ACKNOWLEDGEMENT

The success of the 20th Annual Meeting of the International Institute of Ammonia Refrigeration is due to the quality of the technical papers in this volume and the labor of their authors. IIAR expresses its deep appreciation to the authors, reviewers, and editors for their contributions to the ammonia refrigeration industry.

Board of Directors, International Institute of Ammonia Refrigeration

ABOUT THIS VOLUME

IIAR Technical Papers are subjected to rigorous technical peer review.

The views expressed in the papers in this volume are those of the authors, not the International Institute of Ammonia Refrigeration. They are not official positions of the Institute and are not officially endorsed.

EDITORS

Christopher P. Combs, Project Coordinator
Daniel R. Kuespert, Ph.D., Technical Director
M. Kent Anderson, President

International Institute of Ammonia Refrigeration
1200 Nineteenth St., NW
Suite 300
Washington, DC 20036-2422

+1-202-857-1110 (voice)
+1-202-223-4579 (fax)
www.iiar.org

Ammonia Piping Design

Vernon Alexander, P.E.
Niceville, FL

Introduction

The piping which interconnects the many equipment components is a very important element in a refrigeration system. Its design can have a substantial effect upon the performance and energy efficiency of the system.

This paper deals with the sizing and layout of the various lines through which the ammonia flows. No attempt is made to become involved in the fundamentals related to any particular type of fluid flow which may occur nor any unusual instances. Rather, in keeping with the theme of this meeting, we start with a review of several basic principles related to flow in pipes.

Basic Flow Principles

Flow

1. The substance flowing may be a vapor or a liquid or in some instances two phases, a mixture of liquid and vapor.
2. Flow will occur only when there is a difference in head or pressure between the inlet and a point downstream.
3. Flow increases as the difference in head or pressure increases.
4. Flow is expressed quantitatively in CFM, GPM, lb/min or ft^3/min.

Head

1. For liquids and gases, head can be expressed either in feet or psi. For liquids, head can be increased by means of a liquid pump or an external source of higher pressure.
2. For gases, head can be increased by means of a compressor or an external source of higher gas or vapor pressure.

Pressure Drop

1 Pressure drop is a measure of resistance to flow and is indicated as ΔP.
2 As flow increases in a given size line, pressure drop will increase.
3. Pressure drop is expressed either in psi/100 ft equivalent length or as temperature loss ΔT in °F per 100 ft equivalent length.

Sources of Pressure Drop

Pressure drop occurs in the components of a system through which ammonia flows such as pipe and fittings, shut off and control valves, strainers, evaporative condenser and evaporator coils and static liquid heads.

Figure 1 shows in schematic the various components of the high stage portion of a two stage system and shows locations where pressure drop will occur. The highest pressure will be in the discharge line of a compressor before oil is separated, if a screw compressor, or before passing through the discharge stop valve, if a reciprocating or vane rotary type. Actual discharge pressure will be determined by the pressure at which ammonia is condensed in the condenser. Discharge pressure will always be greater than condensing pressure.

Suction pressure in most instances will be controlled by a pressure-sensitive device which is set to maintain a constant pressure regardless of the evaporator load at any given moment. It will be lowest just after passing through the suction stop valve on the compressor.

Effects of Pressure Drop

To determine how pressure drop affects pressures throughout the system, refer to Figure 1. Starting at the compressor discharge (CD), there is a pressure drop $\Delta P1$ in the piping to the condenser. There is a pressure drop $\Delta P2$ in the condenser coil. The pressure of the liquid as it leaves the condenser coil is shown as (CP). The compressor discharge pressure (DP) can be determined by $CP + \Delta P1 + \Delta P2$.

Since gravity flow occurs in the liquid line from the condenser to the receiver, the head of liquid in the line overcomes pipe resistance so that the pressure in the receiver is the same as that in the condenser (CP). To ensure that the pressure between the two components is the same, an equalizing line is connected between them.

The high temperature liquid (HTL) in the receiver at high pressure can be distributed to the evaporators by various means such as:

1. Directly to the evaporators which operate as direct expansion units.
2. Into a vessel (intercooler or pump receiver) for use as makeup liquid for a refrigerating load.
3. Into a subcooling coil installed within a vessel to cool medium temperature liquid (MTL) which is used as makeup for a refrigerating load.
4. From a pilot receiver into a controlled pressure receiver, from which it is supplied by gas pressure to a refrigerating load.

In this instance, the liquid (HTL) from the receiver flows through piping which has a pressure drop, $\Delta P3 + \Delta P4$, to a liquid throttling device which may have a relatively large pressure drop, $\Delta P8$, and then into the evaporator which has a pressure drop $\Delta P9$.

Leaving the evaporator, the single phase flow passes through the control valve group which has a pressure drop of $\Delta P10$. At this point evaporator pressure (EP) should be indicated by instrument so that evaporating temperature can be determined. Flow continues through piping, which has a pressure drop $\Delta P11$, to the intercooler where there is no pressure drop. Flow continues through piping, which has a pressure drop ΔP12, to the high stage compressor suction (HBS). The compressor suction HBS = EP -(ΔP10 + ΔP11 + ΔP12).

Figure 2 is a schematic flow diagram of the low stage portion of the two stage system. In this instance, liquid is pump recirculated. To avoid the possibility of pump cavitation, there should be sufficient liquid head in the accumulator/pump receiver to adequately overcome the net positive suction head requirements of the pump and its suction piping.

The low temperature liquid (LTL) is supplied by the pump through piping, which has a pressure drop of ΔP14, to the evaporator control group. The liquid throttling device in the line to the evaporator has a pressure drop of ΔP15. The evaporator coil, which has an evaporating temperature (ET), has a pressure drop of ΔP16.

The two-phase flow leaving the evaporator unit passes through the control group, which has a pressure drop ΔP17. It continues through piping which has a pressure drop ΔP18, to the pump receiver. It passes through the pump receiver, which has no pressure drop. It continues through piping, which has a pressure drop ΔP19, to the compressor suction. The compressor suction pressure (LBS) will be the total of ΔP16, ΔP17, ΔP18, and ΔP19 subtracted from the evaporating pressure. LBS = EP - (ΔP16 + ΔP17 + ΔP18 + ΔP19).

As liquid overfeed systems dominate the low temperature applications, it is necessary for the low temperature liquid (LTL) line to be sized for the total amount of liquid to be circulated, that is, 4 times that being evaporated if at a 4 to 1 rate, or 3 times if at a 3 to 1 rate.

Pressure-Volume-Temperature Relationships

Before looking at how various portions of a system are affected by pressure drop a review of three characteristics of ammonia is in order.

1. The refrigerant tables for ammonia show the pressure-temperature-volume relationship for it as a saturated refrigerant.
2. The pressure-volume relationship shows that at low pressures, volume is large and as pressure increases volume decreases.
3. The pressure-temperature relationship shows that as pressure is lowered, temperature also is lowered.

To illustrate the effect these characteristics have on pressure and volume note the variation in pressure change which results when a one °F difference in temperature occurs over a wide range of temperatures. (See Table 1)

Analysis of Table 1 reveals:

<u>At high temperatures</u>:

1. The pressure difference is 3.1 times greater at 95/96 °F, than the 1°F temperature difference.
2. Volume is small because vapor is dense.

At medium temperatures

1. The pressure and temperature differences at 20°F are approximately the same 1° versus 1.1 psi.
2. Volume is larger - approximately 3.9 times greater than compressor discharge volume at 95°F.

At low temperatures 0° to -40°

1. Note the dramatic increase in volume as temperatures become lower. Volume at -40°F is 16.2 times larger than at 95°F.
2. Note that the pressure difference per degree of temperature change becomes increasingly smaller as the temperature becomes colder: 0.31 psi difference at -40°F.

Pressure-Temperature Penalty Chart

To provide a better understanding of the penalty relationship between pressure and temperature differences, in 1980 I prepared Chart 1, Ammonia Pressure/Temperature Penalty due to Pressure Drop. One curve is shown for pressure difference and three curves for temperature differences over a range of +40°F to -50°F. This chart highlights the dramatic effect that lower evaporating temperatures have upon the penalty.

Note the 2 psi ΔP curve at +20°F evaporating temperature: the ΔT (read to left) is 2°F. At -10°F evaporator, the ΔT loss is 3.3°F. At -40°F evaporator temperature, the ΔT loss is 7.0°F. This means that where suction pressures are in the 25 to 35 psig range, a total pressure drop of 2 psi would be considered satisfactory. However, when evaporating temperature is -40°F, a total pressure drop of 2 psi should be reduced. This -40°F evaporator condition is better illustrated by use of the ΔT curves.

The 4°F ΔT curve at -40°F evaporator shows ΔP (read to right) as 1.2 psi. For a total pressure drop of 1.2 psi, suction pressure would be 10.41 minus 1.2 = 9.21 psia and a saturated temperature of -44°F.

Recognition of these relationships at low temperatures is very important when making pipe sizing determinations for applications where suction temperatures are -20°F and lower.

Pipe Sizing Methods

Prior to 1977 piping was sized by various methods, the most common being charts and tables prepared by the Air Conditioning and Refrigeration Institute (ARI). Other data and methods had been developed and used by certain equipment manufacturers and was included in their engineering data. All methods involved the use of charts in which values were indicated along the edges and traced through a maze of parallel lines with the answer interpolated along the edge of another side. It was necessary to repeat this procedure each time a more suitable answer was desired.

ASHRAE Bulletin RP 185

The 1977 ASHRAE Bulletin RP 185 Refrigerant Line Sizing completely changed and simplified the method of sizing pipe. Having sized pipe for many systems prior to its publication, it represented, in my opinion, a major improvement over the methods previously available.

It provides separate charts for suction, discharge and liquid lines along with a table for selecting the refrigerant flow rate as well as other useful tables and charts.

In order to understand how to use these charts, assume we are sizing simplified suction, discharge, liquid and defrosting hot gas lines for a 1000 ton single stage system, evaporator temperature 20°F (33.5 psig), condensing temperature 95°F (181.1 psig), 20°F recirculated liquid at 3 to 1 rate and 100 tons of evaporators defrosted at one time.

Ammonia Flow Rate

In the case of this simple recirculating system, there are two separate sources of liquid supply with each having a different flow rate. One source supplies warm makeup liquid from the high-pressure receiver to the pump receiver, where it flashes and its temperature is lowered. The other source supplies cold liquid from the pump receiver to the evaporators. This pumped cold liquid is recirculated at a designated rate of 3 to 1 in this example.

Now refer to Table 2, which is Table 1-717, Ammonia Flow Rate of ASHRAE Bulletin RP 185. The left hand column, Liquid Temp.°F, lists temperatures of liquid as it enters the expansion device or at the accumulator/pump receiver. Evaporating Temperature °F (shown for the other columns) is the evaporating temperature at the evaporator or leaving the accumulator.

The flow rate for the makeup liquid is obtained by descending down the Liquid Temp column to 95°F (condensing temperature) then moving to the right to the 20°F Evaporating Temperature column. At the point of intersection the mass flow is shown as 0.427 lb/min/ton. The total amount circulated to and from the system will be 1000 tons x 0.427 = 427 lb/min.

The flow rate for the recirculated liquid will be obtained at the intersection of the 20°F Liquid Temp line with the 20°F Evaporating Temperature column which is 0.362 lb/min/ton. The amount required for a 1000 ton load is 1000 x 0.362 = 362 lb/min. The amount to be recirculated at a 3 to 1 rate will be 362 x 3 = 1086 lb/min.

Suction Lines

Refer to Chart 2 which is Chart 1-717 Suction Lines of Bulletin RP 185. One important aspect of this chart is that the curves are based on single phase flow with no excess liquid in the vapor. For systems with two-phase flow, make selection from chart then increase pipe diameter to the next larger size to allow for the combined flow of liquid. Stoecker (1) states some designers follow this practice.

Now to the chart. Note the vertical lines in the right half of the chart. They represent different sizes of pipe with the smallest to the right. Note the numbers just to the left of each vertical pipe size. These represent varying amounts of ammonia flow in lb/min. The amounts are smaller at the bottom of each pipe size line and become larger as each pipe size increases.

The vertical lines on the left half of the chart represent evaporating temperatures with the coldest, -50°F, closest to the center. On the left side of each of these evaporator temperature lines are a series of numbers from 0.1 near the bottom to 10.0 near the top. These numbers indicate pressure drop (ΔP) in psi/100 feet of equivalent length. On the right side of those lines is another series of numbers which range from 0.1 to 8.0 for the 30°F evaporating temperature line to 0.4 to 20 for the -50°F line. These numbers represent °F of temperature loss (ΔT) per 100 ft of equivalent length. The addition of this series of numbers enables a designer to know what the temperature penalty is likely to be and to make a reselection of a more desirable size at that time.

Return to our example of 1000 tons with 362 lb/min. being vaporized at 20°F in the evaporator. The smallest line shown to handle 362 lb/min. is 5 in. Moving horizontally to the vertical 20°F evaporating temperature line, the pressure drop reading on the left side is approximately 4.8 psi/100 ft equivalent length. The temperature penalty reading on the right side is approximately 4.5°F/100 ft equivalent length. Assume that the equivalent length of this simplified header is 400 ft. The pressure drop in this 5 in line would be 4.8 x 400/100 = 19.2 psi and the temperature loss would be 4.5 x 400/100 = 18.0°F. The suction pressure would be 33.5 - 19.2 = 14.3 psig, which is much too low for the system and dictates consideration of a 6 in line. Now, going to the 6 in line with 362 lb/min of vapor and a 20°F evaporating temperature the pressure drop is 1.8 psi/100 ft. equivalent length. The total pressure drop would be 1.8 x 400/100 = 7.2 psi. Suction pressure would be 33.5 - 7.2 = 26.3 psig (12.8°F). This 7.2 psi pressure difference appears to represent a substantial potential penalty to compressor capacity and BHP/ton and warrants consideration of an 8 in line.

The ΔP of an 8 in line with a flow of 362 lb/min is 0.43 psi/100 ft equivalent length. The pressure drop is 0.43 x 400/100 = 1.72 psi. Compressor suction pressure would be 33.5 - 1.72 = 31.78 psig (18.4°F).

A comparison of the capacities and BHP/ton for the 6 in and 8 in lines can be made by referring to Chart 3: <u>Effect of Variable Suction Pressure Upon Compressor Tons & BHP at Constant Discharge Pressure</u>.

The data for these curves was obtained from rating data for a 12 cylinder reciprocating compressor. Its purpose is to show how the capacity and BHP are affected by suction pressure changes.

The suction pressure for the 6 in line is 26.3 psig (12.8°F) and for the 8 in line it is 31.78 psig (18.4°F). Entering at the 12.8°F suction temperature and moving upward to the intersection with the TONS curve the reading on the left scale is 78.5 tons for the 6 in line. Continuing up on that line to the intersection with the BHP/Ton curve and reading the scale to the right shows 1.27 BHP/ton. Repeating this process for the 18.4°F suction of the 8 in line shows capacity as 90.0 tons and 1.13 BHP/ton.

The 11.5 ton increase in capacity for the 8 in. line represents a 14.7% increase in capacity. The 0.14 BHP/ton decrease represents an 11% decrease in BHP/ton. This improvement would appear to indicate use of the 8 in. line size. Since this is a recirculating system, a 10 in. suction line should be considered to the accumulator. From the accumulator to the compressor an 8 in. line is appropriate.

It is my recommendation that all suction line sizing determinations be made using ΔT values rather than ΔP so that reference to ammonia tables for conversion of pressure to temperature is avoided.

Discharge Lines

Refer to Chart 4, which is Chart 2-717, Discharge Lines, of ASHRAE Bulletin RP 185. This chart is quite similar to that for suction lines. The various vertical lines in the right half are pipe sizes with the smallest on the right. The numbers adjacent to the left are ammonia flow in lb/min. The vertical lines in the left half represent condensing temperatures. Once again the numbers to the left indicate pressure drop in psi/100 ft equivalent length while those on the right indicate temperature difference in °F/100 ft equivalent length.

Since a line for 95°F condensing temperature is not shown, one has been drawn in where appropriate. The pressure and temperature loss numbers can be added by connecting a corresponding pressure drop for 100°F condensing with the corresponding pressure drop for 90°F condensing and marking where it crosses the 95°F line.

The discharge line in our example may be sized by initially considering both a 4 in. and 5 in. line handling 427 lb/min. Locating 427 lb/min on the 4 in. line and moving horizontally to the left to the 95°F vertical condensing line, the pressure loss is approximately 7.7 psi/100 ft equivalent length. Assuming the equivalent length is 200 ft, the pressure loss is 7.7 x 200/100 = 15.4 psi. Since the condensing pressure is established by the condensers, the compressor discharge pressure would be condensing pressure plus the pressure loss or 181.1 + 15.4 = 196.5 psig. If a 5 in. line were considered, its pressure loss would be approximately 2.3 psi/100 ft equivalent length and its total pressure loss = 2.3 x 200/100 = 4.6 psi and the discharge pressure would be 181.1 + 4.6 = 185.7 psig. With a pressure loss difference of 15.4 - 4.6 = 10.8 psi, a determination of power savings versus added cost of piping could be considered to determine the most cost effective selection.

Liquid Lines

In this instance, there are three different liquid lines which require sizing. Each has a different design criterion. So as not to confuse their sizing, we shall start with the liquid drain line from the condenser to the receiver, then liquid from receiver to the recirculating vessel, and finally the liquid being recirculated at a 3 to 1 rate to the evaporators.

Condenser to Receiver

Since this line is essentially a gravity drain line it is generally sized for a velocity of 100 ft/minute. Table 3 is Table 5, Condenser to Receiver Piping, of ASHRAE Bulletin RP 185. The vertical columns are nominal pipe sizes. The numbers for R-717 in the horizontal rows for schedule 40 or 80 pipe are maximum lb/min when velocity in the line is 100 ft/min.

The maximum flow in a 4 in. line is 326 lb/min which is less than the 427 lb/min in circulation. The capacity of a 5 in. line is 513 lb/min which is greater than the 427 lb/min and should be selected.

Receiver to Recirculating Vessel

Chart 5 is Chart 3-717, Liquid Lines, of ASHRAE Bulletin RP 185. The vertical lines in the right two-thirds represent pipe sizes, and the numbers to their left indicate flows in lb/min. The vertical lines in the left one third of the page represent liquid temperature. Along the left hand margin are numbers which represent pressure drop in psi/100 ft equivalent length. These are extended slightly upward through the vertical liquid temperature lines.

In our example, the make up liquid will be flowing in this line from the receiver at 95°F to the pump receiver operating in the range of 30 to 33 psig with its temperature approximately 20°F. Examination of the chart shows that a 2 in. or 2 1/2 in. line can be considered for use. Moving horizontally to the left from 427 lb/min on the 2 in. line to the 100°F vertical line (interpolation to 95°F is insignificant) and then slightly downward to the left, the pressure drop is approximately 2.5 psi/100 ft equivalent length. Assuming the equivalent length is 100 ft the pressure drop in the line would be 2.5 x 100/100 = 2.5 psi which is very nominal considering the 30 to 33 psig pressure in the recirculating vessel.

Recirculated Liquid to Evaporator Units

With a recirculating rate of 3 to 1 and the evaporator units vaporizing 362 lb/min, the total flow in this portion of the liquid piping will be 362 x 3 = 1086 lb/min. Reference to Chart 5 indicates 2 1/2 in., 3 in. and 4 in. pipes as being capable of handling a flow of 1086 lb/min.

The pressure drop in a 2 1/2 in. line is 6.3 lb/100 ft equivalent length. The pressure drop in a 3 in. line is approximately 2.10 lb/100 ft equivalent length. The pressure drop in a 4 in. line is approximately 0.57 lb/100 ft equivalent length.

Assuming that the overall equivalent length in this simplified header is 400 ft, then the pressure drop in the lines will be:

2 1/2 in. line is 6.3 x 400/100 = 25.2 psi
3 in. line is 2.1 x 400/100 = 8.4 psi
4 in. line is 0.57 x 400/100 = 2.3 psi

Since pumps are selected for feet of head difference between suction and discharge and other factors, it is necessary to convert the psi pressure loss into feet of head. Converting 1.0 psi of 20°F liquid into feet of head is (1.0/ 40.43) x 144 = 3.57 ft/psi.

When converting pressure drops into head loss, the head loss for each size is:

2 1/2 in. is 25.2 x 3.57 = 89.5 ft
3 in. is 8.4 x 3.57 = 30.0 ft
4 in. is 2.3 x 3.57 = 8.2 ft

It is obvious that the head loss in the 2 1/2 in. line is rather large for a pump to handle in an efficient manner, if at all. Allowance for the 30 ft head loss should be made when establishing the design conditions for the pumps used for the 3 in. main, since other factors also will enter into the pump selection procedure. The small 8.2 ft head loss for the 4 in. line should have no effect upon pump selection for the line.

Defrost Hot Gas

Sizing for these lines is done using Chart 4, Discharge Lines. The 100 tons of evaporators to be defrosted at one time is 10% of the total system tons. Strong has stated that "The time-honored rule of thumb for determining the quantity of hot gas required to properly defrost any evaporator has been to supply a defrost heating effect from the hot gas equal to three times the evaporator's cooling effect when in refrigeration"(2).

The quantity of hot gas needed to defrost 100 tons of evaporator when following the 3 to 1 rule will be 10% x 427 x 3 = 128 lb/min.

Hot gas may be supplied at discharge pressure or at some reduced pressure level to avoid waste of hot gas. This may occur from hot gas blowing through the relief regulator, to the compressor, when a defrost is completed prior to termination of the hot gas cycle. At a lower pressure, the quantity of hot gas blowing through the regulator will be less because of its lighter density. Any blow through of vapor will reduce compressor capacity since expansion of the vapor occurs when pressure is reduced in the suction line.

In this instance, we shall reduce the pressure in the defrosting hot gas header to 125 psig (74.7 °F); the relief regulator being set at 70 psig (47.3 °F). Chart 4 indicates the pressure drop (ΔP) per 100 ft equivalent length for these lines when flow is 128 lb/min as follows: 2 1/2 in. = 13.0, 3 in. = 4.0 and 4 in. = 0.7. Assuming the header has an equivalent length of 400 ft, the total pressure drop in each line will be:

2 1/2 in.: 13.0 x 400/100 = 52.0 psi
3 in.: 4.0 x 400/100 = 16.0 psi
4 in.: 0.7 x 400/100 = 2.8 psi

When the 52 psi pressure drop is subtracted from 125 psig header pressure the net pressure in the line will be 73 psig for the 2 1/2 in. line, which is too close to the pressure setting of the relief regulator and is unsatisfactory. The 3 in. line is a good choice considering that the hot gas supply pressure will be 109.0 (125 -16.0) psig and will allow for some upward adjustment of the setting of the defrost relief regulator, if necessary.

Design Criteria

In the 15 years prior to my retirement in 1977, I was responsible for the design of numerous industrial liquid recirculating refrigeration systems throughout the food industry. In order to ensure that the piping designs were effective and consistent, I developed criteria to be used for sizing all piping installed in those systems (See Table 4). Though it may seem conservative, these were large refrigerated facilities, and it was essential that they perform in accordance with design requirements. Prior to my development of these criteria, tables developed by ARI which listed maximum tons capacity for various type of lines at different pressure drops were published by various equipment manufacturers in their engineering data. More recently, ASHRAE has published maximum capacities for various line sizes and pressure drops for suction, discharge and liquid lines in tabular form. Stoecker has listed preferrable ranges of temperature loss for suction lines at 0.9 to 3.6°F and for discharge lines at 1.8 to 5.4°F(1).

High Stage Suction Line Example

A detailed high stage direct expansion suction line example is shown in Figure 3. Note that several pipe size selections have been made for most line segments and that temperature loss (ΔT) values have been used instead of pressure drop (ΔP) values per 100 ft equivalent length.

Since the final amount of temperature drop is not known until totalled, sizes of certain portions of the line may be interchanged so that a suitable temperature loss will be obtained. To better achieve this objective, temperature loss for both the smaller and larger pipe sizes are shown as well as a cumulative total for each line segment.

In this example, the equivalent length for a globe valve has been used wherever a shut off valve would be installed. These values are separately shown under the equivalent length column. Where a control valve's resistance has been given in psi by a manufacturer it has been converted to temperature loss and shown as an item to be added to the temperature loss for that segment.

Analysis of Figure 3 Example

1. Note the dramatic cumulative temperature loss of <u>8.5°F</u> for the smaller pipe sizes as compared with the <u>3.67°F</u> cumulative temperature loss for the larger pipe size selection.
2. With an evaporating temperature of 20°F, suction temperature at the compressor for the larger pipe sizes would be 20°F minus 3.67° or <u>16.33°F</u> (29.7 psig) versus 20°F minus 8.5° or <u>11.5°F</u> (25.2 psig) for the smaller pipe sizes.
3. The effect of this difference upon compressor capacity also is very dramatic. Reference to Chart 3 indicates a capacity of 76 tons and 1.31 BHP/Ton at a suction temperature of 11.5°F for the smaller pipe sizes and a capacity of 85.7 tons and 1.18 BHP/Ton at a suction temperature of 16.3°F for the larger pipe sizes. These differences represent a <u>12.7% increase in capacity</u> and a <u>10.0% decrease</u> in BHP when larger pipe sizes are considered.
4. Without question, the larger pipe sizes should be selected for use. Since the evaporators are direct expansion units, pipe sizes will remain as shown in the example, rather than being enlarged for liquid recirculation.
5. For a given flow, pressure drop will decrease as line size is increased.
6. There is a much greater pressure drop per 100 ft equivalent length for valves than for pipe.

Low Stage Suction Line Example

A detailed low temperature suction line example for a continuous blast freezer supplied by pump recirculated liquid is shown in Figure 4. In this example, a comparison is made on the effect that a globe valve has upon temperature loss as compared with that of an angle or 45° Y valve. Two pipe sizes are indicated in most segments. However, since the ΔT per 100 ft equivalent length was excessive for the smaller line size, only the larger was used. CG indicates a control valve group in which two shut off valves and a gas powered check valve are installed in the suction header.

Analysis of Figure 4 Example

1. In those segments where only pipe is involved temperature loss is small.
2. Note that the equivalent length for a globe valve is much greater than for an angle or 45° Y valve of the same size. (See Table 7 for equivalent length of valves.)
3. The effect that large size valves have upon temperature loss is substantial.
4. The effect that a control valve having a pressure drop of 1/4 psi has upon temperature loss is substantial. The temperature loss would be twice as large if pressure drop was increased to 1/2 psi.
5. Reference to Chart 6, <u>Effect of Variable Suction Pressure upon Booster Tons and BHP at Constant Discharge Pressure</u>, indicates that the use of angle or 45° Y valves, as compared with globe valves, will result in an increase of 7.7% in compressor capacity and a decrease of 4.2% in BHP/ton.
6. For a liquid recirculating system, consider using pipes one size larger than shown in the example.

Low Temperature Liquid Line Example

A detailed liquid line example is shown in Figure 5. This selection is made for the liquid supplied to the blast freezer suction line example shown in Figure 4. You will note that we have retained the same identification for each segment, but the direction of flow is reversed. The pressure loss is expressed in psi rather than °F since we are more interested in psi and its conversion to feet of head. Since maximum liquid pressure occurs at the pump discharge, the pressure available at the farthest unit in segment A will be the total of all pressure losses subtracted from discharge pressure.

Trapped Suction Line Risers

One source of substantial pressure loss in the suction line of a recirculating system occurs when the line rises (is trapped) after leaving the evaporator outlet. Since the amount of liquid flow in the suction line from any evaporator is only an estimate, the manner in which it passes upward can vary in a number of forms depending upon its loading. Our concern involves its state when flow is occurring under design load conditions.

Though there may be some relationship between flow to pipe size for minimum pressure drop in a riser, the only published information on this subject of which I am familiar is by Richards(3) and Stoecker(1). Stoecker references Richards and uses some of his findings.

Richards predicts temperature loss in 10 ft risers of various size at -40°F with a 5 to 1 recirculating rate. For pipe sizes between 2 1/2 in. and 6 in., the range of minimum temperature loss is between 2.7°F for the smaller size to 3.5°F for the 6 in. size.

To determine the effect a 3°F temperature loss will have upon compressor capacity at -40°F and -43°F, refer to Chart 6. Capacity at -40°F is 48 tons and 1.39 BHP/ton while capacity at -43°F suction is 43.5 tons and 1.51 BHP/ton. This penalty will cause a 9.4% reduction in compressor capacity and an increase of 8.6% in BHP.

The important aspect of this matter is the fact that a substantial temperature penalty occurs whenever a suction line containing recirculated liquid rises or is trapped. The loss will increase as line size or the height of the rise increases. Fortunately, this is the only penalty which a designer has the ability to entirely avoid. This is possible by removing the liquid from the suction line before it rises by means of a liquid transfer system which is located below the unit.

Pressure Loss Due to Height

On occasion, the routing of liquid lines involves their placement on roofs or at high elevations in a building. When this occurs liquid pressure in the line will be reduced by the head loss which results. The amount of this loss is readily obtained by reference to Chart 7 which is Chart 7-717, <u>Pressure Loss from Increase in Height</u>, of Bulletin RP 185.

For example, the pressure loss in a line containing 50°F liquid that rises 30 ft is 8.3 psi. Enter the chart at 30 ft. Move up vertically to the intersection of the horizonal 50°F line. At that point, interpolation between the 8 and 9 psi lines will show pressure drop to be approximately 8.3 psi.

Other Reference Data

For reference purposes there is included from Bulletin RP 185 the following:

1. For pipe fittings - Table 5, <u>Loss in Equivalent Feet of Pipe</u>.
2. For valves - Table 6, <u>Loss in Equivalent Feet of Pipe</u>.
3. Chart 8-717, <u>Gage Pressure at Various Altitudes</u>.

Before concluding this portion of the paper, recognition and comment is in order on one aspect of the Suction Line chart. As that chart was prepared on the basis of all liquid supplied to the evaporator being vaporized it also allowed for 10°F of superheat being added to the vapor returning to the compressor. As a result, the volume of the vapor is increased by approximately 2.75%. Table 2 on page 33 of ASHRAE RP185 indicates that a 10°F reduction in superheat will result in pressure drop being reduced by 2.6%. In this instance, we are dealing with a matter in which knowledge of the actual flow of ammonia is only an estimate and subject to much greater variation than 2.6%. In any event that superheat allowance will provide a 2.6% reduction in pressure drop and render the chart conservative.

Piping Arrangement and Practices

The arrangement of a well conceived piping system involves considerably more engineering effort than that required to provide a schematic flow diagram to a piping contractor. Though any number of arrangements are possible it is important that each:

1. Provides convenient access to items related to system operation such as shut off valves, float valves and chambers, pumps, liquid coolers, condensers etc.
2. Provides convenient access to operating equipment for service or maintenance.
3. Is easily understood and not confusing.
4. Is energy efficient.
5. Avoids interferences or "fixes" where omissions or errors may have caused a problem.

There follows a listing of factors which warrant consideration when arranging piping in specific areas or applications.

In Refrigerated and Non-Refrigerated Areas

Piping Headers

1. Allow ample space between lines.
2. Locate suction lines where valves can be installed with stems horizontal.
3. Locate over product storage areas or close to walls.
4. Install valves for isolation of branch headers where they connect into main header.
5. Avoid locations that will interfere with air distribution.
6. Avoid locating piping over processing equipment.
7. Assure piping is well protected from below.
8. Assure piping is properly braced and supported.

Suction Headers

1. Plan routing.
2. Locate end of each branch at highest point for units located farthest from engine room.
3. Pitch continuously down towards machine room.
4. Branch headers must connect into main header from above, never from below.
5. Avoid use of globe valves in suction lines whenever possible to reduce pressure drop.
6. Do not trap to avoid interference with other piping or equipment.
7. For major rise to another floor or roof, do not trap. Allow suction from unit to drain into an accumulator and transfer pump or a liquid return system located below evaporator unit.

Evaporator Piping

1. Plan piping arrangement from unit to control group to headers.
2. Use shut off valves in all lines between control valves and unit. Locate them so that they are accessible in one location by ladder or fork lift from floor.
3. Defrost relief drain line to take off from underside or end of lower coil connection.

Control Valve Group

1. Locate all valves adjacent to each other where accessible from one location for operation and maintenance.
2. Avoid the use of strainers, and their increased temperature loss penalty, in front of any low temperature suction control valve.
3. Avoid locating over doors.
4. Provide adequate spacing between lines for insulation on valves and accessibility for service.
5. Provide one pressure gauge which will show pressure in the coil at all times.
6. Provide a pressure gauge upstream of liquid throttling device.
7. Restricted spaces may require detailed layout of all valves and piping to avoid interferences and provide access for maintenance.

Engine Room

General

1. Establish equipment locations based upon a planned header arrangement.
2. Establish elevations for major piping headers - HBS, LBS, LLBS, CD, BD.
3. Provide double offsets in suction and discharge piping from headers to compressors to provide flexibility to piping.
4. Locate compressor shutoff valves where accessible.
5. Provide convenient access and work space needed to operate and service items such as: shutoff valves, check valves, control valves, float devices, strainers, pumps, liquid coolers, etc.
6. Avoid interferences or difficult access after installation of insulation.

Individual Items

1. Provide main line liquid solenoid at receiver to stop liquid flow upon emergency shutdown.
2. Feed makeup liquid into vessels - not piping.
3. Combine branch lines into manifolds where possible.

4. Provide stubs or valves for future compressor or condenser additions, if likely.
5. Cross connect booster suctions from different low stage systems for flexibility.
6. Provide starting by-pass for single stage operation of two stage system.
7. Provide suction header drain valves on underside near end of header.
8. Make compressor suction connection to header from above - never from below.
9. Make compressor discharge into header from above at 45° angle.
10. Ensure a check valve is installed in discharge line from each compressor.
11. Provide shut off valves at every connection on every vessel except for safety relief valve.
12. Install valves on vessels with pressure under seat.
13. Never locate a union between a valve and vessel.
14. Provide pump out or drain valves on underside of liquid lines at control or check valves and underside of liquid line to pump suction connection.
15. Provide hydrostatic relief valve where isolation of a liquid line could result in liquid lockup.
16. Locate float columns so that: level eyes are readily visible, float switches can be observed and serviced, and float chambers can be raised or lowered.
17. Provide pump out connection on float columns.
18. Locate liquid level control floats and valves where accessible for service and pressure gauges are readily visible.
19. Provide hand expansion by-pass around all liquid control valves.
20. For low temperature applications, be aware of the effect upon compressor capacity that results from excessive temperature loss in suction lines and make sizing selections on a conservative basis.

Electric Panels

1. Never locate refrigerant or water lines over any switchgear, motor control center, distribution and control panels.

Closing Comment

The recommendations contained herein represent a condensation of the knowledge gained from many years of experience in designing, starting up and troubleshooting numerous refrigeration systems. Of course, there are other ways to do some of the things described with satisfactory results.

The author hopes that system designers, engineers, plant operating supervisors and personnel will find some of this information helpful and beneficial in the course of their work, and that it will contribute to their becoming more knowledgeable and valuable individuals.

References

1. Stoecker, W. F. Industrial Refrigeration, 1988. p. 264 and p. 271-272.
2. Strong, A. P. "Hot Gas Defrost: A one, A more, A time," *Proc. IIAR Ann. Mtg.*, 1984, p 185K.
3. Richards, William, V. "Piping Is Piping ... Or Is It?," *Proc. IIAR Ann. Mtg.*, March 1982, p 20.

Table 1: Pressure-Volume Variation Due to Temperature Change

Temperature °F	Pressure psig	Volume (V_g) ft^3/lb
96	184.2	1.510
95	181.1	1.534
$\Delta T=1°F$	$\Delta P=3.1$ psi	
21	34.6	5.789
20	33.5	5.910
$\Delta T=1°F$	$\Delta P=1.1$ psi	
1	16.5	8.912
0	15.7	9.116
$\Delta T=1°F$	$\Delta P=0.8$ psi	
-19	4.1	14.32
-20	3.6	14.68
$\Delta T=1°F$	$\Delta P=0.5$ psi	
-39	10.72	24.18
-40	10.41	24.86
$\Delta T=1°F$	$\Delta P=0.31$ psi	

Table 2: Ammonia Flow Rate, lb/min/ton

Courtesy of American Society of Heating, Refrigerating and Air-Conditioning Engineers. Reprinted from "Refrigerant Line Sizing," ASHRAE Research Project 185.

Liquid Temp F*	Evaporating Temperature, F**								
	-50	-40	-30	-20	-10	0	10	20	30
120	0.482	0.478	0.473	0.469	0.466	0.462	0.459	0.456	0.453
115	0.475	0.471	0.467	0.463	0.459	0.456	0.453	0.450	0.447
110	0.468	0.464	0.460	0.457	0.453	0.450	0.447	0.444	0.441
105	0.462	0.458	0.454	0.451	0.477	0.444	0.441	0.438	0.435
100	0.456	0.452	0.448	0.445	0.431	0.438	0.435	0.432	0.430
95	0.450	0.446	0.442	0.439	0.436	0.433	0.430	0.427	0.425
90	0.444	0.440	0.437	0.433	0.430	0.427	0.424	0.422	0.419
85	0.439	0.435	0.431	0.428	0.425	0.422	0.419	0.417	0.414
80	0.433	0.430	0.426	0.423	0.420	0.417	0.414	0.412	0.409
75	0.428	0.424	0.421	0.418	0.415	0.412	0.409	0.407	0.405
70	0.423	0.419	0.416	0.413	0.410	0.407	0.404	0.402	0.400
65	0.418	0.414	0.411	0.408	0.405	0.402	0.400	0.398	0.395
60	0.413	0.409	0.406	0.403	0.400	0.398	0.395	0.393	0.391
55	0.408	0.404	0.401	0.398	0.396	0.393	0.391	0.389	0.387
50	0.403	0.400	0.397	0.394	0.392	0.389	0.387	0.385	0.383
45	0.399	0.396	0.393	0.390	0.387	0.385	0.383	0.381	0.379
40	0.395	0.392	0.389	0.386	0.383	0.381	0.379	0.377	0.375
35	0.390	0.387	0.384	0.382	0.379	0.377	0.375	0.373	0.371
30	0.386	0.383	0.380	0.378	0.375	0.373	0.371	0.369	0.367
25	0.382	0.379	0.376	0.374	0.371	0.369	0.367	0.365	
20	0.378	0.375	0.373	0.370	0.368	0.366	0.364	0.362	
15	0.374	0.371	0.369	0.366	0.364	0.362	0.360		
10	0.370	0.368	0.365	0.363	0.361	0.358	0.356		
5	0.366	0.364	0.361	0.359	0.357	0.355			
0	0.363	0.360	0.358	0.356	0.354	0.352			
-5	0.359	0.357	0.354	0.352	0.350				
-10	0.356	0.354	0.351	0.349	0.347				
-15	0.352	0.350	0.348	0.346					
-20	0.349	0.347	0.345	0.343					
-25	0.346	0.344	0.342						
-30	0.343	0.341	0.339						
-35	0.340	0.338							
-40	0.337	0.335							
-45	0.334								
-50	0.331								

* Temperature entering expansion device
** Saturation temperature leaving evaporator or accumulator
***Tons x pounds per (minute, ton) = pounds per minute

Table 3: Condenser to Receiver Line Sizing, 100 ft/min, 90°F liquid temperature

Courtesy of American Society of Heating, Refrigerating and Air-Conditioning Engineers. Reprinted from "Refrigerant Line Sizing," ASHRAE Research Project 185.

	Sch	Nominal Pipe Size ¾	1	1¼	1½	2	2½	3	4	5	6
		Flow Rate, pounds per minute									
R-12	40	30	48	83	113	179	266	411	708	1112	1606
	80	24	40	71	98	164	236	367	639	1012	1450
R-22	40	34	56	96	131	207	308	476	819	1287	1858
	80	28	46	83	114	190	273	425	740	1170	1677
R-502	40	26	42	73	100	158	234	362	623	980	1415
	80	21	35	63	87	145	208	323	563	891	1276
R-717	40					83	123	189	326	513	714
	80	11	18	33	45	76	109	169	295	466	668

Table 4: Design Criteria

H.B. Suction	1 psi per 750 ft equivalent length
L.B. Suction	1 psi per 1000 ft equivalent length
L.L.B. Suction	1 psi per 1000 ft equivalent length
Booster Discharge	1 psi per 200 ft equivalent length
Hot Gas to Condenser	1 psi per 200 ft equivalent length
Hot Gas to Plant	1 psi per 200 ft equivalent length
Liquid Cond-Rec	100 ft/min velocity
Liquid to Plant	1 psi per 200 ft equivalent length
Oil Drain	3/4 in

Table 5: Loss in Equivalent Feet of Pipe

Courtesy of American Society of Heating, Refrigerating and Air-Conditioning Engineers. Reprinted from "Refrigerant Line Sizing," ASHRAE Research Project 185.

Nominal Pipe Size (in.)	Pipe Schedule	Smooth Bend Elbows: 90° Std*	90° Long Rad. †	90° Street*	45° Std*	45° Street*	180° Std*	Smooth Bend Tees: Flow Through Branch	Straight-Through Flow: No Reduction	Reduced ¼	Reduced ½
Screwed Fittings[1]											
½	40	1.6	1.0	2.5	0.8	1.3	2.5	3.0	1.0	1.4	1.6
	80	0.8	0.5	1.3	0.4	0.7	1.3	1.6	0.5	0.7	0.8
¾	40	2.0	1.3	3.2	0.9	1.6	3.2	4.0	1.4	1.9	2.0
	80	1.2	0.8	1.9	0.5	1.0	1.9	2.4	0.8	1.1	1.2
1	40	2.6	1.7	4.1	1.3	2.1	4.1	5.0	1.7	2.3	2.6
	80	1.7	1.1	2.6	0.8	1.3	2.6	3.2	1.1	1.5	1.7
1¼	40	3.2	2.3	5.6	1.7	3.0	5.6	7.0	2.3	3.1	3.3
	80	2.3	1.6	3.7	1.2	2.1	3.9	4.8	1.6	2.1	2.3
1½	40	4.0	2.6	6.3	2.1	3.4	6.3	8.0	2.6	3.7	4.0
	80	2.8	1.8	4.5	1.5	2.4	4.5	5.7	1.8	2.6	2.8
2	40	5.0	3.3	8.2	2.6	4.5	8.2	10	3.3	4.7	5.0
	80	3.2	2.4	6.0	1.9	3.3	6.0	7.3	2.4	3.4	3.7
Welded or Flanged Fittings[11]											
1	40	1.8	1.2		0.8		2.4	5.2	1.6	0.8	1.2
	80	1.2	0.8		0.5		1.5	3.3	1.0	0.5	0.8
1¼	40	2.3	1.6		1.1		3.1	7.1	2.0	1.1	1.6
	80	1.6	1.1		0.7		2.1	4.9	1.4	0.7	1.1
1½	40	2.6	1.8		1.2		3.5	8.4	2.0	1.2	1.8
	80	1.8	1.3		0.9		2.5	6.0	1.4	0.9	1.3
2	40	3.4	2.3		1.5		4.6	10.5	2.5	1.5	2.5
	80	2.5	1.7		1.1		3.3	7.7	1.8	1.1	1.7
2½	40	4.2	2.7		1.8		5.6	13	2.9	1.8	2.7
	80	3.1	2.0		1.3		4.1	10	2.0	1.3	2.0
3	40	5.3	3.4		2.3		7.1	16	3.6	2.3	3.4
	80	4.0	2.6		1.7		5.4	12	2.7	1.7	2.6
4	40	7.2	4.5		3.0		10	22	4.5	3.0	4.5
	80	5.6	3.5		2.3		7.8	17	3.4	2.3	3.5
5	40	9.2	5.7		3.8		12	27	5.1	3.8	9.2
	80	7.3	4.5		3.0		9.5	21	4.0	3.0	7.3
6	40	11	6.8		4.6		15	33	6.1	4.6	11
	80	8.6	5.3		3.5		12	26	4.8	3.5	8.6
8	30	16	10		6.7		21	47	7.5	6.7	16
	40	15	9		6		20	44	7.1	6.0	15
10	30	19	12		8		25	60	9.2	8.0	19
	40	18	11		7		24	56	8.7	7.0	18
	20	25	16		11		34	77	12	11	25
12	30	23	15		10		32	72	11	10	23
	40	22	14		9		30	68	10	9	22

*R/D approximately equal to 1
†R/D approximately equal to 1.5

Table 6: Valve Losses in Equivalent Feet of Pipe

Courtesy of American Society of Heating, Refrigerating and Air-Conditioning Engineers. Reprinted from "Refrigerant Line Sizing," ASHRAE Research Project 185.

Pipe Size In.	Sch.	Globe	60°Y	45°Y	Angle	Ball*	Butter-fly*	Swing Check	Lift Check
½	40	18	9	7	7	4		6	
	80	9	4	3	3	2			
¾	40	22	11	9	9	6		8	
	80	13	7	5	5	4		5	
1	40	29	15	12	12	3		10	
	80	19	10	8	8	2		6	
1¼	40	38	20	15	15	6		14	
	80	26	14	10	10	4		10	
1½	40	43	24	18	18	5		16	Globe and vertical lift same as globe valve
	80	31	17	13	13	3		11	
2	40	55	30	24	24	11		20	
	80	40	22	18	18	8		15	
2½	40	69	35	29	29	9		25	
	80	51	26	21	21	7**		19	
3	40	84	43	35	35	10	26	30	
	80	64	33	27	27	7	20	23	
4	40	120	58	47	47	8	28	40	
	80	94	45	37	37	6	22	31	
5	40	140	71	58	58	19	27	50	
	80	111	56	46	46	15**	22**	40	
6	40	170	88	70	70	41	34	60	Angle lift same as angle valve
	80	133	69	55	55	32	26	47	
8	30	233	122	90	90	44	44	85	
	40	220	115	85	85	42	42	80	
10	30	297	154	111	111	60	58	106	
	40	280	145	105	105	57	55	100	
12	20	362	187	147	147	59	69	136	
	30	339	175	138	138	55	65	128	
	40	320	165	130	130	52	61	120	

*Flow data for ball and butterfly valves vary widely, even from the same manufacturer. Equivalent lengths shown above should not be used without corrections from Table 9.
**Ball and butterfly valves may not be available from some manufacturers in these sizes.

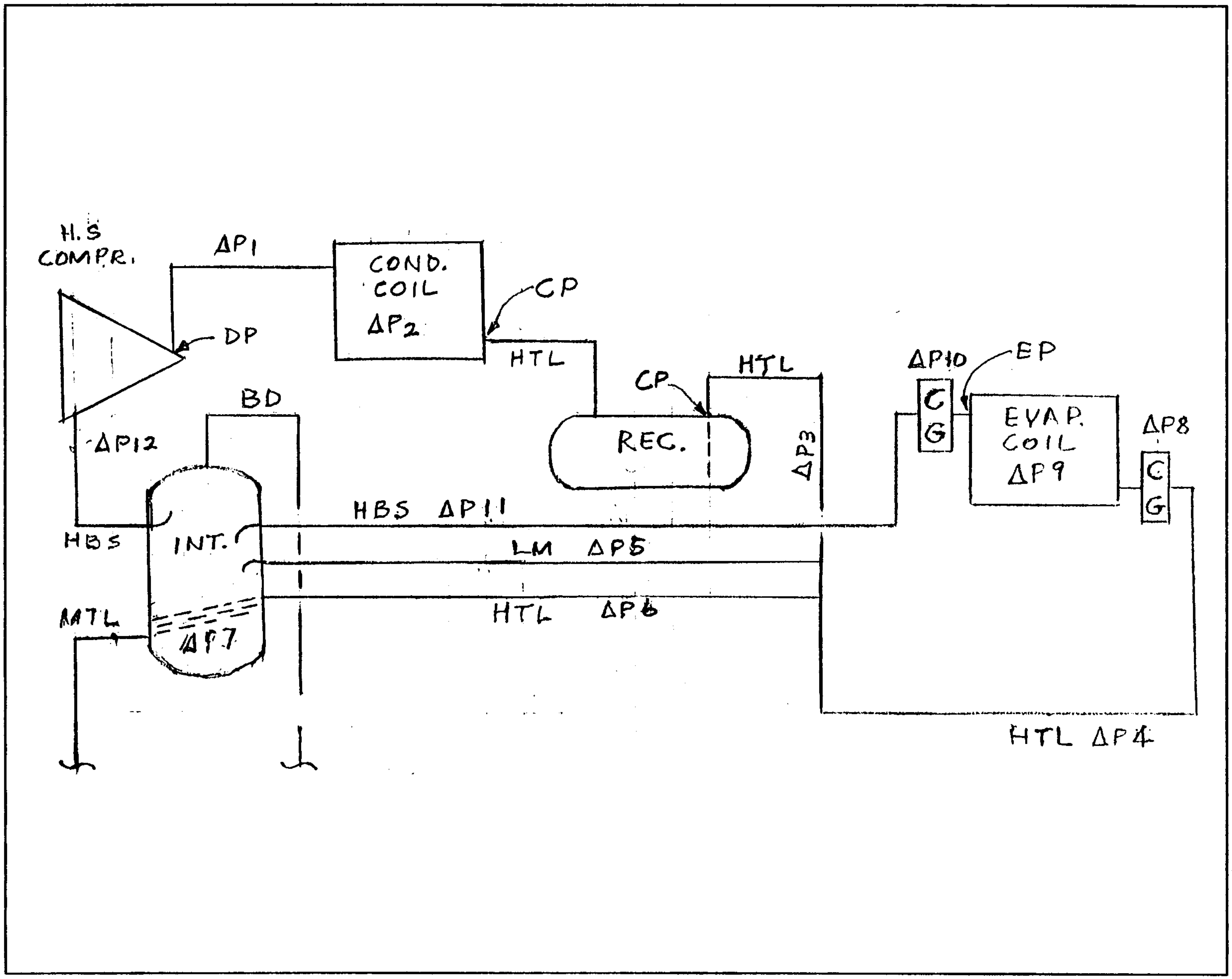


Figure 1: High Stage Schematic

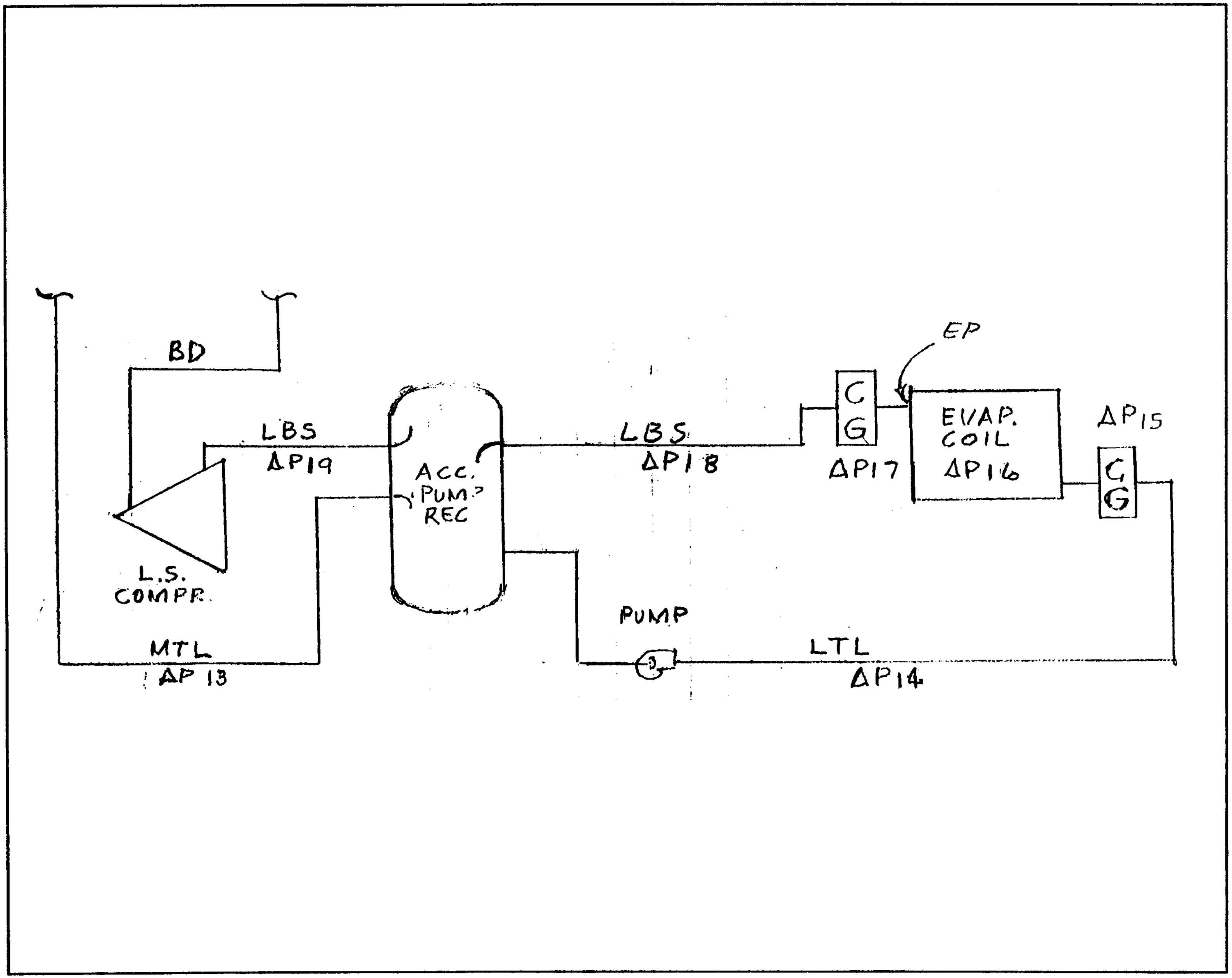
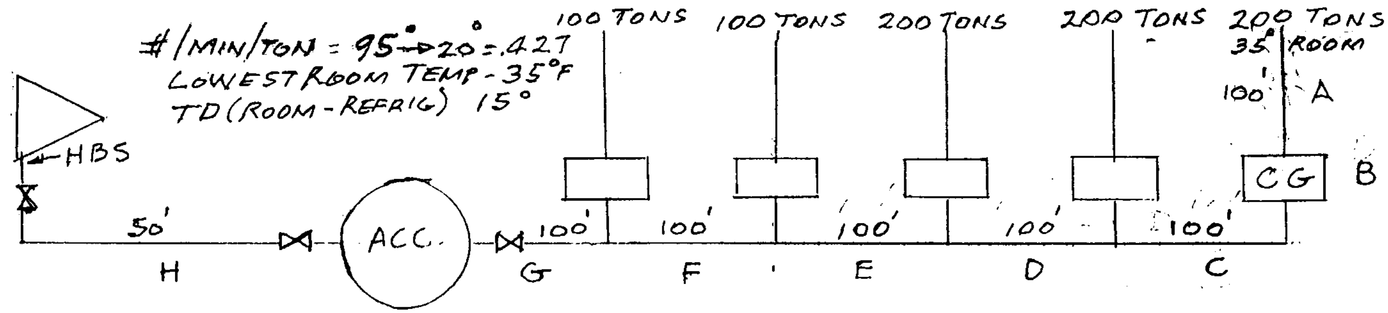


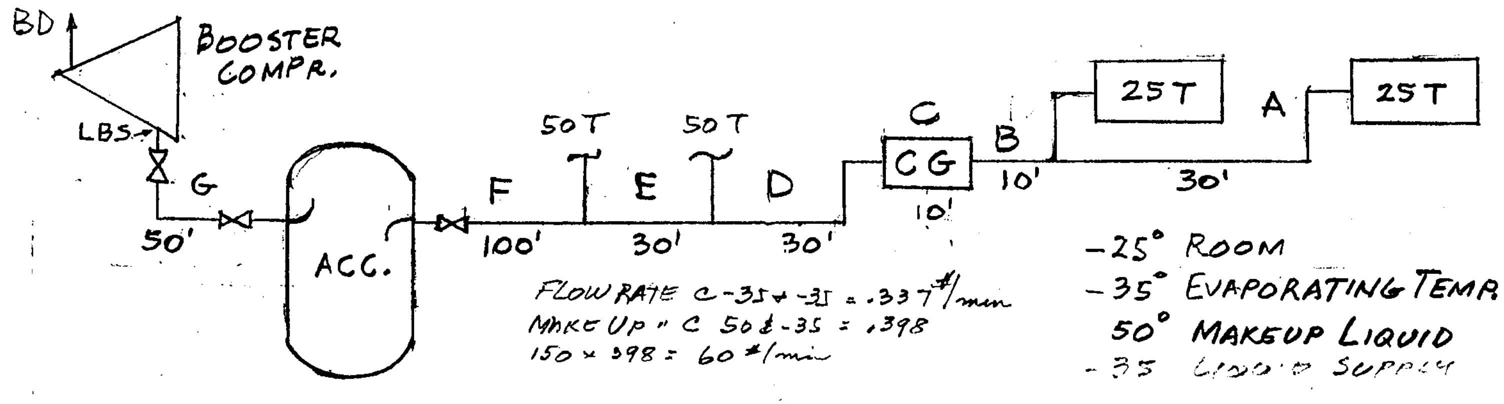
Figure 2: Low Stage Schematic



SEG-MENT	TONS BR.	TONS HDR.	#/MIN. BR.	#/MIN. HDR.	LINE SIZE	ΔT	EQUIV. LGTH.	TEMP. DROP SMALL	TEMP. DROP LARGE	CUM. TEMP DROP SMALL	CUM. TEMP DROP LARGE
A	–	200	–	85	4 5	.80 .27	100'	.80	.27	.80	.27
B	–	200	–	85	4 5	.80 .27	240' + .46 ΔT 280'	2.38	1.22	3.18	1.49
C	–	200	–	85	5	.27	100'	.27	.27	3.45	1.76
D	200	400	85	170	5 6	.90 .38	100'	.90	.38	4.35	2.14
E	200	600	85	255	6 8	.85 .22	100'	.85	.22	5.20	2.36
F	100	700	43	298	8	.27	100'	.27	.27	5.47	2.63
G	100	800	43	341	8 10	.37 .105	100'+220' 100'+280'	1.22	.40	6.69	3.03
H	–	800	–	341	8 10	.37 .105	50'+440' 50'+560'	1.81	.64	8.50	3.67

HBS COMPRESSOR SUCTION 20°F-CTD 11.5°F 16.3°F

Figure 3: Single Stage Suction Line Sizing



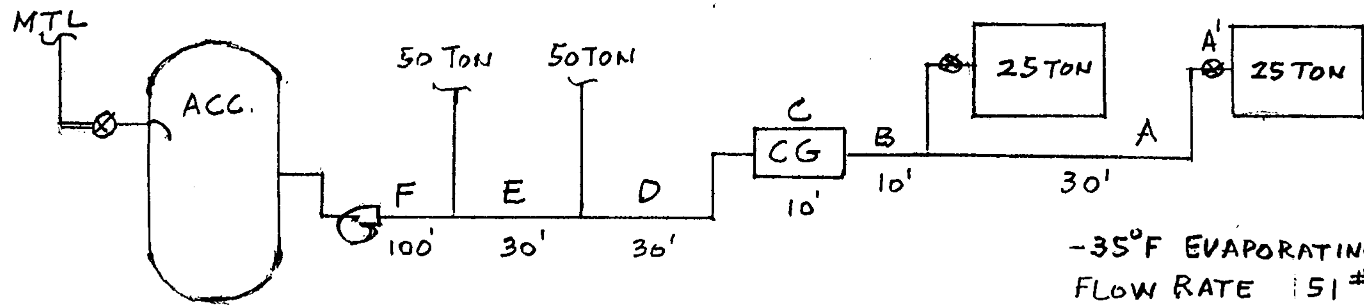
SEG-MENT	TONS		#/MIN.		LINE SIZE	ΔT	EQUIV. LGTH			TEMP. LOSS	
	BR.	HDR	BR.	HDR.			PIPE	G.V.	A.V.	GLOBE	ANGLE
A	25	25	8.5	8.5	2½ 3	1.40 .40	30	69 84	29 35	– .46	– .26
B	25	50	8.5	17	3 4	1.60 .40	10	–	–	– .04	– .04
C	–	50	–	17	4	.40	10	240 CK-2	94 1/4#	1.00 .81	.42 .81
D	–	50	–	17	4	.40	30'	–	–	.16	.16
E	50	100	17	34	4 5	1.40 .50	30	– –	– –	– .15	– .15
F	50	150	17	51	5 6	1.00 .40	100	140 170	58 70	– 1.08	– .68
G	–	150	9	60	6	.60	50	340	240	2.34	1.74

TOTAL 6.04 4.26

(LBS) COMPR. SUCTION =(-35-TEMP. LOSS) -41.0° -39.3°

Figure 4: Low Stage Suction Line Sizing

Figure 5: Low Stage Liquid Line Sizing



-35°F EVAPORATING TEMP.
FLOW RATE 51 #/MIN.
RECIRC. RATE 153 #/MIN.
⊗ THROTTLING DEVICE

SEGMENT	TONS		#/MIN.		LINE SIZE	ΔP PSI/100'	EQUIV. LENGTH	PRESS. LOSS	
	BR.	HDR.	BR.	HDR.				SMALL PIPES	LARGE PIPES
F	–	150	–	153	1½" 2"	2.0 0.37	100' +43' 1/4# +55'	3.11	1.03
E	50	100	51	102	1¼" 1½"	2.0 0.87	30'	0.60	0.26
D	50	50	51	51	1" 1¼"	2.25 0.53	30'	0.67	0.16
C	–	50	–	51	1" 1¼"	2.25 0.53	10' + 58' + ½# + 76'	2.02	0.96
B	50	50	–	51	1" 1¼"	2.25 0.53	10'	0.22	0.05
A	–	25	–	25.5	1"	0.60	30'+29'	0.35	0.35
A'	–	25	–	25.5	1"	5 PSI	–	5.00	5.00

STATIC HEAD TO UNIT ELEV. 20' = 6.0 PSI	6.0	6.0
TOTAL PSI	17.97	13.97
TOTAL FT. HD. = PSI × 3.35 =	60.0	47.0

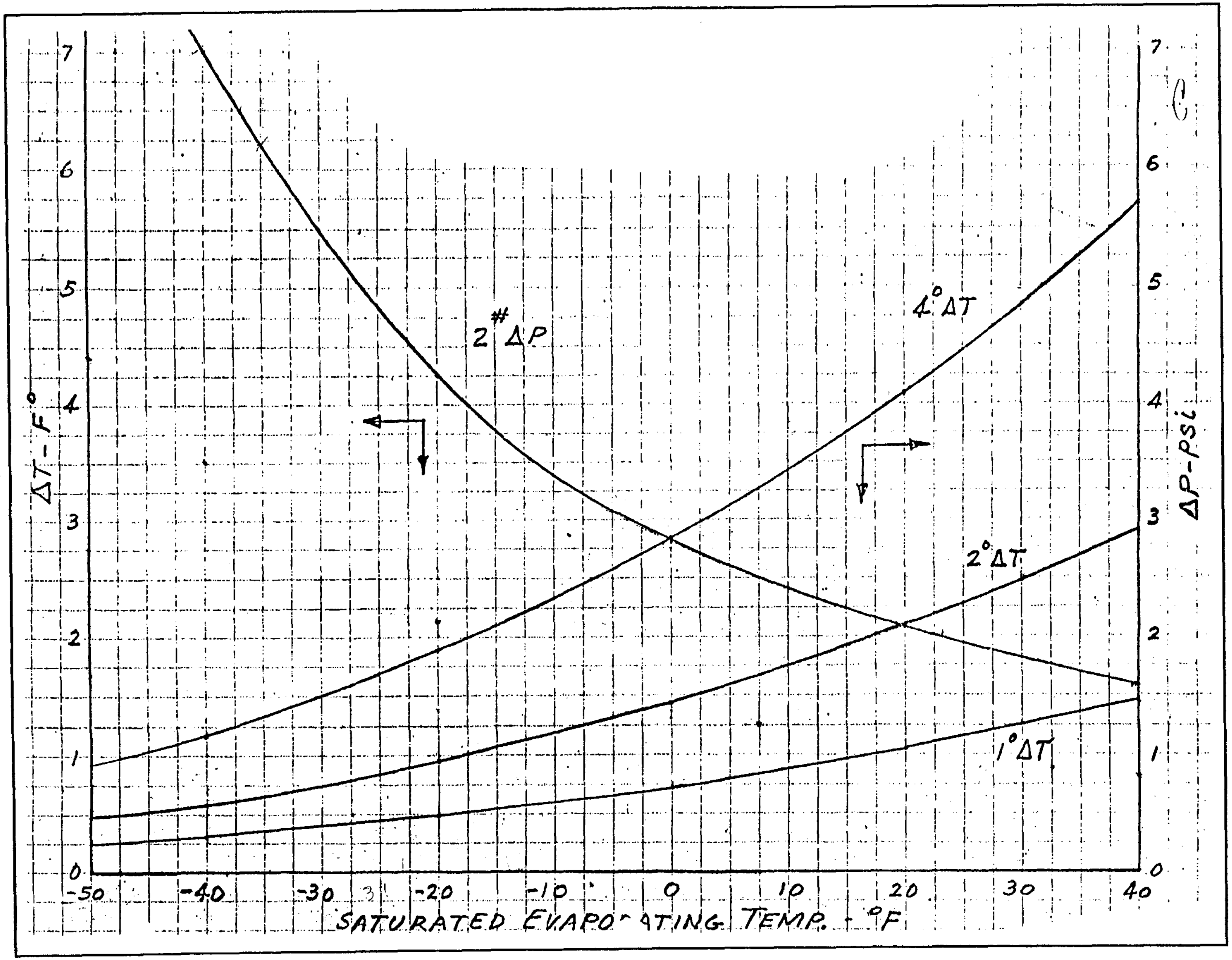
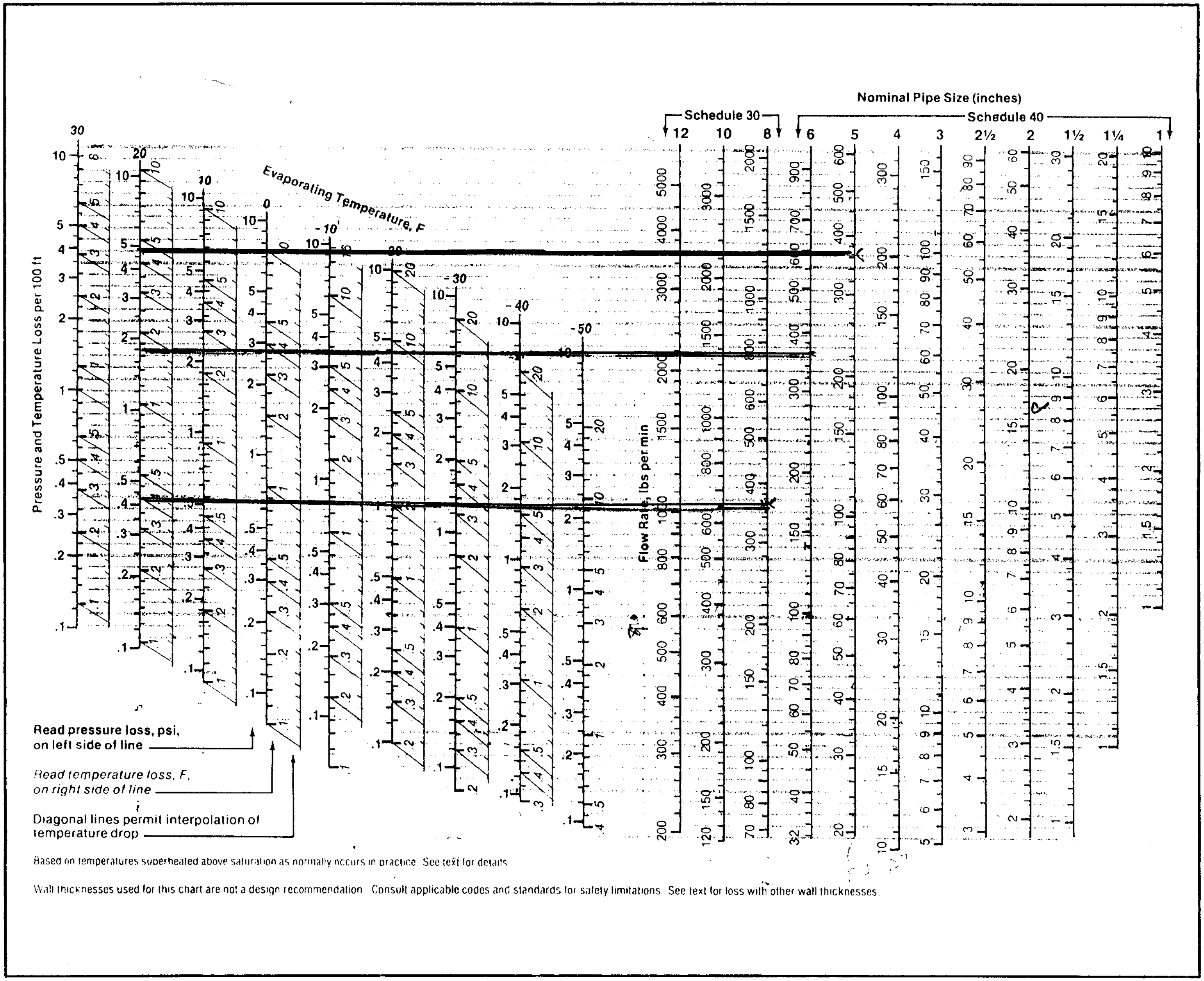


Chart 1: Ammonia Pressure/Temperature Penalty



Courtesy of American Society of Heating, Refrigerating and Air-Conditioning Engineers. Reprinted from "Refrigerant Line Sizing," ASHRAE Research Project 185.

Chart 2: Suction Lines, Pressure and Temperature Losses

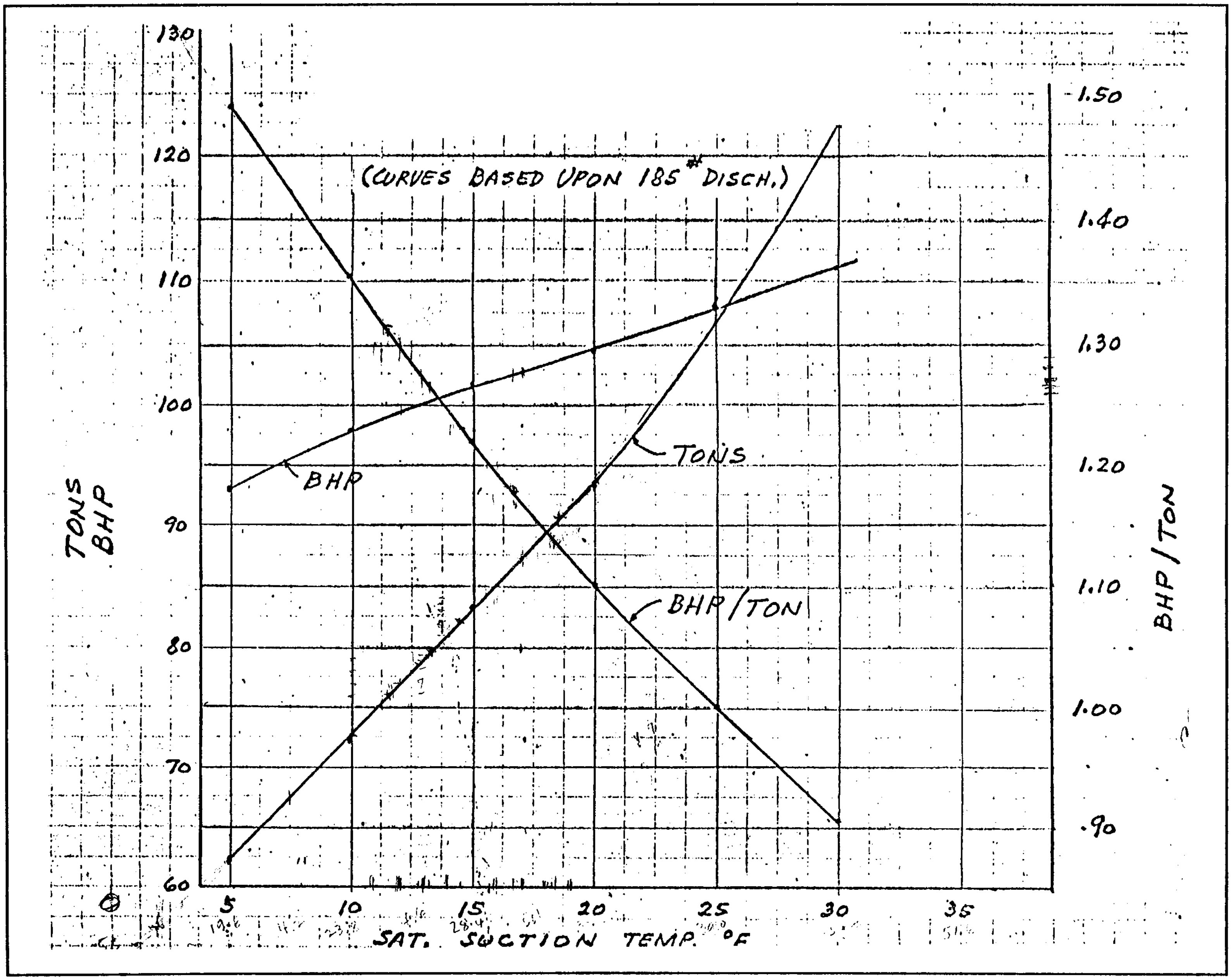
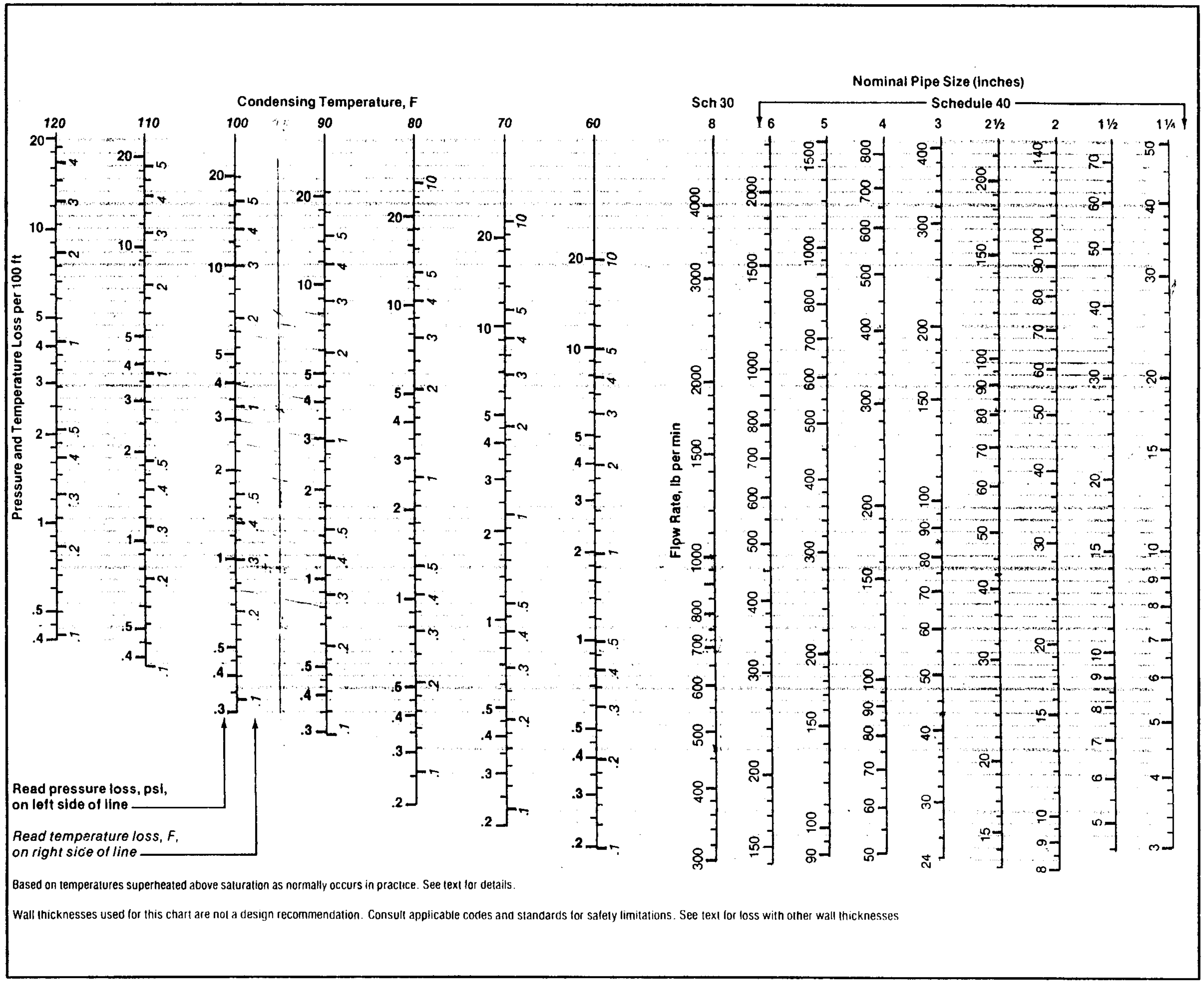
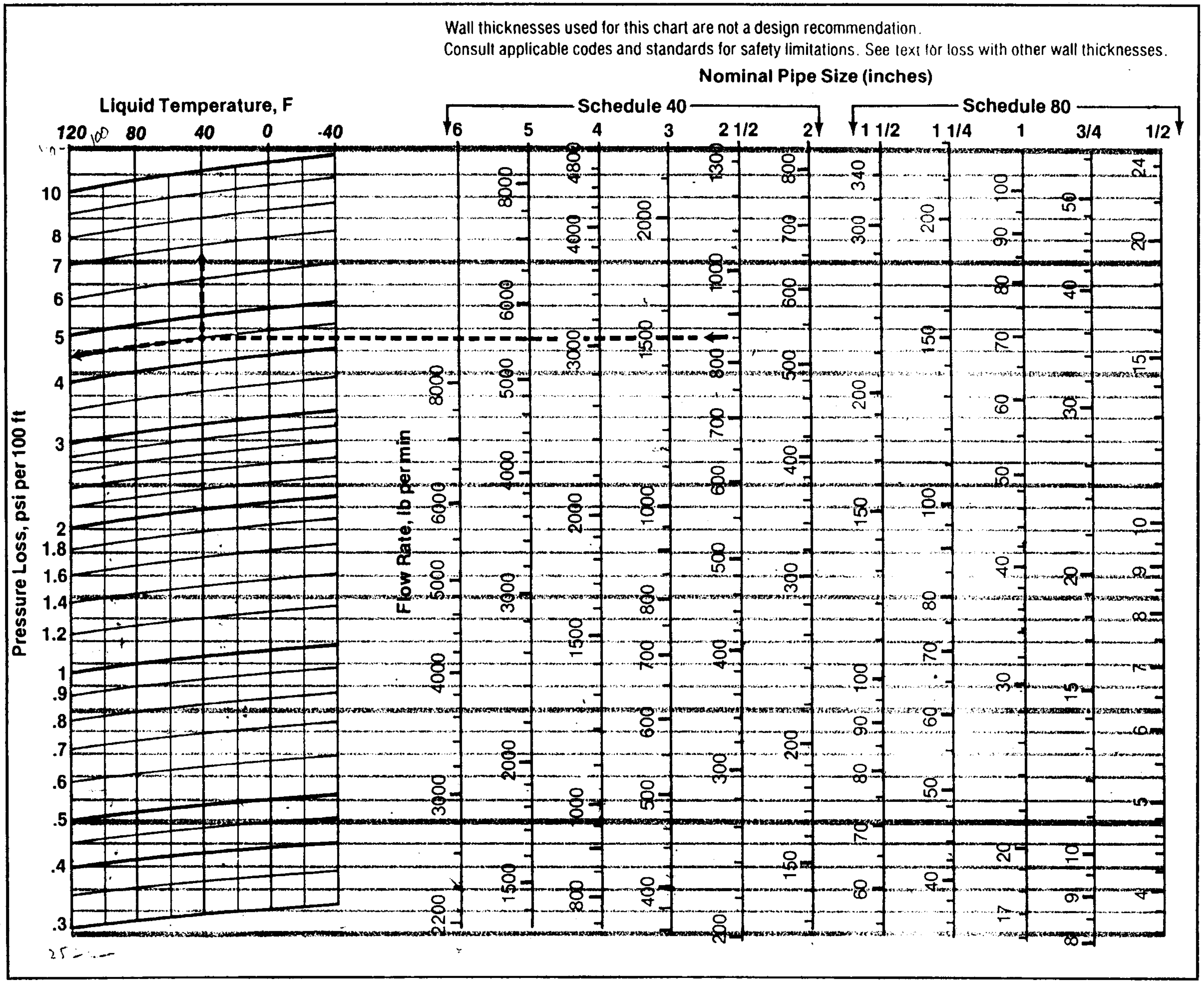


Chart 3: Effect of Variable Suction Pressure Upon Compressor Tons & BHP at Constant Discharge Pressure



Courtesy of American Society of Heating, Refrigerating and Air-Conditioning Engineers. Reprinted from "Refrigerant Line Sizing," ASHRAE Research Project 185.

Chart 4: Discharge Lines, Pressure and Temperature Losses



Courtesy of American Society of Heating, Refrigerating and Air-Conditioning Engineers. Reprinted from "Refrigerant Line Sizing," ASHRAE Research Project 185.

Chart 5: Liquid Lines, Pressure Loss

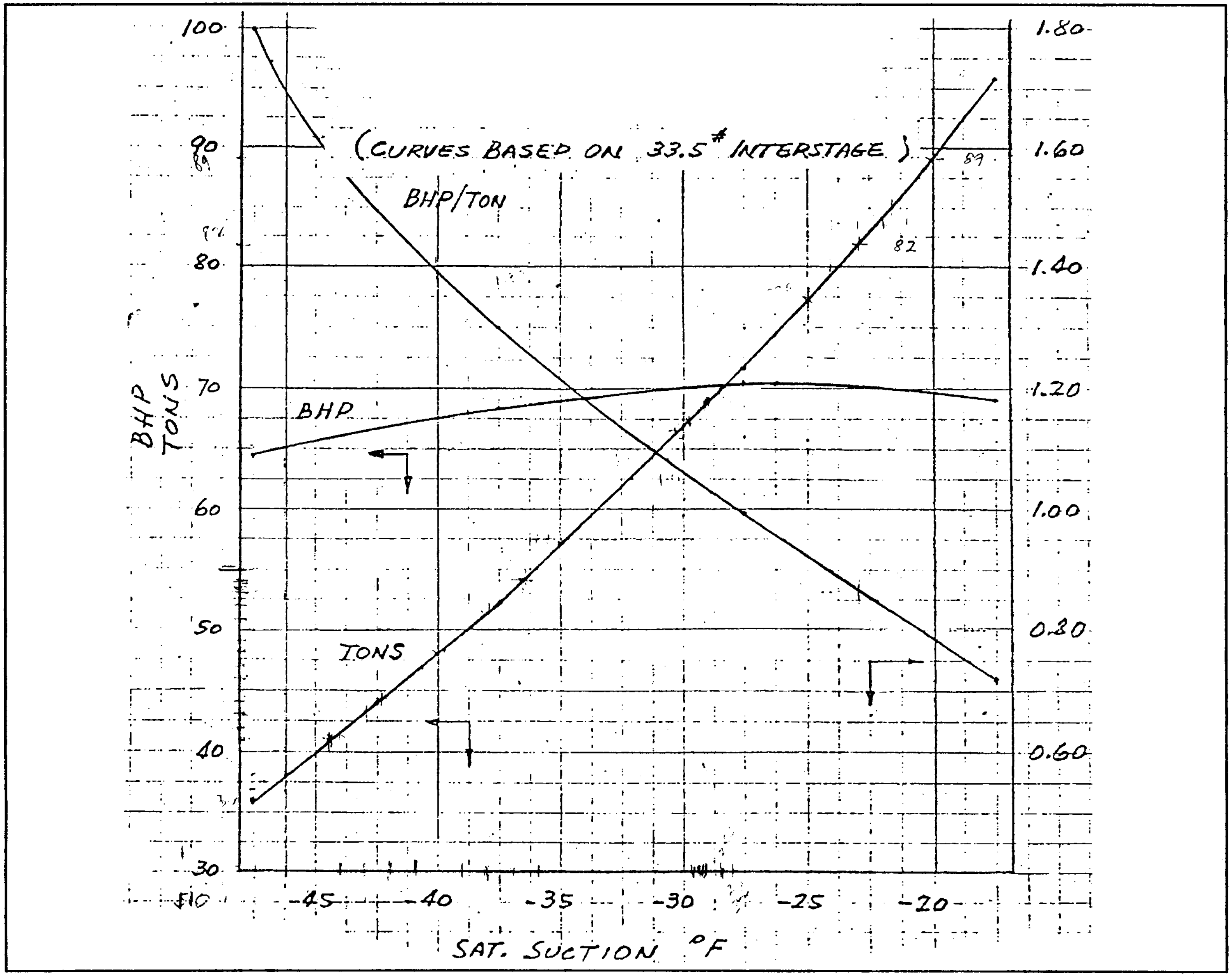
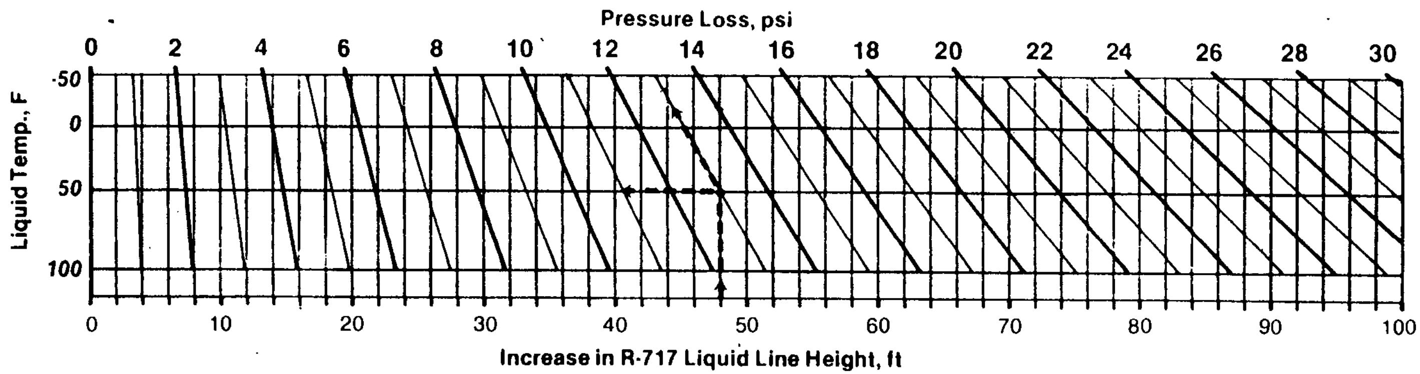


Chart 6: Effect of Variable Suction Pressure Upon Booster Tons & BHP at Constant Discharge Pressure



Courtesy of American Society of Heating, Refrigerating and Air-Conditioning Engineers. Reprinted from "Refrigerant Line Sizing," ASHRAE Research Project 185.

Chart 7: Liquid Lines, Pressure Loss from Increase in Height

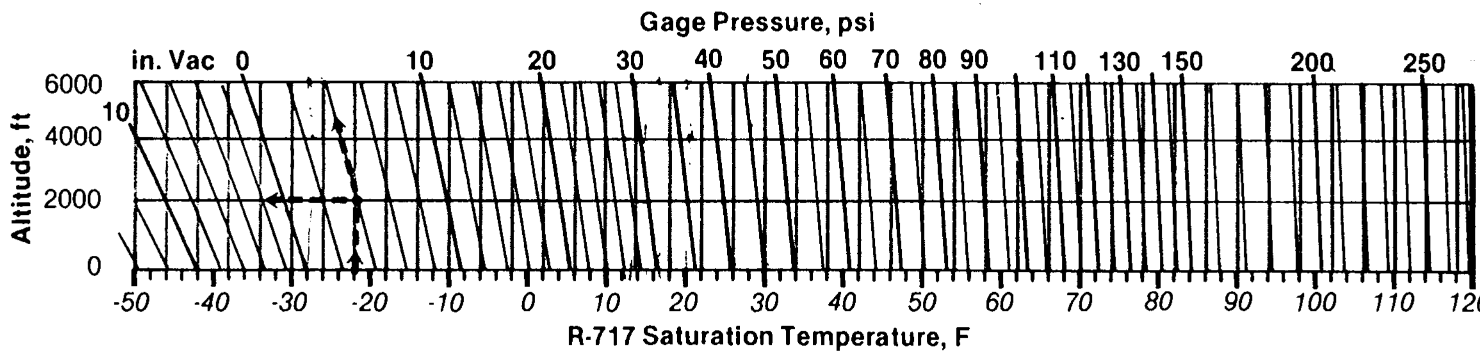


Chart 8: Gauge Pressure at Various Altitudes.
Courtesy of American Society of Heating, Refrigerating, and Air-Conditioning Engineers. Reprinted from "Refrigerant Line Sizing," ASHRAE Research Project 185.