**DEVELOPMENT OF A REFRIGERANT DISTRIBUTION SECTION
FOR ASHRAE STANDARD 152**

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ABSTRACT

On the basis of benchmark calculations on refrigerant distribution systems, as described in an earlier report, the impacts on system efficiency of refrigerant line length, sizing, and insulation are translated into a draft format compatible with ASHRAE Standard 152P, Method of Test for Determining the Design and Seasonal Efficiencies of Residential Thermal Distribution Systems. Refrigerant distribution systems use 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 systems that for one reason or another have the indoor and outdoor coils separated by more than 25 ft. The design and seasonal efficiencies of the distribution system are projected on the basis of design cooling capacity and refrigerant line characteristics.

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INTRODUCTION

In a recent draft report titled "Impacts of Refrigerant Line Length on System Efficiency in Residential Heating and Cooling Systems Using Refrigerant Distribution," (Andrews 2000) some baseline calculations were performed to estimate various impacts on system efficiency of long refrigerant distribution lines. Refrigerant distribution refers to "mini-splits" and other types of space heating and cooling equipment that utilize refrigerant lines, rather than ducts or pipes, to transport heat and cooling effect from the outdoor unit to the building spaces where this heat or cooling is used.

Five factors affecting efficiency were studied in each of the space conditioning modes (heating and cooling) for a total of ten factors in all. Temperature changes and pressure drops in each of the two refrigerant lines accounted for four of the factors, with the remaining one being elevation of the indoor unit relative to the outdoor unit. Of these factors, pressure drops in the suction line in cooling showed by far the largest effect.

This report builds on these baseline calculations to develop a possible algorithm for a refrigerant distribution section of ASHRAE Standard 152. It is based on the approximate treatment of the previous report, and is therefore subject to error that might be corrected using a more detailed analysis, possibly including computer modeling and field testing. However, because the calculated efficiency impacts are generally small (a few percent being typical) it may be that the approximate treatment is sufficient.

That question is left open for discussion. The purpose of this report is not to advocate the adoption of the methodology developed, but rather to present it as an option that could either be adopted as-is or used as a starting point for further analysis.

It is assumed that the reader has available and is familiar with ASHRAE Standard 152P and with the previous analysis referred to above.

APPROACH

The approach used was to select those effects that were expected to be most important on the basis of calculations in Andrews 2000, and then to expand these calculations to include a range of cooling capacities (1.0 to 4.0 tons in half-ton increments) rather than just the baseline 2.5-ton capacity. These results were used to develop tables of design distribution efficiency impacts. A final section considers possible efficiency effects from cycling of the distribution system, and uses these results to develop a table of seasonal distribution efficiency impacts. These are then incorporated into a "straw-man" draft section on refrigerant distribution in ASHRAE Standard 152P format.

In Andrews 2000, the effects of coil elevation were found to be difficult to assess. In addition, it was found that any negative impacts could be eliminated through proper system design. For both these reasons, coil elevation is not considered in the algorithm developed below. That is, only temperature changes and pressure drops in the liquid and vapor lines were considered to be candidates for inclusion.

It should be emphasized that the calculations and resulting algorithm are specific to refrigerant R-22. Other refrigerants can be added once consensus is reached regarding R-22 systems.

A note on table numbering. In the discussion below, tables that are similar to tables in Andrews 2000 are numbered according to their place in that document, with a letter afterward. That is, Table 7 in Andrews 2000 (for a 2.5-ton system) is used here to detail parameters for capacities of 1 to 4 tons. The corresponding tables are numbered 7A through 7E, despite the fact that there are no Tables 1 through 6 preceding them.

COOLING MODE

In the benchmark analysis of Andrews 2000, the effect of temperature drops in the liquid line were found to be beneficial to efficiency. It was recognized, however, that this conclusion depended on the detailed assumptions made in the analysis and could not be depended on generally. In the present analysis, therefore, it was decided to treat this impact as neutral, neither assessing a penalty nor giving a credit for the effect. The second effect, pressure drops in the liquid line, were found to be quite negligible in Andrews 2000 and they will be ignored here. That leaves two potentially significant effects to be studied, both in the suction line: pressure drops and temperature changes.

Pressure Drops in the Suction Line

The calculations of Section 4.4 of Andrews 2000, leading to Table 7 for a 2.5-ton system, are here extended to additional capacities in the one- to four-ton range. The tubing nominal sizes in this table are 3/8", 1/2", 3/4", and 1", respectively.

Table 7A. Pressure Drops Per 10 ft of Suction Line at 40 °F, 1.0 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
Capacity, tons ²	1.0	1.0	1.0	1.0
Calculated saturation ΔT , °F ³	0.60	0.20	0.04	0.005
Calculated pressure drop, psi ⁴	0.85	0.28	0.05	0.01

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.

Table 7B. Pressure Drops Per 10 ft of Suction Line at 40 °F, 1.5 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
Capacity, tons ²	1.5	1.5	1.5	1.5
Calculated saturation ΔT , °F ³	1.26	0.41	0.07	0.02
Calculated pressure drop, psi ⁴	1.76	0.58	0.10	0.03

Table 7C. Pressure Drops Per 10 ft of Suction Line at 40 °F, 2.0 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
Capacity, tons ²	2.0	2.0	2.0	2.0
Calculated saturation ΔT , °F ³	2.11	0.70	0.12	0.04
Calculated pressure drop, psi ⁴	2.96	0.98	0.17	0.05

Table 7D. 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
Capacity, tons ²	2.5	2.5	2.5	2.5
Calculated saturation ΔT , °F ³	3.16	1.04	0.19	0.05
Calculated pressure drop, psi ⁴	4.42	1.46	0.26	0.07

Table 7E. Pressure Drops Per 10 ft of Suction Line at 40 °F, 3.0 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
Capacity, tons ²	3.0	3.0	3.0	3.0
Calculated saturation ΔT , °F ³	4.38	1.44	0.26	0.07
Calculated pressure drop, psi ⁴	6.13	2.02	0.36	0.10

Table 7F. Pressure Drops Per 10 ft of Suction Line at 40 °F, 3.5 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	3.5	3.5	3.5	3.5
Calculated saturation ΔT , °F ³	5.78	1.91	0.34	0.10
Calculated pressure drop, psi ⁴	8.09	2.67	0.48	0.14

Table 7G. Pressure Drops Per 10 ft of Suction Line at 40 °F, 4.0 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	4.0	4.0	4.0	4.0
Calculated saturation ΔT , °F ³	7.35	2.43	0.43	0.12
Calculated pressure drop, psi ⁴	10.29	3.40	0.61	0.17

In the same manner as in Section 5.5 of Andrews 2000, leading to Table 16, the calculated pressure drops are used to estimate efficiency impacts for the full range of cooling capacities.

Table 16A. Efficiency Impacts of Pressure Drops Per 10 ft of Suction Line at 40 °F, 1.0 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	0.85	0.28	0.05	0.01
Added compressor work, ignoring slope change, Btu/lbm ²	0.11	0.04	0.01	0.00
Correction for slope change, Btu/lbm ³	0.02	0.01	0.00	0.00
Total added compressor work, Btu/lbm ⁴	0.13	0.04	0.01	0.00
Percent efficiency impact ⁵	-1.0	-0.3	-0.1	0.0

1. From Table 7. 2. Previous line X 0.130 Btu/lbm/psi. 3. From Equation 9 of Andrews 2000. 4. Sum of rows 3 and 4. 5. Minus previous row / (12.9 Btu/lbm) X 100

Equation 9 of Andrews 2000:

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

Table 16B. Efficiency Impacts of Pressure Drops Per 10 ft of Suction Line at 40 °F, 1.5 Tons Capacity, R-22

Refrigerant Line O.D. Type L Copper (in.)	0.500	0.625	0.875	1.125
Calculated pressure drop, psi	1.76	0.58	0.10	0.03
Added compressor work, ignoring slope change, Btu/lbm	0.23	0.08	0.01	0.00
Correction for slope change, Btu/lbm	0.04	0.01	0.00	0.00
Total added compressor work, Btu/lbm	0.27	0.09	0.01	0.00
Percent efficiency impact	-2.1	-0.7	-0.1	0.0

Table 16C. Efficiency Impacts of Pressure Drops Per 10 ft of Suction Line at 40 °F, 2.0 Tons Capacity, R-22

Refrigerant Line O.D. Type L Copper (in.)	0.500	0.625	0.875	1.125
Calculated pressure drop, psi	2.96	0.98	0.17	0.05
Added compressor work, ignoring slope change, Btu/lbm	0.38	0.13	0.02	0.01
Correction for slope change, Btu/lbm	0.08	0.02	0.00	0.00
Total added compressor work, Btu/lbm	0.46	0.15	0.03	0.01
Percent efficiency impact	-3.6	-1.2	-0.2	-0.1

Table 16D. 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	4.42	1.46	0.26	0.07
Added compressor work, ignoring slope change, Btu/lbm	0.57	0.19	0.03	0.01
Correction for slope change, Btu/lbm	0.11	0.04	0.01	0.00
Total added compressor work, Btu/lbm	0.69	0.23	0.04	0.01
Percent efficiency impact	-5.3	-1.8	-0.3	-0.1

Table 16E. Efficiency Impacts of Pressure Drops Per 10 ft of Suction Line at 40 °F, 3.0 Tons Capacity, R-22

Refrigerant Line O.D. Type L Copper (in.)	0.500	0.625	0.875	1.125
Calculated pressure drop, psi	6.13	2.02	0.36	0.10
Added compressor work, ignoring slope change, Btu/lbm	0.80	0.26	0.05	0.01
Correction for slope change, Btu/lbm	0.16	0.05	0.01	0.00
Total added compressor work, Btu/lbm	0.96	0.31	0.06	0.02
Percent efficiency impact	-7.4	-2.4	-0.4	-0.1

Table 16F. Efficiency Impacts of Pressure Drops Per 10 ft of Suction Line at 40 °F, 3.5 Tons Capacity, R-22

Refrigerant Line O.D. Type L Copper (in.)	0.500	0.625	0.875	1.125
Calculated pressure drop, psi	8.09	2.67	0.48	0.14
Added compressor work, ignoring slope change, Btu/lbm	1.05	0.35	0.06	0.02
Correction for slope change, Btu/lbm	0.21	0.07	0.01	0.00
Total added compressor work, Btu/lbm	1.27	0.42	0.07	0.02
Percent efficiency impact	-9.8	-3.2	-0.6	-0.2

Table 16G. Efficiency Impacts of Pressure Drops Per 10 ft of Suction Line at 40 °F, 4.0 Tons Capacity, R-22

Refrigerant Line O.D. Type L Copper (in.)	0.500	0.625	0.875	1.125
Calculated pressure drop, psi	10.29	3.40	0.61	0.17
Added compressor work, ignoring slope change, Btu/lbm	1.34	0.44	0.08	0.02
Correction for slope change, Btu/lbm	0.28	0.09	0.02	0.00
Total added compressor work, Btu/lbm	1.61	0.53	0.09	0.03
Percent efficiency impact	-12.5	-4.1	-0.7	-0.2

The efficiency impacts in the bottom rows of these tables will be incorporated in the draft refrigerant distribution section for ASHRAE Standard 152P.

Temperature Rises in Suction Line

In Section 4.4 of Andrews 2000, the thermal mass flow rate of refrigerant for a 2.5-ton system was found to be 84.3 Btu/°F-h. When this flow rate is recalculated for the range of cooling capacities considered here, the following values are obtained:

Capacity (tons)	1.0	1.5	2.0	2.5	3.0	3.5	4.0
Thermal Mass Flow Rate (Btu/°F-h)	33.8	50.6	67.5	84.3	101.3	118.2	135.0

The temperature rise in the suction line is then calculated using Equation 3 in Section 4.1 of Andrews 2000, if the line is insulated, or Equation 4 if it is uninsulated. In these equations, $T_{out} - T_{in} = -40$ °F, $h_{conv+rad} = 1.75$ Btu/h-ft²-°F for bare tube or 2.4 Btu/h-ft²-°F for insulation, and $L = 10$ ft. The value of d_i takes on values (in inches) of 0.500, 0.625, 0.875, and 1.125, respectively, corresponding to nominal pipe sizes 3/8", 1/2", 3/4", and 1". Thermal insulation thicknesses are 0.5 in. and 1.0 in., with thermal conductivity $k = 0.02$ Btu/h-ft-°F, characteristic of closed-cell polymer foam. These parameters are in line with the baseline assumptions of Andrews 2000. The calculated temperature rises for capacities ranging from 1 to 4 tons are shown in Table 6A - 6G. The calculated temperature rises are then used to calculate efficiency impacts in the same way as in Section 5.4 of Andrews 2000, leading to Tables 15A - 15F.

Table 6A. Temperature Rises per 10 ft of Suction Line at 40 °F, 1.0 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	2.71	1.21	0.88	3.39	1.38	0.98	4.75	1.72	1.18	6.10	2.04	1.37

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

Table 6B. Temperature Rises per 10 ft of Suction Line at 40 °F, 1.5 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	1.81	0.81	0.59	2.26	0.92	0.66	3.17	1.14	0.79	4.08	1.36	0.91

Table 6C. Temperature Rises per 10 ft of Suction Line at 40 °F, 2.0 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	1.36	0.61	0.44	1.70	0.69	0.49	2.38	0.86	0.59	3.06	1.02	0.68

Table 6D. 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	1.09	0.48	0.35	1.36	0.55	0.39	1.90	0.69	0.47	2.45	0.82	0.55

Table 6E. Temperature Rises per 10 ft of Suction Line at 40 °F, 3.0 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	0.90	0.40	0.29	1.13	0.46	0.33	1.58	0.57	0.39	2.04	0.68	0.46

Table 6F. Temperature Rises per 10 ft of Suction Line at 40 °F, 3.5 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	0.78	0.35	0.25	0.97	0.39	0.28	1.36	0.49	0.34	1.74	0.58	0.39

Table 6G. Temperature Rises per 10 ft of Suction Line at 40 °F, 4.0 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	0.68	0.30	0.22	0.85	0.34	0.25	1.19	0.43	0.30	1.53	0.51	0.34

Table 15A. Efficiency Impacts of Suction Line Temperature Rise, per 10 ft, 1.0 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	2.71	1.21	0.88	3.39	1.38	0.98	4.75	1.72	1.18	6.10	2.04	1.37
Added compressor work, Btu/lbm ²	.081	.036	.026	.102	.041	.029	.142	.052	.035	.183	.061	.041
<-----Ideal-cycle compressor work = 12.9 Btu/lbm ³ ----->												
Efficiency impact (%) ⁴	-0.6	-0.3	-0.2	-0.8	-0.3	-0.2	-1.1	-0.4	-0.3	-1.4	-0.5	-0.3

1. From Table 6A.

2. Using 0.03 Btu/lbm-°F.

3. From Section 2.2.

4. Percent Efficiency Impact = - Added Work / (Ideal Work + Added Work) X 100

Table 15B. Efficiency Impacts of Suction Line Temperature Rise, per 10 ft, 1.5 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	1.81	0.81	0.59	2.26	0.92	0.66	3.17	1.14	0.79	4.08	1.36	0.91
Added compressor work, Btu/lbm	.054	.024	.018	.068	.028	.020	.095	.034	.024	.122	.041	.027
<-----Ideal-cycle compressor work = 12.9 Btu/lbm ³ ----->												
Efficiency impact (%)	-0.4	-0.2	-0.1	-0.5	-0.2	-0.2	-0.7	-0.3	-0.2	-0.9	-0.3	-0.2

Table 15C. Efficiency Impacts of Suction Line Temperature Rise, per 10 ft, 2.0 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	1.36	0.61	0.44	1.70	0.69	0.49	2.38	0.86	0.59	3.06	1.02	0.68
Added compressor work, Btu/lbm ²	.041	.018	.013	.051	.021	.015	.071	.026	.018	.092	.031	.020
<-----Ideal-cycle compressor work = 12.9 Btu/lbm ³ ----->												
Efficiency impact (%) ⁴	-0.3	-0.1	-0.1	-0.4	-0.2	-0.1	-0.6	-0.2	-0.1	-0.7	-0.2	-0.2

Table 15D. 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	0.39	1.90	0.69	0.47	2.45	0.82	0.55
Added compressor work, Btu/lbm ²	.033	.014	.011	.041	.016	.012	.057	.021	.014	.074	.025	.016
<-----Ideal-cycle compressor work = 12.9 Btu/lbm ³ ----->												
Efficiency impact (%) ⁴	-0.3	-0.1	-0.1	-0.3	-0.1	-0.1	-0.4	-0.2	-0.1	-0.6	-0.2	-0.1

Table 15E. Efficiency Impacts of Suction Line Temperature Rise, per 10 ft, 3.0 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	0.90	0.40	0.29	1.13	0.46	0.33	1.58	0.57	0.39	2.04	0.68	0.46
Added compressor work, Btu/lbm ²	.027	.012	.009	.034	.014	.010	.047	.017	.012	.061	.020	.014
<-----Ideal-cycle compressor work = 12.9 Btu/lbm ³ ----->												
Efficiency impact (%) ⁴	-0.2	-0.1	-0.1	-0.3	-0.1	-0.1	-0.4	-0.1	-0.1	-0.5	-0.2	-0.1

Table 15F. Efficiency Impacts of Suction Line Temperature Rise, per 10 ft, 3.5 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	0.78	0.35	0.25	0.97	0.39	0.28	1.36	0.49	0.34	1.74	0.58	0.39
Added compressor work, Btu/lbm ²	.023	.010	.008	.029	.012	.008	.041	.015	.010	.052	.017	.012
<-----Ideal-cycle compressor work = 12.9 Btu/lbm ³ ----->												
Efficiency impact (%) ⁴	-0.2	-0.1	-0.1	-0.2	-0.1	-0.1	-0.3	-0.1	-0.1	-0.4	-0.1	-0.1

Table 15G. Efficiency Impacts of Suction Line Temperature Rise, per 10 ft, 4.0 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	0.68	0.30	0.22	0.85	0.34	0.25	1.19	0.43	0.30	1.53	0.51	0.34
Added compressor work, Btu/lbm ²	.020	.009	.007	.026	.010	.008	.036	.013	.009	.046	.015	.010
<-----Ideal-cycle compressor work = 12.9 Btu/lbm ³ ----->												
Efficiency impact (%) ⁴	-0.2	-0.1	-0.1	-0.2	-0.1	-0.1	-0.3	-0.1	-0.1	-0.4	-0.1	-0.1

The efficiency impacts in the bottom rows of these tables will be incorporated in the draft refrigerant distribution section for ASHRAE Standard 152P.

HEATING MODE

The analysis in Andrews 2000 for the heating mode, using a benchmark 2.5-ton system, is extended to a range of cooling capacities from 1 to 4 tons. The refrigerant flow in the heating mode is assumed to equal 70% of that in the cooling mode, on the basis of consideration of relative refrigerant densities, as explained in Section 2.1 of Andrews 2000. In other words, a heat pump is characterized by its cooling capacity, and heating capacity is then derived for the conditions under consideration.

Pressure Drops in the Heating Mode

The calculations of Section 4.10 of Andrews 2000, leading to Table 12 for a 2.5-ton system, are here extended to the additional capacities and presented as Tables 12A - 12G, below. These results are then applied to calculations similar to those in Section 5.10 of Andrews 2000, which led to Table 19 for 2.5 tons (cooling). These results are presented below as Tables 19A - 19G. The tubing nominal sizes are 3/8", 1/2", 3/4", and 1", respectively.

Table 12A. Calculation of Pressure Drops Per 10 ft of Discharge Line at 20 °F Saturated Suction, Heating Mode, 1.0 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	1.0	1.0	1.0	1.0
Calculated saturation ΔT , °F ³	0.12	0.038	0.007	0.002
Calculated pressure drop, psi ⁴	0.23	0.072	0.013	0.004

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.

Table 12B. Calculation of Pressure Drops Per 10 ft of Discharge Line at 20 °F Saturated Suction, Heating Mode, 1.5 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	1.5	1.5	1.5	1.5
Calculated saturation ΔT , °F ³	0.25	0.080	0.014	0.004
Calculated pressure drop, psi ⁴	0.48	0.15	0.027	0.007

Table 12C. Calculation of Pressure Drops Per 10 ft of Discharge Line at 20 °F Saturated Suction, Heating Mode, 2.0 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	2.0	2.0	2.0	2.0
Calculated saturation ΔT , °F ³	0.41	0.13	0.023	0.007
Calculated pressure drop, psi ⁴	0.78	0.25	0.044	0.012

Table 12D. 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	0.066	0.019

Table 12E. Calculation of Pressure Drops Per 10 ft of Discharge Line at 20 °F Saturated Suction, Heating Mode, 3.0 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	3.0	3.0	3.0	3.0
Calculated saturation ΔT , °F ³	0.86	0.28	0.048	0.014
Calculated pressure drop, psi ⁴	1.63	0.53	0.092	0.026

Table 12F. Calculation of Pressure Drops Per 10 ft of Discharge Line at 20 °F Saturated Suction, Heating Mode, 3.5 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	3.5	3.5	3.5	3.5
Calculated saturation ΔT , °F ³	1.13	0.37	0.064	0.018
Calculated pressure drop, psi ⁴	2.15	0.70	0.121	0.034

Table 12G. Calculation of Pressure Drops Per 10 ft of Discharge Line at 20 °F Saturated Suction, Heating Mode, 4.0 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	4.0	4.0	4.0	4.0
Calculated saturation ΔT , °F ³	1.44	0.47	0.081	0.023
Calculated pressure drop, psi ⁴	2.73	0.89	0.154	0.043

The pressure drops from Tables 12A - 12G are transferred to Tables 19A - 19G and the rest of the calculations carried out in the same manner as in Section 5.10 of Andrews 2000.

Table 19A. Efficiency Impact of Pressure Drops Per 10 ft of Discharge Line at 20 °F Saturated Suction, Heating Mode, 1.0 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	0.23	0.072	0.013	0.004
Added work of compression, Btu/lbm ²	0.010	0.003	0.001	0.000
Percent efficiency impact ³	-0.1	0.0	0.0	0.0

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

Table 19B. Efficiency Impact of Pressure Drops Per 10 ft of Discharge Line at 20 °F Saturated Suction, Heating Mode, 1.5 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	0.48	0.15	0.027	0.007
Added work of compression, Btu/lbm ²	0.020	0.006	0.001	0.000
Percent efficiency impact ³	-0.1	0.0	0.0	0.0

Table 19C. Efficiency Impact of Pressure Drops Per 10 ft of Discharge Line at 20 °F Saturated Suction, Heating Mode, 2.0 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	0.78	0.25	0.044	0.012
Added work of compression, Btu/lbm ²	0.033	0.010	0.002	0.001
Percent efficiency impact ³	-0.2	-0.1	0.0	0.0

Table 19D. 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	0.066	0.019
Added work of compression, Btu/lbm ²	0.049	0.016	0.003	0.001
Percent efficiency impact ³	-0.3	-0.1	0.0	0.0

Table 19E. Efficiency Impact of Pressure Drops Per 10 ft of Discharge Line at 20 °F Saturated Suction, Heating Mode, 3.0 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	1.63	0.53	0.092	0.026
Added work of compression, Btu/lbm ²	0.068	0.022	0.004	0.001
Percent efficiency impact ³	-0.4	-0.1	0.0	0.0

Table 19F. Efficiency Impact of Pressure Drops Per 10 ft of Discharge Line at 20 °F Saturated Suction, Heating Mode, 3.5 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	2.15	0.70	0.121	0.034
Added work of compression, Btu/lbm ²	0.090	0.029	0.005	0.001
Percent efficiency impact ³	-0.5	-0.2	0.0	0.0

Table 19G. Efficiency Impact of Pressure Drops Per 10 ft of Discharge Line at 20 °F Saturated Suction, Heating Mode, 4.0 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	2.73	0.89	0.154	0.043
Added work of compression, Btu/lbm ²	0.115	0.037	0.006	0.002
Percent efficiency impact ³	-0.6	-0.2	0.0	0.0

Temperature Drops in Heating Mode

The heating mode temperature drops are calculated from the cooling mode temperature rises, as in Section 4.9 of Andrews 2000, i.e., using a derived conversion factor of 4.15 which takes account of the greater temperature difference across the line in heating and the greater residence time due to the lower refrigerant flow rate. The values from Tables 6A to 6G, multiplied by the 4.15 factor, are inserted into Tables 18A - 18G and the calculations carried on from there in the same manner as in Section 5.9 of Andrews 2000.

Table 18A. Temperature Drops per 10 ft of Discharge Line at 156 °F, Heating Mode, 1.0 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	11.2	5.0	3.7	14.1	5.7	4.1	19.7	7.1	4.9	25.3	8.5	5.7
Enthalpy loss, Btu/lbm ²	2.58	1.15	0.85	3.24	1.31	0.94	4.53	1.63	1.13	5.82	1.96	1.31
Percent impact on efficiency ³	-3.3	-1.5	-1.1	-4.1	-1.7	-1.2	-5.8	-2.1	-1.4	-7.4	-2.5	-1.7

1. 4.15 times cooling mode temperature rise for same system.
2. Previous line X specific heat.
3. Previous line/78.3 X 100

Table 18B. Temperature Drops per 10 ft of Discharge Line at 156 °F, Heating Mode, 1.5 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	7.5	3.4	2.4	9.4	3.8	2.7	13.2	4.7	3.3	16.9	5.6	3.8
Enthalpy loss, Btu/lbm ²	1.72	0.78	0.55	2.16	0.87	0.62	3.04	1.08	0.76	3.89	1.29	0.87
Percent impact on efficiency ³	-2.2	-1.0	-0.7	-2.8	-1.1	-0.8	-3.9	-1.4	-1.0	-5.0	-1.6	-1.1

Table 18C. Temperature Drops per 10 ft of Discharge Line at 156 °F, Heating Mode, 2.0 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	5.6	2.5	1.8	7.1	2.9	2.0	9.9	3.6	2.4	12.7	4.2	4.8
Enthalpy loss, Btu/lbm ²	1.29	0.58	0.41	1.63	0.67	0.46	2.28	0.83	0.55	2.92	0.97	0.64
Percent impact on efficiency ³	-1.6	-0.7	-0.5	-2.1	-0.9	-0.6	-2.9	-1.1	-0.7	-3.7	-1.2	-0.8

Table 18D. 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	1.5	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	0.34	1.29	0.53	0.37	1.82	0.67	0.46	2.35	0.78	0.53
Percent impact on efficiency ³	-1.3	-0.6	-0.4	-1.6	-0.7	-0.5	-2.3	-0.9	-0.6	-3.0	-1.0	-0.7

Table 18E. Temperature Drops per 10 ft of Discharge Line at 156 °F, Heating Mode, 3.0 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	3.7	1.7	1.2	4.7	1.9	1.4	6.6	2.4	1.6	8.5	2.8	1.9
Enthalpy loss, Btu/lbm ²	0.85	0.39	0.28	1.08	0.44	0.32	1.52	0.55	0.37	1.96	0.64	0.44
Percent impact on efficiency ³	-1.1	-0.5	-0.4	-1.4	-0.6	-0.4	-1.9	-0.7	-0.5	-2.5	-0.8	-0.6

Table 18F. Temperature Drops per 10 ft of Discharge Line at 156 °F, Heating Mode, 3.5 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	3.2	1.5	1.0	4.0	1.6	1.2	5.6	2.0	1.4	7.2	2.4	1.6
Enthalpy loss, Btu/lbm ²	0.74	0.34	0.23	0.92	0.37	0.28	1.29	0.46	0.32	1.66	0.55	0.37
Percent impact on efficiency ³	-0.9	-0.4	-0.3	-1.2	-0.5	-0.4	-1.6	-0.6	-0.4	-2.1	-0.7	-0.5

Table 18G. Temperature Drops per 10 ft of Discharge Line at 156 °F, Heating Mode, 4.0 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	2.8	1.2	0.9	3.5	1.4	1.0	4.9	1.8	1.2	6.3	2.1	1.4
Enthalpy loss, Btu/lbm ²	0.64	0.28	0.21	0.80	0.32	0.23	1.13	0.41	0.28	1.45	0.48	0.32
Percent impact on efficiency ³	-0.8	-0.4	-0.3	-1.0	-0.4	-0.3	-1.4	-0.5	-0.4	-1.9	-0.6	-0.4

The efficiency impacts in the bottom rows of these tables will be incorporated in the draft refrigerant distribution section for ASHRAE Standard 152P.

EFFECTS ON SEASONAL DISTRIBUTION EFFICIENCY

So far all the calculations have been taken assuming steady-state operating conditions, with no consideration of the impact of cycling. A detailed model evaluation of the cycling impacts is beyond the scope of this discussion, but it should be possible to get a reasonable benchmark for the difference between the design and seasonal efficiency of the refrigerant distribution system by considering the thermal losses imposed by the thermal mass of the refrigerant piping under cycling conditions. As an approximate treatment, the impact of cycling will be ignored in deriving the design distribution efficiency. This is the same approach that was taken in the forced-air section of Standard 152P. The number of cycles per hour is an important parameter, because that will determine how many times the refrigerant lines warm up (or cool down) during the off-cycles, with consequent loss of useful energy or degradation of efficiency. The correct number of cycles per hour will depend on field data or detailed modeling results. However, analogy with other types of systems would indicate that the reasonable range is probably ~2 to ~3 cycles per hour. A benchmark 2.5 cycles per hour will be used here.

Cooling Mode

In most systems the liquid line is left uninsulated. In the cooling mode, this line carries liquid from the outdoor unit (condenser) to the indoor unit (evaporator). Generally, there is some subcooling in the condenser, and there will be more subcooling in the first 25 ft of line assigned to the equipment in the usual test procedure. One expects, therefore, that in steady-state operation, the temperature of the rest of this line will be at or near the ambient temperature of whatever space the line is located in. Obviously this will not always be the case, particularly if the refrigerant line passes through more than one buffer zone and these zones are at substantially different temperatures. As a benchmark, though, it would seem reasonable to assume that at the time of compressor shutdown, the "excess" portion of the refrigerant line (beyond 25 ft) is essentially at the ambient temperature of the zone surrounding it. If that is so, that portion of the liquid line will neither heat up nor cool down following compressor shutdown, and there will be no efficiency impact due to cycling, attributable to the liquid line.

The suction line, however, is a different case. The suction line is insulated in an effort to limit any temperature rise above the evaporator leaving temperature. In this analysis we have assumed that this line is at 40 °F during compressor operation. Following compressor shutdown, this temperature will begin to relax toward that of the surrounding space. Under the assumptions made for this analysis, this will be 80 °F for refrigerant line run outside the house. For other locations (e.g. crawl space, attic) it will be different. For the benchmark calculation, however, an initial temperature difference of 40 °F will be assumed.

The next step is to calculate the thermal mass of the four sizes of copper refrigerant line considered in the analysis. The specific heat of copper is 0.092 Btu/lbm- °F. The weight and thermal mass per foot of tubes are given in Table 1.

A calculation of the thermal mass of refrigerant vapor inside the tube was carried out. Saturated R-22 at 40 °F was assumed. Its density is 1.52 lbm/ft³, which is about 20 times that of standard air or 1/40 that of water. The specific heat of the refrigerant vapor is 0.175 Btu/lbm- °F. These parameters produce thermal mass values for refrigerant vapor within the tubes ranging from 1.5% to 2.5% of the thermal mass of the tubing.

A similar calculation was done for saturated liquid refrigerant at 120 °F. The density of the liquid R-22 is 67.9 lbm/ft³, slightly greater than that of water. The specific heat is 0.332 Btu/lbm- °F. These parameters give thermal mass values for the liquid refrigerant in the range 130% to 215% of the thermal mass of the tube. (For comparison, if the tubes were filled with water, the thermal mass of the water would range from three to six times that of the tube.)

Table 1. Thermal Properties of Type L Copper Tubes

Nominal Tube Size	3/8"	1/2"	3/4"	1"
O.D. (in.)	0.500	0.625	0.875	1.125
I.D. (in.)	0.430	0.545	0.785	1.025
Mass per unit length (lbm/ft) ¹	0.198	0.285	0.455	0.655
Thermal mass per unit length (Btu/ft-°F):				
of copper tubing ²	0.018	0.026	0.042	0.060
of refrigerant vapor ³	0.0003	0.0004	0.0009	0.0015
of liquid refrigerant ⁴	0.023	0.037	0.076	0.129
Linear thermal conductance U_{linear} (Btu/h-ft-°F)				
bare tube ⁵	0.20	0.30	0.40	0.50
insulated, 1/2 in. polymer foam (k=0.02 Btu/h-ft-°F) ⁵	0.10	0.12	0.14	0.17
Thermal time constant (min.)				
bare tube with liquid inside ⁶	12	13	18	23
insulated tube with vapor ⁶	11	13	18	22

1. Mark's Standard Handbook for Mechanical Engineers, 8th ed., p. 8-172.

2. Previous line X 0.092 Btu/lbm-°F. 3. Saturated, at 40 °F. 4. Saturated, at 120 °F.

5. Using Equation 1 of Andrews 2000. 6. Thermal mass divided by thermal conductance.

We will also need the linear thermal conductance, or linear U value, of the refrigerant lines. This is calculated using Equation 1 of Andrews 2000, for insulated and uninsulated conditions in each size of tubing.

The above information permits us to calculate the thermal time constant of the (insulated) suction line containing refrigerant vapor. Even though--as discussed above--we do not expect the liquid line to play much of a role in cycling impacts in the cooling mode, this will not be the case in heating, and so it will be useful to have the time constant for this line in hand as well. The time constant is equal to the ratio of the thermal mass per unit length to the linear thermal conductance. The values come out about the same for the lighter, but insulated, suction line as for the heavier (with liquid refrigerant) but uninsulated liquid line. As shown in Table 1, these time constants range from about 10 to 20 minutes. Assuming 2.5 cycles per hour with 20% on-time, this means that the refrigerant lines would relax (heat up or cool down) about 60% to 85% of the way from their initial temperature to the temperature of the surrounding space. As a benchmark, if we assume 80% relaxation, this would be equivalent to two complete relaxations per hour. This is a rough estimate to one significant figure only, essentially saying that 2 relaxations per hour is a better number than 1 and probably better than 3.

The quantities in Table 1 all refer to the pipe itself, and are independent of whether the heating or cooling mode is being addressed. Now we wish to estimate the impact of cycling on cooling-mode efficiency. To do this, we'll first determine the heat gain into 10 ft of tube, per hour of "typical cycling," by multiplying the thermal mass per unit length (from Table 1) by 10 ft times 2.0 relaxations per hour times 40 °F, or 800 °F-ft/h. The results of this calculation are shown in Table 2.

To calculate the efficiency impact, this input of unwanted heat is compared with the hourly cooling rate under cycling conditions. Consistent with previous work on Standard 152P, a cycling fractional ontime of 0.2 is assumed. The cooling rate is then 0.2 times the capacity. The efficiency impact is then the ratio of vapor line heating rate divided by the cooling rate, converted to a percentage.

The efficiency impacts are shown in Table 2 for various tube sizes and rated cooling capacities. These impacts are then available for use in calculating cooling seasonal distribution efficiencies in a refrigerant distribution section of ASHRAE Standard 152.

Table 2. Impact of Cycling on Efficiency in the Cooling Mode, Thermal Relaxation of the Vapor Line, Type L Copper Tube, Insulated.

Nominal Tube Size	3/8"	1/2"	3/4"	1"	
O.D. (in.)	0.500	0.625	0.875	1.125	
I.D. (in.)	0.430	0.545	0.785	1.025	
Thermal mass per unit length (Btu/ft-°F) ¹	0.018	0.026	0.043	0.062	
Hourly heat gain per 10 ft of tube (Btu) ²	14.6	21.1	34.3	49.2	
Capacity (tons)	Cooling Rate (Btu/h) ³	Efficiency Impact, Percent ⁴			
1.0	2400	0.6	0.9	1.4	2.0
1.5	3600	0.4	0.6	1.0	1.4
2.0	4800	0.3	0.4	0.7	1.0
2.5	6000	0.2	0.4	0.6	0.8
3.0	7200	0.2	0.3	0.5	0.7
3.5	8400	0.2	0.3	0.4	0.6
4.0	9600	0.2	0.2	0.4	0.5

1. From Table 1, tube plus refrigerant vapor. 2. At 2.5 cycles/hour, 80% relaxation, 40 °F ΔT.

3. At 0.2 cycling factor. 4. (Hourly heat gain per 10 ft tube / cycling cooling rate) X 100

Heating Mode

In the heating mode, the liquid line is assumed to be at approximately 120 °F during steady-state operation. On compressor shutdown, this line and the liquid refrigerant within it will cool toward the temperature of its surrounding ambient. This will subcool the resident refrigerant, which will place a greater load on the evaporator for a transient period when the compressor comes on again. In the limit of a very large evaporator, this would not affect efficiency. For real evaporators, there will be a second-order impact which will be ignored in this analysis.

Cooldown of the vapor (discharge) line, in contrast, represents a direct loss of heat that otherwise would be available to heat the house. It is possible to make a direct comparison of this loss to that which has already been calculated in the cooling mode, by considering that the calculation will be essentially the same except for the effect of the following differences:

- The initial temperature difference between the tube and its surroundings is 80 °F in the heating mode (120 °F inside the tube, 40 °F ambient) as opposed to 40 °F in the cooling mode. This introduces a factor of two relative to cooling.
- Under the assumptions of this analysis, the cooling refrigerant flow rate per ton is 12000 Btu/h ÷ 62.5 Btu/lbm (see Sec. 2.2 of Andrews 2000) or 192 lbm/h. The heating flow rate (Sec 2.3 of Andrews 2000) is then 0.7 X 192 = 134 lbm/h. This yields a heating capacity of 78.3 Btu/lbm X 134 lbm/h = 10500 Btu/h for each 12000 Btu/h of cooling capacity. The ratio of the two benchmark capacities is 0.875.

The efficiency impact in heating should therefore be $2.0/0.875 = 2.3$ times the efficiency impact in cooling, because of the greater temperature drop impinging on a somewhat smaller thermal delivery rate. The efficiency impacts from Table 2 can therefore be translated into heating efficiency impacts via this conversion factor, as shown in Table 3. These impacts are then available for use in calculating heating seasonal distribution efficiencies in a refrigerant distribution section of ASHRAE Standard 152.

Table 3. Impact of Cycling on Efficiency in the Heating Mode, Thermal Relaxation of the Vapor Line, Type L Copper Tube, Insulated.

Nominal Tube Size		3/8"	1/2"	3/4"	1"
O.D. (in.)		0.500	0.625	0.875	1.125
I.D. (in.)		0.430	0.545	0.785	1.025
Cooling tons	Heating Rate (Btu/h) ¹	Efficiency Impact, Percent ²			
1.0	2100	1.4	2.0	3.3	4.7
1.5	3150	0.9	1.3	2.2	3.1
2.0	4200	0.7	1.0	1.6	2.4
2.5	5250	0.6	0.8	1.3	1.9
3.0	6300	0.5	0.7	1.1	1.6
3.5	7350	0.4	0.6	0.9	1.3
4.0	8400	0.3	0.5	0.8	1.2

1. 87.5% of the cycling cooling rate from Table 2. 2. 2.3 times the cooling efficiency impact as shown in Table 2.

REFERENCES

ASHRAE 1999. Method of Test for Determining the Design and Seasonal Efficiencies of Residential Thermal Distribution Systems. Public Review Draft. American Society of Heating, Refrigerating, and Air-Conditioning Engineers, Inc., Atlanta, Ga.

Andrews, J.W. 2001. Impacts of Refrigerant Line Length on System Efficiency in Residential Heating and Cooling Systems Using Refrigerant Distribution. BNL-68550, April 2001.

DRAFT SECTION FOR ASHRAE STANDARD 152P

A draft section on refrigerant distribution systems, in a format suitable for inclusion in ASHRAE Standard 152P, is given below. The efficiency impacts are taken from the tables developed in the body of this report.

A note is in order on the treatment of refrigerant line lengths. For lengths less than 25 ft, it is assumed in line with current equipment efficiency test methods that the impact of the refrigerant line is included in the equipment efficiency. Design and seasonal distribution efficiencies are set equal to 100%. For refrigerant line lengths greater than 25 ft, the first 25 ft are subtracted from the total length to obtain an "excess" length attributable to the distribution system. Design and seasonal distribution efficiencies are then calculated for 10 ft of excess length. For excess lengths other than 10 ft, the final values of distribution efficiency are calculated by raising the value for 10 ft to a power equal to the excess length divided by 10 ft. For example, if the total length of the refrigerant line pair is 55 ft, the excess length is 30 ft. If the distribution efficiency for 10 ft of excess line happens to be, say, 0.960, then the value for the given case would be 0.96 to the third power, which equals 0.885.

In most cases this procedure will not differ significantly from the more obvious choice of linearly pro-rating the distribution efficiency. In the above case, for example, linearizing the problem yields a value of 0.880. However, in rare cases of poorly designed systems with long refrigerant lines, the procedure recommended here will prevent the derivation of excessively low or even negative distribution efficiencies.

The suggested section is numbered 9, to conform to the current version of the standard, which ends with Section 8.

9 REFRIGERANT DISTRIBUTION SYSTEMS

The use of this section is limited to systems comprising an outdoor unit having compressor(s) and refrigerant-to-air heat exchanger(s), an indoor refrigerant-to-air fan/coil unit located within the conditioned space, and refrigerant lines connecting the two units. At present, the procedure covers only one refrigerant, R-22. As other refrigerants come into use, they may be included in future versions of the standard. Four standard outer diameters for the refrigerant vapor line are given in the tables: 0.500 in. (12.7 mm); 0.625 in. (15.9 mm); 0.875 in. (22.2 mm); and 1.125 in. (28.6 mm). These cover the usual range of sizes found in residential equipment. Other sizes within the range may be evaluated by interpolation. Data in tables shall not be extrapolated outside the range provided.

9.1 Instrumentation Specifications

The instrumentation for the diagnostic measurements shall conform to the following specifications.

9.1.1 Length Measurement

Linear dimension shall have an accuracy of $\pm 5\%$ of measured dimension.

9.2 Apparatus

The measurements for this section are limited to lengths and diameters of refrigerant piping. A tape measure and calipers will be sufficient tools in most cases.

9.3 Procedure

The calculation procedure in the following sections shall be used to determine distribution system efficiencies based on

- Refrigerant line length
- Outer diameters of refrigerant lines
- Insulation level on the line used to transport refrigerant vapor
- Rated capacity of the air conditioner or heat pump

9.3.1 Measured Input

The following parameters shall be measured to an accuracy consistent with the specification in Section 9.1.1:

- The length L_{ref} of the refrigerant line pair (liquid and vapor lines) from the outdoor unit to the indoor unit. If $L_{\text{ref}} < 25 \text{ ft (7.5 m)}$, set $L_{\text{ref}} = 25 \text{ ft (7.5 m)}$.
- The fraction F_{out} of the refrigerant line length that is outside the conditioned space.
- The outer diameter d_{vap} of the refrigerant line used to transport vapor (suction line in cooling, discharge line in heating).
- The outer diameter d_{ins} of the insulation surrounding the refrigerant line used to transport vapor. (If this line is uninsulated, d_{ins} is not used in the calculation.)
- The outer diameter d_{liq} of the refrigerant line used to transport liquid (liquid line).

In addition, the following parameter shall be taken from manufacturer's data:

- Rated capacity in the cooling mode.

9.3.2 Efficiency Impacts, Cooling Mode

Design distribution efficiency is based on the impacts on system efficiency of pressure drops and temperature rises in the suction line. The pressure-drop impact I_p shall be taken from Table 9-1 for the vapor-line size and equipment capacity that pertain to the system under evaluation..

Table 9-1. Pressure-Drop Efficiency Impact I_p (Percent per 10 ft or 3m of refrigerant line) for Four Nominal Line Sizes, Type L Copper. Cooling Mode.

Refrigerant Line O.D. (in.)	0.500	0.625	0.875	1.125
Nominal Pipe Size (in.)	3/8	1/2	3/4	1
Capacity (tons)				
1.0	1.0	0.3	0.1	0.0
1.5	2.1	0.7	0.1	0.0
2.0	3.6	1.2	0.2	0.1
2.5	5.3	1.8	0.3	0.1
3.0	7.4	2.4	0.4	0.1
3.5	9.8	3.2	0.6	0.2
4.0	12.5	4.1	0.7	0.2

The temperature-rise impact I_t shall be taken from Table 9.2 for the vapor-line size, vapor-line insulation condition, and equipment capacity that pertain to the system under evaluation. The values for Insulated = Yes shall only be used if the insulation is at least equivalent to 0.5 in. (1.3 cm) of closed-cell polymer foam ($k=0.02$ Btu/h-ft-°F or 0.035 W/m-K). The formal criterion is that in order for the insulated value of I_t to be used, the ratio of insulation thickness to thermal conductivity shall equal or exceed 2 ft²-h-°F/Btu (0.4 m²-K/W). If the pipe is insulated but to a lesser degree than this, a value for I_t midway between the insulated and uninsulated entries in Table 9.2 shall be used.

Table 9.2. Temperature-Rise Efficiency Impact I_t (Percent per 10 ft or 3m of refrigerant line) for Four Nominal Line Sizes and Three Insulation Thicknesses. Cooling Mode.

Pipe O.D., in.	0.500		0.625		0.875		1.125	
Nominal, in.	3/8		1/2		3/4		1	
Insulated?	No	Yes	No	Yes	No	Yes	No	Yes
Capacity (tons)								
1.0	0.6	0.3	0.8	0.3	1.1	0.4	1.4	0.5
1.5	0.4	0.2	0.5	0.2	0.7	0.3	0.9	0.3
2.0	0.3	0.1	0.4	0.2	0.6	0.2	0.7	0.2
2.5	0.3	0.1	0.3	0.1	0.4	0.2	0.6	0.2
3.0	0.2	0.1	0.3	0.1	0.4	0.1	0.5	0.2
3.5	0.2	0.1	0.2	0.1	0.3	0.1	0.4	0.1
4.0	0.2	0.1	0.2	0.1	0.3	0.1	0.4	0.1

9.3.3 Efficiency Impacts, Heating Mode

Design distribution efficiency is based on the impacts on system efficiency of pressure drops and temperature rises in the discharge line. The pressure-drop impact I_p shall be taken from Table 9-3 for the vapor-line size and equipment capacity that pertain to the system under evaluation. Note that the capacity is benchmarked as the cooling capacity, even though the heating mode is being considered here.

The temperature-drop impact I_t shall be taken from Table 9.4 for the vapor-line size, vapor-line insulation condition, and equipment capacity that pertain to the system under evaluation. The values for Insulated = Yes shall only be used if the insulation is at least equivalent to 0.5 in. (1.3 cm) of closed-cell polymer foam ($k=0.02$ Btu/h-ft-°F or 0.035 W/m-K). The formal criterion is that in order for the insulated value of I_t to be used, the ratio of insulation thickness to thermal conductivity shall equal or exceed 2 ft²-h-°F/Btu (0.4 m²-K/W). If the pipe is insulated but to a lesser degree than this, a value for I_t midway between the insulated and uninsulated entries in Table 9.4 shall be used. Note again that capacity is benchmarked to cooling, even though it is the heating mode that is being considered in this section.

Table 9-3. Pressure-Drop Efficiency Impact I_p (Percent per 10 ft or 3m of refrigerant line) for Four Nominal Line Sizes. Heating Mode.

Refrigerant Line O.D. Type L Copper (in.)	0.500	0.625	0.875	1.125
Nominal Pipe Size (in.)	3/8	1/2	3/4	1
Cooling Capacity (tons)				
1.0	0.1	0.0	0.0	0.0
1.5	0.1	0.0	0.0	0.0
2.0	0.2	0.1	0.0	0.0
2.5	0.3	0.1	0.0	0.0
3.0	0.4	0.1	0.0	0.0
3.5	0.5	0.2	0.0	0.0
4.0	0.6	0.2	0.0	0.0

Table 9-4. Temperature-Drop Efficiency Impact I_t (Percent per 10 ft or 3m of refrigerant line) for Four Nominal Line Sizes and Three Insulation Thicknesses. Heating Mode.

Pipe O.D., in.	0.500		0.625		0.875		1.125	
Nominal, in.	3/8		1/2		3/4		1	
Insulated?	No	Yes	No	Yes	No	Yes	No	Yes
Cooling Capacity (tons)								
1.0	3.3	1.5	4.1	1.7	5.8	2.1	7.4	2.5
1.5	2.2	1.0	2.8	1.1	3.9	1.4	5.0	1.6
2.0	1.6	0.7	2.1	0.9	2.9	1.1	3.7	1.2
2.5	1.3	0.6	1.6	0.7	2.3	0.9	3.0	1.0
3.0	1.1	0.5	1.4	0.6	1.9	0.7	2.5	0.8
3.5	0.9	0.4	1.2	0.5	1.6	0.6	2.1	0.7
4.0	0.8	0.4	1.0	0.4	1.4	0.5	1.9	0.6

9.3.4 Off-Cycle Loss Factors, Cooling Mode

The off-cycle loss factor I_{cyc} shall be taken from Table 9-5 for the vapor-line size and equipment capacity that pertain to the system under evaluation..

9.3.5 Off-Cycle Loss Factors, Heating Mode

The off-cycle loss factor I_{cyc} shall be taken from Table 9-6 for the vapor-line size and equipment capacity that pertain to the system under evaluation..

Table 9-5. Off-Cycle Loss Factor I_{cyc} (Percent per 10 ft or 3m of refrigerant line) for Four Nominal Line Sizes. Cooling Mode.

Refrigerant Line O.D. Type L Copper (in.)	0.500	0.625	0.875	1.125
Nominal Pipe Size (in.)	3/8	1/2	3/4	1
Capacity (tons)				
1.0	0.6	0.9	1.4	2.0
1.5	0.4	0.6	1.0	1.4
2.0	0.3	0.4	0.7	1.0
2.5	0.2	0.4	0.6	0.8
3.0	0.2	0.3	0.5	0.7
3.5	0.2	0.3	0.4	0.6
4.0	0.2	0.2	0.3	0.5

Table 9-6. Off-Cycle Loss Factor I_{cyc} (Percent per 10 ft or 3m of refrigerant line) for Four Nominal Line Sizes. Heating Mode.

Refrigerant Line O.D. Type L Copper (in.)	0.500	0.625	0.875	1.125
Nominal Pipe Size (in.)	3/8	1/2	3/4	1
Cooling Capacity (tons)				
1.0	1.4	2.0	3.3	4.7
1.5	0.9	1.3	2.2	3.1
2.0	0.7	1.0	1.6	2.4
2.5	0.6	0.8	1.3	1.9
3.0	0.5	0.7	1.1	1.6
3.5	0.4	0.6	0.9	1.3
4.0	0.3	0.5	0.8	1.2

9.3.6 Thermal Regain Factor

The thermal regain factor F_{regain} shall be calculated in a manner consistent with the procedure defined in Section 6, with the refrigerant line lengths in each type of buffer space substituting for the duct surface areas in each type of buffer space.

9.3.7 Distribution Efficiencies, Cooling Mode

The cooling design distribution efficiency shall be given by Equations 9-1 and 9-2, using parameters selected from Tables 9-1, 9-2, and 9-5:

For IP

$$\eta_{dist,design} = \left(1 - \frac{I_p + (1 - F_{regain})I_t}{100} \right)^{(L_{ref} - 25)/10} \quad (9-1)$$

For SI

$$\eta_{dist,design} = \left(1 - \frac{I_p + (1 - F_{regain})I_t}{100} \right)^{(L_{ref} - 7.5)/3} \quad (9-2)$$

The cooling seasonal distribution efficiency shall be given by Equations 9-3 and 9-4:

For IP

$$\eta_{dist,seasonal} = \left(1 - \frac{I_p + (1-F_{regain})I_t + (1-F_{regain})I_{cyc}}{100} \right)^{(L_{ref}-25)/10} \quad (9-3)$$

For SI

$$\eta_{dist,seasonal} = \left(1 - \frac{I_p + (1-F_{regain})I_t + (1-F_{regain})I_{cyc}}{100} \right)^{(L_{ref}-7.5)/3} \quad (9-4)$$

9.3.8 Distribution Efficiencies, Heating Mode

The heating design distribution efficiency shall be given by Equations 9-1 and 9-2, using parameters selected from Tables 9-3, 9-4, and 9-6.

The heating seasonal distribution efficiency shall be given by Equations 9-3 and 9-4, using parameters selected from Tables 9-3, 9-4, and 9-6.

EXAMPLE (Not Part of Proposed Standard)

A mini-split system having a rated capacity of 24,000 Btu/h is installed with 60 ft of refrigerant line joining the indoor and outdoor units. Of this, 20 ft is in the conditioned space and 40 ft is outside the conditioned space. The vapor line is ½-in. nominal type L copper tubing with ½-in. polymer foam insulation.

The thermal regain factor F_{regain} is equal to $2/3 * 0 + 1/3 * 1 = 0.333$.

In the cooling mode, the following values are selected from Tables 9-1, 9-2, and 9-5:

$$I_p = 1.2 \quad I_t = 0.2 \quad I_{\text{cyc}} = 0.4$$

In the heating mode, the following values are selected from Tables 9-3, 9-4, and 9-6:

$$I_p = 0.1 \quad I_t = 0.9 \quad I_{\text{cyc}} = 1.0$$

Using equations 9.1 and 9.3 with the heating and cooling parameters, respectively, the distribution efficiencies are calculated to be:

Distribution Efficiencies for Example 1 Refrigerant Distribution System

Mode	Cooling	Heating
Condition		
Design	0.954	0.976
Seasonal	0.945	0.953

The question might be asked, Is it possible ever to get low numbers? The answer is yes, for a poorly designed system. Consider the following: A two-ton system with 3/8-in. nominal refrigerant lines that are 100 ft long. Let the thermal regain factor and insulation conditions be the same as before. Thus, the only things that are changed are the length and diameter of the refrigerant lines.

In the cooling mode, the following values are selected from Tables 9-1, 9-2, and 9-5:

$$I_p = 3.6 \quad I_t = 0.1 \quad I_{cyc} = 0.3$$

In the heating mode, the following values are selected from Tables 9-3, 9-4, and 9-6:

$$I_p = 0.2 \quad I_t = 0.7 \quad I_{cyc} = 0.7$$

Using equations 9.1 and 9.3 with the heating and cooling parameters, respectively, the distribution efficiencies are calculated to be:

Distribution Efficiencies for Example 2 Refrigerant Distribution System

Mode	Cooling	Heating
Condition		
Design	0.756	0.951
Seasonal	0.744	0.918