DDS 562-2
CONTROL SURFACE STRUCTURE DESIGN

DEPARTMENT OF THE NAVY
NAVAL SEA SYSTEMS COMMAND
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Design Data Sheet
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Naval Sea Systems Command

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Control Surface Structure Design

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562-2-b. Introduction

This design data sheet is issued to provide a design procedure for control surface structure design. The procedure is applicable to surface ships and submarines and applies to rudders, fin stabilizers, stern diving planes, fairwater planes and bow planes. The structure considered consists of the control surface, its stock, web spacing, web and plating thickness, hub, gudgeons, pintles, keyways and bearing configuration. The shock design of control surfaces is not considered herein.

562-2-c. Definitions and Nomenclature

Diving Planes - The control surface used for controlling the vertical motion of a submarine. Diving planes include stern diving planes, fairwater planes and bow planes.

Fairwater Planes and Bow Planes - The control surfaces built into the submarine fairwater (sail) or near the bow, used for controlling the vertical motion. Fairwater planes and bow planes have a configuration similar to spade rudders. See Figure 1.

Fin Stabilizers - The control surfaces used for reducing surface ship roll. Fin stabilizers are similar in configuration to spade rudders. See Figure 1.

Gudgeons - The socket on a balanced rudder into which the pintle fits. Normally part of the rudder. See Figure 3. Similarly for horn rudders and stern diving planes.

Hub - The casting at the root of a control surface which mates with the stock. See Figure 1.

Keyway - The matching slots cut into the hub and stock of a spade or horn rudder for receipt of the torque keys. The keys constrain the rudder and stock to rotate together. Similarly for diving planes and fin stabilizers.

Pintle - The pin at the bottom of a balanced rudder which mates with the gudgeon. See Figure 3.

Rudder - The control surface used for control of the horizontal motion of a surface ship or submarine. There are three types of rudders used in Naval ship design practice: Spade, Horn, and Balanced. Figures 1, 2, & 3 illustrate these types.

Stern Diving Planes - The control surfaces located near the stern of a submarine for influencing the vertical motion, primarily by controlling the pitch angle. Stern diving planes have a configuration similar to balanced rudders except that they are usually located behind fixed horizontal stabilizers.

Stock - The rudder stock is the rotating element which articulates the rudder. The rudder is locked to the stock by a system of keys. The stock is rotated by the steering gear machinery located in the ship. The stock passes through the hull of the ship through a system of glands and seals. See Figures 1, 2, and 3. Similarly for other control surfaces.

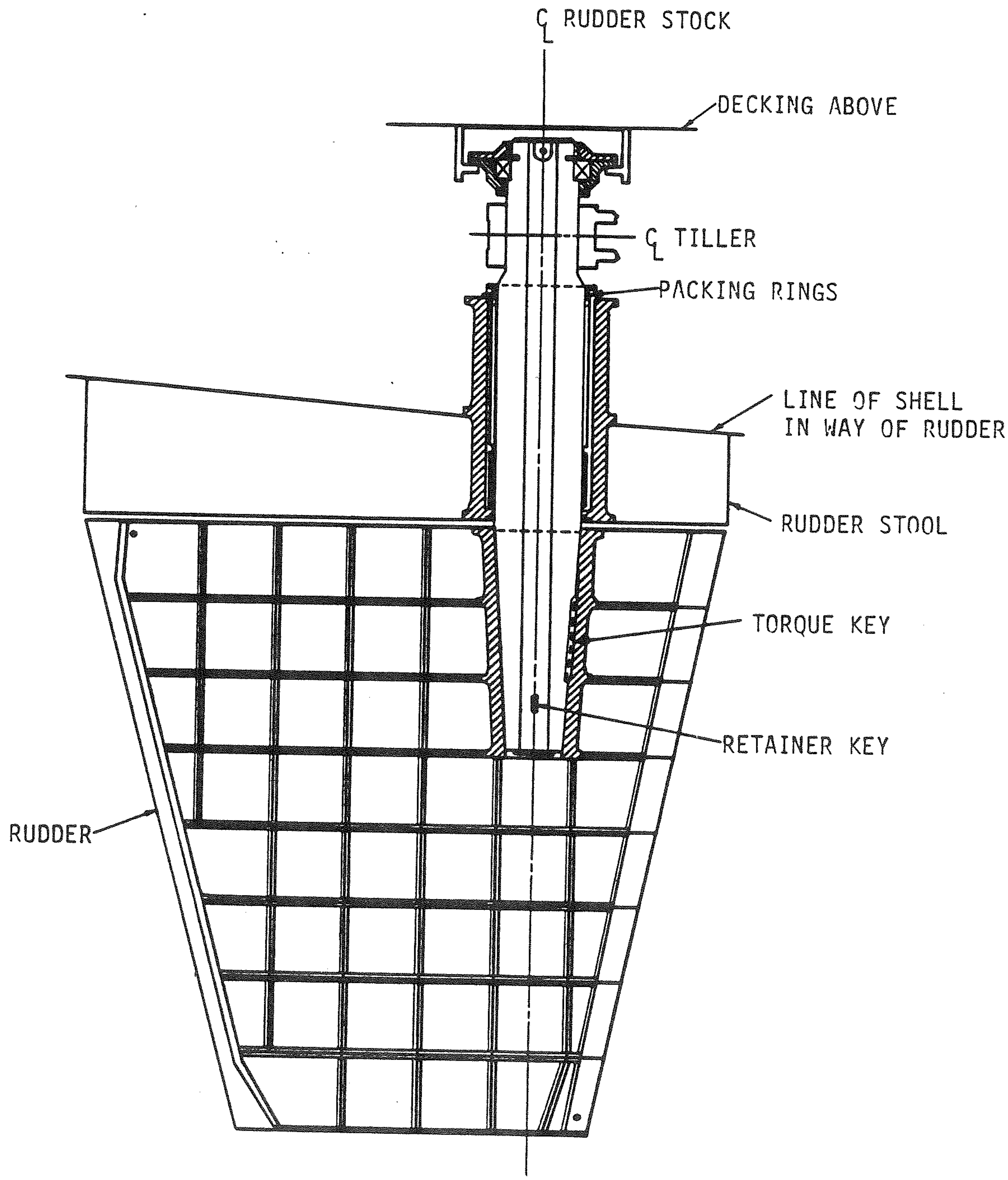

FIGURE 1 SPADE RUDDER

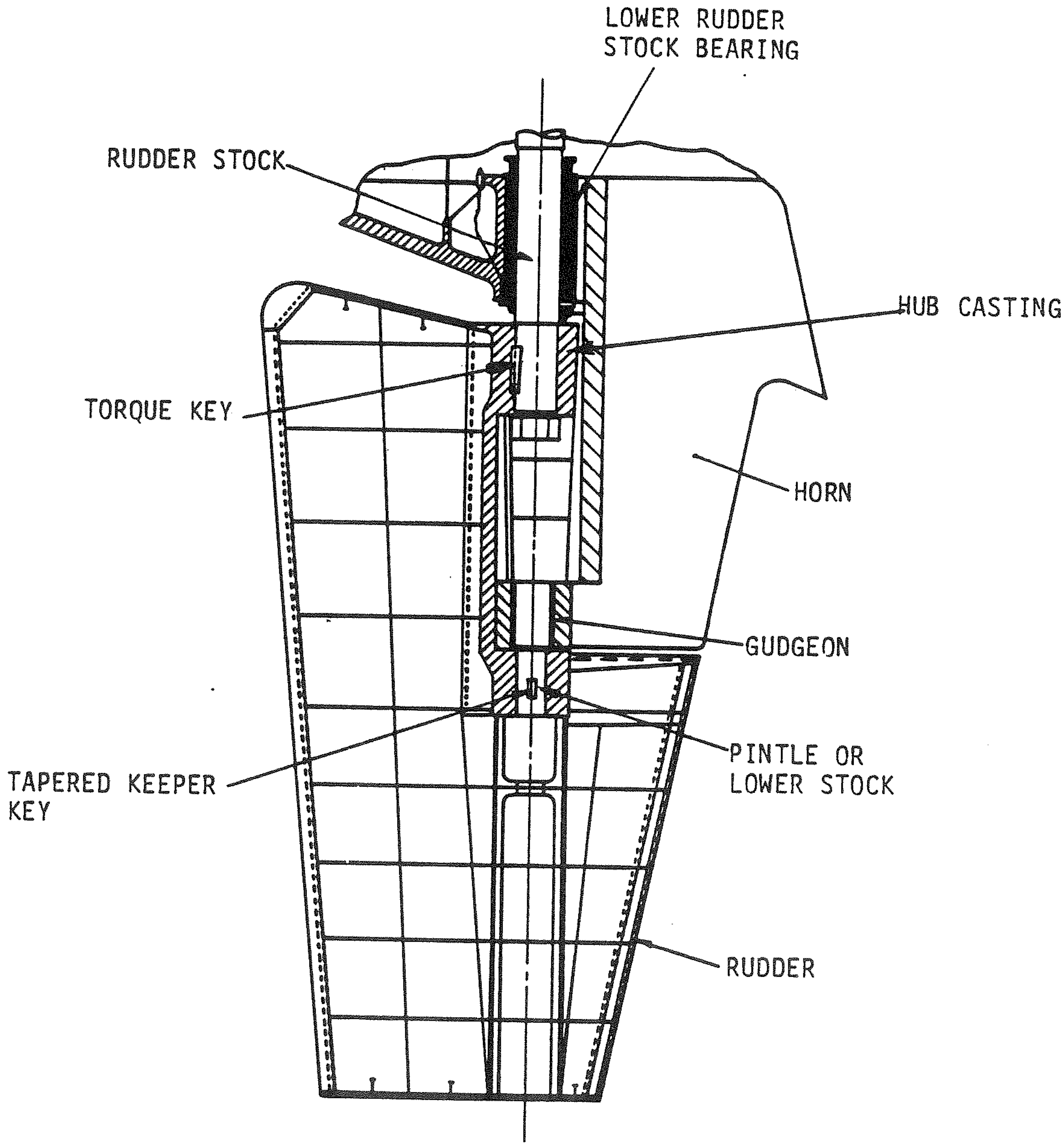

FIGURE 2 HORN RUDDER

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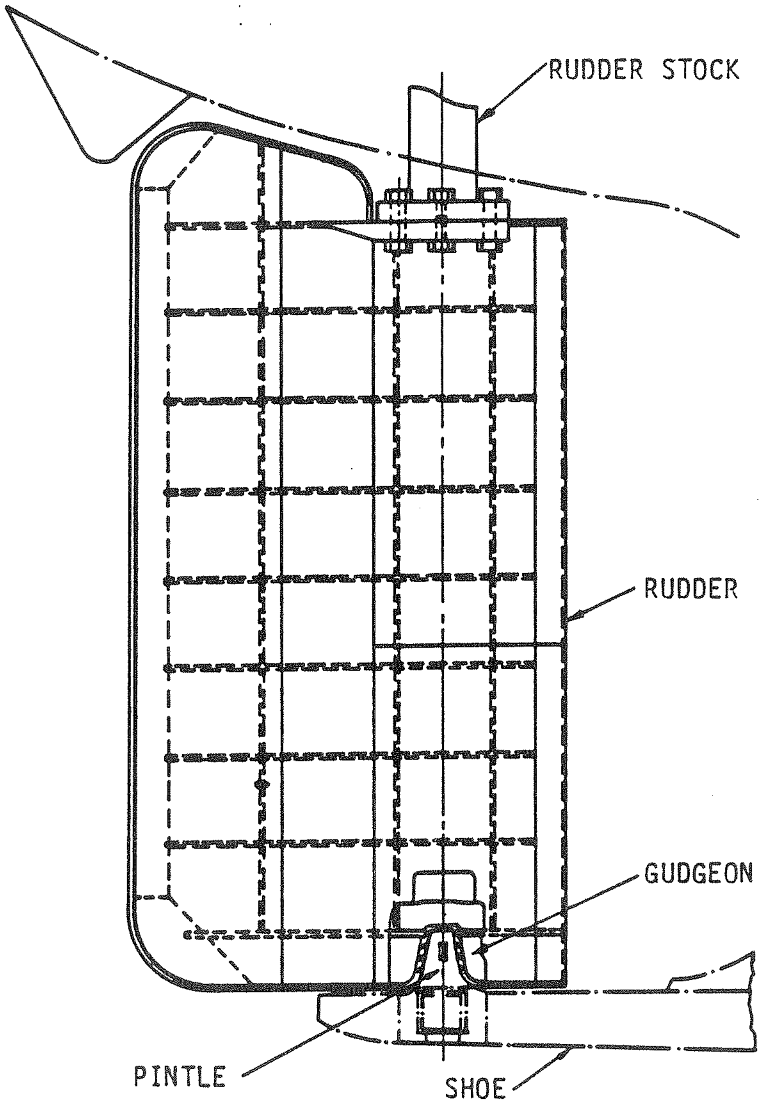

FIGURE 3 BALANCED RUDDER

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The stock passes through the hull of the ship through a system of glands and seals. See Figures 1, 2, and 3. Similarly for other control surfaces.

NOMENCLATURE

a_1 = Length of panel

a = Load at tip, or load at aft end of chord

A = Projected area of control surface

A_h = Area in horizontal section enclosed by the horn.

a_2 = $A/12$; A in ft^2

b = Breadth of panel

b_1 = Load at root, or load at forward end of chord

B = Span

C = Constant depending on material (Table 2)

c = Mean chord length

Cp = Center of Pressure

d = Stock diameter

D = $0.684\ t_m$

E = Modulus of Elasticity

s/t = Circumferential length of control surface horn in horizontal section divided by its mean wall thickness

F = Rudder Force = Area of Load Trapezoid

G = Gudgeon strength factor from Figure 12

H = Maximum total head

I_a = Mean moment of inertia of stock

I_b = Mean moment of inertia of control surface above pintle bearing.

I_d = Mean moment of inertia of sole piece

I_h = Mean moment of inertia of horn

I_H = Moment of inertia of hub and associated stock to be used in strength calculations

I_{H-A} = Actual moment of inertia of the hub and associated stock about centroidal axis of control surface section

J_h = Mean polar moment of inertia of horn

K = Stress factor depending on stock penetration (Figure 16)

k = Rudder horn spring constant

K = Constant depending on b/a (Table 3)

K_d = Factor used in balanced type spanwise load

l = Length of gudgeon

l_a = Defined in Figure 10

l_b = Defined in Figure 10

ld = Distance from centerline of pintle to assumed point of fixity of sole piece to hull or other structure

l_r = Tip chord length

l_F = Distance of line of action of resultant of forces acting on control surface forward of stock centerline from stock centerline

l_R = Distance of line of action of resultant of forces acting on control surface aft of stock centerline from stock centerline

L = Chord length or span

Lu = Root chord length

M = Bending moment in stock

M_F = Maximum bending moment acting at stock centerline due to forces on end of control surface forward of stock centerline

M_h = Bending moment at neck bearing

M_p = Bending moment at pintle bearing

M_R = Maximum bending moment acting at stock centerline due to forces on end of control surface aft of stock centerline

M_t = Torque in stock

n = Factor of safety on stock stresses

N = Resultant force acting at CP

N_1 = Total force on control surface

NF = Resultant of Chordwise Forces acting on control surface forward of stock centerline

N_R = Resultant of chordwise forces acting on control surface aft of stock centerline

σ_G = Maximum stress in gudgeon

σ_y = Yield strength of stock material (KS1)

$$P = \frac{\text{Inner diameter of stock}}{\text{Outer diameter of stock}}$$

P_c = Pintle reaction

P_F = N_F

P_h = Lower bearing reaction

P_L = Iron work labor cost

P_p = Pintle bearing reaction

P_R = N_R

P_S = Material cost

P_u = Upper bearing reaction

P_W = Welder labor cost

R_1 = Outside radius of gudgeon

R_2 = Inside radius of gudgeon

t = Thickness of outside control surface plating

t_1 = Total reaction at gudgeon (or Shear Force at Horn)

t_m = Max. thickness of mean chord

t_1 = Corrosion allowance

t_2 = Cavitation allowance

t' = Thickness of plating based on loading

t_1' = Assumed base thickness of outside plating

W = Chordwise loading

W_b = Uniform load over upper part of control surface

W_c = Uniform load over lower part of control surface

x = Distance of center of pressure from root or from forward edge of chord

Z = Defined in Figure 10

562-2-d. Design Flow Diagram

The flow diagram in Figure 4 indicates the relationships between the various parts of the design procedure. The arrows indicate the sequence of steps to be followed in a normal design.

With horn and balanced type control surfaces, estimated values of stock and control surface moments of inertia are needed in order to compute shear and bending moments. Therefore, the sections on determination of stock size, web spacing, and web and side plating thicknesses will have to be considered first. The resulting calculations will form a loop in the flow diagram until all calculated values are compatible.

562-2-e. Input Information

The information needed to carry out the calculations presented in the following sections is listed below and may be found in the specifications for building of the ship and in the contract and the contract guidance drawings.

562-2-e-1. Control surface configuration and stock location. The size, shape and location of the control surfaces are determined on the basis of previous experience as well as on the particular requirements for the design under consideration. Control surface configuration, location and stock location on the hull are usually shown on the Contract Lines Drawing.

The stock/bearing configuration, bearing locations, control surface internal arrangement and stock/control surface connection are usually shown on the Rudder and Appendages Contract Guidance Drawing. The Rudder and Appendages type drawing (Reference 1) gives guidance for designing and placing the rudder in the ship.

562-2-e-2. Bearings. Bearings are to be designed to take the rudder stock radial and thrust loads and the rudder stock flexural deflections. There are two basic types of bearings which are used: Antifriction and Sleeve.

Sleeve bearings have low initial cost, are relatively easy to maintain, and in most cases are relatively easy to repair in emergencies. Antifriction bearings are fairly expensive, require some precision machining to install and require a careful sealing arrangement to prevent seawater contamination; they have the compensatory advantages of permitting significant reduction of steering gear torque and can be self-aligning (so that rudderstock deflections and hull movement do not cause binding).

The friction coefficients used are 0.01 for antifriction and 0.20 for sleeve type bearings. The sleeve bearing material is usually a reinforced laminated phenolic. This material is best lubricated by grease and is very good for hull bearings or stern post bearings on horn and balanced rudders. Principal control surface internal arrangements and locations of bearings

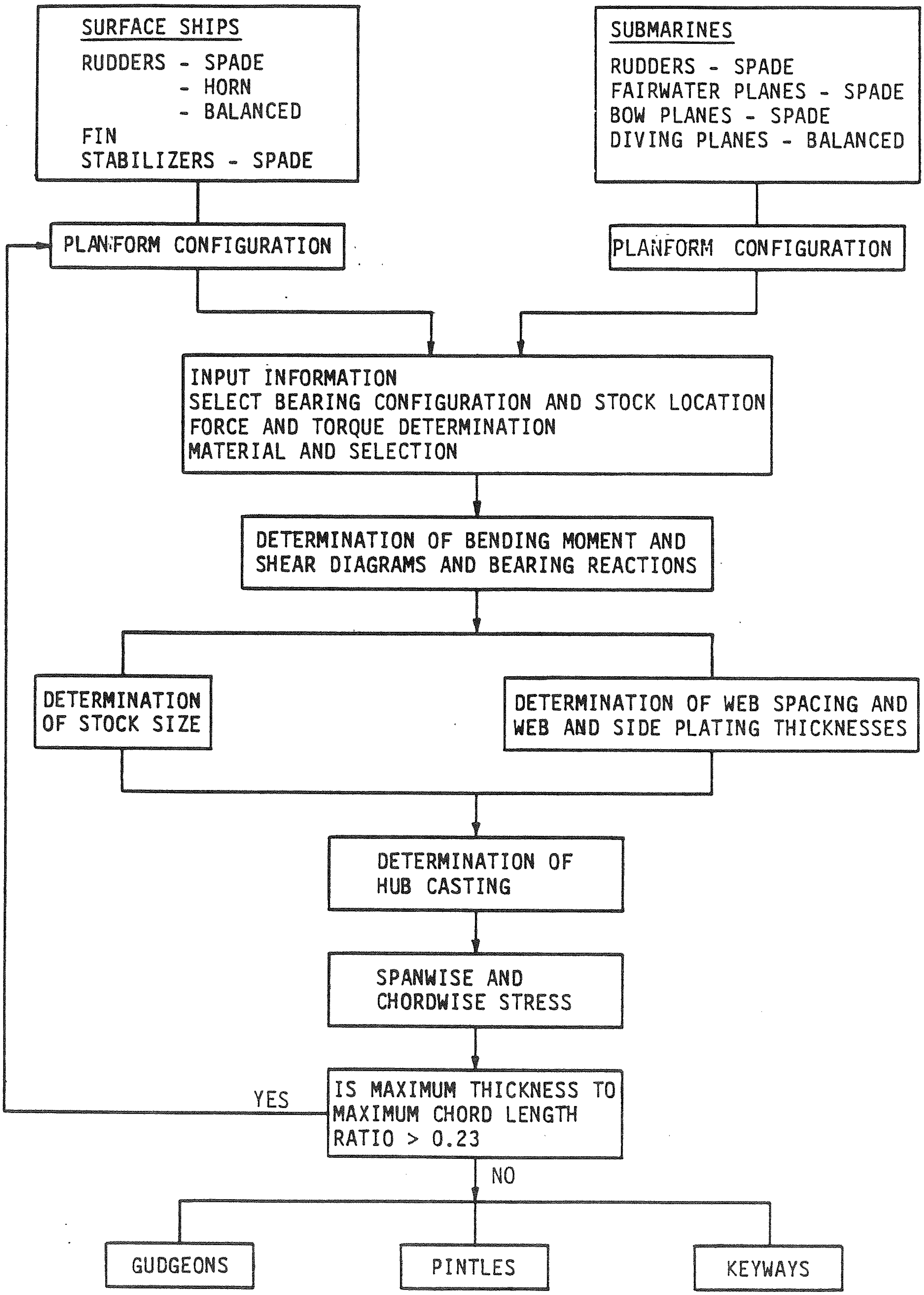

FIGURE 4 FLOW DIAGRAM

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are determined on the basis of previous experience. There are five typical arrangements of rudder stocks and bearings as shown in Figures 5 through 9. Table 1 gives a tabular listing of bearing particulars for each configuration. The upper antifriction bearing is usually a spherical roller bearing which can take thrust as well as radial loads. The lower bearing absorbs radial loads only. The sleeve bearing length (or height) is not to be less than 1.2 times the diameter of the stock. The separation of the upper and lower bearings should not be less than one-third of the distance from the centerline of the lower bearing to the control surface spanwise center of pressure. This is to keep the required stock diameter within reasonable limits.

562-2-e-3. Determination of Forces and Moments Acting on the Control Surface Structure

562-2-e-3(a). Hydrodynamic Forces and Moments

Control Surface Torque

Control surface torque will usually be found in the ship's specification. The torque can be computed by various methods depending on the type of control surface.

Spade Rudders.

The calculation procedure for spade rudders is given in References 2 and 5.

The most important sources of supportive data are References 3, 4, 7 and 12. The results of that calculation are presented in a plot of upsetting and restoring torque versus rudder angle. The torque includes hydrodynamic, frictional and allowance (due to variation of center of pressure) torques. The rudder balance is adjusted until the maximum upsetting torque equals two thirds of the maximum restoring torque.

Horn Rudders.

The calculation procedure for horn rudders, described in Reference 6, is used for determination of the rudder normal force and center of pressure. The rudder is considered as two separate pieces and the normal force and center of pressure curves may be obtained from empirical curves for each piece. The basic calculation procedure is as follows:

- Determine the span to chord ratios for the upper and lower portions of the rudder.
- Obtain coefficients for the normal force and center of pressure for various rudder angles using the graphs of Reference 6.

TABLE 1 - CONTROL SURFACE BEARING CONFIGURATION PARTICULARS

Description - Type of Arrangement	Function of Grading	Type of Bearing	Lubricant	Type of Bearing	Lubricant
A/FIG 5	Radial	Sleeve	Grease	Sleeve	Water and Grease
B/FIG 6	Radial	Sleeve	Grease	Sleeve	Water
C/FIG 7	Carrier and Radial	Antifriction	Grease	Sleeve	Grease
D/FIG 8	Carrier and Radial	Antifriction	Grease	Antifriction	Grease
E/FIG 9	Carrier and Radial	Antifriction	Grease	Antifriction	Grease

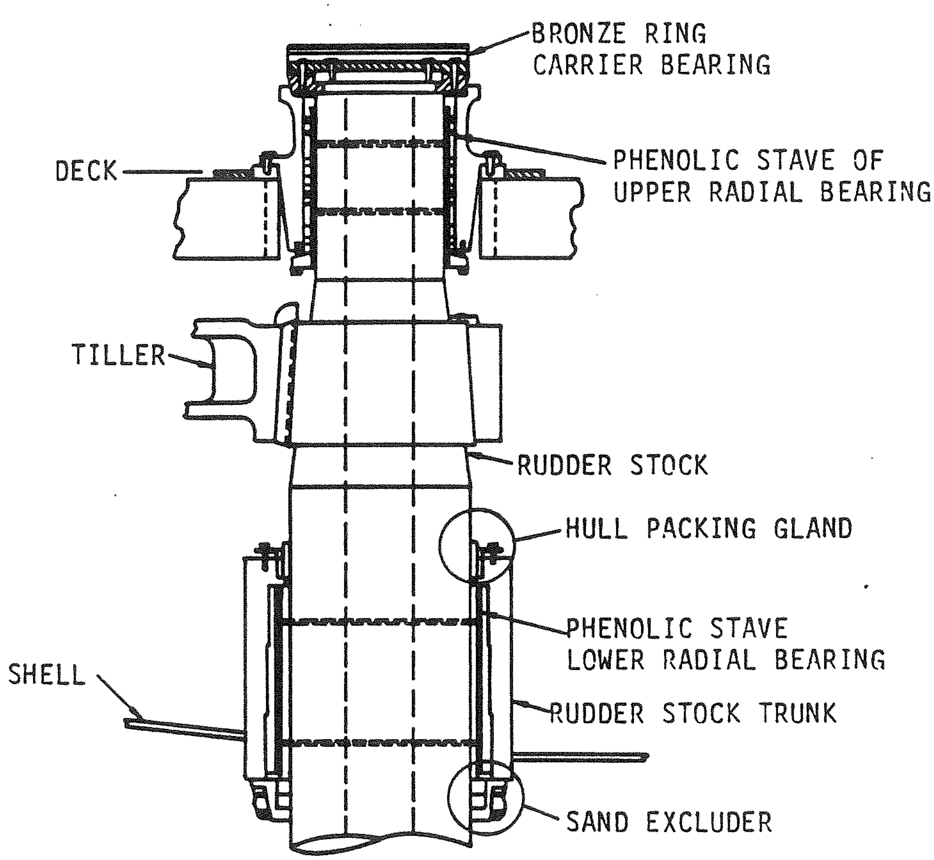

FIGURE 5 RUDDER STOCK BEARING TYPICAL ARRANGEMENT "A"

ARRANGEMENT "A" HAS THE UPPER RADIAL LAMINATED PHENOLIC STAVE BEARING LUBRICATED WITH GREASE ONLY AND THE LOWER BEARING IS FLOODED WITH SEA WATER, AS WELL AS GREASE.

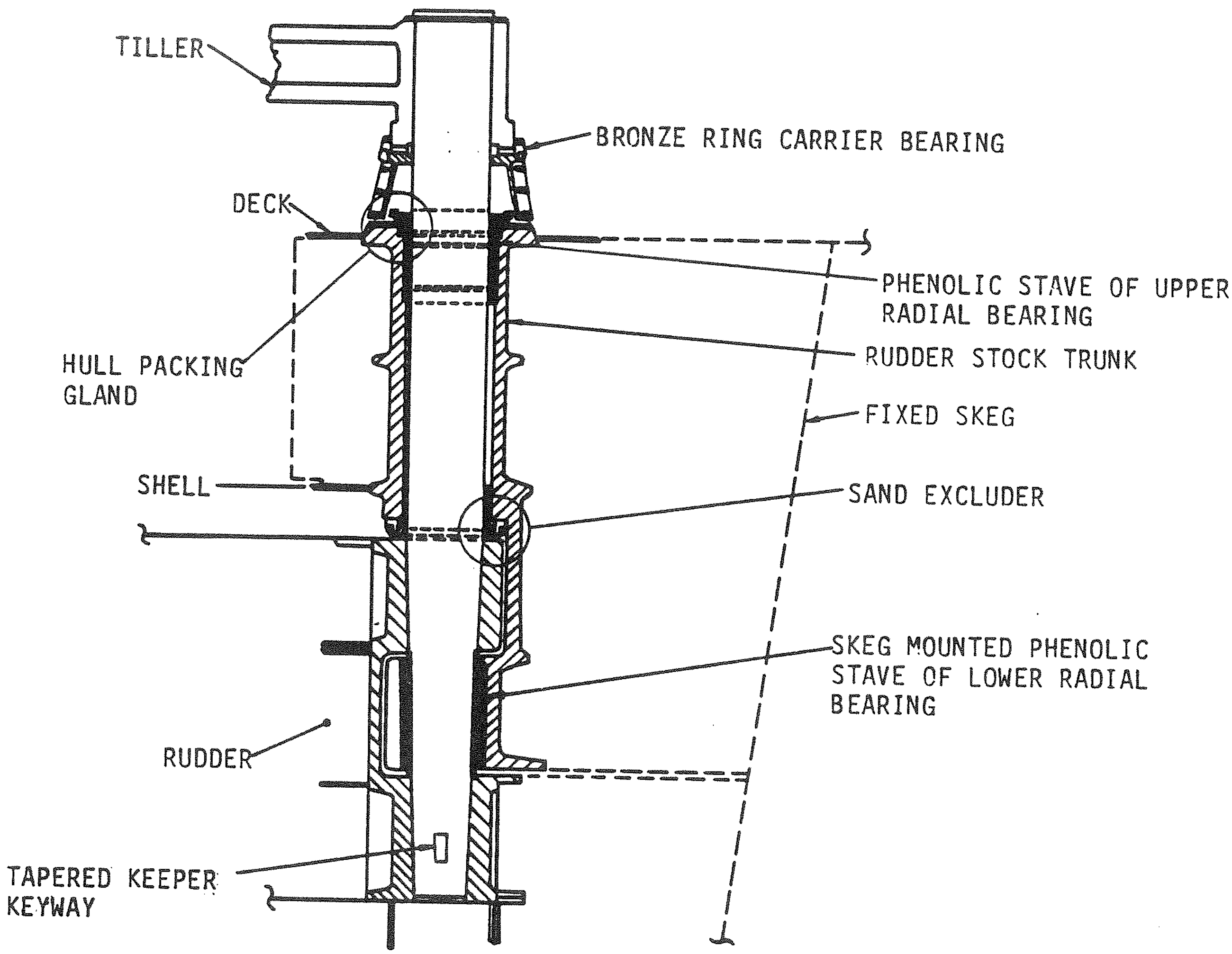

FIGURE 6 RUDDER STOCK BEARING TYPICAL ARRANGEMENT "B"

ARRANGEMENT "B" HAS BOTH UPPER AND LOWER RADIAL BEARINGS FLOODED. THE UPPER BEARING IS LUBRICATED WITH GREASE. THE LOWER BEARING IS LUBRICATED WITH SEA WATER ONLY.

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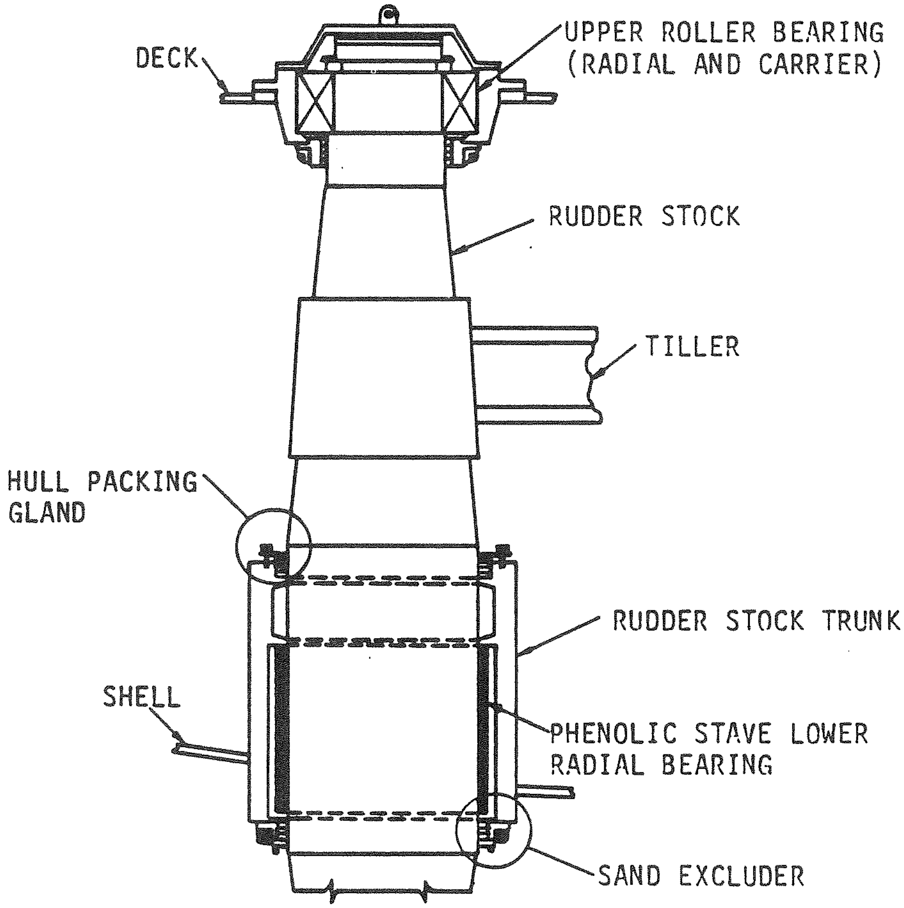

FIGURE 7 RUDDER STOCK BEARING TYPICAL ARRANGEMENT "C"

ARRANGEMENT "C" USES A SPHERICAL ROLLER UPPER BEARING, LUBRICATED WITH GREASE AND A LOWER LAMINATED PHENOLIC BEARING LUBRICATED WITH GREASE.

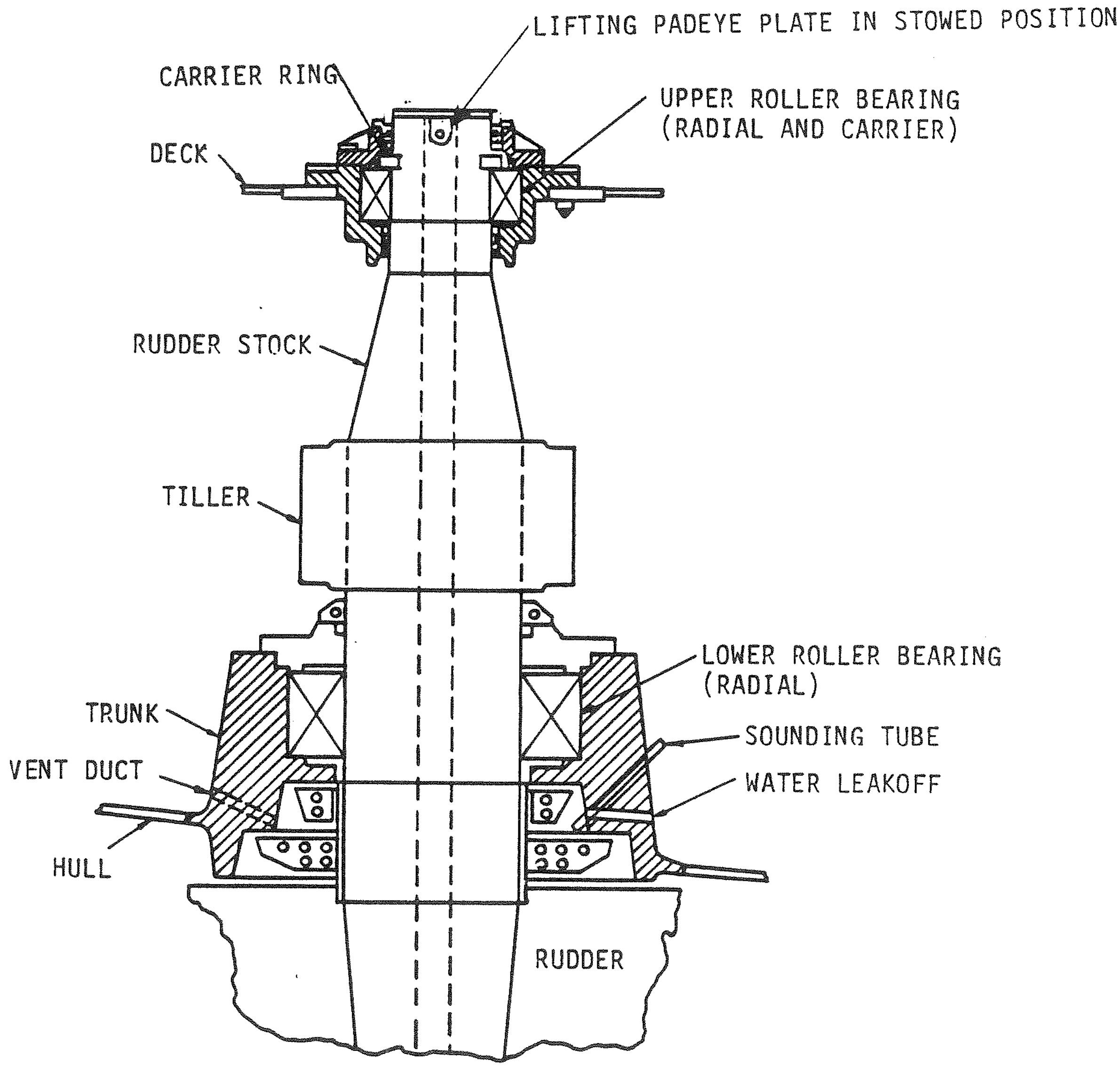

FIGURE 8 RUDDER STOCK BEARING TYPICAL ARRANGEMENT "D"

ARRANGEMENT "D" USES TWO SPHERICAL ROLLER BEARINGS AND AN INTEGRAL BEARING OIL SEAL AND HULL WATER SEAL ON THE LOWER BEARING.

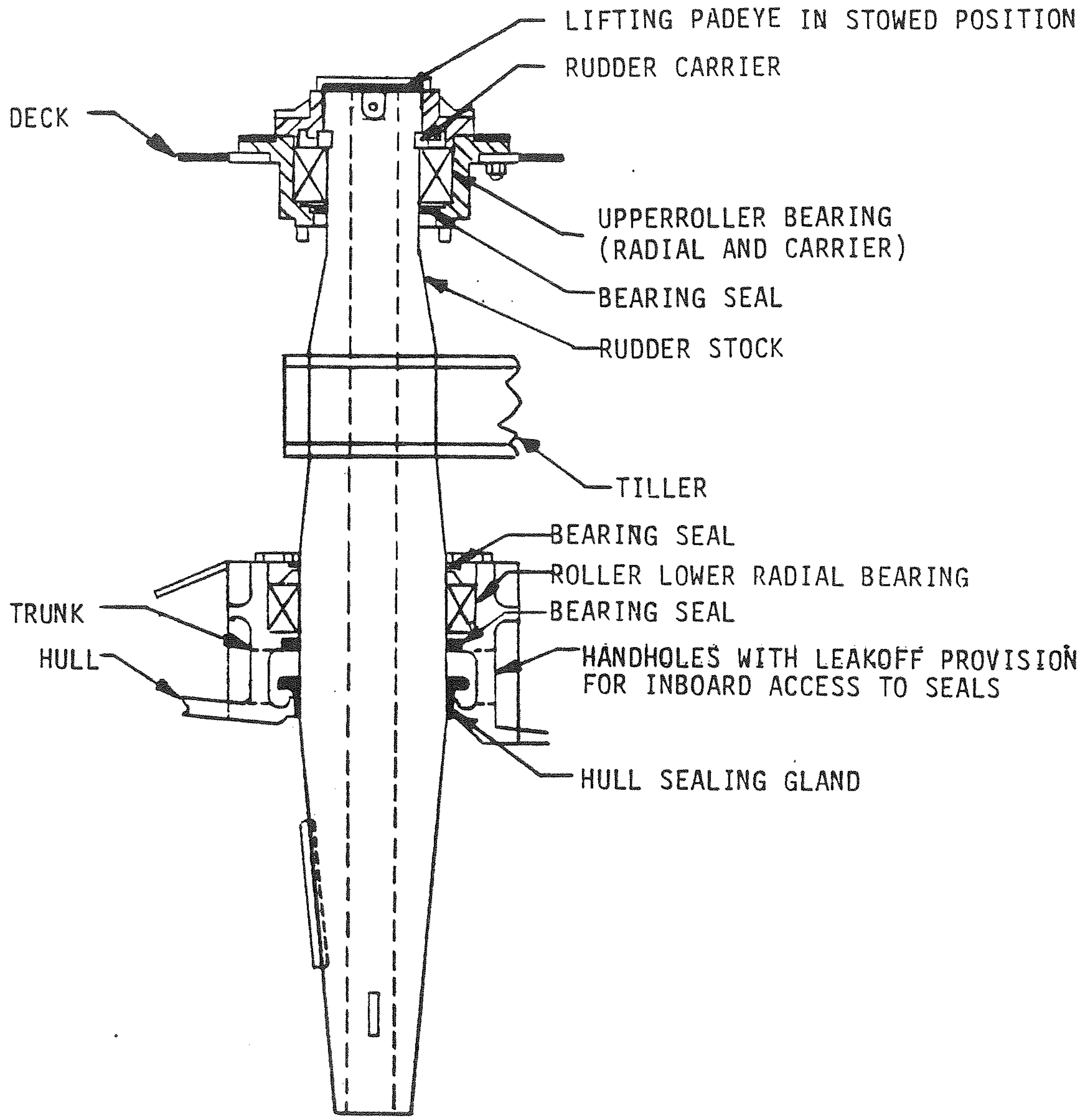

FIGURE 9 RUDDER STOCK BEARING TYPICAL ARRANGEMENT "E"

ARRANGEMENT "E" USES TWO SPHERICAL ROLLER BEARINGS WITH SEPARATE GREASE AND HULL WATER TIGHT SEALS AND AN ADEQUATE ACCESS/LEAK OFF SPACE BETWEEN THE GREASE AND WATER TIGHT SEALS.

- Determine the moment arm and normal force for each rudder portion.

- Determine the torque for each portion of the rudder.

- Sum up the torque values to yield the total hydrodynamic torque.

- Add the frictional torque and allowance torque to the hydrodynamic torque to obtain final design torque. The frictional torque is a function of the rudder bearing reaction, stock bearing radius and bearing friction coefficient. The friction coefficients used are 0.01 for antifriction bearings and 0.20 for sleeve type bearings. The torque allowances are 25 and zero percent of the maximum rudder torque, respectively for roller bearings and sleeve bearings. (Reference 7).

Balanced Rudders.

For this type of rudder the Joessel method (Reference 8) is used when flat plate type structure is used. Reference 2 may be used when airfoil type structure is used. The Joessel method has two formulae, one for the torque variation and one for the variation of the center of pressure with the angle of attack.

Fin Stabilizers, Fairwater Planes & Bow Planes.

The calculation method for these control surfaces is similar to that for spade rudders; however Reference 5 is also used for determining the hydrodynamic coefficients.

Submarine Stern Diving Planes.

The calculation procedure for these planes is given in Reference 7.

562-2-e-3(b). Non-Hydrodynamic Forces and Moments

Sea Slap.

A rectangular shaped spanwise sea slap head distribution may be assumed for surface ship rudders. The maximum value of the sea slap head is given in the ship specifications. For submarines the sea slap and distribution are given in the ship specifications.

Hydrostatic Force.

This pressure is due to the hydrostatic head of water acting on the control surface and is calculated using the depth of water at the point on the control surface for which the pressure is calculated.

Ice Load.

This load is due to the impact of ice on the contral surface. For auxiliary ships, see 562-2-n. For other ships, ice loadings will be contained in the ship's specifications.

562-2-e-4. Materials to be used for the various control surface structural parts.

The minimum yield strength to be used for low carbon alloy steel rudder stocks for auxiliary ships is 65,000 psi, and for combatant ships 100,000 psi. However, when the rudder root thickness to chord ratio becomes greater than 0.23 (after planform alternatives have been considered) 100,000 psi yield point steel should be considered for auxiliary ships' stock material. Where rudder stocks are required to have little or no magnetic permeability, nickel aluminum bronze shall be used. The minimum yield strength for nickel aluminum bronze is 35,000 psi based on specifications. The minimum yield strength for steel hub castings is 30,000 psi. Rudder plating is to be made of HY-80 (for combatant ships) or ABS (for auxiliary ships) steel and internal members are to be made of HS or OS.

For surface ships a factor of safety of 2.0 on yield for spade rudders and 2.5 on yield for horn and balanced rudders are to be minimums.

For submarines, use a factor of safety of 2.0 on yield for hydrodynamic or ice loads, and a factor of safety of 1.25 on yield for sea slap.

For combined hydrodynamic and sea slap loads, refer to the ship specification. The maximum allowable shear stress is to be taken as 60% of yield strength. The factors of safety given above are to be used in conjunction with the shear stress.

562-2-f. Determination of Shear, Bending Moment, and Deflection Diagrams

562-2-f-1. Combining Loads for Spanwise and Chordwise Shear and Bending Moment Diagrams

Combine the given chordwise and spanwise loads as required by the ship specifications. These combined loads will be used for the overall shear and bending moment calculations only.

562-2-f-2. Calculation of Load Distribution over Chord and Span

Using the calculated force and the center of hydrodynamic pressure, the chordwise and spanwise load distributions can be derived using a trapezoidal distribution so that the centroid of the load trapezoid coincides with spanwise or chordwise center of pressure.

The calculation procedure below requires solving equations 1 and 2 below for the loads (a and b) at the forward and aft ends of the mean chord and also at the top and bottom of the span:

$$F = \frac{a + b}{2} L \tag{1}$$

Equation for Centroid of load trapezoid:

$$x = \frac{L(2a + b)}{3(a + b)} \tag{2}$$

Where:

a = load at aft end of mean chord or bottom of span

b = load at forward end of mean chord or top of span

x = distance of center of pressure from forward edge of chord or top of span

L = chord length or span

F = Rudder Force = Area of load trapezoid

Solving equations (1) and (2) calculate the spanwise and chordwise load distributions using:

$$a = \frac{2F}{L}\left(\frac{3x}{L} - 1\right)$$

$$b = \frac{2F}{L}\left(2 - \frac{3x}{L}\right)$$

(a) Spade Type Control Surface - Spanwise Load

The spanwise loading and the resulting shear and moment diagrams for a spade type control surface are shown in Figure 10.

The bearing reaction forces are calculated below:

$$P_u = \frac{N(z + h_n)}{l_a} \tag{3}$$

$$P_h = N + P_u \tag{4}$$

The maximum bending moment is calculated below:

$$M_h = N(z + h_n) \tag{5}$$

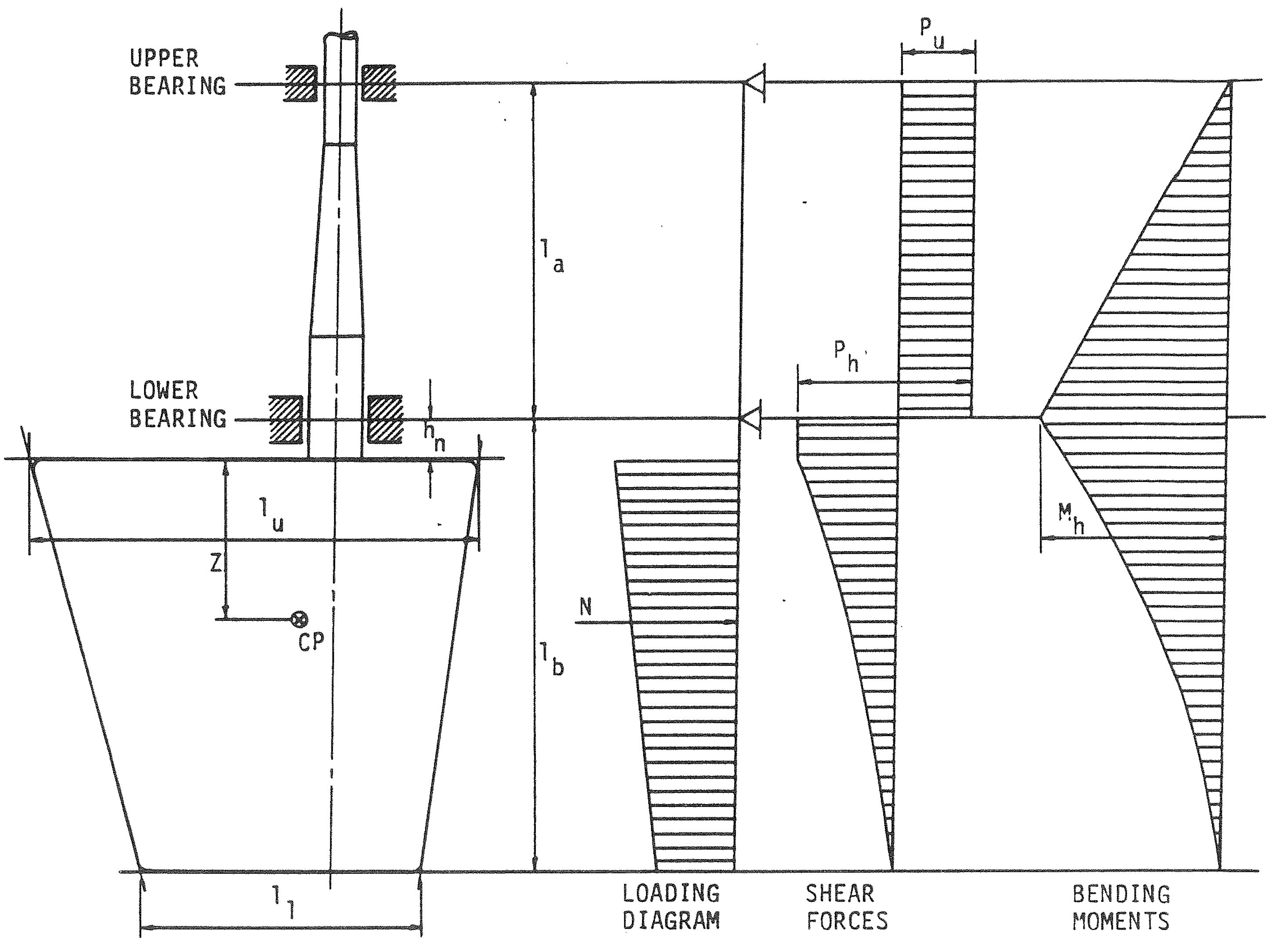

FIGURE 10 SPADE TYPE CONTROL SURFACE - SPANWISE LOAD, SHEAR DIAGRAM AND MOMENT DIAGRAM

l_R = Distance of line action of resultant of forces acting on control surface aft of stock centerline from stock centerline (IN).

l_F = Distance of line of action of resultant of forces acting on control surface forward of stock centerline from stock centerline (IN).

(c) Horn Type Control Surfaces - Spanwise Load

The spanwise load, shear diagram and bending moment diagram for a horn type control surface are shown in Figure 12. The Bearing Reaction forces are calculated as presented below, from Appendix 1, "Rudder Calculations," of Reference 13:

$$P_u = \frac{M_h}{l_a} \tag{8}$$

$$P_h = P_u(1 + l_a/l_b) + (W_b/(2\ l_b))(l_b - h_n)^2 - M_p/l_b \tag{9}$$

$$P_p = N + P_u - P_h \tag{10}$$

The maximum bending moments are:

$$M_p = \frac{W_c\ l_c^2}{2} \tag{11}$$

$$M_h = \frac{W_b(l_b^2 - h_n^2)(\frac{1}{2} + \frac{a}{8}) + W_c l_c^2(\frac{1}{2} + \frac{l_b}{l_c} - \frac{a}{4})}{1 + a(1 + \frac{4}{3}\frac{l_a I_b}{l_b I_a})} \tag{12}$$

where,

$$a = \frac{K\ l_b^3}{3EI_b} \tag{13}$$

$$K = \frac{E}{l_h\ (\frac{2.6e^2}{J_h} + \frac{l_h^2}{3I_h})}\ \text{KIPS/IN} \tag{14}$$

W_b = Uniform load over upper part of control surface (KIPS/IN).

W_c = Uniform load over lower part of control surface (KIPS/IN).

I_a = Mean moment of inertia of stock (IN^4).

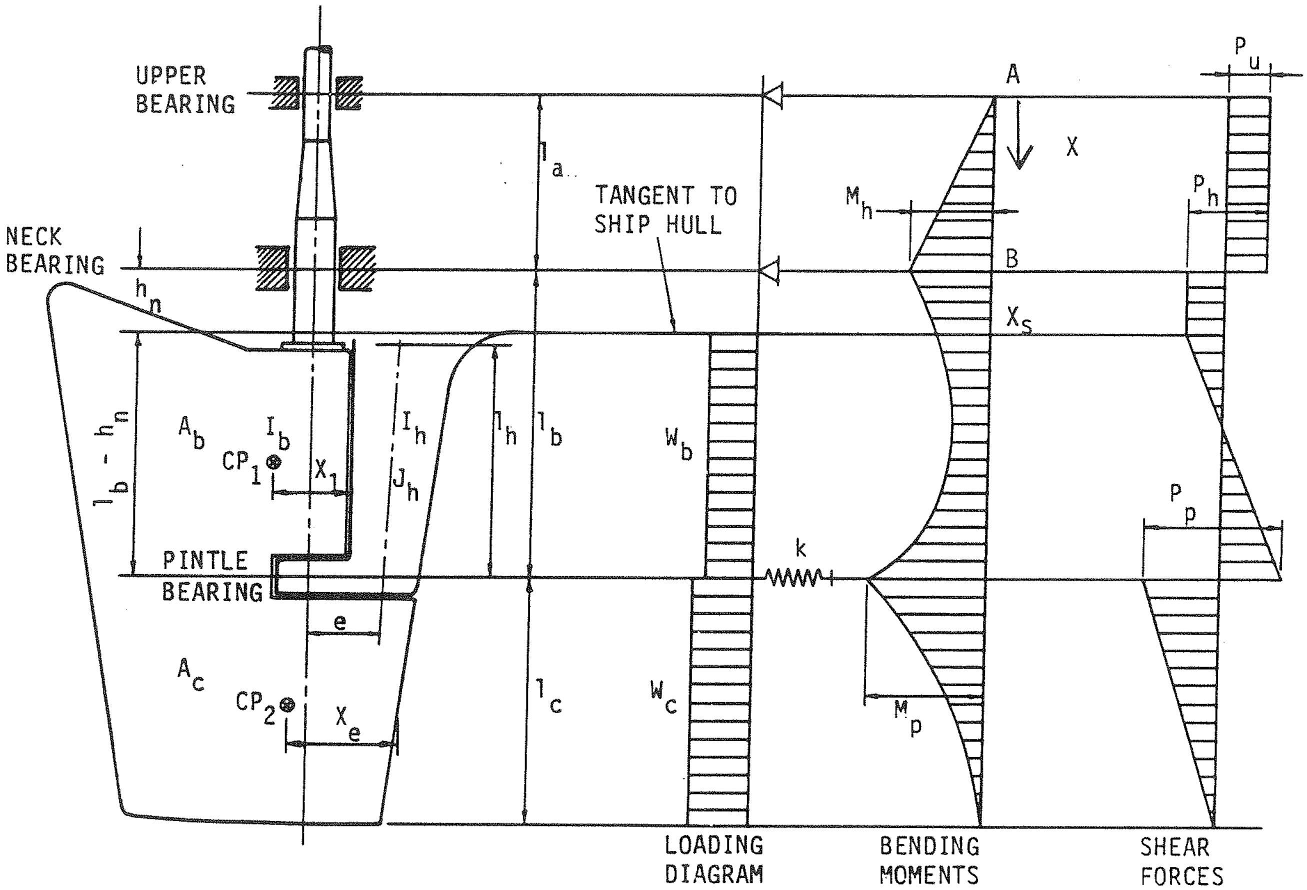

FIGURE 12 HORN TYPE CONTROL SURFACE - SPANWISE LOADING, SHEAR DIAGRAM AND BENDING MOMENT DIAGRAM

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I_b = Mean moment of inertia of control surface above pintle bearing (IN^4).

I_h = Mean moment of inertia of horn (IN^4).

E = Modulus of Elasticity ($KIPS/IN^2$).

P_u = Upper bearing reaction (KIPS).

P_h = Lower bearing reaction (KIPS).

P_p = Pintle bearing reaction (KIPS).

M_h = Bending moment at neck bearing (IN-KIPS).

M_p = Bending moment at pintle bearing (IN-KIPS).

N = Total force on control surface (KIPS).

J_h = Mean polar moment of inertia of horn (IN^4).

$$= \frac{4A_h^2}{\Sigma \frac{s}{t}}$$

A_h = Area in horizontal section enclosed by the horn (IN^2).

$\Sigma \frac{s}{t}$ = Circumferential length of control surface horn in horizontal section divided by its mean wall thickness.

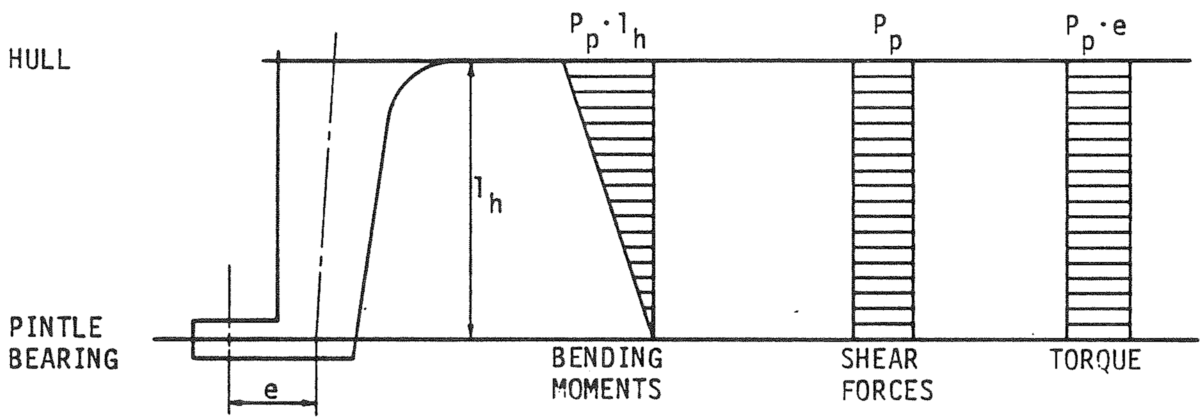
h_n, l_a, l_b and l_n are lengths as shown in Figure 12 (IN).

e = Distance from neutral axis of horn to center of pintle (IN).

NOTE: For horn control surfaces partially in the propeller race a resultant CP and N may be determined from the given loads on the portion within the race and that outside the race. Then the same procedure as outlined above can be used.

(d) Control Surface Horn Load

The shear force, bending moment and torque acting on a horn rudder horn are calculated based on Reference 13 as shown in Figure 13.



MAXIMUM BENDING MOMENT $= P_p \cdot l_n$ KIPS-FT

SHEAR FORCE $= P_p$ KIPS

TORQUE $= P_p \cdot e$ KIPS-FT

FIGURE 13 CONTROL SURFACE HORN - SHEAR FORCE, BENDING MOMENT AND TORQUE DIAGRAM

(e) Horn Type Control Surface - Chordwise Load

Follow the procedure for Spade Type Control Surface. See 562-2-f-2(b).

(f) Balanced Type Control Surfaces - Spanwise Load

The Spanwise loading diagram, shear diagram and bending moment diagram for a balanced type control surface is shown in Figure 14.

The bearing reaction forces are calculated as follows:

$$P_u = \frac{M_h}{l_a} \qquad (15)$$

$$P_h = P_u\left(1 + \frac{l_a}{l_b}\right) + \frac{N}{l_b}\left(\frac{h_r}{2} + h_d\right) \qquad (16)$$

$$P_c = N + P_u - P_h \qquad (17)$$

The bending moments are calculated as follows:

$$M_h = \frac{W_b\left(l_b^2 - h_n^2\right)\left(\frac{1}{2} + \frac{a}{8}\right)}{1 + a\left(1 + \frac{l_a I_b}{l_b I_a}\right)} \qquad (18)$$

$$M_F = \frac{W_b\ h_r^2}{8} \qquad (19)$$

When the neck bearing is located outboard of the control surface root chord and $h_n < 0.1\ h_r$ (see Figure 14), formula (18) may be used if the term "$- h_n^2$" in the numerator is replaced by "$+ h_n^2$".

Where:

$$a = \frac{K_d l_b^3}{3EI_b} \text{ OR } \frac{l_b^3 I_d}{l_d^3 I_b}$$

$$k_d = \frac{3EI_d}{l_d^3}$$

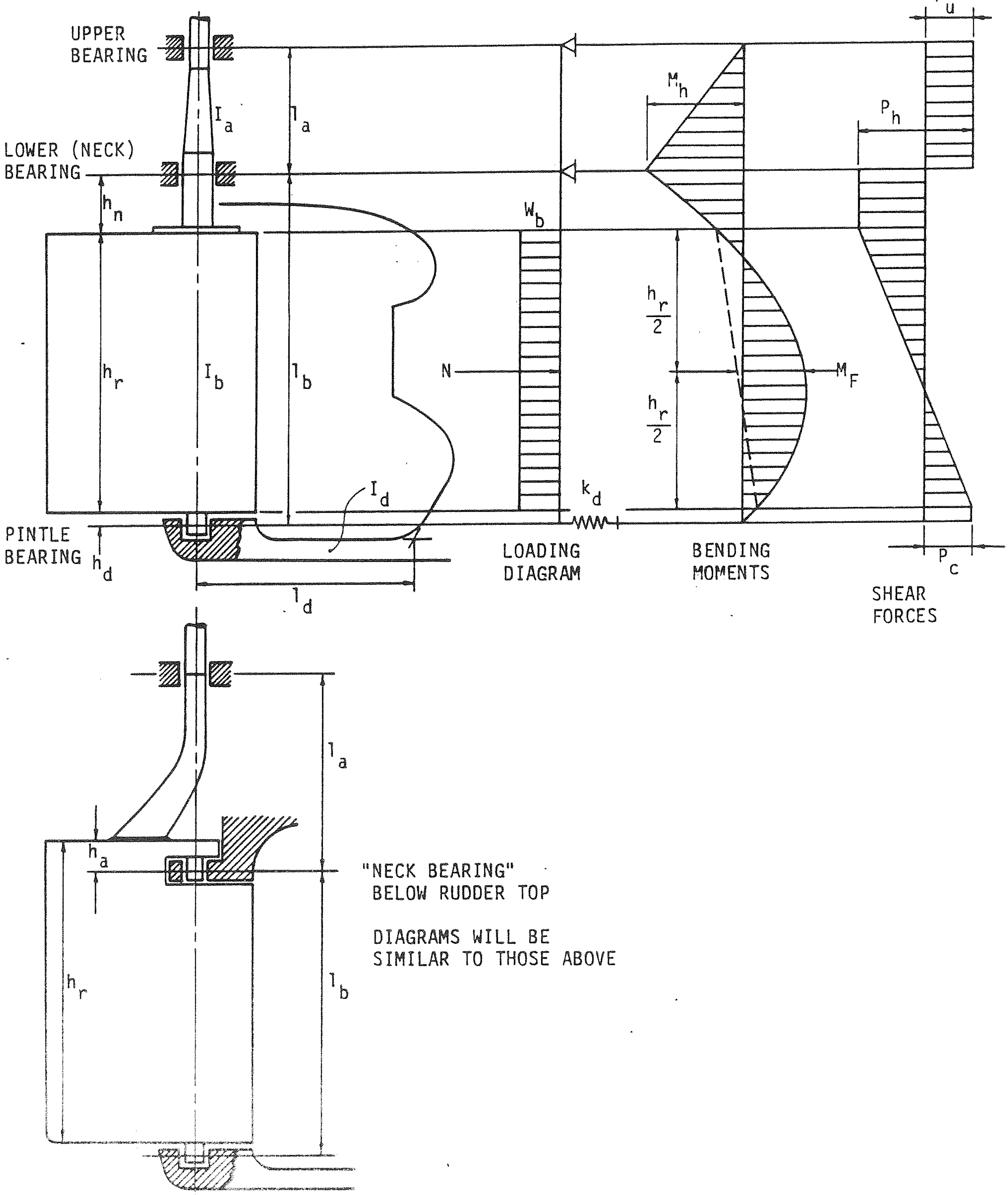


FIGURE 14 BALANCED TYPE CONTROL SURFACE - SPANWISE LOADING, SHEAR DIAGRAM AND BENDING MOMENT

N = Total force on control surface (KIPS).

I_a = Mean moment of inertia of stock (IN^4)

I_b = Mean moment of inertia of control surface (IN^4).

I_d = Mean moment of inertia of sole piece (IN^4).

l_d = Distance from centerline of pintle to assumed point of fixity of sole piece to hull or other structure (IN).

W_b = Uniformly distributed loading on control surface (KIPS/IN).

M_h = Bending moment at lower bearing (IN-KIPS).

M_F = Maximum bending moment on control surface (IN-KIPS).

P_u = Upper bearing reaction (KIPS).

P_h = Lower bearing reaction (KIPS).

P_c = Pintle reaction (KIPS).

E = Modulus of Elasticity ($KIPS/IN^2$).

h_d, h_r, h_n, l_a and l_b are lengths as shown in Figure 14 (IN).

(g) <u>Balanced Type Control Surface - Chordwise Loading</u>

See 562-2-f-2(b) for Spade Type Control Surface amd follow the same procedure.

562-2-f-3. <u>Shear and Bending Moment Diagrams and Bearing Reactions</u>

The shear and bending moment diagrams and bearing reactions should be determined for each combined loading.

The shear curves presented in the above procedures were determined by integrating the spanwise and chordwise load curves over the span and chord respectively. The bending moment curves were determined by integrating the shear curves.

Tiller forces have not been considered in the above procedures. If a tiller force exists, it may be included by determining the shears and bending moments due to the tiller force only, and superimposing them on those outlined herein. Appropriate boundary conditions should be assumed at the bearings when determining tiller shear forces and bending moments.

Mean moments of inertia for horn and balanced control surfaces, horns and sole pieces may be used in determining bending moments for horn and balanced control surfaces only in normal designs where variations in moments of inertia are not great. In other cases the variation of moments of inertia should be taken into account by some acceptable method.

562-2-f-4. Stock Deflection Analysis

The stock deflection at the seal should be determined so that a full load deflection does not cause excessive leakage through the seal.

The moment-area method can be used to find the stock deflection at any desired location. The procedure is as follows:

1) Tabulate the bending moment M from the upper bearing to the location of the seal. Let x be measured along the stock centerline, with $x = A$ at the upper bearing, $x = B$ at the lower bearing ($B > A$), and $x = X_s$ at the seal location, see Figure 12.

2) Determine the moment of inertia I at each x location selected in step 1.

3) Calculate M/(EI) at each x station (E is the elastic modulus).

For $x > B$, the deflection is:

$$v_x = t_{xB} + [(x - B)/(B - A)] * t_{AB}.$$

t_{xB} is the tangential deviation of a point x from a tangent to the elastic curve at B.

t_{xB} = (the area under the M/(EI) curve between x and B) *
(the distance from x to the centroid of that area).

Other deviations are calculated similarly, that is,

t_{AB} = (the area under the M/(EI) curve between A and B) *
(the distance from A to the centriod of that area).

Note that in general, t_{xB} is not equal to t_{Bx}.

For $x < B$, the deflection is:

$$v_x = [x/(B - A)] * t_{BA} - t_{xA}.$$

This method is applicable to control surfaces with or without outboard bearings (pintle bearings), and is valid as long as the M/(EI) curve does not cross through zero in the range between A and X_s. If the curve does cross through zero, the moment area method can still be used, but the formulas above are not valid and a more detailed moment area calculation will have to be used. Refer to a discussion of the moment area method in a strength of materials text for details in that case (for example, Chapter 11 of Reference 14.).

562-2-g Determination of Stock Size

The stock diameter may be determined by the formulas given below. The minimum stock diameter is to be taken as the larger of d_1 or d_2. Stocks may have bores for the purposes of heat treatment and weight reduction.

$$d_1 = \sqrt[3]{\frac{16n}{\sigma_y \pi (1 - p^4)} \left(M + \sqrt{M^2 + M_t^2}\right)} \qquad (20)$$

$$d_2 = \sqrt[3]{\frac{16n}{0.6\sigma_y \pi (1 - p^4)} \sqrt{M^2 + M_t^2}} \qquad (21)$$

Where:

d = Minimum stock diameter - larger of d_1 or d_2 (IN).

M = Bending moment in stock (IN-KIPS).

M_t = Torque in stock (IN-KIPS).

NOTE: In way of keyways the torque is to be increased as outlined in 562-2-i.

σ_y = Yield strength of stock material (KIPS/IN^2)

n = 2.0 - surface ship spade rudder

2.5 - surface ship horn rudder

2.0 - submarine control surface (hydrodynamic or ice loads)

1.25 - submarine control surface (sea slap)

$$p = \frac{\text{Inner diameter of stock (IN)}}{\text{Outer diameter of stock (IN)}}$$

562-2-h. Determination of Web Spacing and Web and Side Plating Thickness

All control surfaces to be constructed with horizontal and vertical webs covered by side plating.

562-2-h-1. Non-Filled Control Surfaces (also see 562-2-h-3)

(a) Panel Breadth to Thickness of Plating Ratio Based on Load

$$\frac{b}{t'} = \frac{C}{k\sqrt{H}} \qquad (22)$$

Where:

b = Breadth of panel (IN).

a = Length of panel (IN).

The following relationship exists between a and b:

$b \approx a$

t' = Thickness of plating based on loading (IN).

H = Maximum total head (FT).

= Σ (Hydrodynamic (or Ice Load) and Hydrostatic Head acting on the panel or sea slap, whichever is the larger).

<u>Hydrodynamic Head</u>

The spanwise head distribution may be determined from the given spanwise load distribution. The chordwise distribution of the head may be assumed similar to the given total chordwise distribution at the mean chord.

<u>Ice Load</u>

See paragraph 562-2-e-3(b).

<u>Sea Slap Head</u>

A rectangular shaped spanwise sea slap head distribution may be assumed for surface ship rudders. The maximum value of the sea slap head will be given in the ship specifications. For submarines the sea slap and distribution will be given in the ship specifications.

<u>Hydrostatic Head</u>

The hydrostatic head should be determined based on the full load draft of the ship.

C = Constant depending on material (Table 2).

K = Constant depending on b/a (Table 3).

(b) <u>Plating Thickness</u>

$$t = t' + t_1 + t_2 \qquad (23)$$

Where t = Thickness of outside control surface plating (not to be less than 0.5 IN unless noted in the specifications)

t' = Assumed base thickness of outside plating

NOTE: t' may be chosen arbitrarily.

Table 2. Values of "C" for Steel Type

Types of Steel	Value of C
OS	550
HS	620
HY80	750
HY100	810

Table 3. Values of "K" vs b/a

b/a	Value of K
0.5 or less	1.00
0.6	.98
0.7	.94
0.8	.89
0.9	.84
1.0	.78

t_1 = 0.06 IN (corrosion allowance)

t_2 = 0.0 IN (Unless otherwise specified in the ship specifications - cavitation allowance)

(c) Panel Breadth to Thickness Ratio, Breadth(b) and Length(a)

The ratio b/t is to be assumed within the following limits:

$40 \leq b/t \leq 50$ For surface ships only.

If b/t is not within this interval, a new t' may be assumed which gives b/t equal to the nearest limit.

"a" should be approximately equal to "b."

562-2-h-2. Filled Control Surfaces (also see 562-2-h-3)

(a) Panel Breadth to Thickness Ratio - b/t

b/t may be chosen between the following limits:

$70 \leq b/t \leq 85$ For submarines only.

t = Thickness of control surface side plating (not to be less than 0.25 IN)

b = Breadth of panel (IN)

(b) Plating Thickness - t

$$t = t' + t_1 + t_2 \qquad (29)$$

Where:

t' = 0.19 IN

t_1 = 0.06 IN (corrosion allowance)

t_2 = 0.0 IN (Unless otherwise specified in the ship specifications - cavitation allowance)

(c) Panel Breadth and Panel Length

The length of a panel "a" should approximate the panel breadth "b."

562-2-h-3. Other Considerations (See Figure 15 also)

(a) Trailing Edge

At the trailing edge of the control surface a rabbeted casting or forging should be used.

(b) Leading Edge

The leading edge of the control surface should be rolled plate of at least the maximum thickness of the outside plating.

(c) Web Thickness

Web thicknesses in non-filled control surfaces shall be not less than 70 percent of the side plating thickness.

Web thicknesses in filled control surfaces shall not be less than 0.25 IN.

(d) Variations in Web Spacing

The web spacings determined by the methods outlined above may require some modifications in way of hub castings.

562-2-i. Determination of Hub Casting Thickness

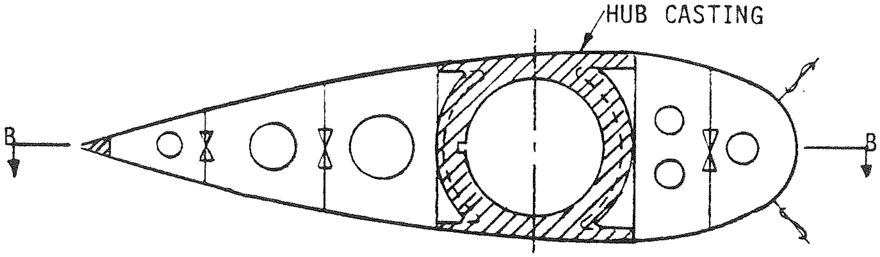
A hub casting must be incorporated into the structure of the control surface for purposes of accepting the stock, transmitting torque between the stock and control surface, helping to resist the bending moment and providing means of attachment of the control surface to the stock. An example of a hub casting and its attachment to webs and plating is shown in Figure 15 for a spade rudder.

An appropriate hub casting should be designed which performs the above-cited functions and has the following characteristics:

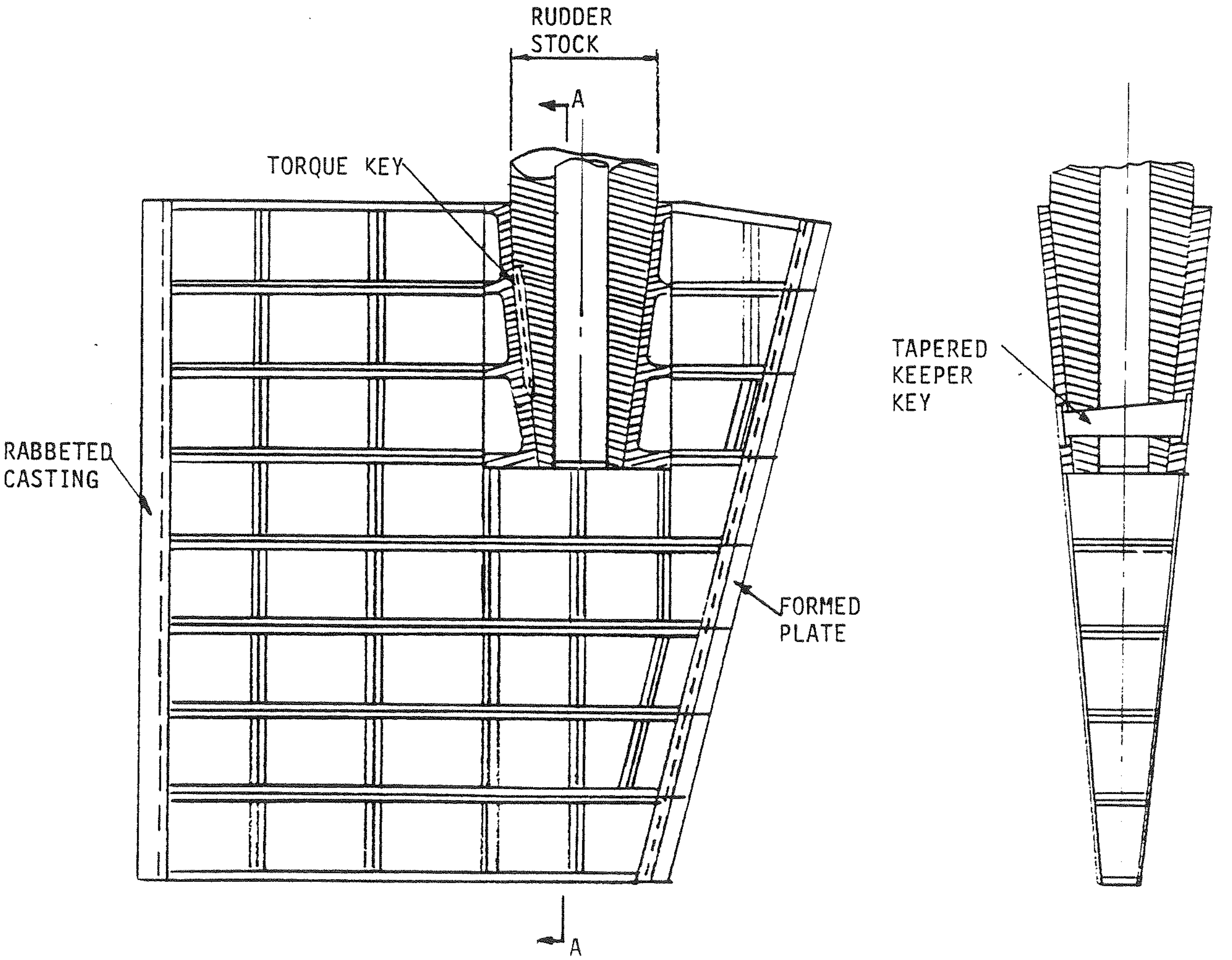
1. The stock should penetrate the hub no less than two stock diameters at the root chord.

2. The minimum hub wall thickness must be no less than 0.15 stock diameter.

3. The taper of the hub-to-stock interface should be approximately two per foot of stock diameter.



TYPICAL HORIZONTAL SECTION



SECTION AT B-B

SECTION AT A-A

FIGURE 15 HUB CASTING

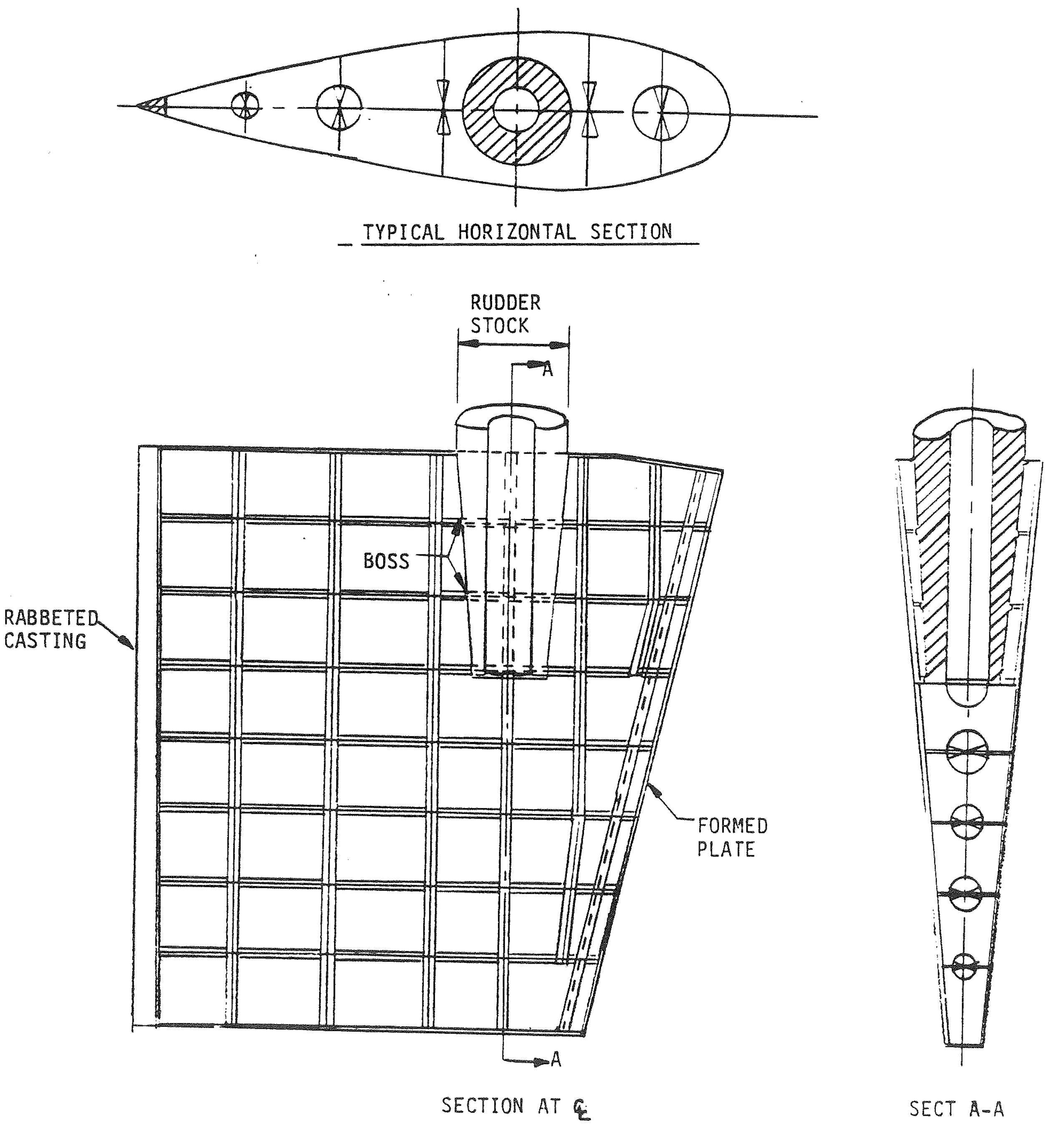


FIGURE 15a - CONTROL SURFACE WITH INTEGRAL STOCK

4. If relief is provided along the hub-stock interface, no more than the middle third of the hub may be relieved.

Local stress calculations of the hub need not be performed except where keys and keyways are located or where some unconventional arrangement is used. However, hub strength should be considered when determining overall bending and shear of the control surface. The moment of inertia of the hub and associated stock used in these calculations should be that given below:

$$I_H = \frac{I_{H-A}}{K} \tag{30}$$

where,

I_{H-A} = Actual moment of inertia of the hub and associated stock about centroidal axis of control surface section (IN^4).

K = Stress Factor depending on stock penetration (from Figure 16).

I_H = Moment of inertia of hub and associated stock to be used in strength calculations (IN^4).

When shipping and unshipping of the rudder in a drydock does not require special high blocking, the hub casting may be eliminated. Then the rudder stock is welded onto the rudder horizontal webs as shown in Figure 15a. In such cases, the rudder stock should have machined lands or bosses onto which the rudder horizontal webs are welded. There should be at least four horizontal webs welded onto the rudder stock, and the penetration of the stock into the rudder should be at least 2.5 times the rudder stock diameter at the top of the rudder. The thickness of the webs attached to the rudder stock should be 50 percent thicker than the webs used with a hub casting; the total spanwise length of these thicker webs should be at least equal to 2.0 times the rudder stock diameter at the top of the rudder, similarly for other types of control surfaces.

562-2-j. Spanwise and Chordwise Stresses on Control Surface Body

The spanwise and chordwise stresses may be determined using the moments and shears from 562-2-f. Stresses should be calculated for suspected critical areas, especially where there is a sudden change of moment of inertia.

562-2-k. Gudgeons, Pintles and Keyways, Grooves and Discontinuities

1. Gudgeons

The stress in a gudgeon may be determined from the following formula:

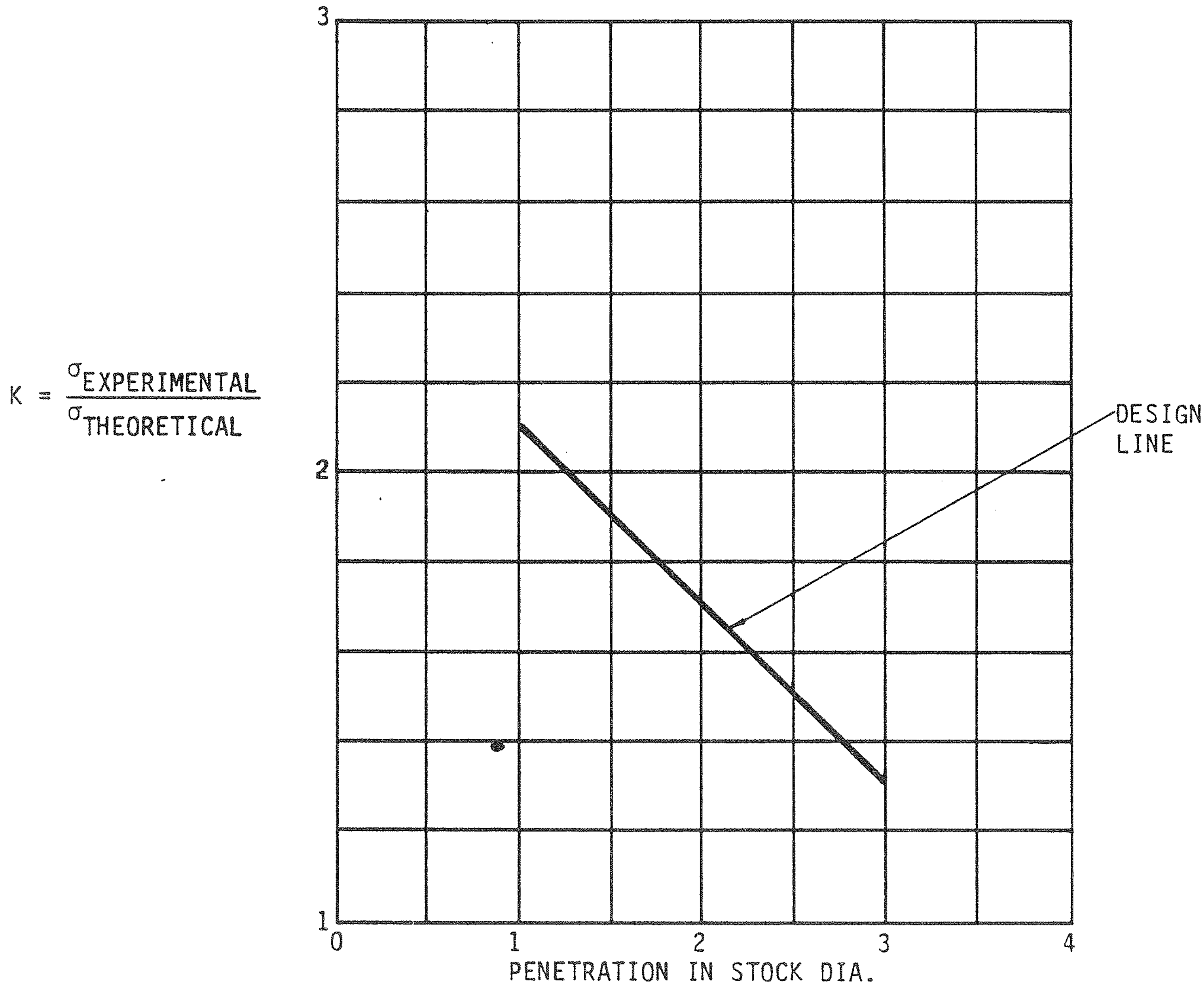


FIGURE 16 "K" STRESS FACTOR VERSUS PENETRATION

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$$\sigma_G = \frac{GT}{10al} \qquad (31)$$

where,

$$a = \frac{R_2 + R_1}{20} \text{ (IN).}$$

R_1 = Outside radius of gudgeon (IN).

R_2 = Inside radius of gudgeon (IN).

l = Length of gudgeon (IN).

T = Total reaction at gudgeon (or shear force at horn) (KIPS)

G = Gudgeon Strength Factor from Figure 17.

σ_G = Maximum stress in gudgeon (KSI). Allowable stress is as per 562-2-e-4.

2. Pintles

In the design of pintles only the shear stresses in the pintle need be considered. A minimum factor of safety of 5.0 on yield is to be provided.

3. Keyways

Whenever keys are used with stocks to transmit torque between the stock and the control surface, stress concentration in the stock due to the keyway should be considered.

A stress concentration factor of 1.0 may be assumed for bending. Torsional stress concentration factors can be determined from Figure 18. Torques and bending moments in way of keyways are to be multiplied by their respective factors in order to determine effective stresses.

4. Grooves and Discontinuities

Of all discontinuities, horizontal grooves for rudder carrier plates are the most often used. The location of this groove on the rudderstock could be critical because it is a high stress riser, see Figure 19 and reference 11. When the situation is such that the location of the groove does cause high stress, a built-in flange for the carrier plate instead of a groove could be the solution; however, shipping and unshipping of the stock must be reanalyzed to ensure ease of maintainability.

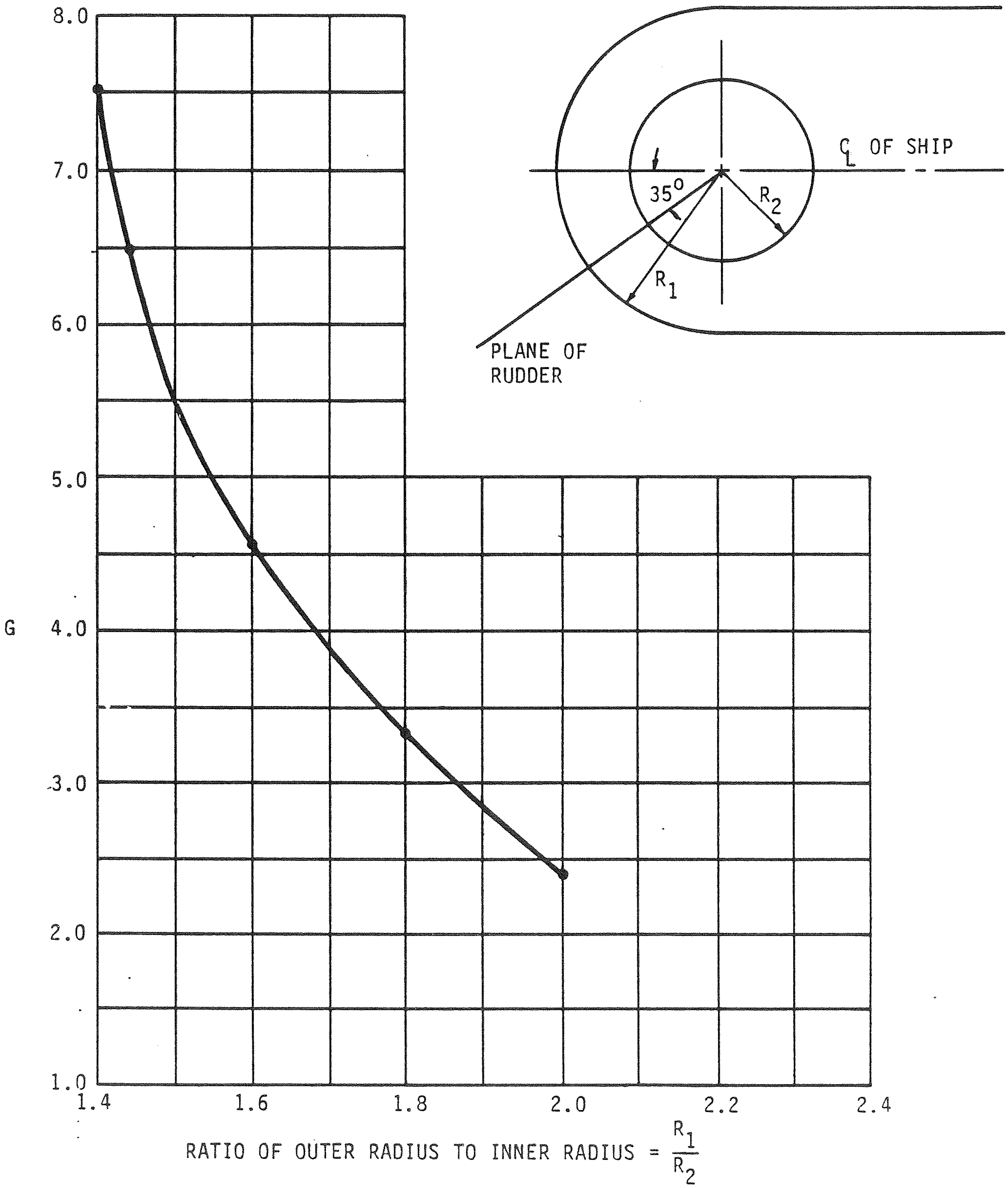

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FIGURE 17 CURVE FOR GUDGEON STRENGTH FACTOR

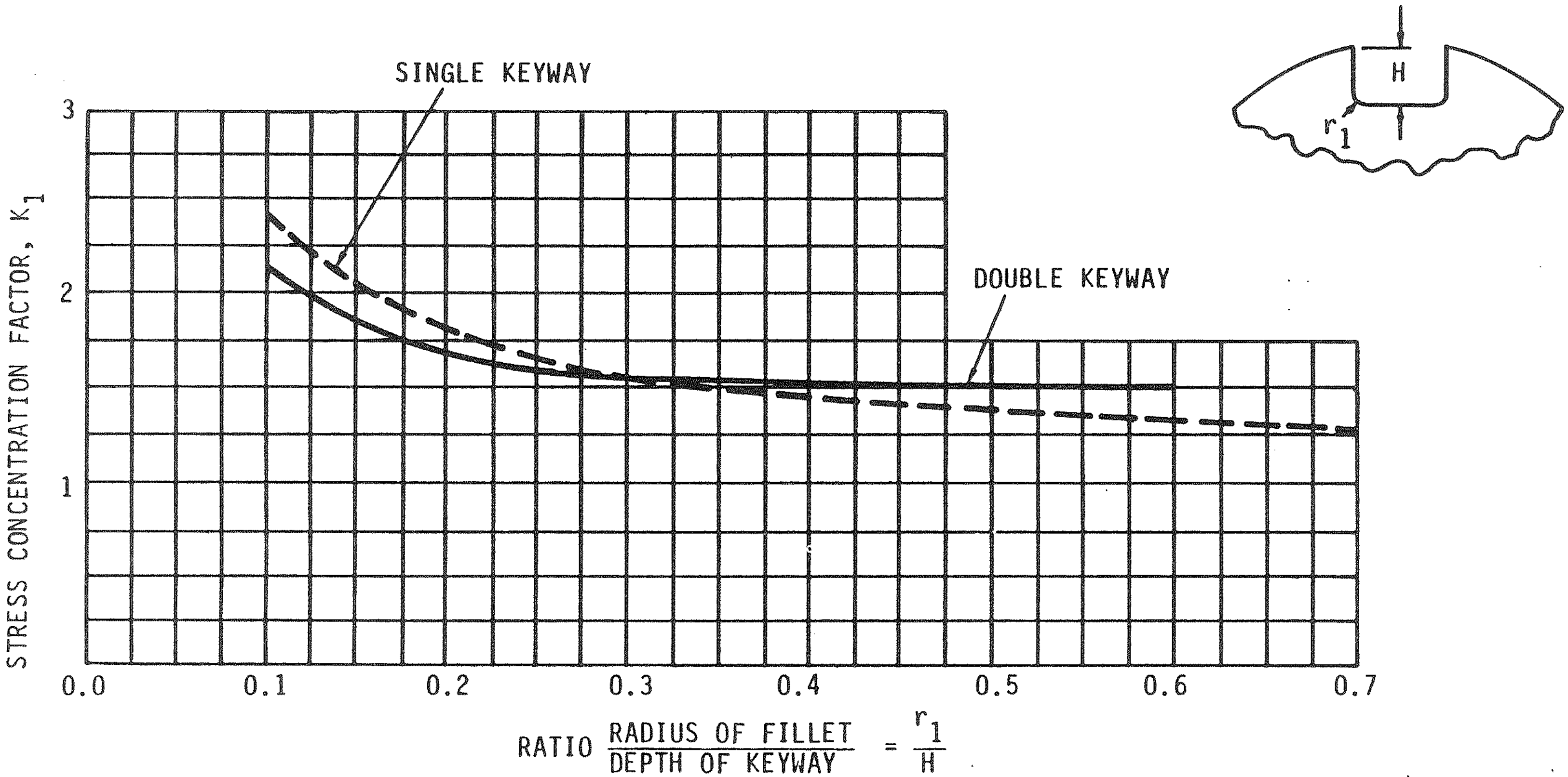


FIGURE 18 STRESS CONCENTRATION FACTOR AT KEYWAY FILLET, IN TORSION

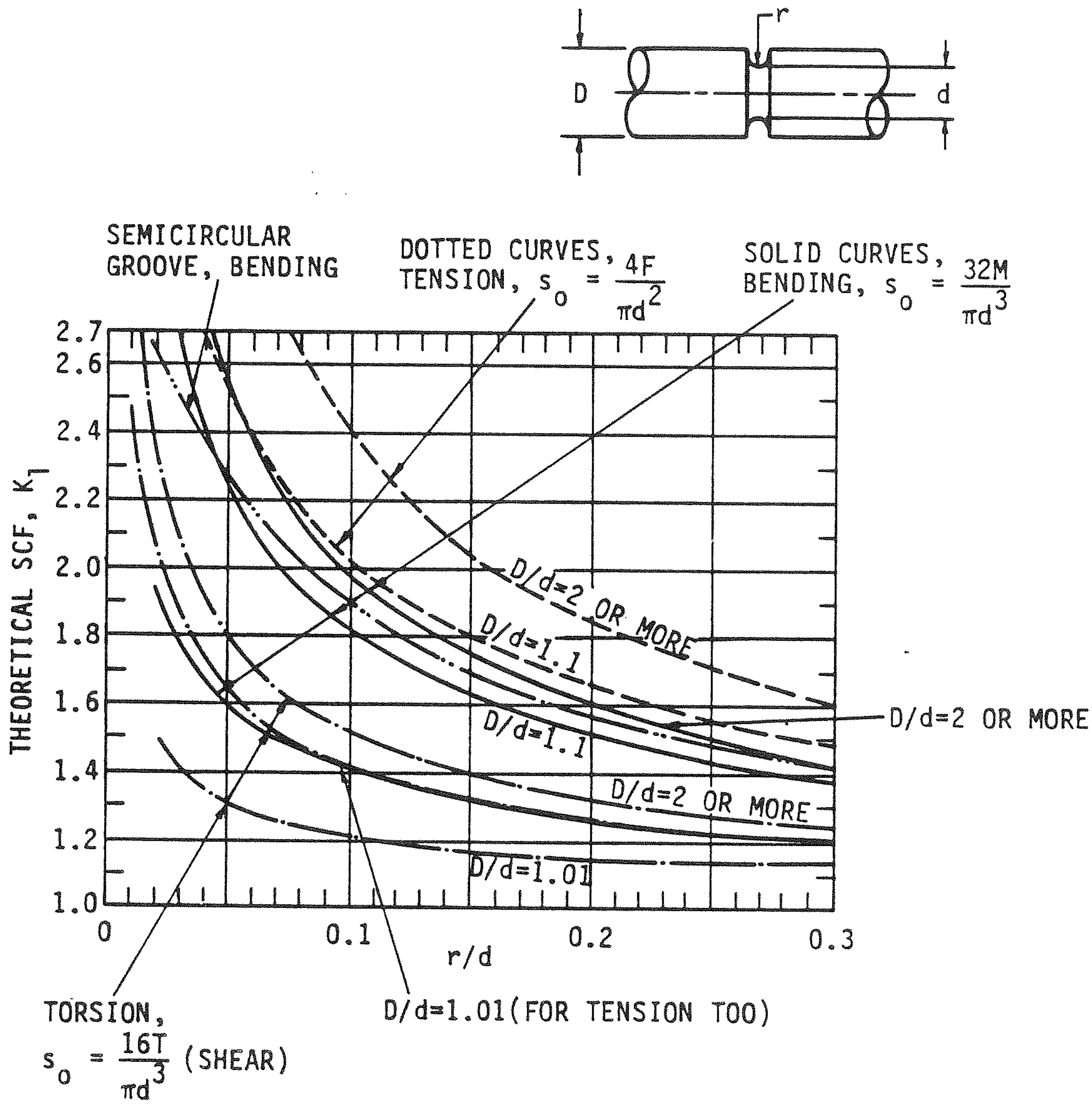


FIGURE 19 STOCK WITH GROOVE. USE THE SOLID CURVE FOR D/d = 1.01 FOR TENSION AS WELL AS BENDING (APPROXIMATE)

562-2-l. Cathodic Protection

Corrosion protection of control surfaces is provided by galvanic anode protectors (Zincs). The quantity of anodes shall be determined from section 633 of the ship specification. Forty percent of the hull anodes shall be installed in the stern area in a location not interfering with the struts, propellers, propeller shafts, or rudders. Anodes shall be installed equally divided, port and starboard, on the hull adjacent to the propellers in locations always submerged when the ship is not moving. For submarines zinc anodes shall be attached to the hull plating in the stern area, approximately five feet forward of the propeller planes. Zinc anodes shall not be attached to rudders and diving planes.

562-2-m. Determination of Rudder Scantlings by ABS RULES

For auxiliary ships that are required to meet the requirements of the American Bureau of Shipping the rudder scantling must meet at least the requirements of Reference 9 or other ABS publications for special ship types. ABS uses empirical formulae without definition of rudder torque. Diameters of stocks and pintles and thicknesses of rudder plates are dependent on the ship's design speed, total projected area of the rudder, and the type of rudder.

562-2-n. Strengthening for Navigation in Ice

If an auxiliary ship is to navigate in ice and to receive special classification for such service then the strength of the rudders and steering gear is to be increased over that required by ABS (Reference 9). ABS Rules specify increased strength of the rudder stock and thickness of rudder plating depending on the class of ice-strengthening required. For the highest class of ice-strengthening the strength of the rudder stock is to be increased 85%. The thickness of rudder plate, and horizontal and vertical diaphrams are to be increased by 50%. For the lowest class of ABS ice-strengthening the rudder stock strength must be increased by 25 percent.

Navy surface ships do not normally require special strengthening for navigation in ice. If ice strengthening is required the guidelines provided in Reference 9 can be used. Guidance on the equivalency of various ice-strengthening classes by the major classification societies can be found in Reference 10. Reference 8 also provides guidelines for determining the degree of ice strengthening required.

562-2-o. Examples

Example calculations are presented for the following control surfaces:

Example 1 - Surface ship spade rudder.

Example 2 - Surface ship horn rudder.

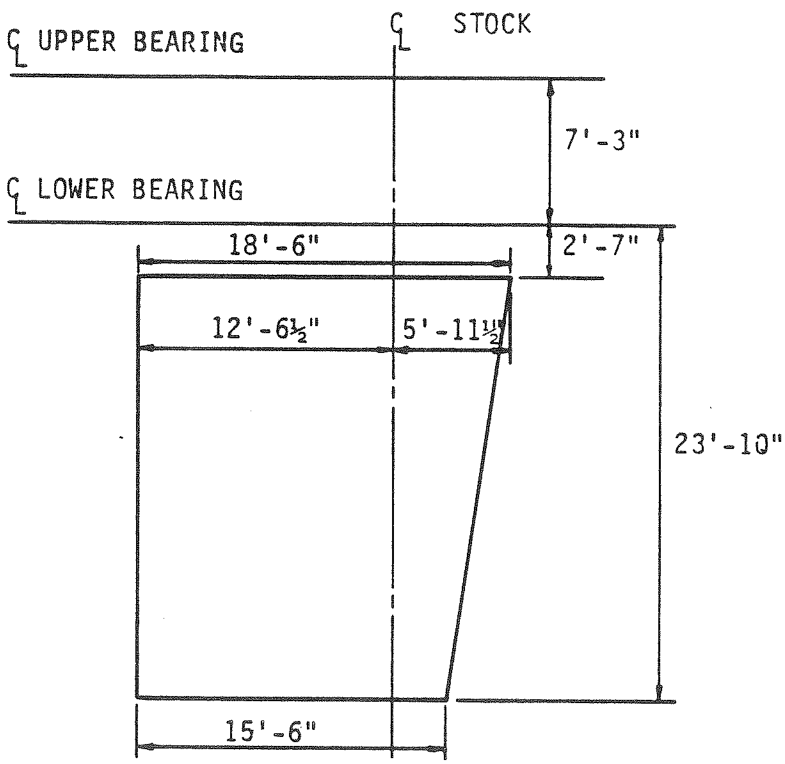
Example 3 - Submarine stern diving plane.

EXAMPLE I

(a) INPUT: The Ship Specification gives the following information:

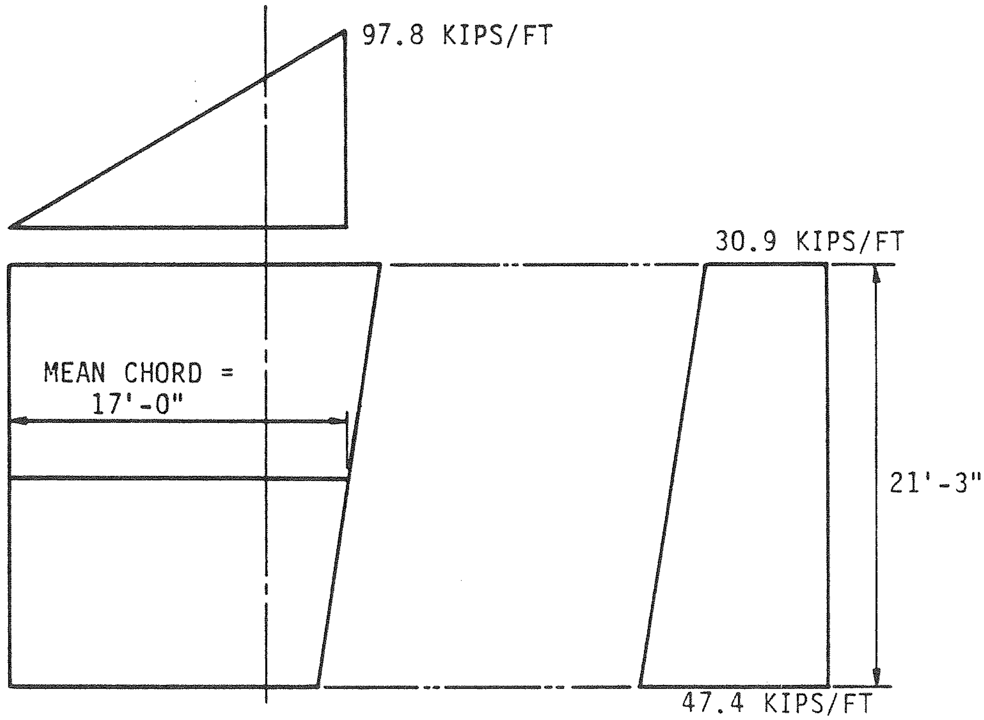
Ship: Surface Vessel

Control Surface: Spade Rudder - Hollow



Torque: Max. Torque = 12,600 IN-KIPS at ahead speed of 25 knots

Loadings: Spanwise and Chordwise Hydrodynamic Load Distribution



Hydrodynamic (@ mean chord)

Chordwise Load Distribution

Hydrodynamic

Spanwise Load Distribution

Load Combinations: For Overall Stress Hydrodynamic Load only.

Material: Rudder Plate - HY 80
Rudder Stock - Steel Forging, MIL-S-23284 and as noted in Table 4

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(b). DETERMINATION OF BENDING MOMENT AND SHEAR DIAGRAMS, AND BEARING REACTIONS:

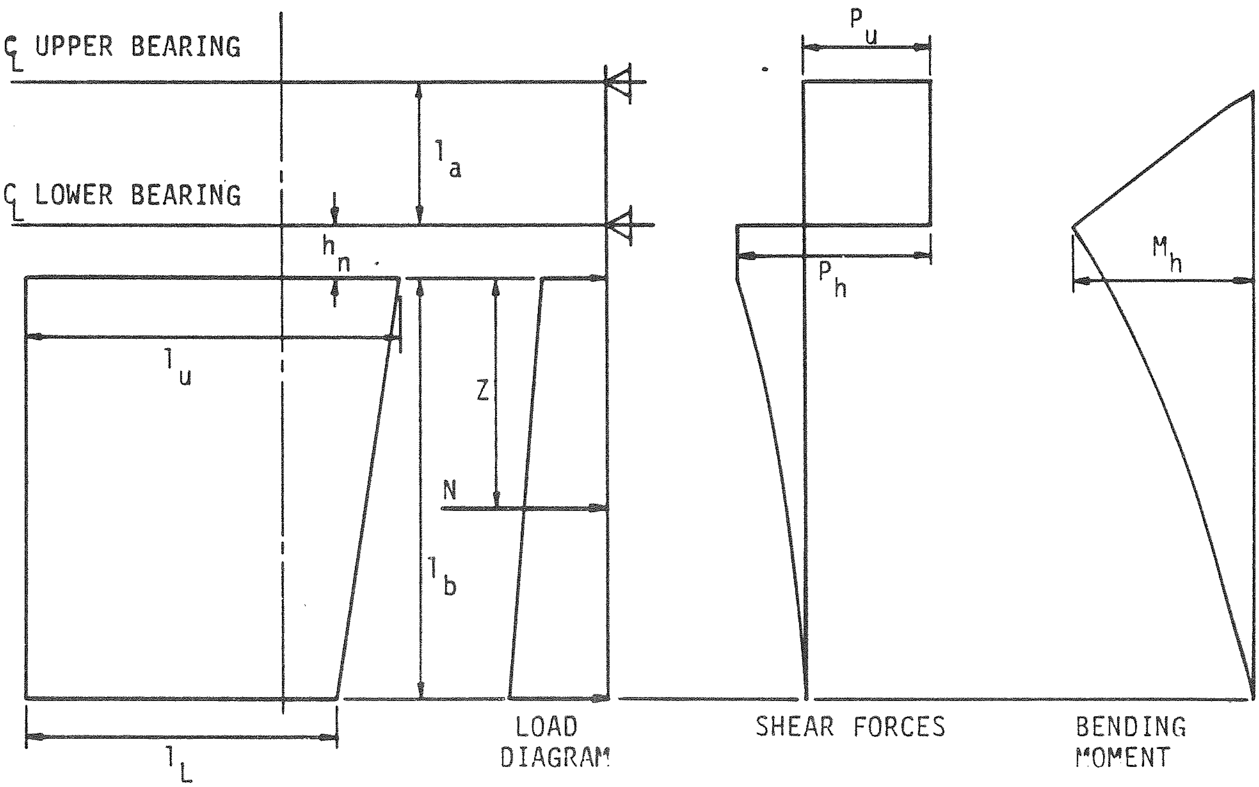
A. Combining Loads for Spanwise and Chordwise Shear and Bending Moment Diagrams:

Hydrodynamic Load only

B. Shear and Bending Moment Diagram and Bearing Reactions:

(Spade Type Control Surface)

a. Spanwise Loading



Reaction Forces:

$$P_u = \frac{N(Z + h_n)}{l_a} \text{ KIPS}$$

$$= \frac{(832) \times (11.35 + 2.58)}{7.25} = 1600 \text{ KIPS}$$

$$P_h = N + P_u = 832 + 1600 = 2432 \text{ KIPS}$$

Max. Bending Moment:

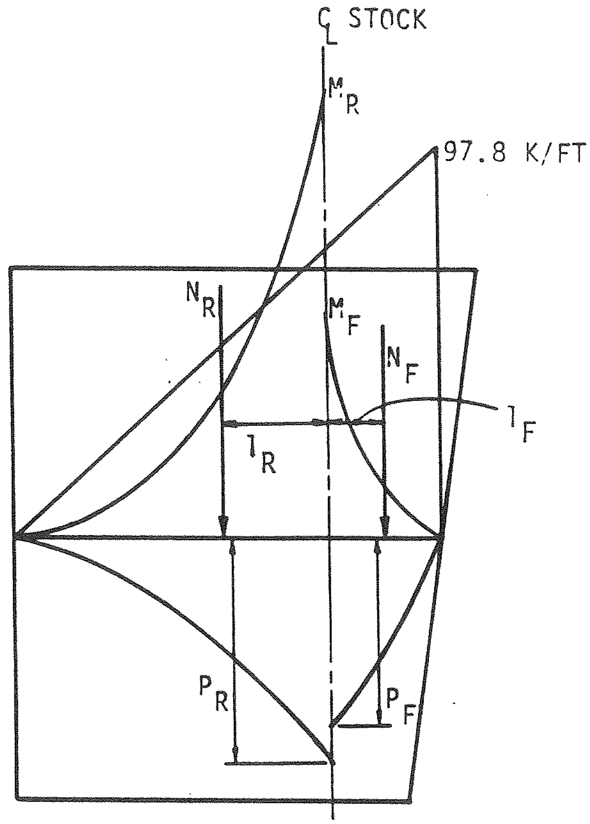
$$M_h = N(Z + h_n)$$
$$= 832(11.35 + 2.58) = 11{,}600 \text{ KIPS-FT}$$

b. <u>Chordwise Loading</u>

N_R = 453 KIPS
l_R = 4.18 FT
N_F = 379 KIPS
l_F = 2.34 FT

Reaction Forces:

$P_R = N_R$ = 453 KIPS
$P_F = N_F$ = 379 KIPS



Bending Moment: $M_R = l_R N_R = (4.18)(453) = 1894$ KIP-FT
$M_F = l_F N_F = (2.34)(379) = 887$ KIP-FT

(c) <u>DETERMINATION OF WEB SPACING AND WEB AND SIDE PLATING THICKNESSES:</u>

<u>(Hollow Control Surfaces):</u>

a. <u>b/t:</u>

$$\frac{b}{t'} = \frac{C}{K\sqrt{H}}$$

Assume b/a = .83. Then K = 0.87
C = 750 for HY 80.

Max. Load (at tip) = 61.3 KIPS/FT

$$H = \frac{61300^{in/ft} \times 2}{15.5^{FT} \times 64^{in/ft^3}} = 123.5 \text{ FT}$$

$$\therefore \frac{b}{t'} = \frac{750}{.87 \times \sqrt{123.5}} = 77.6$$

b. Plating Thickness - t:

Assume: $t' = .5025$ IN

$t = t' + t_1 + t_2 = .5025 + .06 + .5625$ IN

c. b and a:

$b = (b/t')t' = (77.6)(.5025) = 38.99$ IN

$b/t = 63.8$

$a = (38.99)/(.83) = 46.98$ IN

Determination of Stock Size:

$$d_1 = \sqrt[3]{\frac{16n}{\sigma_y \pi(1-P^4)}\left(M + \sqrt{M^2 + M_t^2}\right)} = \sqrt[3]{\frac{(16)\cdot(2)}{(65)(\pi)(1-0.17^4)}\left(139{,}200 + \sqrt{(139{,}200)^2 + (12{,}000)^2}\right)}$$

$$= 35.2 \text{ IN}$$

$$d_2 = \sqrt[3]{\frac{16n}{0.6\pi\sigma_y(1-P^4)}\sqrt{M^2 + M_t^2}} = \sqrt[3]{\frac{(16)(2)}{(0.6)(\pi)(65)(1-0.17^4)}\sqrt{(139{,}200)^2 + (12{,}000)^2}}$$

$$= 33.2 \text{ IN}$$

USE d = 35.5 IN*

* Stock diameter may have to be increased in order to accommodate standard size bearings.

(e) DETERMINATION OF HUB CASTING:

The minimum hub wall thickness and required taper ratio are determined as follows:

$\sigma = KMc/I$

where, $c = d_1/2 + s$, IN

s = hub wall thickness, IN

$I = \pi/64[(d_1 + 2s)^4 - d_2^4)]$

Since d_2, the inner stock diameter, is much smaller than d_1 ($d_2 = 0.17d_1$), it may be neglected.

Then

$I = \pi/64(d_1 + 2s)^4$

$= \pi/64(2c)^4 = \pi/4(c)^4$

Stock penetration into hub =7.3 FT = 87.6 IN

=2.5 stock diameters

From Figure 16, K = 1.5

Therefore,

$$\sigma = \frac{1.5\ Mc}{\pi/4(c)^4} = \frac{6M}{\pi c^3}$$

or

$c^3 = 6M/\pi\sigma$

For $\sigma = 0.6\ \sigma_y = 39{,}000$ PSI

M=139,200 IN-KIPS

and d_1 = 35.5 IN

$c^3 = (6)(139{,}200 \times 1000)/(39{,}000\ \pi)$

=6,817

Therefore,

$c = d_1/2 + s = 18.96$ IN

or

s=18.96-35.5/2 = 1.21 IN

Since this value for the hub wall thickness is less than 15 pecent of the stock diameter, the thickness must be increased to

s=0.15 x 35.5 = 5.325 IN

Finally, the taper ratio of stock diameter = 14(IN)/7.3(FT)=1.92 IN/FT.

(f) SPANWISE AND CHORDWISE STRESS: See Table 4

TABLE 4
STRESS DIAGRAM

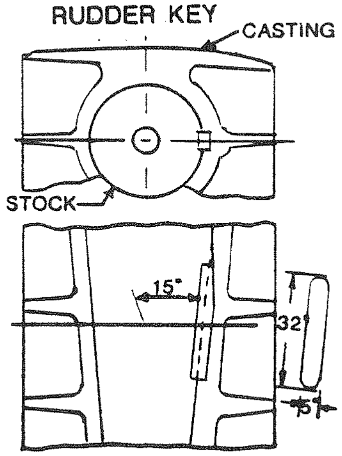
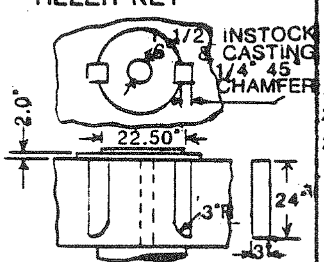
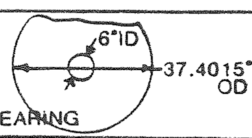
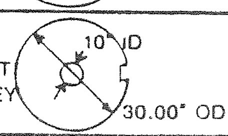
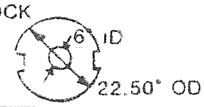
MATERIAL	KIND OF STRESS	SECTION: SHAPE	SECTION: AREA (IN^2)	SECTION: SECTION MODULUS (IN^3)	MAXIMUM LOAD OR BENDING MOMENT	MAX COMBINED STRESS (PSI)	% OF YIELD POINT
HUB CASTING STEEL PLATE	BENDING	CHORDWISE SECTION OF RUDDER	—	3200	49,500K-IN	15,300	51.5
"	BENDING	SPANWISE SECTION OF RUDDER	—	30,774	22,600K-IN	736	1.13
"	BENDING	SPANWISE SECTION OF RUDDER	—	5432	10,200K-IN	1880	626
STEEL MIL-S-21952 TY-B HY80 YP 48,000 PSI	SHEAR	RUDDER KEY 	154.63		$\frac{12000 \times 10^3}{15}$ = 800,000 lbs	5170	10.8
STEEL MIL-S 21952 TY-B HY80 YP 128,000 PSI	BEARING ON STOCK & KEY		27"x1.75= 47.25		$\frac{12000 \times 10^3}{14}$ =857,000 lbs	18,200	14.2
STEEL CASTING MIL-S 15085 CLASS B YP 48,000 PSI	BEARING OF KEY ON RUDDER CASTING		27"x1.75= 47.25		$\frac{12000 \times 10^3}{16}$ = 750,000 lbs	15,900	33.2
STEEL MIL-S-21952 TY B HY80 YP 48,000 PSI.	SHEAR	TILLER KEY 	2(3x24) = 144		$\frac{12000 \times 10^3}{11.5}$ =1,042,000 lbs	7240	15.1
STEEL MIL-S-21952 TY B HY80 YP 129,000 PSI	BEARING ON STOCK & KEY		2(1.25x 22.5) = 56.3		$\frac{12000 \times 10^3}{10.25}$ =1,170,000 lbs	20,800	16.3
STEEL CASTING MIL-S-15083 CLASS B YP 48,000 PSI	BEARING OF KEY ON TILLER CASTING		2(1.25x 24) = 60.0		$\frac{12000 \times 10^3}{12.75}$ =942,000 lbs	15,720	32.8
RETAINER RING STEEL MIL-S-22698B YP 52,800	SHEAR		22.5π x2 = 141.0		165,000 lbs	1170	2.2
STEEL FORGING (RUDDER STOCK)	TORSION BENDING COMBINED	RUDDER STOCK SECTION AT ℄ LOWER BEARING 		10,266 5133	12,000" KIPS 139,304" KIPS	1170 27,100 27,200	41.8
"	TORSION BENDING COMBINED	RUDDER STOCK SECTION AT RUDDER KEY 		5293 $\frac{370,000}{15}$ = 24600	12,000" KIPS 84,720" KIPS	3970 4660 6920	10.7
"	TORSION	RUDDER STOCK SECTION AT CROSSHEAD 		2212	12,000" KIPS	10,800	27.7

TABLE 4
STRESS DIAGRAM

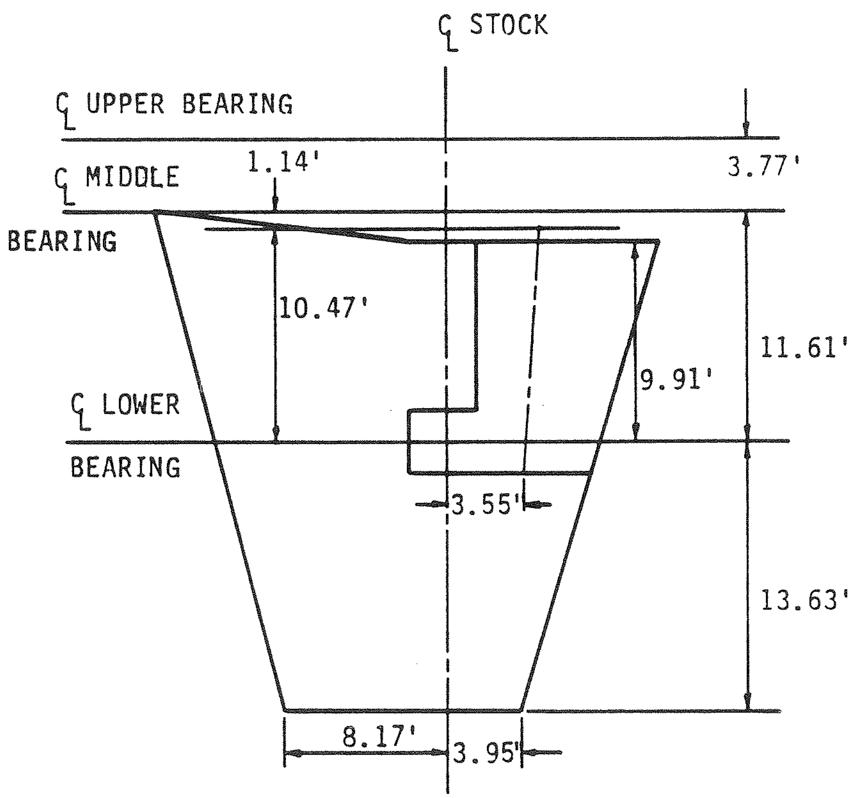
MATERIAL	KIND OF STRESS	SECTION			MAXIMUM LOAD OR BENDING MOVEMENT	MAX COMBINED STRESS (PSI)	% OF YIELD POINT
		SHAPE	AREA (IN^2)	SECTION MODULUS (IN^3)			
	THRUST LOAD	UPPER BEARING			165,000 lbs		
	RADIAL LOAD	UPPER BEARING			1,620,000 lbs		
	RADIAL LOAD	LOWER BEARING			2,452,000 lbs		

<u>EXAMPLE II</u>

(a) <u>INPUT</u>: The Ship Specification gives the following information:

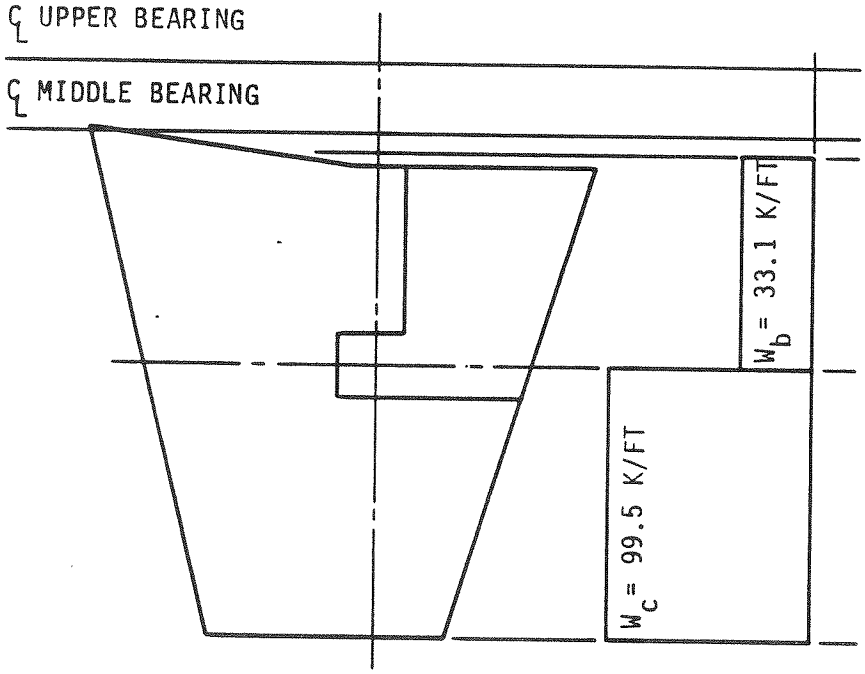
<u>Ship</u>: Surface Vessel

<u>Control Surface</u>: Horn Rudder - Hollow



<u>Torque</u>: Max. Torque = 16,135 IN-KIPS at ahead speed of 33 knots

<u>Loadings</u>: Spanwise and Chordwise Hydrodynamic Load Distribution



Hydrodynamic

Spanwise Load Distribution

NOTE: Only the spanwise loading is shown for example, the chordwise loading will be given similarly to that in Example I in reality.

Load Combinations: For Overall Stress Hydrodynamic Load only.

Material: Rudder Plate - HY 80
Rudder Stock - MIL-S-23284

(b) DETERMINATION OF BENDING MOMENT AND SHEAR DIAGRAMS, AND BEARING REACTIONS:

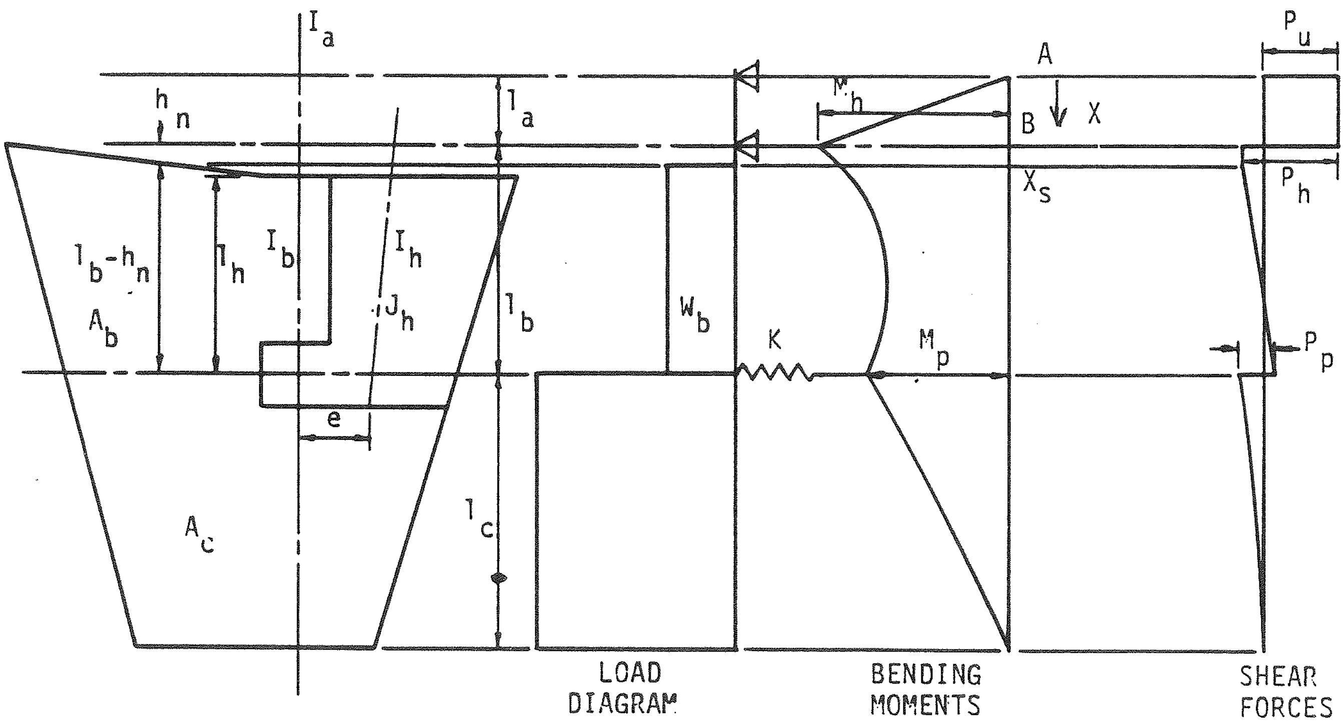
A. Combining Loads for Spanwise and Chordwise Shear and Bending Moment Diagrams:

Hydrodynamic Load only

B. Shear and Bending Moment Diagram and Bearing Reactions:

(Horn Type Control Surface)

a. Spanwise Loading



Where: N = 1702 KIPS
I_a = 27,000 IN^4 (Estimated)
I_b = 160,000 IN^4 (Estimated)
I_h = 760,000 IN^4 (Estimated)
J_h = 37,000 IN^4 (Estimated)

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Max. Bending Moments:

$$M_p = \frac{W_c \cdot l_c^2}{2} = \frac{(99.5)\cdot(13.63)^2}{2} = 9242 \text{ KIPS-FT}$$

$$K = \frac{E}{l_h\left(\frac{2.6e^2}{J_n} + \frac{l_h^2}{3I_h}\right)} = \frac{E}{(9.91)\left(\frac{(2.6)(3.55)^2}{(37{,}000)/(12)^4} + \frac{(9.91)^2}{3\cdot(760{,}000)/(12)^4}\right)}$$

$$= \frac{E}{190.8} \text{ KIPS/FT}$$

$$a = \frac{Kl_b^3}{3EI_b} = \frac{\frac{E}{190.8}\cdot(11.61)^3}{(3)(E)((160{,}000)/(12)^4)} = .35$$

$$M_h = \frac{W_b(l_b^2 - h_n^2)(\frac{1}{2} + \frac{a}{8}) + W_c l_c^2(\frac{1}{2} + \frac{l_b}{l_c} - \frac{a}{4})}{1 + a\left(1 + \frac{4l_a I_b}{3l_b I_a}\right)}$$

$$= \frac{(33.1)(11.61^2 - 1.14^2)(\frac{1}{2} + \frac{.35}{8}) + (99.5)\cdot(13.63)^2(\frac{1}{2} + \frac{11.61}{13.63} - \frac{.35}{4})}{1 + (.35)\left(1 + \frac{(4)\cdot(3.77)((160{,}000)/(12)^4)}{(3)\cdot(11.61)((27{,}000)/(12)^4)}\right)}$$

$$= 11{,}460 \text{ KIPS-FT}$$

Bearing Reaction Forces:

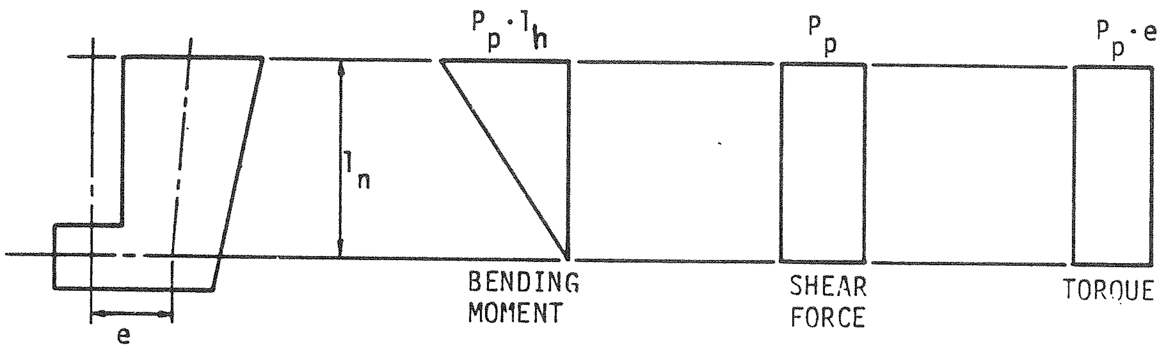
$$P_u = \frac{M_h}{l_a} = (11460)/(3.77) = 3040 \text{ KIPS}$$

$$P_h = P_u(1 + \frac{l_a}{l_b}) + \frac{W_b}{2l_b}(l_b - h_n)^2 - \frac{M_p}{l_b}$$

$$= 3040(1 + \frac{3.77}{11.61}) + \frac{33.1}{2(11.61)}(11.61 - 1.14)^2 - \frac{9242}{11.61} = 3390 \text{ KIPS}$$

$$P_p = N + P_u - P_h = 1702 + 3040 - 3390 = 1352 \text{ KIPS}$$

Control Surface Horn:



Shear Force = P_p = 1352 KIPS

Max. Bending Moment = $P_p \cdot l_h$ = (1352)(9.91) = 13,398 K-FT

Torque = $P_p \cdot e$ = (1352)(3.55) = 4800 KIPS-FT

b. Chordwise Loading: Follow same procedure as in Example I

(c) <u>DETERMINATION OF WEB SPACING AND WEB AND SIDE PLATING THICKNESSES</u>:

Follow same procedure as in Example I.

(d) <u>DETERMINATION OF STOCK SIZE</u>:

Follow same procedure as in Example I.

(e) <u>DETERMINATION OF HUB CASTING</u>:

Follow same procedure as in Example I.

(f) <u>SPANWISE AND CHORDWISE STRESS</u>:

Follow same procedure as in Example I.

(g) <u>GUDGEONS, PINTLES AND KEYWAYS</u>:

1. Gudgeons:

R_1 = 27.125 IN
R_2 = 15.6875 IN
$a = (R_1 + R_2)/20 = 2.14$ IN
l = 3' - 3" = 39 IN
T = 1352 KIPS
σ_c = 3.73 KIPS/SQ. UNIT (For $R_1/R_2 = 1.73$)

$$\sigma_G = \frac{\sigma_c \cdot T}{10a \cdot l} = \frac{3.73 \times 1352}{10 \times 2.14 \times 39} = 6.04 \text{ KSI}$$

(h) DEFLECTION CALCULATION:

Find the stock deflection 6 inches below the lower bearing. Refer to section 562-2-f-4 for nomenclature.

Step 1) At B (see diagram in paragraph (b)), the bending moment is 11460 kip ft, or 1.37E8 inch pounds.

M = (x / 45.24) * 1.37E8 in lbs (x < B)

Between B and X, the shear is $P_h - P_{y_s}$ = 3390 - 3040 = 350 kips, or 3.50E5 pounds. So the moment from B to Xs is

M = 1.375E8 - (3.50E5) (x - B) in lbs (x > B).

Since the moment is linear in this case, only the points A, B, and X_s values need to be tabulated.

	x	M
A	0.0	0.0
B	45.24	1.375E8
X_s	51.24	1.354E8

Step 2) the moment of inertia is assumed constant in this example.

EI = 29E6 psi * 27,000 in^4 = 7.83E11 lb in^2

Step 3) See M/(EI) Diagram below.

	x	M/(EI)
A	0.0	0.0
B	45.24	1.756E-4
X_s	51.24	1.729E-4

$$V_x = t_{xB} + [(X_s - B)/(B - A)] * t_{AB} .$$

t_{xB} = (area between Xs and B) * (distance from Xs to the centroid of that area)

area = (1.729E-4)(6) + (0.5)(1.756E-4 - 1.729E-4)(6)
= 1.037E-3 + 8.100E-6
= 1.045E-3

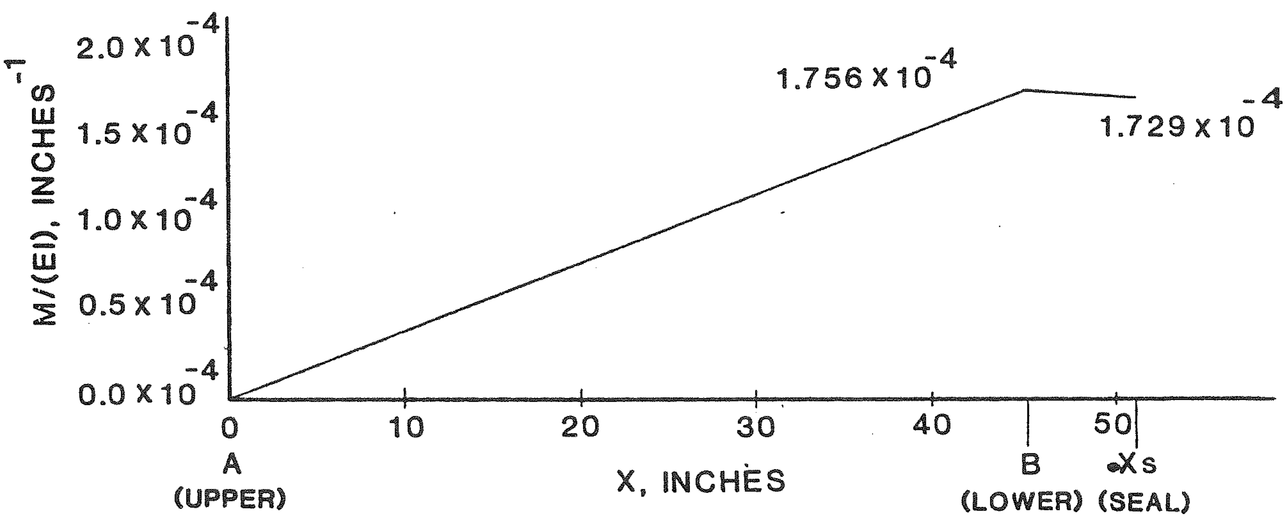
distance = (1.037E-3)(3)/(1.045E-3) + (4)(8.100E-6)/(1.045E-3) = 3.008 inches

so T_{xB} = (1.045E-3)(3.008) = 3.143E-3 inches

(Xs-B)/(B-A) = 6/45.24 = 0.1326

t_{AB} = area * distance to centroid
= (0.5) (1.756E-4)(45.24) *(2/3)(45.24)
= 0.1198 inches

M/(EI) DIAGRAM



M/(EI), INCHES^{-1}
2.0 X 10^{-4}
1.5 X 10^{-4}
1.0 X 10^{-4}
0.5 X 10^{-4}
0.0 X 10^{-4}
1.756 X 10^{-4}
1.729 X 10^{-4}
0
10
20
30
40
50
A
(UPPER)
X, INCHES
B
(LOWER)
Xs
(SEAL)

So, the deflection at the seal = V_x = 3.143E-3 + (0.1326)(0.1198)

= <u>0.019 inches</u>

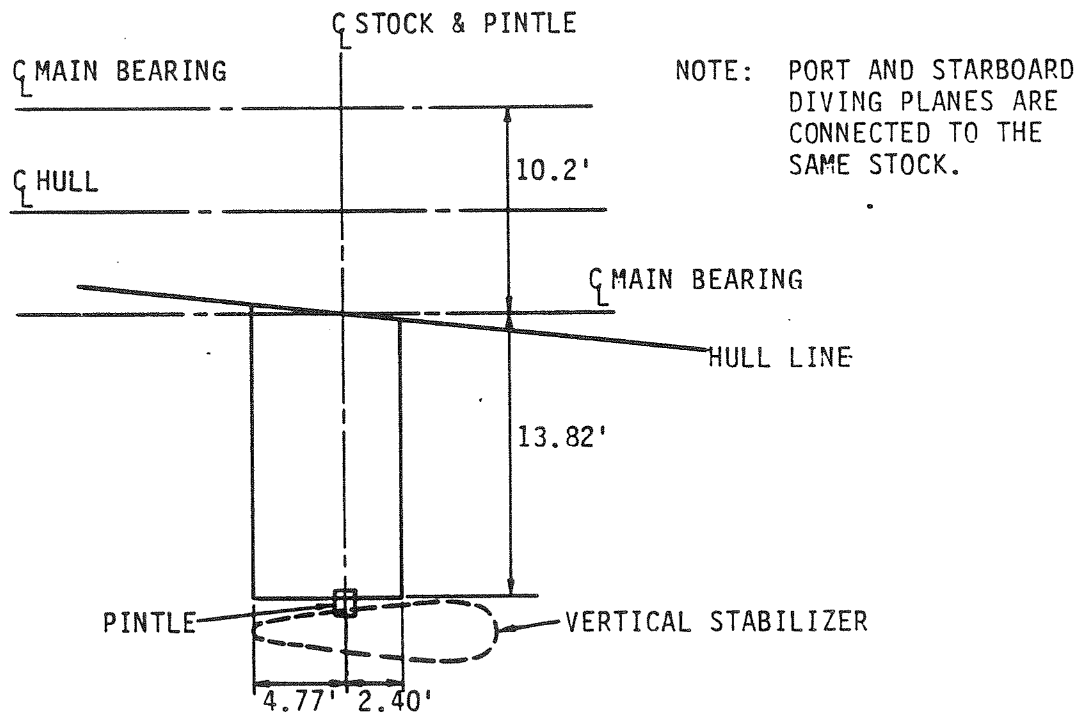
Note that if the moment was nonlinear and/or the moment of inertia were not constant, the M/(EI) curve could be determined at a series of x locations, and the tangential deviations could be evaluated numerically using a trapezoidal integration method.

EXAMPLE III

(a) INPUT: The Ship Specification gives the following information:

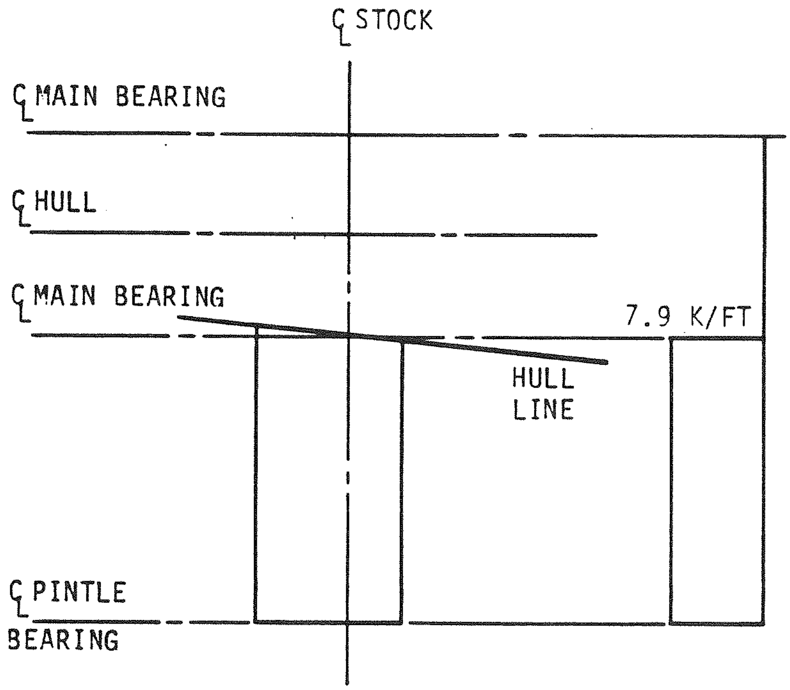
Ship: Submarine

Control Surface: Balanced Diving Plane - Filled



Torque: Max. Torque (Hydrodynamic) = 938 IN-KIPS
Max. Torque (Sea Slap) = 1,720 IN-KIPS

Loadings: Spanwise and Chordwise Sea Slap Loads. Hydrodynamic Loads and Ice Breakthrough Loads (only sea slap loads will be considered in this example however).



Note: Chordwise Load Distribution is given in the specification but is not considered in this example since analysis would be similar to Example I.

Load Combinations: For Overall Stress Sea Slap only.

Note: Combinations of other loads given under "loadings" will not be considered in this example but should be in an actual analysis.

Material: Diving Plane - HY 80
Stock - HY 80

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(b). <u>DETERMINATION OF BENDING MOMENT AND SHEAR DIAGRAMS, AND BEARING REACTIONS:</u>

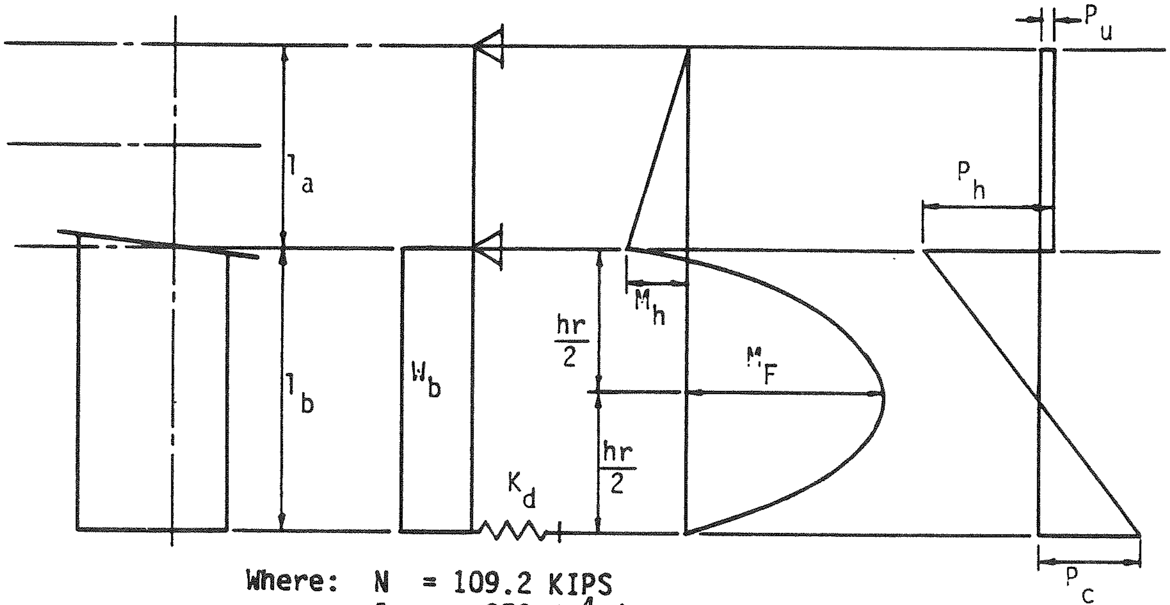
A. <u>Combining Loads for Spanwise and Chordwise Shear and Bending Moment Diagrams:</u>

Sea Slap Forces only

B. <u>Shear and Bending Moment Diagram and Bearing Reactions:</u>

(Balanced Type Control Surface)

a. Spanwise Loading



Where: $N = 109.2$ KIPS
$I_a = 850\ IN^4$ (Estimated)
$I_b = 1{,}600\ IN^4$ (Estimated)
I_d = Very Large (Estimated)

$l_d \simeq 0$

$h_n \simeq 0$

$h_d \simeq 0$

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Bending Moments:

$$a = \frac{l_b^3 I_d}{l_d^3 I_b} \approx \infty$$

$$M_h = \frac{W_b(l_b^2 - h_n^2)(\frac{1}{2} + \frac{a}{8})}{1 + a(1 + \frac{l_a I_b}{l_b I_a})} = \frac{\frac{1}{8}W_b(l_b^2 - h_n^2)}{1 + \frac{l_a I_b}{l_b I_a}}$$

$$= \frac{(\frac{1}{8})\cdot(7.9)((13.82)^2 - 0)}{1 + \frac{(10.2)((1{,}600)/(12)^4)}{(13.82)((850)/(12)^4)}} = 78.9 \text{ K-FT}$$

$$M_F = \frac{W_b h_r^2}{8} = \frac{(7.9)(13.82)^2}{8} = 189 \text{ K-FT}$$

Bearing Reaction Forces:

$$P_u = \frac{M_h}{l_a} = \frac{78.9}{10.2} = 7.7 \text{ KIPS}$$

$$P_h = P_u(1 + \frac{l_a}{l_b}) + \frac{N}{l_b}(\frac{h_r}{2} + h_d) = (5.7)(1 + \frac{10.2}{13.82}) + (\frac{109.2}{13.82})(\frac{13.82}{2})$$

$$= 64.5 \text{ KIPS}$$

$$P_c = N + P_u - P_h = 109.2 + 7.7 - 64.5 = 52.4 \text{ KIPS}$$

The calculation has been performed for one stern plane. Because both stern planes are attached to the same stock, the shears and moments from each are to be superposed in way of the stock between bearings.

b. Balanced Type Control Surface - Chordwise Loading:

Follow same procedure as in Example I.

(c) DETERMINATION OF WEB SPACING AND WEB AND SIDE PLATING THCKNESSES:

(Filled Control Surface):

a. Panel Breadth to Thickness Ratio - b/t_T:

Chose: $b/t = 72$

b. Plating Thickness - t:

$t = t' + t_1 + t_2$
$t' = .19$ IN
$t_1 = .06$ IN
$t_2 = 0$
$t = .25$ IN

c. Panel Breadth and Panel Length

$b = (72)(.25) = 18$ IN
Choose: $a=b=18$ IN

(d) DETERMINATION OF STOCK SIZE:

Follow same procedure as in Example I.

(e) DETERMINATION OF HUB CASTING:

Follow same procedure as in Example I.

(f) SPANWISE AND CHORDWISE STRESS:

Follow same procedure as in Example I.

(g) PINTLES:

$P_L = 50.4$ KIPS (shear force at pintle)
Sectional area of the pintle = 23.5 IN^2

$$\text{Shear stress} = \frac{52.4}{23.5} = 2.23 \text{ KSI}$$

Yield stress = 24 KSI

$$\text{Factor of Safety} = \frac{24}{2.23} = 10.762$$

562-2-p. Metric conversion

Table 5 provides metric conversion factors.

Table 5. 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