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Jim Denkmann / The Windy City / U.S.A.

Selecting Recirculated Liquid & Suction Pipe Diameters

This paper presents examples illustrating use of the ASHRAE RP 185 ammonia piping diagrams; both liquid supply via a recirculation pump and the associated recirculated suction return piping are discussed. The Darcy-Weisbach equation, the Colebrook function, fully-rough friction factor are also used in examples.

1.1 Summary of Equations

(1) $v = \dfrac{\left(\dfrac{\dot{m}\overline{V}}{60}\right)}{A}$ refrigerant velocity, superficial, ft/sec

(2) $\mathrm{Re} = 123.9\dfrac{dv\rho}{\mu}$ Reynolds Number

(3) $\dfrac{1}{\sqrt{f}} = -2.0\log\left(\dfrac{12\dfrac{\varepsilon}{d}}{3.7}\right)$ fully-rough friction factor

(4) $\dfrac{1}{\sqrt{f}} = -2.0\log\left(\dfrac{\varepsilon}{3.7D} + \dfrac{2.51}{\mathrm{Re}\sqrt{f}}\right)$ Colebrook friction factor

(5) $\Delta p = \dfrac{\rho}{144} \cdot f \cdot \dfrac{L_e}{D} \cdot \dfrac{v^2}{2g}$ Darcy-Weisbach equation, psi

(6) $Cv_a = \sqrt{\dfrac{1}{\dfrac{1}{Cv_1^{\,2}} + \dfrac{1}{Cv_2^{\,2}} + \dfrac{1}{Cv_3^{\,2}} + \dots \dfrac{1}{Cv_n^{\,2}}}}$ combining Cv (series flow)

(7) $Cv = \dfrac{0.947\,\dot{m}}{\sqrt{\Delta P\,\rho}}$ flow coefficient using Cv

(8) $Vol_{ref} = \left(\dot{m}_{vap}\,\vec{V}_{vap}\right) + \left(\dot{m}_{liq}\,\dfrac{1}{\rho_{liq}}\right)$ total refrigerant volume flow, wet suction risers, ft^3/min

(9) $\lambda_{liq} = \dfrac{\dot{m}_{liq}\,\dfrac{1}{\rho_{liq}}}{Vol_{ref}}$ decimal percent volume liquid, wet suction risers

(10) $\rho_m = \left(\left(1-\lambda_{liq}\right)\dfrac{1}{\vec{V}_{vap}}\right) + \left(\lambda_{liq}\,\rho_{liq}\right)$ mean refrigerant density, wet suction risers only, lb/ft$^3_{\text{mean}}$

(11) $\Delta p = \left(\dfrac{Q}{Cv}\right)^2 \dfrac{\rho}{62.36}$ valve pressure drop using flow coefficient, psi

(12) $\Delta p_{n-n} = \left[\dfrac{t_n}{t_c + t_w}\left(p_L - p_S\right) + p_S\right] L_e$ interpolation of pressure drop between nodes, (psi)

Note: in order for Eq 12 to be valid, t_n must lie between t_C and t_W and $t_W - t_C \equiv 10$ ºF.

(13) $K = f\,\dfrac{L_e}{D}$ resistance coefficient

(14) $K = \dfrac{891 d^4}{Cv^2}$ resistance coefficient as a function of port diameter and Cv

Nomenclature

A	area, square feet (sq ft, ft^2)
D	diameter, feet (ft)
d	diameter, inches (in)
f	friction factor, $h_L = f\,L\,v^2/D\,2\,g$
g	gravitational constant, 32.2 ft/sec^2
h_L	loss of static pressure due to flow, feet (ft)
K	resistance coefficient, $h_L = K\,v^2/2\,g$
k	ratio of specific heats, constant pressure to constant volume, *Cp/Cv*
L	length of pipe, feet (ft)

L_e	equivalent length of pipe, feet (ft; TEL in text)
L/D	equivalent length of resistance to flow, pipe diameters
$\dot{m}$	mass flow, pounds per minute (lb/min)
P	pressure, pounds per square inch, gauge (psig)
p	pressure, pounds per square inch, absolute (psia)
Q	rate of flow, gallons per minute (gpm)
Re	Reynolds number
T	temperature, degrees Rankine (460 + ºF)
t	temperature, ºF
$\overline{V}$	specific volume, cubic feet per pound (ft^3/lb)
v	fluid velocity, feet per second (fps)
x	refrigerant quality, percent mass vapor ($0 \geq x \leq 1$)
Z	elevation head above reference point, feet (ft)
Δt	differential temperature, ºF
Δp	differential pressure, pounds per square inch (psi)
ε	absolute roughness, feet (ft)
ε/D	relative roughness, feet per foot (ft/ft)
μ	absolute viscosity, centipoise
ρ	weight density of fluid, pounds per cubic foot (lb/ft^3)
λ_{liq}	percent liquid by volume (wet suction risers only)

The following are for use with RP 185:

Δp_{n-n}	= pressure drop, node to node, psi
t_n	= t_{sat} leaving upstream section, ºF
t_C	= t_{sat}, lower evaporating temperature limit, ºF
t_W	= t_{sat}, upper evaporating temperature limit, ºF
p_L	Δp/100 ft TEL, larger value, psia
p_S	Δp/100 ft TEL, smaller value, psia

1.2 The Design Task

An owner of a refrigeration system wishes to build a new addition to an existing refrigeration system. The addition will consist of 3 new 30 TR evaporators for a -10 ºF frozen foods storage freezer. The task undertaken in this example consists of selecting and laying out the pumped liquid supply and recirculated suction return piping, including all valves. As a part of this exercise, the necessary calculations for finding pressure drops will be identified and used in examples.

The piping systems are broken down into "nodes" – that piping between 2 nodes being described as:

- Piping between two divided-flow fittings (tees), or
- Piping separated by a reducing fitting, thus affecting the superficial fluid velocity by means of reducing or increasing the pipe diameter while mass flow in the segment is held constant

No specific allowance will be made for future system expansion in this example, although the piping diameters selected could possibly handle some additional mass flows without undue pressure loss in either the liquid or suction piping, rather typical for many system designs.

1.3 Liquid Velocity in Piping

There appear to be two schools of thought regarding pumped recirculated liquid ammonia piping: the *IIAR Refrigeration Piping Manual* recommends choosing pipe diameters between a range of 325 fpm liquid velocity for a 1" line, up to 450 fpm in a 12" line. A 350 fpm (5.83 fps) threshold has been arbitrarily chosen which closely agrees with IIAR. Chart 3-717 (ASHRAE RP 185) does not reflect fluid velocities, hence the recommendations contained in Table 1 are presented.

TABLE 1 RECOMMENDED MAXIMUM LIQUID MASS FLOW @ 350 FPM (lb_m/min, R717, t_{sat}=-20ºF, x=0)									
Schedule 80				Schedule 40					
¾"	1"	1¼"	1½"	2"	2½"	3"	4"	5"	6"
40	70	130	180	340	430	750	1,300	2,040	3,000

The other school of thought ignores liquid velocity but places an upside limit to the piping friction loss to ≤ 2.0 psi/100 feet TEL. The author has a preference for the former rather than the latter because:

- lower fluid velocities reduce internal pipe wire drawing which is hidden and nearly impossible to detect
- reduces hammering when multiple solenoids close simultaneously

Illustrating the above arguments, consider these possible flow rates in a ¾" sch 80 branch overfeed liquid line to an evaporator:

- 40 lb/min, -20 ºF : 350 fpm (6.5 psi/100 ft TEL)
- 22 lb/min, -20 ºF : 2 psi/100 ft TEL (173 fpm)

The author is of the opinion that branch liquid piping is frequently too large, resulting in over-sized solenoids, check valves and hand expansion valves when installed without piping reducers. When the total branch line friction pressure loss is a comparatively large percent of the total piping system friction loss (excluding static lift Z), it becomes far easier to balance evaporator liquid flow rates. Over-sizing check valves and solenoids can also result in pulsing valve pistons and disks – these send shock waves up and down the piping and should be avoided.

The total dynamic head required for selecting a refrigerant pump at Node 1 has been shown in Figure 1. The example employs the following calculation methods for predicting pressure loss between nodes (divided flow fittings; "tees"):

- simplified method using Chart 3-717 (ASHRAE RP 185)
- Darcy-Weisbach, with a Reynolds number dependent *f*
- Darcy-Weisbach, with a Reynolds number independent (fully-rough) *f* (Colebrook)

1.4 Pumped Liquid Supply Piping & Valves

Problem: Consider the piping diagram shown in Figure 1. This exercise identifies applicable pipe sizes, friction loss and refrigerant pump TDH for the liquid supply piping system shown at Node 1. A refrigerant pump selection is not considered in this example; instead the pressure loss due to friction head and elevation head between nodes 1 and 5 are predicted, comprised of the connection to most remote evaporator, including losses through branch piping, valves and button orifice at the entrance of each evaporator circuit. All liquid flow in excess of that required for the system is assumed bypassed back to the accumulator through a back pressure regulator or orifice, similar to a Hansen pump installation. Note that each evaporator has been initially selected based upon a 4:1 liquid recirculation (3:1 overfeed). A 5 psi pressure drop across each button orifice is assumed plus an additional 5 psi pressure drop across each hand expansion valve in the liquid supply valve control group. The fluid velocity in pumped liquid piping will be limited to 350 fpm (5.83 fps) as previously discussed.

Solution: Start by finding the total system flow for selecting the piping main from Node 1 to Node 2 = 124 lb/min. To check for 5.83 fps velocity, convert mass flow to volume flow, ft^3/sec, using:

$$v = \frac{\left(\dfrac{\dot{m}\dfrac{1}{\rho}}{60}\right)}{A} \qquad (1)$$

The density of liquid ammonia at -20 ºF is 42.23 lb/ft^3, therefore the liquid volume flow supplied to the system is 0.04894 ft^3/sec (numerator value of Eq 1). Start with an initial 1½" sch 80 pipe selection, ID=1.500" (0.01225 ft^2). By dividing volume flow by area, the resulting velocity is 4 fps – sufficient for the service, however this size leaves little “wiggle room”. Now try 2" sch 40, ID=2.067" (0.023 ft^2), v=2.13 fps – this size will be sufficient for the service and can be further justified that the welding cost (pass-inches) will be of minimal additional cost (1½" sch 80 vs 2" sch 40).

1.4.1 Estimating Pressure Drop Using Chart 3-717 (ASHRAE RP 185)

Chart 3-717 will be found at the back of this paper, Figure 7 (ignore the dotted line, entered many years ago). Enter 2" pipe and run down to 124 lb/min. While the intersection is literally “off-the chart”, it can be approximated to the lower-most horizontal line. Then read to the left to the -20 ºF isotherm. Then trace a curve below the 0.3 psi curve and one can approximate ~0.25 psi/100 ft TEL.

To find node-to-node pressure drop, multiply the TEL between nodes by 0.25 psi/100 ft, then multiply by the pipe schedule correction factor (if applicable). Assuming one 2" elbow + 100 lineal feet of piping between these two nodes, the piping TEL becomes 100 ft + 3.3 ft = 103 ft TEL (drop the fraction < .5; round up > 0.5). Therefore the 1 – 2 nodal pressure loss becomes 0.25 psi/100 TEL x 1.03 = 0.26 psi.

Now alternately predict the Node 1 to Node 2 pressure drop using the Darcy Weisbach equation, upon which RP 185 is based.

1.4.2 Node 1- Node 2 - Predicting Pressure Drop Using Darcy-Weisbach Equation

Start by finding the Reynolds Number for liquid ammonia in a 2" sch 40 pipe, flowing at a velocity of 2.13 fps by:

$$\mathrm{Re} = 123.9\frac{dv\rho}{\mu} \tag{2}$$

where:

d = 2.067 inches

v = 2.13 fps

ρ = 42.23 lb_m/ft^3

μ = 0.240273 centipoise

Therefore **Re = 95.875 x 10³**. Then find the pipe relative roughness, *ε/D*. The value of *ε* (epsilon) for commercial steel piping is 0.00015 ft and *D* is 0.1722 ft for 2" sch 40 pipe, therefore the relative roughness is 0.00087 ft/ft.

Using Re = 95.875 x 10^3 with a relative roughness of 0.00087, enter the Moody diagram as shown in Figure 2 to determine whether the resulting friction factor is Reynolds-dependent or Reynolds independent. Since this intersection of relative roughness with Reynolds number occurs to the left of the dotted curve separating these two regions, flow becomes Reynolds dependent and the Colebrook equation will be used to more closely pinpoint *f*. The Moody diagram can be read at $f \approx 0.022$, but we can approach 4 significant figures by using the Colebrook equation. Many practitioners prefer to assume flow is fully rough so this paper finds both and compares the results.

Fully Rough f :

$$\frac{1}{\sqrt{f}} = -2.0\log\left(\frac{12\dfrac{\varepsilon}{d}}{3.7}\right) \qquad (3)$$

Colebrook f :

$$\frac{1}{\sqrt{f}} = -2.0\log\left(\frac{\varepsilon}{3.7D} + \frac{2.51}{\mathrm{Re}\sqrt{f}}\right) \qquad (4)$$

If a spreadsheet program is used to find *f*, Eq 3 will be selected, because in this equation, the value of *f* is *explicit* instead of *implicit* as it is in Eq 4. Therefore, Eq 3 is far faster and easier to execute. Note that pipe diameter in the Eq 3 is expressed in inches and in Eq 4, in feet. The results are:

Fully Rough f : 0.01899

Colebrook f : 0.02183[1]

Therefore, with diameter, viscosity, density and effective pipe length all held constant, Δp in the Darcy-Weisbach equation varies directly with *f* as seen in Eq 5.

$$\Delta p = \frac{\rho}{144} \cdot f \cdot \frac{L_e}{D} \cdot \frac{v^2}{2g} \qquad (5)$$

[1] The author fesses up, – it's easy with EES!

where:

ρ = 42.23 lb/ft^3

f = 0.02183 (Colebrook)

f = 0.01899 (fully rough)

L_e = 103 ft TEL

v = 2.13 ft/sec (Colebrook)[2]

D = 0.172 ft (2.067 in)

g = 32.2 ft/sec^2

Answers:

Fully-rough *f* pressure loss, Node 1 to Node 2:	0.228 psi
Reynolds-dependent *f* pressure loss (Colebrook):	**0.270 psi**

For purposes of this exercise, the Darcy/Colebrook relationship will be used = 0.270 psi.

1.4.3 Node 2 to Node 3 Pressure Drop

Again start by finding the Reynolds Number for this section of pipe using Eq 2. The liquid velocity in this section of pipe, flowing at the rate of 82.4 lb/min is 1.396 fps (see Figure 1); all other values are similar to those used previously. The answer is Re = 62.837 x 10^{-3}. Again, the flow in this section of piping is turbulent and the Colebrook function will be used to find *f*. The answer is ***f* = 0.02294.**

The Darcy-Weisbach equation will again be used to predict pressure drop, where:

L_e = 110 ft (100 ft + 2 elbows + 1 flow-through tee, 2" sch 40)

v = 1.396 fps

D = 0.172 feet

therefore the answer is **Δp = 0.13 psi.**

[2] A fully-rough *f* assumes a higher fluid velocity, however its bounds are unknown because *f* in this region is Reynolds independent, therefore velocity independent.

1.4.4 Node 3 to Node 4 Pressure Drop

This pipe section was selected with a 1-1/4" sch 80 line having 1 elbow + 1 flow-through tee, therefore the TEL is 105 ft. Reading from Chart 3-717 with 41.2 lb/min flowing, the TEL/100 ft = 0.34 psi. Check that the graph was based upon sch 80 pipe – it is – no further correction necessary. Multiplying 0.34 psi/100 ft TEL by 1.05, the Δp for this pipe section yields 0.36 psi.

1.4.5 Node 4 to Node 5 Pressure Drop Using Flow Coefficient, Cv

This pipe section conveys 41.2 lb/min -20 ºF liquid to the 30 TR evaporator; a ¾" line has been selected consisting of the following series resistances to flow:

2 – angle valves, ¾", Hansen Cv = 9

1 – strainer, ¾" (pressure drop normally neglected)

1 – solenoid, ¾" (assume Hansen HS-7, Cv = 8)

1 – hand expansion valve, ¾" (allow 5 psi pressure drop)

1 – button orifice (per mfg instructions allow 5 psi)

1 – check valve (assume ¾" Hansen HCK-4, Cv = 8.2)

15 - lineal feet of ¾" piping + 5 elbows + 1 branch tee, 1-1/4" to ¾" bull, TEL = 26 ft

Valves are treated separately from the piping for this exercise. The valve control group (angle valves, solenoid, check valve) Cv values can be summed using:

$$Cv_a = \sqrt{\frac{1}{\frac{1}{Cv_1^{\ 2}} + \frac{1}{Cv_2^{\ 2}} + \frac{1}{Cv_3^{\ 2}} + \ldots \frac{1}{Cv_n^{\ 2}}}} \qquad (6)$$

therefore Cv_a = 4.26. A prediction of pressure drop for these valves (installed in series) will be found from ($Cv = Cv_a$):

$$Cv = \frac{0.947\,\dot{m}}{\sqrt{\Delta P\,\rho}} \qquad (7)$$

Rearranging terms and solving for Δp, the answer is 2.0 psi. Then to this add 5 psi for an allowable pressure loss across the hand expansion valve + 5 psi additional that the evaporator manufacturer specified for loss through each button orifice at a 4:1 recirculation, our combined valve pressure drop = 2 + 5 + 5 = 12 psi.

The piping TEL is 26 ft. Entering Chart 3-717 for a ¾" sch 80 pipe with 41.2 lb/min flowing, the resulting Δp/100 ft TEL is 5.5 psi. Multiplying this by 0.26, the piping loss is 1.43 psi. Therefore the Node 4 to Node 5 pressure drop = 12 psi + 1.43 psi = 13.43 psi.

1.4.6 Summing the Losses

Node 1 – 2	0.27 psi
Node 2 – 3	0.13 psi
Node 3 – 4	0.36 psi
Node 4 – 5	13.43 psi
Total:	14.19 psi friction loss

To this value, add the force exerted by the static head of a 20 foot column of liquid ammonia having a ρ = 42.23 lb/ft^3 , therefore this force exerted at Node 1 = 5.86 psi$_f$. Summing friction + static head (*Z* feet) = 14.19+ 5.86 = 20.05 psi TDH. This can also be stated in terms of feet head of fluid = 68.4 feet.

When plotting TDH on a pump curve, the static head component *Z* must be treated separately because the force exerted by gravity is a constant. The remaining forces, $h_L = \frac{144p}{\rho} + \frac{v^2}{2g}$ may be plotted as a function of h_L^2 when plotting a system curve.

1.5 Predicting Pressure Loss in Recirculated Suction Return Piping

Recirculated suction return piping presents some interesting phenomena when trying to predict vapor pressure drop. Consider the system shown in Figure 3 – the recirculated return line from the 3 evaporators previously examined. This piping can, for purposes of these calculations, be separated into horizontal piping (suction mains), branch suction piping from evaporators and vertical risers. In all cases, it is desirable that horizontal piping be installed with a pitch in the direction of flow; a minimum 1% pitch is desirable and recommended wherever possible. Horizontal branch suction piping should be selected with a lower vapor velocity (than risers) through the gas-powered suction stop valve and angle valve(s) to minimize pressure drop.

The diameter of a vertical riser must be selected for annular flow at minimum load, a condition upon which the recommendations in Figure 5 are based (Jekel). Consider two cases: a blast freezer (for example a spiral) and a blast cell. The former is continuously-fed; the latter is batch-fed. The performance of recirculated suction risers for these two processes differ in a subtle but important way.

Load profiles for evaporators selected for blast freezers are nearly constant, therefore the vapor velocity up the riser is likewise nearly constant, changing only when product feed is reduced or the belt momentarily stops. However vapor velocity up a riser from a blast cell varies directly with the product temperature inside that particular cell ($\delta t = t_{product} - t_{suct}$ wherein t_{suct} = constant and is measured at the top of a riser). As vapor riser velocity decreases, the drag force exerted by upward moving vapor gradually decreases with a reduction in product temperature with the unfortunate result that liquid begins to accumulate in the riser. Over time this velocity reduction can lead to an entirely liquid-filled riser with bubbles of vapor moving up through the liquid column (known as bubbly-flow or slug flow). When this occurs, the force exerted by liquid in the column equals the total static head force exerted by a completely liquid-filled pipe. This additional force from accumulating liquid static head causes the temperature at which liquid boils inside the evaporator (t_{evap}) to increase. This t_{evap} increase occurs while the product temperature in the blast cell is decreasing, therefore these two temperature changes *oppose one another* while occurring simultaneously. Said another way, this is not a good thing.

The author recommends employing alternate liquid feed methods when long suction risers are encountered in a blast cell process, i.e. a gravity-fed evaporator wherein zero liquid is returned up the riser. When placing these evaporators into defrost, all defrost condensate should drain into a condensate receiver set (similar to those used in a steam heating system) and use the force exerted by a small mechanical-drive pump to move the defrost liquid back uphill. This caveat also applies to any wet suction riser where the turndown ratio ($PLR_{annular}$) is < 0.4 as shown in Figure 5.

For purposes of this exercise, assume the suction risers to be relatively short (10 feet) as shown in Figure 3.

1.5.1 Node 1 to Node 2 Pressure Drop

The analysis begins at the evaporator outlet – at the weld connection the evaporator manufacturer provided – Node 1, and predicts the total back pressure imposed upon this most-remote evaporator by friction losses through the suction piping, valves and fittings, plus the static head imposed by a suction riser partially filled with liquid. The end-point of the analysis concludes in the vapor head space of the pumped liquid accumulator, Node 6.

1.5.1.1 Step 1 – Find Riser Vapor Velocity

Equation 1 can be used to find vapor velocity in a suction riser, however Eq 1 solves for velocity in feet/second. Therefore, the result will be multiplied by 60. First try a 4" sch 40, A = 0.08840 ft^2 with $\overline{V}$ = 14.683 ft^3/lb for -20 ºF vapor. The result is 1711 fpm – this pipe diameter is too large because the resulting velocity falls below the threshold where annular flow begins (Figure 4). Figure 5, for a 30 TR evaporator, recommends a 2½" sch 40, A = 0.03322 ft^2. Solving for v (fpm), the answer is 4553 fpm (75.88 fps). An assumption can be made that flow in this riser will be Reynolds-independent, thus calculating the *Re* becomes unnecessary. Note that this calculated velocity is based upon superficial conditions (without the presence of liquid). The magnitude of error produced by excluding the liquid volume within the pipe is negligible.

1.5.1.2 Step 2 – Determine Fully-Rough f

Equation 3 will be used to solve for f where:

$\varepsilon = 0.00015$

d = 2.469 inches

therefore $\boldsymbol{f}$ **= 0.01821**.

1.5.1.3 Step 3 – Determine Mean Density ρ_m in Suction Riser

This calculation breaks down into three sub-steps, first to find the total volume flow of refrigerant in the suction riser Vol_{ref}, then to compute the liquid fraction by volume λ_{liq}, then compute the refrigerant mean density ρ_m of the flowing stream which has the dimension lb/ft^3_{mean}. This process makes a further assumption that the refrigerant exists in a *metastable* state within a riser. This is not necessarily true, but at a vapor velocity sufficient to shear liquid into fine droplets (ie, a "spray"), it is believed by many in the industry that this two-phase mixture may be treated as such in the following calculation. The author's opinion is that using mean density holds valid when the force of gravity is at 180 degrees to flow (mixed-phase flow in a riser). However, this procedure remains subject to further scrutiny followed up with field verification. Answers to this vexing problem could soon be forthcoming from research currently underway[3]. It is likely that some error (of unknown magnitude) results when applying this method to mixed-phase flow in horizontal piping; vapor velocity, piping pitch angle and rate of overfeed are three strong variables.

Step 3a – Determine Vol_{ref} Flow Rate From Eq 8

$$Vol_{ref} = \left(\dot{m}_{vap}\,\vec{V}_{vap}\right) + \left(\dot{m}_{liq}\,\frac{1}{\rho_{liq}}\right) \tag{8}$$

where:

$\dot{m}_{liq}$ = 30.9 lb/min

$\overline{V}_{vap}$ = 14.683 ft^3/lb

ρ_{liq} = 42.23 lb/ft^3

$\dot{m}_{vap}$ = 10.3 lb/min

Vol_{ref} = 151.2349 ft^3/min vapor + 0.73171 ft^3/min liquid

= **151.967 ft^3/min** refrigerant

Step 3b – Find Liquid Fraction

$$\lambda_{liq} = \frac{\dot{m}_{liq}\,\frac{1}{\rho_{liq}}}{Vol_{ref}} \tag{9}$$

Answer: **0.00481**

Step 3c – Calculate Mean Density, ρ_m of Flowing Stream

$$\rho_m = \left(\left(1-\lambda_{liq}\right)\frac{1}{\vec{V}_{vap}}\right) + \left(\lambda_{liq}\,\rho_{liq}\right) \tag{10}$$

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[3] ASHRAE has commissioned a research project (2007) to the Danish Research Council for examination of pressure drop in a wet suction riser performing under varying load.

Answer: **0.271 lb/ft$^3_{mean}$** (3:1 overfeed, -20 ºF)

*1.5.1.4 **Step 4** – Determine Riser Pressure Drop*

Pipe running length = 10 feet with four 2½" LR elbows and a 5"x2½" reducer: 10 + (4 x 2.7) + 4.8 **= 26 ft TEL**. See Tables 6 and 8, RP 185 for these data.

The piping pressure loss can be predicted using the Darcy-Weisbach Equation:

$$\Delta p = \frac{\rho}{144} \cdot f \cdot \frac{L_e}{D} \cdot \frac{v^2}{2g}$$

Where:

$\rho_m = \rho$

$\rho_m = 0.271$ lb/ft^3

$f = 0.01821$

$L_e = 26$ ft

$v = 75.88$ fps

$D = 0.2057$ ft (2½" sch 40)

$g = 32.2$ ft/sec^2

Answer: **Δp = 0.39 psi**

1.5.2 Step 5 – Node 1 to Node 2 - Summing the Losses

Piping pressure loss = 0.39 psi

Z, psi/10 ft rise: = 0.02 psi

Total: = **0.41 psi**, Node 1 to Node 2 pressure drop

The design evaporating temperature is -20 ºF; the absolute saturated pressure is 18.28 psia. At the top of the suction riser (node 2), the vapor pressure has decreased by 0.41 psi, therefore the vapor pressure entering the downstream section of piping is 18.28 – 0.41 = 17.87 psia. This becomes the entering vapor pressure for the next downstream piping section, Node 2 to Node 3. The resulting saturated vapor temperature has now fallen to -20.84 ºF.

The force exerted by the static head component *Z*, for ρ_{mean} over a 10 foot riser height has been included, however its force is negligible. A warning: if the $PLR_{annular}$ of the riser falls too low ($v < v_{\text{annular flow}}$ caused by an over-sized riser diameter or a too-low riser vapor velocity), then the force exerted by Z can rise to 2.93 psi$_f$ which is equal to a full column of liquid.

1.5.3 Node 2 to Node 3 Pressure Drop

Refer to Figure 3 for this example. This section of piping will have a short piece of piping (maybe 2 feet) plus a strainer, a gas-powered suction stop valve (HCK2 for example), an angle valve and a branch tee – all selected in a 5" sch 40 pipe diameter. The equivalent length would then be:

L_e = 2 + 0 + 140 + 27 = 169 feet.

Equation 12 may be used to find the node-to-node pressure drop:

$$\Delta p_{n-n} = \left[\frac{t_n}{t_c + t_w} \left(\Delta p_L - \Delta p_S \right) + \Delta p_S \right] L_e \tag{12}$$

where:

t_n = -20.84 ºF (the entering temperature)

t_C = -30 ºF

t_W = -20 ºF

Δp_S = 0.21 psi/100 ft

Δp_L = 0.16 psi/100 ft

L_e = 1.69 feet / 100 feet equivalent length

Answer: **0.31 psi**

The saturated vapor temperature for the entering vapor is -20.84 ºF but after undergoing an additional pressure drop of 0.31 psi through the valves and piping, t_{sat} is now = -21.49 ºF. This temperature is then used for the next downstream section, Node 3 to Node 4. Refer to Figure 6, Chart 1-717 for finding the pressure drop data used here.

1.5.4 Node 3 to Node 4 Pressure Drop

This piping section consists of 100 feet of 5" sch 40 piping plus 2 elbows plus one reducer plus one straight tee. Therefore L_e = 122 feet. Again using Eq 12 to find node-to-node pressure drop, the result is 0.22 psi. The entering saturated temperature (t_n) is the same as the leaving saturated temperature of the upstream section, -21.49 ºF. After undergoing a pressure loss equal to 0.22 psi, the resulting downstream pressure becomes 17.34 psia with t_{sat} = -21.95 ºF.

1.5.5 Node 4 to Node 5 Pressure Drop

This piping section consists of 100 feet of 6" sch 40 piping plus 2 elbows plus one 8 x 6 reducer plus one 8" straight tee. Therefore L_e = 126 feet. Again using Eq 12 to find node-to-node pressure drop, the result is 0.35 psi. The entering saturated temperature (t_n) is the same as the leaving saturated temperature of the upstream section, -21.95 ºF. After undergoing a pressure loss equal to 0.35 psi, the resulting downstream pressure becomes 16.99 psia with t_{sat} = -22.71 ºF.

1.5.6 Node 5 to Node 6 Pressure Drop

This piping section consists of 100 feet of 8" sch 40 piping plus 2 elbows plus one straight flow tee plus one 8" pipe projection (the downturned elbow inside the accumulator) and one 8" angle valve.

$$L_e = 100 + (2 \times 9) + 7.1 + 47 + 85 = 257 \text{ feet}$$

Again using Eq 12 to find node-to-node pressure drop, the result is 0.46 psi. The entering saturated temperature (t_n) is the same as the leaving saturated temperature of the upstream section, -22.71 ºF. After undergoing a pressure loss equal to 0.46 psi, the resulting downstream pressure becomes 16.53 psia with t_{sat} = -23.71 ºF.

1.5.7 Node 1 to Node 5 Pressure Drop - Summarizing

The foregoing examples illustrate the interpolation technique when using Chart 1-717, *Suction Lines*, using Eq 12. Before entering Chart 1, it is customary industry practice when selecting the diameter of *horizontal* wet suction piping to *sum the two mass flows,* liquid + vapor, and *treat both as mass vapor*. Then enter Chart 1, a partial exploded view of which is shown in Figure 6. Table 2 is presented to summarize results and information read from Chart 1-717.

TABLE 2 SUCTION MAIN NODE PRESSURE LOSSES, MASS FLOWS (HORIZONTAL PIPING)							
Nodes	Pipe Dia. (in)	$\dot{m}$ (lb/min)	Δp (psi)	t_{enter} (ºF)	t_{lvg} (ºF)	Δp/100ft -20 ºF[1]	Δp/100ft -30 ºF[1]
1 - 2	2½"	41.2	0.41	-20.00	-20.84	pressure drop calculated using ρ_m	
2 - 3	5"	41.2	0.31	-20.84	-21.49	0.160	0.210
3 - 4	5"	41.2	0.22	-21.49	-21.95	0.160	0.210
4 - 5	6"	82.4	0.35	-21.95	-22.71	0.245	0.320
5 - 6	8"	124	0.46	-22.71	-23.71	0.122	0.161
total wet suction Δp =			1.75				

1. Data read from RP 185, Chart 1-717 – Figure 6

1.5.8 Wet Suction Piping and Overfeed Rate - Conclusions

It can be seen from the foregoing arguments that wet suction line pressure drop is strongly affected by the mass fraction of liquid within that piping. More study of these phenomena are needed so that design engineers, system owners and other interested parties can evaluate the impact of reducing the liquid recirculation rate to evaporators upon TR, relative to a reduction in the back pressure imposed by excessive overfeed liquid between a respective evaporator and its associated pumped accumulator.

A force not taken into account in these exercises is the increased pressure imposed upon an evaporator resulting from suction line pressure drop when selecting a refrigerant pump. A*ny* back pressure in a wet suction return line becomes additive to the refrigerant pump TDH. This imposed back pressure is normally insignificant when calculating pump performance; it is usually omitted from pump head calculations. However, consider a case where an evaporator pressure regulator (EPR) is fitted into one or more evaporator suction lines. These particular evaporators then become the highest pressure against which the pump is likely to work and the additional temperature (pressure) lift must be included in TDH calculations. However, exercise great care when applying EPRs to recirculated suction return piping. There is a limit, generally regarded as $t_{evap} - t_{suction}$ = 10 ºF for bottom-fed evaporators.

With only a few exceptions, do not attempt to install an EPR in the recirculated suction line from a top-fed evaporator; severe brining with a significant loss of refrigerating capacity can result.

end of text

References

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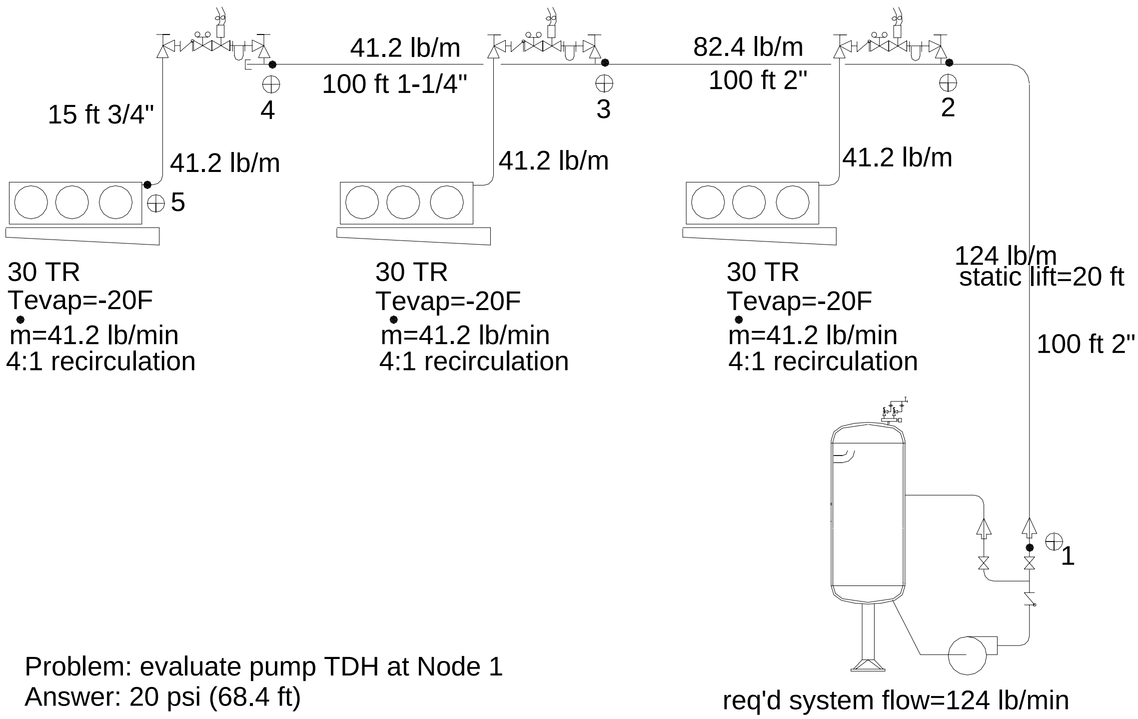
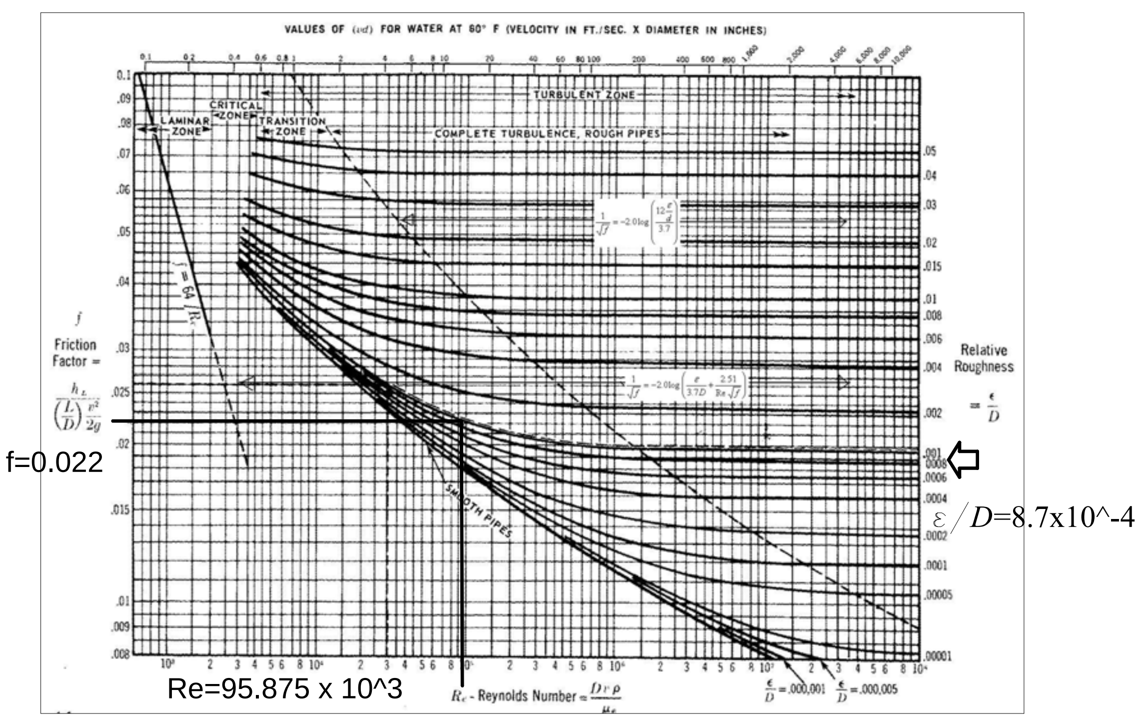
4. Stoecker, W.F, 1988, *Industrial Refrigeration*, Business News Publishing Company
5. IIAR, 2000, *Ammonia Refrigeration Piping Handbook*, International Institute of Ammonia Refrigeration
6. Hansen, 2002, *Collection of Instructions*, Hansen Technologies Corporation

Figure 1 Recirculated Liquid Supply – Example



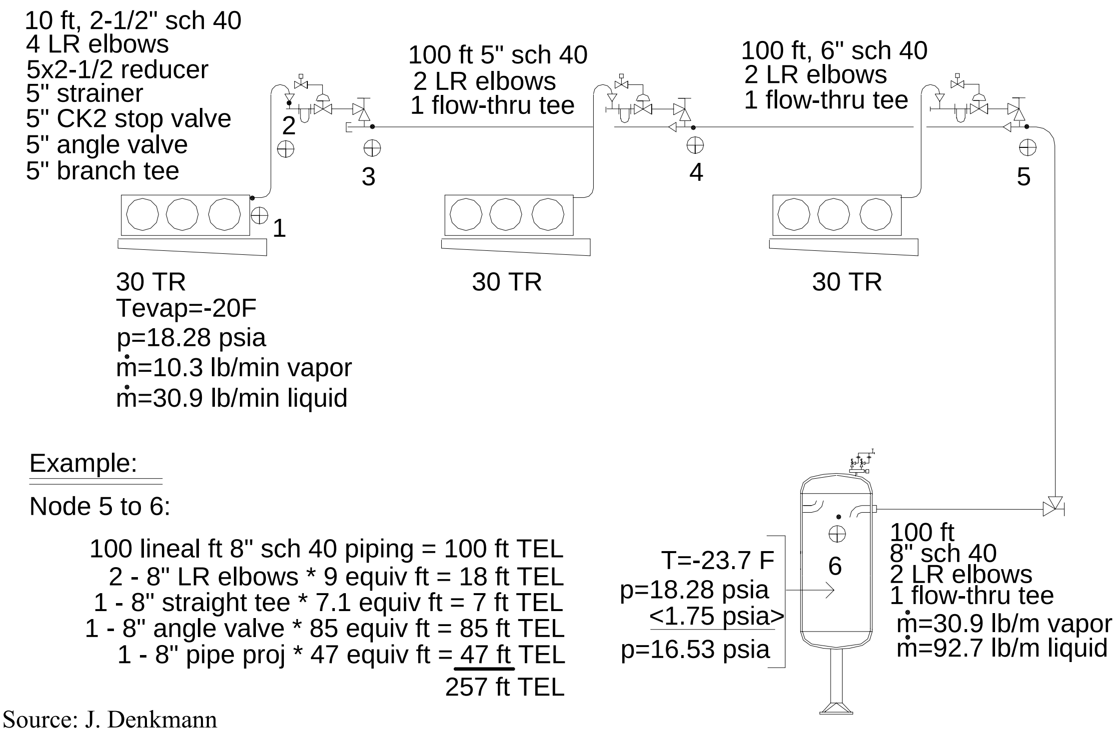
Source: J. Denkmann

Figure 2 Moody Diagram



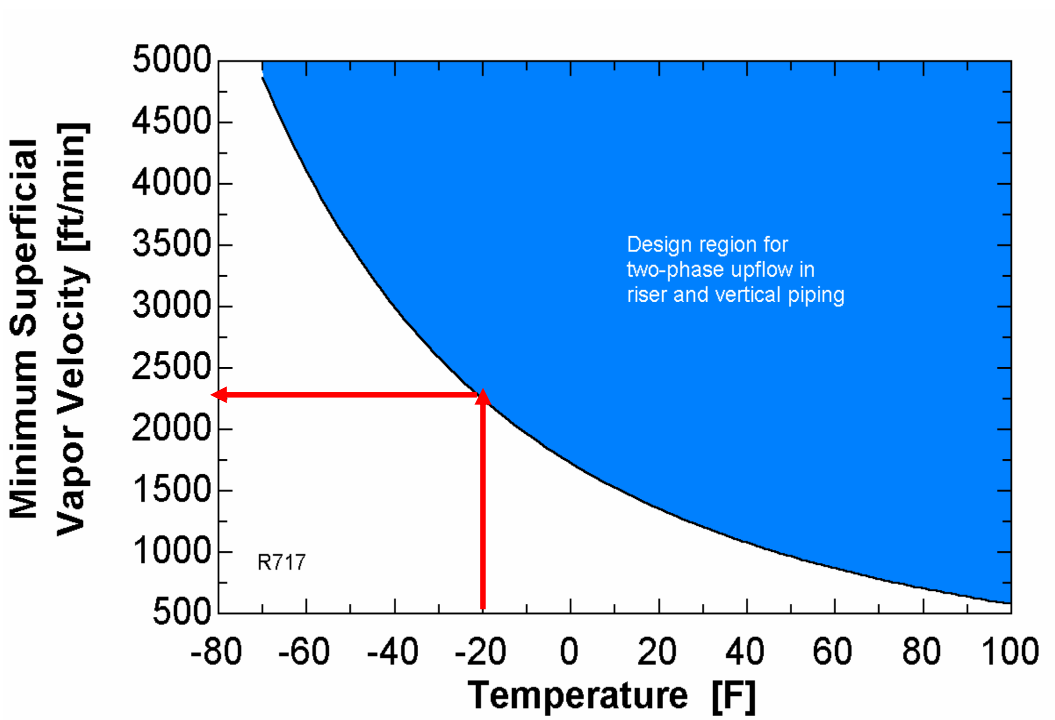
Source: Crane 410

Figure 3 Recirculated Suction Return - Example



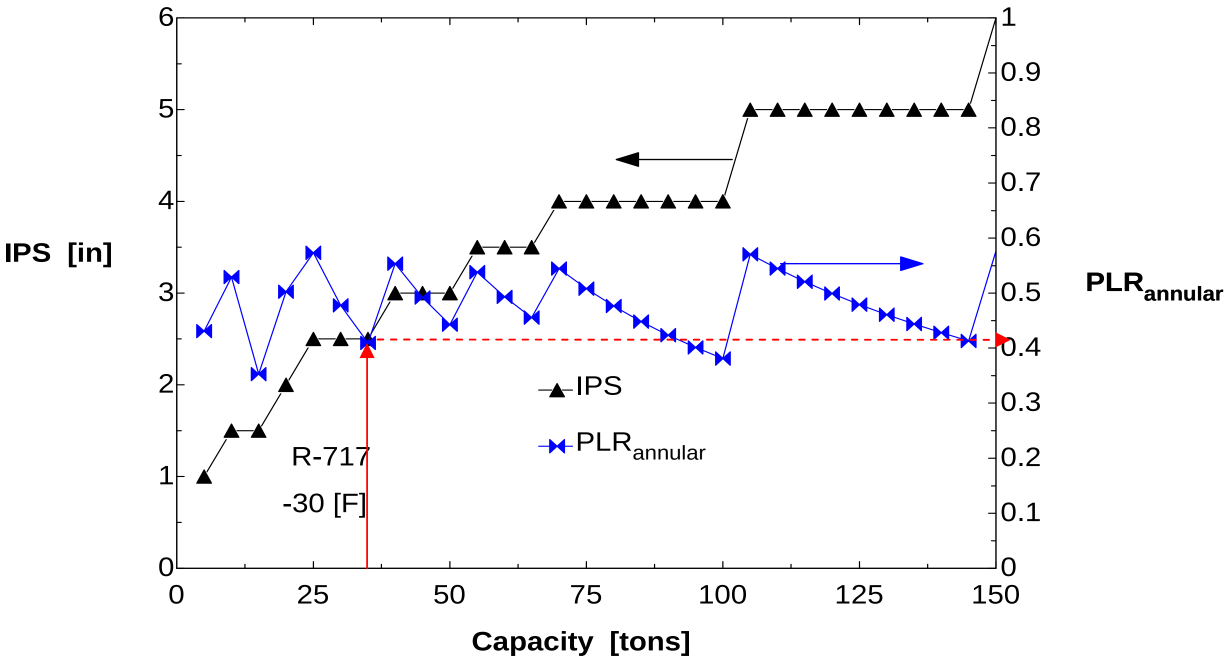
Source: J. Denkmann

Figure 4 Recirculated Suction Risers – Minimum Entrainment Velocity vs Temperature



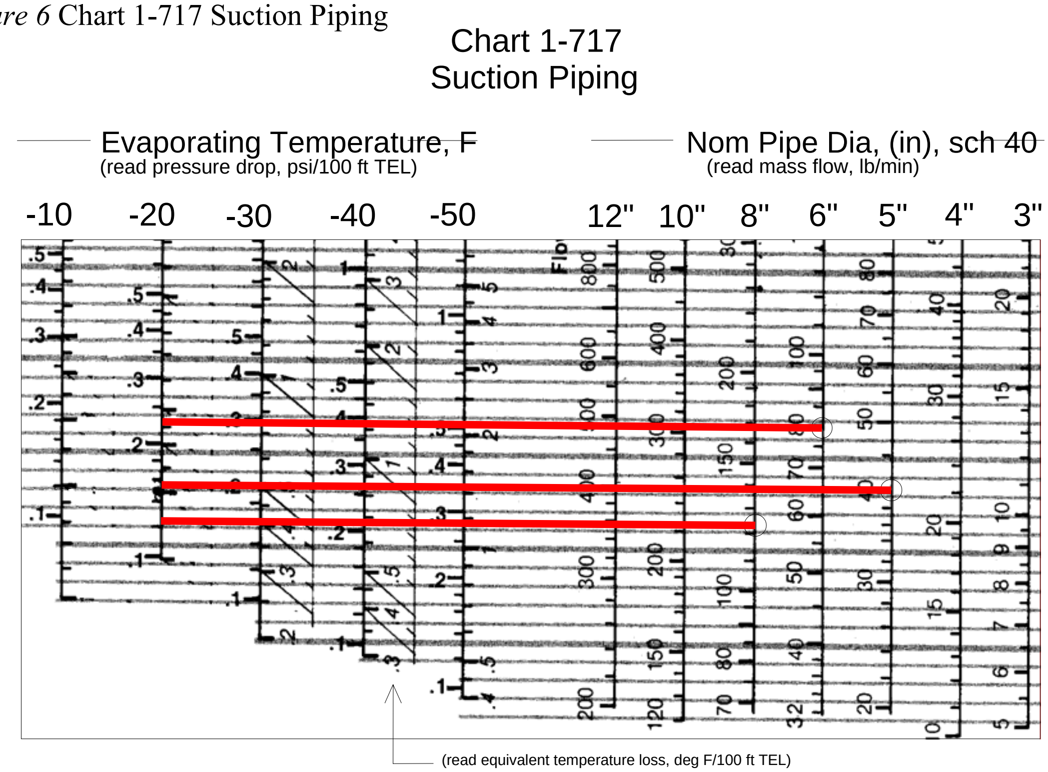
Source: T. Jekel

Figure 5 Recirculated Suction Risers – Recommended Diameters vs $PLR_{annular}$



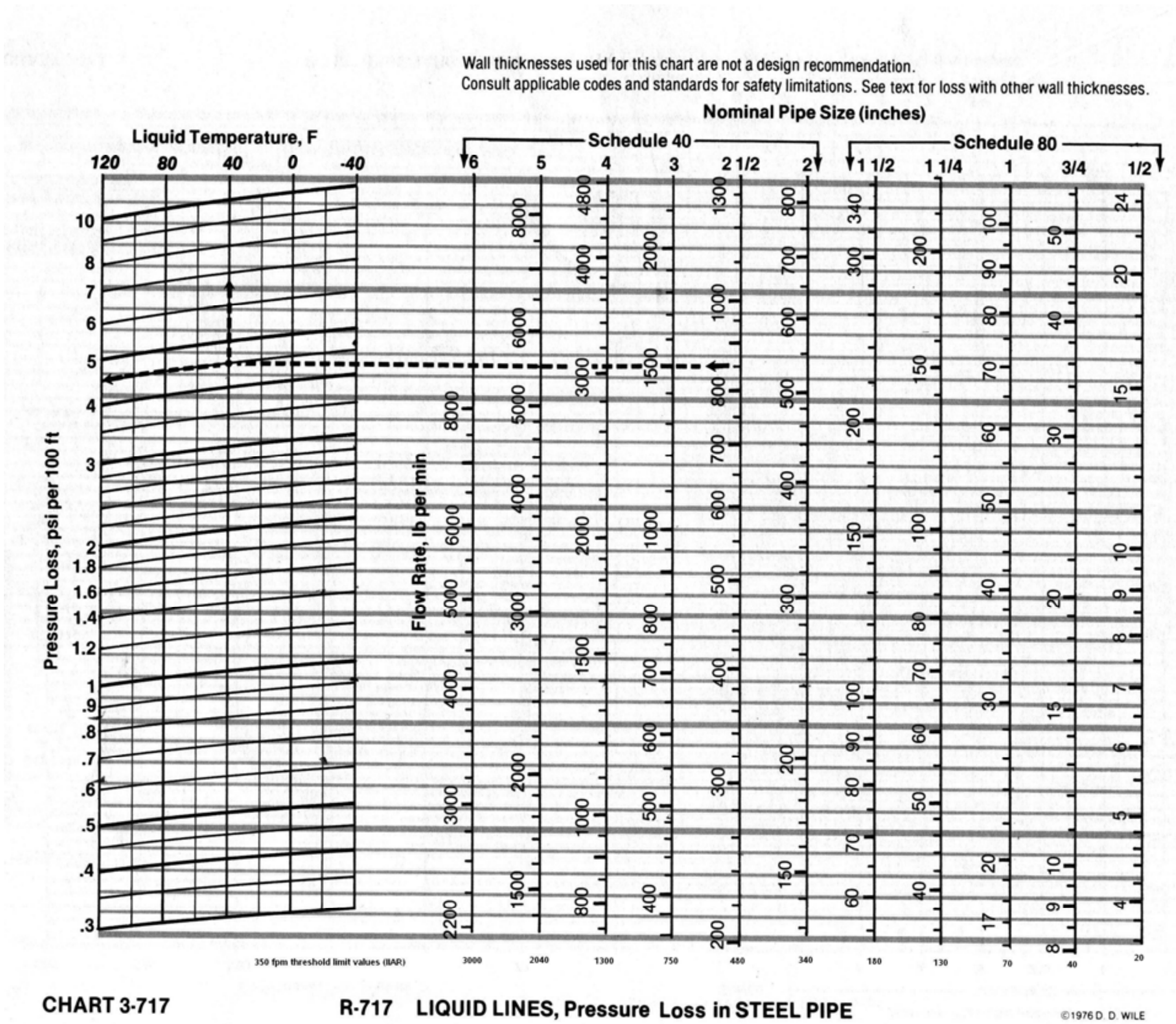
Source: T. Jekel

Figure 6 Chart 1-717 Suction Piping



Source: ASHRAE RP 185

Figure 7 Chart 3-717, Liquid Line Pressure Drop in Steel Piping



Source: ASHRAE, RP185, Wile, 1976