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Informal Report

IMPACTS OF REFRIGERANTLINE LENGTH ON SYSTEM EFFICIENCY IN RESIDENTIAL HEATING AND COOLING SYSTEMS USING REFRIGERANT DISTRIBUTION

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ABSTRACT

The effects on system efficiency of excess refrigerant line length are calculated for an idealized residential heating and cooling system. By excess line length is meant refrigerant tubing in excess of the 25 ft provided for in standard equipment efficiency test methods. The purpose of the calculation is to provide input for a proposed method for evaluating refrigerant distribution system efficiency. A refrigerant distribution system uses refrigerant (instead of ducts or pipes) to carry heat and/or cooling effect from the equipment to the spaces in the building in which it is used. Such systems would include so-called "mini-splits" as well as more conventional split systems that for one reason or another have the indoor and outdoor coils separated by more than 25 ft. This report performs first-order calculations of the effects on system efficiency, in both the heating and cooling modes, of pressure drops within the refrigerant lines and of heat transfer between the refrigerant lines and the space surrounding them.

EXECUTIVE SUMMARY

In American single-family housing, ductwork has been the most common means of transporting thermal energy from the heating or cooling equipment to the building spaces in which the heat or cooling is needed. Hot-water piping has also traditionally found application for heating-only systems, most extensively in the Northeast. Recently, however, a new horse has been entered in the thermal distribution derby: refrigerant distribution.

In a typical refrigerant distribution system, refrigerant lines run directly from an outdoor compressor/coil unit to an indoor fan/coil unit mounted within the space to be cooled or heated. Such a system eliminates the need for ductwork. It is therefore an increasingly popular retrofit option in homes where installing ductwork would be prohibitively expensive or intrusive.

Refrigerant, of course, has long been used as an internal heat transport medium within air conditioners and heat pumps. In "split" systems, the distance over which heat or cooling is carried by refrigerant lines (e.g., from an outdoor coil and compressor mounted on the ground to an indoor coil mounted in the attic) can be a significant fraction of the distance covered by ductwork. So a question arises right away: when is a refrigerant line to be considered part of the equipment and when is it part of the distribution system?

The approach taken here has been to assign to the equipment the first 25 ft of refrigerant line on each side of the system (liquid and vapor), in line with the standard practice in measuring the efficiency of split systems (ARI 210/240-84). Anything over 25 ft is then considered to be part of the distribution system, and the effect of this "excess" refrigerant line in degrading overall system efficiency is attributed to a distribution efficiency value of less than 100%.

This report presents a series of first-order calculations of the efficiency impacts of temperature and pressure changes within the "excess" portions of refrigerant line. The results are presented as proposed input to a refrigerant-distribution section of ASHRAE Standard 152P, Method of Test for Determining the Design and Seasonal Efficiencies of Residential Thermal Distribution Systems.

Ten specific impacts were identified, five in heating systems and five in cooling:

CTL Temperature change due to heat transfer, liquid line, cooling mode
CPL Pressure drop due to added tube length, liquid line, cooling mode
CEL Pressure change due to elevation of indoor coil, liquid line, cooling mode
CTV Temperature change due to heat transfer, vapor (suction) line, cooling mode
CPV Pressure drop due to added tube length, vapor (suction) line, cooling mode

HTL Temperature change due to heat transfer, liquid line, heating mode
HPL Pressure drop due to added tube length, liquid line, heating mode
HEL Pressure change due to elevation of indoor coil, liquid line, heating mode
HTV Temperature change due to heat transfer, vapor (discharge) line, heating mode
HPV Pressure drop due to added tube length, vapor (discharge) line, heating mode.

For each of these impacts, it was necessary to proceed in three steps:

- Identify where and how the impact occurs on a pressure-enthalpy diagram.
- Calculate the change in the temperature or pressure that is caused by a given length (or elevation) of line.
- Calculate the effect of this pressure or temperature change on the efficiency of the cycle. (There may also be effects on capacity, but except insofar as they affect efficiency, these will be of secondary importance in ASHRAE Standard 152.)

Two of the impacts, due to elevation of the indoor coil in the cooling and heating modes, respectively, were evaluated on a rough "worst-case" basis. In the cooling mode, these were found to be only about 1% for a one-story elevation difference and 2% for a two-story elevation difference. In the heating mode, there should be even less negative impact. Furthermore, any such impact can be eliminated by using an actively controlled expansion device (thermostatic expansion valve) or by allowing for the elevation difference in the selection of a fixed device (orifice or capillary tube). It is therefore recommended that no provision for elevation difference be included in ASHRAE Standard 152.

The other eight impacts are defined by the following set of three two-way choices: heating mode vs. cooling mode; temperature vs. pressure changes; and liquid line vs. vapor (suction or discharge) line. These were calculated on the basis of impact in the benchmark case of a 2.5-ton (cooling) unit using refrigerant R-22 for a standard set of outdoor conditions and with four choices of refrigerant line size. A 10 ft section of excess refrigerant line was used as the standard length for which the magnitudes of the impacts were reported. This length was chosen because the impacts for 1 ft were found to be extremely small, while 100 ft (the other obvious choice) is considerably longer than the excess line lengths expected in most residential systems.

This analysis included the four smallest diameters of Type-L copper considered in Chapter 3 of the 1998 ASHRAE Refrigeration Handbook (I-P edition). These have outer diameters, in inches, of 0.500, 0.625, 0.875, and 1.125. Their nominal sizes are quoted in terms of their approximate inner diameters as 3/8", 1/2", 3/4", and 1", respectively.

Only one of these efficiency impacts was found to be greater than 1% per 10 ft of excess refrigerant line. This was the effect of pressure drops in the suction line in the cooling mode (designated as CPV in the above list). For the smallest refrigerant line, 0.500 in. O.D. (3/8 in. nominal) an efficiency degradation of 5.0% per 10 ft of excess refrigerant line was projected. For the second-smallest line, 0.625 in. O.D. (1 /2 in. nominal), a 1.7% degradation in efficiency was projected. Larger lines had insignificant impacts from this cause.

The primary conclusion of this work is that, with the exception of pressure drops in the suction line in the cooling mode, the efficiency impacts are quite small for any reasonable excess lengths of refrigerant line likely to be found in residential applications. Elevation of the indoor coil relative to the outdoor coil may be marginally significant in some cases.

Conceptually, there are at least two possible approaches to a refrigerant distribution efficiency test method. The first is to measure this efficiency directly. The second is to measure those parameters that will provide sufficient basis for calculating an efficiency to a sufficiently good approximation.

It seems likely that measuring the efficiency of refrigerant distribution systems directly will not be feasible in anything other than a highly controlled research environment. Such measurements might have some value as a check on an algorithm that would go into Standard 152, but it is unlikely that they will be useful in a context where time and effort expended are key parameters in an overall optimization of the test method. Direct measurement of efficiency will simply take too much time, equipment, and expertise to be useful in the field.

The alternative is to develop an algorithm for calculating the approximate impact on distribution efficiency of refrigerant line lengths in excess of what is provided for in equipment efficiency test methods. This algorithm would need to provide for the following significant variables:

- Equipment capacity
- Refrigerant line cross-sections and insulation levels
- Excess refrigerant line length
- Refrigerant used
- Evaporating and condensing temperatures
- Temperatures of spaces in which excess refrigerant line length is located, and whether these spaces are in thermal communication with the conditioned space.

The project of developing such an algorithm will be aided by the fact that the required accuracy is reasonably expressed as a fraction of 100% efficiency, not a fraction of the effects themselves. Because the efficiency impacts of excess refrigerant line are mostly small, this will make relatively large fractional errors in most of the individual effects acceptable. The effects due to pressure drops in the suction line in the cooling mode are the major issue. Here, further efforts to refine the accuracy of the calculations could be warranted. If a sufficiently accurate characterization of this effect, coupled with approximations to the others, is achieved, this should result in a useful algorithm for a practical method for evaluating refrigerant distribution efficiency.

The following steps are recommended to proceed with such a project. Each step should be answered satisfactorily and any problem areas addressed before proceeding to the next step.

1. Review the calculations in this report for soundness as to their approximate validity under the conditions for which they were developed.

2. Determine whether any additional impacts may have been ignored.

3. Determine to what extent the characteristics of real cycles, as opposed to the ideal cycle used in this analysis, need to be included.

4. Decide what parameters need to be varied in the Standard 152 algorithm, and what ranges of values these parameters should take on.

5. Extend the calculations in this report to include all these parameters and values.

6. Develop an algorithm for Standard 152 that maximizes accuracy consistent with reasonable measurement effort.

It may be that a quick determination of equipment capacity, length and size of any excess refrigerant lines, where these lines go in the house, and house location (for climate variables) would be enough input for a spreadsheet to determine values for design and seasonal distribution efficiency. In view of the relatively small impacts of everything except pressure drops in the suction line (cooling mode), it appears likely that such a program could succeed.

1 INTRODUCTION

This report develops a first-order quantification of the impacts of excess refrigerant line length on the efficiency of vapor-compression heating and cooling equipment. By "excess" is meant line length over and above that specified in the standard test methods for equipment efficiency. These calculations are meant to serve as input to a planned section on "refrigerant distribution" of ASHRAE Standard 152, Method of Test for Determining the Design and Seasonal Efficiencies of Residential Thermal Distribution Systems (ASHRAE 1999). Refrigerant distribution systems are defined as those that use refrigerant to transport heat and cooling effect from the equipment to the points of use. This would include systems such as "mini-splits" in which refrigerant substitutes for ductwork. It would also include conventional ducted systems in which the refrigerant line length is for some reason longer than the maximum provided for in the equipment test standards. It may also include effects of differential height (elevation) between the indoor and outdoor coils, such as would occur in multistory buildings with attic ductwork and ground-level equipment.

Use has been made of an informal discussion prepared by Keith Rice of Oak Ridge National Laboratory, which is reproduced in Appendix 1, and which is gratefully acknowledged. Any misinterpretations of that discussion are the responsibility of the author of this report.

Following the discussion by Rice, it is assumed that the vapor-compression equipment, whether it be an air conditioner or a heat pump, is tested with 25 ft of interconnecting tubing on each line (liquid and suction in the cooling mode or liquid and discharge in the heating mode). Of these 25 ft, at least 10 ft must be exposed to outdoor conditions. The impact of a refrigerant distribution system on overall efficiency will therefore be calculated as the impact of any incremental length of refrigerant tubing beyond the 25 ft required in equipment testing.

The extra footage of refrigerant line generally can make itself felt in two ways: through added pressure drops, and through temperature changes caused by heat transfer through the walls of the tubing. Additionally, there is the possibility of an effect caused by the pressure difference of elevation of the indoor coil relative to the outdoor coil, if the indoor coil is, say, in an attic while the outdoor coil is ground-mounted. The elevation effect should be significant, if at all, only for lines containing liquid refrigerant, because the density, and therefore any pressure head, in the vapor-containing line will be only ~8% of that in the liquid line in the heating mode and ~2% in the cooling mode, under the conditions studied in this report.

Referring to Figures 1 and 2, therefore, which show schematics of the refrigeration loop in the cooling and heating modes, respectively, it is possible to identify ten specific impacts that need to be evaluated:

CTL Temperature change due to heat transfer, liquid line, cooling mode
CPL Pressure drop due to added tube length, liquid line, cooling mode
CEL Pressure change due to elevation of indoor coil, liquid line, cooling mode
CTV Temperature change due to heat transfer, vapor (suction) line, cooling mode
CPV Pressure drop due to added tube length, vapor (suction) line, cooling mode
HTL Temperature change due to heat transfer, liquid line, heating mode
HPL Pressure drop due to added tube length, liquid line, heating mode
HEL Pressure change due to elevation of indoor coil, liquid line, heating mode
HTV Temperature change due to heat transfer, vapor (discharge) line, heating mode
HPV Pressure drop due to added tube length, vapor (discharge) line, heating mode.

For each of these impacts, it will be necessary to proceed in three steps:

- Identify where and how the impact occurs on a pressure-enthalpy diagram.
- Calculate the change in the temperature or pressure that is caused by a given added length of line (or elevation).
- Calculate the effect of this pressure or temperature change on the efficiency of the cycle. (There may also be effects on capacity, but except insofar as they affect efficiency, these will be of secondary importance in ASHRAE Standard 152.)

These steps are carried out in Sections 3, 4, and 5, respectively, of this report.

In this analysis, the impact on system efficiency of excess refrigerant line is calculated as a percentage of the efficiency of an ideal cycle operating betwen the assumed evaporating and condensing temperatures. By considering percentage changes only, it is hoped that any error caused by the fact that an ideal cycle is less efficient than a real cycle will be kept to an acceptable minimum. In cases where a bias from this procedure can be identified, this will be pointed out and a correction will be made to account for it.

2 DESCRIPTION OF BASE CASE

As discussed above, each of the ten impacts will be evaluated in terms of its effect on the efficiency of an ideal refrigeration cycle. Figure 3 shows a pressure-enthalpy (P-H) diagram of such a cycle. The four major transitions--compression, condensing, expansion, and evaporating--are shown on their corresponding lines in the P-H diagram. Figure 4 adds the locations of the refrigeration lines. It is worth noting that the components of a "split" refrigeration system that are physically the longest, namely the liquid line and the vapor line (suction or discharge in cooling or heating, respectively) are located at compact points on the P-H diagram, whereas functions that are physically compact (compressor, expansion valve, heat exchangers) are extended lines on the P-H diagram.

2.1 Benchmark Assumptions

In the cooling mode, the evaporating (indoor coil) and condensing (outdoor coil) temperatures will be assumed to equal 40 °F and 120 °F, respectively. In the heating mode, the positions of evaporator and condenser are reversed. For these calculations, the outdoor evaporating temperature will be assumed equal to 20 °F and the indoor condensing temperature will be assumed equal to 120 °F.

In a more detailed study, these temperatures will need to be varied as parameters to cover the wide range of conditions to be found in actual applications, but for the present purpose it will suffice to note that in each case the selected temperature is within the range of values expected to be encountered in real systems.

In the cooling mode, refrigerant leaves the condenser at some temperature difference above ambient; the selected 120 °F is within a typical operating range. The evaporating temperature needs to be sufficiently below the indoor-air dew point that adequate dehumidification is provided. The selected 40 °F will generally meet this requirement, although somewhat higher temperatures may meet it as well.

In the heating mode, the indoor condenser needs to be at a high enough temperature that supply air can be provided at ~100 °F. Much lower supply-air temperatures will limit the capacity of the system to deliver heat, and are also likely to cause complaints of drafts. On the evaporator side, the selected temperature of 20 °F will usually be within a real unit's operating range, which will vary depending on the climate.

The equipment cooling capacity will be assumed to equal 30,000 Btu/h (2.5 tons). Refrigerant R-22 will be used as the baseline working fluid. The refrigerant flow rate for the cooling mode will be derived from this specification. The refrigerant flow rate for the heating mode will be assumed to equal 70% of that for the cooling mode. This reflects the ratio of densities of saturated refrigerant vapor at 20 °F and 40 °F.

Where a temperature surrounding refrigerant tubing outside the conditioned space is needed for a benchmark calculation, this will be assumed to equal 80 °F in the cooling mode and 40 °F in the heating mode. The effect of heat transfer on system efficiency will be nearly proportional to the temperature difference between the refrigerant in the line and the space surrounding it, so if in an actual case another surround temperature is needed, the results can be pro-rated linearly.

When the effects on efficiency are finally quantified, they will be expressed on the basis of a 10-ft excess length of refrigerant line. Why ten feet rather than one foot or a hundred? One foot is so little that the effect is generally expected to be insignificant, while 100 ft is a much longer excess length than will usually be encountered in single-family residential applications. Because the excess lengths in these applications are expected to be a few tens of feet at most, it seemed reasonable to base the calculations on a length falling within such a limit but a significant fraction of it. The equations developed will not be specific to these assumptions, but the numerical results will reflect them.

2.2 Ideal Cycle, Cooling Mode

The ideal efficiency in the refrigeration cycle shown in Figure 3 can be calculated from the enthalpies at the "corners." The relevant parameters for the cooling mode are shown in Table 1.

Table 1. Parameters for States Bounding the I deal Cycle, R-22, 40 °F Evaporating Temperature, 120 °F Condensing Temperature. Defining Conditions Boldfaced.

State Description	Pressure (psia)	Temperature (°F)	Entropy (Btu/lbm-°F)	Enthalpy (Btu/lbm)
1. **Saturated** vapor entering compressor (State 1)	83.25	**40**	0.220	108.2
2. Superheated vapor leaving compressor (State 2)	**274.8** (from 3)	156	**0.220** (from 1)	121.1
3. **Saturated** liquid leaving condenser (State 3)	274.8	**120**	N/A	45.7
4. Mixed-phase refrigerant entering evaporator (State 4)	**83.25** (from 1)	40	N/A	**45.7** (from 3)

The ideal work of compression is the difference in enthalpy between State 2 and State 1, or 12.9 Btu/lbm. The ideal cooling effect is the difference in enthalpy between State 1 and State 4, or 62.5 Btu/lbm. The ideal cooling COP is 62.5/12.9 or 4.84. As discussed elsewhere, the efficiency impacts in the baseline calculations will be based on fractional impacts on the ideal cycle efficiencies. It is expected that these will be similar to the fractional impacts on real cycle efficiencies, at least to first order, although further investigation on this point is warranted.

2.3 Ideal Cycle, Heating Mode

Again referring to the refrigeration cycle shown in Figure 3, the ideal heating efficiency can be calculated from a knowledge of the enthalpies at the "corners," with the difference that the evaporating temperature is reduced to 20 °F. The relevant state parameters for the heating mode are shown in Table 2.

Table 2. Parameters for States Bounding the Ideal Cycle, R-22, 20 °F Evaporating Temperature, 120 °F Condensing Temperature. Defining Conditions Boldfaced.

State Description	Pressure (psia)	Temperature (°F)	Entropy (Btu/lbm-°F)	Enthalpy (Btu/lbm)
1. **Saturated** vapor entering compressor (State 1)	57.79	**20**	0.223	106.2
2. Superheated vapor leaving compressor (State 2)	**274.8** (from 3)	156	**0.223** (from 1)	124.0
3. **Saturated** liquid leaving condenser (State 3)	274.8	**120**	N/A	45.7
4. Mixed-phase refrigerant entering evaporator (State 4)	**57.79** (from 1)	20	N/A	**45.7** (from 3)

The ideal work of compression is the difference in enthalpy between State 2 and State 1, or 17.8 Btu/lbm. The ideal heating effect is the difference in enthalpy between State 2 and State 3, or 78.3 Btu/lbm. The ideal heating COP is 78.3/17.8 or 4.40. As discussed elsewhere, the efficiency impacts in the baseline calculations will be based on fractional impacts on the ideal cycle efficiencies. It is expected that these will be similar to the fractional impacts on real cycle efficiencies, at least to first order, although, again, further investigation on this point is warranted.

3 CYCLE IMPACTS

Fully detailed cycle calculations are beyond the scope of this report. It may be that to capture all significant impacts, a complete modeling effort may be required, as suggested by Rice (Appendix 1). However, first-order calculations may provide significant information concerning expected orders of magnitude and relative importance of the various impacts. Where the first-order effect is nil, it may be necessary to consider second order. For the purposes of this report, the terms "first order" and "second order" are defined as follows:

First Order:

- For the baseline system, i.e., without extra tubing or elevation of the indoor coil, an ideal cycle is assumed, with no suction superheat or post-condenser subcooling.
- Heat-exchanger capacity is effectively infinite, that is, condensing and evaporating temperatures are assumed to be unaffected by changes in the throughput caused by the existence of excess refrigerant line length.

Second Order:

- First-order calculations are carried out.
- Condensing and evaporating temperatures are then adjusted to account for changes in heat throughput required by the first-order calculations, and any effect of this on compressor work is accounted for in the efficiency estimate.

The next step will be to locate, on the P-H diagram, the impacts of each of the ten effects defined above.

3.1 Cycle Impacts of Temperature Changes Due to Heat Transfer, Liquid Line, Cooling Mode (CTL)

The ambient temperature in the space surrounding a liquid refrigerant line outside the building is usually expected to be less than the temperature of the refrigerant within the line, the difference being the temperature difference across the condenser minus any subcooling. Any parts of the refrigerant line inside the building will usually be exposed to lower temperatures still, since the conditioned space is generally cooler than the outside ambient during the air-conditioning season. The one major exception would be any portions of the line in a hot attic, whose temperature may significantly exceed ambient.

To the extent that heat leaves the liquid refrigerant line, it will add to any subcooling at the cusp of the condensing and expansion lines in the P-H diagram (Figure 5). This will add to the amount of heat taken up in the evaporator, as shown in the diagram. A second-order effect would be a slight lowering of the evaporating temperature, needed to drive this additional heat across the heat exchanger. This will add somewhat to the work of compression.

3.2 Cycle Impacts of Pressure Drops Due to Added Tube Length, Liquid Line, Cooling Mode (CPL)

Pressure drop in the liquid line, will, everything else being equal, result in less pressure drop through the expansion device (Figure 6). If the device is capable of adjusting the flow, as for example a thermostatic expansion valve, little or no impact on the cycle would be expected. However, for a fixed device such as an orifice or capillary tube, flow will be reduced unless the orifice is resized to account for the pressure drop. This would be partially compensated by a rise in condensing pressure, in turn increasing the compressor work per cycle (although reducing somewhat the capacity of a real compressor).

Detailed consideration of the effect would require a complex calculation; here a worst-case estimate only will be developed. It is expected that liquid-line pressure drops due to friction will be relatively small, but a calculation is needed to provide an order of magnitude for this effect.

3.3 Cycle Impacts of Pressure Drops Due to Coil Elevation, Liquid Line, Cooling (CEL)

The effect of elevating the indoor coil relative to the outdoor coil should be similar to those of friction-caused pressure drops in the liquid line (Figure 6). However, in contrast to those effects (which are determined by line length, cross section, and roughness), in the case of an elevated evaporator, the density of the refrigerant will govern the effect.

3.4 Cycle Impacts of Temperature Rise Due to Heat Transfer, Vapor (Suction) Line, Cooling Mode (CTV)

Heat transfer into the suction line, from outdoors or buffer spaces within the building to the refrigerant, will add to suction superheat at the compressor inlet (Figure 7). This will reduce refrigerant density and therefore capacity. Efficiency impact can be estimated by calculating the impact of the temperature rise on entropy at the compressor inlet, and then getting the work of compression along this isentropic curve, and comparing it with the work of compression along the original isentropic curve that passes through the saturation curve at 40 °F. The work of compression starting at a superheated state will be somewhat higher because the lines of constant entropy get flatter, i.e., have lower slope, as one goes away from the saturation line.

3.5 Cycle Impacts of Pressure Drops Due to Added Tube Length, Vapor (Suction) Line, Cooling Mode (CPV)

Pressure drop in the "excess" portion of the suction line will cause the inlet pressure at the compressor to be lower than it otherwise would have been (Figure 8). This will bring the compressor inlet into the superheat region (assuming saturation at baseline) or further into the superheat region in any real case. This will involve additional compressor work per pound of refrigerant with no effect on the enthalpy change in the evaporator, so efficiency will be reduced. Capacity will also be reduced because of the lowered refrigerant density.

3.6 Cycle Impacts of Temperature Changes Due to Heat Transfer, Liquid Line, Heating Mode (HTL)

In the heating mode, heat loss from the liquid line to the outside, or to buffer spaces in the building, will subcool the refrigerant (Figure 9). This will not add heating capacity (because the heat is lost to the outside), but it will add to the thermal uptake required at the evaporator. In first order, this will have no effect on efficiency, but in second order it will lower the evaporating temperature, requiring some added compressor work. A small drop in efficiency is expected.

3.7 Cycle Impacts of Pressure Drops Due to Added Tube Length, Liquid Line, Heating Mode (HPL)

Friction induced pressure drop in the liquid line will reduce the inlet pressure at the expansion valve, reducing flow in a fixed expansion device (Figure 10). The discussion is similar to that in Section 3.2. Some rise in condensing pressure and temperature may occur to compensate. This effect is expected to be small.

3.8 Cycle Impacts of Pressure Drops Due to Coil Elevation, Liquid Line, Heating (HEL)

In the heating mode, elevation of the indoor coil would increase the pressure drop across the expansion device. (Compare Figure 10 with Figure 6, for the cooling mode.) Friction-induced and elevation-related pressure drops will work in opposite directions here. This does not mean they will cancel, because in any given system one may be much greater than the other.

3.9 Cycle Impacts of Temperature Changes Due to Heat Transfer, Vapor (Discharge) Line, Heating Mode (HTV)

Temperature drops in the discharge line outside the conditioned space will reduce capacity on a one-for-one basis (Figure 11). Since compressor work is unaffected, efficiency will be reduced. Some reduction of condensing pressure and temperature may result from reduced heat transfer in the condenser itself, and this will reduce compressor work somewhat. However, this is a second-order effect that is not expected to mitigate significantly the first-order capacity and efficiency reduction.

3.10 Cycle Impacts of Pressure Drops Due to Added Tube Length, Vapor (Discharge) Line, Heating Mode (HPV)

Pressure drops in the discharge line will have to be compensated for if the same condensing conditions are to be maintained. This will entail added compressor work per pound of refrigerant, as indicated in Figure 12. In a real system, this would reduce capacity, causing the system to equilibrate at a somewhat lower condensing pressure and temperature, reducing somewhat the amount of additional compressor work per pound of refrigerant.

4 TEMPERATURE AND PRESSURE CHANGES IN "EXCESS" REFRIGERANT LINE

In this section, estimates are made of the expected impact of excess refrigerant line length on temperature and pressure in the line.

4.1 Magnitude of Temperature Changes Due to Heat Transfer, Liquid Line, Cooling Mode (CTL)

As indicated above and shown in Figure 5, the major effect of heat transfer from the warm condensed liquid into the generally cooler surrounding regions (whether in a buffer space or outdoors) will be to add subcooling at the intersection of the condensing and expansion lines in the P-H diagram. If the ideal cycle is used as a base, this will push the condensing line into the liquid region. (In a real system, condensing would normally end in the liquid region, i.e., there would be some subcooling in any case, but this would just add to the subcooling.)

As mentioned in Section 3.1, it is also possible that the temperature surrounding the liquid line might be higher than that of the refrigerant in the line. This could happen, for example, in a refrigerant line that passes through a hot attic.

What one needs to calculate here is the amount of temperature drop (or rise) to be expected. Although the excess liquid line is usually uninsulated and will usually experience a modest decline in temperature (unless it is in a hot attic), the calculation will be set up in a general format that can account for liquid or vapor refrigerant, increases or decreases in temperature, and insulated or uninsulated refrigerant lines. That way, subsequent sections dealing with the heating mode and with suction/discharge lines can refer to this one, eliminating the need to repeat much of this discussion.

Because the temperature change is normally expected to be small, the calculation will be based on a constant temperature difference across the annular region representing the refrigerant-line wall. Parameters defining the characteristics of this line are:

d	Inner diameter of refrigerant line, ft
d_1	Outer diameter of refrigerant line (inner diameter of insulation, if any), ft
d_2	Outer diameter of insulation, if any, ft
$h_{conv+rad}$	Radiative/convective heat transfer coefficient to outside, Btu/h-ft^2-°F
k	Thermal conductivity of insulation, if any, Btu/h-ft-°F
L	Excess length of refrigarant line, ft
C_p	Specific heat of refrigerant (at constant pressure), Btu/lbm-°F
$\mathring{m}$	Mass flow rate of refrigerant, lbm/h
Q	Heat transfer rate into excess length of refrigerant line, Btu/h
T_{in}	Refrigerant temperature inside refrigerant line, °F
T_{out}	Temperature of space external to refrigerant line, °F
U_{linear}	Heat transfer coefficient per unit length of refrigerant line, Btu/h-ft-°F

UA	Overall heat transfer coefficient between excess line and outside, Btu/h-°F
ρ	Density of refrigerant, lbm/ft^3
ΔT_{ref}	Temperature change in excess portion of refrigerant line, °F

The heat transfer rate from the outside to the refrigerant in the excess portion of the line will first be calculated, using the expression $Q = UA\ \Delta T_{lm}$, where ΔT_{lm} is the log-mean temperature difference between the refrigerant in the line and the space surrounding the line. In the present case, an expression for the heat transfer coefficient U_{linear} per unit length into an insulated tube will be used, so that UA will equal U_{linear} L. This heat transfer rate will then be divided by the thermal mass flow rate $\mathring{m}\ C_p$ to obtain the temperature rise in the excess portion of the refrigerant line.

Heat transfer equations selected previously for hot-water heating systems (Andrews 1997) will be used here. The heat transfer coefficient per unit length into an insulated tube is given by:

$$U_{linear} = \frac{2\pi}{\frac{1}{k}\ln\left(\frac{d_2}{d_1}\right) + \frac{2}{d_2\, h_{conv+rad}}} \qquad (1)$$

the thermal resistance of the internal film and of the copper itself being assumed negligible relative to that of the external film and, *a fortiori*, the insulation if any.

If one can assume that the temperature change in the excess length of refrigerant tube is an order of magnitude smaller than the temperature difference between the outside and the inside of the tube, only a small error will be introduced by using $\Delta T_{lm} = T_{out} - T_{in}$. The heat transfer rate will then be given by:

$$Q = \frac{2\pi L\,(T_{out} - T_{in})}{\frac{1}{k}\ln\left(\frac{d_2}{d_1}\right) + \frac{2}{d_2\, h_{conv+rad}}} \qquad (2)$$

The temperature change in the refrigerant is then equal to:

$$\Delta T_{ref} = \frac{Q}{\dot{m}\ C_p} = \frac{2\ \pi\ L\ (T_{out} - T_{in})/(\dot{m}\ C_p)}{\frac{1}{k}\ \ln\left(\frac{d_2}{d_1}\right) + \frac{2}{d_2\ h_{conv+rad}}} \qquad (3)$$

If the refrigerant line is uninsulated, $d_1 = d_2$ and Equation 3 reduces to:

$$\Delta T_{ref} = \frac{\pi\ L\ d_1\ h_{conv+rad}\ (T_{out} - T_{in})}{(\dot{m}\ C_p)} \qquad (4)$$

Because the liquid line is usually uninsulated, in this section Equation 4 will be used to obtain ΔT_{ref} for a stated excess length of line, which as discussed above is being taken as 10 ft in this report.

As indicated above, the temperature surrounding the liquid line is assumed to be 80 °F and the temperature leaving the condenser is 120 °F, so $(T_{out} - T_{in}) = -40$ °F.

The refrigerant mass flow rate for 2.5 tons of refrigeration is obtained by noting that the enthalpy of saturated R-22 vapor at 40 °F is 107.9 Btu/lbm, while the enthalpy of saturated R-22 liquid at 120 °F is 45.7 Btu/lbm. The difference of 62.2 Btu/lbm is the evaporator heat uptake in the ideal cycle. Since 2.5 tons of refrigeration requires 30 000 Btu/lbm-h, the mass flow rate is 30 000 divided by 62.2 or 482 lbm/h.

The specific heat of saturated liquid at 120 °F is 0.161 Btu/lbm-°F. When combined with the above mass flow rate, the thermal mass flow rate $\dot{m}C_p$ is 482 X 0.161 = 77.6 Btu/°F-h .

It remains to specify typical values for the outer diameter of the refrigerant line. The ASHRAE Refrigeration Handbook (ASHRAE 1998), Chapter 2, Table 3, gives suction line sizes for R-22 as a function of capacity and saturated suction temperature, for a 2 °F drop in saturated suction temperature. The suction line sizes in this table, consistent with residential-size equipment, range from 0.5 in. to 1.125 in. O.D. (3/8" to 1" nominal), so these sizes are used in the calculations. For bare copper, $h_{conv+rad}$ = 1.75 Btu/h-ft^2-°F (see Andrews 1997). Temperature drops per 10 ft of uninsulated liquid line are given in row 5 of Table 3.
These ideal temperature drops should be considered upper limits for the assumed conditions, because in a real system, there will usually be some subcooling in the condenser and in the 25 ft of

liquid line provided for in the equipment efficiency tests. Therefore, the actual temperature difference between the inside and outside of the refrigerant line will be less than the assumed 40 °F. Revised values of this temperature difference are therefore calculated as 40 °F minus 2.5 times the calculated temperature drop per 10 ft, to account for 5 °F subcooling in the condenser and heat loss in the 25 ft of standard-issue refrigerant line. This process can be iterated until the inside-outside ΔT is consistent with the temperature drop in the refrigerant. These iterated inside-outside temperature differences, together with the resulting temperature drops in the refrigerant per 10 ft of line, are shown in the last two rows of Table 3.

Table 3. Temperature Drops per 10 ft of Liquid Line at 40 °F Temperature Difference, Cooling Mode, 2.5 Ton System, R-22, Type L Copper Pipe.

Pipe O.D., in.	0.500	0.625	0.875	1.125
Pipe I.D., in.	0.430	0.545	0.785	1.025
Nominal Size	3/8"	1/2"	3/4"	1"
Insulation wall thickness, in.	Uninsulated	Uninsulated	Uninsulated	Uninsulated
Temperature drop per 10 ft excess line, uncorrected, °F[1]	1.18	1.48	2.07	2.66
Iterated inside-outside ΔT, °F[2]	32.6	32.0	31.0	30.0
Temperature drop per 10 ft excess line, iterated, °F[3]	0.96	1.18	1.60	1.99

1. From Equation 4. 2. Assuming 5 °F condenser subcooling and calculated temperature drops in standard 25 ft of refrigerant line. 3. Based on recalculated inside-outside ΔTs.

4.2 Magnitude of Pressure Drops Due to Added Tube Length, Liquid Line, Cooling Mode (CPL)

The impact of pressure drops in the liquid line is to reduce the pressure difference across the expansion device (Figure 6). The impact of this on efficiency will be taken up below; here the objective is simply to calculate the magnitude of the pressure drop as a function of line size, excess length, and equipment capacity. In order to do this, the information in the 1998 ASHRAE Refrigeration Handbook, Chapter 2, Table 3 (ASHRAE 1998) will be used. For each of the four smallest refrigerant-line sizes, one first takes the benchmark capacity that will produce a 1 °F drop in the saturated-liquid temperature. Then, using a formula given at the bottom of this table:

$$\Delta T = Table\ \Delta T\ \frac{Actual\ L_e}{Table\ L_e} \left(\frac{Actual\ capacity}{Table\ capacity} \right)^{1.8} \qquad (5)$$

the drop in saturation temperature, per 10 ft of line, for a 2.5 ton capacity is calculated. Care must be taken to include a factor 0.1 for Actual L_e / Table L_e, since 10 ft is being used as the reference line length. Finally it is noted that a drop of 1 °F in saturation temperature is equivalent to a 3.5 psi drop in pressure (at 120 °F condensing), so the calculated ΔT is multiplied by 3.5 to get the calculated ΔP. (The value of 3.05 psi per degree F given in the ASHRAE table is for 105 °F condensing.)

Table 4 shows the inputs and the results of this procedure.

Table 4. Pressure Drops Per 10 ft of Liquid Line in the Cooling Mode, 2.5 Tons Capacity, R-22.

Refrigerant Line O.D. Type L Copper, in.	0.500	0.625	0.875	1.125
Capacity for a 2 °F drop in saturation temperature, tons[1]	3.6	6.7	18.2	37.0
Baseline Capacity, tons[2]	2.5	2.5	2.5	2.5
Calculated saturation ΔT, °F[3]	0.052	0.016	0.003	0.001
Calculated pressure drop, psi[4]	0.18	0.06	0.01	0.002

1. ASHRAE Refrigeration Handbook, 1998, Chapter 2, Table 3. 2. Assumed
3. From Equation 5. 4. Using 3.5 psi/°F change in saturation temperature.

4.3 Magnitude of Pressure Drops Due to Coil Elevation, Liquid Line, Cooling (CEL)

The effect in the liquid line of an elevation in the indoor coil, relative to the outdoor coil, is to reduce the pressure drop across the expansion device. Thus, the elevation impact will need to be folded in with the frictional pressure drop considered in the previous section.

The magnitude of the pressure drop due to elevation difference will simply be the pressure head due to a column of liquid refrigerant. The density of liquid R-22 at 120 °F is 67.9 lbm/ft^3. This corresponds to a pressure of 0.47 psi per foot of elevation. Table 5 gives pressure drops for several situations:

Table 5. Altered Pressure Changes Across Expansion Device for Several Indoor Coil Locations (Outdoor Coil at Ground Level), Cooling Mode, Liquid Line, R-22.

Indoor Coil Location	Basement, Typical	Garage, Typical	Attic, One-Story Home	Attic, Two-Story Home
Indoor Coil Elevation, ft	-4	4	8	16
Change in Pressure Drop Across Expansion Device, psi[1]	+1.9	-1.9	-3.8	-7.5

1. Using 0.47 psi/ft elevation.

4.4 Magnitude of Temperature Rise Due to Heat Transfer, Vapor (Suction) Line, Cooling Mode (CTV)

As indicated above and shown in Figure 7, the major effect of heat transfer into the cool suction line from the warmer surrounding regions (whether in a buffer space or outdoors) will be to add superheat at the cusp of the evaporation and compression lines in the P-H diagram. If the ideal cycle is used as a base, this will push the compression line into the superheat region. (In a real system, compression would normally start in the superheat region, but this would just push it out farther.) The required quantity here is the amount of temperature rise to be expected.

The equations needed for the calculations were developed in Section 4.1. A major difference between that section and this one is that the vapor-containing line is usually insulated. So the next step here is to solve Equation 3 for typical values of the input parameters to obtain benchmark values of the temperature rise ΔT_{ref} for a standard length of 10 ft of excess tubing.

As indicated above, the outdoor temperature is assumed to be 80 °F and the temperature leaving the evaporator is 40 °F, so $(T_{out} - T_{in}) = 40$ °F.

The refrigerant mass flow rate is calculated the same way as in Section 4.1, and equals 482 lbm/h. The specific heat of saturated vapor at 40 °F is 0.175 Btu/lbm-°F, slightly different from the liquid-refrigerant value used in Section 3.1. When combined with the above mass flow rate, the thermal mass flow rate $\dot{m}C_p$ is 482 X 0.175 = 84.3 Btu/°F-h .

The suction line of an air conditioner is typically insulated. Usually this insulation is of a closed-cell polymer foam. A representative value for the thermal conductivity of this material is 0.02 Btu/h-ft-°F. A representative value for the external film heat transfer coefficient $h_{conv+rad}$ is 2.4 Btu/h-ft^2 -°F. These values were obtained from standard references in connection with development of the hydronic portion (Andrews 1996) of ASHRAE Standard 152P (ASHRAE 1999).

Typical inner and outer diameters of the refrigerant line were specified in Section 3.1. It remains to specify typical values for insulation jacket thickness. Table 6 gives temperature rises per 10 ft of refrigerant line for insulation thickness values of 0.5 in. and 1.0 in.

If the line is not insulated, then Equation 4 is employed to find ΔT_{ref}, and the bare-copper value of $h_{conv+rad}$ (1.75 Btu/h-ft^2-°F) is used. Although it is not usual to leave the suction line uninsulated, the temperature rises for no insulation are given in Table 6 for comparison.

Table 6. Temperature Rises per 10 ft of Suction Line at 40 °F, 2.5 Ton System, R-22.

Pipe O.D., in. Type L Copper	0.500			0.625			0.875			1.125		
Pipe I.D., in.	0.430			0.545			0.785			1.025		
Nominal Size	3/8"			1/2"			3/4"			1"		
Insulation wall thickness, in.	No	0.5	1.0	No	0.5	1.0	No	0.5	1.0	No	0.5	1.0
Temperature rise per 10 ft of line, °F[1]	1.09	0.48	0.35	1.36	0.55	0.38	1.90	0.69	0.47	2.45	0.82	0.55

1. Using Equation 3 (insulated) or Equation 4 (uninsulated).

4.5 Magnitude of Pressure Drops Due to Added Tube Length, Vapor (Suction) Line, Cooling Mode (CPV)

The impact of pressure drops in the suction line is to reduce the pressure at the compressor inlet (Figure 8). In order to calculate the impact on efficiency, it is necessary first to determine the magnitude of the pressure drop. As was done for the liquid-line pressure drop (Section 4.2), the information in the 1998 ASHRAE Refrigeration Handbook, Chapter 2, Table 3 (ASHRAE 1998) will be used. For each of the four smallest refrigerant-line sizes, let us first take the benchmark capacity at 40 °F saturated suction temperature that will produce a 2 °F drop in the saturation temperature. Then, using the formula given at the bottom of the ASHRAE table (reproduced above as Equation 5), the drop in saturation temperature, per 10 ft of line, for a 2.5 ton capacity can be calculated.

As indicated, this is similar to the procedure followed in Section 4.2 for calculating the liquid-line pressure drop, but a slight difference here is that, for lines containing vapor, when the condensing temperature differs from 105 °F, it is necessary to multiply the capacity given in the table by a correction factor. A table of these factors, for suction and discharge lines, is given in Note 4 of the ASHRAE Table. For 120 °F condensing and a suction line, the factor is 0.9. Also, it is noted again that the ratio "Actual L_e / Table L_e " equals 0.1 since 10 ft of extra refrigerant line (rather than 100 ft) is being used as the standard amount in the tables. Finally, because a drop of 5 °F in saturation temperature is equivalent to a 7.0 psi drop in pressure, the calculated ΔT is multiplied by 7/5 = 1.4 to get the calculated ΔP.

Table 7 shows the inputs and the results of this procedure.

Table 7. Pressure Drops Per 10 ft of Suction Line at 40 °F, 2.5 Tons Capacity, R-22

Refrigerant Line O.D. Type L Copper, in.	0.500	0.625	0.875	1.125
Capacity for a 2 °F drop in saturation temperature, tons[1]	0.54	1.0	2.6	5.2
Baseline Capacity, tons[2]	2.5	2.5	2.5	2.5
Calculated saturation ΔT, °F[3]	3.2	1.0	0.19	0.05
Calculated pressure drop, psi[4]	4.5	1.4	0.27	0.07

1. ASHRAE Refrigeration Handbook, 1998, Chapter 2, Table 3, including 0.90 correction factor.
2. Assumed. 3. Using Equation 5. 4. Using 1.4 psi/°F change in saturation temperature.

Note that the pressure drops in the suction line are about 25 times greater than those in the liquid line, which is not surprising in view of the higher specific volume of the vapor.

4.6 Magnitude of Temperature Changes Due to Heat Transfer, Liquid Line, Heating Mode (HTL)

The liquid line in the heating mode is the same physical section of tubing that is the liquid line in the cooling mode, but the direction of flow is reversed. Because the liquid line is physically identical in both modes, the temperature changes for the heating mode will be calculated using the results for the cooling mode as a basis, and then including the effects of the reduced refrigerant flow rate (caused by the lower evaporating temperature assumed in the heating mode) and of the greater assumed temperature difference between the warm condensed refrigerant and the space surrounding the refrigerant line.

In any real system, the temperature changes in these lines will be small compared with the temperature differences across them, and so it should be a reasonable approximation to say that the temperature change in the excess refrigerant line is proportional to the residence time of the liquid in this portion of the line, and that in turn will be inversely proportional to the flow rate. Because of the lower evaporating temperature in the heating mode (20 °F instead of 40 °F), the flow rate for heating is assumed to equal 70% of that in cooling, reflecting the difference in densities of saturated vapor entering the compressor in the ideal cycle. The temperature differences found in the cooling mode therefore need to be multiplied by 1/0.70 or 1.43 to account for this increased residence time.

The temperature difference between the refrigerant in the line (leaving the condenser at 120 °F) and the outside (assumed to equal 40 °F) is 80 °F, which is twice the temperature difference assumed in the cooling mode.

Taking these two effects together, the temperature changes in the cooling mode must be multiplied by the factor 1.43 X 2.00 = 2.86. It is recognized that this procedure is an approximation, using the entering temperature difference instead of the log-mean temperature difference, and ignoring any change in heat-transfer film coefficient with flow rate, but considering the limits within which it is applied here, it appears to be reasonable.

Table 8 shows the calculated temperature drops in the liquid line in the heating mode, per 10 ft of excess line, for the conditions assumed.

Table 8. Temperature Drops per 10 ft of Liquid Line at 80 °F Temperature Difference, Heating Mode, 2.5 Ton System, R-22, Type L Copper Pipe.

Pipe O.D., in.	0.500	0.625	0.875	1.125
Pipe I.D., in.	0.430	0.545	0.785	1.025
Nominal Size	3/8"	1/2"	3/4"	1"
Insulation wall thickness, in.	Uninsulated	Uninsulated	Uninsulated	Uninsulated
Temperature drop per 10 ft excess line, °F[1]	3.37	4.23	5.91	7.60

1. Table 3 values X 2.86.

4.7 Magnitude of Pressure Drops Due to Added Tube Length, Liquid Line, Heating Mode (HPL)

Following the plan of relating heating-mode effects to those calculated for the cooling mode, the pressure drops in the heating mode will be less than in cooling because of the reduced refrigerant flow rate. Since the assumed flow rate in heating is 70% of that in cooling, and the pressure drop is proportional to the square of the flow rate (theoretically) or to the 1.8 power (following Table 3 of Chapter 2 of the 1998 ASHRAE Refrigeration Handbook), the pressure drops in heating will be approximately 49% to 52% of those in cooling. A factor of 50% is therefore applied to the cooling-mode pressure drops obtained in Section 4.2, and these are displayed in Table 9.

Table 9. Pressure Drops Per 10 ft of Liquid Line in the Heating Mode, 2.5 Tons Capacity, R-22.

Refrigerant Line O.D. Type L Copper, in.	0.500	0.625	0.875	1.125
Cooling-mode pressure drop psi[1]	0.18	0.06	0.01	0.002
Heating-mode pressure drop, psi[2]	0.09	0.03	0.005	0.001

1. From Table 4. 2. Based on 70% of cooling-mode refrigerant flow.

4.8 Magnitude of Pressure Changes Due to Coil Elevation, Liquid Line, Heating (HEL)

In the heating mode, the indoor coil is the condenser, and any elevation of this coil will result in a larger pressure drop across the expansion device than would be the case for equally elevated indoor and outdoor coils. This change in pressure drop will be equal in magnitude but opposite in sign to that found in the cooling mode. Representative corrections to the pressure drop across the expansion device, for the heating mode, are shown in Table 10.

Table 10. Altered Pressure Changes Across Expansion Device for Several Indoor Coil Locations (Outdoor Coil at Ground Level), Heating Mode, Liquid Line, R-22.

Indoor Coil Location	Basement, Typical	Garage, Typical	Attic, One-Story Home	Attic, Two-Story Home
Indoor Coil Elevation (ft)	-4	4	8	16
Change in Pressure Drop Across Expansion Device, psi[1]	-1.9	+1.9	+3.8	+7.5

1. Based on 0.47 psi per foot of elevation.

4.9 Magnitude of Temperature Changes Due to Heat Transfer, Vapor (Discharge) Line, Heating Mode (HTV)

As indicated above and shown in Figure 11, the major effect of heat transfer from the warm discharge line into the cooler regions (whether in a buffer space or outdoors) will be to reduce the heating capacity. This will reduce efficiency, since the work of compression per cycle is not affected. This is taken up in the next section. Here it is the amount of temperature drop that needs to be calculated.

This is benchmarked against the cooling-mode values in much the same way as was employed for the liquid line. The refrigerant flow rate is 70% of the cooling-mode value, resulting in a 43% increase in residence time, relative to the cooling mode. The discharge-line temperature under the assumption of an ideal cycle is 156 °F. The temperature in the space surrounding this line is assumed to be 40 °F. The temperature difference is therefore 116 °F. This compares with a 40 °F temperature difference for the suction line in the cooling-mode case. Thus, the vapor-containing line (discharge in heating, suction in cooling) is subjected to a temperature difference that is 116/40 or 2.9 times as great in heating as in cooling. Combining this with the effect of the reduced refrigerant flow rate, a factor of 2.9 X 1.43 = 4.15 must be applied to the cooling-mode temperature rises to get the corresponding heating-mode temperature drops. The calculations relative to the cooling-mode results are shown in Table 11.

Table 11. Temperature Drops per 10 ft of Discharge Line at 156 °F, Heating Mode, 2.5 Ton System, R-22, Type L Copper Pipe.

Pipe O.D., in.	0.500			0.625			0.875			1.125		
Pipe I.D., in.	0.430			0.545			0.785			1.025		
Nominal Size	3/8"			1/2"			3/4"			1"		
Insulation wall thickness, in.	No	0.5	1.0	No	0.5	1.0	No	0.5	1.0	No	0.5	1.0
Cooling-mode temperature rise per 10 ft of line, °F[1]	1.09	0.48	0.35	1.36	0.55	0.38	1.90	0.69	0.47	2.45	0.82	0.55
Heating-mode temperature drop per 10 ft of line, °F[2]	4.5	2.0	1.4	5.6	2.3	1.6	7.9	2.9	2.0	10.2	3.4	2.3

1. From Table 6. 2. Using 4.15 heating/cooling mode ratio.

4.10 Magnitude of Pressure Drops Due to Added Tube Length, Vapor (Discharge) Line, Heating Mode (HPV)

The impact of pressure drops in the discharge line is to add to the pressure that the compressor must deliver, in order that condensing can take place at the design temperature and pressure. As discussed above, this adds to compressor work without adding capacity (Figure 12) and reduces efficiency. In this section, the size of the pressure drop will be determined. In a manner similar to that employed for the suction-line pressure drop in the cooling mode, the information in the 1998 ASHRAE Refrigeration Handbook, Chapter 2, Table 3 (ASHRAE 1998) will be used.

For each of the four refrigerant-line sizes, one first selects from the ASHRAE table the capacity at 20 °F saturated suction temperature that will produce a 2 °F drop in the saturation temperature. Then, using the formula at the bottom of the ASHRAE table (reproduced above as Equation 5), the drop in saturation temperature, per 10 ft of line, for a 2.5 ton capacity is calculated. This is very similar to the procedure followed in Section 4.5 for calculating the suction-line pressure drop in the cooling mode, but the correction factor for a discharge line at 120 °F is 1.1, instead of the 0.9 given for a suction line. Also, it is noted again that the ratio "Actual L_e / Table L_e " equals 0.1

since 10 ft of extra refrigerant line (rather than 100 ft) is being used as the standard amount in the tables. Finally, it is noted that a drop of 1 °F in saturation temperature is equivalent to a 1.9 psi drop in pressure, the calculated ΔT is multiplied by 1.9 to get the calculated ΔP.

Table 12 shows the inputs and the results of this procedure.

Table 12. Calculation of Pressure Drops Per 10 ft of Discharge Line at 20 °F Saturated Suction, Heating Mode, 2.5 Tons Capacity (Cooling), R-22

Refrigerant Line O.D. Type L Copper, in.	0.500	0.625	0.875	1.125
Capacity for a 1 °F drop in saturation temperature, tons[1]	0.91	1.7	4.5	9.1
Baseline Capacity, tons[2]	2.5	2.5	2.5	2.5
Calculated saturation ΔT, °F[3]	0.62	0.20	0.035	0.010
Calculated pressure drop, psi[4]	1.17	0.38	0.07	0.02

1. ASHRAE Refrigeration Handbook, 1998, Chapter 2, Table 3, interpolated between -40 °F and 40 °F saturated suction, and including 1.10 correction factor. 2. Assumed.
3. Using Equation 5, with 10 ft of line. 4. Using 1.9 psi/°F drop in saturation temperature.

Note that the pressure drops in the discharge line are about 15 times greater than those in the liquid line.

5 EFFICIENCY IMPACTS OF PRESSURE AND TEMPERATURE CHANGES IN EXCESS REFRIGERANT LINES

In this section, the pressure and temperature changes in excess refrigerant lines that were quantified in the previous section will be applied to the ideal refrigeration cycle as indicated in the section before. Generally, the impact of these pressure and temperature changes on the work of compression and/or on the heat uptakes in the condenser and evaporator will be used to obtain a ratio of modified-cycle COP to ideal-cycle COP. This ratio will then gauge the efficiency impact of the effect under study.

5.1 Efficiency Impact of Temperature Changes Due to Heat Transfer, Liquid Line, Cooling Mode (CTL)

The first-order consequence of heat transfer from the liquid refrigerant outside the conditioned space is to add subcooling, which is a beneficial effect that adds directly to the cooling capacity (Figure 5). The additional enthalpy removed per pound of refrigerant equals the temperature drop multiplied by the specific heat of saturated liquid R-22 at 120 °F, or 0.332 Btu/lbm-°F. The first-order impact on efficiency is then equal to the ratio of this added enthalpy to the enthalpy change in the evaporator in the ideal cycle (62.5 Btu/lbm). The calculations for the four refrigerant-line sizes considered in this report are summarized in Table 13.

Table 13. Efficiency Impacts of Temperature Drops per 10 ft of Liquid Line at 40 °F Temperature Difference, Cooling Mode, 2.5 Ton System, R-22, Type L Copper Pipe.

Pipe O.D., in.	0.500	0.625	0.875	1.125
Pipe I.D., in.	0.430	0.545	0.785	1.025
Nominal Size	3/8"	1/2"	3/4"	1"
Insulation	Uninsulated	Uninsulated	Uninsulated	Uninsulated
Temperature drop per 10 ft excess line, per Section 4.1, °F[1]	0.95	1.16	1.54	1.89
Subcooling enthalpy, Btu/lbm[2]	0.32	0.39	0.51	0.63
Percent improvement in efficiency [3]	+0.51	+0.62	+0.82	+1.01

1. From Table 3. 2. Using 0.332 Btu/lbm-°F.
3. Ratio of subcooling enthalpy to ideal-cycle enthalpy uptake in evaporator (62.5 Btu/lbm)X100.

Although an efficiency improvement due to this subcooling has been projected, this does not mean that added liquid line should be designed into the system. Among the reasons it should not are the smallness of the effect, the fact that the second-order increase in pressure rise in the compressor has not been factored in, and the cost of the added tube length and refrigerant. Also, as shall be seen, the negative impacts should outweigh any benefit from liquid-line subcooling.

5.2 Efficiency Impact of Pressure Drops Due to Added Tube Length, Liquid Line, Cooling Mode (CPL)

Referring to the calculation of liquid-line pressure drops in Section 4.2, one sees that these are extremely small, compared with the pressure drops per foot of elevation as calculated in Section 4.3. It would seem reasonable to ignore an effect that is smaller than that caused by a 0.5 ft elevation difference between the coils, and it will therefore be dropped from further consideration.

5.3 Efficiency Impact of Pressure Drops Due to Coil Elevation, Liquid Line, Cooling (CEL)

The impact on efficiency of an elevated indoor coil is complex. Assuming that the pressure head of the liquid column of refrigerant is small compared with the pressure difference between the evaporator and the compressor (which will be the case for residential applications of R-22), then the main concern will be that the flow rate of refrigerant would be lower than without the elevated indoor coil. This can be dealt with theoretically in two ways. One is to assume that the pressure drop is accounted for in the system design. An actively controlled expansion device, such as a thermostatic expansion valve, could be used. Such a valve should control the refrigerant flow to reflect design values of superheat and subcooling, and thus any elevation difference between the coils will be accounted for automatically. Another possibility is that, if a fixed expansion device is used, it is selected to account for the elevation difference.

If no design provision is made for the elevation of the indoor coil, then an approximate calculation of the impact may proceed by calculating the added work of compression needed to produce an additional discharge pressure sufficient to balance the refrigerant pressure head caused by the elevation. This added work can then be compared with the work of compression in the ideal cycle. This will overestimate the impact for a number of reasons, but it is used here as a worst-case estimate.

The first step in the cycle calculation is to note that if the pressure at the elevated condenser is to be the same as that in an unelevated condenser in the ideal cycle, then an increased compression pressure equal to the pressure head of the elevation must be added (Figure 13). Since everything is being compared with the ideal cycle, this can be done by first finding the slope of the isentropic compression line passing through the saturation condition at 40 °F. In the P-H diagram, the pressure is given on a logarithmic scale, so the line of constant entropy is really a line on a $\log_{10}P$ vs. H plane. A line on this plane will have a slope m, where m is given by the rise in $\log_{10}P$ divided by the run in H. From the 1985 ASHRAE Handbook of Fundamentals, Chapter17, Figure 6, the slope of this line, for s=0.22 Btu/lbm-R, can be obtained graphically. A tenfold increase in pressure results in an increase in $\log_{10}P$ of one unit, and this is accompanied by an increase in H of 25 Btu/lbm. So the slope m is equal to 1/25 or 0.04.

The change in enthalpy ΔH caused by the increase in compression pressure can be calculated by noting that $m = \Delta \log_{10}P / \Delta H$. If the relationship $d \log_{10}P / dP = 0.434/P$ is used to approximate $\Delta \log_{10}P$, then $\Delta H = 0.434 \, \Delta P / (mP_c)$, where P_c is the ideal-cycle condensing pressure. Using m=0.04 as found above and $P_c = 274.8$ psia, one obtains $\Delta H = 0.039 \, \Delta P$

Table 14 shows the calculation of maximum efficiency impacts using the above equation. The percentage impact is derived from the ratio of increased compressor work to the baseline compressor work in the ideal cycle, ignoring effects over the rest of the cycle. It is likely that this calculation overestimates the impact, because the actual operating point would settle down somewhere in between the baseline condensing pressure and the value assuming a full compensation for the elevation pressure.

Again, this impact can be designed out of the system by using an actively controlled expansion device or by sizing a fixed expansion device to take the elevation pressure difference into account.

Table 14. Maximum Efficiency Impact of Indoor Coil Elevation, Uncompensated In Expansion Device Selection, for Several Indoor Coil Locations (Outdoor Coil at Ground Level), Cooling Mode, Liquid Line, R-22.

Indoor Coil Location	Basement, Typical	Garage, Typical	Attic, One-Story Home	Attic, Two-Story Home
Indoor Coil Elevation (ft)	-4	4	8	16
Change in Pressure Drop Across Expansion Device, from Section 4.3, psi[1]	1.9	-1.9	-3.8	-7.5
Added Compressor Work, Btu/lbm[2]	-0.07	0.07	0.15	0.29
Percent Impact on Efficiency3	+0.5	-0.5	-1.2	-2.2

1. From Table 5. 2. Using 0.039 Btu/lbm-psi.
3. Minus Added Compressor Work/Ideal Cycle Compressor Work (12.9 Btu/lbm) X 100

5.4 Efficiency Impact of Temperature Rise Due to Heat Transfer, Vapor (Suction) Line, Cooling Mode (CTV)

The efficiency impact of heat transfer into the suction line can be assessed by obtaining the enthalpy and entropy at 40 °F saturated vapor (pressure 83.25 psia) and 80 °F superheated vapor at the same pressure. Next one follows the respective lines of constant entropy up and to the right on the P-H diagram, until the pressure of saturation at 120 °F is reached, and the respective enthalpy values read off.

The change in enthalpy going from 40 °F to 80 °F at the 40 °F saturation pressure (83.25 psia) is 115.2 - 108.2 = 7.0 Btu/lbm. This can be related to the specific heat because the change in enthalpy caused by each °F rise in temperature is equal to the specific heat. The average specific heat in the 40 °F - 80 °F range is then (115.2 - 108.2)/(80 - 40) = 0.175 Btu/lbm-°F, essentially the same as the value at saturation.

Table 15. Efficiency Impacts of Suction Line Temperature Rise, per 10 ft, 2.5 Ton Capacity, 40 °F Evaporating, R-22, Type L Copper Refrigerant Line

Pipe O.D., in.	0.500			0.625			0.875			1.125		
Insulation wall thickness, in.	No	0.5	1.0	No	0.5	1.0	No	0.5	1.0	No	0.5	1.0
Temperature rise per 10 ft of line, °F [1]	1.09	0.48	0.35	1.36	0.55	0.38	1.90	0.69	0.47	2.45	0.82	0.55
Added compressor work, Btu/lbm[2]	3.3 X 10^{-2}	1.5 X 10^{-2}	1.1 X 10^{-2}	4.1 X 10^{-2}	1.7 X 10^{-2}	1.1 X 10^{-2}	5.7 X 10^{-2}	2.1 X 10^{-2}	1.4 X 10^{-2}	7.4 X 10^{-2}	2.4 X 10^{-2}	1.6 X 10^{-2}
Ideal cycle compressor work, Btu/lbm[3]	<- - - - - - - - - - - 12.9 - - - - - - - - - - ->											
Efficiency impact (%) [4]	-0.2	-0.1	-0.1	-0.3	-0.1	-0.1	-0.4	-0.2	-0.1	-0.5	-0.2	-0.1

1. From Table 6. 2. Using 0.03 Btu/lbm-°F. 3. From Section 2.2.
4. Percent Efficiency Impact = - Added Work /(Ideal Work + Added Work) X 100

At the 120 °F saturation pressure (274.8 psia) the change in enthalpy going between the same two entropy values as obtained in the above, i.e., from 0.220 to 0.233 Btu/lbm-°F , is 8.2 Btu/lbm. This means that the increase in compressor work entailed in an original 40 °F of superheat is 8.2 - 7.0 = 1.2 Btu/lbm. The increase in compressor work per degree of superheat is therefore 1.2/40 = 0.03 Btu/lbm-°F . For each pipe size and insulation level considered in Table 15, it is now possible to calculate the impact of superheat on compressor work. The percentage change over the baseline ideal cycle is used as a proxy for the percentage change in a real cycle.

The efficiency effect is largest for the largest-diameter line. This is as expected, given the larger heat-transfer area of the larger lines.

5.5 Efficiency Impact of Pressure Drops Due to Added Tube Length, Vapor (Suction) Line, Cooling Mode (CPV)

The calculation here is similar to that carried out in Section 5.3, but here the focus is on the suction side of the compressor. The first step is to translate the pressure drop into an increased enthalpy of compression. Since everything is being compared with the ideal cycle, this can be done by first finding the slope of the isentropic compression line passing through the saturation condition at 40 °F (Figure 14.) In the P-H diagram as shown in the ASHRAE Handbook of Fundamentals, the pressure is given on a logarithmic scale, so the line of constant entropy is really a line on a $\log_{10}P$ vs. H plane. A line on this plane will have a slope m given by the rise in $\log_{10}P$

divided by the run in H. Because the lines of constant entropy are not quite straight, a secant passing through the s=0.220 isentrope at P=83.2 psia (40 °F saturation) and 274.8 psia (120 °F saturation) is used. The slope of this secant, for s=0.220 Btu/lbm-R, can be obtained graphically. A tenfold increase in pressure results in an increase in $\log_{10}P$ of one unit, and this is accompanied by an increase in H of 25 Btu/lbm. So the slope m is equal to 1/25 or 0.040.

It now remains to calculate the change in enthalpy entailed in going from line 0 to line 1 in Figure 14. If $y = \log_{10}P$, then line 0 is given by the equation

$$y = k + mH_0 \tag{6}$$

where k is some constant, and line 1 is given y the equation

$$y = k + mH_1 - 0.434\ \Delta P / P_e \tag{7}$$

where the relationship $d \log_{10}P / dP = 0.434/P$ was used to approximate $\Delta \log_{10}P$, with P_e being the pressure at 40 °F saturation (83.25 psia). Setting these equations equal to each other gives the separation in H:

$$H_1 - H_0 = 0.434\ \Delta P /(m\ P_e) = 0.130\ \Delta P\ . \tag{8}$$

The impact on the ideal cycle efficiency can be obtained by taking the ratio of compressor work in the ideal cycle to the compressor work in the modified cycle, i.e., the ideal compressor work plus the value of $H_1 - H_0$ for the particular line size in question. Results of this calculation are shown in the third row of Table 16.

So far, this calculation has ignored any change in slope of the constant-entropy lines as one moves away from the saturation line. The slope of these lines actually decreases as one increases in H (and s), and this will add to the compressor work. This effect can be estimated by noting that a rise in entropy from 0.22 to 0.24 Btu/lbm-°F adds ~2 Btu/lbm to the difference in enthalpy when one goes from 83.2 psia (40 °F saturation) to 274.8 psia (120 °F saturation). The fraction of this additional entropy that will be gained in the isentropic compression can be estimated by interpolation of the decrease in $\log_{10}P$ in the excess portion of the suction line against the decrease in $\log_{10}P$ in going from the s=0.22 to the s=0.24 isentrope. In both cases, these differences are taken straight down the line of constant enthalpy that crosses the vapor saturation line at 40 °F (H=108.2 Btu/lbm). The fractional shift of the compression line, over the range spanned by the s=0.22 and s=0.24 isentropes, is given by

$$\frac{\log_{10}(83.2) - \log_{10}(83.2-\Delta P)}{\log_{10}(83.2) - \log_{10}(32)}$$

since H=108.2 passes through s=0.22 at P=32 psia. This fractional shift is then multiplied by the additional enthalpy change for a shift all the way from s=0.22 to 0.24, i.e., 2 Btu/lbm, to obtain an added or "correction" ΔH:

$$\Delta H_{correction} = 2\,\frac{\log_{10}(83.2) - \log_{10}(83.2-\Delta P)}{\log_{10}(83.2) - \log_{10}(32)} \tag{9}$$

This added compressor work is shown in the fourth row of Table 16. The effect of the total added compressor work on system efficiency is shown in the bottom row of this table.

5.6 Efficiency Impact of Temperature Changes Due to Heat Transfer, Liquid Line, Heating Mode (HTL)

As indicated in Section 3.6, the impact of any heat loss from the liquid line in the heating mode would be to provide or add to condenser subcooling (Figure 9). In contrast to the cooling situation, where this subcooling adds to capacity, in the heating mode it has no beneficial effect. To first order, it will have no negative impact, either. In second order, there will be an added load on the evaporator, lowering the evaporating temperature and increasing compressor work somewhat.

Table 16. Efficiency Impacts of Pressure Drops Per 10 ft of Suction Line at 40 °F, 2.5 Tons Capacity, R-22

Refrigerant Line O.D. Type L Copper (in.)	0.500	0.625	0.875	1.125
Calculated pressure drop, psi[1]	4.5	1.4	0.27	0.07
Added compressor work, ignoring slope change, Btu/lbm[2]	0.58	0.18	0.035	0.010
Correction for slope change, Btu/lbm[3]	0.12	0.04	0.007	0.002
Total added compressor work, Btu/lbm[4]	0.70	0.22	0.041	0.012
Percent efficiency impact[5]	-5.4	-1.7	-0.32	-0.093

1. From Table 7. 2. Previous line X 0.130 Btu/lbm/psi. 3. From Equation 9.
4. Sum of rows 3 and 4. 5. Minus previous row / (12.9 Btu/lbm) X 100

This second-order effect can be estimated by considering the enthalpy represented by the sensible cooling of liquid refrigerant during the subcooling, with temperature drops as calculated in Section 5.6. This is done by multiplying the temperature drop by the specific heat of saturated liquid at 120 °F (0.332 Btu/lbm-°F). This is then used to obtain a fractional increase in the evaporator heat uptake, compared with the ideal-cycle heat uptake of 60.5 Btu/lbm (see Section 2.3). That in turn is used to estimate the amount by which the evaporating temperature and pressure must be reduced. To do this the assumed temperature difference across the evaporator (which is the difference between 40 °F and 20 °F) is increased by the fractional increase in evaporator heat flow. The decline in evaporator temperature is then used to obtain a decline in evaporator pressure, by prorating the difference in saturated vapor pressure between 20 °F and 15 °F, which is 0.43 psia/ °F. With the decline in evaporating pressure in hand, the added compressor work is calculated as in Section 5.5 for the cooling mode, allowing for a slight decrease in the slope of the isentropic compression line starting at 20 °F as compared with 40 °F. This results in a coefficient of ~0.14 Btu/lbm/psi for the added compressor work, compared with 0.0130 in the cooling-mode calculations. From that an estimate of the efficiency impact can be found by comparing the added compressor work with that for the baseline ideal heating cycle.

The steps in the calculation are illustrated in Table 17. The efficiency impacts are small.

Table 17. Efficiency Impact per 10 ft of Liquid Line at 80 °F Temperature Difference, Heating Mode, 2.5 Ton System, R-22, Type L Copper Pipe.

Pipe O.D., in.	0.500	0.625	0.875	1.125
Temperature drop per 10 ft excess line, °F[1]	3.37	4.23	5.91	7.60
Enthalpy loss, Btu/lbm[2]	1.1	1.4	2.0	2.5
Fractional increase in evaporator heat flow[3]	0.018	0.023	0.033	0.041
Decrease in evaporator temperature, °F[4]	0.36	0.46	0.66	0.82
Decrease in evaporator pressure, psi[5]	0.15	0.20	0.28	0.35
Added compressor work, Btu/lbm[6]	0.021	0.028	0.039	0.049
Percent efficiency impact[7]	-0.1	-0.2	-0.2	-0.3

1. From Table 8. 2. Previous line X specific heat 3. Previous line / 60.5 Btu/lbm
4. Previous line X 20 °F 5. Previous line X 0.43 psia/°F 6. Previous line X 0.14 (cf. Sec. 5.5)
7. Minus previous line / ideal cycle compressor work (17.8 Btu/lbm) X 100

5.7 Efficiency Impact of Pressure Drops Due to Added Tube Length, Liquid Line, Heating Mode (HPL)

These pressure drops are small fractions of a psi, smaller even than in the cooling mode (see Section 5.2). These will therefore be ignored.

5.8 Efficiency Impact of Pressure Drops Due to Coil Elevation, Liquid Line, Heating Mode (HEL)

The impact of an elevated indoor coil in the heating mode will be to increase the pressure at the evaporator. This would increase the flow through a fixed expansion device, the reverse effect from that found in the cooling mode. If cooling efficiency is impaired, does that mean heating efficiency is improved? That seems unlikely, but to quantify what actually happens will require a more detailed calculation than can be carried out here. Perhaps it will be adequate, for the purposes of ASHRAE Standard 152, to assume no significant impact, positive or negative.

5.9 Efficiency Impact of Temperature Changes Due to Heat Transfer, Vapor (Discharge) Line, Heating Mode (HTV)

Temperature drops in the discharge line outside the conditioned space will result in a reduction of heating capacity (Figure 11). The work of compression will still be the same, however, so efficiency will be reduced. The reduction in enthalpy can be calculated by multiplying the temperature drops found in Section 4.9 by the specific heat of superheated vapor at and slightly below 156 °F; this is 0.23 Btu/lbm-°F. This is then compared with the enthalpy change in the condenser in the ideal heating cycle (78.3 Btu/lbm) to obtain an efficiency impact.

Table 18 shows the calculations.

Table 18. Temperature Drops per 10 ft of Discharge Line at 156 °F, Heating Mode, 2.5 Ton System, R-22, Type L Copper Pipe.

Pipe O.D., in.	0.500			0.625			0.875			1.125		
Insulation wall thickness, in.	No	0.5	1.0	No	0.5	1.0	No	0.5	1.0	No	0.5	1.0
Heating-mode temperature drop per 10 ft of line, °F [1]	4.5	2.0	1.4	5.6	2.3	1.6	7.9	2.9	2.0	10.2	3.4	2.3
Enthalpy loss, Btu/lbm [2]	1.04	0.46	0.32	1.29	0.53	0.37	1.82	0.67	0.46	2.35	0.78	0.53
Percent impact on efficiency [3]	-0.8	-0.4	-0.3	-1.0	-0.4	-0.3	-1.4	-0.5	-0.4	-1.8	-0.6	-0.4

1. From Table 11. 2. Previous line X specific heat. 3. Minus previous line /78.3 X 100.

5.10 Efficiency Impact of Pressure Drops Due to Added Tube Length, Vapor (Discharge) Line, Heating Mode (HPV)

The impact of pressure drops in the discharge line in heating is to add to the pressure that the compressor must supply, in order to provide for the design value of condensing pressure (Figure 12). To do this one needs a coefficient relating added compressor work to added pressure. In section 5.3, a value of 0.039 Btu/lbm/psi was obtained, at the high-pressure end (275 psia) of the compression cycle joining 40 °F evaporating to 120 °F condensing. This ratio will be slightly higher for the line joining 20 °F evaporating to 120 °F condensing, and it is estimated here to equal 0.042 Btu/lbm/psi. The added work of compression is then found by multiplying the pressure drops found in Section 4.10 by the coefficient 0.042. The efficiency impact is obtained in the usual manner by comparing this added work to the work of compression in the baseline ideal cycle.

Table 19 shows the calculations.

Table 19. Efficiency Impact of Pressure Drops Per 10 ft of Discharge Line at 20 °F Saturated Suction, Heating Mode, 2.5 Tons Capacity (Cooling), R-22

Refrigerant Line O.D. Type L Copper (in.)	0.500	0.625	0.875	1.125
Calculated pressure drop (psi)[1]	1.17	0.38	0.067	0.019
Added work of compression, Btu/lbm[2]	0.049	0.016	0.0028	0.0008
Percent efficiency impact[3]	-0.28	-0.09	-0.02	-0.004

[1]From Table 12. 2. Previous line X 0.042 Btu/lbm/psi. 3. Minus previous line /17.8 X 100.

6 SUMMARY OF RESULTS FOR BENCHMARK CASE

It may be useful to summarize the results for a set of benchmark cases in which there is no elevation of one coil relative to the other (or the design of the system is such that any such elevation can be ignored) and in which the vapor-containing line is provided with 1/2-inch insulation thickness. For these cases, the efficiency impacts from all causes can be added together for both the cooling and heating modes. This is done in Tables 20 and 21.

Table 20. Summary of Efficiency Impacts for the Cooling Mode, Assuming 1/2-Inch Insulation on Vapor Line and No Impact from Coil Elevation, Per 10 ft Each of Liquid and Vapor Line. Cooling Mode, 2.5 Tons Capacity, R-22

Refrigerant Line O.D. Type L Copper (in.) Percent Efficiency Impact From:	0.500	0.625	0.875	1.125
Temperature Drop, Liquid Line	0.5	0.6	0.8	1.0
Pressure Drop, Liquid Line	0.0	0.0	0.0	0.0
Temperature Rise, Suction Line	-0.1	-0.1	-0.2	-0.2
Pressure Drop, Suction Line	-5.4	-1.7	-0.3	-0.1
Total Efficiency Impact	-5.0	-1.2	+0.3	+0.7

Table 21. Summary of Efficiency Impacts for the Heating Mode, Assuming 1/2-Inch Insulation on Vapor Line and No Impact from Coil Elevation, Per 10 ft Each of Liquid and Vapor Line. Heating Mode, 2.5 Tons Capacity (Cooling), R-22

Refrigerant Line O.D. Type L Copper (in.) Percent Efficiency Impact From:	0.500	0.625	0.875	1.125
Temperature Drop, Liquid Line	-0.1	-0.2	-0.2	-0.3
Pressure Drop, Liquid Line	0.0	0.0	0.0	0.0
Temperature Drop, Discharge Line	-0.4	-0.4	-0.5	-0.6
Pressure Drop, Discharge Line	-0.3	-0.1	-0.0	-0.0
Total Efficiency Impact	-0.8	-0.7	-0.7	-0.9

7 CONCLUSIONS AND RECOMMENDATIONS: TOWARD A REFRIGERANT DISTRIBUTION SECTION IN ASHRAE STANDARD 152

The primary conclusion of this work is that, with the exception of pressure drops in the suction line in the cooling mode, the efficiency impacts are quite small for any reasonable excess lengths of refrigerant line likely to be found in residential applications. Elevation of the indoor coil relative to the outdoor coil may be marginally significant in some cases.

These calculations are intended to serve as a starting point for the development of a section in ASHRAE Standard 152 dealing with refrigerant distribution. Such systems are likely to become more commonly used in future years.

Conceptually, there are at least two possible approaches to a refrigerant distribution efficiency test method. The first is to measure this efficiency directly. The second is to measure those parameters that will provide sufficient basis for calculating an efficiency to a sufficiently good approximation.

It seems likely that measuring the efficiency of refrigerant distribution systems directly will not be feasible in anything other than a highly controlled research environment. Such measurements may be useful as a check on an algorithm that would go into Standard 152, but it is unlikely that they will be useful in a context where time and effort expended are key parameters in an overall optimization of the test method. Direct measurement of efficiency will simply take too much time, equipment, and expertise to be useful in the field.

The alternative is to develop an algorithm for calculating the approximate impact on distribution efficiency of refrigerant line lengths in excess of what is provided for in equipment efficiency test methods. This algorithm would need to provide for the following significant variables:

- Equipment capacity
- Refrigerant line cross-sections and insulation levels
- Excess refrigerant line length
- Refrigerant used
- Evaporating and condensing temperatures
- Temperatures of spaces in which excess refrigerant line length is located, and whether these spaces are in thermal communication with the conditioned space.

The project of developing such an algorithm will be aided by the fact that the required accuracy is reasonably expressed as a fraction of 100% efficiency, not a fraction of the effects themselves. Because the efficiency impacts of excess refrigerant line are mostly small, this will make relatively large fractional errors in most of the individual effects acceptable. The effects due to pressure drops in the suction line in the cooling mode are the major issue. Here, further efforts to refine the accuracy of the calculations could be warranted. If a sufficiently accurate characterization of this effect, coupled with approximations to the others, is achieved, this should result in a useful algorithm for a practical method for evaluating refrigerant distribution efficiency.

The following steps are recommended to proceed with such a project. Each step should be answered satisfactorily and any problem areas addressed before proceeding to the next step.

1. Review the calculations in this report for soundness as to their approximate validity under the conditions for which they were developed.

2. Determine whether any additional impacts may have been ignored.

3. Determine to what extent the characteristics of real cycles, as opposed to the ideal cycle used in this analysis, need to be included.

4. Decide what parameters need to be varied in the Standard 152 algorithm, and what ranges of values these parameters should take on.

5. Extend the calculations in this report to include all these parameters and values.

6. Develop an algorithm for Standard 152 that maximizes accuracy consistent with reasonable measurement effort.

It may be that a quick determination of equipment capacity, length and size of any excess refrigerant lines, where these lines go in the house, and house location (for climate variables) would be enough input for a spreadsheet to determine values for design and seasonal distribution efficiency. In view of the relatively small impacts of everything except pressure drops in the vapor (suction or discharge) line, it appears likely that such a program could succeed.

8 REFERENCES

Andrews, J. 1996. Design Predictions and Diagnostic Test Methods for Hydronic Heating Systems in ASHRAE Standard 152P. BNL- 63200, April. See esp. pp. 11-12 and Appendix 3.

ASHRAE 1999. Standard 152P, Method of Test for Determining the Design and Seasonal Efficiencies of Residential Thermal Distribution Systems. American Society of Heating, Refrigerating, and Air-Conditioning Engineers, Atlanta, GA. June.

ASHRAE 1998 Refrigeration Handbook, Chapter 2, Table 3. American Society of Heating, Refrigerating, and Air-Conditioning Engineers, Atlanta, GA.

9 ACKNOWLEDGMENTS

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Cooling Mode

Thermal Effects

1. liquid line losses -- (long line) increase cooling capacity directly by lowering the evaporator inlet enthalpy (if cooled), decrease in capacity if heated by a hot attic location, increased or decreased flow through expansion valve, effect would be incremental beyond 25 ft lengths, bare tubes.
2. suction line gains -- (long line) decrease refrigerant density, cooling capacity, incremental over 25 ft with standard insulation
3. discharge line effect -- (short line inside unit) included in basic test rating

Pressure Drop Effects

1. liquid line losses -- (long line) extra pressure drop from added length and/or elevation rise from ODU to IDU lowers pressure seen by valve, would tend to lower flow rate, effect would be incremental beyond 25 ft lengths.
2. suction line losses -- (long line) extra pressure drop, decrease refrigerant density, cooling capacity, largest negative system effect, incremental only over 25 ft,
3. discharge line effect -- (short line inside unit) included in basic test rating

Heating Mode

Thermal Effects

1. liquid line losses -- lowering evaporator inlet enthalpy increases heat absorbed from ambient in evaporator -- so improves COP, also increases flow through expansion valve, effect would be incremental beyond 25 ft lengths, bare tubes.
2. discharge line losses-- (long line) a direct capacity loss, incremental over 25 ft with standard insulation
3. suction line effect -- (short line inside unit) included in basic test rating

Pressure Drop Effects

1. liquid line losses -- (long line) extra pressure drop from added length, but possible pressure gain from elevation drop from indoor condenser to outdoor evaporator, added length effect would be incremental beyond 25 ft lengths.
2. discharge line losses -- (long line) extra pressure drop may cause condensing pressure to rise to reject heat at same inlet air temp, incremental only over 25 ft.
3. suction line effect -- (short line inside unit) included in basic test rating

Mini-Splits

For mini-split systems, the outdoor unit is typically located in the same manner as a split system. So, the 10 foot allowance for testing would again seem sufficient.

For the indoor side, the 15 ft allowance may not fully account for the line to the distribution box and the average lenths of lines going to each branch from the distribution.Those branches located above the ceilings could have a significant amount of length in an attic. As for split-systems, the mini-splits have the liquid lines uninsulated and the suction/discharge lines insulated. In mini-split systems that can simultaneously heat and cool with 2-pipe indoor loops, 2 of the 3 indoor pipes are insulated. It was informally estimated by one seller of these units that the difference in performance between installations with the shortest and the longest line lengths is no more than 5% in EER and less than this in capacity.

Obtaining Quantitative Estimates

Quantitative estimates could be made fairly accurately with the DOE/ORNL Heat Pump Model but will require some modifications to the line algorithms to properly model the heat loss/gain and elevation head effects. For mini-splits, the effects would probably need to be modeled as a central indoor unit with the average line lengths from the distribution box. To date, we have used the line loss portions of the model only to match to known measured losses from lab tests -- so our comparisons of performance of the major components wasn't thrown off.

To model these effects in the DOE/ORNL Heat Pump Model, we need to add calculations to the program for line heat loss, insulated and uninsulated, to indoor and outdoor ambients. Then one could add incremental line lengths beyond 10 ft outdoor and 15 ft indoor to assess their effects. Pressure drop effects of various line lengths are already in the program, but do not presently account for oil effects which will give some underestimation of pressure drops. A correction factor for this underestimate could be added. A calculation of elevation head loss or gain also could be straighforwardly added to quantify this effect.

We do have a need to add more explicit line loss heat gain/loss calculations and elevation head effects to our model. However, most of these effects (excluding that of elevation differences) are included in the test ratings for existing split system equipment. My rough guess is that the effect of lines running into an attic would be in the 1-2% range for cooling and less for heating with unitary equipment.

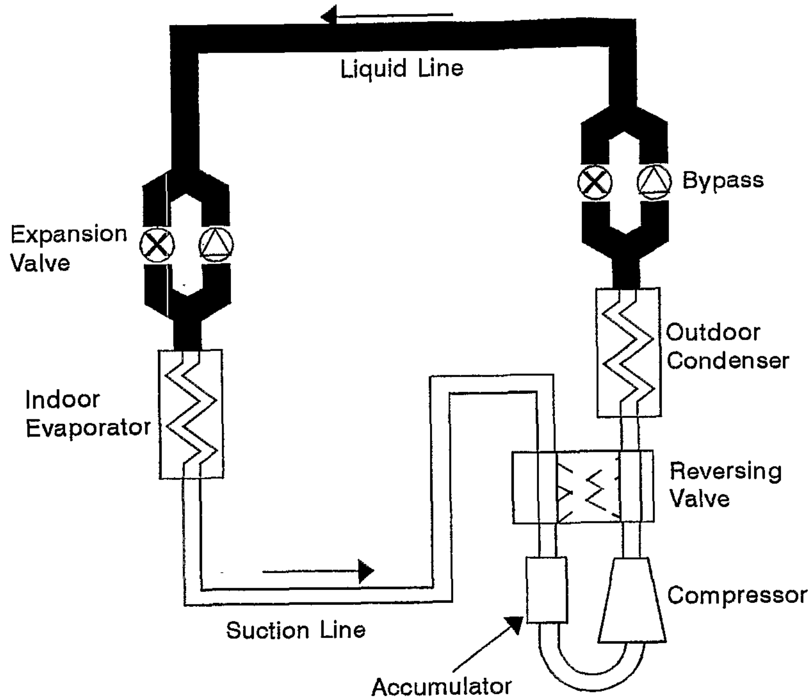

FIGURE 1. SCHEMATIC OF HEAT PUMP IN COOLING MODE.

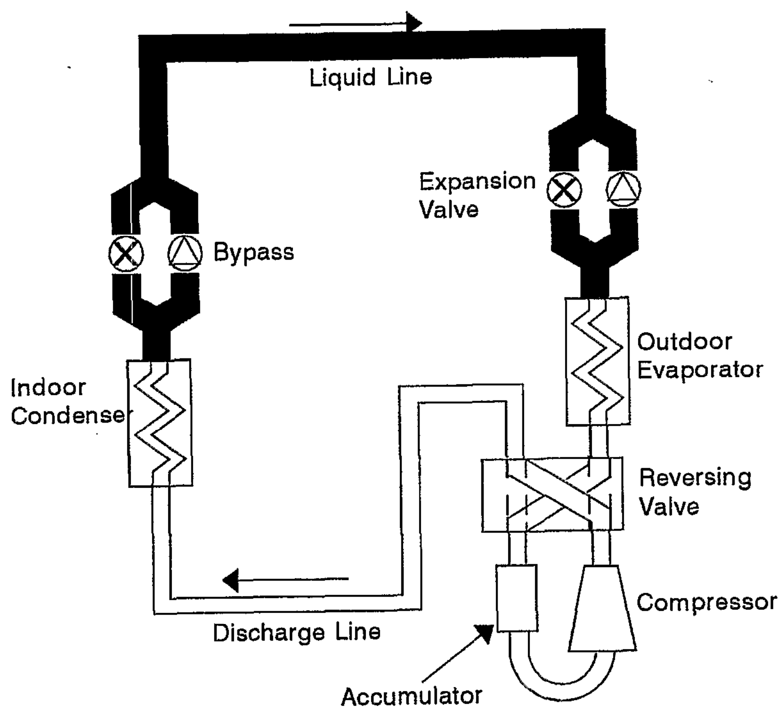

FIGURE 2. SCHEMATIC OF HEAT PUMP IN HEATING MODE.

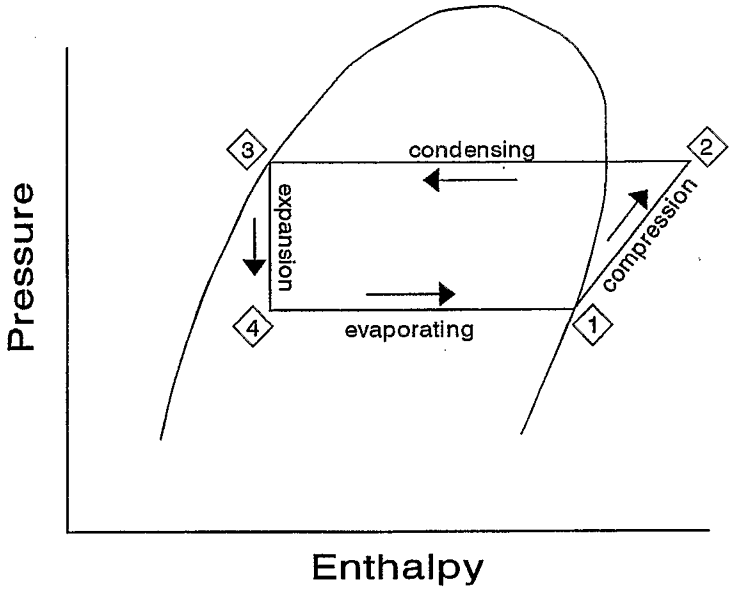

FIGURE 3. IDEAL REFRIGERATION CYCLE

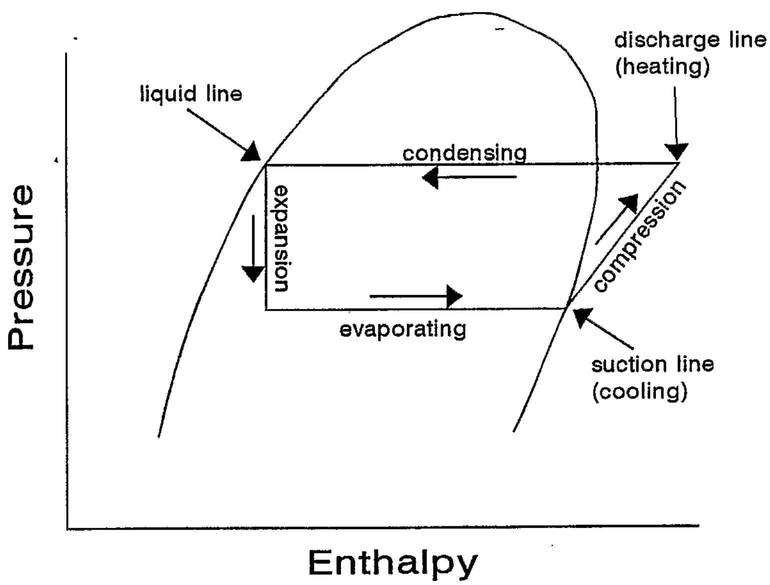

FIGURE 4. REFRIGERANT LINE LOCATIONS ON IDEAL CYCLE

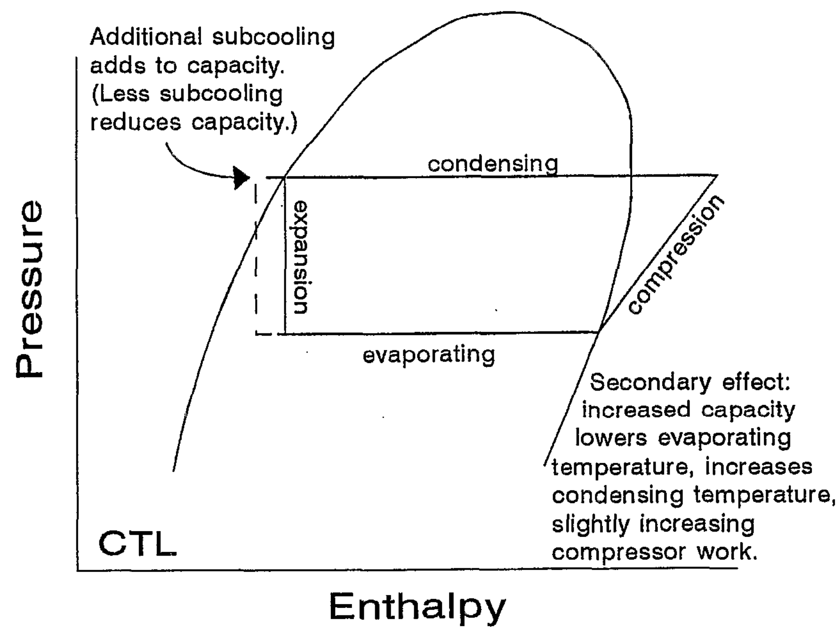

FIGURE 5. CYCLE IMPACT OF TEMPERATURE CHANGE, LIQUID LINE, COOLING MODE

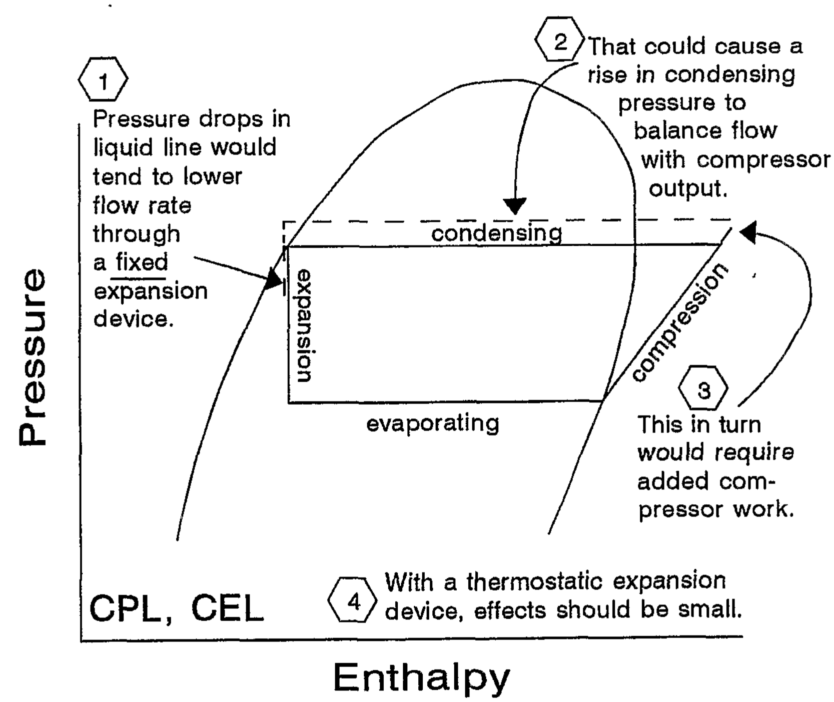

FIGURE 6. CYCLE IMPACT OF PRESSURE DROPS, LIQUID LINE, COOLING MODE

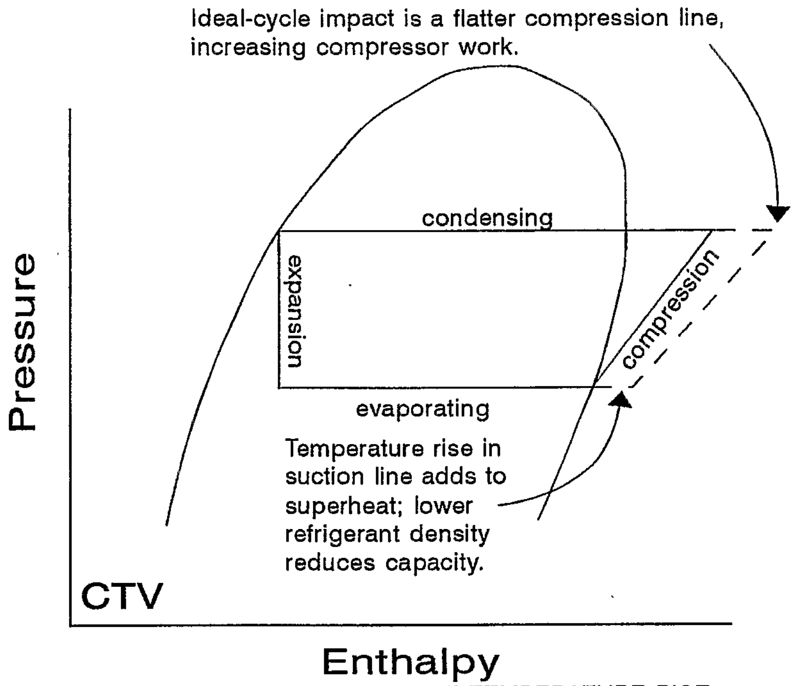

FIGURE 7. CYCLE IMPACT OF TEMPERATURE RISE, SUCTION LINE, COOLING MODE

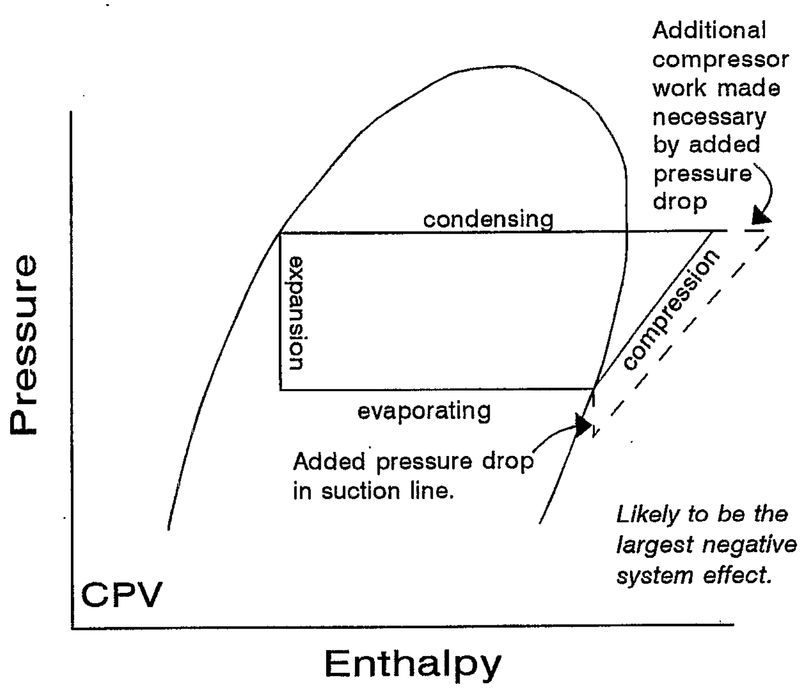

FIGURE 8. CYCLE IMPACT OF PRESSURE DROP, SUCTION LINE, COOLING MODE

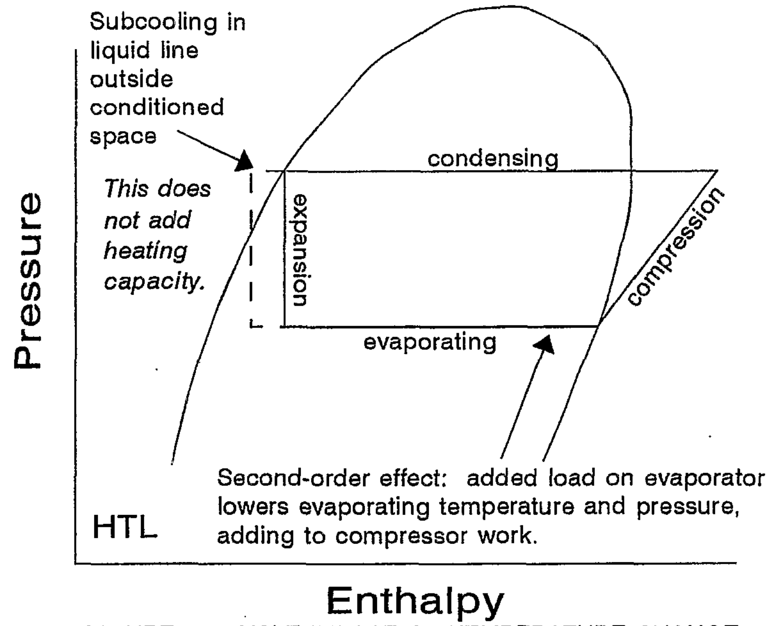

FIGURE 9. CYCLE IMPACT OF TEMPERATURE CHANGE, LIQUID LINE, HEATING MODE

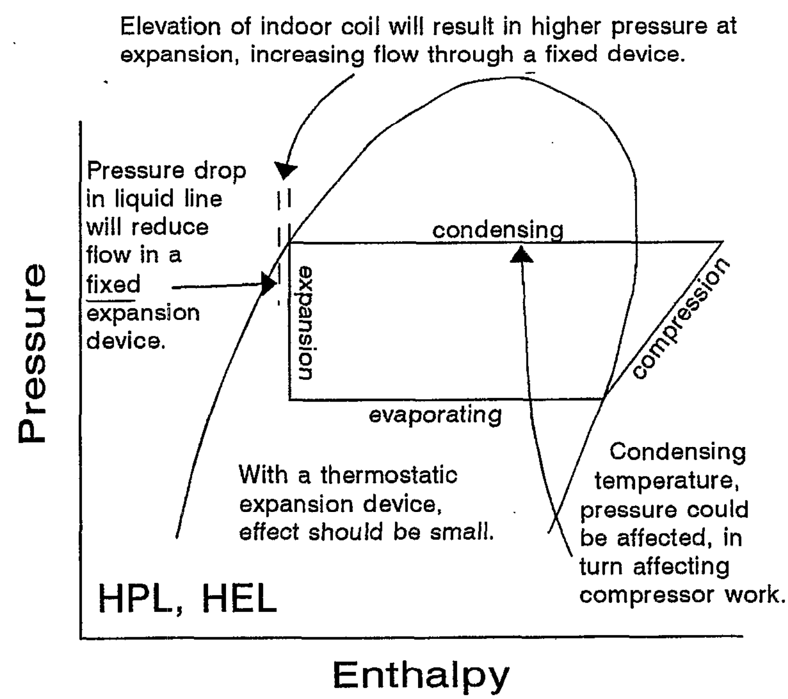

FIGURE 10. CYCLE IMPACT OF PRESSURE CHANGES DUE TO FRICTION, ELEVATION IN LIQUID LINE, HEATING MODE

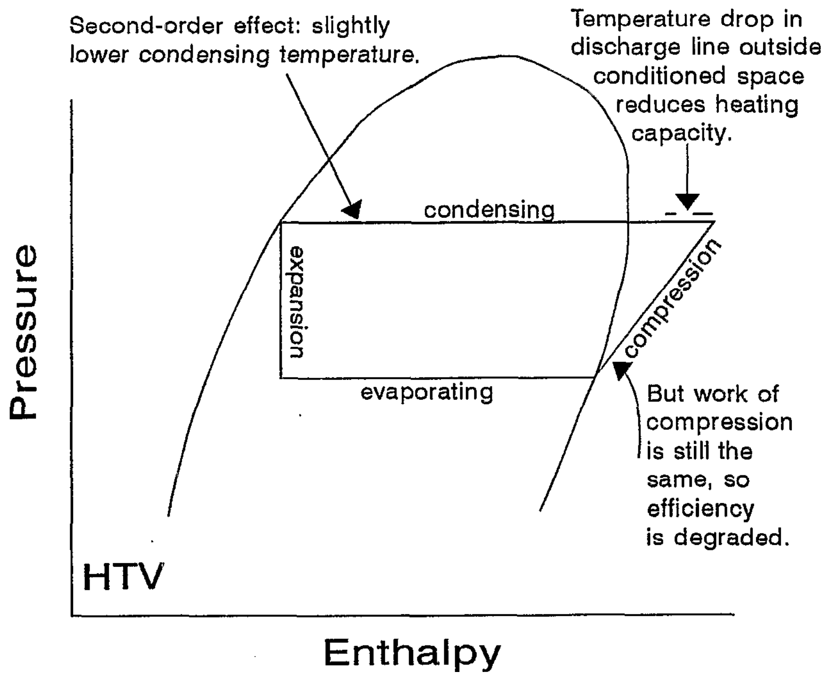

FIGURE 11. CYCLE IMPACT OF TEMPERATURE CHANGE, DISCHARGE LINE, HEATING MODE

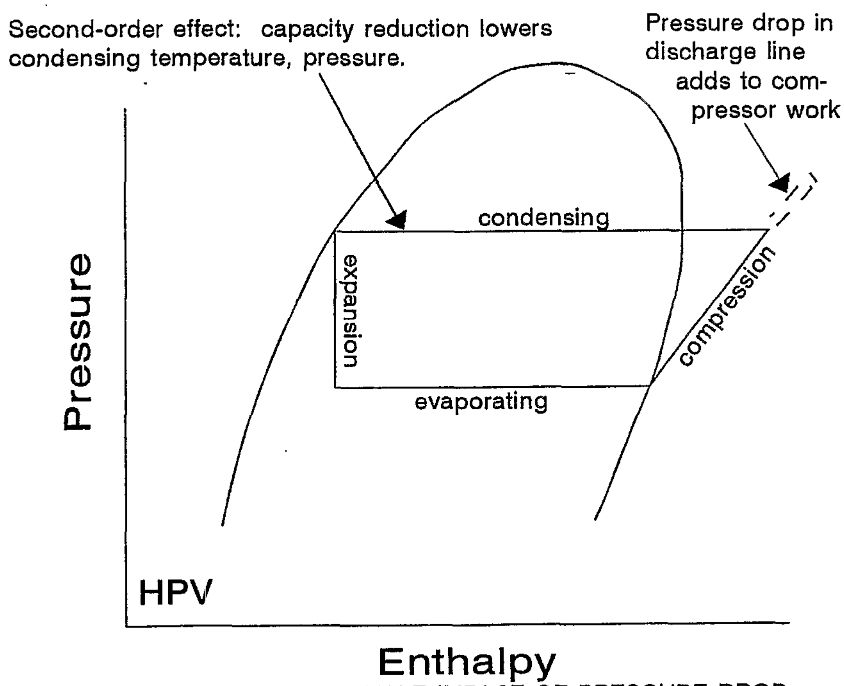

FIGURE 12. CYCLE IMPACT OF PRESSURE DROP, DISCHARGE LINE, HEATING MODE

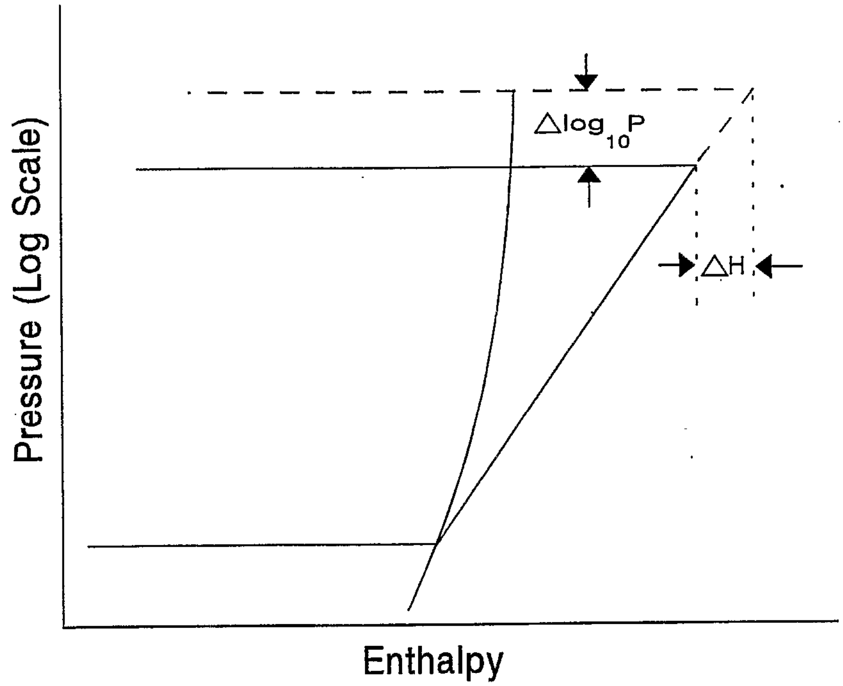

FIGURE 13. ADDED COMPRESSOR WORK DUE TO HIGHER EXIT PRESSURE

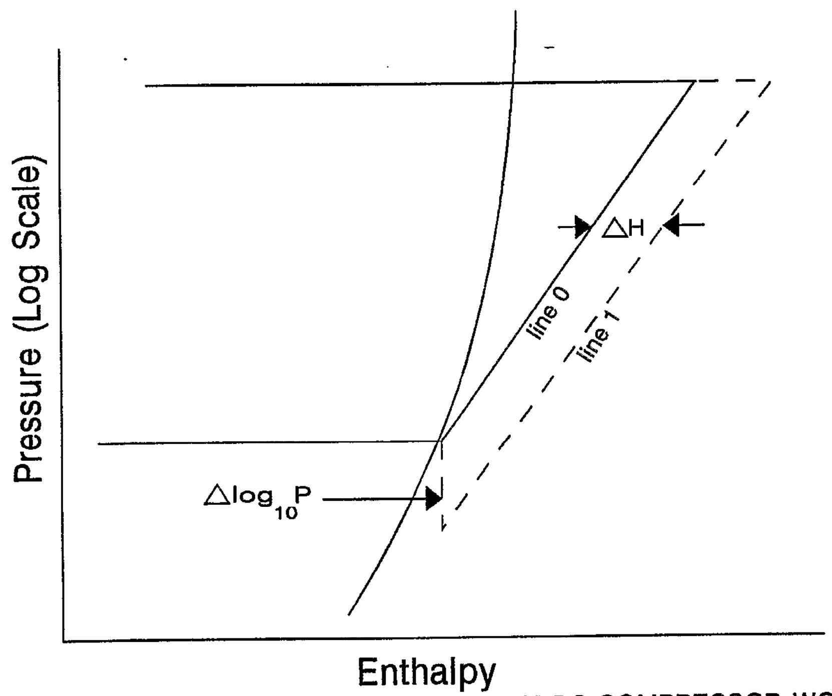

FIGURE 14. SUCTION-LINE PRESSURE DROP ADDS COMPRESSOR WORK, EVEN IF ISENTROPIC COMPRESSION LINES ARE PARALLEL.