed in paragraph 161-1-1, with moments about both axes, the neutral axis is as shown in figure 8, where it is shown how points most distant from the inclined neutral axis can be determined.

$$M_L = K_6 M_x/\sin\theta + M_y/2\cos\theta;\ M_T = K_6 P_x a_T$$

At a point c_L from the neutral axis for longitudinal bending and c_T from the base chord:

$$\text{total bending stress, } f_B = \frac{c_L}{I_L}\left(\frac{K_6 M_x}{\sin\theta} + \frac{M_y}{2\cos\theta}\right) + \frac{c_T}{I_T}(K_6 P_x a_T)$$

If the strut is twisted, an additional component of transverse bending moment will be introduced, as discussed in paragraph 161-1-1, and it must be taken into account in determining the position of the inclined neutral axis. The effect of cant also can be taken into account as discussed in paragraph 161-1-1. Both forward and aft maxima must be considered. Toward the leading edge, axial stresses are opposite to longitudinal bending stresses, assuming that moment is caused by a load aft of the strut.

161-1-n. Ultimate strength of struts

Under extreme conditions, such as grounding, it may be assumed that plastic deformation of the strut arms occurs and the full bending strength of both shaft and strut are developed. Under such conditions, ultimate failure might occur through column action of a strut arm. Under a load of constant magnitude and variable direction in the plane of the neutral axes of the strut arms, column stress is greatest in one arm when the applied force is normal to the other. It is equal to P/A sin 2 θ. Figure 4 shows column strength for steel EPH struts, based on the data of Reference (c).

For tangential struts the transverse bending stress must be considered. It may be assumed that failure takes place when $f_c/F_c + f_T/F_y = 1$.

161-1-o. Stiffness of struts and shafts

Moments to rotate the hub of a vee strut through one radian are as follows (neglecting torsional stresses in the arms):

In a plane normal to the bisector of the vee angle:

$$M_x = \frac{8EI_L \sin^2 \; [1 + (\ell/L) + (\ell/L)^2 \;]}{L[1 - (\ell/L)]^3}$$

In a plane of the bisector of the vee angle:

$$M_y = \frac{2EI_L \cos^2 \theta}{L}$$

The corresponding stiffness of the shaft is somewhat less than $4\ E_s\ I_s/L_s$, the value for fixity at the intermediate strut. In general, I/L will be much greater than I_s/L_s. Although clearance and deformation of the bearing strips will permit the shaft to move through a slightly larger angle than the hub, most of the moment applied to the shaft will normally be transferred to the strut.

161-1-p. Vibration frequency of strut arms

The lowest vibration frequency of the arms of a vee strut is for a mode in which the two arms, bending in their own plane, vibrate in phase about the point of intersection of their axes. For a beam with simple support at one end and fixity at the other:

$$\text{natural frequency} = \frac{2890}{L^2}\sqrt{EI/w} \quad \text{cycles/minute}$$

Using $E = 29{,}600$ ksi (for steel) and $I = 0.0449\ ct^3$ and $w = 0.000212\ ct$ kip/in (for EPH section):

$$F_a = 724 \frac{t}{(L/100)^2} \quad \text{cycles/minute, in air}$$

Water has the effect of increasing the mass of the arm. Adding the mass of a cylinder of sea water of diameter equal to chord length, assuming $c/t = 5$:

$$F_w = 560 \frac{t}{(L/100)^2} \quad \text{cycles/minute, in water}$$

Figure 4 gives frequencies based on this formula. Chord-thickness ratio values of 4 and 6 give constants of 580 and 540, respectively.

161-1-q. Single-arm struts

Single-arm struts of considerable length have comparatively little strength and rigidity. Under load normal to the base chord:

transverse bending stress, $f_T = P_x\, a_s/Z_T$

where a_s is the distance from the load (i.e. centerline of bearing) to the section in question.

The stiffness of a cantilever of variable cross section can be determined by finding the moment of the area of the M/EI diagram about the point of load. Applying Simpson's rule, the following formula results:

$$\text{Stiffness} = \frac{6E}{(L - \ell)\left[\frac{\ell^2}{I_1} + \frac{(L+\ell)^2}{I_2} + \frac{L^2}{I_3}\right]} \quad \text{kip/in}$$

In torsion, stiffness is given by the approximate formula for the moment per radian:

$$M_x = \frac{4GI_T}{(L - \ell)\left[1 + \frac{(4r_T)^2}{c}\right]}$$

For $G = 12{,}000$ ksi, an EPH section with $c/t = 4$ gives:

$M_x = 45{,}300\ I_T/(L - \ell)$ in-kip per radian.

Other values of c/t, unless unusually low, give results only slightly different. The constants are 43,300 and 46,200 for $c/t = 3$ and 5, respectively.

Shear stress in a single-arm strut under torsional moment is approximated as:

$$F_s = M_x/2Z_T,$$

where M_x is the moment actually applied to the hub. In this case, the shaft may be the stiffer member.

Forces and moments in a fore-and-aft plane through the strut-arm axis will have comparatively small effects on stresses.

161-1-r. Connection of strut arms to hull structure

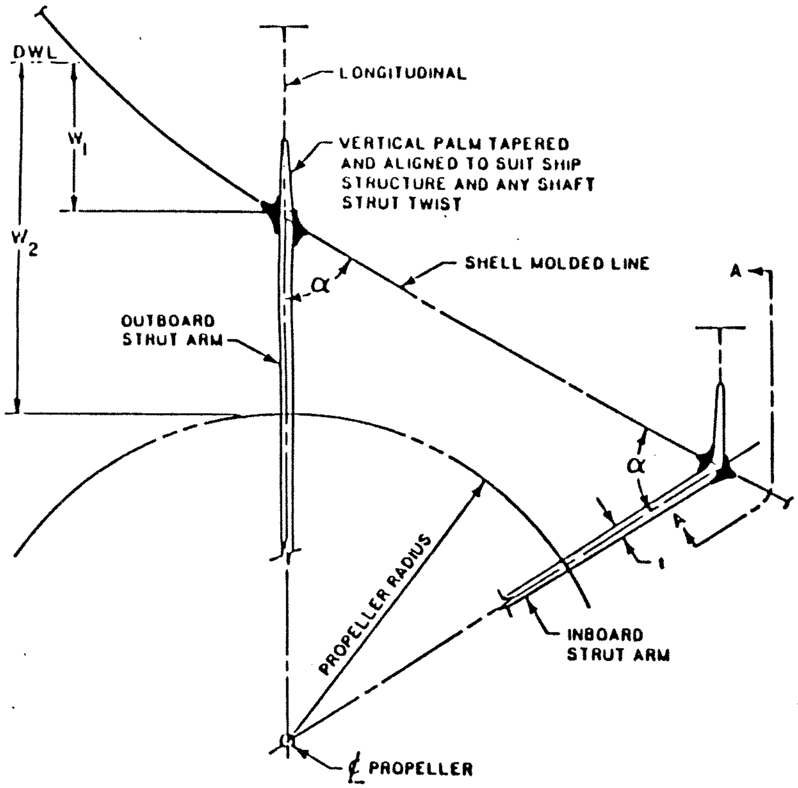
Insofar as practicable, hull structure supporting a strut-arm shall be designed to develop the full strength of the arm. On a steel ship the palms of the two arms of a vee strut are often made approximately parallel to facilitate insertion into the hull after assembly. They shall be attached to longitudinal members whose section moduli (including shell plating) are capable of carrying the moment applied to the strut arms.

The palms are tapered fore-and-aft and vertically, leaving edges thick enough for satisfactory welding. Strength of the connection shall equal the value of AF_c for the arm. With palms a foot or so longer than the strut-arm chord, longitudinal bending strength of the connection will generally be adequate. This may be checked by the approximate formula: stress, in kip/in = $Z_L\ F_y/(0.3\ b^2 + 1.5\ bd)$. This assumes reasonably uniform strength of weld around the periphery of a roughly rectangular palm.

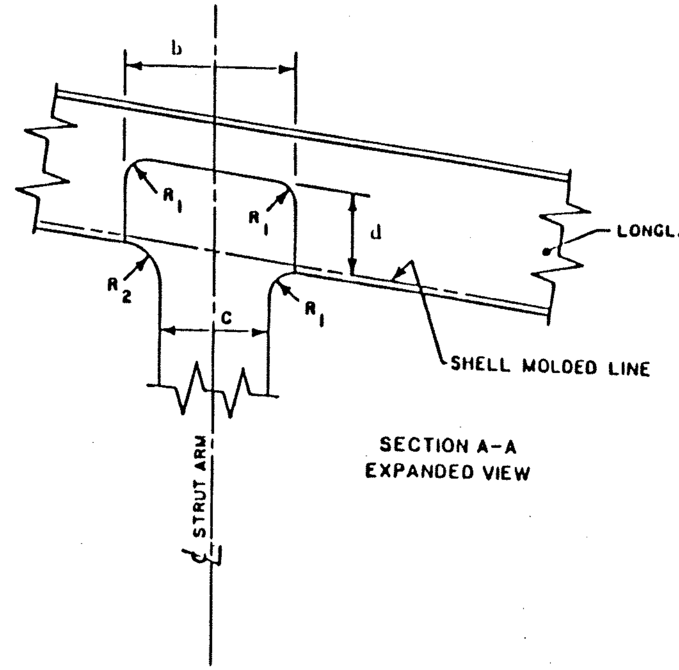
Hull structure shall also be capable of resisting transverse bending moments. This is especially important for tangential, vee struts and, of course, for single-arm struts. For the former, the theoretical moment at the hull is half that at the hub. However, it is desirable to provide for the full value of $Z_T\ F_y$.

Filleting at the connections between hull and strut arms has both structural and hydrodynamic implications. From a structural standpoint, after the scantling requirements of this section have been met by the hull plus internal structure, filleting is required only to provide for gradual transition of stress. From a hydrodynamics standpoint, the hull-strut arm juncture is in boundary layer flow, with its relatively low velocity. Strut arms which are within 45 degrees of being perpendicular to the hull do not require filleting for flow conditions. At more acute angles, filleting or fairing on the inside of the acute angle shall be provided. Typical arrangements of a single palm configuration are shown in figure 11.

On wooden-hull ships, such as minesweepers, the palms are flat against the shell, through-bolted to the exterior of the planking. For such installations it may not be practicable to make the attachment as strong as the strut. However, it should be possible to apply loadings such as those applicable to tangential struts without overstress.



Transverse section, port side, looking fwd



Section A-A, expanded view

<u>Parameters</u>

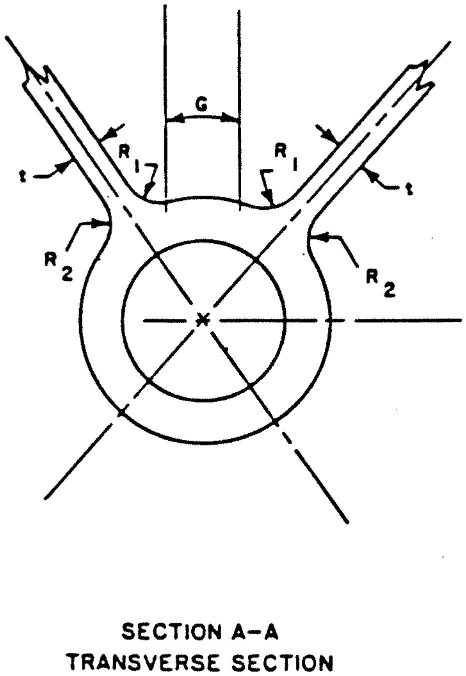
c = Strut arm chord
t = Strut arm maximum thickness
W_1 = Vertical distance from design waterline (DWL) down to juncture of strut arm and hull
W_2 = Vertical distance from design waterline down to top of propeller disc
α = Acute angle between hull and strut arm

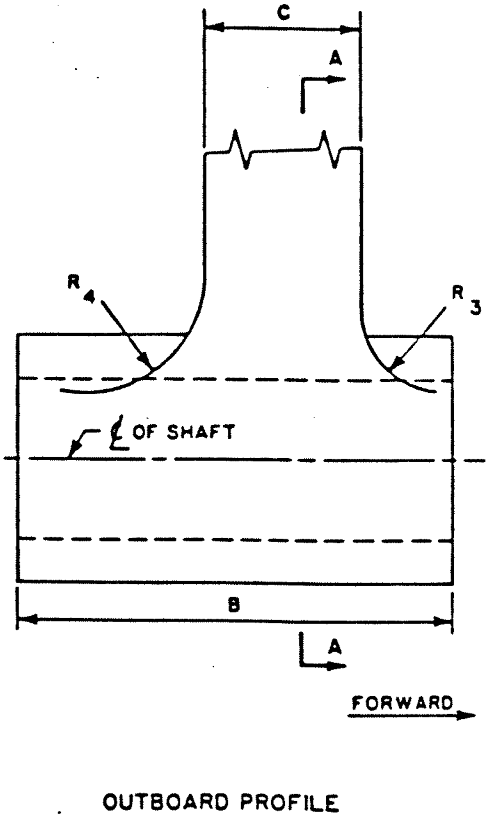
<u>Typical values</u>

For $\alpha \leq 45°$, a fairing shall be installed on the side of the strut arm having that acute reentrant angle. Fairing radius should be $\geq$ 2t.

R_1 between 0.12c and 0.25c
R_2 between 0.3c and 0.6c
d between 0.6c and 1.0c
b between 1.2c and 1.8c
W_1 between $0.2W_2$ and $0.5W_2$

Figure 11. <u>Strut-arm attachment to hull structure</u>





B = Shaft strut barrel length
c = Strut-arm chord
G = Gap between strut arms (measured toe to toe of fillets R_1 on arc along outside of strut barrel)
t = Strut-arm maximum thickness
R_1, R_2 = Radius at strut arm to strut barrel intersection (measured in a transverse plane, at the inner and outer sides, respectively)
R_3, R_4 = Radius at leading and trailing edges of strut arm to strut barrel intersection, respectively (measured in the plane of the strut-arm axis of symmetry)

Typical values are:

$R_1 \geq t$
$R_2 \geq t$
$R_3 \geq t$; typically $\frac{B - C}{4}$

$R_4 \geq t$; typically $\frac{B - C}{2}$

$G \geq t$

Figure 12. Typical fillet radii at connection of strut arms to strut barrel

161-1-s. Connection of strut arms to strut barrel

The connection between a strut arm and strut barrel is usually filleted for both structural and hydrodynamic reasons. In terms of structure, fillets provide for gradual stress transition and also improve end-fixity conditions. In terms of hydrodynamics, the use of fillets provides more uniform flow boundaries, thereby reducing flow irregularities. Figure 12 shows typical values for cases where the strut arms and strut barrel are fabricated as a one-piece casting.

An alternative type of construction that is sometimes selected by a shipbuilder involves welding the strut arms into the strut barrel. In such cases it is appropriate to have less filleting than shown in figure 12, since the deposited weld would form part of the filleting. Such designs require special consideration during review of a shipbuilder's ship construction drawings, and the specific scantlings plus fabrication procedures. Another alternative type of construction is to weld the strut arms into the stub strut arms on the strut barrel.

In some cases the design may call for tangential arm struts, such as shown in figure 10. The fillet radius where the inside of each arm attaches to the barrel is typically equal to or greater than the maximum thickness of the strut arm. The radii at the forward and after ends of the strut arm connections to the strut barrel would be in accordance with foundry practice for the specific dimensions.

161-1-t. Empirical design criteria

Unless otherwise specified these criteria may be used for design.

1. Moments for main struts. Stress in the main strut arms should not exceed yield strength under either of the following longitudinal moments (the formulas include factors of safety on probable loading conditions):

a. A moment (M_1) caused by a centrifugal force assumed to act at the propeller centerline in a plane parallel to the strut-arm axes, together with an unbalanced thrust normal to the plane of the strut-arm axes.

$$M_1 = 0.0035\, D_p [W_p\, a_p (N/100)^2 + 3\, H/V]$$

This represents roughly one-fourth of the moment that would result from complete loss of one blade of a four-bladed propeller with the ship going at full speed.

b. A moment (M_2) in any plane normal to that of the strut-arm axes, proportional to the bending strength of the propeller shaft.

$$M_2 = 0.3 Z_s\, F_{ys}$$

2. Forces for main struts. The main struts shall also be designed to avoid failure of the arms under the following loadings, applied at the shaft centerline, in any direction in the plane of the neutral axes of the strut arms:

a. Loading for radial main struts, a force related to the bending strength of the shaft.

$$P_3 = Z_s \; F_{ys}/D_s$$

b. Loading for tangential struts, a force proportional to the weights of propeller and shafting supported:

$$P_4 = 10W_p + 5w_s \; L_s$$

3. Moment equation for all intermediate struts:

$$M_5 = 0.12Z_s \; F_{ys} \text{ (using same } Z_s \text{ as for main strut)}$$

4. Force for intermediate radial vee struts:

$$P_6 = 0.6Z_s \; F_{ys}/D_s$$

5. Force for intermediate tangential vee struts and single-arm struts:

$$P_7 = 10w_s \; L_s$$

6. Strut natural vibration frequency should be at least 20 percent above propeller blade frequency. Propeller blade frequency is the product of shaft speed (in r/min) and the number of blades. If achieving this natural frequency is impractical, first-mode resonance shall occur below 50 percent of full power and second-mode resonance shall occur above 120 percent of full power.

161-1-u. Sample calculations

Propeller and shaft data:

Number of propeller blades		=	4
Propeller diameter (13.5 ft)	D_p	=	162 in
Propeller weight	W_p	=	19.5 kip
Propeller overhang	a_p	=	70 in
Propeller speed	N	=	290 r/min
Shaft horsepower	H	=	35,000 hp
Ship speed	V	=	30 kt
Outside diameter of shaft	D_s	=	18.5 in
Inside diameter of shaft at after end of bearing		=	4.87 in
Inside diameter of shaft forward of bearing		=	11.0 in
Section modulus of shaft at after end of bearing	Z_s	=	619 in^3
Yield strength of shaft	F_{ys}	=	45 ksi
Unit weight of shaft forward of bearing (0.000284 x area of shaft section forward of bearing)	w_s	=	0.0494 kip/in
Length of shaft between bearings	L_s	=	348 in

1. Main strut, vee-type radial

Applying the loading formulas of paragraph 161-1-t:

$M_1 = 0.0035\ D_p[W_p\ a_p(N/100)^2 + 3\ H/V] = 8490$ in-kip

$M_2 = 0.3\ Z_s\ F_{ys} = 8360$ in-kip

$P_3 = Z_s\ F_{ys}/D_s = 1506$ kip

Assuming a vee angle of 65 degrees, L = 132 in; $\ell/L = 15.5/132 = 0.12$.

Longitudinal bending moment

From figure 7, $K_7 = 0.97$. For $F_y = 30$ ksi, the required longitudinal section modulus for the larger moment (M_1), is $Z_L = K_7\ M_1/F_y = 275$. From figure 4, an EPH section with c = 25.5 in and t = 5.1 in, has $Z_L = 280$, which satisfies the requirements for longitudinal moment.

If the strut is twisted to, for example, 7 degrees, the required longitudinal and transverse section moduli would have to be increased by 8 percent, as discussed in paragraph 161-1-1. An EPH section 26.2 inches by 5.3 inches satisfies this requirement.

Load P_3, if taken to be acting along the neutral axis of a non-canted strut arm, will cause additional longitudinal bending moments if the strut arm is canted. For an angle Φ of 10 degrees, the additional moment at the barrel would be $P_3\ \ell \sin \Phi$, or

$$1506 \times 15.5 \times 0.174 = 4060 \text{ kip/in}$$

The required section modulus at the barrel is then

$$Z_L = K_7(M_1 + 4060)/F_y = 406 \text{ in}^3$$

An EPH section 29 inches by 5.8 inches satisfies this requirement. If the buttock lines slope 10 degrees from horizontal, the length of the canted strut would be reduced to about 129 inches, and the additional moment at the hull is

$$1506 \times 129 \times 0.174 = 33{,}800 \text{ kip/in}$$

The required section modulus at the hull is

$$Z_L = K_7(M_1 + 33{,}800)/F_y = 1367 \text{ in}^3$$

The chord thickness of an EPH strut section having this modulus can be found by multiplying the chord and thickness of the non-canted strut by the factor $[Z_L \text{ required for canted strut}/Z_L \text{ required for non-canted strut}]^{1/3}$. For this particular case the factor is $(1367/275)^{1/3} = 1.71$, and an EPH section 44 inches by 8.8 inches satisfies the requirement.

If, in addition to cant, the strut is twisted by 7 degrees, a further increase in chord and thickness by a factor of 1.026 is required, as shown in paragraph 161-1-1. This would require an EPH section 45 inches by 9 inches. The strut would be tapered in chord and thickness between the hull and the strut barrel.

Column strength

From figure 4, $A = 97\ in^2$ and $F_c = 20$ ksi. The value of P corre-ponding to this stress $= F_c\ A \sin 2\ \theta = 1760$ kip. This exceeds P_3, so a 25.5-in x 5.1-in strut section is satisfactory.

The larger strut sections required for twist and cant would, of course, also be satisfactory.

Vibration frequency

From figure 4, the natural frequency of the arm with a 25.5-inch by 5.1-inch strut section is 1640 cycles/minute, which is more than 20 percent above full power blade frequency (290 x 4 = 1160 cycles/minute for a four-bladed propeller). For the heavier struts required for twist and cant, the natural frequencies would be even higher, and therefore, satisfactory. Note that the frequency portion of figure 4 may be used only if $c/t = 5$; otherwise the formulas in paragraph 161-1-p must be used.

2. Main struts, vee-type, tangential

Tangential main struts will be designed for previously-decribed pro-peller and shaft. Requirements M_1 and M_2 are the same, $P_4 = 10W_p + 5w_s\ L_s =$ 281 kip.

With arms attached to the hull at the same location, the vee angle is 54 degrees, $L = 157$ in, $\ell = 40$ in, $\ell/L = 0.25$, $a_T = 28$ in.

Longitudinal bending moment

From figure 7, $K_7 = 0.90$. This is 8 percent less than the value obtained for the radial struts, indicating slight differences between the two arrangements insofar as the applied longitudinal moments are concerned.

Twisted or canted strut sections found adequate in longitudinal bending for the radial struts, would also be adequate for the tangential struts.

Transverse bending moment

Considering transverse bending stress alone, required section modulus $Z_T = K_6\ P_4\ a_T/f_t$. Allowing 3 ksi for axial stress $Z_T = 0.325 \times 281 \times 28/27 =$ 95 in^3. A section 25.5 in x 5.1 in gives Z_T of only 60 in^3, so thickness must be increased. An EPH section 30 inches by 6 inches would satisfy the requirement, and would have a chord/thickness ratio of 5.

If the strut is twisted 7 degrees, a component

$K_7M_1 \sin 7° = (0.97)(8490)(0.12) = 988$ in-kip

would be added to the transverse bending moment, and Z_T of 131 in^3 would be required. A section 33.5 inches by 6.7 inches would be adequate.

Vibration frequency

To provide a vibration frequency 20 percent above blade frequency, figure 4 indicates that a thickness of about 6.2 in is required.

A section 26 in x 6.5 in, with a Z_T of 99 in^3, appears adequate for vibration frequency and for transverse bending. (A shorter chord would give a undesirably low c/t ratio.) Then $f_T = K_6 \ P_4 \ a_T/Z_T = 25.8$ ksi.

If the strut is twisted 7 degrees, and a section 33.5 inches by 6.7 inches is selected, then

$$f_T = [K_6 P_4 a_T + 988]/131 = 27.1 \text{ ksi}$$

Column strength

Maximum column stress for the 26-in x 6.5-in section:

$$f_c = P/A \sin 2\theta = 281/126 \sin 54° = 2.8 \text{ ksi}$$

From figure 4, $F_c = 21$ ksi

For the twisted 33.5-inch by 6.7-inch section,

$$f_c = 281/168 \sin 54° = 2.1 \text{ ksi}$$

From figure 4, $F_c = 22$ ksi.

Combined stresses

Combining maximum transverse bending stress and maximum column stress, $f_T/F_y + f_c/F_c = 25.8/30 + 2.8/21 < 1.0$. For the 33.5-inch by 6.7-inch section with a 7-degree twist,

$$f_T/F_y + f_c/F_c = 27.1/30 + 2.1/22 < 1.0$$

The combined stress is reduced somewhat below that indicated by the equation if allowance is made for the 27-degree difference between the assumed directions of loading.

The radial design of example 1 is considered preferable to the tangential arrangement of example 2.

3. Intermediate strut, vee-type, radial

For the shaft of example 1, the formulas of paragraph 161-1-t give:

$$M_5 = 0.12 Z_s \ F_{ys} = 3340 \text{ in-kips}$$
$$P_6 = 0.6 Z_s \ F_{ys}/D_s = 903 \text{ kips}$$

Design for these loadings follows the procedure of example 1. With a vee angle of 65 degrees and arm length of 35 in, a section 17 in x 3.4 in is found to be adequate for the above moment and for loads up to 1200 kips.

4. Intermediate strut, single-arm

A single-arm strut 30 in long will be designed for a force of $10 w_s L_s$ (172 kips). A moment equal to M_5 of example 3 will be applied, but it will be assumed that both shafting and strut resist this moment. Allowable torsional shearing stress will be taken as $0.6F_y$. A minimum yield strength of 35 ksi will be specified.

For bending, required transverse section moduli are:

At hull, Pa_s/F_y = 172 x 30/35 = 147 in^3
At hub, 172 x 15.5/35 = 76 in^3
At midlength, 172 x 22.75/35 = 112 in^3

Assuming a chord-thickness ratio of 4, a section 28 in x 7 in gives the required section modulus for midlength. Approximate stiffness under torsional moment is $45{,}300 \times I_T/(L - \ell) = 45{,}300 \times 430/14.5 = 1.34 \times 10^6$ in-kip per radian. The stiffness of the shaft (for the span from intermediate strut to main strut) is $4\ E_s\ I_s/L_s = 4 \times 29{,}600 \times 5030/348 = 1.71 \times 10^6$ in-kip per radian. The moment carried by the strut will then be about $3340 \times 1.34/(1.34 + 2 \times 1.71) = 940$ in-kip. Required section modulus for torsional stress is $M/1.2\ F_y = 940/(1.2 \times 35) = 22.4\ in^3$. This is less than required for bending, so the strut will be proportioned to suit the values calculated above.

At hull, 28 in x 8 in; Z_T = 161 in^3
At hub, 28 in x 6 in; Z_T = 90 in^3
At midlength, 28 in x 7 in; Z_T = 123 in^3

A strut arm of these proportions would be heavier than the two arms of example 3. Weight can be reduced by using a hollow section, increasing the outside dimensions somewhat. Adequate support within the hull is made more difficult by the high transverse and torsional moments. More important, the vee strut provides much better support for the shaft against both forces and moments. Hydrodynamic resistance of the single arm might be somewhat less, but the vee strut is considered the better design for all but extremely short arms.

161-1-v. Aids for inspection requirements

In some cases inspection requirements for smoothness and fairness of strut arms are specified in detail. It may be required to calculate the nose radius for information for an inspector using a template, bridge-dial gage, or other means. For EPH sections of chord, c, and maximum thickness, t, the mathematical value for the elliptical nose radius is

$$R = 0.5732\ t^2/c$$

For example, for an EPH strut arm with a chord length of 47.5 inches and maximum thickness of 9.5 inches, the nose radius is

$$R = 0.5732\ \frac{(9.5)^2}{47.5} = 1.09 \text{ in}$$

161-1-w. Metric conversion

Table I provides metric conversion factors.

Table I. Metric conversion factors

To convert from:	To:	Multiply by:
inches (in)	centimeters (cm)	2.54
in^2	cm^2	6.45
in^3	cm^3	16.39
in^4	cm^4	41.62
kip	newtons (N)	4448.22
ksi	kilopascals (kPa)	6894.76
in-kip	newton meter (N-m)	112.98