THE PROCESS ENGINEER'S POCKET HANDBOOK

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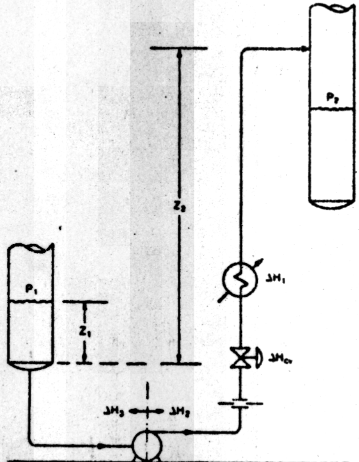

CARL BRANAN

To my family—Margaret, Kay, Ann, and Nana—for their patience and understanding during the preparation of this book.

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CONTENTS

This handbook is intended to be abbreviated, so no long introductions! This information was put together for a traveling man. However, the handbook will be useful for others short of time and space. It is not intended to be a book of conversion factors or property tables. Rather, it is a book of handy, brief approximation methods for a wide range of situations. Equations are used many times rather than charts, graphs, or nomographs. This permits desired miniaturization and with the advent of small hand held calculators the equations can be operated as quickly as graphs; especially so for programmable calculators. The no-nonsense format is aimed at easy retrieval of information and obtaining maximum mileage out of this handbook as a basic source.

1

FLUID FLOW

General

Two of the most useful and basic equations are

$$\Delta h = \frac{u^2}{2g} \tag{1.1}$$

$$\Delta P\,(V) + \frac{\Delta u^2}{2g} + \Delta Z + E = 0 \tag{1.2}$$

where

Δh = Head loss in feet of flowing fluid
u = Velocity in ft/sec
g = 32.2 ft/sec^2
P = Pressure in lb/ft^2
V = Specific volume in ft^3/lb
Z = Elevation in feet
E = Head loss due to friction in feet of flowing fluid

In Equation 1.1 Δh is called the "velocity head." This expression has a wide range of utility not appreciated by many. It is used "as is" for

1. Sizing the holes in a sparger
2. Calculating leakage through a small hole
3. Sizing a restriction orifice
4. Calculating the flow with a pitot tube

With a coefficient it is used for

1. Orifice calculations
2. Relating fitting losses, etc.

For a sparger consisting of a large pipe having small holes drilled along its length Equation 1.1 applies directly. This is because the hole diameter and the length of fluid travel passing through the hole are similar dimensions. An orifice on the other hand needs a coefficient in Equation 1.1 because hole diameter is a much larger dimension than length of travel (say ⅛″ for many orifices). Orifices will be discussed under "Metering" later in this chapter.

For compressible fluids one must be careful that when sonic or "choking" velocity is reached, further decreases in downstream pressure do not produce additional flow. This occurs at an upstream to downstream absolute pressure ratio of about 2:1. Critical flow due to sonic velocity has practically no application to liquids. The speed of sound in liquids is very high. See "Sonic Velocity" in this chapter.

Still more mileage can be gotten out of $\Delta h = u^2/2g$ when using it with Equation 1.2 which is the famous Bernoulli equation. The terms are

1. The PV change
2. The kinetic energy change or "velocity head"
3. The elevation change
4. The friction loss

These contribute to the flowing head loss in a pipe. However, there are many situations where by chance, or on purpose, $u^2/2g$ head is converted to PV or vice versa.

We purposely change $u^2/2g$ to PV gradually in the following situations:

1. Entering phase separator drums to cut down turbulence and promote separation
2. Entering vacuum condensers to cut down pressure drop

We build up PV and convert it in a controlled manner to $u^2/2g$ in a form of tank blender. These examples are discussed under appropriate sections.

Piping Pressure Drop

For most common fluids the following equation works quite well for ballpark checking and feasibility work:

$$W = 370\sqrt{\Delta P\,\rho d^5} \tag{1.3}$$

where

W = Flow in lb/hr
ΔP = Friction loss in lb/in² (psi) per 100 ft
ρ = Density in lb/ft³
d = Inside diameter of pipe in inches

This is a form of the Fanning or Darcy formula with friction factor = 0.0055. This friction factor corresponds to approximately the following:

Commercial steel pipes
Reynolds number = 10^5
Williams and Hazen C factor = 110

For other friction factors multiply the right hand side by

$$\sqrt{\frac{0.0055}{\text{friction factor}}} \tag{1.4}$$

The friction factor can be approximated by

Laminar flow,	$f = 16/\mathrm{Re}$
Commercial pipes,	$f = 0.054/\mathrm{Re}^{0.2}$
Smooth tubes,	$f = 0.046/\mathrm{Re}^{0.2}$
Extremely rough pipes,	$f = 0.013$

where Re is the well known Reynolds number. In this case:

$$\text{friction factor} = \frac{h_L}{4\,(L/D)\,(u^2/2g)} \tag{1.5}$$

where

h_L = Friction head loss in feet
L = Length in feet
D = Diameter in feet

Be careful when applying the friction factor. Sometimes it is defined as

$$\frac{h_L}{(L/D)\,(u^2/2g)}$$

Equation 1.3 applies to liquids. It also applies to compressible fluids for non-critical flow and $\Delta P < 10\%\ P_1$,

where

ΔP = Line pressure drop, psi
P_1 = Upstream pressure, psia

For compressible flow where $\Delta P > 10\%\ P_1$, either break into sections where $\Delta P < 10\%\ P_1$ or use

$$\Delta P = P_1 - P_2 = \frac{2P_1}{P_1 + P_2}\left[0.323\left(\frac{fL}{d} + \frac{\ln(P_1/P_2)}{24}\right) S_1 U_1^2\right] \tag{1.6}$$

from Reference 2 which assumes isothermal flow of ideal gas. In Equation 1.6

P_1, P_2 = Upstream and downstream pressures in psi ABS
S_1 = Specific gravity of vapor relative to water = $0.00150\ MP_1/T$
d = Pipe diameter in inches
U_1 = Upstream velocity, ft/sec
f = Friction factor (assume .005 for approximate work)
L = Length of pipe, feet
ΔP = Pressure drop in psi (rather than psi per standard length as before)
M = Mol. wt.

Table 1.1 gives the equivalent length of straight pipe for various fittings.

Two Phase Flow

Two phase flow is beyond the scope of this small handbook. One word of advice, however, be careful when designing low pressure, and/or flashing, condensate lines. These deserve special care. Reference 10 has a quick method for condensate line sizing.

Piping Rules of Thumb

Tables 1.2, 1.3, and 1.4 give typical piping rules of thumb.

Sonic Velocity

To determine sonic velocity use

$$V_s = \sqrt{KgRT} \tag{1.7}$$

where

V_s = Sonic velocity, ft/sec
K = C_p/C_v the ratio of specific heats at constant pressure to constant volume. This ratio is 1.4 for most diatomic gases.
g = 32.2 ft./sec^2

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Table 1.1. Equivalent Length of Valves and Fittings in Feet
(Reference 9)

Nominal Pipe size in.	Globe valve or ball check valve	Angle valve	Swing check valve	Plug cock	Gate or ball valve	45° ell		Short rad. ell		Long rad. ell		Hard T.		Soft T.		90° miter bends			Enlargement					Contraction				
																			Sudden			Std. red.		Sudden			Std. red.	
																			Equiv. L in terms of small d									
						Weld	thrd	Weld	thrd	Weld	thrd	Weld	thrd	Weld	thrd	2 miter	3 miter	4 miter	d/D=¼	d/D=½	d/D=¾	d/D=½	d/D=¾	d/D=¼	d/D=½	d/D=¾	d/D=½	d/D=¾
1½	55	26	13	7	1	1	2	3	5	2	3	8	9	2	3				5	3	1	4	1	3	2	1	1	—
2	70	33	17	14	2	2	3	4	5	3	4	10	11	3	4				7	4	1	5	1	3	3	1	1	—
2½	80	40	20	11	2	2	..	5	..	3	..	12		3	..				6	5	2	6	2	4	3	2	2	—
3	100	50	25	17	2	2		6		4		14		4					10	6	2	8	2	5	4	2	2	—
4	130	65	32	30	3	3		7		5		19		5					12	8	3	10	3	6	5	3	3	—
6	200	100	48	70	4	4		11		8		28		8					18	12	4	14	4	9	7	4	4	1
8	260	125	64	120	6	6		15		9		37		9					25	16	5	19	5	12	9	5	5	2
10	330	160	80	170	7	7		18		12		47		12					31	20	7	24	7	15	12	6	6	2
12	400	190	95	170	9	9		22		14		55		14		28	21	20	37	24	8	28	8	18	14	7	7	2
14	450	210	105	80	10	10		26		16		62		16		32	24	22	42	26	9	—	—	20	16	8	—	—
16	500	240	120	145	11	11		29		18		72		18		38	27	24	47	30	10	—	—	24	18	9	—	—
18	550	280	140	160	12	12		33		20		82		20		42	30	28	53	35	11	—	—	26	20	10	—	—
20	650	300	155	210	14	14		36		23		90		23		46	33	32	60	38	13	—	—	30	23	11	—	—
22	688	335	170	225	15	15		40		25		100		25		52	36	34	65	42	14	—	—	32	25	12	—	—
24	750	370	185	254	16	16		44		27		110		27		56	39	36	70	46	15	—	—	35	27	13	—	—
30	—	—	—	312	21	21		55		40		140		40		70	51	44										
36	—	—	—		25	25		66		47		170		47		84	60	52										
42	—	—	—		30	30		77		55		200		55		98	69	64										
48	—	—	—		35	35		88		65		220		65		112	81	72										
54	—	—	—		40	40		99		70		250		70		126	90	80										
60	—	—	—		45	45		110		80		260		80		190	99	92										

Table 1.2. Sizing Steam Piping in New Plants Maximum Allowable Flow and Pressure Drop

	Laterals			**Mains**		
Pressure, PSIG	600	175	30	600	175	30
Density, #/CF	0.91	0.41	0.106	0.91	0.41	0.106
ΔP, PSI/100′	1.0	0.70	0.50	0.70	0.40	0.30
Nominal Pipe Size, In.	**Maximum Lb/Hr x 10^{-3}**					
3	7.5	3.6	1.2	6.2	2.7	0.9
4	15	7.5	3.2	12	5.7	2.5
6	40	21	8.5	33	16	6.6
8	76	42	18	63	32	14
10	130	76	32	108	58	25
12	190	115	50	158	87	39
14	260	155	70	217	117	54
16	360	220	100	300	166	78
18	...	300	130	...	227	101
20	...	...	170	...	...	132

Note:

(1) 600 PSIG steam is at 750°F., 175 PSIG and 30 PSIG are saturated.

(2) On 600 PSIG flow ratings, internal pipe sizes for larger nominal diameters were taken as follows: 18/16.5″, 14/12.8″, 12/11.6″, 10/9.75″.

(3) If other actual I.D. pipe sizes are used, or if local superheat exists on 175 PSIG or 30 PSIG systems, the *allowable pressure drop* shall be the governing design criterium.

Table 1.3. Sizing Cooling Water Piping in New Plants Maximum Allowable Flow, Velocity and Pressure Drop

	LATERALS			**MAINS**		
Pipe Size in.	**Flow GPM**	**Vel. ft/sec.**	**ΔP ft/100′**	**Flow GPM**	**Vel. ft/sec.**	**ΔP ft/100′**
3	100	4.34	4.47	70	3.04	2.31
4	200	5.05	4.29	140	3.53	2.22
6	500	5.56	3.19	380	4.22	1.92
8	900	5.77	2.48	650	4.17	1.36
10	1,500	6.10	2.11	1,100	4.48	1.19
12	2,400	6.81	2.10	1,800	5.11	1.23
14	3,100	7.20	2.10	2,200	5.13	1.14
16	4,500	7.91	2.09	3,300	5.90	1.16
18	6,000	8.31	1.99	4,500	6.23	1.17
20	....	....	....	6,000	6.67	1.17
24	....	....	....	11,000	7.82	1.19
30	....	....	....	19,000	8.67	1.11

Table 1.4. Sizing Piping for Miscellaneous Fluids

Dry Gas	100 ft/sec
Wet Gas	60 ft/sec
High Pressure Steam	150 ft/sec
Low Pressure Steam	100 ft/sec
Air	100 ft/sec
Vapor Lines General	Max. velocity 0.3 mach 0.5 psi/100 ft
Light Volatile Liquid Near Bubble Pt. Pump Suction	0.5 ft head total suction line
Pump Discharge, Tower Reflux	3-5 psi/100 ft
Hot Oil Headers	1.5 psi/100 ft
Vacuum Vapor Lines below 50 MM Absolute Pressure	Allow max. of 5% absolute pressure for friction loss

R = 1544/mol. wt.
T = Absolute temperature in °R

To determine the critical pressure ratio for gas sonic velocity across a nozzle or orifice use

$$\textit{critical pressure ratio} = [2/(K+1)]^{k/(k-1)} \tag{1.8}$$

If pressure drop is high enough to exceed the critical ratio, sonic velocity will be reached. When $K = 1.4$, ratio = 0.53.

Control Valve Design

Notes:

1. References 4 and 5 were used extensively for this section. The sizing procedure is generally that of Fisher Controls Company.
2. Use manufacturers' data where available. This handbook will provide approximate parameters applicable to a wide range of manufacturers.
3. For any control valve design be sure to use one of the modern methods, such as that given here, that takes into account such things as control valve pressure recovery factors and gas transition to incompressible flow at critical pressure drop.

Liquid Flow

Recall discussion of converting PV to $u^2/2g$. Across a control valve the fluid is accelerated to some maximum velocity. At this point the pressure reduces to its lowest value. If this pressure is lower than the liquid's vapor pressure, flashing will produce bubbles or cavities of vapor. The pressure will rise or "recover" downstream of the lowest pressure point. If the pressure rises to above the vapor pressure, the bubbles or cavities collapse. This causes noise, vibration, and physical damage.

When there is a choice, design for no flashing. When there is no choice, locate the valve to flash into a vessel if possible. If flashing or cavitation cannot be avoided, select hardware that can withstand these severe conditions. The downstream line will have to be sized for two phase flow. It is suggested to use a long conical adaptor from the control valve to the downstream line.

When sizing liquid control valves first use

$$\Delta P_{allow} = K_m(P_1 - r_c P_v) \tag{1.9}$$

where

ΔP_{allow} = Maximum allowable differential pressure for sizing purposes, psi
K_m = Valve recovery coefficient (see Table 1.7)
r_c = Critical pressure ratio (see Figures 1.1 and 1.2)
P_1 = Body inlet pressure, psia
P_v = Vapor pressure of liquid at body inlet temperature, psia

This gives the maximum ΔP that is effective in producing flow. Above this ΔP no additional flow will be produced since flow will be restricted by flashing. Do not use a number higher than ΔP_{allow} in the liquid sizing formula. Some designers use as the minimum pressure for flash check the upstream absolute pressure minus two times control valve pressure drop.

Table 1.5 gives critical pressures for miscellaneous fluids. Table 1.6 gives relative flow capacities of various types of control valves. This is a rough guide to use in lieu of manufacturer's data.

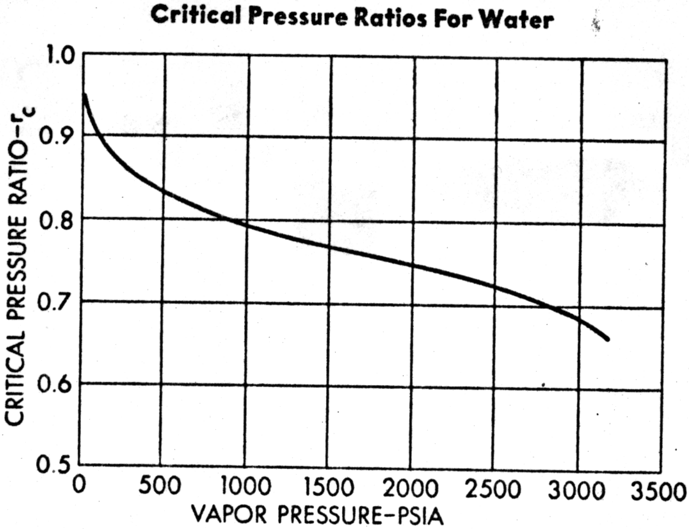

Figure 1.1. Enter on the abscissa at the water vapor pressure at the valve inlet. Proceed vertically to intersect the curve. Move horizontally to the left to read r_c on the ordinate (Reference 4).

The liquid sizing formula is

$$C_v = Q\sqrt{\frac{G}{\Delta P}} \tag{1.10}$$

where

C_v = Liquid sizing coefficient
Q = Flow rate in GPM
ΔP = Body differential pressure, psi
G = Specific gravity (water at 60°F = 1.0)

Two liquid control valve sizing rules of thumb are

Critical Pressure Ratios For Liquids Other Than Water

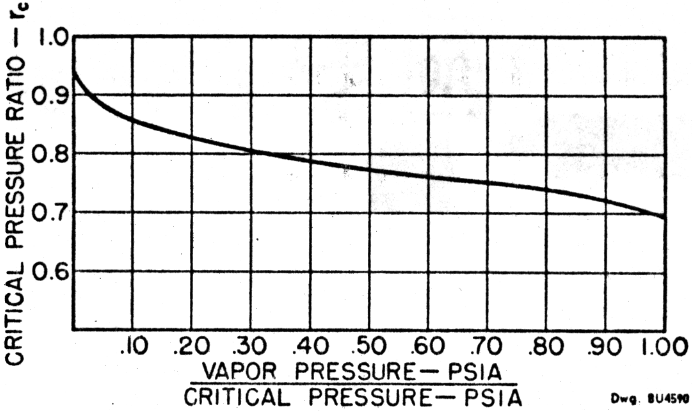

***Figure 1.2.** Determine the vapor pressure/critical pressure ratio by dividing the liquid vapor pressure at the valve inlet by the critical pressure of the liquid. Enter on the abscissa at the ratio just calculated and proceed vertically to intersect the curve. Move horizontally to the left and read r_c on the ordinate (Reference 4).*

1. No viscosity correction necessary if viscosity $\leqq$ 20 centistokes.
2. For sizing a flashing control valve add the C_v's of the liquid and the vapor.

Gas and Steam Flow

The gas and steam sizing formulas are

Gas

$$C_g = \frac{Q}{\sqrt{\frac{520}{GT}}\, P_1 \sin\left[\frac{3417}{C_1}\sqrt{\frac{\Delta P}{P_1}}\right] \text{deg.}} \qquad (1.11)$$

Table 1.5 Critical Pressure of Various Fluids, Psia*

Ammonia	1636	Isobutane	529.2
Argon	705.6	Isobutylene	580
Butane	550.4	Methane	673.3
Carbon Dioxide	1071.6	Nitrogen	492.4
Carbon Monoxide	507.5	Nitrous Oxide	1047.6
Chlorine	1118.7	Oxygen	736.5
Dowtherm A	465	Phosgene	823.2
Ethane	708	Propane	617.4
Ethylene	735	Propylene	670.3
Fluorine	808.5	Refrigerant 11	635
Helium	33.2	Refrigerant 12	596.9
Hydrogen	188.2	Refrigerant 22	716
Hydrogen Chloride	1198	Water	3206.2

*For values not listed, consult an appropriate reference book.

Steam (under 1000 psig)

$$C_s = \frac{Q_s\,(1 + 0.00065\,T_{sh})}{P_1 \sin\left[\frac{3417}{C_1}\sqrt{\frac{\Delta P}{P_1}}\right] \text{deg.}} \tag{1.12}$$

Steam and Vapors (all vapors, including steam under any pressure conditions)

$$C_g = \frac{Q_s}{1.06\sqrt{d_1 P_1}\,\sin\left[\frac{3417}{C_1}\sqrt{\frac{\Delta P}{P_1}}\right] \text{deg.}} \tag{1.13}$$

When the bracketed quantity in the equations equals or exceeds 90 degrees, critical flow is indicated. The quantity must be limited to 90 degrees. This then becomes unity since sin 90° = 1.

Explanation of terms:

$C_1 = C_g/C_v$ (some sizing methods use C_f or Y in place of C_1)
C_g = Gas sizing coefficient

Table 1.6. Relative Flow Capacities of Control Valves (Reference 5)

Valve Type	C_d*	C_dF_p†	C_dF_L**
Double-seat globe	12	11	11
Single-seat top-guided globe	11.5	10.8	10
Single-seat split body	12	11.3	10
Sliding gate	6-12	6-11	na
Single-seat top-entry cage	13.5	12.5	11.5
Eccentric rotating plug (Camflex)	14	13	12
60° open butterfly	18	15.5	12
Single-seat Y valve (300 & 600 lb)	19	16.5	14
Saunders type (unlined)	20	17	na
Saunders type (lined)	15	13.5	na
Throttling (characterized) ball	25	20	15
Single-seat streamlined angle (flow-to-close)	26	20	13
90° open butterfly (average)	32	21.5	18

Note: This table may serve as a rough guide only since actual flow capacities differ between manufacturer's products and individual valve sizes. (Source: ISA "Handbook of Control Valves" Page 17)

*Valve flow coefficient $C_v = C_d \times d^2$ (d = valve dia., in.)

†C_v/d^2 of valve when installed between pipe reducers (pipe dia. 2 × valve dia.)

**C_v/d^2 of valve when undergoing critical (choked) flow conditions.

C_s = Steam sizing coefficient
C_v = Liquid sizing coefficient
d_1 = Density of steam or vapor at inlet, lbs/ft^3
G = Gas specific gravity mol. wt./29
P_1 = Valve inlet pressure, psia
ΔP = Pressure drop across valve, psi
Q = Gas flow rate, SCFH
Q_s = Steam or vapor flow rate, lb/hr
T = Absolute temperature of gas at inlet, °R
T_{sh} = Degrees of superheat, °F

Table 1.7. Average Valve-Recovery Coefficients, K_m and C_l* (Reference 5)

Type of Valve	K_m	C_l
Cage-trim globes:		
Unbalanced	0.8	33
Balanced	0.70	33
Butterfly:		
Fishtail	0.43	16
Conventional	0.55	24.7
Ball:		
Vee-ball, modified-ball, etc.	0.40	22
Full-area ball	0.30	
Conventional globe:		
Single and double port (full port)	0.75	35
Single and double port (reduced port)	0.65	35
Three way	0.75	
Angle:		
Flow tends to open (standard body)	0.85	
Flow tends to close (standard body)	0.50	
Flow tends to close (venturi outlet)	0.20	
Camflex:		
Flow tends to close	0.72	24.9
Flow tends to open	0.46	31.1
Split body	0.80	35

***For use only if not available from manufacturer.**

The control valve coefficients in Table 1.8 are for full open conditions. The control valve must be designed to operate at partial open conditions for good control. Figure 1.3 shows partial open performance for a number of trim types.

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Table 1.8. Correlations of Control Valve Coefficients (Reference 5)

$C_l = 36.59\, C_f$

$C_l = 36.59\sqrt{K_m}$

$C_g = C_l\, C_v$

$K_m = C_f^2 = F_L^2$

$C_s = 1.83\, C_f\, C_v$

$C_v = 19.99\, C_s/C_l$

$C_g = 19.99\, C_s$

Values of K_m calculated from C_f agree within 10% of published data of K_m.

Values of C_l calculated from K_m are within 21% of published data of C_l.

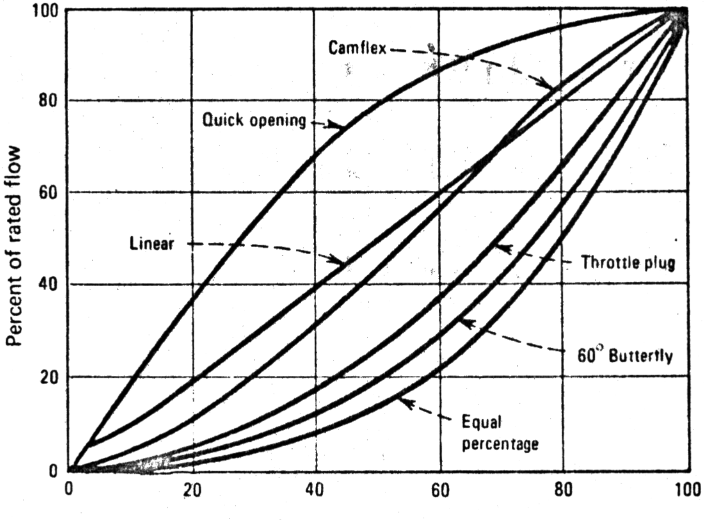

Figure 1.3. These are characteristic curves of common valves (Reference 5).

General Control Valve Rules of Thumb

1. Design tolerance. Many use the greater of the following:

   $Q_{sizing} = 1.3\ Q_{normal}$
   $Q_{sizing} = 1.1\ Q_{maximum}$

2. Type of trim. Use equal percentage whenever there is a large design uncertainty or wide rangeability is desired. Use linear for small uncertainty cases.
   Limit max/min flow to about 10 for equal percentage trim and 5 for linear. Equal percentage trim usually requires one larger nominal body size than linear.
3. For good control where possible, make the control valve take 50%-60% of the system flowing head loss.
4. For saturated steam keep control valve outlet velocity below 0.25 mach.
5. Keep valve inlet velocity below 300 ft/sec for 2″ and smaller, and 200 ft/sec for larger sizes.

Safety Relief Valve Design

The ASME code provides the basic requirements for overpressure protection. Section I, *Power Boilers,* covers fired and unfired steam boilers. All other vessels including exchanger shells and similar pressure containing equipment fall under Section VIII, *Pressure Vessels.* API RP 520 and lesser API documents supplement the ASME code. These codes specify allowable accumulation, which is the difference between relieving pressure at which the valve reaches full rated flow and set pressure at which the valve starts to open. Accumulation is expressed as percentage of set pressure in Table 1.9.

Full liquid containers require protection from thermal expansion. Such relief valves are generally quite small. Two examples are

1. Cooling water that can be blocked in with hot fluid still flowing on the other side of an exchanger.
2. Long lines to tank farms that can lie stagnant exposed to the sun.

Table 1.9. Accumulation Expressed as Percentage of Set Pressure

	ASME Section I Power Boilers	ASME Section VIII Pressure Vessels	Typical Design for Compressors Pumps and Piping
LIQUIDS			
thermal expansion	..	10	25
fire	..	20	20
STEAM			
over-pressure	3	10	10
fire	..	20	20
GAS OR VAPOR			
over-pressure	..	10	10
fire	..	20	20

Sizing

Use manufacturer's sizing charts and data where available. In lieu of manufacturer's data use the formula

$$u = 0.4\sqrt{2g\Delta h} \tag{1.14}$$

where

Δh = Head loss in feet of flowing fluid
u = Velocity in ft/sec
g = 32.2 ft/sec^2

This will give a conservative relief valve area. For compressible fluids use Δh corresponding to ½ P_1 if head difference is greater than that corresponding to ½ P_1 (since sonic velocity occurs). If head difference is below that corresponding to ½ P_1 use actual Δh.

For vessels filled with only gas or vapor and exposed to fire use

$$A = \frac{0.042\,A_s}{\sqrt{P_1}} \tag{1.15}$$

(API RP 520, Reference 8)

Table 1.10. Standard Orifice Sizes for Flanged Steel Safety Relief Valves

Orifice Designation	Effective Area, in.^{2}	Orifice Designation	Effective Area, in.^{2}
D	0.110	L	2.853
E	0.196	M	3.60
F	0.307	N	4.34
G	0.503	P	6.38
H	0.785	Q	11.05
J	1.287	R	16.0
K	1.838	T	26.0

A = Calculated nozzle area, in.^{2}

P_1 = Set pressure (psig) × (1 + fraction accumulation) + atmospheric pressure, psia. For example if accumulation = 10%, then (1 + fraction accumulation) = 1.10

A_s = Exposed surface of vessel, ft^2

This will also give conservative results. For heat input from fire to liquid containing vessels see "Determination of Rates of Discharge" in this chapter.

The set pressure of a conventional valve is affected by back pressure. The spring setting can be adjusted to compensate for *constant* back pressure. For a variable back pressure of greater than 10% of the set pressure, it is customary to go to the balanced bellows type which can generally tolerate variable back pressure of up to 40% of set pressure. Table 1.10 gives standard orifice sizes.

Determination of Rates of Discharge

The more common causes of overpressure are

1. External fire
2. Heat Exchanger Tube Failure
3. Liquid Expansion
4. Cooling Water Failure
5. Electricity Failure
6. Blocked Outlet

7. Failure of Automatic Controls
8. Loss of Reflux
9. Chemical Reaction (this heat can sometimes exceed the heat of an external fire)

Plants, situations and causes of overpressure tend to be dissimilar enough to discourage preparation of generalized calculation procedures for the rate of discharge. In lieu of a set procedure most of these problems can be solved satisfactorily by conservative simplification and analysis. It should be noted also that, by general assumption, two unrelated emergency conditions will not occur simultaneously.

The first three causes of overpressure on our list are more amenable to generalization than the others and will be discussed.

Fire

The heat input from fire is discussed in API RP 520 (Reference 8). One form of their equation for liquid containing vessels is

$$Q = 21{,}000\, F A_w^{0.82} \tag{1.16}$$

where

Q = Heat absorption, Btu/hr
A_w = Total wetted surface, ft^2
F = Environment factor

The environmental factors represented by F are

Bare vessel = 1.0
Insulated = 0.3/insulation thickness, in.
Underground storage = 0.0
Earth covered above grade = 0.03

The height above grade for calculating wetted surface should be

1. For vertical vessels—at least 25 feet above grade or other level at which a fire could be sustained.

2. For horizontal vessels—at least equal to the maximum diameter.
3. For spheres or spheroids—whichever is greater, the equator or 25 feet.

Heat Exchanger Tube Failure

1. Use the fluid entering from twice the cross section of one tube as stated in API RP 520 (Reference 8) (one tube cut in half exposes two cross sections at the cut).
2. Use the old standby, $\Delta h = u^2/2g$, to calculate leakage. Since this acts similar to an orifice, we need a coefficient; use 0.7. So,

$$u = 0.7\sqrt{2g\Delta h} \tag{1.17}$$

If the downstream head is less than ½ the upstream head, use ½ the upstream head as Δh. Otherwise use the actual Δh.

Liquid Expansion

Equation 1.18 can be used for sizing relief valves for liquid expansion.

$$Q = \frac{BH}{500\ GC} \tag{1.18}$$

(API RP 520, Reference 8)

where

Q = Required capacity, gpm
H = Heat input, Btu/hr
B = Coefficient of volumetric expansion per °F:
= 0.0001 for water
= 0.0010 for light hydrocarbons
= 0.0008 for gasoline
= 0.0006 for distillates
= 0.0004 for residual fuel oil

G = Specific gravity
C = Specific heat, Btu/lb °F

Rules of Thumb for Safety Relief Valves

1. Check metallurgy for light hydrocarbons flashing during relief. Very low temperatures can be produced.
2. Always check for reaction force from the tailpipe.
3. Hand jacks are a big help on large relief valves for several reasons. One is to give the operator a chance to reseat a leaking relief valve.
4. Flat seated valves have an advantage over bevel seated valves if the plant forces have to reface the surfaces (usually happens at midnight).
5. The maximum pressure from an explosion of a hydrocarbon and air is 7 × initial pressure, unless it occurs in a long pipe where a standing wave can be set up. It may be cheaper to design some small vessels to withstand an explosion than to provide a safety relief system. It is typical to specify ¼″ as minimum plate thickness (for carbon steel only).

Metering

Orifice

$$(U_o^2 - U_p^2)^{1/2} = C_o\,(2g\Delta h)^{1/2} \qquad \textbf{(1.19)}$$

Permanent head loss % of Δh

D_o/D_p	Permanent Loss
0.2	95
0.4	82
0.6	63
0.8	40

One designer uses permanent loss = $\Delta h\,(1\text{-}C_o)$

where

U_o = Velocity through orifice, ft/sec
U_p = Velocity through pipe, ft/sec
$2g$ = 64.4 ft/sec^2
Δh = Orifice pressure drop, ft of fluid
D = Diameter
C_o = Coefficient. (Use 0.60 for typical application where D_o/D_p is between 0.2 and 0.8 and Re at vena contracta is above 15,000.)

Venturi

Same equation as for orifice:

$C_v = 0.98$

Permanent head loss approximately 3-4% Δh. (1.20)

Rectangular Weir

$$F_v = 3.33\,(L - 0.2H)\,H^{3/2} \qquad (1.21)$$

where

F_v = Flow in ft^3/sec
L = Width of weir, ft
H = Height of liquid over weir, ft

Pitot Tube

$$\Delta h = u^2/2g$$

References

1. *Cameron Hydraulic Data*, Ingersoll-Rand Company, Cameron Pump Division, 13th ed. 1965.
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9. *NGPSA Engineering Data Book,* Natural Gas Processors Suppliers Association, 9th ed., 1972.
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2
PUMPS AND MOTORS

This chapter will not contain detailed piping procedures for pumps. There are many good sources for this type information such as References 3, 4, 5, 6, 7, and 8. Suction piping, especially for liquids near their boiling point, requires very accurate design, which is beyond the scope of this small handbook. Instead, helpful quick methods will be given to estimate pump and motor size.

Affinity (or Fan) Laws

Dynamic type pumps obey the affinity laws:

1. Capacity varies directly with impeller diameter and speed.
2. Head varies as the square of impeller diameter and speed.
3. Horsepower varies as the cube of impeller diameter and speed.

Pump Horsepower

The handiest pump horsepower formula for a process engineer is

$$HP = \frac{GPM\,(\Delta P)}{1715\,(Eff.)} \tag{2.1}$$

where

HP = Pump horsepower
GPM = Gallons per minute
ΔP = Delivered pressure (discharge minus suction), psi
$Eff.$ = Pump efficiency

Pump Efficiency

An equation was developed by the author from the pump efficiency curves in the eighth edition of Reference 1, provided by the M. W. Kellogg Co. The curves were found to check vendor data well. The equation is admittedly bulky appearing, but is easier to use than it appears.

$$\begin{aligned} Eff. = {} & 80 - 0.2855F + 3.78 \times 10^{-4}\,FG - 2.38 \times 10^{-7}\,FG^2 \\ & + 5.39 \times 10^{-4}\,F^2 - 6.39 \times 10^{-7}\,F^2G + 4 \times 10^{-10} \\ & F^2G^2 \end{aligned} \tag{2.2}$$

where

$Eff.$ = Pump percentage efficiency
F = Developed head, ft
G = Flow, GPM

Ranges of applicability:

F = 50-300 ft
G = 100-1000 GPM

Equation 2.2 gives results within about 7% of the aforementioned pump curves. This means within 7% of the curve valve, not 7% absolute, i.e., if the curve valve is 50%, the equation will be within the range 50 ± 3.5%.

For flows in the range 25-99 GPM a rough efficiency can be obtained by using Equation 2.2 for 100 GPM and then subtracting 0.35%/GPM times the difference between 100 GPM and the

Table 2.1. Motor Efficiencies

Standard Motor Horsepower Ratings	Efficiency	Standard Motor Horsepower Ratings	Efficiency
1	80	250	90
2	82	500	93
3	84	600	93
5	85	700	93
7½	85	800	94
10	85	900	94
15	86	1,000	94
20	87	1,250	94
25	88	1,500	94
30	89	1,750	94
40	89	2,000	94
50	89	2,250	94
75	90	2,500	95
100	90	3,000	95
125	90	3,500	95
150	90	4,000	95
200	90	4,500	95
		5,000	95

Table 2.2. Starter Size Versus Horsepower

	Starter Size	00	0	1	2	3	4	5
Three Phase	208-220 volt	1½	3	7½	15	30	50	100
	440-550 volt	2	5	10	25	50	100	200
Single Phase	110 volt	⅓	1	2	3	..	...	...
	220 volt	1	2	3	5	..	...	...

low flow GPM. For flows at the bottom of the range (25-30 GPM), this will give results within about 15% for the middle of the head range and 25% at the extremes. This is adequate for ballpark estimates at these low flows. The horsepower at the 25-30 GPM level is generally below 10.

Motor Efficiency

Table 2.1 will yield ballpark motor efficiencies.

Table 2.1 applies to 3-phase induction motors only. For single phase multiply by about 0.8. Single phase motors are not generally recommended above 10 HP.

Table 2.2 contains handy data for determining starter size.

References

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3

COMPRESSORS AND STEAM TURBINES

Compressors

This chapter discusses centrifugal compressors and steam turbines. Centrifugal compressors are best discussed by going through a calculation procedure and discussing each part.

First, the required head is calculated. Either the polytropic or adiabatic head can be used to calculate horsepower so long as the polytropic or adiabatic efficiency is used with the companion head.

Polytropic Head

$$H_{poly} = \frac{ZRT_1}{(N-1)/N}\left[\left(\frac{P_2}{P_1}\right)^{(N-1)/N} - 1\right] \tag{3.1}$$

Adiabatic Head

$$H_{AD} = \frac{ZRT_1}{(K-1)/K}\left[\left(\frac{P_2}{P_1}\right)^{(K-1)/K} - 1\right] \tag{3.2}$$

where

Z = Average compressibility factor; using 1.0 will yield conservative results
R = 1,544/mol. wt.
T_1 = Suction temperature, °R
P_1, P_2 = Suction, discharge pressures, psia
K = Adiabatic exponent, C_p/C_v ⟶
PARA AIRE
$k = c_p/c_v = 1.4$
N = Polytropic exponent, $\frac{N-1}{N} = \frac{K-1}{KE_p}$
E_p = Polytropic efficiency, use 75% for preliminary work
E_A = Adiabatic efficiency

The polytropic and adiabatic efficiencies are related as follows:

$$E_A = \left[\frac{\left(\frac{P_2}{P_1}\right)^{(K-1)/K} - 1}{\left(\frac{P_2}{P_1}\right)^{(N-1)/N} - 1}\right] = \left[\frac{\left(\frac{P_2}{P_1}\right)^{(K-1)/K} - 1}{\left(\frac{P_2}{P_1}\right)^{(K-1)/KE_p} - 1}\right] \qquad (3.3)$$

The gas horsepower is calculated using the companion head and efficiency.

From Polytropic Head

$$HP = \frac{W\ H_{poly}}{E_p\ 33{,}000} \qquad (3.4)$$

From Adiabatic Head

$$HP = \frac{W\ H_{AD}}{E_A\ 33{,}000} \qquad (3.5)$$

where

HP = Gas horsepower
W = Flow, lb/min

To the gas horsepower is added bearing and oil seal losses. Use 50 horsepower in lieu of manufacturer's data for large machines.

The discharge temperature is calculated as follows:

$$t_2 = \frac{H_{poly}}{ZR\left(\frac{K}{K-1}\right)E_p} + t_1 \tag{3.6}$$

Often the temperature of the gas must be limited. Sometimes temperature is limited to protect against polymerization as in olefin or butadiene plants. At temperatures greater than 450-500°F, the approximate mechanical limit, problems of sealing and casing growth start to occur. High temperature requires a special and more costly machine. Most multistage applications are designed to stay below 250-300°F.

Intercooling can be used to hold desired temperatures for high overall compression ratio applications. This can be done between stages in a single compressor frame or between series frames. Sometimes economics rather than a temperature limit dictate intercooling.

Sometimes for high compression ratio applications the job cannot be done in a single compressor frame. Usually a frame will not contain more than about eight (8) stages (wheels). There is a maximum head that one stage can handle. This depends upon the gas properties and inlet temperature. Usually this will run 7,000 to 11,000 feet for a single stage. In lieu of manufacturer's data use eight (8) maximum stages per frame. Then subtract one stage for every side nozzle such as to and from an intercooler, side gas injection, etc. For many applications the compression ratio across a frame will run 2.5-4.0.

One word of caution—for gas mixtures or air be sure to include the contained water. It can be a healthy percentage for a Gulf Coast air compressor.

Steam Turbines

The process engineer needs to determine the steam requirements for his turbines knowing the horsepower. The theoretical steam rate can be determined from Reference 5 or Mollier charts following a constant entropy path. The theoretical steam rate in Reference 5 is given as lb/hr/kw which is easily converted to lb/hr/hp. One word of caution in using Reference 5–steam pressures are given in *PSIG.* Sea level is the basis. For low steam pressures at high altitudes appropriate corrections must be made. See atmospheric pressure versus altitude in Chapter 7.

The theoretical steam rate (sometimes referred to as the "water rate") must then be divided by an adiabatic efficiency to obtain the actual steam rate. In lieu of manufacturer's data use for preliminary work:

Horsepower	Efficiency, %
500-1,000	50
1,000-1,500	55
1,500-2,000	60
2,000-3,000	65
3,000-5,000	70
5,000 UP	75

The smaller turbines can vary widely in efficiency depending greatly on speed, horsepower, and pressure conditions. Very rough efficiencies to use for initial planning below 500 horsepower at 3,500 rpm are

Horsepower	Efficiency, %
1-10	15
10-50	20
50-300	25
300-350	30
350-500	40

Some designers limit the speed of the cheaper small steam turbines to 3600 rpm.

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1. *Elliott Multistage Compressors,* Bulletin P-25, Copyright© 1973, Carrier Corporation.

2. *NGPSA Engineering Data Book,* Natural Gas Processors Suppliers Association, 9th ed., 1972.
3. Perry and Chilton, *Chemical Engineers' Handbook,* 5th ed., McGraw-Hill Inc., 1973.
4. Ludwig, *Applied Process Design for Chemical and Petro-Chemical Plants,* Vol. 3, Copyright© 1965, Gulf Publishing Company, Houston, Texas.
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4
VACUUM SYSTEMS

This chapter primarily deals with vacuum jets (eductors) and vacuum systems associated with vacuum fractionators. The design of vacuum overhead condensers is discussed in Chapter 7.

Vacuum Jets

Jet design is normally handled by the vendor. However, the process engineer must specify the system into which the jets are incorporated. He must also supply the vendor with operating conditions which include

1. Flows of all components to be purged from the system (often air plus water vapor).
2. Temperature and pressure entering the jets and pressure leaving if not atmospheric.
3. Temperature and pressure of steam available to drive the jets.
4. Temperature and quantity of cooling water available for the intercondensers. Also cooling water allowable pressure drop for the intercondensers.

In addition, the process engineer must be aware of good design practices for vacuum jets.

The vendor will convert the component flow data into an "air equivalent." Since jets are rated on air handling ability, he can then build up a system from his standard hardware. The vendor should provide air equivalent capability data with the equipment he supplies. Determination of air equivalent can be done with Equation 4.1.

$$ER = F\sqrt{0.0345\,(MW)} \tag{4.1}$$

where

ER = Entrainment ratio (or air equivalent). It is the ratio of the weight of gas handled to the weight of air which would be handled by the same ejector operating under the same conditions.
MW = Gas mol. wt.
$F = 1.00$, for MW 1 – 30
$F = 1.076 - 0.0026\,(MW)$, for MW 31 – 140

Equation 4.1 will give results within 2% of the Reference 4 entrainment ratio curve.

The effect of temperature is shown by Equations 4.2 and 4.3.

$$ERTA = 1.017 - 0.00024T \tag{4.2}$$

$$ERTS = 1.023 - 0.00033T \tag{4.3}$$

where

$ERTA$ = The ratio of the weight of air at 70°F to the weight of air at a higher temperature that would be handled by the same ejector operating under the same conditions.
$ERTS$ = Same as above for steam
T = Gas temperature, °F

This information is based on Reference 4.

The vendor should also supply steam consumption data. However, for initial planning the process engineer needs to have an estimate. Use Equation 3.4 or 3.5 to calculate the horsepower required to compress non-condensing components from the jet inlet pressure and temperature to the outlet pressure. For process water vapor handled by the jets with intercondensing, calculate horsepower for the first stage only. After the first stage the condenser will bring the system to the same equilibrium as would have occurred without the process water vapor. Use an adiabatic efficiency of 7% for cases with jet intercondensers and 4% for non-condensing cases. Estimate the steam consumption to be the theoretical amount which can deliver the previously calculated total horsepower using the jet system steam inlet and outlet conditions. These ballpark results can be used until vendor data arrive. This procedure will give conservative results for cases with high water vapor compared to the Ludwig (2) curves for steam consumption.

Following are some general rules of thumb for jets:

1. To determine number of stages required assume 7:1 compression ratio maximum per stage.
2. The supply steam conditions should not be allowed to vary greatly. Pressure below design can lower capacity. Pressure above design usually doesn't increase capacity and can even lower capacity.
3. Use Stellite or other hard surface material in the jet nozzle. For example 316 s/s is insufficient.
4. Always provide a suitable knockout pot ahead of the jets. Water droplets can quickly damage a jet. The steam should enter the pot tangentially. Any condensate leaves through a steam trap at the bottom. It is a good idea to provide a donut baffle near the top to knock back any water creeping up the vessel walls.
5. The jet barometric legs should go in a straight line to the seal tank. A 60°-90° slope from horizontal is best.

Design Recommendations

First one must estimate air or other gas leakage into the system. Of course every effort is made to keep it as tight as possible. The author is aware of possible leak points being sealed with polystyrene which produces an excellent seal. When tests cannot be made, one must use rules of thumb. Many such rough estimating techniques exist.

A close friend with years of experience in designing and operating vacuum systems claims that a system can be made very tight when designed properly. He advocates purging instrument leads with nitrogen. The rate is 1 SCFM maximum per lead. He has designed and successfully operated a number of vacuum systems with provisions for the required instrument purge times a safety factor of about 4. This factor is a contingency for such things as sudden load changes which could bring in fractionator feed that has been more poorly stripped.

Other recommendations are

1. For good control design the pressure drop for the control valve between the fractionating system and the jet system for sonic velocity (approximately 2:1 pressure ratio). This means that the jets suction must be designed for half the absolute pressure of the evacuated system.
2. In even a large fractionator system, under the following conditions:

   a. Properly designed condenser (see Chapter 7)
   b. Vacuum tight
   c. Sonic velocity across the control valve
   d. Well stripped feed

   the control valve trim rarely exceeds ¾″ and usually runs ½″.
3. Set the pressure controller for low proportional band.
4. For applications having column temperature control above the feed point, put the measuring elements for the temperature and pressure controllers on the same tray. This will make for good composition control at varying column loads (varying column differential pressure). In vacuum systems a slight pressure change will produce large equilibrium temperature changes.
5. For big vapor lines and condensers (frequent in vacuum systems) always insulate the line, condenser, and top of

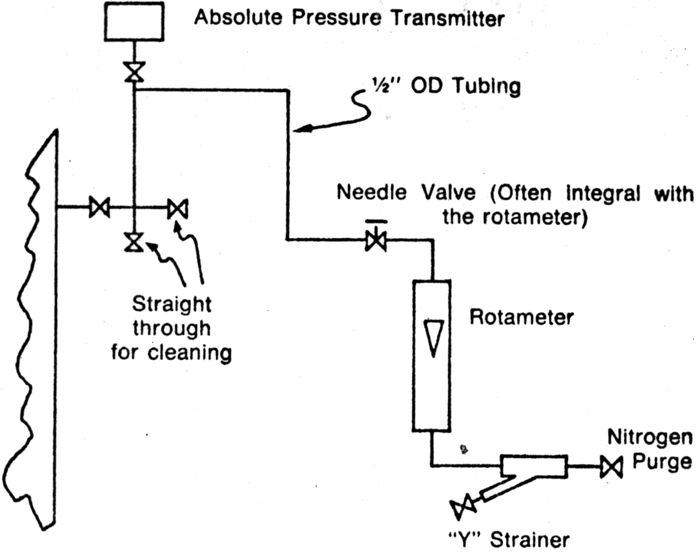

Figure 4.1. Shown here is a vacuum measurement installation.

column. Rain or sudden cold fronts will change column control otherwise. It is possible to have more surface in the overhead line than in the condenser.

6. Never use screwed fittings in any vacuum system regardless of size.
7. Installation of a pressure controller measurement tap is shown in Figure 4.1.
8. Avoid liquid traps in vacuum system piping by never going up after having gone horizontal.
9. Put the vacuum system control valves at the highest point of a horizontal run and the control valve bypass in the same horizontal plane. This is in compliance with item 8.

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5

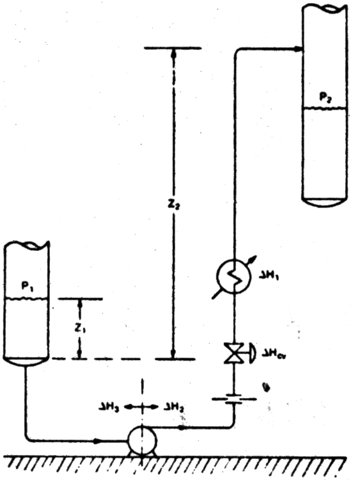
A CONSISTENT METHOD FOR FLOW COMPONENT SIZING

Figure 5.1 illustrates a method that will produce a system in which the parts fit together to accomplish the common goal of good control. Control valve share of total system flowing pressure drop will be 60% at normal flow. The system will still achieve maximum flow as long as the control valve trim selected can pass maximum flow at operating head loss (line 23 of Figure 5.1). The procedure described in Figure 5.1 is intended as a stand-alone device for guiding the calculations, and worksheets can be prepared from it.

	LINE NO.					
1	Material					
2	Phase L, G, V, S					
3	Max. Flow 1000 #/hr					
4	Norm. Flow 1000 #/hr					
5	Min. Flow 1000 #/hr					
6	Specific Gravity (base)					
7	Specific Gravity (flow)					
8	Mol. Wt.					
9	Temp. Flowing °F					
10	Temp. Base °F					
11	GPM Actual Fluid	Based on max flow.				
12	Press. Nominal (psia)					
13	P_2 Destination (psia) }					
14	P_1 Source (psia) }	Press is measured at liquid level.				
15	ΔH_p = [⑬-⑭] [N]/⑦ ft	N = 2.307	ft. H_2O/psi			
16	Z_2 Disch. ft }	Height of liquid level using a common base line. Use				
17	Z_1 Suct. ft }	highest liquid height whether in pipe or vessel.				
18	ΔH_z = ⑯-⑰ ft					
19	ΔH_1 Equip. ft	ΔH_1 = PSI x 2.307/⑦			(Based on ⑪)	
20	ΔH_2 Line (Disch) ft }					
21	ΔH_3 Line (suct) ft }	Based on ⑪				
22	ΔH_F ⑲ + ⑳ + ㉑ ft					
23	ΔH_{cv} = (x) x ㉒ ft	Value of x = 1.5 (④⑤)^2				
24	ΔH_{total} = ㉒ + ㉓ ft	Defined as total flowing loss.				
25	Pump TDH = ⑮+⑱+㉔ + meter permanent loss, ft					
26	Pump HP (Hydraulic)	HP = ③ x ㉕ x 5.05 x 10^{-4}				
27	Actual Pump HP = ㉖ /efficiency					
28	Suct Line IPS/SCH.					
29	Disch Line IPS/SCH.					
30	Suct Line Length, ft					
31	Disch Line Length, ft					
32	Meter Range, "H_2O					
33	d/D Orifice					
34	Full Scale Rdg 1000 #/hr					
	ΔH_p = Static pressure difference in ft. of fluid at flowing conditions.					
	ΔH_z = Liquid head difference from source to destination in ft. of fluid at flowing conditions.					
	ΔH_1 = Flowing loss in exchangers, etc. in ft. of fluid at flowing conditions.					

NOTE: All ft. are reported as ft. of flowing fluid at S.G. corresponding to ⑨.

Figure 5.1. This diagram illustrates a consistent method for flow component sizing.



ΔH_2 = Flowing loss in discharge line in ft. of fluid at flowing conditions.

ΔH_3 = Flowing loss in suction line in ft. of fluid at flowing conditions.

ΔH_{CV} = Pressure drop across control valve equal to 60% of total flowing loss at normal flow.

ΔH_F = Total flowing loss excluding control valve and meter loss in ft. of fluid at flowing conditions.

Figure 5.1. Continued.

6
PNEUMATIC CONVEYING

Types of Systems

This chapter considers conveying solids with air. Materials conveyed with air vary greatly in properties, often requiring special considerations. This chapter will, therefore, give only means for preliminary rough sizing and selection of equipment. Often the entire pneumatic conveying system for a plant is bought as a package from a vendor specializing in such equipment. Even then it is helpful for the process engineer to be familiar with rough equipment sizing methods. Such knowledge can help him plan such things as plot area and utility draws in the initial design phase. The knowledge may also be helpful in later startup or troubleshooting. The methods described are primarily those of References 1, 2, and 3.

The following are the general types of systems and their uses:

1. Negative (vacuum) system; normally used when conveying from several pickup points to one discharge point.
2. Positive-pressure system; normally used when conveying from one pickup point to several discharge points.
3. Pressure-negative (push-pull) combination system; normally used when conveying from several pickup points to several discharge points.

4. Venturi, product systems, or blow tanks will not be discussed here.

The final choice is determined by economics or some special material demand.

Negative System

The negative system usually sucks on a cyclone and/or filter/receiver mounted above the receiving storage hopper or bin. Solids are usually sent down to the hopper with a rotary air lock feeder. Air is sucked into the transfer pipe or duct at the pickup end of the system. A variety of feeders can be used to introduce solids into the flowing air stream such as rotary air lock feeders, pan-type manifolds under rail car hoppers, paddle-type rail car unloaders, screw conveyors, etc.

Positive Pressure System

In the positive pressure system air is blown into the pickup duct often up to a cyclone with atmospheric vent. Usually solids are introduced to the conveying air stream with a rotary air lock feeder. In sending the solids down to the storage bin from the cyclone a simple spout connection can be used instead of the rotary airlock feeder required in the negative pressure system. The positive system doesn't need a vacuum vessel at each receiving location. Also in the positive pressure system conveying to a number of hoppers, a simple bag-type cloth can serve as the filter. So for conveying from one pickup location to several receiving locations, the pressure system is often cheaper than the negative system.

Pressure-Negative System

The pressure-negative system is ideal for unloading rail cars and also for conveying light dusty materials because the suction at the inlet aids the product to enter the conveying line. Positive systems are poorer than negative for handling such materials since feeding into the pressure line can be difficult, and can present a dust-collection problem at the pickup location because of blowback or leakage air through the rotary valve. Also the author has noted that the negative system works better if there are lumps at the pickup end. The positive pressure system tends to pack at the lump while the negative system will often keep solids moving around the lump and gradually wear it away.

For the pressure-negative system a single blower can be used for both the negative and positive sides of the system. However, the lack of flexibility with a single blower usually dictates the need for separate blowers for the negative and positive sides.

Differential Pressures

If the system differential pressure requirements are low sometimes a fan can be used. A fan is limited to a maximum of about 65 inches of water (just over 2 psi) in vacuum service or 77 inches of water in pressure service (just under 3 psi). To estimate fan horsepower use Equation 3.4 or 3.5. In lieu of manufacturer's data use an adiabatic efficiency of 50% for initial work.

For higher differential pressures (or lower differential pressures where it is preferred not to use a fan) the rotary positive-displacement blower is used. These are excellent for conveying systems since they provide

1. Reasonably constant volume at variable pressure discharge.
2. Vacuums of about 8 psi (some can go to 11 psi with water injection) and pressure differentials of 15 psi.
3. A low slippage with improved efficiency.

The higher pressure differentials allow longer and sometimes smaller lines than for fans. To estimate blower horsepower use Equation 3.4 or 3.5. In lieu of manufacturer's data use an adiabatic efficiency of 80% for initial work.

Table 6.1. Capacity Range

Duct Dia. in.	Flow, SCFM at 5,000 ft/min	Friction loss, in. H_2O/100 ft	Usual Capacity, thousands of lbs/hr	
			Negative	Positive
4	440	11.0	2–6	12–40
5	680	8.0	3–10	15–60
6	980	6.3	4–15	20–80
8	1,800	4.5	15–30	30–160

Even though 15 psi is possible from a blower, most positive-pressure systems are limited to 10-12 psi differential pressure because of limits of the rotary air lock feeder valves (deflections in shaft and bearings and increased blowback air).

Equipment Sizing

To rough out line sizes and pressure drop for fan or blower sizing use the following quickie method:

1. Arbitrarily assume air velocity of 5,000 ft/min (good for 90% of conveying situations).
2. Use Table 6.1.
3. Calculate pressure drop. This is in two parts:

   a. Material losses

   E_1 = Acceleration losses
   E_2 = Lifting energy
   E_3 = Horizontal losses
   E_4 = Bends and elbows

   b. Air losses

4. First do material losses in ft-lb/min

$E_1 = MU^2/2g = 108M$ @ 5,000 ft/min
$E_2 = M(H)$
$E_3 = M(L)(F)$
$E_4 = MU^2/gR\,(L)(F)(N) = 342\,(M)(F)(N)$ for 48" radius 90° ell. Assume this to be the case.

where

M = Solids conveyed, lb/min
U = Velocity, ft/min
$2g = 2.32 \times 10^5$ ft/min^2
H = Vertical lift, ft
L = Duct horizontal length, ft
R = 90 ell radius, ft

F = Coefficient of friction and tangent of solids "angle of slide" or "angle of repose." Use 0.8 in lieu of solids data for initial estimating

N = Number of 90° ells. For 45°, 30° etc., express as equivalent 90° ells by direct ratio (Example: A 30° ell is 0.33 of a 90° ell)

5. From Table 6.1 and solids rate, estimate duct size and flow in SCFM (standard cubic feet per minute).
6. Express material losses in inches of water:

$$\frac{ft\text{-}lb/min}{ft^3/min \times 5.2} = in.\ H_2O$$

7. Calculate air losses.

   a. Calculate equivalent length of straight pipe by adding to actual length of straight pipe an allowance for conveying type 90° ells of 1 ft of pipe/in. of diameter. (Example: 4″ 90° ell = 4 ft of pipe)
   b. Assume the following losses for other items in inches of water:

Duct entry loss	1.9
Y branch	0.3
Cyclone	3.0
Collector vessel	3.0
Filter	6.0

8. Add material and air losses.
9. Calculate fan or blower horsepower as explained earlier.
10. Be sure to use the atmospheric suction pressure at the site. Normal blower ratings are given at sea level.

References

1. Fisher, John, "Practical Pneumatic Conveyor Design," *Chemical Engineering*, June 2, 1958, Copyright© (1958) McGraw-Hill, Inc., used with permission.

2. Gerchow, Frank J., "How to Select A Pneumatic Conveying System," *Chemical Engineering,* February 17, 1975, Copyright© (1975) McGraw-Hill, Inc., used with permission.
3. Perkins, Don E. and Wood, Jim E., "Design and Select Pneumatic Conveying Systems," *Hydrocarbon Processing,* March, 1974.

7
HEAT TRANSFER

Tubeside Pressure Drop

This pressure drop is composed of several parts which are calculated as shown in Tables 7.1 and 7.2 (Reference 1).

Shellside Pressure Drop

Tube Patterns

With segmental baffles, where the shellside fluid flows across the tube bundle between baffles, the following tube patterns are usual:

1. Triangular—Joining the centers of 3 adjacent tubes forms an equilateral triangle. Any side of this triangle is the tube pitch *c*.
2. Square inline—Shellside fluid has straight lanes between tube layers, unlike triangular where alternate tube layers are offset. This pattern makes for easy cleaning since a lance can be run completely through the bundle without interference. This pattern has less pressure drop than triangular but shell requirements are larger and there is a

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Table 7.1. Calculation of Tubeside Pressure Drop in Shell and Tube Exchangers

Part	Pressure Drop in Number of Velocity Heads	Equation
Entering plus exiting the exchanger	1.6	$\Delta h = 1.6 \frac{U_p^2}{2g}$ (This term is small and often neglected)
Entering plus exiting the tubes	1.5	$\Delta h = 1.5 \frac{U_T^2}{2g} N$
End losses in tubeside bonnets and channels	1.0	$\Delta h = 1.0 \frac{U_T^2}{2g} N$
Straight tube loss	See Chapter 1 "Piping Pressure Drop"	

Δh = Head loss in feet of flowing fluid
U_p = Velocity in the pipe leading to and from the exchanger, ft/sec
U_T = Velocity in the tubes
N = Number of tube passes

Table 7.2. Calculation of Tubeside Pressure Drop in Air-Cooled Exchangers

Part	Pressure Drop in Approximate Number of Velocity Heads	Equation
All losses except for straight tube	2.9 (calculated from curve on p. 9-11 of Reference 3)	$\Delta h = 2.9 \frac{U_T^2}{2g} N$
Straight tube loss	See Chapter 1 "Piping Pressure Drop"	

Table 7.3. Tube Pattern Relationships

	Triangular	Square Inline	Square Staggered
a	$a = c$	$a = c$	$a = 1.414c$
b	$b = 0.866c$	$b = c$	$b = 0.707c$

lower heat transfer coefficient for a given velocity at many velocity levels. Joining the centers of 4 adjacent tubes forms a square. Any side of this square is the tube pitch *c*.

3. Square staggered, often referred to as square rotated—Rotating the square inline pitch 45° no longer gives the shellside fluid clear lanes through the bundle. Tube pitch *c* is defined as for square inline.

Two other terms need definition: transverse pitch *a* and longitudinal pitch *b*. For a drawing of these dimensions see Reference 1. For our purposes appropriate lengths are shown in Table 7.3.

Turbulent Flow

For turbulent flow across tube banks a modified Fanning equation and modified Reynold's number will now be given. The following method is based on Reference 1.

$$\Delta P_f = \frac{4f'' N_R N_{sp} \rho U^2_{max}}{2g} \tag{7.1}$$

$$R_e' = \frac{D_o U_{max} \rho}{\mu} \tag{7.2}$$

where

ΔP_f = Friction loss in lb/ft^2
f'' = Modified friction factor
N_R = Rows of tubes per shell pass (N_R is always equal to the number of minimum clearances through which the fluid flows in series. For square staggered pitch the maximum velocity, U_{max}, which is required for evaluating R_e' may occur in the transverse clearances *a* or the diagonal clearances *c*. In the latter case N_R is one less than the number of tube rows.
N_{sp} = Number of shell passes
ρ = Density, lb/ft^3

Table 7.4. Determination of f" for 5 Tube Rows or More

C/D_o Both in Same Length Units	Triangular			
$R_e' \times 10^{-3} \rightarrow$	2	8	20	40
1.25 (min)	0.210	0.155	0.130	0.107
1.50	.145	.112	.090	.074
2.00	.118	.096	.081	.066
3.00	.089	.076	.063	.052

	Square Inline			
$R_e' \times 10^{-3} \rightarrow$	2	8	20	40
1.25 (min)	0.139	0.135	0.116	0.099
1.50	.081	.079	.080	.071
2.00	.056	.057	.055	.053
3.00	.052	.050	.045	.038

	Square Staggered			
$R_e' \times 10^{-3} \rightarrow$	2	8	20	40
1.25 (min)	0.130	0.106	0.088	0.063
1.50	.125	.103	.079	.061
2.00	.108	.090	.071	.058
3.00	....	....	....	....

Table 7.5. f" Correction Factor For Less Than 5 Tube Rows

Number of Rows	1	2	3	4
Correction Factor	1.30	1.30	1.15	1.07

U_{max} = Maximum linear velocity (through minimum cross-sectional area), ft/sec
g = 32.2 ft/sec^2
R_e' = Modified Reynold's number
D_o = Outside tube diameter, ft
μ = Viscosity, lb/ft sec; centipoises $\times$ 0.000672

The modified friction factor can be determined by using Tables 7.4 and 7.5.

Laminar Flow

Below $\frac{D_c U_{max} \rho}{\mu} = 40$ where D_c is the tube clearance in feet, the flow is laminar. For this region use

$$\Delta P_f = \frac{1.68 \mu U_{max} L}{D_e^2} \tag{7.3}$$

where

L = Length of flow path, ft
D_e = Equivalent diameter, ft; 4 times hydraulic radius

$$D_e = 4 \frac{(\text{cross-sectional flow area})}{(\text{wetted perimeter})} = D_o\left(\frac{4ab}{\pi D_o^2} - 1\right) \tag{7.4}$$

Pressure Drop for Baffles

Equations 7.1 and 7.3 determine the pressure drop across the tube bundle. For the additional drop for flow through the free area above, below, or around the segmental baffles use

$$\Delta P_f = \frac{W^2 N_B N_{sp}}{\rho S_B^2 g} \tag{7.5}$$

where

W = Flow in lb/sec
N_B = Number of baffles in series per shell pass
S_B = Cross sectional area for flow around segmental baffle, ft^2

Flow Parallel to Tubes

For flow parallel to tubes or in an annular space, e.g. a double-pipe heat exchanger use

Table 7.6. Determination of F_p

Air Face Mass Velocity lb/Hr/ft² Face Area	F_p
1,400	0.033
1,600	.0425
1,800	.0525
2,000	.0625
2,200	.075
2,400	.0875
2,600	.100
2,800	.115
3,000	.130
3,200	.150
3,400	.165
3,600	.185

$$\Delta P = \frac{2f \rho U^2 L}{g D_e} \tag{7.6}$$

where

ΔP = Pressure drop, lb/ft²
f = Friction factor (see Chapter 1)

Air-Cooled Exchangers—Air-Side Pressure Drop

This method will approximate required fan horsepower based on Reference 3.

$$\Delta P_a = \frac{F_p N}{D_R} \tag{7.7}$$

where

ΔP_a = Static pressure drop, inches of water
F_p = Static pressure drop factor from Reference 3, see Table 7.6
N = Number of tube rows
$D_R = \frac{\text{Actual density at average air temperature}}{\text{Air density at sea level and 70°F}}$

Use perfect gas law to calculate D_R

$$\text{Altitude (above sea level), ft} = 25{,}000\ ln\left(\frac{14.7}{P}\right)$$

where

P = Atmospheric pressure in psia

Once ΔP_a is obtained, the pressure that the fan has to provide is then calculated.

$$P_F = \Delta P_a + \left[\frac{ACFM\ Per\ Fan}{3{,}140\ D^2}\right]^2 D_R \tag{7.8}$$

where

P_F = Total pressure that fan has to provide, inches of water
$ACFM$ = Actual cubic feet per minute
D = Fan diameter, feet
$D_R = \dfrac{\text{Actual density at fan temperature}}{\text{Air density at sea level and 70°F}}$

$$HP = \frac{ACFM\ Per\ Fan\ (P_F)}{4{,}460} \tag{7.9}$$

where

HP = Approximate horsepower per fan

Determination of Shell and Tube Heat Exchanger Shell Diameter

For triangular pitch proceed as follows:

1. Draw the equilateral triangle connecting three adjacent tube centers. Any side of the triangle is the tube pitch (recall 1.25 D_o is minimum).

2. Triangle area is ½ *bh* where *b* is the base and *h* is the height.
3. This area contains ½ *tube.*
4. Calculate area occupied by all the tubes.
5. Calculate shell diameter to contain this area.
6. Add one tube diameter all the way around (two tube diameters added to the diameter calculated above).
7. The resulting is minimum shell diameter. There is no firm standard for shell diameter increments. Use 2-inch increments for initial planning.

For square pitch proceed similarly.

Determining Heat Exchanger Temperature Difference

Only countercurrent flow will be considered here. It is well known that the log mean temperature is the correct temperature difference to be used in the expression:

$$q = UA\Delta T_M \tag{7.10}$$

where

q = Heat duty in Btu/hr
U = Overall heat transfer coefficient in Btu/hr ft^2 °F
A = Tube surface area in ft^2
ΔT_M = Mean temperature difference in °F. For our case it is the log mean temperature difference.

$$\Delta T_M = \frac{GTD - LTD}{\ln(GTD/LTD)} \tag{7.11}$$

where

GTD = Greater temperature difference
LTD = Lesser temperature difference

When $GTD/LTD < 2$ the arithmetic mean is within about 2% of the log mean.

These refer to hot and cold fluid terminal temperatures, inlet of one fluid versus outlet of the other. For a cross exchanger with no phase change the ΔT_M gives exact results for true countercurrent flow. Most heat exchangers, however, deviate from true countercurrent so a correction factor, *F*, is needed.

These correction factors are given in various heat transfer texts and are based on Reference 8. In lieu of correction factor curves use the following procedure to derive the factor:

1. Assume shellside temperature varies linearly with length.
2. For first trial on tubeside assume equal heat is transferred in each pass with constant fluid heat capacity.
3. Using the end temperatures of each shell and tube pass calculate ΔT_M for each tube pass. From this the fraction of total duty for each tube pass is determined.
4. For the new end temperatures calculate the new ΔT_M for each tube pass.
5. The arithmetic average of the tube pass ΔT_M's is the ΔT_M corrected for number of passes. $F = \Delta T_M$ corrected/ΔT_M uncorrected.

The above procedure will quickly give numbers very close to the curves.

One thing to be careful of in cross exchangers is a design having a so-called "temperature cross." An example is shown in Figure 7.1.

In Figure 7.1 the colder fluid being heated emerges hotter than the outlet temperature of the other fluid. For actual heat exchangers that deviate from true countercurrent flow the following things can happen under temperature cross conditions:

1. The design can prove to be impossible in a single shell.
2. The correction factor can be quite low requiring an uneconomically large area.
3. The unit can prove to be unsatisfactory in the field if conditions change slightly.

For Figure 7.1 assuming one shell pass and two or more tube passes, the correction factor is roughly 0.7. This shows the undesirability of a temperature cross in a single shell pass.

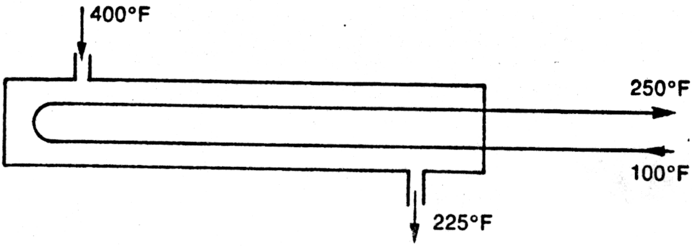

Figure 7.1. Shown here is an example of a temperature cross.

The calculation procedure for temperature correction factors won't work for a temperature cross in a single shell pass, but this is an undesirable situation anyway.

Some conditions require breaking up the exchanger into multiple parts for the calculations rather than simply using corrected terminal temperatures. For such cases one should always draw the q versus temperature plot to be sure no undesirable pinch points or even intermediate crossovers occur.

An example of a multisection calculation would be a propane condenser. The first section could be a desuperheating area, where q versus T would be a steeply sloped straight line followed by a condensing section with a straight line parallel to the q axis (condensing with no change in temperature). Finally, there could be a subcooling section with another sloped line. One can calculate this unit as three separate heat exchangers.

Determining Heat Exchanger Heat Transfer Coefficient

To do this one must sum all the resistances to heat transfer. The reciprocal of this sum is the heat transfer coefficient. For a heat exchanger the resistances are

Tubeside fouling	R_{FT}
Shellside fouling	R_{FS}
Tube metal wall	R_{MW}
Tubeside film resistance	R_T
Shellside film resistance	R_S

For overall tubeside plus shellside fouling use experience factors or 0.002 for most services and 0.004 for extremely fouling materials. Neglect metal wall resistance for overall heat transfer coefficient less than 200 or heat flux less than 20,000. These will suffice for ballpark work.

For film coefficients many situations exist. For the purposes of this small handbook Table 7.7 gives ballpark estimates of film resistance at reasonable design velocities.

Tables 7.8 and 7.9 give overall ballpark *U* values for air cooled heat exchangers.

For liquid boiling the designer is limited by a maximum flux q/A. This small handbook cannot treat this subject in detail. For most applications assuming a limiting flux of 10,000 will give a ballpark estimate.

Air-Cooled Heat Exchanger Rough Rating

The preceding information will suffice to rough rate shell and tube units. For air cooled heat exchangers more is involved in rough rating. A suggested procedure is as follows (References 2 and 3):

1. Calculate exchanger duty (MMBtu/hr).
2. Select an overall U_x from Tables 7.8 and 7.9 (based on finned area). Arbitrarily use ½″-fins, 9 to the inch for determining U_x.
3. Calculate approximate air temperature rise from

$$\Delta T_a = \left(\frac{U_x + 1}{10}\right)\left(\frac{T_1 + T_2}{2} - t_1\right) \tag{7.12}$$

where

ΔT_a = Airside temperature rise, °F
U_x = Overall coefficient based on finned area, Btu/hr ft^2 °F
T_1, T_2 = Process side inlet and outlet temperatures, °F
t_1 = Air inlet temperature, °F

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Table 7.7. Film Resistances

Liquids	**R**
Water	0.0013
Gasoline	.0067
Gas Oils	.0115
Viscous Oils Heating	.0210
Viscous Oils Cooling	.0333
Organic Solvents	.0036
Gases	
Hydrocarbons	
Low Pressure	.0364
High Pressure	.0200
Air	
Low Pressure	.0500
High Pressure	.0250
Vapors Condensing	
Steam	
No Air	0.0006
10% Air by Vol.	.0010
20% Air by Vol.	.0040
Gasoline	
Dry	.0067
With Steam	.0044
Propanes, Butanes, Pentanes	
Pure	.0033
Mixed	.0067
Gas Oils	
Dry	.0133
With Steam	.0090
Organic Solvents	.0030
Light Oils	.0033
Heavy Oils (vacuum)	.0285
Ammonia	.0133
Evaporation	
Water	.0007
Organic Solvents	.0050
Ammonia	.0033
Light Oils	.0044
Heavy Oils	.0333

Table 7.8. Typical Overall Heat-transfer Coefficients for Air Coolers (Reference 3)

Service		Fintube ½ in by 9	Fintube ⅝ in by 10
Water & water solutions			
		U_b U_x	U_b U_x
Engine jacket water ($r_t = .001$)		110 — 7.5	130 — 6.1
Process water ($r_f = .002$)		95 — 6.5	110 — 5.2
50-50 Ethyl glycol-water ($r_f = .001$)		90 — 6.2	105 — 4.9
50-50 Ethyl glycol-water ($r_f = .002$)		80 — 5.5	95 — 4.4
Hydrocarbon liquid coolers			
	Viscosity C_p	U_b U_x	U_b U_x
	0.2	85 — 5.9	100 — 4.7
	0.5	75 — 5.2	90 — 4.2
	1.0	65 — 4.5	75 — 3.5
	2.5	45 — 3.1	55 — 2.6
	4.0	30 — 2.1	35 — 1.6
	6.0	20 — 1.4	25 — 1.2
	10.0	10 — 0.7	13 — 0.6
Hydrocarbon gas coolers			
	Pressure, psig	U_b U_x	U_b U_x
	50	30 — 2.1	35 — 1.6
	100	35 — 2.4	40 — 1.9
	300	45 — 3.1	55 — 2.6
	500	55 — 3.8	65 — 3.0
	750	65 — 4.5	75 — 3.5
	1000	75 — 5.2	90 — 4.2
Air and flue-gas coolers (Use one-half of value given for hydrocarbon gas coolers)			
Steam condensers (Atmospheric pressure & above)			
		U_b U_x	U_b U_x
Pure steam ($r_f = .0005$)		125 — 8.6	145 — 6.8
Steam with non-condensibles		60 — 4.1	70 — 3.3
HC condensers			
	Pressure, psig	U_b U_x	U_b U_x
	0° range	85 — 5.9	100 — 4.7
	10° range	80 — 5.5	95 — 4.4
	25° range	75 — 5.2	90 — 4.2
	60° range	65 — 4.5	75 — 3.5
	100° & over range	60 — 4.1	70 — 3.3
Other condensers			
		U_b U_x	U_b U_x
	Ammonia	110 — 7.6	130 — 6.1
	Freon 12	65 — 4.5	75 — 3.5

Note: U_b is overall rate based on bare tube area and U_x is overall rate based on extended surface.

Table 7.9. Typical Transfer Coefficients for Air-Cooled Heat Exchangers (Reference 6)

Condensing service	U Btu/hr, ft^2, °F
Amine reactivator	90-100
Ammonia	100-120
Freon 12	60-80
Heavy naphtha	60-70
Light gasoline	80
Light hydrocarbons	80-95
Light naphtha	70-80
Reactor effluent — Platformers, Rexformers, Hydroformers	60-80
Steam (0-20 psig)	130-140
Still overhead — light naphthas, steam and non-condensible gas	60-70
Gas cooling service	
Air or flue gas @ 50 psig. ($\Delta P = 1$ psi.)	10
Air or flue gas @ 100 psig. ($\Delta P = 2$ psi.)	20
Air or flue gas @ 100 psig. ($\Delta P = 5$ psi.)	30
Ammonia reactor stream	80-90
Hydrocarbon gases @ 15-50 psig. ($\Delta P = 1$ psi.)	30-40
Hydrocarbon gases @ 50-250 psig. ($\Delta P = 3$ psi.)	50-60
Hydrocarbon gases @ 250-1,500 psig. ($\Delta P = 5$ psi.)	70-90
Liquid cooling service	
Engine jacket water	120-130
Fuel oil	20-30
Hydroformer and Platformer liquids	70
Light gas oil	60-70
Light hydrocarbons	75-95
Light naphtha	70
Process water	105-120
Residuum	10-20
Tar	5-10

Coefficients are based on outside bare tube surface for 1-in. O.D. tubes with 8 extruded Al fins/in., ⅝ in. high, 16.9 surface ratio.

Table 7.10. Design Values for Rough Rating

Typical Section Width, ft	Typical Tube Length, ft
6	6, 10, 15, 20, 24, 30
8	10, 15, 20, 24, 30
12	12, 16, 24, 32, 40
16	16, 24, 32, 40

APF = Extended area ft^2/ft of tube = 3.80
AR = Extended area/bare area = 14.5
APSF = Extended area/bundle face area = No. tube rows (22.8)

4. Calculate ΔT_M and apply appropriate correction factor F.
5. Calculate exchanger extended area from

$$A_x = \frac{q}{U_x \, \Delta T_M} \tag{7.13}$$

where

A_x = Extended (finned) surface, ft^2
q = Duty, Btu/hr
ΔT_M = Log mean temperature difference, °F

6. Estimate number of tube rows from Hudson Company optimum bundle depth curve, Figure 7.2. Use 4 to 6 tube rows if curve comes close to that number.
7. Arbitrarily choose 1″ OD tubes, ½″-fins, 9 to the inch at 2″ Δ pitch. This will give "middle of the road" face area. Use Table 7.10.
8. Calculate face area (F_a), ft^2

$$F_a = \frac{A_x}{APSF}$$

9. Pick a desirable combination of tube length and section width to achieve the approximate face area.

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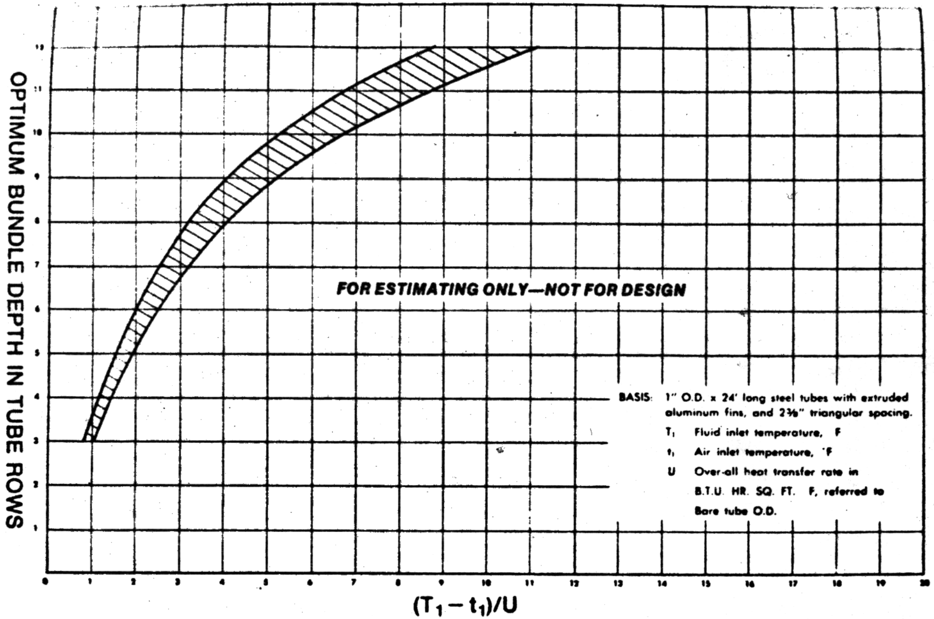


Figure 7.2. This is the effect of temperature level and overall transfer rate upon optimum bundle depth (Reference 6).

10. Estimate number of fans. Use the fact that fanned section length divided by bay width seldom exceeds 1.8. A 16-foot wide bay with 24-foot tubes would have one fan (ratio = 1.5). The same 16-foot wide bay with 32-foot tubes would have 2 fans (ratio would be 2.0 for 1 fan).
11. Estimate minimum fan area by

$$FAPF = Fan\ Area\ Per\ Fan = \frac{0.40\,(F_a)}{Number\ of\ Fans}$$

12. From above calculate fan diameter rounded up to the next even feet.
13. Calculate horsepower per fan from the section in this chapter "Air-Cooled Exchangers—Air Side Pressure Drop."
14. Number of tubes (N_T) can be obtained from

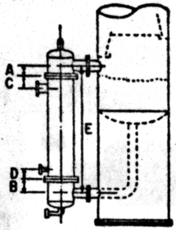
$$N_T = \frac{A_x}{APF\,(Tube\ length)}$$

Reboilers

Some rules of thumb are

1. Use 10,000 maximum flux in lieu of experimental data or detailed calculations as previously discussed in this chapter.
2. For pressure distillation the reboiler duty should run roughly 0.5 MMBtu/hr (D^2) where D is the distillation column diameter in feet. At atmospheric pressure the factor runs roughly 0.3.
3. For vacuum distillation the reboiler duty should run in the neighborhood of 0.15 MMBtu/hr (D^2).
4. Thermosyphon reboilers don't have full vaporization per pass. In fact the exit fluid is mostly liquid. A 4:1 liquid to vapor ratio is about minimum. This ratio can run to 10:1 or higher.
5. The hydraulic aspects of reboiler design are as important as the heat transfer aspects. Installation rules of thumb for the popular vertical thermosyphon reboiler are given in Table 7.11, which is based on 1″ OD tubes on 1.25″ triangular pitch.

Table 7.11. Typical Thermosiphon Reboiler Design Standards (Reference 7)



SHELL O.D., Inches	NO. TUBES	TUBE SHEET FACES	APPROX. AREA	NOZZLE SIZES				DIMENSIONS				
				VAPOR, Inches	LIQUID, Inches	STEAM, Inches	COND., Inches	A Inches	B Inches	C Inches	D Inches	E
16	108	4'-11 3/4"	132 □'	6	4	4	1 1/2	7 15/16	5 15/16	8	5 3/4	6'-1 3/4"
20	176	4'-11 3/4"	215 □'	8	6	4	2	8 15/16	7 15/16	8	5 3/4	6'-4 3/4"
24	272	4'-11 3/4"	333 □'	10	6	6	3	9 15/16	7 15/16	9	7 3/4	6'-5 3/4"
30	431	4'-11 3/4"	527 □'	12	6	6	3	11 7/16	7 15/16	9	7 3/4	6'-7 1/4
36	601	4'-11 3/4"	735 □'	16	8	8	4	13 3/16	9 15/16	10 3/4	8	6'-11"
24	272	6'-7 3/4"	448 □'	10	6	6	3	9 15/16	7 15/16	9	6 3/4	8'-1 3/4"
30	431	6'-7 3/4"	710 □'	12	6	6	3	11 7/16	7 15/16	9	6 3/4	8'-3 1/4"
36	601	6'-7 3/4"	990 □'	16	8	8	4	13 3/16	9 15/16	10 3/4	8	8'-7"
42	870	6'-7 3/4"	1,440 □'	16	10	8	4	17 11/16	10 15/16	10 1/16	6 13/16	9'-0 1/2"
30	431	9'-11 3/4"	1,065 □'	12	6	8	3	11 7/16	7 15/16	9	6 3/4	11'-7 1/4"
36	601	9'-11 3/4"	1,520 □'	16	8	8	4	13 3/16	9 15/16	10 3/4	6 3/4	11'-11"
42	870	9'-11 3/4"	2,180 □'	16	10	8	4	17 11/16	10 15/16	10 1/16	6 13/16	12'-4 1/2"

Vacuum Condensers

Outlet Temperature and Pressure. It is important to have proper subcooling in the vent end of the unit to prevent large amounts of process vapors from going to the vacuum system along with the inerts.

Control. It is necessary to have some over-surface and to have a proper baffling to allow for pressure control during process swings, variable leakage of inerts, etc. One designer adds 50% to the calculated length for the over-surface. The condenser must be considered part of the control system (similar to extra trays in a fractionator to allow for process swings not controlled by conventional instrumentation).

The inerts will "blanket" a portion of the tubes. The blanketed portion has very poor heat transfer. The column pressure is controlled by varying the percentage of the tube surface blanketed. When the desired pressure is exceeded, the vacuum system will suck out more inerts, and lower the percentage of surface blanketed. This will increase cooling and bring the pressure back down to the desired level. The reverse happens if the pressure falls below that desired. This is simply a matter of adjusting the heat transfer coefficient to heat balance the system.

Figure 7.3 shows typical baffling. The inerts move through the first part of the condenser as directed by the baffles. The inerts then "pile-up" at the outlet end lowering heat transfer as required by the controller. A relatively large section must be covered by more or less stagnant inerts which are sub-cooled before being pulled out as needed. Without proper baffles, the inerts build up in the condensing section and decrease heat transfer until the pressure gets too high. Then the vacuum valve opens wider pulling process vapor and inerts into the vacuum system. Under these conditions pressure control will be very poor.

Pressure Drop. Baffling must be designed to keep the pressure drop as low as possible. The higher the pressure drop the higher the energy consumption and the harder the job of attaining proper vent end sub-cooling. Pressure drop is lower at the outlet end because of smaller mass flow.

By-Passing. Baffles should prevent by-pass of inlet vapor into the vent. This is very important.

Typical Condenser. Figure 7.3 illustrates an inlet "bathtub" used for low vacuums to limit pressure drop at entrance to exchanger and across first rows of tubes. Note staggered baffle spacing with large spacing at inlet, and the side to side (40% cut) baffles. Enough baffles must be used in the inlet end for minimum tube support. In the last 25% of the outlet end a spacing of 1/10 of a diameter is recommended.

Fogging

Cause. Fogging occurs in a condenser when the mass transfer doesn't keep up with the heat transfer. The design must pro-

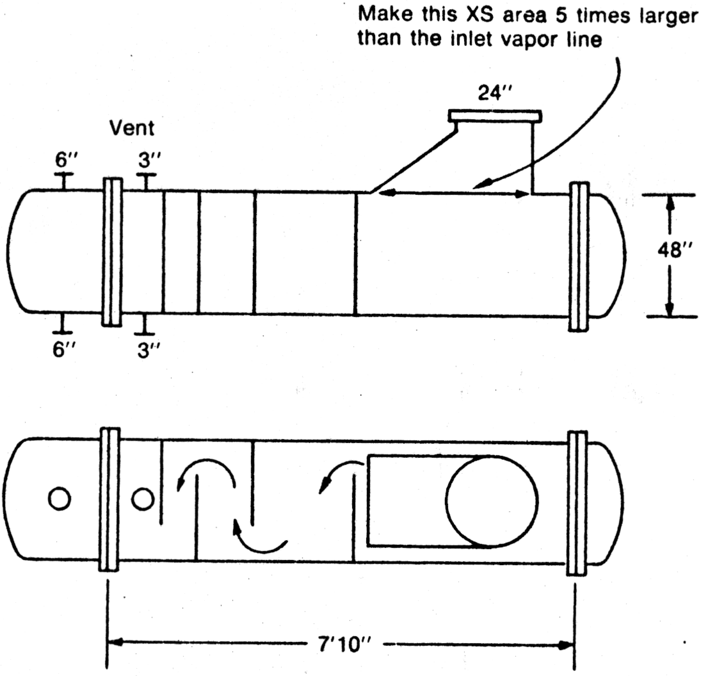


Figure 7.3. *Baffling and inlet "bathtub" are shown in this typical vacuum condenser design.*

vide sufficient time for the mass transfer to occur. A high temperature differential (ΔT) with non-condensibles present or a wide range of molecular weights can produce a fog. The high ΔT gives a high driving force for heat transfer. The driving force for mass transfer, however, is limited to the concentration driving force (ΔY) between the composition of the condensible component in the gas phase and the composition in equilibrium with the liquid at the tube wall temperature. The mass transfer driving force (ΔY) thus has a limit. The ΔT driving force can under certain conditions increase to the point where heat transfer completely outstrips mass transfer, which produces fogging.

Nature of a Fog. Fog, like smoke, is a colloid. Once a fog is formed it is very difficult to knock down. It will go right through packed columns, mist eliminators, or other such devices. Special devices are required to overcome a fog such as an electric precipitator with charged plates. This can overcome the zeta potential of the charged particles and make them coalesce.

A colloid fog will scatter a beam of light. This is called the "Tyndall Effect" and can be used as a troubleshooting tool.

Cures. Eliminate the source of fogging. Use a smaller ΔT and thus more surface for mass transfer. Try to minimize $\Delta T/\Delta Y$.

Calculations. To check a design for possible fogging a procedure is presented that rightly considers mass transfer and heat transfer as two separate processes. Expressions for these are

Heat Transfer

$$Q = UA\Delta T_M \text{ or } Q = h_i A \Delta T_i \tag{7.14}$$

where *i* refers to the condensing side only,

Mass Transfer

$$W = KA\Delta Y = KA\,(Y - Y_i) \tag{7.15}$$

where

h_i = Condensing side film coefficient, Btu/hr ft^2 °F
W = Condensate rate, lbs/hr
K = Mass transfer coefficient, lb/hr ft^2 ΔY
Y = Composition of the condensible component in the gas phase
Y_i = Composition of the condensible component in equilibrium with liquid at tube wall temperature

Mass and heat transfer are related as follows:

$$\frac{h_i}{KC} = \left(\frac{K_T}{C\,\rho\,K_D}\right)^{2/3} \tag{7.16}$$

where

C = Heat capacity (gas phase), Btu/lb °F
ρ = Gas density, lb/ft^3
K_T = Thermal conductivity, Btu/hr ft^2 (°F/ft)
K_D = Diffusivity, ft^2/hr

The ratio of heat transfer to mass transfer is:

$$\frac{Q}{W} = \frac{h_i\,\Delta T_i}{K\Delta Y} \tag{7.17}$$

since

$$\frac{h_i}{K.} = C\left(\frac{K_T}{C\rho\,K_D}\right)^{2/3}$$

then

$$\frac{Q}{W} = C\left(\frac{K_T}{C\rho\,K_D}\right)^{2/3}\frac{\Delta T_i}{\Delta Y}$$

Note that Q refers only to sensible heat transfer. All latent heat is transferred via mass transfer. Likewise, h_i refers only to a dry gas coefficient (no condensation considered).

The calculations are made as follows. The exchanger is divided into small increments to allow numerical integrations. A tube wall temperature is first calculated and then Q/W. The gas temperature and composition from an increment can then be calculated. If the gas composition is above saturation for the temperature, any excess condensation can occur as a fog. This allows the degree of fogging tendency to be quantified. Whenever possible experimental data should be used to determine the ratio of heat transfer to mass transfer coefficients. This can be done with a simple wet and dry bulb temperature measurement using the components involved.

Miscellaneous Heat Exchanger Rules of Thumb

1. For fixed tubesheet design of shell and tube heat exchangers don't allow too high a temperature difference between tubeside and shellside without providing a shellside expansion joint. The author has seen 70°F (one company) and 100°F (another company) used as this limit. An easy way to calculate the maximum stress is as follows:

   a. Assume the tubes are at tubeside bulk temperature and the shell is at shellside bulk temperature.
   b. Calculate the elongation of tubes, if unhampered, and shell, if unhampered, starting at 70°F and using the respective coefficients of expansion.
   c. If the tubes would have grown more or less than the shell, the difference will set up stress in both members, one in tension and the other in compression.
   d. Assume the deformation (strain) in each member is inversely proportional to its cross sectional area. In other words, the fraction of the total strain in inches/inch of length will be proportionally more for the member (tubes or shell) having the smallest cross section.
   e. For each member, multiply its strain by Young's modulus (Modulus of Elasticity) in consistent units to get stress. Strain is dimensionless so Young's modulus in lb/in^2 will yield stress in lb/in^2.
   f. Compare this with the maximum allowable stress for the material used.
   g. The tensile and compressive moduli for steel are essentially the same.

2. Typical handling of design parameters.

   $DP = MOP + 10\%$ but not less than $MOP + 30$ psi

DP for vacuum use 15 psi external pressure for cases having atmospheric pressure on the other side.

DT: Below 32°F *DT* = minimum operating temperature
Above 32°F *DT* = *MOT* + 25°F but not less than 150°F

where

DP = Design pressure
DT = Design temperature
MOP = Max. operating pressure
MOT = Max. operating temperature

3. 40% baffle cut = 40% open area is a good rule of thumb maximum for shell and tube heat exchangers.

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8

DISTILLATION

Because this subject is relatively complicated, only some quick estimating techniques will be discussed.

Relative Volatility

The equilibrium vaporization constant K is defined for a compound by

$$K_i = \frac{Y_i}{X_i} \tag{8.1}$$

where

Y_i = Mole fraction of component i in the vapor phase
X_i = Mole fraction of component i in the liquid phase

To calculate a distillation, the relative volatility α is needed. It is defined as

$$\alpha = \frac{K_i}{K_j} \tag{8.2}$$

where i and j represent two components to be separated.

Raoult's Law for ideal systems is

$$p_i = P_i X_i \tag{8.3}$$

where

p_i = Partial pressure of i
P_i = Vapor pressure of pure component i

By definition

$$p_i = \Pi Y_i \tag{8.4}$$

where

Π = Total pressure of the system

so

$$P_i X_i = \Pi Y_i$$

and

$$\frac{P_i}{\Pi} = \frac{Y_i}{X_i} = K_i$$

Therefore for systems obeying Raoult's Law

$$\alpha = \frac{P_i}{P_j} \tag{8.5}$$

Having a relative volatility one can estimate theoretical distillation stages. There are a number of modern computer programs for distillation. These give excellent results, however, these must be applied by someone with a sound understanding of distillation in general and the accuracy of the input data in particular.

For our purposes there are shortcut equations or one can fairly quickly produce a McCabe-Thiele diagram for binary systems. It helps to have a rough feel for the answer even if a computer program is to be run. For many splits a McCabe-

Thiele diagram is a real help in "seeing" how the system behaves.

Minimum Stages

For binary systems or systems which approach binary, the Fenske-Underwood-Erbar/Maddox method is recommended. For minimum stages use the Fenske equation (Reference 14).

$$N_m + 1 = \frac{\ln \left(\frac{X_{LD}}{X_{HD}}\right)\left(\frac{X_{HB}}{X_{LB}}\right)}{\ln \alpha_{average}} \tag{8.6}$$

where

N_m = Minimum theoretical stages, not including reboiler (total reflux)

$X_{LD}, X_{HD}, X_{HB}, X_{LB}$ = Mole fractions of light and heavy components in the distillate and bottoms

$\alpha_{average} = \sqrt{\alpha_D \, \alpha_B}$, where differences are small in α_D and α_B. For values of α near 1.0 great care must be exercised as a small change in α has a large effect in the results.

Minimum Reflux Ratio

For binary or near binary minimum reflux ratio, *L/D* min, use the Underwood equations (Reference 12).

Bubble Point Liquid Feed

$$R_m = \frac{1}{\alpha - 1}\left[\frac{X_{LD}}{X_{LF}} - \frac{\alpha(1 - X_{LD})}{(1 - X_{LF})}\right] \tag{8.7}$$

Dew Point Vapor Feed

$$R_m = \frac{1}{\alpha - 1}\left[\frac{\alpha X_{LD}}{X_{LF}} - \frac{(1 - X_{LD})}{(1 - X_{LF})}\right] - 1 \tag{8.8}$$

where

R_m = Minimum reflux ratio
L = Reflux rate, lb mols/hr
D = Distillate rate, lb mols/hr
B = Bottoms rate, lb mols/hr
F = Feed rate, lb mols/hr

B and F will be needed later.

For a multicomponent system, a simple technique to yield conservative results is as follows. Combine a light key component and all lighter components, and a heavy key component and all heavier components into two groups to get X_F, X_D, X_B. For the key groups use the α of the keys themselves.

A more accurate but longer method is to disregard presence of components having a D/B greater than 100 or less than 0.01. Light and heavy key groups are generated. The group α's are found by plotting log D/B versus α with a straight line drawn through the major points. The α for each group is read at the D/B for the group. This procedure was introduced by Hengstebeck (Reference 11).

Actual Reflux Ratio

For determining the theoretical stages at an actual reflux ratio, use the Erbar/Maddox relationship in Figure 8.1. N is the theoretical stages and R is the actual reflux ratio L/D. The method is defined in Reference 7.

The actual reflux ratio that one picks should be optimized from economics data. For a ballpark estimate use $1.1 - 1.2\ R_m$ for a refrigerated system and $1.2 - 1.35\ R_m$ for a hot system.

Reflux To Feed Ratio

Heretofore, the reflux ratio has been defined as reflux/distillate, L/D. Another very useful molar ratio is reflux/feed, L/F. For binary systems L/F for all practical purposes is unchanging for wide differences in feed composition, so long as the following hold:

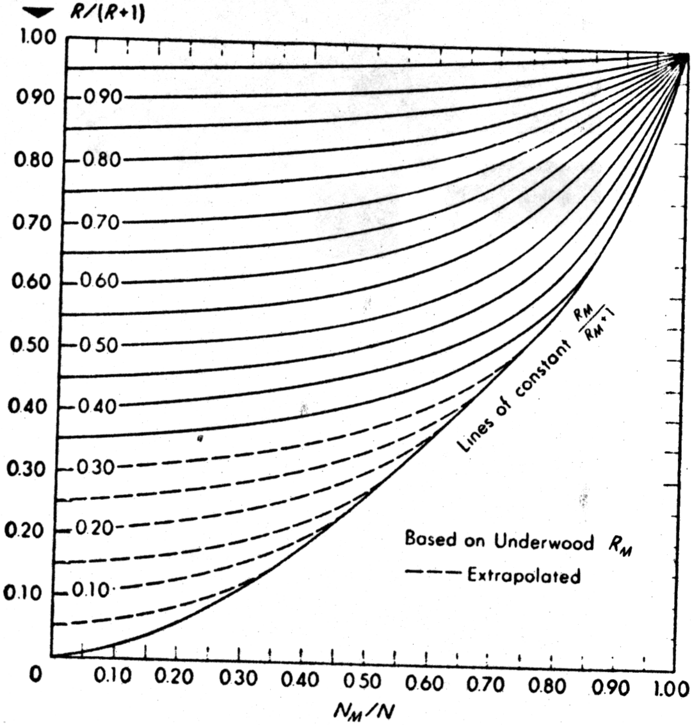


***Figure 8.1.** In this graph of the Erbar-Maddox relationship of reflux versus stages,* N *is the theoretical stages and* R *is the reflux ratio* L/D *(Reference 7).*

1. The distillate and bottoms compositions, but not necessarily the quantities, are held constant.
2. The feed tray is kept matched in composition to the feed (which means the feed tray moves with feed composition changes).

The reader can verify the above using the Underwood equations and the tower material balance. The author recently calculated a case where a large feed change would change L/D by 46%, whereas L/F changed only 1%. Several investigators report that the stability of L/F is well proven in the field. L/F is a good factor to use in predicting the effect of feed changes for design and in an operating plant.

McCabe-Thiele Diagram

In addition to the previously mentioned shortcut equations, plotting a McCabe-Thiele diagram is also a very useful tool. The equation for the equilibrium X-Y diagram and plotting of the operating lines are described next.

Equilibrium Curve

$$Y = \frac{X\alpha}{1 + (\alpha - 1)X} \qquad (8.9)$$

where

Y = Mole fraction of the light component in the vapor
X = Mole fraction of the light component in the liquid

Another useful form is

$$\alpha = \frac{Y(1 - X)}{X(1 - Y)} \qquad (8.10)$$

q Line — The Operating Lines Intersect on This Line

The thermal condition of the feed is designated as q, and is approximately the amount of heat required to vaporize one mole of feed at the feed tray conditions, divided by the latent heat of vaporization of the feed. One point on the q line is on the 45° line at X_F.

Bubble point liquid feed, $q = 1.0$
Dew point vapor feed, $q = 0$
General feed, $q = (L_S - L_R)/F$
Slope of q line $= q/(q - 1)$

where

L_S = Liquid rate in stripping section, lb mols/hr
L_R = Liquid rate in rectifying section, lb mols/hr

Rectifying Section Operating Line

One point is on the 45° line at X_D, and the slope is L_R/V_R, where V_R is rectifying section vapor rate. Another point is on Y axis above the origin at DX_D/V_R.

Stripping Section Operating Line

One point is on the 45° line at X_B, and the slope is L_S/V_S, where S refers to the stripping section. Another point is the intersection of the rectifying section operating line with the q line. And still another point is on the Y axis at location below the origin of $-(B/V_S)(X_B)$.

Tray Efficiency

The foregoing define the number of theoretical stages. Actual stages depend upon the tray efficiency which will probably be the weakest number in the design. Using operating data from a similar system is certainly best where possible. Table 8.1 gives some shortcut correlations.

Ludwig (Reference 1) discusses new work by the A.I.Ch.E. which has produced a method more detailed than the previous short-cut methods. He states that some of the short-cut methods can be off by 15-50% as indicated by the A.I.Ch.E. work. The spread of the Drickamer and Bradford correlation shown in the Ludwig plot is about 10 points of efficiency or ± 5 efficiency points around the curve. Ludwig states that comparisons between short-cut and A.I.Ch.E. values indicate that deviations for the short-cut methods are usually on the safe or low side.

Maxwell's correlation was generated from hydrocarbon data only. Ludwig states that the Drickamer and Bradford correlation is good for hydrocarbons, chloronated hydrocarbons, glycols, glycerine and related compounds, and some rich hydrocarbon absorbers and strippers.

Table 8.1. Fractionator Overall Tray Efficiency, %

Viscosity Centipoises	Gunness (6) and Other Data Plotted Versus Reciprocal viscosity in Maxwell (2). (Average viscosity of liquid on the plates)	Drickamer and Bradford (4) Correlation Plotted in Ludwig (1). (Molal average viscosity of the feed)
0.05•	...	98
0.10	104••	79
0.15	86	70
0.20	76	60
0.30	63	50
0.40	56	42
0.50	50	36
0.60	46	31
0.70	43	27
0.80	40	23
0.90	38	19
1.00	36	17
1.50	30	7
1.70	28	5

•Extrapolated slightly

••Maxwell (2) explains how efficiencies above 100% are quite possible.

Ludwig also presents correlations of O'Connell (Reference 5). He warns that O'Connell may give high results. Ludwig suggests using the O'Connell absorber correlation only in areas where it gives a lower efficiency than the fractionator correlation of Drickamer and Bradford or O'Connell. The O'Connell correlations are given in Table 8.2.

For high values of α low tray efficiency usually results.

Tower Diameter

Several quick methods exist for estimation of tower diameter. For final design, of course, a full rating will be required.

Older Correlations For Bubble Cap Trays

$$U_{STD} = 0.227 \sqrt{\frac{\rho_L}{\rho_V} - 1} \qquad (8.11)$$

Table 8.2. Overall Tray Efficiency, % (O'Connell)

Correlating Factor	Fractionators•	Absorbers $\left(\frac{HP}{\mu}=\frac{\rho}{KM\mu}\right)$••
0.01	..	8
0.05	..	17
0.10	87	22
0.15	80	23
0.20	74	26
0.30	67	29
0.40	62	32
0.50	57	33
0.60	55	35
0.70	52	36
0.80	51	37
0.90	49	38
1	48	39
1.5	44	42
2	41	45
3	37	48
4	35	52
5	33	53
6	32	56
7	32	57
8	31	58
9	..	60
10	..	61

•Fractionators = (Relative volatility of key components) (Viscosity of feed in centipoises). Relative volatility and viscosity are taken at average tower conditions between top and bottom.

••H = Henry's law constant, mols/ ft³ (ATM)
P = Pressure, ATM
μ = Viscosity, centipoises
ρ = Density, lbs/ft³
K = Equilibrium K of key components
M = Mol. wt. of liquid

where

U_{STD} = Standard velocity, ft/sec
$L, V,$ = Refer to liquid and vapor
ρ = Density, lb/ft³

$$G_{STD} = C\sqrt{\rho_V(\rho_L-\rho_V)} \qquad (8.12)$$

(Reference 13)

Table 8.3. Allowable Velocities for Bubble Cap Trays

Service	% U_{STD} (Equation 8.11)	% G_{STD} (Equation 8.12)
Vacuum towers	110**	...
10 PSIG	...	150
50 PSIG	...	125
Refinery Group*	100**	...
Debutaners or other 100-250 PSIG towers	80	100
Depropanizers, stabilizers, other high pressure towers	60	90

*"Refinery Group" = low pressure naphtha fractionators, gasoline splitters, crude flash towers, etc.

**Applies for 24" tray spacing or above.

NOTE: It is frequently found that for towers with high liquid loading or high vapor density, diameter is set to give adequate downcomer area rather than free area for vapor travel, e.g., for absorbers.

where

G_{STD} = Standard mass velocity, lb/hr/ft^2

The values of C in relation to tray spacings are roughly

Tray Spacing	Value of C
18"	460
24"	590
30"	660
36"	700

Percentages are applied to the standard velocities according to the tower service. The percentages are found in Table 8.3. For more accurate C factors see Reference 1 or 13.

Rough Rule of Thumb for Sieve Trays

Hold
$$F = U\sqrt{\rho} \tag{8.13}$$

less than 1.0 to 1.5. F is the correlating factor. U and ρ refer to the vapor.

Table 8.4. System Factors (Reference 10)

Service	System Factor
Non-foaming, regular systems	1.00
Fluorine systems, e.g., BF_3, Freon	.90
Moderate foaming, e.g., oil absorbers, amine and glycol regenerators	.85
Heavy foaming, e.g., amine and glycol absorbers	.73
Severe foaming, e.g., MEK units	.60
Foam-stable systems, e.g., caustic regenerators	.30- .60

Approximation For Quick Estimates With Valve Type Trays

See ballast tray calculation procedure.

Calculation/Procedure for Ballast Tray Minimum Tower Diameter

The following method will give quick approximate results but for complete detailed rating use Reference 10.

1. Determine vapor capacity factor *CAF*.

$$CAF = CAF_o \times \text{System Factor} \qquad (8.14)$$

The proper system factor can be found in Table 8.4.

2. For $\rho_V < 0.17$ use

$$CAF_o = (TS)^{0.65} \times (\rho_V)^{1/6}/12 \qquad (8.15)$$

For $\rho_V > 0.17$ use

$$CAF_o = 0.1392 + 0.01946\,(TS) - 0.000252\,(TS)^2 - 0.000635\,(TS)\,(\rho_V) \qquad (8.16)$$

where

CAF_o = Flood capacity factor at zero liquid load

TS = Tray spacing in inches

ρ_V = Vapor density, lbs/ft^3

Table 8.5. CAF_0 Limit Point (Reference 10)

ρ_V	Tray Spacing (in.) For Maximum Attainable CAF_0
≤2.0	48
2.5	36
2.8	30
3.2	24
3.4	22
3.7	20
3.9	18
4.3	16
4.8	14
5.4	12

Equation 8.16 is the author's correlation of a nomograph in Reference 10. It gives results within about 2% of the nomograph. It is limited to TS of 16 to 36 and ρ_V of 0.17 to 5.5. See item 3 for further limitations.

3. Generally CAF_0 increases with increased tray spacing and decreases with increased vapor density. However, for a given vapor density there is a tray spacing above which CAF_0 doesn't increase. Table 8.5 shows this relationship. Absorbers and strippers frequently operate with a liquid having essentially the same physical characteristics regardless of pressure. An example of this is a gas absorber. The same lean oil is used if the tower is operating at 100 or 1000 psi. This type of system is excluded from the CAF_0 limiting value.

4. Calculate V_{load}.

$$V_{load} = CFS\sqrt{\rho_V/(\rho_L - \rho_V)} \qquad (8.17)$$

where

CFS = Vapor rate, actual ft^3/sec

5. Using V_{load} and GPM (column liquid loading in gallons per minute), obtain approximate tower diameter for calculating flow path length. Use Equation 8.18.

$$(DTA)^2 = 3.025A + 0.0012AB + 7.647 \times 10^{-7} AB^2 + 2.81 \times 10^{-4} B^{1.719} \tag{8.18}$$

where

DTA = Approximate tower diameter, ft
$A = V_{load}$
$B = GPM$

Equation 8.18 is the author's correlation of a nomograph in Reference 10. It gives results within 5% of the nomograph for diameter 4 feet or greater and within 15% for smaller diameters. This is adequate for this first approximation of tower diameter. Equation 8.18 applies for

a. Single pass trays
b. 24″ tray spacing at 80% flood
c. $DT = 2' - 10'$
d. $V_{load} = 0 - 30$
e. $GPM = 0 - 1500$

6. For 2-pass trays

a. Divide V_{load} by 2
b. Divide GPM by 2
c. Obtain diameter from single pass equation
d. Multiply diameter by $\sqrt{2}$

For 4-pass trays replace the 2's by 4's.

7. Calculate flow path length.

$$FPL = 9 \times DTA/NP \tag{8.19}$$

where

FPL = Flow path length, inches
NP = Number of flow paths or passes

8. Calculate minimum active area.

$$AAM = \frac{V_{load} + (GPM \times FPL/13000)}{CAF \times FF} \qquad (8.20)$$

where

V_{load} = Vapor load for any tray in the section
GPM = Liquid load for the same tray
AAM = Minimum active area, ft²
FF = Flood factor or design per cent of flood, fractional. An FF of .65 to .75 should be used for column diameters under 36″.

9. Obtain downcomer design velocity, VD_{dsg}. Use the smallest value from these three equations

$$VD_{dsg} = 250 \times \text{system factor} \qquad (8.21)$$

$$VD_{dsg} = 41 \times \sqrt{\rho_L - \rho_V} \times \text{system factor} \qquad (8.22)$$

$$VD_{dsg} = 7.5 \times \sqrt{TS} \times \sqrt{\rho_L - \rho_V} \times \text{system factor} \qquad (8.23)$$

where

VD_{dsg} = Design velocity, GPM/ft²

For the system factor use values out of Table 8.4 except for the last item in that table (foam-stable systems) use 0.30.

10. Calculate downcomer minimum area.

$$ADM = GPM/(VD_{dsg} \times FF) \qquad (8.24)$$

where

ADM = Minimum downcomer area, ft²

If the downcomer area calculated above is less than 11% of the active area use the smaller of

ADM = Double that of Equation 8.24
ADM = 11% of the active area

11. Calculate minimum column cross-sectional area. Use the larger of

$$ATM = AAM + 2 \times ADM \tag{8.25}$$

$$ATM = \frac{V_{load}}{0.78 \times CAF \times FF} \tag{8.26}$$

where

ATM = Minimum column cross-sectional area, ft². Further detailed design calculations may result in a change in tower diameter.

12. Calculate column minimum diameter.

$$DT = \sqrt{ATM/.7854} \tag{8.27}$$

where

DT = Minimum column diameter, ft

Percentage of Flood for Existing Ballast Tray Columns

Use Equation 8.28

$$\%\ Flood = \frac{V_{load} + (GPM \times FPL/13000)}{AA \times CAF} \tag{8.28}$$

Minimum diameter for multipass trays is given in Table 8.6.

Table 8.6. Minimum Practical Diameter For Multiple Ballast Trays (Reference 10)

Multipass Tray Number of Passes	Minimum Diameter (ft)	Preferred Diameter (ft)
2	5	6
3	8	9
4	10	12
5	13	15

Table 8.7. General Pressure Drop Rules

Service	Design Pressure Drop (inches of water/ft of packed depth)
Absorbers and Regenerators (Nonfoaming system)	0.25 to 0.40
Absorbers and Regenerators (Foaming system)	0.10 to 0.25
Atmospheric or Pressure Stills and Fractionators	0.40 to 0.80
Vacuum Stills and Fractionators	0.10 to 0.40

Holding *GPM/WFP* below about 8 is preferred, although liquid rates as high as 20 *GPM/WFP* can and have been used. *WFP* is the width of the flow path in inches. Some companies like to use trays having no more than two passes.

Packed Towers

by

R. F. Strigle—Norton Company

Most packed towers are designed to operate at a given pressure drop per foot of packed depth. General rules for packed tower design pressure drop are given in Table 8.7. The pressure drop in a packed tower can be determined by using Table 8.8 and Figure 8.2.

The maxmium recommended design pressure drop is 1.0″ of water/ft for liquids having a specific gravity of 0.8 or greater. Lower maximum design pressure drop is recommended for lighter liquids. Packing capacity is independent of liquid surface tension from 4 to 72 dynes/cm.

The capacity of any packing at a constant liquid to gas ratio and a fixed pressure drop is inversely proportional to the square root of the packing factor shown in Table 8.8. To increase the capacity of an existing column by 40%, the packing must be replaced by one having a packing factor one-half as great.

The height equivalent to a theoretical plate *(HETP)* for tower packing is relatively constant for its entire operating range. The following rules of thumb may be used for a design pressure drop range from 0.20 to 1.0″ of water/ft of packed depth.

Table 8.8. Packing Factors,** (Wet and Dump Packed)

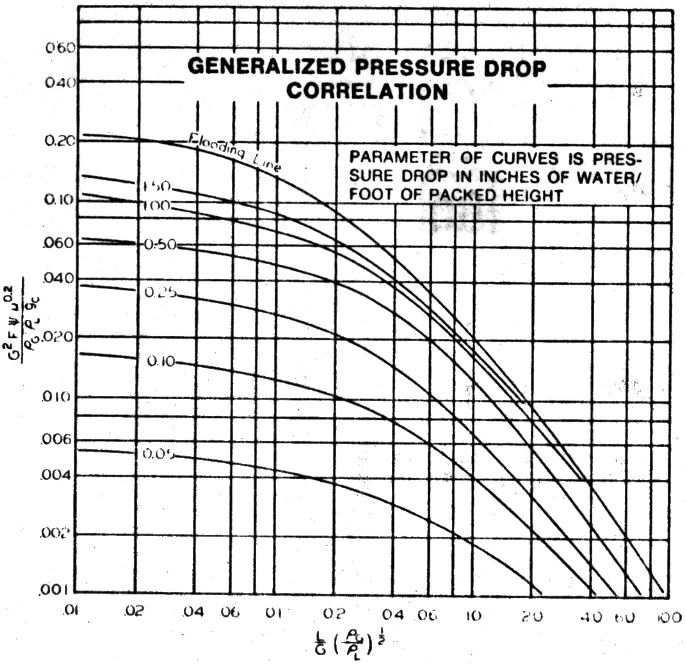
Type of Packing	Mat'l.	Nominal Packing Size (Inches)										
		¼	⅜	½	⅝	¾	1	1¼	1½	2	3	3½
Super Intalox	Ceramic	—	—	—	—	—	60	—	—	30	—	—
Super Intalox	Plastic	—	—	—	—	—	33	—	—	21	16	—
Intalox Saddles	Ceramic	725	330	200	—	145	98	—	52	40	22	—
Hy-Pak Rings	Metal	—	—	—	—	—	42	—	—	18	15	—
Pall Rings	Plastic	—	—	—	97	—	52	—	40	25	—	16
Pall Rings	Metal	—	—	—	70	—	48	—	28	20	—	16
Berl Saddles	Ceramic	× 900	—	× 240	—	170 □	110 □	—	65 □	× 45	—	—
Raschig Rings	Ceramic	× 1600 ⊖	× 1000 ⊖	580 ⦶	380 ⦶	255 ⦶	155 ⊕	× 125 ⊗	95 ⊗	65 ◉	× 37 ●	—
Raschig Rings 1/32" Wall	Metal	× 700	× 390	× 300	170	155	× 115	—	—	—	—	—
Raschig Rings 1/16" Wall	Metal	—	—	410	290	220	137	× 110	83	57	× 32	—

× Extrapolated
○ 1/32" Wall
⊖ 1/16" Wall
⦶ 3/32" Wall
⊕ ⅛" Wall
⊗ 3/16" Wall
◉ ¼" Wall
● ⅜" Wall
×× $F \approx a/\epsilon^3$ Obtained In 16" and 30" I.D. Tower
□ Data by Leva

Nominal Packing Size (in.)	HETP (in.)
1	15-21
1½	22-29
2	30-39

The efficiency of tower packing is insensitive to changes in liquid viscosity from 0.07 to 2.0 cps or changes in relative volatility from 1.03 to 20.

Packing support plates must develop at least as much capacity as the tower packing and, therefore, must have at least as great



L = LIQUID RATE, LBS/SEC., SQ. FT.
G = GAS RATE, LBS./SEC., SQ. FT.
ρ_L = LIQUID DENSITY, LBS/CU. FT.
ρ_G = GAS DENSITY, LBS/CU. FT.
F = PACKING FACTOR

μ = VISCOSITY OF LIQUID, CENTIPOISE
ψ = RATIO, $\frac{\text{DENSITY OF WATER}}{\text{DENSITY OF LIQUID}}$
g_c = GRAVITATIONAL CONSTANT = 32.2

Figure 8.2. ***The pressure drop in a packed tower can be determined by using this generalized pressure drop correlation and Table 8.8.***

an effective free space. Bed limiter and hold down plates are needed to prevent fluidization of the top surface of the packed bed in event the design pressure drop exceeds 0.50″ of water/ft or the column will be operating under surging vapor loads.

Modern packings tend to maintain the liquid distribution imposed upon them. To achieve the proper *HETP* value, if the liquid rate is 3 GPM/ft^2 or higher, a minimum of four distribution points per square foot is required. Columns operating below 2 GPM/ft^2 liquid rate require an increased number of liquid distribution points per square foot. A liquid distributor must be used at every liquid feed point. Packing depths usually are limited to 7-10 column diameters in a single bed. However, bed depth should not exceed 24-30 feet.

Redistributors are used only after two packed bed depths of liquid travel. They normally are specified in towers with low liquid rates or in systems with a high minimum reflux ratio.

In selecting a packing size, consideration must be given to the column diameter. Recommended minimum packing sizes are

Nominal Packing Size (in.)	Minimum Column Diameter (in.)
1	12
1½	24
2	42

Packed towers are widely used in absorption service because of their ability to handle high liquid loads. Maximum allowable liquid loading increases with increasing packing size as follows:

Nominal Packing Size (in.)	Maximum Irrigation Rate (GPM/ft^2)
1	39
1½	55
2	68

Absorption efficiency is more variable than fractionation efficiency. The height of a transfer unit *(HTU)* for 2" nominal size packings in typical absorption systems is given in Table 8.9.

Table 8.9. Typical Absorption Transfer Unit Heights

Solute	Absorbant	HTU (ft)
Ammonia	Water	1.9
Hydrogen Sulfide	16% NaOH Solution	4.3
Propane	Light absorption oil	4.5
Carbon Dioxide	17% MEA solution	4.7

Tower packings are manufactured from ceramics, metals, and plastics. Ceramics offer good corrosion and temperature resistance, but are subject to physical breakage. Metal packings are formed from light gauge strip, thus an alloy giving a corrosion rate no greater than 0.010 inches/year should be specified. Polypropylene is the most widely used plastic packing. It has a maximum continuous operating temperature of 275°F, but should be glass reinforced for service above 245°F.

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9
ABSORPTION

A general study of absorption can be confusing since the calculation methods for the two major types are quite different. First, there is hydrocarbon absorption using a lean oil having hydrocarbon components much heavier than the component absorbed from the gas stream. These absorbers may or may not be reboiled. For designing these, one uses equilibrium vaporization constants (K values) similarly to distillation. Another similarity to distillation is the frequent use of fractionating trays instead of packing. Canned computer distillation programs usually include hydrocarbon absorber options.

The other major type is gas absorption of inorganic components in aqueous solutions. For this type design one uses mass transfer coefficients. Packed towers are used so often for this type that its discussion is often included under sections on packed towers. However, in this book it is included in this chapter on "Absorption."

Hydrocarbon Absorption

Because of its similarity to distillation, many parts of this subject have already been covered, such as

1. Tray Efficiency
2. Tower Diameter Calculations
3. *K* Values

As for distillation, shortcut hand calculation methods exist for hydrocarbon absorption. In distillation, relative volatility (α) values are generated from the K values. For hydrocarbon absorption the K values are used to generate absorption and stripping factors. The 1947 Edmister method (Reference 7) using effective overall absorption and stripping factors and the well known Edmister graphs is very popular for hand calculations. An excellent write-up on this and the Kremser-Brown-Sherwood methods are on pages 38-50 of Reference 1.

Edmister Method (1947)

Briefly, the Edmister absorption method (1947) with a known rich gas going to a fixed tower is as follows:

1. Assume theoretical stages and operating temperature and pressure.

2. Knowing required key component recovery E_a, read A_e from Figure 9.1 at known theoretical trays n.

$$E_a = \frac{A_e^{n+1} - A_e}{A_e^{n+1} - 1} \tag{9.1}$$

$$E_s = \frac{S_e^{m+1} - S_e}{S_e^{m+1} - 1} \tag{9.2}$$

where

n = Number of theoretical stages in absorber
m = Number of theoretical stages in stripper
E_a = Fraction absorbed
E_s = Fraction stripped
A_e = Effective absorption factor
S_e = Effective stripping factor

3. Assume
   a. Total mols absorbed
   b. Temperature rise of lean oil (normally 20-40°F)
   c. Lean oil rate, mols/hr

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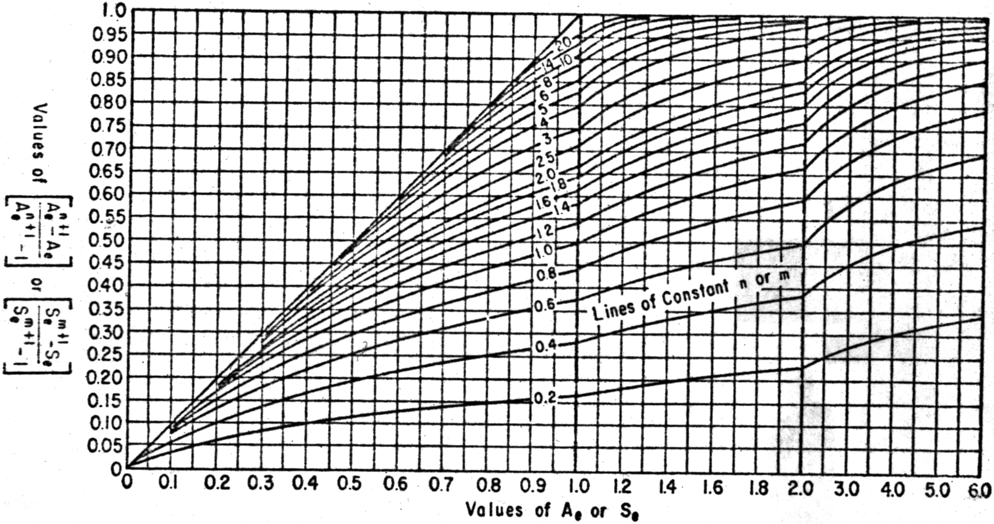

Figure 9.1. *This graph shows the absorption and stripping factors,* E_a *and* E_s, *versus effective values,* A_e *and* S_e *(efficiency functions). (By permission, W. C. Edmister,* Petroleum Engineer, *September, 1947 Series to January, 1948.)*

Table 9.1. Tower Balance

Quantity	Symbol	Equation
Rich gas entering at bottom	V_{n+1}	Known
Gas Absorbed	ΔV	Assumed
Lean gas leaving absorber	V_1	$V_1 = V_{n+1} - \Delta V$
Gas leaving bottom tray	V_n	$V_n = V_{n+1}(V_1/V_{n+1})^{1/N}$
Gas leaving tray 2 from top	V_2	$V_2 = V_1/(V_1/V_{n+1})^{1/N}$
Lean oil	L_o	Known
Liquid Leaving top tray	L_1	$L_1 = L_o + V_2 - V_1$
Liquid leaving bottom tray	L_n	$L_n = L_o + \Delta V$

Table 9.2. Tower Temperatures

Temperature	Symbol	Equation
Rich gas inlet	T_{n+1}	Known
Lean oil	T_o	Known
Temperature rise	ΔT	Assumed
Bottom Tray	T_n	$T_n = T_{n+1} + \Delta T$
Top Tray	T_1	$T_1 = T_n - \Delta T \dfrac{(V_{n+1} - V_2)}{(V_{n+1} - V_1)}$

4. Use Horton and Franklin's relationship (Reference 9) for tower balance in mols/hr. This is shown in Table 9.1 (Reference 1).

5. Calculate L_1/V_1 and L_n/V_n.

6. Use Horton/Franklin method to estimate tower temperatures. This is shown in Table 9.2 (Reference 1).

7. Obtain top and bottom K values.

8. Calculate absorption factors for each component i at the top and bottom

$$A_{1i} = L_1/K_{1i}V_1 \tag{9.3}$$

$$A_{ni} = L_n/K_{ni}V_n \tag{9.4}$$

For stripping factors

$$S_{Ti} = V_1 K_{1i}/L_1 \tag{9.5}$$

$$S_{Bi} = V_n K_{ni}/L_n \tag{9.6}$$

9. Obtain A_{ei} from

$$A_e = [A_B(A_T + 1) + 0.25]^{1/2} - 0.5 \tag{9.7}$$

Similarly

$$S_e = [S_T(S_B + 1) + 0.25]^{1/2} - 0.5 \tag{9.8}$$

10. Read E_{ai} values from Figure 9.1.

11. Calculate mols of each component absorbed.

12. Compare to assumed total mols absorbed and reassume lean oil rate if necessary.

Edmister Method (1957)

Edmister has developed an improved procedure (Reference 8) that features equations combining absorption and stripping functions as follows:

$$V_1 = \phi_a V_{n+1} + (1 - \phi_s) L_0 \qquad \text{(Absorption Section)} \tag{9.9}$$

$$L_1 = \phi_s L_{m+1} + (1 - \phi_a) V_0 \qquad \text{(Stripping Section)} \tag{9.10}$$

where

L_1 = Liquid from bottom stripping tray
L_{m+1} = Liquid to top stripping tray
$\phi_a = 1 - E_a$, fraction not absorbed
$\phi_s = 1 - E_s$, fraction not stripped
V_0 = Vapor to bottom stripping tray

Other symbols are defined in Tables 9.1 and 9.2. Figure 9.1 and Equations 9.3, 9.4, 9.5, 9.6, 9.7, and 9.8 are used as before

V_1 and L_1 are found from Equations 9.9 and 9.10. The improved procedure is better for rigorous solution of complicated absorber designs.

Lean Oil

The selection of lean oil for an absorber is an economic study. A light lean oil sustains relatively high lean oil loss, but has the advantage of high mols/gal compared to a heavier lean oil. The availability of a suitable material has a large influence on the choice. A lean oil 3 carbon numbers heavier than the lightest component absorbed is close to optimum for some applications. In natural gas plant operations, however, the author generally sees a lean oil heavier by about 10-14 carbon numbers.

Presaturators

A presaturator to provide lean oil/gas contact prior to feeding the lean oil into the tower can be a good way of getting more out of an older tower. Absorber tray efficiencies run notoriously low. A presaturator that achieves equilibrium can provide the equivalent of a theoretical tray. This can easily equal 3-4 actual trays. Some modern canned computer distillation/absorption programs provide a presaturator option.

Inorganic Absorption

Design of this type absorber quite often involves a system whose major parameters are well defined such as system film control, mass transfer coefficient equations, etc. Reference 1 gives design data for certain well known systems such as NH_3-Air-H_2O, Cl_2-H_2O, CO_2 in alkaline solutions, etc. Likewise, data for commercially available packings is well documented such as packing factors, *HETP*, *HTU*, etc. Packing parameters are discussed in Chapter 8.

Film Control

The designer needs to know whether his system is gas or liquid film controlling. For commercial processes this is known.

In general an absorption is gas film controlling if essentially all resistance to mass transfer is in the gas film. This happens when the gas is quite soluble in, or reactive with the liquid. Reference 1 gives a listing of film control for a number of commercial systems. If a system is essentially all gas or liquid film controlling, it is common practice to calculate only the controlling mass transfer coefficient. Reference 5 states that for gas absorption, the gas mass transfer coefficient is usually used, and for stripping the liquid mass transfer coefficient is usually used.

Mass Transfer Coefficients

General equations for mass transfer coefficients are given in various references if specific system values are not available. These must, however, be used in conjunction with such things as packing effective interfacial areas and void fractions under operating conditions for the particular packing selected. It is usually easier to find K_{GA} for the packing used with a specific system than effective interfacial area and operation void fraction. Packing manufacturers' data or references, such as Reference 1, can provide specific system K_{GA} or K_{LA} data. Such data are too voluminous for this small handbook.

If K_{GA} values are available for a known system, those of an unknown system can be approximated by

$$K_{GA}\,(unknown) = K_{GA}\,(known)\left[\frac{D_v\ unknown}{D_v\ known}\right]^{0.56} \qquad \textbf{(9.11)}$$

where

K_{GA} = Gas film overall mass transfer coefficient, lb mols/hr (ft^3) *(ATM)*

D_v = Diffusivity of solute in gas, ft^2/hr

Diffusivities

The simplest gas diffusivity relationship is the Gilliland (Reference 1).

$$D_v = 0.0069 \frac{T^{3/2}(1/M_A + 1/M_B)^{1/2}}{P(V_A^{1/3} + V_B^{1/3})^2} \qquad (9.12)$$

where

T = Absolute temperature, °R
M_A, M_B = Molecular weights of the two gases, *A* and *B*
P = Total pressure, *ATM*
V_A, V_B = Molecular volumes of gases

Height of Overall Transfer Unit

Transfer unit heights are found as follows:

$$H_{OG} = \frac{G_m}{K_{GA} P_{AVE}} \qquad (9.13)$$

$$H_{OL} = \frac{L_m}{K_{LA} \rho_L} \qquad (9.14)$$

where

H_{OG}, H_{OL} = Height of transfer unit based on overall gas or liquid film coefficients, ft
G_m, L_m = Gas or liquid mass velocity, lb mols/(hr) (ft^2)
K_{GA}, K_{LA} = Gas or liquid mass transfer coefficients, consistent units
P_{AVE} = Average total pressure in tower, *ATM*
ρ_L = Liquid density, lb/ft^3

Number of Transfer Units

For dilute solutions the number of transfer units N_{OG} is obtained by

$$N_{OG} = \frac{Y_1 - Y_2}{\dfrac{(Y - Y^*)_1 - (Y - Y^*)_2}{\ln \dfrac{(Y - Y^*)_1}{(Y - Y^*)_2}}} \qquad (9.15)$$

where

$(Y - Y^*)$ = Driving force, expressed as mol fractions
Y = Mol fraction of one component (Solute) at any point in the gas phase
Y^* = Mol fraction gas phase composition in equilibrium with a liquid composition, X
X = Mol fraction in the liquid at the same corresponding point in the system as Y
$1, 2$ = Inlet and outlet of the system, respectively

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10

SEPARATORS AND ACCUMULATORS

Liquid Residence Time

Tables 10.1, 10.2, and 10.3 give various rules of thumb for approximate work. For final design additional judgement is required for each individual case.

Vapor Residence Time-Vapor/Liquid Separators

This is usually expressed in terms of maximum velocity which is related to the difference in liquid and vapor densities. The standard equation is

$$U_{vapor\ max} = K\,[(\rho_L - \rho_V)/\rho_V]^{1/2} \tag{10.1}$$

where

U = Velocity, ft/sec
ρ = Density of liquid or vapor, lbs/ft^3
K = System constant

Recommendations for K can be found in Tables 10.4 and 10.5.

Table 10.1. Residence Time for Liquids

Service (times in minutes)	½ Full (Reference 1)	L_{HL} to L_{LL} (minimum) (Reference 2*)	½ Full (Reference 8)	Miscellaneous
Tower reflux drum	See Table 10.2	5-based on reflux flow	5 to 10-based on total flow	....
Vapor-liquid separators	....	....	3-5	....
Product to storage	Depends on situation	Fractionator O.H. Prod. —2	....	....
Product to heat exchanger along with other streams	....	Fractionator O.H. Prod. —5	....	....
Product to heater	....	Fractionator O.H. Prod. —10	....	....
Furnace surge drums	....	....	....	10 min. 20 max.
Tower bottoms FRC control	....	....	....	5 min. 10 max.
LC control	....	....	....	3 min.

*This article deals only with reflux drums. Use only the larger vessel volume determined. Do not add two volumes such as reflux plus product. If a second liquid phase is to be settled, additional time is needed. For water in hydrocarbons, an additional 5 minutes is recommended.

Table 10.5 applies to free-flowing entrained liquids being separated by a standard 12 lb/ft^3 wire mesh York Demister.® Demister® is the registered trademark of the Otto H. York Co. for its mist eliminators and entrainment separators.

Pressure Drop Across Mist Eliminator

Use 1″ H_2O pressure drop.

Pressure Drop Entering Plus Leaving Vessel

One velocity head for inlet and one half for outlet pipe velocity is close.

Table 10.2. Liquid Residence Rules of Thumb for Reflux Drums

½ Full Minutes (Reference 1)

Factor	Reflux Control	Instrument Factor		Labor Factor*		
Base reflux drum rules of thumb		w/ Alarm	w/o Alarm	G	F	P
	FRC	½	1	2	3	4
	LRC	1	1½	2	3	4
	TRC	1½	2	2	3	4

The above are felt to be conservative. For tight design cut labor factors by 50%. Above based on gross overhead.

	Situation	Factor
Multipliers for overhead product portion of gross overhead depending on operation of external equipment receiving the overhead product	Under good control	2.0
	Under fair control	3.0
	Under poor control	4.0
	Feed to or from storage	1.25

	Situation	Factor
Multipliers for gross overhead if no board mounted level recorder	Level indicator on board	1.5
	Gauge glass at equipment only	2.0

*Labor factors are added to the instrument factors G = good, F = fair, P = poor.

Vessel Required Thickness (T), in

Equation 10.2 gives the required vessel thickness based on inside vessel radius. Equation 10.3 gives the thickness based on outside radius.

$$T = \frac{Pr_i}{SE - 0.6P} + C \qquad (10.2)$$

$$T = \frac{Pr_o}{SE + 0.4P} + C \qquad (10.3)$$

where

Table 10.3. Liquid Residence Rules of Thumb for Proper Automatic Control of Interface Level (References 8 and 9)

Flow (gpm)	Reservoir Capacity (gal/in. of depth)
100	2
200	3
600	4
800	8
1,000	10
1,500	15
2,000	24

Table 10.4. *K* Values Based on Flow Ratio (with mist eliminators)

Separation Factor •$W_L/W_V \sqrt{\rho_V/\rho_L}$ (Reference 1)	Vertical at 85% Flooding	Horizontal $K_H = 1.25\,K_{vertical}$
.006	0.25	0.31
.008	0.30	0.37
.01	0.33	0.41
.02	0.40	0.50
.04	0.44	0.55
.06	0.44	0.55
.08	0.43	0.54
0.1	0.42	0.52
0.2	0.37	0.46
0.4	0.29	0.36
0.6	0.22	0.27
0.8	0.18	0.22
1.0	0.16	0.20
2	0.075	0.094
4	0.033	0.041
6	0.018	0.022

•Where *W* is the flow rate of vapor or liquid in lb/sec.

P = Pressure, psig
r = Radius, in.
S = Allowable stress, psi
E = Weld efficiency, fraction.
Use 0.85 for initial work.
C = Corrosion allowance, in.

Table 10.5. *K* Values Based on Disengaging Height (with mist eliminators)

Disengaging Height, in. (Reference 7)	Allowable *K* Value
3	0.12
4	0.15
5	0.19
6	0.22
7	0.25
8	0.29
9	0.32
10	0.35
11	0.38
12	0.40
13	0.42
14	0.43

Typical Vapor-Liquid Separator Calculation Method
(Reference 1)

Vertical Drum

This method uses the separation factor in Table 10.4.

1. Calculate separation factor $= W_L/W_V \sqrt{\rho_V/\rho_L}$
2. Look up K_V in Table 10.4.
3. Calculate $U_{vapor\ max}$ from Equation 10.1.
4. Calculate minimum vessel cross section.
5. Set diameter based on 6-inch increments.
6. Approximate the vapor-liquid inlet nozzle based on the following criteria:

$$(U_{max})\ \text{nozzle} = 100/\sqrt{\rho_{mix}},\ \text{ft/sec} \qquad (10.4)$$

$$(U_{min})\ \text{nozzle} = 60/\sqrt{\rho_{mix}},\ \text{ft/sec} \qquad (10.5)$$

7. Sketch the vessel. For height above center line of feed nozzle to top seam, use 36″ + ½ feed nozzle OD or 48″ minimum. For distance below center line of feed nozzle to maximum liquid level, use 12″ + ½ feed nozzle OD or 18″ minimum.

8. Select appropriate full surge volume in seconds. Calculate the required vessel volume.

$$V = Q_L \text{ (design time to fill), ft}^3 \tag{10.6}$$

where

Q_L = Liquid flow, ft^3/sec

9. Liquid height is

$$H_L = V\,(4/\Pi D^2),\ \text{ft} \tag{10.7}$$

10. Check geometry. Keep

$$(H_L + H_V)/D \tag{10.8}$$

between 3 and 5, where H_V is vapor height in feet.

For small volumes of liquid, it may be necessary to provide more liquid surge than is necessary to satisfy the $L/D > 3$. Otherwise this criteria should be observed. If the required liquid surge volume is greater than that possible in a vessel having $L/D < 5$, a horizontal drum must be provided.

Horizontal Drum

This method is a companion to the vertical drum method based on Reference 1.

1. Calculate separation factor.
2. Look up K_H.
3. Calculate $U_{vapor\ max}$ from Equation 10.1.
4. Calculate required vapor flow area.

$$(A_V)_{min} = Q_V / U_{vapor\ max},\ \text{ft}^2 \tag{10.9}$$

5. Select appropriate design surge time and calculate full liquid volume. The remainder of the sizing procedure is done by trial and error as in the following steps.

6. When vessel is full,

$$(A_{total})_{min} = (A_V)_{min}/0.2 \tag{10.10}$$

$$D_{min} = \sqrt{4(A_{total})_{min}/\Pi}, \text{ ft} \tag{10.11}$$

7. Calculate vessel length.

$$L = \frac{\text{full liquid volume}}{(\Pi/4)\,D^2} \tag{10.12}$$

$$D = D_{min} \text{ to the next largest 6 inches} \tag{10.13}$$

8. If $5 < L/D < 3$, resize.

Quick Rules of Thumb for Gas Scrubbers

Fair Separation

$$G = 900\sqrt{\rho_V(\rho_L - \rho_V)} \tag{10.14}$$

Good Separation

$$G = 750\sqrt{\rho_V(\rho_L - \rho_V)} \tag{10.15}$$

where

G = Allowable mass velocity, lb/hr ft^2
ρ = Density, lb/ft^3

Reflux Drums

For an excellent optimization method see Reference 2. The following are selected comments from that article·

1. Reflux drums are usually horizontal because the liquid load is important.
2. When a small quantity of a second liquid phase is present, a drawoff pot (commonly called a bootleg) is provided to

make separation of the heavy liquid (frequently water) easier. The pot diameter is ordinarily determined for heavy phase velocities of 0.5 ft/min. Minimum length is 3 ft for level controller connections. Minimum pot diameter for a 4- to 8-foot diameter reflux drum is 16 inches. For reflux drums with diameters greater than 8 feet, pot diameters of at least 24 inches are used. The pot must also be placed at a minimum distance from the tangent line that joins the head with the body of the vessel.

3. The minimum vapor clearance height above high liquid level is 20% of drum diameter. If possible this should be greater than 10 inches.

Liquid-Liquid Separators

R. L. Barton ("Sizing Liquid-Liquid Phase Separators Empirically," *Chemical Engineering,* July 8, 1974, Copyright© (1974) McGraw-Hill, Inc., used with permission) provides the following quick method for sizing liquid-liquid phase separators empirically.

The separation of mixtures of immiscible liquids constitutes one of the important chemical engineering operations. This empirical design has proven satisfactory for many phase separations.

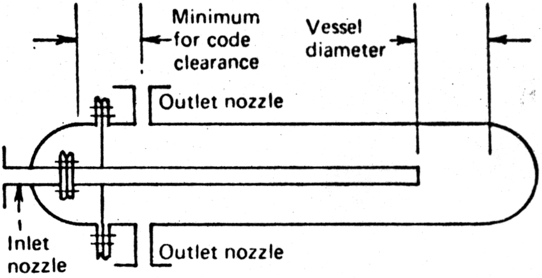
1. Calculate holdup time with the formula

$$T = 0.1\,[\mu/(\rho_b - \rho_t)]$$

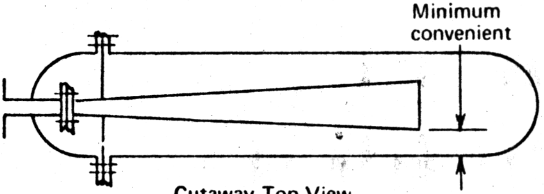
where

T = holdup time, hours
μ = viscosity of the continuous phase, cp
ρ_b = sp. gr. of the bottom phase
ρ_t = sp. gr. of the top phase

2. Assign a length-to-diameter ratio of 5, and size a tank to accommodate the required holdup time.
3. Provide inlet and outlet nozzles at one end, and an internal flat cone (see Figure 10.1).



Cutaway Side View



Cutaway Top View

Notes:

1. For large vessels, a manway and internally dismantled cone may be used where more economical.
2. Inlet and outlet nozzles sized for pump discharge.
3. Gage glass and level instruments to be located at inlet-outlet end.
4. Mechanical design to suit for economy under operating conditions.

Figure 10.1. *This is a recommended design for a liquid-liquid separator (Reference 5).*

While this design procedure is empirical, there is some rationale behind it. The relation between viscosity and specific-gravity-difference of the phases corresponds to those of the equations for terminal settling velocity in the Stokes-law region and free-settling velocity of isometric particles. Also, the dimensions of the tank and cone recognize that the shape of turbulence created by nozzles discharging into liquids spreads at an angle whose slope is about 1 to 5.

This design is not good for emulsions.

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11

TANK BLENDING

This chapter discusses a simple low cost tank blending method. It is referred to as a shrouded blending nozzle system, and it works quite well. The shroud causes the jet nozzle to educt a large quantity of surrounding fluid, improving blending. A simplified diagram of the nozzle portion is shown in Figure 11.1.

Make the nozzle point upward at an angle such that a straight line projected from the nozzle would hit the liquid surface ½ to ⅔ of the way across the tank diameter. The idea is to promote top to bottom turnover. However, tilt the nozzle slightly to the left to promote a slight swirl effect. Aim the nozzle at a point about ⅓ of a radius off-center.

The following rules of thumb apply primarily to the situation where a final tank requires blending after, and perhaps during, transfer from rundown tanks.

1. Base tank size 10,000 barrels.
2. Use about 25 hp circulating pump.
3. Provide roughly 50-75 feet of head or a little higher.
4. For size other than 10,000 barrels, ratio directly for horsepower.
5. Circulation rate in GPM can be calculated from horsepower and head.

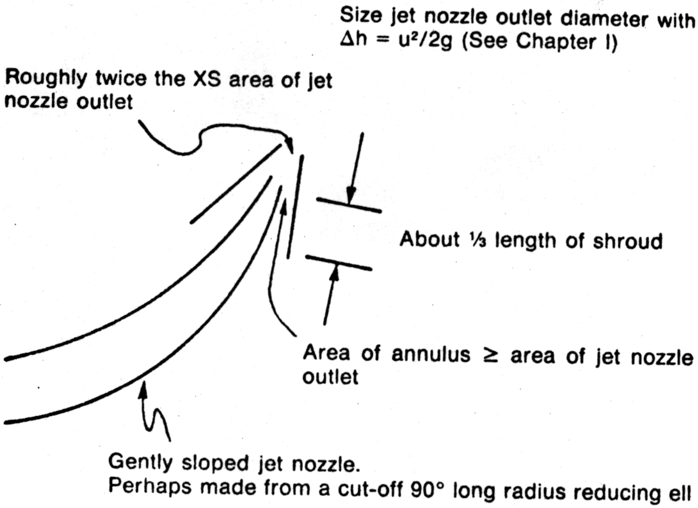

Figure 11.1. *Blending can be improved by a shrouded blending nozzle system because the jet nozzle educts a large quantity of surrounding fluid.*

Often in refrigerated storage light hydrocarbons are involved which have a large gravity change with temperature. This condition makes such storage relatively easy to stratify. Be sure not to put in too large a blending pump for refrigerated storage. Not only does it waste blending pump horsepower, but extra heat is added to the fluid that must be removed by the refrigeration system (a double penalty). Operators often tend to leave blending systems running full time.

Provide a means of filling the tank to operating level prior to operating the blending nozzle system. Damage could result from discharging the high velocity jet into an empty tank.

When circulating the blending system and running down into the tank at the same time, it may be possible to direct the rundown stream into the circulating pump suction for additional blending in the pump.

12
UTILITY SYSTEMS

Cooling Water Systems

To determine cooling water system flows use a heat and material balance and a chloride balance (concentration ratio is usually calculated from chloride concentrations).

$$D = C\ (500)\ \text{lb/hr}\ (1)\ \text{Btu/lb °F}\ (\Delta T)\ \text{°F} \qquad (12.1)$$
$$= 500\ C\Delta T\ \text{Btu/hr}$$

$$E \approx \frac{D\ \text{Btu/hr}}{1000\ \text{Btu/lb}\ (500)\ \text{lb/hr/GPM}} = \frac{C\Delta T}{1000} \qquad (12.2)$$

$$CR = \frac{Cl^-_B}{Cl^-_M};\ \text{also}\ MCl^-_M = BCl^-_B \qquad (12.3)$$

so

$$CR = M/B \qquad (12.4)$$
$$M = E + B\ \text{(overall material balance)} \qquad (12.5)$$
$$E = M - B = CR\,(B) - B = B\,(CR\text{-}1) \qquad (12.6)$$

so

Table 12.1. Windage Loss

Type of Cooling Device	Windage Loss As Percentage of System Circulating Rate
Spray pond	3
Atmospheric cooling tower	0.7
Mechanical draft cooling tower	0.2
(Drift eliminators may do better than 0.2)	

$$B = \frac{E}{CR - 1} \tag{12.7}$$

$$M = E\left(\frac{CR}{CR - 1}\right) \tag{12.8}$$

Use Equation 12.2 to get E, then Equation 12.7 to get B, and finally Equation 12.5 or 12.8 to get M.

where

D = Cooling system duty, Btu/hr
C = System circulation rate, GPM
ΔT = Cooling system temperature difference (hot return water minus cold supply water), °F
E = Cooling system evaporation rate, GPM
CR = Cooling system concentration ratio
Cl^- = Chloride concentration in the makeup or blowdown
M = Cooling system makeup rate, GPM
B = Cooling system total blowdown, GPM. This includes both planned blowdown plus cooling system windage (or drift) losses (of course any system leakage counts as part of "planned" blowdown)

To determine the required amount of planned blowdown, subtract windage losses from B. Use Table 12.1 for windage losses in lieu of manufacturer's or other test data.

When cooling systems are treated, chemicals are sometimes added in shots rather than continuously. Equation 12.9 gives a chemical's half life in a cooling system:

Table 12.2 Comparison of P and M Alkalinities

	OH^-	$CO_3^=$	HCO_3^-
P = O	O	O	M
P = M	M	O	O
P = M/2	O	M	O
P < M/2	O	2P	M-2P
P > M/2	2P-M	2(M-P)	O

$$T_{½} = \frac{S}{B}(.693) \tag{12.9}$$

where

$T_{½}$ = Half life, min.
S = System capacity, gal

Water Alkalinity*

Most water analysis results are rather easily interpreted. However, two simple and useful tests need explanation. These are the *P* and *M* alkalinity. The water is titrated with *N*/30 HCl to the phenolphthalein endpoint at pH 8.3. This is called the *P* alkalinity. Similar titration to the methyl orange end point at pH 4.3 is called the *M* alkalinity. They are reported as ppm $CaCO_3$.

This applies to waters having alkalinity caused by one or all of the following:

1. Bicarbonate (HCO_3^-)
2. Carbonate ($CO_3^=$)
3. Hydroxide (OH^-)

In natural waters the alkalinity is usually caused by bicarbonate. Carbonate or hydroxide is rarely encountered in untreated water. The *M* alkalinity equals the sum of all three forms of alkalinity. The *P* alkalinity equals ½ the carbonate and all the hydroxide alkalinity. Table 12.2 shows what one can deduce from the *P* and *M* alkalinity.

Table 12.3. ABMA Limits For Boiler Water

Boiler Pressure (psig)	Total Solids (ppm)	Alkalinity (ppm)	Suspended Solids (ppm)	Silica (ppm)
0-300	3500	700	300	125
301-450	3000	600	250	90
451-600	2500	500	150	50
601-750	2000	400	100	35
751-900	1500	300	60	20
901-1000	1250	250	40	8
1001-1500	1000	200	20	2.5
1501-2000	750	150	10	1.0
over 2000	500	100	5	0.5

Table 12.4. Target Excess Oxygen

	% Excess Oxygen	
Situation	Gas Firing	Oil Firing
Portable analyzer weekly check	5.0	6.0
Permanently mounted oxygen recorder	4.5	5.0
Oxygen recorder with remote manual damper control	4.0	4.5
Automatic damper control	3.0	4.0

Boiler Blowdown Control

The American Boiler Manufacturers' Association (ABMA) has established limits for boiler water composition. The limits are set to help assure good quality steam (for example limiting silica in the steam to 0.02-0.03 ppm). Blowdown is normally based on the most stringent of these limits shown in Table 12.3

Excess Air for Firing

References 4 and 5 give target excess oxygen to shoot for as a guide in heater efficiency improvement. Table 12.4 summarizes the recommended targets.

In an operating plant the air rate can be adjusted at fixed heat output (constant steam rate for a boiler) until minimum fuel rate is achieved. This is the optimum so long as the warnings below are heeded.

Woodward, References 4 and 5, warns that one must not get so low on excess oxygen that combustibles can get in the flue gas. This can quickly lose the efficiency that one was trying to gain plus pose a safety hazard. In oil firing a fuel rich condition can be detected by smoking. In gas firing, however, substantial loss from unburned combustibles can occur before smoking is seen. Woodward believes that below 3 percent excess oxygen the percentage of combustibles in the flue gas should be monitored.

Reference 6 speaks of controlling the excess air at 10% for plants having highly variable fuel supplies by using gas density to control air/fuel ratio. Reference 6 gives the following equations relating theoretical air and density.

$$\text{Fuel gas/air ratio} = 1 : 14.78\ (1 + 0.0921/r_d)\ \text{mass/mass} \tag{12.10}$$

$$\text{Fuel oil/air ratio} = 1 : 0.115\ (X + 3Y)\ \text{mass/mass} \tag{12.11}$$

where

r_d = Density relative to air; $M.W./M.W.$ air
X = % Carbon
Y = % Hydrogen

Equation 12.10, the fuel gas equation, does not hold for unsaturated hydrocarbons, but for small percentages of unsaturates the error is not serious.

High flame temperature and high excess air increase NO_x emissions.

Process Efficiency

With high fuel costs the search is on for processes with higher thermal efficiency and ways to improve efficiencies of existing processes. One process being pushed lately for its high efficiency is the gas turbine "combined cycle." The gas turbine exhaust heat makes steam in a waste heat boiler. The steam drives turbines, maybe even a gas turbine steam helper. References 10, 11, and 12 treat this subject and mention alternate equipment hookups, some in conjunction with coal gasification plants.

Reference 2 is a well written report which discusses power plant coal utilization in great detail. It gives a thermal efficiency of 80-83% for modern steam generation plants and 37-38% thermal efficiency for modern power generating plants at base load (about 70%). A modern base load plant designed for about 400 MW and up will run at steam pressures of 2400 or 3600 psi and 1000 °F with reheat to 1000 °F and regenerative heating of feedwater by steam extracted from the turbine. A thermal efficiency of 40% can be had from such a plant at full load and 38% at high annual load factor. The 3600 psi case is supercritical and is called a once-through-boiler since it has no steam drum. Plants designed for about 100-350 MW run around 1800 psi and 1000 °F with reheat to 1000 °F. Below 100 MW a typical condition would be about 1350 psi and 950 °F with no reheat. Reference 2 states that below 60% load factor efficiency falls off rapidly and that the average efficiency for all steam power plants on an annual basis is about 33%.

For any process converting heat energy to mechanical efficiency the Carnot efficiency is the theoretical maximum. It is calculated as

$$\frac{T_1 - T_2}{T_1} \times 100 \qquad \textbf{(12.12)}$$

where

T_1 = Temperature of the heat source, °R
T_2 = Temperature of the receiver where heat is rejected, °R

Therefore, the efficiency is raised by increasing the source temperature and decreasing the receiver temperature.

The efficiency for a boiler or heater is improved by lowering its stack temperature. The stack minimum temperature is frequently limited by SO_3 gas dew point. References 7, 8, and 9 discuss this important subject. A stack as hot as 400°F (or perhaps higher) can have problems if the SO_3 concentration is high enough. Reference 14 states that SO_3 condensation will produce a blue-gray haze when viewed against a clear blue sky.

A very useful relationship for determining the maximum available energy in a working fluid is

$$\Delta B = \Delta H - T_0 \Delta S \tag{12.13}$$

where

ΔB = Maximum available energy in Btu/lb
ΔH = Enthalpy difference between the source and receiver, Btu/lb. For a typical condensing steam turbine it would be the difference between the inlet steam and the liquid condensate
T_0 = Receiver temperature, °R
ΔS = Entropy difference between the source and receiver, Btu/lb °F

To obtain lb/hr-hp, make the following division:

$$\frac{2545}{\Delta B}$$

Equation 12.13 will yield the same result as the Theoretical Steam Rate Tables (Reference 13). Therefore, this is a handy way of getting theoretical steam rates having only a set of steam tables sans Mollier diagram.

Energy From a Gas Expander

With high energy costs expanders will be used more than ever. A quickie rough estimate of actual expanoer available energy is

$$\Delta H = C_p T_1 \left[1 - \left(\frac{P_2}{P_1} \right)^{(K-1)/K} \right] 0.5 \tag{12.14}$$

where

ΔH = Actual available energy, Btu/lb
C_p = Heat capacity (constant pressure), Btu/lb °F

T_1 = Inlet temperature, °R
P_1, P_2 = Inlet, outlet pressures, psia
$K = C_p/C_v$

To get lb/hr-hp divide as follows:

$$\frac{2545}{\Delta H}$$

A rough outlet temperature can be estimated by

$$T_2 = T_1 \left(\frac{P_2}{P_1} \right)^{(K-1)/K} + \left(\frac{\Delta H}{C_p} \right) \tag{12.15}$$

For large expanders Equation 12.14 may be conservative. A full rating using vendor data is required for accurate results. Equation 12.14 can be used to see if a more accurate rating is worthwhile.

For comparison the outlet temperature for gas at critical flow across an orifice is given by

$$T_2 = T_1 \left(\frac{P_2}{P_1} \right)^{(K-1)/K} = T_1 \left(\frac{2}{K+1} \right) \tag{12.16}$$

The proposed expander may cool the working fluid below the dew point. Be sure to check for this.

Inverted Bucket Steam Traps

It is easy to improperly design a steam trap. The design must work for two circumstances and often a designer will check only one of these. The circumstance often overlooked is as follows: On startup or upset the steam control valve can open wide so that the steam chest (assume for this discussion that we are speaking of a reboiler) pressure rises to full steam line pressure. At a time like this the steam trap downstream pressure can be atmospheric due to process variations or the

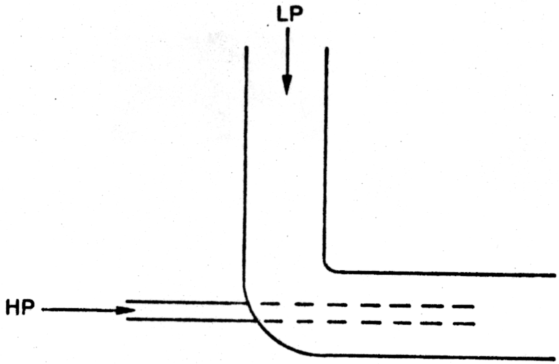

Figure 12.1. *This method is recommended for mixing high and low pressure condensates.*

operators opening the trap discharge to atmosphere in an attempt to get it working.

If the trap orifice has been designed too large, the trap valve cannot open to discharge condensate, creating or amplifying serious plant problems. For any steam trap, for a given trap pressure differential, there is a maximum orifice size above which the bucket can't exert sufficient opening power for the trap to operate. So when designing a trap check manufacturers' data to stay within the maximum sized orifice for full steam line pressure to atmospheric. If a larger orifice is required by the alternate circumstance discussed below, a larger trap body size must be specified whose bucket can service the larger orifice.

The required orifice continuous flow capacity is determined at steam chest pressure to condensate system pressure at a flow 6 to 8 times design. If designed for normal flow the trap would have to be open 100% of the time. Then, as stated above, a body size is selected that can contain the required orifice (not be above the stated maximum for that body size in the manufacturer's sizing tables) at the condition of full steam line pressure to atmospheric. The various vendor catalogs provide sizing charts and tables.

Mixing High and Low Pressure Condensate

Figure 12.1 shows high pressure condensate (small line) being added to low pressure condensate without the usual troublesome "hammer." The high pressure condensate has a chance to cool before emerging into the low pressure condensate line.

Heating Water With Steam

The usual steam in shell versus water in tubes heat exchanger is best designed as follows:

1. Vertical unit.
2. Control valve in outlet condensate.
3. Excess area to allow control of heat balance by varying the level of condensate at the bottom of the tubes.

The above unit is economical, avoids water hammer, has excellent control, and is easy to start up. Contrary to popular belief, heating water with steam is not a highly straightforward design problem. This unit eliminates problems with other designs.

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13

PRESSURE DROP IN A CATALYST BED

Equation 13.1 is a method developed by Girdler Chemical, Inc. for calculating the pressure drop in a catalyst bed. The author has verified it in actual plant operation.

$$\Delta P = 6.58 \times 10^{-10} \times R \times G^{1.9} \times Q \times L/(D_p^{1.1} \times \rho) \qquad \textbf{(13.1)}$$

where

ΔP = Pressure drop, psi
R = Roughness factor. (Smooth surfaces such as glass = 1.0, very rough surfaces = 2.3, Girdler G-56, G-3 = 1.5)
G = Mass flow, lb/hr/ft^2 of open cross section
L = Bed depth, ft
ρ = Gas density, lbs/ft^3 based on the average of inlet and outlet temperatures, pressures, and volumes.
$D_p = 6\ V/A$
V = Actual volume of catalyst pellet or ring, in.^{3} (For rings, does not include volume of hole.)

Table 13.1. Particle Diameter

D_o	L	D_i	D_p	$D_p^{1.1}$
¾	¾	5⁄16	0.5081	0.475
⅝	⅝	¼	0.4326	0.398
⅝	⅝	5⁄16	0.3750	0.339
⅝	⅜	¼	0.3750	0.34
⅝	⅜	5⁄16	0.3309	0.296
⅝	¼	¼	0.3214	0.286
⅝	¼	5⁄16	0.2884	0.255

A = Total surface area, in.^{2}
$Q = (1 - E_p)/E_p^3$
$E_p = E_s + (1 - E_s)\ (D_i/D_o)^2$
D_i = Inside diameter of pellet, in.
D_o = Outside diameter of pellet, in.
$E_s = 0.373 + 0.308\ (D_o/D_t)$
D_t = Inside tube or reactor diameter, in.

Limitation:

1. $D_pG/12\mu > 200$, μ = Fluid viscosity, lb/ft hr
2. ΔP should not be greater than 10-15% of inlet pressure.

The particle diameter, D_p, can be determined from Table 13.1.

References

1. Information from Girdler Chemical, Inc., Louisville, Kentucky, 40201.

14

LEAST SQUARES TECHNIQUE

Frequently data need correlating with the best straight line or curve on a graph. The least squares technique will generate the required equation.

Table 14.1. Linear Relationships*

Graph Paper On Which The Line is Developed	Basic Equation To Be Developed	Simultaneous Equations To Solve to Get The Constants "A" and "B"
Rectangular	$Y = A + BX$	$\Sigma(Y) = N^{**}A + B\,\Sigma(X)$ $\Sigma(XY) = A\,\Sigma(X) + B\,\Sigma(X^2)$
Semi-log (log on X-axis)	$Y = A + B\ln X$	$\Sigma(Y) = NA + B\,\Sigma(\ln X)$ $\Sigma(Y\ln X) = A\,\Sigma(\ln X) + B\,\Sigma(\ln X)^2$
Semi-log (log on Y-axis)	$\ln Y = A + BX$	$\Sigma(\ln Y) = NA + B\,\Sigma(X)$ $\Sigma(X\ln Y) = A\,\Sigma(X) + B\,\Sigma(X)^2$
Log-Log	$\ln Y = A + B\ln X$	$\Sigma(\ln Y) = NA + B\,\Sigma(\ln X)$ $\Sigma(\ln Y\ln X) = A\,\Sigma(\ln X) + B\,\Sigma(\ln X)^2$

*Equations from p. 84 and 102, Herbert Arkin and Raymond R. Colton, *Statistical Methods*, College Outline Series, Barnes and Noble Div., Harper and Row, New York, 1970.

**N = Number of data points.

Table 14.2. Nonlinear Relationships*

General Type of Curve	Basic Equation To Be Developed	Simultaneous Equations To Solve To Get The Constants "A", "B". and "C"
Second Degree Parabola	$Y = A+BX+CX^2$	$\Sigma(Y) = N^{**}A + B\,\Sigma(X) + C\,\Sigma(X^2)$ $\Sigma(XY) = A\,\Sigma(X) + B\,\Sigma(X^2) + C\,\Sigma(X^3)$ $\Sigma(X^2Y) = A\,\Sigma(X^2) + B\,\Sigma(X^3) + C\,\Sigma(X^4)$
Semi-Log version of above	$\ln Y = A+BX+CX^2$	$\Sigma(\ln Y) = NA + B\,\Sigma(X) + C\,\Sigma(X^2)$ $\Sigma(X\ln Y) = A\,\Sigma(X) + B\,\Sigma(X^2) + C\,\Sigma(X^3)$ $\Sigma(X^2\ln Y) = A\,\Sigma(X^2) + B\,\Sigma(X^3) + C\,\Sigma(X^4)$
Reciprocal Relationship convert to	$Y = \frac{1}{A + BX}$ $\frac{1}{Y} = A + BX$ or $Y^1 = A + BX$ where Y^1 is the reciprocal of Y.	$\Sigma(Y^1) = NA + B\,\Sigma(X)$ $\Sigma(XY^1) = A\,\Sigma(X) + B\,\Sigma(X^2)$

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*Equations from p. 84 and 102 Herbert Arkin and Raymond R. Colton, *Statistical Methods,* College Outline Series, Barnes and Noble Div., Harper and Row, New York, 1970.

**N = Number of data points.

Straight Line

Very often a straight line correlation can be used. If a straight line doesn't develop on rectangular paper one can try semi-log or log-log paper. These cases are shown in Table 14.1.

Miscellaneous Curves

Development of equations for several nonlinear relationships is given in Table 14.2.

References

1. Arkin, H. and Colton, R., *Statistical Methods,* College Outline Series, Barnes and Noble Div., Harper and Row, New York, 1970.

15
WEIGHTED SPRAY

A friend developed a unique spray that holds very constant velocity and consistent spray pattern regardless of flow, therefore, overcoming turndown problems in other spray nozzles. This is very useful in quench or desuperheating service. Figure 15.1 illustrates this unique spray.

Knowing the pressure drop across the area producing spray in lbs/in^2 and the lid contact surface in in.^2 one can calculate the weight in pounds. The pressure drop will be the pressure provided to the tee minus the vessel pressure.

The spray velocity can be calculated from $\Delta h = u^2/2g$, the units of which were previously defined. The velocity in ft/sec divided into the horizontal spray distance in feet gives the horizontal time component. The spray fall over that length of time is

$$S = \tfrac{1}{2} g T^2 \tag{15.1}$$

where

S = Fall distance in feet
T = Time in seconds
$g = 32.2\ \text{ft/sec}^2$

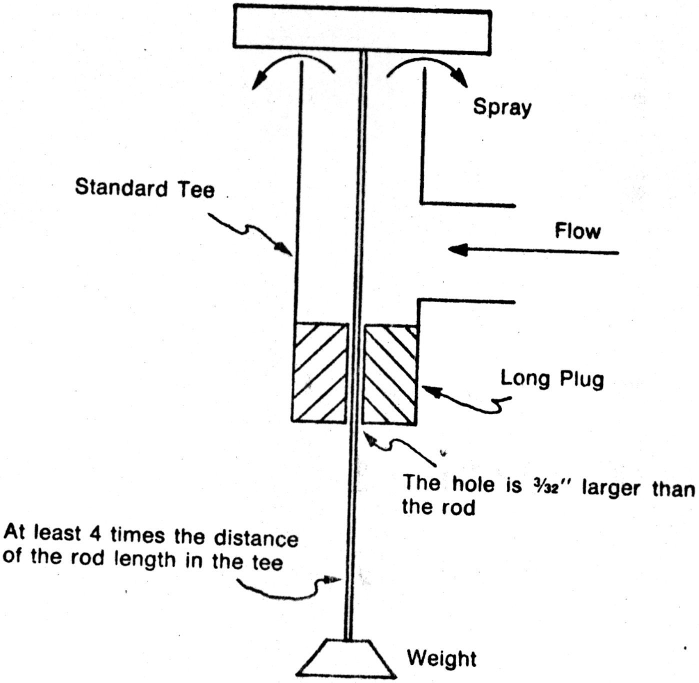

Figure 15.1. *Shown here is the recommended design for a weighted spray.*

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