Chapter – 12 Heat Exchangers for Industrial Refrigeration Systems

Thomas T.S. Wan
温到祥 著
Sept. 3, 2008

This section deals with the heat exchangers used for industrial refrigeration application, such as air cooled condenser, evaporative condenser, water cooled condenser and shell-and-tube evaporators. Other types of heat exchangers such as pipe-coil, unit coolers, product coolers, plate type heat exchanger, coil-and-drum, pipe-in-pipe, air-to-air, fluid-to-gas or gas-to-fluid or oil coolers are not covered in this section.

Condensing Ton and Evaporative Ton:

When dealing with heat transfer, the heat transfer unit is Btu/Hr. Evaporative ton or ton of refrigeration is 12,000 Btu/Hr. However, if the heat transfer ton is referred to the heat rejection from condenser, it is the condensing ton. The condensing ton is not 12,000 Btu/Hr, it is 14,545 Btu/Hr.

Condensing Ton = 1 TR + 1 HP = 12,000 + 2,545 = 14,545 Btu/Hr

The condensing ton is only used for the term of Sq.Ft/ton for water cooled condenser curves. To avoid any confusion, the condensing ton is not used for heat rejection calculation.

System Heat Rejection to Condenser:

Heat rejection = TR x 12000 + BHP x 2545

Heat rejction	=	Btu/Hr
TR	=	Tons of Refrigeration
BHP	=	Compressor BHP at design load

For example: 500 TR, power consumption is 625 BHP
Heat rejection = 500 x 12,000 + 625 x 2545
= 6,000,000 + 1,590,625
= 7,590,625 Btu/Hr

Note: If screw compressor is used and it is with water cooled oil cooling, the heat rejection to condenser should be:

Heat rejection = TR x 12000 + BHP x 2545 – Oil Cooling Heat Removal

Air Cooled Condensers

Basically, air cooled condensers are mostly used for the installation where water is not available or it is economically justified to use air cooled instead of water cooled for large refrigeration installation. In most cases, use of air cooled condenser could result in higher cost and higher power consumption as compared to water cooled condenser project.

Figure 12-1 shows a typical construction of an air cooled condenser. An air cooled condenser consists of a fan section, a coil section, casing and supporting structure. Figure 12-2 shows the belt driving fan assembly for the air cooled condenser. Figure 12-3 is a smaller size air cooled condenser and the Figure 12-4 is a typical larger size air cooled condenser.

The performance of the air cooled condenser is as the following:

$$CT = ADB + TD$$

CT = Condensing Temperature, °F
ADB = Entering Dry Bulb Air Temperature, °F
TD = Approach, °F

The approach for the air cooled condenser is the temperature difference between the design condensing temperature and the design outdoor ambient temperature; smaller approach requires a larger size condenser because more heat transfer coil surface is needed; larger TD permits a smaller air cooled condenser because less heat transfer coil surface is required for the heat rejection.

Figure 12-5 is the typical air cooled condenser performance data sheet. The performance sheet shows the condenser has less heat rejection capacity at a smaller TD and higher heat rejection capacity at a higher TD.

What TD should be used depends on the maximum DWP (design working pressure) of the condensing coil and the refrigerant being used. Most installations use a TD between 15°F to 30°F. The air cooled condenser is getting expensive if the TD is less than 15°F and the DWP of high side is getting too high if the TD is more than 30°F, unless the ambient design dry bulb temperature is low. The power consumption will be higher if the TD is higher for the system; the condenser size is smaller, but the compressor and driving motor is more expensive.

For small and medium size installation, the condenser can have either horizontal fan discharge or vertical discharge. Also, two types of fan draft flow for the condenser: one is Draw Through (Induced Draft) design and the other is Blow Through (Forced Draft) design.

Most large size air cooled condensers are with vertical air discharge. Figure 12-6 shows a typical large air cooled condenser with a forced draft fan; Figure 12-7 is a typical large air cooled condenser with an induced draft fan.

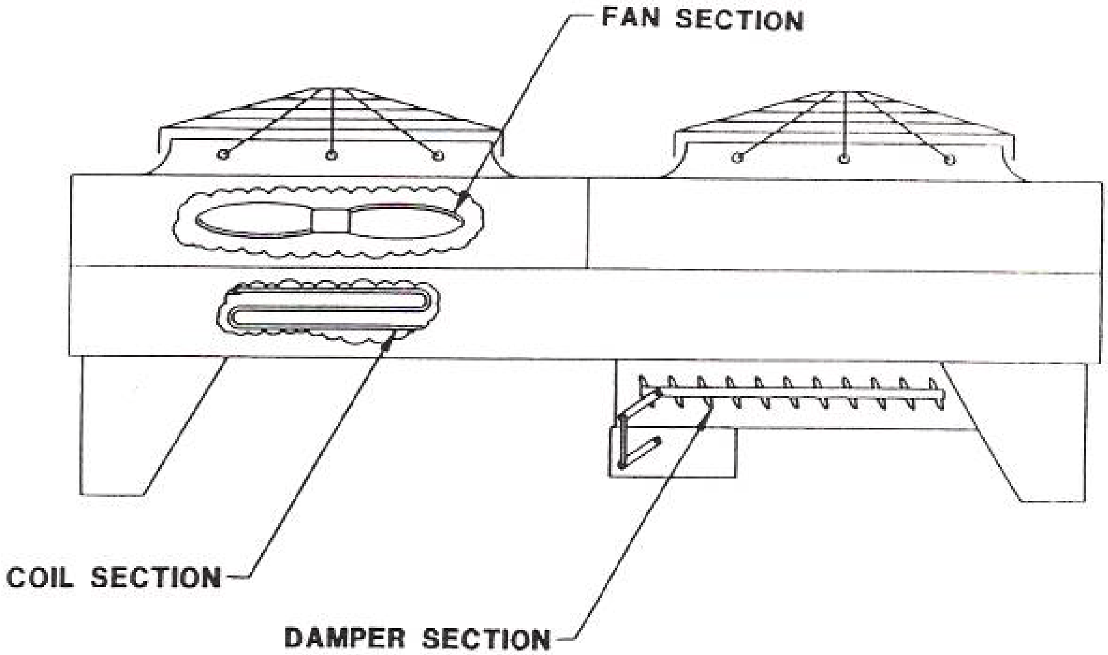


Figure 12-1 Typical Air Cooled Condenser

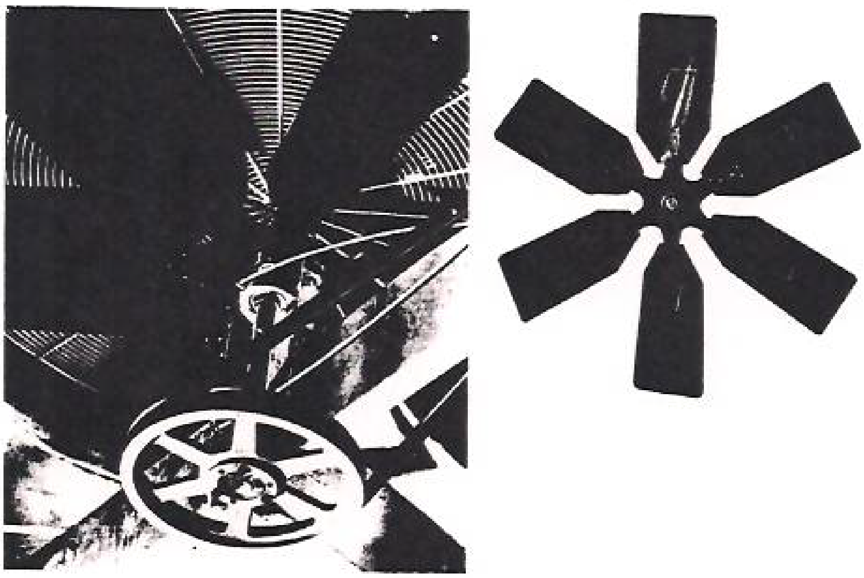


Figure 12-2 Fan Drive Assembly for Air Cooled Condenser

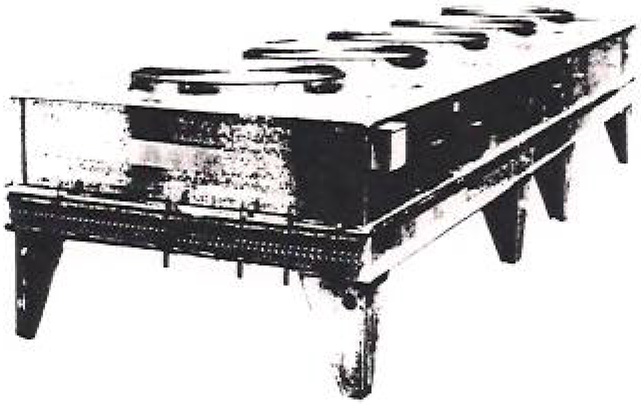


Figure 12-3 Smaller Size Air Cooled Condenser

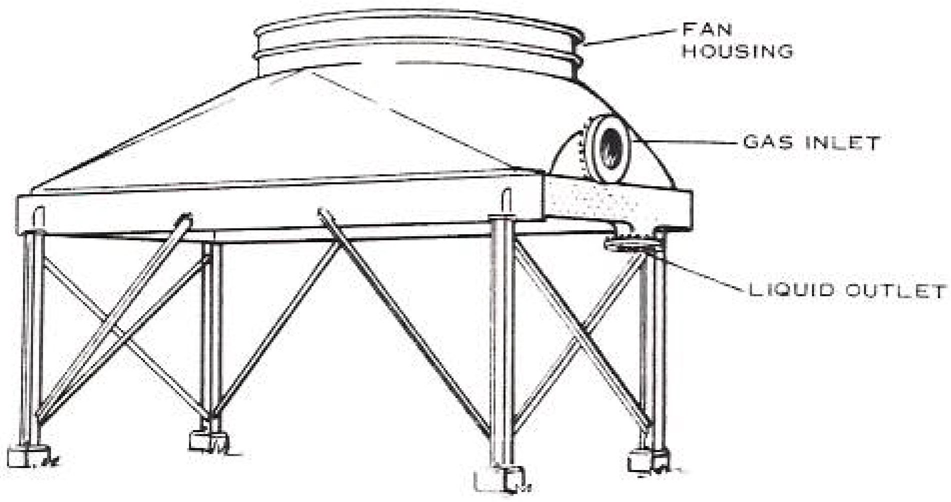


Figure 12-4 Larger Size Air Cooled Condenser

SINGLE CIRCUIT CONDENSERS-REFRIGERANT-12, -22 AND -502
CONDENSER HEAT REJECTION CAPACITY (BTUH)

UNIT SIZE	"TD" (°F) = (CONDENSER CONDENSING TEMP. — ENTERING AIR DRY BULB TEMP.)								
	10	15	20	25	30	35	40	45	50
46	23,350	35,000	46,700	58,250	70,000	81,600	93,400	105,000	116,50
54	27,350	41,000	54,700	68,250	82,000	95,600	109,400	123,000	136,50
61	30,350	45,500	60,700	75,750	91,000	106,100	121,400	131,500	151,50
71	36,000	54,000	72,000	90,000	108,000	126,000	144,000	162,000	180,00
84	42,000	63,000	84,000	105,000	126,000	147,000	168,000	189,000	210,00
93	47,000	70,500	94,000	117,500	141,000	164,500	188,000	211,500	235,00
111	56,000	84,000	112,000	140,000	168,000	196,000	224,000	252,000	280,00
122	61,600	92,500	123,200	154,000	185,000	216,000	246,400	277,500	308,00
142	71,750	107,500	143,500	179,000	215,000	250,500	287,000	322,500	358,00
167	83,750	125,500	167,500	209,000	251,000	293,000	335,000	327,000	418,00
187	93,500	140,000	187,000	233,000	280,000	326,500	374,000	420,000	466,00
218	110,000	165,000	220,000	275,000	330,000	385,000	440,000	495,000	550,00
242	121,500	182,500	243,000	301,000	365,000	426,000	486,000	548,000	608,00
289	145,000	217,500	290,000	362,500	435,000	507,000	580,000	653,000	725,00
339	170,000	255,000	340,000	425,000	510,000	595,000	680,000	765,000	850,00
375	190,000	285,000	380,000	475,000	570,000	665,000	760,000	855,000	950,00
436	221,500	332,500	443,000	554,000	665,000	775,000	886,000	998,000	1,108,00
480	246,500	370,000	493,000	616,000	740,000	864,000	986,000	1,110,000	1,232,00
522	267,000	400,000	534,000	666,000	800,000	934,000	1,068,000	1,200,000	1,332,00
566	290,500	436,000	581,000	726,000	872,000	1,018,000	1,162,000	1,308,000	1,452,00
610	313,000	470,000	626,000	784,000	940,000	1,097,000	1,252,000	1,410,000	1,568,00
636	345,000	490,000	690,000	816,000	980,000	1,142,000	1,380,000	1,470,000	1,632,00
729	363,000	545,000	726,000	907,500	1,090,000	1,270,000	1,452,000	1,635,000	1,815,00
774	384,250	576,000	768,500	960,000	1,152,000	1,345,000	1,537,000	1,730,000	1,920,00
915	451,250	677,500	902,500	1,127,500	1,355,000	1,580,000	1,805,000	2,030,000	2,255,00
982	480,000	720,000	960,000	1,200,000	1,440,000	1,680,000	1,920,000	2,160,000	2,400,00
1135	563,750	845,000	1,127,500	1,407,500	1,690,000	1,970,000	2,255,000	2,535,000	2,815,00
1270	622,500	932,500	1,245,000	1,555,000	1,865,000	2,175,000	2,490,000	2,795,000	3,110,00

Figure 12-5 Typical Performance Data
R-22 Air Cooled Condenser

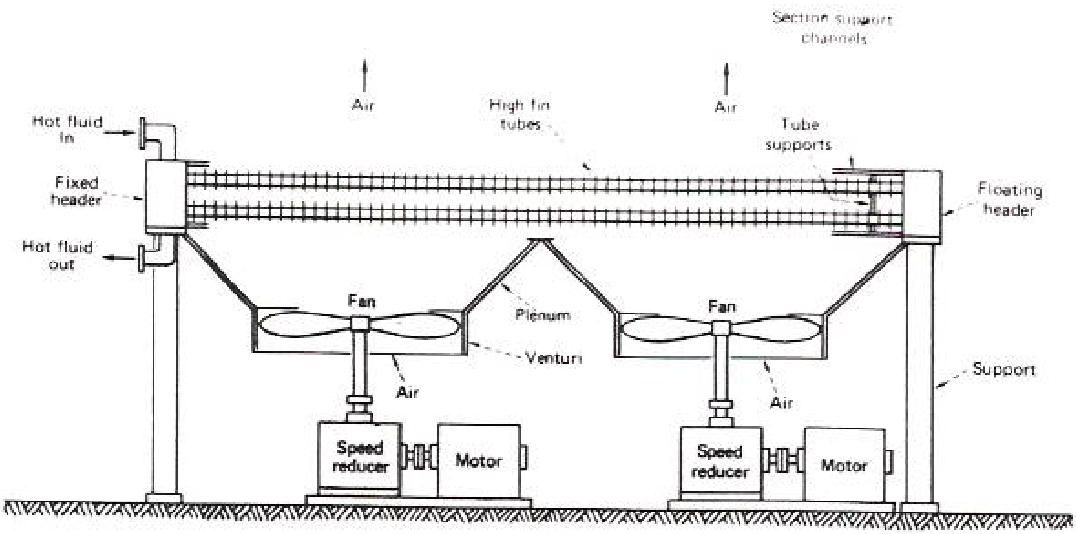


Figure 12-6 Forced Draft Fan Design
Larger Size Air Cooled Condenser

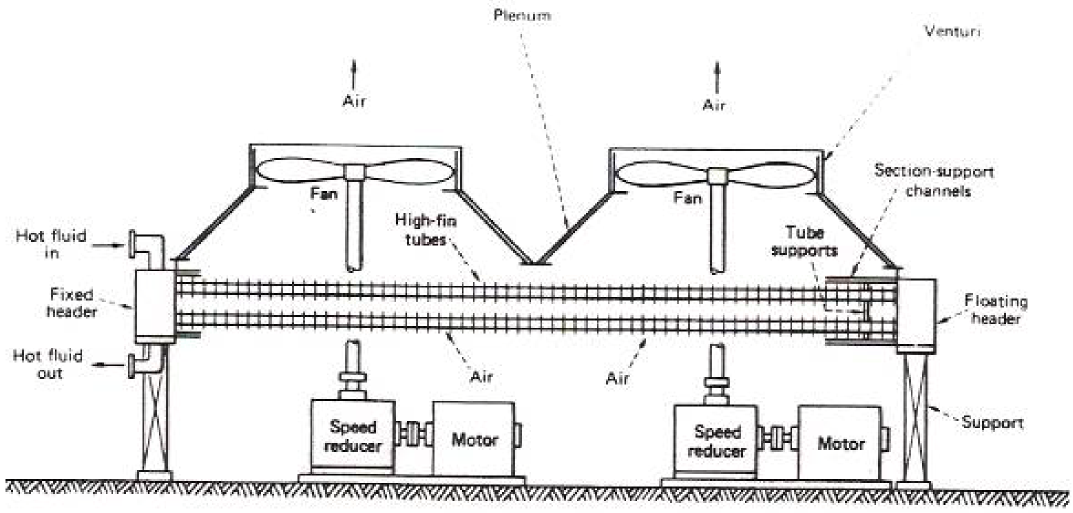


Figure 12-7 Induced Draft Fane Design
Larger Size Air Cooled Condenser

Advantages of Induced Draft Fan:

Less Air Recirculation – The air from induced draft unit exhausts about 2.5 times the velocity of the forced draft unit thereby moving the air further away from the unit, minimize the possibility of recirculation of hot air.

Induced draft units are more efficient – The inlet air more uniformly covers the bottom rows of tubes.

Might require less fan horsepower – Induced Draft unit could move more air at same fan horsepower than Forced Draft unit.

Protect bundles from atmospheric corrosion – The exhaust hoods of the Induced Draft unit cover the top of the units; minimize the amount of rain that drops into the fan ring. It might require less maintenance for corrosion protection.

Better operational heat transfer – Hood cover shields the tube from direct rays of the sun, more uniform cooling.

Minimum data required for air cooled condenser selection and pricing:

The followings are the minimum data required to make an air cooled condenser selection or to make quotation proposal:

1.0 Heat rejection, Btu/Hr.
2.0 Refrigerant.
3.0 Ambient air temperature.
4.0 Condensing temperature.
5.0 Vertical or horizontal arrangement, Blow Through or Draw Through fan.
6.0 Centrifugal, propeller or vane axial fan.
7.0 Type of motor, power supply and code requirements.
8.0 Special coil material requirement, if any.
9.0 Special supporting structure, if any.
10.0 Type of head control, if required.

Head Control Requirement:

Since air cooled condensers are often required to operate over a wide range of ambient air temperatures and variable loading conditions, therefore, air cooled condenser tends to run a very low head pressure when operating in a low ambient air temperature. It is a good practice to take advantage of the falling condensing temperature to save the power consumption. However, the system must maintain a minimum head pressure in order to keep system pressure differential high enough to allow the controls and expansion devices function properly. Therefore, refrigeration application should have head control provision to allow the system operating pressure to vary with the ambient air temperature within certain limits, but without adversely affecting the system operation.

Various Head Controls for Air Cooled Condenser:

A) Fan Cycling Control.

> Individual fan of a multiple fans can be cycled to maintain the condensing temperature of the system. This is most commonly used control for air cooled condenser particularly for small size installations.

B) Face Dampers.

This is good for low ambient temperature installation. Air flow reduction through the condenser coil can be accomplished with closing the face damper. Damper motor(s) is controlled by ambient sensing thermostat or condenser head pressure sensing switch.

C) Flooding-Type Condenser Control and Low Ambient Operation.

Head control valve will partially close the liquid to receiver to flood the condenser coil to maintain the preset pressure for the condenser. An automatic hot gas valve provides pressure differential between compressor discharge and receiver o no less than 30 Psi.

This type of condenser pressure control is effective for cold climates even below 0°F.

The receiver must be sized large enough to hold the extra amount of liquid for the coil flooding.

Low Ambient Bypass Timer - A five minute time delay to bypass the low pressure cutout during low ambient startup.

D) Two-Speed Fan Control or Variable Speed Fan Control.

Two speed fan control or variable fan speed control provides more fine tune of head control for the air cooled condenser.

Variable Fan Pitch Head Control for Large Size Air Cooled Condenser:

In additional to the methods of the head controls available for smaller size air cooled condenser, the other head control available for larger size air condenser is the Automatic Variable Pitch Fan Control. This is to control the pitch of the fan to vary the air volume flow to control the temperature or condensing pressure.

System Shut Down During Cold Ambient:

In case the refrigeration system is shut down during cold ambient, protection against refrigerant migrant in the system should be considered. These protections are such as drain off process fluid in the evaporator if the process fluid will freeze in low ambient; or the refrigerant in the system is to be pumped out into storage receiver or the service valves for the condenser are to be shut off.

Piping and Parallel Multiple Unit Application:

Piping hook-up and the recommendations for parallel multiple units operation are mostly the same as for the evaporative condenser.

Evaporative Condensers

Evaporative condenser is the same as air cooled condenser except water spray is added to wet the entire condensing coil. The water circulation rate for the evaporative condenser is about 1 GPM/Ton.

Most evaporative condensers are with vertical discharge. Fan draft flow is Blow Thru or Induce Draft. Figure 12-8 shows the Blow Thru type evaporative condenser. Figure 12-9 shows the Induce Draft type design. Fans available for evaporative condenser are axial, vane axial or centrifugal.

The performance of an evaporative condenser is rated by °F of Approach.

Deg.F Approach = CT – FWB

CT = Condensing Temperature, °F
FWB = Wetbulb Air Temperature, °F

Therefore:

CT = °F Approach + FWB

For example: Wetbulb Air Temperature is 80°FWB
Approach is 10°F

CT = 10°F + 80°F = 90°F

Figure 12-10 is the typical heat rejection factors for R-22 for various approaches between condensing temperature and air °FWB for a typical evaporative condenser; Figure 12-11 is the typical heat rejection factors for R-717 at various approaches. Those heat rejection factors are to be applied for the determination of the size of the evaporative condenser. Higher the factor, larger size of the evaporative condenser is required. The heat rejection factor will be larger if the approach between the CT and °FWB is smaller.

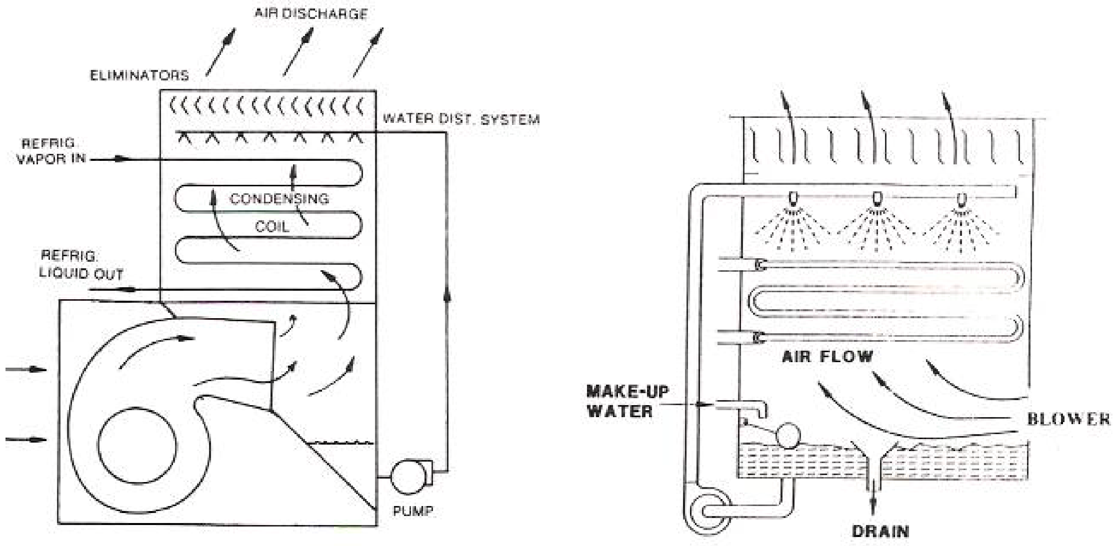
Generally speaking, the minimum approach is about 10°F for most applications. For special application the approach could be 5~6°F; however, the evaporative condenser would be huge and expensive.

Head Control Methods for Evaporative Condenser:

The head controls for evaporative condenser, such as variable speed drive and multiple fan cycling are the same as for the air cooled condenser except the following:

1.0 Automatic Capacity Control Damper control:

(a) Damper installed in the centrifugal fan housings – This is to use the damper to modulate the quantity of air flowing through the unit. When



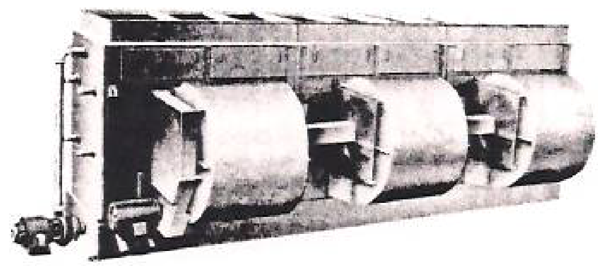
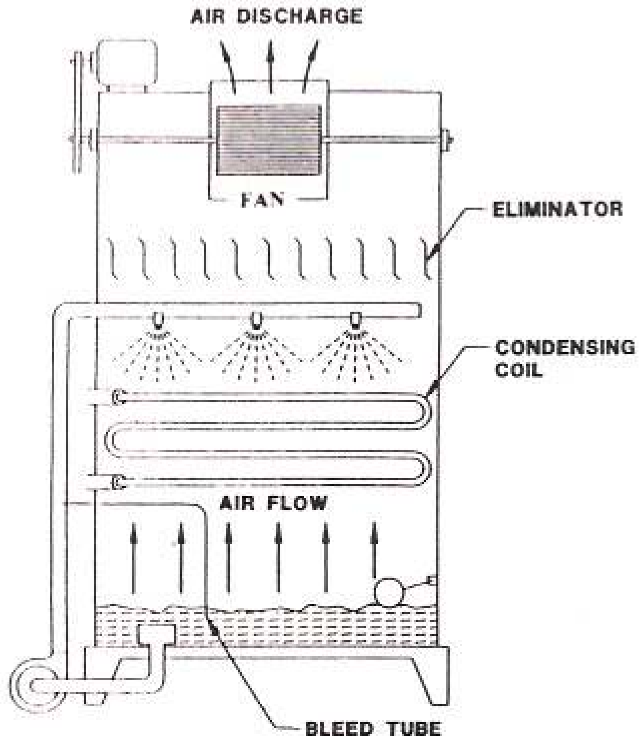


Figure 12-8 Blow Thru Type Evaporative Condenser



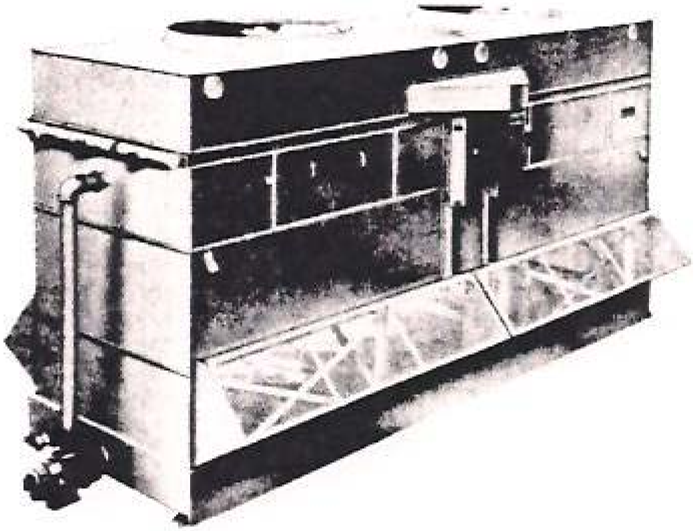


Figure 12-9 Induce Draft Type Evaporative Condenser

Condensing Pressure (PSIG)		Cond. Temp. (°F)	Entering Air Wet Bulb Temperature (°F)											
R-12	R-22		50	55	60	65	68	70	72	75	78	80	85	90
91.8	155.7	85	1.10	1.22	1.39	1.67	1.94	2.13	2.45	2.94	—	—	—	—
99.8	168.4	90	.93	1.02	1.14	1.32	1.47	1.59	1.75	2.00	2.38	2.78	—	—
108.3	181.8	95	.80	.87	.95	1.08	1.16	1.22	1.32	1.45	1.61	1.79	2.56	—
117.2	195.9	100	.71	.76	.82	.89	.93	.98	1.03	1.12	1.23	1.33	1.72	2.50
126.6	210.8	105	.63	.66	.70	.76	.79	.83	.86	.93	1.00	1.05	1.27	1.61
136.4	226.4	110	.56	.59	.62	.66	.70	.71	.75	.79	.84	.88	1.01	1.19
146.8	242.7	115	—	.52	.55	.58	.60	.62	.64	.67	.70	.73	.81	.92
157.7	259.9	120	—	—	—	.51	.53	.54	.55	.57	.60	.62	.68	.75

Figure 12-10 Typical Heat Rejection Factor For Evaporative Condenser, R-22

Cond. Press. (PSIG)	Cond. Temp. (°F)	Entering Air Wet Bulb Temperature (°F)											
		50	55	60	65	68	70	72	75	78	80	85	90
151.7	85	1.00	1.11	1.26	1.52	1.76	1.93	2.23	2.68	—	—	—	—
165.9	90	.85	.93	1.03	1.19	1.33	1.45	1.59	1.82	2.17	2.50	—	—
181.1	95	.73	.79	.87	.98	1.06	1.11	1.19	1.32	1.47	1.61	2.33	—
185.1	96.3	.71	.76	.83	.91	.98	1.04	1.11	1.23	1.36	1.49	2.13	—
197.2	100	.64	.69	.75	.81	.85	.89	.93	1.02	1.12	1.20	1.57	2.27
214.2	105	.57	.60	.64	.69	.73	.76	.79	.84	.91	.96	1.15	1.47
232.3	110	.51	.53	.56	.60	.63	.65	.68	.71	.76	.80	.92	1.08
251.5	115	—	.47	.50	.53	.55	.56	.58	.61	.64	.66	.74	.84
271.7	120	—	—	—	.46	.48	.49	.50	.52	.54	.56	.62	.68

Figure 12-11 Typical Heat Rejection Factor For Evaporative Condenser, R-717

the dampers close to their minimum air flow position, an auxiliary switch will turn off the fan motor.

(b) The discharge hood and damper – The dampers are to be fully open before the fans are running and closed when the fan is off. The damper actuator is controlled by a temperature controller.

2.0 Water Pump Variable Speed Control. And Water Pump On-Off

Reduction of air or water flow would greatly reduce the capacity. Therefore, shutting off the water flow should be done only after other means of control have been used.

The Head Control Function Sequence for evaporative condenser:

On decreasing ambient temperature in winter time:

A. Fan speed reduction.
B. Fans cycling.
C. Damper closing.
D. Pump cycling.

On condensing pressure increasing:

a. Fan speed increase.
b. Fans cycling.
c. Damper opens.
d. Pump cycle on.

Piping Hook-up Suggestion for Single Evaporative Condenser Application:

For single unit of evaporative condenser with single coil, the drain line to receiver can be served as internal equalizer; no external equalizing may be required as shown in Figure 12-12.

However, if liquid drain line is trapped or a surge receiver is used with the evaporative condenser as shown in Figure 12-13, an equalizer line should be used and a liquid trap height (h) should be provided. The liquid drain from the evaporative condenser should be sized for a maximum velocity of 100 Ft/Min. A higher liquid column should be provided if a maximum velocity of 150 Ft/Min and a valve is used for the drain line:

Maximum Velocity of Drain Line, Ft/Min	Valve Between Evaporative Condenser and Receiver	Liquid Column (h), Inches
100 Ft/Min	None, Angle or Globe	14 Inches
150 Ft/Min	None	14 Inches
150 Ft/Min	Angle	16 Inches
150 Ft/Min	Globe	28 Inches

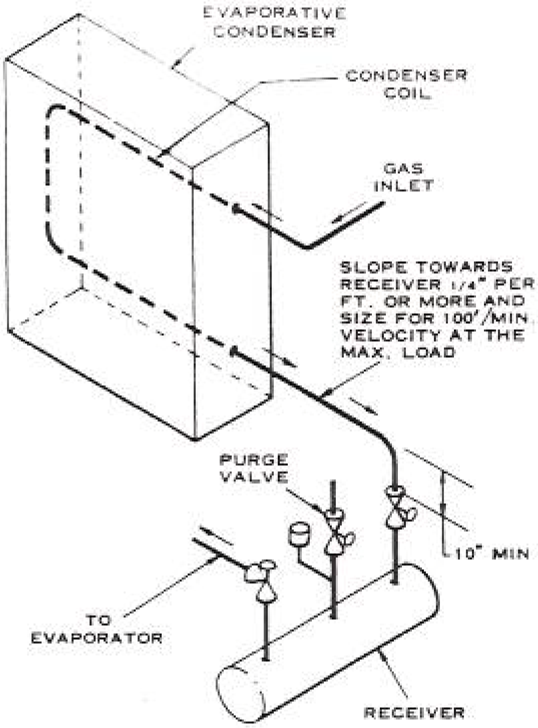


Figure 12-12 Single Coil Evaporative Condenser Top Inlet to Receiver

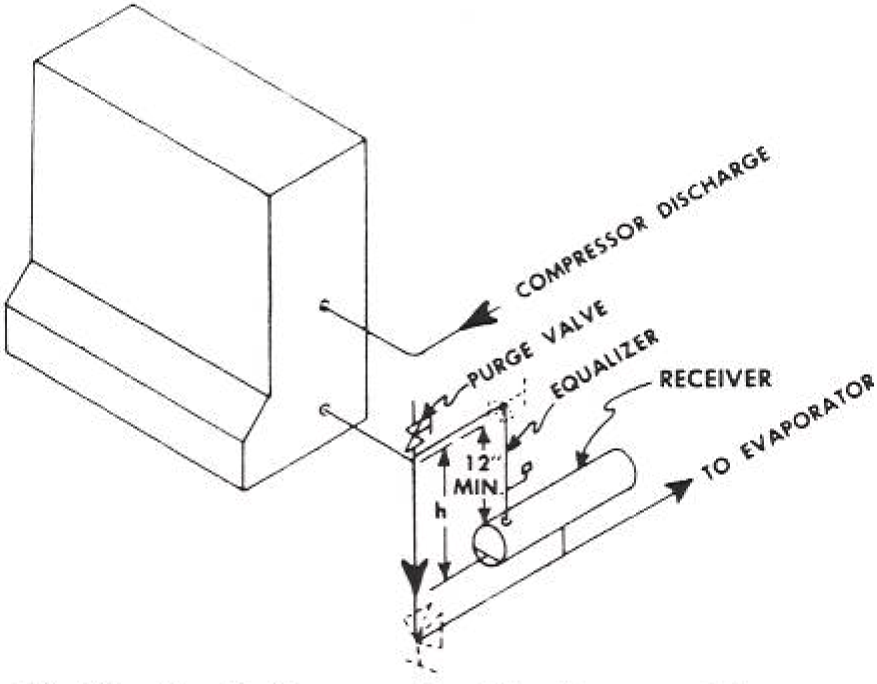


Figure 12-13 Single Evaporative Condenser with Surge Receiver

Liquid Trap Column for Multiple Parallel Units Operation:

Liquid trap column is important for multiple coils or multiple evaporative condenser application when one or more condenser is to be shut off during operation. Figure 12-14 shows one condenser is operating while the other condenser is idling. The pressure at the outlet of the idle condenser is the same as inlet pressure. The liquid shall back up into and flooding the operating condenser coil if no trap liquid column is provided in this case.

Assuming the refrigerant is Ammonia (R-717), CT is 95°F. The liquid column needed to off set the pressure unbalance under this operating condition is calculated as the following:

Condensing Temperature:	95°F
Condensing Pressure:	195.9 Psia
Liquid Density:	36.68 Lbs/Ft^3
Valve P.D.:	0.30 Psi
Pressure after valve:	195.6 Psia
Coil Pressure Drop:	0.8 Psi
Liquid pressure leaving coil:	195.6 – 0.8 = 194.8 Psia
Total pressure differential:	195.9 – 194.8 = 1.1 Psi

$$\text{Pressure Diff. Psi} = \frac{\text{Liquid Density }(Lbs/Ft^3) \times \text{Column (Ft)}}{144}$$

$$1.1 \text{ Psi} = \frac{36.68\ (Lbs/Ft^3) \times \text{Column (Ft)}}{144}$$

$$\text{Column (Ft)} = \frac{1.1 \times 144}{36.68} = 4.32 \text{ Ft}$$

$$= 52 \text{ Inches liquid column of R-717}$$

From the above calculation, 52” liquid trap column is required to balance the pressure drop through the condenser coil and the valve for the R-717 at 95°F CT. The liquid column required for the various common refrigerants for each Psi PD are as the following:

Refrigerant	Liquid Trap Column
R-717 Ammonia	48” for each Psi P.D.
R-290 (Propane)	58” for each Psi P.D.
R-1270 (Propylene)	57” for each Psi P.D.
R-22	22” for each Psi P.D.

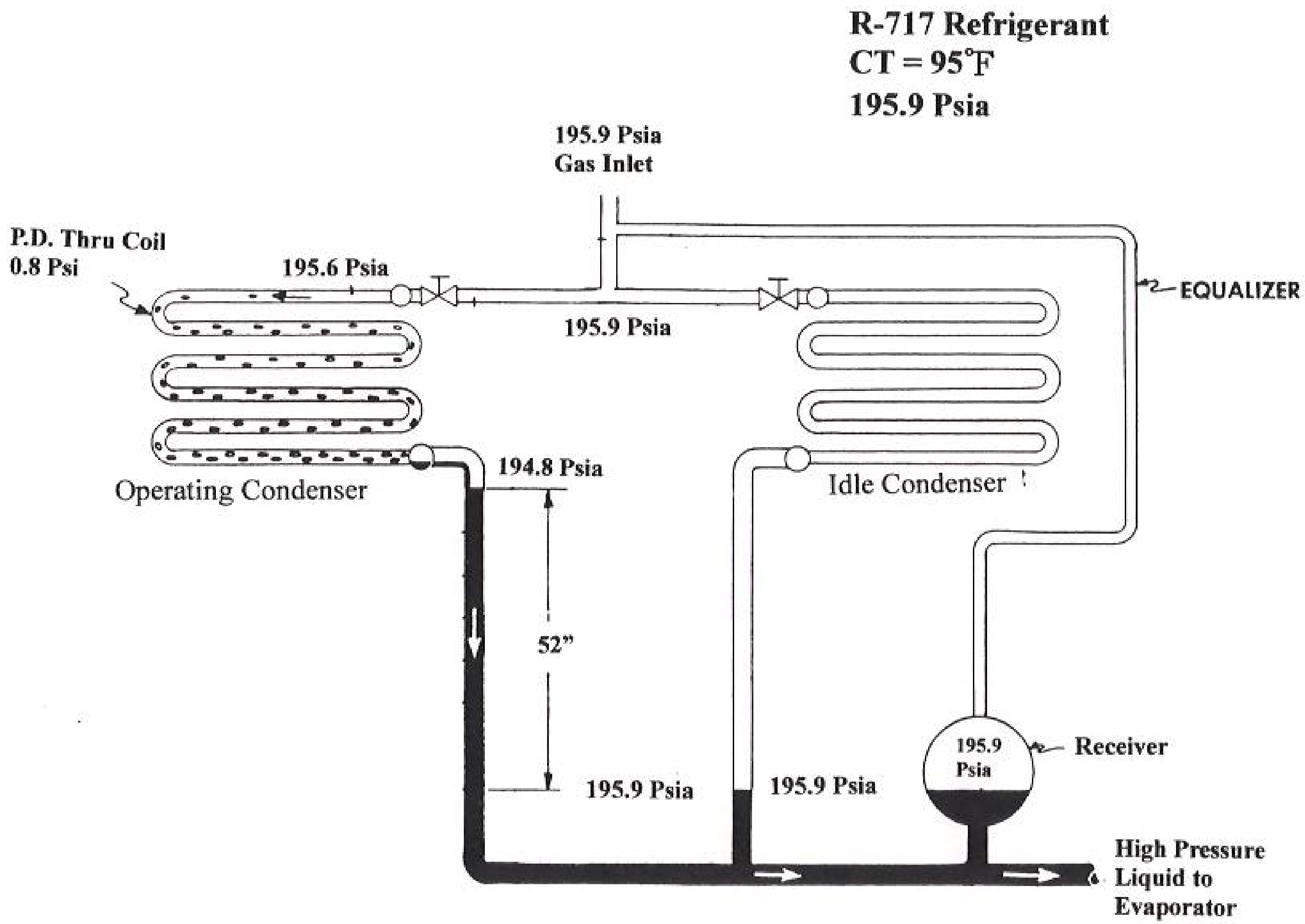


Figure 12-14 Liquid Trap Column for Multiple Parallel Units Evaporative Condenser Operation

It is recommended to check with the evaporative condenser manufacturer to obtain the coil pressure drop for the coil or their recommendation as what is height of the liquid column trap should be provided for the parallel application.

Evaporative Condensers in Parallel Operation:

For the multiple evaporative condensers system, it becomes necessary to shut off one or more evaporative condenser from a multiple system to maintain sufficient condensing temperature and system pressure difference. Therefore, the piping hook-up and the liquid trap column are important for proper operation of the refrigeration system.

Should the multiple evaporative condensers be used for a constant load or where the local wet bulb air temperature variation is not too much or all the condensers are expected to be operated all time even at partial load operation, liquid trap is not required under these circumstances.

For multiple evaporative condenser units operation, the requisites for the arrangements are as the following:

I) A receiver must be used. The receiver must be sized to have storage capacity to hold the operating charge for all the conditions.

II) An external equalizer line must be provided between the receiver and the gas inlet line to the evaporative condensers.

III) Enough vertical height liquid trap column shall be properly provided.

IV) The vertical liquid drain lines from the evaporative condensers to receiver are to be sized to have maximum velocity not more than 150 Ft/Min; the horizontal drain line is to be sized to have maximum velocity not more than 100 Ft/Min.

V) The equalizer line should be connected to the point where the pressure drop in the gas inlet line to each evaporative condenser is approximately the same.

VI) The shut off valves in the evaporative condenser liquid drain lines should be located at the lowest elevation point as possible to minimize flash gas.

VII) Normally, purge valve is to be located at the highest point of the liquid drain line.

The followings are the suggested piping hook-up for various evaporative condensers connected in parallel operation:

Figure 12-15 is the parallel condenser, each is having multiple coil circuits and with a storage liquid receiver.

Figure 12-16 is the piping hookup for parallel evaporative condensers with a surge receiver.

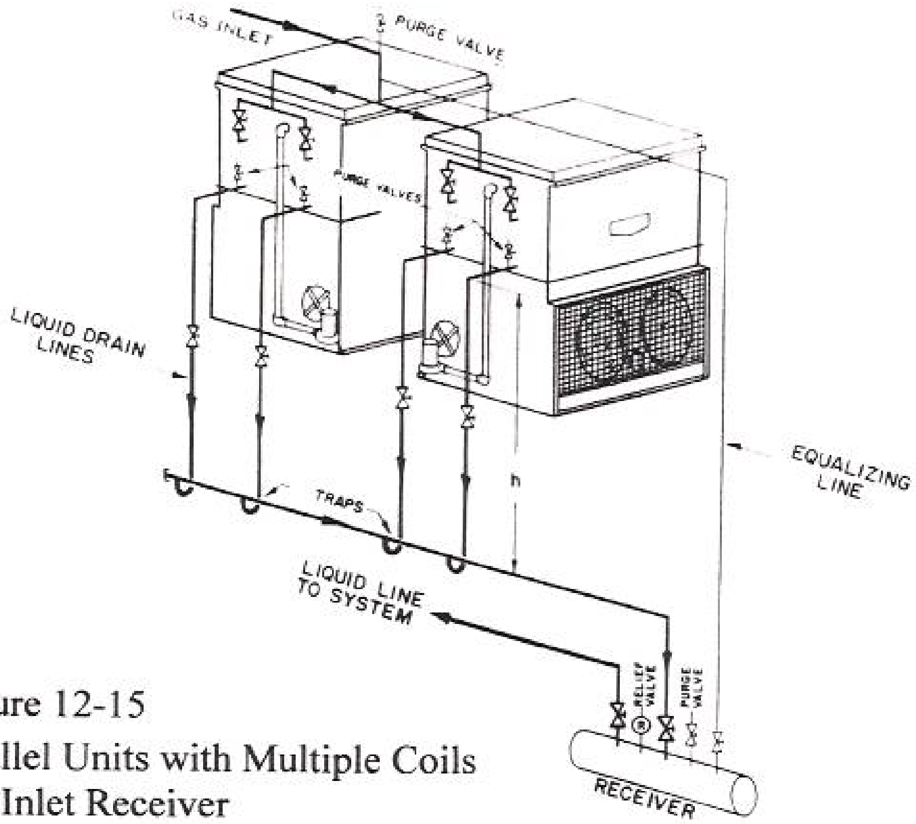


Figure 12-15
Parallel Units with Multiple Coils
Top Inlet Receiver

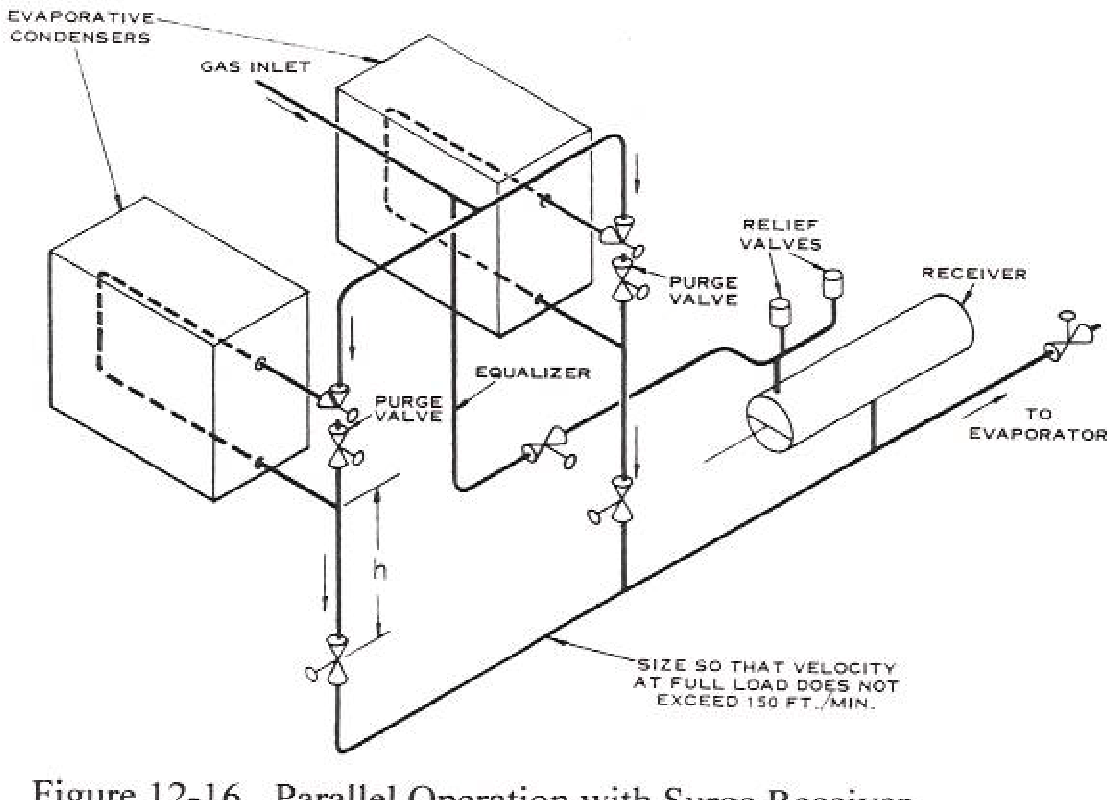


Figure 12-16 Parallel Operation with Surge Receiver

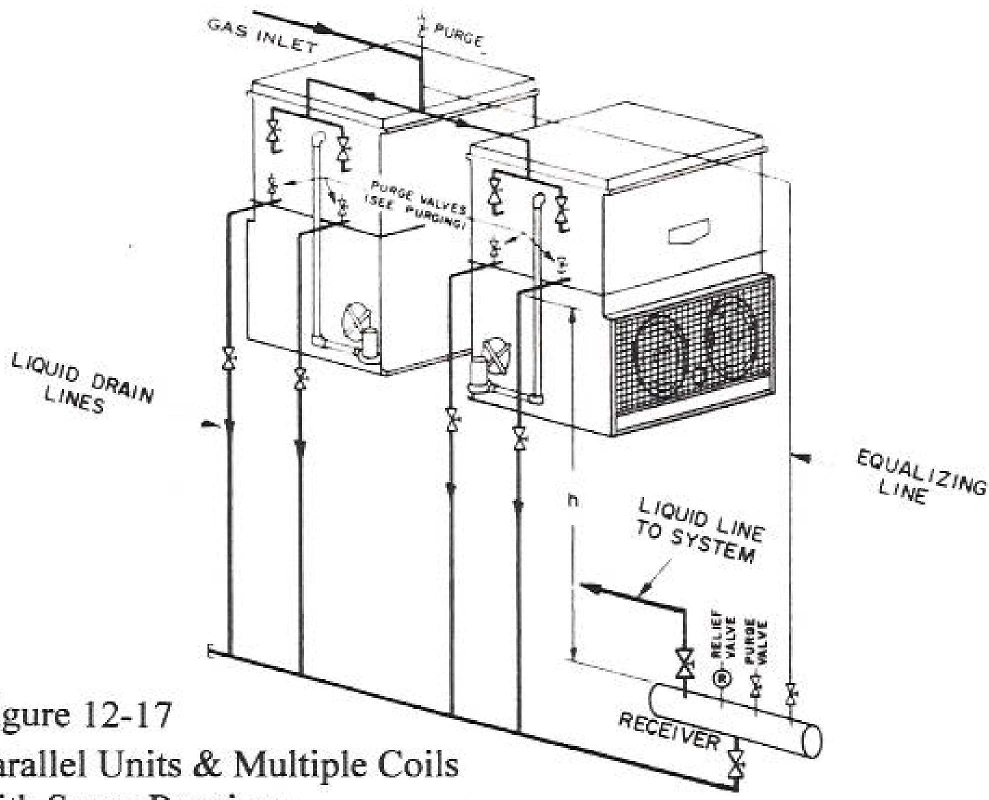


Figure 12-17
Parallel Units & Multiple Coils
with Surge Receiver

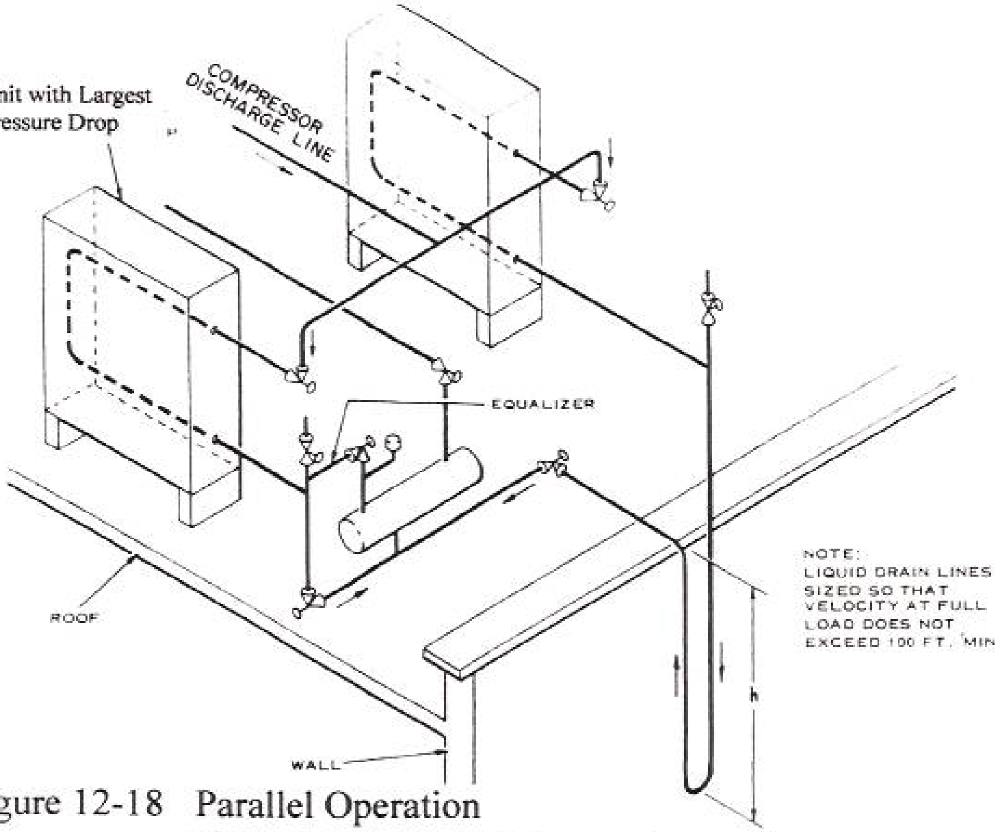


Figure 12-18 Parallel Operation
Condensers and Receiver on Same Floor

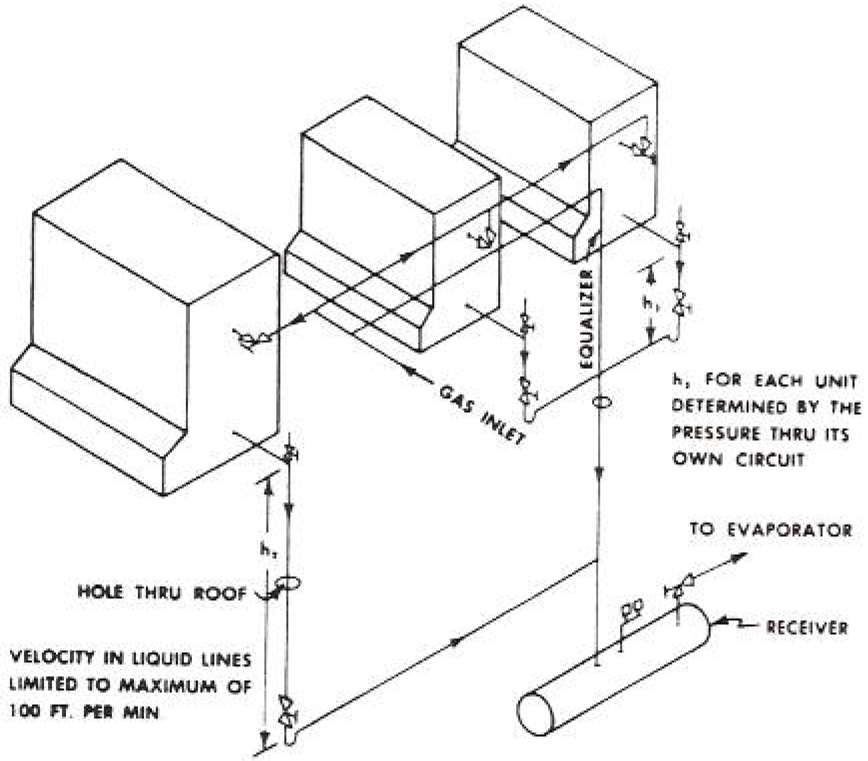


Figure 12-19 Parallel Operation
Mixed Sizes Evaporative Condensers

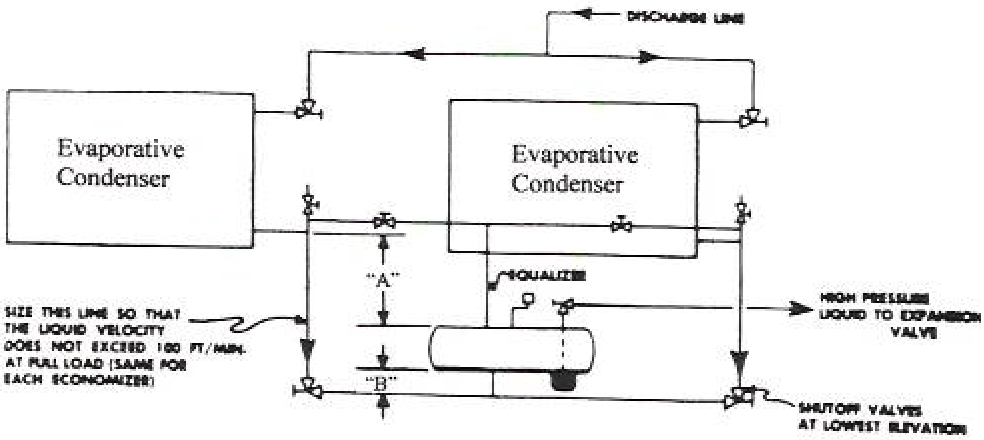


Figure 12-20 Parallel Operation
When Trapping Height not Available

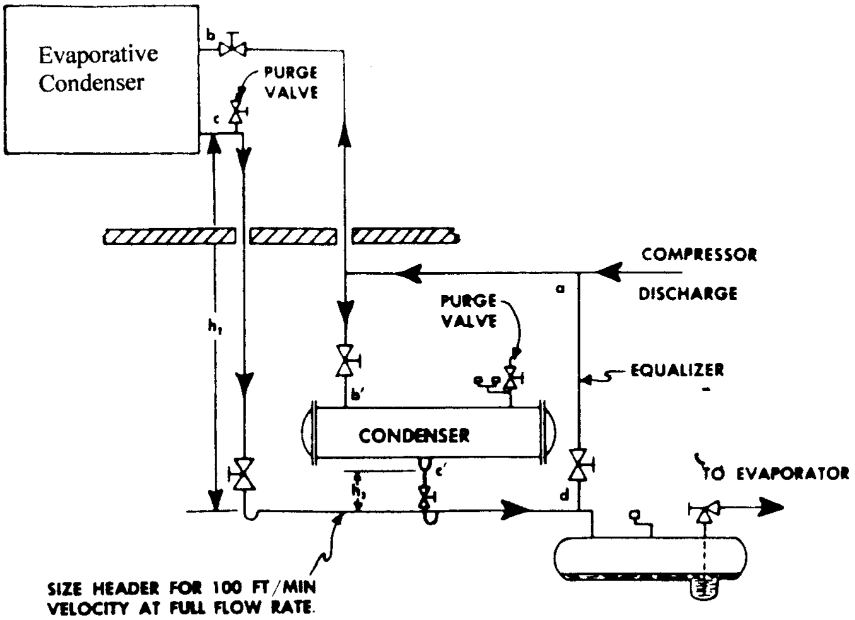


Figure 12-21 Parallel Operation of Evaporative Condenser and Shell-and-Tube Water Cooled Condenser

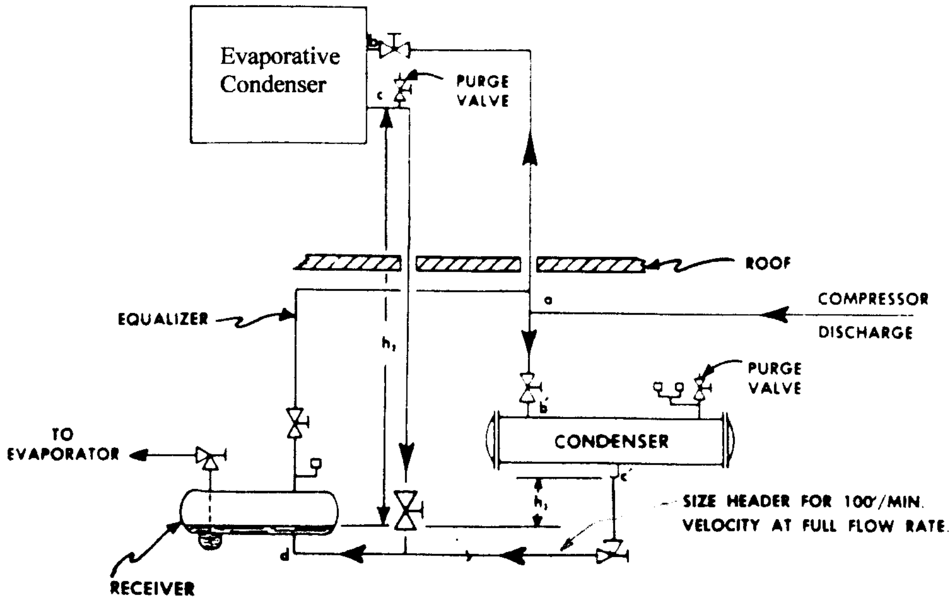


Figure 12-22 Parallel Operation of Evaporative Condenser and Shell-and-Tube Water Cooled Condenser with Surge Receiver

Figure 12-17 is the piping hookup for parallel operation and each unit is having multiple coils with a surge receiver.

Figure 12-18 is the parallel condensers mounted on the same floor level.

Figure 12-19 is the piping hookup for parallel operation but with mixed sizes of evaporative condensers.

Figure 12-20 is the piping hookup suggestion for parallel evaporative condensers on the same floor when the trap liquid column is not available. In this case, dimension "A" is the elevation difference between evaporative condenser outlet and the top of the receiver; dimension "A" must be at least 20 inches. Dimension "B" should be as small as possible. It is suggested that both evaporative condensers are to be remain operated at the same time.

Figure 12-21 shows an evaporative condenser and a shell-and-tube water cooled condenser are connected in parallel with top inlet storage receiver. H_2 in this case is the height between the evaporative condenser and the operating liquid level in the receiver. This height is determined for the pressure drop along the circuit "a-b-c-d" including the pressure drop through the evaporative coil. H_3 is the elevation between the bottom connection of the water cooled shell-and-tube condenser and the operating liquid level in the receiver; this height is determined by the pressure drop through the refrigerant circuit of "a-b'-c'-d".

Figure 12-22 shows the evaporative condenser and shell-and-tube water cooled condenser are connected in parallel with bottom inlet surge receiver. In this case, H_2 is the height between the evaporative condenser and the operating liquid level in the receiver. This height is determined for the pressure drop along the circuit "a-b-c-d" including the pressure drop through the evaporative coil. H_3 is the elevation between the bottom connection of the shell-and-tube condenser and the operating liquid level in the receiver; this height is determined by the pressure drop through the refrigerant circuit of "a-b'-c'-d". The horizontal liquid drain line from the water cooled condenser to the receiver is to be sized for 100 Ft/Min velocity maximum at the design load.

Liquid Subcooling:

Certain degree of liquid subcooling can be obtained by the evaporative condenser. However, the subcooling should be arranged in a separate circuit in the evaporative condenser as shown in the Figure 12-23. The liquid from receiver for the subcooling should be taken out at the downstream of the receiver to minimize the chance that vapor mixes with liquid at this point.

Winter and Low Ambient Operation:

For industrial refrigeration installations, the refrigeration system is required to be operated year round. Therefore, it is necessary to take provisions of maintaining a minimum required condensing pressure for the refrigeration system and also to prevent the water being freeze up in the evaporative condenser.

One of the better control methods for winter operation for evaporative condenser is to have automatic damper control of air flow for the evaporative condenser instead of dry coil operation. Dry coil operation needs change over and the method of control is not entirely satisfactory.

In order to prevent the water in the evaporative condenser freeze up during the system during shut down is to have a remote water recirculation system consists of sump tank and a pump to be located indoor as shown in Figure 12-24.

Data required for evaporative condenser selection and inquiry:

Heat rejection, Btu/Hr.
Condensing temperature.
Refrigerant.
Ambient wetbulb air temperature, °F.
Fan type, vane axial or centrifugal.
Induced draft or blow thru arrangement.
Subcooling coil, if required.
Special coating of additional corrosion protection.
Head control requirement.
Power supply, type of motor and electrical code requirement.

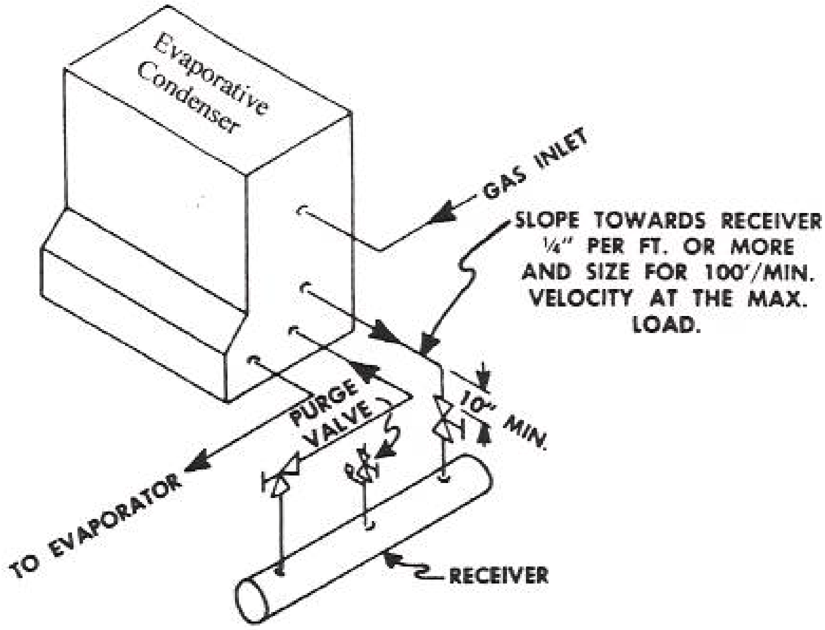


Figure 12-23 Evaporative Condenser with Separate Circuit for Liquid Subcooling

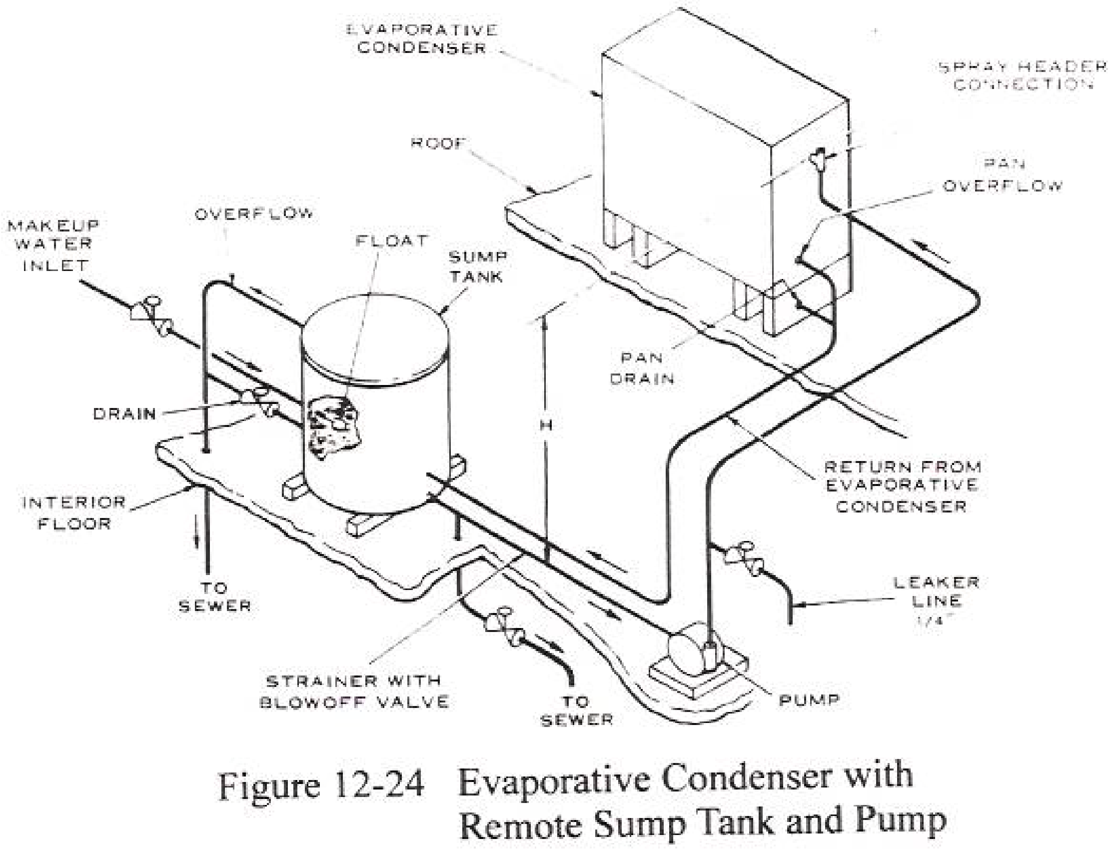


Figure 12-24 Evaporative Condenser with Remote Sump Tank and Pump

Shell-and-Tube Heat Exchangers

Various heat exchangers for refrigeration application under the category of shell-and-tube are as the following:

(I) Water Cooled Condenser. (Figure 12-33)

(II) Evaporators:

Flooded Type:
- Half Bundle type. (Figure 12-39)
- Full Bundle with Surge Drum. (Figure 12-40)

Dry Expansion (DX). (Figure 12-42)
Thermosyphone. (Figure 12-43)
Spray Evaporator. (Figure 12-44)
Overfeed Evaporator. (Figure 12-45)
Kettle Type Evaporator. (Figure 12-46)

Most the shell-and-tube heat exchangers for refrigeration application are horizontal with fixed tube sheet design that includes both condenser and evaporators. The efficiency of a vertical heat exchanger is not as good as the horizontal design; floating tube sheets design heat exchanger is not warranted for refrigeration system. The heat exchangers with U-tubes and/or floating tube sheet design are mainly used for evaporator duty and mostly are provided by the hydrocarbon processing industries users, not by refrigeration system providers.

Small Difference and LMTD:

The heat transfer formula for condenser or evaporator is as the following:

$$Q = U \times A_o \times LMTD$$

Q = Heat transferred, Btu/Hr.
U = Overall heat transfer coefficient, Btu/Hr- Ft^2-°F.
A_o = Total effective outside tube surface, Ft^2.

$$LMTD = \frac{L - S}{LOG_e \left\{ \frac{L}{S} \right\}}$$

The heat transmission and the thermal resistances through the tube are shown in Figure 12-25. Figure 12-26 shows the heat transmission resistances and the temperature gradients for the tube wall and each layer of film through the tube section from t_1 to t_7.

Small difference value is the difference between CT and the leaving cooling water or the

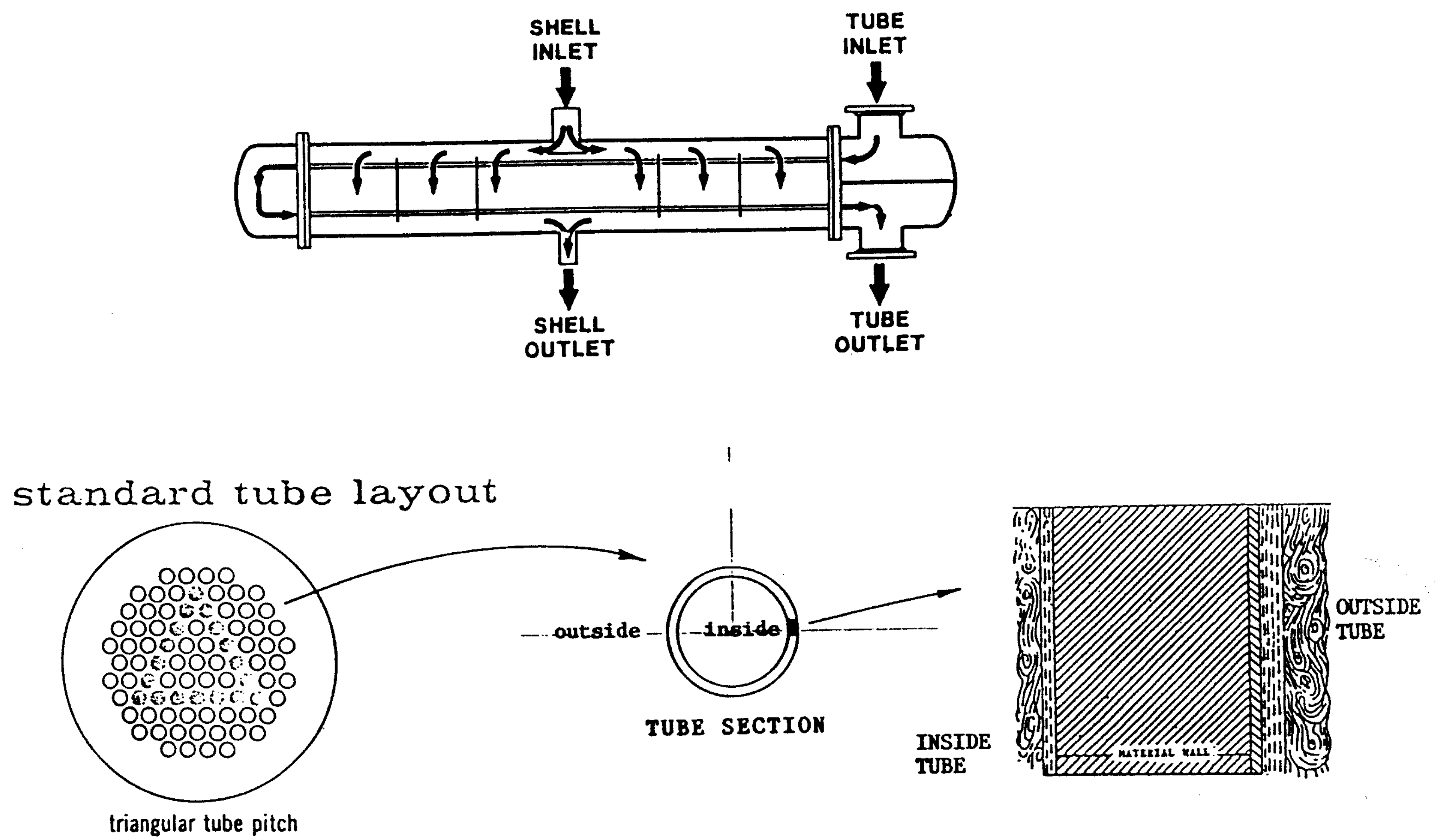
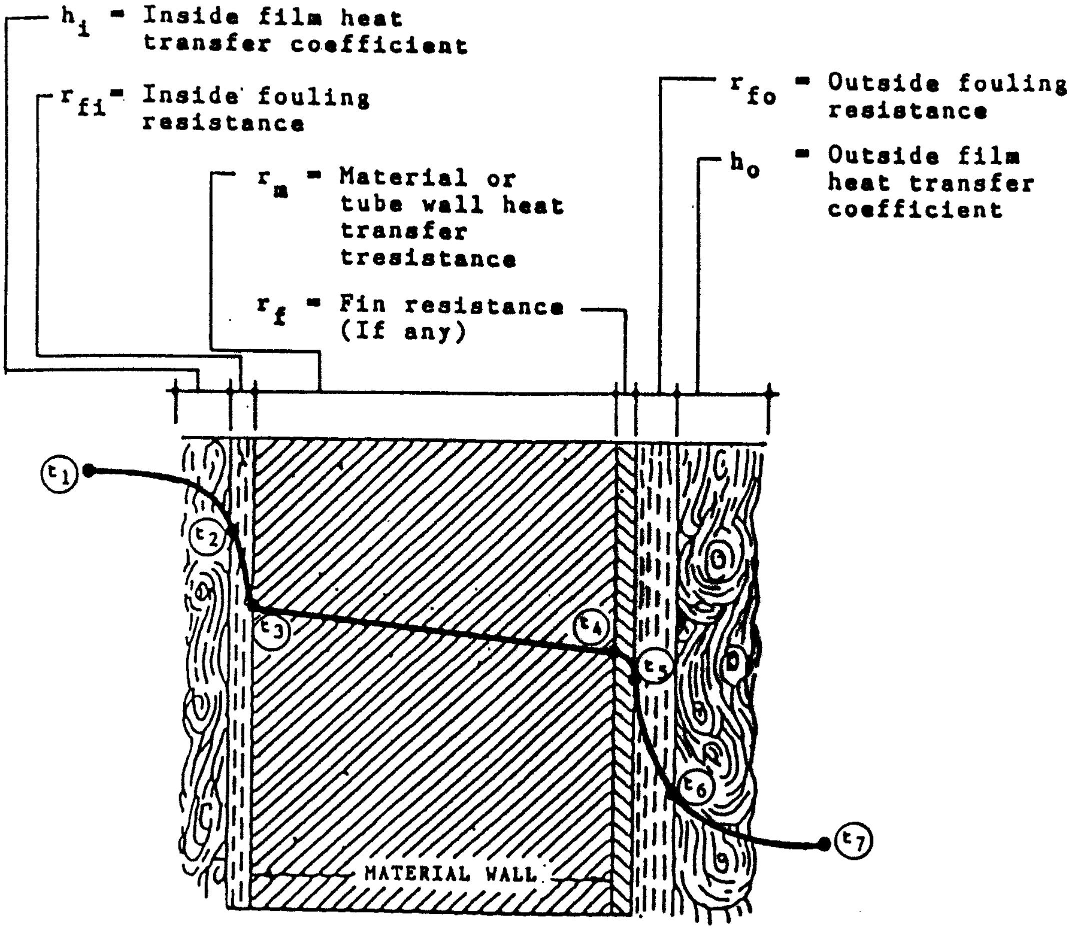


Figure 12-25 Heat Transfer & Thermal Resistances



$$U = \frac{1}{\left(\frac{1}{h_i}\right) R + r_{fi} \cdot R + r_m + r_f + r_{fo} + \frac{1}{h_o}}$$

Figure 12-26 Heat Transmission

difference between the leaving fluid temperature and the ET as shown in Figure 12-27. Some of the condenser or cooler performance curves are expressed in small difference instead of LMTD.

Shell Diameter and Tubes Count:

Figure 12-28 shows the optimum total possible tubes insert for a give shell diameter, the general heat exchanger data and the total external surface for the heat exchanger for given size of the heat exchanger. Figure 12-29 is the table for various shell diameter and number of tube counts for various pass arrangement for the heat exchanger. All the data are based on 3/4” OD tubes on 15/16” Triangular Pitch for both condenser and evaporator. For example: For a shell diameter of 24”OD, 2-Pass arrangement, full bundle flooded evaporator, maximum tubes count is 388.

Figure 12-30 shows the physical data for various commonly used 3/4" OD finned tubes. Figure 12-31 is the commonly used tube material and the corresponding thermal conductivity, Btu/ Ft^2-°F-Hr/Ft at 60°F. Figure 12-32 is the physical data for various commonly used bare tubes.

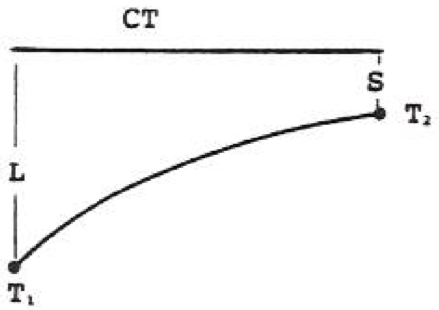
Code and Standard Compliance:

The manufacturing and the construction of shell-and-tube heat exchanger is to confirm with ASME code Section VIII for Unfired Pressure Vessels, see Chapter 13 for further details.

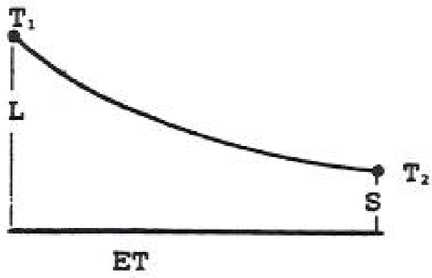
The water heads for the shell-and-tube heat exchanger under ASME code can be either water channel or regular water box or marine water box.

The definition of “marine” water box is that the water box for the heat exchanger shall be constructed in such way without disturbing the water piping connections while cleaning tubes for the heat exchanger.

CONDENSER



EVAPORATOR



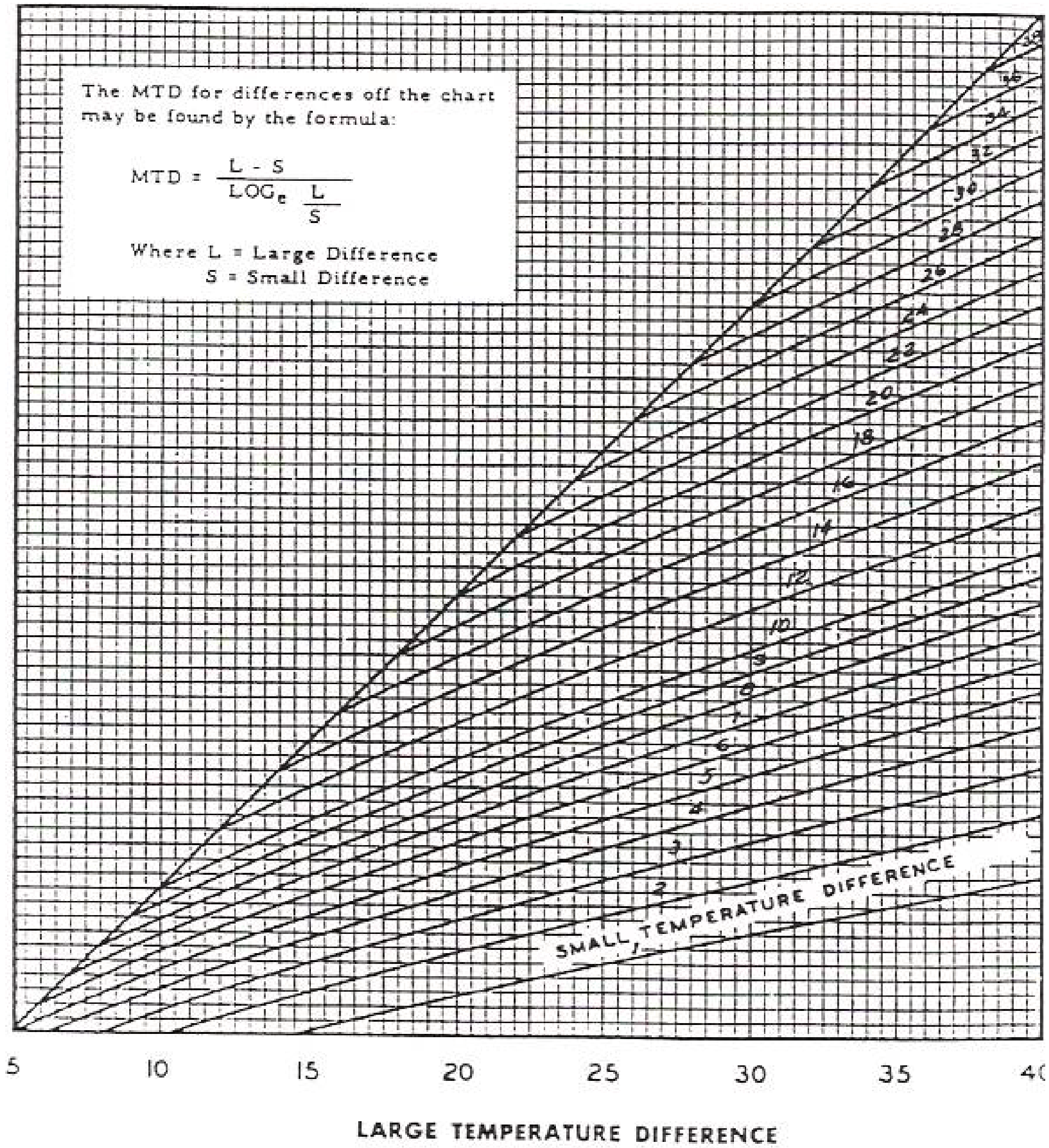


Figure 12-27 LMTD Chart

CONDENSER SIZE	NOMINAL SHELL OD	NO. OF TUBES	AREA, SQ. FT.		R-22, LBS.		WEIGHT, LBS	
			19 FINS/ INCH	26 FINS/ INCH	MIN. CHARGE	PUMPDOWN CAPACITY	SHIPPING	OPERATING
13-X-96	14"	120	475	576	61	302	1112	1277
13-X-120			597	724	77	379	1338	1545
13-X-144			719	872	92	458	1554	1802
13-X-168			841	1020	108	535	1780	2070
13-X-192			963	1168	123	613	1996	2328
13-X-216			1085	1316	211	690	2222	2668
13-X-240			1207	1464	227	768	2438	2926
15-X-120	16"	164	813	986	174	480	1842	2122
15-X-144			979	1188	195	579	2138	2547
15-X-168			1146	1390	216	677	2447	2913
15-X-192			1313	1592	237	776	2743	3265
15-X-216			1479	1794	258	874	3052	3631
15-X-240			1646	1996	279	973	3347	3983
19-X-144	20"	268	1595	1934	273	908	3587	4239
19-X-168			1867	2264	307	1062	4043	4792
19-X-192			2140	2595	340	1281	4478	5325
19-X-216			2412	2925	375	1372	4934	5878
19-X-240			2684	3255	482	1528	5369	6483
23-X-144	24"	404	2396	2906	371	1298	4934	5876
23-X-168			2807	3404	422	1518	5572	6661
23-X-192			3217	3901	473	1742	6179	7414
23-X-216			3628	4399	524	1963	6817	8199
23-X-240			4038	4897	648	2187	7424	9025
25-X-144	26"	460	2728	3309	417	1559	5891	6959
25-X-168			3196	3875	476	1824	6608	7844
25-X-192			3663	4442	535	2093	7289	8693
25-X-216			4131	5009	667	2358	8006	9650
25-X-240			4598	5576	726	2627	8687	10499
27-X-168	28"	600	4134	5013	574	1978	9074	10640
27-X-192			4743	5752	648	2273	9991	11772
27-X-216			5353	6491	795	2562	10951	13020
27-X-240			5962	7230	869	2857	11868	14153
29-X-168	30"	750	5167	6266	683	2162	10934	12856
29-X-192			5929	7190	773	2484	12027	14216
29-X-216			6691	8114	936	2801	13168	15698
29-X-240			7453	9038	1026	3123	14261	17057
31-X-168	32"	850	5856	7101	770	2547	12465	14639
31-X-192			6720	8148	873	2926	13684	16163
31-X-216			7583	9195	1049	3299	14961	17815
31-X-240			8447	10242	1152	3679	16180	19339

NTL – Normal Tube Length, Inches

Finned Tube

Nominal Shell Diameter

Figure 12-28 Shell Size & Tube Count

Shell Diameter, Pass and Maximum Tube Counts of 3/4"OD Tubes

Shell Diameter, Inches OD / ID	Number of Passes of the Shell				
	1-Pass	2-Pass	4-Pass	6-Pass	8-Pass
8.625" / 7.981"	49	42	32	24	NA
10.75" / 10.02"	77	68	60	48	NA
12.75" / 12.00"	111	100	96	84	76
14.00" / 13.25"	129	116	112	100	88
16.00" / 15.25"	177	162	152	138	124
18.00" / 17.25"	227	210	200	174	168
20.00" / 19.25"	273	254	228	204	200
22.00" / 21.25"	339	318	296	280	264
24.00" / 23.25"	411	388	368	346	324
26" OD	475	448	420	422	376
28" OD	559	530	512	502	460
30" OD	649	618	600	574	544
32" OD	749	716	700	672	640
34" OD	875	840	812	750	748
36" OD	995	958	868	864	804
38" OD	1113	1074	992	924	920
40" OD	1245	1204	1112	1072	1036
42" OD	1391	1348	1304	1236	1232
46" OD	1625	1578	1540	1498	1452
48" OD	1849	1800	1764	1690	1676
54" OD	2391	2334	2286	2228	2188
60" OD	2997	2934	2780	2720	2664
66" OD	3657	3588	3432	3364	3304
72" OD	4301	4218	4144	4070	4000

Figure 12-29 Shell Diameter, Pass and Maximum Tube Count

Physical Data for Commonly Used 3/4”OD Tubes

BWG	Nominal Wall at Finned Section	Fins per Inch	External Surface Sq.Ft. per Lin. Ft.	Internal Diameter, Inch	Surface Area Ratio Outside to Inside	Flow Area Square In.	Weight Per Ft. Length (Copper) Lbs.
20	0.035”	19	0.499	0.555	3.43	0.242	0.401
19	0.042”	19	0.499	0.541	3.53	0.229	0.448
18	0.049”	19	0.499	0.527	3.63	0.218	0.493
16	0.065”	19	0.499	0.495	3.86	0.192	0.593
14	0.085”	19	0.499	0.459	4.16	0.162	0.697
16	0.065”	16	0.438	0.495	3.39	0.192	0.665
14	0.085”	16	0.438	0.459	3.65	0.162	0.745
20	0.035”	26	0.630	0.555	4.34	0.242	0.405
19	0.042”	26	0.630	0.541	4.46	0.229	0.452
18	0.049”	26	0.630	0.527	4.58	0.218	--
16	0.065”	26	0.630	0.495	4.87	0.192	--
14	0.083	26	0.630	0.459	5.25	0.162	--

Figure 12-30 Physical Data for Commonly Used 3/4”OD Tubes

Tube Materials & Conductivities

Tube Material	Conductivity
Copper	196
90/10 Copper Nickel	27
70/30 Copper Nickel	17
85% Red Brass	92
Admiralty, Type B	64
Aluminum Brass, Type B	63
Arsenical Copper	196
3003 Aluminum	92
*Aluminum Bronze, 5%	48
Admiralty, Types C and D	64
Low Carbon Steel (Seamless or Welded)	26
Steel, ASTM A334	26
*Monel	15
*SS Steels, 304, 304L, 316, 316L or 321	9
Note: *Available 16 FPI only	

Figure 12-31 Commonly Used Tube Materials and Conductivity

Physical Date for Commonly Used Bare Tubes

Tube Size OD, Inch.	Wall Thickness In.	BWG	Ext. Surf. Sq.Ft./Ft.	I.D. In.	Int. Surf. Sq.Ft./Ft	Surface Ratio	Flow Area Sq. In.	Weight per Ft. (Copper) Lbs.	Displ. Per Ft. Length Cu.In./Ft
3/4”	0.035*	20*							
	0.042	19	0.1964	0.666	0.1740	1.13	0.348	0.360	5.3
	0.049	18	0.1964	0.652	0.1706	1.15	0.334	0.417	5.3
	0.065	16	0.1964	0.620	0.123	1.21	0.302	0.540	5.3
	0.083	14	0.1964	0.584	0.1530	1.25	0.268	0.670	5.3
1”	0.083	14	0.2618	0.834	0.218	1.20	0.546	0.92	9.42
**1-1/4	0.095	13	0.3273	1.06	0.2275	1.18	0.882	1.33	14.75
Note No.1: *0.035” wall bare tube is too thin to be recommended for rolled tube joints.									
Note No.2: **1-1/4”OD, 13 BWG, Black steel tubes, electric resistance welded; mostly used for ammonia horizontal coolers and condensers. Weight per feet length is 1.172 lbs.									

Figure 12-32 Physical Data for Commonly Used Bare Tubes

Water Cooled Condenser

Figure 12-33 shows the typical horizontal design shell-and-tube water cooled condenser.

Cooling tower water is used for most shell-and-tube condenser installations. Besides of cooling tower, the cooling medium for condenser can be river water, well water or sea water. Water softening is commonly used these days for cooling tower to minimize the fouling build-up in the tubes.

The most commonly used tube material for the condenser is copper. The condenser tube material should be changed to other corrosion resistance material if sea water is used. Some cases, the river or well water is also salty and corrosive, special tube material such as 90/10 Cupro Nickel or 70/30 Curpo Nickel should be used instead of copper for these application. Titanium tubes and tube sheet design condenser is the best for corrosion resistance, however it is the most expensive.

Cooling tower water is usually designed to have leaving water temperature 7 to 10°F above the design outdoor ambient wet bulb air temperature. The water range for condenser is usually about 10 to 15°F.

Water Cooled Condenser Performance:

$$\delta T = CT - T_w$$

δT = Small Difference, °F
CT = Condensing Temperature, °F
T_w = Leaving Condenser Water Temperature, °F

Therefore: $CT = \delta T + T_w$

Figure 12-34 is a typical water cooled condenser performance curve. Figure 12-35 is a diagram to show the impact of higher fouling factor for condenser rather than the normal allowance of 0.0005 Ft^2- Hr-°F/Btu. Figure 12-36 is the typical curve to show the impact of using other material for the condenser instead of copper. Those correction factors are in approximate LMTD to be added to the overall LMTD of the heat transfer for the condenser.

Cooling fluid flow calculation:

$$GPM = \frac{Btu/Hr.}{499.8 \times S.G. \times Cp \times (T_2 - T_1)}$$

Btu/Hr = Heat rejection
GPM = Cooling fluid flow, Gal/Min

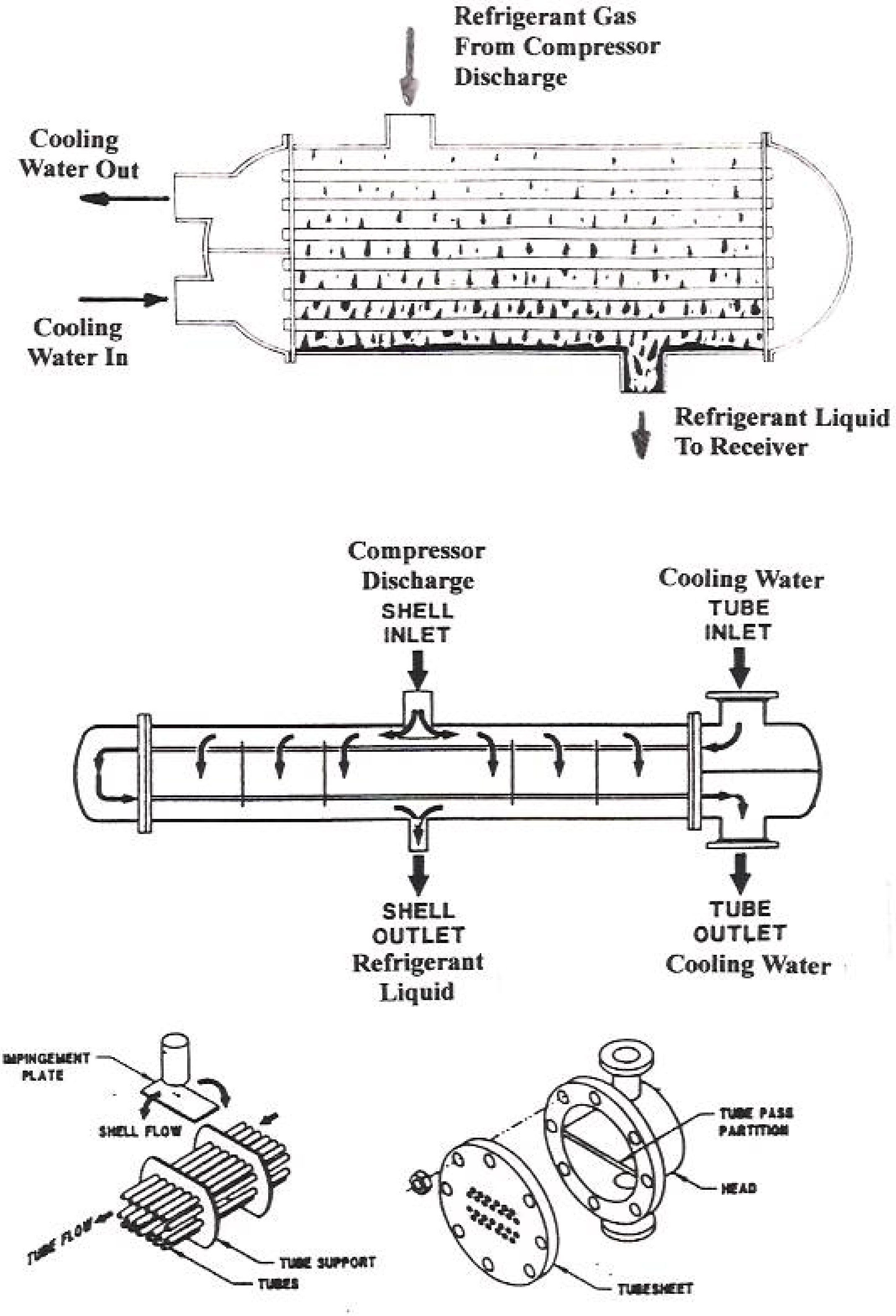


Figure 12-33 Typical Shell-and-Tube Water Cooled Condenser

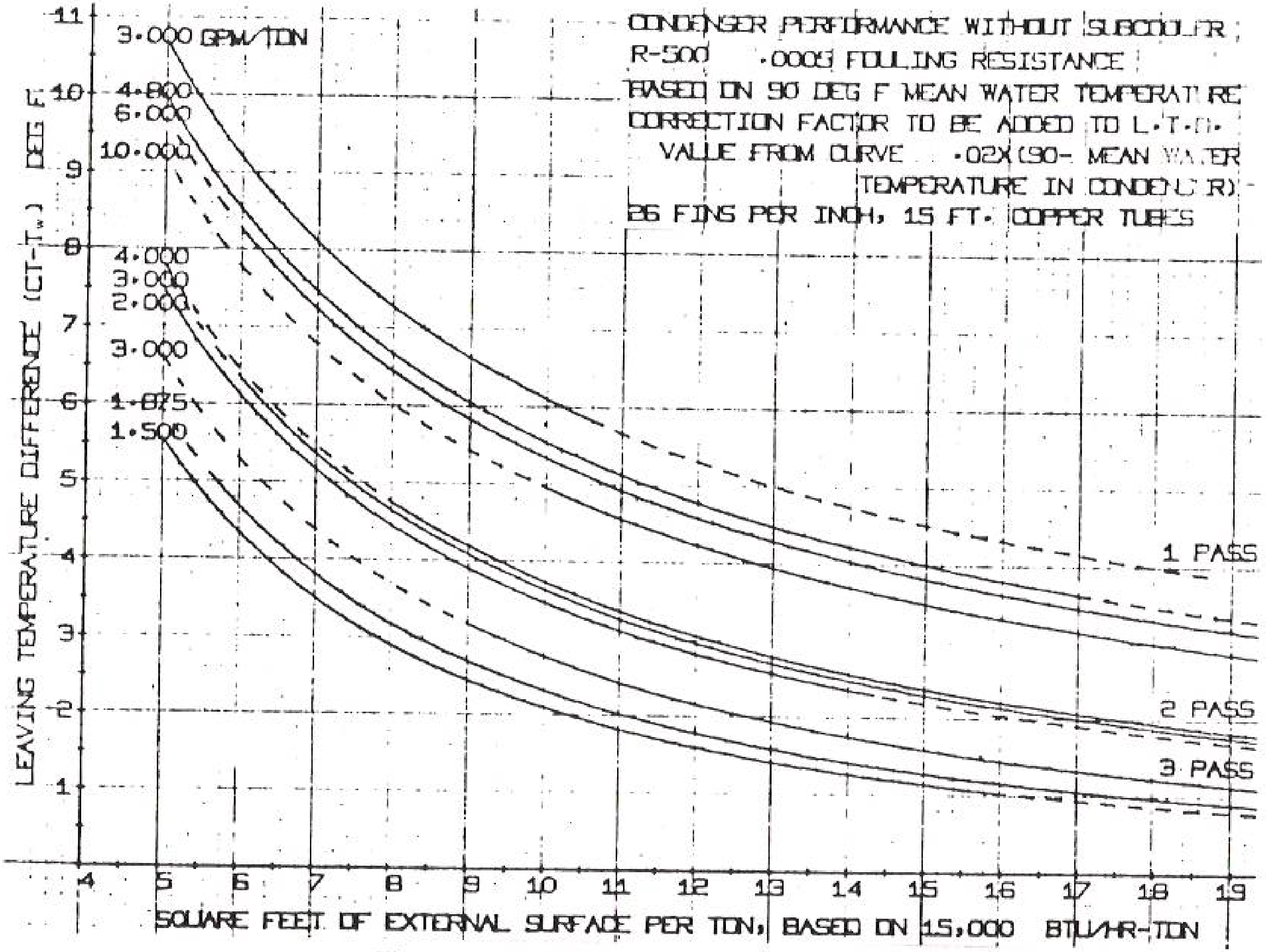


Figure 12-34 Typical Shell-and-Tube Water Cooled Condenser Performance

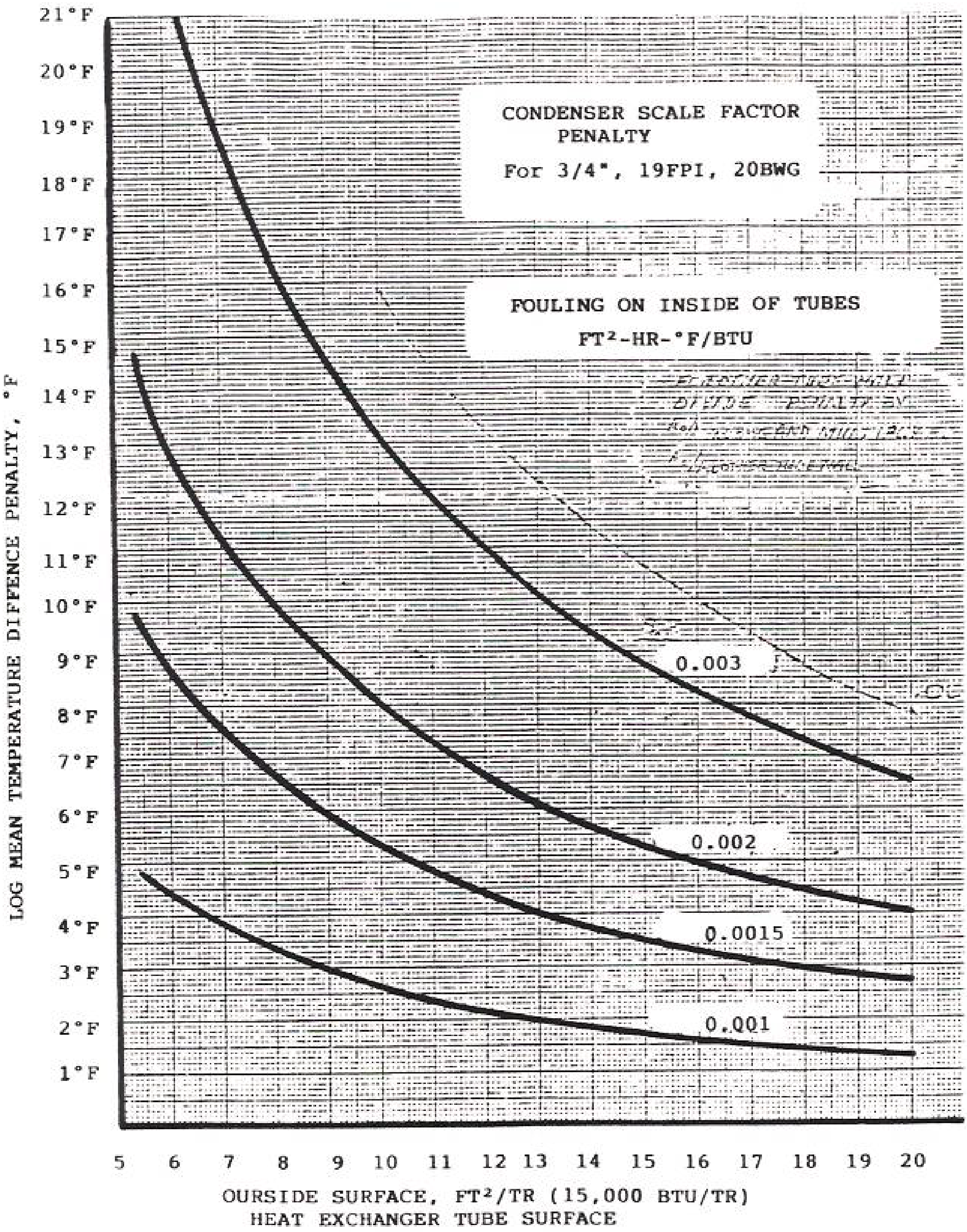


Figure 12-35 Typical Fouling Penalties
For Water Cooled Condenser

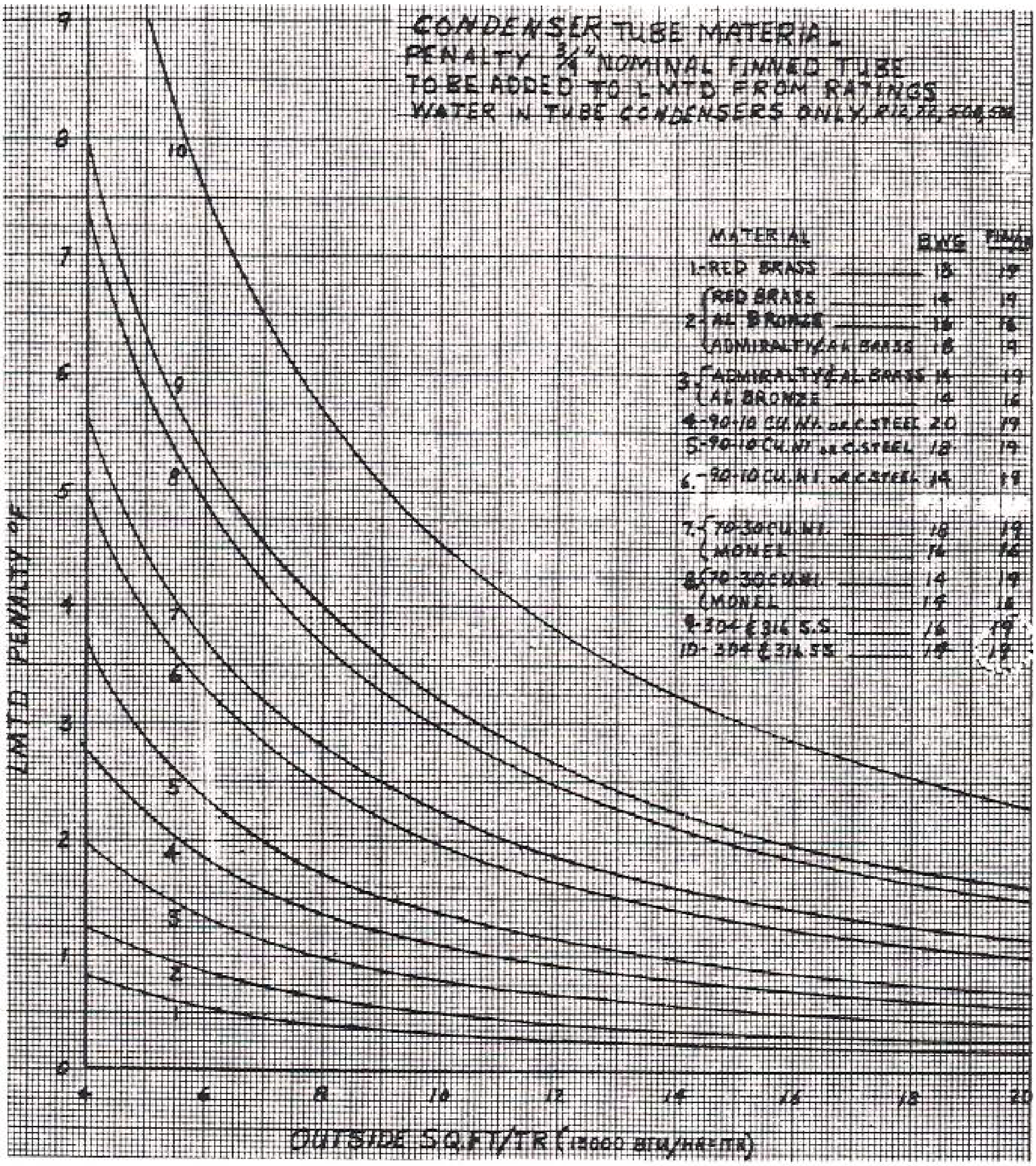


Figure 12-36 Typical Special Material Penalties
For Water Cooled Condenser

S.G..	= Specific Gravity of fluid at average temperature
Cp	= Specific Heat of fluid at average temperature
T_2 - T_1	= Range, fluid temperature difference, °F

If the cooling fluid is water, the formula becomes:

$$GPM = \frac{Btu/Hr.}{499.8 \times (T_2 - T_1)}$$

Data required for water cooled condenser selection and pricing:

Heat Load, Btu/Hr.
Cooling water in temperature, ℉.
Condensing temperature, ℉.
Refrigerant.
Shell side DWP and Tube side DWP.
Overall length or NTL limitation, if any.
Water pressure drop limitation, if any.
Special tube material, Gauge, FPI, Tube OD requirements, if any.
Fouling factor requirement.
Pass arrangement.
Cooling water GPM limitation, if any.

Evaporator

All the heat exchangers are to be selected by the heat exchanger manufacturer because the type of heat exchanger, NTL length, shell diameter, tube type, tube size, tube material and heat transfer coefficients vary widely depending on the brine or product and temperatures handled.

Evaporator Performance:

$$\delta T \quad = \quad T_w - ET$$

δT = Small Difference, °F
ET = Evaporative Temperature, °F
T_w = Leaving Chilled Water Temperature, °F

Therefore: $ET \quad = \quad T_w - \delta T$

Figure 12-37 shows the impact on various fouling factor for evaporator. The Figure 12-38 is the impact on evaporator if other tube material is used instead of copper tubes. These penalties for higher fouling factor or for special tube material are to be included in the value of δT.

Calculation for brine cooling is more complicated than for water because of the specific gravity, viscosity and thermal conductivity and etc. vary widely. Viscosity of the brine impacts greatly on the size of the heat exchanger. More tube heat transfer surface Ft^2/TR is needed for the application if the brine is having higher viscosity. As a rough guide and a rule of thumb, if a flooded evaporator is with 20 BWG 19 FPI 3/4" tubes, the relationships between Ft^2/TR and the viscosity of the brine are as the following:

Ft^2/TR	Brine Viscosity
8 Square Ft. per TR	2 to 3 CP
10 Square Ft. per TR	3 to 5 CP
14 Square Ft. per TR	5 to 8 CP
18 Square Ft. per TR	9 to 12 CP

A different type of evaporator should be considered instead of flooded evaporator if the brine viscosity is above 12 CP.

Fluid or Brine flow for evaporator:

$$GPM = \frac{TR \times 24}{S.G. \times Cp \times (T_2 - T_1)}$$

OR

$$Btu/Hr. = GPM \times 499.8 \times S.G. \times Cp \times (T_2 - T_1)$$

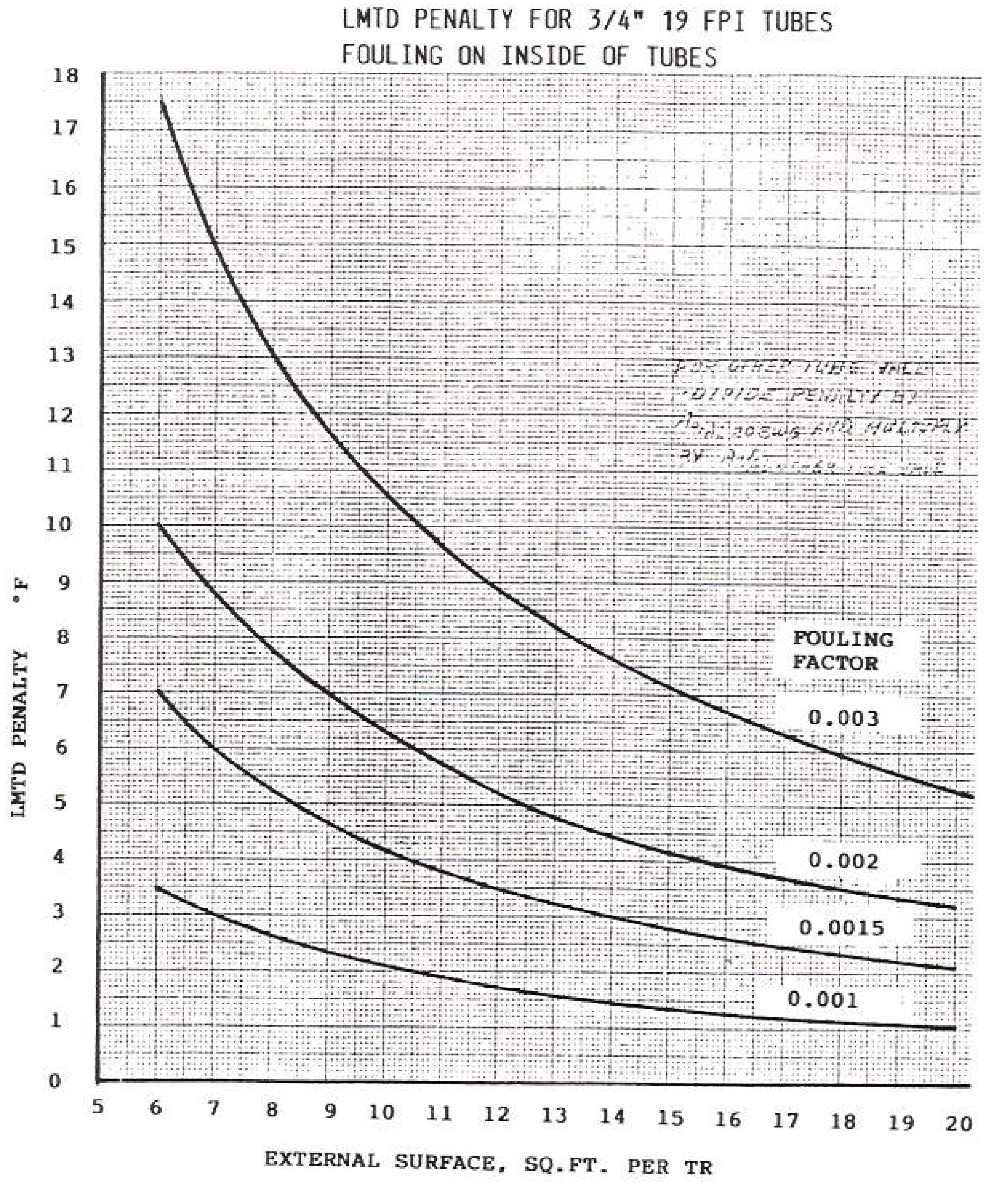


Figure 12-37 Penalties for Fouling Factor for Evaporator

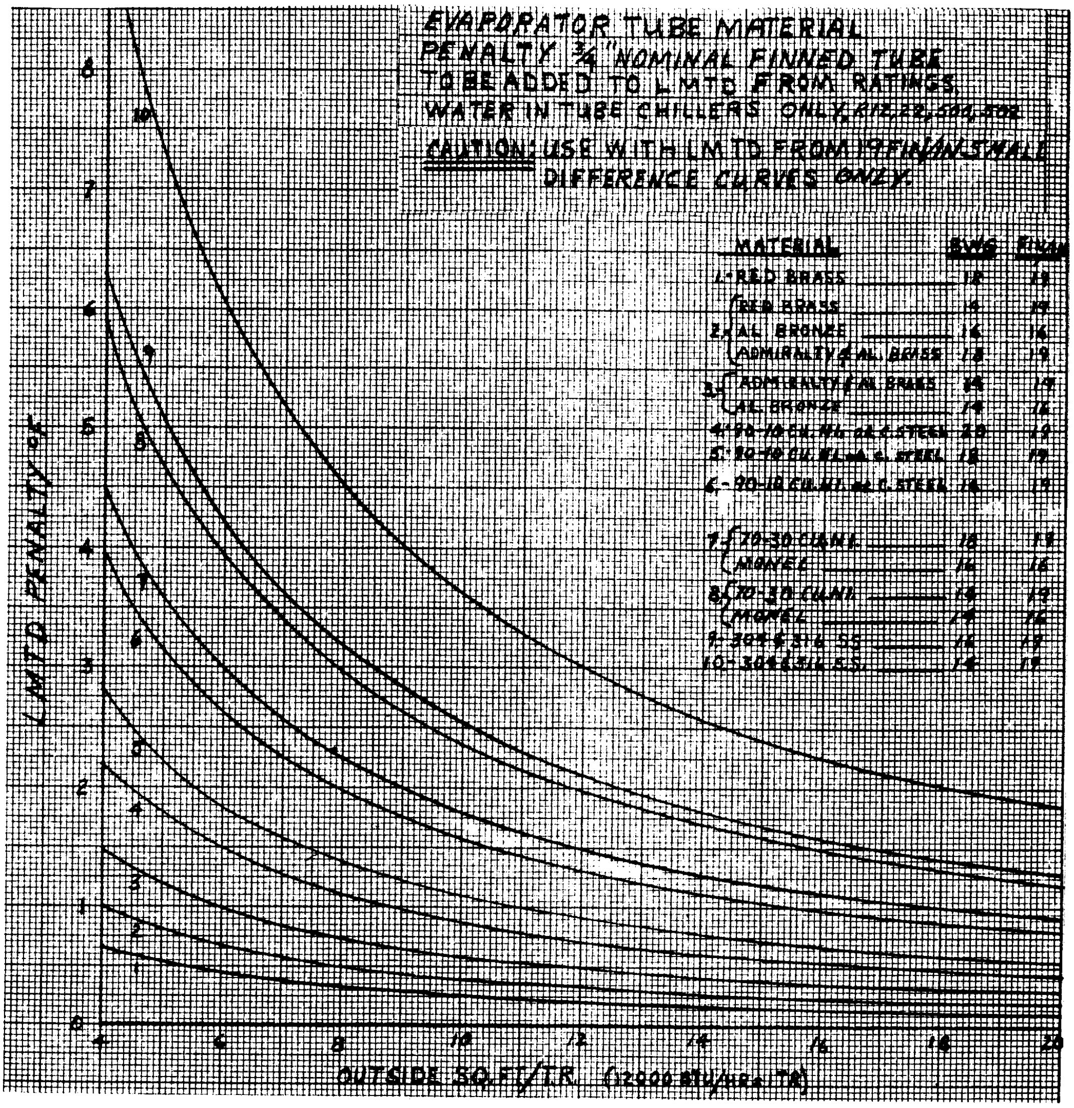


Figure 12-38 Penalties for Special Tube Material
For Evaporator

TR = Tons of refrigeration
GPM = Fluid flow, Gal/Min
S.G. = Specific Gravity of fluid at average temperature
Cp = Specific Heat of fluid at average temperature
$T_2 - T_1$ = Range, Temperature Difference, °F

Submerge Penalty:

For low temperature application, the density of the refrigerant liquid inside the evaporator increases rapidly as the evaporative temperature getting lower, the static head is increasingly greater accordingly. The evaporative temperature is to be lowered to offset this submerge penalty. The larger the shell diameter and lower the ET, more submerge penalty should be allowed. The degree of submergence effect also greatly depends on what refrigerant is used.

Data Required for Evaporator Selection and Pricing:

Refrigeration Load, TR or Btu/Hr.
In and out temperatures, °F.
Refrigerant.
Shell side DWP and Tube side DWP.
Evaporative Temperature, °F.
Pass arrangement.
Fouling factor.
Pressure drop allowed, if any.
Special tube material, if any.
Special requirements for tube OD, FPI, NTL, if any.

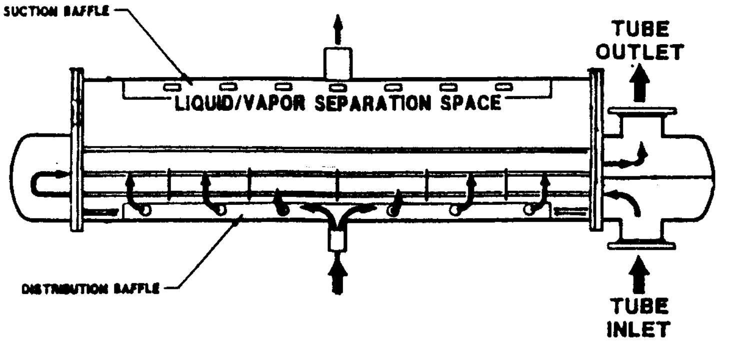
Additional information required if the cooling fluid is common brine instead of water:
Name of the brine, WT% brine concentration, if specified.

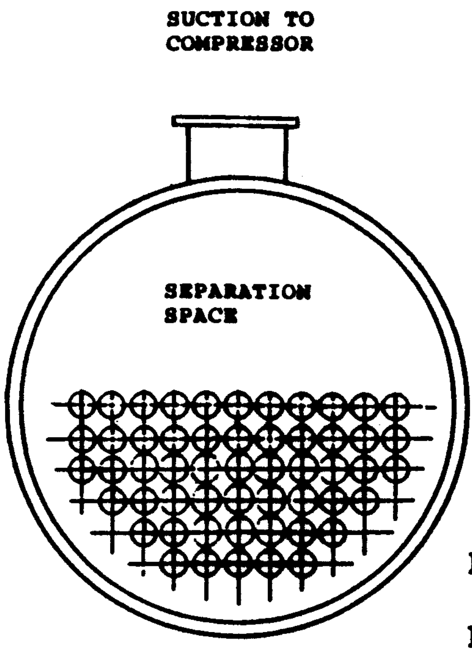
Additional information required if the medium is special fluid or special brine:
Name of the brine for fluid.
Specific heat at the average temperature.
Specific gravity at the average temperature.
Thermal Conductivity at the average temperature.
Viscosity at four temperature points:
At the brine average temperature.
At the leaving brine temperature.
At 10°F below brine leaving temperature.
At 15°F below brine leaving temperature.

Additional information required if the application is for gas condensation:
Name of the gas.
Mole fraction of each components of the gas mixture.

Half Bundle Flooded Evaporator:

Figure 12-39 is the half bundle or partial bundle evaporator design. This heat exchanger is one cylindrical shell. Lower half of the bundle is for tubes insert and the integral upper





PARTIAL BUNDLE
(HALF BUNDLE)
DESIGN

Figure 12-39 Shell-and-Tube Flooded Evaporator
Half Bundle Design

half space is for gas/liquid separation. The advantage of using this half bundle design is for installation where ceiling height is limited. The disadvantages are bigger shell diameter is required; might require higher submerging penalty if the ET is low and operating refrigerant charge is higher.

Full Bundle Flooded Evaporator:

Figure 12-40 is the full bundle shell-and-tube flooded evaporator. This heat exchanger consists of two shells. The lower shell is the evaporator which contains full bundle of tubes insert. The upper shell is the surge drum or the accumulator which provides the space for gas and liquid separation. These two shells are connected by the risers as shown.

Dry Expansion Evaporator:

Figure 12-41 is the DX evaporator. It is a low cost heat exchanger and it simplifies the refrigeration system design. DX system is basically used for small capacity installation.

Thermosypone Evaporator:

Figure 12-42 is a typical thermosyphone evaporator. The thermosyphone evaporator is a shell-and-tube heat exchanger coupled with a surge drum or accumulator. The brine or fluid flows through the shell side and the refrigerant is through the tube side. This heat exchanger is designed to cool high viscous fluid or brine. The refrigerant liquid is supplied to the thermosyphone evaporator from the surge drum on top of the evaporator by gravity force. Portion of the liquid is vaporized to cool the fluid in the heat exchanger; the bubbling mixture of liquid/vapor is circulated back to the surge drum. The refrigerant liquid from high side receiver is supplied to the surge drum through an expansion device such as liquid level control valve. Gas is returned to compressor suction from the surge drum. The surge drum must be located at a certain height above the evaporator in order to generate the thermosyphone effect.

Spray Evaporator:

Figure 12-43 shows the spray type shell-and-tube evaporator. This type of evaporator is used to minimize the refrigerant charge and also is used for very low temperature refrigeration to eliminate the submerge penalty in the evaporator. Liquid is pump recirculated. The liquid refrigerant charge is limited. The shell should be sized to provide gas/liquid separation or moisture eliminator should be provided to prevent liquid carry over back to compressor suction.

Overfeed Evaporator:

Figure 12-44 is the liquid overfeed type evaporator. The refrigerant liquid is forced feed through the tube. This heat exchanger is designed to cool high viscous product or brine because the product or brine is through the shell, not through the tubes. The application of this evaporator is the same as the thermosyphone evaporator except that the liquid is forced fed by a pump instead of by gravity fed, therefore, no height limitation is needed between the separation compartment and the evaporator.

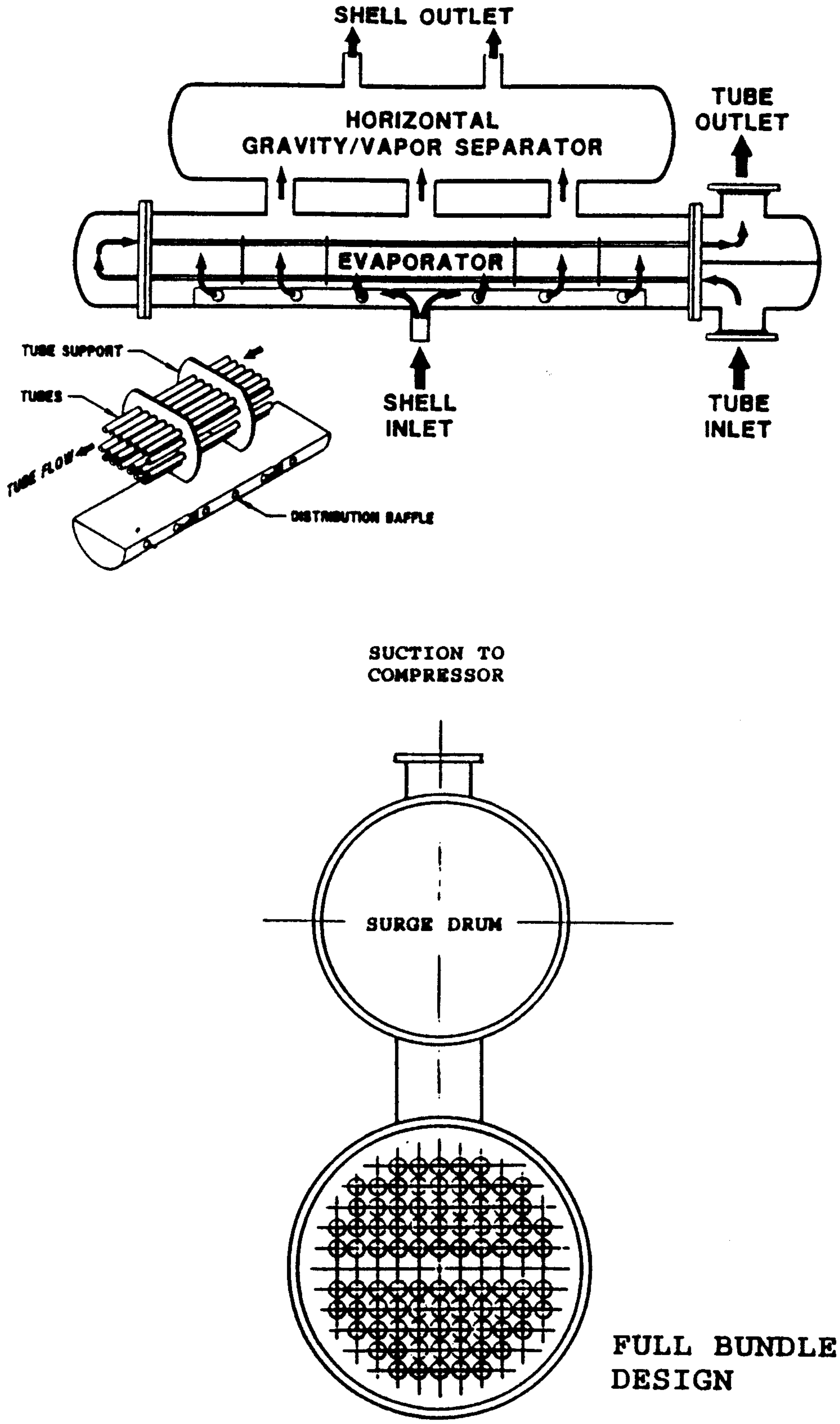


Figure 12-40 Shell-and-Tube Flooded Evaporator
Full Bundle Design

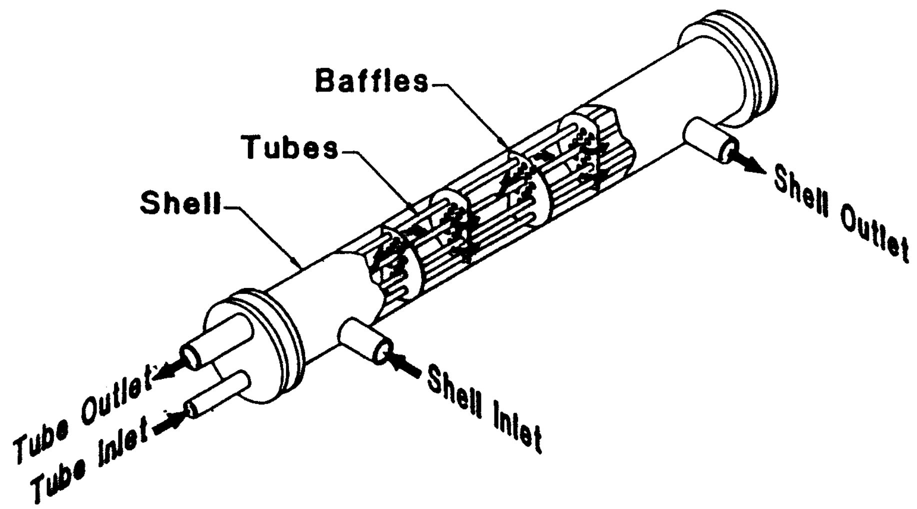


Figure 12-41 Shell-and-Tube Dry Expansion Evaporator

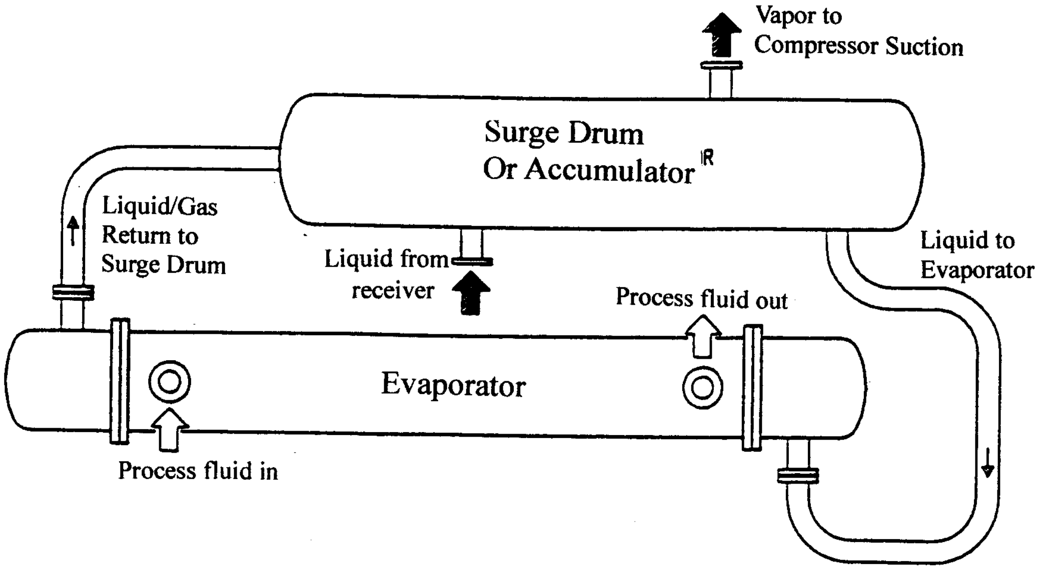


Figure 12-42 Shell-and-Tube Thermosyphone Evaporator

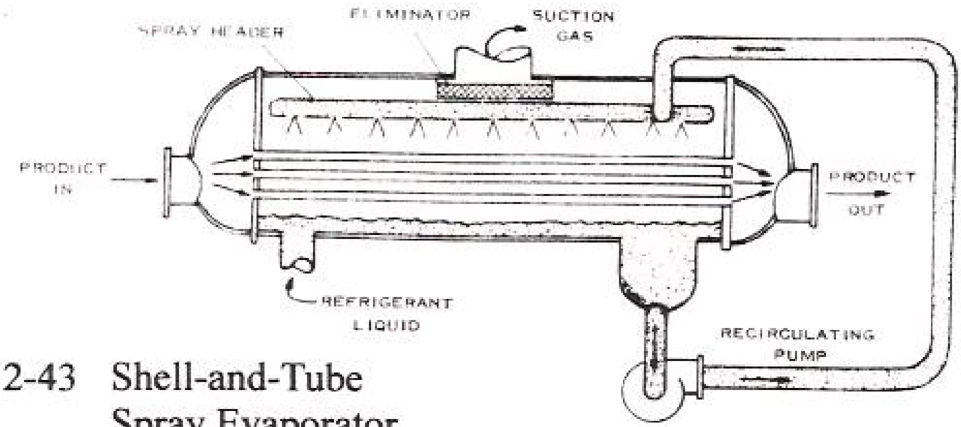


Figure 12-43 Shell-and-Tube Spray Evaporator

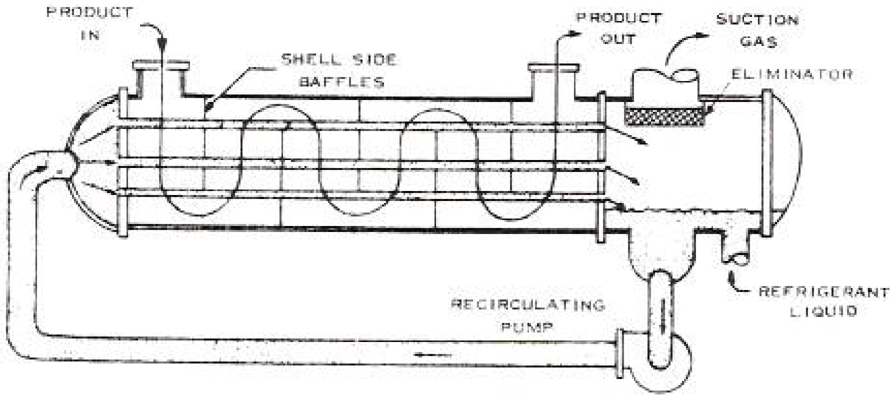


Figure 12-44 Shell-and-Tube Overfeed Evaporator

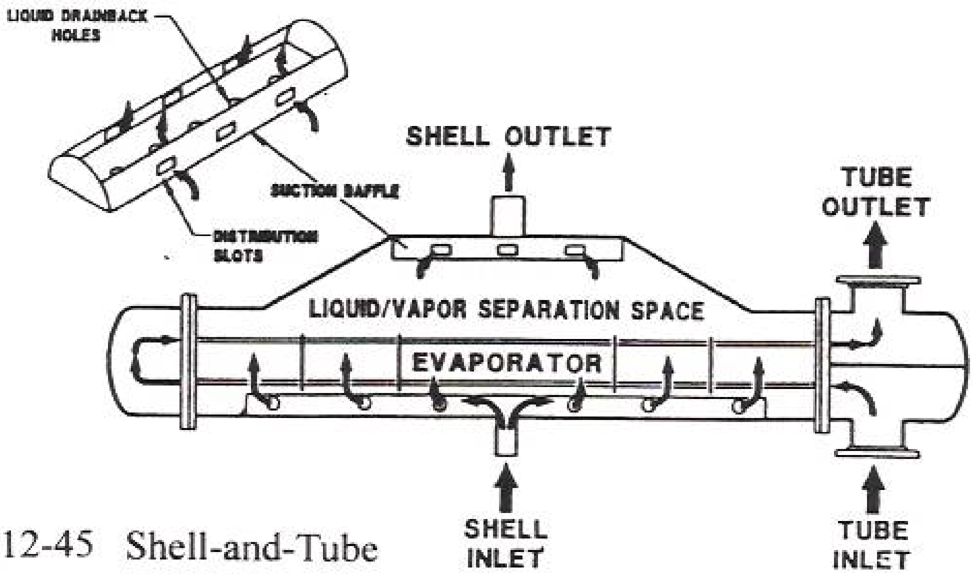


Figure 12-45 Shell-and-Tube Kettle Type Evaporator

Kettle Type Evaporator:

Figure 12-45 is the Kettle type evaporator. This type of evaporator is usually used by oil refinery, petrochemical and hydrocarbon processing industries. It only used for very special application and it usually provided by the user.