EVAPORATIVE CONDENSERS

Evaporative Condenser Piping Manual

BAC Evaporative Condenser Piping Manual

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1. Evaporative condenser piping guidelines

The evaporative condenser piping layout in industrial refrigeration systems is a critical element to proper function of the entire system. Improper pipe sizing and layout can cause increased energy usage, increased refrigerant charge, liquid back-up into the condensers, unstable operation, and other system issues. This manual is intended to provide the user with piping guidelines and methodologies that will aid in maximizing components' life, stable operation, and improving the overall system efficiency.

The depicted piping arrangements are those that, in the opinion and experience of the Baltimore Aircoil Company, provide the simplest and most reliable solutions for piping evaporative condensers. This document is intended for users already familiar with industrial refrigeration systems and cannot be used in isolation. Users of the information contained in this manual must also consider local code requirements, IIAR piping recommendations, local conditions with respect to wind, seismic, and other occasional loads, and proper support of all piping and equipment. Furthermore, to obtain the specified thermal performance from the evaporative condensers, proper airflow spacing shall be maintained between them as per the *BAC Evaporative Condenser Engineering Manual* (https://baltimoreaircoil.com/resources).

The following recommendations are written for evaporative condensers with ammonia as the system refrigerant. However, the rules of thermodynamics, heat transfer, and fluid dynamics apply to all the other similar refrigerants. For most non-natural refrigerants, with much higher mass flow rates compared to ammonia, refrigerant glide, and/or a large pressure/temperature relationship should be considered during the pipe sizing. This is outside the scope of this document, consult your local BAC representative for non-natural refrigerant applications.

NOTICE This document provides guidelines for the optimum design of evaporative condenser piping in ammonia refrigeration systems. The safety focus is on persons and property located at or near the premises where the refrigeration systems are located. Additional precautions may be necessary per the local codes or project specifications.

NOTICE All the ammonia refrigeration pipes shall be labeled with markers in accordance with the ANSI/IIAR standards.

2. High Stage Discharge (HSD) piping

There are several key considerations in the proper design of HSD piping:

- Overall HSD pressure drop between the compressor discharge and the condenser(s) inlet
- Relative HSD pressure drop differences between the common point upstream of all condensers in the array and the outlet of each coil
- Piping layout and connection arrangement to minimize the below-described pressure drop differences

Overall HSD pressure drop

Overall pressure drop affects operation and energy usage of the refrigeration system. This is because conveying refrigerant from the compressor(s) to the condenser(s) requires energy to overcome the pressure drop through pipe, fittings, and valves. Excessive HSD pressure drop requires additional compressor power which increases continuous operational cost and wear-and-tear on the compressor(s). To avoid this, the HSD piping system should be designed to minimize the pressure drop from the compressor outlet to the condenser inlet. The later section on condenser drain piping will explain how HSD pressure drop affects condenser outlet piping. When future capacity additions are planned, layout HSD piping accordingly to prevent rework requiring significant pipe changes and system shutdowns.

Relative pressure drop

The system designer should calculate the actual pressure drop from the first compressor to the last condenser along the piping run. This pressure drop must be either added to the compressor discharge design pressure (or saturated temperature) or subtracted from the design condensing pressure (or saturated temperature).

A convenient starting point is 184.1 Psig (96°F) for the compressor discharge and 181.1 Psig (95°F) for the condensing pressure. This allows for 3 psi of pressure drop in the HSD piping.

If this maximum recommended 3 psi HSD pressure drop is not accounted for, then the compressor will have to operate at a higher discharge pressure than design to overcome this pressure drop and still maintain the design condenser operating pressure and saturated temperature. This increased discharge pressure will consume more compressor energy than designed for, which will increase the motor load and decrease capacity.

The economic analysis of the allowed HSD pressure drop should consider the annual load profile expected for the system. A production facility in a warm climate will operate at or near design conditions most of the year and the HSD pressure drop will consume a considerable amount of energy. Conversely, a cold storage facility or seasonal vegetable freezing facility in a northern climate, will only operate at full design capacity a fraction of the time with the bulk of the operation at part-load with significantly reduced HSD pressure drop and energy consumption.

Note that the pressure drop is proportional to the square of vapor (load) velocity, thus giving part load operation a significant energy advantage.

Table 1, 2 and **3** show an example of how to precisely calculate the pressure drop of the HSD piping using the equations presented in the appendix-A, the appropriate ammonia properties and K or Cv (loss coefficients) values for fittings and valves. In addition, **Table B-2 and B-3** list the maximum recommended refrigerant mass flow rate for HSD header and branch piping for different nominal pipe sizes.

Pipe Segment	Refrigerant mass flow rate	Pipe Size	ΔP per 100’	Velocity	Pipe Length	Loss coefficient K	Segment ΔP	System pressure
	[lb_m/min]	**[in]**	**[psi]**	**[ft/sec]**	**[ft]**	**-**	**[psi]**	**[psig]**
Compressor discharge pressure								**184.10**
Compressor riser	171.9	5	0.33	38.6	5		0.017	
Elbow	171.9	5		38.6		0.20	0.017	
Reducer, 5 x 8	171.9			38.6		0.50	0.043	
T-Branch	171.9	8		46.3		0.58	0.072	
Header end	515.7	8	0.27	46.3	5		0.014	0.28
Elbow	515.7	8		46.3		0.16	0.020	
Header riser	515.7	8	0.27	46.3	20		0.054	
Elbow	515.7	8		46.3		0.16	0.020	
Horizontal pipe run to condenser A	515.7	8	0.33	39.2	8		0.022	
Pressure at the inlet of the common header								**183.82**
Branch pipe pressure drop								0.68
Condenser-A inlet pressure								**183.14**
T-Through	343.8	8		30.9		0.10	0.005	
Horizontal pipe run to condenser B	343.8	8	0.12	31.4	16		0.019	0.025
Branch pipe pressure drop								0.68
Condenser-B inlet pressure								**183.12**
T-Through	171.9	8		15.4		0.10	0.001	
Horizontal pipe run to Condenser C	171.9	8	0.03	15.4	16		0.005	0.01
Branch pipe pressure drop								0.68
Condenser-C inlet pressure								**183.11**
Header extension to equalizer connection								0
Equalizer pressure drop								0

Table 1**: Example of HSD individual section pressure drop at the design mass flow rate without thermosyphon cooling - this table corresponds with **Figure 10

Pipe Segment	Refrigerant mass flow rate	Pipe Size	ΔP per 100’	Velocity	Pipe Length	Cv	Loss coefficient K	Segment ΔP
	[lb_m/min]	[in]	[psi]	[ft/sec]	[ft]	-	[psi]	[psig]
T-Branch pipe	171.9	8	0.03	15.4			0.58	0.008
Reducer, 8 x 4	171.9			15.4			0.40	0.005
Pipe	171.9	4	1.06	60.7	3			0.032
Elbow	171.9	4		61.6			0.22	0.047
Globe Valve	171.9	4				290		0.590
Total ΔP								**0.680**

***Table 2**: Branch pipe pressure drop (same for each condenser) – this table corresponds with **Figure 10**)*

Pipe Segment	Refrigerant mass flow rate	Pipe Size	ΔP per 100’	Velocity	Pipe Length	Cv	Loss coefficient K	Segment ΔP
	[lb_m/min]	[in]	[psi]	[ft/sec]	[ft]	-	[psi]	[psig]
Sharp entrance	47.63	4		13.5			0.5	0.007
Globe valve	47.63	4				290		0.037
Pipe	47.63	4	0.07	13.5	30			0.021
Elbow	47.63	4		13.75			0.22	0.003
Elbow	47.63	4		13.75			0.22	0.003
Elbow	47.63	4		13.75			0.22	0.003
Total ΔP								**0.073**

Table 3**: Thermosiphon vent line pressure drop – this table correspond with **Figure 11

Operating conditions for Table 1, 2 and 3

Pressure drops provided in **Tables 1, 2, and 3** are obtained by using sections A, B, and C in appendix-A and based upon the following operating conditions:

- Three 400-ton compressors discharging into three equally sized condensers. The separator, check valve, and service valve pressure drops are accounted for in the compressor rating.
- Operating conditions are 15°F suction to 96°F saturated discharge with 180°F discharge temperature with water-cooled oil coolers.
- Equalizer flow is zero or minimal, as it is assumed that the liquid volume flow rate into the receiver is equal to the liquid volume flow rate out of the receiver under steady state conditions.

- Compressor discharge vapor density = 0.534 lb_m/ft^3 (Superheated at 180°F)
- Liquid density = 36.624 lb_m/ft^3 (Saturated)
- Thermosiphon vapor density = 0.662 lb_m/ft^3 (Saturated)
- Thermosiphon load = 1,375 kBTU/hr.

Compressors using thermosiphon oil cooling produce two vapor streams to the condensers:

1) Main compressor discharge flow
2) Vapor generated in the oil cooler

This separate vapor flow can vary from 5 to 20% of the main compressor flow. Therefore, it has an impact on the HSD piping design. Water-cooled, and air-cooled compressors require no additional condenser refrigerant piping considerations. Liquid injection oil cooling may increase the discharge mass flow of the compressor and will be reflected in the compressor rating sheet mass flow rate.

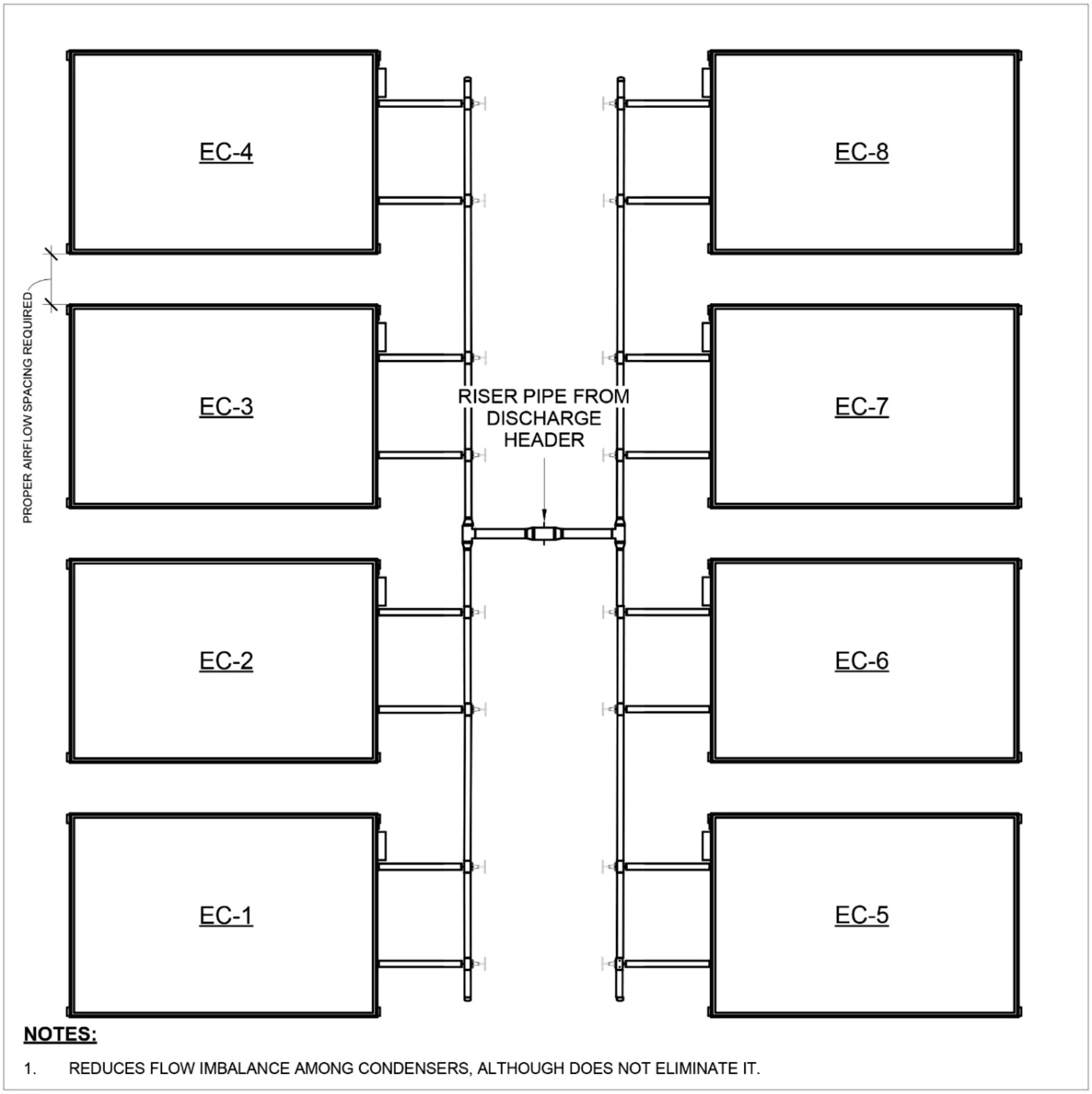
Piping layout and connection size to minimize the pressure drop

In addition to the HSD pressure drop, the HSD piping layout is also very important, especially for multiple condenser installation. Ideally, the discharge header and Condenser Drain (CD) from and to the machine room should enter the condenser field from a central location to provide a balanced refrigerant mass flow to all the condensers. This helps to provide for stable system operation at all operating conditions. Furthermore, a balanced layout for the condensers also has the benefit of reducing the length of, and therefore the total rise in the sloped CD header. This required slope (rise in CD header height at the farthest point) will detract from the available liquid drop leg height for the farthest condenser(s). This point is further explained in the CD piping section. BAC recommends HSD piping layout as depicted in **Figure 1**.

While perfect balance is not possible without introducing an undue pressure drop that would increase energy consumption, methods exist to ensure that sufficient balance is achieved to ensure adequate distribution of refrigerant to each condenser in the array. If necessary, proper HSD sizing along with adequate drop leg heights can be used to overcome less than ideal HSD layouts.

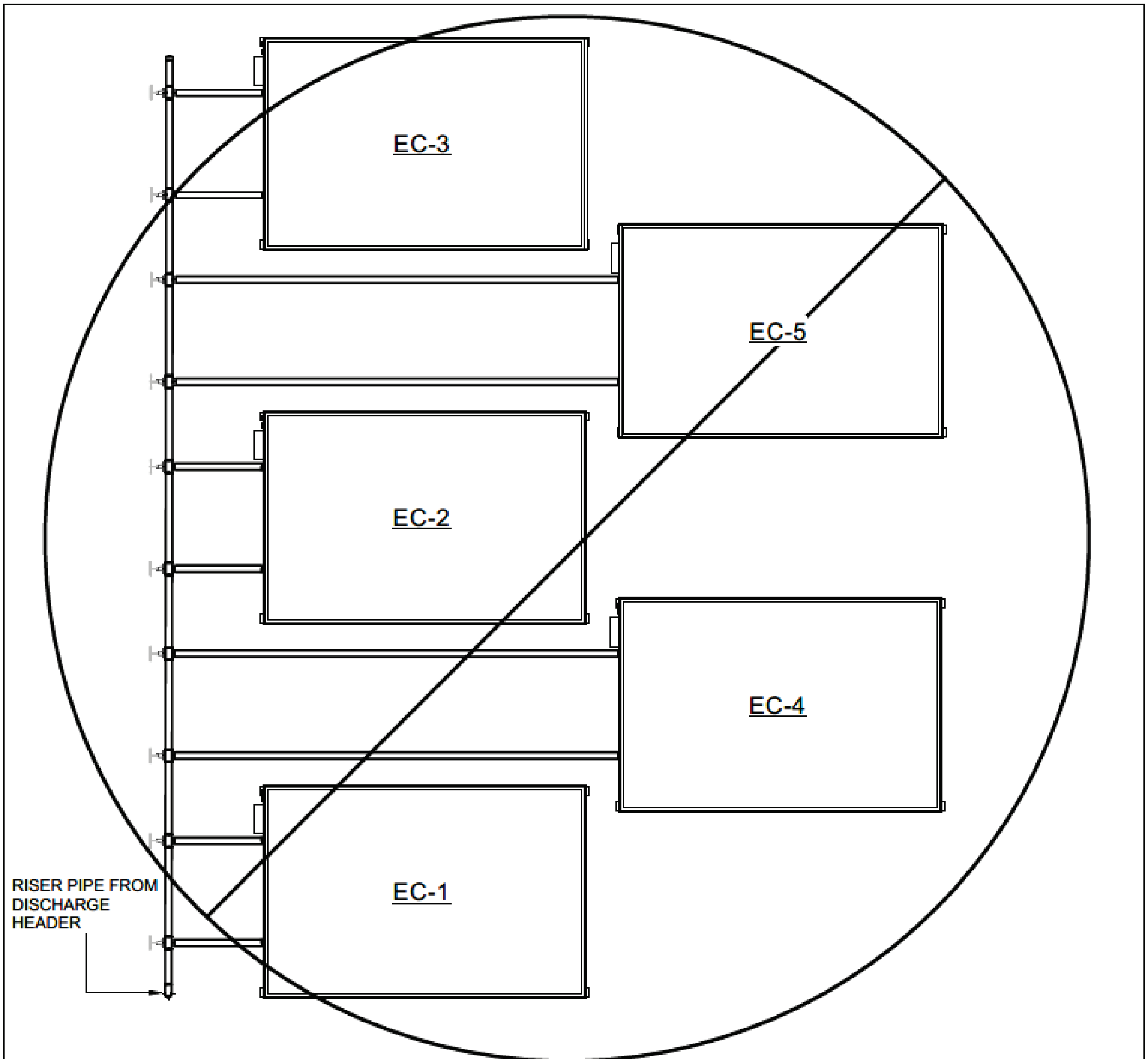
Figure 1 shows the preferred HSD piping layout where the HSD riser from the machine room enters the compressor field from a central area. **Figures 2 and 3** show HSD piping that enters the condenser field with unequal and unbalanced piping lengths which will cause imbalanced discharge flows and pressure drops which is therefore not recommended. If there is no option to have the risers enter the compressor field from a central area, extra precautions in pipe sizing must be accounted for these extra expected pressure drops as the HSD is piped to each condenser.

***Figure 1**: Plan area view of preferred H-Pattern installation showing HSD riser located in the middle of all the evaporative condensers*



NOTICE To obtain the specified thermal performance, proper airflow spacing shall be maintained between the condensers as per the *BAC Evaporative Condenser Engineering Manual* (https://baltimoreaircoil.com/resources).

***Figure 2**: Plan area view of E-Pattern installation (Not preferred because the HSD piping length to EC 3 & 5 is longer than necessary)*

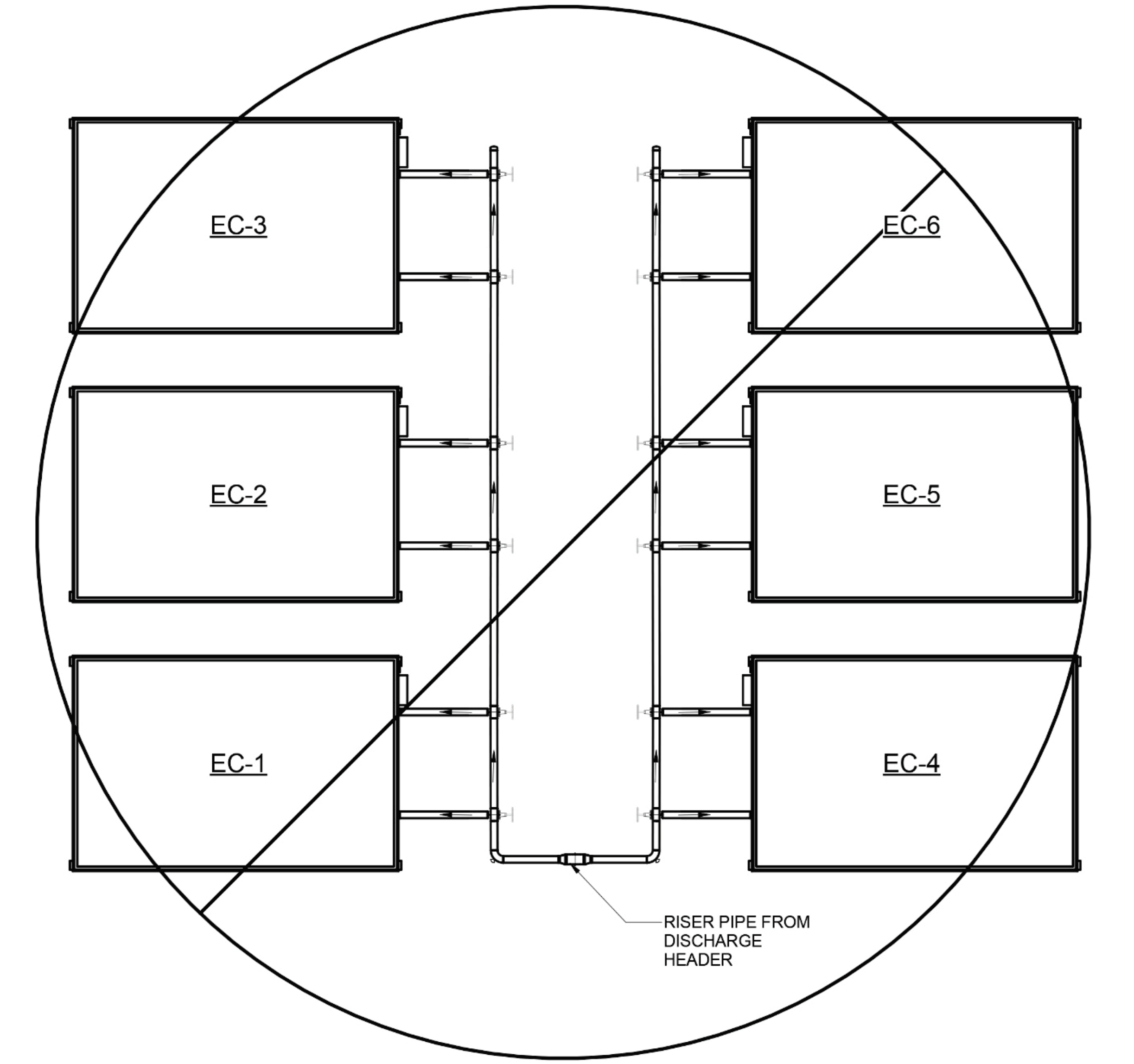


NOTES:

1. FLOW IS IMBALANCED AND WILL FAVOR CONDENSERS WITH SHORTER BRANCH PIPING RUNS, AS WELL AS THOSE CLOSER TO THE DISCHARGE HEADER RISER.

NOTICE To obtain the specified thermal performance proper airflow spacing shall be maintained between the condensers as per the BAC Evaporative Condenser Engineering Manual.

Figure 3: *Plan area view of U-Pattern installation (Not preferred because the HSD piping length to EC 3 & 6 is longer than necessary)*



NOTES:

1. FLOW IS IMBALANCED AND WILL FAVOR CONDENSER CLOSEST TO THE DISCHARGE HEADER RISER.

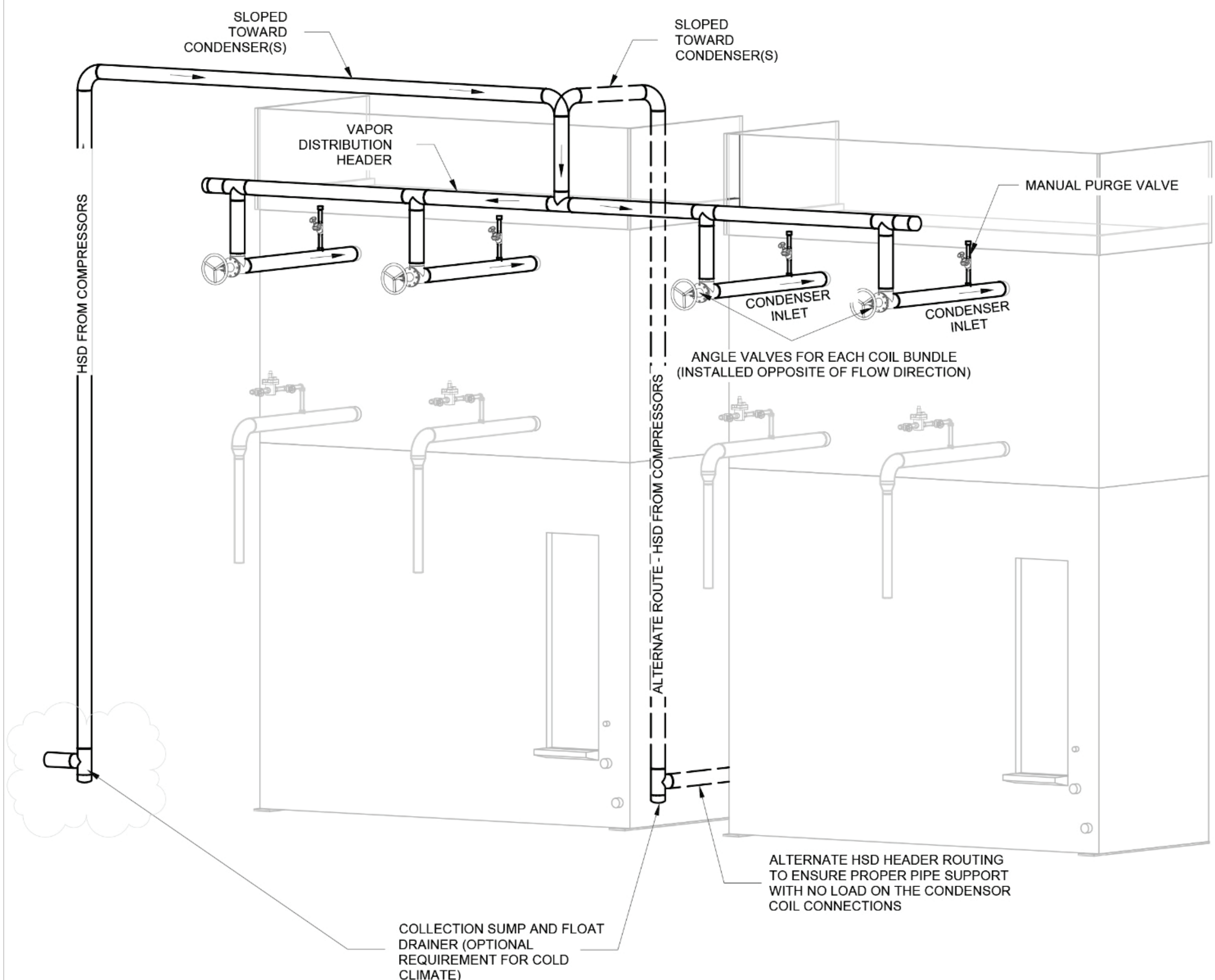
NOTICE To obtain the specified thermal performance proper airflow spacing shall be maintained between the condensers as per the BAC Evaporative Condenser Engineering Manual.

HSD layout at condenser inlet(s)

As shown in **Figure 4**, the vertical portion of the HSD should terminate above the condenser inlets and the horizontal portion should be sloped towards the condensers, so that most internal condensation during idle times in the distribution piping will drain to the condensers rather than back to the compressors. There should be an individual drop to each coil inlet connection. For a lower pressure drop, an angle service valve should be used at the bottom of the drop into the condenser inlet, in lieu of a globe valve followed by a long-radius elbow. In colder climates where ammonia vapor could condense, a collection sump and float drainer should be placed at the bottom of the vertical riser from the machine room to the roof. As shown in **Figure 4,** a ¾” seal cap service valve with a removable cap or plug on the outlet should be installed on the top of the pipe between the service valve and the condenser inlet. This valve will facilitate manual purging, as explained in the later section of this manual.

The piping layout shown here is the ideal setup, provided all the pipe runs can be adequately supported. In installations where the HSD piping configuration cannot be properly supported, it is acceptable to route the HSD header and sub-header(s) in a manner such that there is no load on the condenser coil connections.

***Figure 4**: Piping of multiple condensers - HSD riser pipe and recommended individual drops and angled service valves at each condenser shown with optional collection sump and float drainer for colder climates (Ideal configuration)*



NOTES:

1. IN INSTALLATIONS WHERE THE HSD PIPING CONFIGURATION SHOWN CANNOT BE PROPERLY SUPPORTED, IT IS ACCEPTABLE TO ROUTE THE HSD HEADER/SUBHEADER IN A MANNER THAT CAN BE REASONABLY SUPPORTED.

3. Condenser Drain (CD) line piping

There are several key considerations for condenser outlet/condenser drain piping:

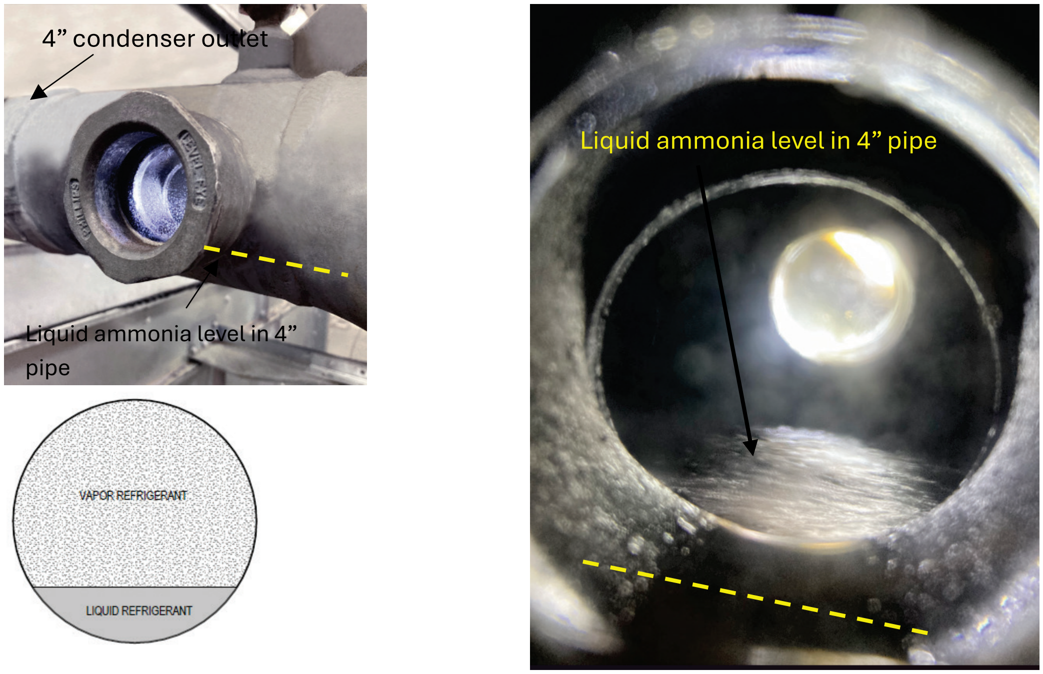
- Facilitation of gravity draining of liquid refrigerant out of the condenser
- Accommodation for automatic purging of non-condensable gases
- Low pressure drop, including through valves
- Balanced HSD and CD layout
- Open-channel flow in the CD header

Facilitation of gravity drain

All the condenser drain lines must be designed to facilitate flow of the refrigerant by gravity. This is because refrigerant vapor is drawn into the condenser bundle inlet by the volume and pressure reduction that occurs when the refrigerant is condensed within the coil bundle. At that point, all the kinetic energy is dissipated, and the liquid flow at the condenser outlet is solely driven by gravity. Therefore, gravity drainage of refrigerant is essential to maintain the correct refrigerant charge, proper flow pattern in the CD lines. This helps to prevent liquid from backing into the condensers that can greatly reduce the evaporative condensers thermal performance and may result in unstable system operation.

As shown in the **Figure 5**, a typical 4” condenser coil outlet at approximately 200-ton high-stage capacity will only have 1” of liquid NH_3 flow depth. This is analogous to water flowing in a culvert or a drainpipe and is termed open-channel or “sewer flow” which, for ammonia refrigerant systems, describes a two-phase flow where a pool of liquid flows at the bottom of the pipe, under the influence of gravity, with free-flow of refrigerant vapor above it (either flowing or stationary). Generally speaking, the flow of the vapor is opposite to the liquid flow direction, but sometimes the vapor may be stationary.

Figure 5*: Ammonia liquid flow depth in the 4" condenser coil bundle outlet*



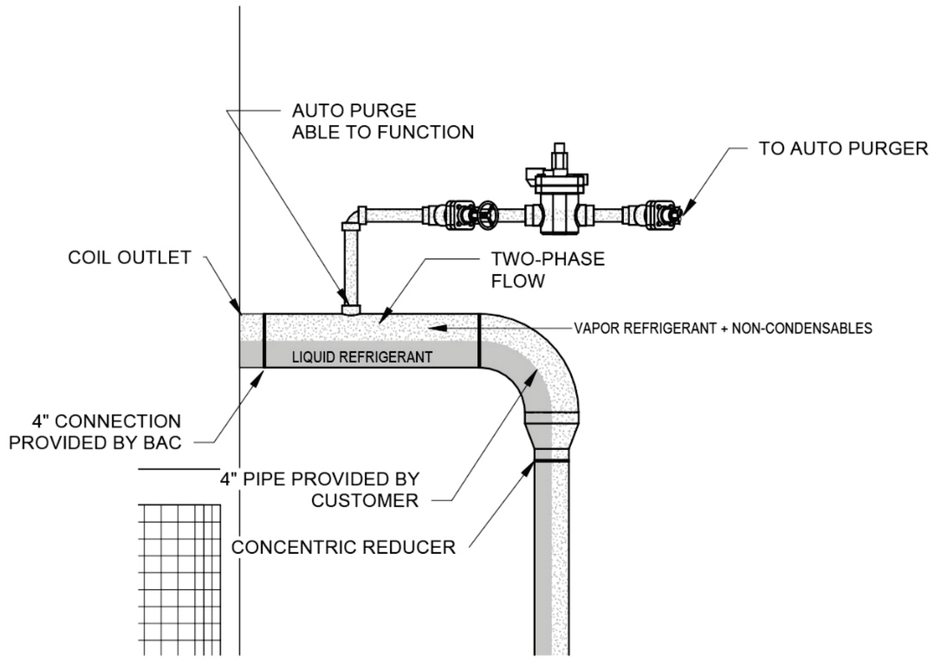
Gravity flow in CD piping must be facilitated in three ways:

1. Sloping of the piping toward the drain point, typically towards the High Pressure Receiver (HPR)
2. Ensuring that vapor being displaced by liquid flowing downstream is provided with a means of flowing back towards the condensers to avoid a vapor-locked condition, which will prevent draining.
3. Ensuring that there is sufficient drop leg height to compensate for the inlet piping, valves, and coil bundle pressure drop and thermosiphon vent line pressure drop (if applicable).

Accommodation for Automatic Purging of Non-Condensable Gases

When the system is operating, non-condensable gases are swept to the condenser outlet because the lowest refrigerant vapor velocity and temperature occur there. Therefore, the highest concentration of the non-condensable gases will accumulate at the condenser outlet, so accommodation must be made to automatically purge there. As shown in **Figure 6**, it is recommended that the CD piping at the coil bundle outlet should remain the same size as the condenser outlet. This will allow non-condensable gases to accumulate for the automatic purging out of the refrigeration system.

Figure 6: *Most effective method for automatic non-condensable purging at the condenser outlet (Recommended)*



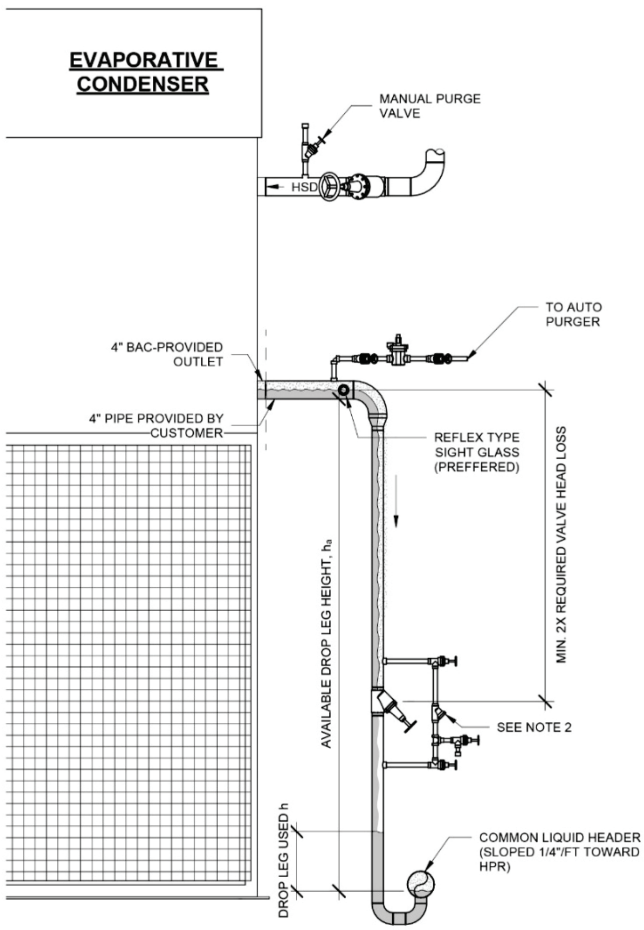
As shown in **Figure 7** below, each coil bundle outlet should have its individual automatic purge point, service valve, liquid drop leg, and P-trap. BAC recommends that the horizontal sloped pipe size should remain the same size as the condenser coil outlet (most commonly used is 4”) until after the down-turning long-radius elbow as testing has showed this is important to promote proper gravity drainage If desired, a concentric pipe reducer should be connected to the bottom of the down-turning long-radius elbow. The size of the smaller diameter drop leg pipe should match the connection size of the outlet service valve.

NOTICE It may appear efficient to use an angle valve in lieu of one of the elbows to construct the P-trap. However, BAC strongly advises against this practice because dirt and mill scale will accumulate at this low point, embedding in the valve seat preventing a tight seal when closed.

Liquid refrigerant piping downstream of the P-trap should be sized using open-channel flow equations presented in appendix-A. This will allow the free flow of vapor above and gravity drainage of the condensed ammonia below. **Table B-5** lists the maximum Tons and NH3 mass flow rate for different pipe sizes of common CD header while maintaining the open-channel flow.

Installing a reflex-type sight glass on the side of the horizontal condenser outlet pipe is an excellent diagnostic tool to determine liquid ammonia backup into the condenser coil.

***Figure 7**: Condenser drain line piping with reduced size shut-off valve (BAC recommended)*



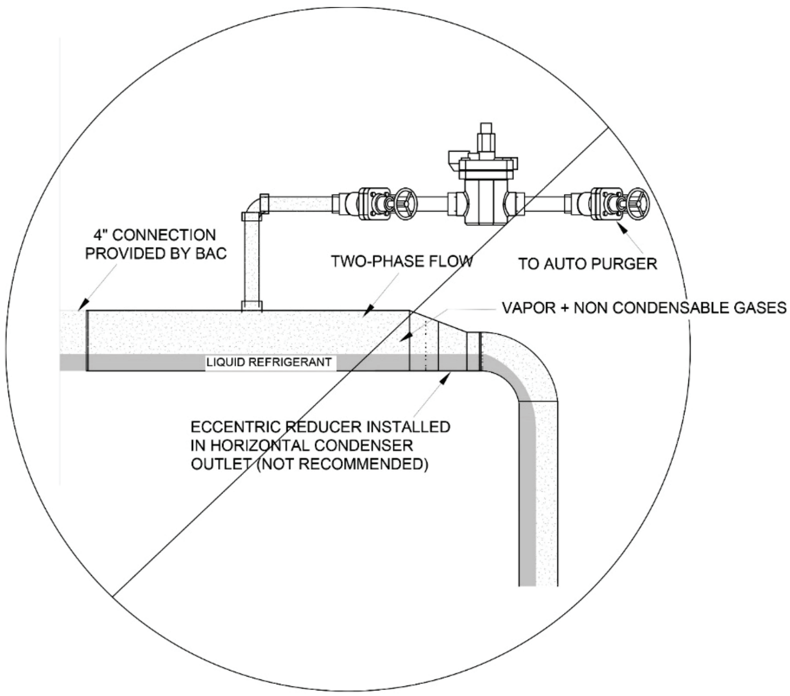
NOTES:

1. *h* IS THE HEIGHT OF LIQUID BACKED UP INTO THE DROP LEG. ITS HEIGHT IS GOVERNED BY THE PRESSURE DROP FROM THE HSD HEADER EQUALIZER CONNECTION (NOT SHOWN) TO THE COIL OUTLET (INCLUDES PIPING AND VALVES FROM EQUALIZER CONNECTION TO THE COIL AND THE COIL ITSELF). LARGER PRESSURE DROPS RESULT IN LARGER VALUES OF *h*.
2. CHECK VALVE FOR HYDROSTATIC OVERPRESSURE PROTECTION.

NOTICE **Do not use eccentric reducer in the horizontal piping section of the condenser outlet.**

Although it has been common practice in the past to install an eccentric reducer in the horizontal line at the condenser outlet. Thorough in-house testing has shown not to do this because the head for flow through the eccentric reducer is achieved by adding to the flow depth in the horizontal pipe which will cause liquid backup into the lower rows of the condenser coil.

***Figure 8**: Based on testing at the BAC ammonia lab, it is no longer recommended to reduce the pipe size in the horizontal pipe coming out of the condenser as liquid will back into the lower rows of the condenser. Therefore, as shown in Figure 7, reduce the pipe from 4" to the properly sized flow size in the vertical line after the 4" down-turning long-radius elbow.*



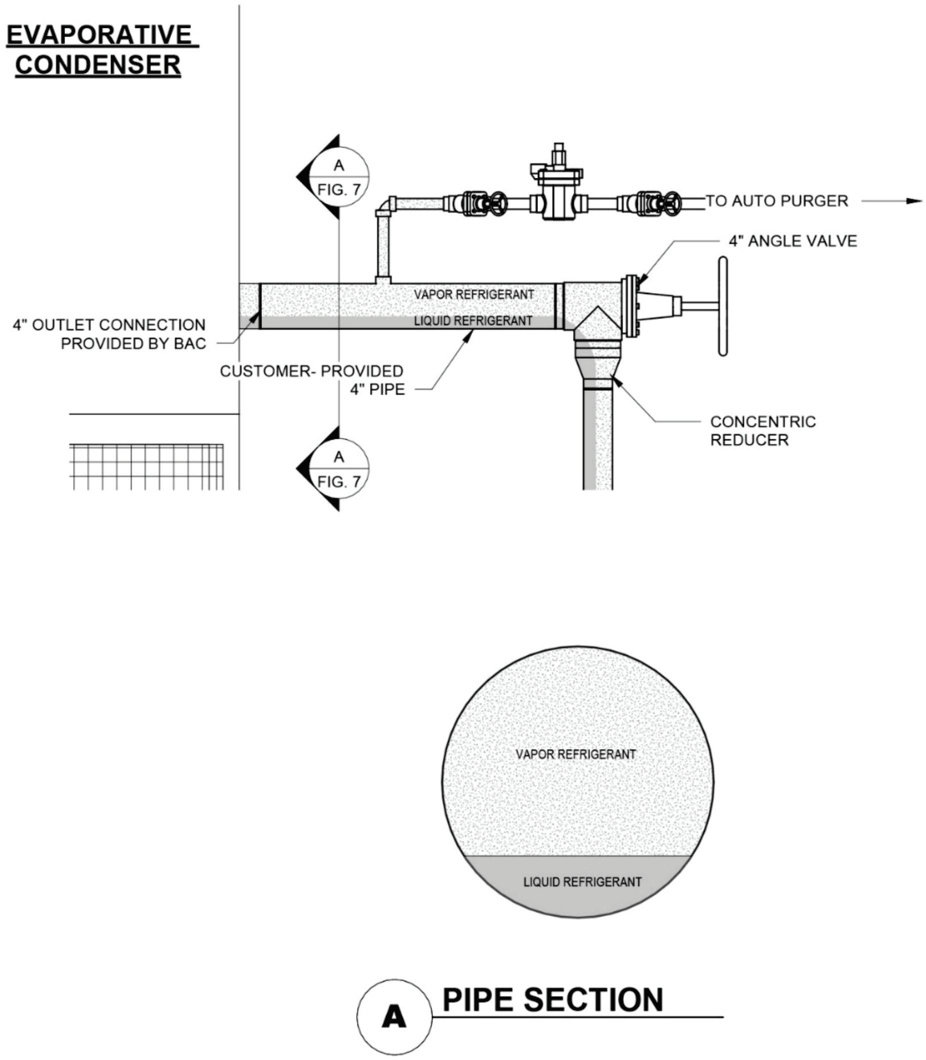
Low pressure drop through the outlet service valve

Using the valve loss equation 1 in the appendix-A, the service valve at the outlet of the condenser coil bundle should be sized for a pressure drop of less than 0.1 psi, which requires less than 5 in. of liquid NH3 static head at the design flow rate.

As shown in **Figure 7**, ideally, the service valve should be mounted chest-high in the vertical down pipe for ease of access while maintaining a minimum 1 ft. drop requirement below the 4" down-turning long-radius elbow to allow flow through the valve at all conditions and avoid liquid backing up into the coil bundle.

Alternatively, as shown in the **Figure 9**, a 4" angle valve can be used, instead of a long-radius elbow and if desired a concentric reducer can be used after the 4" angle valve to reduce the drop leg pipe size.

__Figure 9__: Condenser drain line piping with 4" angle valve at the outlet



Balanced HSD and CD Layout

As mentioned in the earlier HSD section, a balanced HSD and CD line layout for the multiple condensers has the benefit of reducing the pipe-run length of any given CD header section and therefore reducing the total rise in the sloped CD header. If a balanced layout of the main CD vertical line to the receiver cannot be possible, the required slope (rise in CD header height at the farthest point) will detract from the available liquid drop leg height for the farthest condenser(s).

4. Increased condenser capacity and its effect on the pressure drop

During the past decades evaporative condensers have become larger, longer, and more efficient, which increased their capacity and coil pressure drop. The most common improvements to evaporative condenser design are listed below:

- Larger fan(s) and/or higher horsepower per fan to move more air through the coil bundle(s).
- Optimized tube spacing and other heat transfer enhancement features.
- Increase in number of tubes passes and the length of the coil bundles – both resulting in longer circuit length, which caused higher capacity and increased pressure drop.

All these improvements have resulted in higher heat rejection capacity per condenser, which means higher mass flow rate through the coil bundle which has a direct impact on the coil pressure drop. As listed in **Table 4**, as the mass flow rate through the condenser increases so does the pressure drop. Therefore, it is highly recommended to consult with your local BAC representative for the actual coil pressure drop for your specific application and the size the condenser drop leg height accordingly.

Furthermore, the drop leg height should be sized for all ambient and operating conditions. An evaporative condenser that does lower heat rejection at nominal conditions can do much higher heat rejection at lower wet bulb temperatures and will have higher coil pressure drop. It is also prudent to check the pressure drop at the minimum anticipated condensing pressure as well.

			Nominal (96.3°F CT / 78.0°F WB)			Winter (66°F CT / 35.0°F WB)			Arid (95.0°F CT / 70.0°F WB)		
Model	**Nominal Length**	**Fan HP**	**MBH**	$\dot{m}$	**ΔP**	**MBH**	$\dot{m}$	**ΔP**	**MBH**	$\dot{m}$	**ΔP**
	[ft]	**[hp]**	**[MBH]**	**[#/min]**	**[psi]**	**[MBH]**	**[#/min]**	**[psi]**	**[MBH]**	**[#/min]**	**[psi]**
VXMC 620	18	30	6,703	232	0.56	6,644	216	0.80	8,451	292	0.90
C2664N	18	45	8,119	281	0.56	8,048	262	0.80	10,237	353	0.90
VCA-957	18	60	10,344	358	0.88	10,253	334	1.26	13,042	450	1.42
CXV-481	18	45	7,330	254	1.59	7,270	237	2.28	9,243	319	2.57
VCA-N1204	24	80	13,014	451	1.50	12,899	420	2.15	16,408	566	2.42
CXV-T944	26	75	14,387	498	0.77	14,269	464	1.10	18,141	626	1.24
VRC-0717A-1218N-NA	18	75	10920	378	0.98	10821	352	N.A.	13767	475	1.58

Table 4*: Historic preview of condenser growth and coil pressure drop*

NOTICE Some of the listed models in **Table 4** are legacy models and are no longer offered for sale, they are listed here just for comparison purposes. The pressure drops listed in the table are for reference purposes only, consult with your local BAC representative to get the actual pressure drop for the specific evaporative condenser at your design conditions.

5. Liquid drop leg height sizing

Heat rejection and pressure drop in each condenser coil bundle can vary depending on the spray pump and fan operation, vapor flow distribution, condenser(s) layout, and location of the coil bundle inside the condenser. Therefore, minimum liquid drop leg height shall be calculated based on the condenser coil bundle(s) and external pressure drops (i.e., pressure drop in HSD, isolation valve(s), and branch piping). Coil bundles with more heat rejection will have a higher pressure drop which will result in liquid stacking up into the drop leg.

As described earlier, it is important to use the actual pressure drops through the coil, HSD line, fittings, and valves at the design conditions to accurately determine the minimum liquid drop leg height at the design conditions. Using rule-of-thumb estimates or uncorrected tabular data can result in insufficient liquid drop leg height. Your local BAC representative can provide the estimated pressure drop through the coil bundle based on the provided data.

As shown in **Figure 10**, the receiver should be equalized to the lowest pressure point of the HSD header. The equalizer has no or minimal flow during most operating hours, accordingly it will minimally affect the drop leg height. Therefore, the total pressure drop that should be restored by the liquid height column:

$$\Delta P_{total} = \Delta P_A + \Delta P_B + \Delta P_C + \Delta P_D$$

ΔP_{total} = **Minimum** pressure drop restored by the liquid height in the drop leg, psi

ΔP_A = Pressure drop in discharge header from the branch connection to the equalizer connection, psi

ΔP_B = Pressure drop in the condenser inlet branch pipe, fittings, and valves, psi

ΔP_C = Coil bundle pressure drop, psi

ΔP_D = Pressure drop in outlet pipe, fittings, and valve, psi

6. Condenser installations with thermosiphon oil cooling

If the compressors are cooled by thermosiphon oil cooling, it is common practice to use HPR as the thermosiphon vessel. To accommodate this additional function, the receiver will have to be:

- Elevated above the compressors to provide sufficient static head for thermosiphon flow.
- A dedicated bottom outlet connection, Thermosiphon Supply (TSS) to supply liquid ammonia to the oil coolers.
- A dedicated Thermosiphon Return (TSR) connection above the highest liquid level in the vessel, to allow the vapor/liquid mixture to return from the oil coolers.
- Separation space above the operating liquid level for liquid / vapor separation.
- For thermosiphon cooling, the equalizing line becomes the Thermosiphon Vent line (TSV) and is upsized to convey the additional ammonia vapor to the HSD header where it will be directed into the condensers inlet to be condensed.

As explained further in the later section, like equalizer line, TSV should also be connected to the lowest pressure point in the HSD header. This will keep the drop leg height requirement to a minimum. The maximum pressure drop in a TSV line should not exceed 0.25 psi at the design conditions.

NOTICE Always consider the additional thermosiphon mass flow in the discharge header to the nearest condenser.

For thermosiphon oil cooling there will usually be a vapor flow through the TSV vent line which will raise the receiver pressure. The summary equation in the above section remains the same with the addition of the pressure drop of the thermosiphon vent pipe, fitting, and valves:

$$\Delta P_{total} = \Delta P_A + \Delta P_B + \Delta P_C + \Delta P_D + \Delta P_{TSV}$$

ΔP_{TSV} = Pressure drop of the; thermosiphon vent pipe, fitting and valve, psi

ΔP_{total} = **Minimum** pressure drop restored by the liquid height in the drop leg, psi

After calculating the total pressure drop, it should be converted into the static head using the following equation:

$$h_{min}(inches) = \frac{\Delta P_{total}\left(\frac{lb_f}{in^2}\right) * 12\ \left(\frac{in}{ft}\right) * 144\left(\frac{in^2}{ft^2}\right) * \left(\frac{g_c\left(\frac{lb_m.ft}{lb_f.sec^2}\right)}{g\left(\frac{ft}{sec^2}\right)}\right)}{Density\ of\ liquid\ refrigerant\ at\ T_{sat}\ \left(\frac{lb_m}{ft^3}\right)}$$

h_{min} = Minimum liquid drop leg height, inches

This is the minimum drop leg height required to ensure that during routine system operation liquid does not back up into the condenser coils. BAC recommends adding the safety factor of an additional 25% to the minimum drop leg height to account for unexpected ambient and operational conditions.

$$h_{rec}(inches) = h_{min} * 1.25$$

h_{rec} = Recommended drop leg height, inches

The calculated values with 25% safety should be considered absolute minimum and it is good practice to maximize the drop leg height during the installation. Additional drop leg height does not hurt performance and allows for varying loading, unit startup, and unaccounted for piping issues.

Drop leg height calculation example

For Ammonia with 1 psi total pressure drop at 95°F condensing temperature, the minimum and recommended drop leg heights are calculated below:

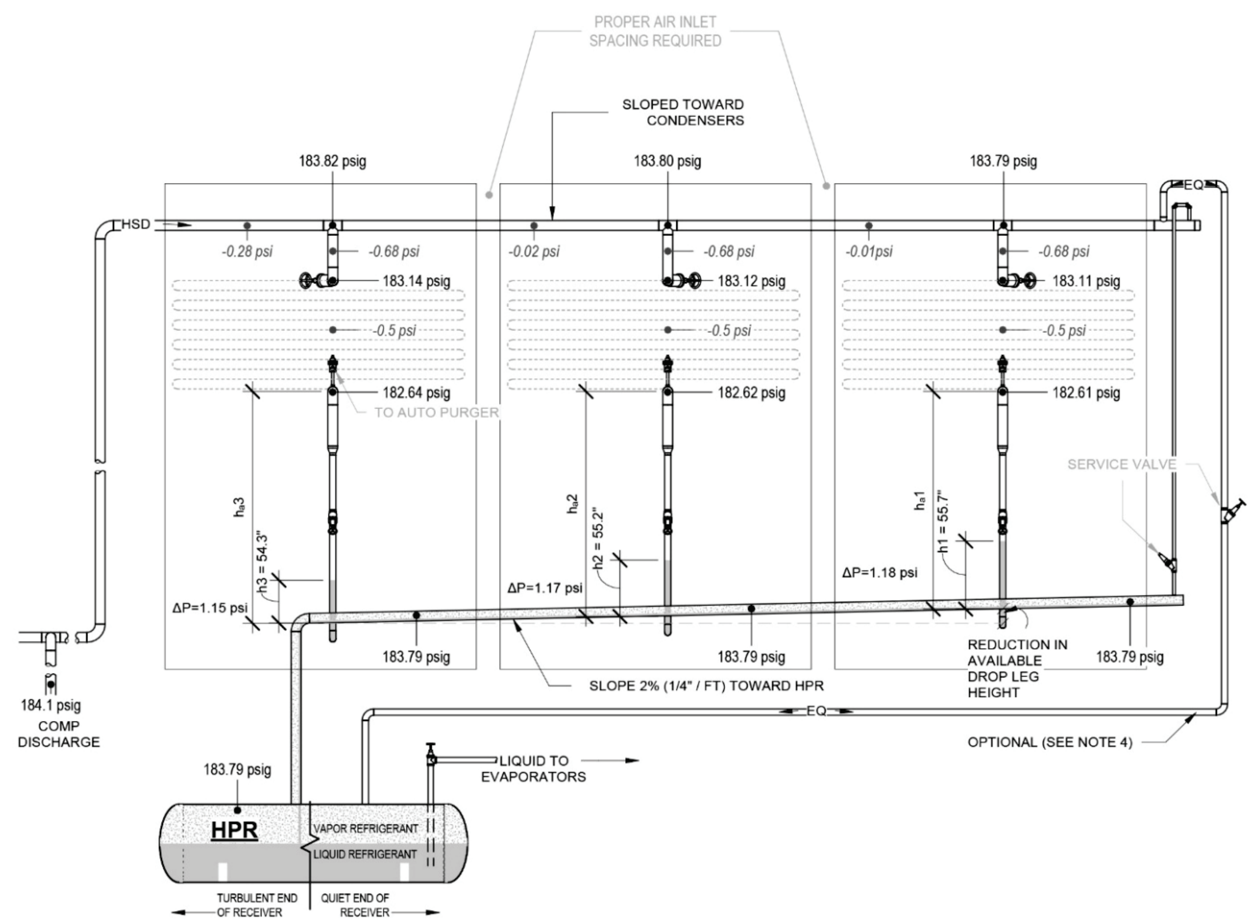
$$h_{min}(inches) = \frac{1 * 12 * 144}{36.67} = 47\frac{1}{8} inches$$

$$h_{rec}(inches) = 47.12 * 1.25 = 59\ inches$$

It is worth mentioning that for an evaporative condenser with 2 psi total pressure drop at 95°F condensing temperature, which can happen when attempting to run one of several condensers in the wintertime, the minimum drop leg heights will double.

$$h_{min}(inches) = 47\frac{1}{8} * 2 = 94\frac{1}{4}\ inches$$

***Figure 10**: Multiple condensers draining to high – pressure receiver*
Example of drop leg calculation with individual pressures and liquid drop leg height

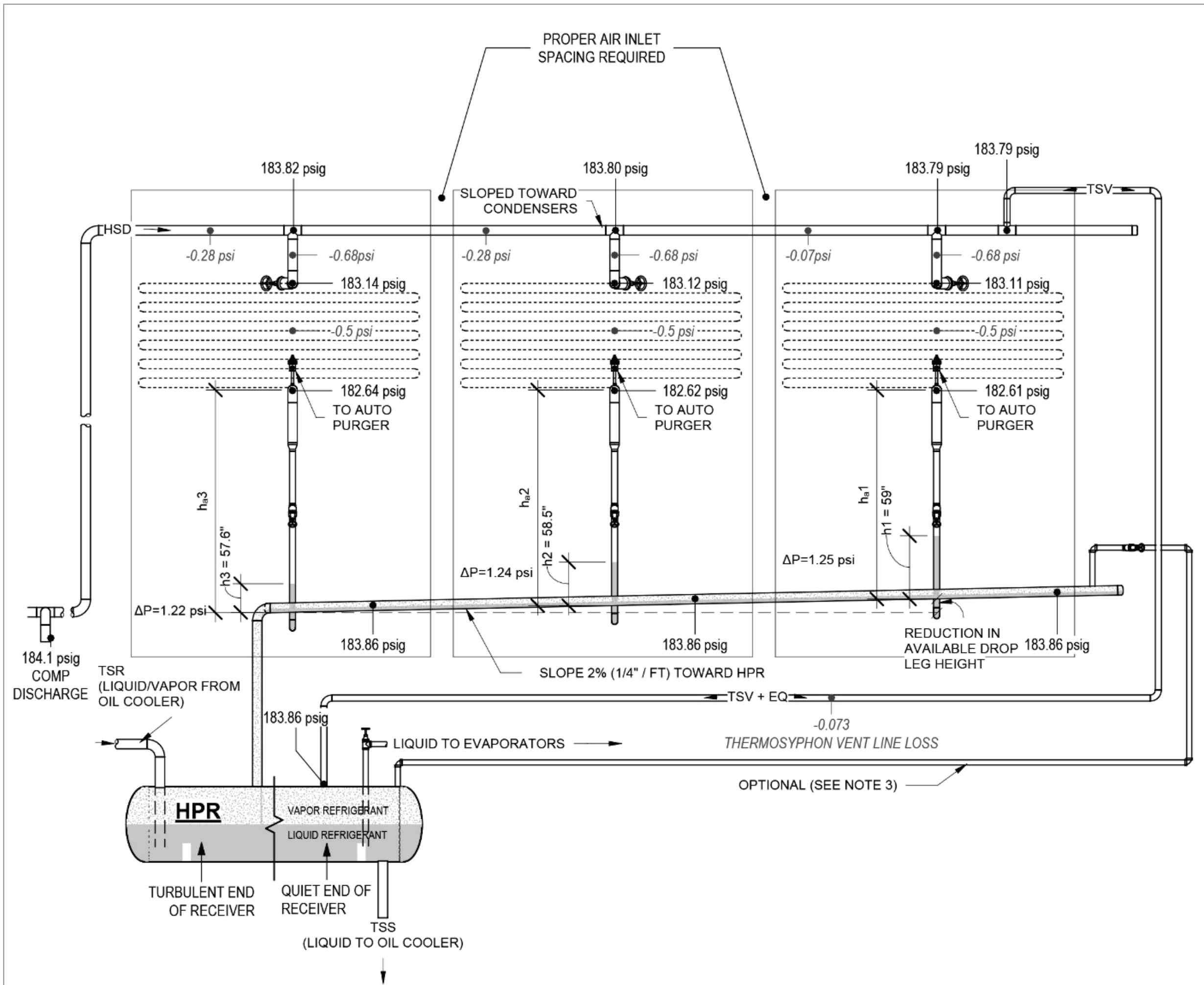


NOTES:

1. H_X = DROP LEG LIQUID HEIGHT IN INDIVIDUAL DROP LEG AT CONDITION SHOWN
2. H_{aX} = AVAILABLE DROP LEG HEIGHT AT INDIVIDUAL DROP LEG
3. PROPER SPACING IS REQUIRED WHER AIR INLETS ARE BETWEEN CONDENSERS.
4. OPTIONAL CD HEADER VENT IF OPEN-CHANNEL FLOW IS QUESTIONABLE
5. ALL PRESSURES ARE IN PSIG

NOTICE If the CD header is not sloped properly or sized large enough, then to ensure a continuous vapor space at the top of the header a valved vent line should be provided at end of the CD header at 12 o'clock position. For a non-thermosiphon system, this vent line can connect to the lowest pressure point of the discharge header or the vessel that the CD header drains into. For thermosiphon cooled compressors, it should vent to the vessel that the CD header drains into, as shown in **Figure 11**.

***Figure 11**: Example of drop leg calculation with individual pressures with Thermosyphon cooling for multiple condensers*



NOTES:

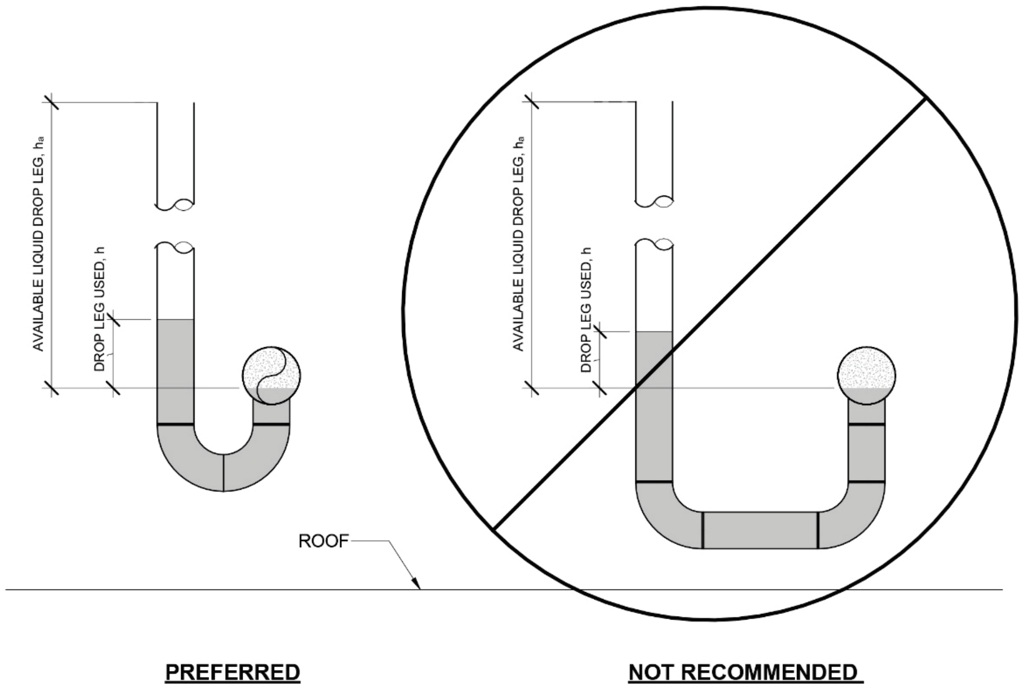
1. H_X = DROP LEG LIQUID HEIGHT IN INDIVIDUAL DROP LEG AT CONDITION SHOWN
2. H_{aX} = AVAILABLE DROP LEG HEIGHT AT INDIVIDUAL DROP LEG
3. OPTIONAL CD HEADER VENT IF OPEN-CHANNEL FLOW IS QUESTIONABLE.

7. P-traps at the bottom of the drop leg

As shown in **Figure 12**, P-trap at the bottom of the drop leg (formed from two long-radius elbows) should connect directly to the bottom of the horizontal CD header. A spacer may be required between the long-radius elbows so the drop leg clears the CD header. An individual P-trap at the bottom of the drop leg establishes a liquid seal which allows the liquid column to develop into the drop leg to offset the coil bundle, HSD line, thermosiphon vent (where applicable), and other pressure drop differences. Furthermore, the liquid seal will not allow the non-condensable gases to flow to the HPR and during the system operation the non-condensable gases will accumulate at the condenser outlets eliminating the need for a purge point at the HPR. Any non-condensable gases inadvertently trapped in the receiver during commissioning or service will migrate to the condensers through the equalizing line and be purged at the condenser outlet(s) through the automatic purger. If P-traps are not present, then there is no liquid seal to initiate the manometer effect on a lower pressure side (i.e. condenser side) of the P-traps.

As explained in **Figure 12**, it is recommended to keep the liquid trap as tight as possible (with long-radius elbows). There is no advantage to adding an extension to the depth of the outlet side of the P-trap as the effective drop leg depth terminates at the liquid level in the drain header. In case of sudden drop in pressure inside the condenser, such as when spray pump turns on, a deeper P-trap will be more difficult to clear out, compared to a standard P-trap, which will hinder the vapor flow up the drop leg for pressure equalization.

***Figure 12**: Condenser trap draining - comparison of standard and extended U-bend*



NOTES:

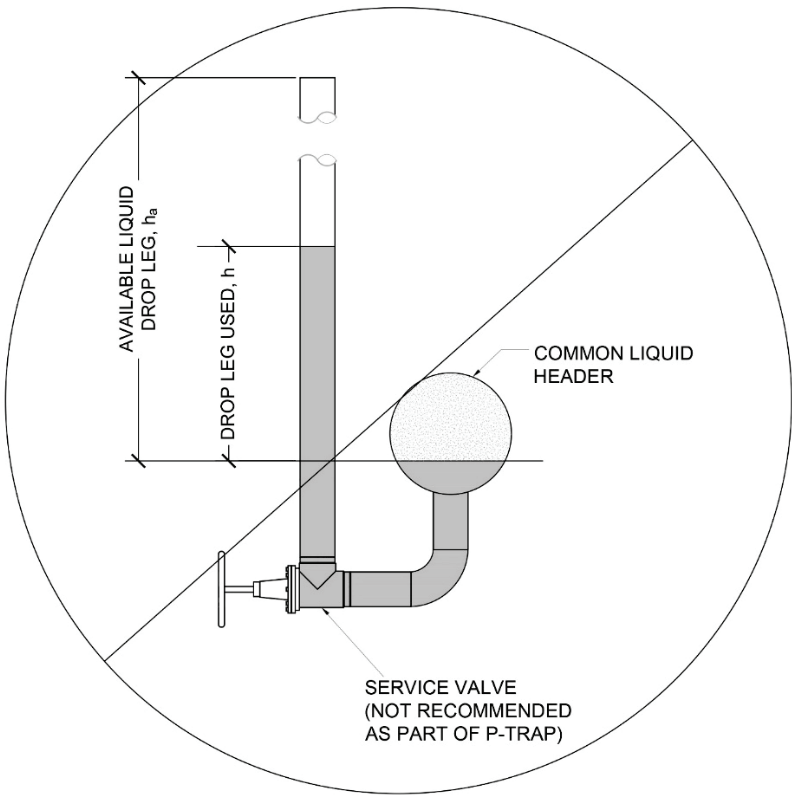
1. *h* IS THE HEIGHT OF LIQUID BACKED UP INTO THE DROP LEG. ITS HEIGHT IS GOVERNED BY THE PRESSURE DROP FROM THE HSD HEADER EQUALIZER CONNECTION TO THE COIL OUTLET (PIPING AND VALVES FROM EQUALIZER CONNECTION TO THE COIL AND THE COIL ITSELF). LARGER PRESSURE DROPS RESULT IN LARGER VALUES OF *h*.
2. THERE IS NO ADVANTAGE TO ADDING A PIPING EXTENSION BETWEEN THE ELBOWS OR THE TEE AND OUTLET ELBOW OF THE TRAP. HOWEVER, DEPENDING ON THE SIZE OF THE CD HEADER OR ITS LOCATION IN A PIPE RACK BELOW THE CONDENSER, A SHORT LENGTH OF SUCH PIPING MAY BE REQUIRED.

NOTICE Although there is no advantage of adding any extension to the outlet side of the P-trap, however, depending on the size of the CD header or its location in a pipe rack below the condenser, a short length of such piping may be necessary. Such piping will not have any negative impact on the system operation.

NOTICE Do not use angle service valve to construct the P-trap

As shown in **Figure 13**, it may appear efficient to use an angle valve in lieu of one of the elbows to construct the P-trap. However, BAC does not recommend this practice because dirt and mill scale will accumulate at this low point, embedding in the valve seat preventing a tight seal during the valve closure.

Figure 13*: It is not recommended to use an angle or any valve to form the P-trap at the bottom of the liquid condensate drop leg*



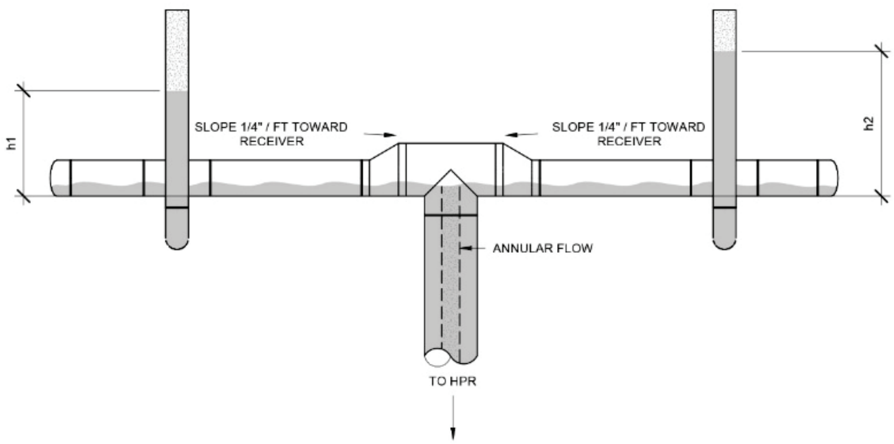
8. Open-channel flow in the liquid Condenser Drain (CD) header

Liquid refrigerant flows up from the P-traps into the sloped header, which flows down by gravity towards the HPR (or TSR). For liquid to flow through a horizontal pipe by gravity, the pipe must either be sloped, or the liquid must accumulate in the pipe until an internal slope is formed on the surface of the flowing liquid. It is recommended that a 2% (i.e., ¼" per foot) slope be used for all horizontal CD piping from the furthest condenser to the vertical drop to the receiver (similar for CD branch headers when present). As shown in **Figure 14**, the depth of liquid flow inside the header should be less than 50% of the inner pipe diameter. This is beneficial for the following reasons:

1. The space above the liquid surface allows free vapor flow ensuring that the entire drain header is at receiver pressure.
2. In case of a rapid drop in pressure inside the condenser coil bundle, the drop leg P-trap will "burp" quickly and vapor from the drain header will flow up the drop leg and restore the coil bundle pressure. This will prevent condenser flooding with liquid in such conditions.
3. It is less likely that, under unsteady conditions, a liquid slug will fill the entire pipe cross-section and prevent a vapor lock in the drain header, which is discussed later in this section.

Table B-5 in appendix-B lists the maximum Tons and NH_3 mass flow rate for open-channel flow in different CD header pipe sizes.

***Figure 14**: Open-channel "sewer" flow in common CD header*



NOTES:

1. h1 AND h2 ARE VARYING LIQUID LEG HEIGHTS REFLECTING VARYING PRESSURE DROPS UPSTREAM.
2. WHEN MULTIPLE CD HEADERS ARE MERGING, THE BULLHEAD T AND VERTICAL PIPE SHOULD BE SIZED APPROPRIATELY TO REFLECT THE TOTAL REFRIGERANT MASS FLOW RATE.

It is recommended that a CD header is sloped 2% towards the HPR, however if height constraints cannot allow uniform 2% slope then consider 1% at the far ends of the drain headers and transition to 2% close to the vertical drain end where there is more refrigerant mass flow rate.

As shown in **Figure 15**, a liquid header with insufficient slope and/or inadequate pipe size will cause the CD header to fill more than 50%, causing the formation of a liquid slug. If liquid slug is formed, it will hinder the free flow of vapor inside the header and the vapor pocket behind the slug up to the header endcap, will inhibit gravity driven liquid flow to the HPR.

This is why BAC recommends that the liquid level in the horizontal header should not exceed 50% of the internal diameter at all conditions. If job site constraints cause the liquid level to be above

50%, then a valved, ¾" vent line shall be installed from the top of the CD header to the lowest pressure point on the HSD header, near the connection point of the equalizer line. The vent line will eliminate vapor lock and allow liquid slugs to drain or disperse.

The CD header should be open-channel flow from the far end(s) of the header to the receiver, allowing for gravity draining while simultaneously allowing vapor to freely flow from the receiver if there is any volume imbalance of the flows into and out of the receiver. This is especially important in the vertical piping where a vapor core must be able to flow upward against a downward annular column of liquid and, in the throat of a service valve at the receiver inlet. If that vapor flow is interrupted, then the header pressure will fluctuate affecting the manometer balance on each side of the P-trap. By keeping the liquid header flow level below 50%, not reducing the vertical pipe size, and using a wye or angle type service valve, equalizing the header to the receiver should not be needed.

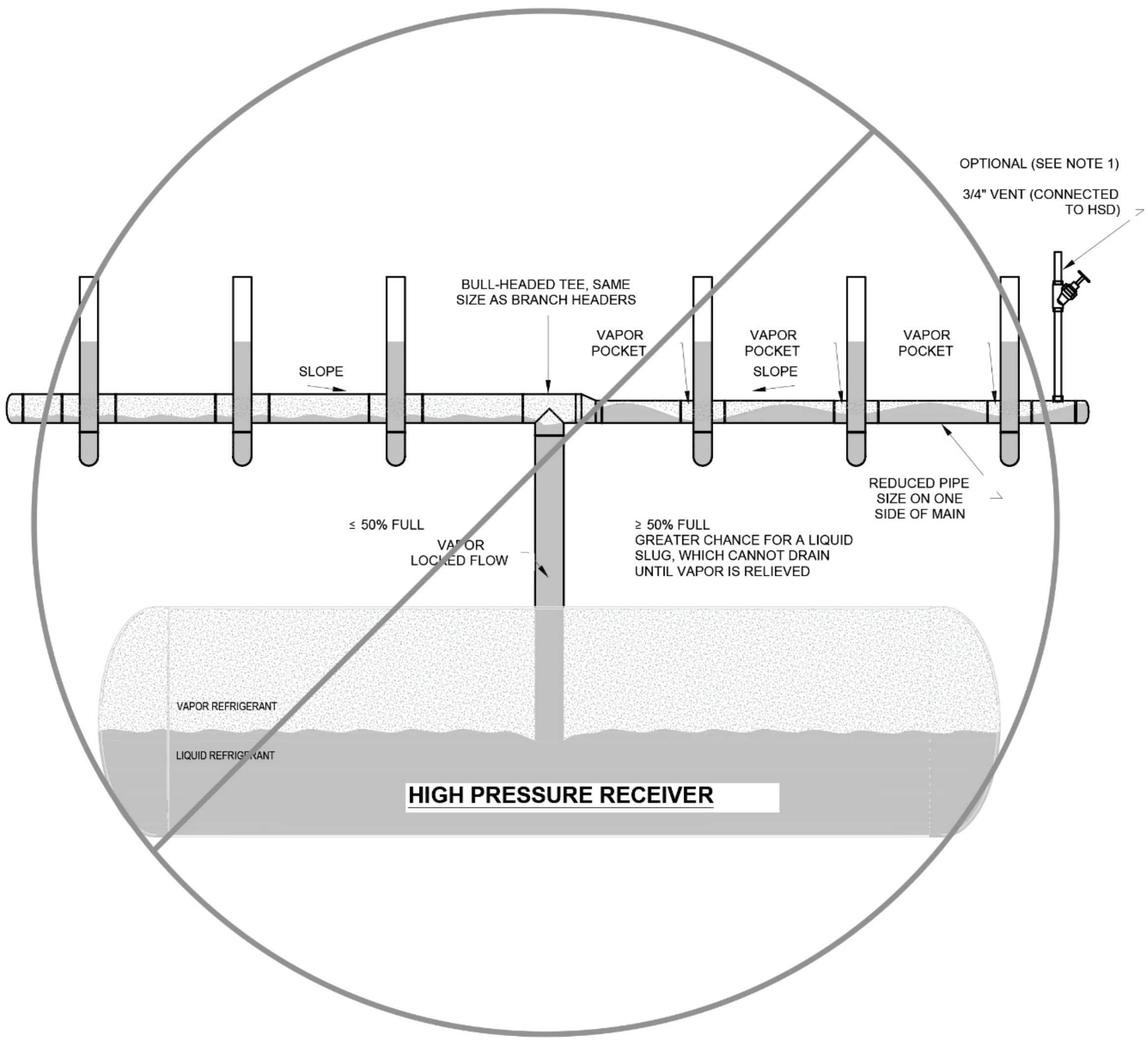
It is the condenser drop leg side of the P-trap that accommodates the pressure drop of the condenser coil, thermosiphon vent line loss (which elevates the receiver and header side of the P-trap pressure) and any pressure difference between the coil inlets and the equalizer connection point.

If the system has thermosiphon oil cooling and the CD header is either not appropriately sloped or sized, then a valved vent line must be connected to the top of the vessel that the CD header drains into. This vessel may be a thermosiphon vessel or a combination thermosiphon vessel and HPR. The reason for this alternate connection point is the pressure drop of the thermosiphon vent line will slightly raise the receiver pressure and by connecting the vent to the receiver, this eliminates that pressure differential and flow resistance, resulting in only gravity being the motivating force for the liquid drain.

As shown above in **Figure 14**, when the liquid from the horizontal drain header flows into the vertical pipe to the receiver below, the liquid flow transitions from a "lazy river" in the horizontal piping to an annular sheet inside the perimeter of the vertical pipe. Therefore, it is important that if two legs of the horizontal drain headers are combined prior to turning vertical, the fittings and vertical pipe size should increase in diameter to reflect the combined mass flow and ensure annular flow in the vertical pipe and space for vapor to travel up. It is not recommended to reduce the vertical pipe size because at high refrigerant mass flow rates conditions, liquid may not be draining freely back to the HPR.

The total pressure drop in the piping run down to the receiver at design flow must be lower than the static head available in the vertical drop, or liquid will be forced to back up into the horizontal drain header(s) at the condensers.

***Figure 15**: Improper piping of multiple condensers*



NOT RECOMMENDED

NOTES:

1. OPTIONAL CD HEADER VENT LINE IF OPEN-CHANNEL FLOW IS QUESTIONABLE.

9. Additional considerations for multiple condenser installation

Balanced HSD and HRP location

As stated previously, for multiple condenser installations it is desirable that the HSD and CD risers and high-pressure receiver be located at a balanced/central location relative to the condenser field to minimize the variance in pressure drop and drop leg height requirements. This will result in balanced refrigerant flow to and from all the condensers which will reduce the variation in required drop leg heights.

Achieving the recommended ¼". per ft. slope with long CD runs may be difficult with the receiver located at the extreme end of a long run, therefore a central HPR location can avoid a situation where slope must be made significantly shallower.

Location of the equalizer/TSV line and its effect drop leg height

As shown in **Figure 10 & 11,** for a multiple condenser installation, the condenser located farthest from the compressor will have a higher HSD line pressure drop compared to the one closest to the compressor. Because the CD horizontal header is also sloped, with its highest elevation at the furthest condenser, the available height for the drop leg will be reduced for that condenser, which increases the need to connect the receiver or thermosiphon vessel's equalizing line to the lowest pressure point within the HSD piping.

Venting

Any receiver(s) must be vented properly, which consists of either an equalizing line or thermosiphon vent line. Sizing methodology for either of these lines is explained in the following sections.

10. Equalizer line sizing

The purpose of the equalizing line is to keep the high-pressure receiver at the lowest pressure. Within the discharge piping it is typically at the end of the longest HSD pipe run. This equalization is critical for the proper drainage of the condenser and ensures that drop legs are used effectively and no liquid backs up into the coil bundle.

During steady-state operation, the liquid volume draining into the receiver is equal to the liquid volume leaving the receiver to feed the downstream loads. In this instance, there is no displaced vapor to flow through the equalizing line. Because the liquid flowing from the condensers may have a slight, typically less than 1°F, amount of subcooling, a negligible amount of the discharge gas may be drawn towards the cooler liquid in the receiver. There is also the consideration of heat gain on the receiver from solar or ambient heat gain (if situated outdoors) or from elevated machine room temperatures (if placed inside), in which case, vapor will flow toward the HSD piping.

The largest and easiest calculated flow occurs when the liquid flow to the plant abruptly stops, and the condensers continue to drain. The displaced vapor from the continued draining must pass through the equalizing line. This vapor flow rate is easily determined by multiplying the maximum design liquid flow rate by the ratio of the liquid density/vapor density. With this vapor mass flow rate, the equalizing pipe, fittings, and valve should be sized for a total pressure drop of 0.25 psi, which equates to 1' of liquid column rise in the drop leg. **Table B-6** in appendix-B lists the maximum system capacity for different equalizer line pipe sizes.

Equalizing line flow rate based upon instant shut down at design load:

Liquid volume in from condensers = Displaced vapor volume out of the receiver

$$m_l \times {}^{\rho_v}/_{\rho_l} = m_v$$

$m_l = Design\ system\ mass\ flow\ rate, lbm/min$

$\rho_v = Vapor\ density, lbm/ft^3$

$\rho_l = Liquid\ density, lbm/ft^3$

$m_v = Design\ equalizer\ line\ mass\ flow\ rate, lbm/min$

For multiple condenser installations, connect the equalizer line adjacent to the lowest pressure point in the HSD, which is typically the inlet to the condenser furthest away from the receiver. This will ensure that the maximum individual drop leg height will be used to offset only the furthest-away condenser coil, and its dedicated inlet pipe, fittings and valve pressure drop and not the common HSD piping and valves pressure drop(s).

11. Thermosiphon vent sizing

As described above, compressors using thermosiphon cooling produce two vapor streams to the condensers:

1) Main compressor discharge flow
2) Vapor generated in the oil cooler

This separate vapor flow from the oil cooler can vary from 5 to 20% of the main compressor flow. Therefore, if the compressor(s) are thermosiphon cooled and use the HPR as the surge drum for the oil cooler(s), then the equalizing line should be sized for the thermosiphon vapor load at 0.25 psi total pressure drop. This will raise the receiver pressure by 0.25 psi during normal operation, and as explained in the drop leg sizing section it must be factored into the drop leg height calculations. In this case, the density ratio calculation method, described above for the vent line, should be ignored because it is superseded by the much larger thermosiphon vapor flow rate. **Table B-7** in appendix-B displays the maximum oil cooler load in kBTU/hr, for a given TSV line size.

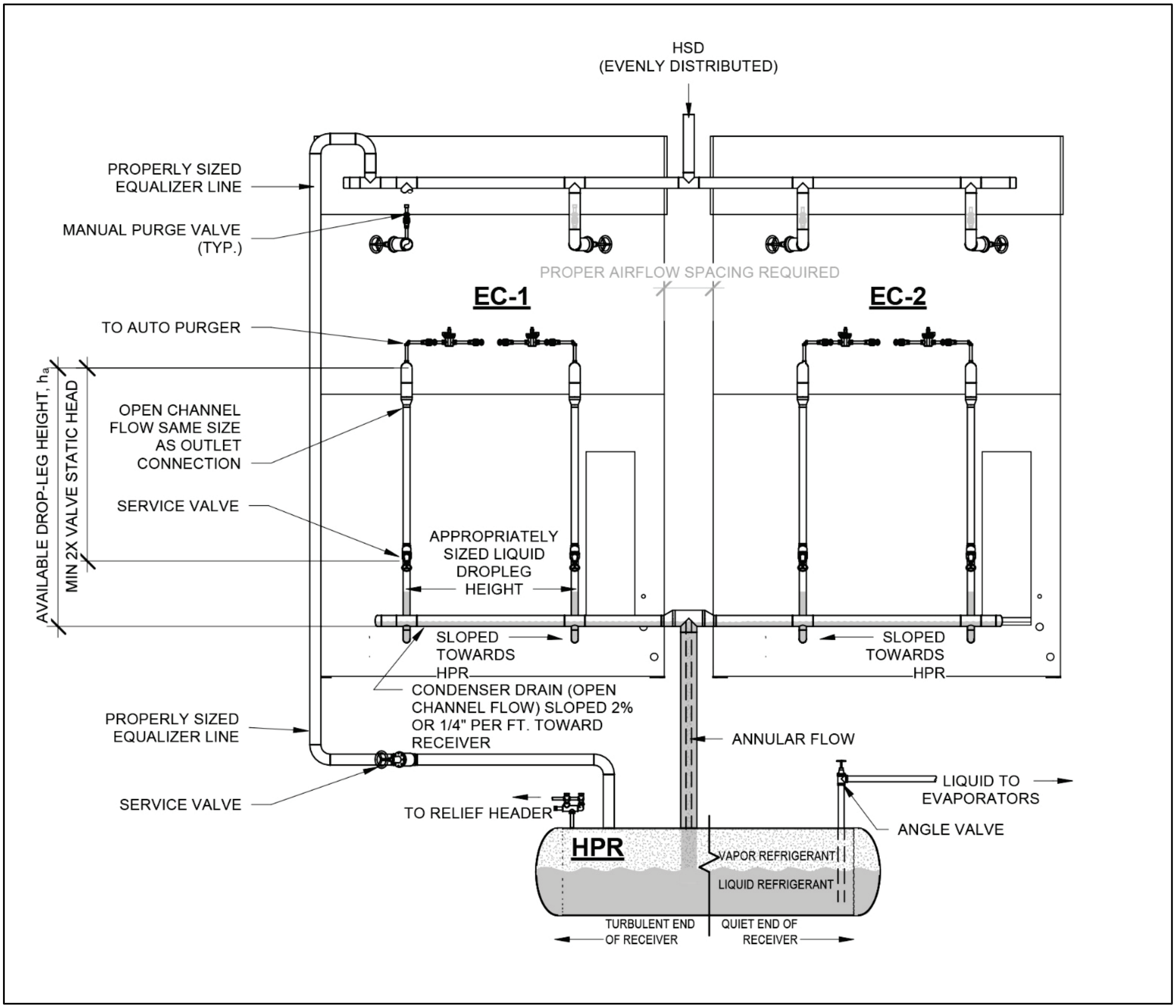
As for the equalizer line, the TSV line should be connected to the lowest pressure point in the HSD header. If it is connected closer to the compressor discharge, that will elevate the pressure in the receiver requiring a greater drop leg height.

12. Multiple condensers

Piping with P-traps and open-channel flow CD header (BAC recommended)

As described in the previous section, in a multiple condenser installation, it is recommended that HSD from the compressor and the high-pressure receiver (HPR) shall be maintained at a balanced location to allow proper supply of the refrigerant and proper condenser drain. For multiple condenser installations, BAC recommends piping the CD line as shown in **Figure 16** and explained in the CD line piping section provided in the earlier part of this manual.

Figure 16: *Multiple condensers CD line piping into the top of the high-pressure receiver (BAC recommended)*



Advantages of CD piping with P-traps and open-channel flow header:

- Stable system operation at all conditions
- Good for unit cycling
- Efficient removal of non-condensable gases
- In case of a sudden pressure drop in the coil bundle, P-traps clear out and the volume of liquid drawn into the coil is minimum.
- Lower charge compared to a flooded header piping

As listed, a very important benefit of this piping configuration over flooded header is that in case of a sudden drop in pressure inside the condenser coil bundle, P-traps will quickly clear out and vapor will flow from the upper portion of the CD header to the condenser(s). This will avoid the possibility of excessive liquid backing up into the condensers and avoid potential operational problems.

Legacy piping configurations for multiple condensers

There are additional CD line piping configurations that have historically been used, which are no longer recommended and are shown here for educational purposes only in **Figures 17, 18 and 19**. It is no longer recommended to flood the CD header with liquid. Not only does flooding increases the ammonia system charge but it also provides a reservoir of liquid, and in the event of a sudden drop in pressure inside the condenser coil bundle(s), for example when a spray pump/fan comes on, the liquid will be drawn up the condenser drop leg and into the coil bundle(s), thus flooding the coil bundle tubes and greatly reducing the condenser thermal capacity and may cause operation problems.

Although less forgiving to capacity fluctuations, these legacy piping configurations may be successfully used if the system control logic avoids frequent spray pump cycling and rapid capacity changes of one condenser relative to another.

***Figure 17**: Multiple condensers piping configuration with bottom inlet HPR*

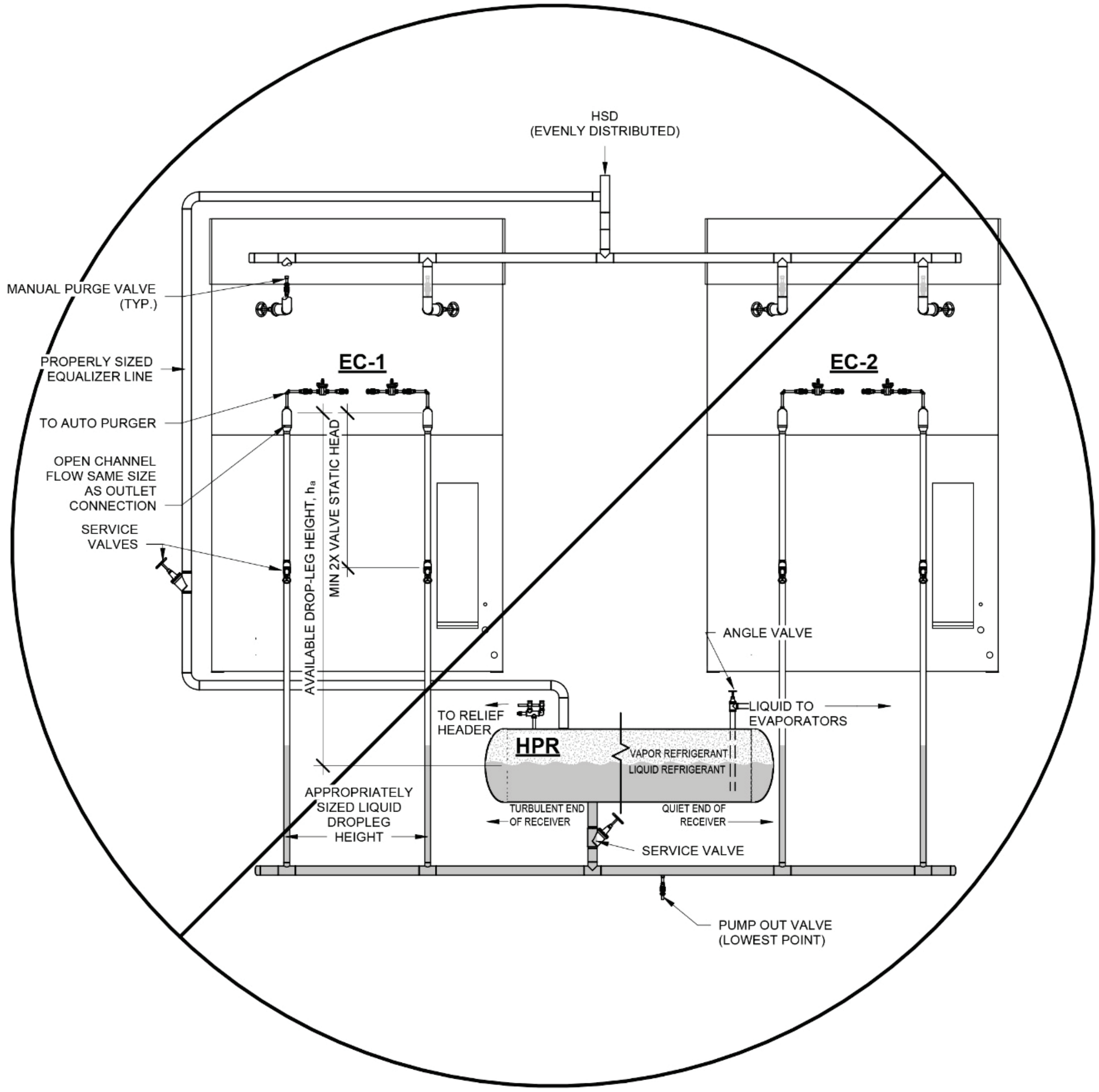
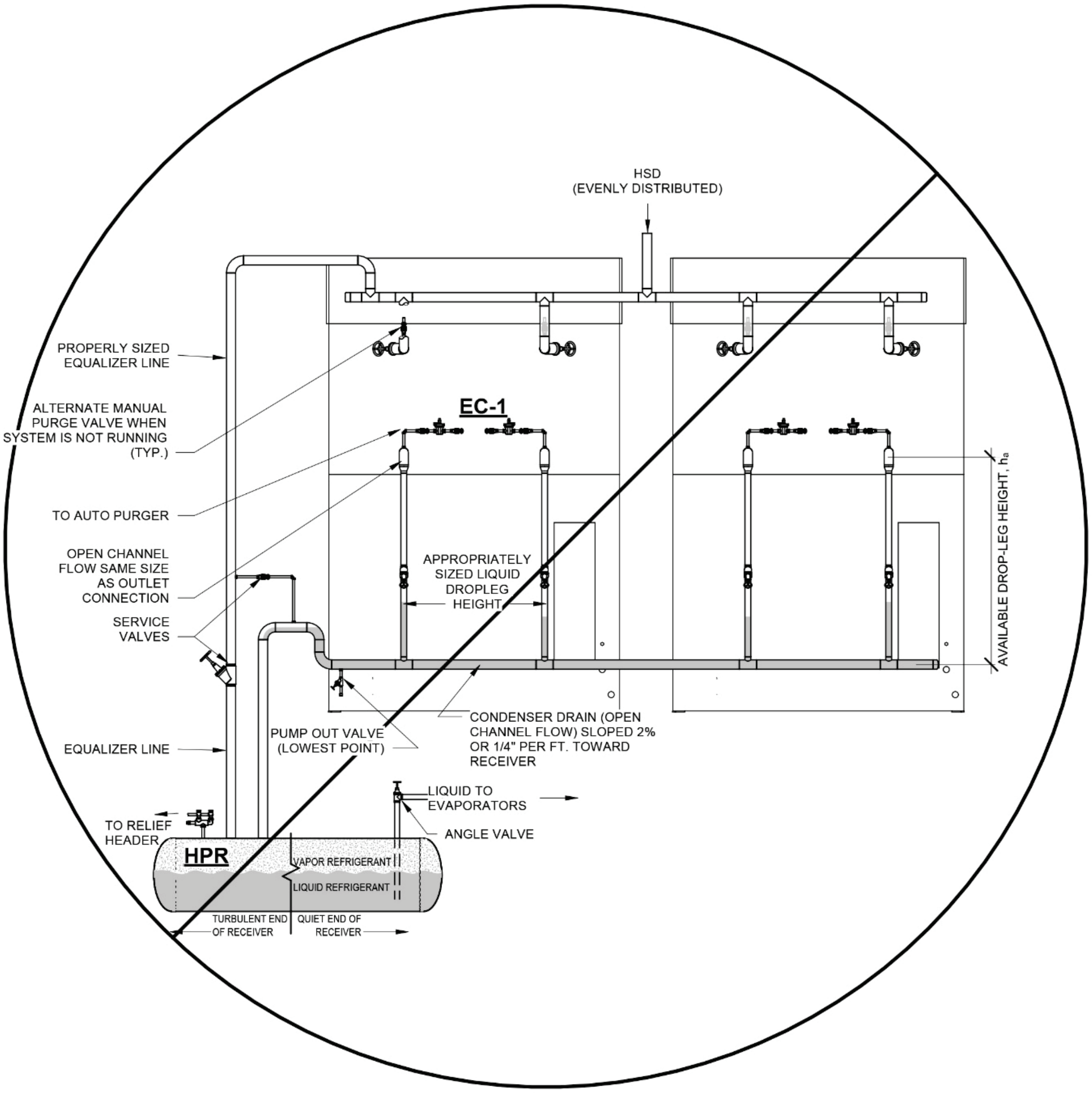


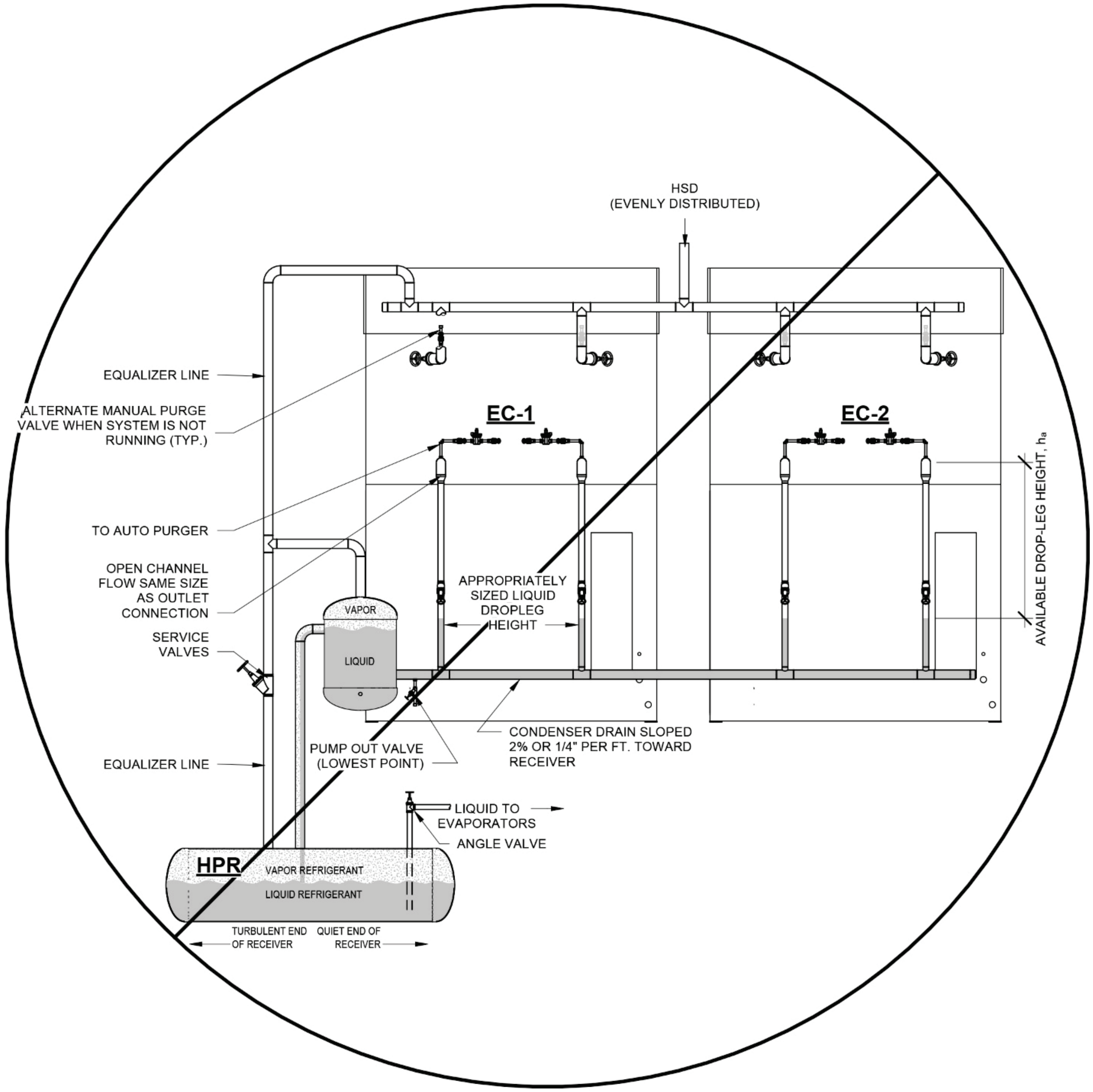
***Figure 18**: Multiple condensers CD piping configuration with inverted P-trap*



NOTES:

1. h_a = LENGTH OF THE SHORTEST DROP-LEG HEIGHT

***Figure 19**: Multiple condensers piping configuration with vertical trap*

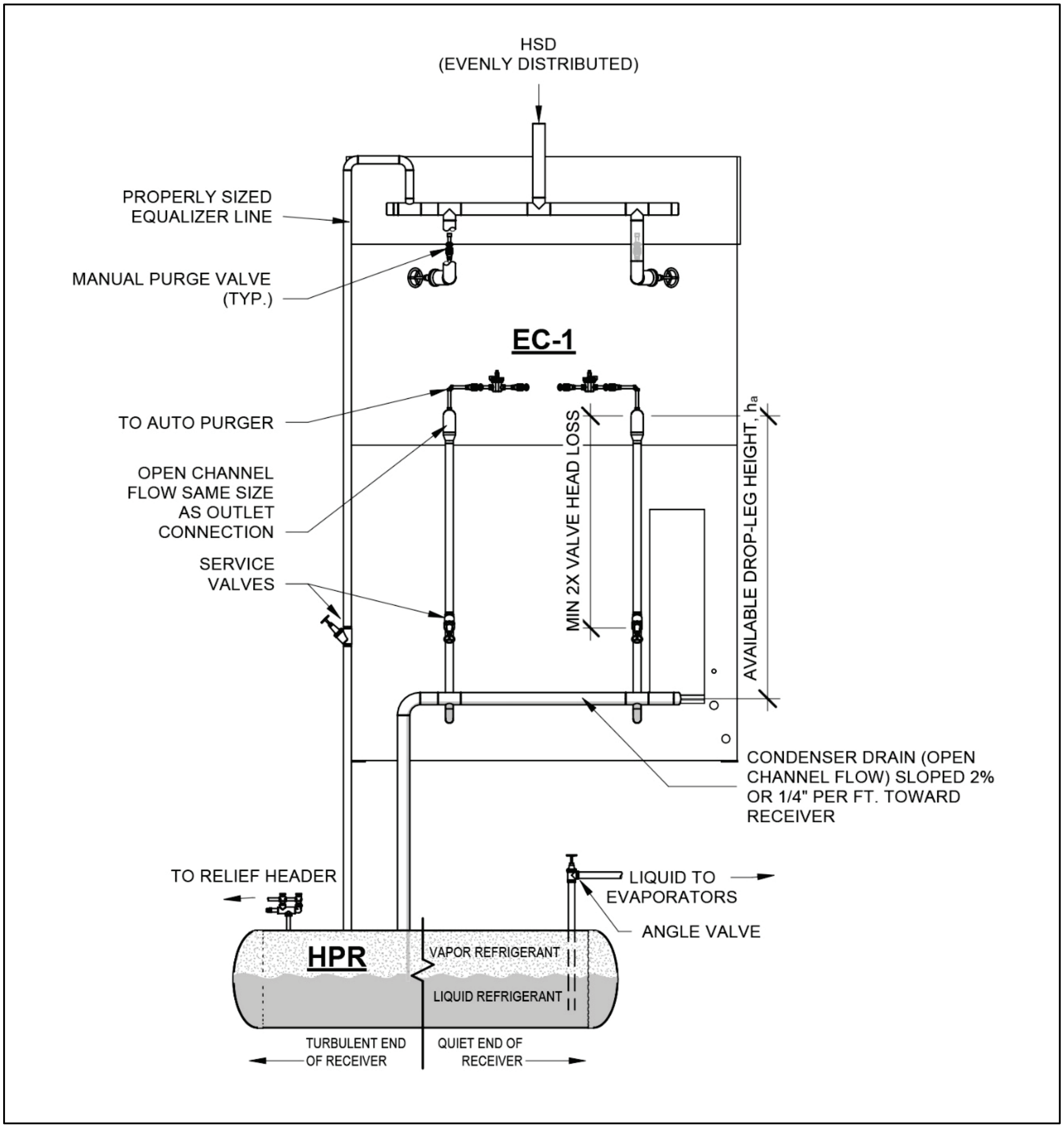


13. Single condenser CD piping

Piping with P-traps and open-channel flow CD header (BAC recommended)

For a single condenser configuration, BAC recommends piping the CD line as shown in **Figure 20** and explained in the CD line guidelines provided in the earlier part of this manual.

***Figure 20**: Single condenser CD piping with individual P-traps and open-channel flow (For clarity purpose, the left hand coil connection is showing the purge valve and the right-hand connection is showing the pipe)*



Advantages of CD piping with P-traps and open-channel flow header:

- Stable system operation at all conditions
- Allows future plant expansion
- Efficient removal of non-condensable gases
- In case of a sudden pressure drop in the coil bundle, P-traps clear out and the volume of liquid drawn into the coil is minimum.
- Lower charge compared to a flooded header piping

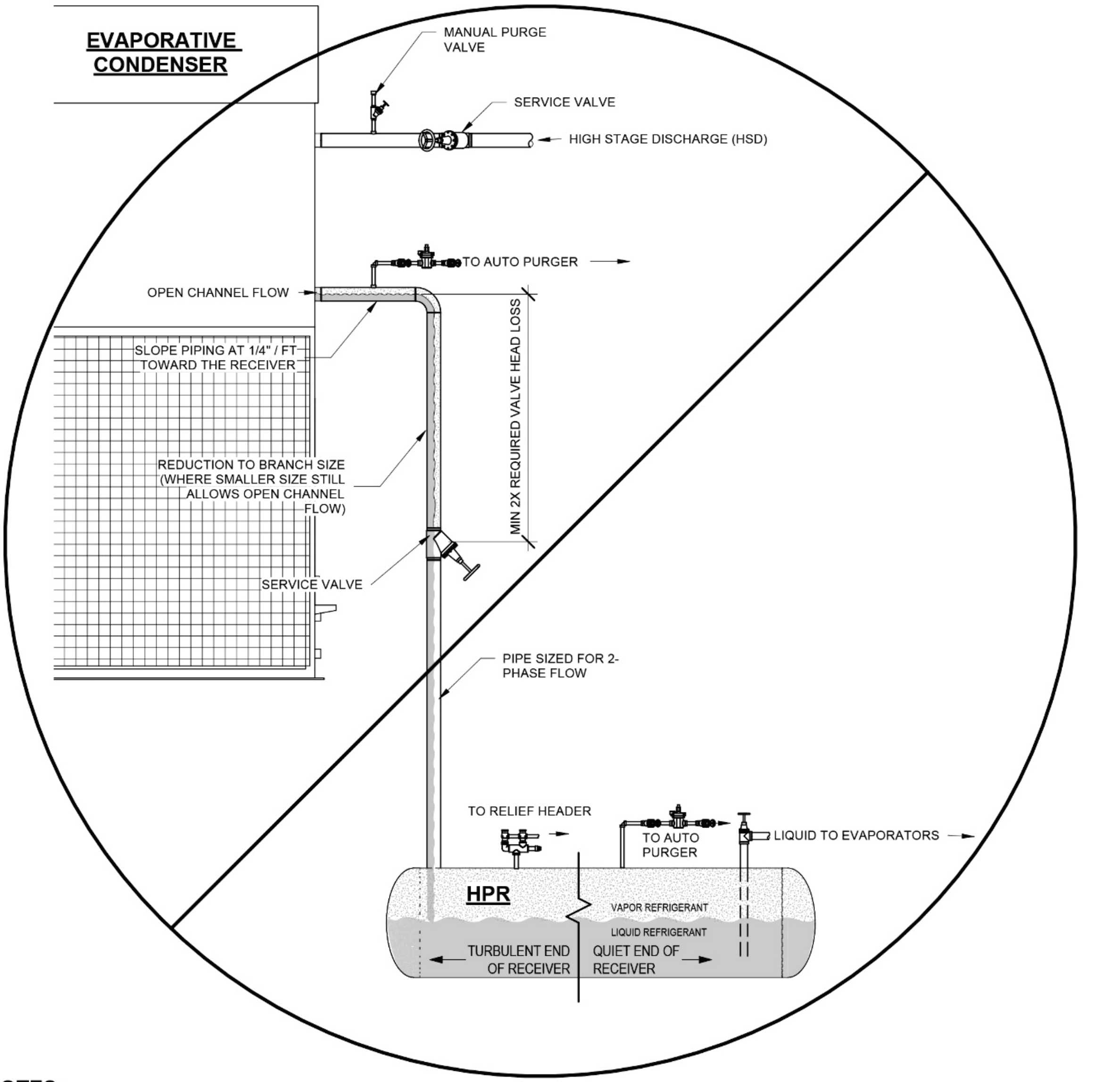
Legacy piping configurations for single condenser

There are additional CD line piping configurations that have historically been used. However, under certain operating conditions, these installations have been known to cause operational issues. Therefore, BAC does not recommend these piping installations and are provided here for educational purposes only.

As shown in **Figure 21**, previous editions of this manual have proposed other methods of piping condenser drains. A single evaporative condenser may be piped with a drain line large enough for open-channel flow. This will allow the receiver to equalize through the drain piping. This method of piping is no longer recommended due to the following reasons:

- It is imperative that the CD line be properly pitched and must not be trapped in any way or liquid accumulation will block the equalization vapor path.
- If a second condenser is added in the future, this style of drain piping must be removed and P-traps with an equalizer line installed in its place.
- Independent coils can have different pressure drops, and without a column of liquid trapped in the outlet piping, the higher-pressure condenser outlets will force liquid into the coil with the lower pressure drop.

***Figure 21**: Piping directly to HPR without P-traps*



NOTES:

1. NO TRAPS REQUIRED TO MAINTAIN A HEIGHT OF LIQUID IN THE DROP LEG.
2. ACCEPTABLE ONLY WITH SINGLE-COIL CONDENSER WHERE SYSTEM EXPANSION IS NOT ANTICIPATED.

14. High side float applications

An alternate method of piping evaporative condensers utilizes single or multiple high-side floats.

They may be installed:

1) On each individual condenser coil drop leg.
2) On the end of a condensate drain header provided there is sufficient capacity
3) On the overflow of a pilot or thermosiphon receiver provided there is sufficient capacity

The advantage of using high side floats is twofold:

1) It reduces the amount of high-pressure liquid in the plant, further adding to plant safety,
2) It removes the expense of a large high-pressure receiver as the liquid level fluctuations and pump-down requirements are accommodated in the low side vessels which have to be sized to accommodate the liquid fluctuations and must be 250 psi DWP.

For such piping, the condenser outlet piping, purge connections, and drop leg pipe sizing remain the same as for a P-trap. However, individual float drainers need to be 2-3’ below the condenser outlet or service valve outlet to allow enough liquid head to overcome the entrance losses of the float drainer.

The float chamber is equalized to the top of the coil outlet piping, completing the hydraulic drain loop.

Exception: Some float drainer designs have a large chamber inlet that discourages the formation of a vapor pocket in the chamber. Consult with the float manufacturer for additional details.

Once the high-pressure condensate passes through the slide gate of the high side float, the pressure is reduced to nearly that of the downstream vessel. When this pressure drop occurs, flash gas is generated, greatly increasing the volume flow and velocity in the outlet piping of the float valve. This must be accounted for when sizing this pipe, and two-phase flow equations must be used. Because of possible temperatures cycling of this pipe, stainless steel with insulation should be considered to prevent corrosion.

If multiple condensers are connected to one large float drainer, then all the design recommendations of section 12 shall be followed, including the condensate drain header sizing.

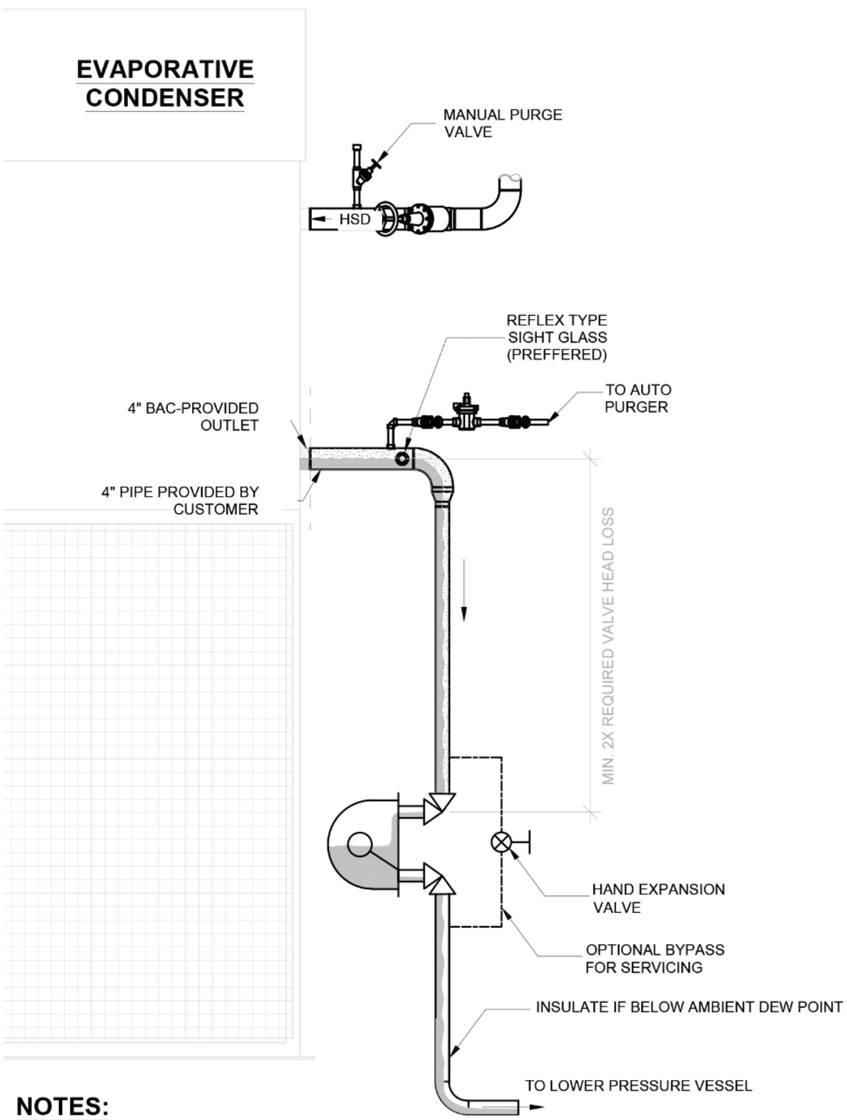
The float drainer shall be connected to the low end of the condensate drain header. Typically, a reducer will be required to transition from the drain header to the float drainer inlet connection.

The reducer should be eccentric, installed flat on bottom. It is also recommended to equalize the float chamber to the far, (elevated) end of the condensate drain header. A service valve into the float drainer would be redundant to the service valves on each drop leg and is not required.

The outlet side of the float drainer will require a service valve and the outlet piping considerations are the same as a single float drainer, described above.

When a float drainer is used to control the level of a pilot or thermosiphon receiver, the piping is beyond the scope of the condenser piping and design guidelines may be found in other sources, such as the IIAR Piping Handbook.

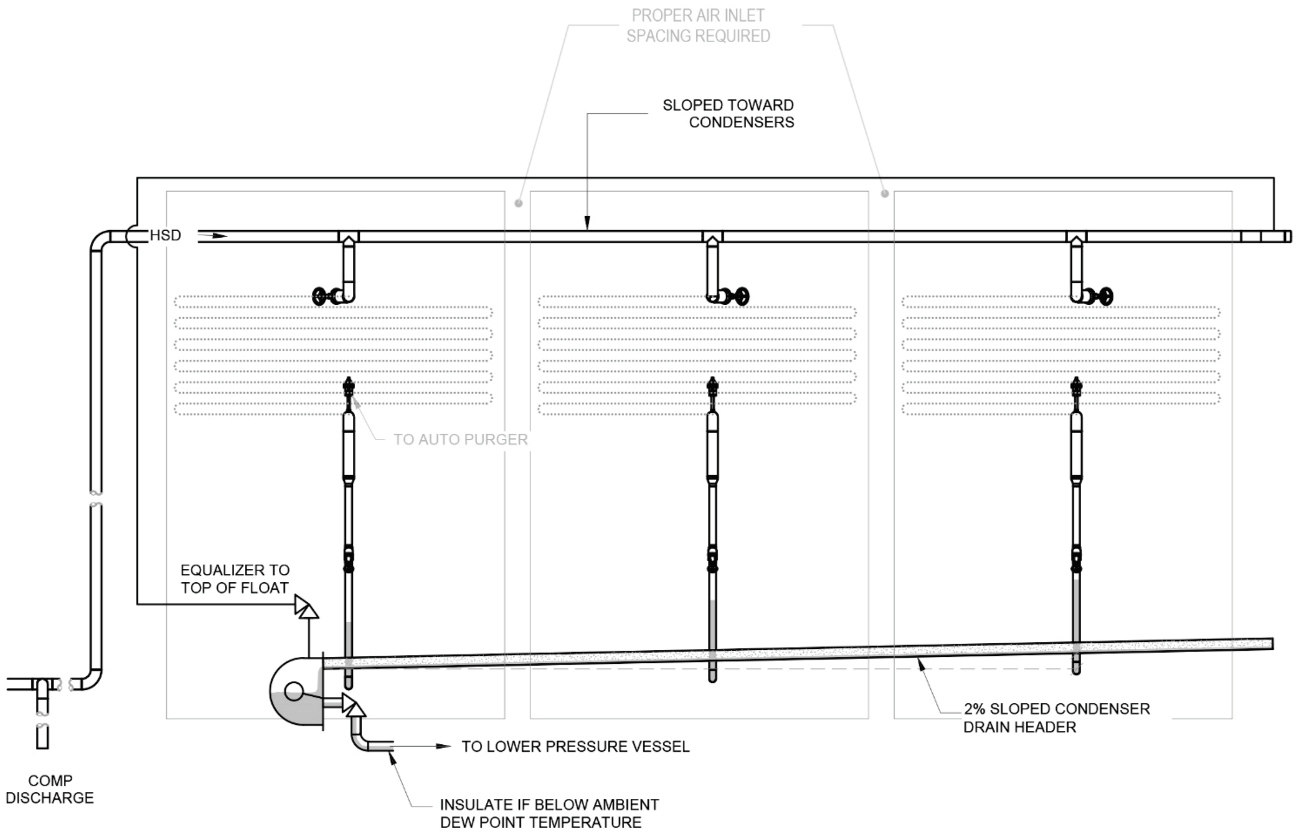
Figure 22*: Float drainer per condenser coil/circuit*



NOTES:

1. FOR THE DETAILED PIPING OF THE HIGH SIDE FLOAT, REFER TO THE MANUFACTURER'S INSTALLATION GUIDELINES.

Figure 23: *One float drainer for multiple condenser coil/circuits*



NOTES:

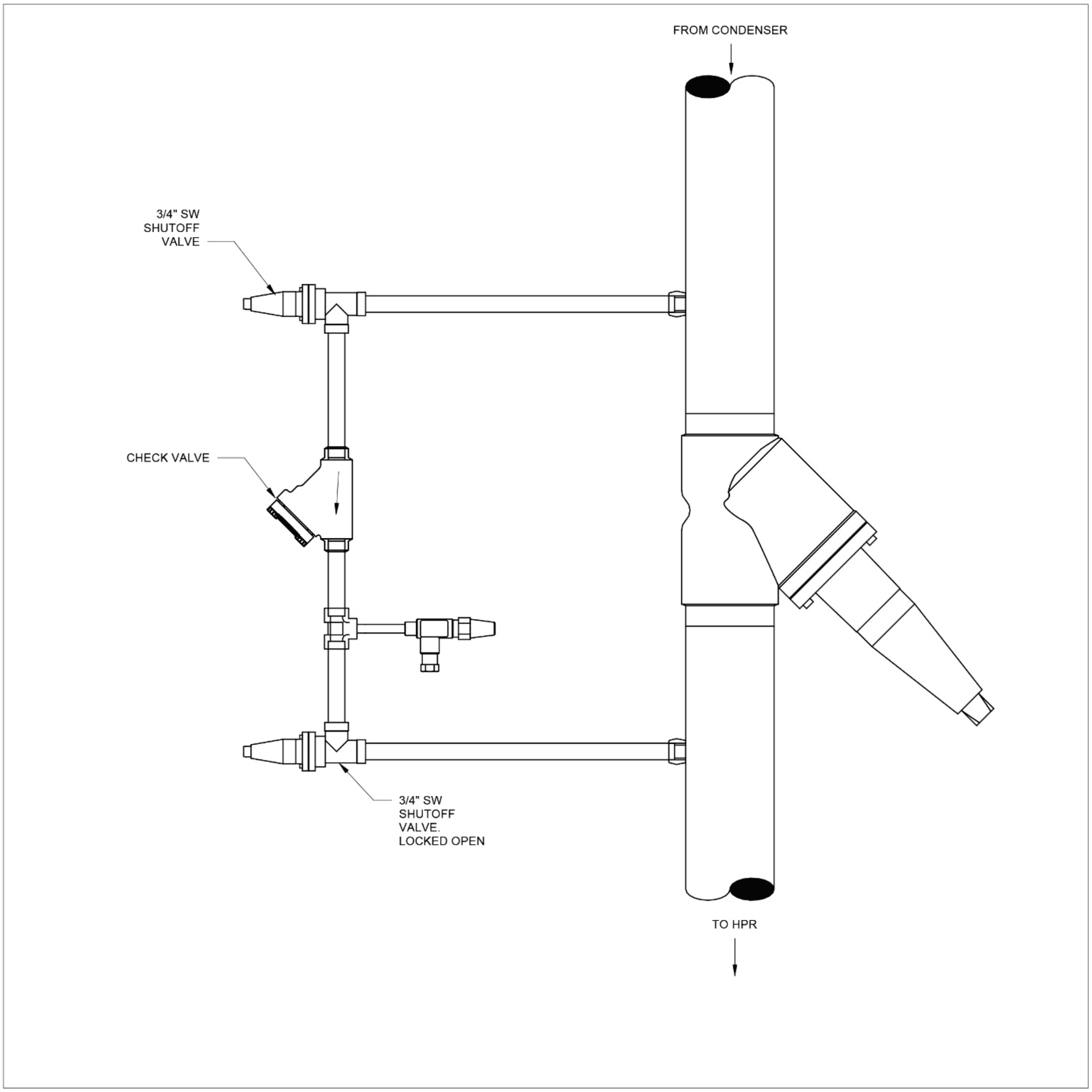
1. FOR THE DETAILED PIPING OF THE HIGH SIDE FLOAT, REFER TO THE MANUFACTURER'S INSTALLATION GUIDELINES.

15. Condenser hydrostatic overpressure protection

If an evaporative condenser is valved off, ammonia vapor may leak by one of the valve seats. In which case it will most likely condense, which has the potential to fill the coil bundle entirely with liquid. Ammonia liquid will expand approximately 0.15% in volume for every 1 °F rise in temperature. This liquid expansion inside a completely filled coil will quickly exert hydraulic pressures far in excess of the design working pressure of the coil. Therefore, when closing the coil bundle(s) vapor inlet and liquid outlet service valves, a means of relieving the coil pressure back to the system piping shall be provided.

As shown in **Figure 24**, this overpressure protection may be accomplished by piping a check valve in parallel to one of the service valves with the flow away from the condenser coil. The appropriate service and bleed valves should be provided as well. The service valves around the check valve must remain open unless the coil is being evacuated for service or check valve is being serviced, in which case, the downstream side of the check valve must be bled. This recommended check valve bypass piping arrangement around the service valve will allow bleeding of pressure around the closed service valve anytime the pressure in the coil becomes greater than the pressure below the service valve preventing hydrostatic coil failure. Another way to accomplish coil overpressure protection is to install a hydrostatic relief valve around the service valve and discharge back into the system. Hydrostatic relief valve requires periodic replacement, per local codes requirement.

Figure 24: *Condenser coil overpressure protection assembly*



NOTICE The overpressure protection assembly is not shown on most figures within the manual, due to the desire for clarity of piping.

⚠WARNING When valving off a condenser bundle for servicing, it is recommended to stop the fans and pumps, close the outlet valve first, followed by immediately closing the inlet service valve. This procedure will trap some liquid in the coil but will prevent filling the coil via liquid ammonia siphoning into the condenser coil. If these steps are not followed, over-pressurization can result in a serious injury, death, or equipment damage.

16. Purge piping

⚠WARNING Ammonia refrigerant venting to the atmosphere shall be performed in accordance with all applicable local, state, and federal laws using accepted industry safety practices. If appropriate safety measures are not taken, it can result in a serious injury, death, or equipment damage.

Non-condensable gases or “foul gas,” is comprised of any vapors, typically air and possibly vapor from the compressor lubricant, in the refrigeration system that do not condense at the same temperature as the refrigerant when subjected to the same pressure. Because these gases will not condense, it acts as a heat transfer resistance inside the inner surface of the condenser tubes and reduces the condenser efficiency. Therefore, they must be removed from the refrigeration system. There are two common methods to purge non-condensable gases from the evaporative condenser(s):

- Automatic purging
- Manual purging

Automatic purging (while condensers are running)

For effective purging of non-condensable gases, purge connections must be located at places where refrigerant velocity and temperature are low. For evaporative condensers, refrigerant gas enters at a very high velocity and temperature. Therefore, with the system running, purging at the inlet of the condenser is not a very effective location to capture the non-condensable gases. On the contrary, at the outlet of the evaporative condenser most of the refrigerant has condensed into a liquid, and the refrigerant temperature is colder, and it is traveling at a very slow speed. This is an ideal location, where non-condensables will accumulate. Therefore, as shown in **Figure 6**, BAC recommends a service valve and automatic purge solenoid valve be situated at the 12 o’clock location of each coil bundle outlet of the evaporative condenser. This coil bundle outlet pipe and the down-turning long-radius elbow should be the same size as the outlet connection. This allows sufficient volume for non-condensable gases to accumulate. Furthermore, BAC strongly recommends using a multi-point, automatic purging system that will efficiently remove non-condensable gases and result in very limited loss of refrigerant to the atmosphere.

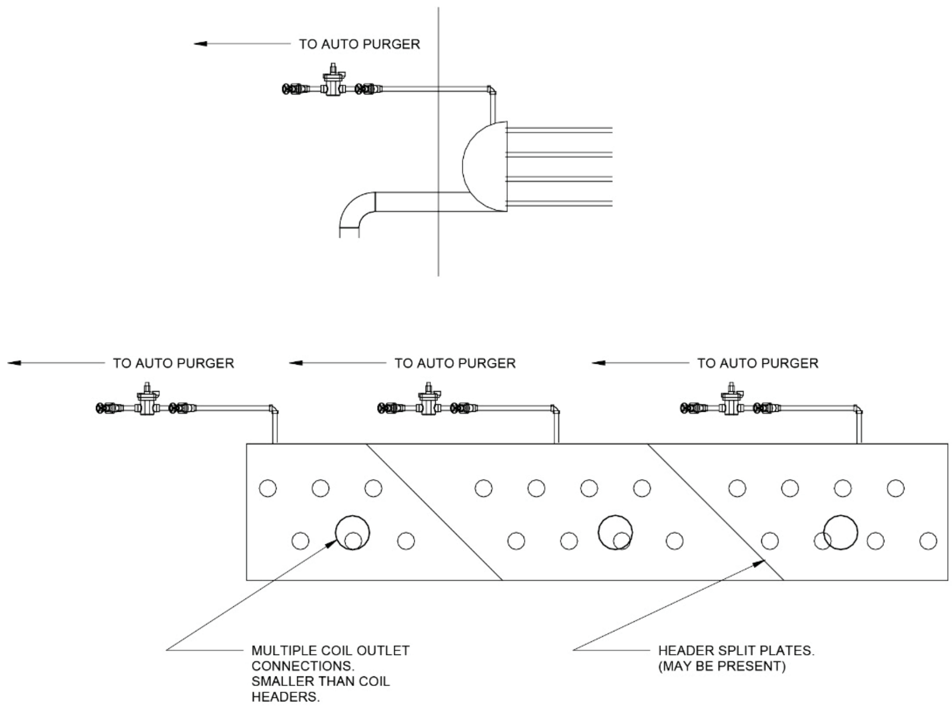
For automatic purging, each coil outlet connection requires its own purge point. Each purge point must be equipped with a solenoid valve. Only one solenoid valve must be opened to the purger at a time to keep the function of the remaining drop legs and P-traps intact. Typically, the automatic purger control system will cycle through each purge point sequentially until no more non-condensable gases are detected.

If liquid backs into the coils due to improper system operation or piping issues, then it can lead to system purging issues. The presence of excessive liquid at the condenser outlet will not allow non-condensable gases to accumulate at the outlet of the condenser which will cause issues with the auto-purger operation. Additionally, the purge line should be cleared of any traps and obstructions. For additional instructions please consult with the auto-purger manufacturer.

If the system was not fully evacuated prior to commissioning or repair, it may be prudent to manually purge the condensers multiple times before relying on the limited capacity of the automatic purger.

Most condenser coils have 4” outlet connections protruding from a similarly sized header, which leaves minimal volume for non-condensable gases to be trapped inside the header. However, as shown in **Figure 25**, coils that are supplied with smaller outlet connections may be equipped with a separate ¾” purge tap into the top of the liquid header. If provided, use this connection for purging the non-condensable gases, which eliminates the need to add a purge connection to the coil outlet.

***Figure 25**: Condenser coil with smaller outlet connection showing a dedicated purge point at the top of the coil header*



NOTES:

1. IN SOME INSTANCES, THE COIL HEADER MAY BE SIGNIFICANTLY LARGER THAN THE OUTLET PIPE. WHEN THIS IS THE CASE, BAC WILL PROVIDE A SEPERATE PURGE CONNECTION CONNECTED TO THE TOP OF THE HEADER. WHEN PROVIDED, ALWAYS USE THIS CONNECTION FOR AUTO PURGING.

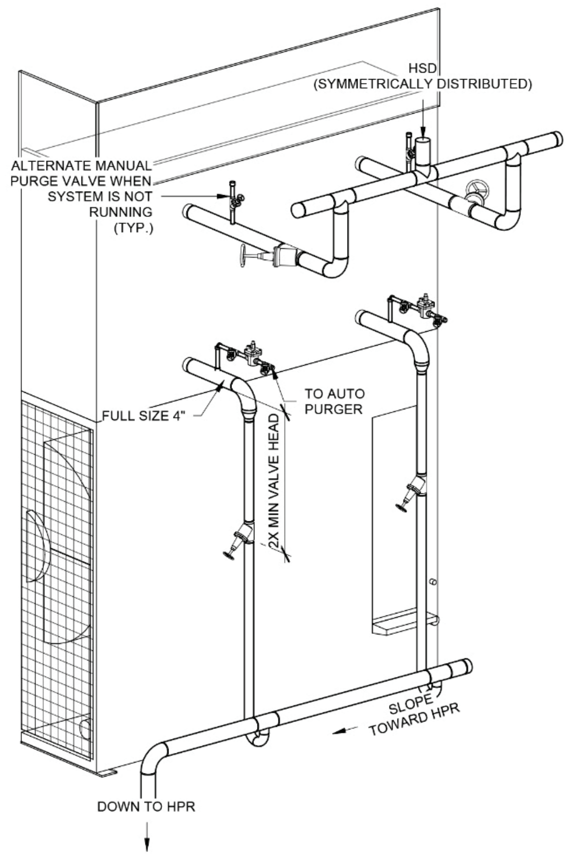
Manual purging (while condenser is shutdown or valved off)

⚠WARNING Venting of ammonia refrigerant to the atmosphere shall be performed in accordance with all applicable local, state, and federal laws using accepted industry safety practices with proper PPEs. If appropriate safety measures are not taken, it can result in a serious injury, death, or equipment damage.

Purging may be accomplished manually while a condenser(s) is idle by following the below procedure. For a single condenser system, the plant has to be shut down but for the multiple condenser system purging can be done one condenser at a time with reduced plant capacity.

1) Close the inlet valve to one of the coil bundles of the multiple condensers with the system running.
2) Leave the condenser fan and pumps ON and the outlet service valve open.
3) After about 10 minutes non-condensable gases will be trapped at the top of the coil bundle. Slowly and partially open the manual purging valve and vent the non-condensable gases (clear), ammonia vapor will puff a gray cloud.
    a. It is recommended to attach the manual purge valve(s) to an appropriately rated refrigerant hose(s) and bring it down to a safe working area.
    b. If venting into a water-filled 55-gallons drum, ammonia vapor will be absorbed into the water, while non-condensable gases will bubble to the top.
4) Repeat step 3 until the clear flow or bubbling stops.
5) Close the manual purge valve, open coil bundle inlet valve.
6) Repeat the procedure on the next coil bundle until all the condensers are done.
7) Return the fan and pumps to automatic control.
8) Try again a few days later if non-condensable gases re-appear.

Figure 26: *Manual purge valves at the top of the condensers*



Steps to check for non-condensable gases in the system

Non-condensable gases, or “foul gas,” is comprised of any vapors in the refrigeration system that do not condense at the same temperature as the refrigerant when subjected to the same pressure. Because it will not condense, it acts as a heat transfer resistance inside the inner surface of the condenser tube's and reduces the condenser efficiency.

Additionally, as per Dalton's law, the total pressure of a refrigerant and non-condensable mixture is equal to the sum of the partial pressures of each of the components.

As an example, if the design ammonia condensing pressure is 181.1 Psig (95°F) and there are non-condensable vapors present, they will have a pressure of their own and the 181.1 Psig becomes the total pressure. The ammonia pressure will now be lower. For instance, 178 psig of ammonia and 3.1 psig of other vapors such as nitrogen or oxygen from air or a hydrocarbon from the lubricating oil in the compressors. Because the ammonia pressure is now 178 Psig, the equivalent condensing temperature will be 94°F. At 78°F wet bulb temperature, this lower condensing temperature will reduce the condenser thermal capacity by 6%. To account for the loss in thermal capacity, the total head pressure will increase to 184.1 psig. This will increase the compressor energy consumption and reduce the compressor capacity. For Ammonia, a 6.3 psi increase in the pressure results in a 2.5% increase in screw compressor input power and a 0.6% reduction in compressor capacity.[7] Furthermore, higher head pressure will lead to higher temperatures and potentially shorter compressor life.

Procedure to tell if non-condensable gases are present when the refrigeration system is running:

1. Measure the condenser outlet pressure
2. Calculate the equivalent saturation temperature at the condenser outlet pressure
3. Measure the refrigerant temperature at the outlet of the condenser
4. If the saturation temperature is more than 3°F higher than the measured temperature, non-condensable gases are likely to be present in the system.

Example to check non-condensable gases

If the ammonia temperature at the condenser outlet is 95°F, the theoretical condenser pressure should be 181.1 psig. If the condenser outlet pressure gauge reads 187.4 psig, then there are likely non-condensable gases in the system which exert an additional 6.3 psi excess pressure. This has the potential to increase power consumption costs by 2.5% and reduces compressor capacity by 0.6%.

After a new installation or repair, BAC recommends that you perform a pressure test and then pull a proper vacuum to remove most of the air and water vapors from the system.

If the system is equipped with an automatic purger a vacuum of 5120 microns (35°F) should be sufficient. For systems that do not have an automatic purger, it is recommended to pull a vacuum of 500 microns. Although Ammonia can absorb a minor quantity of water, it is recommended that any vacuum be held for 24 hours to assure that minimal water vapor remains in the system.

17. References

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18. Appendix-A

This appendix provides the equations necessary for calculating the proper pipe and valve sizes associated with evaporative condenser piping. These equations can easily be inserted into a spreadsheet to aid in the design or analysis process.

A – Vapor or liquid filled pipe loss equations

Use Darcy-Weisbach equation to calculate the pressure drop for non-compressible* vapor or completely filled liquid flow in a pipe:

$$\Delta P = f\left(\frac{L}{D}\right)\left(\frac{\rho}{g_c}\right)\left(\frac{V^2}{2}\right)$$

Where:

ΔP = Pressure drop, lb_f/ft^2

f = Friction factor, dimensionless

Refer to the Moody diagram or use either Churchill or Chen equations to calculate friction factor

L = Pipe segment length, ft

D = Pipe inner diameter, ft

ρ = Refrigerant density at design conditions, lbm/ft3

V = Velocity, ft/sec

g_c = Gravitational constant, 32.2 ft lbm/lbf.sec2

When unit conversions for ΔP and D are factored in, the equation becomes:

$$\Delta P = f\left(\frac{L}{\frac{D}{12}}\right)\left(\frac{\rho}{32.2}\right)\left(\frac{V^2}{2 * 12^2}\right) = f\left(\frac{L}{d}\right)\left(\frac{\rho V^2}{772.8}\right)$$

Where:

ΔP = Pressure drop, lbf/in2

d = Pipe inner diameter, in

12^2 = Conversion from lbf/ft2 to lbf/in2

Note:

- *Because the piping pressure drop is less than 2% of the typical operating pressure, the effects of compressible flow are not considered in the piping calculations.
- Also be mindful that the discharge vapor is superheated, which will lower its density relative to saturated conditions.
- To obtain the friction factor, one must refer to the Moody diagram or use either the Churchill or Chen equations which allows the friction factor to be calculated without resorting to the implicit Colebrook equation.

Churchill Equation

$$f = 8\left[\left(\frac{8}{Re}\right)^{12} + \frac{1}{(B+C)^{1.5}}\right]^{\frac{1}{12}}$$

$$B = \left[2.457\, ln \frac{1}{\left(\frac{7}{Re}\right)^{0.9} + \left(\frac{0.27\varepsilon}{D}\right)}\right]^{16}$$

$$C = \left(\frac{37530}{Re}\right)^{16}$$

Chen Equation

$$\frac{1}{\sqrt{f}} = -2.0\, Log \left\{\frac{\varepsilon}{3.7065D} - \frac{5.0452}{Re} * Log\left[\frac{1}{2.8257}\left(\frac{\varepsilon}{D}\right)^{1.1098} + \left(\frac{5.8506}{Re^{0.8981}}\right)\right]\right\}$$

Where:

f = Friction factor, dimensionless

Re = Reynolds number, dimensionless

ε = Pipe wall roughness, ft – typically 0.00015 for carbon steel pipe

D = Pipe inner diameter, ft

B - Pipe fitting loss equation

The pressure drop through elbows and tees can be calculated by using equivalent pipe length tables or with the head loss coefficient 'K'. Both are available in ASHRAE handbooks, textbooks or numerous online resources.

$$\Delta P = -K\frac{\rho V^2}{2 * g_c}$$

Converting pressure drop from lbf/ft2 to lbf/in2 and reducing yields:

$$\Delta P = -K \frac{\rho V^2}{2{*}12^2 g_c} = -K \frac{\rho V^2}{9273.6}$$

Where:

ΔP = Pressure drop, lbf/in2

K = Pipe fitting head loss coefficient, dimensionless

ρ = Refrigerant density at design conditions, lb/ft3

V = Velocity, ft/sec

g_c = Gravitational Constant, 32.2 ft lbm/lbf.sec2

C - Valve loss equation

The pressure drop for compressible vapor flow through a valve is presented by the Instrument Society of America (ISA) in ANSI/ISA 75.01.01 equation 6, which is:

$$C_v = \frac{W}{63.3\ y\ \sqrt{(xP_1\rho)}}$$

Where:

C_v = Valve flow coefficient, gal/min @1 psi ΔP

W = Mass flow rate, Lb/hr

x = (Desired pressure drop / Inlet pressure), ΔP/P1, dimensionless

ΔP = Pressure drop, lbf/in2

P_1 = Inlet pressure absolute, lb/in2

y = Expansion factor $= 1 - \frac{\left(\frac{\Delta P}{P_1}\right)}{3F_y x_t}$

Where:

F_y = Specific heat ratio, dimensionless 1.32 – 1.4 for ammonia

x_t = Pressure differential ratio factor, ~.30 - .40 for service valve with minor flow restriction

However, the application range for condenser inlet piping is generally 185 psig down to 120 psig and occasionally down to 100 psig, and the desired pressure drop through the service valves is 0.5 psi or less. This creates $\frac{\Delta P}{P_1}$ ratios of 0.0025 to .0037 or 0.0044. When inserted into the expansion factor equation for y above, the resulting value varies between 0.9982 to 0.9969. Which is less than 1% error.

D – Open-channel flow equation

To model liquid flow that partially fills a pipe, as desired in a condenser drain header, it is treated as open-channel flow. The Manning equation is universally accepted for this application, which is:

$$Q = VA = \left(\frac{1.49}{n}\right) AR^{\frac{2}{3}} \sqrt{S}$$

Where:

Q = Flow rate (ft3/sec)

V = Velocity, (ft/sec)

A = Flow area, (ft2)

n = Manning roughness coefficient

R = Hydraulic radius, flow area/wetted perimeter, (ft)

S = Pipe or channel slope, (ft/ft)

The n roughness coefficient has been determined to be 0.0079 to 0.0090 (0.00843 average) for ammonia flowing in a carbon steel pipe at 50 – 250 ft/min velocity range in 3 – 12" pipes. This was determined by comparison to the Darcy-Weisbach equation.

The flow area '*A*' and hydraulic radius '*R*' of a circular pipe can be determined by the following equations:

$$A = (\alpha - \sin\alpha)\frac{D^2}{8}$$

$$R = \left(1 - \frac{\sin\alpha}{\alpha}\right)\frac{D}{4}$$

Where:

α = Center to surface angle in radians

D = Pipe inner diameter, ft

For condensate drain headers, the flow depth should target less than 25% of the pipe inside diameter and not greater than 50%. This will allow sufficient room for vapor flow above the liquid and discourage liquid slugging due to uneven flow.

For the target of 25% flow depth: $A = \frac{0.1535}{4}\,D^2$ and $R = 0.1466\,D$

For the maximum of 50% flow depth: $A = \frac{0.3927}{2}\,D^2$ and $R = 0.25\,D$

The 'n' roughness coefficient varies with the roughness of the channel and the viscosity of the flowing fluid. For ammonia flowing in a carbon steel pipe at 50 – 250 ft/min velocity range in 3-12" pipes, n has been determined to be 0.0079 to 0.0090 (0.00843 average). This was determined by adjusting the flow coefficient, n to agree with the Darcy Weisbach equation results.

The slope of the drain header for gravity dependent flow should be a minimum of 2% which is commonly stated as ¼" of slope per foot of linear run.

To apply the Manning equation to a condenser drain header, one must use the flow rate and slope to determine the minimum pipe diameter, then select a pipe size that exceeds that minimum.

Rearranging and substituting 25% flow depth values for A and R along with n and s, the equation to solve for pipe diameter with mass flow becomes:

$$Q = \frac{w}{60\rho} = \left(\frac{1.49}{n}\right) A R^{\frac{2}{3}} \sqrt{S} = \left(\frac{1.49}{0.0084}\right) * \left(\frac{0.1535\,D^2}{4}\right) (0.1466D)^{\frac{2}{3}} \sqrt{0.02}$$

Which reduces to a minimum pipe ID, d in inches. Round up to the next larger standard pipe ID.

$$d = 12 * \left(\frac{w}{64.38\,\rho}\right)^{\left(\frac{1}{2\frac{2}{3}}\right)} = 12 * \left(\frac{w}{64.38\,\rho}\right)^{(0.375)}$$ For 25% flow depth

$$d = 12 * \left(\frac{w}{234.55\,\rho}\right)^{(0.375)}$$ For 50% flow depth

E – Liquid flow through a valve

Service valves installed on the outlet side of the condenser are either in a horizontal or vertical pipe.

For horizontal pipes, the flow partially fills the pipe and uniform open channel flow should be preserved. Therefore, any service valve should match the pipe size and with the stem in the horizontal plane, thus minimizing any additional liquid elevation necessary to pass over the valve internals.

For vertical pipes, the flow may fill the entire throat area of the valve. Therefore, any service valve should be sized per Equation 1 in Section C above. Once the pressure drop is determined, it should be converted to static head (roughly 1 ft per 0.25 psi) to determine if there is sufficient pipe height above the valve to accommodate this head and not spill over and affect any horizontal piping flow.

19. Appendix-B

Pressure drop listed in all the tables provided in this appendix are calculated using refrigerant mass flow in pounds per minute (lbm/min). The refrigeration Tons indicated in the tables are based upon operating at 10°F Saturated Suction Temperature (SST) and 95°F Saturated Condensing Temperature (SCT). Factors provided in **Table B-1** can be used to convert the mass flow rate into refrigeration Tons.

Important note about valves C_v: To be conservative, Cv values for either globe or angle valves are the lowest Cv value of the commercially available valves, choosing a valve with higher Cv value will result in increased capacity or lower pressure drop through the valve.

Ammonia mass flow rate to refrigeration Tons conversion table

		Saturated Condensing Temperature [°F]									
		70	75	80	85	90	95	100	105	110	115
Saturated Suction Temperature [°F]	40	0.398	0.403	0.407	0.412	0.417	0.422	0.428	0.433	0.439	0.445
	30	0.400	0.405	0.409	0.414	0.419	0.425	0.430	0.435	0.441	0.447
	20	0.402	0.407	0.412	0.417	0.422	0.427	0.432	0.438	0.444	0.450
	10	0.405	0.409	0.414	0.419	0.424	0.430	0.435	0.441	0.447	0.453
	0	0.407	0.412	0.417	0.422	0.427	0.433	0.438	0.444	0.450	0.456
	-10	0.410	0.415	0.420	0.425	0.430	0.436	0.441	0.447	0.453	0.459
	-20	0.413	0.418	0.423	0.428	0.433	0.439	0.445	0.451	0.457	0.463
	-30	0.416	0.421	0.426	0.431	0.437	0.443	0.448	0.454	0.461	0.467

***Table B-1**: NH3 mass flow rate (lbm/min) to refrigeration Tons conversion table at different SST and SCT*

Note: Economizing below 0°F suction raises the mass flow per Ton by less than 0.5%. This is because the mass flow into the economizer port from subcooling the main liquid flow is offset by the reduced suction mass flow from the increased net refrigeration effect of the subcooled liquid.

Example:

NH3 mass flow rate = 100 lbm/min

Refrigeration Tons at 10°F SST and 95°F SCT = 100/0.430 = 233 tons

HSD header maximum recommended capacity

Pressure drop provided in **Table B-2** and **B-3** for the HSD header and branch piping is performed at superheated vapor temperature of 180°F at 181.1 psig (i.e. 95°F SCT). Different discharge pressure and temperature will affect the vapor density which will invalidate the values provided in these tables.

Table B-2 lists the maximum recommended HSD capacity of the listed pipe sizes. It is based upon 100’ pipe length and three long radius elbows. Furthermore, it assumes that there is no main shut-off valve in the header as all the compressors will have discharge service valves and all the condensers will have inlet valves.

As ammonia flow is diverted to the condensers, the header size can be reduced to match the remaining capacity. The loss due to the reducer is not considered in **Table B-2** calculations.

			Pipe loss		Long-radius elbow loss		Total HSD
Nominal pipe diameter	**Tons at 10/95**	**NH_3 $\dot{m}$**	**ΔP/100'**	**Velocity**	**K - LR elbow**	**ΔP**	**ΔP**
		[lb_m/min]	**[psi]**	**[ft/sec]**		**[psi]**	**[psi]**
2	67	29	1.4	44.6	0.30	0.03	1.5
2-1/2	125	54	1.4	51.4	0.28	0.04	1.5
3	220	95	1.4	58.5	0.25	0.05	1.5
4	440	189	1.3	68.0	0.23	0.06	1.5
5	775	333	1.2	76.2	0.24	0.08	1.5
6	1,250	538	1.2	85.1	0.24	0.10	1.5
8	2,525	1,086	1.2	99.3	0.19	0.11	1.5
10	4,425	1,903	1.1	110.4	0.16	0.11	1.5
12	6,800	2,924	1.1	119.5	0.17	0.14	1.5
14	9,000	3,784	1.1	128.0	0.15	0.14	1.5
16	12,700	5,461	1.1	141.4	0.11	0.12	1.5
18	17,000	7,310	1.1	149.5	0.11	0.14	1.5
20	22,100	9,502	1.1	156.4	0.10	0.14	1.5

***Table B-2**: HSD header maximum recommended capacity per 100’ pipe length with 3 long-radius elbows*

Assumptions for Table B-2

- Superheated vapor density at 181.1 psig (i.e. 95°F SCT) and 180°F = 0.525 lbm/ft3
- Total HSD loss is calculated using 100' of pipe length and 3 long-radius elbows.
- No reductions in pipe size considered.
- No main shut-off valve in the header as all the compressors will have discharge service valves.
- All the condensers will have inlet valves, which are considered in the branch piping.
- Pipe thickness: Schedule 80 up to 2", Schedule 40 for > 2”

HSD branch piping maximum recommended capacity per coil bundle

The target total main header pressure loss should be < 1.5 psi, allowing a 0.5 psi pressure loss through the branch to the condenser inlet. Therefore, the total HSD pressure drop should be less than 2-psi.

Branch piping pressure drop, listed in Table 3a, includes the pressure losses from the branch-T in the main HSD header, a globe valve, a long-radius elbow and 10’ of pipe. This provides a conservative pressure drop.

			Pipe loss		T-Branch Loss		Globe valve		Long-radius elbow loss		
Nominal pipe diameter	Tons at 10/95	NH_3 $\dot{m}$	ΔP/100'	Velocity	K – (T-Branch)	ΔP	C_v	ΔP	K – (LR Elbow)	ΔP	Total Loss
		[lb/min]	[psi]	[ft/sec]		[psi]		[psi]		[psi]	[psi]
1-1/4	17	7	0.82	26.1	0.95	0.04	16	0.36	0.37	0.01	0.5
1-1/2	19	8	0.44	21.1	0.90	0.02	36.7	0.09	0.35	0.01	0.5
2	38	16	0.46	25.3	0.84	0.03	61	0.12	0.3	0.01	0.5
2-1/2	70	30	0.44	28.8	0.79	0.04	103	0.15	0.28	0.01	0.5
3	125	54	0.44	33.3	0.76	0.05	114	0.38	0.25	0.02	0.5
4	250	108	0.43	38.6	0.70	0.06	202	0.49	0.23	0.02	0.5
5	450	194	0.42	44.2	0.66	0.07	545	0.22	0.24	0.03	0.5
6	710	305	0.4	48.3	0.62	0.08	693	0.33	0.24	0.03	0.5

Table B-3a: Branch pressure drop with branch-T, globe valve, long radius elbow, and 10’ pipe per coil

Table B-3b provides the branch piping pressure drop by using an angle valve in lieu of a globe valve and a long-radius elbow, doing so will lower the pressure drop.

			Pipe loss		T-Branch Loss		Angle Valve		
Nominal pipe diameter	Tons at 10/95	NH_3 $\dot{m}$	ΔP/100'	Velocity	K – (T-Branch)	ΔP	C_v	ΔP	Total ΔP
		[lb_m/min]	[psi]	[ft/sec]		[psi]		[psi]	[psi]
1-1/4	20	9	1.12	30.7	0.95	0.05	21	0.29	0.5
1-1/2	37	16	1.64	41.2	0.90	0.09	46	0.21	0.5
2	67	29	1.39	44.6	0.84	0.09	80	0.22	0.5
2-1/2	120	52	1.26	49.3	0.79	0.11	140	0.23	0.5
3	190	82	1.01	50.6	0.76	0.11	205	0.27	0.5
4	320	138	0.69	49.4	0.70	0.10	320	0.32	0.5
5	575	247	0.69	56.5	0.66	0.12	596	0.30	0.5
6	820	353	0.54	55.8	0.62	0.11	820	0.32	0.5

Table B-3b: Branch pressure drop with branch-T, angle valve, and 10’ pipe at the condenser coil inlet

Assumptions for Table B-3a and 3b

- Vapor density at 181.1 psig (i.e. 95°F SCT) and 180°F = 0.525 lbm/ft3
- This table was generated with the lowest Cv value of the commercially available valves, higher Cv values will result in increased capacity or lower pressure drop through the valves.

Drop leg maximum recommended flow rate

	Globe valve loss					Pipe loss		Long-radius elbow loss		Valve, 12' pipe and 2 LR elbows	
Nominal pipe diameter	C_v	Tons at 10/95	NH_3 $\dot{m}$	ΔP	Head loss	ΔP/100'	Velocity	K - LR elbow	ΔP	Total ΔP	Total head loss
			[lb_m/min]	[psi]	[in]	[psi]	[ft/min]		[psi]	[psi]	[inches]
3/4	8	35	15	0.09	4.0	0.90	138	0.92	0.02	0.23	10.9
1	15	75	32.2	0.11	5.3	1.07	174	0.41	0.01	0.27	12.6
1-1/4	16	105	45.1	0.19	9.1	0.46	138	0.37	0.01	0.26	12.5
1-1/2	36.7	198	85	0.13	6.2	0.70	192	0.35	0.01	0.24	11.5
2	61	360	155	0.16	7.4	0.47	180	0.30	0.01	0.24	11.1
2-1/2	103	605	260	0.16	7.3	0.47	216	0.28	0.01	0.24	11.3
3	114	780	335	0.21	10.0	0.26	180	0.25	0.01	0.26	12.3
4	202	1,400	602	0.22	10.2	0.20	186	0.23	0.01	0.26	12.2

***Table B-4**: Maximum recommended flow rate for different drop leg pipe sizes*

Assumptions for Table B-4

- Liquid density at 95°F = 36.7 lbm/ft3
- Liquid specific gravity at 95°F = 0.588
- This table was generated with the lowest Cv value of the commercially available valves, higher Cv values will result in increased capacity or lower pressure drop.
- The pipe and P-trap size matches the valve size, and a 12' drop leg height was accounted for in the total pressure drop calcs.
- The total static head loss for most drop legs is less than 12 in. which must be considered in the overall drop leg height

CD header sizing table for open-channel flow

Table B-5 lists the drain header sizing for open-channel flow. Maximum flow rate a 50% pipe depth and at 25% pipe depth for both 2% (¼" per ft) and 1% (1/8" per ft) slope. The flow rates are calculated in volume flow rate (ft3/sec) and converted to mass flow based upon 36.7 lbm/ft3 liquid density at 95°F. The Tons are based upon 0.427 lb/min/Ton (10°F/95°F).

It is recommended that the maximum flow rate in any part of the header stays below 50% depth to avoid liquid slugging and above 25% depth for cost efficiency. In addition, vertical drops should continue in the same pipe size, and the receiver inlet connection size should also match the maximum drain header size.

	2% Slope (Recommended)				**1% Slope***			
	50% pipe depth		**25% pipe depth**		**50% Pipe depth**		**25% Pipe Depth**	
Nominal Pipe Diameter	**Tons at 10/95**	**NH_3 $\dot{m}$**	**Tons at 10/95**	**NH_3 $\dot{m}$**	**Tons at 10/95**	**NH_3 $\dot{m}$**	**Tons at 10/95**	**NH_3 $\dot{m}$**
	[Tons	**[lb_m/min]**	**[Tons]**	**[lb_m/min]**				
2	64	27	18	7	110	47	30	13
2-1/2	297	127	81	35	209	89	57	24
3	530	226	144	62	374	160	102	44
4	1093	467	299	128	773	330	211	90
5	1996	852	547	233	1411	603	387	165
6	3259	1391	893	381	2304	984	631	270
8	6776	2893	1856	793	4791	2046	1313	561
10	12428	5307	3404	1453	8790	3753	2408	1028
12	19828	8467	5433	2320	14022	5987	3841	1640
14	25527	10900	6994	2986	18049	7707	4945	2111
16	36454	15566	9989	4265	25774	11006	7062	3015
18	49914	21313	13676	5840	35294	15070	9669	4129
20	66679	28472	18271	7802	47151	20134	12918	5516

***Table B-5**: Maximum Tons and NH3 mass flow rate for open-channel flow condenser drain header*

Assumptions for Table B-5

- Liquid density at 95°F = 36.7 lbm/ft3
- Tons are based upon 0.427 lbm/min/Ton (10°F/95°F)
- Recommend horizontal pipe slope is 2% (1/4” ft).
- *1% slope should only be used when the refrigerant mass flow rate is very low (<25% flow depth).

Equalizer line sizing

Table B-6 displays the maximum system capacity for different equalizer line sizes. It is based upon the volume of the system liquid flow matched to the same volume of equalizer vapor flow.

The maximum amount of pressure drop allowed for the pipe with fittings and a valve is 0.02 psi which is 1” of static head fluctuation imposed upon the liquid height in the drop leg. This low-pressure drop minimizes any upset to the condenser drain flow dynamics during system liquid flow fluctuations.

				Pipe		Square Entrance Loss		Globe Valve		Long Radius Elbow Loss		Total Pressure Loss	
Nominal Pipe Diameter	**Tons at 10/95**	**System NH_3 $\dot{m}$**	**Equalizer NH_3 $\dot{m}$**	**ΔP/100'**	**Velocity**	**K - Sq. Ent.**	**ΔP**	**C_v**	**ΔP**	**K - LR Elbow**	**ΔP**	**ΔP**	**Liquid head**
		[lb_m/min]	**[lb_m/min]**	**[psi]**	**[ft/sec]**		**[psi]**		**[psi]**		**[psi]**	**[psi]**	**[in]**
3/4	46	20	0.4	0.032	2.99	0.50	0.000	8	0.00	0.92	0.001	0.02	1.0
1	95	41	0.7	0.035	3.71	0.50	0.000	18	0.00	0.41	0.000	0.02	1.0
1-1/4	165	71	1.3	0.023	3.62	0.50	0.000	16	0.01	0.37	0.000	0.02	1.0
1-1/2	290	125	2.2	0.029	4.62	0.50	0.001	36.7	0.01	0.35	0.001	0.02	1.0
2	540	232	4.1	0.026	5.32	0.50	0.001	61	0.01	0.3	0.001	0.02	1.0
2-1/2	950	409	7.3	0.023	5.58	0.50	0.001	103	0.01	0.28	0.001	0.02	1.0
3	1350	580	10.3	0.015	5.13	0.50	0.001	114	0.01	0.25	0.000	0.02	1.0
4	2500	1075	19.1	0.012	5.53	0.50	0.001	202	0.01	0.23	0.000	0.02	1.0
5	5500	2365	42.1	0.017	7.74	0.50	0.002	545	0.01	0.24	0.001	0.02	1.0
6	7800	3359	59.8	0.013	7.59	0.50	0.002	693	0.01	0.24	0.001	0.02	1.0
8	14000	6020	107.1	0.010	7.87	0.50	0.002	1188	0.01	0.24	0.001	0.02	1.0

Table B-6: *Maximum system capacity for different equalizer line sizes*

Assumptions for Tabe B-6

- Saturated vapor density = 0.6527 lbm/ft3
- Liquid density = 36.68 lbm/ft3
- Total Pressure drop reflects 50' pipe, square entrance loss, a globe valve and 3 elbows

Thermosiphon Vent (TSV) line sizing

Table B-7 lists the maximum oil cooler load in kBTU/hr, for a given TSV line size. The pressure drop limit used will result in a 6" additional liquid height requirement over and above the liquid head required in the drop leg to overcome the condenser and drop leg pressure drops.

			Pipe		Square Entrance Loss		Globe Valve		Long Radius Elbow Loss		Total Pressure Loss	
Nominal Pipe Diameter	Oil cooler kBTU/hr at 95°F	NH_3 $\dot{m}$	ΔP/100'	Velocity	K - Sq. Ent.	ΔP	C_v	ΔP	K - LR Elbow	ΔP	ΔP	Liquid Head
		[lb_m/min]	[psi]	[ft/sec]		[psi]		[psi]		[psi]	[psi]	[in]
3/4	26	1	0.19	7.65	0.50	0.002	8	0.02	0.92	0.004	0.13	6
1	53	2	0.2	9.35	0.50	0.003	18	0.01	0.41	0.003	0.12	6
1-1/4	90	3	0.13	8.91	0.50	0.003	16	0.05	0.37	0.002	0.13	6
1-1/2	155	6	0.16	11.13	0.50	0.004	36.7	0.03	0.35	0.003	0.12	6
2	295	10	0.15	12.69	0.50	0.006	61	0.04	0.3	0.003	0.13	6
2-1/2	525	18	0.13	13.92	0.50	0.007	103	0.04	0.28	0.004	0.13	6
3	745	26	0.09	12.80	0.50	0.006	114	0.07	0.25	0.003	0.13	6
4	1375	47	0.07	13.72	0.50	0.007	202	0.08	0.23	0.003	0.13	6
5	2950	102	0.1	18.73	0.50	0.012	545	0.05	0.24	0.006	0.13	6
6	4200	145	0.08	18.48	0.50	0.012	693	0.06	0.24	0.006	0.13	6
8	7500	259	0.060	19.04	0.50	0.013	1188	0.07	0.24	0.006	0.13	6
10	13750	475	0.060	22.14	0.50	0.017	2350	0.06	0.24	0.008	0.13	6

***Table B-7**: Thermosiphon Vent Line sizing*

Assumptions for Table B-7

- Saturated vapor density = 0.6527 lbm/ft3
- Liquid density = 36.68 lbm/ft3
- Total Pressure drop reflects 50' pipe, square entrance loss, a globe valve and 3 elbows.

Valves C_v table

	Lowest C_v*	
Pipe Size	**Angle**	**Straight**
3/4	9	8
1	19	15
1-1/4	21	16
1-1/2	46	36.7
2	80	61
2-1/2	140	103
3	205	114
4	320	202
5	596	545
6	820	693
8	1355	1188
10	1630	2350
12	2169	3270
14	4600	4350
16	5640	
18	6500	

Table B-8: Lowest Cv of commercially available valves

*Lowest Cv value of the commercially available valves, higher Cv values will result in increased capacity or lower pressure drop through the valve.

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