lines if sag and attendant pocketing of the particular small lines are not objectionable. Small lines can be assisted across long spans by providing them with intermediate supports attached to adjacent larger lines; a group of such lines may also be tied together so as to become chords of a simple truss. Often, however, the most practical solution is simply to increase the pipe size to the point of being self-supporting over the required span.

In checking the suitability of support spacing for pipe lines on a horizontal run, the nomographs mentioned subsequently in this section are useful for most purposes. For critical-service piping, the flexibility check for expansion stress can be extended to include weight effects where necessary by using methods given in Chapter 5. As discussed previously in this section, general considerations in locating supports are that they be placed at points suitable for the connections to the pipe (no interference with valves, risers, etc.) and to the structure (in respect both to attachment details and to loading requirements).

Allowable spans for horizontal lines are principally influenced by the need to:

1. Keep stresses within suitable limits. (Instability may be a factor in the case of large thin-walled pipe.)
2. Limit deflections (sagging), if necessary for:
   a. appearance,
   b. avoiding pockets,
   c. avoiding interferences.
3. Control natural frequency (usually by limiting the span) so as to avoid undesirable vibration.

In most cases, an adequate estimate of the stress is readily obtained from the simple beam relationship:

$$S = 1.2(wl^2/Z) \qquad (8.1)$$

where S = maximum bending stress, (psi.)
Z = section modulus, in.^{3}
l = pipe span, ft.
w = total unit weight, lb per ft.

For convenience this formula is given in nomographic form in Chart C–16 of Appendix C. It is based on a maximum moment of $M = \frac{1}{10}wl^2$, and represents a compromise between $M = \frac{1}{12}wl^2$ for a beam with fixed ends and $M = \frac{1}{8}wl^2$ for a free-ended beam, as representative of average runs. Values to suit other end conditions can be obtained by the use of the correction factors given in Chart C–18. Overhang at changes of direction may be beneficial from a structural standpoint; if provided in optimum amount, the maximum moment in a line continuous over a series of equal spans can be held to that of fixed-end conditions. However, substantial overhang is best avoided on lines prone to vibration.

Major concentrated loads such as produced by valves, pipe risers, branches, etc., should be at or near a point of support. The effect of significant concentrated loads, not located at supports, may be approximated from eq. 8.1, by multiplying the stress by the factor $2P/wl$ where P is the concentrated load in pounds and other symbols are as previously defined.

Deflection under weight effects is generally of secondary importance in piping just as it is in structures. In fact, some piping designers are inclined to disregard deflection entirely and to consider the limiting weight stress as the only criterion. In most process units, however, the deflection of the line should be kept within reasonable bounds in order to minimize pocketing and to avoid possible interference in congested areas due to sagging. Appearance, too, will be a factor in many cases. A practical limit for average piping in process units is a deflection on the order of $\frac{1}{2}$ in. to 1 in. For piping in yards or for overland transmission lines a value of $1\frac{1}{2}$ in. or greater is generally acceptable. For power piping a deflection limit as small as $\frac{1}{8}$ in. is specified by some designers.

Perhaps the most important reason for limiting deflection is to make the pipe stiff enough, that is, of high enough natural frequency, to avoid large amplitude response under any slight perturbing force. Although Chapter 9 treats this subject more fully, it can be stated here, as a rough rule, that for average piping a natural frequency of 4 cycles per second will be found reasonably satisfactory. For pulsating lines from compressors, etc., values of 8 cycles per second or higher may be desirable depending on the characteristics of the compressor.

The deflection for a given span may be approximated by the beam relation:

$$\delta = 17.1(wl^4/EI) \qquad (8.2)$$

where I = moment of inertia, in.^{4}
l = pipe span, ft.
δ = deflection, in.
E = modulus of elasticity, psi.
w = total unit weight, lb per ft.

Chart C–17 of Appendix C gives a graphical solution for this equation. Similar to the stress formula, it is based on $M = \frac{1}{10}wl^2$; factors for other conditions of constraint are included in Chart C–18.

When lines are pitched to facilitate drainage, the supports may be spaced so as to completely elimi-

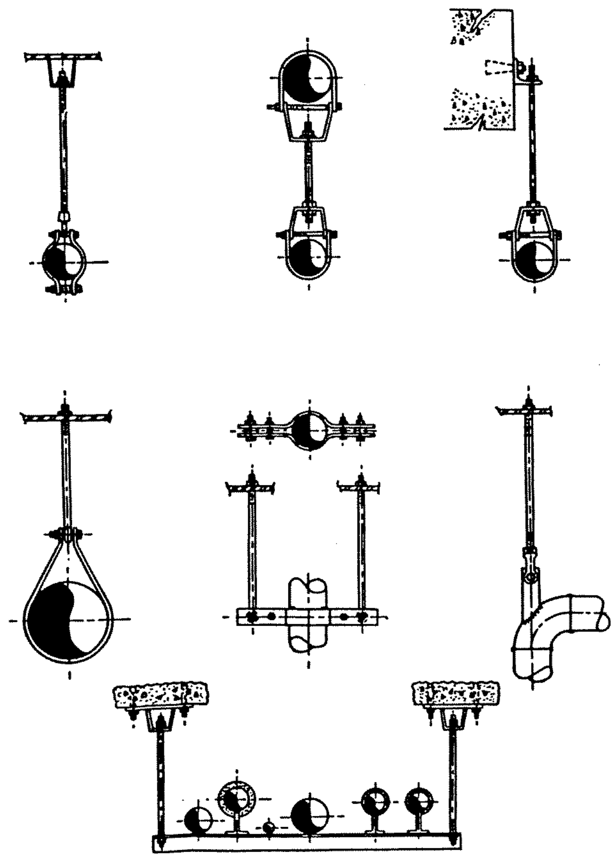


FIG. 8.7 Typical rod hanger assemblies.

nate pocketing due to sag of the piping. Pitching, however, involves considerable added expense of supports and is of limited effectiveness with flowing media which cling in substantial amounts to the pipe wall. Hence, it is becoming a widespread practice to avoid pitching by setting up a regular plant procedure for washing or blowing down the pipe as dictated by safety, corrosion prevention, or contamination requirements. Pitched lines are thus limited to occasional applications where they may be used either generally or in connection with specific pieces of equipment. Where substantial pitch is desired, hanging type supports are generally needed in order to maintain reasonable uniformity in supporting structures.

The minimum pitch of supports required to avoid pocketing due to sag is given by the following formula:

$$h = Kwl^3/EI \qquad (8.3)$$

where h = gradient of supports in feet/100 feet of length, and

K = a constant, depending on constraint.

K = 116 for fixed ends, and

K = 600 for free ends.

Other symbols are as previously defined except that the weight does not include the contents, since the pipe empties as it drains.

The gradient of supports determined by this formula provides that the slope of the deflected line will not be upward in the direction of drainage but will be horizontal or downward. To obtain positive drainage with a given minimum pitch, the support gradient must be further increased by the amount of the minimum pitch. Pitching may also be needed to vent a hot pump suction line back to the source in order to avoid vapor binding.

The advantageous arrangement of support is related to the degree of restraint which can be tolerated, or to the extent and direction of the movements to be allowed at each location. The fundamental types are characterized as ***rigid***, ***resilient***, and ***constant effort***, each of which is capable of wide variation in details and of two basic arrangements, *suspended* and *resting*.

Rigid supports of the suspended arrangement involve solid hangers, while the resting arrangement may function as a sliding contact or be provided with rollers or rockers; for special cases, the support structure may be flexible or of simple- or multiple-hinged design to secure movement in one or two directions, while maintaining constant elevation. Solid hangers eliminate friction and sticking between the pipe and support,[3] but are limited in movement range in proportion to their length, require higher support frames, and involve greater usage of space; however, they are a preferred choice where the general plant arrangement permits their use, particularly on extreme high-temperature or other critical service where unassessable restraint is undesirable. Some typical hanger assemblies are shown in Fig. 8.7. Resting supports, although they involve friction, either sliding or rolling, are widely used and are generally satisfactory, probably due to the friction load resulting from the weight usually being low as compared with the thermal expansion effects; the reduction of friction by using rollers and rockers is not as reliable as by using hangers, due to possible wear and lack of lubrication. Typical resting support assemblies are shown in Fig. 8.8.

Rigid supports are satisfactory for systems involving lengthy horizontal runs with little vertical ex-

[3]It should be noted, however, that freedom of movement renders hangers unsuitable for the support of piping subjected to shock loading, i.e., blow down lines.

SUPPORTS

The basic loading to be considered in the design and spacing of the supports is the *operating* load, which is composed of the weights of the pipe, valves and fittings, the fluid carried, the insulation, and sometimes snow or ice loads. In addition, a check should be made of *test* and *emergency* loadings to assure that these will not lead to failure. Weights of pipe and water contents for use in load calculations will be found in the "Design Properties of Pipe" booklet; weights of Tube Turns fittings are listed opposite each item in the catalog section; weights of other components must be obtained from manufacturers' catalogs and handbooks.[1]

The moments and reactions caused by these loads can be computed by the laws of statics. Where drainage is no factor, the spacing is obtained from stress considerations alone, while for horizontal runs of steam lines, or for other piping which has to be drained completely, the consideration of deflection or sag often assumes controlling influence.

In setting up general formulas based on stress considerations, from which support spacings can be determined for any average horizontal run, it is proper to assume that no more than one-half the allowable *S*-value is absorbed by longitudinal pressure stresses,[2] and it could hence be reasoned that the remaining half could be assigned to bending stresses due to weight loadings. However, in view of possible cumulative effects of the latter and bending stresses caused by thermal expansion, it appears preferable to utilize no more than one-quarter the *S*-value for weight loadings. This has been done in deriving the following formulas for the support spacing *N* (ft), which consider the two normally limiting conditions of end restraint:

For a continuous beam: $$N = \sqrt[2]{\frac{ZS}{4w_t}}$$

For a free span: $$N = \sqrt[2]{\frac{ZS}{6w_t}}$$

Similarly, from a standpoint of drainage, the following limiting support spacings can be established on the basis of maintaining horizontal tangents to the deflection slope (for a given condition where successive supports are set 1 inch lower for every *G* feet horizontal distance between them):

For a continuous beam: $$N = \sqrt[3]{\frac{EI}{13.9\, w_e\, G}}$$

For a free span: $$N = \sqrt[3]{\frac{EI}{72\, w_e\, G}}$$

A free-span condition is rarely, if ever, encountered in piping work, and the assumption of a fully continuous beam (center span of a number of uniform spans) can likewise not be considered representative of average conditions. For this reason, the four charts presented herein, which are intended to serve as a general guide in the selection of reasonable spans for different pipe sizes and materials, types of fluids and temperature conditions, have been based on conditions intermediate between the two extremes.

The spans read from the charts apply directly to standard weight for seamless carbon steel (ASTM A106, Grade A) and to Schedule 10S for 18% chrome 8% nickel stainless steel; for heavier weights, slightly longer spans would be permissible.

The heavy lines on the charts give reasonable support spacings based on stress considerations.

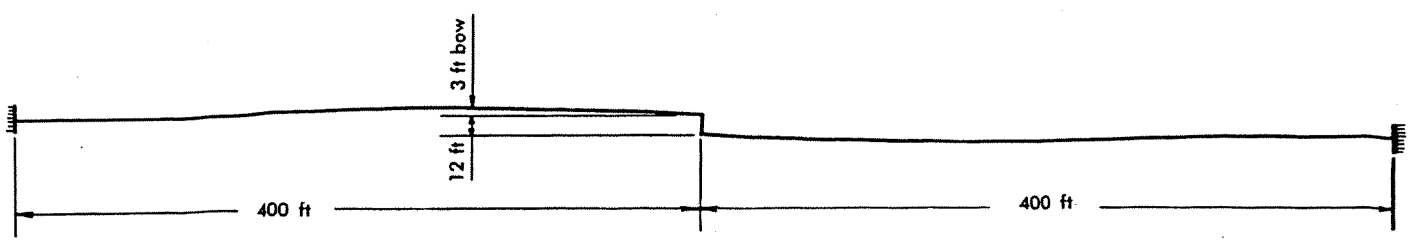
The lighter dash-dot lines give the spans allowing drainage, based on specific gradients; since the slope for drainage becomes important only when the line is nearly empty, the limiting spans based on deflection have been computed for a line which is only 3% full. The selection of a suitable gradient depends upon the speed with which a line is intended to drain, the thoroughness of drainage required, the accuracy of the support line-up, and last, but not least, the extent to which expansion of the line and any attendant yielding or self-springing may alter the position of the line during operation and shutdowns.[3]

It should be noted that the average weight of insulation suitable for the temperature has been included in all computations, but that no allowance is made for the weight of flanges or fittings. Lines with concentrated weights should be considered individually, and heavy fittings, such as valves, should always be supported directly or located close to supports to avoid overstrains or pulsations.

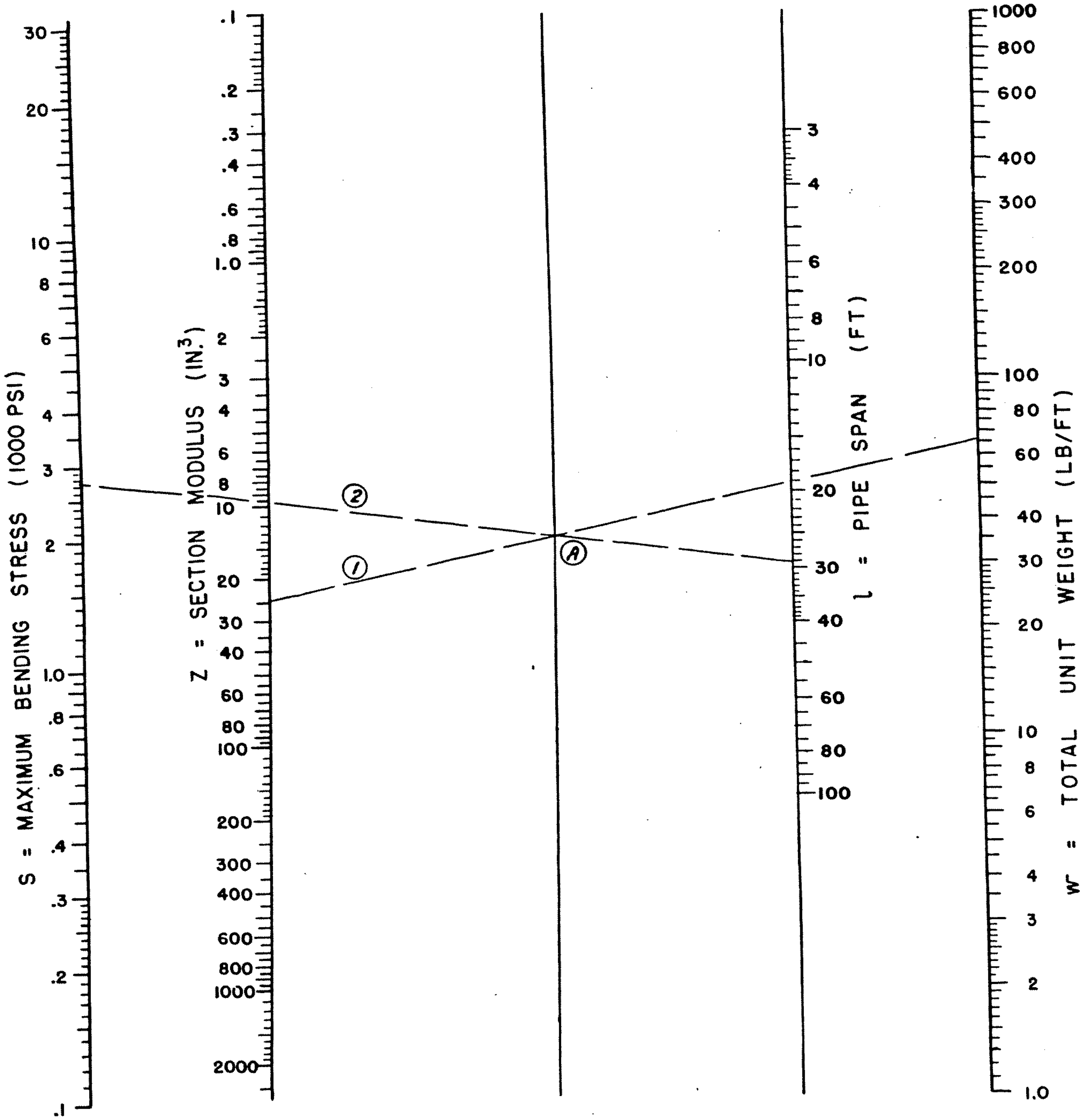
[1]Refer to Crocker's "Piping Handbook," Table III, p. 738 of 1945 Ed., or to "Hanger Load Calculation," published by the Grinnell Co., Providence, R. I., for compilations of weights of all piping components.

[2]The longitudinal pressure stress equals one-half the circumferential pressure stress, and the latter is limited by code to the S-value.

[3]To illustrate that line motion due to expansion may attain unsuspected magnitudes, the displacements at significant locations of the Z-bend shown in the illustration below were calculated on the basis of 1 inch expansion per 100 feet. When it was found that each of the two 400-foot legs would bow out about 3 feet as a result of the resistance of the 12-foot long offset, an expansion loop was recommended for control of motion, even though it was not required from a stress standpoint.



C-16. Span vs. Stress, Horizontal Pipe Lines, Uniform Load

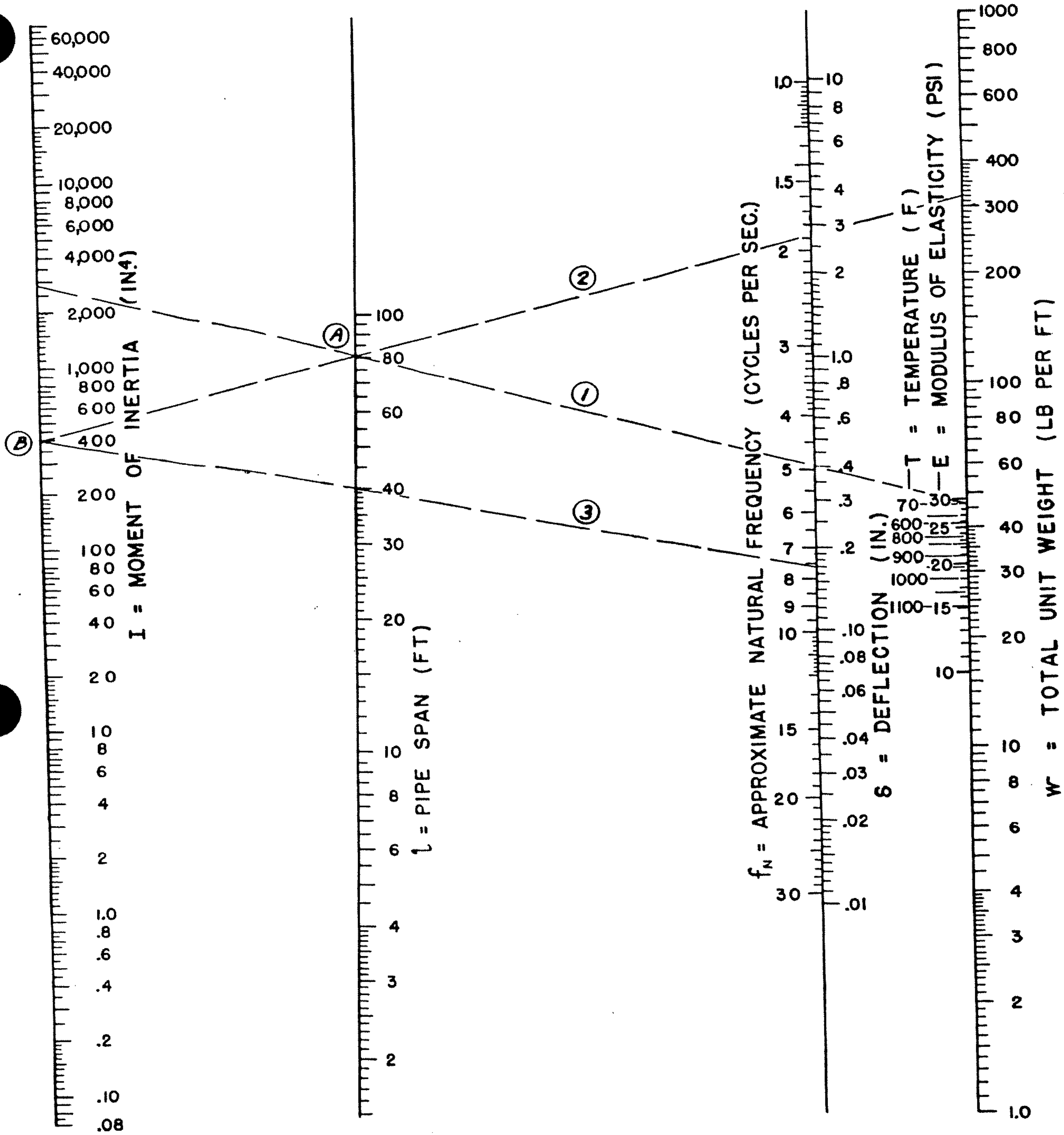


Formula: $S = 1.2wl^2/Z$

Key: (1) Connect Z with w locating turning point (A).
(2) Connect (A) with s locating l.
Connect (A) with l locating s.

Example: Given: $Z = 24.5$ in.^{3}, $w = 66$ lb/ft, $S = 2750$ psi.
Result: $l = 29$ ft.

C-17. Span vs. Natural Frequency and vs. Deflection Horizontal Pipe Lines, Uniform Load

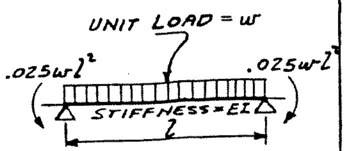
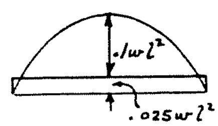
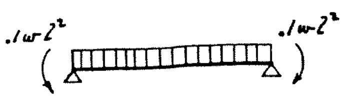
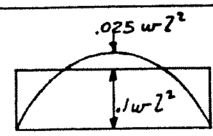
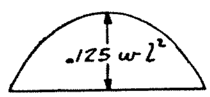
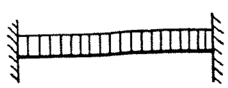
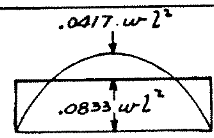
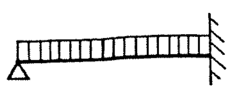
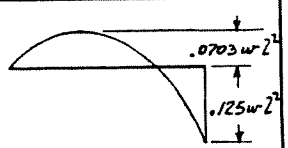
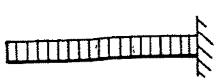
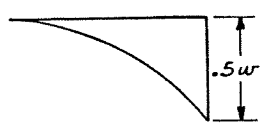


Formulas: $\delta = 17.1wl^4/EI$ $\quad f_n = 3.13/\sqrt{\delta}$

Key: (1) Connect E (or T) with I locating turning point (A) at intersection on l.
(2) Connect (A) with w locating turning point (B) at intersection on I.
(3) Connect (B) with l locating δ (or f_n).
Connect (B) with δ (or f_n) locating l.

Example: Given: $E = 29 \times 10^6$ psi, $I = 2840$ in.^{4}, $w = 320$ lb/ft.
Result: $f_n = 0.76$ cy/sec, $\delta = 0.17$ in.

C-18. Correction Factors for Use with Charts C-16 and C-17

CASE	END FIXATION	MOMENT DIAG.	MAXIMUM MOMENT	MAXIMUM DEFLECTION	F_S	F_δ
1			$\frac{wl^2}{10}$	$\frac{19}{1920}\frac{wl^4}{EI}$	1	1
2			$\frac{wl^2}{10}$	$\frac{1}{1920}\frac{wl^4}{EI}$	1	.0526
3			$\frac{wl^2}{8}$	$\frac{5}{384}\frac{wl^4}{EI}$	1.25	1.316
4			$\frac{wl^2}{12}$	$\frac{1}{384}\frac{wl^4}{EI}$	.833	.263
5			$\frac{wl^2}{8}$	$\frac{1}{185}\frac{wl^4}{EI}$	1.25	.547
6			$\frac{wl^2}{2}$	$\frac{1}{8}\frac{wl^4}{EI}$	5	12.63

For Chart C-16: Multiply S value from chart by F_S to obtain maximum stress for case shown.

For Chart C-17: Multiply δ value from chart by F_δ to obtain maximum deflection for case shown.