5
Vessel Internals

Procedure 5-1: Design of Internal Support Beds

Process vessels frequently have internal beds that must be supported by the vessel shell. Sand filters, packed columns and reactors with catalyst beds are just a few examples of vessels that have internal beds. The beds are typically supported by a combination of beams, grids and a support ring welded to the vessel shell. The support ring supports the periphery of the grating or tray plates. The support ring can be welded to the vessel wall by fillet welds or full penetration welds for larger loaded members. In some cases they are integrally forged into the vessel wall.

This procedure can be used to design all the various components of a support structure consisting of the following;

1. Beams
2. Support ring
3. Grating
4. Beam seats
5. Beam support clips and bolting
6. Vessel wall

Beam Supports

The beams are typically supported by one of three methods;

1. Beam seats
2. Clips
3. Support ring

Beams should never be welded directly to the vessel wall because of the restraint they would impose on vessel growth. All of the methods listed above will allow for the radial expansion of the vessel due to temperature and pressure. Slotted holes in the clip or for the bolting attaching the beam to a beam seat, allow for expansion.

Beam seats and clips allow the top flange, or top of beam, to be level with the top of the support ring. This is a very convenient feature for this type of support. In effect, it creates a level surface to support the grating or tray plates. Conversely, if the beams are supported by the support ring, then the bottom flange of the beam will sit on top of the support ring. For most applications this would be unacceptable. In order to make the top of the bottom flange of the beam to be on the same plane as the top of the support ring, support blocks are used. Support blocks can be used for solid beams, lattice beams or built up beams. For certain applications, the web of the beam can be extended to create the support surface in lieu of support blocks, or support blocks can be added to each side of the web to reinforce the web locally and provide a larger bearing area.

Beams

There is really no limit to the number of cross beams that may be utilized, however this procedure only illustrates the use of up to eight beams. If access through the center of the vessel is a concern, then only an even number of beams can be used. A center beam would impede access. Beams can be solid members, welded I-Beam type, welded T-type, or open, lattice type structures.

One additional support method, not shown here, is the "Hub Ring" or "Ring Beam" approach. Hub rings are ideal for many applications. They consist of two parallel rings attached to radial beams. The radial beams in turn are supported by the vessel shell or support rings. The upper ring is a compression ring. The lower ring is a tension member. The rings are naturally stiff members that are loaded in their strongest direction, perpendicular to the applied load. In theory they employ all the strength aspects of the arch, and thus are sometimes referred to as "arcuate beams". Design of hub rings with radial beams are not detailed in this procedure. Only the design of straight beams are included.

Lattice beams are desirable, if possible, due to their light weight. Where internals are stainless steel to prevent or minimize corrosion, they will also reduce cost. Unlike solid or built-up beams, the top and bottom members of a lattice beam have distinct functions. The bottom chord is the tension only member. The top chord is a compression only member. This is for beams above the line of support. For beams below the line of support, the reverse would be true.

In a lattice beam, the top chord is a compression member and must be capable of supporting a column axial load without buckling. A T-type compression member is sometimes used to provide extra lateral stability in the compression chord.

Solid and built–up beams must be checked for lateral stability against buckling. In the event this is exceeded, then either the design must be changed or some anti-buckling supports installed. One style of anti-buckling devices is the anti-buckling comb.

Each beam supports the load consisting of one half the distance between each adjacent beam or the support ring, though this would result in a non-uniform loading. However, one simplifying assumption, which is conservative, allows for a rectangular loading for each beam based on the overall length of the beams. This method neglects the contribution of the support ring, and assumes the beams support the entire load. While not completely accurate, it is much easier to apply.

It is most convenient from a design standpoint to have the beams equally spaced. However, this is not always possible because of the clearances, accessibility and other internals penetration. This procedure gives standard loads for uniformly spaced beams, but also describes the procedure for the design of beams that are not uniformly spaced.

Whether the beams extend above or below the support level is matter of application. Both methods have their pros and cons. Basically, the method of support is determined by the designer to accommodate the space inside the vessel as well as the process function of the bed or tray that the beams are supporting. If the beam extends up into the bed, then a certain amount of media is displaced and removal of media is more difficult. Conversely, in the case where the beams support a tray, then having the beams extend above the tray may impose restrictions on flow. Beams for catalyst support grid (CSG) applications almost always extend up into the bed to minimize the length of the vessel.

Loadings

The loading for a bed consists of dead load, liquid load and live load. The dead load consists of the weight of the supports and the media it supports, i.e. catalyst, packing, inert balls, sand, clay, etc. The live load consists of the delta P, differential pressure, developed by the restriction of the bed to downward flow. Typically there is some amount of fouling that occurs in beds that cause a buildup of delta P across the bed during operation. The live load can easily become the major design load for the supports.

The last load is the weight of product. This could be either liquid or solid. It could be the liquid on a tray or the liquid hold-up in the bed itself.

The grating is the support members that span between the beams and support the media. The grating may be covered with screen to prevent egress of small particles. The screen may be installed in a single or multiple layers of mesh. It can also be covered with a layer of wedge wire screen for the same purpose. The grating must be designed to support the total of all loads.

In reactors, the support bed configuration is frequently referred to as a CSG. The CSG consists of beams, grating and screen.

Procedure

The basic design procedure for the beams is as follows;

1. Estimate the dead loads to be supported to include the media and weight of support members.
2. Calculate the live load based on the delta P specified.
3. Calculate the liquid (or solids) loading in the bed.
4. Combine the three load cases to get a total load.
5. The total load divided by the cross sectional area is the uniform design load, p.
6. Calculate the area supported by each beam and multiply this area times the uniform load to get the load supported by each beam.
7. A standard AISC type beam type analysis is then performed to size the beam.

Notation

A_r = Area of bolt required, in^2
A_n = Total area supported by beam, in^2
A_c = Cross sectional area of vessel, in
B = Ratio of actual force to allowable force per inch on weld
C_n = Compressive force in beam, Lbs
DL = Dead load, Lbs
E = Modulus of elasticity, PSI

f_b = Stress, bending, PSI
f_C = Stress, compressive, PSI
f_T = Stress, tension, PSI
F = Total load of bed, Lbs
F_b = Allowable bending stress, PSI
F_{br} = Allowable stress, bearing, PSI
F_C = Allowable stress, compression, PSI
F_S = Allowable stress, shear, PSI
F_y = Minimum specified yield strength at design temperature, PSI
I = Moment of inertia, in^4
K = End connection coefficient per AISC
K_1 = Vertical distance from bottom of beam flange to top of fillet on web, in
LL = Live load, Lbs
M = Moment, in-Lbs
n_b = Number of bolts required
n_g = Number of bearing bars per foot
N_b = Minimum bearing length, in
N = Number of beams
P = Concentrated load, Lbs
PL = Product load, Lbs
P_c = Free area in packing/catalyst, %
p = Uniform load over entire bed, PSI
r = Radius of gyration, in
R = End reactions, Lbs
R_a = Root area of bolt, in^2
S = Allowable stress in shell, tension, PSI
S_U = Minimum specified tensile strength, PSI
t = Thickness, in
T_n = Tension force in beam, Lbs
t_w = Thickness of web, in
W_C = Weight, contents, (catalyst, packing, etc) Lbs
W_e = Weight of entrained liquid, Lbs
W_g = Weight, grating, Lbs
W_b = Weight, beams, Lbs
w_f = Size of fillet weld, in
w_n = Uniform load on beam, Lbs/in
V_c = Volume, catalyst/packing, Ft^3
Z = Section modulus, PSI
δ = Deflection, in
ΔP = Differential pressure loading

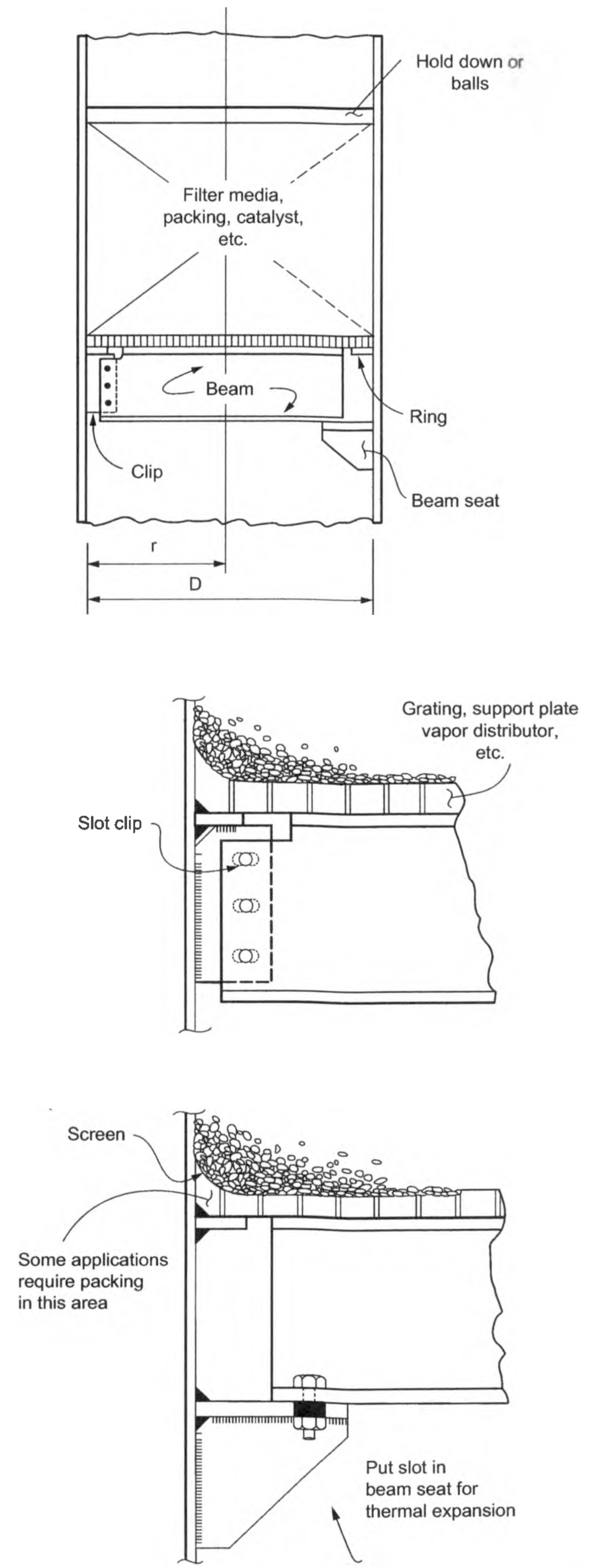


Figure 5-1. Typical support arrangements and details of an internal bed.

CASE 1: SINGLE BEAM	CASE 2: DOUBLE BEAM

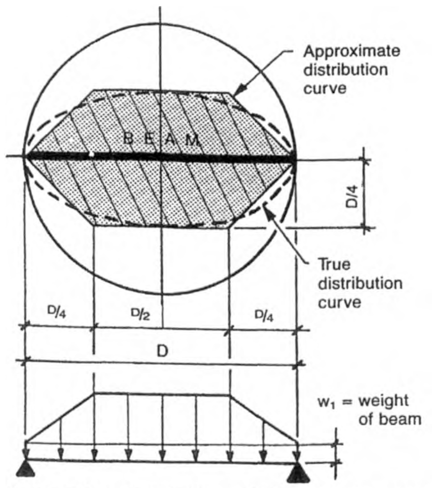


Figure 5.2 Loading diagram of single-beam support. Area of loading = 48%.

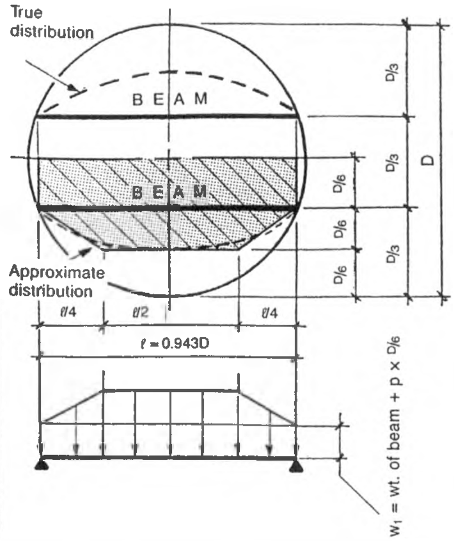


Figure 5.3 Loading diagram of double-beam support. Area of loading = 37% per beam.

BEAM CALCULATIONS

Case 1		Case 2	
UNIFORM LOAD DUE TO BEAM WEIGHT		UNIFORM LOAD DUE TO BEAM WEIGHT	
$M_1 = \frac{w_1 D^2}{8} =$		$M_1 = \frac{w_1 \ell^2}{8}$	
$R_1 = \frac{w_1 D}{2} =$		$R_1 = \frac{w_1 \ell}{2}$	
CONCENTRATED LOADS ON END SECTIONS		CONCENTRATED LOADS ON END SECTIONS	
$P = p\left(\frac{D}{4}\right)^2 =$		$P = \frac{pD\ell}{48}$	
$R_2 = P =$		$R_2 = P$	
UNIFORM LOAD IN CENTER SECTION		UNIFORM LOAD IN CENTER SECTION	
$M_2 = \frac{PD}{6} =$		$M_2 = \frac{P\ell}{6}$	
$w_2 = \frac{pD}{2} =$		$w_2 = \frac{pD}{6}$	
$R_3 = \frac{w_2 D}{4} =$		$R_3 = \frac{w_2 \ell}{4}$	
$M_3 = \frac{R_3 D}{4} + \frac{R_3^2}{2w_2} =$		$M_3 = \frac{R_3 \ell}{4} + \frac{R_3^2}{2w_2}$	

MOMENT AND REACTION CALCULATIONS

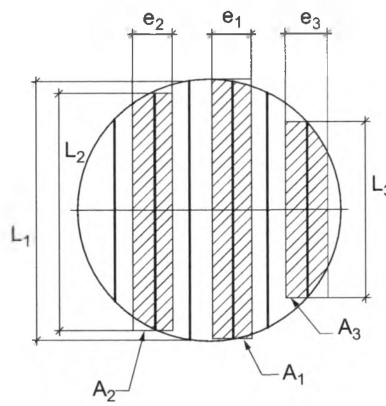
Total moment $M = M_1 + M_2 + M_3$		$Z_{reqd} = \frac{M}{F_b}$	New	
Total end reaction $R = R_1 + R_2 + R_3$			Corroded	
Select beam and add appropriate correction allowance to web and flange				

DESIGN LOADS FOR BEAMS				
DATA			LOADINGS	
ITEM	SYMBOL	VALUE	ITEM	VALUE
Material, beams and grating			Dead load, DL	
Material, vessel shell			a. Weight of beams	
Quantity of beams	N		b. Weight of grating	
Vessel ID	D		c. Weight of tray	
Design temperature	DT		d. Weight of screen	
Differential pressure	ΔP		e. Weight of packing/catalyst	
Bed depth	d_C		Product load, PL	
Liquid/contents unit weight	w_c		a. Weight of liquid on tray	
Corrosion allowance	C_a		b. Weight of liquid in bed	
Specific gravity	S_g		c. Weight of liquid above bed	
Liquid holdup (%)			d. Weight of solids	
Free area in bed (%)	P_C		Live load, LL	
Packing/catalyst unit weight	w_P			
Volume of packing/catalyst	V_P		Total load, F, Lbs	
Packing/catalyst total wt	W_C		F = DL + PL + LL	
Weight of entrained liquid	W_e			
Weight of grating	W_g		Total cross sectional area, A_c	
Weight of beams	W_b		$A_c = \pi r^2$	
Miscellaneous				
MATERIAL PROPERTIES & ALLOWABLE STRESS			Uniform load, p, PSI	
Shell at design temperature	S		$p = F / A_c$	
	E			
	F_y			
Beams at design temperature	E			
	F_y			
Bending = $.66 F_y$	F_b			
Compression, dependent on KL/r ratio	F_C			
Shear = $.4 F_y$	F_S			
Bearing = $.9 F_y$	F_{br}			

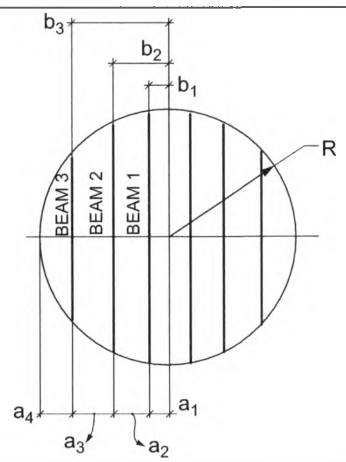
DESIGN OF SUPPORT BEAMS

DIMENSIONS

(6) Beams shown for example only!



DIMENSIONS



EQUATIONS

$e_1 = a_1 + .5\,a_2$

$e_2 = .5\,a_2 + .5\,a_3$

$e_3 = .5\,a_3 + .5\,a_4$

$e_4 = .5\,a_4 + .5\,a_5$

$L_n = 2\,[R^2 - b_n^2]^{1/2}$

$f_T = T_n / A_T$

$f_C = C_n / A_c$

$$C_c = \sqrt{\frac{2\pi^2 E}{F_y}}$$

$$F_C = \frac{\left[1 - \frac{(KL/r)^2}{2\,C_C^2}\right] F_y}{5\backslash 3 + \frac{3(KL/r)}{8\,C_C} - \frac{(KL/r)^3}{8\,C_C^3}}$$

1.0 AREA OF LOADING

BEAM	L_n	e_n	A_n
1			
2			
3			
4			

2.0 LOADS ON BEAMS

BEAM	F_n	R_n	w_n
1			
2			
3			
4			

3.0 MOMENT & FORCES

BEAM	M_n	h_n	T_n or C_n
1			
2			
3			
4			

4.0 BEAM STRESSES

BEAM	f_T	f_C	
1			
2			
3			
4			

5.0 ALLOWABLE STRESS, COMPRESSION

BEAM	L_n	$K\,L_n / r$	F_c
1			
2			
3			
4			

$A_n = L_n\,e_n$

$F_n = A_n\,p$

$R_n = w_n\,L_n / 2$

$w_n = F_n / L_n$

$M_n = w_n\,L_n^2 / 8$

$T_n = C_n = M_n / h_n$

Notes:

1. h will vary depending on the type of beam used.
2. Every beam type, solid, built-up or lattice… will all have a tension and compression zone, section or component.

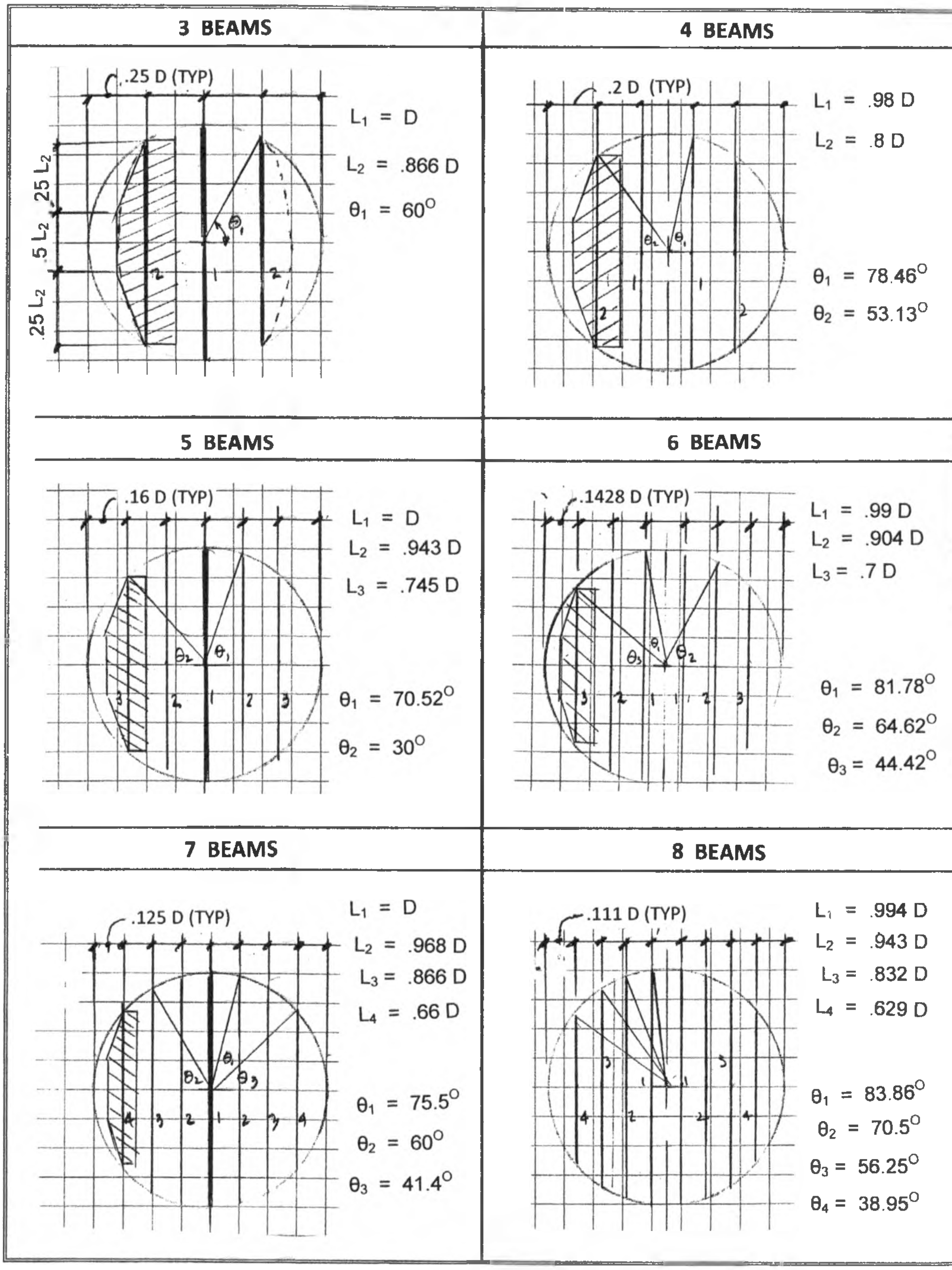

3 BEAMS
.25 D (TYP)
.25 L2
.5 L2
.25 L2
L1 = D
L2 = .866 D
θ1 = 60°
4 BEAMS
.2 D (TYP)
L1 = .98 D
L2 = .8 D
θ1 = 78.46°
θ2 = 53.13°
5 BEAMS
.16 D (TYP)
L1 = D
L2 = .943 D
L3 = .745 D
θ1 = 70.52°
θ2 = 30°
6 BEAMS
.1428 D (TYP)
L1 = .99 D
L2 = .904 D
L3 = .7 D
θ1 = 81.78°
θ2 = 64.62°
θ3 = 44.42°
7 BEAMS
.125 D (TYP)
L1 = D
L2 = .968 D
L3 = .866 D
L4 = .66 D
θ1 = 75.5°
θ2 = 60°
θ3 = 41.4°
8 BEAMS
.111 D (TYP)
L1 = .994 D
L2 = .943 D
L3 = .832 D
L4 = .629 D
θ1 = 83.86°
θ2 = 70.5°
θ3 = 56.25°
θ4 = 38.95°

Table 5-1
Beam supports - Summary of forces and moments

Quantity of Beams	Beam	L_n	A_n	F_n	R_n	M_n	w_n
1	1	D	.3927 D^2	.3927 p D^2	.1864 p D^2	.0565 p D^3	.393 p D
2	1	.943 D	.2698 D^2	.2698 p D^2	.1349 p D^2	.0343 p D^3	.286 p D
3	1	D	.2333 D^2	.2333 p D^2	.1167 p D^2	.0311 p D^3	.233 p D
	2	.866 D	.1850 D^2	.1850 p D^2	.0925 p D^2	.0219 p D^3	.214 p D
4	1	.98 D	.1925 D^2	.1925 p D^2	.0963 p D^2	.0240 p D^3	.196 p D
	2	.8 D	.1405 D^2	.1405 p D^2	.0703 p D^2	.0143 p D^3	.176 p D
5	1	D	.1655 D^2	.1655 p D^2	.0828 p D^2	.0208 p D^3	.166 p D
	2	.943 D	.1548 D^2	.1548 p D^2	.0774 p D^2	.0185 p D^3	.164 p D
	3	.745 D	.1092 D^2	.1092 p D^2	.0546 p D^2	.0107 p D^3	.147 p D
6	1	.99 D	.142 D^2	.142 p D^2	.071 p D^2	.0175 p D^3	.143 p D
	2	.904 D	.129 D^2	.129 p D^2	.0645 p D^2	.0146 p D^3	.143 p D
	3	.7 D	.085 D^2	.085 p D^2	.0425 p D^2	.0074 p D^3	.121 p D
7	1	D	.125 D^2	.125 p D^2	.0625 p D^2	.0156 p D^3	.125 p D
	2	.968 D	.121 D^2	.121 p D^2	.0605 p D^2	.0146 p D^3	.125 p D
	3	.866 D	.108 D^2	.108 p D^2	.0541 p D^2	.0117 p D^3	.125 p D
	4	.66 D	.0722 D^2	.0722 p D^2	.0360 p D^2	.0059 p D^3	.109 p D
8	1	.994 D	.110 D^2	.110 p D^2	.055 p D^2	.0137 p D^3	.111 p D
	2	.943 D	.105 D^2	.105 p D^2	.052 p D^2	.0123 p D^3	.111 p D
	3	.832 D	.0924 D^2	.0924 p D^2	.046 p D^2	.0096 p D^3	.111 p D
	4	.629 D	.0611 D^2	.0611 p D^2	.0031 p D^2	.00048 p D^3	.097 p D

Notes:

1. Table is for uniformly spaced beams only
2. Equations are as follows;

$A_n = L_n e_n$ $F_n = A_n p$ $R_n = F_n/2$ or $w_n L_n/2$ $M_n = w_n L_n^2/8$ $w_n = F_n/L_n$

1.0. Design of Support Clip

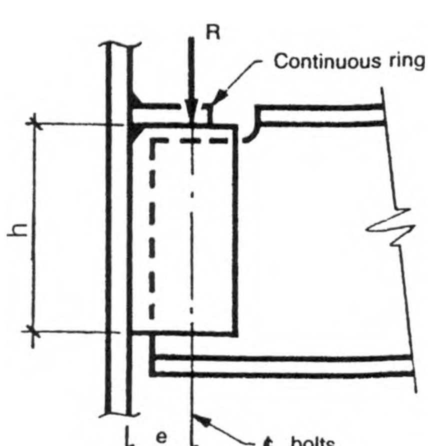


Figure 5-4. Typical clip support.

- Moment in clip, M

$M = R\,e$

- Thickness required, t_r

$t_r = (6\,M)/(h^2\,F_b)$

- Area required per bolt, A_r

$A_r = R / F_s\,n_b$

Use: n = ___________
Size: ___________
Material: ___________

2.0. Design of Beam Seat

- Thickness required, gusset, t_g

$T_g = (R(6\,e - 2\,a))/(F_b\,a^2\,Sin^2\,\phi)$

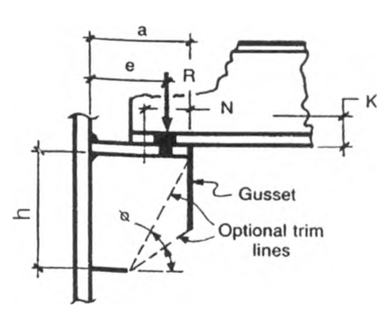


Figure 5-5. Typical beam seat support.

- Bearing length, N_b

  $N_b = [R/(.75\, t_W F_y)] - K_1$

- Ratio B;

  For E-60 Welds;

  $B = R/(23040\, W_f)$

  For E-70 Welds;

  $B = R/(26880\, W_f)$

  For other materials use $B = .384\, S_U$

- Required height of gusset, h_r

  $H_r = \left[.5B\left(B + (B^2 + 64\, e^2)^{1/2}\right)\right]^{1/2}$

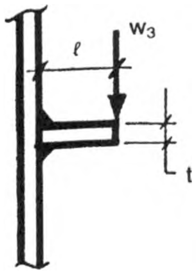


Figure 5-6. Loading diagram of a continuous ring.

3.0. Design of Ring

- Find uniform load, w_3, from table

Qty of Beams	W_3
1	P D / 4
2	P D / 6
3	P D / 8
4	P D /10
5	P D / 12
6	P D / 14
7	P D / 16
8	P D / 18

- Find moment in ring, M_3

  $M_3 = w_3 L$

- Thickness required, ring, t_r

  $t_r = (6\, M_3 / F_b)^{1/2}$

Select appropriate ring size.

4.0. Design of Grating

- Determine maximum span of grating, L_g

  $L_g =$

- Area of loading for a one foot wide panel, A_g

  $A_g = 12\, L_g$

- Total load on panel, F_g

  $F_g = pA_g$

- Uniform loading on one bearing bar, w_g

  $W_g = F_g/L_g\, n_g$

- Bending moment in one bearing bar, M_4

  $M_4 = w_g L_g/8$

- Required section modulus of one bearing bar, Z_r

  $Z_r = M_4/F_b$

- Actual properties of grating;

  $Z = (n_g\, b\, d^2)/6$

  $I = (n_g\, b\, d^3)/12$

  $\delta = \left[5\, p\, L_g (12\, L_g)^3\right] / 384\, E\, I$

Notes

1. Recommended beam ratio, span over depth should be between 15 and 18, 20 maximum.
2. For loading consider packing, catalyst, grating, weight of beams, liquid above packing or filter media, entrained liquid, and differential pressure. The weight of entrained liquid is equal to the volume of bed × the open area × specific gravity × 62.4.
3. Minimum gusset thickness of beam seat should not be less than the web thickness of the beam.
4. Main bearing bars of grating should run perpendicular to the direction of the support beams.
5. Make width of beam seat at least 40% of height.
6. Make fillet weld size no greater than .75 t_w.

BEAMS - LATERAL STABILITY CHECK

SOLID OR BUILT-UP BEAMS

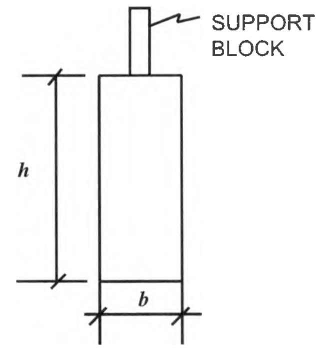
BEAM	L/h	F_c
1		
2		
3		
4		

Determine F_c from the following Table

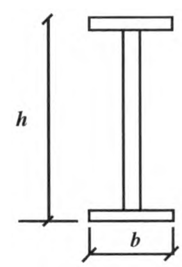
L/h	F_c
≤7	F_y
≥35	.66 [$(\pi^2 E)$ / 12 $(L/h)^2$]
7 < L/h < 35	Interpolate between values

NOTES:

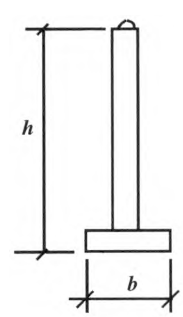
1. The allowable buckling stress, F_c, is a function of dimensions L and h.
2. If the allowable buckling stress, F, is exceeded there are two options;
   a. Redesign the beams
   b. Add anti-buckling devices
3. Anti-buckling devices consist of two types;
   a. Anti-buckling combs
   b. Web stiffeners
4. Anti-buckling devices effectively reduce the "L" dimension and thus increase the allowable stress.
5. Anti-buckling combs are plate devices which fit over the web of multiple beams and reinforce the web.



SOLID BEAM



BUILT-UP BEAM



T-BEAM

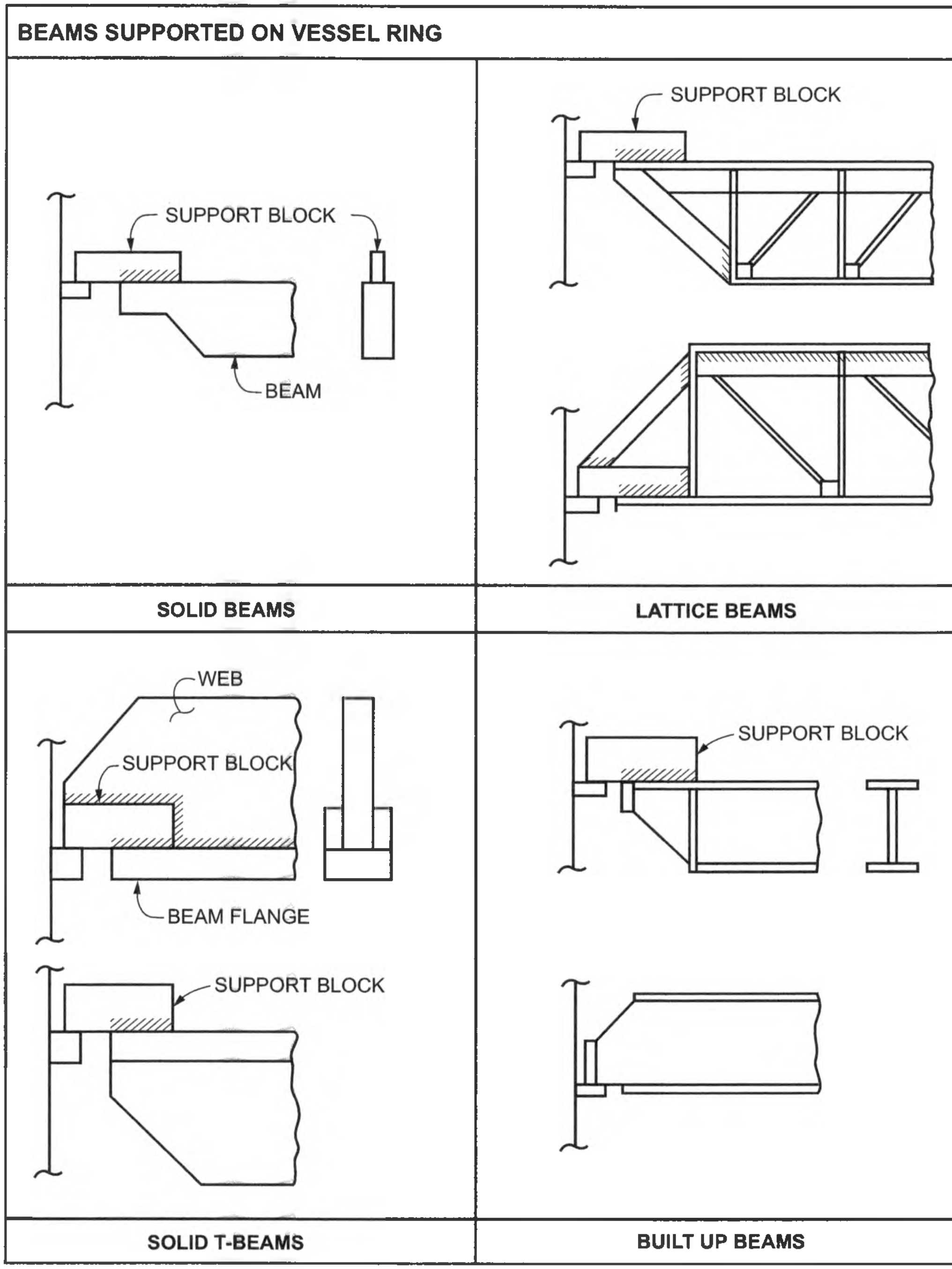
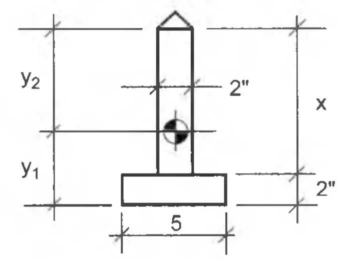

BEAMS SUPPORTED ON VESSEL RING
SUPPORT BLOCK
BEAM
SOLID BEAMS
SUPPORT BLOCK
LATTICE BEAMS
WEB
SUPPORT BLOCK
BEAM FLANGE
SUPPORT BLOCK
SOLID T-BEAMS
SUPPORT BLOCK
BUILT UP BEAMS

Table 5-2
Properties of heavy T-Beams

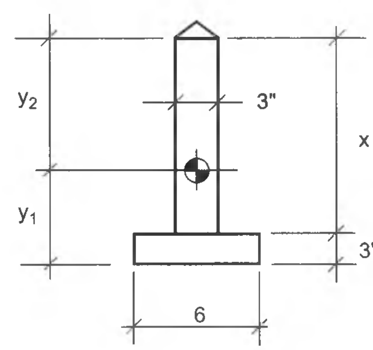
TYPE 1							
No.	X	A	y_1	y_2	I	Z_T	Z_C
1	8	26	4.07	5.93	242.5	59.6	40.9
2	12	34	5.94	8.06	637.2	107.27	79.1
3	16	42	7.86	10.14	1303	165.8	128.5
4	20	50	9.8	12.2	2305	235.2	189
5	24	58	11.75	14.24	3711	315.9	260.6
TYPE 2							
1	8	42	4.64	6.36	452.6	97.55	71.16
2	10	48	5.56	7.44	738.8	132.9	99.3
3	12	54	6.5	8.5	1120.5	172.4	131.8
4	14	60	7.45	9.55	1609.9	216.1	168.6
5	16	66	8.4	10.6	2224	264.7	209.8
6	18	72	9.375	11.625	2960	315.7	254.6
7	20	78	10.34	12.66	3844	371.8	303.6
8	21	81	10.83	13.17	4345	401	329.9
9	24	90	12.3	14.7	6094	495.4	415.6
10	26	96	13.28	15.72	7484	563.6	476.1
TYPE 3							
1	18	100	9.92	12.08	4421	445.6	365.9
2	20	108	10.88	13.12	5701	524	434.5
3	22	116	11.86	14.13	7179	605	508
4	24	124	12.83	15.16	8908	694	588
5	26	132	13.82	16.18	10856	786	671
6	28	140	14.8	17.2	13089	885	761
7	30	148	15.8	18.2	15607	989	857

FIGURES

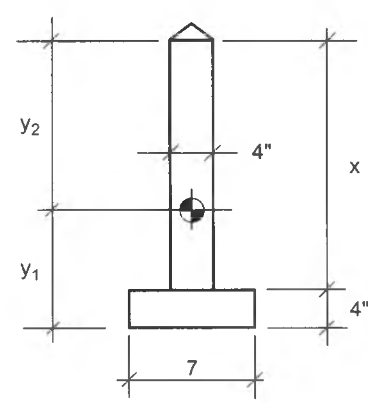
TYPE 1



TYPE 2



TYPE 3

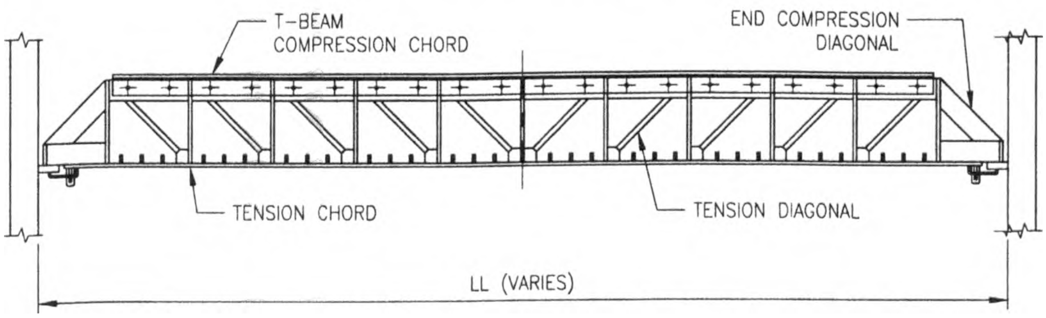


Notes:

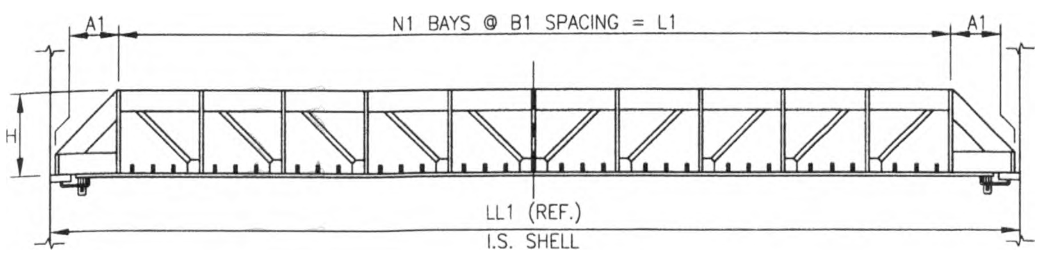
1. Z_T = Section Modulus for tension, = I / y_1.
2. Z_C = Section Modulus for compression, = I / y_2.
3. Designer should review that beams can be inserted through existing manway size.

Procedure 5-2: Design of Lattice Beams

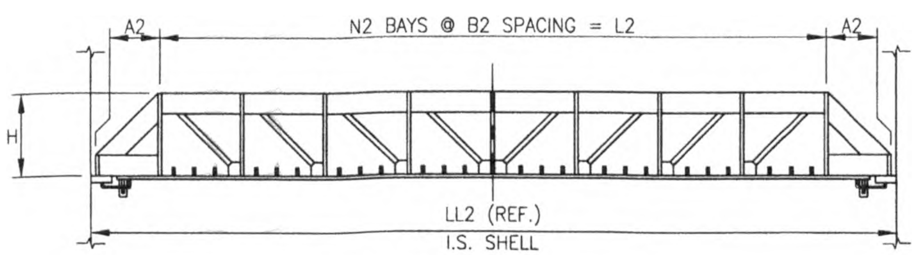
GENERAL CONFIGURATION



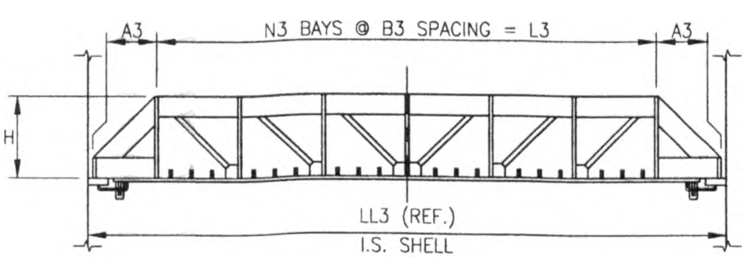
DIMENSIONS OF BEAMS



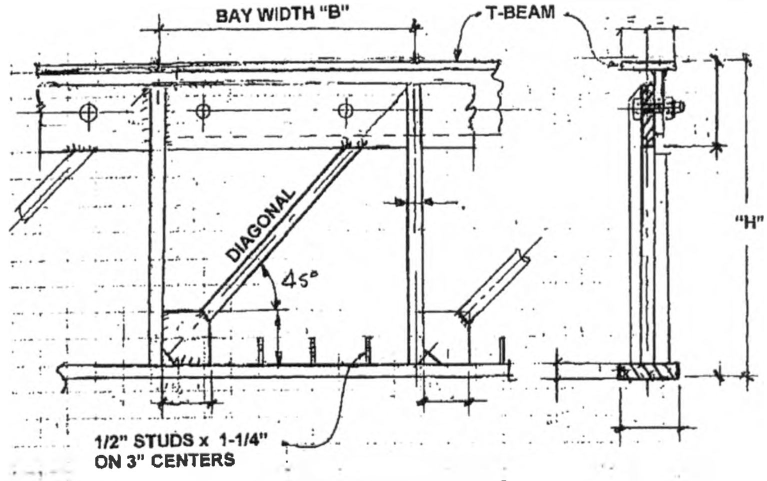
BEAM 1



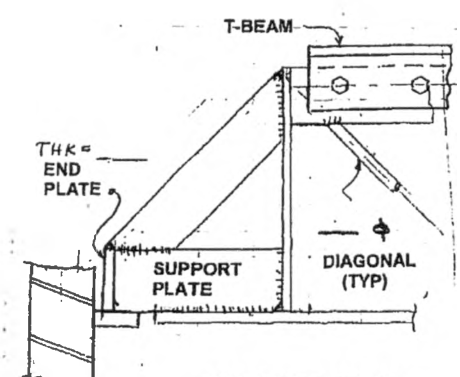
BEAM 2



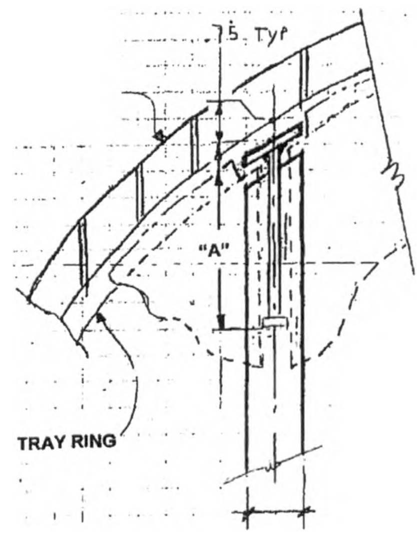
BEAM 3

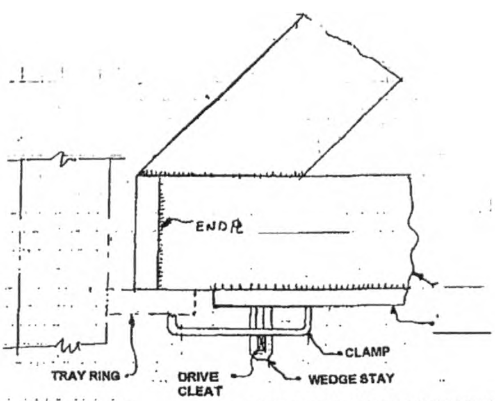


TYPICAL DIAGONAL BRACING

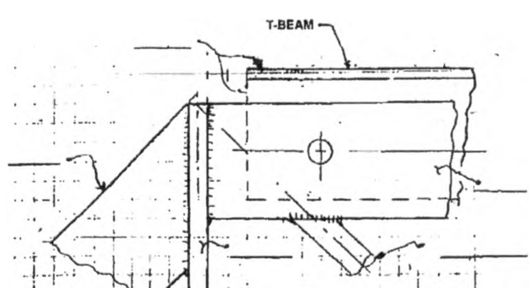


END CONNECTION





TYPICAL END CONNECTION DETAIL



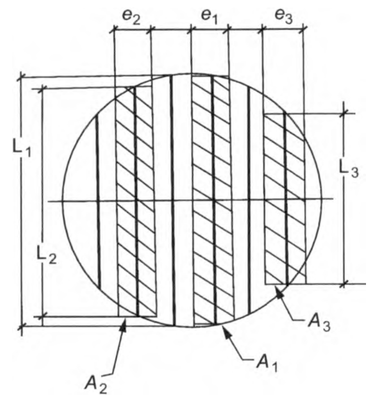
TYPICAL CORNER CONNECTION DETAIL

DESIGN LOADS FOR LATTICE BEAMS				
DATA			LOADINGS	
ITEM	SYMBOL	VALUE	ITEM	VALUE
Material, beams and grating			Dead load, DL	
Material, vessel shell			a. Weight of beams	
Quantity of beams	N		b. Weight of grating	
Vessel ID	D		c. Weight of tray	
Design temperature	DT		d. Weight of screen	
Differential pressure	ΔP		e. Weight of packing/catalyst	
Bed depth	d_C		Product load, PL	
Liquid/contents unit weight	w_c		a. Weight of liquid on tray	
Corrosion allowance	C_a		b. Weight of liquid in bed	
Specific gravity	S_g		c. Weight of liquid above bed	
Liquid holdup (%)			d. Weight of solids	
Free area in bed (%)	P_C		Live load, LL	
Packing/catalyst unit weight	w_P			
Volume of packing/catalyst	V_P		Total load, F, Lbs	
Packing/catalyst total wt	W_C		F = DL + PL + LL	
Weight of entrained liquid	W_e			
Weight of grating	W_g		Total cross sectional area, A_c	
Weight of beams	W_b		$A_c = \pi r^2$	
Miscellaneous				
MATERIAL PROPERTIES & ALLOWABLE STRESS			Uniform load, p, PSI	
Shell at design temperature	S		$p = F / A_c$	
	E			
	F_y			
Beams at design temperature	E			
	F_y			
Bending = .66 F_y	F_b			
Compression, dependent on KL/r ratio	F_C			
Shear = .4 F_y	F_S			
Bearing = .9 F_y	F_{br}			

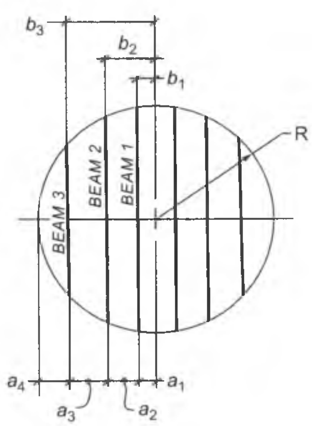
DESIGN OF LATTICE BEAMS

DIMENSIONS

(6) Beams shown for example Only!



DIMENSIONS



EQUATIONS

$e_1 = a_1 + .5\,a_2$

$e_2 = .5\,a_2 + .5\,a_3$

$e_3 = .5\,a_3 + .5\,a_4$

$e_4 = .5\,a_4 + .5\,a_5$

$L_n = 2\,[R^2 - b_n^2\,]^{1/2}$

$f_T = T_n \,/\, A_T$

$f_c = C_n \,/\, A_c$

$$C_c = \sqrt{\frac{2\,\pi^2 E}{F_y}}$$

$$F_c = \frac{\left[1 - \frac{(KL/r)^2}{2\,C_c^2}\right] F_y}{5\backslash 3 + \frac{3(KL/r)}{8\,C_c} - \frac{(KL/r)^3}{8\,C_c^3}}$$

1.0 AREA OF LOADING

BEAM	L_n	e_n	A_n
1			
2			
3			
4			

2.0 LOADS ON BEAMS

BEAM	F_n	R_n	w_n
1			
2			
3			
4			

3.0 MOMENT & FORCES

BEAM	M_n	h_n	T_n or C_n
1			
2			
3			
4			

4.0 BEAM STRESSES

BEAM	f_T	f_C	
1			
2			
3			
4			

5.0 ALLOWABLE STRESS, COMPRESSION

BEAM	L_{cn}	$K\,L_{cn}\,/\,r$	F_c
1			
2			
3			
4			

$A_n = L_n\,e_n$

$F_n = A_n\,p$

$w_n = F_n \,/\, L_n$

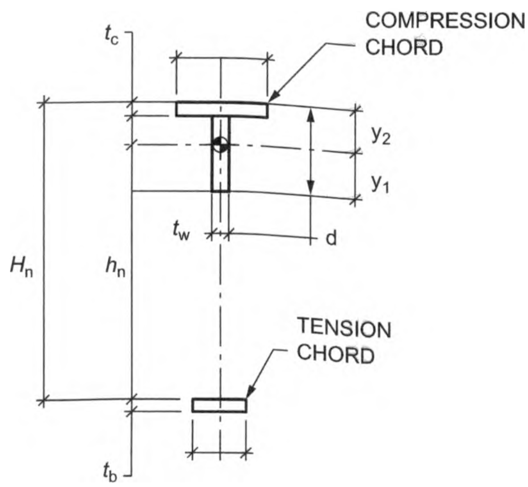
$R_n = F_n \,/\, 2$

$M_n = w_n\,L_n^2 \,/\, 8$

$T_n = C_n = M_n \,/\, h_n$

6.0. Properties of Beam

- Properties of compression chord;



DIMENSIONS OF LATTICE BEAM

Part	A	y	A y	A y^2	I
1					
2					
Σ					

$$y_1 = \Sigma A\, y / \Sigma A$$
$$y_2 = d - y_1$$
$$I = \Sigma A\, y^2 + \Sigma I - y_1\, \Sigma Ay$$
$$Z = I/y_1 \text{ or } Z_r = M_n/F_n$$
$$r = (I/\Sigma A)^{1/2}$$

7.0. Diagonals

- All diagonals are in tension
- Maximum load in diagonals, f_d

Beam	F_n	f_n	d_n
1			
2			
3			
4			

- Axial load in diagonal, tension, f_n

$$f_n = F_n/2 \sin\theta$$

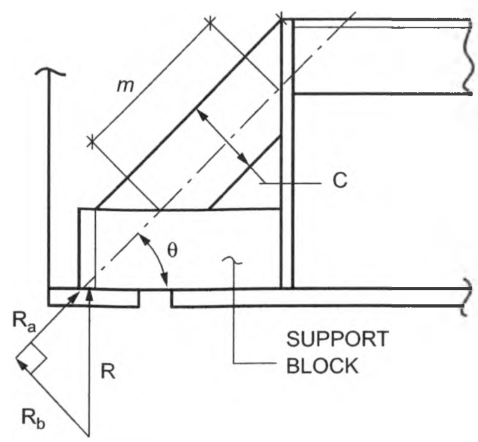
If $\theta = 45^{\circ}$, $f_n = .707\ F_n$

- Diameter of diagonal, d_{min}

$$d_{min} = [(4\ f_n)/(\pi\ F_T)]^{1/2}$$

8.0. Beam End Diagonal

- Determine Loads;



DIMENSIONS OF END CONNECTION

Beam	R_n	R_a	R_b
1			
2			
3			
4			

R_n = Reaction, Lbs

$R_a = R_n \sin\theta$

$R_b = R_n \cos\theta$

- Determine minimum gusset thickness, t_g

$$t_g = (6\ R_b\ m)/F_b\ C^2$$

Use ____________

- Determine properties;

Beam	m	Z	M
1			
2			
3			
4			

- Section modulus, Z

  $Z = (t_g C^2)/6$

- Moment, M

  $M = R_b m$

- Stress in end beam diagonal

Beam	f_a	f_b	F_C
1			
2			
3			
4			

$f_a = R_a/t_g C \leq F_C$

$f_b = M/Z \leq F_b$

$f_a/F_C + f_b/F_b \leq 1$

F_C is based on L/r; L/r = m/r

$r = C/12^{1/2} = .289C$

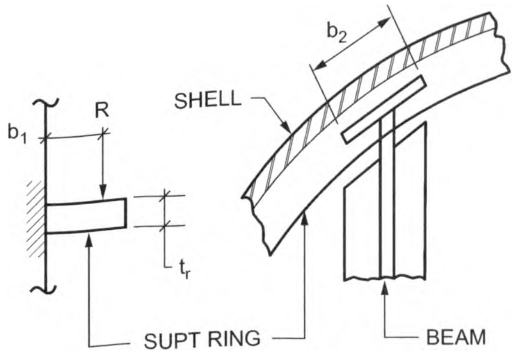
9.0. Stability Check

- The allowable buckling stress for the compression chord, F_{CB}, is dependent on the K L /r ratio (from AISC).

$K L_C / r$	Type	F_{CB}
<50	Short	F_y
50 TO 200	Intermediate	Interpolate
>200	Long	Not Recommended

- For $K L_C/r$ ratios between 50 and 200, the following apply;
  - If $K L_C/r < 4.71 (E/F_y)^{1/2}$
    Or $F_e \geq .44 F_y$
    Then $F_{CB} = (.658^{F_y/F_e}) F_y$
  - If $K L_C / r > 4.71 (E/F_y)^{1/2}$
    Or $F_e < .44 F_y$
    Then $F_{CB} = .877 F_e$
  - Where $F_e = (\pi^2 E I) / (K L_c)^2$

10.0. Ring Analysis



DIMENSIONS AND LOADS - SUPPORT RING

- Uniform load in support ring, w_S

  $w_S = R/b_2$

- Moment, M_S

  $M_S = w_S b_1$

- Bending stress, f_b

  $f_b = (6 M)/t_r^2 \leq F_b$

11.0. Notes

1. The allowable compressive stress in the compression chord should be the lesser of F_C or F_{CB}.
2. See procedure "Design of Internal Support Beds" for terms and definitions not shown.
3. The compression chord can be made without a T-section, with a bolted on T-section or as a built-up T-beam. This is entirely dependent on rigidity of the vertical member. A bolted on T-section is utilized where the assembly of tray plates or grating is difficult with the T-section in place.

Procedure 5-3: Shell Stresses due to Loadings at Support Beam Locations

Notation

A = Bearing area, in^2
D = Flexural rigidity, Lbs-in
E = Modulus of elasticity, PSI
F_y = Minimum specified yield strength at design temperature, PSI
f_{br} = Bearing stress, PSI
F_b = Allowable bending stress in shell, PSI
J_n = Effective width of shell acting at support point, in
M_{Ln} = Longitudinal moment at any point X or Y distance from point of applied load, in-Lbs
M_m = Moment in ring due to uniform load, w, in-Lbs
M_n = Longitudinal moment in shell due to axial load, P, in-Lbs/in
N_n = Minimum bearing length, in
P_n = Uniform compressive load, Lbs/in
R_n = Reaction, Lbs
t_r = Thickness of support ring required, in
w_n = Uniform load in ring, Lbs/in
X = Distance from applied load in axial (longitudinal) direction, in
Y = Distance from applied load in circumferential direction, in
y = Circumferential stress factor ratio
λ = Damping factor
σ_{XM} = Longitudinal membrane stress, PSI
$\sigma_{\phi M}$ = Circumferential membrane stress, PSI
σ_{XB} = Longitudinal bending stress, PSI
$\sigma_{\phi B}$ = Circumferential bending stress, PSI
ν = Poisson's ratio, .3 for steel

Equations

- Angle ϕ_n;

  $\sin \phi_n = m_n/R_i$ therefore $\phi_n =$

- Angle α_n;

  $\alpha_n = 90 - \phi_n$

- Minimum bearing length, N_n;

  $N_n = (R_n \sin \alpha_n)/[(.75\ F_y)\ K_n] < g$

- Effective width of shell, J_n;

  $J_n = [(K_n/\sin \alpha_n)] + 2\ e$

- Distance from centerline of shell to point of applied load, a;

  $a = .5\ t + e$

- Uniform compressive load, P_n, Lbs/in:

  $P_n = R_n/J_n$

- Longitudinal moment in shell due to axial load, P, in-Lbs

  $M_n = P_n\ a$

- Flexural rigidity, D:

  $D = (E\,t^3)/[12(1-\nu^2)] = 0.0916\ E\ t^3$

- Damping factor, λ;

  $\lambda = \left[3(1-\nu^2)/(R_m{}^2\ t^2)\right]^{1/4}$

  $\lambda = 1.285(R_m\ t)^{1/2}$

- Longitudinal moment at any point X distance from point of applied load, M_{Ln}, in-Lbs/in

  $M_{Ln} = M_n/2[e^{-\lambda X} \cos^{\lambda X}]$

  Note: At $X = 0$ and $Y = 0$, $M_{Ln} = M_n/2$

- Longitudinal membrane stress, σ_{XM} PSI

  $\sigma_{XM} = \pm P_n/t$

- Circumferential membrane stress, $\sigma_{\phi M}$ PSI

  $\sigma_{\phi M} = (y\ E)/R_m$

- Longitudinal bending stress, σ_{XB}, PSI

  $\sigma_{XB} = \pm (6\ M_n)/t^2$

- Circumferential bending stress, $\sigma_{\phi B}$, PSI

  $\sigma_{\phi B} = \pm (6\ \nu\ M_n)/t^2$

  $\sigma_{\phi B} = \pm (1.8\ M_n)/t^2$

  $\sigma_{\phi B} = \pm .3\sigma_{XB}$

- Uniform load in ring, w_n, Lbs/in

  $w_n = R_n (\cos \phi_n/K_n)$

- Moment in ring due to uniform load, w, M_m, in-Lbs/in

  $M_m = w_n\ e$

- Thickness of support ring required, t_r, in

  $t_r = [(6\ M_m)/F_b]^{1/2}$

  Use largest value of M_m

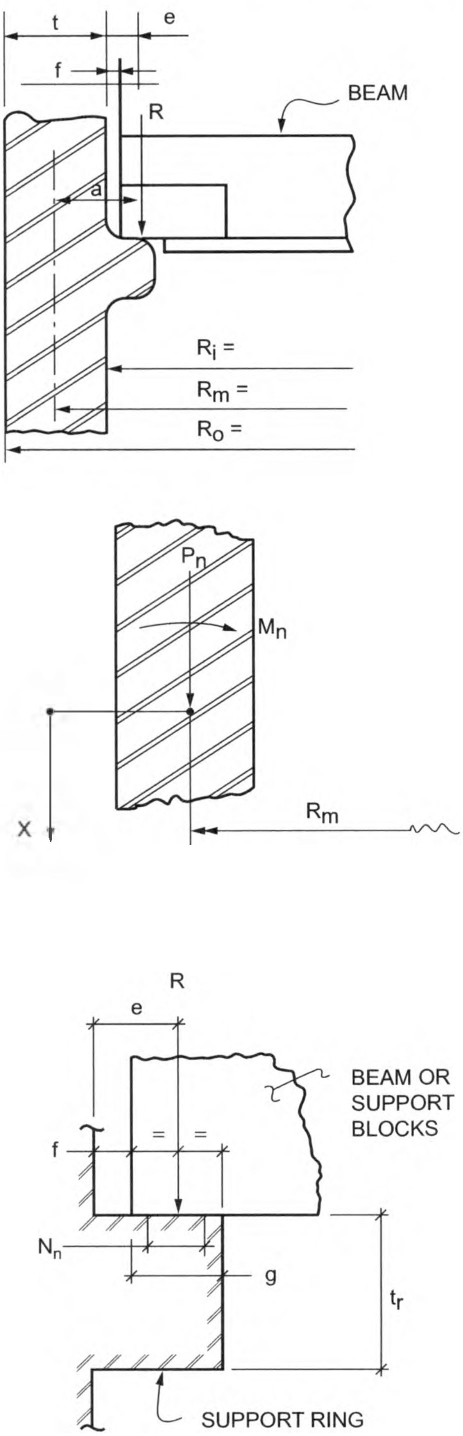
- Bearing area, A_b, in^2;

  $A_b = g\ K_n$

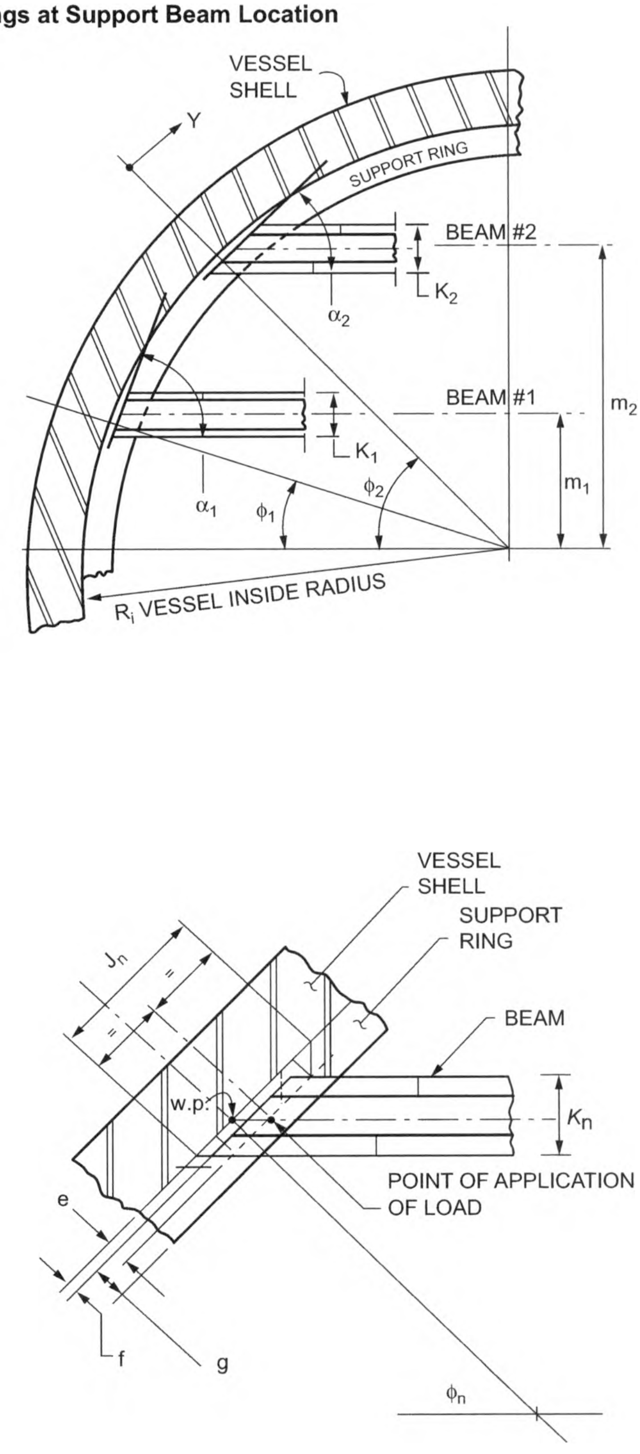
- Bearing stress, f_{br}, PSI

  $f_{br} = R_n/A_b$

Shell Stresses due to Loadings at Support Beam Location



CROSS SECTION OF VESSEL SHELL



PLAN VIEW OF VESSEL QUADRANT

LOADS AND STRESSES IN VESSEL SHELL AT SUPPORT POINT LOCATIONS

1.0 FIND ANGLES ϕ AND α

BEAM	m_n	ϕ_n	α_n
1			
2			
3			
4			

2.0 DIMENSIONS & LOADS

BEAM	R_n	N_n	J_n
1			
2			
3			
4			

3.0 LOADS & FACTORS

BEAM	P_n	M_n	y_n
1			
2			
3			
4			

4.0 MOMENTS & MEMBRANE STRESSES

BEAM	M_{Ln}	σ_{XM}	$\sigma_{\phi M}$
1			
2			
3			
4			

5.0 BENDING STRESSES

BEAM	σ_{XB}	$\sigma_{\phi B}$	
1			
2			
3			
4			

6.0 MAX UNIFORM LOAD IN RING

BEAM	R_n	w_n	M_{rn}
1			
2			
3			
4			

7.0 BEARING STRESS

BEAM	K_n	A_{bn}	f_{br}
1			
2			
3			
4			

Notes:

1. This procedure establishes one method for determining the shell stresses resulting from beams supported directly on a support ring, as opposed to supported off beam seats or clips.
2. This procedure assumes the beam is supported off "support blocks".

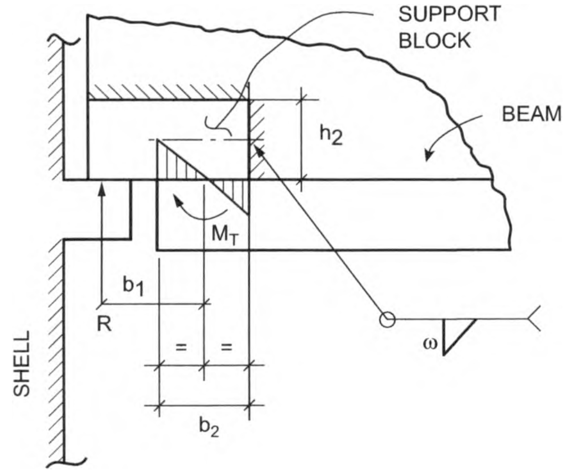
Procedure 5-4: Design of Support Blocks

Often times, when beams are supported on a support ring, the beam is supported by, or reinforced with a support block(s). This procedure shows how to design the support blocks and attachment welds for the various load cases. There are two types of support blocks used.

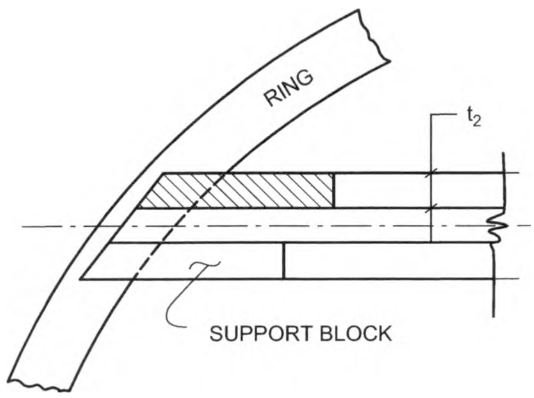
Type 1: Support blocks are welded integral with the web of the beam to expand the bearing area as well as reinforce the web in the area of local load. This type is used primarily with T-Type beams but could also be used with a built-up I beam.

Type 2: This type is used with solid beams where the top of the beam is located level with the top of the support ring to create a uniform support plane. The support block, also called a support plate, is welded to the top of the beam and cantilevers beyond the end of the beam to support the load.

Type 1: Support Blocks Welded Integral with Beam



Type 1-Loads and Dimensions of Support Block



Plan View Beam with Support Blocks

- Moment, M_T

$$M_T = .5\ R\ b_1$$

- Section modulus required, Z_r

$$Z_r = M_T/F_b$$

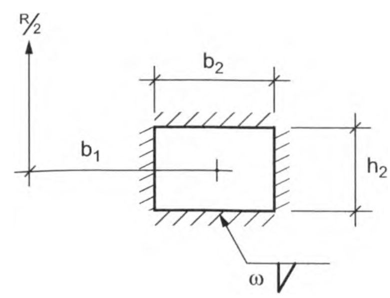
- Actual section modulus, Z

$$Z = (t_2\ h_2^2)/6$$

- Height of block required, h_r

$$h_r = [(6\ Z_r)/t_2]^{1/2}$$

- Maximum force on weld (treated as a line):



Loads on Welds of Support Blocks

- Polar moment of inertia of weld group, J_w

$$J_w = (b_2 + h_2)^3/6$$

- Shear on weld due to twisting moment in horizontal direction, f_{th}, and vertical direction, f_{tv}

$$f_{th} = M_T\ C/J_w = (M_T\ (0.5\ b_2))/J_w$$

$$f_{tv} = M_T\ C/J_w = M_T\ (0.5\ h_2)/J_w$$

- Area of weld group, A_w

$$A_w = 2b_2 + 2h_2$$

- Vertical shear on weld

$$f_{sv} = 0.5R/A_w$$

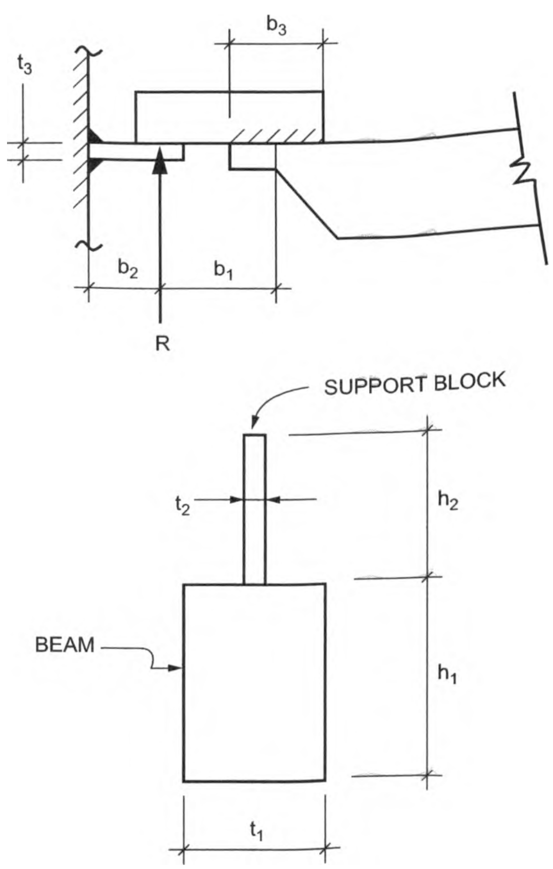
- Resultant shear force on weld, τ_r

$$f_r = \left(f_{th}^2 + (f_{tv} + f_{sv})^2\right)^{1/2}$$

- Size of fillet weld required, w

$$w = f_r/(0.707F_s)$$

Type 2: Support Block Welded to Top of Beam



Type 2 - Loads and Dimensions

- Moment in support block, M_1

$$M_1 = R\,b_1$$

- Required section modulus, Z_r

$$Z_r = M_1/F_b$$

- Actual section modulus, Z

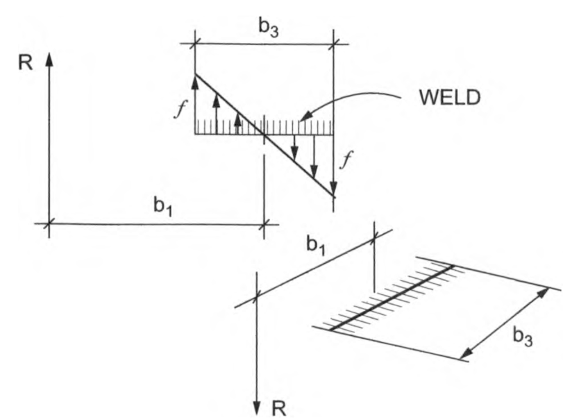
$$Z = \left(t_2\,h_2^2\right)/6$$

- Section modulus of weld group, Z_w

$$Z_w = b_3^2/3$$

- Maximum force on weld group, f

$$f = (R\,b_1)/(b_3^2/3)$$



Loads on Welds

- Size of fillet weld required, w

$$w = f/(0.707\,F_s)$$

Notes

1. An end plate can be welded to the end of the support block to stiffen the end of the plate and distribute the load on the ring.
2. See Procedure "Design of Internal Support Beds" for terms and definitions not shown.

Procedure 5-5: Hub Rings used for Bed Supports

The traditional way of supporting beds inside vessels has been to use straight beams. An alternative approach to the conventional straight beam design is to utilize a circular beam, commonly referred to as a "hub ring", supported by equally spaced radial beams. The radial beams are like the spokes of a wheel.

While this application may be new, the technology or method has been in use for many years. This type of support structure has long been recognized as a very efficient support structure. Efficiency in this case is the weight of the structure versus the weight supported. Hub rings can support up to 150 times their weight, while straight beams are in the range of 100.

The loadings demanded by industry have gone up dramatically over the years. As these loadings have gone up, the size of straight support beams has also increased dramatically. The design of the conventional straight beams is in some cases approaching physical limitations due to manway sizes required to insert or remove the beams.

As design loadings have increased, the spacing of beams has decreased. The height, width and thus weight of beams has increased, as well as the cost. This in turn has led to increasing the manway size to accommodate deeper beams while the access between the beams has become smaller and smaller. The hub ring can provide an innovative solution to this problem.

This type of support structure is ideal for pressure vessels because of the convenience of putting a round structure in a round vessel. The hub ring is comprised of two circular rings separated by the radial beams. The natural couple created by the loadings puts the top ring into compression and the lower ring in tension. The steel works great in tension and the circular beam is equally efficient at resisting the compressive loads. The advantage is achieved by utilizing the best properties of the material as well as the shape of the section. It is merely an application of the old adage, "form follows function".

In the hub ring design the radial beams are typically one quarter of the vessel diameter in length. Since the moment formula for a simply supported, uniformly loaded beam is $wl^2/8$, the beams are proportional, not to the length but to the square of the length. By comparison, straight beams have to span practically the whole diameter of the vessel, while radial beams only span ¼ of the diameter.

Another advantage, besides efficiency, weight, and cost, is safety. Not safety in design, but safety during maintenance periods. The design of straight beams drives the designer to increase the quantity of beams with a corresponding decrease in beam spacing. This reduces the loads on any individual beam, but in some cases narrows the passageway between the flanges of the beams to minimal levels. During plant shutdowns and maintenance periods, this reduction in access space can cost valuable time and create a hazard for maintenance personnel. When you consider the array of workers, ladders, lights, power cords, vacuum lines and loading socks that have to pass between these supports, this restriction is more than a major inconvenience. Since the hub ring is open in the center it can easily provide large access space.

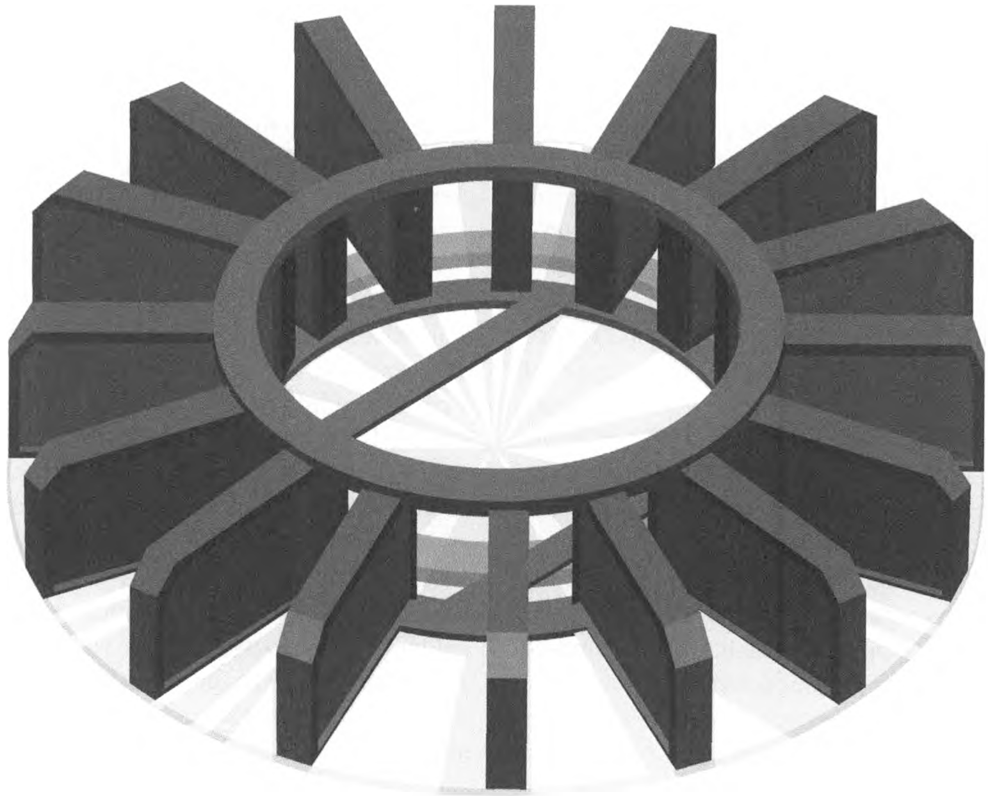
For vessels larger than about 10 feet in diameter, a cross beam, or multiple cross beams, may be required inside the hub ring to support tray decks or grating panels. Once again, these cross beams can be spaced such that more than adequate space is allowed for personnel access.

The ideal proportion for the centerline diameter of the hub rings is $D_m = 0.5\ D$.

The design is based on "N" number of equally spaced, radial loads. The upper ring is in compression. The lower ring is in tension.

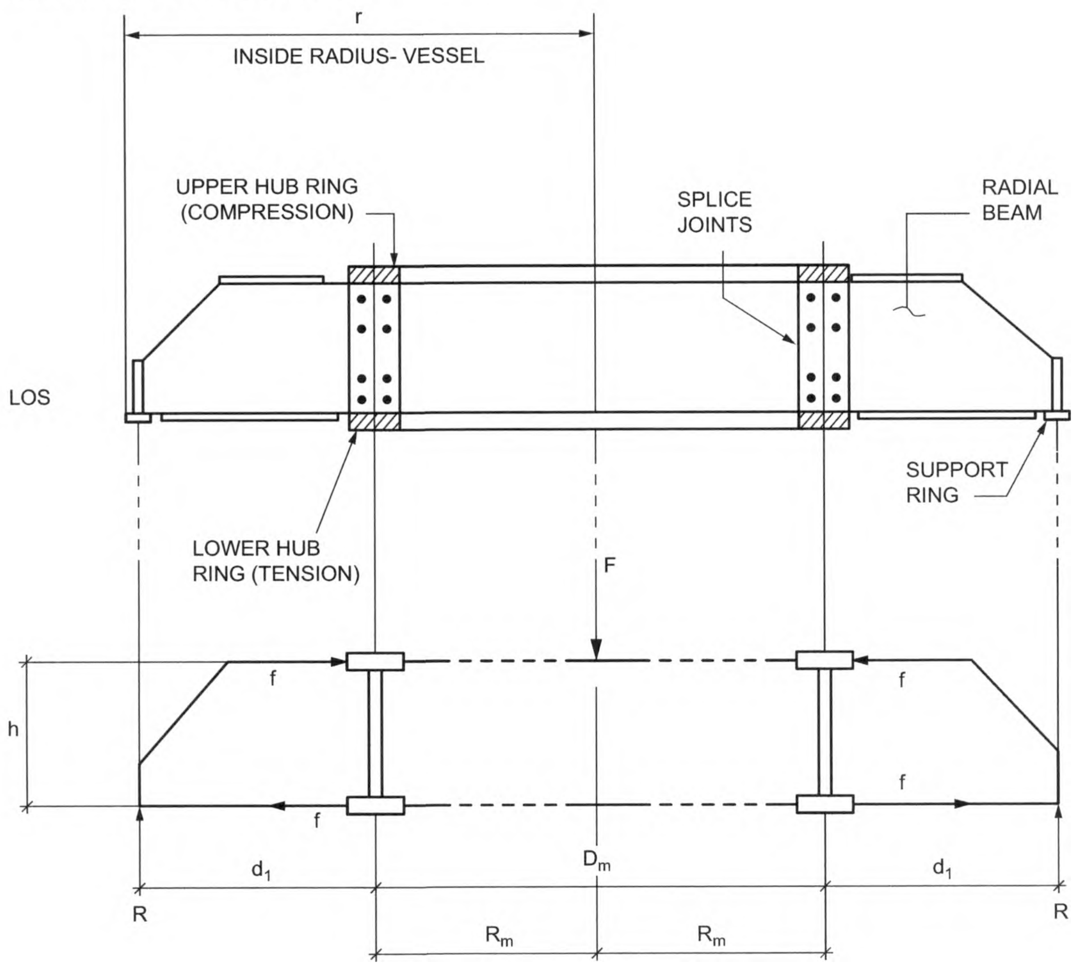
The entire assembly can be welded up and inserted in the vessel prior to installation of the top head or closure seam. Or, the entire thing can be made in pieces to pass through a manway. When made in pieces, the typical splice point is at the radial beams. Thus the quantity of spokes selected is typically a function of what will fit through a given manway size, rather than what loadings the components can handle.

Hub rings really have no limitations. They have been designed to support loads in excess of 1,000,000 pounds and diameters up to 60 feet, but not simultaneously, of course.

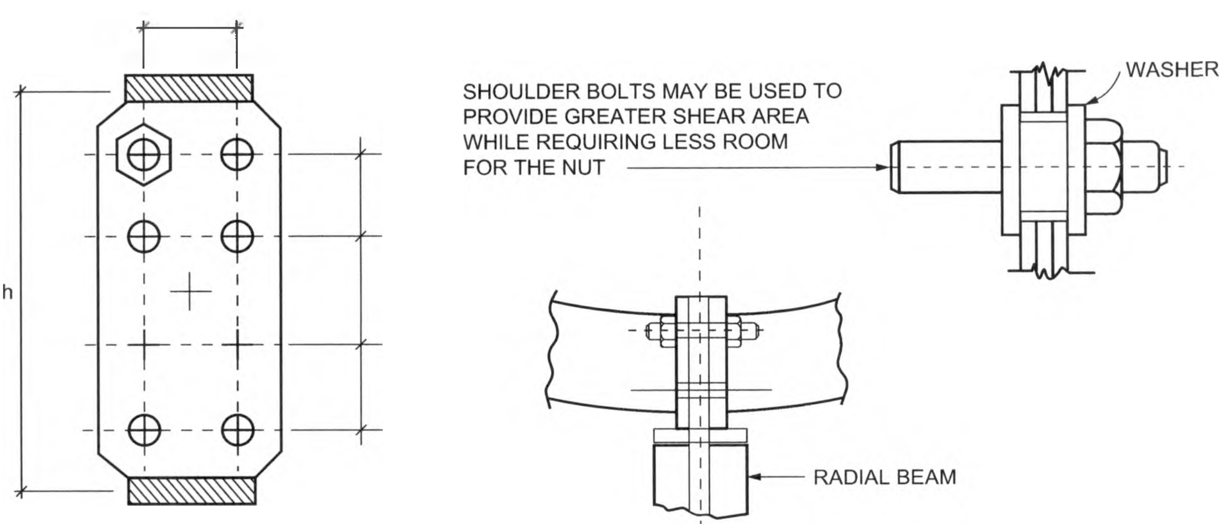


HUB RING WITH RADIAL BEAMS

GENERAL LOADS AND DIMENSIONS

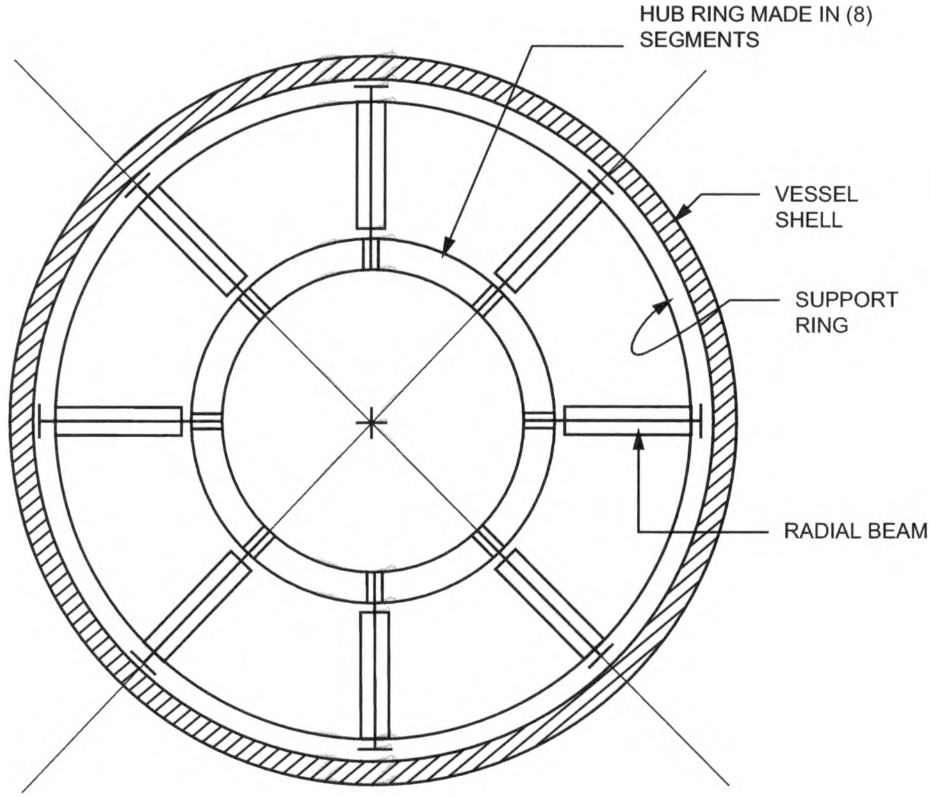


DETAILS OF BOLTED SPLICE CONNECTION

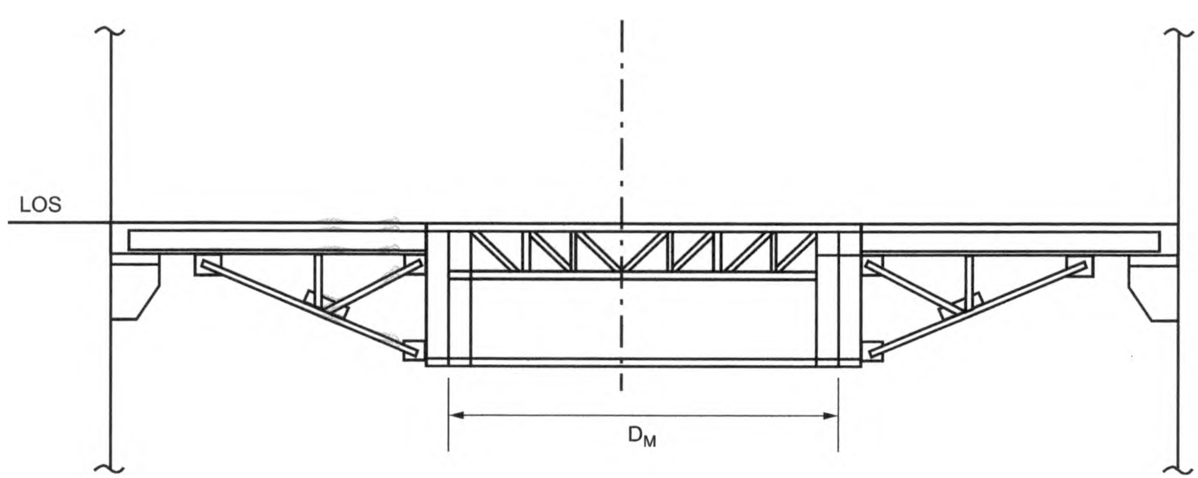


THE SPACING OF BOLTS SHOULD FOLLOW STD BOLTING CONVENTIONS

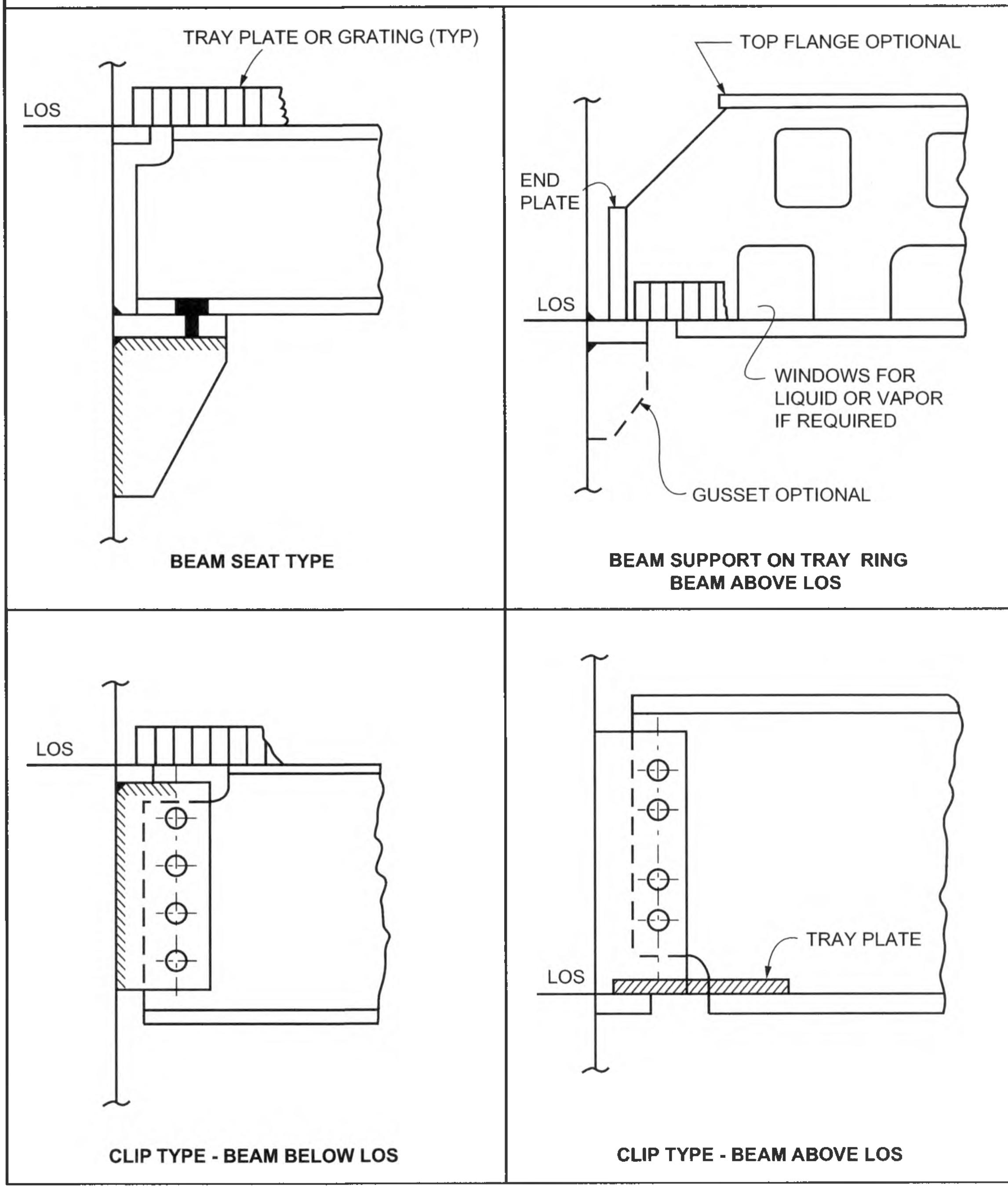
TYPICAL PLAN VIEW
(8) RADIAL BEAMS SHOWN ARE EXAMPLE ONLY



HUB RING SUPPORT FOR LARGE DIAMETER COLUMN
RADIAL BEAMS ARE LATTICE TYPE



SUPPORTS FOR RADIAL BEAMS



Procedure 5-6: Design of Pipe Coils for Heat Transfer [1-9]

This procedure is specifically for helical pipe coils in vessels and tanks. Other designs are shown for illustrative purposes only. Helical coils are generally used where large areas for rapid heating or cooling are required. Heating coils are generally placed low in the tank; cooling coils are placed high or uniformly distributed through the vertical height. Here are some advantages of helical pipe coils.

1. Lower cost than a separate outside heat exchanger.
2. Higher pressures in coils.
3. Fluids circulate at higher velocities.
4. Higher heat transfer coefficients.
5. Conservation of plot space in contrast with a separate heat exchanger.

Manufacture

Helical pipe coils can be manufactured by various means:

1. Rolled as a single coil on pyramid (three-roll) rolling machine. This method is limited in the pitch that can be produced. Sizes to 8 inches NPS have been accommodated, but 3 inches and less is typical. The coil is welded into a single length prior to rolling.
2. Rolled as pieces on a three-roll, pyramid rolling machine and then assembled with in-place butt welds. The welds are more difficult, and a trimming allowance must be left on each end to remove the straight section.
3. Coils can also be rolled on a steel cylinder that is used as a mandrel. The rolling is done with some type of turning device or lathe. The coil is welded into a single length prior to coiling. The pitch is marked on the cylinder to act as a guide for those doing the forming.
4. The most expensive method is to roll the pipe/tubing on a grooved mandrel. This is utilized for very small Dc-to-d ratios, usually followed by some form of heat treatment while still on the mandrel. Grooved mandrels create a very high-tolerance product and help to prevent flattening to some extent.

Coils are often rolled under hydro pressures as high as 85% of yield to prevent excessive ovalling of the pipe or tube. To accomplish this, the hydrotest pump is put on wheels and pulled along during the rolling process. End caps are welded on the pipe to maintain the pressure during rolling.

Stainless steel coils may require solution annealing after forming to prevent "springback" and alleviate high residual stresses. Solution heat treatment can be performed in a fixture or with the grooved mandrel to ensure dimensional stability.

Springback is an issue with all coils and is dependent on the type of material and geometry. This springback allowance is the responsibility of the shop doing the work. Some coils may need to be adjusted to the right diameter by subsequent rolling after the initial forming.

The straight length of pipe is "dogged" to the mandrel prior to the start of the rolling to hold the coil down to the mandrel. Occasionally it may be welded rather than dogged.

Applications for grooved mandrel are very expensive due to the cost of the machining of the mandrel. Mandrels that are solution heat treated with the coil are typically good for only one or two heat treatments due to the severe quench. Thus the cost of the mandrel must be included in the cost of the coil.

Design

There are two distinct aspects of the design of pipe coils for heat transfer. There is the thermal design and the physical design. The thermal design falls into three parts:

1. Determine the proper design basis.
2. Calculating the required heat load.
3. Computing the required coil area.

Physical design includes the following:

1. Selecting a pipe diameter.
2. Computing the length.
3. Determine the type of coil.
4. Location in the tank or vessel.
5. Detailed layout.

To determine the design basis, the following data must be determined:

1. Vessel/tank diameter.
2. Vessel/tank height.

3. Insulated or uninsulated.
4. Indoor or outdoor.
5. Open top or closed top.
6. Maximum depth of liquid.
7. Time required to heat/cool.
8. Agitated or nonagitated.
9. Type of operation.

The type of operation is characterized in the following cases:

1. Batch operation: heating.
2. Batch operation: cooling.
3. Continuous operation: heating.
4. Continuous operation: cooling.

Coils inside pressure vessels may be subjected to the internal pressure of the vessel acting as an external pressure on the coil. In addition, steam coils should be designed for full vacuum or the worst combination of external loads as well as the internal pressure condition. The coil must either be designed for the vessel hydrotest, externally, or be pressurized during the test to prevent collapse.

Pressure Drop

It is important that pressure drop be considered in designing a pipe coil. This will establish the practical limits on the length of pipe for any given pipe size. Large pressure drops may mean the coil is not capable of transmitting the required quantity of liquid at the available pressure. In addition, the fluid velocities inside the coil should be kept as high as possible to reduce film buildup.

There are no set rules or parameters for maximum allowable pressure drop. Rather, an acceptable pressure drop is related to the velocity required to effect the heat transfer. For liquids a minimum velocity of 1-3 feet per second should be considered. For gases "rho-V squared" should be maintained around 4000.

Pressure drop in helical coils is dependent on whether the flow is laminar or turbulent. Typically flows are laminar at low fluid velocities and turbulent at high fluid velocities. In curved pipes and coils a secondary circulation takes place called the "*double eddy*" or *Dean Effect*. While this circulation increases the friction loss, it also tends to stabilize laminar flow, thus increasing the "critical" Reynolds number.

In general, flows are laminar at Reynolds numbers less than 2000 and turbulent when Reynolds numbers are greater than 4000. At Reynolds numbers between 2000 and 4000, intermittent conditions exist that are called the *critical zone*.

For steam flow, the pressure drop will be high near the inlet and decrease approximately as the square of the velocity. From this relationship, combined with the effects of increased specific volume of the steam due to pressure drop, it can be shown that the average velocity of the steam in the coil is three-fourths of the maximum inlet velocity. For the purposes of calculating pressure drop, this ratio may be used to determine the average quantity of steam flowing within the coil.

Heat Transfer Coefficient, U

The heat transfer coefficient, U, is dependent on the following variables:

1. Thermal conductivity of metal, medium, and product.
2. Thickness of metal in pipe wall.
3. Fluid velocity.
4. Specific heat.
5. Density and viscosity.
6. Fouling factor (oxidation, scaling).
7. Temperature differences (driving force).
8. Trapped gases in liquid flow.
9. Type of flow regime (laminar versus turbulent, turbulent being better).

Notes

All of the following apply specifically to helical coils.

1. Overdesign rather than underdesign.
2. The recommended ratio of vessel diameter to pipe diameter should be about 30. However, it has been found that 2 inch pipe is an ideal size for many applications. Pipe sizes of 6 inches and 8 inches have been used.
3. Helical coils are concentric with the vessel axis.
4. Two or more coils may be used, with the recommended distance between the coils of two pipe diameters.
5. Seamless pipe is preferred. Schedule 80 pipe is preferred.

6. Limit maximum pitch to five pipe diameters, with 2 to 2½ recommended. Physical limits should be set between 4 inches minimum and 24 inches maximum.
7. Centerline radius of bends should be 10 times the pipe diameter minimum. (1 inch pipe = 10 inch centerline radius).
8. It is recommended for bend ratios over 5% or fiber elongation greater than 40% that the coils be heat treated after forming. The bend ratio can be computed as follows:

$$\frac{100\ t_p}{R}$$

9. Flattening due to forming should be limited to 10%. Some codes limit ovality to as little as 8%. Ovality may be computed as follows:

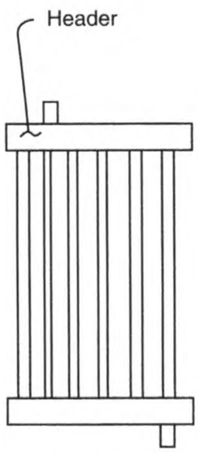
$$100\left(\frac{d_{max} - d_{min}}{d}\right)$$

10. Wall thinning occurs any time a pipe is bent. The inside of the bend gets thicker and the outside of the bend gets thinner. Typically this is not a problem because the outside of the bend that gets thinner will also experience a certain amount of work hardening that can make up for the loss of wall thickness. The tighter the bend, the greater the thinning. Anticipated wall thinning due to forming can be computed as follows:

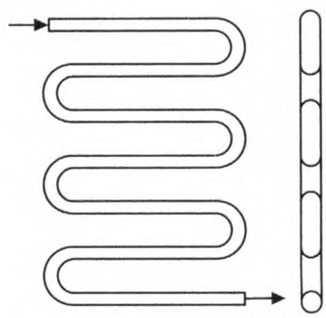
$$t_p\left(1 - \frac{R}{R + 0.5d_0}\right)$$

11. Distance between an internal coil and the side wall or bottom of the tank or vessel is a minimum of 8 inches and a maximum of 12 inches (dimension "c").
12. All coils should be evenly supported at a minimum of three places. Supports should be evenly spaced and allow for thermal expansion of the coil.
13. Coils should be sloped a minimum of ⅛ inch per foot to allow for drainage.
14. Certain flow rates in spiral coils can set up harmonic vibrations that could ultimately be destructive to the coil, supports, etc. In addition, slug flow can cause extreme coil movement. If vibration or movement becomes a problem, then either the flow rate or the coil support arrangement must be changed.
15. Limit velocity to 10 feet per second in coils.
16. The "steady-state" condition requires less coil than any other design condition.
17. If pressure drop is excessive, the coil may be split into multiple coils with manifolds or separate inlets or outlets.

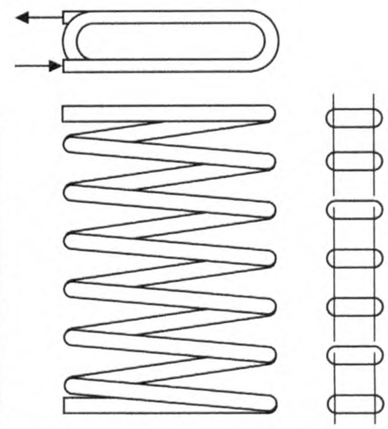
Types of Coils



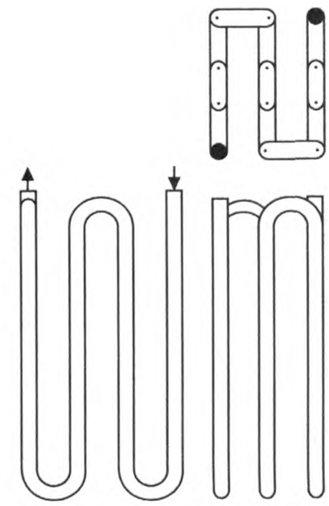
Harp
(flat)



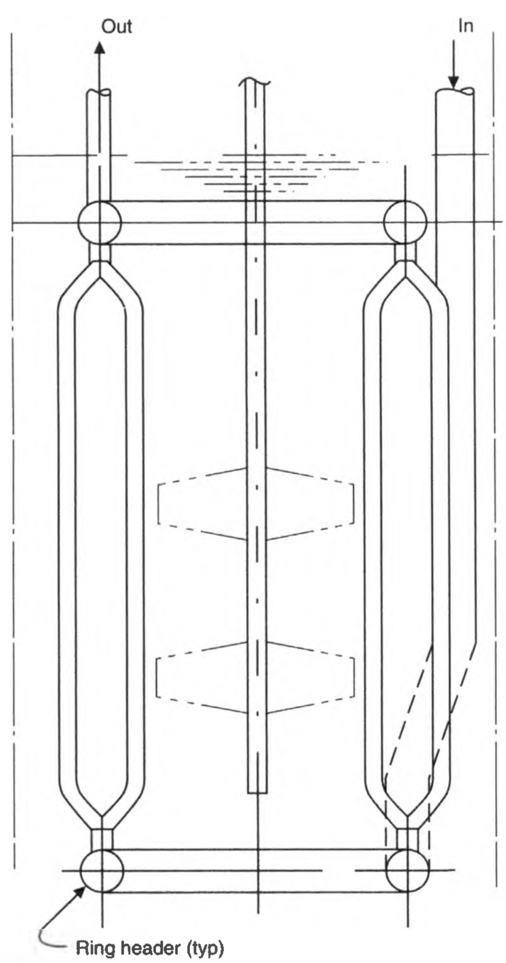
Flat
(serpentine)

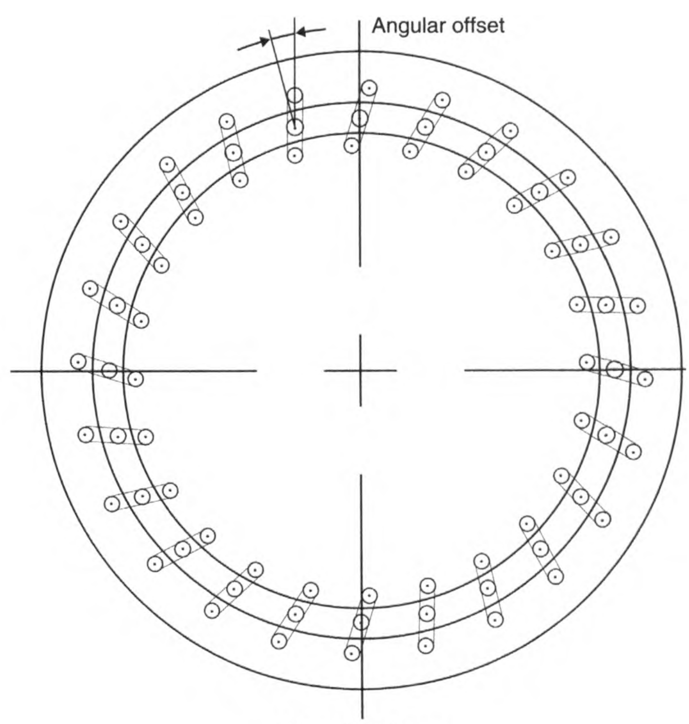


Oval



Box

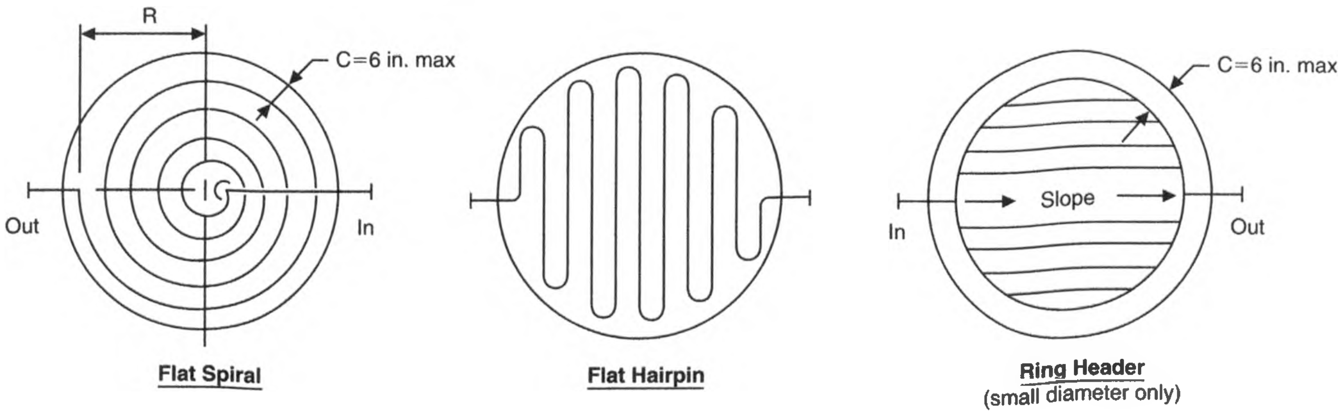




Circular Harp

Note:
Direction of flow will vary depending on the heating or cooling application

Coil Layout for Flat-Bottom Tanks



Developed length of flat spiral coils:

$$L_D = \frac{\pi R^2}{d_0 + C}$$

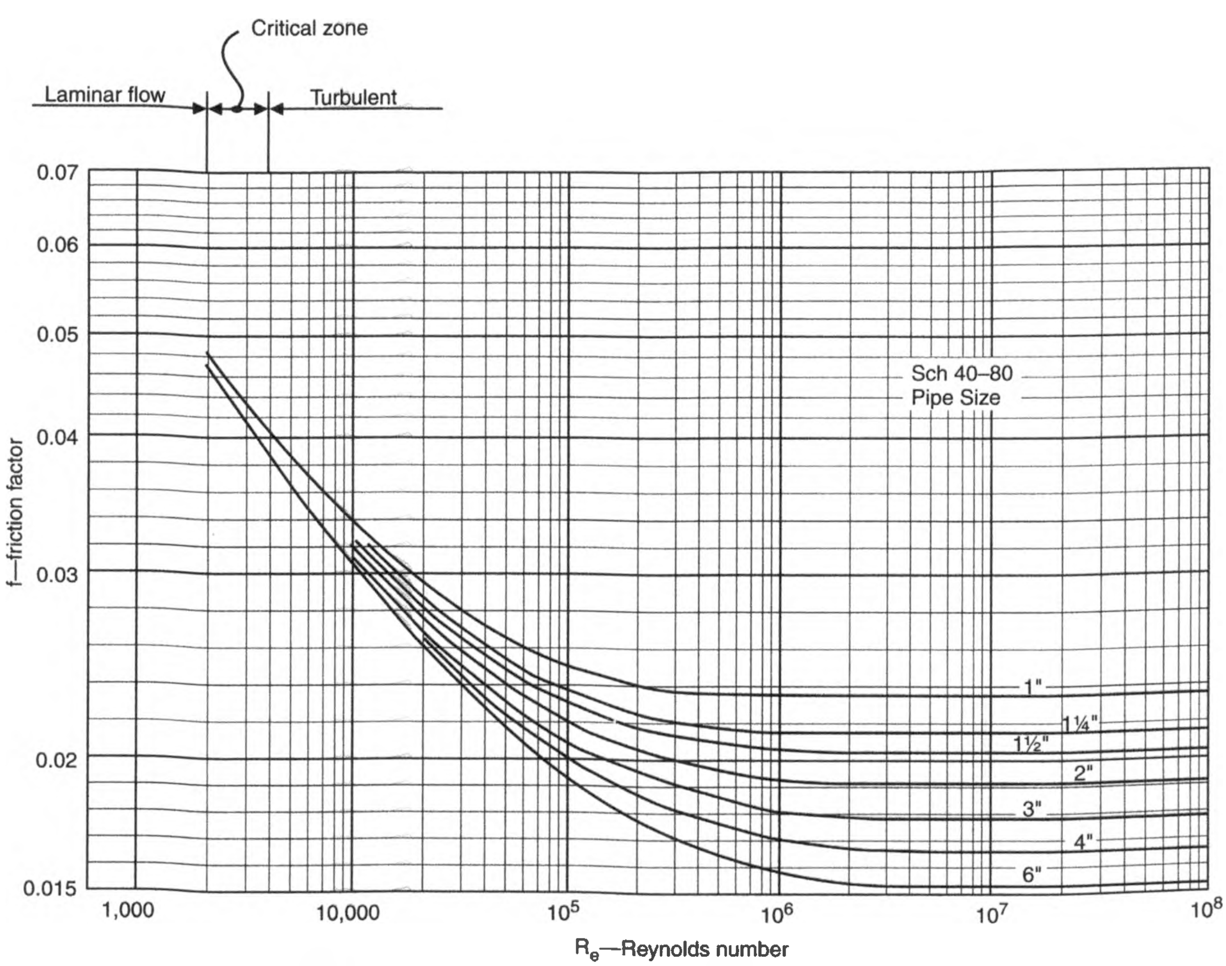
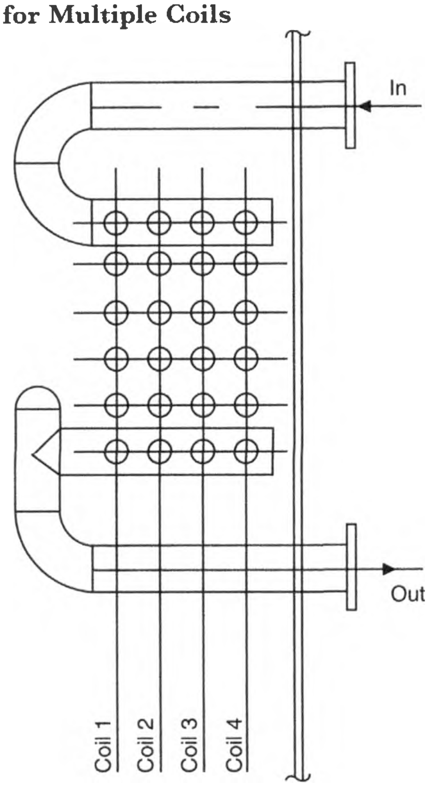


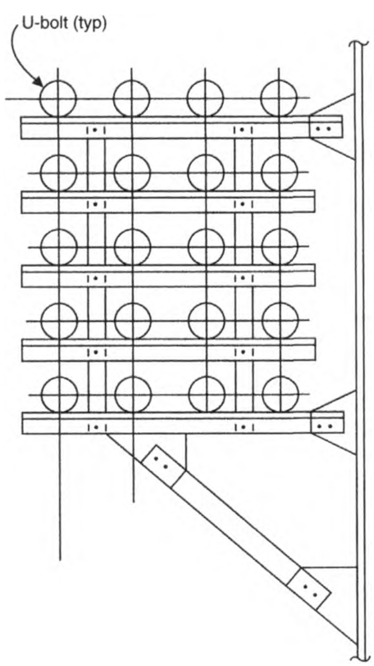
Figure 5-7. Friction factor, f, versus Reynolds number, R_e.

Coil Supports

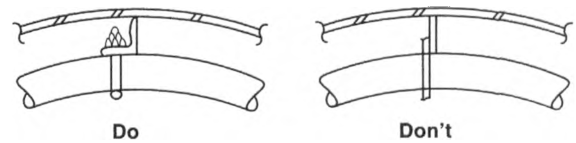
Manifold for Multiple Coils



Support for Multiple Coils

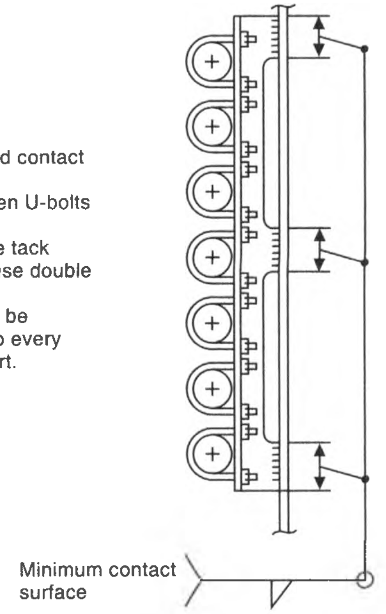


Support for Single Coil

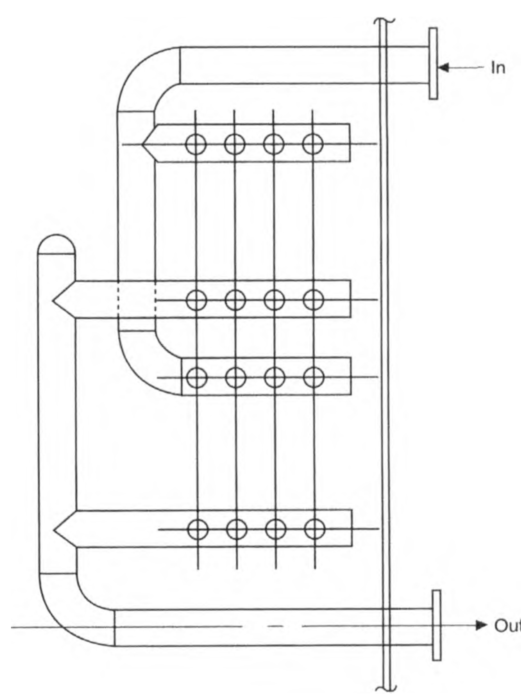


Notes:

1. Provide good contact surface.
2. Do not tighten U-bolts around coil.
3. Nuts may be tack welded, or use double nuts.
4. U-bolts may be alternated to every other support.



Manifold for Multiple Coils, Multiple Series

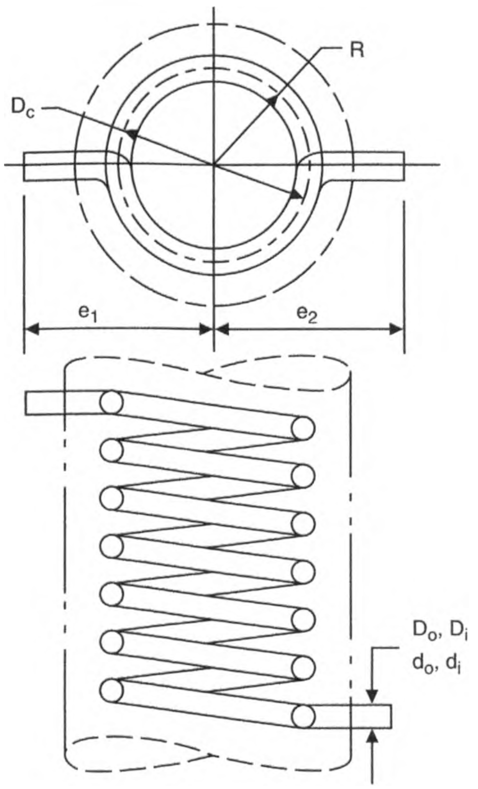


Design of Helical Coils

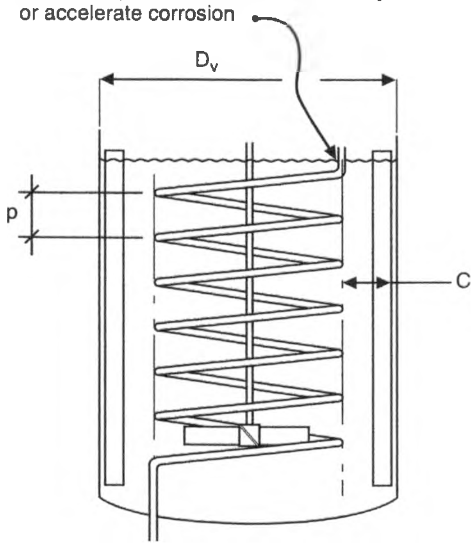
Notation

A = vessel surface area, ft^2
A_r = surface area of coil required, ft^2
C_p = specific heat of coil or vessel contents, BTU/lb/°F
D_c, d_c = centerline diameter of coil, ft (in.)
D_v = inside diameter of vessel, ft
D_o, D_i = OD/ID of pipe, ft
d_o, d_i = OD/ID of pipe, in.
E = enthalpy, latent heat of evaporation, BTU/lb
f = friction factor
F_{LF} = laminar flow factor
G = rate of flow or quantity of liquid to be heated or cooled, ft^3/hr
GTD = greatest temperature difference, °F
g = acceleration due to gravity, $4.17 \times 10^8 ft/hr^2$
g_c = gravitational constant, 32.2 lbm-ft/lbf-sec^2
h_o, h_i = film coefficients, BTU/hr-ft^2-°F
K = thermal conductivity of pipe, BTU-in/hr/ft^2/°F
L_r = minimum required length of coil, ft
L_a = developed length of coil, ft
LTD = least temperature difference, °F
M = mass flow rate, lb/hr
N = number of turns in coil
NPS = nominal pipe size of coil, in.
P = internal pressure in coil, psig
p = pitch of coil, in.
Q = total heat required, BTU/hr
Q_L = heat loss from vessel shell, BTU/hr
q_L = unit heat loss, BTU/hr
R_e = Reynolds number
S = external pipe surface area, ft^2
S_g = specific gravity of liquid
T = time required to heat or cool the vessel contents, hr
t_p = wall thickness of pipe, in.
t_1 = coil temperature, °F
t_2 = initial temperature of vessel contents, °F
t_3 = final temperature of vessel contents, °F
U = heat transfer coefficient, BTU/hr-ft^2-°F
V = velocity in coil, ft/sec
V_T = volume of vessel contents, ft^3
V_s = specific volume, equal to inverse of density, 1/w, ft^3/lb
W = rate of flow, lb/hr
w = density, lb/ft^3
ΔP = pressure drop, psi
ΔP_L = straight-line pressure drop, psi
ΔT = log mean temperature difference, °F
μ = viscosity, cP

Helical Coil with Baffles and Agitators

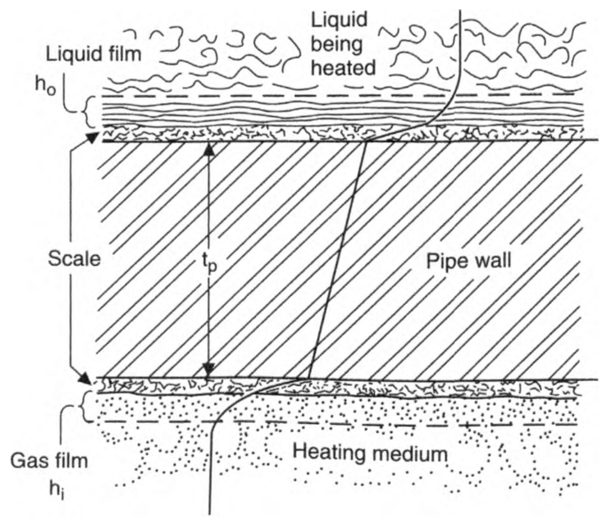


Caution: Splash zone on a hot coil may cause or accelerate corrosion



Calculations

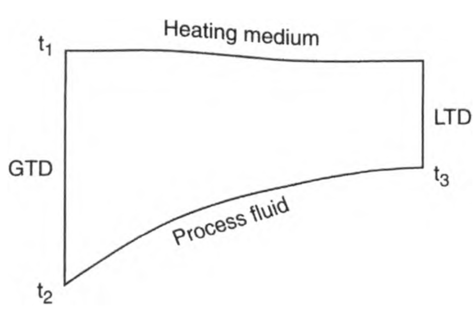
Solving for Heat Transfer Coefficient, U



The value of U can be taken from the various tables or calculated as follows:

$$U = \frac{1}{\frac{1}{h_O} + \frac{t_p}{K} + \frac{1}{h_i}}$$

Heating Applications



- *Determine mass flow rate, M.*

$$M = 62.4GS_g$$

- *Determine Δ T.*

$$GTD = t_1 - t_2$$
$$LTD = t_1 - t_3$$

$$\Delta T = \frac{GTD - LTD}{2.3 \ \log\left(\frac{GTD}{LTD}\right)}$$

- *Heat required, Q.*

$$Q = MC_p \, \Delta T + Q_L$$

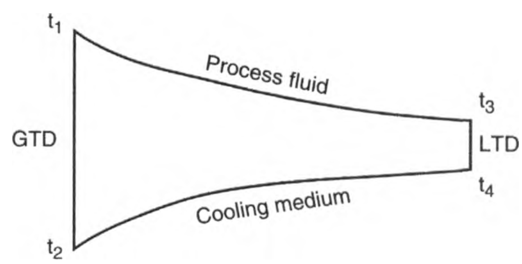
- *Area required, A_r.*

$$A_r = \frac{Q}{U \, \Delta T}$$

- *As an alternative, compute the time required, T.*

$$T = \frac{WC_pGTD}{A_rU\Delta T}$$

Cooling Applications



- *Cooling applications are equivalent to "heat recovery" types of applications. Only the "parallel" type is shown.*
- *Determine mass flow rate, M.*

$$M = 62.4GS_g$$

- *Determine ΔT.*

$$GTD = t_1 - t_2$$

$$LTD = t_3 - t_4$$

$$\Delta T = \frac{GTD - LTD}{2.3 \log\left(\frac{GTD}{LTD}\right)}$$

- *Heat required, Q.*

$$Q = MC_p \, \Delta T - Q_L$$

Subtract heat losses to atmosphere from heat to be recovered.

- *Area required, A_r.*

$$A_r = \frac{Q}{U\,\Delta T}$$

- *As an alternative, compute the time required, T.*

$$T = \frac{WC_pGTD}{A_rU\,\Delta T}$$

Coil Sizing

- *Make first approximate selection of nominal pipe size, NPS.*

$$NPS = \frac{D_v}{30}$$

Preliminary selection:_________

Pipe properties: d_i =_________

D_i =

S =

- *Determine length of coil required, L_r.*

$$L_r = \frac{A_r}{S}$$

- *Check minimum centerline radius, R.*

$$R > 10\ NPS$$

- *Select a pitch of coil, p.* Note: Pitch should be 2 to 2.5 × NPS.

Use p =_______

- *Determine the number of turns required, N.*

$$N = \frac{L_r}{\sqrt{(\pi D_c)^2 + p^2}}$$

Use N =_______

- *Developed length, L_a.*

$$L_a = N\sqrt{(\pi D_c)^2 + p^2}$$

Reynolds Number

- *For steam heating coils.*

1. Given Q, determine the rate of flow, W:

$$W = \frac{Q}{E}$$

2. Reynolds number, R_e:

$$R_e = \frac{6.31W}{d_i\mu}$$

- *For other liquids and gases.*

1. Find velocity in coil, V:

$$V = \frac{0.0509WV_S}{d_i^2}$$

2. Reynolds number, R_e:

$$R_e = \frac{123.9d_iVw}{\mu}$$

- *Find R_e critical.*

For coils, the critical Reynolds number is a function of the ratio of pipe diameter to coil diameter, computed as follows:

$$R_e\ \text{critical} = 20,000\left(\frac{D_i}{D_C}\right)^{0.32}$$

The critical Reynolds number can also be taken from the graph in Figure 5-9.

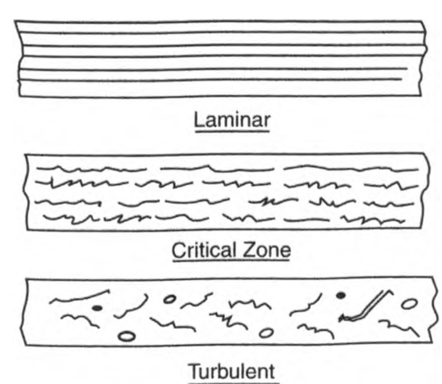


Figure 5-8. Various flow regimes.

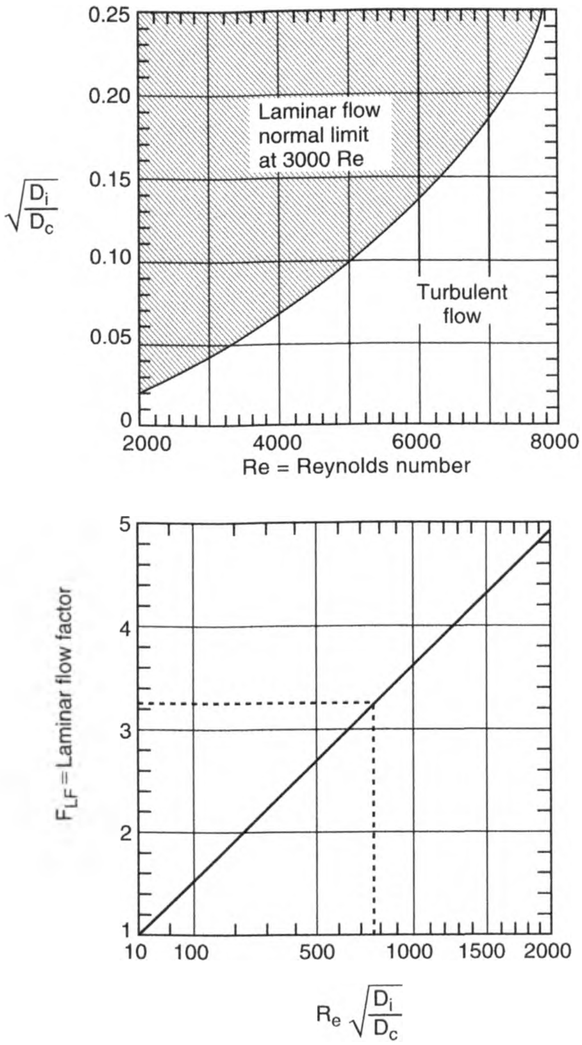


Figure 5-9. Pressure drop factors for flow-through coils. From ASME Transaction Journal of Basic Engineering, Volume 81, 1959, p. 126.

Pressure Drop

- If steam is the heating medium, the pressure drop of condensing steam is;

$$\Delta P = \frac{2fL_aV^2}{3gD_i}$$

The units are as follows;

f = 0.021 for condensing steam
L_a is in feet
V is in ft/hr
g is in ft/hr^2
D_i is in ft

- For other fluids and gases;
  a. If flow is laminar,

$$\Delta P_L = \frac{0.00000336fL_aW^2}{d_i^4 w}$$

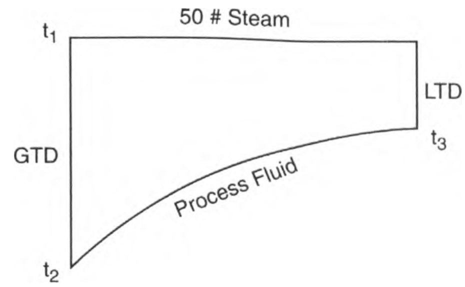
$$\Delta P = \Delta P_L(F_{LF})$$

  b. For turbulent flow,

$$\Delta P_L = \frac{0.00000336fL_aW^2}{d_i^5 w}$$

$$\Delta P = \Delta P_L\sqrt{R_e\left(\frac{d_i}{d_c}\right)^2}$$

Sample Problem 1



Heating Coil: Steam to Oil

- *Batch process.*
- *No agitation (other than natural circulation).*
- *Coil material = carbon steel.*
- *Properties:*

Steam:

$V_s = 6.7$

$E = 912$

$\mu = 0.015$

Oil:

$C_p = 0.42$

$S_g = 0.89$

Vessel:

8 – ft diameter × 30 – ft tan – tan

Liquid height = 15 ft

Volume to liquid height : 700 ft^3 = 5237 gallons

Temperatures:

$t_1 = 300°F$

$t_2 = 60°F$

$t_3 = 200°F$

T = time to heat = 1 hr

- *Log mean temperature difference, ΔT.*

$GTD = t_1 - t_2 = 300 - 60 = 240$

$LTD = t_1 - t_3 = 300 - 200 = 100$

$$\Delta T = \frac{GTD - LTD}{2.3 \quad \log\left(\frac{GTD}{LTD}\right)} = \frac{240 - 100}{2.3 \quad \log\left(\frac{240}{100}\right)}$$

$= 160°F$

- *Quantity of liquid to be heated, G.*

For batch process: $G = \frac{V_T}{T} = \frac{700}{1} = 700 \text{ ft}^3/\text{hr}$

- *Mass flow rate, M.*

$M = 62.4GS_g = 62.4(700)0.89 = 38{,}875 \text{ lb/hr}$

- *Heat required, Q.*

$Q = MC_p\Delta T + Q_L = 38{,}875(0.42)160 + 0$

$= 2{,}612{,}413 \text{ BTU/hr}$

- *Heat transfer coefficient, U.*

U = from Table 5-12: 50–200

from Table 5-13: 20–25
from Table 5-14: 35–60
by calculation: 10–180
Use U = 40.

Table 5-3
Pipe data

Size (in.)	Schedule	d_i (in.)	D_i (ft)	S (ft^2/ft)
1	40	1.049	0.0874	0.344
	80	0.957	0.0797	
1.25	40	1.38	0.115	0.435
	80	1.278	0.1065	
1.5	40	1.61	0.1342	0.497
	80	1.5	0.125	
2	40	2.067	0.1722	0.622
	80	1.939	0.1616	
3	40	3.068	0.2557	0.916
	80	2.9	0.2417	
4	40	4.026	0.3355	1.178
	80	3.826	0.3188	
6	40	6.065	0.5054	1.734
	6	5.761	0.4801	

Table 5-4
Film coefficients

	Medium	Film Coefficient, h_o or h_i
No Change	Water	150–2000
	Gasses	3–50
	Organic solvents	60–500
	Oils	10–120
Condensing	Steam	1000–3000
	Organic solvents	150–500
	Light oil	200–400
	Heavy oil	20–50
Evaporation	Water	1000–2000
	Organic solvents	100–300
	Light oil	200–300
	Heavy oil	100–200

- *Area of coil required, A_r.*

$$A_r = \frac{Q}{U\Delta T} = \frac{(2{,}612{,}413)}{40(160)} = 408 \text{ ft}^2$$

- *Determine the physical dimensions of the coil.*

$NPS = \frac{D_v}{30} = \frac{96}{30} = 3.2$ Use 3-in. pipe

$C = 12$

Table 5-5
Properties of gases

Material	w	C_p 32°F	C_p 212°F	C_p 932°F
Air	0.0808	0.241	0.242	0.245
Ammonia	0.0482	0.52	0.54	
Benzene		0.22	0.33	0.56
Oxygen	0.0892	0.22	0.225	0.257
Nitrogen	0.0782	0.25	0.25	0.27
Methane	0.0448	0.53	0.6	0.92
Ethane	0.0848	0.4	0.5	0.84
Butane	0.1623	0.375	0.455	0.81
Propane	0.1252	0.38	0.46	0.82
Ethylene	0.0783	0.36	0.45	0.72
CO	0.0781	0.25	0.26	0.27
CO_2	0.1235	0.2	0.21	0.26
Steam			0.453	0.507

Table 5-6
Thermal conductivity of metals, K, BTU-in/hr/ft^2/°F

	Temperature, °F								
Material	**200**	**300**	**400**	**500**	**600**	**700**	**800**	**900**	**1000**
Alum—1100-0 annealed	1512	1488	1476	1464	1452	1440	1416		
Alum—6061-0	1224	1236	1248	1260	1272	1272	1272		
Alum—1100 tempered	1476	1464	1452	1440	1416	1416	1416		
Alum—6061-T6	1392	1392	1392	1392	1392	1380	1368		
Carbon steel	360	348	336	324	312	300	288	276	
C-½Mo	348	336	324	312	800	300	288	276	
1Cr–½Mo	324	324	312	300	288	288	276	252	252
2¼Cr–1Mo	300	288	276	276	264	264	252	252	240
5Cr–½Mo	252	252	252	240	240	240	240	228	228
12Cr	168	180	180	180	192	192	192	192	204
18–8 SST	112	118	120	132	132	144	144	156	156
25–20 SST	94	101	107	114	120	132	132	144	144
Admiralty brass	840	900	948	1008	1068				
Naval brass	852	888	924	960	996				
90Cu–10Ni	360	372	408	444	504	564	588	612	636
80Cu–20Ni	264	276	300	324	348	372	408	444	480
70Cu–30Ni	216	228	252	276	300	324	360	396	444
Monel	180	180	192	192	204	216	216	225	240
Nickel	456	432	396	372	348	336	336	348	372
Inconel/incoloy	113	116	119	120	120	132	132	132	144
Titanium	131	128	125	125	126				

Table 5-7
Properties of steam and water

Saturated Steam					Water		
P (PSIG)	**Temp. (°F)**	**V_s (ft^3/lb)**	**E (BTU/lb)**	**μ (centipoise)**	**Temp. (°F)**	**V_s (ft^3/lb)**	**μ (centipoise)**
5	227	20	961	0.014	32	0.0160	1.753
10	240	16.5	952	0.014	40	0.0160	1.5
15	250	14	945	0.014	50	0.0160	1.299
20	259	12	940	0.015	60	0.0160	1.1
25	267	10.5	934	0.015	70	0.0161	0.95
30	274	9.5	929	0.015	80	0.0161	0.85
35	281	8.5	924	0.015	90	0.0161	0.75
40	287	8	920	0.015	100	0.0161	0.68
45	292	7	915	0.015	150	0.0163	0.43
50	298	6.7	912	0.015	200	0.0166	0.3
75	320	4.9	895	0.016	250	0.0170	0.23
100	338	3.9	881	0.016	300	0.0175	0.18
125	353	3.2	868	0.017	350	0.0180	0.15
150	366	2.7	857	0.018	400	0.0186	0.13
200	388	2.1	837	0.019			
250	406	1.75	820	0.019			
300	422	1.5	805	0.02			

Table 5-8
Properties of liquids

Material	S_g	C_p	w
Water	1	1	62.4
Light oils	0.89	0.42	55.5
Medium oils	0.89	0.42	55.5
Bunker "C"	0.96	0.4	59.9
#6 Fuel oil	0.96	0.4	59.9
Tar/asphalt	1.3	0.4	81.1
Molten sulfur	1.8	0.2	112.3
Molten paraffin	0.9	0.62	56.2

Table 5-9
Viscosity of steam and water, in centipoise, μ

°F	1 psia	2 psia	5 psia	10 psia	20 psia	50 psia	100 psia	200 psia	500 psia	1000 psia	2000 psia	5000 psia	7500 psia	10000 psia	12000 psia
saturated steam	0.667	0.524	0.388	0.313	0.255	0.197	0.164	0.138	0.111	0.094	0.078	...	...	...	...
saturated water	0.010	0.010	0.011	0.012	0.012	0.013	0.014	0.015	0.017	0.019	0.023	...	...	...	...
1500°	0.041	0.041	0.041	0.041	0.041	0.041	0.041	0.041	0.042	0.042	0.042	0.044	0.046	0.048	0.050
1450	0.040	0.040	0.040	0.040	0.040	0.040	0.040	0.040	0.040	0.041	0.041	0.043	0.045	0.047	0.049
1400	0.039	0.039	0.039	0.039	0.039	0.039	0.039	0.039	0.039	0.040	0.040	0.042	0.044	0.047	0.049
1350	0.038	0.038	0.038	0.038	0.038	0.038	0.038	0.038	0.038	0.038	0.039	0.041	0.044	0.046	0.049
1300	0.037	0.037	0.037	0.037	0.037	0.037	0.037	0.037	0.037	0.037	0.038	0.040	0.043	0.045	0.048
1250	0.035	0.035	0.035	0.035	0.035	0.035	0.035	0.036	0.036	0.036	0.037	0.039	0.042	0.045	0.048
1200	0.034	0.034	0.034	0.034	0.034	0.034	0.034	0.034	0.035	0.035	0.036	0.038	0.041	0.045	0.048
1150	0.034	0.034	0.034	0.034	0.034	0.034	0.034	0.034	0.034	0.034	0.034	0.037	0.041	0.045	0.049
1100	0.032	0.032	0.032	0.032	0.032	0.032	0.032	0.032	0.033	0.033	0.034	0.037	0.040	0.045	0.050
1050	0.031	0.031	0.031	0.031	0.031	0.031	0.031	0.031	0.032	0.032	0.033	0.036	0.040	0.047	0.052
1000	0.030	0.030	0.030	0.030	0.030	0.030	0.030	0.030	0.030	0.031	0.032	0.035	0.041	0.049	0.055
950	0.029	0.029	0.029	0.029	0.029	0.029	0.029	0.029	0.029	0.030	0.031	0.035	0.042	0.052	0.059
900	0.028	0.028	0.028	0.028	0.028	0.028	0.028	0.028	0.028	0.028	0.029	0.035	0.045	0.057	0.064
850	0.026	0.026	0.026	0.026	0.026	0.026	0.027	0.027	0.027	0.027	0.028	0.035	0.052	0.064	0.070
800	0.025	0.025	0.025	0.025	0.025	0.025	0.025	0.025	0.026	0.026	0.027	0.040	0.062	0.071	0.075
750	0.024	0.024	0.024	0.024	0.024	0.024	0.024	0.024	0.025	0.025	0.026	0.057	0.071	0.078	0.081
700	0.023	0.023	0.023	0.023	0.023	0.023	0.023	0.023	0.023	0.024	0.026	0.071	0.079	0.085	0.086
650	0.022	0.022	0.022	0.022	0.022	0.022	0.022	0.022	0.023	0.023	0.023	0.082	0.088	0.092	0.096
600	0.021	0.021	0.021	0.021	0.021	0.021	0.021	0.021	0.021	0.021	0.087	0.091	0.096	0.101	0.104
550	0.020	0.020	0.020	0.020	0.020	0.020	0.020	0.020	0.020	0.019	0.095	0.101	0.105	0.109	0.113
500	0.019	0.019	0.019	0.019	0.019	0.019	0.019	0.018	0.018	0.103	0.105	0.111	0.114	0.119	0.122
450	0.018	0.018	0.018	0.018	0.017	0.017	0.017	0.017	0.115	0.116	0.118	0.123	0.127	0.131	0.135
400	0.016	0.016	0.016	0.016	0.016	0.016	0.016	0.016	0.131	0.132	0.134	0.138	0.143	0.147	0.150
350	0.015	0.015	0.015	0.015	0.015	0.015	0.015	0.152	0.153	0.154	0.155	0.160	0.164	0.168	0.171
300	0.014	0.014	0.014	0.014	0.014	0.014	0.182	0.183	0.183	0.184	0.185	0.190	0.194	0.198	0.201
250	0.013	0.013	0.013	0.013	0.013	0.228	0.228	0.228	0.228	0.229	0.231	0.235	0.238	0.242	0.245
200	0.012	0.012	0.012	0.012	0.300	0.300	0.300	0.300	0.301	0.301	0.303	0.306	0.310	0.313	0.316
150	0.011	0.011	0.427	0.427	0.427	0.427	0.427	0.427	0.427	0.428	0.429	0.431	0.434	0.437	0.439
100	0.680	0.680	0.680	0.680	0.680	0.680	0.680	0.680	0.680	0.680	0.680	0.681	0.682	0.683	0.683
50	1.299	1.299	1.299	1.299	1.299	1.299	1.299	1.299	1.299	1.298	1.296	1.289	1.284	1.279	1.275
32	1.753	1.753	1.753	1.753	1.753	1.753	1.753	1.752	1.751	1.749	1.745	1.733	1.723	1.713	1.705

Values directly below undescored viscosities are for water.
° Critical point.
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Table 5-10
Heat loss, Q_L, BTU/hr

ΔT	Surface	Still Air	Wind 10 mph	Wind 20 mph	Wind 30 mph
	Uninsulated	1.8	4.1	5.2	6.1
60°	1-in. insulation	0.18	0.2	0.14	0.21
	2-in. insulation	0.1	0.11	0.11	0.11
	Uninsulated	2.1	4.4	5.7	6.5
100°	1-in. insulation	0.18	0.2	0.21	0.21
	2-in. insulation	0.1	0.11	0.11	0.11
	Uninsulated	2.7	5.1	6.4	7.4
200°	1-in. insulation	0.19	0.21	0.22	0.22
	2-in. insulation	0.11	0.11	0.11	0.11

Table 5-11
Heat transfer coefficient, U, BTU/hr-ft²-°F

Fluid Giving up Heat	Fluid Receiving Heat	State of Controlling Resistance: Free Convection, U	State of Controlling Resistance: Forced Convection, U	Typical Fluid
Liquid	Liquid	25–60	150–300	Water
		5–10	20–50	Oil
	Gas	1–3	2–10	Water to Air
	Boiling Liquid	20–60	50–150	Water
		5–20	25–60	Oil
Gas	Liquid	1–3	2–10	Air to Water
	Gas	0.6–2	2–6	Gas to Steam
	Boiling Liquid	1–3	2–10	Gas to Boiling Water
Condensing Vapor	Liquid	50–200	150–800	Steam to Water
		10–30	20–60	Steam to Oil
	Gas	1–2	2–10	Steam to Air
	Boiling Liquid	300–800		Steam to Water
		50–150		Steam to Oil

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Notes:

1. Consider usual fouling for this service.
2. Maximum values of U should be used only when velocity of fluids is high and corrosion or scaling is considered negligible.
3. "Natural convection" applies to pipe coils immersed in liquids under static conditions.
4. "Forced convection" refers to coils immersed in liquids that are forced to move either by mechanical means or fluid flow.
5. The designer should be aware that a natural circulation will arise in the heating mode once the coil is turned on. This natural circulation is not to be confused with forced circulation, which is referred to as "agitated."

Source: W. H. McAdams, Heat Transmission, *McGraw-Hill Book Co. Inc, 1942.*

Table 5-12
Heat transfer coefficient, U, BTU/hr-ft^2-°F

Liquid	Heating Medium		
	150# Steam	10# Steam	180°F Water
Clean fats, oils, etc., 130°F	25	20	17
Clean fats, oils with light agitation	40	40	40
Glycerine, pure, 104°F	40	35	30
Toluene, 80°F	55	47.5	42.5
Methanol, 100°F	70	62	52
Water, soft, 80°F	85	72	66
Water, soft, 160°F	105	82	
Water, soft, boiling	175	108	
Water, hard, 150°F	120	100	

Table 5-13
Heat transfer coefficient, U, BTU/hr-ft^2-°F

Heating Applications		Clean Surface Coefficients		Design Coefficients	
Hot Side	Cold Side	Natural Convection	Forced Convection	Natural Convection	Forced Convection
Steam	Watery solution	250–500	300–550	125–225	150–275
Steam	Light oils	50–70	110–140	40–45	60–110
Steam	Medium lube oils	40–60	100–130	25–40	50–100
Steam	Bunker "C" or #6 fuel oil	20–40	70–90	10–30	60–80
Steam	Tar or asphalt	15–35	50–70	15–25	40–60
Steam	Molten sulfur	35–45	60–80	4–15	50–70
Steam	Molten paraffin	35–45	45–55	25–35	40–50
Steam	Air or gases	2–4	5–10	1–3	4–8
Steam	Molasses or corn syrup	20–40	70–90	15–30	60–80
High temp., hot water	Watery solution	80–100	100–225	70–100	110–160
High temp., heat transf. oil	Tar or asphalt	12–30	45–65	10–20	30–50
Therminol	Tar or asphalt	15–30	50–60	12–20	30–50
Cooling Applications					
Cold Side	Hot Side				
Water	Watery solution	70–100	90–160	50–80	80–140
Water	Quench oil	10–15	25–45	7–10	15–25
Water	Medium lube oils	8–12	20–30	5–8	10–20
Water	Molasses or corn syrup	7–10	18–26	4–7	8–15
Water	Air or gases	2–4	5–10	1–3	4–8
Freon or ammonia	Watery solution	35–45	60–90	20–35	40–60

Notes:

1. Consider usual fouling for this service.
2. Maximum values of U should be used only when velocity of fluids is high and corrosion or scaling is considered negligible.
3. "Natural convection" applies to pipe coils immersed in liquids under static conditions.
4. "Forced convection" refers to coils immersed in liquids that are forced to move either by mechanical means or fluid flow.
5. The designer should be aware that a natural circulation will arise in the heating mode once the coil is turned on. This natural circulation is not to be confused with forced circulation, which is referred to as "agitated."

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Therefore $D_c = 72$ in.

- *Pipe properties.*

Assume 3-in. Sch 80 pipe.

$d_i = 2.9$ in.

$D_i = 0.2417$ ft

$S = 0.916\ ft^2/ft$

- *Length of pipe required, L_r.*

$$L_r = \frac{A_r}{S} = \frac{408}{0.916} = 445 \text{ ft}$$

- *Check minimum radius.*

$$\frac{D_c}{2} > 10\ d_i = \frac{72}{2} > 10(3) = 36 > 30$$

- *Determine pitch, p.*

$$P_{max} = 5NPS = 5(3) = 15$$
$$P_{min} = 2NPS = 2(3) = 6$$

Use $p = 2.5(3) = 7.5$ in.

- *Find number of turns of spiral, N.*

$$N = \frac{L_r}{\sqrt{(\pi D_c)^2 + p^2}} = \frac{445}{\sqrt{[\pi(6)]^2 + 0.625^2}}$$
$$= 23.59$$

Use (24) turns × 7.5 in. = 180 in.—OK.

- *Find actual length of coil, L_a.*

$$L_a = N\sqrt{(\pi D_c)^2 + p^2}$$
$$L_a = 24\sqrt{[\pi(6)]^2 + 0.625^2} = 486 \text{ ft}$$

- *Rate of flow, W.*

$$W = \frac{Q}{E} = \frac{2,612,413}{912} = 2864 \text{ lb/hr}$$

- *Reynolds number, R_e.*

$$R_e = \frac{6.31W}{d_i\mu} = \frac{6.31(2864)}{2.9(0.015)} = 415,445$$

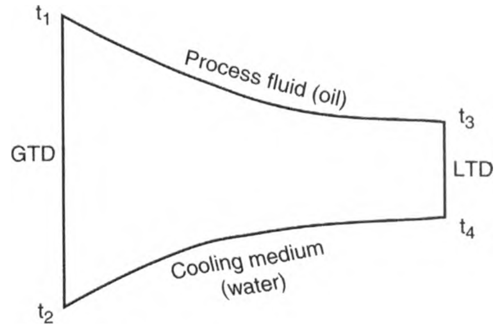
- *Velocity of steam in coil, V.*

$$V = \frac{0.00085WV_s}{d_i^2} = \frac{0.00085(2864)6.7}{2.9^2}$$
$$= 1.94 \text{ ft/sec} = 6982 \text{ ft/hr}$$

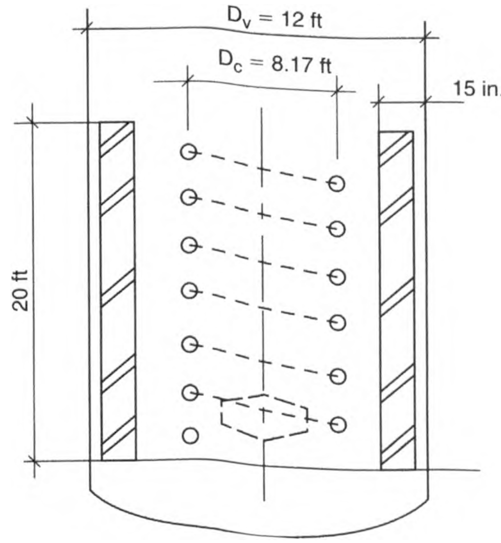
- *Pressure drop, ΔP.*

$$\Delta P = \frac{2fL_aV^2}{3gD_i} = \frac{2(0.021)486(6982^2)}{3(4.17 \times 10^8)0.2417}$$
$$= 1.88 \text{ psi}$$

Sample Problem 2



Cooling application:	Parallel flow
Process fluid:	Hot oil (vessel contents)
Cooling medium:	Water (coil contents)
Vessel indoors:	$Q_L = 0$
Discharge rate:	3000 GPH
Agitation:	Yes
Baffles:	Yes



Properties

Process fluid

$C_p = 0.42$

$\mu = 13\ @\ 110°F$

Coil medium

$\mu = 0.75 @ 90°F$

$V_s = 0.0161$

$w = 62.4$

Temperatures

$t_1 = 140°F$

$t_2 = 60°F$

$t_3 = 110°F$

$t_4 = 90°F$

- *Log mean temperature difference, ΔT.*

$$GTD = t_1 - t_2 = 140 - 60 = 80$$
$$LTD = t_3 - t_4 = 110 - 90 = 20$$
$$\Delta T = \frac{GTD - LTD}{2.3 \log\left(\frac{GTD}{LTD}\right)} = \frac{80 - 20}{2.3 \log\left(\frac{80}{20}\right)} = 43°F$$

- *Mass flow rate, M.*

$$M = 3000\frac{gal}{hr}\left(8.33\frac{lb}{gal}\right) = 24{,}990 \text{ lb/hr}$$

- *Heat required, Q.*

$$Q = MC_p\Delta T - Q_L = 24{,}990(0.42)43 - 0$$
$$= 451{,}319 \text{ BTU/hr}$$

- *Heat transfer coefficient, U.*

U = from Table 5-14: 10 – 20
Use U = 15.

- *Area of coil required, A_r.*

$$A_r = \frac{Q}{U\Delta T} = \frac{451{,}319}{15(43)} = 700 \text{ ft}^2$$

- *Determine baffle sizes.*

Baffle width, B = 0.083D = 12 in.
Off wall, B_c = 0.021D = 3 in.

- *Determine the physical dimensions of the coil.*

$$NPS = \frac{D_v}{30} = \frac{144}{30} = 4.8$$

Use 4-in. pipe.

- *Pipe properties.*

Assume 4-in. Sch 80 pipe:

$d_i = 3.826$ in.

$D_i = 0.3188$ ft

$S = 1.178 \text{ ft}^2/\text{ft}$

- *Determine coil diameter, D_c.*

$$D_c = 144 - 2(15) - 2(8) = 98 \text{ in. } (8.17 \text{ ft})$$

- *Length of pipe required, L_r.*

$$L_r = \frac{A_r}{S} = \frac{700}{1.178} = 595 \text{ ft}$$

- *Check minimum radius.*

$$\frac{D_c}{2} > 10d_i = \frac{98}{2} > 10(4.5) = 49 > 45$$

- *Determine pitch, p.*

$$P_{max} = 5NPS = 5(4) = 20$$
$$P_{min} = 2NPS = 2(4) = 8$$

Use p = 2.5(4) = 10 in. = 0.833 ft

- *Find number of turns of spiral, N.*

$$N = \frac{L_r}{\sqrt{(\pi D_c)^2 + p^2}} = \frac{595}{\sqrt{[\pi(8.17)]^2 + 0.833^2}}$$
$$= 21.6$$

Use (22) turns × 10 in. = 220 in. < 240 in.—OK

- *Find actual length of coil, L_a.*

$$L_a = N\sqrt{(\pi D_c)^2 + p^2}$$
$$L_a = 22\sqrt{[\pi(8.17)]^2 + 0.833^2} = 565 \text{ ft}$$

- *Rate of flow, W.*

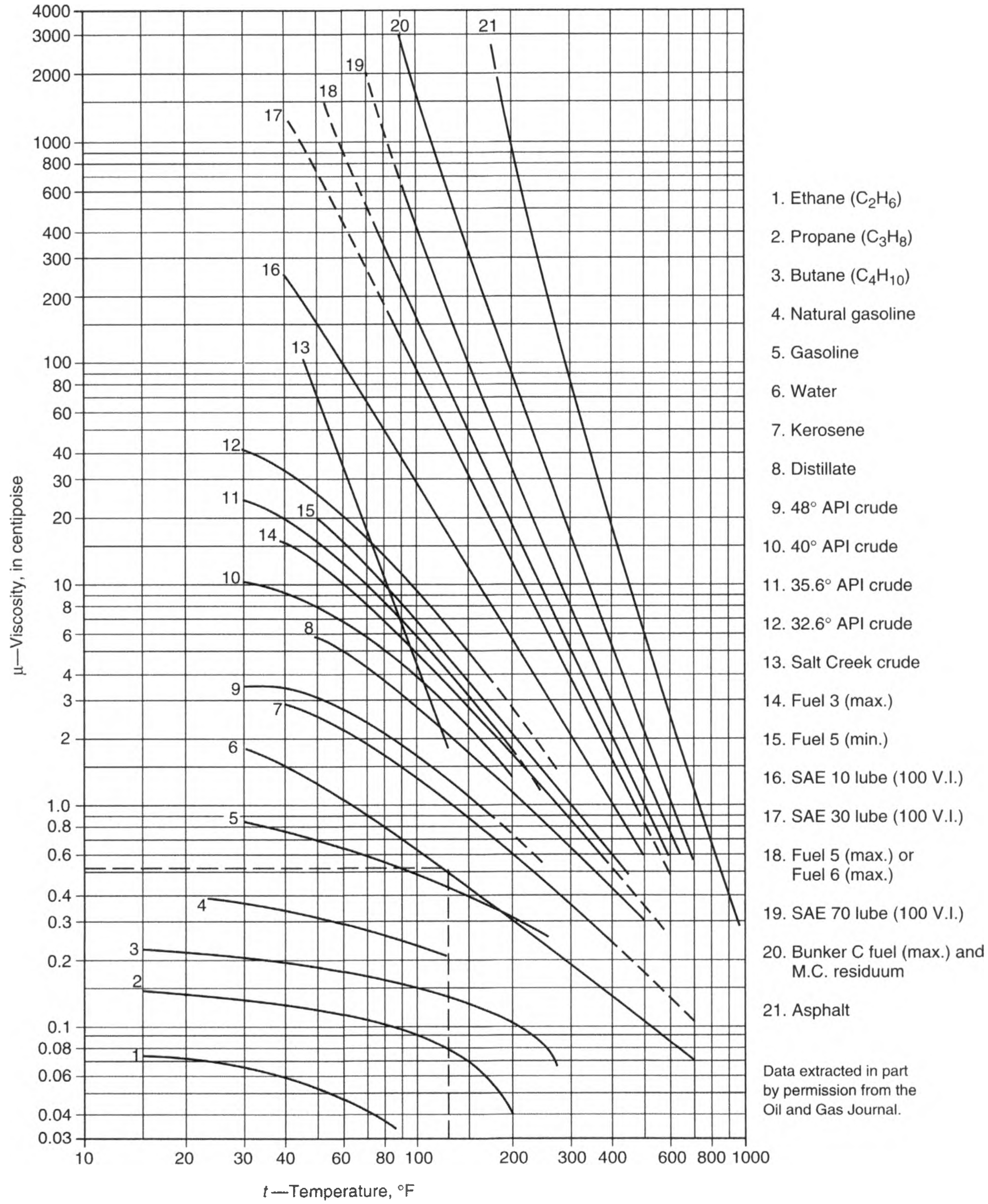
$W = 24{,}990$ lb/hr

- *Velocity, V.*

$$V = \frac{0.0509WV_s}{d_i^2} = \frac{0.0509(24{,}990)0.0161}{3.826^2}$$
$$= 1.4 \text{ ft/sec}$$

- *Reynolds number, R_e.*

$$R_e = \frac{123.9d_iV}{V_S\mu} = \frac{123.9(3.826)1.4}{0.0161(0.75)} = 54{,}961$$



Example: The viscosity of water at 125°F is 0.52 centipoise (Curve No. 6).

Figure 5-10. Viscosity of water and liquid petroleum products. Reprinted by permission by Crane Co., Technical Paper No. 410

Table 5-15
Effect of metal conductivity on "U" values

Application	Material	Film Coefficients		Thermal Conductivity BTU-in/hr/ft²/°F	Metal Thickness (in).	U (BTU/hr-ft²-°F)
		h_o	h_i			
Heating water with saturated steam	Copper	300	1000	2680	0.0747	229
	Aluminum	300	1000	1570	0.0747	228
	Carbon steel	300	1000	460	0.0747	223
	Stainless steel	300	1000	105	0.0747	198
Heating air with saturated steam	Copper	5	1000	2680	0.0747	4.98
	Aluminum	5	1000	1570	0.0747	4.97
	Carbon steel	5	1000	460	0.0747	4.97
	Stainless steel	5	1000	105	0.0747	4.96

Therefore flow is turbulent!

- *Straight-line pressure drop, ΔP_L.*

$$\Delta P_L = \frac{(3.36 \times 10^{-6}) f L_a W^2}{d_i^5 w}$$

$$\Delta P_L = \frac{(3.36 \times 10^{-6}) 0.0218(565)24,990^2}{3.826^5(62.11)}$$

$$= 0.5 \text{ psi}$$

- *Pressure drop, ΔP.*

$$\Delta P = \Delta P_L \sqrt{R_e \left(\frac{d_i}{d_C}\right)^2}$$

$$\Delta P = 0.5 \sqrt{54,961 \left(\frac{3.826}{98}\right)^2} = 4.57 \text{ psi}$$

Procedure 5-7: Agitators/Mixers for Vessels and Tanks

Mixing is defined as the intermingling of particles that produce a uniform product. Hydraulically, mixers behave like pumps. Mixing applications can be either a batch or a continuous process. Although the terms *agitation* and *mixing* are often used interchangeably, there is a technical difference between the two.

Agitation creates a flow or turbulence as follows:

- Mild agitation performs a blending action.
- Medium agitation involves a turbulence that may permit some gas absorption.
- Violent agitation creates emulsification.

Mechanical mixers are used as follows:

- To mix two or more nonhomogeneous materials.
- To maintain a mixture of materials that would separate if not agitated.
- To increase the rate of heat transfer between materials.

The mechanical mixer usually consists of a shaft-mounted impeller connected to a drive unit. Mechanical mixers can be as small as ¼ hp or as large as 200 hp for some gear-driven units. Power consumption over time determines the efficiency and economy of a mixing process. Top-mounted mixers can be located on center (VOC), off center (VOFC), or angled off center (AOC). Mixers on center require baffles.

If the ratio of liquid height to vessel diameter is greater than 1.25, then multiple impellers are recommended. Ratios of 2:1 and 3:1 are common in certain processes. A common rule of thumb is to use one impeller for each diameter of liquid height.

Mixer applications are designed to achieve one of the following:

- *Blending:* combines miscible materials to form a homogeneous mixture.
- *Dissolving:* the dissipation of a solid into a liquid.
- *Dispersion:* the mixing of two or more nonmiscible materials.
- *Solid suspension:* suspends insoluble solids within a liquid.
- *Heat exchange:* promotes heat transfer through forced convection.
- *Extraction:* separation of a component through solvent extraction.

Mounting

Top-entering units can generally be used on all applications. Side-entering units are usually used for low speed, mild blending, and tank cleaning operations. The most efficient mounting is angled off center (AOC).

Tank Baffles

Antiswirl baffles are required in most larger industrial fluid-mixing operations. Baffles are used for center-shaft, top-mounted mixers to prevent vortexing. Baffles also promote top-to-bottom turnover and represent good mixing practice. The most usual arrangement is to have four baffles spaced at 90°. For viscosities up to 500 centipoise, baffles can be mounted directly to the wall. For use in higher-viscosity material or in any mixing application where solids can build up or where other harmful effects develop when the baffle attaches to the wall, the baffles should be spaced off of the wall. Normal spacing is 25% of the actual baffle width. Above 10,000 cP, baffles should be mounted at least 1½ in. off the wall. Above 20,000 cP, no baffles are typically required. Horizontal tanks do not usually require baffles. Baffles should be selected for the minimum viscosity that will occur during a mixing cycle.

As liquid viscosities go up, the need for baffles—and thus the baffle width—decreases. The industrial use of vessels without baffles is limited because unbaffled systems give poor mixing.

Baffle widths and the wall clearance depend on the viscosity of the liquid being mixed:

Viscosity, cP	Baffle Width, B	Off Wall, Bc
Waterlike	0.083 D	0.021 D
5000	0.056 D	0.014D
10,000	0.042 D	0.011 D
20,000	0.021 D	0.005 D

Impellers

Impellers come in the following types:

- Paddle.
- Propeller.
- Anchor.
- Turbine.
- Ribbon.

Paddle-type impellers are the simplest and lowest cost impellers, but they have small pumping capacity. They have very low axial flow, hence the pitched flat blade version is normally used for low-viscosity materials. The ratio of blade diameter to vessel diameter is usually ⅓ to ⅔. A radial flow impeller is used for high shear.

Propeller types pump liquid. Every revolution of a square pitch prop discharges a column of liquid approximately equal to the diameter of the propeller. The flow is axial. Such pumps are used primarily for high-speed applications and side-entry mixers. Dual propellers are used on vessels with H/D ratios greater than 1. The axial flow decreases mix time. They are heavier and cost more than pitched-blade turbines. Propeller-to-vessel-diameter ratio is usually ⅓. A propeller-type impeller is used for high flow.

Anchor-type impellers rotate slowly and have a large surface area. This makes them ideal for batch applications in higher-viscosity materials.

Turbines are always mounted vertically. They are used at low speed where the application requires greater shear than pumping and higher horsepower per unit volume. There are two basic forms of turbines, the flat-blade radial-discharging type and the pitched-blade axial-thrust type. All others are modifications of these basic types. The ratio of blade diameter to vessel diameter is usually ⅓.

Flat-blade turbines pump liquid outward by centrifugal force. Liquid that is displaced by the blade is replaced by flow from the top and bottom. Suction comes from the center, and delivery is on the circumference of the blade. The primary flow is radial. This is the most widely used type of mechanical agitator. The number of blades vary from 4 to 12. This turbine is used primarily for liquid–liquid dispersion. Turbines with curved blades are used for higher-viscosity materials.

The pitched-blade turbine produces a combination of axial and radial flow. The purpose of pitching the blade is to increase radial flow. Blades can be sloped anywhere from 0° to 90°, but 45° is the commercial standard.

Notes

1. All mixers/agitators rotate clockwise.
2. In general, agitators are sized on the basis of the required torque per unit volume. Other factors that affect size and torque are:
   - Viscosity $>$ 100 cP (viscosity can affect blend times).
   - Critical speeds.
   - Tip speed.
   - Impeller diameter.
   - Required degree of agitation.
3. Each shaft is designed for mechanical loads and critical shaft speed. Motor size and shaft design are related. A larger shaft to take the torque will require more horsepower to eliminate wobble.
4. To prevent solid buildup on the bottom, a radial-blade impeller may be used. If elected, then place the blade one blade width off the bottom.
5. Power consumption:
   - Operating speed is back-calculated to ensure delivery of the proper power for a given impeller diameter.
   - The speed and horsepower define the torque required for the system. The torque in turn sets the shaft size and gear box size.
   - Impeller power consumption determines the horsepower and impeller diameter required for a given mixing process.
6. Mixing parameters:
   - Shaft angle.
   - Time.
   - Impeller type and diameter.
   - RPM (pumping capacity).
   - Power.
   - Viscosity, specific gravity.
7. A steady rest bearing may be utilized at the bottom of the tank if the mixing application allows.
8. Other applicable data:
   - Types of seals or packing.
   - Metallurgy.
   - Drain location.
   - Manway size.
   - Indoor/outdoor.
   - Mixer/agitator run times.
   - Head room required above tank.

Vessels with Agitators or Mixers

Notation

H_p = motor horsepower
N = impeller RPM
D = vessel diameter, in.
d = impeller diameter, in.
B = baffle width, in.
B_c = baffle off-wall distance, in.

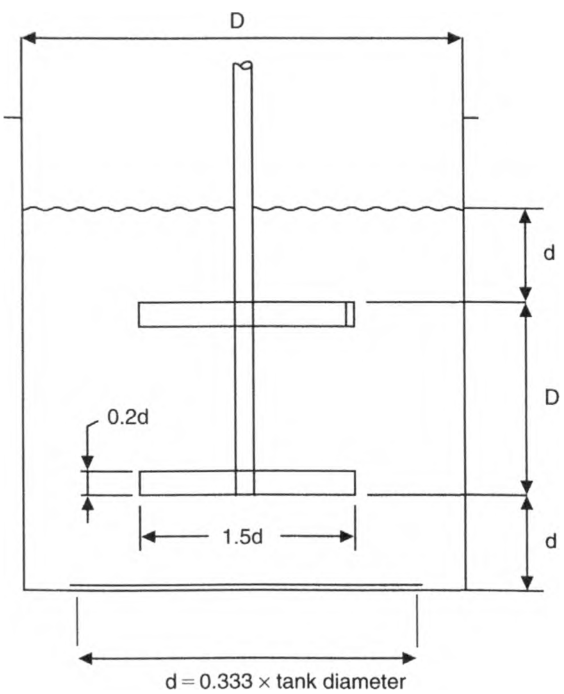
- *Force on baffle, F.*

$$F = \frac{(56,800H_P)}{2N[D - B - (2B_c)]}$$

- *Force per unit area, F_u.*

$$F_u = \frac{F}{Bh}$$

- *Typical ratios.*

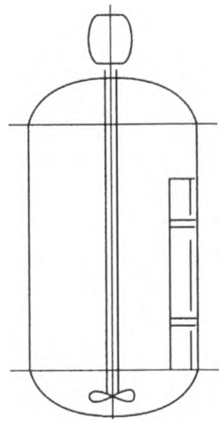


Use wear plate with pitched or propeller blades and suspended solids.

Types of Mixers

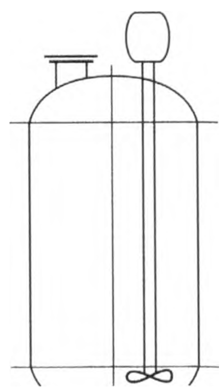
Type 1: Vertical On Center (VOC)

- Requires baffles.

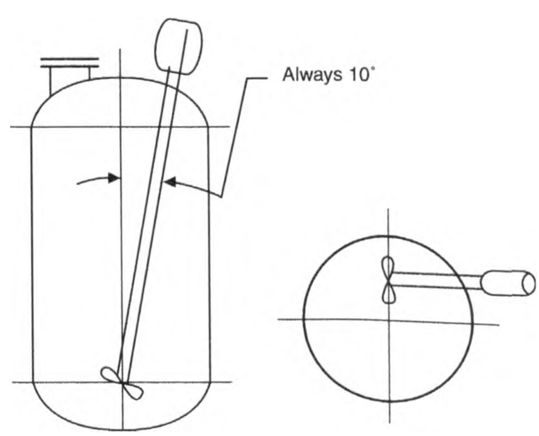


Type 2: Vertical Off Center (VOFC)

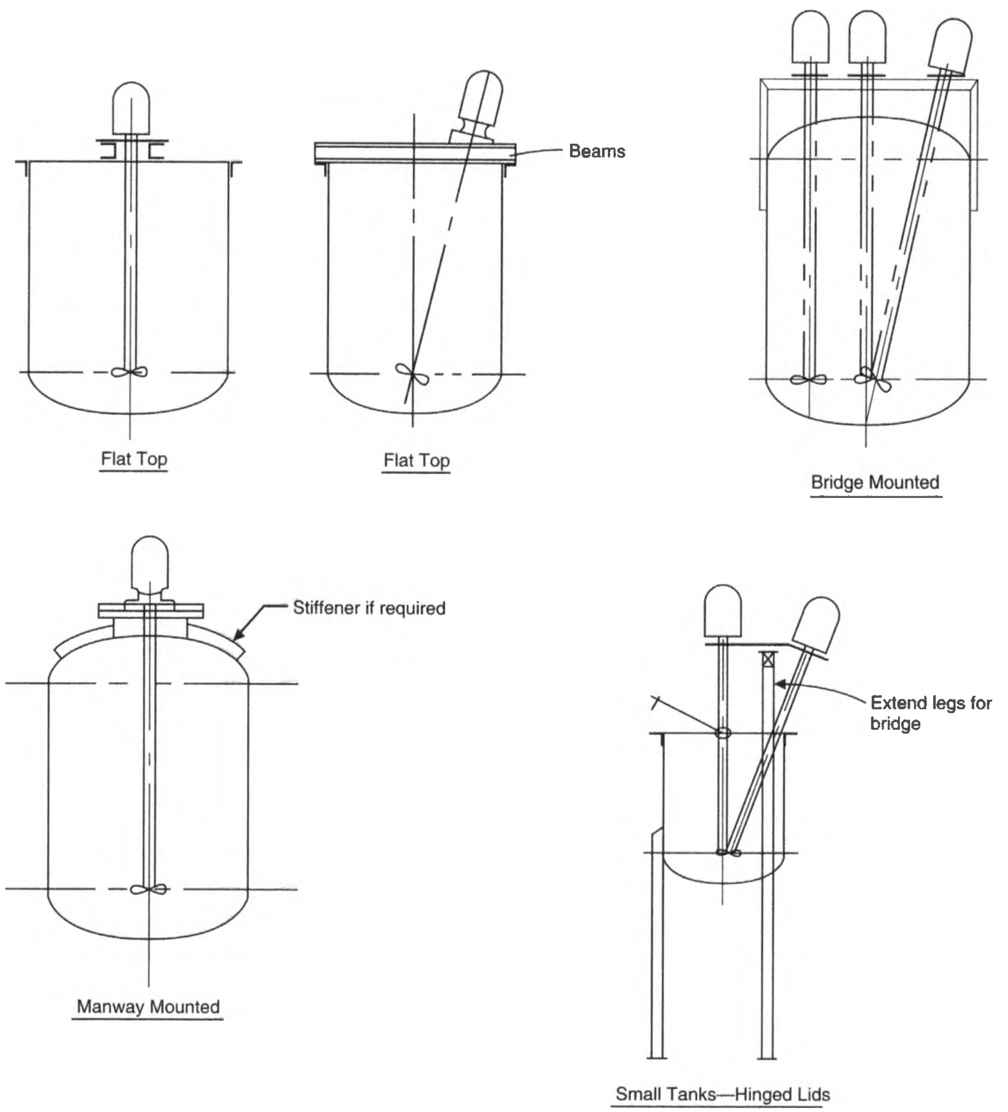
- Least effective.
- Poor mixing.



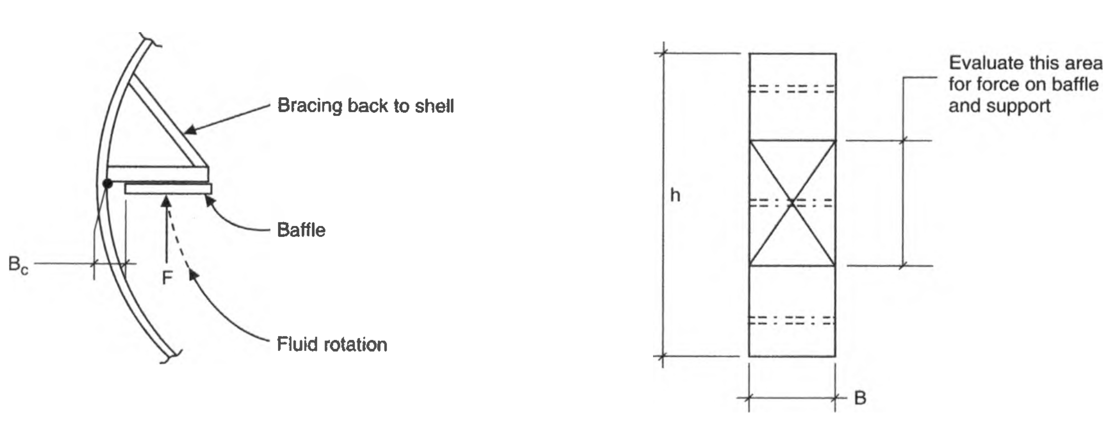
Type 3: Angular Off Center (AOC)



Types of Mounting

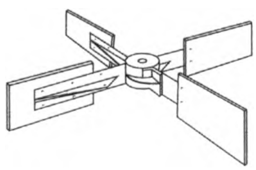


Baffle Supports

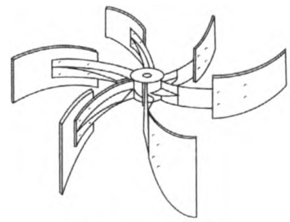


Types of Impellers

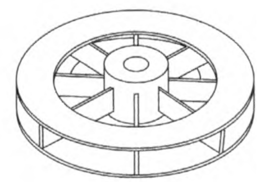
Turbine Impellers



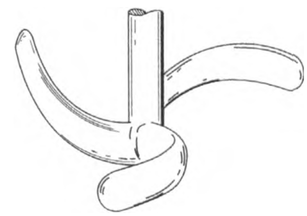
a. Flat blade



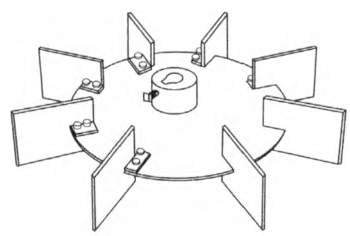
b. Curved blade



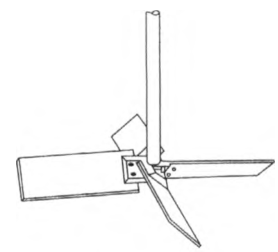
c. Shrouded



d. Retreating blade

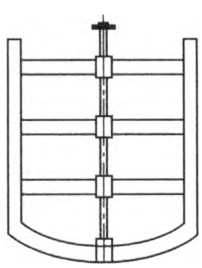


e. Disk flat blade

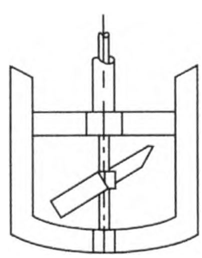


f. Pitched blade

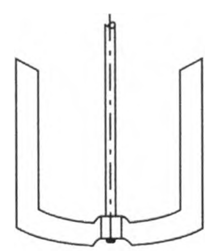
Anchor Impellers



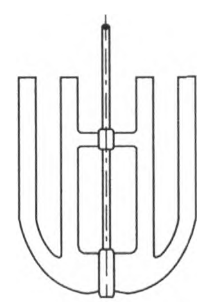
a. Horseshoe with cross members



b. Double-motion horseshoe paddle

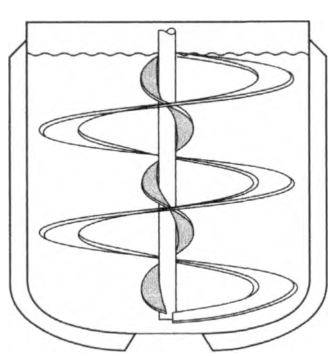


c. Horseshoe

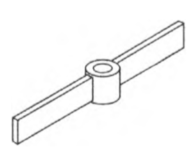


d. Gate type

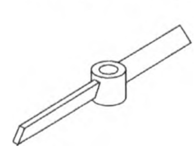
Miscellaneous Impellers



Helical ribbon

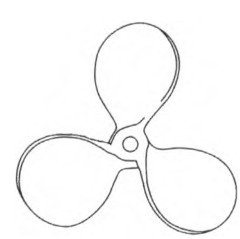


Straight flat blade



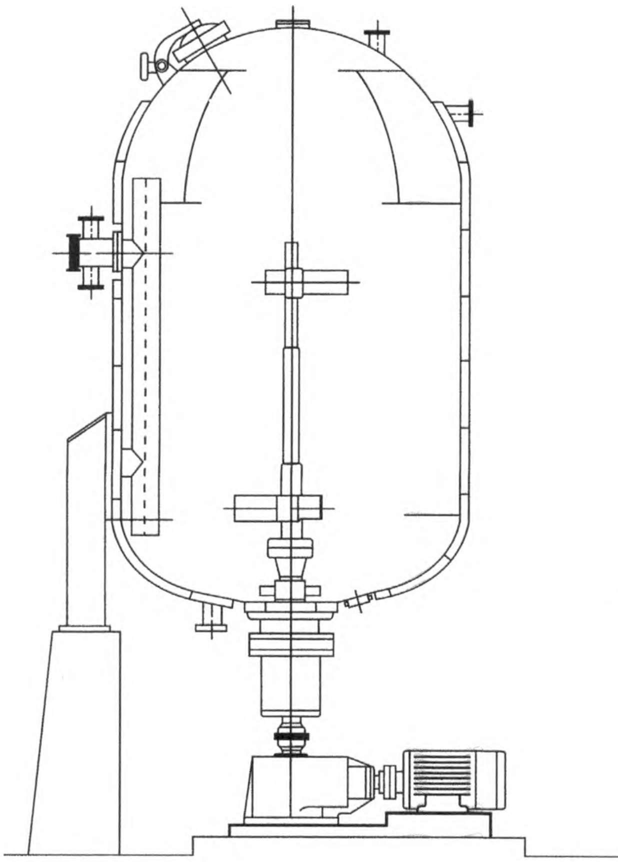
Pitched flat blade

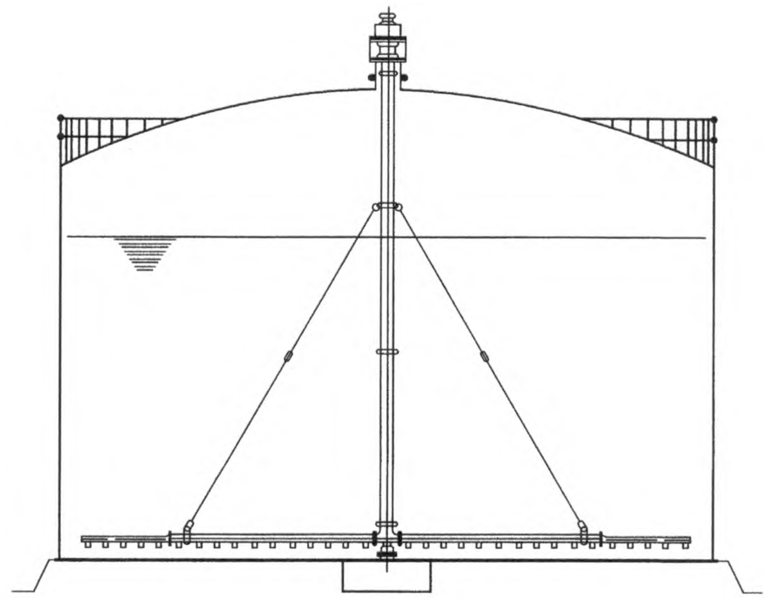
Paddle

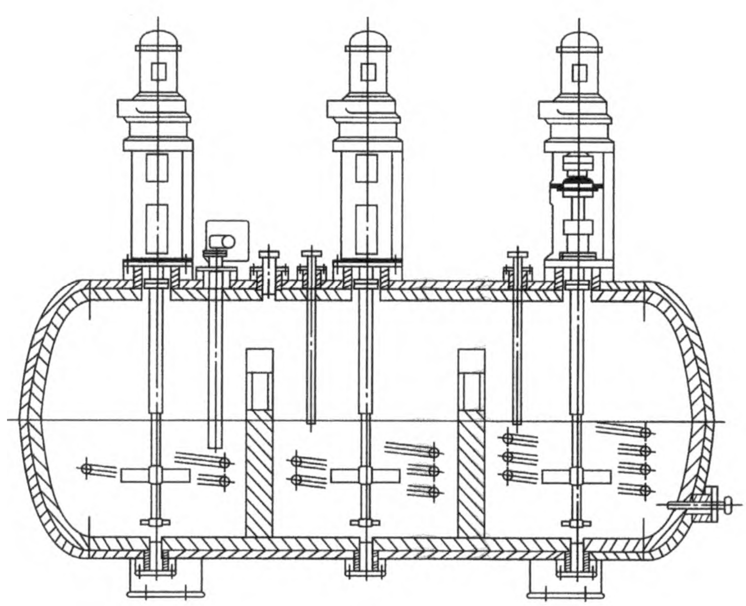


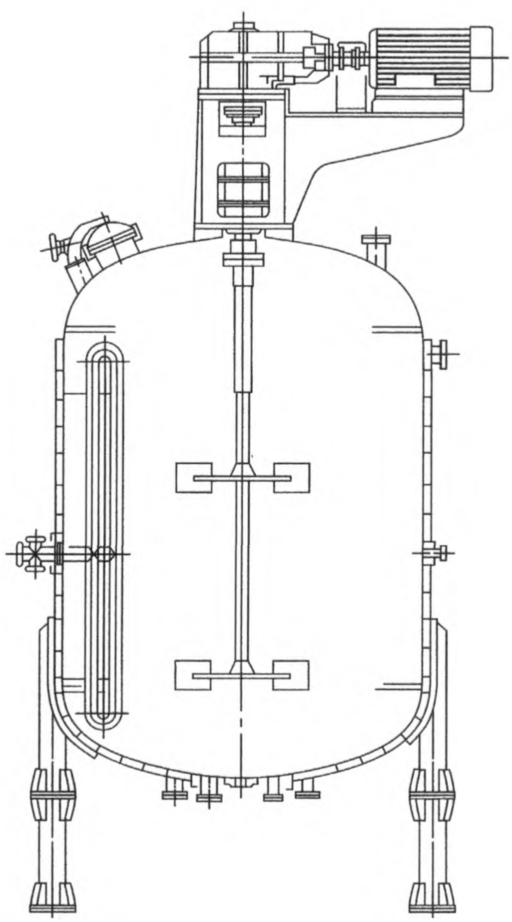
Propeller

Typical Applications



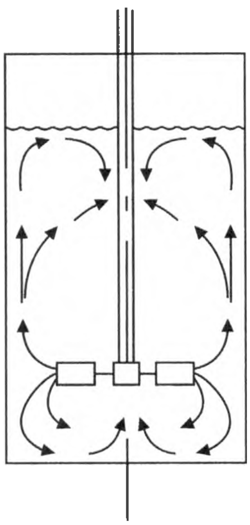




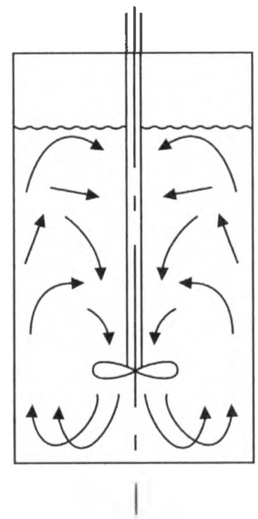


Impeller Actions

Shear Action
- Break up liquid blobs.
- Use radial-flow impeller such as turbine or paddle types without pitched blades.



Pumping Action
- Lift solids from bottom.
- Good for blending solids and liquids.
- Use propeller or turbine or paddle type with pitched blades.



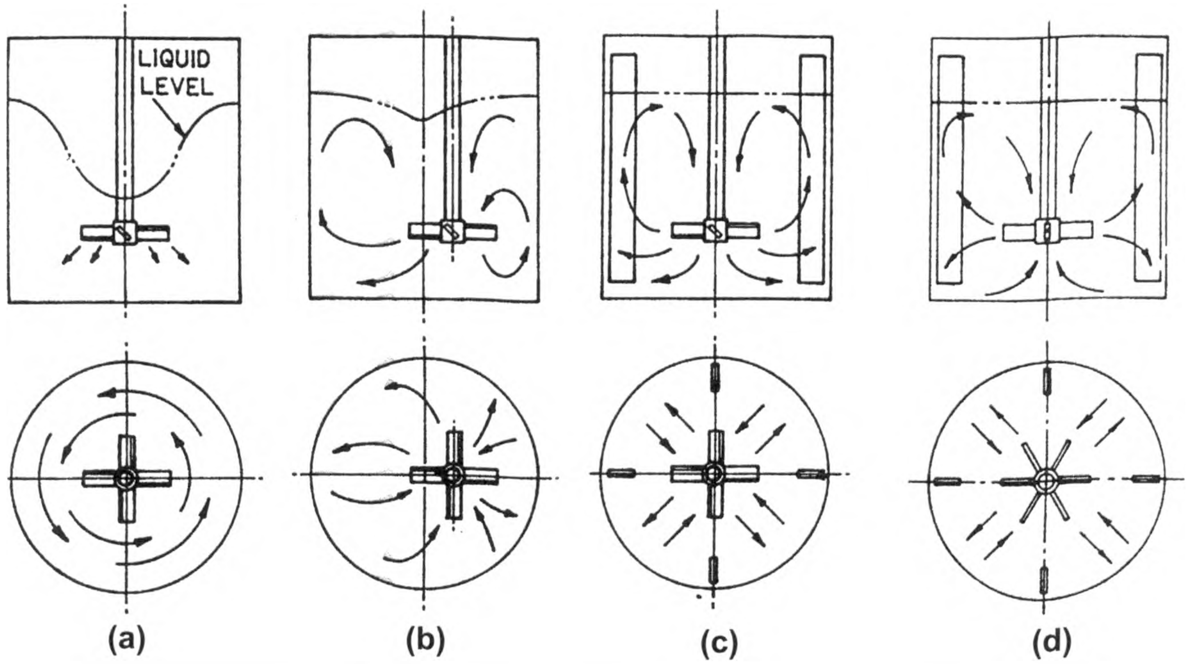


Figure 5-11. Agitator flow patterns. (a) Axial or radial impellers without baffles produce vortexes. (b) Off-center location reduces the vortex. (c) Axial impeller with baffles. (d) Radial impeller with baffles.

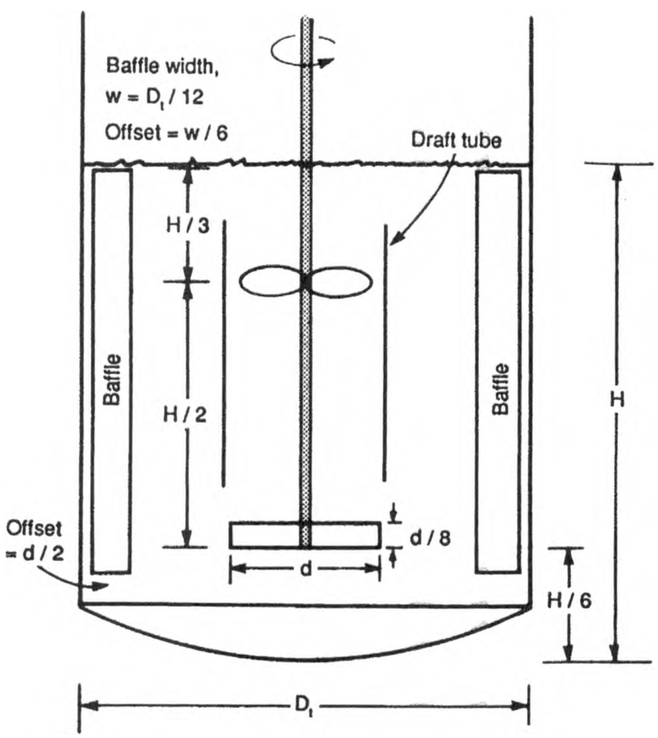


Figure 5-12. Typical proportion of stirred tank design with radial and axial impellers and baffles. The upper axial impeller is housed in a draft tube. For radial impellers, $.3 < d/D_t < .6$

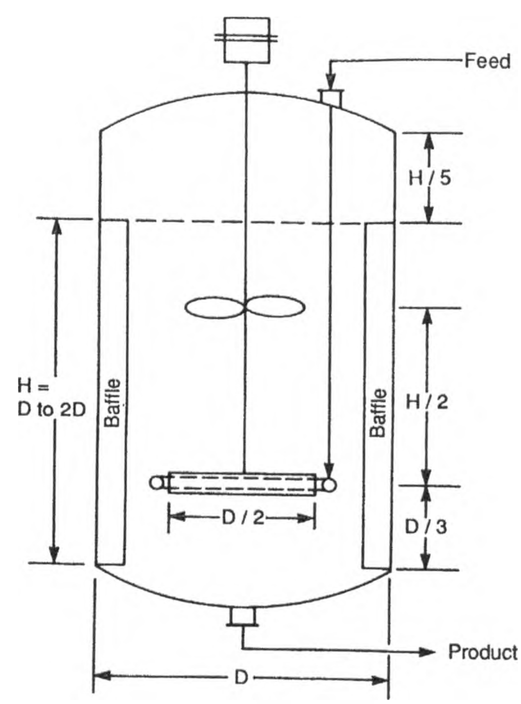


Figure 5-13. Typical proportions of a stirred tank reactor with radial and axial impellers, four baffles, and a sparger feed inlet.

Procedure 5-8: Design of Internal Pipe Distributors

Uniform liquid distribution is essential for efficient operation of chemical processing equipment. To obtain optimum distribution, proper consideration must be given to flow behavior in the distributor itself, flow conditions upstream and downstream of the equipment.

This procedure does not consider flows upstream or downstream of the distributor. However, disturbances upstream of the distributor is dependent on the piping configuration and may increase or decrease the flow to the distribution device.

There are various kinds and types of distributors to handle the following flow regimes;

1. Gas or vapor
2. Liquid
3. Two phase flow

This procedure is only concerned with liquid or two phase flow, and does not consider gas/vapor type distribution devices. There are several types of liquid distributors for packed columns that are also not part of this discussion. These are;

1. Trough distributors
2. Orifice plate distributors

Internal pipe distributors are frequently used for feed or reflux inlets for trayed columns or towers (fractionation or distillation columns). They are used to distribute the inlet stream to a particular point on a tray or uniformly across a tray. The distributor consists of one or more pipes, with or without branches, containing a series of holes, slots or spray nozzles. This procedure addresses the design of these distribution devices.

Internal pipe distributors are also known as "perforated spargers" or "perforated pipe distributors" or headers. They can be designed for either liquid only or two phase flow. A pipe distributor may be considered as liquid service if the volume of vapor is no more than 5%.

Rectangular slots or holes can be used but slots are preferred.

It is not possible to discharge liquid from a distributor pipe with uniform velocities at all discharge points unless the average velocity of the liquid from the holes or slots is double the velocity of the liquid at the inlet. Expressed as an equation, it would look as follows:

Average $V_h \geq 2$ X Average V_P

To get uniform volumetric flow where the above criteria cannot be met, it is necessary to calculate the velocity from each orifice, then calculate the area required to get the necessary volumetric flow. To do this it is necessary to calculate the pressure profile in the distributor at all the various points. This would be a very complicated procedure and is seldom done. This method is not discussed here. Only a simplified procedure is presented and is only valid for inlet velocities between 6 and 14 feet per second.

In a simple perforated pipe or sparger, the flow distribution is uniform. This will be the case if the following has been properly considered;

1. Pressure recovery due to kinetic energy or momentum changes
2. Frictional pressure drop along the length of the pipe
3. Pressure drop across the outlet holes

The procedure presented here is only valid if the following criteria are met;

- The distributor pipe is horizontal
- The slots or holes are located either horizontally along the pipe or pointed downward at a maximum of 30° from horizontal.
- The slots (holes) are equally spaced along the pipe.

The following physical criteria should be met for the piping distributor:

- Slots shall meet the dimensional proportions shown in Fig 5.14.
- The minimum hole size is 0.5 inches (12 mm)
- The length of cut metal should not exceed the length of uncut metal in any given row.
- The ratio of total hole area to pipe cross sectional area should be between 1 and 3.
- Tolerance for hole or slot location is ± 0.25 inches (± 6 mm)
- Slot spacing should not exceed 24 inches (610 mm).

Variables for Distributor Design;

1. End condition
   a. Open end
   b. Closed end
   c. Elbows
   d. Tees
   e. Slotted ends

2. Discharge Type
   a. Slots
   b. Holes
   c. Direction of slots
   d. Spacing of holes
   e. Spray nozzles
3. Discharge Location
   a. Center
   b. Sides
   c. Ends
4. Function
   a. Feed
   b. Reflux
5. Passes
   a. 1 pass
   b. 2 pass
   c. 4 pass
   d. Packed bed
6. Distributor Type
   a. T-Type
   b. Ladder Type
   c. H-Type
   d. Single
   e. Multiple
7. Baffle
   a. With
   b. Without
8. Contents
   a. Liquid flow
   b. Two phase flow
9. Support Type
   a. U-bolts with bracket
   b. Clip
   c. Cross diameter angle
   d. Dummy pipe extension

Notes

1. Total slot/hole area shall be 1-3 times as large as the cross sectional area of the feed pipe.
2. The discharge velocity for any given slot or hole should not exceed 9 FPS. Higher velocities could result in excessive splashing or tray turbulence. If the slot velocity exceeds this value, the main header should be enlarged one size and the slot velocity recalculated. Normally, slot velocity will be satisfactory if the inlet velocity of the header is below 14 FPS
3. Normally, the process engineer will have already sized the inlet line before mechanical begins the detailed design of the internals. Therefore this procedure does not include sizing of the main lines.
4. The long axis of the slots should coincide with the long axis of the pipe.
5. The number and size of branches is typically determined by the configuration of the internals. The discharge locations will dictate single or multiple feeds as well as the basic layout.
6. Spray headers are typically used for packed beds in certain process columns. The design of spray headers is not covered in detail in this procedure. However the basic steps are as follows;
   a. Determine total flow
   b. Select spray nozzles that will give the desired flow, coverage and spray patterns.
   c. Layout nozzles to accommodate coverage.
   d. Develop layout and configuration of header configuration.
   e. Calculate flow rates to all spray nozzles
7. The pipe distributor must be capable of supporting itself, or provide for the support. The support design should be based on deflection rather than stress. Deflection should be limited to 0.5 inches. The formulas for deflection are as follows:
   a. Supported on both ends:

$$\delta = (5\ w\ L^4)/(384\ E\ I)$$

   b. Cantilever;

$$\delta = (w\ L^3)/(8\ E\ I)$$

Notation

A_b = Area required for one branch, in^2
A = Total area, all holes or slots, in^2
a = Area of each hole or slot, in^2
A_P = Cross sectional area of pipe, in^2
A_r = Area required, all holes, in^2
C = Orifice discharge coefficient, .6 to .63
d = Diameter of holes or length of slots, in
D_P = ID of pipe, in
f = Fanning friction factor
f_m = Density of 2-phase flow, liquid/vapor, Lbs / Ft^3
G = Specific gravity
g = Acceleration due to gravity = 32.17 Ft/Sec^2
H = Head, Ft

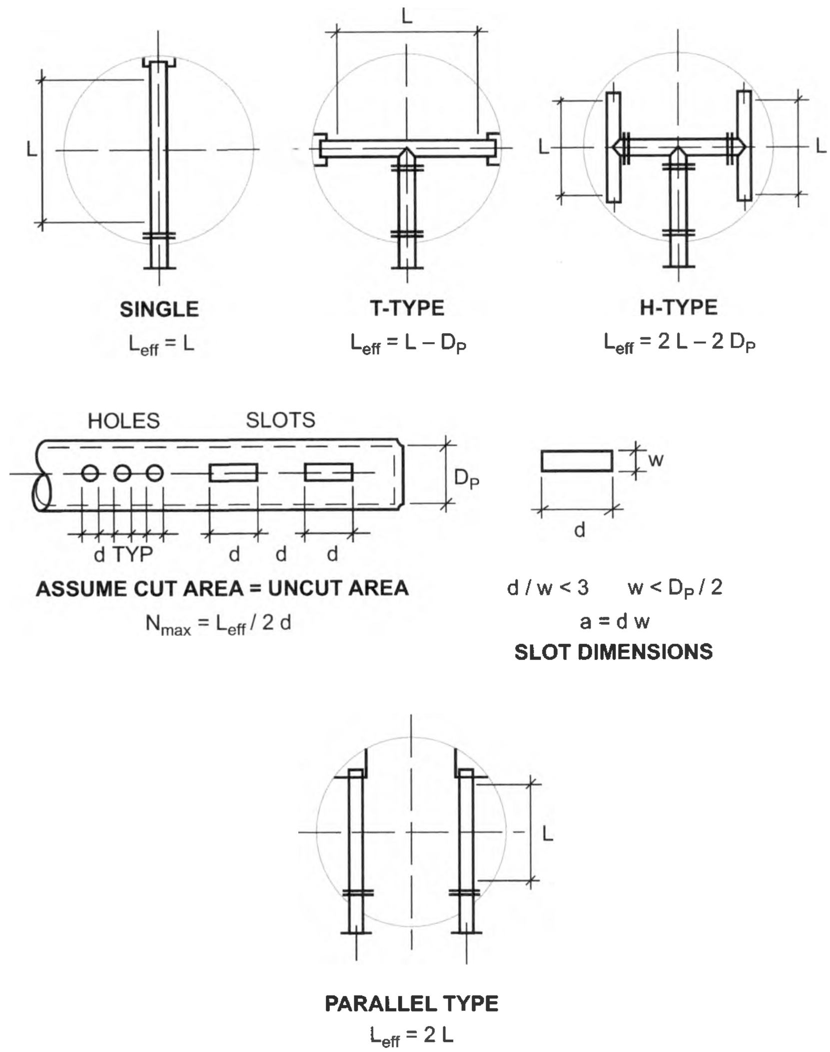


Figure 5-14. Dimensions for piping distributors.

L = Length of pipe or branch where slots or holes can be placed, in
L_{eff} = Total effective length for holes or slots, in
n = Number of branches
N = Number of holes or slots
N_r = Number of holes/slots required
N_{max} = Maximum number of holes/slots that can be placed in the effective length
Q = Flow rate, GPM
R_e = Reynolds number
V = Average inlet velocity, Ft / Sec
V_m = Volumetric 2-phase flow, Ft^3 / Sec
ΔP = Pressure drop, PSI
ρ = Density of liquid, PCF
ρ_m = Density of 2-phase flow, PCF
μ = Viscosity, centepoise

Calculations

Case 1: Liquid Only

- Determine if ΔP is known, assumed or calculated. If assumed, use .25 PSI. It can be calculated as follows;

  $\Delta P = [(4fL_{eff})/(3D_P) - 2][(V^2G)/(4.618g)]$

- Total area of holes/slots required, A_r

  $A_r = [Q/(38\ C)][G/\Delta P]^{1/2}$

  Note; A_r should be 1-3 times A_P

  $A_P =$

- Assume hole or slot size;

  $d =$

  $a =$

- Determine effective length of pipe available, L_{eff}

  $L_{eff} =$ ______

- Determine quantity of holes required, N_r

  $N_r = A_r/a$

- Determine maximum quantity of holes allowed, N_{max}

  $N_{max} = L_{eff}/2\ d$

- Compare qty of holes required with the max allowed, N_r to N_{max}

If $N_r \leq N_{max}$ the design is OK as is

If $N_r > N_{max}$ then another selection of hole or slot size must be selected

Case 2: 2-Phase Flow

- Area required, A_r

  $A_r = [(1.5\ V_m)/C][\rho m/\Delta P]$

- Determine quantity of holes required, N_r

  $N_r = A_r/a$

- Determine maximum quantity of holes allowed, N_{max}

  $N_{max} = L_{eff}/2\ d$

- Compare qty of holes required with the max allowed, N_r to N_{max}

If $N_r < N_{max}$ the design is OK as is

If $N_r > N_{max}$ then another selection of hole or slot size must be selected

Example

GIVEN;

$G = .523$

$\rho = 62.4\ G = 32.63$ PCF

$Q = 640$ GPM

$L_{eff} = 104''$

$\Delta P = 0.25$ psi

PIPE DATA;

Header Size: 6" Sch 40
Branch Size: None

$A_P = 28.9\ in^2$

$D_P = 6.095'' = .5079'$

CALCULATE;

- Velocity, V, FPS

  $V = (.408\ Q)/D_P^2$

  $= (.408(640))/6.095^2$

  $= 7.03$ FPS

- Area of holes required, A_r

  $A_r = [Q/(38\ C)][G/\Delta P]^{1/2}$

  $= [(640)/(38(.6)][.523/0.25]^{1/2}$

  $= 40.6\ in^2$

- Check area proportions;

  $A_r/A_P = 40.6/28.9 = 1.4$

  Between 1 and 3, therefore OK

- Assume hole or slot size;

  $d = 0.5$ in $a = .196$ in^2

- Determine quantity of holes required, N_r

  $N_r = A_r/a = 40.6/.196 = 208$

- Determine the max quantity of holes allowed, N_{max}

  $N_{max} = L_{eff}/2\,d = 104/2(.5) = 104$

  No Good!

  TRIAL 2
  Try slots 0.875 in × 2.25 in

  $d = 2.25$ $a = 1.967$ in^2

- Determine quantity of slots required, N_r

  $N_r = A_r/a = 40.6/1.967 = 21$

- Determine max quantity of slots allowed, N_{max}

  $N_{max} = L_{eff}/2\,d = 104/2(2.25) = 23$

  Therefore, design is acceptable.

Useful Conversion Factors

1. Given flow in GPM, find velocity, V, in Ft/Sec

   $V = [GPM(.408)]/D_P{}^2$

2. Given pressure in feet of head, convert to pressure in PSI;

   $P = (H\,G)/2.309$

3. Convert velocity and head;

   $H = V^2/(2\,g)$

   $V = (2\,g\,H)^{1/2}$

4. Given flow in GPM, find velocity in Ft^3/Sec

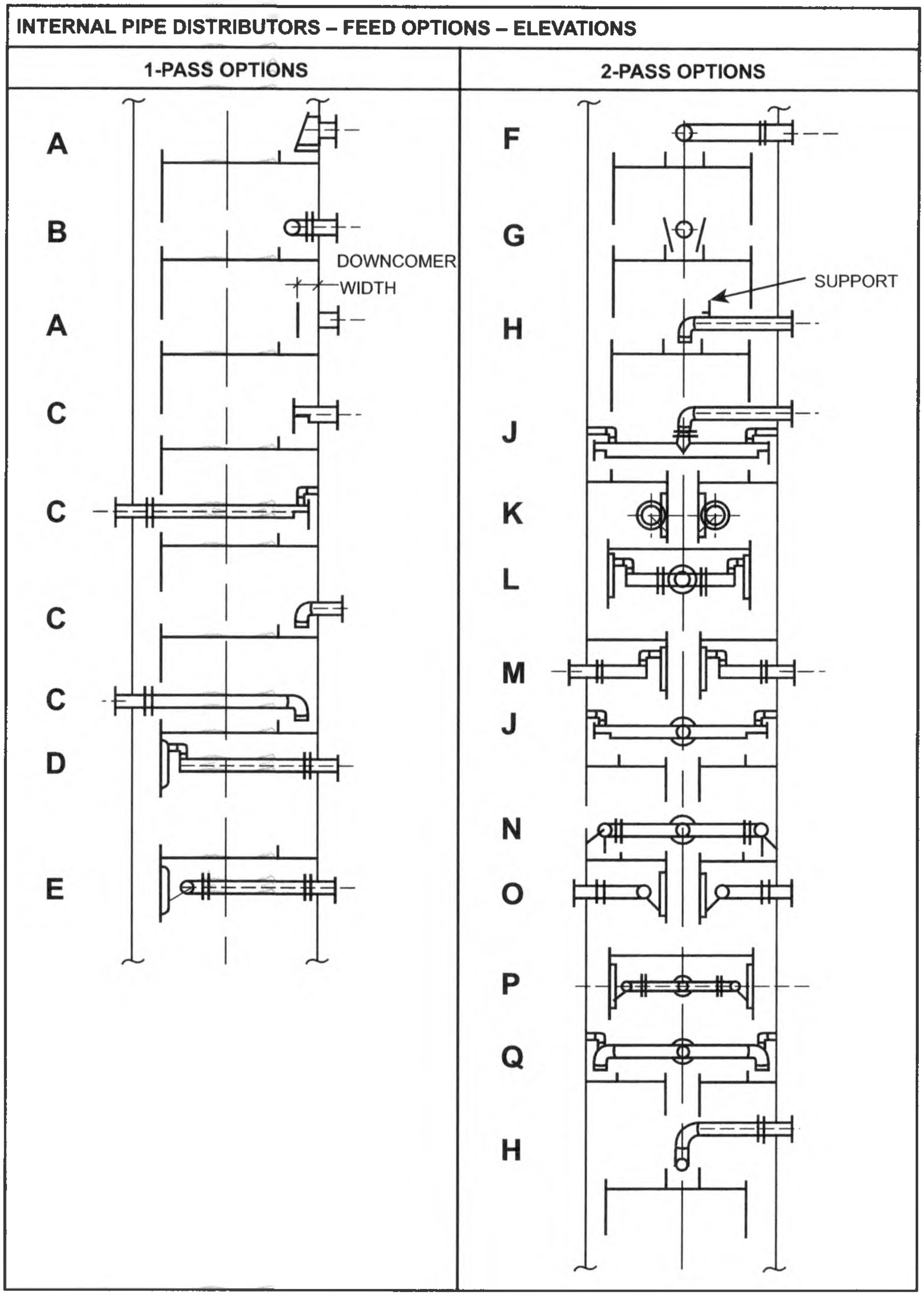
   $V = GPM/449$

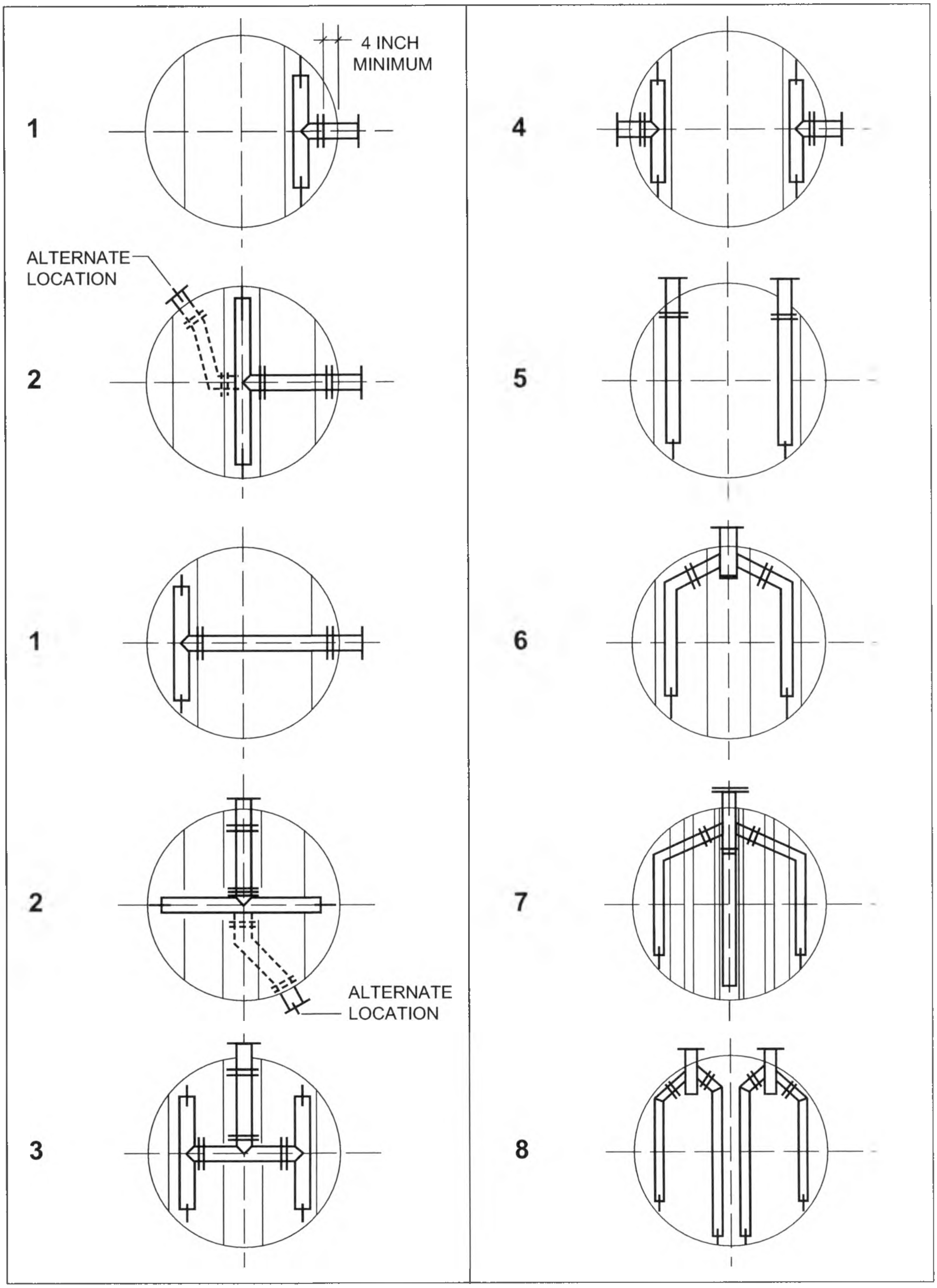
5. GPM equivalents;

   Lbs/Hr/500 G
   Lbs/Min/8.33 G
   Ft^3/Min/7.5
   449 × Ft^3/sec

6. Misc conversions;

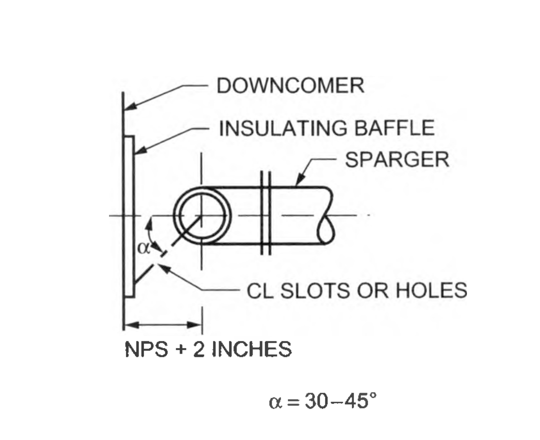
   1 gallon = 231 in^3 = 8.345 Lbs H_2O
   1 Ft^3 = 7.482 Gallons
   = 1728 in^3
   = 62.4 Lbs H_2O
   H = .433 PSI/Ft
   H = .433 G for other medium
   1 PSI = 2.309 feet of head
   $G = 141.5/(131.5 \times API^{o})$


INTERNAL PIPE DISTRIBUTORS – FEED OPTIONS – ELEVATIONS
1-PASS OPTIONS
2-PASS OPTIONS
A
B
A
C
C
C
C
D
E
DOWNCOMER
WIDTH
F
G
H
J
K
L
M
J
N
O
P
Q
H
SUPPORT

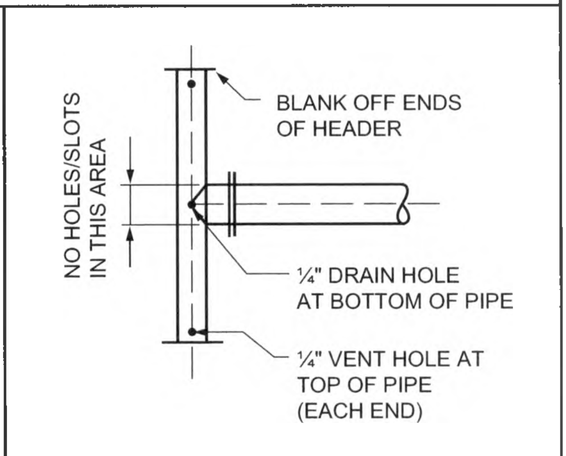

4 INCH
MINIMUM
1
ALTERNATE
LOCATION
2
1
2
ALTERNATE
LOCATION
3
4
5
6
7
8

INTERNAL PIPE DISTRIBUTORS - DESCRIPTIONS			
PLANS		ELEVATIONS	
1	T-TYPE LIQUID FEED FOR 1-PASS TRAY	A	INLET WITH BAFFLE
2	T-TYPE LIQUID FEED FOR 2-PASS TRAY	B	T-TYPE LIQUID FEED/REFLUX FOR 1-PASS TRAY
3	H-TYPE LIQUID/VAPOR FEED FOR 2 OR 4 PASS TRAY	C	SIDE FEED FOR 1-PASS TRAY
4	INDEPENDENT LIQUID/VAPOR FEED FOR 2-PASS TRAY	D	LIQUID/VAPOR FEED FOR 1-PASS TRAY
5	INDEPENDENT DUAL LIQUID/VAPOR FEED FOR 2 OR 4-PASS TRAY	E	T-TYPE LIQUID /VAPOR FEED FOR 1-PASS TRAY
6	COMBINED DUAL LIQUID/VAPOR FEED FOR 2 OR 4 PASS TRAY	F	T-TYPE LIQUID FEED/REFLUX CENTER DISCHARGE FOR 2-PASS TRAY
7	COMBINED TRIPLE LIQUID FEED FOR 4-PASS TRAY	G	SINGLE LINE FEED, CENTER DISCHARGE, WITH BAFFLES FOR 2-PASS TRAY
8	INDEPENDENT DUAL LIQUID/VAPOR FEED FOR 4-PASS TRAY	H	SINGLE LINE, CENTER DISCHARGE, LIQUID, FEED/REFLUX, WITH ELBOW FOR 2-PASS TRAY
		J	T-TYPE, SLOTTED END, LIQUID, FEED/REFLUX, SIDE DISCHARGE FOR 2-PASS TRAY
		K	DUAL, SINGLE LINE, LIQUID/VAPOR FEED AGAINST CENTER DOWNCOMER FOR 2-PASS TRAY
		L	T-TYPE, OPEN END, LIQUID/VAPOR FEED, SIDE DISCHARGE, FOR 2 PASS TRAY
		M	INDEPENDENT, DUAL, OPEN END, LIQUID/VAPOR FEED, CENTER DISCHARGE FOR 2 PASS TRAY
		N	H-TYPE, LIQUID, FEED/REFLUX, SIDE DISCHARGE, FOR 2-PASS TRAY
		O	INDEPENDENT, DUAL, T-TYPE, LIQUID/VAPOR FEED, CENTER DISCHARGE FOR 2-PASS TRAY
		P	H-TYPE, LIQUID/VAPOR FEED, SIDE DISCHARGE FOR 2 PASS TRAY
		Q	T-TYPE, LIQUID FEED/REFLUX, SIDE DISCHARGE WITH ELBOWS FOR 2-PASS TRAY

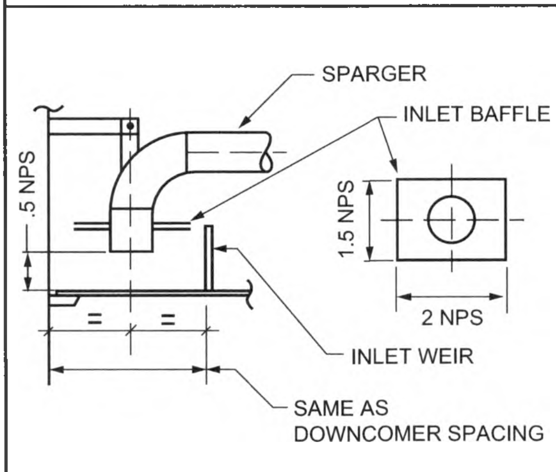
SPARGER DETAILS



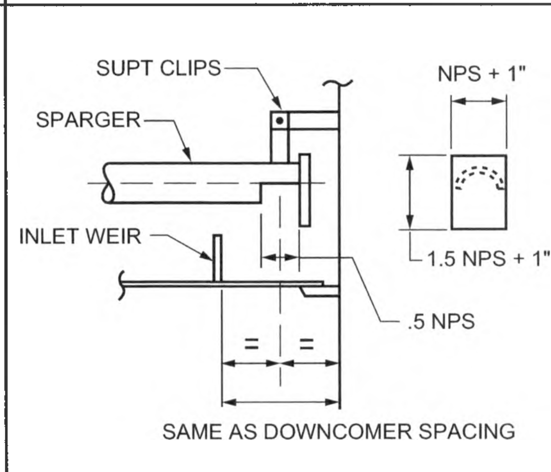
DETAIL 1



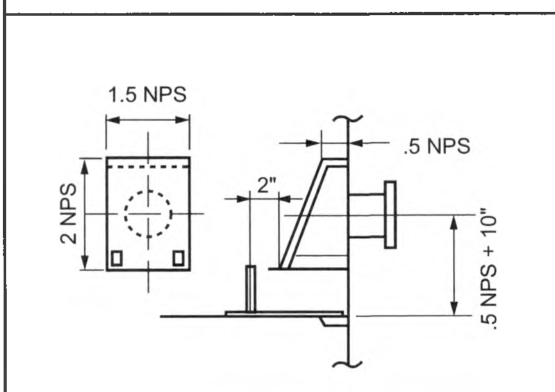
DETAIL 2



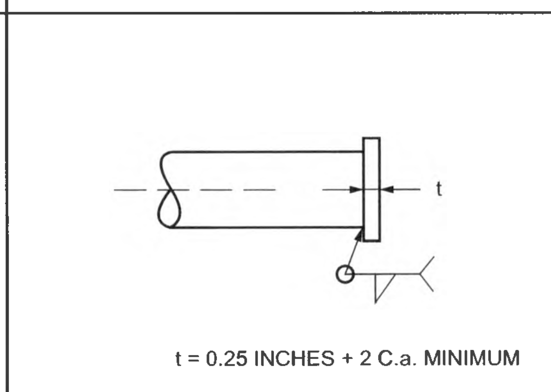
DETAIL 3



DETAIL 4

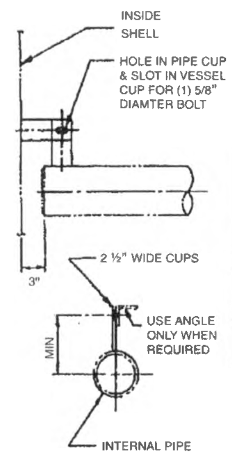


DETAIL 5

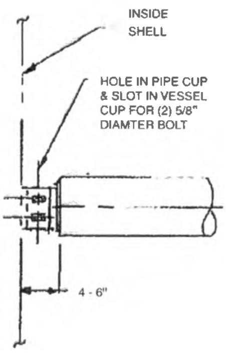


DETAIL 6

INTERNAL PIPE SUPPORT DESIGN

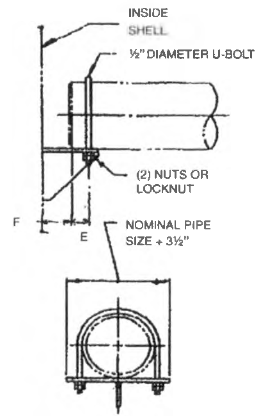


TYPE 1

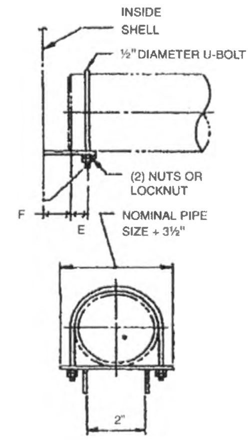


FOR 6 INCH & LARGER ONLY!

TYPE 1 ALTERNATE



TYPE 2



TYPE 3

TABLE 1: SUPPORT SELCTION GUIDE

		VESSEL DIAMETER			
		30-48 IN	48-78 IN	78-144 IN	144-240 IN
PIPE SIZE	2 IN	1	1		
	3 IN	1	1	1	
	4 IN	1	1	1	
	6 IN	1	1	1	1
	8 IN	1	1	2	2
	10 IN		2	2	2
	12 IN			2	2
	14 IN			2	2
	16 IN			2	3
	18 IN			3	3
	20 IN				3

NOTES

1. All plates are 1/4 inch thick + (2) times C.a (3/8 inch minimum)
2. Weld sizes are 1/4 inch + (2) times C.a (3/8 inch minimum)
3. F = 3/16 X pipe length in feet (3 inch minimum) in inches
4. E = 1/8 X pipe length in feet (2 inch minimum) in inches
5. Use Type 1 Alternate as appropriate
6. Table 1 shows recocommendation of pipe support type based on size of pipe and diameter of vessel

Table 5-15
Cross sectional area of pipe, A_p, in^2

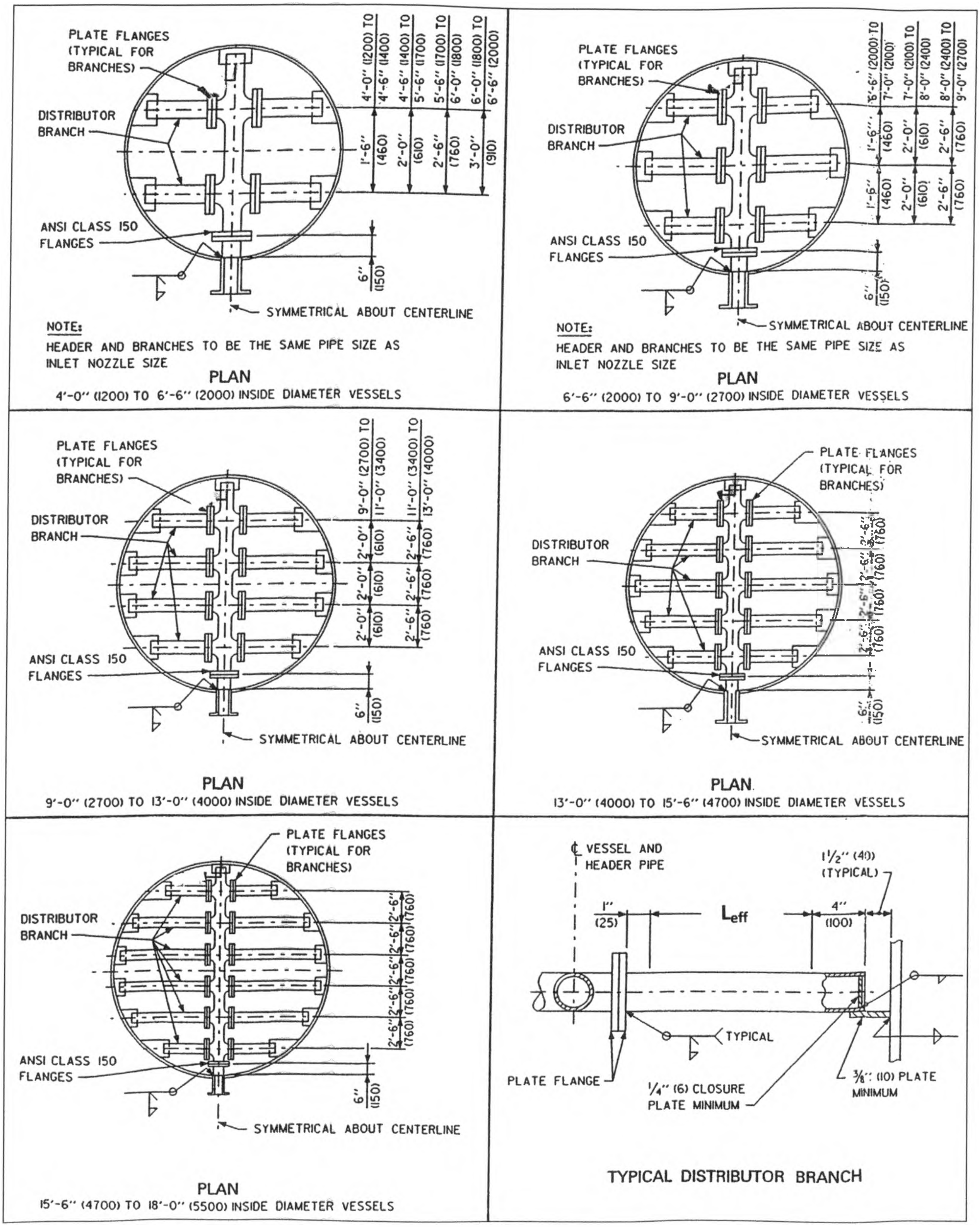
Pipe Size	Schedule			
	10	40	80	160
1	0.945	0.864	0.719	0.522
1.25	1.633	1.496	1.283	1.057
1.5	2.222	2.036	1.767	1.404
2	3.654	3.356	2.953	2.24
2.5	5.45	4.79	4.24	3.55
3	8.35	7.39	6.6	5.41
4	14.25	12.73	11.5	9.28
6	31.7	28.9	26.1	21.1
8	54.5	50	45.7	36.5
10	85.3	78.9	71.8	56.7
12	120.6	111.9	101.6	80.5
14	143.1	135.3	122.7	98.3
16	188.7	176.7	160.9	129
18	240.5	223.7	204.2	163.7
20	298.6	278	252.7	202.7

Table 5-16
Size (ID) of equalizing branches, inches

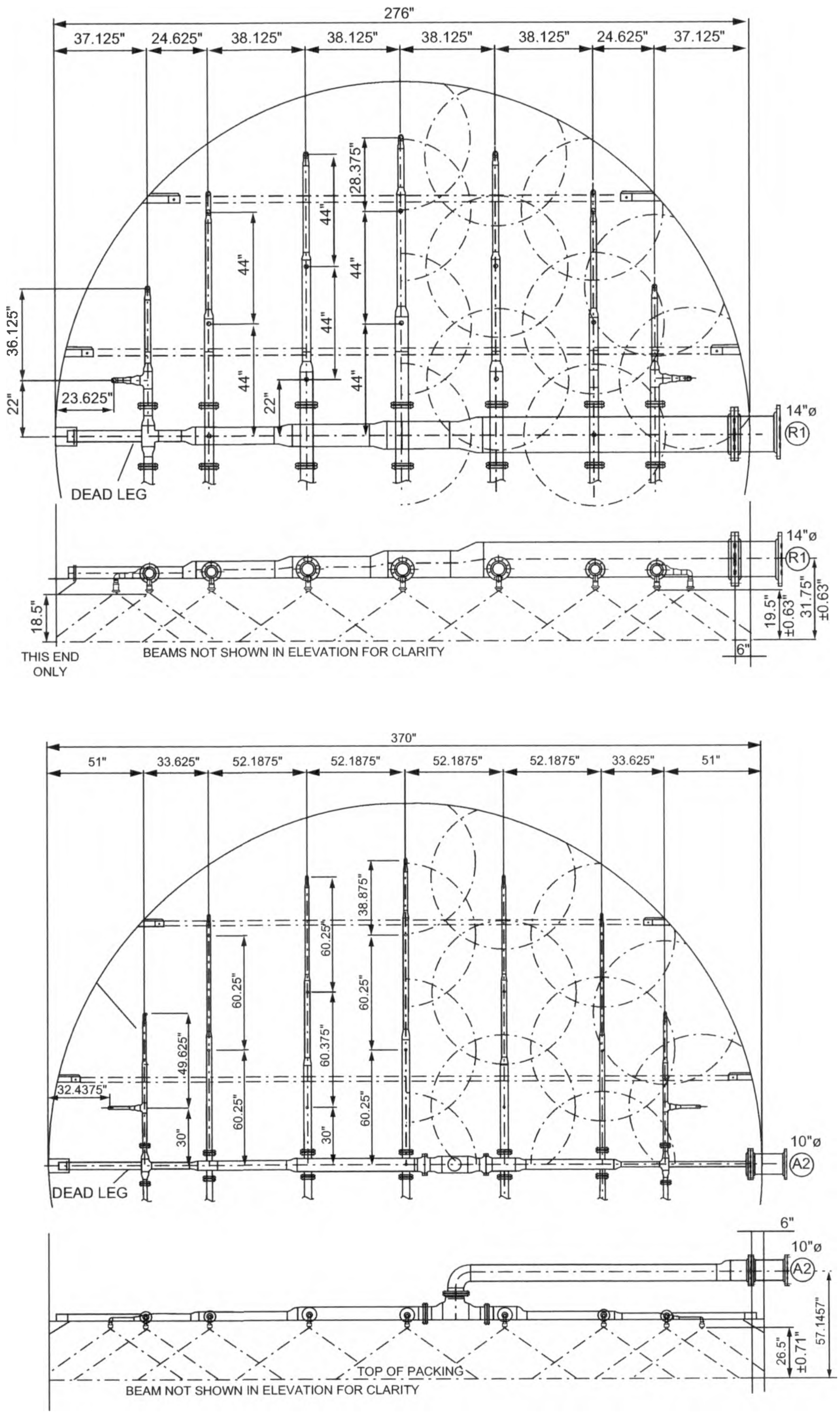
Size of Main Pipe	Area, Ap, in^2 (2)	Quantity of Branches (3)					
		2	4	6	8	10	12
2"	3.356	1.461	1.03				
3"	7.39	2.17	1.53	1.25			
4"	12.73	2.85	2.01	1.64	1.42		
6"	28.9	4.29	3.03	2.48	2.14	1.92	
8"	50	5.64	3.99	3.26	2.82	2.52	
10"	78.9	7.08	5	4.09	3.54	3.17	2.89
12"	111.9	8.44	5.97	4.87	4.22	3.77	3.45
14"	135.3	9.28	6.56	5.36	4.64	4.15	3.79
16"	176.7	10.6	7.5	6.12	5.3	4.74	4.33
18"	223.7	11.93	8.43	6.88	5.97	5.33	4.87
20"	278	13.3	9.4	7.68	7.11	5.95	5.43

Notes:

1. The table lists the exact ID required such that the cross sectional area of the main line and the sum of the cross sectional area of the branches is equal.
2. Assumes that main line is Sch 40.
3. Quantity of branches shown is total. Assume equal quantity for each side.



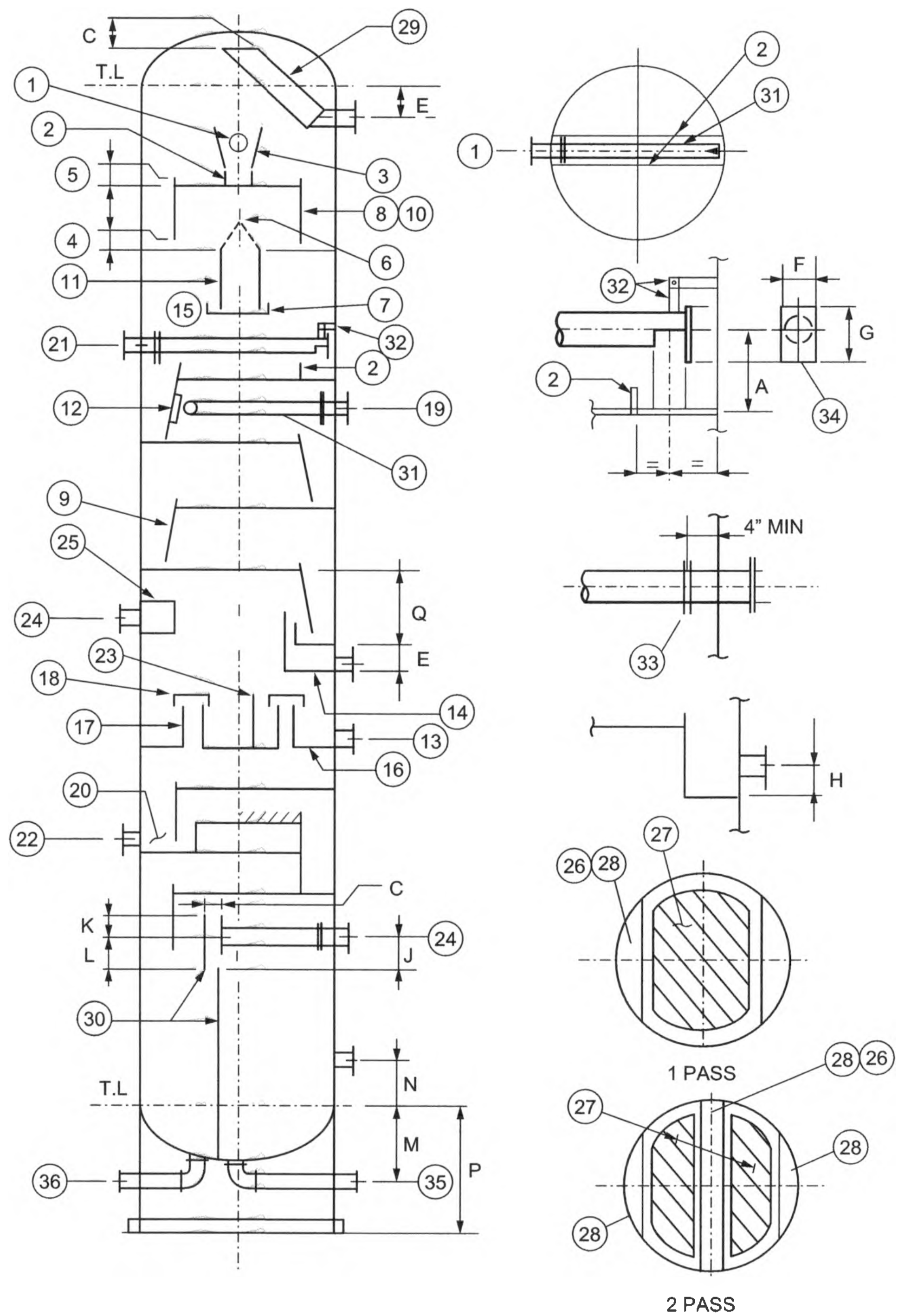
Header types with branches–Examples



SPRAY HEADERS - EXAMPLES

Procedure 5-9: Design of Trays

Typical Tower Dimensions and Nomenclature



Tray Types

1. Valve
2. Bubble Cap
3. Sieve
4. Tunnel
5. Chimney
6. Ripple
7. Shower
8. Shed Rows
9. Disc & Donut
10. Cartridge
11. Jet
12. Kittle
13. Turbogrid
14. Dual Flow
15. Slotted
16. Venturi
17. Cascade
18. Accumulator

Tray Nomenclature

1. Inlet/Feed/Reflux
2. Inlet Weir
3. Inlet Baffle
4. Downcomer Clearance
5. Weir Height
6. Anti-Jump Baffle
7. Seal Pan
8. Straight Downcomer
9. Sloped Downcomer
10. Side Downcomer
11. Center Downcomer
12. Insulating Baffle
13. Draw (Off) Nozzle
14. Draw (Off) Pan
15. Pass Transition
16. Total Draw Chimney Tray
17. Chimney
18. Chimney Hat
19. Return Nozzle
20. Sump
21. Intermediate Feed Nozzle
22. Partial Drawoff Nozzle
23. Partition
24. Vapor Inlet
25. Vapor Inlet Baffle
26. Feed Area
27. Active Tray Area
28. Dowcomer Area (Inactive Tray Area)
29. Vapor Outlet
30. Baffle
31. Internal Pipe
32. Support Clips
33. Internal Flanges
34. End Plate
35. Product Outlet Nozzle
36. Reboiler Draw Nozzle

Dimensions

A = Short = .5 NPS + 4 in

Long = .5 NPS + 10 in

B = Same as middle downcomer

C = .5 NPS

D = Same as weir height

E = NPS + 6 in

F = NPS + 1 in

G = 1.5 NPS + 1 in

H = .5 NPS + 1.5 in

J = 1.5 NPS

K = NPS

L = 2 NPS

M = Minimum with 2:1 S.E. head =

$.25\ D + t_H + 1.5\ NPS + 2\ in$

Minimum with hemi head =

$M = .5\ D + t_H + 1.5\ NPS + 2\ in$

N = Minimum distance from tangent line to nozzle centerline. See Table 5.19

P = Minimum skirt height for 2:1 S.E. Head. See Table 5-20

Q = Minimum seal pan depth = tray spacing + 6 in

R = Downcomer width

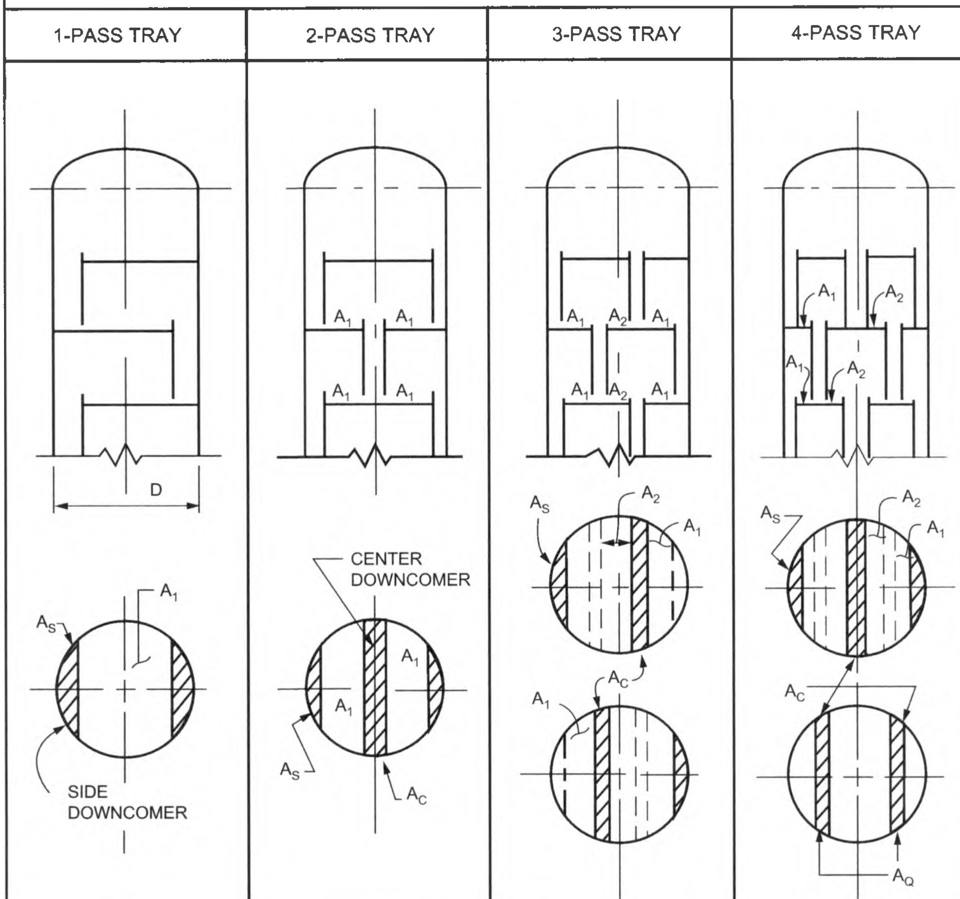
Table 5-18
Minimum Distance From Tangent Line, "N"

Noz Size	N	Noz Size	N
<2 in	5	12 in	16
3 in	7	14 in	18
4 in	8	16 in	20
6 in	10	18 in	24
8 in	12	20 in	27
10 in	14	24 in	30

Table 5-19
Minimum Skirt Height, "P"

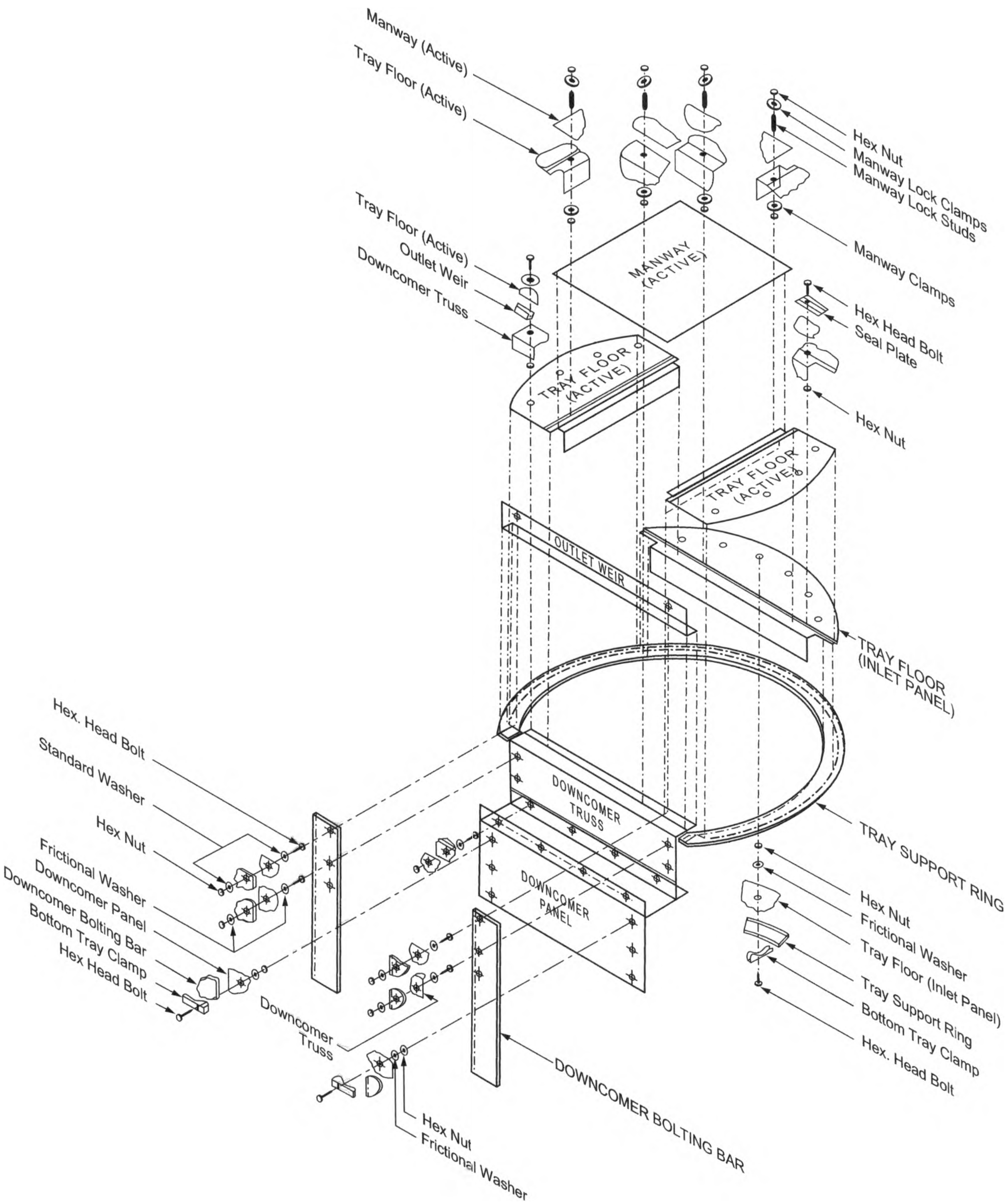
Dia	P	Dia	P
<24 in	2 ft – 6 in	114–120	5 ft – 6 in
30–36	3 ft – 0 in	126–132	6 ft – 0 in
48–54	3 ft – 6 in	138–144	6 ft – 6 in
54–60	3 ft – 6 in	150–156	7 ft – 0 in
66–72	4 ft – 0 in	162–168	8 ft – 0 in
78–84	4 ft – 6 in	174–180	8 ft – 6 in
90–106	5 ft – 0 in		

TRAYS

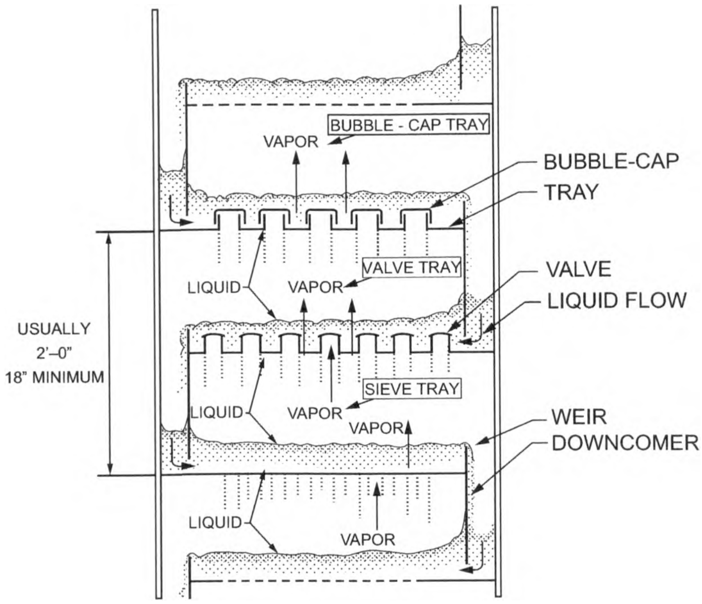


TRAY AND DOWNCOMER AREAS

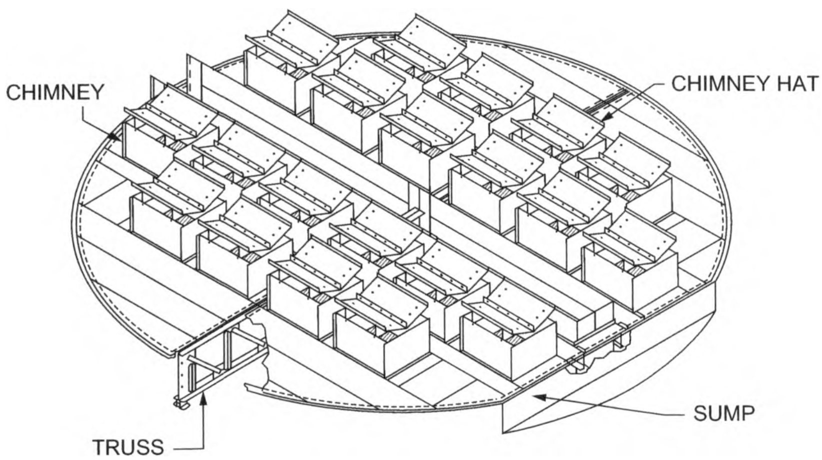
PASSES	TOTAL DC %	A_1	A_2	A_C	A_Q	A_S	MINIMUM DIAMETER	TOTAL WEIR LENGTH
1	8.08	$.6582\ D^2$				$.1268\ D^2$	30 IN	.6834 D
	12.25	$.5927\ D^2$				$.1923\ D^2$	30 IN	.7684 D
	16	$.5337\ D^2$				$.2512\ D^2$	30 IN	.8244 D
2	13.6	$.5715\ D^2$		$.1068\ D^2$		$.1068\ D^2$	48 IN	1.3 D
	17.9	$.5036\ D^2$		$.1407\ D^2$		$.1407\ D^2$	54 IN	1.408 D
	9.21	$.6408\ D^2$		$.0723\ D^2$		$.0723\ D^2$	48 IN	1.156 D
3	17.1	$.2832\ D^2$	$.2389\ D^2$	$.1949\ D^2$		$.0735\ D^2$	78 IN	1.5524 D
	23.65	$.2382\ D^2$	$.1754\ D^2$	$.2646\ D^2$		$.1068\ D^2$	96 IN	1.634 D
4	17.3	$.1712\ D^2$	$.3424\ D^2$	$.0906\ D^2$	$.1359\ D^2$	$.0435\ D^2$	108 IN	3 D
	21.95	$.1561\ D^2$	$.2845\ D^2$	$.1113\ D^2$	$.1724\ D^2$	$.0611\ D^2$	126 IN	3.097 D



TYPICAL TRAY ASSEMBLY
ONE PASS TRAY SHOWN



TYPES OF TRAYS



CHIMNEY/ACCUMULATOR TRAYS

Design of Tray Plates

Stress and Deflection

Case 1: Perforated Plate

Reference: Roark, 5th Edition, Table 26, Case 1A

- Rectangular plate
- Uniformly loaded
- All edges simply supported

DATA

α, β, γ = Coefficients from Table 5-21
E = Modulus of elasticity at design temperature, PSI
t = Corroded thickness of tray, in
C_a = Corrosion allowance, in
n = Hole efficiency
δ = Deflection at center, in
p = Uniform load, PSI
σ = Bending stress, PSI

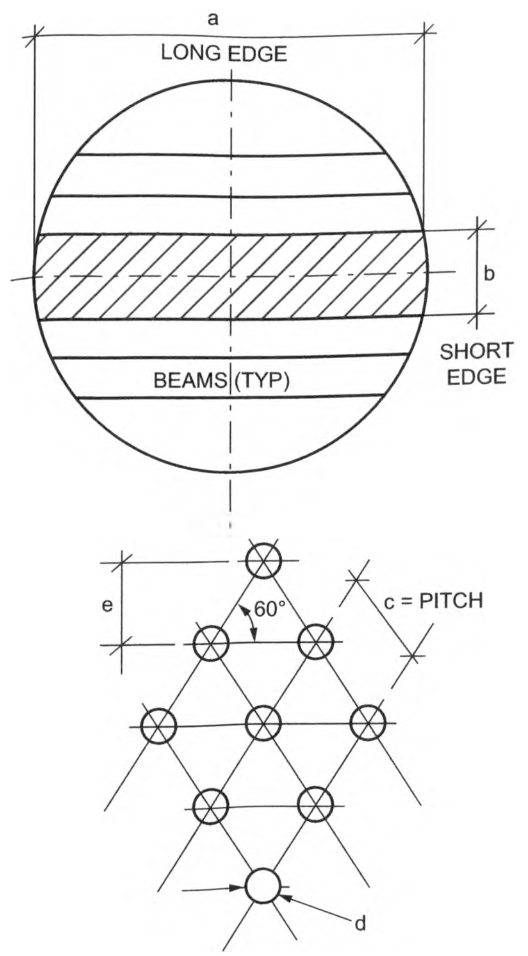
FORMULAS

- Stress, σ

$$\sigma = (\beta \, p \, b^4)/(n \, t^2)$$

Table 5-20
Coefficients

a/b	β	α	γ
1.0	.2874	.044	.420
1.2	.3762	.0616	.455
1.4	.4530	.0770	.478
1.6	.5172	.0906	.491
1.8	.5688	.1017	.499
2.0	.6102	.1110	.503
3.0	.7134	.1335	.505
4.0	.7410	.1400	.502
5.0	.7476	.1417	.501
∞	.7500	.1421	.500



- Deflection, δ

$$\delta = (\alpha\, p\, b^4)/(E\, n\, t^3)$$

- Efficiency of holes for perforated plate, n

$$n = 1 - (.25\,\pi\, d^2)/(e\, C)$$

DIMENSIONAL DATA

a =

b =

t =

a/b =

β =

α =

p =

d =

e =

C =

E =

Tray Design

Case 2: Standard tray plates

LOADS

A. DL = Dead load; weight of trays and beams

B. LL = Live load; dynamic load, ΔP

C. L_L = Liquid load; weight of liquid supported

- Total area, A_T

$$A_T = \pi D^2/4 = .7854\, D^2$$

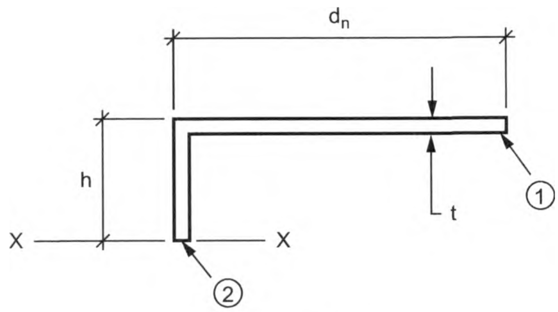
- Total load, P_T

$$P_T = DL + LL + L_L$$

- Uniform load, p

$$p = P_T/A_T$$

PROPERTIES OF SECTIONS



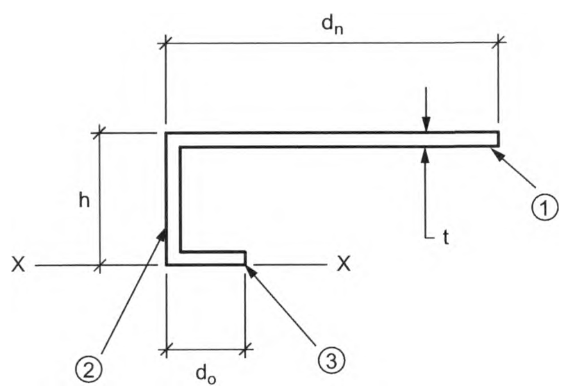
TYPE 1

PART	A	Y	AY	AY2	I
1					
2					
Σ		-----			

$C = \Sigma AY/\Sigma A$

$I = \Sigma AY^2 + \Sigma I - C\,\Sigma AY$

$Z = I/C$



TYPE 2

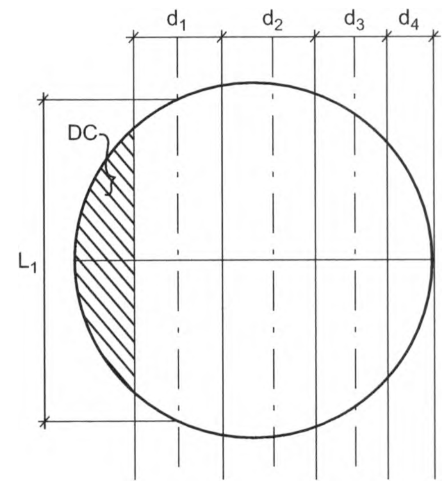
PART	A	Y	AY	AY2	I
1					
2					
3					
Σ		-----			

$C = \Sigma AY/\Sigma A$

$I = \Sigma AY^2 + \Sigma I - C\,\Sigma AY$

$Z = I/C$

CALCULATIONS



DIMENSIONS OF TRAY PANELS

- Determine panel area, A_P

  $A_P = L_n\, d_n$
- Load on panel, F_P

  $F_P = A_P\, p$

Uniform Load

- Uniform load on beam, w

  $w = F_P/L_n$
- Moment, M

  $M = (w\, L_n^2)/8$
- Bending stress, f_b

  $f_b = M/Z$
- Deflection, δ

  $\delta = (5\, w\, L_n^4)/(384\, E\, I)$

Concentrated Load

- Moment, M

  $M = (P\, L_n)/4$
- Bending stress, f_b

  $f_b = M/Z$
- Deflection, δ

  $\delta = (P\, L_n^3)/(48\, E\, I)$

NOTES

1. Design Loads: Trays, pans, drawoff boxes, or similar intemals, shall be designed using a corroded thickness, to support their own weight plus the following live loads;
   a. Fractionation trays: Design live load shall be the greater of 20 PSF (98 Kg/Sq meter) or the weight of water 2" (50 mm) over the highest weir setting.
   b. Areas under downcomers: Design live load shall be the greater of 64 PSF (314 Kg / Sq meter) or a head of water one half the height of the downcomer.
   c. Pans (accumulator and drawoff pans): Design live load shall be the greater of 1 PSI (700 Kg/ Sq meter) or the weight of water at the maximum operating level of the pan.
   d. Baffles: With no operating liquid level shall use a design live load of 1 PSI (700 Kg/Sq meter) on the projected horizontal area or actual impulse force, whichever is greater.
2. Maintenance Loads: Tray support members (all beams, support clips, etc.) shall be designed for a concentrated load of 300 Lbs (135 Kg) at any point on the installed assembly. The design shall be based on the corroded thicknesses and an allowable stress of .9 Fy. For maintenance loads, stresses in the tray plates need not be considered.
3. Uplift resistance: Typical design of uplift for trays and packing is .25 PSI (17 Millibar). For services where excessive uplift can occur, a higher uplift resistance may be used. Examples of higher uplift factors are as follows;
   a. Crude and FCC side strippers: 1 PSI (70 millibar)
   b. Vacuum tower stripping trays, overflash collector, wash trays or beds: 2 PSI (140 Millibar)
   c. Collectiors and packed beds above the wash section of a vacuum Tower: 1 PSI (70 Millibar)
4. Failure sequence: Tray assemblies, whenever possible shall be designed so that failure will occur in the following order;
   a. Tray manways
   b. Tray deck active areas
   c. Minor beams and downcomers
   d. Major beams (defined as beams 10ft (3 Meters) or longer or beams which extend across a vessel without interruption, regardless of length).
5. Allowable stresses: Allowable unit stress shall be based on yield strength, Fy, at design temperature with AISC factors for tension, bearing, shear, etc.
6. Allowable Deflection: The calculated combined deflection of trays and support beams due to operating loads shall not exceed the lesser of 1/900 of the vessel diameter (in inches) or 3/16" (5 mm)

RECOMMENDED TRAY SPACING & MANWAY SIZE

COLUMN DIA (FT)	TRAY SPACING (IN)	MANWAY SPACING	MIN MANWAY SIZE
2.5 to 16	24	Every 10 trays	18 in
16 to 24	30	Every 8 trays	20 in
24 to 32	36	Every 6 trays	24 in
32 and larger	42	Every 4 trays	30 in

Procedure 5-10: Flow Over Weirs

Notation

b = width, ft
H = static head of liquid, ft
Q = discharge rate, cu ft/sec
V = velocity of approach, ft/sec
H′ = head correction per Table 5-22

Table 5-21
Head correction for velocity of approach

V	0.4	0.6	0.8	1.0	1.2	1.4	1.6	1.8	2.0
H'	0.002	0.005	0.01	0.015	0.023	0.03	0.04	0.05	0.062
V	2.2	2.4	2.6	2.8	3.0	3.2	3.4	3.6	3.8
H'	0.075	0.089	0.105	0.122	0.14	0.15	0.179	0.201	0.213

Calculations

Discharge, Q

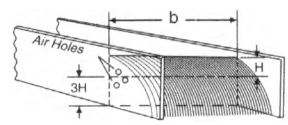
- For a full-length weir (Case 1)
  Q = 3.33b(1.5 H)
- For a contracted weir (Case 2)
  Q = 3.33b(1.5 H)
- For a V-notch weir (Case 3)
  Q = 6.33 H
- For a Cippoletti weir (Case 4)
  Q = 3.367b(1.5 H)

Notes

1. Assumes troughs are level

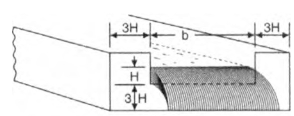
Case 1: Full-Width Weir

V = 1–2 ft/sec at 4 H upstream



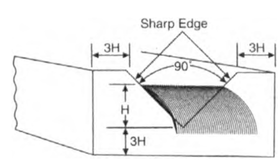
Case 2: Contracted Weir

V = 1–2 ft/sec at 3 H upstream



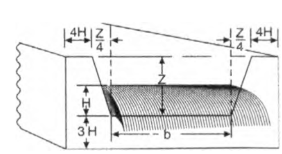
Case 3: V-Notch Weir

V = .5 ft/sec at 5 H upstream



Case 4: Cippoletti Weir

V = 1–2 ft/sec at 4 H upstream



Procedure 5-11: Design of Demisters

Demister pads or mist eliminators, are important internals in process vessels. Anytime there is a continuous two phase flow, vapor and liquid, there is the possibility for liquid entrainment. If it is desirable to separate the liquid and vapor, or prevent liquid carryover in the vapor stream, the velocity must be kept sufficiently low to allow the liquid droplets to fall out of the vapor stream. Demister pads are effective entrainment separators which allow operation at vapor velocities that would otherwise be excessive.

"Vapor disengagement" can be accomplished without a demister, but the vessel must be made much larger to accomplish the same degree of separation. This is known as gravity separation.

Demister pads are used in a wide variety of process vessels. Good performance can be achieved at velocities of 30% to 110% of optimum.

Demister pads can be used in vertical or horizontal vessels. The mesh pad orientation can be either vertical or horizontal in a vertical or horizontal vessel.

A wire mesh demister consists of a knitted wire mat mounted on a light weight support grid. The mesh is made by weaving small diameter metal wire into a mesh. Pads may be made out of metal or plastics. Metal type include carbon steel, stainless steel, monel and others. Plastic materials include PTFE, HDPE and PP.

Pressure drop is usually extremely low, in the range of 1 inch of water column maximum. The pressure drop is normally so low that it is ignored in design.

Vapor disengagement is dependent on the velocity of the stream. The most common equation for determining the allowable vapor velocity for two phase flows is as follows;

$$V_a = K\left[(\rho_L - \rho_V)/\rho_V\right]^{1/2}$$

Where;

V_a = Allowable vapor velocity, Ft/Sec
K = Constant or coefficient
ρ_L = Density of liquid, PCF
ρ_V = Density of vapor, PCF

Notes

1. The mesh pad shall be in one piece or sectional as required for insertion through the vessel manway. Each pad section shall be removable through the vessel manhole.
2. Layers shall not be more than 6 inches thick
3. Where more than one layer is required, all joints shall be staggered.
4. One piece circular pads may be spiral in form.
5. Each section of mesh shall include a grid on one or both sides. The grid, wire and/or fastening device shall be the same material as the mesh pad. Top and bottom grids shall be constructed of ¼ inch diameter rods welded to ⅛ inch by 1 inch bars. The maximum gap between mesh pad and grids shall be ½ inch for diameters less than 84 inches and 1 inch for larger diameters.
6. Estimate grid weights as 3 PSF.
7. Tie down wires shall be 1/16 inch diameter wire thru ¼ inch diameter holes on 4 inch centers. Tie wires should not be pushed through the pad as this could cause leak paths.
8. Pressure drop across mist eliminator shall be assumed as 0.2 PSI for support design.
9. Unless otherwise specified, support rings shall be 2 inches wide for pads 84 inches in diameter and less, and 3 inches wide for larger diameters.
10. The mist eliminator shall be a tight fit against the shell or enclosure as well as between sections. Mesh pads shall be ⅜ inches oversize all around for a force fit against the enclosure. No gaps are allowed.
11. Vessel fabricator shall provide the mesh and grids as well as any stiffeners required.
12. The Mist eliminator shall be specified as either top or bottom removable.
13. "K" is a constant, or coefficient that is a function of the efficiency of the separation. K-values range from .04 to .43, but are generally in the .15 to .35 range. The K-value is affected by pressure, type of pad, size of wire, type of vessel and disengaging height. For pressure applications, the K-value should be decreased by .01 for every 100 PSI of pressure after 100 PSI.
14. Wire diameter for the mesh pad itself vary from 0.003 to 0.016 inches.
15. The free volume of demisters range from 92 to 99.4%
16. The density of the pads ranges from 3 to 33 PCF
17. The NSA (nominal surface area) is the surface area to volume ratio, Ft^2 / Ft^3 and is an important indicator of performance. Ratios range from 50 to 600.
18. To obtain higher separation efficiencies, the wire diameter is decreased and the density and thickness are increased.
19. Typical width of sections for multi-section pads is 12 inches (300 mm) but may vary by manufacturer.

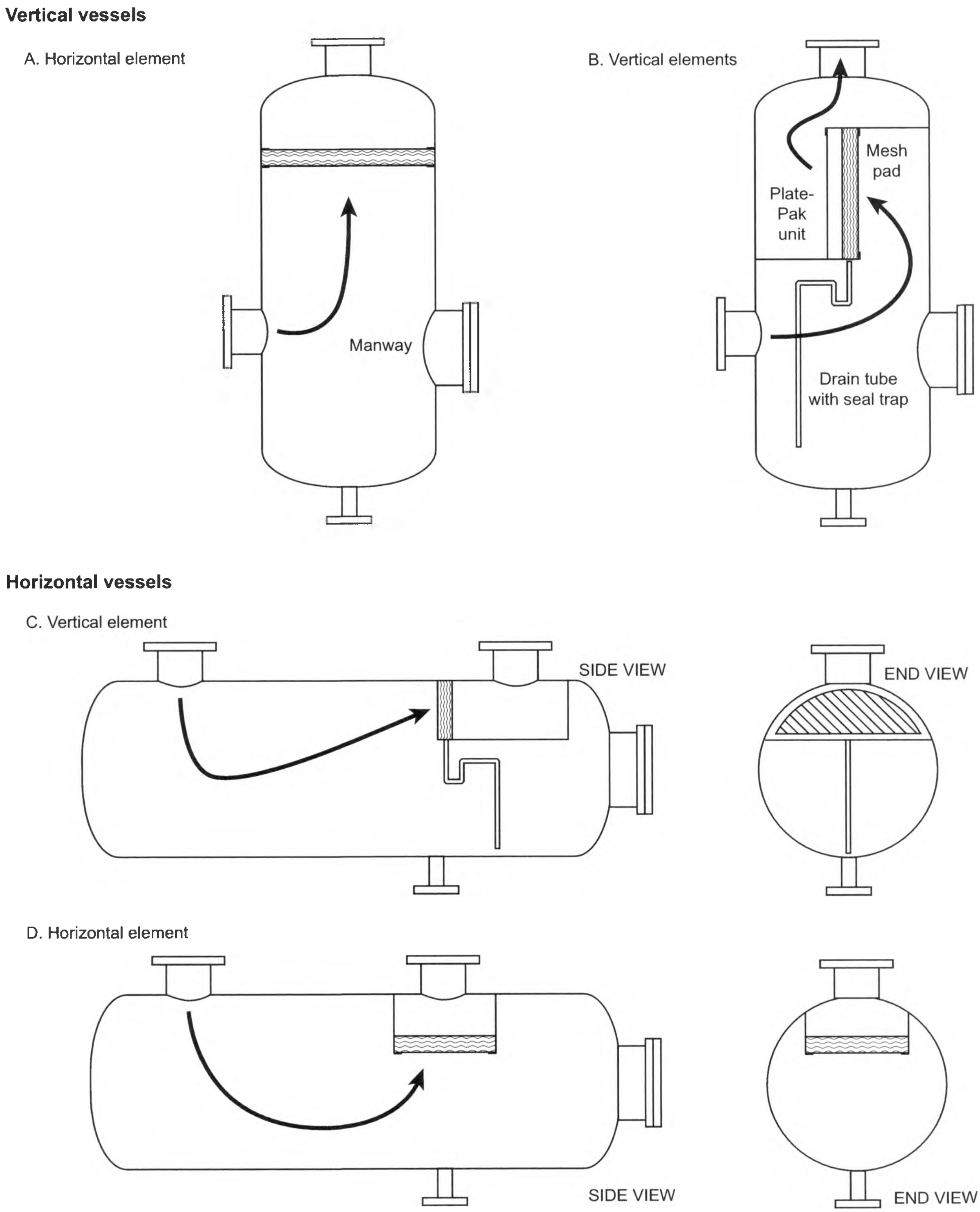


Figure 5-15. Typical demister configurations in vessels.

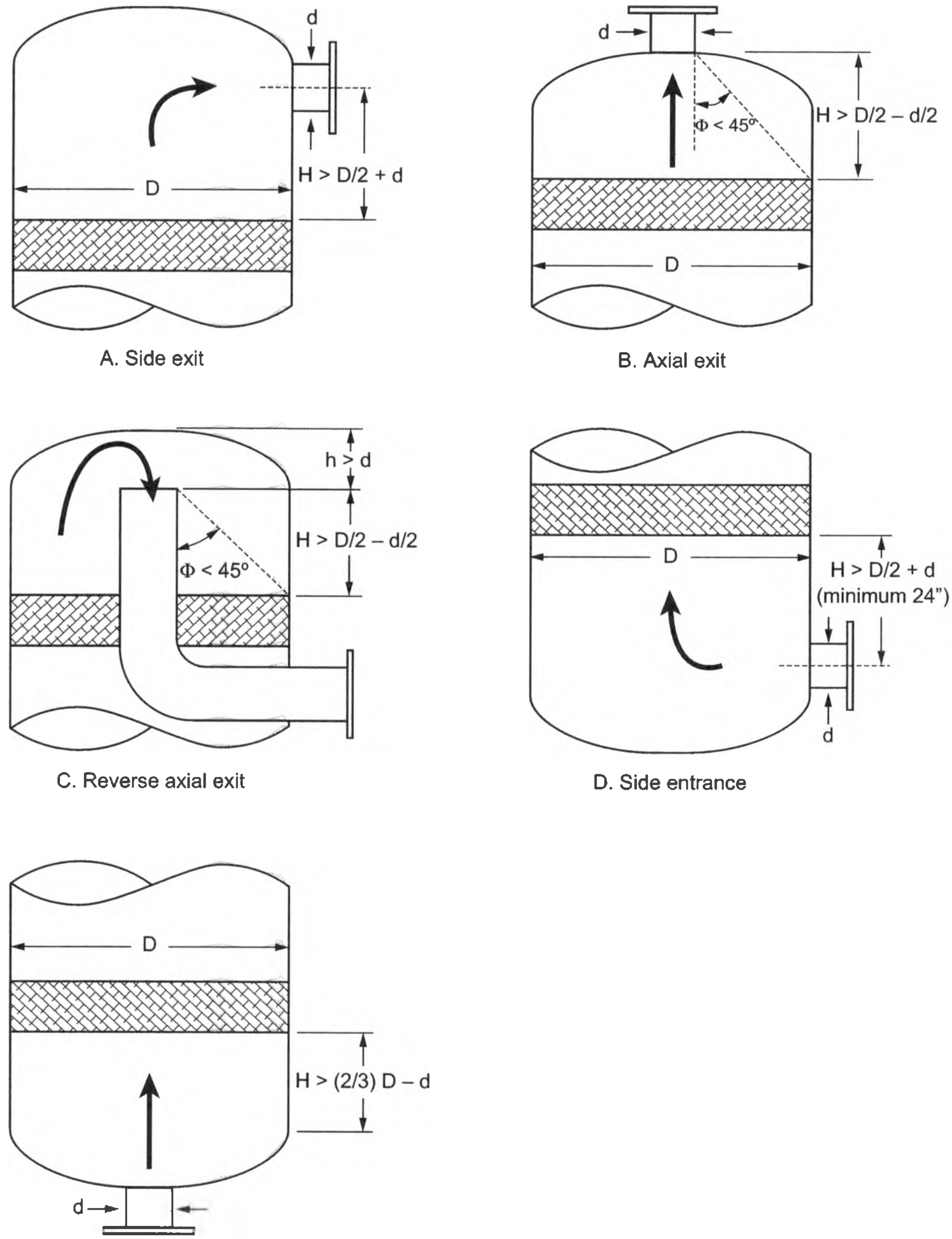


Figure 5-16. Guidelines for maintaining even flow distribution in vessels with axial flow.

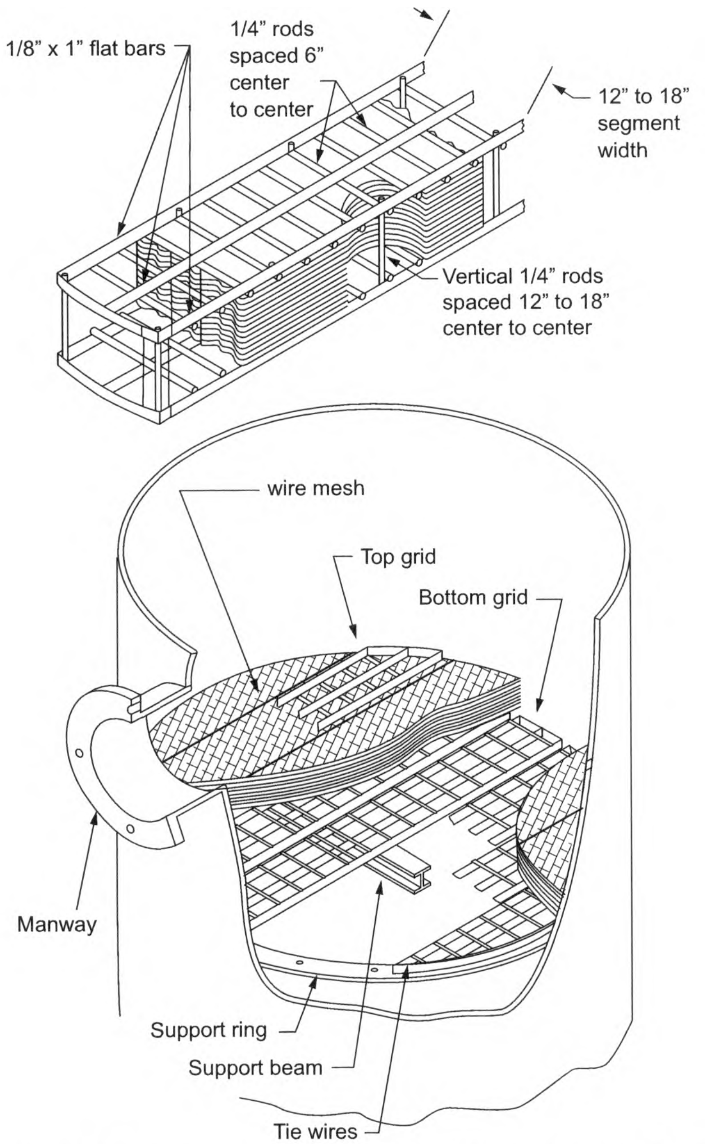


Figure 5-17. Typical mesh pad construction and installation.

Table 5-22
Properties of demister pads

Type	NSA (1) Ft^2 / Ft^3	Density, Pcf	Wire Dia, in	% Free Area	York Style No.	Remarks
A	48	5	0.011	99	931	High Throughput
B	65	7	0.011	98.6	531	Economy, Performance
C	85	9	0.011	98.2	431	Standard - Good All Around
D	110	10.8	0.011	97.7	421	Heavy Duty
E	140	8	0.006	98.4	326	Super High Efficiency - Fine Mist
F	163		0.006	94	371	Liquid - Liquid Coalescer
G	450	20	0.0045	95.9		
H	600	27	0.0045	94.5		

Notes:

1. NSA = Nominal Surface Area

Table 5-23
Values of K (Note 13)

Based on Pressure

Pressure (PSIa)		Vacuum (In Hg)	
Pressure (PSIa)	K	Vacuum (In Hg)	K
15	0.35	30	0.35
50	0.34	20	0.32
100	0.32	10	0.28
200	0.31	5	0.23
300	0.3	1	0.17
500	0.28	<1	0.17
1000	0.27		
>1000	0.27		

Based on Disengaging Height

Disengaging Height, in	K	Disengaging Height, in	K
3	0.12	9	0.32
4	0.15	10	0.35
5	0.19	11	0.38
6	0.22	12	0.4
7	0.25	13	0.42
8	0.29	14	0.43

Based on Application

Vertical Vessels		Horizontal Vessels	
Type	K	Type	K
General	0.35	General	0.35
Compressor Suction Drums	0.25	Steam Drums	0.25
Steam Drums	0.15		

Procedure 5-12: Design of Baffles [10]

Baffles are frequently used in pressure vessels, either vertical or horizontal, to divide the interior volume into different compartments. These compartments may be used to segregate liquids or provide overflow weirs for the separation of liquids. Baffles may be stiffened or unstiffened. When welded across the entire cross section of the vessel, they must be checked that they are not unduly restricting the diametral expansion of the vessel. If the unrestrained radial expansion of the vessel exceeds that of the baffle by more than $^1/_{16}$ in. ($^1/_8$ in. on the diameter), then a "flexible" type of connection between the vessel shell and the baffle should be utilized. Various flexible attachment designs are shown within the procedure.

Baffles should always be designed in the corroded condition. It is typical for welded baffles to be designed with a full corrosion allowance on both sides. If the baffle is bolted in, then one-half the full corrosion allowance may be applied to each side, the logic being that a bolted baffle is removable and therefore replaceable.

The majority of baffles are flat and as a result are very inefficient from a strength standpoint. Deflection is the governing case for flat plates loaded on one side. The preference is to have unstiffened baffles, and they should always be the first choice. This will be acceptable for small baffles. However, for larger baffles, as the baffle thickness becomes excessive, stiffeners offer a more economical design. Therefore stiffeners are frequently used to stiffen the baffle to prevent the thickness of the baffle from becoming excessive. The number, size, and spacing of stiffeners are dependent on the baffle thickness selected. There is a continual trade-off between baffle thickness and stiffener parameters.

The design of a baffle with stiffeners is an iterative process. The procedure for the design of the stiffeners is first to divide the baffle into "panel" sections that are rigid enough to withstand the pressure applied on one side. Each individual panel is checked as a flat plate of the dimensions of the panel. The stiffeners are assumed to be strong enough to provide the necessary edge support for the panel.

The stiffeners themselves are designed next. A section of the baffle is assumed as acting with the stiffener and as contributing to the overall stiffness. This combined section is known as the *composite* stiffener. The composite section is checked for stress and deflection. Both vertical and horizontal stiffeners can be added as required.

If required, an alternate design is assumed based on a thicker or thinner baffle and checked until a satisfactory design is found. There is no "right" answer; however, it should be noted that the thinner the baffle, the greater the number of stiffeners. The lightest overall weight is probably the "best" design but may not be the least expensive due to the welding costs in attaching the stiffeners.

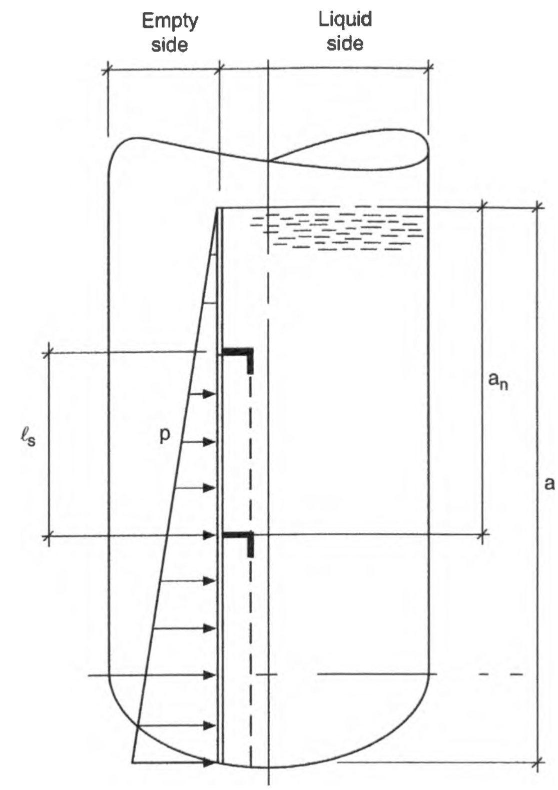
One alternative to a flat baffle with stiffeners is to go to a curved baffle. A curved baffle works best as a vertical baffle in a vertical vessel. The curved baffle takes pressure from either side wall. If the pressure is on the concave side the baffle is in tension. If the pressure is on the convex side, the baffle is in compression.

There are various tables given in this procedure for flat plate coefficients. Flat plate coefficients are utilized to determine the baffle thickness or a panel thickness. Each table is specific for a given condition and loading.

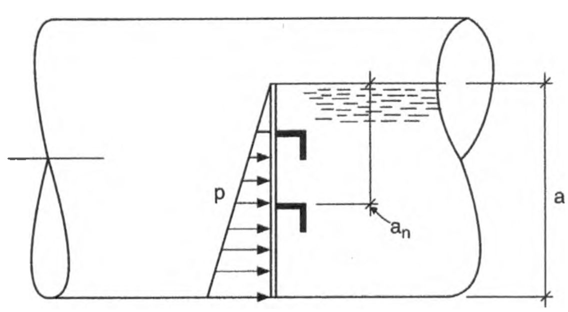
Notation

A_p = area of baffle working with stiffener, $in.^2$
A_s = area of stiffener, $in.^2$
C_p = distance from centroid of composite section to panel, in.
C_s = distance from centroid of composite section to stiffener, in.
E = modulus of elasticity, psi
F_b = allowable bending stress, psi
I = moment of inertia, composite, $in.^4$
I_s = moment of inertia, stiffener, $in.^4$
l = length of baffle that works with the stiffener, in.
M = moment, in.-lb
n = number of welds attaching stiffener
P = vessel internal pressure, psig
p = maximum uniform pressure, psi
p_n = uniform pressure at any elevation, a_n, psi
R_m = vessel mean radius, in.
S_g = specific gravity of contents
t = thickness, shell, in.
t_b = thickness, baffle, in.
t_s = thickness, stiffener, in.
V = shear load, lb
w = required fillet weld size, in.
α = thermal coefficient of expansion, in./in./°F
β, γ = flat plate coefficients
ΔT = differential temperature (design temperature minus 70°F), °F
σ_b = bending stress in baffle, psi
σ_s = bending stress in stiffener, psi
Δ_n = radial expansion, in.
δ = deflection, in.
δ_a = maximum allowable deflection, in.

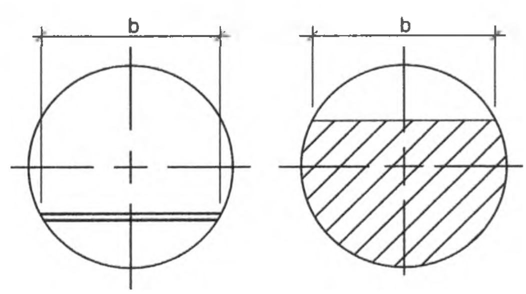
Baffle Dimensions



Vertical Vessel



Horizontal Vessel



Vertical Vessel Horizontal Vessel

Table 5-24
Flat plate coefficients

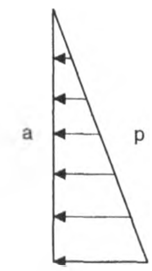
Equations

Case 1: One short edge free, three edges simply supported, uniformly decreasing load to the free edge

a/b / Coefficient	0.25	0.5	0.75	1	1.5
β_1	0.05	0.11	0.16	0.2	0.28
γ_1	0.013	0.026	0.033	0.04	0.05
a/b / Coefficient	**2**	**2.5**	**3**	**3.5**	**4**
β_1	0.32	0.35	0.36	0.37	0.37
γ_1	0.058	0.064	0.067	0.069	0.07

$$\sigma_b = \frac{\beta_1 p b^2}{t^2}$$

$$\delta = \frac{p\gamma_1 b^4}{E t_b^3}$$



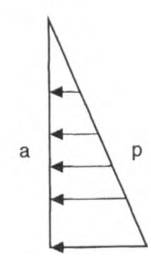
From Ref. 10, Section 6.5-4, Case 4d.

Case 2: All edges simply supported, uniform decreasing load

a/b / Coefficient	0.25	0.5	0.75	1	1.5
β_2	0.024	0.08	0.12	0.16	0.26
γ_2	0	0	0.01	0.02	0.04
a/b / Coefficient	**2**	**2.5**	**3**	**3.5**	**4**
β_2	0.32	0.35	0.37	0.38	0.38
γ_2	0.056	0.063	0.067	0.069	0.07

$$\sigma_b = \frac{\beta_2 p b^2}{t^2}$$

$$\delta = \frac{p\gamma_2 b^4}{E t_b^3}$$

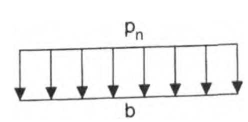


From Ref. 10, Section 6.5-4, Case 4c.

Case 3: All edges simply supported, uniform load

a/b / Coefficient	1	1.25	1.5	1.75	2
β_3	0.287	0.376	0.452	0.569	0.61
γ_3	0.0443	0.0616	0.077	0.1017	0.1106
a/b / Coefficient	**2.5**	**3**	**4**	**5**	**Infinity**
β_3	0.65	0.713	0.741	0.748	0.75
γ_3	0.125	0.1336	0.14	0.1416	0.1422

$$\sigma_b = \frac{\beta_3 p b_n^2}{t^2}$$



$$\delta = \gamma_3\left(\frac{p}{E}\right)\left(\frac{b_n^4}{t_b^3}\right)$$

- Assume p as a uniform load at center of plate.
- $A_n > b_n$

From Ref. 10, Section 6.5-4, Case 4a.

Unstiffened Baffle Check

- *Find load, p.*

$$p = \frac{62.4aS_g}{144}$$

- *Find baffle thickness, t_b.*

$$t_b = \sqrt{\frac{\beta_1 pb^2}{F_b}}$$

- *Find baffle deflection, δ.*

$$\delta = \frac{p\gamma_1 b^4}{Et_b^3}$$

Limit deflection to the smaller of $t_b/2$ or b/360. If deflection is excessive then:

a. Increase the baffle thickness.
b. Add stiffeners.
c. Go to curved baffle design.

If stiffeners are added, the first step is to find the maximum "a" and "b" dimensions that will meet the allowable deflection for a given panel size. This will establish the stiffener spacing for both horizontal and vertical stiffeners. The ultimate design is a balance between baffle thickness, stiffener spacing, and stiffener size.

Thermal Check of Baffle

- Vessel radial expansion due to pressure.

$$\Delta_1 = \frac{0.85PR_m}{tE}$$

- Vessel radial expansion due to temperature.

$$\Delta_2 = R_m\alpha\,\Delta T$$

- Thermal expansion of baffle.

$$\Delta_3 = 0.5b\alpha\,\Delta T$$

- Differential expansion.

$$\Delta_4 = \Delta_1 + \Delta_2 - \Delta_3$$

Stiffener Design

Divide baffle into panels to limit deflection to the lesser of $t_b/2$ or b/360. Deflection is calculated based on the appropriate Cases 1 through 3.

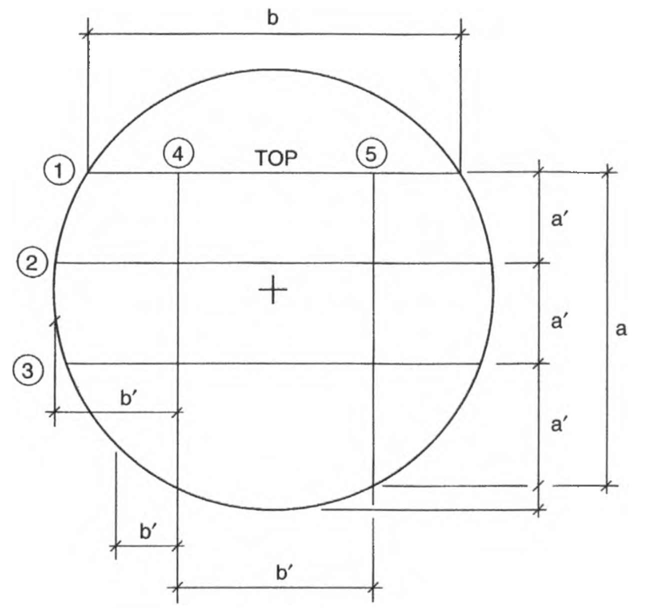
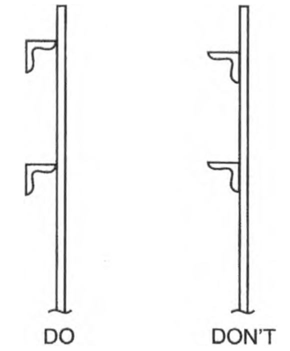


Figure 5-18. Example of stiffener layout.

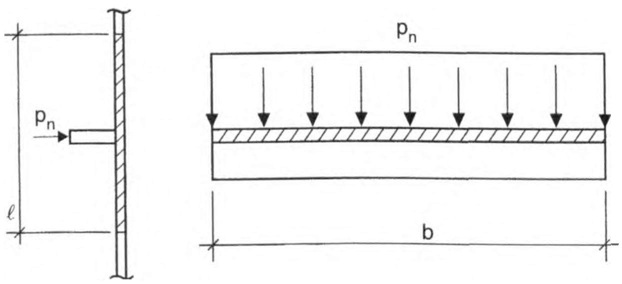
- *Check baffle for panel size a′ × b′.*
- *Check stiffener for length a or b.*

Recommendations for attaching stiffeners



Benefits: Provides added stiffness and no corrosion trap.

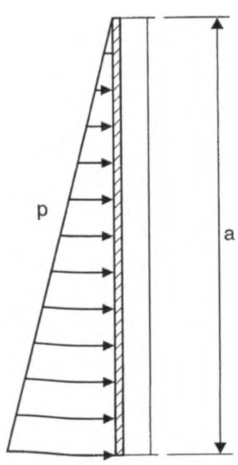
Horizontal Stiffener Design



$$p_n = \frac{a_n 62.4 S_g}{144} \qquad M = \frac{p_n l b^2}{8}$$

$$\delta = \frac{5 p_n l b^4}{384 EI} \qquad V = \frac{p_n l b}{2}$$

Vertical Stiffener Design



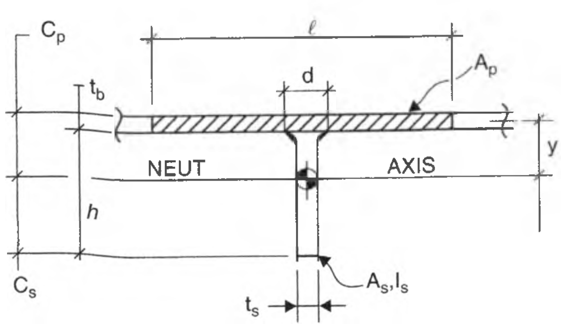
$$p = \frac{a 62.4 S_g}{144}$$

$$\delta = \frac{2.5 p l a^4}{384 EI}$$

$$M = .0642 p l a^2$$

$$V = \frac{p l a}{3}$$

Properties of Stiffener



$$A_s = t_s h$$

$$A_p = t_b l$$

$$I_S = \frac{t_s h^3}{12}$$

$$C_p = \frac{A_s y}{A_s + A_p} + \frac{t_b}{2}$$

$$C_s = (h + t_b) - C_p$$

$$I = I_S + \frac{A_p t_b^2}{12} + \frac{A_s A_p y^2}{A_s + A_p}$$

ℓ = lesser of $32t_b$ or stiffener spacing

Stresses in Baffle/Stiffener

$$\sigma_p = \frac{MC_p}{I}$$

$$\sigma_s = \frac{MC_s}{I}$$

Size Welds Attaching Stiffeners

For E70XX Welds:

$$w = \frac{Vdy}{11{,}200 In}$$

Table 5-25
Intermittent Welds

Percent of Continuous Weld	Length of Intermittent Welds and Distance Between Centers		
75%		3–4	
66			4–6
60		3–5	
57			4–7
50	2–4	3–6	4–8
44			4–9
43		3–7	
40	2–5		4–10
37		3–8	
33	2–6	3–9	4–12

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For E60XX Welds:

$$w = \frac{Vdy}{9600In}$$

Sample Problem

- Given: Horizontal vessel with a vertical baffle

$P = 250$ psig
D.T. $= 500^\circ F$
Material = SA-516-70
C.a. = 0.125 in.
JE = 1.0
$E = 27.3 \times 10^6$ psi
$\alpha = 7.124 \times 10^{-6}$ in./in./°F
$F_y = 30.8$ ksi
D = 240 in.
$F_b = 0.66\ F_y = 20.33$ ksi
$R_m = 120.938$

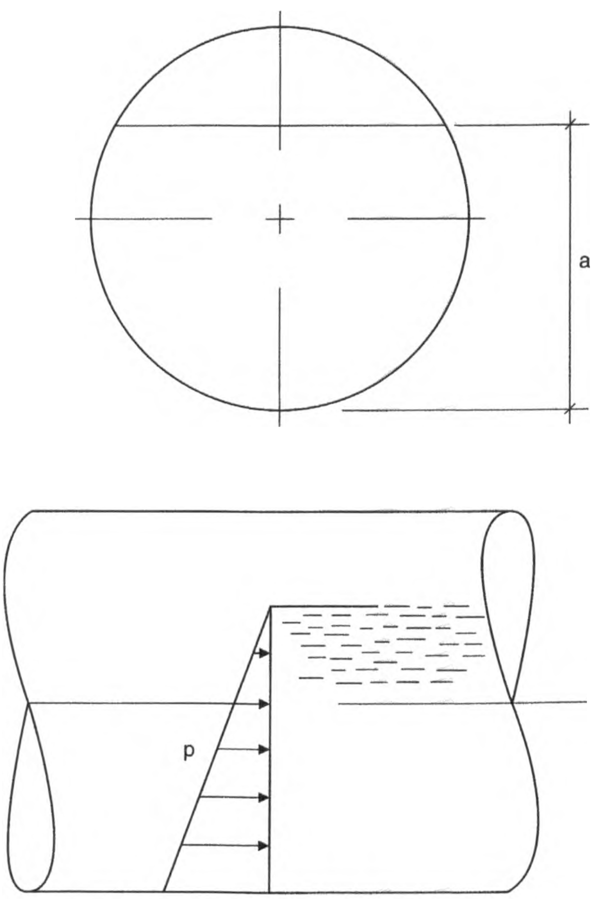


Figure 5-19. Sample problem.

$t_s = 1.75$ in.

$S_g = 0.8$

a = 15 ft

$\Delta T = 500 - 70 = 430^\circ F$

- *Find baffle thickness without stiffener.*

$$p = \frac{62.4aS_g}{144} = 5.2\ \text{psi}$$

$$\frac{a}{b}\text{ratio} = 15/20 = 0.75$$

from Table 5-25, Case 1:

$$\beta_1 = 0.16 \quad \gamma_1 = 0.033$$

- *Thickness of baffle, t_b.*

$$t_b = \sqrt{\frac{\beta_1 pb^2}{F_b}} = \sqrt{\frac{0.16(5.2)240^2}{20,330}}$$

$$t_b = 1.53 + 0.25 = 1.78$$

No good! Use stiffeners.

- *Assume a suitable baffle thickness and determine maximum panel size.*

  $t_b = 0.75$ in. corroded
  maximum panel size = 4ft × 4ft

- Maximum pressure, p.

$$p = \frac{13(62.4)0.8}{144} = 4.5\ \text{psi}$$

$$\frac{a}{b} = \frac{4}{4} = 1$$

See Table 5-25, Case 3:

$$\beta_3 = 0.287 \quad \gamma_3 = 0.0443$$

$$\sigma_b = \frac{\beta_3 pb_n^2}{t^2}$$

$$= \frac{0.287(4.5)48^2}{0.75^2} = 5290\ \text{psi} < 20,333\ \text{psi}$$

$$\delta = \gamma_3\left(\frac{p}{E}\right)\frac{b_n^4}{t^3} = 0.0443\left(\frac{4.5}{27.3\times10^6}\right)\left(\frac{48^4}{0.75^3}\right)$$

$= 0.092$ in. < 0.375 in.

Balance OK by inspection

- *Assume a layout where the maximum stiffener spacing is 4 ft.*

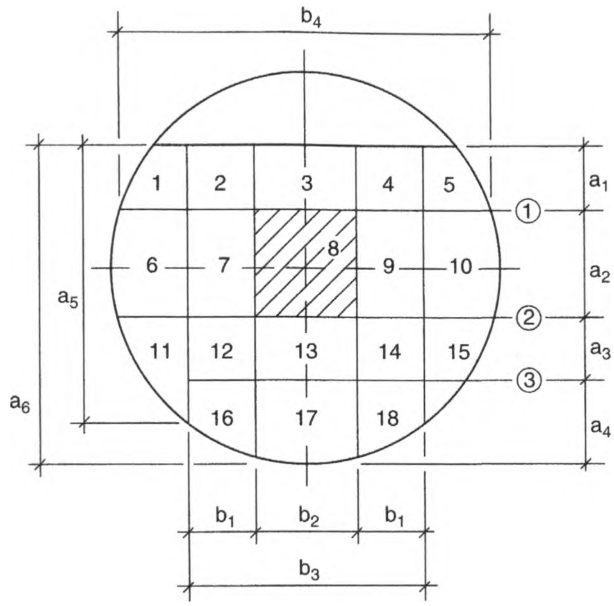


Figure 5-20. Battle layout for sample problem.

(4) horizontal stiffeners
(4) vertical stiffeners
(18) panels

- *Check horizontal stiffeners.*

Dimensions:

$a_1 = 3$ ft $b_1 = 4$ ft

$a_2 = 4$ ft $b_2 = 4$ ft

$a_3 = 4$ ft $b_3 = 12$ ft

$a_4 = 4$ ft $b_4 = 19.6$ ft

$a_5 = 13$ ft

$a_6 = 14.8$ft

- *Assume stiffener size, 1 in.* $\times$ *4 in.*

$y = 2.375$ in.

$A_s = t_s h = 1(4) = 4$ in.^{2}

$l = 32t_b = 32(0.75) = 24$ in. < 48 in.

$A_P = t_b l = 0.75(24) = 18$ in.^{2}

$$I_s = \frac{bh^3}{12} = \frac{1(4^3)}{12} = 5.33 \text{ in}^4.$$

$$I = I_s + \frac{A_p t_b^2}{12} + \frac{A_s A_p y^2}{A_s + A_p}$$

$$= 5.33 + 0.633 + 18.46 = 24.42 \text{ in.}^4$$

$$C_p = \frac{A_s y}{A_s + A_p} + \frac{t_b}{2}$$

$$= \frac{4(2.375)}{22} + \frac{0.75}{2} = 0.807 \text{ in.}$$

$$C_s = (h + tb) - C_p$$

$$= 4 + 0.75 - 0.807 = 3.943$$

Check deflections:

Item	b_n	p_n	δ
1	235.2	1.04	1.49
2	235.2	2.43	3.49
3	144	3.81	0.767

Deflections exceed allowable. No good!

- *Assume a larger stiffener size: WT9* $\times$ 59.3.
  $t_f = 1.06 - 0.25 = 0.81$
  $t_w = 0.625 - 0.25 = 0.375$
- *Check corroded thickness to find properties of corroded section.* This section would be equivalent to a WT9 $\times$ 30. Properties are:

$A_s = 8.82$ in.^{2}

$I_s = 64.7$ in.^{4}

$C_s = 2.16$ in.

$H = 9$ in.

$C_p = h + t_b - C_s$

$= 9 + 0.75 - 2.16 = 7.59$ in.

Table 5-26
Summary of Results for Stress and Deflection in Composite Stiffeners for Sample Problem

Item	Orientation	a_n	b_n	p_n	M	δ	V	σ_p	σ_s
1	Horiz.	235.2	—	1.04	172,595	0.09	2690	3504	997
2	Horiz.	235.2	—	2.43	403,275	0.21	6287	8186	2330
3	Horiz.	144	—	3.81	237,012	0.045	6035	4811	1370
4	Vert.	—	156	4.50	154,674	0.037	5648	3141	894
5	Vert.	—	177.6	5.13	228,539	0.072	6681	6681	1320

$$y = C_p - \frac{t_b}{2}$$

$$= 7.215 \text{ in.}$$

$$I = I_s + \frac{A_p t_b^2}{12} + \frac{A_s A_p y^2}{A_s + A_p}$$

$$= 64.7 + 0.844 + 308.14 = 373.7 \text{ in.}^4$$

- *Check stresses and deflections.* See results in Table 5-27.
- *Stresses and deflections are acceptable.*
- *Check welds.*

$$d = t_s + 2t_w = 0.375 + 2(0.323) = 1.02$$

$$y = 7.215 \text{ in.}$$

$$I = 373.7 \text{ in.}^4$$

$$n = 2$$

$$w = \frac{Vdy}{11{,}200In} = \frac{6681(1.02)7.215}{11{,}200(373.7)2}$$

$$= 0.005 + 0.125 = 0.13 \text{ in.}$$

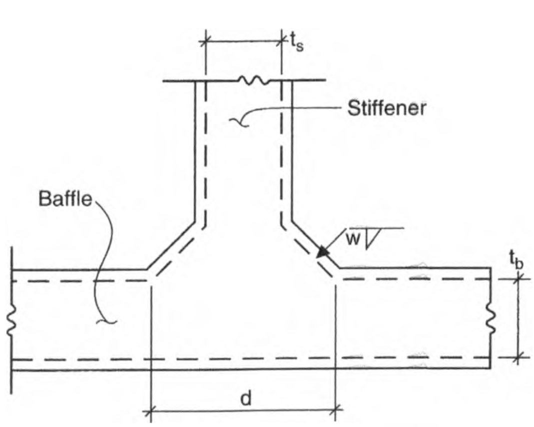


Figure 5-21. Details of weld attaching stiffener.

- *Check thermal expansion of baffle.*

$$\Delta_1 = \frac{0.85PR_m}{tE} = \frac{0.85(250)120.938}{1.75(27.3 \times 10^6)}$$

$$= 0.00054 \text{ in.}$$

$$\Delta_2 = R_m \alpha \; \Delta T = 120.938(7.124 \times 10^{-6})430$$

$$= 0.370 \text{ in.}$$

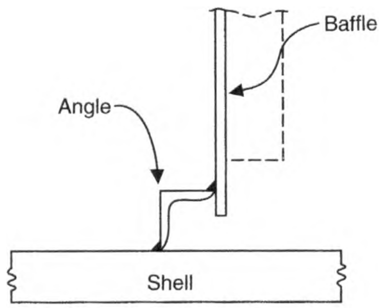
$$\Delta_3 = 0.5b\alpha \; \Delta T = 0.5(240)(7.124 \times 10^{-6})430$$

$$= 0.367 \text{ in.}$$

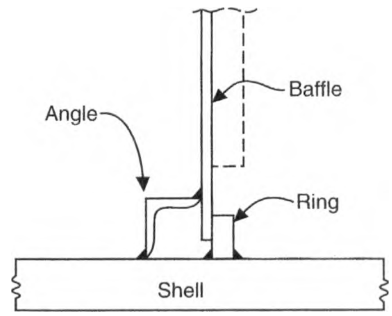
$$\Delta_4 = \Delta_1 + \Delta_2 - \Delta_3 = 0.00054 + 0.370 - 0.367$$

$$= 0.0035 < 0.06 \text{ in.}$$

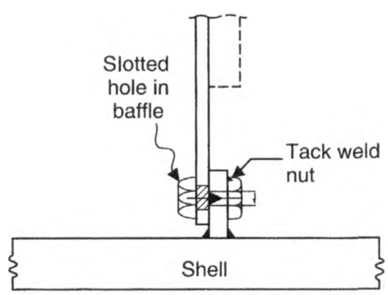
Flexible Baffle Design for Full-Cross-Section Baffles



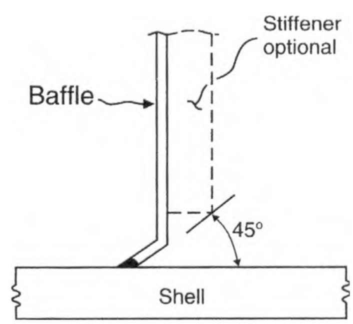
Attached by Angle to Shell



Attached by Angle,
Guided by Ring

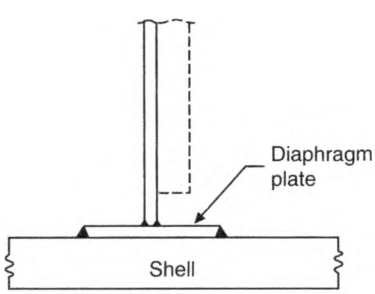


Baffle Bolted to Shell

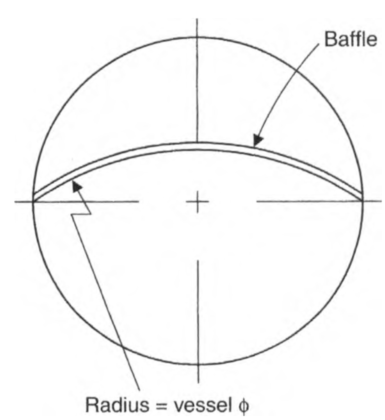


Baffle Welded to Shell

Note: Difficult to fabricate when
t > 3/8" or inside head



Diaphragm Plate

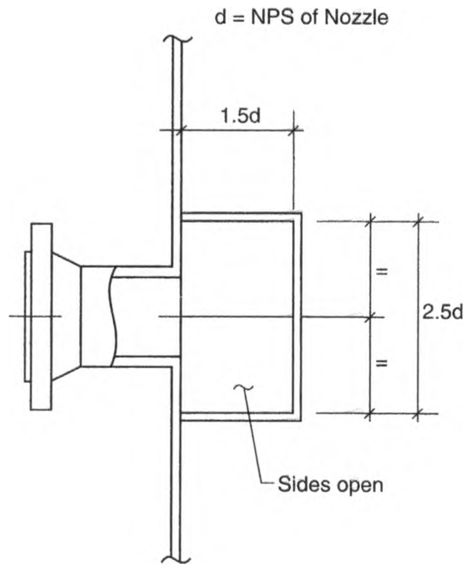


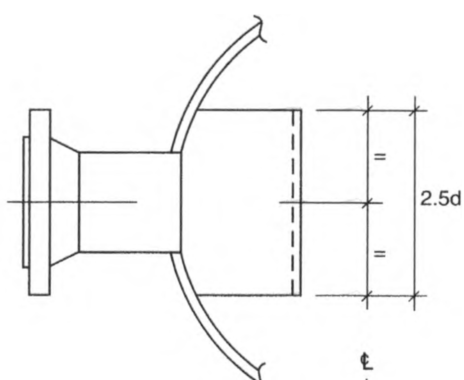
Alternate Construction

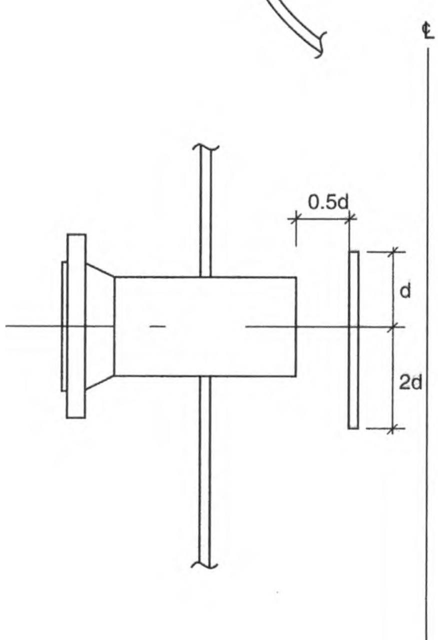
Benefits: Easily takes pressure
from either side and good for thermal
expansion

Miscellaneous Baffle Configurations

Vertical Vessels







Horizontal Vessels

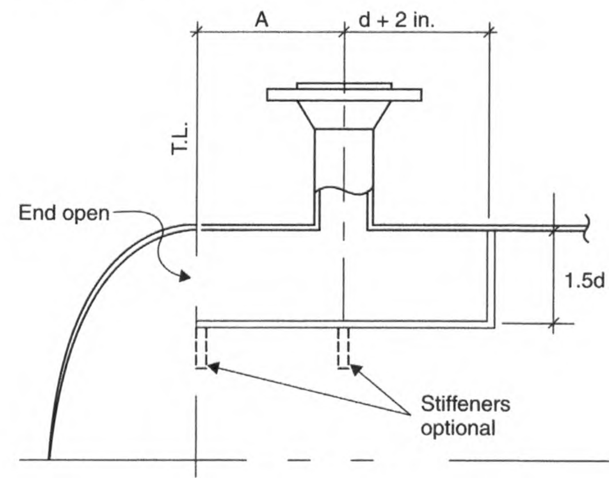
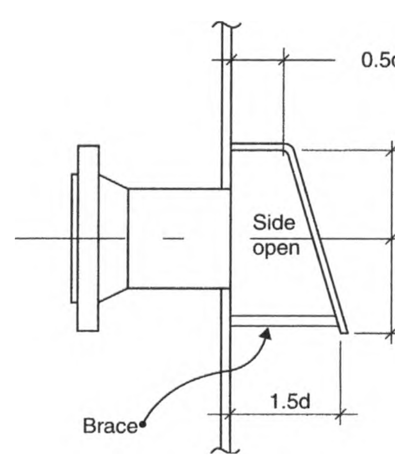


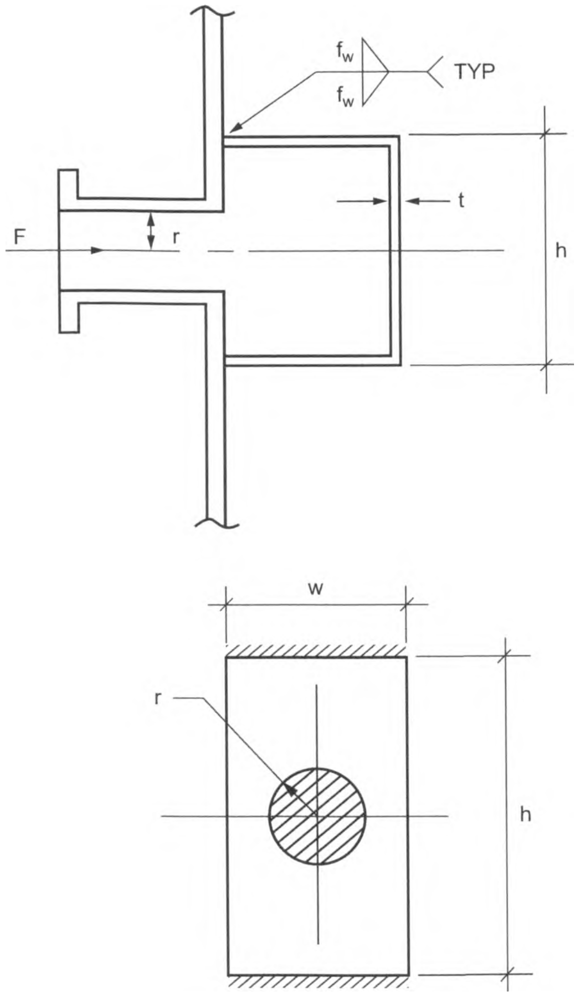
Table 5-27
Dimension "A"

Nozzle Size (in.)	A (in.)	Nozzle Size (in.)	A (in.)
2	5	14	18
3	7	16	20
4	8	18	22
6	10	20	24
8	12	24	28
10	14		
12	16		



Procedure 5-13: Design of Impingement Plates

Notation



Dimensions of baffle impingement plate

Figure 5-22. Dimensions of Baffle Impingement Plate.

A = Cross sectional area of nozzle, in^2
d = Density of liquid, PCF
F = Equivalent static force, Lbs
F_b = Allowable bending stress, PSI
F_y = Minimum specified yield strength, PSI
F_U = Minimum specified tensile strength, PSI
F_W = Allowable shear stress, PSI
f_b = Bending stress, PSI
f_w = Fillet weld size
g = Acceleration due to gravity, 32 Ft/Sec^2
L_W = Length of weld, in
r = Inside radius of nozzle, in
V = Velocity, FPS
τ_S = Shear stress per inch of weld, PSI

Calculation

- Equivalent static force, F

  $F = (V\,A\,d)/g$

  - Maximum bending stress, at center of plate, f_b

    $n = w/h < 1$ (Use 1 for square plate)

    $f_b = (5.3\,F)/(1 + 2.4\,n^2\,t^2)$

  - Allowable stress;

    a. Bending, F_b

    Lesser of… $.6\,F_y$ or $.3\,F_U$

    b. Weld in Shear, F_S

    $= .4\,F_y$

Size of weld required;

- Length of weld, L_W

  a. Welded from one side only;

  $L_W = 2\,w$

  b. Welded both sides, top and bottom;

  $L_W = 4\,w$

- Shear per inch of weld, τ_S

  $\tau_S = F/L_W$

- Size of weld required, fw

  $f_W = \tau_S/.707\,F_S$

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