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by

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(1953)

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STRUCTURE OF LONGITUDINALLY FRAMED TANKERS

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ABSTRACT

In recent years there has been a marked increase of interest and activity in the "super" tanker field. These ships are usually longitudinally framed with lengths approaching 1000 feet. Until late 1958, the Rules of the American Bureau of Shipping covered ships of this type to a maximum length of 550 feet only. The new Rules, covering ship lengths to 1000 feet, are presented in tables and rudimentary formulas which are quite uninformative as to principles and criteria employed. There is no allowance made for advances in metallurgy or novel design.

This thesis analyzes the ABS Rules for sizing scantlings of the midship section in the particulars of section modulus, deck plating, side shell, bottom plating and bottom longitudinals. The object is to discern the principles, criteria and assumptions upon which the Rule requirements are based, especially for ship lengths only recently covered by the Rules.

The general attack of the problem employed background investigation, comparison with information on transverse framing, comparison with requirements of other Classification Societies and development from basic theories. The Rule requirements were reproduced by engineering formulas with criteria and assumptions defined as it was considered most probable the ABS had fixed them.

The most definitive basis for the present ABS Rules analyzed is the work and recommendations of the 1913 Load Line Committee which considered transversely framed ships of 600 feet maximum length. In general, we found the Rules for longitudinally framed tankers to be based on the experience gained from transversely framed ships with a margin of safety added to cover unknowns.

We find that the ABS employs the theory of transverse framing when dealing with longitudinally framed tankers. In numerous cases, the theory no longer applies when the framing pattern changes, and some instances show discrepancies which are of major proportions.

It is our recommendation that further investigation be made using theory applicable to longitudinal framing. Comparison with requirements of other Classification Societies should then be an aid to fixing constants and necessary experience factors on a rational basis.

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Title: Associate Professor of Naval Architecture

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NOTATION

Symbol	Meaning
a	Constant
b	Constant
b_2	Unsupported plating width, in.
c	1. Constant for sizing beams and stiffeners whose magnitude is given in ABS Rules for the case considered 2. Distance from neutral axis to extreme fiber, ft.
c_b	Distance from neutral axis to base line, ft.
c_t	Distance from neutral axis to molded deck line at side amidships, ft.
d	Draft of ship, ft.
e	Constant
f	Tabulated numbers used in figuring required section modulus
f_{tp}	For plating section modulus above neutral axis
f_{bp}	For plating section modulus below neutral axis
f_{tl}	For total section modulus above neutral axis
f_{bl}	For total section modulus below neutral axis
g	Constant
h	Head of water causing hydrostatic loading, ft.
k	Constant
k'	Constant
k_1	Constant
l	Longitudinal distance between supports of longitudinal girders, ft.
m	Correction factor for section modulus requirements

Symbol	Meaning
p	Stress, compressive or tensile, psi
p_1	Ship bending stress, psi
p_2	Girder bending stress, psi
p_3	Plate bending stress, psi
p_{cr}	Critical direct stress, psi
p_y	Yield strength in tension or compression, psi
s	Frame spacing, in.
t	Plating thickness, in.
t_c	Plating thickness assumed to corrode, in.
v	Shear stress, psi.
w	Faying flange width, in.
y	Distance of fiber under consideration from neutral axis, ft.
B	Beam of ship, ft.
D	Depth of ship to strength deck, ft.
E	Young's modulus, psi
I	Moment of inertia of structural cross section, sq. in. x sq. ft.
K	Constant
K_L	Nondimensional coefficient for bending stress in plates under lateral loading
K'	Constant
L	Length of ship (between perpendiculars), ft.
M	Bending Moment, ft-ton or ft-lb
N	Descriptive number used in entering ABS tables to size longitudinals

Symbol	Meaning
Q	First moment of cross-sectional area about neutral axis, sq. in. x ft.
V	Shearing force, ton or lb.
Z	Section modulus, sq. in. x ft.
$\propto$	Indicates proportionality
ρ	Density, lb. per cu. ft.

I. INTRODUCTION

Once the general characteristics, principal dimensions and coefficients of form of a ship design have been established, structural design becomes a logical succeeding consideration. Standard practice dictates that design of a midship section be worked out as a basic and initial structural problem. Once this midship section has been specified, the major part of the hull design will follow logically from the pattern thus established. Since the bending moments and shear loads are greatest between the quarter points of the hull, the scantlings toward each end of the ship need only be given as modifications to the midship section. In ships of usual form, the midship section design is controlling enough to be commonly used in making estimates of hull weight and for purposes of bidding.

In the field of structural design, a ship's hull is rather singular in complexity. To date, science and engineering give an incomplete description of loads and supporting forces acting on the hull. Even if these were determined, we would still be confronted by certain restrictions in our ability to assess the effects of these complex and time varying forces. Our present answer lies in experience with other ships. Classification societies possess a wealth of knowledge, accumulated over the years, to apply when performing their technical services. Ships are classified according to risk involved to insurance organizations that may underwrite policies on the ship and/or its' cargo. The Societies have reduced this experience, in general, to tables and simple, but often

[illegible]

deceiving, formulas for determining scantlings of the midship section. The design of a ship can thus be simplified and hastened since adherence to the Rules gives assurance of conservatism sufficient to avoid known weaknesses of previous ships and attainment of favorable classification for insurance premiums. On the other hand, the "normal" dimensions and coefficients listed by the Societies may be very restrictive on the designer who might wish to depart from them. Since the standards of the Societies are largely based on experience, they may impose an extra measure of conservatism on any design which departs from their known empiricism. This will certainly penalize any new design.

The various Classification Societies had such rules in effect in the late nineteenth century. In 1916 the British Corporation Rules were extensively revised to give them greater flexibility in allowing for departures from the more standard hull forms of the day. In 1947 both Lloyd's Register of Shipping and the American Bureau of Shipping revised their Rules to bring them more into line with the ship design and building practices of the time. We see a very sharp break with the past in 1953 when Det Norske Veritas brought out a new set of Rules based on simple formulas derived from first principles and tempered by experience factors. These were partially explained by Vedeler, the man largely responsible, in a paper (10) in which he invited criticism of the methods used. To reveal the derivation of rules and the standards used was a definite innovation, but to openly invite criticism of them was unheard of.

The present work is primarily directed to longitudinally framed tankers built under the Rules of the American Bureau of Shipping. Therefore it may be assumed that further discussion is confined to this area unless specifically stated otherwise. Until publication of the Tentative Rules in 1958 (2) the maximum length considered for tankers was 550 feet. (7) This was most unfortunate since tankers being designed and built were of the order of 900 feet in length. Since the Tentative Rules cover lengths to 1000 feet and make provisions for variations of overall dimensions and coefficients, it is quite natural for us to be curious as to what principles were used in extending the Rules to cover lengths nearly double those previously considered.

The Tentative Rules were revised in 1959 (3) and now contain tables of plating size and frame spacing with ship length as the entering argument. There is also a table of "f" numbers varying with length. These f numbers are to be used in the formula $Z = fB\ (d + 0.055L)$ to determine the minimum section modulus of the midship section. The basis of these f numbers goes back to a 1913 Load Line Committee (1) appointed by the British Board of Trade. These same basic f numbers have been adopted internationally and are contained in our Load Line Regulations (11) for vessels from 100 to 600 feet in length.

These f numbers, when used with various other sources of information available, yield a great deal of information as to the basic philosophy and methods used in extending the Rules to ships of greater lengths and varying forms. Accordingly, primary emphasis

is placed on them and the information they yield. Other structural members considered are the strength deck, side shell, bottom plating and longitudinals.

Having analyzed the existing American Bureau of Shipping Rules, we will attempt to offer refinements and improvements which may be suggested by first principles and formulations based on them. However, we fully realize that we are not able to draw on the broad experience available to the ABS, and that any actual modification to the Rules would have to insure that no known deficient practice would be allowed or even approached beyond a reasonable factor of safety.

One other serious restriction on the applicability of the ABS Rules should be born in mind throughout this work. Although required properties of the mild steel assumed in the construction of merchant ships are spelled out in the Rules, no allowance is made for advances in metallurgy and more progressive designs such as aluminum deckhouses. By returning to first principles and expanding them to the level of proven ships, it is hoped that we might contribute to the knowledge necessary to extend experience and aid in the design of newer and better ships.

II. PROCEDURE

A. Section Modulus Analysis

The real backbone of the longitudinal strength requirements of the American Bureau of Shipping is in its table of f numbers. These numbers are controlling in requirements for section modulus which is obtained by dividing moment of inertia by distance from the neutral axis to the molded deck at side amidships or to the base line as the case may be. The f numbers were published in the November 1958 "Tentative Rules For The Construction of Tankers" (2) for the general formula

$$Z = f\,d\,B \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (1)$$

In the October 1959 (3) version of these rules the f numbers were altered to fit a different formula and thus showed an influence of draft on section modulus that was less than heretofore specified. In each case there are four different f numbers. The definitions given in the 1959 Rules follow, with the units in inches squared-feet. One will find similar definitions in reference (2) with (d + .055L) replaced by d.

Net plating section modulus above neutral axis = $f_{tp}B\,(d + .055L)$ (2)

" " " " below " " = $f_{bp}B\,(d + .055L)$ (3)

Total section modulus above neutral axis = $f_{tl}B\,(d + .055L)$ (4)

" " " below " " = $f_{bl}B\,(d + .055L)$ (5)

These values are tabulated as functions of length, L.

The use of f numbers goes back to the work of the Load Line Committee of 1913 which was appointed by the British Board of Trade. In the 1916 Transactions of the Institution of Naval Architects,

W. S. Abell presented a paper which summarized the work of this committee (1). The f numbers found by this committee were used in the formula

$$Z = \frac{I}{y} = fdB \quad \ldots\ldots\ldots\ldots\ldots \quad (6)$$

which is the form used in the 1958 A. B. S. "Tentative Rules." (2) This original f number derivation will be examined in more detail later.

By analyzing the f numbers given by the A.B.S. we can discover a great deal about its criteria of longitudinal strength. Let us first consider the values as tabulated in reference (3). Examination of the formulas shows that if we desire to use a minimum of material to achieve the minimum required values of section modulus we are constrained to locate the material and thus the neutral axis in particular positions, provided we also meet the minimum plating thickness requirements. If we take "c" as the distance from the neutral axis to the respective extreme fiber we can derive the following relationship.

$$\frac{f_t}{f_b} = \frac{c_b}{c_t} \quad \ldots\ldots\ldots\ldots\ldots \quad (7)$$

The subscripts t and b refer to top and bottom of the hull girder respectively. By taking the ratios of the tabulated f numbers we thus find that we actually find the ratio of the deck and bottom distances from the neutral axis. It is then a simple matter to find the assumed normal position of the neutral axis. Calculations show that, for a given ship length, the position of the neutral axis will be the same whether we are speaking of the plating alone or of the total section modulus. Figure I shows the results

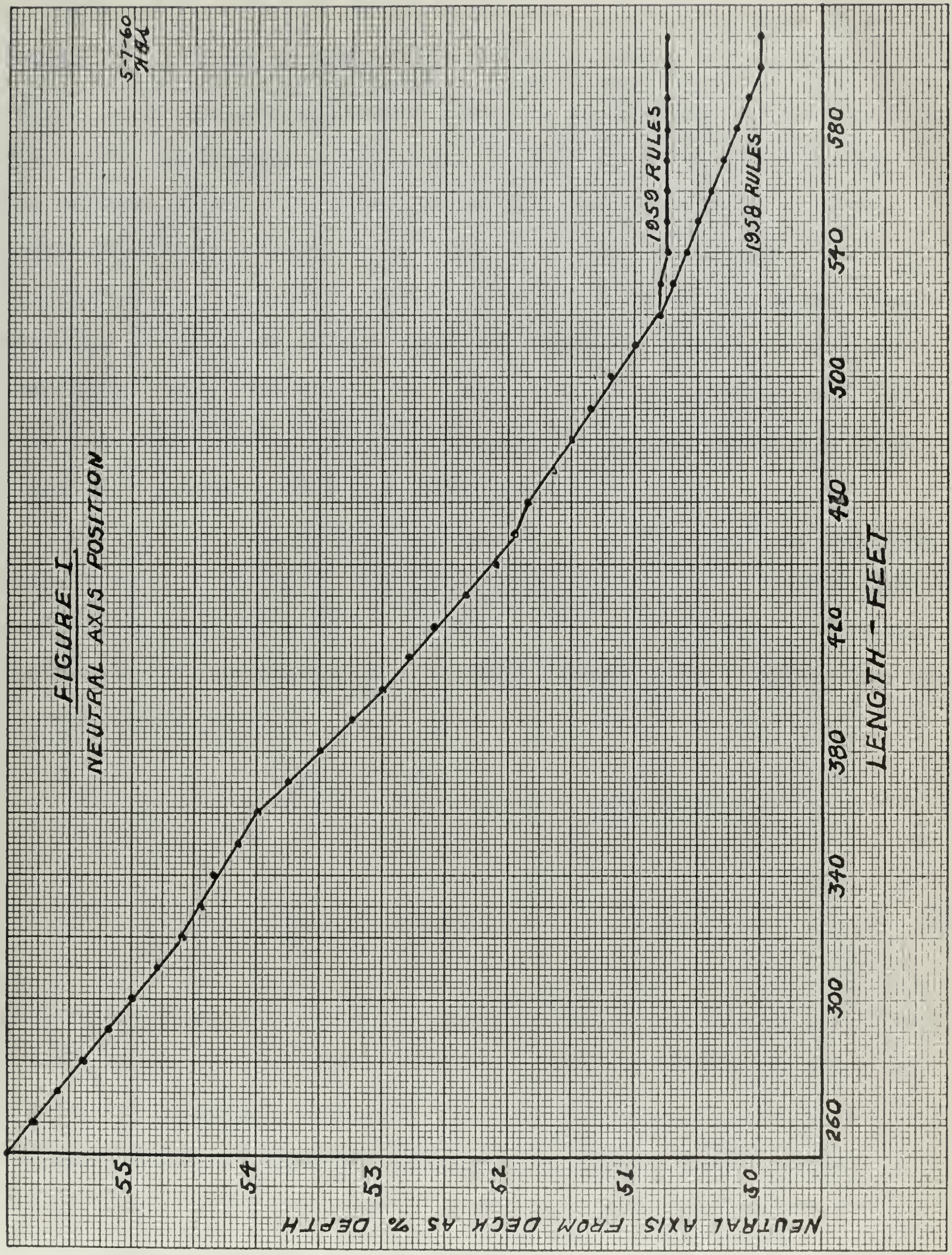


FIGURE I
NEUTRAL AXIS POSITION
5-7-60
1959 RULES
1958 RULES
LENGTH - FEET
260
300
340
380
420
460
500
540
580
NEUTRAL AXIS FROM DECK AS % DEPTH
55
54
53
52
51
50

of these calculations. We see here that the neutral axis at a length of 250 feet is 56% of the depth below the molded deck at side. This distance decreases linearly to 50.8% at a length of 520 feet. At lengths of 540 feet to 1000 feet the position of the neutral axis is constant at 50.75%. This break in a curve composed of two straight line segments will show up again in later analysis. Also included on Figure I is a plot of the neutral axis position as calculated from the 1958 "Tentative Rules." One will immediately see that the A.B.S. has not altered neutral axis position for short ships although the axis has been lowered for long ships from middepth to the aforementioned 50.75%.

Next we will examine the effect of the f values in determining the division of required material between plating and longitudinals. Again disposing the metal so as to use minimal amounts, and using the relation of plating and total section sectional modulus, we get further formulations.

$$\frac{f_{tp}}{f_{tl}} = \frac{f_{bp}}{f_{bl}} \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \qquad (8)$$

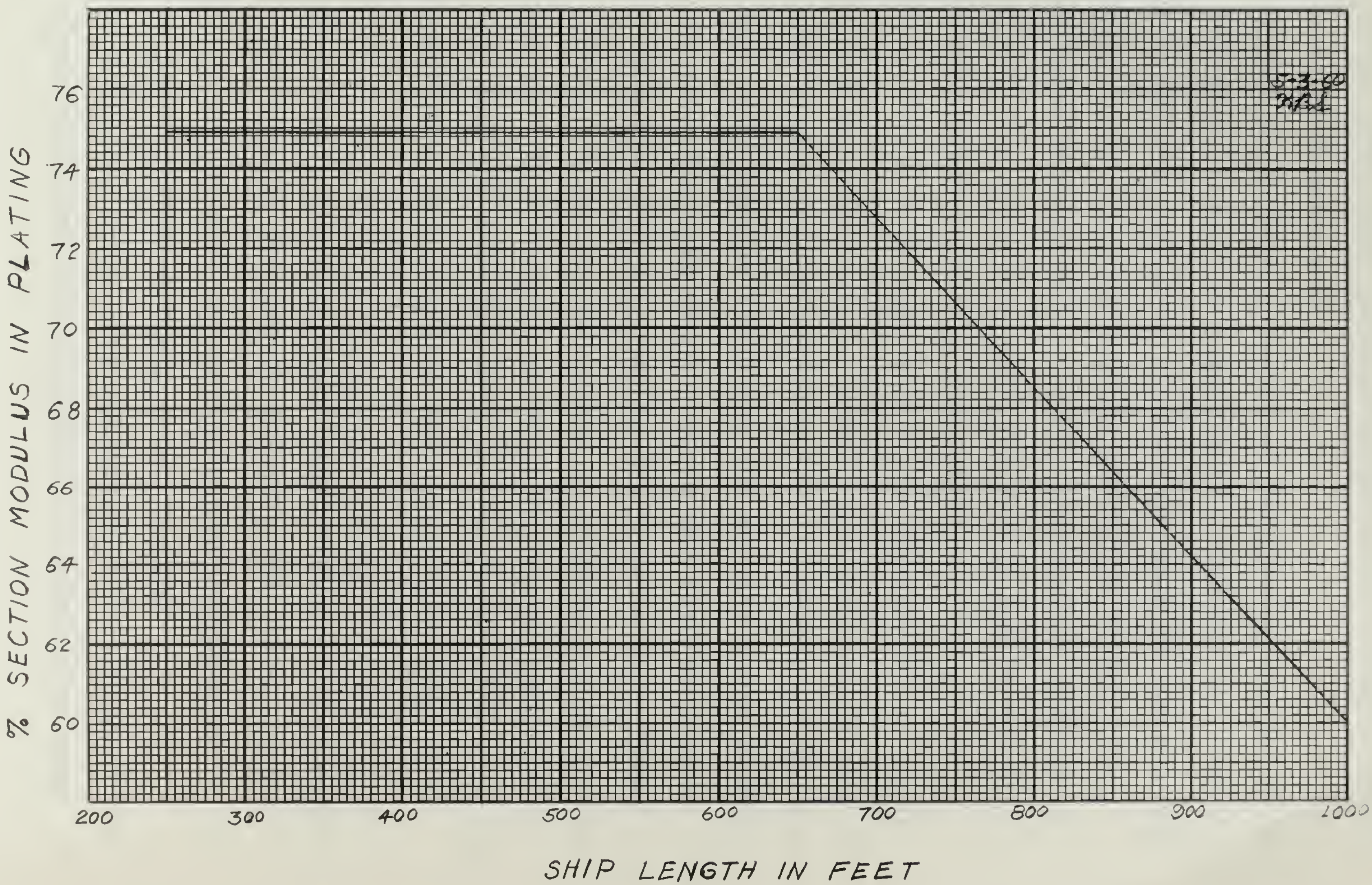
and

$$\frac{f_{tp}}{f_{tl}} = \frac{Z_{tp}}{Z_{tl}} \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \qquad (9)$$

Thus we see that the ratio of f numbers gives the fraction of total section modulus which must be in the plating. Figure II shows the results of these calculations. We note here that from 250 feet to 650 feet we have a constant 74.9% of the total section modulus in the plating. From 650 feet to 1000 feet the percentage drops

FIGURE II

% SECTION MODULUS IN PLATING - 1959 RULES



linearly to 60. Here again later analysis will show this break at 650 feet to be significant.

In 1955, Mr. D. P. Brown, Technical Manager of the American Bureau of Shipping, stated on page 111 of reference (6) that for large tankers and ore carriers, "It is customary to use plating for the shell, decks and longitudinal bulkheads so that when taking only these members into account, the section modulus to the deck is not less than 85%, and that to the bottom not less than 75% of the required section modulus."

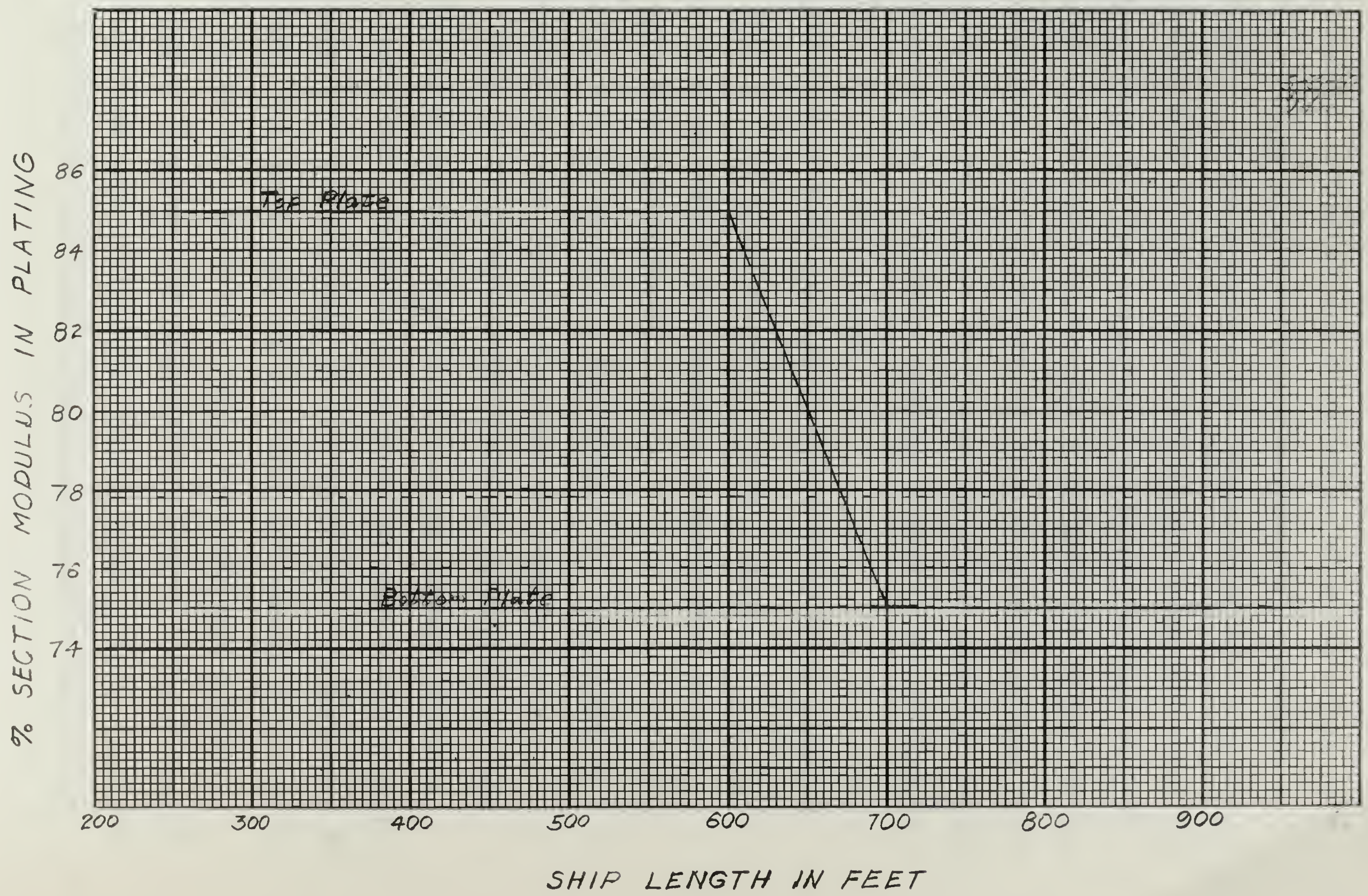
Since these views have apparently been changed or at least tempered by action of the A.B.S. Technical Committee, it will be informative to pursue the matter further. A similar analysis of the 1958 table of f numbers (2) gives the results shown in Figure III. Here we note that, in lengths up to 600 feet (the upper limit of the 1913 Committee (1) f numbers), Mr. Brown's percentages have prevailed. Beyond this the percentage in plating to the bottom is constant, but that in the top plating is decreased linearly between 600 feet and 700 feet from 85% to 75% to coincide with that in the bottom. Once again we find the curves generated by a straight edge.

Although analysis shows no significant change in the requirements for total section modulus in the past few years, it does show a progressive decrease in the percentage required to be in the plating, particularly at the greater lengths.

When the f numbers first were introduced to the shipbuilding industry by the 1913 Load Line Committee (1) various attempts

FIGURE III

% SECTION MODULUS IN PLATING - 1958 RULES



were made to dovetail theory with practice. Engineers, for one reason or another, resisted the philosophy of thinking in terms of sectional modulus and therefore endeavored to calculate allowable stress. If one examines the argument of Sir John Biles (1), it will be much more evident that learned men were reluctant to accept the section modulus criteria for midship design in lieu of the prevailing method of assuming a maximum allowable stress and bending moment. In an effort to find allowable stress from the experience wrapped up in the f numbers Tobin used the section modulus criteria and an assumed bending moment. The procedure used by Tobin (1) was as follows.

1. The f number data was approximated by a curve proportional to the five-thirds power of length.
2. The bending moment was assumed proportional to the length and displacement.
3. Making use of formula 6 the stress, p, was found to be equal to the one-third power of length.

Since the f numbers that were introduced in reference (3) are designed to fit a somewhat different section modulus formula, as has been previously described, we find it once again expedient to investigate the implications of new criteria. In contrast to Tobin's method, the procedure we use will not assume that all f numbers are based on experience. Only recently has interest arisen in "super" tankers and experience factors for the longer ships are therefore not available. Since longitudinal strength modulus is not known, and since extrapolation of f number data

would indeed be shaky, it can be assumed that f factors are derived by assuming bending moment and primary girder stress values. Telfer (5) states that the new Det Norske Veritas Rules account for still water and wave bending moments. Since Vivet (4) introduced a two term bending moment formula that looks suspiciously similar to the A.B.S. formula for section modulus, it seems reasonable to pursue in this direction.

$$M = aBdL^2 + bBL^3 \quad \ldots\ldots\ldots\ldots \tag{10}$$

This formula, which accounts for still water and wave bending moments respectively, can be altered to appear more compatible with formulas 2-5.

$$M = aBdL^2 \left(1 + \frac{b}{a}\frac{L}{d}\right)$$

$$Z = fBd \quad \left(1 + .055\,\frac{L}{d}\right)$$

It is reasonable to assume that the wave and still water moments are of equal magnitude at shorter lengths (5) with the wave moment gaining in importance as the ratio L/d increases. Since 0.055 is a reasonable value for d/L (14), and since Table A and Table 12 in the A.B.S. Rules give values of 0.05 and 0.06 respectively, it is reasonable to assume that 0.055 is a typical d/L value for ships of conventional form. To further substantiate the d/L value, f values as tabulated in reference (2) were compared with the values tabulated in reference (3). If we assume the section modulus required by short ships is the same in both cases a d/L value of .055 is mandatory. Considering the foregoing criteria, b/a is found to equal 0.055. The following relation for f therefore follows.

[illegible]

$$p = M/Z = \frac{aBdL^2\,(1 + .055L/d)}{fBd\,(1 + .055L/d)}$$

$$\therefore \quad f = \frac{aL^2}{p} \qquad \text{. . . . .} \qquad (11)$$

It is here that one is faced with choosing an expression for p, stress. Actually, unless the bending moment formula was known to be accurate for longitudinally framed tankers, serious concern about the formula for stress would not be justified. However, because we are attempting to determine the criteria used by the American Bureau of Shipping in formulating the f numbers, we will proceed although we have found no evidence of the two term bending moment formula's accuracy for longitudinally framed tankers of unusually long length. Because the deck receives the larger bending stress our analysis will be of the total section modulus above the neutral axis. We have already seen the arbitrary way the total modulus is divided between shapes and plates.

We will first let the stress, as determined by Tobin, be equal to $L^{.333}$ and obtain the following expression for f.

$$f = aL^{5/3} \qquad \text{. . . . .} \qquad (12)$$

To support Tobin's value for p it is noted that Mr. Brown (6) stated that $p = 1.19\ L^{1/3}$ for transversely framed ships with machinery amidships. He then stated that f should be increased about 15% if the machinery is located aft and a further increase is needed if the ship is longitudinally framed. These corrections will be applied to f if the bending moment is not altered and the stress is reduced fifteen to twenty per cent. In accordance

[illegible]

[illegible]

[illegible]

[illegible]

[illegible]

with Brown's method (6) "a" can be calculated as follows to satisfy formula 10. This formula assumes a block coefficient of 0.75 which Brown feels is sufficiently accurate due to the precision of the formula.

$$a = \frac{0.75 \times 0.50}{35 \times 35} \qquad a = 0.306 \times 10^{-3}$$

If one calculates the value of "a" by using data for f_{tp} as found in reference (3), at a length of 550 feet one will arrive at the same value.

$$a = \frac{f}{L^{5/3}} = \frac{11.31}{(550)^{5/3}} = 0.306 \times 10^{-3}$$

The fact that the values of "a" calculated are the same merely points out that the Rule stress at 550 feet will be identical with Tobin's. Figure IV shows a plot of f vs L. The f values are obtained from the Rules as f_{tp} and calculated using formula 12. Of primary interest is the fact that at short lengths the Rule values of f are larger than calculated and thus signify an allowable stress less than Tobin's; at lengths above 550 feet the Rule f values are less than calculated and this signifies an allowable stress greater than Tobin's. Figure V shows the variation of stress with length. Because it appears that variation of Rule f from calculated values may be linear with length, a plot, Figure VI, has been made to show this relationship. An examination of this plot clearly shows that the differences in f numbers is not linear. The most significant variation of f numbers, percentage wise, is at the shorter lengths. This would most seemingly be

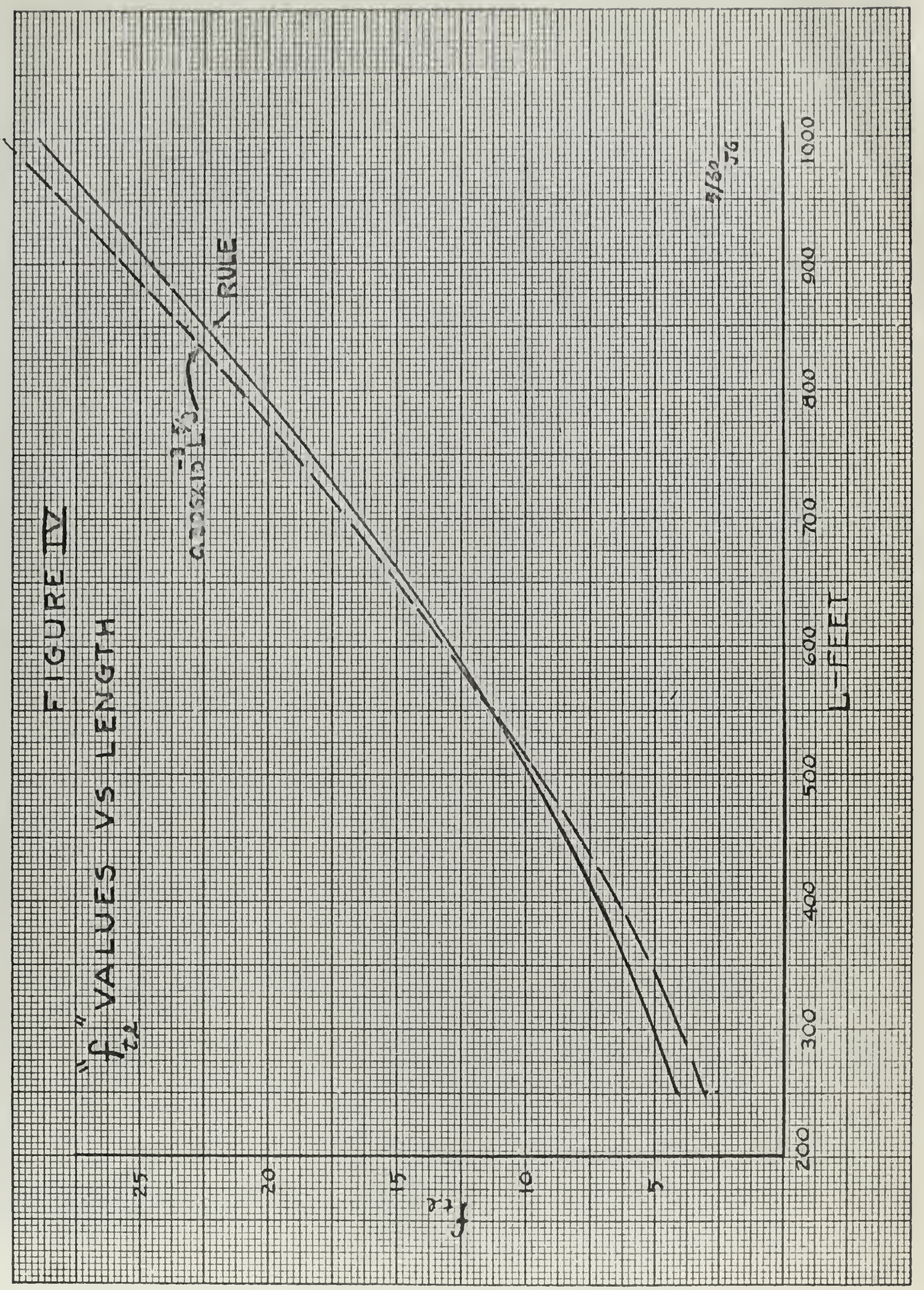


FIGURE IV
"f t2" VALUES VS LENGTH
RULE
L-FEET
200
300
400
500
600
700
800
900
1000
5
10
15
20
25

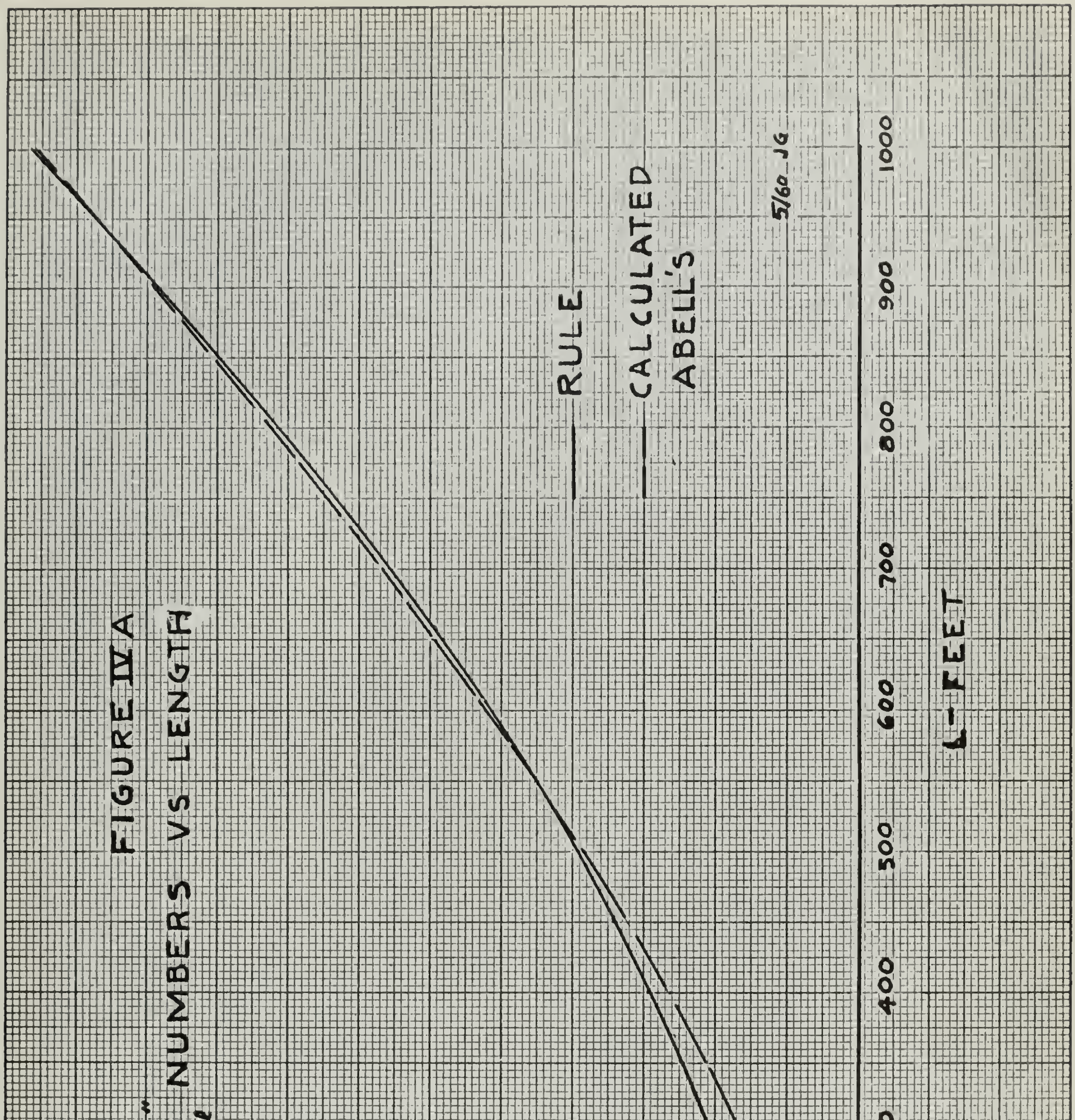


FIGURE IV A
"f_tl" NUMBERS VS LENGTH
RULE
CALCULATED ABELL'S
5/60 JG
L - FEET
f
25
20
15
10
5
200
300
400
500
600
700
800
900
1000

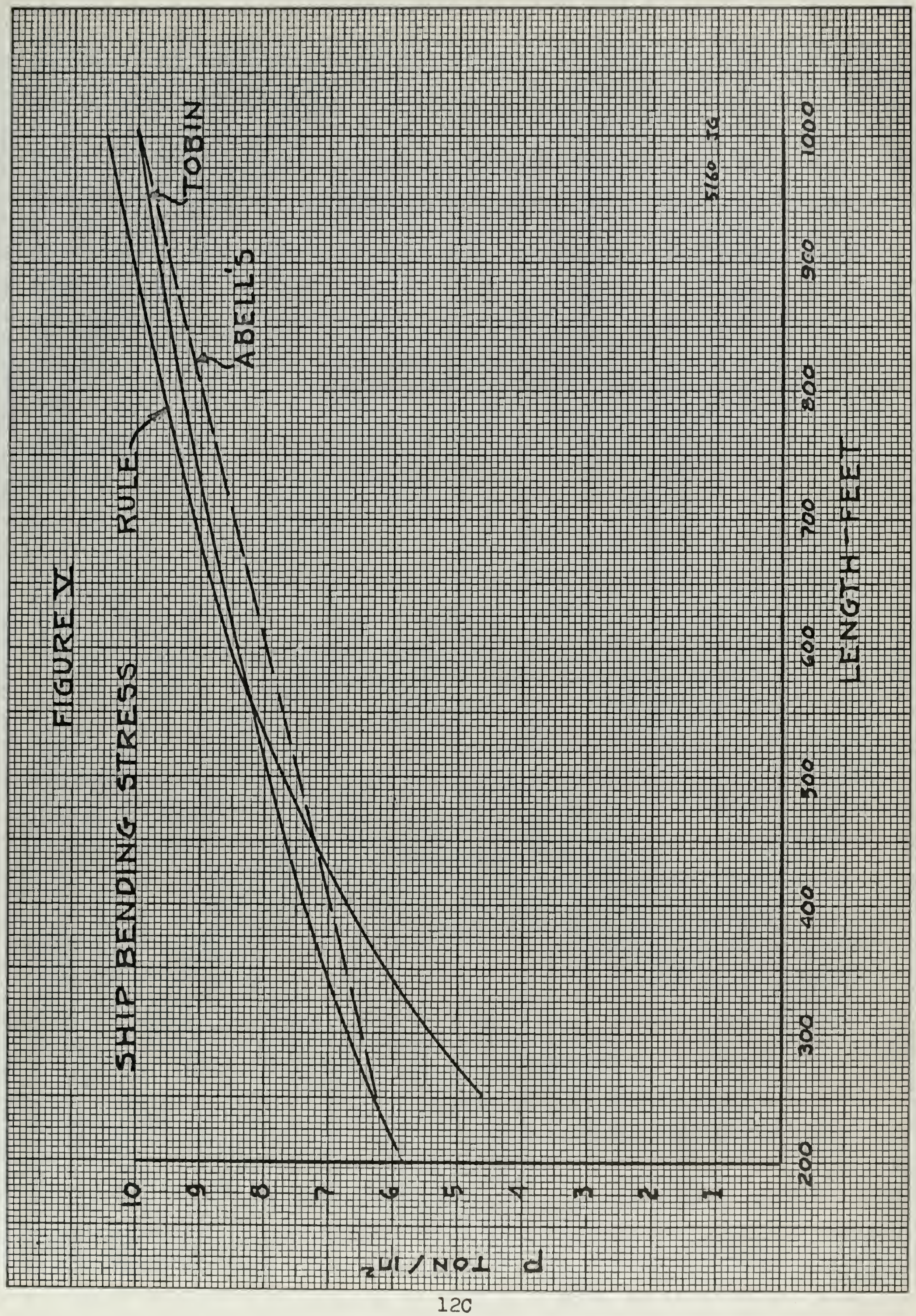


FIGURE V
SHIP BENDING STRESS
RULE
ABELL'S
TOBIN
5160 JG
p TON/IN²
10
9
8
7
6
5
4
3
2
1
200
300
400
500
600
700
800
900
1000
LENGTH-FEET

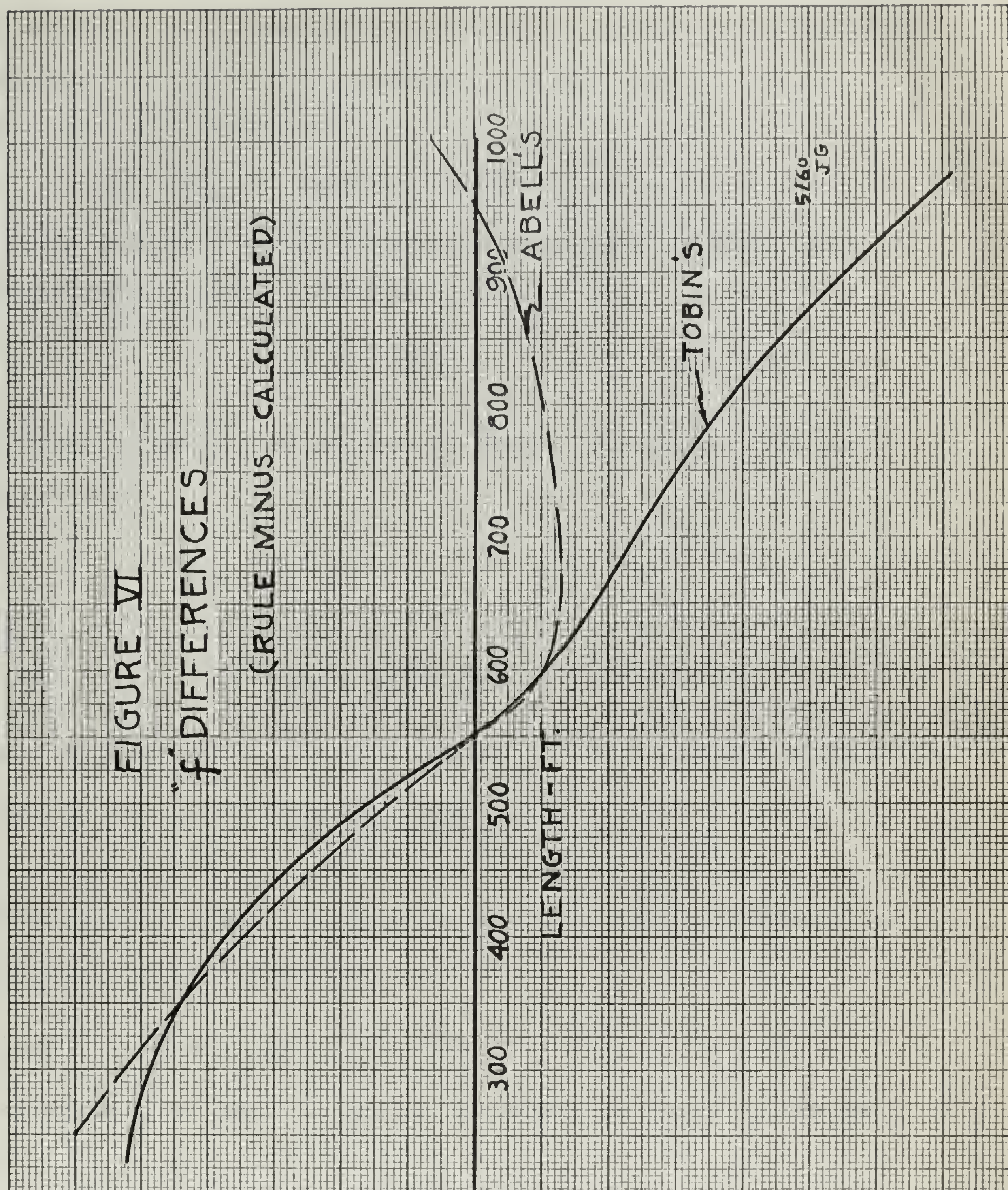


FIGURE VII
"f" DIFFERENCES
(RULE MINUS CALCULATED)
LENGTH - FT.
300
400
500
600
700
800
900
1000
1
.8
.6
.4
.2
-.2
-.4
-.6
-.8
-1
+Δ
ABELL'S
TOBIN'S
5160
JG

explained by a corrosion allowance that was constant, in thickness, with length. It is comparatively easy to check the influence of a constant corrosion thickness on the plate section modulus.

Since

$$Z_{tp} = f_{tp}B\ (d + .055L) \quad \ldots\ldots\ldots\ldots \quad (2)$$

and if we consider the ship girder rectangular in shape with two verticle longitudinal bulkheads

$$Z_c = tD\ (B + \frac{2D}{3}) \quad \ldots\ldots\ldots\ldots \quad (12)$$

where the subscript "c" is used to denote corrosion. We can also say that

$$f_{tp} = f + f_c$$

where

$$f = .306 \times 10^{-3}\ L^{5/3}.$$

If we now solve for f_c we will arrive at the following expression.

$$f_c = \frac{t\ D\ (B + 2D/3)}{B\ (d + .055L)} \quad \ldots\ldots\ldots\ldots \quad (13)$$

Without fear of being too greatly in error we can assume the following relations to hold.

$$L/D = 13 \qquad L/d = 1/.055 \qquad B = L/10 + 18$$

We therefore find that f_c can be expressed in terms of length only.

$$f_c = \frac{12\ t\ (.1512\ L + 18)}{1.43\ (.1000\ L + 18)} \quad \ldots\ldots\ldots \quad (14)$$

It should be noted that t in the above formula is a constant corrosion allowance. Formula 14 tells us that the corrosion allowance will cause an increase in f with length and an examination of Figure IV readily shows that the difference in f numbers decreases with length. Thus, we have shown rather conclusively, that the variation of f numbers from the five-thirds power of length cannot be due to a constant corrosion allowance only.

If we let

$$m = \frac{f_{tp}}{f} = 1 + \frac{f_c}{f}$$

and then plot "m" vs length we will have a plot as shown in Figure VII. It is here that we can mislead ourselves easily. If one solves for "m" at a length of 250 feet and then finds f_c at this length a curve of f_c/f can be plotted which agrees very well with the known "m" curve for lengths less than 550 feet. One could wrongly deduce that this correspondence is because of a constant wastage allowance. A bit of reflection by the reader will show, we feel sure, that this agreement is not because of the corrosion allowance. The overpowering of the 1/f term dominates and the relatively slight variation of f_c with length will have little adverse effect. To what then can we attribute the large f values at shorter lengths? The reasons, we feel, are as follows:

1. Short vessels take much more punishment than the long, large ships. Waves that are the length of the short ship are apt to be quite steep in comparison to longer waves.
2. Any seagoing man will tell us that short (small) ships suffer much higher accelerations than large ships.

[illegible]

[illegible]

[illegible]

[illegible]

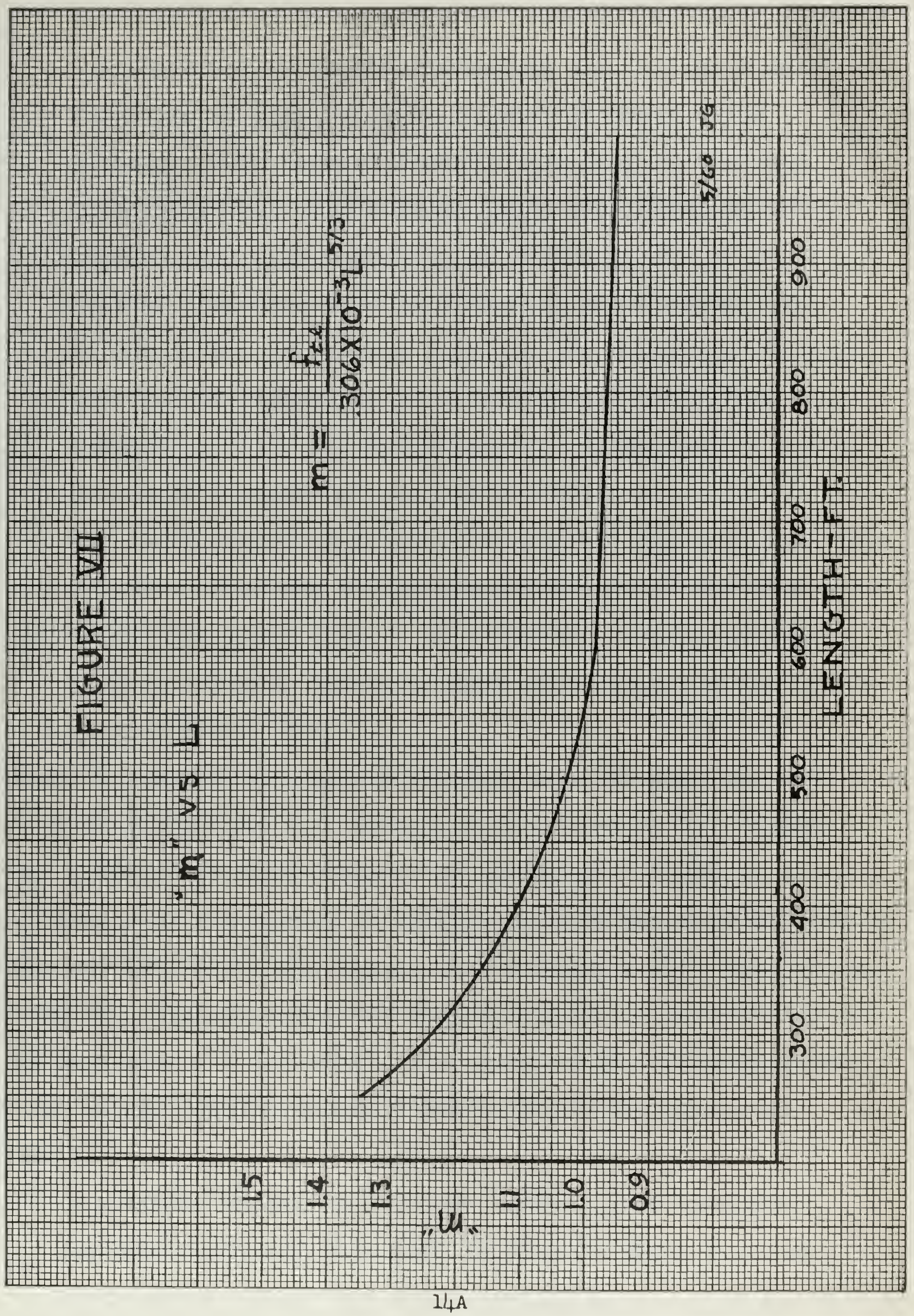


FIGURE VII
"m" vs L
$m = \frac{f_{tt}}{306 \times 10^{-3} L^{5/3}}$
"m"
1.5
1.4
1.3
1.1
1.0
0.9
300
400
500
600
700
800
900
LENGTH-FT.
5/60 JG

Stresses set up as a result of pitching and heaving are therefore apt to be much larger in the shorter ship. The force to mass ratio is greater in short ships.

3. Although the f numbers found do not vary in a manner indicating that corrosion is being considered we nevertheless feel that corrosion allowance is one of the reasons that stress is held down in short ships. It is not at all improbable that the two foregoing reasons decrease in importance at a rate sufficiently fast to overcome the theoretically larger f value that we would expect as a result of corrosion allowance.

4. The rapid reduction of allowable stress in ships less than 300 feet shows the relative unimportance of the bending moment for short ships. As a result, section modulus is an unsuitable criterion upon which to determine requisite strength (20).

Since f numbers for lengths greater than 550 feet are not based on experience but are either extrapolated or calculated, further analysis at these lengths is justified. Since it is elsewhere shown that the amount of section modulus assigned to plating is somewhat arbitrary (see Figure II), attention will here be confined to the total longitudinal section modulus. Immediately one has three avenues of approach. The decrease in modulus requirements at longer lengths can be considered for the following reasons:

1. A constant wastage thickness. Again, this constant wastage thickness will not, by itself, cause f values

to vary from those calculated in the manner indicated in Figure IV. However, a constant wastage thickness would result in an ever increasing allowable stress and therefore this criterion cannot be discounted.

2. The longitudinal bending moment will not increase with length at the long lengths as rapidly as at the short lengths since the length to height ratio of the wave will not be linear with length.
3. The longer and larger the ship the smaller are the dynamic forces due to accelerations. Another way of saying the same thing is that the force to mass ratio can be expected to decrease.

Although not mentioned previously it is known, statistically, that a short ship will encounter waves equal to its length more often than a long ship. Although there is disagreement in the field, we do not feel that this fact justifies a decrease in modulus requirements since a ship should be designed for a maximum situation regardless of occurrence frequency.

Since an analysis using Tobin's expression for longitudinal stress does not clearly give us an exact duplication of f numbers an attempt will be made, using the same procedure as used previously, to satisfy the data using Abell's formula for stress. A glance at Figure V shows strongly the wisdom of such a choice. Although allowable Rule stresses are larger than Abell's they are certainly parallel and differ at the longer lengths by a constant. One can easily deduce that

$$f = \frac{a\ L^2}{5(L/1000} + 1) \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \qquad (15)$$

When one now examines Figure IV (A), a plot of f as calculated using formula 15 and Rule values of f, the striking result is that the Rule values initially decrease relative to the calculated values and then, at about 750 feet in length, increase. It is obvious that this variation cannot be explained, in whole, by a constant corrosion allowance. However, it should be noted that the deviation at lengths above 550 feet are extremely small and surely it appears that the American Bureau of Shipping is allowing stresses for long ships to be more like Abell's than Tobin's. This does not seem to be wholly in agreement with Mr. D. F. MacNaught (14) who states that ships above 650 feet are designed to an allowable stress of 10.3 tons per square inch. However, it should be noted that MacNaught's criteria is conservative for lengths less than 1000 feet. Mr. Brown (6) also infers that a constant value for stress is accepted for ships of length greater than 650 feet when he states that section modulus for these long ships can be approximated by a formula proportional to L^2 b d. However, Mr. Brown may be considering the facts brought out by the two term bending moment formula (formula 10). It should be noted that in formula 10 the influence of draft on the total bending moment decreases in importance and the influence of L therefore increases. At longer lengths then, when the formula for bending moments used by Brown is most apt to be in error, he may have implicitly made the necessary allowance by assuming a stress constant with length. At lengths less than 550 feet the deviation of calculated f numbers from the Rule values is generally the same as when Tobin's criteria was used. A plot of the

calculations of f number deviation, using Tobin's and Abell's stress values, is given in Figure VI.

Before leaving formula 15, however, one should note that p is not proportional to L in the manner assumed by Abell (1). Should one consider p directly proportional to L, as did Abell when he neglected the constant, f would also be proportional to L and this would be in error.

Since the neutral axes of ships constructed using A.B.S. criteria are not at half depth one might suppose that an adjustment to calculated f numbers is necessary. However, one assumes a maximum stress in the deck plating and the f numbers calculated will, theoretically, give that stress when the assumed bending moment is applied. Had we focused our attention on bottom plating the maximum allowed girder stress, disregarding hydrostatic loads, would be reduced. Should the maximum bottom stresses not be reduced, greater stresses than maximum would occur in the deck and failure in the uppermost plating would result.

In an effort to justify the A.B.S. table of f numbers an attempt was made to fit the data with a partial series expansion of f as defined in formula 15. When one performs this expansion the form arrived at is

$$f = K\left(L - c + \frac{c^2}{L} - \frac{c^3}{L^2} + \frac{c^4}{L^3} \ldots .\right) \qquad (16)$$

It can be readily noted that a limited expansion of this sort is good only when c is much less than L and we found that this is not true. Excellent duplication of data was possible over limited ranges of L (i.e. 250 - 550 feet and 550 - 1000 feet) but it was

[illegible]

[illegible]

[illegible]

[illegible]

[illegible]

not possible to arrive at satisfactory results over the complete range of length. Although the wisdom of this part of the procedure can be seriously questioned, since the closed form for f (formula 15) will be always more accurate than the approximate form for f (formula 16), formula 16 does vividly show the effect of including the constant in formula 15 when calculating f. The importance of the terms in formula 16, to the right of L, decrease as the constant "c" decreases in value. The reader may recall that Abell (1) neglected the constant in some of his calculations.

An interesting sideline to the analysis of the f numbers is the expected change in the ratio of draft to length. As has been brought out earlier in this thesis, a d/L ratio of 0.055 is assumed for ships less than 520 feet in length. If one assumes that the section modulus requirements for ships built under the 1958 "Tentative Rules" are the same as requirements specified by the 1959 version of the same rules, and if one assumes that the expected draft is the same in both cases, one can solve for draft at the various lengths.

$$f_{1958} = f_{1959} (1 + .055 \, L/d)$$

The variation of d/L with L is shown in Figure VIII and of interest is the fact that at 1000 feet the draft, as calculated, is about 38 feet giving a d/L ratio of 0.038. We do not claim that the new f numbers were devised from the old using these drafts as criterion.

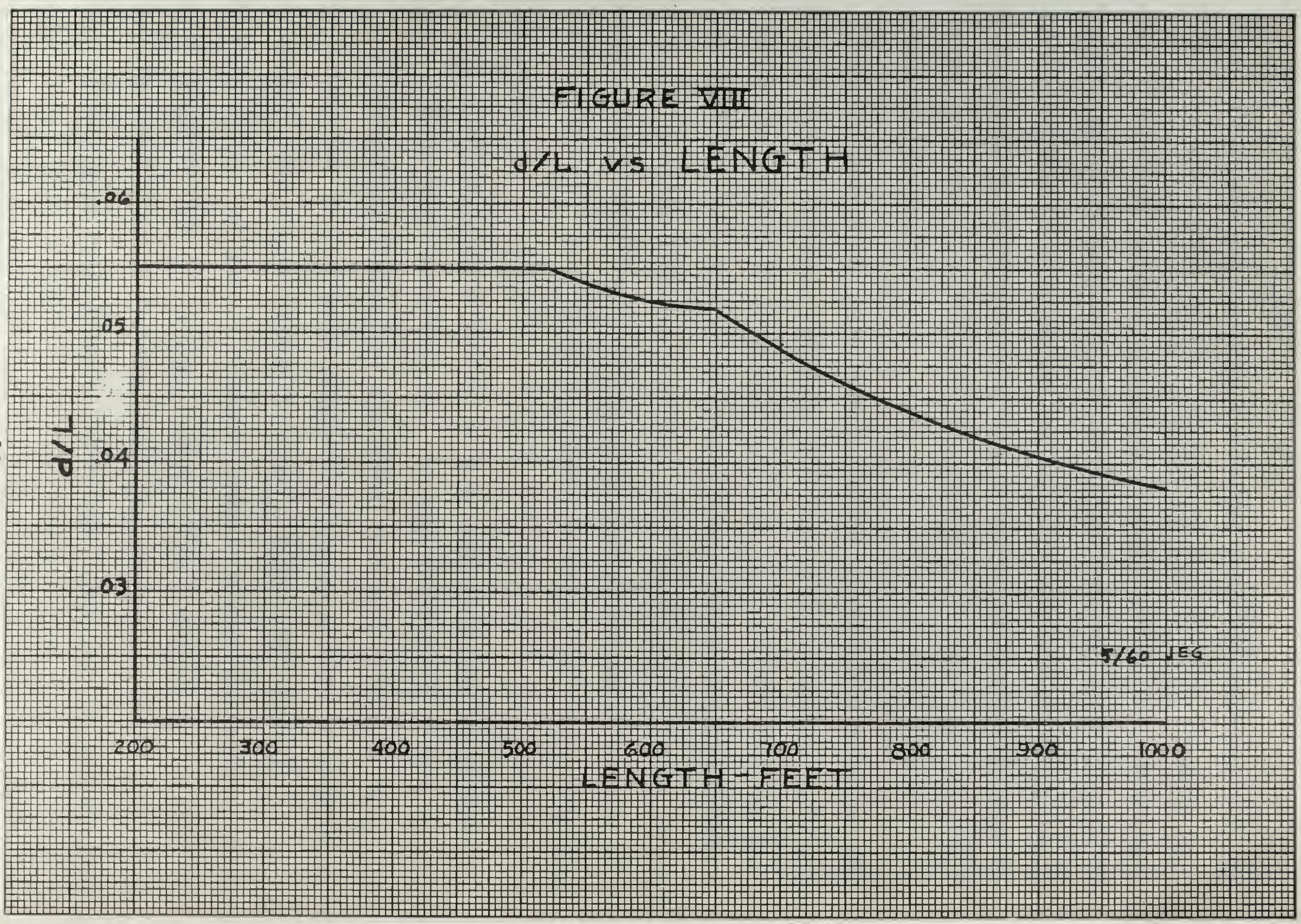


FIGURE VIII
d/L vs LENGTH
.06
.05
.04
.03
d/L
200
300
400
500
600
700
800
900
1000
LENGTH-FEET
5/60 JEG

B. Strength Deck Thickness Analysis

As described previously, a minimum plating modulus, top and bottom, is specified in the "Tentative Rules" (3). Listed in Table A of these Rules, according to ship length, are minimum strength deck thicknesses. In this analysis we will attempt to show the criteria adopted by the American Bureau of Shipping to specify plating requirements. An endeavor will also be made to show the relative importance of plating thickness and plating modulus criteria.

Logical analysis, previously presented in the foregoing section, indicates that section modulus, and therefore f number criterion, is based on primary ship girder stress, p_1. Since this is so, one might question the wisdom of specifying a total section modulus and a net plating section modulus, when seemingly a total section modulus requirement would adequately reduce stresses within allowable limits. We feel, that to question this dual requirement is legitimate.

The shipbuilding industry is old, and with age we generally expect wisdom and conservatism. To the aspirer, the various Rules of Classification Societies are often too restrictive. True, the new "Tentative Rules" were prepared to give greater flexibility to the design, but close examination will show that the architect is not forced, but is gently pushed, toward rather stereotype designs. "Super" tankers are not proven ships. Their machinery is aft, they are longitudinally framed and are normally 650 to 1000 feet long. Structural requirements have been acquired by Societies

[illegible]

[illegible]

[illegible]

[illegible]

from experience with transversely framed ships which almost always have machinery amidships. Thus, when a new type of construction, such as longitudinal, was initiated, the Societies did not know how effective longitudinal shape material would be. The builders, of course, argued for plating reductions, since longitudinals contribute to section modulus. However, the Classification Societies did not possess the same optimism. They knew that plating is effective since transversely framed ships have virtually no useful longitudinal material other than hull plating. Thus, we find that the 1958 "Tentative Rules" (2) require from 85 to 75 per cent of the section modulus in plating while the 1959 "Tentative Rules" require from 75 to 60 per cent. The trend is clear; as longitudinal girders prove effective we expect even greater reductions in plate modulus requirements.

Mr. Brown, of the A.B.S., states (6) that modulus requirements of longitudinally framed ships, where about 60 per cent of the total length of the ship is oil or ore cargo, is about 20 per cent greater than the standard modulus of a transversely framed ship. When Mr. Brown wrote his excellent article, usual procedure was to put from 15 to 25 per cent of the modulus in longitudinals. The end result is rather odd; even when longitudinals can be included as fully effective material, there is not an appreciable reduction in plating requirements. However, using the new Rules (3) modulus requirements are generally reduced. Only if one designs with a draft to length ratio less than plotted in Figure VIII will total, or plate, modulus requirements be greater than necessitated by the 1958 Rules. Since Table A was not

changed from that appearing in the 1958 Rules (2), it is not surprising to find that thicknesses are based on criteria only applicable to transverse framing.

Because a tanker experiences its largest moment sagging, buckling of the deck is a reasonable criterion. The deck may also take a hydrostatic load due to a head of water that varies, according to the Rules, from 4 to 8 feet depending on length. The stress, in plating, resulting from water pressure can be expressed by the formula

$$p_3 = .2225\ h \left(\frac{s}{t}\right)^2 \quad \ldots\ldots\ldots\ldots \quad (17)$$

This plate bending stress, p_3, is orthogonal to primary ships bending stress and occurs midway between web frames. A stress parallel to primary bending stress, occurring midway between longitudinals, is 0.685 times as large. These hydrostatic stresses cannot cause buckling. Two expressions that express critical buckling stress, in terms of frame spacing and plate thickness, follow:

$$p_{cr} = \frac{40,300}{1 + \frac{1}{950}\left(\frac{s}{t}\right)^{1.75}} \quad \ldots\ldots\ldots \quad (18)$$

$$p_{cr} = 1.09 \times 10^8 \left(\frac{t}{s}\right)^2 \quad \ldots\ldots\ldots \quad (19)$$

In reference (13), when explaining formula 18, commonly called Montgomerie's formula, Evans states: "It is based on numerous tests with plates to whose loaded edges double angles were fixed in order to simulate actual riveted attachments at transverse

floors. The unloaded edges were completely unsupported and unrestrained so that the results relate only to actual ship panels whose short, unloaded edges are far apart; at least four or five times the distance between loaded edges." Unfortunately this situation does not exist in longitudinally framed vessels.

Formula 19, credited to Bryan, is more suited to the problem of longitudinally framed ships. This expression is valid for rectangular plates simply supported on all four sides, with short edges loaded in compression. (19) It is applied by Vedeler (10) and St. Denis (17) to their structural problem. Although there is loading from the transverse direction, which this formula neglects, there is also a certain amount of end fixity which hopefully will cancel adverse effects of the long edge load.

Deck thickness, plotted in Figure XII against ship length, follows three straight line segments: one from 250 feet to 390 feet, another from 390 feet to 900 feet, and a constant value for lengths greater than 900 feet.

The ratio of Rule deck thickness to Rule frame spacing is an important buckling criterion. Figure IX is a plot of this relation with length. With the thickness to frame spacing ratios, critical buckling stress of deck plating can be calculated using formulas 18 and 19. Although a slight p_3 stress is present, it can be neglected due to its zero average value. The girder bending stress, p_2, is small and therefore can be neglected. The critical stresses, as calculated by formula, have been plotted in Figure X. Again we will compare this stress with an allowed

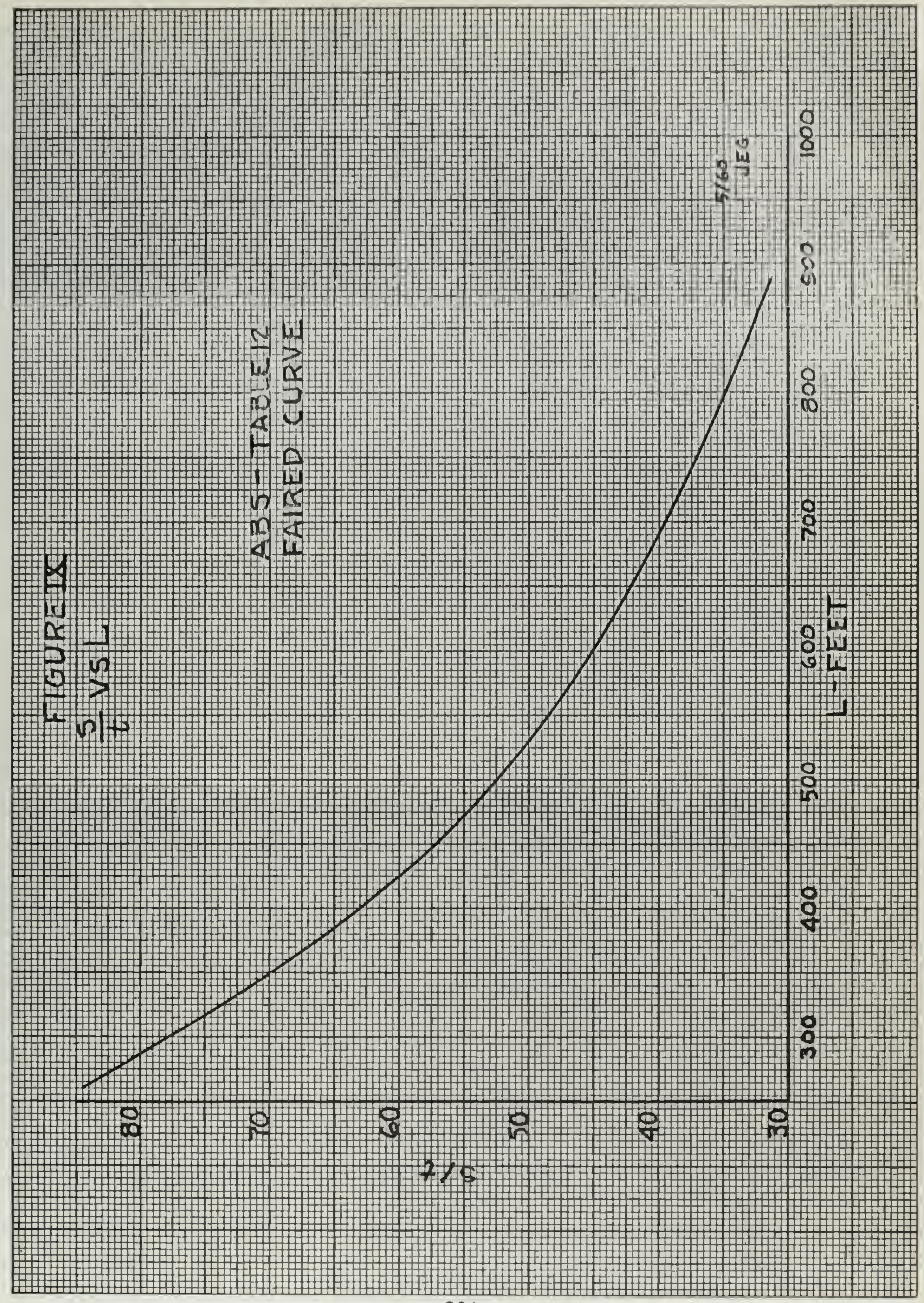


FIGURE IX
S/t VS L
ABS-TABLE 12
FAIRED CURVE
L-FEET
S/t
300
400
500
600
700
800
900
1000
30
40
50
60
70
80
5/60
JEG

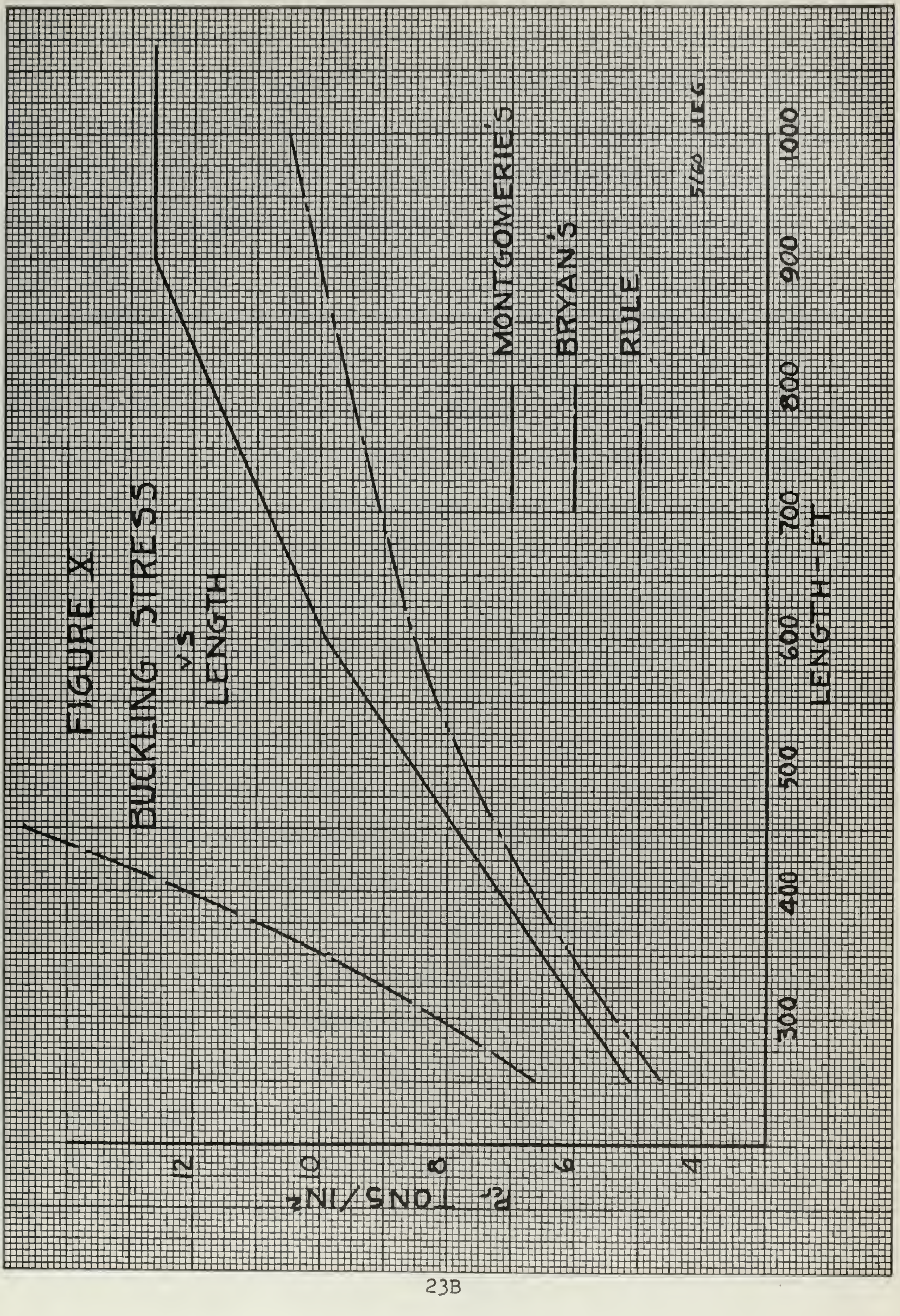


FIGURE X
BUCKLING STRESS
VS
LENGTH
MONTGOMERIE'S
BRYAN'S
RULE
TONS/IN²
12
10
8
6
4
300
400
500
600
700
800
900
1000
LENGTH-FT

ship's bending stress. We have seen, in the section entitled "Section Modulus Analysis," that the stress allowed by the Rules is not exactly Tobin's or Abell's. As a consequence, for the sake of consistency as well as good engineering practice, we will compare critical buckling stresses with allowed Rule ship's bending stress.

Comparison of the three curves of stress shows that Bryan's formula is not used, although it is a much more acceptable criterion for this type of ship. Montgomerie's formula does appear to give results that are somewhat compatible with ship's bending stresses allowed by the Rules, even though there is no known valid reason to justify its use. One will note that critical stresses computed by Mongomerie's formula follow three straight line segments; one from 250 feet to 600 feet, another from 600 feet to 900 feet, and a constant value for lengths greater than 900 feet.

Although the foregoing shows rather conclusively how deck plating thickness was derived by the A.B.S., there still remains a question as to whether this criterion ever need be applied. Should the modulus requirements be great enough, one is forced to make plating thicker than the minimum thickness found in Table A (3). Since deck plating thickness is greater than side shell plating and less than bottom, at lengths above 450 feet, the section modulus of a rectangle shape of deck plating thickness, with and without two longitudinal bulkheads, seemingly would bound most actual ship's moduli. Longitudinal bulkhead plating thickness, found in Table 10 of reference (7) is seldom greater than 0.6 inches.

[illegible]

[illegible]

We therefore expect actual plating modulus to be nearer the rectangular shape modulus with no longitudinal bulkheads. Vedeler uses this shape in arriving at deck area for the Det Norske Veritas Rules (10). Equating section moduli one can arrive at the following formulas.

$$t_p = f_{tp} \frac{B\,(d + .055L)}{12D\,(B + D/3)} \quad \ldots\ldots\ldots\ldots \quad (20)$$

$$t_p = f_{tp} \frac{B\,(d + .055L)}{12D\,(B + 2D/3)} \quad \ldots\ldots\ldots\ldots \quad (21)$$

Formula 20 is considered the upper bound on deck thickness and formula 21 the lower. These two formulas, along with required plating thickness, are plotted in Figure XI. Although this plot infers that section modulus criterion will normally overshadow thickness requirements, the minimum imposed by Table A (3) is realistic and must be checked. We expect greatest concern at lengths of 825 to 950 feet.

C. Side Shell Plating Analysis

Abell (1) gives a development of requirements for thickness of side plating which shows linearity with ship length. If we picture the ship as a girder, and accept the premise that maximum shear in the web (side plating) occurs at the neutral axis, we get the following formula for thickness of side shell:

$$t = \frac{VQ}{vI} \quad \ldots\ldots\ldots\ldots\ldots \quad (22)$$

Where t is the thickness of one side of the plating and Q the first moment of one-half the ship section.

[illegible]

$$[illegible] \qquad (\text{[illegible]})$$

$$[illegible] \qquad (\text{[illegible]})$$

[illegible]

[illegible]

[illegible]

[illegible]

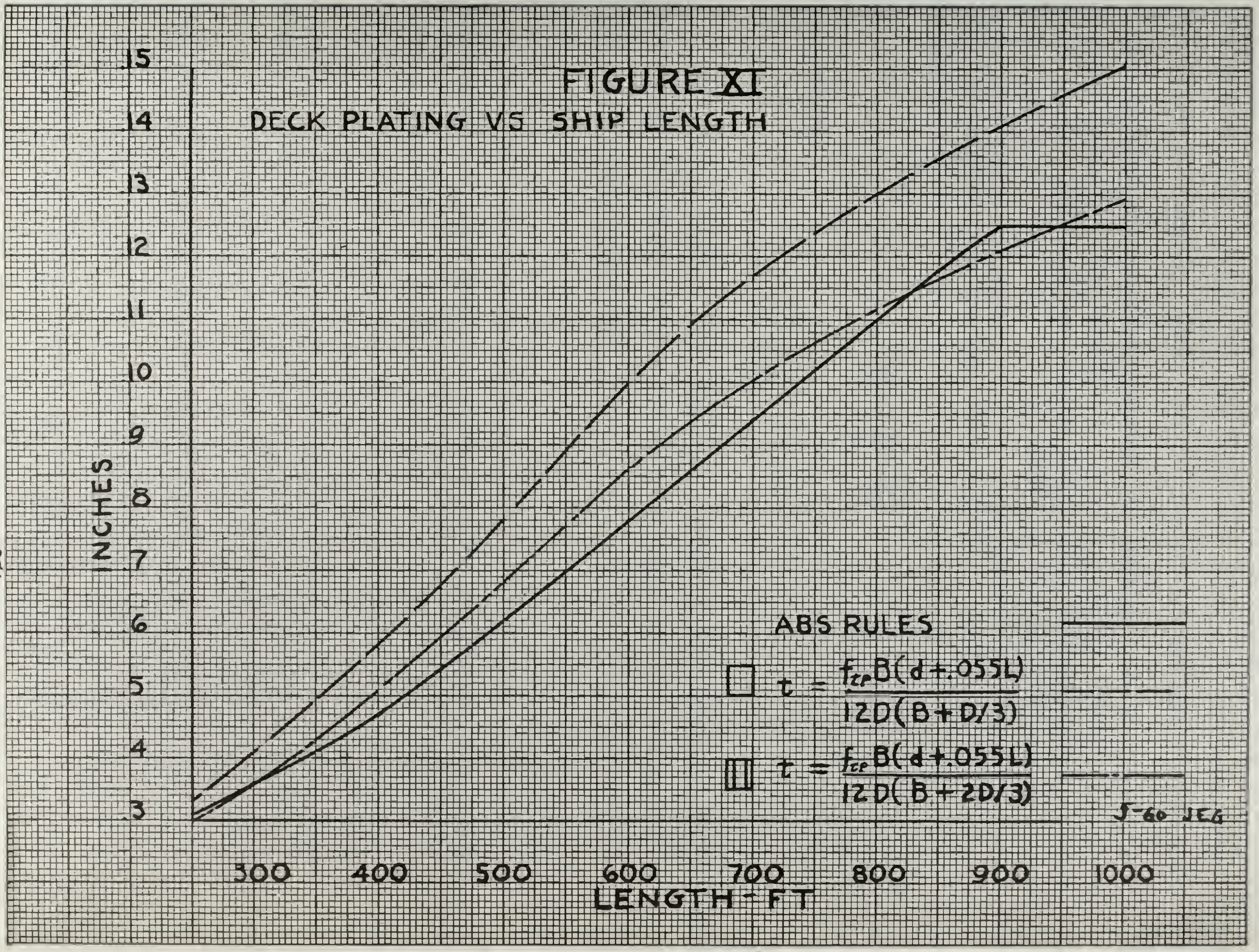


FIGURE XI
DECK PLATING VS SHIP LENGTH
INCHES
3
4
5
6
7
8
9
10
11
12
13
14
15
300
400
500
600
700
800
900
1000
LENGTH - FT
ABS RULES
$t = \frac{f_{cp} B(d + .055L)}{12D(B + D/3)}$
$t = \frac{f_{cp} B(d + .055L)}{12D(B + 2D/3)}$
5-60 JEG

Assuming that the maximum stress due to longitudinal bending varies as the length, we can also say that the maximum shear stress v varies as length. This can be gotten if we assume Abell's formula with stress proportional to length.

$$p = 5\left(1 + \frac{L}{1000}\right) \quad \ldots\ldots\ldots\ldots\ldots \quad (23)$$

We then assume $V \propto$ displacement $\propto LBd$.

Then:

$$t = \frac{VQ}{vI} \propto \frac{Q}{I} \propto \frac{LBd}{L}$$

Therefore:

$$t \propto \frac{Q}{I} Bd$$

Now Abell states that QD/I is constant for similar geometrical sections, and that it was found that this relation applied to actual vessels. This gives us:

$$t \propto B \frac{d}{D}$$

Taking d/D as constant for a particular ship type, and assuming beam a function of length, we get:

$$B = aL + b$$

and:

$$t \propto aL + b \quad \ldots\ldots\ldots\ldots\ldots \quad (24)$$

The actual formula adopted by the Committee was:

$$t = 1.05 \times 10^{-3}L + 0.17 \quad \ldots\ldots\ldots \quad (25)$$

Units of t are inches and L is in feet.

[illegible]

$$[illegible]$$

[illegible]

$$[illegible]$$

[illegible]

$$[illegible]$$

[illegible]

$$[illegible]$$

[illegible]

$$[illegible]$$

[illegible]

$$[illegible]$$

[illegible]

The assumptions of this analysis leave a great deal of room for error, as will be shown later, but apparently it is what the ABS used; for tabulated values of minimum side shell thickness are composed of three straight line segments as is shown in Figure XII which also shows Abell's formula for comparison. We see there is little difference except when L is greater than 800 feet. This is understandable when we note that Abell was considering ships of less than 600 feet. In the length range from 250 feet to 400 feet we see that the points being tabulated to even hundredths of an inch gives a sort of saw tooth effect. We can fit a straight line through the lowest points with a maximum error at any point of 0.0075 inch when we use the formula

$$t = 1.25 \times 10^{-3} L + 0.10 \qquad 250' < L < 400' \tag{26}$$

In the range of ship length from 370 feet to 800 feet we can match every tabulated point exactly with the equation

$$t = 10^{-3} L + 0.20 \qquad 370' < L < 800' \tag{27}$$

For the lengths from 800 feet to 1000 feet the thickness of side plating required is a constant 1.00 inch. Apparently, it is felt that the shear stress will not increase in the side shell at lengths beyond 800 feet. This could be due to either the feeling that wave bending stresses no longer increase at these lengths and/or that d/D will decrease as B increases at these greater lengths. These aspects are treated more fully elsewhere in this paper.

The assumptions of proportionality used by Abell (1) and the Load Line Committee are found to be rather prevalent and widely

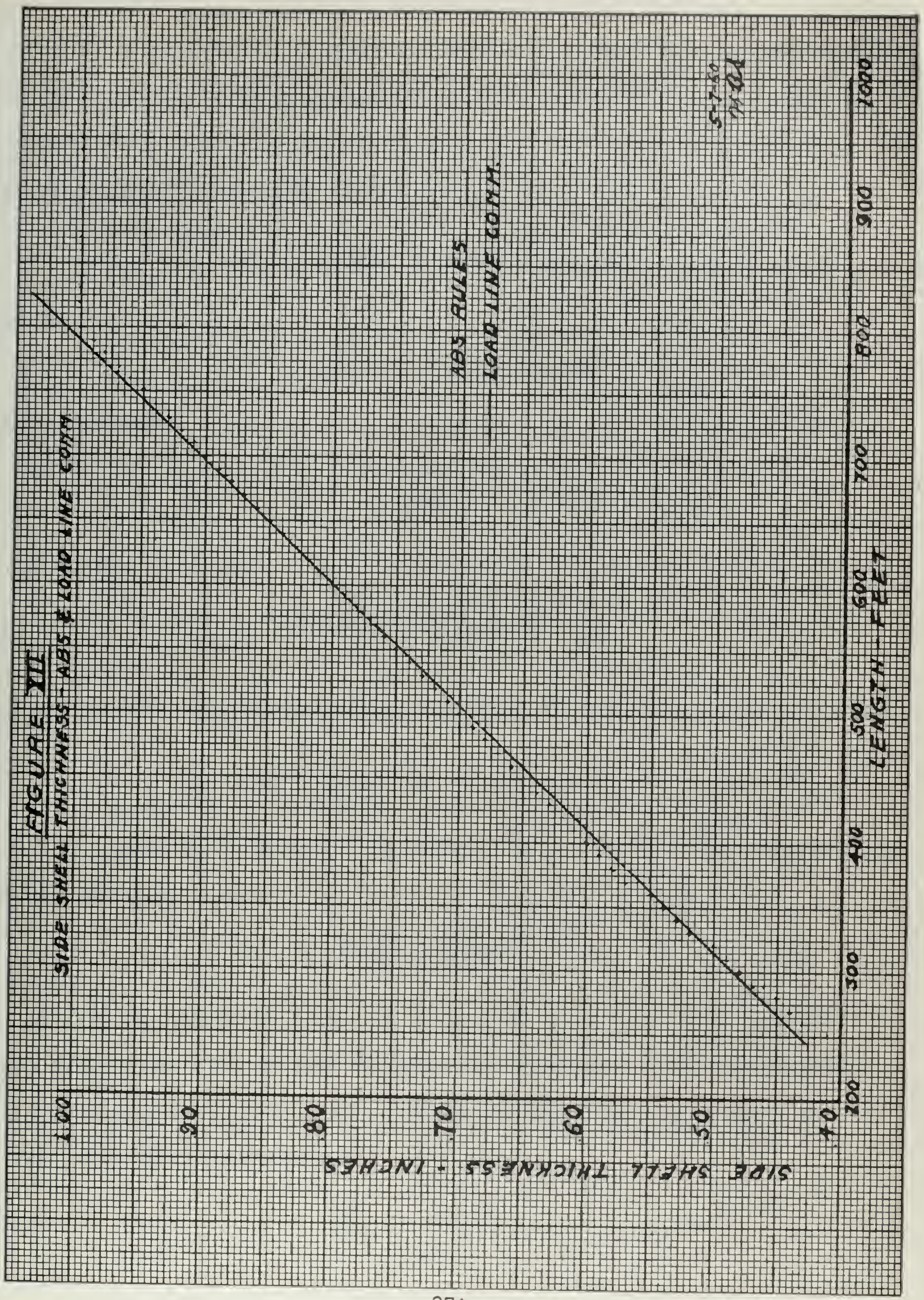


FIGURE XII
SIDE SHELL THICKNESS - ABS & LOAD LINE COMM
ABS RULES
LOAD LINE COMM.
200
300
400
500
600
700
800
900
1000
LENGTH - FEET
1.00
.90
.80
.70
.60
.50
.40
SIDE SHELL THICKNESS - INCHES

accepted in the naval architecture field. We find it instructive to repeat Abell's derivation in a more mathematically rigorous fashion.

Assume Telfer's bending moment (5).

$$M = aBdL^2 + bBL^3 \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \qquad (28)$$

Then:

$$V = \frac{dM}{dL} = 2aBdL + 3bBL^2$$

Again assume:

$$v = kp = k5\left(1 + \frac{L}{1000}\right) = k'(L + 1000)$$

This gives:

$$t = \frac{VQ}{vI} = \frac{(2aBdL + 3bBL^2)}{k'\ (L + 1000)} \frac{QD}{I} \frac{1}{D}$$

Using information contained in Table A of ABS Rules (7), we find:

$$b = \frac{L}{10} + 18 \quad \text{and} \quad \frac{L}{D} = 13$$

for the usual ship forms. Inserting these values and lumping constants we get:

$$t = K\frac{(L + 180)\ L}{(L + 1000)} \quad . \; . \; . \; . \; . \; . \; . \; . \; . \qquad (29)$$

It is of interest here to note that we would have gotten the same end result if we had originally assumed a simpler bending moment formula of $M = aBdL^2$. This is the same as using Abell's assumption $V \propto$ displacement $\propto BdL$ since:

$$V = \frac{dM}{dL} = 2aBdL.$$

The real difference here comes in using a more exact expression for shear and not simply neglecting the constant e in an expression of the form $v = e + gL$. Abell chose to let $B = aL + b$, but let $v = e + gL \approx gL$ for the sake of a simple formula. Since $b = 0.1\ (L + 180)$, while $v = k'\ (L + 1000)$, we can see that retention of the constant in the stress formulation is much more important than in the expression for standard beam.

In order to show the effect Abell's simplification of the formula has, we have solved our formula for a constant K which will bring the thickness into agreement with ABS requirements (3) at a ship length of 400 feet which is a break point in the ABS Rules. Calculations were made with no corrosion allowance and with a corrosion allowance of 0.11 inch. Figure XIII, which displays these results, shows us that, while a corrosion allowance will bring the curves closer to agreement, the slope is very much altered when the more rigorous derivation is used. A corrosion allowance of 0.355 inch would be required to effect coincidence at both 400 and 800 feet.

By requiring coincidence only at 800 feet we could cause the calculated values to fall below the ABS Rules all along, but the differences are not so large as when coincidence was required at 400 feet. The results of these calculations are displayed in Figure XIV. From this we see that we are probably better off requiring coincidence only at a greater length. This takes into account ABS experience which has probably not allowed the plating to be too thin at any length. The rigorous formula then says that

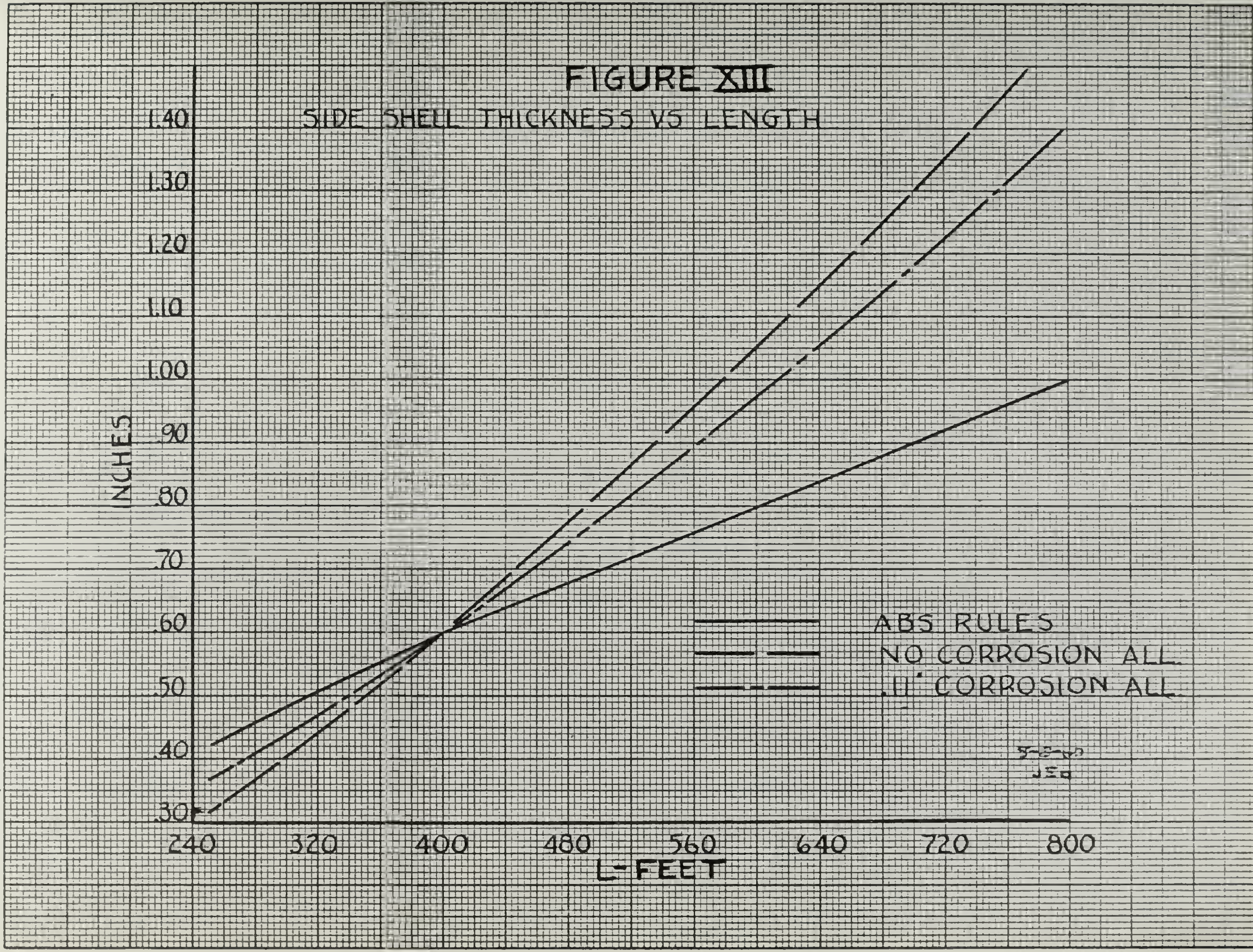
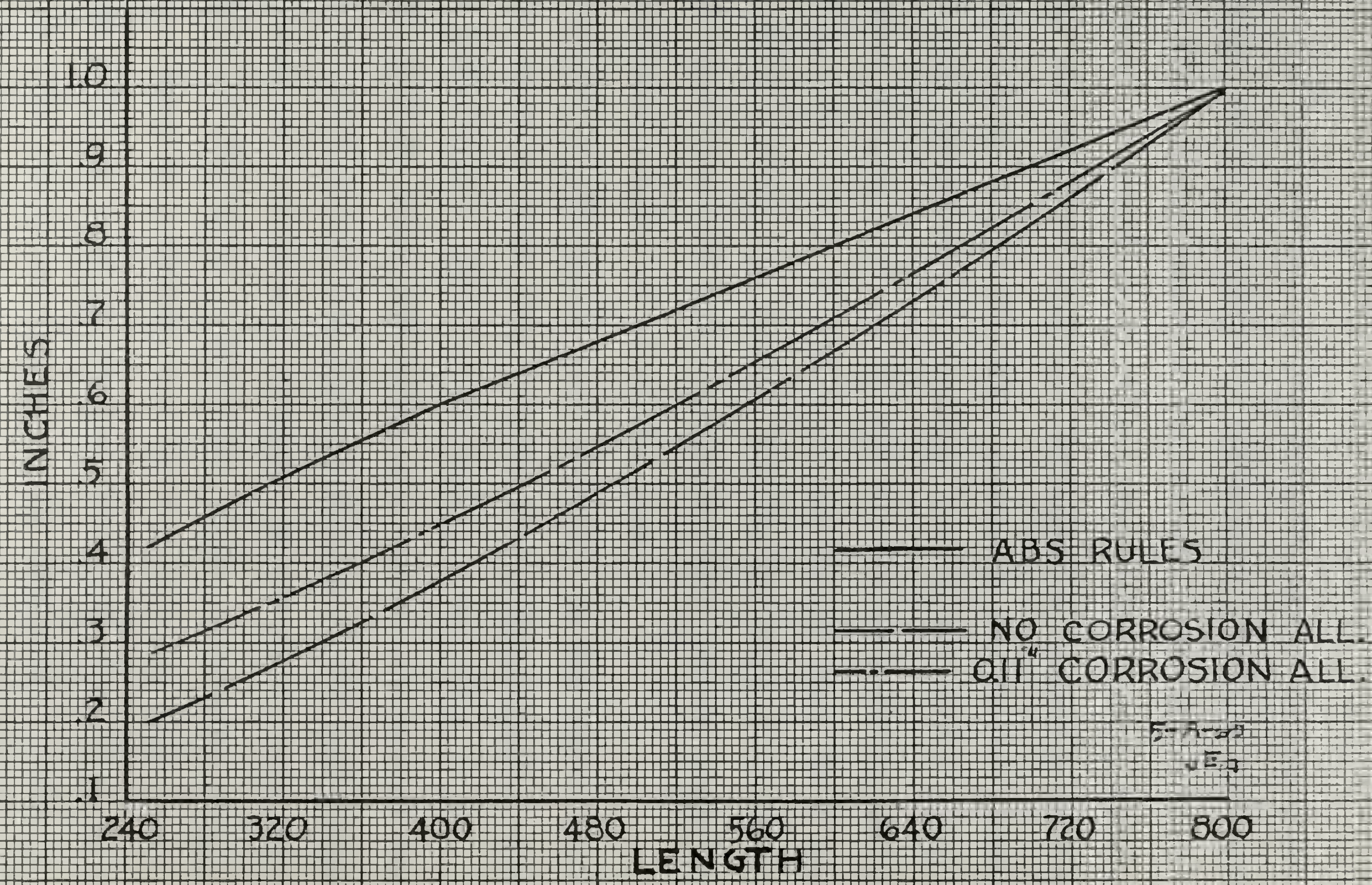


FIGURE XIII
SIDE SHELL THICKNESS VS LENGTH

FIGURE XIV

SIDE SHELL THICKNESS VS LENGTH



the ABS Rules are too stringent as the length decreases. As we get to shorter lengths and rather thin plates, our recently derived formula may no longer be fully applicable. While it may fulfill the shear criterion at the neutral axis, it may give a plate thickness which is entirely inadequate for the hydrostatic loads in the side plates nearest the bottom. It might also give plates too thin to withstand buckling from ship girder bending stresses near the deck or bottom. Presumably, experience has shown the ABS requirements to be at least adequate for these shorter ships. Courtsal (8) and Antoniou (9) examined the hydrostatic pressure aspect but were not able to reach any strong conclusion as to just what its effect was or if it was most important only in a certain range of lengths. This was partly due to the fact that they worked with the 1955 and 1956 ABS Rules respectively, which considered tankers of maximum length of only 550 feet.

To realistically determine the constant in the more rigorously derived formula, and thus relate it to experience, would require more information than is presently available to us. We can adopt the premise that ABS Rules are conservative and that nowhere should we then allow our values to exceed ABS requirements; but then we face the additional problem of assessing the validity of ABS practice of allowing a constant plate thickness above certain lengths. This is probably due to the feeling that with very long ships the L/20 wave is no longer valid and the bending moment used in the derivation is no longer being attained. Somehow then, we would also have to modify the bending moments at longer lengths

[illegible]

so as to give a constant, or nearly constant, plate thickness. A best first guess would probably be to duplicate ABS Rules at 800 feet ship length and then require:

$$K = K' \frac{(L + 1000)}{(L + 180)L} \qquad \text{when } 800 < L < 1000 \tag{30}$$

This is:

$$t = K' \doteq 1.00 \text{ inch} \qquad 800 < L < 1000$$

At the shorter lengths we could only borrow from Courtsal and Antoniou, using hydrostatic head as the test of the plating. To definitely tie this to experience would require a mass of information such as is available to the Classification Societies. Their experience could determine what the usual mode of failure would be in plating which might not be thick enough and this in turn should tell us what the real governing criterion would be.

D. Bottom Shell Plating Analysis

The requirements for bottom shell plating in transversely framed ships under ABS Rules were analyzed by Evans (13). The analysis of longitudinally framed ships shows marked similarity of requirements for the two cases. There are two criteria to be satisfied: instability and hydrostatic pressure. For transversely framed ships, ABS Rules give minimum bottom shell thickness with both instability and hydrostatic head as the governing factor. The Tentative Rules for longitudinally framed tankers give only one value of thickness for each tabulated length. Therefore, it is reasonable to expect that one or the other criterion might apply

[illegible]

to only a segment of the plot of tabulated plating thickness, according to which imposes the more stringent requirement at a given ship length.

Using Montgomerie's expression for instability, Evans found agreement with the tabulated Rule values to within $\pm$ 0.012 inch. Although this formulation was derived with little regard for panel aspect ratios and specific conditions of edge fixity, Evans showed very good historical and technical reasons why the ABS should have chosen to use it. Adopting this same reasoning, we choose as our hull girder stress Tobin's relation, and for an experience factor 1.25 as was used by Evans. This gives us:

$$1.25 \times 2240L^{1/3} = \frac{40,300}{1 + \frac{1}{950}\left(\frac{s}{t}\right)^{1.75}} \quad \ldots\ldots\ldots \quad (31)$$

For frame spacing we use the relationship gotten from Table A of the Rules.

$$s = \frac{L}{40} + 20 \quad \ldots\ldots\ldots\ldots\ldots \quad (32)$$

Figure XV shows the result of the calculations.

Up to a length of 400 feet we find an error of $\pm$ 0.015 inch between the calculated and tabulated values. Beyond 400 feet the error increases sharply with the calculated value becoming increasingly too thin. This break at 400 feet can also be seen in both the deck and side shell thickness curves.

This extension of Montgomerie's formula seems even more probable when we examine Figure XV which is a plot of plate thicknesses for transversely framed ships from Table 12, ABS Rules, corrected

[illegible]

[illegible]

$$[illegible]$$

[illegible]

$$[illegible]$$

[illegible]

[illegible]

[illegible]

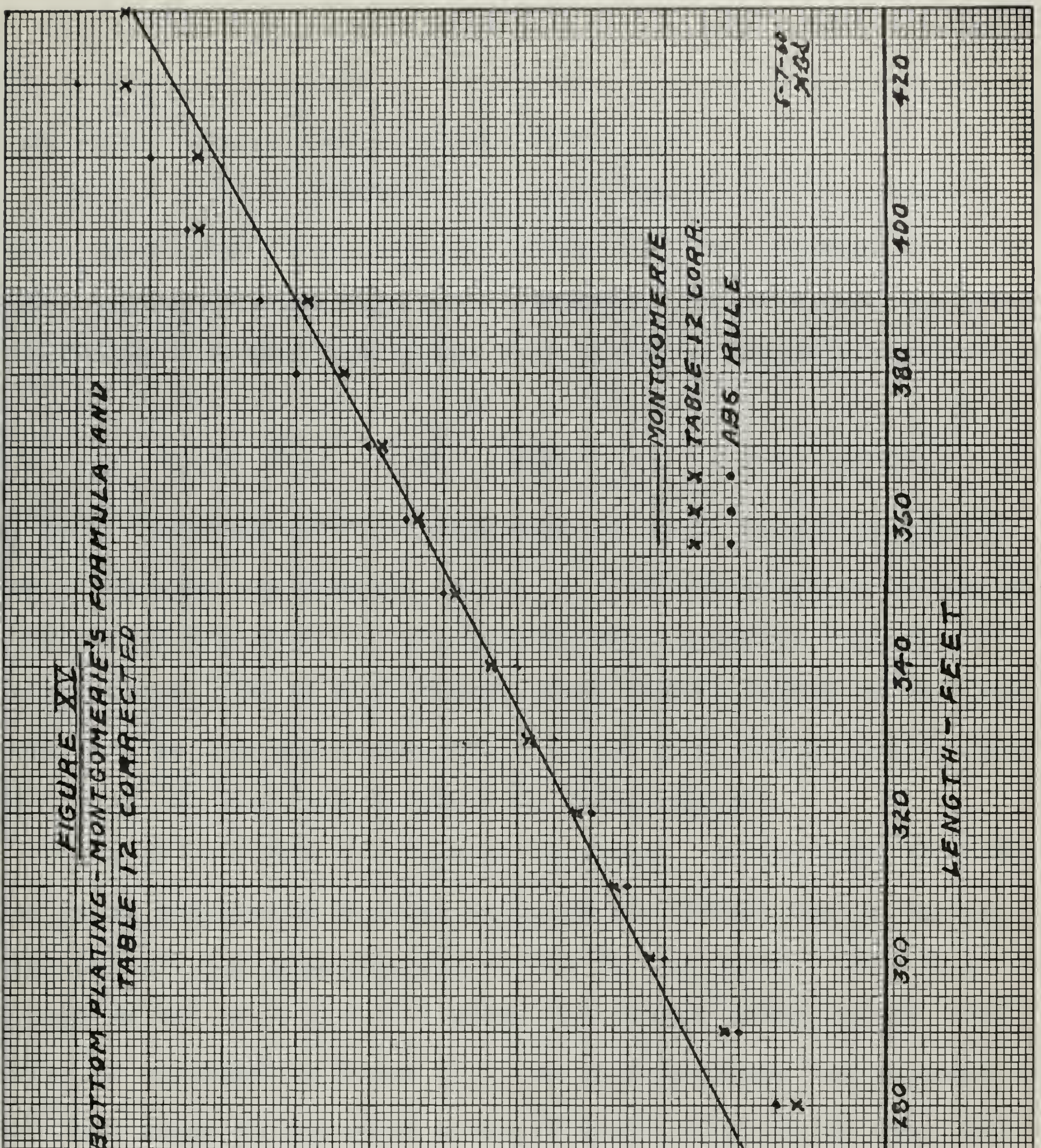


FIGURE XV
BOTTOM PLATING - MONTGOMERIE'S FORMULA AND TABLE 12 CORRECTED

by the Rules to agree with scantlings given for longitudinally framed tankers in Table A. This gives slightly better agreement than using Montgomerie's formula directly.

Having found a divergence of buckling theory and tabulated values above 400 feet, it would naturally seem reasonable to examine hydrostatic loading next. Again we follow Evans' analysis and employ a formula which has been given by several theorists(15,16).

$$p_3 = \tfrac{1}{2}K_L \rho \; h \left(\frac{b_2}{t}\right)^2 \frac{1}{144} \quad \ldots\ldots\ldots\ldots \quad (33)$$

K_L is a modifying constant dependent on panel aspect ratio and the plate edge considered. Since we find the greatest stress at midpoint of the panel's long side we will consider this in our calculations. If we also assume panel aspect ratios to be greater than two, we can use Timoshenko's values (15) and assume K_L equal to unity. We also set b_2 equal to frame spacing.

Next we consider what stress to allow. On the long side of the panel considered we find not only the stress due to hydrostatic load, but also the stress due to the plate's role as a flange of the hull girder. After the design method of St. Denis (17), as expanded by Evans (13), we combine these two stresses and set them equal to the mild steel strength as given by ABS. Again we use Tobin's expression for girder stress.

$$p_3 = 32{,}000 - 2240L^{1/3}$$

This gives us our working formula:

$$32{,}000 - 2240L^{1/3} = \tfrac{2}{9}h \left(\frac{s}{t}\right)^2 \quad \ldots\ldots\ldots \quad (34)$$

[illegible]

Now we have the question of what head of water, h to use. After numerous trials it seemed best, at the outset, to set the head equal to D + 8 for ship lengths over 400 feet. This is given in the section of ABS Rules pertaining to longitudinals and elsewhere in those Rules. Although we will not burden the reader with all the trials made in arriving at this choice, the effect of some of them will be shown in the development which follows.

Examination of the plot, Figure XVI, shows that there are also breaks at lengths of 530, 620 and 760 feet. The range 620 to 760 feet is actually a straight line which can be duplicated by the formula:

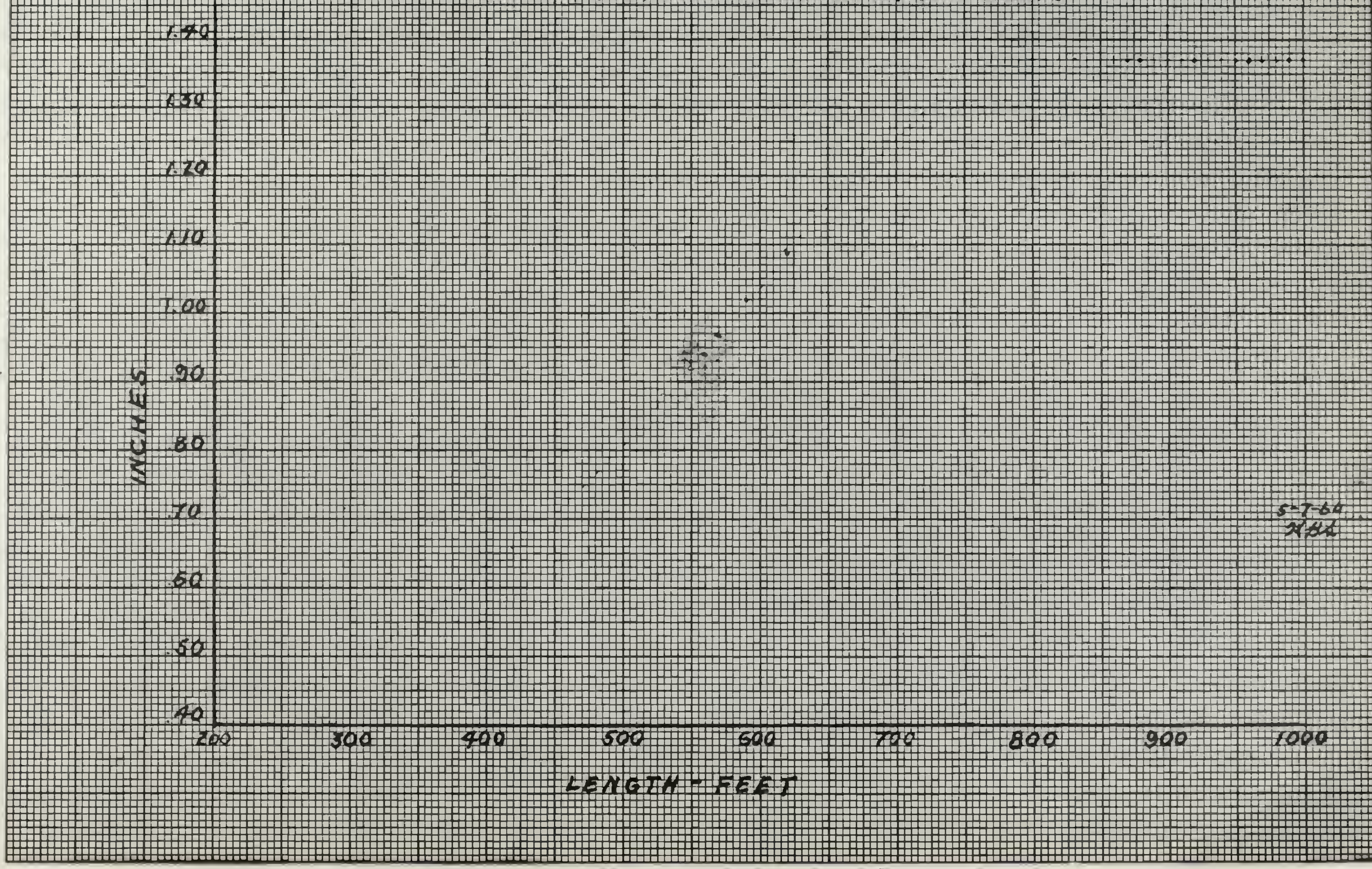
$$t = \frac{L}{500} - 0.15 \qquad 620 < L < 760 \quad \ldots\ldots\ldots \qquad (35)$$

We might dismiss this as another case of straight edge extrapolation, however we desire to examine its rationality and will compare it with theory.

Evans found that, for transversely framed ships, there was a distinct break in the curve of calculated values at a length of 600 feet on account of draft becoming a constant at greater lengths. Our analysis of f numbers showed that certain assumptions would show a decrease in the d/L ratio at greater lengths, but never gave draft as a constant. Using draft as a constant beyond 600 feet length gave an unsatisfactory answer in our case also. To begin our investigation, let us accept the value given in Table A, which is L/D = 13. This will give us values of the head h. The first calculations, shown in Figure XVII, show reasonable agreement with, but are somewhat above the Rule values.

FIGURE XVI

BOTTOM PLATING THICKNESS BY ABS RULES



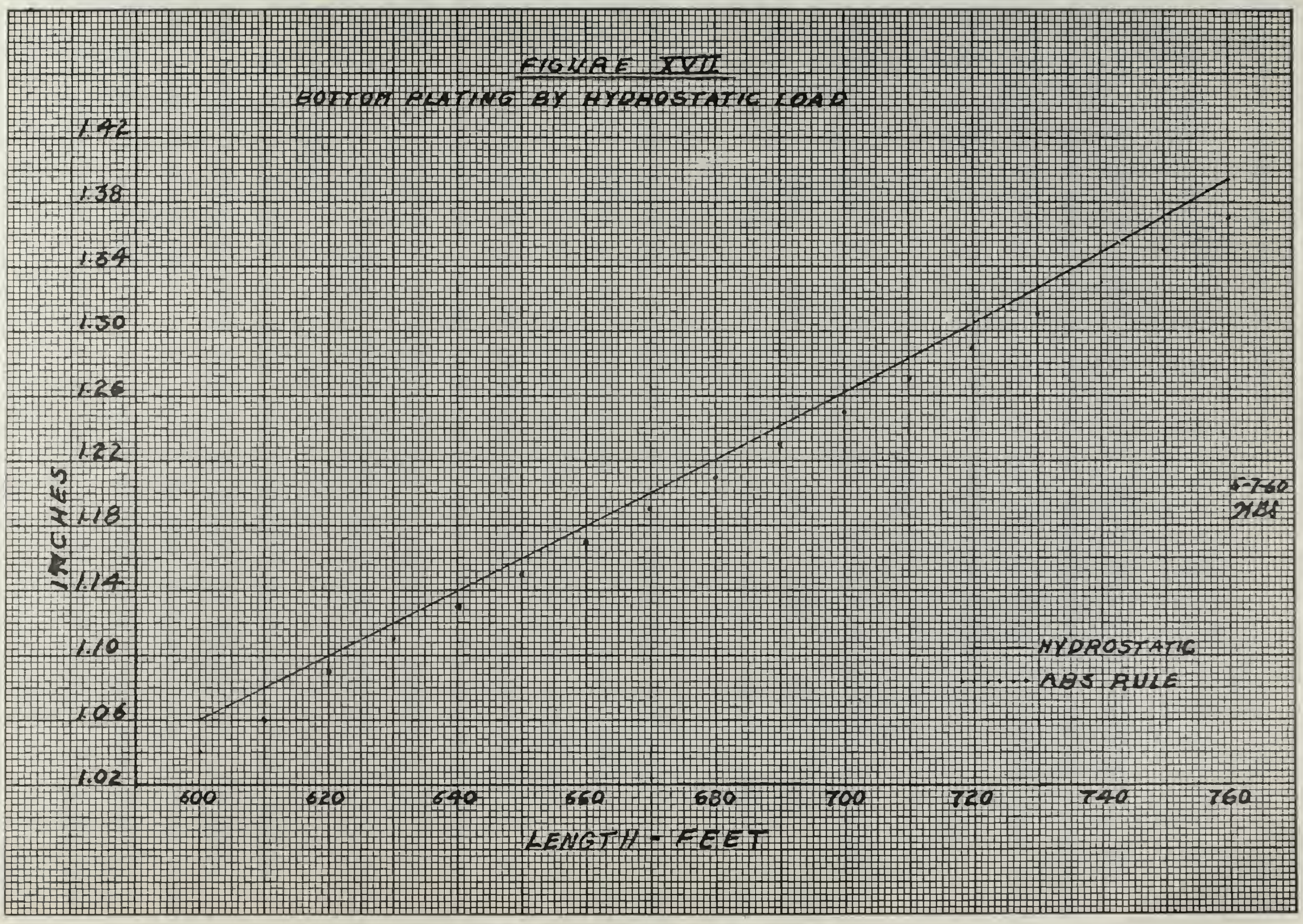


FIGURE XVII

BOTTOM PLATING BY HYDROSTATIC LOAD

They are based on the formula:

$$32{,}000 - 2240L^{1/3} = \frac{2}{9}\left(\frac{L}{13} + 8\right)\left(\frac{\frac{L}{40} + 20}{t}\right)^2 \quad \ldots\ldots \quad (36)$$

One thought is that the value of K_L might be in error. To find out what value of K_L might be satisfactory, the calculations were repeated using the Rule values of plating thickness and solving for K_L. Part of the results are shown in Table I.

TABLE I

Required Value of K_L to Match Rule Values of Bottom Plating

L	760	740	720	700	680	660	640	620	600
K_L	.9659	.9710	.9757	.9791	.9753	.9832	.9836	.9827	.9615

Although the plating thickness is tabulated only in even hundredths of an inch, which gives variations in K_L, it seems reasonable that a value of K_L could be chosen so as to give very good agreement with tabulated values of plating thickness. Examination of the work of Lamble and Shing (18) will show that existing theory gives ample justification of picking such a number. However, we have rejected this for reasons which become more obvious as the investigation continues.

In our analysis we found the assumed position of the neutral axis to be variable with length as shown in Figure I. To take this into account we should multiply the assumed ship girder stress (Tobin's expression) by the ratio of distance from baseline to neutral axis over distance of deck to neutral axis. We note that

this will reduce the girder stress in all cases, and cause a break in the curve at a length of 530 feet if we use the most recent Rules. This gives the formula:

$$32{,}000 - (\text{N.A. Corr.})\ 2240L^{1/3} = \frac{2}{9}\left(\frac{L}{13} + 8\right)\frac{\left(\frac{L}{40} + 20\right)^2}{t^2} \qquad (37)$$

In Figure XVIII we observe that these corrections give values which lie parallel but slightly below the tabulated values at greater lengths and parallel but about 0.04 inch above them at the shorter lengths. Apparently this is due to the fact that the neutral axis position breaks at 530 feet while the plating thickness breaks at 620 feet.

If we now take for our head of water h = D and again use the same neutral axis correction we can expect the calculated plate thickness to be less. A few trials showed that increasing the calculated thicknesses by a corrosion allowance of 0.03 inch gave rather good correspondence with the tabulated values in the range of lengths 400 to 560 feet. This is shown in Figure XIX. Here we are using:

$$32{,}000 - (\text{N.A. Corr.})\ 2240L^{1/3} = \frac{2}{9}\ \frac{L}{13}\left(\frac{\frac{L}{40} + 20}{t - 0.03}\right)^2 \qquad (38)$$

Using the neutral axis location gotten from the 1958 Tentative Rules we find a break point at 620 feet length, but there is no improvement of fit for the curve. Since the plating thickness values have not changed from the 1958 Rules, it may be that the 620 foot break was chosen to match that of the neutral axis position at that time and no account has been taken of the fact that it has since changed.

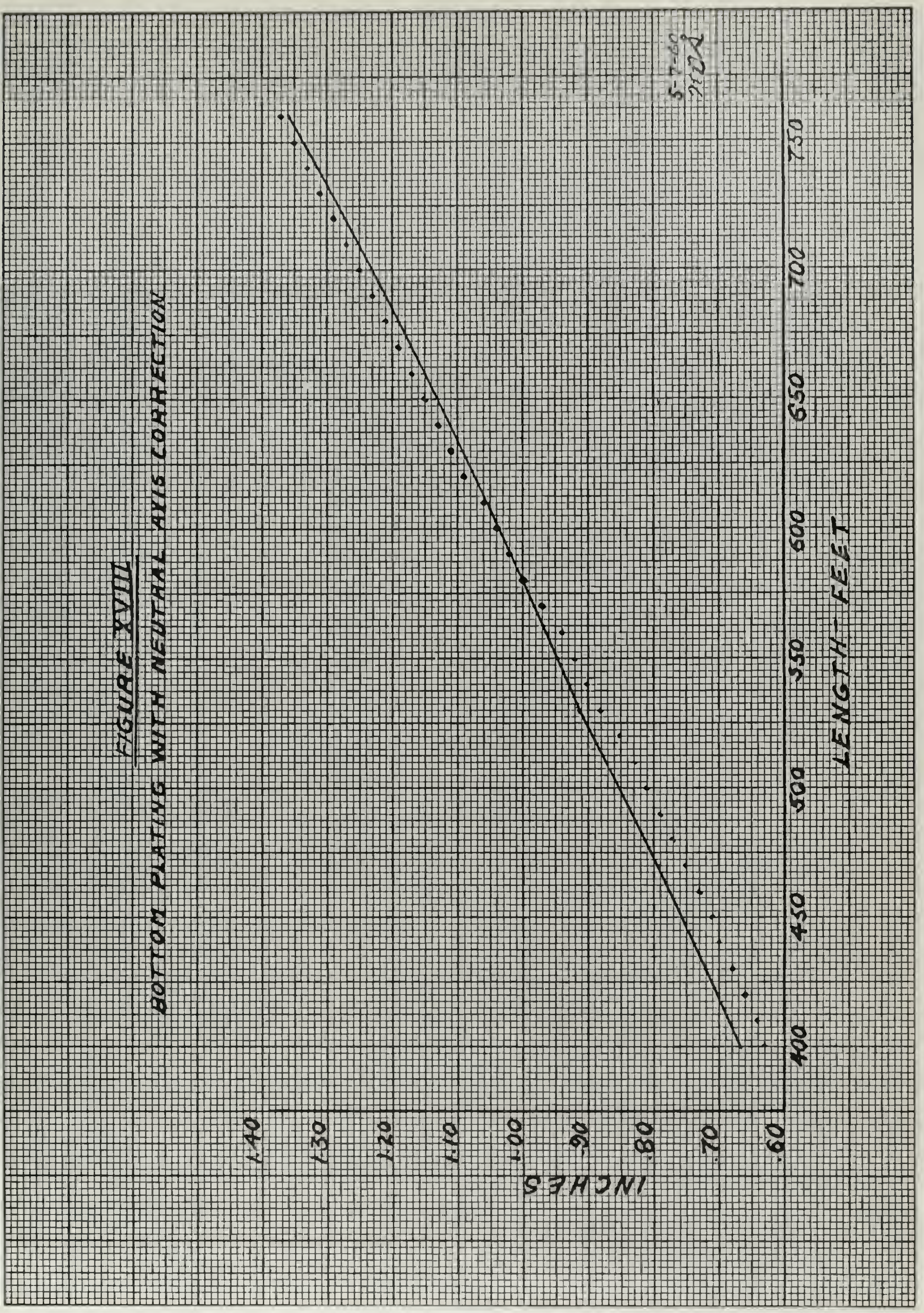


FIGURE XVIII
BOTTOM PLATING WITH NEUTRAL AXIS CORRECTION
LENGTH - FEET
400
450
500
550
600
650
700
750
INCHES
.60
.70
.80
.90
1.00
1.10
1.20
1.30
1.40

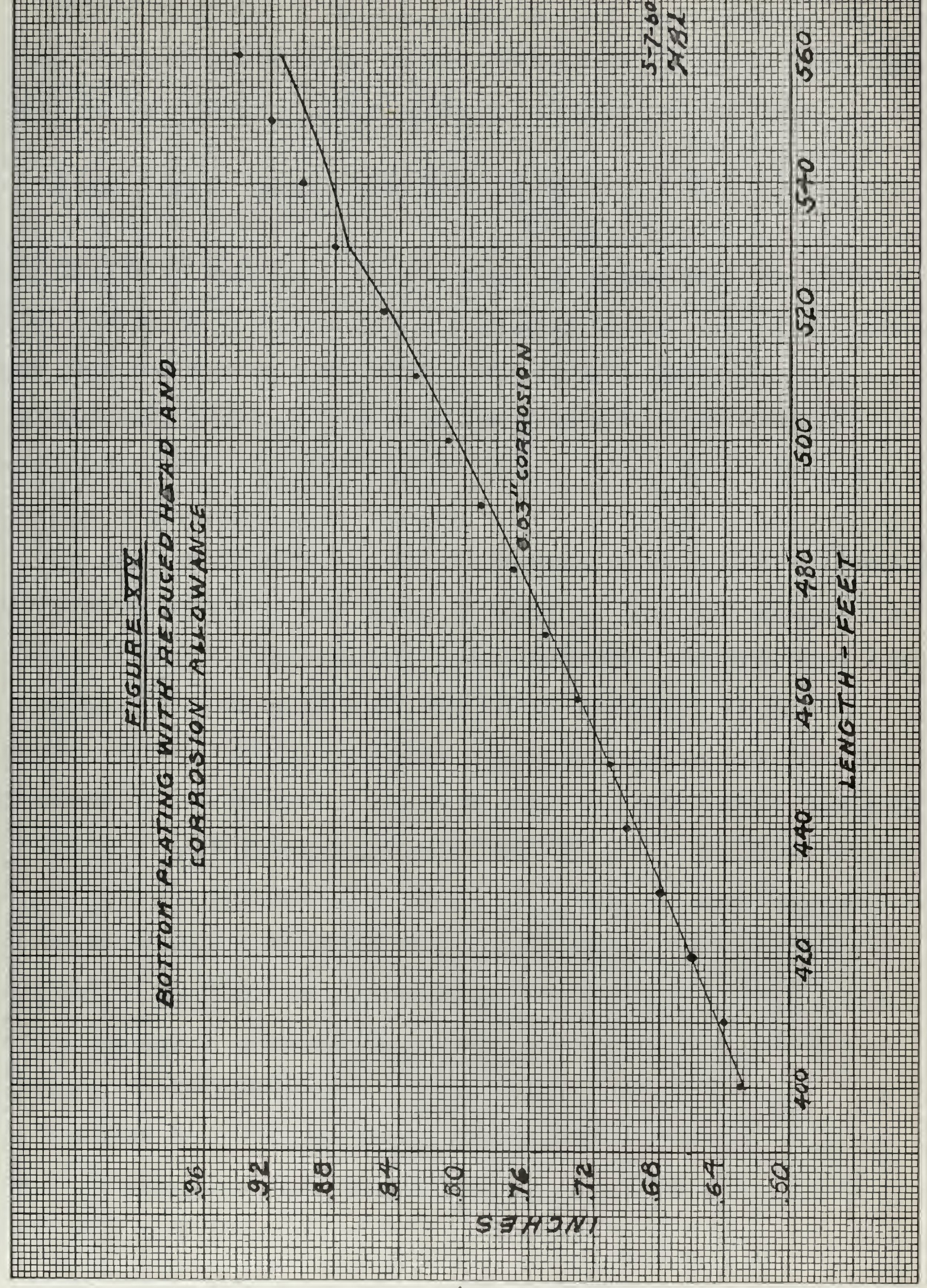


FIGURE XIX

BOTTOM PLATING WITH REDUCED HEAD AND CORROSION ALLOWANCE

Should we assume a faying flange on the longitudinals, we could consider the spacing of the frames reduced by the effective width of the flange. This gives an equation of the form:

$$32{,}000 - (\text{N.A. Corr.})\ 2240L^{1/3} = \frac{2}{9}\left(\frac{L}{13} + 8\right)\left(\frac{\frac{L}{40} + 20 - w}{t}\right)^2 \qquad (39)$$

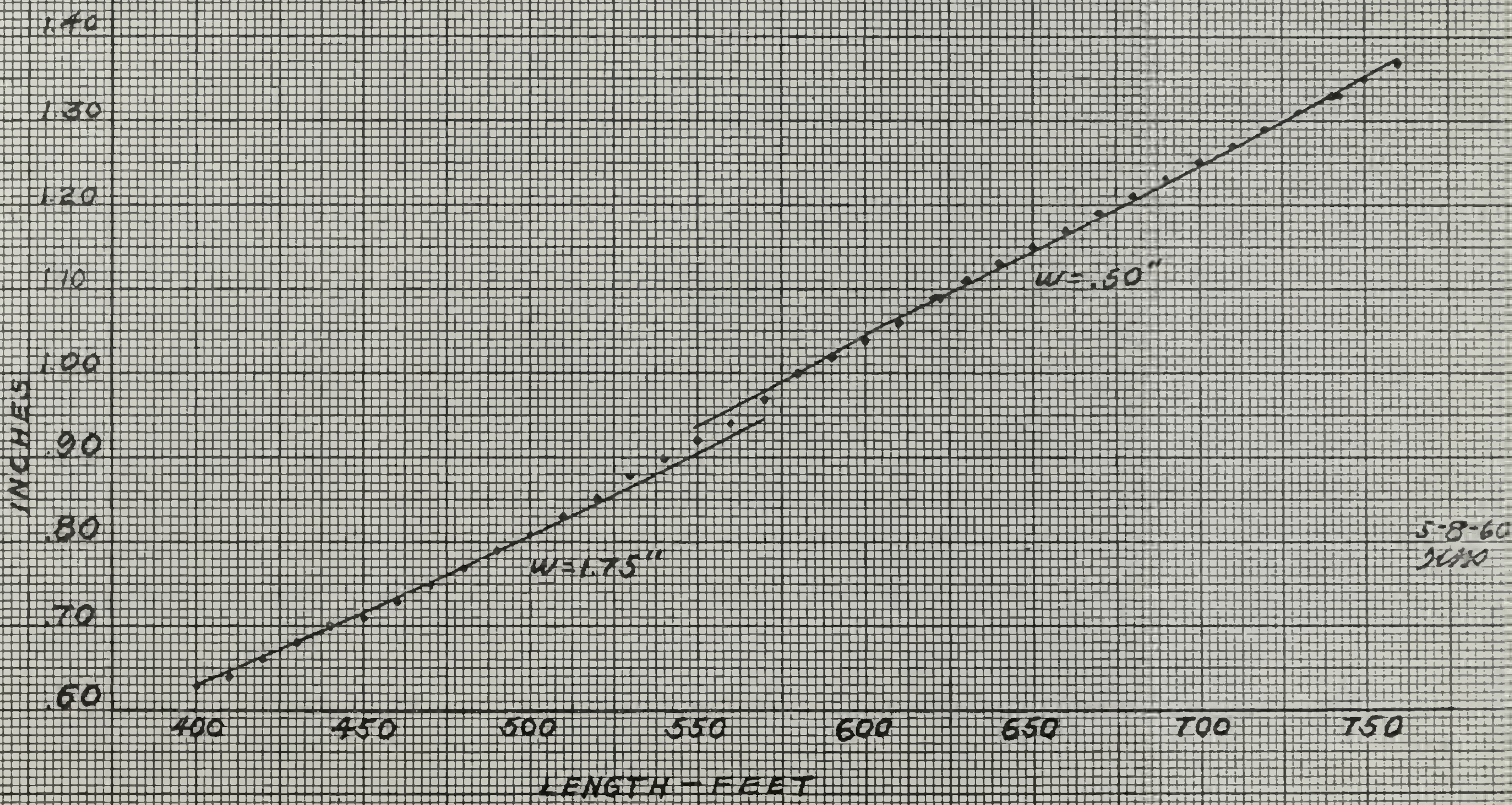
This was tried and found to give good agreement with the tabulated values if w = 1.75 inch for lengths to 560 feet with neutral axis correction from 1958 Rules. For lengths up to 760 feet a faying flange width of 0.50 inch was required as is shown in Figure XX. This inconsistency in faying flange width makes the solution undesirable.

The most reasonable result is obtained if we return to formula 38 assuming head of water equal to depth of ship and correct for neutral axis position with the 1959 f numbers. We have seen that a corrosion allowance of 0.03 inch gives agreement at shorter lengths. If we take a corrosion allowance of 0.10 inch we find agreement at the greater lenghts as seen in Figure XXI. There is a transition zone between 530 feet and 620 feet, the break points of the neutral axis position curves of Figure I, where the plating thickness required increases rapidly. This can be thought of as a faired-in increase in conservatism through this range since the curves are newly extended beyond 550 feet length and the ABS is cautious in extrapolating.

A number of solutions were tried based on draft, draft plus a wave height, and draft plus constants, but none of them gave reasonable results.

FIGURE XX

BOTTOM PLATING WITH FAYING FLANGE CORRECTION



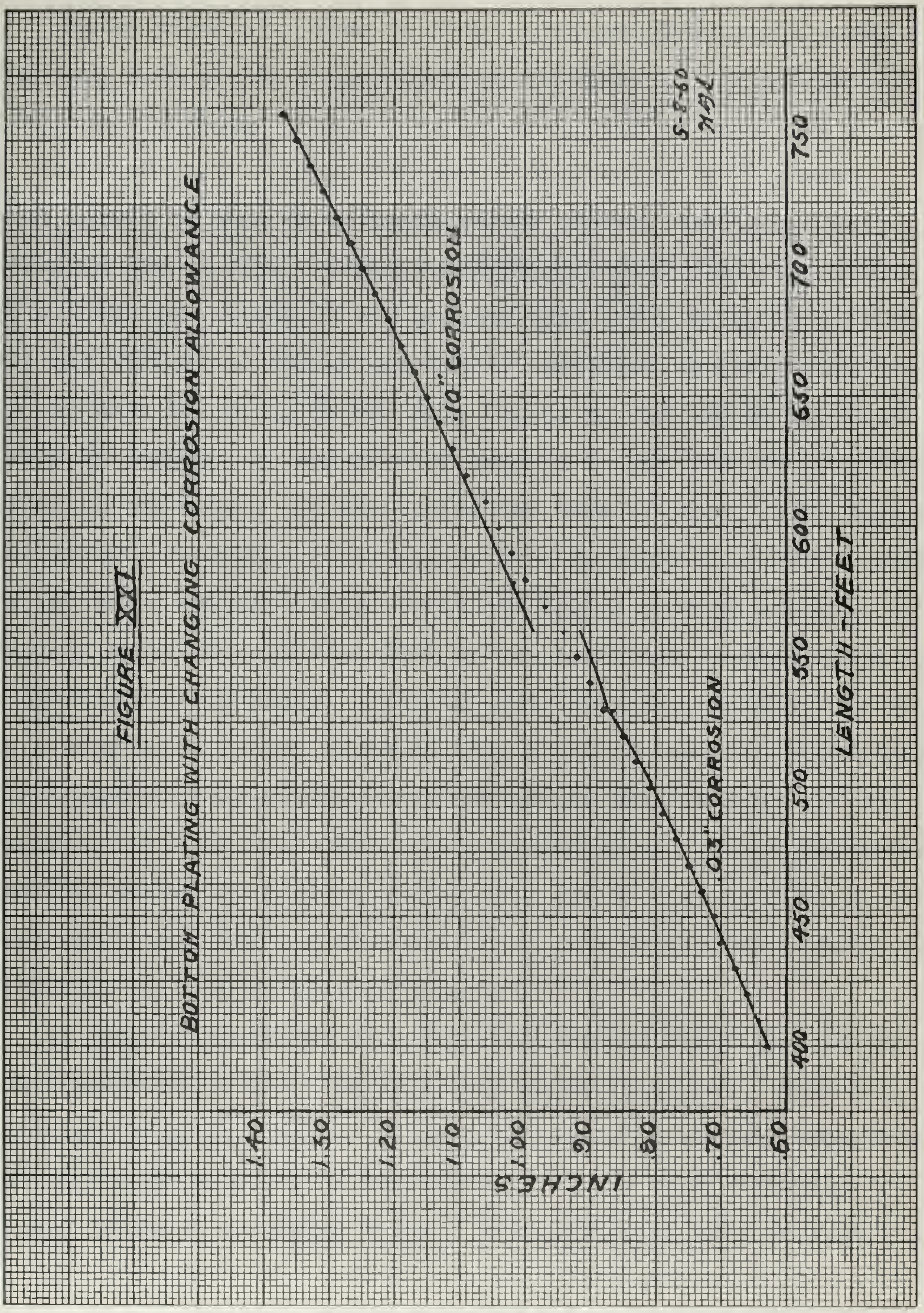


FIGURE XXI
BOTTOM PLATING WITH CHANGING CORROSION ALLOWANCE
.10" CORROSION
.03" CORROSION
INCHES
LENGTH - FEET
.60
.70
.80
.90
1.00
1.10
1.20
1.30
1.40
400
450
500
550
600
650
700
750
5-8-60

Review shows that when we first combined stresses we chose the long side of the plate panel as did Evans in his analysis of transversely framed ships. In our case, this is combining stresses at right angles with each other, which is incorrect. What should be done is to consider the shorter side where the stresses are in line. Using Timoshenko's value (15), we find $K_L = 0.685$ for aspect ratios greater than 1.5. Using this value and all methods of attack previously employed, we are unable even to approach the slope of the Rule values in the 400 to 620 foot range of length. At the greater lengths we are able to come fairly close to the slope, but even using the head of water equal to D + 8 the minimum corrosion allowance we find is about 0.17 inch. This is far above any that Evans found and beyond what we are able to reason, especially in the light of our inability to even match the slope for shorter ships. Although this puts the calculations used on a very shaky basis, it seems very probable that the requirements for longitudinally framed ships are really only "beefed up" extensions of those for transversely framed ships. Since Montgomerie's formula, which was derived emperically for transverse framing, is apparently used for the shortest ships, why should not a parallel extension be used at greater lengths?

At lengths above 760 feet the longitudinal frame spacing becomes constant as does the tabulated plate thickness. This can be interpreted in a number of ways in regard to assumed ship forms and magnitude of stresses and moments. These were discussed in regard to other scantling sizes. The main point here is that the

bottom is the only plate to show a break in tabulated values at the length where frame spacing becomes a constant. This seems a consistentcy which points out inconsistentcies.

E. Bottom Longitudinals

The Tentative Rules do not consider requirements for bottom longitudinals; therefore we will be considering Section 28 of the older Rules (7) in this analysis; however, we will extend to 1000 foot ship lengths. In these Rules the size of longitudinals is given in Table 5 which has entering arguments of web spacing and a number, N.

Where:

s = Spacing of longitudinals in feet.

l = Web frame spacing.

h = Distance in feet from the longitudinal to D + 4 in vessels of 200 feet length and under to D + 8 in vessels 400 feet length and above. At intermediate lengths h is to be taken to intermediate heights above D.

d = A constant equal to 1.40.

This method of tabulation can be quickly analyzed.

If we assume that the longitudinal acts as a simple beam to support the hydrostatic load we can proceed. The total distributed load over the span will be s l h. Assuming the load to be distributed evenly over the span, and assuming the beam to be of uniform section throughout the span, and assuming a constant

allowable stress due to girder bending, we find the required section modulus to be proportional to $s\ h\ l^2$. Since the Rules (7) define a number $N = c\ s\ h$ we can immediately see that the section modulus is proportional to $N\ l^2$. Thus, once the values of N and l are determined, the section modulus of the required beam is uniquely determined. With the foregoing information, and including any desired wastage allowances, the ABS compiled Table 5 (7).

Antoniou (9) made a rather extensive investigation of this problem in which, among other matters, he considered the effect of combining the stresses due to beam loading and the action of the longitudinal as part of the ship girder. This seemed a reasonable approach and yielded acceptable results. One rather cogent objection can be raised to his work in that the head of water used was never that given as h in the formula for N.

Our investigation was based on using the head of water, h, as defined in the formula for N. Since the actual stress in the longitudinal is a combination of ship's bending stress and girder bending stress (p_1 and p_2), and since there is little known about the actual value of ship's bending stress, we feel justified in assuming that longitudinal girders are designed to receive a constant bending stress due to hydrostatic load regardless of ship length. Admittedly, this designed stress must be small since total stress in the longitudinal will have limits imposed by the yield point of material.

For bottom longitudinal analysis it seems reasonable to assume 100 percent "clamped" end conditions. Thus:

$$M = w\, l^2 \quad \text{(in-lb)} \quad \ldots\ldots\ldots\ldots\ldots \tag{40}$$

where

$$w = \rho\, h\, s \quad \text{(lb/ft)}$$

Since

$$N = c\, h\, s$$

we can solve for M and find

$$M = k\, N\, l^2 \quad \text{(in-lb)}$$

and

$$Z = k_1 N\, l^2 = k_1 c\, h\, s\, l^2 \quad (\text{in}^3)$$

Combining formulas, letting $\rho = 64\ \text{lb/ft}^3$, and solving for p_2 we find the expression:

$$p_2 = \frac{46.7}{k_1}\ \text{lb/in}^2 \quad \ldots\ldots\ldots\ldots\ldots \tag{41}$$

Our assumptions are now justified if we are able to find that the value of k_1, when changing from one size girder to the next larger in any column in Table 5, is constant. We find, when examining Table 5, that the same girder is often used for two or three lengths and will still retain the same N value. In any such group, the girder that has the least length will have the smallest stress and the girder with the most length will have the largest stress. The table was analyzed and we find that maximum-stressed girders have a k_1 that has a minimum of 35.5 and minimum stressed girders have a k_1 whose maximum is 51.6. The k_1 values for each size girder is relatively constant considering the modulus change with girder variance. One will find, however, that the maximum allowed stress is dependent on girder size.

Thus, the maximum stress obtained was found in the largest girder and we calculated a smaller maximum for the smallest girder. The maximum allowed stresses for the largest and smallest girders are as follows:

18" x 4.00" x 0.500" x 0.625"	13,150 psi
6" x 2.94" x 0.313" x 0.475"	11,800 psi

The fact that stresses in individual girders do not vary with N, and therefore ship length, is a positive indication of corrosion allowance. In this analysis, due to time considerations, we have examined only the channels listed in Table 5 and we have based the section modulus on a complete channel, disregarding plate effectiveness.

We have checked the reasonableness of our results by adding to the value obtained for p_2 the value of Tobin's stress at 550 feet which is the maximum length covered in the Rules we are working with. This combination stress is approximately equal to the material's yield point although one could account for the smaller hogging moment and have a safety factor. To be consistent, when extending Table 5 for ships greater than 550 feet in length, we feel the value of p_2 should be reduced in accordance with the increase of p_1.

[illegible]

[illegible]

[illegible]

[illegible]

[illegible]

III. RESULTS

A. Section Modulus Analysis

By specifying the section moduli for plating and longitudinals ABS has imposed various restraints on design if a minimum of material is used. Among the implicitly defined requirements we find:

1. Neutral axis location.
2. The percent of total modulus in plating and longitudinals.
3. A restriction on d/L if moduli are within previously specified values.

The analysis showed clearly that a major change of bending moment formula has been accepted by ABS. The formula

$$M = a\,B\,d\,L^2 + b\,B\,L^3$$

is now used and accounts for still water and wave moments respectively. Individual moments are equal when d/L equals 0.055. The relative importance of wave moments will increase with length in ships of normal dimensions.

Although the results are not conclusive we feel that the assumed stress is neither Abell's nor Tobin's but is rather a modification of Tobin's. At short lengths, less than 550 feet, we find a positive increase in modulus from that required to satisfy Tobin's stress as a criterion. At lengths above 550 feet modulus requirements decrease slightly from those forecast.

The decrease of modulus at long lengths is not caused by a constant corrosion allowance.

Stresses calculated by the formula M/Z show allowable bending stress to roughly parallel Abell's at lengths above 550 feet.

B. Deck Plating

When undertaking the analysis of plating we are actually analyzing the ratio of plating thickness to frame spacing since neither buckling stress nor hydrostatic stress is dependent on plating thickness alone. Assuming the Rule value for frame spacing, deck thickness was derived by ABS using Montgomerie's expression as a criterion. Montgomerie's critical stresses plot piecewise linear, with ship length, and are always greater than our calculated Rule stresses. (Figure X)

The deck plating must also do its share to satisfy plating moduli requirements. We have found that in nearly all cases moduli requirements are critical. Approximate procedure shows that most concern about plating thickness will be had at lengths of 850 to 950 feet. At these lengths, the minimum values listed in the Rules may govern.

C. Side Shell

Review of Abell's paper (1) and the present ABS Rule requirements gives strong indications that the 1913 Load Line Committee's conclusion that side shell thickness should increase linearly with length is still considered valid. The present requirements are quite close to those recommended by the Committee.

These recommendations are based on shear stress as strength criterion.

A more mathematically rigorous derivation based on the same principals as Abell's gives a rather different expression for side shell thickness. In this derivation, it matters not whether we divide the assumed bending moment expression into terms representing still water and wave bending moments or consider only a single expression. Comparison of this formula with tabulated Rule values of plate thickness shows the Rule values to be the more exacting and conservative.

D. Bottom Shell

The criteria of instability and hydrostatic pressure are found to be governing here. At ship lengths up to 400 feet Montgomerie's expression of instability reproduces the tabulated Rule values reasonably well.

For ship length 400 to 760 feet hydrostatic pressure is found to be the governing requirement. In order to duplicate the tabulated values, we extend the approach used on transversely framed ships to our case of longitudinal framing. This is not theoretically correct, but we were unable to match Rule values with the more exact treatment. The allowance for corrosion, or safety factor, must be increased to match values of plate thickness at lengths beyond 550 feet which was the maximum length considered in older ABS Rules. Correction is made for the assumed position of the neutral axis.

From 760 feet length upward, both frame spacing and minimum bottom plating thickness are constants.

E. Bottom Longitudinals

The channels listed in Table 5 of the ABS Rules (7) are listed according to physical dimensions. The section moduli of the tabulated channels can be calculated using the following assumptions:

1. Simple beam theory will hold.
2. The complete load is hydrostatic, is distributed uniformly over the span, and has a hydrostatic head, h, identical with h as defined in the Rules.
3. End fixity is 100% clamped.
4. The allowed girder bending stress is independent of N, and therefore independent of ship length.
5. Channel dimensions have been incremented to allow for wastage. Thus, the lighter beams have smaller maximum stresses.
6. The maximum bending stress is approximately 13,500 psi.

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[illegible]

[illegible]

IV. DISCUSSION OF RESULTS

A. Section Modulus

The new criteria for section moduli have the added advantage of accounting for both wave and still water bending moments. These moments have been set equal when a d/L ratio of 0.055 is used. At longer lengths, when the d/L ratio is most apt to decrease, the relative importance of wave bending moment increases. Section moduli, of course, vary directly as bending moments. As a result of the introduction of this two term formula a designer can now more properly optimize draft since total ship's cost is a function of section modulus and payload is a function of draft, all other things being equal.

Neutral axis location varies piecewise linearly (see Figure I) from 56% below the deck at a length of 250 feet to 50.75% below the deck at 530 feet and above. Section modulus is based on ship's bending stress and since maximum bending stress occurs in the deck we assume that the f_{t1} numbers were derived first, or perhaps gleaned from experience, and all other f numbers were proportioned from these. It obviously becomes a simple matter to calculate f_{b1} once a neutral axis position is assumed. If we let

$$\% = \text{neutral axis position from deck}$$

then we find

$$f_{b1} = \frac{\%}{1 - \%} (f_{t1})$$

The percent modulus in plating has been altered, somewhat arbitrarily, from a former minimum value of 75% to the current

[illegible]

minimum value of 60%. Vedeler (10) gives good reason to guard against putting too much material in longitudinals. He claims an optimum figure of 44% but because of horizontal buckling of girders this may be reduced. Figure II shows the sophisticated manner in which the percent of section modulus in plating is varied and for practical considerations seems reasonable. We cannot help but feel, however, that a more optimized and less arbitrary positioning of material could be accomplished with few additional calculations. We believe that the f_{tp} and the f_{bp} values were derived directly from a linear plot such as shown in Figure II. The following relation was used to calculate f_{tp}.

$$\%Z_p = \frac{f_{tp}}{f_{tl}}$$

Since the d/L values, as shown in Figure VIII, were derived by equating moduli in the 1958 and 1959 Tentative Rules, they tend to indicate a realistic value of d/L ratios. These values are not restrictive however, if one designs with a d/L value less than plotted his modulus requirement will be more than required in 1958. The plot says no more.

As in Tobin's time, the chicken and egg problem still exists. It is not possible for us to determine the method used by ABS in arriving at the f_{tl} numbers. Since all other f numbers have been shown to originate with this number the problem is of importance. We have shown how the f_{tl} varies from the simple $L^{5/3}$ law, but these variations need not be corrections. Tobin's stress, and Abell's, reflect the ratio of an arbitrary bending

moment and empirical section moduli for ships that were constructed prior to 1913. Since technological advances have been many in the last 50 years, and since there has been considerable change in ship's characteristics, would we not be in error if we clung to Tobin's or Abell's stress? Our new stress, calculated exactly as Tobin calculated his, should be more up to date than Abell's or Tobin's. We have assumed here that f_{tl} numbers are based on experience but this is not necessarily so since we have had few, if any, failures of 800-1000 foot tankers. Some extrapolation of f numbers was probably taken.

As we analyzed the f numbers, Tobin's stress is equal to our stress at 550 feet. This is an arbitrary move on our part but we feel that either 550 feet or 600 feet should be used due to the discontinuity in the correction curve, Figure VII. A glance at Figure IV, however, would indicate that there are no discontinuities but it does appear reasonable to assume the common intersection at 550 feet. One must not be mislead however; a plot of $f = 0.2913 \times 10^{-3} L^{5/3}$ (match at 1000 feet) will give a smaller mean square error. Nevertheless the end result is the same in all cases: Rule stresses increase at long lengths faster than Tobin's. Would we not be progressing negatively if the reverse were true?

B. Deck Plating

As usual, ABS has based minimum thicknesses found in Table A on a criterion that is applicable to transversely framed ships only. Although Montgomerie's expression will yield conservative

results, there is no known acceptable reason for basing plating thickness on results obtained from his formula. Figure X shows critical stresses calculated with Bryan's and Montgomerie's expressions. Assuming Bryan's formula more nearly correct, since it has accounted for proper edge fixity and aspect ratio, we can see immediately the fallacy in applying Montgomerie's formula for conservative reasons. The factor of safety over Bryan's formula will change, becoming greater with length. We found this factor to increase from 1.3 at a length of 250 feet to 4.26 at 1000 feet. To discuss the application of Montgomerie's formula further would be redundant. Suffice it to say that the expression has here been applied quite incorrectly.

Since ABS has also specified a required section modulus for plating, and clearly deck plating is a major contributor to the modulus, we feel that often one will be forced to plating thicknesses greater than minimum values simply to fulfill moduli criteria. We have shown what we believe to be the upper and lower boundaries of plating thickness that may be required to fulfill the moduli criteria. Our plot, Figure XI, indicates that plating thickness will in fact be governed normally by section modulus. Another bit of information that one can glean is that minimum deck requirements are most apt to be critical for ships of lengths from 850 to 950 feet.

We feel that the method used by ABS to arrive at minimum plating thickness is not correct. In general, to correctly determine plating thickness one should perform the following operations:

1. Express the critical buckling stress of the deck plating by an acceptable expression. Bryan's formula is quite acceptable in this situation.
2. Find the critical buckling stress of the longitudinals. Euler's formula, well known to us all, is applicable and will give acceptable results in both elastic and plastic regions after suitable correction for E.
3. Equate the buckling stress of the plate to the buckling stress of the longitudinal girder. Recall that girder modulus is based on normal loads only. Here one may wish to choose a longitudinal other than selected from Table 5. Of course, one must not choose a longitudinal that will not withstand required hydrostatic loads.

By following a procedure similar to the above, one is free to adjust longitudinal size, frame spacing, and deck thickness. All equations are interdependent and one must bear in mind that total modulus must always be as required by the Rules.

Although it is not the object here to develop a new set of equations for ship construction we feel that a rather complex and interesting engineering problem has been reduced, by the ABS Rules, to an uninteresting technician's task. We hope to eventually see the naval architect receive more design freedom. However, in the meantime it would be logical to base the plating requirements on something more realistic than Montgomerie's expression.

C. Side Shell

We are not able to say whether the Load Line Committee recommendations on side shell thickness have survived due to reinforcement by experience in plating failure and/or devotion to a standard which has not yet allowed failure but which may be needlessly conservative. Since other Classification Societies have Rules which presumably do not allow failures, it might be very instructive to compare their Rules with ABS requirements. This might at least indicate areas of possible over-conservatism or of substantial agreement.

Investigation of the Rules of other Societies might also help in assessing the applicability of our rigorously derived formula. If something other than shear stress becomes a criterion in some range of length, this might be revealed.

Despite the fact that we are unable to conjure up the proper constants to definitely set the limits on required plate thickness, our analysis has shown us an important point. This is that the simplified analysis assumping p directly proportional to L which we have become accustomed to through venerated and perpetuated usage, can give very different results from a more rigorous derivation. We would not argue that the results gotten from using a simplifying assumption have not been reasonably satisfactory through the years of usage. It is rather our desire that when developing basic theory we might be more precise and later temper the theory by experience factors rather than originally compromising our theoretical development to the end that our answer should more closely match past and existing practice.

D. Bottom Shell

Apparently, when giving requirements for longitudinally framed ships, the ABS draws heavily on experience with transverse framing. Using only the Rules themselves we are able to extend the requirements for bottom plating in transversely framed ships to those for longitudinal framing in ship lengths to 400 feet. These requirements are based on Montgomerie's expression for instability which was emperically derived under a given set of conditions. No allowance is made for actual differences of panel aspect ratio or conditions of edge fixity. Though this is not considered theoretically sound, it seems to be the method of the ABS.

For ships of length 400 to 760 feet the governing criterion is found to be hydrostatic pressure. Again satisfactory reproduction of the Rule requirements is gotten by extension of methods used in the case of transverse framing. Although this is again theoretically incorrect, we feel it is what has been done. Since the ABS is an avowed conservative organization, our finding of an increase in margin of safety seems inconsistent in the matter of extending the requirements beyond the maximum length of 550 feet of the old Rules. We further find that the 1958 Tentative Rules defined an assumed position of the neutral axis which is used to correct the p_1 stress seen in the bottom plating. The 1959 Tentative Rules made a slight change in assumed position of neutral axis, but the plating requirements were not changed. Since the transition in margin of safety can be taken to occur from 530 to 620 feet, the range of different behavior of neutral

axis position, we need not be too concerned which neutral axis correction we use as long as we are consistent.

In the range of ship length 760 to 1000 feet we find the plating thickness constant. In this same exact range we find longitudinal frame spacing constant. This is the only plating to have a break in requirements at the same point as frame spacing. This appears to be more a matter of choice and convenience with the ABS than a matter of strict theory. The constant plate thickness may be justified by numerous arguments of frame spacing, wave heights, bending moments, ship proportions, and the like, but these must all be tempered by the scarcity of supporting scientific data and the acceptance of a measure of conjecture.

E. Bottom Longitudinals

The sizing of bottom longitudinals is complex and challenging. To properly determine longitudinal girder size the architect should consider the following:

1. Bending stresses due to normal hydrostatic loads.
2. Verticle buckling.
3. Horizontal buckling.

As a chain is only as strong as its weakest link, so the ship's longitudinal members should not be a lot stronger, or weaker, than the adjacent plating. Hence, the problem is one of optimization.

When using Table 5 of the ABS Rules we find that longitudinal girders are sized to satisfy but one criterion, that criterion being bending due to hydrostatic loads. Although the Rules make

the selection of longitudinals a simple and straightforward procedure we feel that the designer should nevertheless investigate buckling of the shape specified.

Using our calculated Rule stress values for p_1, we have found that total stress in the outermost fibres of the longitudinal will be less than the yield point of mild steel for ships of 550 foot length or less. Because Table 5 was calculated assuming a constant maximum stress for longitudinals, regardless of ship length, the shorter ships are most certainly penalized. We believe that the allowable girder bending stress should be increased for ships shorter than 550 feet at a rate that equals the decrease in our calculated Rule stresses for these ships.

When extending the Rules to cover ships up to 1000 feet in length the value of allowed girder bending stress should be decreased to allow for the increase of p_1 with ship length. We suggest that our calculated Rule stresses, Figure V, be used since they are not only more recent than Tobin's or Abell's but are also more realistic for our uses since they apply to longitudinally framed tankers.

Maximum girder bending stress, although found to be independent of ship length, is dependent on girder dimensions. The lighter girders have smaller maximum stresses. These two facts are positive indication of a wastage allowance. Our calculated range of maximum stresses was bracketed by the following two channels:

18" x 4.00" x 0.500" x 0.625"	13,150 psi
6" x 2.94" x 0.313" x 0.475"	11,800 psi

If we assume that

$$p_1 + p_2 = p_y = 32{,}000 \text{ psi}$$

and for a 550 foot ship p_1 equals 18,350 psi we might expect the actual p_2 maximum to equal 13,650 psi. This is not inconsistent with our calculations since the 18" girder obviously has some corrosion allowance. Although this formula shows implications of careless design, with a safety factor of 1, we do in fact have an implicit safety factor since the ship will have its most severe hydrostatic load amidships in hogging and the primary bending stress is then only eight-tenths of maximum.

We believe that further work on bottom longitudinals will be needed to calculate the exact corrosion allowance used. Although we have had little time to pursue this problem, we do have positive indication that the allowance is not based on a percentage of section modulus but is rather a function of linear dimensions. We expect, of course, a constant corrosion thickness.

In this analysis we considered channels only. The section modulus used was the modulus of a complete, uncut, channel with no effective plating. Thus, the neutral axis was at channel half depth. We believe this assumption realistic; ABS probably assumes that the plating will exactly duplicate the cut-off flange when attachment is by welding. Further thesis work would be useful on this problem and all shapes should be analyzed. We feel that the method of approach outlined herein will, however, be helpful.

[illegible]

V. CONCLUSIONS

A. Section Modulus

1. The new expression for section modulus

$$Z = f\ B\ (d + .055L)$$

was derived from a two term bending moment formula that accounts for still water and wave moments respectively.

$$M = a\ B\ d\ L^2 + b\ B\ L^3$$

When $d/L = .055$ the moments are equal.

2. If minimum requirements are met, the position of the neutral axis is implicitly specified. Figure I.

3. If minimum requirements are met, the percentage of modulus in plating is implicitly specified. Figure II.

4. The adequacy of the Rules modulus requirements cannot be evaluated analytically since modulus, and therefore f, is based on empirical data. Rule stress, including Abell's and Tobin's, is calculated from an assumed moment and is therefore only as accurate as the assumed moment. One may extrapolate either f or p but such extrapolation may give poor results because of a general lack of knowledge of ship's bending moments.

B. Deck Plating

1. Minimum deck thicknesses as given in Table 12 were calculated using Montgomerie's formula and are therefore not correct.

2. Plating modulus requirements are expected to govern plating thickness generally.

C. Side Shell

1. The Rule requirements are based on shearing stress as the governing criterion.

2. The derivation and standards of the 1913 Load Line Committee are still used as a basis of ABS Rules.

3. A rigorous derivation based on shear stress requirements gives results which show that the Committee's neglect of a constant in order to assume p directly proportional to L introduced a serious theoretical error.

4. If ABS conservatism, based on shear stress criterion, were decreased, minimum plating thickness might then be dictated by another criterion such as instability or hydrostatic load, at least in some range of ship length.

D. Bottom Shell

1. In the range of lengths 250 to 400 feet, instability is the criterion.

2. For lengths 400 to 760 feet, hydrostatic pressure is the criterion.

3. In ships over 760 feet, the stresses experienced by the bottom plating are not expected to increase with length.

4. In extending the Rules beyond the old maximum of 550 feet length, the ABS has added an extra measure of conservatism.

5. Bottom plating requirements in longitudinally framed tankers are really only extensions of requirements for bottom shell in transversely framed ships. The theoretical basis for this extension is poor.

6. Allowance is made for the position of the hull's neutral axis when sizing bottom plate.

E. Bottom Longitudinals

1. The ABS Rules base longitudinal modulus on girder bending stress only.

2. Simple beam theory is used with the following assumptions:

   a. Load is hydrostatic of head h, as defined in the Rules.

   b. The load is uniformly distributed.

   c. The end fixity is 100%.

   d. A maximum p_2 of 13,650 psi is allowed.

3. A corrosion allowance is added to longitudinals. Thus, the maximum uncorroded stress varies from 11,800 to 13,150 psi. Maximum stress allowed in any specific channel is constant however, regardless of N or l (length).

VI. RECOMMENDATIONS

A. Section Modulus

1. Future thesis work on an accurate determination of ship's bending moments is needed. To know accurately the load imposed on a ship would enable architects to more intelligently design hull scantlings.

B. Deck Plating

1. Develop a method to design plating and longitudinals so as to have an optimum weight solution. Plating and longitudinals carry much the same loads and they should therefore be designed to act, and to fail, together.

C. Side Shell

1. Compare ABS Rules and our derivation with Rules of other Classification Societies. This may point out discrepancies, inadequacies, over-conservatism, and substantial agreement. It may also set reasonable constants for the derivation.

D. Bottom Shell

1. Re-examine entire length range with both instability and hydrostatic pressure as criteria using methods strictly applicable to longitudinal framing.

2. In both investigations, choose formulations which allow for variations of panel aspect ratio and conditions of edge fixity.

3. Attempt to fix constants in the new formulas and fix degree of conservatism accepted by comparison of requirements of several Classification Societies.

E. Bottom Longitudinals

1. It is recommended that thesis work evaluating actual Rule corrosion allowances be undertaken. This work should include all shapes listed in Table 5 of the ABS Rules.

2. A table similar to Table 5, but correcting the allowable stress due to hydrostatic head to account for primary ship's bending stress, should be compiled.

VII. APPENDIX

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Abbreviations

INA	Institution of Naval Architects
NECIES	North-East Coast Institution of Engineers and Shipbuilders
SNAME	The Society of Naval Architects and Marine Engineers