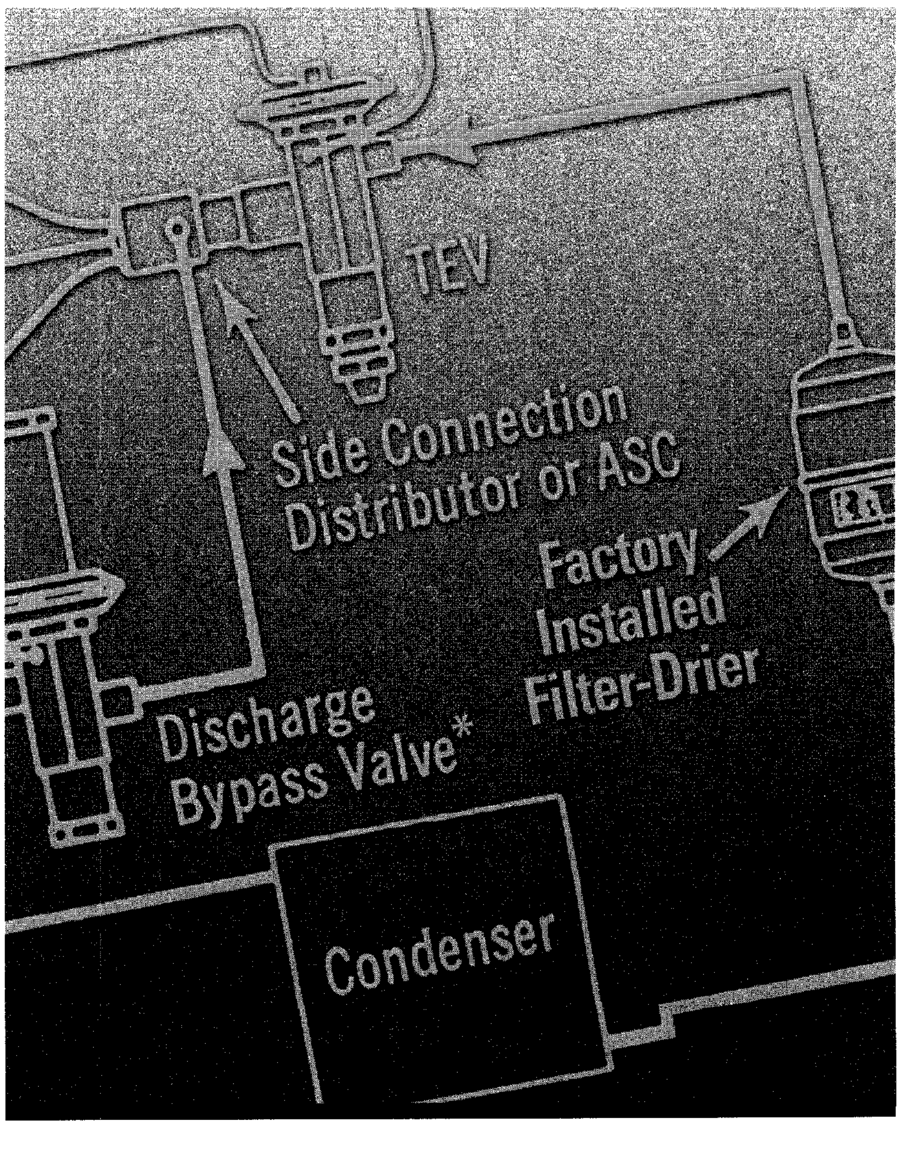
Application Guide

Refrigerant Piping Systems

I – Refrigerant Piping
II – High Rise Applications
III – Hot Gas Bypass



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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:

- Dave Donnelly
- Chuck Erlandson
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- Terry Ryan
- Jim Sharp
- Greg Walters
- Richard Welguisz

Tables and Charts

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Introduction

The purpose of this manual is to assist the user 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 and Chapter III covers Hot Gas Bypass (Capacity Modulation).

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 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 35 PSI for R-22 or 50 PSI for R-410A. Liquid line pressure drop calculations must include friction loss, liquid lifts and refrigerant accessories (solenoid valves, etc.).

There is no penalty for pressure drop in a liquid line, provided it does not exceed 35 PSI (50 PSI with R-410A). The smallest diameter that meets this 35 PSI criteria results in better system reliability (fewer pounds of refrigerant to cause potential damage to the compressor). Note that the 35 PSI allowance is based on 10 degrees of subcooling (no liquid receiver.) Both Trane and American Standard U.P.G. units meet this criteria.

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 "B" 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.

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. Dedicated hot gas lines, such as hot gas lines for hot gas bypass, are covered in Chapter III of this manual.

The new, Windows® based piping program has been released, for both Trane and American Standard. Pub. No. 32-3312-01 covers Trane, American Standard and 50 Hz equipment.

The new 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 knowledgeable expert.

Information provided by the program includes:

- Liquid and suction line sizes
- Liquid and suction line pressure drops
- Net system capacity
- 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.
- Excellent print-outs

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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 and no traps are recommended.

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 in a suction line is a pressure drop equivalent to 2°F (approx. 3 PSI with R-22 in the air conditioning range of evaporating temperatures or approx. 5 PSI for R-410A). 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. (See Chapter III in this manual for information on hot gas lines for hot gas bypass.)

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 10 tons is 35 PSI for R-22 systems and 50 PSI for R-410A systems.

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 the user read all of Chapter I in order to better understand the Tables, Charts, etc.

See the Index 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.

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CHAPTER 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 such as the expansion valve or FCCV Accutron™. 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 gas 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 "C," page 10, 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-22 the static pressure drop will be 0.50 PSI per foot of lift (.43 PSI per foot with R-410A).

As an example, consider an air-cooled R-22 system with 95°F air entering the condenser, the condensing temperature is 120°F (approx. 260 PSI).

After being subcooled in the condenser, the liquid R-22 leaves the condenser at 110°F. Assuming the pressure at the condenser outlet is the same as the condensing pressure of approx. 260 PSIG, the liquid R-22 has been subcooled 10°F. The saturation pressure for R-22 at 110°F is approx. 226 PSIG Substracting 226 PSIG from the 260 PSIG condensing pressure, gives us a difference of 34 PSI.

While this pressure difference is 1 PSI less than the 35 PSI obtained with the 125°F example used in earlier versions of this manual, we will continue to use 35 PSI as the maximum liquid line pressure drop. (A drop in liquid line temperature of 1/3°F by natural cooling, will provide the needed 1 PSI.)

Note that the above mentioned temperatures of 120°F and 110°F represent pressures of 418 PSIG and 365 PSIG with R-410A, a difference of 53 PSI. For the present, we will limit R-410A liquid line pressure drop 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 Product Manual 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.

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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 "A" 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 7.

Note that equivalent lengths are used when calculating pressure drops. Actual (linear) lengths are used when calculating pounds of R-22 in a line set. (An elbow contains about the same amount of R-22 as does the same length of straight tubing.)

Table "C," page 10, 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 (.5 PSI per foot of lift for R-22 systems, .43 PSI per foot with 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-22 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. See page 11 for tubing limits.

Since refrigerant oil is miscible with liquid R-22, 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:

Note: A worksheet for Manual Calculations is provided on page 31.

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Table "A"

Liquid Line Selection Table For R-22 Systems

Maximum Allowable Liquid Line Pressure Drop	=	35 PSI
Subtract .5 PSI for each foot of Liquid Lift (if any)		
Do Not Exceed this value when selecting Liquid Line.		

Pressure Drop (PSI)

		Total Equivalent Length											
Tube O.D.	**Rated BTUH**	**20'**	**40'**	**60'**	**80'**	**100'**	**120'**	**140'**	**160'**	**180'**	**200'**	**220'**	**240'**
1/4"	15000	4.3	8.7	13.0	17.4	21.7	26.0	30.4	34.7	—	—	—	—
	18000	6.0	12.0	18.1	24.1	30.1	—	—	—	—	—	—	—
	24000	10.2	20.3	30.5	—	—	—	—	—	—	—	—	—
5/16"	15000	1.1	2.3	3.4	4.6	5.7	6.8	8.0	9.1	10.3	11.4	12.5	13.7
	18000	1.6	3.2	4.7	6.3	7.9	9.5	11.1	12.6	14.2	15.8	17.4	19.0
	24000	2.7	5.3	8.0	10.6	13.3	16.0	18.6	21.3	23.9	26.6	29.3	31.9
	30000	4.0	8.0	11.9	15.9	19.9	23.9	27.9	31.8	—	—	—	—
	36000	5.5	11.1	16.6	22.2	27.7	33.2	—	—	—	—	—	—
	42000	7.3	14.6	22.0	29.3	—	—	—	—	—	—	—	—
3/8"	18000	.6	1.1	1.7	2.2	2.8	3.4	3.9	4.5	5.0	5.6	6.2	6.7
	24000	.9	1.9	2.8	3.8	4.7	5.6	6.6	7.5	8.5	9.4	10.3	11.3
	30000	1.4	2.8	4.2	5.6	7.0	8.4	9.8	11.2	12.6	14.0	15.4	16.8
	36000	1.9	3.9	5.8	7.8	9.7	11.6	13.6	15.5	17.5	19.4	21.3	23.3
	42000	2.6	5.1	7.7	10.2	12.8	15.4	17.9	20.5	23.0	25.6	28.2	30.7
	48000	3.3	6.5	9.8	13.0	16.3	19.6	22.8	26.1	29.3	32.6	—	—
	60000	4.9	9.8	14.7	19.6	24.5	29.4	34.3	—	—	—	—	—
	72000	6.8	13.7	20.5	27.4	34.2	—	—	—	—	—	—	—
1/2"	36000	.4	.8	1.2	1.6	2.0	2.4	2.8	3.2	3.6	4.0	4.4	4.8
	42000	.5	1.0	1.6	2.1	2.6	3.1	3.6	4.2	4.7	5.2	5.7	6.2
	48000	.7	1.3	2.0	2.6	3.3	4.0	4.6	5.3	5.9	6.6	7.3	7.9
	60000	1.0	2.0	3.0	4.0	5.0	6.0	7.0	8.0	9.0	10.0	11.0	12.0
	72000	1.4	2.8	4.1	5.5	6.9	8.3	9.7	11.0	12.4	13.8	15.2	16.6
	90000	2.1	4.2	6.2	8.3	10.4	12.5	14.6	16.6	18.7	20.8	22.9	25.0
	120000	3.5	7.0	10.5	14.0	17.5	21.0	24.5	28.8	31.5	35.0	—	—
5/8"	90000	.6	1.2	1.9	2.5	3.1	3.7	4.3	5.0	5.6	6.2	6.8	7.4
	120000	1.1	2.1	3.2	4.2	5.3	6.4	7.4	8.5	9.5	10.6	11.7	12.7

Note 1: A blank space indicates a pressure drop of more than 35 PSI.
Note 2: Other existing sources of pressure drop such as solenoid valves, etc. must be accounted for.
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 35 PSI or less results in the most reliable system (fewer pounds of R-22).

Selecting A Liquid Line, Using Table "A"

Step #1 Subtract .5 PSI per foot of liquid lift (if any) from the 35 PSI liquid line pressure drop which can be tolerated by U.P.G. systems (R-22).

Step #2 Calculate the equivalent length of the liquid line (linear length plus an allowance for elbows, etc.). See Table "C," page 10, for equivalent lengths of elbows, etc.

Step #3 Select a liquid line from Table "A" which can handle the system BTUH at the calculated equivalent length within the available pressure drop found in Step #1. (Your first try would normally be the rated liquid line size for the system.)

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 .5 = 10 PSI pressure drop due to liquid lift. 35 PSI (available) minus 10 PSI = 25 PSI available for friction loss.

Step #2 68 + (4 x 3.2) = 80.8 eq. ft. (See Table "C," page 10, for equivalent lengths.)

Step #3 Referring to Table "A" we find:
A – 42000 BTUH with 5/16" O.D. tubing @ 80 eq. ft. = 29.3 PSI (too high)
B – 42000 BTUH with 3/8" O.D. tubing @ 80 eq. ft. = 10.2 PSI (O.K.)

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Suction Lines for Split Cooling and Heat Pump Systems

Suction lines must return refrigerant vapor and oil from the evaporator to the compressor during operation of the system, but should not allow oil or liquid refrigerant to be returned as slugs at any time, because of the danger of broken compressor valves, oil dilution, etc.

Never attempt to operate two hermetic compressors with a common suction line. It is impossible to return oil to each compressor at precisely the same rate as it is being pumped. As a result, one compressor will eventually run low on oil in its sump and proper lubrication is no longer possible. Each hermetic compressor must have its own separate refrigerant system.

(Our two compressor systems may appear to violate this rule, but they do not. These compressors are provided with special piping which allows oil levels in the two compressors to equalize.)

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 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 Tubing Hints on page 13.)

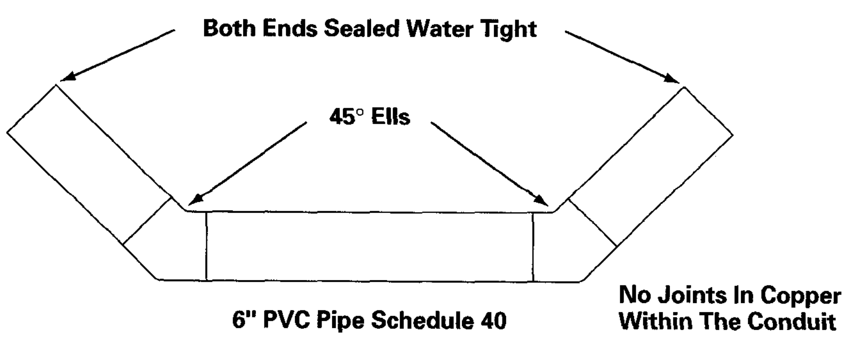
Suction lines must be insulated to prevent condensation and vapor sealed on the outside to prevent a build-up of moisture in the insulation.

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.)

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.

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About Suction Lines and Pressure Drops

ASHRAE recommends that suction line pressure drop be limited to a pressure corresponding to 2°F (approx. 3.0 PSI with R-22). This is usually not a problem with line sets of 100 eq. ft. or less.

A quick look at the pressure drop per 100 ft. listed in Table "B" reveals that using the largest allowable suction line diameter for each tonnage results in a pressure drop of less than 3 PSI per 100 eq. ft. in all cases, except the 1 ton system (3.3 PSI).

Obviously, if line sets exceed 100 eq. ft. significantly, there will be cases where the suction line pressure drop exceeds 3.0 PSI. (This is a price we must pay for long line sets.)

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 "B," page 10, for the system tonnage. To do so would result in refrigerant velocities too low to ensure oil return.

The net capacities indicated in Table "B" for the various equivalent lengths show that there is approx. 1% loss in capacity for each 1.0 PSI of pressure drop. (Efficiency losses are approx. .5% per 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 show in Table "B" are not required in order to select a suction line. They are provided for your information only. One example of their use might be to evaluate an existing system. For instance, careful measurements of an existing 2-1/2 ton system, installed with a 5/16" liquid line and 5/8" suction reveal the following: 110 linear ft., 8 short radius elbows. The equivalent length, 110 + (8 x 5.7*) = 155.6 ft. (*from Table "C" page 10). 155.6 ft. x 12.7/100 = approx. 19.8 PSI suction line pressure drop (more than six times the ASHRAE recommendation).

The approx. 20% loss in capacity tells us that our 2-1/2 ton system is delivering 2 tons. This 20% loss in capacity, together with a 10% loss in efficiency, makes a very strong case for replacing the line set with a properly sized line set.

Assuming that the 5/8" O.D. suction line is to be replaced with a 7/8" O.D. suction line, and the installer was able to reduce the number of elbows to six (long radius) elbows: the equivalent length = 110 + (6 x 5.3) or 141.8 ft., 2.0 PSI/100 ft. (from Table "B") x 141.8/100 = approx. 2.8 PSI suction line pressure drop.

The 30000 BTUH system will now deliver approx. 29500 BTUH (140 eq. ft. value). The loss in efficiency is now less than 1%, which will be reflected in the owner's electric bill (favorably).

(The installer, when replacing the undersized refrigerant lines, should convert the indoor unit to expansion valve flow control and make sure that the compressor is equipped with sump heat, in order to bring the system up to current requirements.)

The Dilemma:

Suppose that a service man is attempting to check the refrigerant charge in the existing 2-1/2 ton system, with 5/16" – 5/8" O.D. refrigerant lines, mentioned above (F.C.C.V. flow control). He is using the superheat method and, based on the existing indoor and outdoor conditions, has determined that 10° superheat, at the outdoor unit is required.

His compound gauge indicates 63 PSIG or 36° evaporating temperature. Adding 10° to the 36° tells him that he should try for a 46° suction line temperature, right? Wrong in this case!

His actual evaporating pressure is 63 PSIG plus 19.8 PSI pressure drop, or approx. 83 PSIG (49° plus.) The desired suction line temperature = 59°. Attempting to charge to the 46° suction line temperature would undoubtedly result in a severe overcharge. The extreme pressure drop not only resulted in very substantial losses in capacity and efficiency, but could easily be the cause of a severe overcharge (and possibly a compressor failure).

Always select one of the suction line sizes listed in Table "B" for the nominal tonnage of your system. Oil return will be assured with any of the listed sizes. The lowest possible capacity losses consistent with adequate oil return are afforded 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 240 ft.

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Table "B"

Allowable Suction Line Diameters and Net Capacities

Nom. Tons	Tube Size (In.)	Nominal Capacity (25 Ft.)	P.D. Per 100 Ft.	Net Capacity For Equivalent Length 40	60	80	100	120	140	160	180	200	220	240
1.0	1/2	14685	11.7	14425	14080	13735	13390	13045	—	—	—	—	—	—
	5/8	15000	3.3	14925	14825	14725	14625	14525	14425	14325	14225	14125	14025	13925
1.5	5/8	18000	4.7	17875	17705	17535	17365	17195	17025	16855	16685	16515	16345	16175
	3/4	19305	1.8	19255	19185	19115	19045	18975	18905	18835	18765	18695	18625	18555
	5/8	23695	8.1	23405	23020	22635	22250	21865	21480	21095	20710	—	—	—
2.0	3/4	24000	3.0	23890	23745	23600	23455	23310	23165	23020	22875	22730	22585	22440
	7/8	24100	1.3	24055	23990	23925	23860	23795	23730	23665	23600	23535	23470	23405
	5/8	29370	12.7	28810	28665	27320	26575	25830	—	—	—	—	—	—
2.5	3/4	30000	4.6	29795	29520	29245	28970	28695	28420	28145	27870	27595	27320	27045
	7/8	30195	2.0	30105	29985	29865	29745	29625	29505	29385	29265	29145	29025	28905
3.0	3/4	35670	6.5	35320	34855	34390	33925	33460	32995	32530	32065	31600	31135	30670
	7/8	36000	2.8	35850	35650	35456	35250	35050	34850	34650	34450	34250	34050	33850
	3/4	41475	8.8	40930	40200	39470	38740	38010	37280	36550	35820	—	—	—
3.5	7/8	42000	3.8	41760	41440	41120	40800	40480	40160	39840	39520	39200	38880	38560
	1-1/8	42295	1.0	42230	42145	42060	41975	41890	41805	41720	41635	41550	41465	41380
4.0	7/8	47570	4.9	47220	46755	46290	45825	45360	44895	44430	43965	43500	43035	42570
	1-1/8	48000	1.3	47905	47780	47655	47530	47405	47280	47155	47030	46905	46780	46655
	7/8	59175	7.5	58510	57620	56730	55840	54950	54060	53170	52280	51390	—	—
5.0	1-1/8	60000	2.0	59820	59580	59340	59100	58860	58620	58380	58140	57900	57660	57420
	1-3/8	60195	.7	60130	60045	59960	59875	59790	59705	59620	59535	59450	59365	59280
6.0	1-1/8	72000	2.8	71700	71295	70890	70485	70080	69675	69270	68865	68460	68055	67650
	1-3/8	72325	1.0	72215	72070	71925	71780	71635	71490	71345	71200	71055	70910	70765
	1-1/8	89390	4.2	88825	88075	87325	86575	85825	85075	84325	83575	82825	82075	81325
7.5	1-3/8	90000	1.5	89795	89525	89255	88985	88715	88445	88175	87905	87635	87365	87095
	1-5/8	90205	.6	90125	90015	89905	89795	89685	89575	89465	89355	89245	89135	89025
	1-1/8	118560	7.4	117245	115490	113785	111980	110225	108470	106715	104960	103205	101450	—
10.0	1-3/8	120000	2.6	119230	118905	118280	117655	117030	116405	115780	115135	114530	113905	113280
	1-5/8	120450	1.1	120250	119985	119720	119455	119190	118925	118660	118395	118130	117865	117600

Note 1: Shaded values = more than 10% capacity loss.
Note 2: Blank space = more than 15% capacity loss.

Suction Line Selection Example

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

Since the equivalent length will probably fall between 140 and 160 ft., common sense tells us that we should select the largest diameter tube listed for a 4 ton system in Table "B," which is 1-1/8" O.D. (This happens to be the rated suction line size for a 4 ton system.) The equivalent length, 132 + (8 x 1.9) = 147.2 ft. The net capacity will be between 47280 and 47155 BTUH (47235 BTUH, by interpolation). This translates to a capacity loss of approx. 1.6%. (good)

Question

Would a 7/8" 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 approx. 2.5% is not a problem, the 7/8" O.D. suction line is O.K. in this case.

$$\left(\frac{48000 - 46755}{48000} = \text{approx. } 2.5\%\right)$$

Table "C"

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.

Piping Limits

1. Compressor Protection

A. Suction line accumulators are no longer required (on 1 through 10 ton systems).

B. Protect reciprocating compressors as follows:
- Up to 80 linear feet of rated tube sizes: OK as shipped.
- Over 80 linear feet, or oversized lines: Apply sump heat① and TXV indoor metering device.①③

C. Protect Scroll compressors as follows:
- Up to 12 lbs. R-22 system charge②: OK as shipped.
- 10.5 lbs. for R-410A.
- Over 12 lbs. R-22 system charge②: Apply sump heat and TXV indoor metering device.①

D. Liquid line solenoid valves (straight cooling systems **only**):*
Cycle solenoid valve with compressor (no pump down).
- If compressor is **above** the indoor unit, locate the liquid line solenoid valve near the indoor unit.
- If compressor is **below** the indoor unit, locate the liquid line solenoid valve within 25 ft. of compressor.

Note: If pump down is desired, a discharge check valve **must** be installed in the discharge line. (Locate the liquid line solenoid valve near the indoor unit.)
* Liquid line solenoid valves are **not required** with TXV systems.

2. Piping Limitations (Heat Pumps and Straight Cooling)

A. Suction line diameter must be one of those listed in Table "B," based on system tonnage.
B. Maximum linear length = 200 ft.
C. Maximum linear liquid lift = 60 ft. (except high rise systems with liquid subcooler).
D. Maximum linear suction lift = 200 ft.
E. If it is necessary to exceed the above limits, contact Application Engineering.
F. Note special limitations for two compressor systems and R-410A systems below.

3. High Rise Systems

A. The high rise system, with liquid subcooler, is limited to heat pump systems only, with the outdoor unit above the indoor unit.
B. The indoor unit on high rise applications must utilize a TXV metering device.

4. 3, 4 and 5 Ton Manifolded Compressor Systems

A. Maximum linear length = 80 ft.
B. Maximum lift (suction or liquid) = 25 ft.

Tube Sizes

3 Ton = 3/8" and 7/8"
4 and 5 Ton = 3/8" and 1-1/8"

5. R-410A Systems

A. Maximum linear length = 200 ft.
B. Maximum linear liquid lift = 60 ft.
C. Maximum linear suction lift = 200 ft.

6. Traps

A. Traps are not recommended.

Notes:
① If not factory furnished.
② System charge = nameplate charge, plus tubing allowance (See page 12 for Tubing Allowance).
③ If a non-bleed TXV is applied to a single phase Reciprocating compressor system, a hard start kit will be required (not required with Scroll compressor systems).
④ Pub. No. 32-3312-01 (Windows® based) computer software contains complete piping data, including high rise systems, R-410A, pressure drops, net capacity, system charge, tube sizes, etc.

Table "E-1"

Refrigerant Piping and Accessories for Trane and American Standard Reciprocating and Scroll Compressors

B Tubing Allowance (Lbs. R-22)									
Tube Size (Inches)	40'	60'	80'	100'	120'	140'	160'	180'	200'
1/4 – 5/8	0.4	0.7	1.1	1.4	1.7	2.1	2.4	2.7	3.1
5/16 – 3/4	0.7	1.3	1.8	2.4	3.0	3.5	4.1	4.7	5.2
5/16 – 7/8	0.7	1.3	1.9	2.5	3.1	3.7	4.3	4.9	5.5
5/16 – 1-1/8	0.8	1.5	2.2	2.9	3.5	4.2	4.9	5.6	6.2
3/8 – 3/4	1.1	1.9	2.8	3.6	4.5	5.3	6.2	7.0	7.9
3/8 – 7/8	1.1	2.0	2.9	3.8	4.6	5.5	6.4	7.3	8.2
3/8 – 1-1/8	1.2	2.2	3.1	4.1	5.1	6.0	7.0	7.9	8.9
3/8 – 1-3/8	1.3	2.3	3.4	4.3	5.4	6.4	7.4	8.4	9.5
1/2 – 1-1/8	2.0	3.6	5.2	6.9	8.5	10.1	11.7	13.3	14.9
1/2 – 1-3/8	2.1	3.9	5.6	7.3	9.0	10.7	12.4	14.1	15.8
1/2 – 1-5/8	2.3	4.1	6.0	7.8	9.6	11.5	13.3	15.1	17.0
5/8 – 1-3/8	3.1	5.6	8.1	10.6	13.1	15.6	18.1	20.6	23.1
5/8 – 1-5/8	3.4	6.1	8.8	11.5	14.3	17.0	19.7	22.4	25.1

Conversion	
Ounces	Pounds
1.00	0.0625
2.00	0.1250
3.00	0.1875
4.00	0.2500
5.00	0.3125
6.00	0.3750
7.00	0.4375
8.00	0.5000
9.00	0.5625
10.00	0.6250
11.00	0.6875
12.00	0.7500
13.00	0.8125
14.00	0.8750
15.00	0.9375
16.00	1.0000

Reciprocating Compressors

Up to 80 linear ft. of rated tubing size	Over 80 linear ft. or oversized tubing
O.K. as shipped	Required: Sump Heat②, Indoor TXV②③

Scroll Compressors

C Up to 12 lbs.* system charge①	C Over 12 lbs.* system charge①
O.K. as shipped	Required: Sump Heat②, Indoor TXV②③

① Nameplate charge (indoor unit, outdoor unit + 15 ft. tubing) plus tubing allowance (over 15 ft.).
② Add, if not factory furnished.
③ Bleed or Non-Bleed.
* 10.5 lbs. for R-410A systems

Note: Failure to add the required components (systems over 12 lbs.) may result in noise complaints and/or compressor damage. Use actual measured length. The 15 ft. is accounted for.

A	+	B	=	C
NAMEPLATE CHARGE (15 Ft. Tubing Included)		TUBING ALLOWANCE		SYSTEM CHARGE*

*Always check refrigerant charge when starting system, as recommended in installer's guide.

Note 1: Refer to unit nameplate, or product specifications page in Product Data Catalog or Service Facts for nameplate charge.

Note 2: Two compressor systems are limited to 80 linear feet and 25 feet linear lifts, suction or liquid.

Note 3: If nameplate charge is given in pounds and ounces, refer to conversion table above.

Note: Hard Start Kits are required for single phase systems with non-bleed TXV. (Reciprocating Compressors Only.)

Table "E-2"

Thermal Expansion Valve for Cooling Units (TXV)

Unit Tonnage	Bleed Type Valve	Non-Bleed Type Valve
1 – 1-1/2	TAYTXVA0B5C	TAYTXVA0B3C
2 – 2-1/2	TAYTXVA0C5C	TAYTXVA0C3C
3 – 3-1/2	TAYTXVA0E5C	TAYTXVA0E3C
4	TAYTXVA0G5C	TAYTXVA0G3C
5 – 6	TAYTXVA0H5C	TAYTXVA0H3C

Note: TXV's are Brazed Type Connections.

Table "E-2"

Thermal Expansion Valve for Heat Pump Units (TXV)

Unit Tonnage	Non-Bleed Type Valve
1 – 1-1/2	TAYTXVH0B3C
2 – 2-1/2	TAYTXVH0C3C
3 – 3-1/2	TAYTXVH0E3C
4	TAYTXVH0G3C
5 – 6	TAYTXVH0H3C

Note: TXV's are Brazed Type Connections.

Sump Heater Kit

Reciprocating Compressors

BAYCCHT003AA
One Size Fits All

Sump Heater Kits for Scroll Compressors

Thermal jackets are no longer required.

BAYCCHT200A	BAYCCHT201A
Used with compressors SSR or SPR 024 through 045 and 047	Used with compressors SSR or SPR 048 through 077 and 046

Refer to optional equipment listing for heater part number or refer to product specifications for compressor type and select from above table.

Application Requirements

Systems within the limits indicated above, or systems beyond those limits which are equipped with compressor sump heat and indoor TXV refrigerant control, are O.K. for:

Linear Lengths up to 200 ft.

Liquid lifts up to 60 linear ft.

Suction lifts up to 200 linear ft.

This includes R-410A systems (recommended line sizes only), but does not include two compressor systems which are limited to 80 linear ft. and to lifts (liquid or suction) of up to 25 linear ft.

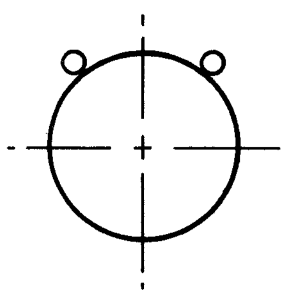
Suction line accumulator is not required on 1 through 10 tons.

Do not apply pump down cycle, unless discharge check valve has been applied. (See page 11.)

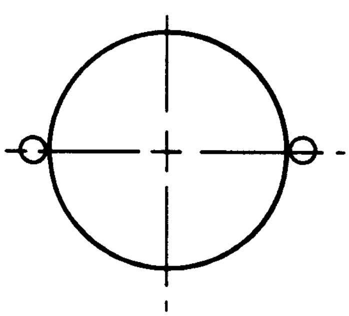
Application Guide

Figure 3

Thermal Bulb Location



SUCTION LINE
7/8" DIAMETER
OR SMALLER



SUCTION LINE
LARGER THAN 7/8"

Figure 2

Typical Straight Cooling System
(Outdoor Unit Above Indoor Unit)

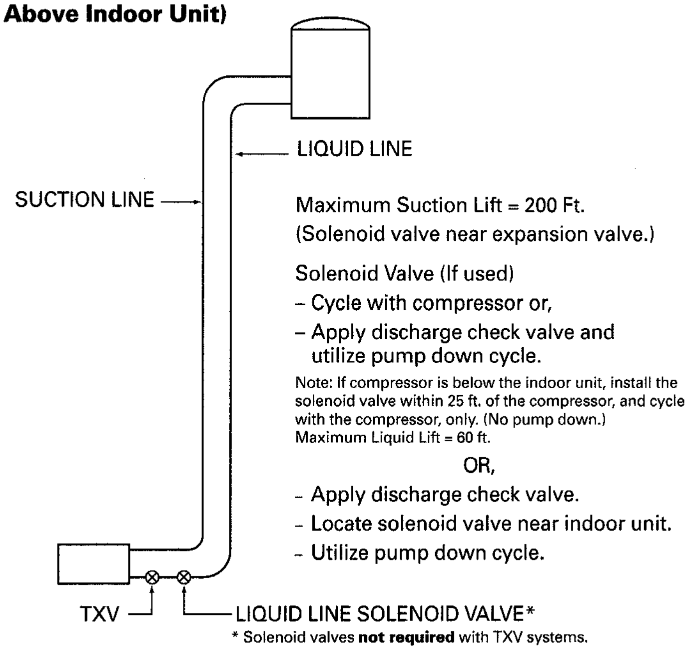


Figure 4

Tubing Hints

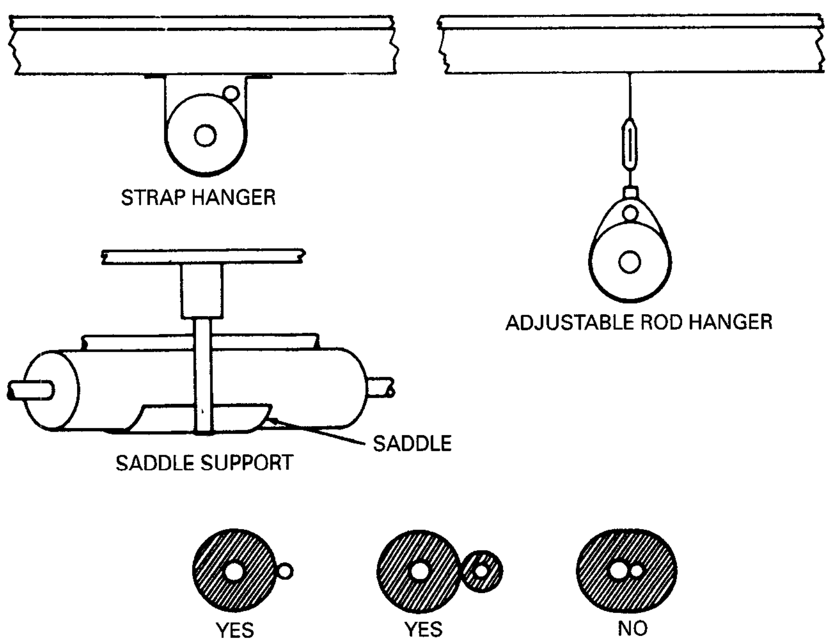


Figure 5

Basic Air Conditioning Formulas

TO DETERMINE	EXPRESSED AS	COOLING	HEATING and/or HUMIDIFYING
Total Airflow	CFM_T	1. $CFM_T = \dfrac{N_T\ V}{60\ \text{min./hr.}}$	1. $CFM_T = \dfrac{N_T\ V}{60\ \text{min./hr.}}$
Infiltration or Ventilation	CFM_o	2. $CFM_o = \dfrac{N_o\ V}{60\ \text{min./hr.}}$	2. $CFM_o = \dfrac{N_o\ V}{60\ \text{min./hr.}}$
Number of Air Changes Per Hour – Total	N_T	3. $N_T = \dfrac{CFM_T\ (60\ \text{min./hr.})}{V}$	3. $N_T = \dfrac{CFM_T\ (60\ \text{min./hr.})}{V}$
Number of Air Changes Per Hour – Outdoor Air	N_o	4. $N_o = \dfrac{CFM_o\ (60\ \text{min./hr.})}{V}$	4. $N_o = \dfrac{CFM_o\ (60\ \text{min./hr.})}{V}$
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. $T_1 = t_1 + \dfrac{CFM_o}{CFM_T} \times (t_2 - t_1) = \text{°F.D.B.}$ ① ① If duct heat gain is a factor, add to T_1: $\dfrac{\text{Duct Heat Gain (Btuh)}}{CFM_T \times 1.08}$	12. $T_1 = t_1 - \dfrac{CFM_o}{CFM_T} \times (t_1 - t_2) = \text{°F.D.B.}$ ② ② If duct heat loss is a factor, subtract from T_1: $\dfrac{\text{Duct Heat Loss (Btuh)}}{CFM_T \times 1.08}$
Leaving Air D.B. Temperature (T_2)	°F. D.B.	13. $T_2 = T_1 - \dfrac{H_S}{CFM_T \times 1.08} = \text{°F.D.B.}$	14. $T_2 = T_1 + \dfrac{H_S}{CFM_T \times 1.08} = \text{°F.D.B.}$
Required Airflow	CFM_T	15. $CFM_T = \dfrac{H_S\ (\text{total})}{1.08 \times (T_1 - T_2)} = \text{CFM}$ OR $CFM_T = \dfrac{H_S\ (\text{internal})③}{1.08 \times (t_1 - T_2)} = \text{CFM}$ ③ Sensible load of outside air not included	16. $CFM_T = \dfrac{H_S}{1.08 \times (T_2 - T_1)} = \text{CFM}$
Enthalpy – Leaving Air (h_2)	Btu/lb. dry air	17. $h_2 = h_1 - \dfrac{H_T}{CFM_T \times 4.5} = \text{Btu/lb. dry air}$	18. $h_2 = h_1 + \dfrac{H_T}{CFM_T \times 4.5} = \text{Btu/lb. dry air}$
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. $\left(\begin{array}{c}\text{Make up}\\ \text{Moisture}\end{array}\right) = \dfrac{\text{Excess Latent Capacity of System} \times \text{\% Run Time}}{1060\ \text{Btu/lb.}} = \text{lbs./hr.}$ (Industrial Process Work)	24. $\left(\begin{array}{c}\text{Make up}\\ \text{Moisture}\end{array}\right) = \dfrac{H_L\ \text{loss Btuh (see \#22)}}{1060\ \text{Btu/lb.}} = \text{lbs./hr.}$

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.)

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

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

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

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

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

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.

Application Guide

R-410A Refrigerant

R-410A is a near-azeotropic mixture of R-32 and R-125 refrigerants. Some separation of the two components can occur in the vapor phase (not enough to cause a significant change in the composition of the refrigerant with a refrigerant leak). However, it is recommended that charging be done in the liquid phase. When adding liquid refrigerant into the low side of the system, a charge metering device is recommended (WATSCO CH200, or equivalent). Allow ample time when adding refrigerant, for the system to balance out, to avoid having to recover refrigerant.

R-410A cylinders are pink in color and dispense liquid when in the upright position. (This may change.)

Gauges, hoses, recovery cylinders, and recovery machines must handle the higher pressures associated with R-410A. (See pressure/temperature chart.) Note that 45° corresponds to 129.7 PSIG, and 115° corresponds to 390.7 PSIG (compared to 76.0 PSIG and 242.7 PSIG for R-22).

R-410A has practically no temperature "Glide." (The temperature remains practically constant when going from 100% liquid to a saturated vapor at a given pressure.)

Existing Halide leak detectors do not work with R-410A. Existing acid test kits do not work with R-410A. (New kits are being developed.) Note that although R-410A does not deplete the ozone layer, all refrigerants must be recovered.

Do not expose R-410A cylinders to temperatures over 125°F.

R-410A systems use a POE oil, which is not compatible with the oils used in R-22 systems. If existing refrigerant lines are to be used with and R-410A system (assuming that the line sizes are acceptable), they must be thoroughly blown out with dry nitrogen to remove the old oil. Blow vertical sections from top to bottom.

POE oils absorb moisture very quickly. Keep container tightly closed, whenever possible, and expose the system to the atmosphere as little as possible. POE oils can also damage a roof, if spilled.

Vacuum pumps can not remove all of the moisture from POE oils. Change the liquid line drier anytime the system is opened to the atmosphere.

Suction line dryers are to be left in the system for no more than 72 hours. Use only liquid and suction line dryers approved for R-410A.

Since all current R-410A systems are expansion valve systems, the refrigerant charge is to be checked by the subcooling method.

Maximum liquid line pressure drop with R-410A systems is 50 PSI (10° subcooling). Recommended suction line pressure drop (2°F) is 4.8 PSI (Round up to 5.0).

At this time, only matched systems are permitted with R-410A.

R-410A boils at –62.9° at atmospheric pressure, so be wary of frostbite!

Figure 6

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

Table "G"

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
1/2" - 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.

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Table "A-R (R-410A)"

Liquid Line Selection Table For R-410A Systems

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 O.D.	Rated BTUH	Pressure Drop (PSI) For Total Equivalent Length											
		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	—	—	—	—	—
5/16"	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
	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	—	—	—	—	—	—
3/8"	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
	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	—	—	—
1/2"	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
	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
	72000	1.4	2.9	4.3	5.8	7.2	8.6	10.1	11.5	13.0	14.4	15.8	17.3
	90000	2.2	4.3	6.5	8.6	10.8	13.0	15.1	17.3	19.4	21.6	23.8	25.9
	120000	3.7	7.4	11.0	14.7	18.4	22.1	25.8	29.4	33.1	36.8	40.5	44.2
5/8"	72000	.4	.9	1.3	1.8	2.2	2.6	3.1	3.5	4.0	4.4	4.8	5.3
	90000	.7	1.3	2.0	2.6	3.3	4.0	4.6	5.3	5.9	6.6	7.3	7.9
	120000	1.1	2.2	3.3	4.4	5.5	6.6	7.7	8.8	9.9	11.0	12.1	13.2

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).

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 "C," page 10, for equivalent lengths.)

Step #3 Referring to Table A-R, 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.)

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Table "B-R (R-410A Refrigerant)"

Allowable Suction Line Diameters and BTUH Loss (R-410A)

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
	7/8	.9	20	45	70	95	125	150	175	200	225	255	280
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
	1-1/8	.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
6.0	3/4	16.5	1070	2495	3920	5345	6770	8195	9625	—	—	—	—
	7/8	7.0	455	1060	1665	2270	2875	3480	4080	4685	5290	5895	6500
	1-1/8	1.8	115	270	430	585	740	895	1050	1205	1360	1515	1670
7.5	7/8	10.8	875	2040	3210	4375	5540	6705	7875	9040	10203	11370	12540
	1-1/8	2.8	225	530	830	1135	1435	1740	2040	2345	2645	2950	3250
	1-3/8	1.0	80	190	300	405	515	620	730	835	945	1055	1160
10.0	7/8	19.3	2085	4865	7645	10420	13200	15980	—	—	—	—	—
	1-1/8	4.9	530	1235	1940	2645	3350	4255	4765	5470	6175	6880	7585
	1-3/8	1.7	185	430	675	920	1165	1410	1650	1895	2140	2385	2630

Note: *Rated tube size.
Note 1: Shaded value indicates more than 10% capacity loss.
Note 2: Blank space indicates more than 15% capacity loss.

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 B-R 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. If capacity is critical, the 1-1/8" O.D. suction line loss is less than 400 BTUH.

Table "C"

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.

CHAPTER II

High Rise Heat Pump Applications (R-22 and R-410A)

The demand for greater vertical separation for the indoor and outdoor sections of heat pump systems, over the years, has lead to the development of the high rise system. Until this time, the maximum allowable liquid lift for R-22 systems was approximately 60 ft. The 60 ft. lift resulted in a pressure drop (60 x .5) of 30 PSI, which left only 5 PSI of the allowable 35 PSI for friction loss. Somewhat higher liquid lifts can be tolerated with R-410A systems, because the allowable (total) liquid line pressure drop for R-410A systems is 50 PSI, also the loss for each foot of liquid lift is .43 PSI versus .50 PSI for R-22.

However, when liquid lifts become high enough to produce a total liquid line pressure drop (lift + friction) of over 35 PSI with R-22 systems or 50 PSI with R-410A systems, the high rise system with subcooler will be 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. The indoor unit must utilize expansion valve flow control.

The high rise system consists of a subcooler (heat exchanger), a capillary tube and associated tubing. The new (Windows) computer program (Pub. No. 32-3312-01) will call for the subcooler, automatically, when the total liquid line pressure drop exceeds 35 PSI (R-22) or 50 PSI (R-410A). The program also selects the proper capillary tube size for the application.

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 (R-22) with no "flashing" (even higher with R-410A.)

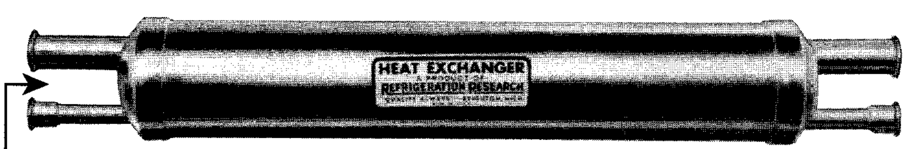
As mentioned earlier, the (windows) computer piping program will call for the subcooler when ever it is required, size the capillary tube and call for any other required accessories. The rest of this chapter is designed to help the system designer who does not have access to the computer program to apply 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 (and is rated for both R-22 or R-410A).

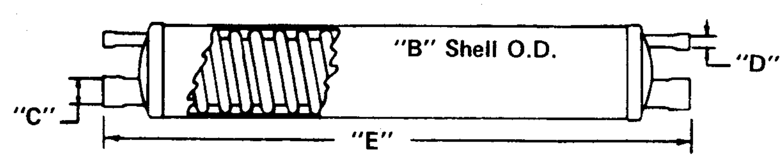
It should be noted that although the heat exchanger used with the high rise system is designed as a suction to liquid heat exchanger, it is not used in that manner. (Suction gas is not routed through the heat exchanger.) Instead, the normal liquid flow is through the suction side of the heat exchanger. 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 R-22 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).

Table "D"

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



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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.

The fact that a small portion of the liquid R-22, 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 R-22 is delivered to the system evaporator, each pound contains less heat, because of the additional subcooling and the net cooling effect is the same. So what have we accomplished? We have delivered 100% liquid to the system expansion valve, in spite of liquid line pressure drops of 100 PSI or more.

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

Table "D," page 18, provides a picture and dimensional information for the heat exchanger.

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

Figure 7 (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 8, page 20, 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 7

Piping Detail – Heat Exchanger

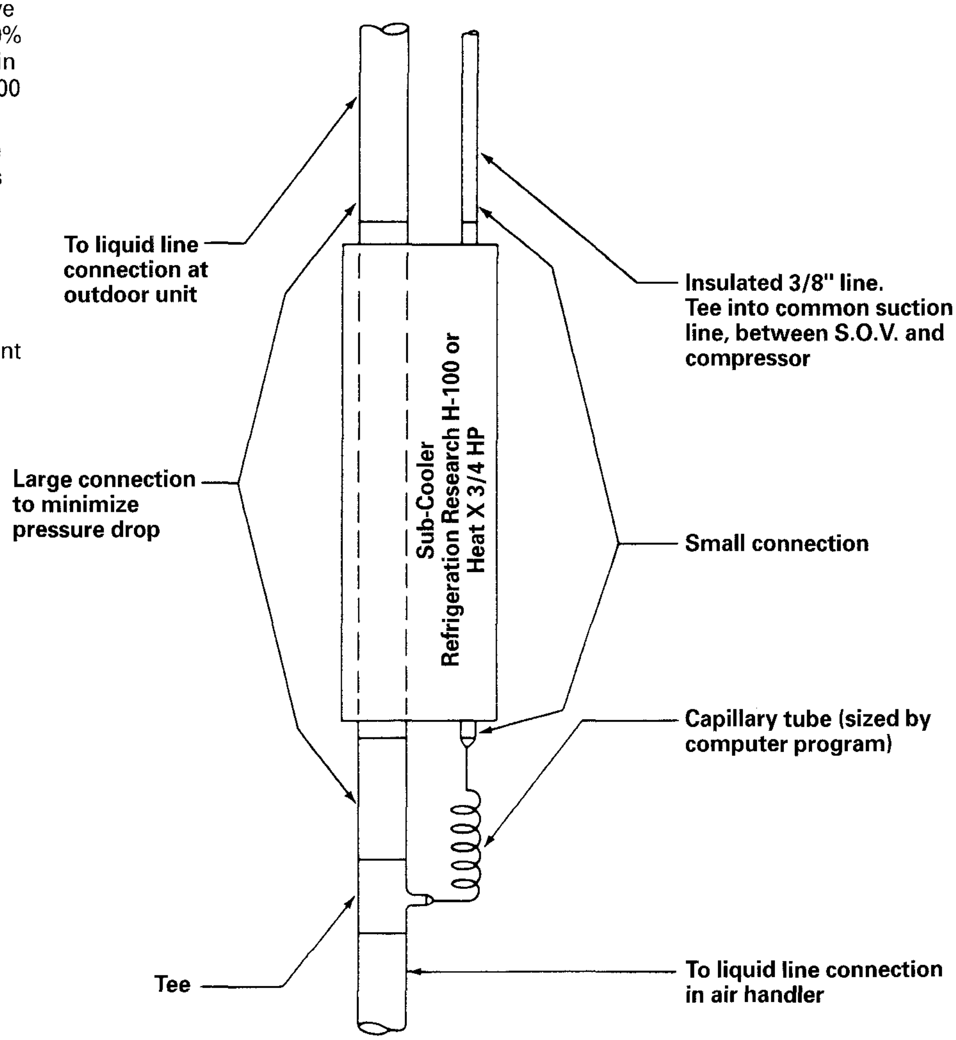
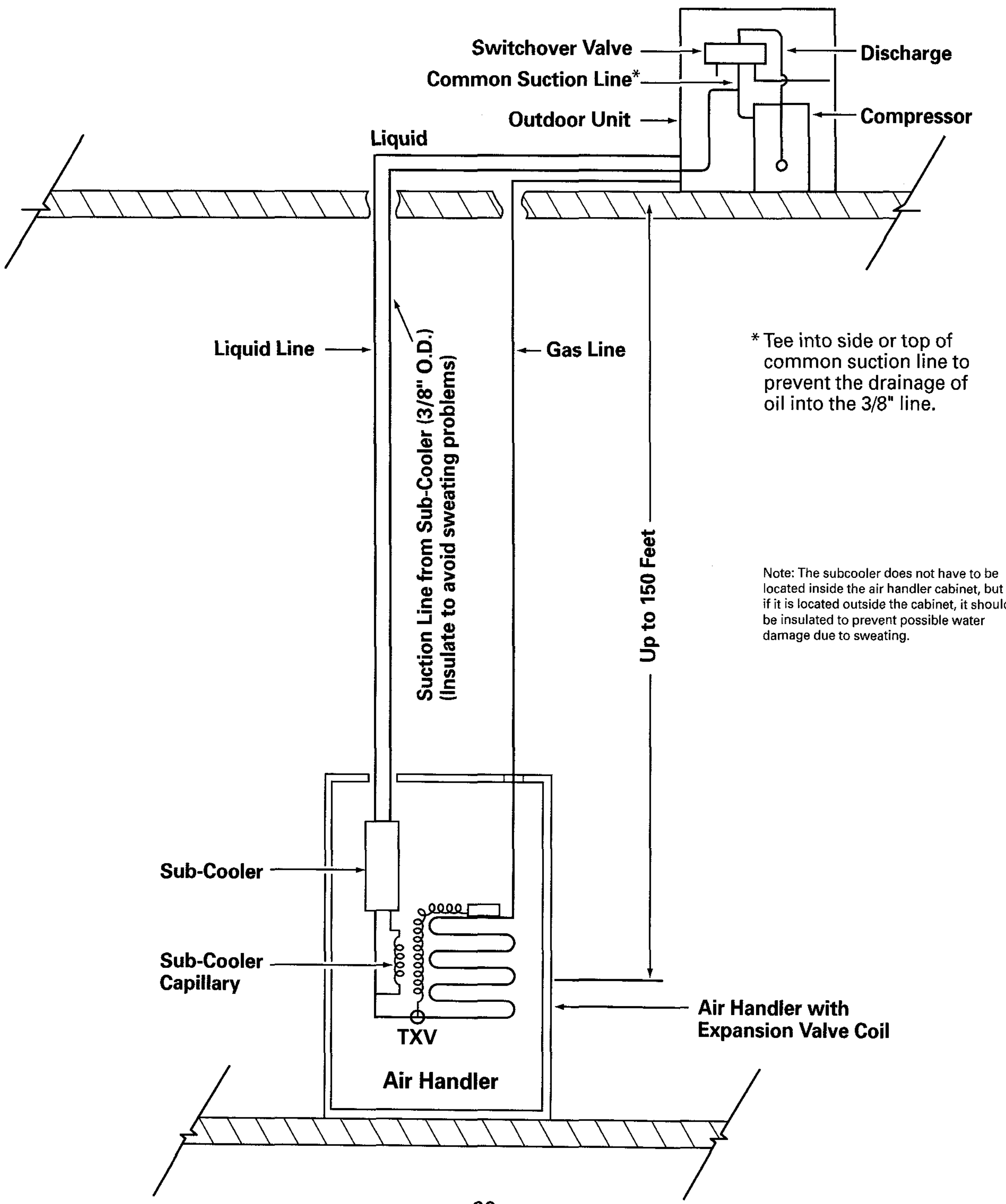


Figure 8

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



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Tables "H" (R-22) and "I" (R-410A) allow you to select the proper capillary tube size, based on excess liquid line pressure drop and system tonnage. The examples below illustrate typical calculations for a system "A" (R-22) and system "B" (R-410A).

Given: R-22 Subcooler, 3/8" O.D. liquid line, 155 equivalent feet, 120 ft. lift (3 tons).

Step #1 Friction loss from Table "A" page 7 = approx. 15 PSI (note: rather than interpolate between 140 and 160 equivalent feet, it is often simpler to multiply the pressure drop for 100 ft. by the equivalent length in hundreds of feet) 9.7 x 1.55 = approx. 15 PSI.

Step #2 Pressure drop due to lift (120 x .5) = 60 PSI.

Step #3 Total pressure drop (15 + 60) = 75 PSI.

Step #4 Excess liquid line pressure drop (75 – 35) = 40 PSI.

Step #5 From Table "H", 3 tons at 40 PSI excess pressure drop requires a 30" x .042" capillary tube.

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 "A-R" (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 "I," 3 1/2 tons at 54 PSI excess pressure drop requires a 30" x .042" capillary tube.

Table "H"

Capillary Tube Selection Table for R-22 Subcooler

(Total Liquid Line Pressure Drop Minus 35 PSI = Excess Pressure Drop)

System Tons	Excess Liquid Line Pressure Drop														
	10	20	30	40	50	60	70	80	90	100	110	120	130	140	150
1.0															
1.5				30" x .042" CAPILLARY TUBE											
2.0															
2.5						34" x .054" CAPILLARY TUBE									
3.0															
3.5										20" x .064" CAPILLARY TUBE					
4.0															
5.0											20" x .080" CAPILLARY TUBE				
6.0															
7.5														(2) 32" x .080" CAPILLARY TUBES	
10.0															

Example: 3 ton system with 40 PSI excess pressure drop requires 30" x .042" capillary tube.

Table "I"

Capillary Tube Selection Table for R-410A Subcooler

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

System Tons	Excess Liquid Line Pressure Drop (PSI)														
	10	20	30	40	50	60	70	80	90	100	110	120	130	140	150
1.0															
1.5					30" x .042" CAPILLARY TUBE										
2.0															
2.5															
3.0								34" x .054" CAPILLARY TUBE							
3.5															
4.0											20" x .064" CAPILLARY TUBE				
5.0															
6.0															
7.5														20" x .080" CAPILLARY TUBE	
10.0															

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

Application Guide

CHAPTER III

Hot Gas Bypass
(Capacity Modulation)

Introduction

Why Capacity Modulation?

The typical residential cooling system performs reasonably well, with a simple on/off control (system thermostat).

While cooling loads vary considerably from design conditions, to mild weather cooling, the changes are usually gradual. Solar and internal loads as well as the flywheel effect of the home and furnishings tend to keep short cycling of the system, in mild weather, within acceptable limits. Humidity control suffers somewhat during mild weather, but with a carefully sized system, it is usually tolerable.

Loads, which vary more dramatically, and over shorter periods of time, however, will often require some means of capacity modulation in order to:

- Provide better humidity control
- Provide better temperature control
- Reduce short cycling problems
- Avoid frosting of evaporator coils

Some examples of loads requiring capacity modulation are:

1 – High percentages of outside air.

The current emphasis on improving indoor air quality often leads to high outside air percentages. Code requirements, or high exhaust requirements are also contributors. It is generally accepted, that outside air percentages greater than 25% require capacity modulation.

2 – On/off type loads such as lighting, industrial processes, etc. can result in wide variations in loads.

3 – Special applications requiring very close control of humidity, temperature, or both.

4 – Unusually high latent loads, which require very low evaporator temperatures, must limit capacity at reduced load or reduced outdoor temperatures. (Evaporator coils may frost.)

5 – Fluctuating loads which may lead to excessive short cycling of equipment.

How Can We Modulate System Capacity?

1 – The application of multiple systems, whenever practical, results in capacity modulation, without sacrificing efficiency, and should be the first consideration. (Multiple systems also have the inherent advantage of not "putting all your eggs in one basket," in the event of a system failure.) If very precise control of capacity modulation is required, hot gas bypass can be applied to the first stage system. An example of this could be two five ton systems matched with a dual circuited 10 ton air handler. Applying hot gas bypass to the first stage system, only, provides modulation from 10 tons to near 5 tons when both stages are called for. When first stage, only, is called for, modulation is from 5 tons to near zero.

2 – **Cylinder unloading.** This method of capacity modulation is very effective on larger systems (factory applied). The industry has found that cylinder unloading is not cost-effective on systems of 10 tons capacity or less.

3 – **Hot gas bypass.** Hot gas bypass, properly applied, provides very precise capacity modulation with a moderate loss in efficiency. (Some energy is expended in pumping the bypassed gas through the system, which provides no cooling effect.)

For applications where cylinder unloading is not available, hot gas bypass is undoubtedly the method of choice, if the application of multiple systems is not practical. (Hot gas bypass can be combined with multiple systems to provide even more precise modulation.)

The remainder of the chapter will cover the field application of hot gas bypass, including, bypass valve selection (sizing), sizing of the hot gas line, and information on accessories, such as solenoid valves, auxiliary side connectors, etc.

Some of the concerns which will be covered are:

- **Oil return**

Introducing the hot gas at the evaporator inlet (side port distributor, or, auxiliary side connector), ensures adequate refrigerant velocity for oil entrainment, through the evaporator, and any suction lifts.

- **Proper superheat in suction gas to compressor**

The refrigerant metering device **must** be a thermostatic expansion valve. The valve will automatically compensate for the highly superheated hot gas, and provide normally superheated suction gas to the compressor (thus preventing compressor overheating or slugging).

- **Undersizing of bypass valve**

Select a valve capable of bypassing the maximum required, at the existing conditions, at the time of the minimum load. (Design load minus minimum load = required bypass.) If the system is significantly oversized, system capacity minus minimum load = required bypass.

Allow for pressure drops through the sideport distributor, or auxiliary side connector, hot gas line, solenoid valve, etc.

(An oversized bypass valve presents no problem, since it is a modulating type valve, and will open only as far as is required. To maintain the desired evaporator pressure.) If more bypass is required than is available from a single valve, two valves can be piped in parallel. (Adjust both valves to open at the same pressure.)

Low head pressure problems are likely to occur with hot gas bypass applications due to:

1 – Operation at low outdoor ambients

2 – The bypassed gas bypasses the condenser, further reducing head pressure. (The published low ambient limit does not apply when hot gas bypass is utilized.) The application of a head pressure control is highly recommended.

Hot Gas Bypass

The purpose of a hot gas bypass system is to artificially load the compressor upon a decrease in evaporator load for one or more of the following reasons:

- Prevent operation of the compressor at excessively low suction pressures. This could cause compressor short cycling, resulting in temperature and humidity control variation.
- Prevent a significant drop in evaporator temperature where reasonably constant conditions must be maintained. This is often necessary for precise temperature control.
- Prevent frosting of the evaporator coil causing serious loss of capacity because of restricted airflow, and potential damage to the compressor.

Types of Discharge Bypass Valves Available

Adjustable and non-adjustable discharge valves are available. The adjustable type uses a spring in the valve head and has the advantage of greater flexibility. The valve can be adjusted at the time of installation and the pressure setting will not be affected by ambient or hot gas temperature. It is recommended that the **adjustable type** with an **external equalizer** be used on all of the applications shown in this guide.

Operation – Hot Gas Bypass Valve

The hot gas bypass regulator or discharge bypass valve automatically responds to changes in suction pressure. When the refrigerant evaporating pressure is above the bypass valve setting, the valve remains closed. When the cooling load drops, the suction pressure drops below the bypass valve setting and the valve begins to open, bypassing a portion of the hot gas directly into the low side, thereby, maintaining the compressor suction pressure at a relatively high level. The amount of valve opening is proportional to the change in the suction pressure, thereby, providing capacity modulation. Capacity reduction over a wide range is possible with proper selection of the components. This is shown in the sample problem. (See page 23)

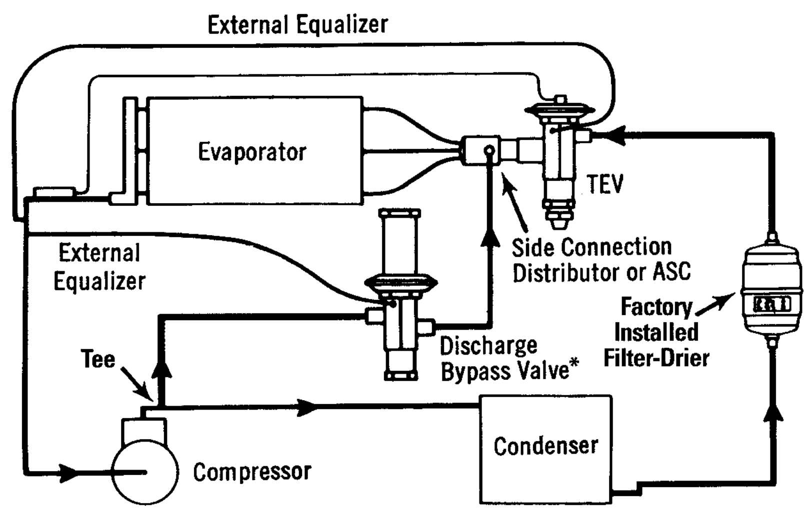
If the suction pressure continues to drop below the valve setting, the valve continues to open until the limit of its stroke is reached. Most applications cannot tolerate sufficient pressure change to open the valve to the limit of its stroke. The amount of pressure change required to move the valve from the closed position to its rated open position varies with the refrigerant used and the minimum evaporator temperature desired. Most manufacturers' valve capacity ratings are based on a 6°F change in suction temperature (approximately 9 PSI for R-22). In other words, a 6°F change in evaporator refrigerant temperature is required to move the valve from the fully closed position to the rated open position or from the rated open position to the fully closed position. This same valve would be able to open further if an 8°F or a 10°F change in evaporator refrigerant temperature, which results in a lower leaving air temperature, could be tolerated.

Note 1: The values given in manufactures' catalogs are valve capacities – not system capacities.

Note 2: Use of discharge bypass valves alone will not maintain adequate head pressure for proper operation under low outdoors ambient operating conditions (below 50 – 55°F).* It will be necessary to use an approved low ambient controlling device to maintain adequate high side pressure under low ambient operating conditions. If bypassing 50% or more, head pressure control is a must.

Figure 9

Recommended Piping Hook-Up for Hot Gas Bypass



• Locate valve at the compressor end of the Hot Gas Line.

Application Guide

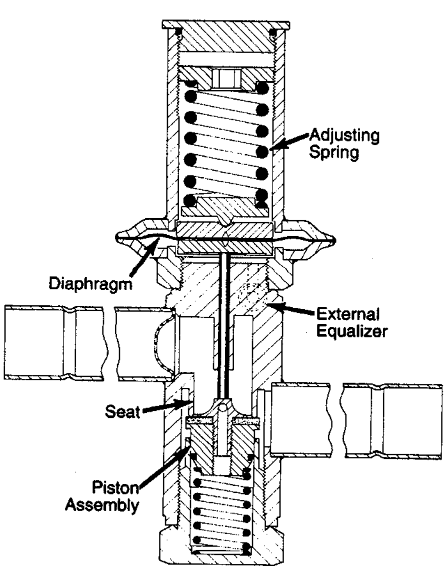
The cutaway view (below) of a typical sporlan hot gas bypass valve illustrates its construction.

When evaporator pressure (sensed through the external equalizer line) falls below the setpoint of the valve, spring pressure against the diaphragm overcomes evaporator pressure and begins to move the piston off its seat. This allows the high pressure, hot gas to flow through the valve to maintain low side pressure. When evaporator pressure has fallen approx. 9 PSI (R-22) below the setpoint, the valve will be open to its rated position.

When evaporator pressure rises above the setpoint, it forces the valve closed.

Figure 10

Typical Sporlan Hot Gas Bypass Valve



U.S. Patent Number 3,402,566

Application

Figure #9 indicates the only recommended piping hook up for hot gas bypass applied to Trane U.P.G. products. Introducing the hot gas at the inlet to the evaporator provides proper refrigerant velocity for oil return through the evaporator and suction lifts, if present (regardless of the amount being bypassed). Good mixing of the bypassed gas with the evaporating refrigerant takes place and the system expansion valve compensates for the added superheat.

This hook-up does require either a sideport distributor or an auxiliary side connector. In some cases, depending on the size of the original distributor and the required amount of bypass, a larger distributor could be required.

An externally equalized bypass valve is recommended for all applications.

Bypass valves should be sized generously, in order to accommodate pressure drops in the hot gas line, etc.

A hand valve installed upstream of the bypass valve facilitates pump down for service operations. The bypass valve must be installed close to the tee in the discharge line, to prevent the accumulation of liquid R-22 in the hot gas line when not bypassing. External equalizer lines should be connected approximately 6" downstream of the TXV Thermal Bulb. (The hot gas bypass valve external equalizer can be connected to the suction line near the outdoor unit if desired.)

Hints to Remember When Applying Hot Gas Bypass

- The refrigerant metering device must be a thermostatic expansion valve.
- Proper refrigerant velocities must be maintained in the evaporator and suction lifts (if any). This dictates the introduction of the hot gas at the evaporator inlet.
- Low ambient cooling problems must be considered. (Head pressure control is often required.)
- Locate the bypass valve close to the compressor to avoid the accumulation of significant amounts of liquid refrigerant in the hot gas line when not bypassing.
- Externally equalized bypass valves are a must.
- Adjustable bypass valves are recommended.
- Do not apply evaporator pressure regulating valves to Trane D.P.G. products (Trane or American Standard).

Heat Pump Systems

When applying hot gas bypass to a heat pump system:

A — The hot gas bypass valve inlet must be connected to the discharge line between the compressor and the switchover valve.

B — A hot gas solenoid valve must be installed upstream of the bypass valve, and wired so that it opens in the cooling mode only.

C — Low head pressure problems during bypass may occur, requiring some means of head pressure control.

D — If a 24 volt solenoid valve is used, be sure that adequate transformer capacity is available.

E — Apply multiple systems whenever possible, to achieve capacity modulation.

F — In most cases, if cooling is required at low outdoor temperatures, it is probably not a good heat pump application.

Application Guide

Selecting The Hot Gas Bypass Valve

The bypass valve must be capable of bypassing the difference between the design load and the anticipated minimum load (at the conditions existing when the minimum load occurs).

Anticipated lower head pressures at the minimum load conditions (further aggravated by the fact that the bypassed gas is bypassing the condenser) will very often, require the application of a head pressure control device.

Pressure drops in the hot gas line, solenoid valve (if used), side-port distributor (or auxiliary side connector plus distributor), all tend to reduce the capacity of the hot gas bypass valve. (Size it generously.)

Table "E" provides a simple method, for the selection of a hot gas bypass valve, and the hot gas line, for average conditions. Table "E" is based on the amount (tons) of bypass required.

Chart "X," page 24, can be used to calculate the pressure drop in the hot gas line, if desired.

Table "F" below, lists information on Sporlan Bypass Valves. (The selections in Table "F," assume a 26°F evaporating temperature, and 80°F condensing temperature, for the minimum load conditions.) The adjustment range of 0 to 80 PSIG is recommended for R-22.

Table "E"

Quick Selection Table For Hot Gas Bypass Valves

R-22 Refrigerant

	Amount of Bypass Required (Design Load Minus Minimum Load)			
	0 – 2 Tons	2.5 – 4 Tons	4.5 – 8 Tons	8.5 – 10 Tons
Hot Gas Bypass Valve	Sporlan ADRSE-2 (or Equiv.)	Sporlan ADRPE-3 (or Equiv.)	Sporlan ADRHE-6 (or Equiv.)	Sporlan DRHE-6 (or Equiv.)
Recommended Hot Gas Line O.D.				
Up to 50 Eq. Ft.	1/2" O.D.	5/8" O.D.	3/4" O.D.	7/8" O.D.
51 to 100 Eq. Ft.	1/2" O.D.	5/8" O.D.	7/8" O.D.	7/8" O.D.

Table "F"

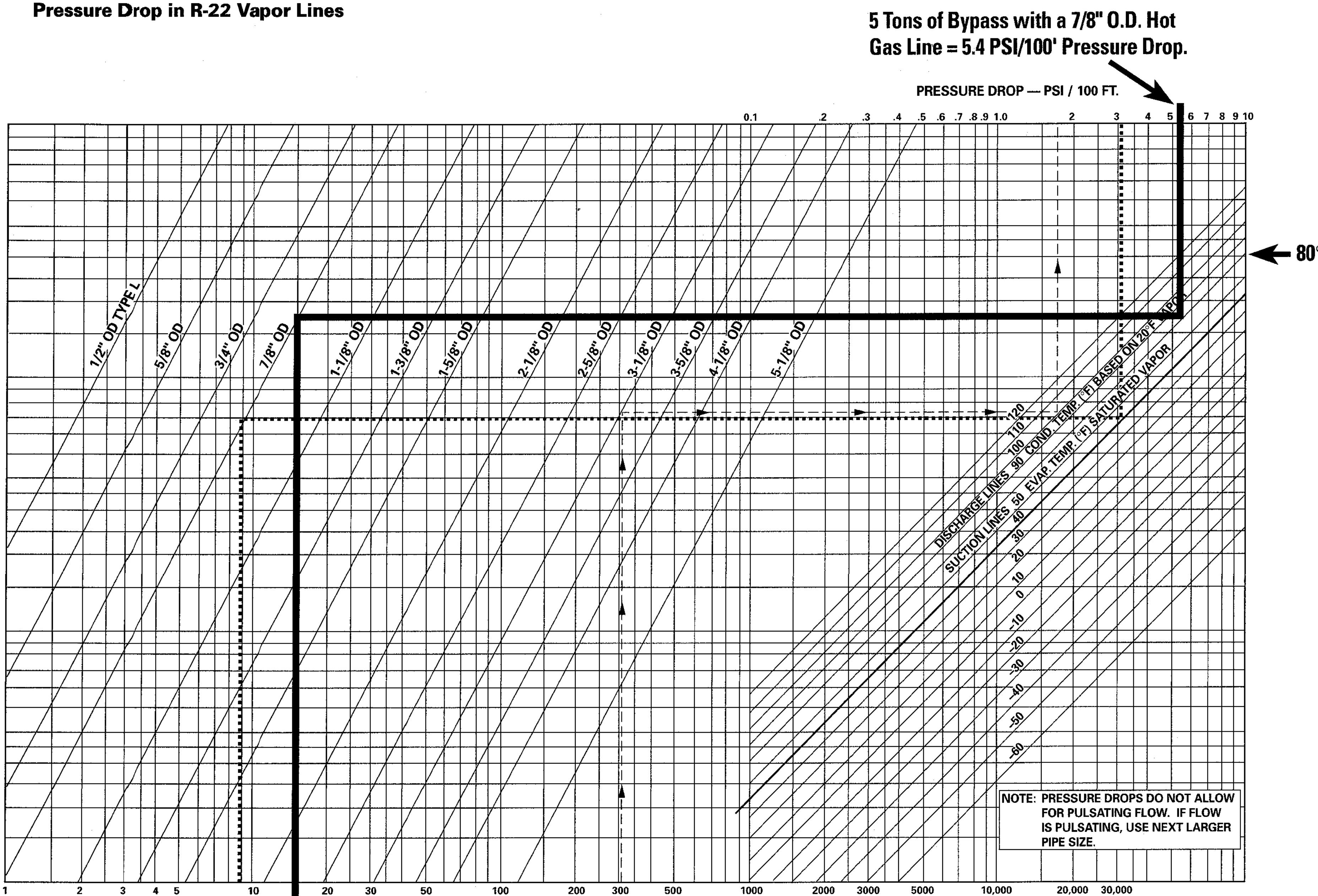
Discharge Bypass Valve Capacities – Tons

Capacities based on 6°F evaporator temperature change from closed to rated opening, discharge temperature 30°F above isentropic compression, 25°F superheat at the compressor, 0° subcooling, and includes both the hot gas bypassed and liquid refrigerant for desuperheating, regardless of whether the liquid is fed through the system thermostatic expansion valve or an auxiliary desuperheating thermostatic expansion valve.

			Minimum Allowable Evaporator Temperature At The Reduced Load – °F																	
		Adjustment	**40**			**26**			**20**			**0**			**-20**			**-40**		
	Valve	**Range**	**Condensing Temperature – °F**																	
Refrigerant	**Type**	**PSIG**	**80**	**100**	**120**	**80**	**100**	**120**	**80**	**100**	**120**	**80**	**100**	**120**	**80**	**100**	**120**	**80**	**100**	**120**
Adjustable Models																				
22	ADRI-1-1/4 ADRIE-1-1/4	0/55	—	—	—	0.35	0.45	0.56	0.41	0.53	0.66	0.58	0.75	0.93	0.54	0.69	0.87	0.49	0.64	0.80
	ADRS-2	0/30	—	—	—	—	—	—	—	—	—	2.93	3.77	4.73	2.82	3.63	4.57	2.72	3.51	4.42
	ADRSE-2	0/80	2.65	3.40	4.26	2.69	3.45	4.34	2.71	3.48	4.37	2.88	3.70	4.66	—	—	—	—	—	—
	ADRP-3	0/30	—	—	—	—	—	—	—	—	—	5.56	7.16	9.00	5.61	7.23	9.10	5.47	7.06	8.90
	ADRPE-3	0/80	4.50	5.78	7.25	4.72	6.06	7.61	4.80	6.17	7.75	5.24	6.73	8.47	—	—	—	—	—	—
	$\boxed{\text{ADRHE-6}}$	0/30	—	—	—	—	—	—	—	—	—	10.5	13.5	17.0	10.6	13.7	17.2	10.3	13.3	16.7
		0/80	6.89	8.84	11.1	7.46	9.58	12.0	7.68	9.87	12.4	8.12	11.4	14.4	—	—	—	—	—	—
Adjustable "Remote Bulb" Models																				
12	DRHE-6	25/35	5.51	7.11	8.96	4.86	6.26	7.89	These models are recommended for air conditioning temperature ranges only.											
22		55/70	14.3	19.0	24.0	12.7	16.4	20.6												
134a		25/35	7.03	9.26	11.9	6.06	7.98	10.2												
502		65/80	13.9	18.2	22.6	12.6	15.8	19.4												

Chart "J"

Pressure Drop in R-22 Vapor Lines



Application Guide

Figures 11 and 12 on this page and 13 on page 26 provide information on accessories for hot gas bypass applications, plus capacity multipliers for evaporating temperature changes other than the standard 6°. (See Figure 12.)

Figure 11 lists compatible distributor/auxiliary side connector combinations, complete with fitting sizes. (O.D.M. = outside diameter, male and O.D.F. = outside diameter, female.)

Figure 12 lists bypass valve capacity multipliers for evaporating temperature changes, other than 6°F. Note that the capacity multiplier for a 2°F evaporating change is .70 (26° evaporating temperature). A 26° refrigerant evaporating temperature, with normal airflows' will result in a coil surface temperature above 32°F. (Frosting should not occur.)

Figure 13 lists capacities of hot gas solenoid valves. Hot gas solenoid valves are not usually required on straight cooling applications, since pump down is not normally permitted. (The hot gas solenoid valve is required on heat pump applications, however, in order to disable the hot gas bypass during the heating mode.)

Note that a ME19S250 (or 270) hot gas solenoid valve (R-22) has a capacity of 9.46 tons x a correction factor of .95 for a 26° evaporating temperature (9.46 x .95 = approx. 9.0 tons). The above values are based on a 10 PSI pressure drop across the solenoid valve. If no more than a 5 PSI pressure drop across the solenoid valve can be tolerated, its capacity is 6.77 x .95 or approx. 6.4 tons of bypass. (Selecting a hot gas bypass valve with some excess capacity, would allow us to use the 10 PSI valve.) The solenoid valve's capacity would be derated somewhat more, assuming a condensing temperature of 80° on a mild day. (The Table in Figure 13 is based on a 100° condensing temperature.)

Figure 14, page 27, lists capacities for liquid line solenoid valves. (Do not confuse Figure 14 with Figure 13, which covers hot gas solenoid valves.) Much larger port sizes are required to handle R-22 vapor.

The circled values in Figure 14 indicate that the ME9S230 (or 240) liquid line solenoid valve has a capacity of 8 tons at a 3 PSI pressure drop across the valve (R-22).

Could we use this solenoid valve on a 10 ton system? The answer is yes, if the 4.7 PSI pressure drop does not result in a total liquid line pressure drop of over 35 PSI ($3.0 \text{ PSIG} \times (10/8)^2 = 4.7 \text{ PSI}$).

Figure 11

Auxiliary Side Connectors

Distributor Type Number	ASC Connection Sizes (Inches) ASC Type Number	Inlet ODM	Outlet ODF	Auxiliary ODF
1620, 1622	ASC-5-4	5/8	5/8	1/2
1112, 1113	ASC-7-4	7/8	7/8	1/2
1115, 1116	ASC-9-5	1-1/8	1-1/8	5/8
1117, 1126	ASC-11-7	1-3/8	1-3/8	7/8
1125, 1127 1143	ASC-13-9	1-5/8	1-5/8	1-1/8

Figure 12

Capacity Multipliers

For Evaporator Temperature Changes Other Than 6°F Nominal Change

Evaporator Temp. Change °F	Refrigerant	Evaporator Temperature °F 40	26	20	0 and Below
2°	12 & 134a	0.65	0.65	0.65	0.65
	22 & 502	0.72	0.70	0.70	
4°	12 & 134a	0.80	0.80	0.80	0.74
	22 & 502	0.87	0.85	0.85	
8°	12 & 134a	1.11	1.11	1.11	1.09
	22 & 502	1.17	1.15	1.11	
10°	12 & 134A	1.22	1.20	1.19	1.11
	22 & 502	1.34	1.27	1.25	

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Figure 13

Hot Gas Solenoid Valve Capacities – Tons

Capacities based on 100°F condensing temperature, isentropic compression plus 50°F, 40°F evaporator and 65°F suction gas.
For other evaporator conditions use the multipliers in the table below.

Valve Type		Connections Inches	Refrigerants							
"A" & "B" Series	"E" Series Extended Connections		12		22		134a		502	
			Pressure Drop Across Valve Port – PSI							
			5	10	5	10	5	10	5	10
A3F1	—	1/4 SAE	0.26	0.35	0.37	0.51	0.31	0.42	0.30	0.42
A3S1	—	1/4 or 3/8 ODF								
—	E5S120	1/4 ODF	0.61	0.84	0.88	1.22	0.74	1.00	0.72	1.00
—	E5S130	3/8 ODF								
MB6F1	—	3/8 SAE	1.01	1.43	1.51	2.10	1.27	1.74	1.30	1.70
MB6S1	ME6S130	3/8 ODF								
MB6S1	ME6S140	1/2 ODF								
MB9F2	—	3/8 SAE	1.50	0.21	2.17	3.04	1.80	2.50	1.80	2.50
—	ME9S230	3/8 ODF								
MB9S2	ME9S240	1/2 ODF								
MB10F2	—	1/2 SAE	2.30	3.20	3.37	4.69	2.80	3.90	2.80	3.80
—	ME10S240	1/2 ODF								
MB10S2	ME10S250	5/8 ODF								
MB14S2	ME14S250	5/8 ODF	3.20	4.40	4.58	6.40	3.80	5.30	3.80	5.30
MB19S2	ME19S250	5/8 ODF	4.70	6.50	6.77	9.46	5.70	7.90	5.70	8.00
MB25S2	ME19S270	7/8 ODF								
MB25S2	ME25S270	7/8 ODF	7.50	10.5	10.8	15.1	9.10	12.7	8.90	12.5
MB25S2	ME25S290	1-1/8 ODF								

Correction Factors

For evaporator temperatures at the reduced load condition						
Evaporator Temperature °F	40°	26°	20°	0°	-20°	-40°
Multiplier	1.00	.95	.93	.87	.81	.75

Application Guide

Figure 14

Liquid Line Solenoid Valve Capacity Selection Table

Type Number						Tons of Refrigeration											
"A" & "B" Series		"E" Series Extended Connections				Pressure Drop – PSI											
With Manual Lift Stem						1				2				3			
Normally Closed				Connections (Inches)	Port Size (Inches)	12	22	134a	502	12	22	134a	502	12	22	134a	502
A3P1	—	—	—	3/8 NPT Female	.101	0.7	0.9	0.8	0.6	1.0	1.3	1.2	0.8	1.2	1.6	1.5	1.0
A3F1	—	—	—	1/4 SAE Flare													
A3S1	—	E3S120	—	1/4 ODF Solder													
A3S1	—	E3S130	—	3/8 ODF Solder													
—	—	E5S120	—	1/4 ODF Solder	.150	1.2	1.6	1.5	1.1	1.8	2.3	2.1	1.5	2.2	2.8	2.6	1.9
—	—	E5S130	—	3/8 ODF Solder													
—	MB6P1	—	—	3/8 NPT Female	3/16	2.2	2.9	2.7	1.9	3.1	4.0	3.8	2.6	3.8	4.9	4.6	3.2
—	MB6F1	—	—	3/8 SAE Flare													
—	MB6S1	—	ME6S130	3/8 ODF Solder													
—	MB6S1	—	ME6S140	1/2 ODF Solder													
—	MB9P2	—	—	3/8 NPT Female	9/32	3.6	4.7	4.4	3.0	5.1	6.6	6.2	4.3	6.2	8.0	7.5	5.2
—	MB9F2	—	—	3/8 SAE Flare													
—	—	—	ME9S230	3/8 ODF Solder													
—	MB9S2	—	ME9S240	1/2 ODF Solder													

Application Guide

How To Estimate Minimum Loads

We are not always given minimum loads, which are necessary, in order to determine the required amount of bypass. A sample calculation follows:

A small commercial application located in Tyler, TX has the following requirements:

Outdoor design = 97° D.B./ 76° W.B. (103 grains water vapor per lb. of dry air.)

Indoor design = 78° D.B./ 64.9° W.B. (72 gr./lb.)

1700 CFM of outside air is required.

We have been given the following information:

Internal sensible load (constant) = 98200 BTUH

Transmission load (sensible) = 6600 BTUH

Internal latent load = 800 BTUH

A — What are the total sensible and latent loads at design conditions?

Outside air sensible load =
1700 x 1.08 x (97 – 78) or 34884 BTUH

Outside air latent load =
1700 x .68 x (103 – 72) or 35836 BTUH

Sensible Loads
Internal = 98200 BTUH
Transmission = 6600 BTUH
Outside air = 34884 BTUH

Total (H_S) 139,684 BTUH

Latent Loads
Internal = 800 BTUH
Outside air = 35836 BTUH

Total (H_L) 36,636 BTUH

B — What are the sensible and latent loads at 60° D.B. outdoors? (Assume 80% rel. humidity.)

Sensible Loads
Internal (constant) = 98200 BTUH
Transmission (loss)

$6600 \times \frac{78-60}{97-78} = -6253$ BTUH or,

$\frac{6600}{19} \times [78-60] = 6253$

Outside air (loss) =
1700 x 1.08 x (78 – 60) = –33048 BTUH

Net sensible load @ 60° outdoors = 58899 BTUH

Latent loads
Internal = 800 BTUH

Outside air (loss) 1700 x .68 x (72 gr. – 62 gr.) = -11560 BTUH

Net latent load @ 60° outdoors –10,760 BTUH

Since the latent load is negative, only the sensible load will be considered.

Design sensible load = 139684 BTUH

Sensible load @ 60° outdoors = 58899 BTUH

Excess sensible capacity = 80785 BTUH (139684 minus 58899)

Note: If the system is significantly oversized at design conditions, use design system capacity minus minimum load.

Assuming 70% sensible capacity, at 60° D.B. the required bypass (total capacity) = 80785 ÷ .70 or 115407 BTUH (total) (approx. 9.6 tons).

One DRHE-6 (or two ADRHE-6, piped in parallel) will provide some excess in rated capacity, to allow for pressure drops in the hot gas line, etc. (See tables "E" or "F")

If two stages of cooling are provided, the first stage, only, would require approx. 5 tons of bypass.

$\frac{115470}{2} = 57704$ BTUH

$\frac{57704}{12000} = 4.8$ tons

Refrigerant Pipe Sizing Worksheet

Project ______________________ Outdoor Model ______________________

Address ______________________ Indoor Model ______________________

______________________ A.R.I. Capacity ______________________

☐ Cooling ☐ Heat Pump

Linear Piping Runs

Horizontal Lengths ______________________ = ________ ft

Vertical Lengths ______________________ = ________ ft

Total Piping Run = (a) ________ ft

Fittings and Refrigerant Specialties

________ Short Elbows ________ 45° Elbows ________ Soleniod Valve

________ Long Elbows ________ Sight Glass[1]

[1] Sight glasses are not recommended on Tyler built units.

Equivalent Lengths

LIQUID LINE

Tubing Size	Fittings/Accessories Quantity		Fittings/Accessories Equivalent Length		Equivalent Length Subtotals
____	____	x	____	=	____
____	____	x	____	=	____
____	____	x	____	=	____
____	____	x	____	=	____

Liquid line fittings/accessories equivalent total = (b) ________ ft

GAS LINE

Tubing Size	Fittings/Accessories Quantity		Fittings/Accessories Equivalent Length		Equivalent Length Subtotals
____	____	x	____	=	____
____	____	x	____	=	____
____	____	x	____	=	____
____	____	x	____	=	____

Gas line fittings/accessories equivalent total = (x) ________ ft

Pressure Losses and Gains

Frictional Losses

	LIQUID LINE	GAS LINE
Actual Piping Lengths [From (a)]	(a) ________ ft	(a) ________ ft
Fittings/Accessories Equivalent Feet (b) or (x)	(b) + ________ ft	(x) + ________ ft
Total Equivalent Piping Run =	________ ft	________ ft
Pressure Drop/Foot $\left[\frac{\text{P.D./100 ft}}{100} = \frac{\_\_\_\_}{100} = \boxed{}\right]$	x ________ psi/ft	x ________ psi/ft
Frictional Pressure Loss =	(c) ________ p.s.i.	________ p.s.i.

Hydrostatic Force Liquid Line Only

Vertical Separation between Indoor/Outdoor = ________ ft

R-22 Hydrostatic Force = x 0.50 psi/ft

If (d) is greater than 25 psi (See Note B) (d) ________ psi

System Loss

Frictional Loss (c) ________

Hydrostatic Force (See Note A) (d) ± ________

Total = ________ psi

Liquid Pressure Drop **Can Not** exceed 35 PSI with R-22 or 50 PSI with R-410A

GAS LINE PRESSURE DROPS

I. If P.D. is less than 3 psi, P.D. in acceptable range.
II. P.D. between 3 and 6 psi, check capacity loss (Chart B – Refrigerant Piping Manual).
III. If P.D. exceeds 6 psi, capacity loss may require oversize pipe or larger equipment size because of capacity losses.

NOTES

A. Heat Pump Systems: Always add (+) (d value) to (c value) for system total.
Air Conditioning Systems:
Condensing unit above Indoor Evaporator: subtract (–) (d value) from (c value).
Condensing unit below Indoor Evaporator: Add (+) (d value) to (c value).

B. A hydrostatic force greater than 25 psi may not provide sufficient pressure available to overcome frictional losses on Heat Pump Systems and Air Conditioning Systems with the outdoor condensing unit below the indoor evaporator unit.

Pub. No. 22-3217-01 P.I. (L)

			Standard Piping Size						Oversized Piping					
			Pipe Size		Pressure Drops[1]		Lbs. R-22 (100 ft.)		Pipe Size		Pressure Drops[1]		Lbs. R-22 (100 ft.)	
Nominal Tonnage	#/Min	ARI [1] Capacity	Gas	Liquid	Gas (100 ft.)	Liquid (100 ft.)	Gas	Liquid	Gas	Liquid	Gas (100 ft.)	Liquid (100 ft.)	Gas	Liquid
1	3.00	12,000	5/8	1/4	2.60	17.0	0.3	1.50	5/8	5/16	2.60	5.0	0.30	2.30
	3.75	15,000			3.40	26.0								
1 1/2	3.75	15,000	5/8	1/4	3.40	26.0	0.3	1.50	5/8	5/16	3.40	7.8	0.30	2.30
	4.50	18,000			4.80	38.0[4]			3/4	5/16	1.80	11.0	0.40	2.30
2	5.25	21,000	3/4	5/16	2.60	14.0	0.4	2.30	7/8	3/8	1.20	3.8	0.50	3.80
	6.00	24,000			3.40	17.0					1.50	4.8		
	6.75	27,000			4.10	20.0					1.80	5.8		
2 1/2	6.75	27,000	3/4	5/16	4.10	20.0	0.4	2.30	7/8	3/8	1.80	5.8	0.50	3.80
	7.50	30,000			5.10	25.0[4]					2.30	7.0		
	8.25	33,000			6.00	32.0[4]					2.75	8.7		
3[5]	8.25	33,000	7/8	5/16	2.75	32.0[4]	0.5	2.30	7/8[2]	3/8	2.75	8.7	0.50	3.80
	9.00	36,000			3.20	36.0[4]			7/8[2]	3/8	3.20	10.2	0.50	
	9.75	39,000			3.70	41.0[4]			7/8 HP[2]		3.70	11.8	0.50[3]	
3 1/2[5]	10.00	40,000	7/8	5/16	3.90	45.0[4]	0.5	2.30	1 1/8	3/8	1.00	12.0	0.90	3.80
	10.50	42,000			4.10	48.0[4]					1.10	13.0		
	11.25	45,000			4.70	54.0[4]					1.20	14.0		
4	11.25	45,000	1 1/8	3/8	1.20	14.0	0.9	3.80	1 1/8	1/2	1.20	2.8	0.90	7.30
	12.00	48,000			1.40	16.0					1.40	3.9		
	12.75	51,000			1.60	18.0[4]					1.60	4.0		
5	14.00	56,000	1 1/8	3/8	1.90	22.5[4]	0.9	3.80	1 3/8	1/2	0.64	4.8	1.30	7.30
	14.75	59,000			2.15	24.5[4]					0.70	5.1		
	15.50	62,000			2.30	26.0[4]					0.76	5.6		
6	16.50	66,000	1 1/8	3/8	2.45	27.0[4]	0.9	3.80	1 3/8	1/2	0.80	6.0	1.30	7.30
	17.25	69,000			2.80	31.0[4]					0.90	6.8		
	18.00	72,000			3.20	35.0[4]					1.00	7.6		
7 1/2	21.50	86,000	1 3/8	1/2	1.50	10.0	1.3	7.30	1 5/8	5/8	0.65	3.2	1.90	11.80
	22.50	90,000			1.65	12.0[4]					0.70	3.7		
	24.50	98,000			1.90	14.0[4]					0.83	4.2		
10	29.00	116,000	1 3/8	1/2	2.80	17.5[4]	1.3	7.30	1 5/8	5/8	1.20	5.5	1.90	11.80
	30.00	120,000			2.90	19.0[4]					1.30	6.0		
	31.25	125,000			3.00	21.0[4]					1.40	6.5		

[1] Adjusted for net capacity.
[2] Max line size on vertical runs and heat pump systems.
[3] 7/8 = 0.50# 1 1/8 = 0.90#
[4] Velocity exceeds 300 FPM — cannot use a quick closing device, such as a solenoid valve.
[5] Standard size for liquid line — pre 1992 models is 5/16 inch, 1992 and later models — 3/8 inch liquid line is standard.

Equivalent Lengths Of Valves And Fittings					
O.D Line Size (in.)	45° ELL	Short Radius ELL	Long Radius ELL	Solenoid Value	Sight Glass
1/4	2.0	4.6	3.1	17	1.2
5/16	2.1	4.6	3.1	20	1.4
3/8	2.2	4.7	3.2	22	1.6
1/2	2.4	4.7	3.2	24	1.7
5/8	2.9	5.7	3.9	25	2.3
3/4	3.3	6.5	4.5	25	2.9
7/8	3.9	7.8	5.3	28	3.7
1 1/8	1.4	2.7	1.9	29	2.5
1 3/8	1.6	3.2	2.2	33	2.7
1 5/8	1.9	3.8	2.6	34	3.0

Notes

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Literature Order Number		
File No.	Pub. No. 32-3009-03	7/00
Supersedes	Pub. No. 32-3009-02	1/00
Stocking Location	-	P.I. (L)