Application Guide

Refrigerant Piping Manual for Small Split Cooling and Heat Pump Systems

I – Refrigerant Piping
II – Microchannel Units
III – High Rise Systems

Trane and American Standard Heating & Air Conditioning Split Systems

February 2011

SS-APG006-EN

Application Guide

Index

This manual is dedicated to improving system performance and reliability. A properly designed refrigerant piping system ensures oil return, minimizes capacity losses, and provides for maximum equipment life.

Our thanks to the following for their valuable contributions:

- Roy Crawford
- Dave Donnelly
- Chuck Erlandson
- Steve Hancock
- Marion Houser
- Dan Joiner
- Red Roley
- Terry Ryan
- Gary Sapp
- Jim Sharp
- Paul Solberg
- Greg Walters
- Richard Welguisz

Tables and Charts

Application Guide

Introduction

The purpose of this manual is to assist in the proper selection of liquid lines and suction lines for straight cooling and heat pump split systems (Chapter I). Chapter II covers High Rise Applications.

Careful use of the tables and charts in Chapter I will ensure:

- minimum pressure drops,
- adequate oil return,
- maximum system reliability,
- delivery of 100% liquid to the metering device.

New selection tables are included for liquid and suction lines covering total linear = 200 ft. equivalent lengths up to 240 ft.

The philosophy in designing a refrigerant piping system can be summed up as follows:

Liquid lines should be sized as small as possible without exceeding the recommended maximum pressure drop of 50 PSI for R-410A. Liquid line pressure drop calculations must include friction loss, liquid lifts and refrigerant accessories (solenoid valves, etc.).

The penalties for liquid pressure drop are minimized provided the pressure drop does not exceed 50 PSI for R410A systems. The liquid line with smallest diameter that meets this 50 PSI criteria, results in better system reliability (fewer pounds of refrigerant to cause potential damage to the compressor). Note that the 50 PSI allowance is based on 10 degrees Fahrenheit of sub-cooling (no liquid receiver.) Both Trane and American Standard Heating and Air Conditioning split systems that utilize Spine Fin™ coils ranging in nominal capacity from 1.0 to 5.0 tons meet this criterion.

Since suction line pressure drop does reduce capacity and efficiency, suction lines should be sized as large as possible, while still maintaining sufficient velocity for oil return. All tubing sizes listed in Table 3 and Table 5 will provide oil return. Using the largest diameter listed for a given tonnage results in the lowest losses in capacity and efficiency consistent with proper oil return. Shorter tubing runs may provide acceptable losses with a smaller diameter. For multistage systems, please reference table 5 for the approved liquid and vapor line diameters and lengths.

Hot gas lines are somewhat less critical insofar as pressure drops and oil return are concerned. In the case of a heat pump, the gas line is sized as a suction line, and although it is somewhat oversized as a discharge line, our experience over many years, indicates that oil return is not a problem, within the published limits.

The Windows® based piping program, Publication 32-3312-03 (or latest version) covers Trane and American Standard Heating and Air Conditioning split systems ranging from 1.0 to 20 ton in nominal capacity.

The piping program is very user friendly and is highly recommended, since it:

- saves valuable time,
- reduces errors,
- reminds the user of the required accessories,
- generates customer confidence,
- establishes the user as a knowledge-able expert.

Information provided by the program includes:

- Liquid and suction line sizes
- Liquid and suction line pressure drops
- Net system capacity loss
- Approximate system charge
- Required system accessories
- High rise requirements
- Reciprocating and scroll compressor requirements
- R-22 and R-410A refrigerants
- Linear lengths to 200 ft.
- Linear lifts to 200 ft. (150 ft. for heat pumps)
- Excellent print-outs

Application Guide

General Information

The four prime considerations in designing a refrigerant piping system are:

A – System Reliability

B – Oil Return

C – Friction Losses (Pressure Drop)

D – Cost

A — The piping system can affect system reliability in a number of ways:

- Oversized liquid lines significantly increase the amount of refrigerant in the system, and thus creating the potential for slugging, oil dilution, or other damage to the compressor.
- Undersized liquid lines and the associated "flashing" of refrigerant causes starving of the evaporator coil. The results can be significant loss in capacity, frosted evaporator coil, high superheat etc.
- Oversized suction lines will result in refrigerant velocities too low to provide adequate oil return to the compressor.
- Undersized suction lines reduce capacity and efficiency and contribute to high superheat.
- Excessive refrigerant line length reduces system capacity and efficiency, as well as system reliability (excessive refrigerant charge). Keep refrigerant lines as short as conditions permit!

B — Oil return must always be considered since some oil is continually being circulated with the refrigerant and **must** be returned to the compressor. If the recommended suction line sizes are used, no oil return problems should be encountered with split systems.

C — Pressure drop or friction losses are important from a performance standpoint. The following general statements point out the effects of pressure drop in the various components of the refrigerant piping system.

1 – Pressure drop in the suction line reduces system capacity significantly and increases power consumption per ton. The most generally accepted value for pressure drop equivalent to 2°F (approximately 5 PSI for R-410A in the air conditioning range of evaporating temperatures). As tubing runs become longer, it is inevitable that the ASHRAE recommendation will be exceeded, at times. This trade-off, of somewhat greater suction line losses, for adequate oil return is an absolute must, in order to preserve system reliability.

2 – Pressure drop in hot gas lines reduces system capacity to a somewhat lesser degree and increases power consumption to a slightly lesser degree than does pressure drop in suction lines. Since the only hot gas lines we are concerned with are in heat pump systems where they also serve as suction lines, we will treat them as suction lines.

3 – There is no direct penalty for pressure drop in a liquid line **provided that 100% liquid is being delivered to the expansion device, and that the liquid pressure available to the expansion device is adequate to produce the required refrigerant flow**. Pressure drop or gain due to vertical lift must be added to the friction loss in liquid lines to determine the total pressure drop. The acceptable pressure drop in the liquid line for equipment through 5.0 tons nominal capacity is 50 PSI for R410A systems utilizing Spine Fin™.

D — Cost is an obvious consideration and dictates that the smallest tubing possible be used that will result in a system with acceptable friction losses.

The following pages cover the selection of liquid lines and suction lines for split heat pump and cooling systems.

It is recommended that Chapter I is read in order to better understand the Tables, Charts, etc.

See the Table of Contents for a complete listing, including page number, for all tables, charts, etc.

All installations must conform to any codes or regulations applying at the site. The Safety Code for Mechanical Refrigeration, ASA-B-9-1 and the Code for Refrigeration Piping, ASA-B31.5 should serve as your guide toward a safe piping system.

Application Guide

Section I
Refrigerant Piping

Liquid Lines for Split Cooling and Heat Pump Systems

The purpose of the liquid line is to convey liquid refrigerant from the condenser to the expansion device. The expansion device in turn throttles the refrigerant from the high side pressure as it exists at the entrance to the device to the relatively low evaporator pressure. The high side pressure varies through a wide range with the cooling load and the outdoor temperature. The expansion device has to handle this situation and the fact that a particular pressure drop is required to produce the flow through the liquid line is not especially critical providing two conditions exist.

The first condition is that the liquid line transports the refrigerant completely as liquid and not allow the refrigerant to flash partly into gas. This requires that the liquid temperature be lower than the temperature which causes refrigerant to vaporize at the pressure prevailing locally in the tube, that is, the refrigerant must be subcooled throughout the length of the liquid line.

The second condition is that the pressure and amount of subcooling at the entrance to the expansion device must be adequate for the device to pass the required flow into the evaporator to suit the cooling load condition. If not, the evaporator is starved for refrigerant. This may cause one part to freeze ice and gradually choke off the indoor airflow even though other parts of the evaporator are warm for lack of refrigerant. When the evaporator is starved, the reduced cooling effect reduces the head pressure in the condenser and throughout the liquid line, which tends still further to reduce the refrigerant flow. This inadequate head pressure situation must be avoided. However, it prevails only when outdoor temperatures are relatively cool and under conditions when air conditioning for most residential applications is not required.

Any situation such as an unusually long liquid line or a large difference in elevation between the indoor and outdoor sections may require consideration as discussed further below.

The flashing of refrigerant to vapor will occur if the refrigerant absorbs heat in the liquid line so that it is no longer subcooled or if its pressure is reduced below the saturation pressure corresponding to its temperature.

Normally, the liquid line temperature is above that of the surrounding ambient so there is no “flashing” as a result of temperature rise and usually there is enough cooling of the refrigerant to compensate for the fact that the pressure gradually drops to maintain flow. In special cases where the liquid line is run through hot attics or other heat sources the liquid line should be insulated.

Table “4” lists the equivalent length of fittings, which must be added to the linear length of the tubing to obtain the equivalent length of the line.

The pressure loss due to vertical lift (evaporator above the condenser) depends on the difference in level between the metering device and condenser (or receiver) and on the density of the refrigerant. At normal liquid line temperatures with R-410A, the static pressure drop will be 0.43 PSI per foot.

As an example, consider an air cooled R-410A system with 95°F air entering the condenser, the condensing temperature is 120°F (approximate 418 PSIG).

After being subcooled in the condenser, the liquid R-410A leaves the condenser at 110°F. Assuming the pressure at the condenser outlet is the same as the condensing pressure of approximately 418 PSIG, the liquid R-410A has been subcooled 10°F. The saturation pressure for R-410A @ 110°F is approximately 365 PSIG. Subtracting 365 PSIG from the 418 PSIG condensing pressure, gives a difference of 53 PSI. Even though the difference is 53 PSI, the formulas in this application guide and in the piping program limit the liquid line pressure drop in an R410A system to 50 PSI.

The foregoing has shown how to figure the liquid line pressure drop and indicated that the heat loss to the surroundings help to maintain adequate subcooling. The amount of refrigerant in the system governs the amount of subcooling of the liquid as it leaves the condenser. The appropriate installation and charging instructions should be followed.

With regard to whether adequate head pressure is available at the expansion device to give the required flow, note that an unusually high pressure drop in a liquid line due to long lengths or large differences in elevation, has the same effect as a reduced head pressure due to cooler outdoor temperatures entering the air cooled condenser. Typically each additional 10 PSI drop in pressure in the liquid line means that the minimum outdoor temperature at which the system will perform satisfactorily is raised by 3 degrees. Allowance for this is significant only for unusual applications where cooling is required at low outdoor temperatures. Performance for such conditions is published in the Performance Tables and is based on 25 feet of line as used for Standard Ratings. For marginal applications where a Head Pressure Control accessory is under consideration, the effect of liquid line pressure drop should be considered.

Application Guide

There are other considerations with regard to the installation of liquid lines.

The use of long radius ells can reduce the equivalent length of a line and thus reduce the friction loss.

Do not add a drier or filter in series with the factory installed drier as the added pressure drop may cause "flashing" of liquid refrigerant.

If a system does not have a liquid receiver, the amount of the refrigerant charge in the system can have a significant effect on the amount of subcooling obtained, which in turn determines the pressure drop which can be tolerated in the liquid line. (An undercharged system will have little or no subcooling while an over-charged system will have high condensing temperatures because of the loss of effective condensing surface.)

Pressure drop due to the weight of the refrigerant is no problem if the evaporator coil is below the condenser, as the weight of the liquid, in this case, causes an **increase** in pressure and aids in subcooling.

Table "2" is used to select a liquid line. The pressure drop is given for the various equivalent lengths (up to 240 eq. ft.).

The actual selection of a liquid line is covered on page 10.

Note that equivalent lengths are used when calculating pressure drops. Actual (linear) lengths are used when calculating pounds of R-410A in a line set. (An elbow contains about the same amount of R-410A as does the same length of straight tubing.)

Table "4" lists equivalent lengths for elbows, etc. for pressure drop calculations.

In addition to friction loss, any pressure drop due to liquid lift must be accounted for (.43 PSI per foot for R-410A).

The importance of a properly charged system cannot be over-emphasized when liquid line pressure drops are being considered. Proper subcooling is dependent on the proper refrigerant charge and the maximum allowable pressure drop in a liquid line is directly dependent on the amount of subcooling obtained.

If the equivalent length of a liquid line is excessive or if vertical lifts use up a large share of the acceptable pressure drop, it may be necessary to go to the next larger tube size in order to keep the pressure drop within acceptable limits. In some instances a slightly oversized expansion valve can compensate for lower than normal liquid pressure at the valve. (Subcooling must be adequate to prevent "flashing" of liquid R-410A to vapor.) Do not oversize liquid lines any more than necessary because this adds very significantly to the amount of refrigerant in the system which adds cost and increases the danger of slugging.

Since refrigerant oil is miscible with liquid R-410A, at the temperatures encountered in the liquid line, there is normally no problem with oil return in liquid lines.

The remaining portion of Chapter I includes:

Application Guide

Suction Lines for Split Cooling and Heat Pump Systems

Suction lines must return refrigerant vapor and oil from the evaporator to the compressor during system operation; however, due to potential damage to the compressor bearings, valves, scroll sets, or diluting of the oil, should not allow oil or liquid refrigerant to be returned as slugs at any time.

Unless two hermetic compressors are factory engineered and factory assembled to operate on the same refrigerant circuit, each hermetic compressor must be connected to a single refrigeration circuit. If two hermetic compressors are field connected to a single refrigeration circuit, oil will eventually return to only one of the compressors, leaving the other compressor with low or no oil level making proper lubrication impossible.

Do not use evaporator pressure regulating valves (EPR valves) or similar throttling valves in the suction line. Hermetic compressors depend on suction gases for cooling and as the EPR valve throttles down to maintain a constant evaporator pressure, the quantity of suction gas returning to the compressor is reduced and its superheat is increased. The only type of capacity modulation for single stage cooling units recommended (other than multiple units) is a hot gas by-pass system properly applied so as to keep suction gas superheat within normal limits, and provide proper velocity through the evaporator and suction lifts (if any) for adequate oil return.

High superheat will result in improper cooling of the hermetic compressor, while excessively low superheat or improper mixing of hot gas and desuperheating liquid may result in slugging of liquid refrigerant.

Do not tape or otherwise fasten liquid lines and suction lines together unless there is insulation between them. The resultant heat exchange would increase suction gas superheat and may cause overheating of the hermetic compressor. (See Figure 8 for tubing hints)

Suction lines must be insulated to prevent condensation and vapor sealed on the outside to prevent a build-up of moisture in the insulation.

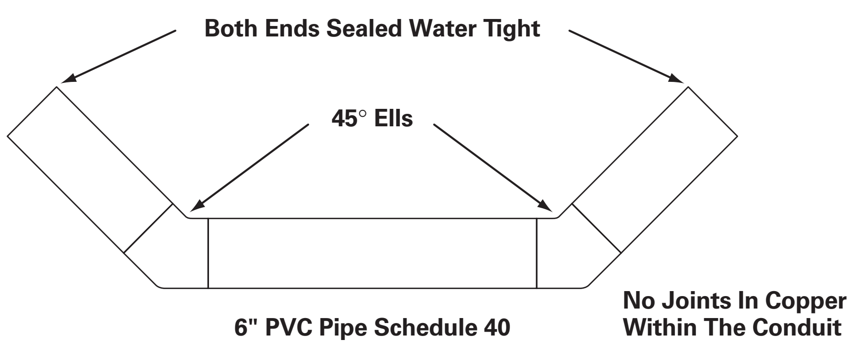
It is advisable to avoid running refrigerant lines underground whenever possible. If it is absolutely necessary to run refrigerant lines underground, they must be run in 6" P.V.C. conduit. (See Figure 1 below.)

Note: Figure 1 is for illustrative purposes only. It is recommended to avoid forming suction line traps within the refrigeration system.

Figure 1

Underground Conduit

(For Underground Refrigerant Lines)



Use 45° elbows to facilitate pulling the tubing through the conduit. The purpose of the conduit is to keep water away from the refrigerant lines. Careful sealing, where the lines enter and leave the conduit is critical. Some installers install a drain in the lower parts of the conduit. Bear in mind, that if the water table rises above the drain, water may be forced **into** the conduit. Vapor and liquid lines must be insulated inside the underground chase.

Application Guide

About Suction Lines and Pressure Drops

ASHRAE recommends that suction line pressure drop be limited to a pressure corresponding to 2°F. (Approximately 5 PSI with R-410A) This is usually not a problem with line lengths of 100 equivalent feet or less.

A quick look at the pressure drop per 100ft. listed in Table 3 reveals that using the largest allowable suction line diameter for each tonnage results in a pressure drop of less than 5 PSI per 100 equivalent feet in all cases.

Obviously, if refrigerant lines exceed 100 equivalent feet significantly, there will be cases where the suction line pressure drop exceeds 5.0 PSI.

In those cases, where long tubing runs result in higher suction line pressure drops than desired, **do not** use a suction line diameter larger than those listed in Table "3" for the system tonnage. To do so would result in refrigerant velocities too low to ensure oil return.

The net capacities indicated in Table 3 for the various equivalent lengths show there is approximately 0.6% loss in capacity for each 1.0 PSI of pressure drop. (Efficiency losses are approximately 0.3% per 1 PSI of pressure drop.)

If the net capacity, indicated for the calculated equivalent length, falls a little short of your requirement (and you have selected the largest allowable tube diameter) one of the following hints may remedy the situation:

1 – Move the outdoor unit closer, if possible.

2 – Use as few elbows as possible, and use long radius elbows to reduce the equivalent length.

3 – Increase the indoor airflow somewhat, within the 350 to 450 CFM per ton limits. (Some latent capacity will be lost.)

4 – Select a different equipment combination that provides the needed capacity.

The pressure drop values shown in Table 3 are not required in order to select a vapor line. They are provided for information only. One example of the pressure drop values use is for evaluating an existing system.

Always select of the vapor line sizes listed in Table 3 for the nominal tonnage of the selected system. These line diameters have been evaluated and will provide the appropriate suction vapor velocity to assure oil return. The lowest possible capacity losses consistent with adequate oil return are addorded by the largest tube size listed. Short tubing runs may provide acceptable losses with a smaller tube size. Net capacities are listed for all approved sizes for equivalent lengths up to 240ft. Please note, for vapor line lengths over 150 linear feet add 2 ounces of oil for every 10 feet over 150 feet.

Application Guide

Table 1

Pounds of R-410A Required for Line Sets

TUBING SIZES	Linear Length								
	40	60	80	100	120	140	160	180	200
1/4" – 5/8"	.4	.7	1.0	1.4	1.7	2.0	2.3	2.6	3.0
5/16" – 3/4"	.7	1.2	1.8	2.3	2.8	3.4	3.9	4.5	5.0
5/16" – 7/8"	.7	1.3	1.9	2.5	3.0	3.6	4.2	4.8	5.4
5/16" – 1-1/8"	.9	1.5	2.2	2.9	3.6	4.3	4.9	5.6	6.3
3/8" – 3/4"	1.0	1.7	2.5	3.2	4.0	4.8	5.5	6.3	7.0
3/8" – 7/8"	1.0	1.8	2.6	3.4	4.2	5.0	5.8	6.6	7.4
3/8" – 1-1/8"	1.1	2.0	2.9	3.8	4.7	5.6	6.5	7.4	8.3
3/8" – 1-3/8"	1.3	2.3	3.4	4.4	5.5	6.5	7.5	8.6	9.6
1/2" – 7/8"	1.7	3.1	4.4	5.8	7.1	8.5	9.9	11.2	12.6
1/2" – 1-1/8"	1.8	3.3	4.7	6.2	7.7	9.1	10.6	12.0	13.5
1/2" – 1-3/8"	2.0	3.6	5.2	6.8	8.4	10.0	11.6	13.2	14.8
5/8" – 1-5/8"	2.2	4.0	5.7	7.5	9.2	11.0	12.8	14.5	16.3
5/8" – 1-3/8"	3.0	5.4	7.7	10.1	12.5	14.9	17.3	19.6	22.0
5/8" – 1-5/8"	3.2	5.7	8.3	10.8	13.3	15.9	18.4	21.0	23.5

Note: The 15 ft. of tubing included in the nameplate charge has been accounted for, use actual linear length with the above table.

Figure 2

R-410A Temperature and Pressure Chart

TEMP.	R-410A	TEMP.	R-410A	TEMP.	R-410A
-60	1.2	16	71.7	44	127.3
-55	3.4	17	73.3	45	129.7
-50	5.8	18	75.0	46	132.2
-45	8.6	19	76.6	47	134.6
-40	11.6	20	78.3	48	137.1
-35	14.9	21	80.2	49	139.6
-30	18.5	22	81.8	50	142.2
-25	22.5	23	83.6	55	155.5
-20	26.9	24	85.4	60	169.6
-15	31.7	25	87.3	65	184.6
-10	36.8	26	89.1	70	200.6
-5	42.5	27	91.0	75	217.4
0	48.6	28	92.9	80	235.3
1	49.9	29	94.9	85	254.1
2	51.2	30	96.8	90	274.1
3	52.5	31	98.8	95	295.1
4	53.8	32	100.8	100	317.2
5	55.2	33	102.9	105	340.5
6	56.6	34	105.0	110	365.0
7	58.0	35	107.1	115	390.7
8	59.4	36	109.2	120	417.7
9	60.9	37	111.4	125	445.9
10	62.3	38	113.6	130	475.6
11	63.8	39	115.8	135	506.5
12	65.4	40	118.0	140	539.0
13	66.9	41	120.3	145	572.8
14	68.5	42	122.6	150	608.1
15	70.0	43	125.0	155	645.0

Application Guide

Table 2 (R-410A) – Liquid Line Selection For R-410A Single Speed

Chart based on 10ºF Sub Cooling

Maximum Allowable Liquid Line Pressure Drop .... = 50 PSI
Subtract .43 PSI for each foot of Liquid Lift (if any) ....
Do Not Exceed this value when selecting Liquid Line. ....

Tube	Rated	Pressure Drop (PSI) For Total Equivalent Length											
O.D.	BTUH	20'	40'	60'	80'	100'	120'	140'	160'	180'	200'	220'	240'
1/4"	15000	4.5	9.0	13.6	18.1	22.6	27.1	31.6	36.2	40.7	45.2	49.7	—
	18000	6.3	12.6	18.8	25.1	31.4	37.7	44.0	—	—	—	—	—
	24000	15.4	30.8	46.2	—	—	—	—	—	—	—	—	—
	15000	1.2	2.4	3.5	4.7	5.9	7.1	8.3	9.4	10.6	11.8	13.0	14.2
	18000	1.6	3.3	4.9	6.6	8.2	9.8	11.5	13.1	14.8	16.4	18.0	19.7
5/16"	24000	2.8	5.5	8.3	11.0	13.8	16.6	19.3	22.1	24.8	27.6	30.4	33.1
	30000	4.1	8.3	12.4	16.6	20.7	24.8	29.0	33.1	37.3	41.4	45.5	49.7
	36000	5.8	11.6	17.3	23.1	28.9	34.7	40.5	46.2	—	—	—	—
	42000	7.7	15.4	23.0	30.7	38.4	46.1	—	—	—	—	—	—
	24000	1.0	1.9	2.9	3.8	4.8	5.8	6.7	7.7	8.6	9.6	10.6	11.5
	30000	1.4	2.9	4.3	5.8	7.2	8.6	10.1	11.5	13.0	14.4	15.8	17.3
3/8"	36000	2.0	4.0	6.1	8.1	10.1	12.1	14.1	16.2	18.2	20.2	22.2	24.2
	42000	2.7	5.3	8.0	10.6	13.3	16.0	18.6	21.3	23.9	26.6	29.3	31.9
	48000	3.4	6.8	10.2	13.6	17.0	20.4	23.8	27.2	30.6	34.0	37.4	40.8
	60000	5.1	10.3	15.4	20.6	25.7	30.8	36.0	41.1	46.3	—	—	—
	42000	.5	1.1	1.6	2.2	2.7	3.2	3.8	4.3	4.9	5.4	5.9	6.5
	48000	.7	1.4	2.0	2.7	3.4	4.1	4.8	5.4	6.1	6.8	7.5	8.2
1/2"	60000	1.0	2.1	3.1	4.2	5.2	6.2	7.3	8.3	9.4	10.4	11.4	12.5

Note 1: A blank space indicates a pressure drop of over 50 PSI.
Note 2: Other existing sources of pressure drop, (solenoid valves, etc.) must be considered.
Note 3: A vertical run with a heat pump system always results in a liquid lift (heating or cooling).
Note 4: The smallest liquid line diameter that results in a total liquid line pressure drop of 50 PSI or less results in the most reliable system (fewer pounds of R-410A).
Note 5: It is recommended to place units where 1/2" liquid line is not required due to the increased refrigerant volume imposed by the larger liquid line.
Note 6: At the time this manual was printed all outdoor units were rated with 3/8" liquid line.

Example

Given: Rated system capacity = 42000 BTUH, 68 linear ft., 4 long radius elbows (no solenoid valve or other source of pressure drop): 20 ft. liquid lift.

Step #1 20 x .43 = 8.6 PSI pressure drop due to liquid lift. 50 minus 8.6 = 41.4 PSI available for friction loss.

Step #2 68 + (4 x 3.2) = 80.8 eq. ft. (See Table 4, page 11, for equivalent lengths.)

Step #3 Referring to Table 2, we find that 80 ft. of 5/16" liquid line, (42,000 BTUH) = 30.7 PSI pressure drop. (Well within our 41.4 PSI limit.)

Allowable Suction Line Diameters for R410A Single Speed Split Systems and BTUH Loss Versus Equivalent Length

Table 3

Nominal Tons	Tube O.D. (Inches)	Press. Drop PSI/100 Ft.	BTUH Loss For Equivalent Length 40'	60'	80'	100'	120'	140'	160'	180'	200'	220'	240'
1.0	1/2	5.0	70	160	250	340	430	520	610	700	790	880	970
	5/8	1.5	20	50	73	100	130	155	180	210	235	265	290
1.5	1/2	10.8	173	410	640	875	1110	1340	1575	1810	2040	2275	2510
	5/8	3.1	50	120	185	250	320	385	450	520	585	655	720
	3/4	1.2	20	45	70	95	125	150	175	200	225	255	280
2.0	5/8	5.4	115	270	430	585	740	895	1050	1205	1360	1515	1670
	3/4	2.0	45	100	160	215	275	330	390	445	505	560	620
2.5	5/8	8.2	220	515	810	1110	1400	1695	1990	2290	2585	2880	3175
	3/4	3.0	80	190	295	405	515	620	730	840	945	1055	1160
	7/8	1.3	35	80	130	175	220	270	315	365	410	455	505
3.0	5/8	11.7	380	885	1390	1895	2400	2905	3410	3915	4425	4930	—
	3/4	4.3	140	325	510	700	880	1070	1255	1440	1625	1810	2000
	7/8	1.9	60	145	225	310	390	470	555	635	720	800	880
3.5	3/4	5.8	220	510	805	1095	1390	1680	1975	2265	2560	2850	3140
	7/8	2.5	95	220	345	475	600	725	850	975	1105	1230	1355
4.0	3/4	7.4	320	745	1170	1600	2025	2450	2875	3305	3730	4155	4580
	7/8	3.2	140	325	510	690	875	1060	1245	1430	1615	1795	1980
	1-1/8③	.9	40	90	145	195	245	300	350	400	455	505	555
5.0	3/4	11.5	620	1450	2280	3105	3935	4760	5590	6415	7245	8073	8900
	7/8	4.9	265	615	970	1325	1675	2030	2380	2735	3080	3440	3795
	1-1/8	1.3	70	165	255	350	445	540	630	725	820	915	1005

Note 1: Shaded value indicates more than 10% capacity loss.
Note 2: Blank space indicates more than 15% capacity loss.
Note 3: Only approved for cooling units, do not use 1 1/8" vapor lines on heat pumps less than 5 ton.
Note 4: If linear length exceeds 150 feet, add 2 ounces of approved compressor oil per every 10 feet in excess of 150 feet. (Example: if the actual line length is 170 feet, add 4 ounces of oil to the system)

Suction Line Selection Example (R-410A)

Given: 4 ton system
132 linear ft.
8 long radius elbows

The equivalent length of the rated, (7/8" O.D.) suction line size = 132 + (8 x 5.3) or 174.4 ft. Table 8 indicates a capacity loss of 1430 BTUH for 180 equivalent feet (approx. 3%). If this loss is acceptable, 7/8" O.D. is the correct size.

Table 4

Equivalent Length (Ft.) of Non-Ferrous Valves and Fittings (Brazed)

O.D. Tube Size (Inches)	Globe Valve	Angle Valve	Short Radius Ell	Long Radius Ell	Tee Line Flow	Tee Branch Flow
1/2*	70	24	4.7	3.2	1.7	6.6
5/8	72	25	5.7	3.9	2.3	8.2
3/4	75	25	6.5	4.5	2.9	9.7
7/8	78	28	7.8	5.3	3.7	12.0
1-1/8	87	29	2.7	1.9	2.5	8.0
1-3/8	102	33	3.2	2.2	2.7	10.0
1-5/8	115	34	3.8	2.6	3.0	12.0

Information for this chart extracted by permission from A.R.I. Refrigerant Piping Data, page 28.
* For smaller sizes, use 1/2" values.

Question

Would a 3/4" O.D. suction line be adequate for a 4 ton system with a piping run of 60 equivalent feet?

Answer

Obviously, oil return would not be a problem with the smaller diameter tube, (higher velocity). So, if the capacity loss of 745 BTUH, (approx. 1.5%) is not a problem, the 3/4" suction line is O.K. for the 60 equivalent feet.

Application Guide

Allowable Vapor and Liquid Line Diameters for Multistage Split Systems

Table 5

OD Unit nominal capacity	Rated Vapor Line OD	Vapor Service Valve OD	Minimum Alternative Vapor Line OD	Maximum Alternative Vapor Line OD	Rated Liquid Line OD	Liquid Service Valve OD	Minimum Alternative Liquid Line OD	Maximum Alternative Liquid Line OD
Two Step Scroll Models								
2 ton CLG	5/8"	5/8"	5/8"	3/4"	5/16"	5/16"	5/16"	3/8"
3 ton CLG	3/4"	3/4"	5/8"	7/8"	3/8"	3/8"	5/16"	3/8"
4 ton CLG	7/8"	7/8"	3/4"	7/8"	3/8"	3/8"	3/8"	3/8"
5 ton CLG	7/8" or 1 1/8"	7/8"	3/4"	1 1/8"	3/8"	3/8"	3/8"	3/8"
2 ton HP	5/8"	5/8"	5/8"	3/4"	5/16"	5/16"	5/16"	3/8"
3 ton HP	3/4"	3/4"	5/8"	7/8"	3/8"	3/8"	5/16"	3/8"
4 ton HP	7/8"	7/8"	3/4"	7/8"	3/8"	3/8"	3/8"	3/8"
5 ton HP	7/8" or 1 1/8"	7/8"	3/4"	1 1/8"	3/8"	3/8"	3/8"	3/8"
Two Compressor Models								
2 ton CLG	3/4"	5/8"	5/8"	3/4"	3/8"	3/8"	5/16"	3/8"
3 ton CLG	3/4"	3/4"	5/8"	7/8"	3/8"	3/8"	5/16"	3/8"
4 ton CLG	7/8"	3/4"	3/4"	7/8"	3/8"	3/8"	3/8"	3/8"
5 ton CLG	7/8"	3/4"	3/4"	7/8"	3/8"	3/8"	3/8"	3/8"
2 ton HP	5/8"	5/8"	5/8"	3/4" - 50 feet max length	3/8"	3/8"	5/16"	3/8"
3 ton HP	3/4"	3/4"	5/8"	3/4"	3/8"	3/8"	5/16"	3/8"
4 ton HP	3/4"	3/4"	3/4"	3/4"	3/8"	3/8"	3/8"	3/8"
5 ton HP	3/4"	3/4"	3/4"	7/8" - 50 feet max length	3/8"	3/8"	3/8"	3/8"

Two compressor models share the same refrigeration circuit, however, do not operate simultaneously. Therefore, it is crucial that refrigerant lines be properly sized and do not exceed the length set forth in this guide. Please note it is recommended to use the service valve connection size tubing.

A. Limitations:

1. Line length limits as shipped:
   A. Vapor line = 80 feet linear length / of the linear length, 25 feet may be installed vertical.
   B. Liquid line = 80 feet linear length / of the linear length, 25 feet may be installed vertical.
   C. When using an alternate diameter tube follow procedure form page 10 and 11 to assure maximum vapor and liquid pressure drop are not exceeded.

B. Explanation:

1. Refrigerant lines shall not exceed 80 feet total line length. / 25 feet of the 80 feet may be vertical.
   A. Liquid sub-cooling may not be achieved on second stage if the liquid line exceeds 80 feet.
   B. Oil return may be sacrificed during first stage operation if the vapor line exceeds 80 feet.

Application Guide

SECTION II

Condensing Units Utilizing Micro-channel Coil Technology

Refrigeration Piping

This section deals specifically with the unique refrigeration piping requirements for the Condensing Unit(s).

A. Piping Limits:

1. Below 110F° outdoor ambient design temperature:
   a) Reference the limits indicated in Table 6
   b) Do not exceed 60 feet vertical change with the outdoor unit below the indoor unit.
   c) Do not exceed 200 feet vertical change with the outdoor unit above the indoor unit
2. Above 110F° outdoor ambient design temperature:
   a) A. Maximum line length = 100 feet.
   b) Maximum vertical length = 60 feet.
3. Compressor crankcase heat is required for line lengths over 80 feet.
4. It is recommended for new installations to use only the line diameters in Table 5. However, in a retrofit application, the line listed in Table 6 may be used.

B. Charging Methods:

1. The recommended charging method is indicated in the Condensing Unit(s) installer's guide. Please refer to the installation manual publication 18-AC78D1-** (latest publication).
2. Reference Table 8 if the unit cannot be started at the time of install. If the charge is weighed in, it is recommended to return to the job site in order to verify the system is charged correctly based on the installation instructions.

Table 6

Unit Size	Line Diameters		Service Valve Connection	
	Vapor Line	Liquid Line	Vapor Line	Liquid Line
1.5 Ton	5/8"	3/8"	5/8"	3/8"
2.0 Ton	5/8"	3/8"	5/8"	3/8"
2.5 Ton	3/4"	3/8"	3/4"	3/8"
3.0 Ton	3/4"	3/8"	3/4"	3/8"
3.5 Ton	7/8"	3/8"	7/8"	3/8"
4.0 Ton	7/8"	3/8"	7/8"	3/8"
5.0 Ton	7/8"	3/8"	7/8"	3/8"

Application Guide

Alternate Liquid Lines

UseTable 7 for retrofit applications. This table should not be used for systems installed in geographical locations where the outdoor ambient exceeds 110F°.

ForTable 7:

Each short radius elbow accounts for 4.7 equivalent feet.

Each long radius elbow accounts for 3.2 equivalent feet.

Table 7

Allowable Liquid Line Diameters For Condensing Units Utilizing Micro-Channel Coil Technology													
Refrigerant Line OD	Nominal Capacity in Tons	Pressure Drop (PSI) Versus Equivalent Length											
		Equivalent Length (feet)											
		10	20	40	60	80	100	120	140	150*	160	180	200
1/4"	1.5	3.1	6.2	12.6	18.8	25.1	31.4	37.7	43.9	47.1			
	2	7.7	15.4	30.8	46.2								
5/16"	1.5	0.8	1.6	3.2	4.9	6.5	8.2	9.8	11.5	12.3	13.1	14.7	16.4
	2	1.3	2.7	5.5	8.2	11	13.8	16.5	19.3	20.7	22.1	24.8	27.6
	2.5	2	4.1	8.2	12.4	16.5	20.7	24.8	28.9	31	33.1	37.2	41.4
	3	2.8	5.7	11.5	17.3	23.1	28.9	34.6	40.4	43.3	46.2		
3/8"	1.5	0.4	0.7	1.5	2.2	2.9	3.7	4.5	5.2	5.6	5.9	6.7	7.4
	2	0.5	0.9	1.9	2.9	3.8	4.8	5.7	6.7	7.2	7.7	8.6	9.6
	2.5	0.72	1.4	2.9	4.3	5.7	7.2	8.6	10.1	10.8	11.5	12.9	14.4
	3	1	2	4	6.1	8.1	10.1	12.1	14.1	15.1	16.1	18.2	20.2
	3.5	1.3	2.6	5.3	7.9	10.6	13.3	15.9	18.6	19.9	21.3	23.9	26.6
	4	1.7	3.4	6.8	10.2	13.6	17	20.4	23.8	25.5	27.2	30.6	34
	5	2.6	5.1	10.3	15.4	20.5	25.7	30.8	35.9	38.5	41.1	46.2	

**If linear length exceeds 150 feet, add 2 ounces of approved compressor oil per every 10 feet in excess of 150 feet. (Example, if the actual line length is 170 feet, add 4 ounces of oil to the system)*

Charge Adjustment

Units ship from the manufacturer with enough refrigerant for 15 feet of the rated line diameters. All micro-channel units are rated with 3/8" liquid lines. Table 8 provides data for removing or adding R-410A.

Table 8

Charge Adjustment Table Based On Liquid Line Diameter and Length														
Factory Connection	Installed Liquid Line	Charge adjustments in pounds of R-410A												
		Actual Line length												
		0	10	15	20	40	60	80	100	120	140	160	180	200
3/8"	1/4"	-0.60	-0.43	-0.35	-0.25	0.05	0.35	0.65	1.05	1.35	1.65	1.95	2.25	2.65
3/8"	5/16"	-0.60	-0.28	-0.15	0.05	0.55	1.05	1.65	2.15	2.65	3.25	3.75	4.35	4.85
3/8"	3/8"	-0.60	-0.40	0.00	0.20	1.00	1.80	2.60	3.40	4.20	5.00	5.80	6.60	7.40

Application Guide

Section III

High Rise Heat Pump Applications (R-410A)

The demand for greater vertical separation between the indoor and outdoor sections of heat pumps systems, over the years, has led to the development of the high rise system. Without using the high rise system, the longest vertical separation between a split system heat pump and indoor coil is limited to a pressure drop of 50 PSI or less. (This assumes a minimum 10°F sub-cooling) If more vertical separation is required than allowed by the unit's installation manual, this guide can be of assistance.

If the liquid pressure drop is calculated (equivalent length + vertical lift) to produce more than 50 PSI, the high rise system is required. The high rise system is to be applied to heat pump systems only and only on systems where the outdoor unit is above the indoor unit. Thermal expansion valve is the only acceptable refrigeration control when applying a heat pump with the high rise system.

The high rise system consists of a properly sized capillary tube and a suction to liquid heat exchanger produced by Refrigeration Research. The Refrigeration Research part number is Model H-100 (This same part can be obtained by ordering Service First part number EXC01082).

The purpose of the subcooler is to provide subcooling beyond the 10° typically provided by standard systems. This is necessary in order to tolerate the higher liquid line pressure drops resulting from high liquid lifts (plus friction loss) without "flashing" of liquid refrigerant to vapor. This "flashing," when it occurs, chokes up the liquid line with large volumes of vapor, as well as substantially reducing the capacity of the metering device because of the mixture of vapor and liquid it would be forced to handle.

It is not unusual for high rise systems to operate with total liquid line pressure drops in excess of 100 PSI without flashing liquid refrigerant to vapor.

As mentioned earlier, the interactive piping program will call for the subcooler when ever it is required, size the capillary tube and call for any other required accessories. The rest of this chapter is designed to help the system designer who does not have access to the computer program to apply the high rise system.

The heat exchanger used with the high rise system (Refrigeration Research #H-100, or Heat-X 3/4 HP) has sufficient heat exchange capacity to provide the required additional subcooling for systems up through 10 tons.

It should be noted that although the heat exchanger used with the high rise system is designed as a suction to liquid heat exchanger, it is not used in that manner. (Suction gas is not routed through the heat exchanger.) Instead, the normal liquid flow is through the suction side of the heat exchanger. A small portion of the liquid is fed through the capillary tube to the other side of the heat exchanger where it is evaporated to chill the liquid refrigerant the required number of degrees. A 3/8" O.D. suction line (insulated) is run from the heat exchanger (located at the bottom of the liquid lift) to the common suction line of the outdoor unit (between the switch-over valve and the compressor).

The 3/8" O.D. suction line is teed into the top of a horizontal common suction line, or into the side of a vertical common suction line, thus preventing the drainage of oil down the 3/8" O.D. tube.

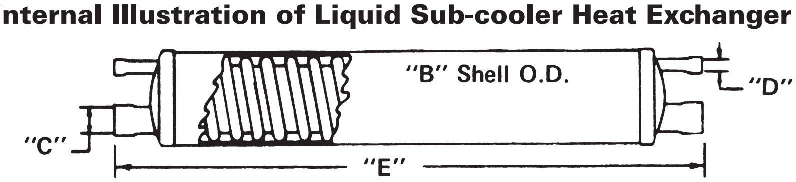
Table 9

Subcooling Heat Exchangers

Catalog Number	H.P.	Shell O.D. (B) (Inches)	Overall Length (E) (Inches)	Suction Line (C) (Inches)	Liquid Line (D) (Inches)	Weight (Pounds)
H 33	1/4 & 1/3	1-1/4	8-5/8	3/8	1/4	.8
H 50	1/2	2	10	1/2	1/4	1.3
H 75	3/4	2	12-1/8	5/8	1/4	1.7
H 100	1	2	13-1/8	5/8	3/8	1.9
H 150	1-1/2	2	17-3/8	7/8	3/8	2.5
H 200	2	3	13-1/4	7/8	3/8	3.1
H 300	3	3	15-1/4	1-1/8	3/8	3.8
H 500	5	5	14-3/8	1-1/8	1/2	7.0
H 750	7-1/2	5	15-5/8	1-5/8	5/8	9.0
H 1000	10	5	18-5/8	1-5/8	5/8	11.0

Figure 3

Internal Illustration of Liquid Sub-cooler Heat Exchanger



Application Guide

The fact that a small portion of the liquid refrigerant, being circulated, is diverted to the heat exchanger, and boiled to a vapor, has no effect on system capacity. While a slightly reduced quantity of liquid refrigerant is delivered to the system evaporator, each pound contains less heat, because of the additional subcooling and the net cooling effect is the same.

The heat exchanger and capillary tube are to be purchased at your local parts wholesaler.

Table "9" page 15, provides a picture and dimensional information for the heat exchanger.

Note that the heat pump indoor unit **must** utilize expansion valve refrigerant control.

SUBCOOLER INSTALLATION NOTES

For Heat Pump High Rise Applications

1. Recover the refrigerant charge from the heat pump unit.
2. Evacuate the heat pump unit to remove any refrigerant that may be absorbed in the compressor oil.
3. Let the pressures equalize.
4. Cut the common suction tube in between the SOV and the Compressor
5. Braze a "T" fitting between the compressor and SOV.
6. Connect the subcooler in the refrigeration circuit as shown in figures 4 & 5.
7. Remove and replace the factory installed liquid line filter drier.

Service First Part Numbers	
Sub-cooler	EXC01082

Note: Do not leave system open more that 4 hours.

Figure 4 (below) indicates the hook-up for the heat exchanger and capillary tube. The heat exchanger is to be located at the bottom of the liquid lift (near the indoor unit).

Figure 5, page 17, shows the piping hook-up between the indoor and outdoor units.

Note that there are now three connecting lines between the indoor and outdoor units (liquid line, gas line and a 3⁄8" insulated suction line) running from the heat exchanger to the common suction line (between the switchover valve and the compressor).

Figure 4

Piping Detail – Heat Exchanger

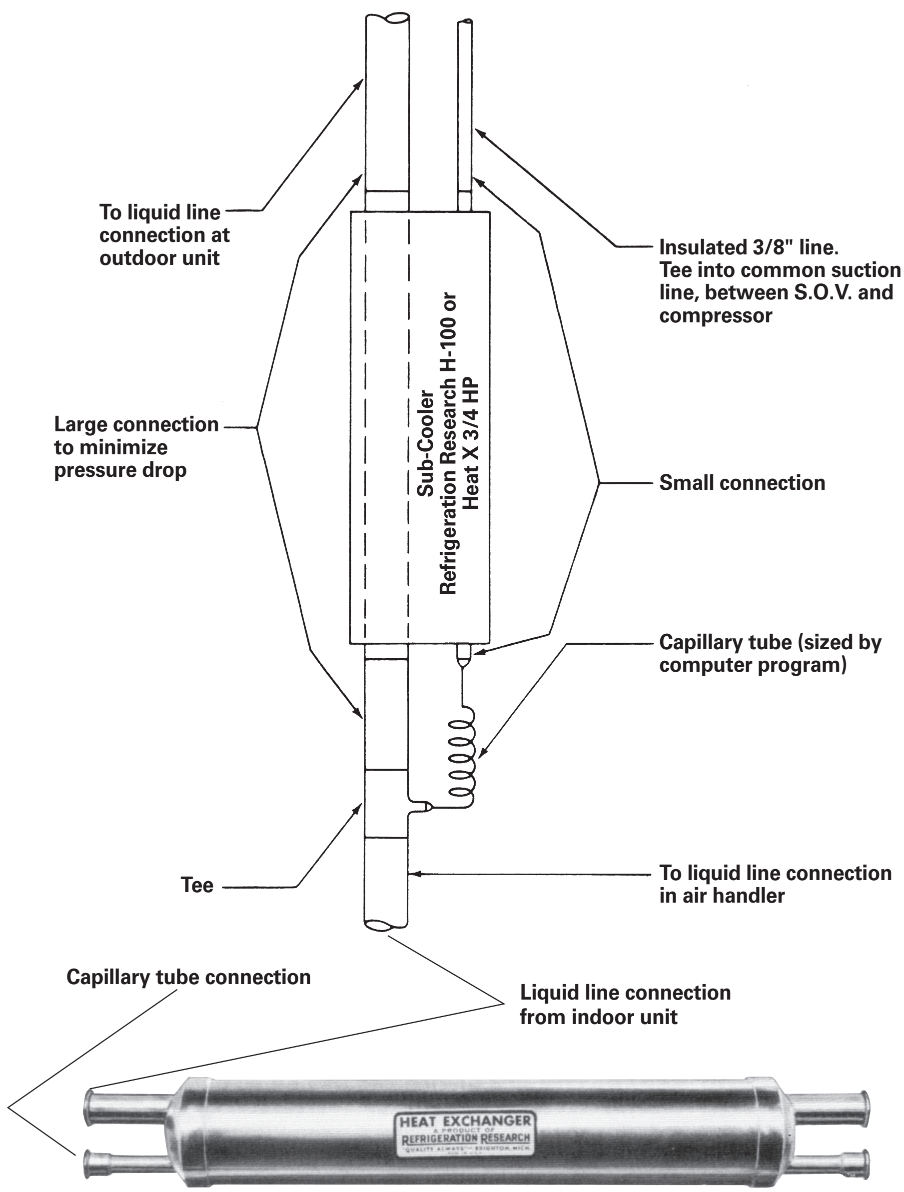
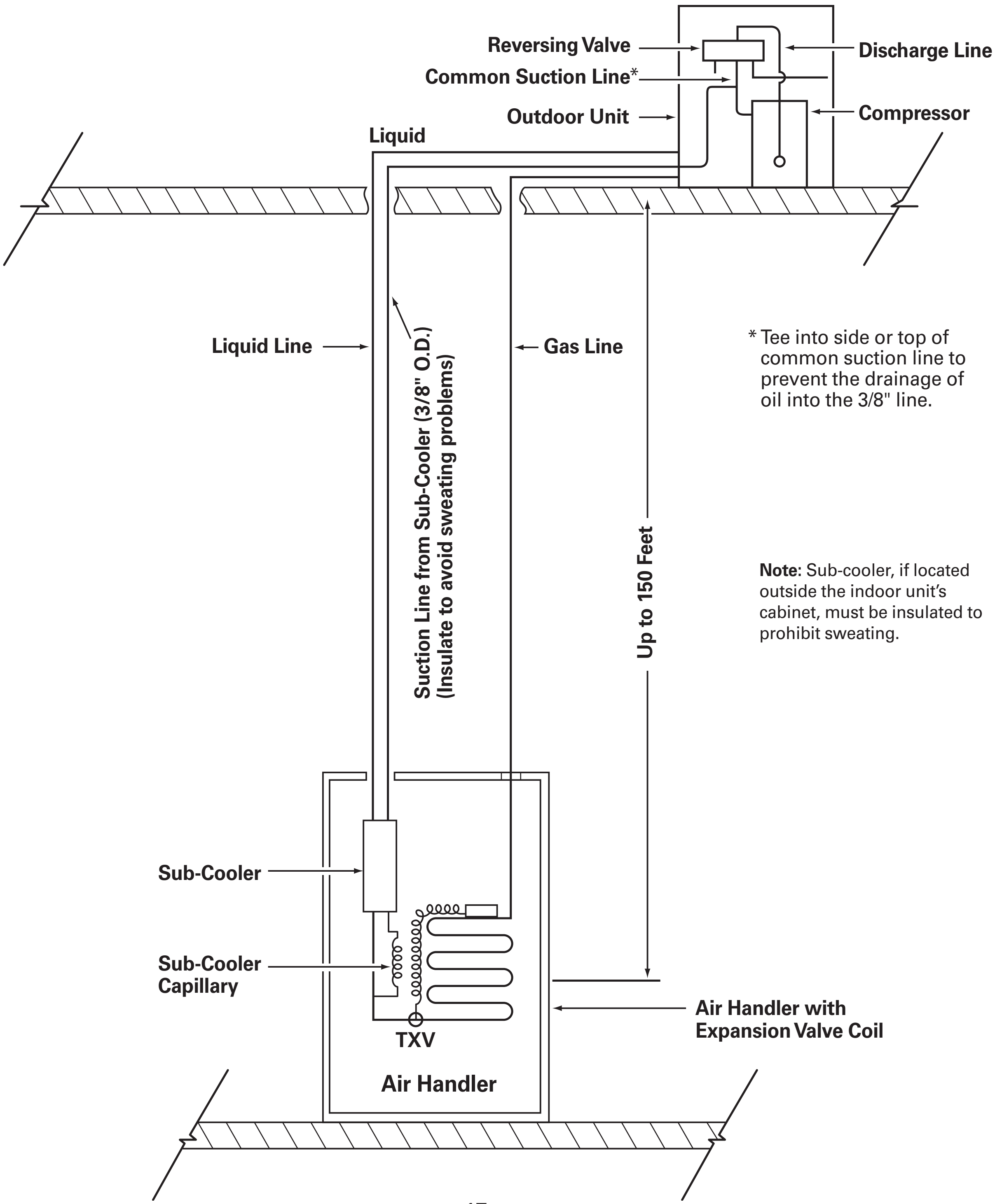


Figure 5

High Lift Heat Pump Piping Schematic (Outdoor Unit above Air Handler)



Application Guide

Table 10 (R-410A) allows you to select the proper capillary tube size, based on excess liquid line pressure drop and system tonnage. The example below illustrates typical calculations for a system utilizing (R-410A).

Given: R-410A Subcooler, 3/8" O.D. liquid line, 195 equivalent feet, 182 ft. liquid lift (3 1/2 tons).

Step #1 Friction loss from Table 7 (13.3 x 1.95) = 25.9 PSI.

Step #2 Pressure drop due to lift (182 x .43) = 78.3 PSI.

Step #3 Total pressure drop (25.9 + 78.3) = 104.2 PSI.

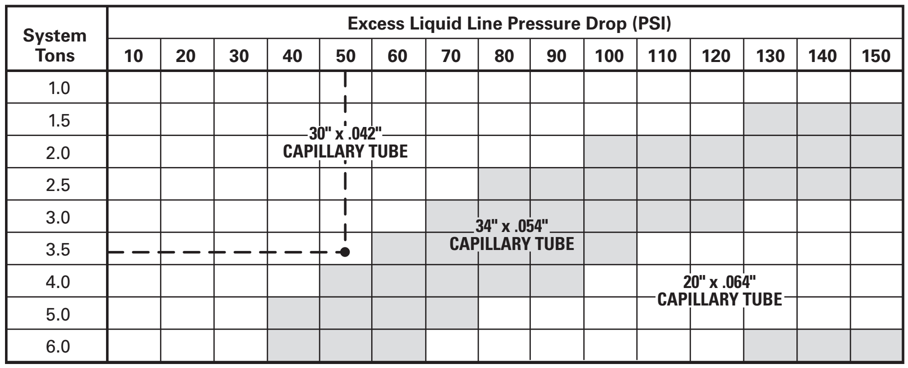
Step #4 Excess pressure drop (104 – 50) = 54 PSI.

Step #5 From Table 12, 3 1/2 tons at 54 PSI excess pressure drop requires a 30" x .042" capillary tube.

Table 10

Capillary Tube Selection Table for R-410A Subcooler

(Total Liquid Line Pressure Drop Minus 50 PSI = Excess Pressure Drop)



Example: 3-1/2 ton system with 54 PSI excess pressure drop requires 30" x .042" capillary tube.

Application Guide

Section IV

Compressor protection and piping limitations

A. Compressor Protection: compressor crankcase heaters, expansion valves, limited line lengths, suction line accumulators, and solenoid valves.

1. Some compressor protection methods are listed below. The two basic devices utilized today are compressor crankcase heaters and expansion valves. Other devices such as solenoid valves are discussed within this section.
   a) Systems with refrigeration lines that exceed 60 feet or in which the total system charge exceeds the limits described in subsection C, require a compressor crankcase heater and a thermal expansion valve or electronic expansion valve where applicable.
   b) Add compressor crankcase heat if system charge exceeds:
      1. 6 lbs in systems utilizing 1 cylinder reciprocating compressors
      2. 12 lbs in systems utilizing 2 cylinder reciprocating compressors
      3. 8 lbs in systems utilizing small diameter scroll compressors
      4. 10 lbs in systems utilizing large diameter scroll compressors
   c) Consult the current product data catalog for factory installed components
2. Lines should be kept as short as possible.
   a) It is recommended to maintain a liquid line pressure drop that is within 50 PSI with the rated liquid line. (In most cases, the rated liquid line is 3/8")
   b) If a larger line diameter than the rated liquid line is required, a suction line accumulator may be necessary, or other means of refrigerant isolation.
   c) If the system charge is greater than 12 lbs. in a heat pump system, a suction line accumulator is required to be installed between the compressor and reversing valve.
      1) Some heat pump units have factory installed suction line accumulators. If additional capacity is required, the factory installed accumulator will need to be replaced with a larger capacity accumulator.
   d) Only use a vapor line that is listed in table 3 for or table 5.
3. Suction line accumulators, if required, should be sized to hold 1/2 of the total system charge. (Unit Nameplate + Additional charge for the refrigerant lines) – See table 1
4. If refrigerant lines are longer than 150 feet, add 2 ounces of oil for every 10 feet over 150 feet. (Example: if line is 170 feet, add 4 ounces of compressor oil).
   a) Obtain oil from Service First Distribution or the local Trane distributor.
5. Liquid line solenoid valves, if used in a heat pump system, shall be bidirectional. The pressure drop imposed by the valve must be considered in both directions of flow. The pressure drop in either direction shall not exceed 50 PSI for systems with 10F subcooling.
6. Liquid line solenoid valves in cooling systems (non-heat pumps).
   a) If the compressor is above the indoor unit, locate the liquid line solenoid valve within 10 feet of the indoor unit.
      1) Solenoid valves shall be used for refrigerant isolation purposes. Wire the solenoid to open when the compressor is energized and to close when the compressor is de-energized by the system controls.
      2. Solenoid valves require a separate transformer. In most cases the factory installed transformer is not large enough to power the solenoid and other low voltage controls.

B. Refrigeration piping limits:

1. Single speed cooling systems: (AHRI rated)
   a) Line length with the condensing unit above the indoor unit:
      1) Maximum of 200 feet, 200 of this may be vertical. (Suction lift)
   b) Line length with the condensing unit below the indoor unit:
      1) Liquid line pressure drop shall not exceed 50 PSI or 60 feet maximum vertical separation.
2. Single speed heat pump systems:
   a) In most cases the lines can exceed 150 feet without any vertical separation. However, it is recommended to maintain a line length in which the liquid line pressure drop does not exceed 50 PSI with the rated liquid line.
   b) Heat pump outdoor section above the indoor unit:
      1) Maximum vertical change is 150 feet using the high rise system. Otherwise, 60 feet or 50 PSI is the pressure drop maximum vertical separation in most cases.
   c) Heat pump outdoor section below the indoor unit:
      1) Maximum vertical separation is 60 feet in most cases.
3. Multistage systems:
   a) Please reference Table 5.

C. Suction Traps

1. If using an AHRI rated system combination and the indoor coil is manufactured by Trane or American Standard Heating and Air Conditioning, suction traps are not required and not recommended.
2. If using a Trane Blower Coil (BCHC / BCVC) or Trane LPCA or M series indoor unit, it is recommended to install a suction trap leaving the evaporator coil.

D. Article D – Reference SS-APG010-EN for piping split small split systems when utilizing a Trane BCHC or BCVC blower coil.

E. Please contact the your Territory Manager, Field Service Representative or Trane Commercial Sales Engineer if there are questions regarding this manual.

1. FSR's, TM's and Trane SE's contact Application Engineering.

Application Guide

Section V

Refrigerant Handling, Retrofit, and Reusing Existing Refrigerant Lines

I. Basic practices for HVAC systems

The following guidelines should be observed when installing any refrigerant bearing system in order to assure reliable and efficient operation.

These practices apply regardless of refrigerant type.

1. Use only compatible indoor and outdoor coil combinations.
   a) Retrofitted systems in most cases, will not be rated in the AHRI directory.
      1) There is no plan to supply ratings to AHRI for retrofitted systems.
   b) Sources for rated system combinatons:
      1. www.ahrinet.org
      2. www.comfortsite.com
      3. Electronic Performance Data
      4. Local distributor or sales office
2. Refrigerant line sets must be sized properly.
   a Refer to the current refrigerant piping publications:
      1) SS-APG006-EN for 1.5 through 5.0 ton split systems.
      2) SS-APG008-EN for 6 through 20 ton split systems.
3. Sealed refrigerant systems must be kept clean!
   a) De-burr all ends of copper tubing to assure free flow of refrigerant.
   b) Use emery cloth or scuff pad to clean the ends of the copper tube to assure good braze joints. Special care should be taken to eliminate any shavings from entering the tubing.
   c) Always allow dry nitrogen to flow through the refrigerant lines to prohibit oxidation while brazing.
   d) Refer to Service Procedures publication 34-1005 and / or Service Procedures publication 34-3458-01 for proper brazing techniques.
4. Moisture compromises performance and operation of all HCFC / HFC refrigerant systems.
   a) Keep all oil bearing containers sealed tight until ready to use.
   b) Do not open system service valves until ready to start up.
      1) Leak check system as soon as possible after brazing joints.

⚠ WARNING

Explosion Hazard!
NEVER leak test with air and R410A. At pressures above 1 atmosphere, just like R-22, mixture of R-410A and air can be combustible! Failure to follow this warning could result in property damage, personal injury or death.

      2) Begin evacuation process immediately after leak check process is completed.
         a) Do not leave service valves open longer than four hours. Moisture absorbed into the system is removable only by driers.
         a) Evacuate to 300 microns, then close all service gauge valves. After one minute, if the reading on the micron gauge rises above 500 microns, leak check and evacuate the system again. *(Please note that a micron gauge must be used, dial gauges cannot read microns.)*
5. Proper refrigerant handling is a must.
   a) Service personnel must be properly certified in order to handle refrigerant!
   b) Refrigerant in existing system must be recovered in accordance with all applicable federal, state and local standards.
      1) New or recycled refrigerant must be used when charging a system
      2) Recovered refrigerant must be recycled at an approved recycling facility or disposed of in accordance with national, state, and local standards.

II. Refrigerant Lines

A. Compatible line diameters and lengths:
   When replacing any HVAC system, the existing refrigerant lines must be evaluated to determine if they are properly sized for the new system. Refer to Table 3 for compatible vapor line and maximum lengths and Table 2 for approved liquid line sizes, length and vertical change. For multistage reference Table 5.
B. Preparation for re-use:
   After verifying the line diameters and length are compatible, the existing line set must be cleared of as much existing mineral oil and contaminants as possible. The lines should be cleared prior to installing new outdoor and / or indoor units.

Existing indoor equipment:

Reference APP-APG011-EN or APP-APG012-EN. In January 2001, the Department of Energy published a rule under NAECA (National Appliance Energy Conservation Act of 1987) to advance the minimum SEER to 13 effective January 23rd, 2006 for single phase HVAC systems. (three phase systems with capacity ratings under 65,000 BTUH followed at a later date). In 1992, the minimum SEER was increased to 10.0. Prior to 1992, SEER ratings were as low as 7.0. In addition, in 2006, HSPF was required to be 7.7. Before 2006 HSPF's were as low as 6.8.

Some of the ways in which the HVAC industry achieved the higher efficiencies include:

A. Increased air to coil surface contact.
   1) Normally, increasing the air to coil surface contact meant the refrigeration coils increased in both physical size as well as refrigerant volume size.
B. Compressor motor, blower motor, and fan motor efficiencies increased.
C. Improved heat transfer technologies
D. Combination of the above.

Application Guide

Regardless of the method, HVAC systems manufacturers typically offer their brand of furnace coils and fan coil units that are specifically designed and tested to operate with compatible outdoor air conditioning units and heat pump units. For the reasons listed, when replacing an outdoor unit, it is in the customer's best interest to replace the indoor furnace coil or fan coil unit. In the case of a heat pump system, the above cannot be overstated since refrigerant flows in both directions, thus requiring the indoor coil and outdoor coil be volumetrically balanced. In order to locate a list of rated HVAC system combinations, the AHRI Directory of Certified Systems should be utilized, or the manufacturer's data be obtained.

However, if, on a retrofit, the building owner determines it is not feasible to replace the indoor section (air handler and coil), some of the previously installed indoor sections may be modified to be compatible with R410A. Refer toAPP-APG011-EN or APP-APG012-EN. Please understand, a retrofitted system may not be listed in the AHRI directory. In addition, the manufacturer will not be able to provide any type of performance data. If the component has surpassed the OEM or extended warranty period, it is recommended to replace the indoor and outdoor unit with an AHRI listed system. If the customer requests an outdoor unit with a nominal rating above 13 SEER, the indoor and outdoor unit shall be replaced unless approved by Application Engineering. *System matches may be located in the Certified HVACR Equipment directories @ www.ahrinet.org*

Reusing existing lines and indoor equipment:

⚠ CAUTION

Wear required personal protective equipment to minimize the risk of oil and debris coming in contact with eyes and skin.

A. Reusing existing refrigeration line and indoor coils.
   1. Drain as much oil as possible from line set and / or indoor coil.
   2. Purging the line set and coil with dry nitrogen may be required to recover oil from the horizontal refrigerant piping and coil circuitry. The indoor coil may require removal in order to purge all existing oil from the coil.
   3. Oil must be captured and recycled or disposed of in accordance with national, state, and local standards. Such a standard is the EPA Clean Air Act. *Reference www.epa.gov*
   4. Conduct an acid test using the appropriate test kit for R-22 / Mineral oil systems.
      a) If the acid test shows negative, then proceed with installing the new OD unit.
      b) If the acid test indicates acid, then treat this system as a burnout.
         1. **If at all possible, replace the refrigerant lines.**
      c) If acid test returns positive and the lines are inaccessible or if there is a concern of debris in the lines, a flush agent may be used.
      d) If the previous system failed as a result of compressor burnout, it is recommended to replace the indoor coil.
      e) It is not necessary to measure the amount of residual mineral oil left in the system. Clearing the indoor coil and or refrigeration lines as detailed in this document is adequate. However, suction line riser traps may pose an issue with mineral oil and debris removal. If the existing vapor line includes suction riser traps, line replacement is recommended if the traps are unable to be eliminated.
      f) The new refrigerant flow control device should be installed after clearing the existing mineral oil from the system. Please reference APP-APG011-EN & APP-APG012-EN for the appropriate TXV / OD unit match.

CAUTION

There are multiple HVAC flush agents available to our industry. Prior to using any flush agent, please read and understand all directions printed by the manufacturer as there may be differences from one manufacturer to another. Flush agents should be used for line cleaning only. Do not flush an indoor or outdoor coil for re-use with Trane HVAC split systems. The coil may trap residue and promote premature compressor and flow control failure. If using a flush product in lines where POE oil will be the system lubricant, such as in R410A systems, it is recommended to use an HFC flush product. In all cases, the system must be evacuated below 300 microns in order to remove residual flush fluid that may contain unstable chlorinated hydrocarbon solvent. At compressor discharge temperatures this solvent will breakdown and form strong acids.

Section VI
Other Applicable Information

Figure 6

Thermal Bulb Location

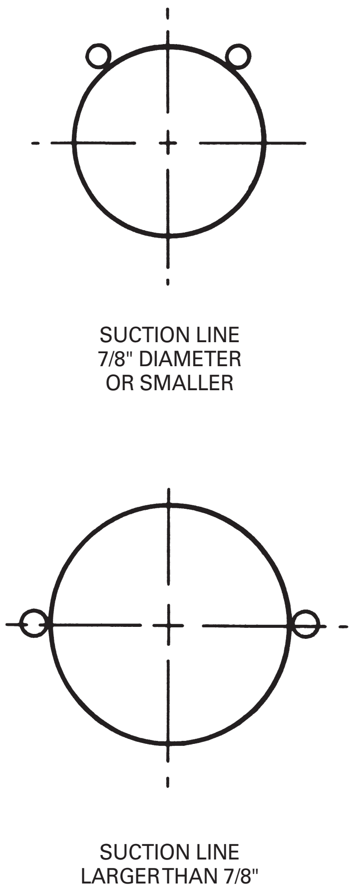


Figure 7

Typical Straight Cooling System
(Outdoor Unit Above Indoor Unit)

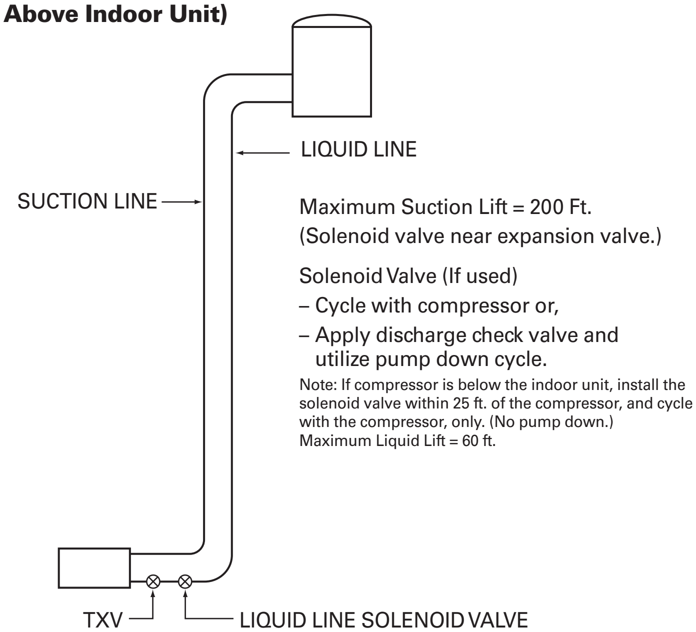


Figure 8

Tubing Hints

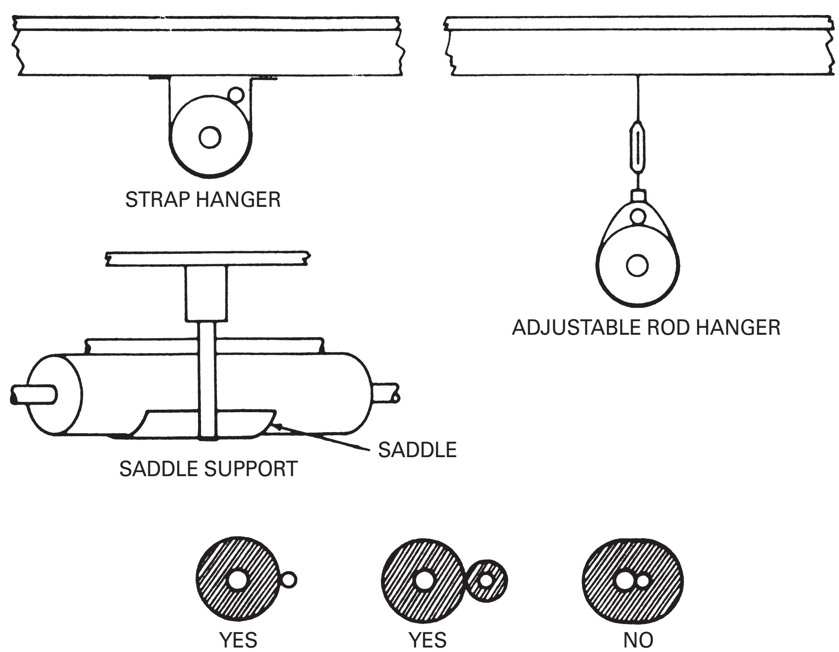


Figure 9

Basic Air Conditioning Formulas

TO DETERMINE	EXPRESSED AS	COOLING	HEATING and/or HUMIDIFYING
Total Airflow	CFM_T	1. $CFM_T = \frac{N_T\ V}{60\ \text{min./hr.}}$	1. $CFM_T = \frac{N_T\ V}{60\ \text{min./hr.}}$
Infiltration or Ventilation	CFM_o	2. $CFM_o = \frac{N_o\ V}{60\ \text{min./hr.}}$	2. $CFM_o = \frac{N_o\ V}{60\ \text{min./hr.}}$
Number of Air Changes Per Hour – Total	N_T	3. $N_T = \frac{CFM_T\ (60\ \text{min./hr.})}{V}$	3. $N_T = \frac{CFM_T\ (60\ \text{min./hr.})}{V}$
Number of Air Changes Per Hour – Outdoor Air	N_o	4. $N_o = \frac{CFM_o\ (60\ \text{min./hr.})}{V}$	4. $N_o = \frac{CFM_o\ (60\ \text{min./hr.})}{V}$
Total Heat (H_T)	Btuh	5. $H_T = CFM_T \times 4.5 \times (h_1 - h_2) = \text{Btuh}$	6. $H_T = CFM_T \times 4.5 \times (h_2 - h_1) = \text{Btuh}$
Sensible Heat (H_S)	Btuh	7. $H_S = CFM_T \times 1.08 \times (T_1 - T_2) = \text{Btuh}$	8. $H_S = CFM_T \times 1.08 \times (T_2 - T_1) = \text{Btuh}$
Latent Heat (H_L)	Btuh	9. $H_L = CFM_T \times .68 \times (W_1 - W_2) = \text{Btuh}$	10. $H_L = CFM_T \times .68 \times (W_2 - W_1) = \text{Btuh}$
Entering Air Temperature (T_1) (Mixed Air)	°F. D.B.	11. $T_1 = t_1 + \frac{CFM_o}{CFM_T} \times (t_2 - t_1) = \text{°F.D.B.}$ ① ① If duct heat gain is a factor, add to T_1: $\frac{\text{Duct Heat Gain (Btuh)}}{CFM_T \times 1.08}$	12. $T_1 = t_1 - \frac{CFM_o}{CFM_T} \times (t_1 - t_2) = \text{°F.D.B.}$ ② ② If duct heat loss is a factor, subtract from T_1: $\frac{\text{Duct Heat Loss (Btuh)}}{CFM_T \times 1.08}$
Leaving Air D.B. Temperature (T_2)	°F. D.B.	13. $T_2 = T_1 - \frac{H_S}{CFM_T \times 1.08} = \text{°F.D.B.}$	14. $T_2 = T_1 + \frac{H_S}{CFM_T \times 1.08} = \text{°F.D.B.}$
Required Airflow	CFM_T	15. $CFM_T = \frac{H_S\ \text{(total)}}{1.08 \times (T_1 - T_2)} = \text{CFM}$ OR $CFM_T = \frac{H_S\ \text{(internal)}③}{1.08 \times (t_1 - T_2)} = \text{CFM}$ ③ Sensible load of outside air not included	16. $CFM_T = \frac{H_S}{1.08 \times (T_2 - T_1)} = \text{CFM}$
Enthalpy – Leaving Air (h_2)	Btu/lb. dry air	17. $h_2 = h_1 - \frac{H_T}{CFM_T \times 4.5} = \text{Btu/lb. dry air}$	18. $h_2 = h_1 + \frac{H_T}{CFM_T \times 4.5} = \text{Btu/lb. dry air}$
Leaving Air W.B. Temperature	°F.W.B.	19. Refer to Enthalpy Table and read W.B. temperature corresponding to enthalpy of leaving air (h_2) (see #17).	20. Refer to Enthalpy Table and read W.B. temperature corresponding to enthalpy of leaving air (h_2) (see #18).
Heat Required to Evaporate Water Vapor Added to Ventilation Air	Btuh	21. $H_L = CFM_o \times .68\ (W_3 - W_o) = \text{Btuh}$	22. $H_L = CFM_o \times .68\ (W_3 - W_o) = \text{Btuh}$
Humidification Requirements	Lbs. water/hr.	23. $\left(\begin{matrix}\text{Make up}\\ \text{Moisture}\end{matrix}\right) = \frac{\text{Excess Latent Capacity of System} \times \text{\% Run Time}}{1060\ \text{Btu/lb.}} = \text{lbs./hr.}$ (Industrial Process Work)	24. $\left(\begin{matrix}\text{Make up}\\ \text{Moisture}\end{matrix}\right) = \frac{H_L\ \text{loss Btuh (see \#22)}}{1060\ \text{Btu/lb.}} = \text{lbs./hr.}$

LEGEND

CFM_T	=	Total airflow cubic feet/min.	
CFM_o	=	Outdoor air cubic feet/min.	
N_T	=	Total air changes per hour	
N_o	=	Outdoor air, air changes per hour	
V	=	Volume of space cubic feet	
H_T	=	Total heat Btuh	
H_S	=	Sensible heat Btuh	
H_L	=	Latent heat Btuh	
* h_1	=	Enthalpy or total heat of entering air	Btu/lb.
* h_2	=	Enthalpy or total heat of leaving air	Btu/lb.
T_1	=	Temperature of entering air	
T_2	=	Temperature of leaving air	
T_{adp}	=	Apparatus dewpoint	°F.D.B.
t_1	=	Indoor design temperature	°F.D.B.
t_2	=	Outdoor design temperature	°F.D.B.
W_1	=	Grains of water/lb. of dry air at entering condition	Grains/lb.
W_2	=	Grains of water/lb. of dry air at leaving condition	Grains/lb.
W_3	=	Grains of water/lb. of dry air at indoor design conditions	Grains/lb.
W_o	=	Grains of water/lb. of dry air at outdoor design conditions	Grains/lb.

DERIVATION OF AIR CONSTANTS

The air constants below apply specifically to standard air which is defined as dry air at 70°F and 14.7 P.S.I.A. (29.92 in. mercury column). They can, however, be used in most cooling calculations unless extremely precise results are desired.

4.5 (To convert CFM to lbs./hr.)

$4.5 = \frac{60\ \text{min./hr.}}{13.33}$ or $60 \times .075$

Where 13.33 is the specific volume of standard air (cu.ft./lb.) and .075 is the density (lbs./cu.ft.)

$1.08 = \frac{.24 \times 60}{13.33}$ or $.24 \times 4.5$

.24 BTU = specific heat of standard air (BTU/LB/°F)

$.68 = \frac{60}{13.33} \times \frac{1060}{7000}$ or $4.5 \times \frac{1060}{7000}$

Where: 1060 = Average Latent Heat of water vapor (BTU/LB.).

7000 = Grains per lb.

* See Enthalpy of air (Total Heat Content of Air) Table for exact values.

Literature Order Number		
File No.	Pub. No. SS-APG006-EN	2/11
Supersedes	Pub. No. 32-3009-03	7/08
Stocking Location		P.I.