UNITED STATES EXPERIMENTAL MODEL BASIN

NAVY YARD, WASHINGTON, D.C.

PROPOSED RULES FOR CONSTRUCTION OF UNFIRED VESSELS SUBJECTED TO EXTERNAL PRESSURE

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RULES for UNFIRED VESSELS SUBJECTED to EXTERNAL PRESSURE

Cylindrical Vessels Subjected to External Pressure

1. The rules for this class of vessels shall apply only to cylindrical vessels of the three types (either with or without stiffening rings) illustrated in Fig. 1, when constructed of steel complying with specifications S-1, S-2, S-4, S-17, S-18, or S-25, and when operated at pressures not in excess of 500 lb. per sq. in. and temperatures not in excess of 700 deg. F. Corrugated shells subjected to external pressure may be used in unfired pressure vessels in accordance with Par. P-243 of the Power Boiler Code.

2. Working pressure shall be the safe operating pressure, which shall be the maximum possible difference in pressure between the outside and inside of the vessel at any time and shall be one-fifth the collapsing pressure.

Shell Thickness

3. The minimum required thickness of the shell plate shall be determined from the chart, Fig. 2.

4. In this diagram the abscissae are L/D, the ordinates are working pressures, and the curves represent different values of t/D, where, as shown in Fig. 3,

L is length of vessel between centers of head seams or between centers of circumferential stiffeners, in.

D is outside diameter, in.

t is minimum required thickness of shell plate, in.

5. In no case shall the external working pressure for which the vessel is designed be taken as less than 15 lb. per sq. in. (corresponding to a collapsing pressure of 75 lb. per sq. in.).

Instructions for Use of Chart

6. To use the chart, Fig. 2, the value of L/D is computed and, with the given working pressure, the corresponding value of t/D is read off. With this value of t/D, the required thickness t is found. When a vessel has an L/D ratio greater than 20, the same t/D ratio shall be used as for a vessel having an L/D ratio of 20.

Example

7. Given: Pressure vessel 12 ft. long between heads, 96 in. outside diameter, external working pressure 15 lb. per sq. in.

Required: Thickness, t.

Solution: L = 12 x 12 = 144 in. D = 96 in. L/D = 144/96 = 1.5

From Fig. 2, for a working pressure of 15 lb. per sq. in. and an L/D ratio of 1.5, the ratio t/D is found to be 0.0046.

Therefore: t = 0.0046 x D = 0.0046 x 96 = 0.44 in.

Out-of-Roundness

8. The out-of-roundness, e, or difference between the maximum and minimum diameters in any plane perpendicular to the longitudinal axis of the vessel, expressed as a fraction of the shell thickness, shall not exceed that given by the chart, Fig. 4, except that on vessels having longitudinal joints of lap construction, the difference between the maximum and minimum diameters may be as great as that given by the chart, Fig. 4, plus the thickness of the plate (See also par. 21). Measurements shall be taken on the completed vessel in a sufficient number of planes to insure that the entire surface of the shell meets the requirements.

Example

9. Given: Pressure vessel considered in previous example.

Required: Maximum out-of-roundness permitted.

Solution: From Fig. 4, for a t/D ratio of 0.0046 and an L/D ratio of 1.5, e is found to be 1.0 t.

This means that the difference between the maximum diameter $D_{max.}$ and the minimum diameter $D_{min.}$ (see Fig. 5) in any plane perpendicular to the longitudinal axis of the vessel, shall not exceed the shell thickness.

Stiffening Rings

10. Circumferential stiffening rings composed of bars or structural shapes secured to the shell of the vessel may be used, in which case the distance L may be considered as the length, measured parallel to the axis of the vessel, between the centers of adjacent stiffening rings, provided the moment of inertia of the rings is not less than that obtained from the chart, Fig. 6.

11. Stiffening rings shall extend completely around the circumference of the vessel. Any joints between the ends or sections of such rings, as shown at C, D, F, and G in Fig. 7, and any connections between adjacent portions of a stiffening ring, lying inside and outside the shell, as shown at H in Fig. 7, shall be so made that the full stiffness of the ring is maintained.

12. Stiffening rings placed on the inside of a vessel may be arranged as shown at A and B in Fig. 7, provided the moment of inertia required by Fig. 6 is maintained within the sections indicated. The moment of inertia of each section shall be taken about its own neutral axis. However, any gap in that portion of a stiffening ring supporting the shell, as shown at A and E in Fig. 7, shall not exceed the length of arc given in Fig. 8 unless additional reinforcement is provided as shown at H in Fig. 7.

13. Particular attention is called to the fact that any arrangement of the structure which does not permit uniform radial contraction of the shell will weaken

the vessel. Internal radial stays or supports for any purpose shall not bear against the shell of the vessel except through the medium of a substantially continuous ring.

Example

14. Given: Pressure vessel 50 ft. long, 200 in. outside diameter working pressure 40 lb. per sq. in.; to be stiffened by circumferential rings.

Required: Thickness, t, best frame spacing L, proportions of stiffening rings, and allowable out-of-roundness.

Solution: If it is desired to use a minimum plate thickness (which will usually be the most economical design), the vessel must be designed with the lowest permissible t/D ratio for the working pressure of 40 lb. per sq. in. From the chart, Fig. 2, it will be found that this t/D ratio is 0.0038, and the corresponding L/D ratio is 0.25. Then, t = 0.0038 x D = 0.0038 x 200 = 0.76 in. and L = 0.25 x D = 0.25 x 200 = 50 in. It should be noted that any further reduction of the L/D ratio below 0.25 does not lower the t/D ratio. Therefore, a spacing of stiffening rings at 0.25 D or 50 in. is the smallest that can be used to advantage.

The moment of inertia of the stiffening ring is obtained from Fig. 6. For L x P = 50 x 40 = 2000, and a diameter D of 200 in., I is found to be 96 in.^4 A 9 in., 30 lb. I-beam is shown in standard handbooks to have a moment of inertia, I, about the neutral axis perpendicular to the web, of 101.4 in.^4. Such a beam is satisfactory. Stiffening rings must be placed every 50 in. along the vessel, and the shell plate must be not less than 0.76 in. thick.

The maximum out-of-roundness permitted is obtained from Fig. 4. For a t/D ratio of 0.0038 and an L/D ratio of 0.25, e is found to be 0.6 t. That is, the difference between the maximum diameter $D_{max.}$ and the minimum diameter $D_{min.}$ in any plane perpendicular to the axis of the vessel shall not exceed 0.6 x 0.76 = 0.46 in. (approximately).

Attachment of Stiffening Rings to Shell

15. Stiffening rings, if used, may be placed on the inside or outside of a vessel and they may be attached to the shell by riveting or welding. If the rings are outside and are riveted, the nominal diameter of the rivets shall be not less than the thickness of the shell plate and the center-to-center distance of the rivet holes shall not exceed that shown in Fig. 9.

16. If the rings are outside and are welded, the total length of the welding on either side of the stiffening ring shall be not less than one-half the outside circumference of the vessel. The arrangement and spacing of such welds, if of the intermittent type, shall be in accordance with Fig. 9.

17. Stiffening rings placed on the inside of a vessel shall be in adequate contact with the shell and sufficiently secured to the shell to hold them in their proper position under any normal condition of operation.

18. All welding for attachment of stiffening rings shall comply with the requirements of the code for the class of vessel in question.

Supports

19. The supports for a vessel shall be such that no concentrated loads are imposed which would cause deformation of the vessel in service exceeding the limits of out-of-roundness permitted by these rules.

> Note: Attention is called to the objection of supporting vessels through the medium of legs or brackets, the arrangement of which may cause concentrated loads to be imposed on the shell. Vertical vessels should be supported through a substantial ring secured to the shell. Horizontal vessels, unless supported at or close to the ends (heads) or at stiffening rings, should be supported through the medium of substantial members extending over at least one-third of the circumference, as shown at K in Fig. 7.
>
> Attention is called also to the danger of imposing highly concentrated loads by the improper support of one vessel on another or by the hanging or supporting of heavy weights directly on the shell of the vessel.

Heads

20. The design of the heads shall comply with the requirements for dished or flat heads as given in this code. Attention is called to the allowable pressure on a dished head when the pressure is on the convex side.

Longitudinal Joints

21. Longitudinal joints may be of any type permitted by these rules, except that if a lap joint is used, either riveted, welded, or brazed, the allowable working pressure shall be 50 per cent of that computed by the rules given herein. Longitudinal joints, if riveted, shall have an efficiency of 50 per cent or greater and in no case less than

$$\frac{P \times D}{S \times 2t}$$

where

P is working pressure, lb. per sq. in.

D is outside diameter, in.

S is working stress lb. per sq. in., given by Table U-3.

Circumferential Joints

22. Circumferential joints may be of any type permitted by these rules. The strength of riveted circumferential joints shall be sufficient, considering all methods of failure, to resist the total longitudinal force acting on the joint with a factor of safety of five.

Nozzle Openings

23. Unreinforced openings in the shell shall conform to the requirements of Par. U-59a where

$$K = \frac{\text{thickness required by Fig. 2}}{\text{actual thickness of shell}}$$

Reinforced openings in the shell shall conform to the requirements of Par. U-59b where t is thickness required by Fig. 2.

Make the following additions to Paragraphs U-64 and U-66 of the Unfired Pressure Vessel Code:-

U-64 Hydrostatic Test. Vessels constructed to operate solely under the external pressure of the atmosphere shall be subjected to an internal pressure of not less than 15 lb. per sq. in.

U-66 Stamping. Vessels constructed to operate solely under the external pressure of the atmosphere shall be stamped "Vacuum".

THREE TYPES OF

UNFIRED CYLINDRICAL VESSELS SUBJECTED TO

EXTERNAL PRESSURE

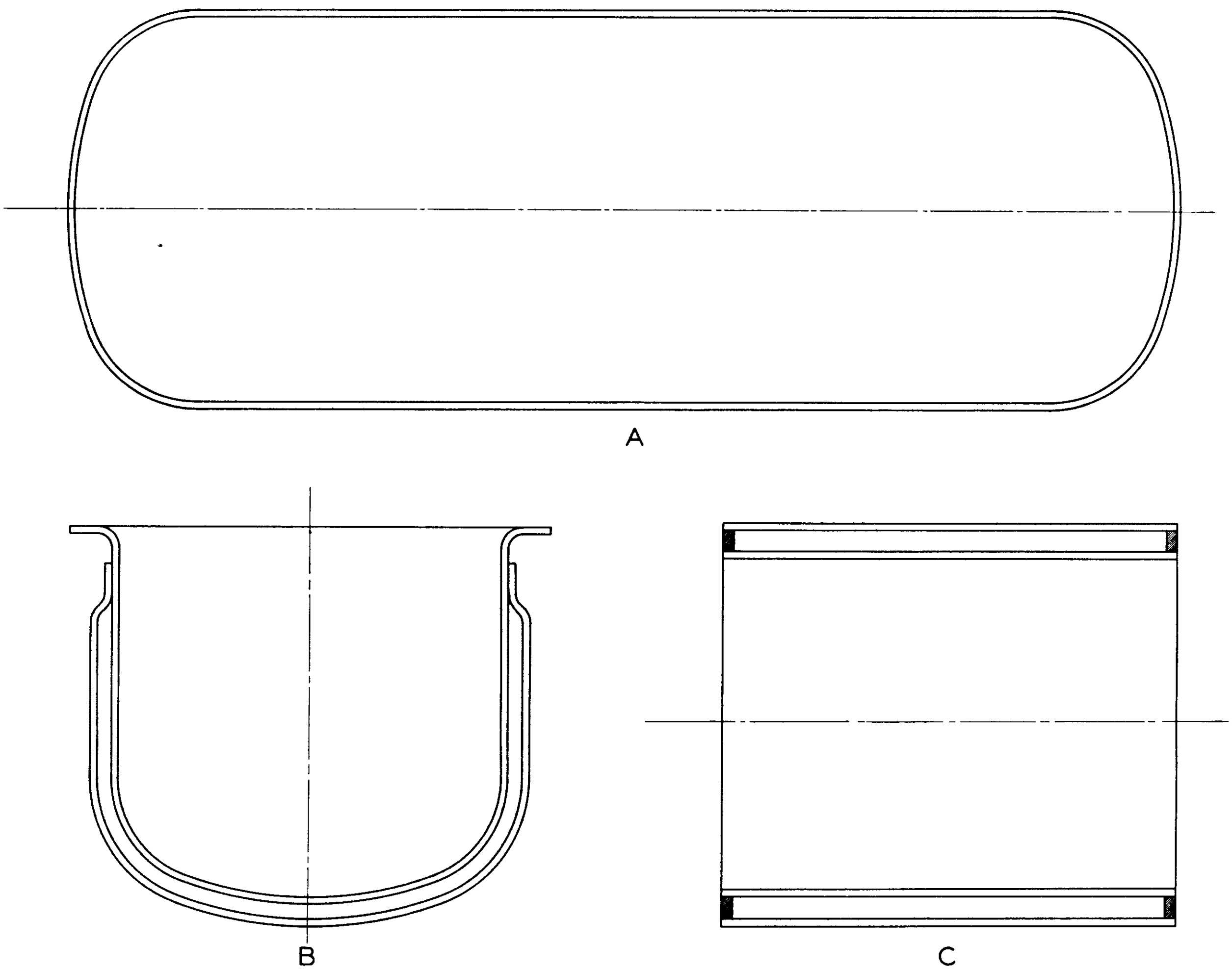

FIG. 1

CHART FOR DETERMINING SHELL THICKNESS OF UNFIRED CYLINDRICAL VESSELS SUBJECTED TO EXTERNAL PRESSURE

WHEN CONSTRUCTED OF STEEL

COMPLYING WITH SPECIFICATIONS S-1, S-2, S-4, S-17, S-18, S-25

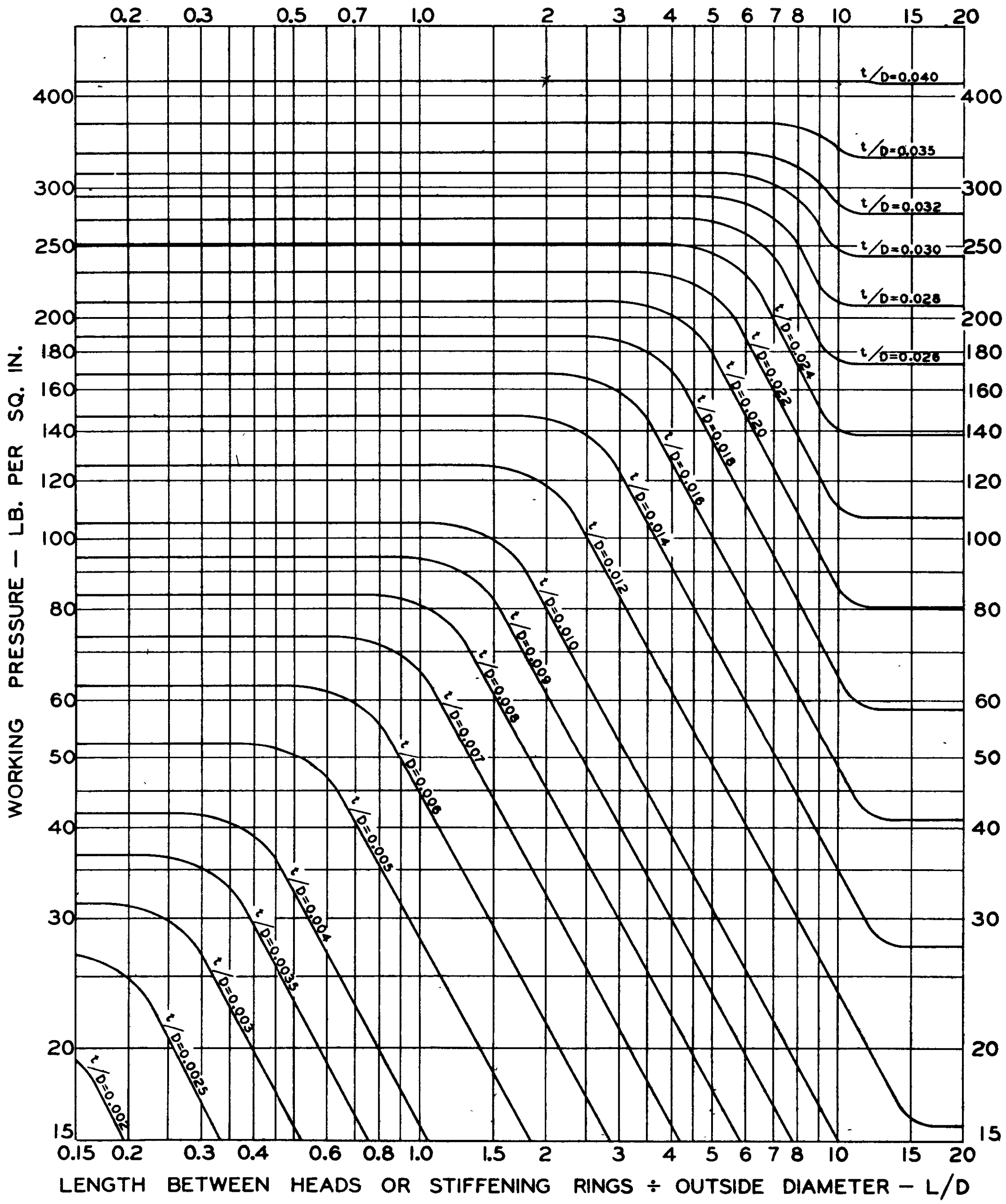

FIG. 2

F = 5

DIAGRAMMATIC REPRESENTATION OF VARIABLES FOR DESIGN OF UNFIRED CYLINDRICAL VESSELS SUBJECTED TO EXTERNAL PRESSURE

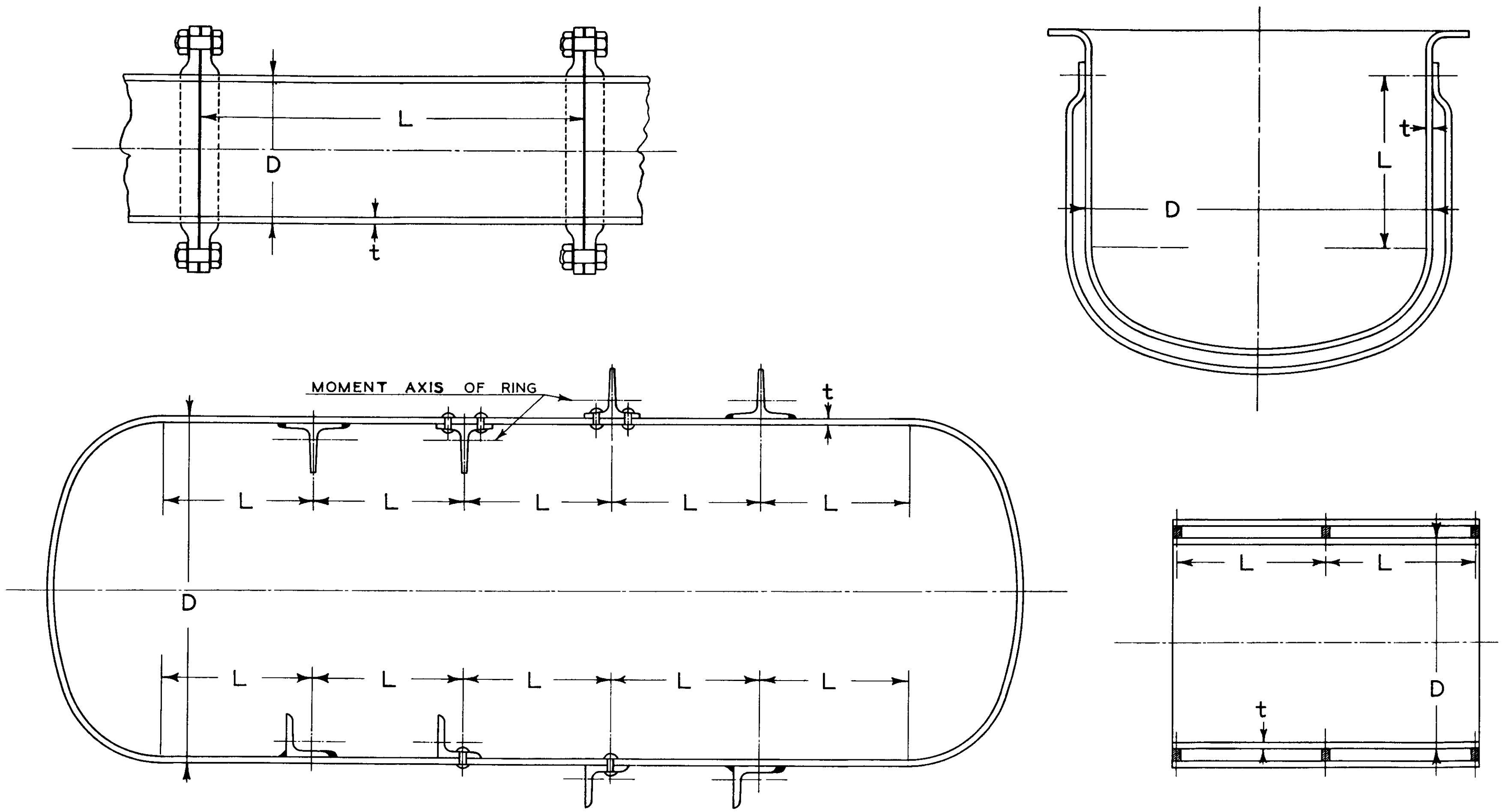

FIG. 3

ALLOWABLE DIFFERENCE IN MAXIMUM AND MINIMUM DIAMETERS FOR UNFIRED CYLINDRICAL VESSELS SUBJECTED TO EXTERNAL PRESSURE

(MAXIMUM DIAMETER – MINIMUM DIAMETER = e)

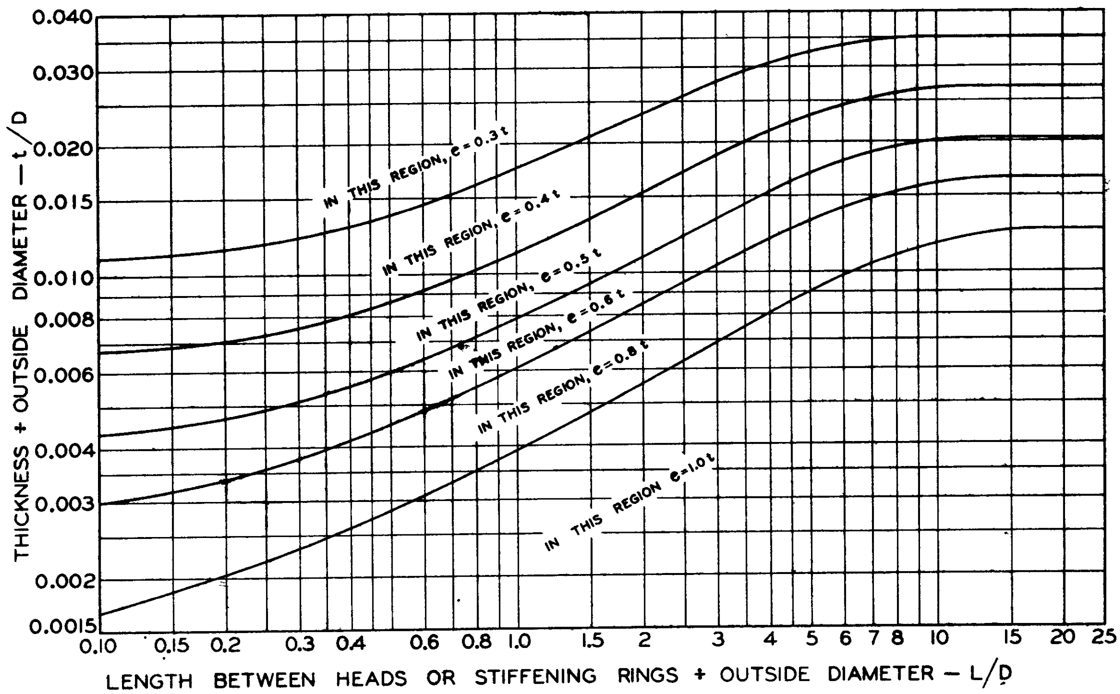

FIG. 4

EXAMPLES OF VARIATION FROM CIRCULAR FORM IN UNFIRED CYLINDRICAL VESSELS SUBJECTED TO EXTERNAL PRESSURE

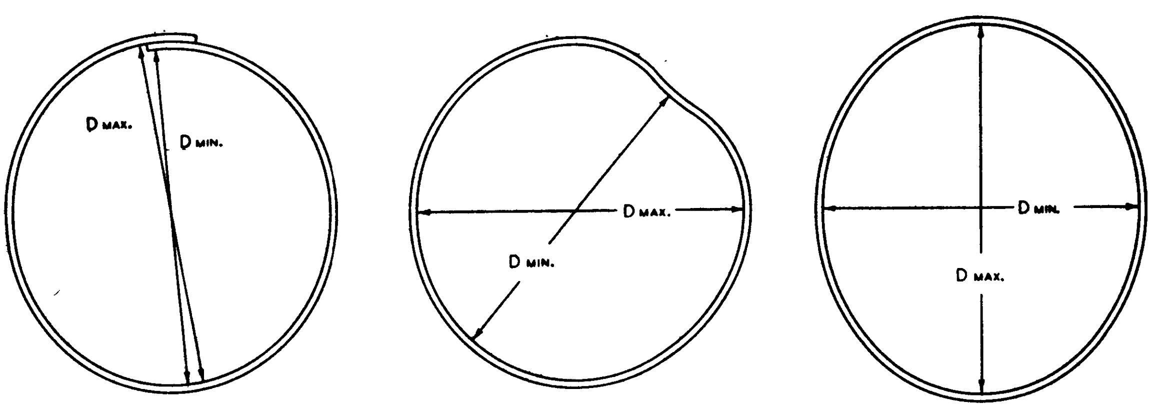

FIG. 5

REQUIRED MOMENT OF INERTIA OF STIFFENING RINGS FOR
UNFIRED CYLINDRICAL VESSELS SUBJECTED TO EXTERNAL PRESSURE

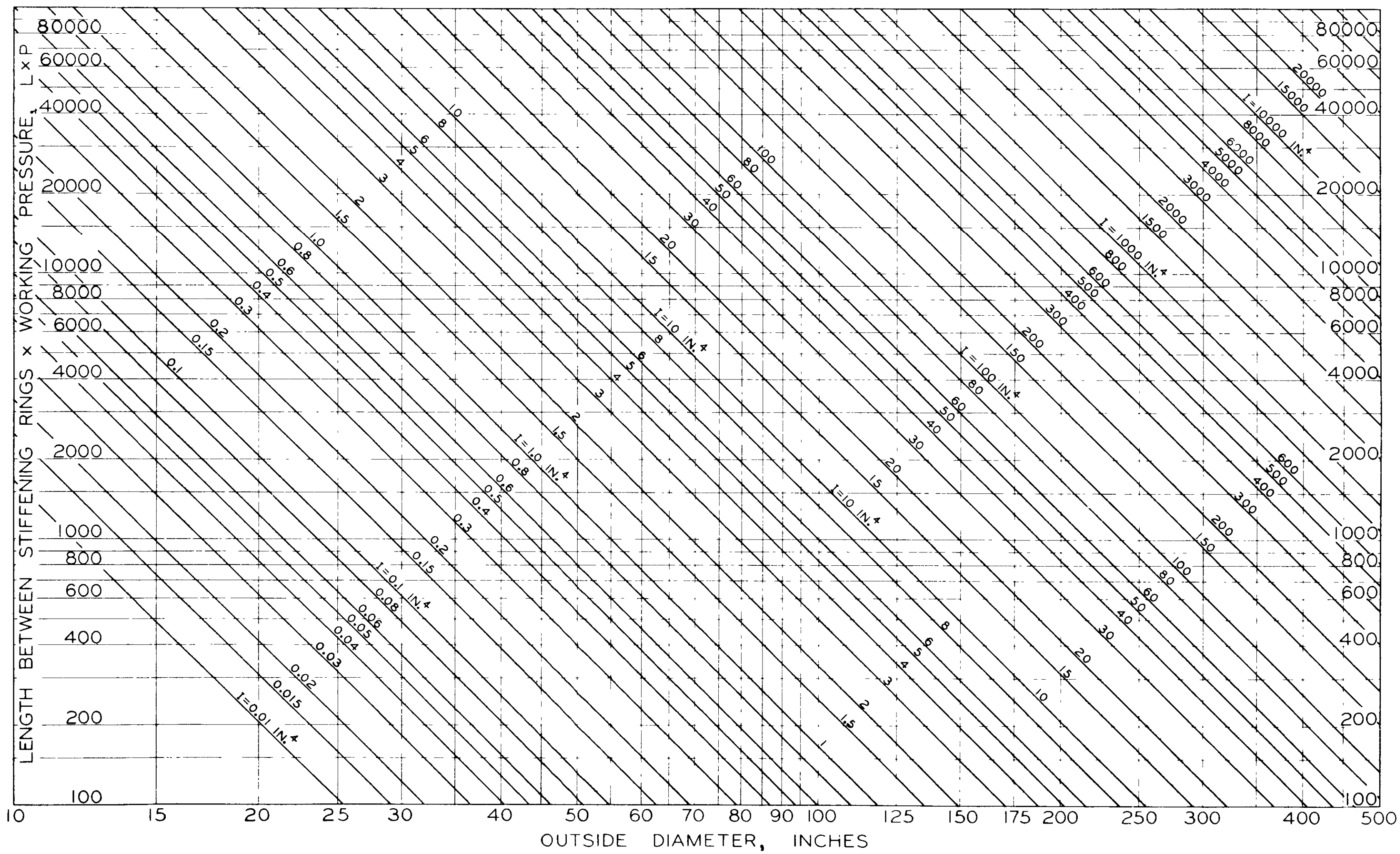

FIG. 6

VARIOUS ARRANGEMENTS OF STIFFENING RINGS FOR UNFIRED CYLINDRICAL VESSELS SUBJECTED TO EXTERNAL PRESSURE

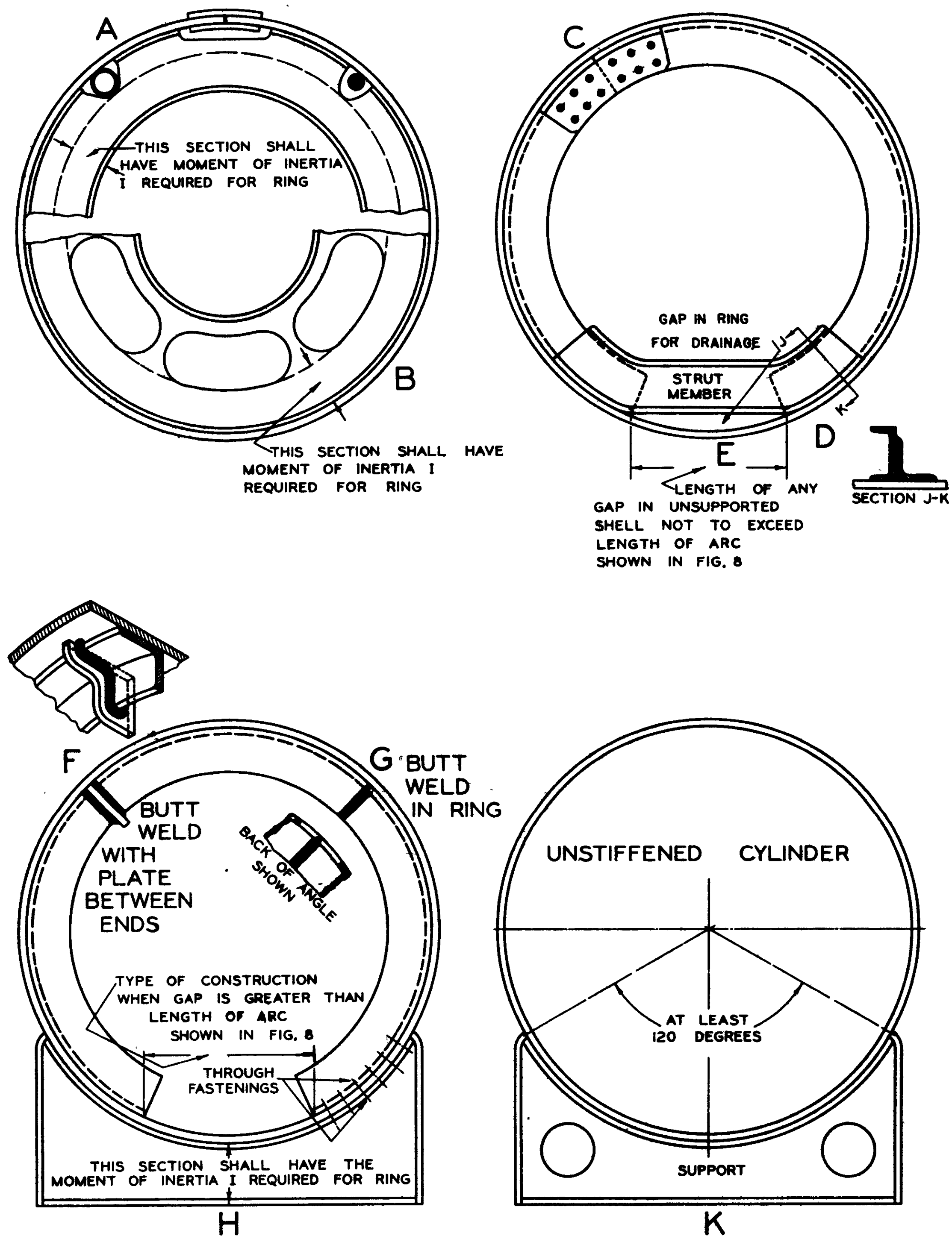

FIG. 7

MAXIMUM ARC OF SHELL LEFT UNSUPPORTED BECAUSE OF GAP IN STIFFENING RING OF UNFIRED CYLINDRICAL VESSELS SUBJECTED TO EXTERNAL PRESSURE

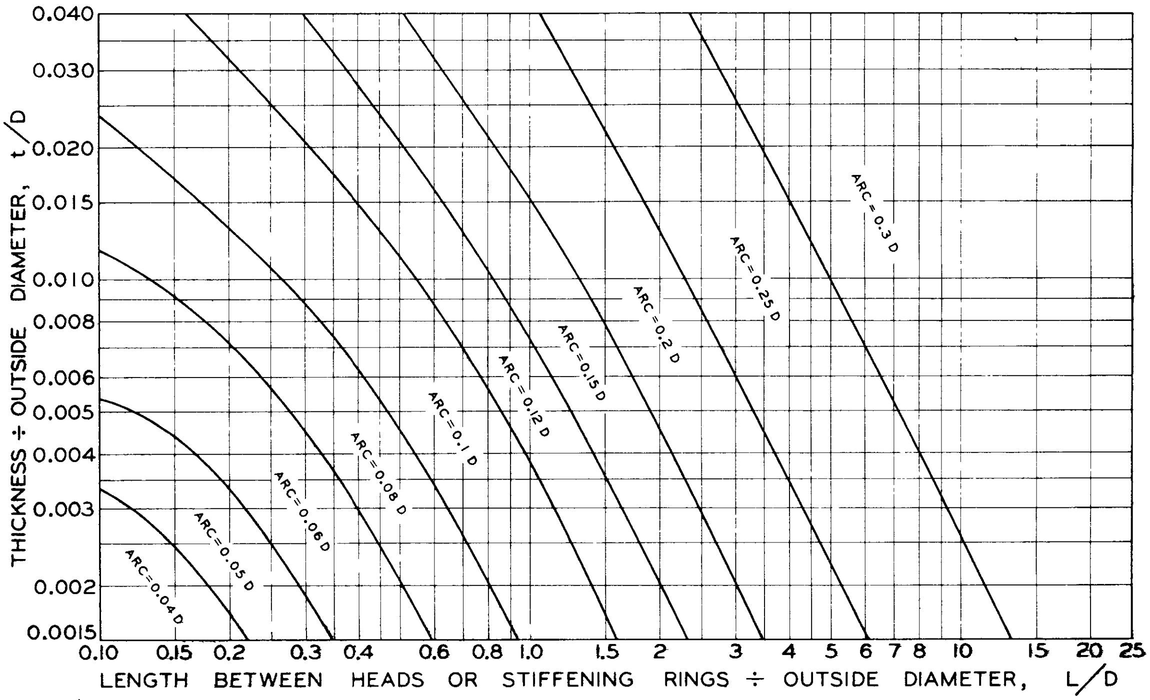

FIG. 8

METHODS OF ATTACHING STIFFENING RINGS TO SHELL OF UNFIRED CYLINDRICAL VESSELS SUBJECTED TO EXTERNAL PRESSURE

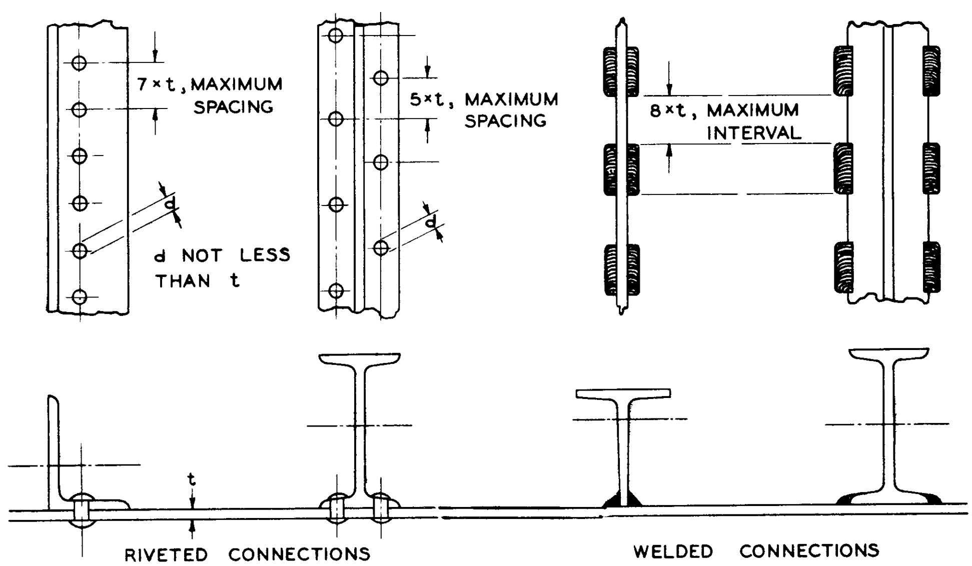

FIG. 9