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Design Criteria for Steel Transmission Poles

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by

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INTRODUCTION

The use of steel poles for high-voltage electrical transmission lines has increased rapidly during the last ten years. The primary reason is an esthetic one, since steel-pole lines cost more than those supported by lattice towers. The large number of users, designers, and manufacturers involved in meeting the increased demand for these structures has created a need for a guide to their design, manufacture, and erection. To this end, a task committee was appointed in 1968 under the ASCE Structural Division's Administrative Committee on Analysis and Design. The Committee submitted its 120-page report at the ASCE National Structural Engineering Meeting in San Francisco in April 1973.[1]

Steel transmission poles are usually unguyed cantilevers, although dead-end or intermediate anchor poles may be guyed. Two-pole bents are used in some cases. The principal loads arise during erection and, in the completed line, from wind and/or ice. Minimum requirements are specified by the National Electric Safety Code. It is industry practice to use load-factor design. The load factors are called overload factors and the pole is designed on a yield-stress basis for various overload combinations.

The principal load on a guyed pole as it occurs in transmission lines is axial compression. The self-supported or unguyed pole is subjected

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primarily to bending. Both circular and polygonal cross sections are used and because of its height the pole is usually tapered. Efficient resistance to bending requires a pole with a large diameter and relatively thin walls, so that local buckling is an important consideration. High-strength steels are usually found to be economical, and steels with a yield stress of 65 ksi are used extensively.

PRIMARY BUCKLING

Since the steel transmission pole is usually tapered its primary buckling load cannot be determined by formulas such as those of the AISC[2] and AISI[3] specifications, which apply to prismatic members. The critical load for such cases can be determined by numerical methods.[4,5] Gere[6] has developed formulas for tapered members. His solution is given in the form:

$$P_{cr} = P^* \frac{\pi^2 E I_o}{L^2} \qquad (1)$$

where P_{cr} = critical load

E = modulus of elasticity

I_o = moment of inertia of cross section at small end

L = length of member

P^* = coefficient which depends on shape of cross section, taper, and boundary conditions

Plots of P^* are given in Ref. 6. However, Eq. 1 holds only for elastic buckling and it is probably unsafe to use it for cases where the predicted buckling stress is close to the yield stress. This is because a lower proportional limit can be expected due to residual stresses and other imperfections, as is the case for prismatic members. An allowance for this effect can be made by using an equivalent radius of gyration in column buckling formulas for prismatic members. The equivalent slenderness ratio $(KL/r)_{eq}$ is found by equating the critical stress according to Eq. 1 to the Euler critical stress (Eq. 3b):

$$P^* \frac{\pi^2 E I_o}{L^2 A_o} = \frac{\pi^2 E}{(KL/r)_{eq}^{\;2}}$$

from which

$$\left(\frac{KL}{r}\right)_{eq} = \frac{1}{\sqrt{P^*}} \frac{L}{r_o} \qquad (2)$$

where A_o = cross-sectional area at small end

K = effective length coefficient

r_o = radius of gyration at small end

The ASCE report suggests that this equivalent radius of gyration be used in the following formulas, which are also to be used for prismatic members,

$$F_a = F_y\left[1 - \frac{1}{2}\left(\frac{KL/r}{C_c}\right)^2\right] \qquad 0 < \frac{KL}{r} < C_c \qquad (3a)$$

$$F_a = \frac{\pi^2 E}{(KL/r)^2} \qquad C_c < \frac{KL}{r} \qquad (3b)$$

where F_a = critical stress

F_y = yield stress

$$C_c = \pi\sqrt{2E/F_y}$$

Equation 3a is the Column Research Council formula for inelastic buckling. This procedure will give conservative results for tapered columns in the inelastic range of buckling because all cross sections do not become inelastic simultaneously. Instead, the small end where the axial stress is largest is the first to become inelastic. If the member does not buckle at this load adjoining sections become successively inelastic as the load increases, until a buckling load is reached. This may occur when only a portion of the length of the column is inelastic, but Eq. 3a is based on the assumption that the entire length is inelastic and that the corresponding tangent modulus determines the buckling load. It is to be noted that the critical stress determined by Eqs. 2 and 3 is to be multiplied by the cross-sectional area at the small end to obtain the buckling load.

The critical load for inelastic buckling of tapered members can also be determined by numerical analysis. However, there is a complication that does not exist for prismatic members, namely, the variation in tangent modulus mentioned above. Therefore, the solution must begin with an estimate of the buckling load in order to determine the stress and corresponding tangent modulus at each node. Equations 2 and 3 can be used to obtain a first approximation to the buckling load.

BENDING

Since the transmission pole and its arms are usually closed sections analysis for bending can usually be made without regard to support against buckling out of the principal plane of bending. This is because the superior torsional stiffness of the closed section makes it highly resistant to lateral-torsional buckling. However, bent-type structures are sometimes made with standard rolled shapes, for which the possibility of lateral-torsional buckling must be considered. The simplest way to handle this problem is to use an equivalent radius of gyration in a column formula. The critical moment $M_{x,cr}$ for doubly symmetrical beams bent about the strong (x) axis and acted upon by end moments and/or transverse loads acting through the shear center is given by[7]

$$M_{x,cr}^2 = \frac{1}{C_m^2}\left(\frac{\pi^2}{(KL)^2} EI_y GJ + \frac{\pi^4}{(KL)^4} EI_y EC_w\right) \qquad (4)$$

where
G = modulus of elasticity in shear
I_y = moment of inertia about weak (y) axis
J = torsion constant
C_w = warping constant
L = distance between points of lateral support
K = effective-length coefficient which depends on y-axis rotation restraint at points of lateral support. $K = 1$ if supports are rotationally free, 0.5 if they are completely restrained, etc., as for columns
C_m = coefficient which depends on variation in moment along the member

The critical bending stress is $f_{cr} = M_{x,cr}/S_x$, where S_x = x-axis section modulus. Equating this critical stress to the Euler stress (Eq. 3b) and solving for r gives the following equation for r_{eq}, the equivalent radius of gyration[7]

$$r_{eq}^{2} = \frac{1}{C_m} \frac{\sqrt{I_y}}{S_x} \sqrt{C_w + 0.04J(KL)^2} \tag{5}$$

The coefficient C_m may be taken equal to unity except that for members acted upon by end moments M_1 and M_2, and with no intermediate loads, the less restrictive value given by

$$C_m = 0.6 + 0.4 \frac{M_1}{M_2} \geq 0.4 \tag{6}$$

may be used, where $M_1 \leq M_2$ and M_1/M_2 is positive if the member is bent in single curvature. Values of C_m for other loadings are given in Refs. 7 and 8.

The critical stress for lateral-torsional buckling is found by substituting KL/r_{eq} into the appropriate column formula (Eqs. 3).

To illustrate the favorable lateral-torsional buckling resistance of closed sections, assume a box beam 6 in. wide by 30 in. deep by 20 ft long, which is an extremely slender member, its depth-width ratio being 5 and its length-width ratio 40. The equivalent slenderness ratio (L/r_{eq}) is 24 and the critical stress according to Eq. 3a is only eight percent below the yield stress.

MEMBERS IN BENDING AND AXIAL COMPRESSION

Members of transmission-pole structures are usually subject to both bending and axial compression, that is, they are beam-columns. In the case of prismatic members bent about the strong axis of the cross section and supported against lateral-torsional buckling, good predictions of beam-column strength are given by the following equations:

$$f_a + f_b \leq F_y \tag{7a}$$

$$\frac{f_a}{F_a} + \frac{f_b}{F_y} \frac{C_m}{1 - f_a/F_E} \leq 1 \tag{7b}$$

where f_a = P/A

f_b = M/S_x

F_a = allowable axial compression in the plane of bending, according to Eqs. 3

F_E = Euler stress in the plane of bending

Formulas of this type are used in both the AISC and the AISI specifications. Equation 7a applies at points of support in the plane of bending (usually the ends of the member) while Eq. 7b checks for maximum stress at points between supports. If there is no support against lateral-torsional buckling Eq. 7b is replaced by

$$\frac{f_a}{F_a} + \frac{f_b}{F_b} \frac{1}{1 - f_a/F_E} \leq 1 \tag{7c}$$

F_a in this equation must be based on weak-axis buckling since the member is free to bend laterally, and F_b must be based on Eqs. 3 using the equivalent radius of gyration according to Eq. 5. C_m does not appear in this equation because it is in Eq. 5.

The strength of tapered beam-columns should be determined by a numerical analysis in which the secondary moment Py, where P is the axial load and y the deflection, is added to the beam moment due to the transverse loads and/or end moments in computing the bending stress f_b. The value of y should include the effect of P. In most cases this effect can be determined with sufficient accuracy by multiplying the deflections due to the beam moments by the amplification factor of Eq. 7b, i.e., by $1/(1 - f/F_E) = 1/(1-P/P_E)$. However, if P/P_E is large, a second cycle of computations of y should be made, based on the sums of the moments $Py/(1 - P/P_E)$ and the beam moments. Several cycles of computation may be needed to achieve the desired accuracy in some cases. The criterion for acceptance is $f = P/A + M/S_x \leq F_y$. Equation 7b is not used since the amplification factor in the second term of that equation has already been applied. Enough points must be checked to make sure that the maximum stress has been found unless the location of the section of maximum stress is self-evident. This procedure must be used with caution if the member is one which may fail by lateral-torsional buckling, since such a member may become unstable at loads less than those which produce yield stress at the most highly stressed cross section. Therefore, the numerical analysis for such cases must be designed to detect lateral-torsional instability.

LOCAL BUCKLING OF CIRCULAR CYLINDERS

The local buckling strengths of axially compressed circular tubes has been reviewed in a paper by Schilling[9] for tubes of moderate length which buckle at a stress equal to or less than the proportional limit. The critical stress is proportional to the parameter Et/R where E = modulus of elasticity, t = thickness of tube, and R = radius of tube. Figure 1, which is adapted from Fig. 4 of Ref. 9, is a non-dimensional plot of a number of test results covering a wide range of yield stress and slenderness and representing the work of three different investigators. The specimens in these tests are classified as manufactured tubes. These are defined in Ref. 9 as "tubes produced by piercing, forming and welding, cupping, extruding, or other methods in a plant devoted specificially to the production of tubes, as distinguished from tubes fabricated from plates in an ordinary fabricating shop." The reason for the distinction is the fact that the local-buckling strength of fabricated tubes may be considerably below that of manufactured tubes. This appears to result from larger imperfections in the geometry of fabricated tubes. The considerable scatter of test results over the entire range of slenderness in Fig. 1 demonstrates the imperfection sensitivity of tubes in axial compression even when they are produced under careful control. Tests on a series of fabricated tubes gave critical stresses ranging from only 40 to 80 percent of yield in the slenderness region 0.14 to 0.22. These tests are not shown in Fig. 1 since they tend to obscure the fact that the Plantema formula[10] which is plotted in the figure is a good lower bound for manufactured tubes. This formula was published in 1946 and has been adopted in the AISC guide in the form

$$F_a = F_y \qquad \frac{D}{t} \geq \frac{3,800}{F_y} \tag{8a}$$

$$F_a = 0.75F_y + \frac{950}{D/t} \qquad \frac{3,800}{F_y} \leq \frac{D}{t} \leq \frac{12,000}{F_y} \tag{8b}$$

where D = diameter of tube

Round tubes in bending tend to be more resistant to local buckling. Results of tests are shown in Fig. 2. Those from Ref. 9 were conducted at the U. S. Steel Laboratories to determine the slenderness limit which enables a tube to develop the plastic moment. The results, which are given in Ref. 9 as the ratio of ultimate moment to plastic moment, are shown in Fig. 2 as the ratio of ultimate moment to yield moment since the latter is taken as the limiting moment of resistance in the design of steel transmission poles. The tests from Ref. 11 were made by the Union Metal Company of Canton, Ohio. Material yield stresses in these tests ranged from 39 to 70 ksi. It is of interest to note how well the two sets of tests fall in line. A conservative lower bound to these test results is given by the Plantema-type formula shown in the figure, which reduces to

$$F_a = F_y \qquad \frac{D}{t} \geq \frac{6,000}{F_y} \tag{9a}$$

$$F_a = 0.70F_y + \frac{1,800}{D/t} \qquad \frac{6,000}{F_y} \leq \frac{D}{t} \leq \frac{12,000}{F_y} \tag{9b}$$

It will be noted that the bending tests show much less scatter than the tests in axial compression. This is probably due to the fact that although geometric imperfections tend to be local in nature and more or less randomly located they reduce uniform axial compressive strength wherever they are but affect bending strength only when they happen to be in the compression zone in the region of maximum moment. The difference in scatter also suggests that the formula which the Committee recommends for round tubes in bending may be more conservative than it need be to give the same reliability as the

Plantema formula for tubes in axial compression. In view of the limited number of test results, however, the Committee thought it best to be conservative.

The Committee did not find any tests on round tubes under combined bending and axial compression. However, a linear interaction formula is known to be conservative[12] and the following equation is proposed for the combined effect.

$$\frac{f_a}{F_a} + \frac{f_b}{F_b} \leq 1 \qquad (10)$$

LOCAL BUCKLING OF POLYGONAL TUBES

Local buckling of flat-plate elements has been the subject of extensive experimental and theoretical investigation and limiting values of the width-thickness ratio to enable such plates to reach yield stress in uniform compression without buckling are given in standard specifications such as those of the AISC and the AISI. The limiting values for plates supported on both unloaded edges are about the same in both these specifications but are expressed in terms of the yield stress in the AISC Specification and the computed service-load stress in the AISI Specification. For example, the limits for flanges of square or rectangular cross sections of uniform thickness are $238/\sqrt{F_y}$ for AISC and $184/\sqrt{f_a}$ for AISI. With $f_a = 0.6F_y$, the maximum allowable value of f_a, the AISI limit becomes $238/\sqrt{F_y}$. The ASCE Task Committee had no reason or evidence to change this limiting value for uniformly compressed members but did round off the numerator. Thus, the slenderness limit of polygonal tubular members is given as

$$\frac{w}{t} \leq \frac{240}{\sqrt{F_y}} \tag{11}$$

where w is defined as the flat width of the side except that the inside bend radius is not to be taken larger than 4t in calculating w.

Since the circular cylinder is more resistant to local buckling when it is subjected to bending instead of axial compression Eq. 11 can be expected to be conservative for polygonal poles subjected to bending. The uniform compression of equal-sided polygonal members produces simultaneous buckling of all sides, but when bending is involved the most highly compressed side receives some rotational edge restraint from the adjacent sides which are under a stress gradient. Figure 3, which is adapted from Fig. 5 of Ref. 13, shows the results of tests by Meyer Industries on poles of 4, 6, 8 and 12 sides. The poles were tested in bending with a transverse load applied at the end. Yield stresses (mill values) ranged from 50.7 to 72.8 ksi. Corner radii were not reported, so the plot is based on w equal to the corner-to-corner width. According to these tests, the limiting slenderness is about $320/\sqrt{F_y}$ based on corner-to-corner width.

Figure 4 shows the results of tests by A. B. Chance Company.[14] These tests were also in cantilever bending. There were five 8-sided and five 12-sided specimens. Yield stresses ranged from 67.6 to 83.1 ksi. Flat widths were measured. Bending stresses were calculated at the point of buckle. All but one member buckled at about half the diameter above the base. The exception was a pole that buckled about six feet above the base. Coupon tests for this specimen disclosed a yield stress of only 62.7 ksi at the buckled section. Coupon yield values at the neutral axis of the buckled section were used to evaluate all the test results. The figure suggests that a limit $240/\sqrt{F_y}$ is well on the conservative side for polygonal tubes in bending. Only the two specimens with w/t greater than $350/\sqrt{F_y}$ failed to reach yield stress. However, there are not sufficient data to adequately define a higher limit. Furthermore, the maximum number of sides the polygon may have in order for the rule to apply is not established. It is clear that this is a factor since the edge support of one face by a neighboring face becomes less and less stiff as the number of sides increases, and a point must eventually be reached at which the polygonal cross section behaves more like a circular cylinder, for which local buckling depends on a different parameter. It is of interest to note that in both the Meyer tests and the Chance tests the compression face buckled oppositely (in a radial sense) to the two adjoining faces for specimens up to eight sides, while the buckle in the 12-sided specimens encompassed the compression face and the two adjoining faces, which is to say that the three contiguous faces buckled in the same radial direction. Thus, it appears that the 12-sided pole is at or above the dividing line between polygonal poles that can be evaluated by plate buckling formulas and those that should perhaps be evaluated by round-tube formulas.

The test results shown in Fig. 4 are plotted in Fig. 5 against slenderness based on corner-to-corner widths. This figure suggests a limiting value on the order of $b/t = 370\sqrt{F_y}$ compared to a w/t limit of about $330/\sqrt{F_y}$ according to Fig. 4. However, it is important to note that these larger limits would be applicable only to members in bending with the limit for axially loaded members to be determined by Eq. 4. It appears that additional tests would be worthwhile and that the effects of combined bending and axial compression should be investigated.

Economical proportioning of polygonal cross sections for transmission poles is not likely to involve plate slendernesses much in excess of the limiting value by Eq. 11. If larger slendernesses are used, design can be based on a reduced local-buckling stress or on the post-buckling strength. It is conservative and simpler to use the former procedure and this is suggested in Ref. 1 according to the following formulas:

$$F_a = 1.45F_y(1 - 0.00129\sqrt{F_y}\,\frac{w}{t}) \qquad \frac{240}{\sqrt{F_y}} \leq \frac{w}{t} \leq \frac{385}{\sqrt{F_y}} \qquad (12a)$$

$$F_a = \frac{108{,}000}{(w/t)^2} \qquad \frac{385}{\sqrt{F_y}} \leq \frac{w}{t} \qquad (12b)$$

Equation 12a for the inelastic range intersects the elastic local-buckling curve, Eq. 12b, at a proportional limit of $0.73F_y$ (Fig. 6). Post buckling strength can be taken into account by using the effective-width concept as in the AISI and AISC specifications. However, it should be noted that it may be unconservative in a polygon of, say, 10 or 12 sides to consider only the face in uniform compression to be partially effective since the adjacent faces are also in compression. This compression is not uniform over the face and there are no formulas for determining the effective width of plates under these stress gradients. Of course, it would be safe to treat them as uniformly compressed plates.

SUMMARY

Recent tests show that the bending strength of round tubes in bending as it is limited by local buckling can be predicted with good accuracy by a formula of the same form as the Plantema formula for round tubes in axial compression. These tests show that round tubes can tolerate a larger slenderness (D/t) in bending than in axial compression. However, the increased strength for a given value of D/t does not appear to be as large as has been suggested by earlier work reported in Ref. 9. More tests are needed to establish the reliability of the formula.

The local-buckling resistance of polygonal tubes in bending has also been the subject of recent investigations, and it appears that an increase in the w/t limits allowed by standard specifications is in order where compression due to bending is involved. However, more tests are needed to establish a reliable limit. Furthermore, tests on polygonal tubes in combined bending and axial compression would seem to be needed to establish the effect of the interaction in reducing limiting values for bending.

ACKNOWLEDGEMENTS

The recommendations for the design of steel transmission pole structures which have been reviewed in this paper are the work of the Task Committee on Steel Transmission Poles, Committee on Analysis and Design of Structures, ASCE Structural Division, which presented its report at the National Structural Engineering Meeting in San Francisco in April 1973.[1] This paper discusses only a part of the report, which covers in addition to the material presented here recommendations for design relative to fasteners, splices, possible beneficial effects of cold work, etc., as well as sections on loading, fabrication, load testing, erection, foundations, and quality assurance.

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NOTATION

A = area of cross section

A_o = area of cross section at small end of tapered column

b = corner-to-corner width of side of polygonal member

C_c = $\pi\sqrt{2E/F_y}$

C_m = coefficient in beam-buckling formula

C_w = warping constant

D = diameter of tube

E = modulus of elasticity

F_a = critical axial stress in column

F_b = critical bending stress

F_E = Euler stress

F_y = yield stress

f_a = axial stress P/A

f_b = bending stress M/S_x

G = modulus of elasticity in shear

I_o = moment of inertia at small end of tapered column

I_x, I_y = moment of inertia for x, y principal axes

J = torsion constant

K = effective-length coefficient

L = length; distance between lateral supports

M = bending moment

$M_{x,cr}$ = critical moment for strong (x) axis of beam

P = axial load

P_{cr} = critical axial load

P^* = coefficient for tapered column

P_E = Euler load

R = radius of tube

r_{eq} = equivalent radius of gyration for lateral-torsional buckling

r_o = radius of gyration at small end of tapered column

S_x = x-axis section modulus

t = thickness

w = flat width of side of polygonal member

y = deflection of beam or column

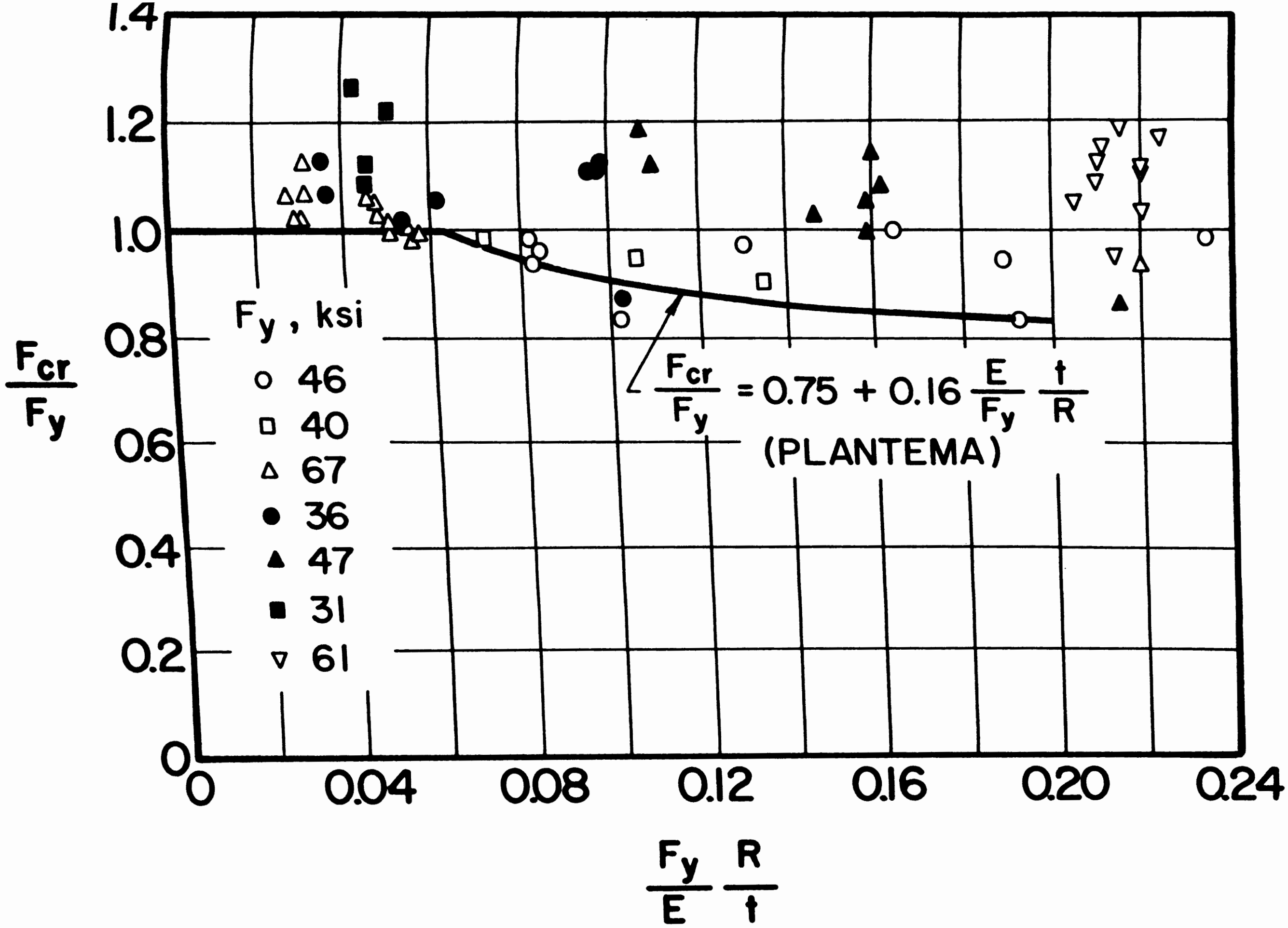

FIG. 1 AXIALLY COMPRESSED ROUND TUBES (Ref. 9)

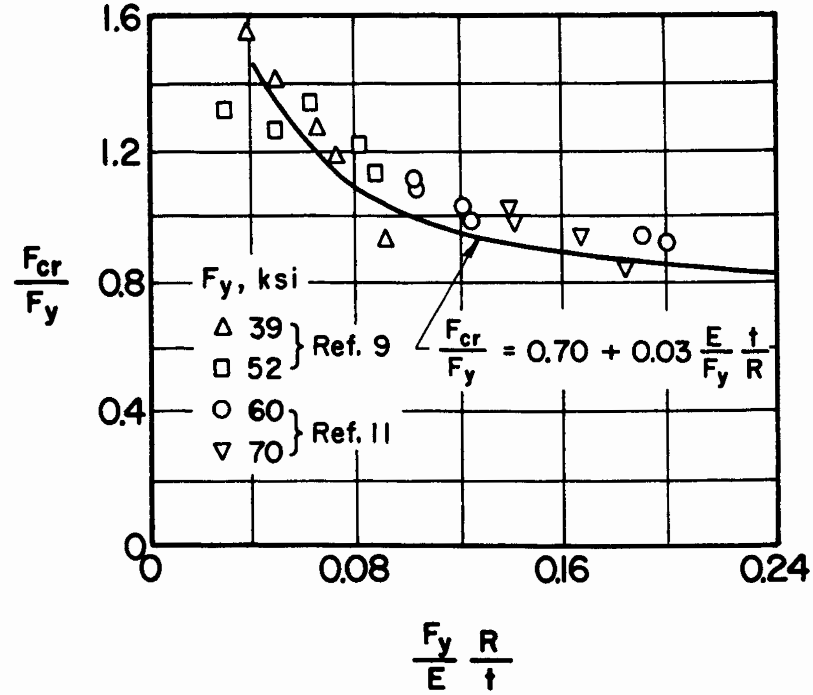

FIG. 2 ROUND TUBES IN BENDING

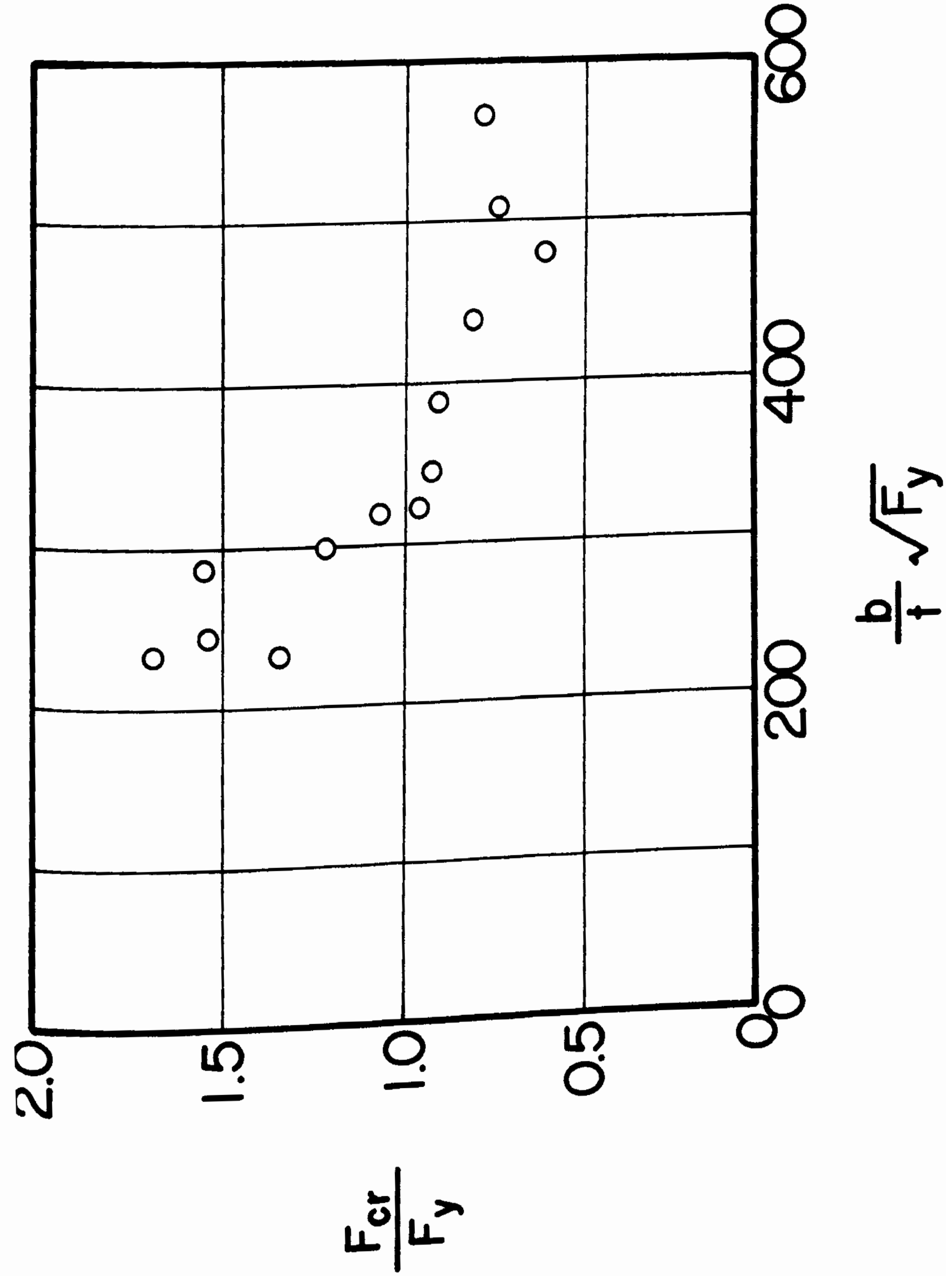

FIG. 3 LOCAL BUCKLING OF POLYGONAL TUBES (Ref. 13)

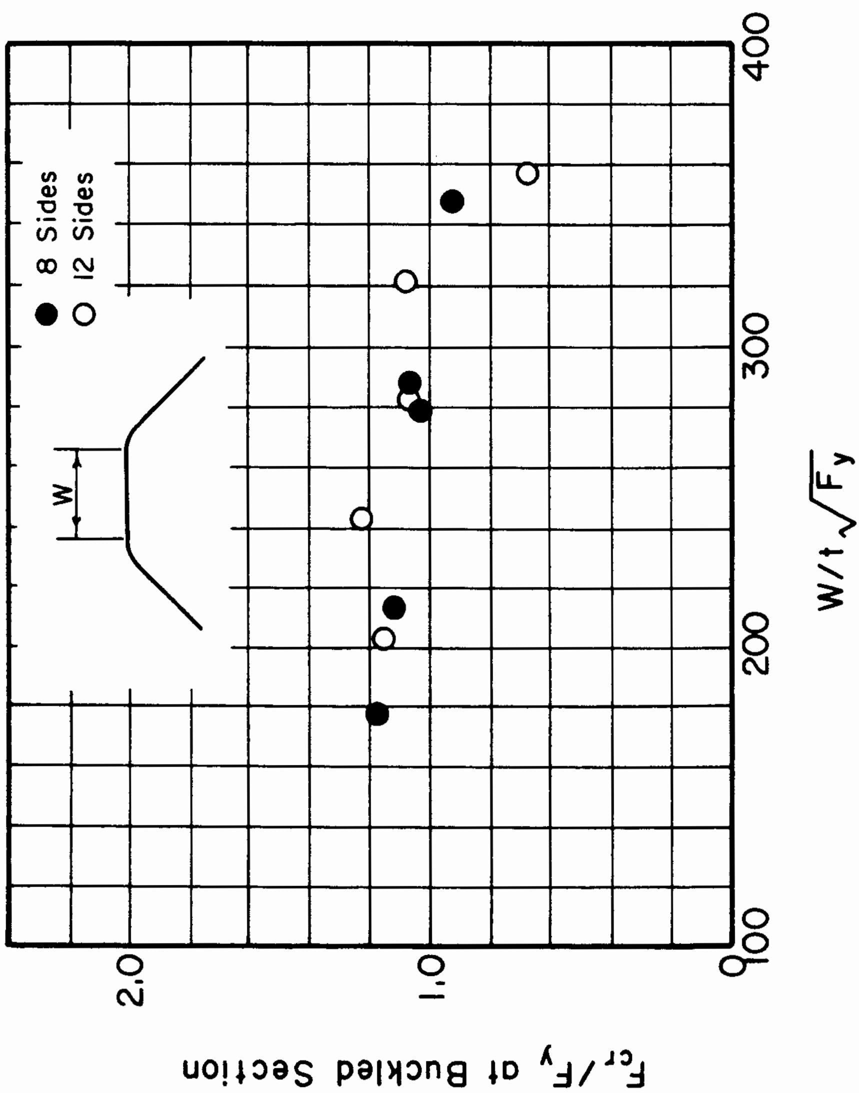

FIG. 4 LOCAL BUCKLING OF POLYGONAL TUBES (Ref. 14)

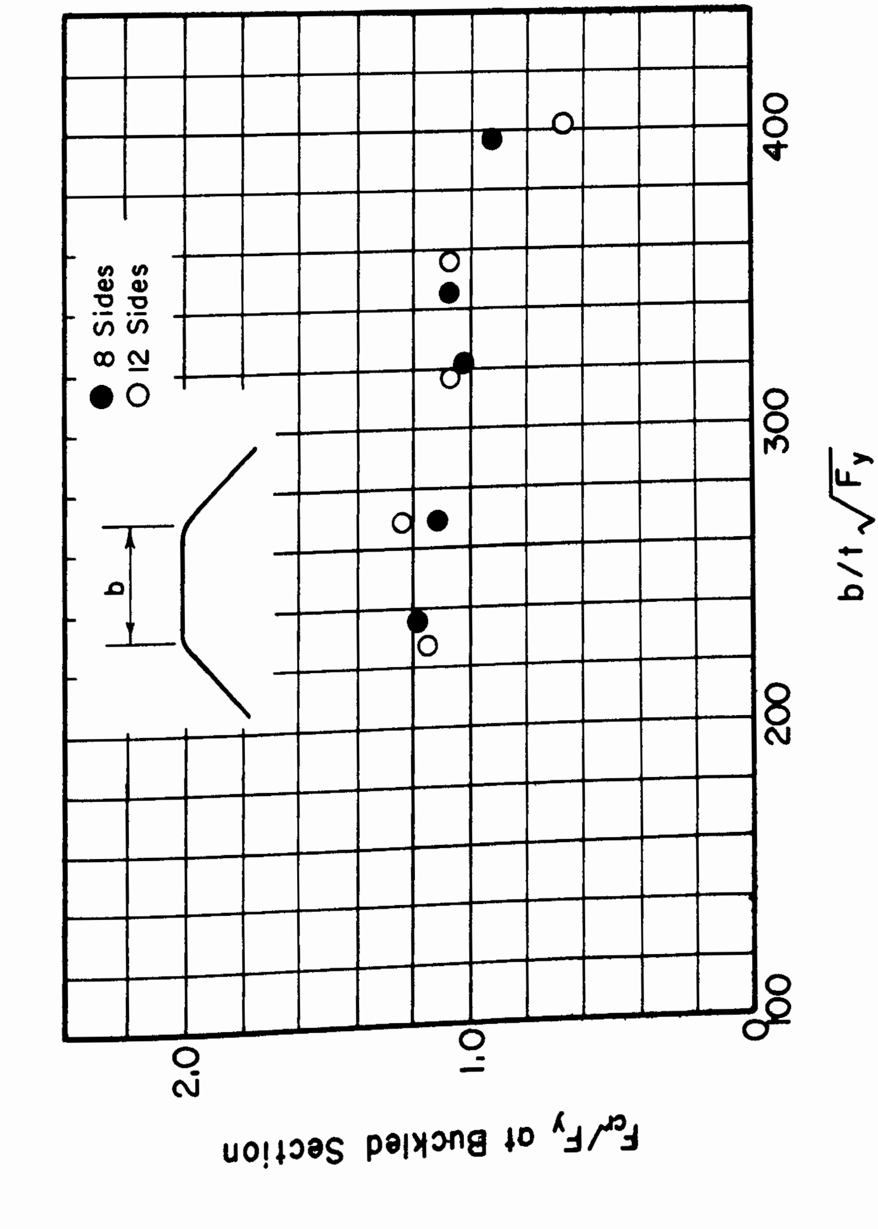

FIG. 5 LOCAL BUCKLING OF POLYGONAL TUBES (Ref. 14)

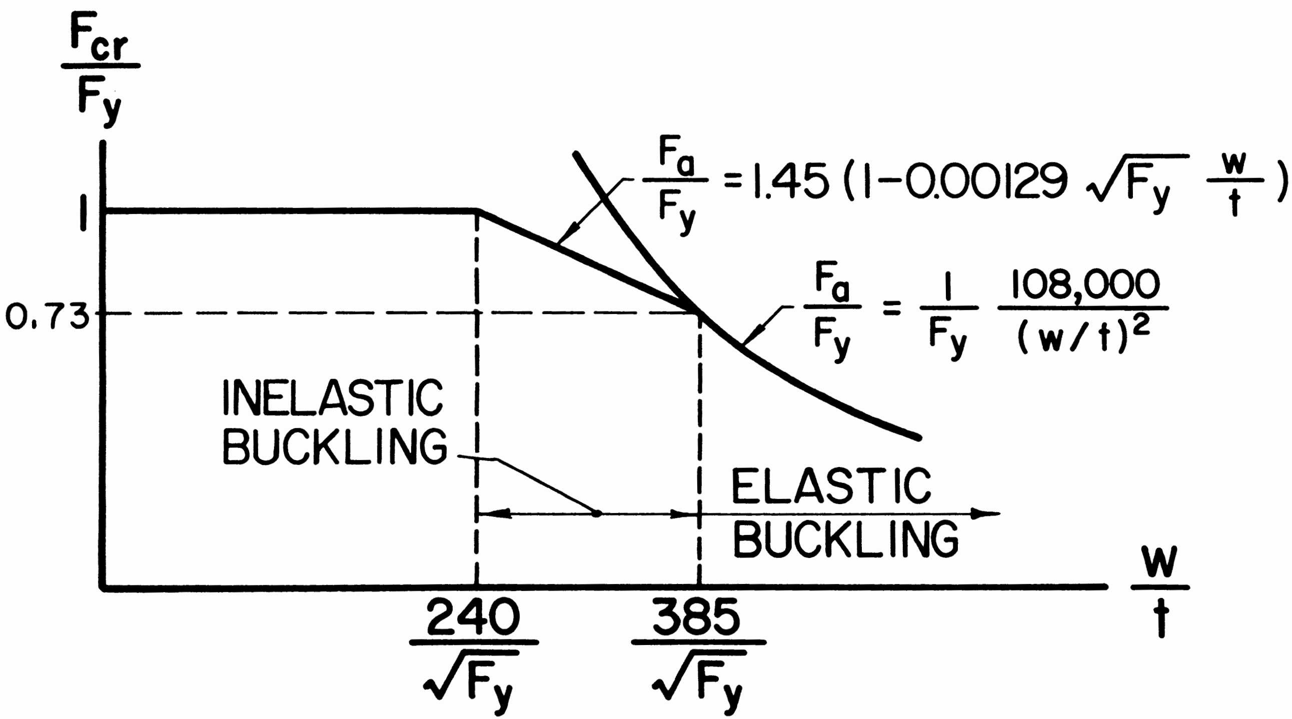

FIG. 6 LOCAL BUCKLING FOR $W/t > 240/\sqrt{F_y}$