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Design of Shell Sections

The symbols used in this article are defined below:

D_o	=	Outside Diameter of the Cylindrical Shell or Tube
E	=	Joint Efficiency
E_m	=	Modulus of Elasticity of Material at Design Temperature
L	=	Design Length of Vessel Section between Lines of Support
L_s	=	One-half the Distance from the Centreline of Stiffening Ring to the next Line of Support on one side, Plus One-half of the Centreline Distance to the Next Line of Support on the Other Side of the Stiffening Ring, Both Measured Parallel to the Axis of the Cylinder
R_o	=	Outside Radius of Spherical Shell
S	=	Maximum Allowable Stress Value
t	=	Minimum Required Thickness of Shell or Tube, or Spherical Shell, mm (in.)
t_s	=	Nominal Thickness of Shell or Tube, mm (in.)

UG-27 Thickness of Shells under Internal Pressure

This paragraph provides formulas to determine minimum required thickness of shells under internal pressure. In addition to these formulas, provisions should also be made for loadings listed in UG-22, when such loadings are expected. Local thin areas on the inside or outside of cylindrical shells under internal pressure are acceptable provided they meet the requirements in Appendix 32 of the Code. Formulas for thick shells from Mandatory Appendix 1 are also listed here.

Please note that the definition of thick shells is arbitrary and different for cylindrical shells and for spherical shells.

Thin Cylindrical Shells

The minimum thickness or maximum allowable working pressure of cylindrical shells shall be greater thickness or lesser pressure as determined for the circumferential stress or the longitudinal stress. For pipes, the inside radius is determined by subtracting the nominal wall thickness from the outside radius.

Circumferential Stress (Longitudinal Joint)

Inside Dimensions:

These formulas are valid for those shells where the thickness does not exceed one-half the inside radius, or inside pressure does not exceed *0.385 SE*:

$$t = \frac{PR}{SE - 0.6P}$$

$$P = \frac{SEt}{R + 0.6t}$$

Outside Dimensions: (Appendix 1-1)

These formulas, in terms of the outside radius are equivalent to and may be used instead of those given for the inside dimensions. The thickness and pressure limitations are same as those for inside dimensions.

$$t = \frac{PR_o}{SE+0.4P}$$

$$P = \frac{SEt}{R-0.4t}$$

Longitudinal Stress (Circumferential Joint)

Inside Dimensions:

These formulas will govern only when the circumferential joint efficiency is less than one-half the longitudinal joint efficiency, or when the effect of the supplementary loadings (UG-22) causing longitudinal bending or tension in conjunction with internal pressure is being investigated. Furthermore, these formulas are valid for those shells where the thickness does not exceed one-half the inside radius, or inside pressure does not exceed *1.25 SE*:

$$t = \frac{PR}{2SE+0.4P}$$

$$P = \frac{2SEt}{R-0.4t}$$

No formulas are provided in the Code for minimum thickness or maximum pressure of shells under internal pressure in terms of outside dimensions for longitudinal stress.

Thick Cylindrical Shells (Appendix 1-2)

Circumferential Stress (Longitudinal Joint)

When the thickness of the cylindrical shell under internal design pressure exceeds one-half the inside radius, or when P exceeds *0.385SE*, the following formulas shall apply:

When P is known and t is desired,

$$t = R\left(Z^{1/2} - 1\right) = R_o \frac{\left(Z^{1/2}-1\right)}{Z^{1/2}}$$

where

$$Z = \frac{SE+P}{SE-P}$$

Where t is known and P is desired,

$$P = SE\left(\frac{Z-1}{Z+1}\right)$$

where

$$Z = \left(\frac{R+t}{R}\right)^2 = \left(\frac{R_o}{R}\right)^2 = \left(\frac{R_o}{R_o-t}\right)^2$$

Longitudinal Stress (Circumferential Joint)

When the thickness of the cylindrical shell under internal design pressure exceeds one-half the inside radius, or when P exceeds *1.25SE*, the following formulas shall apply:

When P is known and t is desired,

$$t = R\left(Z^{1/2} - 1\right) = R_o\left(\frac{Z^{1/2} - 1}{Z^{1/2}}\right)$$

where

$$Z = \left(\frac{P}{SE} + 1\right)$$

Where t is known and P is desired,

$$P = SE(Z - 1)$$

where

$$Z = \left(\frac{R+t}{R}\right)^2 = \left(\frac{R_o}{R}\right)^2 = \left(\frac{R_o}{R_o - t}\right)^2$$

Thin Spherical Shells

The minimum thickness or maximum allowable working pressure of spherical shells shall be as per the formulas given below. These formulas hold when the thickness of the shell of the spherical vessel does not exceed *0.356R*, or the internal pressure does not exceed *0.665SE*.

In terms of inside dimensions,

$$t = \frac{PR}{2SE - 0.2P}$$

$$P = \frac{2SEt}{R + 0.2t}$$

In terms of outside dimensions (Appendix 1-1),

$$t = \frac{PR_o}{2SE + 0.8P}$$

$$P = \frac{2SEt}{R - 0.8t}$$

Thick Spherical Shells (Appendix 1-3)

When the thickness of the shell of a spherical vessel or of a hemispherical head under internal design pressure exceeds *0.356R*, or when P exceeds *0.665SE*, the following formulas shall apply:

When P is known and t is desired,

$$t = R\left(Y^{1/3} - 1\right) = R_o\left(\frac{Y^{1/3} - 1}{Y^{1/3}}\right)$$

where

$$Y = \frac{2(SE + P)}{2SE - P}$$

Where t is known and P is desired,

$$P = 2SE\left(\frac{Y - 1}{Y + 2}\right)$$

where

$$Y = \left(\frac{R+t}{R}\right)^3 = \left(\frac{R_o}{R_o - t}\right)^3$$

UG-28 Thickness of Shells and Tubes under External Pressure

Rules for design of shells and tubes under external pressures in the Code are limited to cylindrical shells, with or without stiffening rings, tubes and spherical shells. Charts used in determining minimum required thickness of these components are given in ASME II-D, Subpart 3, and are reproduced here (See Figure 1).

The design for external pressure uses the concept of design length of a vessel section between lines of support (See Figure 2). A line of support is:

1) A circumferential line on a head (excluding conical heads) at one-third the depth of the head from the head tangent line.
2) A stiffening ring
3) A jacket closure of a jacketed vessel
4) A cone-to-cylinder junction or a knuckle-to-cylinder junction of tori conical head or section that satisfies the moment of inertia requirement of Appendix 1-8 (requirements of this Appendix are not discussed here).

Cylindrical Shells and Tubes

The required minimum thickness of a cylindrical shell or tube under external pressure shall be determined by determined by the following procedure.

Cylinders having D_o/t values ≥ 10

Step 1:

Assume a value for t and determine the ratios L/D_o and D_o/t.

Step 2:

Enter Figure 1 at the value of L/D_o determined in Step 1. For values of L/D_o greater than 50, enter the chart at a value of $L/D_o = 50$. For values of L/D_o less than 0.05, enter the chart at a value of $L/D_o = 0.05$.

Step 3:

Move horizontally to the line for the value of D_o/t determined in Step 1. Interpolation may be made for intermediate values of D_o/t. From this point of intersection, move vertically downward to determine the value of factor A.

Step 4:

Using the value of A calculated in Step 3, enter the applicable material chart in ASME II-D, Subpart 3 (See Figure 3 for sample chart) for the material under consideration. Move vertically to an intersection with the material temperature line for the design temperature. Interpolation may be made between lines for intermediate temperatures.

In cases where the value of A falls to the right of the end of material/ temperature line, assume an intersection with the horizontal projection of the upper end of the material/ temperature line.

For values of A falling to the left of material/ temperature line, the value of P_a can be calculated using the following formula:

$$P_a = \frac{2AE_m}{3(D_o/t)}$$

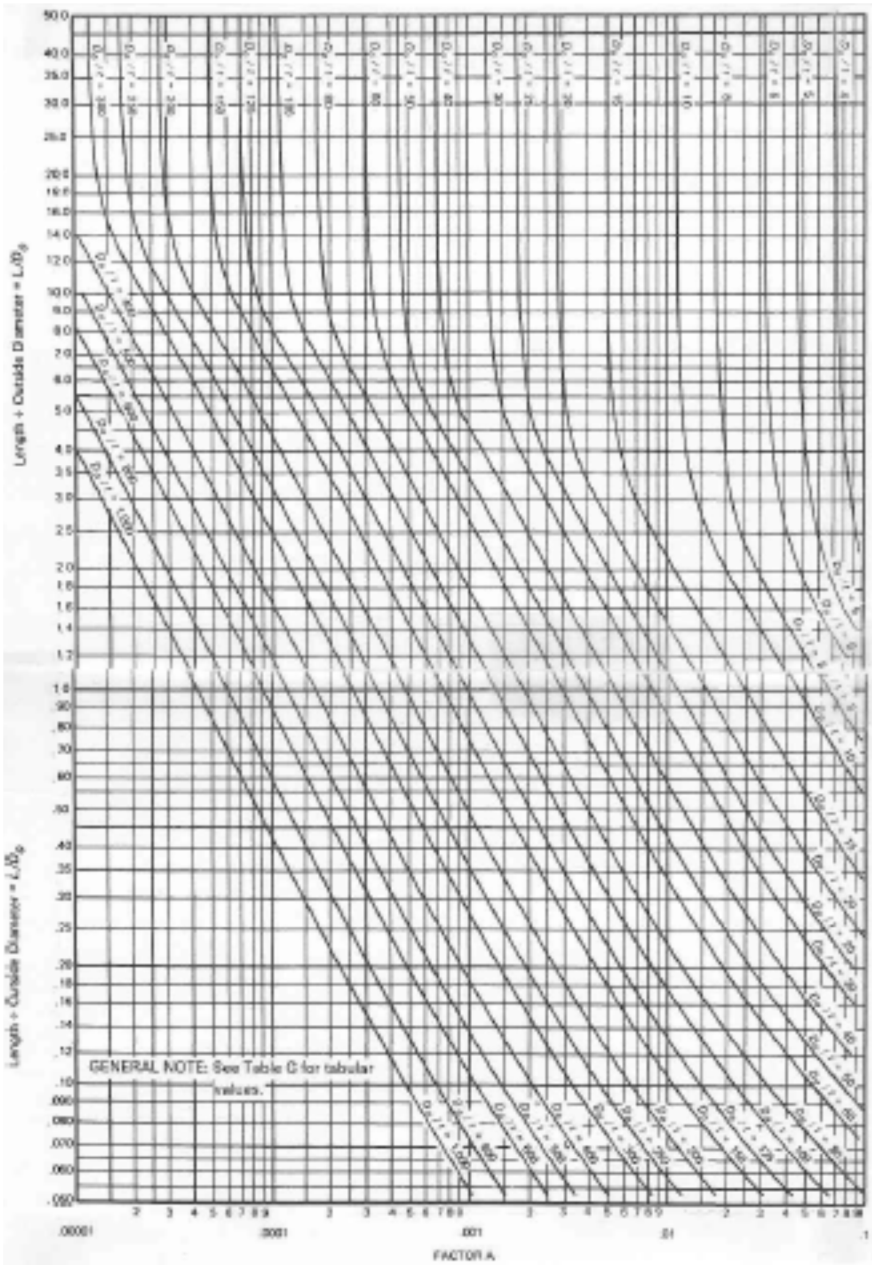

Figure 1: Chart for External or Compressive Pressure Loadings
[Fig. G from ASME Section II, Part D, Subpart 3]

Step 5:

From the intersection obtained in Step 4, move horizontally to the right and read the value of factor B

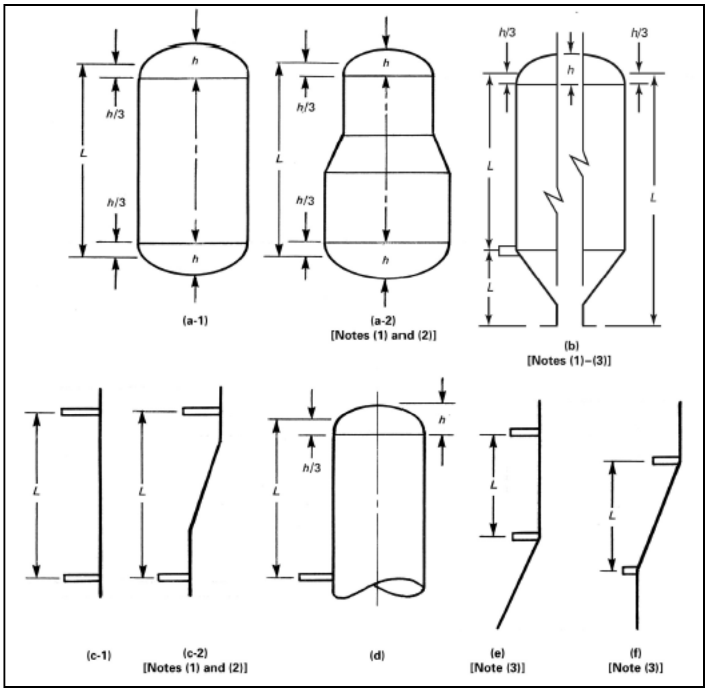

Figure 2: Line of Support for Design of Cylindrical Vessels Subjected to External Pressure

[Fig. UG-28.1]

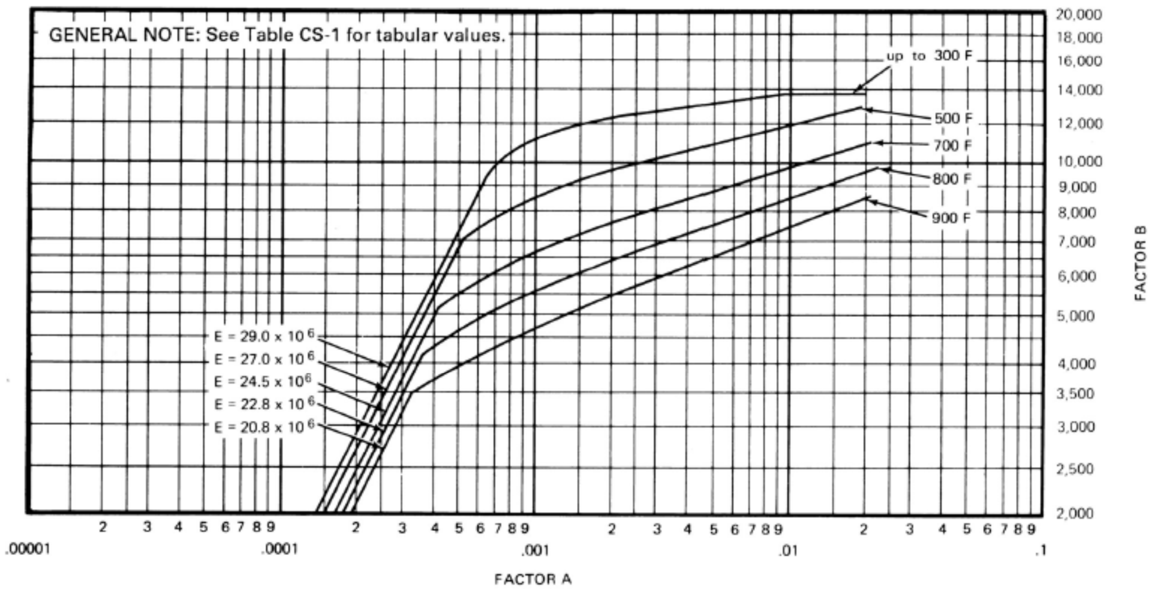

Figure 3: Chart for Determining Shell Thickness for Components under External Pressure - Carbon & Low Alloy Steels with Specified Minimum Yield Strength Less than 30,000 psi

[Fig. CS-1 from ASME Section II, Part D, Subpart 3]

Step 6:

Using this value of B, calculate the value of maximum allowable working pressure Pa using the following formula:

$$P_a = \frac{4B}{3(D_o/t)}$$

Step 7:

Compare the calculated value of P_a - if it is smaller than P, select a larger value for t and repeat the design procedure.

Cylinders having D_o/t values < 10

Step 1:

Using the same procedure as for cylinders with $D_o/t \geq 10$, obtain the value of B. For the values of D_o/t less than 4, the value of factor A can be calculated using the formula:

$$A = \frac{1.1}{(D_o/t)^2}$$

For the values of A greater than 0.10, use a value of 0.10.

Step 2:

Using the value of B obtained in Step 1, calculate a value P_{a1} using the following formula:

$$P_{a1} = \left[\frac{2.167}{(D_o/t)} - 0.0833\right] B$$

Step 3:

Calculate a value P_{a2} using the following formula:

$$P_{a2} = \frac{2S}{(D_o/t)}\left[1 - \frac{1}{D_o/t}\right]$$

S is the lesser of two times the maximum allowable stress value in tension at design metal temperature, or 0.9 times the yield strength of the material at design temperature. Please note that this definition of S is different from that defined earlier.

Values of yield strength are obtained from the applicable pressure chart as follows:

a) For a given temperature curve, determine the B value that corresponds to the right hand side termination point of the curve.
b) Yield stress is twice the B value obtained in B in a).

Step 4:

The smaller of the values of P_{a1} or P_{a2} shall be used as the maximum allowable working pressure P_a. If P_a is smaller than P, select a larger value for t and repeat the design procedure.

Spherical Shells

The minimum required thickness of a spherical shell under external pressure shall be determined by the following procedure:

Step 1:

Assume a value for t and calculate the value for factor A using the following formula:

$$A = \frac{0.125}{(R_o/t)}$$

Step 2:

Using this value of A, enter the material chart in ASME II-D, Subpart 3 for the material under consideration (see Figure 3 for a sample chart). Move vertically to an intersection with the material/ temperature line for the design temperature. Interpolation may be made between lines for intermediate temperatures.

In cases where value of A falls to the right of the end of the material/temperature line, assume and intersection with the horizontal projection of the upper end of the material temperature line.

For values of A falling to the left of the material/temperature line, the value of the maximum external working pressure P_a can be calculated as follows:

$$P_a = \frac{0.0625E_m}{(R_o/t)^2}$$

Step 3:

From the intersection obtained in Step 2, move horizontally to the right and read the value of factor B.

Step 4:

Using this value of B, calculate the value of the maximum external working pressure P_a using the following formula:

$$P_a = \frac{B}{(R_o/t)}$$

Step 5:

Compare the calculated value of P_a - if it is smaller than P, select a larger value for t and repeat the design procedure.

UG-29 Stiffening Rings for Cylindrical Shells under External Pressure

External stiffening rings are attached to the shell mostly by welding, although they can be attached by brazing also in some cases. The adequacy of moment of inertia for a stiffening ring to withstand external pressure shall be determined by the following procedure:

Step 1:

Assuming that the shell has been designed and D_o, L_s, and t are known, select a member to be used for the stiffening ring and determine its cross-sectional area A_s. Then calculate factor B using the following formula:

$$B = 3/4\left(\frac{PD_o}{t+A_s/L_s}\right)$$

Step 2:

Enter the right hand side of the applicable material chart for the material under consideration (see Figure 3 for a sample chart) at the value of B determined in Step 1.

Step 3:

Move horizontally to the left to the material/ temperature line for the design metal temperature. For values of B falling below the left end of the material/ temperature line, the value for A can be calculated using the formula:

$$A = \frac{2B}{E_m}$$

Else, move vertically to the bottom of the chart and read the value of A.

Step 4a:

In those cases where only the stiffening ring is considered, compute the required moment of inertia using the following formula:

$$I_s = \left[D_o{}^2 L_s(t + A_s/L_s)A\right]/14$$

Step 4b:

In those cases where the combined ring-shell is considered, compute the required moment of inertia using the following formula:

$$I_s{}' = \left[D_o{}^2 L_s(t + A_s/L_s)A\right]/10.9$$

Step 5:

In those cases where only the stiffening ring is considered, determine the available moment of inertia I of the stiffener ring cross section about its neutral axis parallel to the axis of the shell.

In those cases where the combined ring-shell section is considered, determine the available moment of inertia I_s of combined ring-shell cross section about its neutral axis parallel to the axis of the shell. The nominal shell thickness t_s shall be used and the width of the shell that is taken as contributing to the moment of inertia of the combined section shall not be greater than $1.10\sqrt{D_o t_s}$ and shall be taken as lying one-half on each side of the centroid of the ring. Portions of the shell plate shall not be considered as contributing area to more than one stiffening ring.

If the stiffeners are so located that the maximum permissible effective shell sections overlap on either or both sides of the stiffener, the effective shell section for that stiffener shall be shortened by one-half of each overlap.

<u>Note: In those cases where the designer chooses to consider only the stiffening ring, Step 4b along with the corresponding moment of inertia in Step 5 is considered. In those cases, where the combined moment of inertia is considered, Step 4b along with the corresponding moment of inertia in Step 5 is considered.</u>

Step 6: If the required moment of inertia is greater than the available moment of inertia for the section selected:

a) For those cases where the combined ring-shell stiffness was not considered, a new section with a larger moment of inertia must be selected, or the ring-shell combination that was not previously considered together shall be considered together.
b) For those cases where the combined ring-shell stiffness was considered, a new section with a larger moment of inertia must be selected.
c) All of the calculations shall be repeated using new section properties of the ring or the ring-shell combination.

If the required moment of inertia is smaller than the actual moment of inertia of the ring or ring-shell combination, whichever is used, that ring section or the combined section is satisfactory.

UG-30 Attachment of Stiffening Rings

Stiffening rings may be placed on the inside or outside of a vessel and shall be attached to the shell by welding or brazing. Brazing may be used if the vessel is not to be later stress-relieved. Majority of the stiffening rings are placed on the outside of the shell and are attached to the shell by welding. Only such rings will be discussed here. Readers interested in other attachment details are requested to refer to the Code.

Strength of the Attachment Welds:

Stiffening ring attachment welds shall be sized to resist the full radial pressure load from the shell between stiffeners, and shear loads acting radially across the stiffener caused by external design loads carried by the stiffener (if any) and a computed radial shear equal to 2% of the stiffening ring's compressive load.

1) The radial pressure load from the shell, lb/in., is equal to PL_s.
2) The radial shear load is equal to $0.01PL_sD_o$.

Minimum Size of Attachment Welds:

The fillet weld leg size shall not be less than the smallest of the following:

1) 6 mm (1/4 in)
2) Vessel thickness at weld location
3) Stiffener thickness at weld location

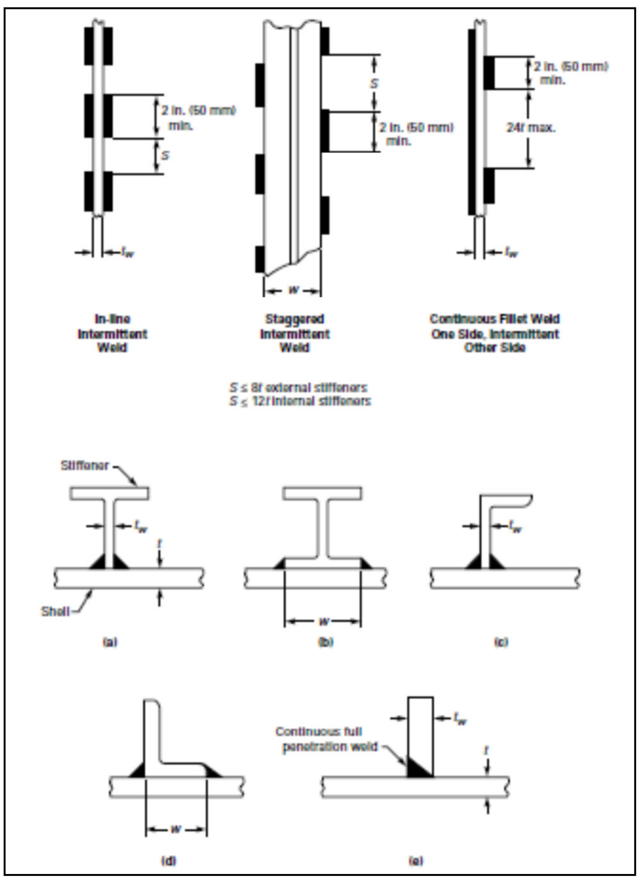

Figure 4: Some Acceptable Methods of Attaching Stiffening Rings [Fig. UG-30]

UG-31 Tubes and Pipe When Used as Tubes or Shells

The procedure for designing tubes and pipes when they are to be used as shells is the same as that discussed so far, with following exception:

Where ends are threaded, additional wall thickness is to be provided in the amount of 20/n mm (0.8/n in) [where n equals the number of threads per 25.4mm (1 in).

Sources:

1. ASME Boiler & Pressure Vessel Code, Section VIII, Division 1: Edition 2010

*** END OF THE ARTICLE ***

About CoDesign Engineering

CoDesign Engineering is involved in providing training and consultancy services as described below:

Training

- Pressure vessel & heat exchanger design (ASME Section VIII, Div. 1)
- Power and process piping design (ASME B31.1 & B31.3)
- Solar PV power plant design

Consultancy

- Engineering solutions related to pressure vessels and heat exchangers
- PMC as well as EPC services for solar PV power plants

We have designed a 3-day training course for ASME BPVC Section VIII, Div. 1 that can be offered at most cities in India. In-house training can also be provided at any location in India or in US upon request. The training is designed as a workshop where the delegates are encouraged to do all calculations using only pencil, paper and calculators. Please contact Ramesh Tiwari at rtiwari123@gmail.com or ramesh.tiwari@codesignengg.com for 2012 training calendar, rates and the contents of the course.

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